INTERNATIONAL STANDARD

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Acoustics and vibration — Laboratory measurement of vibro-acoustic transfer properties of resilient elements —

Part 1:
Principles and guidelines

Acoustique et vibrations — Mesurage en laboratoire des propriétés de transfert vibro-acoustique des éléments élastiques —

Partie 1: Principes et lignes directrices

Reference number
ISO 10846-1:1997(E)

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 10846-1 was prepared jointly by Technical Committees ISO/TC 43, *Acoustics*, Subcommittee SC 1, *Noise*, and ISO/TC 108, *Mechanical vibration and shock*.

Annexes A to E of this part of ISO 10846 are for information only.

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Introduction

Passive vibration isolators of various kinds are used to reduce the transmission of vibrations. Examples are automobile engine mounts, elastic supports for buildings, elastic mounts and flexible shaft couplings for shipboard machinery and small isolators in household appliances.

This part of ISO 10846 serves as an introduction and a guide to parts 2 to 5 of ISO 10846, which describe laboratory measurement methods for the determination of the most important quantities which govern the transmission of vibrations through linear isolators, i.e. frequency-dependent dynamic stiffnesses.

This part of ISO 10846 provides the theoretical background, the principles of the methods, the limitations of the methods and guidance for the selection of the most appropriate standard of the series.

The laboratory conditions described in all parts of ISO 10846 include the application of static preload.

The results of the methods are useful for isolators which are used to prevent low-frequency vibration problems and to attenuate structure-borne sound. The methods are not sufficiently appropriate to characterize completely isolators which are used to attenuate shock excursions.

Acoustics and vibration — Laboratory measurement of vibro-acoustic transfer properties of resilient elements —

Part 1:
Principles and guidelines

1 Scope

This part of ISO 10846 explains the principles underlying parts 2 to 5 of ISO 10846 for determining the transfer properties of vibration isolators from laboratory measurements, and provides assistance in the selection of the appropriate part of this series.

This part of ISO 10846 is applicable to vibration isolators which are used to reduce:

a) the transmission of audiofrequency vibrations (structure-borne sound, 20 Hz to 20 kHz) to a structure which may, for example, radiate fluid-borne sound (airborne, waterborne, or other);

b) the transmission of low frequency vibrations (typically 1 Hz to 80 Hz) which may, for example, act upon humans or cause damage to structures when vibration is too severe.

The data obtained with the measurement methods which are outlined in this part of ISO 10846 and further detailed in parts 2 to 5 of ISO 10846 can be used for:

— product information provided by manufacturers and suppliers;

— information during product development;

— quality control;

— computation of the transfer of vibrations through isolators.

The conditions for the validity of the measurement methods are

a) linearity of the vibrational behaviour of the isolator (this includes elastic elements with non-linear static load-deflection characteristics as long as the elements show approximate linearity for vibrational behaviour for a given static preload);

b) the contact interfaces of the vibration isolator with the adjacent source and receiver structures can be considered as point contacts.

2 Normative reference

The following standard contains provisions which, through reference in this text, constitute provisions of this part of ISO 10846. At the time of publication, the edition indicated was valid. All standards are subject to revision, and parties to agreements based on this part of ISO 10846 are encouraged to investigate the possibility of applying the most recent edition of the standard indicated below. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 2041:1990, *Vibration and shock — Vocabulary.*

3 Definitions

For purposes of this part of ISO 10846, the definitions given in ISO 2041 and the following apply.

3.1
resilient element
(see vibration isolator)

3.2
vibration isolator
isolator designed to attenuate the transmission of vibration in a frequency range [ISO 2041:1990, 2.110]

3.3
elastic support
vibration isolator suitable for supporting a part of the mass of a machine, a building or another type of structure

3.4
blocking force
F_b
dynamic force at the output side of a vibration isolator which results in zero displacement output

3.5
dynamic driving point stiffness
$k_{1,1}$
frequency-dependent complex ratio of the force on the input side of a vibration isolator with the output side blocked to the complex displacement on the input side during simple harmonic vibration

NOTE 1 $k_{1,1}$ may depend on the static preload, temperature and other conditions.

NOTE 2 At low frequencies $k_{1,1,}$ is solely determined by elastic and dissipative forces. At higher frequencies inertial forces in the resilient element play a role as well.

3.6
dynamic transfer stiffness
$k_{2,1}$
frequency-dependent complex ratio of the force on the blocked output side of a vibration isolator to the complex displacement on the input side during simple harmonic vibration

NOTE 1 $k_{2,1}$ may depend on the static preload, temperature and other conditions.

NOTE 2 At low frequencies $k_{2,1}$ is solely determined by elastic and dissipative forces and $k_{2,1} = k_{1,1}$ At higher frequencies inertial forces in the resilient element play a role as well and $k_{2,1} \neq k_{1,1}$

3.7
loss factor of resilient element
η
frequency-dependent ratio of the imaginary part of $k_{2,1}$ to the real part of $k_{2,1}$ (i.e. tangent of the phase angle of $k_{2,1}$) in the low-frequency range where inertial forces in the element are negligible

3.8
point contact
contact area which vibrates as the surface of a rigid body

3.9
linearity
property of the dynamic behaviour of a vibration isolator if it satisfies the principle of superposition

NOTE 1 The principle of superposition can be stated as follows: if an input $x_1(t)$ produces an output $y_1(t)$ and in a separate test an input $x_2(t)$ produces an output $y_2(t)$, superposition holds if the input $\alpha x_1(t) + \beta x_2(t)$ produces the output $\alpha y_1(t) + \beta y_2(t)$. This must hold for all values of α, β and $x_1(t)$, $x_2(t)$; α and β are arbitrary constants.

NOTE 2 In practice the above test for linearity is impractical and a limited check of linearity is done by measuring the dynamic transfer stiffness for a range of input levels. For a specific preload, if the dynamic transfer stiffness is nominally invariant the system can be considered linear. In effect this procedure checks for a proportional relationship between the response and the excitation.

3.10
direct method
method in which either the input displacement, velocity or acceleration and the blocking output force are measured

3.11
indirect method
method in which the transmissibility (for displacement, velocity or acceleration) of an isolator is measured, with the output loaded by a mass/effective mass

3.12
driving point method
method in which either the input displacement, velocity or acceleration and the input force are measured, with the output side of the vibration isolator blocked

4 Selection of appropriate International Standard

Table 1 provides guidance for the selection of the appropriate part of ISO 10846.

Table 1 — Guidance for selection

	International Standard and method type			
	ISO 10846-2 Direct method	ISO 10846-3 Indirect method	ISO 10846-4 Indirect method	ISO 10846-5 Driving point method
Type of vibration isolator	support	support	other than support	support
Examples	resilient mountings for instruments, equipment, machinery and buildings		bellows, hoses, elastic shaft couplings, power supply cables	see under ISO 10846-2 and ISO 10846-3
Frequency range	1 Hz to f_1 f_1 dependent on test rig; typically (but not limited to) 300 Hz < f_1 < 500 Hz	f_2 to f_3 f_2 typically (but not limited to) 20 Hz to 50 Hz. For very stiff mountings f_2 > 100 Hz. f_3 typically 2 kHz to 5 kHz, but dependent on the test rig	f_2 to f_3 f_2 typically (but not limited to) 20 Hz to 50 Hz. For very stiff elements f_2 > 100 Hz. f_3 typically 2 kHz to 5 kHz, but dependent on the test rig	1 Hz to f_4 f_4 typically (but not limited to) < 100 Hz
Translational components	1, 2 or 3	1, 2 or 3	1, 2 or 3	1, 2 or 3
Rotational components	none	informative annex	informative annex	none
Classification of method	engineering	engineering	engineering/survey	engineering/survey
NOTE At the low-frequency end, the direct method and the driving point method yield the same result.				

Further guidance is given in clauses 5 and 6.

5 Theoretical background

5.1 Dynamic transfer stiffness

This clause explains why the dynamic transfer stiffness is most appropriate to characterize the vibro-acoustic transfer properties of isolators for many practical applications. It also indicates briefly for which special situations other vibro-acoustic isolator properties, of which the measurement is not covered in ISO 10846, would be needed in addition.

The dynamic transfer stiffness, as defined in 3.6, is determined by the elastic, inertia and damping properties of the isolator. The reason for choosing a presentation of test results in terms of a stiffness is the practical consideration that it complies with data of static and/or low-frequency dynamic stiffness which are commonly used. The additional importance of inertial forces (i.e. elastic wave effects in the isolators) makes the dynamic transfer stiffness at high frequencies more complex than at low frequencies. Because at low frequencies only elastic and damping forces are important, the low-frequency dynamic stiffness is only weakly dependent on frequency due to material properties.

NOTE — For many vibration isolators, static stiffness and low-frequency dynamic transfer stiffness are different.

In principle the dynamic transfer stiffness of vibro-acoustic isolators is dependent on static preload and temperature. In the following theory linearity, as defined in 3.9, is assumed. See annex D for further information.

Relationships between the dynamic transfer stiffness and other quantities are listed in annex A. These relationships imply that, for the actual performance of the tests, only practical considerations will determine whether displacements, velocities or accelerations are measured. However, for presentation of the results in agreement with the other parts of ISO 10846, appropriate conversions may be needed.

5.2 Dynamic stiffness matrix of vibration isolators

5.2.1 General concept

A familiar approach to the analysis of complex vibratory systems is the use of stiffness — compliance — or transmission matrix concepts. The matrix elements are basically special forms of frequency-response functions; they describe linear properties of mechanical and acoustical systems. On the basis of the knowledge of the individual subsystem properties, corresponding properties of assemblies of subsystems can be calculated. The three matrix forms mentioned above are interrelated and can be readily transformed amongst themselves [5]. However, only stiffness-type quantities are specified in ISO 10846 for the experimental characterization of isolators under static preload.

The general conceptual framework for the proposed isolator characterization is shown in figure 1.

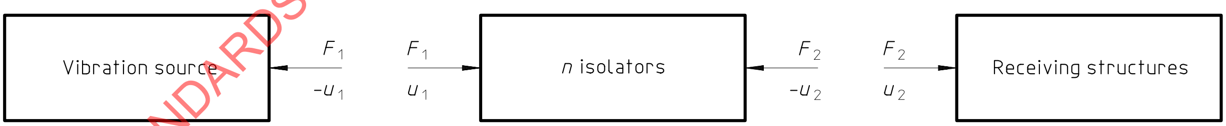


Figure 1 — Block diagram of source/isolators/receiver system

The system consists of three blocks, which respectively represent the vibration source, a number n of isolators and the receiving structures. A point contact is assumed at each connection between source and isolator and between isolator and receiver. To each connection point a force vector $\boldsymbol{F}$ containing three orthogonal forces and three orthogonal moments and a displacement vector $\boldsymbol{u}$ containing three orthogonal translational and three orthogonal rotational components are assigned. In figure 1 just one component of each of the vectors F_1, u_1, F_2 and u_2 is shown. These vectors contain $6n$ elements, where n denotes the number of isolators.

To show that the blocked transfer stiffness, defined in 3.6 as dynamic transfer stiffness, is suitable for isolator characterization in many practical cases, the discussion will proceed from the simplest case of unidirectional vibration to the multidirectional case for a single isolator.

5.2.2 Single isolator, single vibration direction

For unidirectional vibration of a single vibration isolator, the isolator equilibrium may be expressed by the following stiffness equations:

$$F_1 = k_{1,1} u_1 + k_{1,2} u_2 \tag{1}$$

$$F_2 = k_{2,1} u_1 + k_{2,2} u_2 \tag{2}$$

where

$k_{1,1}$ and $k_{2,2}$ are driving point stiffnesses when the isolator is blocked at the opposite side (i.e. $u_2 = 0$, $u_1 = 0$, respectively);

$k_{1,2}$ and $k_{2,1}$ are blocked transfer stiffnesses, i.e. they denote the ratio between the force on the blocked side and the displacement on the driven side. $k_{1,2} = k_{2,1}$ for passive isolators, because passive linear isolators are reciprocal.
Due to additional inertial forces, $k_{1,1}$ and $k_{2,2}$ become different at higher frequencies. At low frequencies only elastic and damping forces play a role, making all $k_{i,j}$ equal.

NOTE — These equations are for single frequencies. F_i and u_i are phasors and $k_{i,j}$ are complex quantities.

The matrix form of equations (1) and (2) is

$$\boldsymbol{F} = [k]\, \boldsymbol{u} \tag{3}$$

with the dynamic stiffness matrix

$$[k] = \begin{bmatrix} k_{1,1} & k_{1,2} \\ k_{2,1} & k_{2,2} \end{bmatrix} \tag{4}$$

For excitation of the receiving structure via the isolator

$$k_{\mathrm{r}} = -\frac{F_2}{u_2} \tag{5}$$

where k_{r} denotes the dynamic driving point stiffness of the receiver. The minus sign is a consequence of the convention adopted in figure 1.

From equations (2) and (5) it follows that

$$F_2 = \frac{k_{2,1}}{1 + \dfrac{k_{2,2}}{k_{\mathrm{r}}}} u_1 \tag{6}$$

Therefore, for a given source displacement u_1, the force F_2 depends both on the isolator driving point dynamic stiffness and on the receiver driving point dynamic stiffness. However, if $|k_{2,2}| < 0,1|k_{\mathrm{r}}|$, then F_2 approximates the so-called blocking force to within 10 %, i.e.

$$F_2 \approx F_{2,\,\mathrm{blocking}} = k_{2,1}\, u_1 \tag{7}$$

Because vibration isolators are only effective between structures of relatively large dynamic stiffness on both sides of the isolator, equation (7) represents the intended situation at the receiver side. This forms the background for the measuring methods of ISO 10846. Measurement of the blocked transfer stiffness (or a directly related function) for an isolator under static preload is easier than measurement of the complete stiffness matrix (or the complete transfer matrix). Moreover it forms the representative isolator characteristic under the intended circumstances.

NOTE — In cases that the condition $|k_{2,2}| << |k_{\mathrm{r}}|$ is not fulfilled, equation (6) also shows that $k_{2,2}$ and k_{r} need to be known to predict F_2 for a given source displacement u_1.

5.2.3 Single isolator, six vibration directions

If forces and motions at each interface can be characterized by six orthogonal components (three translations, three rotations), the isolator may be described as a 12-port [11]. The matrix form of the 12 dynamic stiffness equations is equal to equation (3), where now

$$\boldsymbol{u} = \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix}, \ \boldsymbol{F} = \begin{Bmatrix} F_1 \\ F_2 \end{Bmatrix} \tag{8}$$

are the vectors of the six displacements, six angles of rotations, six forces and six moments. The 12×12 dynamic stiffness matrix may be decomposed into four 6×6 submatrices

$$[k] = \begin{bmatrix} [k_{1,1}] & [k_{1,2}] \\ [k_{2,1}] & [k_{2,2}] \end{bmatrix} \tag{9}$$

where

$[k_{1,1}]$ and $[k_{2,2}]$ are (symmetric) matrices of the driving point stiffnesses;

$[k_{1,2}]$ and $[k_{2,1}]$ are the blocked transfer stiffness matrices.

Reciprocity implies that these transfer matrices equal their transpose.

Again, if the receiver has relatively large driving point dynamic stiffnesses compared to the isolator, the forces exerted on the receiver approximate the blocking forces:

$$F_{2,\text{ blocking}} = [k_{2,1}] \cdot u_1 \tag{10}$$

Therefore, the blocked transfer stiffnesses are appropriate quantities to characterize vibro-acoustic transfer properties of isolators, and also in the case of multidirectional vibration transmission.

5.3 Number of relevant blocked transfer stiffnesses

For the general case the blocked transfer stiffness matrix $[k_{2,1}]$ of a single isolator contains 36 elements. However, structural symmetry causes most elements to be zero. The most symmetrical shapes (a circular cylinder or a square block) have 10 non-zero elements, i.e. five different pairs (see annex B and reference [11]).

In practical situations the number of elements relevant for characterization of the vibro-acoustic transfer is usually even smaller than the number of non-zero elements. In many cases it will be sufficient to take into account only one, two or three diagonal elements for translation vibration, i.e. for only one vibration direction (often vertical) or for two or three perpendicular directions (see annex C for further discussion). For these translational directions, measurement methods will be defined in parts 2 to 5 of ISO 10846.

For some special technical cases, rotational degrees of freedom also play a significant role (see annex C). Although it is not considered as a subject for standardization in ISO 10846, reference is made in 6.3.5 to literature that describes how rotational elements may be handled in the same way as the translational elements. ISO 10846-3 and ISO 10846-4 have informative annexes relevant to this subject.

5.4 Flanking transmission

The model shown in figure 1 and of equations (1) to (10) is correct under the assumption that the isolators form the only transfer path between the vibration source and the receiving structure. In practice there may be mechanical or acoustical parallel transmission paths which cause flanking transmission. For any measurement method of isolator properties, the possible interference of such flanking with proper measurements has to be minimized.

5.5 Loss factor

The objective of ISO 10846 is to standardize measurements of the frequency-dependent dynamic transfer stiffnesses $k_{2,1}$ of resilient elements. Certain users of ISO 10846 also will be interested in the damping properties of isolators. However, ISO 10846 does not standardize the measurement of damping properties of isolators because this would become overly complex. Nevertheless, in parts 2 to 5 of ISO 10846, descriptions are given of how phase data of the complex dynamic transfer stiffness $k_{2,1}$ can be optionally used to give information about the damping properties. The discussion in this subclause is given as background information for the procedures.

For the purposes of the discussion it is sufficient to consider the case of 5.2.2, i.e. a single isolator and a single vibration direction. Because only measurements with a blocked output side are considered in ISO 10846, the phasor equations (1) and (2) are reduced to

$$F_1 = k_{1,1} u_1 \tag{11}$$

$$F_2 = k_{2,1} u_1 \tag{12}$$

At low frequencies, where inertial forces (e.g. wave effects) play no role, there is a simple relationship between the phase angle of the dynamic transfer stiffness and the damping properties of the resilient element. At these low frequencies, the frequency-dependent stiffness can be approximated by

$$k \approx k_{1,1} \approx k_{2,1} \tag{13}$$

This complex low-frequency dynamic stiffness can be written as

$$k = k_0(1+\mathrm{j}\eta) \tag{14}$$

where k_0 denotes the real part. The frequency-dependent loss factor η in equation (14) characterizes the damping of the resilient element at low frequencies (see 3.7).

The relationship between the loss factor and the phase angle ϕ of k is given by

$$\eta = \tan \phi \tag{15}$$

Therefore, the loss factor of a resilient element can be estimated according to

$$\eta \approx \tan \phi_{2,1} \tag{16}$$

where $\phi_{2,1}$ is the phase angle of the dynamic transfer stiffness $k_{2,1}$.

The following points should be kept in mind.

a) The measurement of small loss factors using equation (16), is extremely sensitive to phase measurement errors [12]. However, for rubber-type resilient elements this problem is not critical except at frequencies below a few hertz.

b) For higher frequencies, where the approximations of equation (13) are no longer valid, it is no longer correct to use equation (16) as a characterization of the damping properties of the resilient element. Although there are no simple and strict criteria for when this occurs, a rather sudden change of the slope of η with increasing frequency is usually a good indication that equation (16) can no longer be used.

6 Measurement principles

6.1 Dynamic transfer stiffness

The dynamic transfer stiffness is dependent on frequency. In addition it is also dependent on static preload and, in many cases, on temperature. Three methods are in use to obtain the appropriate test data. Because they are complementary with respect to their strong and weak points, they are all described in ISO 10846.

The direct method requires the measurement of input displacement (velocity, acceleration) and blocking output force. At low frequencies, where the driving point stiffness and the transfer stiffness are equal, both force and displacement may be measured on the driven side of the isolator. This method is called the driving point method.

The indirect method uses a measurement of vibration transmissibility (for displacement, velocity or acceleration). To obtain the blocking output force, the isolator is terminated with a mass which provides a large dynamic stiffness. In a specified frequency range, the product of the measured displacement and the known point dynamic stiffness of the termination should provide a good approximation of the blocking force.

The basic features of these methods and the general requirements for their proper use are described in this part of ISO 10846. Detailed requirements are specified in parts 2 to 5 of ISO 10846.

6.2 Direct method

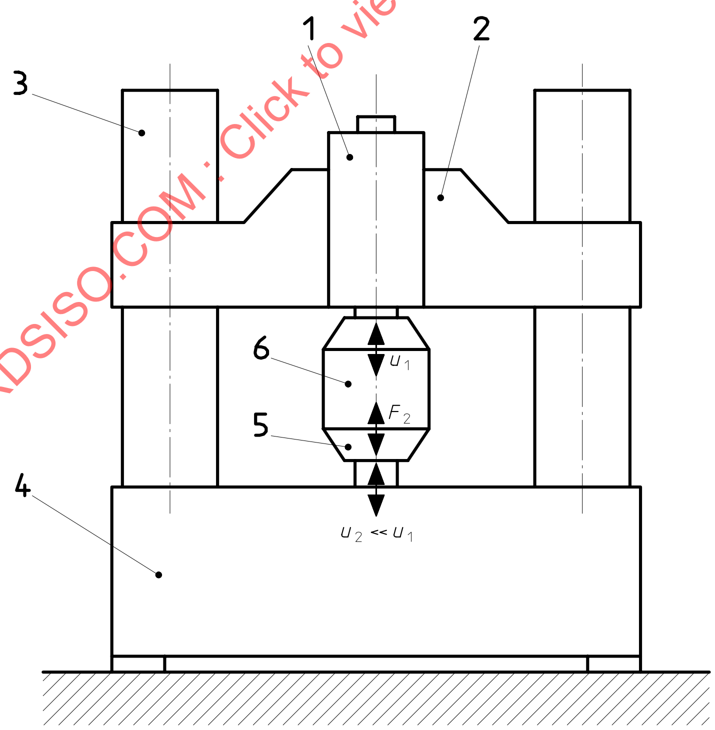
6.2.1 Basic test set-up

The basic principle for the measurement of the dynamic transfer stiffness is shown in figure 2.

The isolator under test is placed between a vibration exciter on the input side and a rigid termination on the output side. A dynamic force transducer is placed between the isolator and the rigid termination. Often it will be necessary to insert force-distribution plates. These serve to approximate point contact conditions and unidirectional motion. For example, in the case of a large isolator flange supported by a small force transducer only, the flange vibration and therefore the dynamic transfer stiffness may deviate significantly from that in practice. For large isolators with a high static preload, stability requirements may make it necessary to measure the force with a number of force transducers.

6.2.2 Measurement quantities

The dynamic quantities to be measured are the force and either the displacement, velocity or acceleration.



Key

1 Hydraulic actuator (static preload and dynamic excitation)
2 Moveable traverse
3 Columns
4 Rigid foundation
5 Force measurement system
6 Test object

Figure 2 — Example of a typical test set-up for the direct method

6.2.3 Measurement under static preload

Because the dynamic transfer stiffness may be heavily dependent on static load, tests should be provided under nominal static load conditions. Often special test rigs are needed to apply such loads. Combined static pre-loading and vibration is typically applied using a hydraulic actuator on top. However, test rigs with separated components for pre-loading and for vibration excitation are also considered in ISO 10846.

6.2.4 Frequency limitations of the direct method

The frequency range of validity of the direct method is mainly determined by the test rig properties. One limitation is given by the actuator bandwidth. Another limitation is often determined by the occurrence of flanking transmission at high frequencies through the frame which is used to apply the static preload. The fundamental frame mode, which usually causes serious problems, is determined by the mass of the traverse and the longitudinal stiffness of the vertical beams. A typical upper frequency of 300 Hz < f < 500 Hz is mentioned in table 1. These values are reported by owners of test rigs with a static load capacity up to 100 kN (see reference [10]). Of course for smaller and more compact rigs this upper limit would move to higher frequencies. For example, for small size elements with small preloads, very simple test set-ups may suffice, having a frequency range for valid measurements up to several kilohertz.

However, generally speaking, the indirect method (see below) gives better possibilities with respect to high frequency measurements. The indirect method gives less flanking transmission because the test isolator is dynamically uncoupled from the load frame.

6.2.5 Directions of vibration

The direct method can be applied for translational and rotational vibration both in the normal load direction and in the transverse directions. However, use of the direct method for rotational vibration is not considered in this part of ISO 10846.

6.3 Indirect method

6.3.1 Basic test arrangement

The basic principle for the measurement of blocked transfer stiffness is illustrated by the examples given in figure 3.

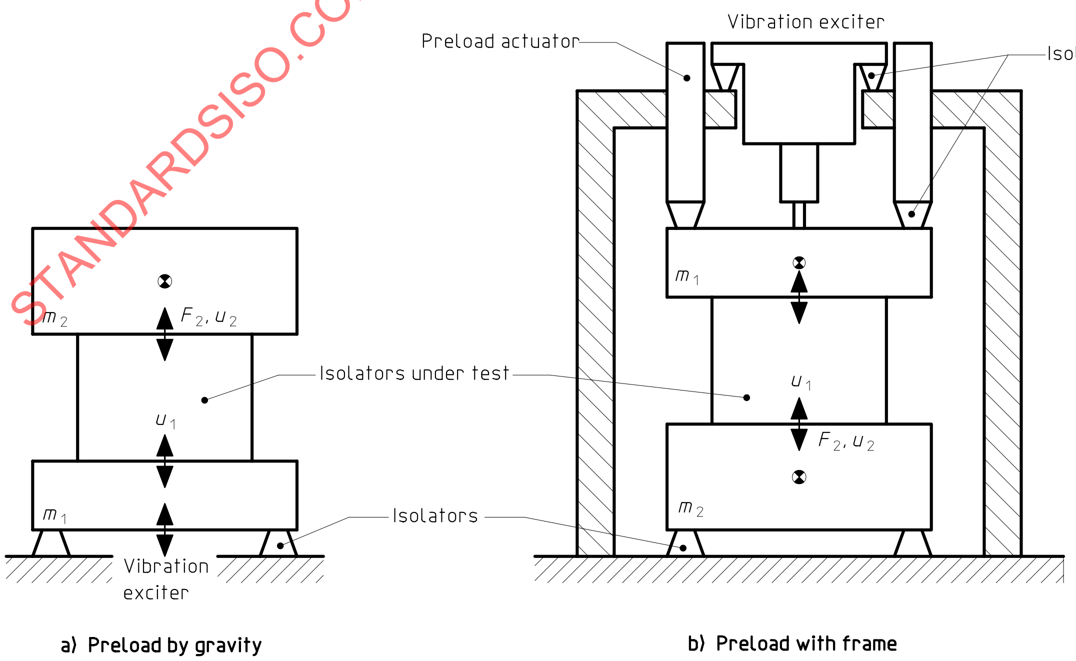


Figure 3 — Examples of typical test set-ups for the indirect method

The isolator under test is fitted between two rigid masses.

The mass on the input side of the isolator has a dual function:

— its rigidity is used to provide point contact conditions;

— it may also be used to obtain unidirectional excitation in different directions (see ISO 10846-3 and ISO 10846-4).

The mass on the output side also has a dual function:

— its rigidity is used for point contact conditions on the receiver side of the isolator;

— its mass and rotational inertias should be large enough to form a high dynamic stiffness termination for all excitation components of the isolator. Therefore, the six natural frequencies of the mass/spring system formed by combination of the test isolator and mass m_2 should be well below the frequency range of interest (see discussion below). The forces exerted by the isolator on the mass are then approximately equal to the blocking forces. These can be derived from the accelerations of the mass on the output side.

The displacements of the masses are denoted by u_1 and u_2. The ratio u_2/u_1 is usually called (displacement) transmissibility. It is equal to the corresponding velocity and acceleration ratio.

The relationship between the dynamic transfer stiffness and the displacement transmissibility is found by using Newton's law. Therefore,

$$k_{2,1} \approx \frac{F_2}{u_1} \approx -(2\pi f)^2 m_2 \frac{u_2}{u_1} \quad \text{for } f >> f_0 \tag{17}$$

where f_0 is the eigenfrequency of the mass/spring system formed by m_2 and the test isolator [and, as in figure 3 b), by the auxiliary isolators].

Equation (17) uses the assumption of equation (7), i.e. that F_2 is approximately equal to the blocking force.

6.3.2 Measurement quantities

The dynamic quantity to be measured is either the displacement, velocity or acceleration.

6.3.3 Measurement under static preload

6.3.3.1 Principle of applying preload

Figure 3 shows basic principles for test rigs in which a static preload can be applied.

In figure 3 a) the gravity force on the mass on the output side is used for static preloading. This test set-up requires either a vibration exciter which can withstand the static load or an auxiliary structure (e.g. vibration isolators) which takes the static load. This test rig principle has the danger of instability, especially for large isolators with high preloads.

In figure 3 b) a frame and an actuator (e.g. hydraulic) are used to apply the static preload. The mass m_2 on the output side of the isolator is dynamically decoupled from the frame using auxiliary isolators. Such auxiliary isolators are also used to decouple the mass on the input side from the test frame. The use of these auxiliary isolators makes the indirect method less vulnerable to flanking transmission via the test frame than the direct method. Further information can be found in ISO 10846-3.

NOTE — In practice the total stiffness of the auxiliary isolators can be of the same order of magnitude as that of the isolator under test.

6.3.3.2 Preloads in other situations

Isolators other than elastic supports need to be tested under nominal static loads as well. For instance, for a flexible shaft coupling this means that a static torque has to be applied. For a liquid-filled bellows or hose, the internal pressure has to be representative.

6.3.4 Frequency limitations of the indirect method

There are conflicting constraints with respect to the frequency range of validity.

One way to extend the range in which valid measurements according to equation (17) can be performed to low frequencies is to use a large mass m_2 to obtain a sufficiently low value for f_0. However, the larger the mass, the lower the upper frequency limit is, due to the non-rigid body behaviour of m_2.

There are many applications where the interest is in measurement data of the dynamic transfer stiffness at audio-frequencies, where it deviates from that of a massless spring. For those cases a compromise may be found along the following lines.

Estimate the frequency of the lowest internal isolator resonance (in stiffest translational direction) using the following approximation:

$$f_e \approx 0{,}5\sqrt{\frac{k_0}{m_{el}}} \qquad (18)$$

where

f_e is the estimated frequency, in hertz;

k_0 is the low frequency dynamic stiffness of the isolator;

m_{el} is the mass of the elastic part of the isolator.

At low frequencies the dynamic transfer stiffness will be nearly equal to k_0. For many isolators this "spring-like" behaviour will be valid for $f < f_e/3$. In general, choosing a mass such that $f_0 \leq 0{,}1 f_e$ gives reliable measurements of the dynamic transfer stiffness for $f \geq f_e/3$. For lower frequencies one might postulate without measurement that the dynamic stiffness is approximately equal to that at $f = f_e/3$, if the main interest for the application of the isolator is for $f \geq f_e/3$.

To obtain a wide frequency range for the measurements, it is desirable to have a low value for f_0 and to maintain rigid body behaviour of mass m_2 up to high frequencies. Such requirements are best fulfilled with steel blocks.

If the frequency range of interest is too wide for a single block, low-frequency measurements and high-frequency measurements require different block sizes.

6.3.5 Directions of vibration

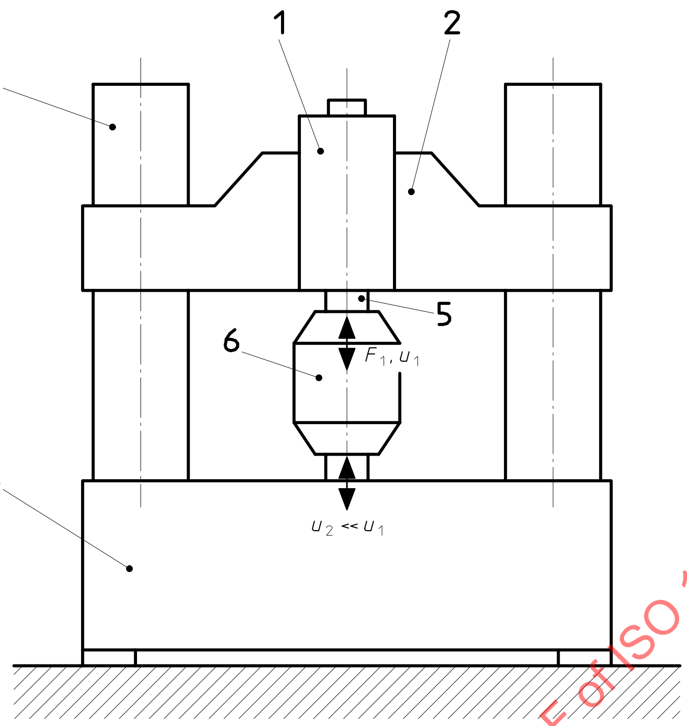
The indirect method is applicable to translational vibration both in the normal load direction and in transverse directions.

The measurement principle can be further extended to rotational excitations on the input side and/or to rotational responses of the mass on the output side [11], [13]. This is discussed in the informative annexes of ISO 10846-3 and ISO 10846-4.

6.4 Driving point method

6.4.1 Basic test arrangement

An example of the basic driving point method is given in figure 4.



Key

1 Hydraulic actuator (static preload and dynamic excitation)
2 Moveable traverse
3 Columns
4 Rigid foundation
5 Force measurement system
6 Test object

Figure 4 — Example of a typical test arrangement for the driving point method

The example given in figure 4 is very similar to that in figure 2 for the direct method. However, instead of the blocking force on the output side, the force on the input side is measured. Therefore, using the driving point method the dynamic transfer stiffness is determined by assuming that at low frequencies it is nearly equal to the driving point dynamic stiffness because the inertial forces are negligible compared to elastic forces, that is,

$$k_{2,1} \approx k_{1,1} = \left. \frac{F_1}{u_1} \right|_{u_2=0} \qquad (19)$$

The static preload is applied in the same way as in the direct method. Measurements may be performed for three orthogonal translational motions. The dynamic quantities to be measured are force and either displacement, velocity or acceleration.

6.4.2 Frequency limitation of the driving point method

The approximation given in equation (19) is only valid at low frequencies, when the inertial forces are small compared to the elastic forces. Quantitative criteria will be given in ISO 10846-5 to estimate the upper frequency limit of validity on the basis of test results. However, given the inherent and not fully known approximation by equation (19), the driving point method has to be considered to be a survey method at higher frequencies.

Annex A
(informative)

Functions related to dynamic transfer stiffness

For linear isolators the quantities that are directly related to dynamic stiffness are mechanical impedance and effective mass. Inverse quantities are compliance, mobility and accelerance.

Table A.1 gives the names and the corresponding symbols for dynamic stiffness and related quantities.

Table A.1 — Symbols for dynamic stiffness and related quantities

Symbol	Name	Inverse	Name
k	dynamic transfer stiffness	$1/k$	compliance
Z	mechanical transfer impedance	$1/Z$	mobility (admittance)
m_{eff}	effective mass	$1/m_{eff}$	accelerance

Table A.2 shows the relating factors, i.e. $k = -\omega^2 m_{eff} = j\omega Z$, etc. A multiplication by $j\omega$ means that, for example, at frequency f the amplitude is multiplied by $\omega = 2\pi f$ and that the phase angle increases by $\pi/2$ rd.

Table A.2 — Factors relating dynamic stiffness with other quantities

Name	Symbol	Definition[1)]	k	Z	m_{eff}
Dynamic transfer stiffness	k	$\boldsymbol{F}/\boldsymbol{u}$	1	$j\omega$	$-\omega^2$
Mechanical transfer impedance	Z	$\boldsymbol{F}/\boldsymbol{v}$	$1/j\omega$	1	$j\omega$
Effective mass	m_{eff}	$\boldsymbol{F}/\boldsymbol{a}$	$-1/\omega^2$	$1/j\omega$	1
1) $\boldsymbol{u}$ = displacement; $\boldsymbol{v}$ = velocity; $\boldsymbol{a}$ = acceleration					

Annex B
(informative)

Effect of symmetry on the transfer stiffness matrix

Equation (9) shows the partitioning of the 12×12 dynamic stiffness matrix for a single isolator into four 6×6 submatrices. ISO 10846 is concerned with the measurement of individual elements of the 6×6 transfer stiffness matrix $[k_{2,1}]$. These are the ratios of the blocking forces at the output side of the element and the displacements at the input.

Using the Cartesian coordinate system of figure B.1 with axes x, y and z, the vector of the six translational and rotational displacements at the input side can be written as

$$u_1 = \{u_{1x}, u_{1y}, u_{1z}, \gamma_{1x}, \gamma_{1y}, \gamma_{1z}\}$$

The vector of six blocking forces and moments at the output side can be written as

$$F_{2,\text{blocking}} = \{F_{2x}, F_{2y}, F_{2z}, M_{2x}, M_{2y}, M_{2z}\}$$

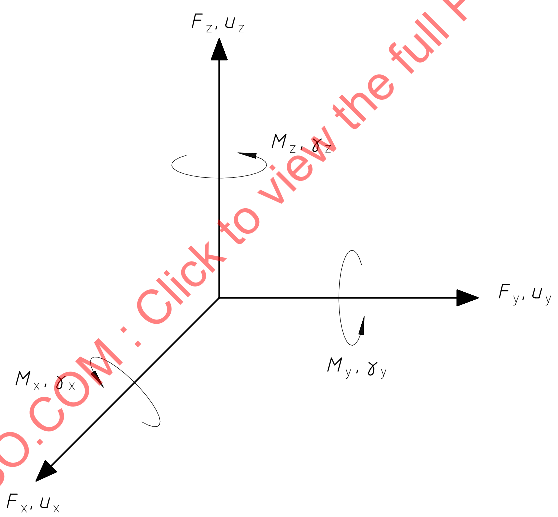


Figure B.1 — Cartesian coordinate system and notation of forces and displacements

Then equation (10) may be expanded as:

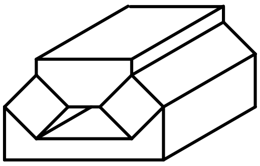
$$\begin{Bmatrix} F_{2x} \\ F_{2y} \\ F_{2z} \\ M_{2x} \\ M_{2y} \\ M_{2z} \end{Bmatrix} = \begin{bmatrix} k_{7,1} & k_{7,2} & k_{7,3} & k_{7,4} & k_{7,5} & k_{7,6} \\ k_{8,1} & k_{8,2} & k_{8,3} & k_{8,4} & k_{8,5} & k_{8,6} \\ k_{9,1} & k_{9,2} & k_{9,3} & k_{9,4} & k_{9,5} & k_{9,6} \\ k_{10,1} & k_{10,2} & k_{10,3} & k_{10,4} & k_{10,5} & k_{10,6} \\ k_{11,1} & k_{11,2} & k_{11,3} & k_{11,4} & k_{11,5} & k_{11,6} \\ k_{12,1} & k_{12,2} & k_{12,3} & k_{12,4} & k_{12,5} & k_{12,6} \end{bmatrix} \begin{Bmatrix} u_{1x} \\ u_{1y} \\ u_{1z} \\ \gamma_{1x} \\ \gamma_{1y} \\ \gamma_{1z} \end{Bmatrix}$$

The shorthand notation of the matrix elements has the following meaning:

$$k_{7,1} = \frac{F_{2x,\,\text{blocking}}}{u_{1x}} \quad \text{and} \quad k_{10,4} = \frac{M_{2x,\,\text{blocking}}}{\gamma_{1x}}, \text{ etc.}$$

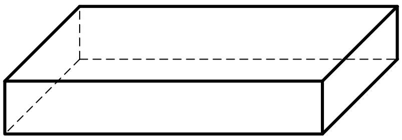
Due to symmetry a large number of elements will be equal to zero and some non-zero elements may be equal in magnitude. The following four examples illustrate this for typical examples of vibration isolator symmetries (see figure B.1 for the orientation of the axes).

EXAMPLE 1 Two orthogonal planes of symmetry (10 different non-zero elements)



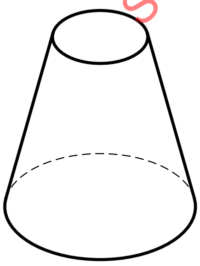
	u_{1x}	u_{1y}	u_{1z}	γ_{1x}	γ_{1y}	γ_{1z}
F_{2x}	$k_{7,1}$	0	0	0	$k_{7,5}$	0
F_{2y}	0	$k_{8,2}$	0	$k_{8,4}$	0	0
F_{2z}	0	0	$k_{9,3}$	0	0	0
M_{2x}	0	$k_{10,2}$	0	$k_{10,4}$	0	0
M_{2y}	$k_{11,1}$	0	0	0	$k_{11,5}$	0
M_{2z}	0	0	0	0	0	$k_{12,6}$

EXAMPLE 2 Three orthogonal planes of symmetry (10 non-zero elements; 8 different elements)



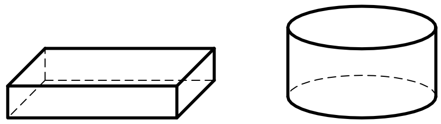
	u_{1x}	u_{1y}	u_{1z}	γ_{1x}	γ_{1y}	γ_{1z}
F_{2x}	$k_{7,1}$	0	0	0	$k_{7,5}$	0
F_{2y}	0	$k_{8,2}$	0	$k_{8,4}$	0	0
F_{2z}	0	0	$k_{9,2}$	0	0	0
M_{2x}	0	$= -k_{8,4}$	0	$k_{10,4}$	0	0
M_{2y}	$= -k_{7,5}$	0	0	0	$k_{11,5}$	0
M_{2z}	0	0	0	0	0	$k_{12,6}$

EXAMPLE 3 Axial symmetry with respect to symmetry planes of example 1 (10 non-zero elements; 6 different elements)



	u_{1x}	u_{1y}	u_{1z}	γ_{1x}	γ_{1y}	γ_{1z}
F_{2x}	$k_{7,1}$	0	0	0	$k_{7,5}$	0
F_{2y}	0	$= k_{7,1}$	0	$= k_{7,5}$	0	0
F_{2z}	0	0	$k_{9,3}$	0	0	0
M_{2x}	0	$k_{10,2}$	0	$k_{10,4}$	0	0
M_{2y}	$= k_{10,2}$	0	0	0	$= k_{10,4}$	0
M_{2z}	0	0	0	0	0	$k_{12,6}$

EXAMPLE 4 Square block or circular cylinder (10 non-zero elements; 5 different elements)



	u_{1x}	u_{1y}	u_{1z}	γ_{1x}	γ_{1y}	γ_{1z}
F_{2x}	$k_{7,1}$	0	0	0	$k_{7,5}$	0
F_{2y}	0	$k_{7,1}$	0	$= k_{7,5}$	0	0
F_{2z}	0	0	$k_{9,3}$	0	0	0
M_{2x}	0	$=-k_{7,5}$	0	$k_{10,4}$	0	0
M_{2y}	$=-k_{7,5}$	0	0	0	$=k_{10,4}$	0
M_{2z}	0	0	0	0	0	$k_{12,6}$