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

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

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

Partie 3: Méthode indirecte pour la détermination de la raideur dynamique en translation des supports élastiques

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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

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.

Attention is drawn to the possibility that some of the elements of this part of ISO 10846 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO 10846-3 was prepared jointly by Technical Committee ISO/TC 43, *Acoustics*, Subcommittee SC 1, *Noise*, and ISO/TC 108, *Mechanical vibration and shock*.

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: Dynamic stiffness of elastic supports for translatory motion — Direct method*

— *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 dynamic stiffness of elastic supports for translatory motion*

Annexes A, B and C of this part of ISO 10846 are for information only.

Introduction

Passive vibration isolators of various kinds are used to reduce the transmission of vibrations. Examples are 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 specifies an indirect method for measuring the dynamic transfer stiffness function of linear resilient supports. This includes resilient supports with non-linear static load-deflection characteristics provided that the elements show an approximate linearity for vibrational behaviour for a given static preload. This part of ISO 10846 belongs to a series of International Standards on methods for the laboratory measurement of vibro-acoustic properties of resilient elements, which also includes parts on measurement principles and on a direct and a driving point method. ISO 10846-1 provides global guidance for the selection of the appropriate International Standard.

The laboratory conditions described in this part of ISO 10846 include the application of static preload, where appropriate.

The results of the indirect method are useful for isolators, which are used to reduce the transmission of structureborne sound (primarily frequencies above 20 Hz). The method does not characterize isolators completely, which are used to attenuate low frequency vibration or shock excursions.

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

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

1 Scope

This part of ISO 10846 specifies a method for determining the dynamic transfer stiffness for translations of resilient supports, under specific preload. The method concerns the laboratory measurements of vibration transmissibility and is called the indirect method. This method is applicable to test elements with parallel flanges (see Figure 1).

NOTE 1 Vibration isolators which are the subject of this part of ISO 10846 are those which are used to reduce the transmission of audiofrequency vibrations (structureborne sound, 20 Hz to 20 kHz) to a structure which may, for example, radiate unwanted fluidborne sound (airborne, waterborne or other).

NOTE 2 In practice the size of the available test rig(s) can give restrictions for very small and for very large resilient supports.

NOTE 3 Samples of continuous supports of strips and mats are included in the method. Whether or not the sample describes the behaviour of the complex system sufficiently, is the responsibility of the user of this part of ISO 10846.

Measurements for translations normal and transverse to the flanges are covered in this part of ISO 10846. Annex A provides guidance for the measurement of transfer stiffnesses which include rotatory components.

The method covers the frequency range from f_2 up to f_3. The values of f_2 and f_3 are determined by the test set-up and the isolator under test. Typically 20 Hz $\leqslant f_2 \leqslant$ 50 Hz and 2 kHz $\leqslant f_3 \leqslant$ 5 kHz.

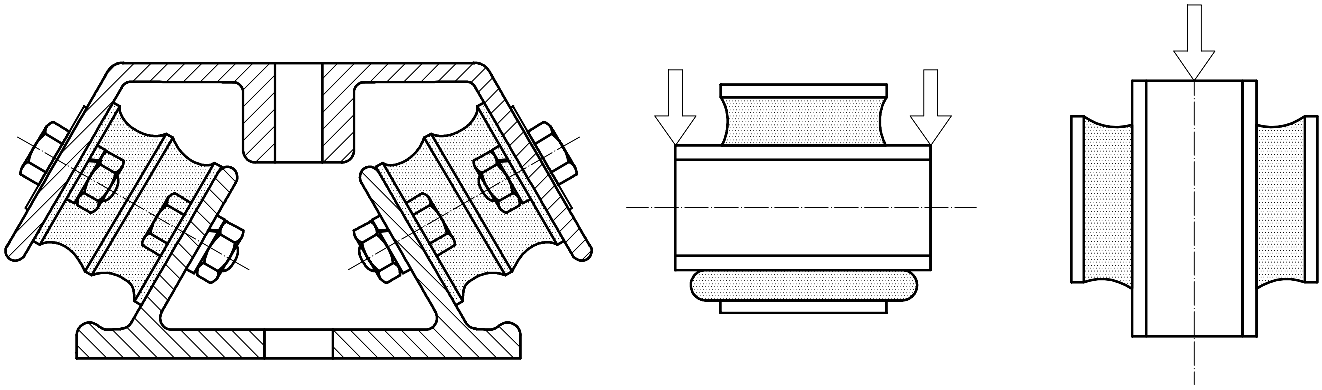
The data obtained according to the method specified in this part of ISO 10846 can be used for

— product information provided by manufacturers and suppliers,

— information during product development,

— quality control, and

— calculation of the transfer of vibration through isolators.



NOTE 1 When a resilient support has no parallel flanges, an auxiliary fixture should be included as part of the test element to arrange for parallel flanges.

NOTE 2 Arrows indicate load direction.

Figure 1 — Examples of resilient supports with parallel flanges

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ISO 10846. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this part of ISO 10846 are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 266, *Acoustics — Preferred frequencies*

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

ISO 5347-3[1)], *Methods for the calibration of vibration and shock pick-ups — Part 3: Secondary vibration calibration*

ISO 5348, *Mechanical vibration and shock — Mechanical mounting of accelerometers*

ISO 7626-1, *Vibration and shock — Experimental determination of mechanical mobility — Part 1: Basic definitions and transducers*

ISO 7626-2, *Vibration and shock — Experimental determination of mechanical mobility — Part 2: Measurements using single-point translation excitation with an attached vibration exciter*

1) To be revised as ISO 16063-21.

3 Terms and definitions

For the purposes of this part of ISO 10846, 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 vibration in a frequency range

[ISO 2041:1990, definition 2.110]

3.2
resilient support
vibration isolator suitable for supporting part of the mass of a machine, a building or another type of structure

3.3
test element
resilient support under test 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 zero displacement output

3.5
dynamic transfer stiffness
$k_{2,1}$
ratio of complex force on the blocked output side of a resilient element to complex displacement on the input side during sinusoidal vibration

NOTE 1 The indices “1” and “2” denote the input and output side respectively.

NOTE 2 The value of $k_{2,1}$ can be dependent upon static preload, temperature and other conditions. At low frequencies $k_{2,1}$ is solely determined by elastic and dissipative forces and $k_{2,1} = k_{1,1}$ ($k_{1,1}$ denotes the ratio of force and displacement on the input side).

NOTE 3 At higher frequencies inertial forces in the resilient element play a role as well and $k_{2,1} \neq k_{1,1}$.

3.6
loss factor of resilient element
η
ratio of the imaginary part of $k_{2,1}$ and 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

3.7
frequency-averaged dynamic transfer stiffness
k_{av}
function of frequency of the average value of the dynamic stiffness over a frequency band Δf (see 8.2)

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

3.9
normal translation
translational vibration normal to the flange of a resilient element

3.10
transverse translation
translational vibration in a direction perpendicular to that of the normal translation

3.11
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 $[a \cdot x_1(t) + b \cdot x_2(t)]$ produces the output $[a \cdot y_1(t) + b \cdot y_2(t)]$. This must hold for all values of a, b and $x_1(t)$ and $x_2(t)$; a and b are arbitrary constants.

NOTE 2 In practice the above test for linearity is impractical and a limited check of linearity is done by measuring the dynamic transfer stiffness for a range of input levels. In effect this procedure checks for a proportional relationship between the response and the excitation (see 7.6).

3.12
direct method
method in which either the input displacement, velocity or acceleration and the blocking output force are measured

3.13
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" may include loads of any known impedance other than a mass-like impedance. However, this part of ISO 10846 does not cover such methods.

3.14
transmissibility
T
ratio $\underline{u}_2/\underline{u}_1$ of the complex displacements $\underline{u}_2$ on the output side and $\underline{u}_1$ on the input side of the test element during sinusoidal vibration

NOTE For velocities v and accelerations a, transmissibilities are defined in a similar way and have the same value.

3.15
force level
L_F
level calculated by the following formula

$$L_F = 10 \lg \frac{F^2}{F_0^2} \text{ dB}$$

where F^2 denotes the mean square value of the force in a specific frequency band and $F_0 = 1\ \mu\text{N}$ is the reference force

3.16
acceleration level
L_a
level calculated by the following formula

$$L_a = 10 \lg \frac{a^2}{a_0^2} \text{ dB}$$

where a^2 denotes the mean square value of the acceleration in a specific frequency band and $a_0 = 10^{-6}\ \text{m/s}^2$ is the reference acceleration

3.17
level of dynamic transfer stiffness
$L_{k_{2,1}}$
level calculated by the following formula

$$L_{k_{2,1}} = 10 \lg \frac{\left|k_{2,1}\right|^2}{k_0^2} \text{ dB}$$

where $|k_{2,1}|^2$ is the square magnitude of the dynamic transfer stiffness (see 3.5) at a specified frequency and $k_0 = 1$ N·m^{-1} is the reference stiffness

3.18
level of frequency band averaged dynamic transfer stiffness
$L_{k_{av}}$
level calculated by the following formula

$$L_{k_{av}} = 10 \lg \frac{k_{av}^2}{k_0^2} \text{ dB}$$

where k_{av} is defined in 3.7 and where k_0 denotes the reference stiffness (= 1 N·m^{-1})

3.19
flanking transmission
forces and accelerations at the output side caused by the vibration exciter at the input side but via transmission paths other than through the test element

4 Principle

The measurement principle of the indirect method is discussed in ISO 10846-1.

The basic principle is that the blocking output force is derived from acceleration measurements on a compact body of mass m_2, which provides sufficiently small vibrations on the output side of the test element. This blocking mass shall be dynamically decoupled from the other parts of the test arrangement to prevent flanking transmission.

For sinusoidal vibration and using complex notation, the relation between the dynamic transfer stiffness (see 3.5) of the element under test and the measured vibration transmissibility (see 3.14) is given by

$$k_{2,1} = \underline{F}_{2,b} / \underline{u}_1 \approx -(2\pi f)^2 (m_2 + m_f) T \quad \text{for } |T| << 1 \tag{1}$$

where

m_f denotes the mass of the output flange of the test element;

indices “1” and “2” denote the input and output side respectively.

A valid indirect determination of a blocking force according to the right-hand term of equation (1) requires that this blocking force solely determines the corresponding vibration measured on the blocking mass. Therefore, in principle, the vibration to be measured is that of the mass centre of the compact body composed of the blocking mass and of the output flange of the test element, and in the direction of the wanted force.

5 Requirements for apparatus

5.1 Normal translations

5.1.1 Overview

In Figures 2 to 4, schematic examples are shown of test arrangements for resilient supports. These are exposed to translatory vibration in the normal load direction. The test element shall be mounted in a way which is representative of its use in practice.

NOTE The collection of examples is by no means exhaustive and is not intended to form a limitation for test arrangement principles.

To be suitable for the measurements according to this part of ISO 10846, a test rig shall include the items described in 5.1.2 to 5.1.6.

5.1.2 Blocking mass

A mass is connected to the output side the test element. One function of this mass is to block the output. The blocking force is determined from measuring the acceleration of the mass. A second function is to provide a uniform vibration of the output flange of the test element.

5.1.3 Static preloading system

Measurements shall be performed with the test element under a representative and specified preload. Examples of methods for applying the static preload are as follows.

a) Use of a hydraulic actuator, which also serves as the vibration exciter. This is mounted in a load frame together with the test element and the blocking mass on the output side of the test element. The blocking mass is supported by auxiliary vibration isolators to decouple the mass from the load frame. The total low frequency dynamic stiffnesss of these auxiliary isolators is of the same order of magnitude as that of the element under test.

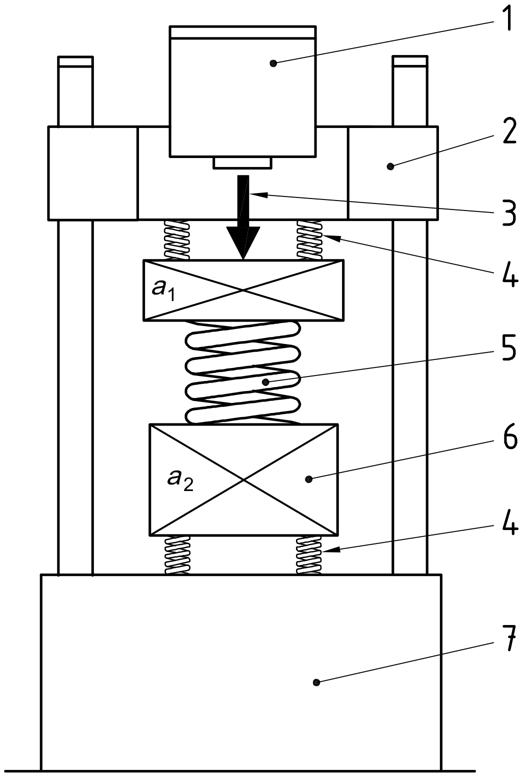
b) Use of a frame which provides static preload only; see Figures 2 and 3. If such a frame is used, auxiliary vibration isolators shall also be applied on the input side of the test element to decouple it from the frame.

c) Use of a gravity load by adding a blocking mass on top of the test element (with or without a support frame; see Figure 4).

5.1.4 Acceleration measurement systems

Accelerometers shall be mounted on the input and output side of the test element and on the foundation which supports the blocking mass. When mid-point positions are not accessible, indirect measurement of the mid-point accelerations shall be performed by making an appropriate signal summation, for example, by taking the linear average for two symmetrically positioned accelerometers.

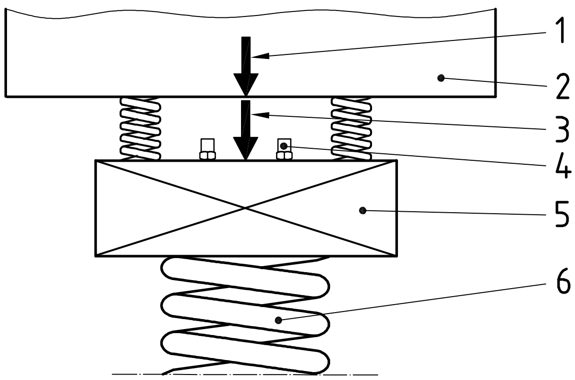
Provided that their frequency range is appropriate, displacement or velocity transducers may be used instead of accelerometers [see Figures 2b) and 4].



Key

1 Exciter
2 Traverse
3 Connection rod
4 Dynamic decoupling springs, static preload
5 Test element
6 Blocking mass
7 Rigid foundation

a) Overview

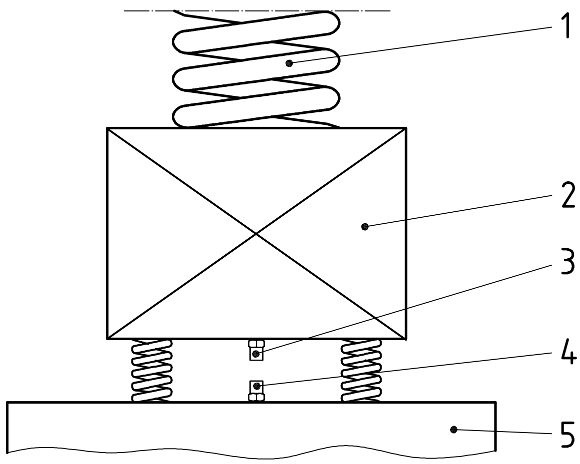


Key

1 Static preload
2 Traverse
3 Dynamic excitation
4 Acceleration measurement (a_1)
5 Excitation mass
6 Test element

b) Input side (details)

Figure 2 — Example 1 of laboratory test rig for measuring the dynamic transfer stiffness for normal translations (*continued*)



Key

1 Test element

2 Blocking mass (m_1)

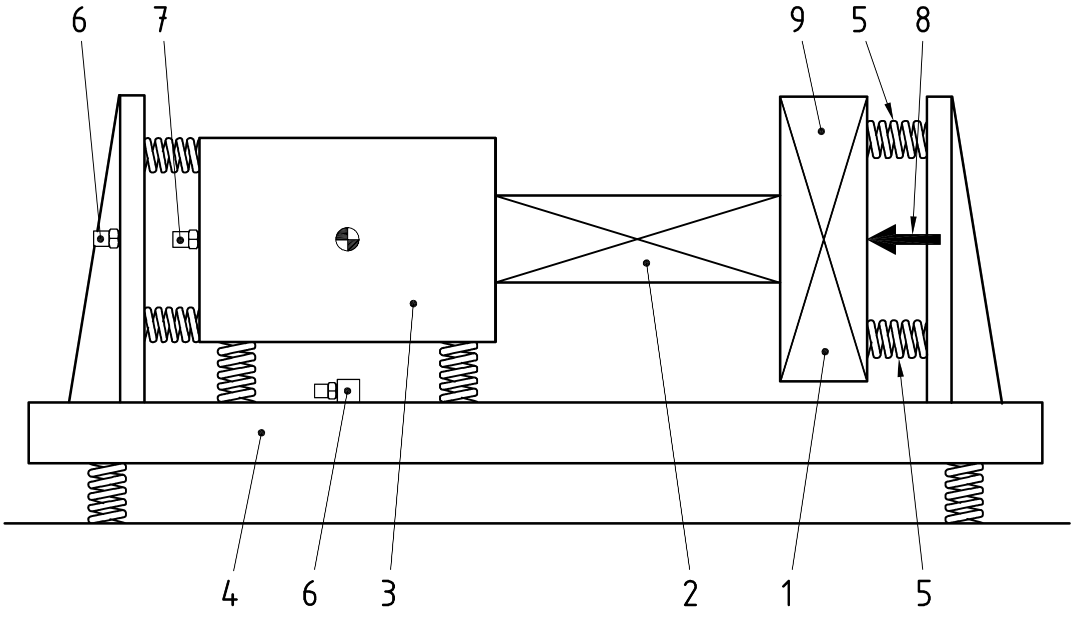
3 Acceleration measurement (a_2)

4 Acceleration measurement (a_3)

5 Rigid foundation

c) Output side (details)

Figure 2 — Example 1 of laboratory test rig for measuring the dynamic transfer stiffness for normal translations



Key

1 Excitation mass

2 Test element

3 Blocking mass

4 Rigid foundation

5 Dynamic decoupling springs

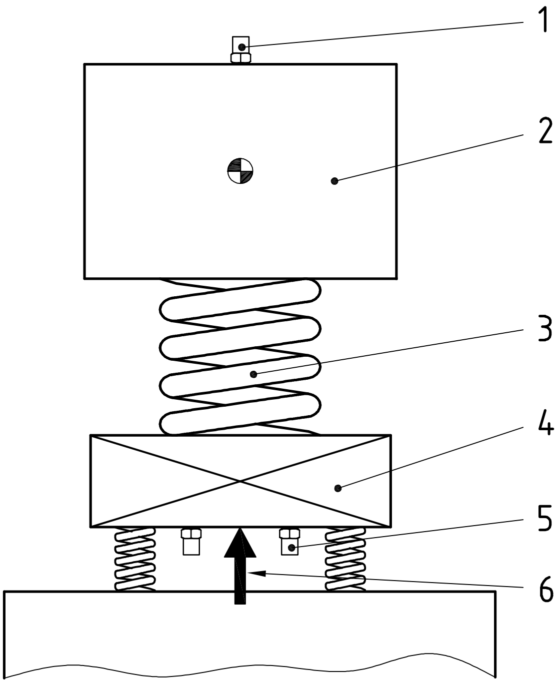
6 Acceleration measurement (a_3)

7 Acceleration measurement (a_2)

8 Dynamic excitation

9 Acceleration measurement (a_1)

Figure 3 — Example 2 of laboratory test rig for measuring the dynamic transfer stiffness for normal translations



Key

1 Acceleration measurement (a_2)
2 Blocking mass
3 Test element
4 Excitation mass
5 Acceleration measurement (a_1)
6 Vibration exciter

NOTE The gravity load due to the blocking mass is used as a static preload for the element under test.

Figure 4 — Example 3 of laboratory test rig for measuring the dynamic transfer stiffness for normal translations

5.1.5 Dynamic excitation system

The dynamic excitation system shall be appropriate for the frequency of interest.

Any suitable type of exciter is permitted. Examples are

a) a hydraulic actuator which also can provide a static preload,

b) one or more electrodynamic exciters (shakers) with connection rods, and

c) one or more piezo-electric exciters.

Vibration isolators may be used for dynamic decoupling of exciters to reduce flanking transmission via the frame for applying static preload. However, in the test rigs which use a hydraulic actuator, for both static and dynamic loading, such a decoupling is usually inconvenient, because of its adverse effects on low frequency measurements.

5.1.6 Excitation mass on the input side

The excitation mass on the input side of the test element has one or both of the following functions:

a) it provides a uniform vibration of the input flange under dynamic forces;

b) it enhances the unidirectional vibration of the input flange.

If the test element contains a solid mass-type input flange, which may provide the above-mentioned functions, the special excitation mass may be omitted.

NOTE Theoretically speaking, the measurements of the dynamic transfer stiffness for normal translation according to this part of ISO 10846 would require excitation with a unidirectional translation on the input side. However, for the majority of resilient supports this requirement is not of practical concern. In annex B of ISO 10846-1:1997, it is shown that the most common symmetry properties of resilient supports cause the blocking output forces in the direction of normal load due to input vibrations other than the normal translations to be negligible. For this to be true, it is required that the test element has two orthogonal planes of symmetry, the line of intersection of which forms the symmetry axis of the resilient support in the normal load direction. See annex A, equation (A.5) and annex B. For resilient supports with exceptional shapes, which do not have these symmetry properties, measurements according to this part of ISO 10846 require excitation with a predominant unidirectional translation (see further 5.3 and 6.4). Using a compact body of sufficiently large mass as a force distribution plate may facilitate such an excitation.

5.2 Transverse translations

5.2.1 Overview

Schematic representations of examples of test arrangements for resilient supports exposed to translatory vibrations perpendicular to the normal load direction are shown in Figures 5 to 7 (see note in 5.1.1). The test element shall be mounted in a way which is representative of its use in practice. To be suitable for the measurements according to this part of ISO 10846, a test rig shall include the items described in 5.2.2 to 5.2.6.

If measurements are performed with an arrangement as shown in Figure 7, the first eigenfrequency of rotational vibrations due to the blocking mass on the test element should be below the lowest frequency of the range of interest.

5.2.2 Blocking mass

See 5.1.2.

5.2.3 Static preloading system

Measurements shall be performed with the test element under a representative and specified preload. Examples of methods for applying the static preload are similar to those described in 5.1.3 and are shown in Figures 5 to 7.

5.2.4 Acceleration measurement systems

Accelerometers shall be mounted on the input and output side of the test element and on the foundation which supports the blocking mass.

On the input side, the accelerometers may be placed on horizontal axes of symmetry of the flange or excitation mass. On the blocking mass, the accelerometers may be placed on the horizontal axis through the mass centre of the compact body composed of the blocking mass and the output flange of the test element (see Figure 8). When such places are not accessible, indirect measurement of mid-point vibrations may be performed by making an appropriate signal summation, for example, by taking the linear average for two symmetrically positioned accelerometers.

Provided that their frequency range is appropriate, displacement or velocity transducers may be used instead of accelerometers.

5.2.5 Dynamic excitation systems

The dynamic excitation system shall be appropriate for the frequency range of interest.

Examples of vibration exciters are given in 5.1.5.

5.2.6 Excitation mass on the input side

The excitation mass on the input side of the test element has one or both of the following functions:

a) it provides a uniform vibration of the input flange under dynamic forces;

b) it enhances the unidirectional vibration on the input flange (see note in this subclause).

If the test element contains a mass-type input flange, which may provide the above-mentioned functions, the special excitation mass may be omitted.

NOTE Predominantly unidirectional translation on the input side of the test element is an essential requirement for the measurement of dynamic stiffnesses according to this part of ISO 10846 (see 6.4). For input translations in transverse directions, predominance of the required translation will be influenced by

a) the symmetry of the vibration excitation and boundary conditions of the excitation mass; see Figure 6;

b) the inertial properties of the excitation mass. In certain cases it will be necessary to apply external constraints such as roller bearings or some other guiding system to prevent vibrations in unwanted directions.

5.3 Suppression of unwanted vibrations

5.3.1 Overview

The test procedures according to this part of ISO 10846 cover measurements of transfer stiffnesses for unidirectional excitations one by one in the normal and transverse directions. However, due to asymmetries in excitation, in boundary conditions and in test elements properties, others than the intended input vibration component may show unwanted strong responses at certain frequencies. Qualitative measures to suppress unwanted input vibrations are discussed next. A special category of test arrangements is that in which two nominally equal test element are tested in a symmetrical configuration. This can be advantageous by suppressing unwanted input vibrations. Quantitative requirements are formulated in 6.4.

5.3.2 Normal direction

For excitation in the normal direction, unidirectional excitation is of practical concern only for those exceptional shapes of resilient supports that couple the normal and other vibration directions; see note in 5.1.6. For such cases the unwanted input vibrations may be strongly suppressed by taking the following measures:

— symmetrical positioning of the exciter or pair of exciters;

— using an axially symmetrical excitation mass;

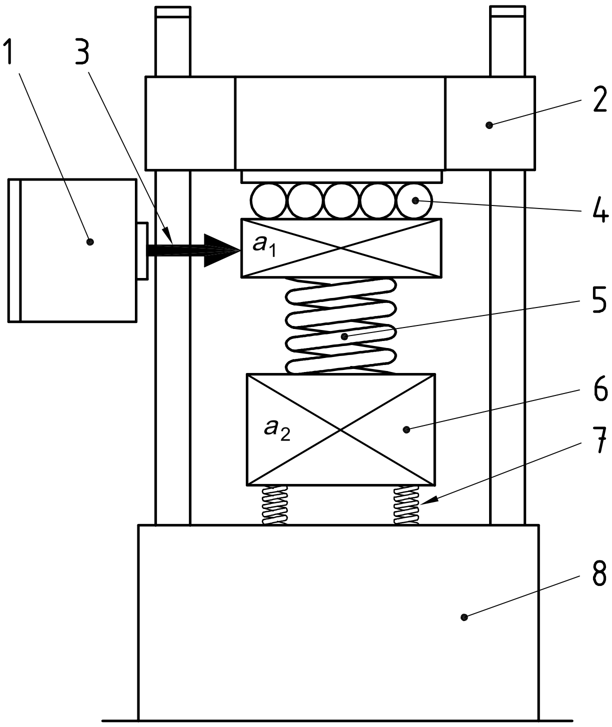
— using an excitation mass which has at the interface with the resilient support, large driving point impedances for transverse and rotational directions, compared to the corresponding driving point impedances of the test element.

Another method of suppressing unwanted input vibrations is the use of a symmetrical arrangement with two nominally identical test elements, or of a ‘guiding’ system on the sides of the excitation mass, for example, roller bearings. These systems are not shown in a figure, but they are rather similar to the examples for transverse excitation, which are shown in Figures 5 to 7.

5.3.3 Transverse direction

For excitation in the transverse direction, coupling between transverse and rotational input vibrations will always occur.

Some examples of measures which may enhance unidirectional vibrations on the input side are discussed. Figure 6 shows a symmetrical arrangement with two nominally identical test elements. Figures 5 and 7 show, as examples, how a guiding system can be used to suppress input rotations. Another method uses a symmetrical excitation block, which is excited along a line through its centre of gravity. In the frequency range where the impedances of the block for transverse and rotational directions exceed those of the test elements and of decoupling springs, the block vibrations will be strongly unidirectional.

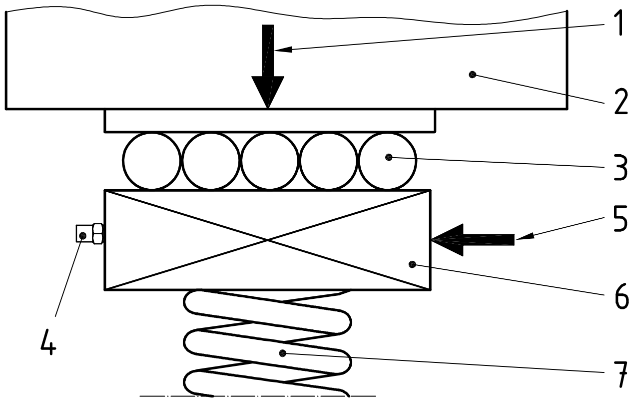


Key

1 Exciter
2 Traverse
3 Connection rod
4 Low friction bearing
5 Test element
6 Blocking mass
7 Dynamic decoupling springs, static preload
8 Rigid foundation

a) Complete system

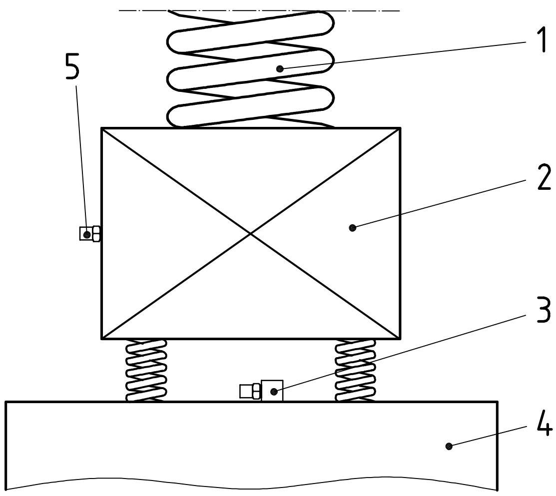
Figure 5 — Example 1 of laboratory test rig for measuring the dynamic transfer stiffness for transverse translations (*continued*)



Key

1 Static preload
2 Traverse
3 Low friction bearing
4 Acceleration measurement (a_3)
5 Dynamic excitation
6 Excitation mass
7 Test element

b) Input side (details)

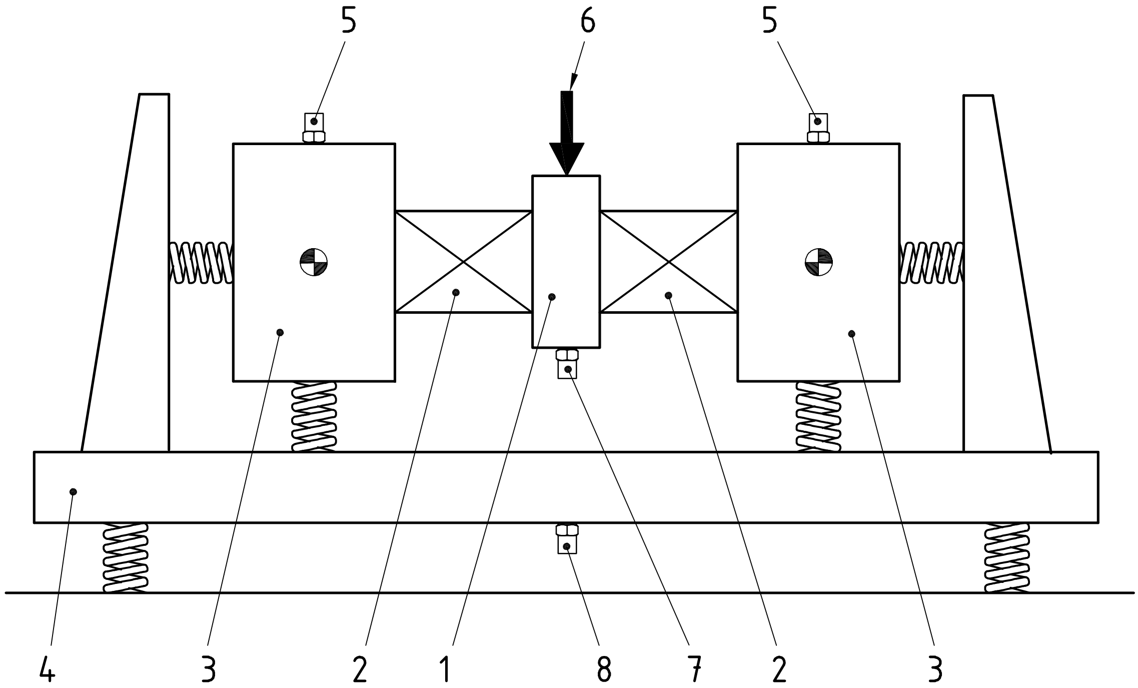


Key

1 Test element
2 Blocking mass (m_2)
3 Acceleration measurement (a_3)
4 Rigid foundation
5 Acceleration measurement (a_2)

c) Output side (details)

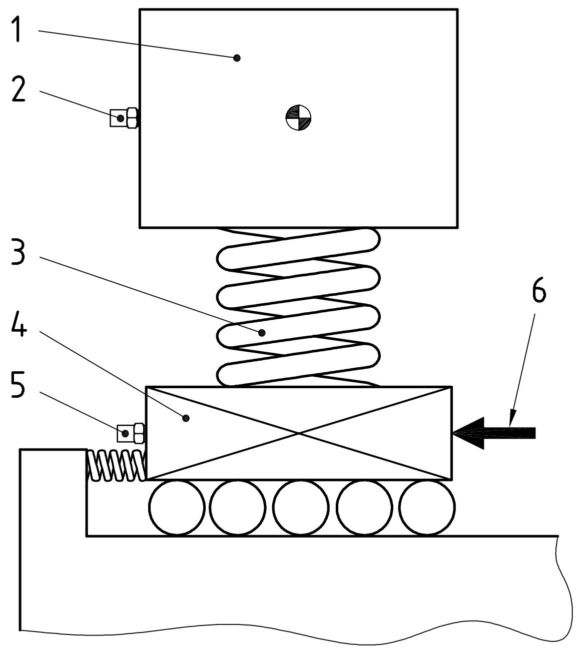
Figure 5 — Example 1 of laboratory test rig for measuring the dynamic transfer stiffness for transverse translations



Key

1 Excitation mass
2 Test element
3 Blocking mass
4 Rigid foundation
5 Acceleration measurement (a_2)
6 Dynamic excitation
7 Acceleration measurement (a_1)
8 Acceleration measurement (a_3)

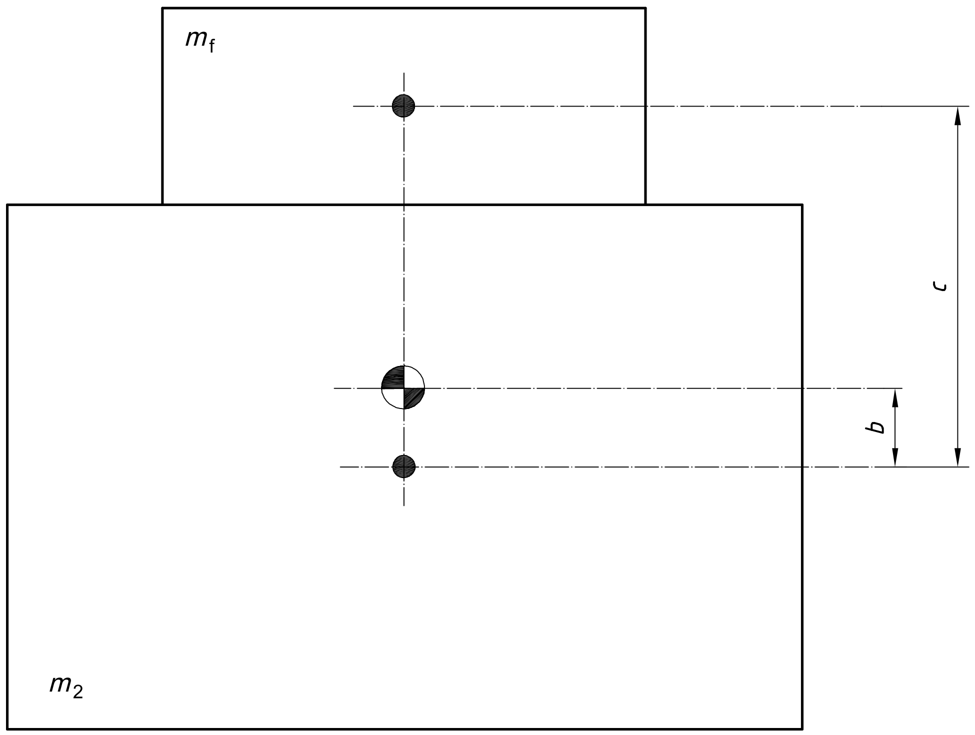
Figure 6 — Example 2 of laboratory test rig for measuring the dynamic transfer stiffness for transverse translations



Key

1 Blocking mass
2 Acceleration measurement (a_2)
3 Test element
4 Excitation mass
5 Acceleration measurement (a_1)
6 Vibration exciter

Figure 7 — Example 3 of laboratory test rig for measuring the dynamic transfer stiffness for transverse translations



NOTE The distance between the mass centres of the blocking mass and the flange equals c. The distance between the mass centre of the blocking mass and that of the compound body equals b:

$$b = \frac{c}{1 + m_2/m_f}$$

Figure 8 — Example of locating the mass centre of the compact body composed of blocking mass and output flange of the test element

6 Criteria for adequacy of the test arrangement

6.1 Frequency range

Each test facility has a limited frequency range in which valid tests can be performed. One limitation is given by the usable bandwidth of the vibration actuator.

Other limitations follow from the accuracy which is required for the approximation in using the transmissibility measurement, as in equation (1). In this part of ISO 10846, this approximation shall be accurate within 1 dB, i.e. within 12 % of the magnitude of the calculated stiffness. This requirement can only be met in a limited frequency range $f_2 < f < f_3$.

One requirement for obtaining this accuracy is a large impedance mismatch between the test element and the blocking mass in the direction of the blocking force which is determined. The measurements according to this part of ISO 10846 are valid only for those frequencies where inequality (2) is valid

$$\Delta L_{1,2} = L_{a_1} - L_{a_2} \geqslant 20 \text{ dB} \tag{2}$$

where a_1 denotes the acceleration of the input side and a_2 the acceleration of the blocking mass.

Below a certain frequency f_2, inequality (2) will be violated because of resonances in the system consisting of the test element, the load distribution plate, the blocking mass and auxiliary springs. Generally speaking, increasing m_2 of the blocking mass can lower f_2.

NOTE For design purposes, the lower natural frequencies of the test arrangements can be estimated with the aid of software for multi-body vibrations. The lower limit f_2 of the frequency range will be about three times the highest natural frequency of the vibration modes (including those with rotations), which can affect the measurement directions. Nevertheless, at certain frequencies above f_2 inequality (2) can be violated. Apart from test rig imperfections, stiffening of the test element due to internal resonance can cause this.

The other requirement for an accurate result using equation (1) is the validity of the assumption that the blocking mass vibrates as a rigid body with mass m_2. The size and the shape of the blocking mass can control the upper limit f_3 of the frequency range of valid measurements. This is discussed in 6.2.

6.2 Determination of upper frequency limit f_3

6.2.1 Effective mass

The upper frequency limit f_3 is a consequence of the fact that above a certain frequency the blocking mass used for the measurement of the blocking forces no longer vibrates as a rigid body. In this case a modified version of equation (1) is valid as follows:

$$k_{2,1} = \underline{F}_{2,\mathrm{b}} / \underline{u}_1 \approx -(2\pi f)^2 \left(m_{2,\mathrm{eff}} + m_\mathrm{f}\right) T \quad \text{for } |T| << 1$$

where $m_{2,\mathrm{eff}}$ denotes the effective mass of the blocking mass. The effective mass is defined as the frequency-dependent ratio between the excitation force which is exerted by the resilient element upon the blocking mass and the acceleration a_2 of the blocking mass. In principle, this quantity depends on the excitation direction, on the area over which the mass is excited, and on the position of the accelerometers.

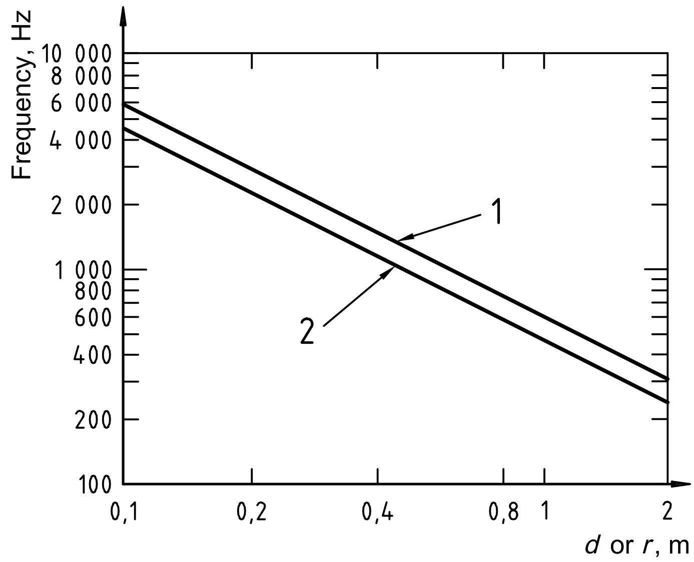
To be suitable for measurements according to this part of ISO 10846, results shall be presented for $f \leqslant f_3$ on basis of the procedures given in 6.2.2 and 6.2.3.

6.2.2 Use of preselected mass

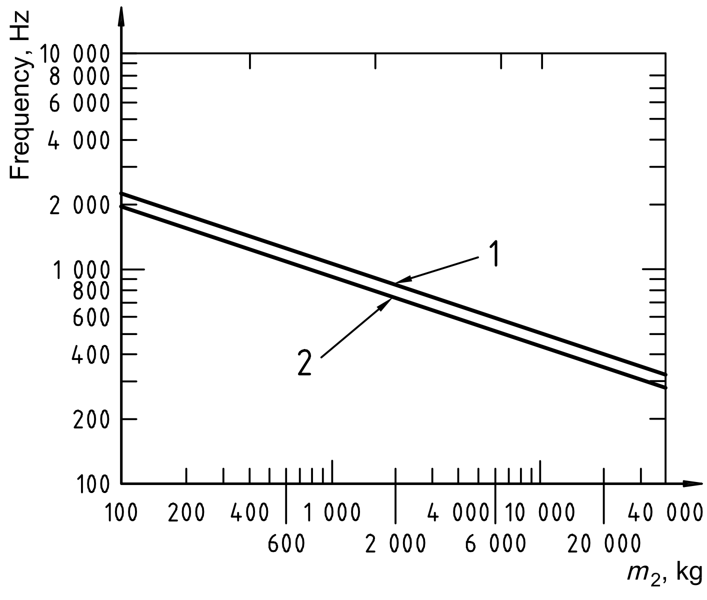
Figure 9 presents nomograms for solid steel blocks with the shape of a cube or a cylinder.

If one of these block shapes is used, transfer stiffness data can be calculated using equation (1) and for frequencies $f \leqslant f_3$, where f_3 for a given dimension is taken from Figure 9a).

NOTE Figure 9b) gives the relationship between the mass m_2 and f_3 for the cylindrical and the cubical blocks. A minimum value of m_2 is needed to obtain an appropriate value for f_2. Therefore, if a block with mass m_2 is to be chosen to obtain a certain value for f_2, then Figure 9b) can be used to determine the corresponding value of f_3 and then Figure 9a) can be used to determine the diameter d or rib length r.



a) f_3 according to this part of ISO 10846 for given block dimensions



b) f_3 according to this part of ISO 10846 for given block mass m_2

Key

1 For a solid steel cylinder

2 For a solid steel cube

r is the edge length of the cube
d is the diameter of the cylinder
h is the height of the cylinder
$d = h$

Figure 9 — Nomograms for solid steel cylinders and cubes

6.2.3 Experimental determination of effective mass

If the size, shape or mass of blocking masses covered by Figure 9 do not suffice for the purpose of measurements, alternative geometries are allowed. However, in this case f_3 must be determined experimentally. To uncouple translations and rotations, the blocking mass should have such a symmetry that, in a system of Cartesian co-ordinates with the mass centre as its origin and with the axes in normal and transverse directions of vibration, these coordinate axes coincide with the principal inertial axes.

To fulfil this requirement, blocks made of homogeneous materials and with shapes such as solid cylinders, annular cylinders, rectangular blocks or combinations of those may be used.

To determine f_3, the effective mass $m_{2,\mathrm{eff}}$ shall be determined as a function of frequency according to a procedure described below. The frequency f_3 is the lowest frequency at which the effective mass m_{eff} deviates more than 12 % (i.e. 1 dB in level) from the mass m_2.

Therefore, dynamic transfer stiffnesses of the element under test shall be calculated using equation (1) and are only presented for frequencies $f \leqslant f_3$, where the following inequality is valid:

$$\left|\Delta L\right| = \left|10 \lg (m_{2,\mathrm{eff}}^2 / m_2^2)\right| \mathrm{dB} \leqslant 1\ \mathrm{dB} \quad (3)$$

The procedure for the measurement of the effective mass is specified with the aid of Figure 10. Figures 10a) and 10b) show examples of a test element which is connected to a blocking mass over the contact area S. During the testing of the resilient element, $a_{2,\mathrm{vert}}$ or $a_{2,\mathrm{hor}}$ is measured depending on the direction of excitation on the input side of the element.

Figures 10c) and 10d) illustrate the determination of the effective mass for excitation in the vertical direction. The blocking mass without the element under test is supported by soft resilient elements. The natural frequency of this mass-spring system shall be below 10 Hz.

On the side where $a_{2,\mathrm{vert}}$ is measured during the test of a resilient element, now an excitation force F_2 is applied in the frequency range needed for testing the resilient element and along the axis through the mass centre. Within the contact area S two accelerometers shall be placed symmetrically to the vertical axis through the mass centre and with a spacing $D = \sqrt{S}$. The effective mass is defined as

$$m_{2,\mathrm{eff}} = \frac{2\underline{F}_2}{(\underline{a}'_1 + \underline{a}''_1)} \quad (4)$$

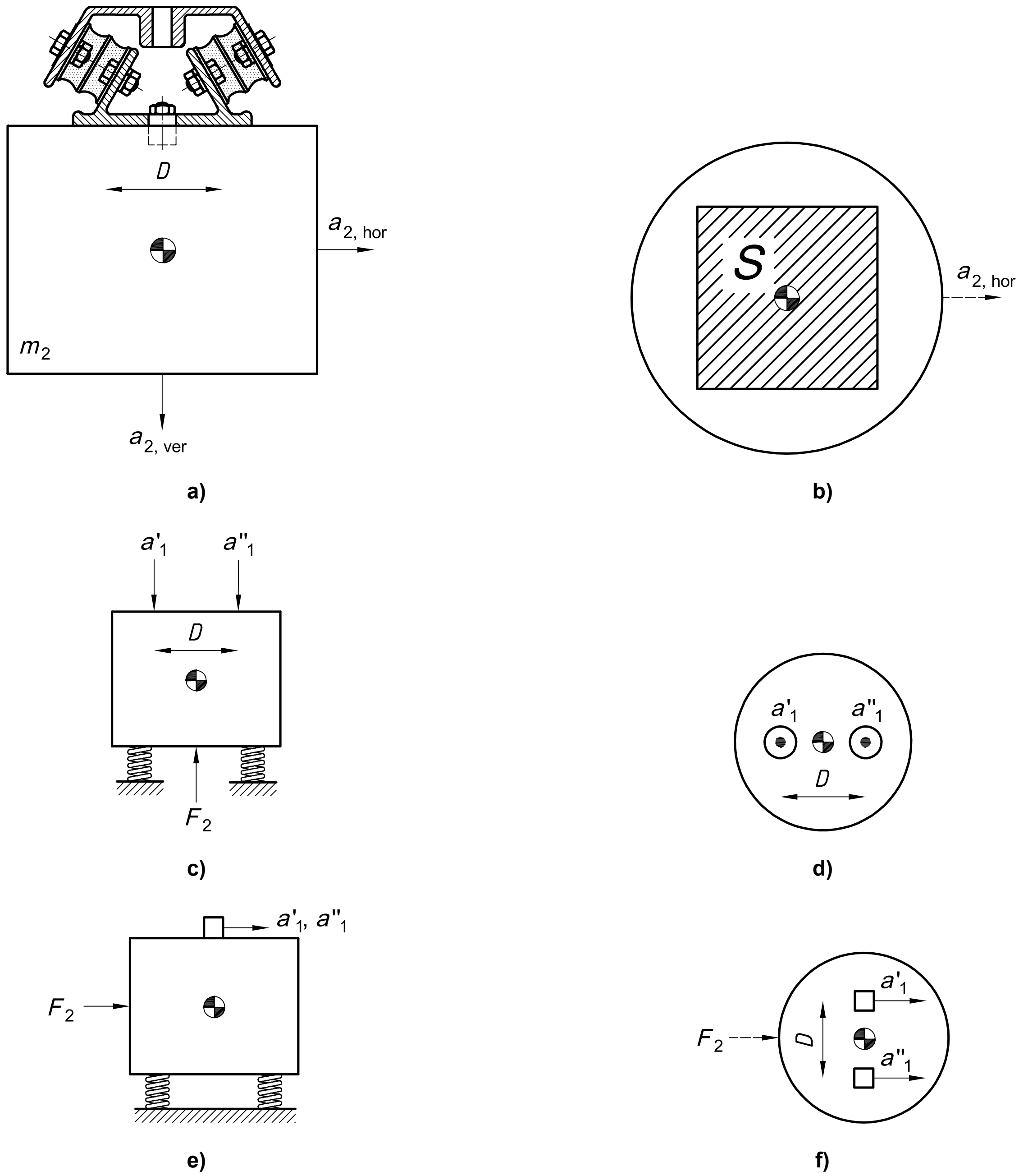
Figures 10e) and 10f) illustrate the determination of the effective mass for excitation in the horizontal direction. A similar procedure is applied as described for the vertical direction, but now for excitation along a horizontal axis through the mass centre and with two accelerometers within S in the horizontal direction. Again the spacing between these accelerometers shall be equal to $D = \sqrt{S}$ and again the effective mass is defined according to equation (4).

If at low frequencies (i.e. $f < 40$ Hz) $m_{2,\mathrm{eff}}$ deviates more than 1 dB in level from m_2, this deviation shall be ignored for the determination of f_3. The reason is that such a deviation will be caused by the mass-spring system behaviour and not by non-rigidity of the block.

The vibration exciter in the horizontal direction needs careful positioning to avoid excitation of block rotations. Otherwise, the measurement according to equation (4) will cause a bias error, which makes it impossible to meet inequality (3) even at lower frequencies.

The force and acceleration measurements shall be performed in accordance with the procedures of ISO 7626-1 and ISO 7626-2.

The realization of a wide frequency range for the measurements would require a low value of f_2, i.e. a heavy block, and at the same time a high value of f_3, i.e. a block as compact as possible. Use of a material with high density and wavespeed is thus preferable, for example, steel. If necessary, different blocking masses shall be used to cover the desired frequency range.



$D = \sqrt{S}$

S is the contact area

Figure 10 — Examples of experimental determination of the effective mass

NOTE For design purposes, it can be simple and cost-effective to determine f_3 on a scale model mass. Using a model of the blocking mass of the same material as the full-scale mass and with its linear dimensions reduced by a scale factor n, one will find f_3 (model) $= n \cdot f_3$ (full scale).

6.3 Flanking transmission

In many test arrangements flanking transmission can limit the applicability or accuracy of the test method. The flanking transmission may be due to airborne sound or due to structure-borne sound. Given the large variety of test arrangements which are allowed, it is the responsibility of the user of this part of ISO 10846 to design tests which make it plausible that the stiffness data which are presented have not been affected by flanking transmission.

These tests and their results shall be described in the test report.

6.4 Unwanted input vibrations

Input accelerations in directions other than those of the excitation shall be suppressed according to 5.3. Measurements according to this part of ISO 10846 are only valid when the input acceleration level in the excitation direction exceeds that in the other directions perpendicular to it by at least 15 dB, i.e.

$$L_a \text{ (excitation)} - L_{a'} \text{ (unwanted)} \geqslant 15 \text{ dB} \tag{5}$$

The measurement positions where this requirement shall hold are shown in Figure 11.

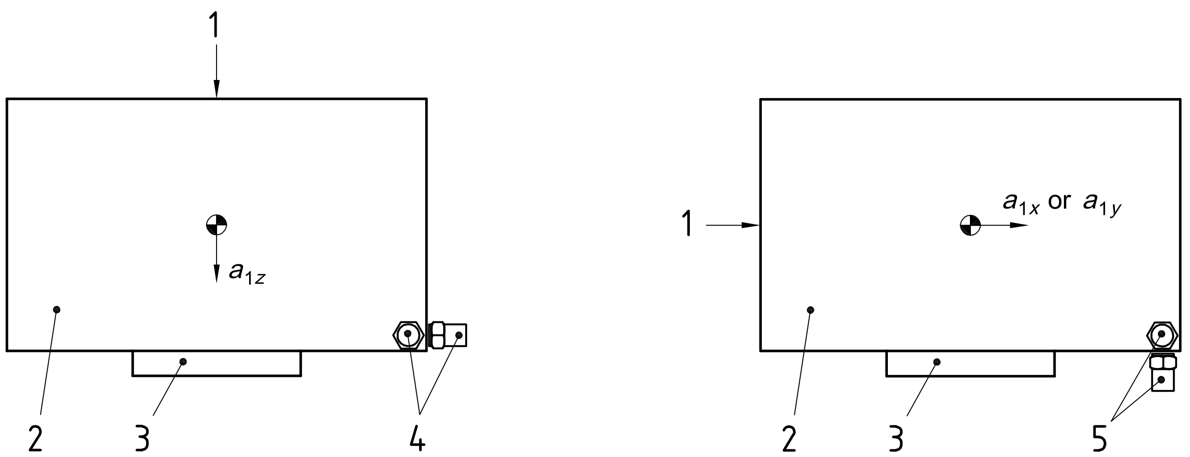
For normal excitation, the input vibration in the excitation direction a_{1z} is along the line of excitation and at the interface between the excitation mass and the input flange. The unwanted inputs in transverse directions a'_{1x} and a'_{1y} shall be measured at the edge of the excitation mass or force distribution plate and in the plane of the interface between the excitation mass and the input flange [see Figure 11a)].

Many types of resilient supports are sufficiently symmetrical about the axis of normal load to uncouple the normal vibrations from the other vibration directions; see note in 5.1.6, and annex B.

If the user of this part of ISO 10846 shows that the test element has such symmetry, it is not required to obey inequality (5) for tests with normal excitation. For this purpose it may be sufficient to report to which class of symmetry the test element belongs (see annex B of ISO 10846-1:1997).

For transverse excitation (x- or y-direction), the input vibration in the excitation direction (a_{1x} or a_{1y}) is measured along a horizontal symmetry axis of the excitation mass. The unwanted inputs a'_{1z} and a'_{1y} or a'_{1x} shall be measured at the edge of the excitation mass and in the plane of the interface between the excitation mass (see Figure 11b).

When the mass-type input flange of the test element replaces the excitation mass (see note in 5.1.6), a configuration similar to that in Figure 11 shall be defined to test the adequacy of the suppression of unwanted inputs according to inequality (5).



a) **Normal excition** (z-direction)

b) **Transverse excitation** (x- or y-direction)

Key

1 Exciter

2 Excitation mass

3 Input flange of test object

4 Unwanted vibrations a'_{1x} and a'_{1y}

5 Unwanted vibrations a'_{1z} and a'_{1y} or a'_{1x}

Figure 11 — Measurement locations for checking the suppression of unwanted input vibrations

6.5 Accelerometers

Accelerometers shall be calibrated in the frequency range of interest and shall have a sensitivity level which is frequency independent to within 0,5 dB. Calibration shall be carried out according to ISO 5347-3.

The accelerometers shall not be sensitive to extraneous environmental effects such as temperature, humidity, magnetic fields, electrical fields, acoustical fields and strain, and the sensitivity to cross-axis accelerations shall be smaller than 5 % of the main axis sensitivity.

If displacement or velocity transducers are used, the same requirements as for accelerometers apply.

6.6 Force transducers

Force transducers shall be used which are calibrated in the frequency range of interest and which have a sensitivity level which is frequency independent to within 0,5 dB. Calibration shall be carried out according to the mass-loading technique as described in ISO 7626-1.

If an appropriate routine is available, i.e. an appropriate transfer function can be applied digitally, the resultant sensitivity function shall meet the 0,5 dB requirement.

The force transducers shall not be sensitive to extraneous environmental effects such as temperature, humidity, magnetic fields, electrical fields, acoustical fields and strain, and the sensitivity to cross-axis forces shall be less than 5 % of the main axis sensitivity.

6.7 Summation of signals

If signals from force transducers or from accelerometers have been added, this shall be performed with a maximum tolerance of 5 %. One way to realise this is to use identical transducers with sensitivities within 5 % of each other. Another way is to perform the summation with the aid of a multi-channel analyser. In that case corrections shall be made to compensate both for differences in transducer sensitivities larger than 5 % and for differences in channel gain factors (see 6.8).

6.8 Analysers

Narrow-band analysers shall be used which fulfil the following requirements.

a) In the frequency range of interest, the spectral resolution shall provide at least five distinct frequencies per one-third-octave band for frequencies above f_2.

b) The difference in frequency responses between the channels (including signal conditioning equipment) which are used for the acceleration measurements on the input and output side shall be less than 0,5 dB for a measurement with the same frequency resolution as used for testing the resilient support. Otherwise corrections shall be made to compensate for the differences in channel gain factors.

One way in which channel gains can be compared is as follows. An identical broadband signal (e.g. white noise) is applied as input on both channels. Then the narrow-band spectrum of the level of the magnitude of the output ratio should be less than 0,5 dB, otherwise the measured gain ratio shall be used as a correction factor for the measured dynamic stiffness.

7 Test procedures

7.1 Installation of the test elements

The test element is attached to the excitation mass and to the blocking mass in a way which ensures good contact over the entire surface of the flanges. Devices which are not part of the resilient element in practical application shall be de-activated and removed.

NOTE 1 To improve a good contact between the resilient support and the mass on both sides, grease or double-sided tape may be added. However, in the latter case problems can occur in the high frequency range. For test elements with big flanges, flattening may be necessary to obtain unambiguous test results.

Test elements that contain rubber-type components will show a change of load or deflection due to creep. For such elements preloading shall be applied to 100 % of the permissible static load. Change of load or deflection due to creep should be less than 10 % per day before valid measurements can be performed.

NOTE 2 Transducers for measuring static load or static displacements and documentation for their application are commonly available. Appropriate selection will take into account the required load or deflection range of the resilient test element.

No particular preloading procedure is required for steel springs, but the appropriate preload shall be applied.

7.2 Mounting and connection of accelerometers

Accelerometers are mounted on the input and output sides of the test element, to measure a_1, a_2 and a_3 respectively. The connection shall be stiff. Mounting shall be carried out in accordance with ISO 5348.

7.3 Mounting and connections of the vibration exciter

A connection rod may be necessary between the vibration source and the excitation mass. It shall be designed in such a way that strong transverse vibrations and sound radiation are avoided due to resonance of this rod.

7.4 Source signal

One of the following source signals may be used:

— a discretely stepped sinusoidal signal;

— a swept sine signal;

— a periodically swept sine signal; or

— a bandwidth-limited noise signal.

The source signal shall be applied for a sufficiently long time to allow for averaging so that the measured results do not differ more than 0,1 dB when the averaging time is doubled. When discretely stepped sinusoidal signals or periodically swept sine signals are used, the spacing of the frequencies of the source signal shall be such that for $f > f_2$ each one-third-octave band of the analysis contains at least five frequencies of the source signal.

7.5 Measurements

7.5.1 General

The measurements shall be carried out under one or more specified load conditions, representing the range of loads in practice.

The measurements shall be carried out under one or more specified environmental temperatures, representing the range of environmental temperatures in practice. The environmental temperature shall be monitored during the measurements. The resilient elements under test shall be exposed for at least 24 h to the appropriate environmental temperature within a tolerance of 3 °C, before they are tested.

If it is known or if it is reasonable to expect that the material properties of the element under test (e.g. damping) are very sensitive to changes in the temperature or humidity of the element, tolerances for the temperature and humidity shall be defined within which the measurement uncertainty according to 7.5.3 is maintained.

In a pre-run, the acceleration level L_{a_2} shall be determined with and without the vibration source in operation. If possible and unless otherwise specified, the source output is adjusted to obtain a minimum level difference of 15 dB in all frequency bands of interest, compared to the measurements with the source switched off.

The main measurements are carried out for the acceleration a_1 on the input side of the test element, for the acceleration a_2 on the output-side and for the acceleration a_3 on the foundation of the test rig. Measurement results which do not meet the conditions of 6.4 shall be excluded from the evaluation of the dynamic stiffness function.

7.5.2 Validity of the measurements

Conditions for the validity of the measurement method are the following:

a) approximate linearity of the vibrational behaviour of the isolator (see 7.6);

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

It is the responsibility of the user to demonstrate the frequency range of validity.

7.5.3 Measurement uncertainty

The standard deviation of the dynamic transfer stiffness of a resilient element measured according to this part of ISO 10846, is approximately 2 dB in level or approximately 26 % in magnitude.

NOTE The standard deviation can be frequency dependent. Given the present state of the art, further testing on measurement accuracy is needed. For this purpose an interlaboratory test will be organized.

7.6 Test for linearity

In the ISO 10846 series, the concept of dynamic transfer stiffness and the methods to measure it are based on linear models for the vibration behaviour of resilient elements. However, real vibration isolators show at best only approximately linear behaviour. Therefore, to define precisely what is accepted in this part of ISO 10846 as approximately linear, the validity of dynamic transfer stiffness data in relation to input vibration levels will be considered.

Because a full test on linearity is impractical, data measured according to this part of ISO 10846 shall be checked with respect to the degree of proportionality between output and input, in terms of the ratio of force output to the input acceleration (or velocity, or displacement); see 3.11, notes 1 and 2.

The validity of dynamic transfer stiffness data measured according to this part of ISO 10846 may only be claimed for input amplitudes which are equal to or lower than those applied in the tests, and for which approximate proportionality between output and input has been proved. This upper boundary of input levels for which valid data may be claimed shall be specified in the test report.

For the proportionality test, the following procedure shall be applied.

a) Let A be a one-third-octave-band spectrum of input levels.

b) Let B be another input spectrum, with one-third-octave-band levels at least 10 dB lower than A.

c) If the transfer stiffness levels for both excitation spectra A and B do not differ by more than 1,5 dB, then the transfer stiffness data shall be considered as valid in the range of input levels (or corresponding input amplitudes) equal to or smaller than those of A.

d) If the maximum levels of A which are possible in the test rig are lower than typical input levels in practical applications of the tested elements, the test rig shall be modified or another test rig shall be used in order to obtain valid data for those applications.

e) If the tests as described under c) lead to unacceptable results, the tests shall be repeated with lower input levels until a valid input level range has been determined for proportionality between output and input.

The range of valid input levels shall be specified as the values of one-third-octave-band levels of the input accelerations (or displacements if input displacements have been measured), which are equal to or lower than those in the test with the higher input levels and with valid results.

NOTE On basis of the upper boundary of input levels, simplified information can be derived which may optionally be presented. For example, this may be a maximum of r.m.s. input displacement.

If a test element fails to meet the above-mentioned criteria for proportionality between input and output, it shall be considered as non-linear. This part of ISO 10846 does not provide a measurement procedure for such cases. Nevertheless, large parts of it may then still be used to define application oriented test procedures, e.g. for sinusoidal excitations with prescribed amplitudes.

8 Evaluation of test results

8.1 Evaluation of dynamic transfer stiffness

The dynamic transfer stiffness is calculated from equation (1).

Within the same limitations and additional requirements on the measurement precision which provide for the vibration transmissibility T, the loss factor $\eta(f)$ of the test element is calculated in accordance with 3.6 from

$$\eta(f) = \mathrm{Im}\{T(f)\} / \mathrm{Re}\{T(f)\} \tag{6}$$

The evaluation of the loss factor is optional. For higher frequencies the resilient support no longer behaves as a massless spring. Then it is no longer correct to use equation (6) as a characterization of the damping properties of the resilient support (see ISO 10846-1).

If the loss factor is very small, then its evaluation using equation (6) will become very sensitive to errors. For example, a loss factor $\eta = 0{,}01$ corresponds to a phase angle $\phi = \mathrm{arctg}(\eta) = 0{,}57°$ for T. In such cases it is recommended to design a test with a so-called half-value bandwidth method.

8.2 One-third-octave-band values of the frequency-averaged dynamic transfer stiffness

One-third-octave-band averages of $k_{2,1}$ are obtained as follows:

$$k_{\mathrm{av}} = \left\{ \frac{1}{n} \sum_{i=1}^{n} \left| k_{2,1}(f_i) \right|^2 \right\}^{\frac{1}{2}} \tag{7}$$

where the summation is performed over a minimum of $n = 5$ frequencies.

NOTE 1 Averaging over the squared magnitude is chosen to emphasize the maxima in the stiffness values, which usually are the most important ones.

NOTE 2 The result of equation (7) is consistent with the frequency-averaged result measured directly with a real time one-third-octave band analyser, in the case of a flat power spectral density function of the input displacement u_1.

NOTE 3 It is evident that the presentation in terms of one-third-octave-band stiffnesses forms a practical reduction of the amount of data. However, phase information is lost.

The results are presented in terms of the level of frequency-averaged dynamic transfer stiffness in accordance with 3.18.

Geometric centre frequencies f_{m} for one-third-octave pass bands shall be used in agreement with ISO 266.

8.3 Presentation of one-third-octave-band results

The presentation of the dynamic transfer stiffness levels for one-third-octave bands may be in the form of tables and/or in graphical form. A table shall contain centre frequencies of one-third-octave bands, levels of dynamic transfer stiffness in decibels, and specification of the reference value (i.e. 1 $N \cdot m^{-1}$).

The format of the graphs shall be as follows:

— vertical scale: 20 mm for 10 dB or equivalent for a factor $10^{1/2}$ in magnitude;

— horizontal scale: 5 mm per one-third-octave band.

In print these dimensions may be enlarged or reduced, provided that the proper ratio is maintained. Grids may be used for the sake of clarity.

NOTE An example of the graphic format is shown in Figure 12. In addition to the decibel scale (vertical scale on the left) a logarithmic vertical scale in newton per metre is given on the right.

The presentations shall include a clear description of the transfer stiffness concerned.

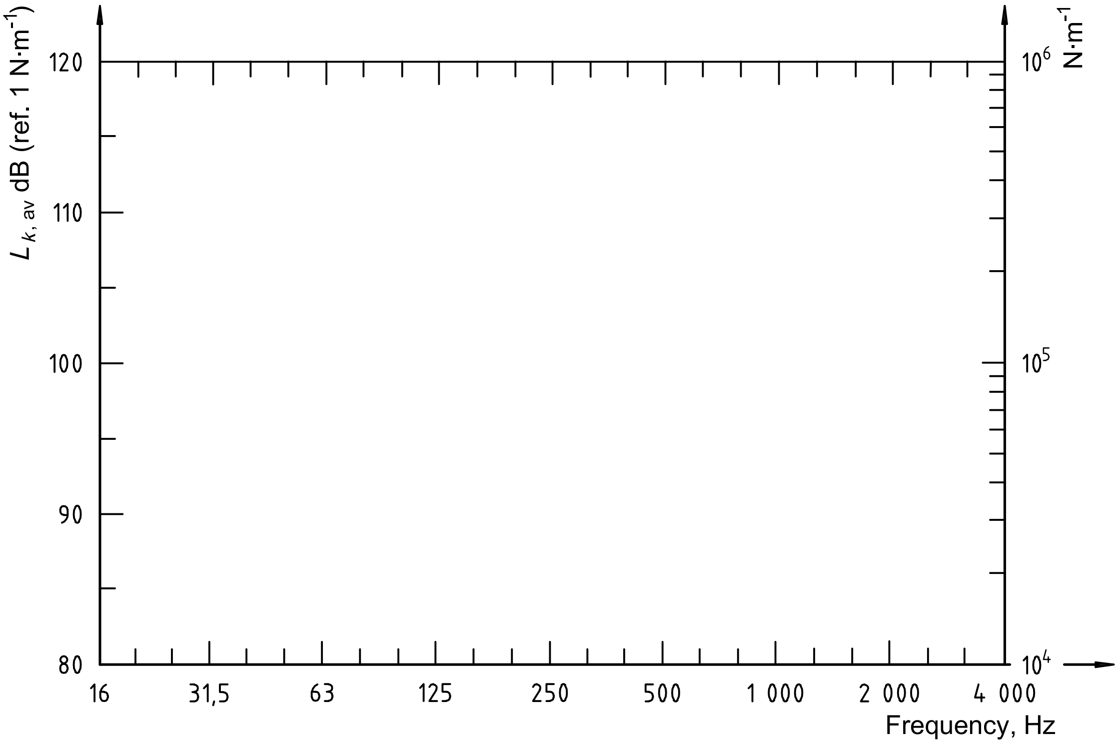


Figure 12 — Example of the format of the graph for presenting one-third-octave-band levels of the dynamic transfer stiffness with an example of scale values

8.4 Presentation of narrow-band data

The magnitude and phase spectra of the dynamic transfer stiffness and spectra of the loss factor may be presented optionally. The frequency resolution of the narrow-band analysis shall then be used.

It is the responsibility of the user of this part of ISO 10846 to provide sufficient complementary information on the accuracy of the narrow-band phase or loss factor data.

The presentation of the level of the magnitude of the dynamic stiffness shall be in graphical form and shall specify the reference value (i.e. 1 $N \cdot m^{-1}$). The format of the graphs shold preferably be as follows:

— vertical scale: 20 mm for 10 dB or equivalently for a factor $10^{1/2}$ in magnitude;

— horizontal scale: 15 mm per octave band.

NOTE See further remarks in 8.3 on printing.

The presentation of the phase data shall be in graphical form.

The format of the graphs shall be preferably as follows:

— vertical scale: 40 mm for the range −180° to +180°;

— horizontal scale: 15 mm per octave band.

NOTE See further remarks in 8.3 on printing.

The presentation of the loss factor shall be in graphical form. The format of the graphs should preferably be as follows:

— vertical scale: 20 mm for a factor 10 in loss factor η;

— horizontal scale: 15 mm per octave band.

NOTE See remarks in 8.3 on printing.

The presentations shall include a clear description of the transfer stiffness concerned.

9 Information to be recorded

If relevant during the measurements, the following data shall be recorded:

— environmental temperature (including its variation during the tests), in degrees Celsius;

— static preload(s), in newtons;

— relative humidity, in percent.

10 Test report

The test report shall make reference to this part of ISO 10846 and shall include at least the following information:

a) the name of the organization that performed the test;

b) information on the test element, including

  — manufacturer, type, serial number,

  — description of the element; in cases where that is not self-evident, the test element and non-test elements (auxiliary parts not included in the tests) shall be clearly defined,

  — data provided by the manufacturer in relation to the application as vibration attenuator;

c) photograph(s) of resilient element and test arrangement; description of auxiliaries for static preload(s);

d) descriptions of the excitation mass, if present, and of the blocking mass (dimensions, material, mass) and of the attachment to the test element;

e) spectra of acceleration level differences to check equation (2) and equation (5) (see 6.1 and 6.4);

f) test conditions

 — environmental temperature(s) and variation during the tests, in degrees Celsius,

 — static preload(s), in newtons or pascals,

 — any other relevant special condition (e.g. static deflection and super-imposed low frequency vibration: amplitude, frequency);

g) description of test signal(s);

h) spectrum of acceleration level L_{a_1} at the input side of the test element (displacement levels if displacements have been measured);

i) the measurement and analysis equipment used, including type, location, serial number, calibration and manufacturer;

j) presentation of frequency-averaged dynamic transfer stiffness, in one-third-octave band levels;

k) description of the linearity test, see 7.6, including data on levels or amplitude range of the acceleration a_1 or the displacement u_1 for which the test data are considered to be valid.

The following are optional:

l) measurement data for 10 lg ($m_{2,\text{eff}}^2/m_2^2$) to determine f_3 (see 6.3);

m) narrow-band magnitude spectra of dynamic transfer stiffness;

n) narrow-band phase spectra of dynamic transfer stiffness;

o) narrow-band spectra of the loss factor, including a statement (with reference to ISO 10846-1) that η is only directly representative of the dissipation losses at low frequencies, where inertial forces inside the test element are negligible;

p) static load-deflection curve, see annex B;

q) real and imaginary parts of transfer stiffness;

r) simplified information of upper boundary of input levels for which the test data are considered to be valid (e.g. a maximum of r.m.s. displacement);

s) tolerances of test element temperature within which the maximum measurement uncertainty according to 7.6.3 is maintained;

t) relative humidity, in percent;

u) description of tests on possible influence of background noise;

v) description of test on possible influence of flanking transmission.

Annex A
(informative)

Transfer stiffness related to rotatory vibration components

A.1 General

In all parts of ISO 10846, only the measurements of transfer stiffnesses for translatory vibration components are standardized. However, the indirect method which is covered in this part of ISO 10846 can be extended to determine transfer stiffnesses for rotatory components as well.

This annex describes the theory, measurement principles and also the adaptation of the test rigs described in clause 5 to perform measurements with rotatory components; see bibliographic references [8] and [9].

A.2 Theory

For a single resilient vibration isolator, the transfer stiffness matrix is a 6 × 6 matrix according to equation (A.1). See ISO 10846-1 for a description of the complete 12 × 12 stiffness matrix.

$$\begin{Bmatrix} \underline{F}_{2x} \\ \underline{F}_{2y} \\ \underline{F}_{2z} \\ \underline{M}_{2x} \\ \underline{M}_{2y} \\ \underline{M}_{2z} \end{Bmatrix} = \begin{bmatrix} k_{F_{2x},u_{1x}} & k_{F_{2x},u_{1y}} & k_{F_{2x},u_{1z}} & k_{F_{2x},\gamma_{1x}} & k_{F_{2x},\gamma_{1y}} & k_{F_{2x},\gamma_{1z}} \\ k_{F_{2y},u_{1x}} & k_{F_{2y},u_{1y}} & k_{F_{2y},u_{1z}} & k_{F_{2y},\gamma_{1x}} & k_{F_{2y},\gamma_{1y}} & k_{F_{2y},\gamma_{1z}} \\ k_{F_{2z},u_{1x}} & k_{F_{2z},u_{1y}} & k_{F_{2z},u_{1z}} & k_{F_{2z},\gamma_{1x}} & k_{F_{2z},\gamma_{1y}} & k_{F_{2z},\gamma_{1z}} \\ k_{M_{2x},u_{1x}} & k_{M_{2x},u_{1y}} & k_{M_{2x},u_{1z}} & k_{M_{2x},\gamma_{1x}} & k_{M_{2x},\gamma_{1y}} & k_{M_{2x},\gamma_{1z}} \\ k_{M_{2y},u_{1x}} & k_{M_{2y},u_{1y}} & k_{M_{2y},u_{1z}} & k_{M_{2y},\gamma_{1x}} & k_{M_{2y},\gamma_{1y}} & k_{M_{2y},\gamma_{1z}} \\ k_{M_{2z},u_{1x}} & k_{M_{2z},u_{1y}} & k_{M_{2z},u_{1z}} & k_{M_{2z},\gamma_{1x}} & k_{M_{2z},\gamma_{1y}} & k_{M_{2z},\gamma_{1z}} \end{bmatrix} \begin{Bmatrix} \underline{u}_{1x} \\ \underline{u}_{1y} \\ \underline{u}_{1z} \\ \underline{\gamma}_{1x} \\ \underline{\gamma}_{1y} \\ \underline{\gamma}_{1z} \end{Bmatrix} \quad \text{(A.1)}$$

The transfer stiffnesses in the matrix of equation (A.1) determine the ratios between the blocking forces and torques on the outside on the one hand, and the translatory and rotatory displacements on the input side on the other hand.

Figure A.1 shows the orthogonal system of the translatory and rotatory forces and displacements.

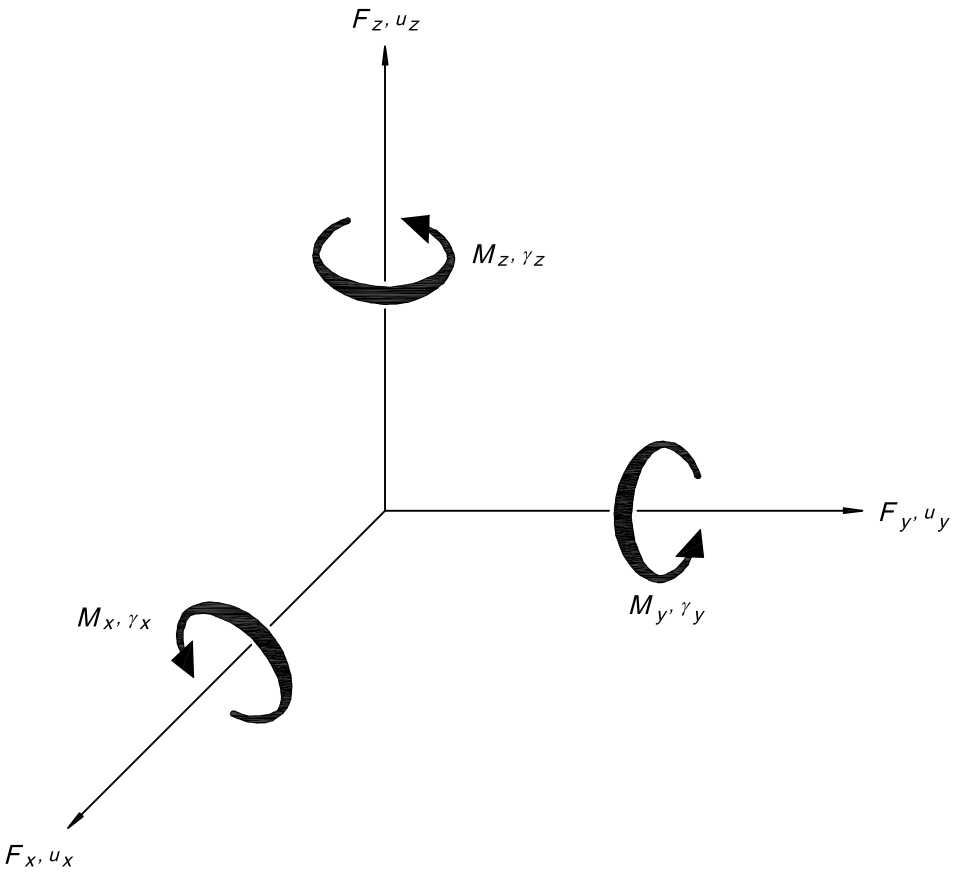


Figure A.1 — Cartesian coordinate system with forces, torques, displacements and rotary displacements

A shorter notation for equation (A.1) is given by

$$\{\underline{F}_2\}_b = [k_{2,1}] \cdot \{\underline{u}_1\} \quad \text{(A.2)}$$

A similar equation is valid when the input and output positions are reversed, i.e. when the displacements are enforced at position 2 and the vibrations are blocked at position 1.

Then equation (A.2) is replaced by

$$\{\underline{F}_1\} = [k_{1,2}] \cdot \{\underline{u}_2\} \quad \text{(A.3)}$$

Because of reciprocity

$$[k_{2,1}] = [k_{1,2}]^T \quad \text{(A.4)}$$

The equality in equation (A.4) has the important practical implication that sometimes a difficult measurement of an element of $[k_{2,1}]$ can be replaced by a simpler measurement of the corresponding elements of $[k_{1,2}]$. This means that a measurement is performed with the resilient element under test turned "upside down".

Usually, for practical isolators, symmetry causes the number of non-zero transfer stiffnesses to be much smaller than 36.

NOTE See Figure A.1 for the corresponding coordinate system.

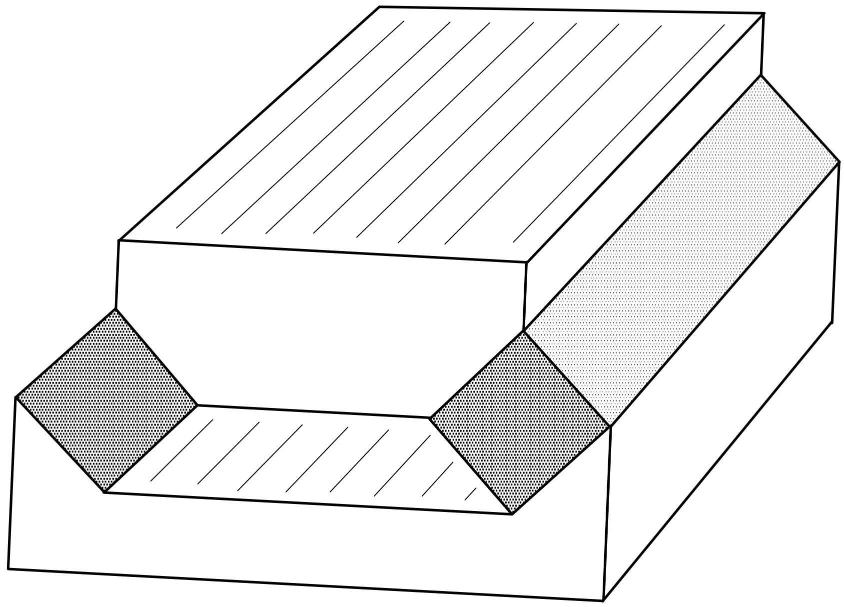


Figure A.2 — Resilient support with ten non-zero transfer stiffnesses in accordance with equation (A.5)

Figure A.2 shows an example of a resilient support which has ten non-zero transfer stiffnesses; see annex B of ISO 10486-1:1997. These are as follows:

$$\left[k_{2,1}\right]=\begin{bmatrix} k_{F_{2x},u_{1x}} & 0 & 0 & 0 & k_{F_{2x},\gamma_{1y}} & 0 \\ 0 & k_{F_{2y},u_{1y}} & 0 & k_{F_{2y},\gamma_{1x}} & 0 & 0 \\ 0 & 0 & k_{F_{2z},u_{1z}} & 0 & 0 & 0 \\ 0 & k_{M_{2x},u_{1y}} & 0 & k_{M_{2x},\gamma_{1x}} & 0 & 0 \\ k_{M_{2y},u_{1x}} & 0 & 0 & 0 & k_{M_{2y},\gamma_{1y}} & 0 \\ 0 & 0 & 0 & 0 & 0 & k_{M_{2z},\gamma_{1z}} \end{bmatrix} \tag{A.5}$$

The ten transfer stiffnesses of equation (A.5) can be divided into three categories, as follows.

a) Diagonal elements for translatory components

The measurements of the three diagonal matrix elements $k_{F_{2x},u_{1x}}$, $k_{F_{2y},u_{1y}}$ and $k_{F_{2z},u_{1z}}$ using the indirect method is the topic of the main body of this part of ISO 10846 and needs no further discussion in this annex.

b) Off-diagonal elements including one translatory and one rotatory component

The four off-diagonal matrix elements $k_{F_{2x},\gamma_{1y}}$, $k_{F_{2y},\gamma_{1x}}$, $k_{M_{2x},u_{1y}}$ and $k_{M_{2y},u_{1x}}$ can be measured using the indirect method and in the same test arrangement described in clause 5. The measurement principle is discussed in A.3.

c) Diagonal elements for rotatory components

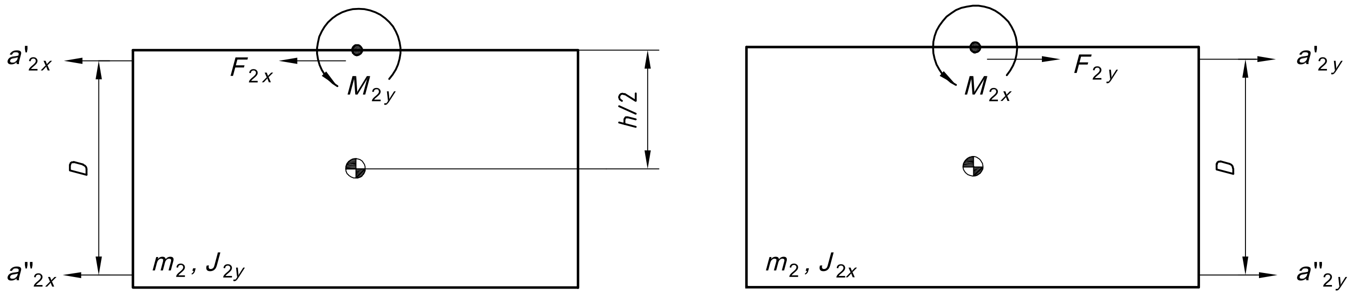
The measurements of the three diagonal matrix elements $k_{M_{2x},\gamma_{1x}}$, $k_{M_{2y},\gamma_{1y}}$ and $k_{M_{2z},\gamma_{1z}}$ can be performed using the indirect method. However, for these measurements the test arrangement described in clause 5 has to be modified. The measurement principle is discussed in A.4.

A.3 Transfer stiffnesses for one translatory and one rotatory component

The off-diagonal terms in the transfer stiffness matrix of equation (A.5) can be measured with the indirect method in the same test arrangement as described in 5.2 for transverse translatory motions. The symmetry of the blocking mass should be the same as discussed in clause 6.

For the measurements of $k_{M_{2y},u_{1x}}$ and $k_{M_{2x},u_{1y}}$ the resilient support under test is installed and excited in the same way as for the measurement of $k_{F_{2x},u_{1x}}$ and $k_{F_{2y},u_{1y}}$. However, now in addition to the blocking forces, also the blocking torques $M_{2y,\mathrm{b}}$ and $M_{2x,\mathrm{b}}$ have to be measured.

The way in which this measurement should be performed is illustrated in Figure A.3.



$\underline{u}_{2x} = -(2\pi f)^{-2}(\underline{a}'_{2x} + \underline{a}''_{2x})/2$, $\underline{\gamma}_{1x} = -(2\pi f)^{-2}(\underline{a}''_{2y} - \underline{a}'_{2y})/D$, etc.

Figure A.3 — Simultaneous measurement of translational and rotational responses of loading mass

In Figure A.3a) the loading mass is simultaneously excited by the blocking force $F_{2x,\mathrm{b}}$ and the blocking torque $M_{2y,\mathrm{b}}$. This excitation is caused by a_{1x} on the input side of the element under test.

Because the blocking mass m_2 is assumed to vibrate as a rigid body, the blocking torque can be derived as follows (see Figure A.3):

$$\underline{M}_{2y,\mathrm{b}} = -(2\pi f)^2 (\underline{J}_{2y}\underline{\gamma}_{2y} - hm_2\underline{u}_{2x}/2) \tag{A.6}$$

where

J_{2y} denotes the principal moment of inertia of the blocking mass about the y-axis;

$-(2\pi f)^2\, u_{2x}$ denotes the translatory acceleration of its mass centre.

As shown in Figure A.3, the translatory and rotatory acceleration can be measured by adding or subtracting the signals of two symmetrically placed accelerometers.

For accurate results, the measurements and evaluation can best be performed in terms of transmissibilities, as in equation (1) for translatory components. For example:

$$k_{M_{2y},u_{1x}} = \frac{\underline{M}_{2y,\mathrm{b}}}{\underline{u}_{1x}} = -(2\pi f)^2 \left(J_{2y} T_{\gamma_{2y},u_{1x}} - hm_2 T_{u_{2x},u_{1x}}/2 \right) \tag{A.7}$$

where

$$T_{\gamma_{2y},u_{1x}} = \frac{1}{D}\left\{ \frac{\underline{a}'_{2x} - \underline{a}''_{2x}}{\underline{a}_{1x}} \right\} \tag{A.8}$$

$$= \frac{1}{D}\left\{ \frac{\underline{a}'_{2x}}{\underline{a}_{1x}} - \frac{\underline{a}''_{2x}}{\underline{a}_{1x}} \right\}, \text{ etc.} \tag{A.9}$$

NOTE Because of the fact that transmissibilities are equal for displacements and for accelerations, equation (A.7) can be determined by using solely acceleration measurements as in equations (A.8) and (A.9).

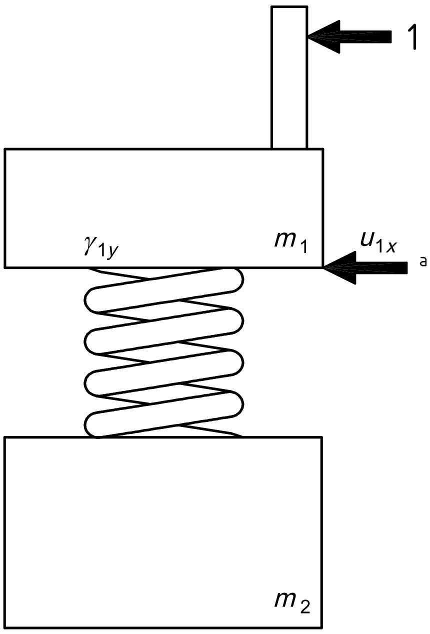
The formulation of equation (A.8) is applicable for a single two-channel measurement, using analog subtraction of a'_{2x} and a''_{2x}. The formulation of equation (A.9) is applicable for two separate measurements of frequency response functions $\underline{a}'_{2x}/\underline{a}_{1x}$ and $\underline{a}''_{2x}/\underline{a}_{1x}$.

For the measurement of the other two off-diagonal terms (i.e. $k_{F_{2x},\gamma_{1y}}$ and $k_{F_{2y},\gamma_{1x}}$) the same test rig may be used as for $k_{F_{2x},u_{1x}}$ and $k_{F_{2y},u_{1y}}$. However, now the resilient support under test has to be turned upside down, i.e. the input and output side are interchanged. On the basis of equation (A.4), the measurement of $k_{F_{2x},\gamma_{1y}}$ is replaced by that of $k_{M_{1y},u_{2x}}$ and the measurement of $k_{F_{2y},\gamma_{1x}}$ is replaced by that of $k_{M_{1x},u_{2y}}$. These are measurements similar to those for the other two off-diagonal terms only, the input and output sides have been interchanged. Of course, results for diagonal terms are unaffected by putting the resilient isolator upside down; see equation (A.4).

A.4 Transfer stiffnesses for two rotatory components

Unless modifications are introduced, the diagonal terms $k_{M_{2x},\gamma_{1x}}$, $k_{M_{2x},\gamma_{1y}}$ and $k_{M_{2z},\gamma_{1z}}$ in equation (A.5) cannot be measured in the test arrangements described in clause 5. These modifications concern the excitation on the input side of the resilient element under test, because on the output side the measurements are similar to those described in A.3.

The simplest modification is needed for the measurement of $k_{M_{2z},\gamma_{1z}}$, if the *z*-direction is considered as the direction of the static load, as in Figures 2 to 4 of this part of ISO 10846. For a test element as in Figure A.2, the excitation with γ_{1z} can be realised in a test arrangement similar to that in Figure 4. Instead of a single and symmetrically placed vibration exciter in the transverse direction, either an asymmetrically placed exciter or, more generally, a pair of exciters, may be used to generate γ_{1z}. When a single asymmetrically placed exciter is used, u_{1y} or u_{1x} will also be excited. However, for a test element shown in Figure A.2 this will not lead to unwanted contributions to γ_{2z}, because the corresponding transfer stiffness $k_{M_{2z},u_{1x}}$ and $k_{M_{2x},u_{1y}}$ are equal to zero [see equation (A.5)].



Key

1 Vibration exciter

a $2u_{1x}$ minimized

Figure A.4 — Excitation with a rotatory vibration while the transverse translatory vibration is simultaneously minimized on the input side of the resilient element under test

For the measurements of $k_{M_{2x},\gamma_{1x}}$ and $k_{M_{2y},\gamma_{1y}}$, Figure A.4 shows one of the possibilities for excitation on the input side.

On the input side, the resilient support under test is loaded with a rigidly vibrating block with a symmetry according to the same principle as discussed in clause 6 for m_2. With the aid of a lever and a vibration exciter in the transverse direction at the appropriate height, the ratio γ_{1x}/u_{1y} or γ_{1y}/u_{1x} is maximized. To apply the static load, auxiliary isolators are placed between the top mass and the load frame in the same way as in Figure 2. For measurements of γ_{1x} or γ_{1y} pairs of normal accelerometers are used in a way similar to that shown in Figure A.3.

If it is not feasible in practice to suppress the transverse translatory motions u_{1x} or u_{1y}, several measurements should be carried out for different heights of the vibration exciter. Each of the experiments provide another combination of γ_{1x} and u_{1y} or γ_{1y} and u_{1x}. Pairs of transfer stiffnesses can be found according to the following example.

EXAMPLE 1

Assume that for two experiments the following equations are valid

$$\text{exp. 1:} \quad \underline{M}_{2x,\text{b}}(1) = k_{M_{2x},u_{1y}} \underline{u}_{1y}(1) + k_{M_{2x},\gamma_{1x}} \underline{\gamma}_{1x}(1) \tag{A.10}$$

$$\text{exp. 2:} \quad \underline{M}_{2x,\text{b}}(2) = k_{M_{2x},u_{1y}} \underline{u}_{1y}(2) + k_{M_{2x},\gamma_{1x}} \underline{\gamma}_{1x}(2) \tag{A.11}$$

To solve these equations on the basis of vibration measurements only, they are rewritten as follows:

$$\text{exp. 1:} \quad \underline{M}_{2x,\text{b}} / \underline{u}_{1y}(1) = k_{M_{2x},u_{1y}} + k_{M_{2x},\gamma_{1x}} R_{\gamma_{1x},u_{1y}}(1) \tag{A.12}$$

$$\text{exp. 2:} \quad \underline{M}_{2x,\text{b}} / \underline{u}_{1y}(2) = k_{M_{2x},u_{1y}} + k_{M_{2x},\gamma_{1x}} R_{\gamma_{1x},u_{1y}}(2) \tag{A.13}$$

The right-hand sides have been written in terms of transmissibilities in the same way as in equations (A.7) to (A.8). The right-hand sides contain vibration ratios $R(i) = \underline{\gamma}_{1x}/\underline{u}_{1y}$. A two-channel FFT-analyser in combination with analog summation or subtraction devices would suffice for measuring them [see also note after equation (A.9)].

Now the vector containing the two transfer stiffness elements is found by using the inverse of the transmissibility matrix, according to equation (A.14):

$$\begin{Bmatrix} k_{M_{2x},u_{1y}} \\ k_{M_{2x},\gamma_{1x}} \end{Bmatrix} = \begin{bmatrix} 1 & R_{\gamma_{1x},u_{1y}}(1) \\ 1 & R_{\gamma_{1x},u_{1y}}(2) \end{bmatrix}^{-1} \begin{Bmatrix} \underline{M}_{2x} / \underline{u}_{1y}(1) \\ \underline{M}_{2x} / \underline{u}_{1y}(2) \end{Bmatrix} \tag{A.14}$$

To prevent a strong amplification of measurement errors by the inversion of an ill-conditioned matrix, the experiments have to be designed in such a way that the condition number

$$\text{cond}\begin{bmatrix} 1 & R(1) \\ 1 & R(2) \end{bmatrix} = \max\left(1+\left|R(1)\right|, 1+\left|R(2)\right|\right) \max\left(\left|\frac{R(2)}{R(1)-R(2)}\right| + \left|\frac{R(1)}{R(1)-R(2)}\right|, \frac{2}{\left|R(1)-R(2)\right|}\right) \tag{A.15}$$

is equal to or smaller than 3, where shorthand notation $R(1)$ and $R(2)$ has been used for the ratios γ_{1x}/u_{1y}. This requirement can be met when both

$$-2 \ \text{m}^{-1} < R(1) < -0{,}5 \ \text{m}^{-1} \tag{A.16}$$

and

$$0{,}5 \ \text{m}^{-1} < R(2) < 2 \ \text{m}^{-1} \tag{A.17}$$

The minus signs in equation (A.16) denote the opposite phase of γ_{1x} and u_{1y}.

EXAMPLE 2

To facilitate the design of appropriate tests, an example of how experiments with $R(1) \approx -1{,}5\ \text{m}^{-1}$ and $R(2) \approx 1{,}5\ \text{m}^{-1}$ can be realized is given. Figure A.5 shows two excitations of a “block” on the input side of the test element. This block is a solid cylinder with diameter d and height h. It is assumed that its mass and its rotatory inertia are large compared to those of the input flange of the test element and that, as a first approximation, the block vibrates as a free body in the frequency range of interest. Then the horizontal excitation shown in Figure A.5a), i.e. with F_y at an appropriate “positive” z-coordinate relative to the position of the mass centre as origin, can provide negative values of the transmissibilty.

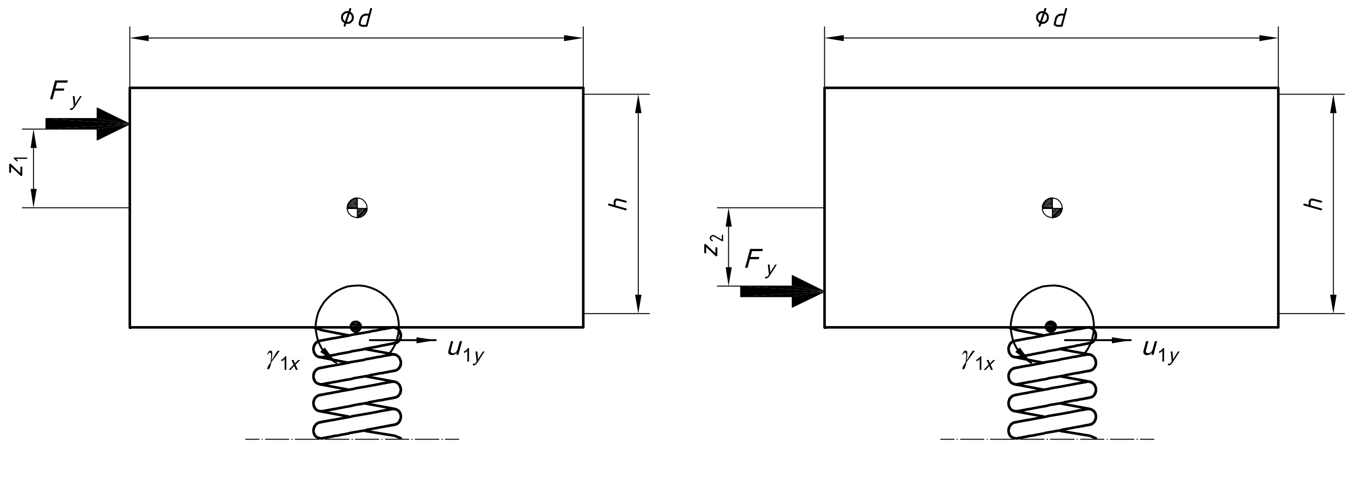


Figure A.5 — Examples of two different excitations on the load distribution “block” on the input side of a test element

The excitation shown in Figure A.5 b), i.e. with F_y at an appropriate “negative” z-coordinate, can provide positive values of the transmissibility.

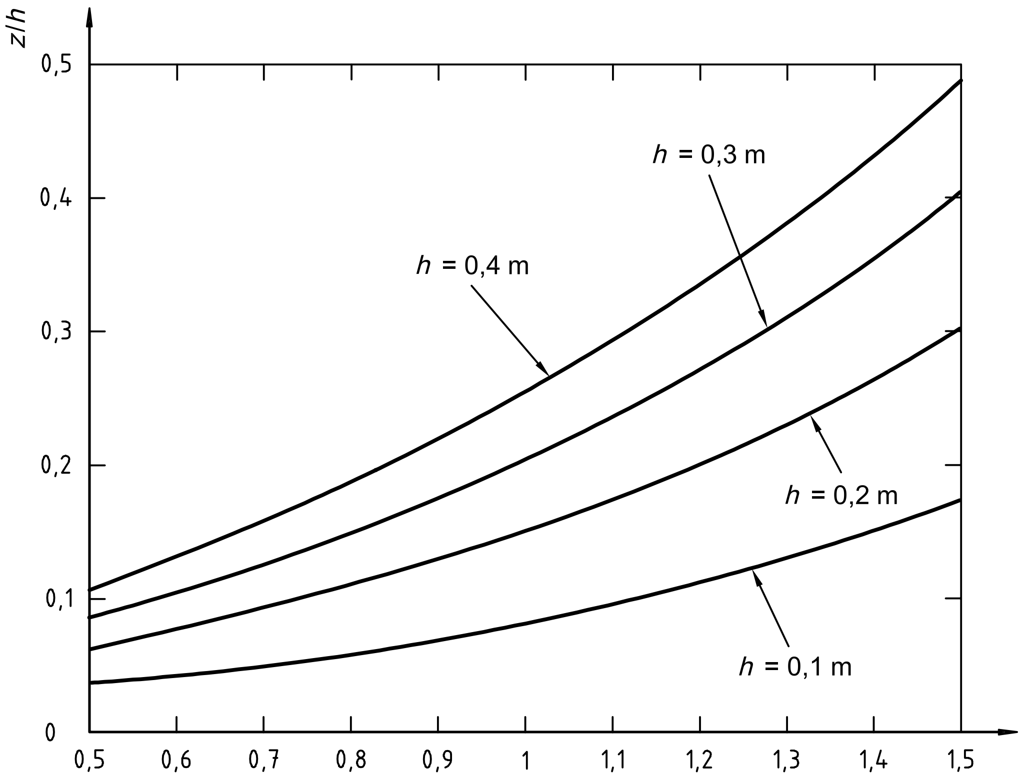
For this type of block

$$R_{\gamma_{1x},u_{1y}} = \frac{-6\alpha}{(2b^2+1)h - 6\alpha h} \tag{A.18}$$

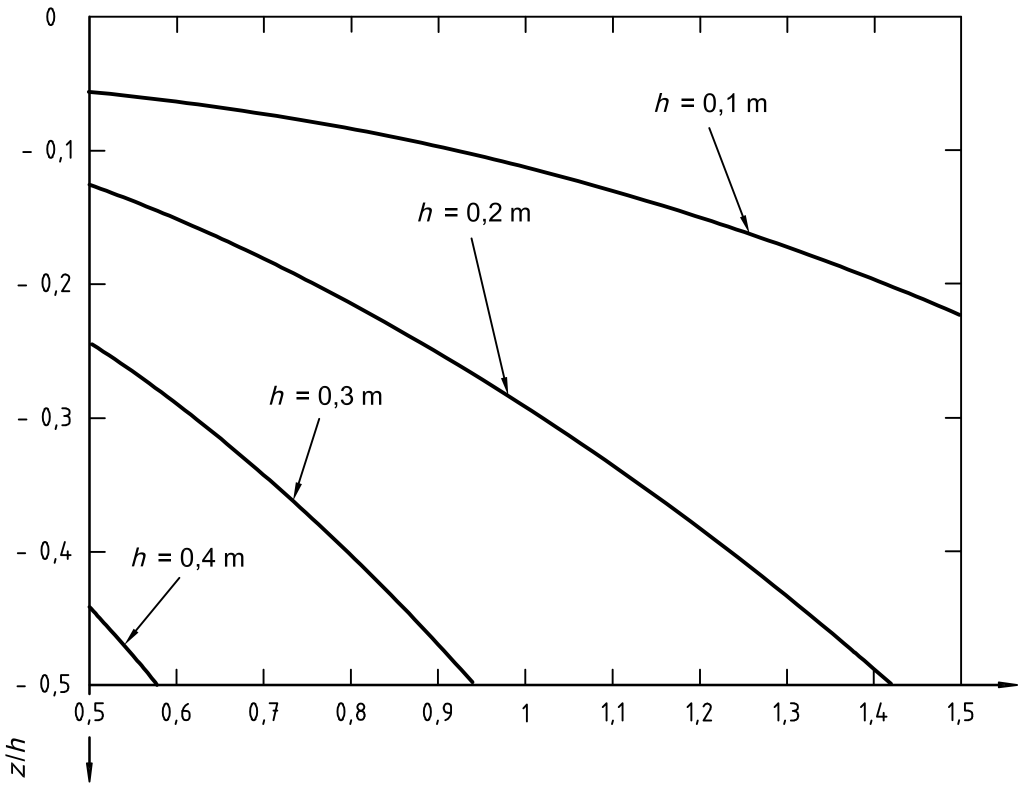
$$\alpha = \frac{(3b^2+1)hR}{6(hR-1)} \tag{A.19}$$

where $\alpha = z/h$ and $b = d/2h$

Figures A.6a) and A.6b) provide nomograms for the selection of cylindrical blocks and excitation positions in the range $-h < z < h$ (i.e. on the block) and for $R(1) = -1{,}5\ \text{m}^{-1}$ and $R(2) = +1{,}5\ \text{m}^{-1}$. Equations (A.18) and (A.19) can be used to find solutions for z-coordinates outside the range of the block, for example, by using a lever and with $z < -h/2$. This is relevant when blocks with a large size are used.



a) $R(1) = -1{,}5\ \mathrm{m}^{-1}$



b) $R(2) = 1{,}5\ \mathrm{m}^{-1}$

Figure A.6 — Nomograms according to equations (A.16) and (A.17) for helping the design of experiments with cylindrical excitation blocks

Annex B
(informative)

Effect of symmetry on the transfer stiffness matrix

ISO 10846 is concerned with the measurement of individual elements of the 6×6 transfer stiffness of resilient elements; see 5.3 and annex B of ISO 10846-1:1997 and annex A, equation (A.1), of this part of ISO 10846.

These individual elements are ratios of a single blocking force at the output side of the elements and a single displacement at the input. Generally speaking, for measurements of these elements according to ISO 10846, only a single input displacement is allowed to be non-zero. However, for the transfer stiffness of resilient supports in the normal load direction, this requirement is usually not necessary due to the symmetry properties of the element. This symmetry eliminates contributions to the blocking force in the normal direction from input displacements other than that in the normal direction.

In Figure A.2 an example of such a resilient element is shown. Equation (A.5) shows the 6×6 matrix of transfer stiffnesses of this resilient element. The third row of this matrix shows that solely the stiffness in the normal load direction can contribute to the blocking force in that direction.

Annex B of ISO 10846-1:1997 discusses four examples of common symmetrical shapes, which show this property.

Annex C
(informative)

Static load-deflection curve

If considered useful, a static load-deflection curve in the range of 0 to 100 % of the maximum permissible load can be added to the report, with a description of, or a reference to, the measurement procedure. See, for example, bibliographic references [3] and [4].

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