ASME BPVC.III.A-2015

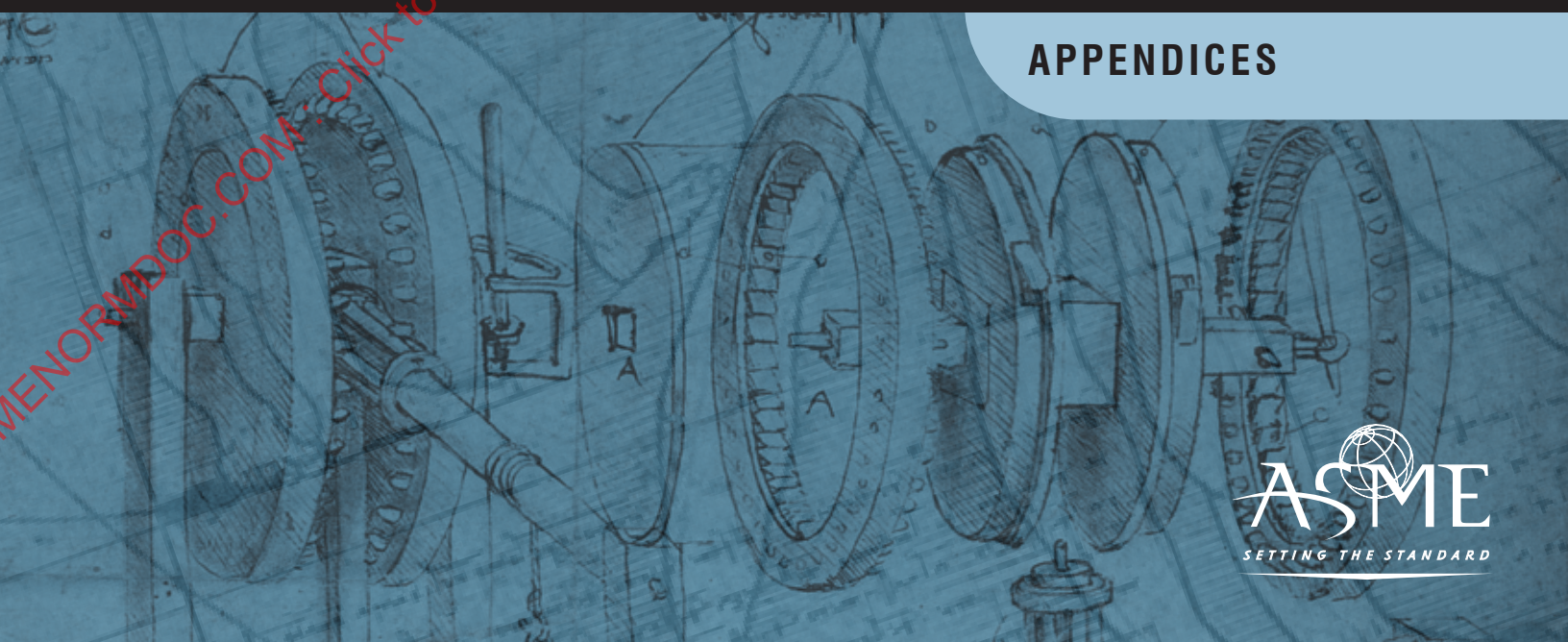
SECTION III

Rules for Construction of Nuclear Facility Components

2015 ASME Boiler and Pressure Vessel Code

An International Code

APPENDICES



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ASME
SETTING THE STANDARD

AN INTERNATIONAL CODE

2015 ASME Boiler & Pressure Vessel Code

2015 Edition July 1, 2015

III

RULES FOR CONSTRUCTION OF NUCLEAR FACILITY COMPONENTS

Appendices

ASME Boiler and Pressure Vessel Committee
on Nuclear Power

The American Society of Mechanical Engineers

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* The 2015 Edition of Section III is the last edition in which Section III, Division 1, Subsection NH, *Class 1 Components in Elevated Temperature Service*, will be published. The requirements located within Subsection NH have been moved to Section III, Division 5, Subsection HB, Subpart B for the elevated temperature construction of Class A components.

INTERPRETATIONS

Interpretations of the Code have historically been posted in January and July at http://cstools.asme.org/interpretations.cfm. Interpretations issued during the previous two calendar years are included with the publication of the applicable Section of the Code in the 2015 Edition. Interpretations of Section III, Divisions 1 and 2 and Section III Appendices are included with Subsection NCA.

Following the 2015 Edition, interpretations will not be included in editions; they will be issued in real time in ASME's Interpretations Database at http://go.asme.org/Interpretations. Historical BPVC interpretations may also be found in the Database.

CODE CASES

The Boiler and Pressure Vessel Code committees meet regularly to consider proposed additions and revisions to the Code and to formulate Cases to clarify the intent of existing requirements or provide, when the need is urgent, rules for materials or constructions not covered by existing Code rules. Those Cases that have been adopted will appear in the appropriate 2015 Code Cases book: "Boilers and Pressure Vessels" or "Nuclear Components." Supplements will be sent or made available automatically to the purchasers of the Code Cases books up to the publication of the 2017 Code.

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FOREWORD[*]

(15)

In 1911, The American Society of Mechanical Engineers established the Boiler and Pressure Vessel Committee to formulate standard rules for the construction of steam boilers and other pressure vessels. In 2009, the Boiler and Pressure Vessel Committee was superseded by the following committees:

(a) Committee on Power Boilers (I)
(b) Committee on Materials (II)
(c) Committee on Construction of Nuclear Facility Components (III)
(d) Committee on Heating Boilers (IV)
(e) Committee on Nondestructive Examination (V)
(f) Committee on Pressure Vessels (VIII)
(g) Committee on Welding, Brazing, and Fusing (IX)
(h) Committee on Fiber-Reinforced Plastic Pressure Vessels (X)
(i) Committee on Nuclear Inservice Inspection (XI)
(j) Committee on Transport Tanks (XII)
(k) Technical Oversight Management Committee (TOMC)

Where reference is made to "the Committee" in this Foreword, each of these committees is included individually and collectively.

The Committee's function is to establish rules of safety relating only to pressure integrity, which govern the construction[**] of boilers, pressure vessels, transport tanks, and nuclear components, and the inservice inspection of nuclear components and transport tanks. The Committee also interprets these rules when questions arise regarding their intent. The technical consistency of the Sections of the Code and coordination of standards development activities of the Committees is supported and guided by the Technical Oversight Management Committee. This Code does not address other safety issues relating to the construction of boilers, pressure vessels, transport tanks, or nuclear components, or the inservice inspection of nuclear components or transport tanks. Users of the Code should refer to the pertinent codes, standards, laws, regulations, or other relevant documents for safety issues other than those relating to pressure integrity. Except for Sections XI and XII, and with a few other exceptions, the rules do not, of practical necessity, reflect the likelihood and consequences of deterioration in service related to specific service fluids or external operating environments. In formulating the rules, the Committee considers the needs of users, manufacturers, and inspectors of pressure vessels. The objective of the rules is to afford reasonably certain protection of life and property, and to provide a margin for deterioration in service to give a reasonably long, safe period of usefulness. Advancements in design and materials and evidence of experience have been recognized.

This Code contains mandatory requirements, specific prohibitions, and nonmandatory guidance for construction activities and inservice inspection and testing activities. The Code does not address all aspects of these activities and those aspects that are not specifically addressed should not be considered prohibited. The Code is not a handbook and cannot replace education, experience, and the use of engineering judgment. The phrase *engineering judgement* refers to technical judgments made by knowledgeable engineers experienced in the application of the Code. Engineering judgments must be consistent with Code philosophy, and such judgments must never be used to overrule mandatory requirements or specific prohibitions of the Code.

The Committee recognizes that tools and techniques used for design and analysis change as technology progresses and expects engineers to use good judgment in the application of these tools. The designer is responsible for complying with Code rules and demonstrating compliance with Code equations when such equations are mandatory. The Code neither requires nor prohibits the use of computers for the design or analysis of components constructed to the

[*] The information contained in this Foreword is not part of this American National Standard (ANS) and has not been processed in accordance with ANSI's requirements for an ANS. Therefore, this Foreword may contain material that has not been subjected to public review or a consensus process. In addition, it does not contain requirements necessary for conformance to the Code.

[**] *Construction*, as used in this Foreword, is an all-inclusive term comprising materials, design, fabrication, examination, inspection, testing, certification, and pressure relief.

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requirements of the Code. However, designers and engineers using computer programs for design or analysis are cautioned that they are responsible for all technical assumptions inherent in the programs they use and the application of these programs to their design.

The rules established by the Committee are not to be interpreted as approving, recommending, or endorsing any proprietary or specific design, or as limiting in any way the manufacturer's freedom to choose any method of design or any form of construction that conforms to the Code rules.

The Committee meets regularly to consider revisions of the rules, new rules as dictated by technological development, Code Cases, and requests for interpretations. Only the Committee has the authority to provide official interpretations of this Code. Requests for revisions, new rules, Code Cases, or interpretations shall be addressed to the Secretary in writing and shall give full particulars in order to receive consideration and action (see Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committees). Proposed revisions to the Code resulting from inquiries will be presented to the Committee for appropriate action. The action of the Committee becomes effective only after confirmation by ballot of the Committee and approval by ASME. Proposed revisions to the Code approved by the Committee are submitted to the American National Standards Institute (ANSI) and published at http://go.asme.org/BPVCPublicReview to invite comments from all interested persons. After public review and final approval by ASME, revisions are published at regular intervals in Editions of the Code.

The Committee does not rule on whether a component shall or shall not be constructed to the provisions of the Code. The scope of each Section has been established to identify the components and parameters considered by the Committee in formulating the Code rules.

Questions or issues regarding compliance of a specific component with the Code rules are to be directed to the ASME Certificate Holder (Manufacturer). Inquiries concerning the interpretation of the Code are to be directed to the Committee. ASME is to be notified should questions arise concerning improper use of an ASME Certification Mark.

When required by context in this Section, the singular shall be interpreted as the plural, and vice versa, and the feminine, masculine, or neuter gender shall be treated as such other gender as appropriate.

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STATEMENT OF POLICY ON THE USE OF THE CERTIFICATION MARK AND CODE AUTHORIZATION IN ADVERTISING

ASME has established procedures to authorize qualified organizations to perform various activities in accordance with the requirements of the ASME Boiler and Pressure Vessel Code. It is the aim of the Society to provide recognition of organizations so authorized. An organization holding authorization to perform various activities in accordance with the requirements of the Code may state this capability in its advertising literature.

Organizations that are authorized to use the Certification Mark for marking items or constructions that have been constructed and inspected in compliance with the ASME Boiler and Pressure Vessel Code are issued Certificates of Authorization. It is the aim of the Society to maintain the standing of the Certification Mark for the benefit of the users, the enforcement jurisdictions, and the holders of the Certification Mark who comply with all requirements.

Based on these objectives, the following policy has been established on the usage in advertising of facsimiles of the Certification Mark, Certificates of Authorization, and reference to Code construction. The American Society of Mechanical Engineers does not "approve," "certify," "rate," or "endorse" any item, construction, or activity and there shall be no statements or implications that might so indicate. An organization holding the Certification Mark and/or a Certificate of Authorization may state in advertising literature that items, constructions, or activities "are built (produced or performed) or activities conducted in accordance with the requirements of the ASME Boiler and Pressure Vessel Code," or "meet the requirements of the ASME Boiler and Pressure Vessel Code." An ASME corporate logo shall not be used by any organization other than ASME.

The Certification Mark shall be used only for stamping and nameplates as specifically provided in the Code. However, facsimiles may be used for the purpose of fostering the use of such construction. Such usage may be by an association or a society, or by a holder of the Certification Mark who may also use the facsimile in advertising to show that clearly specified items will carry the Certification Mark. General usage is permitted only when all of a manufacturer's items are constructed under the rules.

STATEMENT OF POLICY ON THE USE OF ASME MARKING TO IDENTIFY MANUFACTURED ITEMS

The ASME Boiler and Pressure Vessel Code provides rules for the construction of boilers, pressure vessels, and nuclear components. This includes requirements for materials, design, fabrication, examination, inspection, and stamping. Items constructed in accordance with all of the applicable rules of the Code are identified with the official Certification Mark described in the governing Section of the Code.

Markings such as "ASME," "ASME Standard," or any other marking including "ASME" or the Certification Mark shall not be used on any item that is not constructed in accordance with all of the applicable requirements of the Code.

Items shall not be described on ASME Data Report Forms nor on similar forms referring to ASME that tend to imply that all Code requirements have been met when, in fact, they have not been. Data Report Forms covering items not fully complying with ASME requirements should not refer to ASME or they should clearly identify all exceptions to the ASME requirements.

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(15) # SUBMITTAL OF TECHNICAL INQUIRIES TO THE BOILER AND PRESSURE VESSEL STANDARDS COMMITTEES

1 INTRODUCTION

(a) The following information provides guidance to Code users for submitting technical inquiries to the committees. See Guideline on the Approval of New Materials Under the ASME Boiler and Pressure Vessel Code in Section II, Parts C and D for additional requirements for requests involving adding new materials to the Code. Technical inquiries include requests for revisions or additions to the Code rules, requests for Code Cases, and requests for Code Interpretations, as described below.

(1) Code Revisions. Code revisions are considered to accommodate technological developments, address administrative requirements, incorporate Code Cases, or to clarify Code intent.

(2) Code Cases. Code Cases represent alternatives or additions to existing Code rules. Code Cases are written as a question and reply, and are usually intended to be incorporated into the Code at a later date. When used, Code Cases prescribe mandatory requirements in the same sense as the text of the Code. However, users are cautioned that not all jurisdictions or owners automatically accept Code Cases. The most common applications for Code Cases are:

(-a) to permit early implementation of an approved Code revision based on an urgent need

(-b) to permit the use of a new material for Code construction

(-c) to gain experience with new materials or alternative rules prior to incorporation directly into the Code

(3) Code Interpretations. Code Interpretations provide clarification of the meaning of existing rules in the Code, and are also presented in question and reply format. Interpretations do not introduce new requirements. In cases where existing Code text does not fully convey the meaning that was intended, and revision of the rules is required to support an interpretation, an Intent Interpretation will be issued, and the Code will be revised.

(b) The Code rules, Code Cases, and Code Interpretations established by the committees are not to be considered as approving, recommending, certifying, or endorsing any proprietary or specific design, or as limiting in any way the freedom of manufacturers, constructors, or owners to choose any method of design or any form of construction that conforms to the Code rules.

(c) Inquiries that do not comply with these provisions or that do not provide sufficient information for a committee's full understanding may result in the request being returned to the inquirer with no action.

2 INQUIRY FORMAT

Submittals to a committee shall include:

(a) Purpose. Specify one of the following:

(1) revision of present Code rules

(2) new or additional Code rules

(3) Code Case

(4) Code Interpretation

(b) Background. Provide the information needed for the committee's understanding of the inquiry, being sure to include reference to the applicable Code Section, Division, edition, addenda (if applicable), paragraphs, figures, and tables. Preferably, provide a copy of the specific referenced portions of the Code.

(c) Presentations. The inquirer may desire or be asked to attend a meeting of the committee to make a formal presentation or to answer questions from the committee members with regard to the inquiry. Attendance at a committee meeting shall be at the expense of the inquirer. The inquirer's attendance or lack of attendance at a meeting shall not be a basis for acceptance or rejection of the inquiry by the committee.

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3 CODE REVISIONS OR ADDITIONS

Requests for Code revisions or additions shall provide the following:

(a) Proposed Revisions or Additions. For revisions, identify the rules of the Code that require revision and submit a copy of the appropriate rules as they appear in the Code, marked up with the proposed revision. For additions, provide the recommended wording referenced to the existing Code rules.

(b) Statement of Need. Provide a brief explanation of the need for the revision or addition.

(c) Background Information. Provide background information to support the revision or addition, including any data or changes in technology that form the basis for the request that will allow the committee to adequately evaluate the proposed revision or addition. Sketches, tables, figures, and graphs should be submitted as appropriate. When applicable, identify any pertinent paragraph in the Code that would be affected by the revision or addition and identify paragraphs in the Code that reference the paragraphs that are to be revised or added.

4 CODE CASES

Requests for Code Cases shall provide a Statement of Need and Background Information similar to that defined in 3(b) and 3(c), respectively, for Code revisions or additions. The urgency of the Code Case (e.g., project underway or imminent, new procedure, etc.) must be defined and it must be confirmed that the request is in connection with equipment that will bear the Certification Mark, with the exception of Section XI applications. The proposed Code Case should identify the Code Section and Division, and be written as a *Question* and a *Reply* in the same format as existing Code Cases. Requests for Code Cases should also indicate the applicable Code editions and addenda (if applicable) to which the proposed Code Case applies.

5 CODE INTERPRETATIONS

(a) Requests for Code Interpretations shall provide the following:

(1) Inquiry. Provide a condensed and precise question, omitting superfluous background information and, when possible, composed in such a way that a "yes" or a "no" *Reply,* with brief provisos if needed, is acceptable. The question should be technically and editorially correct.

(2) Reply. Provide a proposed *Reply* that will clearly and concisely answer the *Inquiry* question. Preferably, the *Reply* should be "yes" or "no," with brief provisos if needed.

(3) Background Information. Provide any background information that will assist the committee in understanding the proposed *Inquiry* and *Reply.*

(b) Requests for Code Interpretations must be limited to an interpretation of a particular requirement in the Code or a Code Case. The committee cannot consider consulting type requests such as the following:

(1) a review of calculations, design drawings, welding qualifications, or descriptions of equipment or parts to determine compliance with Code requirements;

(2) a request for assistance in performing any Code-prescribed functions relating to, but not limited to, material selection, designs, calculations, fabrication, inspection, pressure testing, or installation;

(3) a request seeking the rationale for Code requirements.

6 SUBMITTALS

Submittals to and responses from the committees shall meet the following:

(a) Submittal. Inquiries from Code users shall be in English and preferably be submitted in typewritten form; however, legible handwritten inquiries will also be considered. They shall include the name, address, telephone number, fax number, and e-mail address, if available, of the inquirer and be mailed to the following address:

Secretary
ASME Boiler and Pressure Vessel Committee
Two Park Avenue
New York, NY 10016-5990

As an alternative, inquiries may be submitted via e-mail to: SecretaryBPV@asme.org or via our online tool at http://go.asme.org/InterpretationRequest.

(b) Response. The Secretary of the appropriate committee shall acknowledge receipt of each properly prepared inquiry and shall provide a written response to the inquirer upon completion of the requested action by the committee.

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INTRODUCTION (15)

Section III appendices are referred to as either Section III Appendices or Subsection Appendices. These appendices are further designated as either mandatory or nonmandatory for use. Mandatory Appendices are referred to in the Section III rules and contain requirements that must be followed in construction. Nonmandatory Appendices provide additional information or guidance when using Section III.

Section III Appendices are contained in this book. These appendices have the potential for multiple subsection applicability. Mandatory Appendices are designated by a Roman numeral followed, when appropriate, by Arabic numerals to indicate the various articles, subarticles, and paragraphs of the appendix, such as II-1500 or XIII-2131. Nonmandatory Appendices are designated by a capital letter followed, when appropriate, by Arabic numerals to indicate various articles, subarticles, and paragraphs of the appendix, such as D-1200 or Y-2410.

Subsection Appendices are specifically applicable to one subsection and are contained within that subsection. Subsection-specific Mandatory and Nonmandatory Appendices are numbered in the same manner as Section III Appendices but with a subsection identifier (e.g., NF, D2, HBB, etc.) preceding either the Roman numeral or the capital letter for a unique designation. For example, NF-II-1100 or NF-A-1200 would be a part of Subsection NF Mandatory Appendix NF-II or Nonmandatory Appendix NF-A, respectively. For Subsection CC, D2-IV-1120 or D2-D-1330 would be a part of Subsection CC Mandatory Appendix D2-IV or Nonmandatory Appendix D2-D, respectively.

A Reference Table (Table 1) has been developed for Section III Appendices to provide additional guidance on appendix usage for the Code user. This Reference Table, reflecting down to a Subsection level, does not take precedence over Code rules.

TABLE 1 (15)
SECTION III APPENDICES REFERENCE TABLE

Appendix Identifier	Sub.	Division 1 Subsections						Div. 2 Sub.	Division 3 Subsections			Division 5 Subsections									
												HA		HB		HC		HF	HG		HH
	NCA	NB	NC	ND	NE	NF	NG	CC	WA	WB	WC	Subpart A	Subpart B	Subpart A	Subpart B (1)	Subpart A	Subpart B (1)	Subpart A	Subpart A	Subpart B (1)	Subpart A
MANDATORY APPENDICES																					
I		X			X	X	X			X				(3)		(3)		(3)	(3)	X	
II		X	X	X	X	X				X	X			(3)	X	(3)		(3)	(3)	X	
III		X			X	X	X			X				(3)		(3)		(3)	(3)		
IV		X												(3)		(3)		(3)	(3)		
V	X	(2)	(2)	(2)	(2)	(2)	(2)	X				(2)	X	(3)	(2)	(3)	(2)	(3)	(3)	(2)	(4)
VI		X	X	X	X		X	X		X	X			(3)	(5)	(3)		(3)	(3)		
VII	Not in use																				
VIII	Not in use																				
IX	Not in use																				
X	Not in use																				
XI		X	X	X	X					X				(3)		(3)		(3)	(3)		
XII														(3)		(3)		(3)	(3)		
XIII			X	X							X			(3)		(3)		(3)	(3)		
XIV			X								X			(3)		(3)		(3)	(3)		

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TABLE 1
SECTION III APPENDICES REFERENCE TABLE (CONT'D)

Appendix Identifier	Sub.	Division 1 Subsections						Div. 2 Sub.	Division 3 Subsections			Division 5 Subsections										
												HA		HB		HC		HF	HG		HH	
	NCA	NB	NC	ND	NE	NF	NG	CC	WA	WB	WC	Subpart A	Subpart B	Subpart A	Subpart B (1)	Subpart A	Subpart B (1)	Subpart A	Subpart A	Subpart B (1)	Subpart A	
MANDATORY APPENDICES																						
XV	Not in use																					
XVI	Not in use																					
XVII	Not in use																					
XVIII		X	X	X	X									(3)		(3)		(3)	(3)			
XIX			X	X										(3)		(3)		(3)	(3)			
XX	Not in use																					
XXI	X	(2)	(2)	(2)	(2)	(2)	(2)	(2)				(2)		(3)	(2)	(3)	(2)	(3)	(3)	(2)		
XXII				X										(3)		(3)		(3)	(3)			
XXIII	X	X	X	X	X	(2)	(2)	(2)	X			(2)	X	(3)	(5)	(3)	(2)	(3)	(3)	(2)	(4)	
XXIV									X					(3)		(3)		(3)	(3)			
XXV																						
NONMANDATORY APPENDICES																						
A		X	X	X	X	X	X	X				X			(3)	X	(3)		(3)	(3)	X	
B	X	(2)	(2)	(2)	(2)	(2)	(2)	(2)	(2)				(2)	X	(3)	(2)	(3)	(2)	(3)	(3)	(2)	(4)
C	X	(2)	X	(2)	(2)	(2)	(2)	(2)	(2)			X	(2)	X	(3)	(2)	(3)	(2)	(3)	(3)	(2)	(4)
D		X	X	X	X	X	X	(5)			X	X			(3)	(5)	(3)		(3)	(3)		
E		X						X			X				(3)	X	(3)		(3)	(3)		
F		X	X	X	X	X	X	X			X	X			(3)	X	(3)		(3)	(3)	X	
G	X	X	X	(2)	X	X	X	X	(2)		X	X	(2)		(3)	X	(3)	(2)	(3)	(3)	X	
H	Not in use																					
I	Not in use																					
J	Not in use																					
K	Not in use																					
L			X	X											(3)		(3)		(3)	(3)		
M															(3)		(3)		(3)	(3)		
N															(3)		(3)		(3)	(3)		
O															(3)		(3)		(3)	(3)		
P															(3)		(3)		(3)	(3)		
Q															(3)		(3)		(3)	(3)		
R			X		X							X			(3)		(3)		(3)	(3)		
S		X	X	X											(3)		(3)		(3)	(3)		
T						X									(3)		(3)		(3)	(3)		
U															(3)		(3)		(3)	(3)		
V	Not in use																					
W	X	(2)	(2)	(2)	(2)	(2)	(2)	(2)	(2)				(2)		(3)	(2)	(3)	(2)	(3)	(3)	(2)	
X	Not in use																					
Y		X	X	X											(3)		(3)		(3)	(3)		
Z															(3)		(3)		(3)	(3)		
AA	X	(2)	(2)	(2)	(2)	(2)	(2)	(2)	(2)				(2)	X	(3)	(2)	(3)	(2)	(3)	(3)	(2)	(4)
BB			X	X											(3)		(3)		(3)	(3)		

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TABLE 1
SECTION III APPENDICES REFERENCE TABLE (CONT'D)

NONMANDATORY APPENDICES																					
CC	X					(2)							(2)		(3)		(3)		(3)	(3)	
DD	X			(2)									(2)		(3)		(3)		(3)	(3)	
EE																					
FF											X	X									

GENERAL NOTE: This reference table is not intended to provide specific Code requirements. It provides general guidance only. Mandatory and Nonmandatory Appendices marked with an 'X' in the table are specifically referred to in the identified Subsection rules. Section III Mandatory Appendices are not to be interpreted as being a requirement for all Section III Subsections. However, certain Mandatory or Nonmandatory Section III Appendices may be appropriate for use in other subsections as long as the respective subsection rules are satisfied.
Sub. = Subsection
Div. = Division

NOTES:
(1) Subpart B for Subsections HB, HC, and HG contain provisions for the use of all or portions of the rules in Division 1, Subsections NB, NC, and NG, respectively, if creep effects are negligible. In those circumstances, the reference to the Appendices is governed by the reference defined for those respective subsections of Division 1.
(2) The Appendix reference determined by Subsection NCA (or, for Division 5, via Subsection HA, Subpart A referencing Subsection NCA) shall apply.
(3) The Appendix reference determined by the referenced Division 1 Subsection shall apply.
(4) The Appendix reference determined by Subsection HA, Subpart B shall apply.
(5) The Appendix reference determined by Subsection NB shall apply.

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(15)

ORGANIZATION OF SECTION III

1 GENERAL

Section III consists of Division 1, Division 2, Division 3, and Division 5. These Divisions are broken down into Subsections and are designated by capital letters preceded by the letter "N" for Division 1, by the letter "C" for Division 2, by the letter "W" for Division 3, and by the letter "H" for Division 5. Each Subsection is published separately, with the exception of those listed for Divisions 2, 3, and 5.

- Subsection NCA — General Requirements for Division 1 and Division 2
- Appendices
- Division 1
  - Subsection NB — Class 1 Components
  - Subsection NC — Class 2 Components
  - Subsection ND — Class 3 Components
  - Subsection NE — Class MC Components
  - Subsection NF — Supports
  - Subsection NG — Core Support Structures
  - Subsection NH — Class 1 Components in Elevated Temperature Service[*]
- Division 2 — Code for Concrete Containments
  - Subsection CC — Concrete Containments
- Division 3 — Containments for Transportation and Storage of Spent Nuclear Fuel and High Level Radioactive Material and Waste
  - Subsection WA — General Requirements for Division 3
  - Subsection WB — Class TC Transportation Containments
  - Subsection WC — Class SC Storage Containments
- Division 5 — High Temperature Reactors
  - Subsection HA — General Requirements
    - Subpart A — Metallic Materials
    - Subpart B — Graphite Materials
    - Subpart C — Composite Materials
  - Subsection HB — Class A Metallic Pressure Boundary Components
    - Subpart A — Low Temperature Service
    - Subpart B — Elevated Temperature Service
  - Subsection HC — Class B Metallic Pressure Boundary Components
    - Subpart A — Low Temperature Service
    - Subpart B — Elevated Temperature Service
  - Subsection HF — Class A and B Metallic Supports
    - Subpart A — Low Temperature Service
  - Subsection HG — Class A Metallic Core Support Structures
    - Subpart A — Low Temperature Service
    - Subpart B — Elevated Temperature Service
  - Subsection HH — Class A Nonmetallic Core Support Structures
    - Subpart A — Graphite Materials
    - Subpart B — Composite Materials

2 SUBSECTIONS

Subsections are divided into Articles, subarticles, paragraphs, and, where necessary, subparagraphs and subsubparagraphs.

[*] The 2015 Edition of Section III is the last edition in which Section III, Division 1, Subsection NH, *Class 1 Components in Elevated Temperature Service*, will be published. The requirements located within Subsection NH have been moved to Section III, Division 5, Subsection HB, Subpart B for the elevated temperature construction of Class A components.

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3 ARTICLES

Articles are designated by the applicable letters indicated above for the Subsections followed by Arabic numbers, such as NB-1000. Where possible, Articles dealing with the same topics are given the same number in each Subsection, except NCA, in accordance with the following general scheme:

Article Number	Title
1000	Introduction or Scope
2000	Material
3000	Design
4000	Fabrication and Installation
5000	Examination
6000	Testing
7000	Overpressure Protection
8000	Nameplates, Stamping With Certification Mark, and Reports

The numbering of Articles and the material contained in the Articles may not, however, be consecutive. Due to the fact that the complete outline may cover phases not applicable to a particular Subsection or Article, the rules have been prepared with some gaps in the numbering.

4 SUBARTICLES

Subarticles are numbered in units of 100, such as NB-1100.

5 SUBSUBARTICLES

Subsubarticles are numbered in units of 10, such as NB-2130, and generally have no text. When a number such as NB-1110 is followed by text, it is considered a paragraph.

6 PARAGRAPHS

Paragraphs are numbered in units of 1, such as NB-2121.

7 SUBPARAGRAPHS

Subparagraphs, when they are *major* subdivisions of a paragraph, are designated by adding a decimal followed by one or more digits to the paragraph number, such as NB-1132.1. When they are *minor* subdivisions of a paragraph, subparagraphs may be designated by lowercase letters in parentheses, such as NB-2121(a).

8 SUBSUBPARAGRAPHS

Subsubparagraphs are designated by adding lowercase letters in parentheses to the *major* subparagraph numbers, such as NB-1132.1(a). When further subdivisions of *minor* subparagraphs are necessary, subsubparagraphs are designated by adding Arabic numerals in parentheses to the subparagraph designation, such as NB-2121(a)(1).

9 REFERENCES

References used within Section III generally fall into one of the following four categories:

(a) References to Other Portions of Section III. When a reference is made to another Article, subarticle, or paragraph, all numbers subsidiary to that reference shall be included. For example, reference to NB-3000 includes all material in Article NB-3000; reference to NB-3200 includes all material in subarticle NB-3200; reference to NB-3230 includes all paragraphs, NB-3231 through NB-3236.

(b) References to Other Sections. Other Sections referred to in Section III are the following:

(1) Section II, Materials. When a requirement for a material, or for the examination or testing of a material, is to be in accordance with a specification such as SA-105, SA-370, or SB-160, the reference is to material specifications in Section II. These references begin with the letter "S."

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(2) Section V, Nondestructive Examination. Section V references begin with the letter "T" and relate to the nondestructive examination of material or welds.

(3) Section IX, Welding and Brazing Qualifications. Section IX references begin with the letter "Q" and relate to welding and brazing requirements.

(4) Section XI, Rules for Inservice Inspection of Nuclear Power Plant Components. When a reference is made to inservice inspection, the rules of Section XI shall apply.

(c) Reference to Specifications and Standards Other Than Published in Code Sections

(1) Specifications for examination methods and acceptance standards to be used in connection with them are published by the American Society for Testing and Materials (ASTM). At the time of publication of Section III, some such specifications were not included in Section II of this Code. A reference to ASTM E94 refers to the specification so designated by and published by ASTM, 100 Barr Harbor Drive, West Conshohocken, PA 19428.

(2) Dimensional standards covering products such as valves, flanges, and fittings are sponsored and published by The American Society of Mechanical Engineers and approved by the American National Standards Institute.[**] When a product is to conform to such a standard, for example ASME B16.5, the standard is approved by the American National Standards Institute. The applicable year of issue is that suffixed to its numerical designation in Table NCA-7100-1, for example ASME B16.5-2003. Standards published by The American Society of Mechanical Engineers are available from ASME (https://www.asme.org/).

(3) Dimensional and other types of standards covering products such as valves, flanges, and fittings are also published by the Manufacturers Standardization Society of the Valve and Fittings Industry and are known as Standard Practices. When a product is required by these rules to conform to a Standard Practice, for example MSS SP-100, the Standard Practice referred to is published by the Manufacturers Standardization Society of the Valve and Fittings Industry, Inc. (MSS), 127 Park Street, NE, Vienna, VA 22180. The applicable year of issue of such a Standard Practice is that suffixed to its numerical designation in Table NCA-7100-1, for example MSS SP-89-2003.

(4) Specifications for welding and brazing materials are published by the American Welding Society (AWS), 8669 Doral Boulevard, Suite 130, Doral, FL 33166. Specifications of this type are incorporated in Section II and are identified by the AWS designation with the prefix "SF," for example SFA-5.1.

(5) Standards applicable to the design and construction of tanks and flanges are published by the American Petroleum Institute and have designations such as API-605. When documents so designated are referred to in Section III, for example API-605–1988, they are standards published by the American Petroleum Institute and are listed in Table NCA-7100-1.

(d) References to Appendices. Section III uses two types of appendices that are designated as either Section III Appendices or Subsection Appendices. Either of these appendices is further designated as either Mandatory or Nonmandatory for use. Mandatory Appendices are referred to in the Section III rules and contain requirements that must be followed in construction. Nonmandatory Appendices provide additional information or guidance when using Section III.

(1) Section III Appendices are contained in a separate book titled "Appendices." These appendices have the potential for multiple subsection applicability. Mandatory Appendices are designated by a Roman numeral followed, when appropriate, by Arabic numerals to indicate various articles, subarticles, and paragraphs of the appendix, such as II-1500 or XIII-2131. Nonmandatory Appendices are designated by a capital letter followed, when appropriate, by Arabic numerals to indicate various articles, subarticles, and paragraphs of the appendix, such as D-1200 or Y-1440.

(2) Subsection Appendices are specifically applicable to just one subsection and are contained within that subsection. Subsection-specific mandatory and nonmandatory appendices are numbered in the same manner as Section III Appendices, but with a subsection identifier (e.g., NF, NH, D2, etc.) preceding either the Roman numeral or the capital letter for a unique designation. For example, NF-II-1100 or NF-A-1200 would be part of a Subsection NF mandatory or nonmandatory appendix, respectively. For Subsection CC, D2-IV-1120 or D2-D-1330 would be part of a Subsection CC mandatory or nonmandatory appendix, respectively.

(3) It is the intent of this Section that the information provided in both Mandatory and Nonmandatory Appendices may be used to meet the rules of any Division or Subsection. In case of conflict between Appendix rules and Division/Subsection rules, the requirements contained in the Division/Subsection shall govern. Additional guidance on Appendix usage is provided in the front matter of Section III Appendices.

[**] The American National Standards Institute (ANSI) was formerly known as the American Standards Association. Standards approved by the Association were designated by the prefix "ASA" followed by the number of the standard and the year of publication. More recently, the American National Standards Institute was known as the United States of America Standards Institute. Standards were designated by the prefix "USAS" followed by the number of the standard and the year of publication. While the letters of the prefix have changed with the name of the organization, the numbers of the standards have remained unchanged.

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SUMMARY OF CHANGES

After publication of the 2015 Edition, Errata to the BPV Code may be posted on the ASME Web site to provide corrections to incorrectly published items, or to correct typographical or grammatical errors in the BPV Code. Such Errata shall be used on the date posted.

Information regarding Special Notices and Errata is published by ASME at http://go.asme.org/BPVCerrata.

Changes given below are identified on the pages by a margin note, **(15)**, placed next to the affected area.

The Record Numbers listed below are explained in more detail in "List of Changes in Record Number Order" following this Summary of Changes.

Page	*Location*	*Change (Record Number)*
xv	List of Sections	Revised
xvii	Foreword	*(1)* Revised *(2)* New footnote added by errata *(13-860)*
xx	Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committees	Revised
xxii	Personnel	Updated
xxxix	Introduction	*(1)* Third paragraph revised *(14-26)* *(2)* In Table 1, "NH" column deleted and "HB Subpart B" column revised *(14-26)*
xxxix	Table 1	
xlii	Organization of Section III	*(1)* New footnote added *(2)* 9(d)(3) added *(13-1032)*
li	Cross-Referencing and Stylistic Changes in the Boiler and Pressure Vessel Code	Added
38	Mandatory Appendix III	Revised in its entirety *(09-207, 12-1105)*
71	Mandatory Appendix VI	Appendix title added *(12-2045)*
72	Table VI-1132-1	Penultimate entry in first column and all the metric values revised *(12-2045)*
88	Table XI-3221.1-1	"Asbestos" in first column changed to "mineral" or "mineral fiber" *(12-1830)*
105	Table XIII-1130-1	Under "Vessel Component" column, for first "Any" entry, references to Notes in third and fourth columns corrected by errata *(13-1943)*
109	XIII-1182	Revised *(12-1105)*
112	XIII-2124	In subpara. (c), in-text table, first entry corrected by errata to D/t *(14-778)*
115	XIV-1221.3	Subparagraph (c), cross-reference to Figures I-9.1 through I-9.6 corrected by errata *(14-778)*

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Page	Location	Change (Record Number)
116	XIV-1322	Subparagraph (b) revised *(12-1105)*
127	XIX-1200	Equations in subparas. (b)(1) and (b)(2) corrected by errata *(14-778)*
129	Mandatory Appendix XXI	*(1)* Appendix title added *(13-1647)* *(2)* Article XXI-1000 title revised *(13-1647)* *(3)* XXI-1110(d) and XXI-1120(c)(1) revised *(13-1647)*
130	XXII-1200	First equation corrected by errata *(14-325)*
134	Mandatory Appendix XXIII	Revised in its entirety *(13-1817)*
152	Mandatory Appendix XXVI	Added *(13-895)*
208	A-2252	Fifth equation corrected by errata *(14-778, 14-1564)*
217	A-5212	Equation (3) corrected by errata *(14-778, 14-1564)*
229	A-8120	Former endnote 18 deleted by errata; subsequent endnotes renumbered *(12-1156)*
230	A-8132.1	Equation (3) number added by errata *(12-1156)*
232	A-8132.4	Equation (13) corrected by errata *(12-1156)*
233	Figure A-8132.4-1	In the graphic, *p* corrected by errata to *P* *(12-1156)*
233	A-8142.2	*(1)* Former subpara. (c) designation deleted, and remaining subparagraph redesignated by errata *(12-1156)* *(2)* Subparagraphs (b) and (c) corrected by errata *(12-1156)*
236	A-8143.2	Subparagraph (a) corrected by errata *(12-1156)*
236	A-8152	Equation (23) corrected by errata *(12-1156)*
236	A-8153	Subparagraph (b) corrected by errata *(12-1156)*
256	B-2123.2	In the last sentence, cross-reference to Subsection NH deleted *(14-26)*
257	B-2150	In B-2151, B-2152, and B-2154, cross-reference to Subsection NH deleted *(14-26)*
257	B-2160	In B-2161.1 and B-2161.2, cross-reference to Subsection NH deleted *(14-26)*
272	Nonmandatory Appendix D	*(1)* Title added *(13-1648)* *(2)* Article D-1000 title revised *(13-1648)*
273	Table D-1210-1	General Note and Note (1) revised *(13-1648)*
275	E-1120	In nomenclature and endnote 26, Tables E-1210-1 and E-1210-2 references revised to Tables XI-3221.1-1 and XI-3221.1-2 *(13-494)*
276	Table E-1210-1	Deleted *(13-494)*
276	Table E-1210-2	Deleted *(13-494)*
281	F-1332.7	First equation corrected by errata *(14-1084)*
281	F-1334.3	Square root symbol in subpara. (b)(1) corrected by errata *(14-1084)*
282	F-1334.6	Subparagraph (d) added *(12-1562)*
284	F-1342	First equation corrected by errata *(14-1084)*

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Page	Location	Change (Record Number)
285	F-1440	Subparagraphs (c) and (d) revised *(12-127)*
289	Figure G-2210-1	General Note deleted by errata *(14-1085)*
290	Figure G-2210-1M	General Note deleted by errata *(14-1085)*
291	Figure G-2214-1	Equation corrected by errata *(14-1660)*
292	Figure G-2214-1M	Equation corrected by errata *(14-1660)*
292	G-2215	Equation (1) corrected by errata *(14-1085)*
308	L-3242	Equation (18) corrected by errata *(14-778)*
309	L-3243	Second equation in subpara. (a)(4)(-b) corrected by errata *(14-778)*
310	L-3244	Equations (23) through (28) corrected by errata *(14-778)*
312	Nonmandatory Appendix M	*(1)* Title added *(13-1649)* *(2)* Article M-1000 title revised *(13-1649)*
327	N-1225.1.1	Subparagraphs (a) and (b) designations and the word "where" before nomenclature deleted by errata *(14-1086)*
340	N-1233.1	Equations (59) and (62) through (64) corrected by errata *(14-1086)*
345	N-1312	Nomenclature term L_i corrected by errata *(14-1086)*
355	N-1342.1	Equation (87) corrected by errata *(14-1086)*
357	N-1343.1	Second paragraph under eq. (98) corrected by errata *(15-70)*
357	N-1343.2	First paragraph corrected by errata *(15-70)*
363	N-1451.1	Last line in eq. (112) corrected by errata *(15-70)*
373	N-1800	Reference [145] corrected by errata *(15-70)*
381	O-1310	Subparagraphs (a) and (b) corrected by errata *(14-1533)*
381	O-1320	Subparagraphs (a) and (b) corrected by errata *(14-1533)*
382	O-1500	Subparagraph (a) corrected by errata *(14-1533)*
385	Q-1130	*(1)* Last four equations in nomenclature term I_h deleted by errata *(14-322)* *(2)* Equation in nomenclature term M_H corrected by errata *(14-778)*
392	Nonmandatory Appendix R	*(1)* Appendix title added *(13-1646)* *(2)* Article R-1000 title revised *(13-1646)* *(3)* R-1110 and R-1200 revised *(13-1646)*
423	W-1100	Revised in its entirety *(12-8)*
426	Article W-2000	Revised in its entirety *(12-372)*
449	Article W-3000	Revised in its entirety *(12-375)*
460	Article W-4000	Revised in its entirety *(12-374)*
469	W-4430	First sentence revised *(14-26)*
472	Y-2300	Definitions of nomenclature terms β_1 and β_2 corrected by errata *(14-778)*
472	Figure Y-2300-1	*(1)* Editorially revised *(2)* Note (2) corrected by errata *(14-778)*
473	Y-2410	Equations (3) through (5) and Note corrected by errata *(14-778)*

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Page	*Location*	*Change (Record Number)*
475	Y-3300	Definitions of nomenclature terms β_1 and β_2 corrected by errata *(14-778)*
476	Y-3400	First paragraph corrected by errata *(14-778)*
476	Y-3410	Equations (3) through (6) and subparas. (b)(1) and (b)(3) through (b)(5) corrected by errata *(14-778)*
477	Y-3420	Equations (7) and (8) corrected by errata *(14-778)*
478	Y-4300	In-text table corrected by errata *(14-778)*
479	Y-4400	Second sentence corrected by errata *(14-778)*
479	Y-4410	Note corrected by errata *(14-778)*
481	Y-5300	In-text table corrected by errata *(14-778)*
482	Y-5400	First paragraph corrected by errata *(14-778)*
482	Y-5410	Subparagraphs (b)(3) through (b)(5) corrected by errata *(14-778)*
513	EE-1250	Second and third paragraphs corrected by errata *(13-1853)*
519	Nonmandatory Appendix GG	Added *(13-621)*
520	Nonmandatory Appendix HH	Added *(11-1552)*

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NOTE: Volume 63 of the Interpretations to Section III Appendices of the ASME Boiler and Pressure Vessel Code follows the last page of Subsection NCA.

LIST OF CHANGES IN RECORD NUMBER ORDER

Record Number	Change
09-207	Revised Mandatory Appendix III to reference Section II, Part D, Appendices 1 and 2, and deleted duplicate information.
11-1552	Added Nonmandatory Appendix HH.
12-8	Revised Nonmandatory Appendix W to add current EPRI, NRC, and other references; typical nuclear power plant environments; materials; and design measures to reduce the susceptibility to environmental effects. Added the GALL report (NUREG 1810) and other PLEX references. Added a Glossary.
12-127	Replaced "ultimate strength greater than 100 ksi (690 MPa) at operating temperature" with "specified minimum tensile strength greater than or equal to 100 ksi (690 MPa)" in F-1440(c). Replaced "ultimate strength less than or equal to 100 ksi (690 MPa) at operating temperature" with "specified minimum tensile strength less than 100 ksi (690 MPa)" in F-1440(d).
12-372	Supplemented existing text and references in Article W-2000.
12-374	Updated EPRI, NRC, industry, and literature references in Article W-4000.
12-375	Updated EPRI, NRC, industry, and literature references in Article W-3000.
12-1105	Revised Mandatory Appendices III, XIII, and XIV to utilize S_y for service stresses to account for the revised criteria for S_m values for bolting.
12-1156	Errata correction. See Summary of Changes for details.
12-1562	Revised F-1334.6 to define the Level D allowable for the first term of NF-3221(e)(1) eq. (21).
12-1830	Replaced "asbestos" with "mineral fiber" in Table XI-3221.1-1.
12-2045	Revised Table VI-1132-1 as necessary to have the proper precision (number of significant figures) considering both the U.S. Customary and SI unit values. Title added to Mandatory Appendix VI, and a minor editorial change to the material thickness description was made.
13-494	Deleted Tables E-1210-1 and E-1210-2, and updated table references to reference the same tables in Mandatory Appendix XI.
13-621	Deleted Tables NB-3642.1(b)-1 and NC/ND-3642.1(c)-1, and replaced with new Nonmandatory Appendix GG.
13-860	In the Foreword, the subtitle has been deleted and replaced with an ANSI disclaimer as a footnote.
13-895	Added Mandatory Appendix XXVI to incorporate Code Case N-755-2 with subsequent committee-approved changes.
13-1032	Added paragraph to the Introduction of Section III, Division 1, Organization of Section III, Article 9(d), References to Appendices to add guidance on the use of nonmandatory appendices for Section III.
13-1646	Added Nonmandatory Appendix R title and the Division 3, WC references into R-1110 and R-1200.
13-1647	Added Mandatory Appendix XXI title and editorial metric equivalents.
13-1648	Added Nonmandatory Appendix D title, and revised General Note and Note (1) of Table D-1210-1 to refer to all the Subsections and Divisions.
13-1649	Added Nonmandatory Appendix M title.
13-1817	Updated Mandatory Appendix XXIII to reflect recent changes and additions to Section III.
13-1853	Errata correction. See Summary of Changes for details.
13-1943	Errata correction. See Summary of Changes for details.
14-26	Revised references of Division 1, Subsection NH to Division 5, Subsection HB, Subpart B.
14-322	Errata correction. See Summary of Changes for details.
14-325	Errata correction. See Summary of Changes for details.
14-778	Errata correction. See Summary of Changes for details.
14-1084	Errata correction. See Summary of Changes for details.
14-1085	Errata correction. See Summary of Changes for details.
14-1086	Errata correction. See Summary of Changes for details.
14-1533	Errata correction. See Summary of Changes for details.

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Record Number	Change
14-1564	Errata correction. See Summary of Changes for details.
14-1660	Errata correction. See Summary of Changes for details.
15-70	Errata correction. See Summary of Changes for details.

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CROSS-REFERENCING AND STYLISTIC CHANGES IN THE BOILER AND PRESSURE VESSEL CODE

There have been structural and stylistic changes to BPVC, starting with the 2011 Addenda, that should be noted to aid navigating the contents. The following is an overview of the changes:

Subparagraph Breakdowns/Nested Lists Hierarchy

- First-level breakdowns are designated as (a), (b), (c), etc., as in the past.
- Second-level breakdowns are designated as (1), (2), (3), etc., as in the past.
- Third-level breakdowns are now designated as (-a), (-b), (-c), etc.
- Fourth-level breakdowns are now designated as (-1), (-2), (-3), etc.
- Fifth-level breakdowns are now designated as (+a), (+b), (+c), etc.
- Sixth-level breakdowns are now designated as (+1), (+2), etc.

Footnotes

With the exception of those included in the front matter (roman-numbered pages), all footnotes are treated as endnotes. The endnotes are referenced in numeric order and appear at the end of each BPVC section/subsection.

Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committees

Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committees has been moved to the front matter. This information now appears in all Boiler Code Sections (except for Code Case books).

Cross-References

It is our intention to establish cross-reference link functionality in the current edition and moving forward. To facilitate this, cross-reference style has changed. Cross-references within a subsection or subarticle will not include the designator/identifier of that subsection/subarticle. Examples follow:

- *(Sub-)Paragraph Cross-References.* The cross-references to subparagraph breakdowns will follow the hierarchy of the designators under which the breakdown appears.
  - If subparagraph (-a) appears in X.1(c)(1) and is referenced in X.1(c)(1), it will be referenced as (-a).
  - If subparagraph (-a) appears in X.1(c)(1) but is referenced in X.1(c)(2), it will be referenced as (1)(-a).
  - If subparagraph (-a) appears in X.1(c)(1) but is referenced in X.1(e)(1), it will be referenced as (c)(1)(-a).
  - If subparagraph (-a) appears in X.1(c)(1) but is referenced in X.2(c)(2), it will be referenced as X.1(c)(1)(-a).
- *Equation Cross-References.* The cross-references to equations will follow the same logic. For example, if eq. (1) appears in X.1(a)(1) but is referenced in X.1(b), it will be referenced as eq. (a)(1)(1). If eq. (1) appears in X.1(a)(1) but is referenced in a different subsection/subarticle/paragraph, it will be referenced as eq. X.1(a)(1)(1).

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MANDATORY APPENDICES

MANDATORY APPENDIX I
DESIGN FATIGUE CURVES

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Table I-9.0
Tabulated Values of S_a, ksi, From Figures I-9.1 through I-9.4

		Number of Cycles [Note (1)]																	
Figure	Curve	1E1	2E1	5E1	1E2	2E2	5E2	8.5E2 [Note (2)]	1E3	2E3	5E3	1E4	1.2E4 [Note (2)]	2E4	5E4	1E5	2E5	5E5	1E6
I-9.1	(See Table I-9.1)	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
I-9.2	(See Table I-9.2)	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
I-9.3	S_y = 18.0 ksi	260	190	125	95	73	52	...	44	36	28.5	24.5	...	21	17	15	13.5	12.5	12.0
I-9.3	S_y = 30.0 ksi	260	190	125	95	73	52	...	44	36	28.5	24.5	...	19.5	15	13	11.5	9.5	9.0
I-9.3	S_y = 45.0 ksi	260	190	125	95	73	52	46	39	24.5	15.5	12	...	9.6	7.7	6.7	6.0	5.2	5.0
I-9.4	MNS ≤ 2.7 S_m [Note (3)]	1150	760	450	320	225	143	...	100	71	45	34	...	27	22	19	17	15	13.5
I-9.4	MNS = 3 S_m [Note (3)]	1150	760	450	300	205	122	...	81	55	33	22.5	...	15	10.5	8.4	7.1	6	5.3

GENERAL NOTES:

(a) All notes on the referenced figures apply to these data.

(b) Interpolation between tabular values is permissible based upon data representation by straight lines on a log-log plot. Accordingly, for $S_i > S > S_j$

$$\frac{N}{N_i} = \left(\frac{N_j}{N_i}\right)^{\left[\log\left(S_i/S\right)\right]/\log\left(S_i/S_j\right)}$$

where S, S_i, and S_j are values of S_a; N, N_i, and N_j are corresponding numbers of cycles from design fatigue data.

Example: From the data given in the Table above, use the interpolation equation above to find the number of cycles N for S_a = 53.5 ksi when UTS ≤ 80 ksi in Figure I-9.1:

$$\frac{N}{2000} = \left(\frac{5000}{2000}\right)^{\log\left(64/53.5\right)/\log\left(64/48\right)}$$

$$N = 3540 \text{ cycles}$$

NOTES:

(1) The number of cycles indicated shall be read as follows: IEJ = I × 10^J, e.g., 5E2 = 5 × 10^2 or 500.

(2) These data points are included to provide accurate representation of curves at branches or cusps.

(3) MNS is the Maximum Nominal Stress.

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Table I-9.0M
Tabulated Values of S_a, MPa, From Figures I-9.1M through I-9.4M

		Number of Cycles [[Note (1)]]																	
Figure	Curve	1E1	2E1	5E1	1E2	2E2	5E2	8.5E2 [Note (2)]	1E3	2E3	5E3	1E4	1.2E4 [Note (2)]	2E4	5E4	1E5	2E5	5E5	1E6
I-9.1M	(See Table I-9.1)	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
I-9.2M	(See Table I-9.2)	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
I-9.3M	S_y = 124 MPa	1793	1310	862	655	503	359	...	303	248	197	169	...	145	117	103	93	86	83
I-9.3M	S_y = 207 MPa	1793	1310	862	655	503	359	...	303	248	197	169	...	134	103	90	79	66	62
I-9.3M	S_y = 310 MPa	1793	1310	862	655	503	359	317	269	169	107	83	...	66	53	46	41	36	34
I-9.4M	MNS < 2.7 S_m [Note (3)]	7929	5240	3103	2206	1551	986	...	689	490	310	234	...	186	152	131	117	103	93
I-9.4M	MNS = 3 S_m [Note (3)]	7929	5240	3103	2068	1413	841	...	558	379	228	155	...	103	72	58	49	41	37

GENERAL NOTES:

(a) All notes on the referenced figures apply to these data.

(b) Interpolation between tabular values is permissible based upon data representation by straight lines on a log-log plot. Accordingly, for $S_i > S > S_j$

$$\frac{N}{N_i} = \left(\frac{N_i}{N_j}\right)^{\left[\log\left(S_i/S\right)\right]/\log\left(S_i/S_j\right)}$$

where S, S_i, and S_j are values of S_a; N, N_i, and N_j are corresponding numbers of cycles from design fatigue data.

Example: From the data given in the Table above, use the interpolation equation above to find the number of cycles N for S_a = 369 MPa when UTS ≤ 552 MPa in Figure I-9.1M:

$$\frac{N}{2000} = \left(\frac{5000}{2000}\right)^{\log\left(441/369\right)/\log\left(441/331\right)}$$

$$N = 3535 \text{ cycles}$$

NOTES:

(1) The number of cycles indicated shall be read as follows: IEJ = I × 10^J, e.g., 5E2 = 5 × 10^2 or 500.

(2) These data points are included to provide accurate representation of curves at branches or cusps.

(3) MNS is the Maximum Nominal Stress.

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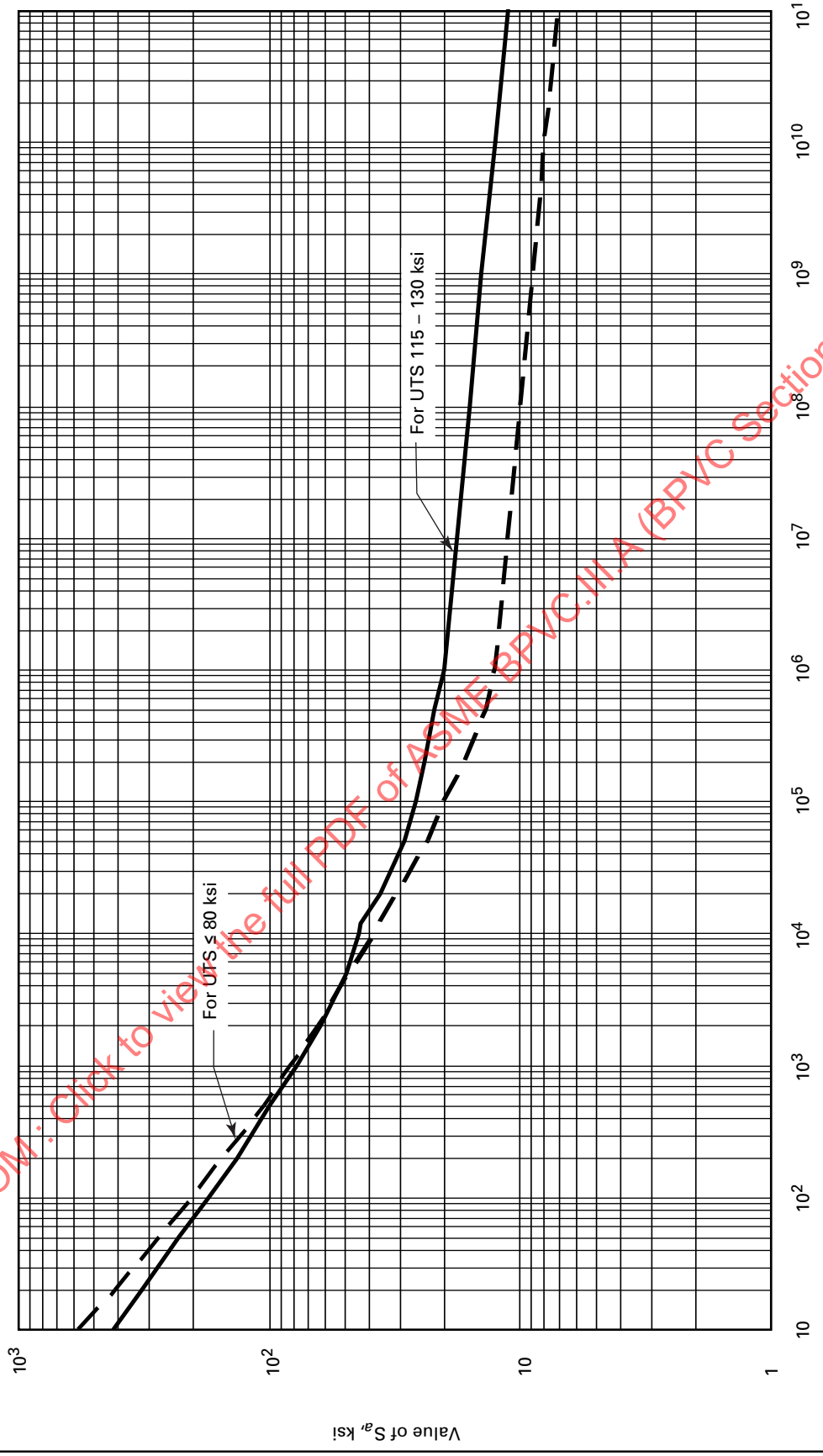
Figure I-9.1
Design Fatigue Curves for Carbon, Low Alloy, and High Tensile Steels for Metal Temperatures Not Exceeding 700°F



GENERAL NOTES:
(a) $E = 30 \times 10^6$ psi
(b) Interpolate for UTS 80.0 ksi to 115.0 ksi.
(c) Table I-9.1 contains tabulated values and an equation for an accurate interpolation of these curves.

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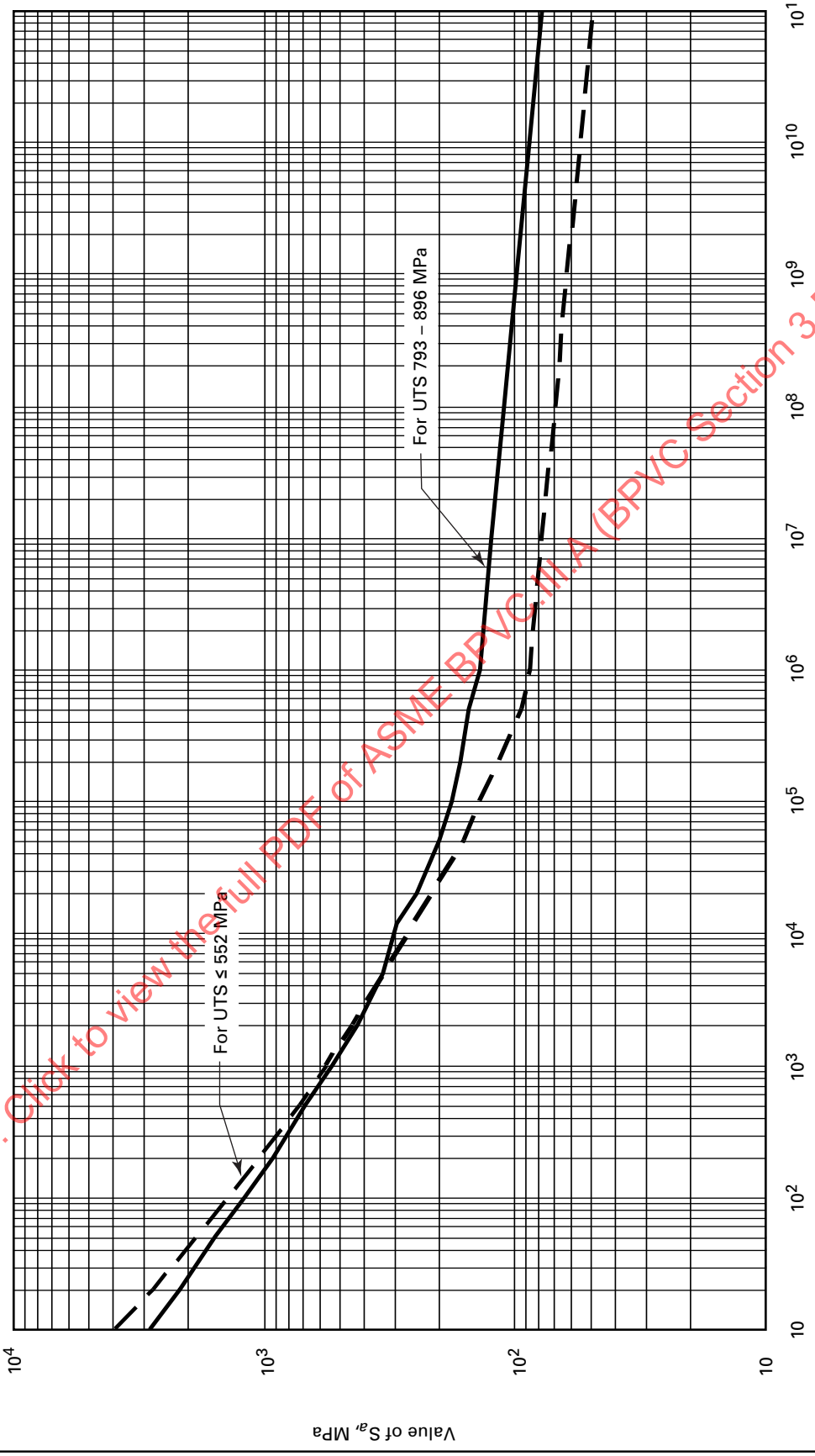
Figure I-9.1M
Design Fatigue Curves for Carbon, Low Alloy, and High Tensile Steels for Metal Temperatures Not Exceeding 370°C



GENERAL NOTES:
(a) $E = 207 \times 10^3$ MPa
(b) Interpolate for UTS 552 MPa to 793 MPa.
(c) Table I-9.1 contains tabulated values and an equation for an accurate interpolation of these curves.

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Table I-9.1
Tabulated Values of S_a, ksi (MPa), From Figures I-9.1 and I-9.1M

Number of Cycles [Note (1)]	UTS 115 ksi to 130 ksi (UTS 793 MPa to 896 MPa)	UTS ≤ 80 ksi (UTS ≤ 552 MPa)
1E1	420 (2 896)	580 (3 999)
2E1	320 (2 206)	410 (2 827)
5E1	230 (1 586)	275 (1 896)
1E2	175 (1 207)	205 (1 413)
2E2	135 (931)	155 (1 069)
5E2	100 (689)	105 (724)
1E3	78 (538)	83 (572)
2E3	62 (427)	64 (441)
5E3	49 (338)	48 (331)
1E4	44 (303)	38 (262)
1.2E4 [Note (2)]	43 (296)	
2E4	36 (248)	31 (214)
5E4	29 (200)	23 (159)
1E5	26 (179)	20 (138)
2E5	24 (165)	16.5 (114)
5E5	22 (152)	13.5 (93)
1E6	20 (138)	12.5 (86)
1E7	17.8 (123)	11.1 (77)
1E8	15.9 (110)	9.9 (68)
1E9	14.2 (98)	8.8 (61)
1E10	12.6 (87)	7.9 (54)
1E11	11.2 (77)	7.0 (48)

GENERAL NOTES:

(a) All notes in Figures I-9.1 and I-9.1M apply to this data.

(b) Interpolation between tabular values is permissible based upon data representation by straight lines on log–log plot. see Table I-9.0 or Table I-9.0M, General Note (b).

NOTES:

(1) The number of cycles indicated shall be read as follows: IEJ = I × 10^{J}, e.g., 5E6 = 5 × 10^{6} or 5,000,000

(2) These data points are included to provide accurate representation of curves at branches or cusps.

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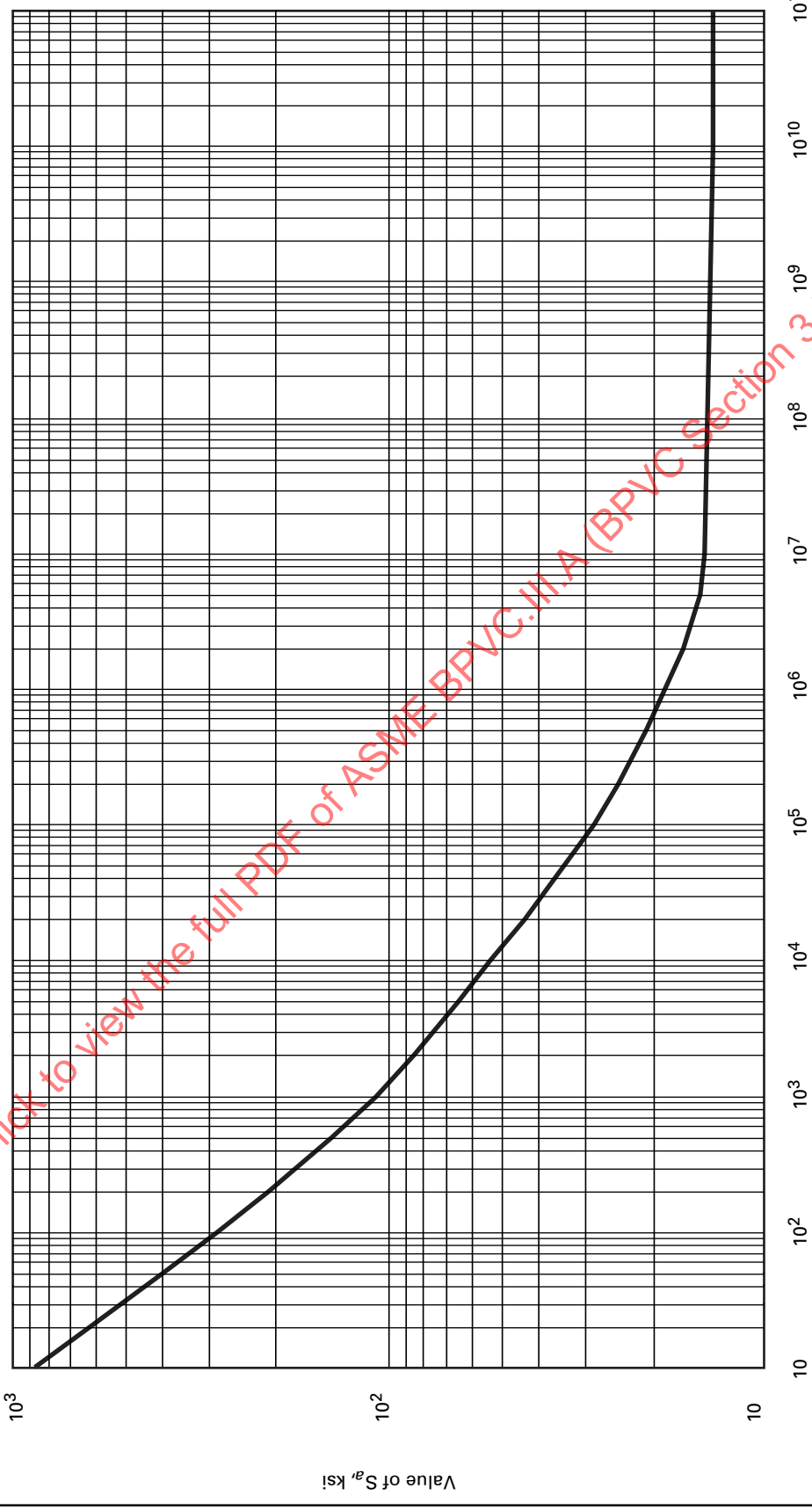
Figure I-9.2
Design Fatigue Curves for Austenitic Steels, Nickel–Chromium–Iron Alloy, Nickel–Iron–Chromium Alloy, and Nickel–Copper Alloy for Temperatures Not Exceeding 800°F



GENERAL NOTES:
(a) E = 28.3 × 10^6 psi
(b) Table I-9.2 contains tabulated values and an equation for an accurate interpolation of this curve.

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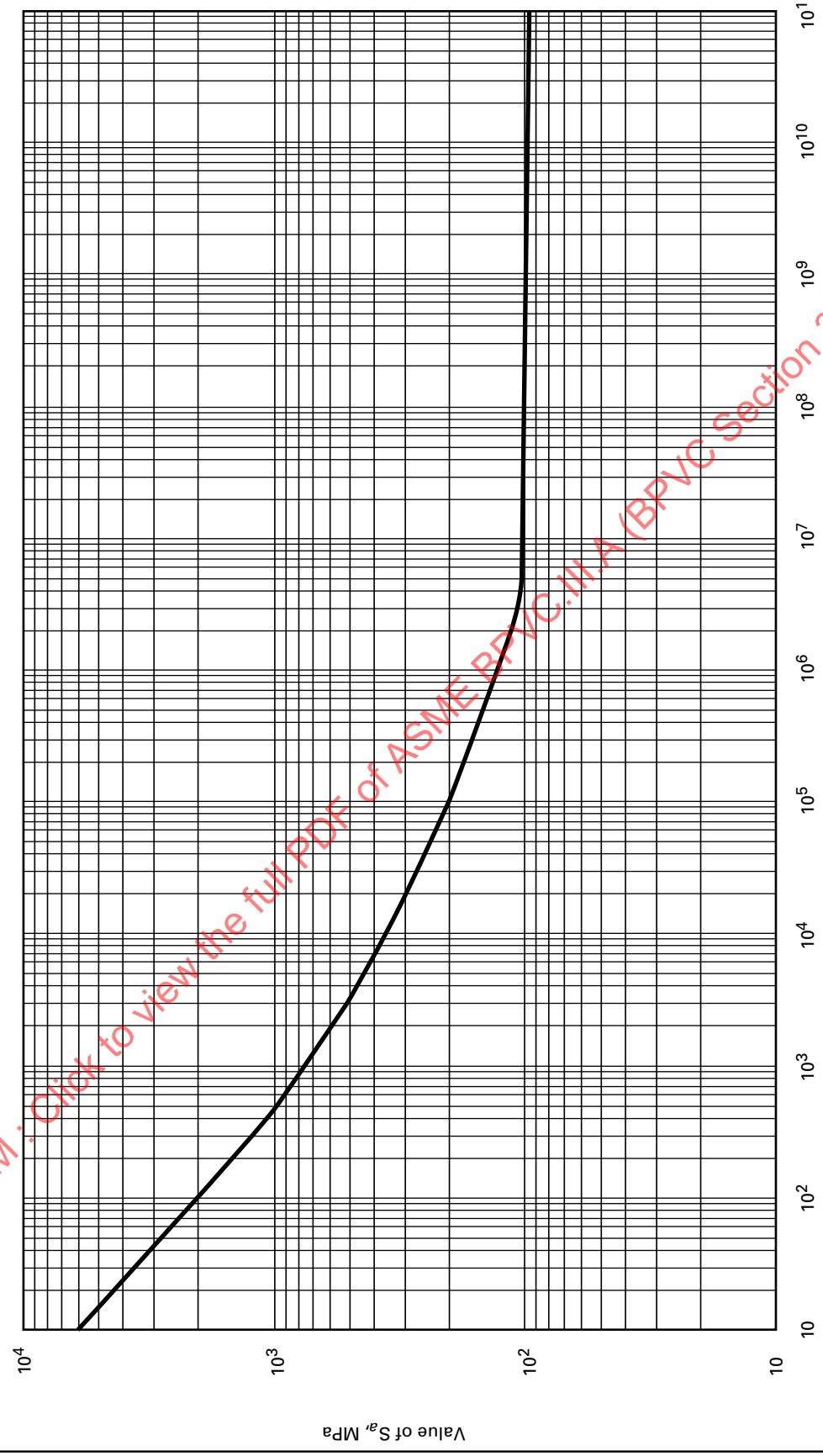
Figure I-9.2M
Design Fatigue Curves for Austenitic Steels, Nickel–Chromium–Iron Alloy, Nickel–Iron–Chromium Alloy, and Nickel–Copper Alloy for Temperatures Not Exceeding 425°C



GENERAL NOTES:
(a) E = 195 × 10^3 MPa
(b) Table I-9.2 contains tabulated values and an equation for an accurate interpolation of this curve.

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Table I-9.2
Tabulated Values of S_a, ksi (MPa), From Figures I-9.2 and I-9.2M

Number of Cycles [Note (1)]	Stress Amplitude
1E1	870 (6 000)
2E1	624 (4 300)
5E1	399 (2 748)
1E2	287 (1 978)
2E2	209 (1 440)
5E2	141 (974)
1E3	108 (745)
2E3	85.6 (590)
5E3	65.3 (450)
1E4	53.4 (368)
2E4	43.5 (300)
5E4	34.1 (235)
1E5	28.4 (196)
2E5	24.4 (168)
5E5	20.6 (142)
1E6	18.3 (126)
2E6	16.4 (113)
5E6	14.8 (102)
1E7	14.4 (99.0)
1E8	14.1 (97.1)
1E9	13.9 (95.8)
1E10	13.7 (94.4)
1E11	13.6 (93.7)

GENERAL NOTES:

(a) All notes in Figures I-9.2 and I-9.2M apply to this data.

(b) Interpolation between tabular values is permissible based upon data representation by straight lines on log–log plot. See Table I-9.0 or Table I-9.0M, General Note (b).

NOTE:

(1) The number of cycles indicated shall be read as follows: IEJ = I × 10^J, e.g., 5E6 = 5 × 10^6 or 5,000,000

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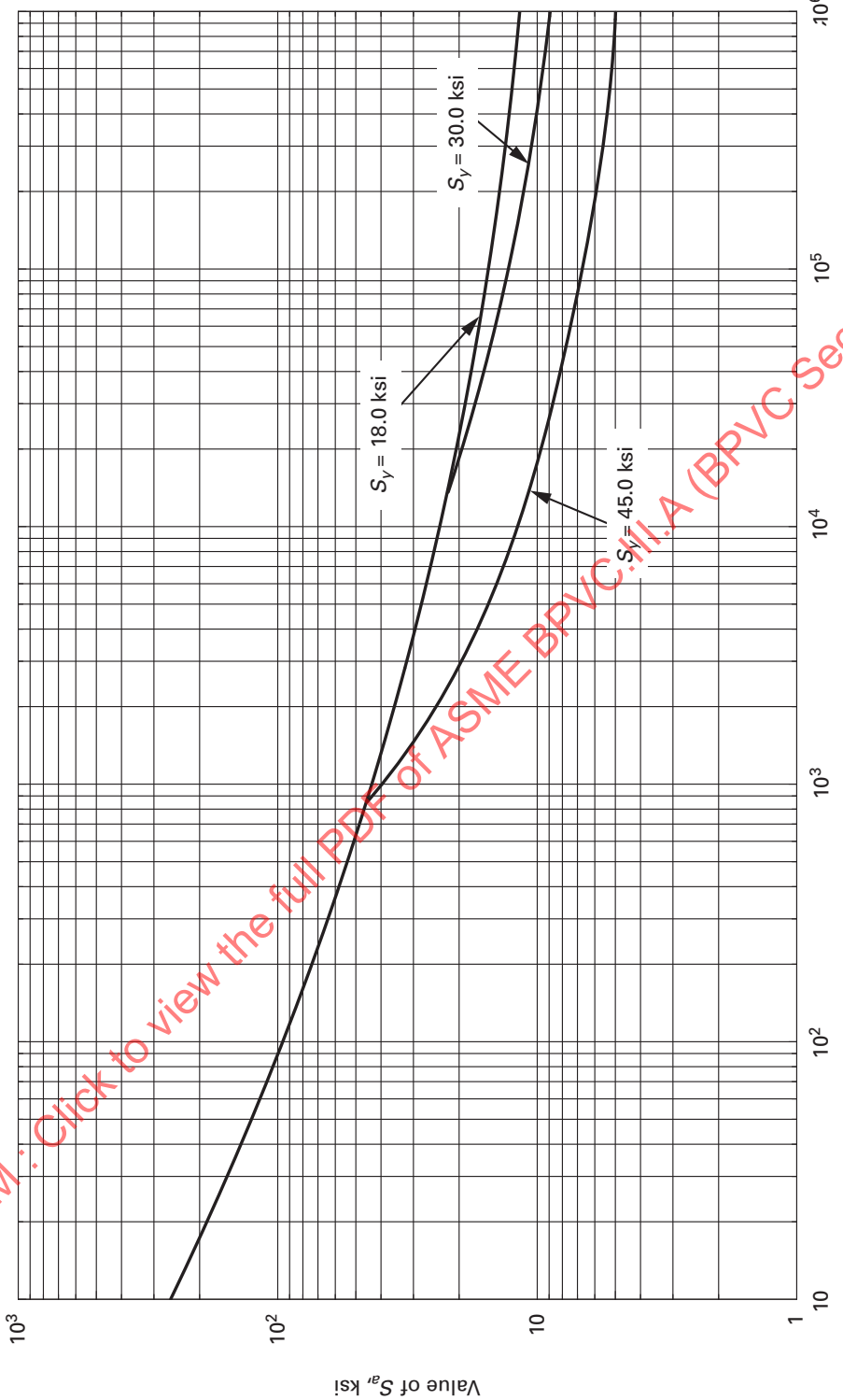
Figure I-9.3
Design Fatigue Curves for Wrought 70 Copper–30 Nickel Alloy for Temperatures Not Exceeding 800°F



GENERAL NOTES:
(a) Care should be exercised in the purchase of this material to ensure that maximum static yield strength is known. These curves may be interpolated for yield strengths between 30.0 ksi and 45.0 ksi.
(b) E = 20 × 10^6 psi
(c) Table I-9.0 contains tabulated values and an equation for an accurate interpolation of these curves.

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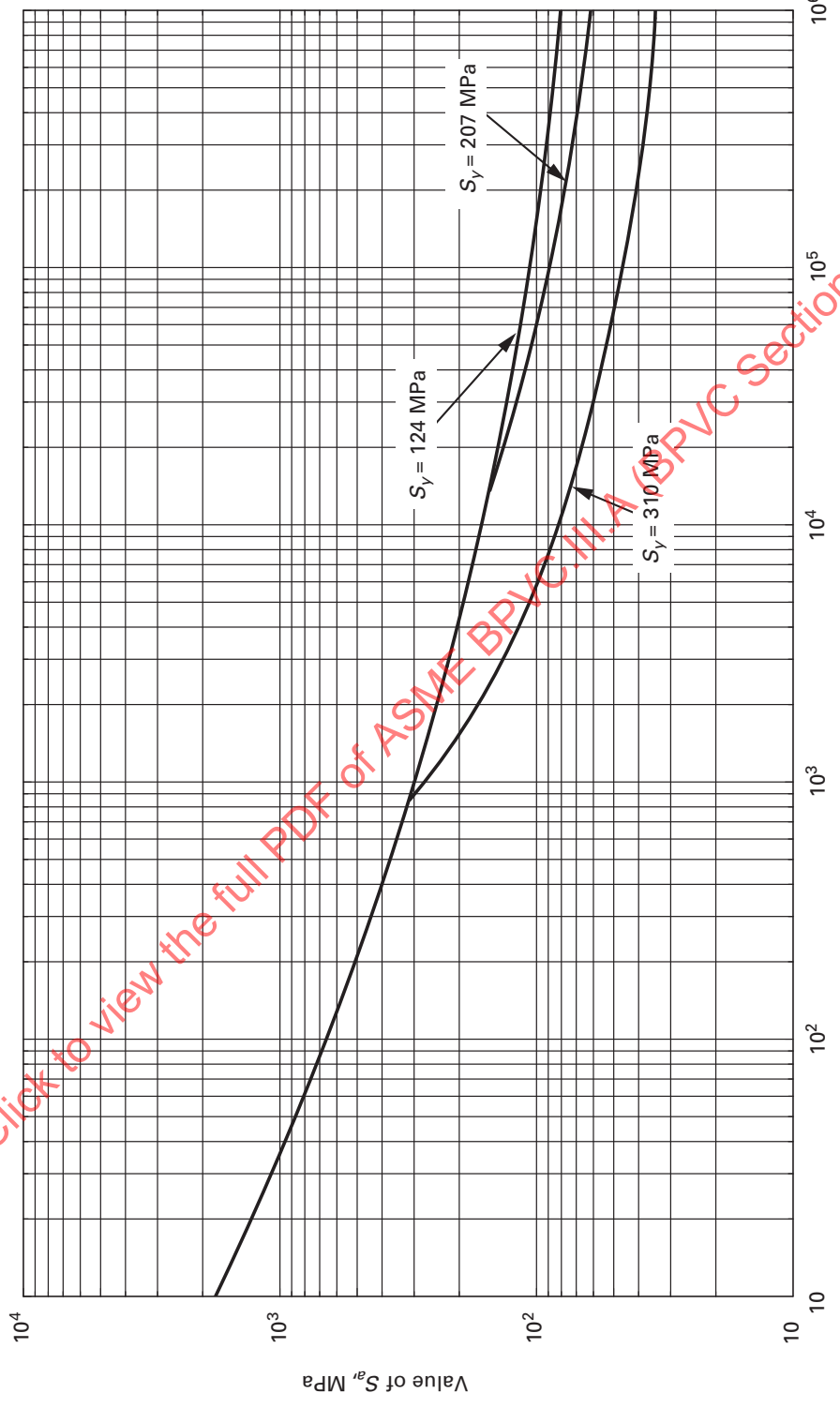
Figure I-9.3M
Design Fatigue Curves for Wrought 70 Copper–30 Nickel Alloy for Temperatures Not Exceeding 425°C



GENERAL NOTES:
(a) Care should be exercised in the purchase of this material to ensure that maximum static yield strength is known. These curves may be interpolated for yield strengths between 207 MPa and 310 MPa.
(b) E = 138×10^3 MPa
(c) Table I-9.0M contains tabulated values and an equation for an accurate interpolation of these curves.

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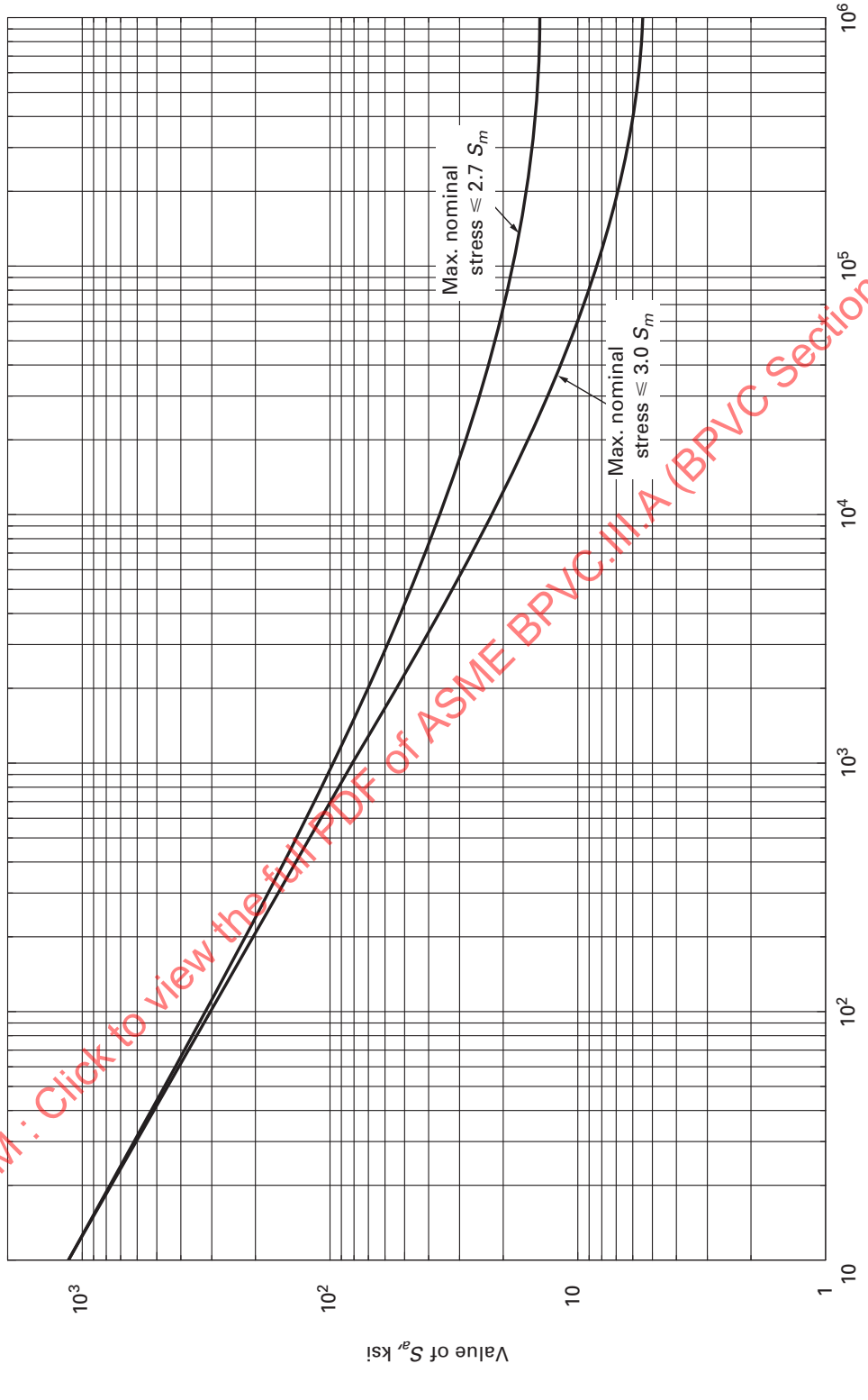
Figure I-9.4
Design Fatigue Curves for High Strength Steel Bolting for Temperatures Not Exceeding 700°F



GENERAL NOTES:
(a) E = 30 × 10^6 psi
(b) Table I-9.0 contains tabulated values and an equation for an accurate interpolation of these curves.

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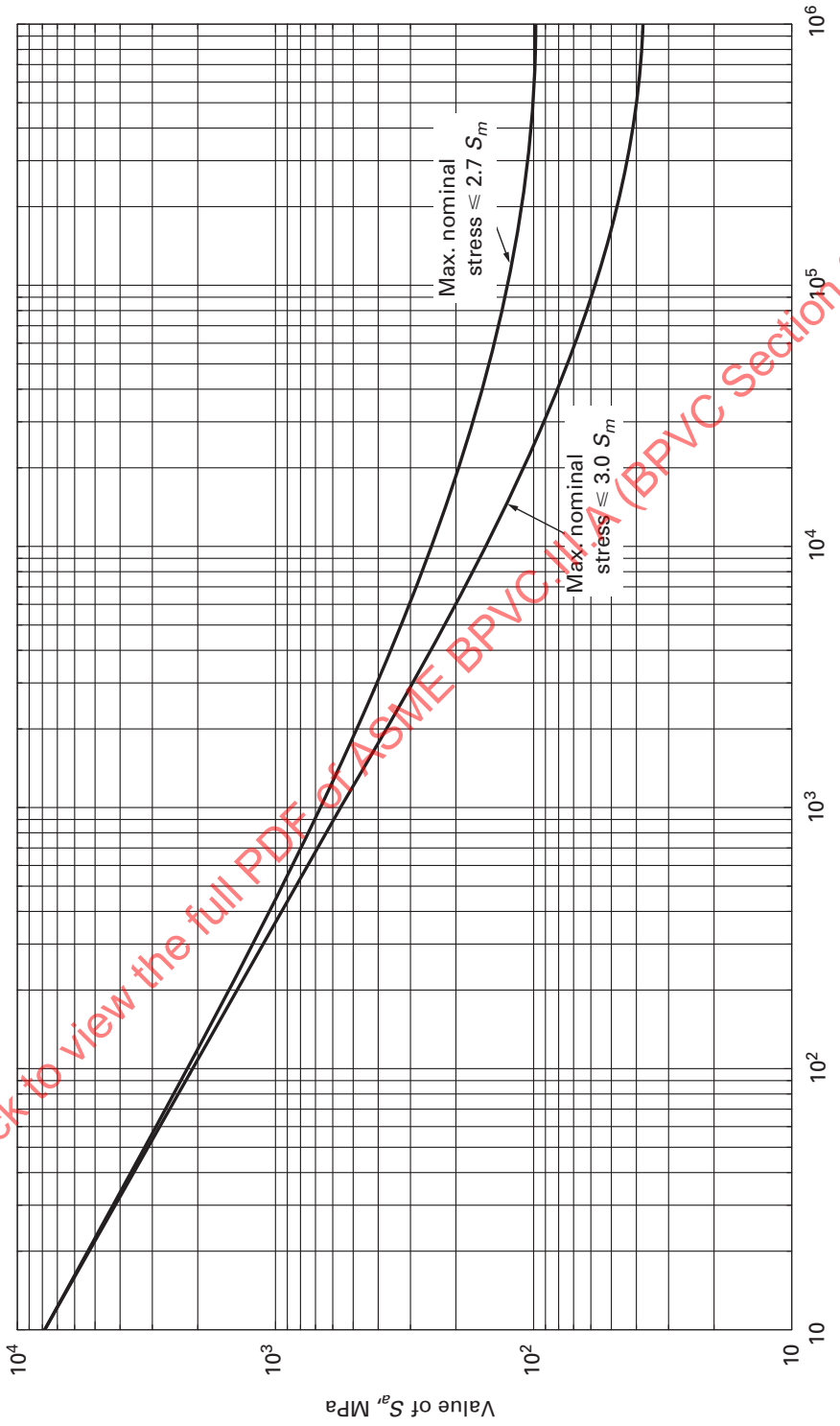
Figure I-9.4M
Design Fatigue Curves for High Strength Steel Bolting for Temperatures Not Exceeding 370°C



GENERAL NOTES:
(a) E = 207 × 10^3 MPa
(b) Table I-9.0M contains tabulated values and an equation for an accurate interpolation of these curves.

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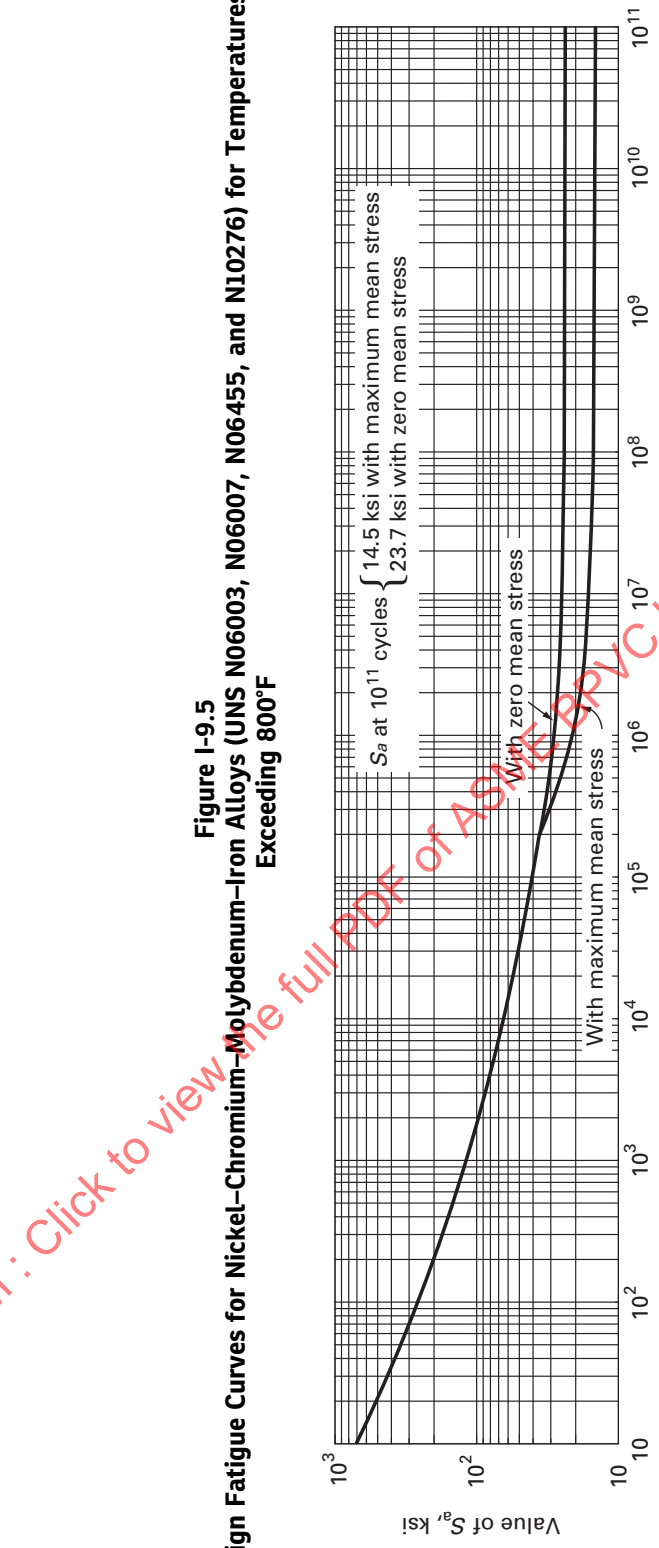
Figure I-9.5
Design Fatigue Curves for Nickel–Chromium–Molybdenum–Iron Alloys (UNS N06003, N06007, N06455, and N10276) for Temperatures Not Exceeding 800°F



GENERAL NOTES:
(a) E = 28.3 × 10^6 psi
(b) Table I-9.5 contains tabulated values and an equation for an accurate interpolation of these curves.

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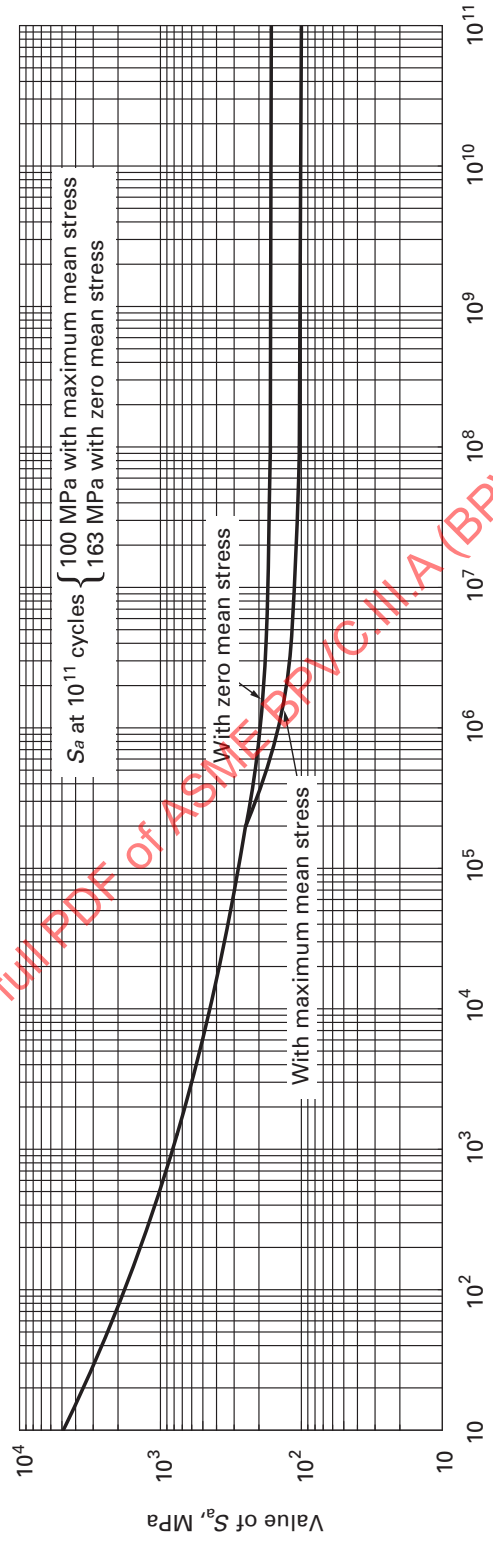
Figure I-9.5M
Design Fatigue Curves for Nickel–Chromium–Molybdenum–Iron Alloys (UNS N06003, N06007, N06455, and N10276) for Temperatures Not Exceeding 425°C



GENERAL NOTES:
(a) E = 195 × 10^3 MPa
(b) Table I-9.5 contains tabulated values and an equation for an accurate interpolation of these curves.

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Table I-9.5
Tabulated Values of S_a, ksi (MPa), From Figures I-9.5 and I-9.5M

Number of Cycles [Note (1)]	Zero Mean Stress	Maximum Mean Stress
1E1	708.0 (4 881)	708.0 (4 881)
2E1	512.0 (3 530)	512.0 (3 530)
5E1	345.0 (2 379)	345.0 (2 379)
1E2	261.0 (1 800)	261.0 (1 800)
2E2	201.0 (1 386)	201.0 (1 386)
5E2	148.0 (1 020)	148.0 (1 020)
1E3	119.0 (820)	119.0 (820)
2E3	97.0 (669)	97.0 (669)
5E3	76.0 (524)	76.0 (524)
1E4	64.0 (441)	64.0 (441)
2E4	56.0 (386)	56.0 (386)
5E4	46.3 (319)	46.3 (319)
1E5	40.8 (281)	40.8 (281)
2E5	35.9 (248)	35.9 (248)
5E5	31.0 (214)	26.0 (179)
1E6	28.2 (194)	20.7 (143)
2E6	26.9 (185)	18.7 (129)
5E6	25.7 (177)	17.0 (117)
1E7	25.1 (173)	16.2 (112)
2E7	24.7 (170)	15.7 (108)
5E7	24.3 (168)	15.3 (105)
1E8	24.1 (166)	15.0 (103)
1E11	23.7 (163)	14.5 (100)

GENERAL NOTE: Interpolation between tabular values is permissible based upon data representation by straight lines on a log–log plot. See Table I-9.1, General Note (b).

NOTE:
(1) The number of cycles indicated shall be read as follows:
IEJ = I × 10^J, e.g., 5E6 = 5 × 10^6 or 5,000,000

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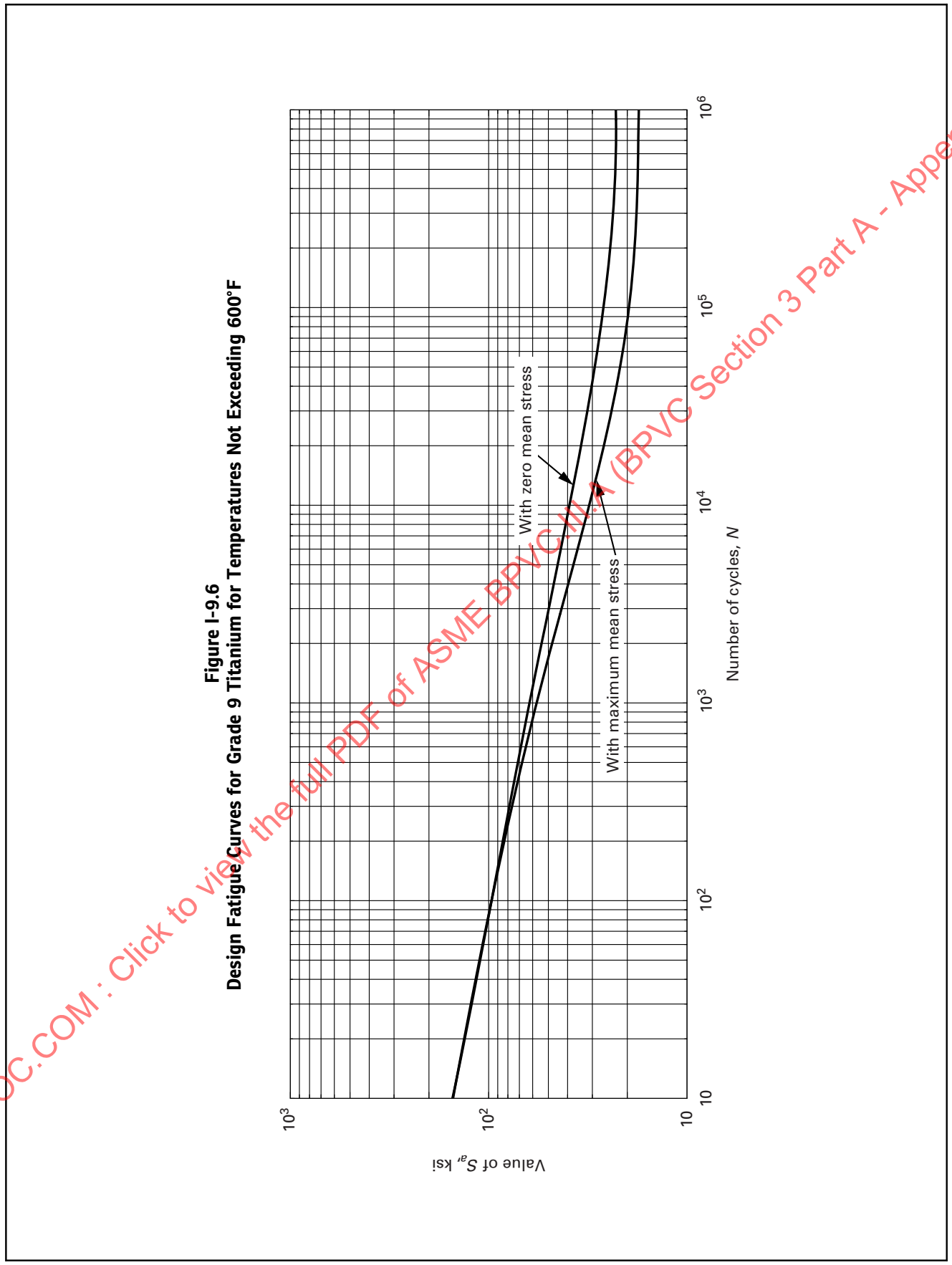

Figure I-9.6
Design Fatigue Curves for Grade 9 Titanium for Temperatures Not Exceeding 600°F

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Figure I-9.6M
Design Fatigue Curves for Grade 9 Titanium for Temperatures Not Exceeding 315°C

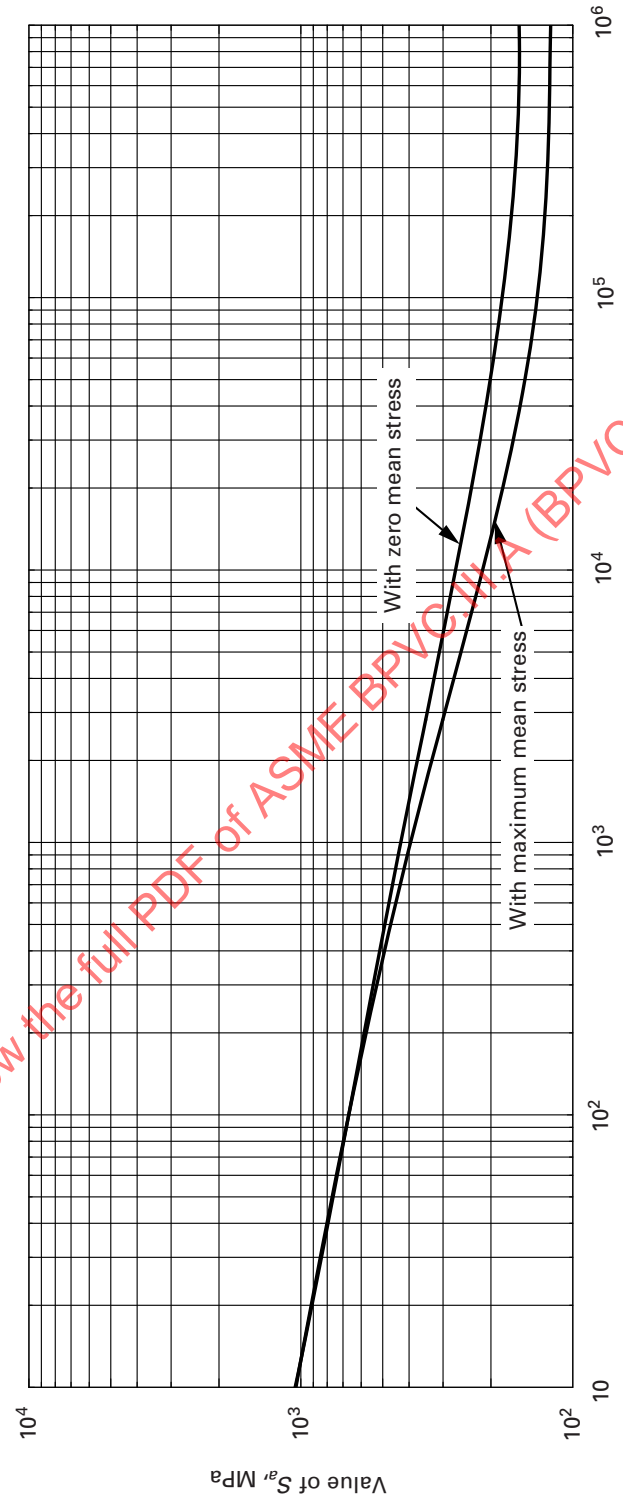

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Table I-9.6
Tabulated Values of S_a, ksi (MPa), for Grade 9 Titanium From Figures I-9.6 and I-9.6M

Number of Cycles	Zero Mean Stress	Maximum Mean Stress
10	151.6 (1 045)	151.6 (1 045)
20	132.4 (913)	132.4 (913)
50	110.8 (764)	110.8 (764)
100	96.8 (667)	96.8 (667)
200	84.6 (583)	84.6 (583)
500	70.8 (488)	67.9 (468)
1000	61.9 (427)	56.7 (391)
2000	54.2 (374)	47.3 (326)
5000	45.2 (312)	37.4 (258)
10000	39.4 (272)	31.4 (216)
20000	34.4 (237)	26.6 (183)
50000	28.9 (199)	21.8 (150)
100000	25.8 (178)	19.1 (132)
200000	24.6 (170)	18.5 (128)
500000	23.4 (161)	17.9 (123)
1000000	22.6 (156)	17.4 (120)

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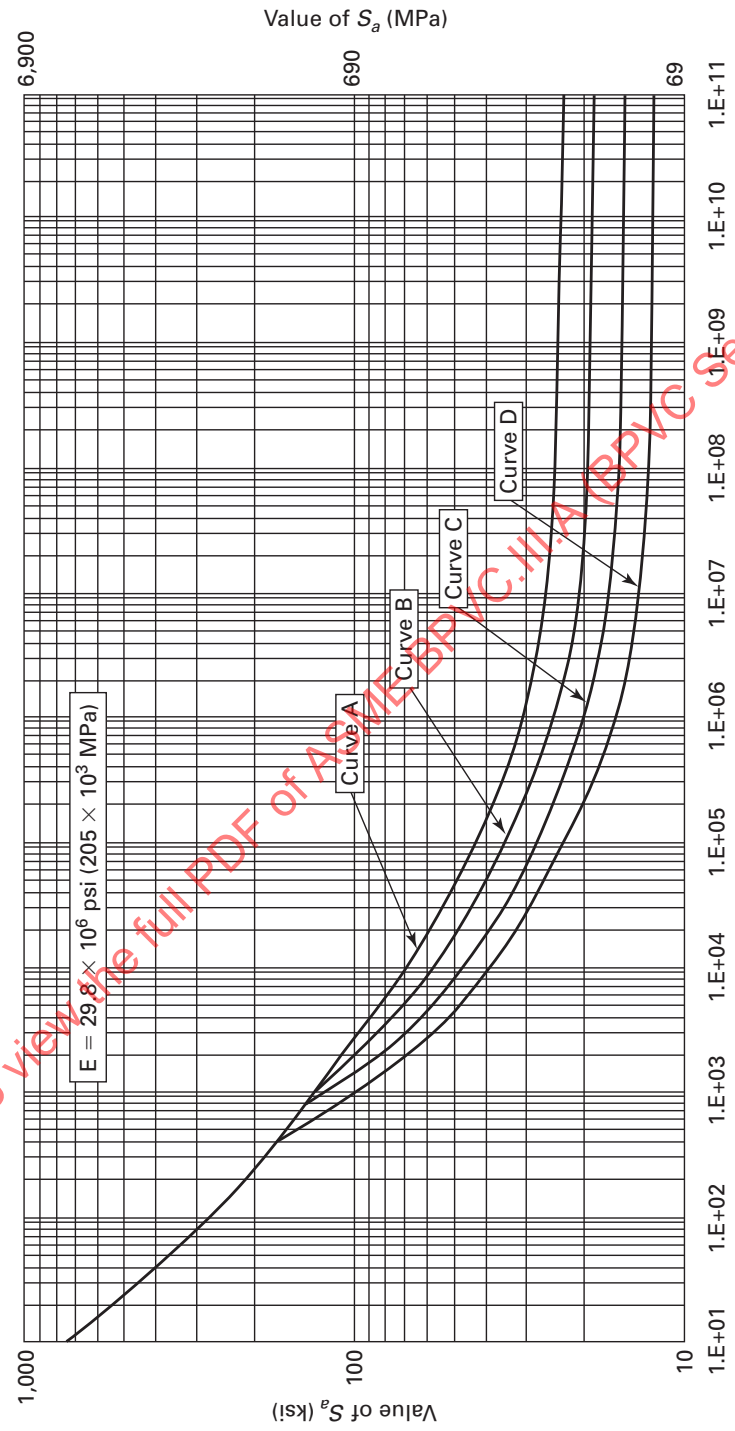
Figure I-9.7
Design Fatigue Curves for Nickel-Chromium Alloy 718 (SB-637 UNS N07718) for Design of 2 in. (50 mm) and Smaller Diameter Bolting for Temperatures Not Exceeding 800°F (427°C)



GENERAL NOTE: Table I-9.7 contains tabulated values for accurate interpolation of these curves.

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Table I-9.7
Tabulated Values of S_a, ksi (MPa), From Figure I-9.7

U.S. Customary Units					SI Units				
Mean Stress	$\sigma_{max} \le 100$ ksi	$\sigma_{max} \le 120$ ksi	$\sigma_{max} \le 135$ ksi	$\sigma_{max} \le 147$ ksi	Mean Stress	$\sigma_{max} \le 690$ MPa	$\sigma_{max} \le 830$ MPa	$\sigma_{max} \le 930$ MPa	$\sigma_{max} \le 1\,015$ MPa
Number of Cycles [Note (1)]	Curve A S_a, ksi	Curve B S_a, ksi	Curve C S_a, ksi	Curve D S_a, ksi	Number of Cycles [Note (1)]	Curve A S_a, MPa	Curve B S_a, MPa	Curve C S_a, MPa	Curve D S_a, MPa
1E1	753	753	753	753	1E1	5 191	5 191	5 191	5 191
2E1	540	540	540	540	2E1	3 723	3 723	3 723	3 723
5E1	365	365	365	365	5E1	2 516	2 516	2 516	2 516
1E2	273	273	273	273	1E2	1 882	1 882	1 882	1 882
2E2	217	217	217	217	2E2	1 496	1 496	1 496	1 496
4E2	173	173	173	173	4E2	1 192	1 192	1 192	1 192
5E2	160	160	160	155	5E2	1 103	1 103	1 103	1 068
8E2	141	141	141	114	8E2	972	972	972	786
1E3	133	133	121	99	1E3	917	917	834	682
2E3	110	100	84	68	2E3	758	689	579	468
5E3	85	71	58	48	5E3	586	489	399	330
1E4	70	58	48	39	1E4	482	399	330	268
2E4	60	49	39	32	2E4	413	337	268	220
5E4	49	40	32	27	5E4	337	275	220	186
1E5	43	35	28	23	1E5	296	241	193	158
2E5	38	31	25	20	2E5	262	213	172	137
5E5	33	27	22	18	5E5	227	186	151	124
1E6	31	25	20	16	1E6	213	172	137	110
2E6	29	23	19	15	2E6	199	158	131	103
5E6	27.1	21.6	17.5	14.2	5E6	186	148	120	97
1E7	26.3	20.6	16.8	13.8	1E7	181	142	115	95
2E7	25.4	20.2	16.5	13.3	2E7	175	139	113	91
5E7	24.8	19.7	15.9	12.9	5E7	170	135	109	88
1E8	24.6	19.6	15.7	12.7	1E8	169	135	108	87
1E11	22.8	18.3	14.8	12.1	1E11	157	126	102	83

GENERAL NOTE: Interpolation between tabular values is permissible based upon data representation by straight lines on a log-log plot. See Table I-9.1, General Note (b).

NOTE:
(1) The number of cycles indicated shall be read as follows: IEJ = I × 10^J, e.g., 5E6 = 5 × 10^6 or 5,000,000.

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Figure I-9.8
Design Fatigue Curves, ksi, for Ductile Cast Iron

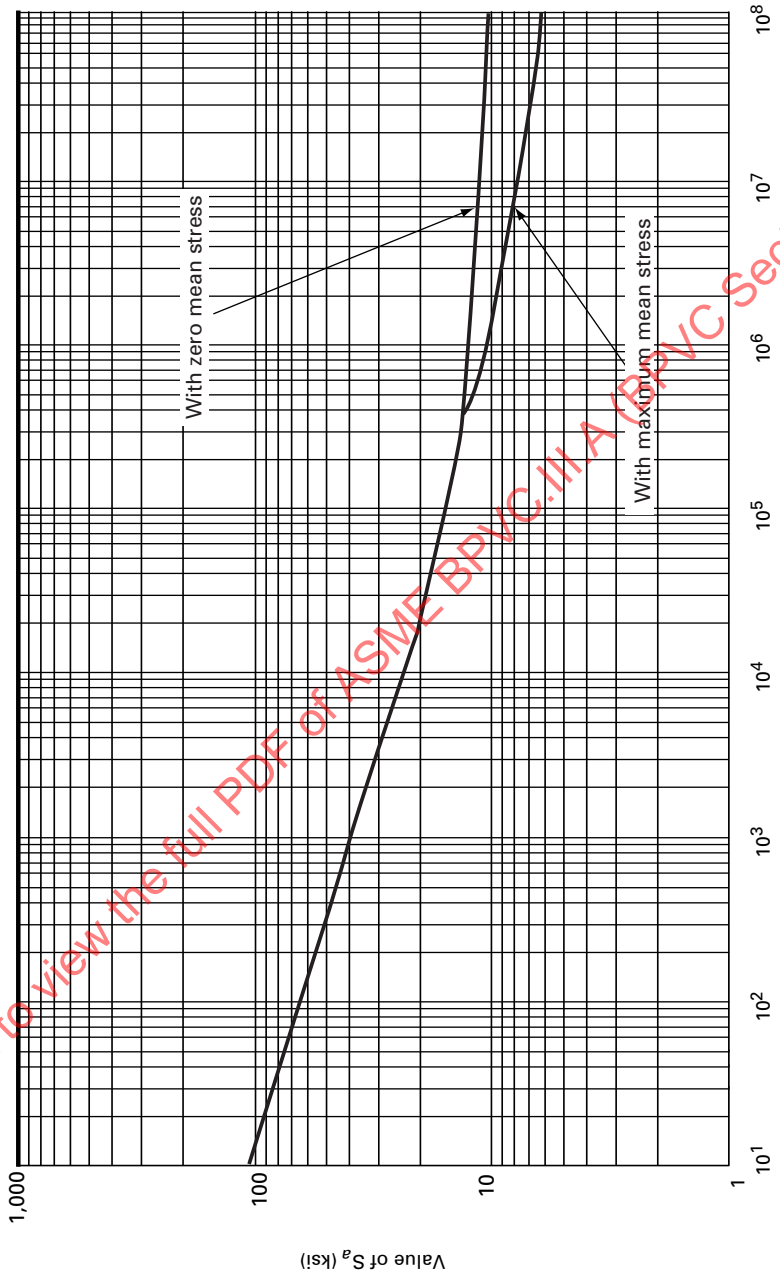

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Figure I-9.8M
Design Fatigue Curves, MPa, for Ductile Cast Iron

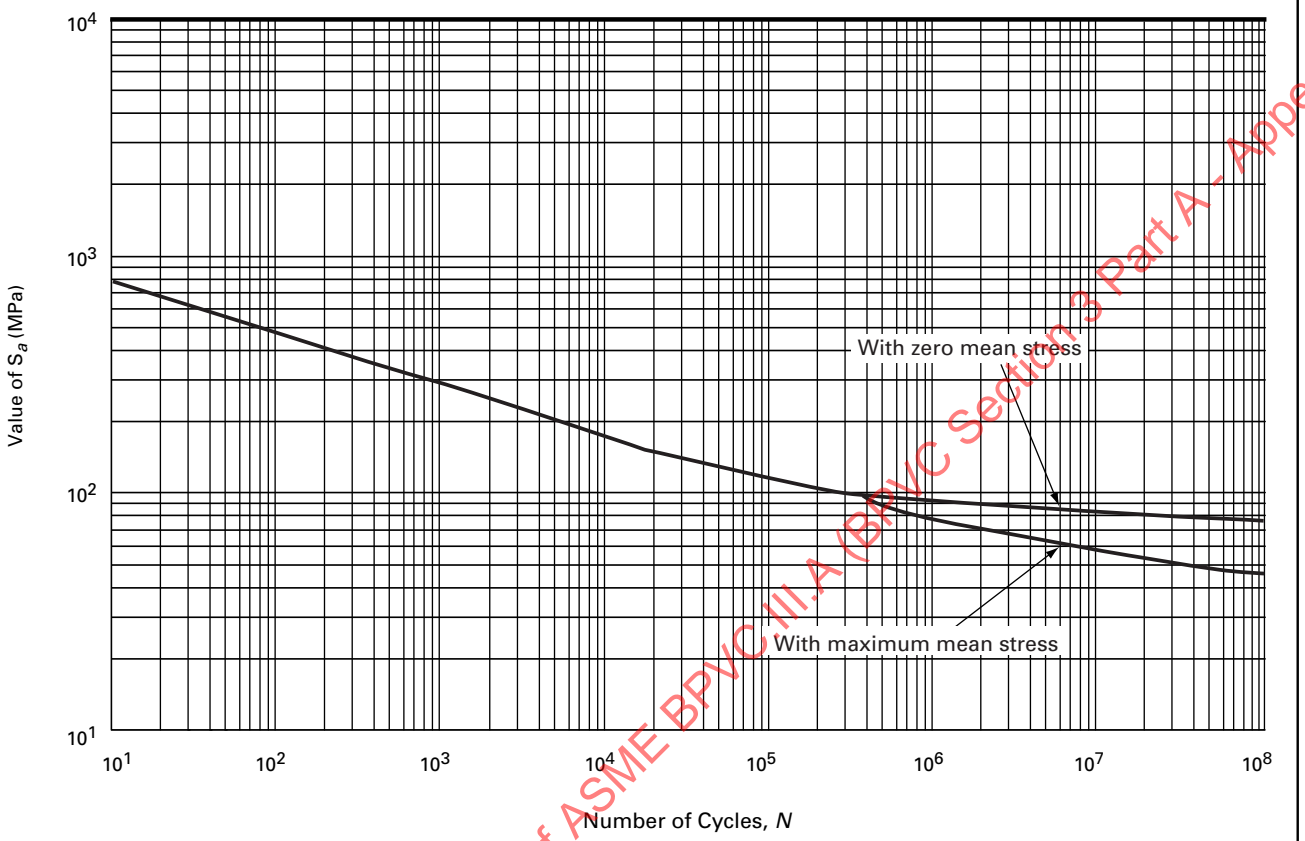

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Table I-9.8
Tabulated Values of S_a, ksi, From Figure I-9.8

Number of Cycles	Values of S_a, ksi	
	Zero Mean Stress	Maximum Mean Stress
1E1	112	112
2E1	94	94
5E1	76	76
1E2	65	65
2E2	56	56
5E2	45	45
1E3	38	38
2E3	32	32
5E3	27	27
1E4	24	24
2E4	21	21
5E4	18	18
1E5	17	17
2E5	15	15
3E5	15	15
5E5	14	13
1E6	13	11
2E6	12	10
5E6	12	9
1E7	11	8
2E7	11	8
5E7	11	7
1E8	11	7

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Table I-9.8M
Tabulated Values of S_a, MPa, From Figure I-9.8M

Number of Cycles	Values of S_a, MPa	
	Zero Mean Stress	Maximum Mean Stress
1E1	772	772
2E1	649	649
5E1	522	522
1E2	447	447
2E2	386	386
5E2	307	307
1E3	261	261
2E3	223	223
5E3	185	185
1E4	162	162
2E4	144	144
5E4	125	125
1E5	114	114
2E5	105	105
3E5	101	101
5E5	96	88
1E6	90	76
2E6	86	67
5E6	82	60
1E7	79	55
2E7	77	52
5E7	75	49
1E8	73	48

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MANDATORY APPENDIX II

ARTICLE II-1000
EXPERIMENTAL STRESS ANALYSIS

II-1100 INTRODUCTION

II-1110 GENERAL REQUIREMENTS

II-1111 When Experimental Stress Analysis Is Required

The critical or governing stresses in parts for which theoretical stress analysis is inadequate or for which design rules are unavailable shall be substantiated by experimental stress analysis.

II-1112 When Reevaluation Is Not Required

Reevaluation is not required for configurations for which there are available detailed experimental results that are consistent with the requirements of this Article.

II-1113 Discounting of Corrosion Allowances

The test procedures followed and the interpretation of the results shall be such as to discount the effects of material added to the thickness of members, such as corrosion allowance, or of other material which cannot be considered as contributing to the strength of the part.

II-1114 Inspection and Reports

Tests conducted in accordance with this Article need not be witnessed by the Inspector. However, a detailed report of the test procedure and the results obtained shall be included with the Design Report (NCA-3551). The report shall show that the instrumentation used was within calibration.

II-1115 Temperature Limits

Experimental methods shall not be used to justify exceeding applicable temperature limits.

II-1200 PERMISSIBLE TYPES OF NONCYCLIC TESTS AND CALCULATION OF STRESSES

II-1210 TESTS FOR DETERMINING GOVERNING STRESSES

Permissible types of tests for the determination of governing stresses are strain measurement tests and photoelastic tests. Brittle coating tests may be used only for the purpose described in II-1310.

II-1220 TESTS FOR DETERMINING COLLAPSE LOAD

Strain measurement tests may be used for the determination of collapse load. Distortion measurement tests may be used for the determination of collapse load, if it can be clearly shown that the test setup and the instrumentation used will give valid results for the configuration on which the measurements are made.

II-1221 Fatigue Tests for Evaluation of Cyclic Loading

Fatigue tests as described in II-1500 may be used to evaluate the adequacy of a component or portion thereof for cyclic loading.

II-1230 TESTS TO DESTRUCTION

Results of tests to destruction are not acceptable except as provided for piping in NB-3649.

II-1240 CALCULATION OF STRESSES

A modified Poisson's ratio is given by NB-3227.6 and identified as being applicable only to local thermal stresses. It should be noted that some situations can arise in which the use of the modified value of Poisson's ratio is indicated for the calculation of stresses of other than thermal origin. Strictly speaking, this modified value should be used in any calculation which results in stresses that exceed a stress intensity range of $2S_y$. For designs which meet the basic stress limits this modification is important only for local thermal stresses such as skin stresses. When calculating the stress range in an element such as a

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fatigue test model which does not meet the basic stress limits, the effect of the modified Poisson's ratio should be considered.

II-1250 STRAIN MEASUREMENT TEST PROCEDURE

II-1251 Requirements for Strain Gages

Strain gages of any type capable of indicating strains to 0.00005 in./in. (0.00005 mm/mm) may be used. It is recommended that the gage length be such that the maximum strain within the gage length does not exceed the average strain within the gage length by more than 10%. Instrumentation shall be such that both surface principal stresses may be determined at each gage location in the elastic range of material behavior at that gage location. A similar number and orientation of gages at each gage location are required to be used in tests beyond the elastic range of material behavior. The strain gages and cements that are used shall be shown to be reliable for use on the material surface finish and configuration considered to strain values at least 50% higher than those expected.

II-1252 Use of Models for Strain or Distortion Measurements

(a) Except in tests made for the measurement of collapse load, strain gage data may be obtained from the actual component or from a model component of any scale that meets the gage length requirements of II-1251. The model material need not be the same as the component material but shall have an elastic modulus which is either known or has been measured at the test conditions. The requirements of dimensional similitude shall be met as nearly as possible.

(b) In the case of collapse load tests, only full scale models, prototypical in all respects, are permitted unless the experimenter can clearly demonstrate the validity of the scaling laws used.

II-1260 PHOTOELASTIC TEST TECHNIQUES

Either two dimensional or three dimensional techniques may be used in photoelastic testing as long as the model represents the structural effects of the loading.

II-1300 TEST PROCEDURES

II-1310 LOCATION OF TEST GAGES

(a) In tests for determination of governing stresses, sufficient locations on the component shall be investigated to ensure that the measurements are taken at the most critical areas and to permit conservative determination of the bending and peak stress components. The location of the critical areas and the optimum orientation of test gages may be determined by a brittle coating test.

(b) In tests made for the measurement of collapse load, sufficient measurements must be taken so that all areas that have any reasonable probability of indicating a minimum collapse load are adequately covered. If strain gages are used to determine the collapse load, particular care should be given to assure that the measured strains (either membrane, bending or a combination) are actually indicative of the load carrying capacity of the structure. If distortion measurement devices are used, care should be given to assure that it is the change in cardinal dimensions or deflections that is measured, such as diameter or length extension, or beam or plate deflections that are indicative of the tendency of the structure to actually collapse.

II-1320 REQUIREMENTS FOR PRESSURE GAGES

Pressure gages shall meet the requirements of NB-6400.

II-1330 APPLICATION OF PRESSURE OR LOAD

(a) In tests for determining governing stresses, the internal pressure or mechanical load shall be applied in such increments that the variation of strain with load can be plotted so as to establish the ratio of stress to load in the elastic range. If the first loading results in strains that are not linearly proportional to the load, it is permissible to unload and reload successively until the linear proportionality has been established. When frozen stress photoelastic techniques are used, only one load value can be applied, in which case the load shall not be so high as to result in deformations that invalidate the test results.

(b) In tests made for the measurement of collapse load, the proportional load shall be applied in sufficiently small increments so that an adequate number of data points for each gage are available for statistical analysis in the linear elastic range of behavior. All gages should be evaluated prior to increasing the load beyond this value. A least square fit (regression) analysis shall be used to obtain the best fit straight line, and the confidence interval shall be compared to preset values for acceptance or rejection of the strain gage or other instrumentation. Unacceptable instrumentation will be replaced and the replacement instrumentation tested in the same manner.

(c) After all instrumentation has been deemed acceptable, the test should be continued on a strain or displacement controlled basis with adequate time permitted between load changes for all metal flow to be completed.

II-1400 INTERPRETATION OF RESULTS

II-1410 INTERPRETATION TO BE ON ELASTIC BASIS

Linear elastic theory shall be used to determine the design load stresses from the strain gage data. The calculations shall be performed under the assumption that the

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material is elastic. The elastic constants used in the evaluation of experimental data shall be those applicable to the test material at the test temperature.

II-1420 REQUIRED EXTENT OF STRESS ANALYSIS

The extent of experimental stress analysis performed shall be sufficient to determine the governing stresses for which design values are unavailable, as described in II-1111. When possible, combined analytical and experimental methods shall be used to distinguish among primary, secondary, and peak stresses so that each combination of categories can be controlled by the applicable stress limit.

II-1430 CRITERION OF COLLAPSE LOAD

(a) For distortion measurement tests, the loads are plotted as the ordinate and the measured deflections are plotted as the abscissa. For strain gage tests, the loads are plotted as the ordinate and the maximum principal strains on the surface as the abscissa.

(b) The least square fit (regression) line as determined from the data in the linear elastic range is drawn on each plot considered. The angle that the regression line makes with the ordinate is called θ. A second straight line, hereafter called the collapse limit line, is drawn through the intersection of the regression line with the abscissa so that it makes an angle $\phi = \tan^{-1}(2\tan\theta)$ with the ordinate. (See Figure II-1430-1.) The test collapse load is determined from the maximum principal strain or deflection value at the first data point for which there are three successive data points that lie *outside* of the collapse limit line. This first data point is called the *collapse load point*. The test collapse load is taken as the load on the collapse limit line which has the maximum principal strain or deflection of the collapse load point. The collapse load used for design or evaluation purposes shall be the test collapse load multiplied by the ratio of the material yield strength at Design Temperature (Section II, Part D, Subpart 1, Table Y-1) to the test material yield strength at the test temperature. Careful attention shall be given to the actual as-built dimensions of the test model when correlating the collapse load of the test model to that expected for the actual structure being designed.

II-1500 CYCLIC TESTS

II-1510 WHEN CYCLIC TESTS MAY BE USED

(a) Experimental methods constitute a reliable means of evaluating the capability of components and appurtenances to withstand cyclic loading. In addition, when it is desired to use higher peak stresses than can be justified by the methods of II-1200 to II-1400 and the fatigue curves of Mandatory Appendix I, the adequacy of a component, appurtenance, or portion thereof[1] to withstand cyclic loading may be demonstrated by means of a fatigue test. The fatigue test shall not be used, however, as justification for exceeding the allowable values of primary or primary plus secondary stresses.

(b) When a fatigue test is used to demonstrate the adequacy of a component or a portion thereof to withstand cyclic loading, a description of the test shall be included in the Design Report. This description shall contain sufficient detail to show compliance with the requirements of this Subarticle.

II-1520 REQUIREMENTS FOR CYCLIC TESTING OF COMPONENTS

The applicable requirements of (a) through (g) below shall be met.

(a) The test component or portion being tested shall be constructed of material having the same composition and subjected to mechanical working and heat treatment that result in mechanical properties equivalent to those of the material in the prototype component. Geometrical similarity must be maintained, at least in those portions whose ability to withstand cyclic loading is being investigated, and in those adjacent areas which affect the stresses in the portion under test.

(b) The test component or portion thereof shall withstand the number of cycles as set forth in (c) below before failure occurs. Failure is herein defined as a propagation of a crack through the entire thickness, such as would produce a measurable leak in a pressure-retaining member.

(c) The minimum number of cycles N_T (hereinafter referred to as test cycles), which the component shall withstand and the magnitude of the loading (hereinafter referred to as the test loading) to be applied to the component during the test, P_T shall be determined by multiplying the specified service cycles N_D by a specified factor K_{TN} and the specified service loads P_D by K_{TS}. Values of these factors shall be determined by means of the testing parameters ratio diagram, the construction of which is given in (1) through (3) below and is illustrated in Figure II-1520(c)-1. When applicable, the requirements of (d) through (f) shall be met.

(1) Project a vertical line from the specified service cycles N_D on the abscissa of S_a of Mandatory Appendix I to an ordinate value of $K_s \times S_{aD}$. The parameter K_s is determined using (g). Label this point *A*.

(2) Extend a horizontal line through the point *D* until its length corresponds to an abscissa value of $K_n \times N_D$. The parameter K_n is determined using (g). Label this point *B*.

(3) Connect the points *A* and *B*. The segment *AB* embraces all the allowable combinations of K_{TS} and K_{TN}. For accelerated testing, see (f). Any point *C* on this segment

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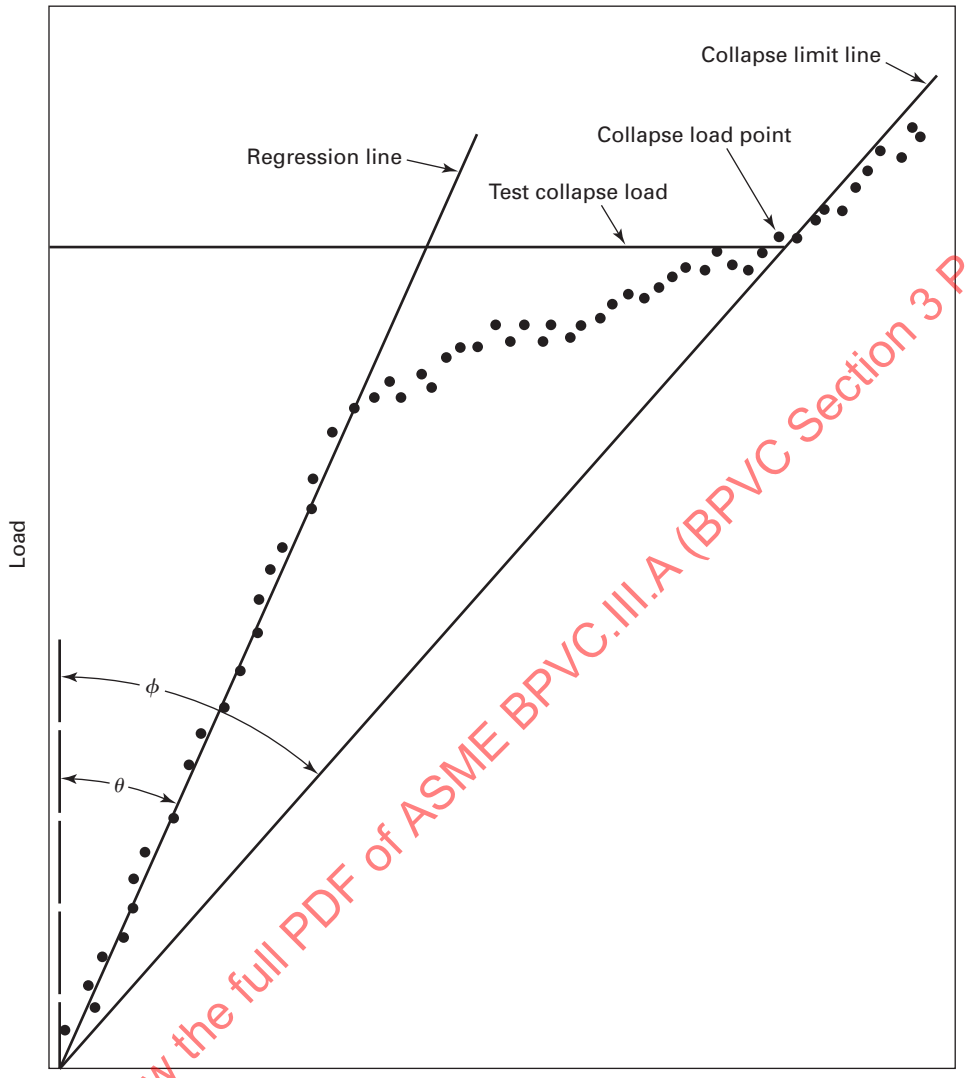
Figure II-1430-1
Construction for II-1430



may be chosen at the convenience of the tester. Referring to Figure II-1520(c)-1, the factors K_{TS} and K_{TN} are defined as follows:

$$K_{TS} = \frac{\text{Value of Ordinate at Point C}}{\text{Value of Ordinate at Point D}} = \frac{S_{aC}}{S_{aD}}$$

and

$$K_{TN} = \frac{\text{Value of Abscissa at Point C}}{\text{Value of Abscissa at Point D}} = \frac{N_C}{N_D}$$

thus,

$$P_T = K_{TS} \times P_D$$

$$N_T = K_{TN} \times N_D$$

(d) It should be noted that, if the test article is not a full size component but a geometrically similar model, then the value P_T shall be adjusted by the appropriate scale factor to be determined from structural similitude principles if the loading is other than pressure. The number of cycles that the component shall withstand during this test without failure shall not be less than N_T while subjected to a cyclic test loading P_T, which shall be adjusted, if required, using model similitude principles if the component is not full size.

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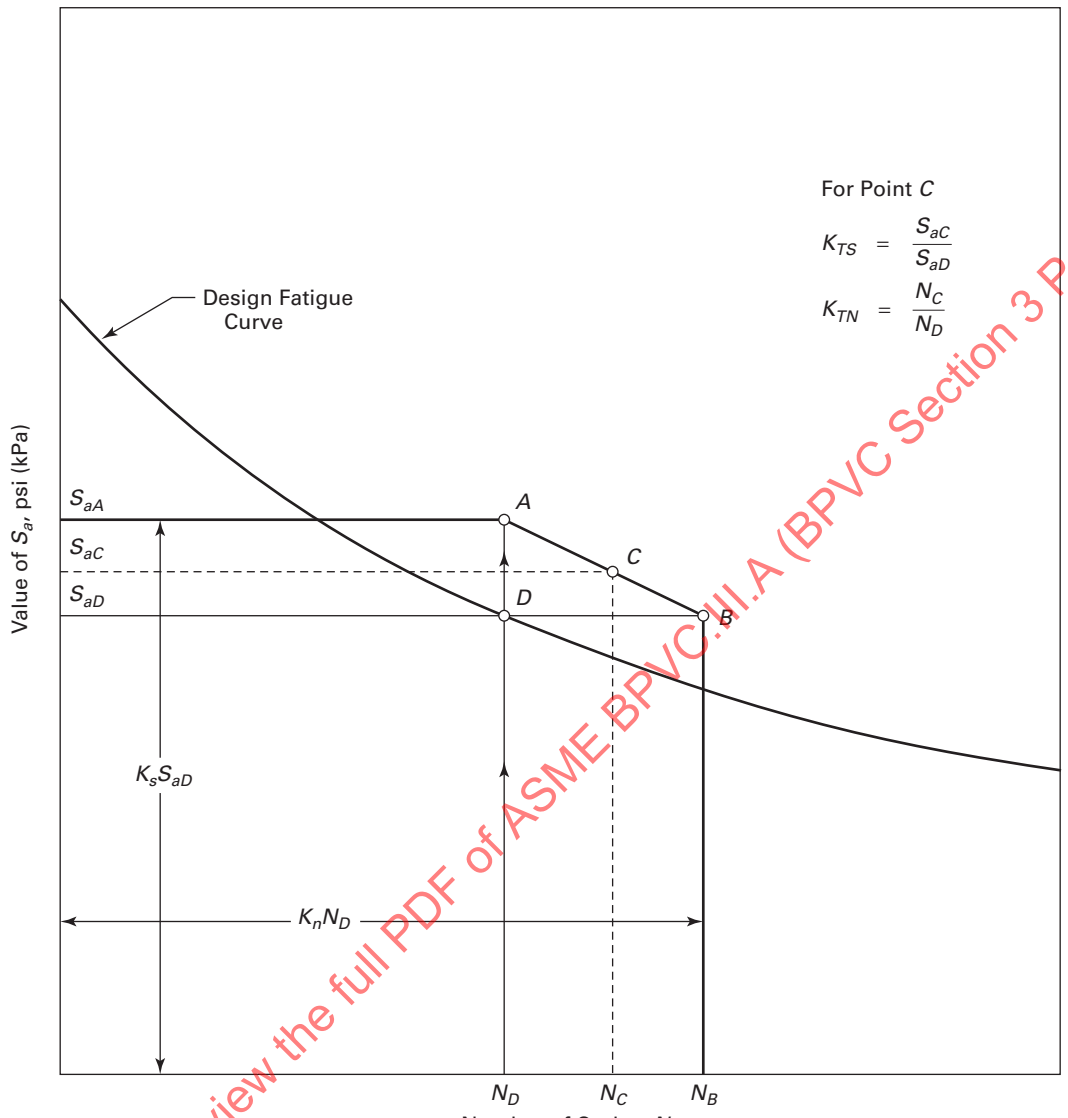
Figure II-1520(c)-1
Construction of the Testing Parameters Ratio Diagram



(e) In certain instances, it may be desirable (or possible) in performing the test to increase only the loading or number of cycles, but not both, in which event two special cases of interest result from the above general case, as described in (1) and (2) below.

(1) Case 1 (Factor Applied to Cycles Only). In this case, $K_{TS} = 1$ and

$$K_{TN} = \frac{\text{Value of Abscissa at Point } C}{\text{Value of Abscissa at Point } D} = \frac{N_C}{N_D}$$

The number of test cycles that the component shall withstand during this test shall not be less than $N_T = K_{TN} \times N_D$, while subjected to the specified service loading, P_D, adjusted as required, if a geometrically similar model is used.

(2) Case 2 (Factor Applied to Loading Only). In this case, $K_{TN} = 1$ and

$$K_{TS} = \frac{\text{Value of Ordinate at Point } C}{\text{Value of Ordinate at Point } D} = \frac{S_{aC}}{S_{aD}}$$

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The component shall withstand a number of cycles at least equal to the number of specified service cycles, N_D, while subjected to a cyclic test loading $P_T = K_{TS} \times P_D$, again adjusted as required, if a geometrically similar model is used.

(f) Accelerated fatigue testing (test cycles N_T are less than specified service cycles N_D) may be conducted if the specified service cycles N_D are greater than 10^4 and the testing conditions are determined by the procedures of (1) through (3) below, which are illustrated in Figure II-1520(c)-2. In this figure, the points *A*, *B*, and *D* correspond to similar labeled points in Figure II-1520(c)-1.

(1) The minimum number of test cycles $N_{T\min}$ shall be

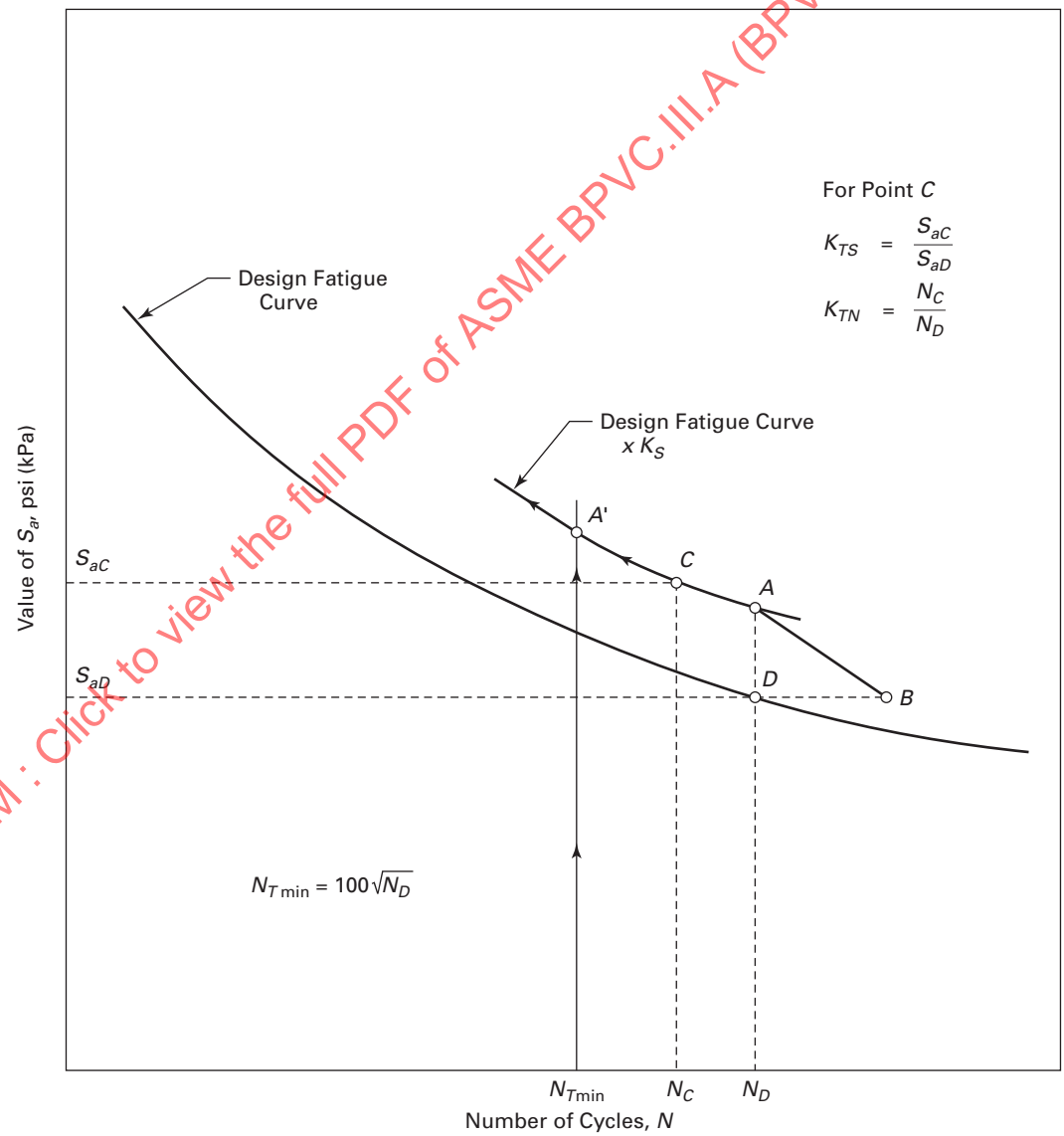
$$N_{T\min} = 100\sqrt{N_D}$$

Project a vertical line through $N_{T\min}$ on the abscissa of S_a versus N diagram such that it intersects and extends beyond the fatigue design curve.

(2) Construct a curve through the point *A* and intersect the vertical projection of N_T [see (1)] by multiplying every point on the fatigue design curve by the factor K_s, which is evaluated according to (g). Label the intersection of this curve and the vertical projection of $N_{T\min}$ as A'.

(3) Any point *C* on the segment A', *A*, *B* determines the allowable combinations of K_{TS} and K_{TN}. The factors K_{TS} and K_{TN} are obtained in the same manner as in (c).

Figure II-1520(c)-2
Construction of the Testing Parameters Ratio Diagram for Accelerated Tests



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(g) The values of K_s and K_n are the multiples of factors that account for the effects of size, surface finish, cyclic rate, temperature, and the number of replicate tests performed. They shall be determined as follows:

$$K_s = K_{sl} \times K_{sf} \times K_{st} \times K_{ss} \times K_{sc}$$

but shall never be allowed to be less than 1.25 and

$$K_n = K_s^{4.3}$$

but shall never be allowed to be less than 2.6, where

K_{sc} = factor for differences in design fatigue curves at various temperatures
K_{sf} = factor for the effect of surface finish
= 1.175 − 0.175 (*SFM/SFP*), where *SFM/SFP* is the ratio of model surface finish to prototype surface finish, in. × 10^{-6} arithmetic average (AA)
K_{sl} = factor for the effect of size on fatigue life
= 1.5 − 0.5 (LM/LP), where LM/LP is the ratio of linear model size to prototype size
K_{ss} = factor for the statistical variation in test results
= 1.470 − 0.044 × number of replicate tests
K_{st} = factor for the effect of test temperature
= (S_{aN} at test temperature)/(S_{aN} at Design Temperature), where S_{aN} equals S_a from applicable fatigue curve at N cycles

No value of K_{sl}, K_{sf}, K_{st}, K_{ss}, or K_{sc} less than 1.0 may be used in calculating K_s.

II-1521 Nomenclature

The symbols defined below are used in II-1520.

K_s, K_n = factors that account for the effects of size, surface finish, cyclic rate, temperature, and the number of replicate tests performed
K_{TS}, K_{TN} = factors used to determine the test loading and test cycles, respectively
N_C = number of cycles at point C
N_D = specified service cycles
N_T = testing cycles
$N_{T\min}$ = minimum number of test cycles
P_D = specified service loading
P_T = test loading
S_{aC} = alternating stress at point C
S_{aD} = alternating stress at point D

II-1600 DETERMINATION OF FATIGUE STRENGTH REDUCTION FACTORS

II-1610 PROCEDURES

Experimental determination of fatigue strength reduction factors shall be in accordance with the procedures of (a) through (e) below.

(a) The test part shall be fabricated from a material within the same P-Number grouping of Section IX, Table QW/QB-422 and shall be subjected to the same heat treatment as the component.

(b) The stress level in the specimen shall be such that the stress intensity does not exceed the limit prescribed by NB-3222.2 and so that failure does not occur in less than 1000 cycles.

(c) The configuration, surface finish, and stress state of the specimen shall closely simulate those expected in the components. In particular, the stress gradient shall not be more abrupt than that expected in the component.

(d) The cyclic rate shall be such that appreciable heating of the specimen does not occur.

(e) The fatigue strength reduction factor shall preferably be determined by performing tests on notched and unnotched specimens and calculated as the ratio of the unnotched stress to the notched stress for failure.

II-1700 EXPERIMENTAL STRESS ANALYSIS OF OPENINGS

II-1710 GENERAL REQUIREMENTS

The stress intensities for opening configurations which do not meet the requirements of NB-3331, NB-3338.2(d), or NB-3339.1 shall be determined in accordance with the methods of this Subarticle.

II-1720 APPLICABILITY OF AVAILABLE EXPERIMENTAL DATA

In accordance with II-1112 reevaluation is not required for configurations for which there are available detailed experimental results that are consistent with the requirements of this Appendix. In order that available experimental data may be interpreted as providing information pertinent to the analysis of slightly different configurations, thereby possibly minimizing the need for additional investigations, the guidelines of II-1730 are presented.

II-1730 GUIDELINES FOR USE OF AVAILABLE EXPERIMENTAL DATA

II-1731 Effect of d/D Ratio

For an unreinforced opening or for an opening where the reinforcement is provided primarily by a uniform increase in component wall thickness, the stresses around the opening will increase with increasing d/D ratio of the opening (diameter of nozzle or opening to diameter of shell). Therefore, experimental data for a small d/D ratio cannot be safely applied to a larger d/D ratio but can be applied to a smaller d/D ratio provided the experiments were made at a d/D ratio <0.5.

II-1732 Effect of D/T Ratio

For an unreinforced opening or for an opening where the reinforcement is provided primarily by a uniform increase in component wall thickness, the stresses around the opening will increase with increasing D/T ratio

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(thinner shell component). Therefore, experimental data for a relatively small D/T ratio cannot be safely applied to a larger D/T ratio but can be applied to a smaller D/T ratio.

II-1733 Proximity to Gross Discontinuities

Generally, the stress data available in the literature are applicable only to single openings. Such data shall be considered valid only for a connection sufficiently removed from another nozzle, opening, flange, or other major discontinuity so that superposition of stresses will not produce an unacceptable value of stress intensity.

II-1734 Requirements for Fillets

Stresses at the outside juncture of a nozzle and shell are greatly influenced by the fillet or transition at the juncture. Generally speaking, stress data available in the literature are for certain specific fillet radii. Other factors being equal, these stress data may be considered valid for fillet radii equal to or greater than those used in the test but shall not be considered valid for smaller fillet radii or undefined fillets and transitions such as for a triangular weld fillet, as commonly used.

II-1800 EXPERIMENTAL DETERMINATION OF STRESS INDICES FOR PIPING

In course of preparation. Pending publication, stress indices for piping shall be determined in accordance with the rules of NB-3680.

II-1900 EXPERIMENTAL DETERMINATION OF FLEXIBILITY FACTORS

In course of preparation. Pending publication, flexibility factors shall be determined in accordance with the rules of NB-3686.

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ARTICLE II-2000
EXPERIMENTAL DETERMINATION OF STRESS INTENSIFICATION FACTORS

II-2100 INTRODUCTION

This Appendix presents a method to experimentally determine stress intensification factors (SIF) of piping components for use in the design of piping systems in accordance with NC/ND-3600.

II-2200 DEFINITIONS

Stress Intensification Factor. A fatigue strength reduction factor which is the ratio of the elastically predicted bending moment producing fatigue failure in a given number of cycles in a butt weld on a straight pipe of nominal dimensions, to that producing failure in the same number of cycles in the component under consideration.

II-2300 TEST PROCEDURE

II-2310 TEST EQUIPMENT

A schematic of a test arrangement is given in Figure II-2310-1.

(a) The machine framework must be sufficiently stiff to prevent anchor rotations.

(b) The pipe component shall be mounted close to the fixed end of the test assembly, but no closer than two pipe diameters.

(c) The free end shall be hinged in a slide capable of applying a fully reversible displacement.

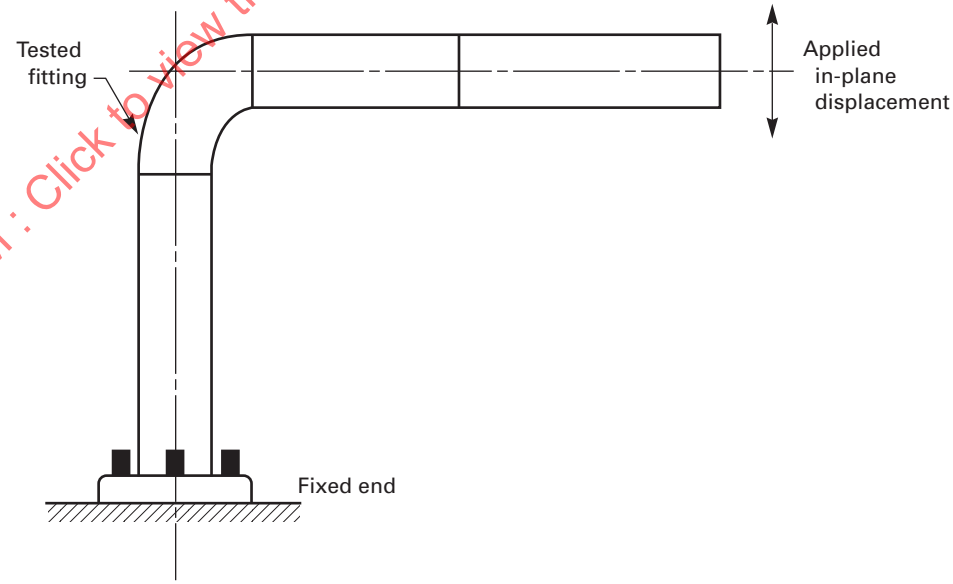
(d) The test equipment shall be calibrated to read displacements with an accuracy of 1% of the imposed displacement amplitude.

II-2320 TEST SPECIMEN

The test specimen shall be SA-106 Grade B pipe and equivalent plates and forgings, otherwise the rules of II-2510 apply.

The fabrication, welding, and examinations of the tested components shall be the same as will be followed in fabrication of the component. Weld contours should be representative of those intended to be used in fabrication.

Figure II-2310-1
Schematic of Test Assembly



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II-2330 APPLIED MOMENT

(a) The test specimen shall be placed in the test configuration and displacements shall be applied in steps to obtain a load-displacement plot analogous to that shown in Figure II-2330-1. At least five points must be recorded in the linear region of the plot.

(b) The loading sequence shall be stopped when the recorded load displacement is no longer linear.

(c) The specimen must then be unloaded, following the same recording sequence as during loading.

(d) The linear region of the load-displacement curve and its straight-line extension will be used in determining the force F_e in II-2400.

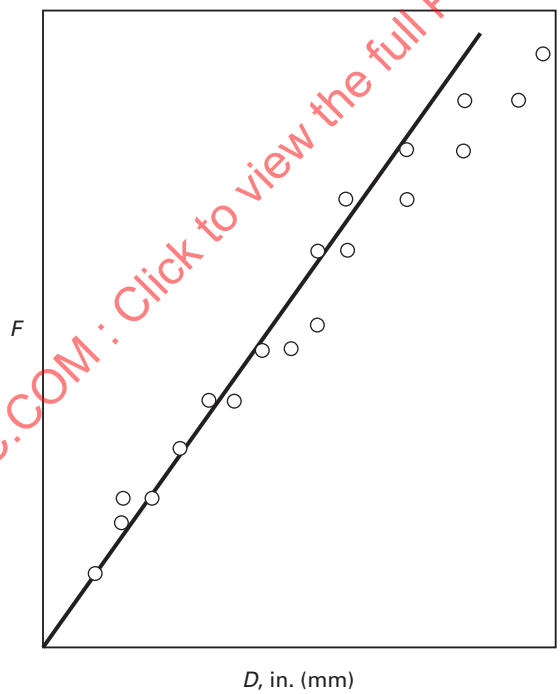
II-2340 CYCLES TO LEAKAGE

(a) The test specimen shall be placed in the test configuration and pressurized with water. The pressure should be sufficient to detect leakage, such as 15 psig to 100 psig (100 kPa to 700 kPa).

(b) The specimen shall be subjected to fully reversed cyclic displacements until a visible through-wall leak develops in the component or its weld to the pipe. Other equivalent methods of through-wall crack detection are permissible.

(c) The fully reversible displacements shall be applied at a frequency not to exceed 120 cycles per minute.

Figure II-2330-1
Displacement *D* and Force *F* Recorded During Loading and Unloading of Test Specimen, With Linear Displacement



(d) The number of cycles N at which the leak occurred shall be recorded. The cyclic displacements shall be selected such that failure occurs in a minimum of N = 500 cycles of reversed displacements.

II-2400 STRESS INTENSIFICATION FACTOR

II-2410 CALCULATED STRESS

(a) The distance L between the point of applied displacement and the leak point is measured.

(b) The imposed displacement is entered on the load-displacement curve established in II-2330, and the corresponding force is noted as F_e.

(c) The applied moment at leakage M_e is to be calculated as

$$M_e = F_e L$$

where

F_e = force corresponding to the applied displacement, read on the straight line of Figure II-2330-1
L = distance between the point of applied displacement and the leak point, in the direction perpendicular to the imposed displacement
M_e = applied elastic moment at leakage

(d) The elastically calculated stress amplitude corresponding to the elastic moment at leakage is

$$S = \frac{M_e}{Z}$$

where

S = leakage stress
Z = section modulus as defined in II-2420

II-2420 SECTION MODULUS

The value of the section modulus, Z, used in calculating the leakage stress in II-2410 shall be that intended to be used in design. The section modulus of the matching pipe is typically used in design. If the leakage stress is computed using Z other than that of the matching pipe, the manner in which Z is computed must be explicitly specified in the definition of the stress intensification factor, and the value of Z at the same location shall be used in design.

II-2430 STRESS INTENSIFICATION FACTOR

The stress intensification factor is established as

$$i = \frac{C}{S\left(N^b\right)}$$

where

b = material exponent; 0.2 for a carbon steel test specimen

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C = material constant; 245,000 psi (1 690 MPa) for a carbon steel test specimen
i = stress intensification factor
N = number of cycles to leakage
S = leakage stress

II-2440 NUMBER OF TEST SPECIMENS

(a) The value of the stress intensification factor i shall be the average value from several, preferably a minimum of four, cyclic displacement tests.

(b) Where less than four tests are conducted, the calculated stress intensification factor i shall be increased by a factor C_i given in Table II-2440-1.

II-2450 DIRECTIONAL STRESS INTENSIFICATION FACTORS

(a) For non-axisymmetric components, a directional stress intensification factor shall be established independently for each direction of bending.

(b) Where the design Code requires the use of a single stress intensification factor, the largest value from the directional stress intensification factors shall be used.

II-2460 VARIABLE AMPLITUDE TEST

If the applied displacement amplitude is changed during a cyclic test, the number of cycles to leakage shall be determined by

$$N = N_j + \sum_{i=1}^{n} (r_i)^{1/b} * N_i$$

where

b = material exponent
= 0.2 for steels
N = equivalent number of cycles to leakage, at maximum amplitude X_j
N_i, N_j = number of cycles at amplitudes X_i, X_j, where all $X_i < X_j$
r_i = X_i/X_j; $r_i < 1$
X_i, X_j = amplitudes of displacement applied during cycles N_i, N_j, in. (mm)

Table II-2440-1
Stress Intensification Increase Factor

Number of Test Specimens	Increase Factor, C_i
1	1.2
2	1.1
3	1.05
≥ 4	1.0

II-2500 VARIATIONS IN MATERIALS AND GEOMETRY

II-2510 MATERIAL CONSTANT AND MATERIAL EXPONENT

When using a test specimen made of Code-listed materials other than carbon steel, a new material constant C and material exponent b shall be established as follows.

(a) A butt-welded test specimen of the tested material shall be fabricated and tested in accordance with II-2300.

(b) The cyclic test of II-2330 shall be repeated for a minimum of eight specimens subject to different applied displacements.

(c) The pairs of values (N,S) shall be plotted on log-log scale.

(d) The material constant C and the material exponent b shall be obtained by tracing a best-estimate straight line through the (N,S) points, in the form

$$\frac{C}{S\left(N^b\right)} = 1$$

II-2520 GEOMETRIC SIMILARITY

(a) The stress intensification factor derived from the tests is applicable to components that are geometrically similar within 20% of the dimensions of the test specimens.

(b) Dimensional extrapolations other than in (a) above shall be identified in the test report, along with their technical justification.

II-2600 TEST REPORT

A test report shall be prepared and certified to meet the requirements of this Appendix by a Professional Engineer competent in the design and analysis of pressure piping systems. The test report shall be complete and written to facilitate an independent review. The report shall contain

(a) description of the tested specimen

(b) nominal pipe and fitting size and dimensions and actual cross-sectional dimensions of importance in interpreting the test results

(c) description and photographs or sketches of the test equipment, including positioning of the test specimens in the machine

(d) calibration of the test equipment. This information may be provided by reference

(e) Certified Material Test Reports for the tested component, including mill-test value of yield and ultimate strength

(f) component and component-to-pipe weld examinations where they are required by the construction Code, with certification of Code compliance of the welds

(g) loading and unloading load-displacement points and line, in accordance with II-2330

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(h) values of material constants C and b, section modulus Z, number of cycles to leakage N, length to leakage point L, force F_e, and moment M_e for each test

(i) derivation of the stress intensification factor i for each test

(j) description, and photograph(s) or sketch(es) of the leakage location

(k) justification for geometrical similarity, if any, in accordance with II-2520

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MANDATORY APPENDIX III
STRESS INTENSITY VALUES, ALLOWABLE STRESS VALUES, FATIGUE STRENGTH VALUES, AND MECHANICAL PROPERTIES

III-1100 LOCATION OF DESIGN STRESS INTENSITY, ALLOWABLE STRESS, YIELD STRENGTH, AND ULTIMATE TENSILE VALUES

All design stress intensity values, allowable stress values, and ultimate and yield strength values for use in design under the rules of this Section are given in Section II, Part D. These values are grouped according to temperature. In every application of the values, the temperature is to be understood as being the actual material temperature. The allowable stress values provided in Section II, Part D are provided for design below the creep regime. They are limited to use at temperatures of 700°F (371°C) and below for ferritic materials, and 800°F (427°C) and below for austenitic materials. See Subsection NH or Section III, Division 5 for design above these temperatures.

III-1200 DERIVATION OF THE DESIGN STRESS INTENSITY AND ALLOWABLE STRESS VALUES

For other than bolting materials, the bases for the design stress intensity and allowable stress values are given in Section II, Part D, Mandatory Appendices 1 and 2. Section II, Part D, Mandatory Appendix 1 provides this information for the allowable stress values; Section II, Part D, Mandatory Appendix 2 provides this information for the design stress intensity. The bases for the design stress intensity and allowable stress values for bolting materials are provided in Section II, Part D, Mandatory Appendix 2.

III-1300 FATIGUE STRENGTH VALUES FOR ALL MATERIALS

The fatigue curves of Mandatory Appendix I are obtained from uniaxial strain cycling data in which the imposed strain amplitude (half range) is multiplied by the elastic modulus to put the values in stress units. A best fit to the experimental data is obtained by applying the method of least squares to the logarithms of the stress values. The curves are adjusted where necessary to include the maximum effect of mean stress. For all figures except Figure I-9.2, the design fatigue strength values are obtained from the best fit curve by applying a factor of 2 on stress or a factor of 20 on cycles, whichever is the more conservative at each point. The design fatigue strength values for Figure I-9.2 are obtained from the best fit curve by applying a factor of 2 on stress or a factor of 12 on cycles, whichever is more conservative at each point.

III-1400 MECHANICAL AND PHYSICAL PROPERTIES

All mechanical and physical design properties, e.g., modulus of elasticity and coefficient of thermal expansion, for use with this Section are given in Section II, Part D. In the absence of available data given in Section II, Part D, the design may be based on available manufacturer's data for that material unless otherwise prohibited by this Section. Substantiation of this data shall be included in the design output documents.

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MANDATORY APPENDIX IV
APPROVAL OF NEW MATERIALS UNDER THE ASME BOILER AND PRESSURE VESSEL CODE

See Appendix 5 of Section II, Part D.

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MANDATORY APPENDIX V
CERTIFICATE HOLDER'S DATA REPORT FORMS, INSTRUCTIONS, AND APPLICATION FORMS FOR CERTIFICATES OF AUTHORIZATION FOR USE OF CERTIFICATION MARK

The instructions for the Data Report Forms (see Table V-1000) are identified by parenthesized numbers corresponding to the circled numbers on the sample Forms in this Appendix.

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FORM N-1 CERTIFICATE HOLDER'S DATA REPORT FOR NUCLEAR VESSELS*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Manufactured and certified by (1) (name and address of N Certificate Holder)

2. Manufactured for (2) (name and address of Purchaser)

3. Location of installation (3) (name and address)

4. Type (4) (horizontal or vertical) (5) (tank, jacketed, heat ex.) (6) (Certificate Holder's serial no.) (7) (CRN) (8) (drawing no.) (9) (National Bd. no.) (10) (year built)

5. ASME Code, Section III, Division 1 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)

Items 6–10 inclusive to be completed for single wall vessels, jackets of jacketed vessels, or shells of heat exchangers.

6. Shell (15) (material spec. no.) (16) (tensile strength) (17) (nominal thickness) (18) (minimum design thickness) (19) (diameter ID) (20) [length (overall)]

7. Seams (21) (long.) (22) (HT[1]) (23) (RT) (24) (eff. %) (25) (girth) (22) (HT[1]) (23) (RT) (26) (no. of courses)

8. Heads (15) [(a) material spec. no.] (16) (tensile strength) (15) [(b) material spec. no.] (16) (tensile strength)

(27)	Location (top, bottom, ends)	Thickness	Corrosion Allowance	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (convex or concave)
(a)										
(b)										

If removable, bolts used (15) (28) (material spec. no., size, quantity) Other fastening (29) (describe or attach sketch)

9. Jacket closure (30) (Describe as ogee & weld, bar, etc. If bar, give dimensions, describe, or sketch)

10. Design pressure[2] (31) at max. temp. (32). Min. pressure-test temp. (33). Pneu., hydro., or comb. test pressure (34)

Items 11 and 12 to be completed for tube sections.

11. Tubesheets (15) (stationary, material spec. no.) (35) [diameter (subject to press.)] (36) (thickness) (37) [attachment (welded, bolted)]
(15) (floating, material spec. no.) (38) (diameter) (36) (thickness) (37) (attachment)

12. Tubes (15) (material spec. no.) (40) (OD) (41) [thickness (inches or gage)] (42) (no.) (43) [type (straight or U)]

Items 13 to 16 inclusive to be completed for inner chambers of jacketed vessels, or channels of heat exchangers.

13. Shell (15) (material spec. no.) (16) (tensile strength) (17) (nominal thickness) (18) (minimim design thickness) (19) (diameter ID) (20) [length (overall)]

14. Seams (21) [long. (welded. dbl., single)] (22) [HT[1] (yes or no)] (23) (RT) (24) (eff. %) (25) (girth) (22) (HT[1]) (23) (RT) (26) (no. of courses)

15. Heads (15) [(a) material spec. no.] (16) (tensile strength) (15) [(b) material spec. no.] (16) (tensile strength) (15) [(c) material spec. no.] (16) (tensile strength)

(27) Location	Thickness	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (convex or concave)
(a) Top, bottom, ends								
(b) Channel								
(c) Floating								

If removable, bolts used (15) (26) (material spec. no., size, quantity) Other fastening (29) (describe or attach sketch)

16. Design pressure[2] (31) at (32). Min. pressure-test temp. (33). Pneu., hydro., or comb. test pressure (34)

[1] If postweld heat treated. [2] List other internal or external pressure with coincident temperature when applicable.

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8\frac{1}{2} \times 11$; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM N-1 (Back — Pg. 2 of (56))

Certificate Holder's Serial No. (6)

17. Nozzles, inspection and safety valve openings (44) (15) (18) (19)

Purpose (inlet, outlet, drain, etc.)	Quantity	Diameter or Size	Type	How Attached	Material	Thickness	Reinforcement Material	Location

18. Supports: Skirt (45) (yes or no) Lugs (45) (quantity) Legs (45) (quantity) Other (45) (describe) Attached (45) (where and how)

19. Remarks: (46)

CERTIFICATION OF DESIGN

Design specification certified by (49) P.E. State _____ Reg. no. _____

Design report certified by (50) P.E. State _____ Reg. no. _____

(69)

CERTIFICATE OF SHOP COMPLIANCE

We certify that the statements made in this report are correct and that this nuclear vessel conforms to the rules for construction of the ASME Code, Section III, Division 1.

N Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (N Certificate Holder) Signed _____ (authorized representative)

(68)

CERTIFICATE OF SHOP INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have inspected the component described in this Data Report on _____, and state that to the best of my knowledge and belief, the Certificate Holder has constructed this component in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

(70)

CERTIFICATE OF FIELD ASSEMBLY COMPLIANCE

We certify that the statements on this report are correct and that the field assembly construction of all parts of this nuclear vessel conforms to the rules of construction of the ASME Code, Section III, Division 1.

N Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (N Certificate Holder) Signed _____ (authorized representative)

(71)

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have compared the statements in this Data Report with the described component and state that parts referred to as data items _____, not included in the certificate of shop inspection, have been inspected by me on _____ and that to the best of my knowledge and belief, the Certificate Holder has constructed and assembled this component in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

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FORM N-1A CERTIFICATE HOLDER'S DATA REPORT FOR NUCLEAR VESSELS*
Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Manufactured and certified by (1) (name and address of N Certificate Holder)
2. Manufactured for (2) (name and address of Purchaser)
3. Location of installation (3) (name and address)
4. Type (4) (horizontal or vertical) (6) (Certificate Holder's serial no.) (7) (CRN) (8) (drawing no.) (9) (National Bd. no.) (10) (year built)
5. ASME Code, Section III, Division 1: (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)
6. Shell (15) (material spec. no.) (16) (tensile strength) (17) (nominal thickness) (18) (minimum design thickness) (19) (diameter ID) (20) [length (overall)]
7. Seams (21) (long.) (22) (HT[1]) (23) (RT) (24) (eff. %) (25) (girth) (22) (HT[1]) (23) (RT) (26) (no. of courses)
8. Heads (15) [(a) material spec. no.] (16) (tensile strength) (15) [(b) material spec. no.] (16) (tensile strength)

(27)	Location (top, bottom, ends)	Thickness	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (convex or concave)
(a)									
(b)									

If removable, bolts used (15) (28) (material spec. no., T.S., size, quantity) Other fastening (29) (describe or attach sketch)

9. Design pressure[2] (31) at max. temp. (32). Min. pressure test temp. (33) Hydro., pneu., or comb. test pressure (34)
10. Nozzles, inspection and safety valve openings (44) (15) (18) (19)

Purpose (inlet, outlet, drain, etc.)	Quantity	Diameter or Size	Type	How Attached	Material	Thickness	Reinforcement Material	Location

11. Supports: Skirt (45) (yes or no) Lugs (45) (quantity) Legs (45) (quantity) Other (45) (describe) Attached (45) (where and how)
12. Remarks (46)

CERTIFICATION OF DESIGN

Design specification certified by (49) P.E. State ____ Reg. no. ____

Design report certified by (50) P.E. State ____ Reg. no. ____

(69)

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that this nuclear vessel conforms to the rules for construction of the ASME Code, Section III, Division 1. N Certificate of Authorization No. ____ Expires ____

Date ____ Name ____ (N Certificate Holder) Signed ____ (authorized representative)

(68)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by ____ of ____ have inspected the component described in this Data Report on ____, and state that to the best of my knowledge and belief, the Certificate Holder has constructed this component in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date ____ Signed ____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

[1] If postweld heat treated. [2] List other internal or external pressure with coincident temperature when applicable.

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8\frac{1}{2} \times 11$; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM N-2 CERTIFICATE HOLDER'S DATA REPORT FOR IDENTICAL NUCLEAR PARTS AND APPURTENANCES*
As Required by the Provisions of the ASME Code, Section III
Not to Exceed One Day's Production

Pg. 1 of (56)

1. Manufactured and certified by (1) (name and address of NPT Certificate Holder)
2. Manufactured for (2) (name and address of purchaser)
3. Location of installation (3) (name and address)
4. Type (8) (drawing no.) (15) (material spec. no.) (16) (tensile strength) (7) (CRN) (10) (year built)
5. ASME Code, Section III, Division 1 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)
6. Fabricated in accordance with Const. Spec. (Div. 2 only) (77) (no.) Revision (77) Date (77)
7. Remarks (46)
8. Nom. thickness (17) Min. design thickness (18) Diameter ID (19) Length overall (20)
9. When applicable, Certificate Holder's Data Reports are attached for each item of this report.

Part or Appurtenance Serial Number (6)	National Board No. in Numerical Order (9)	Part or Appurtenance Serial Number (6)	National Board No. in Numerical Order (9)
(1)		(26)	
(2)		(27)	
(3)		(28)	
(4)		(29)	
(5)		(30)	
(6)		(31)	
(7)		(32)	
(8)		(33)	
(9)		(34)	
(10)		(35)	
(11)		(36)	
(12)		(37)	
(13)		(38)	
(14)		(39)	
(15)		(40)	
(16)		(41)	
(17)		(42)	
(18)		(43)	
(19)		(44)	
(20)		(45)	
(21)		(46)	
(22)		(47)	
(23)		(48)	
(24)		(49)	
(25)		(50)	

10. Design pressure (31) . Temperature (32) . Hydro. test pressure (33) (34) (when applicable) at temp. .

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8^1/_2 \times 11$; (2) information in items 2 and 3 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM N-2 (Back — Pg. 2 of (56))

Certificate Holder's Serial Nos. (6) through _____

CERTIFICATION OF DESIGN

Design specifications certified by (49) (when applicable) P.E. State _____ Reg. no. _____

Design report* certified by (50) (when applicable) P.E. State _____ Reg. no. _____

(69)

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that this (these) _____ conforms to the rules of construction of the ASME Code, Section III, Division 1.

NPT Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (NPT Certificate Holder) Signed _____ (authorized representative)

(68)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have inspected these items described in this Data Report on _____, and state that to the best of my knowledge and belief, the Certificate Holder has fabricated these parts or appurtenances in accordance with the ASME Code, Section III, Division 1. Each part listed has been authorized for stamping on the date shown above.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the equipment described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

(07/11)

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FORM N-3 OWNER'S DATA REPORT FOR NUCLEAR POWER PLANT COMPONENTS*
As Required by the Provisions of the ASME Code, Section III

Pg. 1 of (56)

1. Name of Owner (1)

2. Address of Owner (2)

3. Name of power plant (52) (53) (unit no.) (9) (National Bd. no.)

4. Location of power plant (3)

5. NUCLEAR VESSELS (List all nuclear concrete and metallic vessels or core supports and attach copies of all N and NPT Certificate Holder's Data Reports. Forms N-1, N-1A, N-2, NCS-1, and C-1.)

Certificate Holder and Serial Number (1) (6)	State No. or CRN (7)	National Bd. No. (9)	Year Built (10)

Attach supplemental pages as required.

6. NUCLEAR PIPING (Identify all nuclear piping by listing system identification appearing on Form N-5 and attach copies of all N-5 Data Reports for nuclear piping.)

Certificate Holder and Serial Number (1) (6)	Piping System Identification (51)	National Bd. No. (9)	Year Built (10)

Attach supplemental pages as required.

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8^1/_2 \times 11$; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

(09/06)

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FORM N-3 (Back — Pg. 2 of (56))

Unit No. (53)

7. NUCLEAR PUMPS & VALVES (List and identify all nuclear pumps and valves and attach copies of all N Certificate Holder's Data Reports, Forms NV-1 and NPV-1.)

Certificate Holder	Pump	Valve	Cert. Holder's Serial No.	National Bd. No.	Year Built
(1)	(54)	(54)	(6)	(9)	(10)

Attach supplemental pages as required.

8. Remarks (46)

OWNER'S CERTIFICATE OF COMPLETED INSTALLATION

(69)

I, the undersigned, certify that the statements made in this report are correct and have checked all nuclear components coming under the scope of the ASME Code, Section III, and state that to the best of my knowledge and belief, each Certificate Holder has met all the rules of construction of the ASME Code, Section III. Attached are copies of Certificate Holder's Data Reports covering all nuclear components.

Owner's Certificate of Authorization No. _____ Expires _____

Date _____ Signature _____
(authorized representative)

CERTIFICATE OF INSPECTION

(68)

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____, based on audits of the owner's quality assurance program and supporting documentation for components and/or appurtenances and installation of same described in this Data Report on _____, state that to the best of my knowledge and belief, the owner, or his designee, as applicable, has complied with the requirements of the ASME Code, Section III.

By signing this certificate, neither the inspector nor his employer makes any warranty, expressed or implied, concerning the components and/or appurtenances and installation of same described in this owner's Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this audit activity.

Date _____ Signed _____ Commission (67)
(Authorized Nuclear Inspector) [National Board Number and Endorsement]

(07/11)

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FORM N-5 CERTIFICATE HOLDER'S DATA REPORT FOR INSTALLATION OR SHOP ASSEMBLY OF NUCLEAR POWER PLANT COMPONENTS, SUPPORTS, AND APPURTENANCES*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Installed and certified by (1) (name and address of N or NA Certificate Holder)
2. Installed for (2) (name and address of Purchaser)
3. Location of installation (3) (name and address)
4. System identification (51) (system name) (6) (Cert. Holder's serial no.) (8) (drawing no.) (7) (CRN) (9) (National Bd. no.) (10) (year installed)
5. ASME Code, Section III, Division 1 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)
6. N Certificate Holder having overall responsibility (1) (name and address)
7. Nuclear components, parts, appurtenances, and supports installed: (List each item and attach copies of N Certificate Holders' Data Reports and NPT Certificate Holder's Data Reports.)

Components

(a) Comp. or Appurt.	(b) Name of Certificate Holder	(c) Serial No.	(d) CRN No.	(e) National Bd. No.	(f) Year Built
(62)	(1)	(6)	(7)	(9)	(10)

Piping and part installation

(a) Piping or Part Subassembly	(b) Name of Certificate Holder	(c) Serial No.	(d) CRN No.	(e) National Bd. No.	(f) Year Built—Parts Only
(63)	(1)	(6)	(7)	(9)	(10)

Support installation

(a) Support No.	(b) Name of Certificate Holder	(c) Serial No.	(d) Design Rept./Load Capac. Data Sheet	(e) CRN No.	(f) National Bd. No.	(g) Year Built
(64)	(1)	(6)	(61)	(7)	(9)	(10)

Additional material excluding welding material

(a) Name of Manufacturer	(b) Material Spec. No.	(c) Dimensions—Overall
(59)	(15)	(20)

8. Installation in accordance with

Procedure or Drawing No.	Prepared by
(8)	(60)

9. Hydrostatic test pressure (34) at temp. (33). System design pressure (31) at temp. (32)
10. Remarks (46)

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8\frac{1}{2} \times 11$; (2) information in items 1 and 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM N-5 (Back — Pg. 2 of (56))

Certificate Holder's Serial No. (6)

CERTIFICATION OF DESIGN FOR PIPING SYSTEM

Design information on file at (47)

Design report on file at (48)

Design specification certified by (49) P.E. State _____ Reg. no. _____

Design report certified by (50) P.E. State _____ Reg. no. _____

Design conditions of pressure piping (31) psi. Temp. (32) °F.

(70) **CERTIFICATE OF INSTALLATION COMPLIANCE**

We certify that the statements made in this report are correct and that this installation conforms to the rules for construction of the ASME Code, Section III, Division 1, and was performed in accordance with the documents listed in 8 above.

N or NA Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (N or NA Certificate Holder) Signed _____ (authorized representative)

(71) **CERTIFICATE OF INSTALLATION INSPECTION**

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have inspected the installation of the items described in this Data Report on _____ and state that to the best of my knowledge and belief, the Certificate of Authorization Holder has performed this installation in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the installation described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

(69) **CERTIFICATE OF COMPLIANCE FOR OVERALL RESPONSIBILITY**

Following completion of the above, the Certificate of Authorization Holder accepting overall responsibility for the piping system shall complete the following statement.

We certify that the statements made by this report are correct and that the piping system conforms to the rules for construction of the ASME Code, Section III, Division 1.

N Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (N Certificate Holder) Signed _____ (authorized representative)

(68) **CERTIFICATE OF INSPECTION**

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have inspected the piping system described in this Data Report on _____ and state that to the best of my knowledge and belief, the Certificate Holder has connected this piping system in accordance with the ASME Code, Section III, Division 1.

By signing this certificate, neither the inspector nor his employer makes any warranty, expressed or implied, concerning the piping system described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

(07/11)

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FORM N-6 CERTIFICATE HOLDER'S DATA REPORT FOR STORAGE TANKS*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Manufactured and certified by (1)
(name and address of N Certificate Holder)

2. Manufactured for (2)
(name and address of purchaser)

3. Location of installation (3)
(name and address)

4. Type (4) (horizontal or vertical tank) (6) (Cert. Holder's serial no.) (7) (CRN) (8) (drawing no.) (9) (National Bd. no.) (10) (year built)

5. ASME Code, Section III, Division 1 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)

6. Shell (15) [material (spec. no., grade)] (17) (nominal thickness) (18) (design thickness) (16) (minimum tensile) (19) (diameter ID) (20) [length (overall)]

7. Seams (21) [long. (welded, dbl., sngl., lap. butt)] (23) [RT (spot or full)] (24) [eff. (%)] (22) [HT[1] (yes)] (25) [girth (welded, dbl., sngl., lap. butt)] (23) [RT (spot, partial or full)] (26) (no. of courses)

8. Heads (a) (15) [material (spec. no., grade)] T.S. (16) (b) (15) [material (spec. no., grade)] T.S. (16)

	Location (top, bottom, ends)	Minimum Thickness	Design Thickness	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (convex or concave)	Type of Joint (lap or butt)
(a)											
(b)											

If removable, bolts used (describe other fastenings) (15) (26)
(material, spec. no., gr., size. no.)

9. Design Pressure[2] (31) at max. temp. (32) . Min. pressure-test temp. (33) . Pneu., hydro., or comb. test pressure (34)

10. Nozzles, inspection and safety valve openings (15) (18) (19) (44)

Purpose (inlet, outlet, drain, etc.)	No.	Diameter or Size	Type	Material	Nominal Thickness	Reinforcement Material	How Attached	Location

11. Supports: Skirt (45) (yes or no) Lugs (45) (no.) Legs (45) (no.) Other (45) (describe) Attached (45) (where and how)

12. Remarks (46)

[1] If postweld heat treated. [2] List other internal or external pressures with coincident temperature when applicable.

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8\frac{1}{2} \times 11$; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM N-6 (Back — Pg. 2 of (56))

Certificate Holder's Serial No. (6)

CERTIFICATION OF DESIGN

Design specification certified by (49) P.E. State _____ Reg. no. _____

Design information on file at (42)

(69) **CERTIFICATE OF SHOP COMPLIANCE**

We certify that the statements made in this report are correct and that this storage tank conforms to the rules for construction of the ASME Code, Section III, Division 1.

N Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (N Certificate Holder) Signed _____ (authorized representative)

(68) **CERTIFICATE OF SHOP INSPECTION**

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have inspected the storage tank described in this Data Report on _____, and state that to the best of my knowledge and belief, the Certificate Holder has constructed this storage tank in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the storage tank described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

(70) **CERTIFICATE OF FIELD ASSEMBLY COMPLIANCE**

We certify that the statements on this report are correct and that the field assembly construction of all parts of this storage tank conforms to the rules of construction of the ASME Code, Section III, Division 1.

N Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (N Certificate Holder) Signed _____ (authorized representative)

(71) **CERTIFICATE OF FIELD ASSEMBLY INSPECTION**

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have compared the statements in this Data Report with the described storage tank and state that parts referred to as data items _____, not included in the Certificate of Shop Inspection, have been inspected by me on _____ and that to the best of my knowledge and belief the Certificate Holder has constructed and assembled this storage tank in accordance with the ASME Code, Section III, Division 1.

By signing this certificate, neither the inspector nor his employer makes any warranty, expressed or implied, concerning the installation described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

(07/11)

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FORM NPP-1 CERTIFICATE HOLDER'S DATA REPORT FOR FABRICATED NUCLEAR PIPING SUBASSEMBLIES*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Fabricated and certified by (1) (name and address of NPT Certificate Holder)
2. Fabricated for (2) (name and address of Purchaser)
3. Location of installation (3) (name and address)
4. Type (6) (Certificate Holder's serial no.) (7) (CRN) (8) (drawing no.) (9) (National Bd. no.) (10) (year built)
5. ASME Code, Section III, Division 1: (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)
6. Shop hydrostatic test (34) at (33) (if performed)
7. Description of piping (51) (55) (15)
8. Certificate Holder's Data Reports properly identified and signed by commissioned inspectors have been furnished for the following items of this report (80)
9. Remarks (46)

(69)

CERTIFICATE OF SHOP COMPLIANCE

We certify that the statements made in this report are correct and that the fabrication of the described piping subassembly conforms to the rules for construction of the ASME Code, Section III, Division 1.

NPT Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (NPT Certificate Holder) Signed _____ (authorized representative)

(68)

CERTIFICATE OF SHOP INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have inspected the piping subassembly described in this Data Report on _____, and state that to the best of my knowledge and belief, the Certificate Holder has fabricated this piping subassembly in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the piping subassembly described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

*Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8\frac{1}{2} \times 11$; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM NPP-1 (Back — Pg. 2 of (56))

Certificate Holder's Serial No. (6)

10. Description of field fabrication (51) (55)

11. Pneu., hydro., or comb. test pressure (34) at temp. (33) (if performed)

(70) **CERTIFICATE OF FIELD FABRICATION COMPLIANCE**

We certify that the statements made in this report are correct and that the field fabrication of the described piping subassembly conforms with the rules for construction of the ASME Code, Section III, Division 1.

NPT Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (Certificate Holder) Signed _____ (authorized representation)

(71) **CERTIFICATE OF FIELD FABRICATION INSPECTION**

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have compared the statements in this Data Report with the described piping subassembly and state that parts referred to as data items _____, not included in the Certificate of Shop Inspection, have been inspected by me on _____ and that to the best of my knowledge and belief the Certificate Holder has fabricated this piping subassembly in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the piping subassembly described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

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FORM NPV-1 CERTIFICATE HOLDER'S DATA REPORT FOR NUCLEAR PUMPS OR VALVES*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Manufactured and certified by (1)
(name and address of N Certificate Holder)

2. Manufactured for (2)
(name and address of Purchaser)

3. Location of installation (3)
(name and address)

4. Model No., Series No., or Type (104) Drawing (8) Rev. (8) CRN (7)

5. ASME Code, Section III, Division 1 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)

6. Pump or valve (100) Nominal inlet size (83) Outlet size (83)

7. Material
(a) valve Body (15) Bonnet (15) Disk (15) Bolting (15)
(b) pump Casing (15) Cover (15) Bolting (15)

(a) Certificate Holder's Serial No.	(b) National Board No.	(c) Body/Casing Serial No.	(d) Bonnet/Cover Serial No.	(e) Disk Serial No.
(6)	(9)	(102)	(102)	(102)

FORM NPV-1 (Back — Pg. 2 of (56))

Certificate Holder's Serial No. (6)

8. Design conditions (105) (pressure) (106) (temperature) or valve pressure class (81)

9. Cold working pressure (103)

10. Hydrostatic test (105) . Disk differential test pressure (98)

11. Remarks

CERTIFICATION OF DESIGN

Design Specification certified by (49) P.E. State Reg. no.

Design Report certified by (50) P.E. State Reg. no.

(69)

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that this pump or valve conforms to the rules for construction of the ASME Code, Section III, Division 1.

N Certificate of Authorization No. Expires

Date Name (N Certificate Holder) Signed (authorized representative)

(68)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by

of have inspected the pump, or valve, described in this Data Report on , and state that to the best of my knowledge and belief, the Certificate Holder has constructed this pump, or valve, in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Signed (Authorized Nuclear Inspector) Commission (67) [National Board Number and Endorsement]

(07/11)

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FORM NV-1 CERTIFICATE HOLDER'S DATA REPORT FOR PRESSURE OR VACUUM RELIEF VALVES*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Manufactured and certified by (1) (name and address of NV Certificate Holder)
2. Manufactured for (2) (name and address of Purchaser)
3. Location of installation (3) (name and address)
4. Valve (104) (model no., series no.) Orifice size (82) Nom. inlet size (83) Outlet size (83)
5. ASME Code, Section III, Division 1 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)
6. Type (84) (spring, pilot, or power operated) (85) (set pressure) (86) (blowdown) (87) (rated temp.) (34) (hydro. test., inlet) at (33)
7. Identification (6) (Cert. Holder's serial no.) (7) (CRN) (8) (drawing no.) (9) (National Bd. no.) (10) (year built)
8. Control ring settings (108)
9. Pressure retaining items

	Serial No. or Identification	Material Spec., Including Type or Grade	Tensile Strength
Body	(6)	(15)	(16)
Bonnet or Yoke			
Support Rods			
Nozzle			
Disk			
Spring Washers			
Adjusting Screws			
Spindle			
Spring			
Bolting			
Other Items			

10. Relieving capacity (86) (steam or fluid) @ (65) overpressure as certified by the National Board (66) (date)
11. Remarks (46)

CERTIFICATION OF DESIGN

Design Specification certified by (49) P.E. State Reg. no.
Design Report certified by (50) P.E. State Reg. no.

(69)

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that this valve conforms to the rules for construction of the ASME Code, Section III, Division 1.

NV Certificate of Authorization No. Expires

Date Name (NV Certificate Holder) Signed (authorization representative)

*Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8\frac{1}{2} \times 11$; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM NV-1 (Back — Pg. 2 of (56))

Certificate Holder's Serial No. (6)

(68)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by

of _____ have inspected the valve described in this Data Report on _____ and state that to the best of my knowledge and belief, the Certificate Holder has constructed this valve in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission (67) _____ [National Board Number and Endorsement]

(07/11)

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FORM NCS-1 CERTIFICATE HOLDER'S DATA REPORT FOR CORE SUPPORT STRUCTURES*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Manufactured and certified by (1)
   (name and address of N Certificate Holder)
2. Manufactured for (2)
   (name and address of Purchaser)
3. Location of installation (3)
   (name and address)
4. Type (39) (structure) (6) (C.H.'s serial no.) (7) (CRN) (8) (drawing no.) (9) (National Bd. no.) (10) (year built)
5. ASME Code, Section III, Division 1 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)
6. Manufactured in accordance with Specification (78) Rev. Date (78)
   (Design Report or Load Capacity Data Sheet)
7. List of Drawings (with last revision and date)
8. Remarks (46)

CERTIFICATION OF DESIGN

Design specification certified by (49) P.E. State Reg. no.

Design report certified by (50) P.E. State Reg. no.

(69)

CERTIFICATE OF INTERNAL STRUCTURES

The undersigned, having a valid Certification of Authorization, certify that the construction of the internal structures will not adversely affect the integrity of the core support structures.

N Certificate of Authorization No. Expires

Date Name Signed
(N Certificate Holder) (authorized representative)

(69)

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that this set of core support structures conforms to the rules of construction of the ASME Code, Section III, Division 1.

N Certificate of Authorization No. Expires

Date Name Signed
(N Certificate Holder) (authorized representative)

(68)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by

have inspected the core support structure described in this Data Report on , and state that to the best of my knowledge and belief, the Certificate Holder has constructed this item in accordance with the ASME Code, Section III, Division 1. By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the item described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Signed Commission (67)
(Authorized Nuclear Inspector) [National Board Number and Endorsement]

* Supplemental sheets in the form of lists, sketches, or drawings may be used provided: (1) size is $8\frac{1}{2} \times 11$; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM NF-1 CERTIFICATE HOLDER'S DATA REPORT FOR SUPPORTS*
As Required by the Provisions of the ASME Code, Section III, Division 1 Pg. 1 of (56)

1. Manufactured by (1)
(name and address of NPT Certificate Holder)

2. Manufactured for (2)
(name and address of Purchaser)

3. Location of installation (3)
(name and address)

4. Type (96) (describe) (97) (Design Report or Load Capacity Data Sheet) (10) (Year built)

5. ASME Code, Section III, Division 1 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)

6. Identification

	(a) Support I.D. No.	(b) Material Specification No.	(c) Canadian Registration No.	(d) Applicable Drawings With Last Rev. & Date	(e) National Board No.
	(64)	(15)	(7)	(8)	(9)
(1)					
(2)					
(3)					
(4)					
(5)					
(6)					
(7)					
(8)					
(9)					
(10)					

7. Remarks (46)

*Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8^1/_2 \times 11$; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM NF-1 (Back — Pg. 2 of (56))

Support I.D. Nos. (64) through _____

CERTIFICATION OF DESIGN

Design Specification certified by (49) _____ P.E. State _____ Reg. No. _____

Design Report certified by (50) _____ P.E. State _____ Reg. No. _____

(69)

CERTIFICATION OF COMPLIANCE

We certify that the statements made in this report are correct and that these supports conform to the rules for construction of the ASME Code, Section III, Division 1.

NPT Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (NPT Certificate Holder) Signed _____ (authorized representative)

(68)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have inspected the supports described in this Data Report on _____, and state that to the best of my knowledge and belief, the Certificate Holder has constructed these supports in accordance with the ASME Code, Section III, Division 1.

By signing this certificate, neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component supports described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission _____ [National Board Number and Endorsement]

(07/11)

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FORM NM-1 CERTIFICATE HOLDER'S DATA REPORT FOR TUBULAR PRODUCTS AND FITTINGS WELDED WITH FILLER METAL*
As Required by the Provisions of the ASME Code, Section III, Division 1

Pg. 1 of (56)

1. Manufactured and certified by _____ (1) (name and address of NPT Certificate Holder)
2. Manufactured for _____ (2) (name and address of Purchaser)
3. Location of installation _____ (3) (name and address)
4. Identification _____ (79) (lot, etc.) _____ (8) (drawing no.) _____ (9) (National Bd. no.) _____ (10) (year built)
5. ASME Code, Section III, Division 1 _____ (11) (edition) _____ (12) [Addenda (if applicable) (date)] _____ (13) (class) _____ (14) (Code Case no.)
6. Mat'l. Spec. _____ (15) (SA or SB and no.) _____ (16) (tensile strength) _____ (17) (nominal thickness) _____ (19) (diameter ID) _____ (20) (pipe length and fitting type)
7. Shop hydrostatic test pressure _____ (34) at _____ (33) (if performed)
8. Remarks _____ (46)

(69)

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that the products defined in this report conform to the rules for construction of the ASME Code, Section III, Division 1. The radiographic film and a radiographic report showing film location are attached to the Certified Material Test Reports provided for the material covered by this report.

NPT Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (NPT Certificate Holder) Signed _____ (authorized representative)

(68)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by _____ of _____ have inspected the products described in this Data Report on _____, and state that to the best of my knowledge and belief, the Certificate Holder has constructed this product in accordance with the ASME Code, Section III, Division 1.

By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the products described in this Data Report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ (Authorized Nuclear Inspector) Commission _____ (67) [National Board Number and Endorsement]

*Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8\frac{1}{2} \times 11$; (2) information in items 1 through 4 on this Data Report is included on each sheet; and (3) each sheet is numbered and number of sheets is recorded at the top of this form.

(07/11)

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FORM NS-1 CERTIFICATE HOLDER'S CERTIFICATE OF CONFORMANCE FOR WELDED SUPPORTS*
As Required by the Provisions of the ASME Code, Section III, Division 1 Pg. 1 of (56)

1. Manufactured by (1)
(name and address of NS Certificate Holder)

2. Manufactured for (2)
(name and address of Purchaser)

3. Location of installation (3)
(name and address)

4. Type (96) (describe) (97) (Design Report or Load Capacity Data Sheet) (10) (Year built)

5. ASME Code, Section III, Division 1 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)

6. Identification

	(a) Support I.D. No.	(b) Material Specification No.	(c) Canadian Registration No.	(d) Applicable Drawings With Last Rev. & Date
(1)	(64)	(15)	(7)	(8)
(2)				
(3)				
(4)				
(5)				
(6)				
(7)				
(8)				
(9)				
(10)				

7. Remarks (46)

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8^1/_2 \times 11$; (2) information in items 1 through 4 on this Certificate of Conformance is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM NS-1 (Back — Pg. 2 of (56))

Support I.D. Nos. (64) through _____

CERTIFICATION OF DESIGN

Design Specification certified by (49) P.E. State _____ Reg. No. _____

Design Report certified by (50) P.E. State _____ Reg. No. _____

(69)

CERTIFICATE OF CONFORMANCE

We certify that the statements made in this report are correct and that these supports conform to the rules for construction of the ASME Code, Section III, Division 1.

NS Certificate of Authorization No. _____ Expires _____

Date _____ Name _____ (NS Certificate Holder) Signed _____ (authorized representative)

(07/10)

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FORM C-1 CERTIFICATE HOLDER'S DATA REPORT FOR CONCRETE REACTOR VESSELS AND CONTAINMENTS*
As Required by the Provisions of the ASME Code, Section III, Division 2

Pg. 1 of (56)

1. Constructed and certified by (1)
(name and address of N Certificate Holder)

2. Constructed for (2)
(name and address of Owner)

3. Location (3)
(name and address)

4. (6) (Certificate Holder's serial no.) (7) (CRN) (9) (National Bd. no.) (10) (year built)

Type (5) (reactor vessel or containment) (88) (construction reinforced or prestressed concrete) (8) (drawing no.)

5. ASME Code, Section III, Division 2 (11) (edition) (12) [Addenda (if applicable) (date)] (13) (class) (14) (Code Case no.)

6. Design conditions
(a) Drawing and revision Design pressure (31) Design temp. (32)
(b) Design specification no. (78) Revision Date
(c) Foundation type (91) (soil, rock bearing, piles, etc.) Dome type (90) (spherical, elliptical, flat, etc.)

7. Nominal dimensions (19) (inside diameter) (17) (wall thickness) (20) (foundation top to springline height) (89) (dome height) (17) (dome thickness)

8. Construction specifications (list all construction specifications) (77)

Title	No.	Revision	Date

9. Type of post-tensioning system
(a) Tendon material (15) Min. tensile (16) Diameter or size (38) Corrosion protection (grout, grease, etc.)
(b) Fabricated by (1) Installed by (1)

10. Liner and sleeves (if within constructor's responsibility)
(a) Liner material (15) Min. yield (107) Bottom thickness (17)
Wall thickness (17) Dome thickness (17)
(b) Sleeve material (15) Min. yield (107) Number and sizes (38)

11. Parts (fabricated, installed, or constructed by others)
List each item and attach copy of Certificate Holder's Data Report

Part (63)	Drawing & Rev. (8)	Name of CH (1)	Manufacturer's Serial No. (6)	CRN (7)	National Bd. No. (9)	Year Built (10)

12. Additional material excluding welding material

Name of Supplier (59)	Material Specification (15)	Dimensions (20)

* Supplemental information in the form of lists, sketches, or drawings may be used provided: (1) size is $8^1/_2 \times 11$; (2) information in items 1 through 6 on this Data Report is included on each sheet; and (3) each sheet is numbered and the number of sheets is recorded at the top of this form.

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FORM C-1 (Back — Pg. 2 of (56))

Certificate Holder's Serial No. (6)

13. Construction Report No. (93) Date

14. List of Penetrations (96)

Attach a complete list of penetrations (i.e., personnel locks, equipment hatch, electrical, etc.) to this report. State the type, size, manufacturer, and serial number.

15. Test Pressure (34) Date tested

16. Remarks: (46)

CERTIFICATION OF DESIGN

Design Specification on file at (109)
Design Specification certified by (49) P.E. State Reg. no.
Design Report on file at (48)
Design Report certified by (50) P.E. State Reg. no.

(69)

DESIGNER'S REPORT OF CERTIFICATION

I, the undersigned, representing the Designer and employed by have examined and evaluated the Construction Report for the component described in this Data Report. Following evaluation, the Construction Report has been certified and to the best of my knowledge and belief the Constructor has constructed this component in accordance with the rules of the ASME Code, Section III, Division 2, and the construction specification listed herein, and these construction specifications meet the requirements of the Design Specification.

Date Signed
(authorized representative)

P.E. State Reg. no.

(70)

CERTIFICATE OF CONSTRUCTION COMPLIANCE

We certify that the statements made in this report are correct and that all details of materials, construction, and workmanship of this component conform to the rules for construction of the ASME Code, Section III, Division 2, and the Construction Specifications listed herein.

Certificate of Authorization No. Expires

Date Constructor Signed
(N Certificate Holder) (authorized representative)

(71)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and employed by

of , have inspected the concrete reactor vessel or containment described in this Constructor's Data Report and state that to the best of my knowledge and belief this component has been constructed in accordance with the ASME Code, Section III, Division 2.

By signing this certificate neither the Authorized Inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this report. Furthermore, neither the Authorized Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Signed Commission (67)
(Authorized Nuclear Inspector) [National Board Number and Endorsement]

(07/11)

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Table V-1000
Guide for Preparation of Data Report Forms

Applies to Form													Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1		
X	X	X	X	X	X	X	X	X	X	X	X	X	(1)	Name and address as listed on ASME Certificate of Authorization.
X	X	X	X	X	X	X	X	X	X	X	X	X	(2)	Name and address of purchaser.
X	X	X	X	X	X	X	X	X	X	X	X	X	(3)	Name, address, and unit number of power plant where item is to be installed.
X	X	...	...	...	X	...	...	...	...	...	...	...	(4)	Type of installation intended (horizontal or vertical).
X	...	...	...	...	...	...	...	...	...	...	...	X	(5)	Description or application of vessel (reactor vessel tank, jacketed, heat exch., containment, etc.).
X	X	X	X	X	X	X	X	X	X	...	...	X	(6)	Item serial no.
X	X	X	X	X	X	X	X	X	X	X	...	X	(7)	Canadian Registration No. for item.
X	X	X	...	X	X	X	X	X	X	X	X	X	(8)	Indicate drawing numbers, including applicable revision number, that cover general assembly and list of materials. For Canadian registered vessels, the number of the drawing approved by provincial authorities.
X	X	X	X	X	X	X	X	X	X	...	X	X	(9)	National Board Number from Certificate Holder's series of Numbers to be stamped sequentially without skips or gaps.
X	X	X	X	X	X	X	...	X	X	...	X	X	(10)	Shall be the year certified by the Inspector on the Certificate Holder's Data Report.
X	X	X	...	X	X	X	X	X	X	X	X	X	(11)	ASME Code, Section III, Edition used for construction (e.g., 1986, etc.).
X	X	X	...	X	X	X	X	X	X	X	X	X	(12)	ASME Code, Section III, Addenda used for construction (e.g., A86, A87, etc.).
X	X	X	...	X	X	X	X	X	X	X	X	X	(13)	ASME Code Section III, Class 1, 2, 3, MC, CB, CC, or CS.
X	X	X	...	X	X	X	X	X	X	X	X	X	(14)	All Code Case Numbers and revisions used for construction, including design, fabrication, and materials used, must be listed. Where more space is needed use the "Remarks" section or list on a supplemental page. Code Cases used by Material Manufacturers and Material Suppliers shall be listed on the Data Report.
X	X	X	...	X	X	X	X	X	...	X	X	X	(15)	Material Specification Number. Show complete specification number and grade of actual material used. Material is to be as designated in ASME Code, Section III or as permitted in Code Cases.
X	X	X	...	...	X	...	...	X	...	...	X	X	(16)	For the "N" forms, Section II, Part D, "Min. Tensile Strength." For Form C-1, the ASTM specifications per Section III, Div. 2, CC-2400.
X	X	X	...	...	X	...	...	...	...	...	X	X	(17)	Nominal thickness.
X	X	X	...	...	X	...	...	...	...	...	...	...	(18)	Minimum thickness as specified by design.
X	X	X	...	...	X	...	...	...	...	...	X	X	(19)	Inside diameter.
X	X	X	...	X	X	...	...	...	...	...	X	X	(20)	Length or height overall, including heads.
X	X	...	...	...	X	...	...	...	...	...	...	...	(21)	Type of longitudinal joint (single butt-welded or double butt-welded joint).
X	X	...	...	...	X	...	...	...	...	...	...	...	(22)	Indicate postweld heat treatment (yes or no).
X	X	...	...	...	X	...	...	...	...	...	...	...	(23)	Indicate degree of radiographic examination (full, partial, spot, or none).
X	X	...	...	...	X	...	...	...	...	...	...	...	(24)	Weld joint efficiency as determined by design (%).
X	X	...	...	...	X	...	...	...	...	...	...	...	(25)	Type of girth joint (single butt welded or double butt welded).
X	X	...	...	...	X	...	...	...	...	...	...	...	(26)	Number of sections (courses) joined by girth welds.

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Table V-1000
Guide for Preparation of Data Report Forms (Cont'd)

Applies to Form													Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1		
X	X	...	...	...	X	...	...	...	...	...	...	...	(27)	Location of heads (top, bottom, ends, floating, or channel) and description of head geometry in applicable space.
X	X	...	...	...	X	...	...	...	...	...	...	...	(28)	Diameter and number of bolts.
X	X	...	...	...	...	...	...	...	...	...	...	...	(29)	Other fastenings such as quick opening; describe fully or attach sketch.
X	...	...	...	...	...	...	...	...	...	...	...	...	(30)	Describe type of jacket closure geometry, including dimensions or attach sketch (e.g., ogee and weld, bar, etc.).
X	X	X	...	X	X	...	...	...	...	...	...	X	(31)	Design Pressure specified in Design Specification.
X	X	X	...	X	X	...	...	...	...	...	...	X	(32)	Design Temperature specified in Design Specification.
X	X	X	...	X	X	X	...	X	...	...	X	...	(33)	Minimum pressure-test temperature as specified in Design Specification.
X	X	X	...	X	X	X	...	X	...	...	X	X	(34)	Circle type of test used and specify test pressure (pneumatic, hydrostatic, or combination test, as applicable).
X	...	...	...	...	...	...	...	...	...	...	...	...	(35)	Nominal diameter subject to pressure (refer to design documents).
X	...	...	...	...	...	...	...	...	...	...	...	...	(36)	Nominal thickness of tubesheet.
X	...	...	...	...	...	...	...	...	...	...	...	...	(37)	Method of tubesheet attachments (describe whether bolted, welded or other; attach sketch as necessary).
X	...	...	...	...	...	...	...	...	...	...	...	X	(38)	Specify nominal outside diameter.
...	...	...	...	...	...	...	...	...	X	...	...	...	(39)	Describe type of core support structure component (e.g., bottom grid, fuel support, top guide, etc.).
X	...	...	...	...	...	...	...	...	...	...	...	...	(40)	Nominal outside diameter of tubes.
X	...	...	...	...	...	...	...	...	...	...	...	...	(41)	Nominal wall thickness of tubes or gage size.
X	...	...	...	...	X	...	...	...	...	...	...	...	(42)	Number of tubes.
X	...	...	...	...	...	...	...	...	...	...	...	...	(43)	Actual tube configuration (straight or U-tube, etc.).
X	X	...	...	...	X	...	...	...	...	...	...	...	(44)	Nozzles, inspection, and safety valve openings; list all openings, regardless of size, penetrating pressure boundary.
X	X	...	...	...	X	...	...	...	...	...	...	...	(45)	Describe: (a) type of support (skirt, lugs, legs, etc.); (b) location of support (top, bottom, side, etc.); (c) method of attachment (bolted, welded, etc.).
X	X	X	X	X	X	X	X	X	X	X	X	X	(46)	Describe any additional Code requirements, restrictions, or additional information, including marking in lieu of stamping, not otherwise covered in Data Report. Include any required tests or examination not performed when Data Report is completed.
...	...	...	...	X	...	...	...	...	...	...	...	...	(47)	Specify the name and address of the organization where design information is on file.
...	...	...	...	X	...	...	...	...	...	...	...	X	(48)	Specify the name and address of the organization where Design Report or Design Specification is on file.
X	X	X	...	X	X	...	X	X	X	X	...	X	(49)	Enter the name of engineer who certified the Design Specification. Show state of registration and number. List name of individual only, signature not required. (Applies to Form N-2 only when Design Specification is required.)

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Table V-1000
Guide for Preparation of Data Report Forms (Cont'd)

Applies to Form													Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1		
X	X	X	...	X	...	...	X	X	X	X	...	X	(50)	Enter name of engineer who certified the Design Report. Show state of registration and number. List name of individual only, signature not required in space provided. (Applies to Form N-2 only when Design Report is required.)
...	...	...	X	X	...	X	...	...	...	...	...	...	(51)	System name is identified in the Design Specification (main steam, feedwater, safety injection, etc.).
...	...	...	X	...	...	...	...	...	...	...	...	...	(52)	Name of power plant designated by utility.
...	...	...	X	...	...	...	...	...	...	...	...	...	(53)	Number designation of unit.
...	...	...	X	...	...	...	...	...	...	...	...	...	(54)	Specify whether pump, valve, or safety relief valve.
...	...	...	...	...	...	X	...	...	...	...	...	...	(55)	Describe the piping, flanges, and fittings assembled and covered by Data Report including material specification [see Note (15)] or as an alternative, reference applicable sketch or drawing and attach to the Data Report.
X	X	X	X	X	X	X	X	X	X	X	X	X	(56)	When supplemental sheets are attached to the Data Report, each page is sequentially numbered and the total number of sheets is identified on the top right hand corner of the Data Report in the space provided.
...	...	...	...	...	...	...	...	...	...	...	...	...	(57)	Actual operating pressure for the system. (This may differ from the Design Pressure in the design certification block on back of Form N-5.)
...	...	...	...	...	...	...	...	...	...	...	...	...	(58)	Actual operating temperature for the system. (This may differ from the Design Pressure in the design certification block on back of Form N-5.)
...	...	...	...	X	...	...	...	...	...	...	...	X	(59)	Name of Material Manufacturer.
...	...	...	...	X	...	...	...	...	...	...	...	...	(60)	Name of person approving drawing or procedure.
...	...	...	...	X	...	...	...	...	...	...	...	...	(61)	List report or data sheet number as applicable.
...	...	...	...	X	...	...	...	...	...	...	...	...	(62)	Indicate valve, vessel, pump, or appurtenance.
...	...	...	...	X	...	...	...	...	...	...	...	X	(63)	Indicate piping subassembly or part.
...	...	...	...	X	...	...	...	...	...	X	...	...	(64)	List support identification number, model, or catalog item.
...	...	...	...	...	...	...	...	X	...	...	...	...	(65)	Indicate fluid used for National Board capacity test; the certified relieving capacity; and the percent overpressure used during the capacity test or pressure differential for vacuum relief valves.
...	...	...	...	...	...	...	...	X	...	...	...	...	(66)	Date of National Board capacity test certification.
X	X	X	X	X	X	X	X	X	X	...	X	X	(67)	The Inspector's National Board Commission No. and Endorsement must be shown.
X	X	X	X	X	X	X	X	X	X	...	X	...	(68)	This certificate is to be completed by the Certificate Holder and signed by the Authorized Nuclear Inspector who performs the inspection.
X	X	X	X	X	X	X	X	X	X	X	X	X	(69)	Certificate of Compliance block is to show the name of the responsible Certificate Holder as shown on his ASME Code Certificate of Authorization. This should be signed in accordance with the organizational authority defined in the Quality Assurance Program.

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Table V-1000
Guide for Preparation of Data Report Forms (Cont'd)

Applies to Form													Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1		
X	...	...	...	X	X	X	...	...	...	...	...	X	(70)	Certificate of Compliance block for field installation work or assembly is to be signed by the Certificate Holder's representative in charge of field fabrication. This should be signed in accordance with organizational authority defined in the Quality Assurance Program.
X	...	...	...	X	X	X	...	...	...	...	...	X	(71)	This certificate block is for the Authorized Inspector to sign for any field const ruction or assembly work. See Note (67) for National Board Commission Number requirements.
...	...	...	...	...	X	...	...	...	...	...	...	...	(72)	Specific gravity (density of fluid in relation to water, water being 1.0).
...	...	...	...	...	X	...	...	...	...	...	...	...	(73)	Maximum height of fluid in tank.
...	...	...	...	...	X	...	...	...	...	...	...	...	(74)	Temperature of test.
...	...	...	...	...	X	...	...	...	...	...	...	...	(75)	Impact test lateral expansion.
...	...	...	...	...	X	...	...	...	...	...	...	...	(76)	Minimum temperature [only when below −20°F (−29°C)] for design.
...	...	X	...	...	...	...	...	...	...	...	...	X	(77)	Identification number and title of Division 2 Construction Specification, including applicable revision number and the date of the revision (applies to Division 2 items only).
...	...	...	...	...	...	...	...	...	X	...	...	X	(78)	Design Specification identifying number, including applicable revision number and the date of the revision.
...	...	...	...	...	...	...	...	...	...	...	X	...	(79)	Heat or lot identification number of material used for fabrication.
...	...	...	...	...	...	X	...	...	...	...	...	...	(80)	List items included in the piping subassembly for which Certificate Holder's Data Reports have been completed, including serial number or National Board number and brief identifying description.
...	...	...	...	...	...	...	X	...	...	...	...	...	(81)	Valve pressure class designation per ASME B16.34.
...	...	...	...	...	...	...	...	X	...	...	...	...	(82)	Diameter of orifice opening.
...	...	...	...	...	...	...	X	X	...	...	...	X	(83)	Enter the nominal pipe size of inlet or outlet opening.
...	...	...	...	...	...	...	...	X	...	...	...	...	(84)	Indicate method of valve operation (spring, pilot operated, or power operated).
...	...	...	...	...	...	...	...	X	...	...	...	...	(85)	Set opening pressure of valve.
...	...	...	...	...	...	...	...	X	...	...	...	...	(86)	Difference between opening and reseating pressure.
...	...	...	...	...	...	...	...	X	...	...	...	...	(87)	Temperature rating of valve at the rated relieving capacity.
...	...	...	...	...	...	...	...	...	...	...	...	X	(88)	Concrete construction type (reinforced or prestressed).
...	...	...	...	...	...	...	...	...	...	...	...	X	(89)	Height of dome above springline.
...	...	...	...	...	...	...	...	...	...	...	...	X	(90)	Geometry of dome (spherical, ellipsoidal, conical, flat, etc.).
...	...	...	...	...	...	...	...	...	...	X	...	X	(91)	Foundation type and underlying substrata.
...	...	...	...	...	...	...	...	...	...	...	...	...	(92)	Date Design Report certified.
...	...	...	...	...	...	...	...	...	...	...	...	X	(93)	Construction report identification number and certification date.
...	...	...	...	...	...	...	...	...	...	...	...	X	(94)	Describe corrosion protection and coatings for post-tensioning tendons.
...	...	...	...	...	...	...	...	...	...	...	...	...	(95)	List all penetrations (openings), regardless of size, passing through the pressure boundary. State type (name), size, shape (circular, rectangular, etc.), and serial number.

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Table V-1000
Guide for Preparation of Data Report Forms (Cont'd)

Applies to Form													Ref. to Circled Nos. in Forms	Instructions for Completion
N-1	N-1A	N-2	N-3	N-5	N-6	NPP-1	NPV-1	NV-1	NCS-1	NS-1	NM-1	C-1		
...	...	...	...	...	...	...	...	...	...	X	...	X	(96)	Type of support (plate and shell, linear, standard support).
...	...	...	...	...	...	...	...	...	...	X	...	...	(97)	Design Report or load capacity data sheet (indicate which).
...	...	...	...	...	...	...	X	...	...	...	...	...	(98)	Disk differential pressure. A pressure equal to 110% of the valve pressure rating at 100°F (38°C).
...	...	...	...	...	...	...	...	...	...	X	...	...	(99)	Description of part (support) (snubber, sway brace, clevis, U-bolt, threaded rod with fastener, etc.).
...	...	...	...	...	...	...	X	...	...	...	...	...	(100)	Pump or valve (indicate which).
...	...	...	...	...	...	...	...	...	...	...	...	...	(101)	Brief description of service (feedwater, reactor cooling, safety injection, component cooling, etc.).
...	...	...	...	...	...	...	X	...	...	...	...	...	(102)	Mark number (the unique identification assigned by the Material Manufacturer to provide traceability to the CMTR).
...	...	...	...	...	...	...	X	...	...	...	...	...	(103)	Cold working pressure: the pressure at 100°F (38°C) as established by the pressure–temperature tables in ASME B16.34 (valves only).
...	...	...	...	...	...	...	X	...	...	...	...	...	(104)	Model number, series number, type (either a number traceable to the type or a description of the type for example gate, globe, butterfly, etc.).
...	...	...	...	...	...	...	X	...	...	...	...	...	(105)	Pressure equal to or greater than the Design Pressure specified in the Design Specification.
...	...	...	...	...	...	...	X	...	...	...	...	...	(106)	Coincident temperature per ASME B16.34 to pressure listed in Note (105). This temperature shall be equal to or greater than the Design Temperature specified in the Design Specification.
...	...	...	...	...	...	...	...	...	...	...	...	X	(107)	Minimum allowable yield strength specified in the appropriate Material Specification.
...	...	...	...	...	...	...	...	X	...	...	...	...	(108)	Indicate final ring position(s) with respect to an indicated reference point.
...	...	...	...	...	...	...	...	...	...	...	...	X	(109)	Name of certifying Professional Engineer. Show state of registration and registration number.

GENERAL NOTES:

(a) All blanks on the Data Report must contain an entry. If an entry is not applicable, enter "N/A" into the blank. Any quantity to which units apply shall be entered on the Manufacturer's Data Report with the chosen units.

(b) If space on Data Report is not sufficient for required information, either the remarks section is used or a supplementary sheet shall be attached and information listed by line number.

(c) These instructions constitute a nonmandatory guide for completion of Data Reports for items constructed to Section III Editions and Addenda prior to the Winter 1984 Addenda.

(d) The NS-1 Certificate is a Certificate of Conformance and is used in lieu of a Code Data Report form for welded supports.

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(15)

MANDATORY APPENDIX VI
ROUNDED INDICATIONS

ARTICLE VI-1000
ROUNDED INDICATIONS

VI-1100 ACCEPTANCE STANDARDS FOR RADIOGRAPHICALLY DETERMINED ROUNDED INDICATIONS IN WELDS

VI-1110 APPLICABILITY OF THESE STANDARDS

These standards are applicable to ferritic, austenitic, and nonferrous material.

VI-1120 TERMINOLOGY

VI-1121 Rounded Indications

Indications with a maximum length of three times the width or less on the radiograph are defined as rounded indications. These indications may be circular, elliptical, conical, or irregular in shape and may have tails. When evaluating the size of an indication, the tail shall be included. The indication may be from any source in the weld, such as porosity, slag, or tungsten.

VI-1122 Aligned Indications

A sequence of four or more rounded indications shall be considered to be aligned when they touch a line parallel to the length of the weld drawn through the center of the two outer rounded indications.

VI-1123 Thickness, *t*

t is the thickness of the weld, of the pressure-retaining material, or of the thinner of the sections being joined, whichever is least. If a full penetration weld includes a fillet weld, the thickness of the fillet weld throat shall be included in t.

VI-1130 ACCEPTANCE CRITERIA

VI-1131 Image Density

Density within the image of the indication may vary and is not a criterion for acceptance or rejection.

VI-1132 Relevant Indications (See Table VI-1132-1 for Examples)

Only those rounded indications which exceed the following dimensions shall be considered relevant:

(a) $\frac{1}{10}t$ for t less than $\frac{1}{8}$ in. (3 mm);

(b) $\frac{1}{64}$ in. (0.4 mm) for t equal to $\frac{1}{8}$ in. to $\frac{1}{4}$ in. (3 mm to 6 mm), inclusive;

(c) $\frac{1}{32}$ in. (0.8 mm) for t greater than $\frac{1}{4}$ in. to 2 in. (6 mm to 50 mm), inclusive;

(d) $\frac{1}{16}$ in. (1.5 mm) for t greater than 2 in. (50 mm).

VI-1133 Maximum Size of Rounded Indication (See Table VI-1132-1 for Examples)

The maximum permissible size of any indication shall be $\frac{1}{4}t$ or $\frac{5}{32}$ in. (4 mm), whichever is less, except that an isolated indication separated from an adjacent indication by 1 in. (25 mm) or more may be $\frac{1}{3}t$ or $\frac{1}{4}$ in. (6 mm), whichever is less. For t greater than 2 in. (50 mm), the maximum permissible size of an isolated indication shall be increased to $\frac{3}{8}$ in. (10 mm).

VI-1134 Aligned Rounded Indications

Aligned rounded indications are acceptable when the summation of the diameters of the indications is less than t in a length of $12t$ (see Figure VI-1134-1). The length of groups of aligned rounded indications and the spacing between the groups shall meet the requirements of Figure VI-1134-2.

VI-1135 Spacing

The distance between adjacent rounded indications is not a factor in determining acceptance or rejection, except as required for isolated indications or groups of aligned indications.

VI-1136 Rounded Indication Charts

(a) The rounded indications as determined from the radiographic film shall not exceed that shown in the charts.

(b) The charts in Figures VI-1136-1 through VI-1136-6 illustrate various types of assorted, randomly dispersed, and clustered rounded indications for different weld thicknesses greater than $\frac{1}{8}$ in. (3 mm). These charts represent the maximum acceptable concentration limits for rounded indications.

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Table VI-1132-1
Maximum Size of Nonrelevant Indications and Acceptable Rounded Indications — Examples Only

Thickness t, in. (mm)	Maximum Size of Acceptable Rounded Indication, in. (mm)		Maximum Size of Nonrelevant Indication, in. (mm)
	Random	Isolated	
< 1/8 (<3)	$\frac{1}{4}t$	$\frac{1}{3}t$	$\frac{1}{10}t$
1/8 (3)	0.031 (0.79)	0.042 (1.07)	0.015 (0.38)
3/16 (5)	0.047 (1.19)	0.063 (1.60)	0.015 (0.38)
1/4 (6)	0.063 (1.60)	0.083 (2.11)	0.015 (0.38)
5/16 (8)	0.078 (1.98)	0.104 (2.64)	0.031 (0.79)
3/8 (10)	0.091 (2.31)	0.125 (3.18)	0.031 (0.79)
7/16 (11)	0.109 (2.77)	0.146 (3.71)	0.031 (0.79)
1/2 (13)	0.125 (3.18)	0.168 (4.27)	0.031 (0.79)
9/16 (14)	0.142 (3.61)	0.188 (4.78)	0.031 (0.79)
5/8 (16)	0.156 (3.96)	0.210 (5.33)	0.031 (0.79)
11/16 (17)	0.156 (3.96)	0.230 (5.84)	0.031 (0.79)
3/4 to 2, incl. (19 to 50, incl.)	0.156 (3.96)	0.250 (6.35)	0.031 (0.79)
>2 (>50)	0.156 (3.96)	0.375 (9.53)	0.063 (1.60)

(c) The chart for each thickness range represents full-scale 6 in. (150 mm) radiographs and shall not be enlarged or reduced. The distributions shown are not necessarily the patterns that may appear on the radiograph, but are typical of the concentration and size of indications permitted.

VI-1137 Weld Thickness, t, Less Than 1/8 in. (3 mm)

For t less than 1/8 in. (3 mm), the maximum number of rounded indications shall not exceed 12 in a 6 in. (150 mm) length of weld. A proportionally fewer number of indications shall be permitted in welds less than 6 in. (150 mm) in length.

VI-1138 Clustered Indications

The illustrations for clustered indications show up to four times as many indications in a local area, as that shown in the illustrations for random indications. The length of an acceptable cluster shall not exceed the lesser of 1 in. (25 mm) or $2t$. Where more than one cluster is present, the sum of the lengths of the clusters shall not exceed 1 in. (25 mm) in a 6 in. (150 mm) length of weld.

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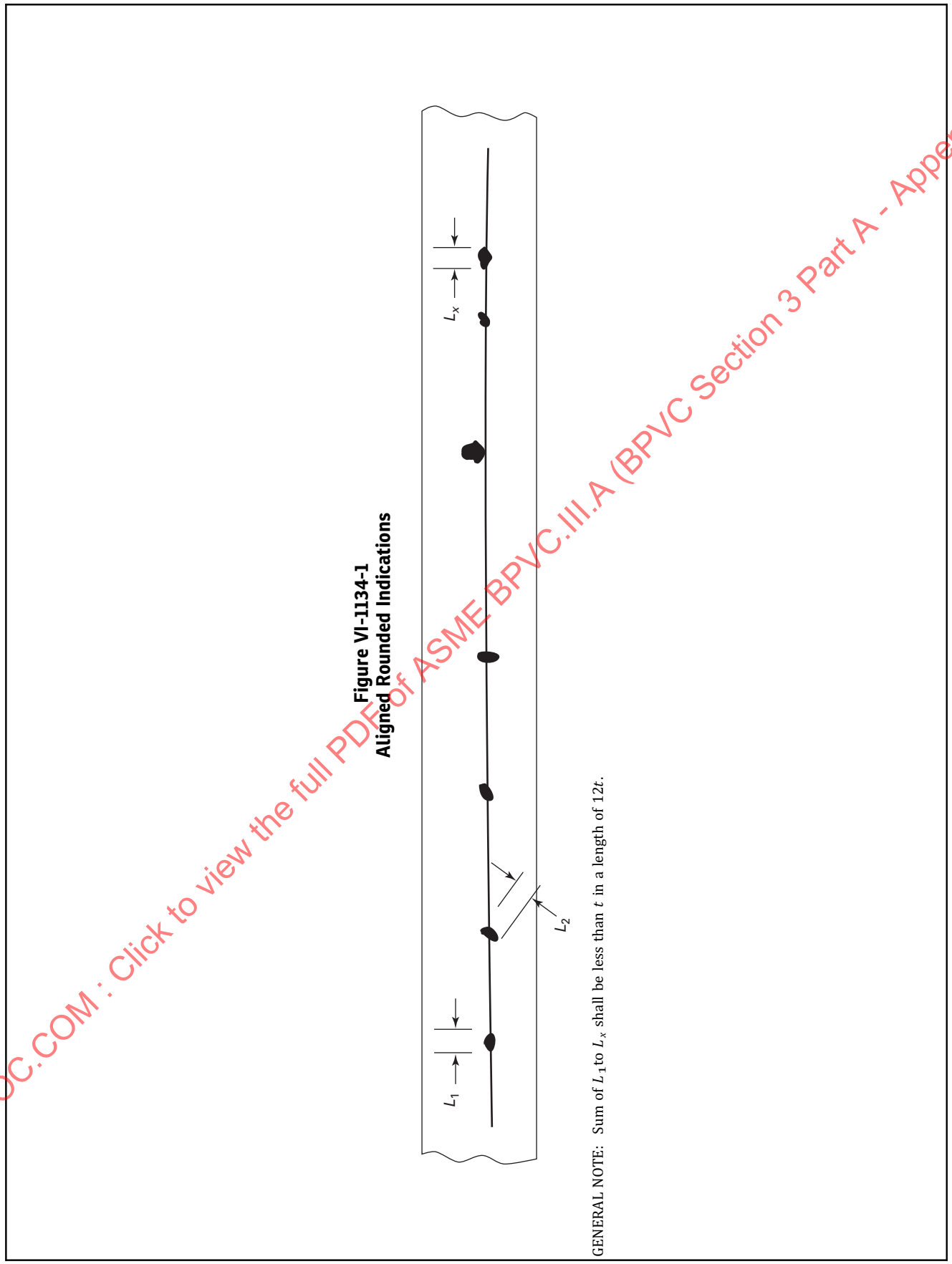

Figure VI-1134-1
Aligned Rounded Indications

GENERAL NOTE: Sum of L_1 to L_x shall be less than t in a length of $12t$.

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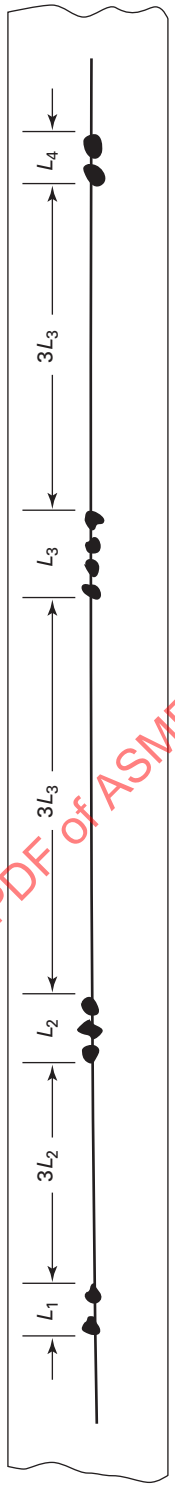
Figure VI-1134-2
Groups of Aligned Rounded Indications



Maximum Group Length	Minimum Group Spacing
L = $\frac{1}{4}$ in. (6 mm) for t less than $\frac{3}{4}$ in. (19 mm)	$3L$ where L is the length of the longest adjacent group being evaluated.
L = $\frac{1}{3}$ t for t equal to $\frac{3}{4}$ in. to $2\frac{1}{4}$ in. (19 mm to 57 mm)	
L = $\frac{3}{4}$ in. (19 mm) for t greater than $2\frac{1}{4}$ in. (57 mm)	

GENERAL NOTE: The sum of the group lengths shall be less than t in a length of $12t$.

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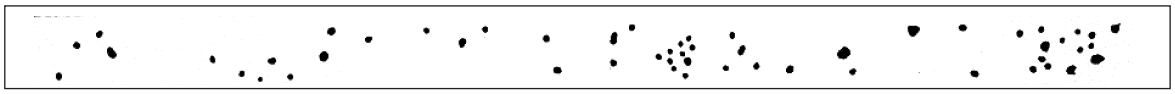
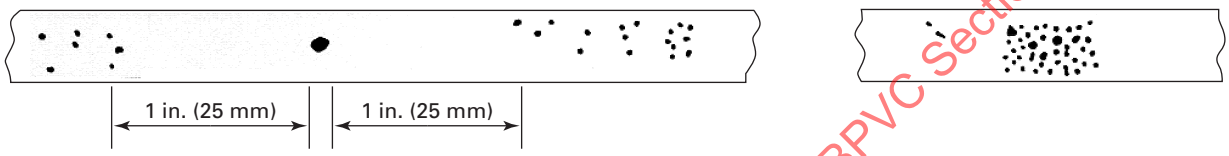
Figure VI-1136-1
Charts for t Equal to $\frac{1}{8}$ in. to $\frac{1}{4}$ in. (3 mm to 6 mm), Inclusive



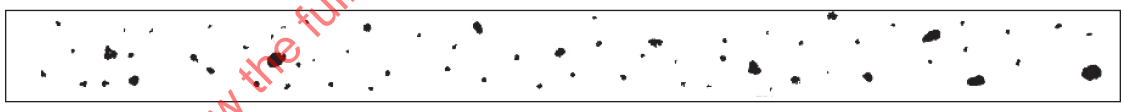
(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



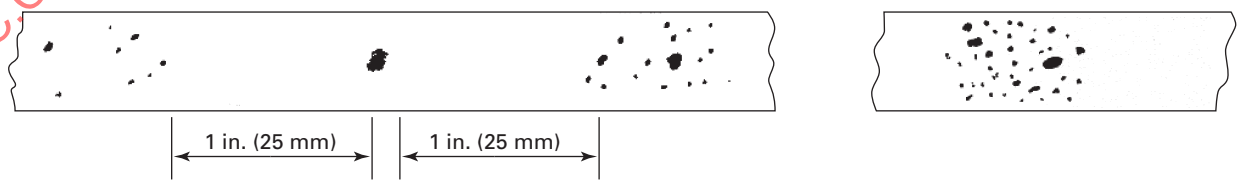
(b) Isolated Indication (Maximum size per Table VI-1132-1.)

(c) Cluster

Figure VI-1136-2
Charts for t Over $\frac{1}{4}$ in. to $\frac{3}{8}$ in. (6 mm to 10 mm), Inclusive



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



(b) Isolated Indication (Maximum size per Table VI-1132-1.)

(c) Cluster

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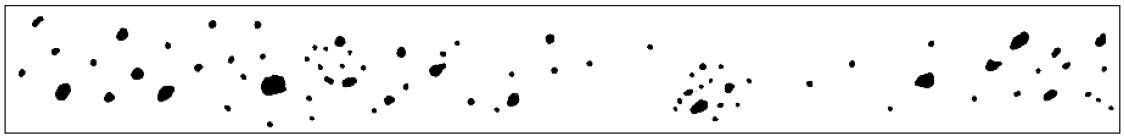
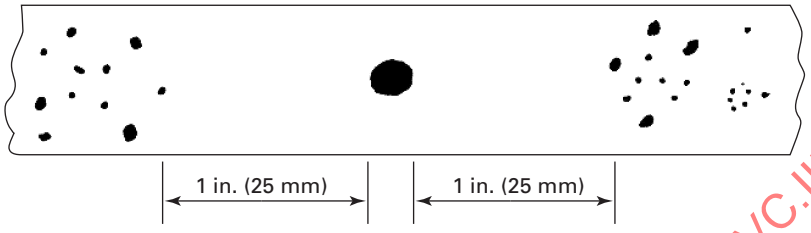
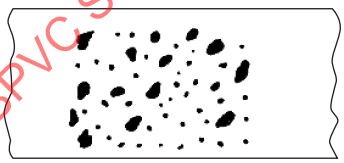
Figure VI-1136-3
Charts for t Over $^{3}/_{8}$ in. to $^{3}/_{4}$ in. (10 mm to 19 mm), Inclusive



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



(b) Isolated Indication (Maximum size per Table VI-1132-1.)



(c) Cluster

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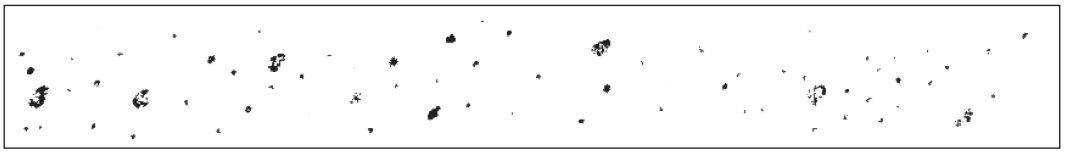
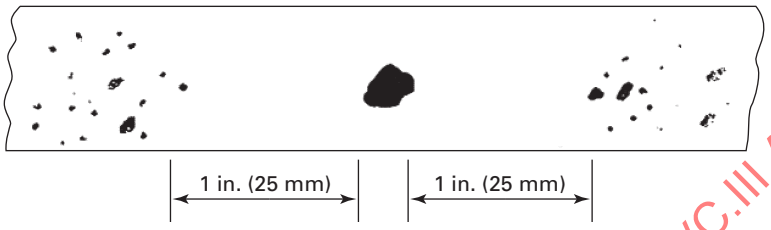
Figure VI-1136-4
Charts for t Over $^{3}/_{4}$ in. to 2 in. (19 mm to 50 mm), Inclusive



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



(b) Isolated Indication (Maximum size per Table VI-1132-1.)

(c) Cluster

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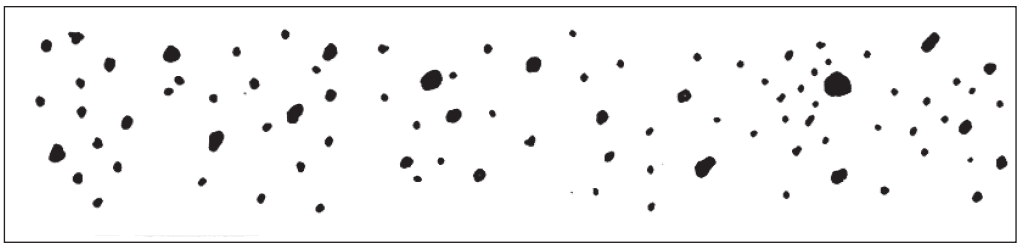
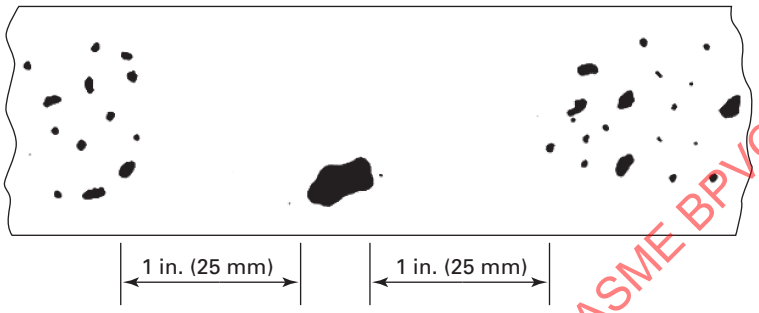
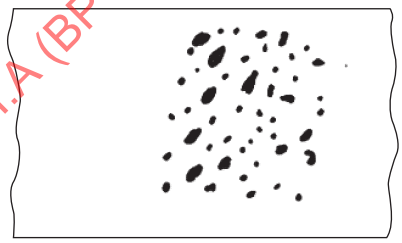
Figure VI-1136-5
Charts for *t* Over 2 in. to 4 in. (50 mm to 100 mm), Inclusive



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



(b) Isolated Indication (Maximum size per Table VI-1132-1.)



(c) Cluster

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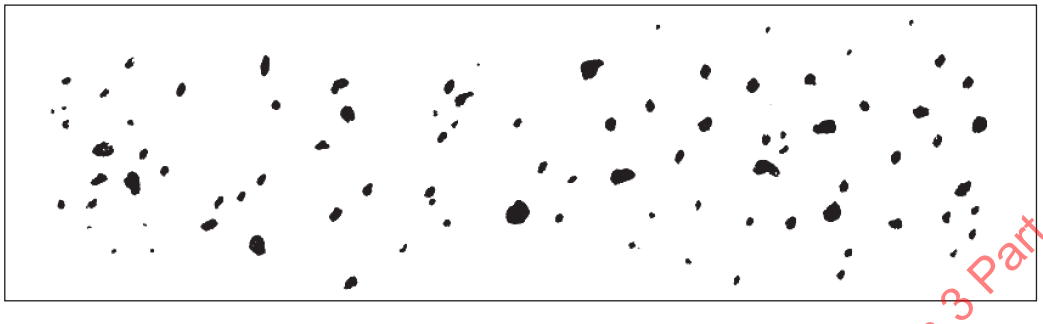
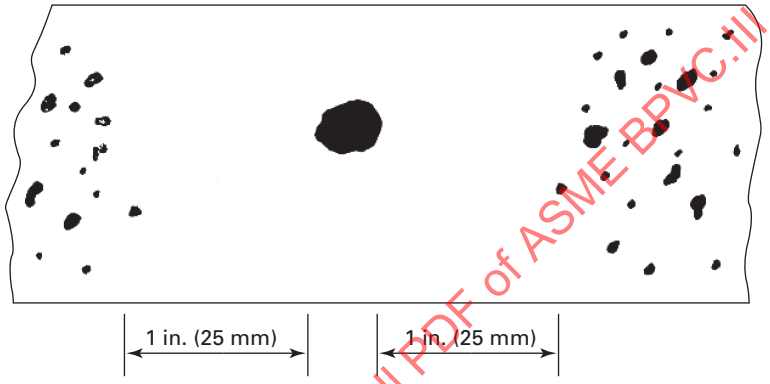
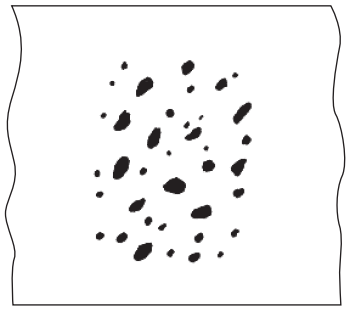
Figure VI-1136-6
Charts for t Over 4 in. (100 mm)



(a) Random Rounded Indications [Typical concentration and size permitted in any 6 in. (150 mm) length of weld.]



(b) Isolated Indication (Maximum size per Table VI-1132-1.)



(c) Cluster

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MANDATORY APPENDIX XI
RULES FOR BOLTED FLANGE CONNECTIONS FOR CLASS 2 AND 3 COMPONENTS AND CLASS MC VESSELS

ARTICLE XI-1000
INTRODUCTION

XI-1100 GENERAL REQUIREMENTS

XI-1110 SCOPE

(a) The rules in Mandatory Appendix XI apply specifically to the design of bolted flange connections for Class 2 and 3 components and Class MC vessels, and are to be used in conjunction with the applicable requirements in Subsections NC, ND, and NE. These rules provide only for hydrostatic end loads and gasket seating loads. For a discussion of design considerations for bolted flange connections, see Mandatory Appendix XII.

(b) Only circular flanges designated Class RF are covered by the rules of Mandatory Appendix XI. The design procedures for Class RF flanges, as defined in XI-3211, are given in XI-3200. For Class FF flanges, see Nonmandatory Appendix L.

(c) The flange design methods stipulated in this Appendix are primarily applicable to circular flanges under internal pressure. However, Class RF flanges may be used under external pressure when designed in accordance with XI-3260.

XI-1120 ELEMENTS INVOLVED IN FLANGE DESIGN

XI-1121 General Considerations

This method of designing a flange involves the selection of the gasket material, type and dimensions, flange facing, bolting, hub proportions, flange width, and flange thickness.[2] Flange dimensions shall be such that the stresses in the flange, calculated in accordance with XI-3240, do not exceed the allowable flange stresses specified in XI-3250 for Class RF flanges. All calculations shall use dimensions in the corroded condition.

XI-1122 Conditions for Which Design Calculations Shall Be Made

In the design of a bolted flange connection, complete calculations shall be made for two separate and independent sets of conditions which are defined in the following subparagraphs.

XI-1122.1 Design Conditions. The Design Conditions are those required to resist the hydrostatic end force of the Design Pressure tending to part the joint and to maintain on the gasket or joint contact surface sufficient compression to assure a tight joint, all at the Design Temperature. The minimum load is a function of the Design Pressure, the gasket material, and the effective gasket or contact area to be kept tight under pressure [as calculated by eq. XI-3221.1(1) and determines one of the two requirements for the amount of bolting A_{m1}. This load is also used for the design of the flange in eq. XI-3223(3).

XI-1122.2 Gasket Seating Conditions. The gasket seating conditions are those existing when the gasket or joint contact surface is seated by applying initial load with the bolts when assembling the joint, at atmospheric temperature and pressure. The minimum initial load considered to be adequate for proper seating is a function of the gasket material and the effective gasket or contact area to be seated [as calculated by eq. XI-3221.2(2)] and determines the other of the two requirements for the amount of bolting A_{m2}. For the design of the flange, this load is modified in eq. XI-3223(4) to take account of the Design Conditions when they govern the amount of bolting required A_m, as well as the amount of bolting actually provided A_b.

XI-1123 Bolted Flange Connections to External Piping

It is recommended that bolted flange connections conforming to the standards[3] listed in NC-3362, ND-3362, and NE-3362, as applicable, be used for connections to external piping. These standards may be used for other

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bolted flange connections within the limits of size in the standards and the pressure–temperature ratings permitted in Subsection NC, ND, or NE.

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ARTICLE XI-2000
MATERIALS FOR BOLTED FLANGE CONNECTIONS

XI-2100 MATERIAL REQUIREMENTS

XI-2110 GENERAL REQUIREMENTS

Materials used in the construction of bolted flange connections shall comply with the material requirements given in Subsection NC, ND, or NE, as applicable.

XI-2120 HEAT TREATMENT OF FLANGES

Flanges made from ferritic steel and designed in accordance with this Appendix shall be given a normalizing or full annealing heat treatment when the thickness of the flange section exceeds 3 in. (75 mm).

XI-2130 WELDABILITY OF FLANGES AND POSTWELD HEAT TREATMENT

Material on which welding is to be performed shall be proved of good weldable quality. Satisfactory qualification of the welding procedure under Section IX is considered as proof. Welding shall not be performed on steel that has a carbon content greater than 0.35%. All welding on flange connections shall comply with the requirements for postweld heat treatment given in Subsection NC, ND, or NE, as applicable.

XI-2140 FABRICATED HUBBED FLANGES

Fabricated hubbed flanges shall be in accordance with (a) through (c).

(a) Hubbed flanges may be fabricated from a hot-rolled or hot-forged billet. The axis of the finished flange shall be parallel to the long axis of the original billet. (This is not intended to imply that the axis of the finished flange and the original billet must be concentric.)

(b) Hubbed flanges, except as permitted in (a), shall not be machined from plate or bar stock material unless the material has been formed into a ring and, further, provided that

(1) in a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange (this is not intended to imply that the original plate surface be present in the finished flange).

(2) the joints in the ring are welded butt joints that conform to the requirements of the applicable Subsection. The thickness to be used to determine postweld heat treatment and radiography requirements shall be the lesser of t or $(A - B)/2$, when these symbols are as defined in XI-3130.

(c) The back of the flange and the outer surface of the hub shall be examined by the magnetic particle method (NC-2545) or the liquid penetrant method (NC-2546) to ensure that these surfaces are free from defects.

XI-2150 BOLTING MATERIALS

Bolts, studs, nuts, and washers shall comply with the requirements of the applicable Subsection. It is recommended that bolts and studs not be smaller than $\frac{1}{2}$ in. (13 mm). If bolts or studs smaller than $\frac{1}{2}$ in. (13 mm) are used, ferrous bolting material shall be of alloy steel. Precautions shall be taken to avoid overstressing small diameter bolts.

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ARTICLE XI-3000
DESIGN REQUIREMENTS

XI-3100 GENERAL REQUIREMENTS

XI-3110 SCOPE

(a) The rules of XI-3200 apply to Class RF flanges as defined in XI-3212.

(b) The flange design methods given in XI-3210 through XI-3250 apply to Class RF flanges under internal pressure. The flange design methods for Class RF flanges under external pressure or under both internal and external pressure are given in XI-3260.

XI-3120 TYPES OF FLANGES

For purposes of computation, there are three types as described in (a), (b), and (c).

(a) Loose Type Flanges. This type covers those designs in which the flange has no direct connection to the nozzle neck, vessel, or pipe wall and designs where the method of attachment is not considered to give the mechanical strength equivalent to integral attachment. Figure XI-3120-1 sketches (1), (1a), (2), (3), and (4) show typical loose type flanges and the location of the loads and moments; welds and other details of construction shall satisfy the dimensional requirements given in the referenced sketches.

(b) Integral Type Flanges. This type covers designs where the flange is cast or forged integrally with the nozzle neck, vessel, or pipe wall, butt welded thereto, or attached by other forms of arc or gas welding of such a nature that the flange and nozzle neck, vessel, or pipe wall is considered to be the equivalent of an integral structure. In welded construction, the nozzle neck, vessel, or pipe wall is considered to act as a hub. Figure XI-3120-1 sketches (5), (6), (6a), (6b), and (7) show typical integral type flanges and the location of the loads and moments; welds and other details of construction shall satisfy the dimensional requirements given in the referenced sketches.

(c) Optional Type Flanges. This type covers designs where the attachment of the flange to the nozzle neck, vessel, or pipe wall is such that the assembly is considered to act as a unit, which shall be calculated as an integral flange, except that for simplicity the designer may calculate the construction as a loose type flange provided none of the following values are exceeded:

$$g_0 = {}^{5}/_{8}\text{ in. (16 mm)}$$

$$B/g_0 = 300$$

$$P = 300\text{ psi (2 MPa)}$$

$$\text{Design Temperature} = 700°\text{F (370°C)}$$

Figure XI-3120-1 sketches (8), (8a), (8b), and (9) show typical optional type flanges. Welds and other details of construction shall satisfy the dimensional requirements given in those sketches.

XI-3130 NOMENCLATURE

The nomenclature defined below and shown in Figure XI-3120-1 is used in the equations for the design of flanges.

A = outside diameter of flange or, when slotted holes extend to the outside of the flange, the diameter to the bottom of the slots

A_b = actual total cross-sectional area of bolts at root of thread or section of least diameter under stress

A_m = total required cross-sectional area of bolts taken as the greater of A_{m1} and A_{m2}

A_{m1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the Design Conditions
= W_{m1}/S_b

A_{m2} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating
= W_{m2}/S_a

B = inside diameter of flange (when B is less than $20g_1$ it will be optional for the designer to substitute B_1 for B in the equation for longitudinal stress S_H)

b = effective gasket or joint contact surface seating width[2]

b_o = basic gasket seating width (Table XI-3221.1-2)

B_1 = $B + g_1$, for loose type hub flanges and also for integral type flanges when $f < 1$
= $B + g_0$, for integral type flanges when $f \geq 1$

C = bolt circle diameter

c = basic dimension used for the minimum sizing of welds; equal to t_n or t_D, whichever is less

C_b = effective width factor
= 0.5 for U.S. Customary calculations
= 2.52 for SI calculations

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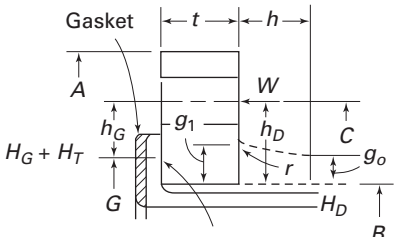
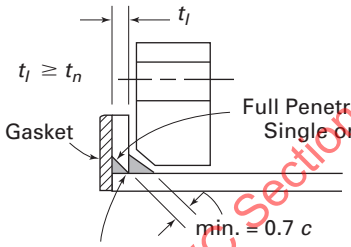
Figure XI-3120-1
Types of Flanges



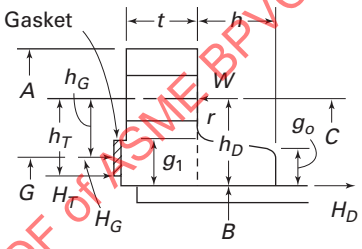
To be taken at mid-point of contact between flange and lap independent of gasket location

(1)



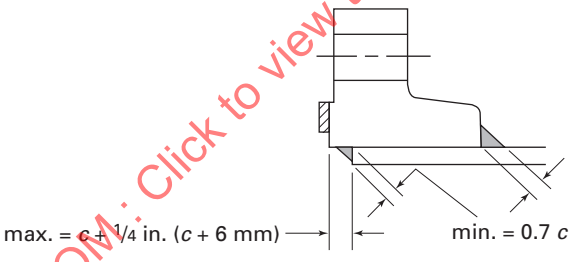
This weld may be machined to a corner radius to suit standard lap joint flanges

(1a)

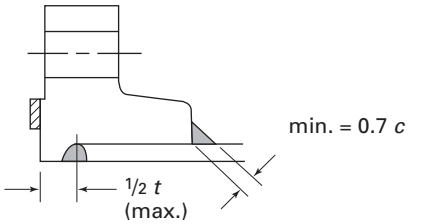


Screwed Flange With or Without Hub

(2)



(3) [Note (1)]



For hub tapers 6 deg or less, use $g_o = g_1$

(4) [Note (1)]

Loose-Type Flanges

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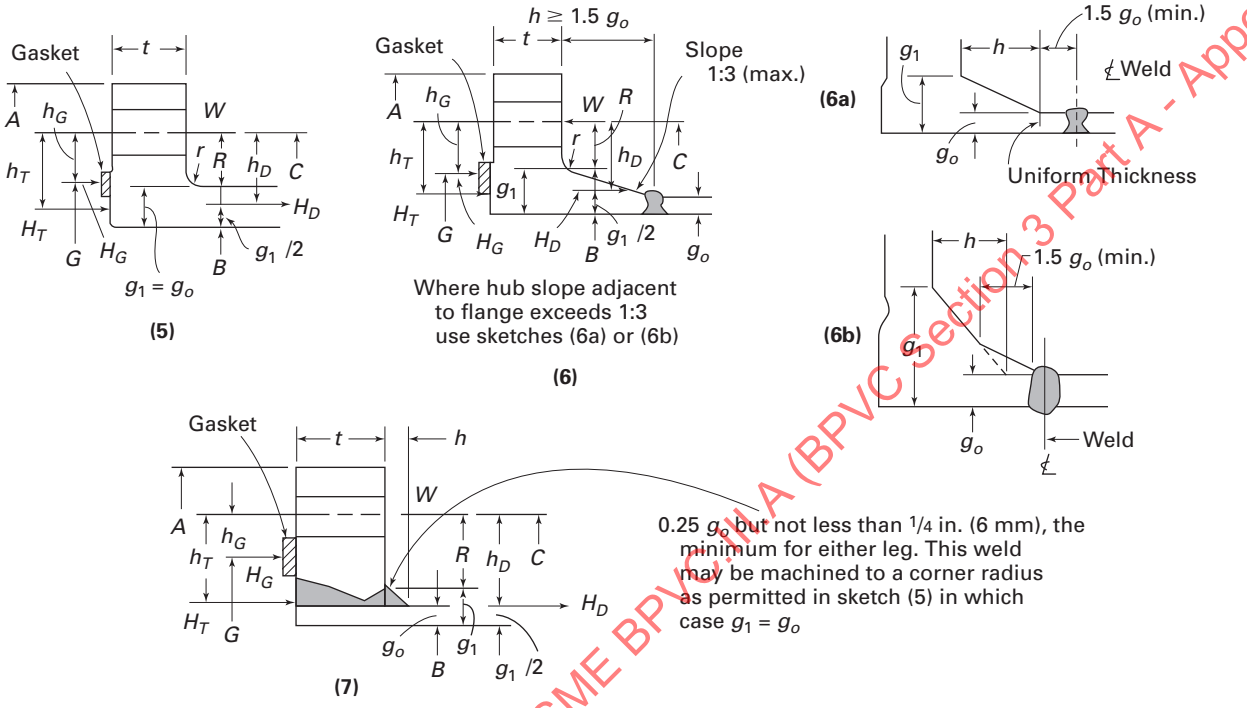
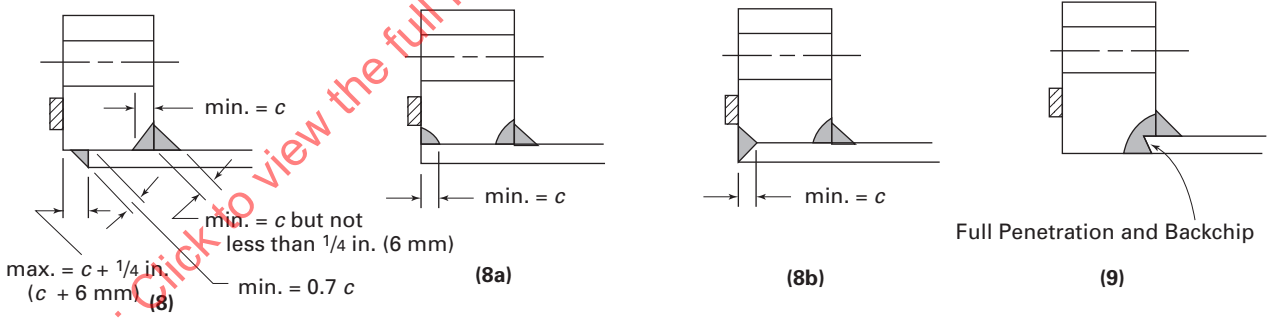
Figure XI-3120-1
Types of Flanges (Cont'd)



Integral-Type Flanges [Note (2)]



Optional-Type Flanges [Notes (3) and (4)]

NOTES:
(1) Loadings and dimensions not shown are the same as for sketch (2).
(2) Fillet radius r to be at least 0.25 g_o but not less than $^3/_{16}$ in. (5 mm). Added thickness greater than $^1/_{16}$ in. (1.5 mm) for raised face, tongue and groove, "O" rings, and ring joint facings shall be in excess of the required minimum flange thickness, t; those less than or equal to $^1/_{16}$ in. (1.5 mm) may be included in the required minimum flange thickness.
(3) These may be calculated as either loose or integral type [(c)].
(4) Loading and dimensions not shown are the same as for sketch (2) for loose type flanges or sketch (7) for integral type.

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d = factor, as follows:

= $\frac{U}{V}h_0{g_0}^2$ for integral type flanges

= $\frac{U}{V_L}h_0{g_0}^2$ for loose type flanges

e = factor, as follows:

= F/h_0 for integral type flanges
= F_L/h_0 for loose type flanges

F = factor for integral type flanges (Figure XI-3240-2)

f = hub stress correction factor for integral flanges from Figure XI-3240-6 (when greater than 1, this is the ratio of the stress in the small end of hub to the stress in the large end; for values below limit of Figure use $f = 1$)

F_L = factor for loose type flanges (Figure XI-3240-4)

G = diameter at location of gasket load reaction; except as noted in sketch (1) of Figure XI-3120-1, G is defined as follows for Class RF flanges (see Table XI-3221.1-2):

(a) when $b_0 \le 1/4$ in. (6 mm), G is the mean diameter of gasket contact face, in. (mm);

(b) when $b_0 > 1/4$ in. (6 mm), G is the outside diameter of gasket contact face less $2b$

g_0 = thickness of hub at small end

g_1 = thickness of hub at back of flange

H = total hydrostatic end force
= $0.785G^2P$

h = hub length

h_0 = factor equal to $\sqrt{Bg_0}$

H_D = hydrostatic end force on area inside of flange
= $0.785B^2P$

h_D = radial distance from the bolt circle to the circle on which H_D acts, as prescribed in Table XI-3230-1

H_G = gasket load (difference between flange design bolt load and total hydrostatic end force)
= $W - H$

h_G = radial distance from gasket load reaction to the bolt circle
= $(C - G)/2$

H_p = total joint contact surface compression load
= $2b \times 3.14GmP$

H_T = difference between total hydrostatic end force and the hydrostatic end force on area inside of flange
= $H - H_D$

h_T = radial distance from the bolt circle to the circle on which H_T acts, as prescribed in Table XI-3230-1

K = ratio of outside diameter of flange to inside diameter of flange
= A/B

L = factor
= $\frac{te + 1}{T} + \frac{t^3}{d}$

m = gasket factor obtained from Table XI-3221.1-1[2]

M_O = total moment acting upon the flange, for the Design Conditions or gasket seating, as may apply (XI-3230)

M_D = component of moment due to H_D
= $H_D h_D$

M_G = component of moment due to H_G
= $H_G h_G$

M_T = component of moment due to H_T
= $H_T h_T$

N = width used to determine the basic gasket seating width b_0, based upon the possible contact width of the gasket (Table XI-3221.1-2)

P = Design Pressure (for flanges subject to external pressure see XI-3260 for Class RF flanges)

R = radial distance from bolt circle to point of intersection of hub and back of flange (integral and hub flanges)
= $(C - B)/2 - g_1$

S_a = allowable bolt stress at atmospheric temperature (given in Section II, Part D, Subpart 1, Table 3)

S_b = allowable bolt stress at Design Temperature (given in Section II, Part D, Subpart 1, Table 3)

S_f = allowable design stress for material of flange at Design Temperature (Design Condition) or atmospheric temperature (gasket seating), as applicable (given in Section II, Part D, Subpart 1, Tables 1A and 1B, as applicable)

S_H = calculated longitudinal stress in hub

S_n = allowable design stress for material of nozzle neck, vessel, or pipe wall at Design Temperature (Design Condition) or atmospheric temperature (gasket seating) as applicable (given in Section II, Part D, Subpart 1, Tables 1A and 1B, as applicable)

S_R = calculated radial stress in flange

S_T = calculated tangential stress in flange

T = factor involving K (Figure XI-3240-1)

t = flange thickness

t_D = two times the thickness g_0, when the design is calculated as an integral flange, or two times the thickness of shell or nozzle wall required for internal pressure when the design is calculated as loose flange, but not less than 1/4 in. (6 mm)

t_n = nominal thickness of shell or nozzle wall to which flange or lap is attached, less corrosion allowance

U = factor involving K (Figure XI-3240-1)

V = factor for integral type flanges (Figure XI-3240-3)

V_L = factor for loose type flanges (Figure XI-3240-5)

W = flange design bolt load for the Design Conditions or gasket seating as applicable (XI-3223)

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w = width used to determine the basic gasket seating width b_0, based upon the contact width between the flange facing and the gasket (Table XI-3221.1-2)

W_{m1} = minimum required bolt load for the Design Conditions (XI-3220)

W_{m2} = minimum required bolt load for gasket seating (XI-3220)

Y = factor involving K (Figure XI-3240-1)

y = gasket or joint contact surface unit seating load[2]

Z = factor involving K (Figure XI-3240-1)

XI-3200 CLASS RF FLANGE DESIGN

XI-3210 GENERAL REQUIREMENTS

XI-3211 Definition of Class RF Flanges

Class RF flanges are circular flanges having gaskets which are entirely within the circle enclosed by the bolt holes and which have no contact outside this circle.

XI-3212 Acceptability

The requirements for acceptability of Class RF flange design are given in (a) and (b).

(a) The design shall be such that the general design requirements of NC-3100, ND-3100, or NE-3100, as appropriate, and the specific design requirements of this Subarticle are met.

(b) The designs shall be limited to the types of flanges defined in XI-3120.

XI-3220 BOLT LOADS AND BOLT AREAS

XI-3221 Determination of Bolt Loads

In the design of a bolted flange connection, calculations shall be made for each of the two conditions, namely, Design Loadings and gasket seating loads, and the more severe condition shall control. In the design of flange pairs used to contain a tubesheet of a heat exchanger, or any similar design where the flanges and/or gaskets may not be the same, loads shall be determined for the most severe condition of Design Loadings and/or gasket seating loads applied to each side at the same time. This most severe condition may be gasket seating on one flange with Design Loadings on the other, gasket seating on each flange at the same time, or Design Loadings on each flange at the same time.

XI-3221.1 Bolt Load for Design Conditions. The required bolt load for the Design Conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the Design Pressure on the area bounded by the diameter of gasket reaction and, in addition, to maintain on the gasket or joint contact surface a compression load H_p which experience has shown to be sufficient to ensure a tight joint. This compression load is expressed as a multiple m of the internal pressure. Its value is a function of the gasket material and construction.[2] The required bolt load for the Design Conditions W_{m1} is determined in accordance with eq. (1):

$$W_{m1} = H + H_p = 0.785G^2P + (2b \times 3.14GmP) \quad (1)$$

XI-3221.2 Bolt Load for Gasket Seating Condition. Before a tight joint can be obtained, it is necessary to seat the gasket or joint contact surface properly by applying a minimum initial load under atmospheric temperature conditions without the presence of internal pressure, which is a function of the gasket material and the effective gasket area to be seated. The minimum initial bolt load W_{m2} required for this purpose shall be determined in accordance with eq. (2):

$$W_{m2} = 3.14bGy \quad (2)$$

The need for providing sufficient bolt load to seat the gasket or joint contact surfaces in accordance with eq. (2) will prevail on many low pressure designs and with facings and materials that require a high seating load and where the bolt load computed by eq. XI-3221.1(1) for the Design Conditions is insufficient to seat the joint. Accordingly, it is necessary to furnish bolting and to pretighten the bolts to provide a bolt load sufficient to satisfy both of these requirements, each one being individually investigated. When eq. (2) governs, flange proportions will be a function of the bolting instead of internal pressure.

XI-3221.3 Bolt Load When Self-Energizing Gaskets Are Used. Bolt loads for flanges using gaskets of the self-energizing type differ from those shown in XI-3221.2 as stipulated in (a) and (b).

(a) The required bolt load for the Design Conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the Design Pressure on the area bounded by the outside diameter of the gasket. H_p is to be considered as zero for all self-energizing gaskets except certain seal configurations which generate axial loads which shall be considered.

(b) W_{m2} = 0. Self-energizing gaskets may be considered to require an inconsequential amount of bolting force to produce a seal. Bolting, however, shall be pretightened to provide a bolt load sufficient to withstand the hydrostatic end force H.

XI-3222 Total Required and Actual Bolt Areas A_m and A_b

The total cross-sectional area of bolts A_m required for both the Design Conditions and gasket seating is the greater of the values for A_{m1} and A_{m2}, where $A_{m1} = W_{m1}/S_b$ and $A_{m2} = W_{m2}/S_a$. A selection of bolts to be used shall be made such that the actual total cross-sectional areas of bolts A_b will not be less than A_m.

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(15)

Table XI-3221.1-1
Gasket Materials and Contact Facings
Gasket Factors, *m*, for Operating Conditions and Minimum Design Seating Stress, *y*

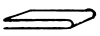
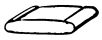
Gasket Material	Gasket Factor, *m*	Min. Design Seating Stress, *y*, psi (MPa)	Sketches	Facing Sketch and Column in Table XI-3221.1-2
Self-energizing types (0 rings, metallic, elastomer, other gasket types considered as self-sealing)	0	0 (0)	...	...
Elastomers without fabric or high percent of mineral fiber:				(1a),(1b),(1c),(1d), (4),(5); Column II
Below 75A Shore Durometer	0.50	0 (0)		
Below 75A Shore Durometer	1.00	200 (1.4)		
Mineral fiber with suitable binder for operating conditions:				(1a),(1b),(1c),(1d), (4),(5); Column II
1/8 in. (3 mm) thick	2.00	1,600 (11)		
1/16 in. (1.5 mm) thick	2.75	3,700 (26)		
1/32 in. (0.8 mm) thick	3.50	6,500 (45)		
Elastomers with cotton fabric insertion	1.25	400 (2.8)		(1a),(1b),(1c),(1d), (4),(5); Column II
Elastomers with mineral fiber fabric insertion (with or without wire reinforcement):				
3-ply	2.25	2,200 (15)		...
2-ply	2.50	2,900 (20)		(1a),(1b),(1c),(1d), (4),(5); Column II
1-ply	2.75	3,700 (26)		...
Vegetable fiber	1.75	1,100 (8)		(1a),(1b),(1c),(1d), (4),(5); Column II
Spiral-wound metal, mineral fiber filled:				(1a),(1b); Column II
Carbon	2.50	10,000 (69)		
Stainless or Monel	3.00	10,000 (69)		
Corrugated metal, mineral fiber inserted; or corrugated metal, jacketed mineral fiber filled:				(1a),(1b); Column II
Soft aluminum	2.50	2,900 (20)		
Soft copper or brass	2.75	3,700 (26)		
Iron or soft steel	3.00	4,500 (31)		
Monel or 4% to 6% chrome	3.25	5,500 (38)		
Stainless steels	3.50	6,500 (45)		
Corrugated metal:				(1a),(1b),(1c),(1d), Column II
Soft aluminum	2.75	3,700 (26)		
Soft copper or brass	3.00	4,500 (31)		
Iron or soft steel	3.25	5,500 (38)		
Monel or 4% to 6% chrome	3.50	6,500 (45)		
Stainless steels	3.75	7,600 (52)		
Flat metal, jacketed mineral fiber filled:				(1a), (1b), (1c) [Note (1)], (1d) [Note (1)], (2) [Note (1)]; Column II
Soft aluminum	3.25	5,500 (38)		
Soft copper or brass	3.50	6,500 (45)		
Iron or soft steel	3.75	7,600 (52)		
Monel	3.50	8,000 (55)		
4-6% chrome	3.75	9,000 (62)		
Stainless steels	3.75	9,000 (62)		
Grooved metal:				(1a),(1b),(1c),(1d), (2),(3); Column II
Soft aluminum	3.25	5,500 (38)		
Soft copper or brass	3.50	6,500 (45)		
Iron or soft steel	3.75	7,600 (52)		
Monel or 4-6% chrome	3.75	9,000 (62)		
Stainless steels	4.25	10,100 (70)		

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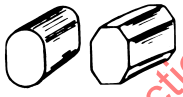
Table XI-3221.1-1
Gasket Materials and Contact Facings
Gasket Factors, *m*, for Operating Conditions and Minimum Design Seating Stress, *y* (Cont'd)

Gasket Material	Gasket Factor, *m*	Min. Design Seating Stress, *y*, psi (MPa)	Sketches	Facing Sketch and Column in Table XI-3221.1-2
Solid flat metal:				(1a),(1b),(1c),(1d), (2),(3),(4),(5); Column I
Soft aluminum	4.00	8,800 (61)		
Soft copper or brass	4.75	13,000 (90)		
Iron or soft steel	5.50	18,000 (124)		
Monel or 4% to 6% chrome	6.00	21,800 (150)		
Stainless steels	6.50	26,000 (180)		
Ring joint:				(6), Column I
Iron or soft steel	5.50	18,000 (124)		
Monel or 4% to 6% chrome	6.00	21,800 (150)		
Stainless steels	6.50	26,000 (180)		

GENERAL NOTE: This Table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b given in Table XI-3221.1-2. The design values and other details given in this Table are suggested only and are not mandatory.

NOTE:
(1) The surface of a gasket having a lap should not be against the nubbin.

XI-3223 Flange Design Bolt Load *W*

The bolt loads used in the design of the flange shall be the values obtained from eqs. (3) and (4). For Design Conditions,

$$W = W_{m1} \quad (3)$$

For gasket seating,

$$W = \frac{(A_m + A_b)S_a}{2} \quad (4)$$

In addition to the minimum requirements for safety, eq. (4) provides a margin against abuse of the flange from overbolting. Since margin against such abuse is needed primarily for the initial bolting up operation, which is done at atmospheric temperature and before application of internal pressure, the flange design is required to satisfy this loading only under such conditions.[4]

XI-3230 FLANGE MOMENTS

(a) In the calculation of flange stresses, the moment of a loading acting on the flange is the product of the load and its moment arm. The moment arm is determined by the relative position of the bolt circle with respect to that of the load producing the moment (Figure XI-3120-1). No consideration shall be given to any possible reduction in moment arm due to cupping of the flanges or due to inward shifting of the line of action of the bolts as a result thereof.

(b) For the Design Conditions, the total flange moment M_0 is the sum of the three individual moments M_D, M_T, and M_G, as defined in XI-3130, and based on the flange design bolt load of eq. XI-3223(3) with moment arms as given in Table XI-3230-1.

(c) For gasket seating, the total flange moment M_0 is based on the flange design bolt load of eq. XI-3223(4), which is opposed only by the gasket load, in which case:

$$M_0 = W\frac{(C - G)}{2} \quad (5)$$

XI-3240 CALCULATION OF FLANGE STRESSES

The stresses in the flange shall be determined for both the Design Conditions and gasket seating, whichever controls, in accordance with the equations in (a) or (b).

(a) For integral type flanges and all hub type flanges

Longitudinal hub stress

$$S_H = fM_0 / Lg_1^2 B \quad (6)$$

Radial flange stress

$$S_R = \left[(1.33te + 1)M_0\right] / Lt^2 B \quad (7)$$

Tangential flange stress

$$S_T = \left(YM_0 / t^2 B\right) - ZS_R \quad (8)$$

(b) For loose type ring flanges including optional type calculated as loose type having a rectangular cross section

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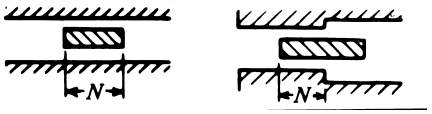
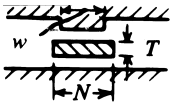
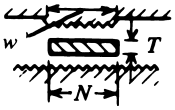
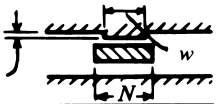
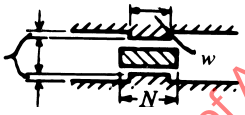
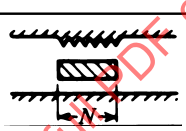
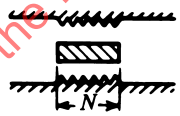
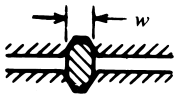
Table XI-3221.1-2
Effective Gasket Width

	Facing Sketch (Exaggerated)		Basic Gasket Seating Width, b_o	
			Column I	Column II
(1a)			$\frac{N}{2}$	$\frac{N}{2}$
(1b) [Note (1)]				
(1c)		$w \le N$	$\frac{w + T}{2}\left(\frac{w + N}{4}\text{max.}\right)$	$\frac{w + T}{2}\left(\frac{w + N}{4}\text{max.}\right)$
(1d) [Note (1)]		$w \le N$		
(2)	$\frac{1}{64}$ in. (0.4 mm) Nubbin 	$w \le N/2$	$\frac{w + N}{4}$	$\frac{w + 3N}{8}$
(3)	$\frac{1}{64}$ in. (0.4 mm) Nubbin 	$w \le N/2$	$\frac{N}{4}$	$\frac{3N}{8}$
(4) [Note (1)]			$\frac{3N}{8}$	$\frac{7N}{16}$
(5) [Note (1)]			$\frac{N}{4}$	$\frac{3N}{8}$
(6)			$\frac{w}{8}$	...

Location of Gasket Load Reaction

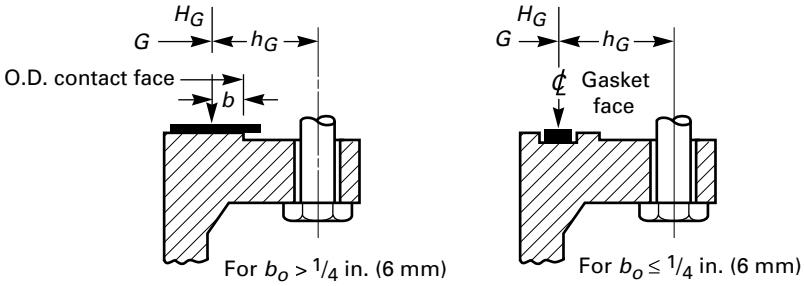


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Table XI-3221.1-2
Effective Gasket Width (Cont'd)

GENERAL NOTES:
(a) Effective Gasket Seating Width:
$b = b_o$ when $b_o \le 1/4$ in. (6 mm)
$b = C_b\sqrt{b_o}$ when $b_o > 1/4$ in. (6 mm)
(b) The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of bolt holes.

NOTE:
(1) Where serrations do not exceed 1/64 in. (0.4 mm) depth and 1/32 in. (0.8 mm) width spacing, sketches (1b) and (1d) shall be used.

$$S_T = YM_0/t^2B$$
$$S_R = 0 \qquad (9)$$
$$S_H = 0$$

XI-3250 ALLOWABLE FLANGE DESIGN STRESSES

The flange stresses calculated by the equations in XI-3240 shall not exceed the values given in (a) through (f).

(a) The longitudinal hub stress S_H shall not be greater than the smaller of $1.5S_f$ or $1.5S_n$ for optional type flanges designed as integral [Figure XI-3120-1 sketches (8), (8a), (8b), and (9)], and also for integral type flanges [Figure XI-3120-1 sketch (7)] where the neck material constitutes the hub of the flange.

(b) The longitudinal hub stress S_H shall not be greater than the smaller of $1.5S_f$ or $1.5S_n$ for integral type flanges with hub welded to the neck, pipe, or vessel wall [Figure XI-3120-1 sketches (6), (6a), and (6b)].

(c) The radial flange stress S_R shall not be greater than S_f.

Table XI-3230-1
Moment Arms for Flange Loads

Flange Type	h_D	h_T	h_G
Integral type flanges [see Figure XI-3120-1 sketches (5), (6), (6a), (6b), (7), (8), (8a), (8b), and (9)]	$R + 0.5g_1$	$\frac{R + g_1 + h_G}{2}$	$\frac{C - G}{2}$
Loose type, except lap joint flanges [Figure XI-3120-1 sketches (2), (3), and (4)]; and optional type flanges [Figure XI-3120-1 sketches (8), (8a), (8b), and (9)]	$\frac{C - B}{2}$	$\frac{h_D + h_G}{2}$	$\frac{C - G}{2}$
Lap joint flanges [Figure XI-3120-1 sketches (1) and (1a)]	$\frac{C - B}{2}$	$\frac{C - G}{2}$	$\frac{C - G}{2}$

(d) The tangential flange stress S_T shall not be greater than S_f.

(e) Also $(S_H + S_R)/2$ shall not be greater than S_f and $(S_H + S_T)/2$ shall not be greater than S_f.

(f) In the case of loose type flanges with laps, as shown in Figure XI-3120-1 sketches (1) and (1a), where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed $0.8S_n$ for the material of the lap, as defined in XI-3130. In the case of welded flanges, shown in Figure XI-3120-1 sketches (3), (4), (7), (8), (8a), and (8b), where the nozzle neck, vessel, or pipe wall extends near to the flange face and may form the gasket contact face, the shearing stress carried by the welds shall not exceed $0.8S_n$. The shearing stress shall be calculated on the basis of W_{m1} or W_{m2} (as defined in XI-3130), whichever is greater. Similar cases where flange parts are subjected to shearing stress shall be governed by the same requirements.

XI-3260 FLANGES SUBJECT TO EXTERNAL PRESSURE

XI-3261 Flanges for External Pressure Only

The design of flanges for external pressure[5] only shall be based on the equations given in XI-3240 for internal pressure except that for Design Conditions,

$$M_0 = H_D(h_D - h_G) + H_T(h_T - h_G) \qquad (10)$$

for gasket seating,

$$M_0 = Wh_G \qquad (11)$$

In eqs. (10) and (11):

$$W = \frac{A_{m2} + A_b}{2}S_a$$
$$H_D = 0.785\,B^2P_e$$
$$H_T = H - H_D$$
$$H = 0.785G^2P_e$$

P_e = external Design Pressure, psi (kPa)

See XI-3130 for definitions of other symbols.

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Table XI-3240-1
Flange Factors in Formula Form

Integral Flange [Note (1)]

For F (Figure XI-3240-2) use:

$$F = \frac{E_6}{\left(\frac{C}{2.73}\right)^{1/4}\frac{(1+A)^3}{C}}$$

For V (Figure XI-3240-3) use:

$$V = \frac{E_4}{\left(\frac{2.73}{C}\right)^{1/4}(1+A)^3}$$

For f (Figure XI-3240-6) use:

$$f = C_{36}/(1+A)$$

Loose Hub Flange [Note (2)]

For F_L (Figure XI-3240-4) use:

$$F_L = -\frac{C_{18}\left(\frac{1}{2}+\frac{A}{6}\right)+C_{21}\left(\frac{1}{4}+\frac{11A}{84}\right)+C_{24}\left(\frac{1}{70}+\frac{A}{105}\right)-\left(\frac{1}{40}+\frac{A}{72}\right)}{\left(\frac{C}{2.73}\right)^{1/4}\frac{(1+A)^3}{C}}$$

For V_L (Figure XI-3240-5) use:

$$V_L = \frac{\frac{1}{4}-\frac{C_{24}}{5}-\frac{3C_{21}}{2}-C_{18}}{\left(\frac{2.73}{C}\right)^{1/4}(1+A)^3}$$

For f (Figure XI-3240-6) use:

$$f = 1$$

Equations

(1) $A = (g_1/g_0) - 1$

(2) $C = 43.68(h/h_0)^4$

(3) $C_1 = 1/3 + A/12$

(4) $C_2 = 5/42 + 17A/336$

(5) $C_3 = 1/210 + A/360$

(6) $C_4 = 11/360 + 59A/5040 + (1 + 3A)/C$

(7) $C_5 = 1/90 + 5A/1008 - (1 + A)^3/C$

(8) $C_6 = 1/120 + 17A/5040 + 1/C$

(9) $C_7 = 215/2772 + 51A/1232 + (60/7 + 225A/14 + 75A^2/7 + 5A^3/2)/C$

(10) $C_8 = 31/6930 + 128A/45,045 + (6/7 + 15A/7 + 12A^2/7 + 5A^3/11)/C$

(11) $C_9 = 533/30,240 + 653A/73,920 + (1/2 + 33A/14 + 39A^2/28 + 25A^3/84)/C$

(12) $C_{10} = 29/3780 + 3A/704 - (1/2 + 33A/14 + 81A^2/28 + 13A^3/12)/C$

(13) $C_{11} = 31/6048 + 1763A/665,280 + (1/2 + 6A/7 + 15A^2/28 + 5A^3/42)/C$

(14) $C_{12} = 1/2925 + 71A/300,300 + (8/35 + 18A/35 + 156A^2/385 + 6A^3/55)/C$

(15) $C_{13} = 761/831,600 + 937A/1,663,200 + (1/35 + 6A/35 + 11A^2/70 + 3A^3/70)/C$

(16) $C_{14} = 197/415,800 + 103A/332,640 - (1/35 + 6A/35 + 17A^2/70 + A^3/10)/C$

(17) $C_{15} = 233/831,600 + 97A/554,400 + (1/35 + 3A/35 + A^2/14 + 2A^3/105)/C$

(18) $C_{16} = C_1C_7C_{12} + C_2C_8C_3 + C_3C_8C_2 - (C_3{}^2C_7 + C_8{}^2C_1 + C_2{}^2C_{12})$

(19) $C_{17} = [C_4C_7C_{12} + C_2C_8C_{13} + C_3C_8C_9 - (C_{13}C_7C_3 + C_8{}^2C_4 + C_{12}C_2C_9)]/C_{16}$

(20) $C_{18} = [C_5C_7C_{12} + C_2C_8C_{14} + C_3C_8C_{10} - (C_{14}C_7C_3 + C_8{}^2C_5 + C_{12}C_2C_{10})]/C_{16}$

(21) $C_{19} = [C_6C_7C_{12} + C_2C_8C_{15} + C_3C_8C_{11} - (C_{15}C_7C_3 + C_8{}^2C_6 + C_{12}C_2C_{11})]/C_{16}$

(22) $C_{20} = [C_1C_9C_{12} + C_4C_8C_3 + C_3C_{13}C_2 - (C_3{}^2C_9 + C_{13}C_8C_1 + C_{12}C_4C_2)]/C_{16}$

(23) $C_{21} = [C_1C_{10}C_{12} + C_5C_8C_3 + C_3C_{14}C_2 - (C_3{}^2C_{10} + C_{14}C_8C_1 + C_{12}C_5C_2)]/C_{16}$

(24) $C_{22} = [C_1C_{11}C_{12} + C_6C_8C_3 + C_3C_{15}C_2 - (C_3{}^2C_{11} + C_{15}C_8C_1 + C_{12}C_6C_2)]/C_{16}$

(25) $C_{23} = [C_1C_7C_{13} + C_2C_9C_3 + C_4C_8C_2 - (C_3C_7C_4 + C_8C_9C_1 + C_2{}^2C_{13})]/C_{16}$

(26) $C_{24} = [C_1C_7C_{14} + C_2C_{10}C_3 + C_5C_8C_2 - (C_3C_7C_5 + C_8C_{10}C_1 + C_2{}^2C_{14})]/C_{16}$

(27) $C_{25} = [C_1C_7C_{15} + C_2C_{11}C_3 + C_6C_8C_2 - (C_3C_7C_6 + C_8C_{11}C_1 + C_2{}^2C_{15})]/C_{16}$

(28) $C_{26} = -(C/4)^{1/4}$

(29) $C_{27} = C_{20} - C_{17} - 5/12 - [C_{17}(C/4)^{1/4}]$

(30) $C_{28} = C_{22} - C_{19} - 1/12 - [C_{19}(C/4)^{1/4}]$

(31) $C_{29} = -(C/4)^{1/2}$

(32) $C_{30} = -(C/4)^{3/4}$

(33) $C_{31} = 3A/2 + C_{17}(C/4)^{3/4}$

(34) $C_{32} = 1/2 + C_{19}(C/4)^{3/4}$

(35) $C_{33} = 0.5C_{26}C_{32} + C_{28}C_{31}C_{29} - (0.5C_{30}C_{28} + C_{32}C_{27}C_{29})$

(36) $C_{34} = 1/12 + C_{18} - C_{21} + C_{18}(C/4)^{1/4}$

(37) $C_{35} = -C_{18}(C/4)^{3/4}$

(38) $C_{36} = (C_{28}C_{35}C_{29} - C_{32}C_{34}C_{29})/C_{33}$

(39) $C_{37} = [0.5C_{26}C_{35} + C_{34}C_{31}C_{29} - (0.5C_{30}C_{34} + C_{35}C_{27}C_{29})]/C_{33}$

(40) $E_1 = C_{17}C_{36} + C_{18} + C_{19}C_{37}$

(41) $E_2 = C_{20}C_{36} + C_{21} + C_{22}C_{37}$

(42) $E_3 = C_{23}C_{36} + C_{24} + C_{25}C_{37}$

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Table XI-3240-1
Flange Factors in Formula Form (Cont'd)

Integral Flange [Note (1)]	Loose Hub Flange [Note (2)]
(43) $E_4 = {}^1\!/_4 + C_{37}/12 + C_{36}/4 - E_3/5 + 3E_2/2 - E_1$	(44) $E_5 = E_1\,({}^1\!/_2 + A/6) + E_2\,({}^1\!/_4 + 11A/84) + E_3\,({}^1\!/_{70} + A/105)$
(45) $E_6 = E_5 - C_{36}\,({}^7\!/_{120} + A/36 + 3\,A/C) - {}^1\!/_{40} - A/72 - C_{37}\,({}^1\!/_{60} + A/120 + 1/C)$	

NOTES:

(1) Except for the case when $g_1 = g_0$, the values used in the Integral Flange equations are determined by using Eqs. (1) through (45), which are based on the values of g_1, g_0, h, and h_0 (see XI-3130 for definitions). When $g_1 = g_0$, Eqs. (1) through (45) are not required and should not be used. For this case ($g_1 = g_0$), $F = 0.908920$, $V = 0.550103$, and $f = 1$.

(2) The values used in the Loose Hub Flange equations are determined by using Eqs. (1) through (5), (7), (9), (10), (12), (14), (16), (18), (20), (23), and (26), which are based on the values of g_1, g_0, h, and h_0 (see XI-3130 for definitions).

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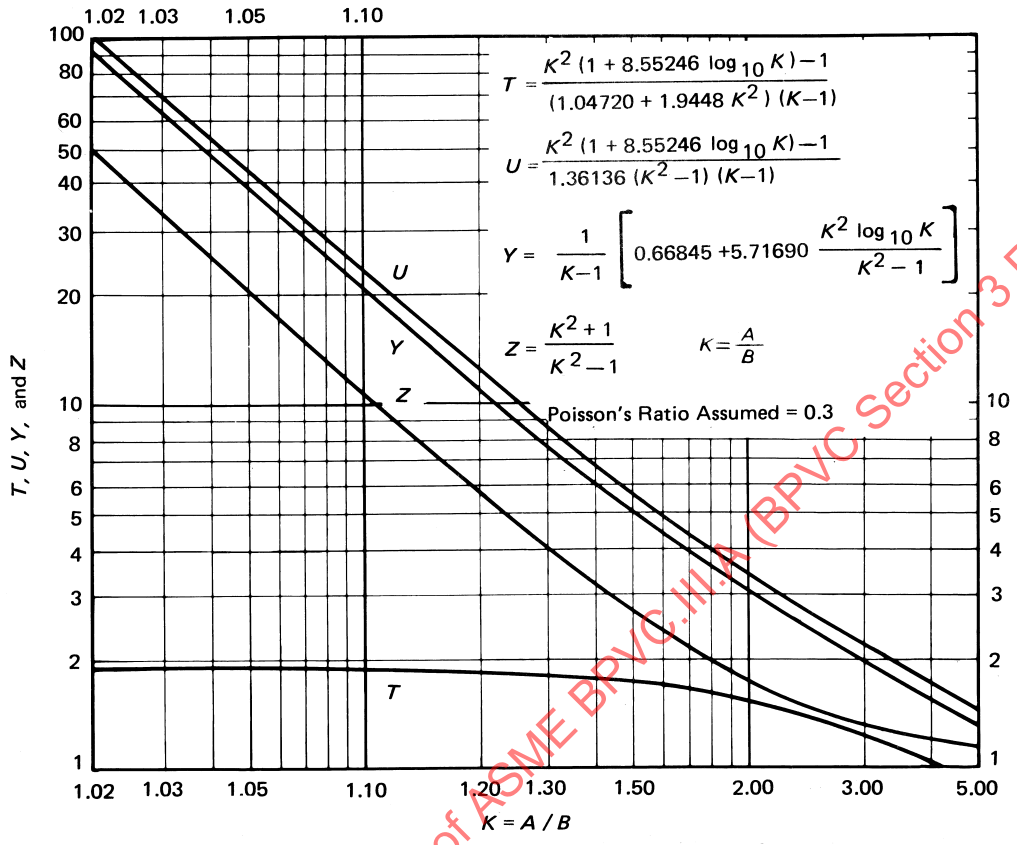
Figure XI-3240-1
Values of T, U, Y, and Z (Terms Involving K)



GENERAL NOTE: The calculation of values of T, U, Y, and Z for values of K outside the boundaries of the graph is acceptable.

XI-3262 Flanges for Both External and Internal Pressure

When flanges are subject at different times during service to external or internal pressure, the design shall satisfy the external pressure design requirements[6] given in XI-3261 and the internal pressure design requirements given elsewhere in this Appendix.

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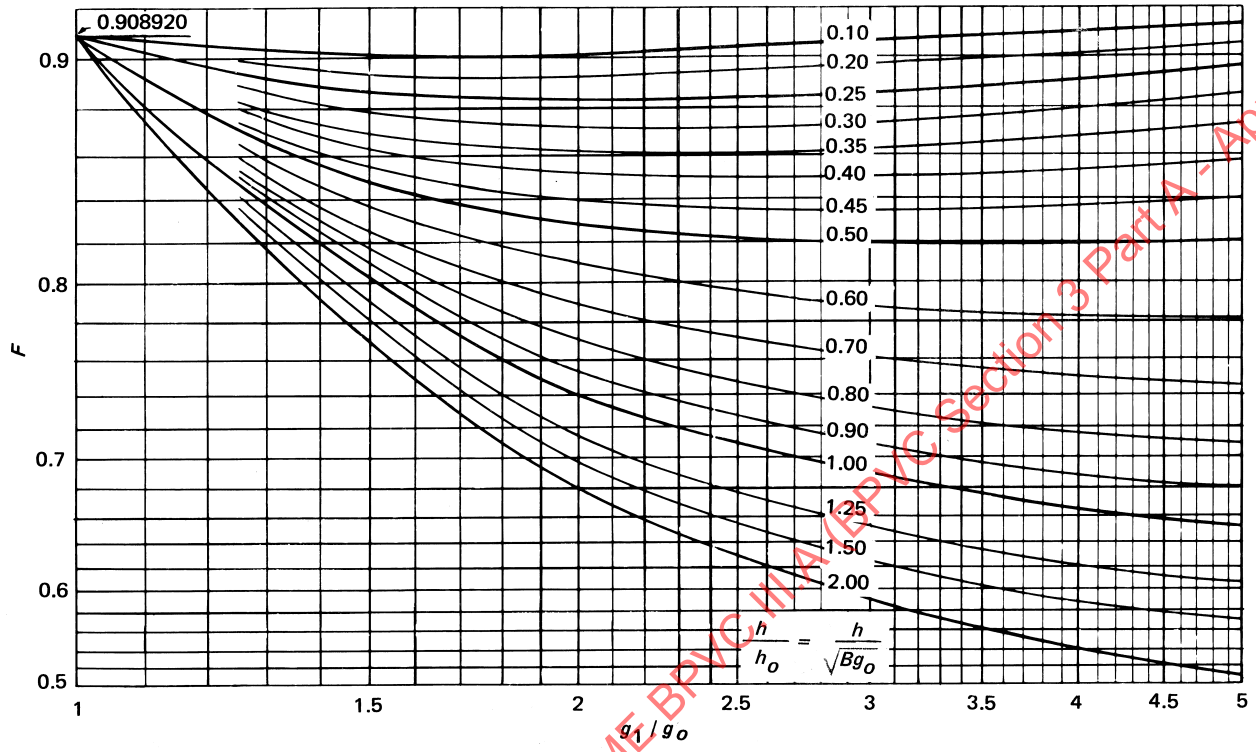
Figure XI-3240-2
Values of *F* (Integral Flange Factors)



GENERAL NOTE: See Table XI-3240-1 for equations.

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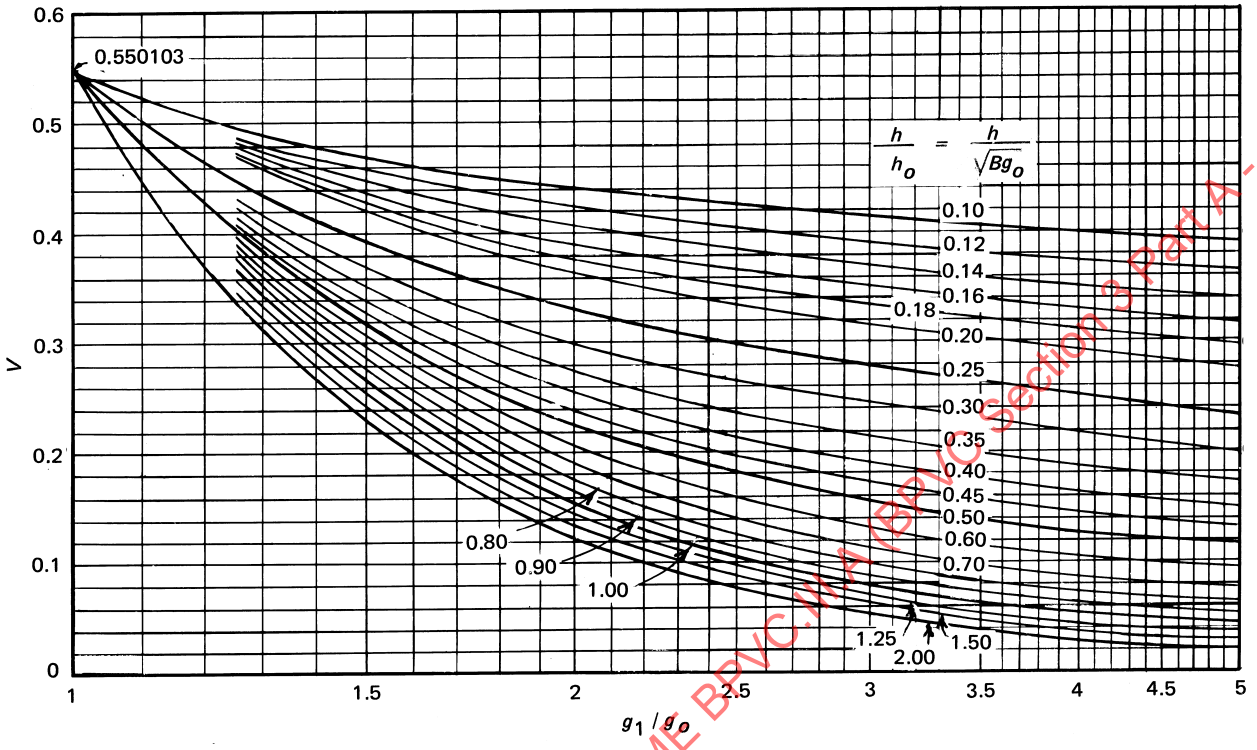
Figure XI-3240-3
Values of *V* (Integral Flange Factors)



GENERAL NOTE: See Table XI-3240-1 for equations.

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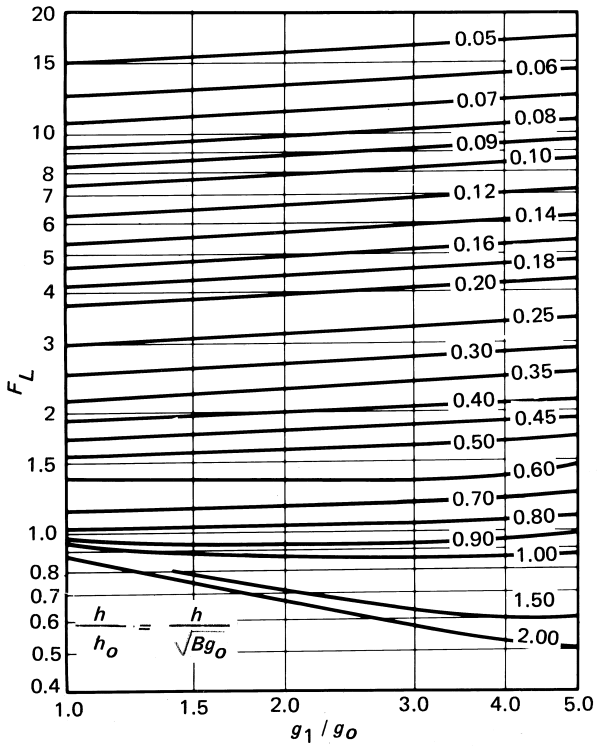
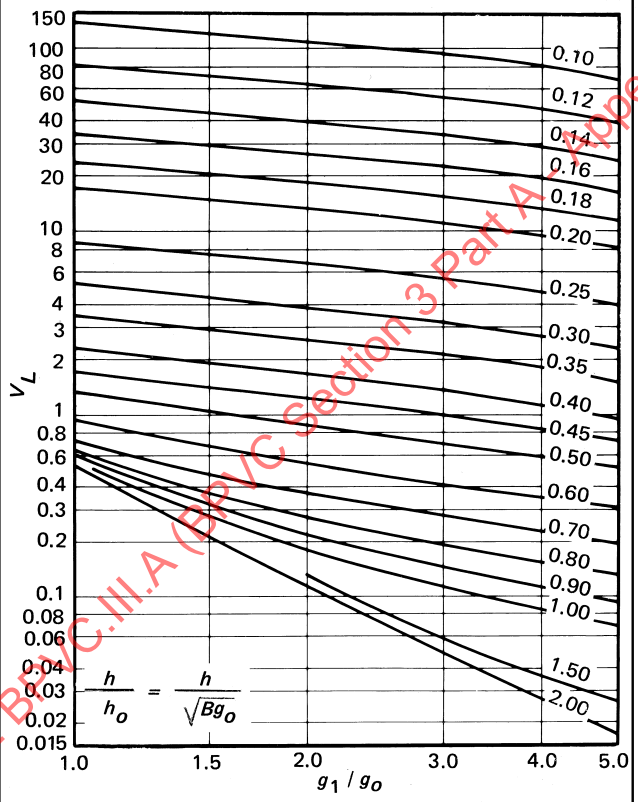
Figure XI-3240-4
Values of F_L (Loose Hub Flange Factors)



GENERAL NOTE: See Table XI-3240-1 for equations.

Figure XI-3240-5
Values of V_L (Loose Hub Flange Factors)



GENERAL NOTE: See Table XI-3240-1 for equations.

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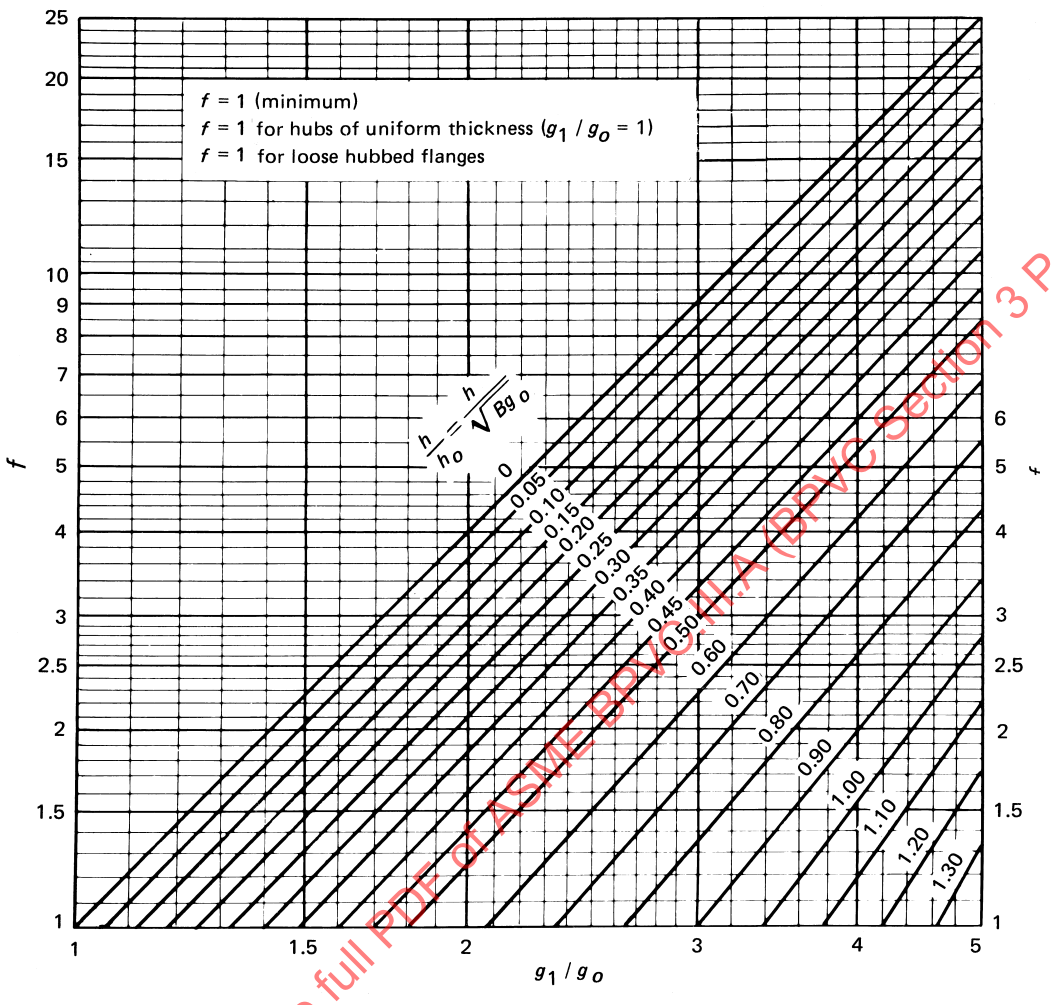
Figure XI-3240-6
Values of f (Hub Stress Correction Factor)



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MANDATORY APPENDIX XII

ARTICLE XII-1000
DESIGN CONSIDERATIONS FOR BOLTED FLANGE CONNECTIONS

XII-1100 CONSIDERATIONS

(a) The primary purpose of the rules for bolted flange connections in Mandatory Appendix XI is to ensure safety, but there are certain other practical matters to be taken into consideration in order to obtain a serviceable design. One of the most important of these is the proportioning of the bolting, i.e., determining the number and size of the bolts.

(b) In the great majority of designs the practice that has been used in the past should be adequate: to follow the design rules in Mandatory Appendix XI and tighten the bolts sufficiently to withstand the test pressure without leakage. The considerations presented in the discussion in (c) through (m) will be important only when some unusual feature exists, such as a very large diameter, a high Design Pressure, a high temperature, severe temperature gradients, an unusual gasket arrangement, and so on.

(c) The maximum allowable stress values for bolting given in Section II, Part D, Subpart 1, Table 3 are design values to be used in determining the minimum amount of bolting required under the rules. However, a distinction must be kept carefully in mind between the design value and the bolt stress that might actually exist or that might be needed for conditions other than the Design Pressure. The initial tightening of the bolts is a prestressing operation, and the amount of bolt stress developed must be within proper limits to ensure, on the one hand, that it is adequate to provide against all conditions that tend to produce a leaking joint and, on the other hand, that it is not so excessive that yielding of the bolts or flanges can produce relaxation that also can result in leakage.

(d) The first important consideration is the need for the joint to be tight in the hydrostatic test. An initial bolt stress of some magnitude greater than the design value therefore must be provided. If it is not, further bolt strain develops during the test, which tends to part the joint and thereby to decompress the gasket enough to allow leakage. The test pressure is usually $1\frac{1}{2}$ times the Design Pressure, and on this basis it may be thought that 50% extra bolt stress above the design value will be sufficient. However, this is an oversimplification because, on the one hand, the safety factor against leakage under test conditions in general need not be as great as under Design Conditions. On the other hand, if a stress–strain analysis of the joint is made, it may indicate that an initial bolt stress still higher than $1\frac{1}{2}$ times the design value is needed. Such an analysis is one that considers the changes in bolt elongation, flange deflection, and gasket load that take place with the application of internal pressure, starting from the prestressed condition. In any event, it is evident that an initial bolt stress higher than the design value may and, in some cases, must be developed in the tightening operation, and it is the intent of Subsections NC, ND, and NE that such a practice is permissible, provided it includes necessary and appropriate provision to ensure against excessive flange distortion and gross crushing of the gasket.

(e) It is possible for the bolt stress to decrease after initial tightening, because of slow creep or relaxation of the gasket, particularly in the case of the softer gasket materials. This may be the cause of leakage in the hydrostatic test, in which case it may suffice merely to retighten the bolts. A decrease in bolt stress can also occur in service at elevated temperatures, as a result of creep in the bolt or flange or gasket material, with consequent relaxation. When this results in leakage under service conditions, it is common practice to retighten the bolts, and sometimes a single such operation or perhaps several repeated at long intervals is sufficient to correct the condition. To avoid chronic difficulties of this nature, however, it is advisable when designing a joint for high temperature service to give attention to the relaxation properties of the materials involved, especially for temperatures where creep is the controlling factor in design.

(f) In the other direction, excessive initial bolt stress can present a problem in the form of yielding in the bolting itself and may occur in the tightening operation to the extent of damage or even breakage. This is especially likely with bolts of small diameter and with bolt materials having a relatively low yield strength. The yield strength of mild carbon steel, annealed austenitic stainless steel, and certain of the nonferrous bolting materials can easily be exceeded with ordinary wrench effort in the smaller bolt sizes. Even if no damage is evident, any additional load generated when internal pressure is applied can

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produce further yielding with possible leakage. Such yielding can also occur when there is very little margin between initial bolt stress and yield strength.

(g) An increase in bolt stress, above any that may be due to internal pressure, might occur in service during startup or other transient conditions, or perhaps even under normal service. This can happen when there is an appreciable differential in temperature between the flanges and the bolts or when the bolt material has a different coefficient of thermal expansion than the flange material. Any increase in bolt load due to this thermal effect, superposed on the load already existing, can cause yielding of the bolt material, whereas any pronounced decrease due to such effects can result in such a loss of bolt load as to be a direct cause of leakage. In either case, retightening of the bolts may be necessary, but it must not be forgotten that the effects of repeated retightening can be cumulative and may ultimately make the joint unserviceable.

(h) In addition to the difficulties created by yielding of the bolts as described above, the possibility of similar difficulties arising from yielding of the flange or gasket material, under like circumstances or from other causes, should also be considered.

(i) Excessive bolt stress, whatever the reason, may cause the flange to yield even though the bolts may not yield. Any resulting excessive deflection of the flange, accompanied by permanent set, can produce a leaking joint when other effects are superposed. It can also damage the flange by making it more difficult to effect a tight joint thereafter. For example, irregular permanent distortion of the flange due to uneven bolt load around the circumference of the joint can warp the flange face and its gasket contact surface out of a true plane.

(j) The gasket, too, can be overloaded, even without excessive bolt stress. The full initial bolt load is imposed entirely on the gasket, unless the gasket has a stop ring or the flange face detail is arranged to provide the equivalent. Without such means of controlling the compression of the gasket, consideration must be given to the selection of gasket type, size, and material that will prevent gross crushing of the gasket.

(k) From the foregoing, it is apparent that the bolt stress can vary over a considerable range above the design stress value. The design stress values for bolting have been set at a conservative value to provide a factor against yielding. At elevated temperatures, the design stress values are governed by the creep rate and stress rupture strength. Any higher bolt stress existing before creep occurs in operation will have already served its purpose of seating the gasket and holding the hydrostatic test pressure, all at atmospheric temperature, and is not needed at the Design Pressure and Temperature.

(l) Theoretically, the margin against flange yielding is not as great. The design values for flange materials may be as high as five-eighths or two-thirds of the yield strength. However, the highest stress in a flange is usually the bending stress in the hub or shell and is more or less localized. It is too conservative to assume that local yielding is followed immediately by overall yielding of the entire flange. Even if a plastic hinge should develop, the ring portion of the flange takes up the portion of the load that the hub and shell refuse to carry. Yielding is far more significant if it occurs first in the ring but the limitation in the rules on the combined hub and ring stresses provides a safeguard. In this connection, reference should be made to Notes G10 and G8 of Section II, Part D, Subpart 1, Tables 1A and 1B, respectively, which provides guidance in the case of high alloy materials to which a strain limiting factor may have to be applied.

(m) Another very important item in bolting design is the question of whether the necessary bolt stress is actually realized and what special means of tightening, if any, must be employed. Most joints are tightened manually by ordinary wrenching and it is advantageous to have designs that require no more than this. Some pitfalls must be avoided, however. The probable bolt stress developed manually, when using standard wrenches, is

(U.S. Customary Units)

$$S = 45,000 / \sqrt{d}$$

(SI Units)

$$S = 1600 / \sqrt{d}$$

where S is the bolt stress (psi, MPa) and d is the nominal diameter of the bolt (in., mm). It can be seen that smaller bolts will have excessive stress unless judgment is exercised in pulling up on them. On the other hand, it will be impossible to develop the desired stress in very large bolts by ordinary hand wrenching. Impact wrenches may prove serviceable, but, if not, resort may be had to such methods as preheating the bolt or using hydraulically powered bolt tensioners. With some of these methods, control of the bolt stress is possible by means inherent in the procedure, especially if effective thread lubricants are employed, but in all cases the bolt stress can be regulated within reasonable tolerances by measuring the bolt elongation with suitable extensometer equipment. Ordinarily, simple wrenching without verification of the actual bolt stress meets all practical needs, and measured control of the stress is employed only when there is some special or important reason for doing so.

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MANDATORY APPENDIX XIII

ARTICLE XIII-1000
DESIGN BASED ON STRESS ANALYSIS

XIII-1100 GENERAL REQUIREMENTS

XIII-1110 SCOPE

This Appendix is applicable only to the design of Class 2 vessels meeting the requirements of NC-3200 and containments meeting the requirements of WC-3200.

(a) When a fatigue evaluation is not required (NC-3219 or WC-3219), a vessel or a vessel part may be designed using the stress analysis methods given in this Appendix. When a fatigue evaluation is required, this Appendix shall be used together with Mandatory Appendix XIV, Design Based on Fatigue Analysis for vessels designed in accordance with NC-3200 or WC-3200.

(b) Use of the analytical methods contained in the Articles of this Appendix following this Article is an acceptable way of obtaining solutions to problems for which these procedures are applicable. Other techniques, analytical or experimental (Mandatory Appendix II), may be used provided they are either more accurate or more conservative than those contained herein.

XIII-1120 DESIGN ACCEPTABILITY

XIII-1121 Requirements for Design Acceptability

The requirements for the acceptability of a design are given in (a) and (b).

(a) The design shall be such that the stresses shall not exceed the limits described in XIII-1140.

(b) For configurations where compressive stresses occur, in addition to the requirement in (a), the critical buckling stress shall be taken into account. For the case of external pressure, see NC-3240 or WC-3133.

XIII-1122 Basis for Determining Stresses

The equivalent stress at any point in a vessel is the value of stress deduced from the stress condition at the point by means of a theory of failure for comparison with the mechanical properties of the material obtained in tests under uniaxial load. The theory of failure, used in the rules of this Appendix for combining stresses, is the maximum shear stress theory. The maximum shear stress at a point is equal to one-half the difference between the algebraically largest and the algebraically smallest of the three principal stresses at the point.

XIII-1123 Terms Relating to Stress Analysis

Terms used in this Appendix relating to stress analysis are defined in (a) through (f).

(a) Stress Intensity.[7] Stress intensity is defined as twice the maximum shear stress, which is the difference between the algebraically largest principal stress and the algebraically smallest principal stress at a given point. Tensile stresses are considered positive, and compressive stresses are considered negative.

(b) Gross Structural Discontinuity. Gross structural discontinuity is a geometric or material discontinuity that affects the stress or strain distribution throughout the entire wall thickness of the pressure-retaining member. Gross discontinuity-type stresses are those portions of the actual stress distributions that produce net bending and membrane force resultants when integrated throughout the wall thickness. Examples of a gross structural discontinuity are head-to-shell junctions, flange-to-shell junctions, nozzles, and junctions between shells of different diameters or thicknesses.

(c) Local Structural Discontinuity. Local structural discontinuity is a geometric or material discontinuity that affects the stress or strain distribution throughout a fractional part of the wall thickness. The stress distribution associated with a local discontinuity causes only very localized deformation or strain and has no significant effect on the shell-type discontinuity deformations. Examples are small fillet radii, small attachments, and partial penetration welds.

(d) Normal Stress. Normal stress is the component of stress normal to the plane of reference. This is also referred to as direct stress. Usually the distribution of normal stress is not uniform through the thickness of a part, so this stress is considered to have two components, one uniformly distributed and equal to the average stress across the thickness under consideration and the other varying from this average value across the thickness.

(e) Shear Stress. Shear stress is the component of stress tangent to the plane of reference.

(f) Membrane Stress. Membrane stress is the component of normal stress that is uniformly distributed and equal to the average stress across the thickness of the section under consideration.

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(g) Bending Stress. Bending stress is the component of normal stress that varies across the thickness. The variation may or may not be linear.

(h) Primary Stress. Primary stress is any normal stress or shear stress developed by an imposed loading that is necessary to satisfy the laws of equilibrium of external and internal forces and moments. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses that considerably exceed the yield strength will result in failure or, at least, in gross distortion. Primary membrane stress is divided into general and local categories. A general primary membrane stress is one that is so distributed in the structure that no redistribution of load occurs as a result of yielding. Examples of primary stress are

(1) general membrane stress in a circular cylindrical shell or a spherical shell due to internal pressure or to distributed loads

(2) bending stress in the central portion of a flat head due to pressure

Refer to Table XIII-1130-1 for examples of primary stress.

(i) Secondary Stress. Secondary stress is a normal stress or a shear stress developed by the constraint of adjacent material or by self-constraint of the structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions that cause the stress to occur and failure from one application of the stress is not to be expected. Examples of secondary stress are

(1) general thermal stress [see (m)]

(2) bending stress at a gross structural discontinuity

Refer to Table XIII-1130-1 for examples of secondary stress.

(j) Local Primary Membrane Stress. Cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a discontinuity would, if not limited, produce excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress.

A stressed region may be considered local if the distance over which the membrane stress intensity exceeds $1.1S_m$ does not extend in the meridional direction more than $1.0\sqrt{Rt}$, where R is the minimum midsurface radius of curvature and t is the minimum thickness in the region considered. Regions of local primary stress intensity involving axisymmetric membrane stress distributions which exceed $1.1S_m$ shall not be closer in the meridional direction than $2.5\sqrt{R_L t_L}$, where R_L is defined as $(R_1 + R_2)/2$ and t_L is defined as $(t_1 + t_2)/2$ where t_1 and t_2 are the minimum thicknesses at each of the regions considered, and R_1 and R_2 are the minimum midsurface radii of curvature at these regions where the membrane stress intensity exceeds $1.1S_m$. Discrete regions of local primary membrane stress intensity, such as those resulting from concentrated loads acting on brackets, where the membrane stress intensity exceeds $1.1S_m$ shall be spaced so that there is no overlapping of the areas in which the membrane stress intensity exceeds $1.1S_m$.

(k) Peak Stress. Peak stress is that increment of stress that is additive to the primary plus secondary stresses by reason of local discontinuities or local thermal stress [see (m)(2)] including the effects, if any, of stress concentrations. The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or a brittle fracture. A stress that is not highly localized falls into this category if it is of a type that cannot cause noticeable distortion. Examples of peak stress are

(1) the thermal stress in the austenitic steel cladding of a carbon steel part

(2) certain thermal stresses that may cause fatigue but not distortion

(3) the stress at a local structural discontinuity

(4) surface stresses produced by thermal shock

(l) Load-Controlled Stress. Load-controlled stress is the stress resulting from the application of a loading, such as internal pressure, inertial loads, or gravity, whose magnitude is not reduced as a result of displacement.

(m) Thermal Stress. Thermal stress is a self-balancing stress produced by a nonuniform distribution of temperature or by differing thermal coefficients of expansion. Thermal stress is developed in a solid body whenever a volume of material is prevented from assuming the size and shape that it normally would under a change in temperature. For the purpose of establishing allowable stresses, two types of thermal stress are recognized, depending on the volume or area in which distortion takes place, as described in (1) and (2) below.

(1) General thermal stress is associated with distortion of the structure in which it occurs. If a stress of this type, neglecting stress concentrations, exceeds twice the yield strength of the material, the elastic analysis may be invalid and successive thermal cycles may produce incremental distortion. Therefore, this type is classified as secondary stress in Table XIII-1130-1 and Figure XIII-1141-1. Examples of general thermal stress are

(-a) stress produced by an axial temperature distribution in a cylindrical shell

(-b) stress produced by the temperature difference between a nozzle and the shell to which it is attached

(-c) the equivalent linear stress[8] produced by the radial temperature distribution in a cylindrical shell

(2) Local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses shall be considered only from the fatigue standpoint and are therefore classified as peak stresses in Table

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XIII-1130-1 and Figure XIII-1141-1. In evaluating local thermal stresses the procedures of XIII-1151.1(b) shall be used. Examples of local thermal stress are

(-a) the stress in a small hot spot in a vessel wall

(-b) the difference between the actual stress and the equivalent linear stress resulting from a radial temperature distribution in a cylindrical shell

(-c) the thermal stress in a cladding material that has a coefficient of expansion different from that of the base metal

(n) Total Stress. Total stress is the sum of the primary, secondary, and peak stress contributions. Recognition of each of the individual contributions is essential to establishment of appropriate limitations.

(o) Service Cycle. Service cycle is defined as the initiation and establishment of new conditions followed by a return to the conditions that prevailed at the beginning of the cycle. The types of service conditions that may occur are further defined in the Design Specification as required by NCA-2142 and WA-2123.

(p) Stress Cycle. Stress cycle is a condition in which the alternating stress difference (XIV-1212) goes from an initial value through an algebraic maximum value and an algebraic minimum value and then returns to the initial value. A single service cycle may result in one or more stress cycles. Dynamic effects shall also be considered as stress cycles.

(q) Fatigue Strength Reduction Factor. Fatigue strength reduction factor is a stress intensification factor which accounts for the effect of a local structural discontinuity (stress concentration) on the fatigue strength. Values for some specific cases, based on experiment, are given elsewhere and in Subsection NC or Subsection WC. In the absence of experimental data, the theoretical stress concentration factor may be used.

(r) Shakedown. Shakedown of a structure occurs if, after a few cycles of load application, ratcheting ceases. The subsequent structural response is elastic, or elastic-plastic, and progressive incremental inelastic deformation is absent. Elastic shakedown is the case in which the subsequent response is elastic.

(s) Deformation. Deformation of a component part is an alteration of its shape or size.

(t) Inelasticity. Inelasticity is a general characteristic of material behavior in which the material does not return to its original shape and size after removal of all applied loads. Plasticity and creep are special cases of inelasticity.

(u) Creep. Creep is the special case of inelasticity that relates to the stress induced time-dependent deformation under load. Small time-dependent deformations may occur after the removal of all applied loads.

(v) Plasticity. Plasticity is the special case of inelasticity in which the material undergoes time-independent nonrecoverable deformation.

(w) Plastic Analysis. Plastic analysis is that method which computes the structural behavior under given loads considering the plasticity characteristics of the materials including strain hardening and the stress redistribution occurring in the structure.

(x) Plastic Analysis — Collapse Load. A plastic analysis may be used to determine the collapse load for a given combination of loads on a given structure. The following criterion for determination of the collapse load shall be used. A load-deflection or load-strain curve is plotted with load as the ordinate and deflection or strain as the abscissa. The angle that the linear part of the load-deflection or load-strain curve makes with the ordinate is called θ. A second straight line, hereafter called the collapse limit line, is drawn through the origin so that it makes an angle $\phi = \tan^{-1}(2 \tan \theta)$ with the ordinate. The collapse load is the load at the intersection of the load-deflection or load-strain curve and the collapse limit line. If this method is used, particular care should be given to ensure that the strains or deflections that are used are indicative of the load carrying capacity of the structure.

(y) Plastic Instability Load. The plastic instability load for members under predominantly tensile or compressive loading is defined as that load at which unbounded plastic deformation can occur without an increase in load. At the plastic tensile instability load the true stress in the material increases faster than strain hardening can accommodate.

(z) Limit Analysis. Limit analysis is a special case of plastic analysis in which the material is assumed to be ideally plastic (nonstrain-hardening). In limit analysis the equilibrium and flow characteristics at the limit state are used to calculate the collapse load. The two bounding methods which are used in limit analysis are the lower bound approach, which is associated with a statically admissible stress field, and the upper bound approach, which is associated with a kinematically admissible velocity field. For beams and frames the term *mechanism* is commonly used in lieu of *kinematically admissible velocity field*.

(aa) Collapse Load — Lower Bound. If, for a given load, any system of stresses can be found which everywhere satisfies equilibrium, and nowhere exceeds the material yield strength, the load is at or below the collapse load. This is the lower bound theorem of limit analysis which permits calculations of a lower bound to the collapse load.

(ab) Limit Analysis — Collapse Load. The methods of limit analysis are used to compute the maximum load that a structure assumed to be made of ideally plastic material can carry. At this load, which is termed the collapse load, the deformations of the structure increase without bound.

(ac) Plastic Hinge. A plastic hinge is an idealized concept used in Limit Analysis. In a beam or a frame, a plastic hinge is formed at the point where the moment, shear, and axial force lie on the yield interaction surface. In plates and shells a plastic hinge is formed where the generalized stresses lie on the yield surface.

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(ad) Strain Limiting Load. When a limit is placed upon a strain, the load associated with the strain limit is called the strain limiting load.

(ae) Test Collapse Load. Test collapse load is the collapse load determined by tests according to the criteria given in II-1430 of Section III.

(af) Ratcheting. Ratcheting is a progressive incremental inelastic deformation or strain which can occur in a component that is subjected to variations of mechanical stress, thermal stress, or both.

XIII-1130 DERIVATION OF STRESS INTENSITIES

One requirement for the acceptability of a design [XIII-1121(a)] is that the calculated stress intensities shall not exceed specified allowable limits. These limits differ depending on the stress category (primary, secondary, etc.) from which the stress intensity is derived. This paragraph describes the procedure for the calculation of the stress intensities which are subject to specified limits. The steps in the procedure are given in (a) through (f).

(a) At the point on the vessel which is being investigated, choose an orthogonal set of coordinates, such as tangential, longitudinal, and radial, and designate them by the subscripts t, l, and r. The stress components in these directions are then designated σ_t, σ_r, and σ_l for direct stresses and τ_{tl}, τ_{lr}, and τ_{rt} for shear stresses.

(b) Calculate the stress components for each type of loading to which the part will be subjected and assign each set of stress values to one or a group of the following categories:[9]

(1) general primary membrane stress P_m [XIII-1123(f) and XIII-1123(h)]

(2) local primary membrane stress P_L [XIII-1123(j)]

(3) primary bending stress P_b [XIII-1123(g) and XIII-1123(h)]

(4) secondary stress[10] Q [XIII-1123(i)]

(5) peak stress F [XIII-1123(k)]

(c) Group the stress components in accordance with (b). Figure XIII-1141-1 is to provide assistance in assigning the stress values to the appropriate category. At any rectangular box calculate the algebraic sum of the values of σ_t which result from the different types of loadings and which have entered the box, and similarly for the other five stress components. The result is a set of six stress components in each box.

(d) Translate the stress components in the t, l, and r directions into principal stresses σ_1, σ_2, and σ_3. (In many pressure vessel calculations, the t, l, and r directions may be so chosen that the shearing stress components are zero and so σ_1, σ_2, and σ_3 are identical to σ_t, σ_l, and σ_r.)

(e) Calculate the stress differences S_{12}, S_{23}, and S_{31} from the relations

$$S_{12} = \sigma_1 - \sigma_2$$
$$S_{23} = \sigma_2 - \sigma_3$$
$$S_{31} = \sigma_3 - \sigma_1$$

The stress intensity S is the largest absolute value of S_{12}, S_{23}, and S_{31}.

(f) The stress intensity calculated as in the preceding (e) from the stress components in any rectangle in Figure XIII-1141-1 shall not exceed the allowable values of XIII-1140 which are shown in the circle adjacent to the rectangle in Figure XIII-1141-1.

NOTE: Membrane stress intensity is derived from the stress components averaged across the thickness of the section. The averaging shall be performed at the component level, in step (b) or (c) above.

XIII-1140 BASIC STRESS INTENSITY LIMITS

XIII-1141 Five Basic Limits

The five basic stress intensity limits which are to be satisfied are given in XIII-1142 through XIII-1146 (Figure XIII-1141-1).

XIII-1142 General Primary Membrane Stress Intensity

This stress intensity is derived from the average value across the thickness of a section of the general primary stresses [XIII-1123(h)] produced by

(a) Design Pressure and specified Design Mechanical Loads for Design Loadings

(b) coincident pressure and mechanical loads associated with each Service Loading as specified in the Design Specification, but excluding all secondary and peak stresses.

Averaging is to be applied to the stress components prior to the determination of the stress intensity values. The allowable value of this stress intensity is kS_m, with S_m as given in Section II, Part D, Subpart 1, Tables 2A and 2B. Values of k, reflecting the specific Design or Service Limit under consideration, are given in Table NC-3217-1 or Table WC-3217-1 as appropriate.

XIII-1143 Local Membrane Stress Intensity

This stress intensity is derived from the average value across the thickness of a section of the local primary stresses [XIII-1123(j)] produced by

(a) Design Pressure and specified Design Mechanical Loads for Design Loadings

(b) coincident pressure and mechanical loads associated with each Service Loading as specified in the Design Specification, but excluding all thermal and peak stresses.

Averaging is to be applied to the stress components prior to the determination of the stress intensity values. The allowable value of this stress intensity is $1.5kS_m$, with S_m as given in Section II, Part D, Subpart 1, Tables

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Table XIII-1130-1
Classification of Stress Intensity in Vessels for Some Typical Cases

(15)

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Cylindrical or spherical shell	Shell plate remote from discontinuities	Internal pressure	General membrane	P_m
			Gradient through plate thickness	Q
		Axial thermal gradient	Membrane	Q
			Bending	Q
	Junction with head or flange	Internal pressure	Membrane	P_L
			Bending	Q [Note (1)]
Any shell or head	Any section across entire vessel	External load or moment, or internal pressure	General membrane averaged across full section	P_m
		External load or moment	Bending across full section	P_m
	Near nozzle or other opening	External load or moment, or internal pressure	Local membrane	P_L
			Bending	Q
			Peak (fillet or corner)	F
	Any location	Temperature difference between shell and head	Membrane	Q
			Bending	Q
Dished head or conical head	Crown	Internal pressure	Membrane	P_m
			Bending	P_b
	Knuckle or junction to shell	Internal pressure	Membrane	P_L [Note (2)]
			Bending	Q
Flat head	Center region	Internal pressure	Membrane	P_m
			Bending	P_b
	Junction to shell	Internal pressure	Membrane	P_L
			Bending	P_b or Q [Note (1)]
Perforated head or shell	Typical ligament in a uniform pattern	Pressure	Membrane (averaged through cross section)	P_m
			Bending (averaged through width of ligament, but gradient through plate)	P_b
			Peak	F
	Isolated or atypical ligament	Pressure	Membrane	Q
			Bending	F
			Peak	F

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Table XIII-1130-1
Classification of Stress Intensity in Vessels for Some Typical Cases (Cont'd)

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Nozzle (XIII-1170)	Within the limits of reinforcement defined by NC-3334 or WC-3230	Pressure and external loads and moments, including those attributable to restrained free end displacements of the attached piping	General membrane	P_m
			Bending (other than gross structural discontinuity stresses) averaged through nozzle thickness	P_m
	Outside the limits of reinforcement defined by NC-3334 or WC-3230	Pressure and external axial, shear, and torsional loads other than those attributable to restrained free end displacements of the attached piping	General membrane stresses	P_m
		Pressure and external loads and moments other than those attributable to restrained free end displacements of the attached piping	Membrane	P_L
			Bending	P_b
		Pressure and all external loads and moments	Membrane	P_L
			Bending	Q
			Peak	F
	Nozzle wall	Gross structural discontinuities	Local membrane	P_L
			Bending	Q
			Peak	F
		Differential expansion	Membrane	Q
			Bending	Q
			Peak	F
Cladding	Any	Differential expansion	Membrane	F
			Bending	F
Any	Any	Radial temperature distribution [Note (3)]	Equivalent linear stress [Note (4)]	Q
			Nonlinear portion of stress distribution	F
Any	Any	Any	Stress concentration (notch effect)	F

NOTES:
(1) If the bending moment at the edge is required to maintain the bending stress in the middle to acceptable limits, the edge bending is classified as P_b. Otherwise, it is classified as Q.
(2) Consideration shall also be given to the possibility of wrinkling and excessive deformation in vessels with a large diameter–thickness ratio.
(3) Consider possibility of thermal stress ratchet.
(4) Equivalent linear stress is defined as the linear stress distribution that has the same net bending moment as the actual stress distribution.

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Figure XIII-1141-1
Stress Categories and Limits of Stress Intensity

Stress Category	Primary			Secondary Membrane Plus Bending	Peak
	General Membrane	Local Membrane	Bending		
Description (For examples, see Table XIII-1130-1)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced by pressure and mechanical loads.	Average stress across any solid sections. Considers discontinuities but not concentrations. Produced by pressure and mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced by pressure and mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by pressure, mechanical loads, or by differential thermal expansion. Excludes local stress concentrations.	(1) Increment added to primary or secondary stress by a concentration (notch). (2) Certain thermal stresses which may cause fatigue but not distortion of vessel shape.
Symbol [Note (1)]	P_m	P_L	P_b	**Q [Note (2)]**	F
Combination of stress components and allowable limits of stress intensities	P_m — kS_m P_L — $1.5kS_m$ $P_L + P_b$ — $1.5kS_m$ $P_L + P_b + Q$ — $3S_m$ (Note (3)) $P_L + P_b + Q + F$ — S_a (Note (4))				

Legend:
- Allowable Value
- Calculated Value
- Use Design and Service Loads as appropriate.
- Use Service Loads only.

NOTES:
(1) The symbols P_m, P_L, P_b, Q, and F do not represent single quantities but rather sets of six quantities representing the six stress components σ_t, σ_l, σ_r, τ_{lt}, τ_{lr}, and τ_{rt}.
(2) The stresses in Category Q are those parts of the total stress which are produced by thermal gradients, structural discontinuities, etc., and do not include primary stresses which may also exist at the same point. It should be noted, however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly and, when appropriate, this calculated value represents the total of P_m (or P_L) + P_b + Q and not Q alone. Similarly, if the stress in Category F is produced by a stress concentration, the quantity F is the additional stress, produced by the notch, over and above the nominal stress. For example, if a plate has a nominal stress intensity S and has a notch with a stress concentration factor K, then $P_m = S$, $P_b = 0$, $Q = 0$, $F = P_m\ (K - 1)$, and the total stress intensity equals $P_m + P_m\ (K - 1) = KP_m$.
(3) This limitation applies to the range of stress intensity. When the secondary stress is due to a temperature excursion at the point at which the stresses are being analyzed, the value of S_m shall be taken as the average of the S_m values tabulated in Section II, Part D, Subpart 1, Tables 2A and 2B for the highest and the lowest temperature of the metal during the transient. When part or all of the secondary stress is due to mechanical load, the value of S_m shall be taken as the S_m value for the highest temperature of the metal during the transient.
(4) S_a is obtained from the fatigue curves (Mandatory Appendix I). The allowable stress intensity for the full range of fluctuation is 2 S_a.

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2A and 2B. Values of k, reflecting the specific Design or Service Limit under consideration, are given in Table NC-3217-1 or Table WC-3217-1 as appropriate.

XIII-1144 Primary Membrane Plus Primary Bending Stress Intensity

This stress intensity is derived from the highest value across the thickness of a section of the general or local primary membrane stresses plus primary bending stresses produced by

(a) Design Pressure and specified Design Mechanical Loads for Design Loadings

(b) coincident pressure and mechanical loads associated with each Service Loading as specified in the Design Specification, but excluding all secondary and peak stresses.

For solid rectangular sections, the allowable value of this stress intensity is $1.5kS_m$, with S_m as given in Section II, Part D, Subpart 1, Tables 2A and 2B. Values of k, reflecting the appropriate Design or Service Limit under consideration, are given in Table NC-3217-1 or Table WC-3217-1 as appropriate. For other than solid rectangular sections, a value of α times the limit established in NC-3217(b) or WC-3217(b) may be used, where the factor α is defined as the ratio of the load set producing a fully plastic section to the load set producing initial yielding in the extreme fibers of the section. In the evaluation of the initial yield and fully plastic section capacities, the ratios of each individual load in the respective load set to each other load in that load set shall be the same as the respective ratios of the individual loads in the specified design load set. The value of α shall not exceed the value calculated for bending only (P_m = 0). In no case shall the value of α exceed 1.5. The propensity for buckling of the part of the section that is in compression shall be investigated. The α factor is not permitted for Level D Service Limits when inelastic component analysis is used as permitted in Nonmandatory Appendix F.

XIII-1145 Primary Plus Secondary Stress Intensity

This is the stress intensity, derived from the highest value at any point across the thickness of a section, of the general or local primary membrane stresses plus primary bending stresses plus secondary stresses produced by specified service pressure and other specified mechanical loads and by general thermal effects. The effects of gross structural discontinuities but not of local structural discontinuities (stress concentrations) shall be included. The allowable value of this stress intensity is $3S_m$ [Note (3) of Figure XIII-1141-1].

XIII-1146 Peak Stress Intensity

This is the stress intensity, derived from the highest value at any point across the thickness of a section, of the combination of all primary, secondary, and peak stresses produced by specified service pressures and other mechanical loads and by general and local thermal effects and including the effects of gross and local structural discontinuities. The allowable value of this stress intensity is dependent on the range of the stress difference from which it is derived and on the number of times it is to be applied. The stress intensity is to be compared with the allowable value obtained by the methods of analysis for cyclic operation described in XIV-1200 through the use of the fatigue curves.

XIII-1150 PLASTIC ANALYSIS, LIMIT ANALYSIS, AND SHAKEDOWN ANALYSIS

XIII-1151 Plastic Analysis

(a) Certain of the allowable stresses permitted in these design criteria are such that the maximum stress calculated on an elastic basis may exceed the yield strength of the material. The limit on primary plus secondary stress intensity of $3S_m$ (XIII-1145) has been placed at a level which ensures shakedown to elastic action after a few repetitions of the stress cycle except in regions containing significant local structural discontinuities or local thermal stresses. These last two factors are considered only in the performance of a fatigue evaluation.

(b) The limits on local membrane stress intensity (XIII-1143) and primary membrane plus primary bending stress intensity of $1.5kS_m$ (XIII-1144) have been placed at a level which conservatively ensures the prevention of collapse as determined by the principles of limit analysis. The following subparagraphs provide guidance in the application of plastic analysis and some relaxation of the basic stress limits which are allowed if plastic analysis is used.

XIII-1151.1 Value of Poisson's Ratio.

(a) In evaluating stresses for comparison with any stress limits other than fatigue allowables, stresses shall be calculated on an elastic basis using the elastic value of Poisson's ratio.

(b) In evaluating stresses for comparison with fatigue allowables, the elastic equations shall be used, except that the numerical value substituted for Poisson's ratio shall be determined from the expression:

$$v = 0.5 - 0.2\left(S_y / S_a\right), \text{but not less than } 0.3$$

where

S_a = alternating stress intensity determined in XIV-1221.3(a) or XIV-1221.3(b) prior to the elastic modulus adjustment in XIV-1221.3(d).

S_y = the yield strength of the material at the mean value of the temperature of the cycle

XIII-1151.2 Plastic Analysis Procedures. The limits on local membrane stress intensity (XIII-1143), primary plus secondary stress intensity (XIII-1145), thermal stress ratchet in shell (XIV-1410), and thermal stress in

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nonintegral connections (XIV-1420) need not be satisfied at a specific location if at that location the procedures of the following (a), (b), and (c) are used.

(a) In evaluating stresses for comparison with the stress limits of XIII-1142, XIII-1144 (general primary membrane plus primary bending stress intensity only), XIII-1180, NC-3216.3(a), XIII-1160, and XIII-1121(b), the stresses are calculated on an elastic basis.

(b) In lieu of satisfying the specific requirements of XIII-1143, XIII-1145, NC-3216.3(b), and XIV-1420 at a specific location, the structural action is calculated on a plastic basis and the design shall be considered to be acceptable if shakedown occurs, as opposed to continuing deformation, and if the deformations which occur prior to shakedown do not exceed specified limits.

(c) In evaluating stresses for comparison with fatigue allowables, the numerically maximum principal total strain range which occurs after shakedown shall be multiplied by one-half of the modulus of elasticity of the material at the mean value of the temperature of the cycle.

XIII-1152 Limit Analysis

The limits on local membrane stress intensity (XIII-1143) and primary membrane plus primary bending stress intensity need not be satisfied at a specific location if it can be shown by means of limit analysis or by tests that the specified mechanical and thermal loadings do not exceed two-thirds the lower bound collapse load.

XIII-1153 Shakedown Analysis

The $3S_m$ stress intensity limit on the range of primary plus secondary stress intensity (XIII-1145) may be exceeded provided that the range of primary plus secondary membrane plus bending stress intensity, excluding thermal bending stresses, shall be $\leq 3S_m$ and that the following requirements of (a) through (e) are met.

(a) The value of S_a used for entering the design fatigue curve is multiplied by the factor K_e, where

K_e = 1.0, for $S_n \leq 3S_m$
= $1.0 + [(1 - n)/n(m - 1)][(S_n)/(3S_m) - 1]$, for $3S_m < S_n < 3mS_m$
= $1/n$, for $S_n \geq 3mS_m$

S_n = range of primary plus secondary stress intensity

The values of the material parameters m and n are given for the various classes of permitted materials in Table XIII-1153(a)-1.

(b) The rest of the fatigue evaluation stays the same as required in Mandatory Appendix XIV, except that the procedure of XIII-1151.1 need not be used.

(c) The component meets the thermal ratcheting requirement of Appendix XIV-1400.

(d) The temperature does not exceed those listed in Table XIII-1153(a)-1 for the various classes of permitted materials.

Table XIII-1153(a)-1
Values of m, n, and T_{max} for Various Classes of Permitted Materials

Materials	m	n	T_{max}, °F (°C)
Low alloy steel	2.0	0.2	700 (370)
Martensitic stainless steel	2.0	0.2	700 (370)
Carbon steel	3.0	0.2	700 (370)
Austenitic stainless steel	1.7	0.3	800 (425)
Nickel-chromium-iron	1.7	0.3	800 (425)
Nickel-copper	1.7	0.3	800 (425)

(e) The material shall have a specified minimum yield strength to specified minimum tensile strength ratio of less than 0.80.

XIII-1160 TRIAXIAL STRESSES

The algebraic sum of the three primary principal stresses ($\sigma_1 + \sigma_2 + \sigma_3$) shall not exceed four times the tabulated value of S_m.

XIII-1170 NOZZLE PIPING TRANSITION

The P_m classification of stresses in nozzles resulting from external load or moment is applicable for that length of nozzle which lies within the limits of reinforcement whether or not nozzle reinforcement is required. Beyond the limits of reinforcement, a P_L classification may be applied to the stresses resulting from external load or moment attributable to thermal expansion of the attached piping. However, the pressure and mechanical load induced membrane stresses retain a P_m classification.

XIII-1180 BOLTING

XIII-1181 Bolt and Gasket Requirements

(a) The number and cross-sectional area of bolts required to resist internal pressure shall be determined in accordance with the procedures of Mandatory Appendix XI. The allowable bolt design stresses, as used in the equations of Mandatory Appendix XI, shall be the values given in Section II, Part D, Subpart 1, Table 4 for bolting materials. When sealing is effected by a seal weld instead of a gasket, the gasket factor m and the minimum design seating stress y may be taken as zero.

(b) When gaskets are used for preservice testing only, the design is satisfactory if the above requirements are satisfied for $m = y = 0$ and the requirements of XIII-1182 are satisfied when the appropriate m and y are used for the test gasket.

XIII-1182 Allowable Maximum Service Stresses in Bolts (15)

It is recognized that actual service stresses in bolts, such as those produced by the combination of preload, pressure, and differential thermal expansion, may be

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higher than the values given in Section II, Part D, Subpart 1, Table 4. The maximum of such service stress, averaged across the bolt cross section and neglecting stress concentrations, shall not exceed two-thirds of the yield strength values of Section II, Part D, Subpart 1, Table Y-1. Except as restricted by XIV-1322(b), the maximum value of such service stress at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations shall not exceed the yield strength values of Section II, Part D, Subpart 1, Table Y-1. Stress intensity, rather than maximum stress, shall be limited to this value when the bolts are tightened by methods other than heaters, stretchers, or other means which minimize residual torsion.

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ARTICLE XIII-2000
PRESSURE STRESSES IN OPENINGS FOR FATIGUE EVALUATION

XIII-2100 METHODS OF EVALUATION

XIII-2110 ACCEPTABLE METHODS

For the purpose of detecting peak stresses around the opening, three acceptable methods are listed in (a), (b), and (c) below.

(a) Analytical Method. This method uses suitable analytical techniques, such as finite element computer analyses, which provide detailed stress distributions around openings. In addition to peak stresses due to pressure, the effects of other loadings shall be included. The total peak stress at any given point shall be determined by combining stresses due to pressure, thermal, and external loadings in accordance with the rules of Article XIII-1000.

(b) Experimental Stress Analysis. This is based on data from experiments (XIII-2130).

(c) Stress Index Method. This uses various equations based on data obtained from an extensive series of tests covering a range of variations of applicable dimensional ratios and configurations (para. XIII-2120). This method covers only single, isolated openings. Stress indices may also be determined by theoretical or experimental stress analysis. Such analysis may be included in the Design Report.

XIII-2120 STRESS INDEX METHOD

XIII-2121 Stress Index

The term *stress index*, as used herein, is defined as the numerical ratio of the stress components σ_t, σ_n, and σ_r under consideration to the computed membrane hoop stress in the unpenetrated vessel material. However, the material which increases the thickness of a vessel wall locally at the nozzle shall not be included in the calculations of these stress components. When the thickness of the wall is increased over that required, the values of r_1 and r_2 in Figure XIII-2124(e)-1 shall be referred to the thickened section.

XIII-2122 Nomenclature

The nomenclature for the stress components are shown in Figure XIII-2122-1 and are defined as follows:

R = inside radius, in corroded condition, of cylindrical vessel, spherical vessel, or spherical head
S = the stress intensity at the point under consideration
t = nominal wall thickness less corrosion allowance, of vessel or head
σ_n = the stress component normal to the plane of the section, such as the circumferential stress around the hole in the shell
σ_r = the stress component normal to the boundary of the section
σ_t = the stress component in the plane of the section under consideration and parallel to the boundary of the section

XIII-2123 Stress Indices for Nozzles

When the conditions of XIII-2122 are satisfied, the stress indices given in Tables XIII-2123-1 and XIII-2123-2 may be used for nozzles designed in accordance with the applicable rules of NC-3230 or WC-3230, and NC-3259 or WC-3259. These stress indices deal only with the maximum stresses, at certain general locations, due to internal pressure. In the evaluation of stresses in or adjacent to vessel openings and connections, it is often necessary to consider the effect of stresses due to external loadings or thermal stresses. In such cases, the total stress at a given point may be determined by superposition. In the case of combined stresses due to internal pressure and nozzle loading, the maximum stresses for a given location shall be considered as acting at the same point and added algebraically unless positive evidence is available to the contrary.

Figure XIII-2122-1
Direction of Stress Components

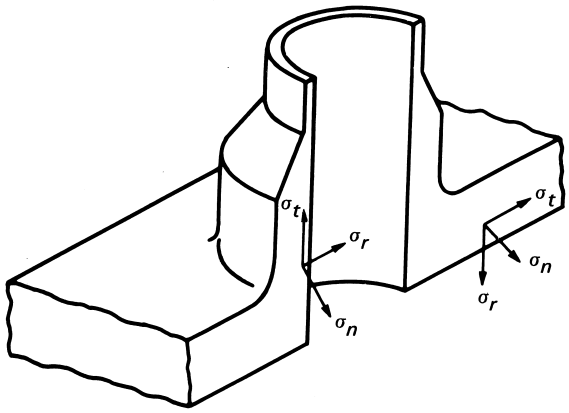

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Table XIII-2123-1
Nozzles in Spherical Shells and Formed Heads

Stress	Inside Corner	Outside Corner
σ_n	2.0	2.0
σ_t	−0.2	2.0
σ_r	$-2t/R$	0
S	2.2	2.0

(15) **XIII-2124 Limitations of Indices of XIII-2123**

The indices of XIII-2123 apply provided the following conditions set forth in (a) through (g) exist.

(a) The opening is for a circular nozzle whose axis is normal to the vessel wall. If the axis of the nozzle makes an angle ϕ with the normal to the vessel wall, an estimate of the σ_n index on the inside may be obtained from one of the following equations, provided $d/D \le 0.15$.

For hillside connections in spheres or cylinders,

$$K_2 = K_1\left(1 + 2\sin^2\phi\right)$$

For lateral connections in cylinders,

$$K_2 = K_1\left[1 + \left(\tan\phi\right)^{4/3}\right]$$

where

K_1 = the σ_n inside stress index of XIII-2123 for a radial connection
K_2 = the estimated σ_n inside stress index for the nonradial connection

(b) The arc distance measured between the centerlines of adjacent nozzles along the inside surface of the shell is not less than three times the sum of their inside radii for openings in a head or along the longitudinal axis of a shell, and is not less than two times the sum of their radii for opening along the circumference of a cylindrical shell. See NC-3231.1(a)(3) when two nozzles are not on a longitudinal or circumferential line.

Table XIII-2123-2
Nozzles in Cylindrical Shells

	Longitudinal Plane		Transverse Plane	
Stress	Inside	Outside	Inside	Outside
σ_n	3.1	1.2	1.0	2.1
σ_t	−0.2	1.0	−0.2	2.6
σ_r	$-t/R$	0	$-t/R$	0
S	3.3	1.2	1.2	2.6

(c) The following dimensional ratios are met:

Ratio	Cylinder	Sphere
D/t	10 to 100	10 to 100
d/D	0.5 max.	0.5 max.
$d/\sqrt{Dt}$	...	0.8 max.
$d/\sqrt{Dtr_2/t}$	1.5 max.	...

where D is the inside shell diameter, t is the shell thickness, and d is the inside nozzle diameter. In the case of cylindrical shells, the total nozzle reinforcement area on the transverse axis of the connections, including any outside of the reinforcement limits, shall not exceed 200% of that required for the longitudinal axis compared to 50% permitted by Figure NC-3332.2-1 or WC-3232.2-1 unless a tapered transition section is incorporated into the reinforcement and the shell, meeting the requirements of NC-3261 or WC-3261.

(d) In the case of spherical shells and formed heads, at least 40% of the total nozzle reinforcement area shall be located beyond the outside surface of the minimum required vessel wall thickness.

(e) The inside corner radius r_1 [Figure XIII-2124(e)-1] is between 10% and 100% of the shell thickness t.

(f) The outer corner radius r_2 [Figure XIII-2124(e)-1] is large enough to provide a smooth transition between the nozzles and the shell. In addition, for opening diameters in cylindrical shells and 2:1 ellipsoidal heads greater than $1\frac{1}{2}$ shell thicknesses and in spherical shells greater than 3 shell thicknesses, the value of r_2 shall be not less than one-half the thickness of the shell or nozzle wall, whichever is greater.

(g) The radius r_3 [Figure XIII-2124(e)-1] is not less than the greater of (1) or (2) below

(1) $0.002\theta d_o$, where d_o is the outside diameter of the nozzle and is as shown in Figure XIII-2124(e)-1 and the angle θ is expressed in degrees

(2) $2(\sin\theta)^3$ times offset for the configuration shown in Figure XIII-2124(e)-1 sketches (a) and (b)

XIII-2130 EXPERIMENTAL STRESS ANALYSIS METHOD

XIII-2131 Determination of Stress Intensities

The stress intensities for opening configurations which do not meet the requirements of NC-3230, WC-3230, or of XIII-2124 shall be determined in accordance with methods of Mandatory Appendix II.

XIII-2132 Evaluation of Data for Similar but Slightly Different Configurations

In accordance with II-1112, reevaluation is not required for configurations for which there are available detailed experimental results that are consistent with the requirements of Mandatory Appendix II. In order that available experimental data may be interpreted as providing information pertinent to the analysis of slightly

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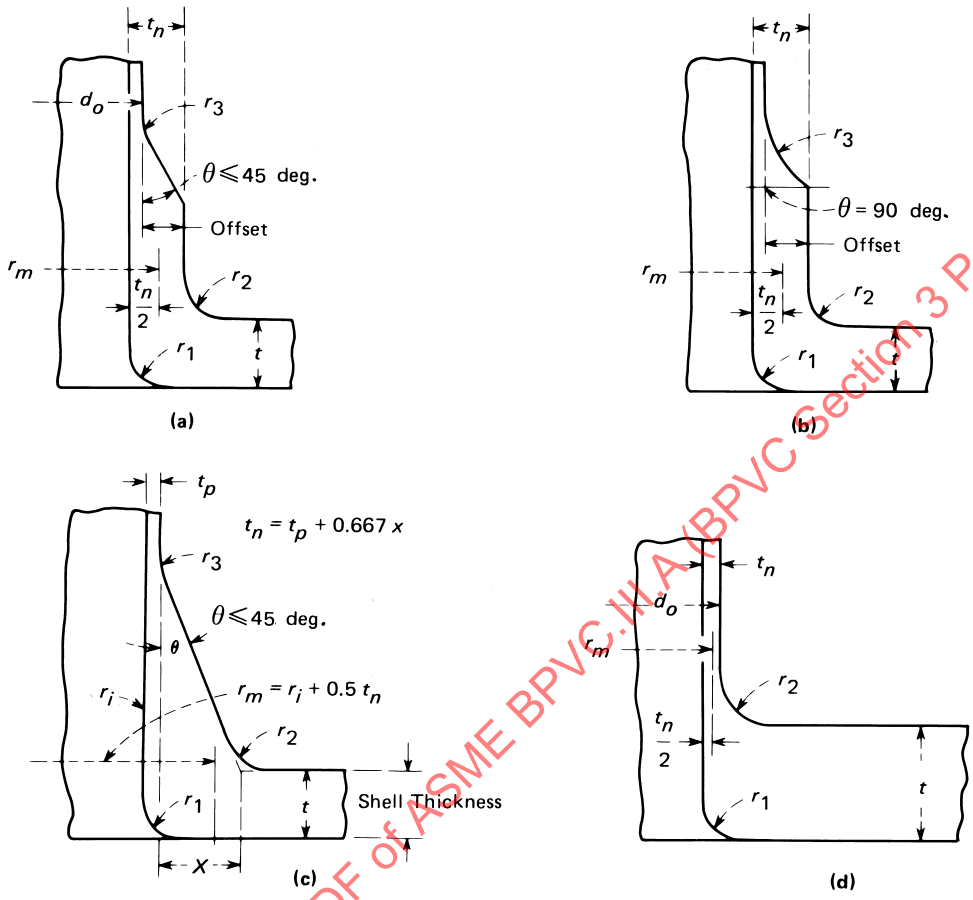
Figure XIII-2124(e)-1
Nozzle Nomenclature and Dimensions



different configurations, thereby possibly minimizing the need for additional investigations, the following guidelines of (a) through (d) are presented:

(a) For an unreinforced opening or for an opening where the reinforcement is provided primarily by a uniform increase in vessel wall thickness, the stresses around the opening will increase with increasing d/D ratio of the opening (diameter of nozzle or opening to diameter of shell). Therefore, experimental data for a small d/D ratio cannot be safely applied to a larger d/D ratio but can be applied to a smaller d/D ratio provided the experiments were made at a d/D ratio <0.5.

(b) For an unreinforced opening or for an opening where the reinforcement is provided primarily by a uniform increase in vessel wall thickness, the stresses around the opening will increase with increasing d/t ratio for thinner shells. Therefore, experimental data for a relatively small d/t ratio cannot be safely applied to a larger d/t ratio but can be applied to a smaller d/t ratio.

(c) Generally speaking, the stress data available in the literature are applicable only to single openings. Such data should be considered valid only for a connection sufficiently removed from another nozzle, opening, flange, or other major discontinuity so that superposition of stresses will not produce an unacceptable value of stress intensity.

(d) Stresses at the outside juncture of a nozzle and shell are greatly influenced by the fillet or transition at the juncture. Generally speaking, stress data available in the literature are for certain specific fillet radii. Other factors being equal, these stress data may be considered valid for fillet radii equal to or greater than used in the test but should not be considered valid for smaller fillet radii or undefined fillets and transitions, such as for a triangular weld fillet, as commonly used.

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MANDATORY APPENDIX XIV

ARTICLE XIV-1000
DESIGN BASED ON FATIGUE ANALYSIS

XIV-1100 INTRODUCTION

XIV-1110 SCOPE

This Appendix is applicable only to the fatigue analysis of Class 2 vessels meeting the requirements of NC-3200 and containments meeting the requirements of WC-3200.

XIV-1200 ANALYSIS FOR CYCLIC SERVICE OF VESSELS

XIV-1210 GENERAL REQUIREMENTS

XIV-1211 Suitability of Vessel for Cyclic Service

The suitability of a vessel component for specified Service Conditions involving cyclic application of loads and thermal conditions shall be determined by the methods described in this Appendix, except that the suitability of high strength bolts shall be determined by the methods of XIV-1300 and the possibility of thermal stress ratchet shall be investigated in accordance with XIV-1400. If the specified service of the vessel meets all of the conditions of NC-3219 or WC-3219, no analysis for cyclic service is required and it may be assumed that the limits on peak stress intensities as governed by fatigue have been satisfied by compliance with the applicable requirements for materials, design, fabrication, examination, and testing of Subsection NC or WC. If the service does not meet all the conditions of NC-3219 or WC-3219, a fatigue analysis shall be made in accordance with XIV-1220 or a fatigue test shall be made in accordance with II-1150.

XIV-1212 Allowable Amplitude of Alternating Stresses

The conditions and procedures of NC-3219 or WC-3219, and XIV-1220 are based on a comparison of peak stresses with strain cycling fatigue data. The strain cycling fatigue data are represented by the design fatigue strength curves of Mandatory Appendix I.[11] These curves show the allowable amplitude S_a of the alternating stress component (one-half of the alternating stress range) plotted against the number of cycles. The stress amplitude is calculated on the assumption of elastic behavior and hence has the dimensions of stress, but it does not represent a real stress when the elastic range is exceeded. The fatigue curves are obtained from uniaxial strain cycling data in which the imposed strains have been multiplied by the elastic modulus and a design margin has been provided, so as to make the calculated stress intensity and amplitude and the allowable stress amplitude directly comparable. The curves have been adjusted, where necessary, to include the maximum effects of mean stress, which is the condition where the stress fluctuates about a mean value which is different from zero. As a consequence of this procedure, it is essential that the requirements of XIII-1145 and XIII-1160 be satisfied at all times with transient stresses included and that the calculated value of the alternating stress intensity be proportional to the actual strain amplitude. To evaluate the effect of alternating stresses of varying amplitudes, a linear damage relation is assumed in XIV-1220.

XIV-1213 Loadings to Be Considered

The loadings to be considered shall include those loads that are due to testing of the vessel when such testing is in addition to that required by NC-6100 or Article WC-6000.

XIV-1220 CYCLIC LOADING

XIV-1221 Design for Cyclic Loading

XIV-1221.1 Determination of Vessel's Ability to Withstand Cyclic Loading. If the specific service of the vessel does not meet the conditions of NC-3219 or WC-3219, the ability of the vessel to withstand the specified cyclic service without fatigue failure shall be determined as provided herein. The determination shall be made on the basis of the stresses at a point of the vessel and the allowable stress cycles shall be adequate for the specified service at every point. Only the stresses due to the specified service cycle need be considered; stresses produced by any load or thermal condition which does not vary during the cycle need not be considered, since they are mean stresses and the maximum possible effect of mean stress is included in the fatigue design curves.

XIV-1221.2 Significance of Compliance With Requirements for Cyclic Loading. Compliance with these requirements means only that the vessel is suitable from

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the standpoint of possible fatigue failure; complete suitability for the specified service is also dependent on meeting the general stress limits of XIII-1140 and any applicable special stress limits of XIV-1410.

(15) **XIV-1221.3 Cyclic Loading Design Procedure.**

(a) When Principal Stress Direction Does Not Change. For any case in which the directions of the principal stresses at the point being considered do not change during the cycle, steps (1), (2), and (3) shall be followed to determine the alternating stress intensity.

(1) Principal Stresses. Consider the values of the three principal stresses at the point versus time for the complete stress cycle, taking into account both the gross and local structural discontinuities and the thermal effects which vary during the cycle. These are designated as σ_1, σ_2, and σ_3 for later identification.

(2) Stress Differences. Determine the stress differences $S_{12} = \sigma_1 - \sigma_2$, $S_{23} = \sigma_2 - \sigma_3$, and $S_{31} = \sigma_3 - \sigma_1$ versus time for the complete cycle. In what follows, the symbol S_{ij} is used to represent any one of these three stress differences.

(3) Alternating Stress Intensity. Determine the extremes of the range through which each stress difference S_{ij} fluctuates, and find the absolute magnitude of this range for each S_{ij}. Call this magnitude S_{rij} and let $S_{\text{alt } ij} = 0.5S_{rij}$. The alternating stress intensity S_{alt} is the largest of the $S_{\text{alt } ij}$ values.

(b) When Principal Stress Direction Changes. For any case in which the directions of the principal stresses at the point being considered change during the stress cycle, it is necessary to use steps (1) through (5).

(1) Consider the values of the six stress components σ_t, σ_l, σ_r, τ_{lt}, τ_{lr}, and τ_{rt} versus time for the complete stress cycle, taking into account both the gross and local structural discontinuities and the thermal effects which vary during the cycle.

(2) Choose a point in time when the conditions are one of the extremes for the cycle, either maximum or minimum algebraically, and identify the stress components at this time by the subscript i. In most cases it will be possible to choose at least one time during the cycle when the conditions are known to be extreme. In some cases it may be necessary to try different points in time to find the one which results in the largest value of alternating stress intensity.

(3) Subtract each of the six stress components σ_{ti}, σ_{li}, etc., from the corresponding stress components σ_t, σ_l, etc., at each point in time during the cycle and call the resulting components σ'_t, σ'_l, etc.

(4) At each point in time during the cycle, calculate the principal stresses σ'_1, σ'_2, and σ'_3 derived from the six stress components σ'_t, σ'_l, etc. Note that the directions of the principal stresses may change during the cycle but each principal stress retains its identity as it rotates.

(5) Determine the stress differences $S'_{12} = \sigma'_1 - \sigma'_2$, $S'_{23} = \sigma'_2 - \sigma'_3$, and $S'_{31} = \sigma'_3 - \sigma'_1$ versus time for the complete cycle and find the largest absolute magnitude of any stress difference at any time. The alternating stress intensity S_{alt} is one-half of this magnitude.

(c) Design Fatigue Curves. Mandatory Appendix I contains the applicable fatigue design curves for some of the materials permitted by Subsection NC or Subsection WC. When more than one curve is presented for a given material, the applicability of each curve to materials of various strength levels is identified. Linear interpolation may be used for intermediate strength levels of these materials. As used herein, the strength level is the specified minimum room temperature value.

(d) Use of Design Fatigue Curve. Multiply S_{alt}, as determined in (a) or (b), by the ratio of the modulus of elasticity given on the design fatigue curve to the value used in the analysis. Enter the applicable design fatigue curve at this value on the ordinate axis and find the corresponding number of cycles on the abscissa. If the operational cycle being considered is the only one which produces significant fluctuating stresses, this is the allowable number of cycles.

(e) Cumulative Damage. If there are two or more types of stress cycles which produce significant stresses, their cumulative effect shall be evaluated as given in (1) through (6).

(1) Designate the specified number of times each type of stress cycle of types 1, 2, 3, ..., n will be repeated during the life of the vessel as n_1, n_2, n_3, ..., n_n, respectively. In determining n_1, n_2, n_3, ..., n_n, consideration shall be given to the superposition of cycles of various origins which produce a total stress difference range greater than the stress difference ranges of the individual cycles. For example, if one type of stress cycle produces 1000 cycles of a stress difference variation from 0 to +60.0 ksi and another type of stress cycle produces 10,000 cycles of a stress difference variation from 0 to −50.0 ksi, the two types of cycle to be considered are defined by the following parameters.

For type 1 cycle:

$$n_1 = 1000$$

$$S_{\text{alt } 1} = (60{,}000 + 50{,}000)/2 = 55{,}000 \text{ psi}$$

For type 2 cycle:

$$n_2 = 9000$$

$$S_{\text{alt } 2} = (50{,}000 + 0)/2 = 25{,}000 \text{ psi}$$

(2) For each type of stress cycle determine the alternating stress intensity S_{alt} by the procedures of the preceding (a) or (b). Call these quantities $S_{\text{alt } 1}$, $S_{\text{alt } 2}$, $S_{\text{alt } 3}$, ..., $S_{\text{alt } n}$.

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(3) For each value $S_{alt\ 1}$, $S_{alt\ 2}$, $S_{alt\ 3}$, ..., $S_{alt\ n}$, use the applicable design fatigue curve to determine the maximum number of repetitions which would be allowable if this type of cycle were the only one acting. Call these values N_1, N_2, N_3, ..., N_n.

(4) For each type of stress cycle, calculate the usage factors U_1, U_2, U_3, ..., U_n from $U_1 = n_1/N_1$, $U_2 = n_2/N_2$, $U_3 = n_3/N_3$, ..., $U_n = n_n/N_n$.

(5) Calculate the cumulative usage factor U from $U = U_1 + U_2 + U_3 + ... + U_n$.

(6) The cumulative usage factor U shall not exceed 1.0.

XIV-1222 Local Structural Discontinuities

These effects shall be evaluated for all conditions using stress concentration factors determined from theoretical, experimental, or photoelastic studies, or numerical stress analysis techniques. Experimentally determined fatigue strength reduction factors may be used when determined in accordance with the procedures of II-1600, except for high strength alloy steel bolting for which the requirements of XIV-1322 shall apply when using the design fatigue curve of Figure I-9.4. Except for the case of crack-like defects, no fatigue strength factor greater than five need be used.

XIV-1223 Fillet Welds

Fillet welds shall not be used in vessels for joints of Category A, B, C, or D (Figure NC-3351-1 or Figure WC-3251-1) except as permitted for joints of Category C for slip on flanges (NC-3252.3 and NC-3262.2) and for the joints of Category D as permitted in NC-3259 or WC-3259. Fillet welds may be used for attachment to pressure vessels using one-half the stress limits of XIII-1142 through XIII-1145 for primary and secondary stresses. Evaluation for cyclic loading shall be made in accordance with Mandatory Appendix XIV using a fatigue strength reduction factor of four and shall include consideration of temperature differences between the vessel and the attachment and expansion or contraction of the vessel produced by internal or external pressure.

XIV-1300 ANALYSIS FOR CYCLIC SERVICE OF BOLTS

XIV-1310 GENERAL REQUIREMENTS

XIV-1311 Suitability of Bolts for Cyclic Service

Unless the vessel on which they are installed meets all the conditions of NC-3219 or WC-3219 and thus requires no fatigue analysis, the suitability of bolts for cyclic service shall be determined in accordance with the procedures of XIV-1320.

XIV-1320 EVALUATION OF BOLTS DEPENDENT ON THEIR MATERIAL PROPERTIES

XIV-1321 Bolts With Tensile Strengths Less Than 100 ksi (689 MPa)

Bolts made of materials which have minimum specified tensile strengths of less than 100.0 ksi (689 MPa) shall be evaluated for cyclic operation by the methods of XIV-1220, using the applicable design fatigue curve, Figure I-9.1 or Figure I-9.2 and an appropriate stress concentration factor (XIV-1324).

XIV-1322 High Strength Bolts (15)

High strength alloy steel bolts and studs may be evaluated for cyclic operation by the methods of XIV-1220, using the design fatigue curve of Figure I-9.4, provided the following requirements of (a) through (e) are met:

(a) The material is one of the following: SA-193 Grade B-7 or B-16; SA-320 Grade L-43; SA-540 Grades B-23 and B-24, heat treated in accordance with SA-540, Section 5.

(b) The maximum value of the service stress at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations shall not exceed $0.9S_y$ if the higher of the two fatigue design curves given in Figure I-9.4 is used. The $^2/_3S_y$ limit for direct tension is unchanged.

(c) Threads shall be of a Vee type, having a minimum thread root radius no smaller than 0.003 in. (0.076 mm).

(d) Fillet radii at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060.

(e) The fatigue strength reduction factor used in the fatigue evaluation shall not be less than 4.0.

XIV-1323 Acceptability for Cyclic Service

The bolts shall be acceptable for the specific cyclic application of loads and thermal stresses provided the cumulative usage factor U, as determined in XIV-1221.3(e), does not exceed 1.0.

XIV-1324 Fatigue Strength Reduction Factor for Threads

Unless it can be shown by analysis or test that a lower value is appropriate, the fatigue strength reduction factor used in the fatigue evaluation of threaded members shall not be less than 4.0.

XIV-1400 ANALYSIS FOR THERMAL STRESS RATCHET

XIV-1410 THERMAL STRESS RATCHET IN SHELL

Under certain combinations of steady state and cyclic loadings there is a possibility of large distortions developing as the result of ratchet action, that is, the deformation

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increases by a nearly equal amount for each cycle. Examples of this phenomenon are treated in XIV-1410 and in XIV-1420.

(a) The limiting value of the maximum cyclic thermal stress permitted in a shell loaded by steady state internal pressure in order to prevent cyclic growth in diameter is as follows. Let

y' = maximum allowable thermal stress, computed on an elastic basis, divided by the yield strength[12] S_y

x = maximum general membrane stress due to pressure divided by the yield strength[12] S_y

(1) Case 1: Linear variation of temperature through the shell wall

$$y' = 1/x \text{ for } 0 < x < 0.5$$

$$y' = 4(1 - x) \text{ for } 0.5 < x < 1.0$$

(2) Case 2: Parabolic constantly increasing or constantly decreasing variation of temperature through the shell wall

$$y' = 5.2(1 - x) \text{ for } 0.615 < x < 1.0$$

and approximately, for $x < 0.615$, $y' = 4.65$, 3.55, and 2.70 for $x = 0.3$, 0.4, and 0.5, respectively.

(b) Use of the yield strength S_y in the above relations instead of the proportional limit allows a small amount of growth during each cycle until strain hardening raises the proportional limit to S_y. If the yield strength of the material is higher than 2 times the S_a value for the maximum number of cycles on the applicable fatigue curve of Mandatory Appendix I for the material, the latter value shall be used, if there are to be a large number of cycles, because strain softening may occur.

XIV-1420 PROGRESSIVE DISTORTION OF NONINTEGRAL CONNECTIONS

Screwed on caps, screwed in plugs, shear ring closures, and breech lock closures are examples of nonintegral connections which are subject to failure by bell mouthing or other types of progressive deformation. If any combination of applied loads produces yielding, such joints are subject to ratcheting because the mating members may become loose at the end of each complete operating cycle and start the next cycle in a new relationship with each other, with or without manual manipulation. Additional distortion may occur in each cycle so that interlocking parts, such as threads, can eventually lose engagement. Therefore primary plus secondary stress intensities (XIII-1145) which result in slippage between the parts of a nonintegral connection in which disengagement could occur as a result of progressive distortion shall be limited to the allowable stress limits given in XIII-1142 and XIII-1143.

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MANDATORY APPENDIX XVIII

ARTICLE XVIII-1000
CAPACITY CONVERSIONS FOR PRESSURE RELIEF VALVES

XVIII-1100 PROCEDURE FOR CONVERSION

XVIII-1110 EQUATIONS FOR CONVERSION (COMPRESSIBLE FLUIDS)

The capacity of a relief valve in terms of a gas or vapor other than the medium for which the valve was officially rated shall be determined by application of the following equations:[13]

For dry saturated steam:

For pressures up to 1,500 psig (10.3 MPa)

$$W_s = C_N K A P \quad (1)$$

where

C_N = 51.5 (5.25)

For pressures over 1,500 psig (10.3 MPa) and up to 3,200 psig (22 MPa), the value of W_s, calculated by the above equation, shall be multiplied by the following factor, F_N:

(U.S. Customary Units)

$$F_N = \frac{0.1906P - 1{,}000}{0.2232P - 1{,}061}$$

(SI Units)

$$F_N = \frac{27.6P - 1{,}000}{33.2P - 1{,}061}$$

For superheated steam:

For superheated steam the value of W_s shall be multiplied by the appropriate superheat correction factor, K_{sh}, of Table XVIII-1110-1 (or Table XVIII-1110-1M for SI calculations).

For wet saturated steam:

For wet saturated steam with a quality (dryness fraction) of 0.90 or greater, the value of W_s shall be corrected by dividing by the quality of the steam used for testing.

For air:

$$W_a = CKAP\sqrt{M/T} \quad (2)$$

where

C = 356 (27.03)
M = 28.97 mol. wt.
T = 520 (288) when W_a is the rated capacity

For any gas or vapor (other than steam):

$$W = CKAP\sqrt{M/T} \quad (3)$$

where

A = actual discharge area of the pressure relief valve, in.^{2} (mm^2)
C = constant for gas or vapor which is a function of the ratio of specific heats, $k = c_p/c_v$ (Figure XVIII-1110-1)
K = coefficient of discharge
M = molecular weight
P = (set pressure + overpressure) plus atmospheric pressure, psia (MPa$_{abs}$)
T = absolute temperature at inlet (°F plus 460) (K)
W = flow of any gas or vapor, lb/hr (kg/h)
W_a = rated capacity,[14] converted to lb/hr (kg/h) of air at 60°F (15°C) inlet temperature
W_s = rated capacity, lb/hr (kg/h) of steam
Z = ratio of deviation of the actual gas from a perfect gas, a ratio evaluated at inlet conditions

These equations shall also be used when the required flow of any gas or vapor is known and it is necessary to compute the rated capacity of steam or air.

(a) Molecular weight of some of the common gases and vapors are given in Table XVIII-1110(a)-1.

(b) In the case of hydrocarbons, the compressibility factor Z shall be included in the equation for gases and vapors as follows:

$$W = CKAP\sqrt{M/ZT} \quad (4)$$

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Table XVIII-1110-1
Superheat Correction Factor, K_{sh}

Flowing Pressure (psia)	Superheat Correction Factor, K_{sh}, Total Temperature, °F, of Superheated Steam																
	400	450	500	550	600	650	700	750	800	850	900	950	1000	1050	1100	1150	1200
50	0.987	0.957	0.930	0.905	0.882	0.861	0.841	0.823	0.805	0.789	0.774	0.759	0.745	0.732	0.719	0.708	0.696
100	0.998	0.963	0.935	0.909	0.885	0.864	0.843	0.825	0.807	0.790	0.775	0.760	0.746	0.733	0.720	0.708	0.697
150	0.984	0.970	0.940	0.913	0.888	0.866	0.846	0.826	0.808	0.792	0.776	0.761	0.747	0.733	0.721	0.709	0.697
200	0.979	0.977	0.945	0.917	0.892	0.869	0.848	0.828	0.810	0.793	0.777	0.762	0.748	0.734	0.721	0.709	0.698
250	...	0.972	0.951	0.921	0.895	0.871	0.850	0.830	0.812	0.794	0.778	0.763	0.749	0.735	0.722	0.710	0.698
300	...	0.968	0.957	0.926	0.898	0.874	0.852	0.832	0.813	0.796	0.780	0.764	0.750	0.736	0.723	0.710	0.699
350	...	0.968	0.963	0.930	0.902	0.877	0.854	0.834	0.815	0.797	0.781	0.765	0.750	0.736	0.723	0.711	0.699
400	...	...	0.963	0.935	0.906	0.880	0.857	0.836	0.816	0.798	0.782	0.766	0.751	0.737	0.724	0.712	0.700
450	...	...	0.961	0.940	0.909	0.883	0.859	0.838	0.818	0.800	0.783	0.767	0.752	0.738	0.725	0.712	0.700
500	...	...	0.961	0.946	0.914	0.886	0.862	0.840	0.820	0.801	0.784	0.768	0.753	0.739	0.725	0.713	0.701
550	...	...	0.962	0.952	0.918	0.889	0.864	0.842	0.822	0.803	0.785	0.769	0.754	0.740	0.726	0.713	0.701
600	...	...	0.964	0.958	0.922	0.892	0.867	0.844	0.823	0.804	0.787	0.770	0.755	0.740	0.727	0.714	0.702
650	...	...	0.968	0.958	0.927	0.896	0.869	0.846	0.825	0.806	0.788	0.771	0.756	0.741	0.728	0.715	0.702
700	...	...	...	0.958	0.931	0.899	0.872	0.848	0.827	0.807	0.789	0.772	0.757	0.742	0.728	0.715	0.703
750	...	...	...	0.958	0.936	0.903	0.875	0.850	0.828	0.809	0.790	0.774	0.758	0.743	0.729	0.716	0.703
800	...	...	...	0.960	0.942	0.906	0.878	0.852	0.830	0.810	0.792	0.774	0.759	0.744	0.730	0.716	0.704
850	...	...	...	0.962	0.947	0.910	0.880	0.855	0.832	0.812	0.793	0.776	0.760	0.744	0.730	0.717	0.704
900	...	...	...	0.965	0.953	0.914	0.883	0.857	0.834	0.813	0.794	0.777	0.760	0.745	0.731	0.718	0.705
950	...	...	...	0.969	0.958	0.918	0.886	0.860	0.836	0.815	0.796	0.778	0.761	0.746	0.732	0.718	0.705
1,000	...	...	...	0.974	0.959	0.923	0.890	0.862	0.838	0.816	0.797	0.779	0.762	0.747	0.732	0.719	0.706
1,050	...	...	...	...	0.960	0.927	0.893	0.864	0.840	0.818	0.798	0.780	0.763	0.748	0.733	0.719	0.707
1,100	...	...	...	...	0.962	0.931	0.896	0.867	0.842	0.820	0.800	0.781	0.764	0.749	0.734	0.720	0.707
1,150	...	...	...	...	0.964	0.936	0.899	0.870	0.844	0.821	0.801	0.782	0.765	0.749	0.735	0.721	0.708
1,200	...	...	...	...	0.966	0.941	0.903	0.872	0.846	0.823	0.802	0.784	0.766	0.750	0.735	0.721	0.708
1,250	...	...	...	...	0.969	0.946	0.906	0.875	0.848	0.825	0.804	0.785	0.767	0.751	0.736	0.722	0.709
1,300	...	...	...	...	0.973	0.952	0.910	0.878	0.850	0.826	0.805	0.786	0.768	0.752	0.737	0.723	0.709
1,350	...	...	...	...	0.977	0.958	0.914	0.880	0.852	0.828	0.807	0.787	0.769	0.753	0.737	0.723	0.710
1,400	...	...	...	...	0.982	0.963	0.918	0.883	0.854	0.830	0.808	0.788	0.770	0.754	0.738	0.724	0.710
1,450	...	...	...	...	0.987	0.968	0.922	0.886	0.857	0.832	0.809	0.790	0.771	0.754	0.739	0.724	0.711
1,500	...	...	...	...	0.993	0.970	0.926	0.889	0.859	0.833	0.811	0.791	0.772	0.755	0.740	0.725	0.711
1,550	...	...	...	...	...	0.972	0.930	0.892	0.861	0.835	0.812	0.792	0.773	0.756	0.740	0.726	0.712
1,600	...	...	...	...	...	0.973	0.934	0.894	0.863	0.836	0.813	0.792	0.774	0.756	0.740	0.726	0.712
1,650	...	...	...	...	...	0.973	0.936	0.895	0.863	0.836	0.812	0.791	0.772	0.755	0.739	0.724	0.710
1,700	...	...	...	...	...	0.973	0.938	0.895	0.863	0.835	0.811	0.790	0.771	0.754	0.738	0.723	0.709
1,750	...	...	...	...	...	0.974	0.940	0.896	0.862	0.835	0.810	0.789	0.770	0.752	0.736	0.721	0.707
1,800	...	...	...	...	...	0.975	0.942	0.897	0.862	0.834	0.810	0.788	0.768	0.751	0.735	0.720	0.705
1,850	...	...	...	...	...	0.976	0.944	0.897	0.862	0.833	0.809	0.787	0.767	0.749	0.733	0.718	0.704
1,900	...	...	...	...	...	0.977	0.946	0.898	0.862	0.832	0.807	0.785	0.766	0.748	0.731	0.716	0.702
1,950	...	...	...	...	...	0.979	0.949	0.898	0.861	0.832	0.806	0.784	0.764	0.746	0.729	0.714	0.700
2,000	...	...	...	...	...	0.982	0.952	0.899	0.861	0.831	0.805	0.782	0.762	0.744	0.728	0.712	0.698
2,050	...	...	...	...	...	0.985	0.954	0.899	0.860	0.830	0.804	0.781	0.761	0.742	0.726	0.710	0.696
2,100	...	...	...	...	...	0.988	0.956	0.900	0.860	0.828	0.802	0.779	0.759	0.740	0.724	0.708	0.694
2,150	...	...	...	...	...	...	0.956	0.900	0.859	0.827	0.801	0.778	0.757	0.738	0.722	0.706	0.692
2,200	...	...	...	...	...	...	0.955	0.901	0.859	0.826	0.799	0.776	0.755	0.736	0.720	0.704	0.690
2,250	...	...	...	...	...	...	0.954	0.901	0.858	0.825	0.797	0.774	0.753	0.734	0.717	0.702	0.687
2,300	...	...	...	...	...	...	0.953	0.901	0.857	0.823	0.795	0.772	0.751	0.732	0.715	0.699	0.685
2,350	...	...	...	...	...	...	0.952	0.902	0.856	0.822	0.794	0.769	0.748	0.729	0.712	0.697	0.682
2,400	...	...	...	...	...	...	0.952	0.902	0.855	0.820	0.791	0.767	0.746	0.727	0.710	0.694	0.679
2,450	...	...	...	...	...	...	0.951	0.902	0.854	0.818	0.789	0.765	0.743	0.724	0.707	0.691	0.677
2,500	...	...	...	...	...	...	0.951	0.902	0.852	0.816	0.787	0.762	0.740	0.721	0.704	0.688	0.674
2,550	...	...	...	...	...	...	0.951	0.902	0.851	0.814	0.784	0.759	0.738	0.718	0.701	0.685	0.671
2,600	...	...	...	...	...	...	0.951	0.903	0.849	0.812	0.782	0.756	0.735	0.715	0.698	0.682	0.664
2,650	...	...	...	...	...	...	0.952	0.903	0.848	0.809	0.779	0.754	0.731	0.712	0.695	0.679	0.664
2,700	...	...	...	...	...	...	0.952	0.903	0.846	0.807	0.776	0.750	0.728	0.708	0.691	0.675	0.661
2,750	...	...	...	...	...	...	0.953	0.903	0.844	0.804	0.773	0.747	0.724	0.705	0.687	0.671	0.657
2,800	...	...	...	...	...	...	0.956	0.903	0.842	0.801	0.769	0.743	0.721	0.701	0.684	0.668	0.653

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Table XVIII-1110-1
Superheat Correction Factor, K_{sh} (Cont'd)

Flowing Pressure (psia)	Superheat Correction Factor, K_{sh}, Total Temperature, °F, of Superheated Steam																
	400	450	500	550	600	650	700	750	800	850	900	950	1000	1050	1100	1150	1200
2,850	...	...	...	...	...	...	0.959	0.902	0.839	0.798	0.766	0.739	0.717	0.697	0.679	0.663	0.649
2,900	...	...	...	...	...	...	0.963	0.902	0.836	0.794	0.762	0.735	0.713	0.693	0.675	0.659	0.645
2,950	...	...	...	...	...	...	...	0.902	0.834	0.790	0.758	0.731	0.708	0.688	0.671	0.655	0.640
3,000	...	...	...	...	...	...	...	0.901	0.831	0.786	0.753	0.726	0.704	0.684	0.666	0.650	0.635
3,050	...	...	...	...	...	...	...	0.899	0.827	0.782	0.749	0.722	0.699	0.679	0.661	0.645	0.630
3,100	...	...	...	...	...	...	...	0.896	0.823	0.777	0.744	0.716	0.693	0.673	0.656	0.640	0.625
3,150	...	...	...	...	...	...	...	0.894	0.819	0.772	0.738	0.711	0.688	0.668	0.650	0.634	0.620
3,200	...	...	...	...	...	...	...	0.889	0.815	0.767	0.733	0.705	0.682	0.662	0.644	0.628	0.614

XVIII-1120 EXAMPLES

XVIII-1121 Example 1

Given: A pressure relief valve bears a rated capacity of 3,020 lb/hr of saturated steam for a pressure setting of 200 psi.

Problem: What is the relieving capacity of that valve in terms of air at 100°F for the same pressure setting?

Solution:

(a) For steam:

$$W_s = 51.5KAP$$

$$3,020 = 51.5KAP$$

$$KAP = 3,020 / 51.5 = 58.6$$

(b) For air:

$$W_a = CKAP\sqrt{\frac{M}{T}}$$

$$= 356\ KAP\sqrt{\frac{28.97}{460 + 100}}$$

$$= (356)(58.6)\sqrt{\frac{28.97}{560}}$$

$$= 4,745 \text{ lb} / \text{hr}$$

XVIII-1222 Example 2

Given: It is required to relieve 5,000 lb/hr of propane from a pressure vessel through a pressure relief valve set to relieve at a pressure of P_s, psi, and with an inlet temperature of 125°F.

Problem: What total capacity in lb/hr of steam in pressure relief valves must be furnished?

Solution:

(a) For propane:

$$W = CKAP\sqrt{\frac{M}{T}}$$

Value of C is not definitely known. Use the conservative value, C = 315:

$$5,000 = 315\ KAP\sqrt{\frac{44.09}{460 + 125}}$$

$$KAP = 57.7$$

(b) For steam:

$$W_s = 51.5\ KAP$$

$$= (51.5)(57.7)$$

$$= 2,970 \text{ lb} / \text{hr set to relieve at } P_s, \text{ psi}$$

XVIII-1123 Example 3

Given: It is required to relieve 1,000 lb/hr of ammonia from a pressure vessel at 150°F.

Problem: What is the required total capacity in lb/hr of steam at the same pressure setting?

Solution:

(a) For ammonia:

$$W = CKAP\sqrt{M / T}$$

Manufacturer and Owner agree to use k = 1.33. From Figure XVIII-1110-1, C = 350:

$$1,000 = 350\ KAP\sqrt{\frac{17.03}{460 + 150}}$$

$$KAP = 17.10$$

(b) For steam:

$$W_s = 51.5\ KAP$$

$$= (51.5)(17.10)$$

$$= 880 \text{ lb} / \text{hr}$$

XVIII-1124 Example 4

Given: A safety valve having a certified rating of 8,000 lb/hr of steam for a pressure setting of 2,000 psi.

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Table XVIII-1110-1M
Superheat Correction Factor, K_{sh}

Flowing Pressure (MPa)	Superheat Correction Factor, K_{sh}, Total Temperature,°C, of Superheated Steam																	
	205	225	250	275	300	325	350	375	400	425	450	475	500	525	550	575	600	625
0.50	0.991	0.968	0.942	0.919	0.896	0.876	0.857	0.839	0.823	0.807	0.792	0.778	0.765	0.752	0.74	0.728	0.717	0.706
0.75	0.995	0.972	0.946	0.922	0.899	0.878	0.859	0.841	0.824	0.808	0.793	0.779	0.766	0.753	0.74	0.729	0.717	0.707
1.00	0.985	0.973	0.95	0.925	0.902	0.88	0.861	0.843	0.825	0.809	0.794	0.78	0.766	0.753	0.741	0.729	0.718	0.707
1.25	0.981	0.976	0.954	0.928	0.905	0.883	0.863	0.844	0.827	0.81	0.795	0.781	0.767	0.754	0.741	0.729	0.718	0.707
1.50	...	...	0.957	0.932	0.907	0.885	0.865	0.846	0.828	0.812	0.796	0.782	0.768	0.755	0.742	0.73	0.718	0.708
1.75	...	...	0.959	0.935	0.91	0.887	0.866	0.847	0.829	0.813	0.797	0.782	0.769	0.756	0.743	0.731	0.719	0.708
2.00	...	...	0.96	0.939	0.913	0.889	0.868	0.849	0.831	0.814	0.798	0.784	0.769	0.756	0.744	0.731	0.72	0.708
2.25	...	...	0.963	0.943	0.916	0.892	0.87	0.85	0.832	0.815	0.799	0.785	0.77	0.757	0.744	0.732	0.72	0.709
2.50	...	...	...	0.946	0.919	0.894	0.872	0.852	0.834	0.816	0.8	0.785	0.771	0.757	0.744	0.732	0.72	0.71
2.75	...	...	...	0.948	0.922	0.897	0.874	0.854	0.835	0.817	0.801	0.786	0.772	0.758	0.745	0.733	0.721	0.71
3.00	...	...	...	0.949	0.925	0.899	0.876	0.855	0.837	0.819	0.802	0.787	0.772	0.759	0.746	0.733	0.722	0.71
3.25	...	...	...	0.951	0.929	0.902	0.879	0.857	0.838	0.82	0.803	0.788	0.773	0.759	0.746	0.734	0.722	0.711
3.50	...	...	...	0.953	0.933	0.905	0.881	0.859	0.84	0.822	0.804	0.789	0.774	0.76	0.747	0.734	0.722	0.711
3.75	...	...	...	0.956	0.936	0.908	0.883	0.861	0.841	0.823	0.806	0.79	0.775	0.761	0.748	0.735	0.723	0.711
4.00	...	...	...	0.959	0.94	0.91	0.885	0.863	0.842	0.824	0.807	0.791	0.776	0.762	0.748	0.735	0.723	0.712
4.25	...	...	...	0.961	0.943	0.913	0.887	0.864	0.844	0.825	0.808	0.792	0.776	0.762	0.749	0.736	0.724	0.713
4.50	...	...	...	...	0.944	0.917	0.89	0.866	0.845	0.826	0.809	0.793	0.777	0.763	0.749	0.737	0.725	0.713
4.75	...	...	...	...	0.946	0.919	0.892	0.868	0.847	0.828	0.81	0.793	0.778	0.764	0.75	0.737	0.725	0.713
5.00	...	...	...	...	0.947	0.922	0.894	0.87	0.848	0.829	0.811	0.794	0.779	0.765	0.751	0.738	0.725	0.714
5.25	...	...	...	...	0.949	0.926	0.897	0.872	0.85	0.83	0.812	0.795	0.78	0.765	0.752	0.738	0.726	0.714
5.50	...	...	...	...	0.952	0.93	0.899	0.874	0.851	0.831	0.813	0.797	0.78	0.766	0.752	0.739	0.727	0.714
5.75	...	...	...	...	0.954	0.933	0.902	0.876	0.853	0.833	0.815	0.798	0.782	0.767	0.753	0.739	0.727	0.715
6.00	...	...	...	...	0.957	0.937	0.904	0.878	0.855	0.834	0.816	0.798	0.783	0.768	0.753	0.74	0.727	0.716
6.25	...	...	...	...	0.96	0.94	0.907	0.88	0.856	0.836	0.817	0.799	0.783	0.768	0.754	0.74	0.728	0.716
6.50	...	...	...	...	0.964	0.944	0.91	0.882	0.859	0.837	0.818	0.801	0.784	0.769	0.754	0.741	0.729	0.716
6.75	...	...	...	...	0.966	0.946	0.913	0.885	0.86	0.839	0.819	0.802	0.785	0.769	0.755	0.742	0.729	0.717
7.00	...	...	...	...	...	0.947	0.916	0.887	0.862	0.84	0.82	0.802	0.786	0.77	0.756	0.742	0.729	0.717
7.25	...	...	...	...	...	0.949	0.919	0.889	0.863	0.842	0.822	0.803	0.787	0.771	0.756	0.743	0.73	0.717
7.50	...	...	...	...	...	0.951	0.922	0.891	0.865	0.843	0.823	0.805	0.788	0.772	0.757	0.744	0.73	0.718
7.75	...	...	...	...	...	0.953	0.925	0.893	0.867	0.844	0.824	0.806	0.788	0.772	0.758	0.744	0.731	0.719
8.00	...	...	...	...	...	0.955	0.928	0.896	0.869	0.846	0.825	0.806	0.789	0.773	0.758	0.744	0.732	0.719
8.25	...	...	...	...	...	0.957	0.932	0.898	0.871	0.847	0.827	0.807	0.79	0.774	0.759	0.745	0.732	0.719
8.50	...	...	...	...	...	0.96	0.935	0.901	0.873	0.849	0.828	0.809	0.791	0.775	0.76	0.746	0.732	0.72
8.75	...	...	...	...	...	0.963	0.939	0.903	0.875	0.85	0.829	0.81	0.792	0.776	0.76	0.746	0.733	0.721
9.00	...	...	...	...	...	0.966	0.943	0.906	0.877	0.852	0.83	0.811	0.793	0.776	0.761	0.747	0.734	0.721
9.25	...	...	...	...	...	0.97	0.947	0.909	0.879	0.853	0.832	0.812	0.794	0.777	0.762	0.747	0.734	0.721
9.50	...	...	...	...	...	0.973	0.95	0.911	0.881	0.855	0.833	0.813	0.795	0.778	0.763	0.748	0.734	0.722
9.75	...	...	...	...	...	0.977	0.954	0.914	0.883	0.857	0.834	0.814	0.796	0.779	0.763	0.749	0.735	0.722
10.00	...	...	...	...	...	0.981	0.957	0.917	0.885	0.859	0.836	0.815	0.797	0.78	0.764	0.749	0.735	0.722
10.25	...	...	...	...	...	0.984	0.959	0.92	0.887	0.86	0.837	0.816	0.798	0.78	0.764	0.75	0.736	0.723
10.50	...	...	...	...	...	...	0.961	0.923	0.889	0.862	0.838	0.817	0.799	0.781	0.765	0.75	0.737	0.723
10.75	...	...	...	...	...	...	0.962	0.925	0.891	0.863	0.839	0.818	0.799	0.782	0.766	0.751	0.737	0.724
11.00	...	...	...	...	...	...	0.963	0.928	0.893	0.865	0.84	0.819	0.8	0.782	0.766	0.751	0.737	0.724
11.25	...	...	...	...	...	...	0.964	0.93	0.893	0.865	0.84	0.819	0.799	0.781	0.765	0.75	0.736	0.723
11.50	...	...	...	...	...	...	0.964	0.931	0.894	0.865	0.84	0.818	0.798	0.78	0.764	0.749	0.735	0.722
11.75	...	...	...	...	...	...	0.965	0.932	0.894	0.865	0.839	0.817	0.797	0.78	0.763	0.748	0.734	0.721
12.00	...	...	...	...	...	...	0.966	0.933	0.894	0.864	0.839	0.817	0.797	0.779	0.762	0.747	0.733	0.719
12.25	...	...	...	...	...	...	0.967	0.935	0.895	0.864	0.839	0.816	0.796	0.778	0.761	0.746	0.732	0.718
12.50	...	...	...	...	...	...	0.967	0.936	0.896	0.864	0.838	0.816	0.796	0.777	0.76	0.745	0.731	0.717
12.75	...	...	...	...	...	...	0.968	0.937	0.896	0.864	0.838	0.815	0.795	0.776	0.759	0.744	0.729	0.716
13.00	...	...	...	...	...	...	0.969	0.939	0.896	0.864	0.837	0.814	0.794	0.775	0.758	0.743	0.728	0.715
13.25	...	...	...	...	...	...	0.971	0.94	0.897	0.864	0.837	0.813	0.792	0.774	0.757	0.741	0.727	0.713
13.50	...	...	...	...	...	...	0.972	0.942	0.897	0.863	0.837	0.813	0.792	0.773	0.756	0.74	0.725	0.712
14.00	...	...	...	...	...	...	0.976	0.946	0.897	0.863	0.835	0.811	0.79	0.771	0.753	0.737	0.723	0.709
14.25	...	...	...	...	...	...	0.978	0.947	0.898	0.862	0.834	0.81	0.789	0.77	0.752	0.736	0.721	0.707
14.50	...	...	...	...	...	...	...	0.948	0.898	0.862	0.833	0.809	0.787	0.768	0.751	0.734	0.72	0.706

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Table XVIII-1110-1M
Superheat Correction Factor, K_{sh} (Cont'd)

Flowing Pressure (MPa)	Superheat Correction Factor, K_{sh}, Total Temperature,°C, of Superheated Steam																	
	205	225	250	275	300	325	350	375	400	425	450	475	500	525	550	575	600	625
14.75	...	...	...	...	...	...	...	0.948	0.898	0.862	0.832	0.808	0.786	0.767	0.749	0.733	0.719	0.704
15.00	...	...	...	...	...	...	...	0.948	0.899	0.861	0.832	0.807	0.785	0.766	0.748	0.732	0.717	0.703
15.25	...	...	...	...	...	...	...	0.947	0.899	0.861	0.831	0.806	0.784	0.764	0.746	0.73	0.716	0.702
15.50	...	...	...	...	...	...	...	0.947	0.899	0.861	0.83	0.804	0.782	0.763	0.745	0.728	0.714	0.7
15.75	...	...	...	...	...	...	...	0.946	0.899	0.86	0.829	0.803	0.781	0.761	0.743	0.727	0.712	0.698
16.00	...	...	...	...	...	...	...	0.945	0.9	0.859	0.828	0.802	0.779	0.759	0.741	0.725	0.71	0.696
16.25	...	...	...	...	...	...	...	0.945	0.9	0.859	0.827	0.801	0.778	0.757	0.739	0.723	0.708	0.694
16.50	...	...	...	...	...	...	...	0.945	0.9	0.858	0.826	0.799	0.776	0.756	0.738	0.721	0.706	0.692
16.75	...	...	...	...	...	...	...	0.944	0.9	0.857	0.825	0.797	0.774	0.754	0.736	0.719	0.704	0.69
17.00	...	...	...	...	...	...	...	0.944	0.9	0.856	0.823	0.796	0.773	0.752	0.734	0.717	0.702	0.688
17.25	...	...	...	...	...	...	...	0.944	0.9	0.855	0.822	0.794	0.771	0.75	0.732	0.715	0.7	0.686
17.50	...	...	...	...	...	...	...	0.944	0.9	0.854	0.82	0.792	0.769	0.748	0.73	0.713	0.698	0.684
17.75	...	...	...	...	...	...	...	0.944	0.9	0.853	0.819	0.791	0.767	0.746	0.728	0.711	0.696	0.681
18.00	...	...	...	...	...	...	...	0.944	0.901	0.852	0.817	0.789	0.765	0.744	0.725	0.709	0.694	0.679
18.25	...	...	...	...	...	...	...	0.945	0.901	0.851	0.815	0.787	0.763	0.742	0.723	0.706	0.691	0.677
18.50	...	...	...	...	...	...	...	0.945	0.901	0.85	0.814	0.785	0.761	0.739	0.72	0.704	0.689	0.674
18.75	...	...	...	...	...	...	...	0.945	0.901	0.849	0.812	0.783	0.758	0.737	0.718	0.701	0.686	0.671
19.00	...	...	...	...	...	...	...	0.946	0.901	0.847	0.81	0.781	0.756	0.734	0.715	0.698	0.683	0.669
19.25	...	...	...	...	...	...	...	0.948	0.901	0.846	0.808	0.778	0.753	0.732	0.713	0.696	0.681	0.666
19.50	...	...	...	...	...	...	...	0.95	0.9	0.844	0.806	0.776	0.75	0.729	0.71	0.693	0.677	0.663
19.75	...	...	...	...	...	...	...	0.952	0.899	0.842	0.803	0.773	0.748	0.726	0.707	0.69	0.674	0.66
20.00	...	...	...	...	...	...	...	...	0.899	0.84	0.801	0.77	0.745	0.723	0.704	0.687	0.671	0.657
20.25	...	...	...	...	...	...	...	...	0.899	0.839	0.798	0.767	0.742	0.72	0.701	0.683	0.668	0.654
20.50	...	...	...	...	...	...	...	...	0.899	0.837	0.795	0.764	0.738	0.717	0.697	0.68	0.665	0.651
20.75	...	...	...	...	...	...	...	...	0.898	0.834	0.792	0.761	0.735	0.713	0.694	0.677	0.661	0.647
21.00	...	...	...	...	...	...	...	...	0.896	0.832	0.79	0.758	0.732	0.71	0.691	0.673	0.658	0.643
21.25	...	...	...	...	...	...	...	...	0.894	0.829	0.786	0.754	0.728	0.706	0.686	0.669	0.654	0.64
21.50	...	...	...	...	...	...	...	...	0.892	0.826	0.783	0.75	0.724	0.702	0.682	0.665	0.65	0.636
21.75	...	...	...	...	...	...	...	...	0.891	0.823	0.779	0.746	0.72	0.698	0.679	0.661	0.646	0.631
22.00	...	...	...	...	...	...	...	...	0.887	0.82	0.776	0.743	0.716	0.694	0.674	0.657	0.641	0.627

Problem: Find the relieving capacity under the following conditions:

- Steam quality (dryness fraction) of 0.93
- Superheat of 100°F (total temperature of 737°F)

Solution:

(a) The wet saturated capacity:

$$W = \frac{W_s}{0.93} = \frac{8,000}{0.93} = 8,600 \text{ lb / hr}$$

(b) The superheat capacity:

$$W = K_{sh}W_s = (0.9152)(8,000) = 7,320 \text{ lb / hr}$$

NOTE: K_{sh} interpolated from Table XVIII-1110-1 for flowing conditions.

XVIII-1130 THEORETICAL FLOW

The theoretical flow for use in the establishment of the coefficient of discharge shall be calculated using eqs. XVIII-1110(1) through XVIII-1110(b)(4) with K being deleted.

XVIII-1140 SATURATED WATER CAPACITY

(a) Because the saturated water capacity is configuration sensitive, the following applies only to those safety valves that have a nozzle type construction (throat to inlet diameter ratio of 0.25 to 0.80 with a continuously contoured change and have exhibited a coefficient K_D in excess of 0.90). No saturated water rating shall apply to other types of construction.

NOTE: The manufacturer, user, and Inspector are all cautioned that for the following rating to apply, the valve shall be continuously subjected to saturated water. If, after initial relief, the flow medium changes to quality steam, the valve shall be rated as per saturated steam. Valves installed on vessels or lines containing steam-water mixture shall be rated on saturated steam.

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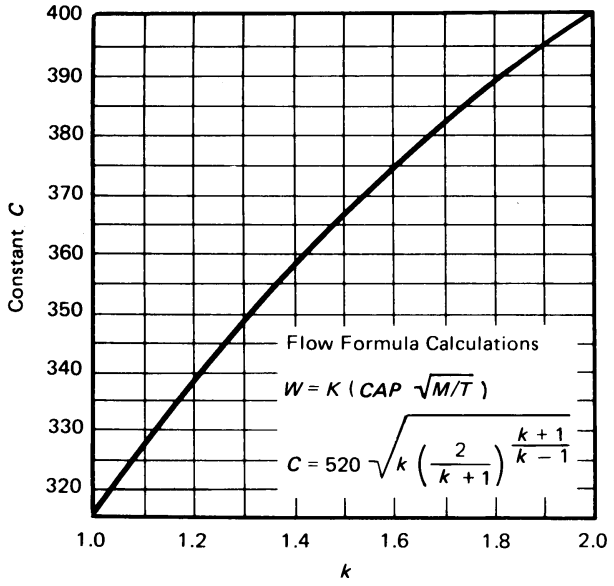
Figure XVIII-1110-1
Constant C for Gas or Vapor Related to Ratio of Specific Heats ($k = c_p/c_v$)



k	Constant C	k	Constant C	k	Constant C
1.00	315	1.26	343	1.52	366
1.02	318	1.28	345	1.54	368
1.04	320	1.30	347	1.56	369
1.06	322	1.32	349	1.58	371
1.08	324	1.34	351	1.60	372
1.10	327	1.36	352	1.62	374
1.12	329	1.38	354	1.64	376
1.14	331	1.40	356	1.66	377
1.16	333	1.42	358	1.68	379
1.18	335	1.44	359	1.70	380
1.20	337	1.46	361	2.00	400
1.22	339	1.48	363	2.20	412
1.24	341	1.50	364	...	...

(b) To determine the saturated water capacity of a valve currently rated and meeting the requirements of (a) above, refer to Figure XVIII-1140-1. Enter the graph at the set pressure of the valve, move upward to the saturated water line, and read horizontally the relieving capacity. This capacity is the theoretical, isentropic value arrived at by assuming equilibrium flow and calculated values for the critical pressure ratio.

XVIII-1150 EQUATIONS FOR CONVERSION (INCOMPRESSIBLE FLUIDS)[15]

The capacity of a pressure relief valve in terms of a liquid other than the medium for which the valve was officially rated shall be determined by application of the following equation:[16]

$$W_t = CKA\sqrt{(P - P_d)W}$$

where

A = actual discharge area of valve, in.^{2} (mm^2)
C = 2,407 (5.092)
K = coefficient of discharge
P = (set pressure × 1.10) plus atmospheric pressure, psia (MPa$_{abs}$)
P_d = pressure at discharge from valve, psia (MPa$_{abs}$)
W = density of liquid at value inlet conditions, lb/ft^3 (kg/m^3)
W_t = rated capacity, lb/hr (kg/h) of any liquid
W_W = rated capacity, lb/hr (kg/h) water @ 70°F

Table XVIII-1110(a)-1
Molecular Weights of Gases and Vapors

Gas or Vapor	Molecular Weight	Gas or Vapor	Molecular Weight
Air	28.97	Freon 22	86.48
Acetylene	26.04	Freon 114	170.90
Ammonia	17.03	Hydrogen	2.02
Butane	58.12	Hydrogen sulfide	34.08
Carbon dioxide	44.01	Methane	16.04
Chlorine	70.91	Methyl chloride	50.48
Ethane	30.07	Nitrogen	28.02
Ethylene	28.05	Oxygen	32.00
Freon 11	137.371	Propane	44.09
Freon 12	120.9	Sulfur dioxide	64.06

XVIII-1160 EXAMPLE

Given: Pressure relief valve bearing a certified rating of 1,500 gpm water @ 70°F with a set pressure of 120 psig.

Problem: Find the flow capacity of this pressure relief valve in gpm of kerosene (G = 0.82) at the same pressure rating.

Solution:

(a) For water at 70°F:

$$KA = \frac{W_W \times 500}{2,407\sqrt{(p - 14.7)(62.3)}}$$

$$KA = \frac{1,500 \times 500}{2,407\sqrt{(146.7 - 14.7)(62.3)}} = 3.436$$

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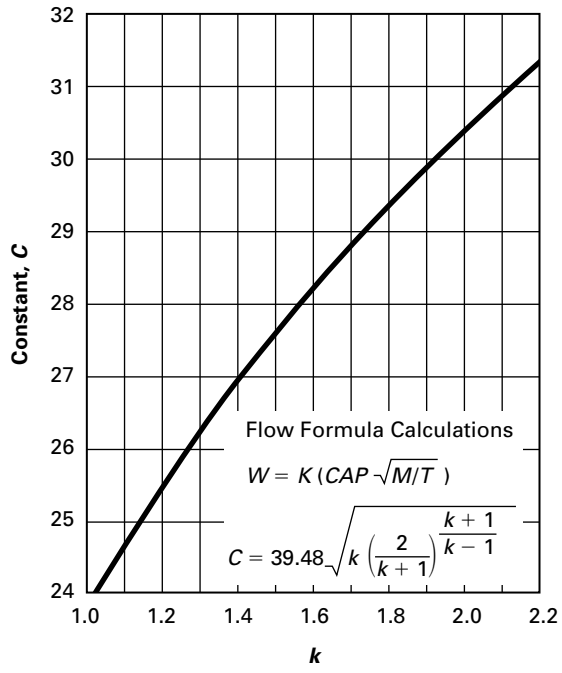
Figure XVIII-1110-1M
Constant C for Gas or Vapor Related to Ratio of Specific Heats ($k = c_p/c_v$)



k	Constant C	k	Constant C	k	Constant C
1.001	23.95	1.26	26.05	1.52	27.80
1.02	24.12	1.28	26.20	1.54	27.93
1.04	24.30	1.30	26.34	1.56	28.05
1.06	24.47	1.32	26.49	1.58	28.17
1.08	24.64	1.34	26.63	1.60	28.29
1.10	24.81	1.36	26.76	1.62	28.40
1.12	24.97	1.38	26.90	1.64	28.52
1.14	25.13	1.40	27.03	1.66	28.63
1.16	25.29	1.42	27.17	1.68	28.74
1.18	25.45	1.44	27.30	1.70	28.86
1.20	25.60	1.46	27.43	2.00	30.39
1.22	25.76	1.48	27.55	2.20	31.29
1.24	25.91	1.50	27.68	. . .	. . .

(b) For kerosene:

$$W_l = 38\,(3.436)\sqrt{\frac{(146.7 - 14.7)}{0.82}}$$

$$W_l = 1,656.60 \text{ gpm}$$

Figure XVIII-1140-1
Flow Capacity Curve for Rating Nozzle Type Safety Valves on Saturated Water (Based on 10% Overpressure)

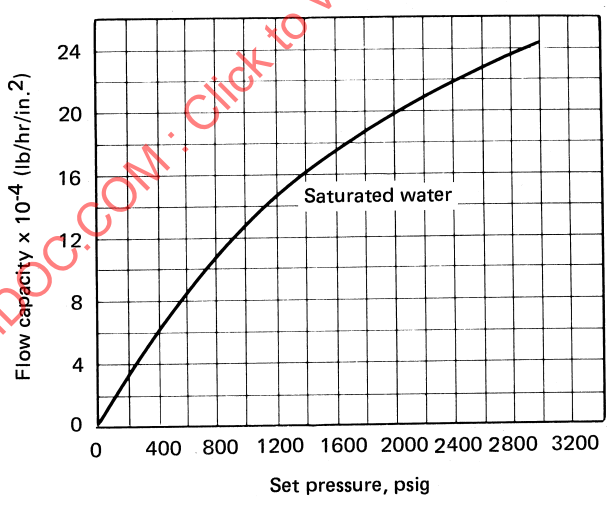


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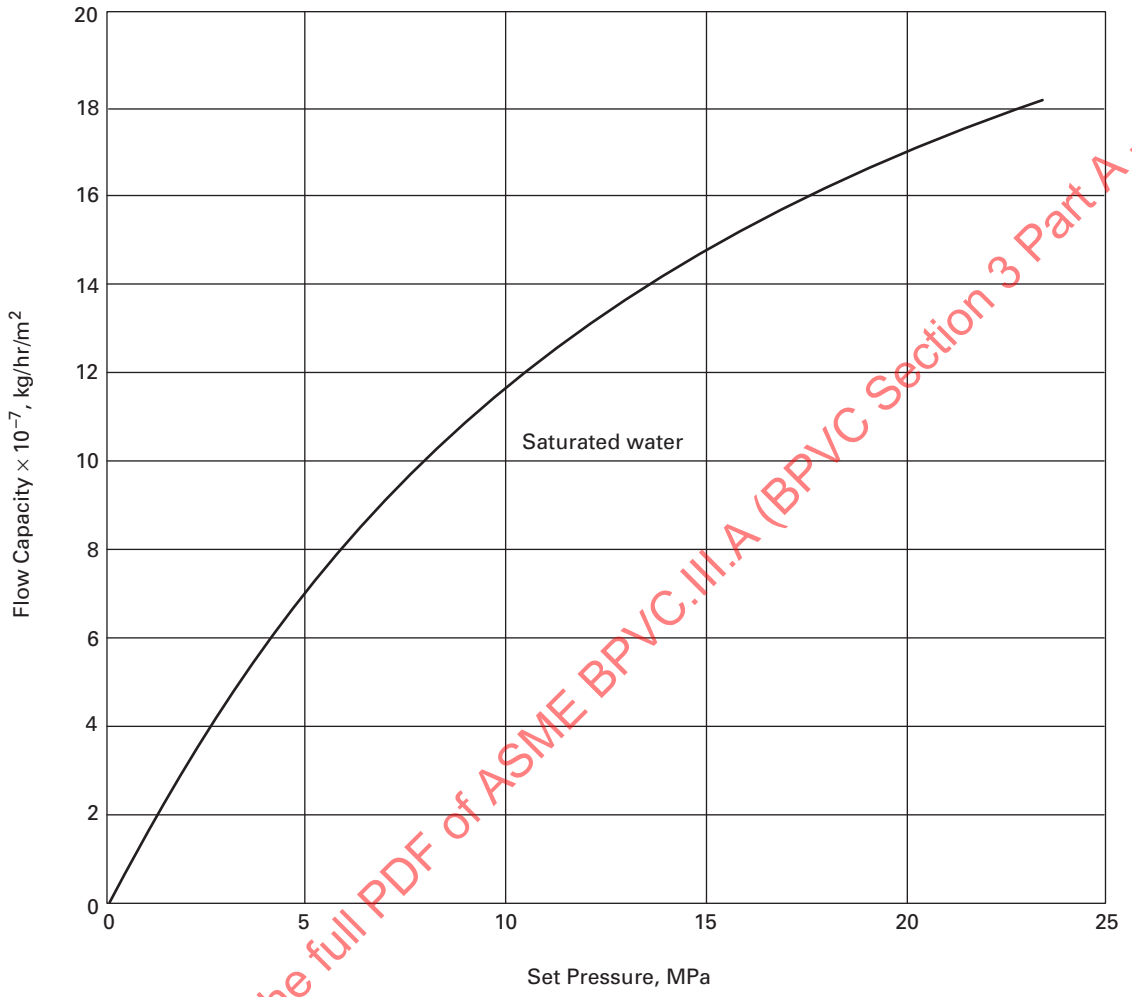
Figure XVIII-1140-1M
Flow Capacity Curve for Rating Nozzle Type Safety Valves on Saturated Water (Based on 10% Overpressure)



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MANDATORY APPENDIX XIX

ARTICLE XIX-1000
INTEGRAL FLAT HEAD WITH A LARGE OPENING

XIX-1100 GENERAL REQUIREMENTS

XIX-1110 SCOPE

(a) Rules of this Appendix apply to flat heads which have a single, circular, centrally located opening that exceeds one-half of the head diameter and have a shell/head juncture which is integrally formed or integrally attached with a full penetration weld similar to those shown in Figure XIX-1110-1. Heads of this type shall be designed according to the rules which follow and related parts of Mandatory Appendix XI.

(b) A general arrangement of an integral flat head with and without a nozzle attached at the central opening is shown in Figure XIX-1110-2.

XIX-1120 NOMENCLATURE

(a) Except as given below, the symbols used in the equations of XIX-1200 are defined in XI-3130:

A = outside diameter of flat head and shell
B_n = diameter of central opening (for nozzle, this is inside diameter, and for opening without a nozzle, diameter of the opening)
B_s = inside diameter of shell (measured below tapered hub if one exists)
M_H = moment acting at shell/head junction
P = internal design pressure
t = flat head nominal thickness

Figure XIX-1110-1
Applicable Configurations of Flat Heads

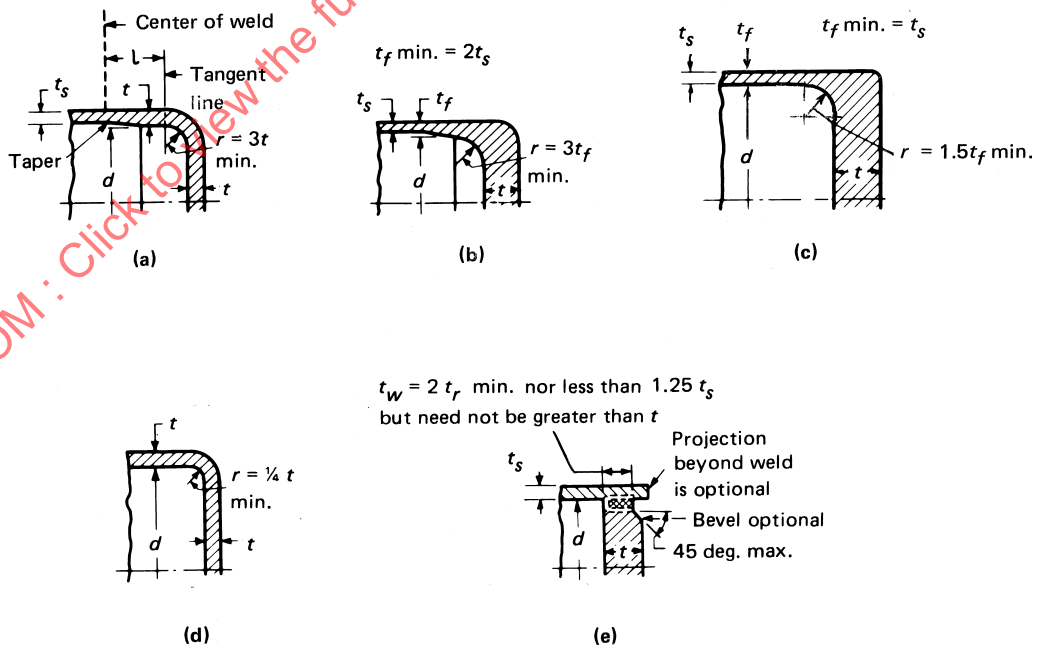

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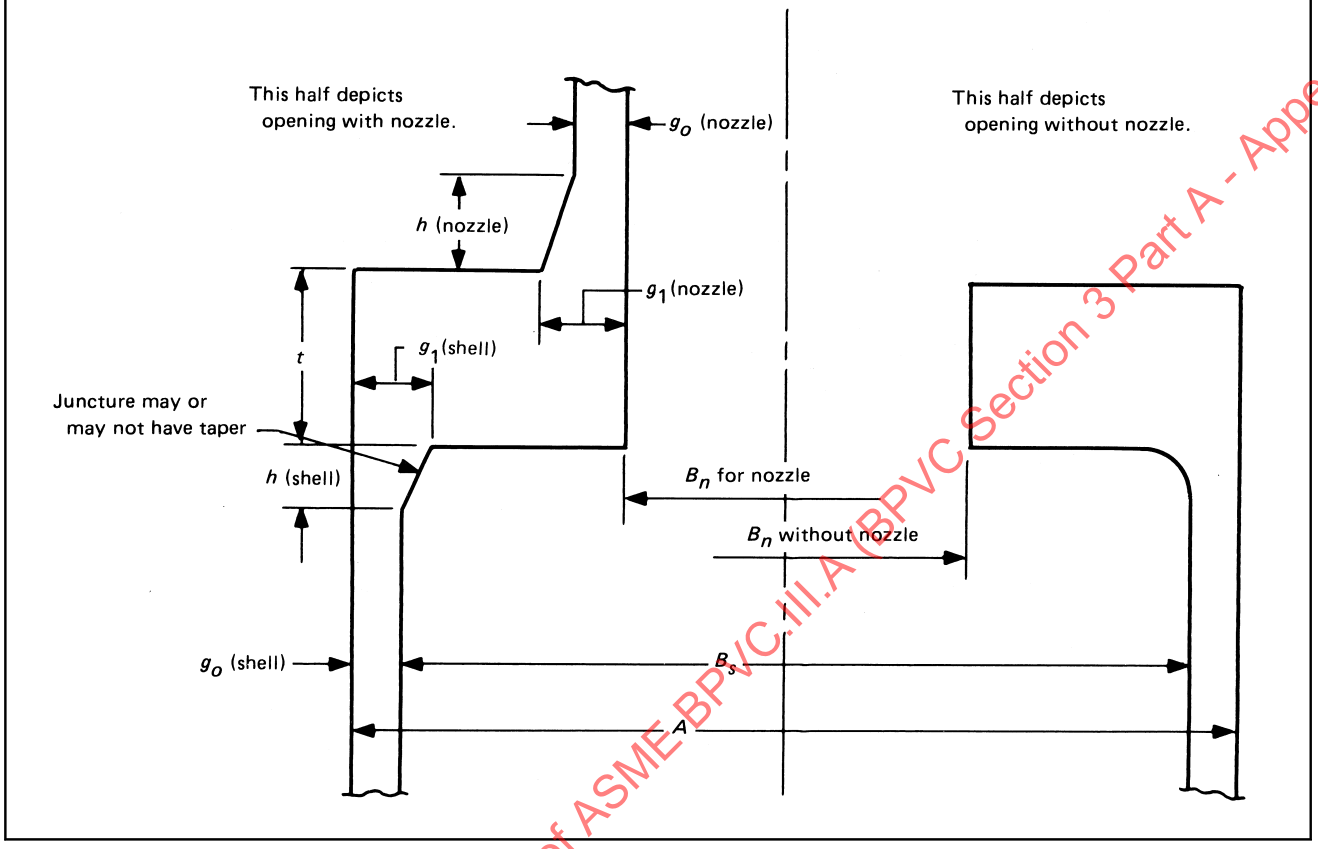
Figure XIX-1110-2
Integral Flat Head With Large Central Opening



B_1, g_0, g_1, h_0, F, V, and f are defined in XI-3130. These terms may refer to either the shell/flat head juncture or to the flat head/central opening juncture and depend upon details of those junctures.

(15) XIX-1200 DESIGN PROCEDURE

(a) Disregard the shell attached to the outside diameter of the flat head and then analyze the flat head with a central opening (with or without nozzle) in accordance with these rules.

(1) Calculate the operating moment, M_0, according to XI-3230 (there is no M_0 for gasket seating to be considered). The equations in Mandatory Appendix XI for loads (XI-3130) and moment arms (Table XI-3230-1) shall be used directly with the following designations and terms substituted for terms in Mandatory Appendix XI. Let

$C = G$ = inside diameter of shell B_s
$B = B_n$, where B_n is as shown in Figure XIX-1110-2 depending on the presence of an integral nozzle or opening without a nozzle

(2) With $K = A/B_n$, use XI-3240 to calculate stresses. Designate the calculated stresses $S_H{}^*$, $S_R{}^*$, and $S_T{}^*$.

(b) Calculate $(E\theta)^*$ as follows:

(1) for integrally attached nozzle:

$$(E\theta)^* = \frac{0.91(g_1/g_0)^2 B_1 V}{f h_0} S_H{}^*$$

(2) for an opening without a nozzle:

$$(E\theta)^* = (B_n/t) S_T{}^*$$

where g_0, g_1, B_1, V, f, h_0, and B_n all pertain to the flat head/opening as described in (a).

(c) Calculate $(E\theta)^*/M_0$.

(d) Calculate M_H:

$$M_H = \frac{(E\theta)^*}{\frac{1.74 h_0 V}{g_0{}^3 B_1} + \frac{(E\theta)^*}{M_0}(1 + Ft/h_0)}$$

where h_0, V, g_0, B_1, and F refer to the shell attached to the outside diameter of the flat head.

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(e) Calculate X_1:

$$X_1 = \frac{M_0 - M_H(1 + Ft/h_0)}{M_0}$$

where F and h_0 refer to the shell.

(f) Calculate stresses at head/shell juncture as follows.

(1) For longitudinal hub stress in shell:

$$S_{HS} = (X_1)(E\theta)^* \frac{1.10 h_0 f}{(g_1/g_0)^2 B_s V}$$

where h_0, f, g_0, g_1, B_s, and V refer to the shell.

(2) For radial stress at outside diameter:

$$S_{RS} = \frac{1.91 M_H(1 + Ft/h_0)}{B_s t^2} + \frac{0.64 F M_H}{B_s h_0 t}$$

where B_s, F, and h_0 refer to the shell.

(3) For tangential stress at outside diameter:

$$S_{TS} = \frac{(X_1)(E\theta)^* t}{B_s} - \frac{0.57(1 + Ft/h_0)M_H}{B_s t^2} + \frac{0.64 F Z M_H}{B_s h_0 t}$$

where B_s, F, and h_0 refer to the shell, and

$$Z = \frac{K^2 + 1}{K^2 - 1}$$

(g) Calculate stresses at opening/head juncture as follows:

(1) For longitudinal hub stress in central opening:

$$S_{HO} = X_1 S_H^*$$

(2) For radial stress at central opening:

$$S_{RO} = X_1 S_R^*$$

(3) For tangential stress at diameter of central opening:

$$S_{TO} = X_1 S_T^* + \frac{0.64 F Z_1 M_H}{B_s h_0 t}$$

where F, B_s, and h_0 refer to the shell, and

$$Z_1 = \frac{2K^2}{K^2 - 1}$$

(h) The preceding calculated stresses shall meet the allowable stresses in XI-3250.

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(15)

MANDATORY APPENDIX XXI
ADHESIVE ATTACHMENT OF NAMEPLATES

ARTICLE XXI-1000
REQUIREMENTS

XXI-1100 INTRODUCTION

XXI-1110 SCOPE

This Appendix provides minimum requirements for the use of adhesive systems for the attachment of nameplates, limited to

(a) the use of pressure sensitive acrylic adhesives that have been preapplied by the nameplate manufacturer to a normal thickness of at least 0.005 in. (0.13 mm) and that are protected with a moisture stable liner

(b) use on items with Design Temperatures within the range of −40°F to 300°F (−40°C to 150°C), inclusive

(c) application to clean, bare metal surfaces, with removal of antiweld spatter compound which may contain silicone

(d) nameplate nominal thickness not less than 0.020 in. (0.5 mm)

(e) use of prequalified application procedures as outlined in this Article

(f) use of the preapplied adhesive within an interval of 2 yr after adhesive application

XXI-1120 NAMEPLATE APPLICATION PROCEDURE QUALIFICATION

(a) The Certificate Holder's Quality Assurance Manual shall require that written procedures, acceptable to the Authorized Inspection Agency, for the application of adhesive backed nameplates shall be prepared and qualified.

(b) The application procedure qualification shall include the following essential variable, using the adhesive and nameplate manufacturers' recommendations where applicable:

(1) description of the pressure sensitive acrylic adhesive system employed, including generic composition

(2) the qualified temperature range [the cold box test temperature shall be −40°F (−40°C) for all applications]

(3) materials of nameplate and item when the mean coefficient of expansion at design temperature of one material is less than 85% of that for the other material

(4) finish of the nameplate and item surfaces to which the nameplate is to be attached

(5) the nominal thickness and modulus of elasticity at application temperature of the nameplate when nameplate preforming is employed. A change of more than 25% in the quantity [(nameplate nominal thickness)^2 × nameplate modulus of elasticity at application temperature] will require requalification

(6) the qualified range of preformed nameplate and companion item contour combinations when preforming is employed

(7) cleaning requirements for the item prior to attachment of the nameplate

(8) application temperature range and application pressure technique

(9) application steps and safeguards

(c) Each procedure used for nameplate attachment by pressure sensitive acrylic adhesive systems shall be qualified for outdoor exposure in accordance with Standard UL-969-82, Marking and Labeling Systems, with the following additional requirements.

(1) Width of nameplate test strip shall not be less than 1 in. (25 mm).

(2) Nameplates shall have an average adhesion of not less than 8 lb/in. (55 kPa) of width after all exposure conditions, including low temperature.

(d) Any change in (b) above shall require requalification.

(e) Each package of nameplates shall be identified with the adhesive application date.

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MANDATORY APPENDIX XXII

ARTICLE XXII-1000
RULES FOR REINFORCEMENT OF CONE-TO-CYLINDER JUNCTION UNDER EXTERNAL PRESSURE

XXII-1100 INTRODUCTION

XXII-1110 SCOPE

(a) The equations of this Appendix provide for the design of reinforcement, if needed, at the cone-to-cylinder junctions for reducer sections and conical heads where all the elements have a common axis and the half-apex angle $\alpha \leq 60$ deg. Subparagraph XXII-1300(d) provides for special analysis in the design of cone-to-cylinder intersections with or without reinforcing rings where α is greater than 60 deg.

(b) In the design of reinforcement for a cone-to-cylinder juncture, the requirements of ND-3336 shall be met.

(15) XXII-1200 NOMENCLATURE

The nomenclature given below is used in the equations of the following subparagraphs:

A = factor determined from the applicable chart in Section II, Part D, Subpart 3 for the material used in the stiffening ring, corresponding to the factor B, below, and the design temperature for the shell under consideration

A_e = effective area of reinforcement due to excess metal thickness

A_{rL} = required area of reinforcement at large end

A_{rS} = required area of reinforcement at small end

A_s = cross-sectional area of the stiffening ring

A_T = equivalent area of cylinder, cone, and stiffening ring

where

$$A_{T_L} = \frac{L_L t_S}{2} + \frac{L_c t_c}{2} + A_s, \text{ for large end}$$

$$A_{T_S} = \frac{L_S t_S}{2} + \frac{L_c t_c}{2} + A_s, \text{ for small end}$$

B = factor determined from the applicable chart in Section II, Part D, Subpart 3 for the material used for the stiffening

D_L = outside diameter of large end of conical section under consideration

D_o = outside diameter of cylindrical shell (In conical shell calculations, the value of D_s and D_L should be used in calculations in place of D_o depending on whether the small end D_s, or large end D_L, is being examined.)

D_s = outside diameter at small end of conical section under consideration

E = lowest efficiency of the longitudinal joint in the shell or head or of the joint in the reducer; $E = 1$ for butt welds in compression

E_c = modulus of elasticity of cone material

E_R = modulus of elasticity of reinforcing material

E_s = modulus of elasticity of shell material

E_x = E_c, E_R, or E_s

f_1 = axial load at large end (excluding pressure P), lb/in. (N/mm)

f_2 = axial load at small end (excluding pressure P), lb/in. (N/mm)

I_s = required moment of inertia of the stiffening ring cross section about its neutral axis parallel to the axis of the shell

I_s' = required moment of inertia of the combined ring-shell-cone cross section about its neutral axis parallel to the axis of the shell, in.^{4} (mm^4). The width of shell which is taken as contributing to the moment of inertia of the combined section shall not be greater than $1.10\sqrt{D_o T}$ and shall be taken as lying one-half on each side of the centroid of the ring. Portions of the shell plate shall not be considered as contributing area to more than one stiffening ring. If the stiffeners should be so located that the maximum permissible effective shell sections overlap on either or both sides of a stiffener, the effective shell section for that stiffener shall be shortened by one-half of each overlap.

$$k = \frac{S_S E_S}{S_R E_R} \text{ but not less than 1.0}$$

L = axial length of cone

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L_c = length of cone between stiffening rings measured along surface of cone. For cones without intermediate stiffeners,

$$L_c = \sqrt{L^2 + (R_L - R_S)^2}$$

L_L = design length of a vessel section taken as the largest of the following:

(a) the center-to-center distance between the cone-to-large-shell junction and an adjacent stiffening ring on the large shell;

(b) the distance between the cone-to-large-shell junction and one-third the depth of head on the other end of the large shell if no other stiffening rings are used.

L_s = design length of a vessel section taken as the largest of the following:

(a) the center-to-center distance between the cone-to-small-shell junction and an adjacent stiffening ring on the small shell;

(b) the distance between the cone-to-small-shell junction and one-third the depth of the head on the other end of the small shell if no other stiffening rings are used.

P = external design pressure

$Q_L = \frac{PR_L}{2} + f_1$; axial compressive force at large end due to pressure and f_1, lb/in. (N/mm)

$Q_s = \frac{PR_S}{2} + f_2$; axial compressive force at small end due to pressure and f_2, lb/in. (N/mm)

R_L = inside radius of large cylinder

R_s = inside radius of small cylinder

S' = the lesser of twice the allowable stress at design metal temperature from Section II, Part D, Subpart 1, Tables 1A and 1B or 0.9 times the tabulated yield strength at design metal temperature from Section II, Part D, Subpart 2, Tables Y-1 and Y-2

S_R = allowable stress of reinforcing material

S_s = allowable stress of shell

T = minimum required thickness of cylinder at cone-to-cylinder junction, exclusive of corrosion allowance (see ND-3133.3)

T_c = nominal thickness of cone at cone-to-cylinder junction, exclusive of corrosion allowance (see ND-3121)

T_L = the smaller of $(T_s - T)$ or $(T_c - T_r)$

T_r = minimum required thickness of cone at cone-to-cylinder junction, exclusive of corrosion allowance

T_s = nominal thickness of cylinder at cone-to-cylinder junction, exclusive of corrosion allowance (see ND-3121)

α = one-half the included (apex) angle of the cone at the centerline of the head

Δ = value to indicate need for reinforcement at cone-to-cylinder intersection having a half-apex angle $\alpha \le 60$ deg. When $\Delta \ge \alpha$, no reinforcement is required at the junction (see Table XXII-1200-1).

XXII-1300 DESIGN PRESSURE

(a) Reinforcement shall be provided at the junction of the cone with the large cylinder for conical heads and reducers without knuckles when the value of Δ obtained from Table XXII-1200-1 using the appropriate ratio $P/S'E$, is less than α. Interpolation may be made in the Table.

The cross-sectional area of the reinforcement ring shall be at least equal to that indicated by the following equation:

$$A_{rL} = \frac{kQ_L R_L \tan\alpha}{S'E}\left[1 - \frac{1}{4}\left(\frac{PR_L - Q_L}{Q_L}\right)\frac{\Delta}{\alpha}\right] \quad (1)$$

When the thickness, less corrosion allowance, of both the reducer and cylinder exceeds that required by the applicable design equations, the minimum excess thickness may be considered to contribute to the required reinforcement ring in accordance with the following equation:

$$A_e = 4T_L\sqrt{R_L T_S} \quad (2)$$

Any additional area of reinforcement which is required shall be situated within a distance of $\sqrt{R_L T_S}$ from the junction of the reducer and the cylinder. The centroid of the added area shall be within a distance of $0.5\sqrt{R_L T_S}$ from the junction.

The reinforcement ring at the cone-to-cylinder junction shall also be considered as a stiffening ring. The required moment of inertia of a circumferential stiffening ring cross section shall not be less than that determined by the following equation:

$$I_s = \frac{AD_L{}^2 A_{T_L}}{14.0}$$

Table XXII-1200-1
Values of Δ for Junctions at the Large Cylinder for α ≤ 60 deg

$P/S'E$	0	0.002	0.005	0.010	0.02
Δ, deg	0	5	7	10	15
$P/S'E$	0.04	0.08	0.10	0.125	0.15
Δ, deg	21	29	33	37	40
$P/S'E$	0.20	0.25	0.30	0.35	
Δ, deg	47	52	57	60	[Note (1)]

NOTE:
(1) Δ = 60 deg for greater values of $P/S'E$.

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The required moment of inertia of the combined ring-shell-core cross section shall not be less than that determined by the following equation:

$$I_s' = \frac{AD_L^2 A_{T_L}}{10.9}$$

The moment of inertia for a stiffening ring at the large end shall be determined by the following procedure:

Step 1. Assuming that the shell has been designed and D_L, L_L, and T are known, select a member to be used for the stiffening ring and determine cross-sectional area A_{TL}. Then calculate factor B using the following equation:

$$B = {}^{3}/_{4}\left(\frac{F_L D_L}{A_{T_L}}\right)$$

where

$$F_L = PM + f_1 \tan\alpha$$

$$M = \frac{-R_L \tan\alpha}{2} + \frac{L_L}{2} + \frac{R_L^2 - R_s^2}{3R_L \tan\alpha}$$

A_{TL} was defined previously.

Step 2. Enter the right-hand side of the applicable material chart in Section II, Part D, Subpart 3 for the material under consideration at the value of B determined by Step 1. If different materials are used for the shell and stiffening ring, use the material chart resulting in the larger value of A in Step 4, below.

Step 3. Move horizontally to the left to the material/temperature line for the design metal temperature. For values of B falling below the left end of the material/temperature line, see Step 5.

Step 4. Move vertically to the bottom of the chart and read the value of A.

Step 5. For value of B falling below the left end of the material/temperature line for the design temperature, the value of A can be calculated using the formula $A = 2B/E_x$.

Step 6. Compute the value of the required moment of inertia from the equations for I_s or I_s' above.

Step 7. Calculate the available moment of inertia of the stiffening ring using the section corresponding to that used in Step 6.

Step 8. If the required moment of inertia is greater than the moment of inertia for the section selected in Step 1, a new section with a larger moment of inertia must be selected and a new moment of inertia determined. If the required moment of inertia is smaller than the moment of inertia of the section selected in Step 1, that section is satisfactory.

The requirements of ND-4430 are to be met in attaching stiffening rings to the shell.

(b) Reinforcement shall be provided at the junction of the conical shell of a reducer without a flare and the small cylinder. The cross-sectional area of the reinforcement ring shall be at least equal to that indicated by the following formula:

$$A_{rS} = \frac{kQ_s R_s \tan\alpha}{S'E} \quad (3)$$

When the thickness, less corrosion allowance, of either the reducer or cylinder exceeds that required by the applicable design formula, the thickness may be considered to contribute to the required reinforcement ring in accordance with the following formula:

$$A_e = \sqrt{\frac{R_S T_c}{\cos\alpha}}\left(T_c - T_r\right) + \sqrt{RT_s}\left(T_s - T\right) \quad (4)$$

Any additional area of reinforcement which is required shall be situated within a distance of $\sqrt{R_S T_S}$ from the junction, and the centroid of the added area shall be within a distance of $0.5\sqrt{R_S T_S}$ from the junction.

The reinforcement ring at the cone-to-cylinder junction shall also be considered as a stiffening ring. The required moment of inertia of a circumferential stiffening ring cross section shall not be less than that determined by the following equation:

$$I_s = \frac{AD_s^2 A_{T_s}}{14.0}$$

The required moment of inertia of the combined ring-shell-cone cross section shall not be less than that determined by the following equation:

$$I'_s = \frac{AD_s^2 A_{T_s}}{10.9}$$

The moment of inertia for a stiffening ring at the small end shall be determined by the following procedure.

Step 1. Assuming that the shell has been designed and D_s, L_s, and T are known, select a member to be used for the stiffening ring and determine cross-sectional area A_{Ts}. Then calculate factor B using the following equation:

$$B = {}^{3}/_{4}\left(\frac{F_s D_s}{A_{T_s}}\right)$$

where

$$F_s = PN + f_2 \tan\alpha$$

$$N = \frac{R_s \tan\alpha}{2} + \frac{L_s}{2} + \frac{R_L^2 - R_s^2}{3R_L \tan\alpha}$$

A_{Ts} was defined previously.

Step 2. Enter the right-hand side of the applicable material chart in Section II, Part D, Subpart 3 for the material under consideration at the value of B determined by Step

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1. If different materials are used for the shell and stiffening ring, use the material chart resulting in the larger value of A in Step 4, below.

Step 3. Move horizontally to the left to the material/temperature line for the design metal temperature. For values of B falling below the left end of the material/temperature line, see Step 5.

Step 4. Move vertically to the bottom of the chart and read the value of A.

Step 5. For values of B falling below the left end of the material/temperature line for the design temperature, the value of A can be calculated using the formula $A = 2B/E_x$.

Step 6. Compute the value of the required moment of inertia from the equations for I_s or I_s' above.

Step 7. Calculate the available moment of inertia of the stiffening ring using the section corresponding to that used in Step 6.

Step 8. If the required moment of inertia is greater than the moment of inertia for the section selected in Step 1, a new section with a larger moment of inertia must be selected and a new moment of inertia determined. If the required moment of inertia is smaller than the moment of inertia for the section selected in Step 1, that section is satisfactory.

The requirements of ND-4430 are to be met in attaching stiffening rings to the shell.

(c) Reducers, such as those made up of two or more conical frustums having different slopes, may be designed in accordance with (d) below.

(d) As an alternative to the rules provided in the preceding (a) and (b) and when half the apex angle is greater than 60 deg, the design may be based on special analysis such as numerical methods or the beam-on-elastic-foundation analysis of Timoshenko, Hetenyi, or Watts and Lang. The stresses at the junction shall meet all of the allowable stress limits of this Division. The effect of shell and cone buckling on the required area and moment of inertia at the joint shall also be considered in the analysis. The theoretical buckling pressure of the junction shall be at least 3.3 times the allowable external design pressure of the junction.

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MANDATORY APPENDIX XXIII
QUALIFICATIONS AND DUTIES OF SPECIALIZED PROFESSIONAL ENGINEERS

ARTICLE XXIII-1000
QUALIFICATIONS AND DUTIES

XXIII-1100 SCOPE

This Appendix presents minimum requirements for the qualification of personnel engaged in the certification activities. The personnel addressed are those who perform the following specialty fields:

(a) for Division 1

(1) certification of the Design Specification on behalf of the Owner

(2) certification of the Design Report on behalf of the N Certificate Holder

(3) certification of the Overpressure Protection Report on behalf of the Owner

(4) certification of the Load Capacity Data Sheet on behalf of the N Certificate Holder

(b) for Division 2

(1) certification of the Design Specification on behalf of the Owner

(2) certification of the Construction Specification, Design Drawings, and Design Report on behalf of the Designer

(c) for Division 3

(1) certification of the Design Specification on behalf of the N3 Certificate Holder

(2) certification of the Design Report on behalf of the N3 Certificate Holder

(3) certification of the Fabrication Specification on behalf of the N3 Certificate Holder

(d) for Division 4 — in the course of preparation

(e) for Division 5

(1) for Class A nonmetallic core support structures

(-a) certification of the Design Specification on behalf of the Owner

(-b) certification of the Construction Specification, Design Drawings, and Design Report on behalf of the Designer

(2) for all other components and supports

(-a) certification of the Design Specification on behalf of the Owner

(-b) certification of the Design Report on behalf of the N Certificate Holder

(-c) certification of the Overpressure Protection Report on behalf of the Owner

(-d) certification of the Load Capacity Data Sheet on behalf of the N Certificate Holder

Also provided are the duties of these personnel in the performance of the activities described above.

XXIII-1200 QUALIFICATIONS

XXIII-1210 GENERAL

One or more Registered Professional Engineers (PE) shall be selected by the Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, to perform Code activities in the appropriate specialized field(s), provided the qualifications of the PE in meeting the requirements of this Appendix have been evaluated and verified by the Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, responsible for the activity being certified or reviewed. Guidelines for demonstrating PE qualifications are contained in Supplement 1 of this Appendix. A record of the qualifications of the PE shall be maintained by the responsible organization or the PE.

XXIII-1220 REGISTRATION AND EXPERIENCE

He shall be a Registered Professional Engineer in at least one state of the United States or Province of Canada and have the following qualifications.

He shall have 4 yr of varied application experience at least 2 of which have been in each specialty field for which he performs certifying or review activities as delineated in XXIII-1230 through XXIII-1270. In addition, he shall keep current his knowledge of Code requirements and continue his professional development in his specialty field through personal study and experience, or by attendance at appropriate courses, seminars, Society meetings, and technical committee meetings. The Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as

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applicable, shall review the qualifications of the PE at least once every 3 yr to assure that his qualifications have been maintained. A continuing record of all such activity shall be included in the qualification records of the PE.

XXIII-1230 CERTIFIER OF THE DESIGN SPECIFICATION FOR ALL DIVISIONS

To qualify as certifier of the Design Specification, the PE shall be experienced in the applicable field of design and related nuclear facility requirements, and in the application of the requirements of the Code relating to the construction of nuclear facility items. This experience shall indicate that the PE has sufficient knowledge of anticipated plant and system operating and test conditions (Divisions 1, 2, and 5) or containment systems operating and test conditions (Division 3) and their relationship to Code design criteria pertinent to the applicable Code item. In addition, he shall be knowledgeable of the specific Code requirements pertaining to his specialty field. Guidelines reflecting the degree of knowledge appropriate for preparation of the Design Specification are contained in Supplement 2, Table S2-1.

The following paragraphs provide additional Division-specific requirements necessary to establish the proper qualifications needed to certify Section III Design Specifications.

XXIII-1231 Certifier of the Design Specification for Divisions 1, 2, and 5

The PE, certifying on behalf of the Owner, shall be experienced in the applicable field of design and related nuclear facility requirements and in the application of the requirements of the Code relating to the construction of nuclear facility items.

For Division 5 applications, the PE shall also be knowledgeable of the additional Design Specification requirements necessary for proper elevated temperature design associated with high temperature reactors. Unique issues are also associated with the proper design and construction requirements for Division 5 nonmetallic core support structures.

XXIII-1232 Certifier of the Design Specification for Division 3

The PE, certifying on behalf of the N3 Certificate Holder, shall be experienced in the applicable field of Division 3 storage and transportation containment and associated internal support structure (basket) design requirements.

XXIII-1233 Certifier of the Design Specification for Division 4

In the course of preparation.

XXIII-1240 CERTIFIER OF THE LOAD CAPACITY DATA SHEET FOR DIVISIONS 1 AND 5 AND THE DESIGN REPORT FOR DIVISIONS 1, 3, AND 5 (EXCLUDING NONMETALLIC CORE SUPPORT STRUCTURES)

To qualify as certifier of the Load Capacity Data Sheet or Design Report, the PE shall be experienced in the applicable field of design and analysis and in the application of the requirements of the Code. In addition, he shall be knowledgeable of the specific Code requirements pertaining to his specialty field. Guidelines reflecting the degree of knowledge appropriate for preparation of the Design Report and the Load Capacity Data Sheet are contained in Supplement 2, Tables S2-2 and S2-3, respectively.

XXIII-1250 CERTIFIER OF THE FABRICATION SPECIFICATION FOR DIVISION 3

To qualify as certifier of the Fabrication Specification on behalf of the N3 Certificate Holder, the PE shall be experienced in the applicable field of design, analysis, fabrication, and the application of Division 3 requirements. Guidelines reflecting degree of knowledge appropriate for the preparation of the Fabrication Specification are contained in Supplement 2, Table S2-4.

XXIII-1260 CERTIFIER OF THE OVERPRESSURE PROTECTION REPORT FOR DIVISIONS 1, 2, AND 5

To qualify as certifier of the Overpressure Protection Report on behalf of the Owner, the PE shall be experienced in nuclear facility systems design, and in facility operation and safety control. In addition, he shall be knowledgeable of the specific Code requirements pertaining to his specialty field. Guidelines reflecting the degree of knowledge appropriate for preparation of the Report on Overpressure Protection are contained in Supplement 2, Table S2-5.

XXIII-1270 CERTIFIER OF THE CONSTRUCTION SPECIFICATION, DESIGN DRAWING, AND DESIGN REPORT FOR DIVISION 2 AND FOR DIVISION 5 NONMETALLIC CORE SUPPORT STRUCTURES

To qualify as certifier of the Construction Specification, Design Drawings, or Design Report, the PE shall be experienced in the applicable field of design and analysis and in the application of the requirements of the Code. In addition, he shall be knowledgeable of the specific Code requirements pertaining to his specialty field. Guidelines reflecting the degree of knowledge appropriate for preparation of the Construction Specification, Design Drawings, and Design Report are contained in Supplement 2, Table S2-6.

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XXIII-1300 DUTIES

XXIII-1310 GENERAL

The certification activities covered in this Appendix may be performed only if the PE has assured himself that he is qualified to do so by virtue of a self-review establishing that his qualifications meet those required by this Appendix. He shall be familiar with the Quality Assurance requirements of the organization responsible for providing the document as these requirements relate to his work. For certification activities, the document being certified must have been reviewed in detail by the certifying PE, or prepared by him or prepared under his responsible direction. He shall prepare a statement[17] to be affixed to the document attesting to compliance with the applicable requirements of the Code.

XXIII-1320 CERTIFICATION OF THE DESIGN SPECIFICATION

It is the responsibility of the PE certifying the Owner's Design Specification (Divisions 1, 2, and 5) or the N3 Certificate Holder's Design Specification (Division 3) to assure that the Design Specification is correct, complete, and in compliance with the requirements of the applicable Edition and Addenda of the Code. As a minimum, the certifier of the Design Specification shall assure that

(a) the function of the item is properly specified

(b) the design requirements, including identification of the item Design and Service Loadings or Operating Conditions and their combinations and associated Limits, are properly specified

(c) the proper environmental conditions, including corrosion, erosion, and radiation, are specified

(d) the Code classification is properly specified

(e) the definition of the specific boundaries and load conditions on these boundaries for each item is specified, and that the boundaries and associated load conditions between adjacent components and structure are compatible with the overall system design

(f) the specified materials for items covered by the Code are permitted by the Code for the applicable item

(g) all requirements with regard to impact testing are specified

(h) any restrictions on or additional requirements for heat treating are specified

(i) any restrictions on cladding materials are specified

(j) any reduction to design stress intensity values, allowable stress, or fatigue curves necessitated by the given environmental conditions are specified

(k) the necessary information concerning the load carrying capacity of structures supporting Code items is given

(l) when operability of a component is a requirement, the Design Specification shall make reference to other appropriate documents that specify the operating requirements

(m) the overpressure protection requirements are specified

(n) the Code Edition, Addenda, and Code Cases to be used for construction are specified

XXIII-1330 CERTIFICATION OF THE DESIGN REPORT FOR DIVISIONS 1, 3, AND 5 (EXCLUDING NONMETALLIC CORE SUPPORT STRUCTURES)

It is the responsibility of the PE certifying the Design Report to assure that the design of the item complies with the requirements of the applicable Edition and Addenda of the Code for the Design, Service Loadings or Operating Conditions, and Test Loadings that have been specified in the Design Specification. As a minimum, the certifier of the Design Report shall assure that

(a) the Design Report reflects the design as shown by the drawings used for construction and that all modifications to the drawings and construction deviations have been reconciled with the Design Report

(b) the design as shown by the drawings is in accordance with the requirements of the Code

(c) the Design Report is in accordance with the requirements of the Code

(d) materials specified for Code items are permitted by the Code, and that any reduction of material impact properties from heat treatments, welding, and forming have been taken into account

(e) the Design Report is based on the Design, Service Loadings or Operating Conditions, and Test Loadings stated in the Design Specification

(f) the specified requirements for protection against nonductile fracture are specified

(g) all special nondestructive examinations required to validate unique features have been specified in appropriate documents/drawings

(h) the specified test pressure and temperature are in compliance with Code requirements

(i) adequate analytical techniques have been employed to assess the structural adequacy of the item of concern for the Design, Service Loadings or Operating Conditions, and Test Loadings specified

XXIII-1340 CERTIFICATION OF THE OVERPRESSURE PROTECTION REPORT FOR DIVISIONS 1, 2, AND 5

It is the responsibility of the PE certifying the Overpressure Protection Report to assure that the report has been reconciled with the system requirements and with the requirements of the applicable Subsection of the Code.

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XXIII-1350 CERTIFICATION OF THE LOAD CAPACITY DATA SHEET FOR DIVISIONS 1 AND 5

It is the responsibility of the PE certifying the Load Capacity Data Sheet on behalf of the N Certificate Holder to determine that the load capacity of the component or piping support is rated in accordance with Subsection NF of the Code. He shall assure that the design of the component or piping support complies with the requirements of the applicable Edition and Addenda of the Code for the Design, Service, and Test Loadings specified in the Design Specification. In addition, his duties shall include the requirements of XXIII-1330(a) through XXIII-1330(i) for the data substantiating the Load Capacity Data Sheet.

XXIII-1360 CERTIFICATION OF THE CONSTRUCTION SPECIFICATION, DESIGN DRAWINGS, OR DESIGN REPORT FOR DIVISION 2 AND FOR DIVISION 5 NONMETALLIC CORE SUPPORT STRUCTURES

It is the responsibility of the PE certifying the Construction Specification, Design Drawings, or Design Report on behalf of the Designer for Division 2 to assure that each of the above principal Code documents is correct, complete, and in accordance with the Design Specification and Section III, Division 2. As a minimum, the certifier of each of the principal Code documents shall assure

(a) for Division 2:

(1) that the Design Drawings contain

(-a) concrete and steel liner thicknesses

(-b) size and location of reinforcing steel, prestressing tendons, and penetrations

(-c) the latest revisions to reflect any change in design

(2) that the Design Report includes, as a minimum, the requirements of XXIII-1330(a) through XXIII-1330(i), as applicable

(3) that the Construction Specification has provided the following in accordance with the Code:

(-a) material specifications

(-b) material shipping, handling, and storage requirements

(-c) requirements for personnel or equipment qualification

(-d) material or part examination and testing requirements

(-e) acceptance and leakage testing requirements

(-f) requirements for shop drawings

(-g) requirements for batching, mixing, placing, and curing of concrete

(-h) requirements for the fabrication and installation of the prestressing system, reinforcing steel, and embedments

(-i) identification of parts requiring a Code stamp

(-j) design life for parts and materials where necessary to establish compliance with the Design Specification

(-k) construction surveillance to be performed by the Designer as required by the Design Specification

(-l) the latest revisions to reflect any change in design

(b) for Division 5:

(1) that the Design Drawings contain

(-a) all details necessary to construct the item in accordance with the requirements of the Design Specification, Construction Specification, and appropriate Division 5 Code rules

(-b) the latest revisions to reflect any change in design

(2) that the Design Report includes, as a minimum, the requirements of XXIII-1330(a) through XXIII-1330(i), as applicable

(3) that the Construction Specification has provided the following, as a minimum, in accordance with the Code:

(-a) material specifications

(-b) material shipping, handling, and storage requirements

(-c) inspection requirements

(-d) appropriate Code references

(-e) requirements for personnel or equipment qualification

(-f) material or item examination and testing requirements

(-g) acceptance testing requirements

(-h) requirements for shop and field drawings

(-i) identification of items

(-j) construction surveillance to be performed by the Designer as required by the Design Specification

(-k) construction documents that require review by the Designer and those that require both review and approval by the Designer

XXIII-1370 CERTIFICATION OF THE FABRICATION SPECIFICATION FOR DIVISION 3

It is the responsibility of the PE certifying the Fabrication Specification on behalf of the N3 Certificate Holder for Division 3 to assure that the Fabrication Specification is correct, complete, and in accordance with the Design Specification, Design Output Documents, and Section III, Division 3. The certifier of the Fabrication Specification shall assure that it contains sufficient detail to provide a complete basis for fabrication.

As a minimum, the Fabrication Specification shall contain the following:

(a) material specifications

(b) material shipping, handling, and storage requirements

(c) requirements for personnel or equipment qualification

(d) weld joint design requirements

(e) fabrication dimensions and tolerances

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(f) identification of Code boundaries
(g) identification of parts requiring a Code stamp
(h) material or part examination and testing requirements
(i) acceptance and leakage testing requirements
(j) requirements for shop drawings
(k) design life for parts and materials where necessary to establish compliance with the Design Specification
(l) fabrication surveillance to be performed by the N3 Certificate Holder as required by the Design Specification
(m) the latest revisions to reflect any change in design
(n) requirements for as-built documentation
(o) a listing of design drawings (by inclusion or reference)

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MANDATORY APPENDIX XXIII
SUPPLEMENTS

SUPPLEMENT 1 NONMANDATORY GUIDELINES FOR DEMONSTRATING PE QUALIFICATIONS

This Supplement provides suggested methods for demonstrating that the requirements for the qualification of personnel engaged in certification activities have been met. The Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, responsible for the activity being certified should establish procedures or instructions for evaluating, verifying, and documenting the qualifications of the PE engaged in certifying activities as required by this Appendix.

1.1 REGISTRATION AND EXPERIENCE

The PE's qualifications for the requirements of XXIII-1220 may be demonstrated as follows:

(a) PE registration in one or more states of the United States or provinces of Canada should be documented on records that, as a minimum, include

(1) PE's identification

(2) state or province of registration

(3) registration number

(4) expiration date

(b) The 4 yr of varied application experience, including 2 yr in his specialty field(s), should be documented in a resume describing the PE's Code experience and places and dates of employment.

(c) In order for the PE to keep current his knowledge of the Code requirements and to continue his professional development in his specialty field(s), as required by this Appendix, he should, in the 36-month period preceding the date of qualification, have performed Code activities requiring certification in his specialty field(s), or have been engaged in the application of Code requirements to an equivalent extent, but not necessarily including Certification. Alternatively, he should have done two or more of the following:

(1) taught or attended an appropriate course or training program

(2) taught or attended an appropriate seminar

(3) attended an ASME or ASME/ACI Code meeting

(4) attended a technical society meeting related to his specialty field

(d) The PE's participation in these activities should be documented in appropriate records that, as a minimum, include

(1) PE's identification

(2) description of Code activities performed

(3) course or training program description, duration, and date completed

(4) seminar description, duration, and date attended

(5) ASME or ASME/ACI Code meeting(s) and date(s) attended

(6) technical society meeting(s) and date(s) attended

(7) the PE's function (i.e., attendee, member, speaker, chairman, etc.) indicating the nature of his participation

1.2 CODE KNOWLEDGE

Supplement 2 of this Appendix provides guidance regarding knowledge of the Code that the PE should have in each specialty field. The PE's qualifications regarding knowledge of the Code may be demonstrated by any one of the following methods:

(a) The Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, upon review of the experience record of the PE, declares in writing that

(1) the PE's knowledge of the Code in his specialty field meets the requirements, and

(2) the PE's experience record reflects successful performance of the applicable Code activities in connection with the construction of ASME Code items.

(b) Another PE previously qualified to this Appendix, designated by the Owner, Designer, N Certificate Holder, or N3 Certificate Holder, as applicable, and familiar with the requirements of the Code, after reviewing the qualifications of the PE to be qualified, attests in writing that the PE's knowledge of the Code in his specialty field(s) meets the requirements of this Appendix.

(c) Attendance of the PE at appropriate courses or seminars that provide instruction in the Code for his specialty field(s) to import knowledge of the Code required by this Appendix. Training should be scheduled as required by this Appendix, at a frequency consistent with significant changes to the Code in his specialty field(s). Training may be accomplished by attending in-house courses or courses presented by others. Training should be documented on appropriate records that, as a minimum, include

(1) attendee's identification

(2) instructor's name and affiliation

(3) outline or description of course or seminar

(4) date and duration of course or seminar

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(d) Examination of the PE in his specialty field(s), either written or oral, to verify his knowledge of the Code as required by this Appendix. The examination may be developed and/or administered either in-house or by others. Examinations should be documented on appropriate records that, as a minimum, include

(1) attendee's identification
(2) examiner's name and affiliation
(3) outline or description of examination
(4) date and results of the examination

SUPPLEMENT 2 NONMANDATORY GUIDELINES FOR ESTABLISHING ASME CODE KNOWLEDGE

This Supplement provides guidelines for establishing the degree of Code knowledge required by the certifying PE in his specialty field. In the paragraphs that follow, the degree of knowledge required by the PE of the requirements of the Code pertaining to his specialty field is indicated by the terminology "general knowledge" and "working knowledge."

As used in this Supplement, "general knowledge" signifies having sufficient acquaintance with the Code to be conversant with other persons involved in its applications, and to make prudent judgments in the application of Code requirements.

As used in this Supplement, "working knowledge" signifies understanding by prior customary involvement in a specialty field of the Code requirements and of the principles on which the Code rules are based, to the extent that the PE may apply or direct others in the application of the requirements.

In this sense, "working knowledge" implies a more thorough understanding of the Code requirements and the ability to apply them than does "general knowledge" of the Code.

The degree of knowledge in the various areas of the Code cited in Tables S2-1 through S2-6 is based upon the more common Code items and activities. There may be special items or activities for which the degree of knowledge in a specific Code area must be more detailed than shown in the applicable table, or may require knowledge of specific Code areas that are not cited.

In the following tables, the degree of knowledge required by the PE of the various requirements of the Code pertinent to his specialty field is indicated by the letter G for "general knowledge" and the letter W for "working knowledge."

Table S2-1
Design Specification — Divisions 1 Through 3 and 5

Division 1		Division 2		Division 3		Division 5 Nonmetallic CSS		Remaining Division 5	
NCA-1000	W	NCA-1000	W	WA-1000	W	HAB-1000	W	HAA-NCA-1000 [Note (1)]	W
NCA-2000	W	NCA-2000	W	WA-2000	W	HAB-2000	W	HAA-NCA-2000	W
NCA- 3100	W	NCA-3100	W	WA-3100	W	HAB-3000	W	NCA-3100	W
NCA-3200	W	NCA-3200	W	WA-3300	W	HAB-4000	G	NCA-3200	W
NCA-3300	W [Note (2)]	NCA-3300	W [Note (2)]	WA-3400	W	HAB-5000	G	NCA-3300	W [Note (2)]
NCA-3400	W [Note (2)]	NCA-3400	W [Note (2)]	WA-3800	G	HAB-7000	G	NCA-3400	W [Note (2)]
NCA-3500	W [Note (2)]	NCA-3500	W [Note (2)]	WA-4000	W	HAB-8000	G	NCA-3500	W [Note (2)]
NCA-3600	W	NCA-3600	W	WA-5000	G	...	...	NCA-3600	W
NCA-3700	G	NCA-3700	G	WA-7000	G	HHA-1000	W	NCA-3700	G
NCA-3800	G	NCA-3800	G	WA-8000	W	HHA-2000	G	NCA-3800	G
NCA-3900	G [Note (2)]	NCA-3900	G [Note (2)]	...	...	HHA-3000	G	NCA-3900	G [Note (2)]
NCA-4000	G	NCA-4000	G	WX-1000	W	HHA-4000	G	NCA-4000	G
NCA-5000	G	NCA-5000	G	WX-2000	G	HHA-5000	G	NCA-5000	G
NCA-6000	G	NCA-6000	G	WX-3000	W	...	...	NCA-6000	G
NCA-7000	G	NCA-7000	W	WX-4000	G	...	...	HAA-NCA-7000	G
NCA-8000	G	NCA-8000	G	WX-5000	G	...	...	HAA-NCA-8000	G
				WX-6000	W	...	...	...	...
NX-1000	W	CC-1000	W	...	...	...	...	HX-1000	W
NX-2100	G	CC-2000	G	...	...	...	...	HX-2000	G
NX-2300	G	CC-3000	G	...	...	...	...	HX-3000	G [Note (2)]

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Table S2-1
Design Specification — Divisions 1 Through 3 and 5 (Cont'd)

Division 1		Division 2		Division 3		Division 5 Nonmetallic CSS		Remaining Division 5	
NX-2500	G	CC-4000	G	...	...	...	...	HX-4000	G
NX-2600	G	CC-5000	G	...	...	...	...	HX-5000	G
NX-3100	G	CC-6000	G	...	...	...	...	HX-6000	W [Note (2)]
NX-3200	G	...	...	...	...	...	...	...	...
NX-3300	G [Note (2)]	...	...	...	...	...	...	HFA-HG-1000	W [Notes (2), (3)]
NX-3400	G [Note (2)]	...	...	...	...	...	...	NF-HG-2000	G [Notes (2), (3)]
NX-3500	G [Note (2)]	...	...	...	...	...	...	NF-HG-3000	G [Notes (2), (3)]
NX-3600	G [Note (2)]	...	...	...	...	...	...	NF-HG-4000	G [Notes (2), (3)]
NX-3700	G [Note (2)]	...	...	...	...	...	...	NF-HG-5000	G [Notes (2), (3)]
NX-3800	G [Note (2)]	...	...	...	...	...	...	...	...
NX-3900	G [Note (2)]	...	...	...	...	...	...	...	...
NX-4100	G	...	...	...	...	...	...	...	...
NX-4210	G	...	...	...	...	...	...	...	...
NX-4220	G	...	...	...	...	...	...	...	...
NX-4240	G	...	...	...	...	...	...	...	...
NX-4620	G	...	...	...	...	...	...	...	...
NX-5100	G	...	...	...	...	...	...	...	...
NX-5200	G	...	...	...	...	...	...	...	...
NX-6000	W [Note (2)]	...	...	...	...	...	...	...	...
NF-NG-1000	W [Note (2)]	...	...	...	...	...	...	...	...
NF-NG-2100	G [Note (2)]	...	...	...	...	...	...	...	...
NF-NG-2300	G [Note (2)]	...	...	...	...	...	...	...	...
NF-NG-3000	G [Note (2)]	...	...	...	...	...	...	...	...
NF-NG-4100	G [Note (2)]	...	...	...	...	...	...	...	...
NF-NG-4200	G [Note (2)]	...	...	...	...	...	...	...	...
NF-NG-5000	G [Note (2)]	...	...	...	...	...	...	...	...

Legend:
CSS = Core Support Structures
G = General Knowledge
HX = HB/HC, as applicable (including Subparts A and B), as well as references to Subsections NB and NC rules, respectively
NX = NB/NC/ND/NE, as applicable
W = Working Knowledge
WX = WB/WC/WD, as applicable

NOTES:
(1) Subsection HA, Subpart A references Subsection NCA for general requirements.
(2) As applicable.
(3) Subsections HF and HG (Subparts A and B) rules as well as references to Subsections NF and NG rules, respectively.

Table S2-2
Design Report — Divisions 1, 3, and 5 (Excluding Nonmetallic CSS)

Division 1		Division 3		Division 5	
NCA-1000	G	WA-1000	G	HAA-NCA-1000 [Note (1)]	G
NCA-2000	G	WA-2000	G	HAA-NCA-2000	G
NCA-3100	G	WA-3100	W	NCA-3100	G
NCA-3200	G	WA-3300	W	NCA-3200	G
NCA-3300	W [Note (2)]	WA-3400	G	NCA-3300	W [Note (2)]

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Table S2-2
Design Report — Divisions 1, 3, and 5 (Excluding Nonmetallic CSS) (Cont'd)

Division 1		Division 3		Division 5	
NCA-3400	W [Note (2)]	WA-3800	W	NCA-3400	W [Note (2)]
NCA-3500	W [Note (2)]	WA-4000	W	NCA-3500	W [Note (2)]
NCA-3600	G	WA-5000	G	NCA-3600	G
NCA-3700	G	WA-7000	G	NCA-3700	G
NCA-3800	G	WA-8000	G	NCA-3800	G
NCA-3900	G	...	...	NCA-3900	G
NCA-4000	G	WX-1000	W	NCA-4000	G
NCA-7000	G	WX-2000	W	HAA-NCA-7000	G
NCA-8000	G	WX-3000	W	HAA-NCA-8000	G
		WX-4000	G	...	...
NX-1000	W	WX-5000	G	HX-1000	W
NX-2100	W	WX-6000	W	HX-2000	W
NX-2300	W	...	...	HX-3000	W [Note (2)]
NX-2500	W [Note (2)]	...	...	HX-4000	G
NX-2600	G	...	...	HX-5000	G
NX-3100	W	...	...	HX-6000	G
NX-3200	W [Note (2)]	...	...	...	...
NX-3300	W [Note (2)]	...	...	HFA-HG-1000	W [Notes (2), (3)]
NX-3400	W	...	...	NF-HG-2000	W [Notes (2), (3)]
NX-3500	W [Note (2)]	...	...	NF-HG-3000	W [Notes (2), (3)]
NX-3600	W [Note (2)]	...	...	NF-HG-4000	G [Notes (2), (3)]
NX-3700	W [Note (2)]	...	...	NF-HG-5000	G [Notes (2), (3)]
NX-3800	W [Note (2)]	...	...	...	...
NX-3900	W [Note (2)]	...	...	...	...
NX-4100	G	...	...	...	...
NX-4210	G	...	...	...	...
NX-4220	G	...	...	...	...
NX-4240	G	...	...	...	...
NX-4620	G	...	...	...	...
NX-5100	G	...	...	...	...
NX-5200	G	...	...	...	...
NX-6000	G	...	...	...	...
NF-NG-1000	W [Note (2)]	...	...	...	...
NF-NG-2100	W [Note (2)]	...	...	...	...
NF-NG-2300	W [Note (2)]	...	...	...	...
NF-NG-3000	W [Note (2)]	...	...	...	...
NF-NG-4100	G [Note (2)]	...	...	...	...
NF-NG-4200	G [Note (2)]	...	...	...	...
NF-NG-5000	G [Note (2)]	...	...	...	...

Legend:
CSS = Core Support Structures
G = General Knowledge
HX = HB/HC, as applicable (including Subparts A and B), as well as references to Subsections NB and NC rules, respectively
NX = NB/NC/ND/NE, as applicable
W = Working Knowledge
WX = WB/WC/WD, as applicable

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Table S2-2
Design Report — Divisions 1, 3, and 5 (Excluding Nonmetallic CSS) (Cont'd)

NOTES:
(1) Subsection HA, Subpart A references Subsection NCA for general requirements.
(2) As applicable.
(3) Subsections HF and HG (Subparts A and B) rules as well as references to Subsections NF and NG rules, respectively.

Table S2-3
Load Capacity Data Sheet — Divisions 1 and 5

Division 1		Division 5	
NCA All	G	HAA-NCA All [Note (1)]	G
NCA-1250	W	NCA-1250	W
NCA-2140	W	NCA-2140	W
NCA-3550	W	NCA-3550	W
NF All	G	HFA-NF All [Note (2)]	G
NF-3100	W	NF-3100	W
NF-3200	W	NF-3200	W
NF-3300	W [Note (3)]	NF-3300	W [Note (3)]
NF-3400	W	NF-3400	W
NF-3500	W [Note (3)]	NF-3500	W [Note (3)]
NF-3600	W [Note (3)]	NF-3600	W [Note (3)]
Mandatory Appendix I	G	Mandatory Appendix I	G
Mandatory Appendix II	G [Note (3)]	Mandatory Appendix II	G [Note (3)]
II-1220	W [Note (3)]	II-1220	W [Note (3)]
II-1430	W [Note (3)]	II-1430	W [Note (3)]
Nonmandatory Appendix F	G	Nonmandatory Appendix F	G
F-1321	W	F-1321	W [Note (3)]
F-1370	W	F-1370	W [Note (3)]

Legend:
G = General Knowledge W = Working Knowledge

NOTES:
(1) Subsection HA, Subpart A references Subsection NCA for general requirements.
(2) Subsection HF, Subpart A references Subsection NF for rules.
(3) As applicable.

Table S2-4
Fabrication Specification — Division 3

Division 3	
WA-1000	W
WA-2000	W
WA-3100	W
WA-3300	W
WA-3400	W

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Table S2-4
Fabrication Specification — Division 3 (Cont'd)

Division 3	
WA-3800	W
WA-4000	W
WA-5000	G
WA-7000	G
WA-8000	W
WX-1000	W
WX-2000	W
WX-3000	W
WX-4000	W
WX-5000	W
WX-6000	W

Legend:
G = General Knowledge
W = Working Knowledge
WX = WB/WC/WD, as applicable

Table S2-5
Overpressure Protection Report — Divisions 1, 2, and 5

Division 1		Division 2		Division 5	
NCA-1000	G	NCA-1000	G	HAA-NCA-1000 [Note (1)]	G
NCA-2000	W	NCA-2000	W	HAA-NCA-2000	W
NCA-3100	G	NCA-3100	G	NCA-3100	G
NCA-3200	G	NCA-3200	G	NCA-3200	G
NCA-3500	G	NCA-3500	G	NCA-3500	G
NCA-3600	G	NCA-3600	G	NCA-3600	G
NCA-4000	G	NCA-4000	G	NCA-4000	G
NCA-7000	G	NCA-7000	G	HAA-NCA-7000	G
NX-1000	G	CC-1000	G	HX-1000	G
NX-3110	W	CC-3100	G	HX-3110	W
NX-3220	G	CC-3200	W	HX-3200	G
NX-3230	G	CC-6100	G	HX-3300	G
NX-3414	G	CC-6211	G	HX-3400	G
NX-3521	G	CC-7000	W	HX-3500	G
NX-3621	G	...	...	HX-3600	G
NX-6200	G	...	...	HX-6000	G
NX-6300	G	...	...	HX-7000	W
NX-7000	W	...	...	...	...

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Table S2-5
Overpressure Protection Report — Divisions 1, 2, and 5 (Cont'd)

Legend:

G = General Knowledge
HX = HB/HC, as applicable (including Subparts A and B), as well as references to Subsections NB and NC rules, respectively
NX = NB/NC/ND/NE, as applicable
W = Working Knowledge

NOTE:
(1) Subsection HA, Subpart A references Subsection NCA for general requirements.

Table S2-6
Construction Specification, Design Drawings, and Design Report — Divisions 2 and 5 (Nonmetallic CSS)

Division 2				Division 5			
Article/ Subarticle	Construction Specification	Design Drawings	Design Report	Article/ Subarticle	Construction Specification	Design Drawings	Design Report
NCA-1000	W	W	W	HAB-1000	W	W	W
NCA-2000	W	W	W	HAB-2000	W	W	W
NCA-3100	W	W	W	HAB-3000	W	W	W
NCA-3200	W	W	W	HAB-4000	W	G	W
NCA-3300	W	W	W	HAB-5000	G	G	G
NCA-3400	W	W	W	HAB-7000	G	G	G
NCA-3500	G	G	G	HAB-8000	G	G	G
NCA-3600	G	G	G	...	...	...	...
NCA-3700	G	G	G	HHA-1000	W	W	W
NCA-3800	W	G	G	HHA-2000	W	G	W
NCA-3900	W	G	G	HHA-3000	W	W	W
NCA-4000	W	G	W	HHA-4000	W	W	W
NCA-5000	G	G	G	HHA-5000	W	W	W
NCA-7000	W	W	W	HHA-8000	W	G	G
NCA-8000	G	G	G	...	...	...	...
CC-1000	W	W	W	...	...	...	...
CC-2000	W	G	W	...	...	...	...
CC-3100	W	W	W	...	...	...	...
CC-3200	G	G	W	...	...	...	...
CC-3300	G	W	W	...	...	...	...
CC-3400	G	W	W	...	...	...	...
CC-3500	W	W	W	...	...	...	...
CC-3600	W	W	W	...	...	...	...
CC-3700	W	W	W	...	...	...	...
CC-3800	W	W	W	...	...	...	...
CC-4100	W	W	W	...	...	...	...
CC-4200	W	W	G	...	...	...	...
CC-4300	W	W	G	...	...	...	...
CC-4400	W	W	G	...	...	...	...
CC-4500	W	W	G	...	...	...	...
CC-5000	W	W	W	...	...	...	...
CC-6000	W	G	G	...	...	...	...

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Table S2-6
Construction Specification, Design Drawings, and Design Report — Divisions 2 and 5 (Nonmetallic CSS) (Cont'd)

Division 2				Division 5			
Article/ Subarticle	Construction Specification	Design Drawings	Design Report	Article/ Subarticle	Construction Specification	Design Drawings	Design Report
CC-7000	W	G	G	...	...	...	...
CC-8000	W	G	G	...	...	...	...

Legend:
CSS = Core Support Structures
G = General Knowledge
W = Working Knowledge

SUPPLEMENT 3 NONMANDATORY SAMPLE STATEMENTS

Form S3-1
Design Specification (Div. 1, 2, and 5)

CERTIFICATION

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of design and related nuclear facility requirements relative to this Design Specification, certify that to the best of my knowledge and belief it is correct and complete with respect to the Design and Service Conditions given and provides a complete basis for construction in accordance with NCA-3250 and other applicable requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division _____, _____ Edition with Addenda (if applicable) up to and including _____.

The Specification and Revision being certified is:

Certified by _____ P.E.

Registration No. _____ State* _____

Date _____

* or Province of Canada

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Form S3-2
Design Report

CERTIFICATION [1]

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of design and using the certified Design Specification and the drawings identified below as a basis for design, do hereby certify that to the best of my knowledge and belief the Design Report is complete and accurate and complies with the design requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division _____ , _____ Edition with Addenda (if applicable) up to and including _____ .

Design Specification and Revision:

Drawings and Revision:

Design Report and Revision:

Certified by _____ P.E.

Registration No. _____ State* _____

Date _____

[1] Similar statement may also be used for certification of Load Capacity Data Sheet when supplied in lieu of Design Report (NCA-3551).
* or Province of Canada

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Form S3-3
Overpressure Protection Report (Div. 1, 2, and 5)

CERTIFICATION

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of design and overpressure protection requirements, do hereby certify that to the best of my knowledge and belief the Overpressure Protection Report complies with the requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division _____ , _____ Edition with Addenda (if applicable) up to and including _____.

Overpressure Protection Report and Revision:

Design Specification and Revision:

Certified by _____ P.E.

Registration No._____ State*_____

Date _____

* or Province of Canada

Form S3-4
Design Specification (Div. 3)

CERTIFICATION

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of Division 3 design requirements relative to this Design Specification, certify that to the best of my knowledge and belief it is correct and complete with respect to the Design and Operating Conditions given and provides a complete basis for construction in accordance with WA-3351 and other applicable requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division 3, _____ Edition with Addenda (if applicable) up to and including _____.

The Specification and Revision being certified is:

Certified by _____ P.E.

Registration No._____ State*_____

Date _____

* or Province of Canada

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Form S3-5
Fabrication Specification (Div. 3)

CERTIFICATION

I, the undersigned, being a Registered Professional Engineer competent in the applicable field of Division 3 fabrication requirements relative to this Fabrication Specification, certify that to the best of my knowledge and belief it is correct and complete with respect to the Design and Operating Conditions given and provides a complete basis for construction in accordance with WA-3361 and other applicable requirements of the ASME Boiler and Pressure Vessel Code, Section III, Division 3, _____ Edition with Addenda (if applicable) up to and including _____.

Design Specification and Revision:

Fabrication Specification and Revision:

Certified by _____ P.E.

Registration No. _____ State* _____

Date _____

* or Province of Canada

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MANDATORY APPENDIX XXIV
STANDARD UNITS FOR USE IN EQUATIONS

Table XXIV-1000
Standard Units for Use in Equations

Quantity	U.S. Customary Units	SI Units
Linear dimensions (e.g., length, height, thickness, radius, diameter)	inches (in.)	millimeters (mm)
Area	square inches ($in.^2$)	square millimeters (mm^2)
Volume	cubic inches ($in.^3$)	cubic millimeters (mm^3)
Section modulus	cubic inches ($in.^3$)	cubic millimeters (mm^3)
Moment of inertia of section	$inches^4$ ($in.^4$)	$millimeters^4$ (mm^4)
Mass (weight)	pounds mass (lbm)	kilograms (kg)
Force (load)	pounds force (lbf)	newtons (N)
Bending moment	inch-pounds (in.-lb)	newton-millimeters (N·mm)
Pressure, stress, stress intensity, and modulus of elasticity	pounds per square inch (psi)	megapascals (MPa)
Energy (e.g., Charpy impact values)	foot-pounds (ft-lb)	joules (J)
Temperature	degrees Fahrenheit (°F)	degrees Celsius (°C)
Absolute temperature	Rankine (°R)	kelvin (K)
Fracture toughness	ksi square root inches ($ksi\sqrt{in.}$)	MPa square root meters ($MPa\sqrt{m}$)
Angle	degrees or radians	degrees or radians
Boiler capacity	Btu/hr	watts (W)

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MANDATORY APPENDIX XXV
ASME-PROVIDED MATERIAL STRESS–STRAIN DATA

ARTICLE XXV-1000
INTRODUCTION

XXV-1100 STRESS–STRAIN DATA

It is recognized that ASME-specified material property data would make implementation of the strain-based acceptance criteria easier for many users. However, at this time, the ASME true stress–strain curves (reflecting minimum yield and ultimate tensile strength values) and the true uniform and fracture strain limits (reflecting a 98% exceedance probability) associated with the strain-based acceptance criteria of Nonmandatory Appendix FF are still under development. Until these data become available, the user shall develop the necessary material data based on tensile testing (see Nonmandatory Appendix EE, EE-1222), and their use shall be justified in the final Design Report.

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(15)

MANDATORY APPENDIX XXVI
RULES FOR CONSTRUCTION OF CLASS 3 BURIED POLYETHYLENE PRESSURE PIPING

ARTICLE XXVI-1000
GENERAL REQUIREMENTS

XXVI-1100 SCOPE

(a) This Appendix contains rules for the construction of Class 3 polyethylene pressure piping systems. The scope is limited to buried portions of Class 3 service water or buried portions of Class 3 cooling water systems, consisting of PE4710 High Density Polyethylene (HDPE) materials at maximum Design and Service Levels A, B, and C temperatures of 140°F (60°C), and a maximum temperature of 176°F (80°C) for Service Level D, with temperatures not exceeding those for which allowable stresses are provided in this Appendix.

(b) Terms relating to polyethylene as used in this Appendix are defined in Article XXVI-9000.

(c) All applicable requirements of Subsection ND shall be met unless modified by this Appendix.

XXVI-1200 QUALIFICATION OF POLYETHYLENE MATERIAL ORGANIZATIONS

The polyethylene material shall be procured in accordance with the requirements of NCA-3970 and this Appendix.

XXVI-1300 CERTIFICATE HOLDER RESPONSIBILITIES

(a) The Certificate Holder shall comply with the requirements of NCA-3970.

(b) The responsible Certificate Holder shall assure that the material complies with the Design Specification and this Appendix.

(c) A Certificate Holder may furnish material when stated in the scope of his certificate. In this case, a Quality System Certificate is not required, nor is the user of the material required to survey, qualify, or audit such a Certificate Holder.

(d) The Certificate Holder shall be responsible for surveying, qualification, and auditing of the Polyethylene Material Organization in accordance with NCA-3970.

(e) The survey and audit of the Polyethylene Material Organization shall establish that the Quality System Program conforms to the Certificate Holder's quality program requirements.

(f) Satisfactory completion of the survey and audit shall allow the Polyethylene Material Organization to supply material to the Certificate Holder for a period of 3 yr. After the 3-yr period, an audit shall be performed to ensure continued program maintenance.

(g) The Certificate Holder shall perform any of the functions specified by his respective Quality Assurance Program that are not performed by the Polyethylene Material Organization. It may elect to perform any other quality program functions, which would normally be the responsibility of the Polyethylene Material Organization. These functions shall be clearly defined in the Certificate Holder's Quality Assurance Program.

(h) The Certificate Holder shall make all necessary provisions so that his Authorized Inspection Agency can perform the inspections necessary to comply with this Appendix.

(i) In accordance with NCA-8120(b), a Certificate of Authorization may be issued by the Society to an organization certifying joining by fusing in accordance with this Appendix.

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ARTICLE XXVI-2000
MATERIALS

XXVI-2100 GENERAL REQUIREMENTS FOR MATERIALS

XXVI-2110 SCOPE

All polyethylene material shall conform to the requirements of this Article.

(a) Polyethylene material shall be selected from specifications listed in Supplement XXVI-I and shall be PE4710 HDPE with material properties as specified in XXVI-2200.

(b) All metallic materials shall conform to the requirements of Article ND-2000.

XXVI-2120 DETERIORATION OF MATERIAL IN SERVICE

Consideration of deterioration of material during service is generally outside the scope of this Appendix. It shall be the responsibility of the Certificate Holder to select material suitable for the conditions stated in the Design Specifications, with specific attention being given to the effects of service conditions on the properties of the material.

XXVI-2200 POLYETHYLENE COMPOUND AND MATERIAL REQUIREMENTS

XXVI-2210 GENERAL REQUIREMENTS

(a) Natural compound, pigment concentrate compound, and polyethylene compound and material shall conform to the requirements of this Article.

(b) Conformance with ASTM Standards referenced in Supplement XXVI-I and herein shall be limited as specified in this Article. In the event of conflict between a referenced standard and this Article, the requirements of this Article shall take precedence.

(c) Natural compound, pigment concentrate compound, and polyethylene compound and material shall be marked in accordance with the marking requirements in Article XXVI-8000 and the applicable ASTM Standard.

XXVI-2220 SPECIFIC COMPOUND REQUIREMENTS

XXVI-2221 Requirements for Certification of Polyethylene Compound

(a) General

(1) Polyethylene compound shall comply with, and be certified in accordance with, this Article and Table XXVI-2221-1.

(2) The required value for each property shall be as specified in Table XXVI-2221-1.

(3) The standard for determining the required value for properties shall be as specified in Table XXVI-2221-1.

(4) The test method for determination of the required value for the physical property shall be as specified in Table XXVI-2221-1.

(b) Polyethylene compound used for the manufacture of polyethylene material shall meet the requirements of the polyethylene compound manufacturer and Table XXVI-2221-1.

(c) Polyethylene compound shall be black except as provided in XXVI-2231(b).

(d) Polyethylene compound is the combination of natural compound and pigment concentrate compound as follows:

(1) When polyethylene compound is combined by the Polyethylene Compound Manufacturer, polyethylene compound is the polyethylene source material.

(2) When polyethylene compound is combined by the Polyethylene Material Manufacturer, natural compound and pigment concentrate compound are the polyethylene source materials.

(3) When polyethylene compound is combined by the Polyethylene Material Manufacturer, the Natural Compound Manufacturer shall provide the Polyethylene Material Manufacturer with a formulation that specifies the weight ratio (proportions) of natural and pigment concentrate compound and with processing equipment setting recommendations that produce polyethylene compound in accordance with Table XXVI-2221-1.

(e) Polyethylene compound shall have an independent listing that is published in PPI TR-4, Table I.A.13. The independent listing shall identify the following:

(1) a standard grade hydrostatic design basis (HDB) rating of at least 1,600 psi (11.03 MPa) at 73°F (23°C)

(2) a standard grade HDB rating of at least 1,000 psi (6.90 MPa) at 140°F (60°C)

(3) a hydrostatic design stress (HDS) rating of at least 1,000 psi (6.90 MPa) for water at 73°F (23°C)

(4) standard grade HDB ratings and HDS ratings shall be determined in accordance with PPI TR-3, Parts A, D, and F

(5) a material designation of PE4710 in accordance with PPI TR-4, Table I.A.13

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Table XXVI-2221-1
Certification Requirements for Polyethylene Compound

No.	Property, Units	Required Value	Requirement Standard	Test Method
1	Density, g/cm^3	0.956 to 0.968 with 2 to 3 wt.% carbon black 0.947 to 0.955 without carbon black or pigment	ASTM D3350	ASTM D1505, ASTM D792, or ASTM D4883
2	High load melt flow rate, g/10 min.	4 to 20	Polyethylene Compound Manufacturer Quality Program	ASTM D1238, Condition 190/21.6
3	Carbon black, wt.%	2 to 3	ASTM D3350	ASTM D4218 or ASTM D1603
4	Slow crack growth resistance (parent material), hr	>2,000	ASTM D3350	ASTM F1473 at 2.4 MPa and 80°C in air
5	Thermal stability, °F (°C)	>428 (>220)	ASTM D3350	ASTM D3350
6	Tensile strength at yield, psi (MPa)	≥3,500 (≥24.14)	ASTM D3350	ASTM D638, Type IV at 50 mm/min. (2 in./min.)
7	Tensile elongation at break, %	≥400	ASTM D3350	ASTM D638, Type IV at 50 mm/min. (2 in./min.)
8	HDB at 73°F (23°C), psi (MPa)	1,600 (11.03)	ASTM D2837, PPI TR-3, and PPI TR-4	ASTM D2837, PPI TR-3, and PPI TR-4
9	HDB at 140°F (60°C), psi (MPa)	1,000 (6.90)	ASTM D2837, PPI TR-3, and PPI TR-4	ASTM D2837, PPI TR-3, and PPI TR-4
10	HDS for water at 73°F (23°C), psi (MPa)	1,000 (6.90)	ASTM D2837, PPI TR-3, and PPI TR-4	ASTM D2837, PPI TR-3, and PPI TR-4
11	Thermoplastic pipe materials designation code	PE4710	Listed in PPI TR-4	N/A

GENERAL NOTE: Only SI units are provided in Table XXVI-2221-1 when the applicable ASTM Standards do not provide U.S. Customary units.

(6) the unique trade name or designation for the compound

(7) the Polyethylene Natural Compound Manufacturer

(f) The Polyethylene Material Manufacturer of polyethylene pipe shall have a dependent listing for black polyethylene compound that is published in PPI TR-4, Table I.A.13. The dependent listing shall identify the following:

(1) a standard grade HDB rating of at least 1,600 psi (11.03 MPa) at 73°F (23°C)

(2) a standard grade HDB rating of at least 1,000 psi (6.90 MPa) at 140°F (60°C)

(3) a HDS rating of at least 1,000 psi (6.90 MPa) for water at 73°F (23°C)

(4) standard grade HDB and HDS ratings in accordance with PPI TR-3, Parts A, D, and F

(5) a unique trade name or designation to the polyethylene compound that is published in PPI TR-4, Table I.A.13

(g) The Certificate of Analysis (C of A) Report shall identify the trade name or designation assigned to the polyethylene compound by the Polyethylene Compound Manufacturer that is published in PPI TR-4.

(h) The Certified Polyethylene Test Report (CPTR) shall identify the trade name for the polyethylene compound assigned by the Polyethylene Material Manufacturer that is published in PPI TR-4, Table I.A.13, and shall identify the following:

(1) the C of A Report trade names for the natural compound and the pigment concentrate compound, or

(2) the C of A Report trade name for the polyethylene compound

(i) If specified, color polyethylene compound shall contain color and ultraviolet (UV) stabilization in accordance with ASTM D3350 Code E. Color polyethylene compound color and UV stabilization duration requirements shall be specified in the Design Specification. Per XXVI-2231(b), color polyethylene compound shall be used only for optional color stripes on polyethylene material in the pipe product form.

XXVI-2222 Natural Compound

(a) Natural compound shall meet requirements specified by the Natural Compound Manufacturer.

(b) Natural compound shall be combined with pigment concentrate compound in accordance with XXVI-2221(d).

(c) The Natural Compound Manufacturer shall assign a unique trade name or designation to the natural compound.

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XXVI-2223 Pigment Concentrate Compound

(a) Black pigment concentrate compound shall meet requirements specified by the Natural Compound Manufacturer.

(b) Black pigment concentrate compound shall be combined with natural compound in accordance with XXVI-2221(d)(3).

(c) The Pigment Concentrate Compound Manufacturer shall assign a unique trade name or designation to the pigment concentrate compound.

(d) Color pigment concentrate compound shall be in accordance with XXVI-2231(b).

XXVI-2230 SPECIFIC MATERIAL REQUIREMENTS

(a) This subsubarticle identifies and provides the specific requirements applicable to the various product forms permitted by this Appendix.

(b) All fabrications produced by fusing shall be produced by a Certificate Holder, using fusing procedures and fusing machine operators trained and qualified in accordance with Section IX and Article XXVI-4000.

(c) All fused joints shall be examined in accordance with Article XXVI-5000.

(d) If a molding process is used, the Product Manufacturer shall certify that the material has been volumetrically examined to ensure that the material is void free. The Owner or his designee shall review and approve the examination technique used by the Product Manufacturer.

XXVI-2231 Polyethylene Material — Pipe

(a) Polyethylene pipe shall be manufactured in accordance with this Appendix and ASTM D3035 for sizes smaller than IPS 3[18] (DN 80) or ASTM F714 for sizes IPS 3 (DN 80) and larger. Elevated temperature sustained pressure test per ASTM D3035 or ASTM F714 shall be successfully completed at least every 6 months by the pipe manufacturer during manufacture of pipe supplied in accordance with these requirements.

(b) Pipe shall be black and manufactured by extrusion. With the exception of optional color stripes per this subarticle, black pipe shall contain 2 wt% to 3 wt% carbon black that is well dispersed through the pipe wall. Samples shall be taken from pipe and tested in accordance with ASTM D1603 or ASTM D4218.

(1) Optional color stripes that are coextruded into the pipe outside surface are acceptable. The depth of optional color stripes into the pipe outside surface shall not infringe upon minimum wall thickness, t_{Design}. Color stripes shall not project above the pipe outside surface and shall not be covered in whole or in part by black pipe material.

(2) Where natural compound and pigment concentrate compound are combined by the Polyethylene Material Manufacturer, the Polyethylene Material Manufacturer shall use the same natural compound with black pigment concentrate compound and with color pigment concentrate compound if optional color stripes are coextruded into the pipe outside surface.

(3) Where black polyethylene compound and color polyethylene compound are used to extrude pipe with optional color stripes, coextruded into the outside surface, the black polyethylene compound and color polyethylene compound shall use the same natural compound.

(c) Pipe print line marking shall be applied in accordance with ASTM D3035 or ASTM F714 during extrusion using heated indentation.

(d) Prior to shipment of the pipe, testing for fusibility of the material shall be performed in accordance with XXVI-2300, unless the Owner or his designee elects to perform the testing.

XXVI-2232 Polyethylene Material — Flange Adapter

(a) Flange adapters shall be fabricated from pipe by machining or by a molding process using polyethylene materials meeting the requirements of XXVI-2200.

(b) The configuration shall be in accordance with XXVI-4520.

(c) The pressure rating, PR, shall be determined in accordance with XXVI-3132.

(d) The dimensions and surface appearance shall be verified and the pressure rating shall be confirmed by testing in accordance with ASTM D3261 or ASTM F2206.

(e) Molded flange adapters shall be certified as void free [refer to XXVI-2230(d)].

(f) Prior to shipment of flange adapters, testing for fusibility of the material shall be performed in accordance with XXVI-2300, unless the Owner or his designee elects to perform the testing.

XXVI-2232.1 Polyethylene Material — Machined Flange Adapter. The polyethylene material used to fabricate flange adapters that are machined from pipe shall meet the requirements of XXVI-2231.

XXVI-2232.2 Polyethylene Material — Molded Flange Adapter. The polyethylene compound used to manufacture molded flange adapters shall meet the requirements of XXVI-2221.

XXVI-2233 Polyethylene Material — Mitered Elbows

(a) The polyethylene material used for mitered elbows shall be pipe meeting the requirements of XXVI-2231.

(b) The configuration of the mitered elbow shall meet the dimensional requirements of the specifications listed in Supplement XXVI-I and the additional requirements of XXVI-3132.1.

(c) Prior to shipment of mitered elbows, testing for fusibility of the material shall be performed in accordance with XXVI-2300, unless the Owner or his designee elects to perform the testing.

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(d) The Data Report Form NM(PE)-2 (Supplement XXVI-III) shall be used for this product form.

XXVI-2234 Polyethylene Material — Thrust Collar

(a) The configuration shall meet the dimensional requirements of Figure XXVI-2234-1.

(b) The Dimension Ratio (DR) shall be equal to or less than that of the attached straight pipe and shall be designed for joining by fusion to the piping.

(c) The pressure rating shall be equal to or greater than the Design Pressure. The dimensions and surface appearance shall be verified and the pressure rating shall be confirmed by testing in accordance with ASTM D3261 or ASTM F2206.

(d) Fabrication fusing shall meet the requirements of Article XXVI-4000.

(e) Prior to shipment of thrust collars, testing for fusibility of the material shall be performed in accordance with XXVI-2300, unless the Owner or his designee elects to perform the testing.

(f) The Data Report Form NM(PE)-2 (Supplement XXVI-III) shall be used for thrust collars fabricated by fusing.

XXVI-2234.1 Polyethylene Material — Fabricated Thrust Collars. The polyethylene pipe material used to fabricate thrust collars by fusing shall meet the requirements of XXVI-2231.

XXVI-2234.2 Polyethylene Material — Machined Thrust Collars. The polyethylene pipe material used to fabricate thrust collars by machining shall meet the requirements of XXVI-2231.

XXVI-2234.3 Polyethylene Material — Molded Thrust Collars.

(a) The polyethylene compound used to manufacture molded thrust collars shall meet the requirements of XXVI-2221.

(b) Molded thrust collars shall be certified as void free by the manufacturer [refer to XXVI-2230(d)].

XXVI-2235 Polyethylene Material — Concentric Monolithic Reducers

Machined or molded concentric monolithic reducers shall be permitted and shall comply with the following requirements:

(a) The configuration shall meet the dimensional requirements of the specifications listed in Supplement XXVI-I.

(b) The material used to fabricate reducers that are machined from pipe shall meet the requirements of XXVI-2231.

(c) The polyethylene compound used to manufacture molded reducers shall meet the requirements of XXVI-2221.

(d) Molded monolithic reducers shall be certified as void free [refer to XXVI-2230(d)].

(e) The pressure rating shall be equal to or greater than the Design Pressure. The dimensions and surface appearance shall be verified and the pressure rating shall be confirmed by testing in accordance with ASTM D3261 or ASTM F2206.

(f) Prior to shipment, testing for the fusibility of the material shall be performed in accordance with XXVI-2300, unless the Owner or his designee elects to perform the testing.

XXVI-2236 Polyethylene Material — Fabricated Fittings (Other)

Fabricated equal outlet mitered tees, equal outlet mitered lateral wyes, and concentric fabricated reducers shall be permitted and shall be in accordance with the following requirements:

(a) The fitting shall be fabricated from polyethylene pipe with the same or lower DR than the attached pipe and shall meet the requirements of XXVI-2231.

(b) The configuration shall meet the dimensional requirements of the specifications listed in Supplement XXVI-I.

(c) The pressure rating shall be equal to or greater than the system Design Pressure. The dimensions and surface appearance shall be verified and the pressure rating shall be confirmed by testing in accordance with ASTM F2206.

(d) All fabrication fusing shall meet the requirements of Article XXVI-4000.

(e) Prior to shipment, testing for fusibility of the material shall be performed in accordance with XXVI-2300, unless the Owner or his designee elects to perform the testing.

(f) The Data Report Form NM(PE)-2 (Supplement XXVI-III) shall be used for these product forms.

XXVI-2237 Polyethylene Material — Molded Fittings (Other)

Injection molded elbows and equal outlet tees shall be permitted and shall be in accordance with the following requirements:

(a) The polyethylene compound shall meet the requirements of XXVI-2221.

(b) The configuration shall meet the dimensional requirements of the specifications listed in Supplement XXVI-I.

(c) The fittings shall be certified as void free [refer to XXVI-2230(d)].

(d) The pressure rating shall be equal to or greater than the system Design Pressure. The dimensions and surface appearance shall be verified and the pressure rating shall be confirmed by testing in accordance with ASTM D3261.

(e) Prior to shipment of the pipe, testing for fusibility of the material shall be performed in accordance with XXVI-2300, unless the Owner or his designee elects to perform the testing.

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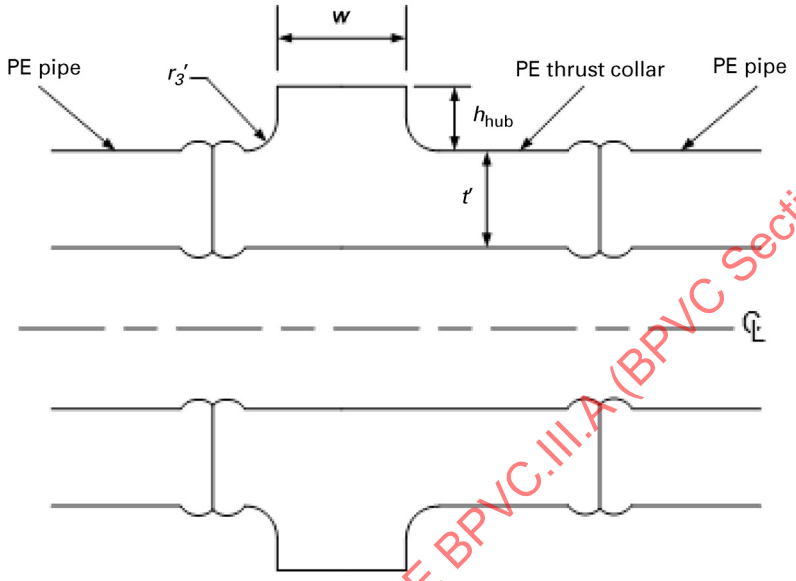
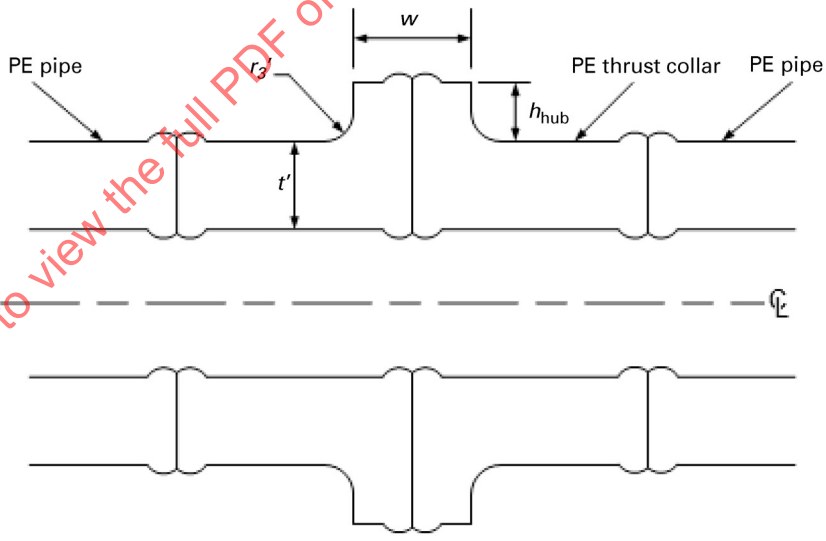
Figure XXVI-2234-1
Thrust Collars



(a) Fabricated Polyethylene Thrust Collar – Side Fused



(b) Fabricated Polyethylene Thrust Collar – Center Fused

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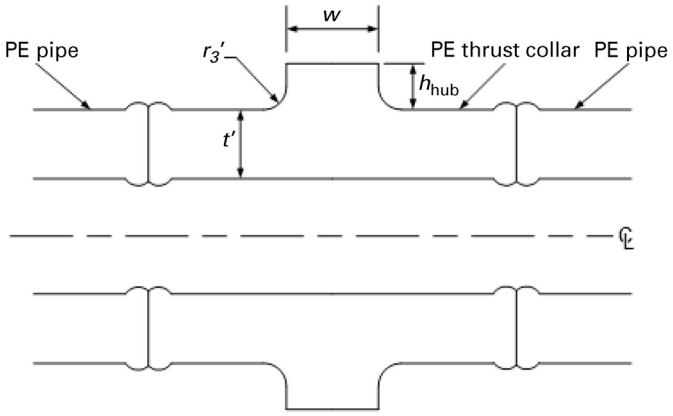
Figure XXVI-2234-1
Thrust Collars (Cont'd)



(c) Machined or Molded Polyethylene Thrust Collar

$r_3'_{min}$ in. (mm)	IPS (DN)
1/8 (3)	≤1 1/4 (32)
3/16 (5)	1 1/2 (38)
1/4 (6)	2 to 3 (50 to 75)
3/8 (10)	4 to 12 (100 to 300)
1/2 (13)	14 to 42 (360 to 1,070)
3/4 (19)	48 to 65 (1,200 to 1,650)

GENERAL NOTE: h_{hub} (minimum) = $0.5t'$; h_{hub} (maximum) = $0.8t'$; W_{min} = $1t'$.

XXVI-2300 POLYETHYLENE MATERIAL FUSING VERIFICATION TESTING

XXVI-2310 GENERAL

(a) All polyethylene material product forms shall be tested for compliance with the Standard Fusing Procedure Specification (SFPS) of Section IX, QF-220, and as specified herein.

(b) The polyethylene materials tested shall be from the same Polyethylene Material Manufacturer's manufacturing facility using the same method of manufacture as the polyethylene materials to be used in production.

(c) Joint fusibility testing shall include each lot of polyethylene source material to be used in production supplied by the same or different polyethylene Material Manufacturers in all combinations of suppliers and in all diameters and thicknesses to be fused in production.

(d) All joint testing shall use the same fusing machine make and carriage model to be used for joining the materials in production [see XXVI-4321(c)].

(e) Joint fusibility testing shall be performed by the Polyethylene Material Manufacturer unless the Owner or his designee elects to perform the testing.

(f) Fusibility testing results shall be included with the CPTR.

XXVI-2320 FUSING PARAMETERS FOR TESTING

One joint shall be made at each of the following conditions to verify fusibility at the pressure/temperature extremes of the fusing procedure:

(a) interfacial pressure of 90 psi (620 kPa) minimum and heater temperature of 450°F (232°C) minimum; heater removal (dwell) time kept to a minimum, not to exceed the specified maximum

(b) interfacial pressure of 60 psi (410 kPa) maximum and heater temperature of 450°F (232°C) minimum; heater removal (dwell) time kept to a minimum, not to exceed the specified maximum

(c) interfacial pressure of 90 psi (620 kPa) minimum and heater temperature of 400°F (204°C) maximum; heater removal (dwell) time at the maximum permitted

(d) interfacial pressure of 60 psi (410 kPa) maximum and heater temperature of 400°F (204°C) maximum; heater removal (dwell) time at the maximum permitted

XXVI-2330 TESTING

Testing of the joints shall be in accordance with Section IX, QF-144.

(a) For pipe and fitting specimens smaller than IPS 4 (DN 100), no fewer than two specimens shall be removed from fused pipe test coupons at intervals of approximately 180 deg apart.

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(b) For pipe and fitting specimens IPS 4 (DN 100) and larger, no fewer than four specimens shall be removed from fused pipe test coupons at intervals approximately 90 deg apart.

XXVI-2400 REPAIR OF MATERIAL

Repair of manufactured or fabricated polyethylene pipe and fittings shall not be permitted. Polyethylene pipe or fittings with gouges, cuts, or other surface conditions that exceed the following requirements shall be rejected and scrapped:

(a) For pipe IPS 4 (DN 100) and smaller in nominal diameter, any indentation greater than 5% of $t_{fab\ min}$ or any indentation resulting in a wall thickness of less than $t_{fab\ min}$ shall be unacceptable.

(b) For pipe larger than IPS 4 (DN 100) in nominal diameter, any indentation greater than 0.040 in. (1.0 mm) or any indentation resulting in a wall thickness of less than $t_{fab\ min}$ shall be unacceptable.

XXVI-2500 GENERAL REQUIREMENTS FOR QUALITY TESTING AND DOCUMENTATION

(a) Through his Quality Systems Program, the Polyethylene Source Material Manufacturer shall ensure that polyethylene compound is certified in accordance with XXVI-2221.

(b) Acceptance of individual lots of polyethylene source material shall be in accordance with XXVI-2510.

XXVI-2510 CERTIFICATE OF ANALYSIS (C OF A) REPORT

The following paragraphs contain requirements for the C of A Report and related traceability documentation.

XXVI-2511 Polyethylene Compound

(a) Polyethylene compound shall be qualified per XXVI-2221.

(b) The Polyethylene Compound Manufacturer shall test polyethylene compound in accordance with Table XXVI-2511-1 and shall provide a C of A Report and related traceability documentation to the purchaser of the lot.

(c) The C of A Report shall include the certified test results in accordance with Table XXVI-2511-1.

(d) The C of A Report and related traceability documentation shall include the following information:

(1) the name of the Polyethylene Compound Manufacturer

(2) the manufacturing location

(3) an identification code that is unique and traceable to the specific lot

(4) the Polyethylene Compound Manufacturer's trade name for the polyethylene compound as published in PPI TR-4

(5) the shipping method or type of container(s) for the lot, such as railcar or boxes, and additional information, such as a railcar number if shipped by rail or the name of the commercial carrier and number of boxes if shipped by commercial carrier

(6) the lot weight of polyethylene compound

(7) the date of shipment

(8) other information that identifies the purchaser (customer), purchaser order, purchaser contact, purchaser delivery location, and contact information for the Polyethylene Compound Manufacturer

(9) if applicable, the Quality System Program statement information per NCA-3974.4

XXVI-2512 Natural Compound

(a) The Natural Compound Manufacturer shall test natural compound in accordance with Table XXVI-2512-1. The Natural Compound Manufacturer shall provide a C of A Report and related traceability documentation to the purchaser of the lot.

(b) The C of A Report shall include the certified test results in accordance with Table XXVI-2512-1.

(c) The C of A Report and related traceability documentation shall include the following information:

(1) the name of the Natural Compound Manufacturer

(2) the manufacturing location

(3) an identification code that is unique and traceable to the specific lot

(4) the Natural Compound Manufacturer's trade name for the natural compound

(5) the shipping method or type of container(s) for the lot, such as railcar or boxes, and additional information, such as a railcar number if shipped by rail or the name of the commercial carrier and number of boxes if shipped by commercial carrier

(6) the lot weight of natural compound

(7) the date of shipment

(8) other information that identifies the purchaser (customer), purchaser order, purchaser contact, delivery location, and contact information for the Natural Compound Manufacturer

(9) if applicable, the Quality System Program statement information per NCA-3974.4

XXVI-2513 Pigment Concentrate Compound

(a) The Pigment Concentrate Compound Manufacturer shall test pigment concentrate compound in accordance with Table XXVI-2513-1. The Pigment Concentrate Compound Manufacturer shall provide a C of A Report and related traceability documentation to the purchaser of the lot.

(b) The C of A Report shall include the certified test results for the lot in accordance with Table XXVI-2513-1.

(c) The C of A Report or related traceability documentation shall include the following information:

(1) the name of the Pigment Concentrate Compound Manufacturer

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Table XXVI-2511-1
Minimum Quality Testing Requirements for Polyethylene Compound Lots

No.	Test	Test Standard	Test Frequency	Test Timing	C of A Reports Test Results
1	High load melt flow rate, Condition 190/21.6, g/10 min.	ASTM D1238 and Table XXVI-2221-1	Once per lot	Before lot shipment	Yes
2	Density	ASTM D792, ASTM D1505, or ASTM D4883 and Table XXVI-2221-1	Once per lot	Before lot shipment	Yes
3	Tensile strength at yield and tensile elongation at break	ASTM D638 and Table XXVI-2221-1	Once per lot	Before lot shipment	Yes
4	Thermal stability	ASTM D3350 and Table XXVI-2221-1	Once per lot	Before lot shipment	Yes
5	Carbon black content	ASTM D1603 or ASTM D4218 and Table XXVI-2221-1	Once per lot	Before lot shipment	Yes

(2) the manufacturing location

(3) an identification code that is unique and traceable to the specific lot

(4) the Pigment Concentrate Compound Manufacturer's trade name for the pigment concentrate compound

(5) the shipping method or type of container(s) for the lot, such as railcar or boxes, and additional information, such as a railcar number if shipped by rail or the name of the commercial carrier and number of boxes if shipped by commercial carrier

(6) the lot weight of pigment concentrate compound

(7) the date of shipment

(8) other information that identifies the purchaser (customer), purchaser order, purchaser contact, delivery location, and contact information for the Pigment Concentrate Compound Manufacturer

(9) if applicable, the Quality System Program statement information per NCA-3974.4

XXVI-2520 CPTR FOR POLYETHYLENE MATERIAL — PIPE

(a) The Polyethylene Material Manufacturer — Pipe:

(1) shall certify the C of A Report values by testing a sample from the polyethylene source material lot in accordance with Table XXVI-2520(a)-1

(2) shall not use the material when certification testing does not verify C of A Report values

(3) shall test pipe in accordance with Table XXVI-2520(a)-2 and shall provide a CPTR and the Compound C of A Report(s) to the purchaser

(b) The CPTR shall include the following per lot:

(1) certified test results for the lot in accordance with Tables XXVI-2520(a)-1 and XXVI-2520(a)-2

(2) the name of the Polyethylene Material Manufacturer

(3) the manufacturing location

(4) an identification code that is unique and traceable to the specific lot

(5) the ASTM Standard for pipe manufacture

(6) the specification for the polyethylene compound

(7) the shipping method and the name of the commercial carrier

(8) the lot length

(9) the date of shipment

Table XXVI-2512-1
Minimum Quality Testing Requirements for Natural Compound Lots

No.	Test	Test Standard	Test Frequency	Test Timing	C of A Reports Test Results
1	High load melt flow rate, Condition 190/21.6, g/10 min.	ASTM D1238	Once per lot	Before lot shipment	Yes
2	Density	ASTM D792, ASTM D1505, or ASTM D4883 and Table XXVI-2221-1	Once per lot	Before lot shipment	Yes
3	Tensile strength at yield and tensile elongation at break	ASTM D638	Once per lot	Before lot shipment	Yes
4	Thermal stability	ASTM D3350	Once per lot	Before shipment	Yes

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Table XXVI-2513-1
Testing Requirements for Pigment Concentrate Compound Lots

No.	Test	Test Standard	Test Frequency	Test Timing	C of A Reports Test Results
1	Carbon black content (black only)	ASTM D1603 or ASTM D4218	Every 24 hr during lot production	Every 24 hr after acceptable product has been produced for given production lot	Yes
2	Color and UV stabilizer (color only)	ASTM D3350	Every 24 hr during lot production	Every 24 hr after acceptable product has been produced for given production lot	Yes

(10) other information that identifies the purchaser (customer), purchaser order, purchaser contact, delivery location, and contact information for the Polyethylene Material Manufacturer

(11) a certification that the polyethylene material was made from only virgin polyethylene source material and that no scrap or reground polyethylene material was used (see NCA-3974.3)

(12) certification of slow crack growth resistance (greater than 2,000 hr per ASTM F1473 completed on a compression molded plaque at 2.4 MPa and 80°C in air per Table XXVI-2221-1 for the polyethylene compound)

(13) results of fusibility testing performed in accordance with XXVI-2300

(14) the Quality System Program statement information per NCA-3974.4

XXVI-2530 MINIMUM QUALITY TESTING REQUIREMENTS FOR POLYETHYLENE MATERIAL — MOLDED PRODUCTS

(a) The Polyethylene Material Manufacturer — Molded Products:

(1) shall certify the C of A Report values by testing a sample from the polyethylene source material lot in accordance with Table XXVI-2512-1

(2) shall not use the polyethylene source material when certification testing does not verify the C of A Report values

(3) shall examine the molded product in accordance with the fabricator procedure to determine product is void free [refer to XXVI-2230(d)] and shall provide a CPTR and the Compound C of A Report(s) to the purchaser

(b) The CPTR shall include the following per lot:

Table XXVI-2520(a)-1
Minimum Quality Testing Requirements for Polyethylene Source Material

No.	Test	Test Standard	Test Frequency	CPTR Reports Test Results
1	High load melt flow rate, Condition 190/21.6, g/10 min.	ASTM D1238	Once per lot upon receipt at the processing facility	Yes
2	Density	ASTM D792 or ASTM D1505 and Table XXVI-2221-1	Once per lot upon receipt at the processing facility	Yes
3	Carbon black concentration percentage for black polyethylene compound or black pigment concentrate compound	ASTM D1603 or ASTM D4218	Once per lot upon receipt at the processing facility	Yes
4	Slow crack growth resistance, hr [Note (1)]	Greater than 2,000 hr per ASTM F1473 completed on a compression molded plaque at 2.4 MPa and 80°C in air per Table XXVI-2221-1	Once per lot prior to shipment of polyethylene material	Yes
5	Thermal stability [Note (1)]	Greater than 428°F (220°C) ASTM D3350 and Table XXVI-2221-1	Once per lot prior to shipment of polyethylene material	Yes

NOTE:
(1) In no case shall any individual test result, used to establish this value in accordance with the reference industry standards, be less than the minimum required value listed in this Table.

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Table XXVI-2520(a)-2
Minimum Quality Testing Requirements for Polyethylene Material — Pipe

No.	Test/Requirement	Manufacturing Standard/ Acceptance Criteria	Test Method	Test Frequency	CPTR Reports Test Results
1	Workmanship	<3 in. IPS (DN 80) ASTM D3035; ≥3 in. IPS (DN 80) ASTM F714	N/A	Hourly or once per length, whichever is less frequent during ongoing production	Yes
2	Outside diameter	<3 in. IPS (DN 80) ASTM D3035; ≥3 in. IPS (DN 80) ASTM F714	ASTM D2122 [Note (1)]	Hourly or once per length, whichever is less frequent during ongoing production	Yes
3	Toe-in	<3 in. IPS (DN 80) ASTM D3035; ≥3 in. IPS (DN 80) ASTM F714	ASTM D2122 [Note (1)]	Once per shift during ongoing production	Yes
4	Wall thickness	<3 in. IPS (DN 80) ASTM D3035; ≥3 in. IPS (DN 80) ASTM F714	ASTM D2122 [Note (1)]	Hourly or once per length, whichever is less frequent during ongoing production	Yes
5	Short-term strength	<3 in. IPS (DN 80) ASTM D3035; ≥3 in. IPS (DN 80) ASTM F714	ASTM D1599 or ASTM D2290	At the beginning of production and weekly thereafter during ongoing production	Yes
6	Carbon black content	XXVI-2231(b)	ASTM D1603 or ASTM D4218	At the beginning of production and weekly thereafter during ongoing production	Yes

NOTE:
(1) Sample conditioning must be as specified in ASTM D3035 or ASTM F714.

(1) certified test results for the lot in accordance with XXVI-2520(a)

(2) the name of the Polyethylene Material Manufacturer

(3) the manufacturing location

(4) an identification code that is unique and traceable to the specific lot

(5) the specification for the polyethylene compound

(6) the shipping method and name of the commercial carrier

(7) the lot quantity in pieces

(8) the date of shipment

(9) other information that identifies the purchaser (customer), purchaser order, purchaser contact delivery location, and contact information for the polyethylene material manufacturer

(10) a certification that the polyethylene material was made from only virgin polyethylene material and that no scrap or regrind polyethylene material was used (see NCA-3974.3)

(11) a certification that the flange adapter is void free [refer to XXVI-2230(d)]

(12) certification of slow crack growth resistance (greater than 2,000 hr per ASTM F1473 completed on a compression molded plaque at 2.4 MPa and 80°C in air per Table XXVI-2221-1 for the polyethylene compound)

(13) results of fusibility testing performed in accordance with XXVI-2300

(14) the Quality System Program statement information per NCA-3974.4

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ARTICLE XXVI-3000
DESIGN

XXVI-3100 SCOPE

The design rules of this Article are limited to buried polyethylene piping systems constructed of straight pipe, the piping items listed in XXVI-2200, fusion joints, and metal-to-polyethylene flanged connections. The maximum Design Temperature and Service Level A, B, or C temperatures shall be 140°F (60°C), and the maximum Service Level D temperature shall be 176°F (80°C). Temperatures shall not exceed those temperatures for which allowable stresses are provided in this Article. Polyethylene piping shall be permitted only for buried Class 3 service water or buried Class 3 cooling water systems.

XXVI-3110 NOMENCLATURE

A = cross-sectional area of pipe at the pipe section where the evaluation is conducted, in.^{2} (mm^2)

a = difference in thickness between pipe walls at a tapered transition joint, in. (mm)

A_s = shear area of thrust collar at the section where the evaluation is conducted, in.^{2} (mm^2)

b = total length of taper at a tapered transition joint, in. (mm)

B_d = trench width, ft (m)

B_1 = stress index, Table XXVI-3311-1

B_2 = stress index, Table XXVI-3311-1

B' = burial factor

c = the sum of mechanical allowances, installation allowance, erosion allowance, and other degradation allowance, in. (mm)

c' = length of counterbore at a tapered transition joint, in. (mm)

D = average outside diameter of pipe in accordance with ASTM F714 or ASTM D3035, in. (mm)

DR = dimension ratio of pipe = average outside diameter of the pipe divided by the minimum fabricated wall thickness = $D/t_{\text{fab min}}$

E_{pipe} = modulus of elasticity of pipe per Table XXVI-3210-3 or Table XXVI-3210-3M, psi (MPa)

E' = modulus of soil reaction, psi (MPa) (data is site specific)

E'_N = modulus of soil reaction of native soil around trench, psi (MPa) (data is site specific)

F_a = axial force due to the specified Design, Service Level A, B, C, or D applied mechanical loads, lb (N)

F_{aC} = axial force range due to thermal expansion, contraction, and/or the restraint of free end displacement, lb (N)

F_{aD} = axial force due to the nonrepeated anchor motion, lb (N)

F_{aE} = axial force range due to the combined effects of seismic wave passage, seismic soil movement, and building seismic anchor motion effects, lb (N)

F_{Alt} = equivalent maximum axial force range due to thermal expansion and contraction and/or the restraint of free end displacement, lb (N)

F_C = axial force due to fully constrained thermal contraction, lb (N)

F_E = axial force due to fully constrained thermal expansion, lb (N)

F_S = soil support factor, per Table XXVI-3210-2

f_o = ovality correction factor, per Table XXVI-3221.2-1

g = acceleration due to gravity, ft/sec^2 (m/s^2)

GSR = Geometric Shape Rating

H = height of ground cover, ft (m)

H_{gw} = height of groundwater above top of the pipe, ft (m)

h_{hub} = thickness of thrust collar hub, in. (mm)

i = stress intensification factor, per Table XXVI-3311-1

K = bedding factor

K' = Design and Service Level longitudinal stress factor from Table XXVI-3223-1

L = deflection lag factor

M = resultant bending moment due to the specified Design, Service Level A, B, C, or D applied mechanical loads, in.-lb (N·mm)

M_C = resultant moment range due to thermal expansion, contraction, and/or the restraint of free end displacement, in.-lb (N·mm)

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M_D = resultant moment due to the nonrepeated anchor motion, in.-lb (N·mm)

M_E = resultant moment range due to the combined effects of seismic wave passage, seismic soil movement, and building seismic anchor motion effects, in.-lb (N·mm)

N = number of equivalent full range temperature cycles

P = internal design gage pressure, plus pressure spikes due to transient events, psig (MPa gage)

P_a = Design or Service Level A, B, C, or D pressure, psig (MPa gage)

P_D = piping system internal Design Pressure at the specified Design Temperature, T_D, both being specified in the piping Design Specification, not including the consideration of pressure spikes due to transients, psig (MPa gage)

P_E = vertical soil pressure due to earth loads, lb/ft^2 (MPa)

P_{gw} = pressure due to groundwater above the top of the pipe, lb/ft^2 (MPa)

P_{hydro} = external hydrostatic pressure, equal to earth plus groundwater pressure plus surcharge load, psi (MPa)

P_L = vertical soil pressure due to surcharge loads, lb/ft^2 (MPa)

P_m = mitered elbow pressure rating, psig (MPa gage)

PR = fitting pressure rating, psig (MPa gage)

R = buoyancy reduction factor

r_1' = radius of curvature at the beginning of a tapered transition joint, in. (mm)

r_2' = radius of curvature at the end of a tapered transition joint, in. (mm)

r_3' = radius of curvature at the thrust collar hub, in. (mm)

S = allowable stress, per Table XXVI-3131-1(a) or Table XXVI-3131-1M(a) and Table XXVI-3131-1(b), psi (MPa)

S_A = allowable secondary stress range value as defined in XXVI-3133 and given in Table XXVI-3133-1 or Table XXVI-3133-1M, psi (MPa)

S_{comp} = allowable sidewall compression stress per Table XXVI-3220-1 or Table XXVI-3220-1M, psi (MPa)

T = temperature, °F (°C)

T_D = Design Temperature, °F (°C)

T_{ground} = temperature of soil around pipe, °F (°C)

T_{water} = temperature of water running through pipe, °F (°C)

t_{Design} = minimum required wall thickness, in. (mm)

t_{elbow} = minimum gore (mitered segment) wall thickness for fabricated elbows

$t_{\text{fab min}}$ = minimum fabricated wall thickness in accordance with ASTM D3035 or F714 (called minimum wall thickness in Table 9 of ASTM F714), in. (mm)

t_{min} = minimum wall thickness for pressure, in. (mm)

t' = wall thickness of thrust collar pipe section, in. (mm)

w = width of thrust collar hub, in. (mm)

W_P = weight of empty pipe per unit length, lb/ft (kg/m)

W_w = weight of water displaced by pipe, per unit length, lb/ft (kg/m)

Z = section modulus of pipe cross section at the pipe section where the moment is calculated, in.^{3} (mm^3)

α = coefficient of thermal expansion of pipe, 1/°F (1/°C)

ΔP = differential pressure due to negative internal pressure of pipe, psi (MPa)

ΔT = $T_{\text{water}} - T_{\text{ground}}$, °F (°C)

ΔT_{eq} = equivalent temperature rise, °F (°C)

$(\varepsilon_a)_{\text{Earthquake}}$ = strain in the pipe from earthquake wave computer analysis

$\varepsilon_{\text{soil}}$ = maximum soil strain due to seismic wave passage

ν = Poisson's ratio

Ω = change in diameter as a percentage of the original diameter, commonly called the change in ring diameter

Ω_{max} = maximum allowable change in diameter as a percentage of the original diameter, commonly called the change in ring diameter, per Table XXVI-3210-1

ρ_{dry} = density of dry soil, lb/ft^3 (kg/m^3)

$\rho_{\text{saturated}}$ = density of saturated soil, lb/ft^3 (kg/m^3)

σ_{Alt} = tensile stress range in the pipe due to the range of thermal expansion and contraction and/or the restraint of free end displacement, psi (MPa)

σ_E = tensile stress in the pipe due to an earthquake, psi (MPa)

σ_{sw} = circumferential compressive stress in the sidewalls of pipe, psi (MPa)

σ_{rc} = tensile stress in the pipe due to fully constrained contraction, psi (MPa)

σ_{re} = tensile stress in the pipe due to fully constrained expansion, psi (MPa)

τ_{Alt} = shear stress range in the thrust collar due to the range of thermal expansion and contraction and/or the restraint of free end displacement, psi (MPa)

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XXVI-3120 DESIGN LIFE

(a) The Design Specification shall specify the design life of the system.

(b) The duration of load shall be specified for each load case, and the polyethylene pipe physical and mechanical properties shall be based on the duration of load.

XXVI-3125 Examination Access

Accessibility to permit the examinations required by the Edition and Addenda of Section XI as specified in the Design Specification for the piping system shall be provided in the design of the piping system.

XXVI-3130 DESIGN AND SERVICE LOADINGS

Design loads shall be as defined in ND-3112.1 through ND-3112.3. Loads applied to buried polyethylene pipe shall be defined in the Design Specification and shall include, as a minimum, the following:

(a) Maximum internal Design Pressure, P_D, for pressure design in accordance with XXVI-3131 and XXVI-3132 and, if applicable, maximum negative internal pressure for evaluation in accordance with XXVI-3221.2.

(b) Maximum and minimum temperature, T, and the number of equivalent full range temperature cycles *(N)* for the selection of allowable stress and design for temperature effects in accordance with XXVI-3300.

(c) Vertical soil pressure, P_E, due to saturated soil, buoyancy, and flotation for the designs in accordance with XXVI-3200.

(d) Vertical pressure due to surcharge loads, P_L, for the design in accordance with XXVI-3200.

(e) Permanent ground movement and soil settlement for design as nonrepeated anchor movements in accordance with XXVI-3300.

(f) Seismic wave passage and seismic soil movement, building anchor motions, and number of seismic cycles for seismic design in accordance with XXVI-3400.

(g) Ground movement caused by frost heave for design for expansion and contraction in accordance with XXVI-3311.

XXVI-3131 Pressure Design of Pipe

XXVI-3131.1 Minimum Required Wall Thickness. The minimum required wall thickness of straight sections of pipe for pressure design shall be determined by the following:

$$t_{\text{Design}} = t_{\text{min}} + c$$

The value of c shall include an allowance for anticipated surface damage during installation.

$$t_{\text{min}} = \frac{P_D D}{(2S + P_D)}$$

The value of $t_{\text{fab min}}$ shall be greater than or equal to t_{Design}.

XXVI-3131.2 Allowable Service Level Spikes Due to Transient Pressures. The sum of the maximum anticipated operating pressure plus the maximum anticipated Service Level B pressure spikes due to transients shall be no greater than 1.2 times the piping system Design Pressure, P_D. The sum of the maximum anticipated operating pressure plus the maximum anticipated Service Level C or D pressure spikes due to transients shall be no greater than 2 times the piping system Design Pressure, P_D.

XXVI-3132 Pressure Design of Joints and Fittings

(a) Polyethylene pipe shall be joined using the butt fusion process. All connections to metallic piping shall be flanged joints.

(b) The design of piping items permitted in XXVI-2200 shall ensure these items have the capacity to withstand a pressure greater than or equal to the Design Pressure, P_D, of the attached pipe.

(c) The design of pipe fittings shall ensure the fitting has the capacity to withstand a pressure greater than or equal to the Design Pressure, P_D, of the attached pipe. The pressure rating (PR) of the fitting shall be determined as follows:

$$\text{GSR} \times \frac{2S}{DR - 1} \geq P_D$$

where GSR is the geometric shape rating per Table XXVI-3132-1

(d) Flanged connections shall include a metallic backup ring and shall provide a leak tight joint up to and including the piping hydrostatic test pressure. In addition, the maximum surge pressure per XXVI-3131.2 shall not cause permanent deformation of the pipe.

XXVI-3132.1 Pressure Design of Miter Elbows.

(a) The design pressure rating of the mitered elbow, P_m, shall be calculated as the lesser of eqs. (1) and (2) (see Figure XXVI-3132-1).

$$P_m = \frac{St_{\text{elbow}}}{r_2}\left(\frac{t_{\text{elbow}}}{t_{\text{elbow}} + 0.622\tan\theta\sqrt{t_{\text{elbow}} r_2}}\right) \quad (1)$$

or

$$P_m = \frac{St_{\text{elbow}}}{r_2}\left(\frac{R_1 - r_2}{R_1 - 0.5r_2}\right) \quad (2)$$

(b) P_m shall be greater than or equal to P_D. Alternatively, the mitered elbow shall be at least one standard dimension ratio (SDR) lower than that of the attached straight pipe. The maximum DR permitted for mitered elbow segments is 13.5.

(c) The minimum fabricated wall thickness of the reinforced sections of the mitered elbow shall be ≥1.25 $t_{\text{fab min}}$ of the attached straight pipe. The additional wall

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thickness shall be provided by enlarging the pipe O.D. while maintaining the pipeline I.D. or by reducing the pipe I.D. while maintaining the pipeline O.D.

(d) The fabrication tolerance of the fitting angular direction shall be ±3 deg. Mitered joints of 3 deg or less (angle α_{elb} in Figure XXVI-3132-1) do not require redesign consideration as mitered elbows.

(e) Mitered elbows shall comply with the requirements of ND-3644 with the following exceptions:

(1) Wall thickness shall be determined as outlined in (c).

(2) ND-3644(e) shall be replaced with butt fusion joints in accordance with this Appendix.

XXVI-3133 Allowable Stress Range for Secondary Stress

The allowable secondary stress range, S_A, is given in Table XXVI-3133-1 or Table XXVI-3133-1M.

(a) The S_A value shall be based on the higher of the Design Temperature or the maximum Service Level A or B temperature.

(b) The S_A shall be selected based on

(1) the total number of temperature cycles, or

(2) the number of equivalent full range temperature cycles, N, as determined in (d).

(c) The maximum number of permitted equivalent full range temperature cycles, N, is 100,000.

(d) The number of equivalent full range temperature cycles, N, is determined as follows:

$$N = N_E + N_1\left(\frac{\Delta T_1}{\Delta T_E}\right)^{5.0} + N_2\left(\frac{\Delta T_2}{\Delta T_E}\right)^{5.0} + \ldots, + N_n\left(\frac{\Delta T_n}{\Delta T_E}\right)^{5.0}$$

where

ΔT_E = maximum temperature change experienced by the pipe, °F (°C)

N_E = number of cycles at maximum temperature change, ΔT_E

$N_1, N_2, \ldots N_n$ = number of cycles at lesser temperature changes, $\Delta T_1, \Delta T_2, \ldots \Delta T_n$

$\Delta T_1, \Delta T_2, \ldots \Delta T_n$ = the lesser temperature changes experienced by the pipe, F (°C)

XXVI-3200 SOIL AND SURCHARGE LOADS

XXVI-3210 RING DEFLECTION

The soil and surcharge loads on a buried polyethylene pipe shall not result in a pipe diameter ring deflection, Ω, beyond the limit of Ω_{max} per Table XXVI-3210-1.

(U.S. Customary Units)

$$\Omega = \frac{100}{144}\frac{KLP_E + KP_L}{\frac{2E_{pipe}}{3}\left(\frac{1}{DR-1}\right)^3 + 0.061F_sE'} \le \Omega_{max}$$

(SI Units)

$$\Omega = 100\frac{KLP_E + KP_L}{\frac{2E_{pipe}}{3}\left(\frac{1}{DR-1}\right)^3 + 0.061F_sE'} \le \Omega_{max}$$

(U.S. Customary Units)

$$P_E = \left[\rho_{saturated}H_{gw} + \rho_{dry}\left(H - H_{gw}\right)\right]$$

(SI Units)

$$P_E = \left[\rho_{saturated}H_{gw} + \rho_{dry}\left(H - H_{gw}\right)\right]g/\left(1 \times 10^6\right)$$

E_{pipe} must be taken at the maximum life specified in the Design Specification, K= 0.1, and L = 1.25 to 1.5 or 1.0 if using soil prism pressure.

XXVI-3220 COMPRESSION OF SIDEWALLS

The circumferential compressive stress in the sidewalls, σ_{sw}, due to soil and surcharge loads shall not exceed S_{comp} per Table XXVI-3220-1 or Table XXVI-3220-1M.

(U.S. Customary Units)

$$\sigma_{SW} = \frac{1}{144}\frac{\left(P_E + P_L\right)DR}{2} \le S_{comp}$$

(SI Units)

$$\sigma_{SW} = \frac{\left(P_E + P_L\right)DR}{2} \le S_{comp}$$

XXVI-3221 External Pressure

XXVI-3221.1 Buckling Due to External Pressure. The following shall be met to ensure the pipe does not fail due to the effects of applied external pressure and possible negative internal pressure:

(a) When the depth of cover is greater than 4 ft (1.25 m) or one pipe diameter, whichever is larger, the external pressure from groundwater (flooding), earth loads, surcharge loads, and air pressure (due to negative internal pressure at minimum internal gage pressure) on a buried polyethylene pipe shall not cause the pipe to buckle. The following equation shall be met:

(U.S. Customary Units)

$$P_{hydro} = \Delta P + \frac{P_E + P_L + P_{gw}}{144} \le 2.8\sqrt{\frac{RB'E'E_{pipe}}{12\left(DR-1\right)^3}}$$

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(SI Units)

$$P_{\text{hydro}} = \Delta P + \left(P_E + P_L + P_{gw}\right) \le 2.8\sqrt{\frac{RB'E'E_{\text{pipe}}}{12\left(DR-1\right)^3}}$$

In addition, the requirements of XXVI-3221.2 shall also be met.

(b) When the depth of cover is less than 4 ft (1.25 m) or one pipe diameter (whichever is larger), the pipe must withstand the combined external pressure of groundwater (flooding), earth, surcharge, and air without credit for the surrounding soil. In this case, the following equation shall be met:

(U.S. Customary Units)

$$\Delta P + \frac{P_E + P_L + P_{gw}}{144} \le \frac{f_0 E_{\text{pipe}}}{\left(1-\upsilon^2\right)}\left(\frac{1}{DR-1}\right)^3$$

(SI Units)

$$\Delta P + P_E + P_L + P_{gw} \le \frac{f_0 E_{\text{pipe}}}{\left(1-\upsilon^2\right)}\left(\frac{1}{DR-1}\right)^3$$

υ = 0.45 for all loads

In this case, the requirements of XXVI-3221.2 do not need to be met.

The buoyancy reduction, R, and burial, B', factors are

$$R = 1 - \left(0.33 \times \frac{H_{gw}}{H}\right)$$

(U.S. Customary Units)

$$B' = \frac{1}{1 + 4\exp\left(-0.065H\right)}$$

(SI Units)

$$B' = \frac{1}{1 + 4\exp\left(-0.213H\right)}$$

XXVI-3221.2 Effects of Negative Internal Pressure. When the depth of cover is greater than 4 ft (1.25 m) or one pipe diameter (whichever is larger), the pipe must withstand the external air pressure resulting from negative internal pressure at the design minimum internal gage pressure without credit for the surrounding soil. This shall be ensured by meeting the following equation:

$$\Delta P \le \frac{f_0 E_{\text{pipe}}}{\left(1-\upsilon^2\right)}\left(\frac{1}{DR-1}\right)^3$$

υ is defined in XXVI-3221.1.

XXVI-3222 Flotation

Buried polyethylene pipe shall have sufficient cover or be anchored to the ground to prevent flotation by groundwater. To ensure this occurs, the following relationship shall be satisfied:

(U.S. Customary Units)

$$W_w < W_P + \left(P_E D/12\right)$$

(SI Units)

$$W_w < W_P + 1\,000\left(P_E D/g\right)$$

XXVI-3223 Longitudinal Stress Design

XXVI-3223.1 Longitudinal Applied Mechanical Loads. Longitudinal stresses due to axial forces and bending moments resulting from applied mechanical loads shall not exceed $K' \times S$

where

$$\left(\frac{B_1 P_a D}{2t}\right) + 2\left(\frac{B_1 F_a}{A}\right) + \left(\frac{B_2 M}{Z}\right) \le K' \times S$$

The value of K' is given in Table XXVI-3223-1. The values of B_1, B_2 are given in Table XXVI-3311-1, and S is per Table XXVI-3131-1(a) or Table XXVI-3131-1M(a) and Table XXVI-3131-1(b).

XXVI-3223.2 Short Duration Longitudinal Applied Mechanical Loads. For the assessment of short duration loads (less than 5 min), the allowable stress, S, may be replaced by one of the following alternatives:

(a) 40% of the material actual tensile strength at yield determined in accordance with ASTM D638 at temperature coincident with the load under consideration, or

(b) the values in Table XXVI-3223-2

XXVI-3300 TEMPERATURE DESIGN

XXVI-3310 MINIMUM TEMPERATURE

The polyethylene material shall not be used at a temperatures below the manufacturer's limit, but in no case shall the temperature be less than −50°F (−45°C).

XXVI-3311 Design for Expansion and Contraction

XXVI-3311.1 Fully Constrained Thermal Contraction. The stress resulting from the assumption of fully constrained thermal contraction of the buried pipe when $T_{\text{water}} < T_{\text{ground}}$, increased by the stress due to axial contraction from Poisson's effect, shall be determined as follows:

$$\sigma_{rc} = \left| E_{\text{pipe}} \alpha \Delta T - \frac{\upsilon PD}{2t} \right|$$

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XXVI-3311.2 Fully Constrained Thermal Expansion. The stress resulting from the assumption of fully constrained thermal expansion of the buried pipe when $T_{water} > T_{ground}$ shall be determined as follows:

$$\sigma_{re} = \left| E_{pipe} \alpha \Delta T \right|$$

XXVI-3311.3 Combined Thermal Expansion and Contraction Stress. The combined thermal expansion and contraction stress shall be

$$\left| \sigma_{rc} \right| + \left| \sigma_{re} \right| \le S_A$$

S_A is per XXVI-3133.

XXVI-3311.4 Alternative Thermal Expansion or Contraction Evaluation. As an alternative to XXVI-3311.1 and XXVI-3311.2, the soil stiffness may be accounted for to calculate pipe expansion and contraction stresses. The stresses shall satisfy the following equation:

$$\sigma_{Alt} = \frac{iM_C}{Z} + \frac{F_{aC}}{A} \le S_A$$

S_A is per XXVI-3133.

XXVI-3312 Nonrepeated Anchor Movements

The effects of any single nonrepeated anchor movements shall meet the requirements of the following equation:

$$\frac{iM_D}{Z} + \frac{F_{aD}}{A} < 2S$$

S is per Table XXVI-3131-1(a) or Table XXVI-3131-1M(a) and Table XXVI-3131-1(b).

XXVI-3313 Design of Thrust Collars

XXVI-3313.1 Fully Constrained Thermal Expansion and Contraction Evaluation. The resulting range of shear stress in the thrust collar resulting from the assumption of fully constrained thermal expansion and contraction of the buried pipe shall be limited to the following:

$$\tau_{Alt} = \left| \frac{F_E}{A_S} \right| + \left| \frac{F_C}{A_S} \right| \le 0.4 S_A$$

$$F_C = A\sigma_{rc}$$

$$F_E = A\sigma_{re}$$

S_A is per Table XXVI-3133-1 or Table XXVI-3133-1M.

XXVI-3313.2 Alternative Thermal Expansion and Contraction Evaluation. As an alternative to XXVI-3313.1, if the soil stiffness is accounted for to calculate pipe expansion and contraction stresses, the shear stress in the thrust collar shall satisfy the following equations:

$$\tau_{Alt} = \frac{F_{Alt}}{A_S} \le 0.40 S_A$$

S_A is per Table XXVI-3133-1 or Table XXVI-3133-1M.

$$F_{Alt} = A\sigma_{Alt}$$

XXVI-3400 SEISMIC DESIGN

XXVI-3410 SEISMIC-INDUCED STRESSES

The stresses in the buried polyethylene piping system due to soil strains caused by seismic wave passage, seismic soil movement, and building seismic anchor motion effects, where applicable, shall be evaluated. The stresses shall satisfy the following equation:

$$\frac{iM_E}{Z} + \frac{F_{aE}}{A} \le S_A$$

S_A is per XXVI-3133.

Seismic wave passage, seismic soil movement, and building seismic anchor motion loads shall be combined by square root sum of the squares.

Supplement XXVI-C provides an alternative method for the analysis of seismic wave passage, seismic soil movement, and building seismic anchor motion effects.

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Table XXVI-3131-1(a)
Long-Term Allowable Stress, *S*, for Polyethylene, psi

Temperature, °F	≤50 yr	Temperature, °F	≤50 yr	Temperature, °F	≤50 yr
≤73	800	96	689	119	587
74	795	97	684	120	582
75	790	98	680	121	578
76	785	99	675	122	574
77	780	100	670	123	570
78	775	101	666	124	565
79	770	102	661	125	561
80	765	103	657	126	557
81	760	104	652	127	553
82	755	105	648	128	549
83	751	106	643	129	545
84	746	107	639	130	540
85	741	108	634	131	536
86	736	109	630	132	532
87	731	110	626	133	528
88	726	111	621	134	524
89	722	112	617	135	520
90	717	113	612	136	516
91	712	114	608	137	512
92	708	115	604	138	508
93	703	116	599	139	504
94	698	117	595	140	500
95	694	118	591	...	...

GENERAL NOTE: The stresses listed in Tables XXVI-3131-1(a) and XXVI-3131-1(b) support a 50-yr operating life; stresses for operating lives longer than 50 yr are under development.

Table XXVI-3131-1M(a)
Long-Term Allowable Stress, *S*, for Polyethylene, MPa

Temperature, °C	≤50 yr	Temperature, °C	≤50 yr	Temperature, °C	≤50 yr
≤23	5.52	36	4.73	49	4.01
24	5.45	37	4.68	50	3.96
25	5.39	38	4.62	51	3.91
26	5.33	39	4.56	52	3.85
27	5.27	40	4.50	53	3.80
28	5.21	41	4.45	54	3.75
29	5.15	42	4.39	55	3.70
30	5.09	43	4.34	56	3.65
31	5.03	44	4.28	57	3.60
32	4.97	45	4.23	58	3.55
33	4.91	46	4.17	59	3.50
34	4.85	47	4.12	60	3.45
35	4.79	48	4.07	...	...

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Table XXVI-3131-1(b)
Elevated Temperature Allowable Stress, *S*, for Polyethylene, psi (MPa)

	≤0.3 yr	
Temperature, °F (°C)	psi	MPa
≤176 (≤80)	341	2.35

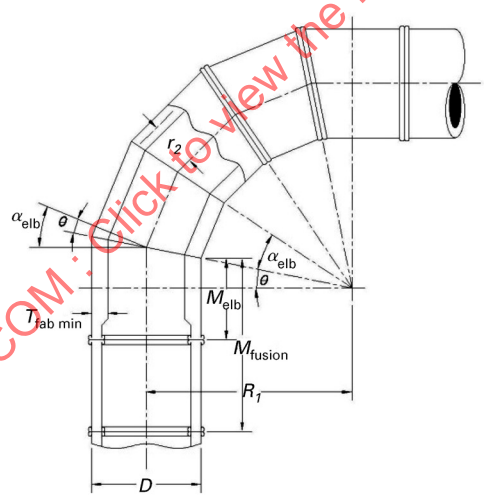
GENERAL NOTE: This allowable stress value is limited to one occurrence during the design life of the system.

Table XXVI-3132-1
Geometric Shape Ratings (GSR)

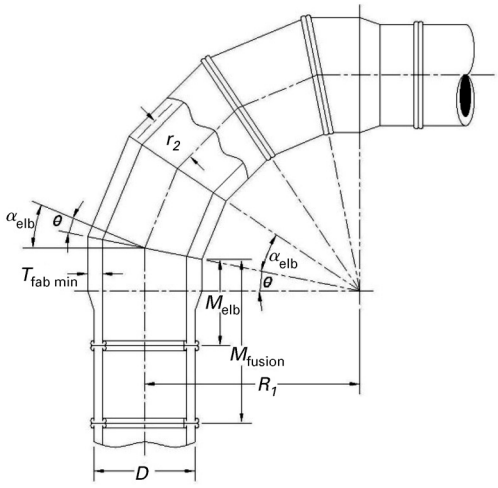
Fitting Description	GSR
Straight pipe	1.0
Molded flange adapters	1.0
Machined flange adapters	1.0
Molded fittings with reinforced body	1.0
Mitered (from one to five segments) *DR* 5.6 to *DR* 9	0.80
Mitered (from one to five segments) *DR* 9.5 to *DR* 13.5 (segments less than or equal to 22.5-deg directional changes per fusion)	0.75 [Note (1)]
Concentric conical monolithic reducer (machined or molded)	1.0
Thrust collar (machined or molded)	1.0
Molded tees equal outlet with reinforced body	1.0
Fabricated tees equal outlet (two *DR* less than pipe) *DR* 5 to *DR* 9	0.65

NOTE:
(1) Alternatively, the GSR factor may be determined by dividing the pressure rating determined by calculation or testing by the pressure rating of the pipe used to make the fitting.

Figure XXVI-3132-1
Nomenclature for Mitered Elbows



(a) Constant O.D., Five-Segment, Reinforced, 90-deg Elbow



(b) Constant I.D., Five-Segment, Reinforced, 90-deg Elbow

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Table XXVI-3133-1
S_A, Allowable Secondary Stress Limit, psi

Number of Equivalent Full-Range Temperature Cycles, N	The Higher of Design Temperature, the Maximum Service Level A Temperature, or the Maximum Service Level B Temperature, °F							
	≤70	80	90	100	110	120	130	140
$N \le 1{,}000$	3,930	3,770	3,610	3,440	3,280	3,110	2,930	2,760
$1{,}000 < N \le 10{,}000$	2,600	2,500	2,400	2,300	2,190	2,084	1,980	1,860
$10{,}000 < N \le 25{,}000$	2,200	2,120	2,040	1,950	1,870	1,780	1,690	1,590
$25{,}000 < N \le 50{,}000$	1,950	1,880	1,800	1,730	1,650	1,580	1,500	1,420
$50{,}000 < N \le 75{,}000$	1,830	1,770	1,700	1,630	1,540	1,470	1,400	1,320
$N_E > 75{,}000$	1,720	1,660	1,600	1,530	1,470	1,400	1,330	1,360

GENERAL NOTE: Linear Interpolation of stress between temperatures is permitted.

Table XXVI-3133-1M
S_A, Allowable Secondary Stress Limit, MPa

Number of Equivalent Full-Range Temperature Cycles, N	The Higher of Design Temperature, the Maximum Service Level A Temperature, or the Maximum Service Level B Temperature, °C								
	≤20	25	30	35	40	45	50	55	60
$N \le 1\,000$	27.3	26.3	25.3	24.3	23.3	22.2	21.2	20.1	19.0
$1\,000 < N \le 10\,000$	18.0	17.4	16.8	16.2	15.5	14.9	14.2	13.5	12.8
$10\,000 < N \le 25\,000$	15.3	14.8	14.3	13.8	13.2	12.7	12.1	11.6	11.0
$25\,000 < N \le 50\,000$	13.5	13.1	12.6	12.2	11.7	11.2	10.8	10.3	9.8
$50\,000 < N \le 75\,000$	12.7	12.3	11.9	11.5	10.9	10.5	10.0	9.6	9.1
$N_E > 75\,000$	11.9	11.5	11.2	10.8	10.4	10.0	9.6	9.1	8.7

GENERAL NOTE: Linear Interpolation of stress between temperatures is permitted.

Table XXVI-3210-1
Maximum Allowable Ring Deflection, Ω_{max}

DR	Ω_{max}, %
13.5	6.0
11	5.0
9	4.0
7.3	3.0

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Table XXVI-3210-2
Soil Support Factor, F_S

E'_N/E'	$(12B_d)/D$, in./in. or $(1{,}000B_d)/D$, mm/mm					
	1.5	2.0	2.5	3.0	4.0	5.0
0.1	0.15	0.30	0.60	0.80	0.90	1.00
0.2	0.30	0.45	0.70	0.85	0.92	1.00
0.4	0.50	0.60	0.80	0.90	0.95	1.00
0.6	0.70	0.80	0.90	0.95	1.00	1.00
0.8	0.85	0.90	0.95	0.98	1.00	1.00
1.0	1.00	1.00	1.00	1.00	1.00	1.00
1.5	1.30	1.15	1.10	1.05	1.00	1.00
2.0	1.50	1.30	1.15	1.10	1.05	1.00
3.0	1.75	1.45	1.30	1.20	1.08	1.00
5.0	2.00	1.60	1.40	1.25	1.10	1.00

Table XXVI-3210-3
Modulus of Elasticity of Polyethylene Pipe, E_{pipe}, psi

Load Duration	Temperature, °F								
	≤73	80	90	100	110	120	130	140	176
0.5 hr	82,000	76,300	67,200	59,900	52,500	47,600	41,000	35,300	18,000
1 hr	78,000	72,500	64,000	56,900	49,900	45,200	39,000	33,500	17,200
10 hr	65,000	60,500	53,300	47,500	41,600	37,700	32,500	28,000	14,200
24 hr	60,000	55,800	49,200	43,800	38,400	34,800	30,000	25,800	13,200
100 hr	55,000	51,200	45,100	40,200	35,200	31,900	27,500	23,700	12,100
1,000 hr	46,000	42,800	37,700	33,600	29,400	26,700	23,000	19,800	10,100
1 yr	40,000	37,200	32,800	29,200	25,600	23,200	20,000	17,200	8,800
10 yr	34,000	31,600	27,900	24,800	21,800	19,700	17,000	14,600	N/A
50 yr	29,000	27,000	23,800	21,200	18,600	16,800	14,500	12,500	N/A

Table XXVI-3210-3M
Modulus of Elasticity of Polyethylene Pipe, E_{pipe}, MPa

Load Duration	Temperature, °C								
	≤23	27	32	38	43	49	54	60	80
0.5 h	566	526	463	413	362	328	283	243	124
1 h	538	519	441	392	344	312	269	231	118
10 h	449	417	368	328	287	260	224	193	99
24 h	414	385	339	302	265	240	207	178	91
100 h	379	353	311	277	243	220	190	163	83
1 000 h	317	295	260	232	203	184	159	137	70
1 y	276	257	226	201	177	160	138	117	61
10 y	234	218	192	171	150	136	117	101	N/A
50 y	200	186	164	146	128	116	100	86	N/A

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Table XXVI-3220-1
Allowable Sidewall Compression Stress, S_{comp} (psi)

Temperature, °F	S_{comp}
≤50	1,180
73	1,015
140	520
160	400
180	325

Table XXVI-3220-1M
Allowable Sidewall Compression Stress, S_{comp} (MPa)

Temperature, °C	S_{comp}
≤10	8.14
23	7.00
60	3.59
71	2.76
82	2.24

Table XXVI-3221.2-1
Ovality Correction Factor, f_o

Ovality, %	Ovality Correction Factor
1	0.91
2	0.84
3	0.76
5	0.64
6	0.59

Table XXVI-3223-1
Design and Service Level Longitudinal Stress Factor, K'

Service Level	Design	A	B	C	D
K'	1.0	1.0	1.1	1.33	1.33

Table XXVI-3223-2
Short Duration (5 min) Allowable Longitudinal Tensile Stress

Temp, °F (°C)	S, psi (MPa)
≤70 (≤21)	1 200 (8.3)
100 (38)	940 (6.5)
120 (49)	770 (5.3)
140 (60)	630 (4.3)
176 (80)	400 (2.7)

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Table XXVI-3311-1
Stress Indices, Flexibility, and Stress Intensification Factors for PE Piping Components

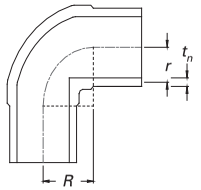
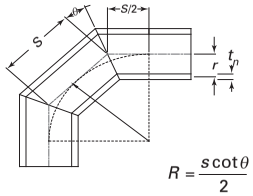
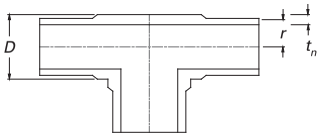
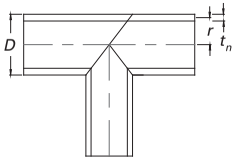
Description	Primary Stress Index B_1	Primary Stress Index B_2	Flexibility Characteristic, h	Flexibility factor, k	Stress Intensification Factor, i	Illustration
Straight pipe	0.5	1.0	N/A	1.0	1.0	N/A
Butt fusion joint	0.5	1.0	N/A	1.0	1.0	N/A
Molded elbow	0.69	[Note (1)]	$\frac{t_n R}{r^2}$	TBD	$\frac{1.25}{h^{2/3}}$	
Mitered elbow $s \geq r\ (1 + \tan\theta)$ [Note (2)], [Note (3)]	0.69	[Note (1)]	$\frac{(1+\cot\theta)}{DR-1}$ or $\frac{t_n(1+\cot\theta)}{2r}$	In-plane loading: $\frac{1.1}{h^{5/6}}$	$\frac{1.7}{h^{2/3}}$	
Equal outlet molded tee [Note (4)]	[Note (5)]	[Note (5)]	$\frac{4.4t_n}{r}$	1.0	$i_b = \frac{1.73}{h^{2/3}}$ $i_r = \frac{1.17}{h^{2/3}}$	
Equal outlet mitered tee	1.0	$B_{2b} = 0.75 i_b \geq 1.0$ $B_{2r} = 0.75 i_r \geq 1.0$	$\frac{4.4t_n}{r}$ or $\frac{8.8}{DR-1}$	1.0	$i_b = \frac{4.45}{h^{2/3}}$ $i_r = \frac{2.21}{h^{2/3}}$	

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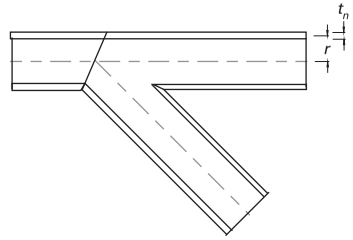
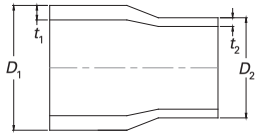
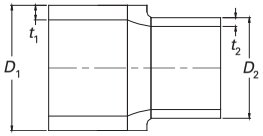
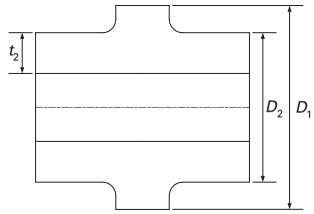
Table XXVI-3311-1
Stress Indices, Flexibility, and Stress Intensification Factors for PE Piping Components (Cont'd)

Description	Primary Stress Index B_1	Primary Stress Index B_2	Flexibility Characteristic, h	Flexibility factor, k	Stress Intensification Factor, i	Illustration
45-deg equal outlet mitered wye [Note (6)]	[Note (5)]	[Note (5)]	$\frac{4.4t_n}{r}$ or $\frac{8.8}{DR-1}$	1.0	$i_b = \frac{2.75}{h^{2/3}}$ $i_r = \frac{1.71}{h^{2/3}}$	
Concentric monolithic reducers	[Note (5)]	[Note (5)]	N/A	1.0	$0.5 + 0.32\left(\frac{D_1}{D_2}\right)\left(\frac{D_2}{t_2}\right)^{1/2} \le 2.5$	
Concentric fabricated reducers	[Note (5)]	[Note (5)]	N/A	1.0	$0.5 + 0.40\left(\frac{D_1}{D_2}\right)\left(\frac{D_2}{t_2}\right)^{1/2} \le 3.0$	
Fabricated, machined, or molded thrust collar	0.5	$B_2 = 0.75i \ge 1.0$	N/A	1.0	$0.5 + 0.40\left(\frac{D_1}{D_2}\right)\left(\frac{D_2}{t_2}\right)^{1/2} \le 3.0$	
Machined or molded metallic to PE bolted flange connection	0.5	1.0	N/A	1.0	1.0	See Figure XXVI-4520-1

GENERAL NOTES:

(a) The following nomenclature applies to this Table only for use in determining stress indices, stress intensification factors, and flexibility factors:

D_1 = nominal outside diameter of the larger side of a concentric fabricated reducer or the diameter of the thrust collar

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Table XXVI-3311-1
Stress Indices, Flexibility, and Stress Intensification Factors for PE Piping Components (Cont'd)

GENERAL NOTES (CONT'D):

D_2 = nominal outside diameter of the smaller side of a concentric fabricated reducer or the nominal pipe diameter of a thrust collar
DR = Pipe Dimension Ratio = D_o/t_n.
R = nominal bend radius of elbow or pipe bend, in. (mm)
r = mean radius of pipe, in. (mm) (matching pipe for elbows and tees)
s = miter spacing at centerline, in. (mm)
T_2 = nominal thickness of the smaller side of a concentric fabricated reducer or nominal pipe thickness of a thrust collar
t_n = nominal wall thickness of pipe, $t_{\text{fab min}}$, in. (mm) (matching pipe for elbows and tees)
T_r = nominal wall thickness of run pipe, $t_{\text{fab min}}$, in. (mm)
θ = one-half angle between adjacent miter axes, deg

(b) The stress indices, i, and the flexibility factors, k, shall not be taken as less than 1.0. They are applicable to moments in any plane for fittings except as noted.
(c) All abutting piping fittings of differing DRs shall meet XXVI-4231.

NOTES:

(1) The B_2 stress indices for mitered bends and molded elbows are dependent upon the DR and shall be as follows:
(a) 1.38 for DR 7
(b) 1.64 for DR 9
(c) 1.91 for DR 11
(d) 2.21 for DR 13.5
(2) One-half miter angle, θ, is limited to ≤11.25 deg.
(3) The flexibility factor, k, is only applicable for in-plane bending moment loading.
(4) The tee thickness, t_n, has to be 1.4 times the pipe thickness, T_r ($1.4T_r$).
(5) Indices in development.
(6) The SIF equation is limited to 45-deg lateral wyes.

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ARTICLE XXVI-4000
FABRICATION AND INSTALLATION

XXVI-4100 GENERAL REQUIREMENTS

XXVI-4110 INTRODUCTION

(a) Fabrication and installation shall be in accordance with the rules of this Article and shall use polyethylene materials that comply with the requirements of Article XXVI-2000. Methods of fabrication and installation shall be by thermal butt fusion and flanged joints. Use of threaded or adhesive joints with polyethylene material shall not be permitted.

(b) Only the thermal fusion circumferential butt joints and miter joints may be used for pressure boundary fusion joints (see Figure XXVI-4110-1).

(c) Branch connections shall not be permitted in polyethylene material.

(d) Hereinafter, all requirements specified in this Article shall apply to fabrication and installation of polyethylene material.

XXVI-4120 CERTIFICATION OF MATERIAL, FABRICATION, AND INSTALLATION BY CERTIFICATE HOLDER

XXVI-4121 Means of Certification

The Certificate Holder for an item shall certify, by application of the appropriate Certification Mark including Designator, if applicable, and completion of the appropriate Data Report in accordance with Article XXVI-8000, that the material used complies with the requirements of Article XXVI-2000 and that the fabrication or installation complies with the requirements of this Article.

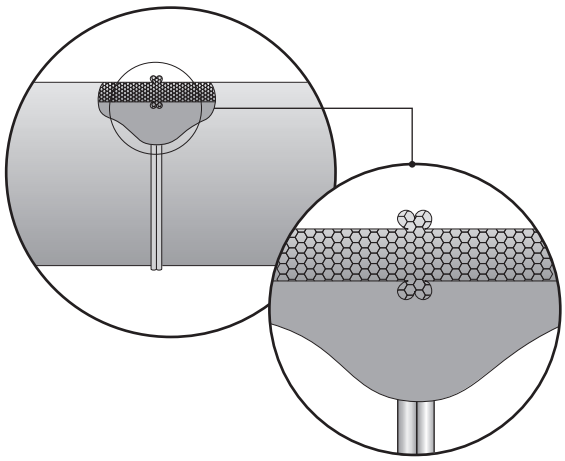
XXVI-4121.1 Certification of Treatments, Tests, and Examinations. If a Certificate Holder or his Subcontractor performs treatments, tests, repairs, or examinations required by other Articles of this Appendix, the Certificate Holder shall certify that this requirement has been fulfilled (NCA-3974). Reports of all required treatments and results of all required tests, repairs, and examinations performed shall be available to the Inspector.

XXVI-4121.2 Repetition of Visual Examination of Surfaces after Material Removal. If, during the fabrication or installation of a pressure-retaining item, new surfaces result, the Certificate Holder shall reexamine the surface of the material in accordance with XXVI-4130 when the original surface was required to be visually examined in accordance with XXVI-4130.

XXVI-4122 Material Identification

(a) Material for pressure-retaining items shall have identification markings that will remain distinguishable until the item is assembled or installed. If the original identification markings are cut off or the material is divided, the same marks shall either be transferred to the items cut or a coded marking shall be used to ensure

Figure XXVI-4110-1
Thermal Fusion Butt Joint



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identification of each piece of material during subsequent fabrication or installation. In either case, an as-built sketch or a tabulation of materials shall be prepared identifying each piece of material with the CPTRs and C of A Reports, where applicable, and the coded marking. Studs, bolts, nuts, flange rings, and other metallic items shall be identified and certified as required by Article ND-4000.

(b) Material from which the identification marking is lost shall be treated as nonconforming material until appropriate verifications are performed and documented to ensure proper material identification. Positive identification shall be made through appropriate evidence, and the material may then be marked; otherwise, it shall be scrapped.

XXVI-4122.1 Marking Material. Material shall be marked in accordance with Article XXVI-8000, as follows:

(a) No indentation stamping is allowed on the polyethylene surface; all marking shall be performed with a metallic paint marker or stenciling marker.

(b) The Polyethylene Material Manufacturer is permitted to apply the standard print line identifier to his piping product using a thermal process in accordance with XXVI-2231(c).

XXVI-4123 Examinations

Visual examination activities that are not specified for examination by XXVI-4130 or Article XXVI-5000, and are performed solely to verify compliance with requirements of Article XXVI-4000, may be performed by the persons who perform or supervise the work. These visual examinations are not required to be performed by personnel and procedures qualified in accordance with the Manufacturer's Quality Assurance Program (XXVI-2500) or to XXVI-5500 unless so specified.

XXVI-4130 REPAIR OF MATERIAL

All polyethylene material shall be inspected upon receipt. Any material not meeting the surface acceptance criteria of XXVI-2400 shall either be scrapped or repaired in accordance with this paragraph. All polyethylene material external surfaces shall be given an additional visual examination prior to installation.

(a) For pipe IPS 4 (DN 100) and smaller, any indentation greater than 5% of $t_{fab\ min}$ shall be unacceptable. Indentations of 5% or less of $t_{fab\ min}$ shall be acceptable provided that the remaining wall thickness is greater than t_{Design}.

(b) For pipe larger than IPS 4 (DN 100), any indentation greater than 0.040 in. (1.0 mm) shall be unacceptable. Indentations of 0.040 in. (1.0 mm) or less shall be acceptable provided that the remaining pipe wall thickness is greater than t_{Design}.

XXVI-4131 Elimination of Surface Defects

Pipe surface gouges or cuts greater than 5% of $t_{fab\ min}$ in pipe IPS 4 (DN 100) and smaller and greater than 0.040 in. (1 mm) in pipe greater than IPS 4 (DN 100) shall be removed by the Certificate Holder by grinding or machining in accordance with the following requirements:

(a) The cavity has a minimum taper of 3:1 (half-width of the overall area to depth) without any sharp edges.

(b) The remaining wall thickness is in excess of t_{Design}.

(c) As an alternative to (a) and (b), the damaged portion may be removed and discarded.

XXVI-4131.1 Additional Requirements — Flange Adapters.

(a) Damage in the pipe section in the transition between the pipe and hub sections shall require flange adapter replacement.

(b) Damage in the flange face (hub) shall be repaired by machining only if, after the repair, the minimum hub dimensional requirements of ASTM F2880 are met.

XXVI-4131.2 Additional Requirements — Thrust Collars. If the damaged area is in the transition between the pipe and hub sections, the entire thrust collar section shall be replaced.

XXVI-4200 FORMING, FITTING, AND ALIGNING

XXVI-4210 CUTTING, FORMING, AND BENDING

XXVI-4211 Cutting

Material shall be cut to shape and size by mechanical methods.

XXVI-4212 Forming and Bending Processes

The material shall not be cold, hot formed, or bent except as follows:

(a) During installation, a pipe radius of curvature greater than or equal to 30 times the outside diameter is permitted for piping with DR 9 through 13.5, except as restricted by (b).

(b) During installation, a pipe radius of curvature for pipe with a DR 14 or higher and all pipe within two outside diameters of a flange connection or mitered elbow (measured from the pipe to fitting fused joint) shall not have a radius of curvature less than 100 pipe outside diameters.

XXVI-4213 Minimum Thickness of Fabricated Items

If any operation reduces the thickness below the minimum required to satisfy the rules of Article XXVI-3000 and XXVI-4130, the material shall be scrapped.

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XXVI-4230 FITTING AND ALIGNING

XXVI-4231 Fitting and Aligning Methods

Items to be joined shall be fitted, faced, aligned, and retained in position during the fusing operation using appropriate fusing machines.

(a) Items of different outside diameters shall not be fused together except as provided in (c).

(b) The alignment surface mismatch shall be less than 10% $t_{fab\ min}$ of the items being fused.

(c) For items with different DRs, the item with the smaller DR shall be counterbored and tapered to equal the wall thickness, or its outside diameter shall be machined and tapered to equal the wall thickness of the item with the larger DR and shall comply with Figure XXVI-4230-1, illustration (a) or illustration (b).

XXVI-4240 JOINT END TRANSITIONS

The joint end transitions of items shall provide a gradual change in thickness from the item to the adjoining items and shall comply with XXVI-4231(c) and Figure XXVI-4230-1.

XXVI-4300 FUSING QUALIFICATIONS

XXVI-4310 GENERAL REQUIREMENTS

XXVI-4311 Types of Processes Permitted

Only those fusing processes that are capable of producing fused joints in accordance with fusing procedure specifications qualified in accordance with Section IX and tested in accordance with XXVI-2300 of this Appendix may be used for fusing pressure-retaining material. Any process used shall be such that the records required by XXVI-4320 can be prepared.

XXVI-4312 Fusing Operator Training

(a) The fusing machine operator shall receive a minimum of 24 hr of training, covering the principles of the fusion process and the operation of the fusing equipment. There shall be a two-part test at the end of this training.

(1) The written theoretical knowledge part of the test shall cover such topics as safety, fundamentals of the fusing process, and recognition of typical joint imperfections.

(2) The practical knowledge portion shall include hands-on training using equipment make and models to be used in production.

(3) Successful completion of this training shall be documented on the performance qualification record.

(4) Performance qualification testing shall be performed and documented in accordance with Section IX and this Article.

(b) Supplement XXVI-A provides guidance for the training program.

XXVI-4320 FUSING QUALIFICATIONS, RECORDS, AND IDENTIFYING STAMPS

XXVI-4321 Required Qualifications

(a) The Certificate Holder shall be responsible for the fusing done by his organization and shall establish the procedure and conduct the tests required by this Article and Section IX, in order to qualify both the fusing procedures and the performance of fusing operators who apply these procedures. Only fusing procedures tested in accordance with XXVI-2300 shall be used.

(b) Procedures and fusing machine operators used to join pressure parts shall also meet the training, testing, and qualification requirements of this Article. Mitered joints shall be fused using procedures and personnel qualified for butt-fused joints in accordance with Section IX and this Article.

(c) The make and model of each fusing machine carriage to be used in production shall be performance tested on all diameters and thicknesses to be fused in accordance with XXVI-2300. The testing — or applicable portions thereof — may be performed by the Certificate Holder prior to installation. Machines to be used at angles exceeding 20-deg slope shall be tested at the maximum slope and maximum estimated drag to be applied. The tested machine make(s) and model(s) shall be documented on the fusing procedure specification.

XXVI-4322 Maintenance and Certification of Records

The Certificate Holder shall maintain records of qualified fusing procedures and the fusing machine operators qualified by him, showing the date and results of testing and the identification mark assigned to each fusing operator. These records shall be reviewed, verified, and certified by the Certificate Holder by signature or some other method of control in accordance with the Certificate Holder's Quality Assurance Program and shall be available to the Inspector.

XXVI-4322.1 Identification of Joints by Fusing Operator. Each fusing operator shall apply the identification mark assigned to him by the Certificate Holder adjacent to all permanent fused joints or series of joints on which he fuses. The marking shall be 1 ft (0.3 m) or less from the fusion bead and shall be done with permanent metallic paint marker or stenciling marker. As an alternative, the Certificate Holder shall keep a record of permanent fused joints in each item and of the fusing machine operators used in fusing each of the joints.

XXVI-4323 Fusing Prior to Qualification

No fusing shall be performed until after the fusing procedure specification which is to be used has been qualified. Only fusing procedures and operators qualified in

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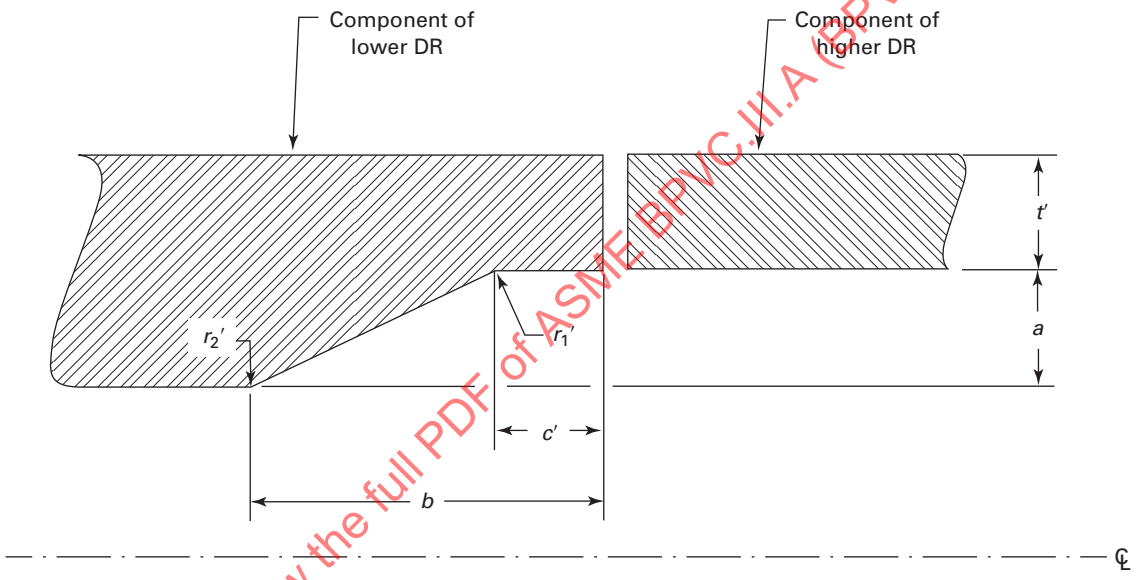
Figure XXVI-4230-1
Tapered Transition Joint



(a) Reinforcement on Inside Diameter

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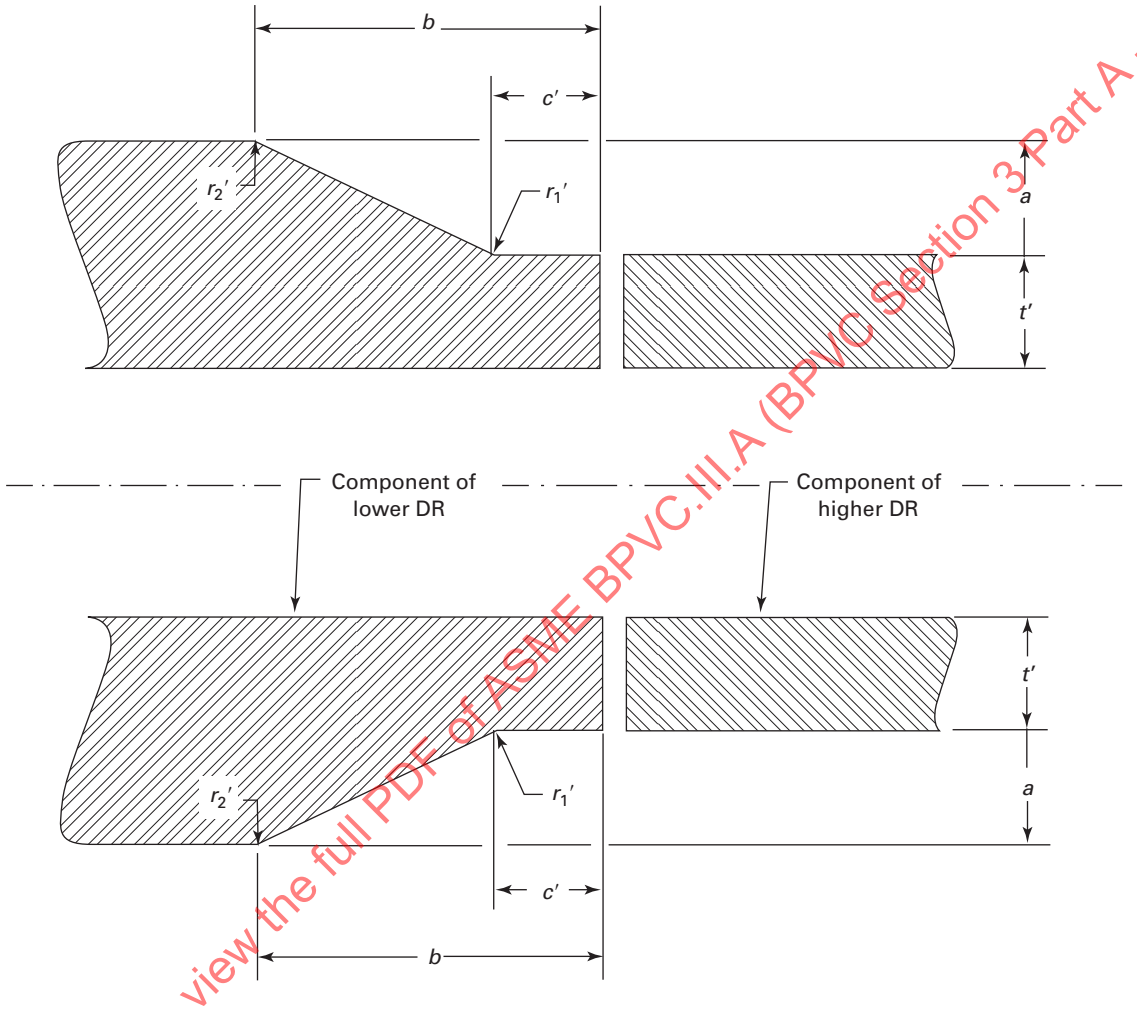
Figure XXVI-4230-1
Tapered Transition Joint (Cont'd)



(b) Reinforcement on Outside Diameter

t' = $t_{fab,min}$ of thinner component

c'_{min} = $2.5t'$

c' = values are after facing

$$0.35 < \left[\frac{a}{(b - c')}\right] < 0.60$$

$r_1'_{min}$ = $0.05t'$

$r_2'_{min}$ = $0.05t'$

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accordance with this Article and Section IX shall be used. Only fusing machines tested in accordance with XXVI-2300 shall be used for production.

XXVI-4324 Transferring Qualifications

The fusing procedure qualifications or performance qualification tests for fusing operators conducted by one Certificate Holder shall not qualify fusing procedures or fusing operators to fuse for any other Certificate Holder.

XXVI-4330 GENERAL REQUIREMENTS FOR FUSING PROCEDURE QUALIFICATION TESTS

XXVI-4331 Conformance to Section IX

All fusing procedure qualification tests shall be in accordance with the requirements of Section IX as supplemented or modified by the requirements of this Appendix, including the testing required by XXVI-2300.

XXVI-4332 Preparation of Test Coupons and Specimens

(a) Removal of test specimens from the fusion test coupon and the dimensions of specimens for procedure qualification and for the testing required by XXVI-2300 shall conform to the requirements of Section IX, except as noted in (b).

(b) Fusion joint test specimens for the Elevated Temperature Sustained Pressure Test described in Section IX, QF-142, shall be made with a minimum of IPS 8 (DN 200) DR 11 pipe or the maximum size to be fused, whichever is less. The completed specimens shall contain pipe on either side of the joint with a minimum length of 1.5 times the pipe outside diameter or 12 in. (305 mm), whichever is greater, from the fused joint to free-end closures on either end.

XXVI-4333 Performance of Testing

Testing shall conform to the requirements of Section IX and the additional requirements of XXVI-2300, except that elevated temperature sustained pressure tests for pipe shall be performed in accordance with ASTM D3035 or ASTM F714 and XXVI-4332(b).

XXVI-4340 GENERAL REQUIREMENTS FOR PERFORMANCE QUALIFICATION TESTS

XXVI-4341 Conformance to Section IX

All fusing operator performance qualification tests shall be in accordance with Section IX, as supplemented or modified by the requirements of this Article, using fusing procedures qualified in accordance with this Article and tested in accordance with XXVI-2300.

XXVI-4342 Additional Requirements

(a) The minimum pipe size shall be IPS 8 (DN 200) DR 11.

(b) A data acquisition device shall be attached to the fusing machine for recording the data concerning the joint required by Section IX, QF-131.

(c) The visual examination required by Section IX, QF-305, shall be performed over the entire inside and outside surfaces of the coupon.

(d) Bend specimens shall be tested in accordance with Section IX, QF-143. The specimens shall not crack or separate in the fused joint.

(e) As an alternative to the bend testing prescribed in Section IX, High-Speed Tensile Impact Testing may be performed in accordance with Section IX. No fewer than four specimens shall be removed from fused pipe test coupons at intervals approximately 90 deg apart.

XXVI-4400 RULES GOVERNING MAKING, EXAMINING, AND REPAIRING FUSED JOINTS

XXVI-4410 PRECAUTIONS TO BE TAKEN BEFORE FUSING

XXVI-4411 Identification, Storage, and Handling of Materials

The Certificate Holder shall be responsible for control of the materials that are used in the fabrication and installation of components (see XXVI-4120). Suitable identification, storage, and handling of material shall be maintained.

XXVI-4412 Cleanliness and Protection of Fusing Surfaces

(a) Precautions shall be taken to prevent contamination of the joint during the fusion process.

(1) The surfaces of the heater used for fusing shall be free of scale, rust, oil, grease, dust, fine particulate, and other deleterious material.

(2) The joint shall be protected from deleterious contamination and from rain, snow, dust, fine particulate, and wind during fusing operations. Fusing shall not be performed on wet surfaces or surfaces containing dust or fine particulate.

(b) Fusing shall not be performed at ambient temperatures less than 50°F (10°C) or greater than 125°F (52°C), unless an environmental enclosure is used to control work area temperature between 50°F (10°C) and 125°F (52°C). For ambient fusing temperatures between 100°F (38°C) and 125°F (52°C), minimum cooling time shall be 13 min/in. of thickness.

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XXVI-4420 RULES FOR MAKING FUSED JOINTS

XXVI-4421 Heating Cycle

(a) Immediately prior to inserting the heater plate between the faced ends to be joined, the temperature shall be verified to be within the required range by measuring at four locations approximately 90 deg apart in the fusing zone, on both sides of the heater plate.

(b) Care shall be taken upon heater removal to ensure uniform flat heated surfaces on both pipe ends of the joint prior to fusing together.

XXVI-4422 Surfaces of Fused Joints

Fused beads shall remain intact after completion of fusing.

(a) When required, fused beads may be removed but only after the visual inspection required by XXVI-5210(b) is completed and documented. The entire surface at the removed bead locations shall be inspected and shall meet the acceptance criteria of XXVI-5322.

(b) The finished joint shall be suitable for required visual and volumetric examinations.

XXVI-4423 Joint Transitions

When items of different diameters are fused together, there shall be a gradual transition between the two surfaces in accordance with XXVI-4231(c).

XXVI-4440 FUSING DATA ACQUISITION RECORDER

The fusing machine shall have a data acquisition recorder attached to it for each joint fused in accordance with this Article. The data acquisition record produced by the device shall include the information specified in Section IX. In addition, job information related to the joints, such as job number, joint number, fusing machine operator, date, and time, shall be recorded. The data acquisition device shall be capable of a minimum of 1 day of butt fusion joint information and capable of downloading this information as a permanent record.

(a) Failure of a recorder to operate properly during the fusion process shall cause removal and replacement of the fused joint.

(b) The data acquisition records shall be compared with the fusing procedure specification to ensure that the proper butt-fusing parameters and procedures were followed. If any parameter is outside the specified range, the fused joint shall be removed and replaced in compliance with the fusing procedure specification, or the item shall be scrapped.

(c) Verification of fusing parameters and variables not included in the data acquisition record shall be documented in accordance with the Certificate Holder's Quality Assurance Program.

XXVI-4450 REPAIR OF FUSED JOINTS

XXVI-4451 General Requirements

(a) Defects in fused joints detected by the examinations required by Article XXVI-5000, or by the testing of Article XXVI-6000, shall cause rejection of the joint. Repair of a fused joint shall not be permitted. All unacceptable joints shall be removed and replaced.

(b) Fusion beads are not required to remain intact. Damaged fusion beads shall be evaluated to verify no infringement upon the fusion joint or adjacent base material. Damaged portions of fusion beads shall be removed if necessary to perform this evaluation.

XXVI-4452 Elimination of Surface Defects

Surface defects may be removed by grinding or machining in accordance with the requirements of XXVI-4131. The removal area shall be reinspected and shall meet the acceptance criteria of XXVI-5322.

XXVI-4500 MECHANICAL JOINTS

XXVI-4510 BOLTING AND THREADING

XXVI-4511 Thread Engagement

The threads of all bolts or studs shall be engaged in accordance with the design. Flange bolting shall be engaged as required by XXVI-4520.

XXVI-4512 Thread Lubricants

Any lubricant or compound used in threaded joints shall be suitable for the service conditions and shall not react unfavorably with either the service fluid, polyethylene material, or any other material in the system.

XXVI-4520 FLANGED JOINTS

(a) Only flanged connections are permitted for joining of polyethylene pipe to metallic pipe or piping items. The polyethylene flange connection shall be constructed using a polyethylene flange adapter having a DR ratio equal to the attached polyethylene pipe and joined by fusion to the attached pipe.

(b) The polyethylene flange adapter shall be connected to the metal flange using a metallic backing ring. The backing ring shall have a pressure rating equal to or greater than the metal flange.

(c) Before tightening, flange faces shall be parallel within $^{1}/_{16}$ in./ft (5.3 mm/m) measured across any diameter, and flange bolt holes shall be aligned within $^{1}/_{8}$ in. (3.2 mm) maximum offset. Damage to the gasket seating surface on the polyethylene flange that would prevent the gasket from sealing shall be evaluated per XXVI-4131.1(b). Use of a gasket is optional.

(d) The flange shall be joined using bolts of a size and strength that conforms to the requirements of ASME B16.5 or ASME B16.47 Series A, as applicable. Bolts or studs should extend completely through their nuts. Any

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bolts or studs which fail to do so are considered acceptably engaged if the lack of complete engagement is not more than one thread. Flat washers shall be used under bolt heads and nuts.

(e) In assembling flanged joints, the gasket, if used, shall be uniformly compressed to the proper design loading. Special care shall be used in assembling flanged joints in which the flanges have widely differing mechanical properties. Tightening to a predetermined torque is recommended. If used, no more than one gasket shall be between contact faces in assembling a flanged joint. The gasket material shall be selected to be consistent and compatible with the service requirements of the piping system.

(f) See Figure XXVI-4520-1 for a typical flange configuration.

XXVI-4600 PIPE SUPPORTS

All installed supports for polyethylene piping shall meet the requirements of Subsection NF and the following:

(a) Piping shall be supported, guided, and anchored in such a manner as to prevent damage thereto. Point loads and narrow areas of contact between piping and supports shall be avoided. Suitable padding shall be placed between piping and supports where damage to piping may occur.

(b) Valves and equipment that would transmit excessive loads to the piping shall be independently supported to prevent such loads.

XXVI-4700 THRUST COLLARS USING POLYETHYLENE MATERIAL

Thrust collars shall be joined by butt fusion to the attached piping. Thrust collars shall comply with Figure XXVI-2234-1.

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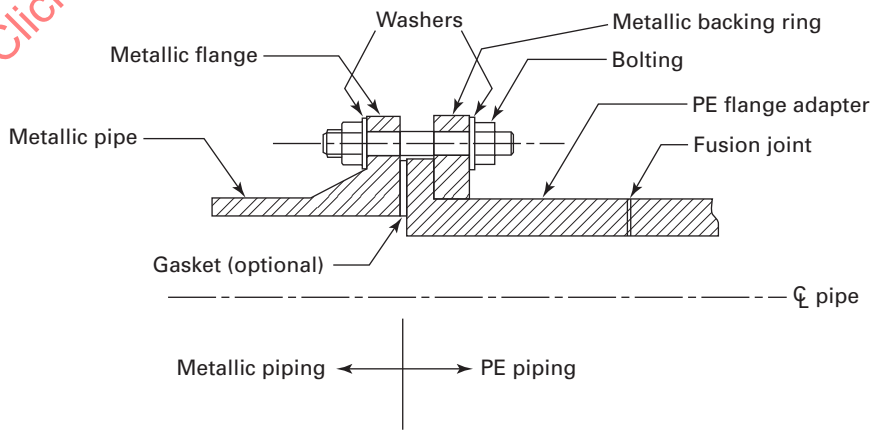
Figure XXVI-4520-1
Transition Flange Arrangement



ARTICLE XXVI-5000
EXAMINATION

XXVI-5100 GENERAL REQUIREMENTS FOR EXAMINATION

XXVI-5110 PROCEDURES, QUALIFICATION, AND EVALUATION

XXVI-5111 General Requirements

(a) Nondestructive examinations shall be conducted in accordance with the examination methods of Section V, except as modified by the requirements of this Article.

(b) Visual examinations shall be conducted in accordance with Section V, Article 9.

(c) Ultrasonic examination shall be in accordance with Section V, Article 4 and Supplement XXVI-II.

(d) The Certificate Holder shall be responsible for reviewing procedure and demonstration results to validate that the range of the essential variables of the procedure were included in the demonstration.

XXVI-5112 NDE Procedures

All nondestructive examinations performed under this Article shall be performed in accordance with detailed procedures, which have been proven by actual demonstration to the satisfaction of the Inspector in accordance with Section V, Article 1[19] and XXVI-5114. Procedures, records of demonstration of procedure capability, and personnel qualification shall be available to the Inspector on request.

XXVI-5113 Volumetric Examination Procedures

(a) The ultrasonic examination shall include the joint volume and includes the joint-to-base material interface and $^{1}/_{4}$ in. (6 mm) from the joint centerline into the joint base material (see Figure XXVI-5220-1).

(b) The ultrasonic examination shall be performed using encoded (position and amplitude) examination techniques.

(c) An ultrasonic examination procedure shall be developed in accordance with the format in Supplement XXVI-II and qualified by performance demonstration per XXVI-5114.

The ultrasonic examination procedure shall

(1) Contain a statement of scope that specifically defines the limits of procedure applicability (e.g., minimum and maximum thickness, minimum and maximum diameter, scanning access).

(2) Specify which parameters are considered essential variables. The procedure shall specify a single value or a range of values for the essential variables.

(3) List the examination equipment, including equipment manufacturer and model or series.

(4) Define the scanning requirements, such as beam angles, scan patterns, beam direction, maximum scan speed, extent of scanning, and access requirements.

(5) Contain a description of the calibration method (e.g., actions required to ensure that the sensitivity and accuracy of the signal amplitude and time outputs of the examination system, whether displayed, recorded, or automatically processed, are repeated from examination to examination).

(6) Contain techniques for data interpretation and plotting.

XXVI-5114 Qualification of Volumetric Examination Procedures

(a) The volumetric examination procedure shall be qualified, and demonstrated to the satisfaction of the Inspector, using specimens conforming to the following requirements:

(1) The specimens shall be fabricated from the same polyethylene material (PE4710) being installed.

(2) The demonstration specimen(s) shall include the same size and type of joint to be examined (e.g., butt joint).

(3) The demonstration specimen scanning and joint surfaces shall be representative of the production surfaces to be examined as specified in the ultrasonic examination procedure.

(4) The demonstration specimens shall include relevant actual or simulated fabrication-type flaws (e.g., lack of fusion, inclusions) consistent with the type of production joint to be examined.

(5) The demonstration set shall include specimens with through-wall flaw sizes including the smallest flaw size of 0.040 in. (1 mm) or 10% of the thickness (or maximum permitted by design), whichever is larger, and the largest flaw size between 30% and 90% of the thickness.

(b) The demonstration set shall include at least one of each minimum and maximum flaw size at each of the following locations: I.D. surface-connected, O.D. surface-connected, and embedded flaw. All flaws shall be separated by a minimum of 1 in. (25 mm).

(c) The demonstration shall be considered acceptable if 100% of the above flaws are identified.

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XXVI-5120 TIME OF EXAMINATION OF COMPLETED FUSED JOINTS

Nondestructive examination of fused joints shall be conducted

(a) after the completion of the cooling period

(b) before the joint becomes inaccessible in the burial trench

XXVI-5200 EXAMINATIONS

XXVI-5210 VISUAL EXAMINATION

Visual examinations shall be performed on the following:

(a) external surface, during receipt inspection, for visual evidence of flaws imposed during packaging, transport, and handling

(b) fusion joints, including review and verification of fusion data for the joint in accordance with XXVI-4440

(c) accessible external surfaces after placement in the trench, for visual evidence of flaws imposed during fabrication and installation

XXVI-5220 ULTRASONIC EXAMINATION

All fused joints in pipe 4-in. (100-mm) O.D. or greater shall be volumetrically examined.

(a) The examination volume shall include 100% of the area of interest shown in Figure XXVI-5220-1.

(b) Each joint shall also be examined 360 deg using the techniques demonstrated in XXVI-5114.

XXVI-5300 ACCEPTANCE STANDARDS

XXVI-5310 GENERAL REQUIREMENTS

Unacceptable fusion joints shall be removed. Repair of unacceptable joints shall not be permitted.

XXVI-5320 VISUAL EXAMINATION ACCEPTANCE CRITERIA OF EXTERNAL SURFACES

XXVI-5321 Fused Butt Joints

Joints shall meet the following:

(a) Fused butt joints shall exhibit proper fusion bead configuration. Supplement XXVI-B depicts unacceptable thermally fused bead configurations.

(b) There shall be no visible evidence of cracks in the cleavage or incomplete fusion as evidenced by cleavage extending beneath the O.D. surface of the piping. The cleavage between fusion beads shall not extend to or below the O.D. pipe surface (see Figure XXVI-5321-1). When cleavage depth cannot be visually verified, pit or depth gages shall be used to verify compliance or else the joint shall be rejected.

(c) Fused joints, except for miter joints, shall not be visually angled or offset by 3 deg or more. The ovality offset shall be less than 10% $t_{fab\ min}$ of the fused items.

(d) The data acquisition record for the fused joint shall be compared with the fusing procedure specification to verify parameters and procedures were followed in fusing the joint.

XXVI-5322 Material Surfaces

Surfaces of all material shall meet the requirements of XXVI-4131.

Figure XXVI-5220-1
Fusion Pipe Joint Examination Volume

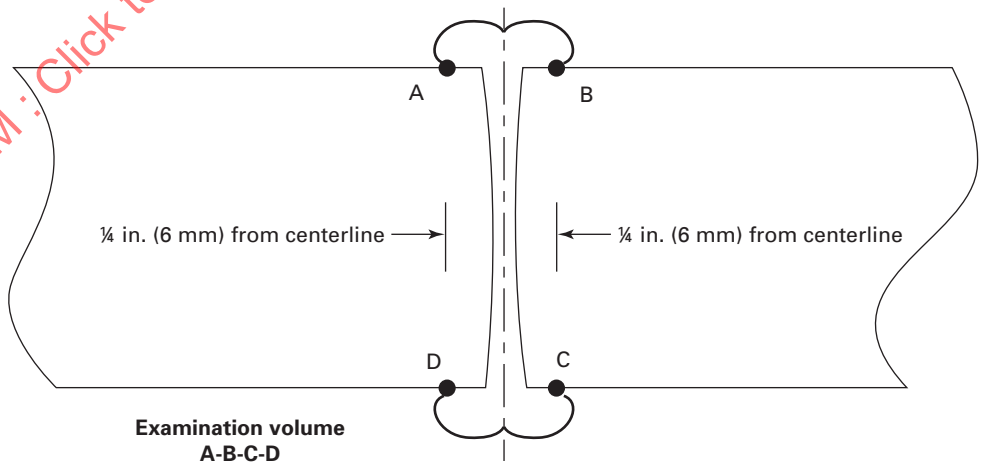

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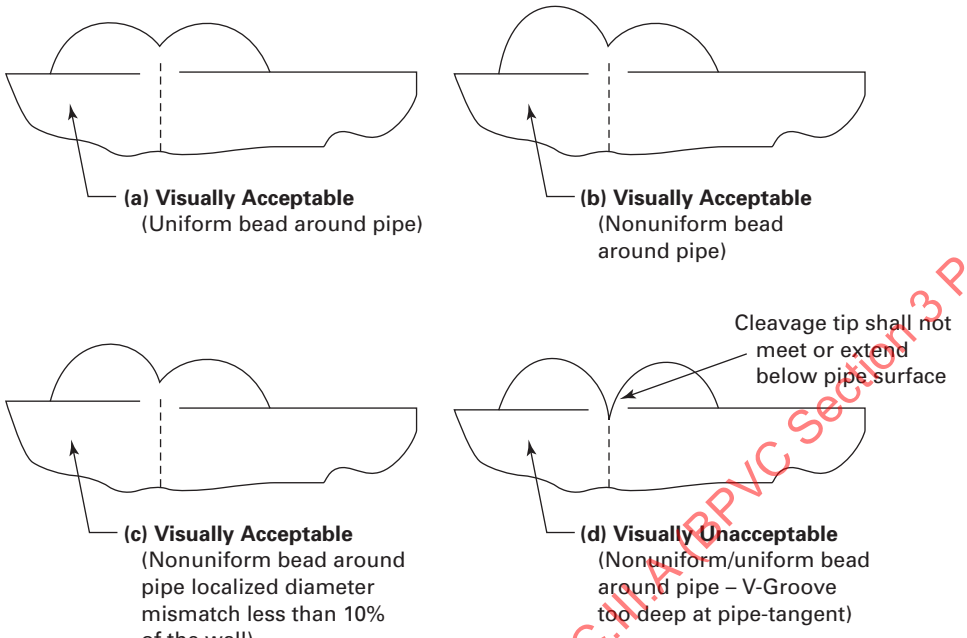
Figure XXVI-5321-1
Polyethylene Pipe Butt Fusion Joint O.D. Bead (Cross-Section View)



XXVI-5330 VOLUMETRIC EXAMINATION ACCEPTANCE CRITERIA

Any indication of a flaw not attributable to configuration that is identified in the examination volume shown in Figure XXVI-5220-1 shall cause the fused joint to be rejected.

XXVI-5400 QUALIFICATION AND CERTIFICATION OF NDE PERSONNEL

XXVI-5410 GENERAL REQUIREMENTS

(a) Organizations performing nondestructive examinations shall use personnel qualified in accordance with XXVI-5420. When these services are subcontracted by the Certificate Holder, he shall verify the qualification of personnel to the requirements of XXVI-5420. All nondestructive examinations shall be performed and the results evaluated by qualified nondestructive examination personnel.

(b) Personnel performing nondestructive examinations required by this Article shall be qualified in accordance with ND-5521 in addition to the requirements herein.

XXVI-5420 PERSONNEL QUALIFICATION REQUIREMENTS

XXVI-5421 Visual Examination

(a) Personnel performing visual examinations on material receipt and of completed fused joints shall be qualified in accordance with XXVI-5410(b) and trained in accordance with (b).

(b) All personnel performing visual examinations required by this Article shall receive the same training as required for the fusing machine operator as described in Supplement XXVI-A. This training shall include the use of a fusing machine to make a fused joint. This joint is not required to be tested for qualification. This training shall be documented on a qualification record.

(c) All personnel performing visual examinations required by this Appendix shall be given a practical examination of physical samples of visually acceptable and unacceptable fused joints. A sample set including flaws representative of unacceptable conditions (e.g., Figure XXVI-5321-1, Supplement XXVI-B) shall be used. The visual examination procedure shall be used, and a passing grade of 80% detection of the intended flaws within the demonstration set is required. The practical examination shall be administered by an individual qualified to Level III for Visual Examination (in accordance with ND-5520) or their designee. The practical examination results shall be documented on a qualification record.

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XXVI-5422 Volumetric Examination

(a) Personnel performing volumetric examinations required by this Appendix shall be qualified in accordance with XXVI-5410(b).

(b) Volumetric examination personnel shall demonstrate their capability to detect flaws by performance demonstration using the qualified procedure in accordance with the following requirements:

(1) The demonstration specimens shall be in accordance with XXVI-5114(a).

(2) The demonstration specimen set shall contain at least one of each minimum and maximum flaw size at each of the following locations: I.D. surface-connected, O.D. surface-connected, and embedded flaw.

(c) The Certificate Holder shall be responsible for reviewing the procedure and demonstration results to validate that the range of the essential variables of the procedure were included in the demonstration.

(d) This examination shall be administered by a Level III for volumetric examination (in accordance with ND-5520) or designee. The practical examination results shall be documented on a qualification record.

XXVI-5423 Certification of Personnel

Certification of NDE personnel shall be in accordance with ND-5522.

XXVI-5424 Verification of NDE Personnel Certification

Verification of NDE personnel shall be in accordance with ND-5523.

XXVI-5500 RECORDS

The following NDE records shall be retained by the Certificate Holder and provided to the Owner upon completion of construction:

(a) all NDE procedure qualification records

(b) visual NDE personnel qualification records and certifications

(c) volumetric NDE personnel qualification records identified in SNT-TC-1A, para. 9.4 including certifications

(d) all visual NDE examination records and results

(e) all volumetric NDE examination records and results, including encoded data

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ARTICLE XXVI-6000
TESTING

XXVI-6100 GENERAL REQUIREMENTS

XXVI-6110 PRESSURE TESTING

XXVI-6111 Scope of Pressure Testing

All pressure-retaining portions of completed piping systems shall be pressure tested except as specified below. Portions of piping systems that are exempt shall be identified in the Design Specification and Data Report Form. The Design Specification shall be available to the Inspector when the balance of the system is hydrostatically tested.

Bolts, studs, nuts, washers, and gaskets are exempt from pressure testing.

XXVI-6112 Pneumatic Testing

A pneumatic test shall not be permitted.

XXVI-6113 Witnessing of Pressure Tests

Pressure testing shall be performed in the presence of the Inspector.

XXVI-6114 Time of Pressure Testing

XXVI-6114.1 Piping System Pressure Test. The pressure-retaining portion of the system shall be pressure tested prior to initial operation. The pressure test may be performed progressively on installed portions of the system, which may then be buried, provided this is documented in the Owner's Quality Assurance Program and is acceptable to the Inspector.

XXVI-6114.2 Piping Subassembly Pressure Test. Piping subassemblies may be tested provided

(a) the test pressure is in accordance with the requirements of XXVI-6221(a)

(b) the pressure test is performed in a manner that, in the subassembly under test, will simulate the loadings present when the completed piping system is installed and pressurized

(c) each piping subassembly pressure test is performed by a Certificate Holder and performed in the presence of the Inspector

XXVI-6115 Machining After Pressure Test

Provided there is no infringement on t_{Design}, removal of an additional amount of material less than or equal to 5% of $t_{fab\ min}$ in pipe IPS 4 (DN 100) and smaller and less than or equal to 0.040 in. (1.0 mm) in pipe greater than IPS 4 (DN 100) shall be permitted after pressure test.

XXVI-6120 PREPARATION FOR TESTING

XXVI-6121 Exposure of Joints

All fused joints shall be uninsulated and exposed for inspection during the test.

XXVI-6122 Restraint or Isolation of Expansion Joints

Expansion joints shall be provided with temporary restraints, if required, for the additional pressure load under test.

XXVI-6123 Isolation of Equipment Not Subjected to Pressure Test

Equipment that is not to be subjected to the pressure test shall be either disconnected from the piping subassembly or system or isolated during the test by a blind flange or similar means. Valves may be used for isolation.

XXVI-6124 Treatment of Flanged Joints Containing Blanks

Flanged joints at which blanks are inserted to isolate other equipment during the test shall not be required to be retested.

XXVI-6125 Precautions Against Test Medium Expansion

If a pressure test is to be maintained for a period of time and the test medium in the system fluid is subject to thermal expansion, precautions shall be taken to avoid excessive test pressure.

XXVI-6126 Check of Test Equipment Before Applying Pressure

The test equipment shall be examined before pressure is applied to ensure that it is tight and that all low-pressure filling lines and other items that should not be subjected to the test pressure have been disconnected or isolated.

XXVI-6200 HYDROSTATIC TESTS

The requirements of this subarticle apply to all piping systems or piping subassemblies.

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XXVI-6210 HYDROSTATIC TEST PROCEDURE

XXVI-6211 Venting During Fill Operation

The piping subassembly or piping system in which the test is to be conducted shall be vented during the filling operation to minimize air pocketing.

XXVI-6212 Test Medium and Test Temperature

(a) Water shall be used for the hydrostatic test.

(b) The test shall be conducted at an ambient temperature that is within the temperature limits of the system design. The test pressure shall not be applied until the piping and pressurizing fluid are at approximately the same temperature.

XXVI-6220 HYDROSTATIC TEST PRESSURE REQUIREMENTS

XXVI-6221 Minimum Hydrostatic Test Pressure

(a) The system shall be hydrostatically tested at no less than 1.5 times the Design Pressure + 10 psi (70 KPa) for 4 hr prior to leakage inspection.

(b) Valves shall be hydrostatically tested in accordance with the rules of ND-3500.

(c) As an alternative to (a), piping between the discharge side of a centrifugal pump and the first shutoff valve may be hydrostatically tested at the shutoff head of the pump. The pressure shall be maintained for a sufficient time to permit examination of all fused joints.

XXVI-6222 Maximum Permissible Pressure

When pressure testing a system, the induced stresses shall not exceed the minimum specified Hydrostatic Design Basis (HDB) for any item in the system.

XXVI-6223 Hydrostatic Test Pressurization and Holding Time

(a) The pressure in the test section shall be gradually increased at a rate of between 5 psig/min (35 KPa gage/min) minimum and 20 psig/min (140 KPa gage/min) maximum. Pressure shall be held at the test pressure for 4 hr, during which time make-up water may be added to maintain pressure due to initial expansion.

(b) After the 4-hr hold time, the test pressure shall be reduced by 10 psig (70 KPa), and make-up water may no longer be added to maintain pressure. The system pressure shall then be monitored for at least 1 hr, during which time there shall be no reduction in pressure greater than 5% of the test pressure.

(c) The total elevated test time greater than normal operating pressure, including initial expansion and time at test pressure, shall not exceed 8 hr. If the pressure test is not completed in that time, the section shall be depressurized and not repressurized for at least 8 hr.

XXVI-6224 Examination for Leakage After Application of Pressure

Following the application of the hydrostatic test pressure for the required time in accordance with XXVI-6223(a), and upon reduction in test pressure in XXVI-6223(b), examination for leakage shall be performed.

(a) Each fused joint and all accessible external exposed surfaces shall be examined for leakage while at the hydrostatic test pressure.

(b) For items whose external surfaces are inaccessible for direct visual examination, only the examination of the surrounding area (including areas or surfaces located underneath the items) for evidence of leakage shall be required.

(c) There shall be no leakage at fused joints or through the pressure boundary except as permitted in (d).

(d) Leakage of temporary gaskets and seals, installed for the purpose of conducting the hydrostatic test that will later be replaced, may be permitted unless the leakage exceeds the capacity to maintain system test pressure during the required examination. Other leaks, such as from permanent seals, seats, and gasketed joints may be permitted when specifically allowed by the Design Specifications. Leakage from temporary seals or leakage permitted by the Design Specification shall be directed away from the surface of the piping to avoid masking leaks from other joints.

(e) The examination shall be witnessed by the Inspector.

XXVI-6300 PRESSURE TEST GAGES

XXVI-6310 REQUIREMENTS FOR PRESSURE TEST GAGES

XXVI-6311 Types of Gages to Be Used and Their Location

Pressure test gages used in pressure testing shall be indicating pressure gages and shall be connected directly to the piping. If the indicating gage is not readily visible to the operator controlling the pressure applied, an additional indicating gage shall be provided where it will be visible to the operator throughout the duration of the test. For systems with a large volumetric content, it is recommended that a recording gage be used in addition to the indicating gage.

XXVI-6312 Range of Indicating Pressure Gages

(a) Analog-type indicating pressure gages used in testing shall be graduated over a range no less than 1.5 times nor more than four times the test pressure.

(b) Digital-type pressure gages may be used without range restriction provided the combined error due to calibration and readability does not exceed 1% of the test pressure.

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XXVI-6313 Calibration of Pressure Gages

(a) All test gages shall be calibrated against a standard dead weight tester or a calibrated master gage. The test gages shall be calibrated before each test or series of tests. A series of tests is that group of tests, using the same pressure test gage or gages, which is conducted within a period not exceeding 2 weeks.

(b) The test gages shall be postcalibrated or verified against a standard dead weight tester or a calibrated master gage after each successful test or series of tests and prior to placing the system in service.

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ARTICLE XXVI-7000
OVERPRESSURE PROTECTION

The requirements of Article ND-7000 shall be met.

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ARTICLE XXVI-8000
NAMEPLATES, STAMPING, AND REPORTS

XXVI-8100 GENERAL REQUIREMENTS

XXVI-8110 SCOPE

The requirements for nameplates, stamping with the Certification Mark and Designator, and reports for components constructed in accordance with this Appendix shall be in accordance with Article NCA-8000 with the following exceptions:

(a) The attachment of nameplates shall be performed using an adhesive or corrosion resistance wire that is compatible with and will not degrade the polyethylene material.

(b) No indentation stamping is allowed on the polyethylene surface, and all marking shall be performed with a metallic paint marker or stenciling marker.

(c) The Polyethylene Material Manufacturer is permitted to apply the standard print line identifier to his piping product using a thermal process.

(d) Fittings fabricated using fusing shall be furnished with Data Report Form NM(PE)-2 as required by XXVI-2230.

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ARTICLE XXVI-9000
GLOSSARY

XXVI-9100 GLOSSARY

Refer to Section IX, QG-109, for definitions applicable to the fusing process. All other definitions shall be as given in Article NCA-9000 with the following additions:

Hydrostatic Design Basis (HDB): one of a series of established stress values for a compound.

Hydrostatic Design Stress (HDS): the estimated maximum tensile stress the material is capable of withstanding continuously with a high degree of certainty that failure of the pipe will not occur. This stress is circumferential when internal hydrostatic water pressure is applied.

lot: the quantity of

(a) Polyethylene Source Material documented on the Certificate of Analysis (C of A) and related traceability documentation.

(b) Polyethylene Material documented on the Certified Product Test Report (CPTR).

modulus of soil reaction, E′: the soil reaction modulus is a proportionality constant that represents the embedment soil's resistance to ring deflection of pipe due to earth pressure. *E′* has been determined empirically from field deflection measurements by substituting site parameters (e.g., depth of cover, soil weight) into Spangler's equation and "back calculating" *E′*.

polyethylene (PE): a polyolefin composed of polymers of ethylene. It is normally a translucent, tough, waxy solid that is unaffected by water and a large range of chemicals. There are three general classifications: low density, medium density, and high density.

void free: free from any detectable voids using the volumetric examination method approved by the Owner or his designee.

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MANDATORY APPENDIX XXVI
SUPPLEMENTS

SUPPLEMENT XXVI-I POLYETHYLENE STANDARDS AND SPECIFICATIONS

XXVI-I-100 ACCEPTABLE POLYETHYLENE (PE) STANDARDS

The PE material standards listed in Table XXVI-I-100-1 are acceptable to the extent invoked by Mandatory Appendix XXVI.

SUPPLEMENT XXVI-II ULTRASONIC EXAMINATION OF HIGH DENSITY POLYETHYLENE

XXVI-II-400

NOTE: Paragraph numbers relate to the applicable paragraphs of ASME Section V, Article 4, similar to existing Mandatory Appendices of that Article. Skipped and omitted numbers indicate no change to the corresponding paragraphs in Article 4.

Table XXVI-I-100-1
PE Standards and Specifications Referenced in Text

Standard	Subject	Edition
	ASTM Standards	
D488	Density Measurement Using Ultrasound	2008
D638	Tensile Properties of Plastics	2010
D792	Density and Specific Gravity (Relative Density) of Plastics	2008
D1238	Melt Flow Rates of Thermoplastics	2010
D1505	Density of Plastics	2010
D1599	Short-Time Hydraulic Pressure	R2005
D1603	Carbon Black Content of Olefin Plastics	2012
D2122	Determining Dimensions of Thermoplastic Pipe	R2010
D2290	Apparent Hoop Tensile Strength	2012
D2837	Obtaining Hydrostatic Design Basis	2011
D3035	Standard Specification for Polyethylene (PE) Plastic Pipe (DR-PR) Based on Controlled Outside Diameter	2012e1
D3261	Standard Specification for Butt Heat Fusion Polyethylene (PE) Plastic Fittings for Polyethylene (PE) Plastic Pipe and Tubing	2012
D3350	Specification for Plastic Pipe and Fitting Material	2012
D4218	Determining Carbon Black Content in PE Compounds	R2008
F714	Specification for Polyethylene Pipe Based on Outside Diameter	2010
F1473	Notch Tensile Test for Slow Crack Growth	2011
F2206	Specification for Fabricated Fittings	2011
F2880	Lap-Joint Type Flange Adapters (applies only to flange adapter hub dimensional requirements)	2011a
	PPI Documents	
PPI TR-3	Developing Hydrostatic Design, Pressure Design, and Strength Design Bases and Minimum Required Strength Ratings	2010a
PPI TR-4	Listing of Hydrostatic Design, Pressure Design, and Strength Design Bases and Minimum Required Strength Ratings. (The latest listing of products in Table I-A.13-PE4710 Materials may be used for PE4710 HDB and HDS equal to 1,600 psi and 1,000 psi, respectively.)	Latest version

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XXVI-II-410 Scope

This Supplement describes the requirements for examination of butt fusion joints in HDPE using encoded pulse echo with either Phased Array (PA) or time of flight diffraction (TOFD) ultrasonic techniques.

XXVI-II-420 General

The requirements of Section V, Article 4, including Mandatory Appendix III or Mandatory Appendix V of that Article, as applicable, shall apply except as modified herein.

XXVI-II-421

XXVI-II-421.1 Procedure Qualification. The requirements of Table XXVI-II-421, plus Section V, Article 4, Table T-421 and either Section V, Article 4, Mandatory Appendix III, Table III-422 or Section V, Article 4, Mandatory Appendix V, Table V-421, as applicable, shall apply.

XXVI-II-422 Scan Plan

A scan plan (documented examination strategy) shall be provided showing search unit placement and movement that provides a standardized and repeatable methodology for the examination. In addition to the information in Section V, Article 4, Table T-421, and, as applicable, Section V, Article 4, Mandatory Appendix III, Table III-422 or Section V, Article 4, Mandatory Appendix V, Table V-421, the scan plan shall include beam angles and directions with respect to the weld axis reference point, weld joint geometry, and examination areas or zones.

XXVI-II-430 Equipment

XXVI-II-431 Instrument Requirements

XXVI-II-431.1 Instrument. The requirements of Section V, Article 4, T-431, and Section V, Article 4, Mandatory Appendix III, III-431.1 shall apply. In addition, when using PA ultrasonic examination, the following shall apply:

(a) An ultrasonic array controller shall be used.

(b) The instrument shall be capable of operation at frequencies over the range of at least 1 MHz to 7 MHz and shall be equipped with a stepped gain control in units of 2 dB or less and a maximum gain of at least 60 dB. The instrument shall have a minimum of 32 channels.

(c) The digitization rate of the instrument shall be at least five times the search unit center frequency.

(d) Compression setting shall be not greater than that used during qualification of the procedure.

XXVI-II-431.2 Data Display and Recording. When performing TOFD, the requirements of Section V, Article 4, Mandatory Appendix III, III-431.2 shall apply. When performing PA ultrasonic examination, the following shall apply:

(a) The instrument shall be able to select an appropriate portion of the time base within which A-scans are digitized.

(b) The instrument shall be able to display A-, B-, C-, D-, and S-scans in a color palette able to differentiate between different amplitude levels.

(c) The equipment shall permit storage of all A-scan waveform data, with a range defined by gates, including amplitude and time-base details.

(d) The equipment shall also store positional information indicating the relative position of the waveform with respect to adjacent waveform(s), i.e., encoded position.

XXVI-II-432 Search Units

The requirements of Section V, Article 4, T-432-1, and Section V, Article 4, Mandatory Appendix III, III-432.1 shall apply. In addition, when using PA ultrasonic examination, the following shall apply:

(a) The nominal frequency shall be from 1 MHz to 7 MHz, unless variables, such as production crystalline microstructure, require the use of other frequencies to ensure adequate penetration or better resolution.

(b) Longitudinal wave mode shall be used.

(c) The number of elements used shall be between 32 and 128.

(d) Search units with angled wedges may be used to aid coupling of the ultrasound into the inspection area.

Table XXVI-II-421
Requirements of an Ultrasonic Examination Procedure for HDPE Techniques

Requirement (as Applicable)	Essential Variable	Nonessential Variable
Scan plan	X	...
Examination technique(s)	X	...
Computer software and revision	X	...
Scanning technique (automated vs. semiautomated)	X	...
Flaw characterization methodology	X	...
Flaw sizing (length) methodology	X	...
Scanner (Mfg. and Model) and adhering and guiding mechanism	...	X
Search unit mechanical fixturing device	X	...

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XXVI-II-433 Couplant

XXVI-II-433.2 Control of Contaminants. Couplants used on HDPE shall not contain oxidizers, grease, and motor oils.

XXVI-II-434 Calibration Blocks

XXVI-II-434.1 General.

XXVI-II-434.1.1 Reflectors. The reference reflector shall be a maximum diameter of 0.08 in. (2 mm).

XXVI-II-434.1.2 Material. The block shall be fabricated from pipe of the same pipe material designation as the pipe material to be used in production.

XXVI-II-434.1.3 Quality. In addition to the requirements of Section V, Article 4, T-434.1.3, areas that contain indications that are not attributable to geometry are unacceptable, regardless of amplitude.

XXVI-II-434.3 Piping Calibration Blocks. The calibration block as a minimum shall contain side-drilled holes (SDH) and shall be at least as thick as the component under examination. Alternative calibration block designs may be utilized provided the calibration is demonstrated as required in XXVI-II-421.1. The block size and reflector locations shall allow for calibration of the beam angles used that cover the volume of interest.

XXVI-II-460 Calibration

XXVI-II-462

XXVI-II-462.6 Temperature.

XXVI-II-462.6.1 The temperature differential between the calibration block and examination surface shall be within 18°F (10°C).

XXVI-II-464 Calibration for Piping

XXVI-II-464.1 System Calibration for Distance Amplitude Techniques.

XXVI-II-464.1.1 Calibration Block(s). Calibrations shall be performed utilizing the calibration block referenced in XXVI-II-434.3.

XXVI-II-464.1.2 Straight Beam Calibration. Not required.

XXVI-II-464.2 System Calibration for Nondistance Amplitude Techniques. Calibrations include all those actions required to ensure that the sensitivity and accuracy of the signal amplitude and time outputs of the examination system (whether displayed, recorded, or automatically processed) are repeated from examination to examination. Calibration shall be by use of the calibration block specified in XXVI-II-434.3.

XXVI-II-467 Calibration Confirmation

XXVI-II-467.1 System Changes. When any part of the examination system is changed, a calibration check shall be made on the calibration block to verify that distance range point and sensitivity setting(s) of the calibration reflector with the longest sound path used in the calibration satisfy the requirements of XXVI-II-467.3.

XXVI-II-467.2 Calibration Checks. A calibration check on at least one of the reflectors in the calibration block or a check using a simulator shall be performed at the completion of each examination or series of similar examinations and when examination personnel (except for automated equipment) are changed. The distance range and sensitivity values recorded shall satisfy the requirements of XXVI-II-467.3.

XXVI-II-467.2.1 Temperature Variations. If during the course of the examination, the temperature between the most recent calibration and component temperature exceeds ±18°F (10°C), calibration is required.

NOTE: Interim calibration checks between the required initial calibration and final calibration check may be performed. The decision to perform interim calibration checks should be based on ultrasonic instrument stability (analog vs. digital), the risk of having to conduct reexaminations, and the benefit of not performing interim calibration checks.

XXVI-II-467.3 Confirmation Acceptance Values.

XXVI-II-467.3.1 Distance Range Points. If the distance range point for the deepest reflector used in the calibration has moved by more than 10% of the distance reading or 5% of full sweep, whichever is greater, correct the distance range calibration, and note the correction in the examination record. All recorded indications since the last valid calibration or calibration check shall be reexamined and their values shall be changed on the data sheets or rerecorded.

XXVI-II-467.3.2 Sensitivity Settings. If any sensitivity setting for the deepest reflector used in the calibration has changed by 4 dB or less, compensate for the difference when performing the data analysis, and note the correction in the examination record. If the sensitivity setting has changed by more than 4 dB, the examination shall be repeated.

XXVI-II-470 Examination

XXVI-II-471 General Examination Requirements

XXVI-II-471.1 Examination Coverage. The examination area of interest is shown in Figure XXVI-5220-1.

XXVI-II-471.6 Recording

A-scan data shall be recorded for the area of interest in a form consistent with the procedure qualification and in recording increments of a maximum of

(a) 0.04 in. (1 mm) for material <3-in. (75-mm) thick

(b) 0.08 in. (2 mm) for material ≥3-in. (75-mm) thick.

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XXVI-II-490 Documentation

XXVI-II-492 Examination Records

For each examination, the required information of Section V, Article 4, T-492, and either Section V, Article 4, Mandatory Appendix III, III-492 or Section V, Article 4, Mandatory Appendix V, V-492, as applicable, shall be recorded. A-scan recorded data shall be retained in accordance with XXVI-5500.

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SUPPLEMENT XXVI-III DATA REPORT FORM

FORM NM(PE)-2 DATA REPORT FOR NONMETALLIC BATCH-PRODUCED PRODUCTS REQUIRING FUSING
As Required by the Provisions of the ASME Section III, Mandatory Appendix XXVI

1. Manufactured by _____
(name and address of manufacturer of nonmetallic products)

2. Manufactured for _____
(name and address of purchaser)

3. (a) Identification–Certificate Holder's Serial No. _____
(Lot No., Batch No., etc.) (print string)

_____ _____
(National Bd. No.) (year of manufacturing)

(b) Owner _____

4. Manufactured according to Material Spec. _____ Purchase Order No. _____
(ASTM)

5. Remarks _____
(brief description of fabrication)

_____

_____

CERTIFICATE OF COMPLIANCE

We certify that the statements made in this report are correct and that the products defined in this report conform to the requirements of the ASME Material specification listed above on line 4. The Certified Material Batch Reports were provided for the material covered by this report.

Certificate of Authorization (NA if Owner) No. _____ to use the _____ Symbol expires _____
(Date)

Date _____ Name _____ Signed _____
(Certificate Holder) (authorized representative)

CERTIFICATE OF INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors employed by _____
of _____ have inspected the products described in this Partial Data Report in accordance with the ASME Section III, Mandatory Appendix XXVI. By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the products described in this Partial Data Report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____ Commission _____
(Authorized Nuclear Inspector) (National Bd. No. and Endorsement)

GENERAL NOTE: Supplemental sheets in form of lists, sketches or drawings may be used provided (1) size is $8^1/_2 \times 11$ in. (2) information on items 1 – 4 on this Data Report is included on each sheet, and (3) each sheet is numbered and number of sheets is recorded at the top of this form.

(07/15)

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NONMANDATORY APPENDIX XXVI
SUPPLEMENTS

SUPPLEMENT XXVI-A FUSING MACHINE OPERATOR QUALIFICATION TRAINING

XXVI-A-100 SCOPE

(a) The major portion of the quality of polyethylene piping is determined by the skills of the fusing machine operators. When installing polyethylene piping, the quality of the fusion joints is essential for the piping system.

(b) It is important that the fusing machine operators are trained and competent in the fusing technology employed in constructing polyethylene piping systems. Continued competence of the fusing operator is covered by periodic retraining and reassessment.

(c) This document gives guidance for the training, assessment, and approval of fusing operators to establish and maintain competency in construction of polyethylene piping for pressure applications. The fusion joining technique covered by this Appendix is thermal butt fusion. This Appendix covers both the theoretical and practical knowledge necessary to ensure high-quality fusion joints.

XXVI-A-110 References

The fusion standards in this Appendix are listed in Table XXVI-A-110-1.

XXVI-A-200 TRAINING

XXVI-A-210 Training Course

(a) The course should cover all aspects of the butt fusion process, including safety, machine evaluation and maintenance, machine operation, fusing procedure specification guidelines, pressure and temperature setting, data log device operation and set-up, in-ditch fusing techniques, visual examination guidance, and data log record evaluation. The minimum course duration is 24 hr.

(b) The course will be delivered by a competent qualified trainer with a minimum of 3 yr of experience in the butt fusion processes and who has mastered the techniques involved.

(c) The trainer should have a range of fusing machines representative of the equipment encountered on worksites for installing pipes, in order for the trainee fusing operator to become acquainted with the fusing equipment commonly used. The trainee fusing operator may be trained on one of these fusing machines or on a machine from his own company if accepted by the training center. The fusing equipment must comply with the fusing machine manufacturer's specifications or ISO 12176-1 (see Table XXVI-A-110-1).

XXVI-A-220 Operator Assessment

The trainee fusing operator who has followed a training course as described above should then pass a theoretical and practical assessment in order to be qualified as a fusing operator for polyethylene systems. The assessor should not be the trainer, but should have the same assessment qualifications as the trainer shown above.

XXVI-A-230 Training Curriculum

(a) The training course should be comprised of any combination of fusing packages based on the requirements of utility or pipeline operators. These packages may be given as individual modules or combined to suit requirements. The course shall include safety training related to the fusing process and equipment.

(b) All consumables and tools necessary for the training package should be available during the training session. The pipes and fittings to be used shall conform to the ASTM product forms permitted by this Appendix.

(c) The lessons should be designed so that the trainee fusing operator learns to master the fusing technique and attains a good working knowledge of the piping system materials and practical problems encountered when fusing pipe in the field. The fusing operator should receive a written manual covering all the elements dealt with in the training.

Table XXVI-A-110-1
Fusion Standards and Specifications Referenced in Text

Standard	Subject	Edition
ASTM F2620	Heat Fusion Joining of PE Pipe and Fittings (for reference only)	2009
ISO 12176-1	Plastic Pipe and Fittings — Equipment for Fusion Joining	2006
ISO TR 19480	Guidance for Training and Assessment of Fusion Operators	2005
PPI TR-33	Generic Butt Fusion Procedure (for reference only)	2012

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(d) The theoretical course should deal with general information in connection with raw materials, pipes and fittings, and also with theoretical knowledge about preparation, tools and devices, joining components, different materials, different diameter ratios, and correct and incorrect parameters. The safety course should include information concerning the fusing process, such as protective clothing, general safety, regulations for electrical equipment, handling heater plates, etc. Areas of study should include but not be limited to the following:

(1) Butt fusion joining.

(-a) principles of fusion

(-b) straight/coiled pipes, service lines, main lines, etc.

(-c) components: pipes, flange adapters, saddle fittings, other fittings

(-d) butt fusing equipment: manual, semiautomatic, and automatic machines

(-e) joint preparation: cleaning, rounding, alignment, facing, etc.

(-f) butt fusion cycle: diagram showing pressure, time, and temperature relationships

(-g) failure modes: understanding and avoiding possible mistakes

(-h) test methods: visual examination; high-speed, tensile-impact test; bending test; hydrostatic test; data log recording/evaluation; etc.

(2) The trainee fusing operator should be familiar with the butt fusion joining technique and fusing procedure specification by making a sufficient number of butt fusion joints. In some cases, the fusing technique may vary slightly according to diameter, material, or other factors. In such cases, the trainee fusing operator should also be made familiar with the various techniques.

(3) The trainee should start by making a butt joint between two pipes and should then learn to make butt fusion joints with pipes and fittings, such as tees, reducers, etc.

(4) The trainee should learn how to detect and avoid typical fusion defects.

(5) The trainee should learn how to assess the quality of a butt fusion joint by doing a visual examination of the butt fusion joint and comparing it with the visual guidelines published in the pipe manufacturer's heat fusion joining procedure booklet. The trainee should also compare the data log record with the fusing procedure specification to ensure the proper parameters and procedures were followed in the butt fusion process.

XXVI-A-300 ASSESSMENT AND TESTING

(a) The training program should end with a theoretical and practical examination (test piece).

(b) The content of the theoretical examination shall consist of no fewer than 20 multiple choice questions about the butt fusion process, fusing machine operation, pipe, quality examination, safety, etc., within a set period of time. A score of 80% or better is considered passing on this examination. Questions to be included but not limited to are

(1) How do you calculate the fusing machine gage pressure?

(2) What is the proper heater surface temperature range from the fusing procedure specification?

(3) What is the proper butt fusion interfacial pressure range from the fusing procedure specification?

(4) How do you calculate the drag pressure?

(5) How do you know when to remove the heater in the heating cycle?

(6) How long do you leave the pipe ends together under pressure in the cooling cycle?

(7) What is the difference between IPS pipe and DIPS pipe?

(8) How do I determine the hydraulic fusing machine's total effective piston area?

(9) How is the total effective piston area of the fusing machine used to determine the fusing machine's gage pressure for a specific pipe?

(10) How do you adjust the machine to improve the alignment of the pipe after facing?

(11) How much material should be removed from the pipe ends in the facing operation?

(12) How do you determine if the fusing machine conforms to the equipment manufacturer's specifications?

(13) How do you align the pipe in the butt fusing machine?

(14) Can you butt fuse pipe in a ditch?

(15) What is interfacial pressure?

(c) The practical examination will require the trainee fusing operator to make a fusion joint with a hydraulic butt fusing machine with a minimum pipe size of IPS 8 (DN 200) DR 11. A data acquisition device must be attached to the fusing machine and the data concerning the joint entered. The data log device shall be used to record the joint made by the trainee. The assessor shall observe the butt fusion joint and note if the proper fusing procedure specification was followed. After the joint is complete, the data log record shall be reviewed by the assessor and compared with the FPS to ensure the proper procedures were followed. The assessor will then conduct a visual examination of the joint to make sure it satisfies the visual acceptance criteria per Figure XXVI-5321-1.

(d) Trainees who pass the theoretical and practical examination shall be documented on a training record. The record should state the technique or techniques and fusing machines that were used. If the fused specimens are used for fusing operator qualification, they shall be tested in accordance with XXVI-4300.

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XXVI-A-400 REASSESSMENT

If the trainee fails one of the examinations, he should retake it after a period not shorter than 1 week. If the trainee fails the examination for the second time, the trainee should repeat the training course before taking the test again.

If the trainee fails fusing operator qualification testing, retesting shall be performed as permitted by Section IX.

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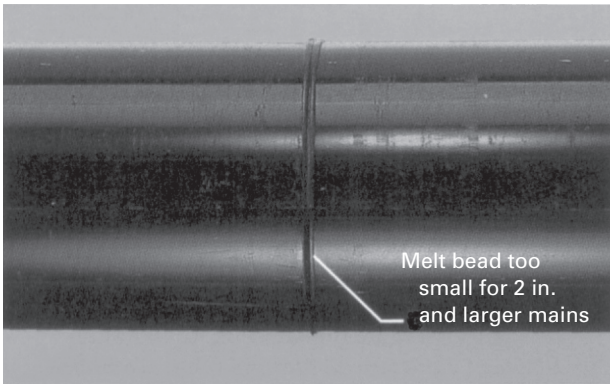
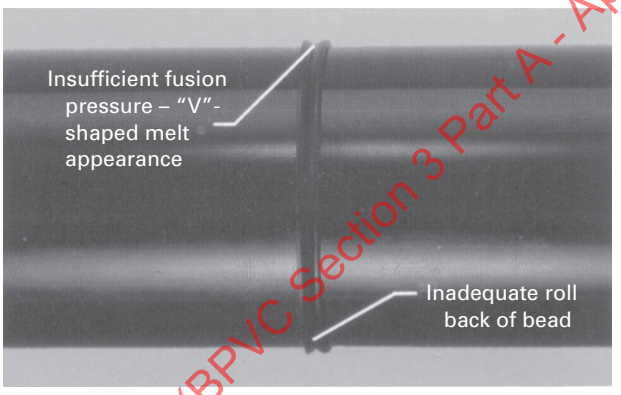
SUPPLEMENT XXVI-B UNACCEPTABLE FUSION BEAD CONFIGURATIONS

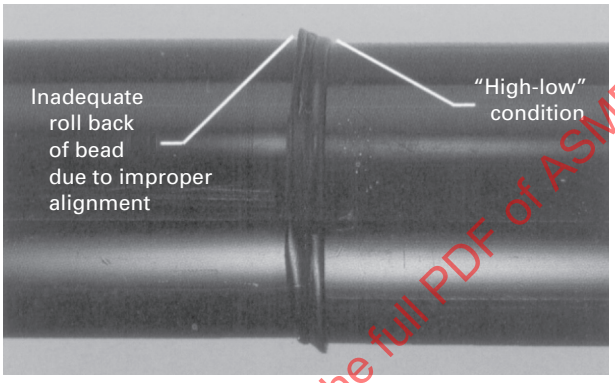
Figure XXVI-B-1
Unacceptable Fusion Bead Configurations



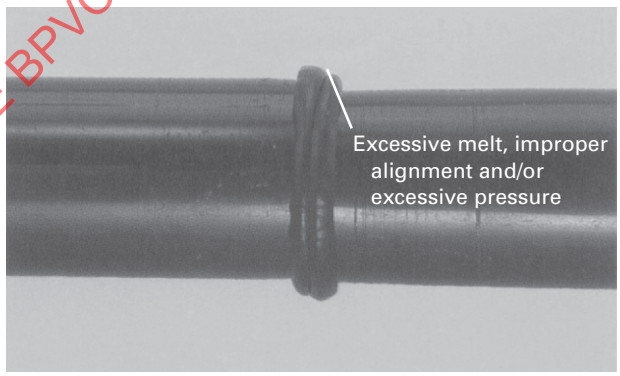
(a) Butt Fusion of Pipe
(Unacceptable appearance – insufficient melt)



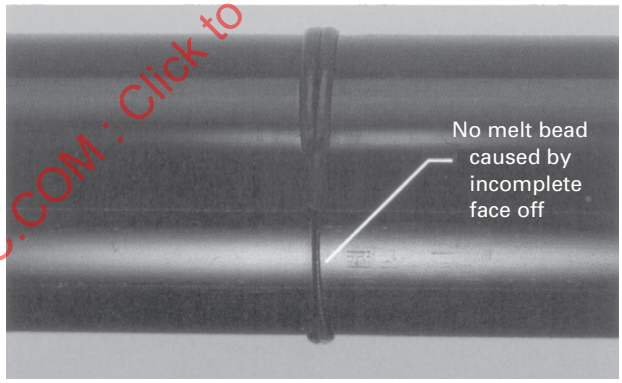
(b) Butt Fusion of Pipe
(Unacceptable appearance – inadequate roll back)



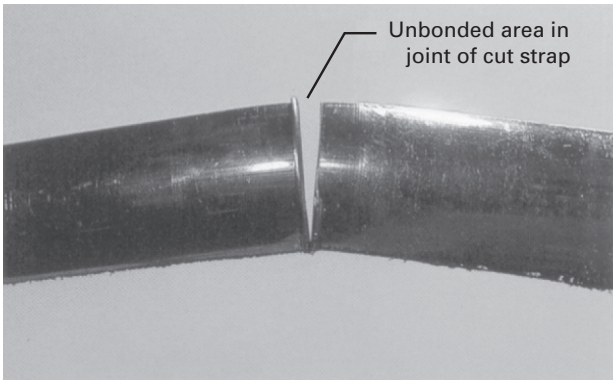
(c) Butt Fusion of Pipe
(Unacceptable appearance – improper alignment)



(d) Butt Fusion of Pipe
(Unacceptable appearance)



(e) Butt Fusion of Pipe
(Unacceptable appearance – incomplete face off)



(f) Butt Fusion of Pipe
(Unacceptable appearance – incomplete face off)

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SUPPLEMENT XXVI-C ALTERNATIVE SEISMIC ANALYSIS METHOD

XXVI-C-100 QUALIFICATION BY ANALYSIS

The buried pipe may be qualified by analysis for the effects of seismic wave passage and any seismically induced permanent or temporary movements, following the method provided in this Supplement.

Step 1. The strains from seismic wave passage and seismically induced permanent or temporary movements, if any, shall be obtained by a plant-specific geotechnical civil investigation.

Step 2. The soil strains shall be converted into an equivalent temperature rise of the buried pipe, as follows:

$$\Delta T_{\text{eq}} = \frac{\epsilon_{\text{soil}}}{\alpha}$$

Step 3. The pipe-soil system shall be modeled as a piping system constrained by soil springs.

(a) The pipe model shall consider two cases: short-term modulus (<10 hr, Table XXVI-3210-3 or Table XXVI-3210-3M) for wave passage and long-term modulus based on system design life.

(b) The soil model shall have at least a bilinear stiffness and shall consider two cases: upper and lower bounds of soil stiffness.

For guidance on modeling pipe-soil interaction, refer to ASCE, Guidelines for the Seismic Design of Oil and Gas Pipeline Systems, 1984; ASCE 4, Seismic Analysis of Safety-Related Nuclear Structures and Commentary; or American Lifelines Alliance, Guidelines for the Design of Buried Steel Pipes, July 2001, with February 2005 addendum.

Table XXVI-C-100-1
Seismic Strain Limits

DR	Allowable Strain
$DR \le 13.5$	$0.025 \times K'$
$13.5 < DR \le 21$	$0.020 \times K'$
$DR > 21$	$0.017 \times K'$

Step 4. The equivalent change of temperature, ΔT_{eq}, shall be applied to the pipe-soil model to obtain forces and moments throughout the system.

Step 5. The anticipated building seismic anchor movements, if any, shall be applied to the pipe-soil model to obtain forces and moments throughout the system.

Step 6. The anticipated seismic movements, if any, shall be applied to the pipe-soil model to obtain forces and moments throughout the system.

Step 7. The results of Steps 4 through 6 shall be combined by SRSS at each point along the piping system to obtain resultant forces and moments.

Step 8. The resultant forces and moments shall be evaluated as follows:

(a) The stresses in pipe, fittings, and fused joints shall comply with the requirements of XXVI-3410.

(b) Alternatively, the seismic-induced strain shall be determined as follows:

$$\left(\varepsilon_{\alpha}\right)_{\text{Earthquake}} = \left[\left|\sigma_E\right| + \left|\nu\left(PD/2t\right)\right|\right]/E_{\text{pipe}}$$

This strain, $(\varepsilon_a)_{\text{Earthquake}}$, shall be limited to the values listed in Table XXVI-C-100-1, where K' is defined in Table XXVI-3223-1.

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NONMANDATORY APPENDICES

NONMANDATORY APPENDIX A

ARTICLE A-1000
STRESS ANALYSIS METHODS

A-1100 INTRODUCTION

A-1110 SCOPE

(a) The Articles of this Appendix illustrate some acceptable methods of analysis to determine the stresses and stress intensities required to ensure the adequacy of a design as defined in NB-3200.

(b) The methods presented here are not intended to exclude others such as computer programs working directly with shell equations or finite element breakdowns of the component under investigation.

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ARTICLE A-2000
ANALYSIS OF CYLINDRICAL SHELLS

A-2100 INTRODUCTION

A-2110 SCOPE

(a) In this Article equations are given for stress and deformations in cylindrical shells subjected to internal pressure only. Refer to NB-3133.3 for cylindrical shells subjected to external pressure.

(b) Equations are given for bending analysis for uniformly distributed edge loads.

A-2120 SIGN CONVENTION AND NOMENCLATURE

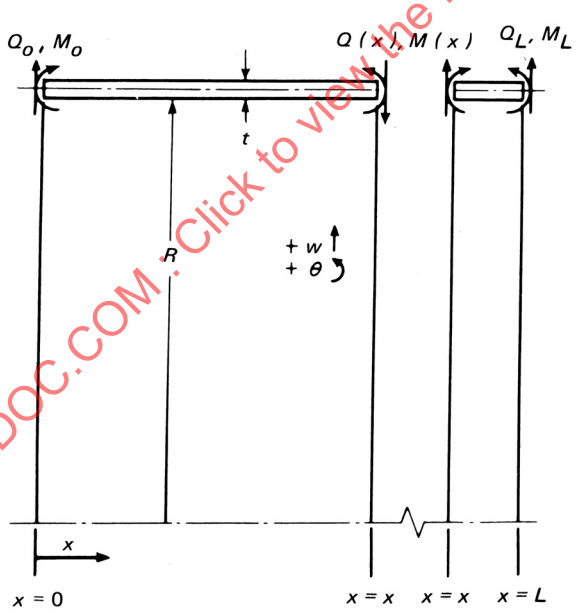
The sign convention arbitrarily chosen for the analysis of cylindrical shells in this Article is as indicated in Figure A-2120-1. Positive directions assumed for pertinent quantities are indicated.

The symbols and sign convention adopted in this Article for the analysis of cylindrical shells are defined as follows:

B_{11} = B_{11} (βL)
= $(\sinh 2\beta L - \sin 2\beta L)/2(\sinh^2 \beta L - \sin^2 \beta L)$

B_{12} = B_{12} (βL)
= $(\cosh 2\beta L - \cos 2\beta L)/2(\sinh^2 \beta L - \sin^2 \beta L)$

Figure A-2120-1



B_{22} = B_{22} (βL)
= $(\sinh 2\beta L + \sin 2\beta L)/2(\sinh^2 \beta L - \sin^2 \beta L)$

D = $Et^3/12(1 - \nu^2)$, in.-lb (N·mm)

E = modulus of elasticity

$f_1(\beta x)$ = $e^{-\beta x} \cos \beta x$

$f_2(\beta x)$ = $e^{-\beta x}(\cos \beta x - \sin \beta x)$

$f_3(\beta x)$ = $e^{-\beta x}(\cos \beta x + \sin \beta x)$

$f_4(\beta x)$ = $e^{-\beta x} \sin \beta x$

$F_{11}(\beta x)$ = $(\cosh \beta x \sin \beta x - \sinh \beta x \cos \beta x)/2$

$F_{12}(\beta x)$ = $\sinh \beta x \sin \beta x$

$F_{13}(\beta x)$ = $(\cosh \beta x \sin \beta x + \sinh \beta x \cos \beta x)/2$

$F_{14}(\beta x)$ = $\cosh \beta x \cos \beta x$

G_{11} = G_{11} (βL)
= $-(\cosh \beta L \sin \beta L - \sinh \beta L \cos \beta L)/(\sinh^2 \beta L - \sin^2 \beta L)$

G_{12} = G_{12} (βL)
= $-2 \sinh \beta L \sin \beta L/(\sinh^2 \beta L - \sin^2 \beta L)$

G_{22} = G_{22} (βL)
= $-2(\cosh \beta L \sin \beta L + \sinh \beta L \cos \beta L)/\sinh^2 \beta L - \sin^2 \beta L)$

L = length cylinder used as subscript to denote evaluation of a quantity at end of cylinder removed from reference end

M = longitudinal bending moment per unit length of circumference, in.-lb/in. (N·mm/mm)

o = used as subscript to denote evaluation of a quantity at reference end of cylinder, $x = 0$

p = internal pressure

Q = radial shearing forces per unit length of circumference, lb/in. (N·mm)

R = inside radius

S = stress intensity

t = thickness of cylinder

w = radial displacement of cylinder wall, in. (mm)

x = axial distance measured from the reference end of cylinder

Y = ratio of outside radius to inside radius

Z = ratio of outside radius to an intermediate radius

β = $[3(1 - \nu^2)/(R + t/2)^2 t^2]^{1/4}$, in.$^{-1}$ (mm^{-1})

θ = rotation of cylinder wall, rad
= dw/dx

ν = Poisson's ratio

σ_l = longitudinal (meridional) stress component

σ_r = radial stress component

σ_t = tangential (circumferential) stress component

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A-2200 STRESS INTENSITIES, DISPLACEMENTS, BENDING MOMENTS, AND LIMITING VALUES

A-2210 PRINCIPAL STRESSES AND STRESS INTENSITIES DUE TO INTERNAL PRESSURE

A-2211 Loading Effects Considered

In this Subarticle equations are given for principal stresses and stress intensities resulting from uniformly distributed internal pressure in cylindrical shells. The effects of discontinuities in geometry and loading are not included and should be evaluated independently. The stresses resulting from all effects shall be combined by superposition.

A-2212 Principal Stresses

The principal stresses developed at any point in the wall of a cylindrical shell due to internal pressure are given by the following equations:

$$\sigma_1 = \sigma_t = p\left(1 + Z^2\right)/\left(Y^2 - 1\right) \quad (1)$$

$$\sigma_2 = \sigma_l = p/\left(Y^2 - 1\right) \quad (2)$$

$$\sigma_3 = \sigma_r = p\left(1 - Z^2\right)/\left(Y^2 - 1\right) \quad (3)$$

A-2220 STRESS INTENSITIES

A-2221 General Primary Membrane Stress Intensity

The general primary membrane stress intensity developed in a cylindrical shell as a result of internal pressure is given by the equation:

$$S = \left(pR/t\right) + \left(p/2\right) \quad (4)$$

A-2222 Maximum Value of Primary Plus Secondary Stress Intensity

The maximum value of the primary plus secondary stress intensity in a cylindrical shell as a result of internal pressure occurs at the inside surface and is given by the equation:

$$S = 2pY^2/\left(Y^2 - 1\right) \quad (5)$$

A-2223 Values of Radial Stress Used

Note that in evaluating the general primary membrane stress intensity, the average value of the radial stress has been taken as $-p/2$. This has been done to obtain a result consistent with burst pressure analyses. On the other hand, the radial stress value used in A-2222 is $-p$, the value at the inner surface, since the purpose of that quantity is to control local behavior.

A-2230 BENDING ANALYSIS FOR UNIFORMLY DISTRIBUTED EDGE LOADS

A-2231 Behavior of Shells Subjected to Bending Moments

The equations in this Subarticle describe the behavior of cylindrical shells when subjected to the action of bending moments, M, in.-lb/in. (N·mm/mm) of circumference, and radial shearing forces, Q, lb/in. (N·mm) of circumference, uniformly distributed at the edges and acting at the mean radius of the shell. The behavior of shells due to all other loadings must be evaluated independently and combined by superposition.

A-2240 DISPLACEMENTS, BENDING MOMENTS, AND SHEARING FORCES IN TERMS OF CONDITIONS AT REFERENCE EDGE ($x = 0$)

A-2241 Equations for Conditions of Any Axial Location

The radial displacement, $w(x)$, the angular displacement or rotation, $\theta(x)$, the bending moments, $M(x)$, and the radial shearing forces, $Q(x)$ at any axial location of the cylinder are given by the following equations in terms of w_o, θ_o, M_o, and Q_o.

$$w(x) = \left(Q_o/2\beta^3 D\right)F_{11}(\beta x) + \left(M_o/2\beta^2 D\right)F_{12}(\beta x) + \left(\theta_o/\beta\right)F_{13}(\beta x) + w_o F_{14}(\beta x) \quad (6)$$

$$\theta(x)/\beta = \left(Q_o/2\beta^3 D\right)F_{12}(\beta x) + 2\left(M_o/2\beta^2 D\right)F_{13}(\beta x) + \left(\theta_o/\beta\right)F_{14}(\beta x) - 2w_o F_{11}(\beta x) \quad (7)$$

$$M(x)/2\beta^2 D = \left(Q_o/2\beta^3 D\right)F_{13}(\beta x) + \left(M_o/2\beta^2 D\right)F_{14}(\beta x) - \left(\theta_o/\beta\right)F_{11}(\beta x) - w_o F_{12}(\beta x) \quad (8)$$

$$Q(x)/2\beta^3 D = \left(Q_o/2\beta^3 D\right)F_{14}(\beta x) - 2\left(M_o/2\beta^2 D\right)F_{11}(\beta x) - \left(\theta_o/\beta\right)F_{12}(\beta x) - 2w_o F_{13}(\beta x) \quad (9)$$

A-2242 Equations When Cylinder Length ≥ $3/\beta$

In the case of cylinders of sufficient length, the equations in A-2241 reduce to those given below. These equations may be used for cylinders characterized by lengths not less than $3/\beta$. The combined effects of loadings at the two edges may be evaluated by applying the equations to the loadings at each edge, separately, and superposing the results.

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$$w(x) = \left(Q_o / 2\beta^3 D\right) f_1(\beta x) + \left(M_o / 2\beta^2 D\right) f_2(\beta x) \quad (10)$$

$$\theta(x) / \beta = -\left(Q_o / 2\beta^3 D\right) f_3(\beta x) - 2\left(M_o / 2\beta^2 D\right) f_1(\beta x) \quad (11)$$

$$M(x) / 2\beta^2 D = \left(Q_o / 2\beta^3 D\right) f_4(\beta x) + \left(M_o / 2\beta^2 D\right) f_3(\beta x) \quad (12)$$

$$Q(x) / 2\beta^3 D = \left(Q_o / 2\beta^3 D\right) f_2(\beta x) - 2\left(M_o / 2\beta^2 D\right) f_4(\beta x) \quad (13)$$

A-2243 Edge Displacements and Rotations in Terms of Edge Loads

The radial displacement w_o and w_L and rotations θ_o and $-\theta_L$, developed at the edges of a cylindrical shell sustaining the action of edge loads Q_o, M_o, Q_L, and M_L, are given by the following equations:

$$w_o = \left(B_{11} / 2\beta^3 D\right) Q_o + \left(B_{12} / 2\beta^2 D\right) M_o + \left(G_{11} / 2\beta^3 D\right) Q_L + \left(G_{12} / 2\beta^2 D\right) M_L \quad (14)$$

$$-\theta_o = \left(B_{12} / 2\beta^2 D\right) Q_o + \left(B_{22} / \beta D\right) M_o + \left(G_{12} / 2\beta^2 D\right) Q_L + \left(G_{22} / 2\beta D\right) M_L \quad (15)$$

$$w_L = \left(G_{11} / 2\beta^3 D\right) Q_o + \left(G_{12} / 2\beta^2 D\right) M_o + \left(B_{11} / 2\beta^3 D\right) Q_L + \left(B_{12} / 2\beta^2 D\right) M_L \quad (16)$$

$$\theta_L = \left(G_{12} / 2\beta^2 D\right) Q_o + \left(G_{22} / 2\beta D\right) M_o + \left(B_{12} / 2\beta^2 D\right) Q_L + \left(B_{22} / \beta D\right) M_L \quad (17)$$

A-2250 LIMITING VALUE OF FUNCTIONS

A-2251 General Limiting Values of Influence Functions

The influence functions B and G, appearing in the equations in A-2243, rapidly approach limiting values as the length L of the cylinder increases. The limiting values are

$$B_{11} = B_{12} = B_{22} = 1$$
$$G_{11} = G_{12} = G_{22} = 0$$

(a) Thus, for cylindrical shells of sufficient length, the loading conditions prescribed at one edge do not influence the displacements at the other edge.

(b) In the case of cylindrical shells characterized by lengths not less than $3/\beta$, the influence functions B and G, are sufficiently close to the limiting values so that the limiting values may be used in the equations in A-2243 without significant error.

A-2252 Limiting Values of Influence Functions for Short Cylinders (15)

In the case of sufficiently short cylinders, the influence functions B and G, appearing in the equations in A-2243, are, to a first approximation, given by the following expressions:

$$B_{11} = 2 / \beta L$$
$$B_{12} = 3 / (\beta L)^2$$
$$B_{22} = 3 / (\beta L)^3$$
$$G_{11} = -1 / \beta L$$
$$G_{12} = -3 / (\beta L)^2$$
$$G_{22} = -6 / (\beta L)^3$$

Introducing these expressions for the influence functions B and G into the equations in A-2243 yields expressions identical to those obtained by the application of ring theory. Accordingly, the resultant expressions are subject to all of the limitations inherent in the ring theory, including the limitations due to the assumption that the entire cross-sectional area of the ring $t \times L$ rotates about its centroid without distortion. Nevertheless, in the analysis of very short cylindrical shells characterized by lengths not greater than $\frac{1}{2}\beta$, the expressions may be used without introducing significant error.

A-2260 PRINCIPAL STRESSES DUE TO BENDING

The principal stresses developed at the surfaces of a cylindrical shell at any axial location x due to uniformly distributed edge loads (Figure A-2120-1) are given by the equations:

$$\sigma_1 = \sigma_t(x) = Ew(x) / (R + t/2) \pm 6\nu M(x)/t^2 \quad (18)$$

$$\sigma_2 = \sigma_l(x) = \pm 6M(x) / t^2 \quad (19)$$

$$\sigma_3 = \sigma_r = 0 \quad (20)$$

In these equations, where terms are preceded by a double sign ±, the upper sign refers to the inside surface of the cylinder and the lower sign refers to the outside surface.

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ARTICLE A-3000
ANALYSIS OF SPHERICAL SHELLS

A-3100 INTRODUCTION

A-3110 SCOPE

(a) In this Article equations are given for stresses and deformations in spherical shells subjected to internal or external pressure.

(b) Equations are also given for bending analysis of partial spherical shells under the action of uniformly distributed edge forces and moments.

A-3120 NOMENCLATURE AND SIGN CONVENTION

(a) The symbols and sign convention adopted in this Article are defined as follows:

A_o = $\sqrt{1 + k_1^2}$

$B(\alpha)$ = $[(1 + \nu^2)(K_1 + K_2) - 2K_2]$

$C(\alpha)$ = $\sqrt{\sin\phi_o / \sin(\phi - \alpha)}$

D = $Et^3 / 12(1 - \nu^2)$, flexural rigidity, in.-lb (N·mm)

E = modulus of elasticity

$F(\alpha)$ = $\sqrt{\sin(\phi_o)\sin(\phi_o - \alpha)}$

H = force per unit length of circumference, perpendicular to centerline of sphere, lb/in. (N·mm)

K_1 = $1 - [(1 - 2\nu) / 2\lambda] \cot(\phi_o - \alpha)$

k_1 = $1 - [(1 - 2\nu) / 2\lambda] \cot\phi_o$

K_2 = $1 - [(1 + 2\nu) / 2\lambda] \cot(\phi_o - \alpha)$

k_2 = $1 - [(1 + 2\nu) / 2\lambda] \cot\phi_o$

l = used as a subscript to denote meridional direction

M = meridional bending moment per unit length of circumference, in.-lb/in. (N·mm/mm)

N = membrane force, lb/in. (N·mm)

o = used as a subscript to denote a quantity at the reference edge of sphere

p = uniform pressure internal or external

Q = radial shearing force per unit of circumference, lb/in. (N·mm)

R = inside radius

R_m = radius of midsurface of spherical shell

S = stress intensity

t = thickness of spherical shell

= used as a subscript to denote circumferential direction

U = ratio of inside radius to an intermediate radius

w = radial displacement of midsurface, in. (mm)

Y = ratio of outside radius to inside radius

Z = ratio of outside radius to an intermediate radius

α = meridional angle measured from the reference edge, rad

β = $[3(1 - \nu^2) / R_m^2 t^2]^{1/4}$, in.$^{-1}$ (mm^{-1})

γ_o = $\tan^{-1}(-k_1)$

δ = lateral displacement of midsurface, perpendicular to centerline of spherical shell, in. (mm)

θ = rotation of midsurface, rad

λ = βR_m

ν = Poisson's ratio

σ_l = longitudinal (meridional) stress component

σ_r = radial stress component

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σ_t = tangential (circumferential) stress component
χ = length of arc for angle α, measured from reference edge of hemisphere
ϕ = meridional angle measured from centerline of sphere, deg
ϕ_L = meridional angle of second edge, deg
= $\pi R_m \alpha / 180$
ϕ_o = meridional angle of reference edge where loading is applied, deg

(b) The sign convention is listed below and shown in Figure A-3120-1 by the positive directions of the pertinent quantities:

Sign	Convention
(p)	pressure, positive radially outward
(δ)	lateral displacement, perpendicular to λ of sphere, positive outward
(θ)	rotation, positive when accompanied by an increase in the radius or curvature, as caused by a positive moment
(M), (M_o)	moment, positive when causing tension on the inside surface
(H), (H_o)	force perpendicular to λ, positive outward
(N_t), (N_l)	membrane force, positive when causing tension

A-3200 STRESS INTENSITIES, BENDING ANALYSIS, DISPLACEMENTS, AND EDGE LOADS

A-3210 PRINCIPAL STRESSES AND STRESS INTENSITIES RESULTING FROM INTERNAL OR EXTERNAL PRESSURE

In this Subarticle equations are given for principal stresses and stress intensities resulting from uniformly distributed internal or external pressure in complete or partial spherical shells. The effects of discontinuities in geometry and loading are not included and should be evaluated independently. The stresses resulting from all effects must be combined by superposition.

A-3220 PRINCIPAL STRESSES AND STRESS INTENSITIES

A-3221 Principal Stresses Resulting From Internal Pressure

The principal stresses at any point in the wall of a spherical shell as a result of internal pressure are given by the following equations:

$$\sigma_1 = \sigma_l = p\left(Z^3 + 2\right) / 2\left(Y^3 - 1\right) \quad (1)$$

$$\sigma_2 = \sigma_t = p\left(Z^3 + 2\right) / 2\left(Y^3 - 1\right) \quad (2)$$

$$\sigma_3 = \sigma_r = p\left(1 - Z^3\right) / \left(Y^3 - 1\right) \quad (3)$$

A-3222 Stress Intensities Resulting From Internal Pressure

A-3222.1 General Primary Membrane Stress Intensity. The general primary stress intensity in a spherical shell as a result of internal pressure is given by the following equation:

$$S = 0.75p\left(Y^3 + 1\right) / \left(Y^3 - 1\right) \quad (4)$$

A-3222.2 Maximum Value of Primary Plus Secondary Stress Intensity. The maximum value of the primary plus secondary stress intensity in a spherical shell as a result of internal pressure occurs at the inside surface and is given by the equation:

$$S = 1.5p\left(Y^3\right) / \left(Y^3 - 1\right) \quad (5)$$

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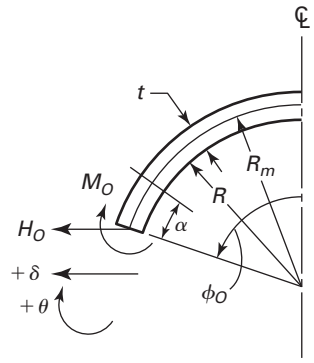
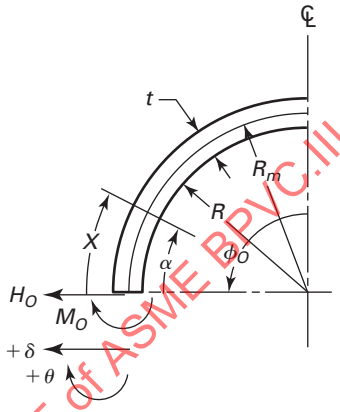
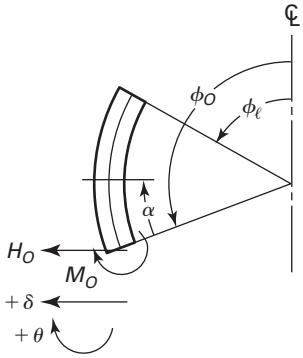
Figure A-3120-1



(a) Spherical Segment, for Values of ϕ_O:
$162/\lambda \leq \phi_O \leq 180 \text{ deg} - 162/\lambda$



(b) Hemisphere for $\phi_O = 90$ deg



(c) Frustum, for Values of ϕ_O:
$\phi_O \leq 180 \text{ deg} - 162/\lambda$ and $|\phi_O - \phi_\ell| \geq 180/\lambda$

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A-3223 Principal Stresses Resulting From External Pressure

The principal stresses at any point in the wall of a spherical shell resulting from external pressure are given by the following equations:

$$\sigma_1 = \sigma_l = -pY^3\left(U^3 + 2\right)/2\left(Y^3 - 1\right) \quad (6)$$

$$\sigma_2 = \sigma_t = -pY^3\left(U^3 + 2\right)/2\left(Y^3 - 1\right) \quad (7)$$

$$\sigma_3 = \sigma_r = pY^3\left(U^3 - 1\right)/\left(Y^3 - 1\right) \quad (8)$$

A-3224 Stress Intensities Resulting From External Pressure

A-3224.1 General Primary Membrane Stress Intensity. The general primary membrane stress intensity in a spherical shell as a result of external pressure is given by the equation:

$$S = 0.75p\left(Y^3 + 1\right)/\left(Y^3 - 1\right) \quad (9)$$

A-3224.2 Maximum Value of Primary Plus Secondary Stress Intensity. The maximum value of the primary plus secondary stress intensity in a spherical shell as a result of external pressure occurs at the inside surface and is given by the equation:

$$S = 1.5p\left(Y^3\right)/\left(Y^3 - 1\right) \quad (10)$$

NOTE: The equations in A-3223 and A-3224 may be used only if the applied external pressure is less than the critical pressure which would cause instability of the spherical shell. The value of the critical pressure must be evaluated in accordance with the rules given in NB-3133.4.

A-3230 BENDING ANALYSIS FOR UNIFORMLY DISTRIBUTED EDGE LOADS

A-3231 Scope and Limitations of Equations Given

(a) The equations in A-3230 describe the behavior of partial spherical shells of the types shown in Figure A-3120-1 when subjected to the action of meridional bending moments M_o, in.-lb/in. (N·mm/mm) of circumference, and forces H_o, lb/in. (N·mm) of circumference, uniformly distributed at the reference edge and acting at the mean radius of the shell. The effects of all other loading must be evaluated independently and combined by superposition.

(b) The equations listed in this paragraph become less accurate and should be used with caution when R_m/t is less than 10 or the opening angle limitations shown in Figure A-3120-1 are exceeded.

A-3232 Displacement, Rotation, Moment, and Membrane Force in Terms of Loading Conditions at Reference Edge

The displacement δ, the rotation θ, the bending moments M_l, M_t, and the membrane forces N_l, N_t at any location of sphere are given in terms of the edge loads M_o and H_o by the following equations:

$$\delta = M_o\left\{\frac{2\lambda^2}{Etk_1}F(\alpha)e^{-\lambda\alpha}\left[\cos(\lambda\alpha) - K_2\sin(\lambda\alpha)\right]\right\} + H_o\left\{\frac{R_m\lambda}{Etk_1}A_o\sin\phi_o\,F(\alpha)e^{-\lambda\alpha}\left[\cos\left(\lambda\alpha + \gamma_o\right) - K_2\sin\left(\lambda\alpha + \gamma_o\right)\right]\right\} \quad (11)$$

$$\theta = M_o\left\{\frac{4\lambda^3}{R_mEtk_1}C(\alpha)e^{-\lambda\alpha}\cos(\lambda\alpha)\right\} + H_o\left\{\frac{2\lambda^2}{Etk_1}A_o\sin\phi_o\,C(\alpha)e^{-\lambda\alpha}\cos\left(\lambda\alpha + \gamma_o\right)\right\} \quad (12)$$

$$M_l = M_o\left\{\frac{1}{k_1}C(\alpha)e^{-\lambda\alpha}\left[K_1\cos(\lambda\alpha) + \sin(\lambda\alpha)\right]\right\} + H_o\left\{\frac{R_m}{2\lambda k_1}A_o\sin\phi_o\,C(\alpha)e^{-\lambda\alpha}\left[K_1\cos\left(\lambda\alpha + \gamma_o\right) + \sin\left(\lambda\alpha + \gamma_o\right)\right]\right\} \quad (13)$$

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$$M_t = M_o\left\{\frac{C(\alpha)}{2\nu k_1}e^{-\lambda\alpha}\left[B(\alpha)\cos(\lambda\alpha) + 2\nu^2\sin(\lambda\alpha)\right]\right\} + H_o\left\{\frac{R_m}{4\nu\lambda k_1}A_o\sin\phi_o\,C(\alpha)e^{-\lambda\alpha}\left[B(\alpha)\cos(\lambda\alpha+\gamma_o) + 2\nu^2\sin(\lambda\alpha+\gamma_o)\right]\right\} \quad (14)$$

$$N_l = -M_o\left\{\frac{2\lambda}{R_m k_1}C(\alpha)e^{-\lambda\alpha}\sin(\lambda\alpha)\cot(\phi_o-\alpha)\right\} - H_o\left[\frac{1}{K_1}A_o\cot(\phi_o-\alpha)(\sin\phi_o)C(\alpha)e^{-\lambda\alpha}\sin(\lambda\alpha+\gamma_o)\right] \quad (15)$$

$$N_t = M_o\left\{\frac{2\lambda^2}{R_m k_1}C(\alpha)e^{-\lambda\alpha}\left[\cos(\lambda\alpha) - \left(\frac{K_1+K_2}{2}\right)\sin(\lambda\alpha)\right]\right\} + H_o\left\{\frac{\lambda}{k_1}A_o\sin\phi_o\,C(\alpha)e^{-\lambda\alpha}\left[\cos(\lambda\alpha+\gamma_o) - \left(\frac{K_1+K_2}{2}\right)\sin(\lambda\alpha+\gamma_o)\right]\right\} \quad (16)$$

A-3233 Displacement and Rotation of Reference Edge in Terms of Loading Conditions at Reference Edge

A-3233.1 At Reference Edge Where $\alpha = 0$ and $\phi = \phi_o$. The equations for the displacement and rotation (A-3232) simplify to eqs. (17) and (18).

$$\delta_o = M_o\frac{2\lambda^2\sin\phi_o}{Etk_1} + H_o\frac{R_m\lambda\sin^2\phi_o}{Et}\left(\frac{1}{k_1}+k_2\right) \quad (17)$$

$$\theta_o = M_o\frac{4\lambda^3}{R_m Etk_1} + H_o\frac{2\lambda^2\sin\phi_o}{Etk_1} \quad (18)$$

A-3233.2 When Shell Is a Full Hemisphere. In the case where the shell under consideration is a full hemisphere, eqs. A-3233.1(17) and A-3233.1(18) reduce to the following:

$$\delta_o = M_o\frac{2\lambda^2}{Et} + H_o\frac{2R_m\lambda}{Et} \quad (19)$$

$$\theta_o = M_o\frac{4\lambda^3}{R_m Et} + H_o\frac{2\lambda^2}{Et} \quad (20)$$

A-3234 Principal Stresses in Spherical Shells Resulting From Edge Loads

The principal stresses at the inside and outside surfaces of a spherical shell at any location, resulting from edge loads M_o and H_o, are given by eqs. (21), (22), and (23).

$$\sigma_1 = \sigma_l(\alpha) = \frac{N_l}{t} \pm \frac{6M_l}{t^2} \quad (21)$$

$$\sigma_2 = \sigma_t(\alpha) = \frac{N_t}{t} \pm \frac{6M_t}{t^2} \quad (22)$$

$$\sigma_3 = \sigma_r(\alpha) = 0 \quad (23)$$

In these equations, where terms are preceded by a double sign ±, the upper sign refers to the inside surface of the shell and the lower sign refers to the outside surface.

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A-3240 ALTERNATIVE BENDING ANALYSIS OF A HEMISPHERICAL SHELL SUBJECTED TO UNIFORMLY DISTRIBUTED EDGE LOADS

A-3241 Nature of Equations Given

If a less exacting but more expedient analysis of hemispherical shells is required, equations derived for cylindrical shells may be used in a modified form. The equations in A-3242 describe the behavior of a hemispherical shell as approximated by a cylindrical shell of the same radius and thickness when subjected to the action of uniformly distributed edge loads M_o and H_o at $\alpha = 0$, $x = 0$, and $\phi_o = 90$ deg.

A-3242 Displacement, Rotation, Moment, and Shear Forces in Terms of Loading Conditions at Edge

$$\delta_o = H_o / 2\beta^3 D + M_o / 2\beta^2 D \tag{24}$$

$$\theta_o = H_o / 2\beta^2 D + M_o / \beta D \tag{25}$$

$$\delta(x) = \frac{H_o \sin^2 \phi}{2\beta^3 D} f_1(\beta x) + \frac{M_o \sin \phi}{2\beta^2 D} f_2(\beta x) \tag{26}$$

$$\theta(x) = \frac{H_o \sin \phi}{2\beta^2 D} f_3(\beta x) + \frac{M_o}{\beta D} f_1(\beta x) \tag{27}$$

$$M(x) = \frac{H_o \sin \phi}{\beta} f_4(\beta x) + M_o f_3(\beta x) \tag{28}$$

$$Q(x) = H_o \sin \phi f_2(\beta x) - 2\beta M_o f_4(\beta x) \tag{29}$$

where f_1, f_2, f_3, and f_4 are defined in Article A-2000, and

$$x = \alpha R_m = \frac{\pi R_m}{180 \text{ deg}} (90 \text{ deg} - \phi)$$

A-3243 Principal Stresses in Hemispherical Shell Due to Edge Loads

The principal stresses in a hemispherical shell due to edge loads M_o and H_o, at the inside and outside surfaces of a hemispherical shell at any meridional location, are given by eqs. (30), (31), and (32).

$$\sigma_1 = \sigma_l(x) = \pm 6M(x) / t^2 \tag{30}$$

$$\sigma_2 = \sigma_t(x) = E\delta(x) / R_m \pm \nu 6M(x) / t^2 \tag{31}$$

$$\sigma_3 = \sigma_r = 0 \tag{32}$$

In these equations, where terms are preceded by a double sign ±, the upper sign refers to the inside surface of the hemisphere and the lower sign refers to the outside surface.

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ARTICLE A-4000
DESIGN CRITERIA AND EQUATIONS FOR TORISPHERICAL AND ELLIPSOIDAL HEADS

A-4100 INTRODUCTION

A-4110 SCOPE

The equations defining the curves in Figure NC-3224.6-1 are summarized in this Article. The analysis is for pressure on the concave portion of the head and does not include effects of thermal gradients and loadings other than pressure.

A-4120 NOMENCLATURE

The nomenclature adopted in this Article is defined as follows:

D = inside diameter of a head skirt, or inside length of the major axis of an ellipsoidal head
L = inside crown radius of torispherical head
P = internal design pressure
r = inside knuckle radius of torispherical head
S = membrane stress intensity limit from Section II, Part D, Subpart 1, Tables 2A and 2B multiplied by the stress intensity factors in Table NC-3217-1, psi (kPa)
t = minimum required thickness of head

A-4130 METHOD USED TO DETERMINE DESIGN PRESSURE

The maximum internal pressure capacity or required thickness of a torispherical and ellipsoidal pressure vessel head is determined from the controlling criterion of primary membrane stress, elastic–plastic collapse load, buckling collapse, and fatigue. For thick heads, where $P/S > 0.08$ (approximately $t/L = 0.04$ to 0.05), primary membrane stress dominates. For thin heads, where $t/L > 0.002$, buckling collapse is the limiting condition. For the intermediate thickness heads, where $t/L > 0.002$ up to t/L, where $P/S < 0.08$ (approximately $t/L = 0.04$ to 0.05), elastic–plastic collapse pressure and fatigue due to pressurization cycles are the determining conditions. At the present time, only design of the intermediate thickness heads is considered in this Division.

A-4140 MATHEMATICAL EXPRESSIONS FOR CURVES IN FIGURE NC-3224.6-1

The equation for computing A for given set of parameters r/D and P/S is as follows:

$$A = a_1 + a_2x + a_3x^2 + \left(b_1 + b_2x + b_3x^2\right)y + \left(c_1 + c_2x + c_3x^2\right)y^2 \quad (1)$$

where

$$x = r/D \quad (2)$$

$$y = \text{logarithm}\left(P/S\right) \quad (3)$$

Constants a_1 through c_3 are given in A-4141 for natural logarithms and in A-4142 for common base logarithms.

A-4141 Natural Logarithms

$$y = \ln\left(P/S\right) \quad (4)$$

$$t/L = e^A \quad (5)$$

and

$$a_1 = -1.2617702$$
$$b_1 = 0.66298796$$
$$c_1 = 0.26878909 \times 10^{-4}$$
$$a_2 = -4.5524592$$
$$b_2 = -2.2470836$$
$$c_2 = -0.42262179$$
$$a_3 = 28.933179$$
$$b_3 = 15.682985$$
$$c_3 = 1.8878333$$

A-4142 Common Base Logarithms

$$y = \log_{10}\left(P/S\right) \quad (6)$$

$$t/L = 10^A \quad (7)$$

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and

$$a_1 = -0.5479782$$
$$b_1 = 0.66298796$$
$$c_1 = 0.61890975 \times 10^{-4}$$
$$a_2 = -1.9771079$$
$$b_2 = -2.2470836$$
$$c_2 = -0.97312263$$
$$a_3 = 12.565520$$
$$b_3 = 15.682985$$
$$c_3 = 4.3468967$$

A-4150 SAMPLE PROBLEM

Consider a torispherical head having the parameters L = 84 in.; D = 90 in.; r = 5.5 in.; P = 200 psi; and for material SA-515 Grade 70, S = 23,300 psi. With these data and using eq. A-4140(1) and the common logarithms and constants of A-4142:

$$P/S = 0.00858369$$
$$r/D = 0.06111111 = x$$
$$y = \log_{10}(P/S) = -2.066325925$$

and

$$y^2 = 4.269702828$$
$$A = -2.013430932$$

Solving eq. A-4140(1):

$$t/L = 10^A = 0.009695474$$
$$t = 0.814 \text{ in.}$$

Direct reading of Figure NC-3224.6-1 gives the following:

$$t/L = 0.0097$$
$$t = 0.814 \text{ in.}$$

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ARTICLE A-5000
ANALYSIS OF FLAT CIRCULAR HEADS

A-5100 INTRODUCTION

A-5110 SCOPE

(a) In this Article equations are given for stresses and displacements in flat circular plates used as heads for pressure vessels.

(b) Equations are also given for stresses and displacements in these heads due to forces and edge moments uniformly distributed along the outer edge, and uniformly distributed over a circle on one face. The radius of this circle is intended to match the mean radius of an adjoining element, such as a cylinder, cone, or spherical segment.

A-5120 NOMENCLATURE AND SIGN CONVENTION

The symbols and sign conventions adopted in this Article are defined as follows:

E = elastic modulus
F = geometry constant, given in Table A-5240-1
ln = $\log_e$
M = radial bending moment, in.-lb/in. (N·mm/mm) of circumference
p = pressure
Q = radial force, lb/in. (N/mm) of circumference
R = outside radius of plate
r = radial distance from center of plate
t = thickness of plate
t_s = thickness of connecting shell at the head junction
w = radial displacement
x = longitudinal distance from midplane of plate
θ = rotation, rad
ν = Poisson's ratio
σ_l = longitudinal (axial) stress
σ_r = radial stress
σ_t = tangential (circumferential) stress

Tensile stresses are positive. The positive directions of the coordinates, radial forces, moments, and displacements are shown in Figure A-5120-1. The pressure is assumed to act on the surface where $x = t/2$.

A-5200 LOADS, DISPLACEMENTS, AND GEOMETRY CONSTANTS

A-5210 PRESSURE AND EDGE LOADS ON CIRCULAR FLAT PLATES

A-5211 Values for Which Equations Are Given

In this Subarticle equations are given for the principal stress and the deformations of flat plates under axisymmetric loading conditions.

A-5212 Pressure Loads on Simply Supported Flat Plates (15)

The principal stresses and deformations for a flat plate, simply supported at its periphery and loaded in the manner shown on Figure A-5212-1 are given for a radial location r at any point x in the cross section by eqs. (1) through (6).

Radial bending stress:

$$\sigma_r = p\frac{3(x)}{4t^3}\left[(3+\nu)(R^2 - r^2)\right] \quad (1)$$

Tangential bending stress:

$$\sigma_t = p\frac{3(x)}{4t^3}\left[(3+\nu)R^2 - (1+3\nu)r^2\right] \quad (2)$$

Longitudinal stress:

$$\sigma_l = (p/t)(x - t/2) \quad (3)$$

Figure A-5120-1

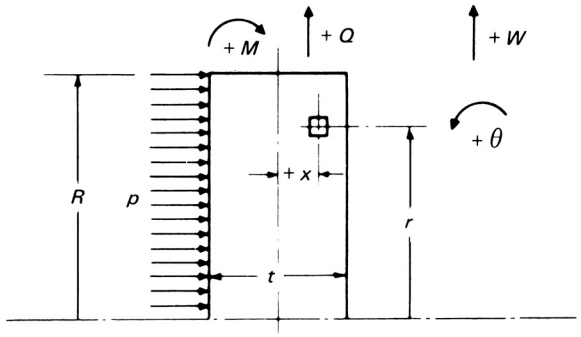

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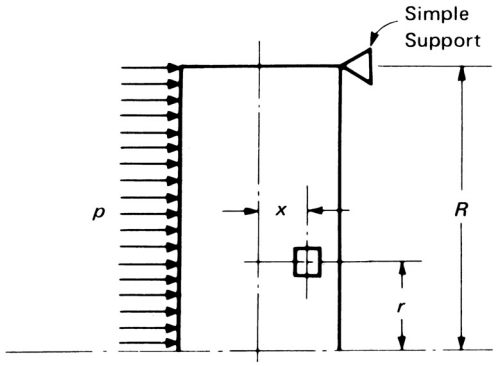
Figure A-5212-1



Rotation of the midplane:

$$\theta = -p\frac{3(1-\nu)}{4t^3E}\left[r^3(1+\nu) - rR^2(3+\nu)\right] \quad (4)$$

Rotation of the midplane at the outer edge:

$$\theta = +p\left[3(1-\nu)/2E\right](R/t)^3 \quad (5)$$

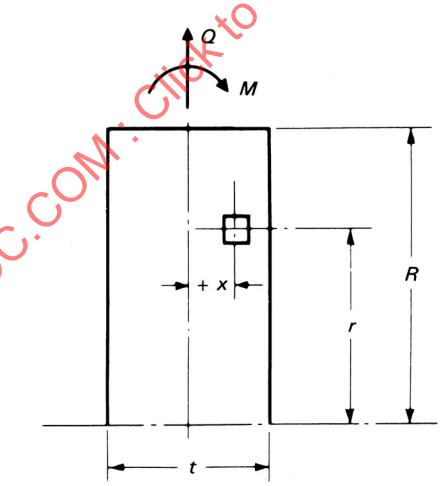
Radial displacement:

$$w = +(x)\theta \quad (6)$$

A-5213 Edge Loads on Flat Plates

The principal stresses and deformations of a flat plate subjected to uniformly distributed edge loads, as shown on Figure A-5213-1 are given for radial location r at any point x in the cross section by the following equations:

Figure A-5213-1



Radial and tangential stresses:

$$\sigma_r = \sigma_t = \frac{Q}{t} - \frac{12(x)}{t^3}M \quad (7)$$

Rotation of the midplane:

$$\theta = \frac{-12(1-\nu)(r)}{Et^3}M \quad (8)$$

Radial displacement:

$$w = \frac{(1-\nu)(r)}{Et}Q + (x)\theta \quad (9)$$

A-5220 FLAT PLATE PRESSURE VESSEL HEADS

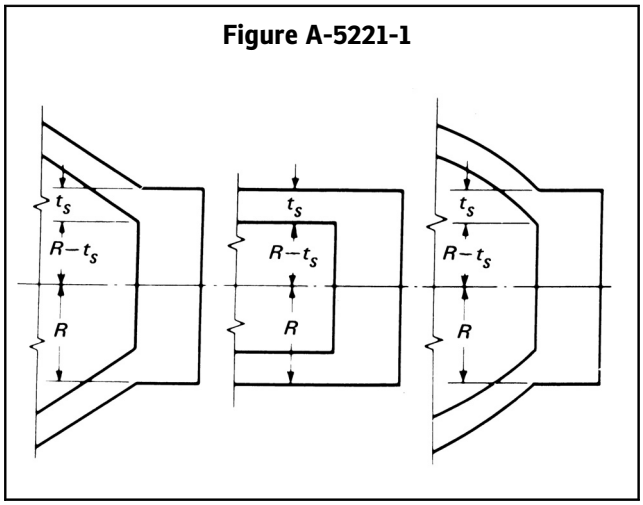
A-5221 Methods of Attachment

Flat plates used as pressure vessel heads are attached to a vessel shell in the manner shown by the typical examples in Figure A-5221-1. Since the support conditions at the edge of the plate depend upon the flexibility of the adjoining shell, the stress distribution in the plate is influenced by the thickness and geometry. The structure formed by the head and the shell may be analyzed according to the principles of discontinuity analysis described in Article A-6000. In the following paragraph, equations are given for the quantities necessary to perform a discontinuity analysis.

A-5222 Displacements and Principal Stresses in a Flat Head

The head is assumed to be separated from the adjoining shell element and under the action of the pressure load. Figure A-5222-1 illustrates this condition. The effects of the adjacent shell are represented by the pressure reaction force, the discontinuity force Q, and the discontinuity moment M. These act at the assumed junction point a. The pressure acts on the left-hand face over a circular area defined by the inside radius of the adjacent shell. The support point lies on this same face at the midradius

Figure A-5221-1



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of the adjacent shell. The equations in this paragraph are given in terms of the head dimensions R and t and multiplying factors F_1 to F_4. These factors reflect the extent of the pressure area and the location of the junction point. The numerical values for F_1 to F_4 are given in Table A-5240-1. These are functions of the ratio of the shell thickness t_s to the head radius R.

A-5223 Displacements of a Flat Head

A-5223.1 Displacements Due to Pressure. For a plate simply supported at a point a, the rotational displacement θ_p and the radial displacement w_p of point a, due to pressure p acting over the area defined by the radius $(R - t_s)$, are given by eqs. (10) and (11).

$$\theta_p = +\frac{F_1}{E(t/R)^3}p \tag{10}$$

$$w_p = -(t/2)\theta_p \tag{11}$$

A-5223.2 Displacements Due to Radial Force and Moment. The rotational displacements θ and the radial displacement w of point a, due to a uniformly distributed radial force Q and moment M acting at point a, are given by eqs. (12) and (13).

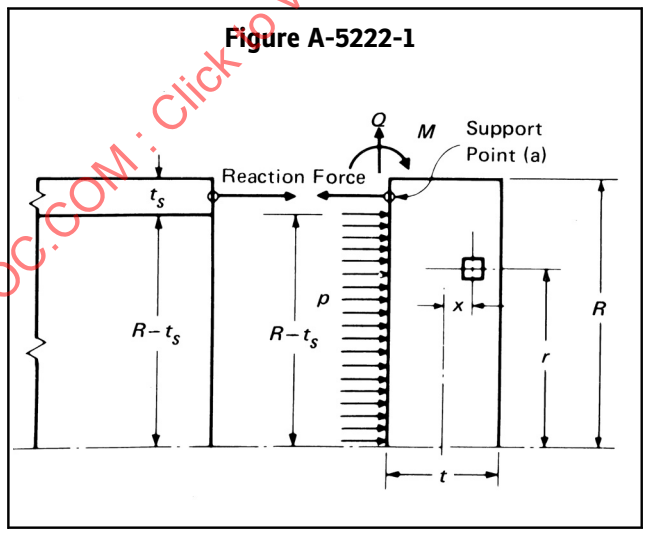
$$\theta = \frac{-F_3}{ER(t/R)^2}Q + \frac{-2F_3}{ER^2(t/R)^3}M \tag{12}$$

$$w = \frac{2F_3}{3E(t/R)}Q + \frac{F_3}{ER(t/R)^2}M \tag{13}$$

A-5224 Principal Stresses in a Flat Head

When the values of the discontinuity force Q and the moment M have been determined by a discontinuity analysis, the principal stresses in a flat plate can be calculated in the following subparagraphs.

Figure A-5222-1



A-5224.1 Radial Stress Due to Pressure. For a plate simply supported at point a, the radial stress σ_r for a radial location r less than $(R - t_s)$ at any point x, due to pressure p acting over the area defined by the radius $(R - t_s)$, is given by the following equation:

$$\sigma_r = \frac{(x)p}{t(t/R)^2}\left[(F_2) - \frac{3(3+\nu)r^2}{4R^2}\right] \tag{14}$$

A-5224.2 Tangential and Axial Stresses Due to Pressure. For these same conditions, the tangential stress σ_t and the axial stress σ_l are given by eqs. (15) and (16).

$$\sigma_t = \frac{(x)p}{t(t/R)^2}\left[(F_2) - \frac{3(1+3\nu)r^2}{4R^2}\right] \tag{15}$$

$$\sigma_l = (p/t)(x - t/2) \tag{16}$$

A-5224.3 Radial and Tangential Stresses Due to Radial Force and Moment. The radial stress σ_r and the tangential stress σ_t for any radial location at any point x in the cross section, due to uniformly distributed radial force Q and a uniformly distributed moment M acting at point a, are given by the equation:

$$\sigma_r = \sigma_t = \frac{F_4}{t}\left[1 - \frac{6(x)}{t}\right]Q - \frac{12F_4(x)}{t^3}M \tag{17}$$

A-5240 GEOMETRY CONSTANTS

The geometry constants F_1 through F_4 are functions of Poisson's ratio and t_s/R. These are given in eqs. (18) through (21).

$$F_1 = \frac{3(1-\nu)(2-f^2)(1-f)^2}{16(2-f)}\left[8 - f(4-f)(1-\nu)\right] \tag{18}$$

$$F_2 = {}^3\!/_8(1-f)^2\left\{(1-\nu)(2-f^2) + 4(1+\nu) \times \left[1 + 2\ln\frac{2-f}{2-2f}\right]\right\} \tag{19}$$

$$F_3 = {}^3\!/_8(1-\nu)(2-f)\left[8 - f(4-f)(1-\nu)\right] \tag{20}$$

$$F_4 = {}^1\!/_8\left[8 - f(4-f)(1-\nu)\right] \tag{21}$$

In these equations

$$f = t_s/R$$

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Table A-5240-1

t_s/R	F_1	F_2	F_3	F_4
0.00	1.0500	2.4750	4.2000	1.0000
0.02	1.0112	2.4149	4.1290	0.9930
0.04	0.9729	2.3546	4.0589	0.9861
0.06	0.9349	2.2943	3.9897	0.9793
0.08	0.8974	2.2338	3.9213	0.9725
0.10	0.8604	2.1734	3.8538	0.9658
0.12	0.8238	2.1129	3.7871	0.9592
0.14	0.7878	2.0524	3.7213	0.9527
0.16	0.7523	1.9919	3.6562	0.9462
0.18	0.7173	1.9315	3.5920	0.9398
0.20	0.6830	1.8712	3.5286	0.9335

Table A-5240-1 lists these functions for various values of t_s/R. These tabular values have been computed using 0.3 for Poisson's ratio.

A-5250 STRESS INTENSITIES IN A FLAT PLATE

The principal stresses due to pressure p, discontinuity force Q, discontinuity moment M, and other coincident loadings should be combined algebraically and the stress differences determined according to the procedures of NB-3215. The calculated stress intensity values should not exceed the stress limits given in NB-3220.

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ARTICLE A-6000
DISCONTINUITY STRESSES

A-6100 INTRODUCTION

A-6110 SCOPE

(a) Pressure vessels usually contain regions where abrupt changes in geometry, material, or loading occur. These regions are known as *discontinuity areas* and the stresses associated with them are known as *discontinuity stresses*. The discontinuity stresses are required to satisfy the compatibility of deformations of these regions.

(b) This Article describes a general procedure for analyzing the discontinuity stresses. A numerical example is included to illustrate the procedure.

(c) To determine the principal stresses at a discontinuity, it is necessary to evaluate the stresses caused by

(1) pressure;

(2) mechanical loads;

(3) thermal loads; and

(4) discontinuity loads. The stress intensities are then obtained by superposition of the stresses according to the rules given in NB-3220.

A-6120 INFORMATION REQUIRED

In order to perform a discontinuity analysis, the following information must be known:

(a) the dimensions of the vessel;

(b) the material properties (E, α, ν) of the component parts of the vessel (A-7120);

(c) mechanical loads, such as pressure, dead weight, bolt loads, and pipe loads;

(d) temperature distribution in the component parts.

A-6200 METHOD OF AND PROCEDURE FOR DISCONTINUITY ANALYSIS

A-6210 METHOD

(a) The analysis of a pressure vessel containing discontinuity areas can be performed in a standard manner similar to the analysis of any statically indeterminate structure. The analysis is initiated by separating the vessel into shell elements of simple geometry, such as rings and cylinders, of which the structural behavior is known. The pressure, mechanical, and thermal loads acting on the structure are applied to the shell elements with a system of forces required to maintain the static equilibrium of each element. These loads and forces cause individual elements deformations, which in general are not equal at the adjoining edges. The deformations at an element edge are defined as:

(1) radial displacement

(2) rotation of the meridian tangent

A redundant moment and shear force must generally exist on the edges of the elements, in order to have compatibility of deformations, and restore continuity in the structure.

(b) At each juncture discontinuity, two equations can be written which express the equality of the combined deformations due to all the applied loads and the redundant forces and moments. One equation will express the equality of rotation, the other equation the equality of displacement of the adjacent elements. The resulting system of simultaneous equations can be solved to obtain the redundant moment and shear force at each juncture.

A-6220 PROCEDURE

A-6221 Basic Steps

The basic steps to follow for determining the redundant shear and moment that may exist at a pressure vessel discontinuity are given in (a) through (f).

(a) Separate the vessel into individual shell elements at locations of discontinuity.

(b) Calculate the edge deformations of each element, caused by a unit shear force and a unit moment at each edge. These values are known as influence coefficients. The deformations due to local flexibilities may be considered in the calculation of these influence coefficients.

(c) Calculate the edge deformations of each element, caused by loads other than redundant loads.

(d) Calculate the edge deformations of each element, caused by the temperature distributions.

(e) At each juncture of two elements equate the total radial displacements and the total rotations of each element.

(f) Solve the final system of simultaneous equations for the redundant shears and moments.

A-6222 Stresses

When the values of the redundant shear forces and moments have been determined, the stresses resulting from the redundant loadings may be computed by conventional methods. The final stresses for each element are

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determined by combining these stresses with the stresses which would exist in the individual shell elements of A-6233, Step 1.

A-6230 EXAMPLE ILLUSTRATING APPLICATION OF A-6221

A-6231 Given

A pressure vessel as shown in Figure A-6230-1. It is constructed of SA-533, Grade B, Class 1 steel and subjected to an internal pressure of 950 psi at 300°F. The vessel consists of the following:

(a) a hemispherical head with:

inside radius	R = 30 in.
thickness	t = 1.375 in.

(b) a cylindrical shell with:

inside radius	R = 30 in.
thickness	t = 1.375 in.
length	L = 10 in.

(c) a flat head with:

outside radius	R = 31.375 in.
thickness	t = 6 in.

The material properties assumed are:

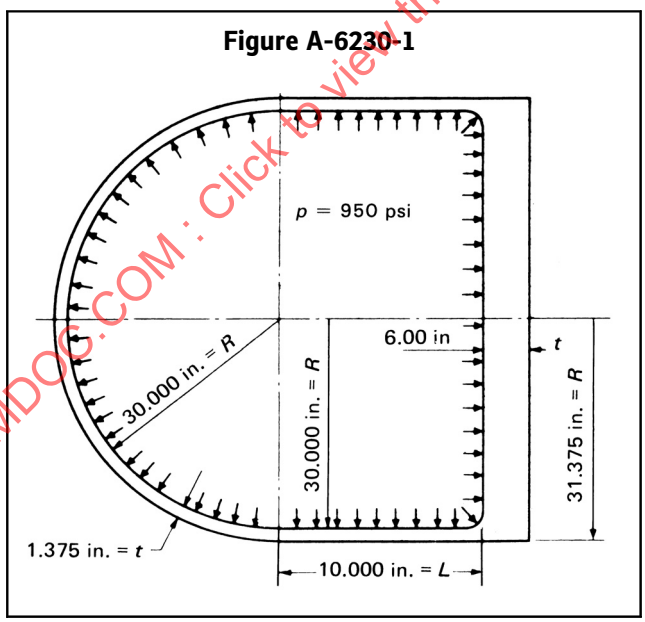
$$E = 29 \times 10^6 \text{ psi}$$

$$\nu = 0.3$$

A-6232 Required

To calculate the discontinuity stresses at the locations of structural discontinuity.

Figure A-6230-1



A-6233 Solution

Step 1. Separate the vessel at locations of discontinuity into individual elements.

Step 2. Calculate the influence coefficients.

(a) Element A, Hemispherical Head. From A-3233.2, the lateral displacement and rotation at juncture O due to edge loads Q_o and M_o are given as:

$$w_{AO} = \delta_{AO} = -\frac{2R_m\lambda}{Et}Q_o + \frac{2\lambda^2}{Et}M_o$$

$$\theta_{AO} = -\frac{2\lambda^2}{Et}Q_o + \frac{4\lambda^3}{R_mEt}M_o$$

NOTE: For this case of a hemispherical shell the lateral force H on the hemispherical head and the radial force Q on the cylindrical shell are equal. Similarly the lateral displacement δ and the radial displacement w are equal.

Substituting the given dimensions and material properties gives:

$$w_{AO} = \delta_{AO} = \left(-9.346755Q_o + 1.849564M_o\right) \times 10^{-6}$$

$$\theta_{AO} = \left(-1.849564Q_o + 0.731995M_o\right) \times 10^{-6}$$

(b) Element B, Cylindrical Shell (See Figure A-6230-2). From A-2243, the radial displacements and rotations at the edges O and L due to edge loadings Q_o, M_o, Q_L, and M_L are given as follows:

$$w_{BO} = \left(B_{11}/2\beta^3D\right)Q_o + \left(B_{12}/2\beta^2D\right)M_o + \left(G_{11}/2\beta^3D\right)Q_L + \left(G_{12}/2\beta^2D\right)M_L$$

$$-\theta_{BO} = \left(B_{12}/2\beta^2D\right)Q_o + \left(B_{22}/\beta D\right)M_o + \left(G_{12}/2\beta^2D\right)Q_L + \left(G_{22}/2\beta D\right)M_L$$

$$w_{BL} = \left(G_{11}/2\beta^3D\right)Q_o + \left(G_{12}/2\beta^2D\right)M_o + \left(B_{11}/2\beta^3D\right)Q_L + \left(B_{12}/2\beta^2D\right)M_L$$

$$\theta_{BL} = \left(G_{12}/2\beta^2D\right)Q_o + \left(G_{22}/2\beta D\right)M_o + \left(B_{12}/2\beta^2D\right)Q_L + \left(B_{22}/\beta D\right)M_L$$

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Substituting given dimensions and material properties gives:

$$w_{BO} = (10.697309Q_o + 2.114906M_o - 3.812809Q_L - 1.025641M_L) \times 10^{-6}$$

$$-\theta_{BO} = (2.114906Q_o + 0.792367M_o - 1.025641Q_L - 0.123127M_L) \times 10^{-6}$$

$$w_{BL} = (-3.812809Q_o - 1.025641M_o + 10.697309Q_L + 2.114906M_L) \times 10^{-6}$$

$$\theta_{BL} = (-1.025641Q_o - 0.123127M_o + 2.114906Q_L + 0.792367M_L) \times 10^{-6}$$

(c) Element C, Flat Head. From A-5223.2, the radial displacement and rotation at juncture L due to edge loadings Q_L and M_L are given as follows:

$$w_{CL} = -\frac{2F_3}{3E(t/R)}Q_L + \frac{F_3}{ER(t/R)^2}M_L$$

$$\theta_{CL} = \frac{F_3}{ER(t/R)^2}Q_L - \frac{2F_3}{ER^2(t/R)^3}M_L$$

Substituting given dimensions and material properties gives:

$$w_{CL} = [-0.486330Q_L + 0.121583M_L] \times 10^{-6}$$

$$\theta_{CL} = [0.121583Q_L - 0.040530M_L] \times 10^{-6}$$

Step 3. Calculate the edge deformations due to the internal pressure.

(a) Element A, Hemispherical Shell (see Figure A-6230-3). The lateral displacement of point O at the midsurface ($r = R_m$) of a hemispherical shell subjected to internal pressure is given by the expression:

$$w_o = \delta_o = p\frac{R^3}{2E(R_o^3 - R^3)R_m^2}\left[2R_m^3(1 - 2\nu) + R_o^3(1 + \nu)\right]$$

Substituting the dimensions, pressure, and material properties gives:

$$w_{O_{(\text{pressure})}} = 7647 \times 10^{-6}$$

NOTE: An alternative expression may be used for the displacement of a thin hemispherical shell:

$$w = (1 - \nu)(pR^2/2Et)$$

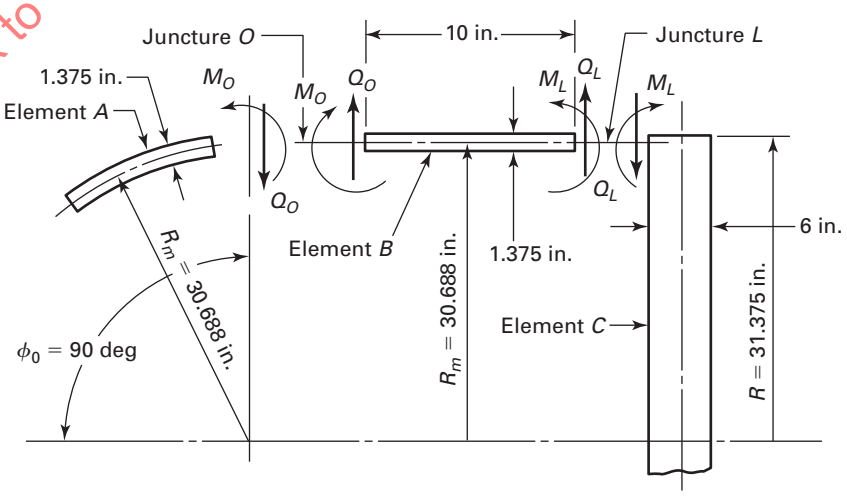
There is no rotation resulting from the internal pressure and membrane forces as shown:

$$\theta_{AO_{(\text{pressure})}} = 0$$

(b) Element B, Cylindrical Shell (see Figure A-6230-4). The radial displacement of the midsurface of a closed end cylindrical shell subjected to internal pressure is given by the expression:

$$w = p\frac{R^2}{E(R_o^2 - R^2)R_m}\left[R_m^2(1 - 2\nu) + R_o^2(1 + \nu)\right]$$

Figure A-6230-2



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Substituting the dimensions, pressure, and material properties gives:

$$w_{BO_{(\text{pressure})}} = w_{BL_{(\text{pressure})}} = 18,857 \times 10^{-6}$$

NOTE: An alternative expression may be used for displacement of a thin cylindrical shell:

$$w = \left[1 - (\nu/2)\right](pRR_m/Et)$$

There is no rotation resulting from internal pressure and the membrane forces as shown:

$$\theta_{BO_{(\text{pressure})}} = \theta_{BL_{(\text{pressure})}} = 0$$

(c) Element C, Flat Head (See Figure A-6230-5). The rotation of a flat head at point L due to internal pressure is given by eq. A-5223.1(10):

$$\theta = \frac{F_1}{E(t/R)^3}p$$

Substituting the dimensions, pressure, and material properties gives:

$$\theta_{CL_{(\text{pressure})}} = 4523 \times 10^{-6}$$

The radial displacement at juncture L is given by eq. A-5223.1(11):

$$w_{CL_{(\text{pressure})}} = -\frac{t}{2}\theta_{CL} = -13,570 \times 10^{-6}$$

Step 4. Calculate the free deformations of the edges of each element caused by temperature distributions.

In this example all parts of the vessel are at the same temperature and are of the same material; therefore, temperature deformations need not be considered.

Step 5. Equate the total lateral displacements and rotations of adjacent elements at each juncture.

Figure A-6230-3

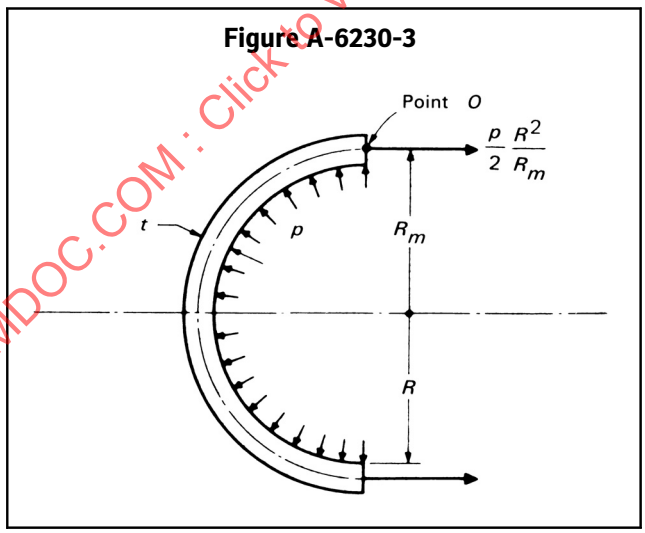
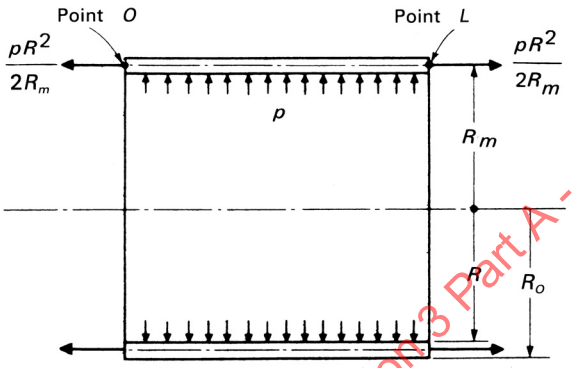


Figure A-6230-4



(a) Juncture O

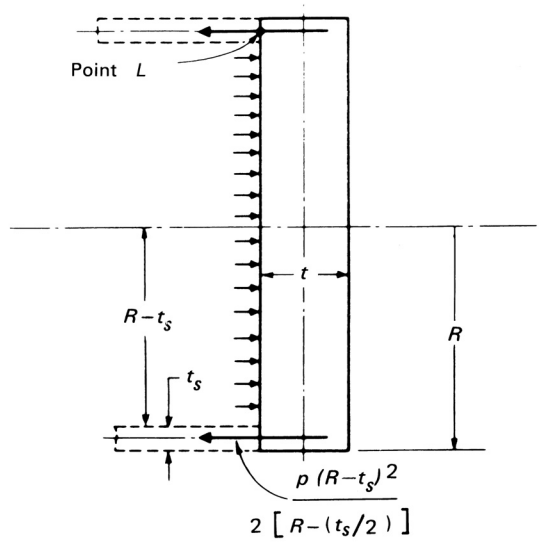
$$w_{AO_{(\text{total})}} = w_{BO_{(\text{total})}}$$

$$\left(-9.346755Q_o + 1.849564M_o + 7647\right) \times 10^{-6} = \left(10.697309Q_o + 2.114906M_o - 3.812809Q_L - 1.025641M_L + 18,857\right) \times 10^{-6} \quad (1)$$

$$\theta_{AO_{(\text{total})}} = \theta_{BO_{(\text{total})}}$$

$$\left(-1.849564Q_o + 0.731995M_o + 0\right) \times 10^{-6} = \left(-2.114906Q_o - 0.792367M_o + 1.025641Q_L + 0.123127M_L + 0\right) \times 10^{-6} \quad (2)$$

Figure A-6230-5



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(b) Juncture L

$$w_{BL_{(\text{total})}} = w_{CL_{(\text{total})}}$$
$$\left(-3.812809Q_o - 1.025641M_o + 10.697309Q_L + 2.114906M_L + 18,857\right) \times 10^{-6} = \left(-0.486330Q_L + 0.121583M_L - 13,570\right) \times 10^{-6} \quad (3)$$

$$\theta_{BL_{(\text{total})}} = \theta_{CL_{(\text{total})}}$$
$$\left(-1.025641Q_o - 0.123127M_o + 2.114906Q_L + 0.792367M_L + 0\right) \times 10^{-6} = \left(0.121583Q_L - 0.040530M_L + 4523\right) \times 10^{-6} \quad (4)$$

Combining like terms and multiplying through by 10^6 results in the following system of simultaneous equations which express compatibility at the junctures:

$$-20.04364Q_o - 0.265342M_o + 3.8128090Q_L + 1.025641M_L = 11,210 \quad (5)$$

$$0.265342Q_o + 1.524362M_o - 1.025641Q_L - 0.123127M_L = 0 \quad (6)$$

$$-3.812809Q_o - 1.025641M_o + 11.183639Q_L + 1.993323M_L = -32,427 \quad (7)$$

$$-1.025641Q_o - 0.123127M_o + 1.993323Q_L + 0.832897M_L = 4523 \quad (8)$$

Step 6. Solve the above equation for Q_o, M_o, Q_L, and M_L. The results are:

$$Q_o = 807 \text{ lb/in.} = -0.807 \text{ kips/in.}$$
$$M_o = -3018 \text{ in.-lb/in.} = -3.018 \text{ in.-kips/in.}$$
$$Q_L = 7259 \text{ lb/in.} = +7.259 \text{ kips/in.}$$
$$M_L = 21,363 \text{ in.-lb/in.} = 21.363 \text{ in.-kips/in.}$$

NOTE: A negative sign indicates that the actual direction of the loading is opposite to that chosen in Step 1.

Step 7. Compute the discontinuity stresses at each juncture due to the redundants Q_o, M_o, Q_L, and M_L. To illustrate the procedure, these stresses will be computed in the cylindrical shell element B at both junctures O and L. From A-2260:

$$\sigma_l(x) = \pm 6M(x)/t^2$$
$$\sigma_t(x) = Ew(x)/(R + t/2) \pm 6\nu M(x)/t^2$$
$$\sigma_r = 0$$

(a) Juncture O. At juncture O, $M(x) = M_o$ and $w(x) = w_o$.

NOTE: When computing $\sigma_t(x)$ only the radial displacement due to the redundant shear forces and moments should be used. The free displacements from Steps 3 and 4 should not be included.

$$w_{BO} = \left(10.697309Q_o + 2.114906M_o - 3.812809Q_L - 1.025641M_L\right) \times 10^{-6} = -0.009007 \text{ in.}$$

$$Ew_{BO} = -261.20 \text{ kips/in.}$$
$$M_o = -3.018 \text{ in.-kips/in.}$$

(1) Inside surface:

$$\sigma_l = 6(-3.018)/(1.375)^2 = -9.6 \text{ ksi}$$

$$\sigma_t = (-261.20/30.6875) + 6(0.3)(-3.02)/(1.375)^2 = -11.5 \text{ ksi}$$

$$\sigma_r = 0$$

(2) Outside surface:

$$\sigma_l = -6(-3.018)/(1.375)^2 = 9.6 \text{ ksi}$$

$$\sigma_t = (-261.20/30.6875) - 6(0.3)(-3.018)/(1.375)^2 = -5.6 \text{ ksi}$$

$$\sigma_r = 0$$

(b) Juncture L. At juncture L, $M(x) = M_L$ and $w(x) = w_L$.

$$w_{BL} = \left(-3.812809Q_o - 1.025641M_o + 10.697309Q_L + 2.114906M_L\right) \times 10^{-6} = -0.026299 \text{ in.}$$

$$Ew_{BL} = -762.67 \text{ kips/in.}$$
$$M_L = 21.363 \text{ in.-kips/in.}$$

(1) Inside surface:

$$\sigma_l = 6(21.363)/(1.375)^2 = 67.8 \text{ ksi}$$

$$\sigma_t = (-762.67/30.6875) + 6(0.3)(21.363)/(1.375)^2 = -4.5 \text{ ksi}$$

$$\sigma_r = 0$$

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(2) Outside surface:

$$\sigma_l = -6(21.363)/(1.375)^2 = -67.8 \text{ ksi}$$

$$\sigma_t = (-762.67/30.6875) - 6(0.3)(21.363)/(1.375)^2$$
$$= -45.2 \text{ ksi}$$

$$\sigma_r = 0$$

NOTE: $E = 29 \times 10^6$ psi.

(c) The discontinuity stresses in the hemispherical shell may be computed by using the expressions given in A-3232 and A-3234.

(d) The discontinuity stresses in the flat head may be computed using the expressions given in A-5224.

Step 8. Compute the total stresses. The total stresses may be computed in any element at any juncture by combining the stresses due to the redundant shear forces and moments as computed in Step 7, with the stresses resulting from all other loadings. In this case the stresses in the cylindrical shell, hemispherical shell, and flat head due to internal pressure may be computed by the expressions given in A-2212, A-3221, and A-5224, respectively. To illustrate the procedure the total stresses in the cylindrical shell at junctures *O* and *L* will be computed. The stresses in the cylindrical shell due to internal pressure may be computed from eqs. A-2212(1), A-2212(2), and A-2212(3):

$$\sigma_l = p/(Y^2 - 1)$$
$$\sigma_t = p(1 + Z^2)/(Y^2 - 1)$$
$$\sigma_r = p(1 - Z^2)/(Y^2 - 1)$$

(a) Juncture O

(1) Inside surface:

$$Y = Z = R_o/R = 31.375/30.000 = 1.04583$$
$$p = 950 \text{ psi}$$
$$\sigma_l = 10,132 \text{ psi} \approx 10.1 \text{ ksi}$$
$$\sigma_t = 21,214 \text{ psi} \approx 21.2 \text{ ksi}$$
$$\sigma_r = -950 \text{ psi} \approx -1 \text{ ksi}$$

The stresses due to the redundant shear forces and moments were computed in Step 7 as:

$$\sigma_l = -9.6 \text{ ksi}$$
$$\sigma_t = -11.5 \text{ ksi}$$
$$\sigma_r = 0$$

The total stresses are:

$$\sigma_l = 10.1 + (-9.6) = 0.5 \text{ ksi}$$
$$\sigma_t = 21.2 + (-11.5) = 9.7 \text{ ksi}$$
$$\sigma_r = -0.95 + 0 \approx -1.0 \text{ ksi}$$

(2) Outside surface:

$$Z = 1$$
$$Y = 1.04583$$
$$\sigma_l = 10,132 \text{ psi} \approx 10.1 \text{ ksi}$$
$$\sigma_t = 20,264 \text{ psi} \approx 20.3 \text{ ksi}$$
$$\sigma_r = 0$$

The stresses due to the redundant shear forces and moments were computed as:

$$\sigma_l = 9.6 \text{ ksi}$$
$$\sigma_t = -5.6 \text{ ksi}$$
$$\sigma_r = 0$$

The total stresses are:

$$\sigma_l = 10.1 + (9.6) = 19.7 \text{ ksi}$$
$$\sigma_t = 20.3 + (-5.6) = 14.7 \text{ ksi}$$
$$\sigma_r = 0 + 0 = 0$$

(b) Juncture L

(1) Inside Surface. The stresses due to the internal pressure are the same as at juncture *O*:

$$\sigma_l = 10.1 \text{ ksi}$$
$$\sigma_t = 21.2 \text{ ksi}$$
$$\sigma_r = -1.0 \text{ ksi}$$

The stresses due to the redundant shear forces and moments were computed as:

$$\sigma_l = 67.8 \text{ ksi}$$
$$\sigma_t = -4.5 \text{ ksi}$$
$$\sigma_r = 0$$

The total stresses are:

$$\sigma_l = 10.1 + (67.8) = 77.9 \text{ ksi}$$
$$\sigma_t = 21.2 + (-4.5) = 16.7 \text{ ksi}$$
$$\sigma_r = -0.95 + 0 \approx -1.0 \text{ ksi}$$

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(2) Outside Surface. The stresses due to the internal pressure are the same as at juncture *O*:

$$\sigma_l = 10.1 \text{ ksi}$$

$$\sigma_t = 20.3 \text{ ksi}$$

$$\sigma_r = 0$$

The stresses due to redundant shear forces and moments were computed as:

$$\sigma_l = -67.8 \text{ ksi}$$

$$\sigma_t = -45.2 \text{ ksi}$$

$$\sigma_r = 0$$

The total stresses are:

$$\sigma_l = 10.1 + (-67.8) = -57.7 \text{ ksi}$$

$$\sigma_t = 20.3 + (-45.2) = -24.9 \text{ ksi}$$

$$\sigma_r = 0 + 0 = 0$$

Step 9. See below.

(a) When evaluating the stresses in accordance with NB-3222.2, the stress intensities at each location should be computed from the total principal stresses determined in Step 8.

(b) When evaluating the stresses in accordance with NB-3222.4(b), it is necessary to consider the influence of local stress concentrations upon the principal stresses determined in Step 8 before computing the stress intensities.

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ARTICLE A-7000
THERMAL STRESSES

A-7100 INTRODUCTION

A-7110 OCCURRENCE OF THERMAL STRESSES AND DISPLACEMENTS

(a) Thermal stresses occur in a system, or part of a system, when thermal displacements (expansions or contractions) which would otherwise freely occur are partially or completely restrained.

(b) Thermal displacements may be induced by temperature distributions caused by heat transfer and internal heat generation.

A-7120 INFORMATION NECESSARY TO CALCULATE THERMAL STRESSES

(a) In order to calculate thermal stresses, the information stipulated in (1), (2), and (3) below must usually be known for each member comprising the system:

(1) the dimensions;

(2) the temperature distribution T as a function of a suitable coordinate system;

(3) the material properties: modulus of elasticity E, Poisson's ratio ν, and coefficient of thermal expansion α, where E and α are temperature-dependent quantities (Section II, Part D, Subpart 2, Tables TE and TM provide these values).

(b) Frequently, it is accurate enough to consider E and α for each material as constant at their instantaneous values for the average temperature of the temperature range under consideration. If the lower limit of the temperature range is ambient, the instantaneous value of α for the average temperature coincides with the mean coefficient of thermal expansion.

(c) During transient conditions, the temperature distribution and thermal stresses vary with time. The analysis, therefore, requires consideration of the thermal stresses as a function of time during the transient.

A-7130 METHOD OF CALCULATION

If closed form expressions for the thermal stresses are not available, the procedure of (a) through (e) below may be used for computing the thermal stress for a specified time θ.

(a) Express the temperature distribution T for each component as a function of the space coordinates t, l, and r.

(b) Divide the system, which may be of irregular shape or of complex geometry, into free bodies of simple shape, such as rings, cylinders, and spherical shells.

(c) Calculate the free body stresses and deformations for each component resulting from the temperature distribution.

(d) Calculate the discontinuity thermal stresses by means of discontinuity analysis as described in Article A-6000.

(e) Superimpose the stresses determined in (c) and (d) above to obtain the combined thermal stresses.

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ARTICLE A-8000
STRESSES IN PERFORATED FLAT PLATES

A-8100 INTRODUCTION

A-8110 SCOPE

(a) This Article contains a method of analysis for flat perforated plates when subjected to directly applied loads or loadings resulting from structural interaction with adjacent members. This method applies to perforated plates which satisfy the conditions of (1) through (5).

(1) The holes are in an array of equilateral triangles.

(2) The holes are circular.

(3) There are 19 or more holes.

(4) The ligament efficiency is greater than 5% ($\eta \geq 0.05$).

(5) The plate is thicker than twice the hole pitch ($t/P \geq 2$). If only in-plane loads or thermal skin stresses are considered, this limitation does not apply.

(b) Credit may be taken for the stiffening effect of the tubes in the perforations. The extent to which the tubes stiffen the perforated plate depends on the materials, the manufacturing processes, operating conditions, and degree of corrosion. This stiffening effect may be included in the calculations by including part or all of the tube walls in the ligament efficiency used to obtain the effective elastic constants of the plate. Such stiffening may either increase or decrease stresses in the plate itself and in the attached shells.

(c) Credit may be taken for the staying action of the tubes where applicable.

(15) ### A-8120 NOMENCLATURE

c = radius of ring load (Figure A-8132.2-1)
E = Young's modulus for plate material
E^* = effective Young's modulus for perforated plate (Figure A-8131-1)
E_t = Young's modulus for tube material
h = nominal width of ligament at the minimum cross section
K = stress multiplier for stresses averaged across the width of the ligament but not through the thickness (Figure A-8142-1)
K_m = ratio of peak stress in reduced ligament to the peak stress in normal ligament
K_r = stress multiplier for circumferential stress in the plate rim (Figure A-8142-6)
K_{skin} = stress multiplier for thermal skin stress (Figure A-8153-1)
ln = $\log_e$
M = radial moment acting at edge of plate, in.-lb/in. (N·mm/mm) of circumference
P = nominal distance between hole centerlines, pitch
p_1, p_2 = pressures acting on surfaces of the plate
p_i = pressure inside tubes
p_s = pressure on surface where stress is computed, p_1 or p_2
Q = radial force acting at edge of plate, lb/in. (N/mm) of circumference
r = designation of radial location in plate
R^* = the effective radius of the perforated plate
= $r_o + \frac{1}{4}(P - h)$
r_o = radial distance from center of plate to center of outermost hole
S = stress intensity (A-8142)
t = thickness of plate exclusive of cladding or corrosion allowance
T_m = mean temperature averaged through the thickness of the plate
T_s = temperature of the surface of the plate
t_t = tube wall thickness
W = total ring load acting on plate (Figure A-8132.2-1), lb (N)
w = radial displacement of plate edge
x = axis of symmetry of hole pattern through the smaller ligament thickness (Figures A-8142-3 through A-8142-5)
Y = stress multiplier for peak ligament stresses (Figure A-8142-1)
y = axis of symmetry of hole pattern, perpendicular to x axis
Δp = differential pressure across the plate
α = coefficient of thermal expansion, in./in.-°F (mm/mm-°C)
β = biaxiality ratio (σ_r/σ_θ or σ_θ/σ_r) or (σ_1/σ_2 or σ_2/σ_1), where $-1 \leq \beta \leq 1$
η = ligament efficiency
= h/P
θ = rotation of plate edge, rad
ν = Poisson's ratio
ν^* = effective Poisson's ratio for perforated plate (Figure A-8131-1)
ρ = radius of holes in the plate
σ_1, σ_2 = principal stress in the plane of the equivalent solid plate (A-8142.2)
σ_{ave} = larger absolute value of σ_r or σ_θ [A-8142.1(b)]

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σ_r = radial stresses in the equivalent solid plate
σ_{rim} = nominal circumferential stress in solid rim
σ_{skin} = thermal skin stress
σ_θ = tangential stress in the equivalent solid plate
$\bar{\sigma}_r$ = radial stress averaged through the depth of the equivalent solid plate

A-8130 ANALYSIS OF CIRCULAR PERFORATED AREA

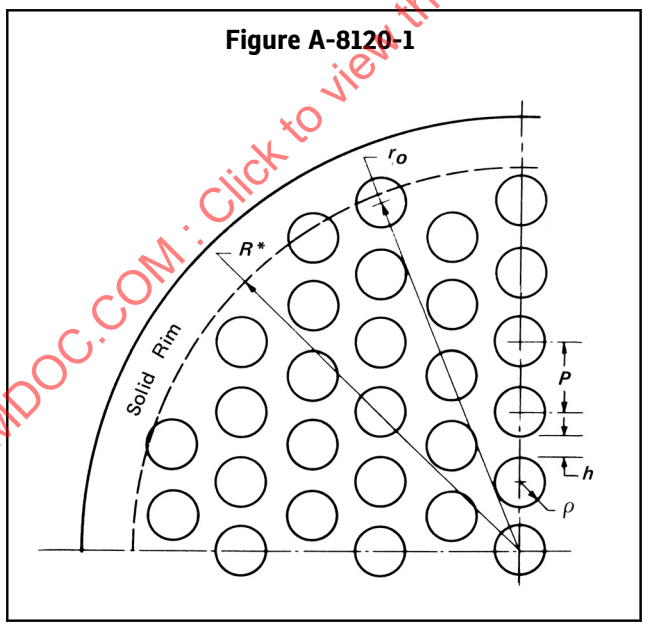
A-8131 Procedure

(a) The analysis method for perforated plates presented in this Article utilizes the concept of the equivalent solid plate. In this method, the perforated plate is replaced by a solid plate which is geometrically similar to the perforated plate but has modified values of the elastic constants.

(b) The elastic modulus E and Poisson's ratio ν are replaced by the effective elastic modulus E^* and effective Poisson's ratio ν^* of the perforated plate, and conventional equations for plates are used to determine the deformations and nominal stresses for the equivalent solid plate. The deformations so computed may be used directly in evaluating interaction effects. The actual values of the stress intensities in the perforated plate are determined by applying multiplying factors to the nominal stresses computed for the equivalent solid plate.

(c) The effective elastic constants are functions of the ligament efficiency η. The values are given in Figure A-8131-1 for the range of $0.05 \le \eta \le 1.0$ in the form of ν^* vs. η for a material with $\nu = 0.3$, and E^*/E vs. η. The stress multipliers are given in Figures A-8142-1 through A-8142-6. The Y factors presented in Figures A-8142-3 and A-8142-4 represent the largest values occurring through the thickness at the given angular position.

Figure A-8120-1



(d) The region of the perforated plate outside the effective radius R^* is called the plate rim. This unperforated portion of the plate may be considered as a separate connecting member, a ring or cylinder, and the structure may be analyzed in accordance with the procedures of Article A-6000.

A-8132 Analysis of Equivalent Solid Plate

In the following subparagraphs, equations are given for the nominal stresses and edge displacements for the equivalent solid circular plate under various axisymmetric load conditions.

A-8132.1 Edge Loads (see Figure A-8132.1-1). (15)

(a) Stresses at any location on the surface of the equivalent solid plate.

$$\sigma_r = \sigma_\theta = \frac{Q}{t} \pm \frac{6M}{t^2} \tag{1}$$

When double signs are used, the upper sign applies to the top surface as shown in Figure A-8132.1-1.

(b) Edge displacements of midplane at R^*:

$$w = \frac{R^*}{E^* t}(1 - \nu^*)Q \tag{2}$$

$$\theta = -\frac{12R^*}{E^* t^3}(1 - \nu^*)M \tag{3}$$

A-8132.2 Ring Loads Transverse to the Plane of the Plate (See Figure A-8132.2-1).

(a) Stresses at any radial location r on the surfaces of the equivalent solid plate: for $r \le c$,

$$\sigma_r = \sigma_\theta = \pm\frac{3W}{2\pi t^2}\left[\frac{1}{2}(1 - \nu^*) - \frac{1}{2}(1 - \nu^*) \times \left(\frac{c}{R^*}\right)^2 + (1 + \nu^*)\ln\frac{R^*}{c}\right] \tag{4}$$

for $r > c$,

$$\sigma_r = \pm\frac{3W}{2\pi t^2}\left[\frac{1}{2}(1 - \nu^*)\left(\frac{c}{r}\right)^2 - \frac{1}{2}(1 - \nu^*) \times \left(\frac{c}{R^*}\right)^2 + (1 + \nu^*)\ln\frac{R^*}{r}\right] \tag{5}$$

$$\sigma_\theta = \pm\frac{3W}{2\pi t^2}\left[(1 - \nu^*) - \frac{1}{2}(1 - \nu^*)\left(\frac{c}{r}\right)^2 - \frac{1}{2}(1 - \nu^*) \times \left(\frac{c}{R^*}\right)^2 + (1 + \nu^*)\ln\frac{R^*}{r}\right] \tag{6}$$

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Figure A-8131-1

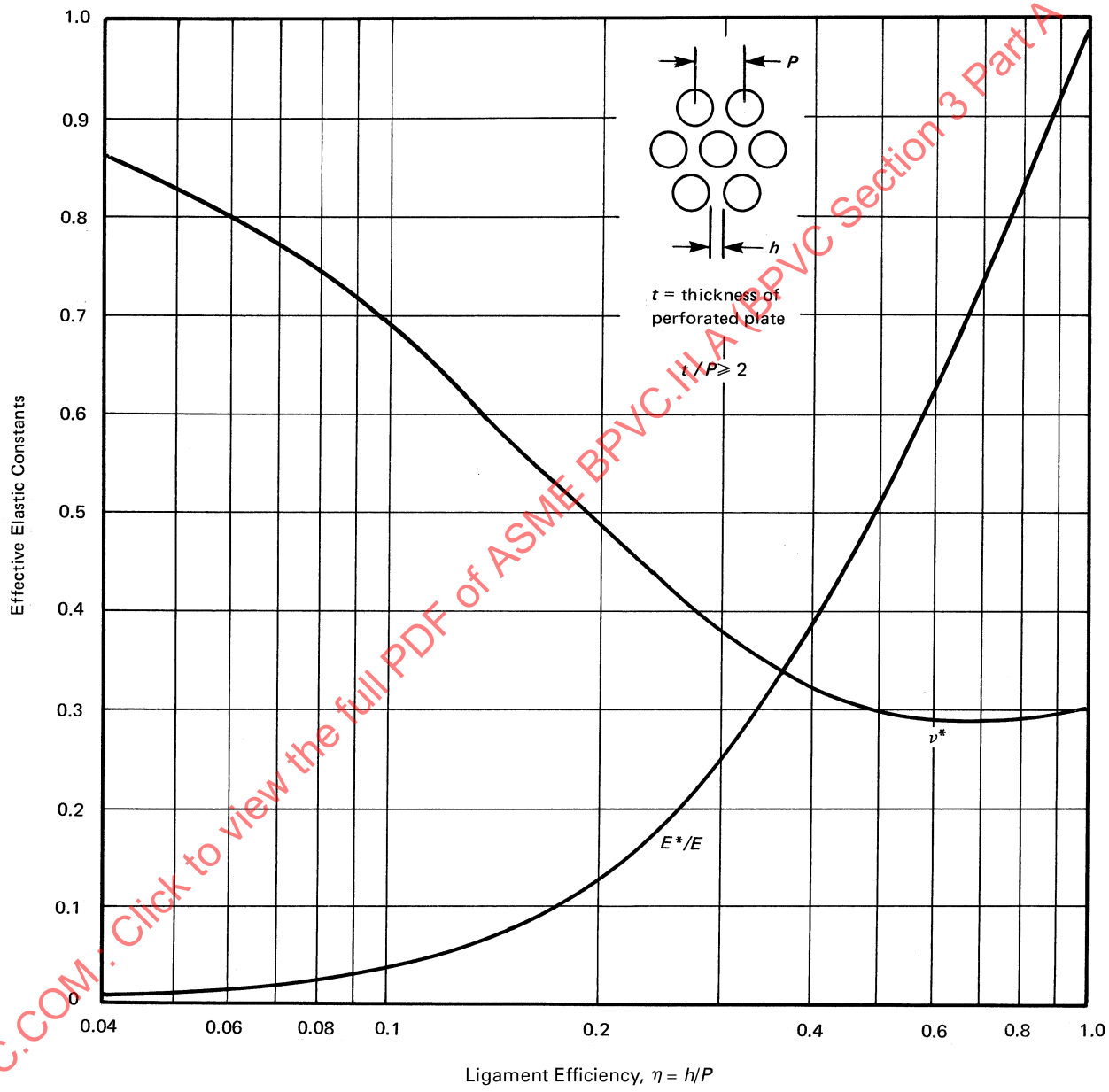

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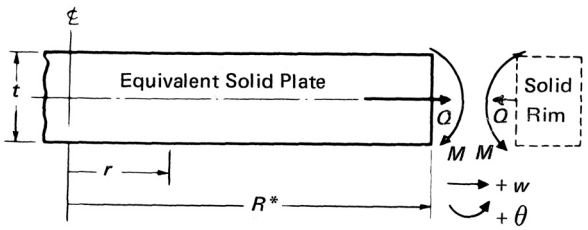
Figure A-8132.1-1



(b) Edge displacements of midplane at $r = R^*$:

$$w = 0 \tag{7}$$

$$\theta = \frac{3R^*}{\pi E^* t^3}(1 - \nu^*)\left[1 - \left(\frac{c}{R^*}\right)^2\right][W] \tag{8}$$

A-8132.3 Uniformly Distributed Pressure Loads (See Figure A-8132.3-1).

(a) Stresses at any location r on the surfaces of the equivalent solid plate:

$$\sigma_r = \pm\frac{3}{8}\left(\frac{R^*}{t}\right)^2(3 + \nu^*)\left[1 - \left(\frac{r}{R^*}\right)^2\right][\Delta p] \tag{9}$$

$$\sigma_\theta = \pm\frac{3}{8}\left(\frac{R^*}{t}\right)^2\left[(3 + \nu^*) - (1 + 3\nu^*)\left(\frac{r}{R^*}\right)^2\right][\Delta p] \tag{10}$$

(b) Edge displacement of midplane at $r = R^*$:

$$w = \frac{\nu R^*}{E}\left(\frac{p_1 + p_2}{2}\right) \tag{11}$$

$$\theta = \frac{3}{2}\frac{(R^*)^3}{E^* t^3}(1 - \nu^*)[\Delta p] \tag{12}$$

A-8132.4 Pressure in Tubes or Perforations (See Figure A-8132.4-1). (15)

(a) Stresses at any location in the equivalent solid plate:

$$\sigma_r = \sigma_\theta = \frac{h}{P}\frac{(P - h - 2t_t)}{\left[h + 2\left(\frac{E_t}{E}\right)t_t\right]}p_i \tag{13}$$

(b) Edge displacements of midplane at $r = R^*$:

$$w = \frac{R^*}{E}\left[\frac{E}{E^*}(1 - \nu^*) - (1 - \nu)\right]p_i \tag{14}$$

where E^*/E and ν^* should be evaluated for the ligament efficiency:

$$\frac{h}{P} = \frac{h + (2E_t/E)t_t}{P} \tag{15}$$

using Figure A-8131-1;

$$\theta = 0$$

A-8140 STRESS INTENSITIES AND STRESS LIMITS FOR PERFORATED PLATES

A-8141 Equations for Stress Intensities

In A-8140 equations are given for the stress intensities in a perforated plate using the stresses determined for the equivalent solid plate.

A-8142 Typical Ligaments in a Uniform Pattern

A-8142.1 Mechanical and Pressure Loads on Circular Plates.

(a) The stress intensity based on stresses averaged across the minimum ligament width and through the thicknesses of the plate is limited according to NB-3221.1 and is computed from the larger of:

$$S = \frac{P}{h}\sqrt{\left(\frac{\Delta p R^*}{t} + \frac{W}{\pi t R^*}\right)^2 + (\bar{\sigma}_r)^2} \tag{16}$$

Figure A-8132.2-1

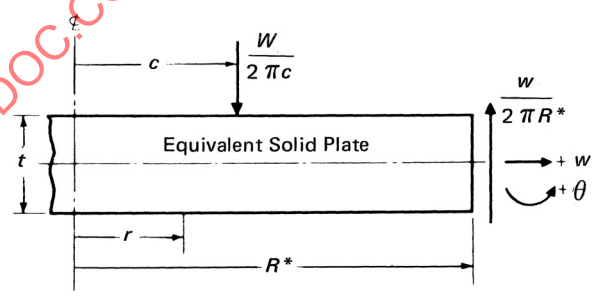
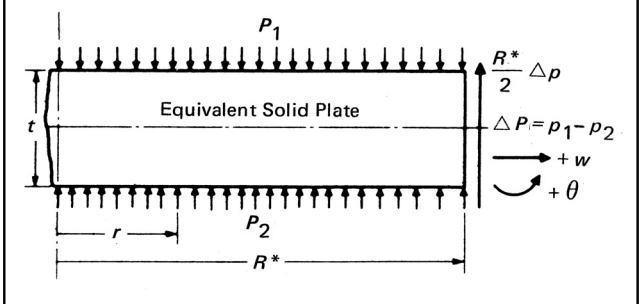


Figure A-8132.3-1



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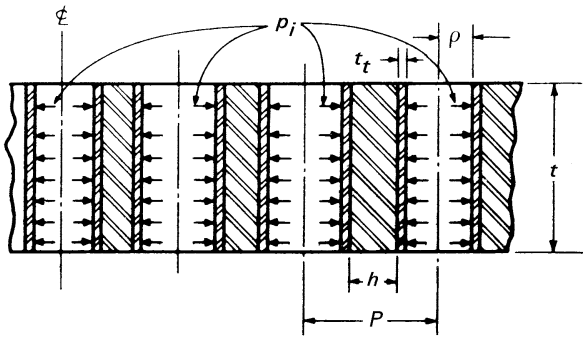
(15)

Figure A-8132.4-1



or

$$S = \frac{1}{2}\frac{P}{h}\left[\sqrt{\left(\frac{\Delta p R^*}{t} + \frac{W}{\pi t R^*}\right)^2 + \left(\bar{\sigma}_r\right)^2} + \bar{\sigma}_r + \left(2p_i h / P\right)\right] \quad (17)$$

where only the positive root is used. The first term under the radical reflects the effect of the transverse shear stress due to the mechanical and pressure loads. It is a maximum in the outermost ligament of the perforated region, but it may be determined for any radius, larger than c, by substituting r for R^* in the expression. For $r > c$, the $W/\pi t R^*$ term should be omitted. $\bar{\sigma}_r$ is the stress resulting from applied in-plane loading averaged through the thickness of the equivalent solid plate. It includes the stresses due to pressure in the tubes or perforations given in A-8132.4. No bending stresses are included.

(b) The stress intensity based on stresses averaged across the minimum ligament width but not through the thickness of the plate is limited according to NB-3221.3 and is computed from

$$S = K\left(P/h\right)\sigma_{\text{ave}} \quad (18)$$

where

K = stress multiplier from Figure A-8142-1

σ_{ave} = larger value of σ_r or σ_θ, psi (MPa), caused by mechanical loading and structural interaction with adjacent members, computed as the sum of the surface stresses in the equivalent solid plate, using the applicable equations in A-8130

However, supporting interactions from adjacent members may only be considered if the primary plus secondary stresses in such members are limited to $1.5S_m$. Effects of temperature are not included.

A-8142.2 Combined Mechanical and Thermal Effects. (15)

(a) The range of the stress intensity based on stresses averaged across the minimum ligament width but not through the thickness of the plate is limited according to NB-3222.2 and is computed from

$$S = K\left(P/h\right)\sigma_1 \quad (19)$$

where

K = stress multiplier from Figure A-8142-1

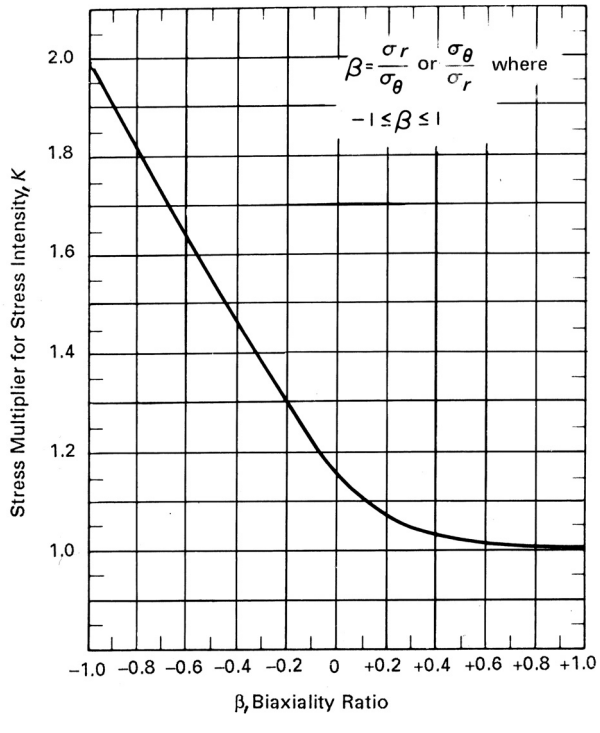
σ_1 = larger absolute value of σ_r or σ_θ, psi (MPa), caused by mechanical loading or structural interaction with adjacent members, computed as the sum of the surface stresses in the equivalent solid plate using the applicable equations in A-8130 and A-8150

The effects of temperature are included in the consideration of the structural interaction with adjacent members.

(b) The peak stress intensity due to all loadings is limited by cumulative fatigue considerations as described in NB-3222.4 and is given by

$$S = Y_{\max}\left(P/h\right)\sigma_1 + p_s \quad (20)$$

Figure A-8142-1



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where

p_s = pressure on the surface where the stress is being computed, psi (MPa)

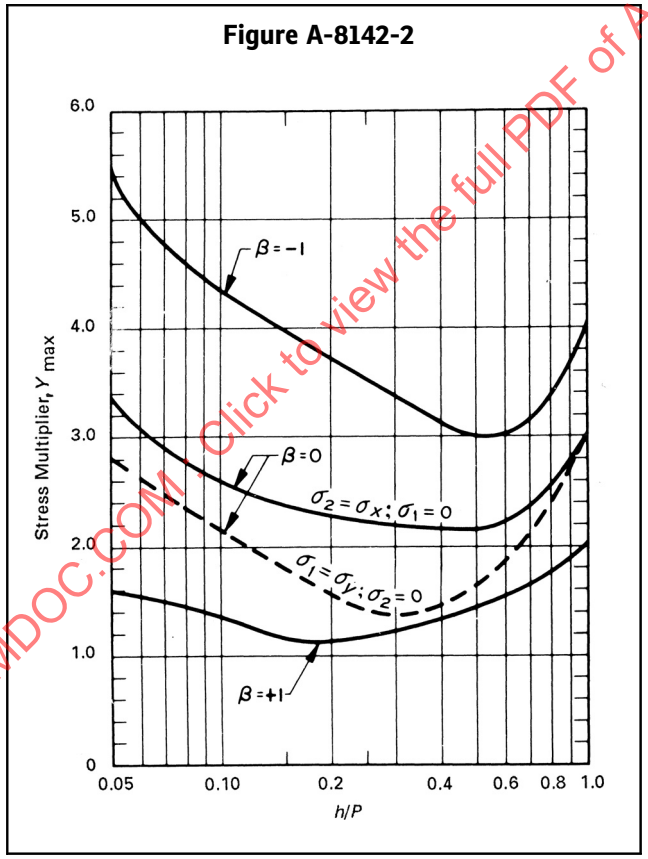
Y_{max} = stress multiplier given in Figure A-8142-2 as a function of the biaxiality ratio $\beta = \sigma_2/\sigma_1$

σ_1 = principal stress being the largest absolute value in the plane of the equivalent solid plate, psi (MPa)

σ_2 = principal stress having the smallest absolute value in the plane of the equivalent solid plate, psi (MPa) (Equivalent solid plate stresses due to various loads shall be superimposed in order to obtain σ_1 and σ_2 before any multipliers are applied, and the signs of σ_1 and σ_2 should be maintained.)

The solid curves in Figure A-8142-2 give the maximum stress multipliers for the worst angular orientation of σ_1 and σ_2 with respect to the axes of symmetry x and y of the hole pattern. In some cases, the worst orientation may not exist anywhere in the plate, and the use of lower stress multipliers is justified. An important case concerns the thermal stress produced by a temperature gradient across the diametral lane in a perforated plate. Such a gradient causes a uniaxial stress oriented parallel to the diametral lane. If the diametral lane is parallel to the y axis as shown in Figure A-8142-3, the stress multiplier given by the dashed line in Figure A-8142-2 may be used.

Figure A-8142-2



Equation (20) will give the maximum stress intensity for any loading system. Equation (20) is not adequate for more complex cyclic histories where the angular orientation of the maximum stress intensity varies during the cycle. In such cases, it is necessary to compute the stress history at each angular orientation ϕ using eq. (21)

$$S\phi = Y_1(P/h)\sigma_1 + Y_2(P/h)\sigma_2 + p_s \qquad (21)$$

where

$S\phi$ = peak stress intensity at the angular orientation ϕ

Y_1, Y_2 = stress multipliers in Figures A-8142-3 through A-8142-5 for various orientations of the principal stresses σ_1 and σ_2 computed for the equivalent solid plate

Note that these figures give stress multipliers for particular angular orientation only. The graph for the angular orientation closest to the actual angular orientation should be used. This is sufficiently accurate since the maximum possible difference between the actual orientation and the nearest orientation given in Figures A-8142-3 through A-8142-5 is only 7.5 deg. Examples for the computation of $S\phi$ are given as follows.

Example 1

The combined stresses in the equivalent solid plate for a perforated plate of 0.05 ligament efficiency were computed at a point as:

$$\sigma_r = 750 \text{ psi}$$

$$\sigma_\theta = -1100 \text{ psi}$$

and σ_r is rotated 12 deg, measured from the y axis of the hole pattern. To determine the value of $\sigma\phi$ at 40 deg from the y axis, use the following procedure: let $\sigma_1 = \sigma_r$, $\sigma_2 = \sigma_\theta$. Since the angular orientation of 12 deg is closest to 15 deg, use Figure A-8142-5 for the stress multipliers. Read at $\phi = 40$ deg on Scale A: $Y_1 = +1.65$; on Scale B: $Y_2 = -0.70$. Then from eq. (21), the peak stress intensity is computed as

$$S\phi = \frac{1}{0.05}(+1.65)(750) + \frac{1}{0.05}(-0.70)(-1100)$$

$$= 40,200 \text{ psi}$$

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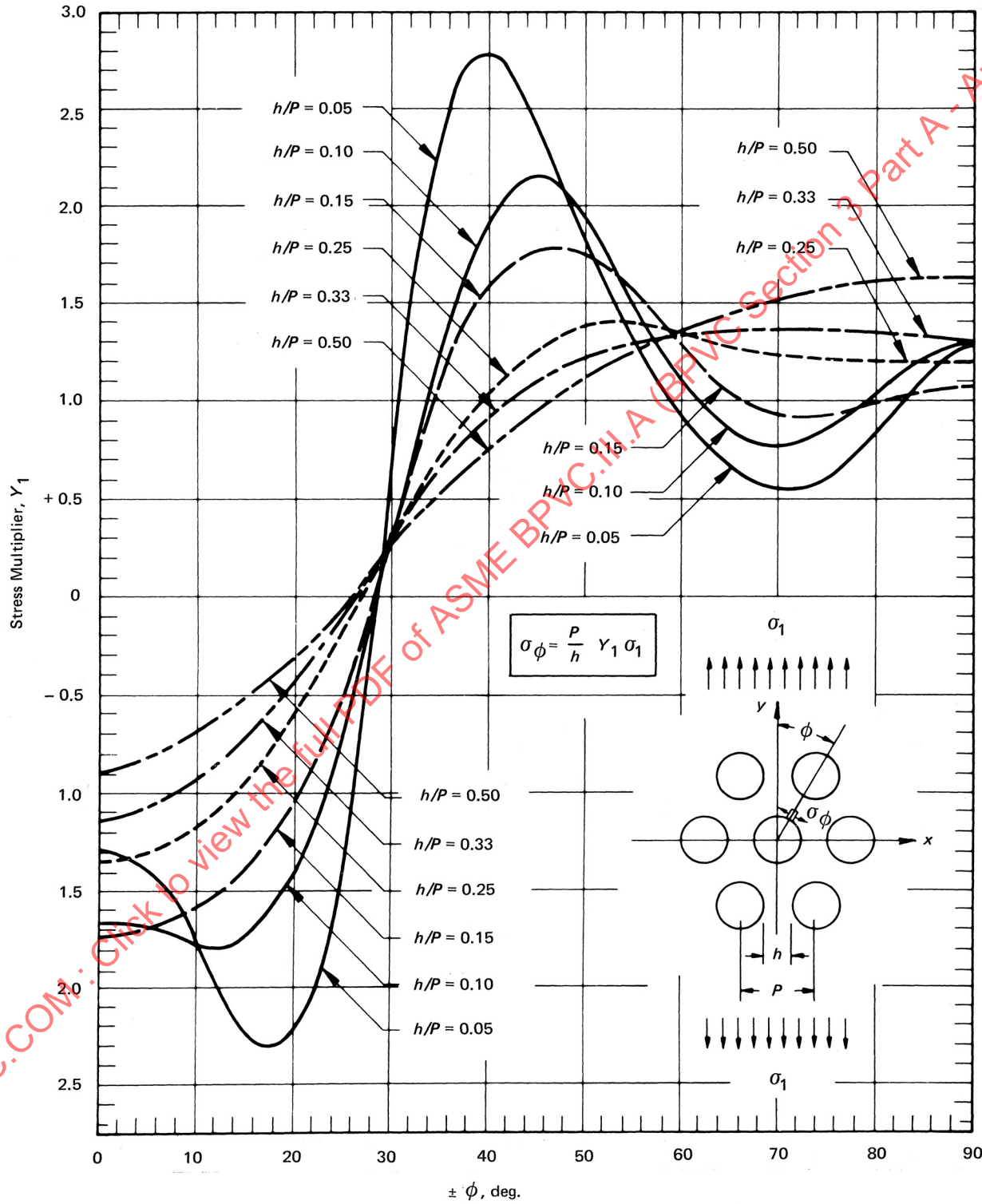
Figure A-8142-3



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Example 2

For the same plate as above at another point, the direction of σ_r coincides with the x axis. Let $\sigma_r = \sigma_2$, $\sigma_\theta = \sigma_1$. Read at $\phi = 40$ deg from Figure A-8142-3, $Y_1 = +2.75$ and from Figure A-8142-4, $Y_2 = -1.80$. Then

$$S\phi = \frac{1}{0.05}(2.75)(-1100) + \frac{1}{0.05}(-1.80)(750)$$
$$= -87,500 \text{ psi}$$

(c) The peak stress intensity at the outermost hole is computed from

$$S = K_r \sigma_{\text{rim}} + p_s \quad (22)$$

where

K_r = a stress multiplier from Figure A-8142-6
σ_{rim} = the nominal circumferential stress in the rim, psi (MPa)

The stresses given by eqs. (b)(20) and (b)(21) and by eq. (22) are limited by cumulative fatigue considerations, as described in NB-3222.4.

A-8143 Irregular Ligament Patterns or Thin Ligaments in a Nominally Uniform Pattern

For irregular ligament patterns or thin ligaments in a nominally uniform pattern, the stresses are determined as given in the following subparagraphs.

A-8143.1 Average Stress Intensity. The stress intensity based upon the ligament stresses averaged across the ligament width and through the plate thickness due to pressure plus other mechanical loads is limited to $3.0S_m$ in accordance with Table NB-3217-1. The appropriate value is computed according to A-8142.1(a), where h_a (the actual width of the thin ligament) is used in place of the nominal width h.

(15) **A-8143.2 Peak Stress Intensity.** The peak stress intensity in the thin ligament due to mechanical loading and structural interaction with adjacent members, including thermal effects, is limited by cumulative fatigue considerations. This peak stress intensity is computed by multiplying the peak stress intensity for a nominal thickness ligament by the K_m value given in Figure A-8143.2-1.

(a) The peak stress intensity in the nominal ligament is calculated as indicated in A-8142.2(b).

A-8150 THERMAL SKIN EFFECT

A-8151 General Considerations

In certain cases, the temperature gradient through the thickness of a perforated plate can be closely approximated by a step change in the metal temperature near the surface of the plate. In such a case, significant thermal stresses develop only in the skin layer of the plate at the surface where the temperature change occurs and the thermal stresses in the remainder of the plate are negligible.

A-8152 Maximum Thermal Skin Stress (15)

The maximum thermal skin stress on the surface of a perforated plate can be computed from the relation:

$$\sigma_{\text{skin}} = K_{\text{skin}} Y_{\max} \left(\frac{P}{h}\right)\left(\frac{E\alpha}{1-\nu}\right)(T_m - T_S) \quad (23)$$

where

E, α, ν = modified material properties
h = ligament width
P = pitch, in. (mm)
T_m = mean temperature of the plate
T_s = temperature of the plate at the surface under consideration
$Y_{\max}$ = stress multiplier from Figure A-8142-2, for $\beta = +1$

A-8153 Peak Stress Intensities When Thermal Skin Stresses Are Included (15)

(a) When thermal skin stresses are to be combined with other stresses to obtain the peak stress intensity, eq. A-8152(23) may not be used. In such a case the thermal stresses at any location on the surface of the equivalent solid plate are given by:

$$\sigma_r = \sigma_\theta = K_{\text{skin}} \frac{E\alpha}{1-\nu}(T_m - T_s) \quad (24)$$

where

E, ν = unmodified material properties (since K_{skin} includes the consideration for E^* and ν^*)
K_{skin} = stress ratio from Figure A-8153-1
T_m = mean temperature of the plate
T_s = surface temperature of the plate

(b) The equivalent solid plate stresses given by eq. (24) can then be combined with other solid plate stresses and the method given in A-8142.2(b) can be used to obtain the peak stress intensity.

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Figure A-8142-4

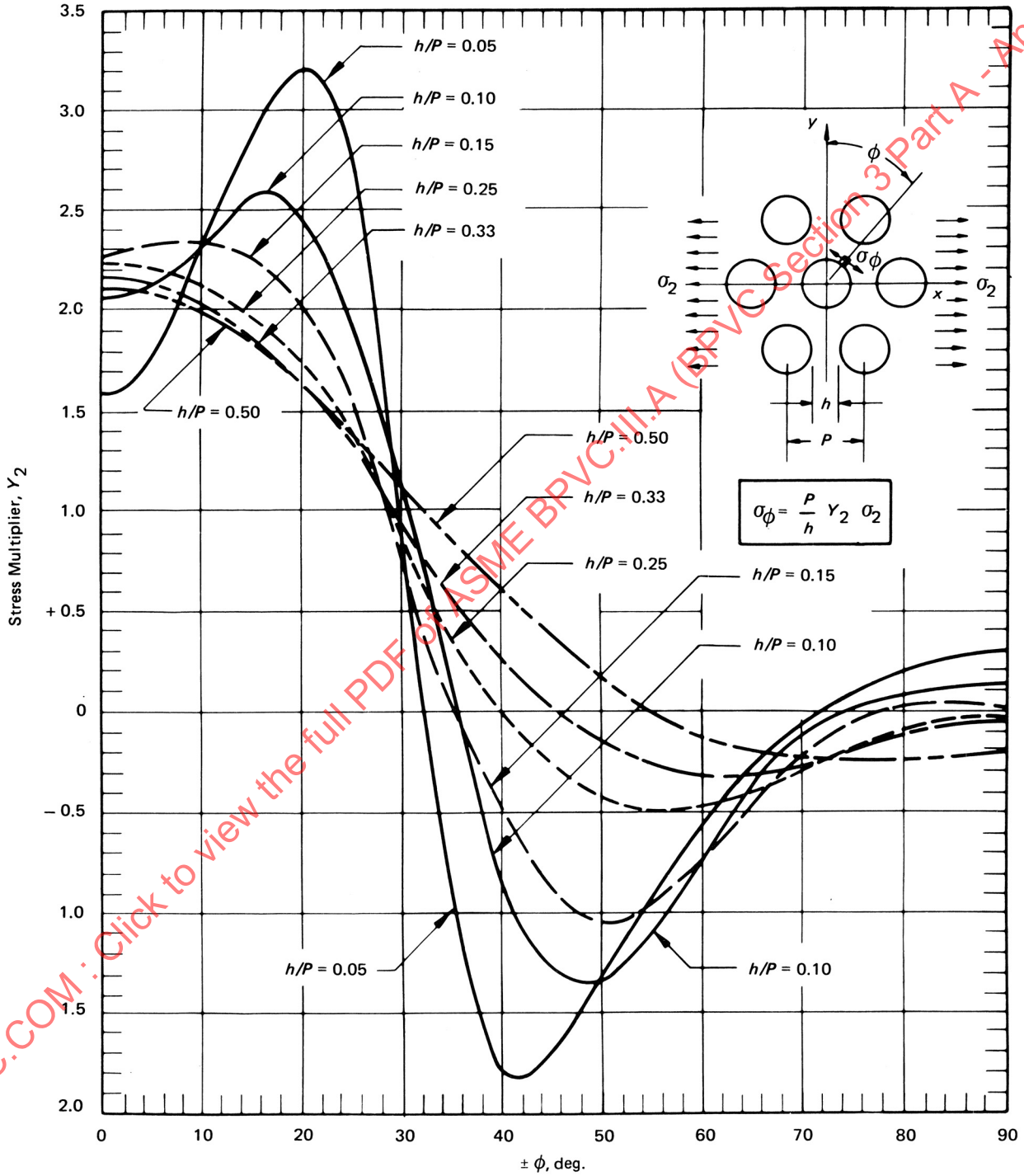

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Figure A-8142-5

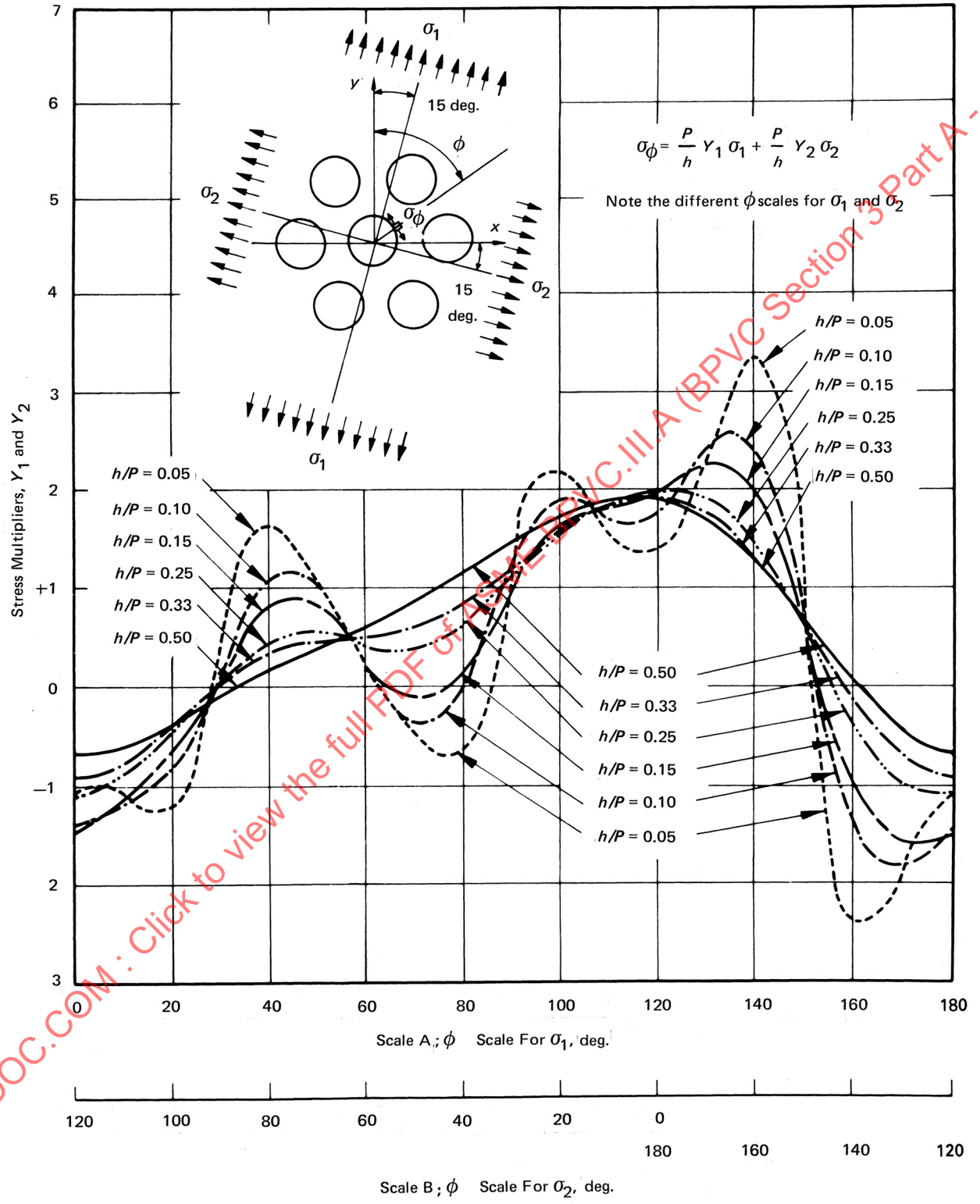

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Figure A-8142-6

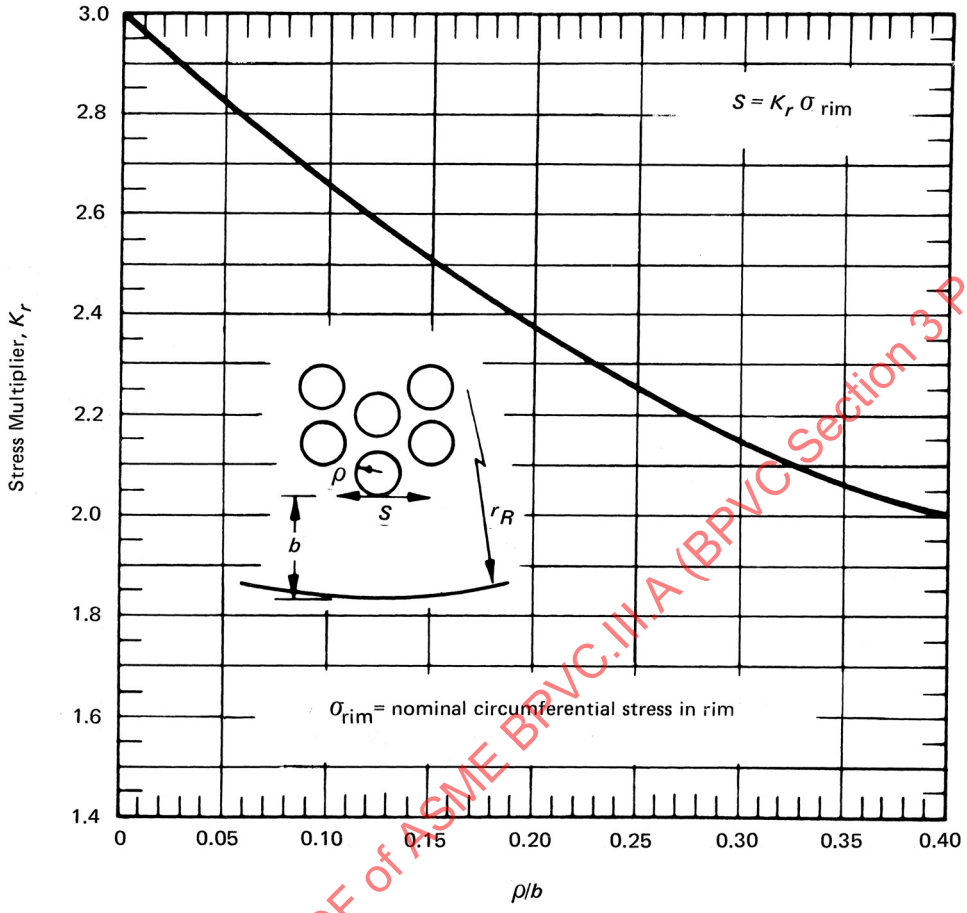

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Figure A-8143.2-1

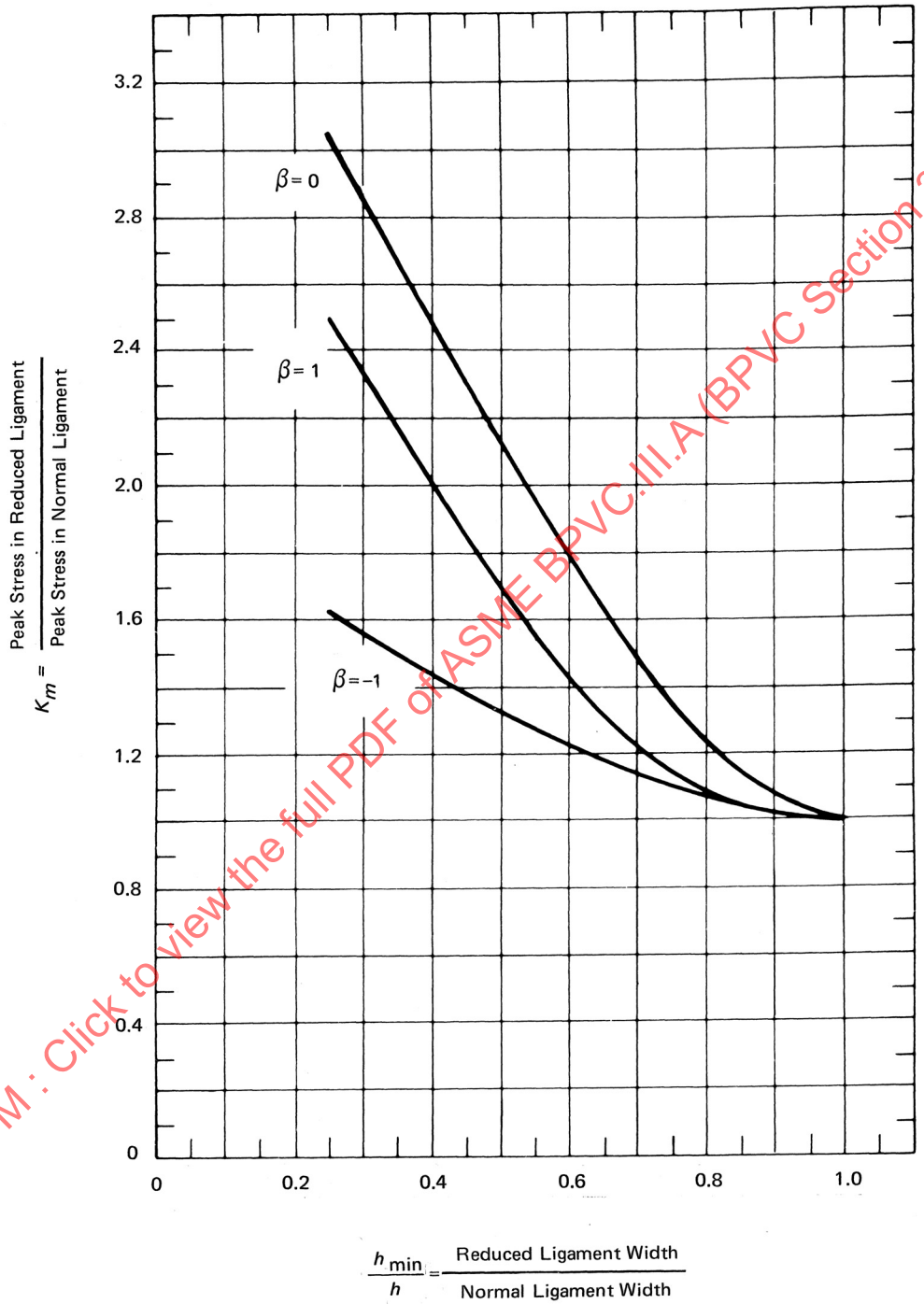

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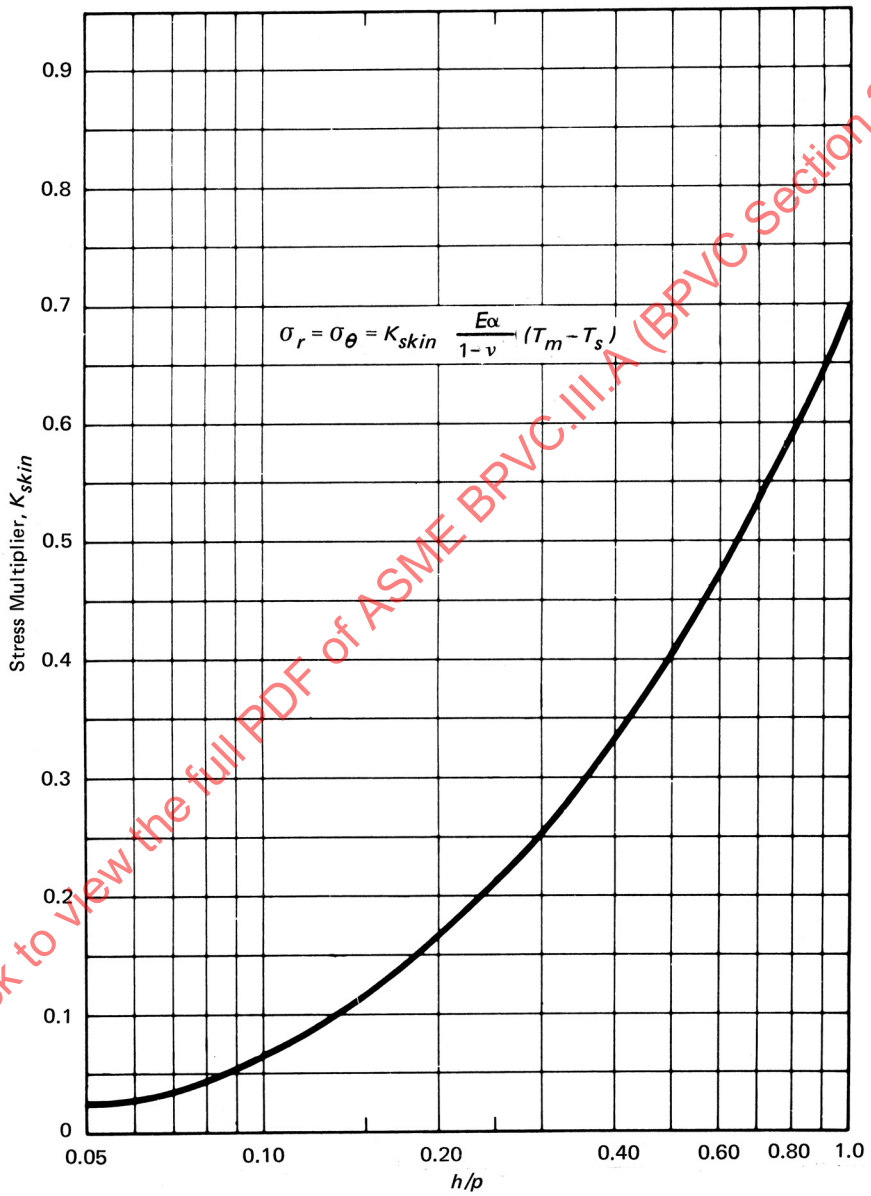
Figure A-8153-1



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ARTICLE A-9000
INTERACTION METHOD

A-9100 INTRODUCTION

A-9110 SCOPE

(a) This Article contains a method for evaluating the adequacy of linear structural elements under combined loads, without determining principal stresses, by use of the stress ratio/interaction curve method. By using an interaction formula for combined stress states the ability of a linear structural element to withstand combined loads can then be determined provided the strength of the element under each individual load is known. The method can be applied to elastic and inelastic problems, including elastic and inelastic stability, and is useful when an exact stress analysis is not practical.

(b) A general interaction formula for three states of stress is given by the following:

$$R_1^{\,p} + \left(R_2^{\,q} + R_3^{\,r}\right)^s \le 1.0 \qquad (1)$$

where R_1, R_2, and R_3 are ratios of either individual stresses, stress resultants, or loads to their respective allowables; and the exponents p, q, r, and s constitute the interaction relationship. These exponents are based upon experimental and/or theoretical considerations. Generally speaking, such an interaction is set up for each individual element in a structure (each beam, column, etc.), and each element will have its own set of exponents for the loads to which it is subjected.

(c) For elastic analysis of compact structures (those in which buckling need not be considered), interaction methods can be used to determine the yield surface. However, classical strength of material methods can also be used to obtain principal stresses, hence an interaction method is not of importance. For ultimate strength, an exact stress analysis is frequently impractical and interaction methods provide a useful alternative. In addition, for structures subject to more than one type of load which can cause instability (e.g., torsional and axial buckling of thin-walled tubes or pipes), interaction methods can again be used.

A-9120 NOMENCLATURE

(a) Definitions of the symbols used in this Article

A = cross-sectional area
c = distance from neutral axis to outermost fiber
e = strain
f_{ap} = linearized allowable bending stress (apparent stress)
f_{uk} = linearized ultimate bending stress for section factor K
f_{yk} = linearized yield bending stress for section factor K
I = moment of inertia
K = section factor
M = allowable bending moment
m = applied moment
n = interaction exponent
P = axial load
Q_m = first moment of the area between the neutral axis and outer fiber

$$= \int_o^c y\,da$$

R = ratio of an individual stress or load to its allowable
S = stress
S_o = trapezoidal intercept stress
U = stress field utilization factor
x = centroidal axis, x direction
x' = principal axis, x' direction
y = centroidal axis, y direction
y_p = distance from principal axis to intermediate fiber
y' = principal axis, y' direction
γ = plasticity factor
ϕ = angle between centroidal and principal axis, deg.

(b) Indices used with the symbols in this Article

al = allowable
ap = apparent
b = bending
bc = buckling
c = compression
pl = proportional limit
s = shear
t = tension
to = torsion
u = ultimate
y = yield
1, 2 = locations across a section

A-9200 INTERACTION EQUATIONS

A-9210 SCOPE

(a) This Subarticle provides interaction equations based on experimental data for a number of common structural shapes.

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(b) Allowable loads and stresses for the interaction equations presented herein shall be determined in accordance with A-9300.

(c) Interaction equations for combinations of loads other than those specified herein may be used, provided they are developed in accordance with the rules of A-9400.

(d) Interaction equations, which may be used for common beam shapes subject to various combinations of loads, are presented in Table A-9210(d)-1. As an alternative to some of the interaction equations given in Table A-9210(d)-1, the curve in Figure A-9210(d)-1 may be used.

(e) All structural shapes subject to buckling shall be governed by the requirements of NF-3300.

(f) Interaction equations which may be used for thin- and thick-walled tubes and pipes, subject to various combinations of loads, are presented in Table A-9210(f)-1 (in the course of preparation).

(g) Interaction equations which may be used for flat, unperforated plates, subject to various combinations of loads, are presented in Table A-9210(g)-1 (in the course of preparation).

A-9300 ALLOWABLE LOADS AND STRESSES

A-9310 SCOPE

This Subarticle provides criteria for determining the allowable loads for components or supports subject to the application of one or more loads. The allowable loads are to be based on the allowable stresses set forth in F-1331.2 or F-1341.5, as appropriate, for either elastic or plastic system analysis.

A-9311 Material Properties

The material properties used in developing the allowable component or support loads or stresses shall be based on Section II, Part D, Subpart 1, and included and justified in the Design Report.

A-9312 Strain Rate Effects

Strain rate effects on material properties may be considered if justified in the Design Report.

A-9313 Temperature Effects

Temperature effects on the allowable component or support loads or stresses shall be considered and justified in the Design Report.

A-9314 Allowable Load

The allowable load of a component or support is defined as the lesser of (a) through (d).

(a) The load at which the most severely stressed fiber reaches the allowable stress defined in F-1331.2 or F-1341.5, as appropriate.

(b) The load at which either strain or deformation exceeds the limits provided by the component or support Design Specification.

(c) The load at which loss of component or support function occurs, as defined by the component or support Design Specification.

(d) The allowable buckling load as defined in F-1334.3.

Figure A-9210(d)-1
Interaction Curve for Beams Subject to Bending and Shear or to Bending, Shear, and Direct Loads

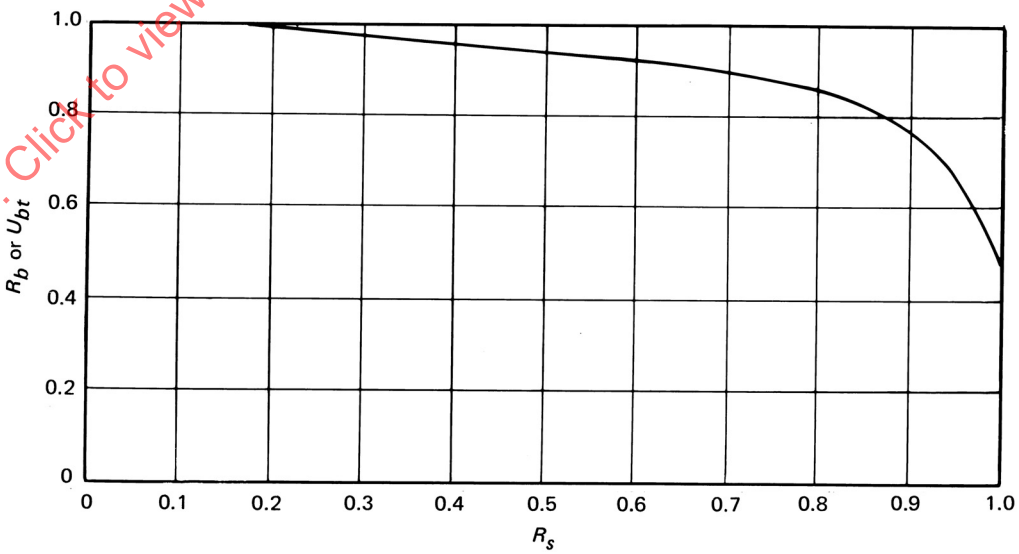

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Table A-9210(d)-1
Interaction Equations for Common Beam Shapes

Type of Load	Interaction Equation [Note (1)] and [Note (2)]	Remarks
Simple bending	$R_b < 1$	$R_b = m/M$
Complex bending	$R_{bx}' + R_{by}' < 1$	$R_{bx}' = m_x'/M_x'$, etc.
Simple shear	$R_s < 1$	$R_s = S_s/S_{sal}$
Complex shear	$\sqrt{R_{sx}'^2 + R_{sy}'^2} < 1$	$R_{sx}' = S_{sx}'/S_{sal}'$ etc.; S_{sx}' and S_{sy}' are maximum shear stresses
Simple bending plus shear	$\sqrt{R_b^2 + R_s^2} < 1$	[Note (3)]
Complex bending plus shear	$\sqrt{R_{b'}^2 + R_{s'}^2} < 1$	$R_{b'} = R_{bx'} + R_{by'}; R_{s'} = \sqrt{R_{sx'}^2 + R_{sy'}^2}$; [Note (1)]
Simple or complex bending plus tension	$R_b' + R_t^n < 1$	$R_t = P_t/P_{tal}$; to determine n use A-9532
Simple or complex bending, tension, and shear	$\sqrt{\left(R_{b'} + R_t^n\right)^2 + R_{s'}^2} < 1$	[Note (3)]; to determine n use A-9532; see A-9533
Simple or complex bending and compression	$R_b' + R_c < 1$	[Note (3)] and [Note (4)]; $R_c = P_c/P_{cal}$
Simple or complex bending, compression, and shear	$\sqrt{\left(R_{b'} + R_c\right)^2 + R_{s'}^2} < 1$	[Note (3)] and [Note (4)]; see A-9535

NOTES:
(1) Allowable loads for use in interaction equations should be based on allowable stresses as defined in A-9300.
(2) All interaction ratios R_i are positive by definition.
(3) As an alternate to the given interaction equation, the curve of Figure A-9210(d)-1 may be used.
(4) Amplification of bending moment by axial load shall be taken into account.

A-9320 METHOD

(a) The allowable load of a component or support under the application of a single load may be determined by experimental or analytical methods or both. The allowable loads of a component or support thus determined are to be modified by the effects discussed in A-9312 and A-9313 for use in the interaction equations of A-9200.

(b) An acceptable method of determining the allowable loads of beam shapes in pure bending or in bending in combination with direct loads and shear is the apparent stress method provided in A-9500.

(c) An acceptable method of determining the allowable loads of pipes and tubes in pure bending or in bending in combination with direct loads and shear is provided in A-9600 (in the course of preparation).

A-9400 NEW INTERACTION EQUATIONS

A-9410 SCOPE

Interaction equations other than those provided in A-9200 may be used for the analysis of components or supports, provided they are developed in accordance with the rules of this Section.

A-9420 METHOD

Any new interaction equations to be used shall be included and justified in the Design Report. They may be justified by one of the following:

(a) common appearance in appropriate technical literature or in industry codes or standards;

(b) experimental development which includes a variation of all types of loads or stresses that appear in the interaction equations. Such a load or stress variance shall bound the loads or stresses to which the component is subjected;

(c) theoretical development which includes testing to verify the interaction equations developed.

A-9500 DETERMINATION OF ALLOWABLE BENDING STRENGTH OF BEAMS BY THE APPARENT STRESS METHOD

A-9510 SCOPE

(a) This Subarticle provides a method to calculate the strength of beams in the plastic range under pure bending or under bending combined with direct loads and shear. It is based on the work of Cozzone[20, 21] and utilizes a fictitious stress called an *apparent stress*. This method shall not be used for the analysis of thin-walled tubes or pipes.

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(b) The conventional beam theory, based on the assumption that a plane section before bending remains plane after bending, gives a linear distribution of strain and stress in the elastic range up to the proportional limit. Beyond the proportional limit, however, although the strain distribution is assumed to remain linear, the stress distribution corresponds with the stress–strain relationship for the material. An approximation of this distribution has been obtained, which enables the prediction of the effects of shape and material properties on bending in the plastic range. This method has the advantage that strain hardening may be taken into account.

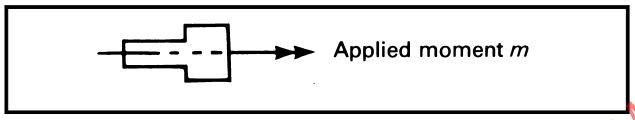
(c) The methods provided herein may also be used for the analysis of beams with cutouts or notches, provided that the geometric properties are based on the net area at the cutout or notch.

(d) The effect of cyclic loading should be evaluated independently, where appropriate.

A-9520 SIMPLE BENDING

A-9521 Simple Bending — Symmetrical Sections

(a) The method given below may be used when the resultant applied moment vector is parallel to a principal axis which is also an axis of symmetry.



(b) The section factor K is given by the formula $K = 2Q_m/(I/c)$. If $K > 2.0$, use $K = 2.0$. Section factors are given in Table A-9521(b)-1 for common structural shapes.

(c) The allowable bending moment shall be determined as given in (1) through (3).

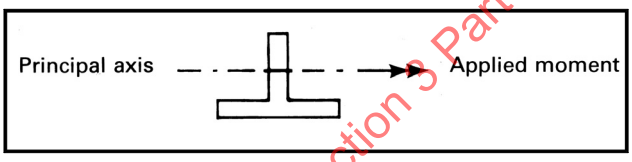
(1) Using the method outlined in A-9540, derive the relationship between the allowable stress S_{al} and the linearized allowable stress f_{ap} for the proper section factor K. An example calculation for SA-672 A50 material at 600°F is provided in A-9542, with the resulting relationship shown in Figure A-9542-1.

(2) Using the value of S_{al} set forth in F-1331.2 or F-1334.7, as appropriate, determine the value of f_{ap} for the proper K.

(3) Multiply f_{ap} from the preceding (2) by I/c to obtain the allowable moment M.

A-9522 Simple Bending — Unsymmetrical Section

(a) Use the following method when the resultant applied moment vector is parallel to a principal axis which is not an axis of symmetry.



(1) Break the section down into the two parts on either side of the principal axis. For each part, compute Q_m, I, and I/c about the principal axis of the original complete section.

(2) Compute $K = Q_m/(I/c)$ for each part. In utilizing the K value for each part, computed as above, it will be the same as for a symmetrical section composed of the given part and its reflection about the principal axis of the original section.

(b) The allowable bending moment shall be determined as given in (1) through (4) below.

(1) Use the method given in A-9521(c) and the K value of the part with the larger c to obtain an f_{ap} value for this part.

(2) Determine the strain e_{ap} associated with S_{al} from the engineering stress–strain curve.

(3) Obtain the allowable maximum strain e_1 in the part having the smaller c by the following equation:

$$e_1 = (\text{smaller } c \,/\, \text{larger } c)e_{ap}$$

Table A-9521(b)-1

SECTION FACTORS

Flanges Only			Solid Round	
$K = 1$	$K = 1$ to 1.5	$K = 1.5$	$K = 1.7$	$K = 2.0$

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Enter this strain e_1 on the stress–strain curve and obtain the corresponding stress from the stress–strain curve. Use this stress value as the allowable stress S_{al} and, with the K value for the part with the smaller c, use the method of A-9521(c) to obtain f_{ap} for this part.

(4) Multiply the f_{ap} value for each part by I/c of each part and add the two to obtain the total allowable moment M.

A-9523 Complex Bending — Symmetrical and Unsymmetrical Sections

This condition occurs when the resultant applied moment vector is not parallel to a principal axis.

A-9523.1 Sign Convention and Nomenclature. In Figure A-9523.1-1, let x and y represent two mutually perpendicular centroidal axes, and let x' and y' represent the principal axes.

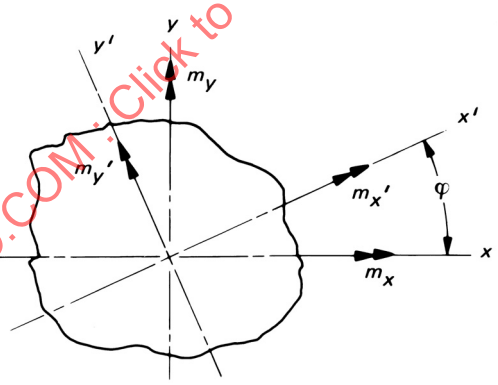
A-9523.2 Resolution of Complex Bending Into Simple Bending. Any case of complex bending may be resolved into two cases of simple bending about the principal axes of the section. The principal axes are defined as mutually perpendicular centroidal axes about which the moments of inertia are a maximum and minimum, respectively, and about which the product moment of inertia is zero. The procedure is given in (a) through (d) below.

(a) Determine the principal axes x' and y'. If they cannot be determined by inspection, obtain I_x, I_y, and I_{xy} about any arbitrary pair of centroidal axes.

$$\tan 2\phi = 2I_{xy} / \left(I_y - I_x\right)$$

(b) Using the x' axis as a reference, determine the allowable moment $M_x{}'$ as described under simple bending (A-9521 and A-9522).

Figure A-9523.1-1
Sign Convention and Nomenclature



GENERAL NOTE: Moment vectors are designated by double-headed arrows and are to be interpreted by the left-hand rule: point left thumb in direction of vector and natural curl of fingers will designate the direction of moment.

(c) Using the y' axis as a reference, determine the allowable moment $M_y{}'$ as described under simple bending (A-9521 and A-9522).

(d) For use in the interaction equations of A-9210, moments in the global axis shall be resolved into moments about the principal axes by use of the following relationships:

$$m_{x'} = m_x \cos\phi + m_y \sin\phi$$

$$m_{y'} = m_x \sin\phi + m_y \cos\phi$$

A-9530 BENDING COMBINED WITH A STRESS FIELD

A-9531 Interaction — Simple or Complex Bending and Shear

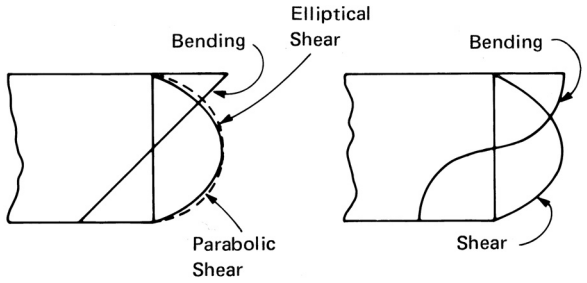
The maximum shear stress in a beam usually occurs at the principal (neutral) axis where the bending stress is zero. The maximum bending stress occurs at an extreme fiber where the shear stress is usually zero.

(a) In the elastic range, the distribution of shear and bending stresses (see Figure A-9531-1) is usually such that the most critical point in the section is at either the principal axis or the extreme fiber. This is true on a rectangular section since the shear distribution across the section is parabolic and the bending distribution is linear. If the shear distribution has been elliptical, every point in the cross section will be equally critical in combined stress based on circular interaction.

(b) In the plastic range, however, the distribution of the shear stress as well as the bending stress differs from that in the elastic range. This results in intermediate points which frequently become more critical in combined stress than either the shear stress at the principal axis or the bending stress at the extreme fiber.

(c) To find the most critical point would require the calculation of combined stresses at a series of points across the section. This procedure would not only be laborious but probably incorrect in the conservative direction, since there would undoubtedly be some redistribution of stress

Figure A-9531-1
Bending and Shear Stresses



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away from the most critical point, although the exact nature of this redistribution appears to be extremely difficult to determine. Therefore, the procedure of (1) through (4) below shall be used.

(1) The method used to determine the shear flows for simple or complex bending shall be included and justified in the Design Report.

(2) For complex bending, the maximum principal shear stresses $S_{sx'}$ and $S_{sy'}$ shall be determined for use in the interaction equations of A-9210.

(3) For simple or complex bending, the allowable shear stress S_{sal} shall be taken as $0.6S_{tal}$.

(4) The allowable moments shall be determined as in A-9521 or A-9522, as appropriate.

A-9532 Interaction — Simple or Complex Bending and Tension

(a) The allowable moments for simple or complex bending shall be determined from A-9521 or A-9522, as appropriate.

(b) The allowable tensile load P_{al} shall be taken as AS_{al}.

(c) The interaction exponent n for use in the interaction equations of A-9210(d) shall be determined from (1) through (3) below.

(1) Determine $Ac/2Q_m$ for use in obtaining the interaction exponent. If the section is unsymmetrical, take c for the side for which axial and bending stresses are of opposite sign. For complex bending, obtain both $Ac/2c_{x'}$ and $Ac/2c_{x'}$. In obtaining $Ac/2c_{x'}$ if the section is unsymmetrical about the x' axis, take $c_{x'}$ for the side for which axial stress and stress due to $m_{x'}$ are of opposite sign. Obtain $Ac/2c_{y'}$ in the same manner.

(2) The material plasticity factor for use in obtaining the interaction exponent is $\gamma = 0.90$ for all materials.

(3) Using $Ac/2Q_m$ determined above in (1) and γ determined above in (2), obtain n from Figure A-9532(c)(3)-1. To determine the interaction exponent for complex bending, obtain both $n_{x'}$ and $n_{y'}$ from Figure A-9532(c)(3)-1 and determine a combined n using $n = (n_{x'}R_{bx'} + n_{y'}R_{by'})/R_b$.

A-9533 Interaction — Simple or Complex Bending, Tension, and Shear

When bending acts in addition to tension and shear, allowables shall be determined as provided in (a) through (c) below.

(a) Follow the procedure outlined in A-9531 and A-9532 to obtain S_{sal}, P_{al}, $M_{x'}$, $M_{y'}$, and n.

(b) Plot the curve of $R_b + R_t{}^n = 1$ [Figure A-9533(b)-1] and the intersection of R_b and R_t. Call this point A. Obtain $a = \overline{0A}$ and $b = \overline{0B}$.

(c) Obtain the bending-tension utilization factor for use in Figure A-9210(d)-1:

$$U_{bt} = a/b$$

A-9534 Interaction — Simple or Complex Bending and Compression

(a) When compression acts in addition to simple or complex bending, the applied moment m shall take into account the additional bending caused by the compressive load.

(b) The allowable moments for simple or complex bending shall be determined from A-9520, as appropriate.

(c) The allowable compressive load P_{cal} shall be taken as the lesser of AS_{al} and the allowable buckling load F-1334.3.

A-9535 Interaction — Simple or Complex Bending, Compression, and Shear

When bending acts in addition to compression and shear, the following procedure may be used to determine the interaction relationships for use in A-9210.

(a) Follow the procedure in A-9534 to obtain the applied and allowable moments and the allowable compressive load.

(b) Obtain the bending-compression utilization factor $U_{bc} = R_b + R_c$.

(c) U_{bc} may be used in Figure A-9210(d)-1 by replacement of U_{bt} by U_{bc}.

A-9540 PROCEDURE FOR DETERMINATION OF ALLOWABLE BENDING STRESS

A-9541 Derivation of Linearized Allowable Bending Stress for Any Material

A fictitious bending stress, called the linearized allowable bending stress f_{ap}, may be used for establishing the bending strength of a material. This method assumes that in the plastic region the nonlinear stress–strain relationship for a particular section and material can be approximated by a trapezoidal shape as shown in Figure A-9541-1.

The stress S_o is a fictitious stress which is assumed to exist at the neutral axis or at zero strain. The value of S_o is determined by requiring that the internal moment of the engineering stress–strain curve must equal the internal moment of the assumed trapezoidal shape. Thus, the total moment capacity of a symmetrical section may be expressed as follows:

$$\begin{aligned} M &= 2S_o\int_0^c yda + (S_{al} - S_o)(I/c) \\ &= 2S_oQ_m + (S_{al} - S_o)(I/c) \\ &= (I/c)\{S_{al} + S_o[2Q_m/(I/c) - 1]\} \\ &= (I/c)[S_{al} + S_o(K - 1)] \end{aligned} \qquad (2)$$

Thus, the linearized allowable stress for any section factor K becomes

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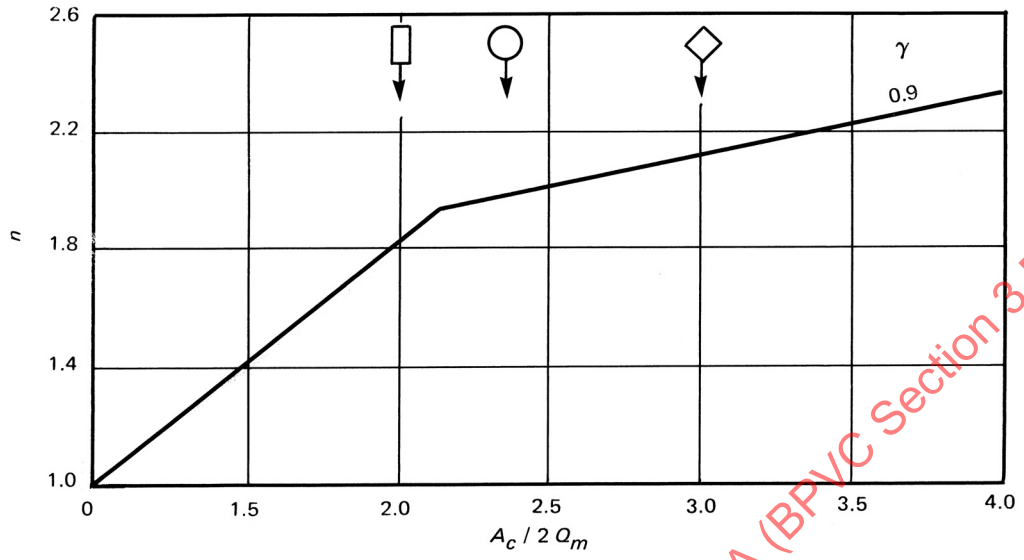
Figure A-9532(c)(3)-1
Interaction Exponent



$$f_{apk} = S_{al} + (K - 1)S_o \quad (3)$$

and the moment capacity of the section is as follows:

$$M = f_{apk} I / c \quad (4)$$

An alternate method of determining f_{ap} for any section factor is to express it in terms of the linearized allowable bending stress for a 1.5 stress factor:

$$f_{apk} = f_{ap\,1.5} + (K - 1.5)S_o \quad (5)$$

When the allowable stress S_{al} is equal to the ultimate stress S_u, eqs. (3) and (5) become

$$f_{uk} = S_u + (K - 1)S_{ou} \quad (6)$$

$$f_{uk} = f_{u1.5} + (K - 1.5)S_{ou} \quad (7)$$

When the allowable stress S_{al} is equal to the yield stress S_y, eqs. (3) and (5) become

$$f_{yk} = S_y + (K - 1)S_{oy} \quad (8)$$

$$f_{yk} = f_{y1.5} + (K - 1.5)S_{oy} \quad (9)$$

where S_{ou} and S_{oy} are the trapezoidal intercept stresses corresponding to S_u and S_y.

In order to calculate the allowable moment of a given beam cross section by the use of eqs. (3) and (4), the intercept stress S_o must first be determined using the allowable moment, determined either by test or an exact stress analysis, that corresponds to a known value of K. Such effort would negate the advantage of this method. On the other hand, the values of S_{ou} and S_{oy} have been calculated for about 50 materials[22] and are shown in Figure A-9541-2. These curves may be utilized for the carbon, low, and high alloy steels given in Section II, Part D, Subpart 1. The corresponding values of $f_{u1.5}$ and $f_{y1.5}$ are shown in Figure A-9541-3.

Once f_{uk} and f_{yk} are determined from Figure A-9541-3 and eqs. (6) through (9), f_{ap} for any value of the allowable stress S_{al} may be determined as shown in (a) through (c) below.

(a) For $S_{al} \le S_{pl}$, where S_{pl} is the proportional limit stress (Figure A-9541-4),

$$f_{apk} = S_{al} \quad (10)$$

(b) For $S_{pl} \le S_{al} \le S_y$,

$$f_{apk} = S_{pl} + Z_{1k}(S_{al} - S_{pl}) \quad (11)$$

where

$$Z_{1k} = (f_{yk} - S_{pl}) / (S_y - S_{pl}) \quad (12)$$

(c) For $S_y < S_{al} \le S_u$,

$$f_{apk} = f_{yk} + Z_{2k}(S_{al} - S_y) \quad (13)$$

where

$$Z_{2k} = (f_{uk} - f_{yk}) / (S_u - S_y) \quad (14)$$

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Figure A-9533(b)-1
Interaction Curve for Bending and Tension

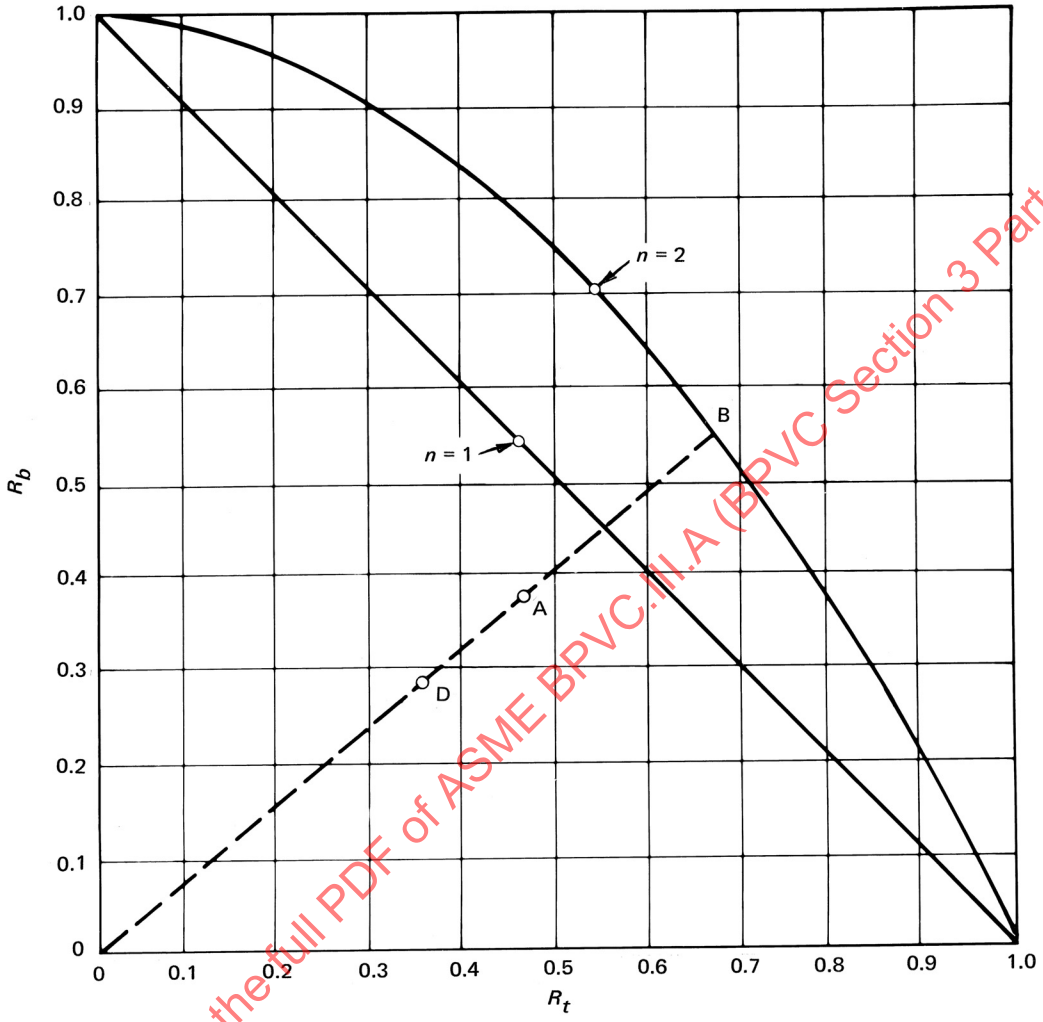
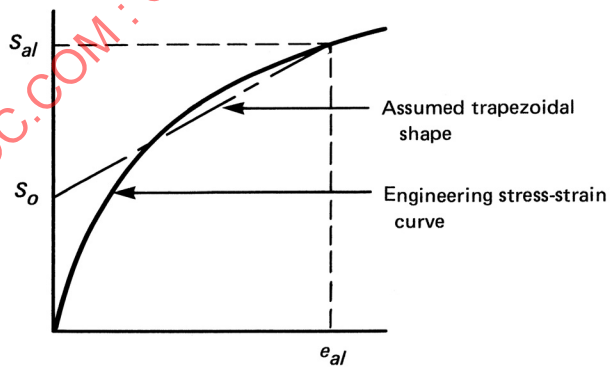


Figure A-9541-1
Trapezoidal Stress–Strain Relationship



A-9542 Example Illustrating the Derivation of Linearized Allowable Bending Stress for SA-672 A50 Material at 600°F

The values of S_y and S_u are given

S_u = 50.0 ksi ultimate tensile strength at 600°F (Section II, Part D, Subpart 1, Table U)

S_y = 20.0 ksi yield strength at 600°F (Section II, Part D, Subpart 1, Table Y-1)

The following values of S_{oy} and S_{ou} are found from Figure A-9541-2:

$$S_{oy} = 11 \text{ ksi}$$

$$S_{ou} = 44 \text{ ksi}$$

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Figure A-9541-2
Ultimate and Yield Trapezoidal Intercept Stresses

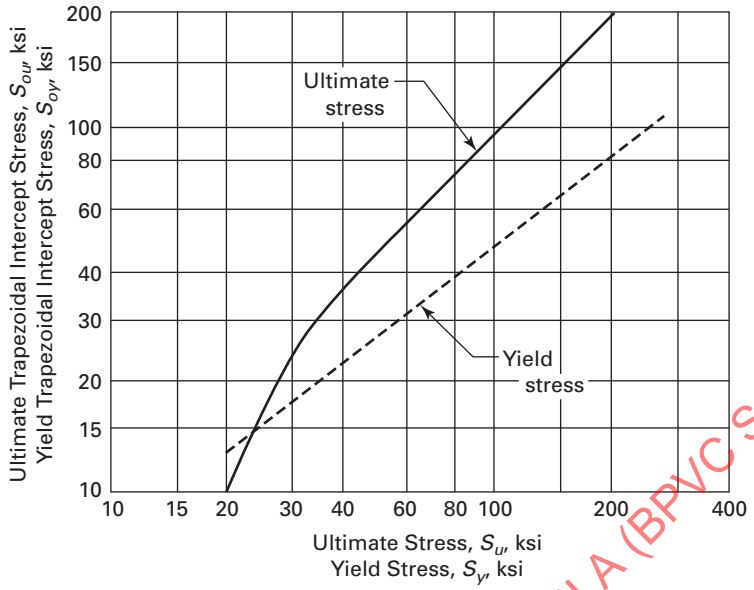

Figure A-9541-3
Linearized Ultimate and Yield Bending Stresses for Rectangular Section

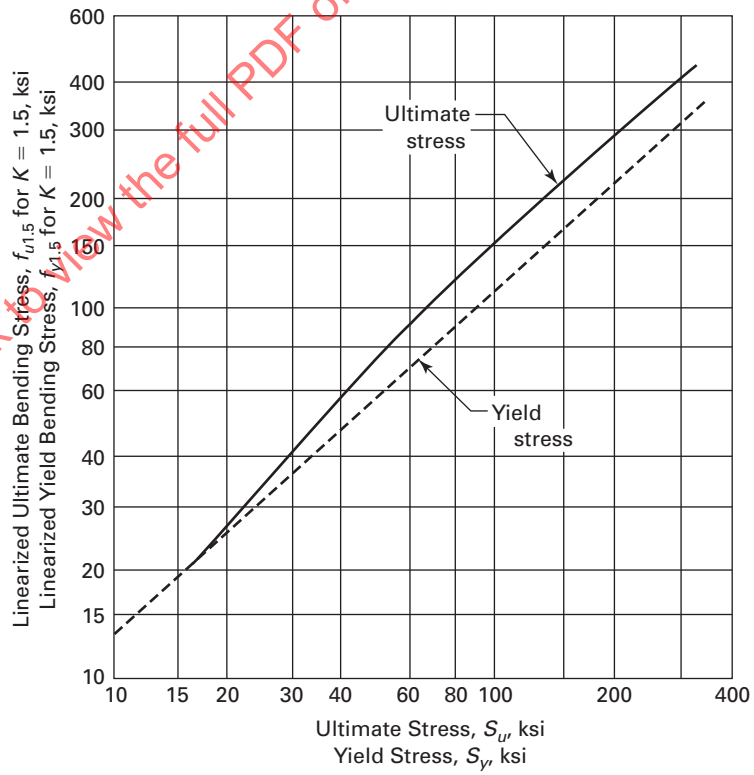

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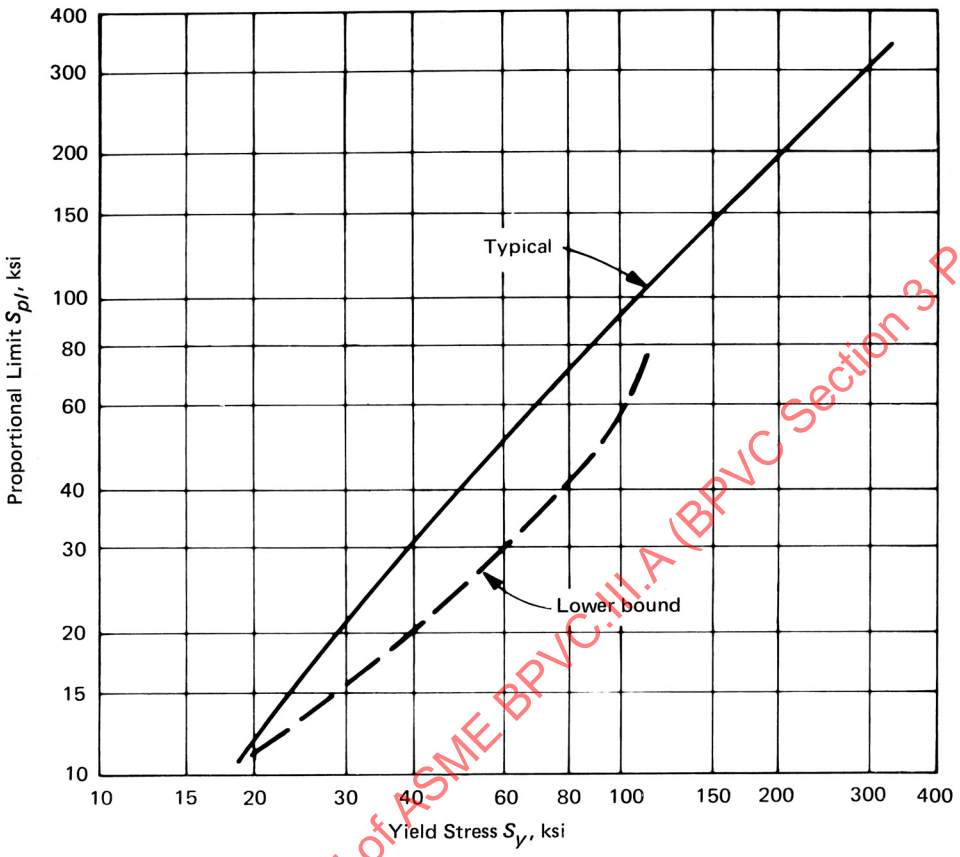
Figure A-9541-4
Proportional Limit As a Function of Yield Stress



Using eqs. A-9541(6) and A-9541(8),

$$f_{u1.5} = 50 + 0.5(44) = 72 \text{ ksi}$$

$$f_{y1.5} = 20.0 + 0.5(11) = 25.5 \text{ ksi}$$

or, using Figure A-9541-3,

$$f_{u1.5} = 72 \text{ ksi}$$

$$f_{y1.5} = 25 \text{ ksi}$$

Using eqs. A-9541(7) and A-9541(9), the value of f_{uk} and f_{yk}, for any K, is determined. For example, if K = 1.9, 1.7, 1.3, and 1.1, then

$$f_{u1.9} = 72.0 + 0.4(44.0) = 89.6 \text{ ksi}$$

$$f_{u1.7} = 72.0 + 0.2(44.0) = 80.8 \text{ ksi}$$

$$f_{u1.3} = 72.0 + (-0.2)(44.0) = 63.2 \text{ ksi}$$

$$f_{u1.1} = 72.0 + (-0.4)(44.0) = 54.4 \text{ ksi}$$

$$f_{y1.9} = 25.5 + 0.4(11.0) = 29.8 \text{ ksi}$$

$$f_{y1.7} = 25.5 + 0.2(11.0) = 27.6 \text{ ksi}$$

$$f_{y1.3} = 25.5 + (-0.2)(11.0) = 23.2 \text{ ksi}$$

$$f_{y1.1} = 25.5 + (-0.4)(11.0) = 21.0 \text{ ksi}$$

From Figure A-9541-4, S_{p1} = 13.0 ksi. Using the above data, Figure A-9542-1 may be obtained for K = 1.9, 1.5, and 1.1.

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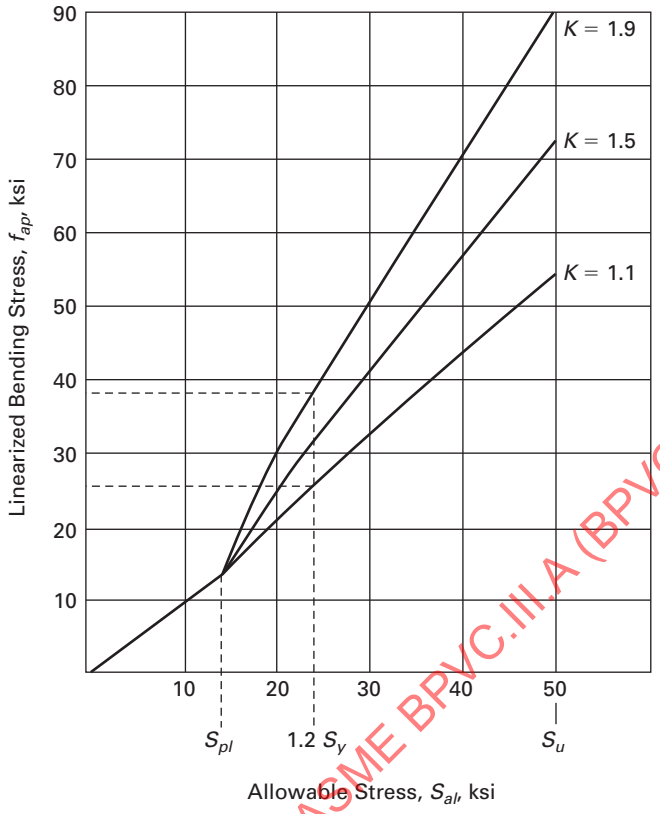
Figure A-9542-1
Linearized Bending Stress Versus Allowable Stress for SA-672 A50 Material at 600°F (316°C)



From F-1323.3, the allowable stress is the lesser of $1.2S_y$ and $0.7S_u$:

$$1.2S_y = 1.2(20) = 24 \text{ ksi}$$

$$0.7S_u = 0.7(50) = 35.0 \text{ ksi}$$

Therefore, S_{al} = 24.0 ksi and

$$S_y = 20 \text{ ksi}$$

$$S_u = 50 \text{ ksi}$$

$$\therefore S_y < S_{al} < S_u$$

From eq. A-9541(c)(14), with K = 1.5,

$$Z = (72 - 25.5)/(50 - 20) = 1.55$$

From eq. A-9541(c)(13),

$$f_{ap} = 25.5 + 1.55(24 - 20)$$

$$= 31.7 \text{ ksi}$$

Alternatively, from Figure A-9542-1, f_{ap} = 31.5 ksi.

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NONMANDATORY APPENDIX B
OWNER'S DESIGN SPECIFICATIONS

ARTICLE B-1000
INTRODUCTION AND SCOPE

B-1100 INTRODUCTION

B-1110 OBJECTIVE

(a) The objective of this Appendix is to provide a guide for the preparation of the Design Specification required by Division 1 of this Section. The writer of the Design Specification is not restricted as to what can be included therein except that, as a minimum, the information required by this Section must be included. Additional, but not less restrictive, requirements which modify the rules of this Section to make them complete for a specific component or to provide more specific or restrictive requirements should be identified.

(b) It is recognized that in order to prepare a document that provides a complete basis for construction of an item for a nuclear facility, a number of considerations outside the scope of this Section may need to be addressed. Some of these which are addressed in this Appendix are

(1) load combinations
(2) operability
(3) regulatory requirements

The additional guidance provided in this Appendix for these considerations is not required by this Section, is not a part of the certification process, and should not be interpreted as extending the duties of the Inspector.

B-1120 FORMAT

B-1120.1 General. Design Specifications should be as uniform throughout the nuclear industry as is reasonably attainable. The format of this Appendix is presented as a guide to uniformity and is divided into major categories as follows:

(a) Generic Requirements applicable to all components (Article B-2000)

(b) Specific Requirements applicable to each component (Article B-3000 through Article B-10000, inclusive)

(c) Included in both the Generic and Specific Requirements are those considerations outside the scope of this Section (operability[23] and regulatory[24] requirements) which have an effect on construction but which are not required by this Section to be a part of the certified Design Specification.

B-1120.2 Nomenclature, Definitions, and Symbols. Nomenclature, definitions, and symbols should be in agreement with those established in the applicable Article. Should a conflict exist between Articles, the Design Specification should be clear as to what is intended in each case.

B-1200 SCOPE OF CERTIFIED DESIGN SPECIFICATION

The certified Design Specification should contain in sufficient detail the information which this Section requires to be provided. Operability[23] and regulatory[24] requirements which are beyond the jurisdiction of this Section are not covered by the Code required certification of the Design Specification (NCA-3252).

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ARTICLE B-2000
GENERIC REQUIREMENTS

B-2100 CERTIFIED DESIGN SPECIFICATION REQUIREMENTS

The information in this Article addresses those portions of the certified Design Specification which are generic in nature and therefore applicable to the construction of all Section III items.

B-2110 GENERAL

B-2110.1 Contents of the Certified Design Specification.

(a) NCA-3252 provides the minimum requirements for the contents of the certified Design Specification.

(b) With respect to NCA-3252(a), it is important to recognize that the boundary defines an interface between two items that are dependent on each other for the transmittal of loads. In order to properly design the item on either side of the boundary, the effect of the attached item is required. The effect may be furnished directly by supplying the forces and moments that are transmitted across the boundary or, alternatively, by providing sufficient information to enable the designer to determine the interaction across the boundary. This Section provides rules to accomplish this in NCA-3254.

(c) Any Code Cases applicable to the construction of an item should be included in the Design Specification.

B-2110.2 Certification. NCA-3255 provides the requirements for certification of the Design Specification. The required certification is not applicable to supplementary, regulatory, or operability requirements which are outside of the scope of this Section.

B-2110.3 Permanent Records. NCA-4134.17 provides the requirements for the continued maintenance and retention location for permanent records.

B-2110.4 Handling, Storage, and Shipping. The Design Specification should include any special measures to control handling, storage, and shipping of the component (NCA-4134.13).

B-2110.5 Identification of Regulatory and Enforcement Authorities. The Design Specification should include identification of regulatory and enforcement authorities at locations of component installation with whom Data Reports must be filed.

B-2110.6 Filing. NCA-3256 provides the requirements for filing of the Design Specification.

B-2110.7 Review of Design Report. NCA-3260 provides the requirements for Owner's review of the Design Report.

B-2111 Classification

B-2111.1 Responsibility. NCA-2110(d) provides the requirements for classification of equipment.

B-2111.2 Multiple Code Class Components. NCA-2133 provides the requirements for multiple Code Class components.

B-2111.3 Optional Use of Code Classes. NCA-2134 provides the requirements for optional use of Code Classes.

B-2111.4 Special Requirements. NCA-2160 provides the requirements for contractual arrangements that are beyond the scope of this Section.

B-2112 Design Basis and Service Limits

B-2112.1 Plant and System Service Conditions. The definition of plant and system service conditions, and the determination of their significance to the design and operability of components and supports of a nuclear facility, may be derived from systems safety criteria documents for specific types of nuclear facilities and may be found in the requirements of regulatory and enforcement authorities having jurisdiction at the site [NCA-2141(b)].

B-2112.2 Design Loadings. The Design Specification shall include the Design Pressure [NCA-2142.1(a)], the Design Temperature [NCA-2142.1(b)], and the Design Mechanical Loads [NCA-2142.1(c)].

B-2112.3 Establishment of Component and Support Design and Service Limits.

(a) For Class 1, MC, and CS components, and for Class 2 and 3 piping and its supports, Design and Service Loads should be specified and appropriate Service Limits designated [NCA-2142(a)].

(b) For Class 2 and 3 components and supports, other than piping and its supports, two options are available as follows:

(1) Design and Service Loads may be specified and appropriate Service Limits designated.

(2) Service Loadings are not required to be identified when the Design Pressure, Design Temperature, and Design Mechanical Loads result in stresses that are at least as high, relative to allowable values, as any which may occur for any Service Loading [NCA-2142(a)].

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B-2112.4 Test Loadings. NCA-2142.3 provides the rules for consideration of Test Loadings.

B-2113 N Certificate Holder's Responsibilities

B-2113.1 Manufacturers of Small Pumps and Valves and of Standard Supports. Manufacturers of small pumps and valves [NPS 4 (DN 100) and smaller] and standard supports (including snubbers) who elect to provide their own Design Specification are responsible for compliance with the requirements of NCA-3252.

B-2113.2 Compliance With N Certificate Holder's Responsibilities. When the completed Code item involves work by more than one organization, the Design Specification shall be provided to the organization having overall responsibility.

B-2120 DESIGN

B-2121 Loadings

The Owner or Owner's designee shall identify the loadings and designate the appropriate Design and Service Limits for each component or support. The loadings that should be taken into account in designing a component include, but are not limited to, the following:

(a) internal and external pressure, including static head

(b) weight of the component and normal contents under service and test conditions

(c) superimposed loads, such as other components, operating equipment, insulation, or corrosion resistant or erosion resistant linings and piping

(d) vibrations and earthquake loads

(e) reactions of supporting lugs, rings, saddles, or other types of supports

(f) temperature effects

(g) restrained thermal expansion

(h) anchor and support movement effects

(i) environmental loads, such as wind and snow

(j) dynamic effects of fluid flow

B-2122 Design Loads

B-2122.1 Design Pressure. NCA-2142.1(a) and NB/NC/ND/NE-3112.1 provide the required definitions for Design Pressure.

B-2122.2 Design Temperature. NB/NC/ND/NE/NF/NG-3112 and NCA-2142.1(b) provide the requirements for Design Temperature. The Design Temperature shall be used in computations involving the Design Pressure and coincidental Design Mechanical Loads. The actual metal temperature at the point under consideration shall be used in all computations where the use of the actual service pressure is required. Where a component is heated by tracing, induction coils, jacketing, or by internal heat generation, the effect of such heating shall be incorporated in the establishment of the Design Temperature.

B-2122.3 Design Mechanical Loads.

(a) The specified Design Mechanical Loads should be selected so that when combined with the effects of Design Pressure, they represent the most severe coincident loadings for which the Level A Service Limits on primary stress are applicable.

(b) The determination of *most severe coincident loadings* may result in specification of pairs of Design Conditions since the one most severe combination may not be readily predicted. The specification may specify the maximum Design Mechanical Load for any situation which, when taken with the Design Pressure, would result in the worst combination of Design Conditions even though they may not be coincident.

(c) The Design Mechanical Loads that are considered are somewhat dependent on the component, its location, its attachment to other components, and for a Class 2 or 3 component, whether Service Loadings are to be specified (refer to B-2112.3 and NCA-2142).

B-2123 Service Loads

In order to properly specify Service Limits for the various types of loadings, the Owner or Owner's designee should recognize the basis for the establishment of those Limits. These are given in NCA-2142.2.

B-2123.1 Service Limits A and B.

(a) For Class 1, MC, and CS components and for Class 2 vessels designed to NC-3200, Service Limits A and B are provided in order to evaluate the effect of system operating loads on the fatigue life of the component. For a fatigue analysis the loads applicable to the component being considered should be described in terms of quantities that the designer may use [NB-3222.4(e)]. The variation with respect to time of pressure, temperature, flow rate, etc., as well as the number of times these changes occur in the life of the component, is needed. In this regard, a service cycle is defined in NB-3213.15 as: "... the initiation and establishment of new conditions followed by a return to the conditions which prevailed at the beginning of the cycle." Thus, as an example, the conditions associated with plant startup do not constitute a service cycle. Startup and shutdown together constitute a service cycle, and if there are n_1 startups in the Design Specification, there should be the same number of shutdowns.

(b) Figure B-2123-1 is an illustration of the time-dependent load information which the designer needs. (Note that it provides only the startup portion of a service cycle.)

(c) Refer to B-6124 for the Class 2 and 3 piping requirements.

(d) For all other Class 2 and 3 components and supports, including piping supports, it is not necessary to define each service cycle in detail since no fatigue analysis is required. It is important for the designer to know the maximum loading condition on the component for these Service Limits.

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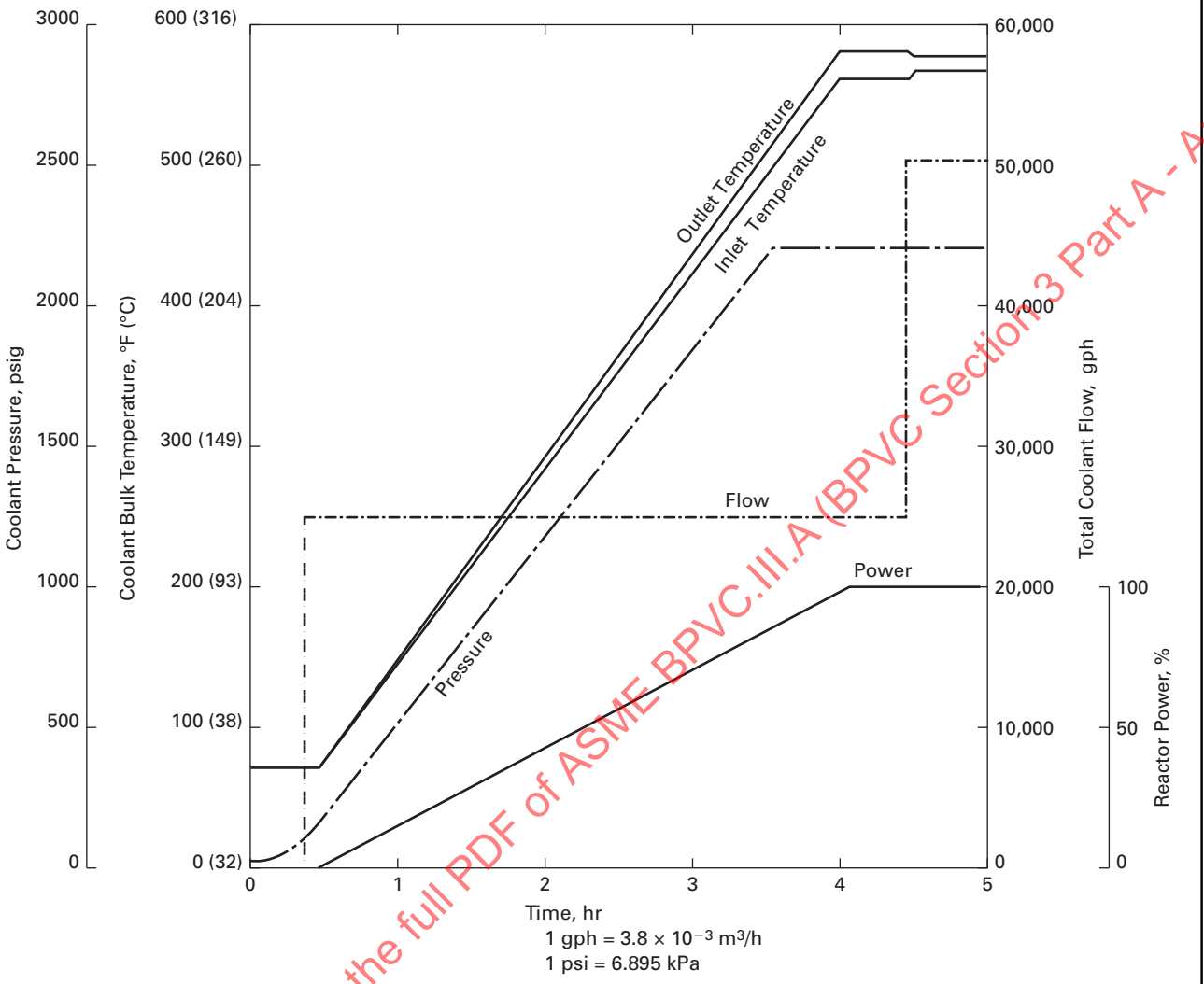
Figure B-2123-1
Time-Dependent Load Information



(15) **B-2123.2 Service Limit C.** Service Limit C is provided in order to evaluate the effect of plant operating loads on the structural integrity of a component for situations which are not anticipated to occur for a sufficient number of times to affect fatigue life and for which large deformations in areas of structural discontinuities are not objectionable. Since the occurrence of stress associated with this Limit may result in removal of the component from service for inspection or repair, the Owner should review the selection of this Limit for compatibility with established system safety criteria. Refer to NB-3113(b) for the limit of number of cycles.

B-2123.3 Service Limit D. Service Limit D is provided in order to evaluate the effect of plant operating loads on the structural integrity of a component for situations in which gross general deformations, loss of dimensional stability, and damage requiring repair, excluding loss of pressure-retaining function, are not objectionable. Since the occurrence of stress associated with this Limit may require removal of the component from service, the Owner should review the selection of this Limit for compatibility with established system safety criteria.

B-2124 Test Loads

Loads due to tests beyond those allowed by this Section should be classified in the appropriate Service Limit in accordance with NCA-2142.3(b) [NCA-2142.4(d)(2)].

B-2125 Load Combinations

In order to provide a complete definition of service loads, the combination of specific events must be considered. Since these combinations are a function of specific systems which make up a part of a specific type nuclear facility, this Section does not directly address this other

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than to provide different Stress Limits for various loadings. Specific guidance is provided in the approved Safety Analysis Report (SAR) for the plant.

B-2126 Deformation Limits

The Code does not provide specific deformation limits other than those that would be associated with a given allowable stress. If control of deformation is a requirement, the deformation limits should be provided.

B-2130 MATERIALS

B-2131 General Requirements

The Design Specification should provide information relative to materials as listed in (a) through (i).

(a) any hydrostatic testing or service temperature limits

(b) any reductions to design stress intensity values, allowable stress, or fatigue curves necessitated by environmental conditions

(c) any restrictions on cladding materials

(d) materials which are acceptable from the standpoints of environment and location

(e) any restrictions on heat treating

(f) any requirements with respect to cleanliness

(g) impact test requirements (B-2132)

(h) any corrosion or erosion allowances

(i) postweld heat treatment times applied to the material or item after it is completed must be specified (NB/NC/ND/NE/NF/NG-4622)

B-2132 Impact Tests

For those cases where impact testing is optional, the Design Specification should state whether or not impact testing of the pressure-retaining material of the component or the support material is required. The test temperature should be specified and the tests become part of the appropriate Subsection.

B-2133 Fracture Mechanics Data

When the methods of Nonmandatory Appendix G are to be used to provide protection against nonductile fracture for materials that have specified minimum yield strengths at room temperature greater than 50 ksi (345 MPa) but not exceeding 90 ksi (620 MPa), the Design Specification shall include additional fracture mechanics data for base metal, weld metal, and heat-affected zone that are required to use Figure G-2210-1 in accordance with G-2110(b). Where these materials of higher yield strengths are to be used in conditions where radiation may affect the material properties, the effect of radiation on the K_{Ic} curve shall be determined for the material prior to its use in construction.

B-2140 FABRICATION

The Design Specification should specify any unusual restrictions on fabrication processes or techniques that would be deleterious to the suitability of the component in the expected service environment.

B-2150 TESTING (15)

B-2151 Pneumatic Test

The Design Specification should identify if a pneumatic test should be used in lieu of hydrostatic testing for those components and appurtenances required to be pressure tested in accordance with the rules of this Section (NB/NC/ND/NE-6111, NB/NC/ND/NE-6112).

B-2152 Restriction on Testing

Any restrictions on the use of the test fluid should be provided (NB/NC/ND/NE-6112). When selecting a fluid for the test, it should be determined that the test fluid does not have deleterious effects and that the test fluid may be safely used at the pressure and temperature specified for the test.

B-2153 Bellows Type Expansion Joints

Any requirements that supplement hydrostatic or pneumatic testing of bellows type expansion joints should be included.

B-2154 Leak Tightness

Leak tightness requirements for areas, such as permanent seals, seats, and gasketed joints for pressure-retaining components or appurtenances, should be included (NB/NC/ND/NE-6224).

B-2155 Additional Testing

If testing in addition to pressure testing is required, the loads due to such testing should be classified in accordance with NCA-2142.3(b).

B-2160 OVERPRESSURE PROTECTION (15)

B-2161 General Requirements

B-2161.1 Scope. For steady state or transient conditions of pressure and coincident temperature that are in excess of design or service loadings and their combinations and associated limits specified in the Design Specifications, system overpressure protection is required for vessels, piping, pumps, and valves in service and subjected to the consequences of the application of these conditions (refer to NB/NC/ND/NE-7110).

B-2161.2 Integrated Overpressure Protection. It should be recognized that the overpressure protection of pressure-retaining components in a system require consideration of the pressure transients which may be imposed on the systems during all service loadings and testing conditions described in the component Design Specifications (refer to NB/NC/ND/NE-7120).

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B-2162 Design Secondary Pressure

The design secondary pressure shall be specified in the Design Specification [refer to NB-7111(d) and NC/ND-7112].

B-2163 Maximum Anticipated Pressure and Temperature

The Design Specification should identify the maximum anticipated pressure and coincident temperature among any systems components under the operating conditions of the system as a consequence of any transients occurring either within the system or in associated systems which may affect the system for which overpressure protection is intended (refer to NB/NC/ND-7300). Service conditions such as at startup and shutdown may require protection against nonductile failure [NB-3211(d)] at pressures lower than the component design pressure.

B-2164 Pressure Relief Valve Operating Requirements

B-2164.1 Blowdown Requirements. The Design Specification may specify blowdown requirements with a greater tolerance than the values stated in NB/NC/ND-7500.

B-2164.2 Popping Point Tolerance. The Design Specification may specify a popping point tolerance greater than the value stated in NB/NC/ND-7500.

B-2165 Pressure Relief Valve Operating Characteristics (Refer to ANSI N278.1)

As applicable, the following pressure relief valve operating characteristics should be specified in the Design Specification when overpressure protection is dependent upon these factors:

(a) set pressure
(b) set pressure range
(c) set pressure tolerance
(d) discharge capacity with due allowance for the effect of the back pressure on the capacity
(e) accumulation
(f) blowdown
(g) static and dynamic back pressure, minimum and maximum
(h) response time (maximum time delay between attainment of set pressure or reception of the energizing signal by the solenoid and valve lift)

B-2166 Rupture Disk Devices

Rupture disk device burst pressure tolerance and manufacturing design range should be specified in the Design Specification.

B-2200 OPERABILITY

B-2210 INTRODUCTION

Operability requirements are outside the scope of this Section [NCA-2142(b)]; however, the Owner or Owner's designee is required to identify any such requirements in the Design Specifications (NCA-3252).

B-2220 ACTIVE PUMPS OR VALVES

The Design Specification should indicate if the specified pump or valve must perform a mechanical motion during the course of accomplishing a system safety function during or following the specified plant event. Such a pump or valve is designated as an active component.

B-2300 REGULATORY REQUIREMENTS

In the process of preparing a Design Specification, it is important to refer to and rely on the requirements contained in SAR documents since they provide the basis for complying with existing regulatory requirements. Conflicts between a Design Specification and the SAR could lead to construction of items not in compliance with the license requirements. A reference list of regulatory documents is available at http://www.nrc.gov/.

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ARTICLE B-3000
SPECIFIC VESSEL REQUIREMENTS

B-3100 CERTIFIED DESIGN SPECIFICATION REQUIREMENTS

The Design Specification for vessels should include requirements indicated in Article B-2000, Generic Requirements.

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ARTICLE B-4000
SPECIFIC PUMP REQUIREMENTS

B-4100 CERTIFIED DESIGN SPECIFICATION REQUIREMENTS

In addition to the Design Specification requirements indicated in Article B-2000, Generic Requirements, the Design Specification for pumps should include the other requirements of B-4110 and B-4120.

B-4110 GENERAL REQUIREMENTS

Covered by B-2110.

B-4120 DESIGN

B-4121 Loads From Connected Piping

The forces and moments produced by the connected piping on each pump inlet and outlet should be included (NB/NC/ND-3415).

B-4122 Earthquake Loadings

NB/NC/ND-3417 provide the requirements for consideration of earthquake loading.

B-4200 OPERABILITY REQUIREMENTS FOR PUMPS

B-4210 GENERAL REQUIREMENTS

B-4211 Applicability

The inclusion of functional operability requirements in the Design Specifications should be based on the functional requirements of the pump being specified. These requirements should be specified only if the pumps are considered to be active pumps.

B-4220 DESIGN

The Design Specification should include all applicable and pertinent information considered important to the functional operability of the pump.

B-4230 QUALIFICATION

B-4231 Methods

The method of pump qualification, if any, for functional operability should be defined in the Design Specification. Qualification by analysis, test, or combinations thereof should be specified. Available codes or standards which cover these areas should be referred to and used to the maximum extent possible.

B-4232 Analysis

Acceptable methods of analysis should be identified. The following areas, as a minimum, should be addressed:

(a) required analysis;

(b) load combinations, including deadweight, thermal loads, nozzle loads, seismic loads, etc.;

(c) allowable stresses for the various loading conditions.

B-4233 Testing

Acceptable methods of testing should be identified. The following areas, as a minimum, should be addressed:

(a) required tests and test sequences

(b) imposed loads and pump function during tests

(c) acceptance criteria

B-4240 FUNCTIONAL OPERABILITY PRODUCTION TESTS

Any special functional operability tests to be conducted on production pumps should be specified in the Design Specification.

B-4250 DOCUMENTATION

Documentation requirements for functional qualification or production tests should be specified.

B-4300 REGULATORY REQUIREMENTS

Regulatory requirements are covered in B-2300.

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ARTICLE B-5000
SPECIFIC VALVE REQUIREMENTS

B-5100 CERTIFIED DESIGN SPECIFICATION REQUIREMENTS

In addition to the Design Specification requirements indicated in Article B-2000, Generic Requirements, the Design Specification for valves should include the other requirements of B-5110 and B-5120.

B-5110 GENERAL REQUIREMENTS

Covered by B-2110.

B-5120 DESIGN

B-5121 Class 1 Valves

B-5121.1 Pipe Reactions for Valves Designed to Alternative Design Rules. NB-3512.2 provides the requirements concerning pipe reactions.

B-5121.2 Earthquake Loadings. NB-3524 provides the requirements concerning earthquake loadings.

B-5121.3 Level C Service Limits. NB-3526 provides the requirements concerning valve function during loading for which Level C Service Limits are specified.

B-5121.4 Pipe Reaction Stress. NB-3526.2 provides the requirements concerning pipe reaction stress computation for Level C Service Limits.

B-5121.5 Level D Service Limits. NB-3527 provides the requirements concerning valve function during loadings for which Level D Service Limits are specified.

B-5121.6 Hydrostatic Tests. NB-3531.2(c) provides the requirements concerning alternative test pressures, seat leakages, and test durations.

B-5121.7 Body Contours at Weld Ends. NB-3544.8 provides the requirements concerning alternative body contours at weld ends of valves.

B-5121.8 Bypass Piping. The Design Specification shall state which organization is responsible for the bypass piping design, if the responsible organization is not the piping system designer [NB-3546.3(b)].

B-5122 Class 2 Valves

B-5122.1 Alternative Rules. The Design Specification shall specify whether the alternative rules of NC-3513 are permitted to be used.

B-5122.2 Hydrostatic Tests. NC-3514 provides the requirements concerning alternative test pressures, seat leakages, and test durations.

B-5123 Class 3 Valves

B-5123.1 Alternative Rules. The Design Specification shall specify whether the alternative rules of ND-3513 are permitted to be used.

B-5123.2 Hydrostatic Tests. ND-3514 provides the requirements concerning alternative test pressures, seat leakages, and test durations.

B-5200 OPERABILITY REQUIREMENTS FOR VALVES

B-5210 INTRODUCTION

Operability requirements are outside the scope of this Section (NCA-2142); however, the Owner or Owner's designee is required to identify any valve operability requirements in the Design Specification [NB-3526(b) and NB-3527].

B-5220 DESIGN

The Design Specification should include all applicable and pertinent information required. A document pertaining to this information is ANSI N278.1. Additional information not covered in ANSI N278.1, but considered important to the functional operability of the valve should also be included. NB-3524, NC-3520, and ND-3520 provide guidance for analysis of valves with extended masses.

B-5230 QUALIFICATION

B-5231 Methods

The method of valve qualification, if any, for functional operability should be defined in the Design Specification. Qualification by analysis, test, or combinations thereof should be specified. Available Codes or Standards which cover these areas should be referenced and used to the maximum extent possible.

B-5232 Analysis

Acceptable methods of analysis should be identified. The following areas, as a minimum, should be addressed:

(a) required analysis

(b) load combinations, including seismic, end loads, mechanical loads, etc.

(c) allowable stresses for the various loading conditions

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B-5233 Testing

Acceptable methods of testing shall be identified. The following areas, as a minimum, should be addressed:

(a) required tests and test sequence

(b) imposed loads and valve function during tests

(c) acceptance criteria Y

B-5240 FUNCTIONAL OPERABILITY PRODUCTION TESTS

Any special functional operability tests to be conducted on production valves shall be specified in the Design Specification.

B-5250 DOCUMENTATION

Documentation requirements for functional qualification or production tests or both should be specified in the Design Specification.

B-5300 REGULATORY REQUIREMENTS

Regulatory requirements are covered in B-2300.

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ARTICLE B-6000
SPECIFIC PIPING REQUIREMENTS

B-6100 CERTIFIED DESIGN SPECIFICATION REQUIREMENTS

In addition to the Design Specification requirements indicated in Article B-2000, Generic Requirements, the Design Specification for piping should include the other requirements of B-6110 and B-6120.

B-6110 GENERAL REQUIREMENTS

Covered by B-2110.

B-6120 DESIGN

B-6121 Seismic

For piping, the loadings, movements, anchor motions, and number of cycles due to seismic events should be given. The associated Service Loadings which occur with, or as a result of, the specified seismic events should be stated.

B-6122 Other Dynamic Loads

Dynamic loadings, such as those resulting from sudden valve or pump operation, should be given. As a minimum, the information needed to determine this loading should be given (such as pressures, temperatures, flow rates, valve operating times).

B-6123 Peak Pressure

In categorizing Service Loadings into appropriate Service Limits, the Design Specification should include the peak pressure.

B-6124 Fatigue Consideration for Class 2 and 3 Piping

For Class 2 and 3 piping, it is not necessary to define each service cycle in detail. However, the maximum range of conditions and the total number of occurrences of all service cycles to which the piping system will be subjected shall be identified [NC/ND-3611.2(e)]. For example, the minimum temperature conditions could be 40°F (5°C) while the maximum is 456°F (235°C). If all other service cycles did not impose a temperature condition less than the minimum or greater than the maximum, it is not required to be specified, unless the total number of occurrences of all service cycles exceeds 7,000. In this case, the range of temperature and the number of occurrences for each service cycle shall be specified. In determining the total number of service cycles, all service cycles shall be considered including those that impose a temperature condition less than the maximum range of temperature.

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ARTICLE B-7000
SPECIFIC CONTAINMENT REQUIREMENTS

B-7100 CERTIFIED DESIGN SPECIFICATION REQUIREMENTS

In addition to the Design Specification requirements indicated in Article B-2000, Generic Requirements, the Design Specification for containment should include the other requirements of B-7110 and B-7120.

B-7110 GENERAL REQUIREMENTS

Covered by B-2110.

B-7120 DESIGN

B-7121 Design Pressure

NE-3112.1 provides the requirements concerning specification of Design Pressure.

B-7122 Design Temperature

NE-3112.2 provides the requirements concerning specification of Design Temperature.

B-7123 Design Mechanical Loads

NE-3112.3 provides the requirements concerning specification of Design Mechanical Loads.

B-7124 Service Conditions

NE-3113 provides the requirements concerning specification of Level A, B, C, and D Service Conditions which satisfy the Generic Requirements for Service Limits in NCA-2142.

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ARTICLE B-8000
SPECIFIC SUPPORT REQUIREMENTS

B-8100 CERTIFIED DESIGN SPECIFICATION REQUIREMENTS

In addition to the Design Specification requirements indicated in Article B-2000, Generic Requirements, the Design Specification for supports should include the other requirements of B-8110, B-8120, and B-8130.

B-8110 GENERAL REQUIREMENTS

In addition to the general requirements of B-2110, the information required by NF-1110 for intervening elements in the support load path should be included in the Design Specification for the supported component.

B-8120 DESIGN

B-8121 Standard Supports

NF-3400 provides the requirements for standard supports.

B-8130 MATERIALS

Requirements for items, such as gaskets, seals, springs, compression spring endplates, bearings, retaining rings, washers, wear shoes, hydraulic fluids, etc., should be stated in the Design Specification. Such items should be made of materials that are not injuriously affected by the fluid, temperature, or irradiation conditions to which the item will be subjected (NF-2121).

B-8300 REGULATORY REQUIREMENTS

Regulatory requirements are covered in B-2300.

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ARTICLE B-9000
SPECIFIC CORE SUPPORT STRUCTURES REQUIREMENTS

B-9100 CERTIFIED DESIGN SPECIFICATION REQUIREMENTS

In addition to the Design Specification requirements indicated in Article B-2000, Generic Requirements, the Design Specification for core support structures should include the other requirements of B-9110, B-9120, B-9130, and B-9140.

B-9110 GENERAL REQUIREMENTS

(a) All information and requirements contained in the specifications which are beyond the jurisdiction of Subsection NG should be so identified.

(b) The Design Specifications should stipulate any specific additional core support structure requirements which the Owner intends to be incorporated in the specific structures covered by the Design Specifications or any additional requirements intended to be more specific or more restrictive than the minimum requirements of this Section.

(c) Where additional terms, definitions, or expressions are required, they should be clearly defined and explained and adequately referenced.

(d) Identification of the core support structure is required as determined by their function and operating requirements (Structures whose purpose is only to limit the motion of the core following the postulated occurrence of a failure in the structure normally supporting the core are not considered core support structures. They are normally designed to meet deformation limits for this postulated condition and are not intended to meet the stress limits of NG-3200.) (NG-1121).

(e) Delineation of those internal structures which are required to be analyzed in order to ensure the structural integrity of the mating core support structures (NG-1122).

(f) The boundaries of the core support structures and their relationship to the support and restraint of the core shall be clearly defined through the use of dimensions, descriptions, or drawings (NG-1120).

B-9120 DESIGN

B-9121 Loading Conditions

The following should be specified:

(a) pressure differences due to coolant flow

(b) weight of the core support structure

(c) superimposed loads, such as those due to other structures, the reactor core, steam separating equipment, flow distributors and baffles, thermal shields, and safety equipment

(d) earthquake loads or other loads which result from motion of the reactor vessel

(e) reactions from the support or restraints, or both

(f) loads due to temperature effects, such as thermal gradients and differential expansion

(g) loads resulting from the impingement or flow of reactor coolant or other contained or surrounding fluids

(h) transient pressure difference loads, such as those which would result from rupture of the main coolant pipe

(i) vibratory loads

(j) reaction loads from control rods

(k) handling loads experienced in preparation for or during refueling or inservice inspection

B-9122 Loading Combinations

The loadings to be simultaneously considered and the applicable Service Limits should be specified (NG-3112).

B-9123 Deformation Limits

In addition to Service Limits given in Subsection NG, static and dynamic deformation limits should be specified to ensure the performance of all safety related functions of the core support structure. These limits are to be those that cause loss of function and are not intended to include a margin of safety (NG-3220).

B-9124 Reinforcement for Openings

The Design Specification should stipulate if the rules for reinforcing applicable to Class 1 vessels may be used for core support structures (NB-3132).

B-9130 MATERIALS

The Design Specifications should provide any special requirements for materials and testing specifically applicable to the core support structures. The Owner is responsible for selecting materials suitable for the conditions stated at the Design Specification with specific attention being given to the effects of service conditions upon the properties of the materials (NB-2160). The requirements for impact testing shall be specified [NG-2311(a)].

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B-9140 FABRICATION

The extent or removal of additional material by mechanical means when P-No. 8 material is prepared by thermal cutting methods should be specified (NG-4211.7).

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ARTICLE B-10000
SPECIFIC PARTS AND MISCELLANEOUS ITEMS REQUIREMENTS

B-10100 CERTIFIED DESIGN SPECIFICATION REQUIREMENTS

In addition to the Design Specification requirements indicated in Article B-2000, Generic Requirements, the Design Specification for parts and miscellaneous items should include the other requirements of B-10110.

B-10110 GENERAL REQUIREMENTS

The following general requirements should be considered when preparing a Design Specification for parts and miscellaneous items.

B-10111 Parts

The Design Specification for components or supports should apply to the parts of such components or supports (NCA-1231).

B-10112 Appurtenances

NCA-1260 provides the requirements for appurtenances.

B-10113 Control Rod Drive Housings

NCA-1271 provides the requirements for control rod drive housings.

B-10114 Heater Elements

NCA-1272 provides the requirements for heater elements.

B-10115 Fluid Conditioner Devices

NCA-1273 provides the requirements for fluid conditioner devices.

B-10116 Rupture Disc Devices

NCA-1275 provides the requirements for rupture disk devices.

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NONMANDATORY APPENDIX C

ARTICLE C-1000
CERTIFICATE HOLDER'S DESIGN REPORT

C-1100 INTRODUCTION

C-1110 OBJECTIVE

The objective of this Appendix is to provide a guide for use by Certificate Holders in the preparation of Design Reports required by NCA-3551.1. Desirably, such Design Reports should be uniform as to format for all of the nuclear industry. Such uniformity is helpful in making for easier review by the Owner (NCA-3260), Inspectors, regulatory agencies, or independent groups. For NF supports designed by load rating (NF-3280), the preparation of a load capacity data sheet in accordance with NCA-3551.2 fulfills the requirements for preparation of a Design Report. The contents of this Appendix constitute only suggestions and are nonmandatory.

C-1120 BASIS

In order to meet the requirements of NCA-3551, the Design Report should be based upon analysis or testing adequate to demonstrate the validity of the structural design to sustain and meet in every respect the requirements and provisions of the relevant certified Design Specifications and the requirements of this Section; the Report should include, as a minimum, the results, conclusions, and other considerations which show that the structural design meets these requirements.

C-1130 FORMAT

C-1131 General Requirements

Since a major purpose of the Design Report is to facilitate an independent review of its content, it is important that it be simple to follow and free from ambiguity. Nomenclature, definitions, and symbols used should be in agreement with those established in Subsection NB for Class 1 components, in Subsection NE for Class MC vessels, in Subsection NF for supports, in Subsection NG for Class CS core support structures, and in Subsection NC for vessels designed in accordance with NC-3200. Where additional terms, definitions, or expressions are required, they should be clearly defined and explained and adequately referenced. It is not the intention to limit the choice of analytical methods or computational techniques used for obtaining the values and results required for the Design Report.

C-1132 Presentation of Analysis

The analysis in the Design Report should be in three sections: Thermal Analysis, Structural Analysis, and Fatigue Evaluation. The desiderata listed in (a) through (j) should be adhered to.

(a) Pages and figures in each section of the Report should be consecutively numbered

(b) Reference data taken from other parts of the calculations should have the proper page number and section of the Report listed

(c) A general description of the method of analysis should be given

(d) All reference sources should be listed

(e) All computer programs should be properly identified and described

(f) Stresses should be tabulated for each area of investigation

(g) Areas which have the most severe stress condition for design conditions or for any specified transient should be listed in the Report, along with the stress values in these areas

(h) Results should be summarized and a general summary of all stresses should be made in each section of the Report

(i) Drawings and sketches necessary for an understanding of the analysis should be part of the Report

(j) The Report should include copies of sufficient computer printouts to justify the governing stress values used in the Design Report and enable independent review. Copies of any manual calculations prepared which establish the final design should also be included.

C-1140 BASIC INFORMATION

It should be noted that the references in this Appendix to basic information which is to be obtained from the certified Design Specifications (NCA-3250) are predicated on the requirement of NCA-3252 that such information be provided.

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C-1150 DISTRIBUTION OF COPIES OF DESIGN REPORT

Copies of the completed certified Design Report (NCA-3355) should be made available for the Owner's review, certification, and distribution as required by NCA-3556, NCA-3260, and NCA-3270. The Certificate Holder shall also make a copy available to the Inspector (NCA-3557).

C-1200 THERMAL ANALYSIS

C-1210 DESCRIPTIONS OF OPERATING CYCLES

Data for the various transients and operating cycles should be obtained from the certified Design Specifications. Typical cycles which should be considered are referenced in Nonmandatory Appendix B.

C-1220 STEADY STATE CONDITION

The steady state condition to provide a thermal equilibrium condition for normal operating transients should be obtained from the certified Design Specifications.

C-1230 GEOMETRY FROM TEMPERATURE DISTRIBUTION POINT OF VIEW

The geometrical structure of the component should be divided into suitable areas for thermal analysis. Sketches of the thermal model should be included in the Report. The areas listed in (a) through (m) are typical of those that should be investigated.

(a) nozzle junctions in the component wall
(b) stud bolts
(c) cylinder junction with cylinder flange
(d) point of support attachment
(e) cylinder junction with head
(f) junction of component wall and internal baffles, tubesheets, and attachments
(g) tube-to-tubesheet junction for heat exchangers
(h) heater penetrations to component junction for pressurizer
(i) junction area between component supports and building structure
(j) external attachments
(k) changes in thickness within a component (such as a reducer, brand connection, component support, etc.) or across a welded joint (such as a socket weld, butt weld of different thickness, etc.)
(l) instrument penetrations to component junction (such as thermowells, flow devices, etc.)
(m) core barrel and core support plate

C-1240 GEOMETRIES FROM STRESS INVESTIGATION POINT OF VIEW

The final breakdown of the geometries, which will correspond to the method of stress calculation, should be indicated in the calculations. Typical areas listed in C-1230 which are applicable to the component under consideration should be included.

C-1250 TEMPERATURE-DEPENDENT DATA FOR EACH THERMAL GEOMETRY AND TRANSIENT

The Certificate Holder should specify values for all parameters, such as coefficients for water and air, which are required for thermal calculations. References to sources should be given for all such data used.

C-1260 TEMPERATURE DISTRIBUTION FOR EACH GEOMETRY

The temperature profile of the applicable areas listed under C-1230 should be calculated and the temperature values attached to the calculations. The temperature distributions should be based on two or three dimensional heat transfer calculations. For calculating through wall (radial) temperature distributions to obtain values of ΔT_1 and ΔT_2 (NB-3650), one-dimensional heat transfer calculations are acceptable.

C-1270 THERMAL GRADIENTS

Individual transients should be investigated separately for each area. The longitudinal, radial, and circumferential gradients should be plotted separately. The temperature gradients used in the stress calculations should be plotted.

C-1300 STRUCTURAL ANALYSIS

C-1310 IMPORTANT THERMAL AND MECHANICAL LOADING ON COMPONENT STRUCTURE

C-1311 Mechanical Loading

The mechanical loads used in the Design Report to calculate primary stresses should be obtained directly from the Design Specification (such as Design Pressure and Temperature) or from information contained in the Design Specification (such as seismic spectra, valve opening and/or closing times, etc.).

C-1312 Thermal and Mechanical Loading

Specific reference should be made to the thermal transients and resulting gradients which are to be used for the Design Report. The internal pressure and external loads to be used should be in accordance with the time and thermal condition analyzed. Values for external nozzle loads should include sign convention of the loadings and be referenced specifically to the geometries.

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C-1320 METHODS OF CALCULATIONS

The Certificate Holder should submit a short description of the calculation methods used in connection with the stress analysis. All computer programs used in making calculations should be verified by comparing the program with the results of an appropriate analytical or experimental solution. The basic theories on which the calculations are based and the assumptions should also be included.

C-1330 PRINCIPAL STRESSES FOR EACH GEOMETRY

In calculating stress components, the requirements of NB-3200, NC-3200, NE-3200, NF-3220, NF-3230, or NG-3200 should be followed. The following are typical of stress components that should be considered:

(a) Mechanical stresses generated by
- *(1)* pressure load
- *(2)* deadweight load
- *(3)* piping load
- *(4)* externally applied load
- *(5)* seismic loads
- *(6)* dynamic loads

(b) Thermal stresses generated by
- *(1)* radial gradient
  - – thermal stress
  - – thermal discontinuity stress
- *(2)* longitudinal gradient
  - – thermal stress
  - – thermal discontinuity stress

C-1340 ALLOWABLE LIMITS

Each individual stress component and combination of the stress components should satisfy the requirements of NB-3220, NC-3220, NE-3220, NF-3220, NF-3230, or NG-3220.

C-1400 FATIGUE EVALUATION

C-1410 SCOPE OF FATIGUE EVALUATION

Fatigue evaluation when required should include the considerations and investigations described in this Subarticle.

C-1420 LOCATIONS OF STRESS CONCENTRATIONS

Stress concentration should be investigated at any geometrical changes in the structure, such as difference in wall thickness, joints and corners, and junctions of dissimilar metals. A list of the locations subject to fatigue evaluation should be included in the Report. For piping, K indices are given for standard piping components in NB-3600 which represent elastic stress concentration factors.

C-1430 FATIGUE STRENGTH REDUCTION FACTORS AS FUNCTION OF LOCATIONS AND TYPES OF STRESS

Fatigue strength reduction factors should be numerically listed for the stresses where they are to be applied. The references and methods of finding the fatigue strength reduction factors should be included in the Report.

C-1440 PROPER STRESS CONCENTRATION OR FATIGUE STRENGTH REDUCTION FACTOR APPLICATION TO STRESSES

The numerical value of the individual stress components should be listed with and without the stress concentration or fatigue strength reduction factor applied. Factors should be applied to each individual stress component and not applied to the total stress at a point or to the stress intensity.

C-1450 COMBINED STRESSES AND ALLOWABLE NUMBER OF CYCLES

Where the rules do not specifically control this, as they do in NB-3500 and NB-3600, methods of combining stresses, determining principal stresses, determining alternating stress intensity, and determining cumulative damage effects and allowable number of cycles should be shown in the Report. These results should be reconciled with the required values.

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NONMANDATORY APPENDIX D
PREHEAT GUIDELINES

ARTICLE D-1000
GUIDELINES

D-1100 INTRODUCTION

D-1110 SCOPE

The preheat temperatures given herein are a general guide for the materials listed in P-Numbers of Section IX. Specific rules for preheating are not given since the need for preheat and the minimum preheat temperatures vary and are dependent on a number of factors, such as chemical analysis, degree of restraint, physical properties, and thickness. Preheat requirements for different materials of the same P-Number may be more or less restrictive depending upon the specific circumstances associated with making a particular weld. The welding procedure specification for the material being welded shall specify the minimum preheating requirements under the welding procedure qualification requirements of Section IX.

D-1120 TEMPERATURE MAINTENANCE

(a) Difficulty may be experienced with certain materials if the temperature is allowed to fall below the preheat temperature between passes. It may be desirable to maintain the preheat temperature or to heat the joint to the postweld heat treatment temperature before allowing it to cool to ambient temperature.

(b) The preheat temperature may be checked by suitable methods, such as temperature-indicating crayons or thermocouple pyrometers, to ensure that the required preheat temperature is maintained during the welding operation. The Certificate Holder should be cautious in the use of temperature-indicating crayons and pellets because some metals may be severely attacked by the chemicals in crayons or pellets at elevated temperatures.

D-1200 FERROUS MATERIALS

D-1210 PREHEAT TEMPERATURES

Suggested minimum preheat temperatures are given in Table D-1210-1.

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Table D-1210-1
Suggested Minimum Preheat Temperatures

(15)

P-No.	Base Metal Thickness, T, in. (mm) and/or Other Description	Nominal Thickness, t, in. (mm) [Note (1)]	Carbon, %	Minimum Preheat, °F (°C)
1 Gr. 1 and Gr. 2	$T \le 1\frac{1}{2}$ (38)	$t \le 1\frac{1}{4}$ (32)	≤ 0.30	50 (10)
	$T \le 1\frac{1}{2}$ (38)	$t \le \frac{3}{4}$ (19)	> 0.30	50 (10)
	$T \le 1\frac{1}{2}$ (38)	$t > 1\frac{1}{4}$ (32) and $\le 1\frac{1}{2}$ (38)	≤ 0.30	200 (95)
	$T \le 1\frac{1}{2}$ (38)	$t > \frac{3}{4}$ (19) and $\le 1\frac{1}{2}$ (38)	> 0.30	200 (95)
	$T > 1\frac{1}{2}$ (38)	$t \le \frac{3}{4}$ (19)	...	200 (95)
	$T > 1\frac{1}{2}$ (38)	$t > \frac{3}{4}$ (19)	...	250 (120)
1 Gr. 3, 3 Gr. 3	Material with maximum tensile strength greater than 70 ksi (485 MPa)	...		250 (120)
	T unlimited	$t > \frac{5}{8}$ (16)	...	250 (120)
	T unlimited	$t \le \frac{5}{8}$ (16)	...	50 (10)
4	Material with specified minimum tensile strength greater than 60 ksi (415 MPa)	...	...	300 (150)
	T unlimited	$t > \frac{1}{2}$ (13)	...	300 (150)
	T unlimited	$t \le \frac{1}{2}$ (13)	...	250 (120)
5A, 5B, 5C	Material with specified minimum tensile strength greater than 60 ksi (415 MPa)	...	...	400 (205)
	T unlimited with Cr > 6.0%	$t > \frac{1}{2}$ (13)	...	400 (205)
	T unlimited with Cr ≤ 6.0%	$t > \frac{1}{2}$ (13)	...	300 (150)
	T unlimited	$t \le \frac{1}{2}$ (13)	...	300 (150)
6	Type 410S welded with A-No. 8, A-No. 9 or F-No. 43 filler metals	$t \le \frac{3}{8}$ (10)	≤ 0.08	Follow Material Manufacturer's recommendations
	All other materials	...	...	400 (205)
7	All materials	...	...	Follow Material Manufacturer's recommendations
8	All materials	...	...	Follow Material Manufacturer's recommendations
9A Gr. 1 and 9B Gr. 1	All welds provided the procedure qualification is made in equal or greater thickness than production weld and need not exceed $1\frac{1}{2}$ in. (38 mm)	$t \le \frac{5}{8}$ (16)	...	200 (95)
	Attachment welds joining nonpressure-retaining material to pressure-retaining materials over $\frac{5}{8}$ in. (16 mm)	$t \le \frac{1}{2}$ (13)	...	200 (95)
	Circumferential butt weld in pipe NPS 4 (DN 100) or less, and tubes with nominal O.D. 4.5 in. (114 mm) or less and attachment welds	$t \le \frac{1}{2}$ (13)	≤ 0.15	250 (120)
	Socket welds in pipe NPS 2 (DN 50) or less and tubes with nominal O.D. $2\frac{3}{8}$ in. (60 mm) or less	$t \le \frac{1}{2}$ (13)	≤ 0.15	250 (120)
	All other materials	...	...	300 (150)
10A Gr. 1	All materials	...	...	200 (95)

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Table D-1210-1
Suggested Minimum Preheat Temperatures (Cont'd)

P-No.	Base Metal Thickness, T, in. (mm) and/or Other Description	Nominal Thickness, t, in. (mm) [Note (1)]	Carbon, %	Minimum Preheat, °F (°C)
10C Gr. 1	$T \le 1\frac{1}{2}$ (38)	$t \le 1\frac{1}{4}$ (32)	≤ 0.30	Follow Material Manufacturer's recommendations
	$T \le 1\frac{1}{2}$ (38)	$t \le \frac{3}{4}$ (19)	> 0.30	Follow Material Manufacturer's recommendations
	$T \le 1\frac{1}{2}$ (38)	$t > 1\frac{1}{4}$ (32) and $t \le 1\frac{1}{2}$ (38)	≤ 0.30	200 (95)
	$T \le 1\frac{1}{2}$ (38)	$t > \frac{3}{4}$ (19) and $t \le 1\frac{1}{2}$ (38)	> 0.30	200 (95)
	$T > 1\frac{1}{2}$ (38)	$t \le \frac{3}{4}$ (19)	...	200 (95)
	$T > 1\frac{1}{2}$ (38)	$t > \frac{3}{4}$ (19)	...	250 (120)
10H	All materials	...	...	Follow Material Manufacturer's recommendations
10I Gr. 1	$T \le \frac{1}{2}$ (13)	$t \le \frac{1}{2}$ (13)	...	Follow Material Manufacturer's recommendations
	$T > \frac{1}{2}$ (13)	...	...	300 (150)
10K	All materials	...	...	Follow Material Manufacturer's recommendations
11A Gr. 1	$T \le \frac{1}{2}$ (13)	$t \le \frac{1}{2}$ (13)	...	Follow Material Manufacturer's recommendations
	$T > \frac{1}{2}$ (13)	...	...	250 (120)
11A Gr. 4	All materials	...	...	250 (120)
11A Gr. 5	All materials	...	...	Follow Material Manufacturer's recommendations
11B Gr. 1, Gr. 2, Gr. 3, Gr. 4, Gr. 8	All materials	...	...	Follow Material Manufacturer's recommendations
15E	All materials	...	...	400 (205)

GENERAL NOTE: Minimum preheat temperature requirements in the respective Subsections and Divisions take precedence over Table D-1210-1.

NOTE:
(1) Nominal thickness, t, for PWHT exemptions is defined in the respective Subsections and Divisions.

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NONMANDATORY APPENDIX E

ARTICLE E-1000
MINIMUM BOLT CROSS-SECTIONAL AREA

E-1100 INTRODUCTION

E-1110 SCOPE

This Article implements NB-3230 by providing specific methods for the determination of the minimum bolt cross-sectional area. Stresses in the bolts during service must also satisfy the requirements of NB-3232.2 and NB-3232.3. Evaluation of service bolt stresses requires analysis in addition to that described by this Article, generally involving the performance of a discontinuity analysis in accordance with the principles described in Article A-6000.

(15) E-1120 NOMENCLATURE[25]

The nomenclature defined below is used in the equations of this Article.

A_b = actual total cross-sectional area of bolts at root of thread or section of least diameter under stress
A_m = total design cross-sectional area of bolts, taken as the greater of A_{m1} and A_{m2}
A_{m1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the Design Conditions
= W_{m1}/S_b
A_{m2} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating
= W_{m2}/S_a
b = effective gasket or joint contact surface seating width[26]
b_o = basic gasket seating width (Table XI-3221.1-2)
C_b = effective width factor
= 0.5 for U.S. Customary calculations
= 2.5 for SI calculations
G = diameter at location of gasket load reaction. G is defined as follows (Table XI-3221.1-2): when $b_o \le 1/4$ in. (6 mm), G is the mean diameter of gasket contact face; when $b_o > 1/4$ in. (6 mm), G is the outside diameter of gasket contact face less $2b$.
H = total hydrostatic end force
= $0.785G^2P$
m = gasket factor obtained from Table XI-3221.1-1[26]
N = width used to determine the basic gasket seating width, b_o, based upon the contact width of the gasket (Table XI-3221.1-2)
P = Design Pressure
S_a = allowable bolt stress at atmospheric temperature (Section II, Part D, Subpart 1, Table 4)
S_b = allowable bolt stress at Design Temperature (Section II, Part D, Subpart 1, Table 4)
T = thickness used to determine the basic gasket seating width, b_o (Table XI-3221.1-2)
W_{m1} = minimum required bolt load for the Design Pressure (E-1210)
W_{m2} = minimum required bolt load for gasket seating (E-1210)
w = width used to determine the basic gasket seating width b_o, based upon the contact width between the flange facing and the gasket (Table XI-3221.1-2)
y = minimum design seating stress

E-1200 DESIGN CROSS-SECTIONAL AREA

E-1210 BOLT LOADS

(a) Design Bolt Loads. The flange bolt loads used in calculating the design cross-sectional area of bolts shall be determined as stipulated in (1) and (2).

(1) The design bolt load for the Design Pressure W_{m1} shall be sufficient to resist the hydrostatic end force H, exerted by the Design Pressure on the area bounded by the diameter of gasket reaction, and, in addition, to maintain on the gasket or joint contact surface a compression load H_p, which experience has shown to be sufficient to ensure a tight joint. This compression load is expressed as a multiple m of the internal pressure. Its value is a function of the gasket material and construction.[26] The design bolt load for the Design Pressure W_{m1} is determined in accordance with eq. (1).

$$W_{m1} = H + H_p = 0.785G^2P + (2b \times 3.14GmP) \quad (1)$$

(2) Before a tight joint can be obtained it is necessary to seat the gasket or joint contact surface properly by applying a minimum initial load, under atmospheric

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(15)

Table E-1210-1

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temperature conditions without the presence of internal pressure, which is a function of the gasket material and the effective gasket area to be seated. The minimum initial bolt load W_{m2}, required for this purpose, shall be determined in accordance with eq. (2).

$$W_{m2} = 3.14bGy \quad (2)$$

The need for providing sufficient bolt load to seat the gasket or joint contact surfaces in accordance with eq. (2) will prevail on many low pressure designs and with facings and materials that require a high seating load, and where the bolt load, computed by (1), eq. (1) for the Design Pressure, is insufficient to seat the joint. Accordingly, it is necessary to furnish bolting and to pretighten the bolts sufficiently to satisfy both of these requirements, each one being individually investigated. When eq. (2) governs, flange proportions will be a function of the bolting instead of internal pressure.

(b) Minimum Required and Actual Bolt Areas A_m and A_b. The minimum cross-sectional area of bolts A_m required for both the Design Pressure and gasket seating is the greater of the values for A_{m1} and A_{m2}, where $A_{m1} = W_{m1}/S_b$ and $A_{m2} = W_{m2}/S_a$. A selection of bolts to be used shall be made such that the actual total cross-sectional area of bolts A_b will not be less than A_m (NB-3230).

(c) Bolt Loads for Flanges Using Gaskets of the Self-Energizing Type

(1) The design bolt load for the Design Pressure W_{m1} shall be sufficient to resist the hydrostatic end force H, exerted by the maximum allowable working pressure on the area bounded by the outside diameter of the gasket. H_p is to be considered as 0 for all self-energizing gaskets, except certain seal configurations that generate axial gasket loads which shall be considered.

(2) $W_{m2} = 0$.

Self-energizing gaskets are considered to be those that require only an inconsequential amount of bolt force to produce an initial seal. The bolting, however, may have to be sufficiently pretightened to prevent extrusion of the gasket, to prevent cyclic fatigue of the bolts, or to resist any external loads or moments that may be imposed on the joint.

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Table E-1210-2

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NONMANDATORY APPENDIX F

ARTICLE F-1000
RULES FOR EVALUATION OF SERVICE LOADINGS WITH LEVEL D SERVICE LIMITS

F-1100 INTRODUCTION

F-1110 SCOPE

This Appendix provides rules and service limits which may be used by Owners and N-Type Certificate Holders for evaluating components and supports subjected to loads for which Level D Service Limits are specified by the Design Specification (NCA-3250).

F-1200 INTENT OF LEVEL D SERVICE LIMITS

(a) The Level D Service Limits and design rules contained in F-1300 are provided for limiting the consequences of the specified event. They are intended (NCA-2142) to assure that violation of the pressure-retaining boundary will not occur, but are not intended to assure operability of components either during or following the specified event.

(b) The limits and rules specified for core support structures are intended to assure maintenance of structural integrity but not to prevent leakage.

(c) The limits and rules of this Appendix need not be applied to that portion of a component or support in which a failure has been postulated.

(d) Table F-1200-1 provides references for component and support elastic system analysis acceptance criteria.

F-1300 LEVEL D SERVICE LIMITS AND DESIGN RULES

F-1310 GENERAL

(a) These limits and design rules are provided to limit the consequences of loads for which Level D Service Limits are specified in the Design Specification.

(b) The contents of F-1320 provide general procedures which are applicable to all components and supports. Specific procedures, which may be used as alternatives to the procedures of F-1320, are provided as follows:

(1) F-1400, Vessels
(2) F-1410, Pumps
(3) F-1420, Valves
(4) F-1430, Piping
(5) F-1440, Core Support Structures

(c) Only limits on primary stresses are prescribed. Unless specifically required by this Appendix, self-relieving stresses resulting from loads for which Level D Service Limits are specified need not be considered.

(d) When compressive stresses are present the stability of the component or component support shall be considered.

(e) Potential for unstable crack growth shall also be considered.

F-1320 DESIGN BY ANALYSIS

F-1321 Terms Related to Analysis

Terms used in this paragraph relating to analysis are defined in the following subparagraphs. Additional definitions are given in NB-3213.

F-1321.1 System Analysis. System analysis is performed to determine loads on components and supports. A system is an assemblage of components, supports, and other interconnected structures. The system analysis is generally dynamic due to the nature of the loads.

F-1321.2 Component and Support Analysis. Component analysis is the calculation of stresses and displacements in a component or support to determine compliance with the service limits listed herein. Loads applied in the component analysis and support analysis shall include those determined in a system analysis plus additional loads as applicable.

F-1321.3 Elastic Analysis. Elastic analysis is based on the assumption of a linear relationship between stress and strain. Consideration of gaps between parts of the structure may cause the relationship between loads and deformations to be nonlinear.

F-1321.4 Inelastic Analysis. Inelastic analysis is a class of methods which computes structural behavior considering nonlinearities in the relationship between stresses and strains. Inelastic analysis as applied in this Appendix shall not be considered to include the time-dependent effects of creep.

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Table F-1200-1
Level D Service Limits — Components and Supports Elastic System Analysis Acceptance Criteria

System	Primary Membrane and Bending Stress	Bearing Stress	Shear	Compressive Loads	Alternative Criteria		
					Interaction Method	Load Rating	Plastic Analysis
Components [Note (1)]	F-1331.1	F-1331.3, F-1336	F-1331.1(d)	F-1331.5	F-1331.2 [Note (2)]	...	F-1340 [Note (3)]
Plate and Shell Type Supports [Note (4)]	F-1332.1, F-1332.2	F-1332.3, F-1336	F-1332.4	F-1331.5(a)	...	F-1332.7 [Note (5)]	F-1340 [Note (3)]
Linear Type Supports [Note (4)], [Note (6)]	F-1334.1, F-1334.4 [Note (7)]	F-1334.10, F-1336	F-1334.2	F-1334.3, F-1334.5	F-1334.7 [Note (8)]	F-1332.7 [Note (9)]	F-1340 [Note (3)]
Bolted Joints [Note (10)], [Note (11)]	F-1335.1	F-1335.2	F-1335.2, F-1335.3	...	...	...	...

GENERAL NOTE: The following Design Rules shall be followed.

(1) Level D self-relieving stresses need not be considered unless as specified within a specific section of Nonmandatory Appendix F.

(2) In addition to consideration of primary stresses, stability of the component or support shall be considered when compressive stresses are present.

(3) Potential for unstable crack growth shall be considered.

(4) Allowables shall be adjusted to reflect any different material behavior from that which the allowables are based. See F-1322.3.

(5) Geometric nonlinearities shall be considered when appropriate.

(6) Design specification requirements shall be satisfied in addition to those given in Nonmandatory Appendix F.

NOTES:

(1) For Vessels and Pumps, the design rules given in F-1330 and F-1340 shall be used; for Valves, as an alternative to the procedures of F-1300, F-1420 can be used; for Piping, as an alternative to the procedures of F-1331, F-1430 can be used. For Core Support Structures, the procedures of F-1300 may be used except as stipulated in F-1440.

(2) As an alternative to the requirements of F-1331.1, the interaction method may be used following F-1331.2 and the procedures given in Article A-9000.

(3) Per F-1322, if elastic system analysis is used, the components and supports may be designed alternatively on the acceptance criteria of F-1340 provided a reevaluation of the system analysis is performed to determine that it has not been significantly invalidated due to load and stress redistribution and changes in geometry. See F-1322.1 for further discussion.

(4) Stresses resulting from constraint of free end displacements and anchor point motion shall be considered primary stresses. Neither peak stresses nor stresses resulting from thermal expansion within the support need be evaluated.

(5) As an alternative to F-1332.1 through F-1332.6 requirements.

(6) Per F-1334, the allowable stress presented in NF-3320 for Level A Service Condition may be increased using the following factors: the smaller of 2 or 1.167 S_u / S_y if S_u > 1.2 S_y. Note that members must be checked for local and general instability.

(7) As an alternative to the requirements in F-1334.4, a collapse load analysis is acceptable following F-1334.6.

(8) As an alternative to the requirements of F-1334.1 through F-1334.5, the interaction method may be used following F-1334.7 and the procedures given in Article A-9000.

(9) As an alternative to the requirements of F-1334.1 through F-1334.5, F-1332.7 may be used.

(10) Threaded structural fasteners used in core support structures shall be evaluated using the rules of F-1440.

(11) Minimum edge distance shall meet the requirements of F-1335.4.

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(a) Limit Analysis. Limit analysis is that method which computes the maximum load or combination of loads a structure made of ideally plastic (nonstrain-hardening) material can carry. Limit analysis for components is described in NB-3213.27, NB-3213.28, and NB-3213.29 and for linear supports in NF-3340.

(b) Plastic Analysis. Plastic analysis is defined in NB-3213.24.

F-1321.5 Interaction Method. Interaction method (formerly Stress Ratio Method) is a method used to evaluate the adequacy of structures under combined loads. Procedures for the method are described in Article A-9000.

F-1321.6 Collapse. Collapse of a structure occurs upon formation of an unstable mechanism of plastic hinges under a given combination of loads. The collapse load is defined as follows:

(a) limit analysis collapse load is defined in NB-3213.28

(b) test collapse load is defined as that load determined by test according to the criteria given in II-1430

(c) plastic analysis collapse load is defined as that load determined by plastic analysis according to the criteria given in II-1430

F-1321.7 Plastic Instability Load. Plastic instability load is defined in NB-3213.26.

F-1321.8 Experimental Stress Analysis. Experimental stress analysis is a method for evaluation of structural behavior, based on direct measurement of response of test specimens, where the configuration of the model and the applied loads are representative of the component or support under consideration. Response parameters measured are selected to provide data appropriate to the acceptance criteria applied. Experimental methods are provided for in Mandatory Appendix II.

F-1321.9 Primary Stress Intensities. The definitions of general primary membrane stress intensity, designated as P_m; local primary membrane stress intensity, designated as P_L; and primary bending stress intensity, designated as P_b, are provided in NB-3221.

F-1322 Methods and Requirements for Analysis

The following requirements shall be satisfied in the evaluation of components or supports under the loads or load combinations for which Level D Service Limits are specified.

F-1322.1 Analysis Combinations.

(a) System analysis may be performed by elastic analysis methods as defined in F-1321.3 or by plastic analysis methods as defined in F-1321.4(b). If elastic system analysis is used, the components and supports shall be designed to meet the acceptance criteria in F-1330. If plastic system analysis is used, the components and supports may be designed based on the acceptance criteria in F-1340.

(b) If elastic system analysis is used, the components and supports may be designed based alternatively on the acceptance criteria of F-1340 provided a reevaluation of the system analysis is performed to determine that it has not been significantly invalidated due to load and stress redistribution and changes in geometry. Some of the conditions under which this analysis combination may be acceptable are

(1) the plastic deformation is highly localized or

(2) the changes in geometry are not significant

This combination may also be considered valid if bounding solutions are established which conservatively account for redistribution of loads and stresses due to plasticity.

(c) If all loads on a component or support are determined independently from system behavior (e.g., specified pressures), then the component or support may be designed based on acceptance criteria in either F-1330 or F-1340.

(d) The Design Specification for the components and supports shall indicate what type of system analysis (if any) has been used to derive the specified loads.

F-1322.2 Dynamic Effects. Postulated events for which Level D Service Limits are specified are generally dynamic in nature. The determination of loads for components and supports shall account for dynamic amplification of structural response, both in the component and in the system.

F-1322.3 Material Behavior.

(a) The mechanical and physical properties shall be taken from Section II, Part D, Subparts 1 and 2 at the actual temperature of the material. The allowable stresses shall be based on material properties given in Section II, Part D, Subpart 1 at temperature. If S_u values at temperature are not tabulated in Section II, Part D, Subpart 1, Table U,[27] the value used shall be included and justified in the Design Report.

(b) The stress and strain allowables given in this Appendix are based on an engineering stress–strain curve. If another type of stress–strain curve (e.g., true stress–strain or Kirchoff stress–strain) is used, the results from the analysis or the allowables given in this Appendix shall be appropriately transformed.

(c) When performing plastic analysis, the stress–strain curve used shall be included and justified in the Design Report. It is permissible to adjust the stress–strain curve to include strain rate effects resulting from dynamic behavior. However, the allowables shall be selected in accordance with the preceding (a).

(d) The yield criteria and associated flow rule used in the inelastic analysis may be either those associated with the maximum shear stress theory (Tresca) or the strain energy distortion theory (Von Mises).

F-1322.4 Geometric Nonlinearities. Geometric nonlinearities may be produced by relatively large deformations and/or rotations and by gaps between parts of the

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structure. Analyses performed for derivation of loads and for evaluation of acceptability of components and supports shall consider geometric nonlinearities if appropriate.

F-1322.5 Strain and/or Deformation Limits. In addition to the limits given in this Appendix, the strain or deformation limits (if any) provided in the Design Specification shall be satisfied.

F-1330 ACCEPTANCE CRITERIA USING ELASTIC SYSTEM ANALYSIS

The acceptance criteria in this Section shall be applied when elastic system analysis is used to determine loads on components and supports. These criteria are subject to the restrictions on methods of evaluation stated in F-1322.

F-1331 Criteria for Components

F-1331.1 Elastic Analysis.

(a) The general primary membrane stress intensity P_m shall not exceed the lesser of $2.4S_m$ and $0.7S_u$ for austenitic steel, high-nickel alloy, and copper-nickel alloy materials included in Section II, Part D, Subpart 1, Tables 2A and 2B, or $0.7S_u$ for ferritic steel materials included in Table 2A.

(b) The local primary membrane stress intensity P_L shall not exceed 150% of the limit for general primary membrane stress intensity P_m.

(c) The primary membrane (general or local) plus primary bending stress intensity $P_L + P_b$ shall be limited in accordance with one of the following provisions:

(1) stress intensity $P_L + P_b$ shall not exceed 150% of the limit for general primary membrane stress intensity P_m

(2) static or equivalent static loads shall not exceed 90% of the limit analysis collapse load using a yield stress which is the lesser of $2.3S_m$ and $0.7S_u$, or 100% of the plastic analysis collapse load or test collapse load (F-1321.6)

(d) The average primary shear stress across a section loaded in pure shear shall not exceed $0.42S_u$.

F-1331.2 Interaction Method. As an alternative to the requirements of F-1331.1 above, acceptability of individual members of components may be demonstrated using the interaction method. Procedures for interaction method analysis are given in Article A-9000. The allowable stress S_{al} shall not exceed the lesser of $2.4S_m$ and $0.7S_u$.

F-1331.3 Bearing Stresses. Except for pinned and bolted joints, bearing stresses need not be evaluated for loads for which Level D Service Limits are specified.

F-1331.4 Stress Limits for Bolts. Bolts shall be evaluated in accordance with the rules of F-1335.

F-1331.5 Requirements for Compressive Loads. Components subjected to compressive loads shall be evaluated against buckling limits. Maximum compressive load (or stress) shall be limited to a value established by (a), (b), or (c).

(a) Two-thirds of the value of buckling load (or stress) determined by one of the following methods:

(1) comprehensive analysis which considers effects such as geometric imperfections, deformations due to existing loading conditions, nonlinearities, large deformations, residual stresses, and inertial forces

(2) tests of physical models under conditions of restraint and loading the same as those to which the configuration is expected to be subjected

(b) a value equal to 150% of the limit established by the rules of NB-3133, except that the pressure is permitted to be 250% of the given value when the ovality is limited to 1% or less

(c) a value determined in accordance with the procedures contained in Code Case N-284 for metal containment shell buckling design methods using a factor of safety of 1.34

F-1332 Criteria for Plate and Shell Type Supports

The criteria presented in this paragraph pertain to primary stresses only. Stresses resulting from constraint of free end displacement and anchor point motion (NF-3121.12 and NF-3121.13) shall be considered primary stresses in the evaluation. Neither peak stresses nor stresses resulting from thermal expansion within the support need be evaluated.

F-1332.1 Primary Membrane Stress Intensity and Primary Membrane Stress Limit.

(a) For Class 1 supports, the general primary membrane stress intensity P_m is limited to the greater of $1.2S_y$ and $1.5S_m$, but may not exceed $0.7S_u$.

(b) For Class 2, 3, and MC supports, the general membrane principal stress is limited to the greater of $1.2S_y$ and $1.5S$, but may not exceed $0.7S_u$.

F-1332.2 Primary Membrane Plus Bending Stress Intensity and Primary Membrane Plus Bending Stress Limit.

(a) For Class 1 supports, the general primary membrane plus primary bending stress intensity, $P_m + P_b$, shall be limited in accordance with one of the following provisions:

(1) 150% of the limit for general primary stress intensity P_m

(2) static or equivalent static loads not exceeding 90% of the limit analysis collapse load (F-1321.6) using a yield strength which is the lesser of $1.2S_y$ and $0.7S_u$, or 100% of the plastic analysis collapse load or test collapse load (F-1321.6)

(b) For Class 2, 3, and MC supports, the local membrane plus bending principal stress is limited to the 150% of the general membrane stress limit.

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F-1332.3 Bearing Stress. Except for pinned and bolted joints, bearing stresses need not be evaluated for loads for which Level D Service Limits are specified.

F-1332.4 Pure Shear. The average primary shear across a section loaded in pure shear shall not exceed $0.42S_u$.

F-1332.5 Requirements for Compressive Stresses. Plate and shell type supports subject to compressive stresses shall be evaluated in accordance with the rules of F-1331.5(a).

F-1332.6 Stress Limits for Bolts. Bolts shall be evaluated in accordance with the rules of F-1335.

(15) **F-1332.7 Load Rating.** As an alternative to the requirements of F-1332.1 through F-1332.6 above, plate and shell type supports may be qualified to Service Level D Limits using the procedure for load rating (NF-3282). The load rating for Level D Service Loadings shall be determined by the following equation:

$$\text{load rating} = TL \times 0.7\frac{S_u}{S_u^*}$$

where

TL = support test load equal to or less than the load under which the support fails to perform its specified support function
S_u = tensile strength of the support material at temperature
S_u^* = tensile strength of the support material at test temperature

but

$$\frac{S_u}{S_u^*} \le 1$$

F-1333 Criteria for Standard Supports

Criteria in F-1332 or F-1334 shall be applied according to whether standard supports are plate and shell or linear type supports.

F-1334 Criteria for Linear Type Supports

The criteria presented in this paragraph pertain to primary stresses only. Stresses resulting from constraint of free end displacement and anchor point motion (NF-3121.12 or NF-3121.13) shall be considered primary stresses in the evaluation.

Neither peak stresses nor stresses resulting from thermal expansion within the support need be evaluated.

Unless otherwise specified, the allowable stresses presented (NF-3320) for Level A Service Condition may be increased using the following factors: the smaller of 2 or $1.167S_u/S_y$ if $S_u > 1.2S_y$, or 1.4 if $S_u \le 1.2S_y$, where S_y is the yield strength, ksi (MPa), and S_u is the ultimate tensile strength, ksi (MPa), both at temperature. In addition, members must be checked for local and general instability.

F-1334.1 Stresses in Tension. The tensile stress on the net section, except at pin holes and in the through-plate thickness direction, shall not exceed the lesser of $1.2S_y$ and $0.7S_u$.

F-1334.2 Stresses in Shear. The shear stress on the gross section shall not exceed the lesser of $0.72S_y$ and $0.42S_u$. Gross section shall be determined in accordance with NF-3322.1(b).

F-1334.3 Axial Compression. Maximum load in axially loaded compression members shall be limited in accordance with either (a) or (b). (15)

(a) Two-thirds of the buckling load, as determined by one of the following methods:

(1) comprehensive stability analysis which considers effects such as large deformations, deformations due to existing loading conditions, material nonlinearities, local buckling, out-of-straightness and other tolerances, load eccentricity, end conditions, residual stresses and inertia loads (for dynamic loading)

(2) testing of a full-scale prototype under conditions of support and loading the same as those to which the actual compression member is expected to be subjected

(b) the maximum allowable load for ferritic steels shall be determined in accordance with the following provided that the initial out-of-straightness does not exceed $^1/_{1000}$ of the unsupported length. Effects of deformations due to existing loads shall also be considered.

(1) Except as noted in (2), the following rules shall be applied:

For $0 \le \lambda \le 1$

$$\frac{P}{P_y} = \frac{1 - \lambda^2/4}{1.11 + 0.50\lambda + 0.17\lambda^2 - 0.28\lambda^3}$$

For $1 \le \lambda \le \sqrt{2}$

$$\frac{P}{P_y} = \frac{2}{3}\left(1 - \lambda^2/4\right)$$

For $\lambda > \sqrt{2}$

$$\frac{P}{P_y} = \frac{2}{3\lambda^2}$$

where

A_g = area of gross section
E = modulus of elasticity
K = effective length factor
L = unbraced length
P = maximum allowable load
P_y = S_yA_g
r = radius of gyration

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$$\lambda = \left(\frac{KL}{r}\right)\frac{1}{\pi}\sqrt{\frac{S_y}{E}}$$

(2) For nonstress-relieved heavy structural shapes [web or flange thickness greater than 1 in. (25 mm)] or for nonstress-relieved built-up members using universal mill plate, the following rules shall be applied:

For $0 < \lambda < 1$

$$\frac{P}{P_y} = \frac{1 - \lambda^2/4}{1.11 + 0.75\lambda + 0.83\lambda^2 - 0.81\lambda^3}$$

For $1 < \lambda < \sqrt{2}$

$$\frac{P}{P_y} = \frac{1 - \lambda^2/4}{1.88}$$

For $\lambda > \sqrt{2}$

$$\frac{P}{P_y} = \frac{1}{1.88\lambda^2}$$

F-1334.4 Combined Axial Tension and Bending.

(a) For members subject to both axial tension and bending stresses, the following equation shall be satisfied:

$$\frac{f_a}{F_a} + \frac{f_{bx}}{F_{bx}} + \frac{f_{by}}{F_{by}} \le 1$$

where

F_a = smaller of $1.2S_y$ or $0.7S_u$

(b) For members qualifying as compact sections under criteria of NF-3322.1(d)(1), the maximum bending stress shall be given by

$$F_b = fS_y$$

where

f = plastic shape factor for the cross section

(c) If members do not meet the compact section requirements, they shall be designed using one of two methods below to determine F_b for use in the preceding equation.

(1) Allowable values for F_b given in NF-3322.1(d)(2) may be increased by a factor of 1.11 (to maintain a nominal factor of safety of 1.5 against instability).

(2) Rigorous analysis of member stability may be used to determine critical bending stress. A factor of safety of 1.5 shall be used in determining allowable design bending stress.

F-1334.5 Combined Axial Compression and Bending. Members subject to combined axial compression and bending shall satisfy NF-3322.1(e)(1) eqs. (20), (21), and (22) with allowable stresses defined as follows.

(a) $F_a = P/A_g$ where P shall be determined in accordance with F-1334.3.

(b) The value of F'_e shall be taken as

$$F'_e = \frac{\pi^2 E}{1.30\left(Kl_b/r_b\right)^2}$$

with terms as defined in NF-3313.1.

(c) F_b shall be determined using F-1334.4(b) or F-1334.4(c) as appropriate.

F-1334.6 Collapse Load Analysis. As an alternative to the requirements in F-1334.4 above, acceptability of linear type supports may be established using one of the following methods: (15)

(a) using lower bound limit analysis given in NF-3340, static or equivalent static loads shall not exceed 90% of the limit analysis collapse load using a yield stress which is the greater of $1.2S_y$ and $1.5S_m$, but not larger than $0.7S_u$

(b) 100% of the plastic analysis collapse load

(c) 100% of the test collapse load (F-1321.6)

(d) In NF-3322.1(e)(1), eq. (21), replace $0.6S_y$ with the smaller of $1.2S_y$ or $0.7S_u$.

F-1334.7 Interaction Method. As an alternative to the requirements of F-1334.1 through F-1334.5 above, acceptability for individual structural members of linear type supports may be demonstrated using the interaction method. Procedures for interaction method analysis are given in Article A-9000. The allowable stress S_{al} shall not exceed the greater of $1.2S_y$ and $1.5S_m$, but not larger than $0.7S_u$.

F-1334.8 Load Rating. As an alternative to the requirements of F-1334.1 through F-1334.5 above, linear type supports may be qualified to Service Level D Limits using load rating criteria given in F-1332.7.

F-1334.9 Stress Limits for Bolts. Bolts shall be evaluated in accordance with the rules of F-1335.

F-1334.10 Bearing Stresses. Except for pinned and bolted joints, bearing stresses need not be evaluated for loads for which Level D Service Limits are specified.

F-1335 Requirements for Bolted Joints

(a) The requirements provided in this Section shall be applied to components and supports. Threaded structural fasteners used in core support structures shall be evaluated using the rules of F-1440.

(b) Allowable stresses for bolts are given in the paragraphs below. These allowable stresses are only applicable if the bolt stresses are calculated using elastic methods.

F-1335.1 Allowable Tensile Stress. The average tensile stress computed on the basis of the available tensile stress area shall not exceed the smaller of $0.7S_u$ and S_y. When high strength bolts or threaded parts having an ultimate tensile strength greater than 100 ksi (700 MPa) at

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operating temperature are used in component applications, the maximum value of the stress at the periphery of the bolt cross section resulting from direct tension plus bending and excluding stress concentrations shall not exceed S_u. The bolt load shall be the sum of the external load and any bolt tension resulting from prying action produced by deformation of the connected parts.

F-1335.2 Allowable Shear Stress.

(a) For bearing type joints, the average bolt shear stress expressed in terms of available shear stress area shall not exceed the smaller of $0.42S_u$ and $0.6S_y$.

(b) Friction type joints shall be evaluated using the rules of NF-3324.6(a)(3)(-b).

F-1335.3 Combined Tensile and Shear Stress.

(a) Bolts subjected to combined shear and tension in bearing type joints shall be so proportioned that the shear and the tensile stresses satisfy the following equation:

$$\frac{f_t^2}{F_{tb}^2} + \frac{f_v^2}{F_{vb}^2} \le 1$$

where

f_t = computed tensile stress
f_v = computed shear stress
F_{tb} = allowable tensile stress at temperature per F-1335.1
F_{vb} = allowable shear stress at temperature per F-1335.2(a)

(b) In friction type joints, the joint clamping force will be reduced by any direct tension load on the joint. Therefore, the bolt clamping force used to calculate the allowable shear load in NF-3324.6(a)(3)(-b) shall be reduced by an equivalent amount.

F-1335.4 Minimum Edge Distance in Line of Load. In both bearing and friction type joints, the minimum distance from the center of the end bolt to that edge of the connected part toward which the load is directed shall satisfy (a) and (b):

(a) $L/d \ge [0.5 + 1.2(f_p/S_u)]$
(b) $f_p/S_u \le 2.1$

where

d = nominal diameter of bolt
f_p = nominal bearing stress
$= P/dt$
L = distance from center of bolt hole to edge of connected part
P = bearing load transmitted by the fastener
S_u = tensile strength of the connected parts at temperature
t = thickness of the connected part

F-1336 Requirements for Pinned Joints

Except for bearing, pinned joints shall be evaluated in accordance with the acceptance criteria of F-1331. The allowable bearing stress shall be $2.1S_u$.

F-1337 Requirement for Support Fillet Welds

For supports, the allowable stresses for fillet welds shall be 1.7 times the limits for the Design Loading as described in NF-3324.5.

F-1340 ACCEPTANCE CRITERIA USING PLASTIC SYSTEM ANALYSIS

The acceptance criteria in this section may be applied provided the system analysis considers effects of material nonlinear behavior. The criteria are subject to the restrictions on methods of evaluation stated in F-1322.

F-1341 Criteria for Components

Acceptability of components may be demonstrated using any one of the following methods:

(a) elastic analysis
(b) plastic analysis
(c) collapse load analysis
(d) plastic instability analysis
(e) interaction method

The primary stress limits for these alternative methods are given in F-1341.1 through F-1341.5. The other limits given in F-1341.6 and F-1341.7 shall also be satisfied as applicable.

F-1341.1 Elastic Analysis. Where the component is evaluated on an elastic basis, the following primary stress limits shall be applied.

(a) The general primary membrane stress intensity P_m shall not exceed the lesser of $2.4S_m$ and $0.7S_u$ for austenitic steel, high-nickel alloy, and copper-nickel alloy materials included in Section II, Part D, Subpart 1, Tables 2A and 2B, or $0.7S_u$ for ferritic steel materials included in Section II, Part D, Subpart 1, Table 2A.

(b) The local primary membrane stress intensity P_L shall not exceed 150% of the limit for general primary membrane stress intensity P_m.

(c) The primary membrane (general or local) plus primary bending stress intensity, $P_L + P_B$, shall not exceed 150% of the limit for general primary membrane stress intensity P_m.

(d) The average primary shear across a section loaded in pure shear shall not exceed $0.42S_u$.

F-1341.2 Plastic Analysis. Where the component is evaluated on a plastic basis the following primary stress limits shall be applied.

(a) The general primary membrane stress intensity P_m shall not exceed $0.7S_u$ for ferritic steel materials included in Section II, Part D, Subpart 1, Table 2A and the greater of

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$0.7S_u$ and $S_y + \frac{1}{3}(S_u - S_y)$ for austenitic steel, high-nickel alloy, and copper-nickel alloy materials included in Section II, Part D, Subpart 1, Tables 2A and 2B.

(b) The maximum primary stress intensity at any location shall not exceed $0.90S_u$.

(c) The average primary shear across a section loaded in pure shear shall not exceed $0.42S_u$.

F-1341.3 Collapse Load. Static or equivalent static loads shall not exceed 90% of the limit analysis collapse load using a yield stress which is the lesser of $2.3S_m$ and $0.7S_u$, or 100% of the plastic analysis collapse load or test collapse load (F-1321.6).

F-1341.4 Plastic Instability Load. The plastic instability load (F-1321.7) is designated P_I and may be determined by one of the following methods:

(a) plastic analysis (NB-3213.24)

(b) experimental analysis (F-1321.8)

The applied load shall not exceed $0.7P_I$.

F-1341.5 Interaction Method. Acceptability for individual members of components may be demonstrated using the interaction method. Procedures for interaction method analysis are given in Article A-9000. The allowable stress S_{al} shall not exceed $0.7S_u$.

F-1341.6 Bearing Stresses. Except for pinned and bolted joints, bearing stresses need not be evaluated for loads for which Level D Service Limits are specified.

F-1341.7 Stress Limits for Bolts. Bolts shall be evaluated in accordance with the rules of F-1335.

F-1341.8 Requirements for Compressive Loads. Components subjected to compressive loads shall be evaluated in accordance with the rules of F-1331.5.

(15) ## F-1342 Criteria for Plate and Shell Type Supports

(a) The criteria presented in this paragraph pertain to primary stresses only. Stresses resulting from constraint of free end displacement and anchor point motion (NF-3121.12 and NF-3121.13) shall be considered as primary stresses in the evaluation. Neither peak stresses nor stresses resulting from thermal expansion within the support need be evaluated.

(b) The allowable stresses for plate and shell type supports shall be per F-1341, with material properties provided by the applicable tables of Section II, Part D per Table NF-2121(a)-1.

(c) As an alternative to the requirements of (b) above, plate and shell type supports may be qualified to Service Level D Limits using the procedure for load rating (NF-3282). The load rating for Level D Service Loadings shall be determined by the following equation:

$$\text{load rating} = TL \times 0.7\frac{S_u}{S_u^*}$$

where

TL = support test load equal to or less than the load under which the support fails to perform its specified support function

S_u = tensile strength of the support material at temperature

S_u^* = tensile strength of the support material at test temperature

but

$$\frac{S_u}{S_u^*} \le 1$$

F-1343 Criteria for Standard Supports

The rules of F-1342 or F-1344 shall be applied according to whether standard supports are plate and shell or linear type supports.

F-1344 Criteria for Linear Type Supports

The criteria presented in this paragraph pertain to primary stresses only. Stresses resulting from constraint of free end displacement and anchor point motion (NF-3121.12 and NF-3121.13) shall be considered primary stresses in the evaluation. Neither peak stresses nor stresses resulting from thermal expansion within the support need be evaluated.

Acceptability of linear type supports may be demonstrated using any one of the following methods:

(a) elastic analysis

(b) plastic analysis

(c) collapse load analysis

(d) plastic instability analysis

(e) interaction method

The primary stress limits for these alternative methods are given in F-1344.1 through F-1344.5. The other limits given in F-1344.6 and F-1344.7 shall also be satisfied as applicable.

F-1344.1 Elastic Analysis. The criteria provided in F-1334 through F-1334.5 shall be applied.

F-1344.2 Plastic Analysis. The criteria provided in F-1341.2 shall be applied. In addition, members shall be checked for local and general instability following the requirements given in F-1334.3.

F-1344.3 Collapse Load Analysis. The criteria provided in F-1334.6 shall be applied.

F-1344.4 Plastic Instability Analysis. The plastic instability load (F-1321.7) is designated P_I and may be determined by one of the following methods:

(a) plastic analysis [F-1321.4(b)];

(b) experimental analysis (F-1321.8).

The applied load shall not exceed $0.7P_I$.

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F-1344.5 Interaction Method. Acceptability for individual structural members of linear type supports may be demonstrated using interaction method analysis. Procedures for interaction method analysis are given in Article A-9000. The allowable stress S_{al} shall not exceed $0.7S_u$.

F-1344.6 Stress Limits for Bolts. Bolts shall be evaluated in accordance with the rules of F-1335.

F-1344.7 Bearing Stresses. Except for pinned and bolted joints, bearing stresses need not be evaluated for loads for which Level D Service Limits are specified.

F-1345 Requirement for Support Fillet Welds

Fillet welds for supports shall be evaluated in accordance with the rules of F-1337.

F-1400 VESSELS

The design rules given in F-1330 and F-1340 shall be used for evaluation of vessels for loads for which Service Level D Limits are specified.

F-1410 PUMPS

The design rules given in F-1330 and F-1340 shall be used for evaluation of pumps for loads for which Service Level D Limits are specified.

F-1420 VALVES

As an alternative to the procedures of F-1300, the criteria of both (a) and (b) may be used.

(a) The maximum internal pressure shall not exceed the lesser of 2.0 times the Design Pressure and the rated pressure at the temperature for which Service Level D Limits are specified.

(b) Calculate the valve crotch (Figure NB-3545.2-1, Section A-A) stress intensity S_n due to all applicable loads

$$S_n = 0.5\,Q_p + P_{es}$$

or

$$S_n = 0.5\,Q_p + nP_{eb}$$

where

n = 1.3 for ferritic steel materials in Section II, Part D, Subpart 1, Table 2A
= 1.5 for austenitic steel, high-nickel alloy, and copper-nickel alloy materials in Section II, Part D, Subpart 1, Tables 2A and 2B

P_{eb} = as defined in NB-3545.2(b)

P_{es} = primary stress in crotch region of valve body caused by piping loads for which Level D Service Limits are specified; includes combined axial, bending, and torsion

Q_p = as defined in NB-3545.2

The allowable value of this stress intensity is the lesser of $3.6S_m$ and $1.05S_u$ for materials in Section II, Part D, Subpart 1, Tables 2A and 2B, or $1.05S_u$ for materials in Section II, Part D, Subpart 1, Table 2A.

F-1430 PIPING

As an alternative to the procedures of F-1331, the criteria of NB-3656 for Class 1, or NC/ND-3655 for Class 2 and 3 piping may be used.

F-1440 CORE SUPPORT STRUCTURES (15)

The procedures of F-1300 may be used except as stipulated in (a) through (d) below.

(a) The specified dynamic or equivalent static loads shall not exceed 80% of the ultimate collapse load as obtained from test P_t, where P_t is defined as the load at which the horizontal tangent to the load deformation curve occurs, or 80% of a load combination used in the test of a prototype or model. In using this method, account shall be taken of the size effect and dimensional tolerances as well as differences which may exist in the ultimate strength or other governing material properties of the actual part and the tested parts to assure that the loads obtained from the test are a conservative representation of the load carrying capability of the actual component under postulated loading conditions for which Level D Service Limits apply.

(b) For core support structures, component inelastic analysis may be combined with elastic system analysis (F-1331). For this analysis, the maximum stress limit shall be $0.67S_u$ for primary membrane stress intensity, and shall be equal to the greater of $0.67S_{ut}$ and $S_y + \frac{1}{3}(S_{ut} - S_y)$, but not to exceed $0.9S_u$ for maximum primary stress intensity, where S_{ut} is defined as the value of ultimate stress obtained from the true stress–strain curve. In this case, the elastic system analysis shall be checked, accounting for component plastic deformation.

(c) The stress limits for high strength threaded structural fasteners with specified minimum tensile strength greater than or equal to 100 ksi (690 MPa) are as given in (1) through (3) below. S_m is as tabulated in Section II, Part D, Subpart 1, Tables 2A and 2B.

(1) For component elastic analysis, combined with either elastic or inelastic system analysis, P_m shall not exceed $2S_m$, and $P_m + P_b$ shall not exceed $3S_m$.

(2) For component plastic analysis, combined with either elastic or inelastic system analysis, P_m shall not exceed $2S_m$, and maximum primary stress intensity shall not exceed the larger of $0.67S_{ut}$ and $S_y + \frac{1}{3}(S_{ut} - S_y)$, but not to exceed $0.9S_u$ where S_{ut} is defined as the value of ultimate stress obtained from the true stress–strain curve.

(3) For component limit analysis, combined with either elastic or inelastic system analysis, P_m shall not exceed $1.33L_L$, and $P_m + P_b$ shall not exceed $1.33L_L$, where

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the shape factor is less than or equal to 1.5, with S_y equal to $1.5S_m$, where L_L is defined in Figure NG-3224-1, Note (6).

(d) The stress limits for threaded structural fasteners with specified minimum tensile strength less than 100 ksi (690 MPa) shall be determined in accordance with F-1331 or F-1341, as appropriate. The requirements of F-1331.4 and F-1341.7 do not apply.

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NONMANDATORY APPENDIX G
FRACTURE TOUGHNESS CRITERIA FOR PROTECTION AGAINST FAILURE

ARTICLE G-1000
INTRODUCTION

This Appendix presents a procedure for obtaining the allowable loadings for ferritic pressure-retaining materials in components. This procedure is based on the principles of linear elastic fracture mechanics. At each location being investigated a maximum postulated flaw is assumed. At the same location the *mode I stress intensity factor*[28] K_I is produced by each of the specified loadings as calculated and the summation of the K_I values is compared to a reference value K_{Ic} which is the highest critical value of K_I that can be ensured for the material and temperature involved. Different procedures are recommended for different components and operating conditions.

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ARTICLE G-2000
VESSELS

G-2100 GENERAL REQUIREMENTS

G-2110 REFERENCE CRITICAL STRESS INTENSITY FACTOR

(a) Figure G-2210-1 is a curve showing the relationship that can be conservatively expected between the critical, or reference, stress intensity factor K_{Ic}, $\text{ksi}\sqrt{\text{in.}}$, $\left(\text{MPa}\sqrt{\text{m}}\right)$ and a temperature which is related to the reference nil-ductility temperature RT_{NDT} determined in NB-2331. This curve is based on the lower bound of static critical K_I values measured as a function of temperature on specimens of SA-533 Type B Class 1, and SA-508 Grade 1, SA-508 Grade 2 Class 1, and SA-508 Grade 3 Class 1 steel. No available data points for static fracture toughness tests fall below the curve. An analytical approximation to the curve is:

(U.S. Customary Units)

$$K_{Ic} = 33.2 + 20.734 \exp\left[0.02\left(T - RT_{\text{NDT}}\right)\right]$$

(SI Units)

$$K_{Ic} = 36.5 + 22.783 \exp\left[0.036\left(T - RT_{\text{NDT}}\right)\right]$$

Unless higher K_{Ic} values can be justified for the particular material and circumstances being considered, Figure G-2210-1 may be used for ferritic steels which meet the requirements of NB-2331 and which have a specified minimum yield strength at room temperature of 50 ksi (350 MPa) or less.

(b) For materials which have specified minimum yield strengths at room temperature greater than 50 ksi (350 MPa) but not exceeding 90 ksi (620 MPa), Figure G-2210-1 may be used provided fracture mechanics data are obtained on at least three heats of the material on a sufficient number of specimens to cover the temperature range of interest, including the weld metal and heat-affected zone, and provided that the data are equal to or above the curve of Figure G-2210-1. These data shall be included in the Design Specification. Where these materials of higher yield strengths (specified minimum yield strength greater than 50 ksi (350 MPa) but not exceeding 90 ksi (620 MPa) are to be used in conditions where radiation may affect the material properties, the effect of radiation on the K_{Ic} curve shall be determined for the material. This information shall be included in the Design Specification.

G-2120 MAXIMUM POSTULATED DEFECTS

The postulated defects used in this recommended procedure are sharp, surface defects normal to the direction of maximum stress. For section thicknesses of 4 in. to 12 in. (100 mm to 300 mm), the postulated defects have a depth of one-fourth of the section thickness and a length of $1\frac{1}{2}$ times the section thickness. Defects are postulated at both the inside and outside surfaces. For sections greater than 12 in. (300 mm) thick, the postulated defect for the 12 in. (300 mm) section is used. For sections less than 4 in. (100 mm) thick, the 1 in. (25 mm) deep defect is conservatively postulated. Smaller defect sizes[29] may be used on an individual case basis if a smaller size of maximum postulated defect can be ensured. Due to the safety factors recommended here, the prevention of nonductile fracture is ensured for some of the most important situations even if the defects were to be about twice as large in linear dimensions as this postulated maximum defect.

G-2200 LEVEL A AND B SERVICE LIMITS

G-2210 SHELLS AND HEADS REMOTE FROM DISCONTINUITIES

G-2211 Recommendations

The assumptions of this Subarticle are recommended for shell and head regions during Level A and B Service Limits.

G-2212 Material Fracture Toughness

The reference critical stress intensity factors for material K_{Ic} values of Figure G-2210-1 are recommended.

G-2213 Maximum Postulated Defects

The recommended maximum postulated defects are described in G-2120.

G-2214 Calculated Stress Intensity Factors

G-2214.1 Membrane Tension. The K_I corresponding to membrane tension for the postulated axial defect of G-2120 is $K_{Im} = M_m \times (PR_i/t)$, where M_m for an inside axial surface flaw is given by

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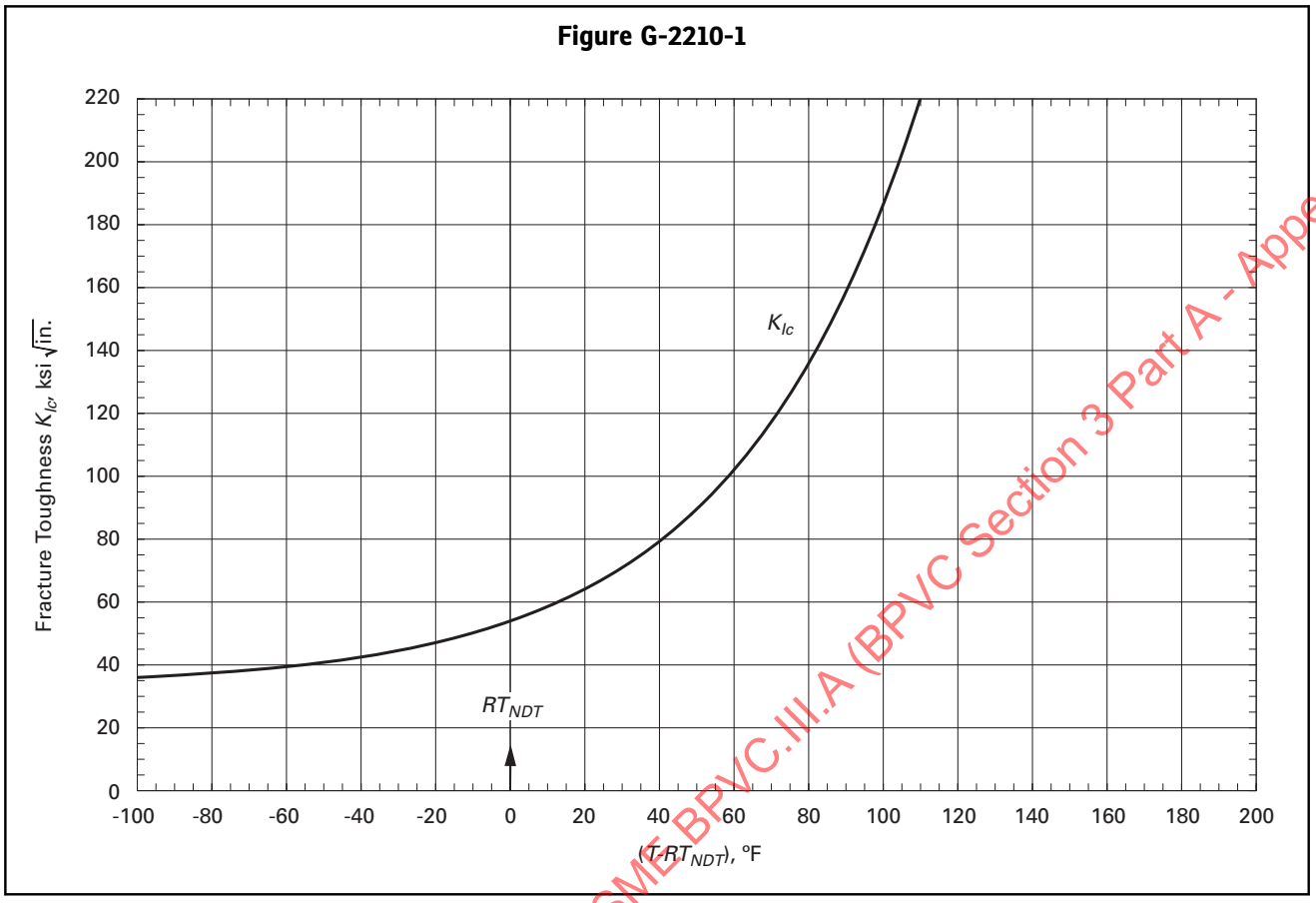
Figure G-2210-1 (15)



(U.S. Customary Units)

$M_m = 1.85$ for $t < 4$ in.
$M_m = 0.926\sqrt{t}$ for 4 in. $\le t \le 12$ in.
$M_m = 3.21$ for $t > 12$ in.

(SI Units)

$M_m = 0.296$ for $t < 102$ mm
$M_m = 0.0293\sqrt{t}$ for 102 mm $\le t \le 305$ mm
$M_m = 0.51$ for $t > 305$ mm

Similarly, M_m for an outside axial surface flaw is given by

(U.S. Customary Units)

$M_m = 1.77$ for $t < 4$ in.
$M_m = 0.893\sqrt{t}$ for 4 in. $\le t \le 12$ in.
$M_m = 3.09$ for $t > 12$ in.

(SI Units)

$M_m = 0.285$ for $t < 102$ mm
$M_m = 0.0282\sqrt{t}$ for 102 mm $\le t \le 305$ mm
$M_m = 0.493$ for $t > 305$ mm

where

p = internal pressure, ksi (MPa)
R_i = vessel inner radius, in. (mm)
t = vessel wall thickness, in. (mm)

The K_I corresponding to membrane tension for the postulated circumferential defect of G-2120 is $K_{Im} = M_m \times (pR_i/t)$, where M_m, for an inside or an outside circumferential surface defect is given by

(U.S. Customary Units)

$M_m = 0.89$ for $t < 4$ in.
$M_m = 0.443\sqrt{t}$ for 4 in. $\le t \le 12$ in.
$M_m = 1.53$ for $t > 12$ in.

(SI Units)

$M_m = 0.141$ for $t < 102$ mm
$M_m = 0.0140\sqrt{t}$ for 102 mm $\le t \le 305$ mm
$M_m = 0.245$ for $t > 305$ mm

G-2214.2 Bending Stress. The K_I corresponding to bending stress for the postulated axial or circumferential defect of G-2120 is $K_{Ib} = M_b \times$ maximum bending stress, where M_b is two-thirds of the M_m for the axial defect.

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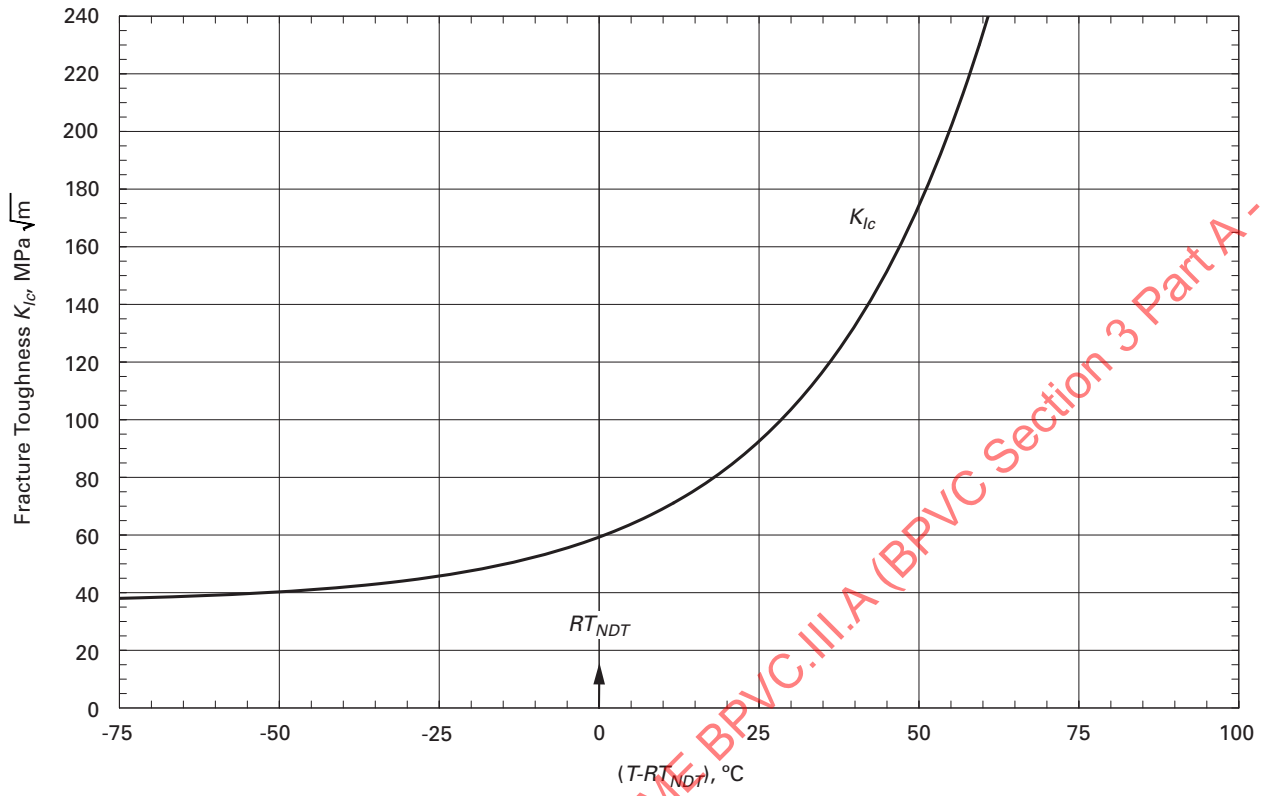
(15)

Figure G-2210-1M



G-2214.3 Radial Thermal Gradient. The maximum K_I produced by a radial thermal gradient for a postulated axial or circumferential inside surface defect of G-2120 is

(U.S. Customary Units)

$$K_{It} = 0.953 \times 10^{-3} \times CR \times t^{2.5}$$

(SI Units)

$$K_{It} = 0.579 \times 10^{-6} \times CR \times t^{2.5}$$

where CR is the cooldown rate in °F/hr (°C/h), t is the thickness in in. (mm), and K_{It} is in ksi $\sqrt{\text{in.}}$ (MPa $\sqrt{\text{m}}$) or, for a postulated axial or circumferential outside surface defect

(U.S. Customary Units)

$$K_{It} = 0.753 \times 10^{-3} \times HU \times t^{2.5}$$

(SI Units)

$$K_{It} = 0.458 \times 10^{-6} \times HU \times t^{2.5}$$

where HU is the heatup rate in °F/hr (°C/h).

The through-wall temperature difference associated with the maximum thermal K_I can be determined from Figure G-2214-1. The temperature at any radial distance from the vessel surface can be determined from Figure G-2214-2 for the maximum thermal K_I.

(a) The maximum thermal K_I and the temperature relationship in Figure G-2214-1 are applicable only for the conditions in (1) and (2).

(1) An assumed shape of the temperature gradient is approximately as shown in Figure G-2214-2.

(2) The temperature change starts from a steady state condition and has a rate, associated with startup and shutdown, less than about 100°F/hr (56°C/h). The results would be overly conservative if applied to rapid temperature changes.

(b) Alternatively, the K_I for radial thermal gradient can be calculated for any thermal stress distribution at any specified time during cooldown for a $^1/_4$-thickness axial or circumferential surface defect.

For an inside surface defect during cooldown

(U.S. Customary Units)

$$K_{It} = \left(1.0359C_0 + 0.6322C_1 + 0.4753C_2 + 0.3855C_3\right)\sqrt{\pi a}$$

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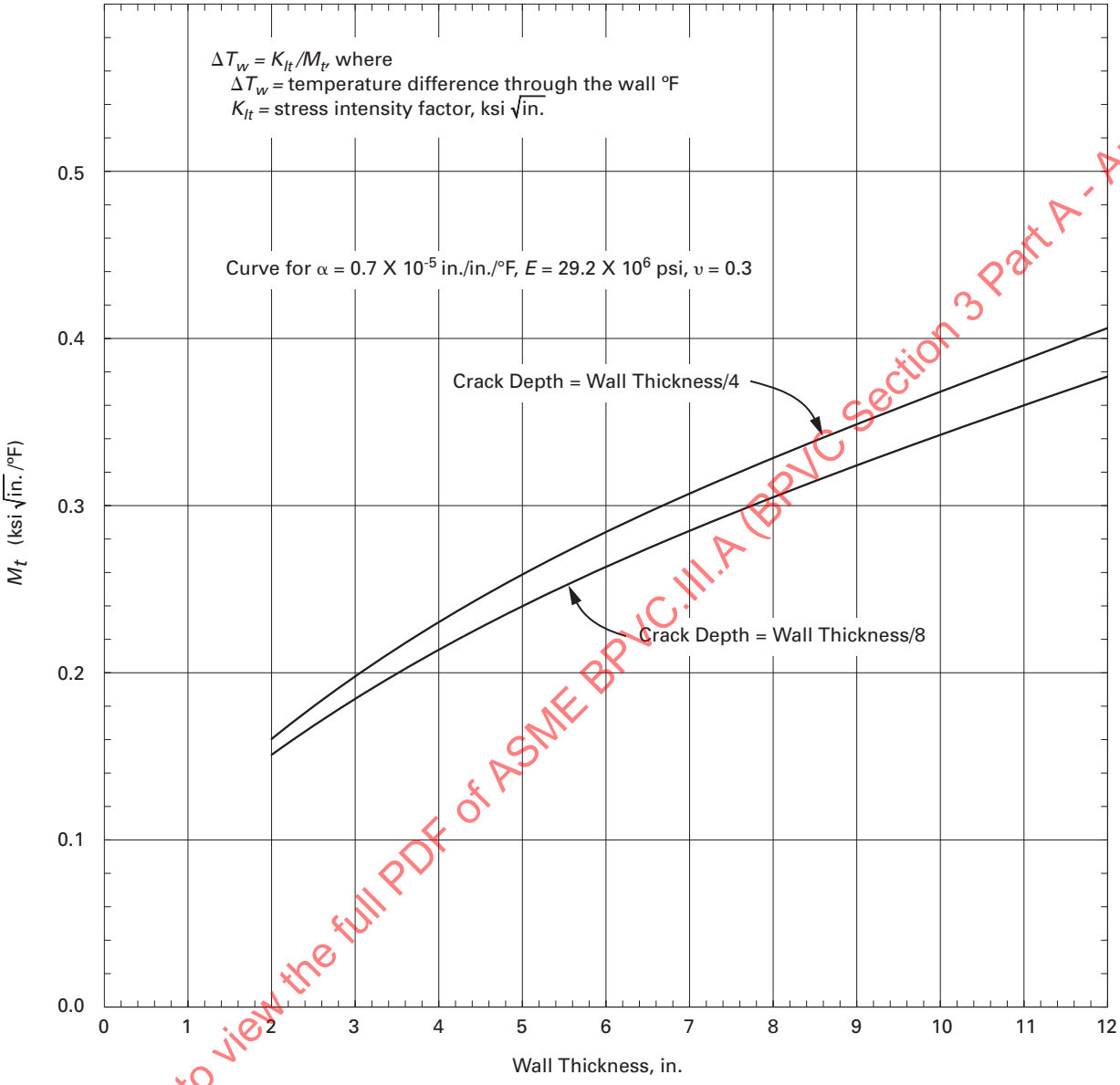
Figure G-2214-1

(15)



(SI Units)

$$K_{It} = \left(0.2259C_0 + 0.1378C_1 + 0.1036C_2 + 0.08405C_3\right)\sqrt{\pi a}$$

For an outside surface defect during heatup

(U.S. Customary Units)

$$K_{It} = \left(1.043C_0 + 0.630C_1 + 0.481C_2 + 0.401C_3\right)\sqrt{\pi a}$$

(SI Units)

$$K_{It} = \left(0.227C_0 + 0.137C_1 + 0.105C_2 + 0.0874C_3\right)\sqrt{\pi a}$$

The coefficients C_0, C_1, C_2, and C_3 are determined from the thermal stress distribution at any specified time during the heatup or cooldown using

$$\sigma(x) = C_0 + C_1(x/a) + C_2(x/a)^2 + C_3(x/a)^3$$

where x is a dummy variable that represents the radial distance, in. (mm), from the appropriate (i.e., inside or outside) surface and a is the maximum crack depth, in. (mm).

(c) For the startup condition, the allowable pressure vs. temperature relationship is the minimum pressure at any temperature, determined from the following:

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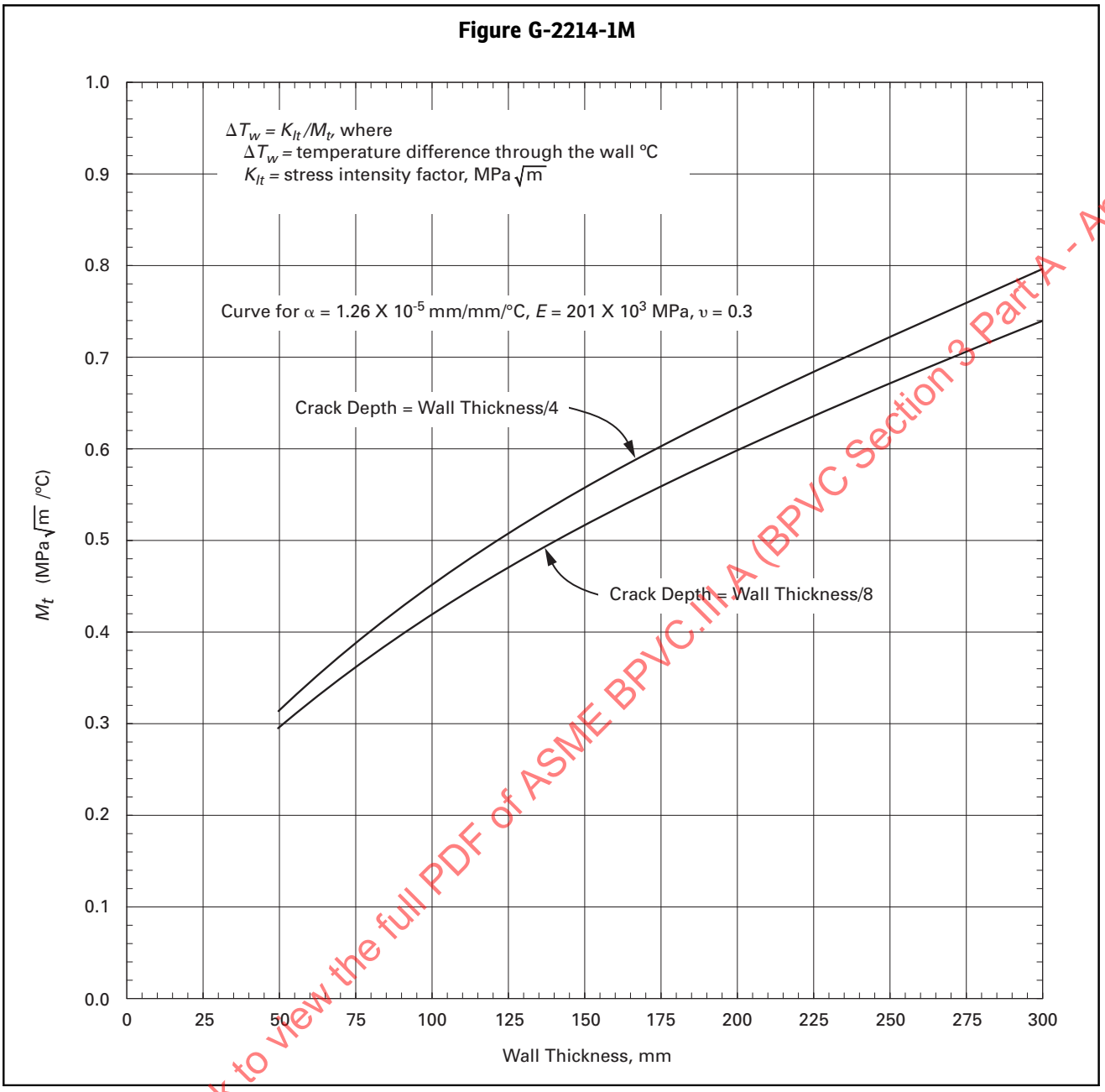
(15)

Figure G-2214-1M



(1) the calculated steady state results for the ¼-thickness inside surface defect,

(2) the calculated steady state results for the ¼-thickness outside surface defect, and

(3) the calculated results for the maximum allowable heatup rate using a ¼-thickness outside surface defect.

(15) **G-2215 Allowable Pressure**

The equations given in this Subarticle provide the basis for determination of the allowable pressure at any temperature at the depth of the postulated defect during Service Conditions for which Level A and B Service Limits are specified. In addition to the conservatism of these assumptions, it is recommended that a factor of 2 be applied to the calculated K_I values produced by primary stresses. In shell and head regions remote from discontinuities, the only significant loadings are: (1) general primary membrane stress due to pressure; and (2) thermal stress due to thermal gradient through the thickness during startup and shutdown. Therefore, the requirement to be satisfied and from which the allowable pressure for any assumed rate of temperature change can be determined is:

$$2K_{Im} + K_{It} < K_{Ic} \quad (1)$$

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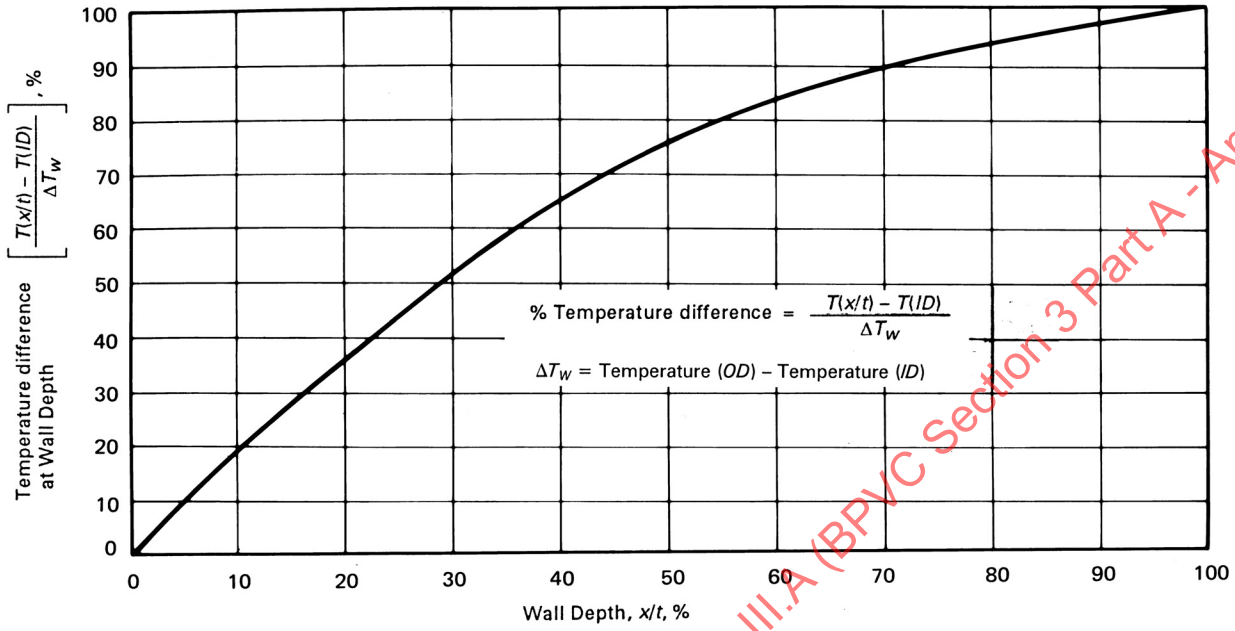
Figure G-2214-2



throughout the life of the component at each temperature with K_{Im} from G-2214.1, K_{It} from G-2214.3, and K_{Ic} from Figure G-2210-1.

The allowable pressure at any temperature shall be determined as follows.

(a) For the startup condition,

(1) consider postulated defects in accordance with G-2120;

(2) perform calculations for thermal stress intensity factors due to the specified range of heat-up rates from G-2214.3;

(3) calculate the K_{Ic} toughness for all vessel beltline materials from G-2212 using temperatures and RT_{NDT} values for the corresponding locations of interest; and

(4) calculate the pressure as a function of coolant inlet temperature for each material and location. The allowable pressure–temperature relationship is the minimum pressure at any temperature determined from

(-a) the calculated steady-state (K_{It} = 0) results for the $^1/_4$-thickness inside surface postulated defects using the equation

$$P = \frac{K_{Ic}}{2M_m}\left(t / R_i\right)$$

(-b) the calculated results from all vessel beltline materials for the heatup stress intensity factors using the corresponding $^1/_4$-thickness outside-surface postulated defects and the equation

$$P = \frac{K_{Ic} - K_{It}}{2M_m}\left(t / R_i\right)$$

(b) For the cooldown condition,

(1) consider postulated defects in accordance with G-2120;

(2) perform calculations for thermal stress intensity factors due to the specified range of cooldown rates from G-2214.3;

(3) calculate the K_{Ic} toughness for all vessel beltline materials from G-2212 using temperatures and RT_{NDT} values for the corresponding location of interest; and

(4) calculate the pressure as a function of coolant inlet temperature for each material and location using the equation

$$P = \frac{K_{Ic} - K_{It}}{2M_m}\left(t / R_i\right)$$

The allowable pressure–temperature relationship is the minimum pressure at any temperature, determined from all vessel beltline materials for the cooldown stress intensity factors using the corresponding $^1/_4$-thickness inside-surface postulated defects.

Those plants having low temperature overpressure protection (LTOP) systems can use the following load and temperature conditions to provide protection against failure during reactor start-up and shutdown operation due to low temperature overpressure events that have been classified as Service Level A or B events. LTOP systems shall be effective at coolant temperatures less than

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200°F (95°C) or at coolant temperatures corresponding to a reactor vessel metal temperature less than RT_{NDT} + 50°F (28°C), whichever is greater.[30, 31] LTOP systems shall limit the maximum pressure in the vessel to 100% of the pressure determined to satisfy eq. (1).

G-2220 NOZZLES, FLANGES, AND SHELL REGIONS NEAR GEOMETRIC DISCONTINUITIES

G-2221 General Requirements

The same general procedure as was used for the shell and head regions in G-2210 may be used for areas where more complicated stress distributions occur, but certain modifications of the procedures for determining allowable applied loads shall be followed in order to meet special situations, as stipulated in G-2222 and G-2223.

G-2222 Consideration of Membrane and Bending Stresses

(a) Equation G-2215(1) requires modification to include the bending stresses which may be important contributors to the calculated K_I value at a point near a flange or nozzle. The terms whose sum must be < K_{Ic} for Level A and B Service Limits are:

(1) $2K_{Im}$ from G-2214.1 for primary membrane stress;

(2) $2K_{Ib}$ from G-2214.2 for primary bending stress;

(3) K_{Im} from G-2214.1 for secondary membrane stress;

(4) K_{Ib} from G-2214.2 for secondary bending stress.

(b) For purposes of this evaluation, stresses which result from bolt preloading shall be considered as primary.

(c) It is recommended that when the flange and adjacent shell region are stressed by the full intended bolt preload and by pressure not exceeding 20% of the preoperational system hydrostatic test pressure, minimum metal temperature in the stressed region should be at least the initial RT_{NDT} temperature for the material in the stressed regions plus any effects of irradiation at the stressed regions.

(d) Thermal stresses shall be considered as secondary except as provided in NB-3213.13(b). The K_I of G-2214.3(b) is recommended for the evaluation of thermal stress.

G-2223 Toughness Requirements for Nozzles

(a) A quantitative evaluation of the fracture toughness requirements for nozzles is not feasible at this time, but preliminary data indicate that the design defect size for nozzles, considering the combined effects of internal pressure, external loading and thermal stresses, may be a fraction of that postulated for the vessel shell. Nondestructive examination methods shall be sufficiently reliable and sensitive to detect these smaller defects.

(b) WRCB 175 provides an approximate method in paragraph 5C(2) for analyzing the inside corner of a nozzle and cylindrical shell for elastic stresses due to internal pressure stress.

(c) Fracture toughness analysis to demonstrate protection against nonductile failure is not required for portions of nozzles and appurtenances having a thickness of 2.5 in. (63 mm) or less, provided the lowest service temperature is not lower than RT_{NDT} plus 60°F (33°C).

G-2300 LEVEL C AND D SERVICE LIMITS

G-2310 RECOMMENDATIONS

The possible combinations of loadings, defect sizes, and material properties which may be encountered during Level C and D Service Limits are too diverse to allow the application of definitive rules, and it is recommended that each situation be studied on an individual case basis. The principles given in this Appendix may be applied, where applicable, with any postulated loadings, defect sizes, and material toughness which can be justified for the situation involved.

G-2400 HYDROSTATIC TEST TEMPERATURE

(a) For system and component hydrostatic tests performed prior to loading fuel in the reactor vessel, it is recommended that hydrostatic tests be performed at a temperature not lower than RT_{NDT} plus 60°F (33°C). The 60°F (33°C) margin is intended to provide protection against nonductile failure at the test pressure.

(b) For system and component hydrostatic tests performed subsequent to loading fuel in the reactor vessel, the minimum test temperature should be determined by evaluating K_I. The terms given in (1) through (4) should be summed in determining K_I:

(1) $1.5K_{Im}$ from G-2214.1 for primary membrane stress

(2) $1.5K_{Ib}$ from G-2214.2 for primary bending stress

(3) K_{Im} from G-2214.1 for secondary membrane stress

(4) K_{Ib} from G-2214.2 for secondary bending stress

K_I, calculated by summing the four values given in (1) through (4), shall not exceed the applicable K_{Ic} value.

(c) The system hydrostatic test to satisfy (a) or (b) should be performed at a temperature not lower than the highest required temperature for any component in the system.

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ARTICLE G-3000
PIPING, PUMPS, AND VALVES

G-3100 GENERAL REQUIREMENTS

In the case of the materials other than bolting used for piping, pumps, and valves for which impact tests are required (NB-2311), the tests and acceptance standards of this Section are considered to be adequate to prevent nonductile failure under the loadings and with the defect sizes encountered under Level A and B Service Limits and testing conditions. Level C and D Service Limits should be evaluated on an individual case basis (G-2300).

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ARTICLE G-4000
BOLTING

G-4100 GENERAL REQUIREMENTS

In the case of bolting materials for which impact tests are required, the tests and acceptance standards of this Section are considered to be adequate to prevent nonductile failure under the loadings and with the defect sizes encountered under Level A and B Service Limits and testing conditions. Level C and D Service Limits should be evaluated on an individual case basis (G-2300). Welding Research Council Bulletin 175 (WRCB 175), "PVRC Recommendations on Toughness Requirements for Ferritic Materials," provides procedures in paragraph 7 for evaluating various defect sizes and associated toughness levels in bolting materials.

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NONMANDATORY APPENDIX L
CLASS FF FLANGE DESIGN FOR CLASS 2 AND 3 COMPONENTS AND CLASS MC VESSELS

ARTICLE L-1000
CLASS FF FLANGES — INTRODUCTION

L-1100 GENERAL REQUIREMENTS

L-1110 ACCEPTABILITY

The requirements for acceptability of Class FF flange design are that the general design requirements of Subsection NC, ND, or NE and the specific requirements of this Appendix are met.

L-1120 DEFINITION OF CLASS FF FLANGES

Class FF flanges are circular flanges having flat faces which are either bolted directly together or are separated by a metal spacer such that there is metal to metal contact between the flange faces and the metal spacer initially or after the flanges have been bolted up.

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ARTICLE L-2000
CLASS FF FLANGES — MATERIALS

L-2100 MATERIAL REQUIREMENTS

The rules of Article XI-2000 apply.

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ARTICLE L-3000
CLASS FF FLANGES — DESIGN

L-3100 GENERAL REQUIREMENTS

L-3110 SCOPE

The rules in this Appendix apply to circular, bolted flanged connections where the assemblage is comprised of identical or nonidentical flange pairs, and where the flanges are flat faced and are in uniform metal-to-metal contact across their entire face during assembly before the bolts are tightened or after a small amount of preload is applied to compress a gasket. The rules also apply when a pair of identical flat faced flanges is separated by a metal spacer. The rules are not intended for cases where the faces are intentionally made nonparallel to each other such that initial contact is at the bore.

Construction details for attachment and configuration of the flange are not covered in this Appendix. Minimum weld sizes and geometric limitations given in Figure XI-3120-1 and Figure NC-4243-2, Figure ND-4243-1, or Figure NE-4243-1, as applicable, apply to Nonmandatory Appendix L flanges. Similarly, when applying the rules of this Appendix, use of the graphs in Mandatory Appendix XI for obtaining applicable design parameters is necessary; namely, Figures XI-3240-1 through XI-3240-6.

L-3120 ASSUMPTIONS AND LIMITATIONS OF RULES

It is assumed that a self-sealing gasket is used approximately in-line with the wall of attached pipe or vessel. The rules provide for hydrostatic end loads only and assume that the gasket seating loads are small and may in most cases be neglected. It is also assumed that the seal generates a negligible axial load under operating conditions. If such is not the case, allowance shall be made for a gasket load H_G dependent on the size and configuration of the seal and design pressure. Proper allowance shall be made if connections are subject to external forces other than external pressure.

L-3130 FATIGUE CONSIDERATIONS

As with flanges with ring type gaskets, the stress in the bolts may vary appreciably with pressure. There is an additional bolt stress generated due to a prying effect resulting from the flanges interacting beyond the bolt circle. As a result, fatigue of the bolts and other parts comprising the flanged connection may require consideration and adequate pretensioning of the bolts may be necessary. It is important to note that the operating bolt stress is relatively insensitive to changes in prestress up to a certain point and that thereafter the two stresses are essentially the same. This is a desirable characteristic of Nonmandatory Appendix L flanges; it means that if the assembly stress (prestress) in the bolts is close to the operating design stress σ_b, then subsequent applications of pressure loadings ranging from zero to full load will have no significant effect on the actual operating stress in the bolts.

Unlike Mandatory Appendix XI flanges and their bolts which are stressed during assembly (although some readjustment in the stresses may occur during pressurization), Nonmandatory Appendix L flanges become stressed during pressurization; however, the effect of pressurization on the operating stress in the bolts depends upon the extent to which the bolts are stressed during assembly.

L-3140 ANALYTICAL APPROACHES

In the case of identical flange pairs, the analytical procedure described in this Appendix considers the flanges to be continuous, annular plates whose flexural characteristics can be approximated by beam theory by considering the flanges to be comprised of a series of discrete, radial beams. For nonidentical flange pairs, beam theory is supplemented by the theory of rigid body rotation so as to preserve equilibrium of moments and forces. Moments associated with beam theory are designated as *balanced moments*, whereas moments used when the theory of rigid body rotations is applied are designated as *unbalanced moments*. Balanced and unbalanced moments are designated M_b and M_u, respectively. When no subscript appears, a balanced moment is intended, i.e., in the equations for the analysis of identical flange pairs (L-3242).

L-3150 REDUCTION IN CONTACT FORCES

A reduction in flange-to-flange contact forces beyond the bolt circle occurs when the flanges are stiff with respect to the bolting and, in the extreme, flange separation occurs. The rules in this Appendix provide little insight into the problem except when the reduction in the contact force is due to the flange–hub interaction moment. The problem is considered to be of little practical significance when the nuts are tightened during assembly using ordinary wrenching techniques.

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L-3160 TANGENTIAL CONTACT BETWEEN FLANGES OUTSIDE THE BOLT CIRCLE

The design procedure is based on the assumption that the flanges are in tangential contact at their outside diameter or at some lesser distance h_C from the bolt circle. [See L-3221(b) and L-3260 when $h_C < h_{C\max}$ for additional requirements.] The diameter of the circle where the flanges are in tangential contact is a design variable; the smaller the diameter of the contact circle $C + 2h_C$, the greater the required prestress in the bolts, the higher the ratio of prestress to operating bolt stress, S_i/σ_b, and the smaller the flange separation at the gasket. The requirement of tangential contact, even when it is assumed to occur at the outside diameter ($C + 2h_{C\max}$) of the flanges, automatically yields a high ratio of S_i/σ_b which means that the possibility of flange separation or an appreciable decrease in the flange-to-flange contact forces is no longer a problem even when the flanges are stiff with respect to the bolts.

L-3170 RELATIVE STIFFNESS OF FLANGES AND BOLTS

The equation for the calculated strain length l of the bolts is generally applicable. However, variations in the thickness of material actually clamped by each bolt, such as sleeves, collars, or multiple washers placed between a flange and the bolt heads or nuts, or by counterboring, must be considered in establishing a value of l for use in the design equations. A large increase in l may cause the flanges to become abnormally stiff with respect to such bolts and the provision of tangential contact may not yield a sufficiently high value of the ratio S_i/σ_b unless h_C is reduced to cause an increase in the ratio.

L-3180 COMBINED STRESSES

Most of the calculated stresses are bending only, so that tensile and compressive stresses of the same magnitude occur on opposite surfaces at the point under consideration. However, when a membrane stress occurs in conjunction with a bending stress, the combined stress represents the maximum absolute value at the point and may be tension or compression [denoted by a minus (−) sign].

L-3190 NOTATION

L-3191 Symbols

The symbols described below are used in the equations for the design of flanges.

A = outside diameter of flange

a = shape factor
= $(A + C)/2B_1$

A_b = cross-sectional area of the bolts using the root diameter of the thread or least diameter of unthreaded portion, if less

A_m = total required cross-sectional area of bolts, taken as the greater of A_{m1} and A_{m2}

A_{m1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions
= W_{m1}/S_b

A_{m2} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating, sq in. (mm^2)
= W_{m2}/S_a

$\overline{AR}$ = bolt hole aspect ratio used in calculating bolt hole flexibility factor r_B
= $\dfrac{nD}{\pi C}$

B = inside diameter of flange. When B is less than $20g_1$, it will be optional for the designer to substitute B_1 for B in the formula for longitudinal stress S_H.

b = effective gasket or joint-contact-surface seating width[2]

b_0 = basic gasket seating width (from Table XI-3221.1-2)

B_1 = $B + g_1$ for loose type flanges and for integral type flanges that have calculated values h/h_0 and g_1/g_0 which would indicate an f value of less than 1.0, although the minimum value of f permitted is 1.0
= $B + g_0$ for integral type flanges when f is equal to or greater than one
= B for Category 3 (loose type) flanges

C = bolt circle diameter

c = basic dimension used for the minimum sizing of welds equal to t_n or t_i, whichever is less(1)

C_1 = factor
$$= -\left(0.748 - 1.567J_S \log \frac{A}{B_1}\right) \div (1 + 1.3J_S) \quad (1)$$

C_2 = factor
$$= \left[\frac{\pi}{32}\left(PB_1{}^3\right) - 1.3J_P M_P\right] \div \left(1 + 1.3J_S\right) \quad (2)$$

C_3 = factor[32]
$$= -\left(0.575 - 1.206J_S \log \frac{A}{B_1}\right) \div \left(J_S + t_1{}^3 / F_1'\right) \quad (3)$$

C_4 = factor[32]
$$= -\left(J_P M_P\right) \div \left(J_S + t_1{}^3 / F_1'\right) \quad (4)$$

D = diameter of bolt hole

d = factor
= $\dfrac{U}{V_L} h_0 g_0{}^2$ for loose type flanges
= $\dfrac{U}{V} h_0 g_0{}^2$ for integral type flanges

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d_b = nominal diameter of bolt

E = modulus of elasticity of flange material, corrected for operating temperature (see Section II, Part D, Subpart 2, Tables TM)

e = factor

$= \frac{F}{h_0}$ for integral type flanges

$= \frac{F_L}{h_0}$ for loose type flanges

$E_I{*}$ = factor

$= E_I t_I{}^3$

$E_{II}{*}$ = factor

$= E_{II} t_{II}{}^3$

F = factor for integral type flanges (from Figure XI-3240-2)

f = hub stress correction factor for integral flanges from Figure XI-3240-6. (When greater than 1, this is the ratio of the stress in the small end of hub to the stress in the large end.) (For values below limit of the figure, use $f = 1$.)

F_L = factor for loose type flanges (from Figure XI-3240-4)

$F' = g_0{}^2(h_0 + Ft) / V$ for Category 1, Group 1 assembly (5a)

$= g_0{}^2(h_0 + F_L t) / V_L$ for Category 2, Group 1 assembly (5b)

$= 0$ for Category 3, Group 1 assembly (5c)

$F_I{}' = g_0{}^2(h_0 + Ft_I) / V$ for Category 1, Group 3 assembly (6a)

$= g_0{}^2(h_0 + F_L t_I) / V$ for Category 2, Group 3 assembly (6b)

$= 0$ for Category 3, Group 3 assembly (6c)

G = diameter at location of gasket load reaction

= mean diameter of gasket

g_0 = thickness of hub at small end

g_1 = thickness of hub at back of flange

H = total hydrostatic end force

$= 0.785\, G^2 P$

h = hub length

h_0 = factor

$= \sqrt{B g_0}$

H_C = contact force between mating flanges

h_C = radial distance from bolt circle to flange-spacer or flange–flange bearing circle where tangential contact occurs. Tangential contact exists from the selected value of h_C to $h_{C\max}$

$h_{C\max}$ = radial distance from bolt circle to outer edge of flange or spacer, whichever is less

H_D = hydrostatic end force on area inside of flange

$= 0.785\, B^2 P$

h_D = radial distance from the bolt circle, to the circle on which H_D acts, as prescribed in Table XI-3230-1

H_G = gasket load due to seating pressure, plus axial force generated by self-sealing of gasket

h_G = radial distance from gasket load reaction to the bolt circle

$= \frac{C - G}{2}$

H_p = total joint-contact-surface compression load

$= 2b \times 3.14\, GmP$

H_T = difference between total hydrostatic end force and the hydrostatic end force on area inside of flange

$= H - H_D$

h_T = radial distance from the bolt circle to the circle on which H_T acts as prescribed in Table XI-3230-1

$J_P = \frac{1}{B_1}\left[\frac{h_D}{\beta} + \frac{h_C}{a}\right] + \pi r_B$

$J_S = \frac{1}{B_1}\left[\frac{2h_D}{\beta} + \frac{h_C}{a}\right] + \pi r_B$

K = ratio of outside diameter of flange to inside diameter of flange

$= A/B$

L = factor

$= \frac{te + 1}{T} + \frac{t^3}{d}$

l = calculated strain length of bolt

$= 2t + t_s + (\frac{1}{2} d_b$ for each threaded end for a Group 1 assembly)

$= t_I + t_{II} + (\frac{1}{2} d_b$ for each threaded end for a Group 3 assembly)

m = gasket factor; obtain from Table XI-3221.1-1 (see XI-3221.1)

M_b = balanced moment acting at diameter B_1 of flange

M_D = component of moment due to H_D

$= H_D h_D$

M_G = component of moment due to H_G

$= H_G h_G$

M_H = moment acting on end of hub, pipe, or shell, at its junction with back face of flange ring

M_P = moment due to H_D, H_T, H_G

$= H_D h_D + H_T h_T + H_G h_G$

M_S = total moment on flange ring due to continuity with hub, pipe, or shell

$= M_H + Qt/2$ where

M_T = component of moment due to H_T

$= H_T h_T$

M_u = unbalanced moment acting at diameter B_1 of flange

N = width used to determine the basic gasket seating with b_0, based upon the possible contact width of the gasket (see Table XI-3221.1-2)

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n = number of bolts
P = design pressure
Q = shear force between flange ring and end of hub, pipe, or shell, positive as indicated in Figure L-3191-2 sketch (b)
R = radial distance from bolt circle to point of intersection of hub and back of flange. For integral and hub flanges,

$$= \frac{C - B}{2} - g_1$$

$$r_B = \frac{1}{n}\left(\frac{4}{\sqrt{1-\overline{AR}^2}}\tan^{-1}\sqrt{\frac{1+\overline{AR}}{1-\overline{AR}}} - \pi - 2\overline{AR}\right)$$

= (see Figure L-3191-1 for a curve of nr_B vs $\overline{AR}$. In the above equation for r_B, $\tan^{-1}$ must be expressed in radians.)
r_E = elasticity factor
= modulus of elasticity of flange material divided by modulus of elasticity of bolting material, corrected for operating temperature (see Section II, Part D, Subpart 2, Tables TM)
r_S = initial bolt stress factor
$= 1 - S_i/\sigma_b$
S_a = allowable bolt stress at atmospheric temperature (given in Section II, Part D, Subpart 1, Table 3)
S_b = allowable bolt stress at design temperature (given in Section II, Part D, Subpart 1, Table 3)
S_f = allowable design stress for material of flange at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (given in Section II, Part D, Subpart 1, Tables 1A and 1B)
S_H = calculated longitudinal stress in hub
S_i = initial bolt stress (always less than S_b)
S_n = allowable design stress for material of nozzle neck, vessel or pipe wall, at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (given in Section II, Part D, Subpart 1, Tables 1A and 1B)
S_R = calculated radial stress in flange
S_T = calculated tangential stress in flange
T = factor involving K (from Figure XI-3240-1)
t = thickness of the flange under consideration (t, t_I, or t_II, as applicable)
= flange thickness of an identical flange pair in a Group 1 assembly
t_I = thickness of the nonreducing flange in a Group 3 assembly (see L-3231)
t_II = thickness of the reducer or flat circular head in a Group 3 assembly (see L-3231)
t_i = two times the thickness g_0, when the design is calculated as an integral flange or two times the thickness of shell or nozzle wall required for internal pressure, when the design is calculated as a loose flange, but not less than $^1/_4$ in. (6 mm)
t_n = nominal thickness of shell or nozzle wall to which flange or lap is attached
t_s = thickness of spacer
U = factor involving K (from Figure XI-3240-1)
V = factor for integral type flanges (from Figure XI-3240-3)
V_L = factor for loose type flanges (from Figure XI-3240-5)
W = flange design bolt load, for the operating conditions or gasket seating, as may apply, lb (kN) (see L-3220)
w = width used to determine the basic gasket seating width b_0, based upon the contact width between the flange facing and the gasket (see Table XI-3221.1-2)
W_{m1} = minimum required bolt load for the operating conditions (see L-3220)
X = factor
$= E_\mathrm{I}^*/(E_\mathrm{I}^* + E_\mathrm{II}^*)$
Y = factor involving K (from Figure XI-3240-1)
y = gasket or joint-contact-surface unit seating load[2]
Z = factor involving K (from Figure XI-3240-1)
β = shape factor for full face metal-to-metal contact flanges
$= (C + B_1)/2B_1$
θ_A = slope of flange face at outside diameter, rad
θ_B = slope of flange face at inside diameter, rad
θ_{rb} = change in slope which flange pair undergoes due to an unbalanced moment, rad

L-3192 Subscripts

Subscripts I and II where noted are used to distinguish between the flanges in a nonidentical flange pair (Group 2 or 3 assemblies). B_1 without a subscript always refers to Flange I (the nonreducing flange) in a Group 2 or 3 assembly.

L-3193 Based on Nonreducing Flange

Unless otherwise noted, B_1, J_S, J_P, and F_I' [equations L-3191(6a), L-3191(6b), and L-3191(6c) and M_P are based on the dimensions of the nonreducing flange (Flange I) in a Group 2 or 3 assembly.

L-3194 Logarithms

All logarithms are to base 10.

L-3200 DESIGN OF FLANGES AND BOLTING

L-3210 ESTIMATING FLANGE THICKNESSES AND BOLTING

L-3211 Equations for Trial Flange Thickness and Bolting

The following simple equations are offered for calculating approximate values of t, t_1, t_II, and A_b before applying the rules in L-3220 through L-3260. The equations are

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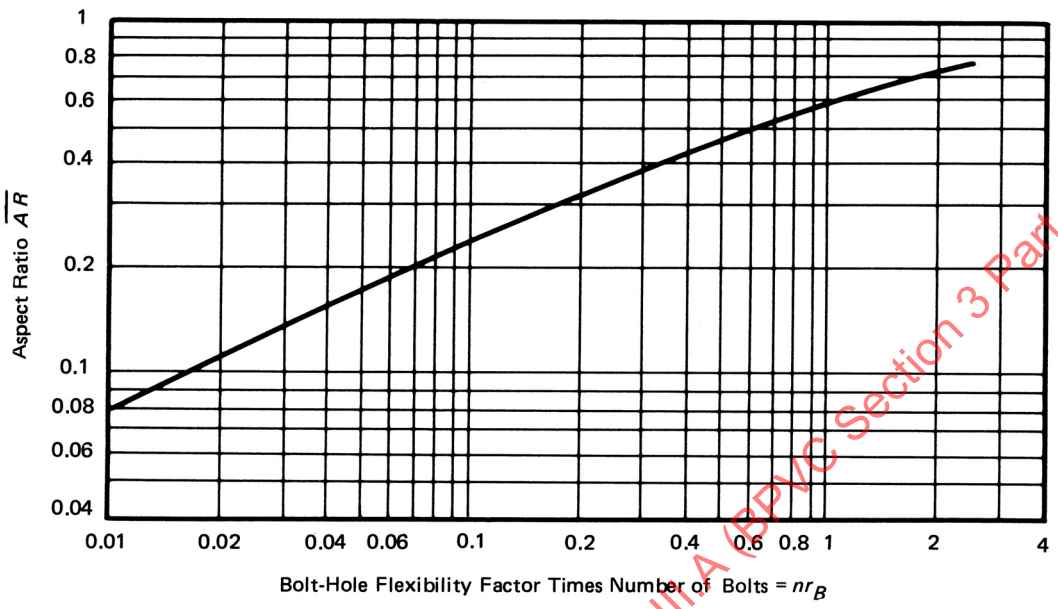
Figure L-3191-1
Bolt Hole Flexibility Factor



not intended to replace the rules; however, they should significantly reduce the amount of work required to achieve a suitable design. Since the flanges are in metal-to-metal contact and interact, the stresses in one flange are influenced by the stiffness of the mating flange and theoretically an unlimited number of designs can be found which satisfy the rules. In practice, however, economics, engineering judgment, and dimensional constraints will show which is the "best" design. It should be noted that the equations in Table L-3212-1 assume that both flanges comprising an assembly have essentially the same modulus of elasticity and allowable stress.

$$t_a = 2.45\sqrt{\frac{M_p}{(\pi C - nD)S_f}} \tag{7}$$

$$t_b = 0.56B_1\sqrt{P/S_f} \tag{8}$$

$$t_c = \text{greater of } t_a \text{ or } t_b$$
$$t_d = t_a + \frac{(B_1 - B_{II})}{(B_1 - 0.5C)}(t_b - t_a) \tag{9}$$

$$t_e = t_a \text{ when } t_b < t_a$$
$$t_e = t_d \text{ when } t_b > t_a \tag{10}$$
$$A_b' = \left[H + 2M_p/(A - C)\right] \div S_b$$

$$t_f = 2.45\sqrt{\frac{H_1 l_1 + H_2 l_2}{(\pi C - nD)S_f}} \tag{11}$$

$$A_b{}^* = 0.95\left[\frac{2(H_1 l_1 + H_2 l_2)}{(A - C)} + 0.785G^2P\right] \div S_b \tag{12}$$

where

$H_1 = 0.785B_{II}{}^2P$
$H_2 = 0.785\ (G^2 - B_{II}{}^2)P$
$l_1 = (C - B_{II})/2$
$l_2 = (C - G)/2 + (G - B_{II})/4$
$t_g =$ smaller of t_c or t_f

L-3212 Trial Values of t, t_I, t_{II}, and A_b

The simple equations given in Table L-3212-1 should yield relatively good trial values of t, t_I, t_{II}, and A_b but they do not assure that the "first trial design" will meet the requirements of L-3240 through L-3250. As a result, it becomes necessary to select new trial values and reanalyze. In order to assist the designer in selecting the second trial values, the following comments concerning the behavior of different groups of Class FF flanges are offered.

(a) The hub of a Category 1 or 2 flange of a Group 1 assembly reduces the radial stress at the bolt circle (due to a negative hub–flange interaction moment) and the longitudinal hub stress. As a result, a pair of Category 1 or Category 2 flanges will be thinner than a pair of identical Category 3 flanges.

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Figure L-3191-2
Flange Dimensions and Forces

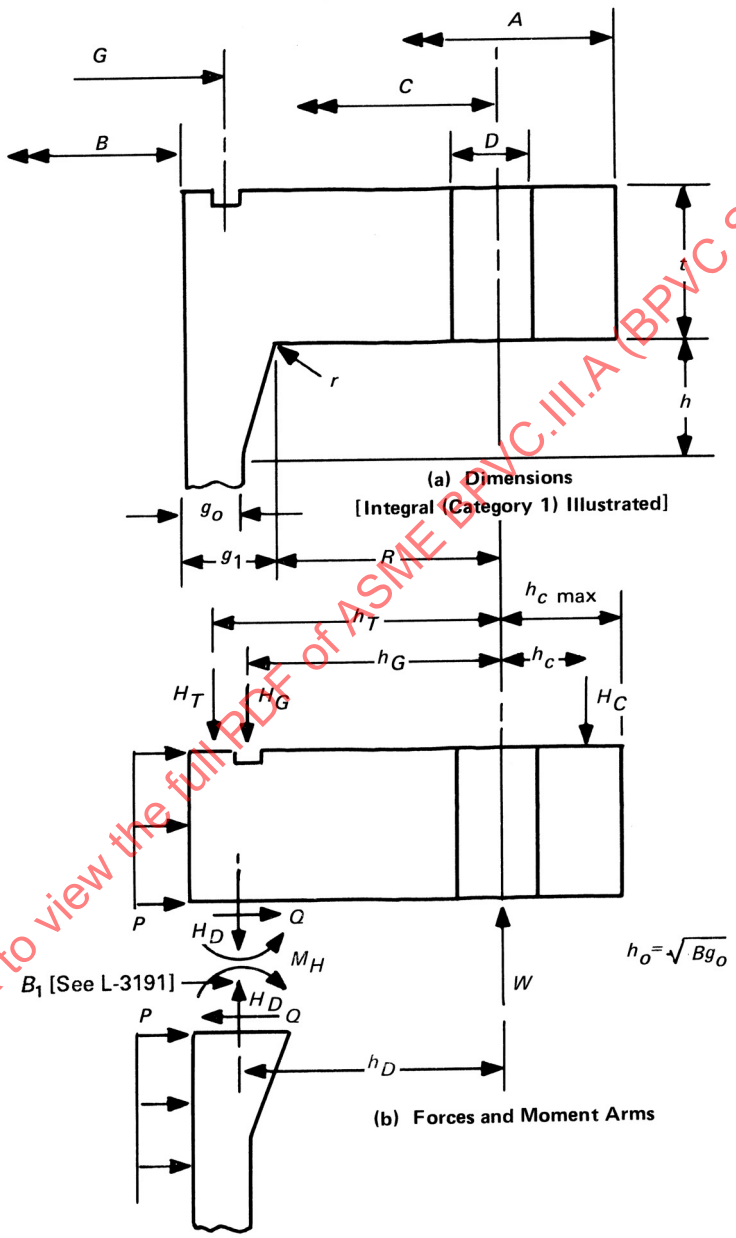

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Table L-3212-1
Trial Flange Thickness and Area of Bolting for Various Groups of Assemblies and Flange Categories

Group (Assembly)	Category of Flanges		Suggested Trial Values		
	Nonreducing	Reducing	t or t_I	t_{II}	A_b
1	1 or 2	...	$0.9t_a$	...	$0.9A_b'$
	3	...	t_a	...	A_b'
2	1 or 2	1 or 2	t_a	t_e	A_b'
	3	3	$1.1t_a$	$1.1t_c$	$1.1A_b'$
	3	1 or 2	t_a	t_c	A_b'
	1 or 2	3	$1.1t_g$	$1.1t_g$	A_b'*
3	1, 2, or 3	...	$1.1t_a$	$1.1t_c$	$1.05A_b'$

(b) Increasing the thickness of the reducing flange of a Group 3 assembly, when the nonreducing flange is Categories 1 and 2, generally reduces the significant stresses in both flanges comprising the assembly. When the stress in Flange I (nonreducing) is excessive, increasing t_I will generally be more effective in reducing the stresses; however, a nominal increase of the stresses in Flange II will occur due to the additional restraint provided by increasing t_I. When the stress in Flange I is excessive and only marginally acceptable in Flange II, both t_I and t_{II} should be increased with the emphasis placed on t_I.

(c) A Category 3 reducing flange bolted to a Category 1 or 2 nonreducing flange produces a large overturning moment which tends to rotate Flange I in a negative direction. As a result, the radial stress at the bolt circle in Flange I will often be excessive due to a large, positive hub–flange interaction moment. As a result, it is usually necessary to increase t_I so that $t_I = t_{II}$. The same problem does not occur when Flange I is Category 3 since there exists no hub–flange interaction moment. When Flange I is an optional type treated as a loose type (Category 3), a hub–flange interaction moment actually exists but is disregarded in the analysis by assigning the flange to Category 3.

(d) When the longitudinal hub stress of a Category 1 or 2 flange is excessive, it can be reduced by increasing the size of the hub, or g_0 when $g_1 = g_0$; however, this will cause an increase in the radial stress at the flange–hub junction. When S_H is excessive and S_R is marginally acceptable, an increase in the thickness of the flange is indicated in which case it may or may not be necessary to alter the size of the hub.

(e) When the longitudinal stress in the hub of the nonreducing flange of a Group 2 or Group 3 assembly is low compared to the allowable stress and the radial stress at the bolt circle is excessive, increasing S_H by making the hub smaller (more flexible) will often reduce the radial stress at the bolt circle to S_f. If it does not, an increase in t_I is indicated.

L-3220 BOLT LOADS

L-3221 Required Bolt Load

The flange bolt load used in calculating the required cross-sectional area of bolts shall be determined as follows.

(a) The required bolt load for the operating condition W_{m1} shall be sufficient to resist the sum of the hydrostatic end force H exerted by the maximum allowable working pressure on the area bounded by the diameter of the gasket reaction, and the contact force H_C exerted by the mating flange on the annular area where the flange faces are in contact. To this shall be added the gasket load H_G for those designs where gasket seating requirements are significant.

(b) Before the contact force H_C can be determined, it is necessary to obtain a value for its moment arm h_C. Due to the interaction between bolt elongation and flange deflection, h_C involves the flange thickness t, operating bolt stress σ_b, initial bolt prestress factor r_s, and calculated strain length l, elasticity factor r_E, and total moment loading on the flange. This Article is based on starting a design by assuming a value for h_C and then calculating the value of the initial bolt stress S_i which satisfies the assumption.

Although the distance h_C from the bolt circle to the flange-to-flange contact circle is a design variable, for the purpose of this Article the use of

$$h_C < h_{C\max}$$

to optimize stresses is considered to be a special situation requiring controlled bolt tightening and verification (see L-3260). Except in special instances, setting h_C equal to $h_{C\max}$ should be satisfactory. It is inherent in the computational process that the flanges will be in tangential contact between the selected bearing circle

$$C + 2h_C$$

and the outside diameter of the flanges

$$C + 2h_{C\max}$$

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(c) The hub–flange interaction moment M_S, which acts on the flange, is expressed by equations L-3242(13), L-3244(a)(25), and L-3244(a)(26); for Category 3 flanges

$$M_S = 0$$

The contact force H_C is determined by equation L-3242(15) or equation L-3244(a)(33).

(d) The required bolt load for operating conditions is determined in accordance with the following equation:

$$W_{m1} = H + H_C + H_G$$

L-3222 Total Required and Actual Bolt Areas, and Flange Design Bolt Load

The total required cross-sectional area of bolts A_m equals W_{m1}/S_b. A selection of bolts to be used shall be made such that the actual total cross-sectional area of bolts A_b will not be less than A_m. The flange design bolt load W shall be taken equal to W_{m1}.

L-3230 CLASSIFICATION OF ASSEMBLIES AND CATEGORIZATION OF INDIVIDUAL FLANGES

It is necessary to classify the different types of flanged assemblies and to further categorize each flange which comprises the assembly under consideration.

L-3231 Classification of a Class FF Flange Assembly

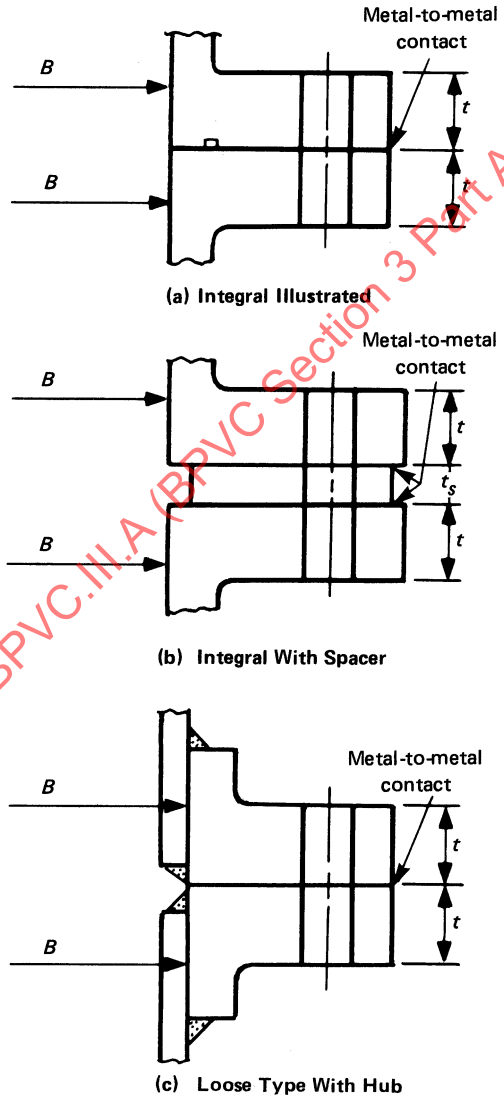
Since the flanges comprising an assembly are in contact outside the bolt circle, the behavior of one flange is influenced by the stiffness of the other. For the purpose of computation it is helpful to classify an assembly consisting of different types of flanges according to the way the flanges influence the deformation of the assembly.

L-3231.1 Group 1 Assembly.[33] A pair of flanges which are bolted together and which are nominally identical with respect to shape, dimensions, physical properties, and allowable stresses[34] except that one flange of the pair may contain a gasket groove. (A Group 1 assembly is also referred to as an identical flange pair.) Figure L-3230-1 illustrates configuration of a Group 1 assembly.

L-3231.2 Group 2 Assembly. Any assemblage which does not fit the description of Group 1 where, in the case of reducers, the inside diameter of the reducing flange exceeds one-half of the bolt circle diameter. Figure L-3230-2 illustrates configuration of a Group 2 assembly.

L-3231.3 Group 3 Assembly. Any assemblage consisting of a reducer or a flat circular head without an opening or with a central, reinforced opening provided the diameter of the opening in the reducing flange or flat cover is less than one-half of the bolt circle diameter. In the analysis the reducing flange is considered to be the equivalent of a flat circular head without an opening. Figure L-3230-3 illustrates configuration of a Group 3 assembly.

Figure L-3230-1
Group 1 Flange Assembly (Identical Flange Pairs)



GENERAL NOTES:
(a) Category 1 flanges illustrated in sketch (a) and (b); Category 2 flanges illustrated in sketch (c).
(b) Permitted weld details are in accordance with Figures XI-3120-1 and NC-4243-2, ND-4243-1, or NE-4243-1, as applicable.

L-3232 Categorization of a Class FF Flange

In addition to classifying an assembly, the individual flanges (except the reducing flange or flat circular head) must be categorized for the purpose of computation as

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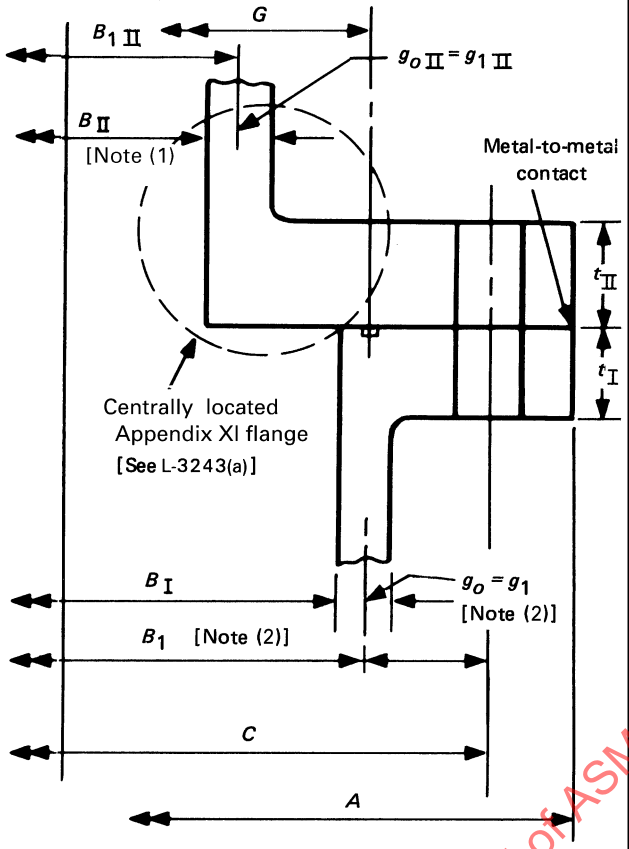
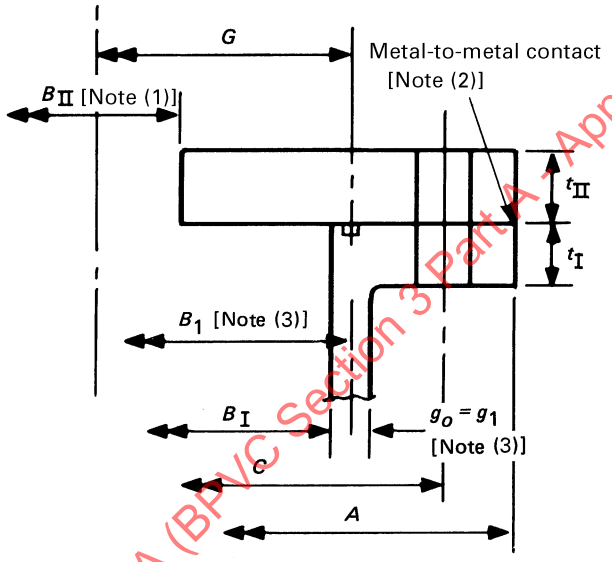
Figure L-3230-2
Group 2 Flange Assembly



GENERAL NOTES:
(a) Category 1 flanges illustrated. Categories II and III permitted.
(b) For purposes of analysis of Flange II by method L-3243(a), assume $A_{II} = G_{II} = B_1$
(c) Permitted weld details are in accordance with Figures XI-3120-1 and NC-4243-2, ND-4243-1 or NE-4243-1, as applicable.

NOTES:
(1) $B_{II} > C/2$.
(2) See L-3192 and L-3193.

Figure L-3230-3
Group 3 Flange Assembly



GENERAL NOTE: Category I flange illustrated. Categories II and III permitted.

NOTES:
(1) $B_{II} \le C/2$.
(2) Permitted weld details are in accordance with Figures XI-3120-1 and NC-3225-3.
(3) See L-3192 and L-3193.

loose type, integral type, or optional type. This can be done using XI-3120; Figure XI-3120-1 is suitable by considering the flanges as flat faced (as a result of removing the raised gasket surface by machining and recessing the gasket in a groove) and by adding a flange-to-flange contact force H_C at some distance h_C outside the bolt circle. Since certain design options exist depending upon the category of the flange, the following categories include both the type of flange and the various design options.

(a) Category 1 Flange. An integral flange or an optional flange calculated as an integral flange.

(b) Category 2 Flange. A loose type flange with a hub where credit is taken for the strengthening effect of the hub.

(c) Category 3 Flange. A loose type flange with a hub where no credit is taken for the strengthening effect of the hub, a loose type flange without a hub, or an optional-type flange calculated as a loose type without a hub. Substitute B for B_1 in the applicable equation for this category of flange.

L-3240 FLANGE ANALYSIS

L-3241 General Method

(a) In order to calculate the stresses in the flanges and bolts of a flanged assembly, classify the assemblage in accordance with L-3231 and then categorize each flange per L-3232.

(b) The method of analyzing various groups and categories of flanges is basically the same. Although many equations appear to be identical, subtle differences do exist and care must be exercised in the analysis. To minimize the need for numerous footnotes and repetitive statements throughout the text, the equations to be used in analyzing the various groups of assemblies and categories of flanges are given in Table L-3240-1. In general, the terms should be calculated in the same order as they

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are listed in the table. It is important to refer to the table before starting an analysis since only a limited number of the equations contained in this Article are used in the design of a particular pair of flanges. Some of the numbered equations appear in L-3191 along with general purpose, unnumbered expressions.

(c) Subscripts I and II refer to the nonreducing flange and the reducer (or flat circular head), respectively, of a Group 3 assembly and of a Group 2 assembly designed using the method of L-3243(a).

(15) L-3242 Analysis of a Group 1 Assembly

The following equations are used for the analysis of Category 1, 2, and 3 flanges of a Group 1 assembly in accordance with Table L-3240-1.

Flange Moment Due to Flange–Hub Interaction

$$M_S = -\frac{J_P F' M_P}{t^3 + J_S F'} \quad (13)$$

Slope of Flange at Inside Diameter Times E

$$E\theta_B = \frac{5.46}{\pi t^3}\left(J_S M_S + J_P M_P\right) \quad (14)$$

Table L-3240-1
Summary of Applicable Equations for Different Groups of Assemblies and Different Categories of Flanges

Group	Category [Note (1)]	Applicable Formulas
1	1	L-3191(5a), L-3242(13) – L-3242(19), L-3242(20a), L-3242(21a), L-3242(22a)
1	2	L-3191(5b), L-3242(13) – L-3242(19), L-3242(20b), L-3242(21b), L-3242(22b)
1	3	L-3191(5c), L-3242(13) – L-3242(19), L-3242(20c), L-3242(21c), L-3242(22c)
2	All	See L-3243
3	1	L-3191(1) – L-3191(4), L-3191(6a), L-3244(a)(23) – L-3244(a)(37), L-3244(a)(38a), L-3244(a)(39a), L-3244(a)(40a), L-3244(a)(41) – L-3244(a)(44)
3	2	L-3191(1) – L-3191(4), L-3191(6a), L-3244(a)(23) – L-3244(a)(37), L-3244(a)(38b), L-3244(a)(39b), L-3244(a)(40b), L-3244(a)(41) – L-3244(a)(44)
3	3	L-3191(1) – L-3191(4), L-3191(6a), L-3244(a)(23) – L-3244(a)(37), L-3244(a)(38c), L-3244(a)(39c), L-3244(a)(40c), L-3244(a)(41) – L-3244(a)(44)

NOTE:
(1) Of the nonreducing flange in a Group 2 or Group 3 assembly.

Contact Force Between Flanges at h_C

$$H_C = \left(M_P + M_S\right)/h_C \quad (15)$$

Bolt Load at Operating Conditions

$$W_{m1} = H + H_G + H_C \quad (16)$$

Operating Bolt Stress

$$\sigma_b = W_{m1}/A_b \quad (17)$$

Design Prestress in Bolts

$$S_i = \sigma_b - \frac{1.159 h_C{}^2\left(M_P + M_S\right)}{a t^3 l r_E B_1} \quad (18)$$

Radial Flange Stress at Bolt Circle

$$S_R = \frac{6\left(M_P + M_S\right)}{t^2\left(\pi C - nD\right)} \quad (19)$$

Radial Flange Stress at Inside Diameter

$$S_R = -\left(\frac{2Ft}{h_0 + Ft} + 6\right)\frac{M_S}{\pi B_1 t^2} \quad (20a)$$

$$S_R = -\left(\frac{2F_L t}{h_0 + F_L t} + 6\right)\frac{M_S}{\pi B_1 t^2} \quad (20b)$$

$$S_R = 0 \quad (20c)$$

Tangential Flange Stress at Inside Diameter

$$S_T = \frac{tE\theta_B}{B_1} + \left(\frac{2FtZ}{h_0 + Ft} - 1.8\right)\frac{M_S}{\pi B_1 t^2} \quad (21a)$$

$$S_T = \frac{tE\theta_B}{B_1} + \left(\frac{2F_L tZ}{h_0 + F_L t} - 1.8\right)\frac{M_S}{\pi B_1 t^2} \quad (21b)$$

$$S_T = \frac{tE\theta_B}{B_1} \quad (21c)$$

Longitudinal Hub Stress

$$S_H = \frac{h_0 E\theta_B f}{0.91\left(g_1/g_0\right)^2 B_1 V} \quad (22a)$$

$$S_H = \frac{h_0 E\theta_B}{0.91\left(g_1/g_0\right)^2 B_1 V_L} \quad (22b)$$

$$S_H = 0 \quad (22c)$$

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(15) L-3243 Analysis of a Group 2 Assembly

(a) The assembly may be analyzed using a variation of the analysis for a Group 3 assembly (L-3244) that accounts for the interaction of nonidentical flanges and the stiffening effect of an integral nozzle or hub centrally located in the reducing flange.

(1) The central nozzle of Flange II with diameter B_{1I} shall be assumed for analysis purposes as an Mandatory Appendix XI flange with outside diameter A, bolt circle C, and gasket circle G all equal to B_1 of Flange I. See Figure L-3230-2.

(2) In addition it is necessary to categorize the centrally located Mandatory Appendix XI flange (nozzle plus the associated over plate to diameter B_1) as a Category 1, 2, or 3 flange in accordance with L-3232.

(3) The moment due to pressure shall be designated M_p' where

$$M_p' = H_D' h_D' + H_T' h_T'$$

$$H_D' = 0.785 B_{II}^2 P$$

$$H_T' = 0.785P\left(B_1^2 - B_{II}^2\right)$$

For Category 1, 2, or 3 flanges [see (2) above],

$$h_T' = \frac{B_1 - B_{II}}{4}$$

For Category 1 or 2 flanges [see (2) above],

$$h_D' = \frac{B_1 - B_{II} - g_{1II}}{2}$$

For Category 3 flanges [see (2) above],

$$h_D' = \frac{B_1 - B_{II}}{2}$$

(4) The rules in L-3244 and the summary of Table L-3240-1 for the analysis of a Group 3 assembly apply to the analysis of a Group 2 assembly with the following additions and substitutions:

C_5 and C_6 and all the symbols in equations in (-a) and (-b) below pertain only to the centrally located Mandatory Appendix XI flange [nozzle plus the associated cover of thickness t_{1I} to diameter B_1 defined in (1) above]. All terms in equations in (-c) and (-d) below, except C_5 and C_6, refer to the nonreducing flange (Flange I). C_1 and C_2 of equations in (-c) and (-d) below replace C_1 and C_2 of eqs. L-3191(1) and L-3191(2).

(-a) Let

$$C_5 = M_P'$$

(-b) Let

$$C_6 = \frac{0.829}{\log\left(B_1 / B_{1II}\right)}$$

for Category 3 flanges.[35] Let

$$C_6 = \frac{0.91 t_{II}^3 V}{L h_0 g_0^2}$$

for Category 1 or 2 flanges.[35]

(-c) Let

$$C_1 = \left[1 - 2.095 J_S \log\left(A / B_1\right)\right] \div \left[-C_6 - 1.738 J_S\right]$$

(-d) Let

$$C_2 = \left(1.738 J_P M_P - C_5 C_6\right) \div \left(-C_6 - 1.738 J_S\right)$$

(-e) Replace eq. L-3244(a)(32) with:

$$E_{II}\theta_{BII} = \frac{5.46}{\pi t_{II}^3}\left(J_S M_{bII} + J_P M_P\right) + \left(E_{II}^* \theta_{rbII}\right) / t_{II}^3$$

(-f) Delete eq. L-3244(a)(44). Subparagraphs (1), (2), and (3) above apply only for calculating C_5 (M_p') and C_6, and subsequently when using (5) below for calculating the stresses in and adjacent to the nozzle in Flange II.

(5) Stresses in the centrally located nozzle of Flange II shall be calculated in accordance with the following equations after M_{S1I} has been found using (4) above. All terms, such as e, Y, and Z, apply to the centrally located Mandatory Appendix XI flange as defined in (1) and (2) above.

For Category 1 or 2 flanges [(2) above]:

Longitudinal Hub Stress

$$S_{HII} = \frac{f\left(M_p' - M_{SII}\right)}{L g_{1\,II}^2 B_{II}}$$

Radial Flange Stress Adjacent to Central Nozzle

$$S_{RII} = \frac{\left(1.33 t_{II} e + 1\right)\left(M_p' - M_{SII}\right)}{L t_{II}^2 B_{II}}$$

Tangential Flange Stress Adjacent to Central Nozzle

$$S_{TII} = \frac{Y\left(M_p' - M_{SII}\right)}{t_{II}^2 B_{II}} - Z S_{RII}$$

For Category 3 Flanges [(a)(2) above]:

Tangential Flange Stress Adjacent to Central Nozzle

$$S_{TII} = \frac{Y\left(M_p' - M_{SII}\right)}{t_{II}^2 B_{II}}$$

Radial and Longitudinal Hub Stress

$$S_{RII} = 0$$

$$S_{HII} = 0$$

(6) The stresses in Flange I and the remaining stresses in Flange II shall be calculated in accordance with L-3244 except as modified by (4).

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(b) As an alternative to the method in (a) above and at the option of the designer, the assembly may be analyzed as if it is one flange of an identical pair in a Group 1 assembly using the procedure in L-3242. All stresses shall satisfy L-3250. The same value of h_C shall be used in both calculations and the strain length l of the bolts shall be based on the thickness of the flange under consideration. This method is more conservative and more bolting may be required than the method in (a) above.

(c) The central nozzle or opening in Flange II of a Group 2 assembly determined by the rules in (a) or (b) above meets the general requirements of this Division and of this Article. The rules for determining thickness and reinforcing requirements of NC-3225, NC-3325, ND-3325, and NE-3325, and NC-3233, NC-3333, ND-3333, and NE-3333, respectively, are not applicable.

(15) **L-3244 Analysis of a Group 3 Assembly**

(a) The following equations are used for the analysis of Category 1, 2, and 3 nonreducing flanges and the reducer (or flat circular head) of a Group 3 assembly:

Rigid Body Rotation of Flanges Times E^*

$$E_{\mathrm{I}}^{*}\,\theta_{rb\mathrm{I}} = \frac{X\left(C_4 - C_2\right)}{1.206\log\left(A/B_1\right) - XC_3 - \left(1 - X\right)C_1} \tag{23}$$

$$E_{\mathrm{II}}^{*}\,\theta_{rb\mathrm{II}} = -E_{\mathrm{I}}^{*}\theta_{rb\mathrm{I}}\left(E_{\mathrm{II}}^{*}/E_{\mathrm{I}}^{*}\right) \tag{24}$$

Total Flange Moment at Diameter B_1

$$M_{S\mathrm{I}} = C_3\left(E_{\mathrm{I}}^{*}\,\theta_{rb\mathrm{I}}\right) + C_4 \tag{25}$$

$$M_{S\mathrm{II}} = C_1\left(E_{\mathrm{II}}^{*}\,\theta_{rb\mathrm{II}}\right) + C_2 \tag{26}$$

Unbalanced Flange Moment at Diameter B_1

$$M_{u\mathrm{I}} = 1.206\,E_{\mathrm{I}}^{*}\,\theta_{rb\mathrm{I}}\log\left(A/B_1\right) \tag{27}$$

$$M_{u\mathrm{II}} = 1.206\,E_{\mathrm{II}}^{*}\,\theta_{rb\mathrm{II}}\log\left(A/B_1\right) \tag{28}$$

Balanced Flange Moment at Diameter B_1

$$M_{b\mathrm{I}} = M_{S\mathrm{I}} - M_{u\mathrm{I}} \tag{29}$$

$$M_{b\mathrm{II}} = M_{S\mathrm{II}} - M_{u\mathrm{II}} \tag{30}$$

Slope of Flange at Diameter B_1 Times E

$$E_{\mathrm{I}}^{*}\,\theta_{B\mathrm{I}} = \frac{5.46}{\pi t_{\mathrm{I}}^{3}}\left(J_S M_{b\mathrm{I}} + J_P M_P\right) + E_{\mathrm{I}}^{*}\,\theta_{rb\mathrm{I}}/t_{\mathrm{I}}^{3} \tag{31}$$

$$E_{\mathrm{II}}\,\theta_{B\mathrm{II}} = \frac{-1.337\left(M_{S\mathrm{II}} - \pi P B_1^{3}/32\right)}{t_{\mathrm{II}}^{3}} \tag{32}$$

Contact Force Between Flanges at h_C

$$H_C = \left(M_P + M_{b\mathrm{I}}\right)/h_C \tag{33}$$

Bolt Load at Operating Conditions

$$W_{m1} = H + H_G + H_C \tag{34}$$

Operating Bolt Stress

$$\sigma_b = W_{m1}/A_b \tag{35}$$

Design Prestress in Bolts

$$S_i = \sigma_b - \frac{1.159 h_C^{2}\left(M_P + M_{b\mathrm{I}}\right)}{2\left(1 - X\right)a t_{\mathrm{I}}^{3} l r_{E\mathrm{I}} B_1} \tag{36}$$

Radial Stress in Flange I at Bolt Circle

$$S_{R\mathrm{I}} = \frac{6\left(M_P + M_{S\mathrm{I}}\right)}{t_{\mathrm{I}}^{2}\left(\pi C - nD\right)} \tag{37}$$

Radial Stress in Flange I at Inside Diameter

$$S_{R\mathrm{I}} = -\left(\frac{2Ft_{\mathrm{I}}}{h_0 + Ft_{\mathrm{I}}} + 6\right)\frac{M_{S\mathrm{I}}}{\pi B_1 t_{\mathrm{I}}^{2}} \tag{38a}$$

$$S_{R\mathrm{I}} = -\left(\frac{2F_L t_{\mathrm{I}}}{h_0 + F_L t_{\mathrm{I}}} + 6\right)\frac{M_{S\mathrm{I}}}{\pi B_1 t_{\mathrm{I}}^{2}} \tag{38b}$$

$$S_{R\mathrm{I}} = 0 \tag{38c}$$

Tangential Stress in Flange I at Inside Diameter

$$S_{T\mathrm{I}} = \frac{t_{\mathrm{I}} E_{\mathrm{I}}\theta_{B\mathrm{I}}}{B_1} + \left(\frac{2Ft_{\mathrm{I}}Z}{h_0 + Ft_{\mathrm{I}}} - 1.8\right)\frac{M_{S\mathrm{I}}}{\pi B_1 t_{\mathrm{I}}^{2}} \tag{39a}$$

$$S_{T\mathrm{I}} = \frac{t_{\mathrm{I}} E_{\mathrm{I}}\theta_{B\mathrm{I}}}{B_1} + \left(\frac{2F_L t_{\mathrm{I}}Z}{h_0 + F_L t_{\mathrm{I}}} - 1.8\right)\frac{M_{S\mathrm{I}}}{\pi B_1 t_{\mathrm{I}}^{2}} \tag{39b}$$

$$S_{T\mathrm{I}} = \frac{t_{\mathrm{I}} E_{\mathrm{I}}\theta_{B\mathrm{I}}}{B_1} \tag{39c}$$

Longitudinal Hub Stress in Flange I

$$S_{H\mathrm{I}} = \frac{h_0 E_{\mathrm{I}}\theta_{B\mathrm{I}} f}{0.91\left(g_1/g_0\right)^{2} B_1 V} \tag{40a}$$

$$S_{H\mathrm{I}} = \frac{h_0 E_{\mathrm{I}}\theta_{B\mathrm{I}}}{0.91\left(g_1/g_0\right)^{2} B_1 V_L} \tag{40b}$$

$$S_{H\mathrm{I}} = 0 \tag{40c}$$

Radial Stress in Flange II at Bolt Circle

$$S_{R\mathrm{II}} = \frac{6\left(M_P + M_{S\mathrm{II}}\right)}{t_{\mathrm{II}}^{2}\left(\pi C - nD\right)} \tag{41}$$

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Radial Stress in Flange II at Diameter B_1

$$S_{RII} = \frac{6M_{SII}}{\pi B_1 t_{II}^2} \quad (42)$$

Tangential Stress in Flange II at Diameter B_1

$$S_{TII} = \frac{t_{II} E_{II} \theta_{BII}}{B_1} - \frac{1.8M_{SII}}{\pi B_1 t_{II}^2} \quad (43)$$

Radial and Tangential Stress at Center of Flange II

$$S_{RII} = S_{TII} = \frac{0.3094PB_1^2}{t_{II}^2} - \frac{6M_{SII}}{\pi B_1 t_{II}^2} \quad (44)$$

(b) The thickness of Flange II of a Group 3 assembly determined by the above rules shall be used in lieu of the thickness that is determined by NC-3225, NC-3325, ND-3325, and NE-3325. However, any centrally located opening in Flange II shall be reinforced to meet the rules of Mandatory Appendix XIX.

L-3250 ALLOWABLE FLANGE DESIGN STRESSES

The stresses calculated by the above equations, whether tensile or compressive (−), shall not exceed the following values for all groups of assemblies:[36]

(a) operating bolt stress σ_b not greater than S_b for the design value of S_i

(b) longitudinal hub stress S_H not greater than S_f for Category 1 and 2 cast iron flanges except as otherwise limited by (1) and (2) below and not greater than 1.5 S_f for materials other than cast iron

(1) longitudinal hub stress S_H not greater than the smaller of 1.5 S_f or 1.5 S_n for Category 1 flanges where the pipe or shell constitutes the hub

(2) longitudinal hub stress S_H not greater than the smaller of 1.5 S_f or 2.5 S_n for integral flanges (Category 1) similar to the Mandatory Appendix XI flanges shown as Figure XI-3120-1, sketches (6), (6a), and (6b)

(c) radial stress S_R not greater than S_f

(d) tangential stress S_T not greater than S_f

(e) also, $(S_H + S_R)/2$ not greater than S_f and $(S_H + S_T)/2$ not greater than S_f

(f) S_R and S_T at the center of the reducing flange in a Group 3 assembly [see eq. L-3244(a)(44)] shall not exceed S_f

L-3260 PRESTRESSING THE BOLTS

The design rules of this Article provide for tangential contact between the flanges at $h_{C\max}$ or some lesser value h_C beyond the bolt circle. As in the case of Mandatory Appendix XI flanges, a Class FF flange must be designed so that the calculated value of the operating bolt stress σ_b does not exceed S_b. Also, as in the case of Mandatory Appendix XI flanges, ordinary wrenching techniques without verification of the actual initial bolt stress (assembly stress) is considered to meet all practical needs with control and verification reserved for special applications. For the purposes of this Article, the use of $h_C < h_{C\max}$ to optimize stresses is considered to be a special application unless it is also shown that all of the requirements of this Article are also satisfied when $h_C = h_{C\max}$.

L-3270 REFERENCES

Additional guidance on the design of flat faced metal-to-metal contact flanges can be found in the following references:

(a) Schneider, R. W., and Waters, E. O., The Background of ASME Code Case 1828: A Simplified Model of Analyzing Part B Flanges, *Journal of Pressure Vessel Technology*, ASME, Vol. 100, No. 2, May 1978, pp. 215–219;

(b) Schneider, R. W., and Waters, E. O., The Application of ASME Code Case 1828, *Journal of Pressure Vessel Technology*, ASME, Vol. 101, No. 1, February 1979, pp. 87–94.

It should be noted that the rules in Nonmandatory Appendix L were formerly contained in Code Case 1828, A Simplified Method for Analyzing Flat Face Flanges with Metal-to-Metal Contact Outside the Bolt Circle/ Section VIII, Division 1.

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(15)

NONMANDATORY APPENDIX M
RECOMMENDATIONS FOR CONTROL OF WELDING, POSTWELD HEAT TREATMENT, AND NONDESTRUCTIVE EXAMINATION OF WELDS

ARTICLE M-1000
RECOMMENDATIONS

M-1100 INTRODUCTION

M-1110 SCOPE

This Appendix provides recommendations for control of welding, postweld heat treatment, and nondestructive examination of welds. The purpose of this Appendix is to identify the areas that should be controlled. It should not be construed that this Appendix describes the only controls that are necessary. Other controls may be necessary depending on the specific application.

M-1120 APPLICABILITY

The recommendations contained in this Appendix may be included in procedures, instructions, specifications, process sheets, shop travelers, checklists, drawings, or other documents that are part of the Quality Assurance or Control Program.

M-1200 WELDING PROCEDURE SPECIFICATIONS

M-1210 RESPONSIBILITY FOR PREPARATION

The responsibility for preparing and approving the Welding Procedure Specification, and the method of handling revisions should be defined in the Quality Assurance or Control Program.

M-1211 Content Requirements

Section IX identifies the essential and also the nonessential variables required to be included in a Welding Procedure Specification. Supplementary information that should be provided, as applicable, is listed in (a) through (k) below. This information may be provided in other specifications, drawings, or documents that are referenced in or used in conjunction with the Welding Procedure Specification, as described in M-1213.

(a) weld joint design, including fit-up

(b) method of weld joint preparation

(c) cleaning requirements prior to welding

(d) purge requirements in addition to Section IX, QW-408.5, such as oxygen content, volume changes, or time of purging prior to welding, and minimum thickness of deposited metal prior to removal or purge

(e) cleaning and examination requirements between weld passes

(f) instructions for measuring preheat and interpass temperatures

(g) preheat temperature control

(h) interpass temperature control

(i) temperature maintenance and control after welding

(j) welding current and voltage control including method and frequency of measurement

(k) welding technique including, as applicable, method of starts and stops, method of arc initiation, and weave or stringer bead

M-1212 Format of Welding Procedure Specification

Welding Procedure Specifications may consist of a single document. Alternatively, there may be a general Welding Procedure Specification that applies to and is referenced by a number of detailed Welding Procedure Specifications. The general Welding Procedure Specification may also reference other specifications to define other requirements such as welding filler materials, purging, and postweld heat treatment.

M-1213 Application of Welding Procedures

Selection of qualified welding procedures should be based on the essential variables applicable for the production weld. Any additional requirements of the Design Specification, such as component classification, applicable Code edition and addenda, fracture toughness

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requirements, and manufacturing considerations, should be evaluated in the selection of welding procedures for specific weld joints.

M-1300 WELDING PERFORMANCE QUALIFICATION AND ASSIGNMENT

It is necessary that a system be established to ensure that properly qualified welders and welding operators are used for specific weld joints. The system should include confirmation that the welder and welding operator qualifications are up to date and valid for the application.

M-1400 CONTROL OF WELDING

A system should be established and maintained that is capable of

(a) maintaining the identification of acceptable electrodes, filler metal, consumable inserts, and fluxes until consumed, and

(b) minimizing moisture absorption of coated electrodes and of fluxes

The system should include requirements for receiving inspection, storage, handling, use of holding ovens, control of exposure time at ambient temperature, and reconditioning.

M-1500 NONDESTRUCTIVE EXAMINATION OF WELDS

M-1510 RESPONSIBILITY FOR PREPARATION

Nondestructive examination procedures should be prepared in accordance with the requirements of the particular Article of this Section used in fabrication, including acceptance standards. The responsibility for preparation and approval of procedures and the method of handling revisions should be defined in the Quality Assurance or Control Program.

M-1520 VERIFICATION OF APPLICABILITY

Prior to assignment of nondestructive examination procedures to specific weld joints, a review should be made of the Code classification of the component and the Code edition and addenda to ensure that the type of examination, procedure and acceptance standards, frequency, and time of examination are all correct for the application.

M-1600 POSTWELD HEAT TREATMENT

A system should be established and maintained that is capable of meeting PWHT requirements for heating and cooling rates, metal temperature, metal temperature uniformity, and temperature control. The location of thermocouples, furnace loading to prevent direct impingement of flame on parts or components, and the furnace atmosphere should be considered in establishing procedures.

M-1700 EXAMINATION AND DIMENSIONAL INSPECTION

Examinations and dimensional inspections should be made to provide assurance that the requirements of this Section are met and that welding conforms with the procedures, specifications, and drawings. As applicable, consideration should be given to (a) and (b).

(a) Before welding
- *(1)* material identification
- *(2)* weld preparation surfaces
- *(3)* cleanliness
- *(4)* root opening
- *(5)* offset
- *(6)* alignment
- *(7)* socket engagement on socket welds

(b) After welding
- *(1)* cracks or linear indications
- *(2)* rounded indications
- *(3)* overlap
- *(4)* undercut
- *(5)* lack of required penetration
- *(6)* surface finish
- *(7)* weld profile

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NONMANDATORY APPENDIX N

ARTICLE N-1000
DYNAMIC ANALYSIS METHODS

N-1100 INTRODUCTION AND SCOPE

Section III does not require dynamic analysis. However, the design of nuclear components requires consideration of the seismic and other dynamic inputs which are defined in the Design Specifications for the component. Component design may be based on the use of static forces resulting from equivalent earthquake acceleration acting at the centers of gravity of the extended masses, or a dynamic system analysis may be used to show how seismic loading is transmitted from the defined ground motions to all parts of the buildings, structures, equipment, and components. N-1100 through N-1200 of this Appendix are presented to illustrate one or more acceptable steps for seismic dynamic analysis. It is not intended that these steps are the only acceptable ones, since the seismic dynamic analysis involves a series of steps, and some of these steps have acceptable alternative methods. Dynamic analysis in general uses techniques which are illustrated in seismic analysis. Those technical areas of dynamic analysis used in nuclear component design which are not specifically illustrated by seismic analysis are included in N-1300 through N-1700.

N-1110 DEFINITIONS AND NOTATIONS

(a) The *magnitude* of an earthquake is a measure of the size of an earthquake and is related to the energy released in the form of seismic waves. Magnitude means the numerical value on a Richter scale. The *intensity* of an earthquake is a measure of its effects on man, on man-built structures, and on the earth's surface at a particular location. Intensity is measured by the numerical value of the modified Mercalli scale.

(b) The *Safe Shutdown Earthquake* (SSE) is that earthquake which is based upon an evaluation of the maximum earthquake potential considering the regional and local geology and seismology and specific characteristics of local subsurface material. It is that earthquake which produces the maximum vibratory ground motion for which those structures, systems, and components important to safety are designed to remain functional.

(c) The *Operating Basis Earthquake* (OBE) is that earthquake which, considering the regional and local geology and seismology and specific characteristics of local subsurface material, could reasonably be expected to affect the plant site during the operating life of the plant.

(d) The *response spectrum* is defined as a plot of the maximum response (acceleration, velocity, or displacement) of a family of idealized linear single-degree-of-freedom damped oscillators as a function of natural frequencies (or periods) of the oscillators to a specified vibratory motion input at their supports.

(e) The *design ground response spectrum* is a smooth response spectrum obtained by analyzing, evaluating, and statistically combining a number of individual response spectra derived from the records of significant past earthquakes.

(f) The *maximum (peak) ground acceleration* (for a given site) is defined as that value of the acceleration which corresponds to zero period in the design response spectra for that site. At zero period the design response spectra acceleration is identical for all damping values and is equal to the maximum (peak) ground acceleration specified for that site.

(g) Normal mode — time history methods use the normal mode theory and a time history of the input motion. When normal mode theory is used, the maximum response is determined by obtaining the combined response of all individual modes at a particular time.

(h) Direct integration — time history methods use numerical step-by-step integration of the equations of motion of a time history of the input motion.

(i) Equivalent statical methods means the use of a statical loading coefficient which gives a definite upper limit to the response.

(j) Coupled structures and plant equipment include those structures and plant equipment which, because of their mass and stiffness properties, significantly influence the dynamic response of each other and must be considered together in a dynamic analysis.

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(k) Uncoupled structures and plant equipment include those structures and plant equipment which, because of their mass and stiffness properties, do not significantly influence the dynamic response of each other and can be considered in separate dynamic analysis.

(l) Seismic systems are all those structures for which loads induced by earthquake should be considered.

(m) Rigid range is used to describe those frequencies of structures, systems, or components whose natural frequencies are greater than some value at which dynamic response acceleration is essentially the same as the impact acceleration. For example, a response to the seismic design response spectra of Figures N-1211(a)-1 and N-1211(b)-1 has unity amplification of acceleration above 33 Hz.

(n) Spectrum consistent time history is a time history which is artificially generated to essentially envelop a given design response spectrum.

N-1200 SEISMIC ANALYSIS

N-1210 EARTHQUAKE DESCRIPTION

N-1211 Ground Response Spectrum

(a) The *horizontal* component ground design response spectra of the Safe Shutdown Earthquake (SSE) or the Operating Basis Earthquake (OBE) on sites underlain by rock or by soil may be linearly scaled from Figure N-1211(a)-1 in proportion to the maximum horizontal ground acceleration specified for the earthquake chosen. [Figure N-1211(a)-1 corresponds to a maximum horizontal ground acceleration of 1.0*g* and accompanying displacement of 36 in. (915 mm).] The applicable multiplication factors and control points are given in Table N-1211(a)-1. For damping ratios not included in Figure N-1211(a)-1 or Table N-1211(a)-1, a linear interpolation may be used.

(b) The *vertical* component ground design response spectra of the SSE or the OBE on sites underlain by rock or by soil may be linearly scaled from Figure N-1211(b)-1 in proportion to the maximum horizontal ground acceleration specified for the earthquake chosen. [Figure N-1211(b)-1 is based on a maximum horizontal ground acceleration of 1.0*g* and accompanying displacement of 36 in. (915 mm).] The applicable multiplication factors and control points are given in Table N-1211(b)-1. For damping ratios not included in Figure N-1211(b)-1 or Table N-1211(b)-1, a linear interpolation may be used.

N-1212 Time History

N-1212.1 Frequency Content of Time History. The design ground response spectra used in the earthquake resistance design, such as those illustrated in Figures N-1211(a)-1 and N-1211(b)-1 for the design of nuclear power plant facilities, are generally based on multicomponent time history motions from a number of major earthquakes. There are no single recorded earthquake ground motions which have such uniform frequency distribution. However, it is often necessary to generate a spectrum-consistent time history motion whose response spectrum matches the design response spectrum for a given damping value. The purpose of developing such a spectrum-consistent time history motion is to provide the analyst with an acceptable basis for generating floor (in structure) response spectra and performing time history analysis of systems and components.

Several acceptable approaches are presented below.

(a) Modified Earthquake Records. An acceptable approach is to modify the components of past earthquake records using spectral raising and suppressing techniques (refs. [71] and [72]).

Spectral raising is accomplished by adding to the original time history a harmonic function at the frequency of interest with a phase angle such that the response spectral value at this frequency will be increased to a desired amount; the time when the maximum vibratory motion occurred will be the same. In this way, the spectral characteristics of the modified time history will be similar to the original earthquake records. Consequently, statistical characteristics of past time history motions can be maintained as discussed in N-1213.

When the time history response spectrum is higher than the design response spectrum at a frequency, spectral suppressing can be carried out by passing the time history through a linearly damped oscillator connected in series with a second damper. This damping arrangement will reduce the response spectral value, locally at the natural frequency of the oscillator, to the desired amount.

This usually requires an iterative procedure. A repetitious application of the raising and suppressing techniques may be used to arrive at a time history motion whose response spectrum is sufficiently close to the design spectrum.

(b) Synthesized Time History Motions. Several methods may be used to generate time history motions without the direct use of an actual earthquake record. One method uses power spectral density functions to generate the time history motions. Power spectral density (PSD) functions may be calculated for the strong motion portion of actual earthquake records by assuming that the strong motion portion is stationary and Gaussian. This (PSD) function may then be used to produce a sample of a pseudo-earthquake ensemble by filtering a white noise record with unit power density through a linear damped system (ref. [73]).

A second procedure which may be used is to pass Gaussian shot noise through selected filters (ref. [74]). This method may produce acceptable results if used in conjunction with the spectral raising and suppressing techniques described earlier.

Another acceptable procedure involves the superposition of continuous waves with the assumption that earthquake motion is stationary for the strong motion portion

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Figure N-1211(a)-1
Horizontal Design Response Spectra Scaled to 1*g* Horizontal Ground Acceleration

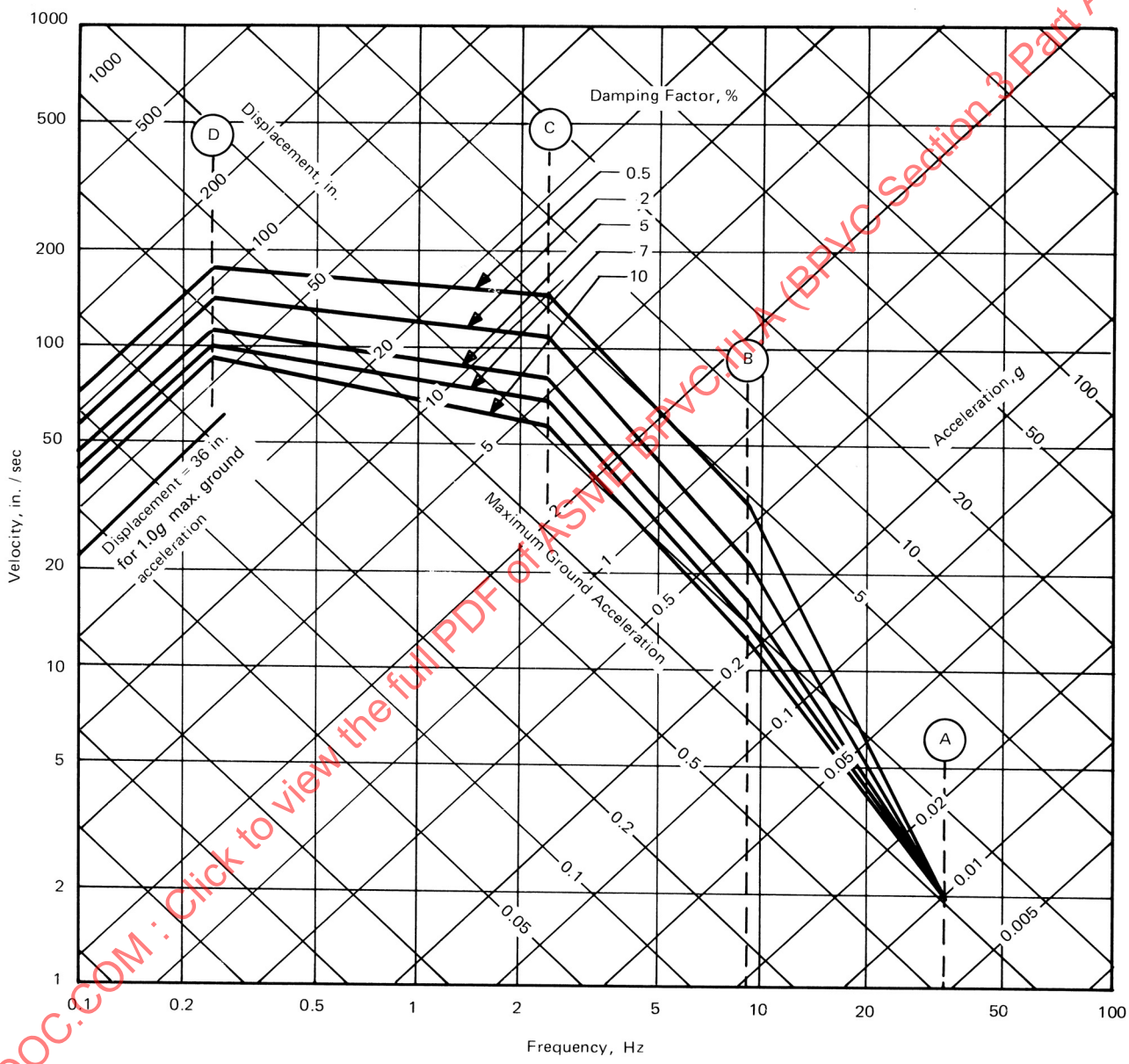

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Table N-1211(a)-1
Horizontal Design Response Spectra Relative Values of Spectrum Amplification Factors for Control Points

Percent of Critical Damping	Amplification Factors for Control Points			
	Acceleration [Note (1)], [Note (2)]			Displacement [Note (1)], [Note (2)]
	A (33 Hz)	B (9 Hz)	C (2.5 Hz)	D (0.25 Hz)
0.5	1.0	4.96	5.95	3.20
2.0	1.0	3.54	4.25	2.50
5.0	1.0	2.61	3.13	2.05
7.0	1.0	2.27	2.72	1.88
10.0	1.0	1.90	2.28	1.70

NOTES:
(1) Maximum ground displacement is taken proportional to maximum ground acceleration and is 36 in. (915 mm) for ground acceleration of 1.0*g*.
(2) Acceleration and displacement amplification factors are taken from recommendations given in ref. [1] and discussed in refs. [2] and [3].

of the total time duration. The earthquake is characterized as the product of a normalized stationary process and a scaling factor that establishes the magnitude of the motion and its envelope shape which consists of three distinct parts: the buildup, the flat or stationary portion, and the motion decay. The random process is represented by a finite number of superimposed sine waves with distinct periods and randomly selected phase angles (ref. [30]).

Synthesized time history motions for the three perpendicular directions of an earthquake may not have time phase relationships similar to those of past earthquake records. The procedures of N-1213.1 may be used to check that time phasing is acceptable.

N-1212.2 Duration of Time History.

(a) It has been observed (ref. [51]) that the earthquake duration effect on the response spectrum shape is small for periods shorter than 0.5 sec, which is the period range significant for nuclear power plant structures, systems and components.

(b) Although actual long duration earthquakes tend to excite a much wider range of frequencies, this effect is included in the nuclear power plant design by using the smoothed, envelope type of design response spectra. The use of a design response spectrum in qualifying structures and components dictates that only maximum response amplitudes, be it displacement or stresses, are computed. The duration of ground motions, which is an influence in prolonging large amplitude structural response, may be important when fatigue life of the structures and components is considered. The fatigue effect is discussed in N-1214.

(c) The earthquakes used in producing the recommended response spectra [Figures N-1211(a)-1 and N-1211(b)-1] occurred almost exclusively in the California area originating along the Circum-Pacific Belt. They are referred to as type 2 earthquakes in ref. [51]. These type 2 earthquakes are associated with moderate distances from the focus and occur only on firm ground. They last for only a moderate time. The type 2 earthquakes are unlike those referred to as type 3 which occur on soft soil such as the Mexico City earthquakes of July 6, 1964, and which have a much longer duration.

(d) Maximum accelerations of type 2 earthquakes generally occur in the first 10 sec of strong shaking. For instance, the north–south component acceleration recorded at the El Centro earthquake, May 18, 1940, shows that the maximum horizontal ground acceleration of 0.33 g occurs at about 2 sec after the instrument started recording.

(e) Time history analysis is necessary to qualify systems and components which exhibit highly nonlinear characteristics, such as the opening and closing of large gaps between members. Long seismic inputs can waste computation resources. Duration of the artificial time histories should be sufficient to allow the structure enough time to achieve its maximum response. For developing response spectra, the input time history used should be chosen long enough so that the resultant response spectra do not significantly change if the time history is increased. For developing spectrum-consistent time histories, the resultant time history should be long enough so that further increases in its length will not produce significantly different response spectra.

(f) Generally the minimum duration of the strong seismic motion required may be taken as 6 sec for Code components. If this minimum 6 sec strong motion duration is used, then a buildup duration of about 4 sec is recommended to precede the strong motion. The buildup region may be taken as the time duration from where motion is zero to the time where stationarity or strong motion begins. In cases where it is uncertain as to the minimum strong motion duration required to produce maximum structural response, the analysis may be extended to longer strong motion durations such as 10 sec in order to

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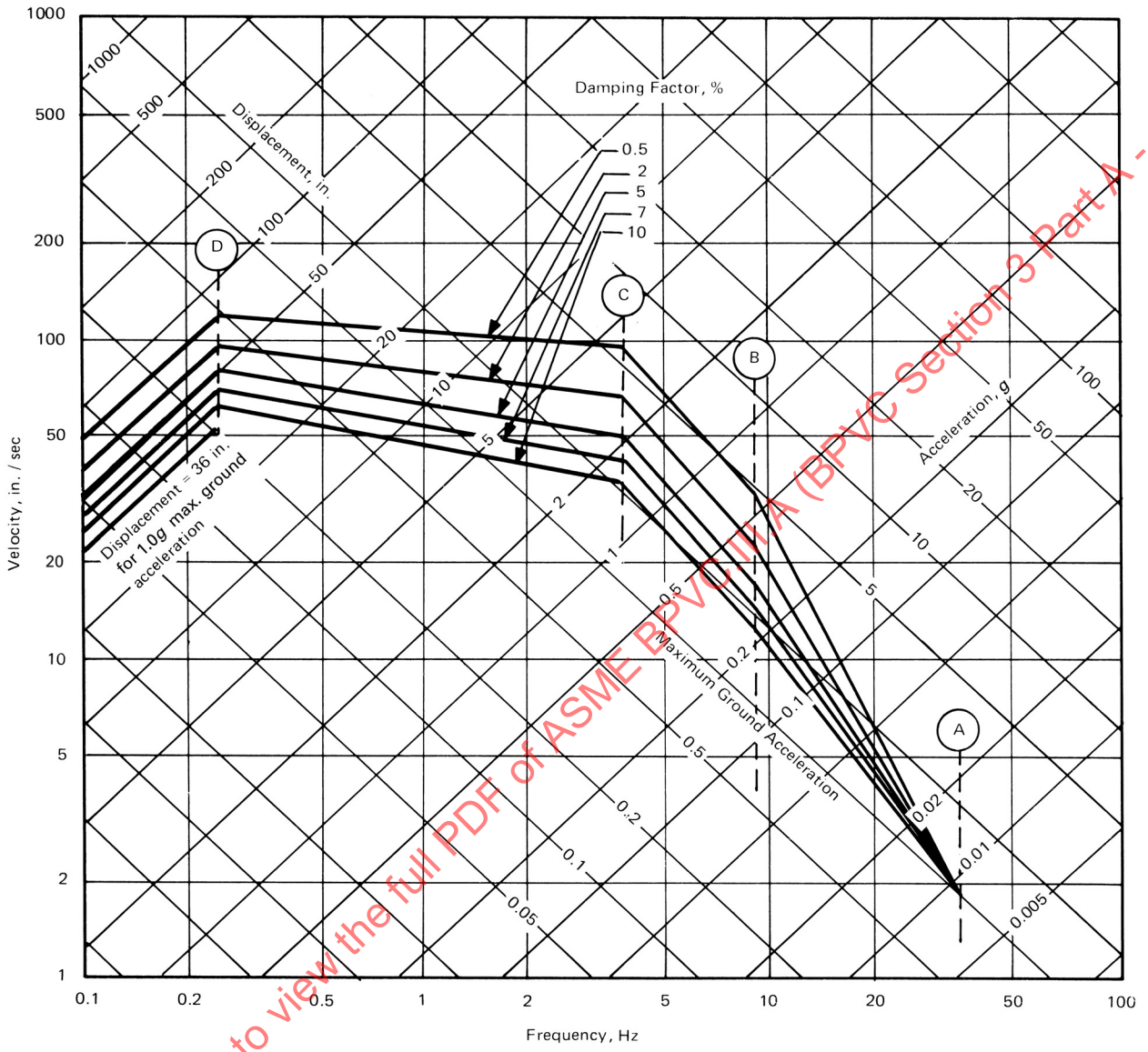
Figure N-1211(b)-1
Vertical Design Response Spectra Scaled to 1*g* Horizontal Ground Acceleration



evaluate whether significant response increases occur above those of the 6 sec duration. The method in ref. [71] may be used when required to include shorter or longer periods of vibratory energy in an artificial time history.

N-1213 Directional and Time Phase Considerations

Three orthogonal components of earthquake excitations are normally considered for nuclear power plant design. Rotational ground motion in the three directions, however, may be neglected. The triaxial input excitations are characterized by the relative magnitudes of the peak accelerations of the three excitations, and the relative values of the response spectra over the frequency range of interest (N-1211). Time histories which are consistent with the design response spectra may be developed through methods discussed in N-1212 and which essentially envelop the design response spectra. Since these artificial time histories may not have time phase relationships similar to those of past earthquake records, the procedures of N-1213.1 may be used to check that time phasing is acceptable.

N-1213.1 Time Phase Relationships. The peak acceleration of the three orthogonal synthetic time histories generally need not occur at the same time. In order to

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Table N-1211(b)-1
Vertical Design Response Spectra Relative Values of Spectrum Amplification Factors for Control Points

Percent of Critical Damping	Amplification Factors for Control Points			
	Acceleration [Note (1)], [Note (2)]			Displacement [Note (1)], [Note (2)]
	A (33 Hz)	B (9 Hz)	C (3.5 Hz)	D (0.25 Hz)
0.5	1.0	4.96	5.67	2.13
2.0	1.0	3.54	4.05	1.67
5.0	1.0	2.61	2.98	1.37
7.0	1.0	2.27	2.59	1.25
10.0	1.0	1.90	2.17	1.13

NOTES:
(1) Maximum ground displacement is taken proportional to maximum ground acceleration and is 36 in. (915 mm) for ground acceleration of 1.0g.
(2) Acceleration amplification factors for the vertical design response spectra are equal to those for horizontal design response spectra at a given frequency, whereas displacement amplification factors are $^2/_3$ those for horizontal design response spectra. These ratios between the amplification factors for the two design response spectra are in agreement with those recommended in ref. [1] and discussed in refs. [2] and [3].

stimulate natural earthquake occurrences, the correlation of the synthesized time histories may be evaluated by calculating the cross correlation coefficients and the coherence functions (refs. [4], [5], and [6]). The artificially generated time histories are acceptable if both their cross correlation coefficients and their coherence functions are approximately equal to the respective functions for past earthquake records. An absolute value of the correlation coefficient less than 0.16 is acceptable. For the coherence function the numerical values ranging between 0.0 and 0.3 with an average of approximately 0.2 are acceptable.

N-1214 Cyclic Criteria

(a) An acceptable cyclical load basis for fatigue analysis of earthquake loading of equipment and components is 10 equivalent maximum stress cycles per earthquake.

(b) The equivalent maximum stress cycle is defined as the full range including plus or minus seismic load calculated by equivalent static or response spectrum modal analysis techniques.

(c) The total usage factor contribution is based upon the number of earthquakes considered for the component.

N-1220 METHODS OF DYNAMIC ANALYSIS

N-1221 Modeling Techniques for Dynamic Analysis

In the course of preparation.

N-1222 Time History Method

(a) The structural system to be analyzed can be generally classified into two forms, linear and nonlinear. In the linear system the structure is idealized in such a way that the responses and motion retain a linear relation with the applied loads. A nonlinear structural dynamic system may be caused by either a material nonlinearity (i.e., a nonlinear stress-strain relationship of the material), a geometrical nonlinearity (an excessive deformation, a support gap, etc. which significantly changes the geometry of the system), or a mixture of the two.

(b) Dynamic analysis for either a linear or nonlinear system is based on the solution of a set of simultaneous, differential equations of motion with given initial and boundary conditions. Acceptable methods are presented for solution to linear and nonlinear equations in N-1222.

N-1222.1 Linear.

(a) The response of a multi-degree-of-freedom linear structural system is described by the differential equation of motion expressed in matrix form

$$[M]\{x''\} + [C]\{x'\} + [K]\{x\} = \{f\} \quad (1)$$

where

C = matrix of viscous damping coefficients
f = force vector and function of time representing external loading at mass points
K = stiffness matrix for the mass points of the linear elastic structure
M = mass matrix
x = displacement vector
x' = velocity vector
x'' = acceleration vector

In all typical linear matrix formulations, matrices M, C, and K are symmetric.

(b) For a seismic analysis where ground motion is the source of excitation, f in eq. (a)(1) can be replaced by $-[M]\{Z''\}$, where Z'' is the acceleration of ground motion in the same direction of x which is the relative displacement

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Figure N-1211(a)-1M
Horizontal Design Response Spectra Scaled to 1*g* Horizontal Ground Acceleration

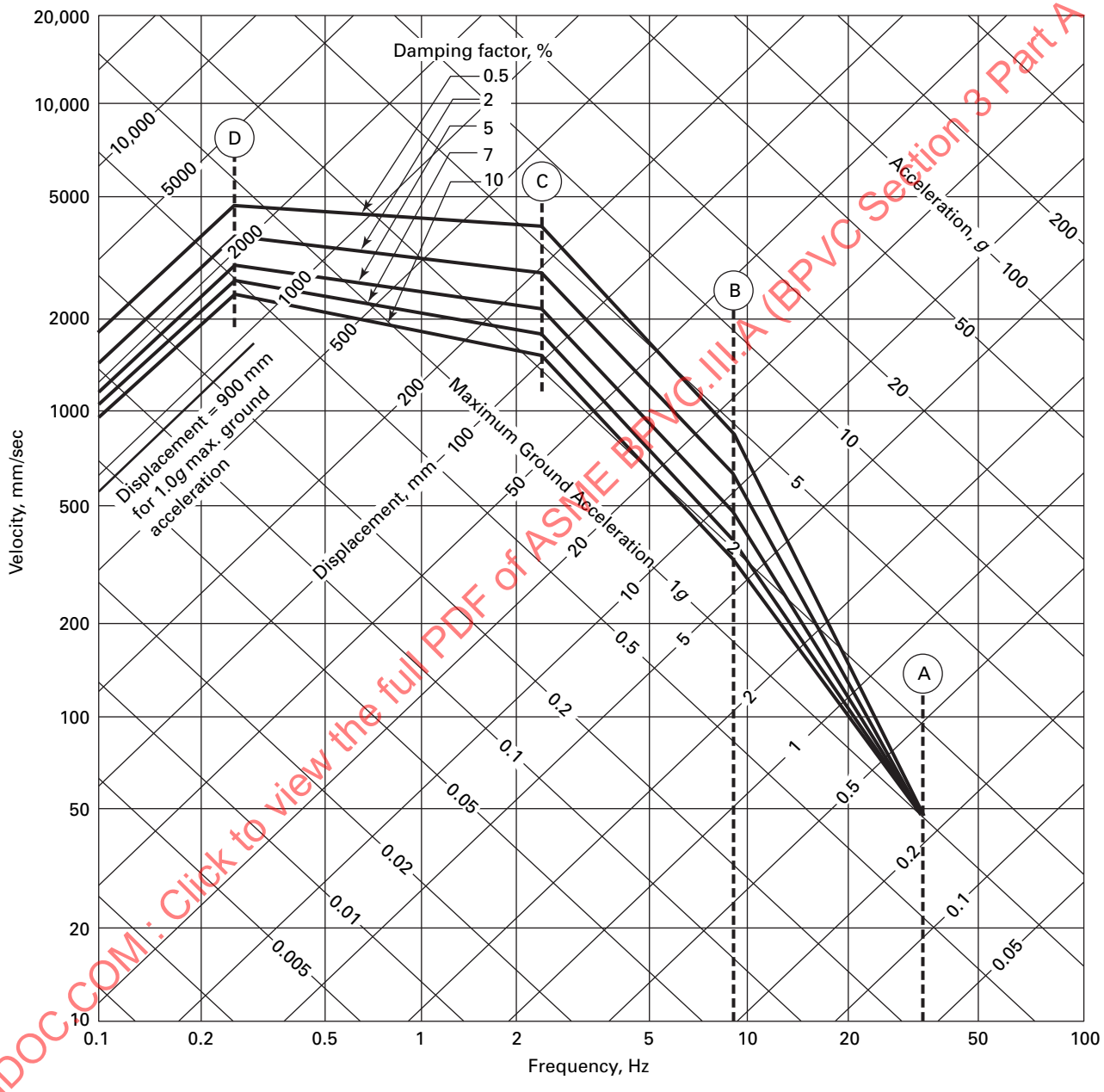

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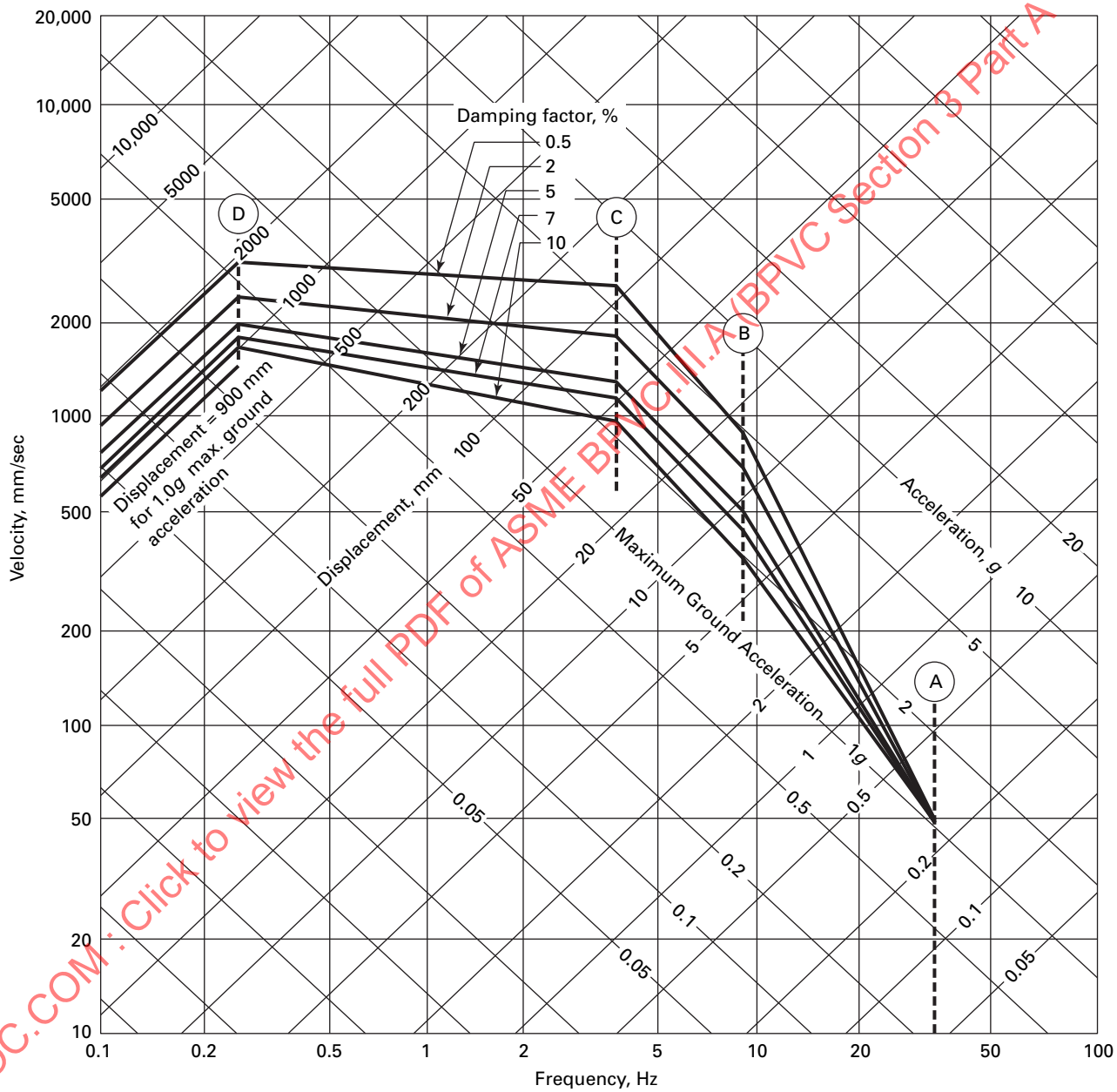
Figure N-1211(b)-1M
Vertical Design Response Spectra Scaled to 1*g* Horizontal Ground Acceleration



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with respect to the ground motion. For systems excited at multiple support locations, further modification of the force vector is necessary (see N-1228).

(c) The differential equation of motion may be solved either analytically or numerically. A rigorous analytical solution of the simultaneous linear differential equations is often impractical and unnecessary. Simplification of the damping matrix may be made to facilitate computations.

(d) For example, in the special case of proportional damping (i.e., when the damping matrix is reduced to a linear combination of the mass and stiffness matrices), the classical method of modal analysis may be applied since eq. (a)(1) can be decoupled.

(e) See N-1230 for acceptable treatments of damping. A numerical solution using step-by-step integration method may be applied to a set of coupled as well as uncoupled differential equations. The direct integration method does not require the process of uncoupling modes; therefore, no calculation of natural frequencies and mode shapes is necessary. The direct integration method permits the handling of proportional damping where this treatment is required.

N-1222.1.1 Method of Modal Superposition (Ref. [8]).

N-1222.1.1.1 Modal Analysis. The method of modal analysis usually assumes that a modal matrix be defined such that

$$\begin{aligned} [\phi]^{\mathrm{T}}[M][\phi] &= [I] \\ [\phi]^{\mathrm{T}}[K][\phi] &= [\omega^2] \\ [\phi]^{\mathrm{T}}[C][\phi] &= [2\xi\omega] \end{aligned} \tag{2}$$

where the right-hand sides are diagonal matrices, ϕ is the modal matrix, ω is the natural frequency of the structure, and ξ is the modal fraction of critical damping. This formation uses damping matrices that are proportional.

Let $\{\eta\}$ be the normal coordinates such that the displacement $\{x\}$ can be defined by the transformation

$$\{x\} = [\phi]\{\eta\} \tag{3}$$

Equation N-1222.1(a)(1) can be written by means of this transformation as

$$\{\eta''\} + 2\xi\omega\{\eta'\} + \omega^2\{\eta\} = [\phi]^{\mathrm{T}}\{f\} \tag{4}$$

which represents a decoupled system of individual modal equations. Response in each mode may be obtained by solving the differential equations individually for the corresponding mode. The total response of the structure is therefore the result of combining the responses of all its component modes.

The equations of motion for a linear system may be uncoupled by means of a transformation to a system of normal coordinates provided that the damping matrix C is a linear combination of M and K matrices.

N-1222.1.1.2 Complex Frequency Response Method. As an alternative approach to the method of modal analysis described above, the complex frequency response method may be used to determine dynamic response of the system represented by eq. N-1222.1(a)(1).

The complex frequency response method requires that the complex frequency responses of the system be determined first and the seismic excitations be transformed into its frequency domain. The time histories of the response may then be found by inverting the response transforms.

Corresponding to every normal coordinate η_r of eq. N-1222.1.1.1(4), a pair of Fourier transforms may be found:

$$\begin{aligned} G_r(\Omega) &= \int_{-\infty}^{\infty} \eta(t) e^{-i\Omega t}\, dt \\ Q_r(\Omega) &= \int_{-\infty}^{\infty} f_r(t) e^{-i\Omega t}\, dt \end{aligned} \tag{5}$$

where

$$f_r(t) = \{\phi_r\}^{\mathrm{T}}\{f(t)\}$$

The Fourier transforms of the excitation and response are related according to

$$G_r(\Omega) = H_r(\Omega)Q(\Omega) \tag{6}$$

and

$$H_r(\Omega) = \frac{1}{1 - (\Omega/\Omega_r)^2 + i2\xi(\Omega/\Omega_r)} \tag{7}$$

The rth mode response is given by evaluating the Fourier integral:

$$\eta_r(t) = \frac{1}{2\pi}\int_{-\infty}^{\infty} H_r(\Omega)Q_r(\Omega)e^{i\Omega t}\, d\Omega \tag{8}$$

and the time history response of the system in the physical coordinate is given by

$$\{x(t)\} = \sum_{r=1}^{\eta}\{\phi_r\}\eta_r(t) \tag{9}$$

Standard fast Fourier transform techniques are available for expedient evaluation of $Q_r(\omega)$ and $\eta_r(t)$ of eqs. (6) and (8), respectively (ref. [9]).

N-1222.1.2 Method of Direct Integration.

(a) This is a numerical method directly applied to the solution of the differential equations of motion of linear structural system, eq. N-1222.1(a)(1), in a step-by-step manner. No uncoupling procedure is necessary to compute the responses. The damping matrix need not be proportional.

(b) There are many acceptable schemes available for numerical integration of the equations of motion such as Newmark β-method (ref. [10]), Houbolt method (ref. [11]), and Wilson θ-method (ref. [12]).

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(c) Using the matrix formulation by Chan, Cox, and Benfield (ref. [13]) and Newmark β-method, eq. N-1222.1(a)(1) may be transformed into a finite difference equation in recurrence form involving displacement:

$$[D]\{x_{n+1}\} = [B]\{x_n\} - [F]\{x_{n-1}\} + \beta h^2 \left\{f_{n+1} + \left(\frac{1}{\beta} - 2\right)f_n + f_{n-1}\right\} \quad (10)$$

where

$[B] = 2[M] - (1 - 2\beta)h^2[K]$
$[D] = [M] + (h/2)[C] + \beta h^2[K]$
$[F] = [M] - (h/2)[C] + \beta h^2[K]$
h = length of the time interval used in the direct integration procedure
n = the number of time intervals lapsed
β = parameter of Newmark β-method in reflecting the type of acceleration function assumed between two time stations and numerical stability of the procedure

(d) The dynamic responses of a structure at any instant of time may be calculated from the time history of excitations and the previous structural responses.

(e) The direction integration method has the advantage of simplicity in computation and elimination of the eigenvalue problem. This method may also be extended to deal with the nonlinear problems.

N-1222.2 Nonlinear.

(a) The linear time history analysis discussed in N-1222.1 is based on the assumptions that external forces are independent of the displacement and velocity, the stress–strain relationship is linear, and the strain–displacement relationship is linear. Many problems of practical consequences exist for which these assumptions are not valid. Some acceptable numerical methods to analyze such nonlinear problems are discussed below.

(b) The following nonlinearities may be introduced into the system:

(1) material nonlinearities (plasticity);

(2) geometric nonlinearities (large displacement);

(3) combination of material and geometric nonlinearities (impact and friction).

(c) The differential equation of motion for nonlinear problems is

$$[M_t]\{U''\} + [C_t]\{U'\} + [K_t]\{U\} = \{P_t\} \quad (11)$$

where

C_t = damping
K_t = stiffness matrices
M_t = time-dependent mass
P_t = force arrays
U = displacement
U' = velocity
U'' = acceleration

The left-hand side of eq. (11) may be made linear by introducing

$$[M_t] = [M] + [M_{nl}]$$
$$[C_t] = [C] + [C_{nl}] \quad (12)$$
$$[K_t] = [K] + [K_{nl}]$$

where

C = proportional damping
C_{nl} = damping
K = stiffness matrices
K_{nl} = stiffness matrices
M = time-independent mass
M_{nl} = time-dependent mass

As an example, the compressible fluid flow in a pipe or a moving mass acting on a structure contributes to $[M_{nl}]$ and the closing and opening of gaps contribute to $[C_{nl}]$ and $[K_{nl}]$. Substituting eq. (12) into eq. N-1222.1(a)(1) yields

$$[M]\{U''\} + [C]\{U'\} + [K]\{U\} = \{F\} \quad (13)$$

where

$$\{F\} = \{P_t\} - [M_{nl}]\{U''\} - [C_{nl}]\{U'\} - [K_{nl}]\{U\}$$

Time history methods to solve eq. (11) or (13) fall into two major categories: (1) mode superposition, and (2) direct integration.

N-1222.2.1 Mode Superposition.

(a) The use of normal modes for linear time history method is discussed in N-1222.1. Mode superposition techniques may also be used for the nonlinear problems (refs. [14], [15], and [16]). A problem of a pendulum having a large amplitude is discussed in ref. [14]. Reference [15] presents an analysis of a structure including the following nonlinear characteristics: variable-stiffness elements, contact (compression only) elements, time-varying boundary conditions, and nonmodal and nonlinear damping elements. A ring problem involving large geometric nonlinearities and plastic strains is discussed in ref. [16].

(b) Let $\{\phi_i\}$ and $\{\omega_i\}$ be the ith orthonormal eigenvector and natural frequency, respectively:

$$[M]\{U''\} + [K]\{U\} = 0 \quad (14)$$

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A transformation

$$\{U\} = [\phi]\{q\} \tag{15}$$

is introduced in eq. N-1222.2(c)(13). The resulting uncoupled equation is as follows:

$$[I]\{q''\} + \left[2\xi_j\omega_j\right]\{q'\} + \left[\omega_j{}^2\right]\{q\} = \{P\} \tag{16}$$

where

I = identity matrix
= $[\phi]^{\mathrm{T}}[M][\phi]$
P = generalized force
= $[\phi]^{\mathrm{T}}\{F\}$
q = generalized displacement vector
ξ_j = jth modal damping ratio
ω_j = jth natural frequency
ϕ = set of significant eigenvectors

(c) Eq. (b)(16) may be solved by either explicit numerical integration schemes such as the fourth order Runge–Kutta method (ref. [17]), Hamming's predictor–corrector method (ref. [17]), or by the analytical integration scheme (ref. [14]). These numerical methods are conditionally stable and available as library programs equipped with an automatic time step adjustment feature. These methods provide a stable solution if the integration time step is appreciably smaller than the smallest period associated with eq. (b)(16). In the analytical integration method, the force vector F is approximated over the time step Δt. This allows for Duhamel integral type solutions for eq. (b)(16).

(d) The generalized forces in eq. (b)(16) are updated at each time increment. If the initial stiffness matrix has a relatively large bandwidth and a large number of eigenvectors are not required for the analysis, it is likely that the mode superposition approach will be quite economical as compared with the direct integration procedures discussed in N-1222.2.2. In cases where the applied load distribution is extremely complex or its time variation contains significant high frequency components, or both, it is necessary to include many modes to obtain adequate accuracy by mode superposition. In these cases the direct integration procedure may be more efficient. This procedure is discussed in N-1222.2.2.

N-1222.2.2 Direct Integration. Direct numerical integration methods can be applied to eq. N-1222.2(c)(11) or eq. N-1222.2(c)(13). When applied to eq. N-1222.2(c)(11), the calculation of time-dependent matrices is necessary at each integration time step. When applied to eq. N-1222.2(c)(13), the time-independent matrices may be calculated once at the beginning of the numerical solution and the time-dependent force array is calculated at each time step. These methods are classified into two groups: explicit schemes and implicit schemes. The explicit schemes are hampered by both numerical instability and convergence problems while the implicit schemes suffer only from convergence problems. These schemes are discussed in N-1222.2.2.1 and N-1222.2.2.2.

N-1222.2.2.1 Explicit Schemes. These schemes convert the differential equations of motion to a set of linear algebraic equations with unknown state variables (at the present time) which are independent of one another. Acceptable explicit schemes include: Runge–Kutta method (ref. [17]), predictor–corrector method (ref. [17]), Nordsieck integration method (ref. [18]), and central difference method (ref. [19]).

As the above methods are conditionally stable, they have a disadvantage of requiring small time step sizes. The first three methods can be made to operate with the variable time step, while the fourth one is a constant time-step procedure. Garnet and Armen (ref. [20]) demonstrate a one dimensional wave propagation problem solution with the aid of a predictor–corrector method, i.e., the modified Adams method. In ref. [21] the Nordsieck integration scheme is demonstrated to solve nonlinear vibration problems of reactor components. Wu and Witmer (ref. [22]) analyzed the problem of large transient elastic–plastic deformation of structures using the central difference method.

N-1222.2.2.2 Implicit Schemes. These schemes convert the differential equations of motion to a set of linear simultaneous algebraic equations and require a matrix inversion to step the solution forward. Acceptable implicit schemes include: Newmark's generalized acceleration method, Wilson's θ method, Nastran's integration method, and Houbolt's method.

(a) Acceptable implicit schemes:

(1) Newmark's Generalized Acceleration Method (Refs. [10] and [23]). The nodal point velocities and displacements are given by the following equations:

$$\{U'_{n+1}\} = \{U'_n\} + (1-\gamma)(\Delta t)\{U''_R\} + \gamma(\Delta t)\{U''_{n+1}\} \tag{17}$$

and

$$\{U_{n+1}\} = \{U_n\} + (\Delta t)\{U'_n\} + (1/2-\beta)(\Delta t)^2\{U''_n\} + \beta(\Delta t)^2\{U''_{n+1}\} \tag{18}$$

Equations (17) and (18) are substituted in eq. N-1222.2(c)(13), which is expressed at the present time point $(n+1)$. This gives the following:

$$\left[[M]+\gamma(\Delta t)[C]+\beta(\Delta t)^2[K]\right]\{U''_{n+1}\} = \{F_{n+1}\} - \left[(1-\gamma)(\Delta t)[C]+(1/2-\beta)(\Delta t)^2[K]\right]\{U''_R\} - \left[[C]+(\Delta t)[K]\right]\{U'_n\} - [K]\{U_n\} \tag{19}$$

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This set of algebraic simultaneous equations in the unknown accelerations is solved and used with eqs. (17) and (18) to obtain the velocity and displacement at the present time. The parameter γ in eq. (19) is a damping parameter. Artificial positive damping is introduced if $\gamma > 0.5$ and artificial negative damping if $\gamma < 0.5$. For linear problems, the method is unconditionally stable if $\beta > (2\gamma + 1)^2/16$ (ref. [12]). For $\gamma = 1/2$, $\beta > 1/4$ gives unconditional stability. For systems with nonlinearities or nonproportional damping or both, there is no analytical expression available for unconditional stability. So to provide a margin of stability for these systems $\beta > 1/4$ is considered for $\gamma = 1/2$. The value of γ slightly larger than 0.5 damp out the highest (and least important) modes while preserving the lower ones. The Newmark method for $\gamma = 1/2$ (no numerical damping) and $\beta = 1/6$ (linear acceleration) is conditionally stable. The Wilson modified method (to make it unconditionally stable) is discussed next.

(2) Wilson's θ Method (ref. [12]). To obtain the solution at the present time $(n + 1)$, this method assumes that the acceleration varies linearly over the time interval $\tau = \theta\Delta t$, where $\theta > 1.0$. With the aid of eq. (1)(19), $\beta = 1/6$ and $\gamma = 1/2$, solution at time point $(n + \theta)$ is obtained. Then the acceleration, velocity, and displacement at the time point $n + 1$ are given by the following expressions:

$$\begin{aligned}
\{U''_{n+1}\} &= (1 - 1/\theta)\{U''_n\} + (1/\theta)\{U''_{n+\theta}\} \\
\{U'_{n+1}\} &= \{U'_n\} + \frac{\Delta t}{2}\{U''_n + U''_{n+1}\} \\
\{U_{n+1}\} &= \{U_n\} + \Delta t\{U'_n\} + \frac{\Delta t^2}{6}(U''_{n+1} + 2U''_n)
\end{aligned} \tag{20}$$

The last two equations are obtained from eq. (1)(18) with the value of β and γ specified above. This method is unconditionally stable for $\theta > 1.37$ (ref. [12]); for $\theta = 1.4$, the numerical damping is less than 1% provided 22 steps are taken within the natural period of the mode of interest (ref. [25]).

(3) Nastran's Integration Method (ref. [26]). In this method the acceleration and velocity are expressed in central finite difference form. They are expressed as follows:

$$\begin{aligned}
\{U''_n\} &= \frac{\{U_{n+1}\} - \{2U_n\} + \{U_{n-1}\}}{(\Delta t)^2} \\
\{U'_n\} &= \frac{\{U_{n+1}\} - \{U_{n-1}\}}{2(\Delta t)}
\end{aligned} \tag{21}$$

The displacement and the forcing function are expressed as the weighted average of the corresponding magnitudes at the three successive time points $n - 1$, n, $n + 1$. Substitution of these expressions in eq. N-1222.2(c)(13) expressed at time point n gives:

$$\begin{aligned}
&\left[\frac{1}{(\Delta t)^2}[M] + \frac{1}{2(\Delta t)}[C] + \frac{1}{3}[K]\right]\{U_{n+1}\} = \\
&\frac{1}{3}\{F_{n+1} + F_n + F_{n-1}\} + \\
&\left[\frac{2}{(\Delta t)^2}[M] - \frac{1}{3}[K]\right]\{U_n\} + \\
&\left[-\frac{1}{(\Delta t)^2}[M] + \frac{1}{2(\Delta t)}[C] - \frac{1}{3}[K]\right]\{U_{n+1}\}
\end{aligned} \tag{22}$$

This set of algebraic simultaneous equations in the unknown displacements is solved and used to find velocity and acceleration.

This scheme is a special form of the Newmark method, eq. (1)(18) for $\beta = 1/3$ and $\gamma = 1/2$ (ref. [13]). The value $\beta = 1/3$ provided some margin of stability and for $\gamma = 1/2$, the artificial damping is zero. Note that the load vector F is averaged over three adjacent time steps in the same manner that K is averaged. This is done in order to provide statically correct solutions for massless degrees of freedom.

(4) Houbolt's Method (ref. [11]). In this method a third-order interpolating polynomial which fits the known displacements at the three previous time points and the unknown displacement at the present time point is formulated. The expressions for acceleration and velocity are given as follows:

$$\begin{aligned}
\{U''_{n+1}\} &= \frac{1}{(\Delta t)^2}[2\{U_{n+1}\} - 5\{U_n\} + 4\{U_{n-1}\} - \{U_{n-2}\}] \\
\{U'_{n+1}\} &= \frac{1}{6(\Delta t)}[11\{U_{n+1}\} - 18\{U_n\} + 9\{U_{n-1}\} - 2\{U_{n-2}\}]
\end{aligned} \tag{23}$$

The eqs. (23) are substituted in eq. N-1222.2(c)(13) which is expressed at the present time point $n + 1$. This gives:

$$\begin{aligned}
&\left[\frac{2}{\Delta t^2}\left(\frac{11}{6\Delta t}\right)[C] + [K]\right]\{U_{n+1}\} = \{F_{n+1}\} + \\
&\left[\frac{5}{\Delta t^2}[M] + \frac{3}{\Delta t}[C]\right]\{U_n\} - \left[\frac{4}{\Delta t^2}[M] + \frac{3}{2\Delta t}[C]\right]\{U_{n-1}\} \\
&+ \left[\frac{1}{\Delta t^2}[M] + \frac{1}{3\Delta t}[C]\right]\{U_{n-2}\}
\end{aligned}$$

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This set of algebraic simultaneous equations in the unknown displacements is solved and used to find velocity and acceleration.

This method is unconditionally stable for linear problems and introduced artificial damping (the amount of such damping increases with the ratio of time step to natural period of the system). The artificial damping is less than 1% provided 50 time steps are taken within the natural period of the mode of interest (ref. [12]). Thus, the Houbolt method effectively removes higher mode response from the system.

(b) Approximation of Load Vectors. All four direct integration methods discussed in the previous section require that the forces at the time point $n + 1$ be known in order to calculate the displacements at that time. These loads, because of the presence of the nonlinear terms, are a function of the displacements which are to be calculated. So, it is not possible to evaluate these terms exactly. These forces may be approximated with the aid of the first-order Taylor series expansion about the motion at time point n as follows:

$$\left\{F(t,U)_{n+1}\right\} = \left\{F(t,U)_n\right\} + \frac{\delta}{\delta t}\left\{F(t,U)_n\right\}\left(\Delta t\right)$$
$$= 2\left\{F(t,U)_n\right\} - \left\{F(t,U)_{n-1}\right\}$$

This expression has an inherent error of order $(\Delta t)^2$ which is the same as the Houbolt and Nastran method (ref. [27]). This expression corresponds to a linear extrapolation of the loads at the two previous time increments which is equivalent to a numerical differentiation. This introduces round-off error (noise) which will set a low bound on the integration time step.

(c) Convergence. For nonlinear systems, the following two steps are recommended to ensure convergence of the solution to the proper value.

(1) Iterative schemes, based on the residual force array derived from the equation of motion at the present time point $n + 1$ may be used to obtain convergence. This residual, obtained by transferring all the terms in eq. N-1222.2(c)(13) to the right-hand side, is a measure of how well the dynamic equilibrium is satisfied at the present time point $n + 1$. The time-dependent matrices and force arrays, in eqs. N-1222.2(c)(11) and N-1222.2(c)(13), respectively, are modified based on the calculated solution at time point $n + 1$ and the iterative scheme is continued until the dynamic equilibrium is satisfied to a prescribed tolerance. References [25], [28], and [29] discuss this approach to improve efficiency in nonlinear solutions.

(2) Successive computation of the time history, employing successively smaller values of the integration time step, may also be used to ascertain convergence.

In the wave propagation problems, the response may contain the spurious oscillations (refs. [20] and [25]), which are due to the finite element discretization, and are not eliminated by reducing the time step Δt. These oscillations may often be reduced by employing more uniform element size.

(d) General Remarks. No best choice of numerical method has been identified for nonlinear problems. Discussions which are helpful to indicate the advantages and limitations of these previously discussed methods are contained in refs. [13], [20], [22], [23], and [27].

Experience obtained from the solution of actual problems is the most reliable indicator of which of the different integration schemes is superior for a particular problem. Experience and checks of methods adopted for a similar type of physical problem where a correct solution is available is recommended to establish validity for a particular nonlinear method.

N-1222.3 Time History Broadening. To account for the effect of possible frequency variation of the structure, the same time history data may be used with at least three different scaled time intervals: Δt (the reference interval) and $(1 \pm \Delta f_j/f_j)\Delta t$, for the analysis of equipment, where f_j is the fundamental structural frequency and Δf_j is a parameter defining the frequency variation due to uncertainties. This variation of the time scale interval has a similar effect to widening the spectral peak when generating the smoothed response spectrum. If one of the equipment frequencies f_e is within the range $f_j \pm \Delta f_j$, it is recommended that the time history also be used with additional scaled time intervals of $[1 \pm (f_e - f_j)/f_j]\Delta t$.

N-1223 Response Spectrum Method

N-1223.1 Modal Combination. In the response spectrum method, the peak values of particular responses of interest (displacement, acceleration, shear, moment, etc.) are determined for each mode. The total response may be obtained by combining the peak modal responses by the square root of sum of squares (SRSS) method. Mathematically this is expressed as follows:

$$R = \pm\left[\sum_{k=1}^{n} {R_k}^2\right]^{\frac{1}{2}} \quad (24)$$

where

R = the total response of interest (strain, displacement, stress, moment, shear, etc.) based on the SRSS method of combining the individual peak modal responses
R_k = the peak response of interest due to the kth mode
n = the number of significant modes

The above method may be used to determine the particular response for all significant modes regardless of frequency spacing.

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N-1223.2 Combination of Effects Due to Triaxial Excitation.

(a) In the response spectrum method of analysis, the natural frequencies, mode shapes, and the load for each mode are first determined. The load for each mode for unit generalized response is the product of the stress matrix, the mode shapes, and the mode participation factors. When this product is multiplied by the generalized response determined from the spectrum curves, the load for each natural mode results.

(b) The following assumptions may be used in combining the loads of each natural mode and for each direction.

(1) The peak responses of the different modes due to any one excitation do not occur at the same time.

(2) The peak generalized responses due to the three different earthquake excitations for the same mode do not occur at the same time.

(3) The peak stresses due to different modes and due to different excitations generally do not occur at the same location on the structure nor at the same angular orientation.

(c) These assumptions are consistent with the use of the SRSS method to compute the resultant responses. It is necessary to use the SRSS method only on scalar components because the SRSS of components orthogonal to each other results in magnitudes equivalent to vector sums thus implying simultaneous occurrence.

(d) The following general procedure should be used to combine the seismic responses due to triaxial excitation:

$$R_{ij} = \pm\left[\sum_{k=1}^{3} R_{ijk}^{2}\right]^{\frac{1}{2}} \quad (25)$$

where

R_{ij} = seismic response of interest for design (strain, displacement, stress, moment, shear, etc.) obtained by the SRSS rule to account for the nonsimultaneous occurrences of R_{ijk}

R_{ijk} = maximum, co-directional seismic response of interest (strain, displacement, stress, moment, shear, etc.) associated with coordinates i and j due to earthquake excitation in the kth direction

N-1224 Component or Equipment Testing

In the course of preparation.

N-1225 Simplified Dynamic Analysis

N-1225.1 Seismic Load Coefficient Method for Piping System Analysis.

(15) **N-1225.1.1 Simplified Seismic Load Coefficient Method Using Floor or Amplified Response Spectra as the Seismic Design Basis Input.**

(a) Piping System Loads. A simplified seismic load coefficient method for earthquake resistant design for equivalent seismic inertia induced resultant piping stresses, displacements, loads, and support reaction loads may be used to describe the earthquake input with a format as follows:

$$F_{phi} = K_{hi} S_{ami} W \quad (26)$$

$$F_{pv} = K_{v} S_{ami} W \quad (27)$$

where

F_{phi} = equivalent static inertia force applied to the piping and piping components in the ith horizontal direction as defined in N-1225.2

F_{pv} = equivalent static inertia force applied to the piping and piping components in the vertical direction as defined in N-1225.2

K_{hi} = the load coefficient applied to the piping and piping components in the ith horizontal direction. The value of K_{hi} shall be determined as follows:
For Soil Sites: V_s < 3500 ft/sec (1100 m/s)

$K_{hi} = 1.0;\ l_{hi} < 3.5\ l_v$
$K_{hi} = 0.65;\ 3.5 l_v \le l_{hi} \le 5.0 l_v$
$K_{hi} = 0.4;\ l_{hi} > 5.0\ l_v$

For Rock Sites: V_s > 3500 ft/sec (1100 m/s)

$K_{hi} = 1.0;\ l_{hi} < 2.5\ l_v$
$K_{hi} = 0.6;\ 2.5 l_v \le l_{hi} \le 4.0 l_v$
$K_{hi} = 0.4;\ l_{hi} > 4.0\ l_v$

Piping systems with l_{hi} in more than one of the categories listed above shall use the highest K_{hi} value applied to two lateral support spans on either side of the span requiring the highest K_{hi} value.

K_v = the load coefficient applied to the piping in the vertical direction. The value of K_v shall be determined as follows:
For Soil Sites: V_s < 3500 ft/sec (1100 m/s)

$K_v = .75;\ l_{hi} < 4.0 l_v$
$K_v = 1.0;\ l_{hi} \ge 4.0 l_v$

For Rock Sites: V_s > 3500 ft/sec (1100 m/s)

$K_v = 1.0$

Piping systems with l_{hi} in more than one of these categories listed above shall use the highest K_v values applied to two lateral support spans on either side of the span requiring the highest K_v value.

l_{hi} = span in the ith direction between lateral supports measured along the axis of the pipe. Lateral supports added to the piping system next to large in-line components whose weights equal or exceed the weight of the pipe between nominal dead weight supports contained in Table NF-3611.1, such as valves, strainers, etc., do not need to be considered in the determination of l_{hi} value

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l_v = deadweight vertical support span from Table NF-3611.1 for the nominal pipe size for the piping system under consideration

S_{ami} = peak acceleration of applicable amplified or floor response spectra in the ith direction (in g's)

V_s = building foundation media shear wave velocity

W = the total piping dead load (weight) which exists during the postulated seismic event. This includes piping weight, water weight and insulation. The units of the term W must be consistent with those of the term F_{phi} and F_{pv} above.

When the piping system under consideration contains more than one pipe size K_{hi} and K_v factors shall be separately determined for and applied to each nominal pipe size in the piping system. In addition, if the K_{hi} and K_v values change as a result of a pipe size change, the highest K_{hi}, K_v values at the point of piping size change shall be applied to the first two lateral support spans at the point piping size change having the lower K_{hi}, K_v values.

(b) Special Considerations for Support Reactions. In using this method to determine piping supports loads for seismic events, the specification of a minimum lateral and vertical support design load is required. This minimum design load is a function of pipe size and shall be determined as follows:

$$R_{Lmin} = (\text{MSLF})(w')\left[\sum_{i=1}^{2}(S_{ami})^2\right]^{\frac{1}{2}} \quad (28)$$

$$R_{Amin} = (\text{MSLF})(w')\left[\sum_{i=1}^{2}(S_{ami})^2\right]^{\frac{1}{2}} \quad (29)$$

where

S_{ami} = values that are the peak accelerations of the applicable amplified or floor response spectra in each of two orthogonal horizontal directions (units are g's)

$$R_{V\min} = (\text{MSLF})(w')(S_{ami})$$

where

S_{ami} = peak acceleration of the applicable amplified or floor response spectra in the vertical direction.

The remaining parameters are defined as follows:

$MSLF$ = the Minimum Support Load Factor as defined in Table N-1225.1.1(b)-1 [MSLF has the units of ft (m)]

$R_{A\min}$ = Minimum Axial Seismic Support Design Load for these axial supports not in the vertical direction. For axial supports in the vertical direction the minimum support design load shall be determined as $R_{V\min}$. [$R_{A\min}$ has units of lb_f (N)].

$R_{L\min}$ = Minimum Lateral Seismic Support Design Support Load [$R_{L\min}$ has the units of lb_f (N)]

$R_{V\min}$ = Minimum Vertical Seismic Support Design Load [$R_{V\min}$ has the units of lb_f (N)]

w' = the total piping dead load per unit length of the piping which exists during a seismic event. This includes piping weight, water weight, insulation weight, etc. [w' has the units of lb_f/ft (N/m)].

In instances where Seismic Support Design Loads determined using the methodology of N-1225.1(a) are less than $R_{L\min}$, $R_{A\min}$, or $R_{V\min}$, the support design load shall be specified as $R_{L\min}$, $R_{A\min}$, or $R_{V\min}$, as determined in this subarticle.

(c) Inline Component Extended Structures. The seismic applied accelerations at the center of gravity (cg) of inline component extended structures may be determined as follows:

$$av_h = \left[\sum_{i=1}^{2}(0.9S_{ami})^2\right]^{\frac{1}{2}} \quad (30)$$

when the S_{ami} values are the peak accelerations of the applicable amplified or floor response spectra in each of two orthogonal horizontal directions.

$$av_v = 1.2\,S_{ami} \quad (31)$$

when the S_{ami} value is the peak of the applicable amplified floor response spectra in the vertical direction.

The remaining parameters are defined as follows:

av_h = the total acceleration applied to the inline component extended structure center of gravity (cg) in the horizontal direction

av_v = the acceleration applied to the inline component extended structure center of gravity (cg) in the vertical direction

Table N-1225.1.1(b)-1
Minimum Support Load Factor

Nominal Pipe Size, ϕ, NPS (DN)	MSLF, ft (m)
$\phi \le 2$ (50)	8.5 (2.6)
2 (50) < $\phi \le 4$ (100)	8.0 (2.4)
4 (100) < $\phi \le 6$ (150)	7.5 (2.3)

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The application of the eqs. (30) and (31) to determine Inline Component Extended Structures applied accelerations shall be limited to $W_v\ h^2/w'$ values less than or equal to 185,000 ft-in.^{2} (36.4 × 10^6 m·mm^2), when:

h = height from the pipe centerline to inline component extended structure center of gravity (cg) [in. (mm)]
W_v = weight of the inline component extended structure considered at its center of gravity (cg) location [lbf (N)]
w' = is as previously defined

For inline components when the extended structure vertical (axial) axis does not correspond to global vertical axis, the method is still applicable but care must be taken to insure that S_{ami} and eqs. (30) and (31), when applied, correspond to the inline component local vertical (axial) and lateral axes.

This method for inline piping component extended structure acceleration prediction is applicable only to those inline piping components where the extended structure of the inline component has a frequency greater than 20 Hz. The inline piping component frequency determination can be made by analysis, test, or judgment, and when making this frequency determination, the inline piping component body shall be or shall be assumed to be rigidly constrained.

(d) This method is limited to NPS 6 (DN 150) and smaller.

N-1225.2 Analysis Using the Seismic Load Coefficient Method. When seismic forces F_{phi} or F_v are determined from the equations of N-1225.1.1(a), they shall be assumed as statically applied inertia loads acting along three orthogonal axes defined relative to the piping system with one of these axes being in the vertical direction. The piping system seismic analysis shall be performed independently for each direction of excitation with the loads applied in the same sense. Resultant internal forces, moments or stresses in the piping products and piping supports determined from seismic loads in the three directions, may be combined by the square root sum of squares basis and will be considered as having both a (+) and (−) sign. In developing the piping system analytical models for this method, consideration shall be given to appropriately simulating the effects of localized concentrated mass effects assumed from such piping products as flanges, inline instrumentation, valves (including extended structures), etc.

The seismic forces defined by the equations of N-1225.1.1(a) consider only inertia effects. They do not include the effects of relative or differential motions between piping supports and motions of equipment to which the piping is attached. These effects are typically referred to as seismic anchor motions.

For this piping analysis method the piping supports may be assumed to be zero displacement points in the direction of restraint.

N-1226 Floor Spectrum Generation

(a) Nuclear facility structures are approximated by mathematical models to permit analysis of responses due to earthquake motions. Considering the large number of degrees of freedom that would be necessary and the possible ill-conditioning of the resulting stiffness matrix if the complete plant were idealized as a single mathematical model, the plant is usually separated into several separate subsystems for analysis purposes. There will usually be one or more primary structural models which support one or more secondary systems. Also, different models of the same structure may be required for different purposes. Specifically, the dynamic model used to generate the seismic excitation data for subsequent, separate analyses of the secondary systems may not be suitable for the detailed, localized stress analysis of the primary structure.

(b) Most equipment will have negligible interaction effects on the primary structure as in the case of equipment with relatively small mass and high frequency, and will only need to be included in the mass distribution of the primary system model. There are, however, major equipment systems, such as a reactor coolant system, whose stiffness, mass, and resulting frequency range should be considered for representation in the building model to account for possible dynamic interaction effects. Some guidelines for determining the extent of interaction are given in refs. [31] through [33].

(c) Equipment may be analyzed by combining the complete equipment model with the support structure model and applying the proper excitation to the base of the support structure. In this method no separate equipment support excitations need be generated because the equipment will be excited directly through the structure (ref. [34]).

(d) For most equipment a separate analysis of the secondary system may be performed using output from the building analysis. If building to equipment interaction is significant, then the equipment should be included in the mathematical model of the structure. The representation of the equipment that is included in the building model should be adequate to consider major interaction effects, but need not be as detailed as the mathematical model used in a separate analysis of the equipment.

(e) For equipment which is not analyzed as part of the building structural model, the response may be obtained by separate analysis using floor response spectra curves, time history excitations, or an appropriately defined power spectral density function at the support locations of the equipment in the structure.

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N-1226.1 Response Spectra.

(a) Both horizontal and vertical response spectra may be computed from the time history motions of the structure at the various floors or other equipment support locations of interest (ref. [35]). The spectrum ordinates should be computed at sufficient frequency intervals to produce complete and accurate response spectra. Spectrum peaks would normally be expected to occur at the frequencies of the peaks on the ground motion spectrum and at the natural frequencies of the supporting structures. In cases involving equipment mounted on equipment, the frequencies of all supporting structures should be included.

(b) Table N-1226-1 provides some systematic methods which may be used for choosing spectrum frequencies. Another acceptable method is to choose a set of frequencies such that each frequency is within 10% of the previous one and then add the natural frequencies of the supporting structures to the set.

(c) A more simplified acceptable method for constructing floor response spectra from the ground spectrum is given in refs. [36] and [37].

N-1226.2 Spectral Peaks. Studies (refs. [38] and [39]) have shown that, in general, the calculation of response of a piece of equipment having a frequency equal to a frequency of the supporting structure will be conservative if the spectrum at the equipment support is generated using a model of the supporting structure that does not include the equipment. This is because the exact resonance of an uncoupled analysis is not possible in the coupled case where the resonant frequencies tend to shift away from each other when the models are coupled. Further, it has been shown in ref. [39] that undamped peak amplification for singly supported equipment structures cannot exceed a maximum $\sqrt{J/m}$, where J is the modal mass for each mode of the primary structure and m is the mass of the supported subsystem. Since this limit is a function of the mass of the particular subsystem, the peak response may be calculated on a case-by-case basis. This rule is recommended only for relatively simple systems.

N-1226.3 Spectrum Peak Broadening.

(a) To account for the effect on structural frequency variation of the possible uncertainities in the material properties of the structure and soil, and the approximations in the modeling techniques used in seismic analysis, the initially computed floor response spectra are usually smoothed, and peaks associated with the structural frequencies are widened. A recommended method of determining the amount of peak widening, associated with the structural frequency, is described below.

(b) Let f_j be the jth mode structural frequency that is determined from the structure model. The variation in the structural frequency is determined by evaluating the individual frequency variation due to the variation in each parameter that is of significant effect, such as the soil modulus, material density, etc. The total frequency variation $\pm\Delta f_j$ is then determined by taking the SRSS of a minimum variation of $0.05f_j$ plus the individual frequency variation $(\Delta f_j)_n$, that is

$$\Delta f_j = \sqrt{\left(0.05f_j\right)^2 + \sum\left(\Delta f_j\right)_n^2}$$

(c) A value of $0.10f_j$ is recommended if the actual computed value of Δf_j is less than $0.10f_j$. Figure N-1226-1 shows a sample of such a smoothed floor spectrum curve. Note that the broadened peak is bounded on each side by lines which are not vertical but parallel to the lines forming the original spectrum peak.

(d) An alternative method for the broadening of the structural peaks can be based on a probabilistic approach, as discussed in ref. [40]. In the particular case where

Table N-1226-1
Suggested Frequencies, Hz, for Calculation of Ground and Floor Response Spectra

Ground Spectra [Note (1)]		Floor Spectra [Note (2)]	
Frequency Range	Increment, Hz	Frequency Range	Increment, Hz
0.5–3.0	0.10	0.5–1.6	0.10
3.0–3.6	0.15	1.6–2.8	0.20
3.6–5.0	0.20	2.8–4.0	0.30
5.0–8.0	0.25	4.0–9.0	0.50
8.0–15.0	0.50	9.0–16.0	1.0
15.0–18.0	1.0	16.0–22.0	2.0
18.0–22.0	2.0	22.0–34.0	3.0
22.0–34.0	3.0	...	...

NOTES:
(1) Calculate response at all frequencies within the ranges shown, at the corresponding increments (results in 72 frequencies).
(2) Calculate response at all frequencies within the ranges shown, at the corresponding increments (results in 46 frequencies); also calculate response at all natural frequencies of the supporting structures with the overall range.

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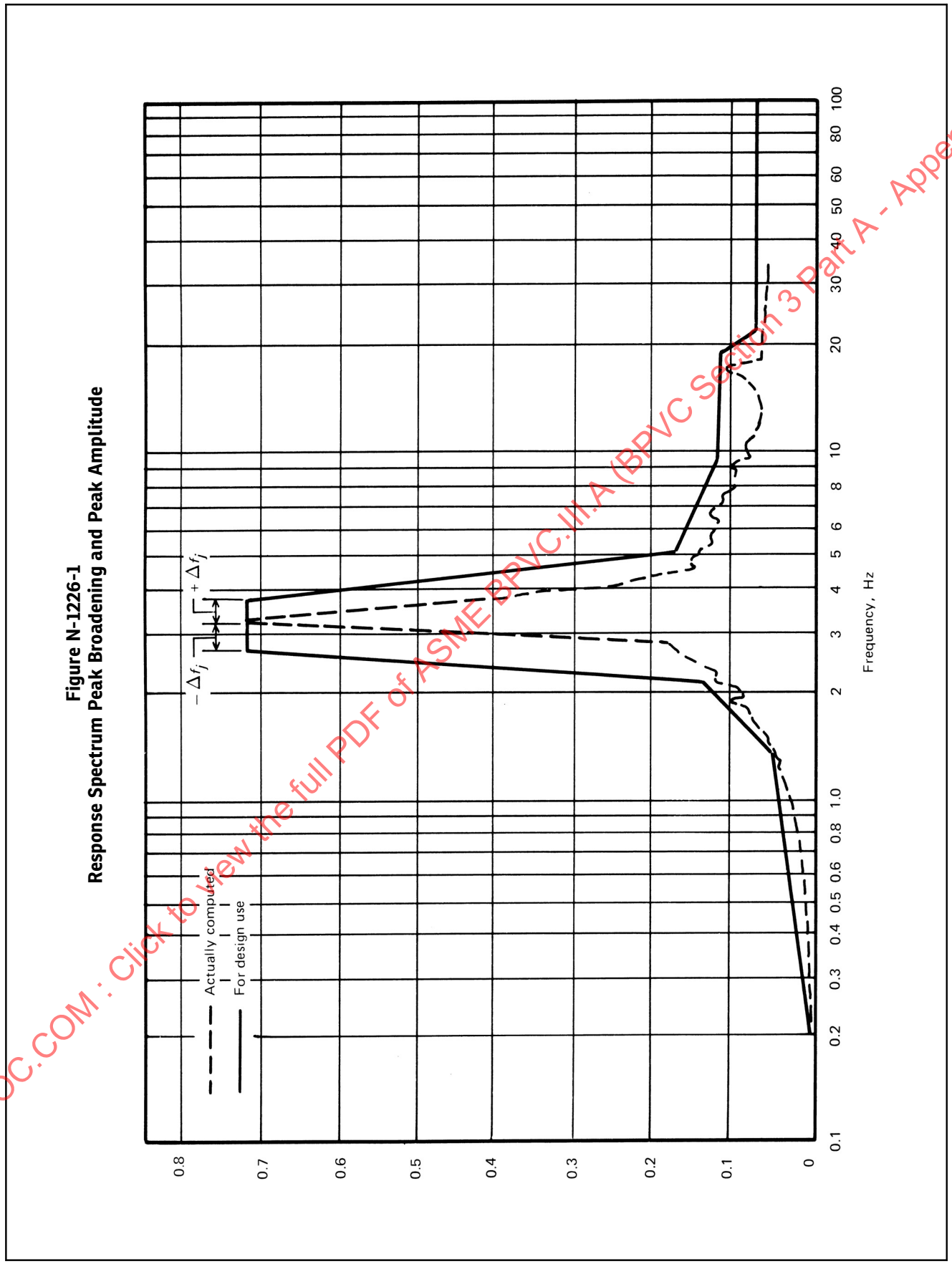

Figure N-1226-1
Response Spectrum Peak Broadening and Peak Amplitude

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there is more than one equipment or piping frequency located within the frequency range of a widened spectrum peak that is associated with a structural frequency f_j, the floor spectrum curve may be more realistically applied in accordance with the following criterion. Based on the fact that the actual natural frequency of the structure can possibly assume only one single value within the frequency range defined by $f_j \pm \Delta f_j$, but not a range of values, only one of these equipment or piping modes can respond with a magnitude indicated by the peak spectral value.

Therefore, seismic analysis of the equipment or piping systems using the broadened floor design response spectra may be accomplished by the following alternative method which can be used for equipment when analytical techniques are justified for predicting the natural frequencies.

Determine the natural frequencies $(f_e)_n$ of the system to be qualified in the broadened range of the maximum spectrum acceleration peak.

If no equipment or piping system natural frequencies exist in the ±15% interval associated with the maximum spectrum acceleration peak, then the interval associated with the next highest spectrum acceleration peak shall be selected and used in the following procedure.

Consider all N natural frequencies in the interval

$$f_j - 0.15f_j \le (f_e)_n \le f_j + 0.15f_j$$

where

f_j = the frequency of maximum acceleration in the unbroadened spectra
n = 1 to N

The system shall then be evaluated by performing $N + 3$ separate analyses using the unbroadened floor design response spectrum and the unbroadened spectrum modified by shifting the frequencies associated with each of the spectral values by a factor of +0.15; −0.15, and

$$\frac{(f_e)_n - f_j}{f_j}$$

where

n = 1 to N

The resultants of these separate seismic analyses shall then be enveloped to obtain the final resultant desired (e.g., stress, support loads, acceleration, etc.) at any given point in the system. If three different floor spectrum curves are used to define the response in the two horizontal and the vertical directions, then the shifting of the spectral values as defined above may be applied to these three spectrum curves.

The criterion is illustrated by the following example. Figure N-1226-2, sketch (a) represents the peak broadening on the floor spectrum curve associated with the jth mode structural frequency. Let there be two equipment or piping frequencies $(f_e)_1$ and $(f_e)_2$ that are within the frequency interval of $f_j \pm 0.15f_j$, Figure N-1226-2, sketch (b). Thus N equals two and therefore five separate floor response spectra must be considered. The unbroadened floor response spectrum as indicated in Figure N-1226-2, sketch (a) would be the first floor spectrum considered. The unbroadened floor response spectrum modified by shifting all of the frequencies associated with the spectral acceleration values by a factor of −0.15 is illustrated in Figure N-1226-2, sketch (c). In Figure N-1226-2, sketches (d) and (e) illustrate the modifications made to the unbroadened floor response spectrum by shifting a factor of

$$\frac{(f_e)_1 - f_j}{f_j} \quad \text{and} \quad \frac{(f_e)_2 - f_j}{f_j}$$

respectively.

Figure N-1226-2, sketch (f) illustrates the modifications made to the unbroadened floor response spectrum by shifting by a factor of +0.15. For this example, the five separate seismic analyses are performed with the individually modified floor response spectrum curves.

The required resultants, stresses, support loads, accelerations, moments, forces, etc., associated with each of the five separate seismic analyses are then obtained. As any of the five modified spectra has an equal probability of occurrence, the resultant utilized to evaluate the seismic loadings shall be the envelope values from the five separate seismic analyses.

N-1227 Multiple Input Response Spectra Analysis

If a structural system is being analyzed by the response spectra method and is supported at intermediate locations with different characteristic response spectra, then the dynamic analysis performed should take into consideration the different response spectra. The effect of relative seismic anchor displacements between the intermediate anchor points should also be considered.

N-1227.1 Inertial Effects Due to Multiple Response Spectra Input. The effect on a structural system of multiple supports with different characteristic response spectra may be accounted for by selecting a single spectra which will effectively produce the critical maximum responses due to different acceleration existing at anchor points and intermediate restraint points of the system. This may be conservatively accomplished by enveloping the response spectra for the different seismic anchor and restraint locations. Other procedures may be used, if applicable. Acceptable alternative procedures are given in refs. [7], [43], [44], and [45].

For a structural system which is restrained in different buildings, the response spectra of the restraint point should be enveloped.

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Figure N-1226-2
Use of Floor Spectra When Several Equipment Frequencies Are Within the Widened Spectral Peak

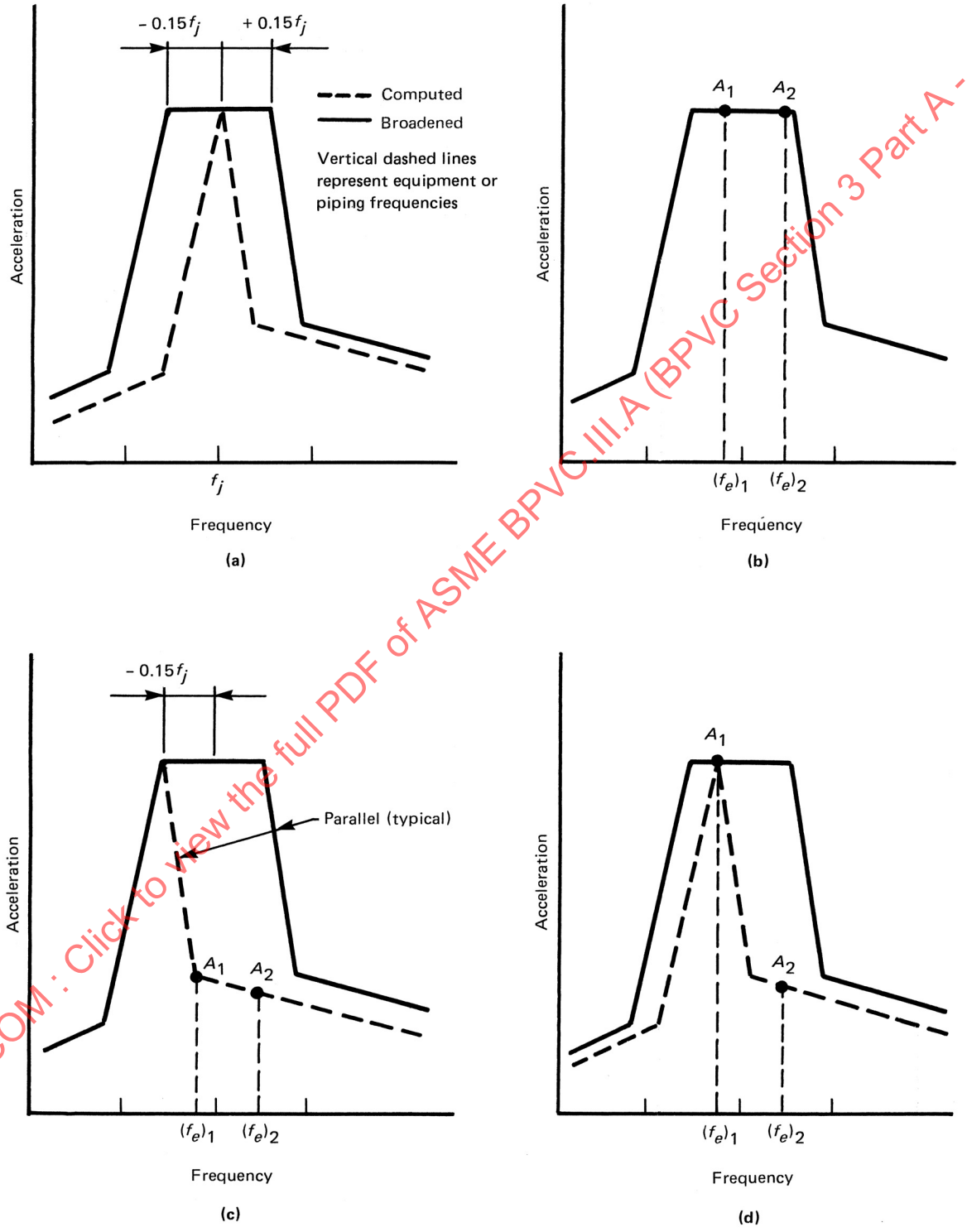

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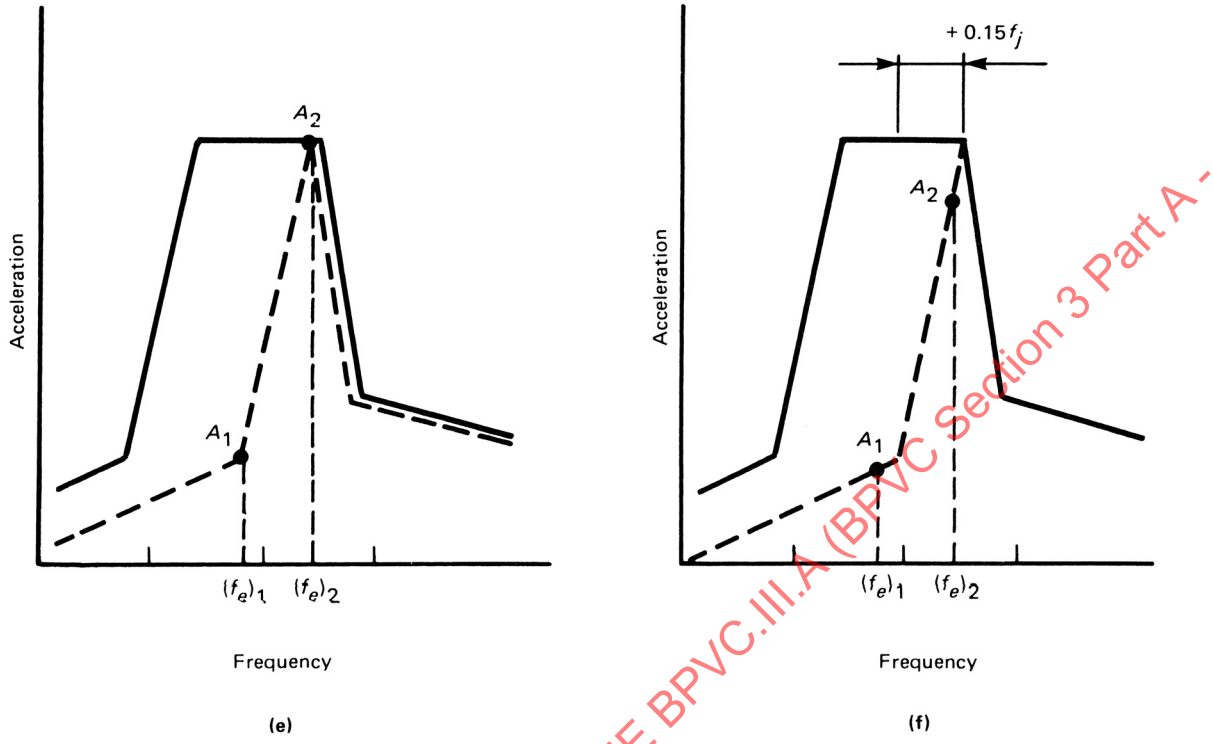
Figure N-1226-2
Use of Floor Spectra When Several Equipment Frequencies Are Within the Widened Spectral Peak (Cont'd)



N-1227.2 Stresses Due to Relative Seismic Support Displacements.

(a) When determining stresses the effects of relative seismic support movements should be considered. When these effects are considered significant, they may be obtained by performing a static structural analysis of the system including anchor movements. For structural systems with several possible combinations of anchor movements, any of the following methods may be used for determining the stresses due to relative support displacements.

(1) Stresses resulting from the differential movement of supports may be calculated for each significant mode using the modal displacement of the supports obtained from the structural response calculations. The maximum model responses are then combined following any of the acceptable methods described in this Appendix. If no support structure modal information is available, then the absolute sum of the displacements may conservatively be used.

(2) If the system is supported on independent structures, then the absolute sum of displacements in each direction should be used unless more detailed analyses are used in accordance with N-1700.

(3) If a displacement time history of the supporting structure is available, the procedures given in N-1228.4 may be used.

(b) Other detail considerations may be specified for dynamic analysis of Section III components, such as NB-3600 for piping, and these component related rules should be considered in the dynamic analysis.

N-1228 Multiple Time History Excitations

For structural systems supported at several locations, the responses due to simultaneous excitations by different motions at each support may be determined exactly by time history methods, as in N-1228.1. Having performed a time history response analysis, reaction loads and motions may also be determined on a time history basis, as in N-1228.2. Since the inertial effects may be considered to produce primary stresses and the differential support motions may be considered to produce secondary stresses, as in NB-3650, the inertial effects may be determined separately from the differential support movement effects, as in N-1228.3.

N-1228.1 Time History Response Analysis.

(a) Several different acceptable methods have been presented in the literature (refs. [7], [43], and [44]) for solving the equations of motion for a structural system excited by multiple simultaneous forcing functions. Since the basic problem to be solved must satisfy the fundamental principles of dynamics, the differences between the various methods are primarily in the approach of

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the solution process. One method of solution is summarized here to provide insight into the basics of time history analysis for multiple excitations.

(b) The general matrix form of the undamped coupled equations of motion may be written (refs. [7] and [45]) as follows:

$$MX'' + KU = F \tag{32}$$

where X'' represents the absolute acceleration of the mass point dynamic degrees of freedom, and U represents those displacements of the mass and support point dynamic degrees of freedom which would tend to cause distortions in the system. Rigid body translations or rotations are included in the mass acceleration terms on the left-hand side of the equation. Such rigid body motions would not tend to distort the system and accounted for by defining one of the system support points to be the *datum* support, and describing the motion of all other *nondatum* supports as motions relative to the datum support.

(c) Equation (b)(32) can be expanded to give

$$\begin{bmatrix} M_m & 0 \\ 0 & M_s \end{bmatrix} \begin{Bmatrix} X''_m \\ X''_s \end{Bmatrix} + \begin{bmatrix} K_{mm} & K_{ms} \\ K_{sm} & K_{ss} \end{bmatrix} \begin{Bmatrix} U_m \\ U_s \end{Bmatrix} = \begin{Bmatrix} 0 \\ F_s \end{Bmatrix} \tag{33}$$

where

F_s = the reaction forces at the system support points due to the response of the system to the motion of the support structure

K = the stiffness matrix of the system model condensed in a manner such that only mass point elements (subscript m) and active, nonreleased, nondatum support elements (subscript s) remain in the matrix

M_m = a diagonal submatrix of the system model lumped masses

M_s = a submatrix of inertia terms associated with the support joints of the system. For the purposes of this analysis, $M_s = 0$ because there is no mass lumped at support joints

U_m = displacement of mass point dynamic degrees of freedom relative to the datum support in each coordinate direction

U_s = displacement of nondatum support points relative to the datum support

X''_m = absolute acceleration of mass point dynamic degrees of freedom of the model

X''_s = absolute acceleration of the system support points

(d) The time history support motions imposed at the nondatum supports include only such displacements as would tend to cause distortions in the system. Rigid body translation or rotation of the supporting structure would not distort the system being analyzed; therefore, such rigid motions of the supporting structure are removed by computing the nondatum support relative motions as follows:

$$U_s = X_s - X_d - R_s\theta \tag{34}$$

where

R_s = a matrix of three vectors representing distances from the datum support point of the nondatum support points

U_s = displacements of nondatum support points relative to the datum support in each direction of excitation

X_d = absolute displacements of datum supports in each direction of excitation

X_s = absolute displacements of nondatum supports in each direction of excitation

θ = rigid body rotations of the supporting structure about each of the three coordinate axes

(e) The first equation of the set of eq. (c)(33) yields:

$$M_m X''_m + K_{mm}U_m + K_{ms}U_s = 0 \tag{35}$$

(f) A separation of variables can be achieved by defining the absolute acceleration of a mass point in terms of acceleration relative to the datum support, such that:

$$U''_m = X''_m - \gamma X''_{sd} - R_m\theta'' \tag{36}$$

where

R_m = a matrix of three vectors representing distances from the datum support point to the mass points for each of the rotational components

X''_{sd} = the absolute accelerations of the datum support in each coordinate direction

γ = matrix defining the direction of each respective translational dynamic degree of freedom

γ_{ij} = 1, if the ith dynamic degree of freedom is in the direction of the jth direction of support translation

γ_{ij} = 0, if the ith dynamic degree of freedom is not in the direction of the jth direction of support translation

θ'' = rotational accelerations of the datum support

(g) Equation (e)(35) then becomes

$$M_m U''_m + K_{mm}U_m = -M_m X''_{sd} - M_m R_m \theta'' - K_{ms}U_s \tag{37}$$

(h) At this point it is to be noted that the equations of motion are in a form expressing three dimensional response of the system mass points due to multiple translational and rotational support excitations in one or more coordinate directions. Normally, the rotations will consist of two components representing rocking about two orthogonal horizontal coordinate axes. The equations, however, are also valid for rotation about the vertical

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axes. It should be noted again that U_m and U_s are only the distortional portions of the total mass and support motions.

(i) The coupled equations of motion, eq. (g)(37), may be solved by one of the methods of direct integration, or, introducing the normal mode coordinate transformation:

$$U_m = \phi q \tag{38}$$

where

ϕ = the matrix of eigenvectors of the supported system
q = the normal mode coordinate

(j) Then making the usual assumption of proportional damping for the modes of the supported system, the equations of motion can be uncoupled and written in the following form:

$$q'' + 2\xi w q' + w^2 q = -\left(\phi' M_m \phi\right)^{-1}\left(\phi^t M_m \gamma X''_{sd} + \phi^t M_m R_m \theta'' + \phi^t K_{ms} U_s\right) \tag{39}$$

where

ω^2 = diagonal matrix of eigenvalues
$2\xi\omega$ = diagonal matrix of modal damping terms

(k) The right-hand side of eq. (j)(39) contains no damping terms in a rigorous sense only if the modal damping is proportional to stiffness. However, Ref. [46] points out that the contribution of these terms to earthquake forces can be expected to be small. This term is usually neglected for systems exhibiting structural damping characteristics within the range of the values in Table N-1230-1.

N-1228.2 Time History Reaction Analysis.

(a) Having solved the dynamics problem for the time histories of the mass point motions, the resultant reactions at locations throughout the system can be determined for each time step by imposing the deflections on the structure in a series of statics problems; one static solution for the deflected shape at each time step. Although this approach is analytically valid, it is generally not practical because of the many thousands of solutions required for the number of time steps in a typical seismic analysis. The problem may be reduced to a single static deflection analysis by applying unit displacements to each mass and support point to determine a set of influence coefficients for each desired reaction. The given support displacements and computed mass point displacements at each time step are multiplied by the set of influence coefficients to perform a complete reaction analysis of the system at each time step.

(b) The desired components of reaction (force, moment, stress, or deflection) are computed at each time step as follows:

$$R(t) = C_m U_m(t) + C_s U_s(t) \tag{40}$$

where

C_m = a matrix of mass point unit displacement influence coefficients (one column per mass point and one row per reaction component)
C_s = a matrix of nondatum support point unit displacement influence coefficients (one column per nondatum support and one row per reaction component)
$R(t)$ = a vector of reaction components at time t
$U_m(t)$ = a vector mass point relative displacements at time t
$U_s(t)$ = a vector of nondatum support relative displacements at time t

(c) In a similar manner, the absolute acceleration of any point in the system may be computed by multiplying the mass point and support relative accelerations by the influence coefficients for displacement reactions, and adding in the datum support rigid body rotational and translational absolute accelerations as follows:

$$R''(t) = \gamma X''_{sd}(t) + C_m U''_m(t) + C_s U''_s(t) + R_p \theta''(t) \tag{41}$$

where

C_m = a matrix of mass point unit displacement influence coefficients for components of displacement reactions
C_s = a matrix of nondatum support unit displacement influence coefficients for components of displacement reactions
R_p = a matrix containing distances to the reaction point from the datum
$R''(t)$ = a vector of absolute acceleration components at time t
$U''_m(t)$ = a vector of mass point accelerations relative to the datum at time t
$U''_s(t)$ = a vector of nondatum support relative accelerations at time t
$X''_{sd}(t)$ = the translational absolute acceleration of the datum support at time t
γ = a vector defining the direction of excitation, where

γ_{ij} = 1, if the ith component of reaction is in the direction of jth support motion
= 0, if the ith component of reaction is not in direction of the jth support motion

$\theta''(t)$ = the rotational absolute acceleration of the datum support at time t

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(d) This method, therefore, permits the calculation of any desired force or moment, or nonmass point motion on a time history basis.

(e) The ability to calculate time history acceleration of nonmass locations in the system permits the subsequent generation of response spectra at any arbitrary point in the system without the necessity of lumping mass at these points.

N-1228.3 Separation of Inertial and Relative Anchor Movement Stress Effects.

(a) The terms of the C_s coefficients matrix can be found by applying a unit displacement to each nondatum support in turn, holding all other supports and mass points fixed, and calculating the desired reaction at a given location. The reactions found in this manner are the combined effect of inertial and relative displacement stress producing effects, suitable for meeting the appropriate Code requirements, such as eq. (10) of NB-3653.1 for Class 1 piping.

(b) Alternatively, one may break up the C terms into two terms C_{s1} and C_{s2} such that

$$R(t) = C_m U_m(t) + C_{s1} U_s(t) + C_{s2} U_s(t) \quad (42)$$

or

$$R_1(t) = C_m U_m(t) + C_{s1} U_s(t) \quad (43)$$

and

$$R_2(t) = C_{s2} U_s(t) \quad (44)$$

where

C_{s1} = a matrix of influence coefficients due to holding all supports fixed and applying the set of forces to all mass points which would be required to hold the mass point in place due to a unit displacement of each nondatum support in turn

C_{s2} = a matrix of influence coefficients to a unit displacement of each nondatum support in turn, allowing all mass points to respond freely as static degrees of freedom in the structure

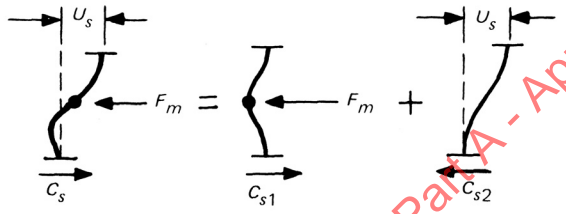
$R_1(t)$ = a vector of stress reaction components at time t due to inertial responses

$R_2(t)$ = a vector of stress reaction components at time t due to relative support motions

(c) The coefficients for a component of shear for a unit displacement of a nondatum support may be visualized as shown in Figure N-1228.3-1.

(d) The C_{s1} term therefore represents inertial effect produced by the differential support motion, and the C_{s2} term represents the relative displacement effect of the differential support motion. The separate stresses determined in this manner may be used to meet the appropriate Code requirements, such as eq. (9) of NB-3652 for Class 1 piping.

Figure N-1228.3-1
Coefficients for a Component of Shear for a Unit Displacement of a Nondatum Support



N-1228.4 Envelope Excitations.

(a) Where the responses are calculated using a single time history excitation for a multiply supported system, the response spectrum of the single time history used should envelop the spectra for the individual support motions. The secondary stress effects of differential support motions may then be determined by a separate static analysis where either

(1) the set of maximum displacements over all time for each support are applied simultaneously in a single analysis, or

(2) the displacements of each support are applied to perform an analysis for each time step, accounting for sign.

(b) Where the maximum displacements over all time are used in a single analysis, then

(1) for systems supported at multiple locations on the same floor/wall where one flexural mode dominates, the relative displacement set for the dominant mode may be used, where justified

(2) for systems supported at multiple locations on different floor/walls or on different structures where different flexural modes can dominate, the displacement may be applied statically as prescribed in N-1227

(c) Note that, in general, different support displacement sets may be required to obtain conservative secondary stress effect at different locations within a system, depending on the complexity of a given system.

N-1230 DAMPING

N-1231 Damping Values

(a) Motions in a structural system will dissipate energy from the system. The phenomenon of this kind of energy loss is called damping. In a structural system, sources of energy loss may be due to: a structural damping, which is caused by internal friction within the material or at connections between elements of a structural system; a viscous damping, which is caused by motions in a fluid; Coulomb damping which results from the sliding friction

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motion of a body on another surface. The following equation of motion is generally used for structural systems and components

$$[M]\{X''\} + \{D\} + [K]\{X\} = \{F\} \quad (45)$$

where

D = damping force vector
F = force vector
K = stiffness matrix
M = mass matrix
X = displacement vector
X' = velocity vector
X'' = acceleration vector

(b) The damping force $\{D\}$ represents the energy losses of the system. To evaluate these losses, the following alternative mathematical expressions are recommended for the vector $\{D\}$.

(1) Viscous damping:

$$\{D\} = [C]\{X'\} \quad (46)$$

where $[C]$ is a viscous damping matrix. The damping force is proportional to the velocity.

(2) Hysteretic damping:

$$\{D\} = iv[K]\{X\} \quad (47)$$

where v represents the structural damping factor (ref. [47]). This damping is also called complex damping. The damping force is proportional to the amplitude of the displacement and opposite in direction to the velocity:

$$\{D\} = g\left(abs[K]\{X\}\right) sgn\ \{X'\} \quad (48)$$

where g is a damping coefficient (ref. [48]). This damping is also called the Reid's damping. The damping force is proportional to the resisting force due to deformation and in the opposite direction of the velocity in harmonic motion.

(3) Coulomb damping:

$$\{D\} = \pm \mu\{N\} \quad (49)$$

where μ is the coefficient of friction and N is the normal force vector. The sign is chosen to be opposite the velocity. The damping force is due to friction and proportional to the normal force.

(4) General dashpot damping:

$$\{D\} = \alpha\{X'\}\left(abs\ \{X'\}\right)^{n-1} \quad (50)$$

where α is a damping constant and n is an integer. This damping force is due to the turbulence and the dashpot geometry.

(c) These expressions permit the simulation of the energy loss for a particular problem. It is extremely difficult to give an accurate analytical expression for the different forms of damping in complex structural systems. The analyst may select a simple expression, such as viscous or hysteretic damping, to solve the problem by assuming that the energy losses are equivalent; i.e., viscous damped system introduced in the analysis will have the same energy loss per cycle as the real structure.

(d) To establish an equivalent viscous damping matrix $[C]$ for the system, experimental data is usually required. Important experimental results, which indicate that energy losses increase with stress or displacement amplitude, or both, have been compiled in existing literature (refs. [49] and [50]). The incorporation of this behavior to the equivalent viscous damping in the transient analysis will usually lead to an expression of the damping force which will make the equations of motion nonlinear. To avoid this inconvenience, conservative modal damping values may be used for nuclear systems and structures for OBE and SSE condition loads, respectively. Table N-1230-1 shows the modal damping values recommended. Damping values higher than the ones given in Table N-1230-1 may be used in a dynamic analysis if the basis is justified.

(e) Methods of incorporating the damping in structural dynamics are usually dependent on the mathematical convenience. The energy loss per cycle is simulated by a convenient mathematical scheme; frequently viscous damping expressions are adopted in the analysis. In N-1232 and N-1233, methods of proportional and nonproportional dampings are discussed and the physical interpretations associated with each method is also examined.

N-1232 Proportional Damping

(a) Consider a viscously damped system of the following form:

$$[M]\{X''\} + [C]\{X'\} + [K]\{X\} = \{F\} \quad (51)$$

Table N-1230-1
Damping Values
Percent of Critical Damping

Structure or Component	Earthquake Magnitude: Operating Basis Earthquake	Earthquake Magnitude: Safe Shutdown Earthquake
Equipment	2	3
Piping systems	5	5
Welded steel structures	2	4
Bolted steel structures	4	7
Prestressed concrete structures	2	5
Reinforced concrete structures	4	7

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Let $[\phi]$ be the modal matrix of the undamped system of

$$[M]\{X''\} + [K]\{X\} = \{0\} \quad (52)$$

such that

$$[\phi]^{\mathrm{T}}[M][\phi] = [I] \quad (53)$$

and

$$[\phi]^{\mathrm{T}}[K][\phi] = [\omega^2]$$

where

$[\phi]^T$ = transpose of the matrix $[\phi]$
I = identity matrix
ω^2 = diagonal matrix
ω_i = circular frequencies

(b) If the matrix $[C]$ in eq. (a)(51) can be diagonalized by the undamped normal modes (i.e., $[\phi]^{\mathrm{T}}[C][\phi]$ is a diagonal matrix), then the damping matrix $[C]$ is called the proportional damping matrix; otherwise, it is called the nonproportional damping matrix.

(c) Reference [52] has illustrated that the matrix $[C]$ can be diagonalized when it is a linear combination of $[M]$ and $[K]$ matrices. Caughey (ref. [53]) has derived more general conditions under which the matrix $[C]$ can be diagonalized.

(d) For proportional damping the equations of motion may be uncoupled into a set of independent one degree of freedom systems. In this case, the damping ratio associated with the different modes can be determined.

(e) Among the category of proportional dampings, two types of damping matrix $[C]$ are commonly adopted; namely, the mass and stiffness damping and the orthogonal modal damping. Both types of damping are discussed in N-1232.1 and N-1232.2.

N-1232.1 Mass and Stiffness Damping (Ref. [53]).

(a) In this damping, the matrix $[C]$ is assumed to be proportional either to the mass matrix $[M]$ or to the stiffness matrix $[K]$ or to a linear combination of the two. That is, it can be written as follows:

$$[C] = \alpha[M] + \beta[K] \quad (54)$$

where

α and β = two real constants
$\alpha[M]$ = mass damping
$\beta[K]$ = stiffness damping

(b) In using the mass and stiffness damping, damping values of the entire system are determined by the two constants α and β. To determine the value of α and β, one can control the damping ratios for two frequencies.

(c) Let ω_r and ω_s be the two frequencies that we want to have a damping ratio of ξ_r and ξ_s, respectively. Then α and β may be determined (ref. [54]) as follows:

$$\begin{aligned} \alpha &= 2\omega_r\omega_s\left(\xi_s\omega_r - \xi_r\omega_s\right)/\left({\omega_r}^2 - {\omega_s}^2\right) \\ \beta &= 2\left(\xi_r\omega_r - \xi_s\omega_s\right)/\left({\omega_r}^2 - {\omega_s}^2\right) \end{aligned} \quad (55)$$

Then, for an arbitrary frequency ω_i the damping ratio ξ_i can be computed by eliminating α and β and is given as:

$$\xi_i = \frac{1}{\left({\omega_r}^2 - {\omega_s}^2\right)}\left[\frac{\omega_r\omega_s}{\omega_i}\left(\xi_s\omega_r - \xi_r\omega_s\right) + \omega_i\left(\xi_r\omega_r - \xi_s\omega_s\right)\right] \quad (56)$$

where

$$\frac{1}{\left({\omega_r}^2 - {\omega_s}^2\right)} \times \frac{\omega_r\omega_s}{\omega_i}\left(\xi_s\omega_r - \xi_r\omega_s\right)$$

is due to mass damping and

$$\frac{1}{\left({\omega_r}^2 - {\omega_s}^2\right)} \times \omega_i\left(\xi_r\omega_r - \xi_s\omega_s\right)$$

is due to stiffness damping.

(d) The application of the mass and stiffness damping concept is illustrated by a following numerical example. Consider a structural system with frequencies of $f_n = n$ Hz for $n = 1, 2, \ldots, 25$. For a damping ratio of $\xi_{10} = 0.05 = \xi_{20}$ at frequencies $f_{10} = 10$ Hz and $f_{20} = 20$ Hz and by using eq. (c)(56), the damping ratios for the other modes may be computed. For example, at the frequencies of 5, 15, and 25 Hz, the corresponding damping ratios are $\xi_5 = 0.075$, $\xi_{15} = 0.047$, and $\xi_{25} = 0.055$.

(e) It should be noted that in eq. (c)(56), the damping ratio is the sum of the contributions due to the mass and stiffness damping. Using mass damping $\alpha \neq 0$, $\beta \neq 0$, the damping ratio will decrease with increase of the frequency. On the other hand, using stiffness damping $\alpha = 0$, $\beta \neq 0$, the damping ratio will increase with the increase of the frequency.

(f) Mass damping introduces the damping forces that are proportional to the velocities of each mass point in the system. Mass damping may be used to represent the energy loss due to impact and friction. Stiffness damping introduces the damping forces that are proportional to the time rate of deformation. Stiffness damping can be used to represent the structural damping. While the mass damping introduces the damping forces due to the rigid body motion (displacements without deformations) of the system, the stiffness damping does not.

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(g) In seismic analysis of a structural system, the equation of motion is generally formulated in terms of the displacements with respect to the base. If the damping ratios are established for the two significant frequencies, the mass and stiffness dampings may be adopted. The damping ratios for the other frequencies are computed with the aid of eq. (c)(56). If only stiffness damping is used, and for the significant frequency Ω_r the established damping ratio is ξ_r, then

$$\beta = \frac{2\xi_r}{\omega_r} \text{ and } \xi_i = \xi_r \frac{\omega_i}{\omega_r}$$

where ξ_i is the damping ratio for the modal frequency ω_i.

N-1232.2 Orthogonal Modal Damping (Ref. [54]). In the orthogonal modal damping method, the damping matrix $[C]$ in eq. N-1232(a)(51) is not formulated in advance. After solving the eigenvalue problem, the system or uncoupled equations are assumed to have the form

$$q''_i + \xi_i(2\omega_i)q'_i + \omega_i{}^2 q_i = f_i \qquad i = 1, 2, \ldots, N \tag{57}$$

where the second term in eq. (57) is added to account for the damping. The damping ratio for each frequency is free to be determined by the analyst without any mathematical restriction. The damping ratio assigned to one mode will have no effect on the damping ratios of the others; the modes are orthogonal. These ratios can be obtained from the experimental results. This method is used often in solving viscously damped systems. The example problems for the various damping methods discussed here are given in ref. [55].

N-1233 Nonproportional Damping

In the foregoing discussion of the proportional damping (N-1232), the damping of a structural system was assumed to have a form that couplings do not exist between the classical modes of vibrations; i.e., the mode shapes obtained from the solution of a free vibration of the undamped system. This approach is appropriate for the correlation of test data for a structural system composed of a single material, or damping mechanism which is homogeneous throughout the system. However, for a system which is composed of different materials, such as a reactor coolant loops, containment building, and soil foundation, the damping mechanism of one type of material may be considerably different from the other. In order to have a better simulation of the damping, it is useful to construct a damping matrix which reflects the material composition rather than the mathematical convenience (refs. [56] and [57]). The damping matrix so constructed will no longer be diagonalized by the classical modes. The term *nonproportional damping* is used for this type of damping. In nonproportional damping, the matrices $[M]$, $[K]$, and $[C]$ are used to solve the complex frequencies and mode shapes. From the complex frequencies, damping ratios associated with each mode can readily be computed. Unlike proportional damping, the matrix $[C]$ of a nonproportional damping system cannot be formulated simply by specifying a set of damping ratios, unless a mathematical process of successive approximations is performed. Due to the mathematical complexity involved in the complex frequencies and mode shapes solution, the process of assigning a set of damping ratios with respect to the complex modes becomes impractical. For certain types of structures such as reactor coolant loops, the system exhibits a pronounced modal behavior (ref. [58]). Instead of using a set of damping ratios based on the complex frequencies and modes, the classical modes and frequencies may be used to avoid the mathematical complexity. N-1233.1 and N-1233.2 describe acceptable methods for the formulation of a nonproportional damping matrix for a structural system which is composed of different materials.

N-1233.1 Composite Mass Damping and Stiffness Damping. (15)

(a) In the proportional damping, eq. N-1232(a)(51), the system damping matrix is assumed to be a linear combination of the system mass and stiffness matrices, without any distinction of the material damping property. Different sets of proportional constants for mass and stiffness may be assumed to formulate a system damping matrix. Using the concept of mass and stiffness damping, the energy dissipation function D for the system can be written as follows:

$$D = \frac{1}{2}\sum_{i=1}^{nc}\left(\alpha_i\{X'\}_i{}^{\mathrm{T}}[M]_i\{X'\}_i + \beta_i\{X'\}_i{}^{\mathrm{T}}[K]_i\{X'\}_i\right) \tag{58}$$

where

$[K]_i$ = stiffness matrix of subsystem i
$[M]_i$ = mass matrix of subsystem i
nc = number of subsystems
α_i = mass proportional constant of subsystem i
β_i = stiffness proportional constant of subsystem i

For the systems with constant damping values, α_i and β_i are the same for all elements. However, when the system is composite, which consists of subsystems with different damping values, α_i and β_i are different for different subsystems but constant within each subsystem. These constants may be established for each subsystem as discussed in N-1232.1.

(b) This decomposition to allow different damping constants α_i and β_i to be used for each different subsystem may be based on the damping values recommended in Table N-1230-1.

(c) To determine the constants α_i and β_i, the frequencies and modes of the free undamped vibration of the ith subsystem are used as a basis. The mathematical process involved is similar to the one of stiffness and mass damping described in N-1232.1 except the constants α_i and β_i are different for each material or subsystem.

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(d) Using eq. (a)(58), the equation of motion of a viscously damped system has the following form:

$$[M]\{X'\} + \sum_{i=1}^{nc}\left(\alpha_i[M]_i + \beta_i[K]_i\right)\{X'\} + [K]\{X\} = \{F\} \tag{59}$$

(e) Equation (d)(59) may be integrated directly to obtain the response. If normal mode method is used, eq. (d)(59) may be solved by neglecting the modal coupling effect to decouple the equations of motions on the normal mode basis.

(f) Let $\{\phi\}$ be the modal matrix of the undamped system of eq. (d)(59) and $\{\phi_j\}$ be the jth modal vector. With the aid of the orthogonality conditions, the equation of motion for the jth mode can be written in the following form:

$$\{\phi_j\}^T[M]\{\phi_j\}q''_j + \{\phi_j\}^T\sum_{i=1}^{n}\left(\alpha_i[M]_i + \beta_i[K]_i\right)[\phi]\{q'_j\} + \{\phi_j\}^T[K]\{\phi_j\}q_j = \{\phi_j\}^T\{F\} \tag{60}$$

where q''_j is the jth generalized coordinate. The second term on the left-hand side of eq. (60) contains the coupling terms. If these coupling terms are neglected, eq. (60) reduces to the following equation:

$$\{\phi_j\}^T[M]\{\phi_j\}q''_j + \{\phi_j\}^T\sum_{i=1}^{n}\left(\alpha_i[M]_i + \beta_i[K]_i\right)\{\phi_j\}q'_j + \{\phi_j\}^T[K]\{\phi_j\}q_j = \{\phi_j\}^T\{F\} \tag{61}$$

or

$$q''_j + 2\omega_j\left(\frac{\sum_{i=1}^{nc}\{\phi_j\}^T\alpha_i[M]_i\{\phi_j\}}{2\omega_j\{\phi_j\}^T[M]\{\phi_j\}} + \frac{\sum_{i=1}^{nc}\omega_j\{\phi_j\}^T\beta_i[K]_i\{\phi_j\}}{2\{\phi_j\}^T[K]\{\phi_j\}}q_j\right) + \omega_j^2 q_j = \frac{\{\phi_j\}^T\{F\}}{\{\phi_j\}^T[M]\{\phi_j\}} \tag{62}$$

Consequently, the effective damping for the jth mode is as follows:

$$\xi_j = \frac{\sum_{i=1}^{nc}\{\phi\}^T\alpha_i[M]_i\{\phi_j\}}{2\omega_j\{\phi_j\}^T[M]\{\phi_j\}} + \frac{\sum_{i=1}^{nc}\omega_j\{\phi_j\}^T\beta_i[K]_i\{\phi_j\}}{2\{\phi_j\}^T[K]\{\phi_j\}} \tag{63}$$

or

$$\xi_j = \frac{\sum_{i=1}^{nc}\alpha_i\left(\begin{array}{c}\text{max. kinetic energy for } i\text{th}\\ \text{subsystem in the } j\text{th mode}\end{array}\right)}{2\omega_j\left(\text{max. kinetic energy for the system}\right)} + \frac{\sum_{i=1}^{nc}\beta_i\left(\begin{array}{c}\text{max. strain energy for } i\text{th}\\ \text{subsystem in the } j\text{th mode}\end{array}\right)}{2\left(\text{max. strain energy for the system}\right)} \tag{64}$$

(g) Equation (f)(63) can be greatly simplified if either α_i or β_i is assumed to be zero. This assumption may be used in the structural analysis for seismic loads where damping is generally assumed proportional either to the kinetic energy distribution of the subsystems or to the strain energy stored.

(h) Once the damping ratio ξ_j is known, the analysis can be carried out using a time history or response spectrum technique.

N-1233.2 Subregional Modal Damping.

(a) In dynamic analysis of soil–structure interaction, the modal damping value of the soil is often higher than that of the superstructure. In addition, the modal damping values of the various subsystems, building, loop, etc., of the superstructure may be different. There are several acceptable approximate methods to simulate this subregional modal damping characteristic. These methods are briefly discussed here. One method from ref. [15] is to relate composite damping to absolute displacement eq. (65), while another method (ref. [59]) is to relate it to relative displacement eq. (66):

$$\xi_j = \frac{\sum_{i=1}^{nc}\{\phi_j\}^T\beta_i[M]_i\{\phi_j\}}{\{\phi_j\}^T[M]\{\phi_j\}} \tag{65}$$

$$\xi_j = \frac{\sum_{i=1}^{nc}\{\phi_j\}^T\beta_i[K]_i\{\phi_j\}}{\{\phi_j\}^T[K]\{\phi_j\}} \tag{66}$$

where

$[K]$ = total system stiffness matrix
$[K]_i$ = stiffness matrix of ith subsystem
$[M]$ = total system mass matrix
$[M]_i$ = mass matrix of ith subsystem
nc = number of subsystems
$\{\phi_j\}$ = jth eigenvector
β_i = % critical damping associated with ith subsystem
ξ_j = % critical damping for jth mode

(b) Another acceptable approximate method employs mass and stiffness damping for each subsystem. This method calculates coefficients α and β, eq. N-1232.1(c)(55), for a subsystem such that its modal

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damping is simulated in the frequency range of interest. The damping matrices of all the subsystems are assembled to formulate a total system damping matrix. Another acceptable approach to generate a total damping matrix is discussed in ref. [62].

(c) All the methods discussed above are approximate. More accurate techniques are also acceptable. A more rigorous method to formulate a damping matrix for a total system is based on the modal synthesis procedure. This procedure is explained by Hurty (ref. [56]). Various investigators have developed another method (ref. [63]) modifying this procedure. In this procedure, the total system having nonproportional damping is divided into subsystems, each having proportional damping. Each subsystem is represented by its modal mass, damping, and stiffness matrices. As each subsystem has a proportional damping, the associated modal damping matrix is formulated without any approximation. The modal matrices of all the subsystems are assembled and the displacement compatibility at subsystem interfaces are satisfied. The resulting equations of motion are solved and with the aid of a transformation matrix, actual response is obtained. References [57], [64], and [65] give the details of the application of this method.

(d) The total damping matrix developed by the preceding methods represents the viscous damping. An acceptable alternative formulation for damping, derived in ref. [66], is an equivalent modal damping ratio which represents the viscous damping in the swaying spring and the hysteretic damping in the rest of the soil-structure system (the superstructure and the rocking spring).

N-1234 Linear Hysteretic Damping

(a) In this paragraph the linear hysteretic damping of the Reid's model (ref. [48]) will be discussed. This nonlinear model leads to well posed mathematical problems for transient and steady-state oscillations of all kinds. The damping forces of the Reid's model are assumed to be proportional to the internal forces of the structure but in phase with the associated velocities. By using the Reid's damping model, the equation of motion [eq. N-1231(a)(45)] becomes:

$$[M]X'' + g\{\{abs[K]\{X\}\} \times \{sgn\{X'\}\}\} + [K]\{X\} = \{F\} \quad (67)$$

where

$abs\ [K]\ \{X\}$ = the absolute value of each component
g = damping coefficient
$sgn\ \{X'\}$ = the algebraic sign of the velocity $\{X'\}$

(b) The Reid's damping model has the following characteristics (ref. [67]):

(1) the energy loss per cycle in a harmonic motion is proportional to the coefficient g and square of the amplitude of displacement and independent of the frequency;

(2) for light damping ($g^2 \ll 1$) the Reid's damping model will be equivalent to the classical modal damping with a damping ratio of $\xi = g/\pi$ for each mode of vibration.

(c) Similar to the case of composite mass damping and stiffness damping, the damping coefficient g in eq. (a)(67) may also be considered as the elemental basis for a structural system which is composed of materials with different damping values.

N-1300 FLOW-INDUCED VIBRATION OF TUBES AND TUBE BANKS

N-1310 INTRODUCTION AND SCOPE

The flow-induced vibration (FIV) potential of structures has been known for a long time (refs. [79] through [84]). FIV analyses are required to determine the adequacy of a design, or in areas of uncertainty, to be aware of the need for experimental verification (refs. [85] and [86]) if high reliability of the component is a necessity. FIV may be due to any one of several excitation mechanisms because power systems include many types of flexible components subject to a variety of fluid flows, such as pipe, channel, and jet flows followed by mixing in plenums and heat exchangers. Since a single component is often subjected to different turbulent flows from several directions because of the influence of adjacent structures and boundaries, FIV analyses for more than one excitation mechanism is not unusual.

The quantitative data and correlations available to perform FIV analyses are unique to the flow geometry created by each component. More quantitative information and design methods are available for some components than others. In particular, the circular cylinder has been studied most. N-1320 through N-1340 of this Appendix are presented to illustrate one or more acceptable steps for the FIV analysis of arrays of cylinders subject to the three most significant excitation mechanisms. The general methods employed are applicable to other types of components, but the data are specifically for single cylinders and cylindrical arrays. Because of the large number of FIV mechanisms, the methodology of analysis is referenced, but enough information is given to understand a mechanism and make design calculations. Because of the developing nature of the subject, more than one set of design data or methods may be recommended with the implication to the designer to use either the more appropriate or the more conservative predictions.

Semiempirical correlations based on experimental data, but guided by the equations of motion, often form the basis of a design method. The state-of-the-art regarding description of the FIV mechanisms is that many mathematical models have been proposed for fluid-structure coupling forces, but general agreement on the physics of many of the phenomena has not been attained, although models simulating the behavior may be available.

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N-1311 Definitions

In this section some commonly used terminologies in flow-induced vibration analysis are defined and briefly described.

(a) Fluid forces can be defined into two broad categories to describe FIV excitation mechanisms (refs. [79] through [83]). Fluid excitation forces are created by the incident flow on a structure, and they would occur, in some form, even without structural motion. Fluid-structure coupling forces are induced by structural motion, and they occur in both flowing and nonflowing fluids.

(b) Added mass and added damping have been successfully used to characterize the fluid-structure coupling forces created by the motion of a structure in a nonflowing fluid (refs. [87] through [93]). Added mass and added damping increase the effective mass and damping of a structure vibrating in a fluid. In addition, the presence of a dense fluid between otherwise unconnected, adjacent structures can couple their vibrations and result in significantly different natural frequencies, mode shapes, and damping from those obtained in a vacuum. For a low density fluid (e.g., air); the added mass is often negligible. Added mass is a function of the geometry of the structural surface exposed to fluid and the presence of adjacent structures, if any. Table N-1311-1 gives equations for added mass for two-dimensional sections and rigid bodies in two-dimensional motion. These equations were determined from exact solution of inviscid potential flow with a moving structural boundary. See N-1400 for dynamics of coupled fluid shells.

(c) In a *weakly coupled fluid-structure system*, the FIV excitation mechanism causes small structural motion and the fluid forces induced by the structural motion can be linearly superimposed onto the fluid excitation forces which are largely independent of the structural motion. The fluid-structure coupling forces can be expressed to a first order of approximation in terms of added mass, stiffness, and damping matrices. The fluid excitation forces can be determined separately from the coupling forces either by analysis or by model tests with only the hydraulics simulated.

Examples of FIV excitation mechanisms producing weakly coupled fluid-structure systems are incident flow turbulence and turbulent boundary layers over rods, plates, and shells (refs. [81] and [82]); some wake flows produced by flow across bluff bodies; and many sources of acoustic noise (refs. [80] and [95]). In these cases, the fluid excitation energy is generated at some point in the fluid circuit and the structure is the recipient of the energy. The forces due to flow turbulence and attached boundary layers typically are broadband random, while separated wake flows that roll into periodically shed vortices can produce very discrete frequency forces (refs. [82], [87], and [97]).

(d) In a *strongly coupled fluid-structure system*, the FIV excitation mechanism causes the structural motion to become large enough to change the flow field; some of the fluid forces amplify, rather than inhibit, the structural motion that produced them. Clearly distinguishing between fluid-structure coupling forces and fluid excitation forces is difficult in strongly coupled fluid-structure systems. In general, the coupling forces are highly nonlinear functions of structural motion and flow velocity.

(e) Fluid-elastic instability of closely packed heat exchanger tube bundles (refs. [80], [81], [82], and [88]) is an example of a strongly coupled fluid-structure system. The motion of each tube affects the fluid forces and the motion of the other tubes to produce self-excitation. The occurrence of the instability has been interpreted as due to adverse changes in the structural mass, damping, and fluid-structure coupling force (ref. [88]). However, most of the expressions for predicting the onset of instability are based on compilations of direct measurements of the critical velocities at the onset of instability.

(f) Cross flow is a flow perpendicular to the structural longitudinal axis. Cross flow is one example where an FIV mechanism is produced that can create either a weakly or a strongly coupled fluid-structure system. Vortex shedding in the wake of a tube in cross flow produces both fluid excitation forces and fluid-structure coupling forces that amplify structural motion. For ideal cross flow, where a long, smooth surface tube is isolated in uniform (2–D) cross flow with little or no turbulence in the approaching flow stream, very periodic, two-dimensional vortices are shed. These vortices produce alternating lift forces normal to the tube axis and flow and are nearly as large as the steady, flow direction drag forces, if the Reynolds number, based on the tube diameter, is below 2×10^5 (refs. [82], [87], and [89]). If the vortex shedding frequency is sufficiently different from the structural natural frequencies, the alternating lift forces act as fluid excitation forces only. However, if the vortex shedding frequency and one of the structural natural frequencies are sufficiently close to each other and the fluid excitation forces can produce large enough motion, then coupled fluid-structure forces occur, which apparently further amplify the motion. Enough experimental data are available to bound the fluid excitation forces, but the representation of the coupled fluid-structure forces is still being researched. Most of the representations are based on highly phenomenological models that stimulate, to various degrees, a small amount of data covering only a narrow range of idealized conditions.

(g) The *joint acceptance* is a measure of the probability that a structure vibrating in one mode will remain in the same mode when excited by a random force; the *cross acceptance* is a measure of the probability that a structure vibrating in one mode will change to another mode when excited by a random force. For many applications only the joint acceptance is assumed to be important. When mode shapes are normalized to unity, the sum of the joint

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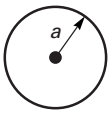
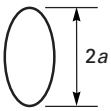
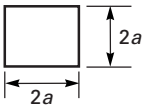
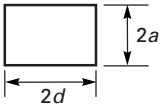
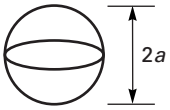
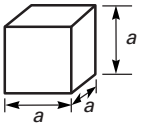
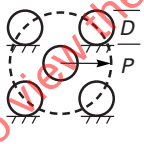
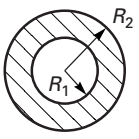
Table N-1311-1
Added Mass for Lateral Acceleration of Structures in a Fluid Reservoir

Geometry	Added Mass for Lateral Acceleration (Acceleration left to right) [Note (1)]
1. Circular Section 	$\rho\pi a^2 b$ [159]
2. Elliptical Section 	$\rho\pi a^2 b$ [159]
3. Square Section 	$1.51\rho\pi a^2 b$ [159]
4. Rectangular Section 	$k\rho\pi a^2 b$ [159] k = 1 for $d << a$
5. Sphere 	$^2/_3\rho\pi a^3$ [159]
6. Cube 	$0.7\rho a^3$ [159]
7. Cylinder Section in an Array of Fixed Cylinders 	$\frac{\rho\pi D^2 b}{4}\left[\frac{(D_e/D)^2+1}{(D_e/D)^2-1}\right]$ where $D_e/D = (1 + 0.5\,P/D)P/D$ Approximate solution from ref. [160]. See refs. [161] and [162] for arrays of flexible cylinders.
8. Circular Section With a Fluid Filled Annulus 	$\rho\pi R_1^2 b\left[\frac{1+(R_1/R_2)^2}{1-(R_1/R_2)^2}\right]$, inner cylinder $\rho\pi R_2^2 b\left[\frac{1+(R_1/R_2)^2}{1-(R_1/R_2)^2}\right]$, outer cylinder [163] See N-1451.1.

GENERAL NOTES:
(a) b = length of section; ρ = fluid mass density. (refs. [159], [160], [161], [162], and [163].)
(b) See N-1400 for finite length shells.

NOTE:
(1) The bracketed numbers refer to the references list following this Appendix.

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acceptances is equal to 1. (See ref. [112].) Therefore, the assumption that the joint acceptance is equal to 1 gives conservative estimates of structural responses.

(h) Damping is the result of energy dissipation during structural vibration. Damping limits resonant vibration amplitude and delays the onset of fluid elastic instability. Damping is the result of material damping within a structure, motion of trapped fluid within joints, and impact, scraping, and friction within joints. The material damping of nickel, copper, and steel is low, less than 0.1% critical damping. Damping at joints and supports dominates the damping of tubes that are supported by passing through oversized holes in support baffles. Table N-1311-2 gives general guidelines for damping in flow-induced vibration. Testing is required to establish more precise estimates of damping for specific designs.

(15) N-1312 Nomenclature

C_n = reduced damping in nth mode
C_L = lift coefficient
D = cylinder diameter
E = Young's modulus
F = force
f_n = natural frequency of nth vibration mode, hertz
f_s = frequency of periodic vortex shedding, hertz
G_f = single-sided power spectral density of the forcing function, in (force/length)^2 per Hz
G_f^i = G_f spectrum for the ith span of a multi-span tube
G_y = single-sided power spectral density of response
H_j = transfer function of jth vibration mode
I = area moment of inertia
J^2 = joint acceptance
J_{jk}^2 = cross acceptance for the jth and kth vibration modes
$\left(J_{jk}^i\right)^2$ = acceptance for the ith span
ℓ_c = axial correlation length
= $2\int_0^L r(x')\,dx'$ where $r(x')$ is the correlation function and x' is the separation distance
ℓ_c^i = correlation length in the ith span
L_e = cylinder length subject to vortex shedding
L_i = span length
m = mass per unit length
m_A = added fluid mass per unit length
m_c = contained fluid mass per unit length
m_f = cylinder displaced fluid mass/length
M_j = modal mass
M_n = effective modal mass/length for nth vibration mode
m_s = structural mass per unit length
m_t = total mass per unit length of tube
= $m_A + m + m_s$
n = vibration mode, $n = 1$ is fundamental mode
P = tube pitch
= distance between tube centers
p = pressure
q = dynamic pressure, $(1/2)\rho V^2$
Re = Reynolds number, VD/ν
R_p = Cross correlation of the pressure field
S = Strouhal number, $f_s\ D/V$
S_f = cross spectral density of the forcing function on a cylinder, (force/length)^2 per Hz
S_{fo} = power spectral density of the forcing function
S_p = cross spectral density of the pressure field
S_y = power spectral density of cylinder response
t = time
U_c = convection velocity

Table N-1311-2
Guidelines for Damping of Flow-Induced Vibration

Description of Tube Installation	Fluid Surrounding Tube	Critical Damping Ratio, ζ Low [Note (1)]	Typical Design Value	High [Note (2)]
Thermowells and single span tubes supported by welded or rolled in ends	Liquid and gas	0.0005	0.002	0.005
Multispan heat exchanger tubes supported by passing through oversized holes in plates	Low density gas	0.008	0.017	0.03 [Note (3)]
	Water and other liquids	0.01	0.02	0.03 [Note (3)]

GENERAL NOTE: This table applies to metallic tubes with 0.5 in. (13 mm) to 2.0 in. (50 mm) outside diameter. For tubes passing through oversized holes in support plates, this table applies to typical diametrical clearance between tube outside diameter and tube support inside diameter of 0.010 in. (0.2 mm) to 0.030 in. (0.8 mm).

NOTES:
(1) Low value: For midspan rms vibration amplitude less than 1% of tube diameter and smaller than the diametrical clearance between the tube and the support plate.
(2) High value: For midspan rms amplitudes comparable to or larger than the diametrical clearance between the tube and the support plate. Tube wear can result.
(3) Critical damping ratios $0.03 < \zeta < 0.05$ can be used if justified by applicable experimental data.

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V = mean velocity
x = axial distance
y_n^* = maximum displacement in nth vibration mode
$\overline{y}^2$ = mean square response of a cylinder
α_n = amplification factor in nth vibration mode
γ_n = mode shape factor in nth vibration mode
Γ = coherence of forcing function on a cylinder
Γ^i = coherence for ith span
δ_m = mass-damping parameter, $2\pi\xi_n m_t/\rho D^2$
δ_n = log decrement for nth vibration mode
= $2\pi\xi_n$
ξ_n = fraction of critical damping for nth mode
ρ = fluid mass density
ϕ_n = nth vibration mode shape
ϕ_n^* = maximum value of ϕ_n
θ = angle between direction of flow and normal to tube axis
ν = kinematic viscosity
ω = frequency, radian/sec

N-1320 VORTEX SHEDDING

N-1321 Vortex Shedding From a Fixed Bluff Body

For a bluff body in uniform cross flow, the wake behind the body is no longer regular, but contains distinct vortices of the pattern shown in Figure N-1321-1 for a circular cylinder. The vortices are shed alternatively from each side of the body in a regular manner and give rise to an alternating lift force. Experimental studies of this vortex shedding process have shown (refs. [94] and [95]) that the frequency in hertz of the alternating lift force can be expressed as:

$$f_s = SV/D \quad (68)$$

Some common types of bodies or structures for which vortex shedding occurs are shown in Figure N-1321-2. The following discussions are based on the circular cylinder; however, the concepts apply equally well to other bluff bodies.

The oscillating lift force produced on an isolated single cylinder of diameter D and length L by uniform cross flow can be expressed as (refs. [96] and [97]):

$$F = C_L J q D L\left[\sin\left(2\pi f_s t\right)\right] \quad (69)$$

Figure N-1321-1
Vortices Shed From a Circular Cylinder

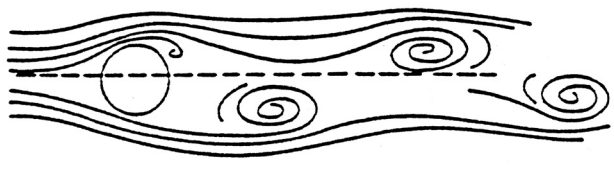
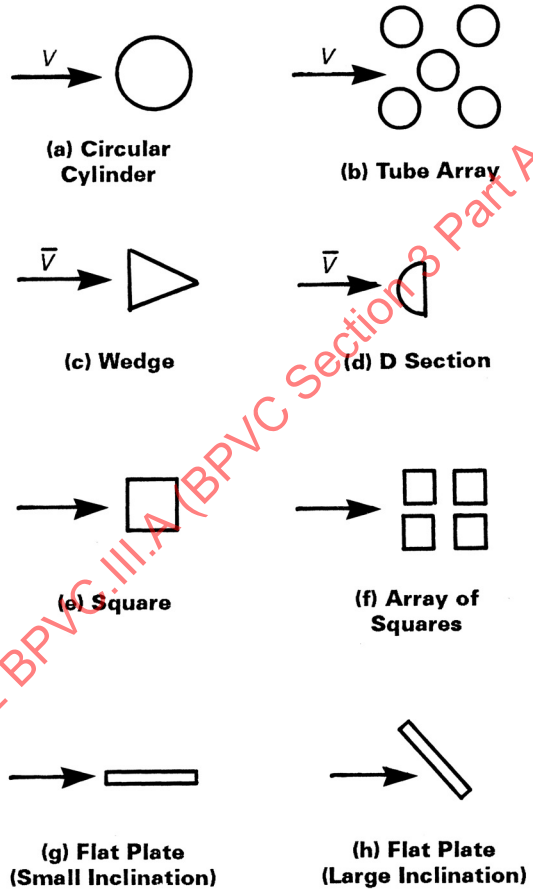


Figure N-1321-2
Some Typical Cross Sections of Bluff Bodies That Can Experience Vortex Shedding



where C_L, f_s, and J are functions of the Reynolds number Re and must be determined experimentally. In uniform cross flow, the energy of vortex shedding occurs over a very narrow frequency band with a center frequency f_s, except over a transition band of Reynolds number (2×10^5 to 3×10^6) where the character of the frequency content may vary from almost periodic to completely random. The measured Strouhal number is $S \approx 0.2$ for $10^3 < Re < 2 \times 10^5$; for larger Re, experimental values of S and C_L show considerable scatter.

The alternating vortex fluid forces are not generally correlated over the entire cylinder length L. As a consequence, two limiting cases of the joint acceptance exist for a uniform rigid-body-mode (ref. [97]).

$$J^2 = \ell_c/L \quad \text{if } \ell_c \ll L$$
$$= 1 \quad \text{if fully correlated} \quad (70)$$

The correlation length in the lift direction for stationary cylinders has been found to be approximately 3 to 7 diameters ($3D < \ell_c < 7D$), for $10^3 < Re < 2 \times 10^5$ (ref. [87]).

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For larger Reynolds numbers, the correlation lengths for stationary cylinders can be expected to be even smaller because the attached boundary layer becomes fully turbulent. J^2 is usually much less than 1 for long stationary cylinders. Motion of the cylinder at the frequency of vortex shedding substantially increases the correlation length (refs. [82] and [87]) as discussed in N-1323 and N-1324.

Vortex shedding also induces a force in the streamwise or drag direction. The drag force occurs at twice the vortex shedding frequency for single cylinders (ref. [87]). However, the magnitude of the oscillating drag force is typically an order of magnitude smaller than the oscillating lift force.

N-1322 Practical Cross Flow

The case of ideal cross flow is rarely found except in the laboratory. Many practical conditions reduce the effectiveness and strength of vortex shedding as an excitation mechanism:

(a) If the body is located in a turbulent flow, or if the tube surface is rough, the turbulence tends to widen the band of shedding frequencies and decrease the energy at the dominant shedding frequency (ref. [129]).

(b) If the cylinder is inclined to the flow, the shedding frequency can be adequately predicted by employing the component of flow velocity normal to the cylinder axis

$$f_S = \left(SV / D\right) \cos\theta \qquad (71)$$

where θ is the angle between the direction of flow and the normal to the cylinder axis. Inclined flow tends to reduce the magnitude of the vortex shedding forces (ref. [98]).

(c) Spanwise variations in flow velocity imply that the vortex shedding frequency also varies in the spanwise direction. This effect will generally reduce the magnitude of the net vortex shedding excitation.

(d) There is some evidence (refs. [99] and [100]) that vortex shedding does not occur in two-phase flow and that vortex shedding is only a concern in single-phase flows.

(e) While the vortex shedding characteristics discussed above have general applicability, the effects of adjacent bodies have not been specifically included. Studies of two (ref. [101]) or more circular cylinders show that vortex shedding does occur, but its character is very sensitive to the relative location and spacing of the cylinders. For the important case of tube arrays, the values of S, J, and C_L to be employed in eq. N-1321(69) are much more uncertain than for single cylinders, as is evident by the considerable scatter in the experimental data (refs. [100], [102], [103], [104], [105], [139], and [140]).

N-1323 Flexible Cylinders

When the vortex shedding frequency f_s is sufficiently different from the structural natural frequencies, a condition called off-resonance, the representation of the vortex shedding lift force by F given in eq. N-1321(69) is valid, and is conservative if $C_L = 1$ and $J = 1$ is chosen. This conservative representation of the force can be extended to nonuniform loading, and modal response can be employed to simplify the analysis of cylinders where many modes are active. Normally, off-resonance response is small. However, as resonance is approached, large motions are encountered.

In the case of a single flexible or resiliently supported tube, once vibration begins the shedding frequency and the tube natural frequency can become synchronized if the two are sufficiently close. For a spring-supported cylinder in an air stream, it was shown (ref. [106]) that the velocity range over which synchronization persists depends upon the damping parameter, $m_t\delta_n/\rho D^2$. In Figure N-1323-1, the shaded area is the region of synchronization. The ordinate, $V/f_n\ D$, is a reduced velocity, where f_n is the natural frequency of the spring-mounted cylinder. Note, in particular, that with increasing $m_t\delta_n/\rho D^2$ the reduced velocity range over which synchronization persists decreases, and no synchronization occurs for $m_t\ \delta_n/\rho D^2 > 32$. Outside the shaded area, the cylinder experiences an alternating lift force at the vortex shedding frequency for a stationary cylinder, as given previously in eq. N-1321(69).

The consequences of synchronization are many. As the flow velocity is either increased or decreased so that the vortex shedding frequency approaches the structure frequency, the following will occur.

(a) The vortex shedding frequency shifts to the structural natural frequency, i.e., it synchronizes with or "locks-in" to the structural frequency even if the flow velocity or the structural frequency is varied within the range of sychronization as indicated in Figure N-1323-1.

(b) The spanwise correlation of the vortex shedding forcing function increases rapidly as structural response increases.

(c) The lift force becomes a function of structural amplitude.

(d) The drag force on the structure increases.

(e) The strength of the shed vortices increases.

Within the synchronization band, substantial resonant vibration in lightly damped structures often occurs. Vibration amplitudes up to three diameters peak-to-peak have been observed in dense fluids, such as water, over cables and tubing. The vibrations are predominantly transverse to the flow and they are self-limiting (refs. [82], [87], and [94]).

Large-amplitude, synchronized vibrations in the drag direction have been observed for a single cylinder in water. These oscillations initiate at relatively low flow velocities corresponding to subharmonic frequencies of vortex shedding, i.e., at $^1/_4$, $^1/_3$, or $^1/_2$ the flow velocity required for synchronization according to eq. N-1321(68) (refs. [107] and [108]). However, the synchronization in the drag direction is not as strong as in the lift direction, and usually occurs only for lightly damped structures in

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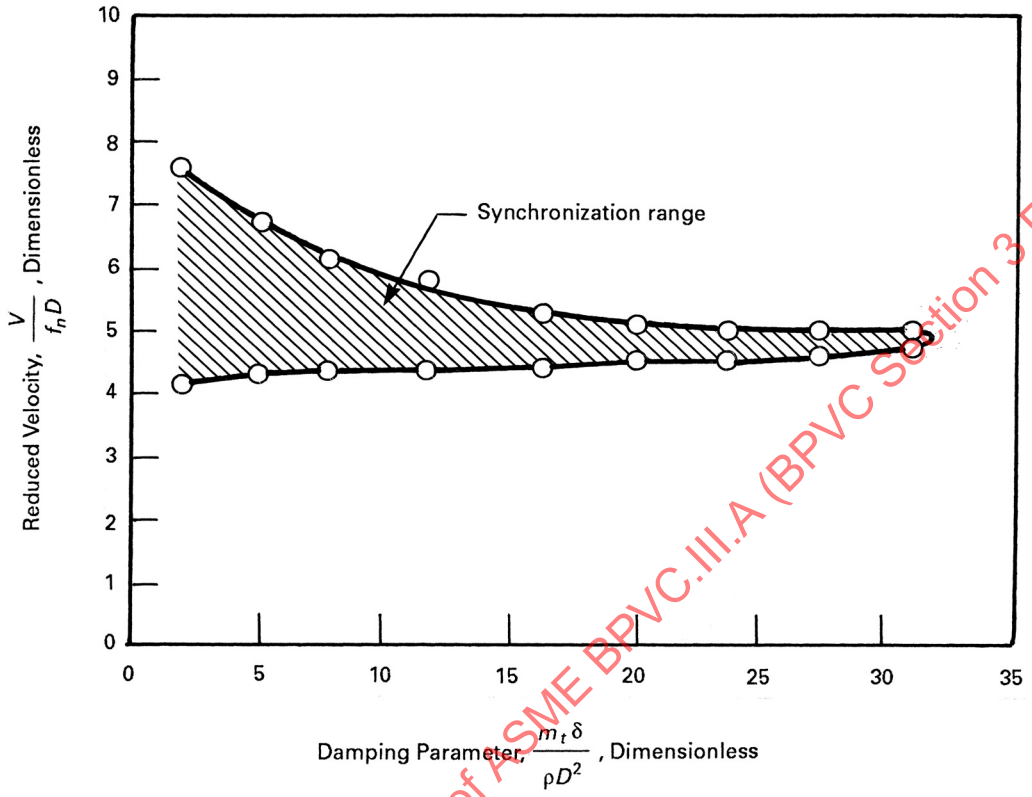
Figure N-1323-1
Synchronization of the Vortex Shedding Frequency and the Tube Natural Frequency for a Single, Flexibly-Mounted Circular Cylinder



GENERAL NOTE: Synchronization occurs within the shaded region. (Ref. [106])

dense fluids (refs. [87] and [110]). Lock-in has not been observed in two-phase flow or deep (more than a few rows) inside a closely spaced tube bundle.

N-1324 Design Procedures for a Circular Cylinder

Whenever possible, lock-in operating conditions should be avoided, but complex designs often make this impossible. Thus, criteria are given for which lock-in can be avoided, and off-resonance structural dynamic analysis can be employed, as well as design procedures to calculate the response during lock-in.

N-1324.1 Avoiding Lock-In Synchronization. Lock-in for a single cylinder can be avoided by one of the following four methods (refs. [82], [106], [108], and [109]). For tube arrays, only (a), (b), and (c) are applicable methods, and V must be the flow velocity in the minimum gap $(P - D)$.

(a) If the reduced velocity for the fundamental vibration mode $(n = 1)$ satisfies:

$$V/f_1 D < 1 \tag{72}$$

then both lift and drag direction lock-in are avoided.

(b) If for a given vibration mode the reduced damping is large enough

$$C_n > 64 \tag{73}$$

then lock-in will be suppressed in that vibration mode.

(c) If for a given vibration mode

$$V/f_n D < 3.3 \tag{74}$$

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and

$$C_n < 1.2 \quad (75)$$

then lift direction lock-in is avoided and drag direction lock-in is suppressed.

(d) If the structural natural frequency falls in the ranges $f_n < 0.7f_s$ or $f_n > 1.3f_s$, then lock-in in the lift direction is avoided in the nth mode.

The reduced damping C_n is calculated according to

$$C_n = \frac{4\pi\xi_n M_n}{\rho D^2 \int_{L_e} \phi_n^2(x)\,dx} \quad (76)$$

where $\xi_n = \delta_n/2\pi$ is the fraction of critical damping measured in air and M_n is the generalized mass

$$M_n = \int_0^L m_t(x)\,\phi_n^2(x)\,dx \quad (77)$$

with ϕ_n the nth mode shape function and $m_t(x)$ is the cylinder mass per unit length. The range L_e in the denominator implies that the integration is over only the region of the cylinder length subject to lock-in cross flow. Note that m_t is calculated according to:

$$m_t(x) = m_s(x) + m_c(x) + m_A(x) \quad (78)$$

For an isolated cylinder, m_A is the displaced fluid mass. If sections of the cylinder are close to other bodies, then the possibility of increased added mass and fluid damping must be taken into account (refs. [81], [82], [90] to [93]).

N-1324.2 Vortex-Induced Response. Off resonance, the response can be calculated using standard methods (ref. [96]) of forced-vibration analysis and eq. N-1321(69) for the forcing function (refs. [82] and [89]). The resultant response is ordinarily small. If operating conditions are such that lock-in cannot be avoided or suppressed, then the resonant vortex induced response must be calculated. Three approaches for calculating the response are recommended for three classes of structures and flows: single uniform cylinder in uniform flow, tube arrays, and nonuniform cylinders in nonuniform flow.

(a) Uniform Structure and Flow. If a uniform cylinder is subject to uniform cross flow over its span, then both the vortex shedding frequency and the vortex force are constant over the span of the cylinder. The periodic vortex induced lift force is given by eq. N-1321(69). At lock-in, the vortex shedding frequency equals the natural frequency of the nth vibration mode, $f_s = f_n$, and the cylinder response is given by (refs. [82] and [89])

$$\frac{y_n^*}{D} = \frac{C_L J \phi_n^*}{16\pi^2 S^2 \left[m_t \xi_n / \left(\rho D^2\right)\right]} \quad (79)$$

This equation provides a conservative upper bound estimate to the amplitude of periodic vortex-induced vibration if the lift coefficient is taken as unity, $C_L = 1$, and the vortex shedding is fully correlated along the span of the cylinder, $J = 1$. Other values of C_L and J may be used in circumstances where experimental data are available. However, eq. (79) with $C_L = 1$ and $J = 1$ has been found to give overly conservative predictions owing to the tendency of the actual lift coefficient to decrease at vibration amplitudes exceeding 0.5 diameter and the lack of perfect spanwise correlation at lower amplitudes. To obtain less conservative predictions, three semiempirical nonlinear methods are given in Table N-1324.2(a)-1. The mode shape factor γ generally varies between 1.0 and 1.3 (ref. [82]) and C_n is determined according to eq. N-1324.1(d)(76) using ξ_n determined in air.

(b) Within Tube Arrays. Coherent vortex shedding has been found to exist only in the first few rows in arrays of cylinders with center-to-center spacing less than 2 diameters, and the design procedures for a single cylinder are applicable using the velocity in the minimum gap $(P - D)$. Within the array vortex shedding exists over a broad range of frequencies rather than at a single distinct frequency. The response within the array is generally less than that of a comparable single cylinder. The techniques that have been developed to predict vibration within the array are based on the theory of random vibration and are given in N-1340.

(c) Nonuniform Structures and Flow. Many cylindrical structures have nonuniform distribution of mass and stiffness, and they are exposed to flow velocities that vary over the span. In this case, only a part of the span of structure will resonate with vortex shedding and contribute to the excitation.

One method for treating nonuniform structures in nonuniform flows is

(1) determine the natural frequencies and mode shapes of the structure.

Table N-1324.2(a)-1
Semiempirical Correlations for Predicting Resonant Vortex-Induced Vibration Amplitude

Reference	Predicted Resonant Amplitude
[111]	$\frac{y_n^*}{D} = \frac{1.29\gamma}{\left[1 + 0.43\left(2\pi S^2 C_n\right)\right]^{3.35}}$
[82]	$\frac{y_n^*}{D} = \frac{0.07\gamma}{\left(C_n + 1.9\right)S^2}\left[0.3 + \frac{0.72}{\left(C_n + 1.9\right)S}\right]^{\frac{1}{2}}$
[110]	$\frac{y_n^*}{D} = \frac{0.32\gamma}{\left[0.06 + \left(2\pi S^2 C_n\right)^2\right]^{\frac{1}{2}}}$

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(2) determine the spanwise distribution of the flow.

(3) identify portions of the structure that can resonate with vortex shedding for each mode. This can be done by calculating the spanwise distribution of the vortex shedding frequency and estimating the potential for resonance by a band of plus or minus 30% from this frequency.

(4) Apply a lift force given by eq. N-1321(69) with $f_n = f_s$ and $C_L = 1$ to those segments of the span that are resonant.

Procedures (1) through (4) are illustrated in refs. [89] and [112]. For a uniform cylinder in uniform cross flow, assumption (4) of complete correlation and $C_L = 1$ gives overly conservative predictions. Other values for C_L may be used where experimental data are available.

N-1330 FLUID-ELASTIC INSTABILITY

Many FIV mechanisms exist wherein as energy supplied to the system is increased, usually as increased flow velocity, a critical value is attained at which a large increase in response occurs. Continued increases in the supplied energy results in continued static or dynamic divergences (rapid increases) of the response. In general, fluid-elastic instability is a result of strong coupling between the structure and the fluid.

N-1331 Instability of Tube Arrays in Cross Flow

Fluid flow across an array of elastic tubes can induce a dynamic instability that can result in very large amplitude vibrations once a critical cross flow velocity is exceeded. Often, motion is limited only by tube-to-tube impacting. The flow of fluid over the tubes results in both fluid excitation and fluid-structure coupling forces on the tubes. The fluid-structure coupling excitation forces fall into several groups

(a) forces that vary approximately linearly with displacement of a tube from its equilibrium position (displacement mechanisms) (ref. [113]).

(b) fluctuations in the net drag forces induced by the oscillating tube's relative velocity with respect to the mean flow (fluid damping mechanism) (ref. [88]).

(c) combinations of the above forces that exhibit step changes as a certain amplitude is exceeded because of the abrupt shift in the point of flow separation (jet switch mechanism) (Ref. [114]). Instability may result from any or all of these fluid forces which are functions of the tube motion.

The general characteristics of tube vibration during instability are as follows.

(d) Tube Vibration Amplitude. Once a critical cross flow velocity is exceeded, vibration amplitude increases very rapidly with flow velocity V, usually as V^n where $n = 4$ or more, compared with an exponent in the range $1.5 < n < 2.5$ below the instability threshold. This can be seen in Figure N-1331-1, which shows the response of an array of metallic tubes to water flow. The initial hump is attributable to vortex shedding that tends to produce larger amplitudes in water flow than air flows.

(e) Vibration Behavior With Time. Often the large amplitude vibrations are not steady in time, but rather beat with amplitudes rising and falling about a mean value in a pseudorandom fashion (ref. [115]).

(f) Synchronization Between Tubes. Most often the tubes do not move as individuals, but rather move with neighboring tubes in somewhat synchronized orbits, as shown in Figure N-1331-2. This behavior has been observed in tests both in water and air (refs. [113], [115], [116], and [117]), with orbit shapes ranging from near circles to near straight lines. As the tubes whirl in their oval orbits they extract energy from the fluid. The stiffness mechanism requires motion of the adjacent tubes, but the damping mechanism does not.

(g) Influence of Structural Variations. Restricting the motion or introducing frequency differences between one or more tubes often increases the critical velocity for instability (refs. [115], [116], and [118]). Such increases are generally no greater than about 40%. Often the onset of instability is more gradual in a tube bank with tube-to-tube frequency differences than in a bank with identical tubes which are free to vibrate.

N-1331.1 Prediction of the Critical Velocity. Dimensional analysis considerations imply that the onset of instability is governed by the following dimensionless groups: the mass ratio $m_t/\rho D^2$; the reduced velocity V/fD; the damping ratio ξ_n, measured in the fluid; the pitch to diameter ratio P/D; the array geometry (see Figure N-1331-3), and the Reynolds number VD/ν. In this section, V is the flow velocity in the gaps between the tubes, and is determined by the product of $P/(P - D)$ and the (approach) flow velocity that would occur if the tubes were not present. Note the added mass part of m_t may be much larger than the displaced fluid mass because of the confining effect of adjacent tubes (refs. [81], [90], and [92]). Also, for most cases, the flow is fully turbulent ($VD/\nu > 2000$) and the Reynolds number is not expected to play a major role in the instability. In such cases, the reduced critical velocity for the onset of instability can be expressed as a function of the remaining nondimensional parameters.

The relationship between the parameters can be investigated theoretically or experimentally. One general form that has been used to fit experimental data is

$$V_c/f_nD = C\left(m_t/\rho D^2\right)^a\left(2\pi\xi_n\right)^b \qquad (80)$$

where C and the indices a and b are functions of the tube array geometry. Experimental data suggest that a and b fall in the range $0.0 < a, b < 1.0$ (refs. [115], [116], [119], [138], and [139]).

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Figure N-1331-1
Response of a Tube Bank to Cross Flow (Ref. [115])

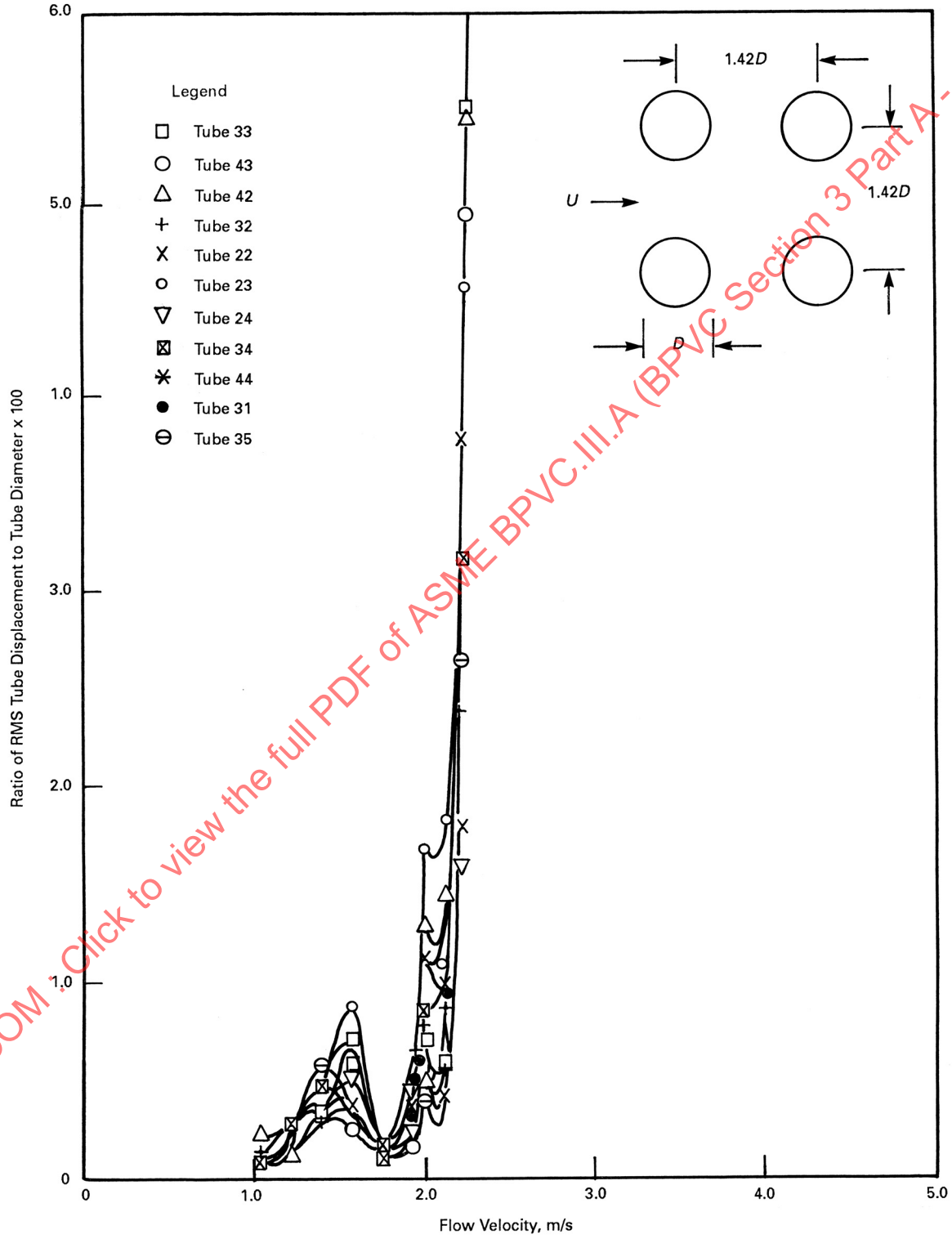

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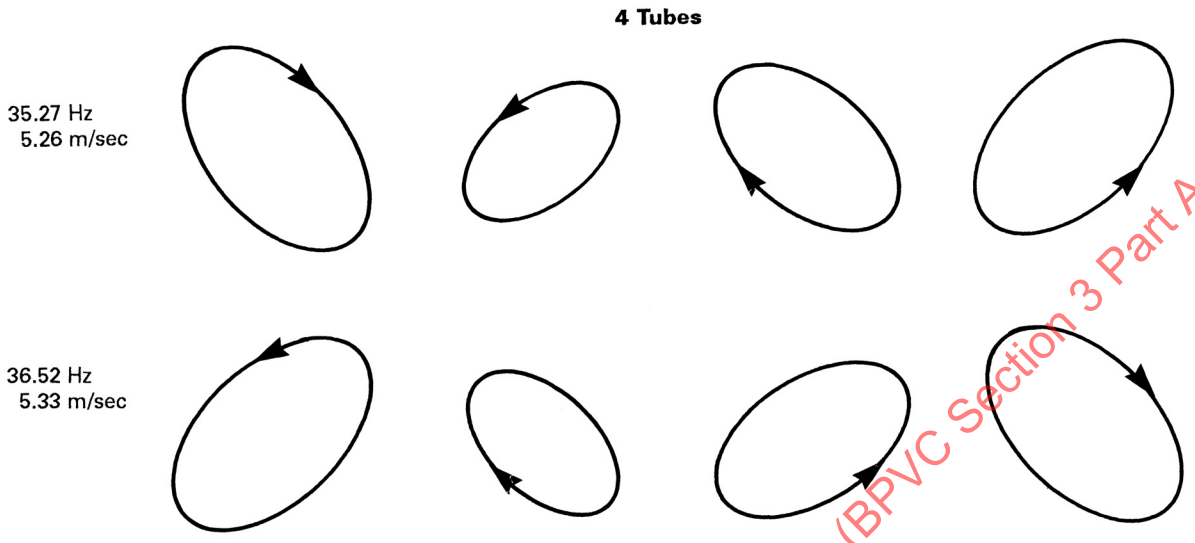
Figure N-1331-2
Tube Vibration Patterns at Fluid-Elastic Instability for a Four-Tube Row (Ref. [118])



N-1331.2 Recommended Formula. Mean values for the onset of instability can be established by fitting semi-empirical correlations to experimental data. The correlation form chosen is

$$V_c / f_n D = C\left[m_t\left(2\pi\xi_n\right)/\rho D^2\right]^a \qquad (81)$$

where

f_n = natural frequencies of the immersed tube
V_c = critical cross flow velocity

For uniform cross flow, the tubes will be stable if the representative cross flow velocity V is less than the critical velocity V_c. If the flow is nonuniform over the tube lengths, an equivalent uniform cross flow gap velocity can be defined as either the maximum cross flow velocity, or the modal weighted velocity:

$$V_e^2 = \int_0^L V^2\left(x\right)\phi_n^2\left(x\right)dx \Big/ \int_0^L \phi_n^2\left(x\right)dx \qquad (82)$$

where $V(x)$ is the cross flow velocity at each axial location of the tube. The tubes will be stable if $V_e < V_c$ for all modes.

The available 170 data points for onset of instability (ref. [120]) are shown in Figure N-1331-4. In the range $m(2\pi\xi_n)/\rho D^2 > 0.7$, there are sufficient data to permit fitting of eq. (81) to data for each array type. The mean values of C are

	Triangle	Rotated Triangle	Rotated Square	Square	All
C_{mean}	4.5	4.0	5.8	3.4	4.0

Based on theory for the displacement mechanism (Ref. [113]), which is active in this parameter range, $a = 0.5$ was chosen in these fits. For $m(2\pi\xi_n)/\rho D^2 < 0.7$, where the fluid damping mechanism is primarily active, neither the theory nor data are sufficient to establish values of C and a in eq. (81). Conservative estimates of the mean values of $V_c/f_n D$ for $m_t(2\pi\xi_n)/\rho D^2 < 0.7$ can be obtained using eq. (81) with $a = 0.5$ and the mean C given in the table above. The use of eq. (81) with $a = 0.5$ and $C = 3.3$ has been recommended (refs. [80], [100]) for the entire mass damping parameter range of Figure N-1331-4.

N-1331.3 Suggested Inputs. Accurately predicting the critical velocity requires scale model testing to determine the value of C and the damping ratio in each application, because practical flow and structural geometries contain features nonexistent in the simpler, controlled laboratory tests used to establish the data base of Figure N-1331-4 (ref. [120]). Usually, industrial tube arrays (bundles) involve multiple spans with intermediate supports provided by plates with holes slightly larger than the tube diameter. Also, flow may pass around the edge of the bundle and does not have the pure cross flow direction shown in Figure N-1331-3, even within the bundle. Furthermore, when the vibration amplitude is small, such as that experienced during subcritical vibration, not all support plates are active. Damping ratios in this vibration mode are typically small, from 0.1% in gas to about 1% in

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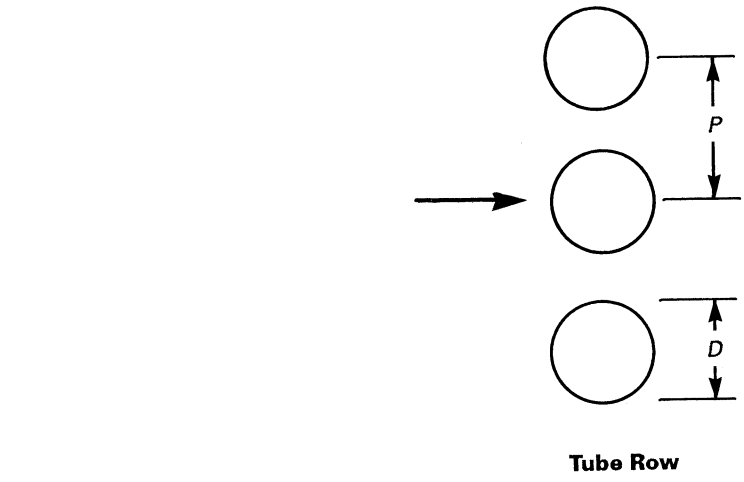
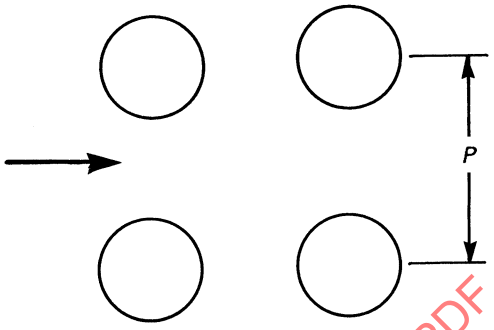
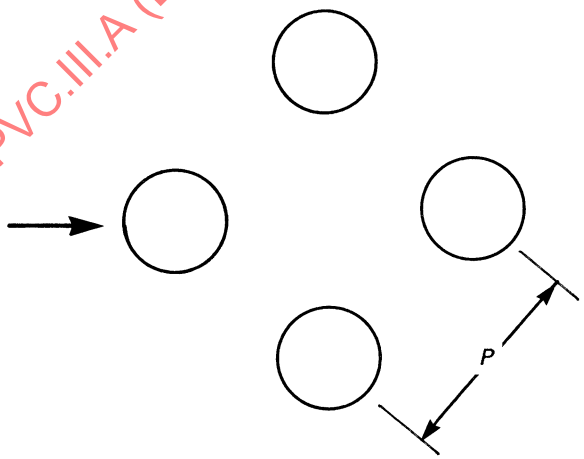
Figure N-1331-3
Tube Arrangements



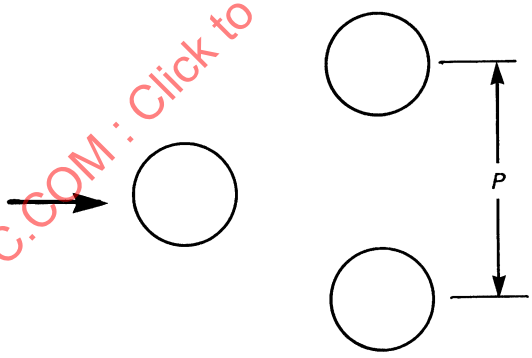
Tube Row



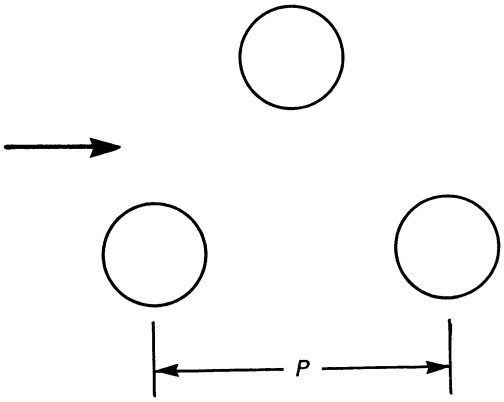
Square Array
(90 Deg)



Rotated Square Array
(45 Deg)



Triangular Array
(30 Deg)



Rotated Triangular Array
(60 Deg)

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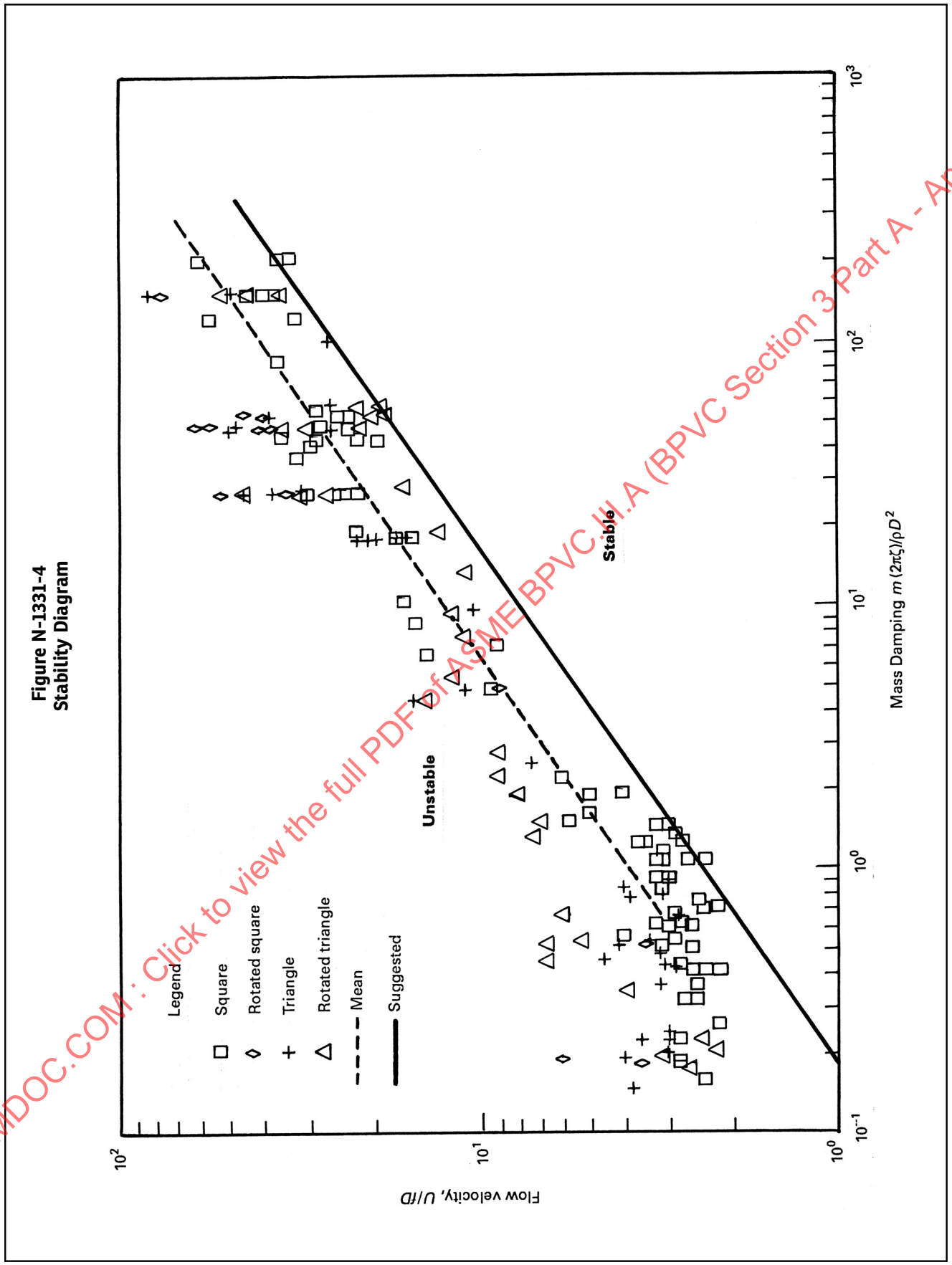

Figure N-1331-4
Stability Diagram

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steam or water. When the vibration amplitude is large, as characterized by the onset of instability, support plate-to-tube interaction greatly increases the damping ratio which can reach 5% or more.

All the practical features discussed above tend to raise the critical flow velocity. Thus, the data base of Figure N-1331-4 can be used to determine a conservative criterion for avoiding fluid-elastic instabilities of tube arrays: if the design equivalent uniform cross flow gap velocity [eq. N-1331.2(82)] is less than the critical velocity [eq. N-1331.2(81)] computed with the suggested design values defined by the solid line (C = 2.4, a = 0.5) in Figure N-1331-4 and with a damping ratio of 0.5% in gas, or 1.5% in "wet" steam or liquid, then instability is almost certainly not a problem, and scale model testing will not be necessary. Otherwise, more accurate values of C and the immersed tube's damping ratios, or the critical velocity itself, must be determined by either model testing or from operational experience.

N-1340 TURBULENCE

In general, the coolant flow paths and flow rates promote and maintain turbulent flows that are optimal for purposes of heat transfer, but provide sources for structural excitation. Also, turbulence in the flow can affect the existence and strength of other excitation mechanisms associated with separating boundary layer flows (wakes), as discussed in N-1320 on vortex shedding. This section will concentrate on turbulence as a source of fluid excitation forces.

N-1341 Random Excitation

Where turbulent flow comes into contact with the surface of a structure some of the momentum in the flow is converted into fluctuating pressures. In addition to any forces produced by the mean flow component, random surface pressure fluctuations are produced by the turbulent velocity component. The time history of the surface pressure fluctuations, like the flow turbulence, is complex and amenable to description only on a statistical basis. However, the fluctuating pressure and the resulting flow-induced response usually can be regarded as ergodic and analyzed with a finite-time record not dependent upon the time origin.

For purposes of structural analysis and design, most useful information on the fluctuating pressures becomes available once the spatial spectral densities of the pressure field, $S_p(x_1, x_2, \Omega)$, are determined. The spectral density is the Fourier transform of the cross correlation of the pressure field

$$S_p\left(x_1, x_2, \Omega\right) = \frac{1}{2\pi}\int_{-\infty}^{\infty}\left\{\lim_{t_0 \to \infty}\frac{1}{2t_0}\int_{\theta t_0}^{t} p\left(x_1, t\right) \times p\left(x_2, t+\tau\right) d\tau\right\} e^{-i\Omega t} dt \quad (83)$$

and provides information about the average products of components of the pressure $p(x, t)$ as a function of the circular frequency Ω in radian/sec for every possible pairing of structural points, x_1 and x_2, including the same point. S_p has units of (pressure)^2 (second). The frequency content of the spectra are band limited, from zero to a maximum frequency determined by the turbulence source. The magnitude of the power spectra increases when the energy of the turbulence at frequency Ω increases. The size of a region of the structure over which the pressures at different points are coherent, or has some cause-effect relationship, is interpreted as a correlation length of the pressure field, or the size of the associated turbulent eddy (ref. [82]).

Only select parts of the surface pressures are effective in exciting dynamic structural response: those parts with frequency content in narrow bands centered on the structural natural frequencies and with correlation lengths similar in size to the spatial wavelength of the associated vibration mode (ref. [122]). The resulting structural response occurs in the narrow frequency bands with random amplitudes, and the widths of the frequency bands are determined by the system damping. Knowledge of the surface pressure statistics enables prediction of the associated structural response statistics utilizing the probabilistic theory of structural dynamics.

N-1342 Structural Response of Tubes and Beams

N-1342.1 Response to Homogeneous Turbulence Excitation. (15) The assumption of a linear structure is justifiable for the small vibrations associated with the turbulence excitation of weakly coupled fluid-structure systems, and the linear structural dynamic analysis theory for arbitrary random loading of beams is highly developed. Since the energy dissipation mechanism of turbulent flow rapidly smooths disturbances caused by the structural boundaries of and in the flow channel, the statistical character of turbulent cross flow often varies gradually over the total length of select spans of single tubes and tube bundles, especially within the bundles. Thus, in many applications, the assumption of a uniform mean velocity and homogeneous turbulence is reasonable.

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Assuming a homogeneous and ergodic pressure field, the equations of motion can be uncoupled to allow solution by modal analysis (see N-1222). The expression for the power spectral density of the cylinder response is (ref. [123])

$$S_y\left(x_1, x_2, \omega\right) = S_{fo} L \sum_{j=1}^{\infty} \sum_{k=1}^{\infty} \phi_j \phi_k J_{jk}^2 H_j H_k^* \quad (84)$$

The mode shapes $\phi_j(x)$ satisfy the orthogonality relation

$$\int_0^L m_t\left(x\right) \phi_i\left(x\right) \phi_j\left(x\right) dx = M_j \delta_{ij} \quad (85)$$

where M_j is the generalized mass, defined here to have the same dimensions as m_t. Thus, if m_t is constant, $M_j = m_t$ and the orthogonality condition reduces to

$$\int_0^L \phi_i\left(x\right) \phi_j\left(x\right) dx = \delta_{ij} \quad (86)$$

The transfer function for the jth mode is

$$H_j\left(\omega\right) = \left[M_j\left(\omega_j^2 - \omega^2 + 2i\xi_j \omega \omega_j\right)\right]^{-1} \quad (87)$$

where ω_j and ξ_j are the modal natural frequency and damping, respectively. The acceptance integral is

$$J_{jk}^2\left(\omega\right) = L^{-1} \int_0^L \int_0^L \phi_j\left(x_1\right) \Gamma\left(x_1, x_2, \omega\right) \phi_k\left(x_2\right) dx_1\, dx_2 \quad (88)$$

where the complex coherence function is

$$\Gamma\left(x_1, x_2, \omega\right) = S_f\left(x_1, x_2, \omega\right) / S_{fo}\left(\omega\right) \quad (89)$$

and S_f is the cross spectral density of the turbulent forcing function per unit length between two different points on the cylinder's length, $x = x_1$ and $x = x_2$. When $x_1 = x_2 = x$, $S_f(x, x, \omega) = S_{fo}(\omega)$ is the power spectral density (or autospectrum) that is independent of location for a homogeneous pressure field. The joint acceptance $J_{jj}(\omega)$ reflects the relative effectiveness of the forcing function to excite the jth vibration mode while the cross acceptance $J_{jk}(\omega)$, $j \neq k$, reflects contributions due to coupling between different modes. In general, the responses in two different modes are dependent upon each other.

The mean square response $\bar{y}^2\left(x\right)$ is the most useful measure for the amplitude or stress and strain design, and is found by integration of the power spectral density of the response, $S_y(x, \omega) = S_y(x, x, \omega)$ over the frequency band, or

$$\bar{y}^2\left(x\right) = \int_{-\infty}^{\infty} S_y\left(x, \omega\right) d\omega \quad (90)$$

The distribution of the positive and negative peaks in displacement has been found for the fundamental mode of a rod in parallel flow to be Gaussian (refs. [81], [82], and [124]). Assuming a Gaussian fluctuating pressure distribution, a Rayleigh distribution is expected for the absolute amplitude of response (refs. [82] and [125]).

Based on physical reasoning and experimental data, the complex coherence function of the homogeneous turbulence pressure field for the tubes in cross flow has been characterized as (refs. [122] and [126])

$$\Gamma\left(x_1, x_2, \omega\right) = \exp\left[-2\left|x_1 - x_2\right| / l_c\right] \times \exp\left[-i\omega\left(x_1 - x_2\right) \sin\theta / U_c\right] \quad (91)$$

where $l_c \ll L$ is the correlation length, which is a measure of the coherence range of the turbulent pressure field; U_c is the convection velocity, or the velocity at which the turbulent eddies move with the flow; and θ is the angle between the direction of the flow and the normal to the axis of the tube. Note that $U_c/\sin\theta$ is the phase velocity of the pressure signal along the tube.

In the case of lightly damped structures with well-separated modes, cross modal contribution to the response can be ignored and eq. (90) can be analytically evaluated to be (refs. [112] and [125]):

$$\bar{y}^2\left(x\right) = \sum_j 2\pi \xi_j \omega_j S_y\left(\omega_j\right) = \sum_j \pi \xi_j f_j G_y\left(f_j\right) \quad (92)$$

In the second equality, the response is expressed in terms of the more commonly used engineering variables frequency f (in Hz) and the single-sided power spectral density G as a function of f, where

$$\begin{aligned} G\left(f\right) &= 4\pi S\left(\omega\right) \text{ for } f = \omega / 2\pi > 0 \\ &= 0 \text{ for } f < 0 \end{aligned} \quad (93)$$

Under the same assumption of light structural damping

$$\bar{y}^2\left(x\right) = \sum_j \frac{L G_f\left(f_j\right) \phi_j^2\left(x\right)}{64\pi^3 M_j^2 f_j^3 \xi_j} J_{jj}^2 \quad (94)$$

where $G_f(f_j)$ is the single-sided power spectral density in $(\text{force/length})^2/\text{Hz}$ generated by the turbulent pressure field at the natural frequency f_j of the jth vibration mode.

N-1343 Design Procedures for Tubes and Beams in Turbulent Cross Flow

In most situations, the component of turbulent flow normal to the axis of a cylinder is a more dominant excitation mechanism than the parallel component. The exception occurs when the flow direction is parallel or barely inclined to the cylinder axis. Thus, cross flow analysis using the component of flow normal to the cylinder should always be made, supplemented by the parallel flow analyses of N-1345 at small angles of inclination.

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The theory in the subsections that follow can be applied to one-dimensional structures in general, but the specific information on the statistics of the pressure field must be limited to tubes (circular cylinders) until adequate information is available for other beam cross sections.

(15) **N-1343.1 Uniform Cross Flow.** In the simplest case of uniform cross flow over the entire length of a lightly damped, rigid cylinder with an evenly distributed total mass and spring supports at both ends, $\phi = L^{-1/2}$, $\theta = 0$ in eq. N-1342.1(91), and the joint acceptance integral of eq. N-1342.1(88), with $i = j = 1$, reduces to (refs. [112] and [127])

$$J_{11}^2 \approx l_c/L, \qquad l_c \ll L \tag{95}$$

The correlation length l_c in most cross flows over a tube (circular cylinder) is no more than three diameters (see N-1322). Also, although eq. (95) were derived for the fundamental mode of the transverse vibrations of a rigid, spring-supported cylinder, they can be used to estimate the joint acceptance of the fundamental mode of cylinders which are simply supported or clamped at both ends. Of course, in determining the RMS response with eq. N-1342.1(94), the mode shapes corresponding to the actual boundary conditions and normalized according to eq. N-1342.1(86) are used. For other boundary conditions and higher modes, the joint acceptance integral will have to be evaluated either numerically (most cases) or in closed form from eq. N-1342.1(91). Since $J_{11}^2 \le 1.0$, (ref. [127]), an upper bound response estimate can be found by setting all the $J_{jj} = 1.0$ in eq. N-1342.1(94).

The random characteristics of the forces exerted on the tubes by the turbulent flow must be obtained from tests. Two expressions for the power spectral density of the turbulent force per unit length on tubes in a tube array are:

$$G(f) = \left[C_R(f)\rho V_g^2 D/2\right]^2 \tag{96}$$

(ref. [100]) and

$$G(f) = \left[C_L(f)\rho V_g^2 D/2\right]^2 (D/V_g) \tag{97}$$

(ref. [128]) where the gap velocity V_g is related to the velocity upstream of the tubes, V_∞, by

$$V_g/V_\infty = P/(P - D) \tag{98}$$

The coefficient $C_R(f)$ in eq. (96) has units of $\sec^{-1/2}$ and is given in Figure N-1343-1 as a function of the frequency f. Therefore, the application of eq. (96) should be limited to the parameter range for which the data were taken, namely, high turbulent water flow (1 to 2 m/sec) entering closely spaced heat exchanger tubes of 12 to 19 mm in diameter. The decrease in $C_R(f)$ with penetration into the bundle is attributed to the highly turbulent inlet flow and possible vortex excitation observed in the first few tube rows. Data for the nondimensional lift coefficient $C_L(f)$ of eq. (97) have not been obtained for as many tube array configurations (refs. [141] and [142]), but use of this alternative expression may predict less conservative responses (ref. [142]).

For an isolated tube in cross flow, which is not subject to conditions of lock-in vortex shedding (see N-1323), the power spectral density $G_f(f)$ and the correlation length are strong functions of the turbulence created in the incident flow stream by the upstream structures. Relatively small amounts of turbulence can cause significant reductions in the effectiveness of vortex shedding as an excitation mechanism, and all periodicity can be eliminated with sufficiently strong incident turbulence. For given turbulence intensities and scale lengths of the incident flow, $G_f(f)$ is available (ref. [129]) and the correlation length l_c may be approximated by the scale length of the incident flow. In the absence of specific information about the incident flow, the random turbulence coefficient for the upstream tube in Figure N-1343-1 can be used to estimate $G_f(f)$ for most isolated tubes in cross flow because of the wide variety of incident flow conditions contained in the data base. In the latter case, the velocity used in eqs. (96) and (97) should be the free stream velocity of the flow.

If upstream structures produce well defined vortices, strong excitation mechanisms may be created on isolated cylinders more than twenty diameters downstream (ref. [130]). Such configurations should be avoided.

(15) **N-1343.2 Multiple Spans of Uniform Cross Flow.** In many applications, a cylinder is subject to one or more partial spans of uniform, but different, velocity and density cross flows that are uncorrelated with each other or with the flow over the remainder of the cylinder's length. These conditions often exist when baffles are used to channel different density flows in the interior of the pressure vessels (heat exchangers, reactors, etc.). The mean square response for such conditions can be determined by simple generalizations of the results given in N-1342.1 for homogeneous turbulence excitation.

Since the uniform cross flow over the span of length L_i is uncorrelated with the uniform cross flows over the other spans, $G_f J_{jj}^2$ in eq. N-1342.1(94) can be calculated (ref. [127]) by summing the products of the locally defined spectra G_f^i and joint acceptances $\left(J_{jj}^i\right)^2$ over all the spans i over which there is significant cross flow. Thus, the mean square response becomes

$$\bar{y}^2(x) \approx \sum_j \sum_i \frac{L_i G_f^i(f_j)\phi_j^2(x)}{64\pi^3 M_j^2 f_j^3 \xi_j}\left(J_{jj}^i\right)^2 \tag{99}$$

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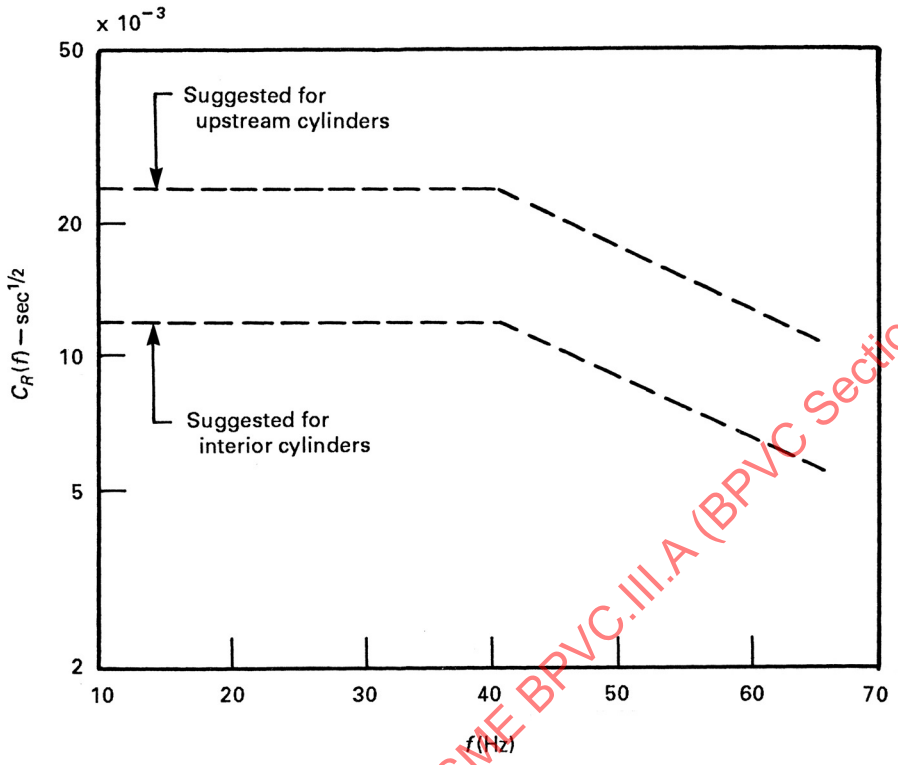
Figure N-1343-1
Random Excitation Coefficient for Arrays in Cross Flow (Ref. [100])



where

$$G_f^i\left(f_j\right) = G_f\left(f_j\right)\int_0^{L_i}\phi_j^2\left(x\right)dx \qquad (100)$$

$\left(J_{jj}^i\right)^2$ are determined with eq. N-1342.1(88) using that part of ϕ_j, denoted by ϕ_j^i, that is active over the ith span with length L_i. As discussed in N-1343.1, if ϕ_j^i is similar to the fundamental mode shape of a one-span beam with simple or clamped supports at each end, then $\left(J_{jj}^i\right)^2 \approx \ell_c^i / L_i$. The correlation lengths inside a tube bundle are smaller than that for an isolated tube, being about 1–2 tube diameters. After specifying $G_f^i\left(f\right)$ and using eqs. N-1343.1(96) and N-1343.1(97) for instance, the mean square response can be determined.

N-1343.3 Nonuniform Cross Flow. In industrial heat exchangers, the cross flow velocities are seldom uniform over the entire length, or even one span of the tubes. While an average cross flow velocity can be used to estimate the force spectra in eqs. N-1343.1(96), N-1343.1(97), and N-1343.1(98), when the velocity distribution is available, better estimates can be obtained by using mode shape-weighted power spectral densities similar to the generalized forces used in deterministic analysis (ref. [127]):

$$G_f\left(f_j\right) = \left(D/2\right)^2 C_R^2\left(f_j\right)\int_0^L\left[\rho U^2\left(x\right)\right]^2\phi_j^2\left(x\right)dx \qquad (101)$$

for a single-span tube of uniform mass density and

$$G_f^i\left(f_j\right) = \left(D/2\right)^2 C_R^2\left(f_j\right)\int_0^{L_i}\left[\rho_i U^2\left(x\right)\right]^2\phi_j^2\left(x\right)dx \qquad (102)$$

for a multi-span of spanwise uniform mass density. These estimates are not rigorously derivable, but they will lead to more accurate estimates of response, especially when the peaks in the velocity distributions are close to the antinodes of the vibration modes.

N-1344 Vortex-Induced Vibrations in a Tube Bundle

The existence of vortex shedding deep in a tube bundle is much less clearly defined than for a single cylinder. Experimental measurements involving tube bundles

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showed that even if a resonance peak exists in the dynamic pressure power spectral density, it is much broader and not as well defined as in the case of a single tube. Furthermore, these peaks are bounded by the pressure power spectral density given by eq. N-1343.1(96). However, if lock-in vortex-induced vibration occurs in a particular span, the forcing function and the tube mode shape will be fully correlated and in-phase for that span. This means that the span joint acceptance J^i_{jj} = 1.0. To be conservative, a lock-in vortex-induced vibration amplitude deep in a multi-span tube bundle can be calculated by substituting J^i_{jj} = 1.0 into eq. N-1343.2(99), for all spans and all modes where lock-in cannot be avoided or suppressed according to N-1324.1.

Classical vortex shedding does occur in the boundary tubes. For the first two to three rows of tubes in a tube bundle, vortex-induced vibration analysis following the procedure outlined in N-1320 is recommended.

N-1345 Cylinders in Axial Flow

Turbulence generally is a much weaker excitation mechanism in axial flow compared with cross flow, where the flow separates from the vibrating body. Also, axial flow is a source of flow damping which increases with flow rate (refs. [81], [82], and [131]). As a result, RMS vibration amplitudes of tubes in axial flow are typically only a few percent of the tube diameter.

The surface pressure fluctuations that excite a tube in axial flow are due to many sources: local turbulence created by the shear flow in the developing boundary layer; free stream turbulence created by upstream disturbances (grid supports, abrupt changes in channel size, elbows, valves, etc.) that quickly attenuate downstream of the disturbance, localized acoustic noise (waves); and system acoustic noise that can propagate long distances (ref. [136]). For pipes and single rods in annuli subject only to fully developed flow, relatively general experimental characterizations of the homogeneous pressure fields are possible (ref. [131]), because they depend only on the local channel geometry and the flow rates. However, general characterizations have not been developed that account for upstream disturbances and adjacent bodies, although many specific systems have been studied (refs. [131] through [137]). Evidently, accurate predictions can be made when the pressure field is characterized in the same system as the response is measured, but the predictions from system to system may vary by an order of magnitude for the same axial flow velocity. Because response is usually much easier to measure than the pressure fluctuations necessary to characterize a pressure field, especially a nonhomogeneous one, empirical correlations of response have been developed for important component geometries (refs. [99] and [137]). The component and prototype tests upon which the correlations are based include all component geometries and excitation sources. Of course, the use of these correlations must be limited to the type of components and parameter variations for which they were developed.

N-1345.1 Recommended Design Procedures.

(a) When the characterization of the pressure field is available, then the response of the structure can be predicted by the general method outlined in N-1342. But, unlike cross flows, the convection velocity U_c is important and must be known in axial flow before the acceptance integral, eqs. N-1342.1(88) and N-1342.1(91), can be evaluated. In axial flows, it is not generally true that the larger the correlation length ℓ_c, the larger the response as in cross flows. Rather, the response is governed by the matching of the structural mode shape and the phase-coherence of the pressure field (refs. [112] and [122]), in addition to its power spectral density.

(b) Regardless of whether a pressure field characterization is available, the maximum amplitude of motion can be estimated to within an order of magnitude using the empirical correlation (ref. [99])

$$\frac{y^*}{D} = \left[\frac{5 \times 10^{-4} K_N}{\alpha^4}\right]\left[\frac{u^{1.6}\epsilon^{1.8}Re^{0.25}}{1+u^2}\right] \times \left[\frac{D_h}{D}\right]^{0.4}\left[\frac{\beta^{2/3}}{1+4\beta}\right] \tag{103}$$

if the cylinder parameters are within the ranges covered by the correlation:

$$2.1 \times 10^{-3} \le u^2 = m_A V^2 L^2 / (EI) \le 8 \times 10^{-1}$$

$$26.8 \le \epsilon = L/D \le 58.7$$

$$2.6 \times 10^4 \le Re = VD/\nu \le 7 \times 10^5$$

$$4.9 \times 10^{-4} \le \beta = m_A / m_t \le 6.2 \times 10^{-1}$$

$$2.10 \le \alpha^2 = \Omega_1 \left[m_t L^4 / EI\right]^{1/2} \le 20.8$$

$$1 \le K_n \le 5$$

where K_n is a noise factor representing a departure from quiet, steady axial flows of K_n = 1. Commercial systems are expected to be bounded by K_n = 5. E is the modulus of elasticity, I is the beam area moment of inertia, and ν is the fluid kinematic viscosity.

N-1400 DYNAMICS OF COUPLED FLUID-SHELLS

N-1410 INTRODUCTION

It is well known that the motion of a solid in a heavy liquid is different from that in a vacuum. In the case of a sphere or a cylinder moving in an infinite fluid medium, for example, the presence of the fluid can be accounted for by simply adding to the physical mass of the solid, the mass of fluid it displaces. Hence the term added mass,

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or hydrodynamic mass, has been used to describe this solid-fluid interaction phenomenon. When the fluid is viscous, then in addition to an added mass, there is an apparent added damping, or hydrodynamic damping, due to the viscosity of the fluid.

In the case of thin cylindrical shells vibrating with fluids entrapped in-between, the effect of fluid-structure interaction is far more complicated. Data from preoperational tests of nuclear plants showed that the natural frequencies of the thermal shields of light water nuclear reactors were much lower than their corresponding values measured in-air. Since then, theoretical studies by Fritz and Kiss (ref. [143]), Horvay and Bowers (ref. [144]), Au-Yang (refs. [93] and [145]), Chen (refs. [90] and [146]), Krajcinovic (ref. [147]), and Levin and Milan (ref. [148]) conclusively showed that the coupling effect of narrow fluid gaps on thin cylindrical shells is much stronger than that of infinite fluid media. Their conclusions are supported by laboratory tests (refs. [90] and [149]). A few review papers on this topic were written by Chen (ref. [90]), Brown (ref. [150]), and Au-Yang (ref. [151]).

This guide describes a simple method to account for the effect of fluid-structure interaction on the response of cylindrical shells coupled by fluid gaps, using the structural priority approach (ref. [155]). In this approach, the effect of fluid-structure interaction is completely accounted for by the "added mass" and the "added damping" terms. The principal advantage of this method is its simplicity. Once the added mass and the added damping terms are computed, the structural response analysis can be carried out by standard methods without any need to revise the computer programs either for calculating the structural response or for calculating fluid forcing function. On the other hand this method is based on linear dynamic theory in which the principal interest is to estimate low-frequency structural responses. The vibration amplitude of the structure must be small compared with the width of the fluid gaps in order that linear dynamic theory holds. Internal structures of commercial nuclear reactors such as the core support structure and the thermal shield are usually restricted by limiter blocks so that their motions are small compared with the fluid annular gap width. In flow-induced vibration, seismic and loss-of-coolant analyses of these components, usually only the responses of the lower few modes are of interest, and the methods described in this guide can be applied. Structural analyses involving large displacements or in which responses to the high frequency components of the forcing function cannot be ignored, or analyses in which the principal interest is to estimate the hydraulic forcing function rather than the structural responses, should not follow the methods described in this guide.

N-1420 NOMENCLATURE

a = radius of inner cylinder or (when used as a subscript or superscript) pertaining to the inner cylinder
b = radius of outer cylinder or (when used as a subscript or superscript) pertaining to the outer cylinder
c = velocity of sound
$[C]$ = equivalent viscous damping coefficient matrix
$C_{\alpha m}$ = Fourier coefficient of the mth cylinder axial mode onto the αth acoustic mode
f = frequency in Hz
$\{f\}$ = reaction force due to fluid shell coupling
$\{f_0\}$ = externally applied force
$f_{m,n}$ = natural frequency of the (m,n)th mode
H = hydrodynamic mass
h = Fourier component of the hydrodynamic mass matrix element
$\bar{H}$ = generalized hydrodynamic mass
I = modified Bessel function of first kind
J = Bessel function of first kind
K = modified Bessel function of second kind
$[K]$ = stiffness matrix
K_{ij} = element of the stiffness matrix
ℓ = length of cylindrical shell
m = axial modal index of the cylindrical shell
n = circumferential modal index of the cylindrical shell
p = pressure
q = (u, v, w), displacement vector of the shell
R = radial function of acoustic mode
r = radial coordinate or (used in Figure N-1451-1, $= a/b$)
u = displacement of the shell in the axial direction
v = displacement of the shell in the tangential direction
w = displacement of the shell in the normal direction
x = axial coordinate
α = axial mode number for the acoustic mode
β = circumferential mode number for the acoustic mode
δ_{33} = $\begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 1 \end{bmatrix}$ matrix in the (u, v, w) space
ϵ = α or $(2\alpha - 1)/2$ as defined in eq. N-1440(108)
θ = azimuthal coordinate
Θ = mode shape function
λ = defined in eq. N-1451.2(115)
μ = surface mass density of the shell
ν = hysteretic damping factor; kinematic viscosity
ρ = mass density of fluid
ϕ = axial acoustic mode shape function
ψ = axial cylinder mode shape function
ω = angular frequency (rad/sec)

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N-1430 FREE VIBRATION OF THIN CYLINDRICAL SHELL IN AIR

Figure N-1430-1 shows the vibration mode of a thin, finite cylindrical shell. Viewed from the ends, the vibration of the cylinder may consist of any number of waves distributed around the circumference. The number of circumferential waves is denoted by n, with $n = 1$ being the beam mode. Throughout this write-up, it is assumed that the circumferential mode shape is normalized to unity,

$$\int_0^{2\pi} \theta^2\left(\theta\right) d\,\theta \;=\; 1 \qquad (104)$$

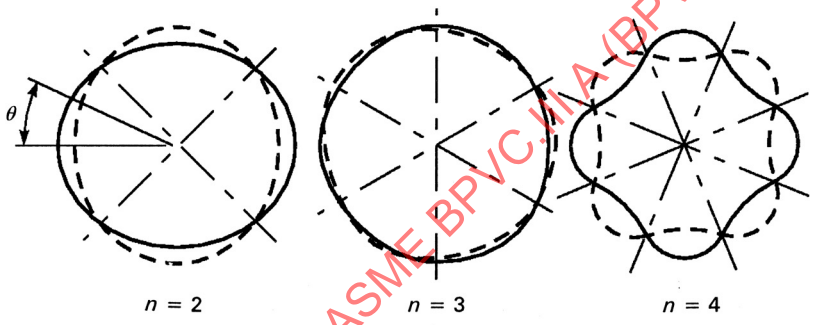
When viewed from the side, the deformation of the cylinder consists of a number of waves distributed along the length of a generator. The number of half waves along a generator is denoted by m, $m = 1, 2, 3,...$ As shown in Figure N-1430-1, the axial wave forms depend on the end conditions of the cylinder.

The equation of motion for the free vibration of the shell is

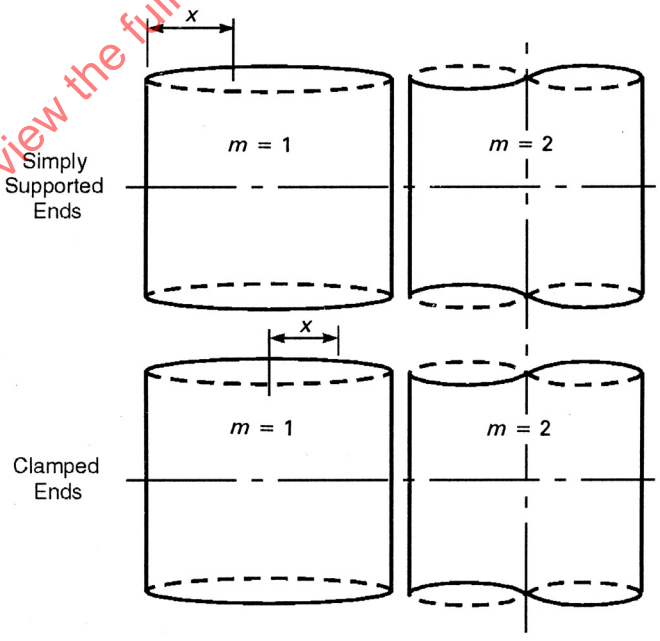
$$[\mu]\{\ddot{q}\} + [K]\{q\} \;=\; 0 \qquad (105)$$

Thus, to determine the natural frequencies, one has first of all to calculate the modal stiffness matrix. In simple cases of cylinders with simply supported or clamped ends, simplified expressions for the stiffness matrix

Figure N-1430-1
Vibration Forms for Circular Cylindrical Shells



Circumferential Vibration Forms



Axial Vibration Forms

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elements have been derived (refs. [92] and [152]), and the natural frequencies of the shell can be computed. There will be three roots for each set of K_{ij}, corresponding to three natural frequencies for each set of m,n. Two of these frequencies are usually much higher than the third one. These are associated with vibrations predominantly in the axial and tangential directions. The lowest frequency, on the other hand, is associated with vibrations predominantly in the normal direction. This is the flexural mode and is usually the mode of interest in structural analysis.

N-1440 ACOUSTIC MODES OF A FLUID ANNULUS BONDED BY RIGID WALLS

The natural frequencies associated with the (α,β)th mode of the fluctuation pressure distribution

$$p_{\alpha\beta} = R_{\alpha\beta}(r)\phi(x)\cos\beta\theta \tag{106}$$

inside the fluid annulus are given by the roots of the equation (ref. [153])

$$J'_\beta(x_\alpha a)Y'_\beta(x_\alpha b) = J'_\beta(x_\alpha b)Y'_\beta(x_\alpha a) \tag{107}$$

where J, Y are Bessel functions and

$$x_\alpha{}^2 = (w/c)^2 - (\epsilon\pi/\ell)^2 > 0 \tag{108}$$

where

ϵ = α if the fluid annulus has both pressure released (p = 0) or both hard (p = max.) ends
= $(2\alpha - 1)/2$ if one end is pressure released and the other end is hard

Note that acoustic modes exist in the fluid annulus only if

$$\omega/c > \epsilon\pi/\ell$$

The frequency at which,

$$\omega/c > \epsilon\pi/\ell$$

is known in acoustics as the coincidence frequency.

N-1441 Frequency Equation and Mode Shape for a Thin Fluid Annulus

For many applications, the fluid annulus is thin compared with its radius, so that the condition $(b - a)/a > 5$ is satisfied. Under this condition, the following approximate equation for the natural frequencies holds true for the plane wave mode (no radial node):

$$f_{\alpha\beta} = (c/2\pi)\left[(\epsilon\pi/\ell)^2 + [2\beta/(a+b)]^2\right]^{1/2}$$

The second radial mode (with one radial node) usually has a lowest frequency much higher than that of the fundamental radial mode up to $\alpha = \beta = 5$. Thus, it can be ignored if only the lowest 20 or so acoustic modes are included in the analysis.

N-1450 FREE VIBRATION OF COUPLED FLUID-SHELL SYSTEMS

When the cylindrical shells bounding the fluid annulus are flexible, then not only are the motions of the shells coupled to the fluid, but they are coupled together by the fluid between them. It was shown that (ref. [145]) there is no cross circumferential model coupling between the fluid and a system of two coaxial, circular cylindrical shells. However, except in the rare case when both the cylinder and the fluid can be represented by the same mode shape function, as is the case of a simply supported cylinder vibrating in a fluid annulus with open ends, there will be coupling between the axial structural and acoustic modes (refs. [93] and [145]).

N-1451 The Hydrodynamic Mass Matrix

It was shown that for small motions of the shell, the pressure induced by the structure in the fluid is proportional to the normal component of the acceleration $\ddot{w}$, of shell (refs. [93], [145], and [155]):

$$\{p\} = [H]\{\ddot{w}\} \tag{109}$$

The "constant" of proportionality is commonly called the hydrodynamic mass or added mass matrix. In general, it is a full matrix, the element of which is dependent on the frequency. For two coaxial cylindrical shells coupled by a fluid gap, the hydrodynamic mass matrix elements are (refs. [93] and [145])

$$\begin{aligned} H^{ab}_{mn} &= \sum_\alpha C^a_{\alpha m} C^b_{\alpha m} h^{ab}_{\alpha n}, \qquad H^a_{mn} = \sum_\alpha \left(C^a_{\alpha m}\right)^2 h^a_{\alpha n} \\ H^{ba}_{mn} &= \sum_\alpha C^a_{\alpha m} C^b_{\alpha m} h^{ba}_{\alpha n}, \qquad H^b_{mn} = \sum_\alpha \left(C^b_{\alpha m}\right)^2 h^b_{\alpha n} \end{aligned} \tag{110}$$

where $C^a_{\alpha m}$, for example, is the projection of the mth axial mode shape, ψ_m of cylinder a (the inner cylinder) onto the αth axial acoustic mode shape ϕ_α of the fluid annulus:

$$C^a_{\alpha m} = \int_0^\ell \psi^a_m(x)\,\phi_\alpha(x)\,dx \tag{111}$$

C^b_α are similarly defined. h (and hence H) are in units of mass per unit area.

The general expressions for the hydrodynamic mass which are valid for any boundary condition and any D/ℓ ratio, are quite complicated, but still can easily be

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computed with the aid of a personal computer. In the following paragraphs, simplified expressions for computing h are given for several commonly encountered special cases.

(15) **N-1451.1 Slender Cylinder Approximation.** When the conditions $|\epsilon\pi a/\ell - \omega a/c| \ll 1$ and $|\epsilon\pi b/\ell - \omega b/c| \ll 1$ are simultaneously satisfied, ref. [145] shows that:

$$\begin{aligned} h^{a}_{\alpha n} &= (\rho a/n)(b^{2n}+a^{2n})/(b^{2n}-a^{2n}) \\ h^{ab}_{\alpha n} &= -(2\rho b/n)(a^{n}b^{n})/(b^{2n}-a^{2n}) \\ h^{b}_{\alpha n} &= (\rho b/n)(b^{2n}+a^{2n})/(b^{2n}-a^{2n}) \\ h^{ba}_{\alpha n} &= -(2\rho a/n)(a^{n}b^{n})/(b^{2n}-a^{2n}) \end{aligned} \qquad (112)$$

Note that the hs are independent of the axial acoustical wave number α in this case. From N-1452.1, the generalized hydrodynamic masses per unit length for the beam mode ($n = 1$) are,

$$\begin{aligned} \bar{H}^{a} &= \rho\pi a^{2}(b^{2}+a^{2})/(b^{2}-a^{2}) \\ \bar{H}^{ab} &= \bar{H}^{ba} = -2\rho\pi ab/(b^{2}-a^{2}) \\ \bar{H}^{b} &= \rho\pi b^{2}(b^{2}+a^{2})/(b^{2}-a^{2}) \end{aligned}$$

N-1451.2 The Ripple Approximation. When the ratio $(b-a)/(b+a)$ is small, curvature effects become unimportant and the annular gap can be treated as a rectangular fluid region. In this case, the expressions for the hydrodynamic masses are greatly simplified to

$$h^{a}_{\alpha n} = h^{b}_{\alpha n} = \rho\left[\lambda_{\alpha n}\tanh(\lambda_{\alpha n}d)\right]^{-1} \qquad (113)$$

$$h^{ab}_{\alpha n} = h^{ba}_{\alpha n} = -\rho\left[\lambda_{\alpha n}\sinh(\lambda_{\alpha n}d)\right]^{-1} \qquad (114)$$

where

$d = b - a$ and

$$\lambda^{2}_{\alpha n} = \left(\epsilon\pi/\ell\right)^{2} + \left[2n/(a+b)\right]^{2} - \left(\omega/c\right)^{2} \qquad (115)$$

N-1451.3 Incompressible Fluid Assumption. When $\omega/c \ll \epsilon\pi/\ell$, the term ω/c can be dropped from the argument of the Bessel function in eq. N-1440(107), the resulting hydrodynamic masses are independent of frequency. Since this condition is satisfied when $c \to \infty$, it is called the incompressible fluid assumption. However, the incompressible fluid assumption must be used with caution for applications to large structures. For a structure 300 in. (7.6 m) long, vibrating in water at 600°F (315°C) (e.g., nuclear reactor internal components), the incompressible fluid assumption generally introduces errors in the hydrodynamic matrix elements of more than 10% for frequencies above 20 Hz. Thus, it should not be used in analyses involving rapid transients in which high frequency contributions to the response are significant.

N-1451.4 Single Beam Mode ($n = 1$) in Narrow Annuli. When a cylinder vibrates inside a stationary, rigid cylinder of slightly larger diameter, its added mass due to the water between the two cylinders can be estimated by a very simple equation (ref. [143]):

$$h^{a} = \rho a^{2}/\left[(b-a)(1+12a^{2}/\ell^{2})\right] \qquad (116)$$

The above equation is valid only for the beam mode ($n = 1$) vibration. Figure N-1451-1 is a comparison between eq. (116) and the exact solution and shows surprisingly good agreement between them as long as the fluid gap is small. From N-1452.1, the generalized, or effective hydrodynamic mass per unit length of the cylinder is:

$$\bar{H}^{a} = \frac{\rho\pi a^{2}}{(b-a)/a}\cdot\frac{1}{1+12a^{2}/\ell^{2}} \qquad (117)$$

N-1451.5 Single Cylinder Containing Fluid. The case when there is only one cylinder containing a fluid is given by (ref. [145]):

$$\begin{aligned} h_{\alpha n} &= \rho I_{n}(x_{\alpha}a)/[x_{\alpha}I'_{n}(x_{\alpha}a)] && \text{for } f < c\epsilon/2\ell \\ &= \rho J_{n}(x_{\alpha}a)/[x_{\alpha}J'_{n}(x_{\alpha}a)] && \text{for } f > c\epsilon/2\ell \\ &= \rho a/n && \text{for } f = c\epsilon/2\ell \end{aligned} \qquad (118)$$

where J_n, I_n are Bessel and modified Bessel functions of the first kind. From N-1452.1, for the beam mode ($n = 1$), the generalized hydrodynamic mass per unit length of the cylinder is,

$$\bar{H}_{1} = \pi ah\alpha l$$

N-1451.6 Single Cylinder in Infinite Fluid. The case where there is only one cylinder in an infinite fluid is given by

$$\begin{aligned} h_{\alpha n} &= -\rho K_{n}(x_{\alpha}a)/[x_{\alpha}K'_{n}(x_{\alpha}a)] && \text{for } f < c\epsilon/2\ell \\ &= -\rho Y_{n}(x_{\alpha}a)/[x_{\alpha}Y'_{n}(x_{\alpha}a)] && \text{for } f > c\epsilon/2\ell \\ &= \rho a/n && \text{for } f = c\epsilon/2\ell \end{aligned} \qquad (119)$$

From N-1452.1, for the beam ($n = 1$) mode, the generalized hydrodynamic mass per unit length is,

$$\bar{H}_{1} = \pi ah\alpha l$$

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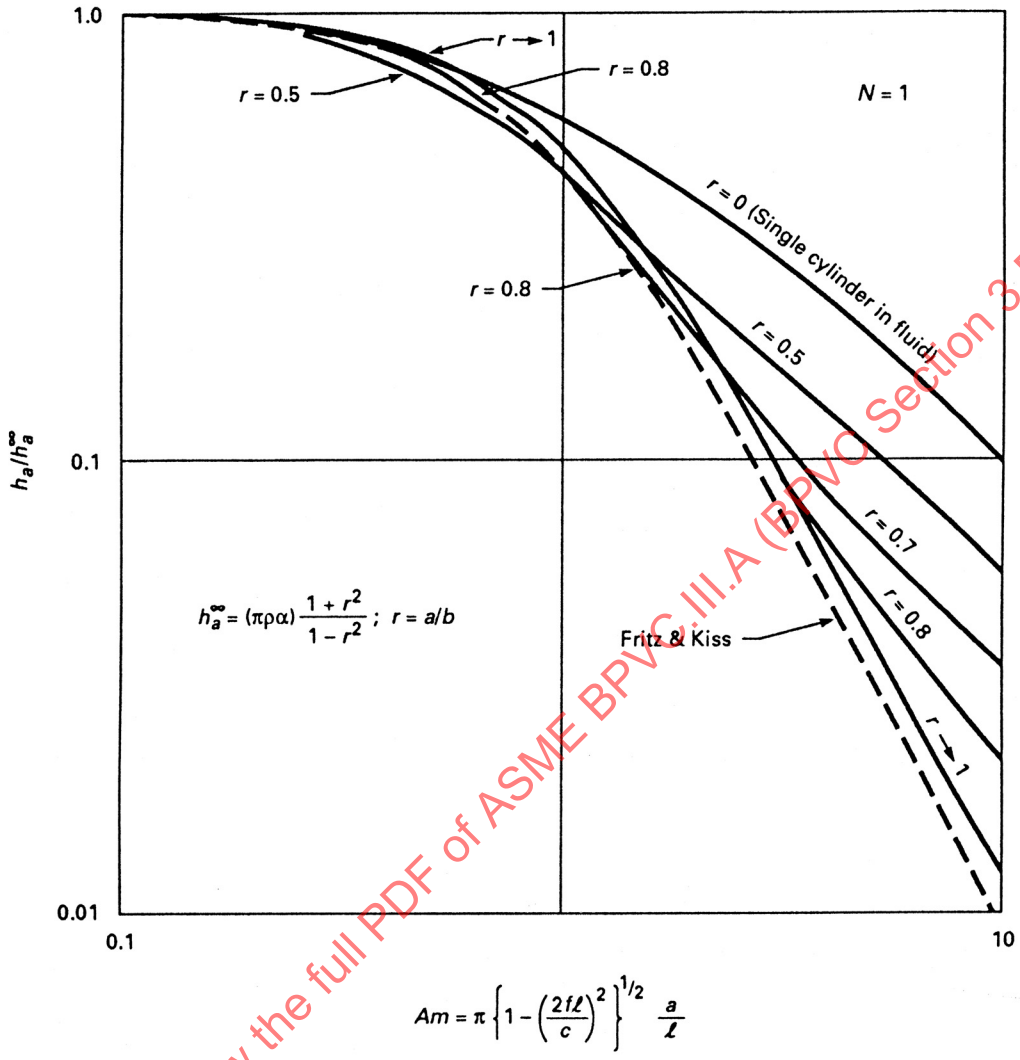
Figure N-1451-1
Comparison of Fritz and Kiss Solution With Exact Solution



GENERAL NOTE: h_a^∞ is the Fourier coefficient for the hydrodynamic mass surface mass density of an infinitely long cylinder.

N-1452 Natural Frequencies of Coupled Fluid-Shells in Special Cases

Once the hydrodynamic mass matrix is computed, it can be input into finite element computer programs to calculate the natural frequencies of fluid coupled shells. First, several special cases in which simplified method of analysis can be used will be considered.

N-1452.1 One-Axial Mode Approximation. In many cases of practical importance, when the D/ℓ ratio of the cylinders are not very small, only the first axial mode is important in the dynamic analysis. Under these circumstances the frequencies of two coaxial cylindrical shells coupled by a fluid gap between them can be estimated by solving the following simplified coupled equations (ref. [154])

$$\begin{aligned}\left(\mu^a + \overline{H}^a_{\ln}\right)\left[\left(f^a_{\ln}\right)^2 - f^2\right]w^a - f^2\overline{H}^{ab}_{\ln}w^b &= 0\\ -f^2\overline{H}^{ba}_{\ln}w^a + \left(\mu^b + \overline{H}^b_{\ln}\right)\left[\left(f^b_{\ln}\right)^2 - f^2\right]w^b &= 0\end{aligned} \qquad (120)$$

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where $[\bar{H}]$ is the generalized hydrodynamic mass matrix with elements:

$$\begin{aligned} \bar{H}^a_{\ln} &= \{q^a\}^T [H^a \delta_{33}] \{q^a\} [\{q^a\}^T \{q^a\}]^{-1} \\ &= H^a_{\ln}/(1+n^{-2}) \\ \bar{H}^b_{\ln} &= H^a_{\ln}/(1+n^{-2}) \\ \bar{H}^{ab}_{\ln} &= H^{ab}_{\ln}/(1+n^{-2}) \end{aligned} \tag{121}$$

$\bar{H}^a_{\ln}$, for example, is the generalized hydrodynamic mass of the inner cylindrical shell, expressed in terms of the hydrodynamic mass defined in eq. N-1451(110). From eq. (121) it can be seen that, because the hydrodynamic mass effects exist only in the normal direction, the effectiveness of H in reducing the natural frequencies of the shell is decreased by the factor $1 + n^{-2}$. For the beam mode ($n = 1$), its effectiveness is reduced by a factor of $\frac{1}{2}$. For high shell modes ($n \gg 1$), it is as effective as the physical mass.

For each n, two coupled frequencies and amplitude ratios for w_a/w_b can be obtained by solving the system of eq. (120). The torsional and longitudinal modes of vibration are not affected and are not coupled by the fluid gap in the inviscid fluid assumption. Note that $[H]$ is dependent on f, and eq. (120) has to be solved by iteration except in the incompressible fluid limit.

The above equations can be generalized to a system of several coaxial cylindrical shells coupled by fluid gaps between them (ref. [155]). However, since the hydrodynamic mass is frequency dependent, many iterations may be necessary to obtain all the coupled frequencies. This procedure may become prohibitive when there are more than three cylinders.

N-1452.2 Cases When Only One Cylinder is Flexible (ref. [156]). If only one cylinder is flexible, coupling between cylinders can be ignored. The only parameter of interest is the "in-water" frequency of the shell, which is also the natural frequency of the fluid–shell system. In this case, the natural frequencies can be obtained by a graphical method. If one plots the two functions

$$g_1 = f/f_{mn} \tag{122}$$

$$g_2 = \left[1 + \bar{H}_{mn}(f)/\mu\right]^{-1/2} \tag{123}$$

then where g_1 and g_2 intercept is a natural frequency of the coupled fluid–shell system. The lowest coupled frequency is usually referred to as the "in-water" frequency of the shell while the higher coupled frequencies are usually referred to as acoustic frequencies. This distinction, however, has no technical basis.

In the incompressible fluid limit, only the "in-fluid" natural frequency of the cylinder is of interest. This is given by

$$\bar{f}_{mn} = f_{mn}\left[1 + \bar{H}_{mn}/\mu\right]^{-1/2} \tag{124}$$

N-1453 Use of Hydrodynamic Mass Matrix in Finite Element Structural Computer Programs

Closed form solutions are possible only in highly idealized cases of perfect circular cylindrical shells with classical boundary conditions coupled by perfectly uniform annular gaps. In many applications, even though the fluid gaps are annular and the uncoupled fluid dynamic problem is amenable to closed form solution, closed form solutions to the coupled fluid shell problem are not possible because the cylindrical shells are not uniform, and the boundary conditions are not classically simply supported or clamped. Examples of this sort are common in the nuclear industry. In these cases, the structural problem is usually solved with the help of finite element computer programs.

The effect of fluid–structure interaction can be readily incorporated into the finite element computer program using closed form solutions for the hydrodynamic mass matrix, provided that the finite element computer program is designed to accept a full mass matrix. Many commercially available finite element computer programs have this capability.

It was shown in ref. [155] that in finite element form, the hydrodynamic mass matrix can be written as:

$$\left[M_H\right] = \left[\begin{array}{c|c} \dfrac{a^{-1}\sum_\alpha h^a_\alpha \left(\phi_\alpha \Delta A_a\right)\left(\phi_\alpha \Delta A_a\right)^T}{a^{-1}\sum_\alpha h^{ba}_\alpha \left(\phi_\alpha \Delta A_b\right)\left(\phi_\alpha \Delta A_a\right)^T} & \\ & \dfrac{b^{-1}\sum_\alpha h^{ab}_\alpha \left(\phi_\alpha \Delta A_a\right)\left(\phi_\alpha \Delta A_b\right)^T}{b^{-1}\sum_\alpha h^{b}_\alpha \left(\phi_\alpha \Delta A_b\right)\left(\phi_\alpha \Delta A_b\right)^T} \end{array}\right] \tag{125}$$

where ΔA_a, ΔA_b are element surface areas. These mass matrices can then be input into the finite element computer program just as the physical mass, with the exception that unlike the physical mass, the hydrodynamic mass, being a pressure, is effective only in the normal direction of the shell. Hence, the hydrodynamic mass should be associated only with the normal degree of freedom.

In the incompressible fluid approximation where the hydrodynamic mass matrix is independent of frequency, solution of the coupled fluid-shell dynamic problem is straightforward. In general, however, the hydrodynamic mass matrix is a full matrix with frequency dependent matrix elements. Solution of the coupled fluid-shell problem involves iteration between calculation of the mass matrix elements and the finite element computer program. When some of the coupled natural frequencies are near to the classical hard-walled acoustic modal

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frequencies, the hydrodynamic masses become extremely sensitive to the frequencies and solution to the coupled fluid-shell system can be difficult.

The method reported in ref. [149] greatly simplifies the above computational procedure by reducing the size of the matrix, while still giving reasonable estimates of the coupled frequencies of the fluid shell system. It consists of calculating the 2 × 2 "generalized" hydrodynamic mass matrix, $\overline{H}^{a}_{mm}$, $\overline{H}^{b}_{mn}$, $\overline{H}^{ab}_{mn}$, and $\overline{H}^{ba}_{mn}$ of the cylindrical shells a and b, and uniformly distributing these hydrodynamic masses over the cylindrical surfaces. For the off diagonal elements, $\overline{H}^{ab}_{mn}$, $\overline{H}^{ba}_{mn}$, it is assumed that coupling exists only for elements of the cylinders that are directly opposite to each other. The resulting mass matrix is tri-diagonal instead of full, and greatly simplifies the computational procedure. The tri-diagonal hydrodynamic mass matrix approach yielded natural frequencies of a coupled fluid-shell system that agree quite well with those obtained from closed form solution.

N-1460 FORCED RESPONSE OF COUPLED-SHELL SYSTEM

When the motion of the shells is small, the equation of motion for forced response of a coupled fluid-shell system can be written as

$$[\mu]\{\ddot{q}\} + [C]\{\dot{q}\} + [K](q) = \{f_0\} + \{f\} \qquad (126)$$

where ν is the equivalent hysteretic damping factor, f_o is the incident force acting on the shells, which is present irrespective of the motion of the shells. Examples of f_o are random pressure caused by turbulent boundary layers, or transient pulses caused by shock waves impinging on the shells. f is the reaction pressure induced on the shell as a consequence of its own motion: a result of the interaction (or coupling) between the fluid and the structure. From eq. N-1451(109),

$$\{f\} = \{p \cdot \Delta A\} = [M_H][\delta_{33}]\{\ddot{q}\} \qquad (127)$$

so that eq. (126) becomes,

$$([\mu] + [M_H][\delta_{33}])\{\ddot{q}\} + [C]\{\dot{q}\} + [K]\{q\} = \{f_0(t)\} \qquad (128)$$

Equation (128) is exactly the same as the equation of motion in-air with the forcing function f_o. The only difference is that added to the physical mass matrix $[\mu]$ is a hydrodynamic mass matrix M_H, which is effective only in the normal direction of the shells. Thus, once the hydrodynamic mass matrix is computed, the dynamic problem can be solved routinely, by finite element structural analysis computer programs or otherwise, as if there were no coupling between the fluid and the structure.

N-1470 HYDRODYNAMIC DAMPING

Just as the fluid mass adds to the effective mass of cylindrical shells, the viscosity of the entrapped fluid adds to the effective damping of the coupled fluid-shell system. This hydrodynamic damping, or added damping, due to coupling of a cylindrical structure and a viscous fluid, was studied by Chen (refs. [90] and [146]), Yeh and Chen (Ref. [157]), and Mulcahy (ref. [158]). It was shown that the equivalent hydrodynamic modal damping ratio is dependent on a dimensionless parameter,

$$S = \omega a^2 / \nu, \qquad (129)$$

where ν is the kinematic viscosity of the entrapped fluid. The general expression for the hydrodynamic damping ratio, like that for the hydrodynamic mass, is quite complicated. Some insight, however, can be gained by studying the special case of the beam mode ($n = 1$) vibration of an infinitely long cylinder of radius a inside a stationary cylinder of radius b. In this case, the equivalent hydrodynamic modal damping ratio is given by

$$\zeta_1 = -\left[0.5\rho\pi a^2 / \left(M_H + \mu\pi a^2\right)\right]\mathrm{Im}(Z) \qquad (130)$$

where Z is a fairly complicated function of modified Bessel functions and S. Figure N-1470-1, reproduced from ref. [146], gives theoretical values of $-\mathrm{Im}(Z)$ as a function of the radius ratio b/a, for different values of S, and shows that as $S \to \infty$, $-\mathrm{Im}(Z)$ becomes infinitesimal even for small ratios of b/a, indicating that hydrodynamic damping is negligible for large values of S. As an example, for the vibration of the core support structure of a typical nuclear reactor inside the reactor vessel, $S \approx 10^9$. Thus, the hydrodynamic damping in this case is negligible. This conclusion, however, may not be true for the vibration of a thin rod inside a fluid-filled jacket of slightly larger diameter.

Although the above example is for the beam mode vibration of an infinitely long cylinder, the same conclusion holds true qualitatively for the shell mode vibration of finite cylindrical shells. This theoretical deduction is also supported by experimental observations. Damping ratios of thermal shields measured during pre-operational tests of pressurized water reactors show little difference from their corresponding values measured in-air. Thus, hydrodynamic damping in large cylindrical structures can be ignored.

N-1500 FLUID TRANSIENT DYNAMICS

In the course of preparation.

N-1600 MISCELLANEOUS IMPULSIVE AND IMPACTIVE LOADS

In the course of preparation.

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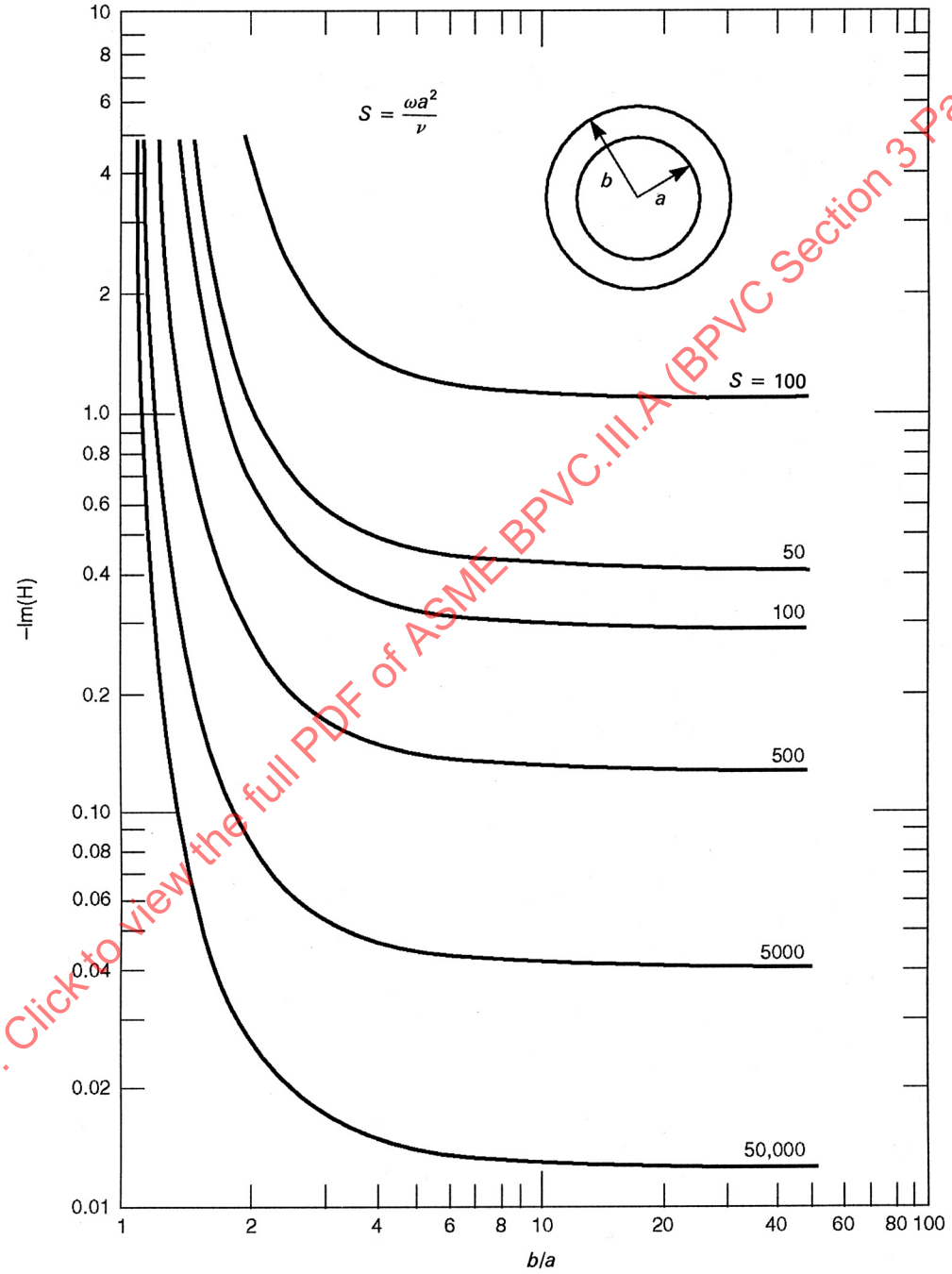
Figure N-1470-1
Imaginary Part of Z as a Function of b/a for Selected Value of S (Ref. [146])



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N-1700 COMBINED RESPONSES

N-1710 DYNAMIC RESPONSE COMBINATION

In the design of nuclear power components it is necessary to consider the combined responses caused by two or more different sources of dynamic loading. There are two situations which arise: one in which the time phase relationship between the two or more responses is known (deterministic); and a second, where the time phase relationship is said to be random. In the first situation, the response may be determined by the algebraic summation method of N-1720. In the second case where the time phase relationship is random, one of the three methods N-1721, N-1722, or N-1723 may be used depending upon satisfying the qualifying conditions contained in N-1721, N-1722, or N-1723. The owner, either directly or through his designee, may prescribe whether the time phase relationship of individual load responses is to be considered deterministic or random. Consideration may be given to the probability of occurrence of the associated events which cause the combined loading in arriving at detailed procedures for combining randomly phased responses (see discussion in N-1725).

N-1720 ALGEBRAIC SUMMATION

When the time phase relationship between two or more time history responses is known and the dynamic system is linear, the individual collinear responses may be algebraically added to determine the combined maximum and minimum structural response. If the dynamic system is nonlinear, then the combination loading should be input into the dynamic system analysis in order to determine the combination response.

N-1721 Peak Combined Response

In those cases where the dynamic system is linear and where two or more responses are to be combined, the peaks of the individual collinear responses can always be conservatively added to obtain the combined response irrespective of whether the phase relationship of the responses is undefined, deterministic, or random.

N-1722 SRSS (Square Root of Sum of Squares) Method

N-1722.1 For Responses With Nearly Equal Dominant Frequencies. In those cases where the dynamic system is linear and where (a) the time phasing of the peak individual collinear responses is random and (b) all of the individual responses have nearly an equal dominant[37] frequency with arbitrary amplitude, the SRSS method may be used to determine the combined design response (refs. [68], [69], and [70]).

N-1722.2 For Uncorrelated Dynamic Responses. The intent of this method for combination of transient dynamic responses is to achieve a conditional nonexceedance probability of at least 50% for the peak combined response of the system, component, or element considered. That is, there is a 50% probability that the magnitude of the summed responses will not exceed the SRSS value, given that the combined events have occurred. This goal is achieved by compliance with the following criteria provided that the amplitude of loads or accelerations for each input are conservatively represented (approximately at the level of the 84th percentile or the mean plus one standard deviation of the input amplitude).

Criterion. Dynamic or transient responses of linear structures, components, and equipment arising from combinations of dynamic loading or motions may be combined by SRSS provided that each of the dynamic inputs or responses has a *limited number of peaks* of force or acceleration and *approximately a zero mean*, and that the individual component inputs can be considered to be *relatively uncorrelated* (refs. [75] to [78]). Specific criteria for these three requirements are given below.

The requirement of a *limited number of peaks* is satisfied if any one of the following three criteria is met.

(a) For each response time history in the combination:

$$\frac{T_{75}}{\Delta T} \leq 0.02$$

and

$$\frac{T_{50}}{\Delta T} \leq 0.08$$

where T_{75} represents the summation of time intervals for all peaks over which the response exceeds 75% of either the maximum or minimum response (whichever is greater), T_{50} represents the time over which the response exceeds 50% of either the maximum or minimum response (whichever is greater), and ΔT represents the total time interval over which the maximum responses are expected to occur. For the case of random relative start times, ΔT represents the time interval during which the strong responses may overlap (generally taken as the time interval from the zero crossing time just preceding the first peak that exceeds 50% of the absolute maximum response to the zero crossing time just following the last peak that exceeds 50% of the absolute maximum response, of the longer time history). The strong motion of a seismic time history is discussed in N-1212.2(f). In determining the time T_{75} and T_{50}, only the durations of peaks with the same sign (positive or negative) are additive. These definitions are illustrated in Figure N-1722.2-1.

(b) For each input-time history in the combination:

$$\frac{T_{75}}{\Delta T} \leq 0.01$$

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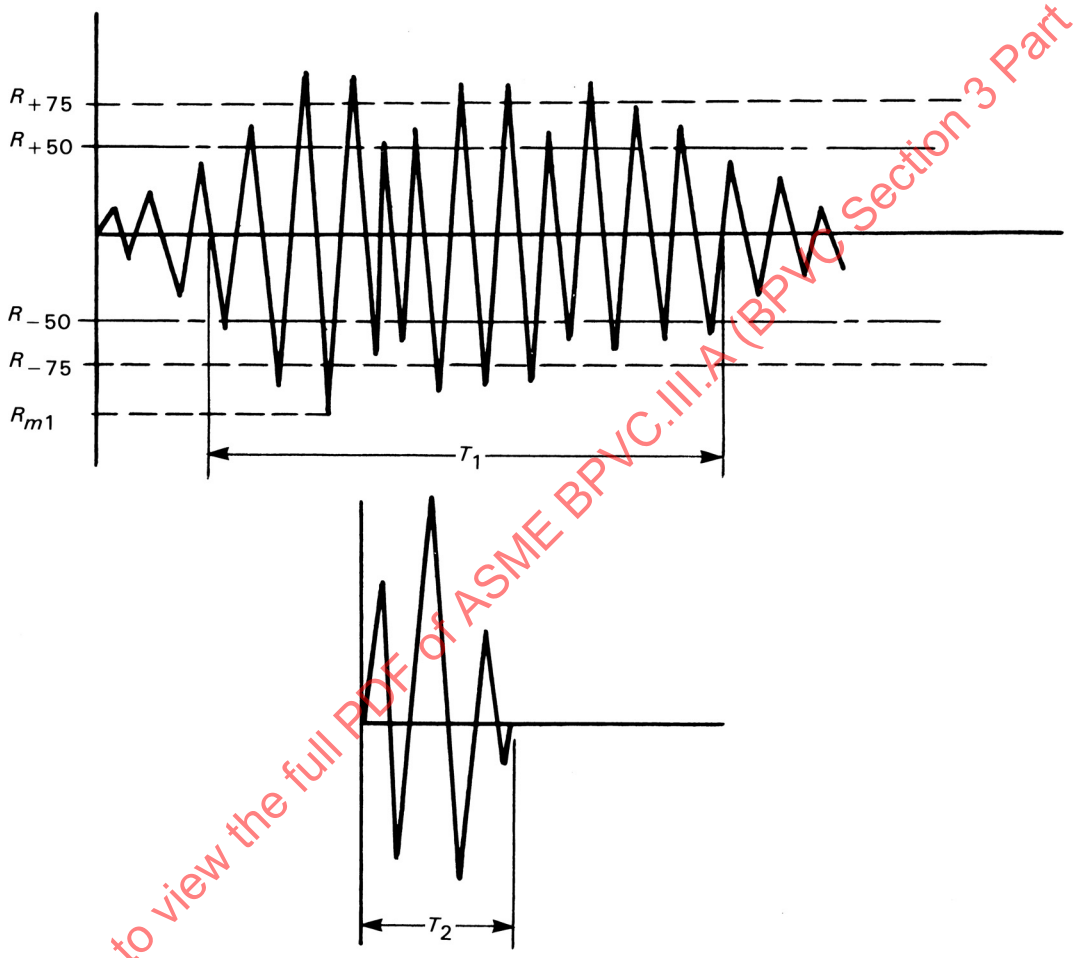
Figure N-1722.2-1
Definition of Notation



Legend:

R_{m1} = absolute maximum response
R_{+75} = 0.75 $|R_{m1}|$
R_{-75} = −0.75 $|R_{m1}|$
T_{+75} = summation of time intervals that response exceeds R_{+75}
T_{-75} = summation of time intervals that response exceeds R_{-75}
T_{75} = larger of T_{+75} or T_{-75}
ΔT = T_1, since $T_1 > T_2$

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and

$$\frac{T_{50}}{\Delta T} \leq 0.04$$

where the definitions are the same as above, except for the substitution of input for response.

(c) If criteria (a) or (b) above are not met by each of the responses or inputs, respectively, in the combination, this requirement is still satisfied if the effective time ratios $(T_{75}/\Delta T)_e$ and $(T_{50}/\Delta T)_e$ for the response combination meet the requirements of (a) above. The effective time ratios for the response combination are given by:

$$\left(\frac{T_{75}}{\Delta T}\right)_e = \sqrt{\frac{\sum_{i=1}^{N}\left(R_m T_{75}/\Delta T\right)_i^2}{\sum_{i=1}^{N}\left(R_m\right)_i^2}}$$

where i represents each individual response in the combination and $(R_m)_i$ represents the maximum amplitude of that response component. The effective time ratio $(T_{50}/\Delta T)_e$ is obtained by substituting 50 for 75 in the above equation.

The requirement of *approximately a zero mean* is satisfied if, for the duration of strong input (or response), the ratio of the mean to maximum input (or response) is less than 0.20. The mean value is:

$$\mu = \frac{1}{N}\sum_{j=1}^{n} R_j$$

where μ is the mean value, R_j is the response at time increment j, and N is the total number of equally spaced time increments.

If this requirement is not met, responses can be divided into two parts (nonzero mean part and peak response relative to mean response) provided the response function is not skewed from the axis of the mean μ. The nonzero mean part of response can be combined algebraically (accounting for the sign of the mean response) and the peak responses relative to the mean responses can be combined SRSS if these relative responses meet the provisions of this section.

The requirement of *relatively uncorrelated* inputs is met if either of the following is satisfied:

(1) the individual dynamic inputs or responses are from events such that their relative start times are uncertain;

(2) if the relative start times are known, then the coefficient of correlation must be less than 0.3.

N-1723 SRSSE (Equivalent SRSS) Method

(a) In those cases where individual linear system time history responses have randomly defined time phase relationships and condition (b) of N-1722 is not satisfied, the SRSSE method may be used to determine the combined design response. As an illustration, N-1723.1 and N-1723.2 describe a detailed numeric procedure for determining the combinations for the case where two individual time history responses have a uniform random time phase relationship.

(b) For the case when three or more linear time history responses are to be combined, the procedures of N-1724 may be used to determine the combined SRSSE design responses (ref. [68]).

N-1723.1 SRSSE Method for Two Time History Responses. A method may be constructed to obtain the SRSSE response combinations as given in (a), (b), and (c).

(a) Determine one time history as a reference function. Bring the origin of the second function coincident with the origin of the reference function. Care should be taken to determine the reference function so that the time shift of the second function is consistent with the actual sequence of events expected. The origin of the second function could occur at, before, or after the origin of the reference function. The case where the origins are coincident will be discussed in (b) and (c). The procedure for the other two cases are similar.

(b) Shift the second function in small time steps relative to the reference function until the origin of the second function goes to the end of the reference function (see Figure N-1723.1-1). For each time phase difference τ, a scan is made from the time zero to $\tau + T_2$, if $(\tau + T_2) \geq T_1$, or from the time zero to $(\tau + T_2) < T_1$, to determine the largest amplitude (peak) of the sum of the two functions. T_1 and T_2 are the time duration of the reference function and second function, respectively. The algebraic maximum and minimum of these peaks, Maximum Combined Amplitude (FF), are plotted as functions of the time phase difference τ. The resulting function is called the Amplitude Phase Function (APF) which is shown in Figure N-1723.1-3.

(c) The functional relationship between the APF and the Probability Density Function (PDF) for the time shift (Figure N-1723.1-2) is then used to compute the Cumulative Distribution Functions (CDF) of the maximum and minimum combined amplitudes as detailed in N-1723.1.1 (Figure N-1723.1-4). The choice of the appropriate PDF for τ should be developed on a case-by-case basis. The uniform PDF is frequently a reasonable assumption when there is a weakly known cause and effect between the initiation of the causative events.

N-1723.1.1 Cumulative Distribution Function (CDF). Both the maximum and minimum Cumulative Distribution Functions need to be constructed.

(a) From Figure N-1723.1-3, which illustrates the maximum APF function, it can be seen that any possible combined amplitude FF falls between two extreme limits FF_{max} and FF_{min}. It is clear that the probability that the value of FF is less than FF_{max} is one, and the probability that the value of FF is less than FF_{min} is zero. Hence, in Figure N-1723.1-4, we can immediately plot these two extreme points.

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Figure N-1723.1-1

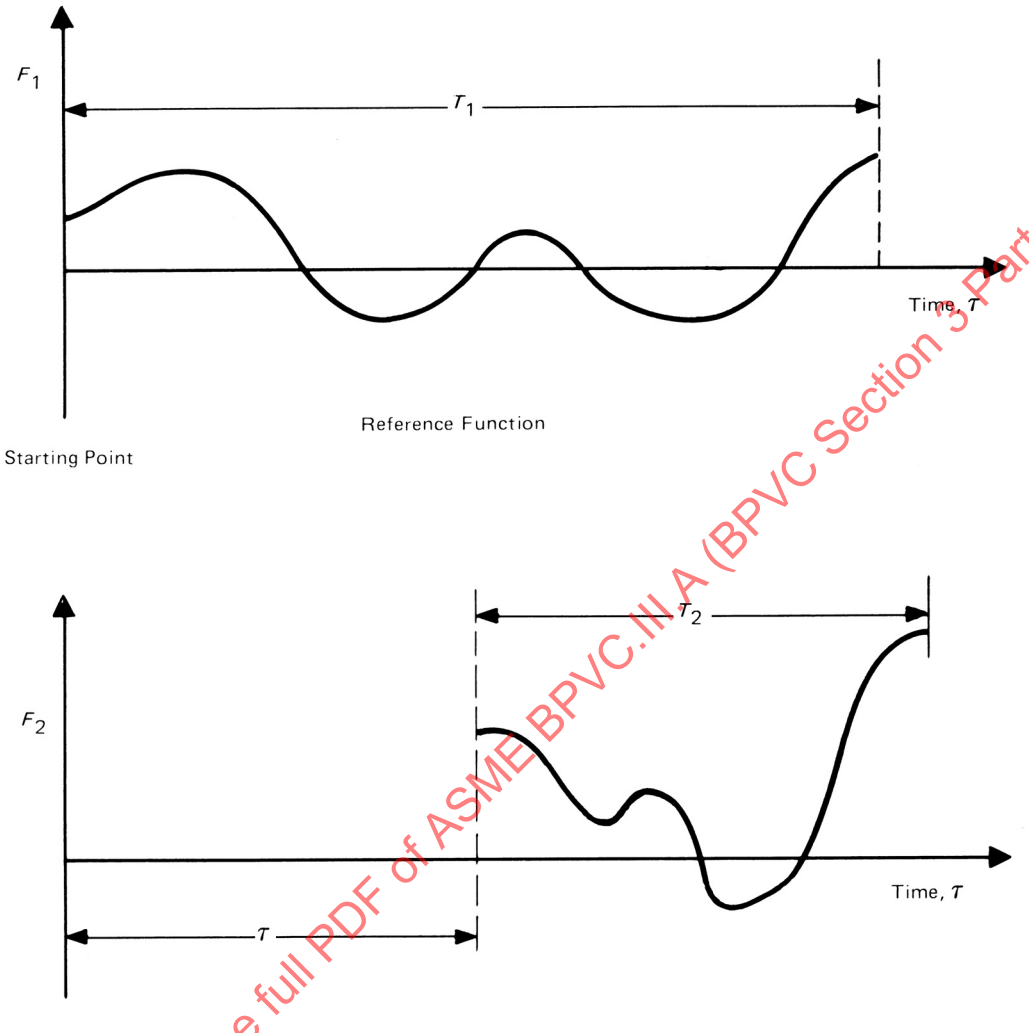

Figure N-1723.1-2

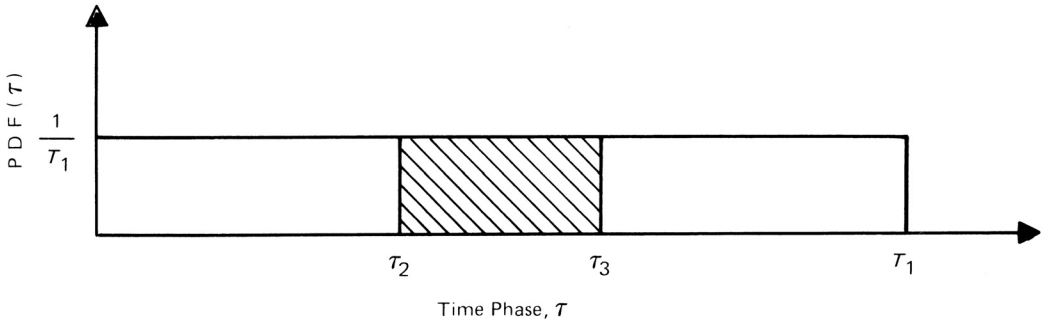

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Figure N-1723.1-3

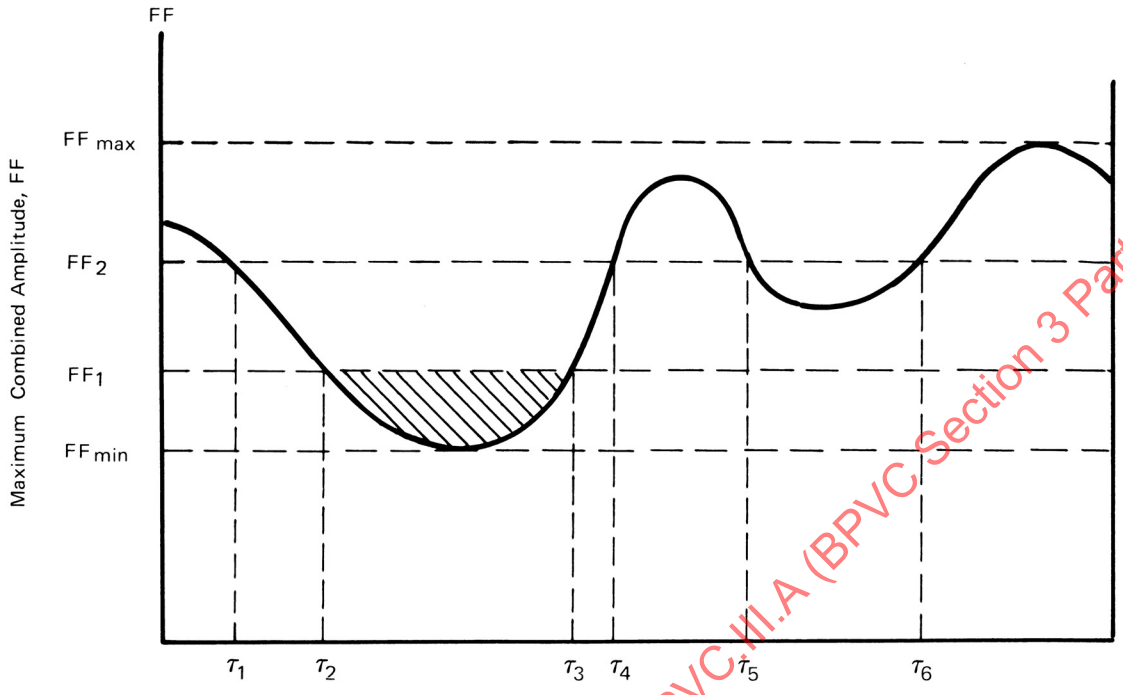
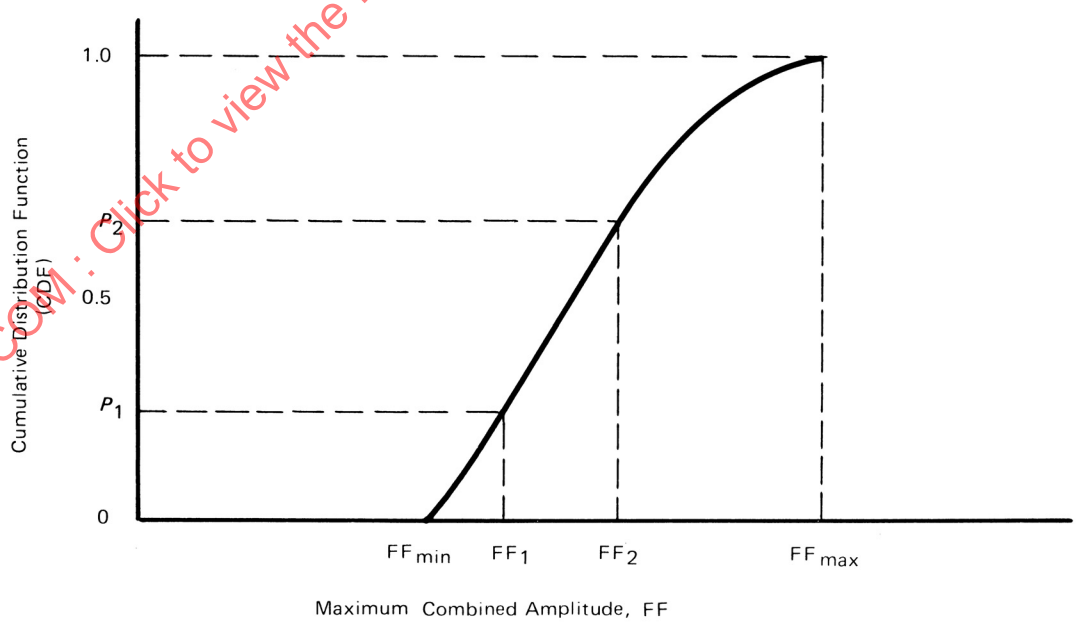


Figure N-1723.1-4



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(b) The intermediate points on the CDF are found by integration of the distribution of the time shift τ between appropriate limits. Hence, at this point, the choice of the distribution for τ enters the calculations.

(c) Calculation of the intermediate points on the CDF requires the consideration of the functional relationship between the maximum combined amplitude FF and the time phase shift τ. For a given value of FF on Figure N-1723.1-3 (in this case FF_1) there is an interval on the τ axis (τ_2, τ_3) in Figure N-1723.1-3. This same τ interval (τ_2, τ_3) can also be observed on the Probability Density Function for τ in Figure N-1723.1-2. The shaded area of Figure N-1723.1-2 represents the probability that τ falls in the interval (τ_2, τ_3). The event of τ falling in the interval (τ_2, τ_3) corresponds exactly to the event that FF is less than or equal to FF. Therefore, the ordinate P of Figure N-1723.1-4 corresponding to FF can be computed as

$$P_1 = \int_{\tau_2}^{\tau_3} \mathrm{PDF}(\tau) d \tag{131}$$

and for the case where

$$\mathrm{PDF}(\tau) = 1 / T_1 \tag{132}$$

then

$$P_1 = (\tau_3 - \tau_2) / T_1 \tag{133}$$

(d) The same procedure is followed for other levels of FF. Note that care must be exercised when integrating multiple valued functions to avoid a loss of information. If the level of FF selected contains more than one interval of τ, all intervals must be integrated to obtain the associated probability. For example, in the case of FF, the integration limits in eq. (c)(131) would be from τ_1 to τ_4 and from τ_5 to τ_6. The result of the integration is designated P_2 and plotted in Figure N-1723.1-4.

(e) Selecting different levels of FF and repeating the above procedure, the CDF function as shown in Figure N-1723.1-4 can be constructed.

(f) This entire procedure is then repeated for the minimum APF function to find its corresponding CDF.

N-1723.1.2 Cumulative Distribution Function for the Absolute Maximum. An alternative and more conservative procedure than that given in N-1723.1.1 may also be used to generate the CDF of combined responses. In this alternative procedure, instead of summing the positive and negative peaks of the response separately as in N-1723.1.1, the absolute value of the peak combined response is identified for each value of τ when generating the APF. In this alternative procedure, a single APF is generated which corresponds to a combination of both positive or negative combined responses, whichever is greater in absolute magnitude for each value of τ. The procedure to construct the absolute maximum CDF is the same as stated in N-1723.1.1, except that only one CDF is generated for each response combination instead of two.

N-1723.2 SRSSE Values. From the maximum and minimum CDF determined in N-1723.1.1, select the values for SRSSE maximum and SRSSE minimum combinations corresponding to the 50% probability levels. It should be noted that these 50% levels are conditional probability levels given that the causative events have occurred. In order to determine total probability levels associated with the SRSSE values, it is necessary to multiply by the corresponding simultaneous combined event occurrence as illustrated in ref. [70].

N-1724 SRSSE Method for Three or More Functions

A Monte Carlo simulation procedure (refs. [69] and [70]) will generally be required to generate the CDF if there are more than two time history response functions to be combined. The SRSSE maximum, minimum, or absolute maximum values are at the same 50% probability levels used in N-1723.2.

N-1725 Alternative CDF Values

The SRSSE values of N-1723.2 and N-1724 are at a conditional nonexceedance probability level of 50%. That is, there is a 50% probability that the magnitude of the summed responses will not exceed the SRSSE values, given that the combined events have occurred. When the peak combined response values of N-1721 are used, the 100% nonexceedance values result. Since the use of either of these levels (50%, 100%) must be specified through consideration of systems requirements and service levels, alternative nonexceedance probability levels between 50% and 100% may also be used.

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NONMANDATORY APPENDIX O

ARTICLE O-1000
RULES FOR DESIGN OF SAFETY VALVE INSTALLATIONS

O-1100 SCOPE AND DEFINITIONS

O-1110 SCOPE

(a) The scope of Nonmandatory Appendix O is confined to the design of the safety valve installations as defined in O-1120. The loads acting at the safety valve station will affect the bending moments and stresses in the complete piping system, out to its anchors and/or extremities, and it is the designer's responsibility to consider these loads. This Appendix, however, deals primarily with the safety valve installation and not the complete piping system.

(b) The design of the safety valve installation requires that careful attention be paid to all loads acting on the system, the forces and bending moments in the piping and piping components resulting from the loads, the loading and stress criteria, and general design practices. All components in the safety valve installation must be given consideration, including the complete piping system, the connection to the main header, the safety valve, valve and pipe flanges, the downstream discharge or vent piping, and the system supports. The scope of this Appendix is intended to cover all loads on all components.

(c) This Appendix has application to either safety, relief, or safety relief valve installations. For convenience, however, the overpressure protection device is generally referred to as a safety valve. The loads associated with relief or safety relief valve operation may differ significantly from those of safety valve operation, but otherwise the rules contained herein are equally applicable to each type of valve installation.

(d) Pressure relief safety valve stations require detailed analysis. In performing its design function the station is subject to dynamic loading from the structural response to both thermal and hydraulic forces and, in some instances, significant impact loading from the valve mechanism. The resultant loading is a mechanical load.

(e) This Appendix provides guidance for design and analysis of the piping components of a safety valve station. The guidance is presented by discussing four areas of consideration

(1) load computation — open discharge system
(2) stress evaluation — open discharge system
(3) closed discharge system
(4) general design considerations

O-1120 DEFINITIONS

(a) Safety Valve. An automatic pressure-relieving device actuated by the static pressure upstream of the valve and characterized by full opening pop action. It is used for gas or vapor service.

(b) Relief Valve. An automatic pressure-relieving device actuated by the static pressure upstream of the valve which opens further with the increase in pressure over the opening pressure. It is used primarily for liquid service.

(c) Safety Relief Valve. An automatic pressure actuated relieving device suitable for use either as a safety valve or relief valve, depending on application.

(d) Power-Actuated Pressure-Relieving Valve. A relieving device whose movements to open or close are fully controlled by a source of power (electricity, air, steam, or hydraulic). The valve may discharge to atmosphere or to a container at lower pressure. The conditions, and such effects, shall be taken into account. If the power-actuated pressure-relieving valves are also positioned in response to other control signals, the control impulse to prevent overpressure shall be responsive only to pressure and shall override any other control function.

(e) Open Discharge Installation. An installation where the fluid is discharged directly to atmosphere or to a vent pipe that is uncoupled from the safety valve. Figure O-1120(e)-1 shows a typical open discharge installation with an elbow installed at the valve discharge to control direction of flow. Figure O-1120(e)-2 shows a typical open discharge system discharging into a vent pipe. The values for l and m in Figure O-1120(e)-2 are upper limits for which the rules for open discharge systems may be used.

(f) Closed Discharge Installation. An installation where the effluent is carried to a distant spot by a discharge pipe which is connected directly to the safety valve.

(g) Safety Valve Installation. The safety valve installation is defined as that portion of the system shown on Figures O-1120(e)-1 and O-1120(e)-2. It includes the run pipe, the branch connection, the inlet pipe, the valve,

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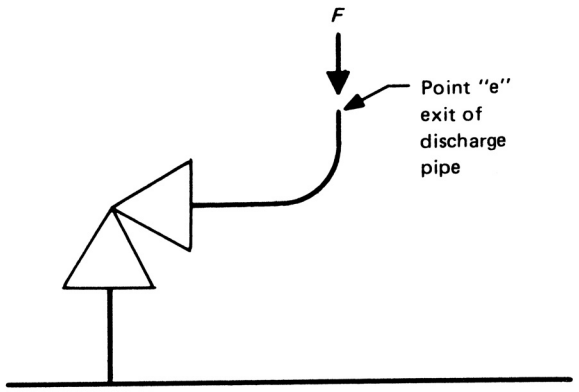
Figure O-1120(e)-1
Application Point of Venting Force *F*



the discharge piping, and the vent pipe. Also included are the components used to support the system for all static and dynamic loads.

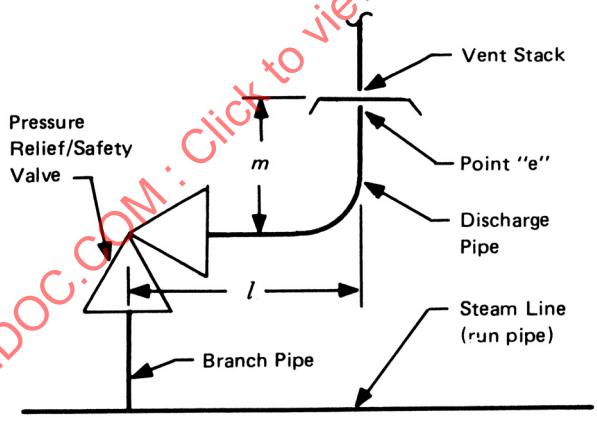
O-1200 METHOD OF AND PROCEDURE FOR LOAD COMPUTATION

O-1210 BASIC CONSIDERATIONS

(a) The load computation includes two thermodynamic computations: evaluation of momentum effects and evaluation of pressure effects. The response computation of the piping system includes consideration of transient, dynamic loading effects of the sudden opening and closing valve action. The loads computation is combined with other specified loads and translated to a stress computation which is then compared to the Code allowable stress for acceptability.

(b) The basic principles of analysis of a pressure relief valve station are generally applicable to both open and closed discharge systems. The application of these basic principles may be quite different.

Figure O-1120(e)-2
Limiting Safety Valve Arrangements and Dimensions



$l \leq 4.0$ X discharge pipe diameter

$m \leq 6.0$ X discharge pipe diameter

O-1220 OPEN SYSTEM — DISCHARGE THRUST

(a) The limiting dimensions for safety valve arrangements are shown by Figure O-1120(e)-2. The determination of the reaction force F value(s) is the responsibility of the piping system designer.

(b) The steady-state load due to steam reaction force from the opening and subsequent venting of the safety valve shall include consideration of both momentum and pressure effects and may be computed by the formula

$$F = (W/g)V_e + (P_e)A$$

where

A = exit flow area at point e
F = reaction force
g = gravitational constant, 32.2 lbm-ft/lbf-sec^2 (9.81 m/s)
P_e = static gage pressure at point e
V_e = exit velocity at point e
W = mass flow rate (relieving capacity stamped on the valve × 1.11 — adjust for units to be compatible, if necessary)

The reaction force F is a design mechanical load that requires structural equilibrium for system stability and is applied as shown in Figure O-1120(e)-1.

(c) To ensure consideration of the effects of the suddenly applied load F, a dynamic load factor DLF, based on the relief/safety valve opening time and system dynamic characteristics, shall be applied to the forces and moments due to the reaction force F.

(d) Instead of a simplified dynamic analysis with the application of the DLF a dynamic hydraulic/structural system analysis may be performed.

O-1230 OTHER MECHANICAL LOADS

Other mechanical loads to be considered should include, as a minimum

(a) interaction loads on the run pipe when more than one valve releases, and

(b) the transient impacting of the valve mechanism opening and closing, if applicable

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O-1300 STRESS EVALUATION OPEN SYSTEM

Evaluation of stresses due to the design mechanical loads may be made by using the rules for Class 1 piping, O-1310 and for Class 2 or Class 3 piping, O-1320.

(15)

O-1310 CLASS 1 PIPING

(a) Whenever any of the equations of NB-3650 are used in the analysis of Class 1 piping systems, the value of M_i shall include the reaction force moment. The contribution from the reaction force F to the branch moment M_b, as defined in NB-3683.1, shall be no less than the product F × nominal discharge pipe size × DLF.

(b) Note that the use of the equations of NB-3650 for branch connections requires a nozzle spacing as defined by NB-3683.8(a)(2).

(c) When NB-3650 eq. (9) is used in the analysis, the value of M_i shall be defined as:

M_i = resultant moment due to a combination of primary loads. Loads to be considered include: weight; earthquake, considering only one-half the range of the earthquake and excluding the effects of anchor displacement due to earthquake; thrusts from relief and safety valve loads from pressure and flow transients; and other sustained mechanical loads.

(d) The combination of loads shall be specified in the Design Specification. In the combination of loads, all directional moment components in the same direction shall be combined before determining the resultant moment; i.e., resultant moments of loads shall not be combined.

(e) When NB-3650 eq. (10) is used in the analysis, the value of M_i shall be defined as:

M_i = resultant range of moment due to a combination of primary plus secondary loads. Loads to be considered include: thermal expansion; anchor movement from any cause; earthquake effects; thrusts from relief and safety valve loads from pressure and flow transients; and other mechanical loads.

(f) The combination of loads shall be specified in the Design Specification. The earthquake loading shall be considered in conjunction with the operating conditions. Weight effects need not be considered in the range loading because they are noncyclic in character. In the combination of loads, all directional moment components in the same direction shall be combined before determining the resultant moment; i.e., resultant moments of loads shall not be combined. If a combination includes earthquake effects, M_i shall be the greater of the resultant range of moment due to the combination of all loads considering one-half the range of the earthquake or the resultant range of moment due to the full range of the earthquake.

O-1320 CLASS 2 OR CLASS 3 PIPING (15)

(a) For Class 2 or Class 3 piping, NC-3653.1(a) eq. (9a) or ND-3653.1(a) eq. (9a) is to be used with M_b to include the reaction force moment. The contribution from the reaction force F to the branch moment, as defined in NC-3653.3 or ND-3653.3, shall be no less than the product, F × nominal discharge pipe size × DLF.

(b) Note that the use of NC-3653.1(a) eqs. (9a) and (9b) or ND-3653.1(a) eqs. (9a) and (9b) for branch connections requires a nozzle spacing, as defined by NC-3643.3 (c)(6) or ND-3643.3(c)(6).

O-1400 CLOSED DISCHARGE SYSTEMS — OPEN DISCHARGE SYSTEMS WITH LONG DISCHARGE PIPES — SYSTEMS WITH SLUG FLOW

(a) For closed discharge systems, open discharge systems with long discharge pipes, and systems with slug flow, the state of the art does not lend itself to a well defined method of load computation. For these systems the dynamic interaction forces of the total system including the attached discharge piping must be considered.

(b) When a safety valve discharge is connected to a relatively long run of pipe and is suddenly opened, there is a period of transient flow until the steady-state discharge condition is reached. During this transient period, the pressure and flow will not be uniform. When the safety valve is initially opened, the discharge pipe may be filled with air. If the safety valve is on a steam system, the steam discharge from the valve must purge the air from the pipe before steady state steam flow is established and, as the pressure builds up at the valve outlet flange and waves start to travel down the discharge pipe, the pressure wave initially emanating from the valve will steepen as it propagates, and it may steepen into a shock wave before it reaches the exit.

(c) Relief valves discharging into an enclosed piping system create momentary unbalanced forces which act on the piping system during the first few milliseconds following relief valve lift. The pressure waves traveling through the piping system following the rapid opening of the safety valve will cause bending moments in the safety valve discharge piping and throughout the remainder of the piping system. In such a case, the designer must compute the magnitude of the loads and perform appropriate evaluation of their effects.

(d) Particular attention should be given to the large forcing functions acting on the pipe if it contains water seals, two phase flow, or if there is a water column in the discharge piping.

(e) The reaction force effects are dynamic in nature. A time history dynamic solution, incorporating a multi-degree-of-freedom model solved for the transient hydraulic forces is considered to be a preferred method of analysis.

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(15) O-1500 DESIGN CONSIDERATIONS

Reference should be made to NB/NC/ND-7000.

It is recommended that the following be included as part of the total design consideration.

(a) Where not required by the Code, it is recommended that the header penetrations for relief valves be in accordance with the nozzle spacing recommendation of NB-3683.8(a)(2).

(b) No more than one penetration should be made around the circumference of the run pipe (i.e., no two penetrations in the same transverse plane), the spacing to be in accordance with the preceding (a).

(c) The stress analysis of the pipe could require additional thickness for membrane protection above that required by the thickness equation for pressure load only.

(d) Detail design should preclude sharp notches that may be generated by the use of saddles, gussets, ribs, etc.

(e) Contoured outlets are often advantageous.

(f) The direction of discharge of several pressure relief valves on the same run pipe should be such as to tend to balance one another for all modes of operation specified in the piping design specification.

(g) Supports may require a detailed analysis to determine their role in restraint as well as support. Considerations should be given to the possibility that, under load, snubbing devices may permit significant deflections.

(h) The reaction force moment arm on the outlet piping should be minimized in accordance with the valve manufacturer's recommendation.

(i) The relief valve outlet piping stack clearance should be checked for interference from thermal expansion, earthquake displacements, etc. The vent stack and valve discharge piping system should be arranged such that pull out of the valve discharge pipe does not occur.

(j) Thermal expansion effects are to be considered as they presently are defined in the Code.

(k) The force due to venting should be included in the evaluation of the stack forces. The effects of back pressure in the discharge stack can be significant.

(l) The station should be arranged such that the discharge piping is void of collected water. The discharge piping from each valve or device should be at least of the same size as the valve outlet.

(m) Drains shall be provided so that condensed leakage, rain, or other water sources will not collect on the discharge side of the valve and adversely affect the reaction force. Safety valves are generally provided with drain plugs that can be used for a drain connection. Discharge piping shall be sloped and provided with adequate drains if low points are unavoidable in the layout.

(n) Where water seals are used ahead of the safety valve, the total water volume in the seals should be minimized. To minimize forces due to slug flow or water seal excursion, the number of changes of direction and the lengths of straight runs of piping should be limited.

(o) Often safety valves are full lift, pop-type valves and are essentially full flow devices with no capability for flow modulation. In actual pressure transients, the steam flow required to prevent overpressure is a varying quantity, from zero to the full rated capacity of the safety valves. As a result, the valves may be required to open and close a number of times during the transient. Since each opening and closing produces a reaction force, consideration should be given to the effects of multiple valve operations on the piping system, including supports.

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NONMANDATORY APPENDIX P

ARTICLE P-1000
CERTIFIED MATERIAL TEST REPORTS

P-1100 INTRODUCTION

The requirements for a Certified Material Test Report (CMTR) are stated in NCA-3860. However, the material requirements vary with the class of construction, the product form of material, the requirements of the material specification, and the manufacturer's procedures. Since changes in any of these requirements may be made by Addenda or new editions of the Code, it is important for the purchaser and the supplier to know the requirements of the applicable edition and Addenda, so that the reported results can be compared with the requirements to determine whether or not the material is in compliance with the Code.

P-1200 GENERAL REQUIRED INFORMATION

The items of information given in (a) through (j) below are required in the CMTRs for metallic material (as defined by NCA-1220 and NB/NC/ND/NE/NF/NG-2100) used in Section III, Division 1, construction.

(a) Name of certifying organization [P-1200(j)].

(b) Number and expiration date of the organization's Certificate of Authorization or Quality System Certificate (Materials). Alternatively, if the organization was qualified by a party other than the Society, the revision and date of the written program under which the material is being certified.

(c) Purchaser's order or contract number.

(d) Description of the material, including specification number, grade, class, type, and nominal size, as applicable. For pipe made to specifications which include both seamless and welded pipe, the report shall designate which type it is.

(e) Description of material identification marking.

(f) Actual results of chemical analyses, tests, and examinations required by the Material Specification and this Section.

(g) Reports of weld repairs performed, if any, as required by NB/NC/ND/NE/NF/NG-2500, including radiographic films, when radiography is required.

(h) Charpy V-notch and drop-weight test results required by NB/NC/ND/NE/NF/NG-2320, when this testing is required by NB/NC/ND/NE/NF/NG-2311. When Charpy V-notch impact tests are required, the report shall include the test temperature; the absorbed energy when required; the lateral expansion; and the location and orientation of the specimens used. When drop-weight tests are required, the report shall include the test temperature; the type, location, and orientation of the specimens used; and the results of the tests (break or no break).

(i) Nondestructive examinations performed and accepted as required by NB/NC/ND/NE/NF/NG-2500.

P-1300 INFORMATION REQUIRED UNDER SPECIFIC CIRCUMSTANCES

The information given in (a) through (f) is required under specific circumstances.

(a) Heat treatment data, as follows:

(1) temperatures (or temperature ranges) and times at temperature used when the material specification requires specific temperatures and times

(2) heat treatment conditions when no specific temperatures (or temperature ranges) or times are required by the material specification

(3) the minimum solution annealing temperature used for austenitic stainless steels and high nickel alloys

(4) recorded temperature ranges and actual times at temperature, and heating and cooling rates, for postweld heat treatment of materials that are repaired by welding, when postweld heat treatment is required by NB/NC/ND/NE/NF/NG-2500

(5) recorded temperatures and actual times at temperature for test coupons when required by NB/NC/ND/NE/NF/NG-2210, when those paragraphs are applicable

(b) Hydrostatic test pressure, when a hydrostatic test is required by the material specification, or notation that the hydrostatic test has not been performed, if it is deferred.

(c) Ferrite Number for all A-No. 8 welding material except type 16-8-2, as required by NB/NC/ND/NE/NF/NG-2433.

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(d) The grain size, reported in accordance with ASTM E112, when a fine grain size is specified for metallic materials.

(e) For welding materials, in addition to applicable paragraphs above, the process, the preheat and interpass temperature, the type of chemical analysis (filler metal/undiluted deposit), the shielding gas composition, and the Ferrite Number shall be reported as required by NB/NC/ND/NE/NF/NG-2400 and as applicable.

(f) A list of chemical analyses, tests, examinations, and heat treatments required by the Material Specification that were not performed.

P-1400 EXECUTION

(a) All requirements of the material specification and this Section need not be performed by the same organization. In that case the certifying organization shall ensure that each organization, providing material or services, is identified on the CMTR along with the activities for which it is responsible. Alternatively, the CMTR's furnished by the other organizations may be referenced on and attached to the CMTR of the organization which supplies the material.

(b) The CMTR shall include a dated statement affirming that the contents of the report are correct and accurate.

(c) Signing or notarization of the CMTR is not required.

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NONMANDATORY APPENDIX Q

ARTICLE Q-1000
DESIGN RULES FOR CLAMP CONNECTIONS

Q-1100 INTRODUCTION

Q-1110 SCOPE

(a) The rules in Nonmandatory Appendix Q apply specifically to the design of clamp connections for pressure vessels and vessel parts and may be used in conjunction with the applicable requirements in Subsections NC, ND, and NE. These rules are not to be used for the determination of the thickness of supported or unsupported tubesheets integral with a hub nor for the thickness of covers. These rules provide only for hydrostatic end loads and gasket seating and do not consider external loads or thermal effects.

(b) The design of a clamp connection involves the selection of the gasket, bolting, hub, and clamp geometry (see Figure Q-1130-1). Bolting shall be selected to satisfy the requirements of Q-1140. Connection dimensions shall be such that the stresses in the clamp and hub, calculated in accordance with this Appendix, do not exceed the allowable stresses specified in Q-1180. All calculations shall be made on dimensions in the corroded condition. Calculations for both assembly and operating conditions are required.

(c) It is recommended that either a pressure energized or a low seating load gasket, or both, be used to compensate for possible nonuniformity in the gasket seating force distribution. Hub faces shall be designed so as to have metal-to-metal contact outside the gasket seal diameter. This may be provided by recessing the hub faces or by use of a metal spacer (see Figure Q-1130-1). The contact area shall be of sufficient cross-sectional area to prevent yielding of either the hub face or spacer under both operating and assembly axial loads.

(d) It is recognized that there are clamp designs which utilize no wedging action during assembly since clamping surfaces are parallel to the hub faces. These designs should satisfy the bolting and corresponding clamp and hub requirements of a clamp connection design with a total included clamping angle of 10 deg. This will provide some safety against loads imposed by angular deflections of the connection faces during operation and also provide some compensation against mechanical and thermal ratcheting.

(e) The rules of this Appendix should not be construed to prohibit the use of other types of clamp connections provided that they are designed in accordance with good engineering practice and the method of design is acceptable to the Inspector. These rules shall apply only to new construction.

(f) Clamps designed to the rules of this Section shall be provided with a bolt retainer. The retainer shall be designed to hold the clamps together independently in case of failure of the primary bolting. An appropriate external yoke or multiple bolting is considered satisfactory for this requirement. Clamp–hub friction shall not be considered as a retainer method.

Q-1120 MATERIALS

(a) Materials used in the construction of clamp connections shall comply with the requirements given in Article NC-2000, Article ND-2000, or Article NE-2000, as applicable.

(b) Hubs and clamps shall not be machined from plate.

Q-1130 NOTATION (15)

The symbols defined below are used in the equations for the design of clamp-type connections (see also Figures Q-1130-1 and Q-1130-2).

A = outside diameter of hub

A_1 = partial clamp area
$= (C_w - 2C_t)(C_t)$

A_2 = partial clamp area
$= 1.571{C_t}^2$

A_3 = partial clamp area
$= (C_w - C_g)l_c$

A_b = total cross-sectional area of bolts per clamp lug using the root diameter of the thread or least diameter of unthreaded portion, whichever is less. Cross-sectional area of bolt retainer shall not be included in calculation of this area.

A_c = total effective clamp cross-sectional area
$= A_1 + A_2 + A_3$

A_m = total required cross-sectional area of bolts per clamp lug taken as the greater of A_{m1}, A_{m2}, or A_{m3}

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A_{m1} = total cross-sectional area of bolts per clamp lug at root of thread or section of least diameter under stress, required for the operating conditions
= $W_{m1}/2S_b$

A_{m2} = total cross-sectional area of bolts per clamp lug at root of thread or section of least diameter under stress, required for gasket seating
= $W_{m2}/2S_a$

A_{m3} = total cross-sectional area of bolts per clamp lug at root of thread or section of least diameter under stress, required for assembly conditions
= $W_{m3}/2S_a$

B = inside diameter of hub

b = effective gasket or joint-contact surface seating width (see Table XI-3221.1-2)

B_c = radial distance from connection centerline to effective center of bolts

b_o = basic gasket or joint-contact surface seating width (see Table XI-3221.1-2)

C = diameter of effective clamp–hub reaction load
= $(A + C_i)/2$

C_g = effective clamp gap taken at clamp–hub contact center (C)

C_i = inside diameter of clamp

C_t = effective clamp thickness

C_w = effective clamp width

e_b = radial distance from effective center of the bolts to the centroid of the clamp body
= $B_c - (C_i/2) - l_c - X$

f = hub stress correction factor from Figure XI-3240-6. (This is the ratio of the stress in the small end of the hub to the stress in the large end.) (For values below limit of the Figure, use f = 1.0.)

G = diameter at location of gasket load reaction. Except as noted in Figure Q-1130-1, G is defined as follows (see Table XI-3221.1-2):
(a) when $b_0 \le 1/4$ in. (6 mm), G = mean diameter of gasket or joint contact face
(b) when $b_0 > 1/4$ in. (6 mm), G = outside diameter of gasket contact face less $2b$

g_0 = thickness of hub neck at small end

g_1 = thickness of hub neck at intersection with hub shoulder

g_2 = height of hub shoulder (g_2 shall not be larger than T)

$\bar{g}$ = radial distance from the hub inside diameter to the hub shoulder ring centroid

$$= \frac{Tg_1^2 + h_2 g_2\left(2g_1 + g_2\right)}{2\left(Tg_1 + h_2 g_2\right)}$$

H = total hydrostatic end force
= $0.785\,G^2P$

h = hub taper length

$\bar{h}$ = axial distance from the hub face to the hub shoulder ring centroid

$$= \frac{T^2 g_1 + h_2^2 g_2}{2\left(Tg_1 + h_2 g_2\right)}$$

h_2 = average thickness of hub shoulder
= $T - (g_2 \tan\phi)/2$

H_D = hydrostatic end force on bore area
= $0.785\,B^2P$

h_D = radial distance from effective clamp–hub reaction load to the circle on which H_D acts
= $[C - (B + g_1)]/2$

H_G = difference between total effective axial clamping preload and the sum of total hydrostatic end force and total joint contact surface compression
= $[1.571W/\tan(\phi + \mu)] - (H + H_p)$

h_G = radial distance from effective clamp–hub reaction load to the circle on which H_G acts (for full face contact geometries h_G = 0)

H_m = total axial gasket seating requirements for make-up ($3.14bGy$ or the axial seating load for self-energizing gaskets, if significant)

h_n = hub neck length [minimum length of h_n is $0.5g_1$ or 1/4 in. (6 mm), whichever is larger]

h_o = $\sqrt{Bg_o}$

H_p = total joint-contact surface compression load
= $2b \times 3.14GmP$ (for self-energized gaskets, use H_p = 0 or actual retaining load if significant)

H_T = difference between total hydrostatic end force and hydrostatic end force on bore area
= $H - H_D$

h_T = radial distance from effective clamp–hub reaction load to the circle on which H_T acts
= $[C - (B + G)/2]/2$

I_c = effective moment of inertia of clamp relative to axis of entire section

$$= \left(\frac{A_1}{3} + \frac{A_2}{4}\right)C_t^2 + \frac{A_3 l_c^2}{3} - A_c X^2$$

I_h = effective moment of inertia of hub shoulder ring relative to its neutral axis

$$= \left(T^2 g_1 + h_2^2 g_2\right)\frac{\bar{h}}{6} + \frac{Th_2 g_1 g_2\left(T - h_2\right)^2}{3\left(Tg_1 + h_2 g_2\right)}$$

L_a = distance from W to the point where the clamp lug joins the clamp body

l_c = effective clamp lip length

L_h = clamp lug height

l_m = effective clamp lip moment arm
= $l_c - (C - C_i)/2$

L_w = clamp lug width

m = gasket factor from Table XI-3221.1-1

M_D = component of moment due to H_D
= $H_D h_D$

M_F = offset moment
= $H_D(g_1 - g_o)/2$

M_G = component of moment due to H_G
= $H_G h_G$

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M_H = reaction moment at hub neck

$= M_O / \left\{ 1 + \frac{1.818}{\sqrt{Bg_1}} \left[T - \bar{h} + \frac{3.305 I_h}{g_1^2 (B/2 + \bar{g})} \right] \right\}$

M_O = total rotational moment on hub (see Q-1150)

M_P = pressure moment

$= 3.14 \times PBT(T/2 - \bar{h})$

M_R = radial clamp equilibrating moment

$= 1.571\, W \left\{ \bar{h} - T + \left[(C - N) \tan\phi \right] / 2 \right\}$

M_T = component of moment due to H_T

$= H_T\, h_T$

N = outside diameter of hub neck

P = Design Pressure

Q = reaction shear force at hub neck

$= 1.818 \frac{M_H}{\sqrt{Bg_1}}$

r = clamp body radius (shall be less than or equal to C_t)

S_1 = hub longitudinal stress on outside at hub neck

S_2 = maximum Lame hoop stress at bore for hub neck section

S_3 = hub axial shear stress (maximum) across the hub shoulder

S_4 = hub radial shear stress (maximum) across the hub neck

S_5 = clamp longitudinal stress at clamp body inner diameter

S_6 = clamp tangential stress at clamp body outer diameter

S_7 = shear stress (maximum) across clamp lips

S_8 = clamp lug bending stress

S_9 = effective bearing stress between clamp and hub

S_a = allowable bolt stress at atmospheric temperature (see Section II, Part D, Subpart 1, Table 3)

S_{AC} = allowable design stress for clamp material at (assembly condition) atmospheric temperature (see Section II, Part D, Subpart 1, Tables 1A and 1B)

S_{AH} = allowable design stress for hub material at (assembly condition) atmospheric temperature (see Section II, Part D, Subpart 1, Tables 1A and 1B)

S_b = allowable bolt stress at Design Temperature (see Section II, Part D, Subpart 1, Table 3)

S_{OC} = allowable design stress for clamp material at (service condition) Design Temperature (see Section II, Part D, Subpart 1, Tables 1A and 1B)

S_{OH} = allowable design stress for hub material at (service condition) Design Temperature (see Section II, Part D, Subpart 1, Tables 1A and 1B)

T = thickness of hub shoulder for design purposes. The hub shoulder ring is the ring with cross-sectional dimensions T by $(A - B)/2$.

W = total design bolt load required for service or assembly, as may apply

W_e = total effective axial clamping preload on one clamp lip and hub shoulder (gasket seating or assembly)

$= 1.571W/\tan(\phi + \mu)$

W_{m1} = minimum required total bolt load for the service conditions [see Q-1140(b)(1)]

W_{m2} = minimum required total bolt load for gasket seating [see Q-1140(b)(2)]

W_{m3} = minimum required total bolt load for assembly [see Q-1140(b)(3)]

X = radial distance from inside surface of clamp body to the centroid of the clamp body

$= \left[\left(\frac{C_w}{2} - \frac{C_t}{3} \right) C_t^2 - \frac{(C_w - C_g)}{2} l_c^2 \right] / A_c$

y = gasket or joint-contact surface unit seating load (see Table XI-3221.1-1)

Z = effective clamp–hub taper angle, deg (for gasket seating and preload, $Z = \phi + \mu$; for operating, $Z = \phi - \mu$) [see Q-1140(b)(4)]

α = hub–neck pipe transition taper, deg (α shall not be greater than 45 deg)

μ = effective friction angle, deg

ϕ = clamp–hub taper angle, deg (ϕ shall not exceed 35 deg)

Q-1140 BOLT LOADS

(a) General. During assembly of the clamp connection, the design bolt load W is transferred via the clamp–hub taper angle to an axial load (effective clamp preload W_e). In addition, the effect of friction will cause W_e to be reduced for a given W. Friction effects can be reduced by lubrication or by jarring the clamps during assembly. An appropriate friction angle shall be established for both assembly and operating conditions.

(b) Calculations. In the design of the bolting for a clamp connection, complete calculations shall be made for three separate and independent sets of conditions which are defined as follows.

(1) The required bolt load for the service conditions W_{m1} shall be sufficient to:

(-a) resist the hydrostatic end force H exerted by the maximum allowable working pressure on the area bounded by the diameter of gasket reaction; and

(-b) maintain on the gasket or joint-contact surface, a compression load H_p which experience has shown to be sufficient to ensure a tight joint.

The minimum operating bolt load W_{m1} shall be determined in accordance with eq. (1).

$$W_{m1} = 0.637 \left(H + H_p \right) \tan\left(\phi - \mu \right) \qquad (1)$$

(2) Before a tight joint can be obtained, it is necessary to seat the gasket or joint-contact surface properly by applying a minimum initial load (under atmospheric temperature conditions without the presence of internal pressure), which is a function of the gasket material and

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Figure Q-1130-1
Typical Hub and Clamp

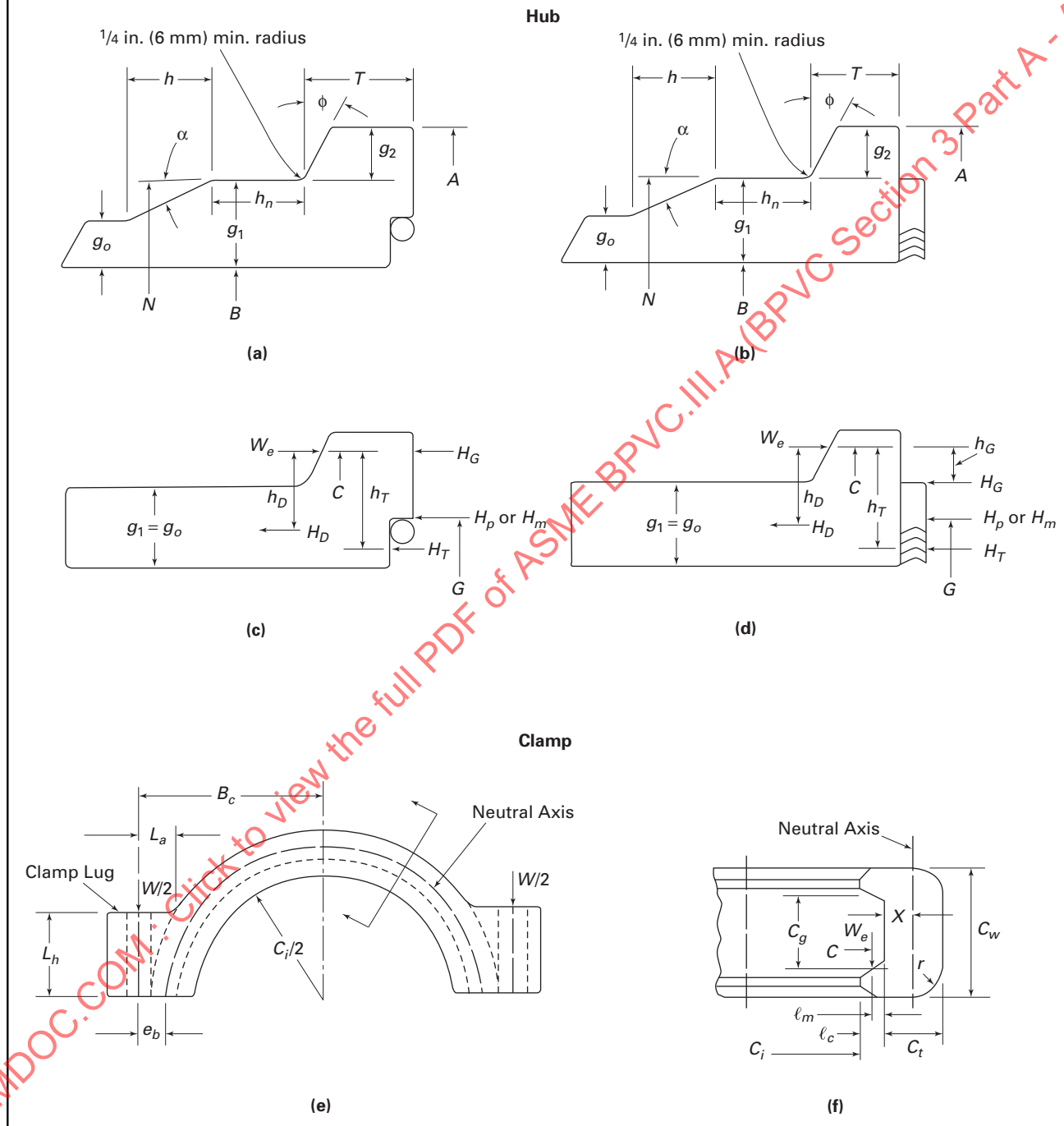

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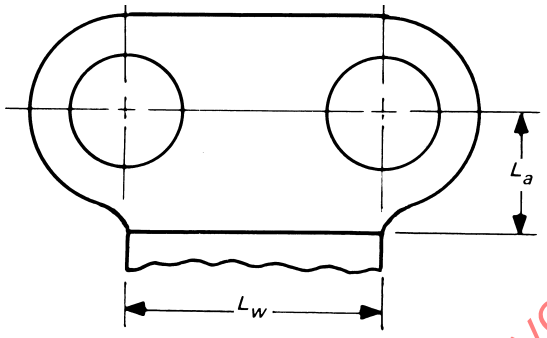
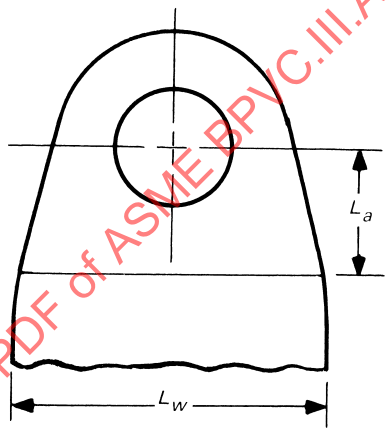
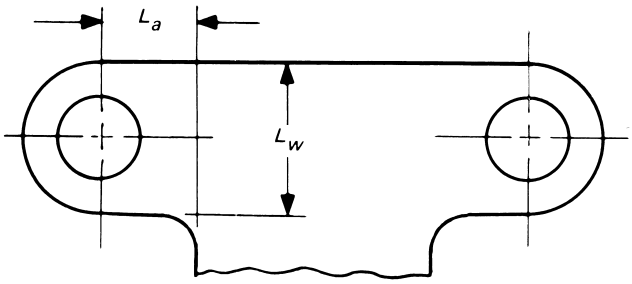
Figure Q-1130-2
Typical Clamp Lug Configurations



(a)



(b)



(c)

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the effective gasket area to be seated. The minimum initial bolt load required for gasket seating W_{m2} shall be determined in accordance with eq. (2).

$$W_{m2} = 0.637\, H_m \tan\left(\phi + \mu\right) \tag{2}$$

(3) To ensure proper preloading of the clamp connection against service conditions, an assembly bolt load W_{m3} shall be determined in accordance with eq. (3).

$$W_{m3} = 0.637\left(H + H_p\right)\tan\left(\phi + \mu\right) \tag{3}$$

(4) In eq. (1)(1) credit for friction is allowed based on clamp connection geometry and experience, but shall be limited to a value in which $(\phi - \mu)$ is equal to or greater than 5 deg. In eqs. (2)(2) and (3)(3), friction shall be considered and be such that μ is equal to or greater than 5 deg. This will then satisfy the requirements of Q-1110(d).

(5) The need for providing sufficient bolt load for either gasket seating in accordance with eq. (2)(2) or assembly in accordance with eq. (3)(3) will prevail on many low pressure designs and with facings and materials that require a high seating load where the service bolt load computed by eq. (1)(1) is insufficient to properly preload the connection.

(c) Required Bolt Area. The total cross-sectional area of bolting A_m required shall be the greater of the values for service conditions A_{m1}, gasket seating conditions A_{m2}, or assembly condition A_{m3}. Bending of the bolting due to nonparallel nut bearing surfaces shall be compensated for by use of a stress correction factor in bolt area calculations or by use of spherically seated nuts and/or washers.

(d) Clamp Connection Design Bolt Load W. The bolt load used in the design of the clamp connection shall be the values obtained from eqs. (4) and (5).

For service conditions:

$$W = W_{m1} \tag{4}$$

For assembly conditions:

$$W = \left(A_m + A_b\right)S_a \tag{5}$$

Q-1150 HUB MOMENTS

In the calculation of hub stresses, the moment of a load acting on the hub is the product of the load and its moment arm. The moment arm is determined by the relative position of the effective hub–clamp reaction diameter with respect to that of the load producing the moment [see Figure Q-1130-1, sketches (c) and (d)]. In addition to the load moments, additional reaction moments with relation to the hub shoulder ring centroid are considered to compensate for hub-to-pipe transition, radial pressure, and clamp radial equilibrating effects.

For the service conditions, the rotational hub moment M_O is the sum of six individual moments M_D, M_G, M_T, M_F, M_P, and M_R based on the design bolt load of Q-1140(d) eq. (4) with moment arms as given in Figure Q-1130-1.

For assembly, the rotational hub moment M_O is based on the design bolt load of Q-1140(d) eq. (5) in which case

$$M_O = \frac{0.785\, W\left(C - G\right)}{\tan\left(\phi + \mu\right)} \tag{6}$$

Q-1160 CALCULATION OF HUB STRESSES

The stresses in the hub shall be determined for both the service and the assembly conditions.

(a) The reaction moment M_H and reaction shear Q as defined in Q-1130 shall be calculated at the hub neck for rotational moment M_O.

(b) Hub stresses are to be calculated from the following equations:

Hub longitudinal stress

$$S_1 = f\left[\frac{PB^2}{4g_1\left(B + g_1\right)} + \frac{1.91\, M_H}{{g_1}^2\left(B + g_1\right)}\right] \tag{7}$$

Hub hoop stress

$$S_2 = P\left(\frac{N^2 + B^2}{N^2 - B^2}\right) \tag{8}$$

Hub axial shear stress

$$S_3 = \frac{0.75\, W}{T\left(B + 2g_1\right)\tan Z} \tag{9}$$

Hub radial shear stress

$$S_4 = \frac{0.477\, Q}{g_1\left(B + g_1\right)} \tag{10}$$

Q-1170 CALCULATION OF CLAMP STRESSES

The stresses in the clamp shall be determined for both the service and the assembly conditions. Clamp stresses are to be calculated from the following equations:

Clamp longitudinal stress

$$S_5 = \frac{W}{2C\left(\tan Z\right)}\left[\frac{1}{C_t} + \frac{3\left(C_t + 2l_m\right)}{{C_t}^2}\right] \tag{11}$$

Clamp tangential stress

$$S_6 = \frac{W}{2}\left[\frac{1}{A_c} + \frac{\left|e_b\right|\left(C_t - X\right)}{I_c}\right] \tag{12}$$

Clamp lip shear stress

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$$S_7 = \frac{1.5\,W}{\left(C_w - C_g\right) C \tan Z} \quad (13)$$

Clamp lug bending stress

$$S_8 = 3\,W \frac{L_a}{L_w L_h^{\,2}} \quad (14)$$

In addition, a bearing stress calculation is to be made from eq. (15) for either the clamp or hub.

$$S_9 = \frac{W}{\left(A - C_i\right) C \tan Z} \quad (15)$$

Q-1180 ALLOWABLE DESIGN STRESS FOR CLAMP CONNECTIONS

Table Q-1180-1 gives the allowable stresses that are to be used with the equations of Q-1160 and Q-1170.

Table Q-1180-1
Allowable Design Stress for Clamp Connections

Stress Category	Allowable Stress
S_1	$1.5S_{OH}$ or $1.5S_{AH}$
S_2	S_{OH}
S_3	$0.8S_{OH}$ or $0.8S_{AH}$
S_4	$0.8S_{OH}$ or $0.8S_{AH}$
S_5	$1.5S_{OC}$ or $1.5S_{AC}$
S_6	$1.5S_{OC}$ or $1.5S_{AC}$
S_7	$0.8S_{OC}$ or $0.8S_{AC}$
S_8	S_{OC} or S_{AC}
S_9	[Note (1)]

NOTE:
(1) 1.6 times the lower of the allowable stresses for hub material (S_{OH}, S_{AH}) and clamp material (S_{OC}, S_{AC}).

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(15)

NONMANDATORY APPENDIX R
DETERMINATION OF PERMISSIBLE LOWEST SERVICE METAL TEMPERATURE FROM T_{NDT} FOR DIVISION 1, CLASSES 2 AND MC; AND DIVISION 3, CLASS WC CONSTRUCTION

ARTICLE R-1000
PERMISSIBLE LOWEST SERVICE METAL TEMPERATURE

R-1100 INTRODUCTION

R-1110 SCOPE

These rules provide the method for determining the permissible lowest service metal temperatures for materials having nil-ductility transition temperatures (T_{NDT}) determined in accordance with NC-2311(a)(8), NE-2311(a)(8), WC-2311(a)(7) or NC-2331(b), NE-2331(a)(2), WC-2331(a)(2), as applicable.

R-1200 DETERMINATION OF PERMISSIBLE LOWEST SERVICE METAL TEMPERATURE

The permissible lowest service metal temperature is defined as:

$$T_{NDT} + A$$

where T_{NDT} is determined in accordance with NC-2311(a)(8), NE-2311(a)(8), WC-2311(a)(7) or NC-2331(b), NE-2331(a)(2), WC-2331(a)(2), as applicable, and A is determined from Figure R-1200-1 for the thickness of the material.

R-1210 MATERIAL ACCEPTABILITY

For the material to be acceptable, the permissible lowest service metal temperature shall not be higher than the specified lowest service metal temperature.

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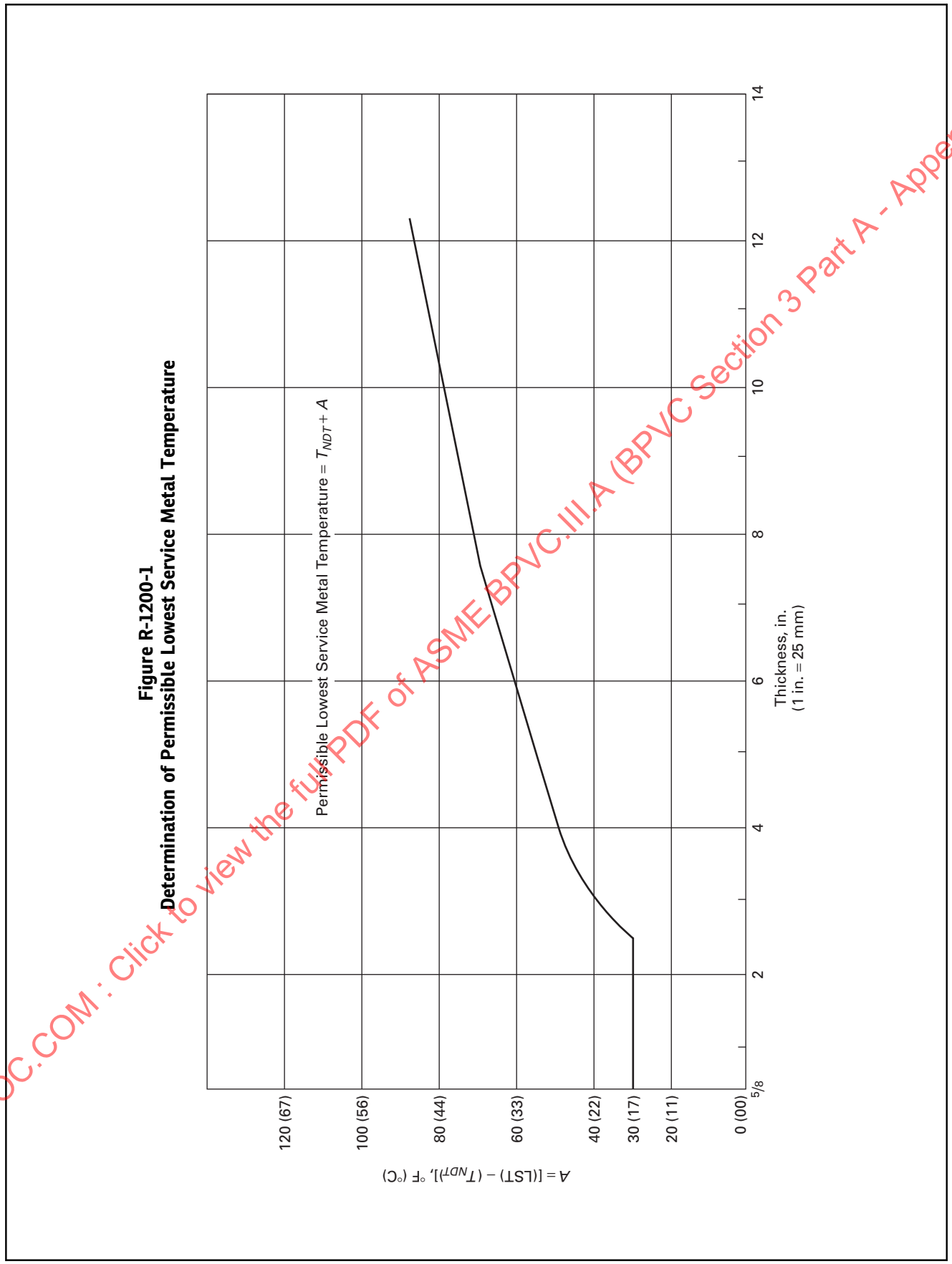
Figure R-1200-1
Determination of Permissible Lowest Service Metal Temperature



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NONMANDATORY APPENDIX S

ARTICLE S-1000
PUMP SHAFT DESIGN METHODS

S-1100 INTRODUCTION

(a) This Appendix provides guidelines for the design and evaluation of shafts for Section III nuclear pumps. They include suggestions for the loads to be considered and the method for arriving at a design that will sustain these loads for the life of the pump.

(b) The method presented is not intended to exclude other methods which may be equally satisfactory. Experience and expertise will be factors in determining the best method. In the final analysis the skill and judgment of the designer determine the quality of the design.

S-1200 SCOPE

The guidelines presented are intended to cover the design of pump shafts, including rigid couplings, up to, but not including, separable parts which attach the shaft to the driver.

S-1300 DESIGN REQUIREMENTS

S-1310 DESIGN METHODS

The design rules presented in this section are provided as guidance to the designer of pump shafts. Alternative design methods may be used based on the pump manufacturer's experience with pumps in similar service.

S-1320 OPERATING CONDITIONS

Pump shaft assemblies are subject to combinations of steady state and variable or transient loads. These loads include torsional, lateral, bending, axial, and thermal components. They may occur as a result of power input, hydraulically imposed forces, static or dynamic unbalance, rotating element runout, internal misalignment, thermal distortion, system and component vibration, and resonance. External system applied loads, such as seismic and other loads associated with A, B, C, and D Service Level Loadings must also be considered.

S-1400 RESPONSIBILITY

(a) Owner's Responsibility. It shall be the responsibility of the Owner or his designee to include in the Design Specification all external forces and operating conditions that may have an effect on the operability of the pump shaft.

(b) Pump Designer Responsibility. It is the responsibility of the pump Designer to include in the evaluation the specified pump loads and identify the type and magnitude of the internal operating loads on the shaft assembly.

S-1500 OPERATING LOADS

S-1501 MAXIMUM SHAFT OPERATING LOADS

In establishing maximum shaft operating loads, the design should take into account plant service that the pump will experience and external loads associated with these conditions. Other forms of off-normal operating loadings may include inservice testing, inadvertent starting and stopping, loss of coolant accident, etc.

S-1502 THERMAL LOADS

Thermal loads shall be considered as one of the internal operating loads.

S-1503 OFF-NORMAL OPERATING LOADS

The maximum steady state and transient loads will usually occur in the pump when it is operating away from its best efficiency point. Typically it will be the combination of loads producing stresses at regions of high stress concentration that may result in high cycle fatigue failures of shafts. The very nature of these loads makes it difficult to quantify them and in some cases a bounding estimate must be made. Examples of load sources include low flow recirculation, flow separation, and other hydraulic instabilities which cause radial and axial alternating or transient loads on the shaft. Lacking a thorough understanding of these loads, conservative design practices based on years of operating experience must be used to insure successful design of pump shafts.

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S-1600 SHAFT FAILURE MODES

Shaft failures usually occur at points of high stress concentration or structural discontinuities. The most common locations of shaft failures are threaded regions, shaft grooves, shoulders, keyways, couplings, and collars (see Figure S-1600-1). Areas susceptible to erosion/corrosion, stress corrosion cracking, thermal transients, and steep temperature gradients are also possible locations for shaft failure.

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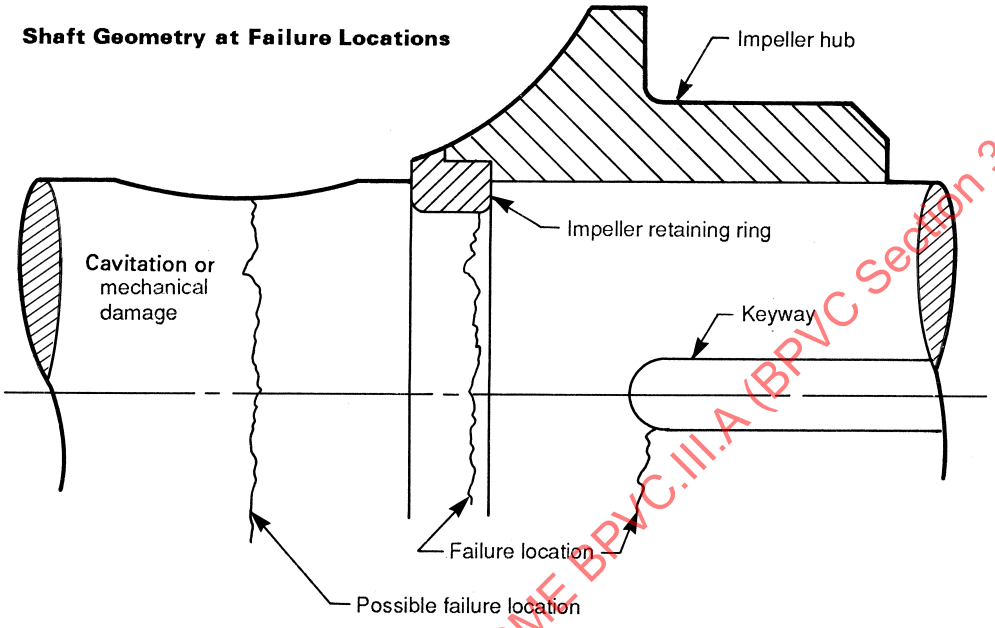
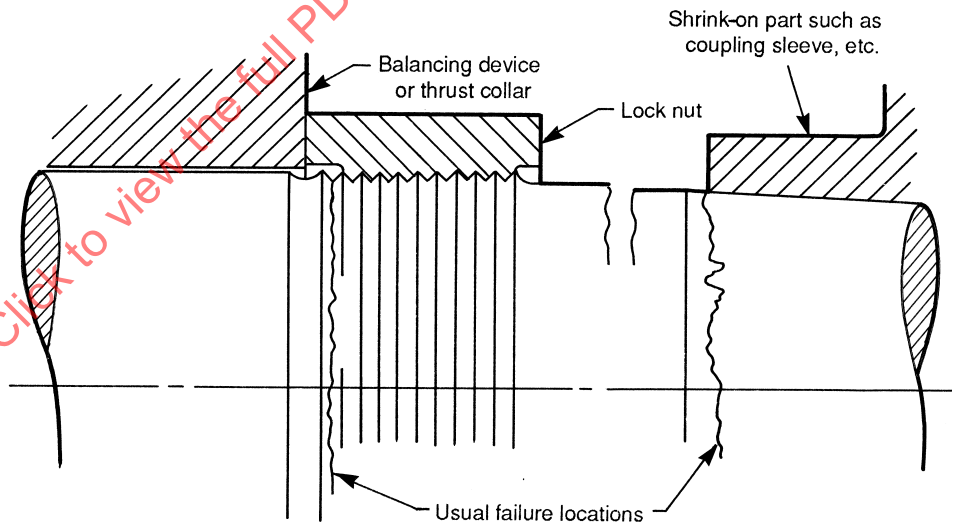
Figure S-1600-1
Typical Centrifugal Pump Shaft Failure Locations



(a) Shaft Groove or Interstage External Damages



(b) Threaded or Shrunk-on Parts

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ARTICLE S-2000
DESIGN PROCEDURE

Any method of evaluation (analytical or experimental) which can be substantiated by data from pumps in service, experiencing conditions similar to the specified operating limits, may be used.

S-2100 CRITICAL SPEEDS

The evaluation shall address both torsional and lateral, and where applicable, axial critical speeds, shaft deflections, and stresses. Critical speeds and shaft deflections shall be such as to avoid any difficulties for the specified range of the design and operating conditions. The actual percentage difference between critical and operating speeds shall take account of the method of determination of critical speed. The percentage difference between stress allowed and calculated shall also take account of the accuracy of the design and the analysis method.

S-2200 MAXIMUM TORSIONAL LOAD

The maximum torsional load shall be defined. The maximum torsional shear stress for this load (stress resulting from this load without application of concentration factors) shall be based on design experience or experimental evidence for the particular class of pump involved. The maximum driver horsepower may determine the maximum torsional loading for units with short shafts (typical of pump types A, B, and C). The motor startup torque may determine the maximum torsional load for pumps with relatively long shafts (typical of type L pumps). Torsional alternating or transient loads shall be considered if applicable.

S-2300 SHAFT EVALUATION

The flow chart, Figure S-2300-1, outlines a procedure for evaluation of pump shafts to meet load requirements. This procedure establishes a basic sizing criterion as well as a detailed fatigue evaluation method.

The basic shaft sizing criterion is based on maximum shear stress and conservative cyclic loading factors. These fatigue factors include an evaluation of the endurance limit of the unnotched and polished test specimen reverse bending test data in air (S_e), which in the absence of specified data can be approximated as:

$$S_e = 0.5S_u$$

This polished specimen test endurance limit is then factored for the product of reduction factors that account for such items as environment, reliability, size, finish, duty cycles, etc., which is conservatively estimated as one-third. Consequently, this corrected material endurance limit stress can be represented as:

$$S'_e = \frac{1}{K_t}\frac{S_e}{3}$$

where the terms used on the chart, Figure S-2300-1, are as follows:

K_t = stress concentration factors. An initial value of 6.0 is suggested where reasonable stress riser control is exercised. Higher values may be required for designs with severe discontinuities (e.g., small fillet radii relative to shaft diameter). Lower values may be used if justified by design methods and/or testing that accounts for the specific shaft discontinuities under consideration. Notch sensitivity values, when available, may be used. For additional information, see ANSI/ASME B106.1M-1985.

S_a = alternating axial stress

S_b = alternating component of the shaft bending stress

S_e = material endurance limit

S'_e = design endurance limit

S_s = maximum shear stress

S_{ss} = material allowable shear stress
= $S_y/3$ unless higher values are justified through specified shear test data

S_u = material ultimate strength at design temperature

S_y = material yield stress at design temperature

U_t = summation of usage cycles. Each usage cycle shall be determined as the ratio of the maximum alternating stress in the shaft divided by the design fatigue stress of the material for that number of cycles. U_t will be determined by summing all of the above ratios.

S-2400 OTHER CONSIDERATIONS

The fatigue life of a shaft is not always the limiting factor in its design. The effect of misalignment and deflection of a shaft on the performance of support bearings, seals, and couplings as well as on other key power transmission components must also be taken into account. Shafts can be strong enough to meet fatigue life requirements, yet not stiff enough to satisfy natural frequency and operational requirements.

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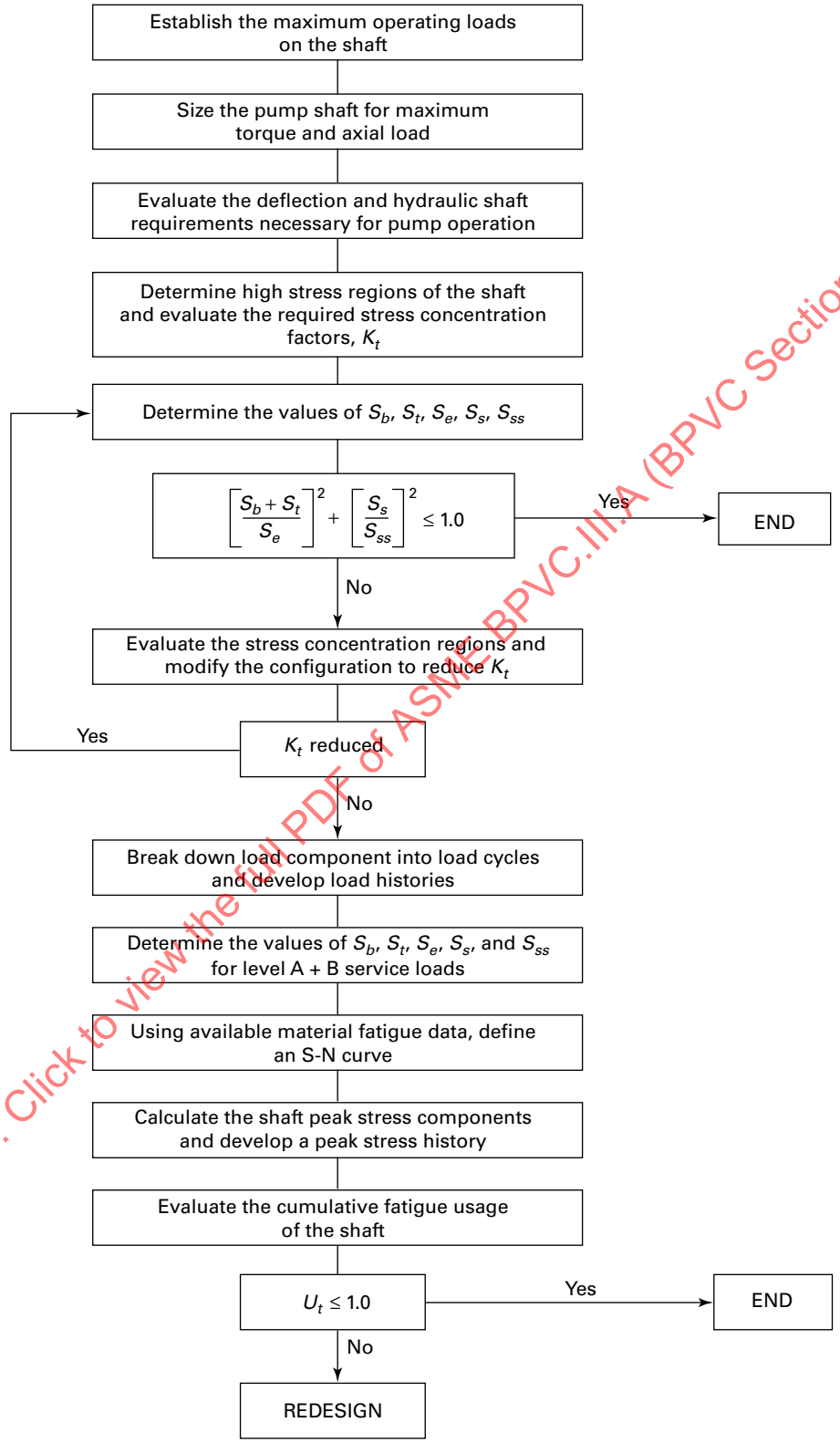
Figure S-2300-1
Steps in the Design of a Pump Shaft



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NONMANDATORY APPENDIX T

ARTICLE T-1000
RECOMMENDED TOLERANCES FOR RECONCILIATION OF PIPING SYSTEMS

T-1100 INTRODUCTION

The building structure and major components of a power plant are constructed according to rules that permit varied tolerances. Since piping system installation follows construction of the building and installation of the major components, the piping systems must be permitted to vary within the space allotted to them. In addition, a large number of systems are often installed in a limited space. Interferences often occur and changes within Installation Tolerances may be used to eliminate the interference. The tolerances provided in this Appendix bridge the gap between the exactness associated with a design by analysis, and a practical and acceptable installation.

The basis for the tolerances and guidance in this Appendix was developed by the PVRC Technical Committee on Piping Systems.[38] Additional guidance on implementation of these tolerances has been published by EPRI.[39]

T-1110 SCOPE

This Appendix provides recommended tolerances and methods for satisfying the requirements of NCA-3554, "Modification of Documents and Reconciliation With Design Report" for piping systems designed to the rules of NB-, NC-, or ND-3600. This Appendix provides:

(a) identification of dimensions and weights significant to the piping stress analysis; and

(b) acceptable tolerances for these dimensions and weights such that if piping is installed within these tolerances, the reconciliation is accomplished.

These tolerances have been established such that their effect on the accuracy of analysis results is minimal and is consistent with accepted practices and the use of tolerances in the Code. These tolerances are applicable to most situations; however, specific situations where more restrictive tolerances may be needed are identified.

The tolerances in this Appendix are applicable to piping systems where conventional seismic analysis methods were used for the original design, i.e., modal response spectrum analysis methods. For piping systems where seismic analysis methods that are significantly more sensitive to the tolerances were used in the original design (i.e., seismic time history analysis methods), the Designer shall review the applicability of these tolerances and establish more stringent guidelines if necessary. Further, this Appendix shall be restricted to piping systems analyzed using linear elastic methods.

This Appendix does not relieve the Designer of responsibility for consideration of other unique situations where more restrictive tolerances may be required to satisfy the intent of the design bases or the Code.

Installation Tolerances more restrictive than the Total Tolerances recommended in this Appendix may be specified. Less restrictive tolerances may be specified when engineering justification is provided to demonstrate that the design requirements have been satisfied.

Tolerances for complete, installed piping systems are addressed. Tolerances provided for manufacturing or fabricating the individual items or subassemblies that make up piping systems are not addressed, but the effect of these tolerances on the as-installed condition shall be within the Total Tolerance. Other design and construction areas which may be included in reconciliation such as design or operating conditions, support details, and gaps are not addressed.

Tolerances for support erection, including length and orientation of individual members and pipe location on the support, are specified in Subsection NF, Appendix NF-D, Tolerances.

T-1120 TERMS RELATED TO RECONCILIATION

Definitions of terms used in this Appendix are given in the following paragraphs.

T-1121 Nominal Dimension

This is the dimension which provides configuration and/or spatial information on piping drawings within specified tolerances.

T-1122 As-Analyzed Configuration

This is the configuration of piping components and supports, defined by nominal sizes, weights, cross section properties, and dimensions, which forms the basis for

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the piping stress analysis. In the case of piping systems which are qualified by simplified rules, the design drawings for the systems are considered as the As-Analyzed Configuration.

T-1123 Critical-to-Design (CTD) Dimension

A dimension that must be satisfied, within a specified tolerance, in order for the piping stress analysis to remain valid. These dimensions define the relative configuration of the piping. They may also include dimensions which define the global or spatial position of the piping.

Examples of CTD dimensions are: the location of a pipe support relative to in-line pipe components such as valves, anchors, and other supports; the orientation of the pipe support centerline relative to the pipe centerline; length of pipe runs; and spacing between supports.

T-1124 Fit/Clearance (F/C) Dimension

This is a dimension that is specified to provide reasonable assurance that the piping fits into the allocated building space. These dimensions define the global position of the piping in three-dimensional space and deviations from these dimensions do not have a significant effect on the validity of the piping stress analysis. Examples of F/C dimensions are the locating dimensions of pipe centerlines from column lines, walls, and floors. However, dimensions defining clearance required to allow thermal expansion or to protect critical components from adverse interaction due to dynamic (earthquake) loads are CTD dimensions.

T-1125 As-Built Documents

These are the drawings, sketches, or other documents which define the as-installed piping configuration (nominal dimensions and tolerances) and which have been reconciled with the stress analysis.

T-1126 Installation Tolerance

This is the specified acceptable departure from nominal piping dimensions, support locations and orientations, and component weights to be used during installation to provide for practical installation limitations. Installation Tolerances for CTD dimensions shall be less than or equal to the corresponding Total Tolerances.

T-1127 Total Tolerance

This is the maximum allowable departure between corresponding as-built and as-analyzed CTD piping dimensions, support locations and orientations, and component weights for the piping stress analysis to be applicable. The Total Tolerance is applicable only to CTD dimensions since F/C dimensions are not critical to the stress analysis. Acceptable Total Tolerances are provided in T-1200.

T-1130 MEASUREMENT ACCURACY

Measurements of the as-built configuration for use in reconciliation shall be made using methods capable of producing accuracy to the nearest inch for linear dimensions and to the nearest 2 deg for angular dimensions (Figure T-1213-1 shows examples of angular dimensions).

T-1140 EVALUATION OF OUT-OF-TOTAL TOLERANCE CONDITIONS

CTD values that exceed Total Tolerances shall be recorded and evaluated to assure the design bases, including the applicable design code, have been satisfied. This evaluation may be made using engineering judgment, simplified models, or by reanalyzing the complete original model. The objective should be to determine if the condition being evaluated had any significant effect on the response of the piping systems to the design loadings. The use of engineering judgment shall be documented with the technical reasoning described so that a technically qualified third party reviewer will understand how the evaluation was justified. The documentation requirements for evaluation based on engineering judgment shall be consistent with other engineering calculations.

If it is determined that there is no significant effect, then the original analysis remains valid and the results (stress summaries, support design loads, etc.) need not be revised. If changes are found to be significant, then the affected results shall be revised.

T-1200 TOTAL TOLERANCES

T-1210 GENERAL

The Total Tolerances are given for the reconciliation of the piping stress analysis in terms of the maximum allowable plus or minus departure between the as-analyzed (nominal) value and the as-built value. These tolerances should not be used to circumvent specific requirements established by the Design or Installation Specification nor other applicable material or fabrication standards.

As-built piping systems that conform with the piping stress analysis within the Total Tolerances are acceptable and evaluation of the as-built condition is not required.

T-1211 Altering Relative Position

Tolerances shall not be applied in such a manner that would result in the alteration of the relative position of piping components and supports. For example, supports shall not be moved across valves, tees, elbows, etc. In no case shall the permissible location change the directions of a support or its function.

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T-1212 Independence

The tolerances given are independent of each other and are not interrelated to position of adjacent items except as noted in T-1211. Each tolerance may be applied independently of any other.

T-1213 Methods of Dimensioning

Examples of angular dimensions are shown in Figure T-1213-1. Two common methods of dimensioning piping installation are shown in Figure T-1213-2. Either method or some combination of these methods are usually used on the Design Drawings. The chain method references a dimension to the adjacent dimensions. The common point method references a number of dimensions to a common base. All measurements should be rounded off to the nearest inch for linear dimensions and the nearest 2 deg for angular dimensions.

T-1220 PIPING CONFIGURATION TOLERANCES

T-1221 Centerline Lengths

Centerline length to fittings, flanges, valves, and piping specialties including branch line piping shall be within the tolerance specified below. (Locations of branch connections on the run pipe are not included. See T-1222.)

Specified Nominal Dimension, ft (m)	± Total Tolerance, in. (mm)
0 to < 5 (0 to < 1.5)	3 (75)
5 to < 10 (1.5 to < 3.0)	6 (150)
10 to < 15 (3.0 to < 4.5)	9 (225)
15 to < 20 (4.5 to < 6.0)	12 (300)
20 to < 25 (6.0 to < 7.5)	15 (375)
25 to < 30 (7.5 to < 9.0)	18 (450)
30 to < 35 (9.0 to < 10.5)	21 (525)
35 and over (10.5 and over)	24 (600)

NOTE: The 3 in. (75 mm) tolerance for nominal dimensions of 0 to less than 5 ft (1.5 m) may not be applicable in all cases. For example, cantilevered vents and drains, relief valve inlet and outlet piping, and instrument piping near the connection to the process piping are cases where more restrictive tolerances may be needed. If so, they shall be specified on the design documents.

When common point dimensions (see Figure T-1213-2) are used, the tolerances along the pipe centerline are to be applied to each pipe leg, i.e., piping between changes in direction. There may be cases where the nominal dimensions must be determined from the difference in common point dimensions for the ends of the pipe leg.

T-1222 Centerline Location of Branch Connections (Relative to Run)

(a) This section applies to branch connections independent of the connection detail or type of fitting used.

(b) The location of branches for branch/run size combinations indicated by an asterisk (*) in Table T-1222-1 have no design related tolerance restrictions except when the stress intensification factor (SIF) on the run pipe exceeds 1.0, in which case the maximum tolerance is ±2 ft 0 in. (600 mm) and the requirements of (d) shall be satisfied.

(c) The location of branches for the branch/run size combinations without an asterisk (*) in Table T-1222-1 (including sizes not listed in the table) have the same tolerance as in T-1221.

(d) For all branch/run size combinations, changes in location of the branch connection that exceed 2 ft (600 mm) shall be approved by the Designer and should not

(1) change the flow path defined on the piping system diagram.

(2) affect system functional flow and pressure characteristics, such as branch connections used for differential pressure and flow measurement, for safety and relief valves, and for core safety injection.

(3) be allowed without considering the effect of run pipe movement on the branch pipe. (Not significant if the tolerances of T-1221 are satisfied.)

T-1223 Angular Deviation of Pipe Centerline

Angular deviation of pipe centerline from theoretical centerline is limited to ±10 deg. See Figure T-1213-1, Rolled Elbows (“B°”) and Bends; Nonstandard Elbows (“C°”).

T-1224 Angular Tolerance on Power Operated Valves

Angular tolerance on power operated valves over NPS 2 (DN 50) (Nominal Pipe Size) is limited to ±15 deg. See Figure T-1213-1, Valves (“D°”). For power operated valves NPS 2 (DN 50) and smaller, the angular tolerance of ±15 deg applies only when the weight of the valve operator equals or is less than the valve analyzed weight. When the weight of the valve operator exceeds the valve’s analyzed weight for an NPS 2 (DN 50) or smaller valve, the angular tolerance is not provided in this Appendix. There are no CTD restrictions on angular orientation of manually operated valves.

T-1230 PIPING SUPPORT LOCATION/ ORIENTATION TOLERANCES

The tolerance provided as piping support tolerances assume the support is the specified type and rating. The support type shall not be changed without approval of the Designer.

These tolerances are applicable to the reconciliation of the piping stress analysis. They are not applicable for the reconciliation of the pipe support design.

T-1231 Location of Supports

Tolerances on the location of supports, anchors, and restraints along the pipe centerline of the horizontal or vertical straight runs that do not contain significant concentrated weight are

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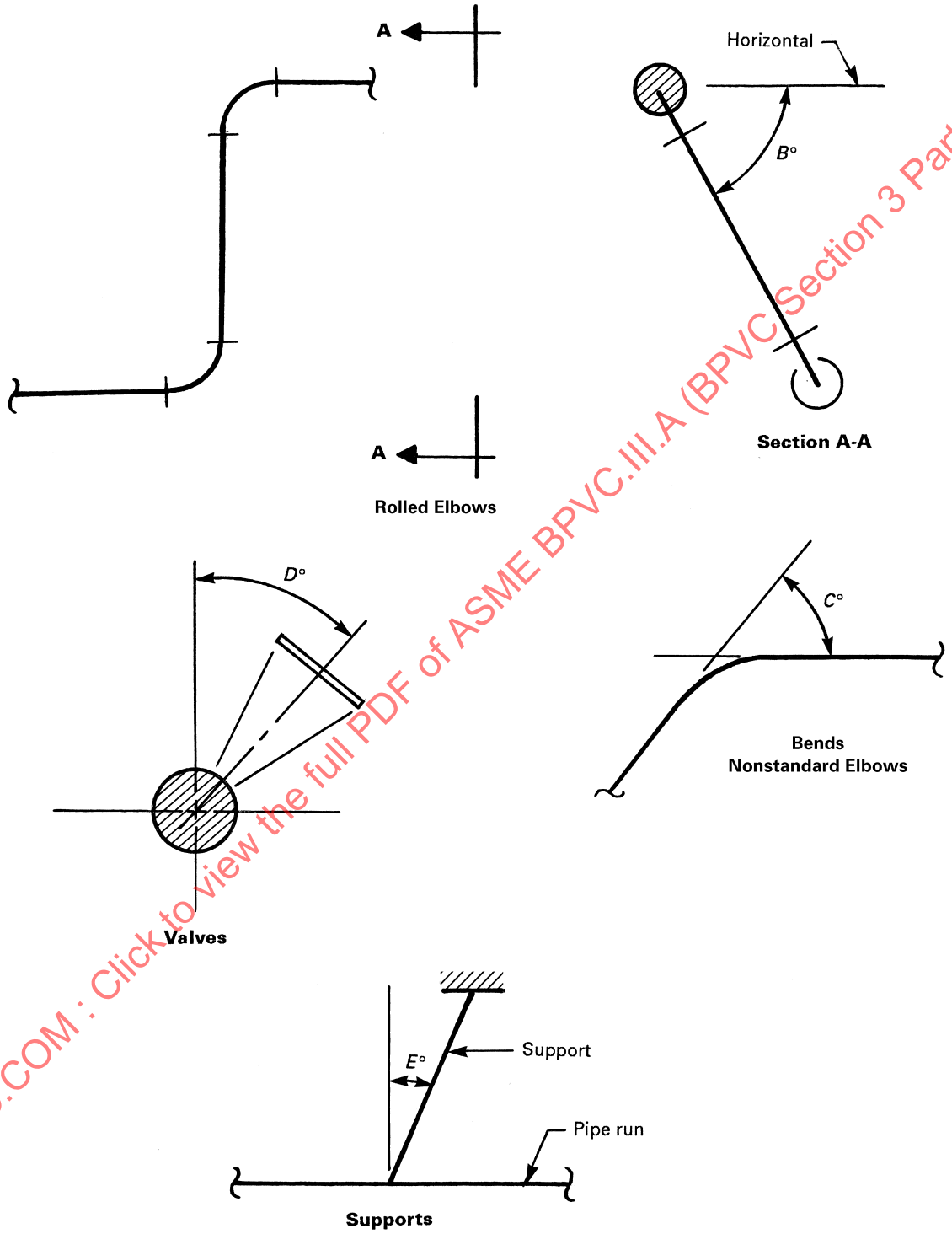
Figure T-1213-1
Illustrations of Angular Dimensions — Pipe Legs, Valves, Supports, Bends



GENERAL NOTE: $B°$, $C°$, $D°$, and $E°$ indicate angular dimensions.

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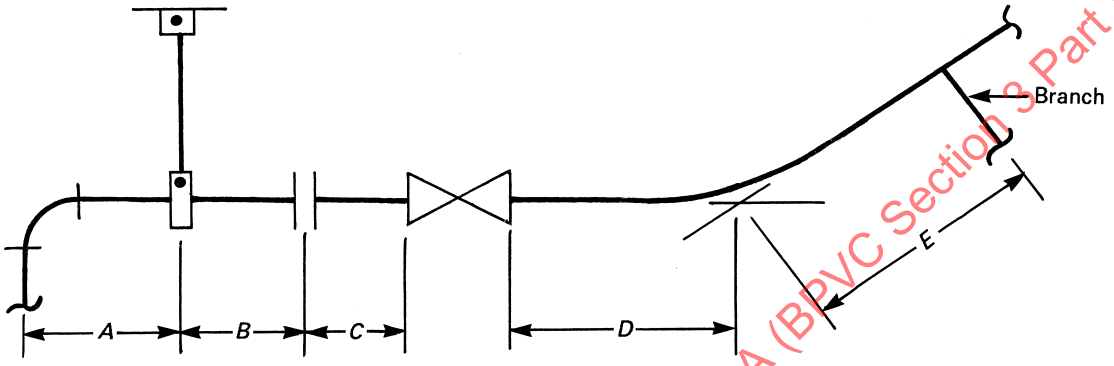
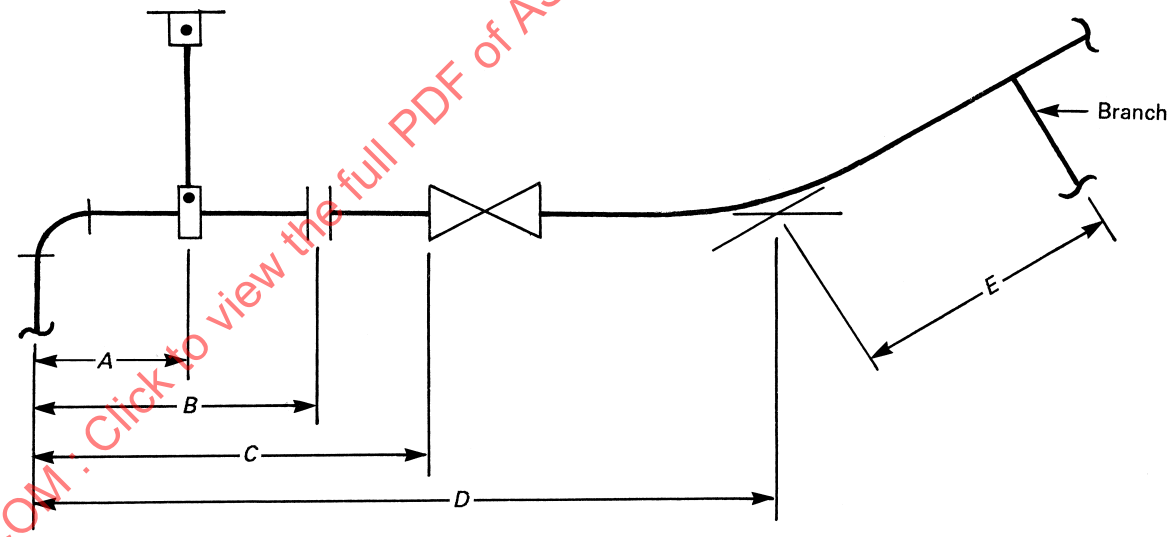
Figure T-1213-2
Illustrations of Linear Dimensions



Method 1 — Chain Dimensions



Method 2 — Common Point Dimensions

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Table T-1222-1
Branch/Run Size Combinations

Run Size — NPS (DN)	Branch Size — NPS (DN)						
	¾ (20)	1 (25)	1½ (40)	2 (50)	2½ (65)	3 (80)	4 (100)
¾ (20)							
1 (25)							
1½ (40)							
2 (50)							
2½ (65)	*						
3 (80)	*	*					
4 (100)	*	*					
6 (150)	*	*	*	*			
8 (200)	*	*	*	*	*		
10 (250)	*	*	*	*	*	*	
12 (300)	*	*	*	*	*	*	
14 (350)	*	*	*	*	*	*	
16 (400)	*	*	*	*	*	*	
18 (450)	*	*	*	*	*	*	
20 (500)	*	*	*	*	*	*	
24 (600)	*	*	*	*	*	*	

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Pipe Size	Tolerance
NPS 2 (DN 50) and smaller	6 in. (150 mm)
NPS 2½ (DN 65) and larger	One pipe diameter or 12 in. (300 mm), whichever is greater

T-1232 Location of the First Support on Either Side of Spans That Contain Concentrated Weights

Tolerances on the location of the first support or restraint on either side of spans that contain concentrated weights such as valves, flanges, risers, fittings, bends, or other concentrated loads are

Pipe Size	Tolerance
NPS 12 (DN 300) and smaller	Smaller of 3 pipe diameters or 12 in. (300 mm)
Larger than NPS 12 (DN 300)	One pipe diameter

These tolerances may not be applicable to the first restraint adjacent to a bend, which is oriented in the plane of the bend, when the piping is subjected to waterhammer, steamhammer, or relief valve discharge loading. In these cases, if more restrictive tolerances are needed they must be specified on the design documents.

T-1233 Location of the First Support From Rotating Equipment Nozzles

Tolerances for the location of the first support or restraint in each direction from rotating equipment nozzles are

Pipe Size	Tolerance
NPS 2 (DN 50) and smaller	3 in. (75 mm)
NPS 2½ (DN 65) and larger	One-half pipe diameter or 6 in. (150 mm), whichever is greater

T-1234 Snubbers

The location of snubbers with axis coincident to the pipe leg centerline is not a critical dimension. However, snubbers with tolerances greater than those of T-1231 should be approved by the Designer who must assure they are not affected by the anticipated thermal movements.

T-1235 Angular Orientation of Supports

The tolerance for the angular orientation of supports is ±5 deg [see Figure T-1213-1, Supports (“E°”)]. The angular tolerance may be increased to ±10 deg if this value plus the angularity change from thermal expansion of the system does not exceed functional limitations established by the support manufacturer and the Designer assures that any increased loadings on the connection are compatible with the design.

T-1240 WEIGHT TOLERANCES

T-1241 General

Tolerances on both uniformly distributed weights and concentrated weights are included in this Appendix. These tolerances do not imply that components should be weighed, but provide for reconciliation of weight changes identified in the design process. As a practical matter, the concentrated weight of items is a concern primarily for valves and other in-line components of significant weight, but particularly power operated valves. Changes to the distributed weight of insulation systems and other piping items should also be considered.

T-1242 Uniformly Distributed Weight

The uniformly distributed weight for the piping system may vary by ±20% from the as-analyzed weight.

This weight tolerance may need to be reduced for piping primarily supported by constant force supports or springs.

T-1243 Concentrated Weight

Concentrated weights of in-line items such as valves may vary by the greater of ±20% of the analyzed weight or 20 lb (9 kg).

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NONMANDATORY APPENDIX U

ARTICLE U-1000
RULES FOR PUMP INTERNALS

U-1100 INTRODUCTION

U-1110 SCOPE

These rules apply to materials, fabrication and examination of internal items for Class 1, 2, and 3 pumps. Pump internal items are those parts other than pump cases, inlets and outlets, covers clamping rings, seal housings, related bolting and other items as covered in Subsections NB, NC, and ND.

U-1120 CATEGORIES

Category as set forth in Table U-1600-1 is the grouping of various pump internal items for the purpose of applying the rules of this Appendix. Categories for typical pump types are shown in Figures U-1500-1 through U-1500-7. The figures are not to scale, and are not intended to convey any preference for pump type or design, but are provided as a guide to the manufacturer to identify the various internal items of a pump for categorization. In determining categories for items of pump types not specifically illustrated, a pump or pump detail which is most nearly representative shall apply. Categories 1 and 2 are pressure-retaining items presently covered by Subsections NB, NC and ND. Categories 3 through 6 are pump internal items which may be constructed in accordance with this Appendix except that Material Manufacturers and/or Material Suppliers for Category 3 through 6 items are not required to comply with NCA-3800. Material which forms an integral welded extension to Category 1 material in Figure U-1500-1, items 1–5, may be classified as Category 4 or 5 material when it serves a nonpressure-retaining function and its structural effect is considered in the design of the Category 1 material. For use in Class 1, 2, and 3 pumps the extension material shall satisfy the requirements of NB-2300, NC-2300, or ND-2300.

U-1200 GENERAL REQUIREMENTS

U-1210 RESPONSIBILITIES AND DUTIES

It is the responsibility of the Certificate Holder manufacturing pumps to assign each item of a pump to the proper category and to indicate the categories in a report or on a drawing.

U-1220 CODE STAMPING

Certification Mark is not required for pump internal items.

U-1300 MATERIALS

U-1310 GENERAL REQUIREMENTS FOR MATERIAL

U-1311 Scope of Principal Terms Employed

The term *materials* as used in this Appendix applies to those items produced to material specifications permitted by Section III, Division 1, and/or other material permitted by this Appendix.

U-1312 Permitted Material Specifications

(a) Materials used for Category 3 and 4 items shall conform to the requirements of one of the specifications for materials given in Table U-1610-1 of this Appendix for Class 1, 2, and 3 pumps; materials listed in Section II, Part D, Table 2A for Class 1 and 2 pumps; materials listed in Table 2A for Class 2 pumps; or materials listed in Table 1A for Class 3 pumps; and to the special requirements of this Appendix which apply to the item for which the material is used.

(b) Category 5 and 6 items may be made from any material suitable for the intended service.

(c) The Certificate Holder manufacturing pumps shall provide a list which identifies the material used for each Category 3, 4, 5, or 6 item. This list may be a bill of materials or a separate list.

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Table U-1600-1
Summary of Requirements

Typical Items	Category No. & Pump Class	Stress Report	Certified Material Test Report	Nondestructive Examination	Impact Testing	Material Identif.
Pressure-Retaining Items	Category 1: Class (All)	X	Subsection NB, NC, ND			
Pressure-Retaining Bolting	Category 2: Class (All)	X	Subsection NB, NC, ND			
Shafting Line Shaft Couplings	Category 3: Class 1	—	X	[Note (5)] [Note (7)]	X	X
Impeller Nuts or Impeller Locking Screw	Class 2	—	X	—	[Note (4)]	X
	Class 3	—	X	—	[Note (4)]	[Note (1)]
Impellers Bearing Support	Category 4: Class 1	—	[Note (2)] [Note (3)]	[Note (6)] see U-1120	— See U-1120	X
Bearings Journals	Class 2	—	[Note (2)]	— See U-1120	— See U-1120	X
	Class 3	—	[Note (2)]	— See U-1120	— See U-1120	[Note (1)]
Im./Case Rings Keys Mech. Seal (Met.) Parts Bolting (Internal) Bearings Journals	Category 5: Class 1	—	[Note (2)]	— See U-1120	— See U-1120	[Note (1)]
	Class 2	—	—	— See U-1120	— See U-1120	[Note (1)]
	Class 3	—	—	— See U-1120	— See U-1120	[Note (1)]
Packing Gaskets "O" Rings Carbon	Category 6: Class 1	—	—	—	—	—
	Class 2	—	—	—	—	—
	Class 3	—	—	—	—	—

NOTES:
(1) Quality control system.
(2) Materials Manufacturer's Certificate of Compliance.
(3) Certified Material Test Reports required for impellers.
(4) When required for the pump per Design Specification.
(5) Ultrasonic Examination required for Class 1 shafting.
(6) Magnetic Particle or Liquid Penetrant Examination required for Class 1 impellers.
(7) Magnetic Particle or Liquid Penetrant Examination required for all Class 1, Category 3 items.

(d) Where the tensile strength, yield strength, tempering temperature or aging temperature listed in Table U-1610-1 differ from the requirements of the material specification, the minimum requirements listed in Table U-1610-1 shall apply.

U-1313 Special Requirements Conflicting With Permitted Material Specifications

(a) Special requirements stipulated in this Appendix shall apply in lieu of the requirements of the material specifications wherever the special requirements conflict with the material specification requirements. Where the special requirements include an examination, test, or treatment which is also required by the material specification, the examination, test, or treatment need be performed only once. Any required nondestructive examinations shall be performed as specified in U-1430. Any examination, repair, test, or treatment required by the material specification or this Appendix may be performed by the Material Manufacturer, Material Supplier, or the Certificate Holder manufacturing pumps. The Material Manufacturer or Material Supplier shall obtain approval from the Certificate Holder manufacturing pumps for the weld repair of materials (see U-1440).

(b) For materials listed in Table U-1610-1 for Category 3 and 4 items, the tensile test requirements of the material specification may be performed on representative samples of each heat of material used, for each specified heat treatment. The tensile strength and yield strength results shall satisfy the specified values and be below the

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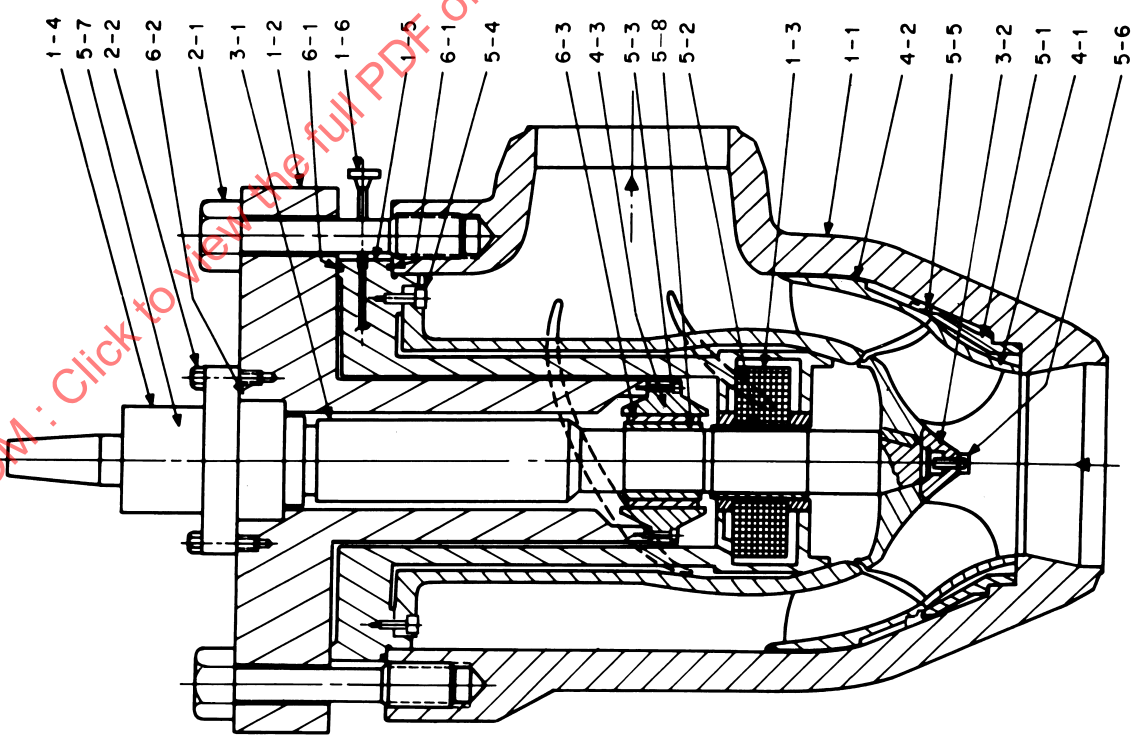
Figure U-1500-1
Typical for Type A, C, E, F, and/or Some J (NB-3400) Pumps



Key to Fig. U-1500-1

Cat.	Pump Item	Name (Typical)
1	1-1	Casing
	1-2	Main Flange
	1-3	Heat Exchanger
	1-4	Seal Housing
	1-5	Thermal Barrier
	1-6	TB Connection Piping (Include Flange)
2	2-1	Main Flange Bolts
	2-2	Seal Housing Bolts
3	3-1	Shaft
	3-2	Impeller Nut
4	4-1	Impeller
	4-2	Diffuser
	4-3	Bearing Support
5	5-1	Diffuser Adapter (Casing Wear Ring)
	5-2	Thermal Sleeve
	5-3	Radial Bearing Bolts
	5-4	Turning Vane/Thermal Barrier Bolts
	5-5	Diffuser/Diffuser Adapter Bolts
	5-6	Impeller Nut Locking Bolt
	5-7	Seals—Mechanical Seal (Metallic)
	5-8	Bearings/Journals and Shaft Sleeves
6	6-1	Flange Gaskets
	6-2	Seal Housing O-Ring
	6-3	Radial Bearing
	6-4	Mechanical Seal (Nonmetallic)

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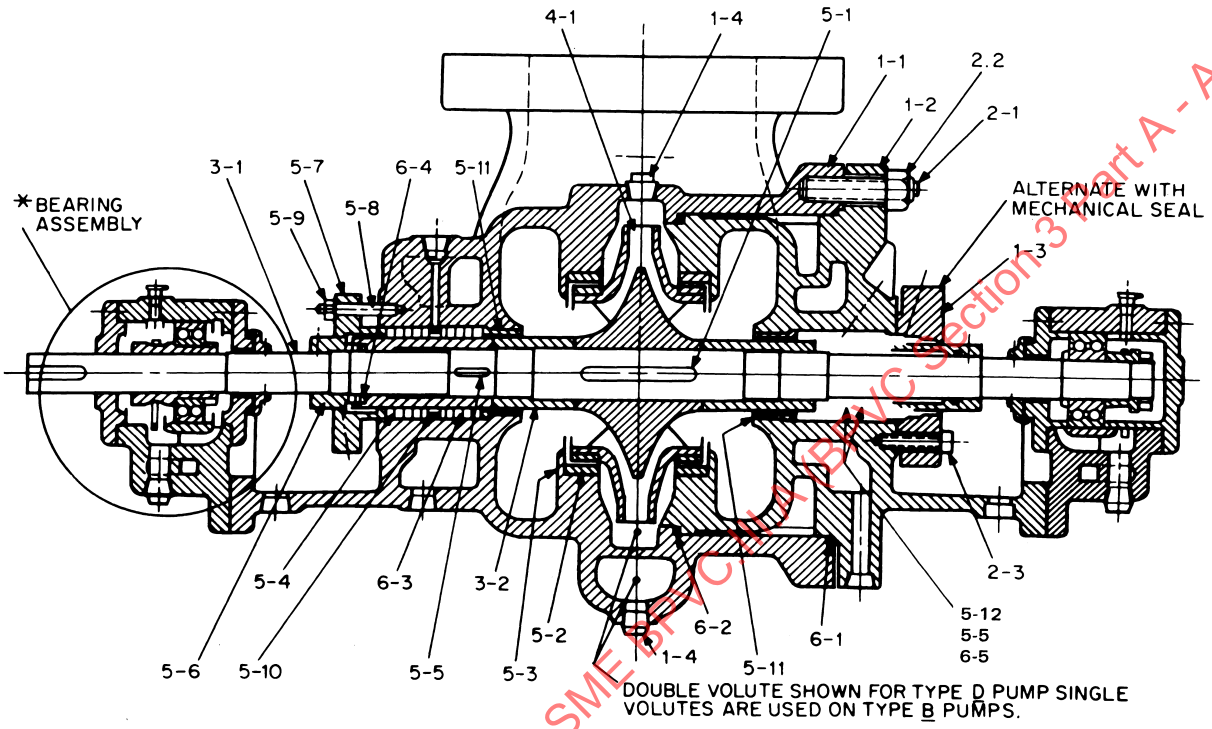
Figure U-1500-2
Typical for Type B and D Pumps (NC-3400 and ND-3400)



Key to Fig. U-1500-2

Cat.	Pump Item	Name (Typical)
1	1-1	Case
	1-2	Head
	1-3	Seal Plate
	1-4	Case Plug
2	2-1	Case Stud
	2-2	Case Stud Nut
	2-3	Seal Plate Bolt
3	3-1	Pump Shaft
	3-2	Impeller Locknut
4	4-1	Impeller
5	5-1	Impeller Key
	5-2	Case Wearing Ring
	5-3	Impeller Wearing Ring
	5-4	Shaft Sleeve
5	5-5	Shaft Sleeve Key
	5-6	Shaft Sleeve Nut
	5-7	Packing Gland
	5-8	Packing Gland Stud
	5-9	Packing Gland Nut
	5-10	Lantern Ring
	5-11	Packing Box Bushing
	5-12	Mech. Seal Metallic Parts
6	6-1	Head Gasket
	6-2	Inner Head Gasket
	6-3	Packing
	6-4	Shaft Sleeve Seal Ring
	6-5	Mech. Seal Nonmetallic Parts
*		Bearing Assembly

*Part not covered by this Appendix

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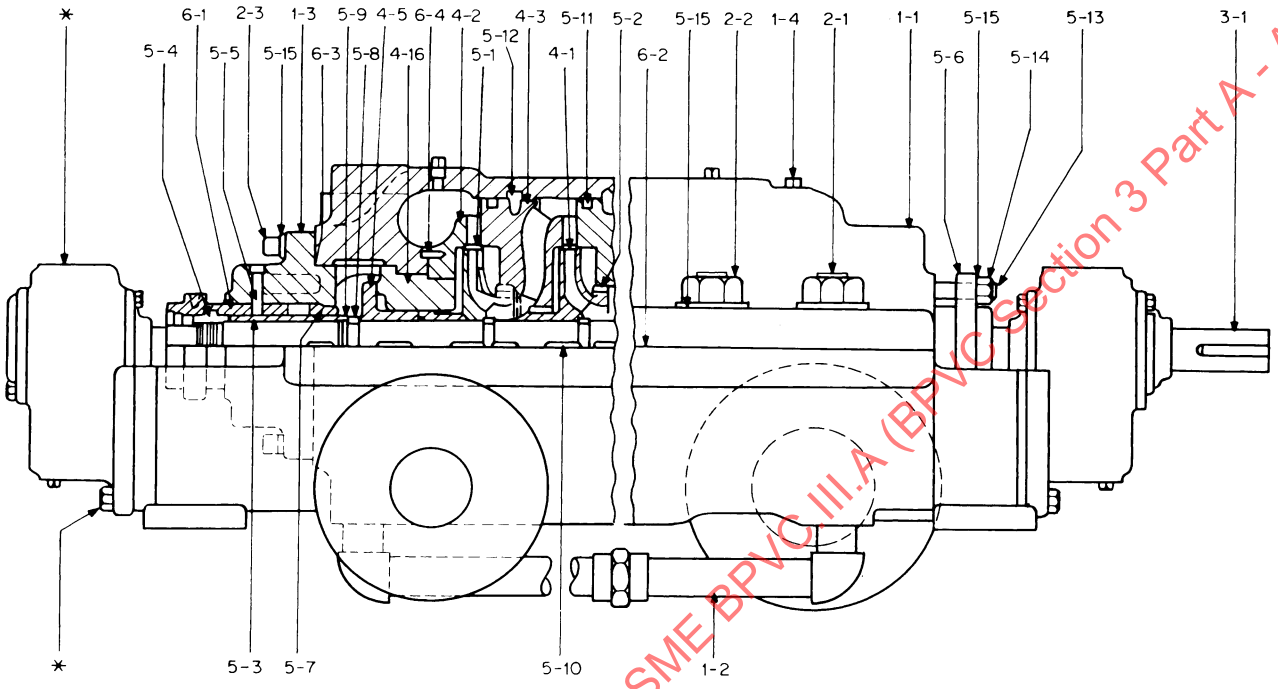
Figure U-1500-3
Typical for Type G and H Pumps (NC-3400 and ND-3400)



Key to Fig. U-1500-3

Cat.	Pump Item	Name (Typical)
1	1-1	Casing
	1-2	Piping, Balance Line (Including Fittings)
	1-3	Stuffing Box
	1-4	Plugs, Casing
2	2-1	Studs, Main Flange
	2-2	Nuts, Main Flange
	2-3	Cap Screws, Stuffing Box
3	3-1	Shaft
4	4-1	Impeller
	4-2	Diffuser
	4-3	Channel Ring
	4-4	Balance Sleeve
	4-5	Balance Drum
5	5-1	Impeller Ring
	5-2	Casing Ring
5	5-3	Shaft Sleeve
	5-4	Nut, Shaft Sleeve
	5-5	Seal Cage (Metallic)
	5-6	Gland Halves
	5-7	Bushing, Stuffing Box
	5-8	Split Ring
	5-9	Retaining Ring
	5-10	Key
	5-11	Sealing Ring, Channel
	5-12	Pin
	5-13	Studs, Gland
	5-14	Nut, Gland
	5-15	Washer
6	6-1	Packing, Stuffing Box
	6-2	Gasket, Main Flange
	6-3	Gasket, Metallic-Asbestos
	6-4	O-Ring

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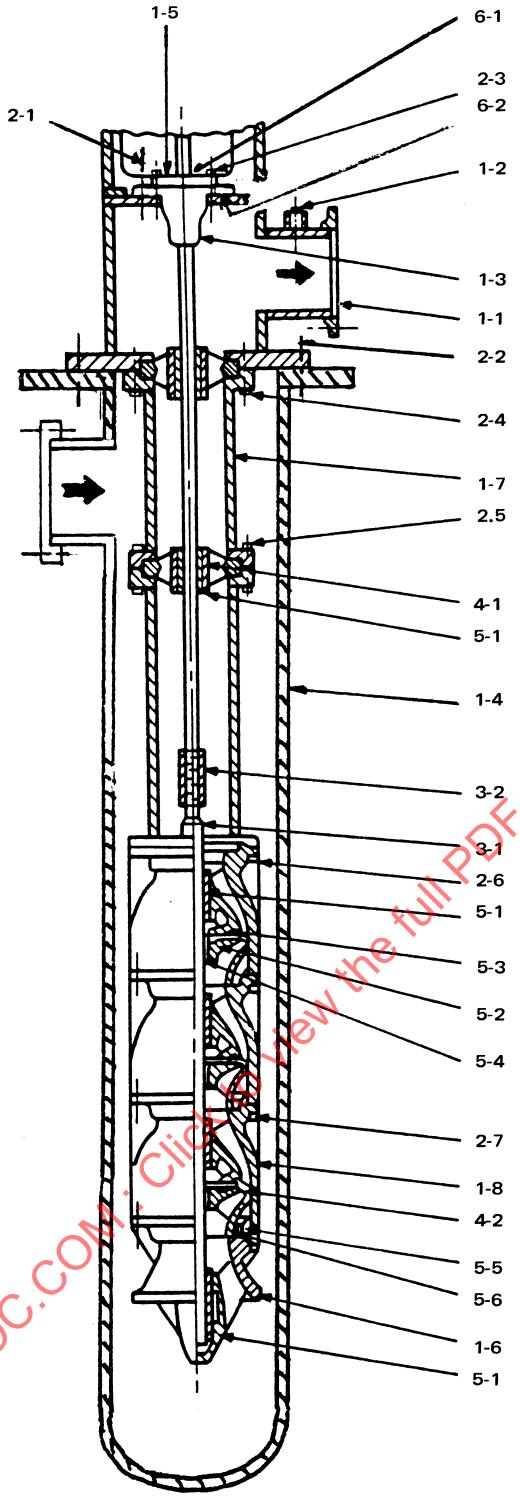
Figure U-1500-4
Typical for Type K Pumps (NC-3400 and ND-3400)



Key to Fig. U-1500-4

Cat.	Pump Item	Name (Typical)
1	1-1	Head
	1-2	Pipe Plug—Discharge
	1-3	Stuffing Box
	1-4	Barrel
	1-5	Packing Gland
	1-6	Suction Ball
	1-7	Column
	1-8	Bowl
2	2-1	Bolting—Stuffing Box to Head
	2-2	Bolting—Head to Barrel
	2-3	Bolting—Packing Gland to Stuffing Box
	2-4	Bolting—Column to Head
	2-5	Bolting—Column to Column
	2-6	Bolting—Bowl to Column
	2-7	Bolting—Bowl to Bowl
3	3-1	Shaft
	3-2	Coupling—Shaft
4	4-1	Spider
	4-2	Impeller
5	5-1	Bearing
	5-2	Split Ring—Thrust
	5-3	Capscrew—Split Ring
	5-4	Key—Impeller
	5-5	Wear Ring—Bowl
	5-6	Wear Ring—Impeller
6	6-1	Packing
	6-2	Gasket—Stuffing Box to Head

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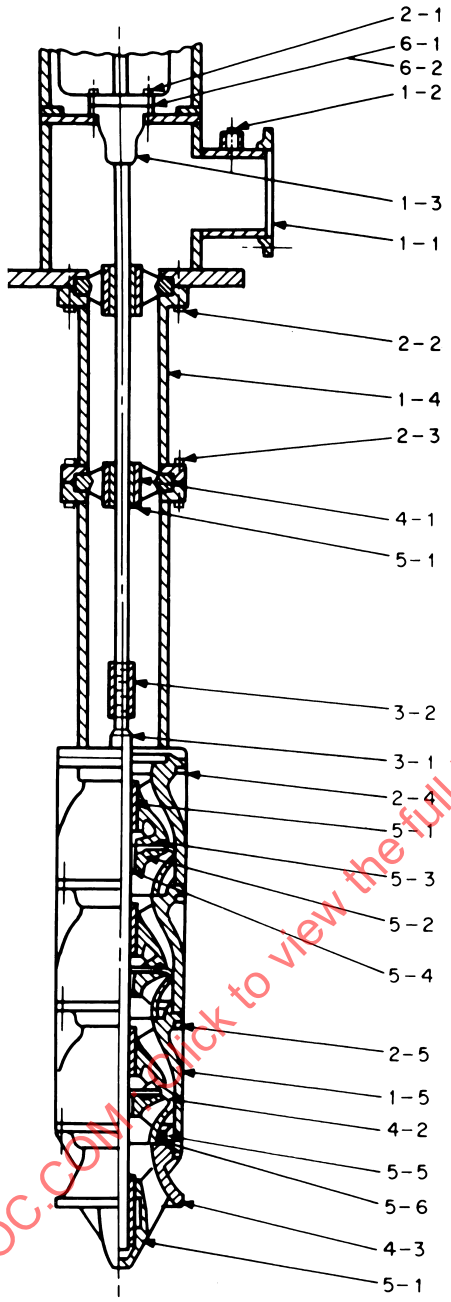
Figure U-1500-5
Typical for Type L Pumps (NC-3400 and ND-3400)



Key to Fig. U-1500-5

Cat.	Pump Item	Name (Typical)
1	1-1	Head
	1-2	Pipe Plug—Discharge
	1-3	Stuffing Box
	1-4	Column
	1-5	Bowl
2	2-1	Bolting—Stuffing Box to Head
	2-2	Bolting—Column to Head
	2-3	Bolting—Column to Column
	2-4	Bolting—Bowl to Column
	2-5	Bolting—Bowl to Bowl
3	3-1	Shaft
	3-2	Coupling—Shaft
4	4-1	Spider
	4-2	Impeller
	4-3	Suction Bell
5	5-1	Bearing
	5-2	Split Ring—Thrust
	5-3	Capscrew—Split Ring
	5-4	Key—Impeller
	5-5	Wear Ring—Bowl
	5-6	Wear Ring—Impeller
6	6-1	Packing
	6-2	Gasket—Stuffing to Head

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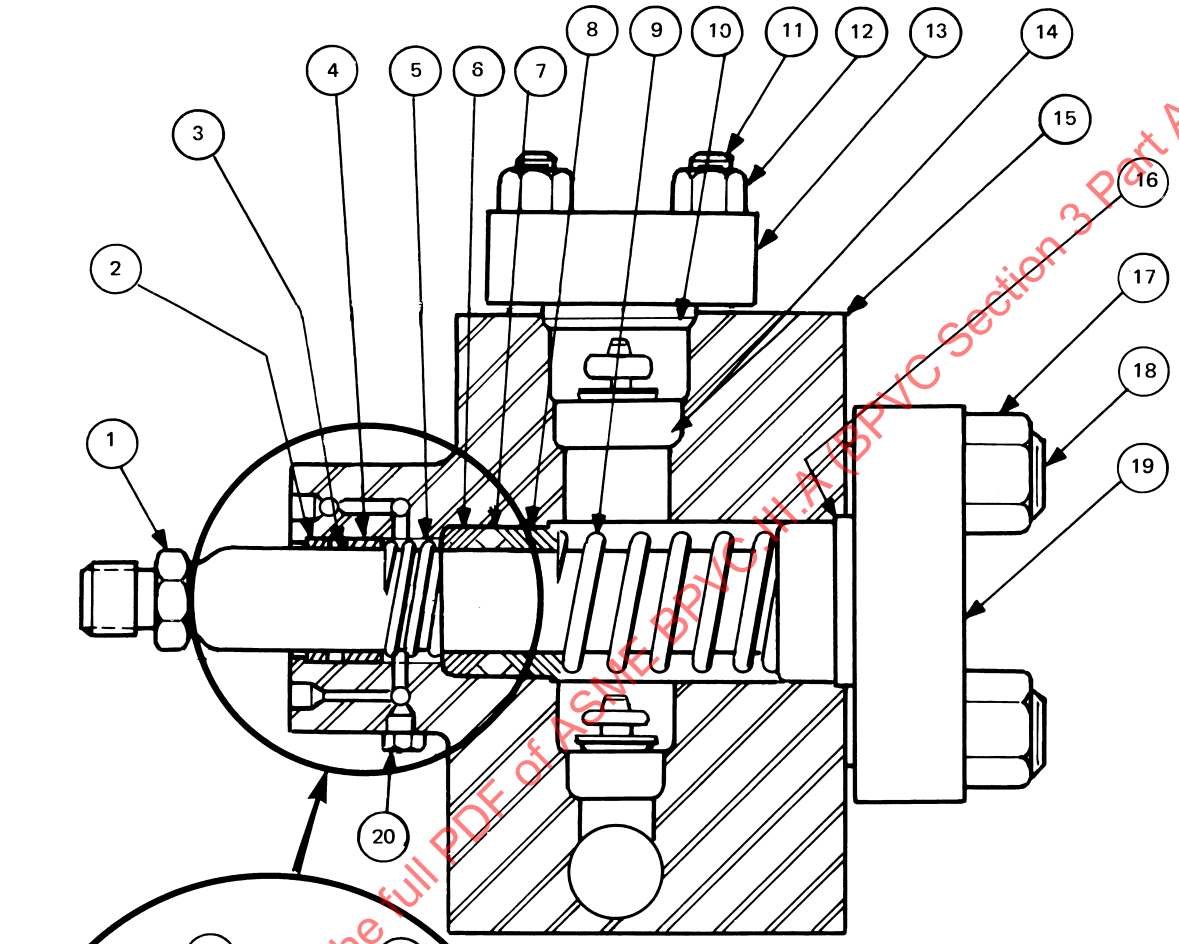
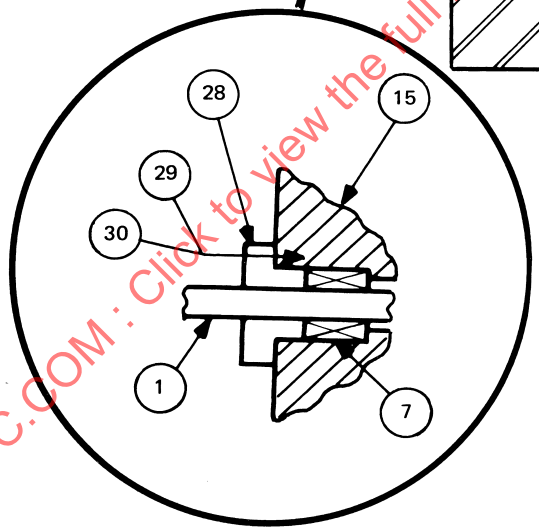
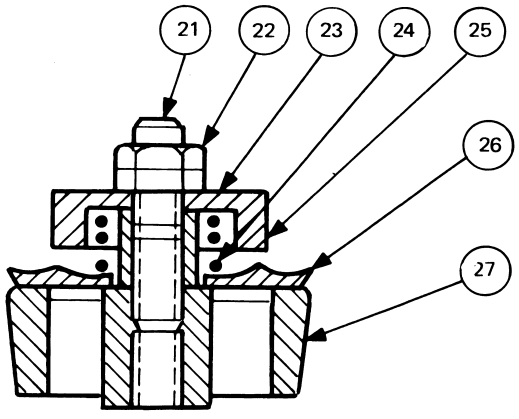
Figure U-1500-6
Reciprocating Plunger Pump (NC-3400 and ND-3400)





Alternate Arrangement
With Packing Gland



Valve Assembly

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Figure U-1500-6
Reciprocating Plunger Pump (NC-3400 and ND-3400) (Cont'd)

Key to Fig. U-1500-6

Cat.	Pump Item	Name (Typical)
1	13	Upper Cap
	15	Cylinder Block
	19	Front Cap
	28	Packing Gland
2	11	Upper Cap Stud
	12	Upper Cap Stud Nut
	17	Front Cap Stud Nut
	18	Front Cap Stud
	29	Stud (Gland)
	30	Nut (Gland)
3	1	Plunger
5	2	Packing Adj. Ring (Secondary)
	4	Plunger Ring (Secondary)
	5	Packing Spring (Secondary)
	6	Packing Adj. Ring (Primary)
	8	Plunger Ring (Primary)
	9	Packing Spring (Primary)
	14	Valve Assembly
	20	Plug
	21	Stud
	22	Double Locknut
	23	Sleeve
	24	Spring
	25	Guard Stop
	26	Valve
	27	Seat
6	3	Packing (Secondary)
	7	Packing (Primary)
	10	Upper Cap Gasket
	16	Front Cap Gasket

maximum specified values listed in Table U-1610-1. Where the material will be used to fabricate various item sizes in different heat-treated thicknesses, the manufacturer shall assure himself that the heat treatment specified will be effective for the entire size range.

U-1314 Certification of Materials

(a) Certification by Material Manufacturer. The Material Manufacturer shall provide a Certified Material Test Report for Category 3 items and Category 4, Class 1 impellers, including all welding and brazing materials used on these items. The Material Manufacturer shall certify that the contents of the report are correct and accurate, and that all operations performed by him or his subcontractors are in compliance with the requirements of the material specification and this Appendix. Alternatively, the Material Manufacturer shall provide a Certified Material Test Report for operations he performed and at least one Certified Material Test Report from each of his subcontractors for operations they performed. Chemical analysis, tests, examinations, and heat treatments required by the material specification that were not performed shall be listed on the Certified Material Test Report. A Material Manufacturer's Certificate of Compliance with the material specification, grade, class, and heat-treated condition, as applicable, may be provided in lieu of a Certified Material Test Report for material used for pumps with inlet connections 2 in. nominal pipe size and less, and bolting 1 in. nominal diameter and less. Material

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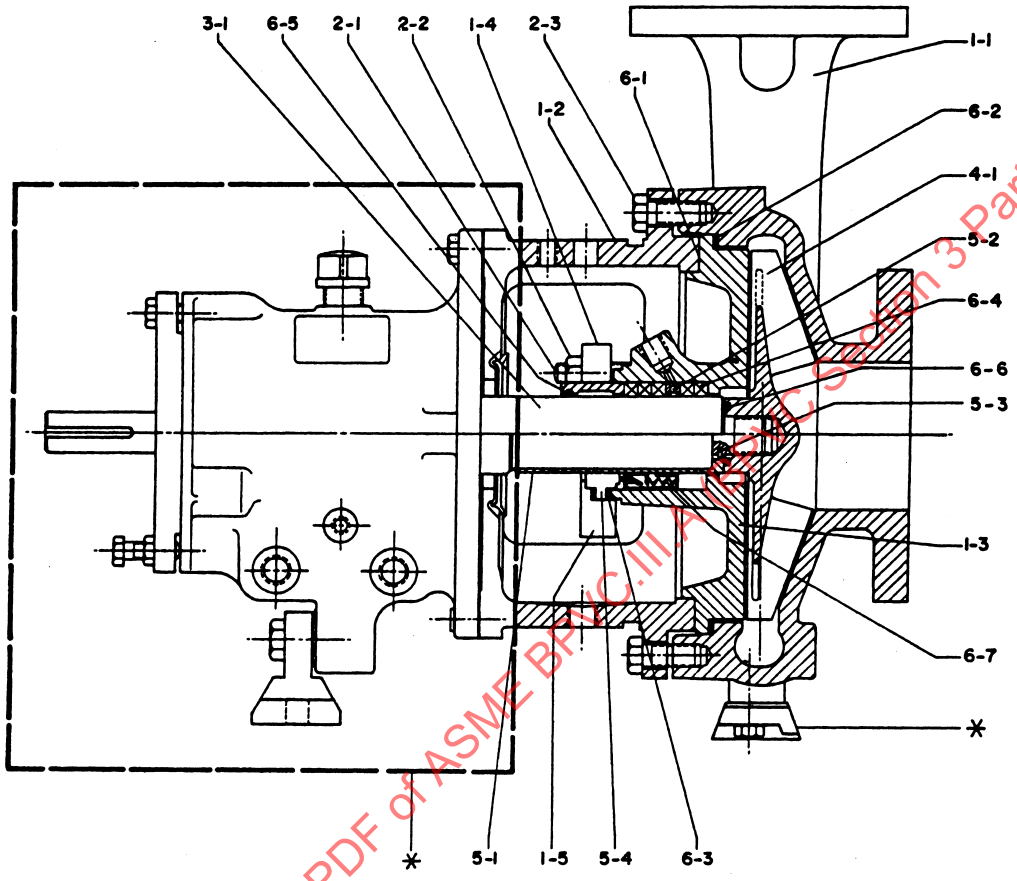
Figure U-1500-7
Typical for Type A and C Pumps (NC-3400 and ND-3400)



Key to Fig. U-1500-7

Cat.	Pump Item	Name (Typical)
1	1-1	Casing
	1-2	Frame Adapter
	1-3	Stuffing Box Cover
	1-4	Gland—Packing
	1-5	Gland—Mech. Seal
2	2-1	Stud-Gland
	2-2	Nut—Gland Stud
	2-3	Cap Screw—Case
3	3-1	Shaft
4	4-1	Impeller
5	5-1	Shaft Sleeve
	5-2	Lantern Ring
	5-3	Pin—Sleeve
	5-4	Mech. Seal (Metallic Parts)
6	6-1	Gasket—Adapter/Cover
	6-2	Gasket—Casing
	6-3	Gasket—Gland
	6-4	Packing—Stuffing Box
	6-5	Packing—Gland
	6-6	O-Ring—Shaft Sleeve
	6-7	Mech. Seal (Nonmetallic Parts)

*Part not covered by this Appendix

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Table U-1610-1
Materials for Pump Internal Items for Class 1, 2, and 3 Pumps

Material	Product Form	Spec. No. (8)	Type or Grade	Notes	Diameter or Thickness, in. (mm)
1Cr-0.2Mo	Bar, Rod	A-331	4140	(2)(5)	Up to 2 (50) incl.
	Bar, Rod	A-331	4340	(2)(5)	
1Cr-0.2Mo-0.3Pb	Bar, Rod	A-331	41L40	(2)(5)	
13Cr	FFBS	A-182	F6	(1)(4)(5)	Up to 8 (200) incl.
	Plate	A-240	410, 410S	(1)(4)(5)	Up to 8 (200) incl.
	Tube	A-268	TP410	(4)(5)	Up to 8 (200) incl.
	Bar, Shapes	A-276	403, 410	(4)(5)	
	Billets, Bars	A-314	403, 410	(4)(5)	Up to 8 (200) incl.
	Billets, Bars	A-314	416, 416Se	(2)(4)(5)	Up to 8 (200) incl.
	Castings	A-217	CA15	(4)(5)	
	Bar, Shapes	A-479	410	(4)(5)	Up to 8 (200) incl.
	Bars	A-582	416, 416Se	(2)(4)(5)	Up to 8 (200) incl.
	Bars	A-582	403, 410	(4)(5)	Up to 8 (200) incl.
	Bar, Shapes	A-276	414	(4)(5)	Up to 8 (200) incl.
	Castings	A-296	CA15	(4)(5)	
	Castings	A-296	CA40	(4)(5)	
	Castings	A-487	CA15M	(4)(5)(6)	
13Cr-4Ni	Castings	A-296	CA6NM		
	Castings	A-487	CA6NM		
13Cr	Bar, Shapes	A-276	420	(2)	
	Bar, Shapes	A-276	420	(1)2)(5)	
18Cr	Bar, Shapes	A-276	440A	(1)(2)(5)(7)	Up to 8 (200) incl.
	Bar, Shapes	A-276	440C	(1)(2)(5)(7)	Up to 8 (200) incl.
13Cr	Bar, Shapes	A-565	615	(2)(5)(7)	
	Bar, Shapes	A-565	616	(2)(5)(7)	
17Cr-4Ni-4Cu	Bar, Shapes	A-564	630	(1)(2)(4)(5)	8 (200) max
	Castings	AMS 5355B		(1)(2)(4)(5)	
		AMS 5398A		(2)(4)(5)	
				(2)(4)	
				(2)(4)	
15Cr-6Ni-Cu-Mo	Bar, Shapes	A-564	XM-25	(2)(5)	8 (200) max
	Bar, Shapes	A-564	XM-25	(2)	8 (200) max
	Bar, Shapes	A-564	XM-25	(2)	8 (200) max
25Ni-15Cr-2Ti	Bolting	A-453	660	(2)	
25Ni-15Cr-2Ti	Bar, Shapes	A-638	660	(2)(5)	
16Cr-12Ni-2Mo	Bar, Shapes	A-276	316	(5)	Up to $^{3}/_{4}$ (19) $^{3}/_{4}$ to 1 (19 to 25) 1 to $1^{1}/_{4}$ (25 to 32) $1^{1}/_{4}$ to $1^{1}/_{2}$ (32 to 38)
Phos-Bronze	Bar, Rod, Shapes	B-139	544		Up to 2 (50) incl.
Al-Bronze	Castings	B-148	955		
		B-148	952		
Pb-Sn-Bronze	Castings	B-584	932		
		B-584	937		
Cu-Sn-Pb-Zn	Castings	B-584	836		
Ni-Cr-Fe	Bar, Shapes	B637-84A	718	(2)(5)	4 (100) max
	Bar, Shapes	B637-84A	688 Type 2	(2)(5)	

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Table U-1610-1
Materials for Pump Internal Items for Class 1, 2, and 3 Pumps (Cont'd)

Cond.	Tensile Strength, psi (MPa), min	Yield Strength, psi (MPa), min	Minimum Tempering or Aging Temp., °F (°C)	Spec No.
HT	180,000 (1241)	155,000 (1069)	800 (425)	A-331
HT	178,000 (1227)	156,000 (1076)	1000 (540)	A-331
HT	180,000 (1241)	155,000 (1069)	800 (425)	A-331
	165,000 (1138)	132,000 (910)	900 (480)	A-182
	136,000 (938)	112,000 (772)	1025 (550)	A-240
	125,000 (862)	100,000 (690)	1075 (580)	A-268
	120,000 (827)	90,000 (620)	1100 (595)	A-276
	116,000 (800)	92,000 (634)	1125 (605)	A-314
	110,000 (758)	90,000 (620)	1175 (635)	A-314
	100,000 (690)	80,000 (552)	1275 (690)	A-217
	90,000 (620)	65,000 (448)	1375 (745)	A-479
				A-582
				A-582
				A-276
				A-296
				A-296
				A-487
	110,000 (758)	80,000 (552)	1100 (595)	A-296
	110,000 (758)	80,000 (552)	1100 (595)	A-487
A	95,000 (655)	50,000 (345)	NA (NA)	A-276
QT	250,000 (1724)	195,000 (1345)	750 (400)	A-276
HT	285,000 (1965)	275,000 (1896)	500 (260)	A-276
HT	285,000 (1965)	275,000 (1896)	600 (315)	A-276
HT	140,000 (965)	110,000 (758)	1150 (620)	A-565
HT	140,000 (965)	110,000 (758)	1150 (620)	A-565
PH	190,000 (1310)	170,000 (1172)	900 (480)	A-564
PH	155,000 (1069)	145,000 (1000)	1025 (550)	AMS 5355B
PH	145,000 (1000)	125,000 (862)	1075 (580)	AMS 5398A
PH	140,000 (965)	115,000 (793)	1100 (595)	
PH	135,000 (931)	105,000 (724)	1150 (620)	
PH	180,000 (1241)	170,000 (1172)	900 (480)	A-564
PH	125,000 (862)	75,000 (517)	1150 (620)	A-564
STr	125,000 (862)	95,000 (655)	NA (NA)	A-564
A or B	130,000 (896)	85,000 (586)	1325 (720)	A-453
1 or 2	130,000 (896)	85,000 (586)	1300 (700)	A-638
B	125,000 (862)	100,000 (690)	NA (NA)	A-276
B	115,000 (793)	80,000 (552)	NA (NA)	
B	105,000 (724)	65,000 (448)	NA (NA)	
B	100,000 (690)	50,000 (345)	NA (NA)	
Hard	50,000 (345)	35,000 (241)		B-139
	90,000 (620)	40,000 (276)	NA (NA)	B-148
	65,000 (448)	25,000 (172)	NA (NA)	B-148
	30,000 (207)	14,000 (97)	NA (NA)	B-584
	25,000 (172)	12,000 (83)	NA (NA)	B-584
	30,000 (207)	14,000 (97)	NA (NA)	B-584
PH	185,000 (1276)	150,000 (1034)	1325 (720)	B-637
PH	170,000 (1172)	115,000 (793)	1350 (730)	B-637

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Table U-1610-1
Materials for Pump Internal Items for Class 1, 2, and 3 Pumps (Cont'd)

Material	Product Form	Spec. No. (8)	Type or Grade	Notes	Diameter or Thickness, in. (mm)
Ni-Cu	Bar, Rod	B-164	400 Class A		
	Pipe, Tube	B-165	400		Up to 5 (125)
	Pipe, Tube	B-165	400	(2)	Over 5 (125)
	FFBS	AMS 4676	Alloy 500	(5)	12 (300) max
	Rounds	AMS 4676	Alloy 500	(5)	4 (100) max
Cr-Ni-Cu-Mo	Casting	A-351	CN7M		

TABLE U-1610-1
MATERIALS FOR PUMP INTERNAL ITEMS FOR CLASS 1, 2, AND 3 PUMPS (CONT'D)

Cond.	Tensile Strength, psi (MPa), min	Yield Strength, psi (MPa), min	Minimum Tempering or Aging Temp., °F (°C)	Spec No.
A	70,000 (483)	25,000 (172)	NA	B-164
A	70,000 (483)	28,000 (193)	NA	B-165
A	70,000 (483)	25,000 (172)	NA	B-165
PH	140,000 (965)	100,000 (690)		AMS 4676
PH	135,000 (931)	95,000 (655)		AMS 4676
SHT	62,000 (427)	25,000 (172)	NA	A-351

NOTES:
(1) Not to be used with Category 3 items, except for pumps with inlet piping connections NPS 2 (DN 50) and less.
(2) Welding of these materials is not permitted.
(3) Where the tensile strength, yield strength, tempering temperature, or aging temperature listed in Table U-1610-1 differ from the requirements of the material specification, the minimum requirements listed in Table U-1610-1 shall apply. The material shall be identified with this Appendix number in addition to the requirements for identification of para. U-1316.
(4) Cross-bracketing indicates that any of the bracketed materials may be used with any of the bracketed properties.
(5) The maximum tensile strength shall not exceed the minimum specified tensile strength listed in this Table by more than 40,000 psi (275 MPa).
(6) Welding of this material is permitted provided the carbon content is 0.05% or less.
(7) Service temperatures shall not exceed temperatures of 100°F (38°C) below the aging or tempering temperature.
(8) Material shall conform to Edition specified by N-Certificate Holder.

identification including any marking code (see U-1316) shall be described in the Certified Material Test Report or Certificate of Compliance as applicable.

(b) Certification by Material Supplier. The Material Supplier who completes any operation not performed by the Material Manufacturer shall provide a Certified Material Test Report for all operations performed by him. This certification affirms that the contents of the report are correct and accurate, and that all test results and operations performed by him or his subcontractors are in compliance with the material specification and the applicable material requirements of this Section as designated by the purchaser.

(c) Certified Material Test Reports or Material Manufacturer or Material Supplier Certificates of Compliance are not required for Category 5 and 6 items.

U-1315 Welding and Brazing Materials

All welding and brazing materials used on Category 3 and 4 items shall meet the requirements of NB-2400, NC-2400, or ND-2400, as applicable.

U-1316 Material Identification

U-1316.1 Class 1 and 2 Pump Items.

(a) The identification of Materials for Category 3 items used for Class 1 and 2 pumps shall consist of marking or tagging the material with the applicable specification number, grade, heat number or heat code, and any additional marking required to facilitate traceability of the reports of the results of all tests and examinations performed on the material, except that heat number identification is not required for pumps with inlet connections 2 in. nominal pipe size (DN 50) and less. Alternatively, a Certification Mark and/or code may be used which identifies the material with the Materials Certification and such Certification Mark and/or code shall be explained in the certificate (see U-1314). For identification and marking during fabrication by the Pump Manufacturer, see U-1420.

(b) The identification of Materials for Category 4 items used for Class 1 and 2 pumps shall consist of marking or tagging the material or its container in accordance with

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the marking requirements of the applicable material specification. Category 5 items shall be identified as set forth in the Manufacturer's Quality System Program.

(c) Materials may be marked by any method which will not result in any harmful contamination or sharp discontinuities. Stamping, when used, shall be done with blunt-nosed-continuous or blunt-nosed-interrupted-dot die stamps.

U-1316.2 Class 3 Pump Items. The identification of materials for Category 3 through 5 items used for Class 3 pumps shall consist of marking the material or its container in accordance with the requirements of the Manufacturer's Quality System Program.

U-1316.3 Welding and Brazing Material Identification. Welding and brazing materials shall be clearly identified by legible marking on the package or container so that they are identifiable as acceptable material until the material is actually consumed in the process.

U-1320 FRACTURE TOUGHNESS REQUIREMENTS FOR CATEGORY 3 MATERIALS

U-1321 Materials to Be Impact Tested

Impact tested specimens shall be representative of the final heat treatment of the finished part.

U-1321.1 Materials for Which Impact Testing Is Required. Materials for Category 3 items for Class 1 pumps, and for Class 2 and 3 pumps when required by the Design Specification, shall be impact tested in accordance with the requirements of U-1320 except that the following materials do not require impact testing:

(a) all thickness of materials for pumps with a nominal inlet pipe size 6 in. (150 mm) diameter and smaller

(b) materials for pumps with all pipe connections of $^{5}/_{8}$ in. (16 mm) nominal pipe wall thickness and less

(c) materials with a nominal section thickness of $^{5}/_{8}$ in. (16 mm) and less

(d) bars with a nominal cross-sectional area of 1 in.^{2} (650 mm^2) and less

(e) austenitic stainless steels

(f) nonferrous materials

U-1321.2 Impact Test Procedure.

U-1321.2.1 Charpy V-Notch Tests. The Charpy V-Notch Test shall be performed in accordance with SA-370. Specimens shall be in accordance with SA-370, Figure 11, Type A. A test shall consist of a set of 3 full-size 10 × 10 mm specimens. The test temperature and lateral expansion shall be reported in the Certified Material Test Report.

U-1321.2.2 Location and Orientation of Test Specimens. Impact test specimens shall be removed from the locations and orientations specified by the material specification for tensile test specimens in each product form.

U-1321.2.3 Test Requirements and Acceptance Standards. Three Charpy V-Notch specimens shall be tested at a temperature equal to or lower than the lowest service temperature. All three specimens shall meet or exceed 15 mils (0.45 mm) lateral expansion. Lowest service temperature is the minimum temperature of the fluid retained by the pump. The lowest service temperature shall be specified in the Design Specification.

U-1321.2.4 Retests. One retest at the same temperature may be conducted provided

(a) not more than one specimen per test is below the minimum requirements, and

(b) the specimen not meeting the minimum requirements is not lower than 5 mils (0.13 mm) below the specified requirements

U-1400 FABRICATION REQUIREMENTS

Category 3 through 5 items shall be fabricated in accordance with the requirements of U-1400 and shall be manufactured from materials which meet the requirements of U-1300.

U-1410 CERTIFICATION OF MATERIALS AND FABRICATION BY PUMP MANUFACTURER

The pump manufacturer shall provide certification that all treatments, tests, repairs, or examinations performed on pump items are in compliance with the requirements of this Appendix. Reports of all required treatments and the results of all required tests, repairs, and examinations performed shall be maintained in accordance with NCA-3862.1 or NCA-4134.17.

U-1420 MATERIAL IDENTIFICATION

Material for Category 3 and 4 items for Class 1 and 2 pumps shall carry identification markings, including heat treatment grade either directly on the item or on a separate tag that accompanies the item, which will be maintained during and after assembly.

U-1430 EXAMINATION OF MATERIALS

Bars and forgings for Category 3 pump shafting for Class 1 pumps, shall be ultrasonically examined in accordance with NB-2542 and NB-2547. Materials for Class 1 pumps, Category 3 items and Category 4 impellers for Class 1 pumps shall be examined on all external and accessible internal surfaces by the magnetic particle or liquid penetrant method in accordance with Section V. The examination may be performed by the Material Manufacturer, Material Supplier, or the pump manufacturer (see U-1313). Acceptance standards for magnetic particle and/or liquid penetrant examination shall be as follows.

(a) Only indications with major dimensions greater than $^{1}/_{16}$ in. (1.5 mm) shall be considered relevant.

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(b) The following relevant indications are unacceptable:

(1) any linear indications greater than $^{1}/_{16}$ in. (1.5 mm) long for materials less than $^{5}/_{8}$ in. (16 mm) thick; greater than $^{1}/_{8}$ in. (3 mm) long for materials from $^{5}/_{8}$ in. (16 mm) thick to under 2 in. (50 mm) thick; and $^{3}/_{16}$ in. (5 mm) long for materials 2 in. (50 mm) thick and greater;

(2) rounded indications with dimensions greater than $^{1}/_{8}$ in. (3 mm) for thicknesses less than $^{5}/_{8}$ in. (16 mm) and greater than $^{3}/_{16}$ in. (5 mm) for thicknesses $^{5}/_{8}$ in. (16 mm) and greater;

(3) four or more indications greater than $^{1}/_{16}$ in. (1.5 mm) in a line separated by $^{1}/_{16}$ in. (1.5 mm) or less edge to edge;

(4) ten or more indications greater than $^{1}/_{16}$ in. (1.5 mm) in any 6 in.^{2} (4000 mm^2) of area whose major dimension is no more than 6 in. (150 mm) with the dimensions taken in the most unfavorable location relative to the indications being evaluated;

(5) linear nonaxial indications.

Materials for Category 4 and 5 items for Class 1 pumps and for Category 3, 4, and 5 items for Class 2 and 3 pumps shall be examined in accordance with the material specification. When Category 4 or 5 material forms an integral or welded extensions to Category 1 material (see U-1120), the examination required for the Category 4 or 5 portion shall be in accordance with this Appendix. In addition, the requirements for the examination of the Category 1 portion shall be applied for a distance of at least 2t (t is thickness of the Category 4 or 5 material) from the Category 1 boundary as defined by the pump designer.

U-1431 Time of Examination

Magnetic particle or liquid penetrant examination shall be performed in the finished conditions, after all heat treatment operations and postweld heat treatment, except that threaded items may be examined prior to threading. Examinations shall be performed prior to any coating or plating. Lapping of seating surfaces to reduce leakage or lapping of bearing surfaces shall not require reexamination.

U-1432 Elimination of Surface Defects

(a) Unacceptable surface defects shall be removed by grinding or machining, provided:

(1) the remaining thickness of the section is not reduced below the minimum required by the design;

(2) the depression, after grinding or machining, is blended uniformly into the surrounding surface and the depression does not affect the function of the item;

(3) after grinding or machining, the area is examined by the method which originally disclosed the defect to assure that the defect has been removed or the indication reduced to an acceptable size.

(b) If grinding or machining reduces the thickness of the section below the minimum required by the design, the item may be repaired and returned to an acceptable size.

U-1440 REPAIR BY WELDING OF CATEGORY 3, 4, AND 5 PUMP ITEMS

(a) Category 3, 4, and 5 items for Class 1, 2, and 3 pumps may be repaired by welding using the provisions of Section IX for materials in Table U-1610-1 with assigned P-numbers provided the requirements of the following subparagraphs are met.

(b) Until such time as P-numbers are assigned, welding of those materials without P-numbers shall be separately qualified as required by Section IX.

(c) Bolts, studs, nuts, and material for which welding is prohibited by Note (2) of Table U-1610-1 shall not be repair welded.

U-1441 Defect Removal

The defect shall be removed or reduced to an acceptable size by suitable mechanical or thermal cutting or gouging methods and the cavity prepared for repair.

U-1442 Qualification of Welding Procedures and Welders

(a) When impact tests of Category 3 items are required, the impact testing requirements of U-1321.2.3 shall be met in the heat-affected zone regardless of filler metal used and in the weld metal for all material except austenitic and nonferrous filler metal.

(b) Except as permitted in (c) below, the welding procedure and welders or welding operators shall be qualified in accordance with Section IX.

(c) Heat-treated materials listed in Table U-1610-1 which are not capable of passing bend tests required by Section IX for procedure or performance qualification may be qualified by a Fillet Weld Test in accordance with Section IX, QW-180. In addition, a minimum of two cross sections of the qualification test plate (assembly) shall be ground and etched with a suitable etchant and visually examined at 10× magnification. The weld metal and adjacent base material of the ground and etched cross sections shall be free of cracks.

U-1443 Blending of Repaired Areas

After repair, the surface shall be blended into the surrounding surface.

U-1444 Examination of Repair Welds

Each repair weld of materials for Category 3 items for Class 1 pumps shall be examined by the method that originally disclosed the defect. The finished surface shall be examined by either the magnetic particle or liquid penetrant method in accordance with Section V. The acceptance standards shall be those specified in U-1430(a)

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and U-1430(b). Repair welds of materials for the other categories shall be in accordance with the material specification.

U-1445 Heat Treatment After Repair

(a) Materials listed in Table U-1610-1 which are repaired by welding shall be heat treated in accordance with the requirements of NB-4600, NC-4600, and ND-4600 as applicable, or as allowed by (c) below.

(b) Materials listed in Table U-1610-1 which are repaired by welding shall be heat treated and tempered or aged after repair, except as allowed by (c) below. The minimum tempering or aging temperature shall be as specified in Table U-1610-1 for the finished item.

(c) Repair weld procedures for welds not exceeding the lesser of $^{3}/_{8}$ in. (10 mm) or 10% of the section thickness shall be qualified by test weld specimens meeting the tensile and bend test requirements of Section IX, without PWHT, for the material repaired.

U-1446 Repair Weld Report

A record shall be made of each defect repair of Category 3 items for Class 1 pumps in which the depth of the repair cavity exceeds the lesser of $^{3}/_{8}$ in. (10 mm) or 10% of the section thickness. The record shall include the location and size of the repaired cavity, the welding material, the welding procedure, the heat treatment, and the examination results.

U-1450 WELDING REQUIREMENTS

Except as permitted in U-1442 and U-1451, all welds shall be made using qualified welding procedures and welders or welding operators in accordance with Section IX.

U-1451 Special Fabrication Welds

Fillet welds and partial penetration welds $^{1}/_{4}$ in. (6 mm) and less in size may be made in the fabrication of pump items or between pump items where either of the items is a material listed in Table U-1610-1 provided welding is not prohibited by Note (2) in Table U-1610-1, and the procedures and welders are qualified as follows:

(a) a test assembly shall be made for each combination of materials to be welded.

(b) the test assembly shall be a duplicate of the production weld joint or a groove butt weld $^{1}/_{4}$ in. (6 mm) minimum thickness.

(c) the test assembly shall be sectioned (a minimum of four cross sections), ground, etched with a suitable etchant, and visually examined at 10× magnification. All surfaces of the weld and adjacent base material(s) shall be free of cracks.

U-1452 Examination of Welds

All welds shall be examined by the magnetic particle or liquid penetrant method in accordance with Section V. The time of examination shall be in accordance with NB-5120, NC-5120 and ND-5120. Acceptance standards shall be as follows.

U-1453 Acceptance Standards

(a) Only indications with major dimensions greater than $^{1}/_{16}$ in. (1.5 mm) shall be considered relevant.

(b) The following relevant indications are unacceptable:

(1) any cracks or linear indications

(2) rounded indications with dimensions greater than $^{3}/_{16}$ in. (5 mm)

(3) four or more rounded indications in a line separated by $^{1}/_{16}$ in. (1.5 mm) or less edge to edge

(4) ten or more rounded indications in any 6 in.^{2} (4 000 mm^2) of surface with the major dimensions of this area not to exceed 6 in. (150 mm), with the area taken in the most unfavorable location relative to the indications being evaluated

U-1454 Heat Treatment of Welds

(a) Postweld heat treatment of welds which join materials listed in Table U-1610-1 shall be in accordance with the postweld heat treatment requirements of NB-4620, NC-4620 or ND-4620, as applicable.

(b) Postweld heat treatment of welds which join materials listed in Table U-1610-1 shall be in accordance with the postweld heat treatment requirements of NB-4620, NC-4620, or ND-4620, as applicable. Special techniques, such as local postweld heat treatment, may be necessary to avoid changing the base material properties of the item in location not adjacent to the weld. A change in the specified postweld heat treatment temperature will require requalification of the WPS in accordance with Section IX, QW-407.1.

(c) Postweld heat treatment of welds which join materials listed in Table U-1610-1 to materials listed in Section II, Part D, Table 1A shall be in accordance with the postweld heat treatment requirements of NB-4620, NC-4620, or ND-4620, as applicable. Special techniques, such as local postweld heat treatment, may be necessary to avoid changing the base material properties of the item in location not adjacent to the weld. A change in the specified postweld heat treatment temperature will require requalification of the WPS in accordance with Section IX, QW-407.1.

(d) Postweld heat treatment of welds for joining materials listed in Table U-1610-1 shall be in accordance with the heat treatment specified for the material of the finished item, i.e., the heat treatment required to obtain the tensile strength and/or yield strength listed in Table U-1610-1.

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(e) Materials listed in Table U-1610-1 subject to material specification heat treatment after repair welding shall be welded to a procedure which shall demonstrate that the required strength can be met in the weld without affecting the properties of the base material.

(f) For fillet welds and partial penetration welds $^{1}/_{4}$ in. (6 mm) and less in size, postweld heat treatment is neither required nor prohibited, provided the requirements of U-1451 are met.

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NONMANDATORY APPENDIX W

ARTICLE W-1000
ENVIRONMENTAL EFFECTS ON COMPONENTS

(15) W-1100 INTRODUCTION

Design Specification preparers and component designers must consider environmental degradation factors to eliminate, reduce, mitigate, or delay adverse environmental effects on Code components to an appropriate level by applying experience, research, and conservative design principles in the design and appropriate selection of materials and manufacturing practices. Owners must consider environmental degradation in defining appropriate inservice inspection, repair and replacement, and life extension programs.

This Appendix is organized by degradation mechanism and includes the following:

(a) the degradation mechanism description (refs. [1] through [4])

(b) the materials susceptible to the degradation mechanism

(c) the design measures to avoid component degradation

(d) the mitigating actions, such as material changes, stress mitigation, and coatings

The references cited in this Appendix cover mitigation strategies for long-term operation considering the interactions between materials, environments, stress, irradiation, etc. The Electric Power Research Institute (EPRI) Issue Management Tables and Materials Degradation Matrix (MDM) provide a prioritized comprehensive review of degradation mechanisms and possible mitigation actions. Other technology-specific reports are listed that supplement the EPRI and Nuclear Regulatory Commission (NRC) reports.

W-1110 SCOPE

This Appendix is intended to provide information related to degradation mechanisms experienced by operating plants but is not all inclusive. The Design Specification for new components must include appropriate requirements. Each degradation mechanism is described in brief terms; refs. [1] through [7], [34], and [35] provide additional information that should be consulted for particular component design problems.

W-1120 MATERIALS

This Appendix is not intended to prescribe or prevent the use of any material provided there is due consideration of environmental conditions, and it includes examples of materials that are expected to perform acceptably in the environments to be found in new light water reactors (LWRs) (see refs. [18] through [21]). Materials to avoid based on their history of performance are also highlighted or referenced.

W-1130 DESIGN

Design measures to cope with adverse environmental conditions include

(a) minimizing thermal stratification at nozzles

(b) minimizing stagnant dead-legs, crevices, socket welds, and sharp bends

(c) designing for complete drainage, layup, and nitrogen purge

(d) using full-penetration welds

(e) reducing bolt yield strength [less than approximately 120 ksi (827 MPa)] to reduce intergranular stress corrosion cracking (IGSCC)

(f) minimizing applied loads, peak stresses, and residual stresses

(g) specifying fabrication methods and controls to minimize residual strains and stresses at wetted surfaces

W-1140 MITIGATING ACTIONS

Environmental effect-mitigating actions may include changing materials, the residual stress state, and/or the environmental conditions; actions specific to the degradation mechanism; and the following general mitigating actions:

(a) using appropriate welding techniques, including minimizing the heat input from welding and use of pre- and postweld heating, where appropriate

(b) minimizing cold work, such as that produced by heavy grinding, or following cold work with corrective procedures, such as solution heat treatment or surface treatments (e.g., properly qualified peening)

(c) maintaining cleanliness throughout manufacturing

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(d) controlling water chemistry and including additives to reduce electrochemical potential (refs. [11] through [18])

(e) using clean hydrostatic test fluid to prevent microbiologically influenced corrosion (MIC) and IGSCC

(f) applying solution heat treatment to stainless steel welds

(g) using weld inlays, onlays, and overlays

(h) using mechanical stress improvement

(i) using electropolishing

(j) using passivation or preconditioning to establish a corrosion protective layer

(k) avoiding weld repairs that leave tensile stress on a wetted surface

W-1150 RELATION TO SECTION XI AND PLEX APPLICATIONS

Consideration of damage mechanisms and the service-related degradation of components should be considered when establishing Section XI inservice examination and surveillance programs and in projecting plant life extension limits (refs. [7] through [10]). Environmental conditions that exist during activities such as layup, cleaning, surveillance, maintenance, and repair can affect component integrity, service life, and plant life extension (PLEX) (refs. [1] through [19]). This Appendix addresses new component design, long-term operation (LTO), and PLEX for existing components. The GALL report (ref. [4]) is an extensive evaluation of existing components. Risk-informed inservice inspection (RI-ISI) methodology has been developed for existing LWRs to identify high safety-significant inspection elements that are subject to environmental degradation; this is informative for new component design (see refs. [28] through [31]; also, see ref. [32] for enhanced screening criteria for crevice corrosion). Component degradation mechanisms, mitigation, and research are further described in the Issue Management Tables for the respective technology (e.g., refs. [5] and [6]). Additional areas of concern may emerge in the future, e.g., in research reports, NRC generic letters and bulletins, and other industry event reports. It is the designer's responsibility to take such new information into consideration.

W-1160 REFERENCES

The following references apply to new component design and may also be applicable for repair and/or replacement activities performed under Section XI. Each Article in this Appendix lists additional references specifically applicable to that degradation mechanism. These references consist of extensive current and historical information.

[1] NUREG/CR-6923, Expert Panel Report on Proactive Materials Degradation Assessment, 2007.

[2] NEI 03-08, Rev. 2, Guideline for the Management of Materials Issues, 2010.

[3] EPRI Materials Degradation Matrix, Rev. 3, EPRI 3002000628, 2013.

[4] NUREG-1801, Rev. 2, Generic Aging Lessons Learned (GALL) Final Report, 2010.

[5] Materials Reliability Program: Pressurized Water Reactor Issue Management Tables, Rev. 3 (MRP-205), EPRI 3002000634, 2013.

[6] BWRVIP-167NP, Rev. 3: BWR Vessel and Internals Project Boiling Water Reactor Issue Management Tables, EPRI 3002000690, 2013.

[7] Advanced Nuclear Technology: Near-Term Deployment of Advanced Light Water Reactors, EPRI 1024870, 2012.

[8] Life Cycle Management Sourcebook for Nuclear Plant Service Water Systems, EPRI 1008282, 2005.

[9] NUREG-1061, Vol. 3, Evaluation of Potential for Pipe Breaks, 1984.

[10] NUREG-0313, Rev. 2, Technical Report on Material Selection and Processing Guidelines for BWR Coolant Pressure Boundary Piping, 1987.

[11] PWR Primary Water Chemistry Guidelines, Vol. 1, Rev. 4, EPRI TR-105714, 1999.

[12] PWR Secondary Water Chemistry Guidelines, Rev. 5, EPRI TR-102134, 2000.

[13] Pressurized Water Reactor Secondary Water Chemistry Guidelines, Rev. 6, EPRI 1008224, 2004.

[14] BWRVIP-130: BWR Vessel and Internals Project, BWR Water Chemistry Guidelines, EPRI TR-1008192, 2004.

[15] Closed Cooling Water Chemistry Guideline (Rev. 1 to TR-107396, Closed Cooling Water Chemistry Guideline), EPRI TR-1007820, 2004.

[16] Boric Acid Corrosion Guidebook: Managing Boric Acid Corrosion Issues at PWR Power Stations, Rev. 1, EPRI 1000975, 2001.

[17] Materials Reliability Program, Reactor Vessel Head Boric Acid Corrosion Testing (MRP-165): Task 3 — Separate Effects Testing, EPRI 1011807, 2005.

[18] Materials Reliability Program: Pressurized Water Reactor Internals Inspection and Evaluation Guidelines (MPR-227-A), Technical Report, EPRI 1022863, 2005.

[19] EPRI Materials Management Matrix Project: Advanced Light Water Reactor — Boiling Water Reactor Degradation Matrix (ALWR BWR DM), Rev. 0, EPRI 1019611, 2009.

[20] EPRI Materials Management Matrix Project: KHNP Advanced Pressurized Water Reactor (APR1400) Materials Management Tables, Rev. 0, EPRI 1024568, 2011.

[21] Fontana, M. G., and Greene, N. D. *Corrosion Engineering*, 3rd Ed., McGraw-Hill, 1986.

[22] Uhlig, H. H. *Corrosion and Corrosion Control*, 3rd Ed., Wiley & Sons, Inc., 2011.

[23] *ASM Handbook*, Volume 13A, "Corrosion: Fundamentals, Testing, and Protection," 2003; Volume 13B, "Corrosion: Materials," 2005; Volume 13C, "Corrosion: Environments and Industries," 2006, American Society for Metals (ASM) International.

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[24] Carbon Steel Handbook, EPRI 1014670, 2007.
[25] Landrum, R. J. *Fundamentals of Designing for Corrosion Control: A Corrosion Aid for the Designer*, ISBN: 0915567342, NACE International, 1992.
[26] Meyers, M. A., and Chawla, K. K. *Mechanical Metallurgy: Principles and Application*, 1st Ed., Prentice-Hall Inc., 1983.
[27] K. R. Rao. *Companion Guide to the ASME Boiler and Pressure Vessel Code*, 4th Ed., The American Society of Mechanical Engineers (ASME), 2012.
[28] Code Case N-577-1, Risk-Informed Requirements for Class 1, 2, or 3 Piping, Method A, Section XI, Division 1, 2000.
[29] Code Case N-578-1, Risk-Informed Requirements for Class 1, 2, or 3 Piping, Method B, Section XI, Division 1, 2000.
[30] Code Case N-560, Alternative Examination Requirements for Class 1, Category B-J Piping Welds, Section XI, Division 1, 2000.
[31] Code Case N-716, Alternative Piping Classification and Examination Requirements, Section XI, Division 1, 2006.
[32] Enhanced Crevice Corrosion Criteria in RI-ISI Evaluations, EPRI 1011945, 2005.
[33] Boiling Water Reactor Chemistry Performance Monitoring Report, 2005 Ed., EPRI 1009932, 2005.
[34] ASME BPVC Section II, Nonmandatory Appendix A, Issues Associated With Materials Used in ASME Code Construction.
[35] Approaches to Ageing Management for Nuclear Power Plants: International Generic Ageing Lessons Learned (IGALL) Final Report, IAEA-TECDOC-1736, 2014.

W-1200 SECTION XI AND PLEX APPLICATIONS

Along with providing guidance and a starting point for ASME Section III material and design considerations to minimize service degradation, this report is relevant to ASME Section XI and plant life extension (PLEX) evaluations. Consideration of these damage mechanisms and the service degradation of components should be a part of establishing Section XI in-service inspection (ISI) and surveillance programs, and in projecting plant life extension limits.

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(15)

ARTICLE W-2000
SUMMARIES OF CORROSION DAMAGE MECHANISMS

W-2100 STRESS CORROSION CRACKING

W-2110 GENERAL DESCRIPTION

Stress corrosion cracking (SCC) includes intergranular, transgranular, irradiation-assisted, strain-induced, and hydrogen embrittlement mechanisms (ref. [42]). Several of these mechanisms are discussed separately in this Appendix. SCC can lead to leakage or component failure in some materials in a light water reactor (LWR) environment. A combination of the following conditions is required for initiation of SCC:

(a) a susceptible material

(b) a suitable environment

(c) a high surface tensile stress [As little as 5 ksi (34.5 MPa) in the presence of chlorides may cause SCC of susceptible types of stainless steel.]

SCC has been experienced in all pressurized water reactors (PWRs) and boiler water reactors (BWRs) (ref. [1]). SCC of Alloy 600 steam generator tubing has resulted in replacement of most original steam generators in the PWR fleet (refs. [2] through [4]). Similarly, SCC of Alloy 600 control rod drive mechanism nozzles in PWR reactor vessel heads has resulted in augmented inspections and replacement of reactor vessel heads. BWR austenitic stainless steel recirculation piping and reactor vessel internals have been subject to SCC, which resulted in extensive inspections and repairs, and sometimes complete piping replacements (refs. [5] and [6]; see also W-2500).

W-2120 PREVENTIVE MEASURES

SCC preventive measures include the following:

(a) careful selection of materials that have a minimum susceptibility to SCC in the expected service environmental conditions

(b) an understanding of the Owner-approved specifications for water chemistry

(c) use of design and fabrication techniques that minimize surface tensile stress or that mitigate residual stresses

The elimination, or reduction to some threshold level, of one of the factors in Figure W-2120-1 eliminates SCC (ref. [6]). See the Utilities Requirement Document for aggressive species to avoid, such as low-melting-point metals (lead, antimony, cadmium, indium, mercury, zinc, bismuth, copper, and tin) and their alloys and high-sulfur materials (with the exception of strong acid cation resin). Impurity levels (e.g., levels of chlorine, fluorine, mercury, arsenic, lead, sulfur, and zinc) in nonmetallic materials such as gaskets, lubricants, and cleaning fluids should be controlled to given individual and combined part-per-million levels, and polyvinyl chloride (PVC), Teflon, and fluorosilicates should not be allowed at all (ref. [40])

W-2130 MATERIALS

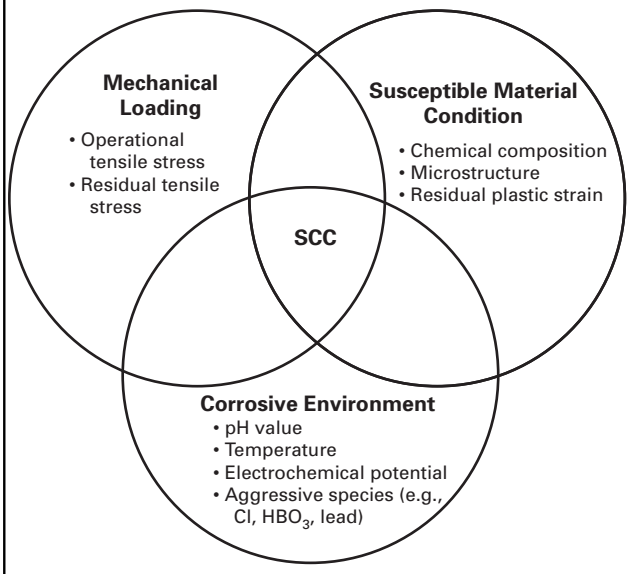
SSC potential can be significantly reduced by selecting resistant materials known to offer corrosion resistance in a LWR environment.

W-2131 Austenitic Stainless Steels

The austenitic stainless steels (e.g., Types 304 and 316 stainless steels) have proven to be resistant to SCC in PWR primary coolant but have had significant SCC problems in BWR reactor coolant and in many aqueous environments containing chlorides and other contaminants.

Analysis of SCC in austenitic stainless steel welds removed from BWR service has shown that cracking initiates in the weld heat-affected zone (HAZ) in the base metal near the weld fusion line. The weld metal itself rarely contains SCC, and cracking in the base metal outside the HAZ has not been observed. It has been

Figure W-2120-1
Environmental Conditions Required for SCC



GENERAL NOTE: See W-3200 for irradiation effects.

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concluded that the heat of welding "sensitizes" the HAZ through the precipitation of chromium carbides on grain boundaries. (See W-2510 for more discussion on sensitization of stainless steels.) Base metal that is free from carbide precipitation is resistant to cracking, and weld metal has been found to be resistant as long as delta ferrite levels exceed ~5%.

In the HAZ, cracking appears to follow the microstructural grain boundaries because precipitation of chromium carbides at the grain boundaries reduces the local free chromium content and, consequently, the grain boundary corrosion resistance. This type of cracking is referred to as intergranular stress corrosion cracking (IGSCC).

The industry action to mitigate IGSCC in BWRs through material selection has been to develop and use nuclear grade (NG) alloys based on the traditional Types 304, 304L, 316, and 316L stainless steel grades. NG alloys contain very low carbon levels as well as controls on nitrogen content and other trace elements. The nitrogen content is increased to further improve IGSCC resistance and to recover strength lost by the reduced carbon content.

NG alloys successfully resist IGSCC in weld HAZs by limiting the degree of carbide precipitation on grain boundaries during welding, i.e., they are difficult to sensitize. However, it is emphasized that fabrication techniques also play a significant role in the corrosion resistance of these materials, as in the following examples:

(a) Control of heat input when welding austenitic stainless steel is important in maintaining SCC resistance.

(b) Care should be taken when performing postweld heat treatment (PWHT) of low alloy steel components containing stainless steel parts since the specified PWHT temperature for many low alloy steels is within the sensitization temperature range 800°F to 1,500°F (427°C to 826°C) for austenitic stainless steels.

(c) Solution annealing should be considered for materials that are subject to high levels of cold work during fabrication.

(d) Machining techniques can introduce high residual surface tensile stresses if not carefully controlled.

The SCC resistance of austenitic stainless steels should always be confirmed by testing in accordance with ASTM A262, Practice E, with Practice A screening permitted (ref. [45]). For temperatures below 200°F (93°C), IGSCC is not a significant concern for austenitic stainless steels in normal BWR water environments without chlorides and sulfates. In the presence of chlorides or sulfates, SCC can occur at temperatures as low as room temperature. The presence of sulfates and chlorides or fluorides can lead to transgranular SCC in all austenitic stainless steels, even in unsensitized material and in the absence of oxygen (ref. [32]). The use of low- carbon nuclear grade material is not sufficient to prevent SCC in the presence of these contaminants. SCC mitigation is restricted to water purity control in this case.

W-2132 Stainless Steel Weld Metal

Industry actions to prevent IGSCC in stainless steel weld metals (typically Types 308 and 308L alloys) focus on controlling weld metal ferrite content. Traditionally, ferrite content in the weld metal, as depicted by a ferrite number (FN), has been restricted to values between 5 and 20. The lower limit provides sufficient ferrite to avoid microfissuring during welding, and the upper limit reduces the potential for thermal aging effects (ferrite decomposition). NRC Regulatory Guide 1.31 (ref. [1]) notes certain SFA weld filler metal specifications that do not require FN determinations. A cooperative study group was formed by ASME, the American National Standards Institute (ANSI), and the U.S. Nuclear Regulatory Commission (NRC) to investigate controls for ferrite content in austenitic stainless steel weld material to minimize IGSCC in weldments; the results were incorporated in the Code and endorsed by the NRC (refs. [1] and [4]). In BWR applications, the ferrite number for ferrite in weld metals is normally specified to be 8 FN minimum in Owner ordering-data requirements; NRC guidelines have also specified a minimum ferrite level of 7.5 FN for welding-resistant material (ref. [10]).

W-2133 Ni–Cr–Fe Alloys 600 and 690

Early on, the industry experienced widespread chloride-induced SCC and pitting in austenitic stainless steel steam generator tubes, and a move was made to the nickel-based alloys, which are resistant to chloride SCC. The primary alloy selected was Ni–Cr–Fe Alloy 600 (Inconel® 600). Later, it was found that Alloy 600 itself was subject to SCC in primary coolant without contaminants (PWSCC). By the time this behavior was detected, Alloy 600 and its associated weld filler materials, primarily Alloys 82 and 182, had been extensively applied in steam generator tubes and other LWR components, such as reactor vessel control rod drive nozzles, reactor vessel nozzle buttering for safe ends, and pressurizer heater sleeves.

Material changes to prevent SCC in nickel-based Ni–Cr–Fe Alloy 600 occurred in two stages. Early research on SCC in Alloy 600 steam generator tubes revealed that cracking often initiated in the primary side of the tubes in areas of high residual stress, e.g., tubesheet roll transitions, inner U-bends, and support plate dents. Laboratory work confirmed that the cracking was PWSCC. In response, heat treatments for Alloy 600 were developed that included a high-temperature solution anneal followed by a thermal treatment at a lower temperature. Use of Alloy 600 in the late 1970s and early 1980s involved replacing as-produced "mill annealed" Alloy 600 components with "thermally treated" Alloy 600 components (ref. [31]).

Unlike the nuclear grade stainless steels, the thermally treated Alloy 600 replacement material did not involve any significant material chemistry changes. However,

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further testing showed that thermally treated Alloy 600 was improved but not immune to PWSCC. A new alloy was required and developed by International Nickel Co. to provide a high level of PWSCC resistance. The alloy is designated Alloy 690 and differs from Alloy 600 primarily in its very high chromium content (e.g., SB-166 UNS N06990). The nuclear power industry has generally adopted additional controls on the production of this material that include tighter chemistry controls and special solution annealing and thermal heat treatments (TT) that have been demonstrated to provide optimal PWSCC resistance. Alloy 690 made to these specifications has proven to be very resistant to PWSCC and has become the standard replacement for Alloy 600 in new PWR designs and for repairs in existing plants. There is no significant operating history for Alloy 690 in BWR service.

The performance of Alloy 600 in BWR service has been mixed. IGSCC has been observed in applications involving crevices, such as crevices behind nozzle thermal sleeves and welded crevices (including a welded sleeve on shroud head bolts) or cracks initiated in Alloy 182 nozzle buttering that propagate into Alloy 600. The alloy has generally performed well in uncreviced applications such as core shroud support legs. A niobium-modified Ni–Cr–Fe Alloy 600 that is resistant to IGSCC has been developed for advanced boiling water reactor (ABWR) core support structure–support shroud service (refs. [7], [11], and [31]).

W-2134 Ni–Cr–Fe Weld Metals

Alloys 182 (covered electrode) and 82 (bare wire) weld fillers are used in Ni–Cr–Fe Alloy 600 construction and for buttering in bimetallic welds such as those joining low alloy steel nozzles to stainless steel safe-ends. They have proven to be susceptible to IGSCC and PWSCC in LWR service. Because of its higher chromium content, Alloy 82 is considered to be more resistant to SCC; advanced BWRs are still using Alloy 82 (with control of stabilizing ratio) at present. In BWRs, no Alloy 82 crack initiation has been experienced, but Alloy 182 cracks readily. Cracking has been primarily observed in full penetration nozzle-to-piping welds and partial penetration welded nozzles. However, neither of these materials should be used in new PWR applications where they will be exposed to primary water.

Alloys 52 and 152 filler metals were developed for use with Alloy 690 and have proven to be resistant to PWSCC in PWRs and IGSCC in BWRs. However, industry experience and laboratory testing have shown that these alloys are susceptible to hot cracking or ductility dip cracking during welding, so improvements to these alloys continue to be developed. Alloys 52M, 52MSS, and 152M have all been developed to improve weldability while maintaining high levels of SCC resistance. Recommendations to prevent hot cracking have been developed (refs. [38] and [39]).

The presence of susceptible material in nozzle welds (primarily buttering) and piping welds has required the industry to develop mitigating actions to prevent or control PWSCC in PWRs and IGSCC in BWRs. These include externally deposited weld overlays, which replace the structural function of the susceptible material; internally deposited weld inlays and onlays that isolate the susceptible material from the corrosive environment; and various stress improvement techniques, such as induction-heating stress improvement and mechanical stress improvement. ASME has published Code Cases for many of these techniques (refs. [42] and [43]). These mitigating actions are discussed further in W-2140 and W-2150.

W-2135 Other Materials

Nickel-based Alloy X-750 is a high-strength nickel alloy frequently used for bolting, springs, and other applications requiring high strength and general corrosion resistance in reactor-coolant environments. The alloy has demonstrated IGSCC in BWR jet pump hold-down beams and some PWR bolting applications. IGSCC in this material has been mitigated by a combination of actions, including redesign to reduce stresses, reduction of the preloads, refinements to the solution annealing and age-hardening heat treatments, and surface conditioning to reduce residual tensile stresses. Procurement specifications for Alloy X-750 parts should impose the manufacturing restrictions recommended in refs. [12] and [13].

Alloy 718 has been used in some cases as a replacement for Alloy X-750. A single-step aging treatment appears to result in greater SCC resistance as opposed to the conventional two-step aging normally applied to this alloy to generate higher strength. Alloy 718 with a single-step aging treatment was recently added to Code Case N-60-6 (ref. [44]) for use in core support structures and has been used at one BWR for jet pump hold-down beams.

High-strength low alloy steel (e.g., AISI 4130, AISI 4140) bolting also experienced degradation in operating plants. It was first observed in nuclear service in steam generator manway bolting failures and equipment anchor bolt failures in damp environments. The manway failures were often observed in bolts lubricated with molybdenum disulfide. This lubricant decomposes to produce aggressive sulfides in the presence of steam, which is often present due to leaking manway gaskets. More important, bolts most susceptible to SCC were observed to be the hardest or strongest. It was found that bolts with a Rockwell hardness of HRC 40 and higher were very susceptible to cracking in all environments. Consequently, refs. [20] and [21] require that the material tensile strength be limited to 150 ksi (1 034 MPa) in critical high-strength bolting. Field experience and tests have shown that this restriction greatly reduces the potential for SCC in high-strength bolting materials. Use of martensitic stainless

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steels, such as those of chromium grades from 11% to 13%, should be avoided for reactor stud-bolting applications (ref. [21]).

SCC has been observed in Type 416 martensitic stainless steel pump shaft couplings exposed to raw-water service. The cause was the presence of sulfur inclusions in this resulfurized, free-machining steel. Use of free-machining grades of martensitic stainless steels and low alloy steels should be avoided in critical applications (refs. [16] and [21]).

Finally, it should be noted that manufacturing process controls at the mill production and component fabrication levels are required to produce components resistant to SCC. Many of these controls do not appear in materials specifications and must be imposed via Owner ordering data or Owner-generated specifications. Examples of required controls include

(a) limits on welding heat input and heat treatment

(b) limits on cold work and cold forming in austenitic stainless steels

(c) limits on pickling and intergranular attack on austenitic stainless steel and nickel-based alloy surfaces

(d) cleanliness requirements and limits on chlorides and low-melting-point metals in lubricants, paints, marking crayons, insulation, and tape adhesives (refs. [3] and [4])

(e) surface-finish requirements and limits on abusive grinding

W-2140 STRESS CONTROL

High peak-service stresses at geometric discontinuities have led to SCC failures in hold-down beams, springs, and bolting. High residual stress also contributes to SCC. Residual stresses are present in most structures as a result of fabrication practices (e.g., machining, forming, and welding) and are impacted by factors such as welding processes and resulting shrinkage stresses.

Residual stress, often combined with the sensitization of the HAZ near the weld fusion line, produces two of the three criteria required for IGSCC in stainless steels: tensile stress and susceptible material. The contribution of the reactor coolant (aggressive environment) is discussed in W-2150. As noted in W-2130, different materials are susceptible to SCC in different LWR environments, so weldments susceptible to SCC in BWR environments may not be similarly affected in PWR environments and vice versa.

Depending on the joint configuration, weld sequence, and weld materials, large through-wall residual stresses may result from the welding process. In BWR piping, tensile stresses have been found to be near yield stress levels on the inside (I.D.) surface (ref. [9]) due to weld high residual stress at the I.D. surface. Weld residual stresses can be minimized with proper weld joint designs (narrow groove is better), low heat input, weld sequencing, and sufficient filler metal feed to reduce the local diametrical shrinkage at the joint. Residual stresses at the I.D. surface can be minimized by avoiding final welding steps at the I.D. surface, such as an I.D. repair.

Many operating plants contain welds in piping and nozzles that are sensitized or that contain susceptible materials. In these cases, action can be taken to reduce tensile stresses on the I.D. surface of the weld, including

(a) induction heat stress improvement (IHSI) method with application of induction heat to the weld outside surface while keeping the pipe filled with water (cooled). The thermal gradient generated by induction heating causes compressive yielding on the outside surface and weld residual stress reversal once heating is terminated.

(b) mechanical stress improvement process (MSIP), which uses a hydraulic clamp to compress the pipe outside diameter (O.D.) surface near, but not on, the weld, leaving compressive residual stresses in the weld on the inside diameter (I.D.) surface.

(c) structural weld overlay on the O.D. surface of the pipe, which induces compressive stresses on the pipe I.D. surface. A weld inlay/overlay also provides a barrier to the propagation of cracks if composed of resistant weld filler (e.g., Type 308L stainless steel with controlled ferrite or Alloy 52 nickel-based filler).

(d) shot peening, laser peening, or water-jet peening, all of which generate compressive residual stresses at the component surface (ref. [36]). Water-jet- and laser-peening techniques have both been applied at commercial LWR plants to mitigate SCC of thick-wall components. These methods are typically applied underwater. In each method, a shock pressure is produced at the treated surface resulting in a layer of compressive residual stress to a depth of roughly at least 0.04 in. (1 mm).

In cases such as nozzles welded to large components (e.g., pressurizer heater sleeve welds or reactor vessel head control-rod-drive-mechanism nozzles) where stress improvement cannot be applied, either a modification to isolate the susceptible material from reactor coolant or component removal, redesign, and replacement are required to effect a permanent solution.

The expansion of tubes in the tubesheet of steam generators generates an expansion transition region in the tube that can generate high residual stresses. The design and method of expansion have a major impact on those stresses. Early steam generators demonstrated extensive PWSCC in the Alloy 600 tube expansion transitions because the tubes were rolled into the tubesheet. This produced a sharp transition with high tensile residual stresses. Expansions are currently performed hydraulically with carefully controlled transition geometries. This is another example of stress mitigation through careful fabrication.

Other fabrication activities can have a large impact on SCC. For example, significant cold work results in high residual surface tensile stresses. For Type 304 austenitic stainless steel in BWR service, cold-work levels above ~5% can impact SCC resistance by producing a

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martensite phase transformation in the alloy that becomes sensitized at operating temperatures (no welding required). In addition, cold work itself has been associated with BWR IGSCC failures in Type 316 stainless steel. Thus, cold work should be limited to less than 5% for all austenitic varieties used in BWR coolant.

Stainless steels seem to be resistant to SCC in PWR primary coolant, even at high cold-work levels. In fact, 20% cold-worked Type 316 stainless steel is the preferred material to replace Alloy X-750 bolting in PWR primary coolant service. Cold work can be limited by process controls during fabrication and by using material with maximum hardness limited to HRB 90 or less.

Grinding and machining should also be controlled. Abusive grinding can often be detected by surface color changes, which indicate overheating. For critical surfaces, all grinding should be followed by flapping to remove cold-worked surface material. For machined parts, surface cold work can be controlled by specifying a smooth surface finish on fabrication drawings. Single-point machining with a sharp tool is preferred. In critical bolting, rolled threads are preferred to machined threads because rolling leaves compressive residual stresses at the thread root.

W-2150 ENVIRONMENT

In this section, environmental effects on SCC are considered along with SCC mitigation through environmental modification.

The standard BWR coolant chemistry up through the 1980s was pure water. However, testing showed that the propensity for IGSCC in austenitic stainless steels was sensitive to water purity. Consequently, water purity requirements increased continuously through the decade, and conductivity limits decreased from 2 μS/cm to 0.2 μS/cm (2 micromhos/cm to 0.2 micromhos/cm) during that time (ref. [35]).

Water purity, while important, was not the primary cause of IGSCC in BWRs. Testing by General Electric in the 1980s showed that IGSCC cracking was driven by the electrochemical potential (ECP) of the coolant, which in a BWR is quite high due to the radiolytic decomposition of water in the core. This decomposition produces oxidizing species, such as peroxides and dissolved oxygen. The typical range in BWR water is 200 ppb to 300 ppb. Originally, the typical value for the ECP in the recirculation system was +200 mV_{SHE} (millivolts, standard hydrogen electrode). Tests demonstrated that IGSCC crack growth rate is reduced to a manageable level over the life of the plant with an ECP of –230 mV_{SHE} or lower. It was determined that the addition of hydrogen gas to the coolant via feedwater injection on the order of 1 ppm to 2 ppm could reduce the potential to –230 mV_{SHE} or lower. This practice (hydrogen water chemistry) is currently being used by most, if not all, U.S. BWRs.

Platinum electrodes placed in the recirculation loop flow under hydrogen injection conditions achieved very low ECP values at low feedwater hydrogen levels. For platinum and other noble metals, it appeared to be sufficient merely to reach stochiometry with the oxidizing species, which required feedwater injection levels of hydrogen in the tenths of a part per million range rather than the 1-ppm to 2-ppm levels required to markedly reduce the potential on stainless steel surfaces. This became the basis for noble metal chemical addition chemistry now being used in BWRs. Noble metal solutions are injected into the feedwater to deposit on recirculation piping surfaces where they catalyze the hydrogen recombination reaction with the oxidizing species generated in the core. Noble metal chemistry achieves the required ECP to inhibit IGSCC while minimizing the required amount of hydrogen injected into the feedwater. Currently, online noble metal chemical injection systems are preferred over the batch process.

The primary coolant in PWRs contains boric acid to control core reactivity, ammonium or lithium hydroxide additions to control pH, and hydrogen additions to levels of ~25 cc/kg to ~50 cc/kg with a typical operating range of 30 cc/kg to 40 cc/kg to reduce the corrosion potential. At 572°F (300°C), a neutral pH_T is 5.7, and the coolant is maintained in the mildly alkaline range (e.g., pH_T 6.9 to pH_T 7.4), to minimize corrosion of structural materials and deposition of corrosion products on core heat transfer surfaces.

The coolant pH seems to have little impact on PWSCC in Alloy 600. The high levels of added hydrogen create very low ECP conditions on plant surfaces (e.g., –600 mV_{SHE}), but this also has only a moderate impact on PWSCC crack initiation rates in Alloy 600. On the other hand, the very low potential has protected sensitized stainless steel welds quite well over the years, which explains the excellent historic performance of the austenitic stainless steels in PWR primary systems.

Operating experience and laboratory tests on Alloy 600 have demonstrated that PWSCC is highly temperature dependent. The crack initiation time is reduced sharply at high temperatures. The behavior is of the Arrhenius type, with activation energy of ~50 kCal/mole. Therefore, a 15°F (8°C) reduction in temperature doubles the lifetime of Alloy 600 components. As a result of this finding, some plants in the past reduced the reactor outlet temperature to extend the life of Alloy 600 steam generator tubing and other Alloy 600 components.

Finally, it should be noted that certain chemical impurities can cause SCC in reactor plant components and must be controlled. Chlorides, even at low levels, can rapidly crack austenitic stainless steels at reactor operating temperatures. This is true even for nuclear grade material and other stainless materials that are not sensitized. Reduced sulfur compounds, which could be produced by the decomposition of ion exchange resins, can crack sensitized stainless steels and Alloy 600 in minutes, even at

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