INTERNATIONAL STANDARD

ISO 10813-2

First edition
2019-07

Vibration-generating machines — Guidance for selection —

Part 2: Equipment for dynamic structural testing

Générateurs de vibrations — Lignes directrices pour la sélection — Partie 2: Moyens pour les essais dynamiques

Reference number
ISO 10813-2:2019(E)

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Published in Switzerland

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Foreword

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The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

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This document was prepared by Technical Committee ISO/TC 108, *Mechanical vibration, shock and condition monitoring*, Subcommittee SC 6, *Vibration and shock generating systems*.

A list of all parts in the ISO 10813 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

A proper selection of vibration generating system, when purchasing new test equipment or updating existing equipment for the purposes of a specific test or when choosing between the equipment proposed by a test laboratory or even selecting a laboratory which offers its service to carry out such a test, is very important. When making this type of selection, several factors should be considered simultaneously, as follows:

— the type of the test to be carried out (e.g. environmental testing, normal and/or accelerated, dynamic structural testing, diagnosis, calibration, etc.);

— the motions to be generated during the test;

— the test conditions (e.g. single or multiple excitations, one mode of vibration or combined vibration, single or combined test, for example, dynamic plus climatic, etc.);

— the objects to be tested and their mounting.

This document deals only with equipment intended to be used for dynamic structural testing, and selection procedures are predominantly designed to meet the requirements of this testing. However, specific test conditions and the specific object to be tested can significantly influence the selection.

If the equipment is expected to be used for different types of tests, all possible applications should be accounted for when selecting. Thus, if the vibration generator is acquired to be applied during both environmental and dynamic structural testing, ISO 10813-1 and this document should be used simultaneously. In this document, it is presumed that a system can be selected if it enables to swing the test object up to a specified level. To generate an excitation without undesired motions, a suitable control system should be used. The selection of a control system is not considered in this document.

Vibration generating systems are complex machines, so the correct selection always demands a certain degree of engineering judgement. Consequently, the purchaser, when selecting the vibration test equipment, can resort to the help of a third party. In such a case, this document can help the purchaser to ascertain if the solution proposed by the third party is acceptable or not. Designers and manufacturers can also use this document to assess the market environment.

Vibration-generating machines — Guidance for selection —

Part 2: Equipment for dynamic structural testing

1 Scope

This document provides guidance to select a vibration generator that will be used to evaluate frequency responses of a test structure or to study how vibration grows/decreases along the structure. These structural dynamics tests can be carried out under field or laboratory conditions (see the ISO 7626 or ISO 10846[4][5][6][7] series).

This document describes the selection procedure in terms of the force developed by a single vibration generator. Meanwhile, to move massive structures such as dams or bridges, an assembly of vibration generators is usually applied. Properly phased generators produce in total the same force as calculated for a single vibration generator (see 6.2.6).

Guidance also can be applied for the selection of equipment to be used for modal testing to determine natural frequencies, modal shapes and damping in a structure; however, for such a test, more factors than covered by this document usually need to be considered.

This document deals only with translational excitation. For equipment applied to generate angular vibration, see Reference [8].

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 2041, *Mechanical vibration, shock and condition monitoring — Vocabulary*

ISO 7626-1, *Mechanical vibration and shock — Experimental determination of mechanical mobility — Part 1: Basic terms and definitions, and transducer specifications*

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

ISO 15261, *Vibration and shock generating systems — Vocabulary*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 2041, ISO 7626-1, ISO 10846-1 and ISO 15261 apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

4 Dynamic structural testing

4.1 General

Dynamic structural testing is performed to evaluate such characteristics of a structure as

— frequency responses over a wide range of frequencies,

— modal characteristics (mode shapes, natural frequencies, damping ratios, etc.),

— amplification/attenuation of vibration along the structure.

Knowledge of the dynamic behaviour of structures allows for the, for example:

— design of low-vibration mechanical systems (buildings, machinery, transport and their elements), and

— calculation of isolation systems and means used to reduce vibration.

Level of excitation during the structural testing is not so important as compared with environmental testing (see ISO 10813-1) provided that linear behaviour of the test structure is maintained. However, this level should be sufficient to produce the response of the test structure far above the noise floor at frequencies where the mechanical impedance of the structure reaches its maximum values. For a non-linear structure, vibration should be excited at the same level as observed under actual operating conditions.

In a laboratory environment, the test structure can be freely suspended or rigidly supported, whichever is defined in a relevant specification (see, for example, ISO 7626-2). Single-point or multi-point force excitation may be used.

Under field conditions, the vibration generator may be used either coupled or uncoupled to the test structure. Generally, the coupling should be rigid in the direction of excitation but flexible (to allow rotational motions) in a transverse direction.

4.2 Excitation types

ISO 7626-2 and ISO 7626-5 define various possible types of vibration and shock excitation applicable for dynamic structural testing.

In case of linear behaviour of the test structure, any type of excitation specified by ISO 7626-2 and ISO 7626-5 can be used. In the contrary case, only sine vibration should be generated during the test.

5 Vibration generators

5.1 Main types of equipment

A vibration generator is an executive device of a vibration generating system designed to produce some force or kinematic excitation of a test structure. Excitation parameters depend on the purposes and conditions of the test.

Electrodynamic, electromagnetic, piezoelectric and magnetostrictive vibration generators are the most common types of equipment used for dynamic structural testing. To resolve some problems in generating vibration at low frequencies, pneumatic, hydraulic and mechanical vibration generators may be preferred.

5.2 Principal characteristics of vibration generators

Among other measurable parameters of vibration generators used in dynamic structural testing, the following ones are of principal interest when selecting the test equipment:

— rated force;

— permissible static load;

— frequency range;

— limit values for displacement, velocity and acceleration;

— harmonic distortions;

— spurious motions;

— resonance frequencies.

Another key characteristic of the vibration generator is the manner in which it can be fastened to the test structure, whether it requires external mechanical constraints or not. In addition, to provide a specified force excitation, the driving-point impedance of the vibration generator should be far less than the ones of the test structure.

5.3 Features of different vibration generators

5.3.1 Electrodynamic generator

An electrodynamic vibration generator is a device that transforms electric energy of interaction between a static magnetic field and an alternating current conductor into mechanical motion. The current conductor is usually designed as a coil connected to a test table and placed in a circular gap of an iron circuit with a magnetic bias induced by a direct current coil or by a permanent magnet.

The electrodynamic generator produces a force proportional to the excitation current. This property makes this type of equipment rather convenient for use in the broadest range of tests.

Another advantage of the electrodynamic generator is that it enables excitation in a broad range of frequencies (up to 15 000 Hz) and that its input electrical impedance can be easily matched to the output impedance of the power amplifier over the operating band.

In most of electrodynamic generator designs, the iron circuit is rigidly secured in the case and the coil with the rigidly connected table is mounted in the case by means of flexible membranes of non-uniform stiffness, for example, elastic in the axial direction and rigid in a transverse direction.

The principle characteristics of typical electrodynamic generators are given in ISO 10813-1.

Installation and fastening of electrodynamic vibration generators depend on the test purpose and conditions. The electrodynamic vibration generator cannot withstand a considerable static load. Therefore, if the test structure needs to experience a static deformation during the testing, such a deformation should be provided by means of a separate device.

5.3.2 Electromagnetic vibration generator

An electromagnetic vibration generator is a device designed to drive a test structure through the interaction between an electromagnetic field and a ferromagnetic body. The vibratory force grows in a quadratic correlation with the exciting current. To linearize the process, polarization of magnetic flow in the vibration generator iron circuit is applied by means of, for example, a permanent magnet field. The electromagnetic vibration generator can be designed as either a one-gap or two-gap (differential) device. The two-gap generator allows harmonic distortions much less than the one-gap device.

While preparing the test, the constant force of the electromagnetic interaction as well as the mass of generator's armature should be considered, particularly when testing light-weight and/or flexible structures. In these cases, it is recommended to use some unloading device and compensation of a negative stiffness caused by the constant electromagnetic force.

The main advantage of the electromagnetic generator lies in its efficiency and reliability. However, high distortions and a limited operation range confine the scope of its use.

5.3.3 Piezoelectric vibration generator

A piezoelectric vibration generator is a device in which the principle of piezoelectric element straining as affected by electrical field is applied to produce a mechanical excitation of the test object. The magnitude of the strain in the direction of the forcing electric field can achieve up to thousandths of the piezoelectric element dimension in the same direction.

A specific feature of the piezoelectric vibration generator is its capacity to produce broadband kinematic excitation (up to 15 kHz) of heavy and complex mechanical structures. Other advantages are that it can work under very high static loads (up to 2 000 kg/cm^2) and is capable of producing well-directed excitations. The disadvantages are associated with some technological problems in designing an amplifier that can effectively transmit a broadband signal to the piezoelectric generator.

5.3.4 Magnetostrictive vibration generator

The operation of a magnetostrictive vibration generator is based on the magnetostrictive effect — deformation of a ferromagnetic body when it is magnetized.

Like the piezoelectric generator, the magnetostrictive generator is capable to produce broadband kinematic excitation (up to 1 kHz) of heavy and complex mechanical structures. Also, it can be driven with a relatively cheap power amplifier. Significant power consumption is its disadvantage.

5.3.5 Hydraulic vibration generator

A hydraulic vibration generator is a device that produces excitation of the test object by pulsing the pressure of fluid in the hydraulic system controlled by one (or several) servovalve(s).

The basic advantages of the hydraulic vibration generator are the following:

— high magnitudes of displacements (up to 200 mm) and forces (up to 10 MN);

— low transverse vibration of the actuator (vibration table);

— large permissible static loading (up to several tons);

— simple and reliable design.

The basic disadvantages are the following:

— high level of non-linear distortions at low frequencies (up to 15 %);

— rather narrow frequency range not exceeding, as a rule, 200 Hz.

The principle characteristics of typical hydraulic vibration generators are given in ISO 10813-1.

5.3.6 Mechanical vibration generator

A mechanical vibration generator transforms energy of a mechanical driving device. According to their operation, mechanical vibration generators are divided into direct-drive and reaction-type generators.

The basic advantages of mechanical vibration generators are as follows:

— high efficiency;

— simple and reliable design.

The basic disadvantages are the following:

— limited frequency range (usually 5 Hz to 100 Hz);

— high non-linear distortions;

— capability to excite sine vibration only.

The principle characteristics of mechanical vibration generators are given in ISO 10813-1.

5.3.7 Impactor

A typical hammer-type impactor consists of a rigid mass with a force sensor on one side of the impactor mass and a flexible tip on the other side. The force sensor may be replaced with an accelerometer. If the impactor mass oscillates as a rigid body then the output signal of accelerometer is proportional to the force applied to the test subject.

The hammer provides forces up to 10^5 N over the frequency range from 2 Hz to 10 000 Hz. The frequency range of excitation can be adjusted by means of tips (see ISO 7626-5).

To excite large massive objects, a large mass either suspended on cables or free-falling downwards is used.

6 Selection procedure

6.1 General

The selection of test equipment depends first on its functionality, reasonable practicality for specific measurement tasks to be carried out, including requirements related to size, mounting, access to points of excitation, etc. The test equipment should also guarantee the response to be measured to a specified accuracy. This means that the equipment should be capable of generating sufficiently high vibration over the whole frequency range of interest within specified tolerances (on direction, distortions, etc.).

In its turn, the minimum level of excitation depends on background vibration of the test structure and electrical noise in the measurement circuit. Vibration generated by the test equipment should be significantly higher than the background vibration at every point of measurement and at every frequency of interest. The output signal of the vibration transducer should be significantly higher than the electrical noise.

If a vibration generator enables the production of a constant force over the whole frequency range of interest, then the force depends on the mechanical impedance $Z(f)$ of the test structure and a minimum vibration velocity $v(f)$ [see Formula (1) below].

6.2 Procedure

6.2.1 The measurement task, including the test structure, characteristics to be evaluated and frequency range of interest, is specified.

6.2.2 Data concerning the test structure is collected, including:

— the type of the structure (framed structure, machine, isolator, foundation, etc.);

— the size and mass;

— the test conditions (field or laboratory measurement, part of the structure to be tested, access to excitation and measurement points, vibration generator mounting, compatibility of the vibration generator and the test structure interfaces, etc.);

— the background vibration.

6.2.3 Factors such as directivity of excitation, transverse motions, distortions, etc. which can affect the measurement result are evaluated and limited.

NOTE The influence of some factors can be reduced by means of repetitive tests and averaging. For example, if the test specification assumes the structure is to be impacted by a hammer several times, then the excitation directivity can be improved to be within ±5° from a specified axis. Such a limit on the directivity provides an error of the measured mechanical impedance magnitude to be less than 1 %.

6.2.4 Using knowledge of background vibration, the minimum motion to be developed by the vibration generator is calculated in terms of root mean square (RMS) values of acceleration $a_{\mathrm{RMS,min}}$ (f), velocity $v_{\mathrm{RMS,min}}$ (f) or displacement $s_{\mathrm{RMS,min}}(f)$.

Generally, vibration developed by the vibration generator should be three to ten times higher than the background vibration at every frequency of interest.

NOTE 1 If acceleration of the background vibration is uniformly spaced over the frequency range of interest then velocity $v_{\mathrm{RMS,min}}(f)$ and displacement $s_{\mathrm{RMS,min}}(f)$ reach their maximums at the lower limit of the range.

NOTE 2 High level of uniformly distributed background acceleration impedes application of vibration generators unable to develop significant displacements at low frequencies such as piezoelectric ones.

6.2.5 The mechanical impedance $Z(f)$ of the test structure over the frequency range of interest is roughly estimated on the basis of physical modelling, prototype testing, handbook data, and so on. Some recommendations on the rough estimation of $Z(f)$ are given in Annex A for some types of structures.

NOTE Annex A establishes a simple way to evaluate impedances from mass-spring-damper models. There are more complicated methods of a rough estimation of $Z(f)$ for specific structures, for example, using FEM-models[9][10].

6.2.6 The vibratory force $F_{\mathrm{RMS,min}}(f)$ is given by Formula (1):

$$F_{\mathrm{RMS,min}}(f)=v_{\mathrm{RMS,min}}(f)Z(f) \tag{1}$$

where

$v_{\mathrm{RMS,min}}(f)$ is the minimum velocity to be developed by the vibration generator during the testing;

$Z(f)$ is the mechanical impedance of the test structure, driving point or transfer depending on the testing purpose;

f is the frequency.

The frequency $f_{\max}$ at which $F_{\mathrm{RMS,min}}(f)$ reaches its maximum $F_{\max}$, i.e. $F_{\max} = F_{\mathrm{RMS,min}}(f_{\max})$ is fixed.

If several vibration generators are applied to excite the test structure then they are to develop the same vibratory force $F_{\mathrm{RMS,min}}(f)$ as calculated according to Formula (1), i.e.

$$F_{\mathrm{RMS,min}}(f)=\sqrt{\sum_{n=1}^{N} F_{n;\mathrm{RMS,min}}^{2}(f)} \tag{2}$$

where

$F_{n;\mathrm{RMS,min}}(f)$ is the force developed by the n^{th} vibration generator;

N is the number of vibration generators applied during the testing.

6.2.7 The vibration generator is selected according to criteria related to:

a) test conditions and generator's performance (see 6.2.1 to 6.2.3);

b) vibration parameters in terms of:

— vibratory force F_{max} at the frequency f_{max} and forces $F_{RMS,min}(f)$ over the whole frequency range of interest;

— minimum acceleration $a_{RMS,min}(f)$, velocity $v_{RMS,min}(f)$ and displacement $s_{RMS,min}(f)$.

Examples of the selection of a vibration generator depending on the measurement task are given in Annex B.

6.2.8 The vibration transducer is selected.

Vibration transducer and conditioning devices of the measurement circuit should be selected on the assumption that the transducer output $U_{g\ min}$ for the minimum velocity $v_{RMS,min}(f)$ is at least three to ten times higher than the internal circuit noise U_n. Given $U_{g\ min}$, the transducer's sensitivity $S(f)$ can be calculated from Formula (3):

$$S(f) = U_{g\ min}(f) / v_{RMS,min}(f) \quad (3)$$

Annex A
(informative)

Prognosis of mechanical impedance for some types of structures

A.1 General

This annex provides envelopes of magnitude of mechanical impedances for some types of structures. These envelopes lie above real curves of mechanical impedances, obtained as a result of extensive studies (see, for example, Reference [8]), so as to involve some margin of safety. Thus, the envelope curve associated with some structure can be dissimilar to an actual mechanical impedance curve for that structure. Nevertheless, such an envelope allows to obtain a somewhat overestimated value of $Z(f)$.

Envelopes for driving-point and transfer impedances of isolators are considered in A.2. Clauses A.3, A.4 and A.5 deal only with driving-point impedances of coupled or single machines, damped frames and foundations, respectively.

Intrinsically, isolators possess input and output points (input and output flanges) that are used to define a transfer frequency-response function, while such points on other structures are unknown beforehand and should be defined following test specifications.

Structure behaviour differs in different frequency regions. At low frequencies, the change of driving-point/transfer impedance as a function of the frequency f depends on whether elastic or inertial forces prevail in the mechanical system. In the former case, $Z(f)$ decreases as $1/f$ and reaches a maximum at a lower limit of the frequency range. Typical examples of such a system are isolators (see A.2), damped frames (see A.4) and foundations (see A.5). In the case of a rigidly mounted machine which is excited via its feet or frame (see A.3), $Z(f)$ increases as f.

At middle frequencies, the structure behaves like a multiple mass system with springs and dampers. The curve of input mechanical impedance exhibits a train of peaks and valleys that usually can be attributed to natural oscillations of the system and its elements. In this case, valleys are associated with the system resonances while peaks correspond to the system antiresonances. The mechanical impedance $Z(f)$ has maximums at antiresonance frequencies.

Further increase in the frequency leads the driving-point impedance to become dependent on the wave properties in the excitation point. The vibration magnitude is governed by the sizes and elastic moduli of a structural element that is directly excited. If the machine under test is mounted on its feet or elastic supports, a metal supporting plate serves as such an element.

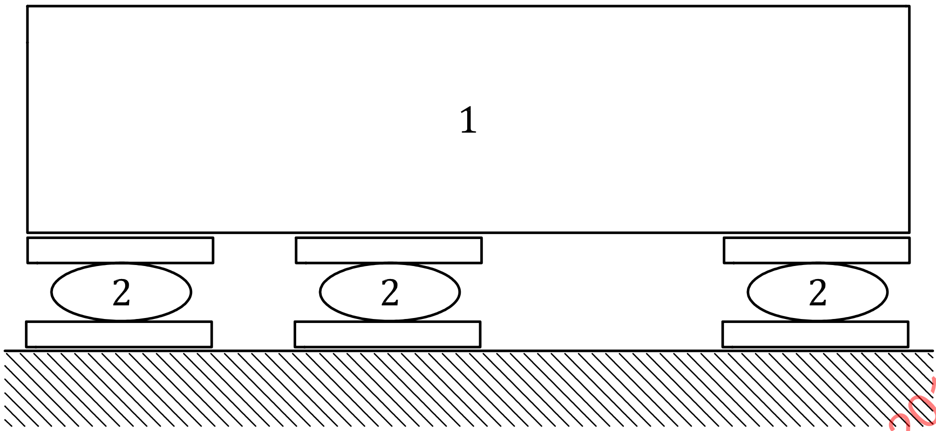
NOTE 1 Frequency regions where the test structure demonstrates essentially different dynamic behaviour are separated by corner frequencies in the figures in Clauses A.2 to A.5.

Practically, when testing machines, it is rather difficult to provide the boundary conditions necessary to measure blocked impedances that form the complete matrix of impedances (see ISO 7626-1). Thus Clause A.3 deals with free impedances.

NOTE 2 Numerical coefficients in the formulae of this annex are empirical ones. They were obtained following numerous research works[8].

A.2 Isolators

The maximum value of the driving-point impedance Z_{11} (Z_{22}) and transfer impedance Z_{12} (Z_{21}) of an isolator loaded by mass m (see Figure A.1) can be evaluated using Figure A.2.



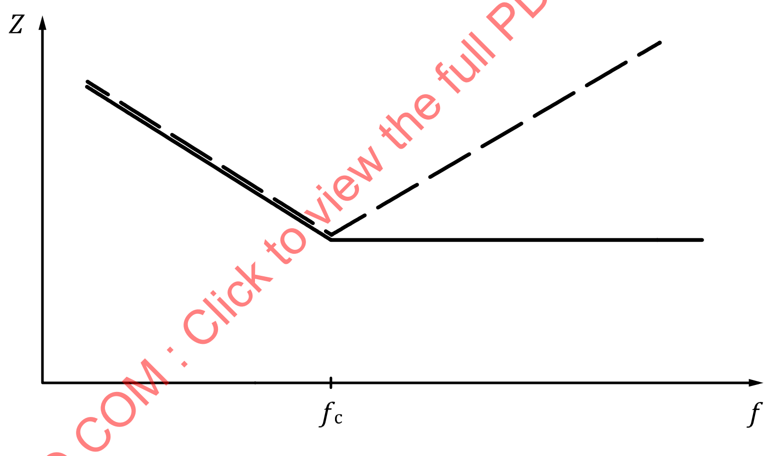
Key

1 machine (loading mass)

2 vibration isolator

NOTE m is the mass of the load per isolator.

Figure A.1 — Isolators loaded by a machine mass



Key

Z	mechanical impedance (driving-point or transfer)	- - -	driving-point impedance Z_{11} (Z_{22})
f	frequency	——	transfer impedance Z_{12} (Z_{21})
f_c	corner frequency		

NOTE The envelopes decrease as $1{,}5C/2\pi f$ right up to the corner frequency f_c (see Table A.1).

Figure A.2 — Envelopes of the driving-point and transfer impedances for a vibration isolator

Given the static stiffness C of the isolator in the direction of the applied excitation, the natural frequency f_0 of the loaded isolator oscillations can be calculated from Formula (A.1):

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{C}{m}} \qquad \text{(A.1)}$$

Then the corner frequency f_c (see Figure A.2) can be taken from Table A.1.

Table A.1 — Relationship between the natural frequency of oscillation and corner frequency for a loaded isolator (see Figure A.1)

f_0 Hz	2 to 5	5 to 6	7 to 11	12 to 17	18 to 30	>30
f_c Hz	163	250	315	630	800	1 000

A.3 Machinery

A.3.1 General

The log-log plot of a typical envelope of free driving-point impedances for a single machine (see Figure A.3) or a machine system is given in Figure A.4.

The upper estimate of mechanical impedances for different frequency regions shown in Figure A.4 can be determined from Formulae (A.2) to (A.4):

$$Z_1(f) = 2\pi f m \quad \text{(A.2)}$$

$$Z_2 = \frac{Z_1(f_1)}{\eta} \quad \text{(A.3)}$$

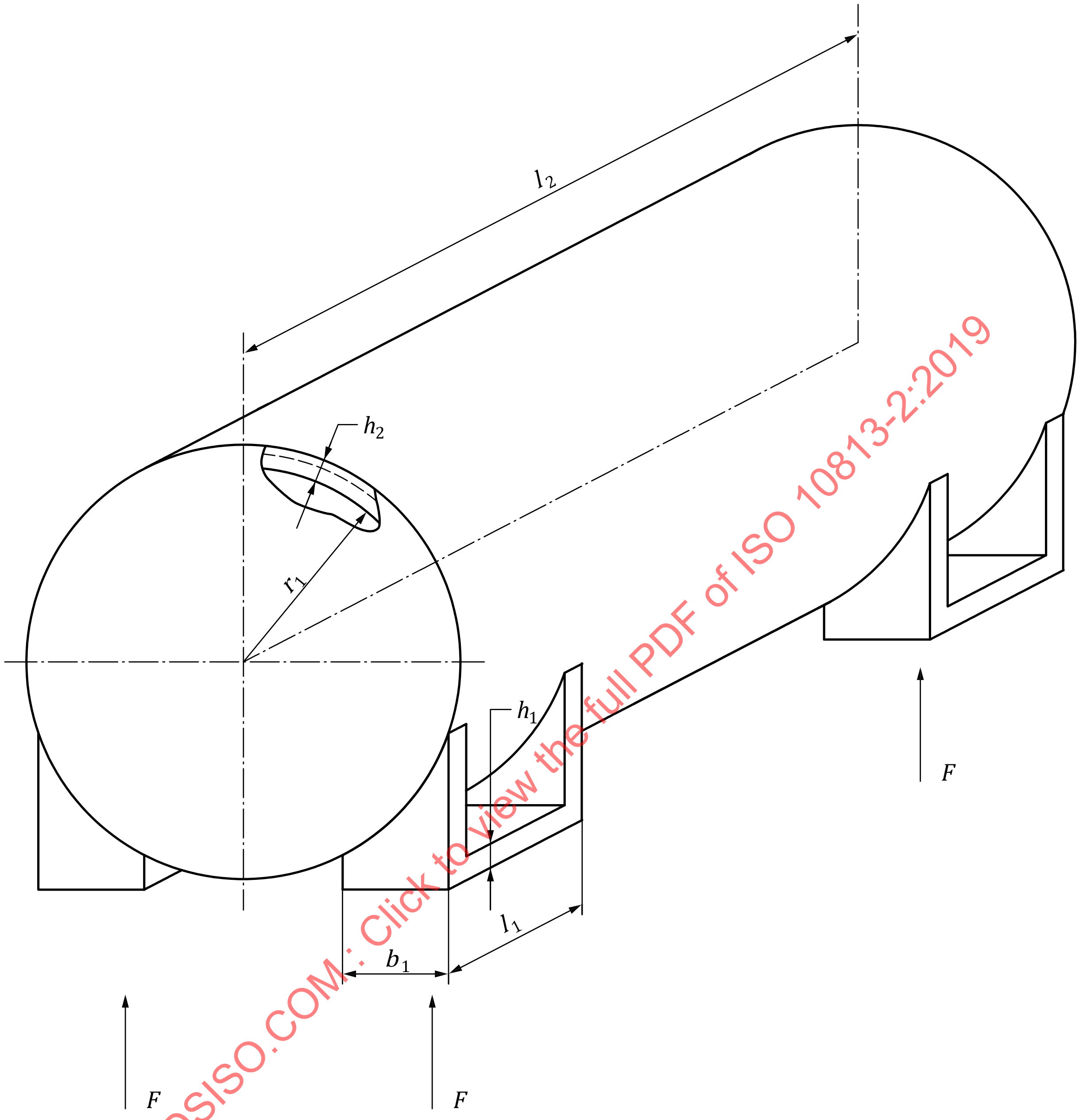
$$Z_4(f) = 2{,}3 h^2 \sqrt{E\rho} \quad \text{(A.4)}$$

where

- f is the frequency;
- m is the mass of the machine or machine system;
- h is the thickness of the plate in the excitation point;
- E is the Young's modulus of the plate;
- ρ is the density of the plate;
- η is the loss factor of the material.

$Z_3(f)$ is plotted in Figure A.4 as the straight line through points $Z_1(f_1)$ and $Z_4(f_2)$, and Z_2 is the envelope value at the antiresonance frequency f_1. The corner frequencies f_1 and f_2 depend on the kind of the machine and the excitation point (see A.3.2 and A.3.3).

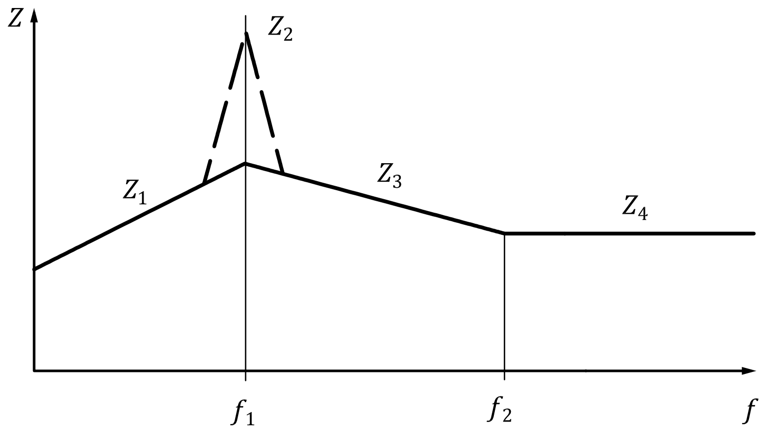
NOTE To some extent of idealization, an antiresonance on the driving-point impedance curve corresponds to resonance oscillations of a certain element of the mechanical system. The less oscillation losses, the higher the antiresonance peak. The peak Z_2 at the frequency f_1 corresponds to the so-called ring frequency of the machine shell.



Key

F	applied force
l_2	machine's shell length
h_2	machine's shell thickness
r_1	machine's shell radius
l_1	machine's foot length
h_1	foot plate thickness
b_1	machine's foot width

Figure A.3 — Sketch of a machine



Key

Z mechanical impedance

f frequency

f_1, f_2 corner frequencies

Figure A.4 — Typical envelope of the driving point impedances for machines

A.3.2 Single machines

A.3.2.1 Excitation through the shell

In this case, the corner frequencies f_1 and f_2 are obtained from Formulae (A.5) and (A.6):

$$f_1 = \frac{0{,}2\,h_2}{r_1^2}\sqrt{\frac{E_s}{\rho_s}} \quad \text{(A.5)}$$

$$f_2 = \frac{0{,}03}{h}\sqrt{\frac{E_s}{\rho_s}} \quad \text{(A.6)}$$

where

h_2 and r_1 are the thickness and the radius of the machine's shell, respectively (see Figure A.3);

E_s and ρ_s are the Young's modulus and density of the shell material, respectively.

A.3.2.2 Excitation through a foot

In this case, the antiresonance frequency f_1 is determined from Formula (A.5) and another corner frequency f_2 is obtained from Formula (A.7):

$$f_2 = 0{,}65\frac{h_1}{b_1}\sqrt{\frac{E_{pl}}{\rho_{pl}}} \quad \text{(A.7)}$$

where

h_1 is the thickness of the foot plate (see Figure A.3);

b_1 is the width of the foot plate (the distance between the internal fixed edge and the external free edge of the plate); see Figure A.3;

E_{pl} and ρ_{pl} are the Young's modulus and density of the foot material, respectively.

A.3.3 Coupled machines

A.3.3.1 Force applied to a machine

In this case, f_1 = min{f_m, f_a}, where f_m is determined from Formula (A.5) and f_a = 40 Hz. The frequency f_2 is determined from Formula (A.6).

NOTE The antiresonance frequency f_1 equals either the frequency of the first machine shell resonance or the frequency of the first machine assembly resonance, whichever is lower. Examples seem to indicate that the latter falls into the one-third octave band with the centre frequency of 40 Hz.

A.3.3.2 Force applied to the common base plate

In this case, f_1 = 40 Hz and the frequency f_2 is obtained from Formula (A.7), in which the base plate parameters are substituted.

A.4 Damped frames mounted on isolators

Maximums of driving-point impedances of a loaded damped frame (see Figure A.5) can be evaluated using the log-log plot in Figure A.6.

The envelope curve consists of four straight lines corresponding to four frequency regions numerated with Roman numbers from I to IV in Figure A.6. Upper estimates $Z_{fr}^{(I)}(f)$, $Z_{fr}^{(II)}(f)$ and $Z_{fr}^{(IV)}(f)$ of the driving-point impedance for the frequency regions I, II and IV, respectively, can be calculated from Formulae (A.8) to (A.10):

$$Z_{fr}^{(I)}(f) = \frac{C}{2\pi f} \tag{A.8}$$

$$Z_{fr}^{(II)}(f) = 2\pi f\, m_t \tag{A.9}$$

$$Z_{fr}^{(IV)}(f) = 46\, h_{fr}^2 \sqrt{E_{fr}\rho_{fr}} \tag{A.10}$$

where

f is the frequency;

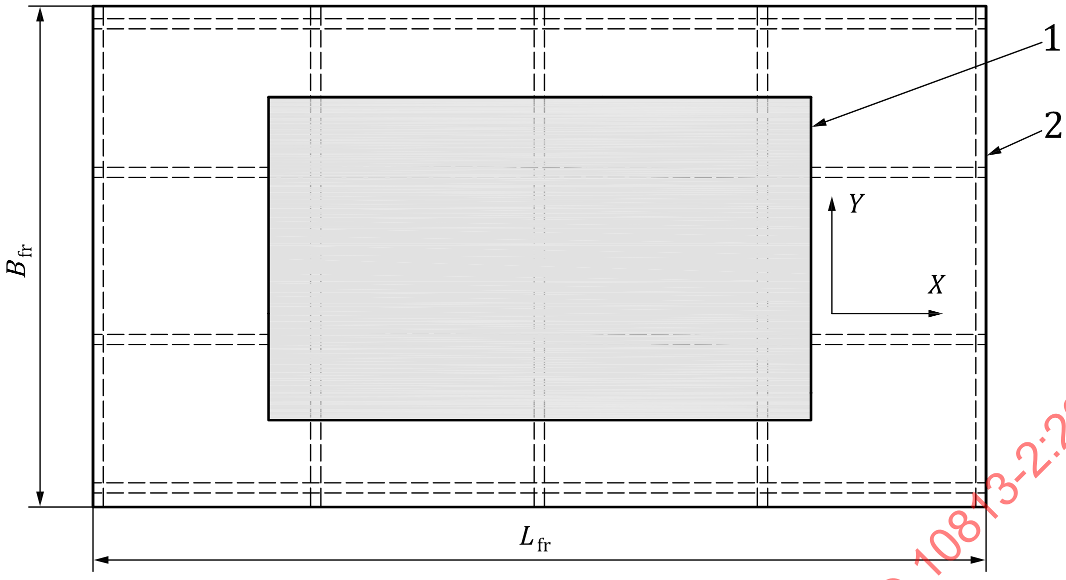
C is the total static stiffness of the isolators;

m_t is the total mass of the frame and machine;

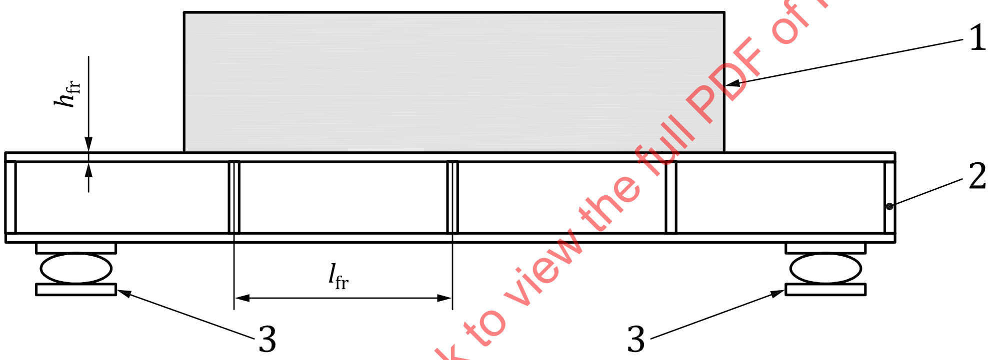
h_{fr} is the thickness of the frame base plate;

E_{fr} is the Young's modulus of the frame material;

ρ_{fr} is the density of the frame material.



a) Top view



b) Front view

Key

1 machine

2 frame

3 isolator

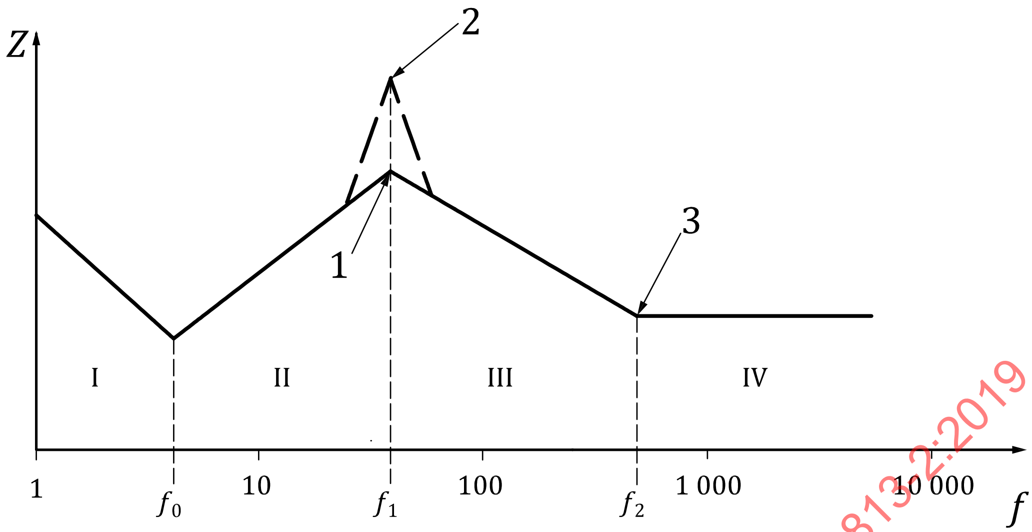
L_{fr} frame length

B_{fr} frame width

l_{fr} distance between strengthening ribs

h_{fr} thickness of the base plate

Figure A.5 — Sketch of a loaded damped frame



Key

Z	mechanical impedance
f	frequency
f_0, f_1, f_2	corner frequencies

Figure A.6 — Typical envelope of driving point impedances of the loaded frame

The straight lines $Z_{\text{fr}}^{(\text{I})}(f)$ and $Z_{\text{fr}}^{(\text{II})}(f)$ are met at the point which corresponds to the natural frequency f_0, and the straight line $Z_{\text{fr}}^{(\text{III})}(f)$, which is the upper estimate of the impedance over the frequency region III, connects points 1 and 3 in Figure A.6, where point 1 corresponds to $Z_{\text{fr}}^{(\text{II})}(f_1)$ and point 3 corresponds to $Z_{\text{fr}}^{(\text{IV})}(f_2)$. The corner frequencies f_0, f_1 and f_2 can be calculated from Formulae (A.11) to (A.13):

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{C}{m_{\text{t}}}} \tag{A.11}$$

$$f_1 = 0{,}6 \cdot \alpha \sqrt{\frac{E_{\text{fr}} J_{\text{fr}}}{m_{\text{t}} L_{\text{fr}}}} \tag{A.12}$$

$$f_2 = 1{,}2 \frac{h_{\text{fr}}}{l_{\text{fr}}^2} \sqrt{\frac{E_{\text{fr}}}{\rho_{\text{fr}}}} \tag{A.13}$$

where

α is a dimensionless factor depending on the ratio δ between the frame length L_{fr} and width B_{fr}, $\delta = L_{\text{fr}}/B_{\text{fr}}$, as shown in Figure A.7;

J_{fr} is the second moment of the frame profile along the axis Y [see Figure A.5 a)];

l_{fr} is the average distance between strengthening ribs of the frame [see Figure A.5 b)].

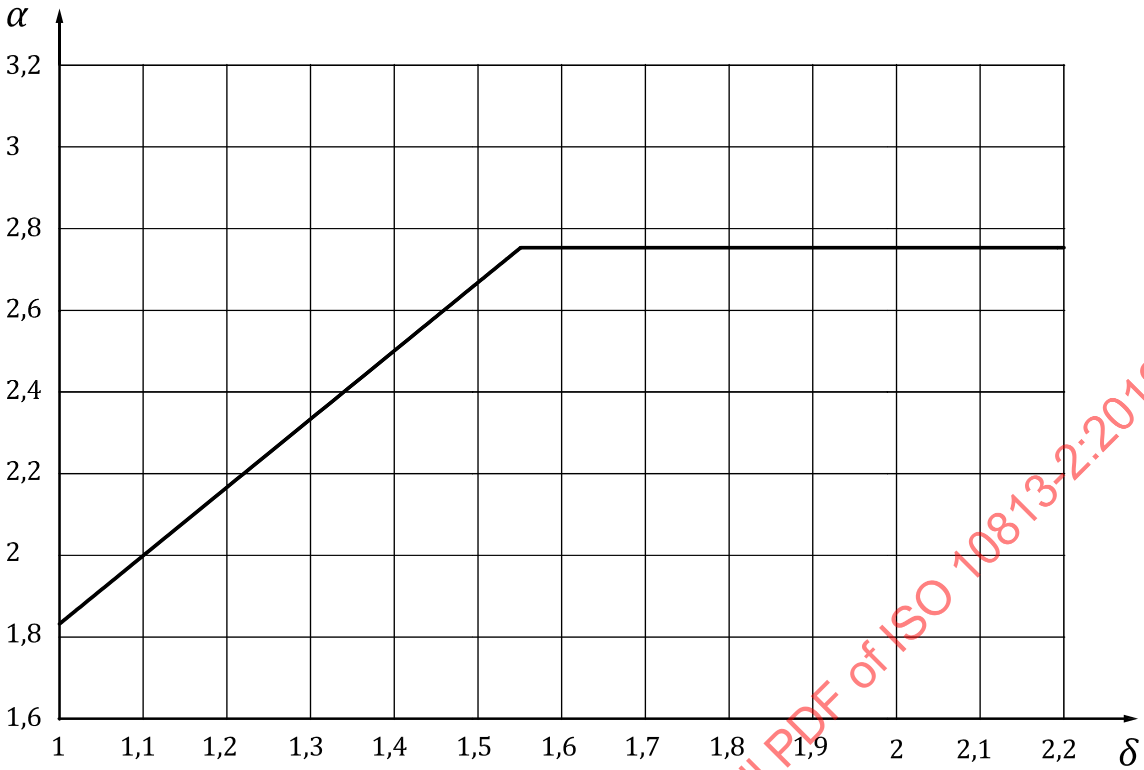


Figure A.7 — Relationship between the factor α and the frame dimension ratio L_{fr}/B_{fr}

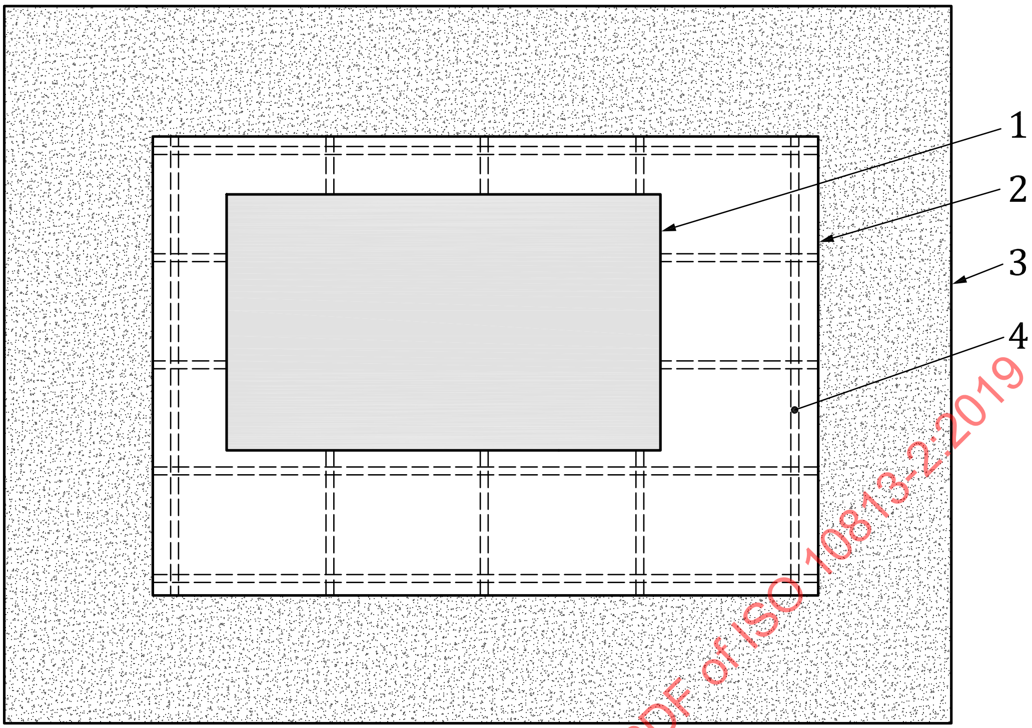
In the vicinity of the antiresonance frequency f_1 (approximately between 0,7f_1 and 1,3f_1), the envelope lies higher than specified by Formulae (A.9) and (A.10) and reaches a peak value Z_{frmax} (point 2 in Figure A.6) which can be evaluated from Formula (A.14):

$$Z_{fr\,max} = Z_{fr\,max}(f_1) = \frac{2\pi f_1 m_t}{0{,}15} \quad \text{(A.14)}$$

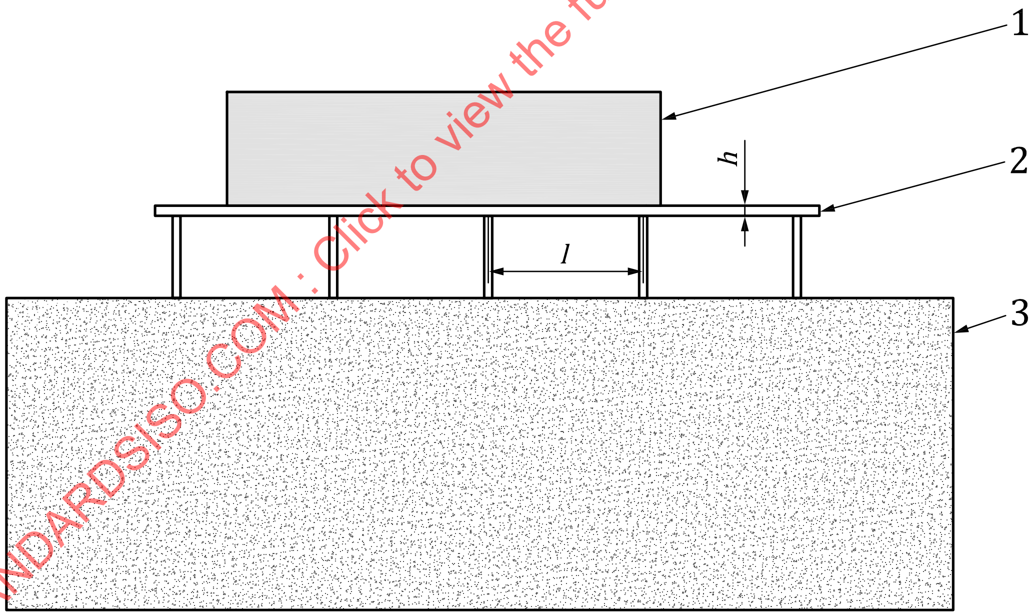
The driving-point impedance of an unloaded suspended frame can be evaluated using the same envelope curve as shown in Figure A.6. However, in this case, the natural frequency f_0 becomes less than 1 Hz, and the first part of the curve which corresponds to the region I should be removed from the plot.

A.5 Machine's base plate

Maximum possible values of driving-point impedances of a steel plate that works as an intermediate structure between a rigidly mounted machine and a blocked heavy foundation of a building or transport facility (see Figure A.8) can be evaluated using the log-log plot in Figure A.9.



a) Top view

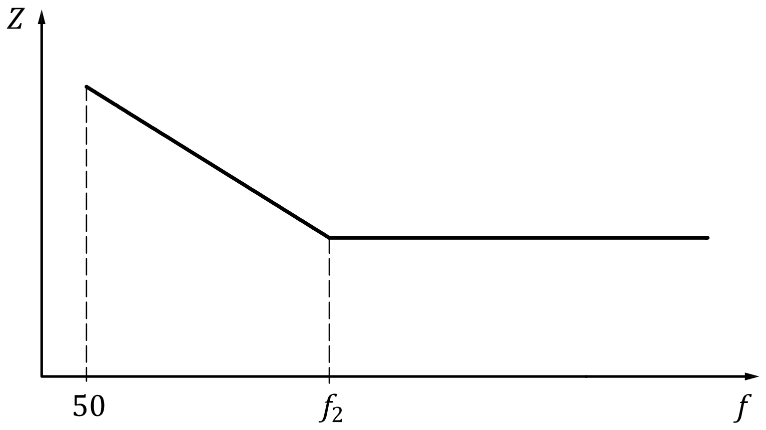


b) Front view

Key

1	machine	4	strengthening rib
2	base plate	*l*	distance between strengthening ribs
3	heavy foundation	*h*	thickness of the base plate

Figure A.8 — Sketch of a loaded base plate



Key

Z mechanical impedance

f frequency, Hz

f_2 corner frequency

Figure A.9 — Typical envelope of driving point impedances of the machine's base plate

The envelope of driving-point impedances $Z(f)$ at frequencies higher than 50 Hz can be determined from Formula (A.15):

$$Z(f)=\begin{cases}\dfrac{f_2}{f}46\,h^2\sqrt{E\rho}\,, & f\leq f_2\\ 46\,h^2\sqrt{E\rho}\,, & f>f_2\end{cases} \tag{A.15}$$

where

f is the frequency;

h is the thickness of the base plate;

E is the Young's modulus of the base plate material;

ρ is the density of the base plate material;

f_2 is the corner frequency which can be calculated approximately from Formula (A.16):

$$f_2\approx 1{,}2\frac{h}{l^2}\sqrt{\frac{E}{\rho}} \tag{A.16}$$

where l is the average distance between strengthening ribs of the base plate [see Figure A.8 b)].

NOTE At frequencies over 50 Hz, the driving-point impedance $Z(f)$ depends on the mechanical properties of the base plate itself. At lower frequencies, the properties of the heavy foundation also need to be considered. It would be safe to assume for the envelope at frequencies below 50 Hz that it equals the driving-point impedance $Z(f)$ at the frequency of 50 Hz.