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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
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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.

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The main task of technical committees is to prepare International Standards. 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.

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

This second edition cancels and replaces the first edition (ISO 10846-1:1997), which has been technically revised.

ISO 10846 consists of the following parts, under the general title *Acoustics and vibration — Laboratory measurement of vibro-acoustic transfer properties of resilient elements*:

— *Part 1: Principles and guidelines*

— *Part 2: Direct method for determination of the dynamic stiffness of resilient supports for translatory motion*

— *Part 3: Indirect method for determination of the dynamic stiffness of resilient supports for translatory motion*

— *Part 4: Dynamic stiffness of elements other than resilient supports for translatory motion*

— *Part 5: Driving point method for determination of the low-frequency transfer stiffness of resilient supports for translatory motion*

Introduction

Passive vibration isolators of various kinds are used to reduce the transmission of vibrations. Examples include automobile engine mounts, resilient supports for buildings, resilient 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 ISO 10846-2, ISO 10846-3, ISO 10846-4 and ISO 10846-5, which describe laboratory measurement methods for the determination of the most important quantities which govern the transmission of vibrations through linear resilient elements, i.e. frequency-dependent dynamic transfer 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, where appropriate.

The results of the methods are useful for resilient elements, which are used to prevent low-frequency vibration problems and to attenuate structure-borne sound. However, for complete characterization of resilient elements that are used to attenuate low-frequency vibration or shock excursions, additional information is needed, which is not provided by these methods.

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 ISO 10846-2, ISO 10846-3, ISO 10846-4 and ISO 10846-5 for determining the transfer properties of resilient elements from laboratory measurements, and provides assistance in the selection of the appropriate part of this series. It is applicable to resilient elements that are used to reduce

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

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

The data obtained with the measurement methods, which are outlined in this part of ISO 10846 and further detailed in ISO 10846-2, ISO 10846-3, ISO 10846-4 and ISO 10846-5, can be used for

— product information provided by manufacturers and suppliers,

— information during product development,

— quality control, and

— calculation of the transfer of vibrations through resilient elements.

The conditions for the validity of the measurement methods are

a) linearity of the vibrational behaviour of the resilient elements (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), and

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

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 2041:—[1], *Mechanical vibration, shock and condition monitoring — Vocabulary*

1) To be published. (Revision of ISO 2041:1990)

ISO/IEC Guide 98-3 [2], *Uncertainty of measurement — Part 3: Guide to the expression of uncertainty in measurement (GUM 1995)*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 2041 and the following apply.

3.1
vibration isolator
resilient element
isolator designed to attenuate the transmission of the vibration in a certain frequency range

NOTE Adapted from ISO 2041:—[1], definition 2.120.

3.2
resilient support
vibration isolator(s) suitable for supporting a machine, a building or another type of structure

3.3
test element
resilient element undergoing testing, including flanges and auxiliary fixtures, if any

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

3.5
dynamic driving point stiffness
$k_{1,1}$
frequency-dependent ratio of the force phasor $\underline{F}_1$ on the input side of a vibration isolator with the output side blocked to the displacement phasor $\underline{u}_1$ on the input side

$$k_{1,1} = \underline{F}_1 / \underline{u}_1$$

NOTE 1 The subscripts “1” denote that the force and displacement are measured on the input side.

NOTE 2 The value of $k_{1,1}$ can be dependent on the static preload, temperature, relative humidity and other conditions.

NOTE 3 At low frequencies, elastic and dissipative forces solely determine $k_{1,1}$. At higher frequencies, inertial forces play a role as well.

3.6
dynamic driving point stiffness of inverted vibration isolator
$k_{2,2}$
dynamic driving point stiffness, with the physical input and output sides of the vibration isolator interchanged

NOTE At low frequencies, where elastic and dissipative forces solely determine the driving point stiffness, $k_{1,1} = k_{2,2}$. At higher frequencies inertial forces play a role as well and $k_{1,1}$ and $k_{2,2}$ will be different in case of asymmetry.

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2) ISO/IEC Guide 98-3 will be published as a re-issue of the *Guide to the expression of uncertainty in measurement* (GUM), 1995.

3.7
dynamic transfer stiffness
$k_{2,1}$
frequency-dependent ratio of the blocking force phasor $\underline{F}_{2,\mathrm{b}}$ on the output side of a resilient element to the displacement phasor $\underline{u}_1$ on the input side

$$k_{2,1} = \underline{F}_{2,\mathrm{b}} / \underline{u}_1$$

NOTE 1 The subscripts “1”and “2” denote the input and output sides, respectively.

NOTE 2 The value of $k_{2,1}$ can be dependent on the static preload, temperature and other conditions.

NOTE 3 At low frequencies, $k_{2,1}$ is mainly determined by elastic and dissipative forces and $k_{1,1} \approx k_{2,1}$. At higher frequencies, inertial forces in the resilient element play a role as well and $k_{1,1} \neq k_{2,1}$.

3.8
loss factor of resilient element
η
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

3.9
point contact
contact area which vibrates as the surface of a rigid body

3.10
linearity
property of the dynamic behaviour of a resilient element, 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 $a\,x_1(t) + b\,x_2(t)$ produces the output $ay_1(t) + b\,y_2(t)$. This must hold for all values of a, b and $x_1(t)$, $x_2(t)$; a and b are arbitrary constants.

NOTE 2 In practice, the above test for linearity is impractical and a limited check of linearity is performed 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.

3.11
direct method
method in which either the input displacement, velocity or acceleration and the blocking output force are measured

3.12
indirect method
method in which the vibration transmissibility (for displacement, velocity or acceleration) of a resilient element is measured, with the output loaded by a compact body of known mass

NOTE The term “indirect method” can be permitted to include loads of any known impedance other than a mass-like impedance. However, the ISO 10846 series does not cover such methods.

3.13
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 resilient element blocked

3.14
flanking transmission
forces and accelerations at the output side caused by the vibration exciter on the input side but via transmission paths other than through the resilient element under test

3.15
upper limiting frequency
f_{UL}
frequency up to which results for $k_{1,2}$ are valid, according to the criteria given in various parts of ISO 10846

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 Direct or indirect method**	**ISO 10846-5 Driving point method**
Type of resilient element	support	support	other than support	support
Examples	resilient mountings for instruments, equipment, machinery and buildings		bellows, hoses, resilient shaft couplings, power supply cables	see under ISO 10846-2 and ISO 10846-3
Frequency range of validity	1 Hz to f_{UL} f_{UL} dependent on test rig; typically (but not limited to) 300 Hz $< f_{UL} <$ 500 Hz	f_2 to f_3 f_2 typically (but not limited to) between 20 Hz and 50 Hz. For very stiff mountings $f_2 >$ 100 Hz. f_3 typically 2 kHz to 5 kHz, but dependent on the test rig	Direct method: see under ISO 10846-2; Indirect method: see under ISO 10846-3	1 Hz to f_{UL} f_{UL} typically (but not limited to) $<$ 200 Hz f_{UL} is dependent both on test rig and on test element properties;
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
Expanded measurement uncertainty for 95 % coverage probability	To be estimated according to ISO/IEC Guide 98-3	4 dB (considered as the upper limit)	4 dB (considered as the upper limit)	To be estimated according to ISO/IEC Guide 98-3
NOTE Within coinciding frequency ranges of validity, and within the uncertainty ranges of the methods, the direct method, the indirect 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 resilient elements for many practical applications. It also describes special situations where other vibro-acoustic properties, not covered in ISO 10846, would also be necessary.

The dynamic transfer stiffness, as defined in 3.7, is determined by the elastic, inertia and damping properties of the resilient element. Describing the test results in terms of stiffness properties allows for compliance 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. At low frequencies, only elastic and damping forces are important. Because in general the modulus of elasticity and the damping properties are only weakly dependent on frequency in this range, this holds also for the low-frequency dynamic stiffness.

NOTE For many resilient elements, static stiffness and low-frequency dynamic transfer stiffness are different.

In principle, the dynamic transfer stiffness of vibro-acoustic resilient elements is dependent on static preload, temperature and relative humidity. In the following theory, linearity, as defined in 3.10, 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 resilient elements

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 resilient elements under static preload.

The general conceptual framework for the specified characterization of resilient elements is shown in Figure 1.

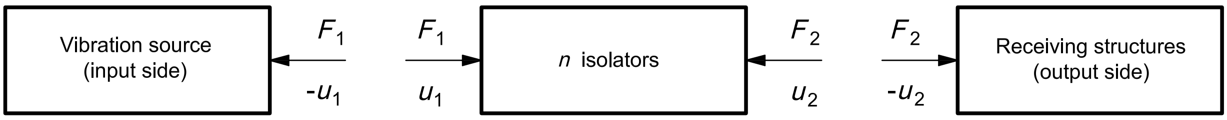


Figure 1 — Block diagram of source/isolators/receiver system

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[3)] $\boldsymbol{u}$ containing three orthogonal translational components and three orthogonal rotational components are assigned. In Figure 1, just one component of each of the vectors $\boldsymbol{F}_1$, $\boldsymbol{u}_1$, $\boldsymbol{F}_2$ and $\boldsymbol{u}_2$ is shown. These vectors contain $6n$ elements, where n denotes the number of isolators.

To show that the blocked transfer stiffness, defined in 3.7 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.

3) Linear algebra: a vector is a linear array of elements.

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:

$$\underline{F}_1 = k_{1,1}\,\underline{u}_1 + k_{1,2}\,\underline{u}_2 \tag{1}$$

$$\underline{F}_2 = k_{2,1}\,\underline{u}_1 + k_{2,2}\,\underline{u}_2 \tag{2}$$

where

$k_{1,1}$ and $k_{2,2}$ are driving point stiffnesses when the isolator is blocked at the opposite side (i.e. $\underline{u}_2 = 0$, $\underline{u}_1 = 0$, 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 increasing 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 1 These equations are for single frequencies. $\underline{F}_i$ and $\underline{u}_i$ are phasors and $k_{i,j}$ are complex quantities.

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

$$\boldsymbol{F} = \boldsymbol{K}\boldsymbol{u} \tag{3}$$

with the dynamic stiffness matrix

$$\boldsymbol{K} = \begin{bmatrix} k_{1,1} & k_{1,2} \\ k_{2,1} & k_{2,2} \end{bmatrix} \tag{4}$$

For excitation of the receiving structure via the isolator

$$k_t = -\frac{\underline{F}_2}{\underline{u}_2} \tag{5}$$

where k_t denotes the dynamic driving point stiffness of the termination. The minus sign is a consequence of the convention adopted in Figure 1.

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

$$\underline{F}_2 = \frac{k_{2,1}}{1 + \dfrac{k_{2,2}}{k_t}}\,\underline{u}_1 \tag{6}$$

Therefore, for a given source displacement $\underline{u}_1$, the force $\underline{F}_2$ depends both on the isolator driving point dynamic stiffness and on the receiver driving point dynamic stiffness. However, if $|k_{2,2}| < 0,1|k_t|$, then $\underline{F}_2$ approximates the so-called blocking force to within 10 %, i.e.

$$\underline{F}_2 \approx \underline{F}_{2,b} = k_{2,1}\underline{u}_1 \tag{7}$$

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 measurement 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 2 In cases where the condition $|k_{2,2}| << |k_t|$ is not fulfilled, Equation (6) also shows that $k_{2,2}$ and k_t need to be known to predict $\underline{F}_2$ for a given source displacement $\underline{u}_1$.

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, Reference [11]. The matrix form of the 12 dynamic force equations is equal to Equation (3), where now

$$\boldsymbol{u} = \begin{Bmatrix} \boldsymbol{u}_1 \\ \boldsymbol{u}_2 \end{Bmatrix}, \boldsymbol{F} = \begin{Bmatrix} \boldsymbol{F}_1 \\ \boldsymbol{F}_2 \end{Bmatrix} \tag{8}$$

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

$$\boldsymbol{K} = \begin{bmatrix} \boldsymbol{K}_{1,1} & \boldsymbol{K}_{1,2} \\ \boldsymbol{K}_{2,1} & \boldsymbol{K}_{2,2} \end{bmatrix} \tag{9}$$

where

$\boldsymbol{K}_{1,1}$ and $\boldsymbol{K}_{2,2}$ are (symmetric) matrices of the driving point stiffnesses;

$\boldsymbol{K}_{1,2}$ and $\boldsymbol{K}_{2,1}$ are the blocked transfer stiffness matrices.

The symmetry of the dynamic stiffness matrix in Equation (3) implies that these transfer matrices equal the transpose of each other.

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:

$$\boldsymbol{F}_2 \approx \boldsymbol{F}_{2,\mathrm{b}} = \boldsymbol{K}_{2,1}\,\boldsymbol{u}_1 \tag{10}$$

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

In general, the blocked transfer stiffness matrix $\boldsymbol{K}_{2,1}$ 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 ISO 10846-2, ISO 10846-3, ISO 10846-4 and ISO 10846-5.

For some special 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 has an informative annex 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 resilient elements 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 ISO 10846-2, ISO 10846-3, ISO 10846-4 and ISO 10846-5, 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

$$\underline{F}_1 = k_{1,1}\underline{u}_1 \tag{11}$$

$$\underline{F}_2 = k_{2,1}\underline{u}_1 \tag{12}$$

At low frequencies, where inertial forces 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 3.8).

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

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

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

$$\eta = \tan \varphi_{2,1} \tag{16}$$

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

The following points should be kept in mind.

a) The measurement of loss factors, using Equation (16), is sensitive to imperfections in the blocked output condition $u_2 = 0$. In Reference [18], correction procedures are described. The measurement of small loss factors is also sensitive to phase measurement uncertainties (Reference [12], pp. 216-218).

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. It may also be dependent on relative humidity. 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 covered in ISO 10846.

The direct method requires the measurement of input displacement (velocity, acceleration) and blocking output force.

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 driving point method for determination of the transfer stiffness is used in test rigs in which the dynamic force on a resilient element with a blocked output side can only be measured on the input side. The stiffness resulting from measuring input displacement (velocity, acceleration) and input force, is the dynamic driving point stiffness. Only at low frequencies, where the driving point stiffness and the transfer stiffness are equal, this method can be used for determination of the dynamic transfer stiffness. Although the direct method for measuring the dynamic transfer stiffness has a wider frequency for valid measurements than the driving point method, including those at low frequencies, the latter is covered by ISO 10846 as well. In this way, owners of (often expensive) test rigs for driving point stiffness measurements, are enabled to use their facilities for determination of the low-frequency dynamic transfer stiffness as well.

The basic features of these three methods and the general requirements for their proper use are described in this part of ISO 10846. Detailed requirements are specified in ISO 10846-2, ISO 10846-3, ISO 10846-4 and ISO 10846-5.

6.2 Direct method

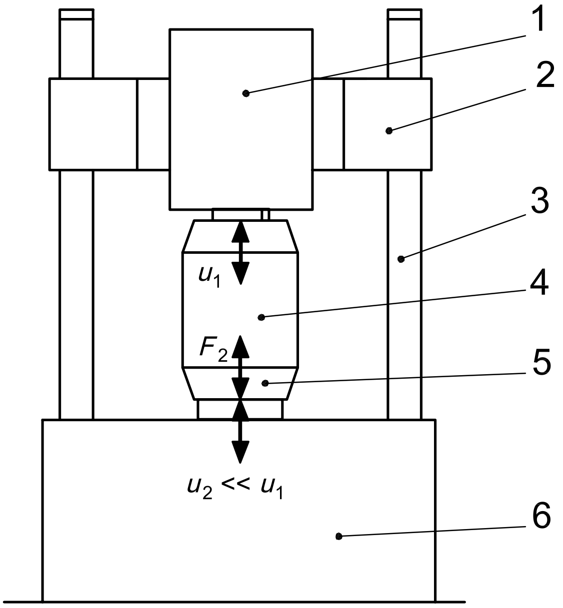
6.2.1 Basic test set-up

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.

The dynamic transfer stiffness is determined as

$$k_{2,1} = \frac{\underline{F}_2}{\underline{u}_1} \text{ for } \underline{u}_2 << \underline{u}_1 \tag{17}$$



Key

1 hydraulic actuator (static preload and dynamic excitation)
2 moveable traverse
3 columns
4 test element
5 force measurement system
6 rigid foundation

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

6.2.2 Measurement quantities

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

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 preloading and vibration is typically applied using a hydraulic actuator on top. However, test rigs with separated components for preloading 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 determined 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 columns. A typical upper frequency of 300 Hz $< f_{UL} <$ 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 resilient elements with small preloads, valid measurements have been made up to several kilohertz, using very simple test set-ups.

However, generally speaking, the indirect method (see 6.3) 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

Although Figure 2 presents an example for stiffness measurement in the normal load direction, the direct method can be applied for translational and rotational vibration in all directions. In ISO 10846-2 stiffness measurements for three perpendicular translations are standardized. Use of the direct method for rotational vibration is not considered in 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.

The resilient element under test is fitted between two rigid masses.

The mass on the input side of the element 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, ISO 10846-4 and ISO 10846-5).

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 element 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,

$$k_{2,1} \approx \frac{F_2}{\underline{u}_1} \approx -(2\pi f)^2 m_2 \frac{u_2}{\underline{u}_1} \text{ for } f >> f_0 \tag{18}$$

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

Equation (18) 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 bears the static load. This test rig principle carries 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 resilient 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

Conflicting constraints exist with respect to the frequency range of validity.

One way to extend the range in which valid measurements according to Equation (18) 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 the 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, f_e (in the stiffest translational direction), using the following approximation:

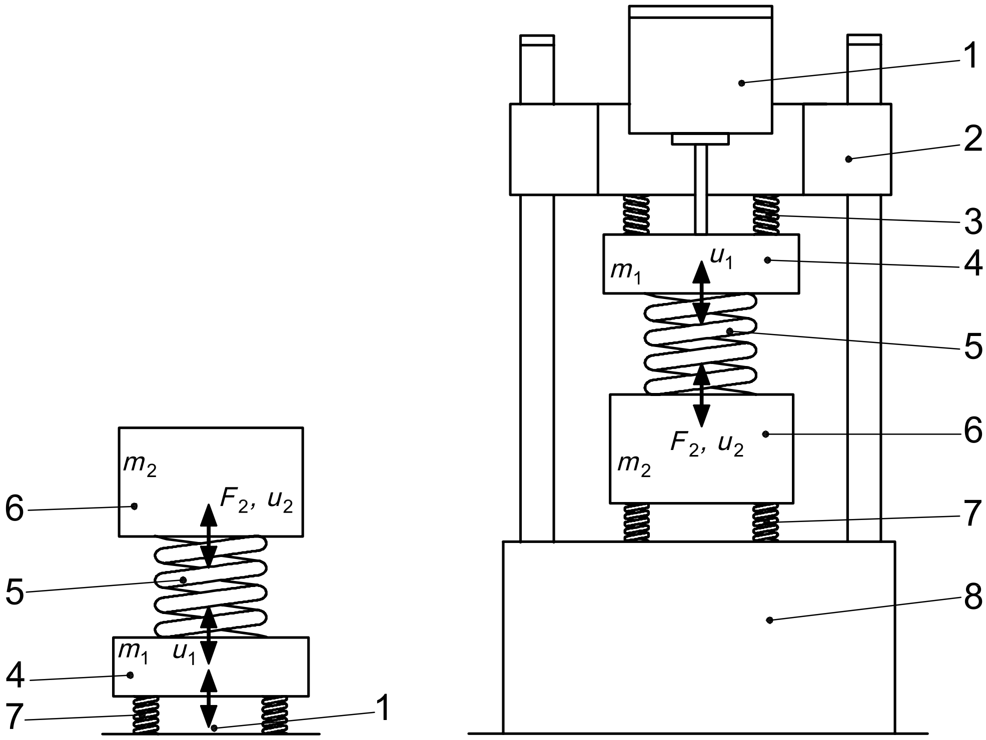
$$f_e \approx 0{,}5 \sqrt{\frac{k_0}{m_{el}}} \tag{19}$$

where

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 $f_0 \leqslant 0{,}1 f_e$ gives reliable measurements of the dynamic transfer stiffness for $f \geqslant f_e/3$. 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 in the application of the isolator is for $f \geqslant f_e/3$.



a) Preload by gravity **b) Preload with frame**

Key

1 vibration exciter
2 moveable transverse
3 dynamic decoupling springs, static preload
4 force distribution mass
5 test element
6 blocking mass
7 vibration isolators
8 rigid foundation

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

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

Although Figure 3 presents an example of stiffness measurement in the normal load direction, the indirect method can be applied for translational and rotational vibration in all directions. In ISO 10846-3, the stiffness measurements for three perpendicular translations are standardized.

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, References [11], [13], but this is not standardized. Instead, this is discussed in informative annexes of ISO 10846-3 and ISO 10846-4.

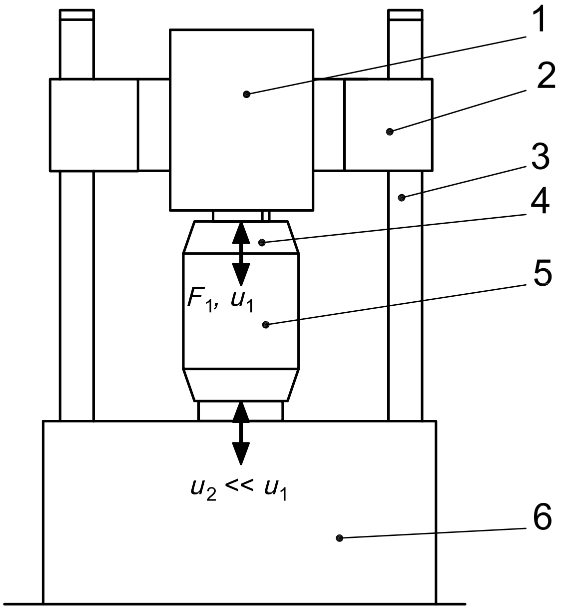
6.4 Driving point method

6.4.1 Basic test arrangement

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

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,

$$k_{2,1} \approx k_{1,1} = \left. \frac{\underline{F}_1}{\underline{u}_1} \right|_{u_2 = 0} \tag{20}$$



Key

1 hydraulic actuator (static preload and dynamic excitation)
2 moveable traverse
3 columns
4 force measurement system
5 test element
6 rigid foundation

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

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 (20) is only valid at low frequencies (typically $f < 200$ Hz), when the inertial forces are small compared to the elastic and damping forces. Quantitative criteria will be given in ISO 10846-5 to estimate the upper frequency limit of validity on the basis of test results.

6.4.3 Directions of vibration

Although Figure 4 presents an example for stiffness measurement in the normal load direction, the driving point method can be applied for translational and rotational vibration in all directions. In ISO 10846-5, stiffness measurements for three perpendicular translations are standardized. Use of the driving point method for rotational vibration is not considered in ISO 10846.

Annex A
(informative)

Functions related to dynamic 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 stiffness	$1/k$	compliance
Z	mechanical impedance	$1/Z$	mobility (admittance)
m_{eff}	effective mass	$1/m_{\mathrm{eff}}$	accelerance

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

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

Name	Symbol	Definition[a]	k	Z	m_{eff}
Dynamic stiffness	k	$\underline{F}/\underline{u}$	1	$\mathrm{j}\omega$	$-\omega^2$
Mechanical impedance	Z	$\underline{F}/\underline{v}$	$1/\mathrm{j}\omega$	1	$\mathrm{j}\omega$
Effective mass	m_{eff}	$\underline{F}/\underline{a}$	$-1/\omega^2$	$1/\mathrm{j}\omega$	1
[a] Phasors: $\underline{F}$ = force; $\underline{u}$ = displacement; $\underline{v}$ = velocity; $\underline{a}$ = 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 $\boldsymbol{K}_{2,1}$. These are the ratios of the blocking forces at the output side of the resilient 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 phasors at the input side can be written as

$$\boldsymbol{u}_1 = \{\underline{u}_{1x}, \underline{u}_{1y}, \underline{u}_{1z}, \underline{\gamma}_{1x}, \underline{\gamma}_{1y}, \underline{\gamma}_{1z}\}$$

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

$$\boldsymbol{F}_{2,\mathrm{b}} = \{\underline{F}_{2x}, \underline{F}_{2y}, \underline{F}_{2z}, \underline{M}_{2x}, \underline{M}_{2y}, \underline{M}_{2z}\}$$

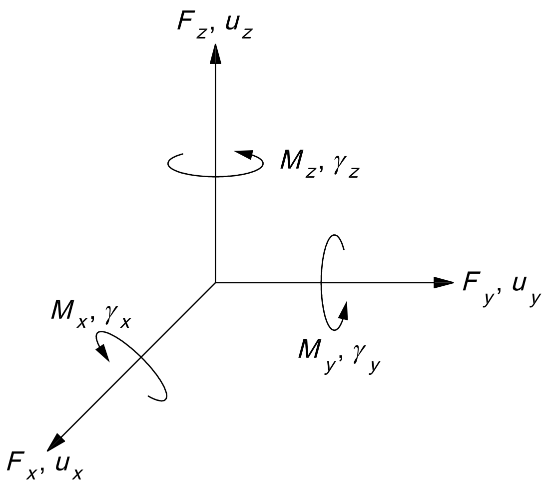


Figure B.1 — Cartesian coordinate system and notation of forces and displacements

Then Equation (10) may be expanded as:

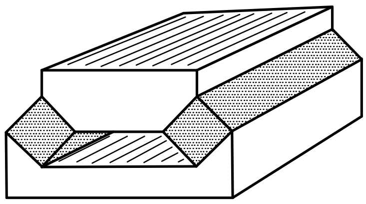
$$\begin{Bmatrix} \underline{F}_{2x} \\ \underline{F}_{2y} \\ \underline{F}_{2z} \\ \underline{M}_{2x} \\ \underline{M}_{2y} \\ \underline{M}_{2z} \end{Bmatrix} = \begin{bmatrix} k_{7,1} & k_{7,2} & k_{7,3} & k_{7,4} & k_{7,5} & k_{7,6} \\ k_{8,1} & k_{8,2} & k_{8,3} & k_{8,4} & k_{8,5} & k_{8,6} \\ k_{9,1} & k_{9,2} & k_{9,3} & k_{9,4} & k_{9,5} & k_{9,6} \\ k_{10,1} & k_{10,2} & k_{10,3} & k_{10,4} & k_{10,5} & k_{10,6} \\ k_{11,1} & k_{11,2} & k_{11,3} & k_{11,4} & k_{11,5} & k_{11,6} \\ k_{12,1} & k_{12,2} & k_{12,3} & k_{12,4} & k_{12,5} & k_{12,6} \end{bmatrix} \begin{Bmatrix} \underline{u}_{1x} \\ \underline{u}_{1y} \\ \underline{u}_{1z} \\ \underline{\gamma}_{1x} \\ \underline{\gamma}_{1y} \\ \underline{\gamma}_{1z} \end{Bmatrix}$$

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

$$k_{7,1} = \frac{\underline{F}_{2x,\,\mathrm{b}}}{\underline{u}_{1x}} \text{ and } k_{10,4} = \frac{\underline{M}_{2x,\,\mathrm{b}}}{\underline{\gamma}_{1x}}, \text{ etc.}$$

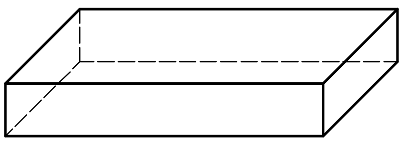
Due to symmetry, a large number of elements will be equal to zero and some non-zero elements may be equal in magnitude. The following four examples illustrate this for typical examples of vibration isolator symmetries (see Figure B.1 for the orientation of the axes).

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



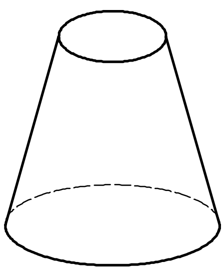
	$\underline{u}_{1x}$	$\underline{u}_{1y}$	$\underline{u}_{1z}$	$\underline{\gamma}_{1x}$	$\underline{\gamma}_{1y}$	$\underline{\gamma}_{1z}$
$\underline{F}_{2x}$	$k_{7,1}$	0	0	0	$k_{7,5}$	0
$\underline{F}_{2y}$	0	$k_{8,2}$	0	$k_{8,4}$	0	0
$\underline{F}_{2z}$	0	0	$k_{9,3}$	0	0	0
$\underline{M}_{2x}$	0	$k_{10,2}$	0	$k_{10,4}$	0	0
$\underline{M}_{2y}$	$k_{11,1}$	0	0	0	$k_{11,5}$	0
$\underline{M}_{2z}$	0	0	0	0	0	$k_{12,6}$

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



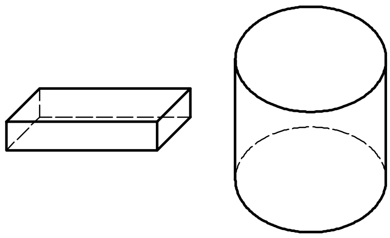
	$\underline{u}_{1x}$	$\underline{u}_{1y}$	$\underline{u}_{1z}$	$\underline{\gamma}_{1x}$	$\underline{\gamma}_{1y}$	$\underline{\gamma}_{1z}$
$\underline{F}_{2x}$	$k_{7,1}$	0	0	0	$k_{7,5}$	0
$\underline{F}_{2y}$	0	$k_{8,2}$	0	$k_{8,4}$	0	0
$\underline{F}_{2z}$	0	0	$k_{9,3}$	0	0	0
$\underline{M}_{2x}$	0	$-k_{8,4}$	0	$k_{10,4}$	0	0
$\underline{M}_{2y}$	$-k_{7,5}$	0	0	0	$k_{11,5}$	0
$\underline{M}_{2z}$	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).



	$\underline{u}_{1x}$	$\underline{u}_{1y}$	$\underline{u}_{1z}$	$\underline{\gamma}_{1x}$	$\underline{\gamma}_{1y}$	$\underline{\gamma}_{1z}$
$\underline{F}_{2x}$	$k_{7,1}$	0	0	0	$k_{7,5}$	0
$\underline{F}_{2y}$	0	$k_{7,1}$	0	$k_{7,5}$	0	0
$\underline{F}_{2z}$	0	0	$k_{9,3}$	0	0	0
$\underline{M}_{2x}$	0	$k_{10,2}$	0	$k_{10,4}$	0	0
$\underline{M}_{2y}$	$k_{10,2}$	0	0	0	$k_{10,4}$	0
$\underline{M}_{2z}$	0	0	0	0	0	$k_{12,6}$

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



	$\underline{u}_{1x}$	$\underline{u}_{1y}$	$\underline{u}_{1z}$	$\underline{\gamma}_{1x}$	$\underline{\gamma}_{1y}$	$\underline{\gamma}_{1z}$
$\underline{F}_{2x}$	$k_{7,1}$	0	0	0	$k_{7,5}$	0
$\underline{F}_{2y}$	0	$k_{7,1}$	0	$k_{7,5}$	0	0
$\underline{F}_{2z}$	0	0	$k_{9,3}$	0	0	0
$\underline{M}_{2x}$	0	$-k_{7,5}$	0	$k_{10,4}$	0	0
$\underline{M}_{2y}$	$-k_{7,5}$	0	0	0	$k_{10,4}$	0
$\underline{M}_{2z}$	0	0	0	0	0	$k_{12,6}$

Annex C
(informative)

Simplified transfer stiffness matrices

C.1 General

In 5.3 it is stated that, for many practical cases, it is sufficient to determine only one, two or three diagonal elements for translations to characterize the transfer stiffness matrix. A few examples are discussed in this Annex to elucidate this statement.

C.2 Translations

As the block diagram in Figure 1 of the source/isolator/receiver system shows, the vibro-acoustic transmission through isolators depends on the source vibration, the isolator transfer stiffnesses and the receiver driving point stiffnesses. Usually, vibrational sources do not vibrate unidirectionally. Therefore, the measurement of isolator stiffnesses for orthogonal directions is of practical interest.

Let a representative case be considered where the translation amplitudes of the source in three orthogonal directions are of the same order of magnitude. Then, *a priori*, at least three transfer stiffnesses, i.e. the diagonal elements for translation of the transfer stiffness matrix, are of interest for characterizing the isolator. Of course, depending on symmetry, this may be reduced to two transfer stiffnesses, i.e. one vertical and one transverse.

The question of whether it is allowed to neglect transverse vibration now depends on two factors:

— the ratio of the transverse stiffness to the vertical stiffness of the isolators;

— the ratio of the transverse stiffness to the vertical stiffness of the receiver.

For example, equipment is often installed on a thick concrete floor on isolators with the transverse stiffness equal to or smaller than the vertical stiffness, while the vertical point stiffness of the floor is smaller than that for the transverse direction. In this case, it is fully justified to analyse the vibro-acoustic transmission for vertical vibration only. Therefore, only the vertical transfer stiffness of the isolator is needed.

In another example, the vibration isolators form part of a structure that allows for thermal expansion (e.g. an exhaust system). Cone-type mountings are used, which are perhaps ten times stiffer for the transverse direction than for the direction of the main load. Moreover, the receiver structure is quite flexible in all directions. For such a situation, it becomes clear that an analysis for transverse vibration is at least as important as that for the direction of the main static load.

In conclusion, a good judgement of the type of vibration isolators and their typical installation environment is needed to decide on the number of transfer stiffnesses that have to be determined for an isolator.

C.3 Rotations

Rotational stiffnesses and cross-stiffnesses that connect a translational component on one side with a rotational component on the opposite side of an isolator are not covered in ISO 10846. An important reason for this is the complexity of the measurements involved, especially where standardized transducers are not available.

The question is in which type of situations might this lead to inaccurate characterization of vibro-acoustic transmission via isolators.

With respect to the source and receiver structures, in general terms, the bending wavelengths determine the ratio between the translational and rotational responses. For shorter wavelengths, the rotational vibration or the sensitivity for rotational excitation increases comparatively. This implies that thin plate-beam structures and relatively high excitation frequencies display stronger rotations. On the other hand, in the above-mentioned example of a machine on a heavy concrete floor, the influence of rotations is usually negligible.

An example of a relative increase of the importance of moment excitation is for isolators on top of T- or I-type beam structures. The driving point dynamic stiffness for point force F may be quite large, whereas for excitation by moment M this is not the case (see Figure C.1).

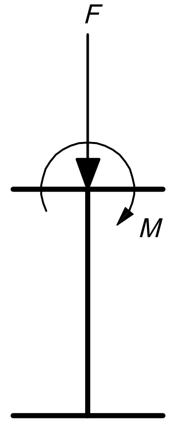


Figure C.1 — Excitation on top of an I-type beam for point force F and moment M

With respect to the isolator properties, geometrical shapes which have large transverse stiffnesses may also display large rotational stiffnesses. When isolators of such shapes are applied in thin-walled structures and for rather high frequency isolation, transfer stiffnesses which include rotational components may become quite important. In such cases, the above-mentioned simplification, which takes only one, two or three translational stiffnesses into account, could be inaccurate for the purpose of analysing the vibro-acoustic transmission.

Annex D
(informative)

Linearity of resilient elements

In principle, the dynamic properties of a vibro-acoustic isolator are dependent on static preload, vibration amplitude, frequency and temperature.

The assumption of linearity implies that the principle of superposition holds and that the dynamic stiffness at a given frequency is independent of vibration amplitude. For many isolators, this assumption is approximately satisfied when, under the appropriate static preload, the dynamic deformation amplitudes are small compared with the static deformation. However, it should be noted that this depends on the materials of which the isolators are composed, and a simple check should be carried out by comparing the dynamic stiffness characteristics for a range of input levels. If these are nominally invariant then linearity may be assumed to hold.

For butyl rubber (IIR), Reference [14] presents data for the in-phase component and the phase angle of the dynamic shear modulus, as a function of shear strain amplitude and of the percentage of carbon black. For shear strain amplitudes smaller than about 10^{-3}, the in-phase component and phase angle are hardly dependent on the vibration amplitude. However, a significant decrease of dynamic stiffness is seen when shear strain amplitudes exceed about 2×10^{-3}, especially for rubber with a high percentage of carbon black.

Therefore, it is important to consider strain amplitudes which occur in practice and to check whether the test conditions are appropriate for the testing of rubber isolators. For strain amplitudes smaller than about 10^{-3}, the assumption of linearity (implying, for example, amplitude-independent stiffness and also reciprocity) seems justified.

Hydraulic mounts are increasingly used, especially for automotive applications. This type of isolator may also show a very non-linear behaviour, i.e. stiffness strongly dependent on the vibration amplitude. Because of their two-fold purpose, i.e. damping of low-frequency engine vibration caused by road excitation and isolation of engine generated structure-borne sound at higher frequencies, appropriate test amplitudes have to be applied for the whole frequency range of interest [15], [16].

It is sometimes known *a priori* that linearity does not hold. In such cases, it may still be advantageous to apply many of the procedures described in ISO 10846. Often this will imply that, in addition, special test requirements will be formulated with respect to preloads, signal amplitudes and measured quantities.

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