TECHNICAL REPORT

ISO/TR 11688-2

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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

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Contents

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International 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 Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.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.

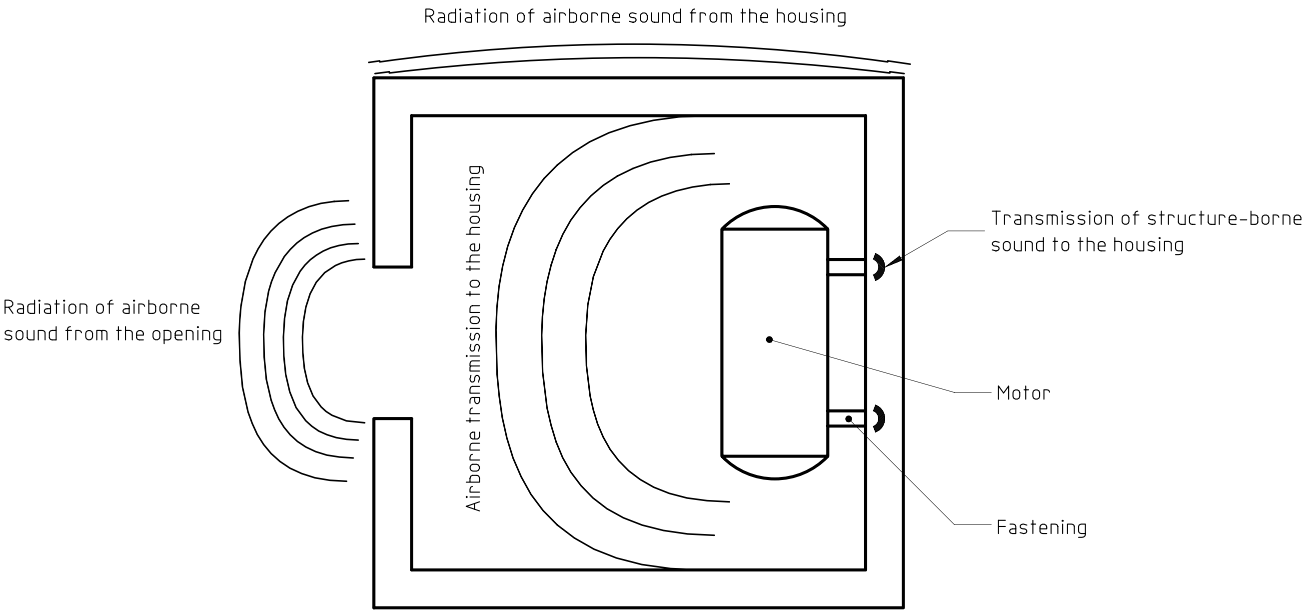


Figure 1 — Simplified machine for the illustration of acoustical modelling

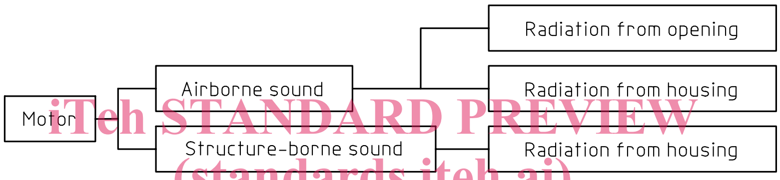


Figure 2 — Block diagram for the illustration of generation, transmission and radiation of sound in the "machine" of Figure 1

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:

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

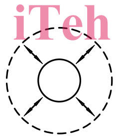
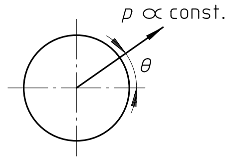
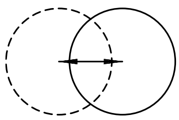
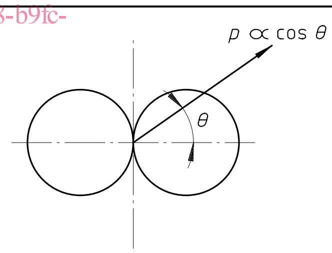
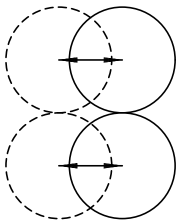
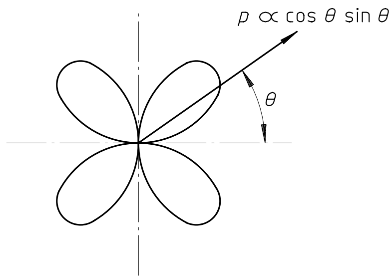
where

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]):

$$= \rho \quad ^{2} \quad ^{3}\left(\frac{}{}\right)^{k} = \rho \quad ^{2} \quad ^{3}(\quad)^{k} \tag{2}$$

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])

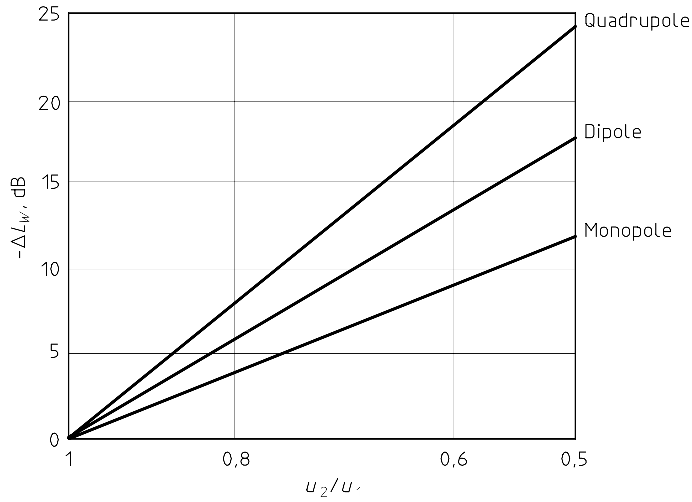
	Dimension n of flow field		
	$n = 1$	$n = 2$	$n = 3$
Mass flow fluctuation (monopole)	$W \sim \rho a u^2$	$W \sim \rho u^3$	$W \sim \frac{\rho}{\alpha} u^4$
Force fluctuation (dipole)	$W \sim \frac{\rho}{\alpha} u^4$	$W \sim \frac{\rho}{\alpha^2} u^5$	$W \sim \frac{\rho}{\alpha^3} u^6$
Turbulence (quadrupole)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

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:

$$\frac{\quad}{\text{mech}} \qquad (4)$$

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

An empirical estimation for the sound power level is

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

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:

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

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

$$St = \frac{fD}{u} \qquad (7)$$

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

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

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 $p = \rho gz$.

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.

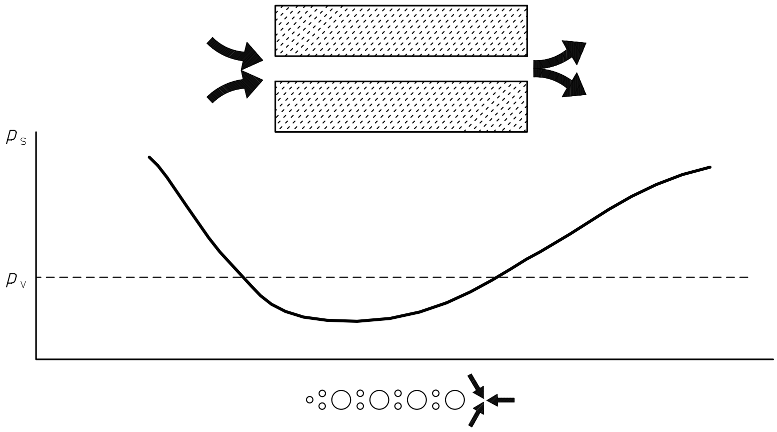


Figure 4 — Generation and implosion of cavitation bubbles

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,

$$\zeta_W = \frac{F_W}{\frac{\rho}{2} u^2 A} \tag{9}$$

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

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

with $A_0 = 1\ \mathrm{m}^2$ 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;

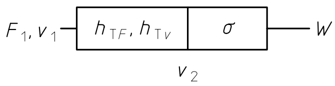
— frequency shift by replacing large openings by smaller ones.

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.

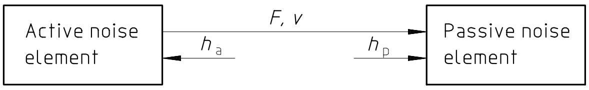
Furthermore the following assumptions are made:

— 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

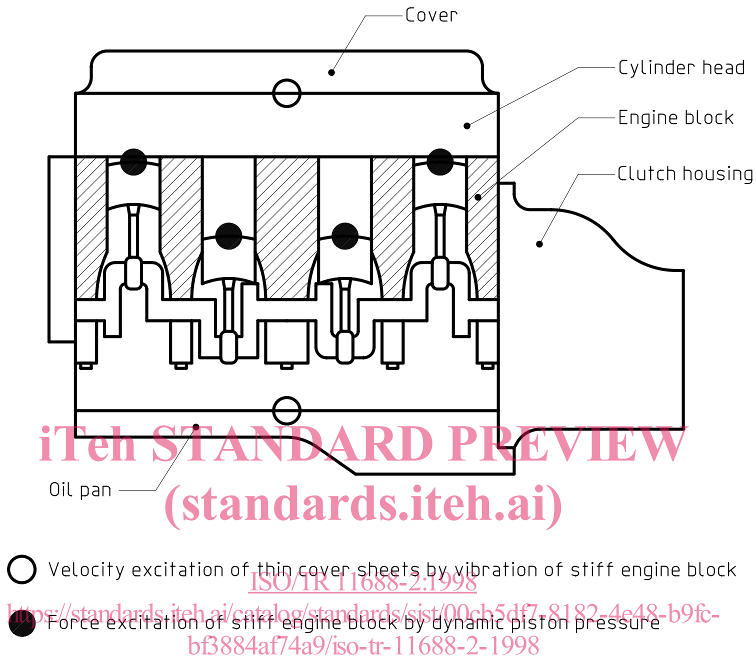


Figure 7 — Schematic cross-section of diesel engine

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