

Systematic Optimization of an Exhaust System to Meet Noise Radiation Criteria at Idle

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ABSTRACT

Exhaust noise is a major contributor to the radiated noise level of a vehicle, especially at idle. The radiated noise level has to meet a certain criteria based on regulation and consumer demand. In many cases, the problem appears after the vehicle is manufactured and the tailpipe noise measurement is performed indicating a high noise level that needs to be reduced. This paper describes one of those cases where the radiated noise level of a certain passenger car at idle was required to be reduced by 6 dB(A). The exhaust system consists of one main muffler and one auxiliary muffler. A 1D two-port model of the exhaust system including the two mufflers was built using commercial software. This model was validated against the measurement of the two-port matrix of both mufflers. The model was then used together with tailpipe noise measurements to estimate the characteristics of the source strength and impedance. Using a 1D model of the complete system, it was possible to propose several modifications for the vehicle manufacturer to choose from. A systematic optimization approach is presented in this paper describing several suggestions to reduce the radiated tailpipe noise.

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INTRODUCTION

The increased environmental demands for reduced noise and CO₂ emissions, put in focus effective design methods for IC-engine exhaust systems to provide high acoustic damping and low pressure drop. Noise control can be accomplished at three locations - at the source, in the path between the source and the receiver and at the receiver. In this paper the noise is controlled in the path between the source and the receiver through the optimization of the exhaust system mufflers. The specific design of mufflers is driven by three paramount considerations which are: appropriate outer geometry related to space constraints, low pressure drop and sufficient sound attenuation [1]. The concern in this paper is at idle and therefore the pressure drop of the exhaust system is neglected here.

Some research has been performed on shape optimization of mufflers in the past. In 1986, Bernhard [2] introduced a method for shape optimization of simple expansion mufflers by using so called design sensitivity matrices. The volume of the reactive silencer is still non-constrained. Yeh [3] presented results for optimal design of a single chamber muffler with side

inlet and outlet, and optimal design of a single expansion muffler with concentric extended inlet and outlet pipes [4]. For both cases, space constraints were applied by using the exterior penalty function method, interior penalty function method and the method of feasible direction, which also depends on sensitivity analysis. The calculation of design sensitivity matrices is difficult especially for complex mufflers.

Yeh introduced an optimization method for constrained single expansion mufflers [5], double chamber mufflers [6], double chamber mufflers with extended inlet and outlet tubes [7], and multi-segments mufflers [8], all using a Genetic Algorithm (GA) based on the principles of natural biological evolution. Later [9], he carried out a study to determine the optimal shape design of a single expansion muffler with side inlet/outlet using the boundary element method for calculating the acoustic properties, combining the gradient method and the genetic algorithm to get the optimal design. Although with the Genetic Algorithm the sensitivity analysis is avoided, one disadvantage of the GA is their higher computational cost [10], especially for large number of design variables



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Barbieri [11] presented an approach for shape optimization of single expansion mufflers with extended tubes using the Zoutendijk's feasible directions method, in order to get the maximum transmission loss at certain frequency ranges using finite element analysis. Ih [12], presented results of shape optimization of a double chamber muffler using simulated annealing algorithm, in order to achieve a maximum insertion loss. Chie Chiu presented the shape optimization of multi-chamber cross-flow mufflers by simulated annealing optimization [13], and studied multi-chamber perforated mufflers by genetic algorithm optimization [14].

Elsaadany, Elnady and Abom [15] presented a case study to optimize an exhaust system that consists of a diesel engine, two mufflers, intermediate pipes, and a tailpipe. The first muffler is a typical EU-regulation compliant. The dimensions and location of the second muffler were to be optimized. A limit for the system back pressure is imposed by the engine manufacturer. An optimum design was investigated for different engine speeds and loads.

In this paper, a systematic optimization approach is presented based on the simple data available to any design engineer. This approach is applied to the exhaust system of a passenger car. High tailpipe noise was observed at idle, and the objective was to reduce the overall level at idle by 6 dB. The following steps were used throughout the paper to achieve the optimization goal.

1. Tailpipe Noise Measurement
2. Measurement of Transfer Matrix of different mufflers in the system
3. Modeling of the exhaust system
4. Validation of the model against the transfer matrix measurements
5. Estimation of the source characteristics of the engine
6. Validation of the full model
7. Suggested modifications to the system
8. Build a prototype with the suggested modifications
9. Validation Measurement

In this work, the acoustic performance is analyzed assuming low frequency 1-D sound propagation and incompressible flow in complex duct networks based on the two-port theory. The acoustic modelling is done using the commercial software SIDLAB [16].

THE TWO-PORT THEORY

Duct systems or networks acting as acoustic wave guides are assuming linear wave propagation suitable for analysis using so called building block or matrix methods. When the low frequency regime is considered, and plane wave propagation can be assumed, this approach leads to the two-port transfer

matrix method, as described in [17]. This method splits the system into several smaller duct parts, acoustic elements, in which the sound propagation is well defined. Since plane acoustics waves propagate between different elements the sound field can be characterized by two state variables. One convenient choice is to use acoustic pressure p and volume velocity q . The sound propagation inside each element is analyzed separately and higher order modes can exist inside the element.

There exists a complex frequency dependent 2×2 matrix, T , the so called two port transfer matrix, which describes the sound transmission within a certain two port element. The acoustic pressure and volume velocity on each side of the element can be related with the following expression

$$\begin{bmatrix} p_1 \\ q_1 \end{bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \cdot \begin{bmatrix} p_2 \\ q_2 \end{bmatrix} \quad (1)$$

where p and q are the acoustic pressure and volume velocity, the subscript 1 refers to the inlet side (or node) and 2 refers to the outlet side (or node), and T_{ii} are the elements of the two-port transfer matrix. It can be noted that the two-port in Equation (1) is passive, i.e., it does not contain any sources or an active port.

Sources can be introduced into the model as a one-port element that only affects the network through one opening (port). It is usually connected to a single node, i.e., a connection point between one or several two-ports in a duct network. A one-port connected to a node can be completely described by the acoustic source impedance and the source strength (constant pressure source). The relation between pressure and volume flow for an active one-port can be written as

$$p = p_s - z_s q \quad (2)$$

where p_s is the source strength and z_s is the source impedance. When all the elements (one-ports + two-ports) in a network are generated, they are connected to each other as defined by the network structure and the sound propagation in the complete duct system can be analyzed. If the elements are connected in cascade ("a chain"), the problem is simple and the transfer matrix for the system is the result of the successive multiplication of the transfer matrices of all elements. If the elements are connected in an arbitrary fashion, the analysis developed by Glav and Abom [18] is used to set up a global network matrix. This is implemented in SIDLAB which is based on the representation of a duct network as a network of arbitrary connected two-ports. It can be noted that SIDLAB also allows for active two-ports but since this is not of interest here the details have been omitted.

For the work presented in this paper, five elements are used; pipe, horn, expansion chamber, lined duct and perforate. These elements as implemented in SIDLAB will be described next.

2.1.1 The Pipe Element

The transfer matrix can be deduced for a pipe element using basic equations of linear acoustics for plane wave propagation, and is given by:

$$T = \begin{bmatrix} \frac{1}{2}(e^{ik_+L} + e^{-ik_-L}) & \frac{\rho_0 c_0}{2S}(e^{ik_+L} - e^{-ik_-L}) \\ \frac{S}{2\rho_0 c_0}(e^{ik_+L} - e^{-ik_-L}) & \frac{1}{2}(e^{ik_+L} + e^{-ik_-L}) \end{bmatrix} \quad (3)$$

where L is the pipe length, S is the cross sectional area, and k_+ and k_- are the wave numbers in the positive and negative directions, respectively. The effects of viscous and turbulent damping, included through complex wave numbers, are taken according to Dokumaci [19] and Howe [20].

2.1.2 The Horn Element

The horn element is a smooth expansion or contraction in a flow duct. A horn is often used to decrease the pressure drop in an exhaust system. In the modelling used in this paper, effects of flow separation are assumed negligible. In order to allow for an arbitrarily varying cross section, the horn is accordingly divided into a number, N , of short "ordinary straight pipes" connected in cascade and approximating the area variation.

2.1.3 The Expansion Chamber Element

The expansion chamber element considered here is a conventional expansion chamber with extended and concentric inlet and outlet pipes. All walls but the end plates, which can be given wall impedance, are assumed to be rigid. The effects of higher order modes are included inside the chamber. This analysis, using the mode-matching technique is explained in detail by Åbom [21]. Compared to Ref. [21], the SIDLAB acoustic model also includes flow losses (dissipation) at the inlet and outlet expansion/contraction. It should be noted that higher order modes are included not only in the main chamber but also in quarter wavelength resonator regions and in the inlet and outlet ducts, although only plane waves are assumed to be incident on the chamber in these ducts. The number of included higher order modes is prescribed for the main chamber and the number of modes in the other regions, required for accurate analysis, is then given by the so called 'edge condition' [22]. At the 'bottom' of the quarter wavelength resonator, i.e., at the end plates, it is possible to allow for a locally reacting boundary, as in the case of a wall covered with porous material.

2.1.4 The Perforated Element

The perforated element impedance is calculated based on the dimensions of the perforate: thickness, porosity, hole diameter, grazing and through flow, and total area. The used model is based on the expressions developed by Elnady and Bodén [23].

A segmentation approach to model perforated pipes inside a cavity (four-port element) was developed by Elnady & Åbom [24]. It is based on the conversion of the four-port element into a number of two-port elements. It uses a two-port transfer matrix to model both the perforated branches and the intermediate hard pipe segments. This is illustrated in Figure 1. Each perforated tube section is represented by point-wise acting perforate impedances represented by lumped impedance two-ports (shaded rectangles) separated by hard-walled straight pipes (un-shaded rectangles). A split into three perforate elements (two pipe segments) is illustrated below.

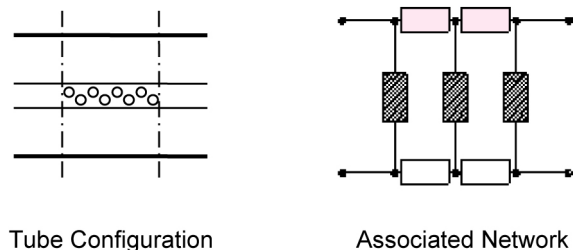


Figure 1. The tube configuration of a four-port element and the associated two-port network. SIDLAB will automatically generate the network with a number of sections that depends on the perforate length and maximum frequency.

2.1.5 The Lined Duct Element

The lined duct element is similar to the hard pipe element except that the main passage for flow is surrounded by a cavity filled with absorbing material. The same equation to calculate the transfer matrix is used. The only difference is the calculation of the complex axial wavenumbers. The wavenumbers are calculated based on Cummings and Chang [25].

DESCRIPTION OF THE EXHAUST SYSTEM

The exhaust system under consideration is shown in Figure 2. The system consists of the manifolds, a catalytic converter, an auxiliary muffler, and a main muffler. The total length of the system is 3750 mm. For simplicity, the source plane is taken directly after the catalytic converter exit. The manifold section and the catalytic converter are both considered part of the source.

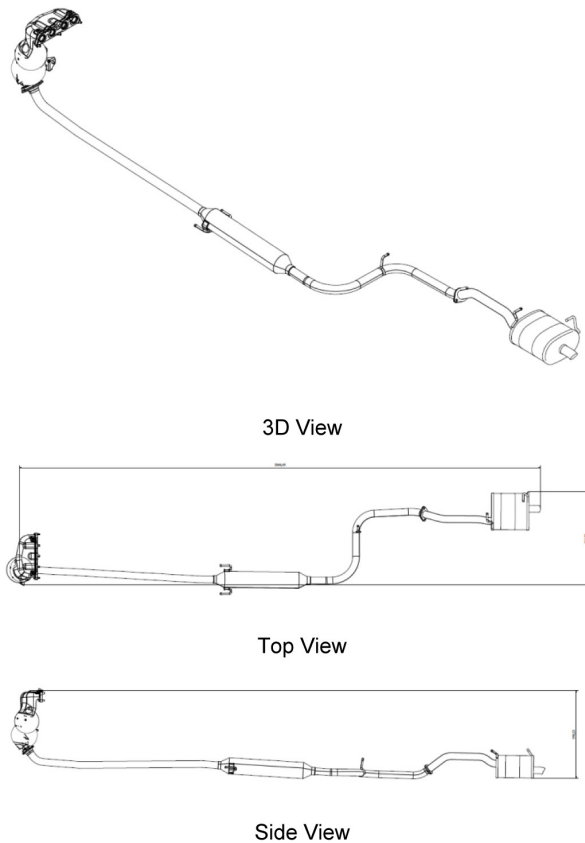


Figure 2. Layout of the exhaust system showing the location of the auxiliary and main mufflers.

TAILPIPE NOISE MEASUREMENT

A good indication of how much noise a vehicle generates is the measurement of the tailpipe noise. A microphone is located in the same horizontal plane as the exhaust outlet, as seen in Figure 3. This gives a measurement of the radiated noise from the exhaust outlet while the vehicle is stationary. In this project, the measurement was taken at idle condition only. Other measurements were taken when the engine is accelerating but the results are outside the scope of this paper.



Figure 3. The location of the microphone with respect to the tailpipe exit during the tailpipe noise measurements. This photo shows the case with 700 mm extension.

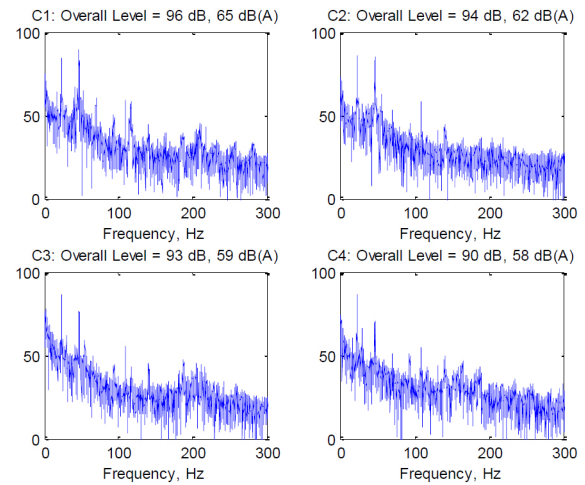


Figure 4. The measured Sound Pressure Level for different extensions 0 (C1), 200 (C2), 500 (C3) and 700 (C4) mm respectively.

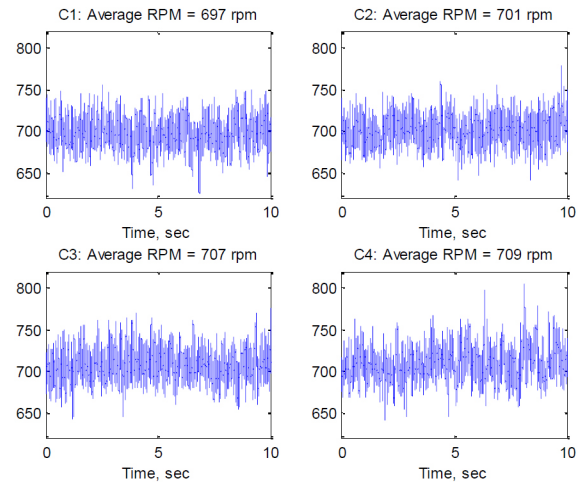


Figure 5. The recorded Engine RPM for different extensions 0 (C1), 200 (C2), 500 (C3) and 700 (C4) mm respectively.

The radiated tailpipe noise was measured for different lengths of the tailpipe. The tailpipe was extended with 0, 200, 500 and 700 mm. The objective of this procedure is to obtain different measurements with different end conditions in order to validate the estimation of the source characteristics. During the measurement, the engine was running at idle. The engine rotational speed varied from 650 to 750 RPM with an average of 700 RPM as shown in Figure.

The firing frequency of an internal combustion engine can be calculated by the following formula

$$\text{Firing Frequency} = \frac{\text{RPM}}{60} \times \frac{2}{N_{st}} \times N_{cyl} \quad (4)$$

Where RPM is the number of revolutions per minute for the engine, N_{st} is the number of strokes, and N_{cyl} is the number of cylinders.