BRITISH STANDARD

BS EN ISO/TR 11688-2:2001

Acoustics — Recommended practice for the design of low-noise machinery and equipment —

Part 2: Introduction to the physics of low-noise design

The European Standard EN ISO 11688-2:2000 has the status of a British Standard

ICS 17.140.20; 21.020

National foreword

This British Standard is the English language version of EN ISO/TR 11688-2:2000. It is identical with ISO/TR 11688-2:1998.

The UK participation in its preparation was entrusted by Technical Committee EH/1, Acoustics, to Subcommittee EH/1/4, Machinery noise, which has the responsibility to:

— aid enquirers to understand the text;

— present to the responsible international/European committee any enquiries on the interpretation, or proposals for change, and keep the UK interests informed;

— monitor related international and European developments and promulgate them in the UK.

A list of organizations represented on this subcommittee can be obtained on request to its secretary.

Cross-references

Attention is drawn to the fact that CEN and CENELEC Standards normally include an annex which lists normative references to international publications with their corresponding European publications. The British Standards which implement these international or European publications may be found in the BSI Standards Catalogue under the section entitled “International Standards Correspondence Index”, or by using the “Find” facility of the BSI Standards Electronic Catalogue.

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Summary of pages

This document comprises a front cover, an inside front cover, the EN ISO title page, the EN ISO foreword page, the ISO/TR title page, pages ii to iv, pages 1 to 46, an inside back cover and a back cover.

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EN ISO 11688-2

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English version

Acoustics - Recommended practice for the design of low-noise machinery and equipment - Part 2: Introduction to the physics of low-noise design (ISO/TR 11688-2:1998)

Acoustique - Pratique recommandée pour la conception de machines et équipements à bruit réduit - Partie 2: Introduction à la physique de la conception à bruit réduit (ISO/TR 11688-2:1998)

Akustik - Richtlinien für die Gestaltung lärmarmer Maschinen und Geräte - Teil 2: Einführung in die Physik der Lärmminderung durch konstruktive Maßnahmen (ISO/TR 11688-2:1998)

This European Standard was approved by CEN on 17 September 1999.

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This European Standard exists in three official versions (English, French, German). A version in any other language made by translation under the responsibility of a CEN member into its own language and notified to the Management Centre has the same status as the official versions.

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EUROPEAN COMMITTEE FOR STANDARDIZATION
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Management Centre: rue de Stassart, 36 B-1050 Brussels

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Foreword

The text of the International Standard from Technical Committee ISO/TC 43 "Acoustics" of the International Organization for Standardization (ISO) has been taken over as an European Standard by Technical Committee CEN/TC 211 "Akustik", the secretariat of which is held by DS.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by 2001, and conflicting national standards shall be withdrawn at the latest by June 2001.

This European Standard has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association, and supports essential requirements of EU Directive(s).

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Czech Republic, Denmark, Finland, France, Germany, Greece, Iceland, Ireland, Italy, Luxembourg, Netherlands, Norway, Portugal, Spain, Sweden, Switzerland and the United Kingdom.

Endorsement notice

The text of the International Standard ISO/TR 11688-2:1998 has been approved by CEN as a European Standard without any modification.

EN ISO 11688–2:2000

TECHNICAL REPORT

ISO/TR 11688-2

First edition
1998-09-01

Acoustics — Recommended practice for the design of low-noise machinery and equipment —

Part 2:
Introduction to the physics of low-noise design

Acoustique — Pratique recommandée pour la conception de machines et équipements à bruit réduit —

Partie 2: Introduction à la physique de la conception à bruit réduit

Reference number
ISO/TR 11688-2:1998(E)

Contents

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 organisations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The main task of technical committees is to prepare International Standards, but in exceptional circumstances a technical committee may propose the publication of a Technical Report of one of the following types:

— type 1, when the required support cannot be obtained for the publication of an International Standard, despite repeated efforts;

— type 2, when the subject is still under technical development or where for any other reason there is the future but not immediate possibility of an agreement on an International Standard;

— type 3, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example)

Technical Reports of types 1 and 2 are subject to review within three years of publication, to decide whether they can be transformed into International Standards. Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.

ISO/TR 11688-2, which is a Technical Report of type 3, was prepared by Technical Committee ISO/TC 43, *Acoustics*, Subcommittee SC 1, *Noise*.

ISO 11688 consists of the following parts, under the general title *Acoustics — Recommended practice for the design of low-noise machinery and equipment*:

— *Part 1: Planning*

— *Part 2: Introduction to the physics of low-noise design*

Introduction

The objective of this part of ISO/TR 11688 is noise reduction in existing machinery and noise control at the design stage of new machinery.

It is important that non-acoustic engineers are engaged in noise control practice. It is of great importance for these engineers to have a basic knowledge of noise generation and propagation characteristics and to understand the principles of noise control measures.

Acoustics — Recommended practice for the design of low-noise machinery and equipment —

Part 2:
Introduction to the physics of low-noise design

1 Scope

This part of ISO/TR 11688 provides the physical background for the low-noise design rules and examples given in ISO/TR 11688-1[1)] and supports the use of extensive special literature.

It is intended for use by designers of machinery and equipment as well as users and/or buyers of machines and authorities in the field of legislation, supervision or inspection.

Equations given in this Technical Report will improve the general understanding of noise control. In many cases they allow a comparison of different versions of design, but they are not useful for the prediction of absolute noise emission values.

Information on internal sound sources, transmission paths and sound radiating parts of a machine is the basis for noise control in machines. Therefore measurement methods and computational methods suitable to obtain this information are described in clauses 7 and 8 and annex A.

2 References

See ISO/TR 11688-1 and the bibliography.

3 Definitions

See ISO/TR 11688-1 and annex A.

4 Acoustical modelling

In order to facilitate the understanding of complex sound generation and propagation mechanisms in machinery and equipment or vehicles (the latter are also called "machines" in this part of ISO/TR 11688), it is necessary to create simple acoustical models. The models provide a basis for noise control measures at the design stage.

[1)] ISO/TR 11688-1:1995, *Acoustics — Recommended practice for the design of low-noise machinery and equipment — Part 1: Planning.*

A universal approach is to distinguish between

- internal sources;
- transmission paths inside the machine;
- radiation from its boundaries.

The internal sources and the transmission paths can each be assigned to three categories according to the media used:

- airborne;
- liquid-borne;
- structure-borne.

Radiation is considered for air only.

Figures 1 and 2 serve to illustrate the principle of acoustical modelling. Figure 1 shows a simplified machine consisting of an electric motor and a housing with an opening in it.

The motor is the only internal source. It generates airborne and structure-borne sound.

There are three internal transmission paths:

- through the air inside the housing to the opening;
- through the air inside the housing to the walls of the housing;
- through the fastenings to the walls of the housing.

Radiation occurs from the opening and from the walls of the housing.

Figure 2 illustrates this in a block diagram.

The total sound power emitted from the machine is the sum of the three contributions.

A systematic approach starts with an assessment of the relative importance of these contributions. The next step is examining the blocks in Figure 2 looking for possibilities to reduce source strength, transmission and/or radiation (see also following clauses). This should be done in relation to the various aspects of the design process (see ISO/TR 11688-1:1995, Figure 1).

5 Control of airborne and liquid-borne noise

The basic principles of generation, transmission and radiation of sound in air (or other gases) and liquids are basically identical and are therefore considered together in this clause. There is only one important exception: cavitation. Occurring in liquids only, this phenomenon is considered separately in 5.1.3.

5.1 Generation of fluid-dynamic noise

Important noise-generating phenomena in gases and liquids are turbulence, pulsation and shock. Fluid-dynamic processes generate noise if flow rate and pressure vary over time in a limited volume of a liquid or a gas, for example in a turbulent flow. This leads to the transmission of sound from the disturbed volume of the fluid to the surrounding medium. A classic example of this is the escape of compressed air from a nozzle.

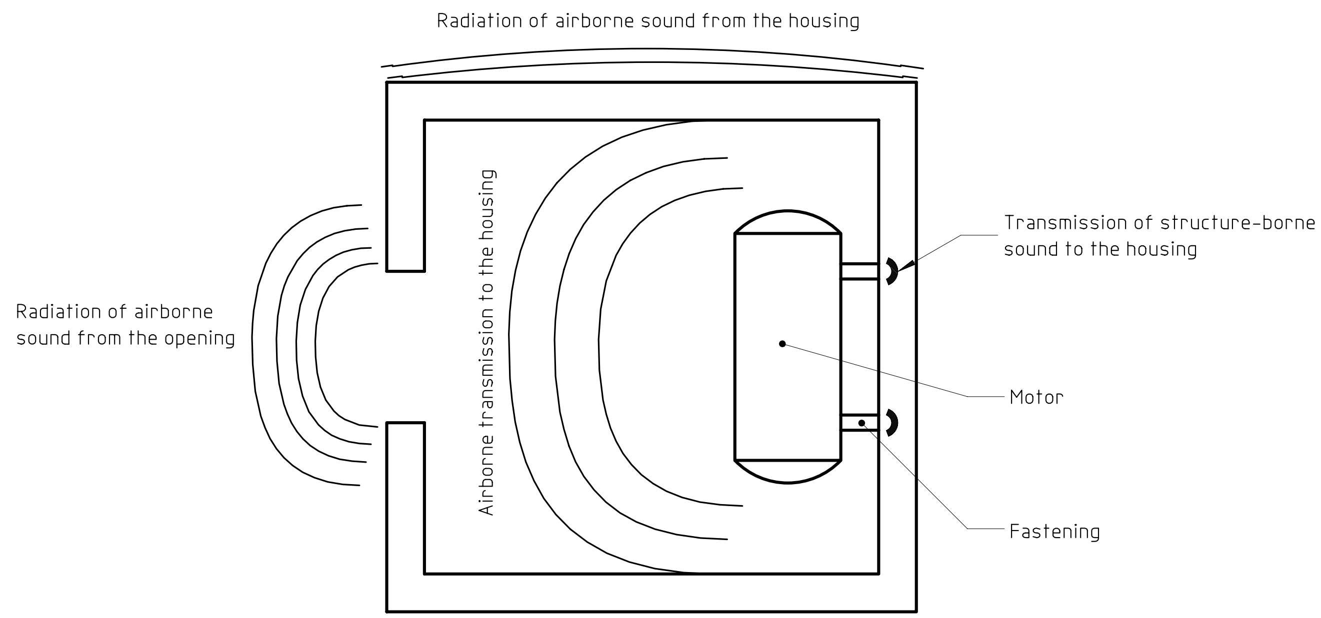


Figure 1 — Simplified machine for the illustration of acoustical modelling

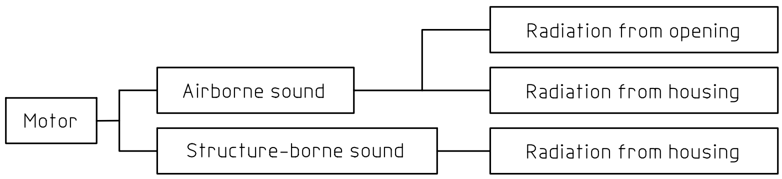


Figure 2 — Block diagram for the illustration of generation, transmission and radiation of sound in the "machine" of Figure 1

Mechanisms of fluid-dynamic sound generation can be related to properties of elementary sound sources with known characteristics:

— monopoles;

— dipoles;

— quadrupoles.

5.1.1 Elementary model sources

A monopole source is an in-phase volume change, such as a pulsating volume of any shape or a piston in a large rigid surface. In the far field, monopoles have a spherical radiation pattern. The sound radiated from a monopole source can be reduced by reducing the temporal variation in the volume flow rate.

EXAMPLE 1: Outlets of internal combustion engines, rotary piston fans, multi-cell compressors, piston pumps, piston compressors, flares.

A dipole source arises as a result of external time-variable forces acting on a fluid without volume change, such as in an oscillating rigid body of any shape. The dipole source can be replaced by two monopole sources of equal strength and opposite phase situated very closely together. The far-field directivity pattern of a dipole is shown in Table 1. Radiation from a dipole can be reduced by reducing the temporal variation of the forces acting on the fluid.

EXAMPLE 2: Vibrating rigid parts of machinery, parts of machinery running out of balance, ducts, propellers and fans.

A quadrupole source can be represented by a time-variable deformation of a body without change of its volume or position. It can be replaced by two dipole sources of equal strength and opposite phase situated very closely together. The far-field directivity pattern is shown in Table 1. Radiation from a quadrupole is reduced when the time-variable deformation is reduced.

EXAMPLE 3: Free turbulent flow as in safety valves, compressed-air nozzles, pipe fittings.

Most sound sources encountered in machinery contain aspects of more than one elementary source.

NOTE Because of the stochastic nature of turbulence the sound spectrum is broad-band. An example is the turbulent flow in the mixing zone of a free jet, particularly for Mach numbers Ma > 0,8. The definition of the Mach number is:

$$Ma = \frac{u}{c} \tag{1}$$

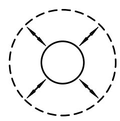
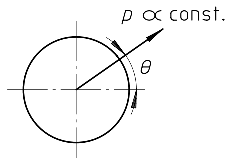
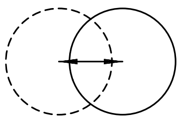
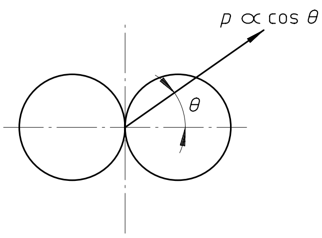
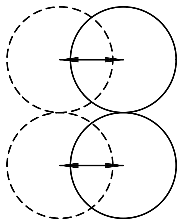
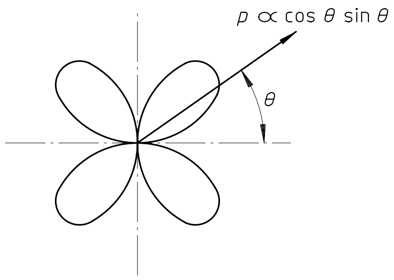
where

u is the flow velocity;

c is the speed of sound.

Table 1 summarizes and illustrates the information on the properties of the elementary sources.

Table 1 — Properties of elementary model sources

Type of source	Schematic illustration	Example(s)	Far-field directivity
Monopole "Breathing" sphere		Siren, piston compressor or pump, exhaust of internal combustion engine, cavitation phenomena, compressed air engine, gas burner	
Dipole Oscillating sphere		Slow machines (axial and centrifugal fans), obstacles in the flow (flow separation), ventilating or air-conditioning systems, ducts with flow	
Quadrupole Two oscillating spheres with an opposite phaseshift (two dipole sources)		Turbulent flow (mixing zone of a free jet), compressed-air nozzles, steam jet equipment, safety valves	

5.1.2 Influence of main parameters

The sound power radiated by aerodynamic sound sources (e.g. the elementary source models monopole, dipole, quadrupole) can be approximated by (see reference [17]):

$$W = \rho D^2 u^3 \left(\frac{u}{c}\right)^k = \rho D^2 u^3 (Ma)^k \tag{2}$$

where

ρ is the density of the liquid,

D is the characteristic dimension of the elementary source,

u is the flow velocity,

k the exponent of the Mach number, which depends on the type of elementary source.

NOTE 1 The following is typical:

— $k = 1$ for a monopole source;

— $k = 3$ for a dipole source;

— $k = 5$ for a quadrupole source.

NOTE 2 Stüber and Heckl [18] have shown that for a three-dimensional sound field and three-dimensional sound propagation the following relationship applies:

$$k = (n - 3) + (2e - 1) \tag{3}$$

where

n is the dimension of the flow field and

e is the parameter of elementary sources (monopole: $e = 1$, dipole: $e = 2$, quadrupole: $e = 3$).

Table 2 shows a summary of the influence of flow velocity and flow field dimension on sound power emission.

Table 2 — Summary of functional relationship between the sound power, W, flow velocity, u, and dimension of flow field, n (see reference [18])

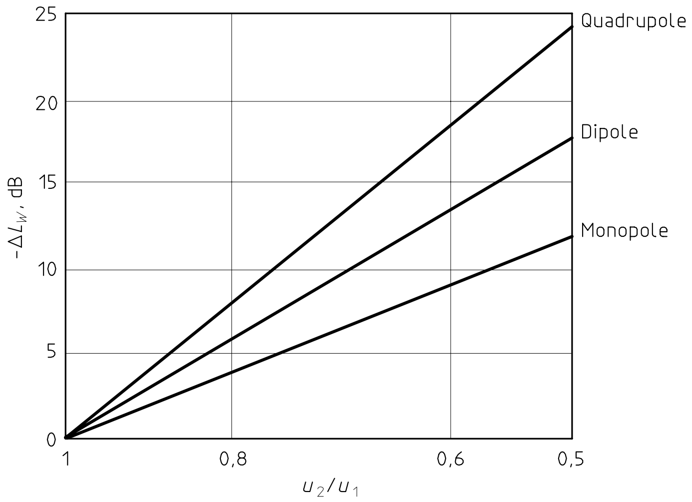
	Dimension n of flow field		
	$n = 1$	$n = 2$	$n = 3$
Mass flow fluctuation (monopole)	$W \sim \rho a u^2$	$W \sim \rho u^3$	$W \sim \frac{\rho}{\alpha} u^4$
Force fluctuation (dipole)	$W \sim \frac{\rho}{\alpha} u^4$	$W \sim \frac{\rho}{\alpha^2} u^5$	$W \sim \frac{\rho}{\alpha^3} u^6$
Turbulence (quadrupole)	$W \sim \frac{\rho}{\alpha^3} u^6$	$W \sim \frac{\rho}{\alpha^4} u^7$	$W \sim \frac{\rho}{\alpha^5} u^8$

Since the sound power of a fluid-dynamic noise source (in a three-dimensional flow field) increases in proportion to the fourth power for a monopole source, the sixth power for a dipole source and the eighth power for a quadrupole source, a reduction in flow velocity leads to a considerable reduction of the sound energy emitted. For machines with rotors, the demand for lower flow velocities also means that lower rotational speeds, i.e. lower peripheral velocities, are required.

Figure 3 shows how the sound power level of a source varies along with a variation of the flow rate. If a characteristic fluid-mechanical value (e.g. mass flow rate, volume flow rate, mechanical power consumption) is to be conserved, a reduction of flow velocity must be compensated by an increase of the characteristic dimension D.

Examples of the characteristic dimension are

- duct diameter for duct flow,
- impeller diameter in flow machines,
- smallest dimension of obstacles in flow,
- diameter of inlet or outlet nozzle.



u_1 = given flow rate; u_2 = reduced flow rate

Figure 3 — Reduction of sound emission by reduction of flow rate
(for three-dimensional sound propagation)

For a simple prediction or estimation of the sound power W of an aeroacoustic sound source mechanism, the acoustic efficiency is an important value:

$$\eta = \frac{W}{W_{\text{mech}}} \tag{4}$$

where W_{mech} is the mechanical or aerodynamic power of the flow.

An empirical estimation for the sound power level is

$$L_W = 120 \text{ dB } + 10 \text{ lg } \eta \frac{W_{\text{mech}}}{W_0} \text{ dB} \tag{5}$$

where W_0 = 1 W.

Examples of acoustic efficiencies in aeroacoustics are summarized in Table 3.

Theoretical methods of high accuracy for predicting or estimating the sound power level or the sound power spectra of fluid-borne sound are not generally available. Equation (2) can be written in a logarithmic form:

$$L_W = L_{W\text{sp}} + 20 \text{ lg} \frac{D}{D_0} \text{dB} + k \cdot 10 \text{ lg } Ma \text{ dB} \tag{6}$$

If the specific sound power level $L_{W_{\text{sp}}}$ is known, acoustical data measured for certain configurations can be scaled using similarity laws, to apply to other configurations with different geometry, dimensions, flow velocities, static pressure levels or flowing media.

For the conversion of spectra, a distinction must be made between broad band and tonal components. The frequency of tonal noise is to be normalized with the Strouhal number St

$$St = \frac{fD}{u} \tag{7}$$

Table 3 — Typical values of the acoustic efficiency

Aeroacoustic sound source	Type of elementary source	Acoustic efficiency η
Piston compressor (radiation in long duct system)	monopole	$\eta = \frac{p'}{\Delta p}$ *)
Siren	monopole	1×10^{-1}
Trumpet	dipole	1×10^{-2}
Propeller aircraft	dipole	1×10^{-3}
Outlet flow (subsonic flow $Ma < 1$)	mixed	$1 \times 10^{-4}\, Ma^5$
Diesel engine (outlet flow noise)	mixed	1×10^{-4}
Gas turbine	mixed	1×10^{-5}
Flow machine (at design point)	dipole	1×10^{-6}
Free turbulent jet	quadrupole	$1 \times 10^{-4}\, Ma^5$
Propeller of a ship with cavitation	monopole	1×10^{-7}
*) p is the maximum value of variable pressure, Δp is the pressure difference.		

5.1.3 Cavitation

Cavitation is a special effect occurring exclusively in liquids. Where local pressure drops below the vapour pressure, cavitation will occur in flowing liquids. Bubbles are generated, which will collapse in a region of higher pressure. This is illustrated in Figure 4. In a flowing liquid the pressure is determined by the Bernoulli equation

$$\frac{u^2}{2} + \frac{p}{\rho} + gz = \text{const} \qquad (8)$$

where

u is the flow velocity

p is the static pressure

ρ is the density of the liquid

g 9,81 m/s^2

z is the height of liquid on top of the region of interest

NOTE $p = \rho g z$.

Equation (8) will allow the determination of low pressure regions where cavitation can occur. When entering a region where the pressure exceeds the vapour pressure, the bubbles implode.

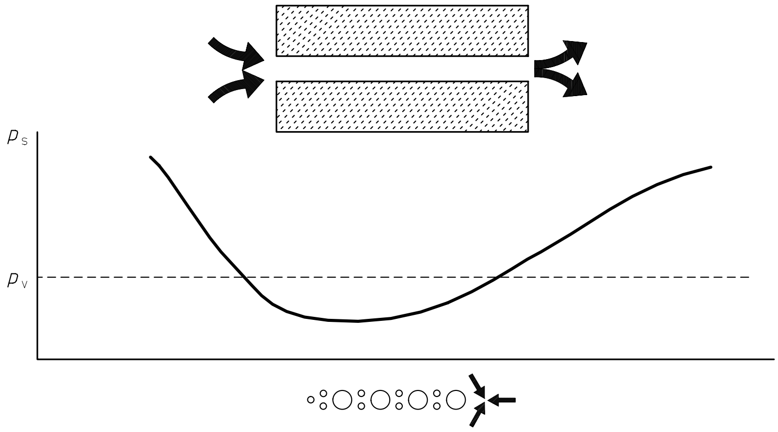


Figure 4 — Generation and implosion of cavitation bubbles

Cavitation is avoided by maintaining low flow velocities in the suction line of the system. In low-pressure tubes, bubbles can continue to exist if no pressure increase takes place after the generation. These bubbles are transported to the reservoir and will enter the pump which results in sound generation. Separation of the bubbles can be effected by placing a mesh in the reservoir between the inlet and outlet. To avoid cavitation, increase static pressure and keep pressure differences low. Cavitation is a monopole source. For further measures to avoid cavitation, see ISO/TR 11688-1.

5.2 Noise control measures

Some noise control measures and mechanisms for the generation of sound are described hereafter by using important fluid-borne industrial sound sources as examples.

Obstacles in the flow

The obstacle is characterized in terms of fluid mechanics by the dimensionless drag coefficient ζ_w, the drag force F_w, which is caused by the flow acting on the body,

$$\zeta_{\mathrm{w}} = \frac{F_{\mathrm{w}}}{\frac{\rho}{2} u^2 A} \tag{9}$$

where, in general, A is the main cross-sectional area of the body.

An analysis of dimension shows that ζ_w is a function of the Reynolds number Re

$$Re = \frac{u \cdot D}{\nu} \tag{10}$$

and the length ratio L/D where L is the length and D is the characteristic dimension of the obstacle. ν is the kinematic viscosity.

For Reynolds numbers Re > 100 the relation between sound power level and drag coefficient for a dipole source is

$$L_W = L_{W\mathrm{sp}} + 10 \lg \frac{A}{A_0} \mathrm{dB} + 30 \lg \zeta_{\mathrm{w}} \mathrm{dB} + 60 \lg \frac{u}{u_0} \mathrm{dB} \tag{11}$$

with A_0 = 1 m^2 and u_0 = 1 m/s.

The specific sound power level $L_{W_{sp}}$ of ventilation grids, for example, is 10 dB.

The above equations show that noise reduction is achieved by

— reduction of flow speed (u),
— downscaling of the bodies in the flow (A),
— disturbance of vortex street (A, ζ_w),
— streamlining of the outer shape of the body (A constant, reduce ζ_w).

Duct and pipe flow

At ducts and pipes with installations (bends, diffusers, changes of cross-sectional area) sound sources are:

— separation which causes secondary flow regions;
— turbulence due to shear layers of different speed (or density).

Separated flow regions and pulsating secondary flows have a dipole source character.

Flow machines

The rotating pressure field of the impeller is one important sound source in flow machines. In centrifugal fans the radial gap between the impeller and the casing is the most important value for (discrete) noise emission. In axial fans the number of impeller and vane blades have a large influence on the sound power level of the blade passing frequency. The tip clearance ratio (the gap between impeller and casing wall) in axial turbo-machines is also important for noise emission.

Generally a flow machine with high aerodynamic efficiency has a low noise emission.

The fan installation has an important influence on sound generation, because disturbed inlet flow profiles causes high pressure fluctuations in addition to the sound of the rotating pressure field.

For Mach numbers $Ma < 0{,}3$ the predominant sound generation mechanisms have dipole character.

Measures for noise control are:

— low tip speed of impeller;
— large casing (centrifugal fans);
— small tip clearance (axial fans);
— no blade and vane numbers which are multiples or submultiples (spinning modes generated by the fan cannot propagate).

Free jets

The noise of a free jet originates in the turbulent mixing region approximately four to five diameters downstream of the outlet nozzle. That means that the sound is mainly generated in the mixing layer through differences in Mach numbers.

Noise control is possible by

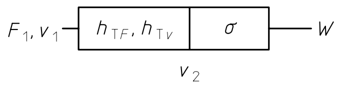
— reducing flow speed (increasing outlet dimensions);
— avoiding obstacles in the flow;
— reducing differences of Mach numbers between outlet flow and surrounding ambient flow field;
— frequency shift by replacing large openings by smaller ones.

6 Control of structure-borne sound

6.1 Model of sound generation

Structure-borne sound is generated when a structure (e.g. a machine housing) is excited by a time-variable force or velocity.

The causal chain of the structure-borne sound generation can be described by the model displayed in Figure 5. According to this model the radiated sound power of a machine can be determined from an excitation function (the force or velocity spectrum) together with quantities representing the vibrational transmission (h_{TF}, h_{Tv}) and radiation efficiency (σ) of an excited structure



F_1, v_1: excitation force and velocity
h_{TF}, h_{Tv}: transmission quantities
v_2: velocity of the radiating surface
σ: radiation efficiency
W: radiated sound power

Figure 5 — Mechanical sound generation

The excitational quantities not only include all quantities arising from mechanical processes such as impacts or unbalances, but all physical processes which can excite vibrations in mechanical structures, such as magnetic flux (electric motors) or non-stationary forces of fluid flow (pumps, internal combustion engines). Excitation by airborne sound is not considered.

Machine structures include housings, frames and claddings as well as moving parts insofar as these transmit and/or radiate sound when the machine is in operation.

For problems of low-noise design, it is convenient to use a simplified model with the following features:

— The excitational quantity is described by the frequency spectrum corresponding to its time function.

— The point of force excitation is characterized by its mechanical impedance or admittance (mobility).

— The passive machine structure is characterized by a frequency-dependent overall transfer function for frequency-bands.

— Complex machine components are simplified by reduction to basic structural elements (plates, beams etc.).

— The structure-borne sound transmission is described by approximations.

— The radiation behaviour of vibrating structures is described by approximations.

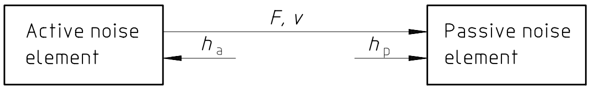
Furthermore the following assumptions are made:

— linearity of the transmission characteristics of the structure; noisy "loose connections" (for example, the rattling of cladding sheets etc.) in the passive structure have to be treated as new sources;

— point-like and unidirectional excitation;

— incoherence of excitations, i.e. the effect of each excitation can be examined separately and the overall effect calculated by energetic addition.

On the basis of these assumptions two special cases of structure-borne sound excitation can be distinguished:



F_1, v_1: excitation force or velocity
h_a: internal source admittance
h_p: load admittance

Figure 6 — Source-receiver model

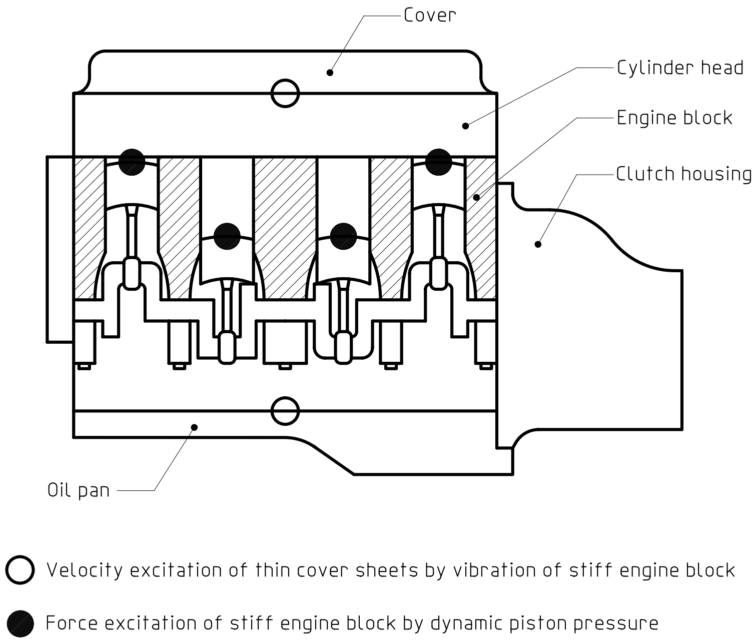


Figure 7 — Schematic cross-section of diesel engine

Force excitation:

The excitation can be described approximately by force alone, i.e. without the complex values of the frequency-dependent admittances h_a and h_p (see Figure 6). This is allowed if the source admittance is much greater than the admittance of the excited system (passive noise element). Thus $h_a \gg h_p$ for high admittance sources (force sources). For mass controlled structures with h = 1/jω m_a it follows that $m_p \gg m_a$. Example: Piston pressure excites engine structure (see Figure 7).

Velocity excitation:

The velocity at the connection between source and load does not depend upon the respective admittance if $h_a \ll h_p$ (for mass controlled systems $m_a \gg m_p$). This is often true at the periphery of machines. Example: Thick-walled cast iron housing excites thin sheet metal (see Figure 7).

The decision "force or velocity excitation" is of great practical importance for the choice of effective noise control measures for machines.

A mathematical description of the model in Figure 5 relies upon the definition of the radiation efficiency σ which is defined indirectly through the radiated sound power:

$$W(f) = \rho c \overline{v^2(f)} S \sigma(f) \tag{12}$$

where

ρc is the characteristic impedance;

ρ is the density of air;

c is the speed of sound;

$\overline{v^2(f)}$ is the spatially averaged mean square value of the velocity of the radiating surface;

σ is the radiation efficiency for a typical frequency characteristic, see Figure 29;

S is the radiating surface area of the structure.

For the case of force excitation an energetic model (Figure 8) can be described by $h_{\mathrm{T}F}^2(f) = \frac{\overline{v}_2{}^2}{F_1^2}$, the squared velocity transmission mobility, which is the squared ratio of the mean velocity of the radiating surface and the exciting force.

For velocity excitation refer to Figure 9 with $h_{\mathrm{T}v}^2 = \frac{\overline{v}_2{}^2}{\overline{v}_1^2}$ the squared velocity transmission. By applying the definition of σ the radiated sound power due to force excitation reads

$$\frac{W(f)}{F_1^2(f)} = \rho c \cdot h_{\mathrm{T}F}{}^2(f)\sigma(f)S = \rho c \cdot h_{\mathrm{T}v}{}^2(f)\frac{1}{Z_1^2(f)} \cdot \sigma(f) \cdot S \tag{13}$$

with $Z_1(f) = \frac{F_1(f)}{v_1(f)}$ the driving point impedance, i.e. the impedance at the point of force excitation. Thus a reduction of the sound power radiated by a force-excited structure can be achieved by

— increasing the driving point impedance;

— reducing the radiating surface S;

— lowering the radiation efficiency σ or the velocity transmission $h_{\mathrm{T}v}$.

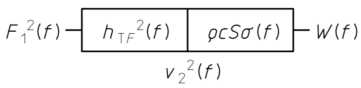


Figure 8 — Energetic model of force excitation

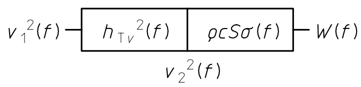


Figure 9 — Energetic model of velocity excitation

In level notation

$$W(f) = \rho c \cdot F_1{}^2(f) \cdot h_{\mathrm{T}F}{}^2(f) \cdot \sigma(f) \cdot S \tag{14}$$

results in

$$L_W(f) = L_F(f) + 10 \lg \frac{S}{S_0} \cdot \frac{h_{TF}^2(f)}{h_{TF_0}^2} + 10 \lg \sigma(f) \tag{15}$$

where $\rho c = (\rho c)_0$ and

$$S_0 = 1 \text{ m}^2, \quad h_{TF_0}^2 = 2{,}5 \times 10^{-15} \text{ m}^2 \text{s}^{-2} \text{ N}^{-2} \tag{16}$$

and:

Force level

$$L_F(f) = 20 \lg \frac{F(f)}{F_0} \text{ dB} \qquad F_0 = 1 \text{ N} \tag{17}$$

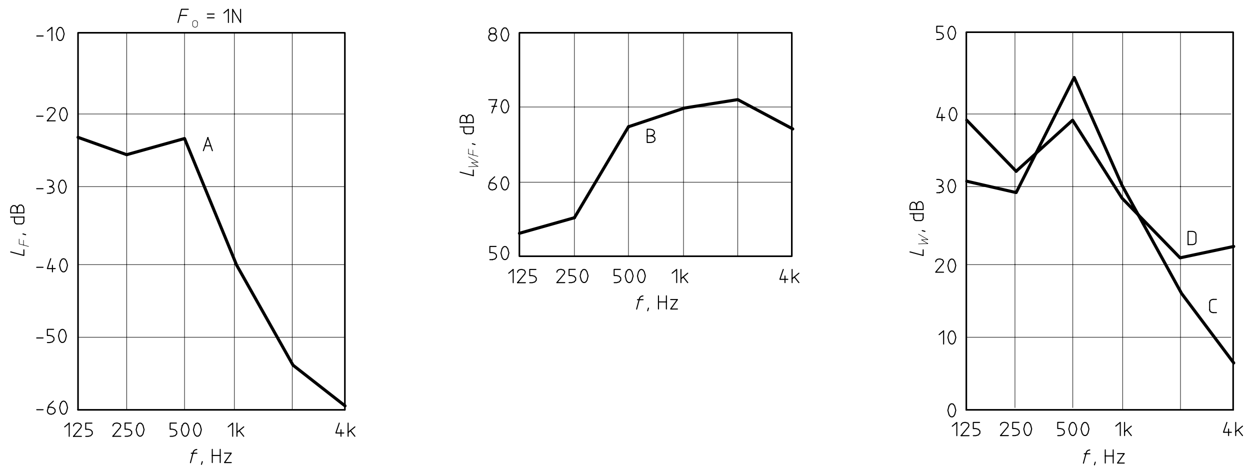
Sound power level

$$L_W(f) = 10 \lg \frac{W(f)}{W_0} \text{ dB} \qquad W_0 = 10^{-12} \text{ W} \tag{18}$$

Introducing the sound power level per unit force L_{WF} to characterize the passive machine structure, this can be re-written as

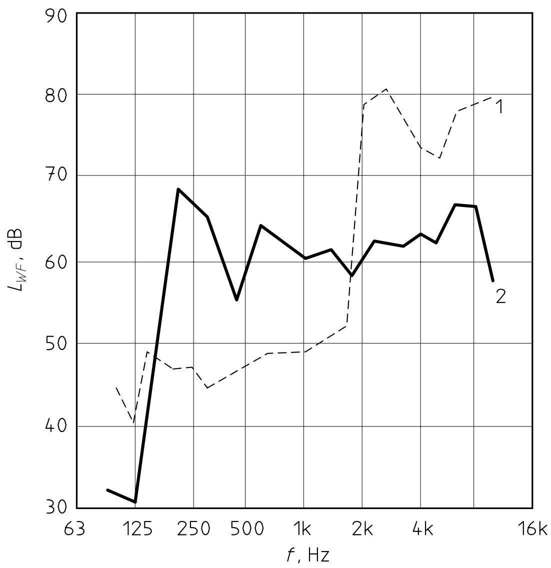
$$L_W = L_F(f) + L_{WF}(f) \tag{19}$$

Because of the frequency-dependent shape of the sound power level per unit force the excitation spectrum is subjected to a frequency weighting. Figure 10 shows a typical variation in shape of an excitation spectrum of an internal source by the function L_{W_F} of the passive machine structure. The result is the spectrum of the radiated sound power level L_W. Typical measured spectra of L_{WF} are shown in Figure 11.



A: measured equivalent exciting force level of the compressor
B: L_{WF} determined(for σ = 1) from measurements at the mounting points of the compressor feet
C: L_W calculated from L_F and L_{WF}
D: L_W measured directly

Figure 10 — Spectra of refrigerator compressor



1 Hermetically sealed compressor capsule, 1,5 mm steel sheet, approximately 0,2 m × 0,2 m × 0,2 m

2 Housing made of cast iron with a thickness between 20 mm and 40 mm, approximately 0,5 m × 0,5 m × 0,5 m

Figure 11 — L_{WF} for different compressor housings

In the case of velocity excitation

$$\frac{W(f)}{v_1^2(f)} = \rho c \, {h_{\mathrm{T}v}}^2(f)\, \sigma(f) S \tag{20}$$

Thus, a reduction of the sound power of velocity-excited structures is achieved by

— minimizing the velocity transmission $h_{\mathrm{T}v}(f)$;

— reducing the radiation efficiency $\sigma(f)$;

— reducing the radiating surface S.

In level notation and for $\rho c = (\rho c)_0$ the equation reads:

$$L_W(f) = L_{v1}(f) + 10 \lg \left(\frac{S}{S_0} \cdot {h_{\mathrm{T}v}}^2(f) \right) \mathrm{dB} + 10 \lg \sigma(f) \quad \mathrm{dB} \tag{21}$$

where L_{v1} is the velocity level:

$$L_{v1}(f) = 10 \lg \frac{v_1^2(f)}{v_0^2} \mathrm{dB}$$

where ($v_0 = 5 \times 10^{-8}$ m/s) and by introducing the sound power level per unit velocity L_{Wv}, a quantity describing the passive machine structure, finally:

$$L_W(f) = L_{v1}(f) + L_{Wv}(f) \tag{22}$$

On the basis of these equations, it is obvious that an easy estimation of the radiated sound power is only possible if it is clear which kind of excitation, force or velocity, is involved.

6.2 Internal sources

6.2.1 Classification of excitation

The classification described here reduces the multitude of excitation systems for mechanical sound generation to five basic ones, which can be clearly distinguished. These are shown in Table 4.

Table 4 — Classification of structure-borne sound excitation

Type of excitation	Subgroups/examples	Measures of influencing excitation
1. Free mass force Line spectrum	Undesired mass forces: Rotors (turbines, electric motors) Desired mass forces: Vibration technology such as conveying, sieving, cleaning, compressing	Lowering rotational speed, improvement of balance, mass balance to the highest possible order Change of technology, e.g. rigid body motion of the compressing container by in phase excitation
2. Impact Continuous spectrum for individual impacts	Impact technologies: Drop forging, percussion riveting, typing, mould sand compacting, transport of rigid objects Impact due to design: Positioning stops, play	Change of technology; or improvement of time function such as time dilation of the impact Using an elastic impact surface (application of resistant rubber), reduction of moved masses and impact speeds, avoidance of play
3. Irregular force curves — Periodic process: line spectrum — Transient process: continuous spectrum	Interaction between machine components: Gear boxes, rolling processes (bearings), electric motors Interaction between machine and work piece: Separation, cutting and forming processes Excitation of machine structure by internal pulsation: Piston machines, internal combustion engines and pumps	Increasing the accuracy of manufacturing and the uniformity, e.g. by skewing (helical gearing) Influencing of the spatial distribution of excitation forces (distance between opposing equal forces small compared to the bending wavelength of the structure)

(continued)

Table 4 *(concluded)*

4. Forces due to non-stationary flow Broad band continuous spectrum for stochastic processes: cavitation Single tone components: Vortex formation Pressure impulses at rotating grids with k elements $f = kn/60$ *)	Flow machines in closed pipe systems: compressors, turbines, pumps	Optimized flow design
5. Self-excitation Pure tone; sudden start or end of high sound level with small change of parameter	Stick-slip processes: Squeaking friction brakes; chisel squeaking of a lathe during cutting, forming, sliding; curving of track vehicles	Meeting the stability requirement of feed-back system; stiffening; stiffening and damping of the chisel and/or work piece lubrication
*) n: rotational speed		

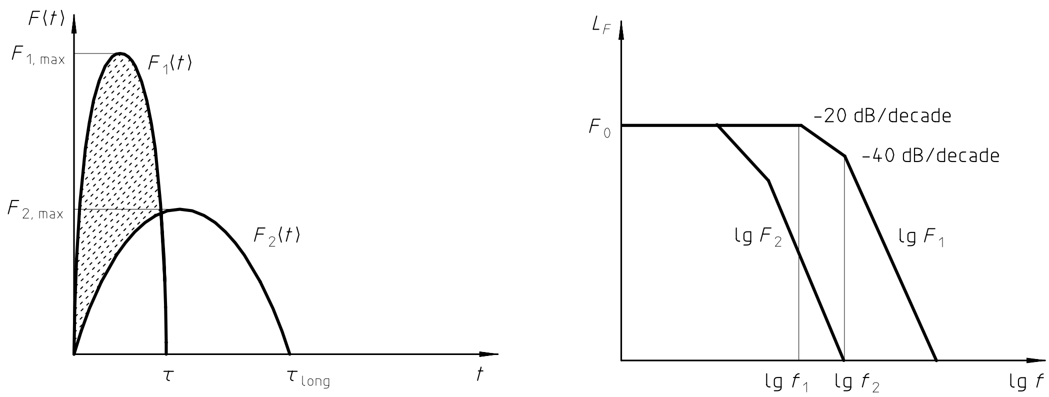
6.2.2 Excitation by various force-time functions

Many excitation processes are force-time functions with a Fourier transform which produces

— a continuous spectrum over the frequency range for a single event (e.g. for an impact)

— a line-spectrum for a periodic process (e.g. for an unbalance).

It is sufficient for an approximate description to determine the enveloping curve for these spectra with characteristic values of the time function. Presented in log-log form (L_F against lg f) this will form straight sections of varying gradients, as shown in Figure 12 and Figure 13.



$$\int_{\tau} F_1(t)\mathrm{d}t = \int_{\tau_{long}} F_2(t)\mathrm{d}t$$

Figure 12 — Schematic spectrum of short impact

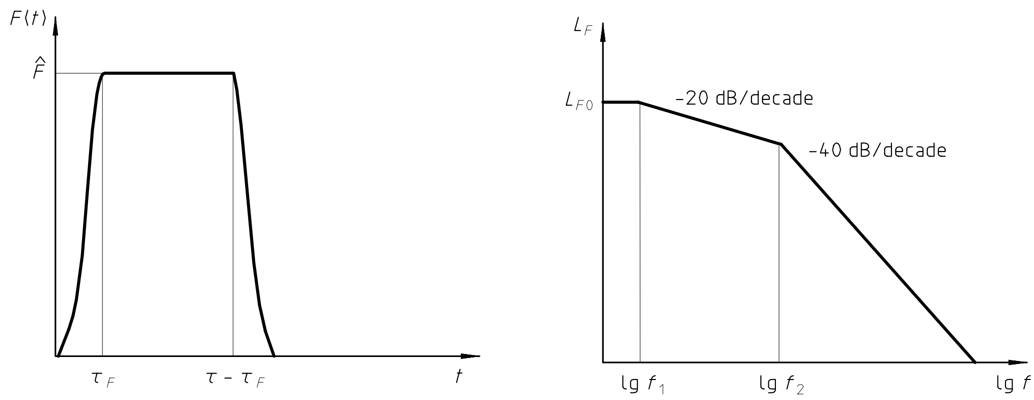


Figure 13 — Schematic spectrum of force-time function with continuous first derivative

The characteristic quantities of the spectrum presented in Figure 12 are obtained according to the following equations:

$$I = \int_0^{\tau} \left| F(t) \right| \mathrm{d}t = F_0 \tag{23}$$

$$L_{F0} = 20 \lg \left| \frac{I}{1\mathrm{Ns}} \right| \quad \mathrm{dB} \tag{24}$$

$$f_1 = \frac{F_{\mathrm{max}}}{2\pi I} \tag{25}$$

$$f_2 = \frac{1}{2\pi\, F_{max}} \left| \frac{\mathrm{d}F}{\mathrm{d}t} \right|_{\mathrm{max}} \tag{26}$$

So for short impacts (with a duration of less than 0,5 ms) of any form (Figure 12) occurring from impact technology, play between moving components, positioning stops etc. the enveloping curve of the excitation spectrum is constant over a wide frequency range (e.g. 0 Hz to 5 kHz for impact durations $\tau \approx$ 0,1 ms). For such impacts influence in the sense of a less exciting process is possible by reducing the masses m involved, the impact speed v, the play width S and the rotational speed n.

$$\Delta L = \left[20 \lg \frac{m_2}{m_1} + 20 \lg \frac{v_{02}}{v_{01}} + 20 \lg \frac{\sqrt{S_2}}{\sqrt{S_1}} + 20 \lg \frac{n_2}{n_1} \right] \mathrm{dB} \tag{27}$$

where

m_2 is the reduced mass

v_2 is the reduced impact speed

S_2 is the reduced play width

n_2 is the reduced rotational speed

(Subscript 1 denotes the original parameters.)

Another way to reduce excitation is to prolong the impact duration τ by the use or introduction of elastic materials. For example: Assuming a half-sinus impulse in Figure 12, a doubling of the impulse duration while maintaining the impulse I results in a reduced excitation of higher frequencies. It should be noted, however, that for impacts which are necessary for technological reasons (compacting, jolt-ramming, impact printers) the necessary peak force is given and it is not allowed to decrease it as it happens when prolonging the impact duration while maintaining I.

For non-continuous force-time-functions, like slow processes ($\tau \geq 2{,}5$ ms, Figure 13) with steep edges, as is typical for piston machines, for cutting processes or pressure curves of pumps, the characteristic gradient of the amplitude envelope of the force spectrum in the acoustically relevant frequency range is –20 dB per decade. A lower excitation of the structure can be achieved by avoiding quick force time changes thus reducing the edge steepness of the force-time curve by for example:

— applying damping slits in pumps;

— avoiding sudden pressure changes in pipes or valves by making area changes smooth and gradual;

— using bevel grinded cutting tools;

— separating or cutting instead of chopping.

The characteristic quantities of the spectrum in Figure 13 are calculated from:

$$L_{F_0} = 20 \lg \left| \int_0^{\tau} \frac{F(t)}{F_0} \, \mathrm{d}t \right| \text{dB} \tag{28}$$

$$F_0 = 1 \text{ Ns} \tag{29}$$

$$f_1 = \frac{1}{\pi(\tau - \tau_{\mathrm{F}})} \tag{30}$$

$$f_2 = \frac{1}{2\tau_{\mathrm{F}}} \tag{31}$$

Continuous force-time functions, like slow processes ($\tau \geq 2{,}5$ ms) which occur with cam drive mechanisms or radial cams show an amplitude spectrum in the acoustically relevant frequency range, which depends on the continuity of the force time function and of its derivatives, respectively. Thus, for a continuous force-time function the amplitude spectrum decreases with –40 dB per decade.

Consequently, measures to reduce the excitation include enhancing the continuity for cam drives and their precise balancing.

Furthermore rolling and sliding surfaces should be very smooth and well lubricated.

To reduce the maximum amplitude of the force spectrum, rotational speeds should be as low as possible, because their influence on the radiated sound power is considerable. For example, a doubling of the rotational speed of a hydraulic pump results in a 15 dB higher sound power level. Therefore, instead of increasing rotational speed, the alternative option of parallel arrays or multi-line operation should be considered for higher performance of a machine.

6.3 Transmission of structure-borne sound

6.3.1 Introduction

The sound radiated by the point or area where the structure is excited is usually negligible as compared to the sound radiation of the boundary surfaces of the machine. The transmission of structure-borne sound from the internal sources to these surfaces must thus be prevented or reduced by suitable measures.

6.3.2 Driving point impedance

The driving point impedance Z characterizes the resistance of a mechanical structure against the excitation by an oscillating force acting at the excitation point.

NOTE 1 The impedance is called point impedance if the dimension of the area where the force is exciting the structure is smaller than one sixth of the relevant wavelength (bending wavelength, shear wavelength).

Assuming a harmonic force excitation where F is the force and v is the velocity at the excitation point, the mechanical power W input into the structure is

$$W = \frac{1}{2} Re\left(F \cdot \overset{*}{v} \right) \tag{32}$$

(The asterisk (*) denotes the complex conjugate.)

$$W = \frac{1}{2} \cdot |F|^2 \, Re(Y) = \frac{1}{2} \cdot |F|^2 \, Re\left(\frac{1}{Z} \right) \tag{33}$$

where

$$Y = Z^{-1} = \frac{v}{F} \tag{34}$$

Y is the admittance (see note 2).

Thus an increase of the driving point impedance Z or a lowering of the driving point admittance Y leads to reduced excitation of the structure and finally to reduced sound radiation.

Assuming a harmonic velocity excitation, the equation for the mechanical W power input reads

$$W = \frac{1}{2} \cdot |v|^2 \, Re(Z) \tag{35}$$

Thus for velocity excited structures the mechanical driving point impedance must be reduced. This can be done by applying springs or other soft isolators.

NOTE 2 The driving point impedance Z as well as the driving point admittance Y are actually matrices because of the multitude of vibrational degrees of freedom of a structure submitted to an exciting force. For practical reasons (measurements) the driving point impedance is normally defined as the ratio of the exciting force to the resulting velocity of the structure in the direction of the force. Thus the admittance $Y = v/F$ of the structure at the driving point is indeed measured, and the reciprocal of the admittance is called the driving point impedance Z. For theoretical investigations Y is the more convenient quantity.

The mechanical driving point impedance is a complex quantity. So, for a freely vibrating rigid mass, $\boldsymbol{Z}$ (see Figure 14) can be written as

$$\boldsymbol{Z} = Re(\boldsymbol{Z}) + jlm(\boldsymbol{Z}) \tag{36}$$

$$\boldsymbol{Z} = \frac{F}{v} = j\omega m \tag{37}$$

where

m is the mass;

$\omega = 2\pi f$.

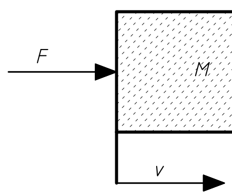
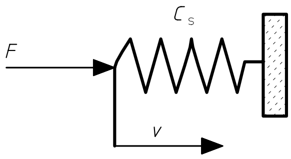


Figure 14 — Force acting on a mass

For an idealized spring (see Figure 15) (no damping, no mass) $\boldsymbol{Z}$ reads

$$\boldsymbol{Z} = \frac{C_s}{j\omega} \tag{38}$$

where C_s is the stiffness of the spring.



NOTE Steel plate, 480 mm × 340 mm × 5 mm, supported at the edges, excited at a point close to the centre.

Figure 15 — Force acting on a spring

Many structural elements of a machine can be approximated by beams or plates. Thus, knowing the impedance of a plate or a beam can be helpful when estimating whether alterations to impedances are effective.

Although the driving point impedance of a plate with defined dimensions and support as seen in Figure16 fluctuates erratically due to the different eigenfrequencies in the relevant frequency range, an averaging over the frequency bands shows a mean dependency which can be described by a simplified equation, if the thickness of the plate, d, is less than one sixth of the bending wavelength λ_B:

$$\boldsymbol{Z}_{\text{plate}} = \omega \rho d \frac{\lambda_B^2}{5} \tag{39}$$

where ρ is the density.

For a beam the simplified equation reads

$$\boldsymbol{Z}_{\text{beam}} = 2{,}67 \rho A \sqrt{c_L \cdot d \cdot f}\,(1 + j) \tag{40}$$

where

A is the cross-sectional area;

c_L is the longitudinal wave velocity;

d is the thickness of the beam.

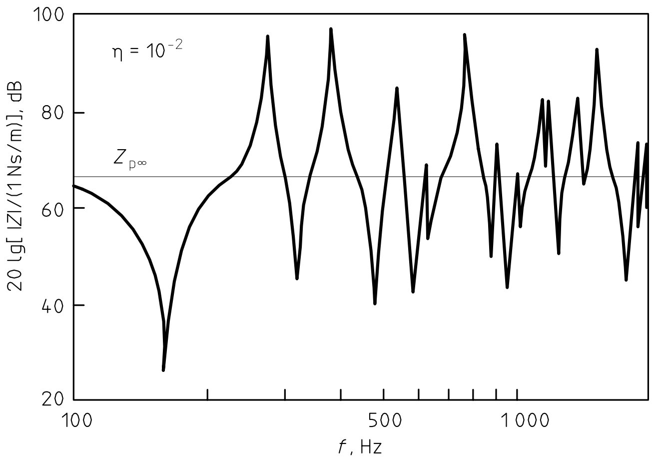


Figure 16 — Characteristic behaviour of a plate supported at the edges

It is assumed that in the case of the plate as well as that of the beam the exciting force is applied perpendicular to the surface of the structural element.

Figure 17 gives an indication of the behaviour of the driving point impedance of the above-mentioned elements.

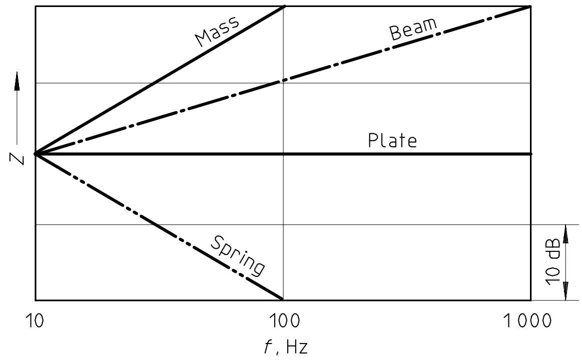


Figure 17 — Characteristic behaviour of idealized structural elements

The easiest way to effect changes in the driving point impedance of a structure is by adding mass to the driven part.

The curve in Figure 18 gives an impression of the driving point impedance of a real structure.

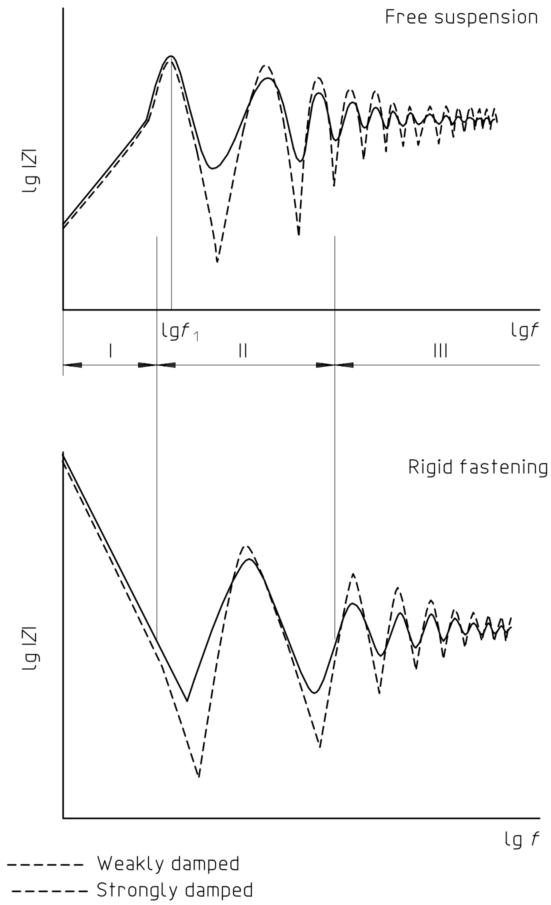
Thus for the lower frequency range the impedance is characterized by a mass ($Z_M \propto f$) or spring ($Z_S \propto 1/f$) behaviour leading to a straight line with a slope of +1 or –1. In the mid-frequency range the structure can be considered as a beam ($Z_B \propto \sqrt{f}$) or plate ($Z_P \neq F(f)$), the respective slopes for the straight lines being 1/2 and 0. For the high frequency range of usual structures the impedance can be approximated by equation (39). For very heavy and stiff structures the impedance is equal to $Z = 2\pi f m_a$ where m_a is the associated mass of a sphere cut from the plate or beam with the radius of one-quarter of the transverse wavelength λ_T

with

$$\lambda_T = \frac{1}{f} \cdot \sqrt{\frac{G}{\rho}} \tag{41}$$

where G is the shear modulus; this means that in the eigenfrequency range the impedance is mainly determined by local characteristics. The resulting straight line for the high frequency range has a slope of –2, and the impedance is proportional to $1/f^2$.

In the lower frequency range (Figure 18, range I) where the machine structure behaves like a mass or spring an additional damping is useless, doubling of the mass or stiffness, however, leads to an increase of 6 dB of the impedance.



Frequency ranges

I: effect of a small component
(top: pure mass, bottom: pure spring)

II: resonant response range, f_1 is the first eigenfrequency

III: multi-resonant response range, effect of a large component

Figure 18 — Driving point impedance |Z| **of a real structure** (schematic) (Note the response ranges and the effect of damping)

A locally applied additional mass and/or stiffness results in a shift of one or more relevant eigenfrequencies in the lower eigenfrequency range leading to a possible weakening of specific sound sources. Use of additional damping may have a similar effect but will mainly lead to a reduction of resonance peaks.

In the multi-resonant response range adding mass or stiffening to specific parts is very effective, e.g. a mounting of additional masses on the surface of the structure leads to a shifting of the first eigenfrequency of the structure to lower frequencies and thus to a reduction of the velocity level on the surface in the resonant response range.

In cases where the mass is attached directly in the line of force, for instance an additional mass at the excitation point on a plate-like structure, the driving point impedance tends to behave mass-like with increasing frequency. Therefore radiation is reduced.

In Figure19 the influence of an additional mass in the line of force is shown for the case of a low-noise design of a thrust bearing.

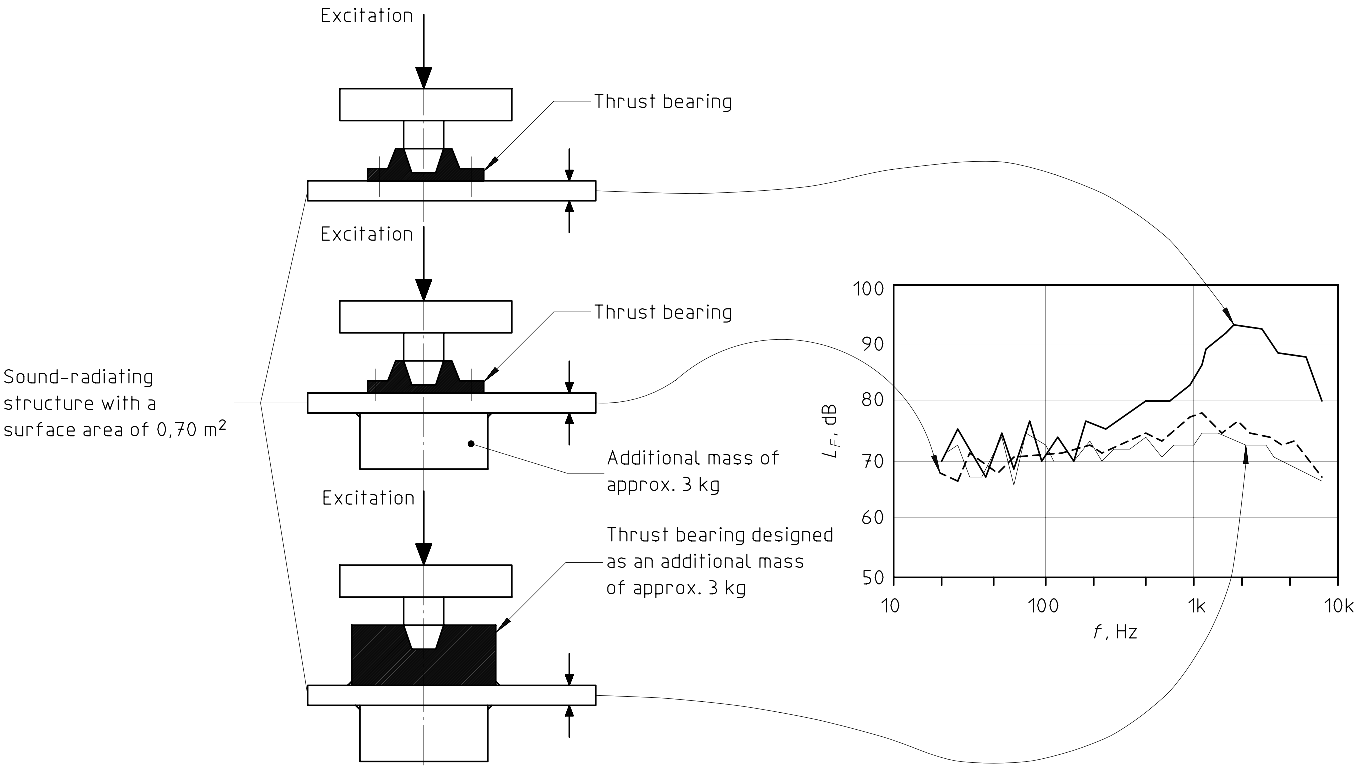


Figure 19 — Reduction of radiation by application of an additional mass to a thrust bearing

The achievable insertion loss D for an additional mass on a plate-like structure for $f \gg f_1$ (f_1 is the first eigenfrequency of the structure) can be described by:

$$D = 10\,\lg\left(1+\left(\frac{\omega\, m}{Z_{\text{plate}}}\right)^2\right)\text{dB} \tag{42}$$

where m is the additional mass.

The increase of local structural stiffness in the multi-resonant response range is only effective in the case of force excitation. For velocity excitation this will have no effect. For the frequency range below the first eigenfrequency f_1, however, stiffening of the structure (e.g. by ribbing the plate surface) reduces sound radiation.

Due to additional stiffening, for $f < 0{,}5\, f_1$ reads:

$$D = 10\,\lg\left(1+\frac{C_{\text{ad}}}{C_{\text{s}}}\right)^2 \text{dB} \tag{43}$$

where

C_{ad} is the additional stiffness;

C_{s} is the initial stiffness of the structure at the driving point.

Introducing a spring and adding mass to the excited structure will convert velocity excitation to force excitation.

Table 5 presents a synopsis of noise control measures for the three response ranges based on changing the driving point impedance.

Table 5 — Noise reduction based on changing point impedance

Type of excitation	Quasi-static	Resonant	Multi-resonant
Force excitation	Add mass, add stiffness	Add mass, add stiffness, add damping	Add mass, add stiffness, add damping
Velocity excitation	Insert a spring and add mass or add stiffness to the structure	Insert a spring and add mass or add stiffness to the structure	Insert a spring and add mass or add stiffness to the structure

6.3.3 General aspects of structure-borne sound transmission

Shape and material of passive noise components in machines determine their behaviour concerning structure-borne sound transmission. For the understanding of general physical laws of structure-borne sound transmission the different shapes of passive components can be substituted, at least partially, by plates or beams, e.g. the walls of a box-like housing by plates. The bending wavelength of a plate is

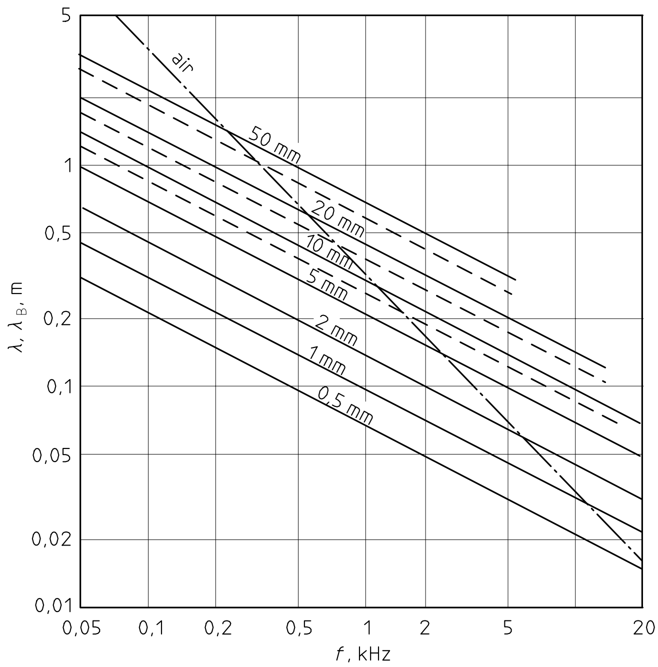
$$\lambda_{\mathrm{B}} = 1{,}4\sqrt{\frac{d c_{\mathrm{L}}}{f}}$$

where

d is the thickness of the plate;

c_{L} is the longitudinal speed of sound, about 5 100 m/s in steel.

Examples can be seen Figure 20.



Key

—— steel

– – – grey cast iron

–·–·– air

Figure 20 — Variation of bending wavelengths for plates and the wavelength of sound in air

If the bending wavelength is small compared to the length and width of the plate (or similarly shaped structure), structure-borne sound energy transmission takes place by wave propagation with all the typical wave effects (reflection, damping etc.). This happens in the multi-resonant response range.

If the bending wavelength is greater than the length and width of the plate, there are no wave effects but in-phase vibrations of the structure, the response of which is then mass-controlled or stiffness-controlled, depending on the kind of excitation and suspension. This is typical for the quasi-static response range.

The transition between both ranges of transmission behaviour is given by the first bending wave eigenfrequency f_1.

NOTE Although the radiation of airborne sound is dominated by transmitted bending waves, longitudinal waves can also play an important part because bending waves can be generated by conversion of longitudinal waves where two plates join at an angle (e.g. floor and wall or two walls).

6.3.4 Control of structure-borne sound transmission by isolation

6.3.4.1 General

One way of reducing the transmission of structure-borne sound is to change the mechanical parameters of the structural elements considerably so that a relevant reflection of structure-borne sound occurs. This can be done by locally increasing or generating an impedance mismatch or mobility mismatch by, for example:

— implementation of soft or elastic elements between relatively stiff structural components;

— application of additional masses in the path of structure-borne sound transmission;

— sudden changes of cross-sectional areas or application of deflections.

Concerning the implementation of elastic elements between structural components remember that

— the stiffness of the elastic element should be much less than that of the structural component;

— for higher frequencies the insulation will not always increase,

— it is recommended to apply an additional mass close to the elastic element (e.g. decoupling of a machine structure from its outer surface plates by a mass-spring system).

In the case of additional masses consider that the added mass must be greater than the local mass of the original structure component.

The effect of sudden changes of the cross-sectional area depends on the type of structural wave travelling through the mechanical structure.

It is important, however, to consider that any reflection of structure-borne sound increases the vibrations in front of the point of local changes. Thus damping must be increased on that side.

6.3.4.2 Isolation by use of elastic elements

If an elastic element is inserted between an active element and a radiating structure (see Figure 21), the effect of this isolation is characterized by the insertion loss D:

$$D = 10 \lg \left| \frac{\boldsymbol{h}_1 + \boldsymbol{h}_2 + \boldsymbol{h}_3}{\boldsymbol{h}_1 + \boldsymbol{h}_3} \right|^2 \text{ dB} \tag{44}$$

Figure 21 — Block diagram illustrating isolation by elastic elements

The isolation will be effective if the mobility $\boldsymbol{h}_2$ of the elastic element is considerably greater than that of the source or the structure to be decoupled, i.e. $\boldsymbol{h}_2 \gg \boldsymbol{h}_1, \boldsymbol{h}_3$.

It is possible to distinguish two distinct cases:

a) decoupling of the source (Figure 22 a);

b) decoupling of the radiating structure (Figure 22 b).

The source is decoupled when the radiating structure has a very small mobility, i.e. $\boldsymbol{h}_3$ is very small, the driving point impedance Z_3 thus very large. The insertion loss D will then read:

$$D = 10 \lg \left| 1 + \frac{\boldsymbol{h}_2}{\boldsymbol{h}_1} \right|^2 \text{ dB} \tag{45}$$

Examples can be seen in motors, gear boxes, pumps, fans which, as sources of structure-borne sound, are all mounted using elastic elements.

For a machine mounted resiliently on a rigid foundation the insertion loss can be calculated using

$$D = 10 \lg \left| 1 - \left(\frac{f}{f_0} \right)^2 \right|^2 \text{ dB} \tag{46}$$

where

$$f_0 = \frac{1}{2\pi} \sqrt{\frac{C_2}{m_1}} \tag{47}$$

with the spring constant C_2 in newtons per metre, N/m, and m_1 the mass of the source in kilograms, kg.

For rubber springs use the dynamic spring constant C_{dyn} instead of the static spring constant C_{stat} for C_2. As a rule, $C_{dyn} = kC_{stat}$, with $1 \le k \le 3$.

A radiating structure is decoupled when its mobility h is large in comparison to the small mobility h_1 of the active element. The insertion loss is then calculated according to the equation:

$$D = 10 \lg \left| 1 + \frac{\boldsymbol{h}_2}{\boldsymbol{h}_3} \right|^2 \text{ dB} \tag{48}$$

or

$$D = 10 \lg \left(1 - \left(\frac{f}{f_0} \right)^2 \right)^2 \text{ dB} \tag{49}$$

with the mass m_3 of the radiating structure to determine f_0.

Examples: Decoupling of an enclosure from the machine inside, or decoupling of a driver's cabin from the vibrating body of a vehicle.

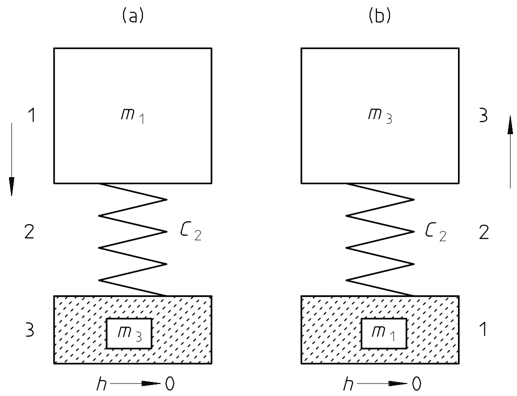


Figure 22 — Decoupling of source and radiating structure

Low-frequency vibrations must also be considered to ensure safe operation and long service life of isolators. If the relevant excitation frequency of the machine is f_{ex}, f_0 should be $f_{ex}/3$ or less. An effective isolation can be expected for the range

$$0{,}47\, f_{ex} < f < 0{,}25 \sqrt{\frac{C_{dyn}}{m_s}} \tag{50}$$

where m_s is the mass of the spring.

The first eigenfrequency of a rubber spring depends on its length. Typical values lie

— between 250 Hz and 500 Hz for 100-mm-long rubber springs, and

— between 1 300 Hz and 1 600 Hz for 15-mm-long rubber springs, and

— between 100 Hz and 500 Hz for steel springs.

Figure 23 shows the frequency characteristic of the insertion loss D of an elastically mounted mass upon a rigid foundation.

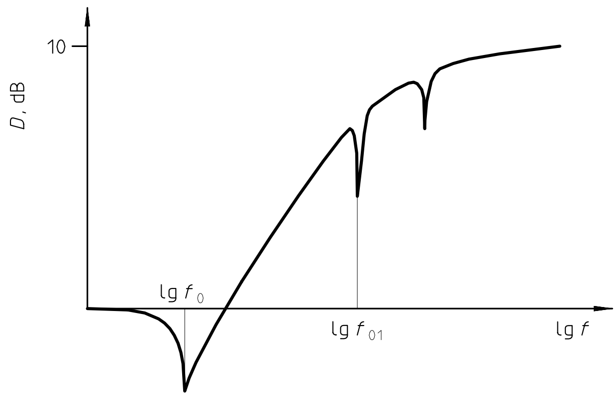


Figure 23 — Insertion loss *D* of an elastically mounted mass upon a rigid foundation (f_{01} is the first spring eigenfrequency)

6.4 Control of structure-borne sound transmission by damping

In the acoustical frequency range metals like steel, grey cast, aluminium, etc. show a poor internal damping. Thus, plate-like constructions can easily be excited to perform flexural vibrations.

Increasing the damping of the structure is therefore a widely applied method to reduce the velocity transfer function h_{Tv} and the radiation efficiency.

Damping is characterized by the loss factor η, which is defined as

$$\eta = \frac{W_{\text{diss}}}{2\pi W_{\text{rev}}} = \frac{\text{energy dissipation per vibration period}}{2\pi \times \text{reversible mechanical energy}} \quad (51)$$

For some materials like metal or concrete η does not depend on the frequency, temperature or the type of wave propagation. For plastics, however, it can vary considerably with temperature and to some degree also with frequency. Table 6 shows loss factors for different materials (temperature = 20°C, mid-frequency range).

Table 6 — Loss factors for different materials of freely vibrating plane plates

Material	η
Steel, aluminium, brass	$\approx 10^{-4}$
Cast iron	$\approx 10^{-2}$
Special copper-manganese alloys	$\approx 3 \times 10^{-2}$
Plastic (construction material)	$\approx 10^{-2}$ to 10^{-1}
Glass fibre reinforced polyester	$\approx 10^{-2}$
Plexiglass	$\approx 2 \times 10^{-2}$
Concrete	$\approx 10^{-2}$
Glass	$\approx 10^{-3}$ to 10^{-2}

The increase of η can only be effective in the resonant response range. It leads to a shorter decay time after impact excitation and to a reduction of resonance vibrations for steady state excitation. For large and complex structures not only the increase of η is important, but also the location where the damping treatment is applied.

As a rule of thumb one may state that damping treatments are most effective on plates close to the source resulting in a reduced transmission and on plates that radiate sound to the environment owing to a reduction of radiation.

There are, in principle, two options to increase the damping:

— using material with a high internal damping;

— adding special damping treatment to the structure.

Although cast iron or polymeric materials have a much greater loss factor than metals like steel or aluminium their application does not always result in structures radiating less sound. This owes to the much larger loss factor of complex structures as compared to the loss factor of the materials from which they are made (see Table 7), because of the losses through friction as in connections or added components.

Table 7 — Typical loss factors of structures

Types of structure	η
Metal structures composed of a small number of thick-walled parts (e.g. ship hull)	3×10^{-3} for $f < 500$ Hz 10^{-3} for $f > 1\,000$ Hz
Metal structures composed of many thick and a few thin-walled parts (e.g. engine, car)	10^{-2}
Metal structures composed of many thin-walled components (small and complex machinery)	5×10^{-2} for $f < 500$ Hz 10^{-2} for $f > 1\,000$ Hz

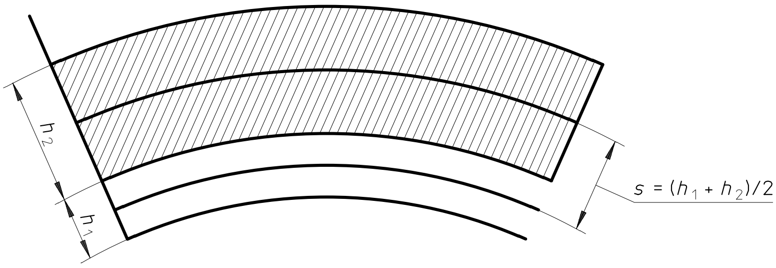


Figure 24 — Geometry of a single-layer damping system

In order to exceed the usual loss factors of "undamped" structures significantly special damping treatments are necessary. For plates there are (see Figure 24)

— damping by single layers;

— damping by constrained layers;

— damping by a plate-to-plate system.

The single-layer damping system consists of a carrier plate to which on one side a visco-elastic layer is attached, which has a loss factor much higher (up to 1 000 times) than that of the metal carrier plate. The damping relies on the fact that part of the vibrational energy in the carrier is needed to bend the visco-elastic layer. From this part of the mechanical energy a large fraction is converted into heat during each vibration period. Therefore, an efficient layer should be stiff to increase the energy needed for bending and have a great loss factor η to increase conversion into heat.

The damping effect of a single-layer system does not only depend on the loss factor η of the layer but also on the ratio d_2/d_1 (layer thickness/plate thickness, see Figure 24) and the ratio of the ratio E_2/E_1 where E_1 and E_2 are Young's moduli for layer and plate, respectively.

The total loss factor η_t of a single layer damping system can be approximated by

$$\eta_{\mathrm{t}} \approx \eta_2 \cdot \frac{E_2 \cdot d_2 \cdot s^2}{B_{\mathrm{t}}'} \tag{52}$$

where

$$s = \frac{d_1 + d_2}{2} \tag{53}$$

and the total bending stiffness

$$B_{\mathrm{t}}' \approx \frac{E_1\, d_1^3}{12} + E_2\, d_2\, s^2 \tag{54}$$

Summarizing the basic ideas from the equations and from Figure 25, one can say:

— the effect of damping increases with the square of the ratio d_2/d_1 of the thicknesses;

— E_2 (Young's modulus of the layer) and η_2 (total loss factor of the layer) must be high.

— the thickness of the layer should be twice or three times the thickness of the metal plate

In cases where the visco-elastic layer is lying between the carrier plate and a top plate one speaks of damping by a constrained layer (Figure 26).

A distinction is made between symmetrical and unsymmetrical designs, the latter, however, being more important for practical use.

The principle of this damping system is the conversion into heat of a large fraction of the vibrational energy that is needed for shear deformation of the visco-elastic layer. In contrast to the single layer system the total loss factor η_t of the constrained-layer system varies significantly with temperature and also much more with frequency (Figure 27). Its calculation is rather complicated.

The major advantage of a constrained layer is that it adds less weight to the machine than a single-layer damping system.

The plate-to-plate damping system consists of two flat plates which are attached to one another by spot welds, bolts or other point-like connections. The damping is based on frictional losses in the residual air layer between the plates. Investigations have shown that damping is increased when plates of different thicknesses are used (a ratio of 3 : 1 is appropriate) and when connection points are more than half a bending wavelength apart. An advantage of this damping method is its applicability even at high temperatures. Corrosion between the two plates may be a problem.

6.5 Radiation

The sound radiation from the surface of a structure is a result of the conversion of vibration of the excited structure into a pulsating compression of the surrounding air. Thus small, compact, vibrating bodies do not radiate sound as efficiently as large vibrating bodies, particularly not in the lower frequency range.

The radiation efficiency σ is the principal quantity for the characterization of the capability of a structure which is excited by structure-borne sound to radiate sound.

$$\sigma = \frac{W}{\rho c \cdot S \cdot \overline{v^2}} \tag{55}$$

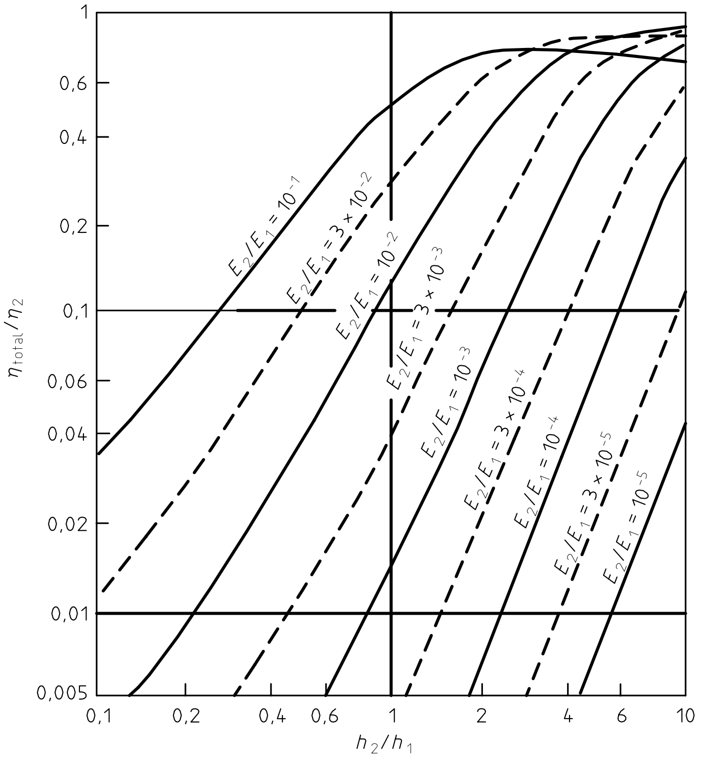


Figure 25 — Loss factors of homogeneous flat plates with single-layer damping

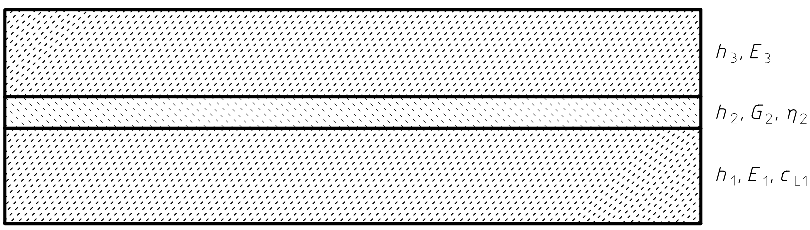
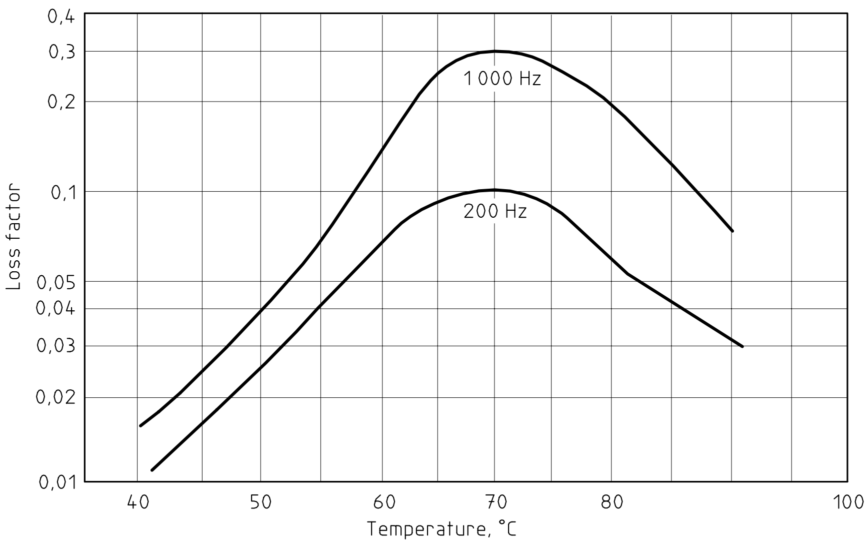


Figure 26 — Geometry of a constrained layer



Carrier plate 4 mm, viscoelastic layer 1 mm, top plate 1 mm.

Figure 27 — Effect of temperature on loss factor of a constrained layer

Depending on the kind of radiator and the frequency range of interest the values of σ range between approximately zero (e.g. perforated plates $\rho = 10^{-5}$) and unity (e.g. in the higher frequency range for oscillating compact rigid bodies like hydraulic pumps). (In level notation, this corresponds to $-\infty$ dB and 0 dB, respectively).

Figure 28 shows measured radiation efficiencies for a number of structures. It is apparent that large differences occur. They owe to the different distribution of vibrations with regard to their amplitude and phase on the surface of the structure. Basically, σ is greatest in the higher frequency range and decreases for lower frequencies. The transition from $\sigma < 1$ to $\sigma = 1$ is usually denoted by a critical frequency f_c which, together with σ, can be derived from theory for some elementary sources.

For a pulsating sphere, for example,

$$f_{\mathrm{c}} = c \,/\, 2\pi a \qquad (56)$$

where

c = 340 m/s, the speed of sound in air;

a is the radius of the sphere.

For a piston vibrating in-phase in a baffle

$$f_{\mathrm{c}} = 0{,}7c \,/\, \sqrt{\pi S} \qquad (57)$$

where S is the surface area of the piston.

For an oscillating sphere

$$f_{\mathrm{c}} = 1{,}41c \,/\, 2\pi a \qquad (58)$$

where a is the diameter of the sphere.

Figure 29 shows the resulting radiation efficiency as a function of the frequency normalized to the relevant f_c for both elementary sources.

The pulsating sphere is a good approximation for all compact sources with in-phase surface vibrations. Compact machinery components oscillating as rigid bodies can be described by the oscillating sphere.

Many vibrating surfaces of machinery components that are excited by structure-borne sound are plate-like structures which normally do not vibrate as rigid bodies but have complicated distributions of the surface velocity (both amplitude and phase). Plate-like components of machinery will mainly be excited to bending waves above the fundamental eigenfrequency. The critical frequency is therefore determined by the frequency for which the bending wavelength λ_B equals the wavelength of the radiated sound in air.

So for supported plates f_c reads

$$f_c = \frac{c_{air}^2}{2\pi} \cdot \sqrt{\frac{m''}{B'}} \approx \frac{64\,000}{c_L \cdot d} = \frac{64\,000}{d} \cdot \sqrt{\frac{\rho}{E}} \qquad (59)$$

where

c_L is the longitudinal wave speed;

d is the thickness of the plate;

m'' is the mass per unit area;

B' is the bending stiffness of the plate.

Approximating σ for $f < f_c/2$

$$\sigma \approx \frac{Uc_{air}}{\pi^2 S \cdot f_c} \sqrt{\frac{f}{f_c}} \qquad (60)$$

for $f = f_c$:

$$\sigma \approx 0{,}45 \cdot \frac{\sqrt{Uf_c}}{c_{air}} \qquad (61)$$

for $f > 2f_c$:

$$\sigma \approx 1 \qquad (62)$$

where

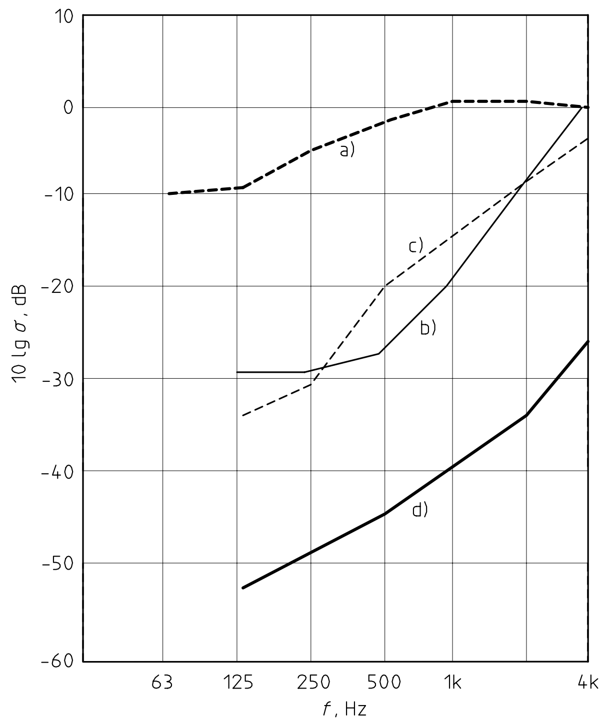
U is the perimeter;

S is the surface area.

Although the ratio U/S has a weak influence on the stiffening of plates, ribs increase the length of the boundaries and therefore the ratio U/S (where U is the length of the perimeter plus twice the total length of the ribs). Stiffening thus increases σ except when the force excites the panel via a rib. In this case the driving point impedance of the plate is increased by the stiffening.

Figure 30 shows the influence of the thickness of a plate on its radiation efficiency.

To summarize, a reduction of radiation is related to the reduction of the product of $S\bar{v}^2$ and of the radiation efficiency σ of the structure.



Key

a) Cylinder block of diesel engine
b) Steel pipe, diameter 0,7 m, wall thickness 1,3 mm
c) Steel plate, 0,5 m × 0,5 m, thickness 1,5 mm
d) Steel plate like c, perforated, 30 % of the area open

Figure 28 — Measured radiation efficiencies

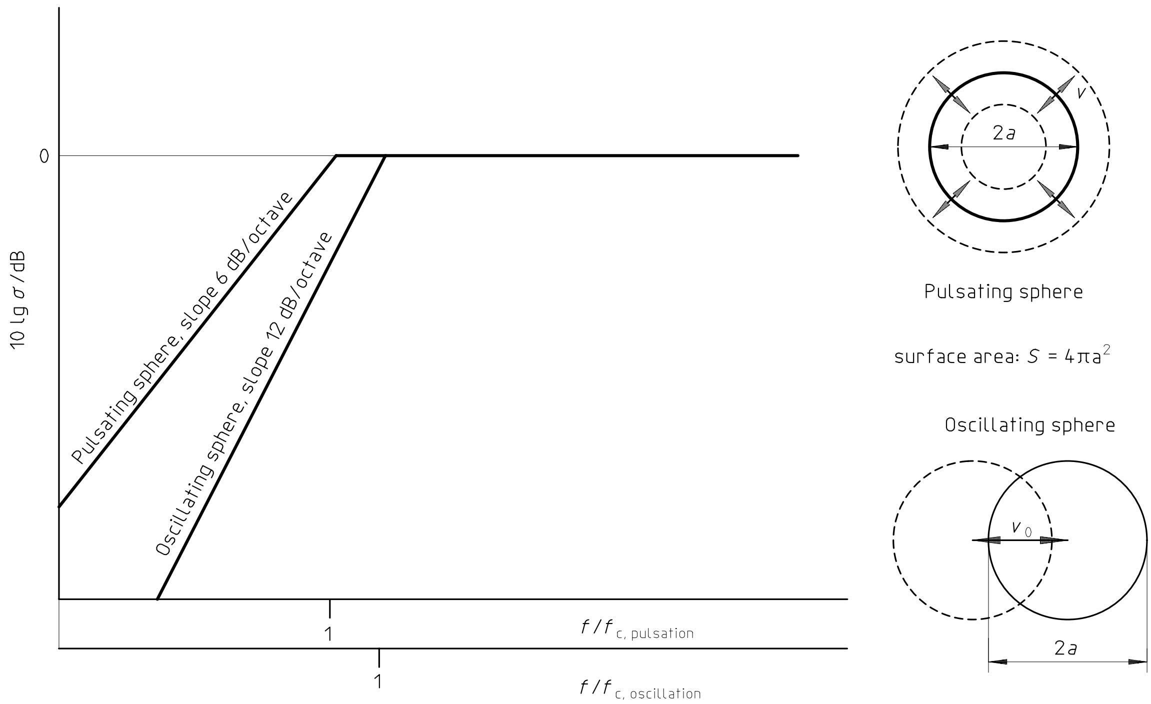


Figure 29 — Radiation efficiences of ideal model sources

Main measures are:

- compact design;
- application of perforated plates;
- use of flexible plates (heavy, if possible) to raise the critical frequency f_c, if high-frequency sound must be reduced.

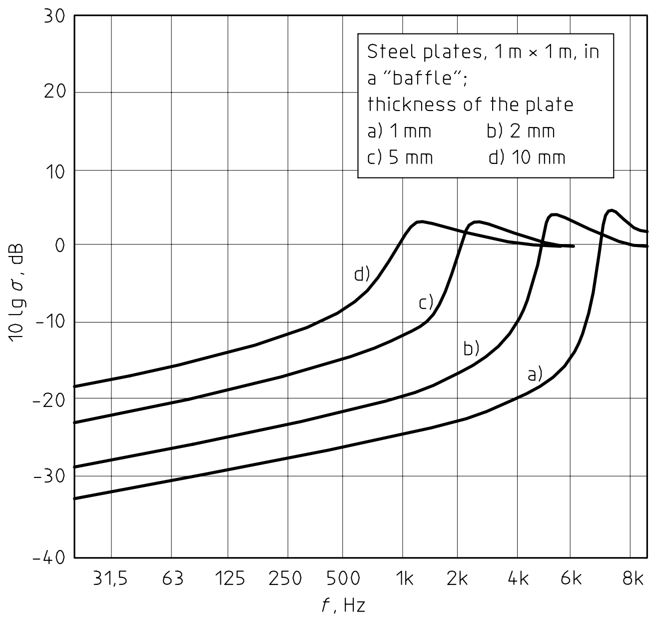


Figure 30 — Effect of plate thickness on radiation efficiency

7 Analysis by measurement methods

7.1 Purpose of the analysis

The analysis of existing machinery is important at various stages of the design:

Stage 1: Clarification of task: measurements on existing competitor's machines or on own machines of earlier design may be very useful to define a realistic design task.

Stage 2: Conceptional design: measurements on existing machines may be very useful to compare different principles.

Stage 3: Design and detail: at this stage, measurements can help in the choice of dimensions, materials and components.

Stage 4: Prototyping: the analysis of prototype(s) is a central element of every design process. The results can be used to improve details of the design (back to stage 3).

The analysis should be focused on the basic elements mentioned in clause 1, i.e:

- internal sources;
- transmission paths in the machine;
- radiation from the boundaries.

A cost-effective strategy for the reduction of noise can only be developed when a certain amount of information on all three aspects has been made available. The minimum information that should be obtained is the following:

- the most important internal sound sources;
- the most important transmission paths for these sources;
- the most important sound-radiating parts of the machine.

Answers to these questions may differ for different frequency ranges.

In the simple case of one dominating internal source, and a small number of transmission paths and radiating parts, qualitative information may be sufficient to decide which measures are to be taken. In many cases, however, it will be necessary to obtain quantitative information as well (i.e. the "strength" of the internal sources, the value of the "transfer functions" of the transmission paths, and the magnitude of the radiation of various machinery parts into the connected receiving systems (air, structures, etc.).

7.2 Internal sources

A first analysis of the internal sources can be done on the basis of the spectrum of the radiated sound (one-third-octave or narrow band), combined with the knowledge of the machine structure (potential internal sources, rotation speeds, numbers of blades, poles, gear-wheel teeth, etc.). The source of important peaks in the spectrum can often be identified in that way. Additional information can be obtained from the analysis of the time history of the radiated sound pressure. In both approaches variation of the operational parameters (especially speed) may be very helpful in the interpretation of the signals. The same is true concerning listening to the signal.

If the information obtained from the spectrum and from time history is not sufficient, the next steps may be

- successive switching off of internal sources (if possible);
- successive blocking of the transmission of sound from the internal sources to the boundaries of the machine (by isolation of airborne, liquid-borne or structure-borne sound transmission).

In situations where more detailed quantitative information is required there are many possible approaches. Examples can be seen in Table 8 and the references [8] and [9].

A final possibility for the analysis of sources is the combination of computed or measured data on transmission and radiation with the measured sound pressure, resulting in quantitative information on internal sources (see 7.3 and 7.4).

7.3 Transmission paths

The most straightforward method to obtain information on the relative importance of sound paths is to successively increase the isolation in these paths, for example by the introduction of more efficient shielding, isolators or silencers.

Quantitative information can be obtained experimentally, using theoretical models as an aid, or with a mixed approach. For examples of such techniques see Table 9.

For more information on these and other methods, see references [8], [10], [11] and [12].

7.4 Radiation

The analysis may concern the radiation into air, into a gas in a pipe, into a liquid in a pipe, a structure or a combination of these. For the analysis of the radiation into the air surrounding a machine the methods of table 10 may be applied.

For the analysis of the "radiation" into gas or liquid filled piping systems, or structures, or into combined systems, some of the methods of Table 10 may also be applicable.

For more information see references [8] through [14].

7.5 Summary of procedures for the analysis of existing machinery by measurement methods

Information is summarized in Tables 8 through 10.

Table 8 — Procedures for the analysis of internal sources

No.	Designation	Description	Remarks
1	Frequency spectrum analysis	Determination of the sound spectrum at an arbitrary point in the far field of the machine. Arbitrary surroundings are allowed.	If the mechanical structure and the working principles of the machine are known, frequency analysis may indicate which internal sources are important.
2	Time analysis of radiated sound pressure	Measure the sound pressure as a function of time at an arbitrary point in the far field of the machine. Arbitrary surroundings are allowed.	In addition to frequency analysis, time domain analysis may give further insight concerning the excitation mechanism.
3	Switching off of partial sound sources	It is sometimes possible to operate a machine with certain internal sources switched off. If the sound pressure as a function of frequency or time is measured in the far field the contribution of the partial source may be determined.	
4	Successive shielding or blocking of partial internal sources	It is often possible to decrease the transmission of sound from a particular partial internal source to the boundaries of the machine. Measure the effect on the radiated sound pressure (as a function of frequency and/or of time) in arbitrary surroundings.	A partial airborne sound source may be put into an improvised enclosure. A partial structure-borne source may be isolated from the machine structure by improvised resilient elements.
5	Variation of operational parameters	Variation of speed or load. Measurement of radiated sound pressure as a function of time and/or of frequency, in arbitrary surroundings.	
6	Signature analysis	Measure the acceleration at a point at the boundaries of the machine, synchronise with the signal of the rotating shaft or axle. Use special software for the analysis.	For rotating machinery this is a very suitable method to distinguish internal sources from the response of the machine structure.
7	Direct substitution method	Replace the partial source by an artificial source with known properties. Measure the radiated sound pressure.	Method to determine the (equivalent) source strength of an internal source (airborne or structure-borne).
8	Reciprocal substitution method	Reciprocal version of method 7, using a loudspeaker at the receiving position and a microphone or an accelerometer at the position of the internal source.	A convenient alternative when the internal source cannot be replaced by an artificial source. Assumes linearity.
9	Correlation/coherence procedure	Determine the correlation between the radiated sound pressure and a reference signal from the partial source.	The usefulness of this method may be limited because in practice it may be difficult to generate a suitable reference signal.
10	Sound intensity measurements	Determine partial sound powers of potential internal sources.	Suitable for the determination of direct contributions to airborne sound emission by various components.

Table 9 — Procedures for the analysis of sound transmission inside the machine

No.	Designation	Description	Remarks
1	Direct measurement of transfer functions	Apply an artificial source at the position of the internal source and measure accelerations at the machine's boundaries or sound pressures in the far field.	
2	Reciprocal measurement of transfer functions	Reciprocal version of method 1. The places of source and receiver are interchanged.	Useful when it is impossible to apply an artificial source inside the machine.
3	Successive blocking of transfer paths inside the machine	Apply isolation or insulation techniques in the various sound paths.	The method is suitable for the identification of important sound paths.
4	Measurement of power flow	Apply special techniques to determine the airborne, liquid-borne or structure-borne sound power flow along a path.	Complicated methods; specialists knowledge required.
5	Modal analysis (experimental)	Measure the deterministic vibrational behaviour of the structure.	Not suitable for multi-resonant response range.

Table 10 — Procedures for the analysis of airborne sound radiation

No.	Designation	Description	Remarks
1	Selective shielding of parts which radiate airborne sound.	Successively shield radiating parts of the machine and measure radiated sound pressure.	Suitable for the determination of direct contributions to the airborne sound from various parts of the machine.
2	Sound intensity measurements	Sample over the machine surface with an intensity probe.	(see above)
3	Near field sound pressure measurement	Measure the sound pressure close to the radiating surface.	Easier to perform but less accurate than method 2.
4	Directional sound measurements	Measure the sound pressure in the free field with a strongly directional microphone or microphone array.	A straightforward method, but seldom applied.
5	Vibration measurements on machinery parts which radiate airborne sound	Measure the vibration at the vibrating surface. $W = \rho c S \overline{v^2}\, \sigma \varpi$ where ρc is the characteristic airborne sound impedance; S is the radiating surface area; $\overline{v^2}$ is the mean value of squared velocity; σ is the radiation efficiency.	Suitable if a good estimate of the radiation efficiency σ is available.
6	Modal analysis	Analyse the deterministic behaviour of the outside boundaries.	Suitable at low frequencies if the radiation efficiency σ is known

8 Analysis by computational methods

8.1 Purpose of the analysis.

In clauses 4, 5 and 6 a considerable number of simple analytical models are outlined which describe the basic behaviour of a type of source, the radiation of a source or the transmission of sound through a certain type of sound path. These models are relevant at all stages of the design. Additionally, however, there are many other computational methods, which can be classified in the following way:

1) methods to compute excitation;

2) methods to compute the transmission and radiation of sound.

The advanced methods to compute excitation are very specific for a particular type of excitation (flow, electromagnetic, impact, etc.) and are not considered any further in this Technical Report.

The methods which describe the transmission and radiation of sound can be divided into two major classes:

1) deterministic (numerical) methods.

2) statistical methods.

Important examples of the deterministic methods are the Finite Element Method (FEM) and the Boundary Element Method (BEM). A well-known example of the statistical methods is the Statistical Energy Analysis (SEA). There are also statistical methods based on generalised measurement results (empirical methods). Furthermore there are simplified versions of SEA.

8.2 Deterministic methods

The deterministic methods are useful to compute the detailed response of a specific system to a specific excitation. Consequently such computations may be useful for low and medium frequencies at the detailed design stage (stage 3) and sometimes for the analysis of some low or medium frequency aspects at the prototyping stage (stage 4). Application to multi-resonant response ranges or during the design stages 1 and 2 is not recommendable.

Generally, the deterministic methods require considerable experience, powerful hardware and a significant modelling effort.

8.3 Statistical methods

Statistical methods are suitable to determine the average of transfer and response quantities over frequency bands for a specific system and to determine the average of such quantities over an ensemble of more or less similar systems. They are thus particularly valuable for high and medium frequencies. Most statistical approaches are easy to apply and are useful at all stages of the design process, most of all at stage 2 (conceptional design stage).

8.4 Applicability of computational methods

Table 11 gives hints on the applicability of some types of computational methods.

Table 11 — Survey of deterministic and statistical computational methods

Method	Characteristics	Applicability	Frequency region
Finite Element Method (FEM)	Capable of providing detailed information about the response, the input mobility and transfer functions. A well-established method for experts.	Structures, enclosures filled with gases or liquids	Low to medium modal density
Boundary Element Method (BEM)	Suitable for the computation of the radiation from a vibrating surface. Expert knowledge required.	Radiating surfaces	Low to medium modal density
Matrix Method for exhaust and intake systems	Computes the characteristics of sound transmission through a gas filled piping system. Suitable for the design of silencers.	Exhausts and intakes	Low to medium modal density
Statistical Energy Analysis (SEA)	Provides approximate information about the transmission of sound power. Modelling requires expertise. Interaction with experiments required. Not yet fully established in engineering world.	Structures, gas, liquid and combinations	High and medium modal overlap
Simplified SEA	As SEA but for a more limited range of applications. Easy to apply.	As above	As above
Empirical models	Models based on measured data in similar systems, sometimes combined with analytical models (for example for the radiation). Suitable to determine the transmission of sound energy. Simple to apply.	As above	As above

Annex A

Example of the estimation of airborne sound emission of a machine caused by structure-borne and airborne sound emission from a component

A.1 Given data

Basic design of a given machine as in Figure 1; total dimensions (e.g. 0,5 m × 0,5 m × 0,5 m); component (e.g. electric motor).

A.2 Steps of calculation

Calculation using the model in Figure 2 with input data and equations according to clauses 5 and 6 as well as common basics of technical acoustics.

Path 1 — Airborne sound, W_1, L_W

$$L_{W1} = L_W - \Delta L$$

Necessary data:

a) Sound power level L_W of the component measured in frequency bands according to an International Standard.

b) Insertion loss ΔL of the housing around the machine, in frequency bands.

Origin of data (items refer to above list):

a) L_W from the manufacturer of component or by measurement through the user.

b) Estimation of ΔL

— if (different from Figure 2) component outside of the housing ΔL = 0 dB
— if (in accordance to Figure 2) inside component
  — housing without openings, in most cases path 1 negligible in relation to path 2
  — housing with openings

$$\Delta L = 10 \lg \frac{A}{S_0}$$

where

S_0 is the area of openings (in square metres, m²)

A is the equivalent sound absorbing area inside the housing (in square metres, m²), see technical literature

ΔL is the insertion loss (in decibels, dB)

Path 2 — Structure-borne sound, W_2, L_{W_2}

$$L_{W2} = L_F + L_{WF}$$

Valid for force excitation; for velocity excitation an analogous equation can be derived according to reference [5].

Necessary data:

a) Emission value of structure-borne sound L_F of the component, no standardised measuring method available.

b) Transmission quantity L_{WF} of machine structure.

Origin of data (items refer to above list):

a) Manufacturer of component (agreement on measuring conditions necessary) or measurement by user.

b) Measurement by manufacturer on a prototype or similar machine; approximative calculation for simple structures, e.g. plates.

Remark concerning measuring conditions (items refer to above list):

a) component mounted on test structure similar to the machine to be designed; measurement of acceleration (a) near mounting point and measurement of point admittance Y_F at these points in frequency bands.

$$F = \frac{a}{2\pi f Y_F}$$

b) excitation of machine by shaker in frequency bands with a force F and measurement of radiated sound power L_W

$$L_{W\mathrm{F}} = L_W - L_F$$

A.3 Resulting sound emission

$$W_{\mathrm{tot}} = W_1 + W_2$$

$$L_{W\,\mathrm{tot}} = 10\ \lg\left(10^{L_{W1}/10} + 10^{L_{W2}/10}\right)$$

($L_{W\ \mathrm{tot}}$, L_{W_1}, L_{W_2} in decibels, dB)

A.4 Example with given values

Path 1

L_W = 80 dB

S_0 = 0,01m²

A = 0,1 m²

ΔL = 10 dB

L_{W_1} = 70 dB

Path 2

Case a)	Case b)
F = 0,5 N	F = 2 N
L_F = –6 dB	L_F = 6 dB
L_{W_F} = 68 dB	L_{W_F} = 68 dB
L_{W_2} = 62 dB	L_{W_2} = 74 dB
Airborne sound is dominating	Structure-borne sound is dominating
The total airborne sound emission is	The total airborne sound emission is
$L_{W'\mathrm{tot}}$ = 71 dB	$L_{W'\mathrm{tot}}$ = 76 dB

Actions for noise control at:

Path 1	Path 2
Absorbing material inside the housing	Decoupling of motor
A =0,1 m^2 increases to 0,8 m^2	equation (31) with f_0 = ½ f
$\Delta L''$ = 19 dB	D = 10 dB
L_{W_1}' = 61 dB	$L_{W_2}' = L_{W_2} - D$
	L_{W_2} = 64 dB
$L_{W'\mathrm{tot}}$ = 65 dB	$L_{W\mathrm{tot}}$ = 71 dB

Annex B

Glossary

A-weighting

Electric filter implemented in sound level meters with the standardized frequency response taking into account that the sensitivity of hearing is frequency dependent. Used for measurement of A-weighted sound pressure levels.

Characteristic airborne sound impedance

Ratio of sound pressure to particle velocity at a point on a propagating plane sound wave.

Critical frequency f_c

At the critical frequency the bending wavelength of a plate equals the wavelength in air.

Eigenfrequency

One of the possible frequencies of free vibration of a mechanical system resulting from only elastic and inertial forces applied to the system.

First eigenfrequency

The eigenfrequency with the lowest possible frequency.

Mass/stiffness controlled system

An in-phase moving mechanical system without/with deformation when force is applied.

Plate admittance h_μ

Frequency independent point admittance of the infinite homogeneous plate. Applicable to finite real plates if exited in frequency bands and plates being larger than bending wave length.

Point admittance h

At a point in a mechanical system the ratio of the resulting vibration velocity to the applied force at a specified frequency.

Radiation efficiency σ

Ratio of sound power really radiated by a structure of a specified area vibrating with a given root-mean-square velocity over the area to that sound power, which would be emitted as a plane wave by a plane plate of the same area vibrating on phase with the same velocity.

Spring eigenfrequency f_{01}

For springs, the lowest eigenfrequency of the spring-element itself (characteristic wavelength equals $\lambda_B/2$).

Transfer function

The computed or measured relation between the output (response) and the input (excitation) of a system expressed as a function of frequency.

Transmission admittance h_T

The ratio of the root-mean-square velocity at the sound radiating area of a structure to the applied vibration force at a specified point and frequency.

Velocity transmission T_I

The ratio of the root-mean-square velocity at the sound radiating area of a structure to the applied vibration velocity at a specified point and frequency.

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