INTERNATIONAL STANDARD

ISO 10813-1

First edition
2004-08-01

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

Reference number
ISO 10813-1:2004(E)

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Published in Switzerland

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

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 10813-1 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration and shock*, Subcommittee SC 6, *Vibration and shock generating systems*.

ISO 10813 consists of the following parts, under the general title *Vibration generating machines — Guidance for selection*:

— *Part 1: Equipment for environmental testing*

Further parts are under preparation.

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 quite easily 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 part of ISO 10813 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 part of ISO 10813, 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 part of ISO 10813.

If the equipment is expected to be used for tests of different types, all possible applications should be considered when selecting. Later 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 part of ISO 10813 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 part of ISO 10813 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 part of ISO 10813 to assess the market environment.

Vibration generating machines — Guidance for selection —

Part 1: Equipment for environmental testing

1 Scope

This part of ISO 10813 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 Some examples are given in Annex A.

2 Normative references

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

ISO 2041, *Vibration and shock — Vocabulary*

ISO 5344, *Electrodynamic vibration generating systems — Performance characteristics*

ISO 8626, *Servo-hydraulic test equipment for generating vibration — Methods 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.

4 Requirements for vibration tests

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.

Strength 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. In these tests, vibration might cause mechanical damage (fatigue) and may be used to predict the lifetime of the object under vibration.

Endurance 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 the IEC (see [1] to [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 standardized by the IEC are given for information in Annex B.

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 constant 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 500 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 and test duration.

4.2.3 Wide-band random vibration

The wide-band random excitation, specified by the shape of 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 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.

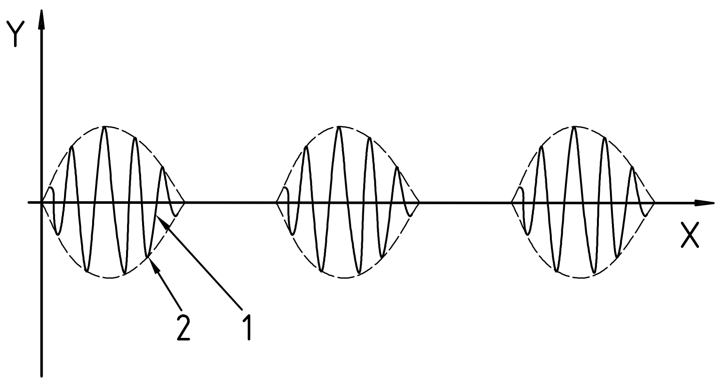
This test is applied to specimens which in service can be subjected to short-duration random-type dynamic forces induced, for example, by earthquakes, explosions or transportation.

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 preset 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.



Key

X time

Y vibration aplitude

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, servohydraulic, mechanical, electromagnetic, piezoelectric, magnetostrictive, etc. The most common types of vibration generators being used for environmental testing are electrodynamic, servohydraulic 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 magnetic field. The alternating magnetic field is produced by 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 Servohydraulic vibration generators

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

A vibration generating system including a servohydraulic vibration generator is called a servohydraulic 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 servohydraulic 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

Typical parameters for electrodynamic vibration generators are given in the Table 1. Manufacturers offer various series or steps of force ratings for the vibration generating system. 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 N	Output of the power amplifier VA	Frequency range Hz	Maximum displacement mm	Maximum velocity m/s	Maximum acceleration without load m/s^2	Maximum load kg	Mass of moving system kg
31,5	6,3	5 to 13 000	2,5	0,4	200	1,0	0,16
63	19	5 to 10 000	2,5	0,4	300	1,5	0,2
125	62,5	5 to 8 000	5,0	0,8	500	2,0	0,25
250	165	5 to 8 000	8,0	1,3	650	4,0	0,38
500	400	5 to 7 000	8,0	1,3	800	10,0	0,62
1 000	1 000	5 to 5 000	12,5	2,0	1 000	25,0	1,0
2 000	2 000	5 to 5 000	12,5	2,0	1 000	75,0	2,0
4 000	4 000	5 to 4 000	12,5	2,0	1 000	200,0	4,0
8 000	8 000	5 to 3 500	12,5	2,0	1 000	300,0	8,0
16 000	16 000	5 to 3 000	12,5	2,0	1 000	400,0	16,0
32 000	32 000	5 to 2 500	12,5	2,0	1 000	500,0	32,0
64 000	64 000	5 to 2 000	12,5	2,0	1 000	1 000,0	64,0
128 000	128 000	5 to 1 800	12,5	2,0	1 000	2 000,0	128,0
200 000	200 000	5 to 1 600	12,5	2,0	1 000	3 125,0	200,0
NOTE Upper limits for different vibration parameters cannot be achieved simultaneously.							

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), etc.;

— ease of control (manual and automatic);

— wide frequency range: 0,5 Hz up to 15 000 Hz (typically 5 to 5 000 Hz); in general, the lower the rated force the higher the upper limit of the frequency range;

— high displacement: up to ± 25 mm (typically up to ± 12,5 mm), and acceleration: up to 1 500 m/s^2 (typically up to 1 000 m/s^2);

— 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);

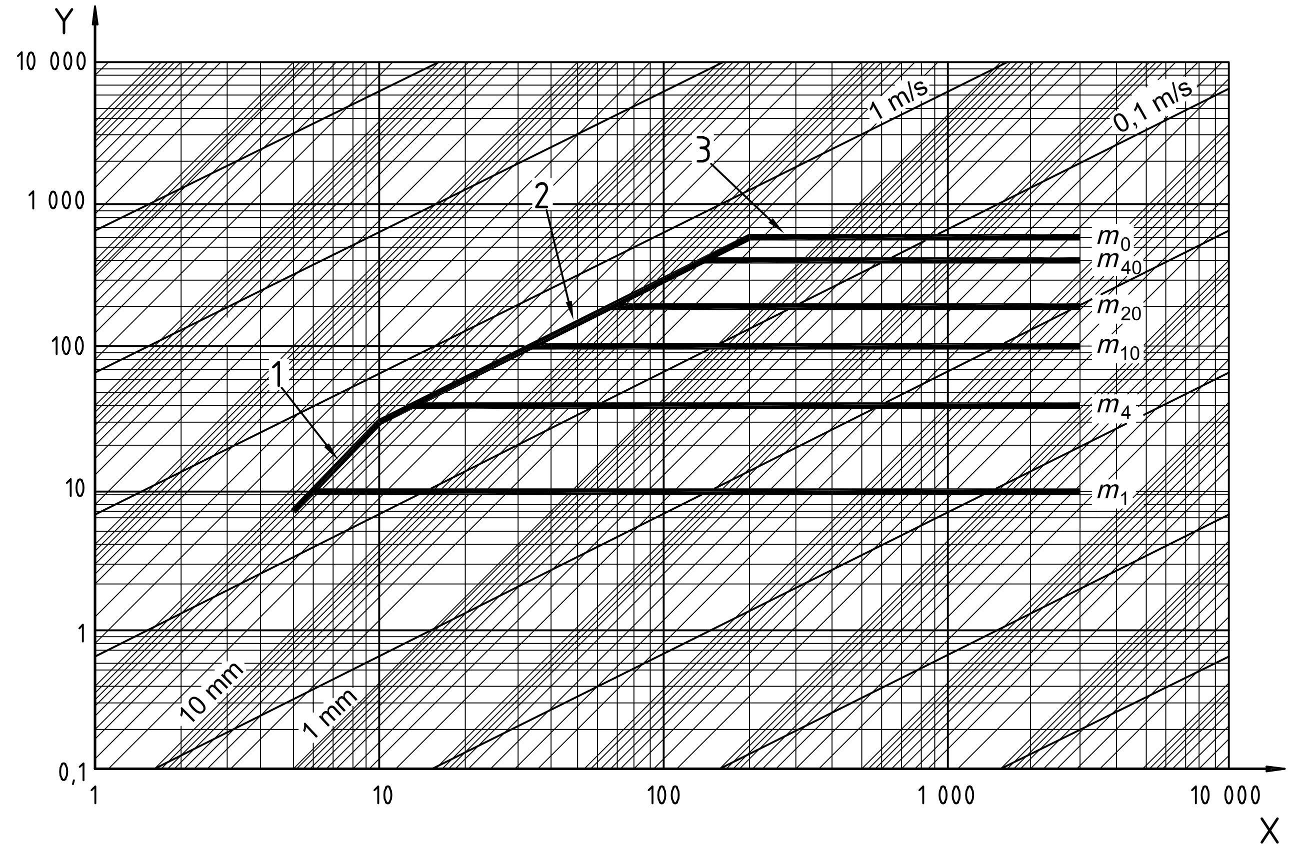
— low harmonic distortion: about 5 %, excluding frequency bands where distortion increases because of resonances between the vibration generator and the load;

— 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 0,001 T 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}, 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.



Key

X frequency, Hz

Y acceleration, m/s²

1 displacement limit

2 velocity limit

3 maximum acceleration

Figure 2 — Typical curves for electrodynamic vibration generators

In the case of random vibration, the rated force is defined in terms of the acceleration spectral power density $\Phi_a(f)$, in (m/s²)²/Hz (see ISO 5344):

$$\Phi_a(f) = 0 \qquad f < 20\ \text{Hz}$$

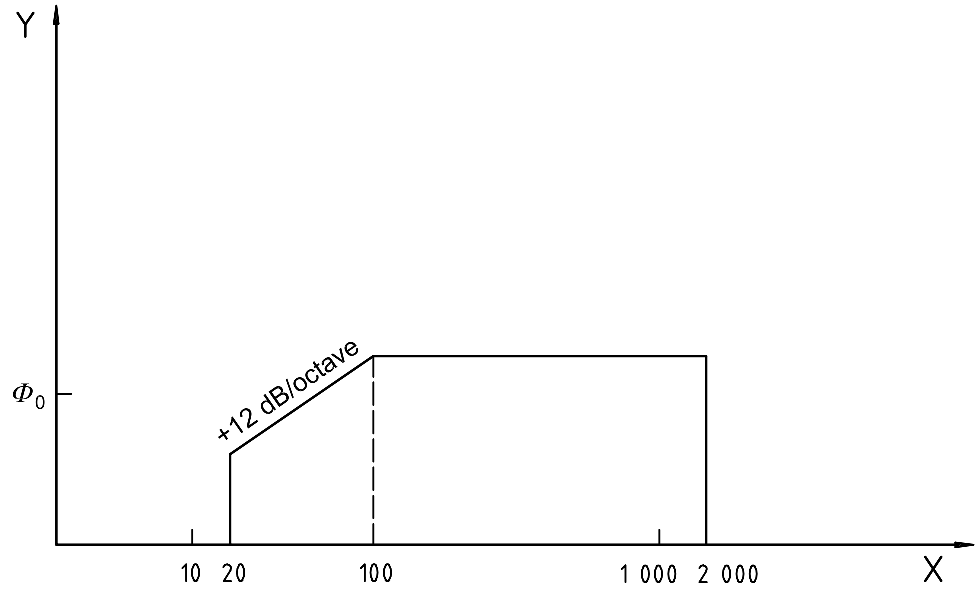
$$\Phi_a(f) = (f/100)^2\ \Phi_0 \qquad 20\ \text{Hz} < f < 100\ \text{Hz}$$

$$\Phi_a(f) = \Phi_0 \qquad 100\ \text{Hz} < f < 2\,000\ \text{Hz}$$

$$\Phi_a(f) < \Phi_0 (2\,000/f)^4 \quad \text{or} \quad 10^{-4}\Phi_0 \qquad f > 2\,000\ \text{Hz}$$

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

Crest factor should not be less than 3.



Key

X frequency, Hz

Y acceleration spectral power density, Φ_a

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

5.3.2 Servohydraulic vibration generators

Typical features for servohydraulic vibration generators are given in Table 2. Manufacturers offer various series or steps of force ratings for the vibration generating system. When a system is purchased from a manufacturer, or selected from several systems of purchaser's own, it is recommended to use actual specification sheets.

The main features of servohydraulic vibration generators are the following:

— any type of excitation is possible;

— ease of control (manual and automatic);

— frequency range extended down to d.c. 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/s^2; 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 servohydraulic vibration generators

Rated force N	**Frequency range** Hz	**Maximum displacement** mm	**Maximum velocity** m/s	**Maximum acceleration** m/s^2	**Mass of moving system** 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

Curves for servohydraulic 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.

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 spectral power density $\Phi_a(f)$, in (m/s^2)^2/Hz, or the displacement spectral power density $\Theta(f)$, in m^2/Hz:

$$\Phi_a(f) = 0 \qquad \Theta(f) = 0 \qquad f < f_1$$

$$\Phi_a(f) = \frac{f^4}{(f_2 f_3)^2}\Phi_0 \qquad \Theta(f) = \Theta_0 \qquad f_1 < f < f_2$$

$$\Phi_a(f) = \frac{f^2}{f_3^2}\Phi_0 \qquad \Theta(f) = \frac{f_2^2}{f^2}\Theta_0 \qquad f_2 < f < f_3$$

$$\Phi_a(f) = \Phi_0 \qquad \Theta(f) = \frac{(f_2 f_2)^2}{f^4} \qquad f_3 < f < f_4$$

$$\Phi_a(f) = \frac{f_4^2}{f^2}\Phi_0 \qquad \Theta(f) = \frac{(f_4 f_3 f_2)}{f^6}\Theta_0 \qquad f_4 < f < f_5$$

$$\Phi_a(f) = \frac{(f_4 f_5)^2}{f^4} \qquad \Theta(f) = \frac{(f_5 f_4 f_3 f_2)^2}{f^8}\Theta_0 \qquad f_5 < f < f_6$$

$$\Phi_a(f) = 0 \qquad \Theta(f) = 0 \qquad f > f_6$$

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;

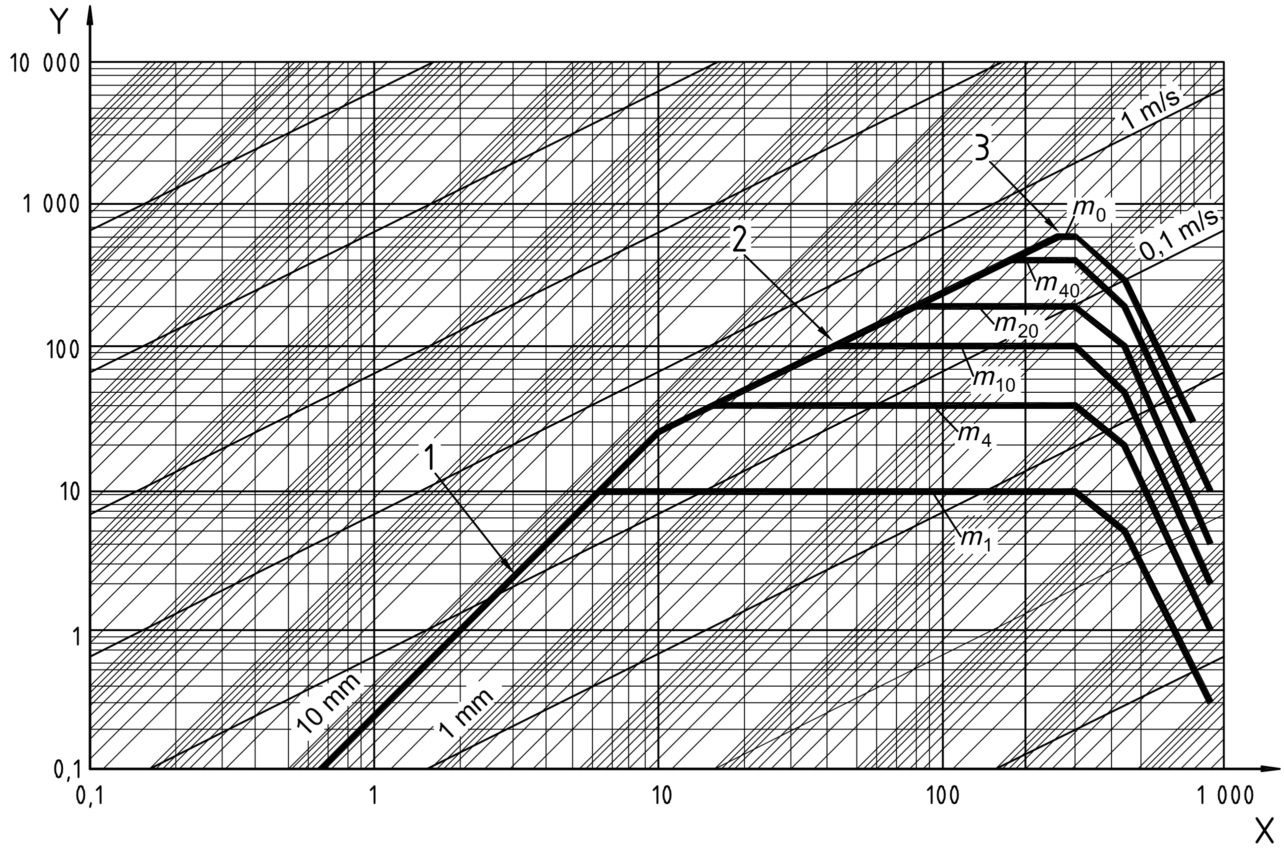
f_3 is the cross-over frequency between constant velocity and constant acceleration ranges;

f_4 is the frequency of the first spectral power density limitation;

f_5 is the frequency of the second spectral power density limitation;

f_6 is the upper limit of the frequency range.

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



Key

X frequency, Hz

Y acceleration, m/s^2

1 displacement limit

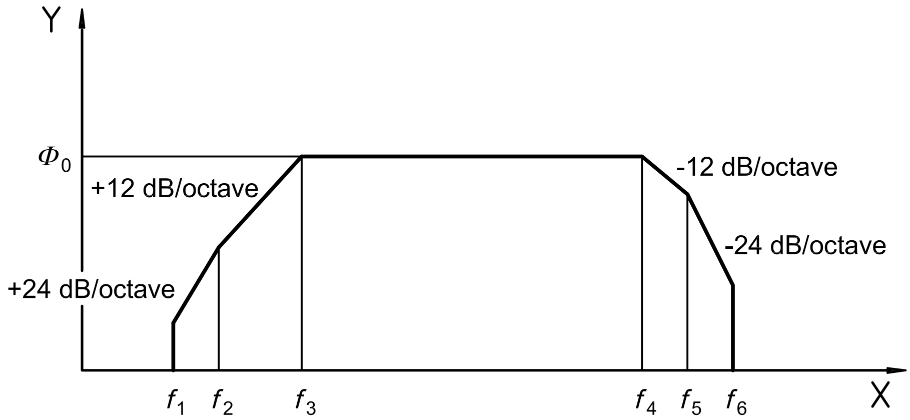
2 velocity limit

3 maximum acceleration

Figure 4 — Typical curves for servohydraulic vibration generators

The curve of the acceleration spectral power density for servohydraulic vibration generators is shown in Figure 5.

The crest factor should not be less than 3.



Key

X frequency, f, Hz

Y acceleration spectral power density, Φ_a

Figure 5 — Shape of acceleration power spectral density for servohydraulic 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	Max. displacement mm	Max. 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/s^2 (typically not exceeding 150 m/s^2);

— 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, servohydraulic and mechanical vibration generators

Characteristic limits for vibration generators of these three types are given in Table 4.

Table 4 — Comparison between electrodynamic, servohydraulic and mechanical vibration generators

Parameter	Type of vibration generator		
	Electrodynamic	Servohydraulic	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	± 25, typically ± 12,5	± 200, typically ± 50	± 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 induction, T	0,001 and more	No	No
Sensitivity to load misalignment	Significant	Low	Significant
Types of excitaton	All	All	At fixed frequencies
Cost	High	High	Low

Figure 6 presents ranges of their typical use.

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, acceleration a and acceleration spectral power density Φ_a at a fixed frequency f or 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 servohydraulic vibration generator is preferable; at middle frequencies (up to 200 Hz), all three types may be used.

For a significant displacement (more than ± 25 mm), a servohydraulic generator is usually applied.

Random vibration can be reproduced by both electrodynamic and servohydraulic vibration generators.

For extremely large loads (more than 1 000 kg), servohydraulic vibration generators are predominantly used.

If there is a possibility of using vibration generators of various types, one should take into account 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 a magnetic field is not allowed, this excludes practically or impedes greatly the application of an electrodynamic vibration generator, which otherwise would be preferable. However, 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 servohydraulic 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 servohydraulic vibration generators may be used. The rated characteristics should be stricter than 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 spectral density $\Phi_a\left(f\right)$.

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.

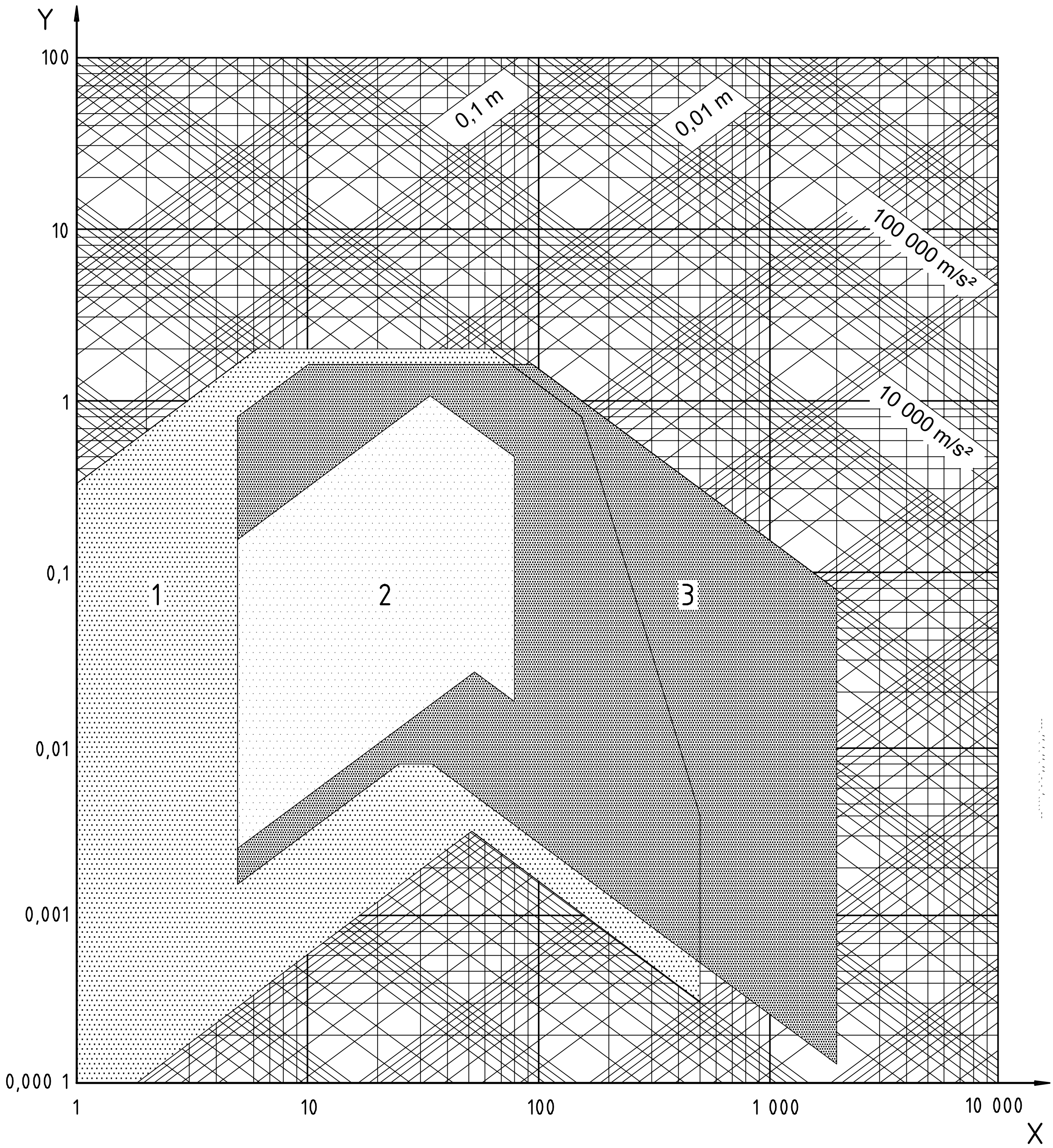
All calculations to be performed in accordance with Equations (1) to (10) demand numerical data, which should be supplied by the manufacturer.

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, such as vibrator 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.



Key

X frequency, Hz

Y velocity, m/s

1 servohydraulic vibration generators

2 electrodynamic vibration generators

3 mechanical vibration generators

Figure 6 — Comparison of electrodynamic, servohydraulic and mechanical vibration generating systems for typical application

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 to be moved

$$m_\mathrm{z} = m_\mathrm{e} + m_\mathrm{s}$$

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:

$$F = m_\mathrm{z} a = \left(m_\mathrm{e} + m_\mathrm{s}\right) a \qquad (1)$$

For sinusoidal test conditions, the force F is expressed as F_s:

$$F_\mathrm{s} = \left(m_\mathrm{e} + m_\mathrm{s}\right) a_\mathrm{max} \qquad (2)$$

where a_max is the highest acceleration stated in the specification.

For a random test, the force F_r is expressed through r.m.s. acceleration a_r by the formula:

$$F_\mathrm{r} = \left(m_\mathrm{e} + m_\mathrm{s}\right) a_\mathrm{r} \qquad (3)$$

where

$$a_\mathrm{r} = \left[\varPhi_a\left(f\right)\Delta f\right]^{1/2} \qquad (4)$$

$\varPhi_a\left(f\right)$ is the acceleration spectral density in Δf.

Equation (4) was obtained assuming that, for the purpose of testing, $\varPhi_a\left(f\right)$ 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 determined from the specimen mass m_s and the maximum load of the vibration generator in accordance with Tables 1, 2 and 3. As the first approximation, m_e may be assumed to be equal to the mass of the moving system taken from the row for the load mass equal to the specimen mass or from the next one.

Another way to estimate m_e is based on values of m_s, a_max and a_0 by the formula

$$m_\mathrm{e} = m_\mathrm{s} \frac{a_\mathrm{max}}{a_0 - a_\mathrm{max}} \qquad (5)$$

where a_0 is the maximum acceleration of the vibration generator without the load ($m_\mathrm{s} = 0$) taken from Tables 1, 2 and 3.

6.2.6 Rated travel

For an electrodynamic system, 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, depends on the maximum displacement amplitude d_{max}, the total mass m_z and the natural frequency f_z of the moving system loaded by the mass m_s:

$$S_L > d_{max} + l_{st} = d_{max} + \frac{g}{4\pi^2 f_s^2} \tag{6}$$

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.

Equation (6) may be written in terms of the natural frequency f_e of the unloaded moving system as follows:

$$S_L > d_{max} + l_{st} = d_{max} + l_1 + l_2 = d_{max} + \frac{\left(1 + m_s/m_e\right)g}{4\pi^2 f_e^2} \tag{7}$$

or

$$S_L > d_{max} + \frac{a_0}{a_{max}} \frac{g}{4\pi^2 f_e^2} \tag{8}$$

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 servohydraulic system, the rated travel S_L, defined as the rated piston stroke, depends on the peak-to-peak displacement:

$$S_L > 2d_{max} = \frac{2a_{max}}{4\pi^2 f_L^2} \tag{9}$$

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. Otherwise, for example when vibrator isolators are used, the rated travel should be less than that calculated by the value of the body displacement d_b:

$$d_b = d_{max} \frac{m_z}{m_b}$$

where m_b is the body mass.

NOTE 1 If the vibration generators are equipped with a compensating device for the moving system, $S_L = d_{max}$.

NOTE 2 If the test frequency range passes through the isolation system resonant frequency and the total moving mass is heavy, the vibrator body will vibrate vigorously, making the test extremely difficult to control.

6.2.7 Maximum velocity

For an electrodynamic and servohydraulic system, the maximum velocity v_{max} is determined from the maximum displacement d_{max} and the cross-over frequency f_t (about 20 Hz) between the range of constant displacement and the range of constant velocity:

$$v_{max} = 2\pi\, f_t\, d_{max} \tag{10}$$

Usually, the maximum velocity for electrodynamic and servohydraulic systems is 2 m/s.

NOTE To find the amplitude of displacement d and velocity v, it is advisable to use a nomogram for displacement, velocity and acceleration as functions of frequency.

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 servovalves for servohydraulic systems) should be selected properly.

All calculations to be performed in accordance with Equations (11) to (22) demand numerical data or graphs like Figures 7 to 9 which should be 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 operation in the case of failure of the moving system coil.

6.3.2.2 Output power

The apparent power of the power amplifier P, in volt amps, is determined by the formula

$$P = U \cdot I \tag{11}$$

where

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

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

The output voltage is determined by the formula

$$U = E_{\mathrm{c}} + \left(R + j\,\omega\,L\right)I \tag{12}$$

where

E_{c} is the back-electromotive force, $E_{\mathrm{c}} = B\,l\,v$;

R is the moving coil resistance;

ω is the angular frequency;

L is the moving coil inductance;

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.

In practice, it is necessary to know the maximum output power of the amplifier to ensure that the required force is available at the cross-over frequency between the ranges of constant velocity to constant acceleration, which is usually placed in the frequency range from 30 Hz to 100 Hz.

NOTE At 30 Hz: $v_{max} = 2$ m/s, $a_{max} = 400$ m/s^2; at 80 Hz: $v_{max} = 2$ m/s, $a_{max} = 1\,000$ m/s^2; at 100 Hz: $v_{max} = 1{,}7$ m/s, $a_{max} = 1\,000$ m/s^2.

In this case $E_c >> \left(R + j\,\omega\,L\right) I$. Thus,

$$U \approx E_c = B\,l\,v = \frac{F\,v}{I} \tag{13}$$

where F is the exciting force.

Then the required maximum apparent power is determined as

$$P = U\,I = F\,v \tag{14}$$

All the parameters of the Equation (14) 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

$$P = 0{,}5 F_{peak} v_{peak} \tag{15}$$

6.3.2.3 Coil current

The current in the moving coil is determined by the following formula:

$$I = 0{,}5 \frac{F_{peak}\ v_{peak}}{U} \tag{16}$$

The output voltage U is limited by the supply voltage for elements of the output stage of the power amplifier. If they are transistors, the limiting voltage is 50 V to 150 V.

6.3.2.4 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.

where

R_c is the resistive load of the coil;

L_c is the inductive load of the coil;

ω is the angular frequency;

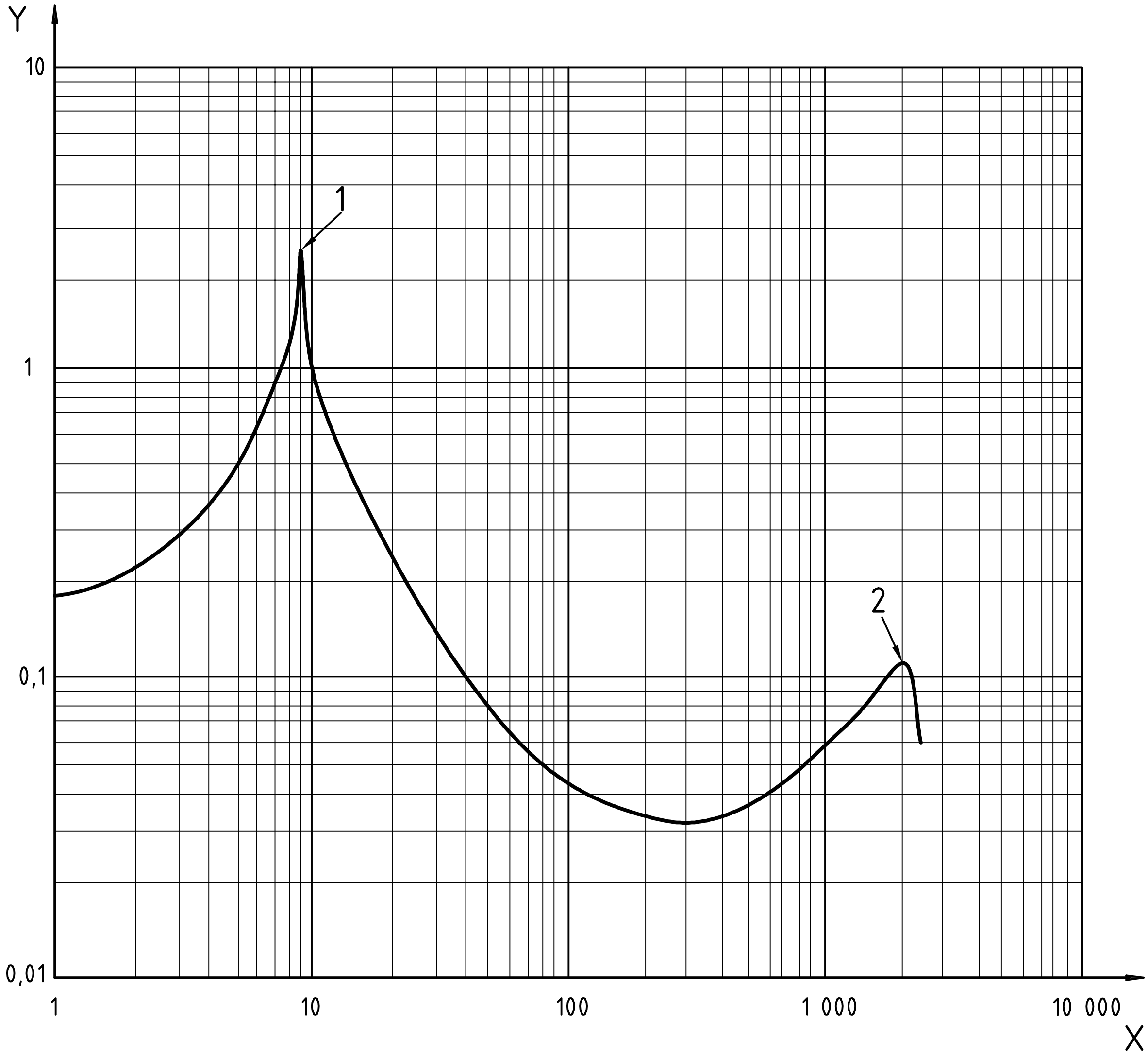
Z_m is the additional capacitance load due to coil motion, $Z_m = 0{,}5 F_{peak}\, v_{peak} / I^2$;

F is the force;

v is the coil velocity;

I is the current.

At low frequencies in the vicinity of d.c., 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 7 Hz to 60 Hz. 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.



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 is determined by the electric parameters of the coil and by its motion:

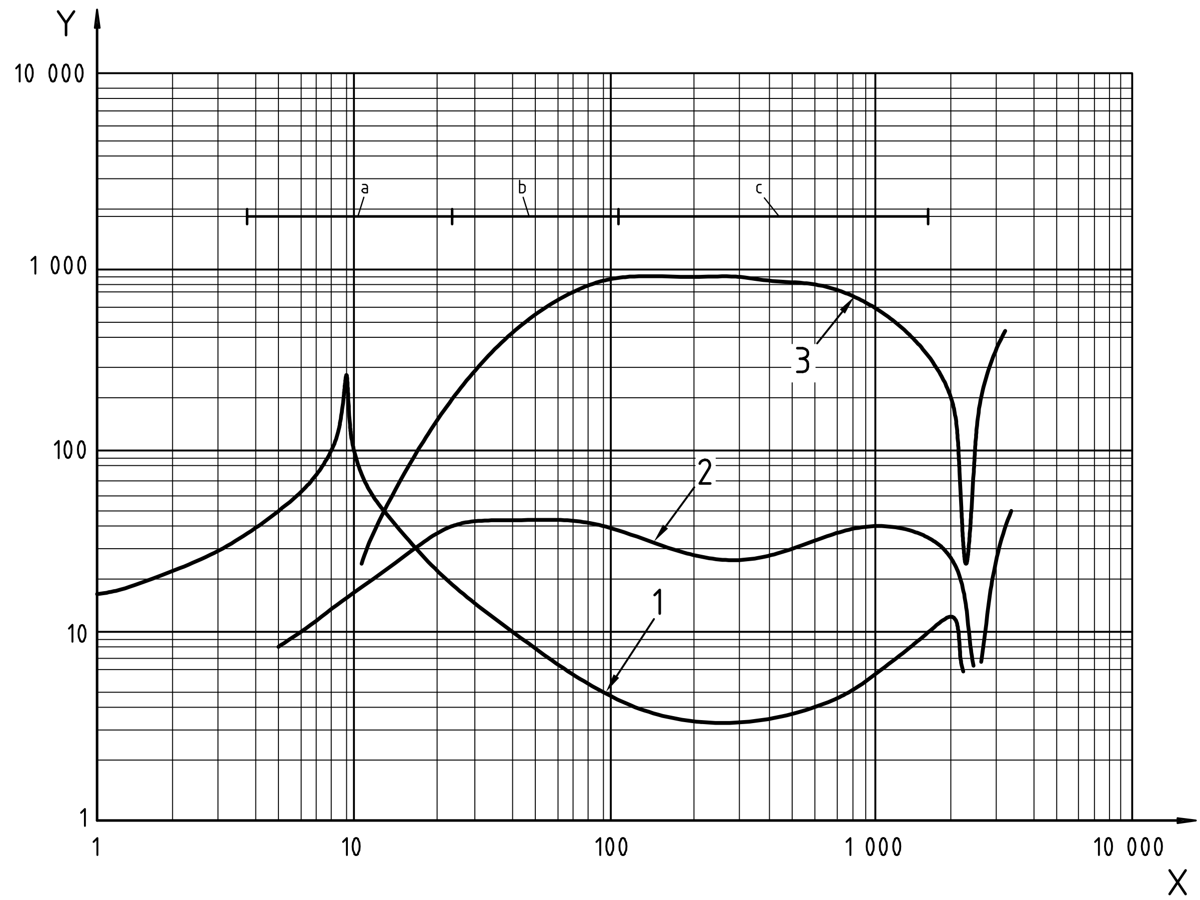
$$Z = U/I = R_{\mathrm{c}} + j\,\omega\,L_{\mathrm{c}} + Z_{\mathrm{m}} \tag{17}$$

6.3.2.5 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.



Key

[a] Constant displacement.

[b] Constant velocity.

[c] Constant acceleration.

X frequency, Hz

Y impedance, voltage or current

1 impedance

2 voltage

3 current

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 $v_{\text{max,L}}$ 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.

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_{peak}, P_{peak} and static, F_{st}, P_{st} components and are related by the formulae

$$F_{peak} = a_{max} m_z = S_p P_{peak} \tag{18}$$

$$F_{st} = g \cdot m_z = S_p P_{st} \tag{19}$$

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.

Assuming the pressure loss in the supply system to be negligible, the required mean oil flow Q_m can be calculated by formula

$$Q_m = \frac{2}{\pi} \cdot v_{peak} \cdot S_p \tag{20}$$

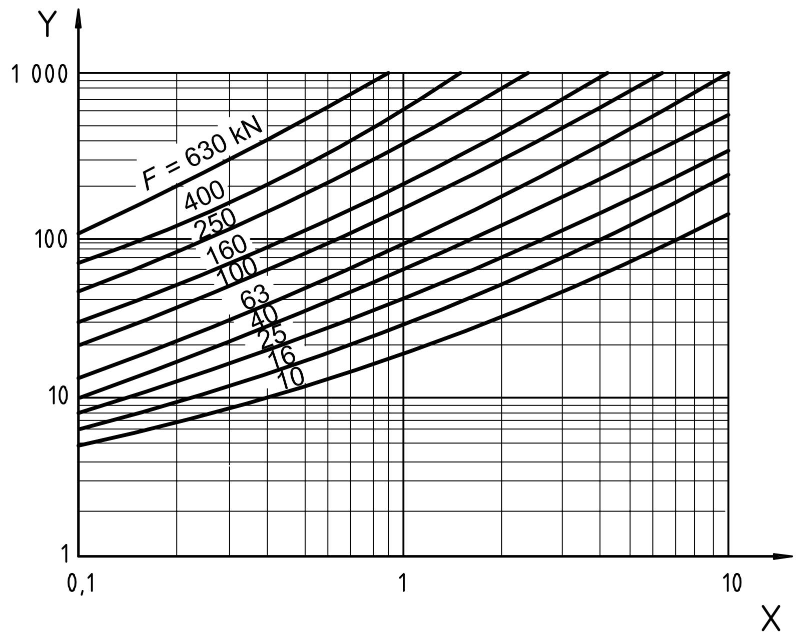
Equations (18) to (20) yield

$$Q_m = \frac{2}{\pi} \cdot \frac{(a_{max} + g) \cdot v_{peak} \cdot F_{peak}}{a_{max} \cdot P} \tag{21}$$

If the model is provided with a gravity compensation device, Equation (21) is transformed to the form

$$Q_m = \frac{2}{\pi} \cdot \frac{v_{peak} \cdot F_{peak}}{P} \tag{22}$$

In order to take account of the system losses, the result obtained according to Equation (21) or (22) should be multiplied by a factor of 1,1 to 1,2.



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)

Annex A
(informative)

Examples of selections

A.1 General

The process of determining the system requirements can be simplified into a mechanistic approach. However, there are many considerations that have to be made which can only come from experience and detailed knowledge of the subject.

The examples below consider that the device under test is symmetrical and a true mass, which it may not be in practice. For these examples, only the basic system parameters are considered. Such parameters as load support capability, cross-axial stiffness and isolation system resonance are not considered as it would make the example too complex.

A.2 Example 1: Selection of electrodynamic vibration generator

A.2.1 Test conditions

The test conditions are specified as follows:

$$m_\mathrm{s} = 80 \text{ kg};\ a_\mathrm{max} = 40 \text{ m/s}^2;\ d_\mathrm{max} = 12{,}5 \text{ mm};\ f_\mathrm{L} = 20 \text{ Hz};\ f_\mathrm{H} = 2\,000 \text{ Hz};\ \Phi_a(f) = 0{,}5\ (\text{m/s}^2)^2/\text{Hz}$$

A.2.2 Determination of the system requirements

System parameters are calculated in accordance with 6.2 and 6.3:

— mass of the moving system (in accordance with Table 1): $m_\mathrm{e} = 4$ kg

— total mass: $m_\mathrm{z} = m_\mathrm{e} + m_\mathrm{s} = 84$ kg

— required force (sinusoidal excitation): $F_\mathrm{s} = m_\mathrm{z}\, a_\mathrm{max} = 3{,}36$ kN

— required force (random excitation): $a_\mathrm{r} = \sqrt{\Phi_a(f)(f_\mathrm{H} - f_\mathrm{L})} = 31{,}5$ m/s^2; $F_\mathrm{r} = m_\mathrm{z}\, a_\mathrm{r} = 2{,}65$ kN

— required travel (assuming $f_\mathrm{e} = 20$ Hz): $S_\mathrm{L} = d_\mathrm{max} + \dfrac{(1 + m_\mathrm{s}/m_\mathrm{e})}{4\pi^2 f_\mathrm{e}^2} g = 25{,}5$ mm (given compensating device, $S_\mathrm{L} = d_\mathrm{max}$)

— maximum velocity (assuming the cross-over frequency, $f_\mathrm{t} = 25$ Hz): $v_\mathrm{max} = 2\pi f_\mathrm{t} d_\mathrm{max} = 2{,}0$ m/s (see 6.2.7)

— output power of amplifier: $P \approx 0{,}5\, F_\mathrm{s}\, v_\mathrm{max} = 3\,360$ VA

— coil current (assuming $U = 50$ V): $I = P/U = 67{,}2$ A

— amplifier load: $Z = U/I = 0{,}74\ \Omega$ (load of resistive character in the vicinity of f_t).

A.3 Example 2: Verification of capabilities of proposed electrodynamic vibration generator

A.3.1 Test conditions

The test conditions (sinusoidal vibration) are specified as follows:

$m_{\text{s}} = 35$ kg (including 25 kg of the fixture); $f_{\text{L}} = 5$ Hz; $f_{\text{H}} = 500$ Hz; $d_{\max} = 10{,}0$ mm for the range of 5 Hz to 19,49 Hz; $a_{\max} = 150$ m/s^2 for the range of 19,49 Hz to 500 Hz.

A.3.2 Vibration generator proposed to be used for testing

The vibration generator, whose capabilities are to be verified, has the following parameters:

— mass of the moving system: $m_{\text{e}} = 12$ kg

— rated force (sinusoidal excitation): $F_{\text{s}} = 10$ kN

— rated travel: $S_{\text{L}} = 25{,}4$ mm

— rated velocity: $v_{\max,\text{s}} = 2{,}0$ m/s

— maximum allowable acceleration: $a_{\max,\text{s}} = 750$ m/s^2.

A.3.3 Determination of the system requirements

System parameters are calculated in accordance with 6.2 and 6.3:

— total mass: $m_{\text{z}} = m_{\text{e}} + m_{\text{s}} = 47$ kg

— required force: $F_{\text{s}} = m_{\text{z}}\, a_{\max} = 7{,}05$ kN

— maximum velocity: $v_{\max} = 2\pi\, f_{\text{t}} d_{\max} = 1{,}225$ m/s (or $v_{\max} = \sqrt{a_{\max}\, d_{\max}} = 1{,}225$ m/s)

— required travel (assuming $f_{\text{e}} = 15$ Hz): $S_{\text{L}} = d_{\max} + \dfrac{\left(1 + m_{\text{s}}/m_{\text{e}}\right)}{4\pi^2 f_{\text{e}}^{\,2}} g = 14{,}3$ mm (given compensating device, $S_{\text{L}} = d_{\max} = 10$ mm).

A.3.4 Conclusions

The force, acceleration, velocity and travel required are well within the maximum of the proposed system. It allows the testing loads which do not behave as a true mass, and will also increase the time between maintenance prolonging the life of the vibrator.

A.4 Example 3: Selection of servohydraulic vibration generator

A.4.1 Test conditions

The test conditions are specified as follows:

$m_{\text{z}} = 1\ 000$ kg; $a_{\max} = 40$ m/s^2; $f_{\text{L}} = 6$ Hz; $f_{\text{H}} = 60$ Hz

A.4.2 Determination of the system requirements

The requirements are as follows:

— required force (sinusoidal excitation): $F = m_{\text{z}}\, a_{\max} = 40$ kN

— maximum displacement: $d_{\max} = a_{\max} \Big/ \left(4\pi^2 f_{\text{L}}^{\,2}\right) = 28$ mm

— required travel: $S_{\text{L}} = 2\, d_{\max} = 56$ mm

— maximum velocity: $v_{\max,\,\text{L}} = a_{\max} \big/ \left(2\pi\, f_{\text{L}}\right) = 1{,}06$ m/s; $v_{\max,\,\text{H}} = a_{\max} \big/ \left(2\pi\, f_{\text{H}}\right) = 0{,}106$ m/s

— mean oil flow (assuming a model with characteristics shown in Figure 9): $Q_{\text{m}} = 60$ l/min.

Annex B
(informative)

Vibration severity in test methods standardized by the IEC

B.1 General

The vibration severity of a test method is usually specified in terms of the frequency range of interest, test duration and vibration magnitude.

Common requirements for all the test methods are the following:

— total harmonic distortion for acceleration: less than 32 %;

— transverse motion: less than 25 % below 500 Hz and less than 50 % above 500 Hz;

— uniformity of the table motion at the points of load attachment: less than 25 % below 500 Hz and less than 50 % above 500 Hz.

B.2 Sinusoidal vibration (IEC 60068-2-6:1995)

The test requirements include:

— frequency range of interest, in hertz: 1 to 5 000 (1 to 35, 1 to 100, 10 to 55, 10 to 150, 10 to 500, 10 to 2 000, 10 to 5 000, 55 to 500, 55 to 2 000, 55 to 5 000, 100 to 2 000);

— displacement amplitude, in millimetres: 0,035 to 100 (35 mm and over are only applied for frequency band of 1 Hz to 10 Hz);

— acceleration amplitude, in metres per second squared: 1 to 500;

— test duration in each axis as number of sweep cycles: selected from the series of 1, 2, 5, 10, 20, 50, 100;

— tolerance on vibration amplitude, in percent: ± 15.

B.3 Random vibration (IEC 60068-2-64:1993)

The test requirements include:

— frequency range of interest, in hertz: 1 to 5 000 (1 to 100, 5 to 500, 20 to 2 000, 50 to 5 000);

— acceleration spectral density $\Phi_a(f)$, in $(\text{m/s}^2)^2/\text{Hz}$: 0,05 to 100;

— r.m.s. acceleration a, in metres per second squared, calculated on the basis of given acceleration spectral density $\Phi_a(f)$ by the formula

$$a = \left[\int_F \Phi_a(f)\,\mathrm{d}f \right]^{1/2} \approx \left[\Phi_a \Delta f \right]^{1/2}$$

where

F is the frequency range of interest, in hertz;

Δf is the width of F, in hertz;

— tolerance on acceleration spectral density, in decibels: ± 3;

— tolerance on r.m.s. acceleration, in percent: ± 10.

B.4 Time-history method (IEC 60068-2-57:1999)

The test requirements include:

— frequency range of interest, in hertz: 0,1 to 2 000 (0,1 to 10, 1 to 35, 1 to 100, 5 to 35, 10 to 100, 10 to 150, 10 to 500, 10 to 2 000, 55 to 2 000);

— response spectrum:

  — zero period acceleration (i.e. the largest peak value of acceleration of the test table), in metres per second squared: normally selected from the series of 2, 5, 10, 20, 50, 100, 200…

  — maximum value of acceleration, in metres per second squared: 2,24 to 6,5 times that of the zero period acceleration depending on the damping ratio and application;

— number and duration of time-histories:

  — number of time-histories: selected from the series of 1, 2, 3, 5, 10, 20, 30, 50…;

  — time-history duration, in seconds: selected from the series of 1, 2, 5, 10, 20, 50…;

  — duration of the strong part of the time-history, expressed as percentages of the total duration: 25, 50, 75;

— number of high peaks of the response: 3 to 20 (for a threshold value of 70 % of the required response spectrum value at specific natural frequencies).

B.5 Sine-beat method (IEC 60068-2-59:1990)

The test requirements include:

— frequency range of interest, in hertz: 0,1 to 100 (0,1 to 10, 1 to 35, 1 to 100, 5 to 35, 10 to 100);

— displacement s and acceleration a amplitudes: selected in accordance with the cross-over frequency as shown in Table B.1;

— number of cycles in the sine beat: selected from the series of 3, 5, 10, 20;

— number of sine beats: selected from the series of 1, 2, 5, 10, 20, 50 …

Table B.1 — Test levels for different cross-over frequencies

	Cross-over frequency, Hz																	
	0,8				**1,6**						**8,0**							
s, mm	40	80	120	200	10	20	30	50	100	200	0,4	0,8	1,2	2	4	8	12	20
a, m/s^2	1	2	3	5	1	2	3	5	10	20	1	2	3	5	10	20	30	50

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