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Origin and Control of Noise

1. Fundamentals

Industrialized progress has presented humanity with a host of increasingly acute environmental challenges. These must be deemed to include the noise loads caused by machinery and flow media.

Among such noise sources are fans and the fluid flows in their upstream and downstream ducting. The following explanations and comments are intended to give a brief outline of "sound"-related phenomena and the associated issues.

The sound perceivable by the human ear is generated by vibrations (oscillation) of material constituents (particles) of an elastic medium. These oscillations about a given zero level vary in frequency from about 16 to 16000 Hz, with 1 Hz representing a frequency of one oscillation per second. According to the medium in which this sound propagates, we distinguish between airborne, structure-borne, and waterborne sound transmission.

A tone is a sound that oscillates in the form of a sine-wave (compression and decompression).

A sound of identical frequency gets louder with increasing amplitude, while a rise in frequency is perceived as giving a "higher" sound.

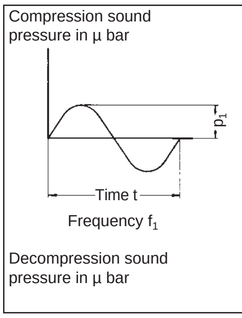


Fig. 1

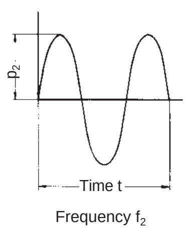


Fig. 2

The tone in Fig. 2 (sound pressure p_2) is perceived as "higher" and generally louder (for details refer to section 2) than its counterpart in Fig. 1 (sound pressure p_1).

Several tones emitted at the same time may blend harmoniously, as in a chord; by contrast, "noise" refers to a random distribution of sound pressures over the frequency range detectable by the human ear. Such noise may easily reach an intensity that is found unpleasant or annoying.

2. Human perception of sound

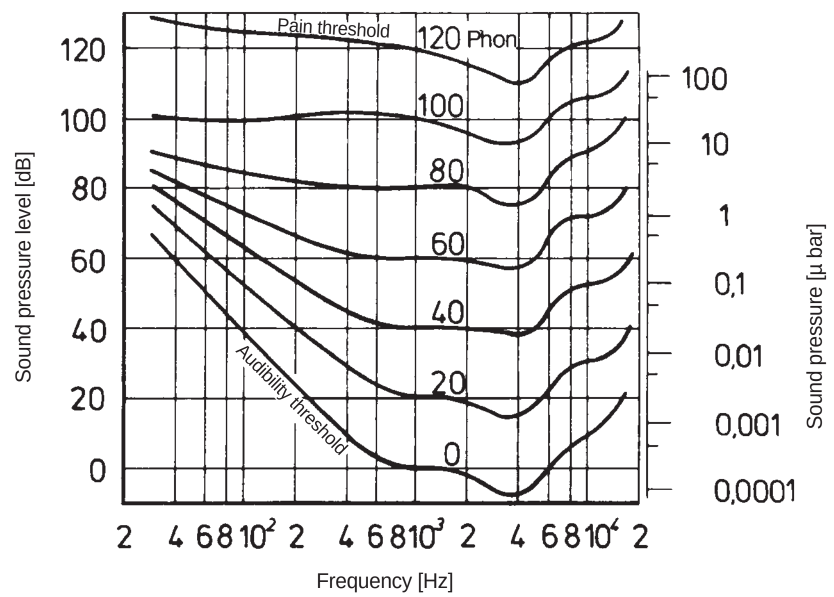
Sound pressure can be accurately measured using appropriate instruments. However, its physiological effects on humans are much more difficult to determine, e.g., because our hearing, when detecting two tones of identical sound pressure but different frequency, will perceive one to be "louder" than the other.

Studies have been carried out in which numerous respondents were asked to rate the loudness of tones of different frequencies against the sound pressure of a 1000-Hz tone. The aim of these experiments was to determine the sound pressure px (or the corresponding sound pressure level in dB, respectively) which, at a frequency of 1000 Hz, was perceived as equally loud as a given sound pressure p_n (or the corresponding sound pressure level in dB, respectively) at a frequency f_m (in Hz). In other words, an identical loudness (expressed in phon) was assigned to the sound pressures p_n and p_x or the corresponding sound pressure levels, respectively. By definition, the magnitude of the sound pressure level and loudness coincide at 1000 Hz.

Fig. 3 shows the iso-loudness curves (curves of identical loudness) obtained in this manner.

Since the shape of these curves changes as a function of both frequency and sound pressure, it proves difficult to devise a convenient measuring instrument yielding objective loudness measurements. But this is only one reason why experts have been seeking to come up with another assessment system. Another reason lies in the fact that the phon curve method can only compare single tones, but our ear perceives single tones differently from noise.

1) For details refer to section 3, "Basic acoustic terminology".



Iso-loudness curves in phon according to Fletcher and Munson

Fig. 3

A solution addressing this state of affairs was found with the so-called “A-weighting curve” which has by now been internationally adopted. It approximates the human frequency response in the range of medium sound pressure levels. To account for the fact that single tones are often perceived as more “piercing” (unpleasant) than multi-tone noise of the same loudness[2]), specifications demanding compliance with a specific overall noise threshold are often combined with a further, more stringent limit for single tones. The emission of single tones by machines is a physical phenomenon which cannot be avoided entirely. A typical single tone of this type is the “blade tone” of a fan, ther frequency of which depends on the number of blades and the fan’s rotational speed (rpm). This primary tone and its whole-numbered multiples (= harmonics) together form the so-called “rotary sound”.

2) The reason why individual tones are perceived as particularly unpleasant or “noisy” lies in their information content (e.g., of a siren)

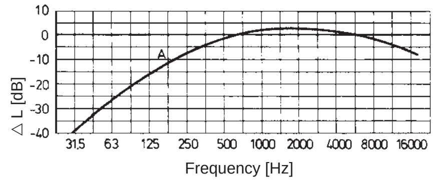
3. Basic acoustic terminology

Units of sound parameters

In acoustics, it is standard practice to work with “levels”, i.e., to consider logarithmic magnitude ratios stated in bels (B) or decibels (dB) (provided that a base 10 logarithm is used) rather than the original parameters with their associated units (= effective values).

$$\text{Level of the sound parameter} = \lg \frac{\text{effective value of the sound parameter}}{\text{reference value of the sound parameter}} \text{ in dB} = 10 \lg \frac{\text{effective value}}{\text{reference value}} \text{ in dB}$$

Thus, since all sound parameters are measured in the same units, it is important to note their designations and to distinguish clearly between, e.g, sound pressure level and sound power level.



A-weighting curve according to DIN 45633, Sheet1

Fig. 4

Sound pressure level L

The sound pressure level L (commonly referred to as “sound level”) indicates the magnitude of the sound pressure at a given measuring point.

By definition, we can write

$$L = 10 \lg \frac{p^2}{p_0^2} = 20 \lg \frac{p}{p_0} \quad \text{in dB}$$

where p = effective value of the sound pressure at the measuring point, e. g., in N/m^2

$p_0 = 2 \times 10^{-5}\ N/m^2$

$= 20\ \mu\ Pa$

$= 2 \cdot 10^{-4}\ \mu\ bar$

(reference sound pressure, audibility threshold for the 1000 Hz tone)

Weighted sound pressure level L_A

The weighted sound pressure level L_A, expressed in dB(A), is obtained from the sound pressure level L by weighting according to DIN EN 60651, Table 4. In the weighting process, the sound pressure level L measured at each frequency is augmented by adding the ΔL value from the A-weighting curve corresponding to that frequency (Fig. 4).

It is evident from the weighting curve that the L_A values remain clearly below the L values at lower frequencies while exceeding these L values, albeit only by a narrow margin, in the higher frequency range.

Measuring-surface sound pressure levels $\overline{L}$ and $\overline{L}_A$

The “measuring-surface sound pressure level” $\overline{L}$ refers to the average sound level, in acoustic energy terms, determined over the measuring surface and corrected to eliminate stray noise and ambient factors (reflections). $\overline{L}_A$ is the corresponding “weighted” measuring-surface sound pressure level.

The measuring surface S is a theoretical surface deemed to enclose the noise-emitting machine at a specific distance (typically 1 m). Following the exterior shape of the machine, this surface is construed from simple geometrical surfaces or surface elements such as spheres, cylinders or cubes. Individual projecting parts that do not contribute materially to noise output are ignored altogether. Sound-reflecting boundary surfaces such as floors or walls are not included in the measuring surface. Measuring points should be sufficient in number and evenly distributed over the measuring surface. Their number depends on the size of the machine and the homogeneity of the sound field.

Since it is common in acoustics to work with logarithmic ratios, as outlined above, the measuring surface (in m^2) is related to a reference surface, and the measuring surface level L_S is defined as a characteristic variable:

$$L_S = 10 \lg \frac{S}{S_0} \quad \text{in dB}$$

S = measuring surface in m^2

$S_0 = 1\ m^2$ (reference surface)

Sound power level L_W

The total magnitude of the acoustic power emitted by the sound source is expressed by the sound power level L_W.

$$L_W = 10 \lg \frac{W}{W_0} \quad \text{in dB}$$

W = acoustic power in watts emitted as airborne sound

$W_0 = 10^{-12}$ W (reference sound power at the 1000 Hz audibility limit)

Weighted sound power level L_{WA}

By subjecting the sound power level LW to the weighting method explained above for the sound pressure level (again using the A-weighting curve), we obtain the weighted sound power level L_{WA}.

1) Multiple level measurements taken on a given sound source are averaged over a given area or time interval using the following formula:

$$\overline{L} = 10 \lg \left(\frac{1}{n} \cdot \sum_{i=1}^{i=n} 10^{0,1 L_i} \right)$$

If the difference between the individual levels is smaller than 6 dB, an approximation may be carried out by determining the arithmetic mean thus:

$$\overline{L} \approx \frac{1}{n} \cdot \sum_{i=1}^{i=n} 10 \, L_i$$

Interrelationship between sound pressure and sound power level

Unlike the sound pressure p, the sound power W is not measured directly but calculated from the sound pressure p, the sound particle velocity ν and the measuring surface S:

$$W = p \cdot \nu \cdot S$$

with $\nu = \frac{p}{\varrho \cdot c}$

ρ = air density
c = velocity air sound
becomes:

$$W = \frac{p^2}{\varrho \cdot c} \cdot S$$

Assuming that both and ρ and c are constant, we obtain the proportionality law

$$W \sim p^2 \cdot S$$

In level terms, the above yields the practically important expression

$$L_W \approx \overline{L} + 10 \lg \frac{S}{S_0} = \overline{L} + L_S \text{ in dB}$$

and, accordingly,

$$L_{WA} \approx \overline{L}_A + 10 \lg \frac{S}{S_0} = \overline{L}_A + L_S \text{ in dB}$$

Thus, by way of approximation, the sound pressure level L_W can be calculated as the sum of the measuring-surface sound pressure level $\overline{L}$ and the measuring-surface level L_S.

From this equation it can be derived that, for a given sound power level and a spherical or hemi-spherical sound emission into free space (ideal sound propagation conditions), the sound pressure level will decrease by 6 dB when the distance to the sound source is doubled.

This value may increase due to sound absorption by the air or ground, or decrease due to reflection from obstacles. In addition, sound pressure level may be augmented or attenuated by weather factors.

Acoustics	Electricity
$W = p \cdot \nu \cdot S$	N = power U = voltage I = current intensity R = resistance
$\frac{W}{S} = I$	$N = U \cdot I$
$I = p \cdot \nu$	$I = \frac{U}{R}$
$\nu = \frac{p}{\varrho \cdot c}$	$N = \frac{U^2}{R} = I^2 \cdot R$
$I = \frac{p^2}{\varrho \cdot c} = \nu^2 \cdot \quad \cdot c$	$N \sim U^2$
$I \sim p^2$	

Sound intensity is proportional to the square of the sound pressure.

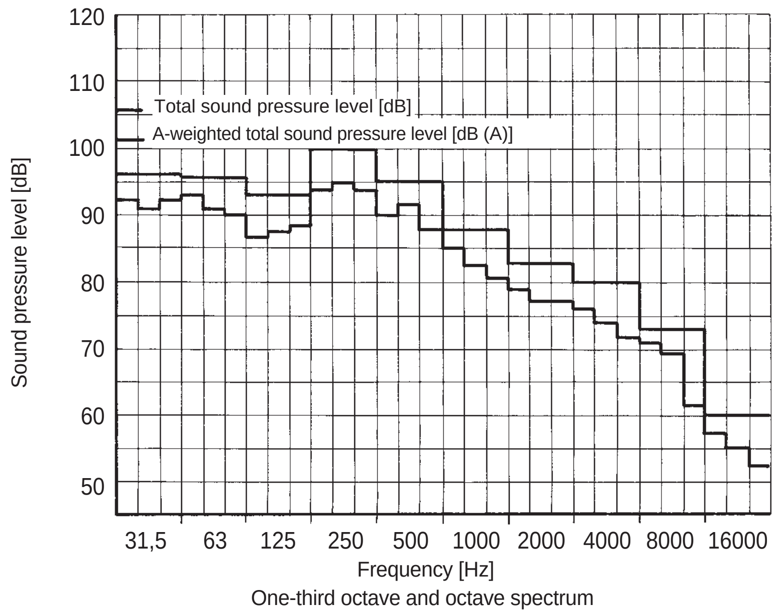


Fig. 5

Sound intensity level L_I

At this point, the acoustic power output per square meter, or “sound intensity” I, should be briefly mentioned:

$$I = \frac{W}{S} \text{ in Watt/m}^2$$

This parameter can be used to illustrate an analogy to the laws of electricity:

The sound intensity is proportional to the square of the sound pressure.

The definition of the corresponding sound intensity level is

$$L_I = 10 \lg \frac{I}{I_0} \text{ in dB}$$

with $I_0 = 10^{-12}$ Watt/m²
(reference sound intensity)

4. Sound analysis

The total or cumulative sound power level of a given acoustic emission is obtained by logarithmically adding many individual sound power levels of diverse frequencies (Fig. 5). For the purpose of acoustic measurements, the audible frequency range has been divided into 10 octave bands.

The width of each octave is characterized in that the upper limit frequency f_o of the spectrum is in a 2:1 ratio to its lower limit frequency f_u.

Octave: $\frac{f_o}{f_u} = 2$

For one-third of an octave, the corresponding ratio is

One-third octave: $\frac{f_o}{f_u} = \sqrt[3]{2}$

The division of the spectrum into one-third octaves is common practice in acoustics.

For the mid-frequencies, the following equation applies:

$f_m = \sqrt{f_u \cdot f_o}$

Hence, we can write:

Octave: $f_m = \sqrt{2} \cdot f_u = \frac{f_o}{\sqrt{2}}$

One-third octave: $f_m = \sqrt[6]{2}\, f_u = \frac{f_o}{\sqrt[6]{2}}$

The individual octave band mid-frequencies are approximately distributed as follows:

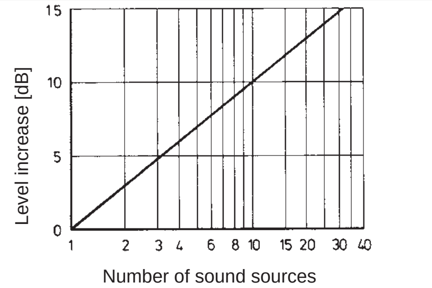
31.5 Hz	1000 Hz
63 Hz	2000 Hz
125 Hz	4000 Hz
250 Hz	8000 Hz
500 Hz	16000 Hz

In practice, the first and last octave band are normally of minor importance only.

Commercially available sound measuring instruments capable of determining sound pressure levels in dB and dB(A) are equipped with switchable octave or one-third octave filters to facilitate frequency analysis. If the resolution of the octave band analysis is too low, one-third octave filters can be activated to provide a more selective analysis.

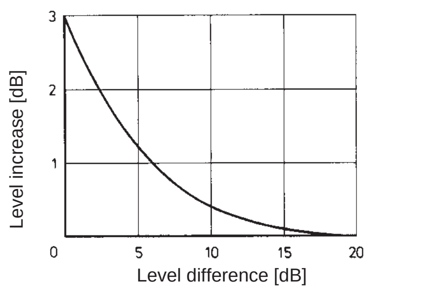
For single tones or noise extending only over a one-third octave band, the one-third octave and octave analysis will yield the same value.

An example of an octave band analysis and one-third octave band analysis is given in Fig. 5.



Addition of multiple sound sources of identical level

Fig. 6



Addition of two sound sources of different levels

Fig. 7

Where the one-third octave band analysis is not selective enough, filters may be used which give an even higher band resolution (search tone analyzer).

5. Addition of levels

From the individual (sound pressure or sound power) levels L_i, the total level L_{tot} is formed according to the following equation:

$$L_{tot} = 10 \lg \sum_{i=1}^{i=n} 10^{\,0{,}1 L_i}$$

When adding sound pressure levels, care must be taken to ensure that all individual levels relate to the same location.

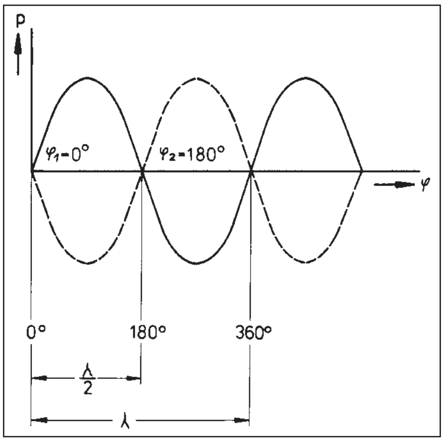


Fig. 8

For the special case where there are n sound sources of identical sound output W_1, the total level L_{Wtot} can thus be calculated as

$L_{Wtot} = L_{W1} + 10 \lg n$

The level increase produce by a rise in the number of sound sources is also illustrated in Fig. 6.

In the special case of two individual sound sources of different levels, the total level is obtained by adding the level difference to the higher level.

As can be seen from Fig. 7, there is virtually no level increase if the level difference exceeds 10 dB. In the special case of two sound sources having the same level (level difference = 0), the level increase is 6 dB (refer also to Fig. 6).

In calculating total levels, the case of two single tones of identical sound pressure p_1, identical frequency f_1 and identical phase angle φ_1 being mutually superimposed merits separate consideration. Contrary to the summation law described above, the total level in this case will be 6 dB higher than the sound pressure level of the single tone:

$$L_{tot} = 10 \lg \left(2 \cdot \frac{p_1}{p_0}\right)^2 = 20 \lg 2 \cdot \frac{p_1}{p_0}$$
$$= L_1 + 20 \lg 2$$

If the phase angles of the two tones are offset by 180 deg. ($\varphi_1 = 0°, \varphi_2 = 180°$) or $\lambda/2$, interference will occur, i.e., the two tones extinguish each other.

These two cases are of practical importance where two fans operate at near-identical rotational speeds on a common duct system. In this case, sound wave superimposition effects will result in periodic level variations referred to as “beat”. The beat frequency can be obtained from the difference between the two rotational speeds.

6. Formation of fan noise

The operating noise of a fan is composed of several sound constituents.

In the boundary zone of fast-moving gas flows, eddies will form as a result of viscosity phenomena. In the case of fans these flow separation and vortex (turbulence) effects take place at the trailing edge of the impeller blade. The resulting "vortex noise" caused by the rotating impeller must be deemed to constitute the "primary noise" which is superimposed by the noise of flow (usually highly turbulent) in the fan casing and ductwork. The "vortex noise" and "flow noise" generally extend over a broad frequency range, with acoustic power increasing approximately with the 5th - 7th power of the peripheral speed.

Over and beyond such broadband noise, "pulsating" noise may occur at specific frequencies as a result of periodic pressure fluctuations of the conveyed medium. These fluctuations are due to relative movement between the impeller blade and an object in the air flow. Pulsating noise is generated when the flow is impaired in the impeller area by projecting edges, bracing members, or the like (e.g., the cut-off lip on centrifugal fans, guide vanes of axial-flow units). Fan engineers refer to this interference noise as the "blade tone" or "rotary sound"; the main interference frequency can be determined by multiplying the number of blades with the rotational speed (rpm). Whole-numbered multiples of this basic frequency may likewise be emitted (harmonics). Depending on the type and magnitude of the interference, this "rotary sound" phenomenon may cause a significant increase in acoustic power output in specific frequency ranges.

7. Fan sound pressure and sound power levels

Sound pressure level L emitted by a fan can be calculated in advance from its peripheral speed, impeller diameter, and specific constants. Depending on the fan type and performance specifications, a mean weighted sound pressure level L_A between 90 and 110 dB(A) will be obtained (always measured at a distance of 1 m and a 45-deg. angle to the intake direction).

The A-weighted sound power level can be calculated, by way of approximation, using the following equation:

$$L_{WA} = K + 10 \lg \frac{\mathring{V}}{\mathring{V}_0} + 20 \lg \frac{p}{p_0} \quad \text{in dB (A)}$$

where
p = total pressure difference in μ bar
p_0 = 100 μ bar
$\mathring{V}$ = volume flow in m³/h
$\mathring{V}_0$ = 1 m³/h
K ≈ 11 dB (A) for centrifugal fans with backward-curved blades
K ≈ 16 dB (A) for axial-flow fans

The basic magnitude determining the propagation of fan noise is the overall acoustic power W or the corresponding sound power level L_W.

When it comes to noise emissions, a distinction must be made between

- "primary" acoustic power output originating in the fluid flow and transmitted to the environment through the intake and outlet connections;
- "secondary" emissions by components excited by structure-borne noise.

Primary acoustic power emissions W_S and W_D:

L_{WS} : Level of sound power emitted from fan inlet opening, against the fluid flow direction.

L_{WD} : Level of sound power emitted via the outlet connection, along with the fluid flow direction.

Secondary acoustic power emissions W_G, W_U, W_{SL} and W_{DL}:

L_{WG} : Acoustic power W_G impinging on the casing wall excites structure-borne noise in it. The casing emits this noise to the environment as airborne noise. The corresponding sound power level is L_{WG}.

L_{WU} : Structure-borne noise is transmitted from the casing to its supporting structure, then emitted to the environment from here as airborne noise. The corresponding sound power level is L_{WU}.

LWSL, LWDL: Acoustic power emitted through the fan inlet and outlet openings (W_S, W_D) excites structure-borne noise in the ducting system which is connected to (and hence, mechanically decoupled from) it by compensators. This structure-borne noise is radiated off into the surrounding space in the form of airborne noise. The corresponding sound power levels are L_{WSL}, L_{WDL}

One method of quantitating these individual acoustic power parameters is to assume appropriate measuring surfaces at a distance of 1 m from the respective components, as described above. The acoustic power emitted as airborne noise can then be calculated by the approximative equation

$$L_{Wi} = L_i + 10 \lg \frac{S_i}{S_0}$$

For the specific case, S, SL, D, DL, G or U must be substituted for i as appropriate.

By way of example, the foregoing is illustrated in Fig. 10 to Fig. 13 for a fresh-air fan (with and without noise control features). The propagation of sound is symbolized by arrows of different colour (primary and secondary sound sources) and arrows of different length (level).

8. Noise control measures

Noise loads emitted by a fan can be reduced by means of sound-proofing material and anti-noise covers ("hoods") on the one hand (sound insulation), and by means of silencers on the other (sound absorption).

A silencer attenuates the propagation of sound in the ducting system without interfering materially with the flow of the conveyed medium (reduction of L_{WS} and L_{WD} through conversion of sound energy into thermal energy).

Sound-proofing materials and anti-noise covers go a significant way towards isolating the fan environment against the propagation of airborne noise emitted by components in which structure-borne noise has been excited (reduction of L_{WG}, L_{WSL}, L_{WDL} by reflecting sound energy back towards the source and, additionally, by converting some of it into heat).

To dampen the noise output of fans, silencers are employed. Depending on the application case, these may be of the non-tuned absorption type or chamber-type silencers tuned to a specific frequency (interference, resonance or λ/4 silencers).

Both designs rely on multiple channels arranged parallel to the direction of flow.

Channel designs differ according to the underlying operating principle (i.e., friction or a combination of reflection and interference).

In an absorption silencer the space between the (perforated sheet) channels is filled with a noise-absorbing mineral fibre material.

Mineral fibre packing decelerates the flow molecules previously excited to oscillate; as a result, the sound energy propagating through the perforations is converted into thermal energy by molecular friction.

Absorption silencers are used to dampen a broadband noise spectrum. However, they can only be used successfully with low-dust flow media, since dust loads in the medium will clog up the perforations in the channel wall. In this condition, the effectiveness of the silencer is greatly reduced.

For high-dust media, preference will therefore be given to a chamber-type unit (λ/4 silencer), despite its limited effectiveness in damping broadband noise. By its very operating principle, such a silencer primarily dampens prominent single tones to which it must therefore be tuned. In addition to this single-tone attenuation, a certain broadband damping capability is achieved by applying sound-absorbing mineral fibre mats to the underside of partitions forming the silencer chambers (Fig. 9). Single-tone damping action is based on the principle of reflection and interference. The key parameter in the design of chamber silencers is the chamber depth, which must be approximately one-fourth of the wavelength of the interference tone (t = λ/4). Only then can the following process take place:

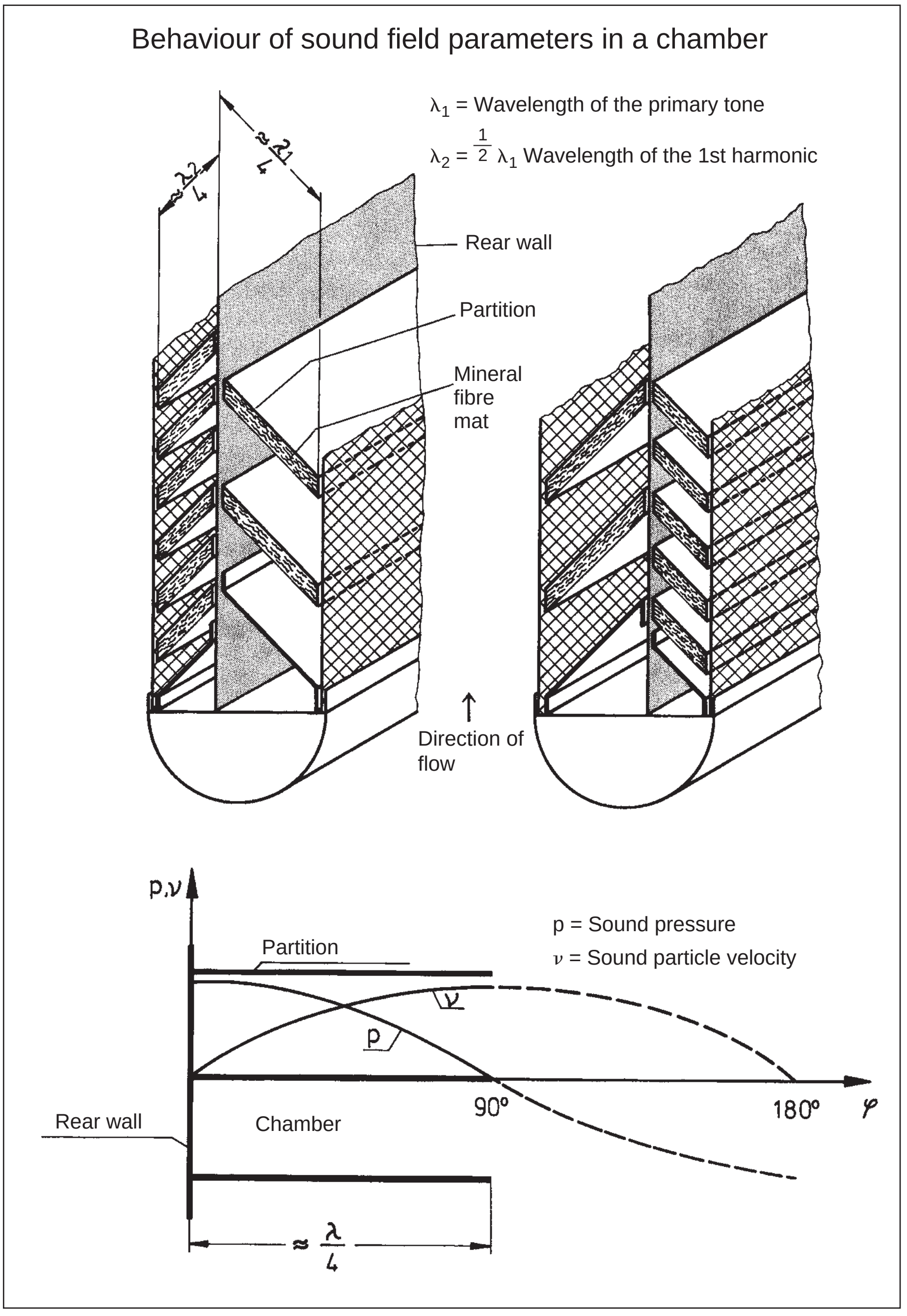


Fig. 9

At a distance of λ/4 from the outer chamber wall the sound wave impinges on the “acoustically hard” rear wall of the chamber, from which it is reflected. It then travels back by a longer distance of λ/4 towards the sound source, arriving there with its phase angle offset by λ/2 (180°) against the next following sound wave. As a result, interference occurs, i.e., the tones extinguish each other.

Sound emission on a fresh-air fan without noise control features (Fig. 10)

L_W : Total sound power generated by the fan

L_{WD} : Sound power emitted in the direction of flow via the outlet connection.

L_{WDL} : Sound power emitted as airborne noise by the outlet duct as a result of LWD and structure-borne noise transmission.

L_{WS} : Sound power emitted against the direction of flow via the inlet connection.

L_{WSL} : Sound power emitted as airborne noise by the inlet duct as a result of L_{WS} and structure-borne noise transmission (Figs. 11 and 12)

L_{WG} : Sound power emitted as airborne noise by the casing due to excitation of structure-borne noise by sound energy in the flow.

L_{WU} : Sound power emitted as airborne noise by the support structure as a result of structure-borne noise transmitted from the casing,

L_{WM} : Sound emissions from adjoining equipment (e.g., fan motor)

Noise control on a fresh-air fan using an inlet silencer and insulation (Fig. 11)

Damping of the L_{WS} sound power level by means of an inlet silencer.

Reduction of the L_{WG}, L_{WDL}, L_{WSL} sound power levels by means of acoustic insulation. Since the sound energy within the system is not consumed, all surfaces not lined with insulation will emit this sound power in unmitigated form.

L_{WD} sound power is emitted into the ducting.

Sound damping: (by absorption): Sound energy penetrates porous walls and is converted into thermal energy through viscous friction.

Sound insulation: Sound energy impinges on non-porous walls and is reflected.

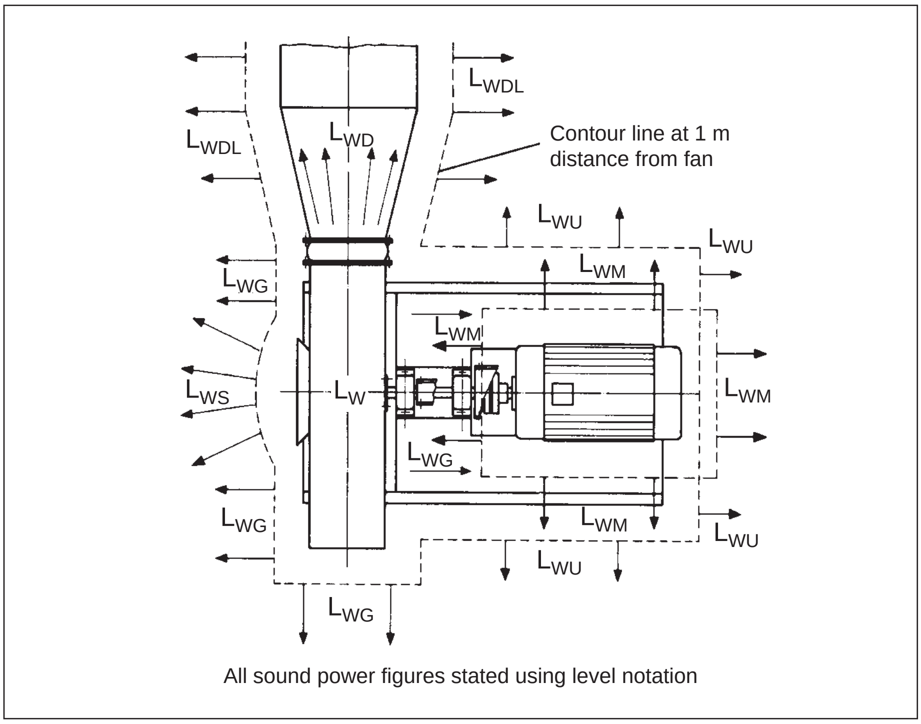


Fig. 10

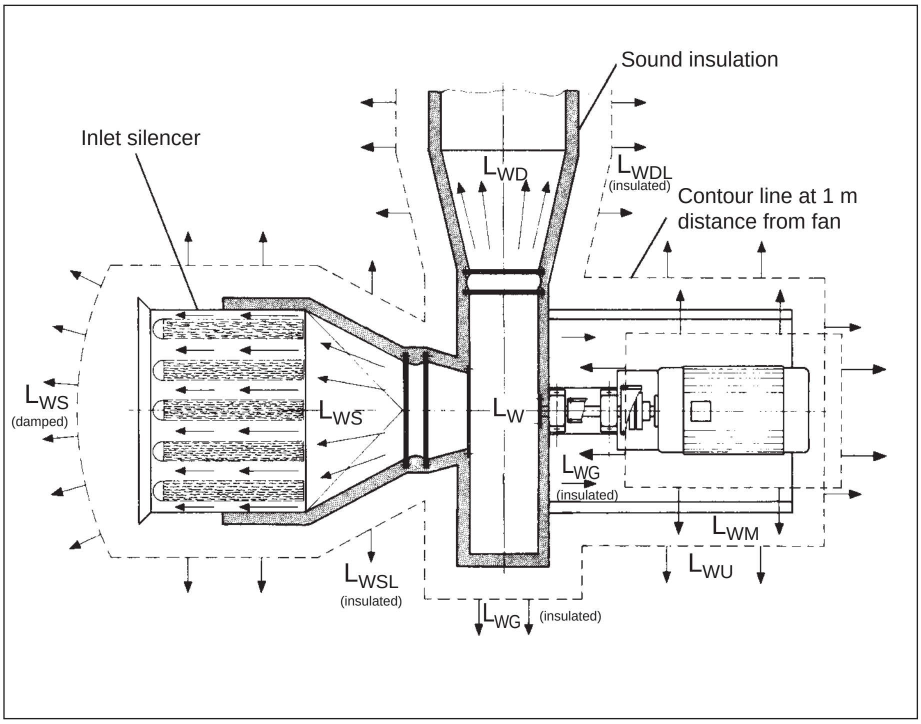


Fig. 11

Noise control on a fresh-air fan using an inlet- and outlet silencer and insulation (Fig. 12)

In addition to the measures detailed in Fig. 11, sound power L_{WD} emitted via the outlet connection is reduced by an outlet silencer.

Noise control on a fresh-air fan using an anti-noise hood with integrated inlet silencer (Fig. 13)

L_{WG} and L_{WU} sound power emitted as airborne noise from the fan and L_{WM} sound power coming from the motor are damped by an enclosure ("hood").

Sound power leaking through the inlet opening (L_{WS}) is reduced to the permitted level by the silencer integrated into the enclosure.

The emission of L_{WDL} sound power by the outlet duct to the environment can be reduced by insulation (resulting in an emission of L_{WD} sound power into the ducting) or by installation of an outlet silencer.

Design of the anti-noise hood (enclosure) must ensure, through an appropriate arrangement of air inlet and outlet points, that the heat generated the motor and fan will be dissipated to the necessary extent, i.e., that a specified maximum temperature within the hood will not be exceeded. If necessary, an exterior fan system may have to be provided for this purpose (e.g., in the case of hot-gas fans).

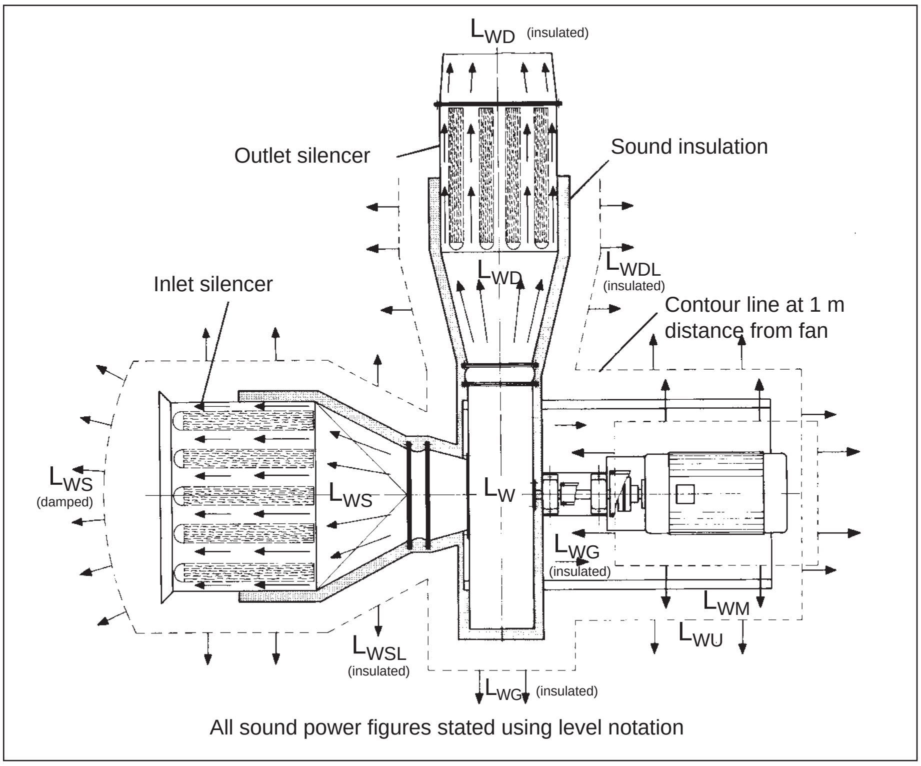


Fig. 12

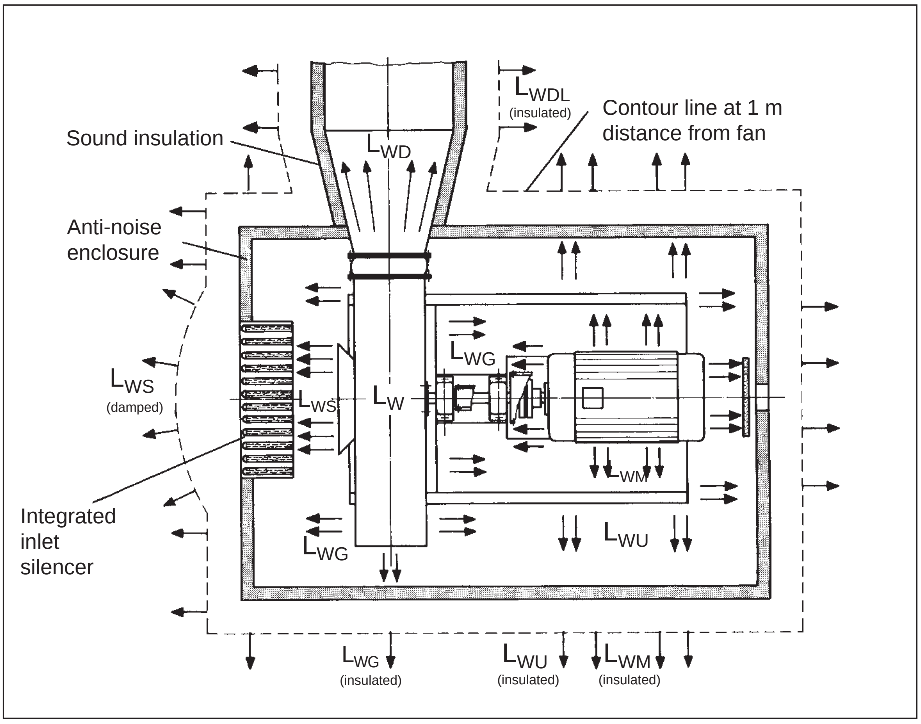


Bild 13