
**Vibration generating machines —
Guidance for selection —**

**Part 1:
Equipment for environmental testing**

*Générateurs de vibrations — Lignes directrices pour la sélection —
Partie 1: Moyens pour les essais environnementaux*

STANDARDSISO.COM : Click to view the full PDF of ISO 10813-1:2023



STANDARDSISO.COM : Click to view the full PDF of ISO 10813-1:2023



COPYRIGHT PROTECTED DOCUMENT

© ISO 2023

All rights reserved. Unless otherwise specified, or required in the context of its implementation, no part of this publication may be reproduced or utilized otherwise in any form or by any means, electronic or mechanical, including photocopying, or posting on the internet or an intranet, without prior written permission. Permission can be requested from either ISO at the address below or ISO's member body in the country of the requester.

ISO copyright office
CP 401 • Ch. de Blandonnet 8
CH-1214 Vernier, Geneva
Phone: +41 22 749 01 11
Email: copyright@iso.org
Website: www.iso.org

Published in Switzerland

Contents

Page

Foreword	iv
Introduction	v
1 Scope	1
2 Normative references	1
3 Terms and definitions	1
4 Vibration test requirements	2
4.1 Vibration test purposes	2
4.2 Test methods	2
4.2.1 General	2
4.2.2 Sinusoidal vibration	2
4.2.3 Wide-band random vibration	3
4.2.4 Time-history method	3
4.2.5 Sine-beat method	3
5 Types and characteristics of vibration generators	4
5.1 Main types of vibration generators	4
5.1.1 General	4
5.1.2 Electrodynamic vibration generators	4
5.1.3 Servo-hydraulic vibration generators	4
5.1.4 Mechanical vibration generators	5
5.2 Major parameters	5
5.3 Features	5
5.3.1 Electrodynamic vibration generators	5
5.3.2 Servo-hydraulic vibration generators	8
5.3.3 Mechanical vibration generators	11
5.4 Comparison between electrodynamic, servo-hydraulic and mechanical vibration generators	13
6 Recommendations for the selection of vibration generators	14
6.1 Selection of type	14
6.2 Selection of the model	15
6.2.1 General	15
6.2.2 Frequency range	15
6.2.3 Maximum acceleration	16
6.2.4 Force	16
6.2.5 Mass of the moving element	16
6.2.6 Size of the moving element	17
6.2.7 Rated travel	17
6.2.8 Maximum velocity	18
6.3 Selection of components	19
6.3.1 General	19
6.3.2 Selection of power amplifier	19
6.3.3 Selection of hydraulic power supply	23
6.4 Verification of adequate infrastructure	25
Annex A (informative) Examples of selections	27
Bibliography	30

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.

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 document 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).

ISO draws attention to the possibility that the implementation of this document may involve the use of (a) patent(s). ISO takes no position concerning the evidence, validity or applicability of any claimed patent rights in respect thereof. As of the date of publication of this document, ISO had not received notice of (a) patent(s) which may be required to implement this document. However, implementers are cautioned that this may not represent the latest information, which may be obtained from the patent database available at www.iso.org/patents. ISO shall not be held responsible for identifying any or all such patent rights.

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation on the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT) see the following URL: www.iso.org/iso/foreword.html.

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

This second edition cancels and replaces the first edition (ISO 10813-1:2004), which has been technically revised. It also incorporates the Technical Corrigendum ISO 10813-1:2004/Cor.1:2006.

The main changes are as follows:

- tables and figures are updated to reflect advances in typical system capability;
- selection calculations are rearranged to use values which are more likely to be available;
- a non-mathematical treatment of armature size and infrastructure requirements is included.

A list of 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

To select a suitable vibration generating system is an urgent problem if it is necessary for a certain test to purchase new test equipment or to update the equipment already available, or to choose between equipment proposed by a test laboratory or even a laboratory itself which offers its service to carry out such a test. A problem like this can be resolved acceptably if a number of factors are considered simultaneously, as follows:

- the type of the test to be carried out (environmental testing, normal and/or accelerated, dynamic structural testing, diagnosis, calibration, etc.);
- the requirements to be followed;
- the test conditions (one mode of vibration or combined vibration, single vibration test or combined test, for example, dynamic plus climatic);
- the objects to be tested.

This document deals only with equipment to be used during environmental testing, and those selection procedures that are predominantly to meet the requirements of this test. However, the user should keep in mind that a specific test condition and a specific object to be tested can significantly influence the selection. Thus, to excite a specimen inside a climatic chamber imposes limitations on the vibration generator interface, and a specimen of a large size and/or of a complex shape, having numerous resonances in all directions, demands larger equipment than that specified for the procedures of this document, assuming that excitation is to be applied to the rigid body of the same mass. Unfortunately, such aspects cannot easily be formalized and, thus, are not covered by this document.

If the equipment is expected to be used for tests of different types, all possible applications should be considered when selecting. Other parts of ISO 10813 will address the problem of the case where the vibration generator is acquired to be applied during both environmental and dynamic structural testing. It is presumed in this document that the system selected will be able to drive the object under test up to a specified level. In order to generate an excitation without undesired motion, a suitable control system should be used. The selection of a control system will be considered in a further International Standard.

It should be emphasized that vibration generating systems are complex machines, so the correct selection always demands a certain degree of engineering judgement. As a consequence, 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.

STANDARDSISO.COM : Click to view the full PDF of ISO 10813-1:2023

Vibration generating machines — Guidance for selection —

Part 1: Equipment for environmental testing

1 Scope

This document gives guidance for the selection of vibration generating equipment used for vibration environmental testing, depending on the test requirements.

This guidance covers such aspects of selection as

- the equipment type,
- the model, and
- some main components, excluding the control system.

NOTE 1 Some examples are given in [Annex A](#).

NOTE 2 This document is primarily focused on determining functional specifications for equipment based on the requirements of a specific environmental test. More practical aspects of the selection (including target test definition and fixturing considerations) are covered in IEST-RP-DTE009.1^[1].

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 5344, *Electrodynamic vibration generating systems — Performance characteristics*

ISO 8626, *Servo-hydraulic test equipment for generating vibration — Method of describing characteristics*

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 5344, ISO 8626 and ISO 15261 apply.

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

- IEC Electropedia: available at <https://www.electropedia.org/>
- ISO Online browsing platform: available at <https://www.iso.org/obp>

4 Vibration test requirements

4.1 Vibration test purposes

The purpose of vibration tests is to estimate the capability of an object to maintain its operational characteristics and to stay intact under vibration loading of defined severity. The tests are subdivided, in accordance with their tasks, into functional, strength and endurance tests.

Endurance tests are carried out to estimate the capability of an object to withstand vibration of defined severity and to stay in working order when the excitation is removed. Vibration endurance testing is oftentimes referred to as strength testing. In these tests, vibration might cause mechanical damage (fatigue) and may be used to predict the lifetime of the object under vibration.

Functional tests are carried out to estimate the capability of an object to function and maintain the operational parameters within the acceptable limits under vibration. Usually during those tests, the object is working for a defined period in its normal condition and is being exposed to vibration not causing mechanical damage to it. Faults and malfunctions in the operation of the object should be registered.

4.2 Test methods

4.2.1 General

Laboratory test methods may use both sinusoidal and multifrequency excitation in various forms, such as sinusoidal at a fixed frequency, swept sinusoidal, random (narrow-band or wide-band), as well as in a mixed mode. The excitation may be multidirectional and/or multipoint.

Test specifications usually deal with the following waveforms:

- sinusoidal at a fixed frequency;
- swept sinusoidal;
- wide-band random;
- time history;
- sine-beat.

The above waveforms are briefly described in [4.2.2](#) to [4.2.5](#) primarily in aspects as standardized by IEC 60068-2-6^[2], IEC 60068-2-57^[3] and IEC 60068-2-64^[4], however the user should be aware that other variants of a waveform may be used for specific applications.

Requirements for the test excitation (and, hence, for the test equipment) for test methods are standardized by IEC 60068-2-6^[2], IEC 60068-2-57^[3] and IEC 60068-2-64^[4].

4.2.2 Sinusoidal vibration

4.2.2.1 Sinusoidal vibration at fixed frequencies

This excitation consists of a set of discrete-frequency sinusoidal processes of defined amplitude, applied sequentially to the test object within the frequency range of interest. Frequency and amplitude are adjusted manually. A control system maintains the displacement or acceleration amplitude. The test conditions to be set include the frequency range (bands) and individual fixed frequencies, test duration and displacement, velocity or acceleration amplitude.

4.2.2.2 Swept sinusoidal vibration

This excitation is a sinusoidal signal of a specified amplitude, commonly defined in displacement terms at low frequencies and in acceleration terms at high frequencies. The frequency is continuously swept from the lower to the upper limit of the frequency range of interest and vice versa. Cross-over frequency usually lies in the range of 10 Hz to 100 Hz. A control system maintains the displacement or acceleration amplitude. During the frequency sweep, the mechanical resonances and undesirable mechanical and functional behaviour of the test object can be observed and identified. The test conditions to be set include the frequency range of interest, displacement and acceleration amplitudes, cross-over frequency, sweep rate, sweep type (linear or logarithmic) and test duration.

NOTE 1 Test amplitude is uniquely specified at each excitation frequency; however, it can vary across the frequency band.

NOTE 2 Any two of displacement amplitude, acceleration amplitude and cross-over frequency is sufficient to derive the third value. It is common to provide only the amplitudes.

4.2.3 Wide-band random vibration

The wide-band random excitation, specified by the shape of power spectral density of acceleration to be close to real operational conditions in the frequency range of interest, is generated at the control point of the table or the object. The test conditions to be set include the acceleration power spectral density levels for the frequency bands in which tests are carried out.

4.2.4 Time-history method

This test consists of subjecting the specimen to a time-history specified by a response spectrum with characteristics simulating the effects of short-duration random-type forces. A time-history may be obtained from a natural event (natural time-history), or from a random sample, or as a synthesized signal (artificial time-history). The use of a time-history allows a single test wave to envelop a broad-band response spectrum, simultaneously exciting all modes of the specimen on account of the combined effects of the coupled modes.

This test is applied to specimens which in service can be subjected to short-duration random-type dynamic forces induced, for example, by earthquakes, explosions or transportation.

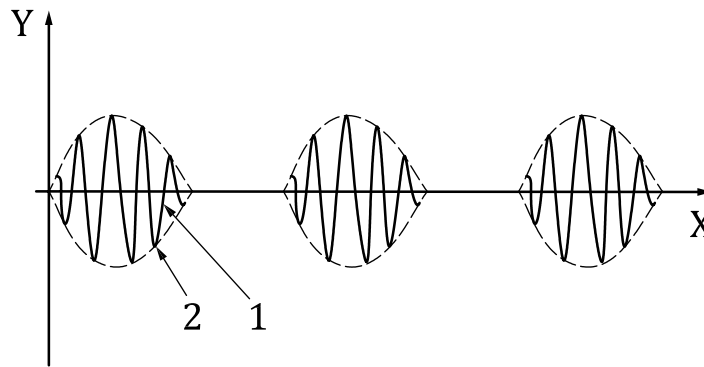
The test conditions to be set include the frequency range of interest, required response spectrum, number and duration of time-histories, number of high peaks of the response.

4.2.5 Sine-beat method

In this test the specimen is excited at fixed frequencies (to be experienced in the practical application or to be changed with a step of not greater than one-half octave) with a pre-set number of sine beats (see [Figure 1](#)). These fixed frequencies may be critical frequencies identified by means of vibration response investigation.

The test conditions to be set include the frequency range, test level, number of cycles in the sine beat, number of sine beats. A control system maintains the displacement amplitude below the cross-over frequency and the acceleration amplitude above the cross-over frequency.

Each channel of the instrumentation needs to be checked by mechanical excitation of the transducer prior to and after each measurement series to ensure proper functioning.

**Key**

- X time
 Y vibration magnitude
 1 carrier wave (test frequency)
 2 envelope curve (modulating frequency)

Figure 1 — Typical sequence of sine beats

5 Types and characteristics of vibration generators

5.1 Main types of vibration generators

5.1.1 General

A vibration generator is the final control element of a vibration generating system, providing generation of the desired vibration and transmission of it to the object being tested. The type and performance of a vibration generator determine the main system characteristics, such as force generation capabilities, permissible loads, displacement/velocity/acceleration amplitudes, frequency ranges and accuracy characteristics (tolerances, distortions, transverse motions, etc.). Depending on their design, vibration generators are subdivided into electrodynamic, servo-hydraulic, mechanical, electromagnetic, piezoelectric, magnetostrictive, etc. The most common types of vibration generators being used for environmental testing are electrodynamic, servo-hydraulic and mechanical.

5.1.2 Electrodynamic vibration generators

This type of vibration generator produces a vibration force by interaction of a static magnetic field and an alternating current in the moving coil, which is an actuator.

A vibration generating system including an electrodynamic vibration generator is called an electrodynamic system. It consists of a power amplifier, input signal source and control system, measuring instrumentation, field power supply and auxiliaries. The system may also include an auxiliary table.

5.1.3 Servo-hydraulic vibration generators

This type of vibration generator produces a vibration force by application of a liquid pressure being changed in a predetermined manner. In servo-hydraulic vibration generators, force and motion are transmitted to the object by a hydraulic actuator (piston pushed by fluid) controlled by servo valves.

A vibration generating system including a servo-hydraulic vibration generator is called a servo-hydraulic system. It consists of a hydraulic power supply system, signal source, close-loop control system, and measurement and auxiliary equipment.

5.1.4 Mechanical vibration generators

This type of vibration generator produces a vibration force by transformation of mechanical rotation energy.

Mechanical vibration generators are classified into kinematic and reaction-type vibrators.

In kinematic vibrators, the test object is moved by some control unit directly, for example by a crank, a rocker or a cam.

In reaction-type vibrators, the centrifugal force is generated by rotational movement (sometimes by reciprocal movement) of unbalanced masses.

A vibration generating system including a mechanical vibration generator is called a mechanical system.

5.2 Major parameters

ISO 5344 and ISO 8626 deal with characteristics of electrodynamic and servo-hydraulic vibration generators respectively. They cover the following main characteristics:

- rated force;
- permissible static load;
- frequency range;
- limits for displacement, velocity and acceleration;
- distortion;
- transverse motion ratio;
- non-uniformity of table motion;
- resonance frequencies.

5.3 Features

5.3.1 Electrodynamic vibration generators

[Table 1](#) gives typical parameters for electrodynamic vibration generators of a “traditional” design. Force ratings for sinusoidal (peak value) and random (r.m.s. value) performance are assumed to be the same, although this is not always the case. Manufacturers offer various series or steps of force ratings for the vibration generating system which will not line up exactly with [Table 1](#). Significant deviation from the patterns established in the table may indicate a vibration generator designed for a special purpose, such as specialized shock transient performance. When a system is being purchased from a manufacturer, or being selected for usage from several systems of purchaser's own, it is recommended to use actual specification sheets.

Table 1 — Typical parameters for electrodynamic vibration generators

Rated force	Output of the power amplifier	Frequency range	Maximum displacement	Maximum velocity	Maximum acceleration without load	Maximum load	Mass of moving system
N	VA	Hz	mm	m/s	m/s ²	kg	kg
30	6,3	5 to 13 000	±12,5	1,8	200	1,0	0,15
60	19	5 to 10 000	±12,5	1,8	300	1,5	0,2
125	62,5	5 to 8 000	±12,5	1,8	500	2,0	0,25
250	165	5 to 8 000	±12,5	1,8	650	4,0	0,38
500	400	5 to 7 000	±12,5	1,8	800	10,0	0,62
1 000	1 000	5 to 5 000	±25	2,0	1 000	25,0	1,0
2 000	2 000	5 to 5 000	±25	2,0	1 000	75,0	2,0
4 000	4 000	5 to 4 000	±25	2,0	1 000	200,0	4,0
8 000	8 000	5 to 3 500	±38	2,0	1 000	300,0	8,0
16 000	16 000	5 to 3 000	±38	2,0	1 000	400,0	16,0
32 000	32 000	5 to 2 500	±38	2,0	1 000	500,0	32,0
64 000	64 000	5 to 2 000	±38	2,0	1 000	1 000,0	64,0
128 000	128 000	5 to 1 800	±38	2,0	1 000	2 000,0	128,0
200 000	200 000	5 to 1 600	±38	2,0	1 000	3 125,0	200,0

NOTE 1 Upper limits for different vibration parameters generally cannot be achieved simultaneously for extended durations due to thermal constraints. See ISO 5344 for more details on electrodynamic vibration system specification.

NOTE 2 When the rated force is an r.m.s. value in random mode, this value is limited by electrical power draw and heat dissipation. It is understood that higher instantaneous peak forces (often six times the r.m.s. level or more) will occur. The manufacturer is responsible for designing components to survive these brief transients. This does not affect the selection process.

The main features of electrodynamic vibration generators are the following:

- any type of excitation is possible: sinusoidal (at fixed frequencies and swept), random (broad-band and narrow-band), shock, etc.;
- ease of control (manual and automatic);
- wide frequency range: 0,5 Hz up to 15 000 Hz (typically 5 Hz to 5 000 Hz); in general, the lower the rated force the higher the upper limit of the frequency range;
- high displacement: up to ±50 mm (typically up to ±25,5 mm or ±38 mm), and acceleration: up to 1 500 m/s² (typically up to 1 000 m/s²);
- high force: up to 400 kN (typically up to 200 kN);
- relatively large permissible load: up to 4 000 kg (typically up to 1 000 kg);
- low harmonic distortion: about 5 %, excluding frequency bands where distortion increases because of resonances between the vibration generator and the load;
- acceptable transverse motion and uniformity of table motion: about 10 %, excluding frequency bands where an undesired motion arises due to moving system resonances or off-set test loads.

One disadvantage of electrodynamic generators is caused by the presence of a magnetic field in the area of the vibration table. This, however, may be reduced to the order of 0,001T by means of special compensation devices.

Also rated force cannot be generated over the whole frequency range. It is limited by the rated travel at low frequencies, by the rated velocity at middle frequencies and by the resonances of the moving

system at high frequencies. Achievable acceleration depends on the load mass. ISO 5344 states six test loads m_0 , m_1 , m_4 , m_{10} , m_{20} and m_{40} , where the first load is zero and the following are those permitting maximal accelerations of 10 m/s², 40 m/s², 100 m/s², 200 m/s² and 400 m/s² respectively.

Figure 2 shows typical curves of acceleration (displacement, velocity) against frequency for various loads.

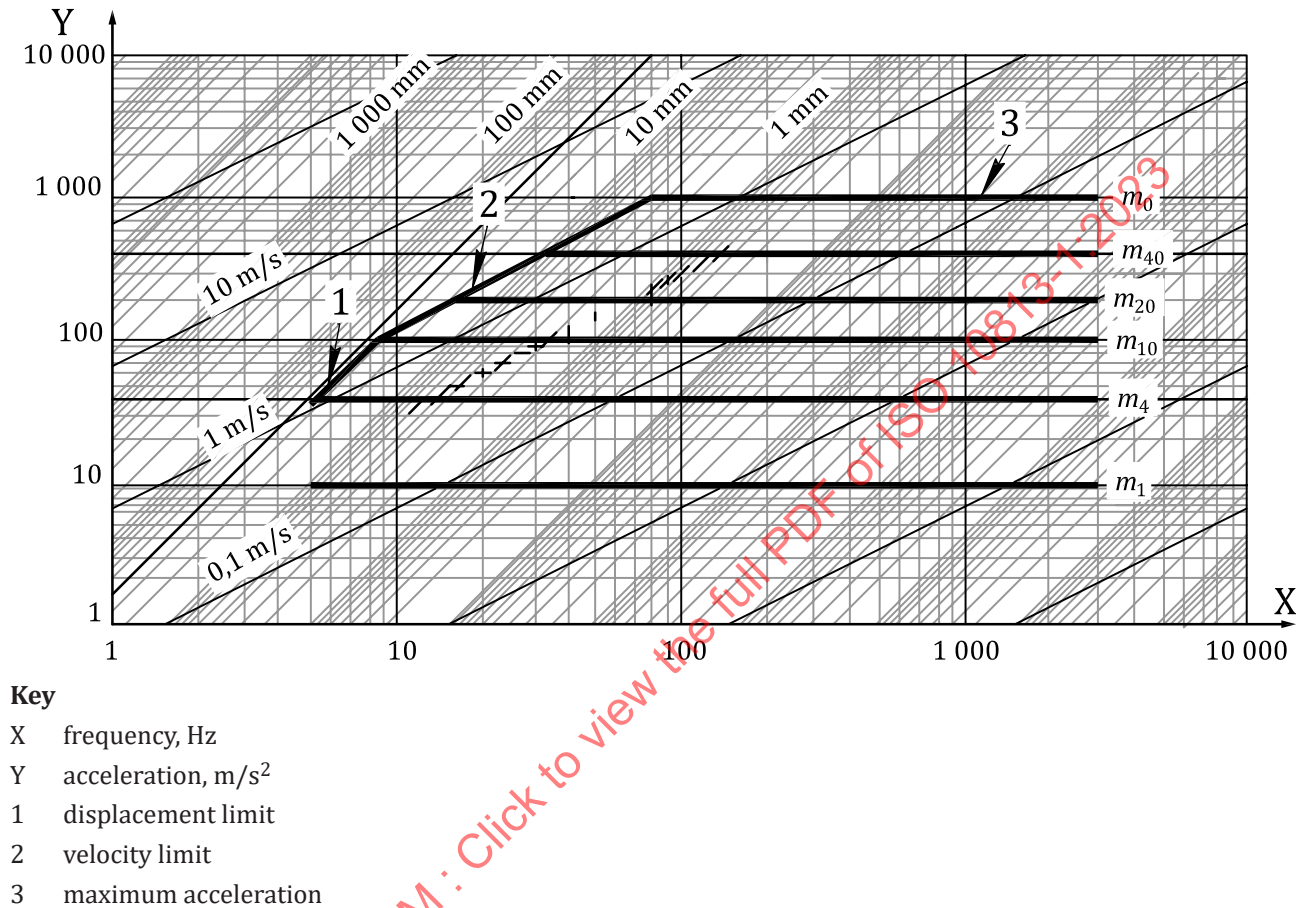
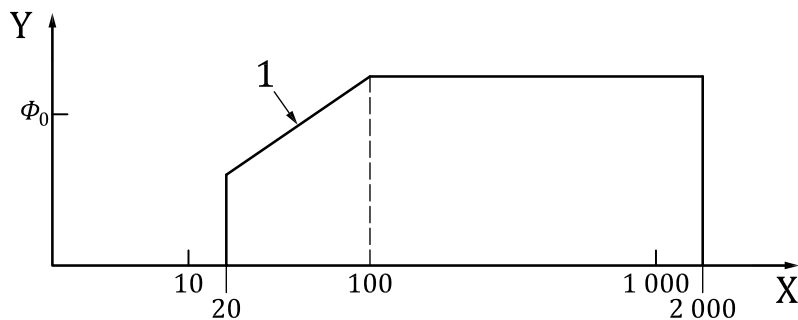


Figure 2 — Typical curves for electrodynamic vibration generators

In the case of random vibration, the rated force is defined in terms of the root-mean-square acceleration achievable while running the acceleration power spectral density $\Phi_a(f)$, in (m/s²)²/Hz, given by Formula (1) (see ISO 5344):

$$\begin{aligned}
 \Phi_a(f) &= 0, & f < 20 \text{ Hz} \\
 \Phi_a(f) &= (f/100)^2 \Phi_0, & 20 \text{ Hz} \leq f < 100 \text{ Hz} \\
 \Phi_a(f) &= \Phi_0, & 100 \text{ Hz} \leq f < 2\,000 \text{ Hz} \\
 \Phi_a(f) &< \Phi_0 (2\,000/f)^4 \text{ or } 10^{-4} \Phi_0, & f > 2\,000 \text{ Hz}
 \end{aligned} \tag{1}$$

The corresponding curve of acceleration power spectral density for electrodynamic vibration generator is shown in Figure 3.

**Key**

- X frequency, Hz
 Y acceleration power spectral density, Φ_a
 1 +12 dB/octave

Figure 3 — Shape of acceleration power spectral density for electrodynamic vibration generator (from ISO 5344)

If explicitly desired, a validation may be performed to ensure that the crest factor of the drive signal is at least 2,5 and the crest factor of the acceleration signal at the reference point is at least 3,0. Artificial reduction of crest factor (such as sigma clipping) should not be employed during validation testing.

5.3.2 Servo-hydraulic vibration generators

Table 2 gives typical parameters for servo-hydraulic vibration generators of a “traditional” design. Force ratings for Sine (peak) and Random (rms) performance are assumed to be the same, although this is not always the case. Manufacturers offer various series or steps of force ratings for the vibration generating system which will not line up exactly with Table 2. Significant deviation from the patterns established in the table may indicate a vibration generator designed for a special purpose. When a system is being purchased from a manufacturer, or being selected for usage from several systems of purchaser's own, it is recommended to use actual specification sheets.

The main features of servo-hydraulic vibration generators are the following:

- any type of excitation is possible;
- ease of control (manual and automatic);
- frequency range extended down to DC and limited at high frequencies to 800 Hz (typically not exceeding 100 Hz);
- high displacement, up to 200 mm; acceleration up to 1 000 m/s²; velocity up to 10 m/s (typically up to 2 m/s);
- very high force, up to 10 MN (typically up to 1 MN);
- very large permissible load, up to several tonnes;
- low transverse motion, about 5 % to 10 %;
- absence of a magnetic field in the area of the table;
- low sensitivity to load misalignment;
- increased harmonic distortion at the low-frequency range (below the natural frequency of the actuator), up to 15 % and more;
- low harmonic distortion at frequencies above the natural frequency, in the order of 5 %.

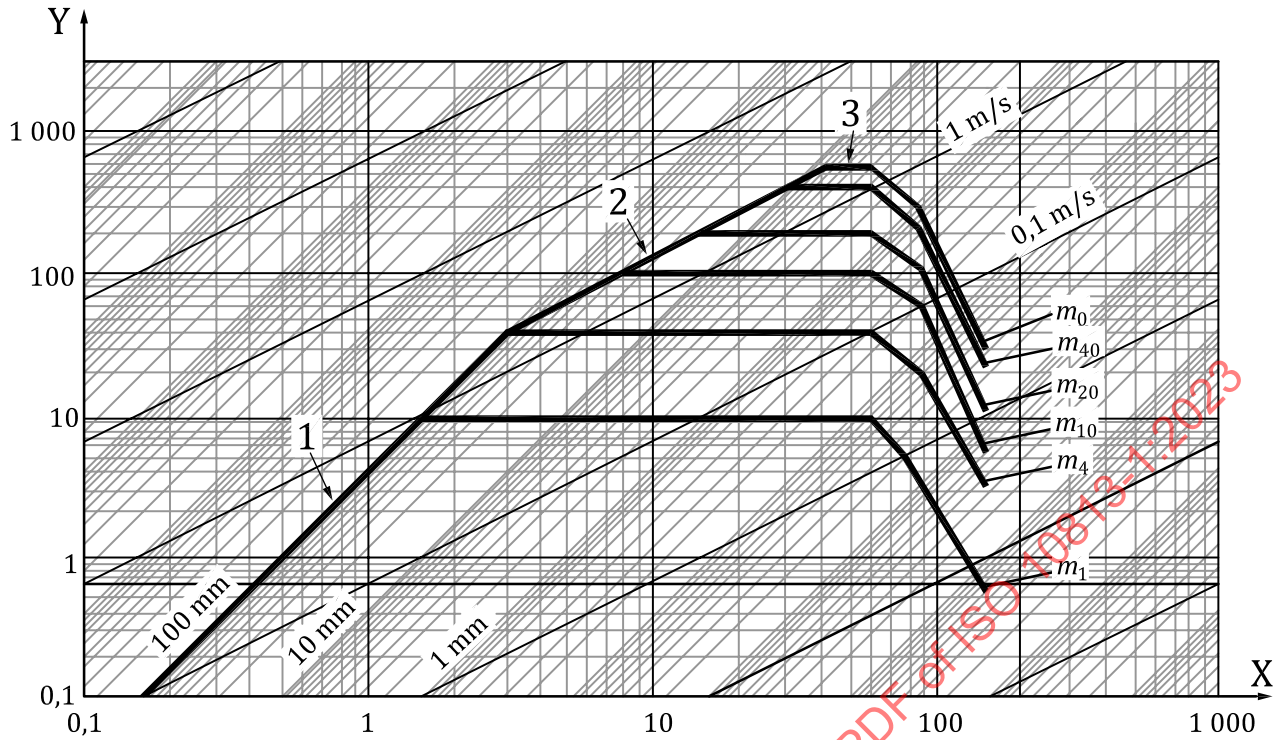
Table 2 — Typical parameters for servo-hydraulic vibration generators

Rated force	Frequency range	Maximum displacement	Maximum velocity	Maximum acceleration	Mass of moving system
N	Hz	mm	m/s	m/s ²	kg
5 000	0,1 to 140	±100	2,0	1 000	5
8 000	0,1 to 100	±100	2,0	1 000	8
10 000	0,1 to 100	±100	2,0	1 000	10
15 000	0,1 to 100	±100	2,0	1 000	15
20 000	0,1 to 100	±100	2,0	1 000	20
30 000	0,1 to 60	±100	2,0	1 000	30
50 000	0,1 to 60	±100	2,0	1 000	50
100 000	0,1 to 60	±100	1,7	600	167
200 000	0,1 to 60	±100	0,8	300	667
500 000	0,1 to 30	±100	0,3	100	5 000
1 000 000	0,1 to 30	±100	0,1	30	33 333

NOTE 1 Upper limits for different vibration parameters generally cannot be achieved simultaneously for extended durations. See ISO 8626 for more details on servo-hydraulic vibration system specification.

NOTE 2 When the rated force is an r.m.s. value in random mode, this value is limited by electrical power draw and heat dissipation. It is understood that higher instantaneous peak forces (often six times the r.m.s. level or more) will occur. The manufacturer is responsible for designing components to survive these brief transients. This does not affect the selection process.

Curves for servo-hydraulic system characteristics are presented in [Figure 4](#). They are the same as those for electrodynamic vibration generators excluding the sharp fall in force (acceleration) at high frequencies.



- Key**
- X frequency, Hz
 - Y acceleration, m/s²
 - 1 displacement limit
 - 2 velocity limit
 - 3 maximum acceleration

Figure 4 — Typical curves for servo-hydraulic vibration generators

In the case of random vibration, the rated values are similar to those for electrodynamic vibration generators. The rated force is defined in terms of the acceleration power spectral density $\Phi_a(f)$, in (m/s²)²/Hz, or the displacement power spectral density $\Theta(f)$, in m²/Hz, by [Formula \(2\)](#):

$$\begin{aligned}
 &\Phi_a(f)=0, & \Theta(f)=0, & f < f_1 \\
 &\Phi_a(f)=\left[f^4/(f_2f_3)^2\right]\Phi_0, & \Theta(f)=\Theta_0, & f_1 \leq f < f_2 \\
 &\Phi_a(f)=\left(f^2/f_3^2\right)\Phi_0, & \Theta(f)=\left(f_2^2/f^2\right)\Theta_0, & f_2 \leq f < f_3 \\
 &\Phi_a(f)=\Phi_0, & \Theta(f)=\left[(f_3f_2)^2/f^4\right]\Theta_0, & f_3 \leq f < f_4 \\
 &\Phi_a(f)=\left(f_4^2/f^2\right)\Phi_0, & \Theta(f)=\left[(f_4f_3f_2)^2/f^6\right]\Theta_0, & f_4 \leq f < f_5 \\
 &\Phi_a(f)=\left[(f_4f_5)^2/f^4\right]\Phi_0, & \Theta(f)=\left[(f_5f_4f_3f_2)^2/f^8\right]\Theta_0, & f_5 \leq f < f_6 \\
 &\Phi_a(f)=0, & \Theta(f)=0, & f \geq f_6
 \end{aligned} \tag{2}$$

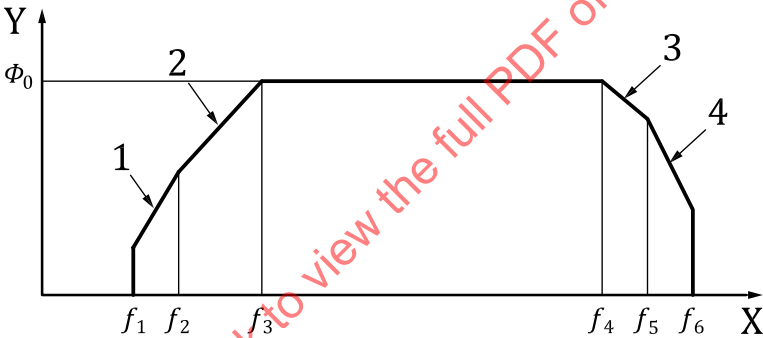
where

- f_1 is the lower limit of the frequency range;
- f_2 is the cross-over frequency between constant displacement and constant velocity ranges;
- f_3 is the cross-over frequency between constant velocity and constant acceleration ranges;
- f_4 is the frequency of the first power spectral density limitation;
- f_5 is the frequency of the second power spectral density limitation;
- f_6 is the upper limit of the frequency range.

$$\Theta_0 = \Phi_0 \frac{1}{(2\pi f_2)^4}$$

The curve of the acceleration power spectral density for servo-hydraulic vibration generators is shown in [Figure 5](#).

The crest factor should not be less than 3.



Key

- X frequency, f , Hz
- Y acceleration power spectral density, Φ_a
- 1 +24 dB/octave
- 2 +12 dB/octave
- 3 -12 dB/octave
- 4 -24 dB/octave

Figure 5 — Shape of acceleration power spectral density for servo-hydraulic vibration generator (from ISO 8626)

5.3.3 Mechanical vibration generators

Typical parameters for mechanical vibration generators are given in [Table 3](#).

Table 3 — Typical parameters for mechanical vibration generators

Rated load kg	Frequency range Hz	Maximum displacement mm	Maximum acceleration m/s ²	Mass of moving system kg
5	5 to 100	±5	150	0,33
25	5 to 100	±5	150	1,66
50	5 to 100	±5	100	5,0
100	5 to 80	±3	100	10,0
250	5 to 80	±3	50	50,0
500	5 to 80	±3	50	100,0
1 000	5 to 80	±2,5	50	200,0

The main features of mechanical vibration generators are the following:

- possibility of sinusoidal excitation at fixed frequencies only;
- difficulty of control;
- small frequency range, 0,1 Hz to 300 Hz (typically 5 Hz to 100 Hz);
- low displacement, typically in the order of 5 mm; in the infrasonic range it may be increased up to 100 mm;
- low acceleration, up to 300 m/s² (typically not exceeding 150 m/s²);
- permissible load up to several tonnes (typically tens or hundreds of kilograms);
- increased harmonic distortion (about 15 % to 25 %, background narrow-band noise at high frequencies);
- increased transverse motion in the order of 25 %;
- absence of a magnetic field in the area of the table;
- simple design;
- low cost;
- displacement (velocity, acceleration) does not depend on the mass of the load;
- displacement does not depend on the frequency.

5.4 Comparison between electrodynamic, servo-hydraulic and mechanical vibration generators

Characteristic limits for vibration generators of these three types are given in [Table 4](#).

Table 4 — Comparison between electrodynamic, servo-hydraulic and mechanical vibration generators

Parameter	Type of vibration generator		
	Electrodynamic	Servo-hydraulic	Mechanical
Rated force, kN	0,1 to 400	5 to 10 000	0,5 to 150
Load, kg	1 000	5 000	500
Displacement, mm	±50, typically ±38 or ±25	±200, typically ±100	±100, typically ±5
Acceleration (without load), m/s ²	1 500, typically 1 000	1 000, typically 100	300, typically 100
Frequency, Hz	0,5 to 20 000, typically 5 to 5 000	0 to 800, typically 1 to 200	0,1 to 300, typically 5 to 100
Harmonic distortion, %	2 to 5; up to 25 and more at some frequencies	15 below natural frequency 5 above natural frequency	20 and more
Transverse motion, %	5 to 10, up to 25 and more at some frequencies	5 to 10	20 and more
Uniformity over table, %	5 to 10, up to 25 and more at some frequencies	2 to 5	5 to 10
Magnetic field at test article, T	0,001 and more	No	No
Sensitivity to load misalignment	Significant	Low	Significant
Types of excitation	All	All	At fixed frequencies
Cost	High	High	Low

[Figure 6](#) presents ranges of their typical use.

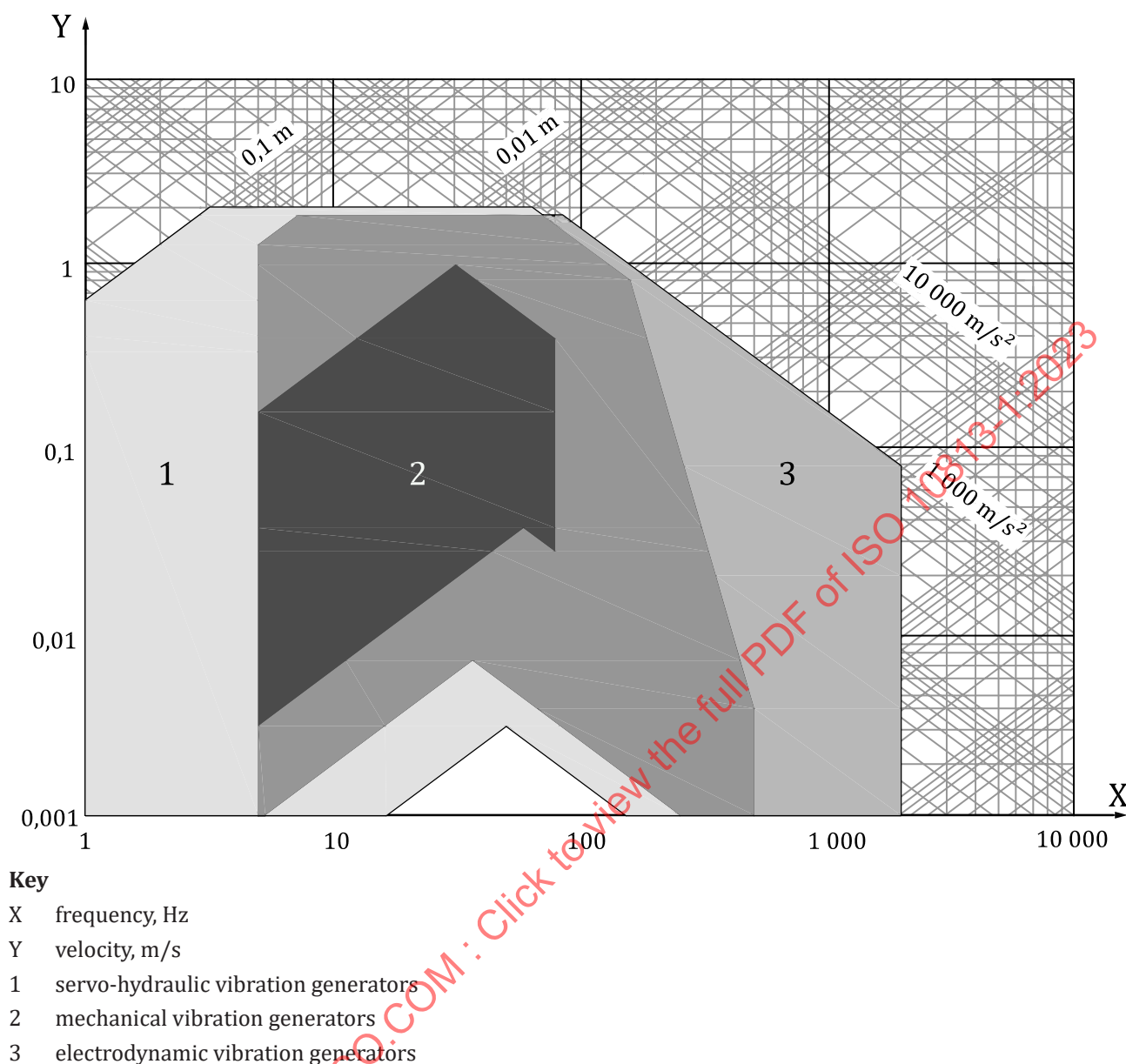


Figure 6 — Comparison of electrodynamic, servo-hydraulic and mechanical vibration generating systems for typical application

6 Recommendations for the selection of vibration generators

6.1 Selection of type

According to test requirements, the vibration generator should be capable of generating vibration with specified parameters: displacement, d , velocity, v , or acceleration, a , at a specified frequency, f ; and acceleration power spectral density Φ_a over a frequency range f_L to f_H under loading by mass m_s , with permissible harmonic distortion, transverse motion, and uniformity of the table motion and magnetic field in the area of the table.

The type of vibration generator should be selected on the basis of these requirements as well as from features and characteristics of vibration generators described in [5.3](#) and [5.4](#).

At high frequencies (above 1 000 Hz), only an electrodynamic vibration generator may be used; at low frequencies (infrasonic range), a servo-hydraulic vibration generator is preferable; at middle frequencies (up to 200 Hz), all three types may be used.

For significant displacement (more than 100 mm peak-to-peak), a servo-hydraulic generator is usually applied.

Random vibration can be reproduced by both electrodynamic and servo-hydraulic vibration generators.

For extremely large loads (more than 1 000 kg), electrodynamic or servo-hydraulic vibration generators may be used, but external load support may be necessary.

If there is a possibility of using vibration generators of various types, one should consider the accuracy characteristics, such as permissible harmonic distortion, transverse motion, and the uniformity of table motion and magnetic field in the area of the load. If the magnetic field generated by an electrodynamic vibration generator causes undesired effects for the test article, testing may require an alternative vibration source unless additional protection (such as shielding or active degauss) can be provided. If no severe restrictions are imposed on accuracy characteristics, a mechanical vibration generator may be used in order to bring down the testing cost.

A servo-hydraulic vibration generator should be used at low frequencies if high displacement and high force are to be developed.

An electrodynamic vibration generator should be used during testing within a wide frequency band with low displacement and high acceleration. Its features are low signal distortion under sinusoidal excitation and precise control under random excitation.

For random tests, both electrodynamic and servo-hydraulic vibration generators may be used. The rated characteristics should exceed those that are acceptable for the test conditions.

6.2 Selection of the model

6.2.1 General

The major requirement for a vibration generating system defined by test conditions is its capability to produce oscillation of the mass, m_s , at the frequency, f , or within frequency range f_L to f_H with maximum displacement, $d_{\max}$, or with maximum velocity, $v_{\max}$, or with maximum acceleration, $a_{\max}$, or with the specified acceleration power spectral density $\Phi_a(f)$.

The procedure for the definition of system performance meeting the specified test conditions is given in 6.2.2 to 6.2.7. The defined performance determines the specific model of vibration generator which should be selected.

All calculations to be performed in accordance with [Formulae \(3\) to \(15\)](#) demand numerical data, which are supplied by the manufacturer according to ISO 5344 or ISO 8626, depending on which one is relevant.

6.2.2 Frequency range

The frequency range of a vibration test system should be greater than the frequency range of the test. The defined frequency range of a system is governed by several factors.

In the case of low frequency operation, factors such as vibration and shock isolation systems, the low frequency response of the amplifier and even the low frequency response of the accelerometer should be considered.

The stated upper frequency of operation of a vibration test system is normally based on 1,1 times the resonant frequency of the moving element. This figure is a generalization since the dynamics of the moving element and the product under test will be such that extreme vibration in an axis other than that desired will occur. The term “usable” will be found in system performance specifications, which

implies that the end result could be unpredictable and the suitability of running tests at the upper frequency range can only be determined by the end user. In general terms, the upper frequency of operation depends on the dynamics of the fixture, auxiliary table and the device under test.

6.2.3 Maximum acceleration

The maximum acceleration of a vibration test system determines the maximum acceleration level which, under continuous operation, would not significantly fatigue or overstress the moving element. This maximum level will ensure that the moving element will have a reasonable working life, which is proportional to the period of use and the magnitude of operation.

Since the maximum acceleration and force of a system are also proportional to the current provided by the amplifier, most systems are designed so that the amplifier will provide only enough current to produce the maximum system force and acceleration. There are situations, however, where the current available from the amplifier is much greater than that required to drive the system to its maximum force and acceleration. If this were the case, it would be possible to drive the system beyond the stated limits, which would be inadvisable as it would shorten the working life of the vibrator.

6.2.4 Force

The force, F , to be developed by a vibration generator is determined from the total mass, m_z , to be moved, which is calculated by [Formula \(3\)](#):

$$m_z = m_e + m_s \quad (3)$$

and the specified acceleration, a , in the test specification, where m_e is the mass of the moving element of the vibration generator (see [Tables 1](#) and [2](#)), and m_s is the mass of the specimen, including the test fixture and auxiliary table, by [Formula \(4\)](#):

$$F = m_z a = (m_e + m_s) a \quad (4)$$

For sinusoidal test conditions, the peak force, F , is expressed as F_s and calculated by [Formula \(5\)](#):

$$F_s = (m_e + m_s) a_{\max} \quad (5)$$

where $a_{\max}$ is the highest acceleration stated in the specification.

For a random test, the r.m.s. force, F_r , is expressed through r.m.s. acceleration, a_r , by [Formula \(6\)](#):

$$F_r = (m_e + m_s) a_r \quad (6)$$

where

$$a_r = [\Phi_a(f) \Delta f]^{1/2} \quad (7)$$

$\Phi_a(f)$ is the acceleration power spectral density in Δf .

[Formula \(7\)](#) was obtained assuming that, for the purpose of testing, $\Phi_a(f)$ is usually specified as a flat-top curve of the rectangular shape over the operational frequency range Δf .

6.2.5 Mass of the moving element

The mass of the moving element, m_e , is specified by the manufacturer, but can be estimated from [Tables 1](#), [2](#) and [3](#) depending on the force rating and load capacity required.

If not provided, m_e , can be estimated by [Formula \(8\)](#):

$$m_e = m_s \frac{a_{\text{rate}}}{a_0 - a_{\text{rate}}} \quad (8)$$

where a_0 is the maximum acceleration of the vibration generator without the load ($m_s = 0$) and $a_{\text{rate}} = F_{\text{rate}} / m_s$ is the acceleration derived from a sine rating with load, m_s .

6.2.6 Size of the moving element

Though [Tables 1, 2](#) and [3](#) do not list a typical footprint of the moving element, it is important to consider whether or not the specimen, including the test fixture, will anchor securely to the provided attachments without overhanging. If not, a head expander or additional fixturing will be required, further increasing the specimen mass, m_s . This will in turn increase the required force rating according to [Formula \(1\)](#). In the case of horizontal motion, the size of the auxiliary table must be determined and its mass included in m_s .

6.2.7 Rated travel

Most electrodynamic systems are equipped with a compensating (often pneumatic) device to support the moving load. In this case, the rated travel, S_L , of the moving element, defined as its maximum down travel with reference to the equilibrium position for the unloaded system, is equal to the maximum displacement amplitude, d_{max} . For systems in which an adequate compensating device is absent, S_L is calculated by [Formula \(9\)](#):

$$S_L > d_{\text{max}} + l_{\text{st}} = d_{\text{max}} + \frac{g}{4\pi^2 f_s^2} \quad (9)$$

where

l_{st} is the static displacement with reference to the equilibrium position under loading;

g is the standard acceleration due to the earth's gravity;

f_s is the natural frequency of the moving system loaded by the mass m_s .

[Formula \(9\)](#) may be rewritten in terms of the natural frequency, f_e , of the unloaded moving system as [Formula \(10\)](#):

$$S_L > d_{\text{max}} + l_{\text{st}} = d_{\text{max}} + l_1 + l_2 = d_{\text{max}} + \frac{\left(1 + \frac{m_s}{m_e}\right)g}{4\pi^2 f_e^2} \quad (10)$$

or [Formula \(11\)](#):

$$S_L > d_{\text{max}} + \frac{a_0}{a_{\text{max}}} \frac{g}{4\pi^2 f_e^2} \quad (11)$$

where

l_1 is the static displacement of the moving system under its own mass;

l_2 is the additional static displacement of the moving system when loaded.

For a servo-hydraulic system, the rated travel, S_L , defined as the rated piston stroke, depends on the peak-to-peak displacement according to [Formula \(12\)](#):

$$S_L > 2d_{\max} + \frac{2a_{\max}}{4\pi^2 f_L^2} \quad (12)$$

where f_L is the lower limit of the frequency range.

All the above is valid only in the case of rigid mounting of a vibration generator to a base. When vibration isolators are used between the body of the vibration generator and the ground, the rated travel should be reduced by the value of the body displacement, d_b , calculated in assumption that the body moves out of phase with the load by [Formula \(13\)](#):

$$d_b = d_{\max} \frac{m_z}{m_b} \quad (13)$$

where m_z is the total moving mass and m_b is the body mass.

NOTE If the test frequency range passes through the isolation system resonant frequency and the total moving mass is heavy, the body will vibrate vigorously, making the test extremely difficult to control.

6.2.8 Maximum velocity

For electrodynamic and servo-hydraulic vibration generators, the maximum velocity, $v_{\max}$, specified by the manufacturer or estimated from [Tables 1](#) and [2](#) takes precedence over the specified displacement, $d_{\max}$, above some cross-over frequency, f_{dv} . The maximum acceleration, $a_{\max}$ (given a particular load) takes precedence over the specified velocity at some higher cross-over frequency, f_{va} . These cross-over frequencies can be calculated by [Formulae \(14\)](#) and [\(15\)](#), respectively:

$$f_{dv} = \frac{v_{\max}}{2\pi d_{\max}} \quad (14)$$

$$f_{va} = \frac{a_{\max}}{2\pi v_{\max}} \quad (15)$$

Usually, the maximum velocity for electrodynamic and servo-hydraulic systems is 2 m/s. Outside of impulsive shocks (which will be dealt with in other relevant parts of ISO 10813), this is more than adequate for most tests.

For tests which specify a maximum velocity segment, the rating of the vibration generator may be compared directly to the test requirement. In practice, the test which drives generator selection is often specified only in terms of a peak displacement, $d_{\max}$ and a peak acceleration, $a_{\max}$. In this case, the maximum velocity requirement can be calculated by [Formula \(16\)](#):

$$v_{\max} = \sqrt{a_{\max} d_{\max}} \quad (16)$$

In this case, $f_{dv} = f_{va}$ for the test and the maximum test velocity occurs only at this single cross-over frequency.

NOTE To find and compare requirements of displacement, d , velocity, v , and acceleration $a = F/m$ it is advisable to use a nomogram displaying these quantities as functions of frequency.

[Formulae \(14\)](#) to [\(16\)](#) refer to sine tests which sweep through a range of frequencies. For random tests specified by a power spectral density, maximum velocity is rarely an issue.

6.3 Selection of components

6.3.1 General

In order to provide the required characteristics of the vibration generating system, the components of the system other than the vibration generator (such as power amplifier for electrodynamic systems, and hydraulic power system and servo valves for servo-hydraulic systems) should be selected properly.

All calculations to be performed in accordance with [Formulae \(17\)](#) to [\(28\)](#) demand numerical data or graphs like [Figures 7](#) to [9](#) which are supplied by the manufacturer.

6.3.2 Selection of power amplifier

6.3.2.1 General

A power amplifier is used to supply an electrodynamic vibration generator with current. Its output, required to develop a force from tens of newtons to hundreds of kilonewtons, varies from tens of watts to hundreds of kilowatts.

A power amplifier is loaded by the complex impedance of the moving coil varying with frequency.

A power amplifier should meet the following:

- output power in accordance with the specified force of the vibration generator at the specified frequency range;
- low distortion (less than 1 % to 2 %);
- low background noise (signal-to-noise ratio not less than 50 dB);
- continuous operation (typically for 8 h);
- stable shut down in the case of failure of the moving system coil.

6.3.2.2 Output voltage

The output voltage, U , needed to drive the moving element of a direct-coupled vibration generator is determined by [Formula \(17\)](#):

$$U = E_c + (R + j\omega L)I \quad (17)$$

where

E_c is the back-electromotive force, $E_c = Blv$;

B is the magnetic flux density in the operating gap of the moving coil;

l is the total length of the moving coil wire;

v is the velocity of the moving coil;

R is the moving coil resistance;

ω is the angular frequency;

L is the moving coil inductance;

I is the current in the moving coil.

The maximum voltage requirement for sinusoidal vibration typically occurs at the cross-over frequency between the ranges of constant velocity to constant acceleration (typically in the range of 30 Hz to 100 Hz). However, the voltage needed to run random vibration at high levels may, depending on the geometry of the vibration generator, exceed the voltage requirement for sinusoidal testing.

NOTE At 30 Hz: $v_{\max} = 2 \text{ m/s}$, $a_{\max} = 377 \text{ m/s}^2$; at 80 Hz: $v_{\max} = 2 \text{ m/s}$, $a_{\max} = 1\,000 \text{ m/s}^2$; at 100 Hz: $v_{\max} = 2 \text{ m/s}$, $a_{\max} = 1\,250 \text{ m/s}^2$.

Since several of the parameters in [Formula \(17\)](#) are generally unavailable, the manufacturer should be consulted to determine the minimum voltage capability of a third-party amplifier. Alternatively, ISO 5344 gives guidance on the use of an over-sized amplifier to experimentally determine the minimum required output voltage.

The output voltage, U , is limited by the supply voltage for elements of the output stage of the power amplifier. For transistors, the limiting voltage may be anywhere from 50 V to 600 V. This voltage 'rail' is maintained by capacitors, which must be continuously recharged by a rectifier. It is important that the power amplifier be capable of maintaining the necessary output voltage while outputting the required current.

6.3.2.3 Output power

The apparent power of the power amplifier, S , in volt amps, is determined by [Formula \(18\)](#):

$$S = U \cdot I \quad (18)$$

where

U is the voltage at moving coil terminals, in volts;

I is the current traversing the moving coil, in amps.

When estimating power at velocity-acceleration cross-over, $E_c \gg (R + j\omega L)I$. Thus, the voltage, U , can be approximately evaluated by [Formula \(19\)](#):

$$U \approx E_c = B \cdot l \cdot v = \frac{F \cdot v}{I} \quad (19)$$

where F is the exciting force.

Then the required maximum apparent power is determined by [Formula \(20\)](#):

$$S = U \cdot I = F \cdot v \quad (20)$$

All the parameters in [Formula \(20\)](#) are expressed in terms of r.m.s. Usually, the force and velocity developed by a vibration generator are expressed in terms of peak values, F_{peak} and v_{peak} respectively. In this case S can be approximately estimated by [Formula \(21\)](#):

$$S = 0,5 F_{\text{peak}} \cdot v_{\text{peak}} \quad (21)$$

Because amplifiers are often rated by output power in VA or kVA, [Formula \(21\)](#) can be used to determine if a proposed amplifier is too small. In practice an amplifier capable of outputting the necessary voltage, U , and current, I , across the desired frequency range will be rated for more output power than specified by [Formula \(21\)](#).

6.3.2.4 Coil current

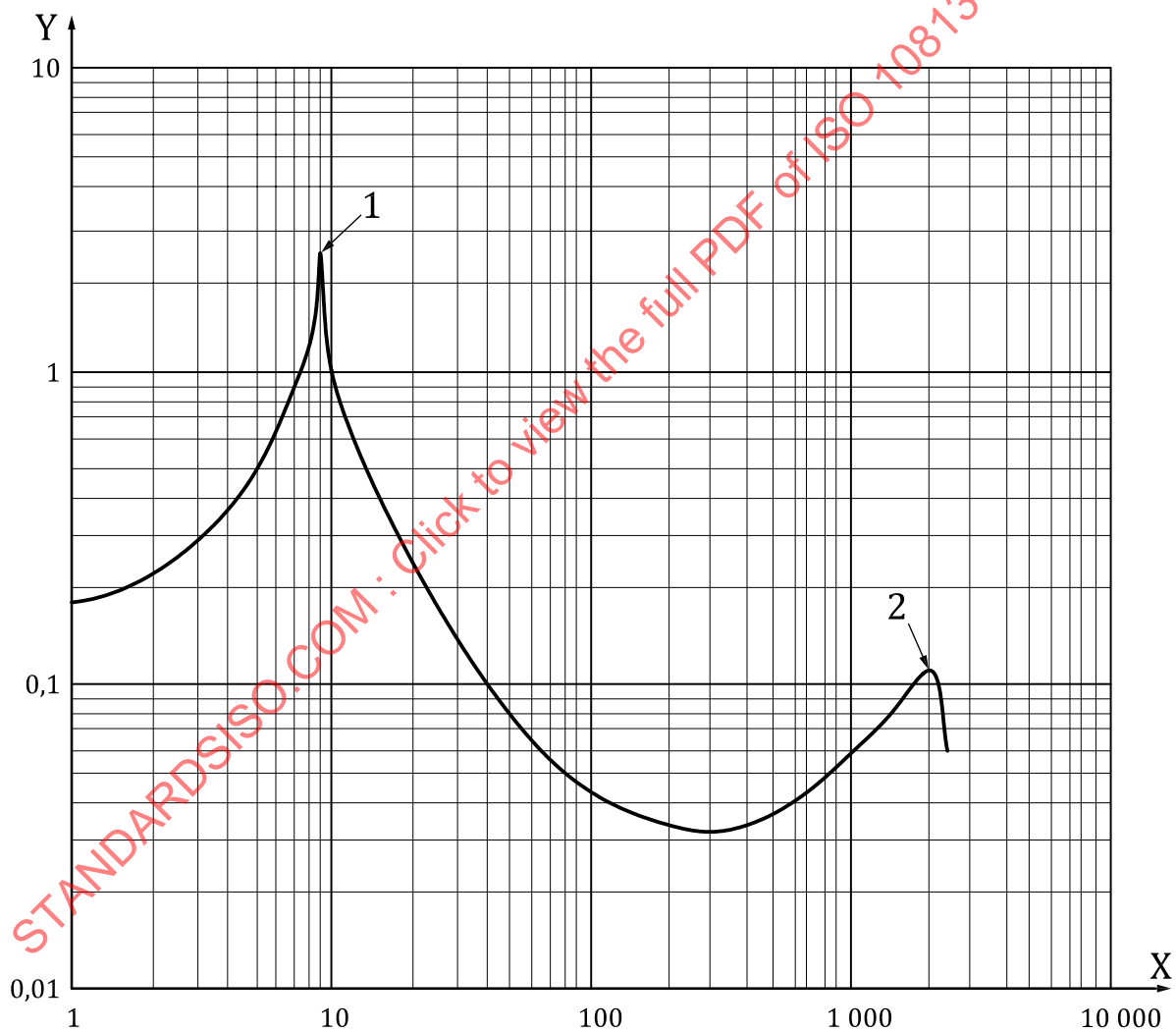
The current in the moving coil is determined by [Formula \(22\)](#):

$$I = \frac{F}{l \cdot B} \quad (22)$$

The manufacturer should provide guidance with regard to the minimum current required to run all specified tests. Alternatively, ISO 5344 gives guidance on the use of an over-sized amplifier to experimentally determine the minimum required coil current.

6.3.2.5 Amplifier load

Output power, as well as output current and voltage, depends on the amplifier load (e.g. moving coil), whose impedance varies with frequency as shown in [Figure 7](#).



Key

- X frequency, Hz
- Y impedance, Ω
- 1 mechanical resonance of suspension
- 2 first resonance of moving coil assembly

Figure 7 — Example of the amplifier load impedance

Load impedance, Z , is determined by the electric parameters of the coil and by its motion according to [Formula \(23\)](#):

$$Z = \frac{U}{I} = R_c + j\omega L_c + Z_m \quad (23)$$

where

R_c is the resistive load of the coil;

L_c is the inductive load of the coil;

ω is the angular frequency;

Z_m is the additional capacitance load due to coil motion,

$$Z_m = 0,5F_{\text{peak}}v_{\text{peak}}^2 / I;$$

F is the force;

v is the coil velocity;

I is the current.

At low frequencies in the vicinity of DC, the coil impedance approaches the resistive load R_c . The inductive component rises with frequency. The impedance comes to a maximum at the frequency f_s of the mechanical resonance of the coil suspension due to the back-electromotive force. This frequency falls within 3 Hz to 4 Hz for pneumatic suspension systems or 7 Hz to 60 Hz for elastic suspension. At the frequency of so-called electromechanical resonance (200 Hz to 400 Hz), the load becomes resistive again and equals approximately R_c . When the frequency increases further, the coil impedance rises again by the inductive component.

6.3.2.6 Recommendations for selection of amplifier

Typical curves for impedance, Z , current, I and voltage, U against frequency are shown in [Figure 8](#). Maximum output power is reached at the corner frequency, f_2 , of change from constant velocity to constant acceleration.

In order to use a power amplifier in an efficient manner, its parameters should be matched with the vibration generator impedance by a matching transformer. Modern transistor amplifiers usually need no matching transformers and may be connected to electrodynamic vibration generators directly.

The user may select a power amplifier from two main types: linear and switching. The latter exhibits a higher efficiency (85 % to 90 % against 40 % to 55 % for a linear power amplifier of the same output capability) and this significantly reduces operational costs, particularly for large amplifiers with power of a few tens of kilowatts.

6.3.2.7 Field coil power supply

With the exception of small vibration exciters made of permanent magnets, electrodynamic vibration generators rely on powered coils embedded within the generator body to supply the DC magnetic field in the operating gap. Rated performance assumes a certain current supplied to these coils. The impedance of the field coils generally increases with temperature. Care should be taken to ensure that any third-party power supply used to drive the field coil(s) have adequate voltage overhead to maintain full current after long run times. The field coil power supply may require power which is a significant fraction of the amplifier power.

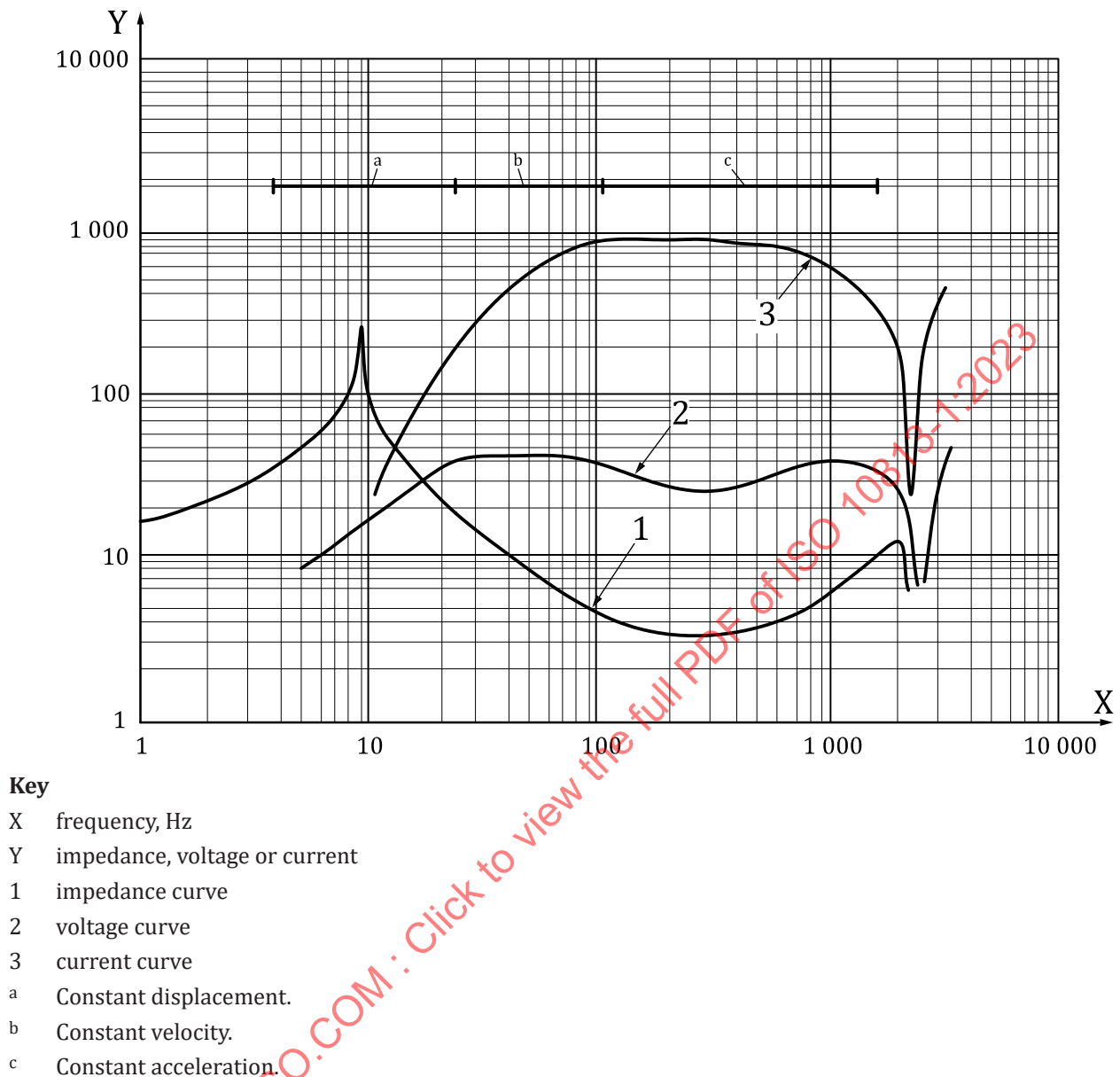
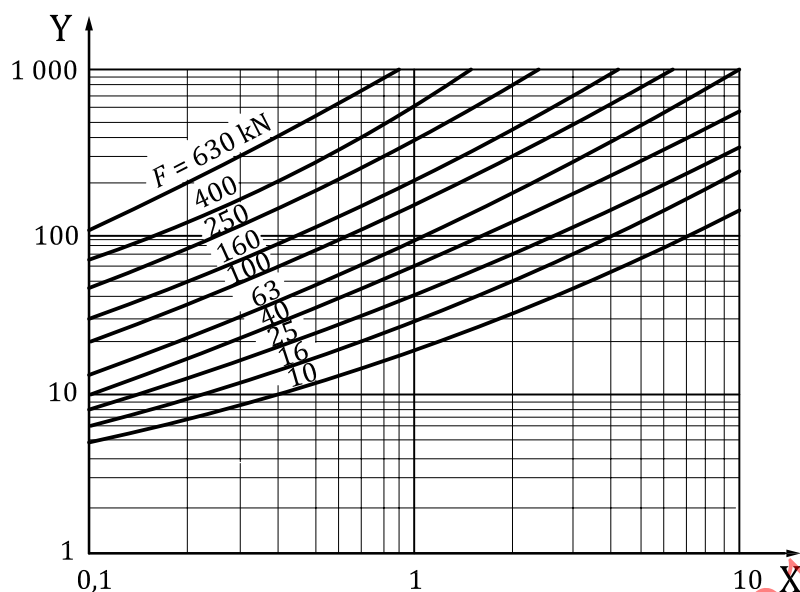


Figure 8 — Typical curves for impedance, current and voltage

6.3.3 Selection of hydraulic power supply

The main parameter of a hydraulic power supply is the mean oil flow, Q_m , which depends on the maximum vibration velocity $v_{\max,L}$ and rated force under a given supply pressure. Various models differ slightly in the form of such a dependence. Information about the relationship between the mentioned values for a specific model is usually provided by the manufacturer as tables or graphically (charts, nomograms, etc.). An example of a graphic presentation is shown in [Figure 9](#). Such a nomogram enables determination of the required mean oil flow, Q_m , on the basis of the velocity, v , and the rated force, F .



Key

X v_{peak} , m/s
Y Q_m , l/min

Figure 9 — Example of a nomogram for evaluation of the mean oil flow, Q_m , ($P = 28$ MPa)

A rough calculation of Q_m can be fulfilled as follows. Usually, the force (and, therefore, the pressure) is partially consumed to act against gravity. Thus, the maximum force, F and pressure, P are subdivided into dynamic, F_{dyn} , P_{dyn} and static, F_{st} , P_{st} components and are related by [Formula \(24\)](#) and [Formula \(25\)](#) respectively:

$$F_{\text{dyn}} = a_{\text{max}} \cdot m_z = S_p \cdot P_{\text{dyn}} \quad (24)$$

$$F_{\text{st}} = g \cdot m_z = S_p \cdot P_{\text{st}} \quad (25)$$

where

S_p is the useful cross-section of the piston of the actuator;

g is the standard acceleration due to the earth's gravity.