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Mechanical vibration and shock — Vibration and shock in buildings with sensitive equipment —

Part 1:
Measurement and evaluation

Vibrations et chocs mécaniques — Vibrations et chocs dans les bâtiments abritant des équipements sensibles —

Partie 1: Mesurage et évaluation

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Foreword

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

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

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

In other circumstances, particularly when there is an urgent market requirement for such documents, a technical committee may decide to publish other types of normative document:

- an ISO Publicly Available Specification (ISO/PAS) represents an agreement between technical experts in an ISO working group and is accepted for publication if it is approved by more than 50 % of the members of the parent committee casting a vote;
- an ISO Technical Specification (ISO/TS) represents an agreement between the members of a technical committee and is accepted for publication if it is approved by 2/3 of the members of the committee casting a vote.

An ISO/PAS or ISO/TS is reviewed every three years with a view to deciding whether it can be transformed into an International Standard.

Attention is drawn to the possibility that some of the elements of this part of ISO/TS 10811 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO/TS 10811-1 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration and shock*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

ISO/TS 10811 consists of the following parts, under the general title *Mechanical vibration and shock — Vibration and shock in buildings with sensitive equipment*:

- *Part 1: Measurement and evaluation*
- *Part 2: Classification*

Annexes A and B of this part of ISO/TS 10811 are for information only.

Introduction

This part of ISO/TS 10811 proposes a new method for specifying and evaluating the performance of buildings housing equipment which is sensitive to shock and vibration. The purpose is to encourage users and manufacturers of sensitive equipment to collect data using this method to facilitate exchange of information and assessment of the usefulness of the methodology.

TECHNICAL SPECIFICATION **ISO/TS 10811-1:2000(E)**

Mechanical vibration and shock — Vibration and shock in buildings with sensitive equipment —

Part 1:
Measurement and evaluation

1 Scope

This part of ISO/TS 10811 defines methods of measurement, algorithms for analysis and the report of shock and vibration data for equipment in buildings which is sensitive to shock and vibration. The methods are applicable to the quantification of a future installation or the verification of an existing one.

Accurate vibration data acquisition, analysis and uniform reporting methods are needed to evaluate vibration relative to manufacturers' and generic vibration criteria. Expected levels of vibration can be found in IEC 60721. Procedures for testing of equipment can be found in IEC 60068.

NOTE A classification system of environmental vibration conditions established from measurements according to this part of ISO/TS 10811 should serve as guidelines for designers, manufacturers and users of shock-and-vibration-sensitive equipment and for building constructors (see ISO/TS 10811-2).

The types of shock and vibration considered are those transmitted from floors, tables, walls, ceilings or isolation systems into an equipment unit. The vibration and shock response of individual mechanical or electronic parts inside the unit are not considered explicitly.

The types of sensitive equipment envisaged include, but are not limited to, the following:

a) stationary computer systems, including the peripherals;

b) stationary telecommunication equipment;

c) stationary laboratory equipment such as electron microscopes, equipment using scanning probe methods, biotechnical instrumentation, mass spectrometers, etc.;

d) mechanical high-precision instruments (tools) such as equipment for microelectronics production;

e) optical high-precision instruments, systems for photoreproduction;

f) electromechanical systems in traffic control centres for trains;

g) security equipment (fire intrusion) and equipment for access control.

The types of shock and vibration considered herein can be generated by the following:

— external sources, for example traffic (by road, rail or air), or building and construction activities such as blasting, piling and vibratory compaction; the vibration response to sonic booms and acoustical excitations is also included as well as weather-induced vibration;

— equipment for indoor use, such as punch presses, forging hammers, rotary equipment (air compressors, air conditioner systems, etc.) and heavy equipment transported or operated inside a building;

- human activities in connection with the service or operation of the equipment, for example, people walking, especially on raised floors.

The measurement and evaluation of shock and vibration effects on sensitive equipment in buildings covered in this part of ISO/TS 10811 does not directly consider the human operators' capability to observe, operate or maintain the equipment. For vibration effects on human beings, see ISO 2631.

The frequency range of interest is normally 2 Hz to 200 Hz. Normally the dominant frequencies are less than 100 Hz because they represent the response of the elements in the building. For special purposes, another frequency range may be used and the numbers referring to frequency range should be changed accordingly.

The vibration amplitude and duration depend mainly upon the source, its distance from sensitive equipment, and the response of the elements of the building containing the sensitive equipment. Expressed in terms of vibration velocity, the values to consider are in the range 0,001 mm/s to 10 mm/s.

This part of ISO/TS 10811 deals only with vibration from a maximum amplitude point of view. The concept of vibration dose (e.g. estimation of fatigue life) is not treated.

2 Normative reference

The following normative document contains provisions which, through reference in this text, constitute provisions of this part of ISO/TS 10811. For dated references, subsequent amendments to, or revisions of, this publication do not apply. However, parties to agreements based on this part of ISO/TS 10811 are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

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

3 Vibration wave forms

The vibration wave forms in a building that could affect sensitive equipment may be of any kind: sinusoidal (periodic), random or transient. Therefore, to be effective, measurement and analysis of the vibration shall address all three types of motion. Typical examples of sources for different wave forms are the following:

a) rotating machinery for sinusoidal vibration;

b) road traffic (many vehicles) for random vibration;

c) single road vehicles, piling, impacting and blasting for transient vibration.

The frequency content of the vibration is determined by the source but is also influenced by the dynamics of the building.

The main purpose of this part of ISO/TS 10811 is to define methods for measurement and analysis of vibration that can be used for any types of vibration wave form.

4 Measurement and analysis

4.1 General considerations

In order to determine the vibration and shock conditions to which equipment may be exposed, accurate and comprehensive measurements in the field shall be made.

Measurement of the vibration shall be performed at points as close as possible to the points of contact of the equipment or its support with a floor or wall. For large equipment, multiple points may be necessary.

The time history of the vibration should be recorded and analysed along three orthogonal axes (preferably in the vertical direction and in two orthogonal horizontal directions).

The measurements shall, where practical, be made with the shock-and-vibration-sensitive equipment in place, or with a dummy having the same mass and similar dynamic behaviour as that of the equipment under consideration. (The effective mass of the equipment on raised floors or tables may significantly change the response levels and frequencies. At the equipment's resonance frequencies, it will act as a "dynamic absorber" and reduce the vibration.)

It is recommended that the measurement be performed, whenever possible, with the sensitive equipment (including auxiliary systems) in both the operating and the non-operating modes in order to distinguish between the various possible vibration sources.

4.2 Instrumentation requirements

Velocity transducers or accelerometers may be used. Because vibration is often at very low amplitudes, the transducers shall be highly sensitive.

Depending on the application, transducer sensitivity as high as 100 mV/(m/s^2) (for accelerometers) and 25 mV/(mm/s) (for velocity transducers) may be required. Transducers shall be mounted in accordance with ISO 5348.

In any case, the instrumentation noise (including electrical disturbances) measured as total r.m.s. value in the frequency range 2 Hz to 200 Hz should be less than 5 % of the maximum measured vibration amplitude. The use of battery-powered instrumentation is recommended to prevent power line disturbances.

The frequency range of the measurement system shall be 1 Hz to 315 Hz (–3 dB each), with a roll-off at both ends of 12 dB per octave. This may be accomplished by using a second-order high-pass filter of the Butterworth type with a cut-off frequency of 1 Hz in the low end and a second-order low-pass filter of the Butterworth type with a cut-off frequency of 315 Hz in the high end. For a digital system, the recommended sampling frequency is at least 2 000 Hz. Additional anti-alias filters with a proper cut-off frequency should be used.

4.3 Analysis

The time history shall be analysed using a set of filters corresponding to the response-equivalent peak velocity. The rationale for the method and description of the filters are given in annex A. Each filter corresponds to the Q normalized pseudo velocity response of a single-degree-of-freedom system with a defined resonance frequency and Q value. The method gives the amplitude of a sine wave having the same maximum relative displacement response as the studied vibration. The analysis result is presented as peak velocity versus resonance frequency.

The (resonance) frequencies of the filters shall be 40 per decade, logarithmically distributed. If possible the appropriate value of Q should be obtained from the instrumentation manufacturer. In the absence of such information, it is recommended that three Q values be used in the analysis: 5, 10 and 20. If only one Q value is used, the recommended value is 10.

The algorithms for the digital computation of the filtered time series are given in annex B.

CAUTION: When the time history is fed to the filters, a transient response will occur at the start. It is recommended that the calculation of maximum response be delayed until the initial transient has died away.

4.4 Comparison with other analysis methods

4.4.1 Power spectral density

The idea here is to calculate the response-equivalent peak velocity for a random signal with the given power spectral density, given as the acceleration spectral density. The noise bandwidth of the response-equivalent peak velocity filter with the resonance frequency f_n and Q value is given by

$$\frac{1}{4Q \cdot 2\pi f_n} \tag{1}$$

If the power spectral density at the frequency f_n is P (m/s^2)^2/Hz, the r.m.s. velocity of the random vibration with that spectral density fed through the filter is

$$\sqrt{\frac{P}{4Q \cdot 2\pi f_n}} \tag{2}$$

To calculate the expected maximum value, an estimate of the measurement time (or the testing time) T is needed. Let the crest factor (i.e. maximum value divided by r.m.s. value) be designated by C. The mean value C of the expectation can be shown to be

$$C = C_1 + \frac{0{,}577\,2}{C_1} \tag{3}$$

where

$C_1 = \sqrt{2\ln(f_n \cdot T)}$;

f_n is the resonance frequency, in hertz;

T is the measurement time, in seconds.

The calculation is performed for each frequency value of interest.

4.4.2 One-third-octave band spectrum

If the spectrum is given in r.m.s. values (averaged, not peak hold), the estimation of the maximum value from the r.m.s. velocity given in 4.4.1 may be applied for each frequency in the spectrum. If the vibration is known to be periodical, the factor of 1,4 may be used. The result should be presented as the vibration velocity.

4.5 Statistical considerations

Each vibration event (e.g. from a passing train, from a blasting, from a series of piling blows or a recording of vibration from rotating machinery) will result in one response-equivalent peak velocity spectrum according to 4.3. The number of events that should be analysed must be subject to sound engineering judgement. It is recommended that a complete cycle of events be treated (for example, a full day and night cycle). The final result is then calculated by taking the maximum value for each frequency over the calculated spectra.

4.6 Calibration and accuracy

All instrumentation should be calibrated to a known and traceable source at the time of purchase and at least annually thereafter. End-to-end calibration is recommended, which means that the transducer is subjected to a known (sinusoidal) vibration and the resulting signal is analysed according to the method given in 4.3.

Additionally, it is a good practice to subject transducers to a known or reference vibration immediately prior to a test programme to confirm the accuracy of the entire measurement system. Recalibration is recommended whenever an instrument has been shocked or dropped.

For the total system, the response in the frequency range 2 Hz to 200 Hz should be flat to within ±10 %.

5 Test report

The data and information to be reported are as follows:

a) a description of the sensitive equipment installation, including

  — site location,

  — building construction and floor plan, room size and layout,

  — equipment make and type,

  — mounting of equipment,

  — vibration isolators, if any;

b) a definition of any equipment failures or malfunctions;

c) a description of construction activity, traffic or any other source of shock and vibration conditions;

d) a description of the shock and vibration measuring instrumentation, including the instrument model and manufacturer, calibration equipment, transducers, amplifiers, recorders and analysers;

e) the transducer location and direction of axes;

f) the analysis results.

Annex A
(informative)

Definition of response-equivalent peak velocity spectrum

Consider the response to a base acceleration, a_1, of a mechanical single-degree-of-freedom system (Figure A.1). The parameters of the system are conventionally as follows:

m is the mass, in kilograms (which can also be written as $\mathrm{Ns^2/m}$);

c is the damper constant, in Ns/m;

k is the spring constant, in N/m.

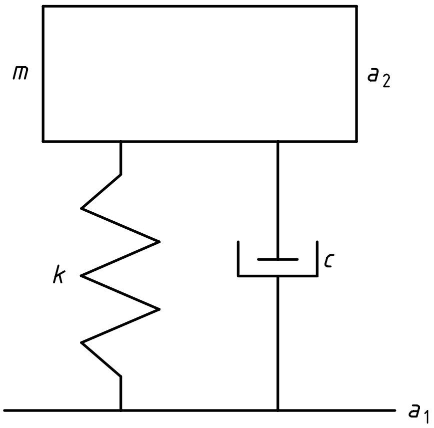

Figure A.1 — Single-degree-of-freedom system

If the response is measured as the acceleration, a_2, then the transfer function is given by:

$$\frac{a_2}{a_1} = \frac{cs + k}{ms^2 + cs + k} \qquad \text{(A.1)}$$

where s is the Laplace variable (complex frequency), in radians per second.

The single-degree-of-freedom system is normally characterized by its resonance frequency, f_0, in hertz, and the resonance gain Q (Q-factor):

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

$$Q = \sqrt{\frac{km}{c}} \qquad \text{(A.3)}$$

The transfer function may then be rewritten as

$$\frac{a_2}{a_1} = \left(\frac{\omega_0 s}{Q} + \omega_0^2\right) \Bigg/ \left(s^2 + \frac{\omega_0 s}{Q} + \omega_0^2\right) \tag{A.4}$$

with $\omega_0 = 2\pi f_0$ being the angular resonance frequency, in radians per second.

The transfer function is given versus frequency in Figure A.2, where the resonance frequency is set to 1 Hz and $Q = 10$ as an example. Note the gain of Q at resonance.

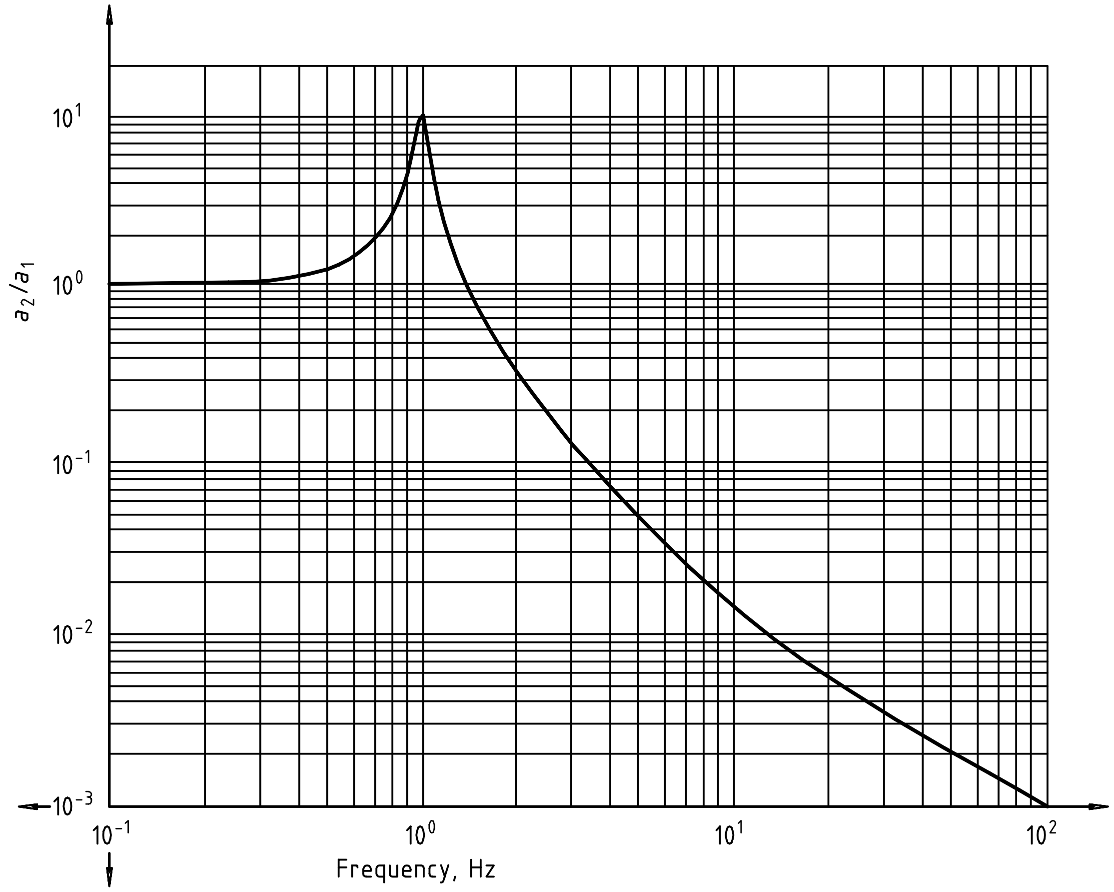

Figure A.2 — Transfer function of a single-degree-of-freedom system
(f_0 = 1 Hz, Q = 10)

For motion-sensitive equipment, the relative displacement error often is a good measure to estimate the risk of malfunction. A mechanical system may have a number of resonances, but it is common practice to consider one resonance at a time, using the simple single-degree-of-freedom model. To describe the severity of the applied external vibration, the relative displacement between the base and the mass in the model is calculated for several different resonance frequencies.

If the response is calculated as relative displacement, $d_2 - d_1$, given base acceleration, a_1, the transfer function is given by:

$$\frac{d_2 - d_1}{a_1} = -1 \Bigg/ \left(s^2 + \frac{\omega_0 s}{Q} + \omega_0^2\right) \tag{A.5}$$

If the transfer function for relative displacement is multiplied by ω_0, the result is the pseudo velocity transfer function:

$$\frac{d_2 - d_1}{a_1} \times \omega_0 = -\omega_0 \Bigg/ \left(s^2 + \frac{\omega_0 s}{Q} + \omega_0^2 \right) \qquad \text{(A.6)}$$

The pseudo velocity transfer function (filter function) is plotted versus frequency in Figure A.3. There is still a gain of Q at resonance.

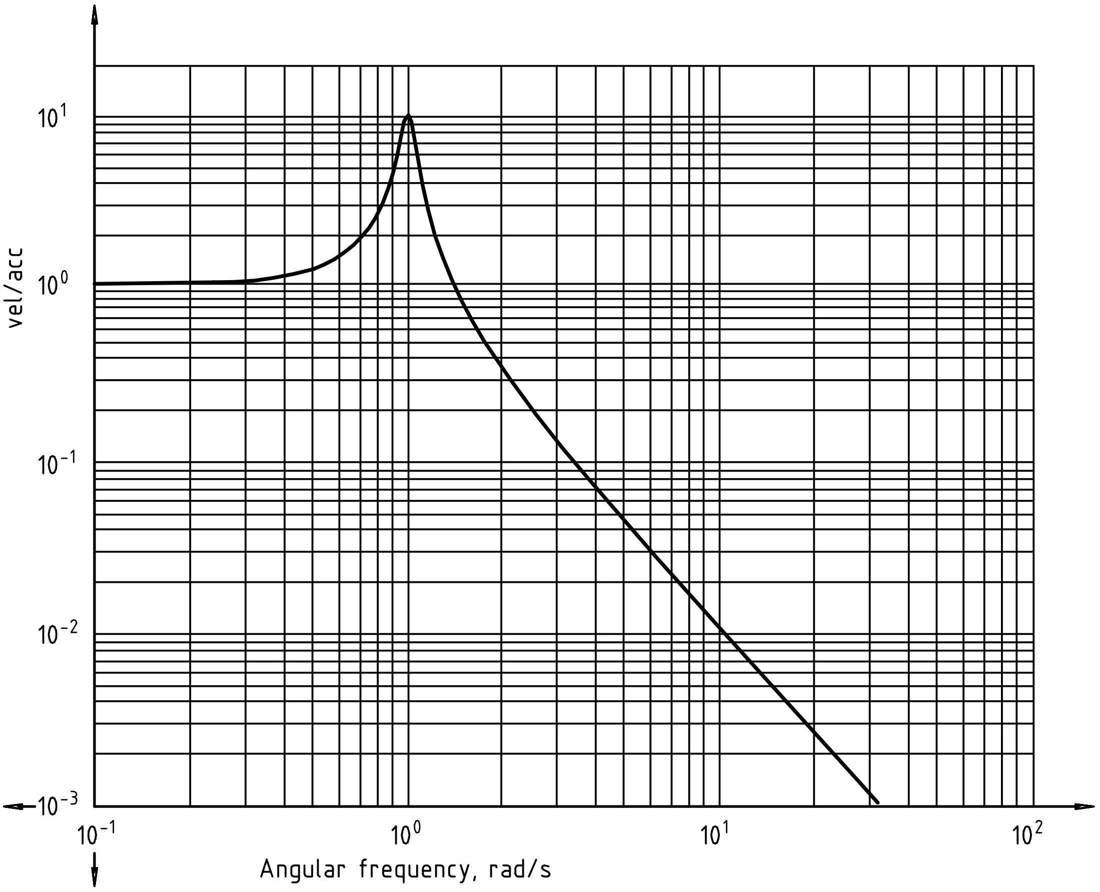

Figure A.3 — Pseudo velocity transfer function with acceleration input
(f_0 = 1 Hz, Q = 10)

To obtain a filter function with maximum response equal to 1, the pseudo velocity response function is divided by Q, and the resulting transfer function for acceleration input $H_a(s,\omega_0,Q)$ is

$$H_a(s,\omega_0,Q) = \frac{-\omega_0}{Q} \Bigg/ \left(s^2 + \frac{\omega_0 s}{Q} + \omega_0^2 \right) \qquad \text{(A.7)}$$

The normalized filter function is given in Figure A.4.

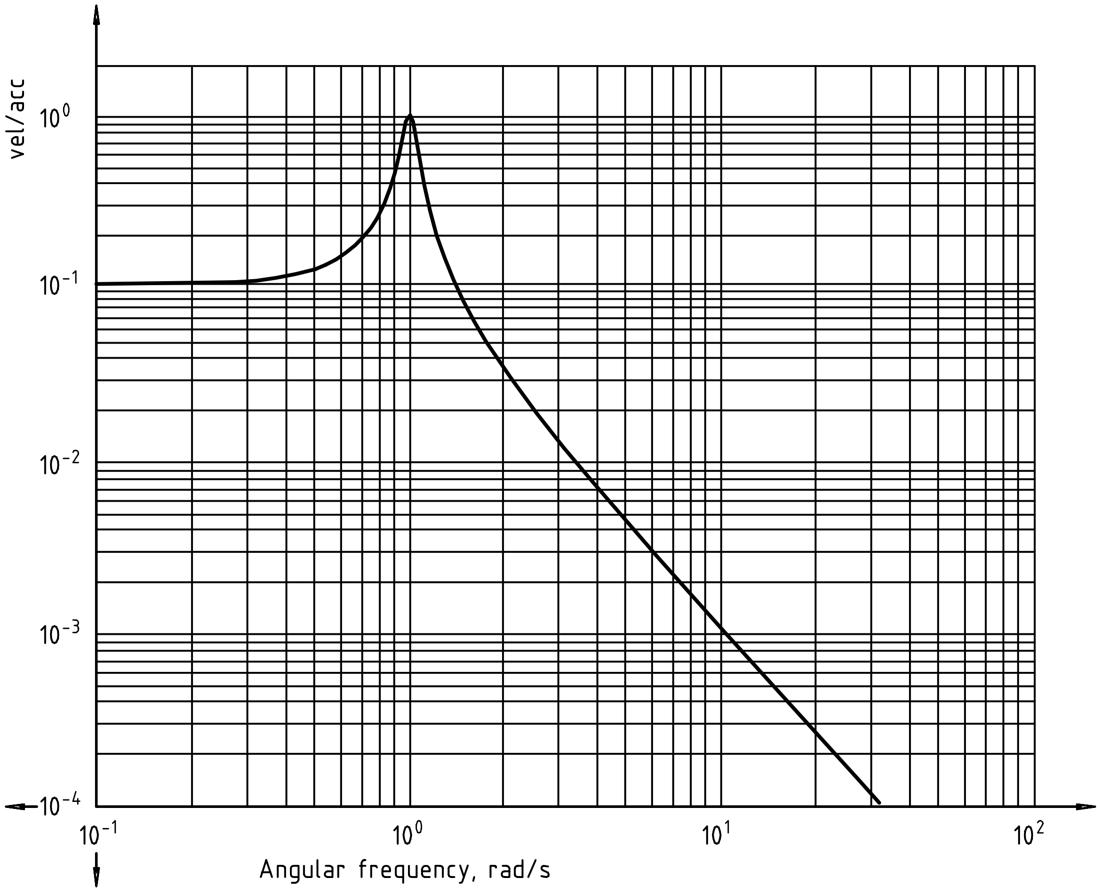

Figure A.4 — Normalized pseudo velocity transfer function with acceleration input
(f_0 = 1 Hz)

As this normalized filter function has a gain of 1, the maximum response of the filter, v_{max}, to the applied excitation will be the same as if a sinusoidal vibration with peak velocity v_{max} (and a frequency equal to the resonance frequency) were applied. This is the rationale for using the method, and the reason for the term response-equivalent peak velocity.

To analyse a given base acceleration, the time history is fed into a set of filters, as defined above, and the maximum response (maximum of the absolute value of the response) is noted for each filter. The resonance frequencies should be 40 per decade and logarithmically distributed between 2 Hz and 200 Hz. The diagram showing maximum response versus resonance frequency is the response-equivalent peak velocity spectrum.

Due to the design, one obvious interpretation of the resulting spectrum is as follows. If a slowly swept sine wave vibration, with the peak velocity as a function of frequency given by the spectrum, is applied to any single-degree-of-freedom system, the maximum response will be the same as that one resulting from the vibration under analysis. If the primary vibration is measured as velocity, the filter must be changed accordingly. The transfer function for response-equivalent peak velocity with velocity input is then

$$H_v(s,\omega_0,Q) = \frac{-s \cdot \omega_0}{Q} \Bigg/ \left(s^2 + \frac{\omega_0 s}{Q} + \omega_0^2 \right) \tag{A.8}$$

The transfer function is given in Figure A.5.

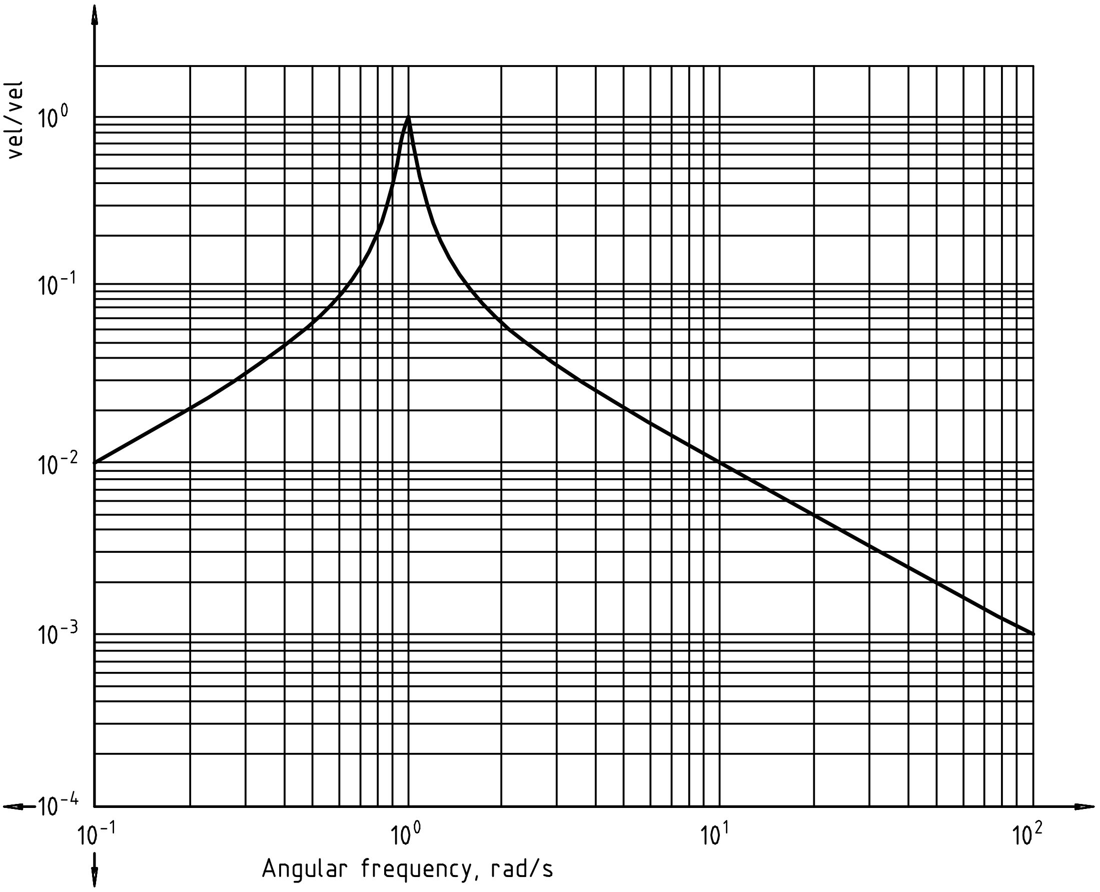

Figure A.5 — Normalized pseudo velocity transfer function with velocity input
(f_0 = 1 Hz)

As a further example, the spectrum is calculated with a vibration of 20 Hz. The amplitude is set to 1 mm/s for velocity or 125,7 mm/s^2 for acceleration. The resulting response-equivalent peak velocity spectrum is given in Figure A.6.

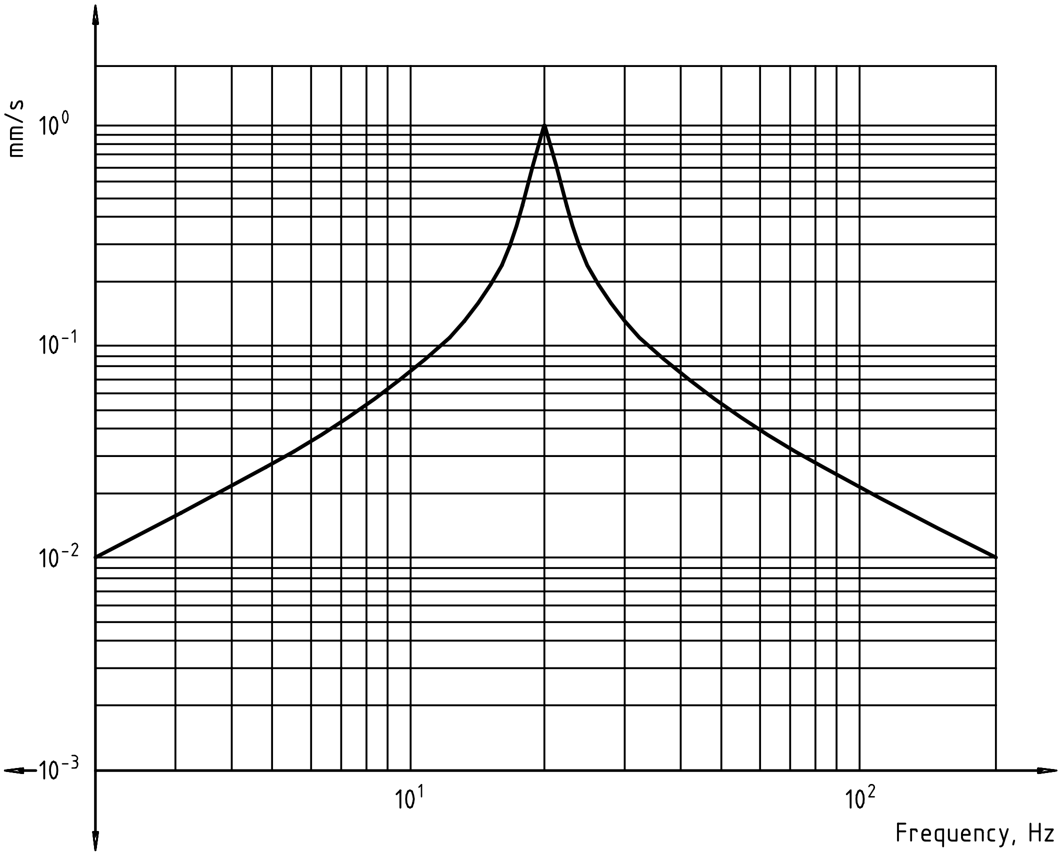

Figure A.6 — Response-equivalent peak velocity spectrum for 20 Hz sine wave with velocity amplitude 1 mm/s

Annex B
(informative)

Algorithms for filter calculations

B.1 General

Many methods exist to convert given analog filters into digital. The simple expressions given here are calculated using a bilinear transform. For another widely used transform, the ramp-invariant transform, see reference [9].

All methods have their limitations, so it is recommended that the accuracy of the method used be checked by an end-to-end calibration, including the algorithm used.

B.2 Algorithm for acceleration input

The response time series y_n to the input acceleration time series x_n is calculated by the difference equation:

$$y_n = b_0 \cdot x_n + b_1 \cdot x_{n-1} + b_2 \cdot x_{n-2} - a_1 \cdot y_{n-1} - a_2 \cdot y_{n-2} \qquad \text{(B.1)}$$

with the coefficients

$$b_0 = \frac{A}{2Q \cdot B \cdot f_s}$$

$$b_1 = 2b_0$$

$$b_2 = b_0$$

$$a_1 = \frac{2A^2 - 2}{B}$$

$$a_2 = \left(1 - \frac{A}{Q} + A^2\right) \Big/ B$$

where

$$A = \tan\left(\pi \times \frac{f_0}{f_s}\right)$$

$$B = 1 + \frac{A}{Q} + A^2$$

f_0 is the resonance frequency;

f_s is the sampling frequency;

Q is the resonance gain (Q-factor).

B.3 Algorithm for velocity input

The response time series y_n to the input velocity time series v_n is calculated by the difference equation:

$$y_n = b_0 \cdot v_n + b_1 \cdot v_{n-1} + b_2 \cdot v_{n-2} - a_1 \cdot y_{n-1} - a_2 \cdot y_{n-2} \qquad \text{(B.2)}$$

with the coefficients

$$b_0 = \frac{A}{Q \cdot B}$$

$$b_1 = 0$$

$$b_2 = -b_0$$

$$a_1 = \frac{2A^2 - 2}{B}$$

$$a_2 = \left(1 - \frac{A}{Q} + A^2\right) \Big/ B$$

where

$$A = \tan\left(\pi \times \frac{f_0}{f_s}\right)$$

$$B = 1 + \frac{A}{Q} + A^2$$

f_0 is the resonance frequency;

f_s is the sampling frequency;

Q is the resonance gain (Q-factor).

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