Technical Acoustics and Noise Control *(lecture notes for self-learning)*

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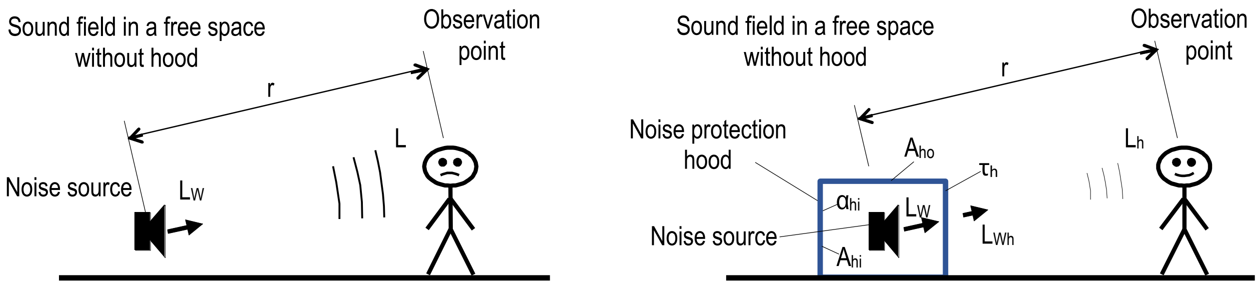
Lecture 14.

Content:

14.1. Noise protection hood: A soundproof wall built around the noise source, which completely separates the noise source from the surrounding space, is called a noise protection hood. The noise protection hood is an effective noise reduction method that can be used both in free field and in a closed space bounded by sound-reflecting walls. Depending on the noise source, the size of the noise protection case can range from a box that is a pair times ten centimeters in size to a noise protection machine house of several tens of meters. The noise-reducing effect of the case is expressed by the insertion loss. The insertion loss of the housing is the difference between the sound pressure levels determined at the same measuring point without the hood and in the presence of the hood. The derivation of the insertion loss (ΔL_h) of the noise protection hood is presented for a noise source placed in a free field (see Figure).



Reducing noise with noise protection hood in free acoustic field

Calculate sound field without hood:

The calculation of the sound field of a point-like sound source radiating uniformly in all directions has been presented in detail earlier. Accordingly, as a function of the sound power emitted by the point source (P), the distance between the sound source and the observation point (r) and the directivity coefficient (D) (see left side of the previous figure), the square of the effective sound pressure,

$$p_{eff}^2 = P\frac{\rho_0 a D}{4\pi r^2}$$

To calculate the sound field in the presence of the hood:

Build a closed case made of soundproof (resist against sound propagation) material around the point-like sound source placed in the free field. Cover the internal surface of the case with a sound-absorbing material (causing loss during sound propagation). (High sound insulation can be achieved with high rigidity, internal attenuation and mass per unit area. Such a wall usually has acoustic hard, sound-reflecting properties.) In this case, the sound wave emitted from the noise source passes through the inner sound-absorbing layer and then reflected back from the acoustic hard surface, and passes again the sound-absorbing material. Thus, during each reflection, the sound wave passes twice through the sound-absorbing layer, while a part of the mechanical worker's ability is dissipated, that means, irreversibly transformed into an increase in internal energy (heating up). If the sound-reflecting ability of the outer wall is good enough, a lot of reflection develops, a significant portion of the sound power emitted from the source is converted to heat, and only a small portion passes through the outer wall of the case. Due to the lower sound power output in the presence of the case, the sound field is weaker and the sound intensity, or the square of the effective sound pressure at the same observation point is smaller.

The sound power passing through the wall of the noise hood is performed similar to the calculation of the sound field bounded by the sound-reflecting walls. A diffuse sound field was assumed in the calculation of the reflected sound field. The inner surface of the case is usually small, the case wall is good sound-reflecting (behind the inner sound-absorbing cover there is a sound-reflecting outer wall), so the room constant is small and the reflected sound field is dominant, comparing to the direct sound field. After switching on the noise source and when the steady state occurs, the sound powers entering and leaving the reflected sound space of the case (P_{rhin} and P_{rhout}) are the same.

$$P_{rhin} = P_{rhout}$$

The power entering the reflected sound space, is the part after the first reflection, which is the product of the sound power output (P) and the surface-weighted average sound reflection coefficient ($\overline{r}_{hi}$). The sound power output from the reflected sound field is the product of the sound intensity incident on the case wall (I_{hi}), the inner surface of the case (A_{hi}) and the surface-weighted average sound absorption coefficient ($\overline{\alpha}_{tb}$) on the inner surface of the case,

$$P\overline{r}_{hi} = I_{hi}A_{hi}\overline{\alpha}_{hi}$$

To reduce the number of the unknown variables, take into consideration, that ($\overline{r}_{hi} = 1 - \overline{\alpha}_{hi}$). After replacement,

$$P(1 - \overline{\alpha}_{hi}) = I_{hi}A_{hi}\overline{\alpha}_{hi}$$

The sound intensity, passes the wall of the hood (I_{ho}), is the product of the intensity incident to the internal surface of the hood wall (I_{hi}) and the surface-weighted average sound transmission coefficient ($\overline{\tau}$),

$$I_{ho} = \overline{\tau}\, I_{hi} = \overline{\tau}\frac{P}{\frac{A_{hi}\overline{\alpha}_{hi}}{(1 - \overline{\alpha}_{hi})}} = \overline{\tau}\frac{P}{R_{ch}}$$

Where, the room constant, related to the internal surface of the hood,

$$R_{ch} = \frac{A_{hi}\overline{\alpha}_{hi}}{(1 - \overline{\alpha}_{hi})}$$

The acoustic power radiated by the outer surface of the hood to the free acoustic field (P_{ho}), is the product of the sound intensity passes the wall of the hood (I_{ho}) and the hood outer surface contacting the ambient free field (A_{ho}),

$$P_{ho} = I_{ho}\, A_{ho} = \overline{\tau}\frac{P\, A_{ho}}{R_{ch}}$$

The square of the effective sound pressure as a function of the sound power (P_{ho}) emitted from the outer surface of the hood into the free space, the distance between the sound source and the observation point (r) and the directivity coefficient (D) (see right side of the previous figure),

$$p_{effh}^2 = P_{ho}\frac{\rho_0 aD}{4\pi r^2} = \bar{\tau}\frac{P\,A_{ho}}{R_{ch}}\frac{\rho_0 aD}{4\pi r^2}$$

The insertion loss (ΔL_h) of the soundproof walled hood is the difference between the sound pressure levels determined at the same observation point without the hood and in the presence of the hood,

$$\Delta L_h = L - L_h = 10lg\frac{p_{eff}^2}{p_0^2} - 10lg\frac{p_{effh}^2}{p_0^2} = 10lg\frac{p_{eff}^2}{p_{effh}^2} = 10lg\frac{R_{ch}}{\bar{\tau}\,A_{ho}} = 10lg\frac{1}{\bar{\tau}} + 10lg\frac{R_{ch}}{A_{ho}}$$

$$\Delta L_h = R + 10lg\frac{R_{ch}}{A_{ho}}\ [dB]$$

Comments:

- The insertion loss (ΔL_h) of the noise protection hood is a function of the sound transmission loss (R) of the case wall, the room constant (R_{ch}) for the inner surface of the case and the outer surface (A_{ho}) of the case. The transmission loss of the wall and the room constant (sound absorption capacity) for the inner surface vary as a function of frequency, so the insertion loss of the hood varies as a function of frequency.

- The insertion loss was derived for a noise source placed in an open space, but in a space bounded by sound-reflecting walls we get practically the same result.

- The insertion loss (ΔL_h) of the hood is high, if the enclosure wall sound transmission loss (R) is high (low sound transmission factor ($\overline{\tau}$)), high enclosure room constant (R_{ch}), that means high enclosure internal surface area (A_{hi}) and high enclosure internal surface-weighted average sound absorption coefficient ($\overline{\alpha}_{tb}$)), and small outer surface of the case adjacent to the free space (A_{ho}).

- If ($\overline{\alpha}_{tb}$) is large (the case wall is a good sound absorber) and ($\overline{\tau}$) is small (the case wall resists against sound propagation), this means that a large internal propagation loss occurs in the wall. Such a property usually does not have a wall made of a single layer of material, so the good case wall is made with a complex layer order. Important parts of the layer order are the "acoustically hard" layer (e.g. sheet steel, reinforced concrete) on the outer side of the case wall, which prevents the propagation of sound by reflection, and the sound-absorbing layer on the inner (at the source-side) surface of the wall (e.g. glass or rock wool).

- The inner surface of the hood can be wedge-shaped or can be enlarged with a corrugated surface design. The outer surface adjacent to the free space can be reduced by placing it on the base level, edge or corner.

- Practical advice, the wall of the case and its connection to the base level must be made without gaps. There must be no mechanical connection between the noise source and the housing wall to avoid the transmission of vibrations.

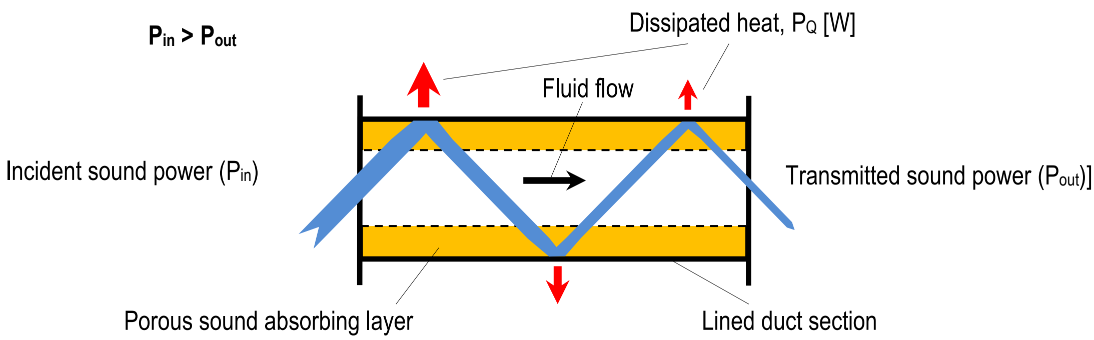
- If there is a noise source in the housing that heats up during operation (e.g. electric motor), the supply and exhaust air of the cooling air must be implemented via a silenced duct section. The insertion losses of the silencers and the noise protection hood must be equal. The same applies to the construction of other technological openings.

14.2. Duct silencer: The propagation of noise in pipes and ducts is reduced by installing a silencer. The muffler is a duct section characterized by low flow resistance and high acoustic resistance (insertion loss). Duct silencers are generally used to reduce noise in gaseous, rarely liquid media. Typical applications are mufflers built into the intake and discharge ports of flow technology machines (e.g. fans, blowers or compressors) or, for energy machinery and equipment, internal combustion engine flue gas exhaust, gas turbine air intake and exhaust or boiler flue gas chimney mufflers.

Reactive (left) and absorber (right) type duct silencers

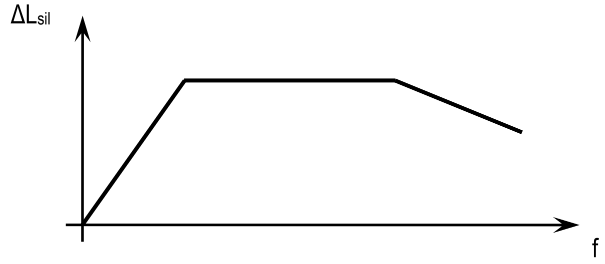
From the point of view of noise protection, an important feature of silencers is the insertion loss (difference in sound pressure levels measured in a duct without a silencer and with a silencer at the fixed point). There are two main types of duct silencers, absorber and reactive silencers (see the previous figure), in addition to which we also consider special noise reduction devices that can be used in pipes and ducts (e.g. active duct silencer, blow-down silencer).

Absorber type silencers: The inner surface of the absorber type silencers is covered with a layer of porous (usually glass or rock wool, less often open-cell polyurethane foam,…) sound-absorbing material. Due to the design of the muffler, a significant portion of the incoming sound waves are sooner or later projected onto the surface of the porous material. In the porous material there is a dissipative loss during the propagation of sound between the thin fibers and in the small holes (the working ability of the sound is dissipated, it is irreversibly transformed into an increase in internal energy), the following figure shows the schematic drawing of the process.



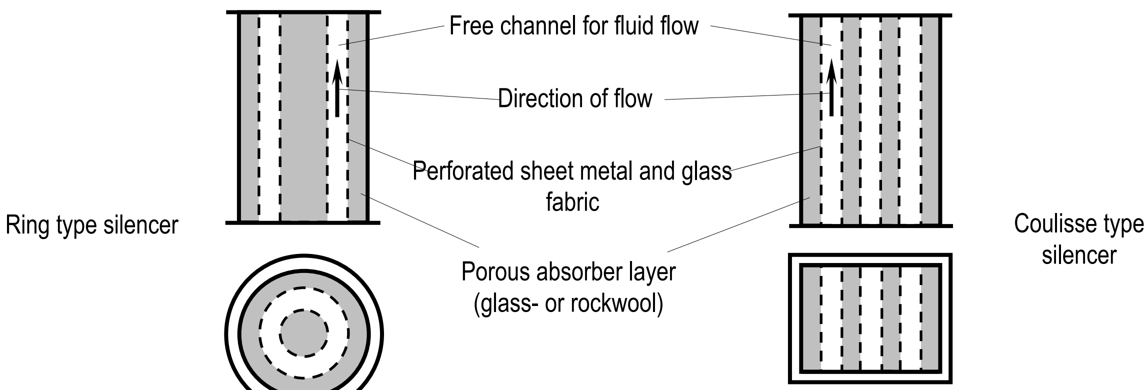
Loss in porous material during sound propagation in a lined channel

The formation of dissipative loss in the porous material layer is a complex phenomenon, in which viscous fluid friction, thermal conduction and in some cases even Coulomb dry friction play an important role. The loss is governed by the characteristics of the boundary layer between the solid body and the air, as well as by the large interaction surface (for fluid friction and heat transfer). For the engineering design work an easy-to-use mathematical model for describing losses in the porous material layer and thus in mufflers is currently not available. The sizing of absorber silencers is performed on the basis of dimensionless sound absorption curves determined by empirical methods and model measurements, or on the basis of type measurements of specific products. The following figure shows the change in the insertion loss of an absorber muffler as a function of frequency for a real (stochastic) incident sound propagation direction distribution. The (classical) dissipative losses due to fluid friction and thermal conduction during sound propagation in a porous medium are proportional to the square of the frequency, so that at low frequencies the insertion loss of the absorber silencers is small. At high frequencies, the wavelength is much smaller, than the typical size of the muffler channels, so less sound deflection occurs, the sound waves “radially overshot” the free channel cross-sections, which causes a reduction in the insertion loss. Between low and high frequency ranges, in the medium frequency range, important for mechanical noise protection, absorber silencers are effective broadband noise reduction devices.



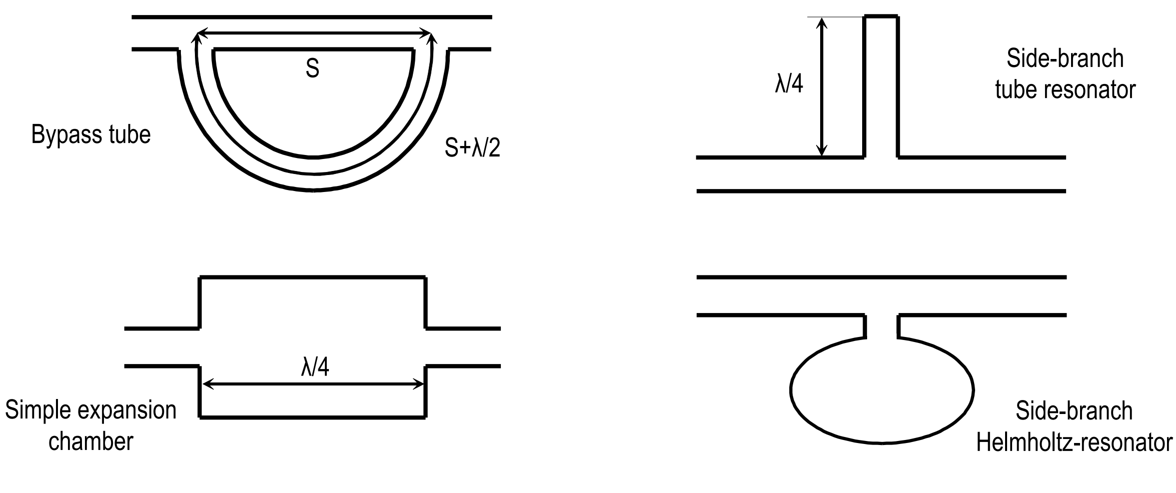
Typical sections of the absorber type muffler insertion loss as a function of frequency in the case of a stochastic incident sound distribution

The basic types of absorber mufflers are coulisse and ring shaped tubular mufflers, their schematic drawing is shown in the following figure. In order to increase the propagation losses, the absorbing layer is placed not only on the inner wall of the channel, but in the case of a square cross-section channel into internal plates (coulisse) or in the case of a circular cross-section channel into a core tube placed in the symmetry axis.



Schematic drawing of ring and coulisse type absorber silencers

Reactive mufflers: Reactive mufflers reduce the propagation of sound by creating reflections, destructive interference, and in some cases increasing dissipative loss during resonant behaviour. In the case of reactive silencers, the geometric design and the wave acoustic behaviour of the sound are of paramount importance. Basic types are the expansion chamber, side-branch Helmholtz and tube resonators, and bypass (Quincke) tube, their schematic drawing is shown in the following figure.



Schematic drawing of simple reactive silencers

Reactive mufflers are generally effective in a narrow frequency range (or ranges). Depending on the sizing, they can achieve high insertion losses even at low frequencies too. If the noise reduction task is to be performed on several discrete frequencies or frequency ranges, the required installation loss can be created by combining the basic types appropriately. The resulting installation loss of complex silencers made up of basic types cannot be determined by simple network calculation methods.

14.3. Test questions:

T.Q.1. Using a simple energetic acoustic model, deduce the formula of the insertion loss of the noise protection hood. For the solution, make a drawing, describe the simplification conditions and the application limitations of the relation. Based on the derived relation, explain the needed wall layer structure to make an effective noise protection!

T.Q.2. List the most important types of duct silencers!

T.Q.3. Recommend a duct muffler type for ventilation systems in the case of broadband noise emissions or to reduce the exhaust noise of a fixed speed internal combustion engine.
