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Acoustics — Guidelines for noise control by silencers

Acoustique — Lignes directrices pour la réduction du bruit au moyen de silencieux

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

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

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.

International Standard ISO 14163 was prepared by Technical Committee ISO/TC 43, *Acoustics*, Subcommittee SC 1, *Noise*.

Annexes A to C of this International Standard are for information only.

Introduction

Whenever airborne sound cannot be controlled at the source, silencers provide a powerful means of sound reduction in the propagation path. Silencers have numerous applications and different designs based on various combinations of absorption and reflection of sound, as well as on reaction on the sound source. This International Standard offers a systematic description of principles, performance data and applications of silencers.

Acoustics — Guidelines for noise control by silencers

1 Scope

This International Standard deals with the practical selection of silencers for noise control in gaseous media. It specifies the acoustical and operational requirements which are to be agreed upon between the supplier or manufacturer and the user of a silencer. The basic principles of operation are described in this International Standard, but it is not a silencer design guide.

The silencers described are suitable, among others,

— for attenuating system noise and preventing crosstalk in heating, ventilation and air-conditioning (HVAC) equipment;

— for preventing or reducing sound transmission through ventilation openings from rooms with high inside sound levels;

— for attenuating blow-off noise generated by high-pressure lines;

— for attenuating intake and exhaust noise generated by internal combustion engines; and

— for attenuating intake and outlet noise from fans, compressors and turbines.

They are classified according to their types, performance characteristics and applications. Active and adaptive passive noise-control systems are not covered in detail in 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 3741, *Acoustics — Determination of sound power levels of noise sources using sound pressures — Precision methods for reverberation rooms.*

ISO 3744, *Acoustics — Determination of sound power levels of noise sources — Engineering methods for free-field conditions over a reflecting plane.*

ISO 7235, *Acoustics — Measurement procedures for ducted silencers — Insertion loss, flow noise and total pressure loss.*

ISO 11691, *Acoustics — Measurement of insertion loss of ducted silencers without flow — Laboratory survey method.*

ISO 11820, *Acoustics — Testing of silencers in situ.*

3 Terms and definitions

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

3.1
silencer
device reducing sound transmission through a duct, a pipe or an opening without preventing the transport of the medium

3.2
dissipative silencer
absorptive silencer
silencer providing for broad-band sound attenuation with relatively little pressure loss by partially converting sound energy to heat through friction in porous or fibrous duct linings

3.3
reactive silencer
general term for reflective and resonator silencers where the majority of the attenuation does not involve sound energy dissipation

3.4
reflective silencer
silencer providing for single or multiple reflections of sound by changes in the cross-section of the duct, duct linings with resonators, or branchings to duct sections with different lengths

3.5
resonator silencer
silencer providing for sound attenuation at weakly damped resonances of elements

NOTE   The elements may or may not contain absorbent material.

3.6
blow-off silencer
silencer used in steam blow-off and pressure release lines throttling the gas flow by a considerable pressure loss in porous material and providing sound attenuation by lowering the flow velocity at the exit and reacting on the source of the sound (such as a valve)

3.7
active silencer
silencer providing for the reduction of sound through interference effects by means of sound generated by controlled auxiliary sound sources

NOTE   Mostly low-order modes of sound in ducts are affected.

3.8
adaptive passive silencer
silencer with passive sound-attenuating elements dynamically tuned to the sound field

3.9
insertion loss,
D_i
difference between the levels of the sound powers propagating through a duct or an opening with and without the silencer

NOTE 1   The insertion loss is expressed in decibels, dB.

NOTE 2   Adapted from ISO 7235.

3.10
insertion sound pressure level difference
D_{ip}
difference between the sound pressure levels occurring at an immission point, without a significant level of extraneous sound, without and with the silencer installed

NOTE 1   The insertion sound pressure level difference is expressed in decibels, dB.

NOTE 2   Adapted from ISO 11820.

3.11
transmission loss
D_t
difference between the levels of the sound powers incident on and transmitted through the silencer

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

NOTE 2   For standard test laboratories D_t equals D_i, whereas results for D_t and D_i obtained from *in situ* measurements may often differ due to limited measurement possibilities.

NOTE 3   Adapted from ISO 11820.

3.12
discontinuity attenuation
D_s
that portion of the insertion loss of a silencer or silencer section due to discontinuities

NOTE   The discontinuity attenuation is expressed in decibels, dB.

3.13
propagation loss
D_a
decrease in sound pressure level per unit length which occurs in the midsection of a silencer with constant cross-section and uniform longitudinal design, characterizing the longitudinal attenuation of the fundamental mode

NOTE   The propagation loss is expressed in decibels per metre, dB/m.

3.14
outlet reflection loss
D_m
difference between the levels of the sound power incident on and transmitted through the open end of a duct

NOTE   The outlet reflection loss is expressed in decibels, dB.

3.15
modes
spatial distributions (or transverse standing wave patterns) of the sound field in a duct that occur independently from one another and suffer a different attenuation

NOTE   The fundamental mode is least attenuated. In narrow and in lined ducts, higher-order modes suffer substantially higher attenuation.

3.16
cut-on frequency
lower frequency limit for propagation of a higher-order mode in a hard-walled duct

NOTE 1   The cut-on frequency is expressed in hertz, Hz.

NOTE 2   In a duct of circular cross-section, the cut-on frequency for the first higher-order mode is $f_{cC} = 0{,}57c/C$ where c is the speed of sound and C is the duct diameter. In a rectangular duct with larger dimension H, $f_{cH} = 0{,}5c/H$.

3.17
pressure loss
Δp_t
difference between the mean total pressures upstream and downstream of the silencer

NOTE 1   The pressure loss is expressed in pascals, Pa.

NOTE 2   Adapted from ISO 7235.

3.18
regenerated sound
flow noise
flow noise caused by the flow conditions in the silencer.

NOTE   Sound power levels of regenerated sound and pressure losses measured in laboratory tests are related to a laterally uniform flow distribution at the inlet of the silencer. If this uniform flow distribution is not attainable under *in situ* conditions, for example because of the upstream duct design, higher levels of regenerated sound and higher pressure losses will occur.

4 Specification, selection and design considerations

4.1 Requirements to be specified

4.1.1   In general, the sound pressure level (A-weighted, one-third-octave or full-octave) shall not exceed a specified value at a specified position (e.g. at a work station, in the neighbourhood, or in a recreation room). The permissible contribution from a sound source can then be determined in terms of the sound power level and the directivity index of that source using sound propagation laws and requirements concerning the allocation of contributions to several partial sound sources. The required insertion loss of the silencer is given by the difference between the permissible and the actual sound power level of the source.

In simple cases where the sound immission is determined solely by the sound source to be attenuated, the necessary insertion sound pressure level difference of the silencer can be calculated directly from the difference between the permissible and the actual sound pressure level at the immission point. When the difference in directivity indices with and without the silencer is negligible, the insertion sound pressure level difference equals the insertion loss of the silencer.

4.1.2   The permissible pressure loss shall not be exceeded.

NOTE   This requirement should be specified as clearly as possible. Instead of the imprecise specification "as small as possible", a sensible limit value has to be found. Even if the pressure loss is considered as "not critical", a limit value should be determined from the maximum permissible flow velocity that may not be exceeded for reasons of mechanical stability, regenerated sound or energy consumption costs.

4.1.3   The permissible size of the silencer shall be kept as small as possible (for reasons of cost and weight).

NOTE   There is a minimum size which (given the state of the art) cannot be reduced. This size depends on the required reduction in sound level, the permissible pressure loss and on other restrictions concerning materials to be used (or avoided), resistance to different kinds of stress, etc.

4.1.4   Additional requirements (concerning materials, durability, leakages, etc.) result from the application of the silencer in hot, dusty, humid or aggressive gases, in pressure lines or for high sound pressure levels and vibration levels, and from the combination of silencers with devices for the control of exhaust gas, sparks and particles.

4.2 Selection and layout of silencers

Specific information on silencers can be drawn from

— laboratory measurements made in accordance with ISO 7235;

— silencer manufacturers' test data;

— theoretical models to calculate propagation loss and insertion loss for silencers with circular or rectangular cross-section;

— pressure loss and regenerated sound prediction methods.

The selection of a dissipative, a reactive or a blow-off silencer will be determined by its application or by reference to the experience presented in this International Standard.

Results obtained from computer programs for the insertion loss of dissipative silencers depend on the assumptions made concerning the magnitude and distribution of airflow resistance in the silencer and the acoustical effect of the cover [18]. Certain geometrical features like off-setting of splitters or subdividing of absorbers are not easily accessible for calculation. Calculations are most accurate for parameter variations concerning design as well as operating conditions. Effects of flow on the performance of reactive silencers are taken into account by special highly sophisticated computer software.

4.3 Design of special silencers

The design of a special silencer is usually an iterative process featuring the following stages:

a) rough specification of the dimensions of free ducts for the flow and of connected spaces for the distribution of sound, for example using the manufacturers' declarations for similar silencers and taking into account the essential requirements and restrictions;

b) construction of a model for predictive calculation or measurements;

c) use of the model and comparison of the results with requirements concerning sound level reduction and pressure loss;

d) change of dimensions and sound-absorbent materials to fulfil requirements or to optimize the design;

e) constructional consideration of special requirements.

5 Types of silencers, general principles and operational considerations

5.1 Overview

Silencers are used to

— prevent pulsations and oscillations of the gas at the source,

— reduce conversion of the pulsations and oscillations into sound energy, and

— provide conversion of sound energy into heat.

Table 1 — Typical advantages and shortcomings of different types of silencers

Type of silencer	Advantages	Shortcomings
Dissipative silencer	Broad-band attenuation, little pressure loss	Sensitive to contamination and mechanical destruction
Reactive silencers:		
Resonator type	Tuned attenuation, insensitive to contamination	Narrow-band attenuation, sensitive to flow
Reflective type	Robust element, application for large pressure pulsations, high sound levels, contaminated flow, strong mechanical vibrations	Greater pressure loss, acoustic pass bands (frequency bands with little or no attenuation), flow sensitivity of acoustical performance

The resulting insertion loss for a silencer mounted in a duct will in general depend on all three of these mechanisms. According to the dominant attenuation mechanisms involved, silencers may be classified as (see table 1):

— dissipative silencers,

— reactive silencers, including resonator and reflective silencers,

— blow-off silencers, and

— active silencers.

5.1.1 Dissipative silencers

These provide broad-band sound attenuation by conversion of sound energy into heat with relatively little pressure loss. Precautions shall be taken to prevent coating or clogging of the surface of the absorbent material when dissipative silencers are used in ducts carrying gases contaminated with dust or encrusting material. Porous absorbers made of fine fibrous material or thin-walled structures may be mechanically destroyed by high amplitudes of alternating pressure.

5.1.2 Resonator silencers (reactive)

These reduce the conversion of gas pulsations and oscillations into sound energy and absorb sound. Single resonators are mounted as side branches in duct walls. Groups of resonators are used as duct linings or splitter elements (baffles) in ducts, thus causing a limited pressure drop. Resonances are mostly tuned to low and intermediate frequencies, where attenuation is needed. The performance is limited to a narrow frequency band, is sensitive to grazing flow and may (under certain unfavourable conditions) be negative so that a tone is generated.

5.1.3 Reflective silencers (reactive)

These reduce the conversion of gas pulsations and oscillations into sound energy. They are usually chosen for their robustness in applications where purely dissipative silencers are less suitable, and where greater pressure loss is permissible. This is the case, for example, with gas flows carrying dust, or with higher flow velocities and pressure pulsations, and for applications with strong mechanical vibrations. The maximum attenuation and the frequency where it occurs will be affected by the flow. It is possible that in some frequency bands only little or even negative attenuation is encountered.

5.1.4 Blow-off silencers

These are mounted on steam and pressurized air release lines and are effective by reaction on the source of sound, such as a valve, and by lowering the exit flow velocity through an expanded surface area while conversion of sound into heat is usually of little significance. Large pressure losses require the silencer to have a good mechanical stability. Its performance can be affected by material carried by the gas. There is also a danger of icing.

5.1.5 Active silencers

These mainly consist of speaker sets driven by amplifiers with input from suitable microphones. Control is effected by a high-performance computer, the controller. These are specialist devices not dealt with in this International Standard. Active silencers are most effective at low frequencies where passive dissipative silencers offer little attenuation [32].

NOTE   Active systems are presently offered exclusively as individual solutions tailored for particular applications and are thus not discussed in this International Standard.

5.2 Acoustic and aerodynamic performance of silencers

The attenuation required from a silencer is described in terms of the insertion loss, D_i, if no particular immission point is defined, or in terms of the insertion sound pressure level difference, D_{i_p}, at a particular position. It is specified in one-third-octave bands or full-octave bands. According to the laboratory standard ISO 7235, the attenuation shall be measured in one-third-octave bands. Full-octave-band values may be calculated using equation (1):

$$D_{1/1} = -10\lg\left(\frac{1}{3}\sum_{k=1}^{3} 10^{-\frac{D_{1/3,k}}{10\text{dB}}}\right)\text{dB} \qquad (1)$$

where $D_{1/3,1}$ to $D_{1/3,3}$ are the attenuation values in the three one-third octaves of a full-octave band, in decibels, and $D_{1/1}$ is the resulting full-octave-band value. Declaring attenuation values in full octaves will suffice for broad-band noise and for silencers with broad-band effect. For tonal noise and for resonator silencers with narrow band effect, the attenuation data should be given in one-third-octave bands.

NOTE 1   Octave-band attenuation data may strongly depend upon the spectrum of the sound (see annex B).

A necessary parameter for the selection of a silencer is the permissible pressure loss in the flow. It shall not exceed the total pressure loss Δp_t which depends on the mean flow velocity and density of the gas and on the flow condition as described by equation (2):

$$\Delta p_t = (\zeta + \Delta\zeta)\frac{\rho}{2}v_1^2 \qquad (2)$$

where

ζ is the total pressure loss coefficient as defined in ISO 7235 for uniform flow conditions at both ends of the silencer;

$\Delta\zeta$ is the additional pressure loss coefficient due to flow conditions *in situ* deviating from the laboratory conditions (values are to be estimated empirically);

ρ is the density of the gas, in kilograms per cubic metre, kg/m^3;

v_1 is the mean flow velocity in the inlet cross-section, in metres per second, m/s.

NOTE 2   It is common for definitions of the total pressure loss coefficient to differ from the one given in ISO 7235. It is therefore necessary to check the definitions before using any values. For example, a different definition is the one considering the flow velocity in the narrowest cross-section of the silencer instead of v_1. This will result in much lower values for ζ.

Other parameters to be considered which affect the acoustic and aerodynamic performance are

— the regenerated sound,

— the maximum dimensions available for the silencer, and

— the necessary durability of the silencer under exposure to flow, pressure pulsations and mechanical vibration.
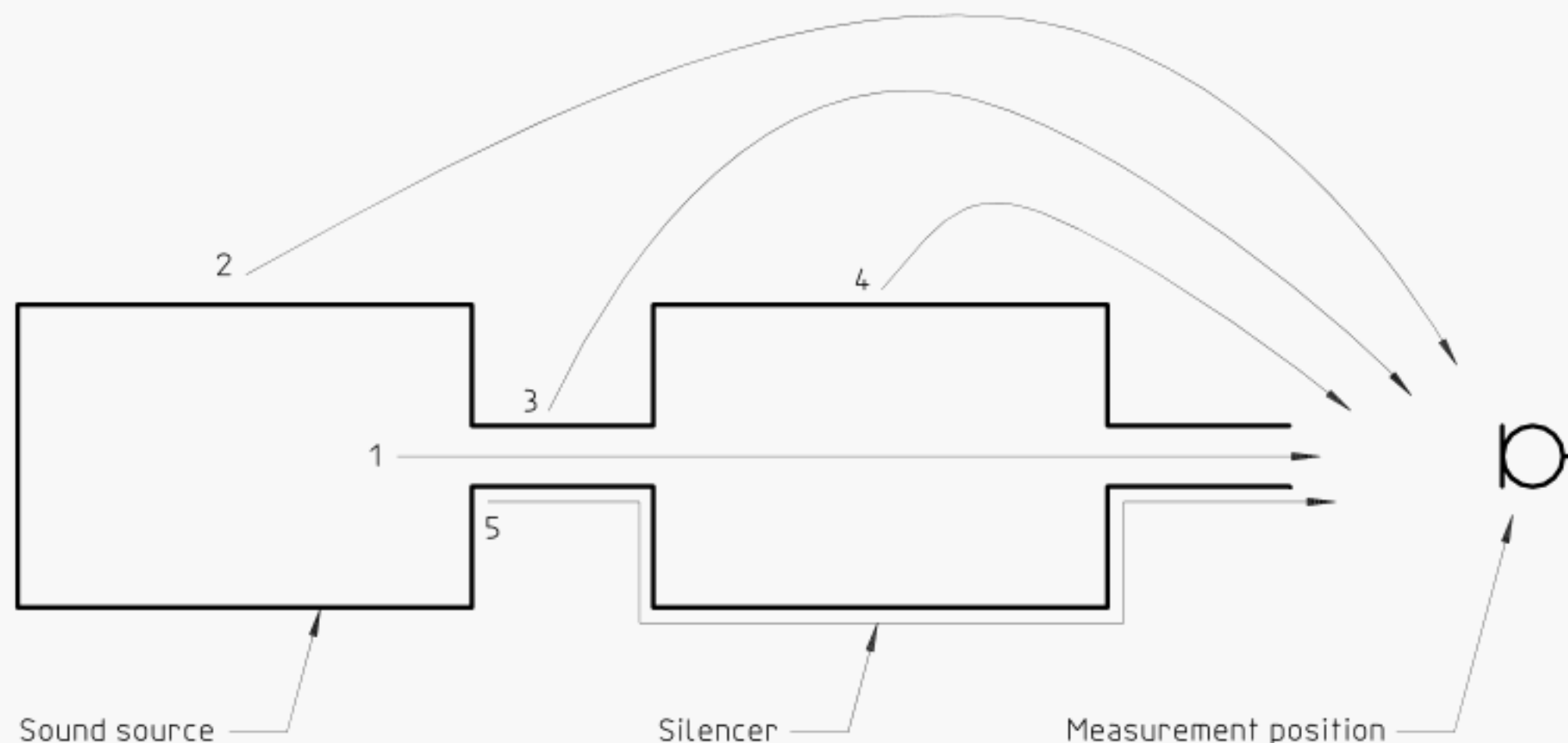
5.3 Sound propagation paths

It is possible for sound propagating from a source to an immission point to follow several paths beside the direct one through the silencer (Figure 1, path 1). The additional paths are:

a)   radiation from the housing of the source (path 2);

b)   radiation from duct walls before the silencer (path 3);

c) radiation from the shell of the silencer (path 4); and

d) propagation of structure-borne sound along and past the silencer (path 5).

Sound propagation along these flanking paths shall be prevented by providing housings and duct walls with sufficient sound insulation and by inserting vibration isolation devices for interrupting the propagation path for structure-borne sound.

Figure 1 — Sound propagation paths (schematic)

5.4 Acoustic installation effect

For certain applications and silencer types, the sound attenuation provided by a silencer depends on the characteristics of the source connected to the inlet side and the characteristics of the termination connected to the outlet side. Such an installation effect occurs especially on reactive silencers or on all types of silencers for low frequencies.

It is also important that either the source or the termination is reactive, i.e. non-absorbing. When these conditions are fulfilled, unfavourable resonance effects can be expected in the system that will lead to strong coupling between different parts of the system. Formally, this type of installation effect can be described via equation (3):

$$L_W(\text{rad}) = L_W(\text{source}) - D_t - D_m + E \qquad\qquad (3)$$

where

$L_W(\text{rad})$ is the level of sound power radiated from duct end, in decibels, dB,

$L_W(\text{source})$ is the level of sound power radiated from source into duct with anechoic termination, in decibels, dB;

D_t is the transmission loss (see 3.11), in decibels, dB;

D_m is the reflection loss at the duct outlet (see 3.14 and 6.2.2.2), in decibels, dB;

E is the acoustic installation effect, in decibels, dB; in dissipative systems; the magnitude of E generally does not exceed 10 dB.

The reaction of reflected sound on the source described by E can result in an increase or a decrease of sound emission.

NOTE   For strongly reactive systems, E can be a large positive quantity in narrow frequency bands, which implies that the silencer system actually amplifies the sound power radiated from the source.

5.5 Abrasion resistance and protection of absorbent surfaces

Abrasion of the materials used in dissipative silencers may lead to particles of the infill being carried in the gas flow.

NOTE   Little is known about the particle number concentration in the gas stream for longer operation of silencers.

If the surface of a sound-absorbent material is mechanically damaged, low flow velocities will suffice to carry away large numbers of particles through erosion. This process may even result in the depletion of a whole absorbent element (such as a splitter).

To protect the sound-absorbing infill of silencers against moisture, water or pollutants carried in the gas (in particular in hospitals and in the food processing industry), foils are used for airtight sealing. Such foils not only reduce the attenuation performance at high frequencies (typically above 1 kHz) but may also rupture during plant operation. A difference in total (i.e. static and dynamic) pressures inside and outside the sealed element causes stress in the foil. High temperatures and impacting sharp (and hot) particles increase the risk of damage. Thus, the protection of sound-absorbing infill by means of foil needs careful consideration of foil thickness, temperatures, flow velocities and contamination of the gas.

5.6 Fire hazards and protection against explosion

There is a particular danger of fire being started or transmitted by ventilation silencers for technical equipment if oil aerosols are carried. This applies particularly to chemical laboratories, large kitchens and engine-testing installations. Organic substances like flour or milk powder may form explosive mixtures with air, and this shall be taken into account where dust-carrying gases flow through the silencer.

In all these applications, and in accordance with many building codes, "non-combustible" materials shall be used for the silencer. Collections of grease, oil or dust in the absorbent material shall be prevented by using appropriate shapes and arrangements of silencers. Resonator silencers without absorbent material and with precautions against dust deposit are also suitable to meet fire- and explosion-protection requirements.

5.7 Starting-up and closing-down of plants

Silencers in technical plants may cause problems when the plant is started up or closed down. Sufficient space shall be provided for the expansion of components of the silencer to allow for considerable changes in pressure and/or temperature. Particularly in the case of pressure variations and for foil covers, pressure relief shall be possible in the absorbent lining.

In the starting-up and closing-down phases of plants, there are frequently temperatures below the dew point, especially inside the absorbent linings and on the inside of the silencer housing. Collection of moisture should be prevented (for instance by "dry-running" the plant); particular corrosion problems may arise. Condensed liquid should be allowed to drain.

5.8 Corrosion

Sheet metal shells, covers and partitions of silencers as well as mounting flanges shall be protected from the effects of weather, acids in exhaust gases, and differences in voltage potentials of different materials. Corrosion can be prevented by selection of particular material (e.g. aluminium) or by application of protecting covers (e.g. rubber).

5.9 Hygienic requirements and risk of contamination

Special requirements shall be met, for example,

— in cleanrooms,

— in food-processing plants,

— in hospitals,

—   in power plants.

Hygienic problems can arise when dust is deposited on the adhesive surfaces of sound-absorbent linings, particularly in combination with humidity. Viable particles (bacteria) can also pose a problem, especially if the air temperature is elevated. Nuclear contamination may occur in nuclear power plants.

Smooth surfaces shall be used for silencer linings in such critical plants. Large cavities and protruding edges shall be avoided because they encourage the collection of dust and damp and enhance the pressure loss.

5.10 Inspection and cleaning, decontamination

Provision for the inspection, cleaning or replacement of silencers or splitters should be made where needed.

The special requirements in most applications of HVAC equipment make cleaning or decontamination necessary at intervals. It is therefore necessary that elements (splitters) can be dismounted for cleaning (decontamination) or replacing. In this case, the silencer housing shall be cleaned as well. Splitters can be cleaned using pressurized air, steam jets, brushes and solvents, or decontamination fluid, depending on the design.

A dust deposit forming on splitters after a certain operating time in dusty flow will lead to a considerable decrease in insertion loss. Here too, provisions should be made to allow for cleaning at intervals.

6 Performance characteristics of types of silencers

6.1 Dissipative silencers

6.1.1 Simple dissipative silencers

A simple dissipative silencer is a straight duct with a sound-absorbent lining, of circular or rectangular cross-section and without any fittings (see Figure 2).
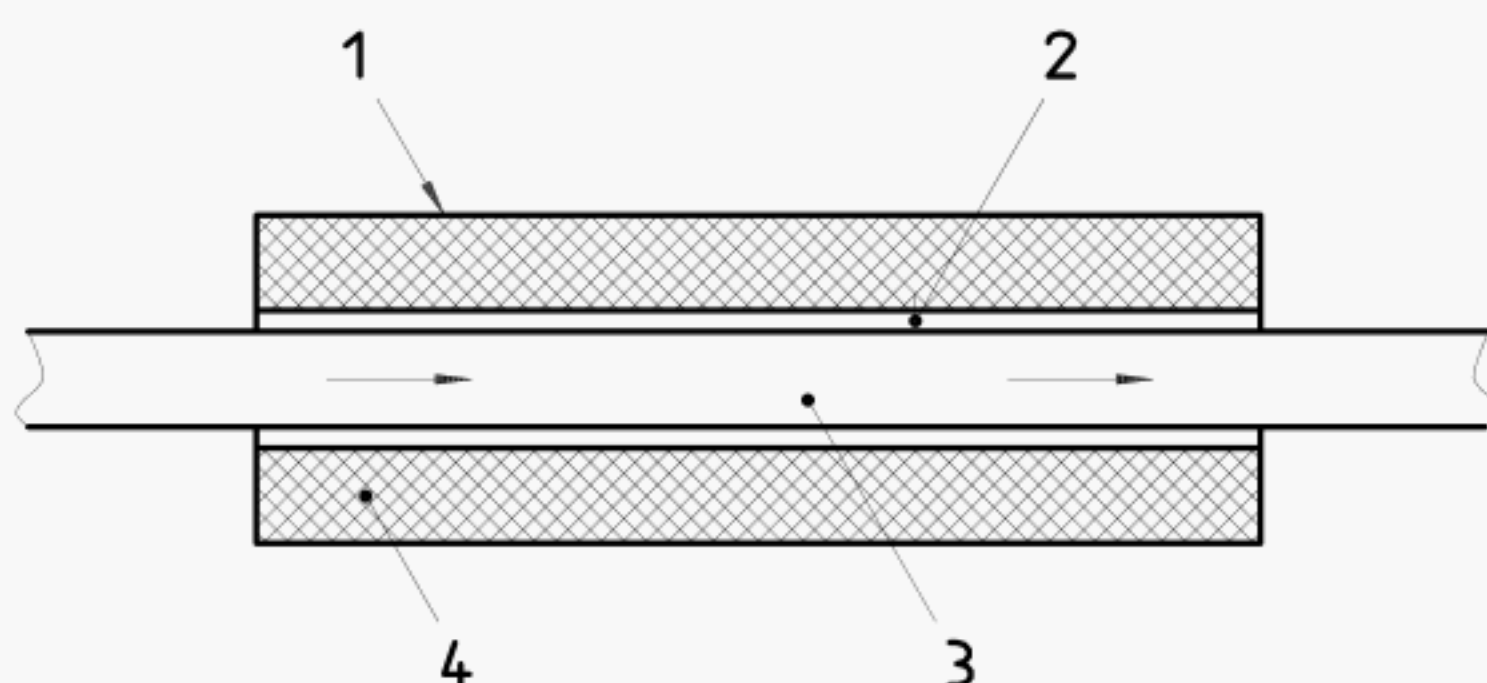
Key:

1   Shell
2   Sound-permeable cover
3   Flow duct
4   Sound-absorbent material

Figure 2 — Dissipative silencer (schematic)

A sound-absorbent element consists of one or more layers of absorbent material and a sound-permeable cover. Fine mineral, metal or plastic fibres and open-pore structures made of foam, sintered metal or concrete are used as absorbent material. In coarse-grained structures, the viscosity of the air will have a smaller effect than turbulence. In this case, pressure differences will increase with the square of the flow velocity. Such non-linear effects can be found in silencers with flow through or tangential to the absorber. For covering fibre materials and foams subject to high stress, perforated sheet metal, diamond mesh or rib mesh combined with close-meshed wire screen. glass or steel fibre cloth should be used. For moderate stress conditions, thin foil, fibre glass or plastic fleece should be used.

The transmission loss D_t (or insertion loss D_i;. see 3.11) of the simple dissipative silencer can be described by

$$D_t = D_s + D_a l \qquad (4)$$

where

D_s is the discontinuity attenuation, in decibels, dB;

D_a is the propagation loss along the silencer, in decibels per metre. dB/m;

l is the length of the silencer, in metres, m.

The discontinuity attenuation can be calculated from laboratory measurements on two different lengths l_1 and l_2 of a type of silencer. If the insertion losses D_{i1} and D_{i2} are measured for l_1 and l_2 without the influence of flanking transmission within or around the silencer, the discontinuity attenuation D_s can be determined from equation (5):

$$D_s = \frac{D_{i1} l_2 - D_{i2} l_1}{l_2 - l_1} \qquad (5)$$

The propagation loss is determined from such measurements as:

$$D_a = \frac{D_{i2} - D_{i1}}{l_2 - l_1} \qquad (6)$$

For a qualitative estimate of the propagation loss D_a, Piening's proportionality can be used:

$$D_a \propto \frac{U}{S} \alpha \qquad (7)$$

where

U is the length, in metres, m, of the duct perimeter lined with sound-absorbent material;

S is the cross-sectional area of the duct, in square metres, m^2;

α is the sound absorption coefficient of the lining.

The greater the ratio of the surface area Ul of the absorber to the cross-section S of the duct, and the higher the absorption coefficient α of the duct lining, the more effective the dissipative silencer will be. Small sound-reflecting surfaces will reduce the effect only slightly.

The free area S of the cross-section is dependent on the maximum permissible flow velocity. This flow velocity shall not be exceeded because of its relationship to the service life of the silencer, the pressure loss and the regenerated sound. If the area is adapted to connecting ducts, the cross-section may also be round or rectangular. Equation (7) indicates that narrow, rectangular openings with the larger sides being sound-absorbent are to be preferred. Openings like this will also suppress beam formation which occurs when the distance between the walls exceeds half the wavelength of the sound.

A high sound absorption coefficient is only possible when the thickness of the lining is at least one-eighth of the sound wavelength. This criterion can be fulfilled in simple dissipative silencers even for low frequencies if a sufficiently large cross section is available at the location where the silencer is to be mounted. The proportionality in equation (7) to the sound absorption coefficient of the lining breaks down when the duct width becomes significantly smaller than half the wavelength of the sound to be attenuated. Furthermore, the formula does not apply at high frequencies when the sound propagates like a beam without hitting the lining at all.

A sound-absorbent material is characterized by its airflow resistivity r [29] (ranging for silencer applications from 5 kN·s/m^4 to 50 kN·s/m^4). The airflow resistivity is related to the fibre diameter and material bulk density according to equation (8):

$$r \propto \frac{\eta}{a^2}\left(\frac{\rho_c}{\rho_u}\right)^{3/2} \tag{8}$$

where

ρ_c is the bulk density, in kilograms per cubic metre, kg/m^3, of the compressed absorber material;

ρ_u is the bulk density, in kilograms per cubic metre, kg/m^3, of the uncompressed absorber material,

η is the viscosity of the gas, in newton seconds per square metre, N·s/m^2;

a is the average diameter of the fibres, in metres, m.

The influence of temperature and pressure on the specific airflow resistance $R_S = rd$ of a material layer of thickness d is approximately described bv equation(9):

$$\left[\frac{R_S}{\rho c}\right]_{T,p} = \left(\frac{T}{T_0}\right)^{1,2}\frac{p_0}{p}\left[\frac{R_S}{\rho c}\right]_{T_0,p_0} \tag{9}$$

where

T is the absolute temperature, in kelvins, K;

T_0 is the reference temperature, in kelvins, K;

p is the gas pressure, in pascals, Pa;

p_0 is the reference pressure, in pascals, Pa;

ρc is the characteristic impedance of the gas, in N·s/m^3, for plane waves.

Typical temperatures to be expected for various sound sources and temperature limits for various sound-absorbent materials are listed in annex B.

Examples for the propagation loss in ducts of circular cross-section with linings of different thickness are shown in Figure 3. They are based on rigorous calculations without flow and typical data for the airflow resistivity of mineral wool. The thickness of the lining has a strong effect on the attenuation performance at low frequencies.

In some circumstances it is necessary to protect the environment from the silencer infill or the infill from the gas flow. This can be done by thin impervious or perforated covers. For broad-band attenuation, the effective mass per unit area of the cover should be kept as small as possible. The effective mass is either the weight of an impervious cover or the mass of the air oscillating near the perforated cover divided by the fraction of open area.

NOTE   Often a surface weight of the impervious cover of less than 0,033 kg/m^2 or a porosity of the perforated cover of more than 30 % is sufficient.

Ensure that impervious covers do not stick to the infill or, in the case of multiple layers of different covers to the perforated cover which will reduce the mobility.

Key:

t

----- 0,15 m

——— 0,10 m

········· 0,05 m
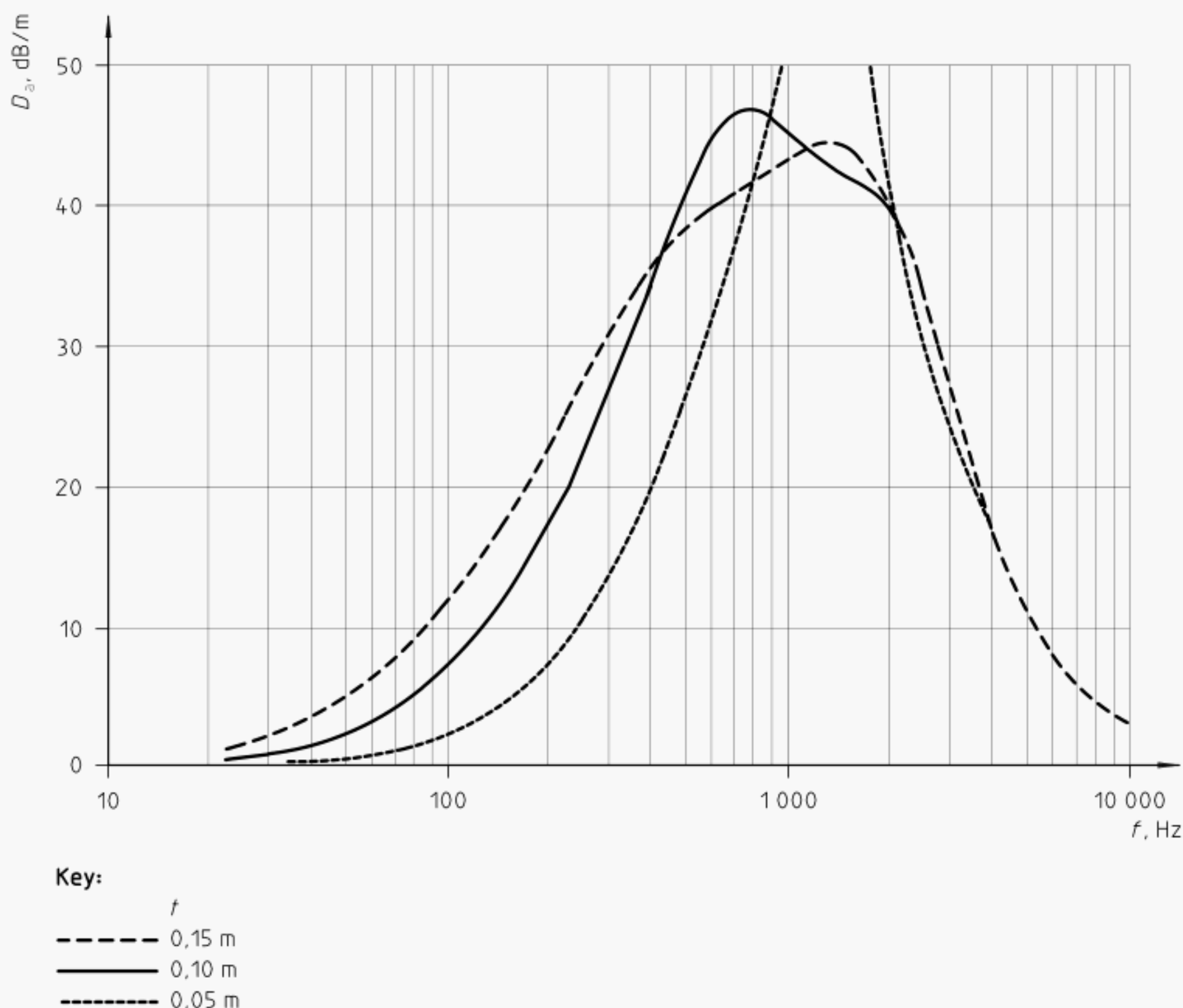
Free duct diameter:	$D = 0,2$ m

Airflow resistivity of isotropic absorber: $r = 12$ kN·s/m^4

Specific airflow resistance of lining surface modelling the effect of a dust deposit or a close-fitting porous cover: $R_s = 0,2$ kN·s/m^3

Figure 3 — Calculated propagation loss D_a vs. frequency f for a simple dissipative silencer with circular cross-section and lining thickness t

For enhanced low-frequency attenuation, thicker impervious covers or perforated covers with lower porosity are sometimes used.

Frequent starting-up and closing-down of furnaces may lead to humidity collecting in flue gas silencers (see A.2.4). Plastic foil cannot completely prevent steam diffusion and will allow water to collect in the absorber, particularly when the foil is damaged.

Absorbers shall be mechanically and thermally stable and their shape or structure shall not change due to mechanical vibrations throughout their agreed service life.

6.1.2 Splitter silencers

6.1.2.1 General considerations

The factors that determine the acoustic performance of splitter silencers are, essentially, the same as those for simple dissipative silencers described in 6.1.1.

A splitter silencer generally consists of a transition element which serves to expand the duct cross-section, a midsection containing sound-absorbent splitters (or baffles) and gaps or airways to channel the flow, and a second transition element to concentrate sound and flow to the original duct cross-section. This is illustrated in Figure 4. In special cases, the transition elements at both sides are omitted or are not to be considered part of the silencer if so agreed by the parties involved.
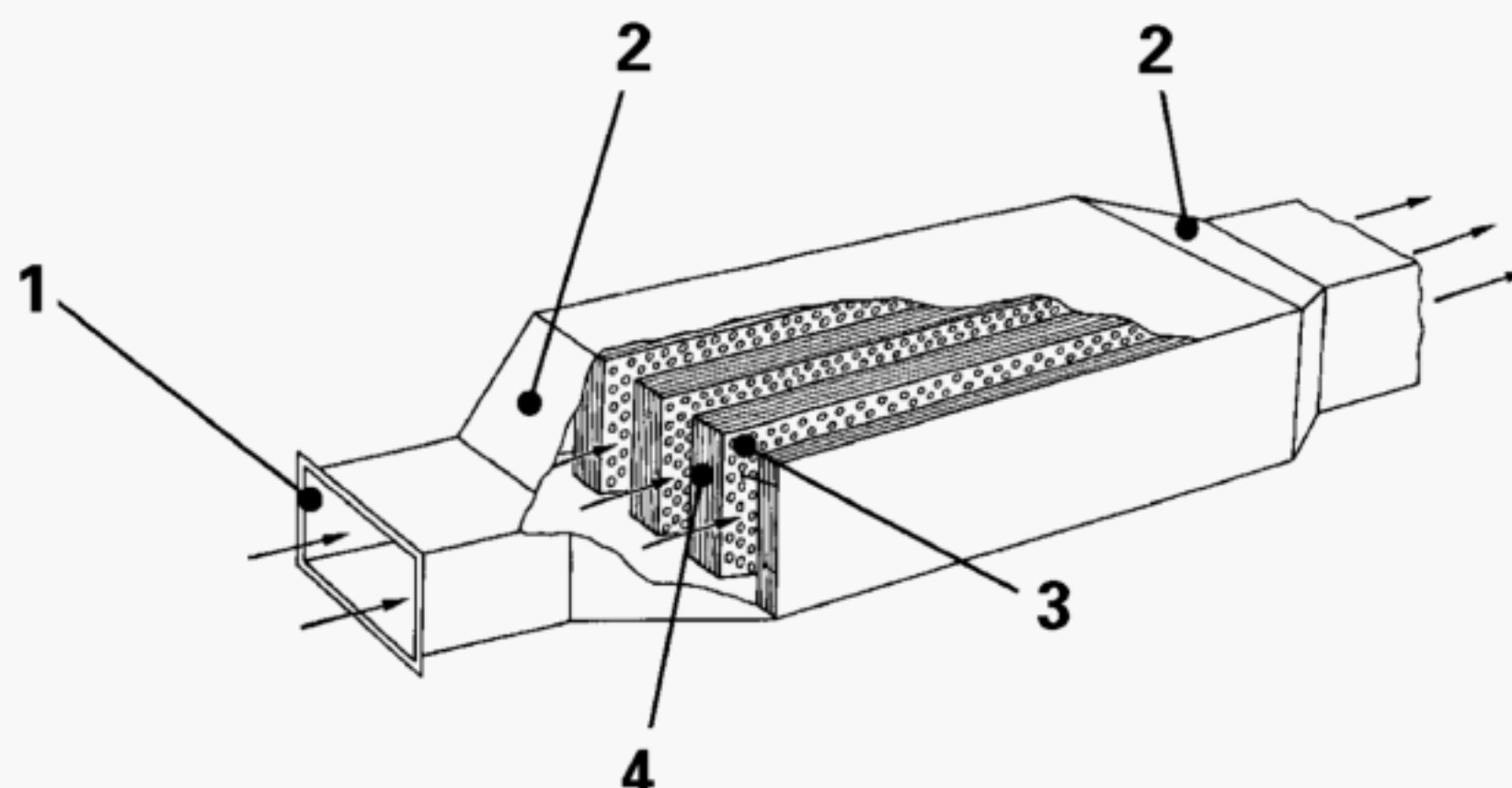
Key:

1   Entry cross-section
2   Transition elements
3   Sound-permeable cover
4   Sound-absorbent material (splitter)

Figure 4 — Splitter silencer

Providing a number of parallel splitters and a sufficient free area S can help to achieve high sound attenuation according to equation (7) at small pressure loss.

Depending on the frequency range, the insertion loss of a splitter silencer results from the contributions of a discontinuity attenuation at the inlet and the propagation loss along the splitters (see Figure 5). At low frequencies, when the diameter of the connected duct is less than half a wavelength and propagation of higher-order modes is inhibited, the discontinuity attenuation is negligible. At high frequencies, when the transition element allows for nearly random sound incidence on the splitters, the discontinuity attenuation usually lies between 6 dB and 10 dB and may exceed the propagation loss.

An additional discontinuity attenuation effective for splitters, where the internal structure changes along the propagation path, is usually small.
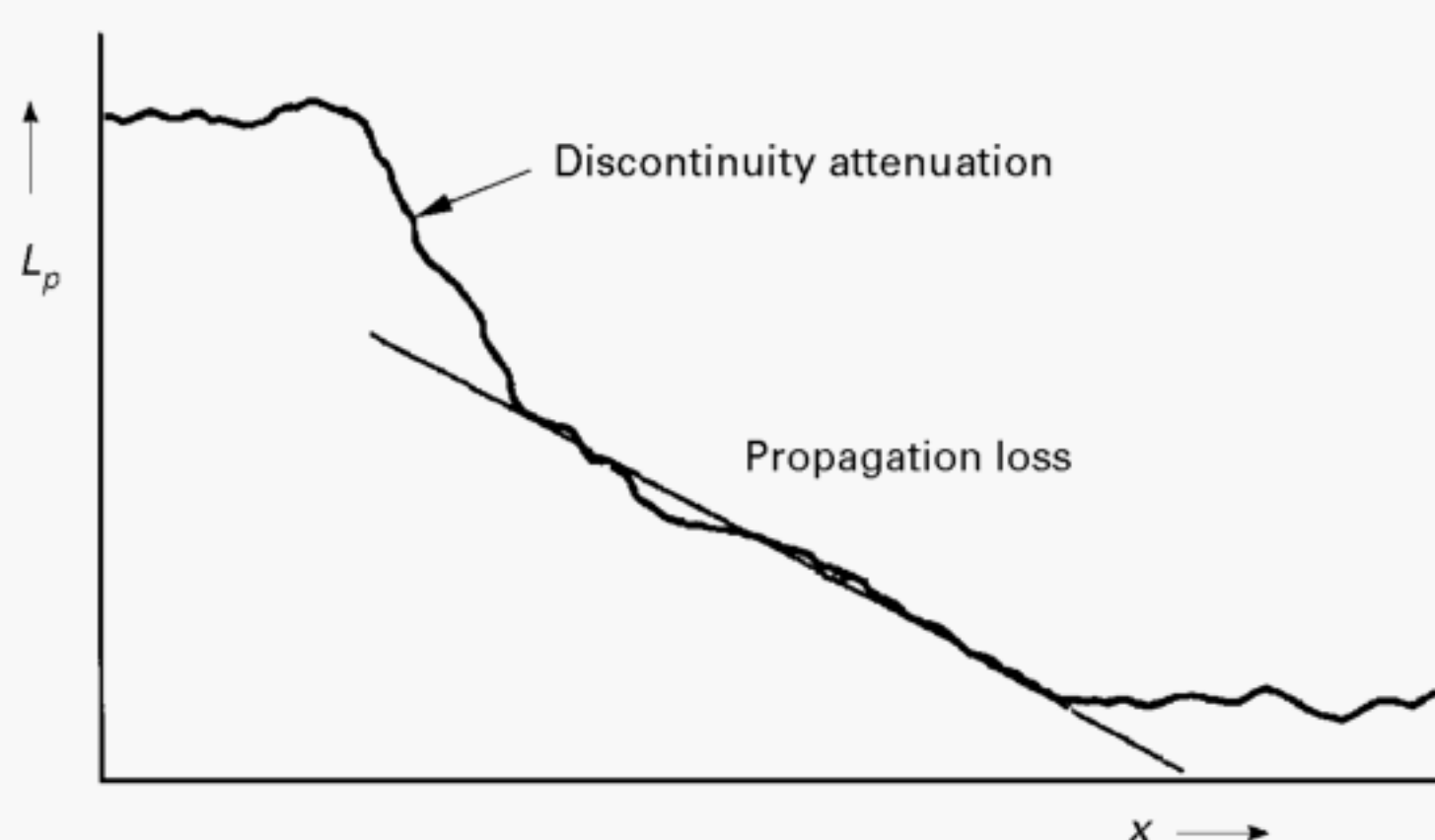
Figure 5 — Decay of sound pressure level L_p along a path length x taken by the sound in a splitter silencer

All joints between the duct walls and bottoms or tops of the splitters, which are sometimes needed as expansion gaps, shall be sealed to avoid flanking sound transmission. Airways between a splitter and a wall may only be half as wide as those between two splitters. If reduced flow through the side airways is to be avoided, a boundary splitter should be mounted to the wall.

NOTE   From the acoustical point of view, this splitter need only be half as thick if it is uniformly structured.

When mounting splitters with non-uniform structure, such as splitters with partial cover, attention should be paid to mounting instructions. As a rule, the surfaces of two splitters forming an airway shall have the same structure, i.e. the surfaces may vary along but not across the airway.

To ensure the durability of splitters subject to flow with velocities in the airway exceeding 5 m/s, precautions shall be taken to ensure uniform flow conditions, for instance by using flow rectifiers. Transverse flow through the splitters will result in splitter material being carried away and shall be avoided. Therefore, it is not recommended to place splitters closely behind large changes in cross-section and/or bends in the duct, otherwise guide vanes shall be provided to ensure a uniform flow distribution.

Splitters completely covered with foil for application in a humid atmosphere may be subject to internal overpressure (see 5.5). Foils may rupture during plant operation. They also reduce the high-frequency attenuation performance.

In many cases, it is helpful to provide access to the splitters for inspection and maintenance. Openings for sound-measuring purposes shall be considered in the design. Where particular hygienic requirements shall be met, it shall be possible to remove the splitters for decontamination.

### 6.1.2.2 Splitters for broad-band attenuation

Splitters with uniform absorber filling will provide for several octave bands of attenuation, depending on splitter thickness, width of airway, protective cover, off-set of alignment, and contamination. For low frequencies, a high absorption coefficient requires thick splitters, whereas thin splitters will suffice for high frequencies. Typical frequency characteristics for the performance of a splitter silencer can be seen from the example in Figure 6. At low frequencies the propagation loss will increase as absorber thickness and frequency are increased. In the mid-frequency range where the duct width and half the sound wavelength coincide, a maximum value will be found which is inversely proportional to the airflow resistance of the absorber. The total specific airflow resistance perpendicular to the splitter should not significantly exceed 2 kN·s/m^3. The propagation loss will drop to very low values at higher frequencies where the duct width or width of airway between splitters are greater than half the sound wavelength.

Key:

	R_s
	kN·s/m^3
- - - - -	0
————	0,2
--------	0,4
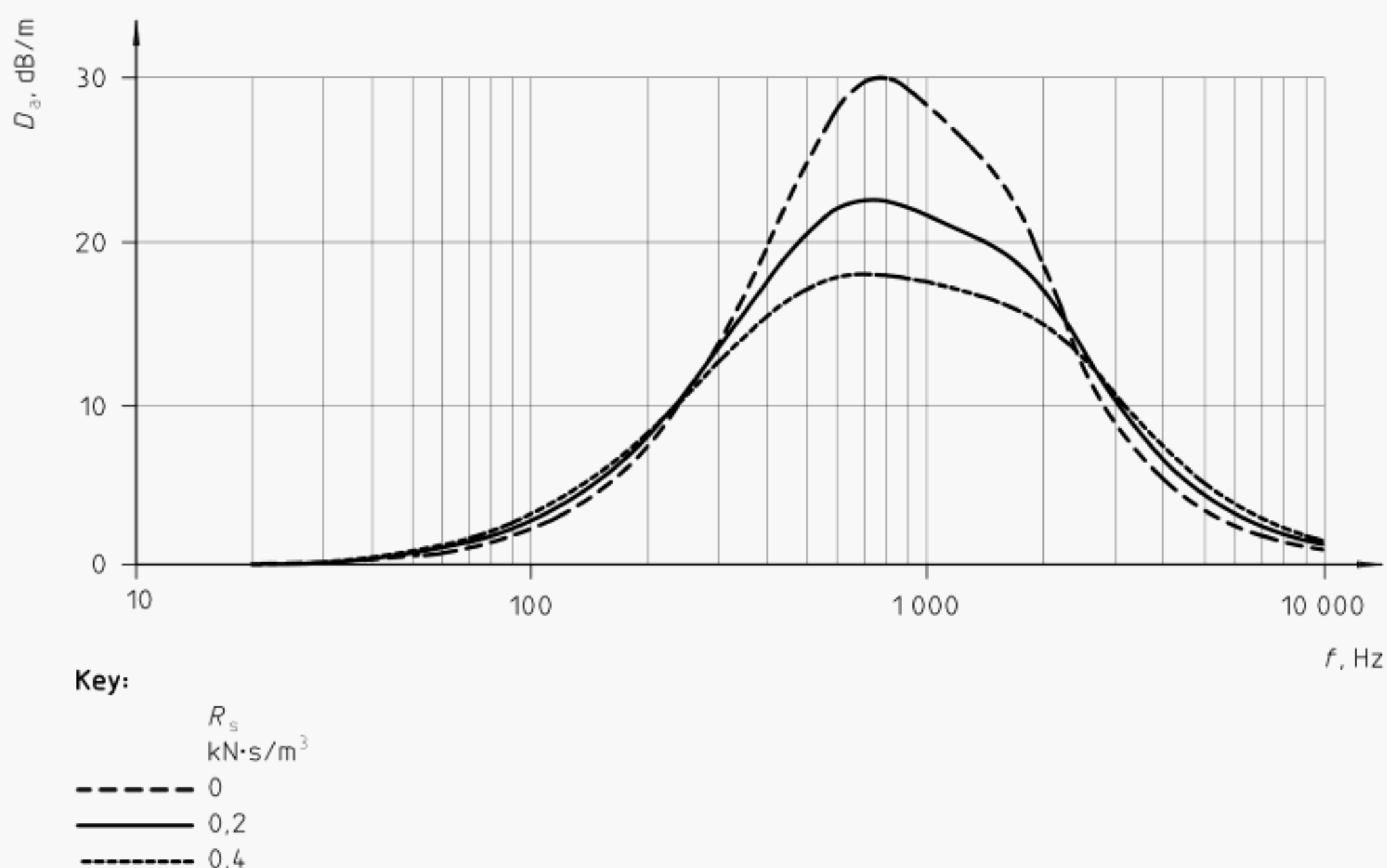
Thickness of splitter: $d = 0,2$ m

Width of airway between splitters: $s = 0,2$ m

Airflow resistivity of isotropic absorber: $r = 12$ kN·s/m^4

NOTE   The effect of a dust deposit or a close-fitting porous cover is modelled by the specific airflow resistance R_S of a covering layer.

Figure 6 — Calculated propagation loss D_a vs. frequency f for a splitter silencer

Key:

	d
--------	0,15 m
————	0,2 m
– – – –	0,3 m
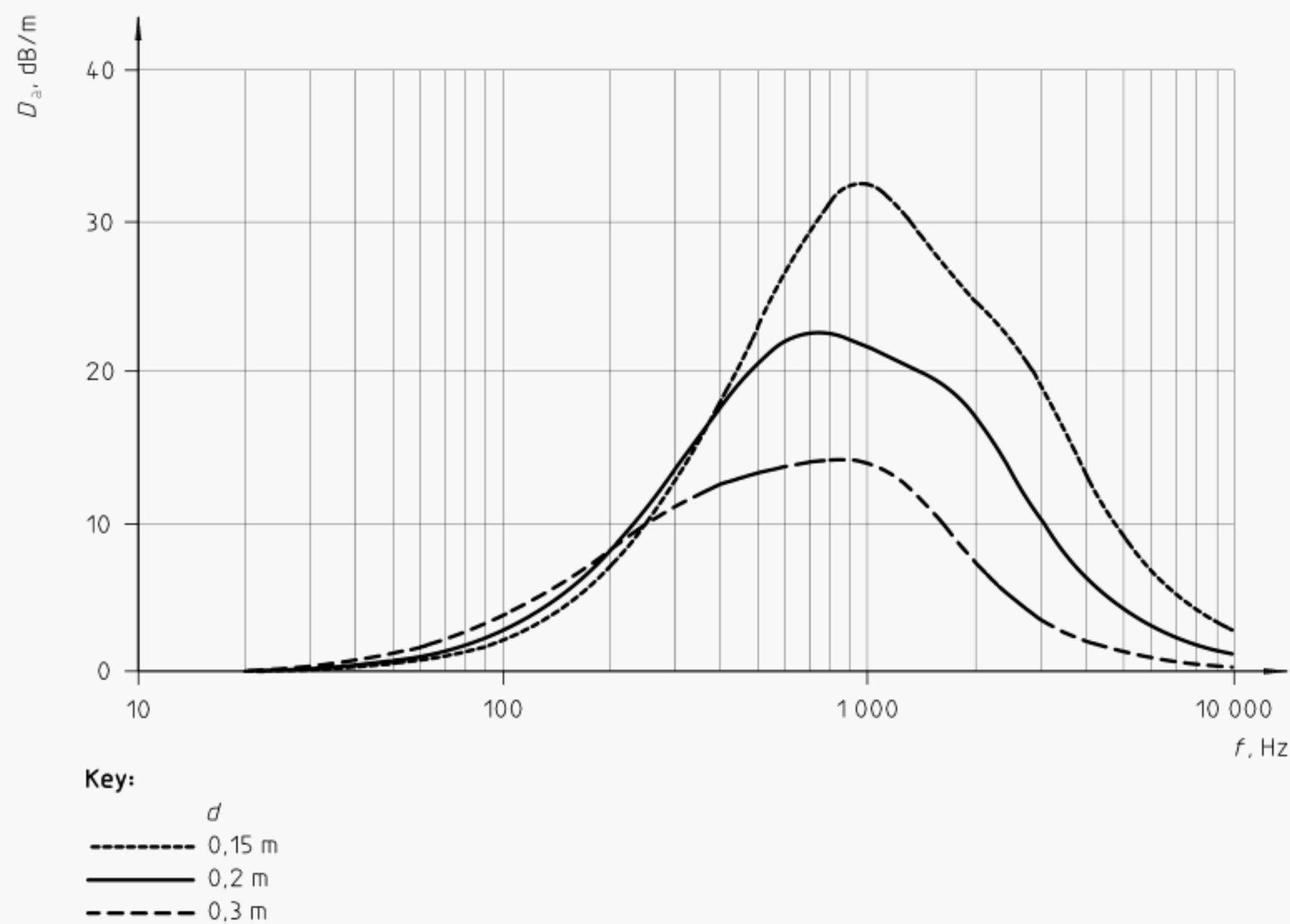
Airflow resistivity of isotropic absorber: $r = 12$ kN·s/m^4

Flow resistance of lining surface: $R_s = 0,2$ kN·s/m^3

Figure 7 — Calculated propagation loss D_a vs. frequency f for a silencer with splitters of different thickness d and airway width between splitters equal to the splitter thickness

The effect of splitter thickness is shown in Figure 7. When the same fraction of the duct cross section is blocked by splitters (i.e., s/d = constant), thicker splitters slightly enhance the silencer performance at low frequencies but yield only a moderate attenuation at mid frequencies and a marginal propagation loss at high frequencies.

To enhance absorption at low frequencies at the expense of high-frequency attenuation, covers with greater area masses are used (see Figure 8).

When selecting and optimizing splitter silencers for low-frequency attenuation, attention shall be paid to the choice of infill, cover and internal partitioning of the splitters. To improve the high-frequency attenuation performance. the airway width can be reduced or separate splitters can be positioned along the duct with an off-set. Both measures result in increased pressure loss. While the off-set provides an additional attenuation of less than 6 dB. the pressure loss may almost double (see Figure 9).

A marked decrease in attenuation at high frequencies can be expected if there is a free line of sight from the inlet of the silencer to its outlet.

Contamination of splitters generally results in reduction of attenuation performance at mid and high frequencies.

Dimensions in millimetres
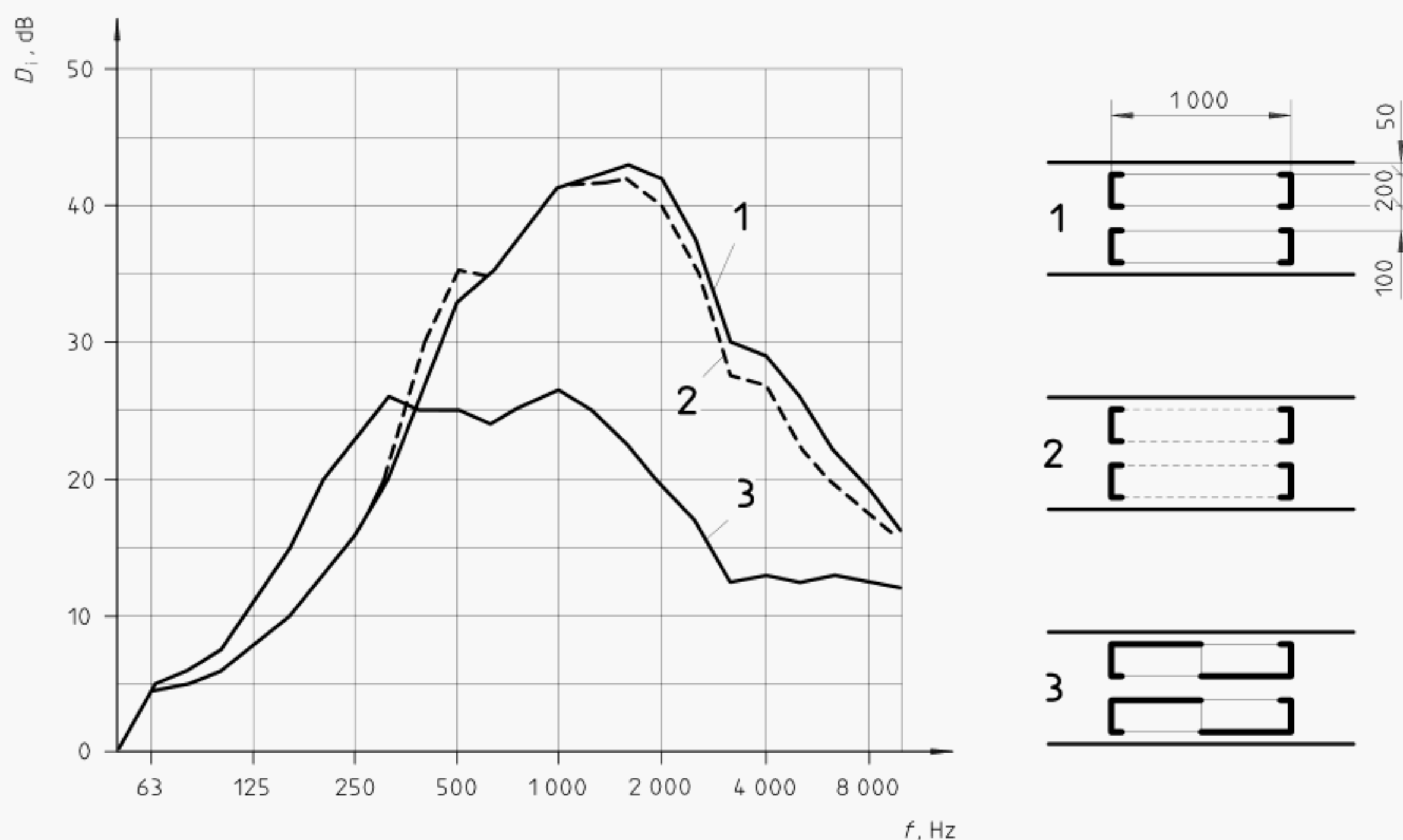
Key:

1   Splitters without cover
2   Splitters with perforated-plate cover
3   Splitters with alternating cover

Figure 8 — Insertion loss D_i vs. frequency f of silencers with common splitters according to laboratory measurements

6.1.2.3 Pressure loss

The total pressure loss produced by a silencer [see equation (2)] is a decisive factor in the choice of splitters and airway widths. It comprises the pressure losses at the inlet, outlet and along the airway between splitters. For the selection of a silencer, the permissible total pressure loss must be known. In the case of uniform non-rotating inflow to the silencer and ducts where the cross-section remains constant, an estimate for the pressure loss at both ends can be drawn from the pressure loss coefficient ζ_s (in relation to the total duct cross-section):

$$\zeta_s = \left(\frac{d}{s}\right)^2 \left[0{,}5\zeta_1\left(\frac{s}{d}+1\right) + \zeta_2 \right] \tag{10}$$
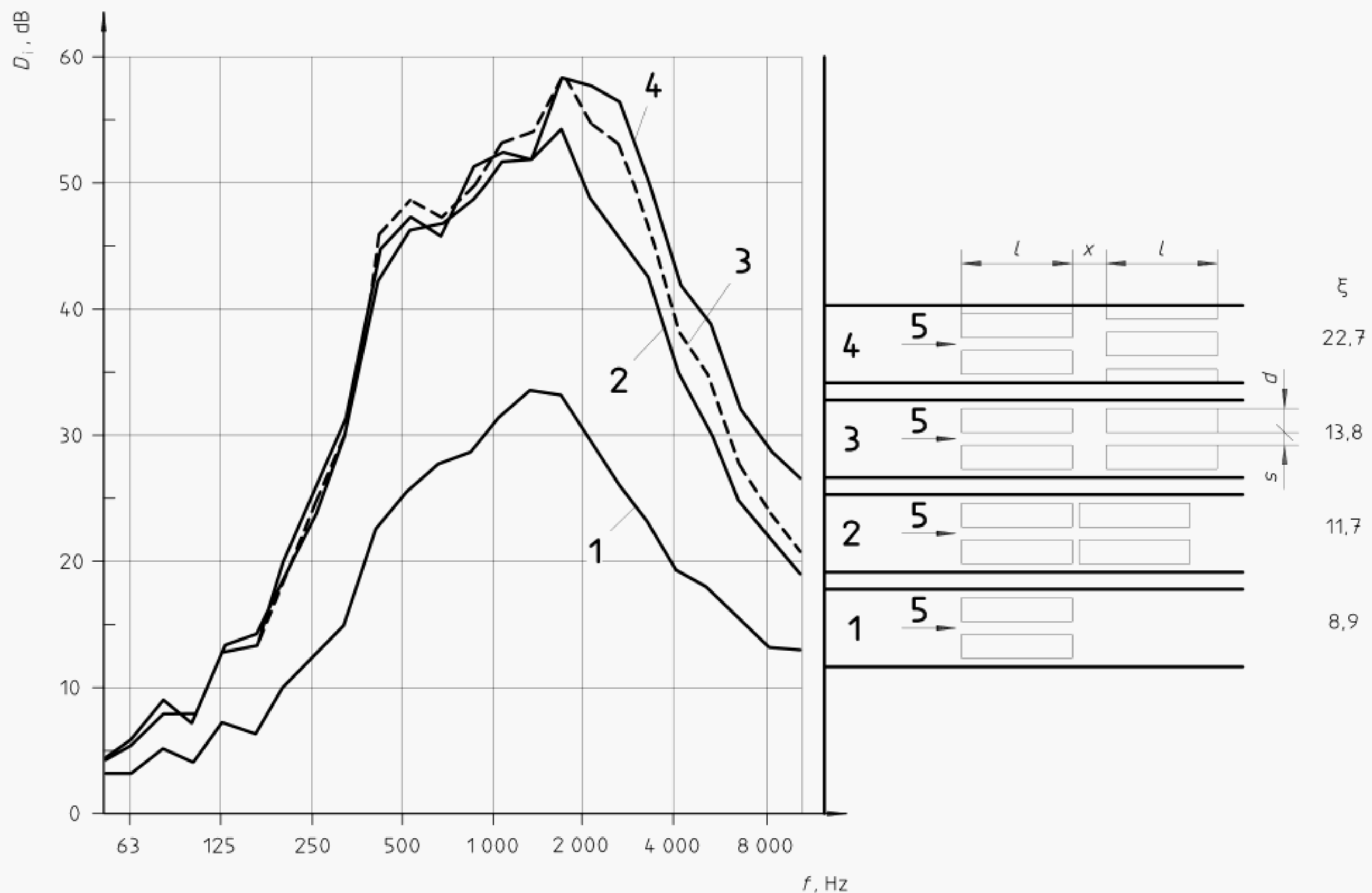
where

ζ_1 is the form factor for the upstream side; for rectangular splitters, $\zeta_1 = 1$; for splitters with semicircular upstream side profile $\zeta_1 \approx 0{,}1$;

ζ_2 is the form factor for the downstream side; for rectangular splitters $\zeta_2 = 1$; for splitters with semicircular end profile $\zeta_2 \approx 0{,}7$ (small effect);

s is the width of the airway, in metres, m;

d is the thickness of the splitter, in metres, m.

Key:

l = 0,75 m	s = 0,1 m	1 to 4	Splitters
x = 0,1 m	d = 0,2 m	5	Direction of flow

NOTE   Values larger than 40 dB are subject to effects of flanking transmission.

Figure 9 — Insertion loss D_i vs. frequency f and pressure loss coefficient ζ for various splitter arrangements according to laboratory measurements

On the whole, these pressure losses will grow with the square of the ratio d/s. Frictional losses increase with increasing ratio of splitter length, l, and with the hydraulic cross-section which is proportional to the airway width s. For sound-absorbent splitters with or without a perforated plate cover, the pressure loss coefficient ζ_f due to friction can be estimated from:

$$\zeta_f = 0,025 \frac{l}{s}\left(1 + \frac{d}{s}\right)^2 \tag{11}$$

The value 0,025 is typical for half the friction coefficient of dissipative splitters.

Therefore, to keep pressure losses within acceptable bounds, splitters shall not be too thick and airway widths shall not be too small.

For comparison with laboratory measurements in accordance with ISO 7235, the total pressure loss is calculated from

$$\Delta p_t = \left(\zeta_S + \zeta_f\right)\frac{\rho}{2}v_1^2 = \zeta \frac{\rho}{2}v_1^2 \tag{12}$$

NOTE   The measurement conditions in accordance with ISO 7235 result in $\Delta\zeta = 0$ [see equation (2)].

6.1.2.4 Effects of flow on attenuation and regeneration of sound

Flow with a velocity up to about 20 m/s in the airway will barely have an impact on dissipative attenuation.

Flow can affect sound dissipation in splitter silencers in two ways. First, the speed of sound is different in the upstream and downstream directions. Secondly, velocity profiles cause refraction effects. Both effects depend on the Mach number Ma and are negligible for $Ma < 0,05$.

Most important is the regeneration of sound by flow. The regenerated sound (or flow noise) is measured in laboratory tests and is characterized by sound power levels which are closely associated with flow velocities. These levels are related to a non-rotating inflow to the silencer. If this condition is not valid *in situ*, for example because of the upstream duct design, higher levels of regenerated sound will occur.

The level of the sound power emitted from the silencer cannot be less than the sound power level of the regenerated sound. Attenuations *in situ* are therefore often less than laboratory values which are determined without giving consideration to the regenerated sound. An estimate for the octave-band sound power level of regenerated sound can be obtained from equation (13):

$$L_{W,\text{oct}} = B + \left\{ 10 \lg \frac{pcS}{W_0} + 60 \lg Ma + 10 \lg \left[1 + \left(\frac{c}{2fH} \right)^2 \right] - 10 \lg \left[1 + \left(\frac{f\delta}{v} \right)^2 \right] \right\} \text{ dB} \qquad (13)$$

where

B is a value, in decibels, dB, depending on the type of silencer and the frequency;

v is the flow velocity, in metres per second, m/s, in the narrowest cross-section of the silencer;

c is the speed of sound in the medium, in metres per second; m/s;

Ma is the Mach number ($Ma = v/c$);

p is the static pressure in the duct, in pascals, Pa;

S is the area, in square metres, m^2, of the narrowest cross-section;

f is the octave-band centre frequency, in hertz, Hz;

H is the maximum transverse dimension of the duct, in metres, m;

δ is a length scale, in metres, m, characterizing the high frequency spectral content of the regenerated noise;

$W_0 = 1$ W.

The sound power level of regenerated sound will vary with the temperature T approximately with $-25 \lg(T/T_0)$ dB. For smooth-walled dissipative splitter silencers used in heating, ventilation and air-conditioning equipment, an approximation is given by $B = 58$ dB and $\delta = 0,02$ m. For this case, a graph of equation (13) is shown in Figure 10, and the A-weighted sound power level of a duct cross-section of 1 m^2 is then calculated from:

$$L_{WA} = \left(-23 + 67 \lg \left[\frac{v}{v_0} \right] \right) \text{ dB} \qquad (14)$$

where $v_0 = 1$ m/s.

NOTE   For other types of silencers, particularly for resonator silencers, B may be larger in certain frequency bands. However, no general information can be given on the values of B and δ.

6.1.3 Dissipative silencers with bends

Bends occur at intake (e.g. absorbent disk valves, see Figure 11) or blow-off openings and in the course of long duct systems (corners). At low frequencies where the cross-sectional diameter is small compared to the wavelength of sound, bends in the duct (as in flexible tube silencers) do not affect the transmission of sound. At high frequencies, when the wavelength of sound is less than the duct width, the sound will hit the front wall almost like a beam and can therefore be strongly attenuated by a sound-absorbent lining.

NOTE   Basically, the attenuation of a bend may be defined as the additional attenuation observed in a bent duct as compared to a straight one. For the time being, however, there are no specifications for measurements of this attenuation. In practice, the effect of the lining may be determined by comparing hard-walled and absorber-lined structures.
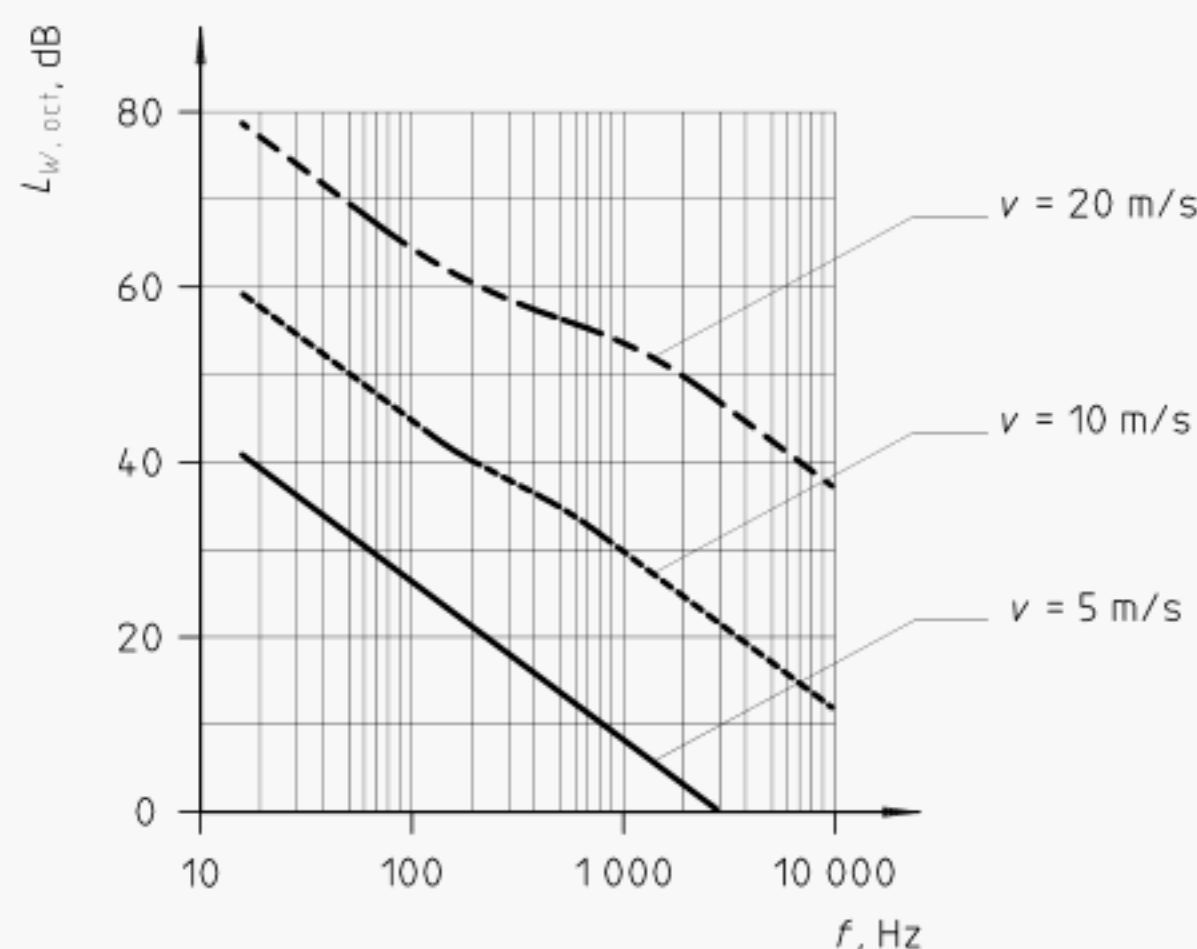
Figure 10 — Octave-band sound power level $L_{W,oct}$ of regenerated sound vs. frequency f for air under ambient conditions in a ducted silencer with narrowest cross-section $S = 0,5$ m², maximum transverse duct dimension $H = 1$ m, and different flow velocities v
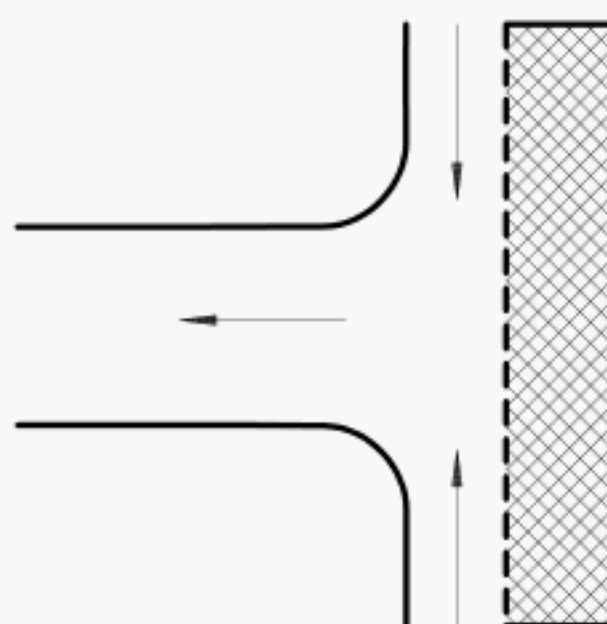

Figure 11 — Silenced intake opening with absorbing baffle (schematic)
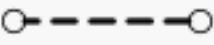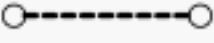
The effect of disk valves may be described using equation (7). The ratio U/S does not change along the radial sound propagation path. The effective length is determined by the disk radius. As dimensions are usually small, suppression of beam formation is of minor importance. The duct end is formed like a curved funnel to reduce the relatively high pressure losses.

In the case of corners, it is important to distinguish between aerodynamically shaped ducts and ordinary ventilation ducts. In aerodynamically shaped ducts, sound-absorbent guide vanes are used, which require only little space but can have considerable effects at high frequencies. In ordinary ventilation ducts, wall linings near the corner are suitable. The more protrusions the walls have that are comparable with the wavelength of sound, the more the sound will be scattered and the greater will be the attenuation. Figure 12 shows an example of the transmission losses measured in a bend with different surfaces lined or left bare.

Dimensions in metres
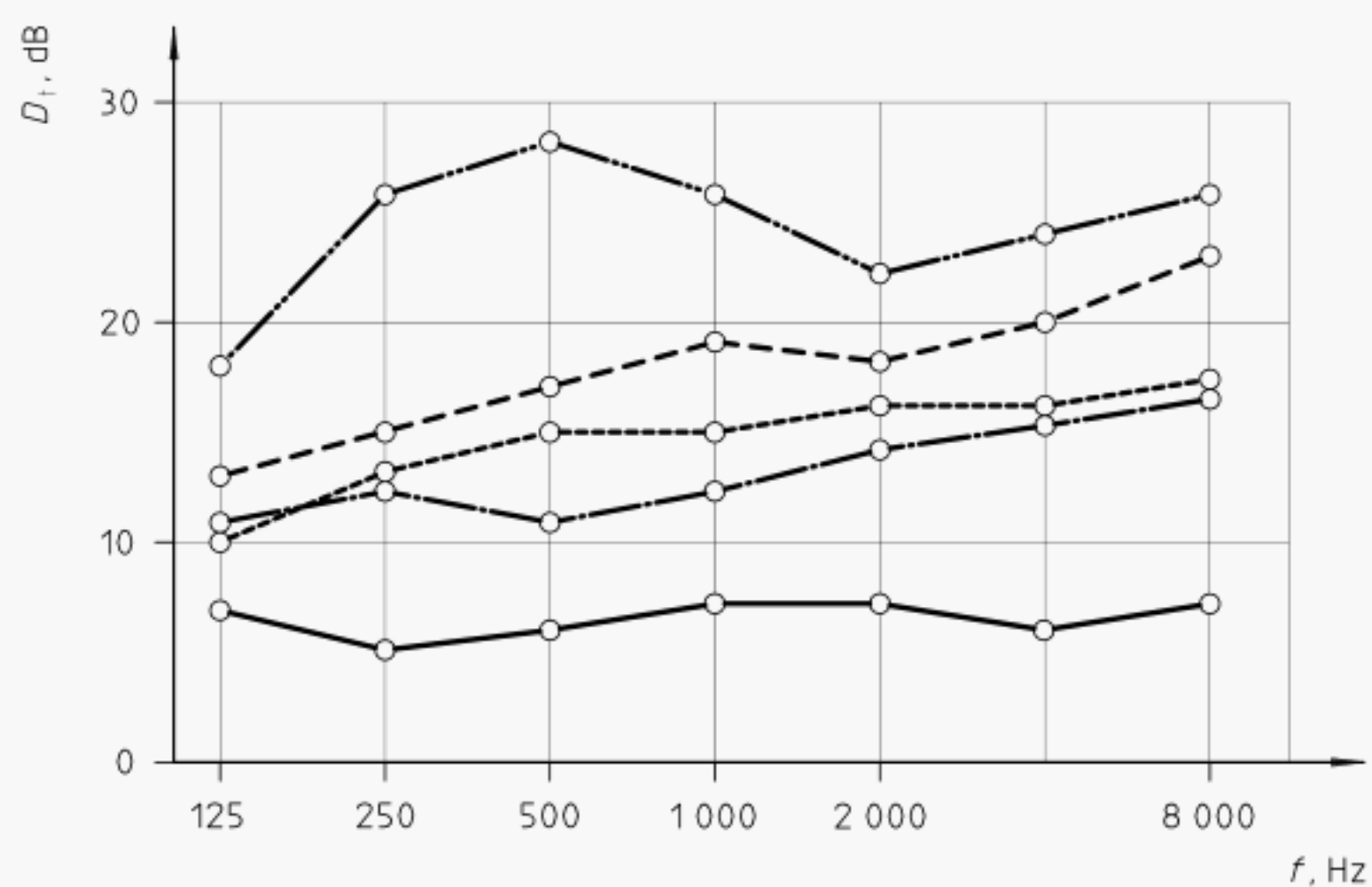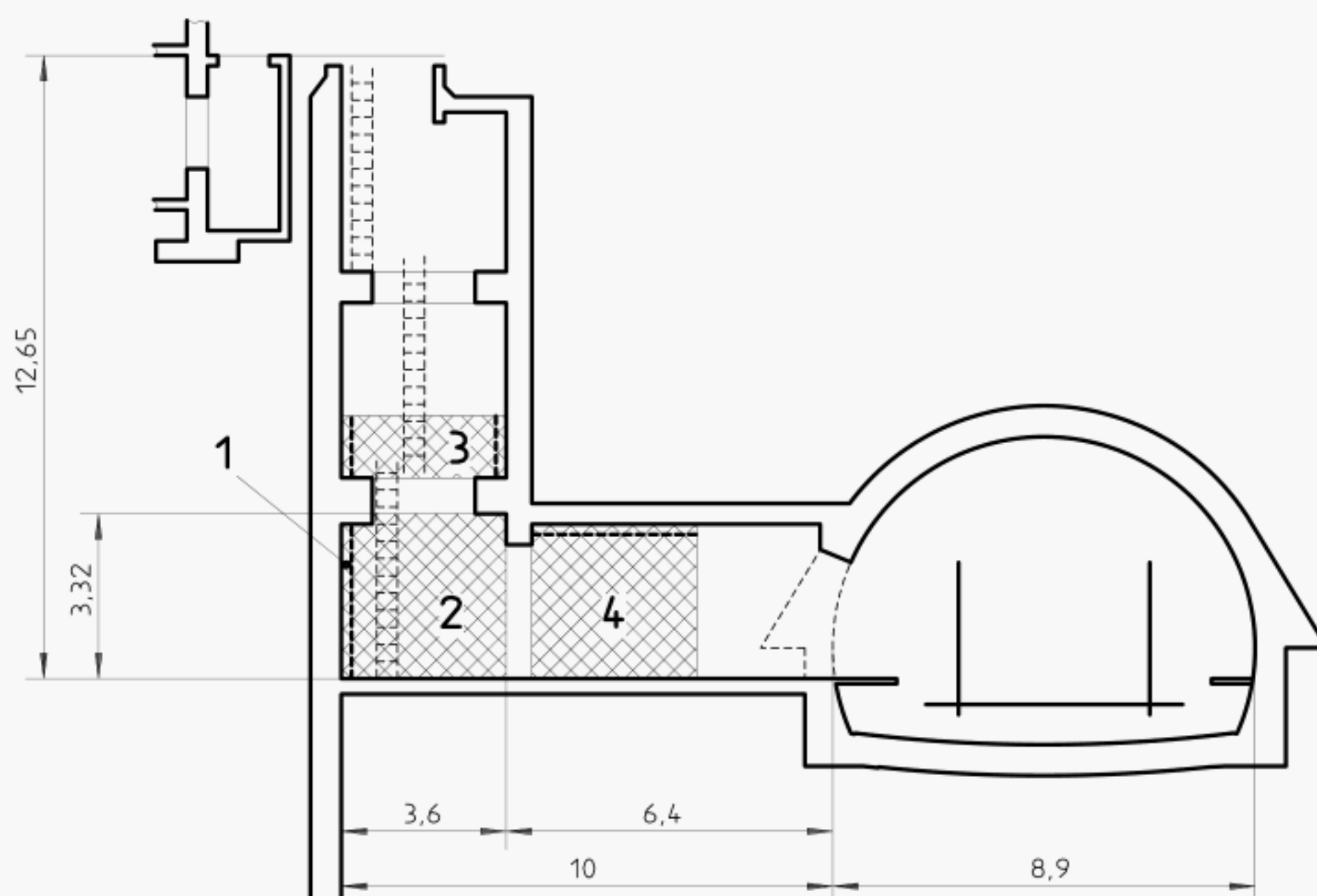
Key:

Sound-absorbent wall lining with 50 mm mineral fibre, density 70 kg/m³

Linings 1 + 2 + 3 + 4
Linings 1 + 2 + 3
Linings 1 + 2
Lining 1
No lining

Figure 12 — Transmission loss D_t vs. frequency f of a bend with various arrangements of sound-absorbent wall linings (ventilation shaft of a subway tunnel)

6.2 Reactive silencers

6.2.1 Resonator silencers

6.2.1.1 General considerations

For the lining of ducts as well as for the construction of splitters, attenuating elements in the form of absorbers or resonators are used. The combination of both types may be useful for particular applications.

Different types of resonators, shown in Figure 13, are as follows:

a) a sound-absorbent layer of low flow resistance with rigid backing and lateral partitioning, performing as a quarter-wavelength resonator;

b) a similar device equipped with a perforated or slotted plate of low porosity, providing "bottle necks" in the path of airborne sound (Helmholtz resonators);

c) a similarly partitioned lining with sound-absorbent layers on the partition walls or without any absorbent material, also performing as a quarter-wavelength resonator; and

d) a similar device covered with a light foil or plate.
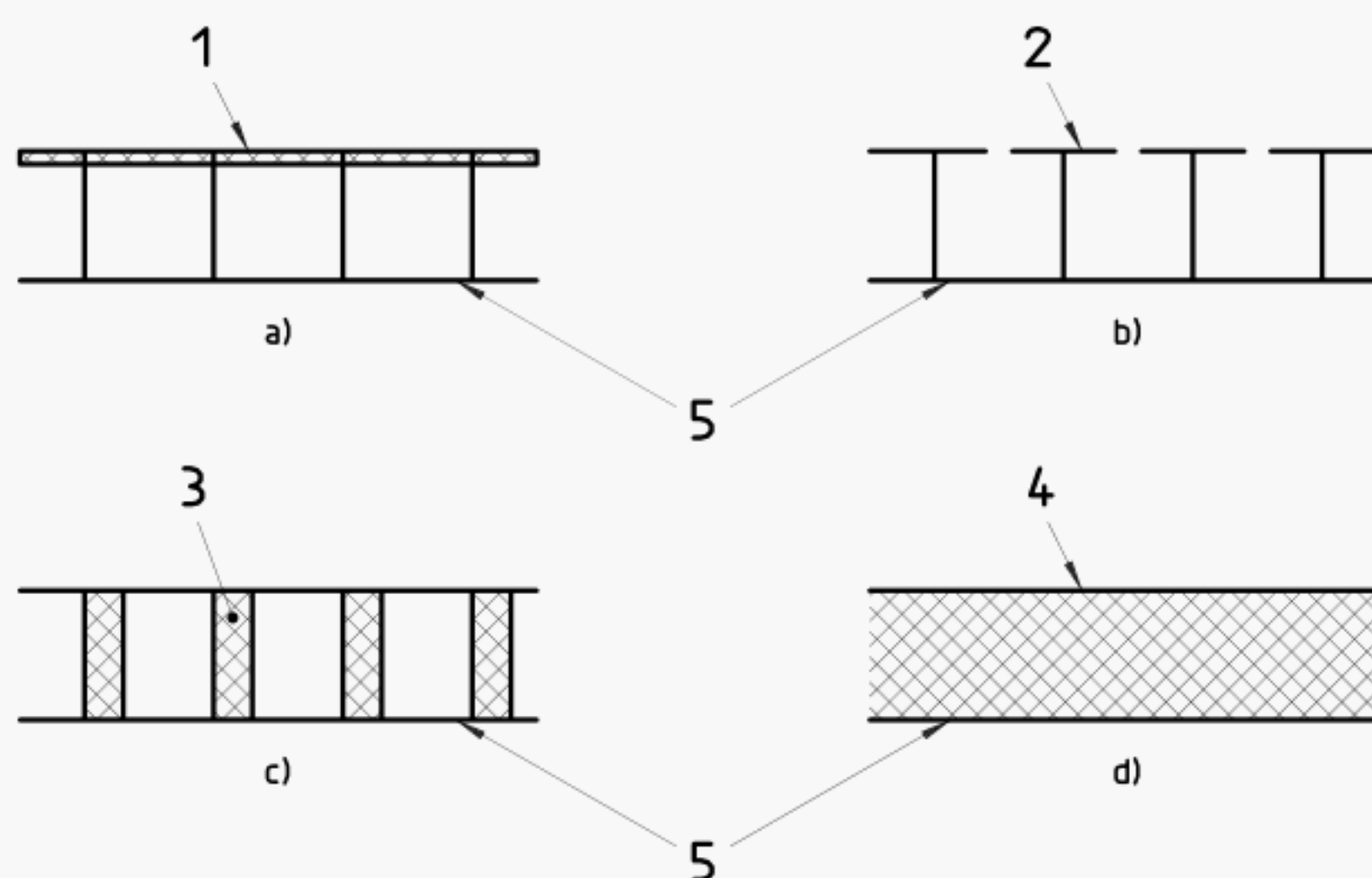
Key:

1 Resistive layer
2 Perforated or slotted plate
3 Sound-absorbent layer
4 Foil or plate
5 Rigid backing or plane of symmetry

Figure 13 — Types of resonator linings (schematic)

Combinations of Helmholtz and plate resonators, which do not require any absorbent material, are in practical use [25].

6.2.1.2 Quarter-wavelength resonators

The quarter-wavelength resonance frequency f_0, in hertz, is determined by equation (15):

$$f_0 = \frac{c}{4t}$$

(15)

where

    c is the speed of sound, in metres per second, m/s;

    t is the effective thickness of the lining, in metres, m.

An example for multiple quarter-wavelength resonators is shown in Figure 18. The width of the lateral partitioning, which may be oriented perpendicularly or inclined towards the cover, shall be less than t (preferably less than $t/2$) in the direction of sound propagation. The sound-absorbent material, if used, shall be protected from contamination or abrasion caused by the flow. Quarter-wavelength resonators are also effective at odd multiples of the natural frequency f_0 if the width of the chamber is sufficiently small.

6.2.1.3 Helmholtz resonators

The natural frequency of a Helmholtz resonator is given by equation (16):

$$f_0 = \frac{c}{2\pi} \sqrt{\frac{\varepsilon}{t(l + \Delta l)}} \qquad\qquad (16)$$

where

    ε is the fraction open of the area of the covering plate;

    l is the plate thickness, in metres, m;

    Δl is the end correction for the openings, in metres, m;

c and t are defined as in equation (15).
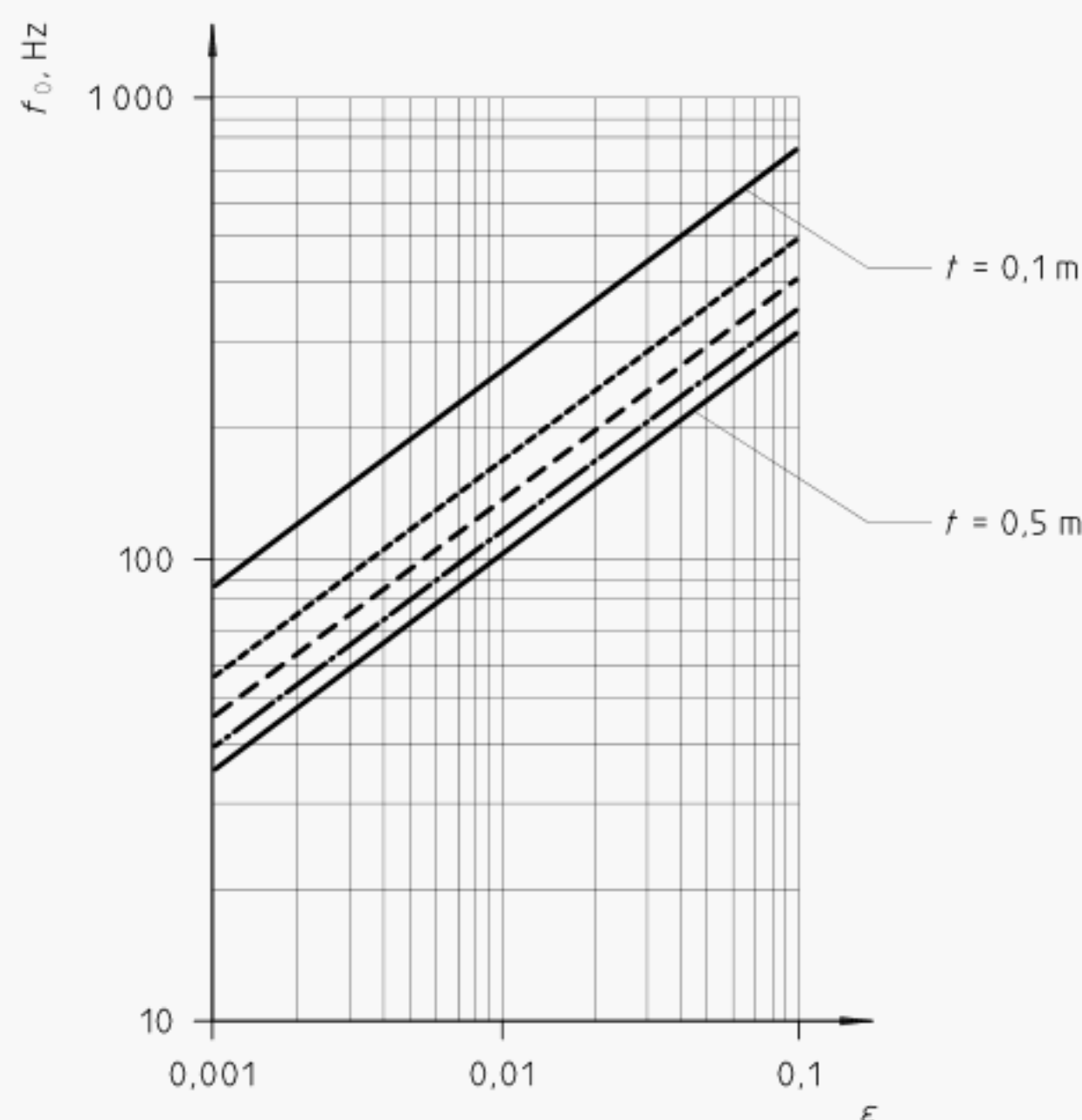
Figure 14 — Resonance frequency f_0 of a Helmholtz resonator vs. fraction open of the area ε of a perforated plate (1 mm thick, 5 mm diameter holes) in front of a partitioned lining of depth t ($c = 340$ m/s)
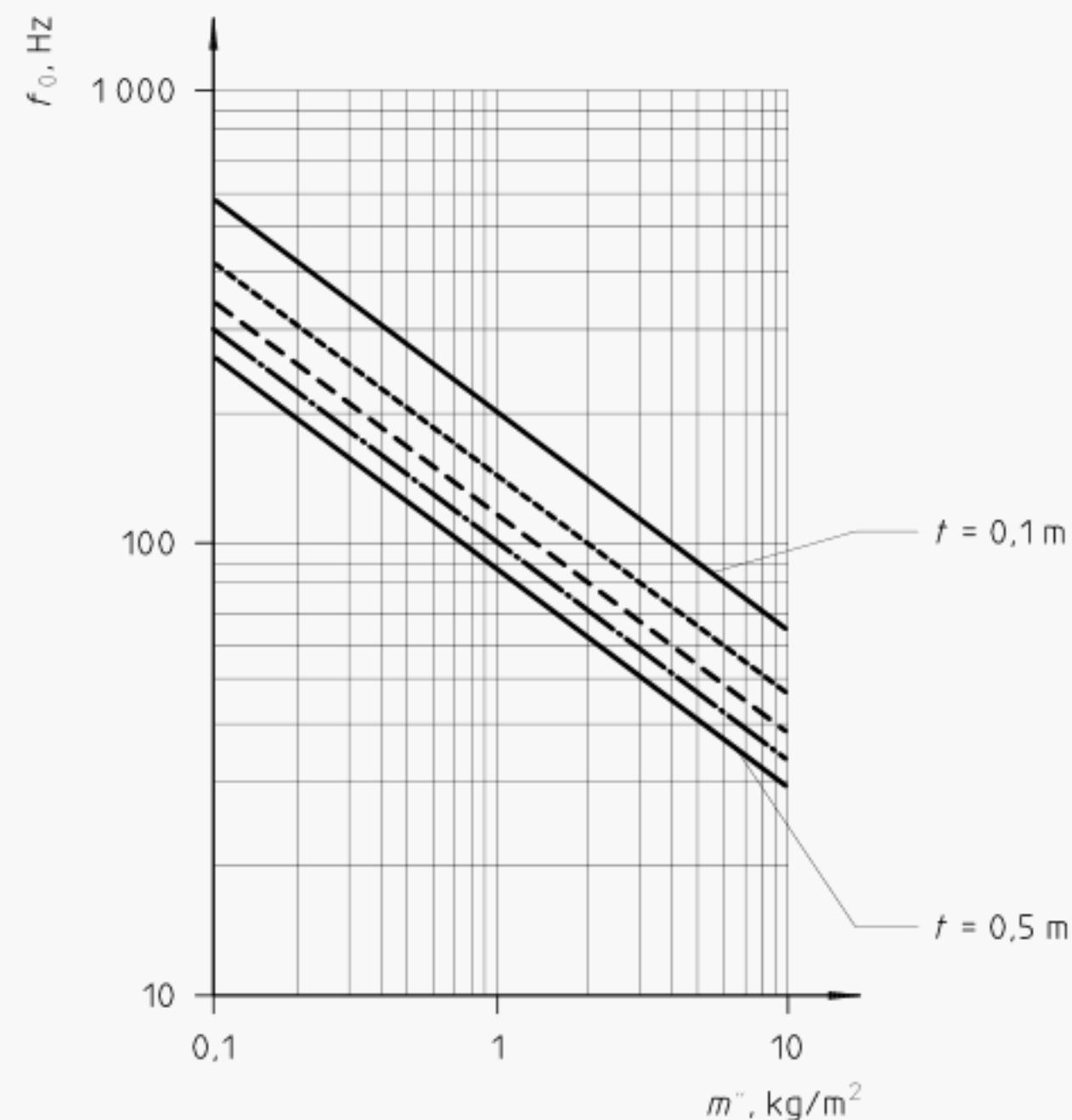
Figure 15 — Resonance frequency f_0 of a plate/foil resonator vs. mass per unit area m'' of a limp plate in front of a partitioned lining of depth t (c = 340 m/s, p = 1,2 kg/m³)

A graph of equation (16) is shown in Figure 14. The end correction depends on the diameter of the openings and their relative positions and on the velocity of grazing flow, should this exceed 15 m/s.

NOTE   For identical depth of the chamber, the Helmholtz resonator is always tuned to a lower frequency and acts in a narrower frequency band than the quarter-wavelength resonator. Damping the chamber will not result in significant changes in bandwidth. By comparison, porous material applied to the cover acts as an effective dampener but is highly sensitive to contamination.

6.2.1.4 Plate/foil resonators

To compute the resonance frequency of a plate/foil resonator, replace $\varepsilon/(l+\Delta l)$ in equation (16) by ρ/m'':

$$f_0 = \frac{a}{\sqrt{m''t}} \tag{17}$$

where

$$a = \frac{c}{2\pi}\sqrt{\rho}$$

ρ is the density of the gas, in kilograms per cubic metre, kg/m³;

m'' is the mass per unit area of the plate or foil, in kilograms per square metre kg/m²;

c and t are defined in equation (15).

For air under normal conditions, $a = 60\sqrt{kg/m}$ Hz . A graph of equation (17) is shown in Figure 15. Suitable choices shall be made in material and design to avoid the resonator being detuned through deposits, and to prevent the co-vibrating cover from becoming sensitive to mechanical damage. Special plastic or metal foils are used. With thin foil there is the danger of flutter noise caused by the flow.

At higher frequencies, characteristic vibrations of the cover are exploited to generate acoustically soft walls in further frequency bands. Broad-band attenuation is more reliably achieved by using differently tuned resonators on the path along the duct. Distances between the resonator groups shall then be at least one-quarter of the (largest) resonant wavelength in order to avoid any interaction between them, which is usually detrimental. This rule also applies for the different sides of ducts. Since the resonators are most effective in the frequency range where the airway between linings is less than half a wavelength, differently tuned resonators shall not be specified for opposite walls.

For all types of resonators, the natural frequency depends on temperature according to the speed of sound c:

$$c = c_0 \sqrt{T / T_0} \qquad\qquad (18)$$

where

T is the absolute temperature, in kelvins, K;

T_0 is the ambient temperature, in kelvins, K;

c_0 is the speed of sound, in metres per second, m/s, at ambient temperature.

To tune a resonator to a specified natural frequency at an elevated temperature T, the dimensions shall be enlarged by the factor $\sqrt{T / T_0}$ as compared to the dimensions at ambient temperature. Damping of resonators increases with increasing temperature.

6.2.2 Reflective silencers

6.2.2.1 General considerations

Reflective silencers are mostly designed to attenuate the fundamental mode in ducts below the cut-on frequency of higher-order modes, i.e. for relatively narrow ducts. In wider ducts, the propagation of higher-order modes may be prevented by using rigid axial duct intersections (so-called mode filters). A part of the incident sound will be reflected. This effect is not at present put to practical use.

Reflective silencers may comprise

— a simple expansion or contraction,

— a housing containing several interconnected expansion chambers,

— duct branching, and

— reactive-type splitters.

Distinction shall be made between reflective silencers for stationary plant and for automobiles and other mobile equipment.

When selecting reflective silencers for stationary plant, primary consideration is given to achieving a simple construction of sufficient mechanical stability. In silencers for pressure lines, the housing is designed as a pressure vessel. Radiation of sound from the housing is suppressed by providing a sufficiently heavy or stiff circular cross-section.

Reflective silencers for automobiles are designed to meet restrictions concerning mass and transverse dimensions. As a consequence, lightweight housings are built with an oval or generally non-circular cross-section. Radiation of sound from these housings shall be suppressed by special measures, such as double-shell constructions with damping layers in between, construction of chamber walls as stiffening bulkheads, and the use of special ribs to ensure stability.

All transverse dimensions and often the axial dimensions of fittings are small in comparison to the wavelength of the low-frequency sound to be attenuated. When tuning the elements, raised temperatures in the exhaust flow of internal combustion engines and compressor lines should be considered. Non-linearities (shock waves) and regenerated sound are decisive factors in high-frequency attenuation.

Dimensions in millimetres
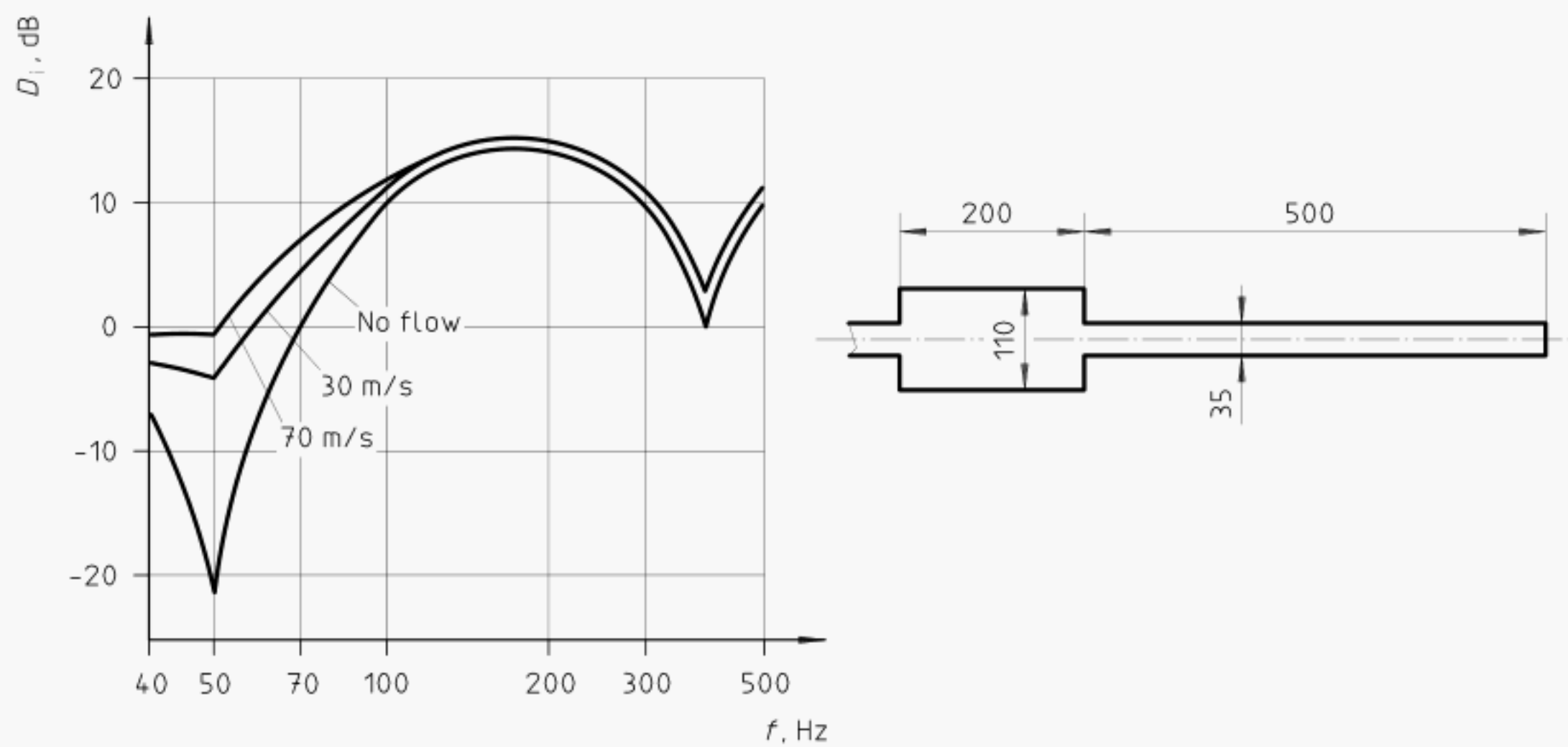
Figure 16 — Calculated insertion loss D_i vs. frequency f of a single-chamber reflective silencer for varying flow velocities in the direction of sound propagation

In the absence of flow and high sound levels, good agreement can be obtained between transmission line calculations and laboratory measurements with loudspeakers [6]. In practice, however, the effects of flow are very important. Flow causes damping in perforated pipes and expansion chambers (see Figure 16). Resonators become de-tuned and their damping enhanced or reduced, depending on the direction of the flow.

6.2.2.2 Expansions and expansion chambers

From an open end, sound is reflected back towards the source if the diameter of the outlet cross-section is small compared to the wavelength $\lambda = c/f$. Reflection will be stronger, and hence the sound radiated outwards will be less, the greater the solid angle Ω of radiation. To enhance reflection loss D_m at the outlet, the area S shall be as small as possible and the outlet is better positioned far away from a wall ($\Omega = 4\pi$) than in a wall ($\Omega = 2\pi$), in an edge ($\Omega = \pi$), or in a corner ($\Omega = \pi/2$); see equation (19):

$$D_m = 10 \lg \left[1 + \left(\frac{c}{4\pi f} \right)^2 \frac{\Omega}{S} \right] \text{ dB} \qquad\qquad (19)$$

where

c is the speed of sound, in metres per second, m/s;

f is the frequency, in hertz, Hz;

Ω is the solid angle of radiation, in ster radians, sr;

S is the outlet area, in square metres, m^2.

Flow leaving ducts that end in the open or in a large enclosed space will regenerate sound if a significant pressure drop occurs at the duct outlet. To keep the regeneration of sound small in critical cases, the outlet area shall be as large as possible and free of obstacles.

NOTE   When the linear dimension of an expansion chamber in any direction is small compared to the wavelength of sound, its volume relative to the open area causes a spring-like reaction of the enclosed gas. The larger the volume the softer the spring is. The element has a high-pass frequency characteristic.

When the linear dimension of an expansion chamber in any direction is large compared to the wavelength of sound, diffuse sound fields occur which provide for a de-coupling of different openings. Multiple reflections can be used to provide significant attenuation even when there is little absorption in the chamber.

6.2.2.3 Contractions

A pipe inserted in a partition between two chambers is acoustically effective with the mass of the gas enclosed (with end corrections), provided that the pipe is carrying no flow and that its length is short compared to the wavelength. For a hole in a thin wall or a perforated plate, the mass is essentially that of the end corrections. Such elements have a low-pass frequency characteristic and can be used to tune resonances of the silencer.

For a pipe carrying flow, resistive properties arise particularly from the pressure drop at the outlet. Flow grazing over a perforated plate also increases the acoustical resistivity of the element.

Special contractions are Venturi nozzles which are used as single elements or in perforated plates. In a suitable dimension such elements exhibit considerably less resistance for the average direct flow than for the superimposed oscillation peaks, thus acting as non-linear elements.

6.2.2.4 Multi-chamber housings

A reflective silencer may consist of a housing with several flanges to be connected to the source and inlet or outlet ducts, and of fittings mounted in the housing. These fittings form changes in the cross-section, branchings or dead ends (see Figure 17). The changes in cross-section are expansions or contractions. The acoustic performance is mainly determined by the ratio of the linear dimension l to the sound wavelength λ. This ratio is inversely proportional to the square root of the absolute temperature:

$$\frac{l}{\lambda} = \sqrt{\frac{T_0}{T}} \frac{l}{\lambda_0} \tag{20}$$

where

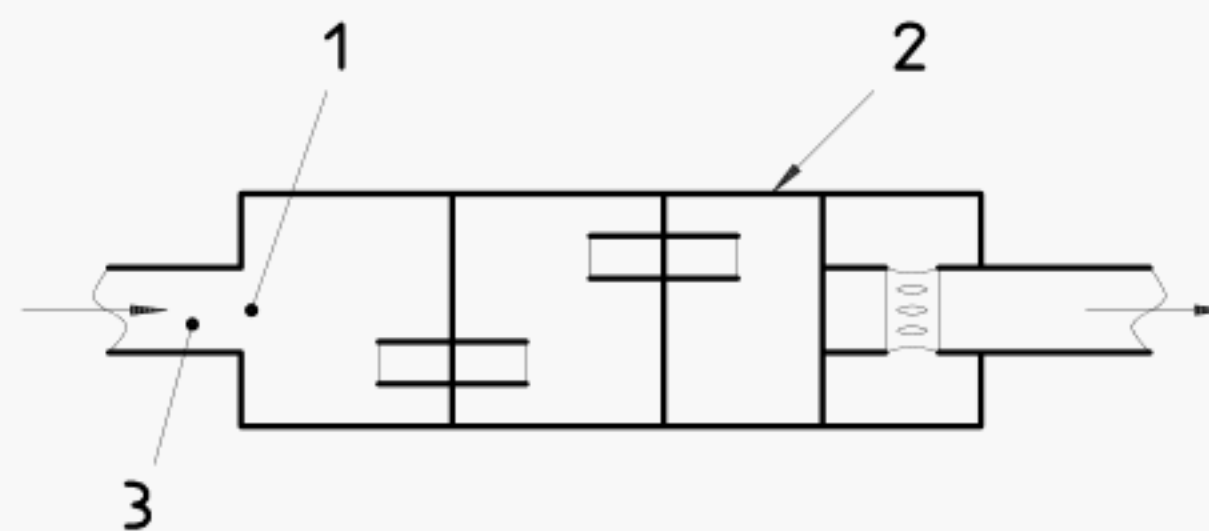T is the temperature, in kelvins, K, of the gas in the flow duct;

T_0 is the ambient temperature, in kelvins, K;

λ is the wavelength, in metres, m, of sound at temperature T;

λ_0 is the wavelength, in metres, m, of sound at ambient temperature T_0.

Key:

1   Sudden change in cross-section
2   Outer casing
3   Flow duct

Figure 17 — Multi-chamber reactive silencer (schematic)

6.2.2.5 Branchings

If a duct splits into branches where the lengths differ by Δl before recombination, a high degree of attenuation through interference will be achieved for odd multiples of the frequency $c/(2\Delta l)$, where c is the speed of sound. This interference gives rise to marked reflections in several narrow frequency bands at the branching point.

A special form of branches is the side-branch, where one branch length is small compared to a quarter wavelength. The performance of a side-branch is similar to that of a quarter-wavelength resonator.

6.2.3 Reactive splitters

Duct linings or splitters with resonators which are not significantly damped by sound-absorbent material will result in high insertion loss mainly for frequencies near the natural frequency of the resonator (see Figures 18 and 19).

Dimensions in millimetres
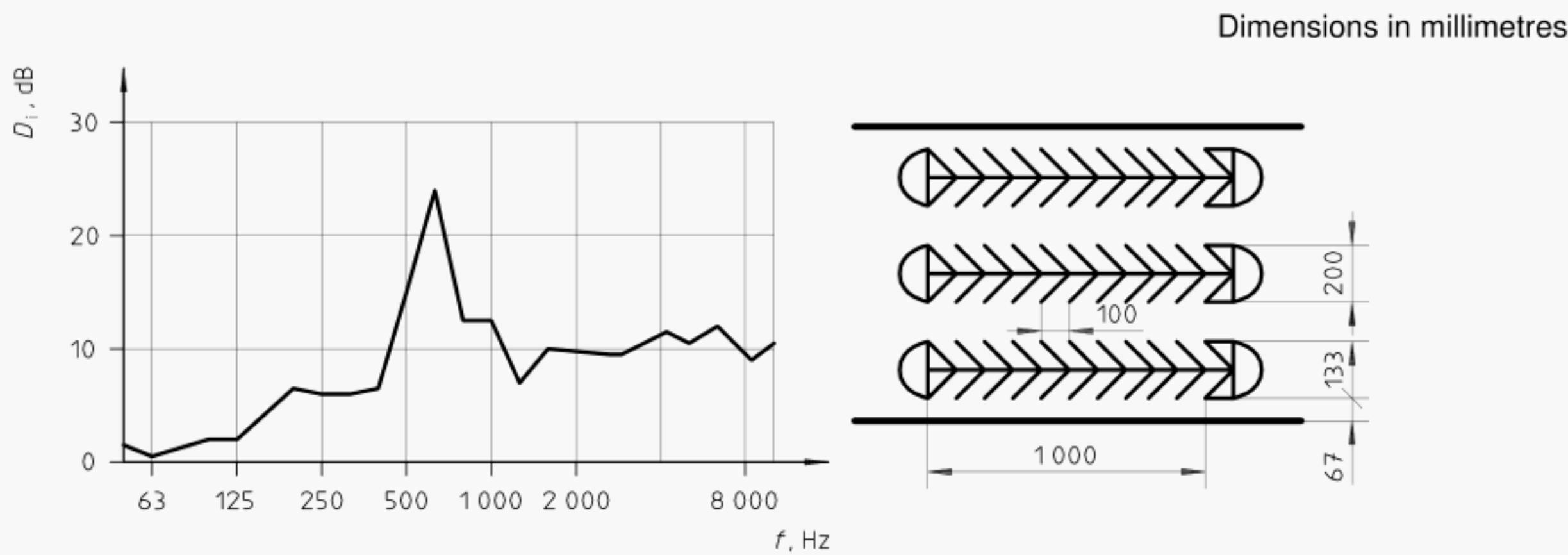
Figure 18 — Typical insertion loss D_i vs. frequency f of a splitter silencer with quarter-wavelength resonators

Dimensions in metres
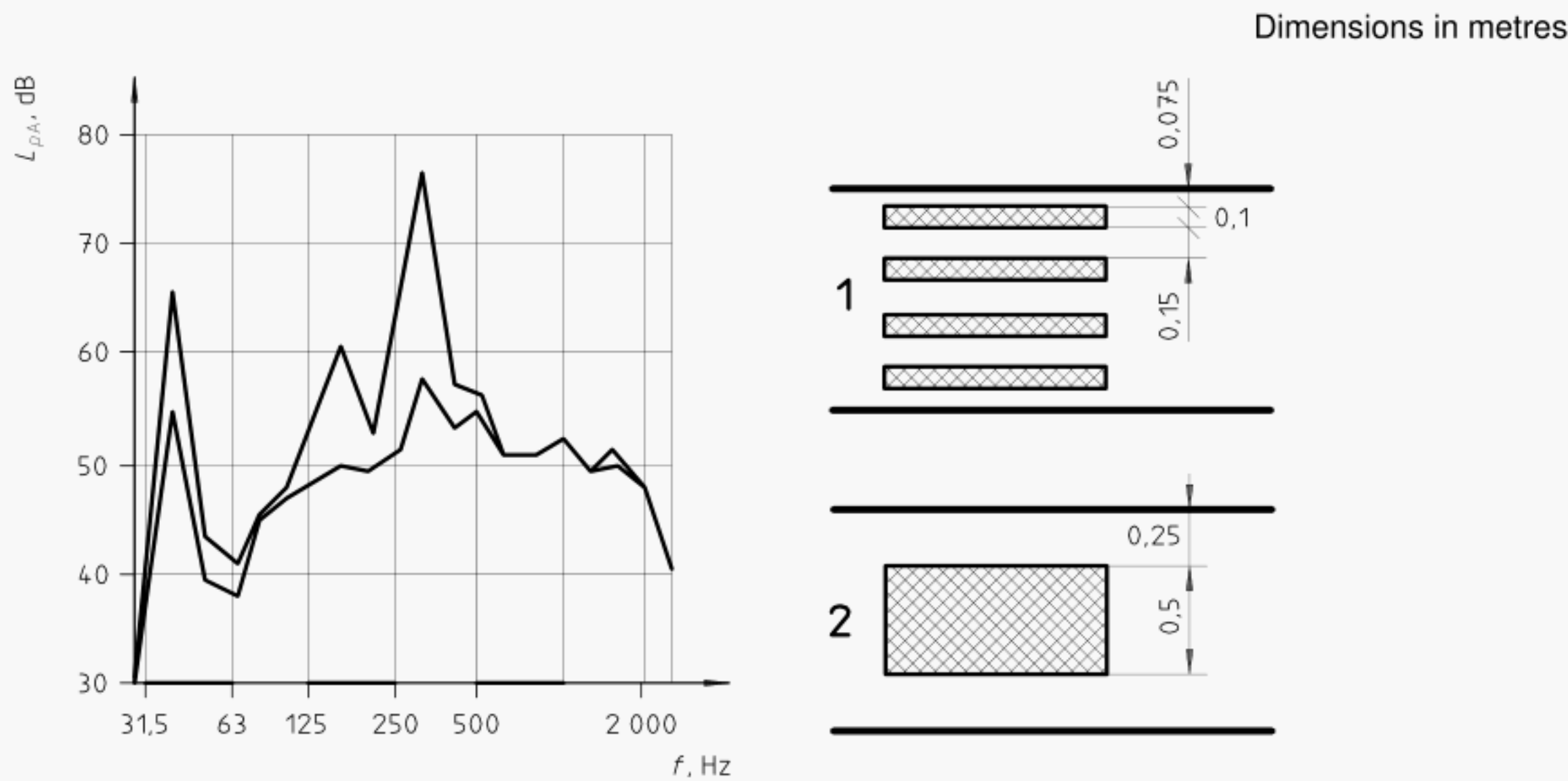
Key:

1   Splitters with resonators tuned to 160 Hz and 315 Hz
2   Splitters with resonators tuned to 40 Hz

NOTE   Top curve is without silencer; bottom curve is with silencers 1 and 2 installed in series.

Figure 19 — A-weighted sound pressure level L_{pA} vs. frequency f, 1 m to the side of the opening of exhaust funnel

NOTE   Flow with velocities exceeding 10 m/s may de-tune resonators towards higher frequencies by up to a one-third octave, and may increase or decrease damping, depending on the shape of the resonators. The maximum attenuation will decrease and regenerated sound will increase.

At high frequencies, depending on the roughness of the surface compared to the wavelength of sound, the attenuation will become independent of frequency; it will be negligible for plate resonators.
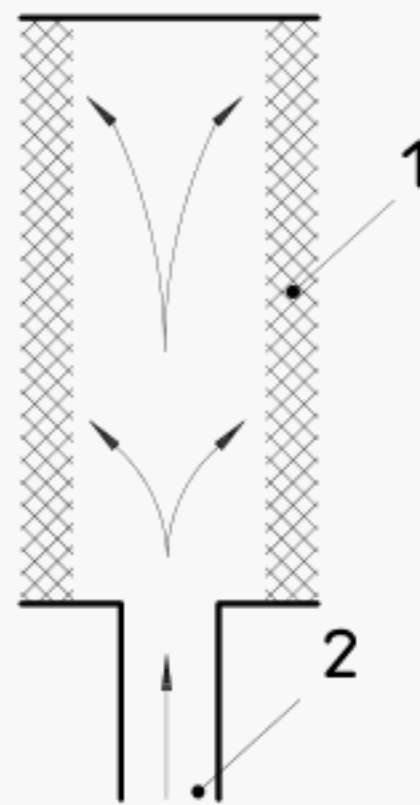
Key:

1   Flow-permeable material (e.g. sintered metal)
2   Highly pressurized medium

Figure 20 — Throttle silencer for pneumatic systems (schematic)

Laboratory measurements are only reliable if the influence of flow on tuning, damping (increase or decrease) and regenerated sound can be taken into account in a realistic way. The influence of temperature on absorption cannot usually be measured.

Calculations are often limited to the near vicinity of the natural frequency. It is difficult to account for the influences of flow and roughness of the surface. In Figure 18, the effective roughness is mainly determined by the ratio of the side-branch width to the wavelength.

Reactive silencers, such as plate/foil resonators and Helmholtz resonators or combinations of both, that dissipate sound solely through boundary layer effects (viscosity and heat conduction) or through structure-borne vibrations, are of particular interest because the gas cannot become contaminated by absorption material even if the silencer is damaged. This type of silencer is particularly suitable for areas with stringent hygienic requirements because of its closed surface.

6.3 Blow-off silencers

It should always be ensured that the silencer does not affect the safety of the machinery to which it is fitted.

Blow-off silencers to be fitted to the outlets of pneumatic valves are small elements. They consist of a cylindrical element with a surface which is large compared to the cross-section of the duct and is permeable to the flow (see Figure 20). A sufficient flow resistance of the surface shell, which consists of fibrous material or sintered metal, provides for an almost uniform distribution of the flow over the surface area. Such throttle silencers have a threaded end for connection to the duct. When contaminated, they are replaced unless cleaning by steam, chemicals or burning-out is possible.

Silencers for blow-off lines of large safety valves are designed for multi-stage relief. This is achieved by using perforated sheet metal which ensures that the permissible rise in pressure is observed. The structure shall be of high mechanical stability, so that it will not be compressed or destroyed when a blow-off occurs. Condensed liquid in the attenuator shall be prevented from freezing. Perforated plates are often combined with dissipative silencers to meet the strict requirements concerning sound level reduction.

7 Measurement techniques

7.1 Laboratory measurements

7.1.1 Overview

The following measurement methods are in use:

— laboratory measurements for ducted silencers as specified in ISO 7235;

— determination of the insertion loss of silencers in ducts without flow; laboratory survey method as specified in ISO 11691;

— additional laboratory measurements for development and detailed analysis of ducted silencers;

— determination of the insertion loss of a silencer on a machine by measurement of the sound power level of the machine with and without the silencer as specified in the ISO 3740 series.

The method chosen is determined by the application of the silencer and by the purpose for which the results shall be used.

7.1.2 Measurements in accordance with ISO 7235

High expenditure is needed, particularly for the determination of the insertion loss with flow. Therefore, for flow velocities below 20 m/s in the airway between dissipative splitters, the usually small effect of flow is often ignored.

As long as the sound field in the silencer consists of plane waves, different test facilities with adjoining reverberant rooms will find very small reproducibility standard deviations of insertion loss. A different excitation of higher-order modes, which is hardly avoidable at their cut-on frequency and at higher frequencies, may cause greater deviations.

Laboratory measurements will not usually allow the extrapolation of the silencer performance to higher temperatures, pressures or flow velocities.

7.1.3 Measurements in accordance with ISO 11691

Laboratory measurements of insertion loss without flow in smaller silencers for HVAC plant and similar applications shall be carried out in accordance with ISO 11691. The reproducibility standard deviation for measurements in different laboratories is kept small by specifying the dimensions of the sound source and of the test ducts in front of and behind the silencer as well as those of the substitution duct.

7.1.4 Further measurements on ducted silencers

Silencers with straight ducts or airways for the gas flow can be examined in detail by moving the microphone along the duct. Sound pressure distributions in frequency bands like the one shown in Figure 5 can be determined using this method.

If silencer models are built in accordance with the model laws for geometrical, operational and material similarities (see 6.1.1), laboratory measurements can serve to predict attenuation *in situ*. This method is used for large, complex geometries and for silencers intended for use under special operating conditions.

7.1.5 Measurements on silencers for small machines

In order to determine the insertion losses of silencers for use with small machines, laboratory measurements using either the free-field method (enveloping surface method of ISO 3744) or the reverberation room method in accordance with ISO 3741 can be used, depending on the significance and separability from other machine noise. The method is chosen according to the required accuracy. It shall be specified whether measurements are to be carried out with and without the silencer or whether the silencer is to be replaced by a hard-walled duct with inlet and outlet cross-sections identical to those of the silencer.

7.2 Measurements *in situ*

Measurements *in situ* differ from laboratory measurements because

— the measurement method has to be adapted to the conditions *in situ*;

— details of the measurements, such as the number and position of measurement points, cannot be specified independently from the conditions *in situ*;

— the conversion of measurement quantities, such as sound pressure levels, into values characterizing the silencer, such as transmission loss, cannot be carried out referring to specified testing conditions but using solely pragmatic values;

— the separation of regenerated noise and other limitations to the attenuation is hardly possible;

— flow measurements, if possible at all, can only be carried out to obtain survey precision.

These differences are allowed for in ISO 11820. Interested parties should, prior to commencing measurement, select the appropriate measurement method from a schematic list of conditions and agree upon the pragmatic correction values for the conversion of the measured quantities. Regenerated noise is regarded as a property of the silencer in the specific application and as such part of its attenuation performance. Flow measurements serve chiefly to detect non-uniform flow distributions which may cause silencer malfunction.

For special analysis *in situ*, measurements in accordance with ISO 11820 may be supplemented by measurements taken with a drag microphone or at various fixed positions in the free duct or airway. This can help to detect flanking sound transmission paths.

7.3 Measurements on vehicles

No specific standards exist for silencer measurements on moving vehicles.

8 Information on silencers

8.1 Information to be provided by the user

As a minimum, the following information, if applicable, shall be provided by the user/purchaser in order to specify the requirements for a silencer:

a) type of machine or plant (information concerning representative modes of operation), for example

    — for piston machines: power, engine speed, operating principle, number of cylinders, ignition sequence or number of stages, respectively,

    — for air-moving devices: power or volume flow and pressure difference, engine speed, operating principle, number of guiding and rotating blades per stage, number of stages, shape and type of blades, intake and outlet cross-sectional dimensions;

b) displaced medium

    — identification,

    — mass or volume flow,

    — temperature, pressure, humidity, gas constant or density,

    — type and quantity of contaminations,

    — materials to be used or to be avoided for the construction of the silencer;

c) spatial mounting conditions of the complete plant including the silencer and pipes (sketch with indication of dimensions);

d) required attenuation as

— A-weighted sound level reduction for a specified spectrum, or

— insertion loss in one-third-octave or octave bands between 50 Hz and 10 kHz, or

— insertion sound pressure level difference for a specified immission point in frequency bands between 50 Hz and 10 kHz;

e) permissible pressure loss;

f) additional requirements concerning, for example

— fire protection,

— emergency conditions,

— opportunities for maintenance, maintenance cycles and standstill time;

— further specific information (if necessary).

8.2 Information to be provided by the manufacturer

For the specification of the operational properties of a silencer, the supplier/manufacturer shall provide at least the following information, if applicable:

a) sound attenuation under specified operating conditions, in one-third-octave or octave bands, given as

— insertion loss, or

— transmission loss with correction terms in accordance with ISO 11820 and measurement positions, or

— insertion sound pressure level difference for a given immission point;

b) pressure loss under specified operating conditions, taking into account intake and exhaust flow conditions;

c) geometry of the silencer (drawing);

d) materials used, particularly information suited to indicate compliance with cleanroom specifications and potential health hazards by comparison with current limit or recommended values;

e) weight, mounting, inspection and maintenance conditions;

f) further specific information, if necessary.

Annex A
(informative)

Applications

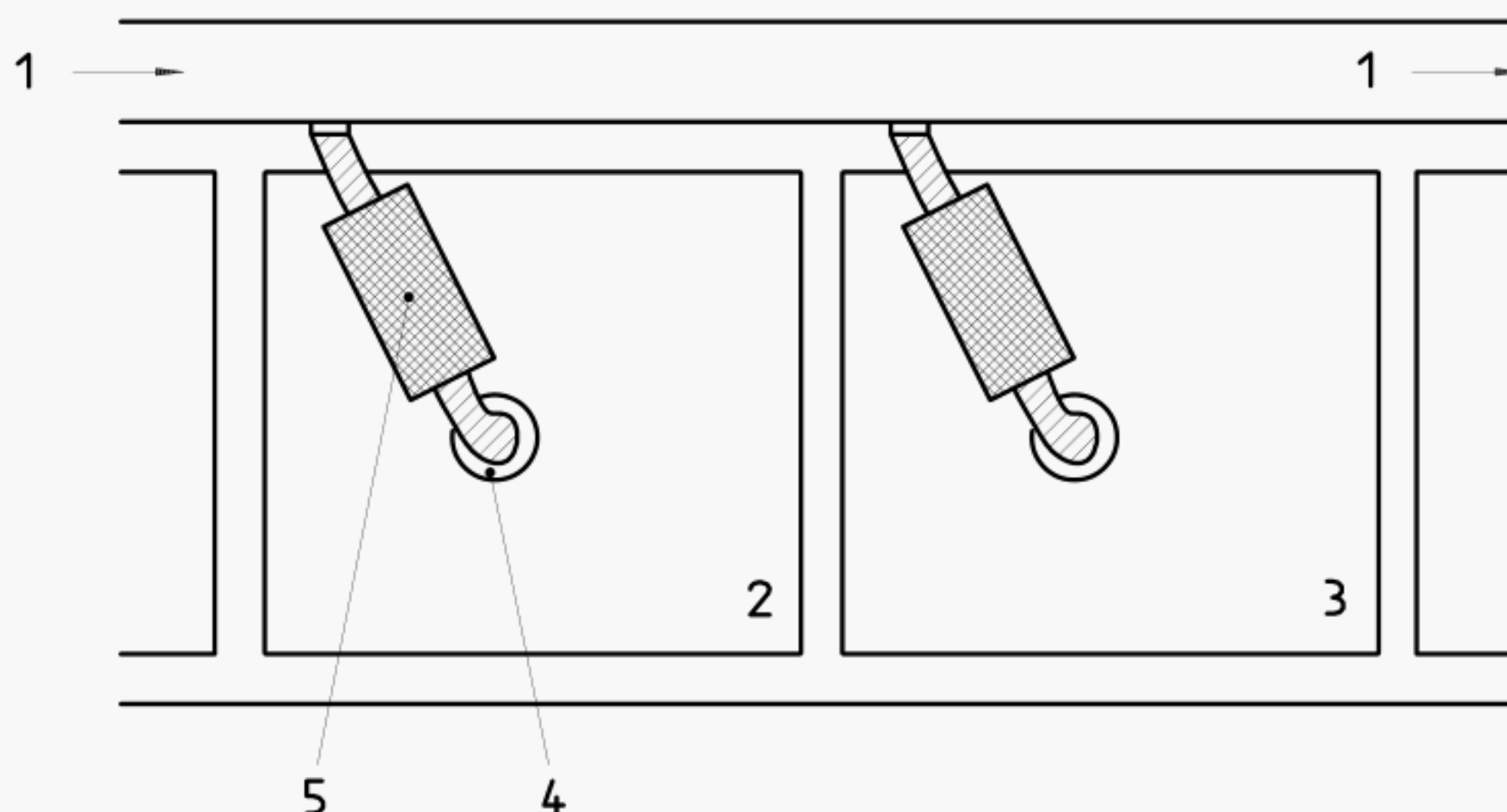## A.1 HVAC equipment

A.1.1 General considerations

Heating, ventilation and air conditioning (HVAC) technology is a major field of application for silencers. It is their task here to keep fan noise and regenerated sound caused by duct fittings out of rooms where low sound levels are required. Air leakages should be observed. Furthermore, so-called crosstalk silencers are used to achieve compliance with requirements for sound insulation between adjacent rooms (see A.1.4). Resonator silencers may be needed in addition to dissipative silencers if very stringent acoustical requirements are to be met by the HVAC equipment. From the acoustic as well as the economic point of view, a suitable arrangement consists of a resonator silencer near the fan (primary silencer) and a dissipative silencer near the outlet (secondary silencer).

A.1.2 Prevention of regenerated sound

Since the sound power of broad-band flow noise is roughly proportional to the sixth power of the flow velocity (see 6.1.2), it is most important for the prevention of regenerated sound to keep the maximum flow velocity in the duct cross-section and along the duct sufficiently small. Elements inserted in ducts which cause periodic vortex shedding provide pure tones. This can be prevented by particular shapes and special orientation with respect to the direction of flow. Guide vanes applied on splitters and in bends for the reduction of pressure loss will cause additional regenerated sound unless they are sound-absorbent constructions.

A.1.3 Flexible tube silencers

Connections between non-aligned ducts can be achieved by means of radially stiff but axially flexible tubes. As long as they are perfectly round in cross-section and not deformed by damage during installation or too narrow bends, the walls provide for a high transmission loss. Internal lining yields attenuation especially at high frequencies.

Key:

|			
1	Direction of airflow	4	Disk valve
2	Room 1	5	Absorptive silencer
3	Room 2		

Figure A.1 — Crosstalk silencer with disk valves in HVAC equipment (schematic)

A.1.4   Crosstalk attenuation

In ventilation technology, crosstalk is the sound transmission from one room to another through a ventilation duct which is open at both ends. If there are requirements concerning airborne sound insulation between the two rooms, this flanking transmission should be suppressed by placing crosstalk silencers in the duct between the two rooms. Figures A.1 and A.2 give an example showing the schematic and efficacy of a crosstalk silencer. Crosstalk attenuation includes the transmission loss of the ductwork, the insertion loss of the silencer and the end reflection.

A.1.5   Ventilation of industrial workshops

Workshops and enclosed plants with ventilation outlets in the façade will be fitted with silencers for ventilation outlets in the façade if noise protection in the neighbourhood is needed. If, for economic reasons, natural draught ventilation shall be used, outlets shall be large in area and fitted with dissipative silencers. For moderate noise control requirements, attenuating louvres are sufficient (Figure A.3). Where requirements are higher, thought should be given to weather-protection devices, as sound can be generated under certain weather conditions (through wind and rain).
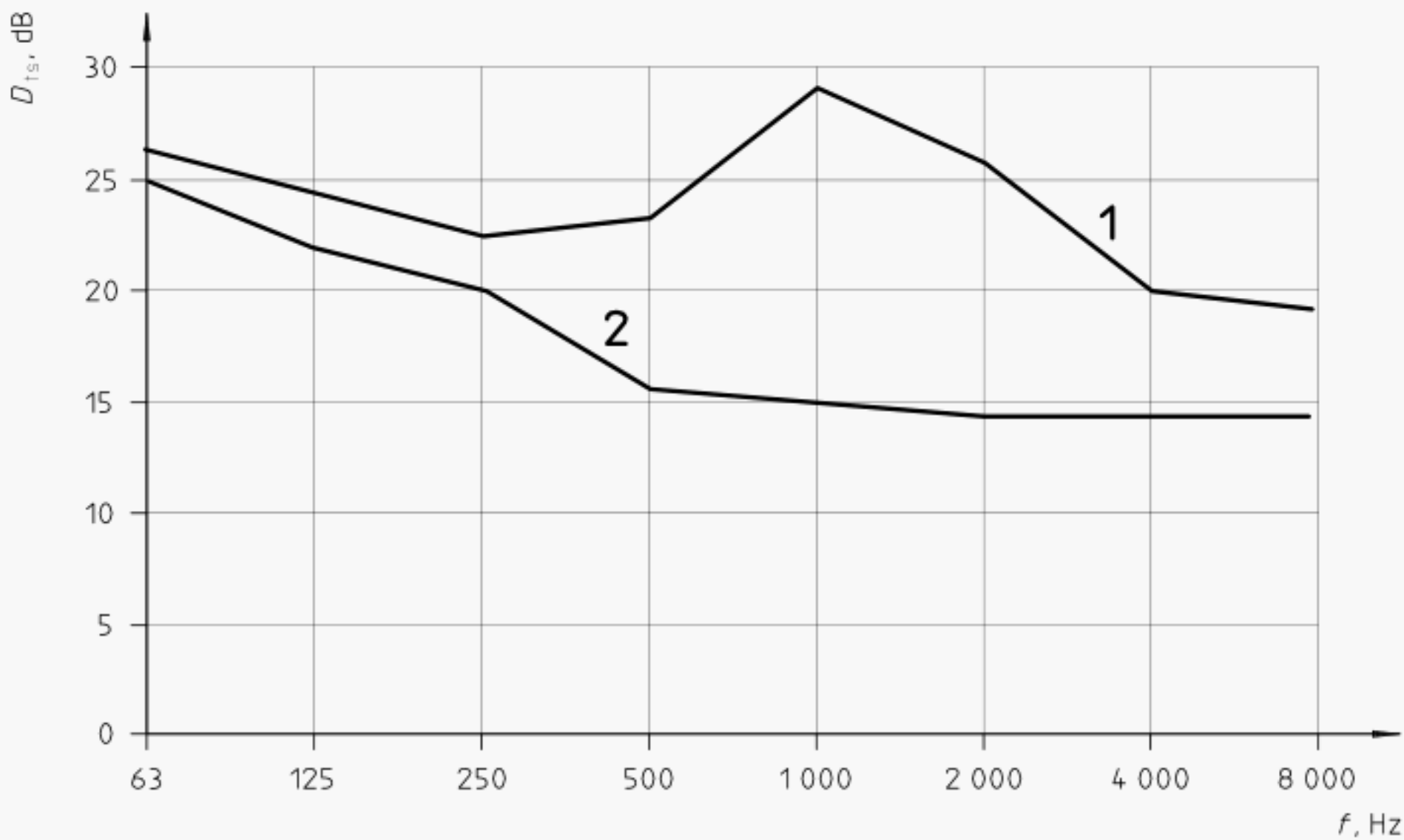
NOTE   Curve 1 with silencer, curve 2 without silencer; measured with disk valve half open.

Figure A.2 — Transmission loss D_{ts} vs. frequency f of a crosstalk silencer (flexible tube silencer, 500 mm long, 25 mm thick sound-absorbent lining) on a disk valve with nominal width of 150 mm
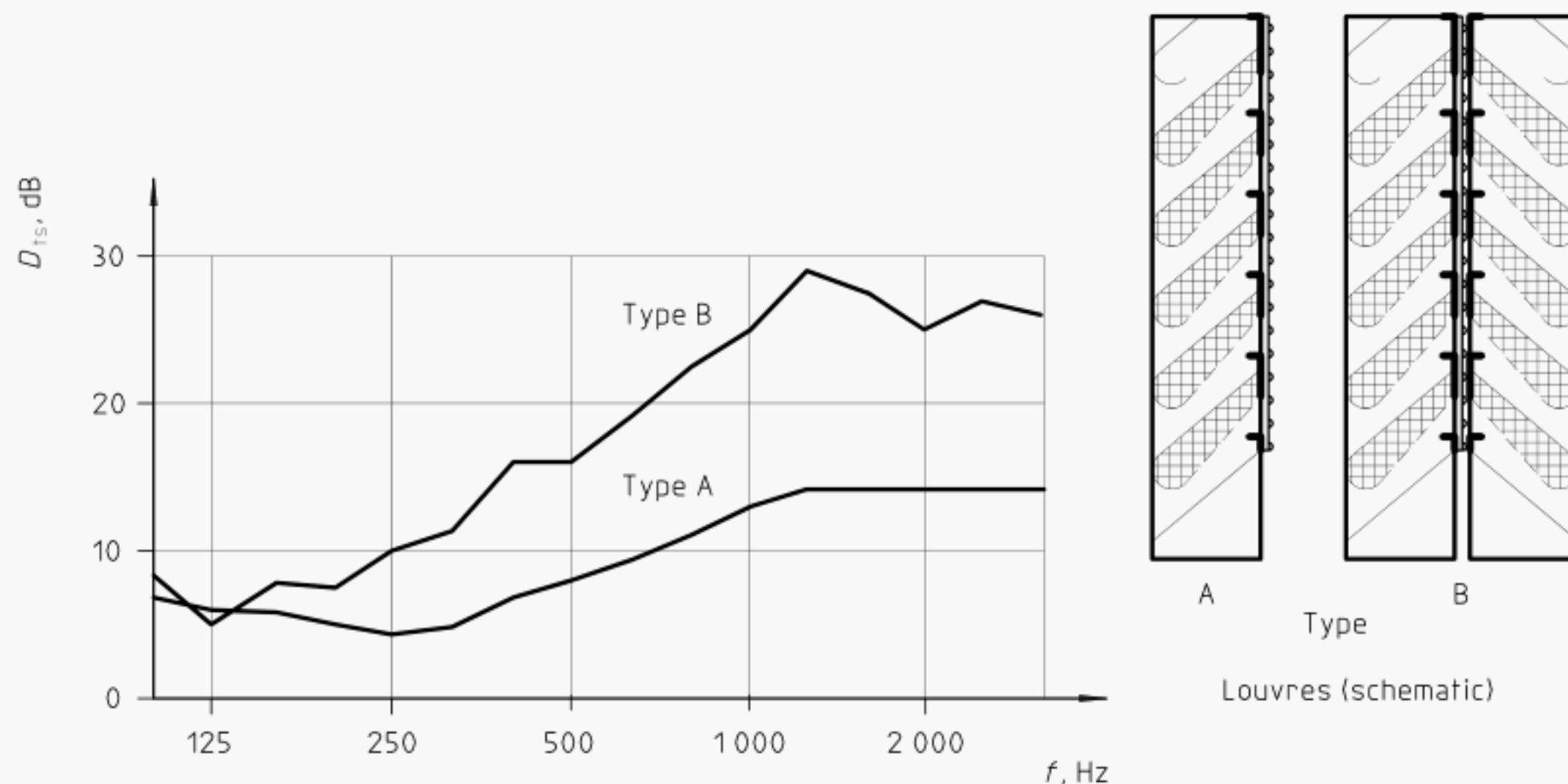
Figure A.3 — Transmission loss D_{ts} vs. frequency f for two types of louvres

A.2 Industrial plants

A.2.1 Fields of application

Noise control is applied, for example, in power plants, chemical processing plants, mining and mineral processing plants.

Silencers are generally needed

— on the suction and pressure sides of air moving devices,

— in the conveying system of mills and other processing equipment,

— on the suction and exhaust sides of furnaces and gas turbines,

— in the conveying system of pneumatic conveying and hoisting plant,

— behind control valves in pipes,

— for safety valves,

— in the ventilation system of enclosures and cabins.

A.2.2 Fans

Fans are considered among the most abundant noise sources in plant technology. Depending on noise control requirements, silencers should be fitted on the intake and/or outlet side. A peak emission in the low frequency range is a characteristic of most air-moving devices. In addition to a broad-band sound, emission many have tonal components as well.

The frequency characteristic of attenuation and the pressure loss of a silencer should be matched to the characteristics of the air-moving device. For superpositions of broad-band noise and tonal components, the combination of broad-band dissipative silencers with tunable resonator and reactive silencers is recommended. Sufficient space should be provided for the silencer if low frequencies are to be attenuated, as low-frequency attenuation requires thick linings. For tonal components, resonator silencers may present a solution needing little space.

The pressure loss in the silencer should be taken into account when determining the operating point of a fan. Significant pressure loss will require more fan power, which in turn will result in increased sound emission and higher operating costs.

If silencers are mounted directly in front of or behind a fan, the fact that structure-borne sound generated by the fan may excite the housing of the silencer should be considered. A strong excitation of structure-borne sound in the silencer housing can cause sound radiation into the duct. The performance of the silencer is then limited by this flanking path (see 5.3). It is recommended that flexible connections be built into the duct wall before the silencer. If the silencer is mounted to the same supporting structure as the fan, additional elastic mounts should be provided for the silencer housing, if necessary. For severe noise control requirements, resilient elements in the silencer housing are required to avoid flanking structure-borne sound transmission limiting the silencer's performance.
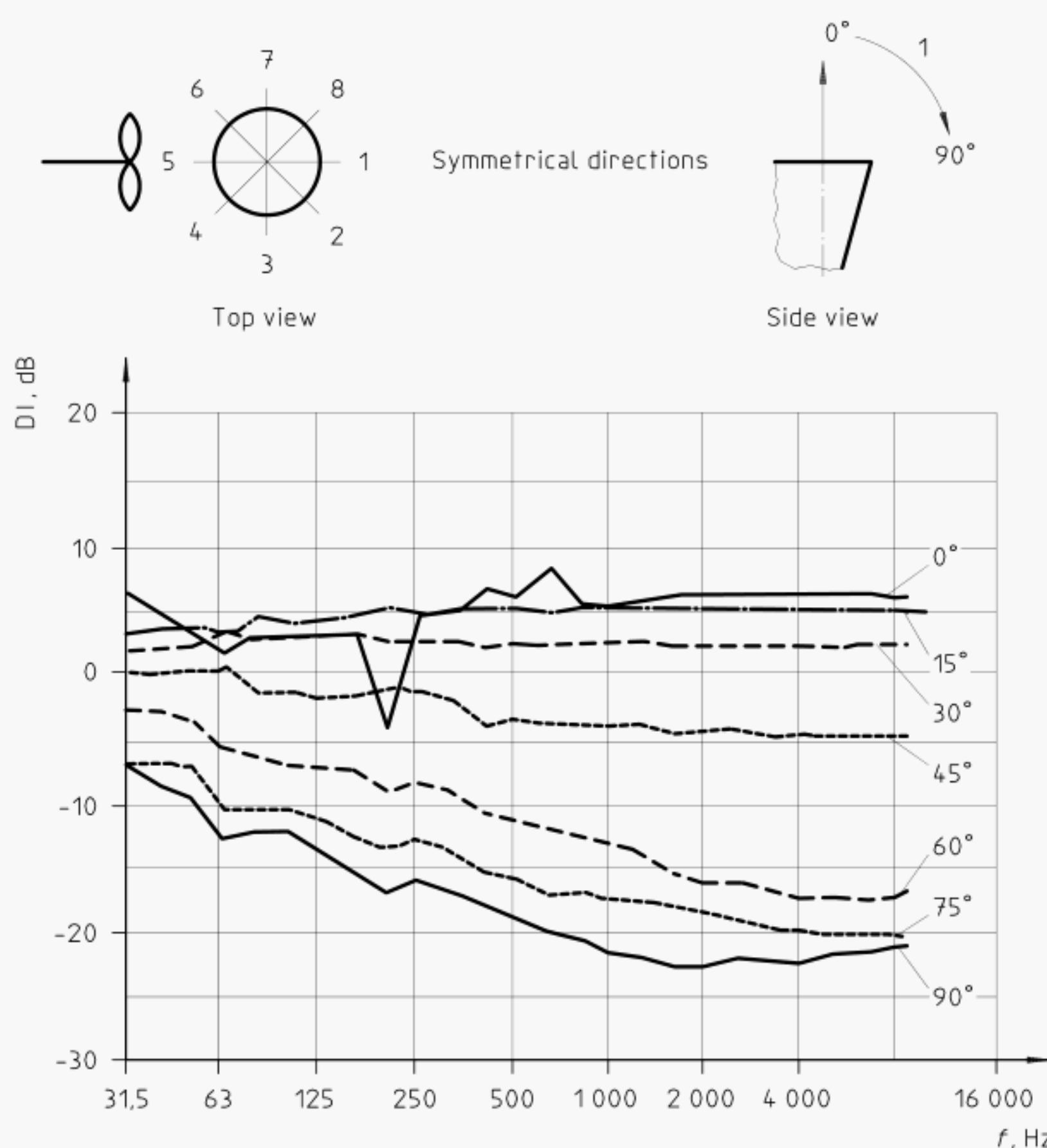
NOTE   Determined from measurements at a height of 20 m to 25 m above the ground; average values over eight horizontal directions; diameter of diffusor: 9 m; average flow velocity: 4 m/s; average wind speed: 3 m/s or less.

Figure A.4 — Directivity index (DI) vs. frequency f of sound radiation from round diffusor outlets in the vertical direction θ

A.2.3   Mining ventilation

Mining ventilation is generally based on large axial fans placed above ground and capable of coping with large volume flows. Splitter silencers are mounted in vertical or horizontal diffusors. They suffer from corrosive and abrasive wear and shall be highly resistant to dynamic stress. Splitters containing quarter-wavelength resonators and/or Helmholtz resonators made of stainless steel and concrete elements are used.

The directivity of sound radiation should be considered in the selection. As is generally the case for large outlet cross-sections, it is determined by diffraction of sound related to the geometry of the outlet. An example is given in Figure A.4.

A.2.4 Induced-draught fans

To assist natural (convective) draught in power plants, fans are used to carry flue gas out of the plant past filtering elements and through the chimneys. Despite filtering, the flue gas still carries ashes and other combustion residue which can render absorbent elements ineffective by dust deposits formed on the surface. Therefore, resonator elements as specified in 6.2.2.1 are used in silencers. As operating temperatures range between 100 °C and 200 °C, the effect of temperature should be taken into account as described by equations (18) and (20).

A.2.5 Cooling towers

Silencers in wet cooling towers are subject to corrosive stress because of a high degree of humidity. The cooling water droplets cause noise with a maximum emission between 1 kHz and 2 kHz. This is the dominating noise in natural-draught cooling towers. In induced-draught cooling towers, there is also the low-frequency emission from the fans.

In general, splitter silencers built of hydrophobic porous absorbers are used for noise control. An acoustically transparent protective covering of the absorber is absolutely necessary. To avoid corrosion, frames and covers of the splitters should be made of stainless steel, aluminium or plastic material.

In natural-draught cooling towers, the pressure loss in the silencer should not exceed 10 Pa, whereas for induced-draught cooling towers up to 70 Pa is permissible.

A.2.6 Compressors

Compressors are machines for the compression of gases. Silencers are used for noise control on the suction side (e.g. atmosphere) and the pressure side (e.g. pipework). The selection of the silencer depends on the type of compressor. A distinction is mainly made between

— turbo compressors, and

— piston compressors.

Silencers for turbo compressors are usually dissipative. They may be large in size as, for example, in intake systems for large stationary gas turbines generating electric power.

Turbo compressors generate tones, the frequency of which is given by the product of the number of turbine blades and the rotational frequency. In the selection of splitter silencers, it is important to ensure that the wavelength of tones at blade passage frequencies is less than twice the width of the airway. Silencers for turbo compressors require a particular mechanical stability because of the excitation of vibrations from the fluid and by structure-borne sound. Intake silencers for turbo compressors should be sufficiently robust so that components do not shake loose and damage the compressor.

Piston compressors generate a pulsating flow which causes noise and mechanical vibration. Plenum chambers and/or damped quarter-wavelength resonators are used. Plenum chambers are expansion chambers with a volume of preferably 12 times the piston-swept volume. Resonators tuned to the same frequency or groups of resonators tuned to different frequencies are damped to yield broad-band attenuation. The silencers are often designed as pressure vessels (adsorbers). A different kind of design employs one or more perforated plates mounted in the duct cross-section which may be shaped to act as Venturi nozzles.

On the suction side, too, the forces acting should be taken into account in the design. If aerosols or dust are transported, it is important to ensure that no harmful deposits can form in the sound-absorbent layer.

A.2.7 Enclosure, cabin and machine room ventilation

When machines are enclosed for noise control purposes, the heat generated inside the enclosure should be disposed of, thus requiring ventilation. Silencers should be provided for the ventilation system to maintain the effect of the enclosure. Their attenuation should match the required insulation of the enclosure. The same applies for the fresh air supply to cabins for personnel and ventilation of machine rooms.

A.2.8 Pneumatic control

Silencers are usually provided where air escapes from tools and valves. They should be small and shall not affect the functioning of the equipment even if they are oiled-up or contaminated in any other way. A great number of commercially available designs comply with this requirement.

A.2.9 Safety valves

The requirements for blow-off silencers for safety valves are usually determined by the considerable volume flow of the medium, large pressure loss in the silencer and sudden changes in pressure during starting-up. Special safety requirements apply to guarantee operation even after long standstill. It is important to ensure that parts of the silencer (such as compressed sound-absorbent material) or ice do not cause blockages. In the operation of the blow-off silencer, considerable forces act; this should be taken into account in the selection. See also 6.3.

A.2.10 Furnaces

Silencers are provided in the exhaust line of furnaces to reduce combustion noise and noise generated by the induced-draught fans. Special requirements are set because of the generally high temperature during operation, and frequently because of chemically aggressive dust carried by the flue gas. This also applies to desulfurisation and denitrification plants.

It is important to select carefully the shape and material to prevent the acoustic performance of the silencer being reduced by dust deposits. The use of resonator silencers is therefore preferred in this area. The occurrence of chemically aggressive liquids during starting-up and closing-down should be taken into account.

A.2.11 Gas turbines and engine test facilities

In the exhaust flow of gas turbines, silencers are often subjected to elevated temperatures, high flow velocities and deposits. These operating conditions require a careful choice of materials. Fibre absorbers should be heat resistant, with long fibres to prevent their being carried away by the alternating forces caused by the flow. Chambers holding the sound-absorbent material should not be too big and should be filled tightly so that no cavities occur. Covers (usually several layers) of perforated plate, mesh and/or cloth should be provided. Mostly, only little pressure loss is permitted in such silencers.

A.2.12 Pneumatic conveyors

For silencers in pneumatic conveyors, for silo ventilation, for the flow of ground stock in crushing plant and other processing plant, functional safety requirements are higher due to the danger of dust deposits. Therefore, resonator silencers are commonly used. The chemical properties of the transported material and explosion safety requirements should be considered (see also 5.6).

A.3 Internal combustion engines

A.3.1 Vehicles

In the operation of internal combustion engines, intake and exhaust noises arise which need to be attenuated by silencers so that the legally specified noise limits are met by the vehicle as a whole and so that sufficient sound comfort for passengers is attained.

Intake noise control is mostly obtained using reactive silencers combined with air filter elements, the whole then being named an attenuator filter. Further attenuation can be obtained by means of additional quarter-wavelength resonators, changes in cross-section and sound-absorbent wall linings of the silencer chamber.

The spectrum of the exhaust noise is determined by the pulsating volume flow emanating from the cylinder outlet elements. Reactive silencers are predominantly used for noise control. The slightly greater pressure loss in comparison with dissipative silencers is acceptable for low and medium power engines. Only high-performance engines with turbo chargers, shock wave chargers and the like are almost exclusively provided with dissipative

silencers for the exhaust system. It is important that the absorber (preferably basalt wool, occasionally combined with stainless steel wool) meet the strict requirements concerning stress caused by gas pulsation, mechanical vibration, high temperatures and chemical influences. It should not harden or be sealed by deposits from the exhaust gas. Even empty expansion chambers without absorber should be designed to allow condensed liquid to drain with the flow. Reactive and dissipative silencers are also used together.

For the low-frequency domain, attenuation is determined by the size and location of the silencer pots in the exhaust line. Venturi nozzles are also used for low-frequency attenuation. In the mid- and high-frequency domain, side branches, perforated tubes, screens and bends are effective. Pronounced minima in the frequency characteristic of airborne sound and sound radiation from the housing should be avoided. Requirements are more difficult to comply with in this application because they should be met for varying operating temperatures depending on engine load, engine speed and on the cooling on the path along the exhaust line.

A.3.2 Stationary engines

Stationary internal combustion engines differ in some aspects from internal combustion engines for automobiles: the speed range of individual engines is much smaller, mostly limited to some fixed modes of operation, which facilitates the selection of the silencer system. In contrast to automobile engines, power output ranges can differ greatly for different plants (up to several megawatts) so that different types of silencers are used. Acoustical requirements are often stricter, for instance for plant in hospitals. Furthermore, sometimes only little pressure loss is permissible, in which case certain types of silencers used for automobiles cannot be used here. In stationary plant with high power output, ignition frequencies are often low. This requires careful layout of the silencer for low frequencies (below 100 Hz).

Annex B
(informative)

Effect of spectral distribution of sound on the declaration of attenuation in one-third-octave or octave bands

According to ISO 7235, attenuation values $D_{1/3,k}$ for a silencer are to be determined for one-third-octave bands. The conversion to full-octave-band values ($D_{1/1}$) can be carried using equation(1). However, the results are accurate for pink noise exclusively. If the source emits sound with a spectrum which varies markedly with the one-third-octave-band centre frequency, the attenuation for a given octave band may deviate significantly from the calculated value.

Deviations in the octave band around 63 Hz are of major importance in practical cases. Table B.1 gives an example of the conversion of one-third-octave-band attenuation values to octave-band values. The one-third-octave-band values (for 50 Hz, 63 Hz and 80 Hz) were drawn from laboratory measurements. The octave-band values refer to actual spectra in two applications.

As can be seen from Table B.1 and Figure B.1, the manufacturer's declaration of 7 dB for the octave-band attenuation is exceeded for the spectrum of the axial fan. The attenuation is much better than claimed. In contrast, the fan with bent blades emits such an unfavourable spectrum that an attenuation of merely 5 dB remains in the octave band. The declared value of 7 dB is not found in laboratory measurements taken in compliance with ISO 7235.

Table B.1 — Example of the conversion of attenuations in one-third-octave bands to the attenuation in the corresponding octave band

	One-third-octave bands			Octave band
Centre frequency, Hz	50	63	80	63
Attenuation in one-third-octave bands, dB	3	12	21	
Sound power level of the source in the laboratory (pink noise), dB	90	90	90	95
Attenuated sound power level, dB	87	78	69	88
Attenuation in the octave band, dB				7
Practical case 1:				
Sound power level of an axial fan, dB	84	88	93	95
Attenuated sound power level, dB	81	76	72	83
Attenuation in the octave band, dB				12
Practical case 2:				
Sound power level of a centrifugal fan, dB	93	88	84	95
Attenuated sound power level, dB	90	76	63	90
Attenuation in the octave band, dB				5

a) Pink noise

b) Axial fan

c) Centrifugal fan

Key:

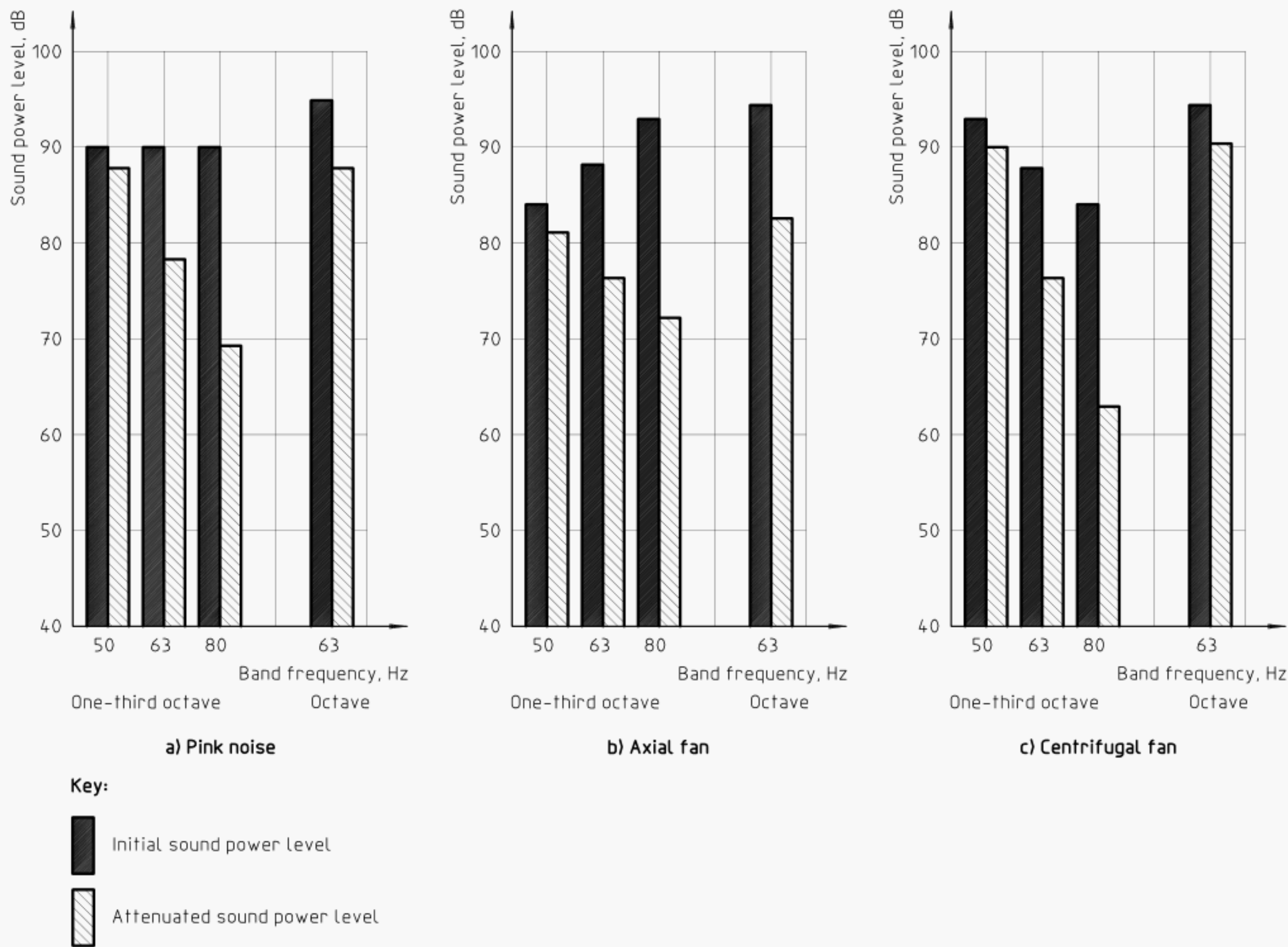Initial sound power level

Attenuated sound power level

Figure B.1 — Graphical presentation of the example shown in Table B.1

Annex C
(informative)

Operating temperatures of sound sources and temperature limits of sound-absorbent materials

Table C.1 — Temperatures to be expected for various sound sources

Sound source	Temperature °C
Steam valve	530
Gas turbine	600
Jet engine	800
Compressor	200
Car engine	400 to 800

Table C.2 — Temperature limits for various sound-absorbent materials

Material	Approximate temperature limit °C
Wool	50
Polymeric foam	150 to 200
Glass fibre cloth	300
Mineral fibre	
with binder	220
without binder	500
Special basalt fibre	750
Sintered metal	
Bronze	400
Stainless steel	600
Special metal	1 000
Stainless steel cloth	500 In special cases: 600

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