INTERNATIONAL STANDARD

ISO 12241

First edition
1998-03-01

Thermal insulation for building equipment and industrial installations — Calculation rules

Isolation thermique des équipements du bâtiment et des installations industrielles — Méthodes de calcul

Reference number
ISO 12241:1998(E)

Contents

International Organization for Standardization
Case postale 56 • CH-1211 Genève 20 • Switzerland
Internet central@iso.ch
X.400 c=ch; a=400net; p=iso; o=isocs; s=central

Printed in Switzerland

Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

International Standard ISO 12241 was prepared by Technical Committee ISO/TC 163, *Thermal insulation*, Subcommittee SC 2, *Calculation methods*.

Annexes A to C of this International Standard are for information only.

Introduction

Methods relating to conduction are direct mathematical derivations from Fourier´s Law of Heat Conduction, so international consensus is purely a matter of mathematical verification. No significant difference in the equations used in the member countries exists. For convection and radiation, however, there are no methods in practical use which are mathematically traceable to Newton´s Law of Cooling or the Stefan-Boltzman Law of Thermal Radiation, without some empirical element. For convection, in particular, many different equations have been developed, based on laboratory data. Different equations have become popular in different countries, and no exact means are available to select between these equations.

Within the limitations given, these methods can be applied to most types of industrial thermal insulation heat transfer problems.

These methods do not take into account the permeation of air or the transmittance of thermal radiation through transparent media.

The equations in these methods require for their solution that some system variables be known, given, assumed, or measured. In all cases, the accuracy of the results will depend on the accuracy of the input variables. This International Standard contains no guidelines for accurate measurement of any of the variables. However, it does contain guides which have proven satisfactory for estimating some of the variables for many industrial thermal systems.

It should be noted that the steady-state calculations are dependent on boundary conditions. Often a solution at one set of boundary conditions is not sufficient to characterize a thermal system which will operate in a changing thermal environment (process equipment operating year-round, outdoors, for example). In such cases local weather data based on yearly averages or yearly extremes of the weather variables (depending on the nature of the particular calculation) should be used for the calculations in this International Standard.

In particular, the user should not infer from the methods of this International Standard that either insulation quality or avoidance of dew formation can be reliably assured based on minimal simple measurements and application of the basic calculation methods given here. For most industrial heat flow surfaces, there is no isothermal state (no one, homogeneous temperature across the surface), but rather a varying temperature profile. This condition suggests the need for numerous calculations to properly model thermal characteristics of any one surface. Furthermore, the heat flow through a surface at any point is a function of several variables which are not directly related to insulation quality. Among others, these variables include ambient temperature, movement of the air, roughness and emissivity of the heat flow surface, and the radiation exchange with the surroundings (often including a great variety of interest). For calculation of dew formation, variability of the local humidity is an important factor.

Except inside buildings, the average temperature of the radiant background seldom corresponds to the air temperature, and measurement of background temperatures, emissivities, and exposure areas is beyond the scope of this International Standard. For these reasons, neither the surface temperature nor the temperature difference between the surface and the air can be used as a reliable indicator of insulation performance or avoidance of dew formation.

Clauses 4 and 5 of this International Standard give the methods used for industrial thermal insulation calculations not covered by more specific standards. In applications where precise values of heat energy conservation or (insulated) surface temperature need not be assured, or where critical temperatures for dew formation are either not approached or not a factor, these methods can be used to calculate heat flow rates.

Clauses 6 and 7 of this International Standard are adaptations of the general equation for specific applications of calculating heat flow temperature drop and freezing times in pipes and other vessels.

Thermal insulation for building equipment and industrial installations — Calculation rules

1 Scope

This International Standard gives rules for the calculation of heat transfer related properties of building equipment and industrial installations, predominantly under steady-state conditions, assuming one-dimensional heat flow only.

2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this International Standard. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this International Standards are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 7345:1987, *Thermal insulation — Physical quantities and definitions*

ISO 9346:1987, *Thermal insulation — Mass transfer — Physical quantities and definitions*

NOTE — For further publications, see annex C.

3 Definitions, symbols and abbreviations

For the purposes of this International Standard, the definitions given in ISO 7345 and ISO 9346 apply.

3.1 Physical quantities, symbols and units

Physical quantities	Symbol	Unit
heat flow rate	Φ	W
density of heat flow rate	q	W/m^2
linear density of heat flow rate	q_l	W/m
thermodynamic temperature	T	K
Celsius temperature	θ	°C
temperature difference	$\Delta\theta$	K
thermal conductivity	λ	W/(m·K)
design thermal conductivity	λ_d	W/(m·K)
surface coefficient of heat transfer	h	$W/(m^2 \cdot K)$
thermal resistance	R	$m^2 \cdot K/W$
linear thermal resistance	R_l	m·K/W
linear thermal surface resistance	R_{le}	m·K/W
surface resistance of heat transfer	R_s	$m^2 \cdot K/W$
thermal resistance for hollow sphere	R_{sph}	K/W
thermal transmittance for hollow sphere	U_{sph}	W/K
thermal transmittance	U	$W/(m^2 \cdot K)$
linear thermal transmittance	U_l	W/(m·K)
specific heat capacity at constant pressure	c_p	kJ/(kg·K)
thickness	d	m
diameter	D	m
temperature factor	a_r	K^3
radiation coefficient	C_r	$W/(m^2 \cdot K^4)$
emissivity	ε	-
Stefan Boltzmann constant (see reference [9])	σ	$W/(m^2 \cdot K^4)$
height	H	m
length	l	m
thickness parameter (see 4.2)	C'	m
perimeter	P	m
area	A	m^2
volume	V	m^3
velocity	v	m/s
time	t	s
mass	m	kg
mass flow rate	$\dot{m}$	kg/h
density	ρ	kg/m^3
specific enthalpy; latent heat of freezing	h_{fr}	kJ/kg
relative humidity	ϕ	%

3.2 Subscripts

ambient	a
average	av
cooling	c
convection	cv
design, duct, dewpoint	d
exterior,external	e
effective	ef
final medium	fm
freezing	fr
interior, internal	i
initial medium	im
laboratory	lab
linear	l
pipe	p
radiation	r
reference	ref
surface	s
exterior surface	se
interior surface	si
spherical	sph
soil	E
total	T
vessel	v
water	w
wall	W

4 Calculation methods for heat transfer

4.1 Fundamental equations for heat transfer

The formulae given in this clause apply only to the case of heat transfer in the steady-state, i.e. to the case where temperatures remain constant in time at any point of the medium considered.
Generally the thermal conductivity design value is temperature dependent (see figure 1, dashed line).
For further purposes of this International Standard, the design value for the mean temperature for each layer shall be used.

NOTE —This may imply iterative calculation.

4.1.1 Thermal conduction

Thermal conduction normally describes molecular heat transfer in solids, liquids and gases under the effect of a temperature gradient.

It is assumed in the calculation that a temperature gradient exists in one direction only and that the temperature is constant in planes perpendicular to it.

The density of heat flow rate q for a plane wall in the x-direction is given by:

$$q = -\lambda \cdot \frac{d\theta}{dx} \quad \mathrm{W/m^2} \tag{1}$$

For a single layer

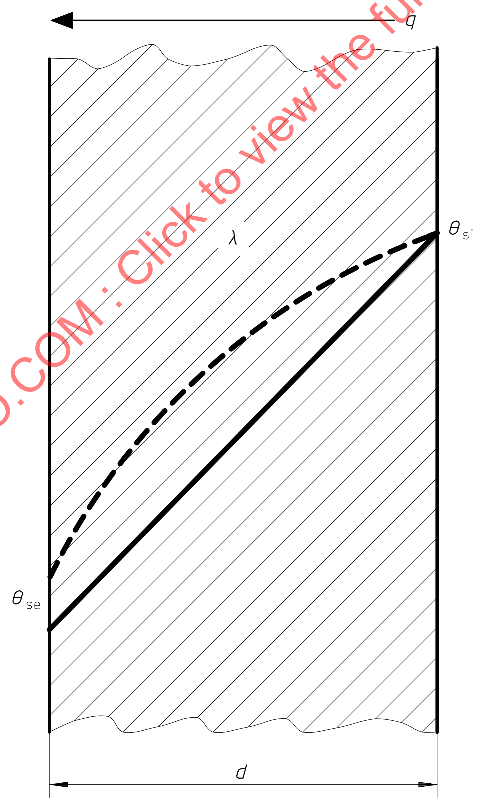
$$q = \frac{\lambda}{d} \cdot (\theta_{si} - \theta_{se}) \quad W/m^2 \qquad (2)$$

or

$$q = \frac{\theta_{si} - \theta_{se}}{R} \quad W/m^2 \qquad (2a)$$

where

λ is the thermal conductivity of the material, in W/(m·K);
d is the thickness of the plane wall, in m;
θ_{si} is the temperature of the internal surface, in °C;
θ_{se} is the temperature of the external surface, in °C;
R is the thermal resistance of the wall in (m²·K)/W.



NOTE — The straight curve shows the negligible, the dashed one the strong temperature dependence of λ.

Figure 1: Temperature distribution in a single layer wall

For multi-layer insulation

$$q = \frac{\theta_{si} - \theta_{se}}{R'} \quad \text{W/m}^2 \qquad (3)$$

where R' is the thermal resistance of the multi-layer wall

$$R' = \sum_{j=1}^{n} \frac{d_j}{\lambda_j} \quad \text{m}^2 \cdot \text{K/W} \qquad (4)$$

NOTE — The prime denotes a multi-layer quantity.

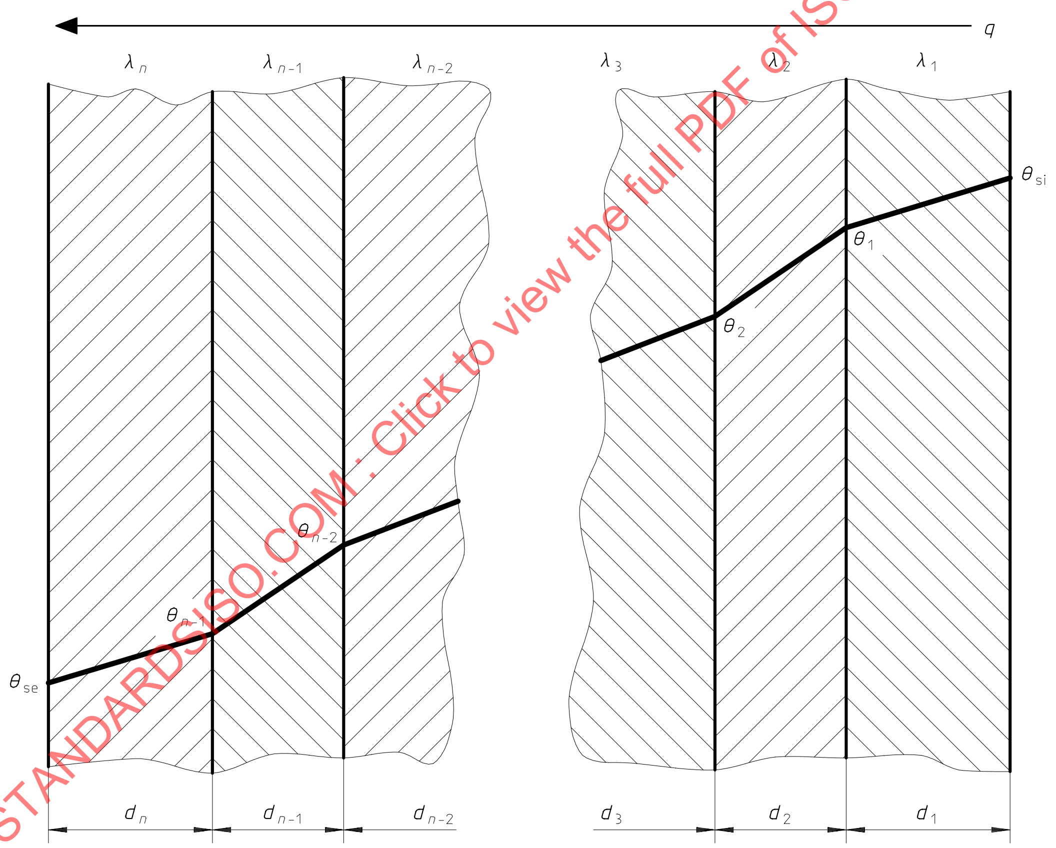


Figure 2: **Temperature distribution in a multi-layer wall**

The linear density of heat flow rate q_l of a single layer hollow cylinder:

$$q_l = \frac{\theta_{si} - \theta_{se}}{R_l} \quad \text{W/m} \tag{5}$$

where R_l is the linear thermal resistance of a single layer hollow cylinder:

$$R_l = \frac{\ln \frac{D_e}{D_i}}{2 \cdot \pi \cdot \lambda} \quad \text{m} \cdot \text{K/W} \tag{6}$$

D_e is the exterior diameter of the layer, in m;
D_i is the interior diameter of the layer, in m.

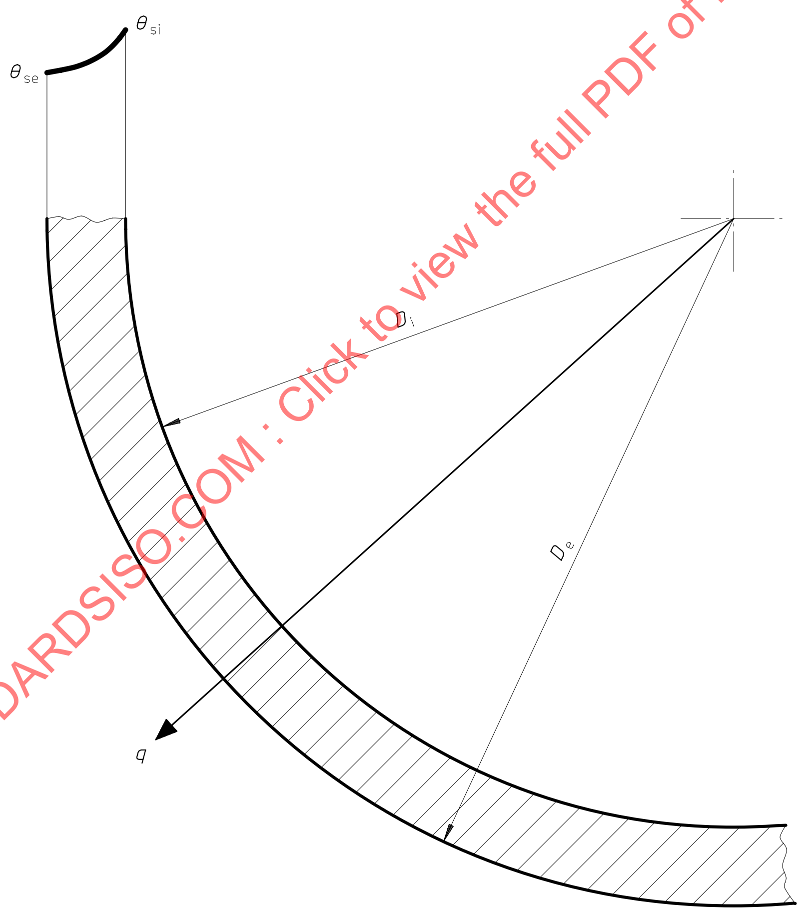


Figure 3: **Temperature distribution in a single layer hollow cylinder**

For multi-layer hollow cylinder:

$$q_l = \frac{\theta_{si} - \theta_{se}}{R_l'} \qquad \text{W / m} \qquad (7)$$

where

$$R_l' = \frac{1}{2 \cdot \pi} \sum_{j=1}^{n} \left(\frac{1}{\lambda_j} \cdot \ln \frac{D_{ej}}{D_{ij}} \right) \qquad \text{m} \cdot \text{K / W} \qquad (8)$$

with $D_0 \equiv D_i$ and $D_n \equiv D_e$

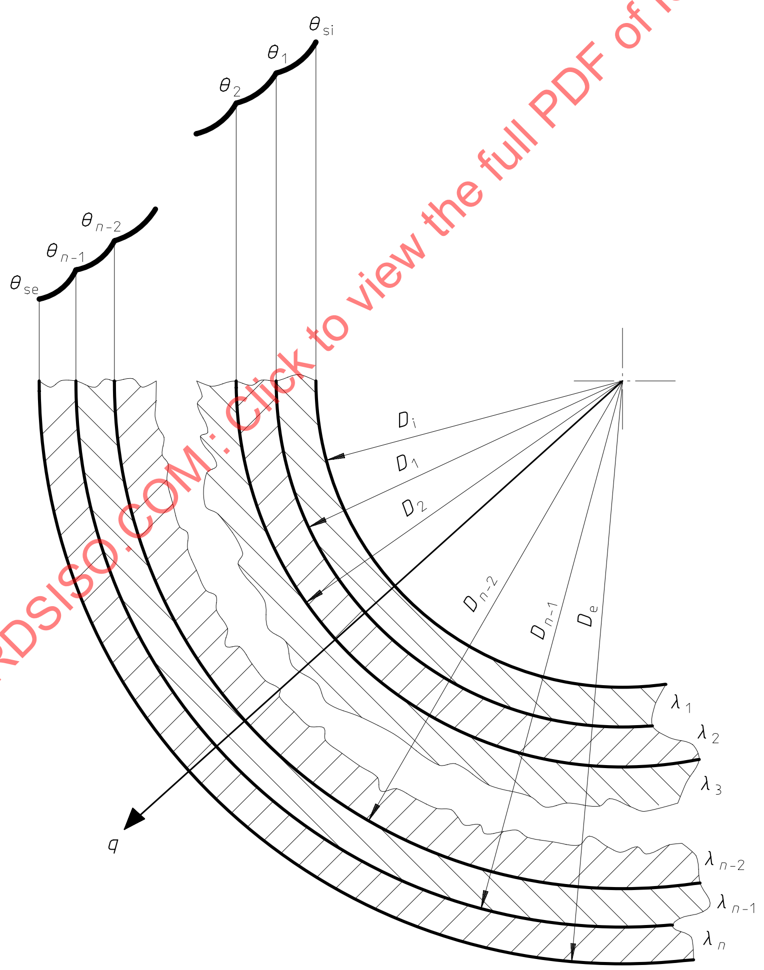


Figure 4: **Temperature distribution in a multi-layer hollow cylinder**

The heat flow rate of a single layer hollow sphere is

$$\Phi_{sph} = \frac{\theta_{si} - \theta_{se}}{R_{sph}} \quad \text{W} \tag{9}$$

where R_{sph} is the thermal resistance of a single layer hollow sphere in K/W.

$$R_{sph} = \frac{1}{2 \cdot \pi \cdot \lambda}\left(\frac{1}{D_i} - \frac{1}{D_e}\right) \quad \text{K/W} \tag{10}$$

D_e is the outer diameter of the layer, in m;
D_i is the inner diameter of the layer, in m.

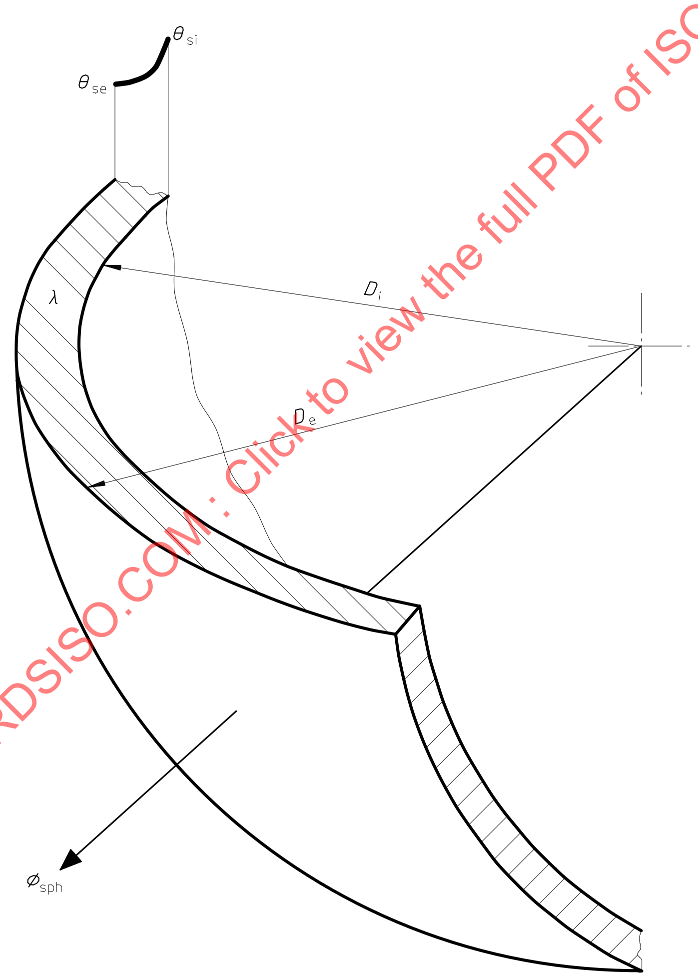


Figure 5: **Temperature distribution in a single layer hollow sphere**

The heat flow rate of a multi layer hollow sphere is

$$\Phi_{sph} = \frac{\theta_{si} - \theta_{se}}{R'_{sph}} \quad W \qquad (11)$$

where

$$R'_{sph} = \frac{1}{2\pi} \sum_{j=1}^{n} \frac{1}{\lambda_j} \cdot \left(\frac{1}{D_{j-1}} - \frac{1}{D_j} \right) \quad K/W \qquad (12)$$

with $D_0 \equiv D_i$ and $D_n \equiv D_e$.

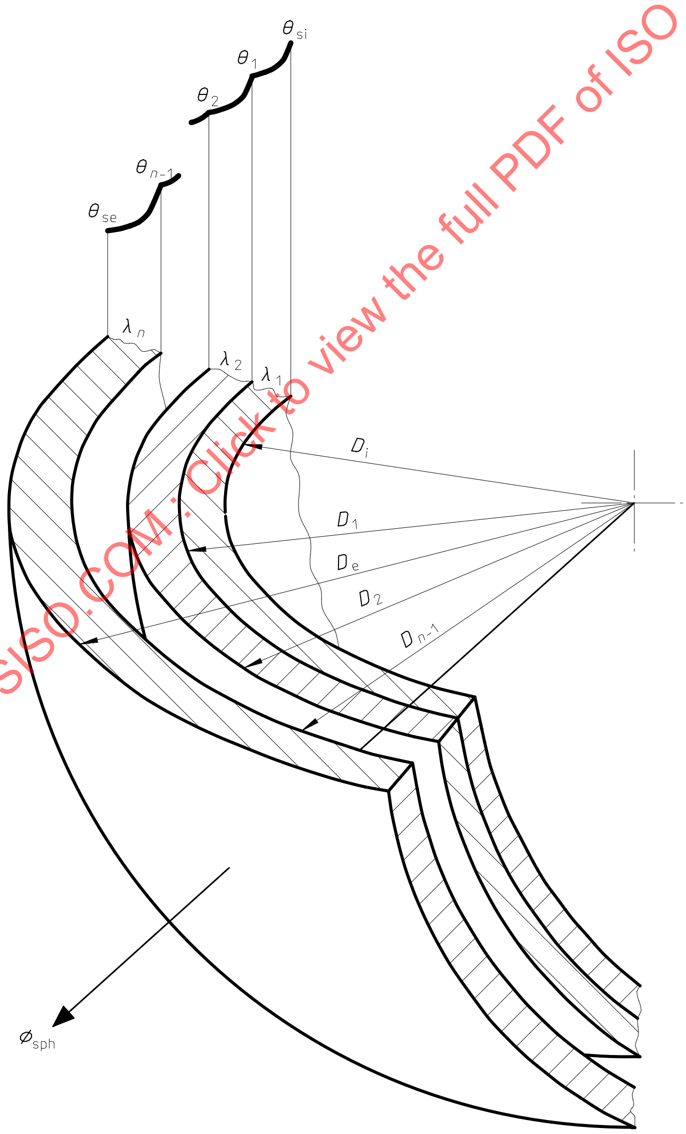


Figure 6: **Temperature distribution in a multi-layer hollow sphere**

The heat flow rate through the wall of a duct with rectangular cross section is given by

$$q_d = \frac{\theta_{si} - \theta_{se}}{R_d} \quad \text{W/m} \tag{13}$$

The linear thermal resistance of the wall of such a duct can be approximately calculated by

$$R_d = \frac{2 \cdot d}{\lambda \cdot (P_e + P_i)} \quad \text{m} \cdot \text{K/W} \tag{14}$$

where

P_i is the inner perimeter of the duct, in m;

P_e is the external perimeter of the duct, in m;

d is the thickness of the insulating layer, in m.

$$P_e = P_i + (8 \cdot d) \tag{14a}$$

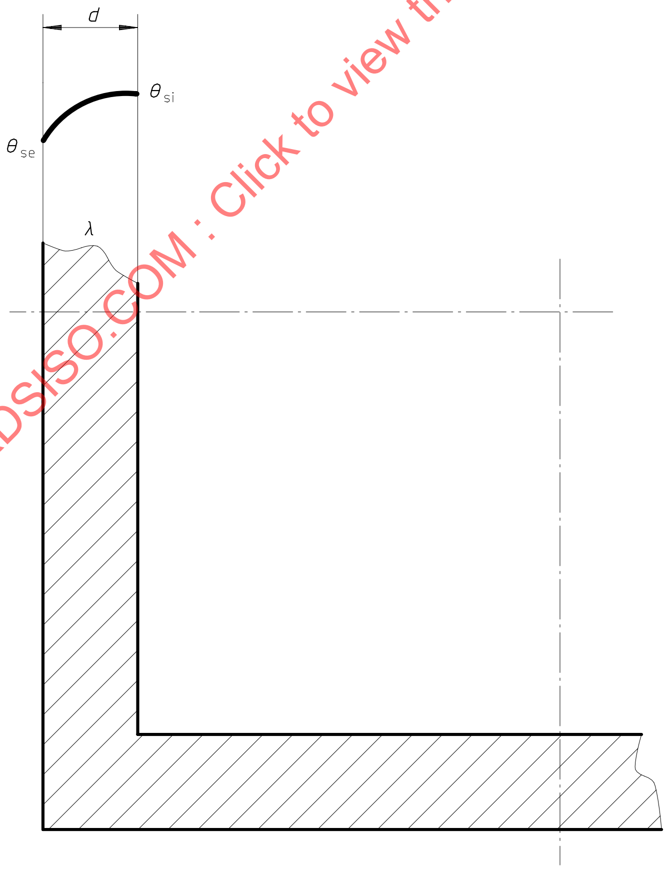


Figure 7: **Temperature distribution in a wall of a duct with rectangular cross section**

4.1.2 Surface coefficient of heat transfer

In general the surface coefficient of heat transfer is given by:

$$h = h_r + h_{cv} \quad W/\left(m^2 \cdot K\right) \qquad (15)$$

where

h_r is the radiative part of the surface coefficient of heat transfer;
h_r is dependent on the temperature and the degree of emissivity of the surface.

NOTE — The emissivity is defined as the ratio between the radiation coefficient of the surface and the black body radiation constant (see ISO 9288).

h_{cv} is the convective part of the surface coefficient of heat transfer.
h_{cv} is in general dependent on a variety of factors such as air movement, temperature, the relative orientation of the surface, the material of the surface and other factors.

4.1.2.1 Radiative part of surface coefficient h_r

h_r is given by:

$$h_r = a_r \cdot C_r \quad W/\left(m^2 \cdot K\right) \qquad (16)$$

a_r is the temperature factor. It is given by:

$$a_r = \frac{(T_1)^4 - (T_2)^4}{T_1 - T_2} \quad K^3 \qquad (17)$$

and can be approximated up to a temperature difference of 200 K by

$$a_r \approx 4 \cdot (T_{av})^3 \quad K^3 \qquad (17a)$$

where

T_{av} is 0,5 × (surface temperature + ambient or surface temperature of a radiating surface in the neighbourhood), in K;
C_r is the radiation coefficient, in $W/(m^2 \cdot K^4)$.
C_r is given by

$$C_r = \varepsilon \cdot \sigma$$
$$\sigma = 5{,}67 \cdot 10^{-8} \quad W/(m^2 \cdot K^4)$$

4.1.2.2 Convective part of surface coefficient h_{cv}

For convection a distinction has to be made between surface coefficient inside buildings and in open air. For pipes and containers there is a difference as well between internal surface coefficient h_i and the external surface coefficient h_{se}.

a) Inside buildings:

In the interior of buildings h_{cv} can be calculated for **plane vertical walls and vertical pipes** for laminar free convection ($H^3 \cdot \Delta\theta \leq 10$ m^3·K) by

$$h_{cv} = 1{,}32 \cdot \sqrt[4]{\frac{\Delta\theta}{H}} \ \mathrm{W/(m^2 \cdot K)} \qquad (18a)$$

where

$\Delta\theta = |\theta_{se} - \theta_a|$, in K;

θ_{se} is the surface temperature of the wall, in °C;

θ_a is the temperature of the ambient air inside the building, in °C;

H is height of the wall or diameter of a pipe, in m.

For **vertical plane walls, vertical pipes** and in approximation for large spheres inside buildings the convective part h_{cv} for turbulent free convection ($H^3 \cdot \Delta\theta > 10$ m^3·K) is given by:

$$h_{cv} = 1{,}74 \sqrt[3]{\Delta\theta} \quad \mathrm{W/(m^2 \cdot K)} \qquad (18b)$$

For **horizontal pipes inside** buildings h_{cv} is given by

- laminar airflow ($D_e^3 \cdot \Delta\theta \leq 10$ m^3·K)

$$h_{cv} = 1{,}25 \cdot \sqrt[4]{\frac{\Delta\theta}{D_e}} \quad \mathrm{W/(m^2 \cdot K)} \qquad (18c)$$

- turbulent airflow ($D_e^3 \cdot \Delta\theta > 10$ m^3·K)

$$h_{cv} = 1{,}21 \cdot \sqrt[3]{\Delta\theta} \quad \mathrm{W/(m^2 \cdot K)} \qquad (18d)$$

b) Outside buildings:

For **vertical plane walls outside** of buildings and in approximation for large spheres the convective part h_{cv} of the surface coefficient is given by:

laminar airflow ($v \cdot H \leq 8$ m^2/s):

$$h_{cv} = 3{,}96 \cdot \sqrt{\frac{v}{H}} \ \mathrm{W/(m^2 \cdot K)} \qquad (18e)$$

- turbulent air flow ($v \cdot H > 8$ m²/s):

$$h_{cv} = 5{,}76 \cdot \sqrt[5]{\frac{v^4}{H}}\ \mathrm{W/(m^2 \cdot K)} \qquad (18f)$$

For horizontal and vertical pipes which are outside buildings the following equation applies:

- laminar airflow ($v \cdot D_e \leq 8{,}55 \times 10^{-3}$ m²/s) :

$$h_{cv} = \frac{8{,}1 \times 10^{-3}}{D_e} + 3{,}14 \cdot \sqrt{\frac{v}{D_e}}\quad \mathrm{W/(m^2 \cdot K)} \qquad (18g)$$

- turbulent airflow ($v \cdot D_e > 8{,}55 \times 10^{-3}$ m²/s) :

$$h_{cv} = 8{,}9 \cdot \frac{v^{0{,}9}}{D_e^{0{,}1}}\quad \mathrm{W/(m^2 \cdot K)} \qquad (18h)$$

where

D_e is the external insulation diameter, in m;

v is the wind velocity, in m/s.

NOTE — For calculation of surface temperature, formulas (18a) to (18d) should be used for wall and pipe instead of formulas (18e) to (18h) when the presence of wind is not established.

Table 1 gives a selection of appropriate equations to be used for calculation of h_{cv}.

Table 1 —Selection of h_{cv}

Location	Walls				Pipes			
	vertical		horizontal		vertical		horizontal	
	laminar	turbulent	laminar	turbulent	laminar	turbulent	laminar	turbulent
inside buildings	18a	18b	1)	1)	18a	18b	18c	18d
outside buildings	18e	18f	18e	18f	18g	18h	18g	18h
1) Not important for most practical purposes								

All the equations for the convective part of the outer thermal surface coefficient inside buildings apply for the heat transfer between surfaces and air at temperature differences $\Delta T < 100$ K.

4.1.2.3 Approximation for the calculation of h_{se}

For approximate calculations the following equations for the outer surface coefficient h_{se} can be used **inside** buildings.

For **horizontal** pipes

$$h_{se} = C_A + 0{,}05 \cdot \Delta\theta \quad \mathrm{W/(m^2 \cdot K)} \qquad (19)$$

For **vertical** pipes and walls

$$h_{se} = C_B + 0{,}09 \cdot \Delta\theta \quad W/(m^2 \cdot K) \qquad (20)$$

using the coefficients in table 2.

Equations 19 and 20 can be used for horizontal pipes in range of D_e = 0,25 m to 1,0 m and for vertical pipes for all diameters.

Table 2 — Coefficients C_A and C_B for approximate calculation of total exterior thermal surface coefficient

Surface	C_A	C_B	ε	$C_r \times 10^{-8}$ W/(m²·K⁴)
aluminium, bright rolled	2,5	2,7	0,05	0,28
aluminium, oxidized	3,1	3,3	0,13	0,74
galvanized sheet metal, blank	4,0	4,2	0,26	1,47
galvanized sheet metal, dusty	5,3	5,5	0,44	2,49
austenitic steel	3,2	3,4	0,15	0,85
aluminium - zinc sheet	3,4	3,6	0,18	1,02
nonmetallic surfaces	8,5	8,7	0,94	5,33

For cylindrical ducts with a diameter less than 0,25 m the convective part of the external surface coefficient can be calculated in good approximation by equation (18 c). For larger diameters i.e. D_e>0,25 m the equation for plane walls (18 a) can be applied. The respective accuracy is 5 % for diameters D_e>0,4m and 10% for diameters 0,25<D_e<0,40m. Equation (18 a) is also used for ducts with rectangular cross-section, having a width and height of similar magnitude.

4.1.2.4 External surface resistance

The reciprocal of the outer surface coefficient h_{se} is the external surface resistance.

For plane walls the surface resistance R_{se}, in m²·K/W, is given by

$$R_{se} = \frac{1}{h_{se}} \quad m^2 \cdot K/W \qquad (21)$$

For pipe insulation the linear thermal surface resistance R_{le} is given by:

$$R_{le} = \frac{1}{h_{se} \cdot \pi \cdot D_e} \quad m \cdot K/W \qquad (22)$$

For hollow spheres the thermal surface resistance $R_{sph\,e}$ is given by

$$R_{sph\,e} = \frac{1}{h_{se} \cdot \pi \cdot D_e^2} \quad K/W \qquad (23)$$

4.1.3 Thermal transmittance

Thermal transmittance U is given by

$$U = \frac{q}{\theta_i - \theta_a} \quad \mathrm{W/(m^2 \cdot K)} \tag{24}$$

For plane walls the thermal transmittance U can be calculated

$$\frac{1}{U} = \frac{1}{h_i} + R + \frac{1}{h_{se}} = R_{si} + R + R_{se} \quad \mathrm{m^2 \cdot K/W} \tag{25}$$

For pipe insulation the linear thermal transmittance U_l can be calculated

$$\frac{1}{U_l} = \frac{1}{h_i \cdot \pi \cdot D_i} + R_l + \frac{1}{h_{se} \cdot \pi \cdot D_e} = R_{li} + R_l + R_{le} \quad \mathrm{m^2 \cdot K/W} \tag{26}$$

For hollow spheres the thermal transmittance U_{sph} is given by:

$$\frac{1}{U_{sph}} = \frac{1}{h_i \cdot \pi \cdot D_i^2} + R_{sph} + \frac{1}{h_{se} \cdot \pi \cdot D_e^2} \quad \mathrm{K/W} \tag{27}$$

The surface resistance of flowing media in pipes R_{si} (in the cases predominantly considered here) is small and can be neglected. For the external surface coefficient h_{se}, equations (19) and (20) apply. For ducts one also has to use the internal surface coefficient.

The reciprocal of thermal transmittance U is the total thermal resistance R_T for plane walls and respectively the total linear thermal resistance R_{Tl} for pipe insulation and $R_{T\ sph}$ for hollow spheres insulations.

The thermal transmittance of a duct with rectangular cross sections can be obtained by eq. (25) by replacing R by R_d (eq. 14).

4.1.4 Temperatures of the layer boundaries

The general equation for the heat loss in a multi-layer wall may be written in the following general form:

$$q = \frac{\theta_i - \theta_a}{R_T} \quad \mathrm{W/m^2} \tag{28}$$

and

$$R_T = R_{si} + R_1 + R_2 + \ldots + R_n + R_{se} \quad \mathrm{m^2 \cdot K/W} \tag{29}$$

where R_l, R_2 ... are the thermal resistances of the individual layers and R_{si}, R_{se} are the thermal surface resistances of the interior and exterior surface.

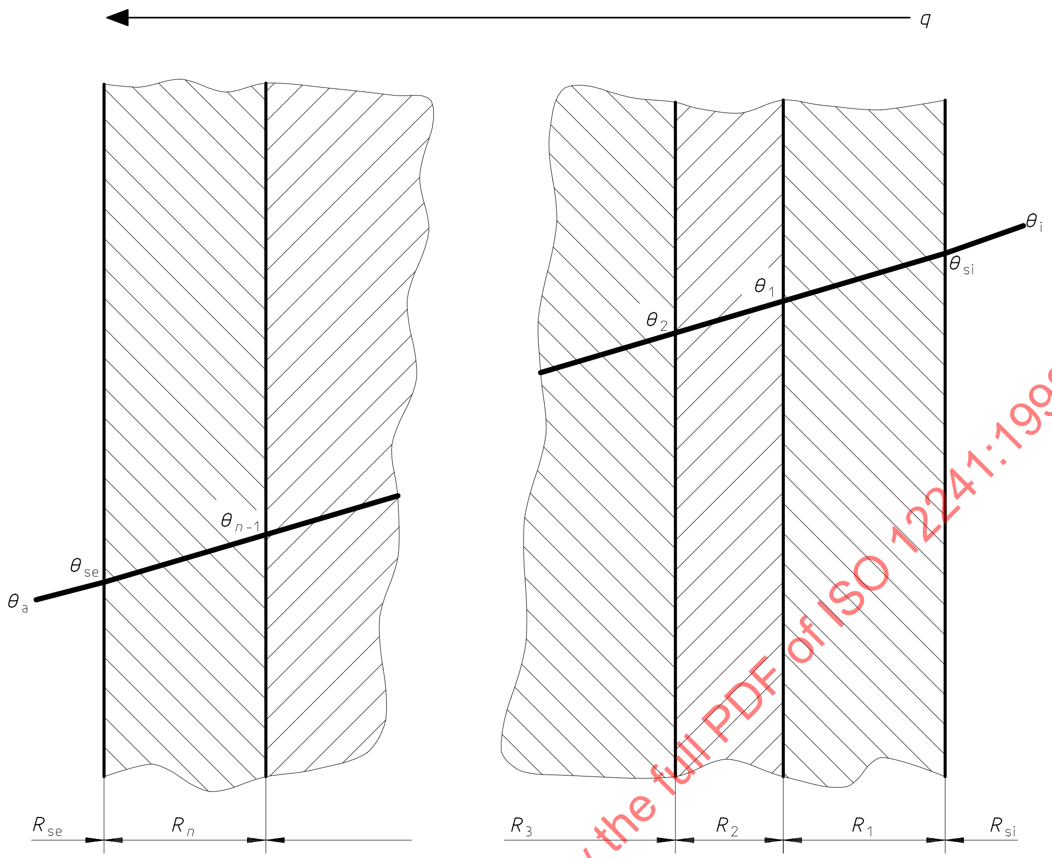


Figure 8: The temperature distribution for a multi-layer plane wall in relation to the thermal surface resistance and the thermal resistances of layers

The ratio between the resistance of each layer or the surface resistance with respect to the total resistance will give a measure of the temperature change across the particular layer or surface in K.

$$\theta_i - \theta_{si} = \frac{R_{si}}{R_T} \cdot (\theta_i - \theta_a) \quad \text{K} \qquad (30)$$

$$\theta_{si} - \theta_1 = \frac{R_1}{R_T} \cdot (\theta_i - \theta_a) \quad \text{K}$$

$$\theta_1 - \theta_2 = \frac{R_2}{R_T} \cdot (\theta_i - \theta_a) \quad \text{K}$$

.
.
.

$$\theta_{se} - \theta_a = \frac{R_{se}}{R_T} \cdot (\theta_i - \theta_a) \quad \text{K}$$

R_T is defined for plane walls according to equation (25), for cylindrical pipes according to eq. (26), and for spherical insulations by equation (27).

4.2 Surface temperature

The surface temperature can be calculated by using eq. (30)

For operational reasons it is often stipulated in practice that a certain surface temperature or temperature of the surface higher than that of the ambience should be maintained. The surface temperature is no measure for the quality of the thermal insulation. This depends not only on the heat transmission but also on operating conditions which cannot be readily determined or warranted by the manufacturer. These include among other things: ambient temperature, movement of the air, state of the insulation surface, effect of adjacent radiating bodies, meteorological conditions etc. Further, it will be necessary to make assumptions for the operating parameters. With all these parameters it is possible to estimate the required insulation thickness using equation (30) or diagram 1 (see reference [10]). It must be pointed out, however, that these assumptions will correspond to the subsequent operating conditions only in very rare cases.

Since an accurate registration of all relevant parameters will be impossible, the calculation of the surface temperature is inexact and the surface temperature cannot be warranted. The same restrictions apply to the warranty of the temperature difference between surface and air, also called excess temperature. Although it includes the effect of the ambient temperature on the surface temperature it assumes that the heat transfer by convection and radiation can be covered by a total heat transfer coefficient whose magnitude must also be known (see 4.1.2). However, this condition is generally not fulfilled because the air temperature in the immediate vicinity of the surface, which determines the convective heat transfer, mostly departs essentially from the temperature of other surfaces with which the insulation surface is in radiative exchange.

Diagram 1: Determination of insulating layer thickness for a pipe at a given heat flux density or for a set surface temperature (see next page)

$$C' = 2 \cdot \lambda \cdot \left[\left(\frac{|\theta_{im} - \theta_a|}{q} \right) - \frac{1}{h_{se}} \right] \qquad \text{(a)}$$

$$C' = \frac{2 \cdot \lambda}{h_{se}} \cdot \left[\left(\frac{|\theta_{im} - \theta_a|}{\theta_{se} - \theta_a} \right) - 1 \right] \qquad \text{(b)}$$

Example a: Set heat flux q

Θ_{im} = 300 °C λ = 0,068 W/(m·K)
Θ_a = 20 °C D = 0,324 m
h_{se} = 5,7 W/(m²·K) q = 63 W/m²

in accordance with Diagram 3:

$$C' = 2 \cdot \lambda \cdot \left(\frac{|\theta_{im} - \theta_a|}{q} - \frac{1}{h_{se}} \right)$$

$$= 2 \cdot 0{,}068 \left(\frac{|300 - 20|}{63} - \frac{1}{5{,}7} \right) = 0{,}58 \text{ m}$$

Result: d = 200 mm

Example b: Set surface temperature for dew prevention

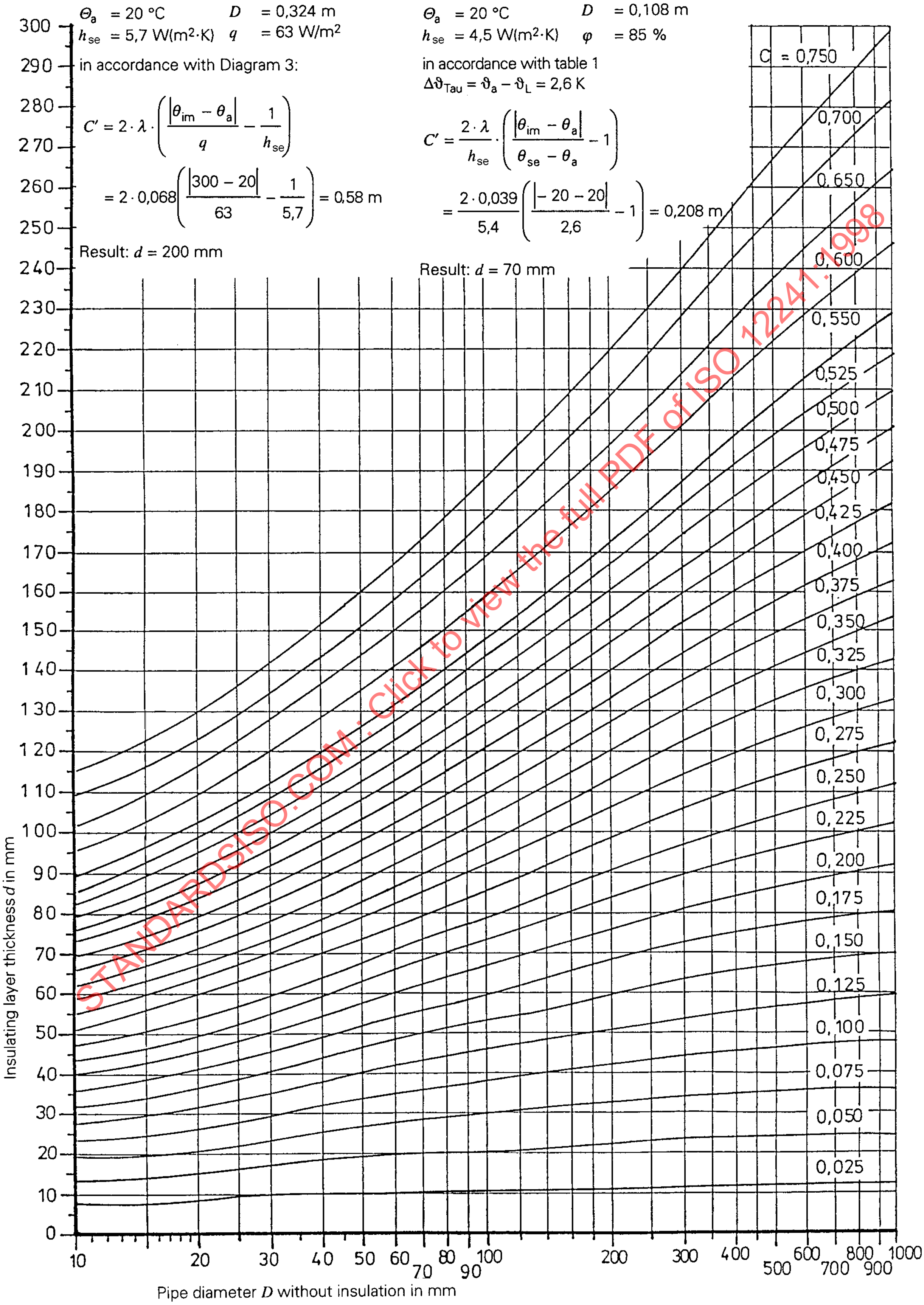
Θ_{im} = – 20 °C λ = 0,039 W/(m·K)
Θ_a = 20 °C D = 0,108 m
h_{se} = 4,5 W/(m²·K) φ = 85 %

in accordance with table 1
$\Delta\vartheta_{Tau} = \vartheta_a - \vartheta_L = 2{,}6$ K

$$C' = \frac{2 \cdot \lambda}{h_{se}} \cdot \left(\frac{|\theta_{im} - \theta_a|}{\theta_{se} - \theta_a} - 1 \right)$$

$$= \frac{2 \cdot 0{,}039}{5{,}4} \left(\frac{|-20 - 20|}{2{,}6} - 1 \right) = 0{,}208 \text{ m}$$

Result: d = 70 mm



The equation for the thickness parameter C' is derived from equations (24) and (26) by elementary transformations. Equation (a) permits calculation of the necessary insulation thickness for a given linear density of heat flow rate, whereas equation (b) allows calculation of the required insulation thickness for a given temperature difference between the pipe surface (with insulation) and the ambient temperature.

In both cases h_{se} must be assumed or calculated (see example B.7).

4.3 Prevention of surface condensation

The surface condensation does not only depend on the parameters affecting the surface temperature but also on the relative humidity of the surrounding air which very often cannot be stated accurately by the customer. It is all the more difficult to state the higher the relative humidity is, in which case fluctuations of the humidity or of the surface temperatures make themselves felt strongly. Unless other data are available assumptions have to be made as in diagram 3 (see clause 9) to calculate the necessary insulation thickness to prevent dew formation on pipes. Using equation (30) the necessary insulation thickness to prevent dew formation can be obtained by iterative techniques. The allowed temperature difference (in °C) between surface and ambient air for different relative humidities at the onset of dew formation is given in table 3.

5 Calculation of the temperature change in pipes, vessels and containers

5.1 Longitudinal temperature change in a pipe

To obtain the accurate value of the longitudinal temperature change in a pipe with a flowing medium, i.e. liquid or gas, the following equation applies:

$$\left|\theta_{fm} - \theta_a\right| = \left|\theta_{im} - \theta_a\right| \cdot e^{-\alpha \cdot l} \quad \text{K} \qquad (31)$$

where

$$\alpha = \frac{U_l \cdot 3{,}6}{\dot{m} \cdot c_p} \quad \text{m}^{-1} \qquad (32)$$

- θ_{fm} is the final temperature of the medium, in °C;
- θ_{im} is the initial temperature of the medium, in °C;
- θ_a is the ambient temperature, in °C;
- c_p is the specific heat capacity at constant pressure of the flowing medium, in kJ/(kg·K);
- $\dot{m}$ is the mass flow rate of the flowing medium, in kg/h;
- l is the length of the pipe, in m;
- U_l is the linear thermal transmittance, in W/(m·K).

Equations (31) and (32) can also be used for ducts with rectangular cross section. Then U_l has to be replaced by U_d (eq. 25).

Since, in practice, the allowed temperature change is often small, for approximate calculation the following equation applies:

$$\Delta\theta = \frac{q_l \cdot l \cdot 3{,}6}{\dot{m} \cdot c_p} \quad \text{K} \qquad (33)$$

- $\Delta\theta$ is the longitudinal temperature change, in K;
- q_l is the linear density of heat flow rate, in W/m.

For the calculation of q_l equation (24) can be used or equations (5) and (7) if the external surface coefficient can be neglected.

Equation (33) will yield results of satisfying accuracy only for relatively short pipes and a relatively small temperature change [$\Delta\theta \leq 0{,}06 \cdot (\theta_{im} - \theta_a)$].

5.2 Temperature change and cooling times in pipes, vessels and containers

The allowed cooling time t_v for a given temperature change is calculated by

$$t_v = \frac{(\theta_{im} - \theta_a) \cdot (m \cdot c_p) \cdot \ln \dfrac{(\theta_{im} - \theta_a)}{(\theta_{fm} - \theta_a)}}{q \cdot 3{,}6 \cdot A} \quad \text{h} \qquad (34)$$

where

$$q = \frac{(\theta_{im} - \theta_a)}{\dfrac{d}{\lambda} + \dfrac{1}{h_{se}}} \quad \text{W/m}^2 \qquad (35)$$

q is the density of heat flow rate, in W/m²;
A is the surface area of the container or vessel, in m²;
t_v is the cooling time, in h, producing the temperature drop;
m is the mass of contents, in kg;
c_p is the specific heat capacity, in kJ/(kg·K), of the medium.

For a spherical container $q \cdot A$ is replaced by the heat flow rate Φ_{sph} from equation (11).

The accurate calculation of the time-dependent temperature change is performed according to section 5.1, using equation (31) and replacing l by t and α by α'.

The approximate time-dependent temperature drop can be calculated by equation (36):

with $\alpha' = \dfrac{U \cdot A \cdot 3{,}6}{m \cdot c_P}$

$$\Delta\theta = \frac{q \cdot A}{m \cdot c_p} \cdot t \cdot 3{,}6 \quad \text{K} \qquad (36)$$

NOTE — In calculating the cooling time it is assumed that no heat is absorbed by the media during cooling. The obtained cooling time is the fastest, which means there is a safety factor built in by modelling (design calculation). For small containers the heat capacity of the container itself is taken into account and in equation (34) an analogous term as in equation (37) is added.

6 Calculation of cooling and freezing times of stationary liquids

6.1 Calculation of the cooling time for a given thickness of insulation to prevent the freezing of water in a pipe

It is impossible to prevent the freezing of a liquid in a pipe, although insulated, over an arbitrary long period of time.

As soon as the liquid (normally water) in the pipe is stationary the process of cooling starts. The linear density of heat flow q_l of a stationary liquid is determined by the energy stored in the liquid $c_{pw} \cdot m_w$ and in the pipe material $c_{pp} \cdot m_p$ as well as by the freezing enthalpy required to transform water to ice. If $m_p \cdot c_{pp} \ll m_w \cdot c_{pw}$ then $m_p \cdot c_{pp}$ may be neglected.

The time until freezing starts is calculated using the following equations :

$$t_{wp} = \frac{(\theta_{im} - \theta_a) \cdot (m_w \cdot c_{pw} + m_p \cdot c_{pp}) \cdot \ln \dfrac{(\theta_{im} - \theta_a)}{(\theta_{fm} - \theta_a)}}{q_{wp} \cdot 3{,}6 \cdot l} \quad \text{h} \qquad (37)$$

where

$$q_{wp} = \frac{\pi \cdot (\theta_{im} - \theta_a)}{\dfrac{1}{2 \cdot \lambda} \cdot \ln \dfrac{D_e}{D_i} + \dfrac{1}{h_{se} \cdot D_e}} \quad \text{W/m} \qquad (38)$$

and

l	is the length of the pipe, in m;
θ_{im}	is the initial medium temperature, in °C;
θ_{fm}	is the final medium temperature, in °C;
θ_a	is the ambient temperature in °C;
c_p	is the specific heat capacity, in kJ/(kg·K);
m_w	is the mass of water, in kg;
m_p	is the mass of the pipe, in kg.

In practice, for the calculation of q_{wp} the exterior thermal surface resistance should be neglected for insulated pipes.

If a comparison is made between uninsulated and insulated pipes the influence of the surface coefficient of the uninsulated pipe must be taken into consideration. The density of heat flow rate of the uninsulated pipe is given by:

$$q_l = h_{se} \cdot (\theta_{im} - \theta_a) \cdot 2 \cdot \pi \cdot D_e \quad \text{W/m} \qquad (39)$$

As an approximation the cooling time is given by:

$$t_{wp} = \frac{(m_w \cdot c_{pw} + m_p \cdot c_{pp}) \cdot (\theta_{im} - \theta_{fm})}{q_{wp} \cdot 3{,}6 \cdot l} \quad \text{h} \qquad (40)$$

The time until freezing starts is calculated by using the procedure above with θ_{fm} equal to the freezing point of the liquid.

Diagram 2 shows examples for the cooling time before freezing starts for a range of pipe diameters and ambient temperatures, for water initially at 5 °C.

6.2 Calculation of the freezing time of water in a pipe

The freezing time is dependent on the heat flow and the diameter of the pipe. It is given by:

$$t_{fr} = \frac{f}{100} \cdot \frac{\rho_{ice} \cdot \pi \cdot D_{ip}^2 \cdot \Delta h_{fr}}{q_{fr} \cdot 3{,}6 \cdot 4} \quad \text{h} \qquad (41)$$

with

$$q_{fr} = \frac{\pi \cdot (-\theta_a)}{\frac{1}{2 \cdot \lambda} \cdot \ln \frac{D_e}{D_i}} \quad \text{W/m} \qquad (42)$$

and

f is the percentage of water that is frozen;
D_{ip} is the interior pipe diameter, in m;
h_{fr} is the specific enthalpy = latent heat of ice formation = 334 kJ/kg;
ρ_{ice} is the density of ice at 0°; ρ_{ice} = 920 kg/m^3.

The percentage f of water that is frozen shall be chosen according to a requirement, i.e 25% (f = 25).

The allowable cooling time may be taken as well from diagram 2.

Due to the reduction of the cross-section of slides, taps and fittings cooling and freezing times are reduced as well. It is advised to decrease the cooling and freezing times t_{wp} and t_{fr} given in 6.1 and 6.2 by 25%. The allowed cooling times may also be taken from diagram 2.

7 Thermal bridges

Pipe mountings, supports and armatures may be thermal bridges which cannot be calculated by normal means. They cause additional heat losses, which can be taken into consideration in different ways. For pipes, components in the insulating layer like spacers and supports are taken into account by an additional term $\Delta\lambda$ to the reference thermal conductivity λ of the insulation material (see clause 9):

$$\lambda_{eff} = \lambda + \Delta\lambda \qquad (43)$$

The effect of valves, slide valves and flanges may be taken into account according to table 4 by adding a fictitious pipe length Δl to the given length l:

$$l_{eff} = l + \Delta l \qquad (44)$$

Like in pipes, the real temperature drop in containers is much affected by thermal bridges. A substantial increase in thickness of the insulation in containers is necessary.

8 Underground pipelines

Pipelines are laid in the ground with or without thermal insulation either in channels or directly in the soil.

8.1 Calculation of heat loss (single line)

At present, the following two methods are used for pipe-laying without channels:

The heat flux per metre of an underground pipe is calculated from

$$q_{l,E} = \frac{\theta_i - \theta_{sE}}{R_l' + R_E} \quad \text{W/m} \qquad (45)$$

θ_i is the medium temperature, in °C;
θ_{sE} is the surface temperature of the soil, in °C;
R_E is the thermal resistance for a pipe laid in homogeneous soil, in m·K/W;
λ_E is the thermal conductivity of the ambient soil, in W/(m·K);
H_E is the distance between the centre of the pipe and the surface, in m.

The thermal resistance for the ground (figure 9) is calculated in accordance with equation (46)

$$R_E = \frac{1}{2 \cdot \pi \cdot \lambda_E} \cdot \text{arcosh} \frac{2 \cdot H_E}{D_i} \quad \text{m} \cdot \text{K/W} \qquad (46)$$

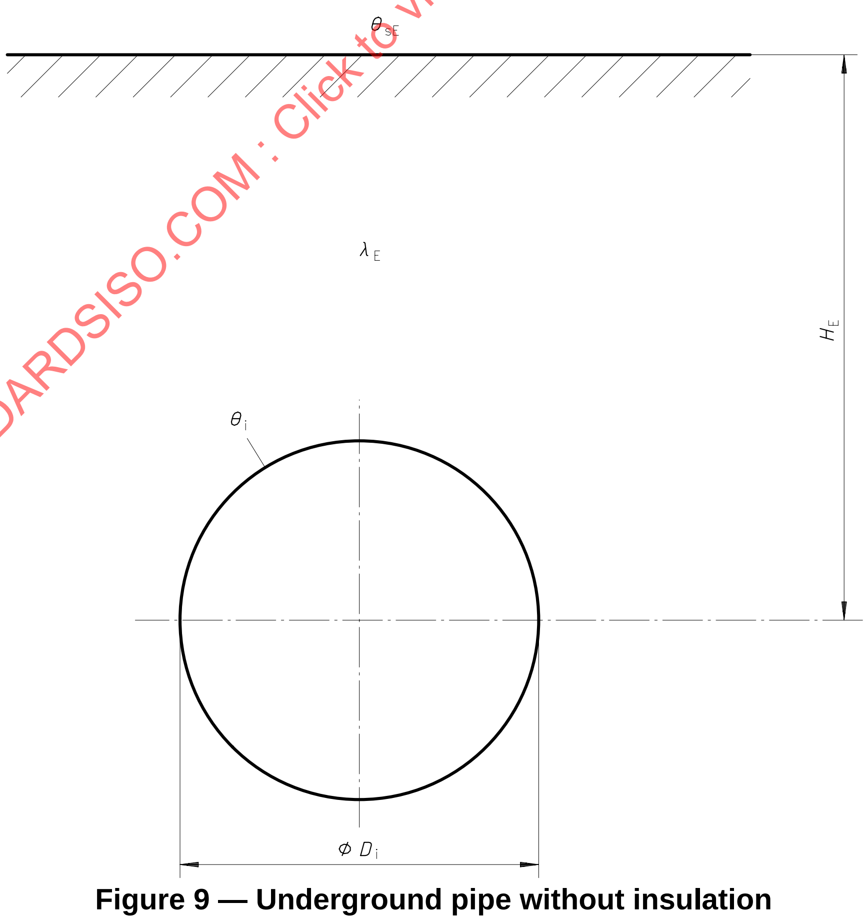


Figure 9 — Underground pipe without insulation

whereby equation (46) is simplified for $H_E/D_i > 2$ to

$$R_E = \frac{1}{2 \cdot \pi \cdot \lambda_E} \cdot \ln \frac{4 \cdot H_E}{D_i} \quad \text{m} \cdot \text{K/W} \tag{47}$$

For underground pipes with insulating layers in accordance with figure 10, the thermal resistance is calculated in accordance with equation

$$R_l{}' = \frac{1}{2 \cdot \pi} \sum_{j=1}^{n} \left(\frac{1}{\lambda_j} \cdot \ln \frac{D_{ej}}{D_{ij}} \right) \quad \text{m} \cdot \text{K/W} \tag{48}$$

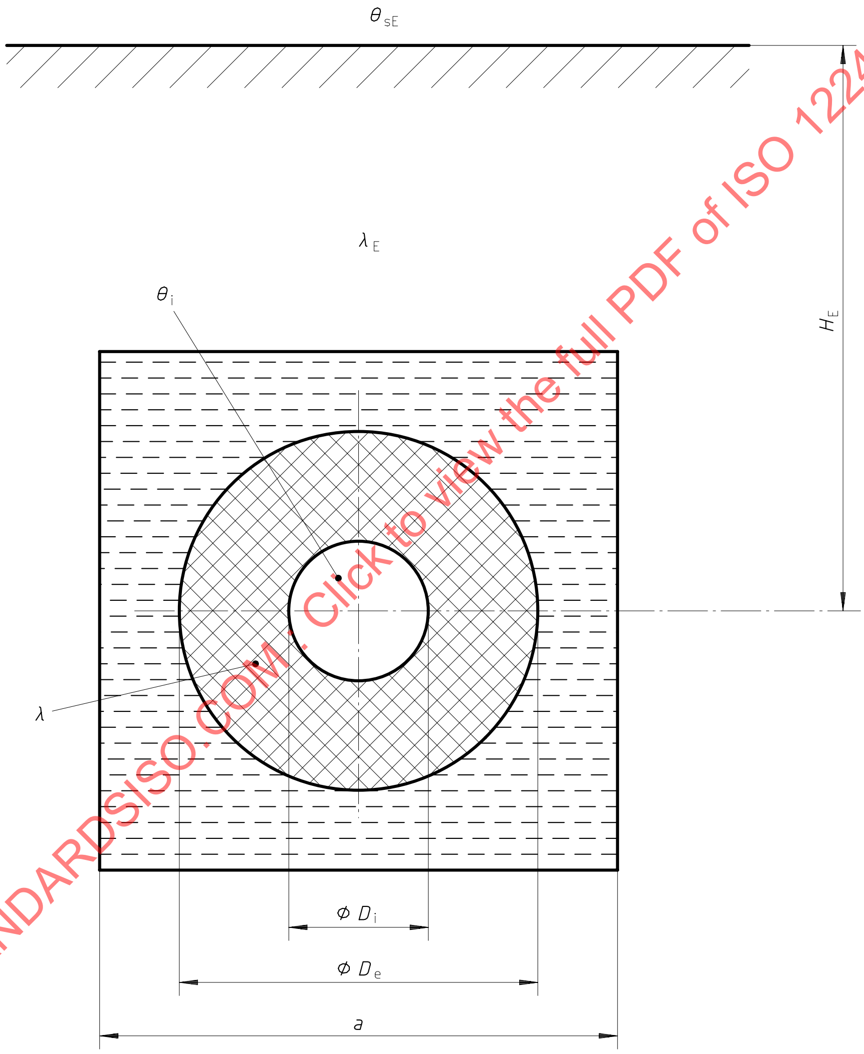


Figure 10 — Underground pipe comprising several concentric layers, e.g. consisting of insulating material and sheathing (e.g. jacket pipe), embedded in a bottoming (e.g. sand) with a square cross-section

The square cross-section of the outer layer with side length *a* is thereby taken into consideration with an equivalent diameter

$$D_n = 1{,}073 \cdot a \tag{49}$$

Internal diameter D_i is identical to D_o (where $j = 1$). The thermal resistance of the ground R_E results for this case at

$$R_E = \frac{1}{2 \cdot \pi \cdot \lambda_E} \cdot \operatorname{arcosh} \frac{2 \cdot H_E}{D_n} \quad \mathrm{m \cdot K/W} \tag{50}$$

Calculation methods are available for the determination of the heat flow rate and temperature field in the ground for several adjacent pipes, i.e double lines or laid systems, see references [12] to [14] in Annex C.

In the case of commonly used jacket pipes which are laid adjacent to each other, if $\lambda_1 \ll \lambda_E$, calculation as an individual pipe is generally sufficient as an initial approach, as the mutual effects can be disregarded.

For pipes embedded in insulating masses without additional insulation, the simplified calculation is not permissible.

9 Tables and diagrams

Table 3: The allowed temperature difference in °C between surface and ambient air for different relative humidities at the onset of dew formation

Ambient air temperature °C	Relative air humidities, %													
	30	35	40	45	50	55	60	65	70	75	80	85	90	95
-20	-	10,4	9,1	8,0	7,9	6,0	5,2	4,5	3,7	2,9	2,3	1,7	1,1	0,5
-15	12,3	10,8	9,6	8,3	7,3	6,4	5,4	4,6	3,8	3,1	2,5	1,8	1,2	0,6
-10	12,9	11,3	9,9	8,7	7,6	6,6	5,7	4,8	3,9	3,2	2,5	1,8	1,2	0,6
-5	13,4	11,7	10,3	9,0	7,9	6,8	5,8	5,0	4,1	3,3	2,6	1,9	1,2	0,6
0	13,9	12,2	10,7	9,3	8,1	7,1	6,0	5,1	4,2	3,5	2,7	1,9	1,3	0,7
2	14,3	12,6	11,0	9,7	8,5	7,4	6,4	5,4	4,6	3,8	3,0	2,2	1,5	0,7
4	14,7	13,0	11,4	10,1	8,9	7,7	6,7	5,8	4,9	4,0	3,1	2,3	1,5	0,7
6	15,1	13,4	11,8	10,4	9,2	8,1	7,0	6,1	5,1	4,1	3,2	2,3	1,5	0,7
8	15,6	13,8	12,2	10,8	9,6	8,4	7,3	6,2	5,1	4,2	3,2	2,3	1,5	0,8
10	16,0	14,2	12,6	11,2	10,0	8,6	7,4	6,3	5,2	4,2	3,3	2,4	1,6	0,8
12	16,5	14,6	13,0	11,6	10,1	8,8	7,5	6,3	5,3	4,3	3,3	2,4	1,6	0,8
14	16,9	15,1	13,4	11,7	10,3	8,9	7,6	6,5	5,4	4,3	3,4	2,5	1,6	0,8
16	17,4	15,5	13,6	11,9	10,4	9,0	7,8	6,6	5,4	4,4	3,5	2,5	1,7	0,8
18	17,8	15,7	13,8	12,1	10,6	9,2	7,9	6,7	5,6	4,5	3,5	2,6	1,7	0,8
20	18,1	15,9	14,0	12,3	10,7	9,3	8,0	6,8	5,6	4,6	3,6	2,6	1,7	0,8
22	18,4	16,1	14,2	12,5	10,9	9,5	8,1	6,9	5,7	4,7	3,6	2,6	1,7	0,8
24	18,6	16,4	14,4	12,6	11,1	9,6	8,2	7,0	5,8	4,7	3,7	2,7	1,8	0,8
26	18,9	16,6	14,7	12,8	11,2	9,7	8,4	7,1	5,9	4,8	3,7	2,7	1,8	0,9
28	19,2	16,9	14,9	13,0	11,4	9,9	8,5	7,2	6,0	4,9	3,8	2,8	1,8	0,9
30	19,5	17,1	15,1	13,2	11,6	10,1	8,6	7,3	6,1	5,0	3,8	2,8	1,8	0,9
35	20,2	17,7	15,7	13,7	12,0	10,4	9,0	7,6	6,3	5,1	4,0	2,9	1,9	0,9
40	20,9	18,4	16,1	14,2	12,4	10,8	9,3	7,9	6,5	5,3	4,1	3,0	2,0	1,0
45	21,6	19,0	16,7	14,7	12,8	11,2	9,6	8,1	6,8	5,5	4,3	3,1	2,1	1,0
50	22,3	19,7	17,3	15,2	13,3	11,6	9,9	8,4	7,0	5,7	4,4	3,2	2,1	1,0

Example: At an ambient temperature of 20° C and 70 % relative humidity the allowed surface temperature is 20 °C − 5,6 °C = 14,4 °C.

a) Valves and slide valves

To account for the presence of valves and slide valves in a piping system, add additional length in metres from table 4 to the real length of the pipeline before calculating the heat loss. These values account for the valve and its own flanges, but not for the flanges where the valve mounts in the piping system [see b)].

Values in table 4 assume typical industrial insulation thicknesses for the temperatures given, and thermal conductivities of λ = 0,08 W/(m·K) at 100 °C mean temperature, and λ = 0,10 W/(m·K) at 400 °C mean temperature.

Table 4: Additional heat losses due to components in a pipeline

Pipe diameter	D_i in m	0,10		0,50	
Medium temperature	θ_i in °C	100	400	100	400
Pipe inside	non-insulated valve 2/3 insulated valve 3/4 insulated valve	6 3,0 2,5	16 6,0 5,0	9 4,0 3,0	25 10,0 7,5
Pipe outside	non-insulated valve 2/3 insulated valve 3/4 insulated valve	15 6,0 4,5	22 8,0 6,0	19 7,0 6,0	32 11,0 8,5

b) Pair of flanges

To account for the heat losses from a pair of flanges in a piping system (including the flange pair when a valves is mounted):

Non-insulated flanges: From the table above, use one third of the length given for a valve of the same diameter. Add this to the real length of the piping before calculating the heat losses.

Insulated with flange boxes: To the real length of the piping, add one meter for each flange with flange box, before calculating the heat losses.

Insulated flanges: No adjustment required; calculate heat losses based on real length.

c) Pipe suspensions

Add to calculated heat loss (without previous compensation for other components).

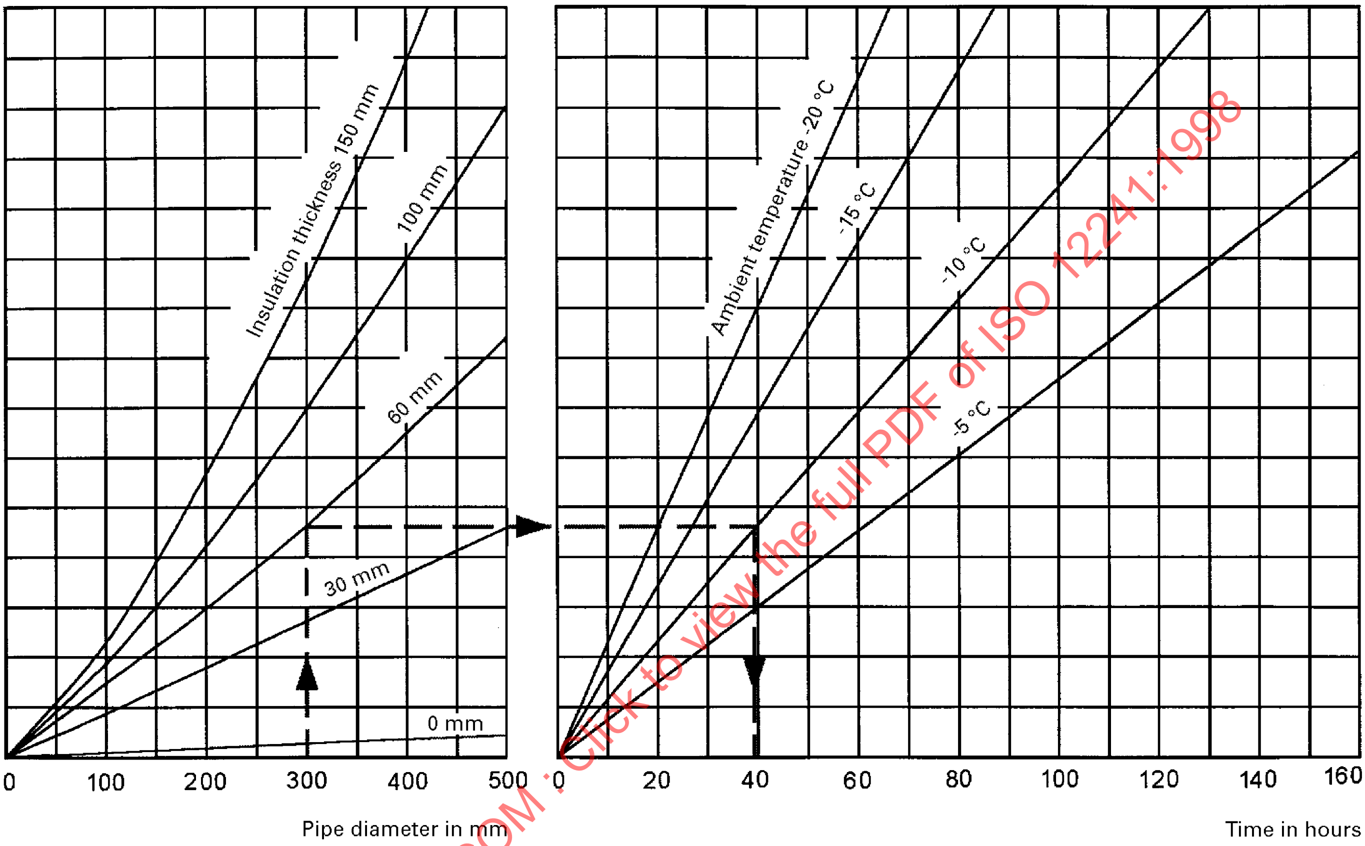
in interior spaces:	15 % of the heat loss
in the open air without wind:	20 % of the heat loss
in the open air with wind:	25 % of the heat loss

d) Supports for sheet-metal pipelines jackets

Additions to thermal conductivity:	for steel supports	0,010 W/(m·K)
	for ceramic supports	0,003 W/(m·K)

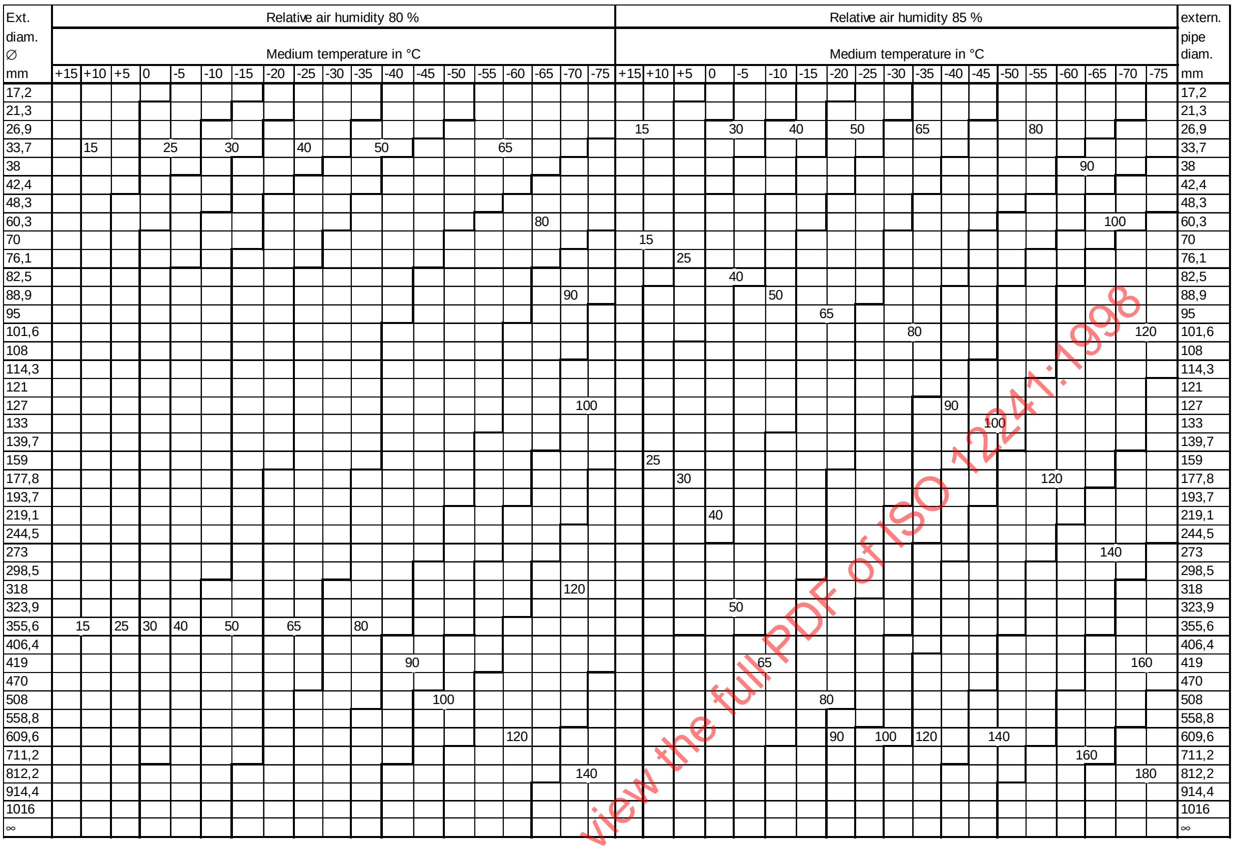
Diagram 2: Determination of cooling times from 5 °C to 0 °C

The maximum allowed cooling time of water in pipes of different diameter and with different insulation thicknesses at different ambient temperatures to avoid freezing of the water in the pipe. The initial water temperature θ = 5 °C, the wind speed v = 5 m/s, λ = 0,04 W/(m·K), h_e=20 W/(m^{2}K)



Example: For a given pipe diameter of 300 mm with an insulation thickness of 60 mm and an ambient air temperature of −10 °C, the maximum allowed cooling time is 40 h.

Diagram 3: The required insulation thickness to prevent dew formation



The required insulation thickness in mm for refrigerant pipes of different diameters and different temperatures at different relative humidities of the ambient air.

The thermal conductivity of the insulation at $\theta = 10$ ° is $\lambda = 0{,}04$ W/(m·K),

The thermal conductivity of the insulation at $\theta = -100$ °C is $\lambda = 0{,}033$ W/(m·K),

Ambient air temperature is 20 °C, $h_{se} \approx 6$ W/(m^2·K)

Annex A
(informative)

Comments on thermal conductivity

There is a distinction between the laboratory and the declared thermal conductivity as well as the design value of the thermal conductivity of an insulation.

A.1 Laboratory thermal conductivity λ_{lab}

An important description of the quality of a thermal insulation (see ISO 8302) is the laboratory thermal conductivity, measured with a guarded hot plate in accordance with ISO 8302 or the heat flow meter in accordance with ISO 8301 on plane samples. It is dependent on the kind of thermal insulation, its composition, structure and density and on the temperature.

The laboratory thermal conductivity is measured with unused, dry samples in the given temperature regime in steps of 20 K, 50 K or 100 K (see ISO 9251). The laboratory thermal conductivity for plane products is called $\lambda_{lab,\,P}$ and is given as a function of the mean test temperature.

The laboratory thermal conductivity of dry thermal insulations of hollow cylindrical specimens like pipe sections (see ISO 9229) in different diameters and thicknesses is determined with the pipe testing apparatus according to ISO 8497. This value includes besides the temperature difference parameters, which are due to the test conditions, for example the influence of longitudinal or transverse joints and of single or multiple insulation layers, i.e. effects of workmanship. It is given in the relevant temperature regime as a function of the mean temperature. This value is called $\lambda_{lab,R}$.

A.2 Declared thermal conductivity λ_{dec}

The declared thermal conductivity stated by the manufacturer must take production-related fluctuations into consideration. The declared value for plane products is based on the laboratory thermal conductivity $\lambda_{lab,p}$ and the declared value of pipe sections on $\lambda_{lab,R}$. A method to derive the declared thermal conductivity from the laboratory thermal conductivity is given in ISO 13787. Another approach for the derivation of the declared and designed values is given in reference [8].

A.3 Design value λ_d

The design value of an insulation is warranted by the contractor who does the actual application. The evaluation of the design value is done either on the basis of the declared thermal conductivity or on the basis of the laboratory thermal conductivity.

In addition the thermal conductivity has to be increased by allowance factors taking into account the influences of the actual temperature difference of the installed material, of workmanship, changes in density or structural changes (see BS 5422).

If, in the case of pipe insulation, the declared thermal conductivity is used as basis, the value might already include these parameters. However, one has to prove that they are of sufficient magnitude.

If a vertical thermal insulation is permeable to air and if there exists an air layer within the insulation, then an appropriate allowance has to be considered.

The influence of other thermal bridges, which are due to the installation of the insulation material such as spacers, carrying and support constructions has to be included with allowance factors according to clause 7 and table 4 of this standard.

The resultant design value is only a warrantable quantity if the allowance factors due to thermal bridges are known with sufficient accuracy.

For other kinds of constructions the allowance factors have to be determined either experimentally or by calculation.

NOTE — More advanced calculation techniques for thermal bridges are given in reference [14].

The laboratory thermal conductivity of a specimen, taken from an installed insulation may be only checked if no material or structural changes occurred during mounting.