ICSV13 - Vienna
The Thirteenth International Congress on Sound and Vibration
Vienna, Austria, July 2-6, 2006

EXPERIMENTAL INVESTIGATION OF THE ACOUSTIC EFFECT OF NON-RIGID WALLS IN IC-ENGINE INTAKE SYSTEMS

Magnus Knutsson*[1], Hans Bodén[2], and Ravi Varma Nadampalli[2]

[1]Volvo Car Corporation, 405 31 Göteborg, Sweden
[2]KTH Aeronautical and Vehicle Engineering, The Marcus Wallenberg Laboratory for Sound and Vibration Research, SE-100 44, Stockholm, Sweden
mknutsso@volvocars.com (e-mail address of lead author)

Abstract
This paper presents the results of an experimental study of the acoustic properties of an automotive air intake system. Modern air intake systems for personal cars are largely made of plastic materials. When trying to make an acoustic model for the sound transmission through the intake system it is questionable if the walls of different components can be modelled as acoustically rigid. The acoustic losses which determine the transmitted sound level at break-through frequencies may be especially difficult to model. To find out if acoustic losses associated with wall vibrations or with flow interaction are dominating for a typical intake system an experimental investigation has been performed. The acoustic plane-wave transmission matrices have been measured, using the two-source replacement technique, for the complete intake system and for separate parts. Measurements were made in a flow test rig without flow and for two different flow speeds. The measurements were then repeated with sand loading on the walls to reduce wall vibrations. The results indicate that flow related losses dominates at low frequencies while losses associated with acoustically non-rigid walls dominate at higher frequencies, giving an additional transmission loss in the order of 2 dB for the complete system. Both effects will therefore have to be taken into account in an accurate model. Comparisons will also be made between results from an acoustic FE-model and experimental results.

INTRODUCTION

An intake system for a naturally aspirated multi-cylinder IC-engine is mainly composed of an intake manifold, a throttle, an air cleaner, ducts and occasionally resonators. The primary task of the system is to provide the engine with clean air at appropriate temperature with as small pressure drop as possible. Among the secondary tasks, the ability to reduce sound is crucial since the orifice intake noise is one of the main contributors to the Pass-by noise legislation for passenger cars. The dominating part of the intake noise from a naturally aspirated engine is originating from the movement of the pistons and valves and travels as plane waves upstream the ducts to the intake orifice, where it is radiated to the surroundings. The component with the largest potential to reduce noise is the air cleaner box acting as an expansion chamber. Due to geometrical restrictions in the surrounding engine compartment, the air cleaner has a complicated geometry that normally is not possible to describe analytically. Predictions are possible either from measurements or by numerical calculations. However using acoustic finite elements as a tool, the losses in the system are not predictable. The losses can be caused by several physical mechanisms such as deviations from adiabatic changes of state, flow and fluid structure interaction. This project aims to study the losses due to fluid structure interaction and flow for a typical air cleaner system.

Several parameters can be used to describe the acoustic performance of a duct element such as an air intake system. These include the transmission loss (TL), the noise reduction (NR), and the insertion loss (IL). The parameter chosen for comparisons in this investigation is the transmission loss which is the difference in sound power level between the incident and the transmitted sound wave when the test object termination is anechoic. The standard technique today for measuring acoustic plane wave properties in ducts, such as absorption coefficient, reflection coefficient and impedance is the two-microphone method (TMM) [2-5,9]. The sound pressure is decomposed into its incident and reflected waves and the input sound power may then be calculated. Many papers have been devoted to the analysis of the accuracy of the TMM for example [2-4]. Transmission loss can in principle be determined from measurement of the incident and transmitted power using the two-microphone method on the upstream and downstream side of the test object provided that a fully anechoic termination can be implemented on the outlet side, which is very difficult in the low frequency region and with flow. Instead the so-called two-source replacement technique [8] has been used. In this technique sufficient information for determining the two-port matrix is obtained from two sets of measurements, one with the source on the upstream side and one with the source on the downstream side [7].

EXPERIMENTAL TECHNIQUES

The technique used for determining the two-port data in this study is the two-source replacement method [5,6,8], using the test set-up shown in figure 1. The first test state is obtained by turning loudspeaker A on and B off and the second independent test state is obtained by turning loudspeaker B on and A off. If the input and the output vectors for the transfer matrix are measured, the following matrix equation can be established

$$\begin{bmatrix} \hat{p}_a^1 & \hat{p}_a^2 \\ \hat{q}_a^1 & \hat{q}_a^2 \end{bmatrix} = \begin{bmatrix} T_{aa} & T_{ab} \\ T_{ba} & T_{bb} \end{bmatrix} \begin{bmatrix} \hat{p}_b^1 & \hat{p}_b^2 \\ \hat{q}_b^1 & \hat{q}_b^2 \end{bmatrix} \tag{1}$$

where 1 denotes the first test state and 2 the second. Once the four-pole parameters have been established, the transmission loss of the test object can be obtained from equation (2).

$$TL(f) = 20\log_{10}\left[abs\left(T_{aa} + T_{ab}/Z + T_{ba}Z + T_{bb}\right)\right] - 10\log_{10}(4) \qquad (2)$$

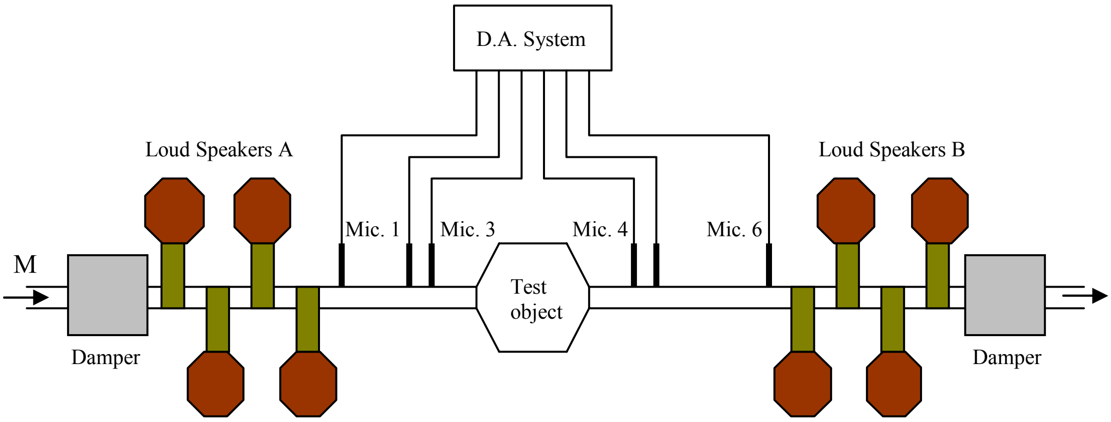

Figure 1 – Layout for test rig for mufflers at MWL/KTH

An efficient way of suppressing turbulent pressure fluctuations is to use a reference signal, which is uncorrelated with the disturbing noise in the system and linearly related to the acoustic signal in the duct. A good choice for the reference signal is to use the electric signal driving the external sources as a reference. Deviation from a linear relation between the reference signal and the acoustic signal in the duct can for instance be caused by non-linearity in amplifiers and loudspeakers at high input amplitudes, temperature drift and non-linearity of the loudspeaker connections to the duct at high acoustic amplitudes. One possibility is to put an extra reference microphone close to a loudspeaker or even in the loudspeaker box behind the membrane i.e., without contacting the flow. Otherwise one of the measurements microphones can be used as a reference. The disadvantage of this technique is that one will get a minima's at the reference microphone at certain frequencies or poor signal to noise ratio. To solve this problem one can use the microphone with the highest signal-to-noise ratio as the reference [1]. In this work the electronic signals driving the loudspeakers was used as the reference.

NUMERICAL TECHNIQUES

An uncoupled finite element model of an acoustic cavity with real valued speed of sound and without mean flow will represent the situation with the least losses possible. However, this situation will not be possible to reproduce in measurements since there will always exist small amounts of losses due to wave propagation and deviation from adiabatic changes of state that are not represented by the a linear acoustic FE-model. Obviously, the comparison between experimental and numerical results in this study will suffer from this effect.

About 200 000 linear elements and 40 000 nodes were used to calculate a harmonic solution for every 10 Hz between 10 and 1000 Hz. The element size was approximately 10 mm yielding more than 30 elements per wavelength at 1000 Hz which was the highest frequency studied. The method to obtain the four-pole parameters was the same as in the experiments using the two-source replacement technique. Finally, the frequency dependent transmission loss for the test object was calculated using equation (2). All FEM calculations were performed using the commercial software LMS/Sysnoise.

TEST SET-UP

Experiments were carried out at room temperature using the flow acoustic test facility at The Marcus Wallenberg Laboratory for Sound and Vibration Research, KTH. The test ducts used during the experiments were made of standard steel with wall thickness of 2 mm. The duct diameters were chosen to fit the test objects. Eight loudspeakers were used as acoustic sources, as shown in figure 1. The loudspeakers were divided equally between the upstream and downstream side. Fluctuating pressures were measured by using six condenser microphones flush mounted in the duct wall. The measurements were carried out using random noise with different number of frequency domain averages. The flow velocity was measured using a Pitot tube. Once the flow velocity was measured the Pitot tube was removed from the duct before taking the acoustic measurements as it might disturb the flow. The flow up-and down-stream of the test object was measured separately before and after the acoustic measurements and the average result was used. The transfer functions between the reference signal and the microphone signals was measured and used to estimate the transfer matrix components.

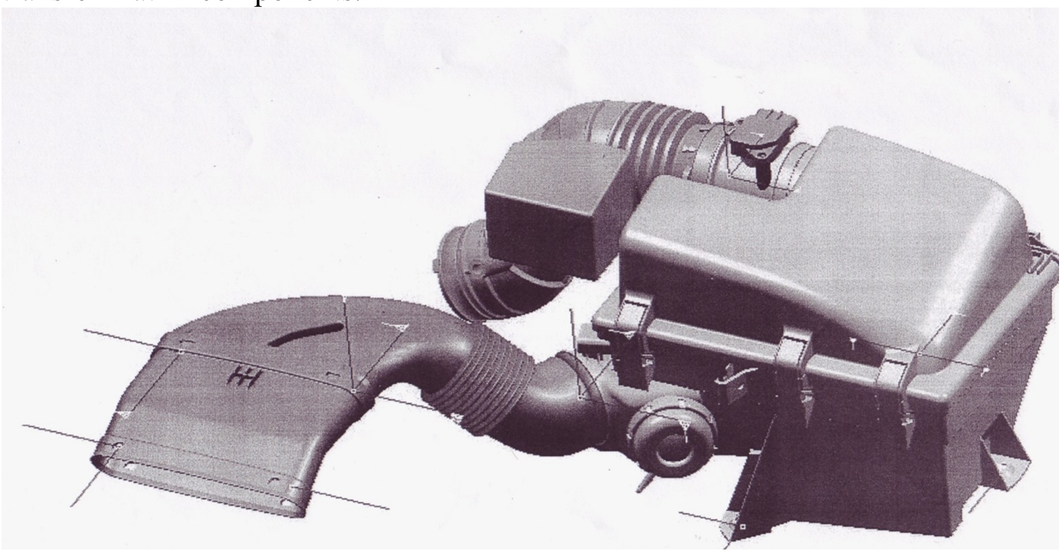

Figure 2 - Complete air intake system

The intake system that was studied in this investigation is originally developed to be fitted in a large size personal car with a five cylinder naturally aspirated petrol engine installed. Figure 2 shows a 3D CAD image of all components in the system. However, in this study when referring to the complete system, the dirty air duct with the non-circular cross section to the left in the picture is excluded. All parts are made of stiff plastic material except for the rubber hose connecting the air cleaner to the engine. To clean the incoming air, a filter paper is mounted in the air cleaner box. Figure 3 shows the opened air cleaner box with the filter paper mounted. The acoustic transmission properties of the filter paper were studied by repeating the measurements with the filter paper removed. The flow speed was chosen to represent engine operating conditions at full load when performing Pass-by noise certification.

Figure 3 – Air cleaner unit with filter paper

RESULTS AND DISCUSSION

Effects of yielding walls and flow

To study the effect of yielding walls, the transmission loss for the complete air intake system (without the dirty air duct) is shown in figure 4. The system is modified by removing the Helmholtz resonator and the filter paper in figures 5 – 6.

In figure 4 both the resonator and the filter paper is present. Here the effect of yielding walls is clearly visible between 500 and 1000 Hz. Yielding walls will add some 2-3 dB extra to the transmission loss in this frequency region independent of the flow conditions tested. In this figure, the effect of flow is also obvious. At the peak value at 310 Hz, defined by the Helmholtz resonator, the flow will decrease the transmission loss more than 10 dB. Increasing the mean flow speed will aggravate this effect.

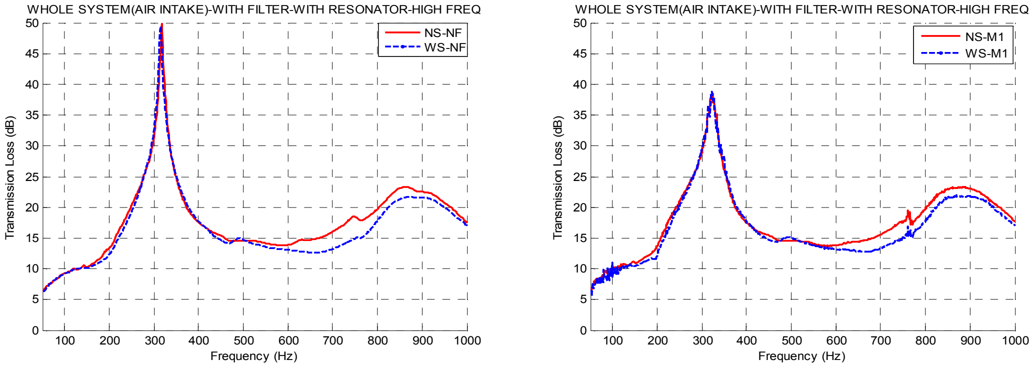

Figure 4 – Effect of yielding walls and flow. The picture to the left shows transmission loss for the complete system without flow, (NS-NF with yielding walls, WS-NF with sand loaded walls). The picture to the right shows the same set-up with flow, (NS-M1 with yielding walls, WS-M1 with sand loaded walls). M1 = 20.5 m/s.

In figure 5, where the resonator is removed, the observed effect of yielding walls between 500 – 1000 Hz is still present. A smaller amount of additional transmission loss can also be noticed between 200 and 500 Hz. However the large change due to flow is not present. This observation is in line with earlier knowledge about influence of mean flow on transmission properties for Helmholtz resonators.

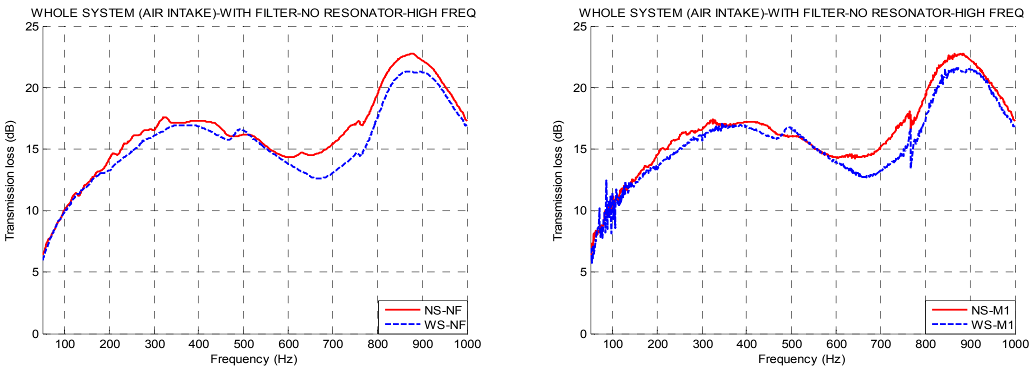

Figure 5 – Effect of yielding walls and flow. The picture to the left shows transmission loss for the complete system without resonator and without flow, (NS-NF with yielding walls, WS-NF with sand loaded walls). The picture to the right shows the same set-up with flow, (NS-M1 with yielding walls, WS-M1 with sand loaded walls). M1 = 20.5 m/s.

In figure 6 both the resonator and the filter paper is removed. The observation about the effect of yielding walls and flow is still valid. The extra contribution from this figure is the large effect of removing the filter paper. Comparing figure 5 and 6 shows that the filter paper adds more than 10 dB in the break through frequency at 780 Hz. It also shifts the frequency from 780 Hz to 650Hz, indicating both a resistive and a reactive effect. Removing the filter paper will also reveal a small effect of flow below 300 Hz, where additional losses of about one dB are appearing.

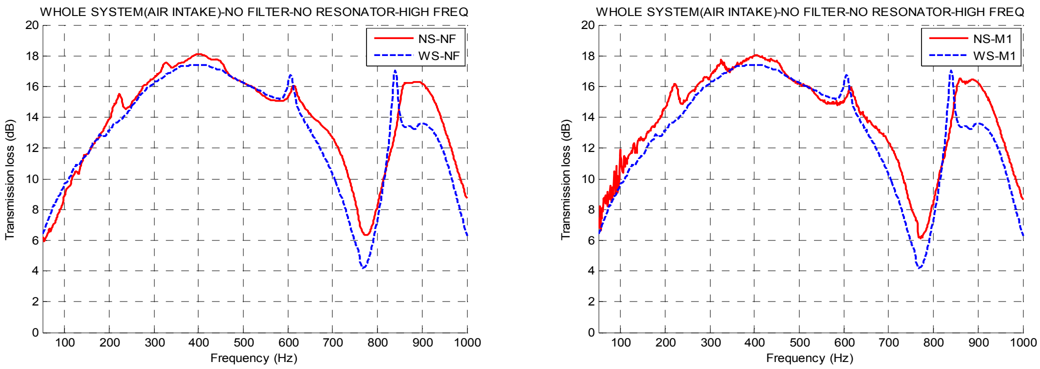

Figure 6 – Effect of yielding walls and flow. The picture to the left shows transmission loss for the complete system without resonator, without filter paper and without flow, (NS-NF with yielding walls, WS-NF with sand loaded walls). The picture to the right shows the same set-up with flow, (NS-M1 with yielding walls, WS-M1 with sand loaded walls). M1 = 20.5 m/s

Comparison with numerical results

Finally to verify the use of finite element to predict transmission loss, a comparison between numerically obtained and measured results is shown in figure 7. To simplify the calculations just results for the air cleaner box is presented.

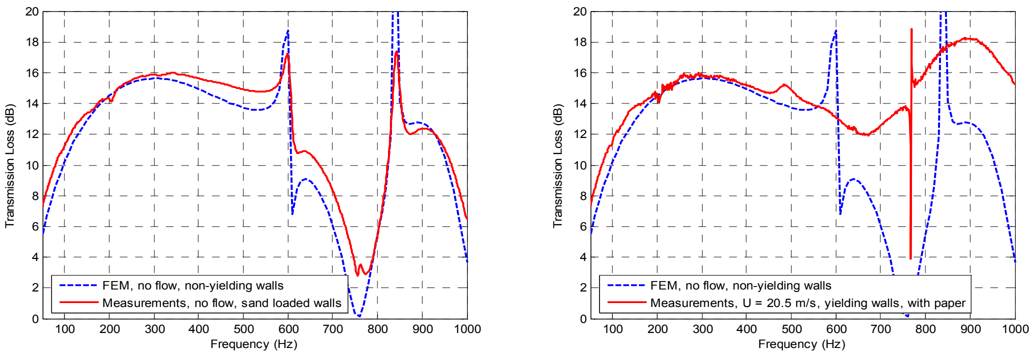

Figure 7 – Comparison of numerical results and measurements for air cleaner box. FE-calculations in both pictures with rigid walls, without filter paper and without flow. Measurements to the left with sand loaded walls, without filter paper and without flow. Measurements to the right with yielding walls, with filter paper and with mean flow 20.5 m/s.

In the picture to the left, the measured values are obtained with sand loading but without mean flow, aiming to get as close as possible to those obtained by numerical simulations where no fluid structure interaction and no mean flow is present. The agreement is reasonably good, with a deviation of about 2 dB. Of interest is to notice that numerical calculations under predict the transmission loss almost everywhere. It is not possible to conclude if this deviation is due to damping of the propagating sound waves or due to inability to establish totally rigid walls using the sand loading.

The picture to the right shows the same FE results, here compared to measurements corresponding to the actual design case where yielding walls, mean flow and filter paper all are present. Up to 400 Hz the prediction is reasonably good with a maximum deviation of about 2dB. Above 400 Hz the prediction is more or less useless with deviations of more than 10dB. From figure 5 and 6 it can be concluded that most of the deviation originates from the effects of the filter paper.

CONCLUSIONS

From the results obtained it can be concluded that the method used to find the transmission properties for the air intake system gives reasonably good results. It seems that to model the complete system, non-rigid wall effect will have to be taken into account as well as sound flow interaction effects. Interaction between the sound field and the walls could give 2-3 dB increase of transmission loss for the complete intake system in the frequency region 550-1000 Hz. The effect of flow is mainly important for the resonator but also adds an extra dB to the transmission loss below 300 Hz when the filter paper is not present. It can also be concluded from the experiments that a good model for the acoustic transmission through the filter paper is crucial for the results above 500 Hz.

REFERENCES

[1] Allam, S., "Acoustic Modeling and Testing of Advanced Exhaust System Components for Automotive Engines", PhD thesis, The Royal Institute of Technology (KTH), 2004

[2] Bodén H., Åbom M., "Influence of Errors on the Two-Microphone Method for Measuring Acoustic Properties in Ducts", J. Acoust. Soc. Am., **79(2),** 541-549 (1986)

[3] Bodén H., Åbom M., "Error analysis of Two-Microphone Measurements in Ducts with Flow", J. Acoust. Soc. Am., **83,** 2429-2438 (1988)

[4] Chung J.Y., Blaser D. A., "Transfer Function Method of Measuring In-Duct Acoustic Properties I. Theory." J. Acoust. Soc. Am., **68,** 907-913 (1980)

[5] Munjal M.L., *Acoustics of Ducts and Mufflers,* (Wiley-Interscience, New York, 1987).

[6] Munjal M. L., Doige A. G., "Theory of a Two Source-location Method for Direct Experimental Evaluation of the Four-pole Parameters of an Aeroacoustic Element" J. Acoust. Soc. Am., **141(2),** 323-333 (1990)

[7] Nadampalli R.V., "Acoustic properties of IC-engine intake system components an experimental study", TRITA-AVE 2006:10 ISSN 1651-7660

[8] Åbom M., "Studies of Sound Propagation in Ducts and Mufflers" PhD thesis, The Royal Institute of Technology (KTH), Report Number 8902 (1989)

[9] "Acoustic-Determination of Sound Absorbing Coefficient and Impedance in Impedance Tubes, Part I: Method Using Standing Wave Ratio." ISO10534-1: 1996.