Analysis and design of a semi-active muffler

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Abstract
In this work, a flow reversal resonator fitted with a short-circuit duct connecting the inlet and outlet is analysed and used as a tuneable muffler element, aimed to be used use in a semi-active muffler on an IC-engine. The work done can be divided into 3 main parts. 1), a study of what type of valve that could be used to change the acoustical properties of the short-circuit duct. 2), Design of a flow reversal resonator with a controllable valve as the short-circuit. 3), experimental validation in a flow acoustic test rig.

The flow reversal resonator with a controllable valve as short-circuit is successfully validated to work as tuneable muffler element during laboratory conditions. The same valve concept is simulated in a full scale concept but not validated experimentally on a running IC-engine.

The theory used to describe the acoustics of a flow duct element is also presented together with three simulation techniques and the two microphone technique used to determine the acoustic properties of the investigated flow reversal resonator.
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Introduction

Background
In flow duct applications, such as e.g. IC-engine exhaust systems on cars and trucks, high sound pressure levels are a common problem. The sound is usually attenuated with a muffler in which two different types of silencing elements is used, reflective and resistive. Resistive elements are elements where the acoustic energy is dissipated as heat, such material is e.g. mineral wool and perforated plates. Reflective elements attenuate the sound by destructive interference between sound waves travelling in different directions. Some examples of these are area discontinuities and resonators like the quarter wave resonator and the Helmholtz resonator. Depending of the geometry, the reflective elements will have one or more eigenfrequencies with maximum attenuation. In this work, a reflective type of muffler, the flow reversal resonator, has been studied and modified to act as a semi-active muffler.

The Flow reversal resonator
It is shown in [1] that a flow reversal chamber may be modified with a short-circuit duct, connection the inlet and outlet, thus creating a resonator. The eigenfrequency of this resonator will change depending of the ratio between area and length of the short circuit duct, L/S, [1], if the rest of the system is left unchanged.

![Flow reversal resonator concept](image)

The lowest possible eigenfrequency for this system is when the inlet and outlet pipes are in full connection, a zero length short-circuit duct, [1]. This eigenfrequency will correspond to a Helmholtz resonator having the same volume as the flow reversal chamber and the same neck length as the distance between the flow reversal chamber and the short-circuit duct. The highest eigenfrequency is given when the short circuit is closed. For this case the eigenfrequency will be decided by the standing quarter wave in the chamber. In [1], the flow reversal resonator is proposed to be suitable for use as a semi-active muffler if the geometry of the short-circuit can be varied. The idea is to control the eigenfrequency to match the firing frequency of an IC-engine, given by

$$f_f = \frac{\text{engine speed (in rpm)} \times \text{no.of cylinders}}{60 \times 2}$$

(1)
For a six cylinder engine running between 1000 and 2000 rpm, this gives a frequency range of 50 to 100Hz.

**Muffler basics**

**Muffler performance**
The performance of a muffler is characterized using 3 different measures.

- Insertion loss, IL
- Transmission loss, TL
- Noise reduction, NR

*Insertion loss* is defined as “the ratio (in dB) between the acoustic power radiated at the outlet of a reference system and the system investigated, with both systems driven by the same source” [2] The reference is usually a straight pipe with the same length as the investigated system.

*Transmission loss* is defined as “the ratio (in dB) of the incident power to the power transmitted for a given termination” [2]. Normally a reflection free pipe.

*Noise reduction* is defined as the “difference in sound pressure level (in dB) at two arbitrarily selected points in the exhaust pipe and tailpipe” [3], (upstream and downstream side of the muffler).

In this work, transmission loss, TL, will be used as the measurement of the muffler performance.

**Multi-port representation**
A multi-port representation is a way of characterizing a duct system in the plane wave region with a matrix relation between its input and output state variables

\[ y = Gx \]  

Where \( x \) is the input state variables, \( y \) the output state variables and \( G \) is the relation between \( x \) and \( y \). These state variables are functions of time and completely define the state at the input and output. The system is also assumed to be linear, time-invariant, adiabatic and passive i.e. source free.

![Figure 2: Multi-port](image)

The plane wave region is frequencies below the first cut-on frequency for higher order modes, which for a circular duct can be calculated from

\[ f_{01}^c = \frac{1.841 \times c_0}{\pi \times D} \]

Where \( D \) is the duct diameter.
**Two-port representation**

A special case of a multi-port is the two-port which has one inlet and one outlet. This is the case for a duct system in the plane wave regime. The state variables used to represent the acoustics are acoustic pressure and particle velocity or the amplitudes of the acoustic pressure waves travelling inward and outward from the two-port. The choice of state variables determines how to mathematically describe a two-port or several in cascade. The applied experimental technique may influence the choice of state variables. When using acoustic pressure, \( p \), and volumetric flow rate, \( q \), the relation between input and output is called the transfer matrix, \( T \). The equation for the transfer matrix formulation for a passive two port element is then

\[
\begin{bmatrix}
\hat{p}_a \\
\hat{q}_a \\
\end{bmatrix} = \begin{bmatrix}
T_{aa} & T_{ab} \\
T_{ba} & T_{bb} \\
\end{bmatrix} \begin{bmatrix}
\hat{p}_b \\
\hat{q}_b \\
\end{bmatrix}
\]

With

\[
T = \begin{bmatrix}
T_{aa} & T_{ab} \\
T_{ba} & T_{bb} \\
\end{bmatrix}
\]

The transmission loss \([2]\) is then calculated from

\[
TL = 10 \log_{10} \left[ \frac{1}{4} \left( T_{aa}^2 + T_{ab} T_{ba} Z_{in} + T_{bb}^2 \right) \right]
\]

Where

\[
Z_{in} = \frac{\rho c_0}{S_{in}}
\]

\[
Z_{out} = \frac{\rho c_0}{S_{out}}
\]

Where \( S_{in} \) and \( S_{out} \) is the inlet and outlet area and the mach number is assumed to be equal at inlet and outlet.

![Figure 3: Example of a two-port duct element with variables for transfer matrix representation](image)

When using the wave amplitude for waves travelling in both directions at each port the relation between input and output is called the scattering matrix, \( S \). The equation for the scattering matrix formulation for a passive two port element is then
With

\[
\begin{bmatrix}
\hat{p}_a^+ \\
\hat{p}_b^-
\end{bmatrix} = \begin{bmatrix}
r_a & t_{ba} \\
t_{ab} & r_b
\end{bmatrix} \begin{bmatrix}
\hat{p}_a^- \\
\hat{p}_b^-
\end{bmatrix}
\]

(9)

The transmission loss is then calculated from

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

(11)

The mach number is assumed to be equal at inlet and outlet.

**Method**

In this work, several methods were used to design and evaluate a semi-active muffler. First a study of a previous work done by Mikael Karlsson and Ragnar Glav [1] was done. To evaluate a new prototype of the semi-active flow reversal resonator, different valve types were considered and evaluated by simulations and measurements. These simulation- and measurement techniques are described in this chapter.

**Simulation**

Simulations were done to predict the largest possible variation of the lowest eigenfrequency. From the bottom end frequency with open short circuit to the top end frequency occurring with the short-circuit closed. Three different techniques were used, lumped model simulation, transfer matrix simulation in SIDLAB and FEM using commercial code COMSOL. They are all different in both accuracy and the demand of the computational power.

**Lumped model**

The flow reversal resonator was modelled using a transfer matrix representation. In [1], the transfer matrix for the flow reversal resonator is derived from the acoustic impedance of each element. The transmission loss is then calculated from the elements in the transfer matrix and the input and output impedance, see Eq. (12) to (27). This simulation technique is fast with a calculation time below one second.
Figure 5: The flow reversal resonator with variables used in the lumped model

\[ Z_s = \frac{p_a - p_b}{q_s} = \frac{i\omega p L_s}{s_s} \]  
\[ Z_1 = \frac{p_a - p_c}{q_1} = \frac{i\omega p L_1}{s_1} \]  
\[ Z_2 = \frac{p_c - p_b}{q_2} = \frac{i\omega p L_2}{s_2} \]  
\[ Z_c = \frac{p_c}{q_1 - q_2} = \frac{\rho c^2}{i\omega V_c} \]  
\[ q_a = q_s + q_1 \]  
\[ q_b = q_s + q_2 \]  
\[ \begin{bmatrix} \dot{p}_a \\ \dot{q}_a \end{bmatrix} = \begin{bmatrix} T_{aa} & T_{ab} \\ T_{ba} & T_{bb} \end{bmatrix} \begin{bmatrix} \dot{p}_b \\ \dot{q}_b \end{bmatrix} \]  
\[ T_{aa} = \frac{BD}{A} - \frac{Z_1}{Z_2} \]  
\[ T_{ab} = \frac{CD}{A} - D \]  
\[ T_{ba} = \frac{B}{A} \]  
\[ T_{bb} = \frac{C}{A} \]  
\[ A = Z_s Z_c + Z_1 Z_2 + Z_2 Z_c + Z_c Z_3 \]
\[ B = Z_2 + Z_c + Z_1 \]  
(24)

\[ C = Z_1 Z_c + Z_s Z_2 + Z_1 Z_c + Z_2 Z_c + Z_1 Z_c \]  
(25)

\[ D = Z_1 + Z_c + \frac{Z_c Z_2}{Z_2} \]  
(26)

\[ TL = 10 \log_{10} \left[ \frac{Z_{out}}{4 Z_{in}} \left( T_{in} + \frac{T_{ab}}{Z_{out}} + T_{ba} Z_{in} + \frac{T_{bb} Z_{in}}{Z_{out}} \right)^2 \right] \]  
(27)

**Transfer matrix simulation in SIDLAB**

SIDLAB is a commercial software used to model the propagation of low frequency sound in duct networks. It is based on the two-port theory using transfer matrix approach. This software is based on predefined muffler elements which can be connected in an arbitrary way. It is also possible to define elements by experimental determination of the two-port, this is something which was used in the later stage of this work. SIDLAB is easy to use with its drag and drop function when constructing a duct network and the calculation time is around a couple of seconds. Simulations can also be done with a mean flow present but this was not done in the current work.

**FEM simulation in COMSOL**

FEM simulations were done in COMSOL multiphysics acoustic module. A normal sized physics-controlled mesh was used giving around 16000 elements. A pressure acoustic model was used with sound hard walls as boundary conditions. The transmission loss was calculated from the ratio, in dB, of incident to transmitted power, calculated from surface integration over the input and output duct. See Eq. (26).

\[ W = \oint \iota dS \]  
(28)

\[ TL = 10 \log \frac{\bar{W}_i}{\bar{W}_r} \]  
(29)
Measurements
To validate the simulated concepts, measurements were done in a flow acoustic test rig.

Two microphone technique
The two microphone wave decomposition method with source switching technique \[4\] was used to characterize the test object. This is the common method used for experimental determination of the scattering matrix and the transfer matrix. The acoustic plane wave pressure amplitudes are decomposed into waves traveling in the upstream and downstream direction. To do this, the sound pressure has to be measured at two points on each side for two different source- or load cases. Upstream and downstream speaker excitation, or excitation at one side with varying load at the other side. A matrix relation between the acoustic pressure wave amplitudes and the acoustic pressure measured by the microphones can then be formed.

\[
\begin{bmatrix}
    e^{-ik_{m1}x_1} & e^{ik_{m1}x_1} \\
    e^{-ik_{m2}x_2} & e^{ik_{m2}x_2}
\end{bmatrix}
\begin{bmatrix}
    \hat{p}_{a+} \\
    \hat{p}_{a-}
\end{bmatrix} =
\begin{bmatrix}
    \hat{p}_1 \\
    \hat{p}_2
\end{bmatrix}
\]

(30)

\[
\begin{bmatrix}
    e^{-ik_{b1}x_1} & e^{ik_{b1}x_1} \\
    e^{-ik_{b2}x_2} & e^{ik_{b2}x_2}
\end{bmatrix}
\begin{bmatrix}
    \hat{p}_{b+} \\
    \hat{p}_{b-}
\end{bmatrix} =
\begin{bmatrix}
    \hat{p}_3 \\
    \hat{p}_4
\end{bmatrix}
\]

(31)
In practice, instead of the pressure amplitudes, the transfer functions between a reference signal and the microphone signals are used. Here, the reference signal was the loudspeaker voltage. The two independent measurement cases now yields four equations for each measurement, sufficient for solving the equation system giving the scattering matrix $S$.

\[
H_a = \frac{H_1 e^{ik_a s} - H_2 e^{-ik_a s}}{e^{ik_a s} - e^{-ik_a s}}
\] (32)

\[
H_a = \frac{H_2 - H_1 e^{-ik_a s}}{e^{-ik_a s} - e^{-ik_a s}}
\] (33)

\[
H_b = \frac{H_3 e^{ik_b s} - H_4 e^{-ik_b s}}{e^{ik_b s} - e^{-ik_b s}}
\] (34)

\[
H_b = \frac{H_4 e^{-ik_b s}}{e^{-ik_b s} - e^{-ik_b s}}
\] (35)

\[
S = \begin{bmatrix}
H^1_a & H^2_a \\
H^1_b & H^2_b
\end{bmatrix}
\begin{bmatrix}
H^1_a & H^2_a \\
H^1_b & H^2_b
\end{bmatrix}^{-1}
\] (36)

Where $H^{\pm}_y$ are the transfer functions, index a for upstream wave, b for downstream wave. 1 and 2 is the two source cases, upstream and downstream excitation, $s$ is the microphone spacing and $k_{y \pm}$ is the wave number in each direction at each side.

\[
k_{y \pm} = \frac{\omega}{c_0} \frac{K_0}{1 \pm K_0 M}
\] (37)

Where

\[
K_0 = 1 + ((1-i)k_0(1+(\gamma-1)/\sqrt{Pr})/\sqrt{2})
\] (38)
Where

\[ k_s = r / \sqrt{\rho_0 \omega / \mu} \]  

(39)

\( k_s \) is the shear wave number, \( \mu \) is the dynamic viscosity, \( r \) is the duct radius, \( M \) is the mach number, \( \rho_0 \) is the ambient density, \( \gamma \) is the ratio of specific heat and \( Pr \) is the Prandtl number. To perform the inversion needed in Eq. (34), the matrices have to be linearly independent. This implies that the usable frequency range for one microphone pair is restricted. With a microphone separation close to multiples of a half wavelength this inversion becomes sensitive to random errors. As proposed in [5], the wave number \( k \) should be in the range of

\[ 0.1 \pi < \frac{k}{1 - M} < 0.8 \pi \]  

(40)

This usually results in the need for more than one pair of microphones on each side of the test object to cover the frequency span of interest.

The transfer functions was estimated from

\[ \hat{H}_1 = \frac{\hat{G}_{sy}}{\hat{G}_{sx}} \]  

(41)

where \( \hat{G}_{sx} \) is the averaged cross power spectral density between the reference signal and the microphone signal and \( \hat{G}_{sx} \) is the averaged auto spectrum for the reference signal. This is the best way of estimating the transfer functions if uncorrelated noise is on the output side in the measurement system [6]. The estimation

\[ \hat{H}_2 = \frac{\hat{G}_{sy}}{\hat{G}_{sy}} \]  

(42)

can also be used if noise is expected on the input side. This was not used here. The coherence function was estimated from

\[ \gamma^2 = \frac{\left| \hat{G}_{sy} \right|^2}{\hat{G}_{sx} \hat{G}_{sy}} \]  

(43)

**Concept study and prototypes**

Several valve types were studied together with the flow reversal resonator to get an overview of which type of geometric shape and size that could be suitable for changing the acoustic properties. Control systems for varying the valve are also studied in this stage. Two concepts, prototype one and two was simulated and evaluated experimentally, a third concept, prototype three was only evaluated by simulations. Prototype one is a reference system for prototype two. Prototype two is
the same as prototype one but with a valve as short-circuit duct connecting the inlet and outlet of the flow reversal chamber, instead of a fixed open duct in prototype one. Prototype one and two was designed as down scaled test objects with eigenfrequencies higher than what is needed for a full scale prototype.

The knowledge from simulations and experimental evaluation of prototype one and two was used to design a third concept. This concept was designed as a full scale prototype with eigenfrequencies in the range of the firing frequencies generated in a heavy duty six cylinder diesel engine.

**Flow reversal resonator with open short-circuit, prototype one**
Prototype one was designed to have an eigenfrequency at 100 Hz. The dimensions of this prototype were decided with simulations of the system using the lumped model described in the method section. The eigenfrequency for this prototype is supposed to be lower than the bottom end eigenfrequency for prototype two. With this eigenfrequency, it is possible to design prototype two with eigenfrequencies from 100 Hz an upwards. The dimensions of this prototype was also decides with the need of space for a valve between the inlet and outlet in mind. The simulated transmission loss for prototype one can be seen in Figure 12.

![Figure 9: Prototype one dimensions](image)

![Figure 10: Prototype one](image)
Valves
The purpose of the valve in the flow reversal resonator was to acoustically short circuit the inlet and outlet duct and by this changing the acoustic properties of the system. Valve types considered were regular valves, e.g. butterfly valve and ball valve but also a new concept. The most important property of the valve is the ability to achieve a frequency shift large enough to follow the firing frequency of an IC-engine when going from idle speed to full speed. In the case of a heavy-duty six-cylinder diesel engine for commercial vehicles this is around 1000 to 2000 rpm. This means at least 100 Hz in frequency shift. Some other important properties are response time and power needed to control the valve.

Regular valves
Regular valves were considered because of simplicity with respect to prototype development and future commercial realization. Most of the valves available at the market are sold with different types of drives and control units which can handle signals from the truck, for example engine speed which could be used to set the position of the valve.

The ball valve, the butterfly valve and the weir-type diaphragm valve works in different ways but they are all constricting the opening over a limited length. The straight-through type diaphragm valve is constricting the opening over a larger length but the rest of the valve, inlet and outlet with its flanges are bulky. The ball valve and the butterfly valve are fast to control and a low force is needed. The diaphragm valves are slower but have got a lower flow resistance compared to the other valves.
Figure 12: a) Ball valve, b) Butterfly valve, c) Diaphragm valve, straight-through type, d) Diaphragm valve, weir type

Rod valve concept, prototype two

Figure 13: Rod valve concept
In this concept, a rod is inserted in the short-circuit duct. By placing the short-circuit duct on top of the inlet and outlet, the flow is left undisturbed and the flow losses are minimized. The force needed to control the rod is low and suitable for control by a linear electric motor.

Figure 14: Rod valve prototype

Figure 15: Prototype 2 without the rod valve mounted
Results of concept study

The different concepts were simulated using FEM and SIDLAB, the regular valves were simulated as area constrictions in the middle of the short-circuit duct with the other dimensions same as for prototype 1. These simulations showed that the regular valves were not suited for changing the acoustic properties of the flow reversal resonator in a desired way, see Figure 16, where the short-circuit duct in prototype one (40 mm in diameter, 60 mm length) has been modified with an area constriction representing an almost closed 40 mm butterfly valve in the middle of a 60 mm long short-circuit duct.

All these regular valves are varying the area over a very limited length, this is not enough to achieve the desired frequency shift. To achieve a large shift, the acoustic impedance of the short-circuit has to be changed more than what is possible with regular valves. Like the mass of air in the neck pipe of a Helmholtz resonator, the mass of air in the short-circuit duct has to be changed. The concept where a solid rod is inserted in a duct connecting the inlet and outlet, see Figure 17 and Figure 18, showed to be a successful and decided to be tested experimentally. To get a large shift of the eigenfrequency for a given length change, it is proposed in [1] that the diameter of short-circuit should be as small as possible. In the rod valve concept, the diameter of the short-circuit duct was 0.02 m, the length was 0.14 m and the inserted rod had a diameter of 0.018 m. These dimensions gave a possible range for the L/S ratio from 446 to 2345, which gave a frequency range for the eigenfrequency from 150 to 295 Hz when simulated.

As can be seen in Figure 17 and Figure 18, the simulated transmission loss differs from FEM simulation and transfer matrix simulation. The difference in amplitude at frequencies giving maximum transmission loss is due to difference in the frequency resolution and due to viscous losses included in the transfer matrix simulation but not in the FEM simulation. Another difference is the difference in frequency resolution, 1 Hz was used in the transfer matrix simulation, 5 Hz in the FEM simulation. The difference in frequency for maximum transmission loss at a certain valve position depends of that the transfer matrix simulation does not calculate the near field around the orifices to the short-circuit duct properly.

The whole model used in COMSOL was meshed with COMSOL’s physics-controlled mesh function. This generated a total of approx. 16000 elements of different size.
Figure 16: FEM and transfer matrix simulation of prototype one and prototype one modified with a 2 mm long, 5 mm in diameter constriction representing an almost closed butterfly valve

Figure 17: FEM simulations (COMSOL) of the transmission loss for the rod valve in different positions
Control system
To control the rod in the rod valve concept, a servo tube linear actuator could be used. This is an electronic device which is used to create linear movement. There are many different brands and types on the market to choose between but most of them has in common that they are sold together with a combined power supply and control unit. One example is a Copley type 2504 servo tube linear actuator. It can generate a constant force up to 51 N with peak force at 281 N. The step resolution is 20 µm with a maximum stroke of 309 mm. Maximum speed is 5.9 m/s and maximum acceleration is 355 m/s² [7].

Full scale implementation
To implement the rod valve in a full scale concept, a Scania euro IV muffler was modified and the transfer matrix was measured in a flow acoustic test rig. The modification consisted of removing the inner element so that a free volume was created, the catalyst element was left in its original place. The measured transfer matrix was used as a user defined element in SIDLAB to design a rod valve type short-circuit between the inlet and outlet on this muffler. The desired frequency shift for this implementation was from 50 to 100 Hz with mean flow present.
Figure 20: Modified Scania euro IV muffler

Figure 20 shows the modified euro IV muffler, the outlet pipe can be turned to be directed in the same way as the inlet. This should be favorable when mounting the rod valve between these pipes.

Figure 21: Full scale implementation concept

Experimental setup

MWL flow acoustic test rig setup

The measurements on prototype one and two were done in the flow acoustic test rig at MWL-KTH. This test facility consist of a fan connected to an anechoic room, the flow generated by the fan is led in to the anechoic room, used as a stagnation chamber, and then further to the measurement setup consisting of the test object placed between microphones and loudspeakers. 6 Brüel & Kjaer type 4938 ¼ inch microphones were used, three on each side to cover the frequency range of interest. The microphones were connected to Brüel & Kjaer type Nexus conditioning amplifiers with a 20-3kHz band pass filter. Data acquisition was done with HP VXi system consisting of a E1421B mainframe, E1432A input card, E1434A output card and Agilent E8491B card for PC communication. This system was connected to a PC with MATLAB used for data recording and post processing. On the
output side a NAD C370 amplifier in bridge mode was used. Between the output amplifier and the data acquisition system a 1/10 signal down step was used to prevent overload in the data acquisition system. The microphones were calibrated relative one of the microphones used since absolute calibration is not needed when measuring the transfer functions needed for further calculations.

Figure 22: MWL-KTH flow acoustic test facility

Figure 23: Microphone calibrator
The measurements were done without flow and with several flow rates, \( M = 0.025, 0.05, 0.1, 0.15 \) and 0.2. During no flow condition, random excitation was used, with flow, swept sine excitation was used. The flow rate was measured with a pitot tube on both sides of the test object, after microphone 3 on upstream side and between microphone 6 and the first loudspeaker at downstream side. The pipes upstream and downstream to the test object had the same diameter as the test object inlet and outlet, 0.056 m, except of the bend on the downstream side which had a diameter of 0.06 m.
**SWENOX flow acoustic test rig**

The measurements in the development of the full scale concept were done at a flow acoustic test rig at SWENOX. This test rig was similar to the test rig at MWL-KTH but can produce a higher mass flow and instead of an anechoic room a large muffler was used. 6 GRAS 40AP ½ inch microphones were used, three on each side to cover the whole frequency range of interest. The microphones were connected to 3 GRAS 12AA power modules with low pass filter set to 20 kHz. Data acquisition was done with HP VXI system consisting of E1421B mainframe, E1433B input card, E1433B output card and Agilent E8491B card for PC communication. This system was connected to a PC with MATLAB used for data recording and post processing. The microphones were calibrated in the same way as for the MWL experimental setup. The microphone separation was 80 mm for the high frequency microphone pairs and 500 mm for the low frequency pairs. Distance between test object and first microphone was 280 mm. The flow rate was calculated from a mass flow meter integrated in the fan system.

**Results and discussion**

**Experimental validation**

The results from the measurements are presented together with the results from the simulations used to design the prototypes. The measurements on prototype one and two were done without flow and with five different flow rates, \( M=0.025, M=0.05, M=0.01, M=0.15 \) and \( M=0.2 \).

**Prototype one – Fixed short-circuit**

![Graph showing transmission loss comparison](image)

*Figure 26: Comparison of the transmission loss for different flow rates and lumped model simulation*

In Figure 26, the measured transmission loss is presented together with the result from the lumped model simulation. The measured eigenfrequency agrees well with the simulated. At \( M=0.025 \) and \( M=0.05 \) the eigenfrequency is the same as for the no flow case but increases for higher flow rates. With flow, the maximum attenuation is strongly influenced and decreases already at low flow rates.
For frequencies not close to the eigenfrequency, the measured attenuation level is almost the same for all flow rates, slightly higher for the highest flow rates due to turbulence.

An enlarged view of the measured and simulated transmission loss around the first eigenfrequency can be seen in Figure 27.

The pressure drop over prototype two at different flow rates can be seen in Table 1.

<table>
<thead>
<tr>
<th>Flow rate</th>
<th>M=0.025</th>
<th>M=0.05</th>
<th>M=0.1</th>
<th>M=0.15</th>
<th>M=0.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure drop [Pa]</td>
<td>10.5</td>
<td>42.2</td>
<td>86.6</td>
<td>195.0</td>
<td>261.7</td>
</tr>
</tbody>
</table>
Prototype two – Rod valve short-circuit

Figure 28: Comparison of measured and simulated transmission loss for prototype two, simulations with the lumped model

Figure 28 shows the frequency range where it is possible to control the attenuation maximum with the rod valve. The transmission loss measured at M=0 and M=0.1 is presented together with the transmission loss simulated with the lumped model. The rod valve was further investigated in 5 positions during no flow and five flow rates. These results are presented in Figure 30 to Figure 34. The investigation was extended to include 9 valve positions during the measurements at M=0.1, these results are presented in Figure 35 to Figure 37. The nine valve positions is visualized in Figure 29.

Regarding the compliance of the measured and simulated transmission loss for the rod valve, it is clear that it agrees best for the case when the valve is fully open. In the transfer matrix simulations in SIDLAB, straight duct elements were used to model the rod valve, but this is not really what it is, except of when the valve is fully open. The transfer matrix simulation also has a problem to take the near field around the orifices to the short-circuit duct into the simulation. This is extra clear when the valve is almost close. At this position, the near field is large compared to the free volume in the short circuit duct. In the FEM simulations the deviation of the results can be a due to a too coarse mesh, a finer mesh was not possible because of lack in computational power.

The measured transmission loss for prototype two has the same general behavior as for prototype one when the rod valve is fully opened, 0 mm closed, the flow rate has a strong influence on the maximum attenuation. But with the valve in the end position, 140 mm closed, the flow rate has a small influence on the maximum attenuation.

The change in frequency for the maximum attenuation with varying flow rates is important to understand if the rod valve is implemented in a full scale muffler application. This frequency change for M=0.1 compared with M=0 is presented in Figure 38. For the valve positions 0 mm, 85 mm and 140 mm, the difference is small, almost 0 Hz. For the positions between 0 mm and 85 mm the
The eigenfrequencies measured with flow are higher compared to the no flow measurements. For the positions between 85 and 140 mm, the eigenfrequencies measured with flow are lower compared to the no flow measurements.

**Figure 29: Evaluated valve positions**

**Figure 30: Measured transmission loss compared with simulated transmission loss, rod valve 0 mm closed**
Figure 31: Measured transmission loss compared with simulated transmission loss, rod valve 20 mm closed

Figure 32: Measured transmission loss compared with simulated transmission loss, rod valve 70 mm closed
Figure 33: Measured transmission loss compared with simulated transmission loss, rod valve 120 mm closed

Figure 34: Measured transmission loss compared with simulated transmission loss, rod valve 140 mm closed
Figure 35: Transmission loss measured at M=0 compared with simulated transmission loss for the evaluated valve positions.

Figure 36: Transmission loss measured at M=0 compared with transmission loss measured at M=0.1 for the evaluated valve positions.
Figure 37: Transmission loss measured at $M=0$ compared with transmission loss measured at $M=0.1$ for the evaluated valve positions.

Figure 38: Difference in frequency giving the maximum attenuation for the evaluated valve positions.

The pressure drop over prototype two at different flow rates can be seen in Table 2.

Table 2: Pressure drop measured at different flow rates

<table>
<thead>
<tr>
<th>Flow rate</th>
<th>M=0.025</th>
<th>M=0.05</th>
<th>M=0.1</th>
<th>M=0.15</th>
<th>M=0.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure drop [Pa]</td>
<td>10.5</td>
<td>42.1</td>
<td>127.9</td>
<td>257.4</td>
<td>346.3</td>
</tr>
</tbody>
</table>
Full scale implementation
In Figure 39, the transmission loss simulated for the full scale concept is presented. The rod valve used in this simulation had a duct diameter of 100 mm, 90 mm rod and a length of 300 mm. With this dimensions, the the ratio between area and length of the short circuit duct, L/S, could be shifted from 38 to 201. The simulated transmission loss indicates a frequency shift large enough to follow the firing order of a six cylinder diesel engine, but this simulation does not include any flow. Based on the measurements on prototype two, the first peak in transmission loss with open valve will decrease with flow. For this concept it means that the controllable frequency span will be too small to cover a frequency span of 50 Hz. Based on this, no prototype was built and tested on a truck engine. When comparing the simulated transmission in Figure 39 with the measured transmission loss presented in Figure 40, it is clear that the high frequency attenuation is decreased too much to be acceptable. In Figure 40, it can also be seen that the peak in transmission loss is too low. If this should be implemented, this peak must be higher, this also indicates that this modified muffler is not suitable for use as a part of a flow reversal resonator with short-circuit duct.

![Simulated transmission loss for the full scale concept](image-url)
The valve concept where a rod is inserted in a duct connecting the inlet and outlet of a flow reversal chamber has been proven to give a frequency shift for the maximum attenuation. The valve positions that provide a smooth change (approx. 15-25 Hz, see Figure 41) of the frequency giving maximum transmission loss are the positions 20 mm closed to 120 mm closed. In other words, the positions when the rod is between the orifices of the short-circuit duct, see Figure 29.

The frequency range shown in Figure 41 indicates some important conclusions. The measured transmission loss for valve position 100 mm closed covers the measured transmission loss for valve
position 20 to 85 mm closed, at frequencies below maximum transmission loss. This means that there is no real winning in using these positions. This was something which was not shown in the simulation of the system, which was performed without flow. If one would redesign this resonator system with a similar valve, the initial L/S ratio should probably be lower. The system seems to be less sensitive to flow at higher L/S ratios. If this is a general behavior or just due to the valve design is not clear and need further investigation. Though the eigenfrequency of the system depends on all dimensions, it is possible to design the flow reversal resonator with different L/S ratio but with the same eigenfrequency. This could be one way to further investigate the flow effect on different short-circuit ducts.

In the full scale concept, no combination of valve dimensions was considered good enough to design a valve that could work in the same way as prototype two. Because of this, the full scale concept was not realized, although a full scale prototype was the goal with this work. But with this said, a full scale implementation should be possible if the flow reversal chamber can be designed together with the rod valve short-circuit. As mentioned in [1], the short circuit to be controlled should have a small diameter, but in a full scale muffler implementation a small short-circuit diameter can be a problem when the goal is to achieve an eigenfrequency as low as 50 Hz. A small short-circuit can be compensated with a large volume of the flow reversal chamber, but in an implementation on a truck, the space is limited. Another way of achieving a low eigenfrequency is to have long inlet and outlet pipes. Here, further investigation is needed to design a full scale flow reversal resonator with controllable short-circuit which fulfills all requirements (outer dimensions, enough transmission loss over the whole frequency span, flow losses, etc.).
References


