A NEW TYPE OF COMPACT SILENCER FOR HIGH FREQUENCY NOISE
Kabral, R., Auriemma, F., Knutsson, M., Åbom, M.

Abstract: In modern IC engine design super-chargers are utilized to increase the fuel conversion efficiency. Nevertheless, these components are also recognized as strong high frequency noise sources in the engine compartment. For installations under such limited space and high sound pressure conditions innovative noise control concepts are essential. To reduce this type of noise a new type of silencer based on micro-perforated plates and optimized using the so called Cremer’s acoustic impedance is proposed and investigated experimentally. The experimental data is also used to validate modelling done on the new silencer. Key words: Compact silencer, Micro-perforated, Acoustic impedance, Super-Charger.

1. INTRODUCTION
Since 2015 the vehicle manufacturers have been obliged to implement engines complying the Euro 6 emission standard [1]. To this aim, the super-charging of the engine is almost un-avoidable, as it increases the indicated fuel conversion efficiency. On the other hand, additional concerns related with high frequency noise generation, will arise from compressors.

Traditional solutions to reduce intake noise of the IC engine are based on the well-known Helmholtz resonator, which reflects sound generated back to the source. Moreover, sound reflections also occur at the opening of the duct termination (See e.g. [7] or [9]). Therefore, in such solutions, the noise dissipation relies on the source ability to absorb the reflected noise.

To overcome this dependence, and to design a robust noise control solution, the sound has to be dissipated by properly designed absorptive elements. Traditionally, dissipative silencers are based on fibrous materials, which pollute the medium and, when integrated to the air inlet, can cause failure of the IC unit.

Therefore, producing acoustic absorption with non-fibrous materials is of interest. The idea of producing acoustic resistance in room applications by employing the viscosity in circular small apertures (in order of acoustic boundary layer) is originated from D. Y. Maa [4]. Usually, producing such acoustic elements is a time consuming and expensive process. These issues can be overcome by using mass produced sound absorbing panels called Acustimet™ [5]. These panels were studied also as one possible noise control solution in vehicle applications (See e.g. [6]). In addition, a new type of silencer based on the micro-perforated panels (MPP) was proposed and studied by Allam and Åbom in [7]. In order to eliminate the drawback of having transmission loss minima at half-wave length multiples, the locally reacting limit was formulated by Åbom and Allam in [8]. The proposed design consists of straight-flow channel made of MPP that is adjoined to locally reacting cavity, thus resulting in a locally reacting surface.

Kabral et. al. proposed the optimization technique for such compact silencer in the [9]. The essence of the method is matching the acoustic wall impedance with the Cremer optimum impedance [10]. The latter was derived to obtain the highest sound damping in an infinite channel. The
concept was developed further by Tester in [11], who derived the expression for the circular cross-section, and also added the plug-flow correction as follows:

\[
Z_{Cc} = (0.88 - 0.38i) \frac{kr}{\pi(1 + M)^2}, \quad (1)
\]

\[
Z_{Cc} \quad \text{-- the norm. surface impedance for theor. maximum sound damping in a circular duct;
}
\]

\[
k \quad \text{-- the wave number, m}^{-1};
\]

\[
r \quad \text{-- the radius of the channel, m;}
\]

\[
M \quad \text{-- the Mach number.}
\]

The investigation in [9] was carried out by employing simplified FEM model where the cavity and perforation of the compact silencer were defined as an acoustic impedance boundary condition. The model was been validated with hi-resistance configuration under no flow conditions. The results indicated that, by implementing this optimization technique, very high controlled sound damping is achievable.

In the present work, a systematic validation of the modelling, and a detailed investigation of the sound damping mechanisms, will be performed. The same prototype cavity, used in [9], will be implemented in configuration with three different Acustimet™ MPPs (see Fig. 1). In addition, the same simplified FEM model is employed to compute acoustic quantities corresponding to the Cremer’s optimum.

2. METHOD

In order to evaluate if the MPP and cavity combination is optimal, the acoustic impedance of the absorbing surface has to be determined and compared to the Cremer’s optimum. In the compact silencer this impedance is formed by contributions from MPP and the locally reacting cavity as

\[
Z_{surf} = Z_{MPP} + Z_{cav}, \quad (2)
\]

where

\[
Z_{surf} \quad \text{-- a normalized surface impedance;}
\]

\[
Z_{MPP} \quad \text{-- the normalized MPP impedance;}
\]

\[
Z_{cav} \quad \text{-- the normalized cavity impedance.}
\]

The geometrical parameters are known in case of two of the three MPP Acustimet included in the investigation (Tab. 1). As a consequence, the acoustic transfer impedance can be determined by means of existing semi-empirical models. On the other hand, exact parameters of the Acustimet panel with largest apertures are unknown, except the thickness, which is 1mm for all the panels. Therefore, the acoustic transfer impedance of this panel will be determined experimentally.

<table>
<thead>
<tr>
<th>Name</th>
<th>Perforation Ratio</th>
<th>Slit width [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Res. 0.05</td>
<td>Unknown</td>
<td>Unknown</td>
</tr>
<tr>
<td>Res. 0.25</td>
<td>6.5</td>
<td>0.240</td>
</tr>
<tr>
<td>Res. 1.50</td>
<td>4.3</td>
<td>0.095</td>
</tr>
</tbody>
</table>

Table 1. Geometrical parameters of Acustimet MPP’s.

2.1 Acoustic impedance models

The comprehensive overview of the existing semi-empirical models for MPP transfer impedance has been given by Guo et al. in [7]. The resulting model of slit type MPP, utilized also herein, was validated experimentally by testing different Acustimet™ panels. The acoustic resistance \( Re(Z) \) and reactance \( Im(Z) \) of Acustimet MPP is being computed as:
\[ r_s = \text{Re} \left\{ \frac{i \omega t}{\sigma c} \left[ 1 - \frac{\tanh(k \sqrt{i})}{k \sqrt{i}} \right]^{-1} \right\} + \frac{4 R_s}{\sigma \rho c} + \frac{|u_h|}{\sigma c} + \frac{\beta M}{\sigma} \] 

(3)

and

\[ x_s = \text{Im} \left\{ \frac{i \omega t}{\sigma c} \left[ 1 - \frac{\tanh(k \sqrt{i})}{k \sqrt{i}} \right]^{-1} \right\}, \] 

(4)

where

\[ \omega \] – a radial frequency, s\(^{-1}\);
\[ \rho \] – the density of the medium, kg/m\(^3\);
\[ k \] – shear wave number, m\(^{-1}\);
\[ t \] – the thickness of perforated panel, m;
\[ \sigma \] – the porosity of the perf. surface;
\[ \beta \] – a factor for grazing flow effects;
\[ u_h \] – peak particle velocity in apertures, m/s; and
\[ R_s \] – surface resistance, Pas/m.

The shear wave number which is used to relate the acoustic boundary layer thickness with the dimensions of the aperture, is defined as:

\[ k = d \left( \frac{\omega}{4 \nu} \right)^{1/2}, \] 

(5)

where

\[ d \] – the slit width, m; and
\[ \nu \] – the kinematic viscosity of medium, m\(^2\)/s.

The consequence of the oscillating motion of the fluid on the perforated surface is the increase of the acoustic resistance, which contribution is given by \[^6\]:

\[ R_s = \frac{1}{2} \sqrt{2 \rho \omega \eta}, \] 

(6)

where

\[ \eta \] – the dynamic viscosity of the medium, kgm2/s.

The equation for cavity impedance, implemented herein, was derived in \[^8\] as:

\[ Z_{cav} = \frac{i}{H_0^{(1)}(ka_r) - H_1^{(1)}(ka_r) R_0^{(2)}(ka_r)} \left[ \frac{H_1^{(1)}(ka_r) - H_1^{(2)}(ka_r) R_1^{(2)}(ka_r)}{H_1^{(2)}(ka_r) R_1^{(2)}(ka_r)} \right], \] 

(7)

where

\[ ka \] – the axial wave number, m\(^{-1}\);
\[ R \] – the radius of the expansion chamber, m;
\[ r \] – the radius of main duct, m;
\[ H_m^{(n)} \] – the Hankel function of n:th kind and m:th order.

### 2.2 Experiments

The experimental procedures in the present work is considering plane wave 0\(^{th}\) duct mode and assuming time dependency of \(\exp(i\omega t)\). Consequently, the well-known acoustic two-port model \[^{12}\] for flow-duct elements is appropriate for the investigation.

Depending on the selection of acoustic state variables, the linear relation of the states between the ports are given though either acoustic scattering (Eq. 8) or transfer matrix (Eq. 9) \[^{12}\]:

\[
\begin{bmatrix}
    p_{a+} \\
    q_{a+}
\end{bmatrix}
+ i
\begin{bmatrix}
    T_{11} & T_{12} \\
    T_{21} & T_{22}
\end{bmatrix}
\begin{bmatrix}
    p_{a-} \\
    q_{a-}
\end{bmatrix}
=
\begin{bmatrix}
    r_a & T_b \\
    T_a & R_b
\end{bmatrix}
\begin{bmatrix}
    p_{b+} \\
    q_{b+}
\end{bmatrix}
\]

(8)

\[
\begin{bmatrix}
    p_{a'} \\
    q_{a'}
\end{bmatrix}
+ i
\begin{bmatrix}
    T_{11} & T_{12} \\
    T_{21} & T_{22}
\end{bmatrix}
\begin{bmatrix}
    p_{a''} \\
    q_{a''}
\end{bmatrix}
=
\begin{bmatrix}
    r_{a'} & T_{b'} \\
    T_{a'} & R_{b'}
\end{bmatrix}
\begin{bmatrix}
    p_{b'} \\
    q_{b'}
\end{bmatrix}
\]

(9)

where \(p_{a+}\) and \(p_{b-}\) are complex acoustic pressure wave amplitudes at port a and b of the first and second set of state variables. The subscript + and – indicate the propagation direction; \(R\) and \(T\) are complex reflection and transmission coefficients. \(p_{a'}\) and \(q_{a'}\) are total acoustic pressure and acoustic volume flow at port a and b of first and second set of state variables.

By considering the number of unknowns in these matrixes, and by assuming not
symmetric setup, two sets of linearly independent state vectors have to be experimentally determined. While the elements of scattering matrix are straightforward description of the wave interaction problem, the transfer matrix formulation is more appropriate for estimation of transfer properties, e.g. acoustic impedance. Moreover, one can be obtained from the other one by linear transformation of the state vectors. In order to determine the necessary state vectors of two-port, the acoustic pressures at both branches are measured with duct wall mounted microphones. Finally, the wave decomposition is carried out according to technique described in [13]. For perforated elements, whose thickness is much smaller than the acoustic wave length, the air inside the apertures can be considered as lumped mass. In this case it can be shown that the transfer matrix elements (Eq. 9) become \( T_{11} = 1, T_{21} = 0, T_{22} = 1 \) and \( T_{12} = Z_{MPP} \rho_0 c/A \).

2.2 Performance parameters

The most common quantity used to evaluate the acoustic performance of a silencer is sound transmission loss (TL). The TL can be interpreted as the loss of acoustic power in sound transmission through the two-port element. This can be obtained from the transmission elements of the scattering matrix (Eq. 8) according to:

\[
TL = 10 \log_{10} \left( \frac{1}{|T|^2} \right),
\]

The Eq. 10 is valid in case of same cross-sectional area in both ports \((a \text{ and } b)\), negligible flow pressure loss and temperature gradient.

The transmission loss is produced by the absorption and reflection of incident waves. The latter is not favorable, in terms of robust noise control solution, since it relies on the ability of the source to absorb the reflected waves. Hence, the TL alone is not sufficient to evaluate the silencers acoustic performance.

In addition to the TL, the absorption coefficient spectra are commonly studied to evaluate the silencer ability to absorb sound. This can be computed by utilizing the scattering matrix elements and normalizing the input sound power to 1W as:

\[
A = 1 - |R|^2 \frac{(1 - M)^2}{(1 + M)^2} - |T|^2, \quad (11)
\]

The absorption coefficient provides insight of how much incident sound power is absorbed. This is not adequate to evaluate the performance of the absorbing element of the silencer as soon as \( R \) becomes not negligible. In addition, the absorption coefficient is computed in linear domain while the sound perceived by the receiving person is in logarithmic scale.

Therefore, for optimization purposes, the sound actually entering the silencer has to be considered. This can be done by computing sound absorption (SA) instead of absorption coefficient as:

\[
SA = 10 \log_{10} \left[ 1 - |R|^2 \frac{(1 - M)^2}{(1 + M)^2} \right],
\]

Also the sound reflected can be obtained in similar fashion as:

\[
SR = -10 \log_{10} \left[ 1 - |R|^2 \frac{(1 - M)^2}{(1 + M)^2} \right],
\]

The set of TL, SA and SR will give detailed description of the silencer acoustic performance and quantification of dampened, absorbed and reflected sound power. Hence, the set of these logarithmic quantities is adequate to evaluate the goodness of the optimized silencer.
3. RESULTS

The normalized acoustic resistance and reactance of locally reacting surface inside the compact silencer prototype has been computed by means of Eq. 3, 4 and 7 for Res. 0.25 and Res 1.50 MPPs (See Tab. 1).

In the Fig. 3 the acoustic surface impedance of the configuration with the Res. 0.25 MPP is computed and compared to the Cremer optimum in case of 0.05 Mach mean flow condition.

The respective quantities of Res. 0.05 MPP are obtained by combining the experimentally determined transfer impedance with the Eq. 7 according to Eq. 2. These results are plotted in the comparison with the Cremer’s optimum for no mean flow case in the Fig. 2.

In the Fig. 2 one can observe that, for none of the configurations, the total (Eq. 2) acoustic impedance is not matching exactly the Cremer’s optimum. Nevertheless, the prototype provided with the MPP of Res. 0.25 is matching the above optimum at around 2.5 kHz, since the delivered normalized resistance is only 0.1 lower.

However, as the grazing flow on the surface of MPP generates additional contribution to the resistance (See Eq. 3), it is expected to match the Cremer condition even better when a mean flow through the silencer is introduced.

It can be seen in the plot (Fig. 3) that, in case of 0.05 Mach mean flow, the Cremer condition is closely fulfilled.

In the following figure (Fig. 4) the set of acoustic performance quantities (See Eq. 12 and 13) are plotted for the three compact silencer configurations in no mean flow conditions.

The wide and low TL peak of Res. 1.50 configuration in Fig. 4 indicates the absorptive type of damping which can be confirmed by observing the SA curve of the same configuration. In this setup the acoustic resistance is too large, i.e. the access to the cavity is restricted and, therefore, the cavity is not efficiently utilized.

Although this type of low reflection behavior (See the SR curve in the Fig. 4) is desirable in anechoic terminations, it is not
optimum in perspective of space, and hence it will not be considered further.

The other configurations in Fig. 4 have both high TL peaks. For this reason, by looking only at the TL spectrums, it would be hard prefer one over another. Nevertheless, by studying the SA and SR spectrums, the mechanism on sound damping is revealed. One has to note that the cut-off of the peak of the configuration Res. 0.05 is believed related with the experimental difficulties to realize the completely sealed cavities. This means that even higher reflection properties are expected.

The transfer impedance of the MPPs has been determined for planar elements, whereas the plates inside prototype are bended to tubular shape. Nevertheless, the numerical results of simplified FEM model presented in [9] are validated reasonably well for both configurations in the Fig. 4.

Accordingly, this modelling technique can be efficiently utilized in the optimization process.

In the Fig. 5 it is observable that the absorption properties of the prototype configurations are unaffected and the reflection of sound has been reduced further at the resonance frequency.

4. CONCLUSIONS

In the present work a new type of compact silencer was proposed and experimentally investigated. In addition, the numerical model of this silencer has been validated.
It was shown that the sound transmission loss is not adequate for the acoustic performance assessment of the compact silencer.

To get better insight of the sound damping mechanism, another set of quantities, describing the sound power distribution, were derived and analyzed. This enabled to evaluate whether in a certain configuration the absorption is optimal.

It was confirmed experimentally that the proposed compact silencer, optimized according to the Cremer’s impedance, is a very effective solution for noise control.

5. REFERENCES


6. ADDITIONAL DATA ABOUT AUTHORS

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