Fundamentals of silencing and their practical application in screw compressor plants

Dr.-Ing. Johann Lenz

Reducing sound emissions of machinery and its equipment is considered to be increasingly important in many areas of industry. To achieve this, in general, a metrological determination of the initial sound situation is performed. This includes the acquisition of air-borne, structure-borne and fluid-borne sound. Evaluating and analysing these data the dominant sound sources can be identified and the responsible propagation mechanisms are revealed. Based upon this information an effective mitigation can be engineered.

This article is divided into two sections. The first section covers the theoretical background of sound engineering. The second section illustrates their application by means of practical examples.

Theory of sound

Air or fluid-borne sound is generally considered as weak pressure fluctuations. These are related to relative movement of individual particles. The speed at which these pressure pulsations propagate is known as the speed of sound α. The speed of sound is dependent on thermophysical properties of the fluid (isentropic exponent κ and specific universal gas constant R_s) and the temperature T according to the following relationship:

$$a = \sqrt{\kappa \cdot R_s \cdot T}$$

The movement of the particles always takes place in the propagation direction; these are known as the so-called longitudinal waves (Fig. 1).

Amplitude and frequencies are the major characterisation properties of sound pressure fluctuations in addition to material-dependent sound velocity. Sound is perceptible to the human ear in a frequency range from approximately 16 to 20,000 Hz.

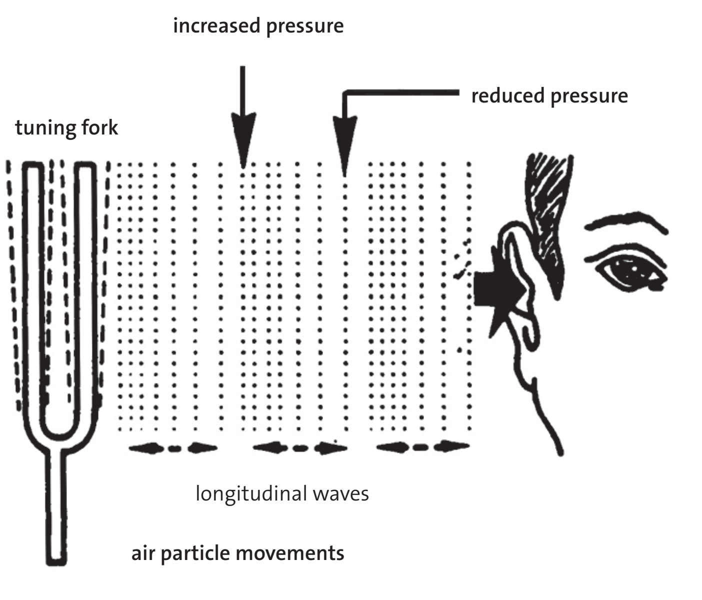

Fig. 1: Propagation of sound waves

Mechanical vibrations, which are propagated in solids, are referred to as structure-borne sound. Structure-borne sound can be emitted from surfaces. In this case, the surface behaves like the moving diaphragm present in a loudspeaker. This diaphragm causes the air to vibrate. The structure-borne sound is transformed into air-borne sound perceivable to the human ear.

The so-called sound pressure is used to quantify air-borne sound. This corresponds to the dynamic fluctuation of the sound pressure around a static mean value. The human ear is able to perceive very low and even very high pressure fluctuations. The sound pressure level L_p is used to divide this wide listening range into a practical scale which is specified in decibels (dB). This relates the actual, effective sound pressure p_{eff} to a reference sound pressure p_0= 20 µPa:

$$L_p = 10 \cdot lg\left(\frac{p_{eff}{}^2}{p_0{}^2}\right) = 20 \cdot lg\left(\frac{p_{eff}}{p_0}\right)$$

According to this conversion of the sound pressure into a sound pressure level, the hearing range of the human ear is approximately 0 to 140 dB. In order to account the different sensitivities of the human ear with regard to the frequency, it is common practice to use a frequency-weighted sound pressure level. One weighting curve for this is the A-weighting curve (Fig. 2). Figure 2 illustrates that low-frequency noise below approx. 500 Hz is perceived to be significantly less disruptive by humans than noise in the medium frequency range of approximately 2 kHz.

Another important parameter in the context of acoustics is sound intensity I. It describes the sound energy, which penetrates a unit of area perpendicular to the propagation direction of the sound waves per unit of time. The intensity can also be expressed in a sound intensity level L_I analogue to the sound pressure level.

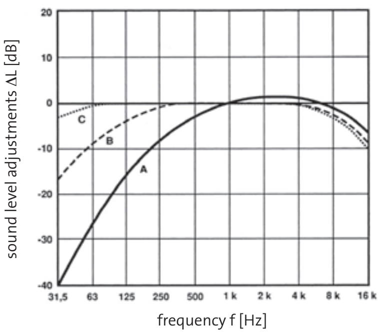

Fig. 2: Weighting curves

$$I = \overline{p \cdot \vec{v}} \rightarrow L_I = 10 \cdot lg\frac{I}{I_0}$$

The reference sound intensity here is I0 = 10^{-12} W/m².

A key parameter to evaluate a sound source is the sound power. Unlike sound intensity and pressure this parameter is independent from the position of source and receiver. Although the sound power level is in general obtained by measurement of intensity and pressure.

When measuring the sound pressure, a subsequent adjustment needs to be performed to account for external and ambient noise. External noise is the term used for all the noises that are not generated or emitted by the machine that is being examined. Furthermore, acoustic effects with an influence on the measured value are taken into account in the ambient noise adjustment. These can be acoustical resonances caused by the surrounding building.

Determining the sound power using intensity technology is beneficial as measurements can be taken even in the case of severe external noises. The methodology is based on a closed measuring surface covering the device under test. Sound which penetrates the measuring surface from outside will exit at the opposite side and offset with spatial averaging of the intensity values. This makes it possible to determine the sound power under real operating and set-up conditions, even if other machines are being operated in the immediate vicinity. Accordingly, the sound power is determined by integrating the sound intensity over surface A of the structure that is being examined using the reference power P_0 = 10^{-13} W.

$$P = \oint_A p \cdot \vec{v} \cdot \overrightarrow{dA} = \oint_A \vec{I} \cdot \overrightarrow{dA}$$

$$\rightarrow L_W = 10 \cdot lg\frac{P}{P_0}$$

The sound power level is used in particular for the site-independent description of a sound source and to calculate the sound pressure level as a function of the distance from the noise source.

The relation between sound power and sound intensity is illustrated in figure 3. While the sound intensity decreases as the distance from the sound source increases, the sound power remains constant because it is calculated by the projected sound intensity normal to a surface.

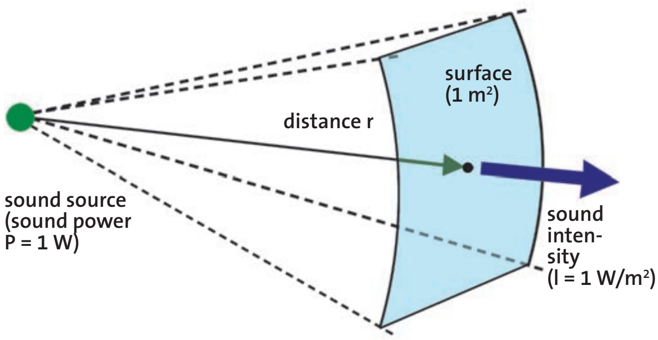

Fig. 3: Illustration of the sound power and the sound intensity of an idealised point source

Example of a sound emission investigation of a screw compressor plant

In the following section the significant correlations of the acoustic principles are illustrated within the scope of a sound optimisation of a screw compressor which is used to provide compressed air.

Application-related noise abatement measures are then shown on the basis of the recorded test results.

Initial situation

A screw compressor and its connected electric motor is located on a resiliently mounted base frame placed in an adapted acoustic hood (Fig. 4). The hood is ventilated by means of an axial fan. Based on a metrological examination, an assessment of the plant design is demanded. If reasonable, improvements to the design in terms of sound emission mitigation should be recommended.

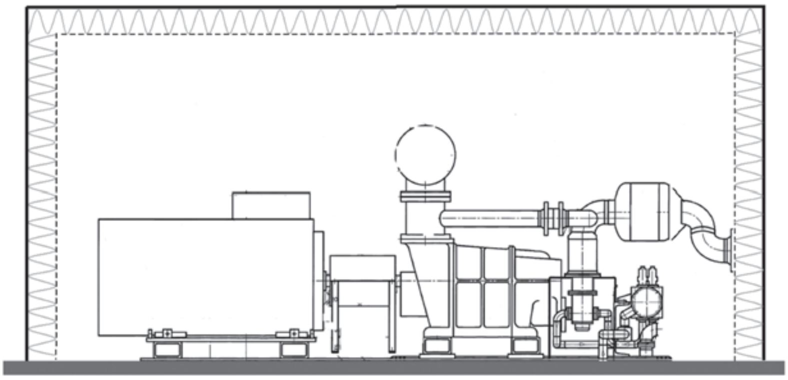

Fig. 4: Compressor system with acoustic hood (schematic construction)

Procedure

The screw compressor plant was examined on a test bench, which was acoustically separated from adjacent hall areas by absorbing partition walls. The power-based average sound pressure level was recorded at a total of 12 monitoring points around the compressor plant (height 1.6 m; distance 1 m). Figure 5 shows the A-weighted 1/3-octave band spectrum of the calculated sound power level of the entire system. The influence of the external and ambient noises was taken into account for this.

A "whistling noise" with the third/one-third octave mid-band frequency of 5,000 Hz was emitted on a flange on the pressure pipe. In addition, the sound emission of the plant was dominated by a low-frequency sound of about 125 Hz. The low-frequency sound as well as the discharge frequencies of both screw compressor stages (approx. 1,200 Hz) and their harmonics can be distinctively identified in the non-weighted narrow band spectrum of the sound pressure (Fig. 6).

It turned out that the low-frequency tone at about 125 Hz matched the rotational frequency of the v-belt pulleys of the two compressor stages. After analysing the structure-borne sound, it became obvious that inadequate alignment or a residual imbalance in the rotors were causing this noise. A further tonal component with f = 375 Hz was emitted by the axial fan of the acoustic hood.

To investigate the efficiency of the soundproofing capsule, acoustic measurements were made while operating the compressor with open and with closed doors at a distance of one metre. The following third-octave band spectrum shows the estimated frequency-related insulation loss D_e of the soundproofing capsule (Fig. 7).

The determined level difference during operation of the compressor is D_e = 16.1 dB. At the mid-band frequency of f = 125 Hz, the difference between an open and closed door was found to be only 4.5 dB.

To analyse the cause of this, the insertion loss of the soundproofing capsule was determined during a further experiment with external excitation while the compressor was shut off (Fig. 8). For this experiment a speaker was used.

The determined level difference is D_e = 19.1 dB. In the third-octave band of f = 125 Hz, the determined level difference rose from approx. 4.5 dB to 9.5 dB. This reveals that the soundproofing capsule is directly stimulated in the 125 Hz range to produce structure-borne noise and emits this as air-borne noise.

Further analysis identified the partial sound power level related to air-borne sound of individual components of the acoustic hood in addition to the total sound power level. The analysis was performed using the in-

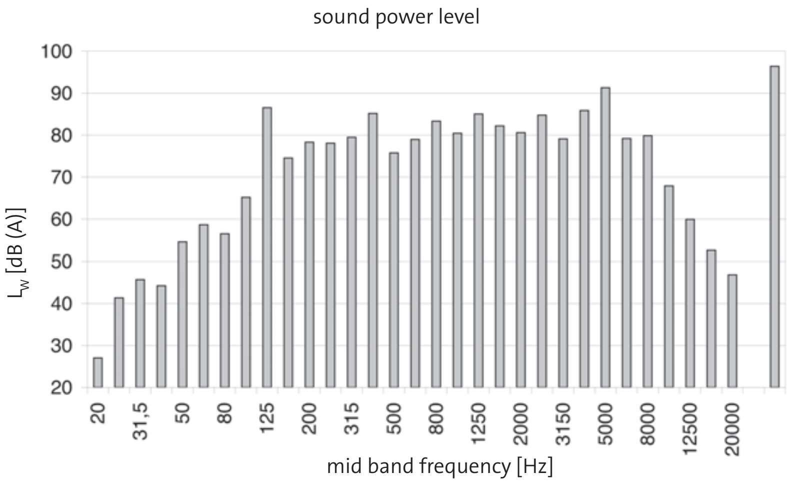

Fig. 5: A-weighted 1/3-octave band spectrum of the sound power level (initial state)

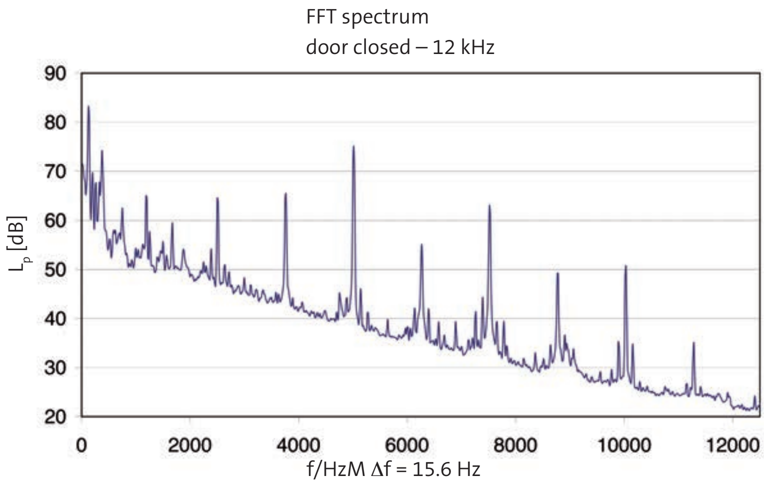

Fig. 6: FFT narrow band frequency spectrum

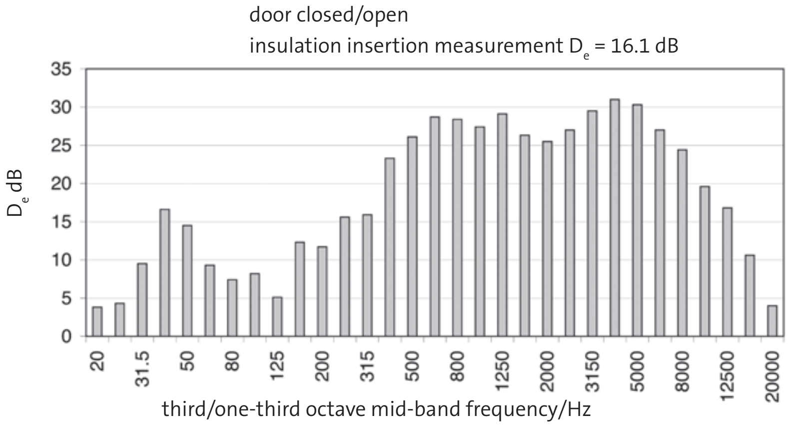

Fig. 7: Insulation loss of the soundproofing capsule

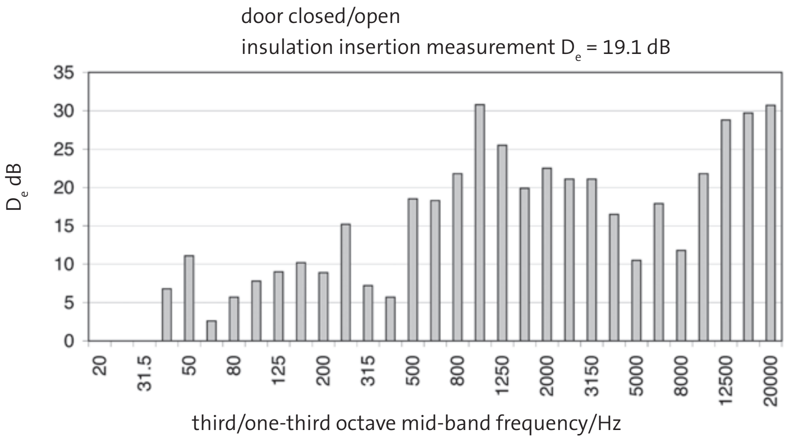

Fig. 8: Insertion loss of the acoustic hood gathered by loudspeaker excitation

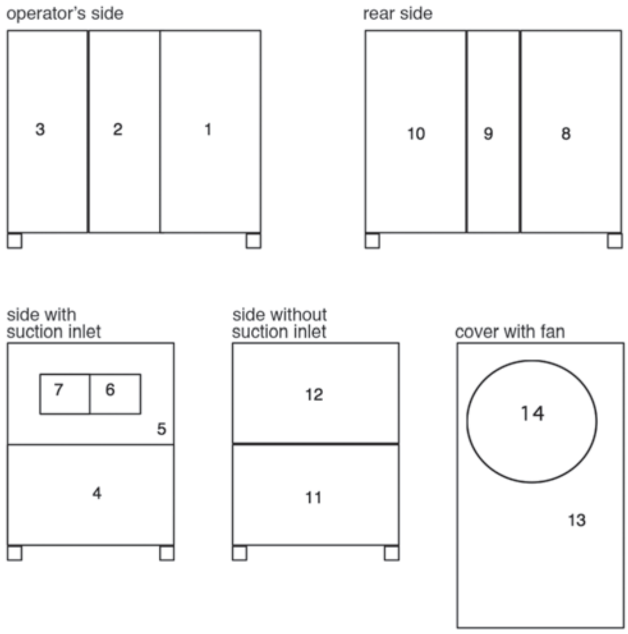

Fig. 9: Examined sub-areas (sketches)

tensity measuring process according to DIN ISO 9614-2 for individual sub-areas.

By balancing the sound power level, noise abatement measures can be simulated for individual components and it is possible to separate their effect on the total sound power level.

Sub-area	l_{eq} [dB(A)]	S [m²]	L_W [dB(A)]
1	78.1	2.0	81.1
2	74.7	1.5	76.5
3	79.4	1.6	81.4
8	75.1	2.0	78.1
9	78.4	1.1	78.7
10	80.7	2.0	83.7
11–12	81.7	3.3	86.9
13	75.2	4.2	81.4
Floor gap 10 cm	81.7	0.8	80.7
4–5	75.1	2.9	79.7
6–7	80.2	0.4	76.2
14	95.9	1.2	96.7
Total:			**97.9**

Table 1: Sound power level of the sub-areas

Result

The sub-area 14 (axial fan) with a level of 96.7 dB(A) has a decisive effect on the total sound power level of 97.9 dB(A) and could therefore be localised as the dominant area determining the sound profile of the hood.

The following sketch shows the layout of the axial fan. The fan freely emits the air-borne sound to the outside.

A comparable situation for the airflow can be produced if the axial fan is mounted on the inside of the soundproofing capsule and the remaining sheet metal duct is lined with absorbing material (see Fig. 11). This creates a highly effective silencer route that dampens the fan noise as well as the compressor noise.

Under the proviso, that the axial fan is moved inside and the existing duct is transformed into a silencer line with absorption material, a noise reduction in the valve noise of about 10 dB can be expected. The calculated total sound power level in the compressor plant is reduced to L_W = 93.4 dB(A) by implementing this mitigation measure.

The advantage of this approach is that the effect of different types of noise reduction can be specifically determined in advance using a simple calculation model.

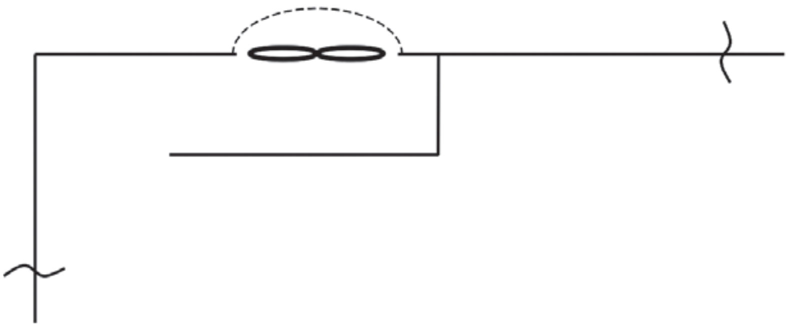

Fig. 10: Sketch of the existing situation with sound-reflecting metal duct.

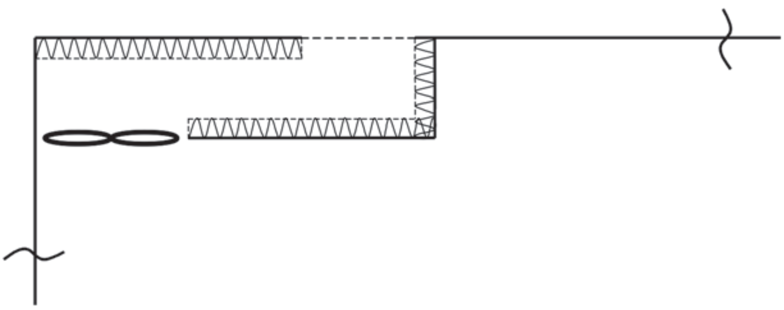

Fig. 11: Sketch of a situation with silencer

Summary

When reducing the noise from machinery and equipment, the following also applies: “The situation can only be mastered if one knows the cause”.

The noise emission of relevant sound sources can be determined in a frequency-selective manner on the basis of a metrological investigation. It is possible to assess noisy areas to allow sound optimisation when sound transmission, propagation and emission are known.

Bibliography

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Author:
Dr.-Ing. Johann Lenz,
KÖTTER Consulting Engineers GmbH & Co. KG, Rheine