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Acoustics — Guidelines for noise control by enclosures and cabins

Acoustique — Lignes directrices pour la réduction du bruit au moyen d'encoffrements et de cabines

Reference number
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Foreword

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

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International Standard ISO 15667 was prepared by Technical Committee ISO/TC 43, *Acoustics*, Subcommittee SC 1, *Noise*.

Annexes A and B of this International Standard are for information only.

Introduction

Acoustic enclosures and cabins provide a reduction of airborne sound on the propagation path from the machine (or a set of machines) to nearby work stations or to the environment. This International Standard describes criteria which determine the acoustic performance of enclosures and cabins with consideration of operational aspects.

Acoustics — Guidelines for noise control by enclosures and cabins

1 Scope

This International Standard deals with the performance of enclosures and cabins designed for noise control. It outlines the acoustical and operational requirements which are to be agreed upon between the supplier or manufacturer and the user of such enclosures and cabins. This International Standard is applicable to two types of acoustic enclosures and cabins, as follows.

a) Cabins for noise protection of operators: free-standing cabins and cabins attached to machines (e.g. vehicles, cranes).

b) Free-standing enclosures covering or housing machines: enclosures with a fraction of acoustically untreated open area of less than 10 % of the total surface are the main subject of this International Standard.

In this International Standard, emphasis is put on lightweight constructions. However, thick, massive structures as, for example, brick walls, are not excluded.

Enclosures and cabins with more than 10 % open and untreated area belong to the category of partial enclosures. They are not the subject of this International Standard.

A third type of enclosure, integrated enclosures which form a part of the machine and are firmly attached to it, is not the subject of this International Standard.

2 Normative references

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

ISO 140-3, *Acoustics — Measurement of sound insulation in buildings and of building elements — Part 3: Laboratory measurements of airborne sound insulation of building elements*.

ISO 717-1, *Acoustics — Rating of sound insulation in buildings and of building elements — Part 1: Airborne sound insulation.*

ISO 3740 series, *Acoustics — Determination of sound power levels of noise sources using sound pressure*.

ISO 9614 (all parts), *Acoustics — Determination of sound power levels of noise sources using sound intensity*.

ISO 11200 series, *Acoustics — Noise emitted by machinery and equipment.*

ISO 11546-1:1995, *Acoustics — Determination of sound insulation performance of enclosures — Part 1: Measurements under laboratory conditions (for declaration purposes)*.

ISO 11546-2:1995, *Acoustics — Determination of sound insulation performance of enclosures — Part 2: Measurements* in situ *(for acceptance and verification purposes)*.

ISO 11957:1996, *Acoustics — Determination of sound insulation performance of sound protecting cabins — Laboratory and* in situ *measurements*.

ISO 14163, *Acoustics — Guidelines for noise control by silencers*.

3 Terms and definitions

For the purposes of this International Standard, the following terms and definitions apply.

3.1
enclosure
structure covering or housing a sound source (machine) for protection of the environment from this sound source (machine)

NOTE The shape may be box-like or follow the contour of machine parts. Box-shaped enclosures consist of walls and a roof. The enclosure may have openings for doors, windows, ventilation, material flow, etc.; see Figure 4.

3.2
cabin
construction specially designed for the protection of human beings (e.g. machine operators) from environmental noise, consisting of a fully enveloping structure

NOTE 1 Adapted from ISO 11957:1996.

NOTE 2 A floor is not always a component of the cabin.

3.3
sound power insulation of the enclosure
insertion loss of the enclosure
D_W
difference between the levels of the sound powers emitted from the sound source (machine) with and without the enclosure, in one-third-octave bands or octave bands, measured according to ISO 11546-1 or ISO 11546-2

NOTE 1 The sound power insulation (or insertion loss) is expressed in decibels, dB.

NOTE 2 This spectrum of values is useful for general planning of environmental noise control for locations at some distance from the source, e.g. in the reverberant field of an industrial hall or in the neighbourhood of a plant.

3.4
weighted sound power insulation of the enclosure
$D_{W,\mathrm{w}}$
single-number value determined in accordance with the method stated in ISO 717-1, except that the sound reduction index (or transmission loss) is replaced by the insertion loss, D_W

NOTE 1 The weighted sound power insulation is expressed in decibels, dB.

NOTE 2 The single-number value is useful for a rough comparison of different enclosures and for general acoustical planning inside buildings without detailed knowledge of the source spectrum.

NOTE 3 Adapted from ISO 11546-2:1995.

3.5
panel transmission loss
R
sound reduction index (or transmission loss) of individual panels from which the enclosure is made, in accordance with ISO 140-3

NOTE 1 The panel transmission loss is expressed in decibels, dB.

NOTE 2 In a limited range of medium frequencies (typically 250 Hz to 1 000 Hz), the insertion loss, D_W, of a completely sealed enclosure is approximately related to the panel transmission loss, R, by

$$D_W \approx R + 10 \lg(\alpha) \text{ dB} \quad (1)$$

where α denotes the average absorption coefficient of the internal side of the panels. While spectral information on R and α is often provided, the relation (1) primarily gives an upper limit and is not a reliable foundation for predicting the actual insertion loss, D_W. Leakages, insufficiently acoustically treated openings, and flanking transmission of structure-borne sound result in smaller values of the actual insertion loss.

NOTE 3 For measurements of the airborne sound insulation of small building elements with openings, see ISO 140-10 [11].

3.6 Sound pressure insulation, D_p

3.6.1
sound pressure insulation for enclosures
D_p
difference between the levels of the sound pressures at a specified position with and without an enclosure, in one-third-octave bands or octave bands

NOTE 1 The sound pressure insulation is expressed in decibels, dB.

NOTE 2 This spectrum of values is useful for the detailed analysis of the acoustic performance of an enclosure in different directions.

NOTE 3 For measurements of the sound pressure insulation of an enclosure, see ISO 11546-1 and ISO 11546-2.

3.6.2
sound pressure insulation for cabins
D_p
difference between the levels of the sound pressures in an external diffuse sound field and in a cabin located in this field, in one-third-octave bands or octave bands

NOTE 1 The sound pressure insulation is expressed in decibels, dB.

NOTE 2 For measurements of the sound pressure insulation of a cabin see ISO 11957.

NOTE 3 Adapted from ISO 11957:1996.

3.7
apparent sound pressure insulation of a cabin
D'_p
difference between the levels of the sound pressures in a room with arbitrary sound field distribution and in a cabin located in the room, in one-third-octave bands or octave bands

NOTE 1 The apparent sound pressure insulation of a cabin is expressed in decibels, dB.

NOTE 2 The sound field in the room may not necessarily be diffuse.

NOTE 3 For measurements of the apparent sound pressure insulation of an enclosure, see ISO 11957.

NOTE 4 Adapted from ISO 11957:1996.

3.8
A-weighted sound pressure insulation
$D_{p\mathrm{A}}$
single-number value determined for the actual sound source spectrum, describing the reduction in the A-weighted sound pressure level at a specified position due to the enclosure or in a cabin located in a diffuse sound field

NOTE 1 The A-weighted sound pressure insulation is expressed in decibels, dB.

NOTE 2 This single-number value is most relevant for describing the actual acoustic performance of an enclosure for a particular machine, e.g. at a distance of 1 m from a machine enclosure or at any position inside a cabin.

3.9
estimated noise insulation due to the enclosure
$D_{p\mathrm{A,e}}$
single-number value determined for a specific sound source spectrum, describing the reduction in the A-weighted sound pressure level at a specified position due to the enclosure

NOTE 1 The estimated noise insulation due to the enclosure is expressed in decibels, dB.

NOTE 2 This single-number value is most relevant for estimating the acoustic performance of an enclosure without detailed knowledge about the source spectrum.

3.10
leak ratio
θ
ratio between the area of all acoustically untreated openings of the enclosure and the total interior surface area of the enclosure (including openings)

NOTE Adapted from ISO 11546-1:1995 and ISO 11546-2:1995.

4 General principles and operational considerations

4.1 Sound source

The sound source (or sources) to be acoustically treated by an enclosure shall be clearly identified. The radiated airborne sound shall be measured according to the relevant International Standards of the ISO 3740, ISO 9614 or ISO 11200 series.

The provision of an enclosure will result in a build-up of internal heat. Air-moving devices and auxiliary equipment supplied with the enclosure for removing the heat and for air-conditioning shall be considered as additional sound sources.

4.2 Sound propagation paths

Several paths of sound propagation from a sound source in an enclosure to the environment can be grouped into four categories as shown in Figure 1.

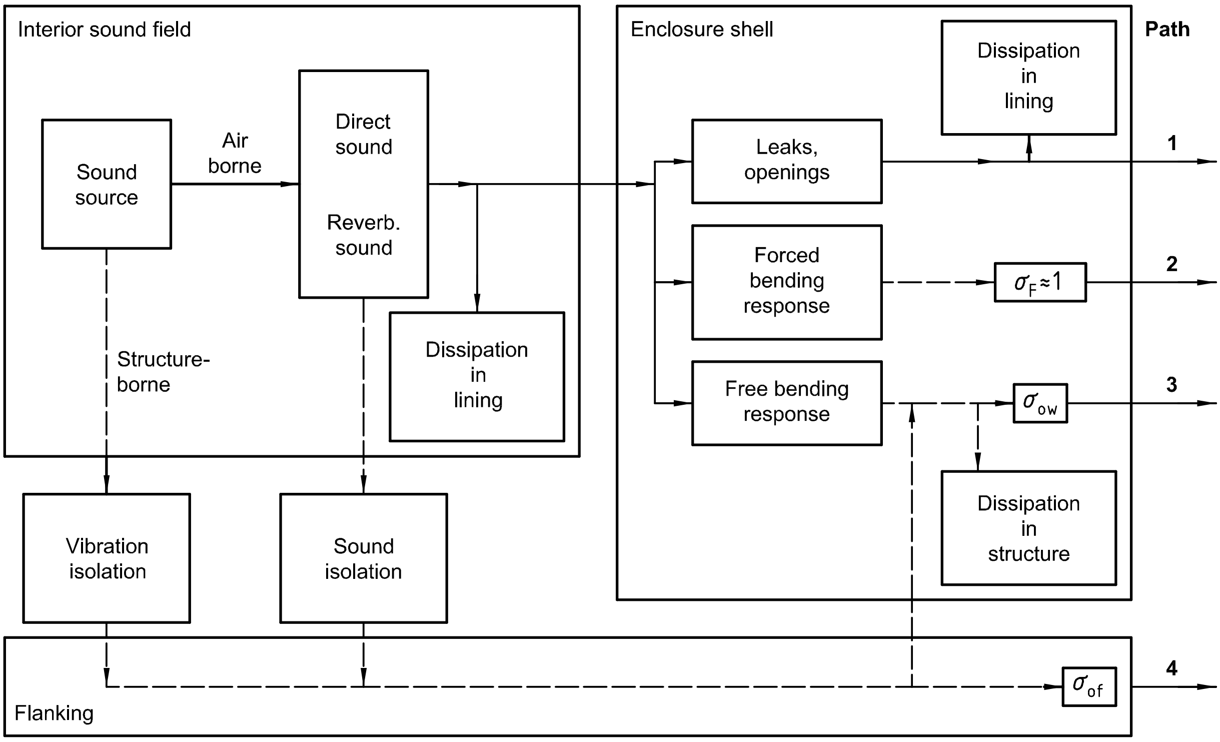


Figure 1 — Block diagram of sound propagation paths

a) **Path 1** for airborne sound through openings (or leaks) of the enclosure requires most attention. At very low frequencies, where the dimensions of the enclosure are small when compared to the wavelength and where there is little or no absorption by the enclosure lining, the volume of the enclosure and the constriction of the openings form a Helmholtz resonator, which may result in a negative insertion loss of the enclosure. At high frequencies, where the enclosure provides for substantial dissipation, the leak ratio θ and the dissipation of sound in linings close to the openings determine the transmission of sound along path 1. For acoustically untreated openings, the high frequency sound reduction index (or transmission loss) R_1 along path 1 is estimated from:

$$R_1 \approx -10\lg(\theta)\ \text{dB} \qquad (2)$$

b) **Path 2** for sound propagation through the enclosure walls is typically controlled by laboratory tests on well-sealed enclosures without flanking transmission of structure-borne sound. At very low frequencies, the ratio of the compliance of the air inside the enclosure and the volume compliance of the enclosure walls determines the insertion loss of the enclosure [see equation (3)]. At low frequencies, the compliance of the air between the machine and a nearby enclosure wall may resonate with the mass of the wall, which results in a minimum of the insertion loss.

At intermediate and higher frequencies, the panel transmission loss is effective. It is determined by the impedance of the impervious shell and the attenuation on the propagation path through the inside lining. Single-wall constructions exhibit a sound reduction index (or transmission loss) which is mass controlled up to a panel weight of about 15 kg/m^2 and frequencies of about 2 kHz. Double-wall constructions are used to improve the sound reduction index (or transmission loss) at intermediate frequencies above the double-wall resonance frequency. Minima of the sound reduction index (or transmission loss) due to coincidence of the incident sound with free bending waves on the panel are mostly avoided by sound damping by the lining at frequencies above 2 kHz. At all but very low frequencies where the perimeter of the enclosure is smaller than the wavelength of airborne sound, the radiation efficiency of forced bending response $\sigma_F \cong 1$.

NOTE The radiation efficiency is defined in ISO/TR 7849 [14].

c) **Path 3** contains the radiation of free bending waves from the enclosure walls. Since mostly thin panels are used for the enclosure, the radiation efficiency σ_{ow} of limp panels is small and predominantly determined by their clamped edges or attachment points. Free bending waves are largely caused by flanking transmission of structure-borne and airborne sound. Damping of the panels provides for dissipation of such waves. Free bending waves on the enclosure frame may need to be considered at frequencies above 1 kHz.

d) **Path 4** is for the radiation with efficiency σ_{of} of structure-borne and airborne sound from flanking components which is unaffected by the enclosure. The floor, unenclosed parts of the machine, material supplied to the machine, and pipework connected to the machine are examples of flanking components. The contribution from this path finally limits the acoustic performance of an otherwise well-designed enclosure.

In critical cases, the sound transmission via all the different paths needs to be considered. The individual contributions may be determined from appropriate measurements or calculations. The distinction between contributions from path 2 and path 3 is the most difficult. In addition, if possible, the background sound pressure level $L_{p\text{b}}$ should be determined for the case where the sound source to be enclosed is turned off.

4.3 Efficient noise control

NOTE For concerns to be addressed for efficient noise control by enclosures and cabins, see also references [1], [2], [6], [9].

4.3.1 Select an enclosure or cabin which is matched to the particular task of housing a machine or protecting a work station under general operating criteria, including availability of space, safety aspects, material flow, etc.

4.3.2 Generally, the acoustic performance of panels mounted on a mechanically stable frame is sufficient in terms of absorption and sound reduction index (or transmission loss) if common materials are used. Typical components shown in Figure 2 are

— outer shell: 1,5 mm steel sheet metal; where material other than steel is used for the outer shell, the thickness should be selected so as to result in a minimum mass per unit area of 10 kg/m^2 to 15 kg/m^2;

— absorbent lining on the inside: 50 mm mineral wool;

— perforated plate covering the absorbent lining: ≥ 30 % open;

— safety glass pane for windows: 6 mm thick.

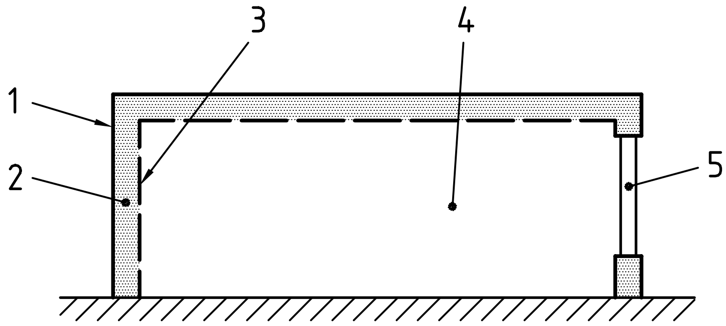
NOTE For the sake of brevity, the term "mineral wool" has been chosen throughout this International Standard to denote "mineral wool or fibre glass".

A typical spectrum of the sound pressure level close to a machine with and without such an enclosure is shown in Figure 3. The maximum A-weighted sound emission around 500 Hz determines the A-weighted sound pressure insulation.

Special requirements for enhanced low-frequency insertion loss, protective covers on the mineral wool, use of particular shapes and materials for the impervious surface, and the absorbing material, etc. need detailed investigations.

4.3.3 Devote full attention to leaks and openings. Avoid leaks between panels by making use of special single or double sealing constructions, depending on the acoustic requirements. If the panels are frequently removed, make sure that the sealing constructions can be used repeatedly. Where leaks are unavoidable, as in sliding doors, use absorbing linings or slot silencers. Minimize all openings for ventilation, cables, pipes, transport of material, etc. and equip them with silencers or sound-absorbent lined tunnels. Openings for maintenance purposes shall be closed carefully during operation.

4.3.4 To avoid flanking transmission of structure-borne sound, the sound source should be mounted on resilient elements. Panels making up the enclosure should not make contact with the sound source. Where this is unavoidable, the number of mounting points should be kept to a minimum and these should be provided with resilient elements between the sound source and contact points.



Key

1 Outer shell
2 Absorbent lining
3 Perforated cover
4 Space for sound source of work station
5 Window

Figure 2 — Acoustic enclosure or cabin (schematic)

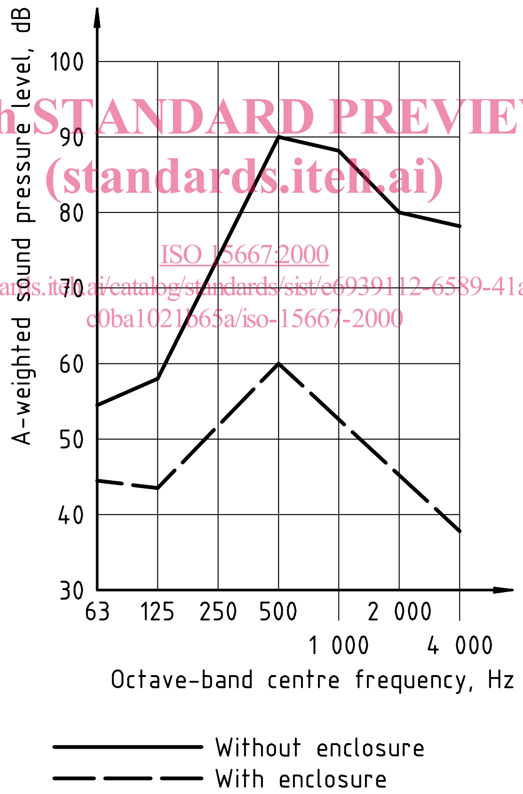


Figure 3 — Typical example of A-weighted octave-band spectrum of sound pressure level close to a machine

4.3.5 To avoid flanking transmission of airborne sound via the floor, use enclosures fully enveloping the machine, if necessary for a particularly high acoustical performance.

4.3.6 Coat the panels with damping material to increase the weight-dependent sound reduction index (or transmission loss) and the attenuation of free bending waves, if necessary for special applications.

5 Types of enclosures and cabins and particular requirements

5.1 Enclosures

5.1.1 Small enclosures (hoods)

Enclosures may be considered as being small for low-frequency sound when the largest dimension is less than one-quarter of the wavelength of airborne sound. Low-mass walls and transparent walls allow for easy handling, appropriate use and long life. The supporting structure is often the frame of the machine.

At low frequencies, the insertion loss of an airtight enclosure is

$$D_W = 20\lg\left(1+\frac{C_v}{\sum_{i=1}^{n} C_{\mathrm{w}i}}\right)\ \mathrm{dB} \qquad (3)$$

where

$C_v = V_0/(\kappa P_0)$ is the compliance of the gas volume inside the enclosure, in metres to the fifth power per newton, m^5/N;

V_0 is the volume of the gas inside the enclosure, in cubic metres, m^3;

κ is the ratio of the specific heats of the gas inside the enclosure; for air $\kappa = 1{,}4$;

P_0 is the static pressure of the gas inside the enclosure, in pascals, Pa; for air under ambient conditions $P_0 = 10^5$ Pa;

$C_{\mathrm{w}i} = \Delta V_{\mathrm{p}i}/p$ is the volume compliance of the ith enclosure panel in response to the sound pressure inside the enclosure, in metres to the fifth power per newton, m^5/N;

$\Delta V_{\mathrm{p}i}$ is the volume displacement of the ith enclosure panel in response to the sound pressure inside the enclosure, in cubic metres, m^3;

p is the uniform sound pressure inside the enclosure, in pascals, Pa;

n is the number of panels forming the enclosure.

For the special case of a cubical enclosure with clamped flat panels, the insertion loss is

$$D_W = 20\lg\left[1+41\left(\frac{h}{a}\right)^3 \frac{E}{\kappa P_0}\right]\ \mathrm{dB} \qquad (4)$$

where

h is the panel thickness of the enclosure, in metres, m;

a is the edge length of the enclosure, in metres, m;

E is Young's modulus for the panel material in pascals, Pa;

κ, P_0 are as in equation (3).

For simply supported rather than clamped panel edges, the insertion loss is typically 10 dB lower. For small enclosures of the same mass, equation (4) indicates that aluminium and glass are superior to steel by more than 10 dB in insertion loss, while lead is a very poor choice for low-frequency sound [1].

Except for special constructions, all small enclosures are likely to have leaks and do not provide a positive insertion loss at frequencies below 1,4 f_L, where for a cubicle enclosure with a round opening

$$f_L = \frac{c}{2\pi}\sqrt{\frac{\theta}{(h+\Delta h)a\left(1+\frac{\sum_{i=1}^{n} C_{wi}}{C_v}\right)}} \qquad (5)$$

where

c	is the speed of sound in the air volume inside the enclosure, in metres per second, m/s;
θ	is the leak ratio;
a, h	are as in equation (4);
$\Delta h \approx 1{,}6 a_L$	is the end correction for both ends of the opening in the enclosure, in metres, m;
a_L	is the radius of the opening in the enclosure, in metres, m;
C_{wi}, C_v	are as in equation (3).

At frequencies above f_L, the insertion loss of the leaky enclosure approaches that of the sealed enclosure. Leaks between the enclosure and the frame should be sealed by resilient strips suitable for frequent use. Since efficient silencers cannot be installed due to lack of space, openings should be kept as small as possible. Flanking transmission of structure-borne sound (e.g. via paper from a mechanical printer) shall be controlled, preferably by vibration damping.

5.1.2 Enclosures for single stationary machines

5.1.2.1 In workshops

The size of an enclosure is often determined by the available space around the machine. In some cases it may be considered that a partial enclosure, surrounding the dominant sound source, is more suitable.

The size and construction of the enclosure needs to be chosen by taking into consideration many aspects including the need for access, maintenance, adjustments, or removal/replacement of tools, etc. In some cases the size and mass of individual panels may require stiffening and the provision of hooks to accommodate lifting and removal.

Additionally, enclosures may need to be treated externally to resist the effects of the environment, e.g. the effects of oil and water. They should also be capable of being cleaned. The internal surfaces of the enclosure and all openings should be provided with absorbent linings. These linings can be protected from the ingress of oil and water by the provision of plastic films or metal foils. Where these coverings are used, it must be appreciated that in some cases the coverings may affect the acoustical performance of the absorbent linings, especially at high frequencies.

NOTE 1 For an area-related mass of film or foil of more than 50 g/m^2, or a thickness of plastic film of more than 50 μm, reduced absorption occurs at frequencies above 2 kHz.

For protection of the absorptive lining from mechanical damage, sound-transparent covers are necessary.

NOTE 2 Sufficient sound transparency is generally obtained by use of aluminium mesh or perforated steel plate, 30 % open area, and with holes of 3 mm to 5 mm diameter.