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EVALUATIONS OF SILENCERS

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Abstract

In most cases HVAC equipment has to be provided with silencers. In compliance with development of technology most buildings are now currently equipped with air conditioning systems which require large sections for distribution ducts with built-in silencers. At present this creates great pressure on a minimization of the overall size of silencers in ducts. The present paper deals with mutual relations between acoustic requirements on silencers and determination of their hydraulic resistances. In conclusion the paper presents a concept of the evaluation of silencers by means of a so-called quality factor which essentially covers both their acoustic and hydraulic properties.

INTRODUCTION

A number of engineering facilities produce noise, which, in areas inhabited by people, is assessed as excessive according to the health and safety regulations. If noise is being generated in a protected area from a duct system it is possible to ensure the necessary noise reduction by inserting a silencer. The silencer should not however place a disproportionately high constriction on the flow of the flowing liquids. In essence, a silencer in a duct is a channel or conduit placing resistance to noise transferral. When constructing a silencer, or rather during its design in the course of planning, it is essential to take into consideration the basic conditions that will ensure its successful application in the engineering facility:

A. *Acoustic requirements*, requiring the level of the sound power in the individual frequency bands to be depressed. Most often it works with a spectrum of soud power level in octave or 1/3 octave bands, which is controlled in various locations on the duct. Very often the term used for reducing noise is insertion loss. In some cases sound transmission loss is used.

In essence the acoustic properties of silencers can be expressed in a dual manner. There is the term *transmission loss*, which is defined as the difference in the sound power level before silencing L_{W1} [dB] and after silencing L_{W2} [dB]

$$D = L_{W1} - L_{W2} \tag{1}$$

Transmission loss itself has the effect of retuning the ducting system before silencing, which can cause a different acoustic result after silencing.

Insertion loss is given by the difference in the sound power level in the same place on the duct before installing the silencer L_{W1} [dB] and afterwards L_{W2} [dB]. For air-conditioning equipment the term insertion loss is used because this gives the designer the surety of an expected noise reduction by installing the chosen silencer.

B. *The aerodynamic requirement*, which requires the silencer's hydraulic resistance to be brought down to a level that does not significantly effect the actual engineering equipment's working. Usually it is assigned a maximum permissible pressure drop Δp_{max} [Pa] established for the specified flow rate of the liquid. There is a different permissible pressure drop for air-conditioning equipment and for exhausts form a combustion engine.

C. *The geometric requirement,* takes into account the spatial potential for building a silencer into the duct system. This is not just a matter of the overall silencer volume, but above all the maximum permissible cross-section or length of the silencer.

D. *The requirement of the resilience of the environment,* in which the silencer will work. The difference is obviously that between a silencer built into air-conditioning equipment, through which clean air flows, and a silencer that has to reduce the noise generated by combustion equipment releasing into a smoke flue.

E. An indispensable requirement is the *question of silencer maintenance*, if it is exposed to the action of impurities, e.g. in exhaust systems for woodworking machines, where the threat of clogging by impurities arises.

F. *The price requirement* is now an essential parameter during silencer design. It is linked to the conditions mentioned above. The crucial factor is the necessity of using a certain quality of materials that are resistant to corrosion.

Silencers can be divided up, according to the principle of their function, into resonator-type silencers and absorption-type silencers. The principle difference between these silencers is the aspect of disseminating the acoustic energy through the duct system. Where a resonator-type silencer is installed it reflects the acoustic energy back to the source. This means that the acoustic energy is not converted into heat energy but partial standing waves are created in the duct system between the silencer and the noise source, which can decrease the machine's efficiency. A typical example of using resonator-type silencers are piston combustion engines whose silencers must be tuned so that the removal of exhausts from the machine's combustion centre is not deleteriously affected

Absorption silencers are distinguished by the fact that a predominant part of the demonstrable silencing is brought about by changing acoustic energy into heat energy. This transferral occurs in the layers of the absorption material, the quality of which is the basic assumption for successful silencer design. The majority of silencer users only assess the silencer used according to its insertion loss.

ABSORPTION SILENCERS

The simplest construction for the construction of absorption silencers is based on using materials suitable for absorbing sound, which are placed on the duct walls, as shown in *Figure 1*.

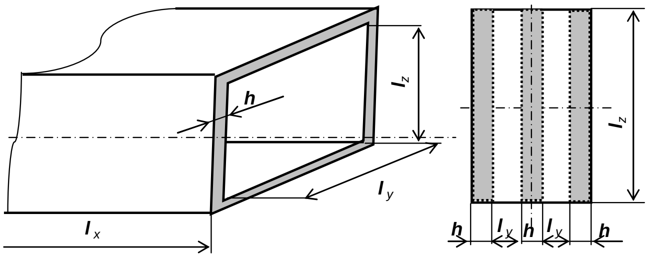


Figure 1 – An absorption silencer with lined walls (a practical solution for baffle-type silencers)

Absorption silencers' noise damping must be divided up into two basic frequency bands. If the lateral dimensions of the air conduit are more than half the wavelength of the transmitted noise signal ($\lambda > l_y$), then a plane wave will expand along the duct. For this case the valid computational relationship is one, which can be used to express the noise attenuation

$$D = 1{,}1\,\alpha \frac{O}{S} l_x \tag{2}$$

Where α [-] is a factor of the wall material's absorption, O [m] is the wetted perimeter of the duct diameter, $O = 2\,(l_y + l_z)$, S [m^2] is the duct cross-section, $S = l_y\, l_z$, l_x[m] is silencer length.

This solution is most commonly used in practice. Only the longer sides of the duct are lined i.e. within the duct cross section additional absorption baffle-type silencers are inserted as documented in *Figure 3*. For example where the dimension l_z is markedly larger than the gap between the absorption baffle-type silencers l_y, formula (2) can be simplified to

$$D = 2{,}2 \frac{\alpha}{l_y} l_x \tag{3}$$

It follows from the last relationship that there is an unambiguous hyperbolic dependence of noise attenuation in the absorption silencer to the width of the gap l_y between the absorption boards. The absorption coefficient α is a frequency dependent quantity as shown in the diagram for porous materials in *Figure 2*.

It also follows from the diagram that there is a dependence on the thickness of the absorption material h [mm]. It is generally true that the variable for porous materials' absorption has low values in bands with low frequencies. This is all projected onto the development of the absorption silencer's insertion loss. The use of absorption layer material with a thickness h is recommended so that at the lowest frequency $f_{\min}$ the silencer demonstrates an attenuation of 3 to 5 dB/m.

The change in the diameter of the flow channel due to the inserted absorption baffle-type silencers, as shown by the scheme for the silencer in *Figure 3*, gives rise to a partial reflective attenuation, which adds to the absorption attenuation. For this reason there is an increase in the insertion loss especially in the case of narrow channels between the baffle-type silencers and when using wider baffle-type silencers as documented by the diagram in *Figure 4*, which is drawn for multiple attenuation with a gap width of $D_1.l_y$ [dB.mm/m]. For longer silencers the effect of sudden changes in the channel's diameter does not play such a large role.

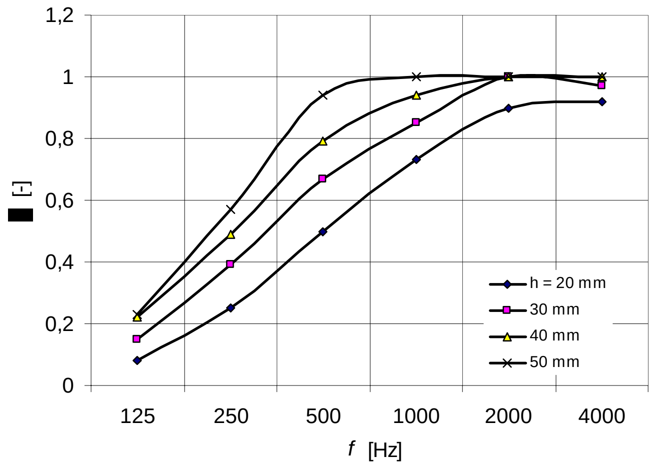


Figure 2 – Frequency development for the absorption variable of ISOVER SSP2 material

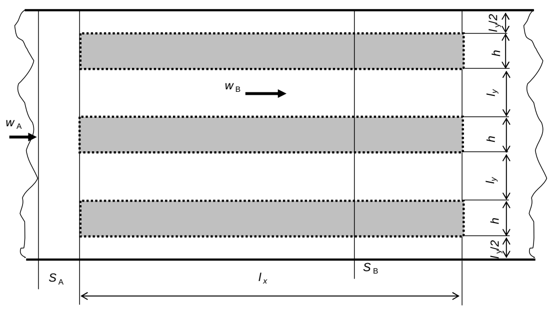


Figure 3 – Scheme for an absorption silencer without leading and trailing metal

In the band of higher frequencies the relationships given above do not give such reliable results because the condition for spreading flat waves is not maintained. If the wave length is comparable or greater than the transverse dimension of the channel, the waves begin to spread by variously directed rays (modes).

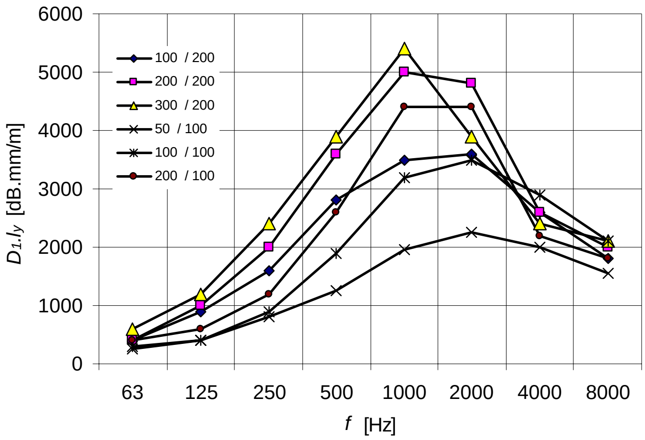


Figure 4 – Insertion loss for a silencer of length m and various built in dimensions for the parameters ly/h, correlated to the gap width ly

From this frequency the attenuation of the absorption silencers begins to show a marked drop. According to Cremer [3] it is possible to set a noise attenuation for which $\boldsymbol{f} \gg \boldsymbol{c}/\boldsymbol{l}_y$, from the relationship

$$D = 4{,}35(n+1)^2 \frac{c^2}{l_y^3 f^2}\Theta \tag{4}$$

Where Θ [-] is the real component of the material's relative acoustic impedance, n [-] is the mode number (1,2,3,…), c [m/s] is the speed of sound in air.

It follows from the information presented here that for absorption silencers to dampen in low frequency bands the attenuation is primarily decided by the thickness of the absorption material h and the gap width l_y. In bands with higher frequencies the effect of gap width l_y is decisive. To estimate the specific damping $D_1.l$y it is possible to use the diagram in *Figure 4*.

The problem of noise from air-conditioning equipment using radial ventilators is usually focussed on the frequency band 63 to 500 Hz. This is elicited by the shape of the spectrum of the acoustic output level from the noise source, which is, on the whole, a radial or axial ventilator.

OPTIMIZATION OF THE SILENCER

Concerning the facts presented above the question arises as to how to meet the basic requirements for designing a silencer, which has high noise attenuation, low hydraulic resistance and minimum dimensions (apart from the other requirements presented in the introduction to this article).

The overall attenuation in a certain octave band can be expressed as a function of a silencer's specific attenuation D_1 (dB/m) and the silencer length l_x (m)

$$D = D_1\ l_x \tag{5}$$

The silencer's maximum permissible hydraulic resistance Δp_{max} [Pa] is usually determined as the local pressure drop according to the relationship

$$\Delta p_{\max} = \xi_1 l_x \frac{w_A^2}{2} \rho = \xi_1 l_x \frac{V^2}{S_A^2} \frac{\rho}{2} \tag{6}$$

Where ξ_1 [1/m] is the coefficient of the specific pressure drop, l_x [m] is silencer length, w_A [m/s] is air speed in the cross-section S_A, V [m^3/s] is air flow, S_A [m^2] is the silencer's flow cross-section.

It is necessary to note that the pressure drops can be expressed either for speed in the diameter S_A or S_B see *Figure 3*. From the standpoint of assessing the silencer's size the authors prefer the calculation for diameter S_A.

On the basis of the scheduled necessary noise attenuation in the given critical octave band, the air flow and the silencer's maximum permissible pressure drop, the necessary diameter of the silencer can be determined by solving equations (5) and (6). The silencer diameter is determined from the relationship

$$S_A = \sqrt{\frac{\rho}{2}} \sqrt{\frac{D}{\Delta p_{\max}}} \sqrt{\frac{\xi_1}{D_1}} = \sqrt{\frac{\rho}{2}} \sqrt{\frac{D}{\Delta p_{\max}}} \frac{1}{K} \tag{7}$$

Through a breakdown of relationship (7) we arrive at the finding that the quality of the silencer for the given frequency range can be ascertained according to the value of parameter K, which can be called the silencer quality factor

$$K = \sqrt{\frac{D_1}{\xi_1}} = \sqrt{\frac{D}{\xi}} \tag{8}$$

Where D [dB] is the silencer's attenuation in the critical frequency band, ξ [-] is the overall coefficient of the silencer's local pressure drop.

The ability of the silencer to dampen noise with minimum pressure drops clearly follows from the relationship.

BREAKDOWN OF THE PRESSURE DROPS

It is quite clear that the same silencer (same baffle-type silencer width and gap between the baffle-type silencers) at various lengths gives differing values for the quality factor. This is caused by the partial reflection attenuation and, for pressure drops, by the existence of a pressure drop due to sudden contraction and sudden expansion. The overall real attenuation of a baffle-type silencer can be expressed as the sum of the reflection attenuation D_{reflex} (it only occurs at each silencer the once) and the absorption attenuation D_{absp}, which is a function of the silencer length

$$D = D_{reflex} + D_{1absp} l_x \tag{9}$$

Similarly the overall coefficient of the silencer's local pressure drop ξ can be determined as the sum of the local pressure drops ξ_m (sudden expansion, sudden contraction) and the effect of pressure drops through friction.

$$\xi = \xi_m + \frac{\lambda\, l_x}{d_{ekv}} \tag{10}$$

Where d_{ekv} [m] is the equivalent diameter of the corresponding average flow cross section between the baffle-type silencers

$$d_{ekv} = \frac{2 l_y l_z}{l_y + l_z} \tag{11}$$

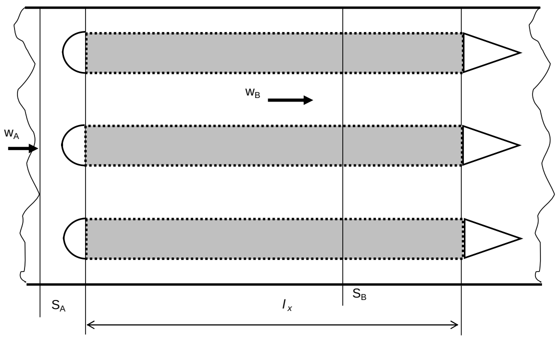
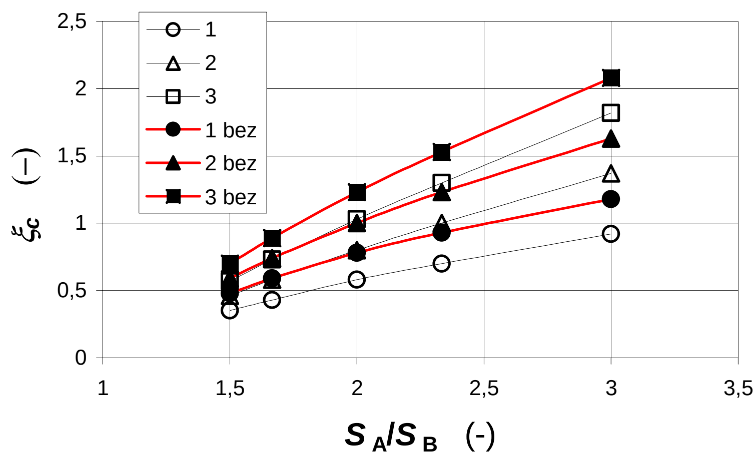


Figure 5 – Baffle-type silencer with leading-edge metal

With regards to the great importance of pressure drops for baffle-type silencers experiments were carried out to clarify the proportion of local drops with the entry and exit of air from the silencer at sudden changes in the cross-section and pressure drops due to friction. At the same time these parameters were checked for silencers, which are aerodynamically modified at the leading and trailing sides of the baffle-type silencer see *Figure 5*.



Fiigure 6 – Overall coefficient of the pressure drop for baffle-type silencers

In *Figure 6* the overall pressure drop coefficients are compared, these are divided into two sections according to relationship (10). The results given here indicate the expedience of using leading-edge metal. For small silencers (for small airflows) the design of the optimal silencer cross-section is not so crucial. In contrast for silencers with a large air flow it is possible to make savings not only in the investment costs, but also in the ventilators' electricity usage. In these situations it pays to design untraditional aerodynamic measures at the input and output of the silencer. The M 200/40 silencer can be used as an example; during its design emphasis was placed on minimising pressure drops during the air's through flow (see *Figure 7*).

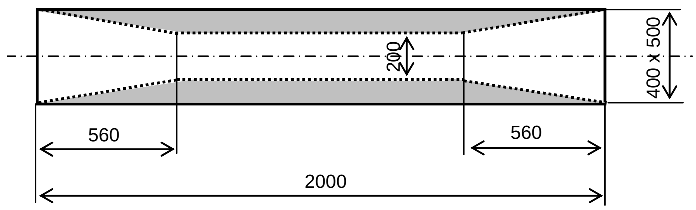


Figure 7 – The M 200/40 silencer

In conclusion it is possible to make a comparison of the silencers presented here from the aspect of the quality factor *K*, which is summarised for the most important frequency bands in *Table 1*.

Table 1 Comparing the quality factor for baffle-type silencers

Silencer	63	125	250	500
200/100 from fig. 6	1.36	1.67	2.36	3.47
200/100 from fig. 10	1.59	1.95	2.76	4.06
M200/40	1.754	2.11	3.093	3.604

CONCLUSIONS

The published computational relations enable planners to make better decisions during silencer design. On the basis of the values in the tables it is possible to arrive at an optimisation of the silencer from the standpoint of the overall volume and consequently the price of the silencer. The presented results of research were obtained with the support of Grant MSM 6840770011.

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