

A Compact Silencer for the Control of Compressor Noise

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ABSTRACT

Current trends for IC-engines are driving the development of more efficient engines with higher specific power. This is true for both light and heavy duty vehicles and has led to an increased use of super-charging. The super-charging can be both in the form of a single or multi-stage turbo-charger driven by exhaust gases, or via a directly driven compressor. In both cases a possible noise problem can be a strong Blade Passing Frequency (BPF) typically in the kHz range and above the plane wave range.

In this paper a novel type of compact dissipative silencer developed especially to handle this type of problem is described and optimized. The silencer is based on a combination of a micro-perforated (MPP) tube backed by a locally reacting cavity. The combined impedance of micro-perforate and cavity is chosen to match the theoretical optimum known as the Cremer impedance at the mid-frequency in the frequency range of interest. Due to the high damping achieved at the Cremer optimum (hundreds of dB/m) it is easy to create a compact silencer with a significant damping (say > 30 dB) in a range larger than an octave. Both simulations and experimental tests of the novel silencer are presented based on a light duty vehicle application.

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INTRODUCTION

In the field of internal combustion (IC) engines the marketing success depends on complying with the continuously increasing environmental regulations, e.g., EURO 6 [1], and fulfilling the market expectations for fuel efficiency and vehicle performance. Therefore, the industry main focus today is on developing more efficient IC engines side by side with other energy saving solutions, e.g., automatic engine start-stop or brake energy regeneration systems.

It is a well-known fact that in order to increase the IC engine efficiency, one approach is to increase the charge air pressure which results in a better fuel conversion efficiency and specific power of the IC unit while keeping the low level of nitrogen emission. There are two main types of such devices that can increase the charge air pressure: one which is mechanically driven by the IC engine itself or by an additional electric motor, and another which uses a turbine to extract otherwise wasted energy from the hot high pressure exhaust gases. On the modern IC engine, both solutions are commonly implemented for covering the high pressure charge air delivery demand in a wide mass flow range while ensuring a good transient response.

The expected dominating sound when such rotating devices are used, is the blade passing frequency tone (BPF) and its multiples. If in the audible range, such tones can be highly annoying. This can either happen for larger compressors on heavy duty vehicles or light duty vehicles when a combination of a directly driven compressor, for moderate load conditions, and a turbo-charger, for high power output operation, is used. In the latter case tones in the audible frequency range are caused by the compressor often of Roots type.

In order to address these noise issues, relatively robust and lightweight compact silencer solutions are needed. As a starting point to find such a silencer, the expansion chamber muffler with a micro-perforated sheet tube proposed by Allam and Åbom in Ref. [2] will be used here (See Fig. 1). It was demonstrated in Ref. [2] that such a configuration can provide a non-fibrous alternative to the traditional dissipative silencers, with the drawback of having transmission loss (TL) minima at half wave-length multiples of the cavity length.

The effects of mean flow and high sound pressure levels causing non-linear effects on the micro-perforate impedance were also investigated by Allam and Åbom in Ref. [2]. The modelling in Ref. [2] was based on FEM neglecting convective

flow effects but including flow effects for the micro-perforate impedance, which is reasonable for low Mach-numbers and will also be used here. The micro-perforate impedance was used as a boundary condition between the two domains, main duct and the outer cavities, see Figure 1. In order to eliminate the minima at half-wave length multiples of the cavity length, a solution in the form of a locally reacting limit was recently suggested by Åbom and Allam [3]. Also two different MPPs were considered, and it was concluded that higher damping of the sound is achievable when the acoustic resistance is smaller than the optimum for room acoustic applications. The conclusion being that one cannot naively use standard MPPs for a silencer, which are optimized for applications involving 3D sound fields in rooms [4].

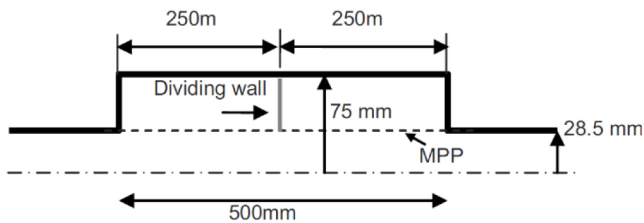


Figure 1. Geometry for the MPP muffler proposed in Ref. [2]. Circular symmetry is assumed and the outer chamber is sub-divided into two or several cavities by rigid walls.

In the present work, the dissipative silencer concept proposed in Refs. [2-3] is further studied in order to find an optimum design. Based on the proposed optimization a design for a Compact Silencer for a light-duty vehicle charger problem is considered. The design constraints for this case are summarized in Table 1 and Figure 2.

Table 1. Design constraints for the compressor noise problem studied.

Parameter	Inlet/outlet duct radius, m	Average Mach number	Temperature, °C	Target frequency, kHz
Value	0.03	0.05	20	2.0

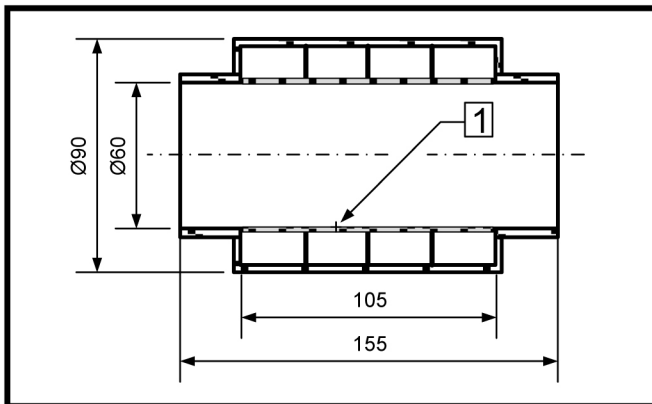


Figure 2. Geometry of the prototype MPP silencer studied. The arrow, marked 1, points at the micro-perforated sheet tube (Acustimet™), separating the main duct from the outer cavities. Note all outer dimensions [mm] except the chamber diameter are fixed during the optimization.

A photo of the prototype built by using rapid prototyping is presented in Figure 3.

MODELING

First a prototype silencer (See Fig. 3) will be evaluated. This silencer was built ensuring local reaction in the outer cavities in the frequency range of interest, but *not* using an optimized cavity plus micro-perforate impedance. After validating modelling procedures for this prototype an optimization will be carried out by varying the chamber diameter and the micro-perforate impedance. Through all the current work the 0th ("plane wave") duct mode with time dependence of $\exp(i\omega t)$ is assumed and air (1.21 kg/m³) with a sound speed of 343 m/s.

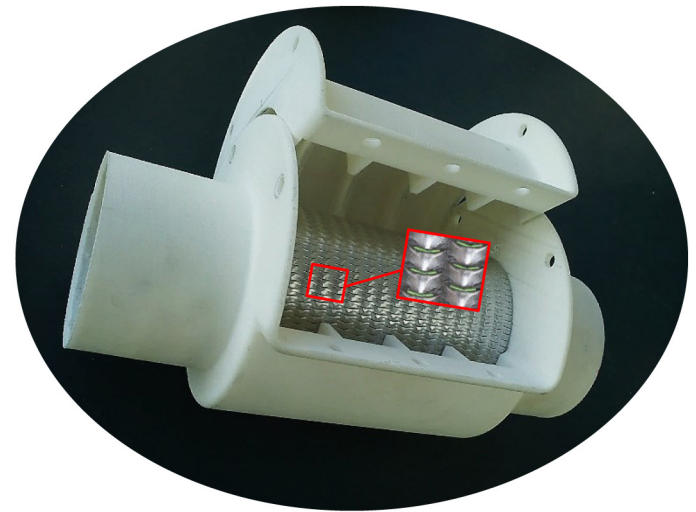


Figure 3. Photo of the compact silencer rapid prototype with magnified view of the micro-perforated (Acustimet™) sheet tube. Dimensions as in Figure 2.

Experimental Validation Data Determination

The prototype is constructed by implementing a standard MPP aluminum sheet (Acustimet™) under the constraints summarized in Table 1 and Figure 2. In order to evaluate its acoustic attenuation ability, the transmission loss was experimentally determined using a two-port test rig [5]. The obtained TL curve can be found in Figure 5.

In principle, a flow-duct component as the compact silencer treated here can be completely modeled by a finite element method (FEM) approach. Nevertheless, the relatively small aperture dimensions and complicated surface geometry of the MPP would require very fine grid elements and therefore, is computationally expensive. Moreover, since wave propagation in the cavity can be neglected, except normal to the perforated surface, its acoustic effect can be described by an impedance. Hence, a more efficient approach is a simplified model, where a locally reacting surface will be entered into the FEM model using an impedance boundary condition representing the MPP and the outer cavity.

Wall Impedance Determination

The wall impedance is a consequence of two parts a perforate (MPP) and an adjoining cavity, acting together so that:

$$Z_{wall} = Z_{MPP} + Z_{cav}, \quad (1)$$

where

Z_{wall} - a normalized surface impedance,

Z_{MPP} - the normalized MPP impedance; and

Z_{cav} - the normalized cavity impedance.

Guo et al. gave an overview in Ref. [6] on the semi-empirical modeling of micro-perforate transfer impedance. They also investigated the effect of flow and based on experiments at KTH, they suggested correction terms for the acoustic impedance under grazing flow conditions. In addition, they showed that the acoustic mass end correction term for the slit types of MPP, e.g. as the Acustimet™ panel used here, can be neglected. The acoustic resistance $\text{Re}(Z_{MPP})$ of the slit type MPP can be computed from:

$$r_s = \text{Re} \left\{ \frac{i\omega t}{\sigma c} \left[1 - \frac{\tanh(k_s \sqrt{i})}{k_s \sqrt{i}} \right]^{-1} \right\} + \frac{4R_s}{\sigma \rho c} + \frac{|u_h|}{\sigma c} + \beta \frac{M}{\sigma} \quad (2)$$

and the reactance $\text{Im}(Z_{MPP})$ from:

$$x_s = \text{Im} \left\{ \frac{i\omega t}{\sigma c} \left[1 - \frac{\tanh(k_s \sqrt{i})}{k_s \sqrt{i}} \right]^{-1} \right\}, \quad (3)$$

where

ω - a radial frequency, s^{-1} ;

ρ - the density of the medium, $\text{kg} \cdot \text{m}^{-3}$;

k_s - shear wave number, m^{-1} ;

t - the thickness of perforated panel, m;

σ - the porosity of the perf. surface;

β - a factor for grazing flow effects;

u_h - peak particle velocity in apertures, $\text{m} \cdot \text{s}^{-1}$;

R_s - surface resistance, $\text{Pa} \cdot \text{s} \cdot \text{m}^{-1}$; and

M - Mach number of grazing flow.

The shear wave number, which relates the aperture dimensions with the acoustic boundary layer thickness, is defined as:

$$k_s = d \sqrt{\frac{\omega}{4\nu}}, \quad (4)$$

where

d - the slit width, m; and

ν - the kinematic viscosity of medium, $\text{m}^2 \cdot \text{s}^{-1}$.

Finally, the surface resistance which is a consequence of oscillating motion of the fluid on the surface of a perforated panel, can be found from:

$$R_s = \frac{1}{2} \sqrt{2\rho\omega\eta}, \quad (5)$$

where

η - the dynamic viscosity of the medium, $\text{kgm}^2 \cdot \text{s}^{-1}$.

The normalized resistance (Eq. 2) is a sum of four terms which relates acoustic losses with viscous effects inside the apertures, first term, and outside, the last three terms. The non-linear effects (the second last term in Eq. 2) proportional to the velocity in the holes u_h , can be important for applications on IC-engines but will for the present study be omitted. As also seen from Eq. 2 and Eq. 3 a grazing flow will affect (increase) the resistance but not affect the reactance.

The analytical model for the acoustic cavity impedance has been derived and proposed in the earlier paper [3] and is given by:

$$Z_{cav} = \frac{i \left[H_0^{(1)}(k_a r) - \frac{H_1^{(1)}(k_a R)}{H_1^{(2)}(k_a R)} H_0^{(2)}(k_a r) \right]}{H_1^{(1)}(k_a r) - \frac{H_1^{(1)}(k_a R)}{H_1^{(2)}(k_a R)} H_1^{(2)}(k_a r)}, \quad (6)$$

where

k_a - the axial wave number, m^{-1} ;

R - the radius of the expansion chamber, m;

r - the radius of main duct, m; and

$H_m^{(n)}$ - the Hankel function of n :th kind and m :th order.

FEM Model Description

For all the FEM computations the commercial software COMSOL Multiphysics® [7] is employed herein. The wave equation solved in a lossless medium herein, is known as the inhomogeneous Helmholtz equation: