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

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

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

Partie 4: Raideur dynamique en translation des éléments autres que les supports élastiques

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.

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

ISO 10846-4 was prepared by Technical Committee ISO/TC 43, *Acoustics*, Subcommittee SC 1, *Noise*, in collaboration with 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 the determination of the low frequency dynamic stiffness of elastic supports for translatory motion*

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 a direct and an indirect method for measuring the dynamic transfer stiffness function of linear resilient elements (other than resilient supports) such as resilient bellows, hoses, shaft couplings, power supply cables and pipe hangers. This part of ISO 10846 belongs to a series of International Standards on methods for the laboratory measurement of the vibro-acoustic properties of resilient elements, which also includes documents on measurement principles and on a direct, an indirect and a driving point method for resilient supports. 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 method described in this part of ISO 10846 are useful for resilient elements that are used to reduce the transmission of structure-borne sound (primarily frequencies above 20 Hz). The method does not characterize completely elements that are used to attenuate low-frequency vibration or shock excursions.

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

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

1 Scope

This part of ISO 10846 specifies two methods for determining the dynamic transfer stiffness for translations of resilient elements other than resilient supports. Examples are resilient bellows, shaft couplings, power supply cables, hoses and pipe hangers (see Figure 1). Elements filled with liquids, such as oil or water, are excluded.

NOTE 1 Pipe hangers are extensionally deflected, as opposed to elastic supports which are compressed. Therefore, the test conditions are different from those described in ISO 10846-2 and ISO 10846-3.

The methods are applicable to resilient elements with flat flanges or flat clamp interfaces. It is not necessary that the flanges be parallel.

Resilient elements which are the subject of this part of ISO 10846 are those that are used to reduce

a) the transmission of audiofrequency vibrations (structure-borne sound, 20 Hz to 20 kHz) to a structure which may, for example, radiate unwanted 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.

In practice, the size of the available test rig(s) determines restrictions for very small and for very large resilient elements.

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

The direct method can be applied in the frequency range from 1 Hz up to a frequency that is usually determined by the lowest resonance frequency of the test arrangement frame (typically 300 Hz for test rigs with dimensions of the order of 1 m).

NOTE 2 In practice, the lower frequency limit depends on the dynamic excitation system.

The indirect method covers a frequency range that is determined by the test set-up and the isolator under test. The range is typically from a lower frequency between 20 Hz and 50 Hz, to an upper frequency between 2 kHz and 5 kHz.

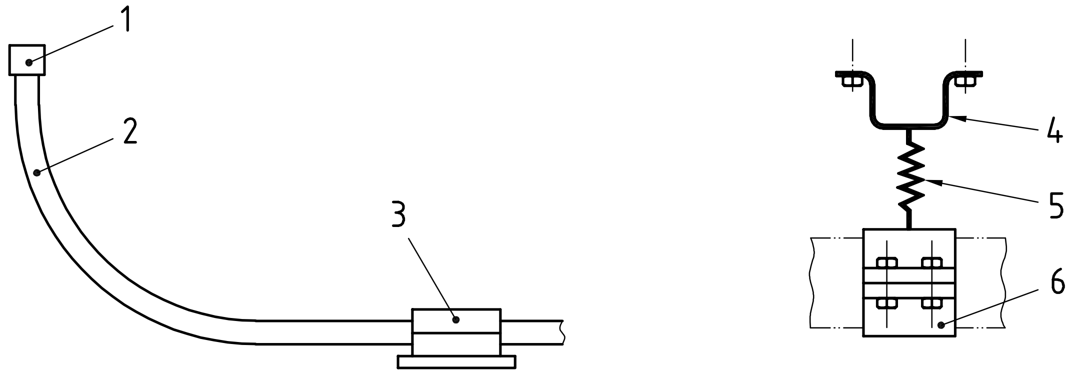
The data obtained according to the methods specified in this part of ISO 10846 can be used for

— product information provided by manufacturers and suppliers,

— information during product development,

— quality control, and

— calculation of the transfer of vibration through resilient elements.



a) Power cable including connector and clamping device **b) Pipe hanger**

Key

1 connector
2 cable
3 clamp
4 fixture
5 flexible element
6 pipe clamp

Figure 1 — Examples of resilient elements with flat flanges or clamps

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 266, *Acoustics — Preferred frequencies*

ISO 2041, *Vibration and shock — Vocabulary*

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*

ISO 10846-1, *Acoustics and vibration — Laboratory measurement of vibro-acoustic transfer properties of resilient elements — Part 1: Principles and guidelines*

ISO 16063-21, *Methods for the calibration of vibration and shock transducers — Part 21: Vibration calibration by comparison with a reference transducer*

GUM:1993, *Guide to the expression of uncertainty in measurement.* BIPM/IEC/IFCC/ISO/IUPAC/IUPAP/OIML

3 Terms and definitions

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

3.1
resilient element
element of which one of the functions is the reduction of the vibration transmission in a certain frequency range

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

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

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

3.5
dynamic transfer stiffness
$k_{2,1}$
frequency-dependent ratio of the complex blocking force $\underline{F}_{2,b}$ on the output side of a resilient element to the complex displacement $\underline{u}_1$ on the input side during simple harmonic motion, defined by the following formula

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

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

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 the frequency of the average value of the modulus of the dynamic transfer stiffness over a frequency band Δf

NOTE See 8.3.

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·$x_1(t)$ + b·$x_2(t)$ produces the output a·$y_1(t)$ + b·$y_2(t)$. 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 7.7).

3.12
direct method
method in which 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

3.14
transmissibility
T
ratio of the complex displacements on the output side $\underline{u}_2$ to those on the input side $\underline{u}_1$ of the test element during simple harmonic motion, defined by the following formula

$$T = \underline{u}_2 / \underline{u}_1$$

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 $F_0 = 10^{-6}$ 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 $a_0 = 10^{-6}$ 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 denotes the reference stiffness (= 1 N·m^{-1})

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 k_0 denotes the reference stiffness (= 1 N·m^{-1})

3.19
flanking transmission
transmission of vibrations to the output side via paths other than through the resilient element under test

4 Principles

The measurement principles of the direct and the indirect method are discussed in ISO 10846-1.

In the *direct* method, the basic principle is that the blocking output force is measured between the output side of the resilient element and a foundation. The foundation shall provide a sufficient reduction of the vibrations on the output side of the test object compared to those on the input side.

In the *indirect* method 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 relationship between the dynamic transfer stiffness of the element under test and the measured vibration transmissibility (3.14) is given by the following approximation

$$k_{2,1} \approx -(2\pi f)^2 (m_2 + m_f)T \qquad \text{for } |T| \ll 1 \qquad (1)$$

where m_f denotes the mass of the output flange of the test element. The 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 centre of mass of the compact body composed of the blocking mass and the output flange of the test element, and in the direction of the wanted force.

5 Test arrangements

5.1 General

In Figures 2 to 8, examples are given of test arrangements for resilient elements other than resilient supports. The sketches are schematic. Examples are given of test arrangements for single elements as well as for symmetrically paired ones.

NOTE The collection of examples is by no means exhaustive and is not intended to form a limitation for test arrangement principles. It is meant as an illustration of solutions that have been applied to meet requirements for the adequacy of the test arrangements (see Clause 6).

To be suitable for measurements according to this part of ISO 10846, a test arrangement shall include the components given in 5.3, when applicable. Other aspects concerning test rig properties are discussed in 5.4 and 5.5.

5.2 Local coordinate systems

For the resilient elements which are tested according to this part of ISO 10846, the directions normal to flanges or fixtures on the input and on the output side may be not the same (see Figures 7 and 8). For non-planar test elements, they are even out-of-plane. Therefore, for each test configuration, the local Cartesian coordinate systems and the local corresponding forces, torques, displacements and rotatory displacements shall be defined in agreement with Figure 9. The positive directions of the z-axes shall coincide with the directions normal to the input and output flanges and shall point away from the test element. In the case of a "planar" test element, the x-axis shall be chosen out-of-plane on the input as well as on the output side. In the case of a non-planar test element, the transverse axis directions shall be defined according to the requirements of the applications. The naming of the x- and y-directions is the responsibility of the user of this part of ISO 10846. Thus, the definition of dynamic transfer stiffnesses of cables and hoses is dependent on the test element and on the test arrangement.

The naming of the dynamic transfer stiffnesses shall be in agreement with the following notation:

$k_{2x,1x}$; $k_{2x,1y}$; $k_{2x,1z}$

$k_{2y,1x}$; $k_{2y,1y}$; $k_{2y,1z}$

$k_{2z,1x}$; $k_{2z,1y}$; $k_{2z,1z}$

where the subscripts $2x$, $2y$, $2z$ refer to the local coordinate system for the blocking output forces, and the subscripts $1x$, $1y$, $1z$ to the local coordinate system for the input displacements.

In cases where confusion is unlikely, simpler notations may be used. For example, for an axially symmetrical test component as in Figure 2, it can suffice to define two transfer stiffnesses as follows: $k_{2,1}$(axial); $k_{2,1}$(radial).

5.3 Test rig components

5.3.1 Resilient elements under test

The test element shall be mounted in a way that is representative of its use in practice. This shall include the static preload and the fixture arrangements on the input and output sides. Auxiliary fixtures shall be considered as parts of the test element (see 3.3).

NOTE Resilient elements with a strongly non-linear static load deflection curve show strongly preload dependent dynamic behaviour as well. However, in contrast to the resilient supports covered in ISO 10846-3, the static preloads in this part of ISO 10846 are not primarily due to gravity. For example, the static preload for a resilient shaft coupling may be a torque load [Figure 3 b)].

5.3.2 Force measurement system on the output side

When the *direct* method is used, the force measurement system on the output side of the resilient element shall consist of one or more force transducers.

It may be necessary to apply a force distribution plate between the test element and the force transducers (see Figure 8).

NOTE Besides its function of load distribution, the force distribution plate also provides a high contact stiffness to the force transducers. Moreover, it provides a uniform vibration of the output flange or clamp.

5.3.3 Blocking mass on the output side

When the *indirect* method is used, one function of the blocking mass is the estimation of the blocking output force by measuring the acceleration of the mass. A second function is to provide a spatially uniform vibration of the output flange of the test object over the frequency range of interest.

5.3.4 Acceleration measurement systems

Accelerometers shall be mounted on the input and output side of the test object and on the foundation of the test arrangement. 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.

When the *indirect* method is used, the transverse accelerometers of the blocking mass that are needed are those along the x- and y-axes through the centre of mass of the compact body composed of the blocking mass and the output flange of the test element (see Figure 10).

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

5.3.5 Dynamic excitation system

The dynamic excitation system shall be appropriate for the frequency range of interest. Any suitable type of exciter is permitted. Examples are

a) a hydraulic exciter,

b) one or more electrodynamic vibration 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.

5.3.6 Excitation mass on the input side

The excitation mass on the input side of the test object has one or more of the following functions:

a) to provide a uniform vibration of the input flange under dynamic forces;

b) to enhance unidirectional vibration of the input flange.

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

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 6.5). The predominance of unidirectional vibration of the input flange, will be influenced by

a) the symmetry of the vibration excitations and boundary conditions of the excitation mass [see Figure 2 b)], and

b) the inertial properties of the excitation mass [see Figure 3 a)].

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.

5.4 Suppression of unwanted vibrations

5.4.1 General

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, boundary conditions and test element properties, components other than the intended input vibration component may show unwanted strong responses at certain frequencies. Qualitative measures to suppress unwanted input vibrations are discussed in 5.4.2 and 5.4.3. A special category of test arrangements is that in which two nominally equal resilient elements are tested in a symmetrical configuration. This may help to suppress unwanted input vibrations. Quantitative requirements are given in 6.5.

5.4.2 Normal direction

For excitation in the normal direction, symmetrical positioning of the exciter or a pair of exciters and use of an axially symmetric excitation mass is the preferred method for suppressing transverse and rotational vibrations on the input side.

Nevertheless, the properties of the test object itself can cause coupling between the normal and other vibration directions. Such unwanted responses may be strongly suppressed by using an excitation mass which has, at the interface with the test element, large driving point impedances for transverse and rotational directions compared to the corresponding driving point impedances of the test element [see, for example, Figure 8 a)].

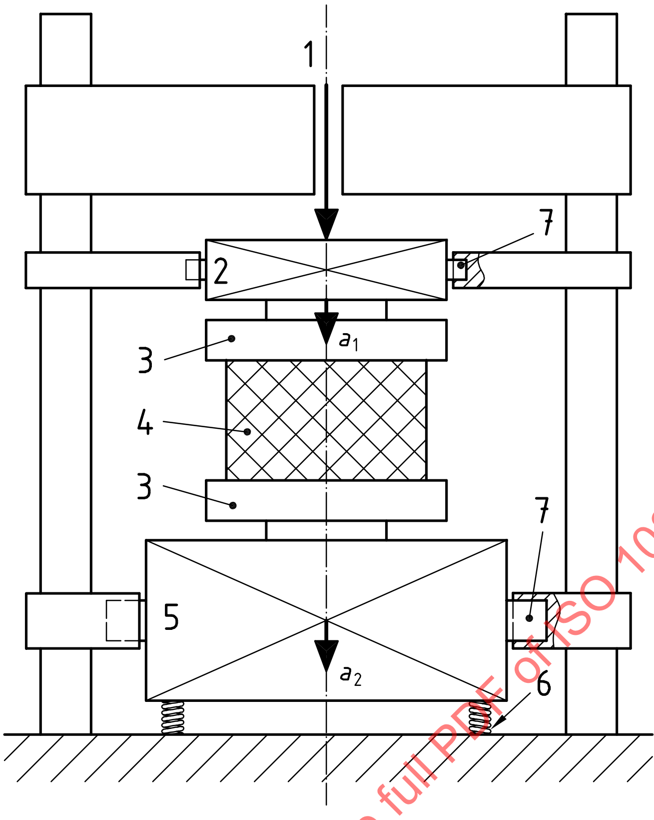
Another method of suppressing unwanted input vibrations is the use of a symmetrical arrangement with two nominally identical test objects, or of a “guiding” system on the sides of the excitation mass, for example, roller bearings. These systems are not shown in a figure, but they are rather similar to the examples for transverse excitation shown in Figures 2 b), 3 a) and 5.

5.4.3 Transverse direction

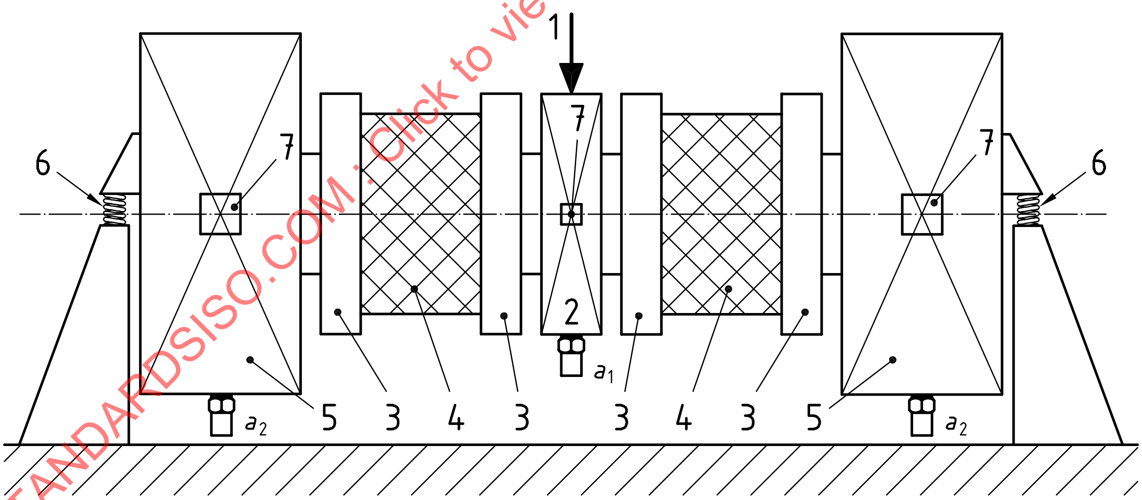
For excitation in the transverse direction, coupling between transverse and rotational input vibrations will always occur.

Some examples are discussed of measures which may enhance unidirectional vibrations on the input side. Figures 2 b), 6 and 8 d) show a symmetrical arrangement with two nominally equal test objects. Figures 3 a) and 4 show, as examples, how a guiding system can be used to suppress input rotations. Figure 7 b) shows an example without a guiding system. In this latter case, a symmetrical excitation block 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.

An alternative to the application of conventional methods might be the use of active vibration control. Using multiple actuators and sensors in combination with a control system, the ratio between the wanted and unwanted vibration levels can be improved.



a) Test rig in a frame and with axial excitation

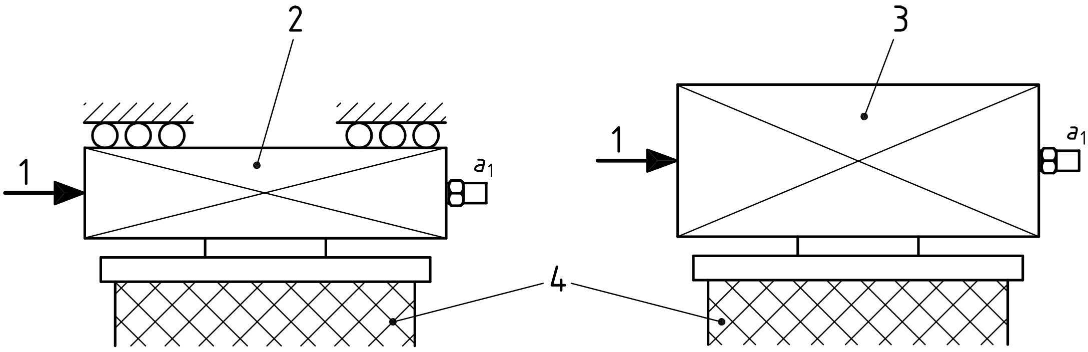


b) Symmetric configuration including two nominally equal couplings and with transverse excitation

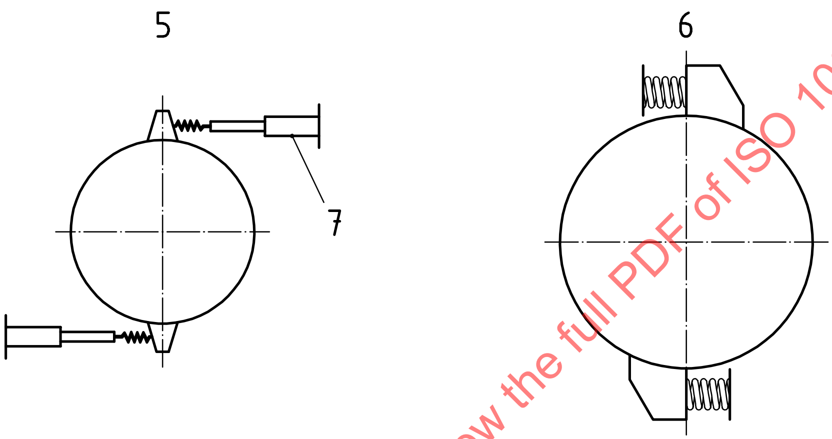
Key

1 exciter
2 excitation mass
3 coupling flanges
4 flexible part of coupling
5 blocking mass
6 dynamic decoupling springs
7 torque preload attachments [see Figure 3 b)]

Figure 2 — Examples of laboratory test rigs for measuring the dynamic transfer stiffness of a resilient shaft coupling with static torque preload and using the *indirect* method



a) Radial excitation for frame arrangement [see Figure 2 a)]

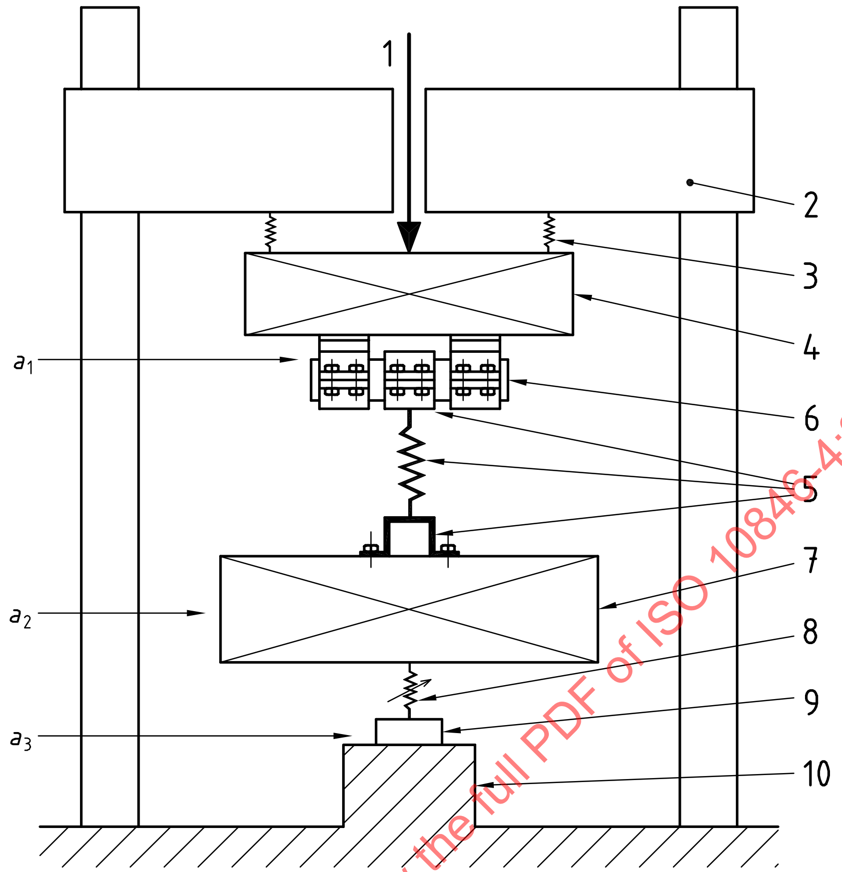


b) Torque preload

Key

1 exciter
2 thin force distribution plate with guiding system
3 large excitation mass with suppressed rotation
4 coupling
5 input side
6 output side
7 pneumatic cylinder

Figure 3 — Examples of details of a laboratory test rig as in Figure 2

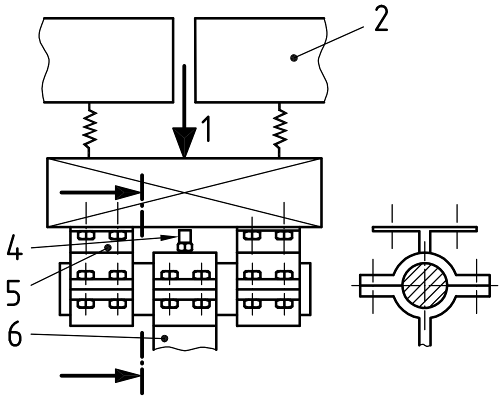


Key

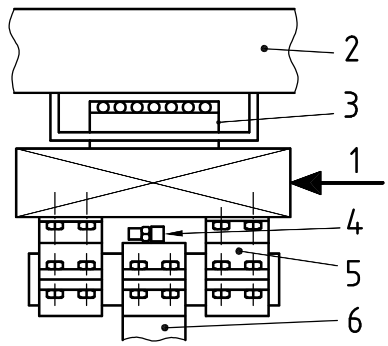
1 exciter
2 traverse
3 dynamic decoupling springs
4 excitation mass
5 test element with fixture and pipe clamp
6 solid cylinder with pipe clamps
7 blocking mass
8 controllable air spring
9 load cell
10 rigid foundation

Figure 4 — Example of laboratory test rig for measuring the dynamic transfer stiffnesses of a resilient pipe hanger with gravity load using the *indirect* method

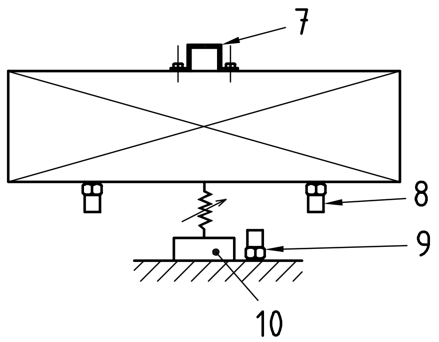
(Overview with normal excitation, the pipe hanger is mounted upside down)



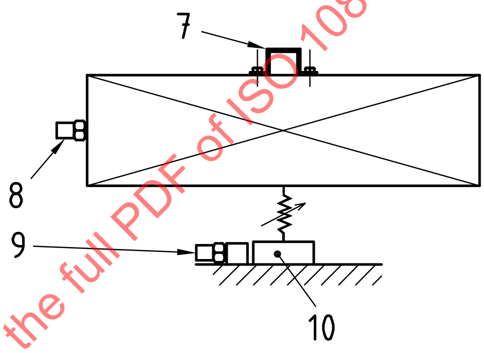
a) Input side, normal excitation



b Input side, transverse excitation



c) Output side, normal excitation

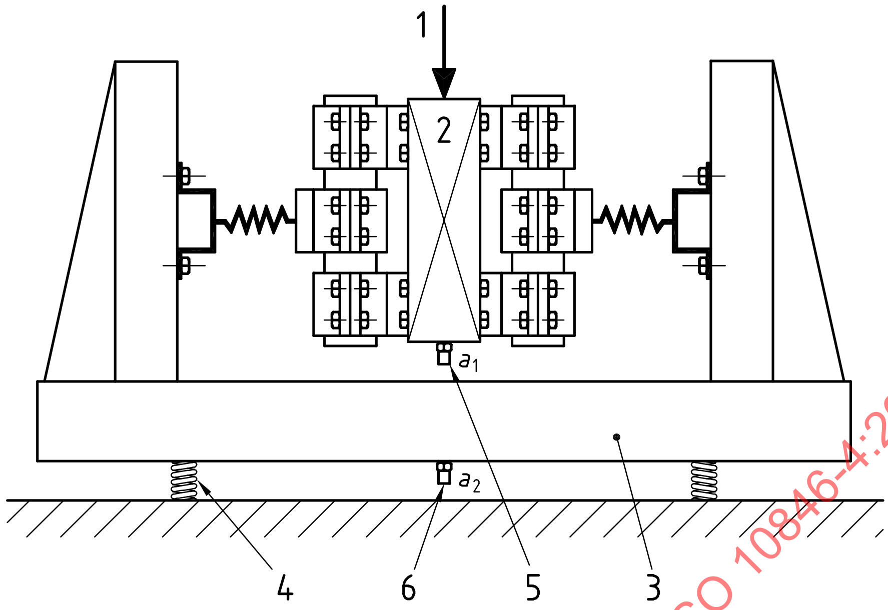


d) Output side, transverse excitation

Key

1 exciter
2 traverse
3 optional guiding system
4 acceleration measurement (a_1)
5 pipe clamps for excitation mass
6 pipe clamp test element
7 pipe hanger fixture
8 acceleration measurement (a_2)
9 acceleration measurement (a_3)
10 load cell

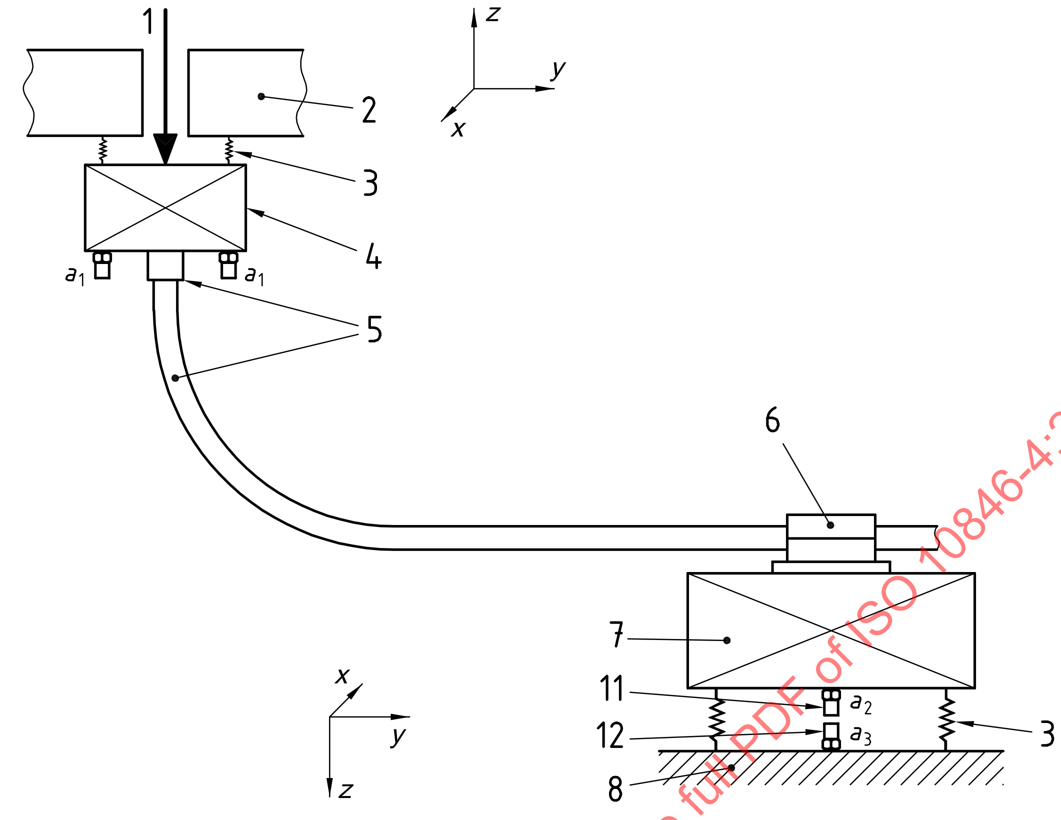
Figure 5 — Examples of details of a laboratory test rig as in Figure 4



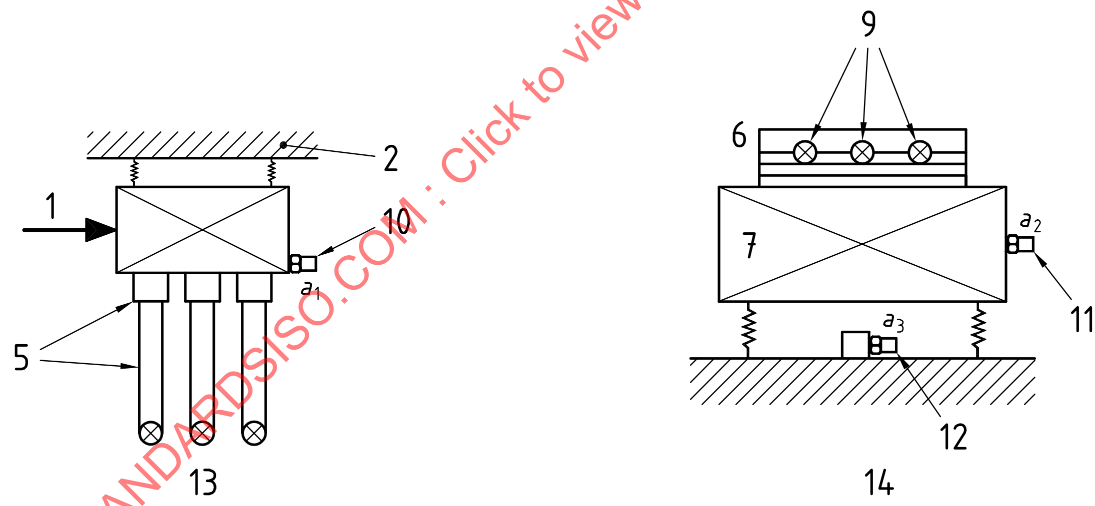
Key

1 exciter
2 excitation mass
3 blocking mass
4 dynamic decoupling springs
5 acceleration measurement (a_1)
6 acceleration measurement (a_2)

Figure 6 — Example of laboratory test rig for measuring the dynamic transfer stiffnesses of a resilient pipe hanger using the *indirect* method
(Symmetrical configuration with two nominally equal hangers)



a) Overview and axial excitation

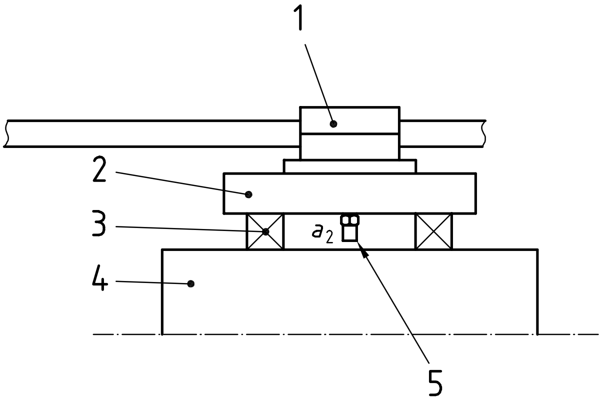


b) Transverse excitation

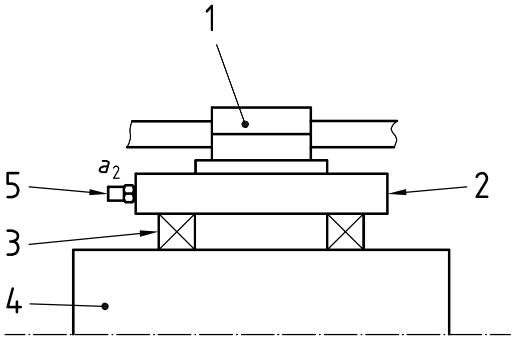
Key

1 exciter
2 frame
3 dynamic decoupling springs
4 excitation mass
5 connectors and cables
6 clamp
7 blocking mass
8 rigid foundation
9 cables
10 acceleration measurement (a_1)
11 acceleration measurement (a_2)
12 acceleration measurement (a_3)
13 input side
14 output side

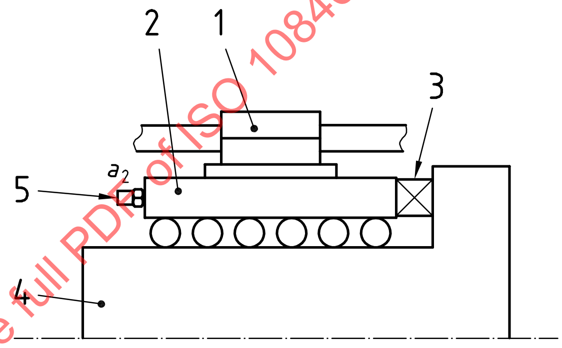
Figure 7 — Example of laboratory test rig for measuring the dynamic transfer stiffnesses of an electric cable bundle (e.g. three cables), using the *indirect* method



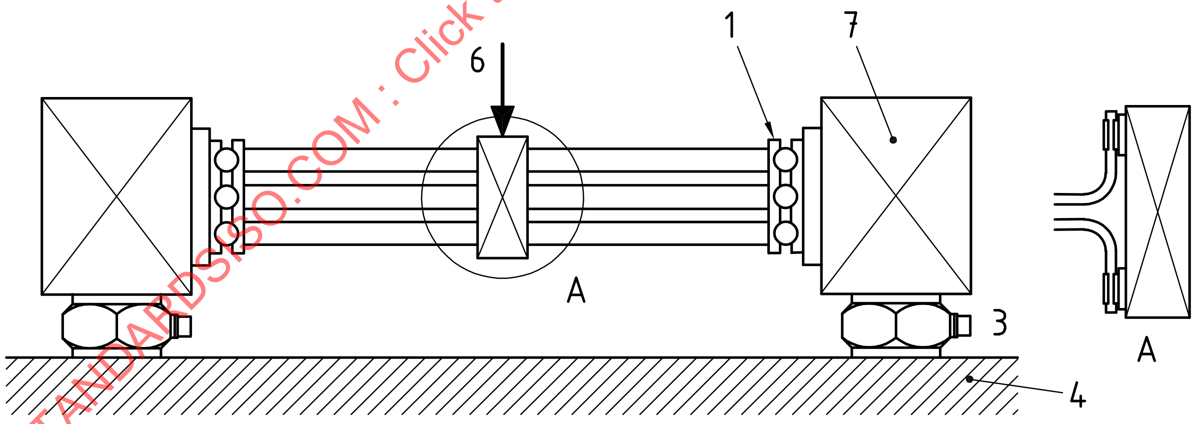
a) Output side for blocking force measurement



b) Output side for transverse measurements with shear force measurement system



c) Output side for transverse measurements with low friction guiding system



d) Symmetrical configuration of two bundles for the benefit of unidirectional transverse vibration input

Key

1 clamp
2 output force distribution plate
3 force measurement system (F_2)
4 rigid foundation
5 acceleration measurement (a_2)
6 exciter
7 blocking mass

Figure 8 — Example of laboratory test rig for measuring the dynamic transfer stiffnesses of an electric bundle (e.g. three cables), using the *direct* method

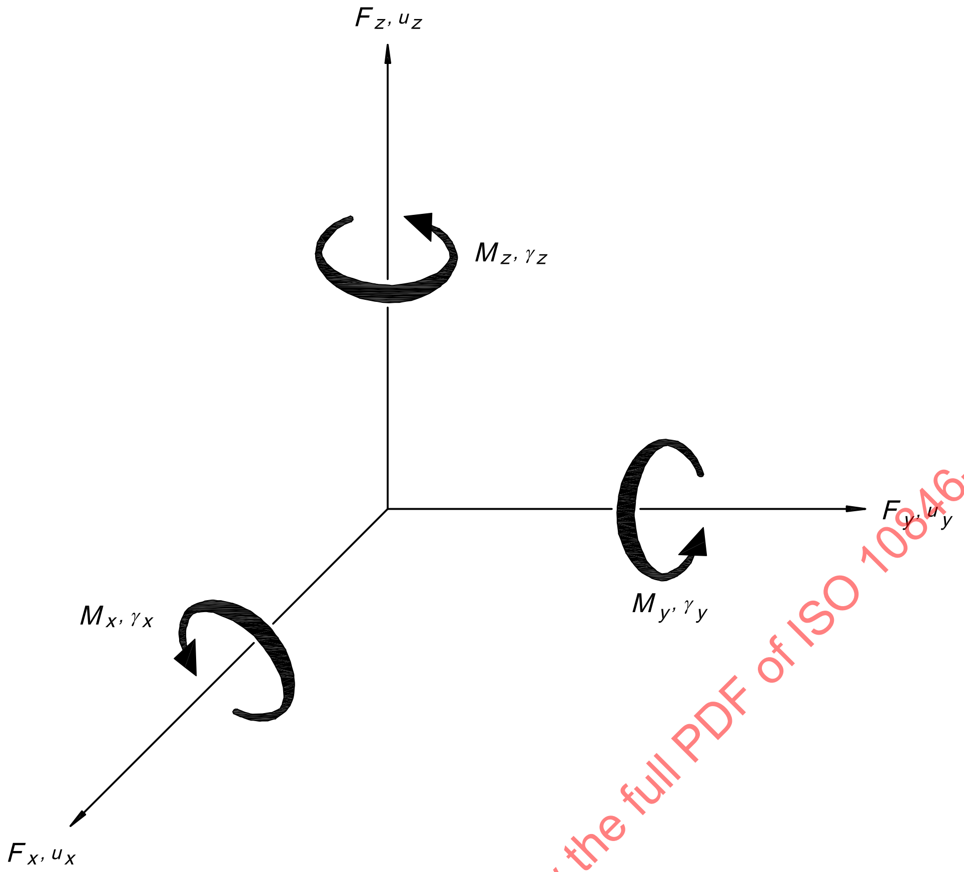
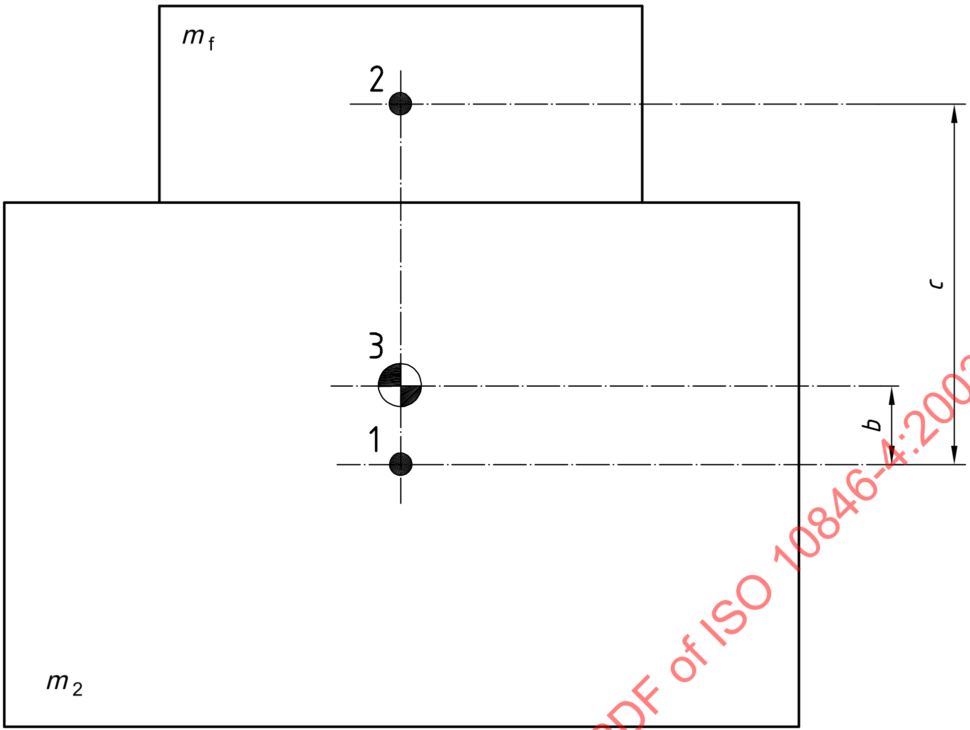


Figure 9 — Cartesian coordinate system with forces, torques, translatory and rotatory displacements



Key

1 centre of mass of the blocking mass
2 centre of mass of the flange
3 centre of mass of the compound body

NOTE 1 The distance between the centres of mass 1 and 2 equals c.

NOTE 2 The distance between the centres of mass 1 and 3 equals b.

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

Figure 10 — Example of locating the centre of mass of the compact body composed of blocking mass and output flange of the test element

5.5 Special requirements

5.5.1 Provisions for testing

According to 5.1.1, the test element shall be mounted in a way which is representative of its use in practice. This may require special provisions for the test arrangement. The most common situations are treated in 5.5.2 to 5.5.5. In situations which are not specifically described in this part of ISO 10846, similar provisions shall be taken. These shall be described in detail in the test report and it shall be made clear in such a report that the test conditions are representative of those found in practice.

5.5.2 Static torque preload

The vibro-acoustic transfer properties of flexible couplings in drive shafts (e.g. in ships) are often highly dependent on the preload. Therefore, the dynamic transfer stiffnesses shall be determined with the test object preloaded with an appropriate static torque. Figure 3 b) shows an example of how the torque can be applied. Static rotation of the excitation mass may be obtained by using, for example, two pneumatic or hydraulic cylinders which are connected to the mass via dynamic decoupling springs. The rotation of the blocking mass is restricted using dynamic decoupling springs and “foundation-mounted” stops.

In Figure 2 a) the rotation axis of the test coupling is vertical. Although this could be different from practice, such testing is permitted as long as an unrealistic gravity preload on the flexible test element is prevented. Use of such a “vertical” set-up can be attractive, for example, when a test frame as described in ISO 10846-3 is already available.

5.5.3 Fixtures, etc.

Test objects which have no flat flanges shall be provided with auxiliary fixtures to arrange for appropriate connections to the excitation mass and to the blocking mass. Figures 4 to 8 show examples of a pipe hanger and a cable bundle.

A pipe hanger shall be tested upside down. For the connection to the excitation mass, a pipe clamp shall be used as part of the pipe hanger together with a solid cylindrical rod [see Figure 5 a)]. This rod shall be rigidly connected to the excitation mass, for example, with the aid of two clamps. The clamp of the pipe hanger shall contain a resilient layer inside when this is normal practice. The pipe hanger shall be connected to the blocking mass, or to the output force distribution plate, using a representative fixture [see Figure 5 b)].

A cable or cable bundle shall be connected to the excitation mass using representative connectors. Connection to the blocking mass or to the output force distribution plate shall be made with the aid of a representative clamping device (see Figures 7 and 8).

5.5.4 Gravity load on pipe hangers

Pipe hangers shall be investigated under representative tensional loading. For example, in the indirect method the gravity load may be applied with the aid of the blocking mass. However, to prevent overloading, it may be necessary to compensate part of the gravity load, for example, with the aid of a controllable air mount underneath the blocking mass (see Figure 4). In the example in Figure 6, other type of auxiliaries are needed for the application of preloads. It is the responsibility of the user of this part of ISO 10846 to use a test rig with appropriate preload auxiliaries.

5.5.5 Gas-filled test elements

Gas-filled flexible hoses or bellows shall be tested under representative internal static pressure. The auxiliary equipment needed for this purpose shall not affect the measurement by its mechanical connections. A description of this equipment shall be given in the test report and it shall be made clear that appropriate measures have been taken to avoid any unwanted influence on the measurements.

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.

In cases where the *direct* method is used, another limitation follows from the requirement that the output force shall be a close approximation of the blocking force.

The measurements according to this part of ISO 10846 are valid only for those frequencies where

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

where

a_1 is the acceleration of the input flange;

a_2 is the acceleration of the output flange and output force distribution plate.

NOTE 1 A too small value for the level difference $\Delta L_{1,2}$ can be explained by the stiffness mismatch between the test element and the foundation table being insufficient, or by flanking transmission. When a test frame is used, the frequency range of valid measurements is often limited by an upper frequency f_1, which is the lowest frequency at which the frame can exhibit resonance. Typically f_1 is about 300 Hz for test frames with dimensions of the order of 1 m.

In cases where the *indirect* method is used, 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 to within 1 dB, i.e. to within 12 % of the magnitude of the calculated stiffness. This requirement can only be met in a limited frequency range ($f_2 < f < f_3$). The lower frequency limit is determined by resonances in the test arrangement. Typically this frequency f_2 is in the range from 20 Hz to 50 Hz. The upper frequency is limited by the vibration properties of the blocking mass. Typically this frequency f_3 is in the range 2 kHz to 5 kHz.

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, but where a_1 is the acceleration of the input side and a_2 is 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 excitation mass, the blocking mass and auxiliary springs. Generally speaking, f_2 can be lowered by increasing the blocking mass m_2.

NOTE 2 For design purposes, the lower natural frequencies of the test arrangements can be estimated with the aid of analysis for multibody 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 affect the measurement directions. Nevertheless, at certain frequencies above f_2, Inequality (2) can be violated. Except for test rig imperfections, stiffening of the test element due to internal resonances 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 upper limit f_3 of the frequency range of valid measurements can be controlled by the size and the shape of the blocking mass. This is discussed in 6.3.

6.2 Measurement of blocking force in the direct method

When the *direct* method is used, the mass between the test isolator and the output force transducers causes a bias error in the measurement of the blocking force. The difference between the blocking mass force F_b and the measured force F_2 is approximately equal to the inertia force $m_0 a_2$.

The mass m_0 is the sum of the masses of the output flange of the test element, that of the output force distribution plate and half the mass of the force transducers, and shall respect the following inequality:

$$m_0 \leqslant 0{,}06 \times \frac{10^{L_{F_2}/20}}{10^{L_{a_2}/20}} \text{ kg} \tag{3}$$

NOTE 1 Inequality (3) is equivalent to the requirement that $| L_{F_b} - L_{F_2} | \leqslant 05$ dB.

NOTE 2 If Inequality (3) is not respected, then either a decrease of m_0 or an increase of force transducer(s) stiffness is needed. The latter may imply the use of more transducers or a larger size of transducer.

6.3 Determination of upper frequency limit f_3 in the indirect method

6.3.1 Effective mass

When the *indirect* method is used, 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:

$$k_{2,1} = \underline{F}_{2,b}/\underline{u}_1 \approx -(2\pi f)^2 (m_{2,\text{eff}} + m_f)T \qquad \text{for } |T| << 1 \tag{4}$$

where $m_{2,\text{eff}}$ denotes the effective mass of the blocking body, which still is named “blocking mass”. The effective mass is defined as the frequency-dependent ratio between the excitation force that is exerted by the resilient element upon the blocking mass and the acceleration a_2 of the blocking mass [see Figure 12 a)]. 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.

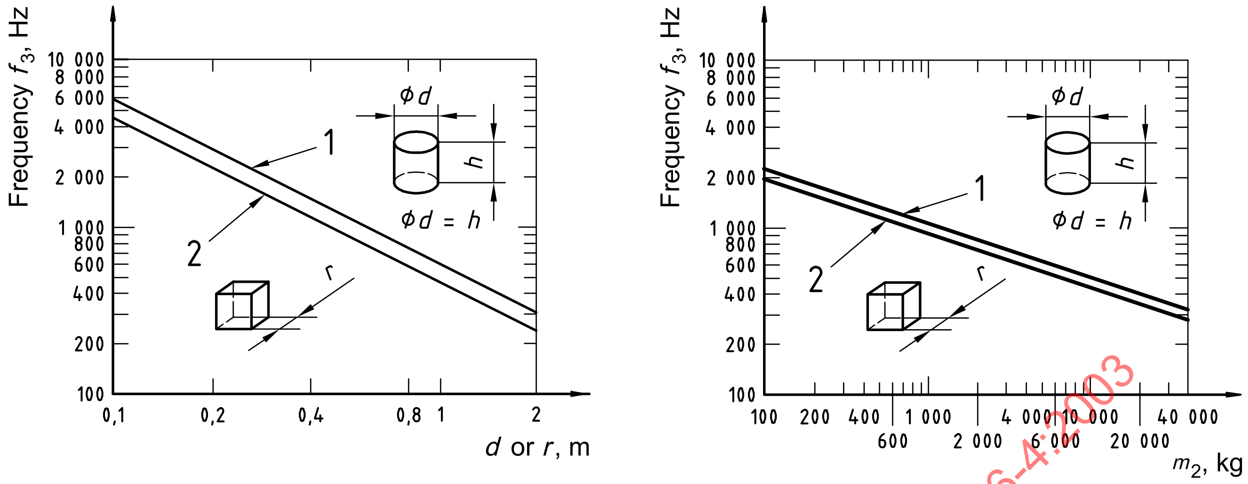
To be suitable for measurements according to this part of ISO 10846, results shall be presented for $f \leqslant f_3$ on the basis of the procedures given in 6.3.2 and 6.3.3.

6.3.2 Use of pre-selected blocking mass

Figure 11 presents nomograms for solid steel blocks with the shape of a cube (side length r) or a cylinder (diameter d, height h and with $d = h$).

If one of these block shapes is used, transfer stiffness data may be calculated using Equation (1) and for frequencies $f \leqslant f_3$, where f_3 for a given dimension is taken from Figure 11 a).

NOTE Figure 11 b) gives the relationship between the mass m_2 and frequency 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 11 b) can be used to determine the corresponding value of f_3, and then Figure 11 a) can be used to determine d or r.



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

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

Key

1 solid steel cylinder

2 solid steel cube

Figure 11 — Nomograms for solid steel cylinders and cubes

6.3.3 Experimental determination of effective mass

6.3.3.1 If the size, shape or mass of blocking masses covered by Figure 11 do not suffice for the measurement purposes, alternative geometries are allowed. However, in this case f_3 has to be determined experimentally. To uncouple translations and rotations, the blocking mass should have such a symmetry that, in a system of Cartesian coordinates with the centre of mass 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 $m_{2,\text{eff}}$ shall be determined as a function of frequency according to the procedure described in 6.3.3.2. 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 have to be calculated using Equation (1) and are only presented for frequencies $f \leqslant f_3$, where the following inequality is valid:

$$| \Delta L | = | 20 \lg (m_{2,\text{eff}}/m_2) | \text{ dB} \leqslant 1 \text{ dB} \qquad (5)$$

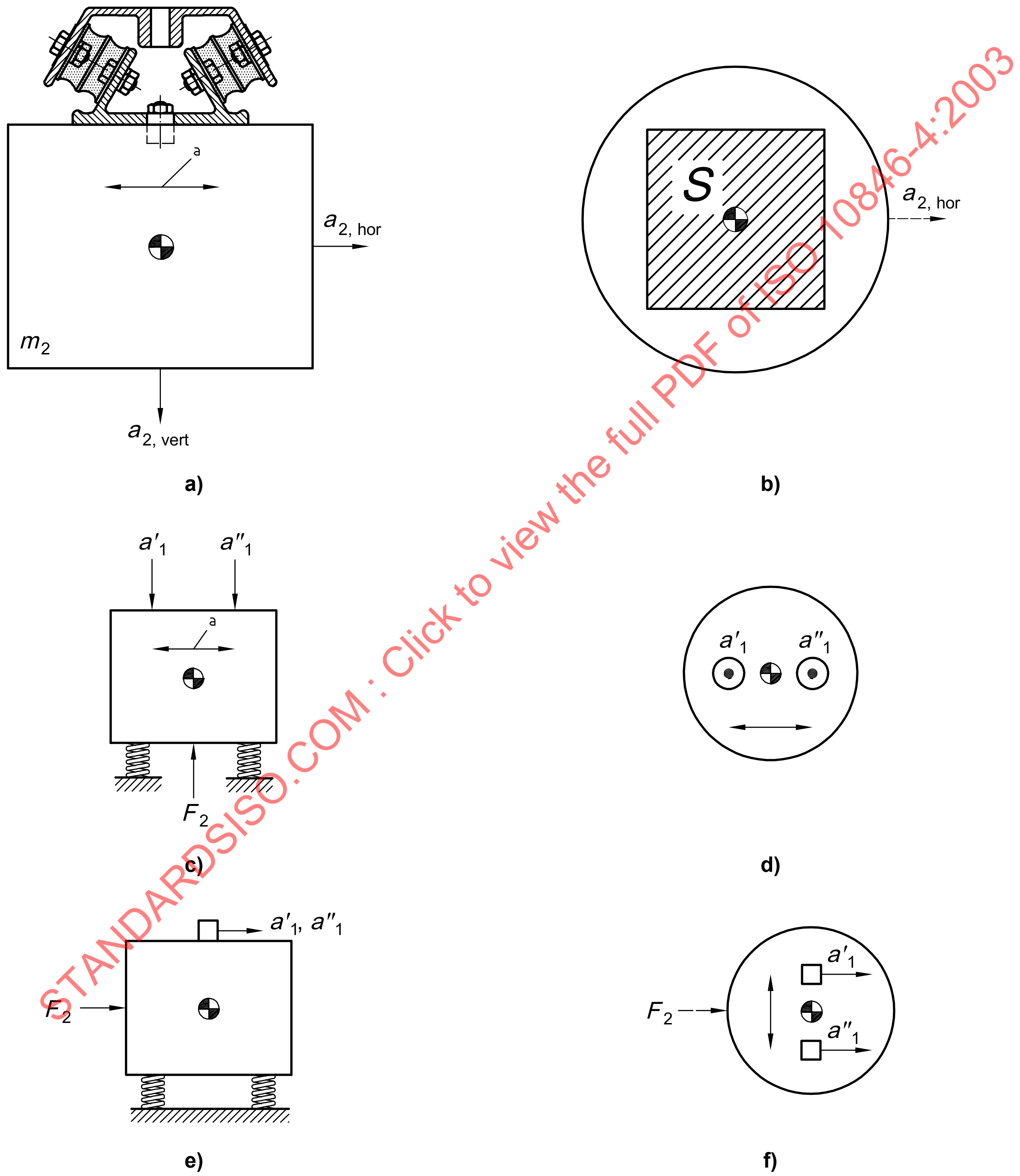
6.3.3.2 The procedure for the measurement of the effective mass is specified with the aid of Figure 12. Figures 12 a) and 12 b) show an example of a test element which is connected to a blocking mass over the contact area S. During the testing of the resilient element, $a_{2,\text{vert}}$ or $a_{2,\text{hor}}$ is measured depending on the direction of excitation on the input side of the element.

Figures 12 c) and 12 d) 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 $a_{2\text{vert}}$ 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 centre of mass and with a spacing of $\sqrt{S}$. The effective mass is defined as

$$m_{2,\mathrm{eff}} = \left| \frac{2F_2}{a_1' + a_1''} \right| \qquad (6)$$

Figures 12 e) and 12 f) 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 centre of mass and with two accelerometers within the contact area S in the horizontal direction. Again the spacing between these accelerometers shall be equal to $\sqrt{S}$ and again the effective mass is defined according to Equation (6).



a Spacing equals $\sqrt{S}$.

Figure 12 — Examples of experimental determination of the effective mass

If at low frequencies (i.e. $f < 40$ Hz) $m_{2,\text{eff}}$ 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 (6) will cause a bias error which makes it impossible to meet Inequality (5) 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). The use of a material with high density and wavespeed is thus preferable (e.g. steel). If necessary, different blocking masses shall be used to cover the desired frequency range.

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, then f_3 (model) = = nf_3 (full scale).

6.4 Flanking transmission

In many test arrangements, flanking transmission can limit the applicability or accuracy of the test method. The flanking transmission can be due to airborne sound or 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.

6.5 Unwanted input vibrations

Input accelerations in directions other than those of the excitation shall be suppressed according to 5.4. Measurements according to this part of ISO 10846 are only valid when the level of the input acceleration in the excitation direction exceeds that in the other directions perpendicular to it by at least 15 dB, i.e.

$$L_{a,\text{wanted}} - L_{a',\text{unwanted}} \geqslant 15 \text{ dB} \tag{7}$$

The measurement positions where this requirement shall hold are shown in Figure 13.

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 the 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 13 a)].

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 with the input flange [see Figure 13 b)].

When the mass-type input flange of the test object replaces the excitation mass (see 5.3.6), a configuration similar to that in Figure 13 shall be defined, to test the adequacy of the suppression of unwanted inputs according to Inequality (7).

6.6 Accelerometers

Accelerometers shall be calibrated in the frequency range of interest and shall have a sensitivity which is frequency independent to within 0,5 dB. Calibration shall be carried out according to ISO 16063-21.

The accelerometers shall be insensitive to extraneous environmental effects such as temperature, humidity, magnetic fields, electrical fields, acoustical fields and strain. Also, 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 apply as for accelerometers.

6.7 Force transducers

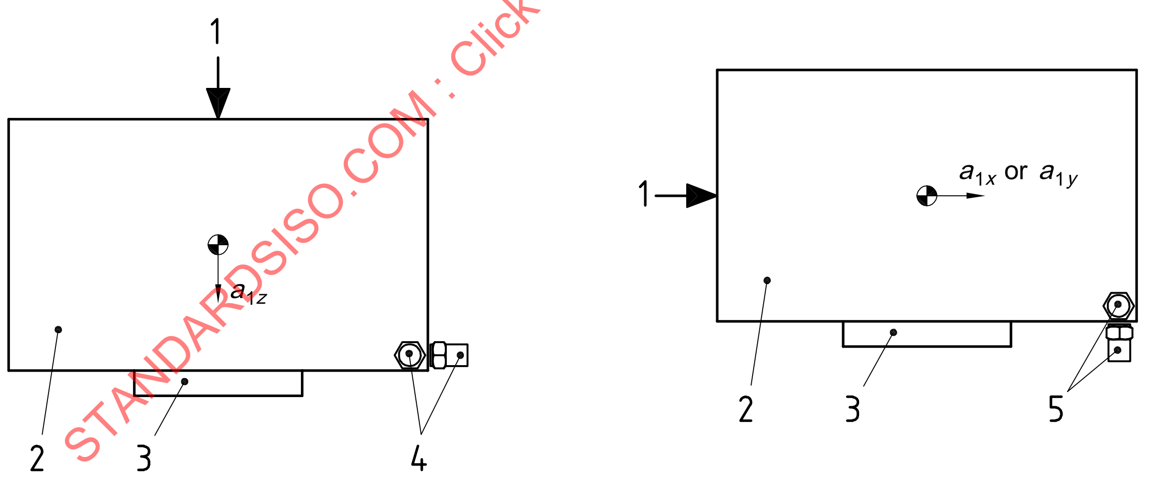
Force transducers shall be used which are calibrated in the frequency range of interest and have a sensitivity level which is frequency independent to within 0,5 dB. Calibration shall be carried out according to the mass-loading technique described in ISO 7626-1.

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

The force transducers shall be insensitive to extraneous environmental effects such as temperature, humidity, magnetic fields, electrical fields, acoustical fields and strain. Also, the sensitivity to cross-axis forces shall be smaller than 5 % of the main axis sensitivity.

6.8 Summation of signals

If signals from force transducers or from accelerometers need to be added, this shall be performed with a maximum uncertainty of 5 %. One way to realize 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 and for differences in channel gain factors (see 6.9).



a) Normal excitation (*z*-direction) **b) Transverse excitation (*x*-or *y*-direction)**

Key

1 exciter
2 excitation mass
3 input flange of test object
4 transducers for measurement of unwanted vibrations in *x*- and *y*-directions
5 transducers for measurement of unwanted vibrations in *z*-direction and in *y*- or *x*-directions

Figure 13 — Measurement locations for checking the suppression of unwanted input vibrations

6.9 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.

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 may 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 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 output force distribution plate or blocking mass in a way that 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.

To improve contact between the resilient test element 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 which contain rubber-type components will show a change of load or deflection due to creep. For such objects, 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 measurements are performed.

7.2 Selection of force measurement system and force distribution plates

When the *direct* method is used, depending upon the size and symmetry of the test isolator and on the maximum permissible load, one or more force transducers are applied.

The force distribution plate shall be as small and as light as possible, but stiff enough to avoid resonances of the system occurring in the frequency range of interest. The minimum lateral dimension is determined by the size of the test object.

To check the rigid body behaviour of the force measurement system, excite the system by a point force in the centre. The transfer function determined from this point force (measured with a calibrated force transducer and the output signal of the force measurement system) shall be flat in the frequency range of interest.

7.3 Mounting and connection of accelerometers

Mount accelerometers on the input and output sides of the test object 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.

7.4 Mounting and connection 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 due to resonance of this rod are avoided.

7.5 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 sufficiently long to allow for averaging so that the measured results do not differ by 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 each one-third-octave band for which stiffness data are determined contains at least five frequencies of the source signal.

7.6 Measurements

7.6.1 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 at one or more specified temperatures, representing the range of environmental temperatures in practice. During the measurements, the environmental temperature shall be monitored. The resilient elements under test shall be exposed for at least 24 h to the appropriate environmental temperature, within ± 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 (such as damping) are very sensitive to changes in temperature or humidity of the element, tolerances for the temperature and humidity shall be defined within which the measurement uncertainty according to 7.6.3 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, adjust the source output 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 on the input side of the test object for the acceleration a_1, on the output-side for the acceleration a_2, and on the foundation of the test rig for the acceleration a_3. Measurement results that do not meet the conditions of 6.5 shall be excluded from the evaluation of the dynamic stiffness function.

7.6.2 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 7.7);

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

NOTE Observation of the coherence function between the appropriate input and output signals is useful, because its value is indicative of low signal-to-noise ratio, non-linearities, or other causes reducing the measurement accuracy.

