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BRITISH STANDARD

BS 6897-2:
1990
ISO 7626-2:
1990

Method for

Experimental determination of mechanical mobility —

Part 2: Measurements using single-point translation excitation with an attached vibration exciter

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Committees responsible for this British Standard

The preparation of this British Standard was entrusted by the General Mechanical Engineering Standards Policy Committee (GME/-) to Technical Committee GME/21, upon which the following bodies were represented:

Electricity Supply Industry in England and Wales
Institute of Sound and Vibration Research
Institution of Mechanical Engineers
Lloyds Register of Shipping
Ministry of Defence
Power Generation Contractors Association (BEAMA Ltd.)
Society of British Aerospace Companies Ltd.
Society of Environmental Engineers
Society of Motor Manufacturers and Traders Ltd.

This British Standard, having been prepared under the direction of the General Mechanical Engineering Standards Policy Committee, was published under the authority of the Board of BSI and comes into effect on 30 November 1990

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The following BSI references relate to the work on this standard:
Committee reference GME/21
Draft for comment 86/75548 DC

ISBN 0 580 19045 5

Amendments issued since publication

Amd. No.	Date	Comments

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National foreword

This British Standard has been prepared under the direction of the General Mechanical Engineering Standards Policy Committee. It is identical with ISO 7626:1990 "*Vibration and shock — Experimental determination of mechanical mobility — Part 2: Measurements using single-point translation excitation with an attached vibration exciter*", which was prepared by Technical Committee ISO/TC108 of the International Organization for Standardization (ISO) and in the development of which the UK played an active part.

Cross-references

International Standard	Corresponding British Standard
ISO 5344:1980	BS 6140 *Test equipment for generating vibration* Part 1:1981 *Methods of describing characteristics of electrodynamic test equipment for generating vibration* (Identical)
ISO 7626-1:1986	BS 6897 *Experimental determination of mechanical mobility* Part 1:1987 *Specification for transducers* (Identical)

The Technical Committee has reviewed the provisions of ISO 2041:1975, to which reference is made in the text, and has decided that they are acceptable for use in conjunction with this standard.

A related British Standard to ISO 2041:1975 is BS 3015:1976 "*Glossary of terms relating to mechanical vibration and shock*".

ISO 4865 is currently under preparation.

Parts 3, 4 and 5 of ISO 7626, referred to in the introduction to this standard, are currently in preparation.

A British Standard does not purport to include all the necessary provisions of a contract. Users of British Standards are responsible for their correct application.

Compliance with a British Standard does not of itself confer immunity from legal obligations.

Summary of pages

This document comprises a front cover, an inside front cover, pages i and ii, pages 1 to 22, an inside back cover and a back cover.

This standard has been updated (see copyright date) and may have had amendments incorporated. This will be indicated in the amendment table on the inside front cover.

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Introduction

General introduction to ISO 7626 on mobility measurement

Dynamic characteristics of structures can be determined as a function of frequency from mobility measurements or measurements of the related frequency-response functions, known as accelerance and dynamic compliance. Each of these frequency-response functions is the phasor of the motion response at a point on a structure due to a unit force (or moment) excitation. The magnitude and the phase of these functions are frequency-dependent.

Accelerance and dynamic compliance differ from mobility only in that the motion response is expressed in terms of acceleration or displacement, respectively, instead of in terms of velocity. In order to simplify the various parts of ISO 7626, only the term "mobility" will be used. It is understood that all test procedures and requirements described are also applicable to the determination of accelerance and dynamic compliance.

Typical applications for mobility measurements are for:

a) predicting the dynamic response of structures to known or assumed input excitation;

b) determining the modal properties of a structure (natural frequencies, mode shapes and damping ratios);

c) predicting the dynamic interaction of interconnected structures;

d) checking the validity and improving the accuracy of mathematical models of structures;

e) determining dynamic properties (i.e. the complex modulus of elasticity) of materials in pure or composite forms.

For some applications, a complete description of the dynamic characteristics may be required using measurements of translational forces and motions along three mutually perpendicular axes as well as measurements of moments and rotational motions about these three axes. This set of measurements results in a 6 × 6 mobility matrix for each location of interest. For N locations on a structure, the system thus has an overall mobility matrix of size $6N \times 6N$.

For most practical applications, it is not necessary to know the entire $6N \times 6N$ matrix. Often it is sufficient to measure the driving-point mobility and a few transfer nobilities by exciting with a force at a single point in a single direction and measuring the translational response motions at key points on the structure. In other applications, only rotational mobilities may be of interest.

In order to simplify the use of the various parts of ISO 7626 in the various mobility measurement tasks encountered in practice, ISO 7626 will be published as a set of five separate parts.

ISO 7626-1 covers basic definitions and transducers. The information in ISO 7626-1 is common to most mobility measurement tasks.

ISO 7626-2 (this part of ISO 7626) covers mobility measurements using single-point translational excitation with an attached exciter.

ISO 7626-3 covers mobility measurements using single-point rotational excitation with an attached exciter. This information is primarily intended for rotor system rotational resonance predictions.

ISO 7626-4 covers measurements of the entire mobility matrix using attached exciters. This includes the translational, rotational and combination terms required for the 6 × 6 matrix for each location on the structure.

ISO 7626-5 covers mobility measurements using impact excitation with an exciter which is not attached to the structure.

Mechanical mobility is defined as the frequency-response function formed by the ratio of the phasor of the translational or rotational response velocity to the phasor of the applied force or moment excitation. If the response is measured with an accelerometer, conversion to velocity is required to obtain the mobility. Alternatively, the ratio of acceleration to force, known as accelerance, may be used to characterize a structure. In other cases, dynamic compliance, the ratio of displacement to force, may be used.

NOTE Historically, frequency-response functions of structures have often been expressed in terms of the reciprocal of one of the above-named dynamic characteristics. The arithmetic reciprocal of mechanical mobility has often been called mechanical impedance. It should be noted, however, that this is misleading because the arithmetic reciprocal of mobility does not, in general, represent any of the elements of the impedance matrix of a structure. This point is elaborated upon in ISO 7626-1.

Mobility test data cannot be used directly as part of an impedance model of the structure. In order to achieve compatibility of the data and the model, the impedance matrix of the model shall be converted to mobility or *vice versa* (see ISO 7626-1 for limitations).

Introduction to this part of ISO 7626

For many applications of mechanical mobility data, it is sufficient to determine the driving-point mobility and a few transfer mobilities by exciting the structure at a single location, in a single direction, and measuring the translational response motions at key points on the structure. The translational excitation force may be applied either by vibration exciters attached to the structure under test or by devices that are not attached.

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Categorization of excitation devices as "attached" or "unattached" has significance in terms of the ease of moving the excitation point to a new position. It is much easier, for example, to change the location of an impulse applied by an instrumented hammer than it is to relocate an attached vibration exciter to a new point on the structure. Both methods of excitation have applications to which they are best suited. This part of ISO 7626 deals with measurements using a single attached exciter; measurements made by impact excitation without the use of attached exciters are covered by ISO 7626-5.

1 Scope

This part of ISO 7626 specifies procedures for measuring mechanical mobility and other frequency-response functions of structures, such as buildings, machines and vehicles, using a single translational vibration exciter attached to the structure under test for the duration of the measurement.

It is applicable to measurements of mobility, accelerance, or dynamic compliance, either as a driving-point measurement or as a transfer measurement. It also applies to the determination of the arithmetic reciprocals of those ratios such as free effective mass. Although excitation is applied at a single point, there is no limit on the number of points at which simultaneous measurements of the motion response may be made. Multiple-response measurements are required, for example, for modal analyses.

2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this part of ISO 7626. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this part of ISO 7626 are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 2041:1975, *Vibration and shock — Vocabulary.*

ISO 4865:—, *Vibration and shock — Methods for analysis and presentation of data*[1].

ISO 5344:1980, *Electrodynamic test equipment for generating vibration — Method of describing equipment characteristics.*

ISO 7626-1:1986, *Vibration and shock — Experimental determination of mechanical mobility — Part 1: Basic definitions and transducers.*

[1] To be published.

3 Definitions

For the purposes of this part of ISO 7626, the definitions given in ISO 7626-1 and ISO 2041 apply; certain terms pertaining to digital data analysis are defined in ISO 4865. For convenience, the most important definitions used in this part of ISO 7626 are given in **3.1** to **3.5**.

3.1
frequency-response function

the frequency-dependent ratio of the motion-response phasor to the phasor of the excitation force

NOTE 1 Frequency-response functions are properties of linear dynamic systems which do not depend on the type of excitation function. Excitation can be harmonic (i.e. sinusoidal), random or transient functions of time. The test results obtained with one type of excitation can thus be used for predicting the response of the system to any other type of excitation. Phasors and their equivalents for random and transient excitation are discussed in Annex B of ISO 7626-1:1986.

NOTE 2 Linearity of the system is a condition which, in practice, will be met only approximately, depending on the type of system and on magnitude of the input. Care has to be taken to avoid non-linear effects.

NOTE 3 Motion response may be expressed in terms of either velocity, acceleration, or displacement; the corresponding frequency-response function designations are mobility, accelerance, and dynamic compliance, respectively.

NOTE 4 This definition has been taken from ISO 7626-1:1986.

3.2
mobility

the frequency-response function formed by the ratio of the velocity-response phasor to the excitation-force phasor or, in other words, the ratio of the velocity-response spectrum to the excitation-force spectrum

the required boundary conditions are that no forces are applied to any point on the structure other than the exciting force at the driving point

3.3
driving-point mobility, Y_{jj}

the frequency-response function formed by the ratio, in metres per newton second, of the velocity-response phasor at point j to the excitation force phasor applied at the same point with all other measurement points on the structure allowed to respond freely without any constraints other than those constraints which represent the normal support of the structure in its intended application

NOTE 1 The term "point" designates a location and a direction. The term "coordinate" has also been used with the same meaning as "point".

NOTE 2 This definition has been taken from ISO 2041:1975.

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3.4
transfer mobility, Y_{ij}

the frequency-response function formed by the ratio, in metres per newton second, of the velocity-response phasor at point *i* to the excitation force phasor applied at point *j* with all points on the structure, other than *j*, allowed to respond freely without any constraints other than those constraints which represent the normal support of the structure in its intended application

NOTE This definition has been taken from ISO 2041:1975.

3.5
frequency range of interest

span, in hertz, from the lowest frequency to the highest frequency at which mobility data are to be obtained in a given test series

NOTE This definition has been taken from ISO 7626-1:1986.

4 Overall configuration of the measurement system

Individual components of the system used for mobility measurements carried out in accordance with this part of ISO 7626 shall be selected to suit each particular application.

However, all such systems should include certain basic components arranged as shown in Figure 1. Requirements for the characteristics and usage of those components are given in the relevant clauses.

5 Support of the structure under test

5.1 General

Mobility measurements are performed on structures either in an ungrounded condition (freely suspended) or in a grounded condition (attached to one or more supports), depending on the purpose of the test. The constraints on the structure induced by the application of the vibration exciter are dealt with in **6.4**.

5.2 Grounded measurements

The support of the test structure shall be representative of its support in typical applications unless it has been specified otherwise. A description of the support should be included in the test report.

5.3 Ungrounded measurements

A compliant suspension of the test structure shall be used. The magnitudes of all relevant elements of the driving-point mobility matrix of the suspension, at its point(s) of attachment to the structure under test, should be at least ten times greater than the magnitudes of the corresponding elements of the mobility matrix of the structure at the same attachment point(s). Details of the suspension system used shall be included in the test report.

In the absence of quantitative information, design of the suspension is largely a matter of judgment. As a minimum requirement, all resonance frequencies of the rigid-body modes of the suspended structure shall be less than half the lowest frequency of interest.

Items commonly used to provide compliant suspension include shock cords and resilient pads of material such as foam and rubber. Since some suspension systems have mass but little damping, care shall be taken to ensure that the frequencies of the suspension resonances are well away from the modal frequencies of the test structure itself. The masses of any suspension components, such as hooks and turnbuckles, located close to the structure under test shall also be less than one-tenth of the free effective mass of the structure at each frequency of interest.

Preliminary testing should be performed to identify locations for the attachment of the suspension with the minimum possible effect on the intended measurements. Suspension near nodal points of the structure under test will minimize the interaction of the suspension system with the structure. Suspension cables should run normal to the direction of excitation, if practical, and even in this case, transverse string vibrations of suspension cables can affect the data.

NOTE Attention should also be paid to any added damping of the structure due to the suspension system.

6 Excitation

6.1 General

Any excitation waveform, the spectrum of which covers the frequency range of interest, can be used provided that the excitation and response signals are processed properly.

Early investigators used sinusoidal excitation signals; under ideal conditions, the steady-state response then is also a sinusoidal signal. The ratio of the amplitudes of the sinusoidal response and the excitation signals yields the modulus of the mobility at that particular frequency and the phase difference is the argument.

This technique works because the amplitude of a sinusoidal signal is the modulus of the Fourier transform of that signal, so that excitation in itself accomplishes the same end as Fourier transformation of more complex signals. However, it is necessary to dwell at each excitation frequency long enough to reach the steady-state response. This is not necessary if the Fourier transforms of the excitation signal and of the response velocity are determined. A short duration sine burst can then be used and the ratio of the response and force spectra gives a correct mobility value over a limited frequency range.

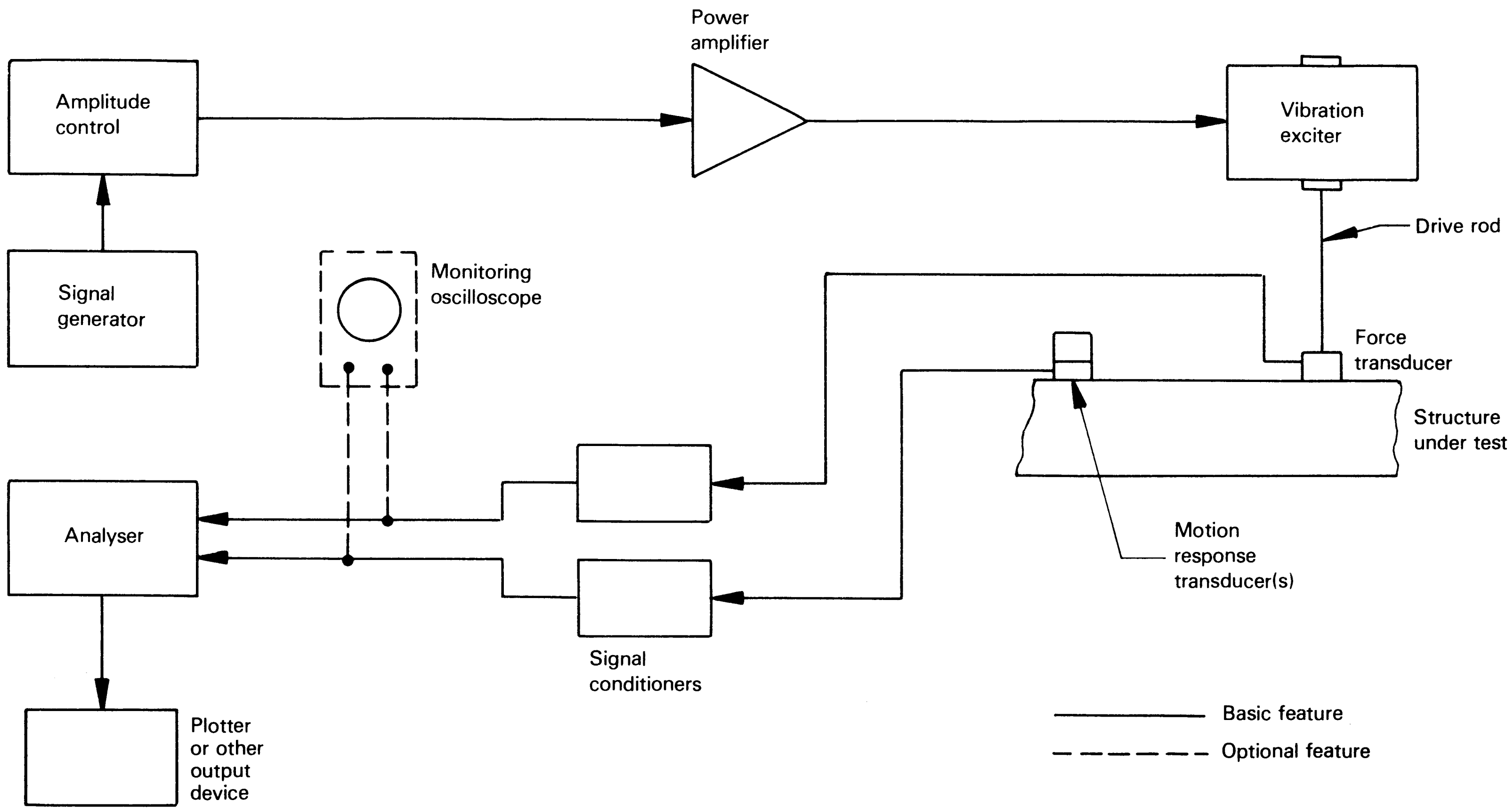

Figure 1 — Block diagram of mobility measurement system

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The same will hold in case of swept-sine excitation: if Fourier transforms are applied, the sweep rate limitations mentioned in **9.2.3** are no longer relevant and the slowly swept-sine signal can be replaced by a fast swept-sine signal.

When applying digital Fourier transforms it is rather easy to use periodic excitation signals, for example periodic chirp or periodic random. The advantage is that time-domain leakage can be prevented easily.

6.2 Excitation waveforms

6.2.1 *General*

Applicable excitation waveforms include, but are not limited to, those described in **6.2.2** to **6.2.5**. This part of ISO 7626 reflects technology in wide use during its drafting and no attempt was made to include emerging or research-oriented measurement methods. Comparative advantages and disadvantages of the different types of waveforms are discussed in [1].

6.2.2 *Discretely stepped sinusoidal excitation*

The excitation for a given measurement consists of a set of individual discrete-frequency sinusoidal signals, applied sequentially. The frequencies of the signals are incrementally spaced over the frequency range of interest; requirements for selecting the frequency increment are given in **9.2.2**. At each frequency, the excitation is applied over a small interval of time. The length of the time interval shall be sufficiently long to achieve steady-state response of those natural vibration modes of the structure that are excited at the particular frequency and to achieve proper processing of the signal.

6.2.3 *Slowly swept sinusoidal excitation*

The excitation for a given measurement is a sinusoidal signal continuously swept in frequency from the lower to the upper limit of the frequency range of interest. The rate at which the frequency is swept shall be slow enough to achieve quasi-steady-state response of the structure; requirements for selecting the sweep rate are given in **9.2.3**. Over a small interval of time, the energy of excitation is concentrated in the small frequency band swept during that interval.

6.2.4 *Stationary random excitation*

The waveform of stationary random excitation has no explicit mathematical representation, but does have certain statistical properties. The spectrum of the excitation signal shall be specified by the spectral density of the exciting force. Recommendations for shaping the spectral density to concentrate the excitation in the frequency range of interest are given in **9.4.3**. All vibration modes having frequencies within this frequency range are excited simultaneously.

6.2.5 *Other excitation waveforms*

Additional types of waveforms, described in **6.2.5.1** to **6.2.5.4**, also simultaneously excite all vibration modes within a frequency band of interest. The methods of signal processing and excitation control used in conjunction with these waveforms are similar to those used with stationary-random excitation. These waveforms are repetitive and are recommended when synchronous time-domain averaging of the response waveform is necessary to measure properly the motion response of the structure.

6.2.5.1 *Pseudo-random excitation*

The excitation signal is synthesized digitally in the frequency domain to attain a desired spectrum shape. An inverse Fourier transformation of the spectrum may be performed to generate repetitive digital signals which are then converted to analogue electrical signals to drive the vibration exciter.

6.2.5.2 *Periodic-chirp excitation*

A periodic chirp is a rapid repetitive sweep of a sinusoidal signal in which the frequency is swept up or down between selected frequency limits. The signal may be generated either digitally or by a sweep oscillator and should be synchronized with the signal processor for waveform averaging to improve the signal-to-noise ratio.

6.2.5.3 *Periodic-impulse excitation*

A suitably shaped impulse function, usually generated digitally, is periodically repeated. The signal processor should be synchronized with the signal generator. The impulse function shape (typically half-sine or decaying step functions) shall be chosen to meet the excitation frequency requirements.

6.2.5.4 *Periodic-random excitation*

A periodic-random excitation combines the features of pure random and pseudo-random excitation in that it satisfies the conditions for a periodic signal yet changes with time so that it excites the structure in a purely random manner; this is done by using different pseudo-random excitation for each average.

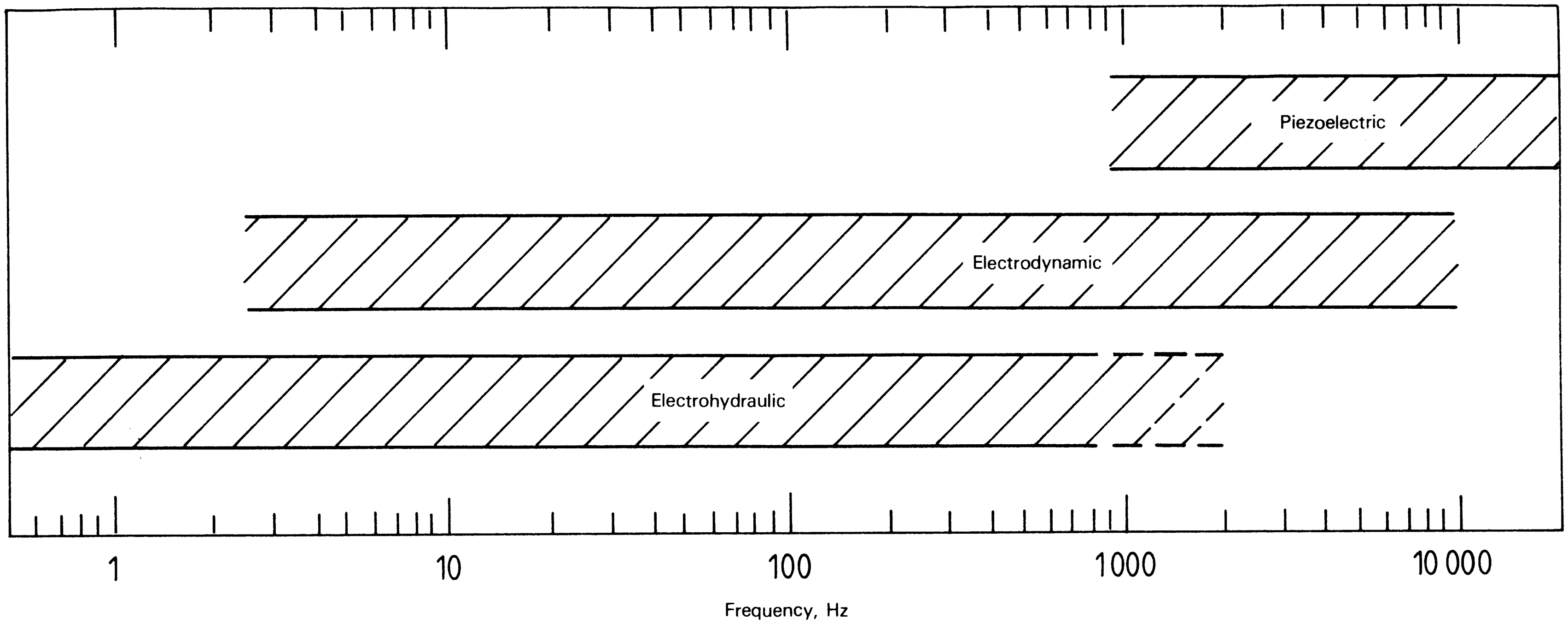

Figure 2 — Frequency ranges of general applicability for vibration exciters

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6.3 Vibration exciters

Devices commonly attached to the structure under test to apply input forces having desired waveforms include electrodynamic, electrohydraulic, and piezoelectric vibration exciters (see ISO 5344). The frequency ranges of general applicability for each type of exciter are shown in Figure 2.

The basic requirement of a vibration exciter is that it shall provide a sufficient force and displacement capability so that mobility measurements may be made over the entire frequency range of interest with an adequate signal-to-noise ratio. A larger vibration exciter may be required to apply adequate broad-band random excitation to a given structure than is needed for sinusoidal excitation. Smaller exciters may be used if a band limiting of the random noise is selected or if time-domain averaging of the excitation and response signal waveforms is used (see **6.2.5**).

NOTE The coherence function may be used as a measure of the adequacy of the vibration exciter in relation to background and electronic noise.

The excitation-force input to a structure gives rise to a reaction force which is provided either by the exciter support or by the inertia of the exciter itself; these approaches are illustrated in Figure 3 a) and Figure 3 b). If necessary, an additional mass should be attached to the exciter. An incorrect set-up which would allow transmission of exciter reaction forces to the structure via a path other than through the force transducer, i.e. through a common base on which both the exciter and the structure are mounted, is illustrated in Figure 3 c).

6.4 Avoidance of spurious forces and moments

6.4.1 *General*

A basic requirement for mobility measurements is that the excitation force be applied in a single direction at a single point on a structure.

Any spurious moments or forces (other that the intended excitation force along the intended direction) will cause errors in the resulting mobility data. The driving point and all other measurement points on the structure shall be free to respond by moving in any direction without restraint. Dynamic interactions between the structure, the motion, and the force transducers as well as between the structure and the exciter shall be avoided. In order to ensure that spurious forces and moments are avoided the factors dealt with in **6.4.2** to **6.4.4** shall be taken into consideration.

6.4.2 *Transducer mass loading*

Spurious forces are generated at each transducer attachment point as a result of the acceleration of the transducer mass. Measurement errors caused by mass loading shall be minimized by selecting transducers having the smallest mass consistent with sensitivity requirements. When measuring driving-point mobility, such loading by a force transducer can be electronically compensated to a certain extent (see **7.3**).

6.4.3 *Transducer rotational inertia loading*

Spurious moments are generated at each transducer attachment point as a result of the rotational acceleration of the transducer, especially impedance heads which may have a large rotational inertia. Such spurious moments shall be minimized by selecting transducers having low moments of inertia about their mounting points.

6.4.4 *Exciter attachment restraints*

Spurious moments and cross-axis forces are generated at the exciter attachment point by restraints imposed on the rotational and lateral driving-point responses of the structure under test. For example, clamping constraints introduced by the exciter/impedance head assembly could adversely effect the measurement of low-order modes in test structures. The use of area-reducing cones may be required to approximate more closely a point driving force.

NOTE 1 Area-reducing cones can further increase the likelihood of introducing a spurious moment if careful consideration is not given to their use.

Avoidance of exciter attachment restraints is often the most difficult problem encountered when using fixed vibration exciters to measure the mobility of lightweight structures.

To avoid measurement errors caused by attachment restraints, the magnitudes of the lateral and rotational driving-point mobilities of the exciter attachment, when the exciter and attachment hardware are disconnected from the structure, shall be at least ten times larger, at all frequencies of interest, than those of the corresponding elements of the driving-point mobility matrix of the structure itself.

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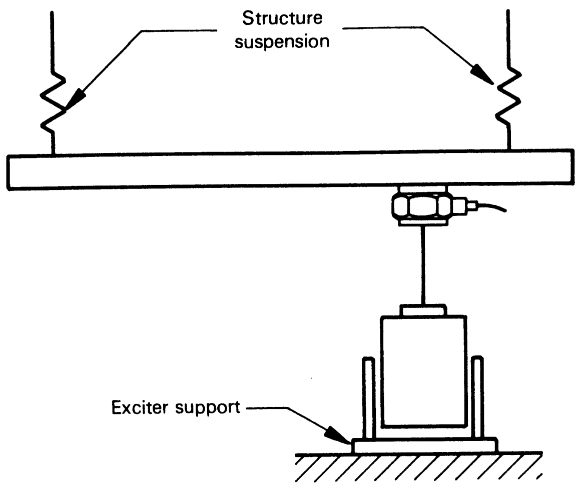
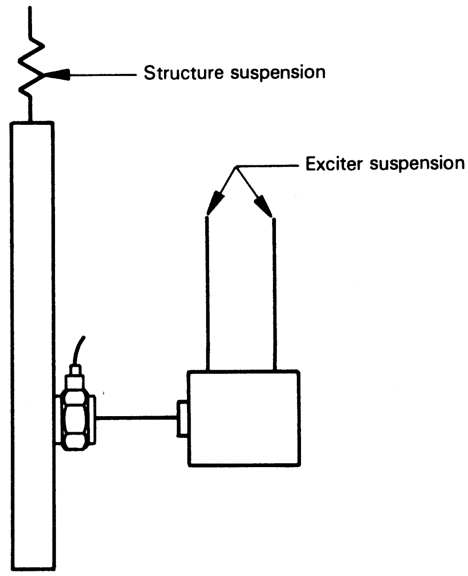
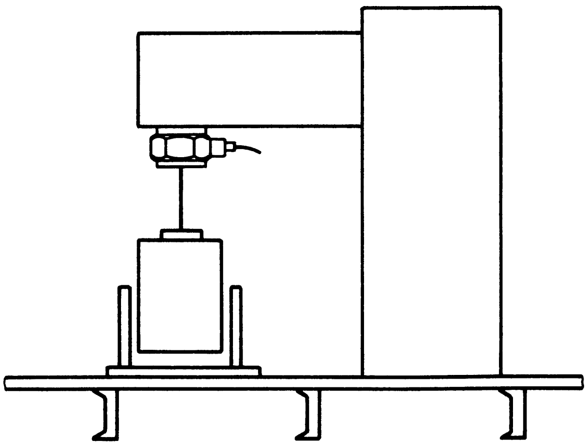


a) Reaction by external support



b) Reaction by exciter inertia



c) Example of an incorrect arrangement

Figure 3 — Exciter reaction forces

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In the absence of quantitative data for either lateral or rotational driving-point mobility, determination of whether a particular test set-up avoids measurement errors caused by significant attachment restraints is often a matter of judgment; the following items shall be taken into consideration:

a) the use of a free-floating voice-coil exciter as described in [2];

b) the design of the support system for an inertia-controlled exciter such that the reaction to the force applied to the structure under test will not result in any rotational motion of the exciter nor in any motion transverse to the axis of the force transducers;

c) the installation of a drive rod connecting the exciter to the force transducers.

The drive rod shall be designed [3] to provide a high stiffness in the axial direction and sufficient flexibility in all other directions. Slender short rods are frequently used for this purpose; however, thick rods with thin flexible sections near each end may give better results. Care shall be taken to ensure that the exciter and drive rod are aligned with the force transducer axis.

If flexible drive rods are used, the accelerometer shall be attached directly to the structure in all cases. The accelerometer shall not be connected to the structure via intermediate devices, such as drive rods, the axial compliance of which would render motion-response measurements invalid [see Figure 4a)]. The force transducer shall be arranged so that it always measures the force transmitted from the drive rod to the structure [see Figure 4b)]. Only with extreme caution can the force transducer be located at the exciter end of the rod [see Figure 4c)]. If the arrangement illustrated in Figure 4c) is unavoidable, the effect of the drive rod compliance shall be checked as described in ISO 7626-1 and the compensation for the rod mass shall be applied using the procedure specified in **7.3**.

NOTE 2 Drive-rod bending modes having natural frequencies within the frequency range of interest may interfere with the mobility test.
Furthermore, bending vibrations of the moving systems of the exciter may introduce moments into the structure which are not detected by the force transducer, but which can affect the response measurements.

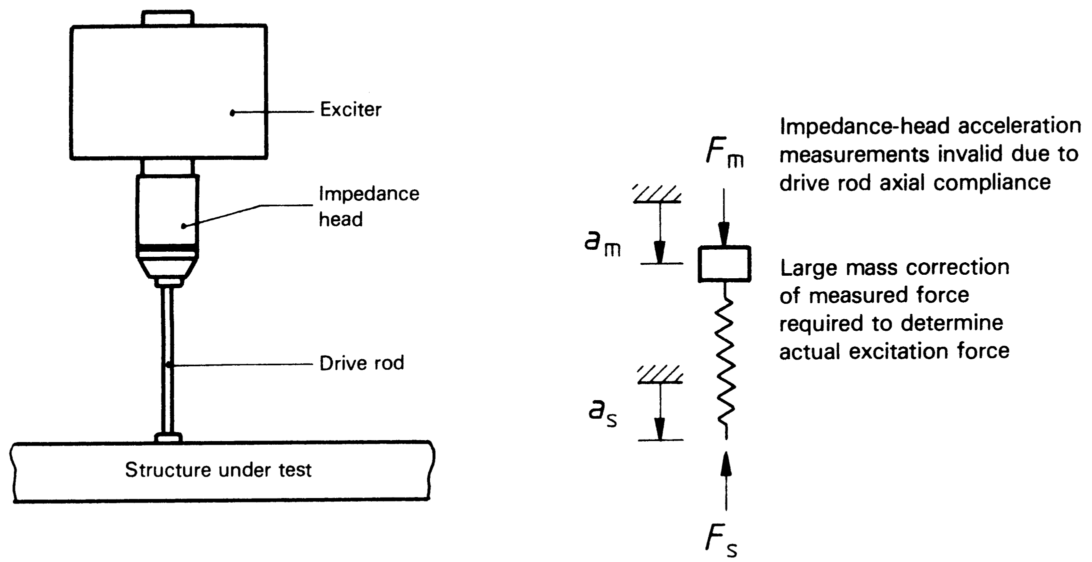

Figure 4a) — Connection of exciter drive rod — Incorrect method

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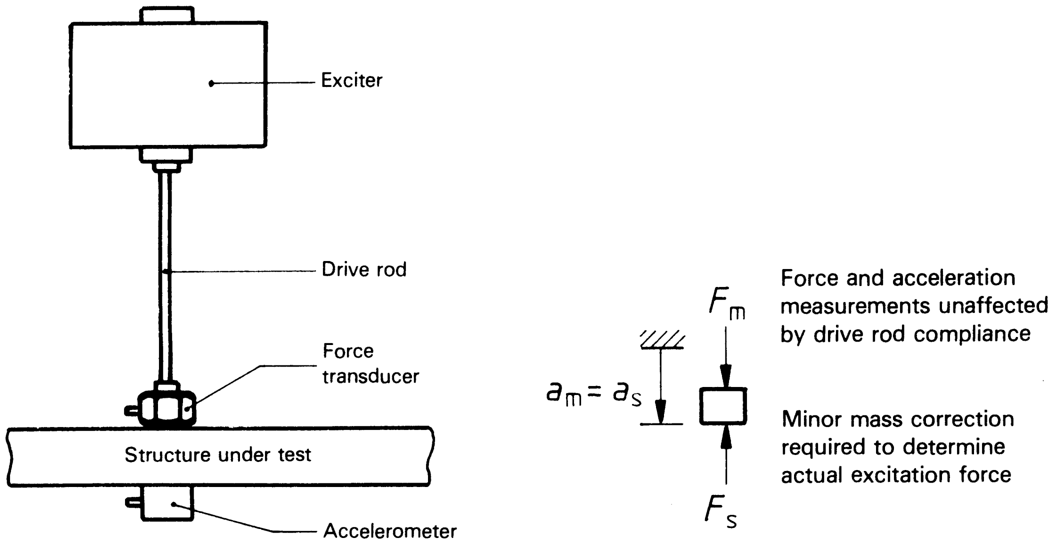

Figure 4b) — Connection of exciter drive rod — Best method

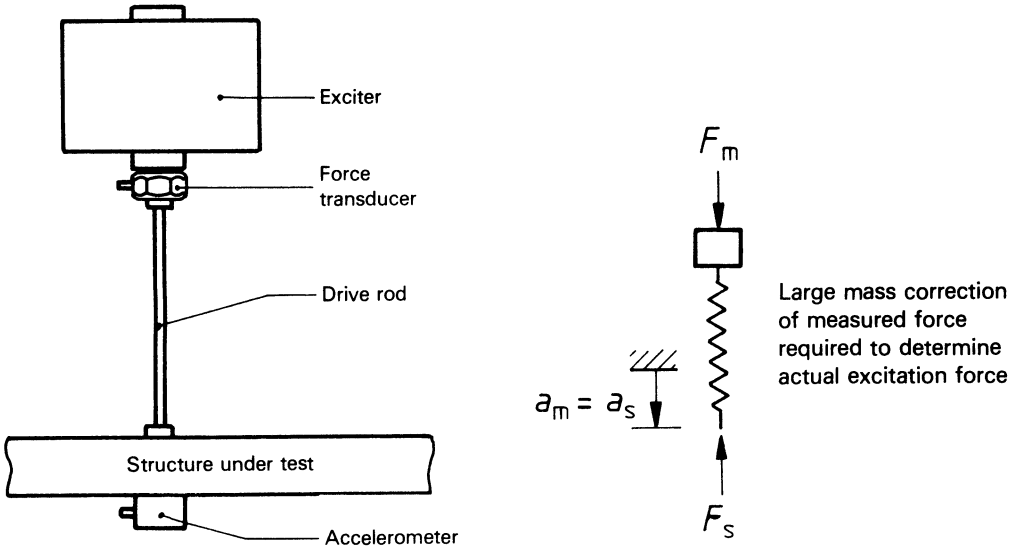

Figure 4c) — Connection of exciter drive rod — Compromise method A

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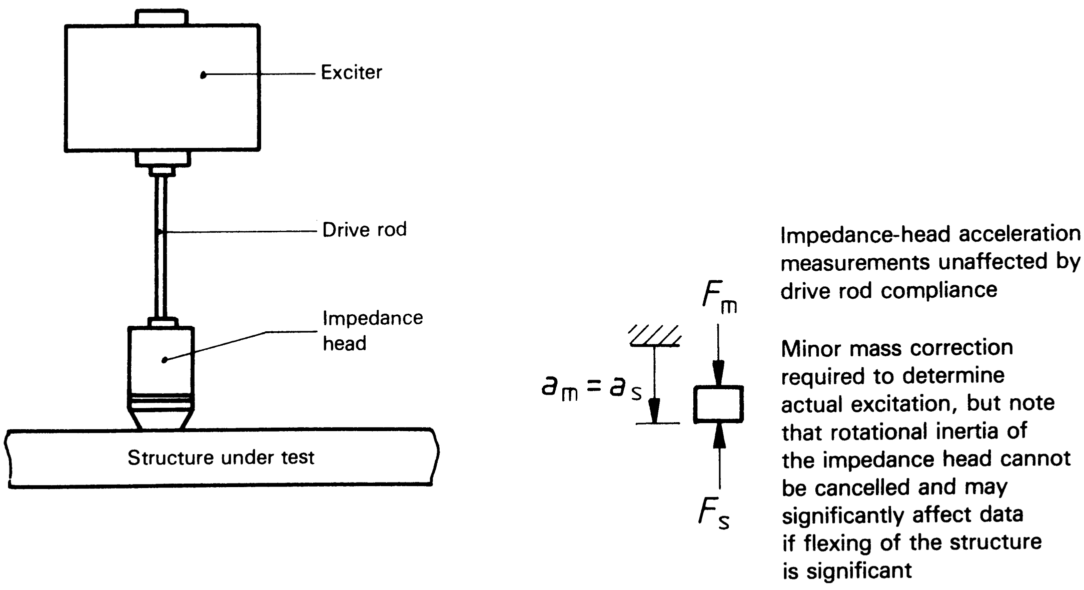

Figure 4d) — Connection of exciter drive rod — Compromise method B

7 Measurement of the exciting force and resulting motion response

7.1 General

Basic criteria and requirements for the selection of motion transducers, force transducers and impedance heads, and methods for determining the characteristics of those transducers are specified in ISO 7626-1. Measurements of exciter current or voltage shall not be used to infer excitation force amplitudes; excitation forces shall always be measured by a suitable transducer.

The types of transducers most commonly used in structural frequency-response measurements are piezoelectric accelerometers, piezoelectric force transducers, and impedance heads combining those devices in one assembly. Displacement or velocity transducers may be used in lieu of accelerometers. Some displacement transducers offer the advantage of a non-contacting design. Piezo-resistive accelerometers have certain advantages when impulsive excitation waveforms are used. Care shall be taken to ensure that the frequency response and linear range of any candidate transducer are sufficiently broad.

Any of the three types of motion (displacement, velocity and acceleration) can be determined using any type of motion transducer by multiplying the measurement result, at each frequency *f*, using the following factor raised to the appropriate positive or negative integer exponent:

$$j\,2\,\pi\,f$$

where

$j = \sqrt{-1}$

f is the frequency concerned.

7.2 Attachment of transducers

Two methods commonly used to attach force and motion transducers to structures are threaded studs and cement; detailed guidance on transducer attachment methods is given in ISO 5348 and in [4], [5] and [6].

The excitation force shall be transmitted as directly as possible through the force transducer or impedance head to the structure with as few intervening components as practical. If the surface of the structure is not flat at the transducer attachment points, appropriately shaped metallic mounting pads shall be used. A thin film of a viscous fluid (such as heavy oil or grease) between the transducer and the mounting surface may improve the attachment coupling at high frequencies. The effects of attachment compliance should be checked as described in ISO 7626-1. Force transducers mounted on studs shall be tightened to the torque recommended by the transducer manufacturer.

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7.3 Mass loading and mass cancellation

As outlined in **6.4.2**, the amount of mass added to a structure for test purposes shall be minimized. When testing lightweight structures, it may be desirable to compensate electronically for the total effective mass, m_t, of the transducers and attachment hardware at the driving point of the structure. Electronic compensation for this mass shall be considered when the magnitude of the driving-point mobility of the structure being tested is greater than $0,01/fm_t$ at all frequencies, f, in hertz, within the frequency range of interest; the mass, m_t, in kilograms, is the sum of the mass of the hardware used to attach the force transducer to the structure being tested and the effective end mass of the force transducer or impedance head as defined in ISO 7626-1.

If the above criterion cannot be met, the following compensation procedure, commonly known as "mass cancellation", can be considered. The acceleration signal at the point of excitation is obtained and multiplied, either in an analogue circuit or digitally, by the total effective mass to be compensated. The product represents that part of the exciter output force which is required to accelerate the effective mass added to the structure for the purpose of carrying out the test. This force signal is subtracted, either in an analogue circuit or digitally, from the force transducer signal to obtain the net exciting force acting on the structure under test.

NOTE 1 If a separate driving-point accelerometer is used below the force transducer [see Figure 4b) and Figure 4c)], the mass of the accelerometer shall also be included in the determination of the total effective mass, m_t.

NOTE 2 During driving-point mobility measurements, the accelerometer used for measuring the response of the structure also provides the signal for determining the force required to accelerate the effective mass. During transfer-mobility measurements, however, a separate accelerometer at the driving point of the structure is required in order to obtain the signal to be used for mass cancellation.

NOTE 3 Electronic mass cancellation cannot compensate for rotational inertia loadings; it can only compensate for translational inertia loadings at the driving point and in the direction of the excitation. All other spurious forces can only be minimized by choosing transducers of low inertia. Measurement errors caused by uncompensated inertia loadings of the structure include shifts in the frequencies of the response peaks.

NOTE 4 It is strongly recommended that reconsideration of the transducer selection and redesigning of the attachment hardware be given higher priority than the use of mass cancellation. In addition, to avoid large measurement errors, mass cancellation should only be used in the range where the ratio of the effective mass of the attachment hardware and transducers to the free effective mass of the test structure at the driving point is greater than 0,06 and less than 0,5.

7.4 Signal amplifiers

Piezoelectric force and motion transducers require charge amplifiers or high-impedance voltage amplifiers.

NOTE 1 Some piezoelectric transducers are equipped with built-in electronic circuitry, requiring amplifiers which are compatible with this circuitry.

NOTE 2 The sensitivity of a voltage amplifier tends to be affected by the impedance of the transducer cable. Voltage amplifiers have more severe low frequency response limitations than charge amplifiers.

7.5 Calibrations

7.5.1 *General*

Requirements for basic and supplementary transducer calibrations are given in ISO 7626-1; both are essential for determining the suitability of piezoelectric transducers for mobility measurements. The basic calibrations of each transducer should be performed once a year.

The actual overall measurement system shall be calibrated at the beginning of each day's test series by performing the operational calibration described in **7.5.2**. The operational calibration shall also be checked at the end of each test series. Additional calibrations may be performed during the test, as required.

7.5.2 *Operational calibration*

Operational calibrations shall be performed by measuring either the mobility or the accelerance of a freely suspended rigid calibration block of known mass. All components of the measurement system shall be connected in the same manner as they will be in the test series. The measured frequency response for the calibration block shall agree within ± 5 % of its known correct value, for example with the magnitude of the accelerance $1/m$ or with the magnitude of the mobility $1/(2 \pi f m)$, where m is the known mass of the calibration block. The same attachment hardware that is to be used in the measurement series shall be used for the operational calibration so that any errors caused by the attachment compliance may be detected (see ISO 7626-1). The mass of the calibration block shall be selected so that its mobility is representative of the range of mobilities involved in the measurement series. If necessary, several operational calibrations shall be performed with appropriate calibration blocks to cover the range of mobilities involved.

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8 Processing of the transducer signals

8.1 Determination of the frequency-response function

8.1.1 *General*

The motion and force signals shall be processed by an analyser which filters the signals (and performs mass cancellation, if required) and determines the ratio of their magnitudes as well as the phase angle between the signals, both as functions of frequency. The analyser should also perform the mathematical operations which may be required for converting the measured frequency-response function to another type (for example converting accelerance to mobility) (see **7.1**). Processing requirements pertaining to the various excitation waveforms described in **6.2** are specified in **8.1.2** and **8.1.3**.

8.1.2 *Sinusoidal excitation*

The magnitude of the frequency-response function shall be determined by analogue or digital means as the ratio of the phasor amplitudes of the two sinusoidal signals. The phase of the frequency-response function shall be determined by measuring the difference between the phase angles of the two signals.

NOTE If discretely stepped sinusoidal excitation is used, several frequency-response functions may be obtained during a given measurement run by switching a single response channel from one response transducer to the next. If slowly swept sinusoidal excitation is used, only a single frequency-response function per response channel can be measured in a given run.

8.1.3 *Random excitation*

Transducer signals generated with random, periodic random, pseudo-random, periodic chirp or impulse excitation should be processed using digital Fourier transform analysers. As described in ISO 7626-1, the frequency-response function may be obtained by suitable computations such as the ratio of the cross-spectral density between the motion response and the excitation force divided by the auto-spectral density of the excitation force. Estimates of those spectra shall be computed by discrete Fourier transformation of the properly time-domain weighted excitation and response signals (see **8.4.3**). A sufficient number of spectra shall be averaged to achieve at least 90 % confidence that, at each resonance frequency, the random error in the computed driving-point mobility is less than 5 % (see Annex A). At least the same number of spectra shall be averaged when computing the corresponding transfer mobilities.

NOTE 1 When measuring transfer mobilities, it may not be possible to achieve the level of confidence specified above, especially when the response is measured at a point and in a direction where the transfer mobility has a small magnitude. Little is gained in such cases by increasing the number of spectra that are averaged, beyond those required to meet the above criterion for the corresponding driving-point mobility test.

NOTE 2 Two-channel Fourier analysers can obtain only a single frequency-response function in a given measurement run. Analysers with additional response channels may be used if simultaneous measurement of several frequency-response functions is desired.

8.2 Filtering

8.2.1 *Sinusoidal excitation*

The frequency-response function shall be computed using only those components of the response and excitation signals corresponding to the excitation frequency. Suitable filters or synchronous digital sampling shall be used for minimizing noise and harmonics without altering the phase between the excitation and response signals.

NOTE Tracking filters are traditionally used for this purpose. Tracking filters are phase-matched, narrow passband analogue devices which use a heterodyne process to adjust automatically to the excitation frequency. Alternatively, digital signal processing devices may be used to synchronize the data sampling function with the frequency of excitation signal.

8.2.2 *Random excitation*

Filtering to remove noise and harmonics from the excitation and response signals is not feasible when random excitation is used. The signal-to-noise ratio can be enhanced by using band limiting techniques, however. Suitable filters should be used to limit the excitation bandwidth as described in **9.4.3**. Phase-matched, anti-aliasing filters having sharp high-frequency roll-off rates are always required when using digital analysers to avoid errors caused by signal components having frequencies above the highest frequency of the analysis range.

8.3 Avoidance of saturation

Systematic checks on gain settings to avoid saturation of the signal amplifiers are vitally important to ensure the validity of the measurement. The overload indicator of an analyser responds only to saturation which occurs within, but not prior to, the analyser. As shown in Figure 1, an oscilloscope should be used to monitor the signals at suitable stages prior to the analyser unless the preamplifiers are equipped with overload indicators.

NOTE Saturation can be detected visually by the clipped appearance of the waveform as it appears on the oscilloscope.

8.4 Frequency resolution

8.4.1 *General*

The resolution shall be fine enough to resolve all the eigenfrequencies of the test structure within the frequency range of the interest and to evaluate adequately the modal damping.

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8.4.2 *Sinusoidal excitation*

For slowly swept and discretely stepped sinusoidal excitations adequate resolution of resonance frequencies requires that the time rate of change of the excitation frequency be sufficiently slow (see **9.2**).

8.4.3 *Random excitation*

For the excitation waveforms described in **6.2.4** and **6.2.5**, adequate frequency resolution requires sufficiently small frequency increments in the discrete-frequency Fourier transform analysis. The frequency increments (line spacing), in hertz, shall be determined from consideration of the modal density and modal damping of the structure under test. Furthermore, time-domain weighting of the signals using the Hanning or other suitable time-weighting function should be used to maximize frequency resolution (see [7]).

NOTE 1 For structures that have light mechanical damping, the required number of samples of digitized data (i.e. the sample "block size") is large if the spectra of the excitation and response signals are computed for the entire frequency range of interest (i.e. by a "baseband" Fourier analysis). Alternatively, a Fourier analysis limited to a selected frequency band (zoom), or a combination of both, may be used. In either case, the total time (record length) required, in seconds, will be the reciprocal of the required frequency increment (resolution), in hertz.

NOTE 2 Random excitation can be considered as a time series of impulse functions (Duhamel's approach). It is easily understood that the response at the start of the sampling of a block of data is mainly the result of earlier excitation; the response is thus not corresponding (coherent) to the excitation. Near the end of the datablock there is still excitation, but the response is truncated, depending on the ratio of the decay time and the total time for sampling the datablock and the place of the "excitation impulse" in the block; this again produces a bad coherence. The better part of the coherent excitation and response data is found in the middle of the datablocks. The application of Hanning windows is sometimes recommended to improve the coherence of mobility data; however, it remains a compromise inherent to random excitation for mobility measurements.

8.4.4 *Periodic excitation*

Periodic random, chirp and impulse excitations do not suffer from the problems described in **8.4.3**. Since periodic excitation produces a head-to-tail connection of the subsequent data-blocks, the starting transients shift over to the next block and, after a certain time, each datablock includes all the response data. In principle, no averaging is necessary. In some cases, the coherence function may be used to provide an estimate of the extraneous noise and, thus, provide guidance in selecting the appropriate signal-averaging parameters (see Annex A).

9 Control of the excitation

9.1 General

Control of the excitation time is necessary to achieve adequate frequency resolution; control of the excitation amplitude is usually necessary to achieve an adequate dynamic range.

9.2 Time required for sinusoidal excitation

9.2.1 *General*

With either swept- or stepped-frequency sinusoidal excitation, the rate of change of the excitation frequency (or the step size and rate) shall be controlled to achieve the frequency resolution required. Fine resolution is required in the vicinity of resonances (response peaks) and anti-resonances (response dips) of the structure in order to determine accurately the magnitude and phase and to obtain accurate information for calculating natural frequencies and structural damping.

9.2.2 *Discretely stepped sinusoidal excitation*

When making use of stepped sinusoidal excitation, the excitation frequency nearest to each resonance frequency of the structure will differ from the true resonance frequency by as much as one-half of the frequency stepping increment. Thus, the maximum error in the determination of a resonance frequency is half the frequency increment. In addition, the measured magnitude of the peak response of the structure will likely be less than the true resonance peak. Upper bounds on the error are given in Table 1. Errors in the measurements of the true magnitude of the response peaks of a structure lead to excessive estimates of the damping of the structural modes.

The frequency increment in the frequency range of ± 10 % of a resonance frequency shall be chosen so that the measured peak response amplitude and the modal damping ratio are within 5 % of their true values.

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Table 1 — Maximum error in measurements of the magnitude of the motion response of a structure at resonance when using discretely stepped sinusoidal excitation

Ratio of stepped frequency increment to true half-power bandwidth of a structural mode	Maximum error in the measurement of the magnitude of the response peak	
	%	dB
Greater than 1	More than 29,3	More than 3
1	29,3	3
1/2	10,6	1
1/3	5,1	0,5
1/4	3	0,3
1/5	1,9	0,2
1/6	1,4	0,1
1/7	1	0,1
1/8	0,7	0,1

NOTE The errors given in the table were calculated by means of an equation published in [8]; however, additional terms were carried in the computations to improve the accuracy.

Equations for determining the maximum frequency increment meeting this requirement are given in Annex B as a function of resonance frequency and modal damping coefficient. The minimum time duration for excitation at each frequency (dwell time) is also given in Annex B. For frequencies outside the range of ± 10 % of a resonance or anti-resonance frequency, larger frequency increments and shorter dwell times may be used.

9.2.3 ***Slowly swept sinusoidal excitation***

If slowly swept sinusoidal excitation is used, the frequency shall be varied either as a linear or logarithmic function of time. The sweep rate shall always be chosen so that, in the frequency range within ± 10 % of a resonance frequency, the measured magnitude of the motion response of the structure is within 5 % of the true value.

For linearly swept excitation, the maximum sweep rate, $(df/dt)_{max}$, in hertz per minute, shall be as follows:

$$(df/dt)_{max} \leqslant 54\ (f_n)^2/Q^2$$

For logarithmically swept excitation, the maximum sweep rate, $(df/dt)_{max}$, in octaves per minute, shall be as follows:

$$(df/dt)_{max} \leqslant 77{,}6\ f_n^2/Q^2$$

In these two relationships,

f_n is the estimated resonance frequency;

Q is the estimated dynamic amplification (quality) factor of the structural mode of concern at the resonance frequency.

NOTE The two relationships given above are taken from [8], and are intended to ensure that essentially steady-state measurements are obtained.

9.3 Time required for random excitation

The excitation shall be applied and the motion response measured for a duration long enough to permit averaging the number of spectra specified in **8.1.3**.

The number of spectra to be averaged is a function of the signal-to-noise ratio of the measurement system. The coherence function between the excitation force signal and the motion-response signal shall be used to determine the minimum number of spectra that have to be averaged to ensure that the random error is less than 5 % within 90 % confidence limits (see clause **A.1**).

The excitation time, in seconds, required for obtaining each spectrum is the reciprocal of the frequency increment, in hertz, of the discrete Fourier transform (see notes to **8.4.3**).

9.4 Dynamic range

9.4.1 ***General***

The mobility of lightly damped structures typically covers a magnitude range of more than $10^5 : 1$ (100 dB) over the frequency range of interest. Each data channel, in addition to having a maximum working voltage above which saturation occurs, has a minimum voltage below which electronic noise and, for digital systems, noise associated with the digitizing process becomes significant in comparison with the signal. For accurate measurements, the control of the excitation shall be such that the voltages in both channels remain within those bounds. Guidelines for excitation amplitude control to achieve adequate dynamic range for mobility measurements using the various types of excitation waveform are given in **9.4.2** and **9.4.3**.

9.4.2 ***Sinusoidal excitation***

With a constant excitation force amplitude, the maximum dynamic range that can be achieved for a mobility measurement is the dynamic range of the response signal channel of the measurement system (typically approximately 300 : 1 or 50 dB). In order to increase the attainable range, the excitation amplitude should be reduced in the vicinity of each resonance frequency (response peak) and increased in the vicinity of each anti-resonance frequency (response dip). The limitation of attainable dynamic range as a result of the use of constant-amplitude excitation is illustrated in Figure 5 a). With constant-amplitude excitation, the maximum measured motion-response magnitude is less than the true maximum motion response because of amplifier saturation. Likewise, the measurement of the true motion response at structural anti-resonances is limited by electronic noise. The improvement in dynamic range obtainable by proper control of the amplitude of the exciting force is illustrated in Figure 5 b).

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9.4.3 ***Random excitation***

The concept of excitation control described in **9.4.2** and illustrated in Figure 5 b) should also be used when the excitation force has one of the random waveforms described in **6.2.4** or **6.2.5**. As a minimum, the excitation spectrum shall be truncated sharply at the maximum frequency of interest to eliminate excitation and response signals above the maximum frequency of interest. If band-limited analysis is used to increase the frequency resolution of the measurements, the excitation shall be limited by bandpass (or by high- and low-pass) filters to the frequency band selected for the high-resolution measurements (see **8.4.3**).

10 Tests for valid data

When mobility data are measured with random excitation of the vibration exciter, the procedure described in clause **A.1** may be used to estimate the minimum number of spectra that need to be averaged to achieve a specified level of confidence in the results. Additional tests, applicable to all types of excitation, are described in clauses **A.2** to **A.4**. The additional tests provide useful information on linearity, reciprocity, and the general validity of the test results.

If the data are plotted on pre-printed mobility graph paper (see ISO 7626-1), proper alignment of the graph paper with the plotter should be verified, using a procedure appropriate for the plotter being used.

11 Modal parameter identification

The purpose of many mobility tests is to identify the modal parameters of the structure under test. Interpretation of mobility data for that purpose is beyond the scope of this part of ISO 7626; general principles are discussed, however, in Annex C.

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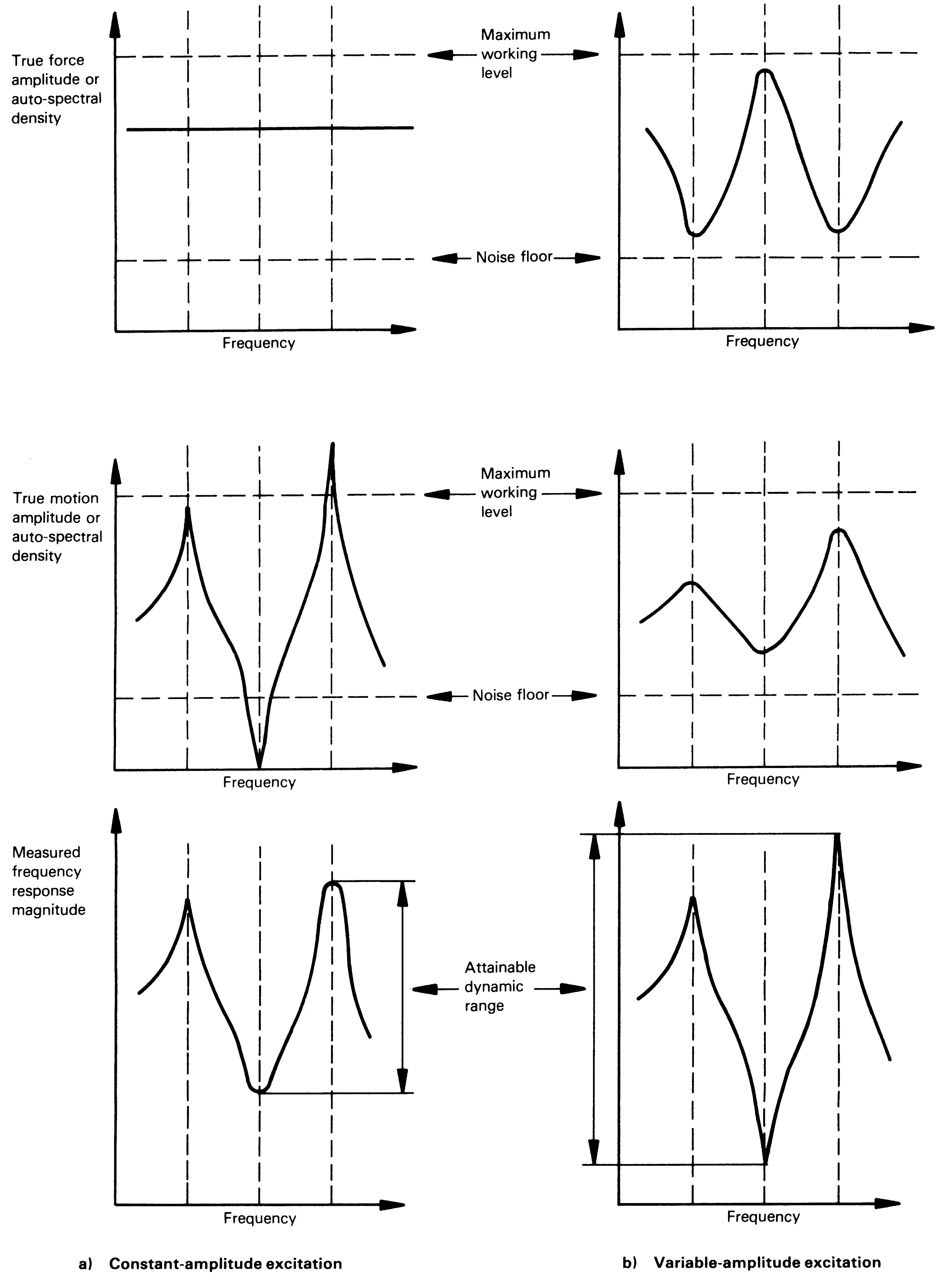

Figure 5 — Illustration of increase in attainable dynamic range through use of variable-amplitude excitation rather than constant-amplitude excitation

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Annex A (normative) Tests for validity of measurement results

A.1 Coherence

When non-sinusoidal excitation (see **6.2.4** and **6.2.5**) is used, the coherence function, $\gamma^2(f)$, between the force signal and the response signal should always be computed as a check on certain potential errors in the computed frequency-response function. At least two ensembles shall be averaged to obtain a usable estimate of the coherence function.

Mathematically, the coherence function, $\gamma^2(f)$, is defined as the ratio of the square of the magnitude of the cross-power spectrum between the excitation force (input) and the motion response of the structure (output), divided by the product of the input power spectrum times the output power spectrum, as given in the following formula:

$$\gamma^2(f) = \frac{|G_{xy}(f)|^2}{G_x(f)\,G_y(f)}$$

By definition, the value of the coherence function is $0 \leqslant \gamma^2(f) \leqslant 1$. A typical coherence function plot is illustrated in Figure A.1.

Substantially lower values of the coherence function are indications of the following problems:

a) Notches in the coherence function at the resonance and anti-resonance frequencies of the structure under test are an indication that either

1) the frequency resolution is inadequate (i.e. the analysis bandwidth is excessive, as discussed in [9]); or

NOTE Inadequate frequency resolution will cause bias errors in the computed frequency-response function. The guidance given in **8.4.3** should be considered for methods of improving the frequency resolution.

2) inadequate time-domain weighting is used in the analyser (see **8.4.3**); or

3) the structure under test has non-linearities (see clause **A.2**) or the signal amplifiers are saturating (see **8.3**); or

4) there are multiple input forces; or

5) there is contamination by noise associated with the digitizing process or by electronic noise, possibly resulting from inadequate force input by the vibration exciter.

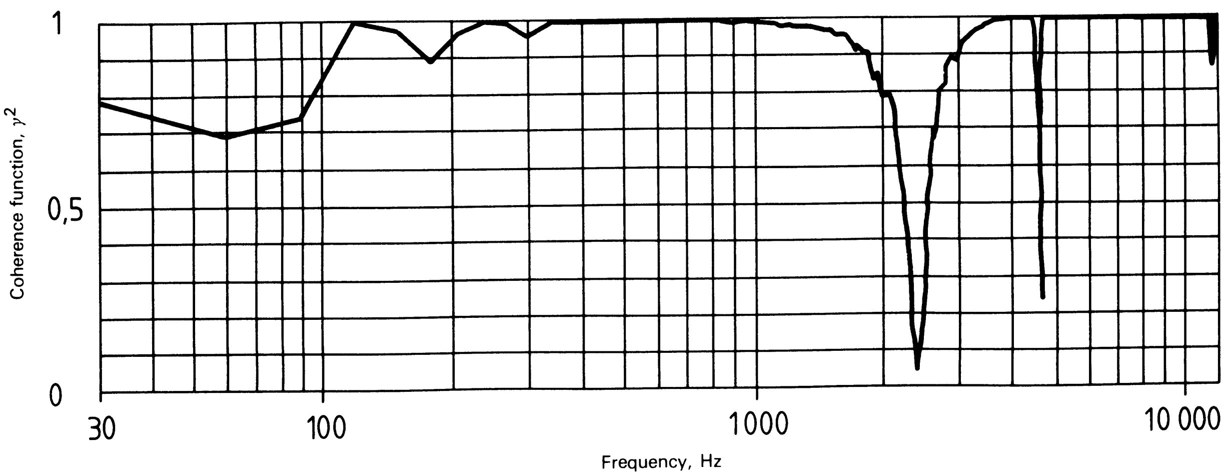

Figure A.1 — Typical coherence function plot

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b) Notches in the coherence function at resonance frequencies can also be an indication that, at that frequency, the exciting force supplied by the vibration exciter drops because of an inherent limitation of the exciter. If the force signal drops below the noise floor, the calculated magnitude of the frequency-response function will be too low if the calculation is based on the ratio of the cross-spectral density between the motion response and the excitation force divided by the auto-spectral density of the excitation force (see **8.1.3**). Possible remedies are discussed in [10].

c) Low coherence over a broad range of frequencies is an indication of a poor signal-to-noise ratio. This is often due to inadequate dynamic range and can be alleviated by suitable shaping of the excitation spectrum (see **9.4**). If the excitation waveforms described in **6.2.5** are used, the signal-to-noise ratio can also be enhanced by synchronous time-domain averaging of the waveforms (see [1]).

d) Non-linearities within the structure can result in low coherence any place in the frequency range. These effects depend upon the type of structural non-linearity encountered.

Poor signal-to-noise ratios lead to random errors in the computed frequency-response function. For a given signal-to-noise ratio (i.e. for a given value of the coherence function), the magnitude of the expected error can be reduced by frequency-domain averaging.

The minimum number of spectra to be averaged, for a 90 % confidence that the random error is below any specified value, may be read from Figure A.2 as a function of the measured value of the coherence function. For example, if the measured coherence is 0,8, then at least 178 spectra shall be averaged to be 90 % confident that the random error of the computed magnitude of the frequency-response function is less than ± 5 %.

NOTE 1 The curves shown in Figure A.2 were derived from an analysis in [11].

NOTE 2 The coherence function cannot be calculated properly from single samples of the excitation and response signals since this always results in a fictitious value of unity.

NOTE 3 A high value of the coherence function does not always prove validity of the data; it could be an indication of cross-talk between the force and response channels.

A.2 Linearity

A.2.1 *General*

The presence of bolted joints, support clearances and other features of practical structures may cause non-linear motion responses. In order to detect non-linear effects, the check procedure described in **A.2.2** should be made in the course of each test series.

A.2.2 *Check procedure*

First measure the frequency-response function using a given schedule of excitation amplitude versus frequency. Then repeat the measurement with the excitation amplitude significantly increased or decreased. If the results of the two tests agree, both adequate dynamic range in the measurement equipment and freedom from non-linear effects have been established. If the results do not agree, the cause should be determined and eliminated.

A.3 Reciprocity

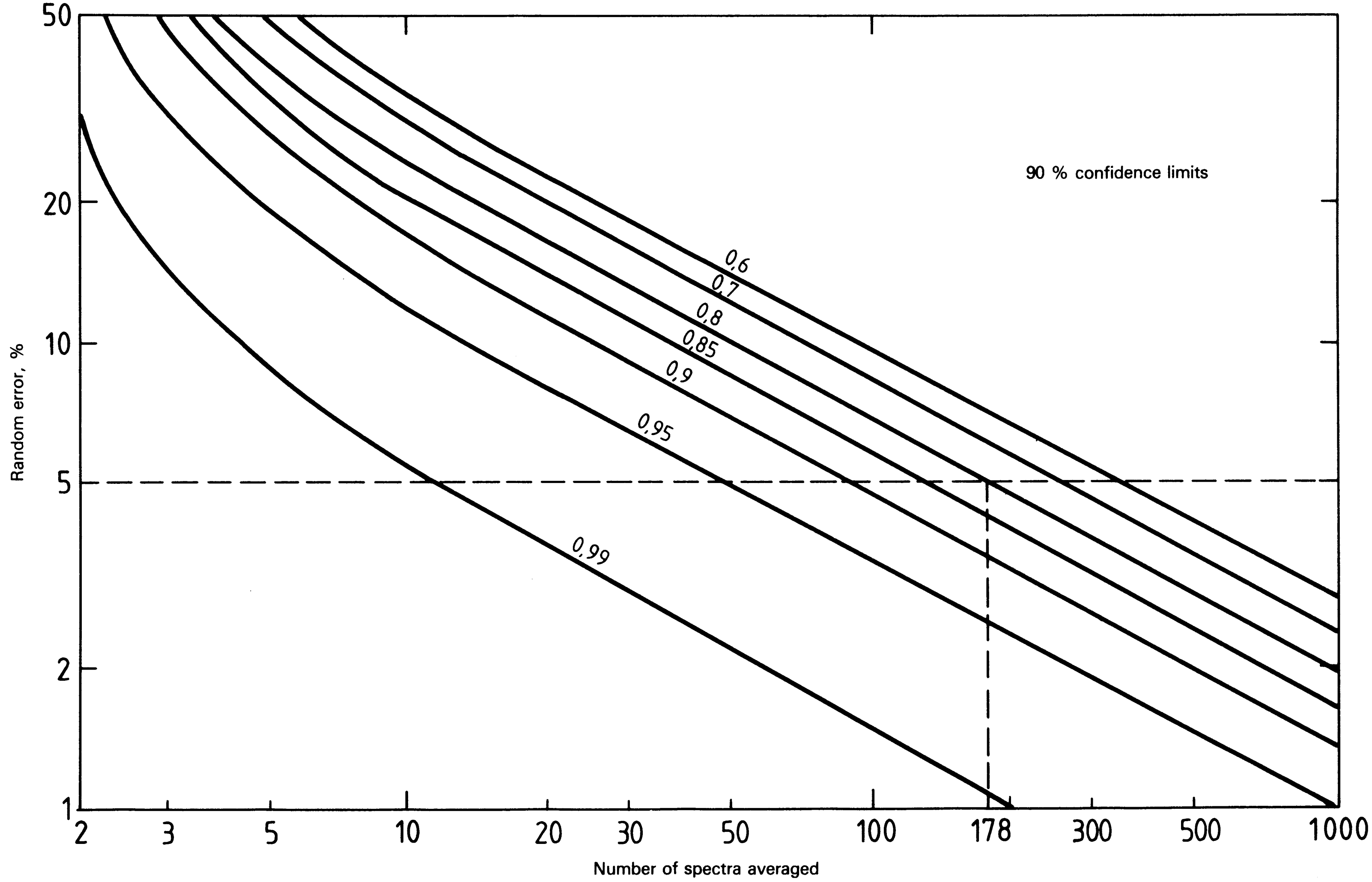
The principle of dynamic reciprocity requires equality between corresponding pairs of transfer mobilities for linear elastic structures. Let a measurement $Y_{ij}(f)$ be performed, establishing the complex ratio of velocity response at a point i on a structure to the force applied at point j at frequency f. The principle of dynamic reciprocity states that $Y_{ij}(f) = Y_{ji}(f)$, where the new measurement $Y_{ji}(f)$ denotes the ratio of the velocity response measured at point j, in the direction of the previous excitation, to a force applied at point i in the direction of the previous response. Agreement between such pairs of measurements for a given linear structure confirms that proper test equipment and procedures are being used.

Although the transfer mobility of most structures exhibits dynamic reciprocity, some system elements, such as hydro-dynamic bearings, do not behave reciprocally. When a non-reciprocal element is located between the driving point and the response point, $Y_{ij}(f)$ will not be equal to $Y_{ji}(f)$.

A.4 Driving-point measurements versus transfer measurements

Any driving-point frequency-response function should exhibit an anti-resonance between each pair of resonances. There is, however, not always an anti-resonance between adjacent resonances in a transfer measurement. If any of the anti-resonances between adjacent resonances is missing in a driving-point measurement, this is an indication of an imperfection in the test set-up (such as a small offset between the force and motion transducers) which should be corrected.

Another check on the validity of driving-point mobility measurements is that the phase angles should always fall between – 90° and + 90°. Transfer mobility phase angles, however, may fall into any one of the four quadrants.



NOTE The value on the curves are the measured values of the coherence values of the coherence function y^2.

Figure A.2 — Number of spectral measurements that need to be averaged to provide 90 % confidence that random error in the magnitude of mobility data does not exceed a specified value, for example 5 %

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Annex B (normative) Requirements for excitation frequency increments and duration

B.1 Frequency increments

In order to determine the response magnitude of the structure under test accurately, at any of its resonance frequencies, it is necessary to measure the response at a sufficient number of frequencies in the vicinity of that resonance frequency. The error caused by the use of a finite number of equally spaced measurement frequencies will fall between zero and the values shown in Table 1 (see **9.2.2**), depending on whether the resonance frequency coincides with one of the measurement frequencies (zero error) or falls midway between two adjacent frequencies (maximum error).

In order to make use of Table 1 to determine the maximum allowable frequency increment in the vicinity of a resonance frequency, f_n, the true half-power bandwidth, B, has to be known. This half-power bandwidth, B, in hertz, is related to the resonance frequency and the estimated amplification factor, Q, of the structure as given in the following relationship:

$$B = f_n/Q$$

From a maximum error of 5 % in the magnitude of the frequency-response function at resonance, the maximum frequency increments, Δf_{max}, within ± 10 % of a structural resonance frequency, should be less than 0,32 B.

NOTE Although Table 1 was derived for discretely stepped sinusoidal excitation, it also gives an indication of the resolution required in the Fourier transform analyser when using other excitation waveforms.

B.2 Duration of the excitation

After each step change in excitation frequency, the motion response of the structure consists of a transient superimposed on the steady response. The amplitude of the transient, $x(t)$, will decay according to the following formula:

$$|x(t)| = X_i e^{-\pi B \tau}$$

where

- X_i is the initial amplitude;
- B is the half-power bandwidth;
- τ is the duration of amplitude decay.

The initial amplitude of the transient may be assumed to be typically less than 10 % of the steady-state amplitude, X_s.

With that assumption, the time required for the amplitude of the transient to decay to less than 5 % of the steady value, $\tau_{0,05}$, may be estimated from the following formula:

$$e^{-\pi B \tau_{0,05}} = \frac{|x(\tau_{0,05})|}{X_i} = \frac{0,05\ X_s}{0,1\ X_s} = 0,5$$

which results in

$$\tau_{0,05} = \frac{\text{In } 0,5}{-\pi B} = \frac{0,221}{B}$$

The excitation shall therefore be applied for at least (0,221/B) s before measuring the motion response at any of the discretely stepped frequencies. At each frequency, the total duration for application of the exciting force is the sum of $\tau_{0,05}$ and the time required for the instruments to measure the force- and motion-response signals.

Annex C (informative) Modal parameter identification

C.1 Requirements for modal parameter identification

Techniques for extracting modal damping and natural frequency values from frequency-response measurements require adequate dynamic range and sufficiently fine resolution of both magnitude and phase data in the vicinity of each resonance of the structure to obtain the necessary definition of the frequency response function, including the peak amplitude. The effect of errors in the measurement of the true amplitude and phase at the resonance peaks may be minimized by using advanced data reduction procedures such as circle fitting (see [12]).

C.2 Methods for determining mode shapes

Using mobility measurements or measurement equipment, vibration mode shapes may be obtained by one of the procedures outlined in **C.2.1** to **C.2.3**.

C.2.1 With a common exciter location, make transfer-mobility measurements for all response locations and directions of interest. Use the measured magnitude and phase data at each resonance frequency to determine the corresponding mode shapes.

NOTE Because of dynamic reciprocity, the same result should be obtained if a single response measurement location and direction were used with various exciter locations and directions (see clause **A.3** for exceptions to dynamic reciprocity).

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C.2.2 First determine the resonance frequencies of the structure under test from a single mobility measurement. Then excite the structure at each resonance frequency, one at a time. While the structure is being excited at a resonance frequency, use a single transducer to measure the magnitude and phase of the motion response at each location of interest, one at a time.

NOTE This method has the disadvantage that modes may be missed if one of their nodes happens to coincide with either the location of the vibration exciter or the location of the motion response transducer.

C.2.3 The methods described in **C.2.1** and **C.2.2** may be modified by using a separate transducer at each location of interest and a multi-channel motion response analyser or similar signal processor.

C.3 Method for obtaining rotational motion responses

A method for obtaining rotational motion responses during mobility tests that use both translational excitations and translation-response measurements is described in [13].

Annex D (informative) Bibliography

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