IC-Engine Intake Noise Predictions Based on Linear Acoustics

Magnus Knutsson

Licentiate Thesis

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The Marcus Wallenberg Laboratory for Sound and Vibration Research

<table>
<thead>
<tr>
<th>Postal address</th>
<th>Visiting address</th>
<th>Contact</th>
</tr>
</thead>
<tbody>
<tr>
<td>Royal Institute of Technology</td>
<td>Teknikringen 8</td>
<td>Tel:  +46 31 325 40 64</td>
</tr>
<tr>
<td>MWL / AVE</td>
<td>Stockholm</td>
<td>Email: <a href="mailto:magnuskn@kth.se">magnuskn@kth.se</a></td>
</tr>
<tr>
<td>SE-100 44 Stockholm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sweden</td>
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Abstract

Shorter product development cycles, densely packed engine compartments and intensified noise legislation increase the need for accurate predictions of IC-engine air intake noise at early stages. The urgent focus on the increasing CO₂ emissions and the efficiency of IC-engines, as well as new techniques such as homogeneous charge compression ignition (HCCI) might worsen the noise situation. Non-linear one dimensional (1D) computational fluid dynamic (CFD) time domain prediction codes are used within the automotive industry to predict intake and exhaust orifice noise. The inherent limitation of 1D plane wave propagation, however, limits this technique to sufficiently low frequencies where non-plane wave effects are small. Therefore this type of method will first fail in large components such as air cleaners. Further limitations, that might not be important for simulation of engine performance but indeed for acoustics, include difficulties to apply frequency dependent boundary conditions and losses as well as to include effects of vibrating walls. This thesis deals with the use of linear acoustics to improve the accuracy of intake orifice noise predictions.

In order to predict intake noise using linear acoustics, knowledge is required about the engine as an acoustic source. The first part of this thesis describes how a linear time invariant one-port source model can be extracted using non-linear 1D CFD. Predicted source data for a six cylinder naturally aspirated engine is validated using experimental data obtained from engine test bench measurements.

Acoustic 3D finite elements (FE) or boundary elements (BE) can be used to predict sound transmission through duct systems and include effects of non-plane waves. However, acoustic losses can not be predicted by linear theory. The second and third paper in this thesis include experimental investigations dealing with the influence of mean flow, yielding walls and filter paper on the acoustics of air intake systems.

The third paper also describes how a linear source, extracted from 1D CFD, can be coupled to acoustic FE describing the intake system and to BE describing the radiation to the surroundings. These couplings create an entirely virtual methodology to predict intake orifice noise. Simulations and measurements are performed for a large number of engine revolution speeds in order to make the first systematic validation of a complete intake noise model for a wide engine speed range.

Charge air coolers (CACs) are used in the intake system on many turbo-charged engines to increase the volumetric efficiency. The acoustics of these devices has, however, so far gained interest from very few authors. In the last paper in this thesis a detailed acoustic analysis of CACs is presented. Models to predict sound transmission in narrow cooling tubes, including losses due to viscous and thermal boundary layers as well as turbulence, are discussed. An efficient matrix formalism based on multi-ports and including 3D effects is proposed. From this the first linear frequency domain model for CACs, which includes a complete treatment of losses in the narrow tubes and 3D effects in the connecting tanks, located on each side of the bundle of tubes, is extracted in the form of a two-port. The frequency dependent transmission loss is calculated and validated for a passenger car CAC with good accuracy.

Keywords: IC-engine, intake noise, 1D CFD, linear source data, frequency domain, 2-port, losses, air-filter, flow, FEM, BEM, charge air cooler, narrow duct, turbulence, multi-port
Licentiate thesis

This licentiate thesis consists of an introduction with a summary and the following papers:

*Paper A*
IC-engine acoustic source data from non-linear simulations.

*Paper B*
Experimental investigation of the acoustic effect of non-rigid walls in IC-engine intake systems.

*Paper C*
Prediction of IC-engine intake orifice noise using 3D acoustic modelling and linear source data based on non-linear CFD.
M. Knutsson, J. Lennblad, H. Bodén. *To be submitted.*

*Paper D*
Sound propagation in narrow tubes including effects of viscothermal and turbulent damping with application to charge air coolers.

**Division of work between the authors:**
The formulation of the problems and proposals for methodology described in this thesis has been identified in co-operation between Magnus Knutsson and the supervisors Mats Åbom and Hans Bodén. Hans Bodén also supervised and participated in the engine test bench measurements described in Paper A. The basic WAVE engine model used in the simulations in Paper A and C was designed by Johan Lennblad and refined by Magnus Knutsson. The measurements described in Paper B were performed by Ravi Varma Nadampalli as a part of his MSc thesis that was supervised by Magnus Knutsson and Hans Bodén. Magnus Knutsson has performed all acoustic simulations in the thesis, the experiments described in Paper D, and has been responsible for writing all papers.
The content of this thesis has been presented at the following conferences:

1. 12th International Congress on Sound and Vibration (ICSV12), Lisbon, Portugal, 11-14 July, 2005. “On extraction of IC-engine intake acoustic source data from non-linear simulations”.

2. 13th International Congress on Sound and Vibration (ICSV13), Vienna, Austria, 2-6 July, 2006. “Experimental investigation of the acoustic effect of non-rigid walls in IC-engine intake systems”.


5. 14th International Congress on Sound and Vibration (ICSV14), Cairns, Australia, 9-12 July, 2007. “Sound propagation in narrow channels with arbitrary cross sections and superimposed mean flow with application to charge air coolers”.


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1. Introduction

1.1. Background

Air intake noise is created in an internal combustion (IC) engine when the intake valves open to reveal the pumping motion of the pistons. High amplitude pressure pulsations travel upstream through the different components in the air intake system, see Fig. 1. The sound waves are finally radiated either through the air intake as orifice noise or through the plastic walls of the intake system as shell noise due to fluid-structural interaction. The amplitude of these resulting intake noise sources is strong enough to make them possibly significant contributors to the Pass-by-noise as well as to the sound quality impression of the vehicle. The sound reducing measures available for the intake system are either relatively space consuming or are reducing the efficiency of the engine breathing process which is directly linked to fuel consumption and CO₂ emissions. In order to meet customer demands of attractive design and reduced vehicle weight the space dedicated for the powertrain installation is often decreased. New techniques such as homogeneous charge compression ignition (HCCI), where the engine load can be controlled without a throttle, might add further complexity to the situation due to increased intake noise emissions at part load conditions. To meet these requirements, together with intensified noise legislation and the demand for shorter product development cycle times, better methods are needed to enable acoustic optimization of the air intake system. In this thesis some aspects of using linear acoustic techniques to improve predictions of intake noise are treated.

**Figure 1:** Schematic representation of the gas exchange system of a six-cylinder naturally aspirated petrol engine.
1.2. Modelling intake noise

The gas dynamics of an IC-engine can essentially be described by a set of non-linear equations for conservation of mass, momentum and energy [1]. In the general case analytical solutions to those equations can not be found and numerical models based on various approximations are necessary. A very powerful and, for IC-engine intake or exhaust ducts, often used simplification is that of considering one dimensional fields only. This assumption basically implies that the variables of pressure, density, velocity and temperature are treated as being constant over the cross-section of the duct under consideration. From this the solution of the coupled non-linear equations will be greatly simplified. Another convenient simplification is that of assuming small perturbations and perform a linearization of the governing equations. When there is a homogeneous mean flow present the final result will be the convective wave equation and then also 3D effects can be treated without too much difficulty. If only plane waves are considered, the wave equation will transfer to a 1D linear wave problem, which can be efficiently analysed via so called two-port or four-pole methods.

Within the automotive industry the most widely adopted technique for gas exchange studies is to solve the one dimensional coupled set of non-linear equations using the finite volume or finite difference method. This technique is used in several commercial softwares e.g., Ricardo/WAVE, GT-Power and AVL/BOOST, which also provide easy to use graphical user interfaces. The main purpose of these codes is for tuning of cycle averaged parameters, such as the torque and power output from engines, but unsteady pressures and flow velocities are additionally provided at positions distributed throughout the intake and exhaust systems. Hence, they can also be used for acoustic studies [2]. The boundary condition usually prescribed at any duct orifice is a fixed pressure corresponding to the ambient conditions. The predicted fluctuating velocities can thereafter, together with the assumption of spherical or hemispherical radiation, be used to calculate the noise that is emitted from the intake or exhaust orifice in a post processing step. The inherent limitation of one dimensional plane wave propagation, however, limits this technique to sufficiently low frequencies where non-plane wave effects are small. Therefore this type of model will first fail in large components such as air cleaners. Further limitations, that might not be important for simulation of engine performance but indeed for acoustics, include difficulties to apply frequency dependent boundary conditions and include effects of vibrating walls. As a result the accuracy of the predictions for intake noise is not fully reliable, neither concerning prediction of absolute sound pressure levels nor resonance frequencies.

Several authors have proposed strategies to improve the predictions of sound based on non-linear 1D gas exchange simulations. Basically there exist three main groups of methods for how this can be done; methods that use information from non-linear simulations as input to linear acoustic simulations, hybrid methods where linear information is inter-changed between the non-linear calculations and the acoustic simulations and the extension from 1D to a full solution of the 3D non-linear equations. A recent example of the first group is the work by Shaw, Moenssen and Montgomery [3], where fluctuating velocities predicted from 1D gas exchange simulations were used as input to linear boundary element simulations, but without taking into account the frequency dependence of the boundary impedance at the coupling section. Another example is the work by Fairbrother, Bodén and Glav [4] who studied the exhaust noise from a turbo-charged truck engine by using non-linear 1D computational fluid dynamics (CFD) to extract a linear time invariant source and thereafter coupled that to linear acoustic two-ports. The predictions of in-duct sound pressure levels shown were of
reasonable agreement to measurements but the free field predictions were not as good. A similar exhaust noise study was performed on a four-cylinder naturally aspirated diesel engine from a passenger car by Hota and Munjal in Ref. [5]. Here, the results at free field appear to be more accurate; however, only three discrete values of engine speed were reported. Recent examples of hybrid methods include the work by Payri, Desantes and Torregrosa [6], who also gives a good review of earlier work, and Chiavola [7]. Earlier work at MWL/KTH dealing with hybrid methods is described in Refs. [12]-[15]. An example of how non-linear effects can be included in a one-port model was proposed in Ref. [16] which also includes a literature review of different source models. The method of coupling 1D to 3D non-linear CFD is provided as a built in function in some of the commercial softwares [8] but is still not very useful for engineering noise predictions due to extremely long computational times.

The approach used in Paper C in this thesis is based upon coupling a linear source, which has been extracted from non-linear 1D gas exchange simulations, to linear acoustic transmission as was done in Ref. [4] and [5]. The validation of the source data in Paper A and of the radiated intake noise at free field in Paper C are performed using a six-cylinder naturally aspirated engine taken from a passenger car. Simulations and measurements are performed for a large number of engine revolution speeds in order to make the first systematic validation of a complete intake noise model for a wide engine speed range. The transmission models are taken from experimentally validated 3D acoustic finite element simulations in order to create a completely virtual engine.

An important limitation of linear acoustic simulations is that they can only be used when the amplitudes of the fluctuating pressures are small. Often small amplitudes are taken as a relative fluctuation amplitude of 1 % of the steady state value. This implies that a linear acoustic model is accurate enough up to at least 154 dB [ref. 2·10^5] for a steady state value of 1 bar. The sound pressure level in the duct just upstream the throttle might exceed this value why non-linear effects can be of importance, see Fig. 2. The amplitudes close to the inlet valves are definitely above this limit why the choice of a non-linear technique seems more appropriate in this region. In principle there are two effects of non-linearities on the wave propagation; wave steepening (shock forming) and too high local amplitude velocity pulsations e.g. at perforated elements and narrow constrictions. The propagation distance \( x_s \) where an initial harmonic wave becomes a shock wave is approximately [9]

\[
x_s/\lambda \approx 0.2 \cdot p_0/\hat{p}
\]

where \( \lambda \) is the wave length, \( p_0 \) the steady state pressure and \( \hat{p} \) the amplitude of the pressure pulsations. Considering the highest amplitude for the 3rd engine order, at 4400 rpm, in Fig. 2 and a steady state value of 1 bar the shock forming distance is approximately 14 m, far exceeding the length of the intake system on any passenger car. This implies that non-linear wave propagation is not very important and linear models can be used at least downstream the inlet manifold. The local non-linearities appearing at narrow constrictions or perforated elements, which can occur at lower pressure fluctuation levels, might be more important for intake systems equipped with this type of elements.
1.3. The linear time-invariant one-port source model

A model that can be used to represent an engine as a source must be able to describe the power input from the source and how incoming waves are reflected by the source. If only plane waves are considered at the source cross-section the simplest model that can be used is the linear time invariant one-port source model [10]. The condition of plane waves restricts its validity to frequencies below the cut-on frequency for the first non-planar mode. For a circular duct of radius $a$ the cut-on frequency can be calculated as

$$ f_{\text{cut-on}} = \frac{\alpha'_{01} c_0}{2\pi a} $$

where $\alpha'_{01} = 1.841$ is the first root of the Bessel function $J_0$ and $c_0$ is the isentropic speed of sound in the gas mixture. For a duct with a radius of 40 mm, which is a typical duct dimension in an intake system on passenger cars, this corresponds to a cut-on frequency of more than 2500 Hz at room temperature. For a naturally aspirated engine, where the main part of the acoustic energy from the breathing process is below this limit, the assumption of plane waves is justified. In the literature the linear time invariant one-port source model is often expressed in terms of source strength and source impedance. Convenient choices of source strength variables, often used in the literature, are pressure or volume velocity.
The relations between source variables and acoustic pressure $p'$ and volume velocity $q'$ can be expressed, with reference to the electric analogy in Fig. 3, as

$$p_S = p' + Z_0 \zeta_S q'$$  \hspace{1cm} (1.3)

or

$$q_S = p' \frac{1}{Z_0 \zeta_S} + p' \frac{1}{Z_0 \zeta_L}$$  \hspace{1cm} (1.4)

for the case of pressure source and volume velocity source respectively. Here $\zeta_S$ is the normalized source impedance, $\zeta_L$ the normalized impedance of the acoustic load, $Z_0 = \rho_0 c_0 / S$ the characteristic impedance for the gas with the density $\rho_0$ and $S$ is the area of the duct cross-section. For a perfectly linear and time invariant source the relationship between the source pressure and the source velocity is simply

$$p_S = q_S Z_0 \zeta_S.$$  \hspace{1cm} (1.5)

Several procedures to extract one-port source data are described in the literature. Basically they can be divided into direct methods, where an external source is required, and indirect or multi-load methods. A review of these different techniques is given in Ref. [11]. The approach used in Paper A and C in the present work is the indirect method. Here the two unknowns of the source are determined via a multi-load procedure. At least two different known acoustic loads are applied to the source and the acoustic pressure is extracted at the source-cross section for each load. The position of this section is where the linear source is located and it is normally just upstream the throttle, see Fig. 1. For the case of a pressure source the resulting system of equation in matrix form based on Eq. (1.3) becomes

$$\begin{bmatrix} \zeta_1 & -p_1' \\ \zeta_2 & -p_2' \end{bmatrix} \begin{bmatrix} p_S \\ \zeta_S \end{bmatrix} = \begin{bmatrix} p'_{\zeta_1} \\ p'_{\zeta_2} \end{bmatrix}.$$  \hspace{1cm} (1.6)

The corresponding system of equations for a volume velocity source using Eq. (1.4) is
\[
\begin{bmatrix}
1 & -p'_1/Z_0 \\
1 & -p'_2/Z_0
\end{bmatrix}
\begin{bmatrix}
q_s \\
1/\zeta_s
\end{bmatrix} =
\begin{bmatrix}
p'_1/(\zeta_s Z_0) \\
p'_2/(\zeta_s Z_0)
\end{bmatrix}.
\]

(1.7)

In order to reduce the effect of measurement errors and deviation from source linearity it is possible to use more than two known acoustic loads which results in an over-determined system of equations.

1.4. Acoustic modelling of special elements

In order to make accurate predictions of intake orifice noise the source as well as the transmission and the radiation must be described accurately. Examples of interesting components that are supposed to have large resistive or reactive properties affecting the transmitted sound are the air cleaner and the charge air cooler.

Several parameters can be used to describe the acoustic performance of a component in 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 Paper B, C and D 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) [17]-[20]. 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 [18]-[20]. 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 difficult in the low frequency region and with flow. Instead the so-called two-source replacement technique [21] can be 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.

1.4.1. Air cleaner boxes

The component with the largest potential to reduce noise is the air cleaner box, acting as an expansion chamber, which therefore deserves careful treatment in order to not reduce the prediction accuracy. 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 in the low frequency region are possible from treating the air cleaner as an expansion chamber where the area ratio and the length must be correctly given, see e.g. the book by Munjal [17] for an extensive description. Contribution from higher order modes to the response at higher frequencies, as is described by Glav in Ref. [22], is not possible to include analytically due to the non regular shape. More accurate sound transmission predictions can be obtained from numerical calculations using 3D linear acoustical finite elements or boundary elements, as has been shown by several authors, see e.g. the work by Herrin et al. [23] for a recent example. However, when using acoustic finite elements or boundary 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. Examples of publications describing theories for losses due to flow include Boij and Nilsson [24], Alfredsson and Davies [25], Glav [22] and Allam and Åbom [26]. Effect on sound transmission in air induction components due to fluid structural interaction has recently been treated by Marion and Ye [27] and Alex [28] who stated that the influence of non-rigid walls can be large on the intake orifice. Both works were, however, based on generic components that where not optimized by means of wall stiffness. In contrast, air intake system components are normally thoroughly reinforced by stiffeners in order to reduce the amount of sound that is radiated into the engine compartment. These reinforcements are also reducing the shift of resonance frequencies that can occur in resonators with yielding walls. The work in Paper B aims to give information on the importance of including flow and yielding walls when analysing an air cleaner box that is properly designed and is taken from a car in series production. The study also includes sound transmission measurements with and without air filter paper. Surprisingly, as it normally is neglected in 1D CFD models for predicting engine performance, the filter paper has a large effect on the transmission loss for frequencies exceeding 500 Hz and must be taken into account in a complete model. A suggestion for a filter paper model is given by Ih and Kang in Ref. [29] where a 1D model, originally developed by Allam and Åbom [26] for diesel particulate filters, is applied to extract a two-port. At frequencies above 500 Hz, where cross-modes will appear in the air cleaner and were the filter was shown to be most important, this model will not be sufficient.

1.4.2. Charge air coolers

Many turbocharged engines are equipped with charge air coolers (CAC), a device used to increase the overall performance of the engine. The cooling of the charged air results in higher density and thus volumetric efficiency. Important for petrol engines is also that the knock margin increases with reduced temperature. The parameters of main interest when designing a CAC are normally the pressure drop and the heat exchange efficiency. However, what seem to have been overlooked are the acoustic properties which are still not very well investigated. To the authors knowledge the sound attenuation properties are only dealt with in two previous publications [30], [31]. The models in both these references are making use of acoustical two-ports (or four-poles) to assemble a complete model for a CAC. However, none of them includes a complete treatment of the losses in the cooling tubes. According to the literature survey in Ref. [30], there are predictive models available describing the thermal efficiency [32], and also models treating flow unsteadiness [33]-[35] in CACs. Still they are only evaluated in terms of heat transfer performance, pressure drop and gas-exchange properties mainly affecting lower frequencies. In Ref. [36] and [37] Knutsson and Åbom have presented some initial parts of the work presented in Paper D in this thesis, which aims to make a complete description of the sound attenuating properties of a CAC when there is a mean flow present.
Most CACs consists of two of the most widely used sound attenuation measures, the reactive expansion chamber, with reference to Fig. 4 denoted by inlet/outlet tank, and the dissipative narrow cooling tubes. The assembled component offers thereby possibly underestimated capabilities for broadband noise silencing that could be used for noise optimization. Important is that the low frequency engine breathing noise resulting from the motion of the intake valves, as well as the overhearing of exhaust noise through the EGR-system, will be reduced by the CAC. It thereby opens a possibility to reduce the size of the air cleaner since less volume will be required for low frequency damping. A smaller air cleaner is easier to position in the engine compartment. As a result it will be easier to design a lay-out of the ducts with low pressure drop which is important for breathing efficiency. Since the CAC is installed downstream the compressor, see Fig 5, the noise that is radiated from the compressor and travels upstream towards the intake orifice, is of course not affected by the CAC. However, the CAC will attenuate the compressor noise that is transmitted downstream towards the engine and will thereby reduce the amount of break-out noise that radiates through the walls of the ducts situated downstream the CAC.

Figure 4: Schematic representation of a generic air-to-air charge air cooler.

Figure 5: Schematic representation of the gas exchange system of a turbo-charged diesel engine with charge air cooler.
One challenge when describing a charge air cooler is to correctly treat the propagation of sound in narrow cooling tubes with non-circular geometries for different flow speeds. The propagation of sound in narrow ducts was found by Kirchoff [38] to be dissipative because of viscous and thermal effects at the duct walls. In this work he formulated the solution to the problem, without any flow present, as a complicated, complex transcendental equation which so far has not been solved analytically. In the work by Zwikker and Kosten [39] an approximate solution to the problem was found from a set of simplified equations. An extensive overview of other models available for a wide variety of shear wave numbers was presented by Tijdeman in Ref. [40]. When flow is present the situation is somewhat more complicated and no complete theory exists. Several authors see Refs. [41]-[46] have, however, derived solutions based on simplified equations or numerical calculations. For practical applications the most useful is perhaps the ones proposed by Domumaci in Ref. [41]-[42]. In Ref. [41] he showed that the equations for sound propagation in a thermo-viscous fluid, simplified in the manner of Zwikker and Kosten theory [38], could be solved analytically for a circular pipe with a mean flow profile that is constant over the cross-section. In a later paper [42] Dokumaci extended the model from [41] to rectangular cross-sections by expanding the solution in terms of a double Fourier sine series. Other works starting out from essentially the same equations as used by Dokumaci; include Astley and Cummings [43] and Peat [44], Ih and Park and Kim [45], and Jeong and Ih [46]. At operating conditions CACs as well as catalytic converters experience temperature and pressure gradients. The effect of axial axial pressure and temperature gradients has been treated by Peat [47], Peat and Kirby [48], and Dokumaci [49]. All the models mentioned above assume laminar flow and do therefore not take into account any effect of turbulence on the propagation of sound waves. The model by Howe in Ref. [50] combines the effects of turbulence and viscothermal sub-layers on wave propagations in circular cross-sections. Although it requires the cross-section to be “wide” and the turbulent flow to be fully developed with a constant flow profile, it provides useful information and understanding concerning the low frequency damping for the cases where the Reynolds number is large. In Paper D the solution for circular ducts by Dokumaci [41], a modified version of the numerical solution scheme for arbitrary cross-sections derived by Astley and Cummings [43] and the solution accounting for turbulence in circular ducts by Howe [50] are used to model cooling tubes in order to find the most accurate solution. Two-ports representing a cooling tube are extracted from the three solutions, which has not to the authors’ knowledge previously been done for the two later. The effect of approximating cross-sections that are shaped as isosceles trapeziums with circular geometries, where the hydraulic diameter is equivalent, is studied for cases where laminar or turbulent flow is present.

Another challenge concerning charge air cooler modelling concerns coupling of several ducts into one volume, where higher order modes are present, in order to extend the frequency range. The commercial FE software LMS Sysnoise provides the ability to couple two arbitrary volumes together via two-ports [51] and could be used for this purpose. This procedure requires that the complete model is solved for each frequency for each load case; for models with high element resolution time consuming as well as expensive. A good method to shorten iteration time, if the properties of the air mixture and the geometry of the tanks are fixed, is to use the multi-port approach that is described in Paper D. The algebra for the multi-ports, based on the admittance relationships between the ports, is derived and presented in an attractive form for easy implementation.
2. Summary of the papers

2.1. Paper A: IC-engine acoustic source data from non-linear simulations

Non-linear one-dimensional (1D) CFD time domain prediction codes are used to calculate the performance of the gas exchange process for IC-engines. These softwares give time-varying pressures and velocities in the exhaust and intake systems. They could therefore in principle be used to predict radiated orifice noise. However, the accuracy is not sufficient for them to be used as a virtual design tool. More accurate results might be provided by dividing the problem into a source domain and a transmission domain and use linear 3D frequency domain codes to describe the transmission part. Radiated shell noise and frequency dependent damping could also be included in the frequency domain models. The simplest source model for simulation of the dominating engine harmonics is the linear time invariant 1-port model. The 1-port source data is usually obtained from experimental tests where the multi-load methods and especially the two-load method are most commonly used. The main limitations of these tests are that they are time consuming, expensive and require the engine, which prevents them from being used for early predictions. It would therefore be of interest to extract the acoustic source data from the existing 1D CFD gas exchange models. This paper presents a comparison between 1-port source data, obtained by measurements on a six-cylinder personal car petrol engine, and by 1D simulations of identical intake systems on the same engine. The degree of non-linearity in the results is discussed as well as the choice of source type and its relation to engine properties. The results show that it is possible to obtain reasonably accurate source strength as well as source impedance estimates, for the intake side, from 1D gas exchange simulations.

2.2. Paper B: Experimental investigation of the acoustic effect of non-rigid walls in IC-engine intake systems

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 two-port 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. Finally the measurements were repeated with the filter paper removed. The results indicate that flow related losses dominates at low frequencies while losses associated with acoustically non-rigid walls are more important at higher frequencies, giving an additional transmission loss in the order of 2 dB for the complete system. However, most important at higher frequencies is the filter paper that can add more than 10 dB at frequencies where the reactive effects in the system are small. All three effects will have to be taken into account in an accurate model. Comparisons are also made between results from an acoustic FE-model and experimental results.
2.3. Paper C: Prediction of IC-engine intake orifice noise using 3D acoustic modelling and linear source data based on non-linear CFD.

Shorter product development cycles, densely packed engine compartments and intensified noise legislation increases the need for accurate predictions of engine air inlet noise at early stages. Non-linear one dimensional (1D) computational fluid dynamic (CFD) time domain prediction codes are used for calculation of the performance of the gas exchange process of internal combustion (IC) engines. These codes give time varying pressures and velocities in the exhaust and intake system. From using the calculated velocity fluctuations at the air intake and the assumption of monopole radiation the noise emanating from the orifice can be predicted in a linear post processing step. However, the accuracy is unreliable for intake systems exhibiting a three dimensional nature. The use of linear three dimensional acoustic frequency domain methodologies such as finite elements for the transmission might provide more accurate results. In the present study linear acoustic one-port source data for a six cylinder naturally aspirated engine is extracted from 1D gas exchange analysis. The sound transmission from the source reference cross-section and through the air cleaner is represented by one single two-port that is extracted from acoustic 3D finite element analysis as well as from experiments. The radiated intake orifice noise is obtained from combining the source data and the two-port with the terminating inlet duct (“the dirty air duct”), represented by boundary elements. Finally the complete coupled model is validated using intake orifice noise measurements obtained in an engine test bench. The results are as good as, or for the case of higher frequencies even better, than the predictions from using non-linear 1D CFD for the complete system together with a linear monopole radiation model.

2.4. Paper D: Sound propagation in narrow channels including effects of viscothermal and turbulent damping with application to charge air coolers

Charge air coolers (CACs) are used on turbo charged internal combustion engines to enhance the overall gas exchange performance. The cooling of the charged air results in higher density and thus volumetric efficiency. Important for petrol engines is also that the knock margin increases with reduced charge air temperature. A property that is still not very well investigated is the sound transmission through a CAC. The losses, due to viscous and thermal boundary layers as well as turbulence, in the narrow cooling tubes results in frequency dependent attenuation of the transmitted sound that is significant and dependent on the flow conditions. Normally, the cross-sections of the cooling tubes are neither circular nor rectangular, why no analytical solution accounting for a superimposed mean flow exists. The cross-dimensions of the connecting tanks, located on each side of the cooling tubes, are large compared to the diameters of the inlet and outlet ducts. Three dimensional effects will therefore be important at frequencies significantly below the cut-on frequencies of the inlet/outlet ducts. In this study the two-dimensional finite element solution scheme for sound propagation in narrow tubes, including the effect of viscous and thermal boundary layers, originally derived by Astley and Cummings [43] is used to extract two-ports to represent the cooling tubes. The approximate solutions for sound propagation, accounting for viscothermal and turbulent boundary layers derived by Dokumaci [41] and Howe [50], are additionally calculated for corresponding circular cross-sections for comparison and discussion. The two-ports are thereafter combined with numerically obtained multi-ports, representing the connecting tanks, in order to obtain the transmission properties for the charged air
when passing the complete CAC. An attractive formalism for representation of the multi-ports based on the admittance relationship between the ports is presented. From this the first linear frequency domain model for CACs, which includes a complete treatment of losses in the cooling tubes and 3D effects in the connecting tanks is extracted in the form of a two-port. The frequency dependent transmission loss is calculated and compared to corresponding experimental data with good agreement.
3. Future research

Interesting future research concerning source data extraction includes more investigations on how to choose acoustic loads for the multi-load method. Using 1D CFD opens the opportunity of optimization of this process in order to further increase the accuracy of the estimated source data. Furthermore, it would be interesting to verify the linear time-invariant source model on a four-cylinder engine, were the reflections from the expansion at the air cleaner will affect the performance of the engine in larger extent than on a six-cylinder engine.

The task of developing an orthotropic model for air cleaner paper is the most important in order to improve simulation results, for sound propagation through air intake systems on naturally aspirated engines, above 500 Hz. An interesting reference is the work by Ih and Kang [29] who used a 1D model to simulate wave propagation through a folded filter paper. In order to extend the accuracy of this model to exceed 500 Hz three dimensional effects will have to be taken into account. The vibration of the filter paper might also influence the sound transmission and should therefore also be included in a complete model.

The charge air cooler measurements and simulations in Paper D expose the need for a wave propagation model that fully includes the effect of turbulence damping for low frequencies.

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References


Paper A
IC-Engine Intake Acoustic Source Data from Non-Linear Simulations

Magnus Knutsson
Volvo Car Corporation, Sweden

Hans Bodén
KTH CICERO, the Marcus Wallenberg Laboratory for Sound and Vibration Research, Sweden

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ABSTRACT

Non-linear 1-D CFD time domain prediction codes are used to calculate the performance of the gas exchange process for IC-engines. These softwares give time-varying pressures and velocities in the exhaust and intake systems. They could therefore be used to predict radiated orifice noise. However, the accuracy is not sufficient for them to be used as a virtual design tool. More accurate results might be provided by dividing the problem into a source domain and a transmission domain and use linear 3-D frequency domain codes to describe the transmission part. Radiated shell noise and frequency dependent damping could also be included in the frequency domain models. The simplest source model used in the low frequency plane wave range for simulation of dominating engine harmonics is the linear time invariant 1-port model. This acoustic source data is usually obtained from experimental tests where the multi-load methods and especially the two-load method are most commonly used. The main limitations of these tests are that they are time consuming, expensive and require physical hardware which prevents them from being used for early predictions. It would therefore be of interest to extract the acoustic source data from the existing 1-D CFD gas exchange models. This paper presents a comparison between acoustic source data, obtained applying the two-load technique to measurements on a six-cylinder personal car petrol engine, and to 1-D simulations of identical intake systems on the same engine. The degree of non-linearity in the results is discussed as well as the choice of source type and its relation to engine properties. The results show that it is possible to obtain reasonably accurate source strength as well as source impedance estimates, for the intake side, from 1-D gas exchange simulations.

INTRODUCTION

Air inlet noise is created in an IC-engine when the intake valves reveal the pumping motion of the pistons. High amplitude pressure pulsations travel upstream in the air inlet system and are thereafter radiated either through the air inlet as orifice noise or through the plastic walls of the intake system as shell noise due to fluid-structural interaction. The amplitude of these intake noise sources is strong enough making them possibly significant contributors to the Pass-by-noise legislation as well as interior sound quality. The sound reducing measures available for the induction system are either relatively space consuming or reducing the efficiency of the engine breathing and thereby increasing the fuel consumption as well as reducing maximum power output. The ever-decreasing demand for higher engine performance and the trend of smaller, more densely packed engine compartments therefore increases the demands for early prediction accuracy of intake noise.

One-dimensional CFD-codes such as AVL-BOOST, Ricardo-WAVE and GT-Power are used within the automotive industry for IC-engine gas exchange studies. Except for performance quantities such as volumetric efficiency, torque and power they can provide unsteady pressures and flow velocities at different positions in the intake and exhaust systems. Some of the codes also provide the ability to predict orifice noise, using calculated volume velocity at the orifice and monopole radiation as a linear post processing step. However, the prediction accuracy is not fully reliable neither concerning absolute sound pressure level nor resonance frequency estimation. Possible reasons for this might be the monopole assumption, the zero-pressure boundary condition applied at the orifice in the non-linear solution, or just the fact that the geometry of the air-cleaners and the ducts are highly three-dimensional. An interesting approach to improve the accuracy of the predictions is to use linear three-dimensional frequency domain codes to describe the sound transmission in the air-cleaner, the dirty air duct and for the orifice, sound radiation to free space [1]. To be able to calculate orifice noise and insertion loss as well as radiated shell noise using these codes, information about the engine as an acoustic source is needed. The objective of the work presented here is to
study if such source data can be extracted from 1-D CFD simulations. This idea is not completely new, it has been attempted, for exhaust systems, in [2] with low success and in [3, 4] with reasonable accuracy. However, the later work concerned the exhaust side of a turbo charged truck diesel engine while this study deals with the intake side of a naturally aspirated personal car petrol engine with an intake manifold tuned to support pulse charging. The obvious differences between those two studies are the higher temperatures and sound pressure levels appearing in the exhaust system. Less obvious but nevertheless, for source data estimation, important differences are the larger source-load interaction for the intake side and the wider engine speed range for the personal car petrol engine. Also interesting to investigate is what type of linear source model that best represents the behavior of the intake side of an IC-engine. In the following these three later aspects will be studied and discussed.

THEORY

LINEAR TIME INVARIANT SOURCE MODEL

If only plane waves are considered in the clean air intake duct, the most simple source model that can be used to represent the engine is the linear, time invariant, frequency dependent, active one-port source model [5]. The plane wave assumption restricts its validity to frequencies below the cut-on frequency. For a circular duct with radius $a$ the cut-on frequency can be calculated using Bessel’s equation as

$$f_{\text{cut-on}} = \frac{1.841 c_0}{2 \pi a},$$  (1)

where $c_0$ is the speed of sound in the medium. For a duct with a radius of 50 mm this corresponds to a cut-on frequency of 2000 Hz at room temperature. For a naturally aspirated engine, the main part of the acoustic energy will be below this limit, justifying the choice of the one-port model.

In the literature the one-port source model is often expressed in terms of source strength and source impedance. For the case of the pressure source in Figure 1, this can expressed as

$$P = P_S - Z_0 \zeta_S Q,$$  (2)

where $P_S$ denotes the source pressure, $P$ and $Q$ the acoustic pressure and volume velocity respectively, $Z_0$ the characteristic impedance of the fluid and finally $\zeta_S$ the normalized source impedance.

For the case of a velocity source, as shown in Figure 2, the corresponding expression yields

$$Q_S = P_S \frac{1}{Z_0 \zeta_S} + P \frac{1}{Z_0 \zeta_r},$$  (3)

where $Q_S$ is the source volume velocity and $\zeta_r$ the normalized load impedance. If the source model is perfectly linear and time invariant the relationship between the source pressure and the source velocity is simply

$$P_S = Q_S Z_0 \zeta_S.$$  (4)

The source strength and source impedance are known to be strongly frequency dependent; additionally they depend on the engine speed as well as the engine load. For intake systems on naturally aspirated petrol engines the most interesting engine load case is full load, when the throttle is wide open, which is the situation occurring during a Pass-by-noise verification. Moreover, most of the part load conditions will suffer from choked flow through the throttle, preventing any pressure waves from propagating upstream in the intake system.
TWO-LOAD METHOD

Equation (2) and (3) has two unknowns each, which means that they can be solved from applying two complex equations. These two equations can be obtained from using two acoustic loads; in this context two different air intake systems connected to the source. This implies that the unknown variables are independent of the acoustic load. For the case of a pressure source the resulting system of equation in matrix form is

\[
\begin{bmatrix}
-\frac{P_1}{Z_0} & P_S \\
-\frac{P_2}{Z_0} & \frac{1}{Z_S}
\end{bmatrix}
\begin{bmatrix}
\zeta_1 \\
\zeta_2
\end{bmatrix}
= \begin{bmatrix}
P_1 \\
P_2
\end{bmatrix}
\begin{bmatrix}
\zeta_1 \\
\zeta_2
\end{bmatrix}.
\]

(5)

This is the basis for the so-called Two-Load Method for extracting source data [5, 6]. If more than two acoustic loads are applied an over-determined system of equations arises which can be used to reduce effects from deviation from source linearity and measurement errors. The corresponding system of equations for a volume velocity source is

\[
\begin{bmatrix}
1 & -\frac{P_1}{Z_0} & \frac{1}{Z_S} \\
1 & -\frac{P_2}{Z_0} & \frac{1}{Z_S}
\end{bmatrix}
\begin{bmatrix}
Q_S \\
\frac{1}{Z_S}
\end{bmatrix}
= \begin{bmatrix}
P_1/\zeta_0 \\
P_2/\zeta_0
\end{bmatrix}.
\]

(6)

To validate the chosen source model it is of interest to get an estimate of to what degree the source deviates from linearity. For this purpose a linearity test, proposed by Bodén and Albertsson [7], similar to the coherence function can be used. If a problem with \( m \) complex unknowns is assumed and \( n \) measurements are performed, an over-determined equation system can be written as

\[
A \cdot x = b,
\]

(7)

where \( A \) is an \( n \times m \) matrix, \( x \) an \( m \times 1 \) vector and \( b \) an \( n \times 1 \) vector. The used linearity coefficient is defined as

\[
\gamma^2 = x^{-1} \cdot x = b^{-1} \cdot A^{-1} \cdot b,
\]

(8)

where \( A^{-1} \) is interpreted as the pseudo-inverse of \( A \). This linearity coefficient will have a value in the interval \( 0 \leq \gamma^2 \leq 1 \), where the upper limit represents a perfect linear relationship.

NON-LINEAR 1-D CFD

The commercial software used for all numerical calculations in this study is Ricardo/WAVE. This non-linear code uses the finite volume approach to solve the 1-D compressible gas dynamics equations for mass, energy and momentum. The solver, which is explicit in the time domain, calculates flow quantities at the finite volume boundaries, while scalar properties such as pressure and temperature are predicted at the volume centers. As the calculated time data is aimed for transformation to the frequency domain, the solver has been forced to take time steps of equal length in order to avoid interpolation and resampling. Bearing in mind the effect of aliasing, all calculated time steps are requested as output and used in the fast Fourier transform. The time step size used in this study is 0.0001 s; chosen since it is the smallest step normally used by the solver to find a converged solution. Altogether with the Nyquist theorem this yields a maximum frequency of 5000 Hz. As this is far above the cut-on frequency for the ducts, the frequency range of interest for this study is well covered.

TEST METHODS

SOURCE CHARACTERISATION USING MEASUREMENTS AND 1-D CFD

To verify the possibility to extract linear source data from non-linear 1-D CFD, a test case has been measured on an engine as well as predicted using Ricardo/WAVE. The engine used in the test set up was a 6 cylinder petrol engine with a swept cylinder volume of 3.2 liters. A generic air intake system consisting of PVC ducts with a wall thickness of 3.6 mm and a cubic shaped air cleaner box with a volume of 10 liters, see Figure 3, was used as acoustic load to simplify the corresponding 1-D modeling. Acoustic pressures were measured in a straight duct at three positions with different separation distances in order to extend the valid frequency range. The acoustic load impedances were thereafter evaluated using the two-microphone method. In the numerical non-linear simulations the acoustic pressures and velocities were extracted at positions close to the left microphone in Figure 3 and thereafter transferred, using the 4-pole matrix for a straight duct with flow, to match the measurement evaluation section. The numerical acoustic load impedances were obtained directly by Fourier transformation of the unsteady pressures and velocities.

Figure 3. Generic air intake system used for experimental and numerical source estimation. The section where the source data is evaluated is located at the microphone to the left. The engine is connected to the left side of the system.
Measurements and simulations were made for an over-determined system with six different acoustic loads, obtained from using the same intake system with a variable length quarter wave resonator connected to the clean air duct, 1500 mm upstream the throttle. The six different lengths for the quarter-wave resonator were: 0, 380, 635, 985, 1430 and 1910 mm. To capture the behavior of an engine run-up, a large number of operation points were measured. The engine was held at steady state operating conditions and the pressures were measured at 1700 rpm and at every 100 rpm between 2000 and 3700 rpm at full load (WOT). Finally the predicted and measured order contents of the source data were compared in the frequency domain.

**INTERACTION BETWEEN SOURCE AND ACOUSTIC LOAD**

The tuning of the torque curve for a naturally aspirated IC-engine is known to be highly sensitive to the basic volumes and duct lengths in the air intake system. Most IC-engines with small overlap between the inlet valves are, as a matter of fact designed bearing in mind the super charging effect of the reflection from the air-cleaner box or the air inlet orifice. Since the engine in this study has six cylinders and thereby a relatively large inlet-valve-overlap, it can be expected that there is less favorable effects from the air-cleaner reflections. Nevertheless, by comparing the torque curves obtained from calculations where different acoustic loads have been applied, this possible source-load interaction can be checked and its effect on the source data can be investigated.

![Figure 4. 6-2-1 configuration of the inlet manifold mounted on the six-cylinder engine used in all measurements and simulations.](image)

The attached intake manifold is variable with the possibility to use several configurations of plenums and runners. However, all the measurements and simulations presented in this paper were performed in the same operating mode. This manifold configuration, as is shown in Figure 4, consists of a two-bank system where two groups of three cylinders are connected to one plenum each, followed by two longer secondary runners joining in a second plenum just downstream the throttle. Since the secondary runners are of almost equal length there will be cancellation effects taking place in the secondary plenum, making the 3rd harmonic of the engine speed the first frequency with high amplitudes. If the secondary plenum is large enough and the manifold is completely symmetric with equal distances between all cylinders and the plenum, there will be little interaction between the performance of the engine and the reflections from the air-cleaner volume. Due to geometrical limitations and production purposes, this completely symmetrical system is almost impossible to achieve whereby there will always be some influence on the torque output from the reflections at the air-cleaner unit.

There is an obvious concern that this in-advance known source-load interaction effect is in contradiction to the requirement that the acoustic source properties are independent of the applied acoustic load. To investigate this issue, two completely different air intake systems were built and simulated using the 1-D software. The geometry of the first virtual system is similar to the air-intake system intended for the final installation of this engine (Figure 5) and consists of a dirty air duct with the length 430 mm; a clean air duct of 750 mm and an air-cleaner unit of 8.5 liters volume. Additionally the previously mentioned variable quarter-wave resonator is connected to the clean air duct 70 mm upstream the source section to achieve the six different acoustic loads. The source data for this new system is calculated in a section located about 70 mm upstream the throttle and compared to source data from the generic system calculated at the same distance from the throttle. The second system is the simplest possible, consisting of just a 290 mm straight duct upstream the source section. The ability to vary the acoustic load for this system is achieved from the same variable quarter-wave resonator as for the other two systems, connected 70 mm upstream the source section.

![Figure 5. Production-like air-intake system used as acoustic load in numerical study of source-load interaction.](image)
LINEARITY OF INTAKE SYSTEM SOURCE MODEL

The final purpose of this study is to investigate to what extent the calculated source model deviates from linearity. For this purpose the linearity coefficient defined in Equation (8) can be used. Another interesting method to estimate non-linear effects is to compare the source impedance calculated using the pressure source, to the corresponding obtained from the volume velocity source formulation. For a perfectly linear solution this will indeed yield exactly the same result. However, since a small deviation from linearity is expected, at least for weaker engine orders, it is useful to know whether the pressure formulation or velocity formulation yields the best solution. From this information it is thereafter possible to choose the most accurate source data as input for further calculations.

RESULTS AND DISCUSSION

SOURCE CHARACTERISATION USING MEASUREMENTS AND 1-D CFD

A personal car petrol engine is normally operated at an engine speed somewhere between 600 and 6500 rpm with maximum torque output somewhere between 3000 and 5000 rpm when naturally aspirated. For a typical truck engine the corresponding figures would be 500 to 2100 rpm with maximum torque output at 1200 rpm. For the personal car petrol engine the ability to predict a wider engine speed range with good resolution is evident. The measurements and simulations in this investigation were made at steady state conditions and the results are therefore not the same as from a slow transient run-up. However, the character of the results will be similar. Also worth highlighting is the fact that just full load conditions are studied. This choice was made due to the fact that the flow through the throttle is choked during a wide range of part load conditions and therefore no sound transmission is possible. Additionally the Pass-by noise legislation, where intake noise is known to be one of the main contributors, is based on full load conditions.

To verify the confidence level of the source data obtained from numerical predictions, they are plotted versus corresponding data obtained from measurements in Figure 6 to 9. All comparisons are made in the frequency domain since a time domain plot would be highly dominated by the strongest engine order, in this case normally the 3rd. As it is easier to relate to a sound pressure level than a sound velocity level, the source strength shown is the source pressure. The source impedance is separated into a real and an imaginary part making evaluation of the properties more straightforward. Source strength as well as source impedance data is achieved from using the pressure formulation in Equation (5).

The engine orders chosen for the comparison are the 3rd, 6th, 9th and 5th. This choice is based upon the large amplitude of the source strength for the 3rd order and its harmonics. The 5th order is chosen as it is, for this particular engine, the strongest order except for these before mentioned harmonics. The level of the half orders, such as the normally quite strong 4.5th order, could not be verified due to lack of information from the cam-shaft sensor during the tests. The upper and lower limits of the source strength plots in Figure 6 to 9 are all the same, enabling comparisons between their relative strength, although the absolute level is not shown due to company secrecy policies.

Source data for the 3rd engine order

The source data for the 3rd engine order is shown in Figure 6. The source strength data predicted by WAVE is in good agreement with those obtained by experiments. Important is that the absolute level is relatively strong in comparison to all other orders as expected. The strongest value occurs at a peak at 2500 rpm followed by a slightly weaker local maximum at 3500 rpm. For all studied engine speeds the WAVE model over-predicts the source strength data with an offset in the order of 5 dB except at the strong peak at 2500 rpm where the predicted and measured data is in close agreement.

The overall behavior of the calculated normalized source impedance is also relatively good except for just before and after 2500 rpm where the imaginary part shifts from
positive to negative and a peak appears in the real part. At these engine speeds the imaginary as well as the real part of the impedance are under-predicted by the WAVE model. The shift of sign for the imaginary part indicates the behavior of an anti-resonance; a state where losses are known to be highly important. As flow losses are mainly caused by flow separation, wall friction and turbulence which are not resolved by a 1-D solution, this is not a surprising shortcoming. The strong source pressure at this engine speed might also explain this discrepancy since the level is actually in a range where linear theory is known to be less valid.

Source data for the 6th engine order

In Figure 7 the source data for the 6th engine order is shown. The predicted source strength is in good agreement with the measured. The maximum deviation is about 10 dB, occurring at 2400 rpm where there is a local minimum. Elsewhere the prediction of the source pressure stays within 5 dB from the measured. At the higher engine speeds the numerical data tends to display a small shift in frequency indicating some frequency dependent end correction not being accurately modeled by the software. The source strength is indicated to be weaker than the 3rd order but still stronger than the remaining orders in Figure 8 and 9, making it highly important for the resulting orifice noise.

The accuracy of the predicted source impedance is very good at most engine speeds. At 2400 and 2500 there is a deviation for both the real as well as the imaginary part, coincident to negative real parts of the experimentally obtained impedance. This un-physical behavior of this source model has been noticed in previous research [6] and has been reported to still yield good results when used as input to linear calculations. A possible explanation for the negative value is the low value of the source strength corresponding to the high amplitude in the 3rd engine order at the same engine speed resulting in non-linear effects.

Source data for the 9th engine order

The confidence of the predicted source data for the 9th engine order can be judged in Figure 8. The numerically obtained source strength is less accurate than the 3rd and 6th orders; however, the deviation remains less than 10 dB and the overall trends are relatively well described. The source pressure level is about 10 dB lower than the lower harmonics making this source data less important for the overall level. Still it remains important for studies concerning sound quality.

The source impedance is reasonably well predicted. There are some negative real parts occurring in the measured as well as the predicted data. The importance of these remains to be verified in studies were the source data is coupled to linear transmission models and compared to engine test bench measurements.
Source data for the 5th engine order

The 5th engine order is supposed to show rather weak source strength. For lower engine speeds this expectation is verified in Figure 9. At higher speeds the source strength is of almost the same order of magnitude as the 6th order making the prediction accuracy important. It is indeed satisfying to conclude that the measured data is well predicted by the WAVE model. The accuracy stays within 5 dB except at 2400 rpm, an engine speed where there have been some discrepancies noticed for the earlier commented orders. At 3300 rpm, corresponding to 275 Hz, there is an unexpected dip in the experimentally obtained source pressure level. The same type of behavior is also found in Figure 7 for the 6th order at 2800 rpm, which is almost the same frequency (280 Hz). The wavelength for this frequency is 1.2 meters, possibly indicating measurement problems due to a full wavelength between the source section and the quarter wave resonator.

The source impedance displays the same order of accuracy as the previous judged orders. Negative real part of the impedance occurs at 3000 rpm with larger amplitudes in the measured data than the numerical.

INTERACTION BETWEEN SOURCE AND ACOUSTIC LOAD

The linear active one-port source model requires the acoustic source properties to be independent of the acoustic load. The most important sign of interaction between the source and load is a changed torque output from the engine when using different loads. A different torque represents different volumetric efficiency and consequently a different gas exchange process. To find out if there is such a variation between the different acoustic loads, the calculated torque is studied in this section. The curves in Figure 10 represent the maximum difference in torque output between the six possible configurations for the three different air intake systems analyzed separately. The generic system displays the largest differences; for low engine speed almost twice as much as the production-like and the short system. Common for all three systems are the large differences appearing at low engine speed followed by a decreasing trend for higher engine speed. The conclusion from this is that there is interaction taking place between the source and the load when changing the length of the quarter wave resonator; however, it affects mainly the lower engine speeds while the difference at higher engine speeds is quite small. This is not surprising since the reflections from the quarter wave resonator correspond to larger wavelengths known to be more efficient for super charging at lower engine speeds.

Figure 10. Maximum difference in torque between acoustic loads as a function of engine speed for the three different air-intake systems. Generic system indicated by solid line, production-like system by dashed line, short system by +-marked line.

Figure 9. Source data obtained from measurements (solid line) and WAVE calculations (dashed line) for the 5th engine order.
The difference in torque output from using different air intake systems can be judged in Figure 11. The curves represent the mean torque from using all six load cases for each intake system. As can be seen, the difference is in the same order of magnitude as the internal difference shown in Figure 10. However, there is a tendency of increasing differences at higher engine speed, although the differences are small.

In the following 4 figures the source data, calculated at a section positioned 70 mm upstream the throttle for the three different set-ups, is presented. The plots are arranged according to the previous pattern with source strength and normalized impedance for engine order 3, 6, 9 and 5 in separate figures. The upper and lower limits of the source strength plots are all the same, enabling comparisons between their relative strength. From Figure 12 and 13, it can be concluded that the data for the 3rd and 6th engine order from the three systems are in close agreement. The only discrepancy worth mentioning occurs for the generic system between 2300 and 2400 rpm for the 6th order. The corresponding frequency is 230 Hz with a wavelength just below 1.5 meters. The distance between the volume just downstream the throttle and the opening of the quarter wave resonator is 1460 mm in the generic system; thus making a standing wave between the two volumes a possible explanation for the discrepancy in source impedance between the generic and the production-like system. However, this theory will not explain the difference in source strength at 2400 rpm between the production-like and the short system.
Following the 9th engine order in Figure 14 it is possible to conclude that the differences appearing between the systems are at least in the same order of magnitude as in the measurement validation plot in Figure 8. The deviation in source strength is mostly less than 10 dB and the trends are relatively well described.

The 5th engine order source data described in Figure 15 exhibits more differences than order 3, 6 and 9 in the engine speed range between 2400 and 3000 rpm, a behavior not observed in the measurement validation plot in Figure 9. In all other points the correspondence between the systems is good. The deviations almost exclusively appear in a range where the source strength is weak and the impedance behaves as would be expected for an anti-resonance.

LINEARITY OF INTAKE SYSTEM SOURCE MODEL

In order to make judgments of the linearity of the source model, the linearity coefficient calculated using Equation (9) is plotted in Figure 16-19 for the earlier identified engine orders separately. The systems used for this study is the generic and the production like. Additionally the absolute value of the normalized source impedances calculated using the pressure and the volume velocity formulations are shown. It is known that an absolute value of the normalized impedance approaching infinity represents a volume velocity source, while approaching zero represents a pressure source. The unity value will represent a non reflecting boundary.

Figure 16. Absolute value of normalized source impedance and linearity coefficient for the 3rd engine order calculated from pressure formulation and velocity formulation. Generic system pressure formulation data is indicated by blue solid line, velocity data by red dashed. Corresponding blue curves are from the production-like air-intake system.
As expected from the previously obtained results, the 3rd engine order data shown in Figure 16 does not indicate any deviations from linearity. This can also be observed as very good correspondence between the impedances calculated from the two formulations. The two studied intake systems behave almost identically.

The linearity of the 6th engine order, shown in Figure 17, is not as good as the 3rd. Especially the curves representing the generic system indicates non linear effects at 2300 to 2400 rpm and above 3400 rpm. Interesting to note is that where there are disagreements between the impedances from the two formulations there are also deviations from unity for the linearity coefficient. Also interesting is the shape of the impedance curves. If the engine speed is transformed to frequency, the first part of the curve will be almost identical to the impedance from the 3rd order with corresponding frequency. One can observe a frequency dependence of the data making it shift from volume velocity source to pressure source and to velocity source again as the engine speed increases. There is, however, no coupling between the linearity coefficient, the model used to calculate the impedance and the behavior of the impedance.

The 9th order data is shown in Figure 18; here the linearity is less good compared to the 3rd and 6th order results. There are large discrepancies between the two impedance formulations apparent above 3000 rpm for both systems, consequently found in the linearity coefficient. Once again the generic system is showing the largest discrepancies indicating that the choice of this set up is less good than the shorter. Furthermore the same type of shifting between velocity and pressure source observed for the lower orders is present.

The 5th order data is shown in Figure 19; here the linearity is less good compared to the 3rd and 6th order results. There are large discrepancies between the two impedance formulations apparent above 3000 rpm for both systems, consequently found in the linearity coefficient. Once again the generic system is showing the largest discrepancies indicating that the choice of this set up is less good than the shorter. Furthermore the same type of shifting between velocity and pressure source observed for the lower orders is present.

Figure 18. Absolute value of normalized source impedance and linearity coefficient for 9th engine order calculated from pressure formulation and velocity formulation. Generic system pressure formulation data is indicated by red solid line, velocity data by red dashed. Corresponding blue curves are from the production-like air-intake system.

Figure 19. Absolute value of normalized source impedance and linearity coefficient for 5th engine order calculated from pressure formulation and velocity formulation. Generic system pressure formulation data is indicated by red solid line, velocity data by red dashed. Corresponding blue curves are from the production-like air-intake system.
Finally the 5th engine order data in Figure 19 displays a reasonably good correspondence between impedances up to 2300 rpm. Above this speed there are discrepancies apparent for either system, although the generic system shows the largest. The frequency dependent shifting between velocity and pressure source is once more displayed.

CONCLUSIONS

The possibility to extract linear acoustic source data for intake systems from 1-D CFD gas exchange simulations using Ricardo/WAVE has been tested and validated. The simulated source data for a 6-cylinder petrol engine provides reasonably accurate results when predicting fluctuating pressure and velocities in the intake system, especially for lower engine orders. This therefore seems to be a promising technique which could replace expensive, and time consuming experimental source data determination. Since noise is most efficiently reduced at the source this ability to predict source data opens up interesting possibilities for sound optimization of engine parameters affecting this source data such as intake valve profile and timing.

It has been shown from additional 1-D CFD calculations that the choice of intake system and the variation of acoustic load from using a variable quarter wave resonator affect the performance of the engine, in this paper referred to as source-load interaction. However, for the engine used in the study, this variation is comparably small. Although this interaction is existent, the predicted source data for the 3rd and 6th engine order remains almost the same for the investigated three different intake systems. The weaker 9th and 5th engine order source data is more affected by the choice of system indicating that care should be taken not to deviate in too large extent from the intended system when choosing system for source data prediction. This finding indicates that predicting intake source data may be a more delicate task than predicting exhaust source data since the tuning of the torque output for the engine is known to be less sensitive to the reactive properties of the exhaust system than the intake system.

Finally the linearity of the source data has been studied by means of a linearity coefficient and by comparisons between source impedances calculated using pressure source or velocity source formulations. The 3rd and 6th engine order shows almost no deviation from linearity and good correspondence between impedances while the weaker 5th and 9th order are occasionally more non-linear with corresponding discrepancies between source impedances. The used production-like system provides better results for these weaker orders than the generic system, once again highlighting a possible limitation in choices of systems suitable for predicting linear source data.

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CONTACT

e-mail: mknutsso@volvocars.com or hansbod@kth.se
web: www.volvocars.com or www.ave.kth.se
Paper B
EXPERIMENTAL INVESTIGATION OF THE ACOUSTIC EFFECT OF NON-RIGID WALLS IN IC-ENGINE INTAKE SYSTEMS

Magnus Knutsson*1, Hans Bodén2, and Ravi Varma Nadampalli2
1Volvo Car Corporation, 405 31 Göteborg, Sweden
2KTH Aeronautical and Vehicle Engineering, The Marcus Wallenberg Laboratory for Sound and Vibration Research, SE-100 44, Stockholm, Sweden
mknutsson@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 breakthrough 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

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

(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).

\[
TL(f) = 20\log_{10}\left[\text{abs}\left(\frac{T_{aw} + T_{ab}}{Z + T_{ba}Z + T_{bb}}\right)\right] - 10\log_{10}(4) \quad (2)
\]

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.

![Figure 3 – Air cleaner unit with filter paper](image)

**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.
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.

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 650 Hz, 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).

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.

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

Paper C
Prediction of IC-Engine Intake Orifice Noise Using 3D Acoustic Modelling and Linear Source Data Based on Non-Linear CFD

Magnus Knutsson¹,² Johan Lennblad¹ and Hans Bodén²

¹Volvo Car Corporation, SE-405 31 Göteborg, Sweden
²KTH CICERO, The Marcus Wallenberg Laboratory, Royal Institute of Technology, SE-100 44 Stockholm, Sweden

Abstract

Shorter product development cycles, densely packed engine compartments and intensified noise legislation increase the need for accurate predictions of engine air inlet noise at early stages. Non-linear one dimensional (1D) computational fluid dynamic (CFD) time domain prediction codes are used for calculation of the performance of the gas exchange process of internal combustion (IC) engines. These codes give time varying pressures and velocities in the exhaust and intake system. From using the calculated velocity fluctuations at the air intake orifice and the assumption of monopole radiation the noise emanating from the orifice can be predicted in a linear post processing step. However, the accuracy is unreliable for intake systems exhibiting a three dimensional nature. The use of linear three dimensional acoustic frequency domain methodologies such as finite elements for the transmission might provide more accurate results. In the present study linear acoustic one-port source data for a six cylinder naturally aspirated engine is extracted from 1D gas exchange analysis. The sound transmission from the source reference cross-section and through the air cleaner is represented by one single two-port that is extracted from acoustic 3D finite element analysis as well as from experiments. The radiated intake orifice noise is obtained from combining the source data and the two-port with the terminating inlet duct (“the dirty air duct”), represented by boundary elements. Finally the complete coupled model is validated using intake orifice noise measurements obtained in an engine test bench.

KEYWORDS

IC-engine, CFD, linear source data, frequency domain, two-port, FEM, BEM, intake noise
1. Introduction

Air intake noise is created in an internal combustion (IC) engine when the intake valves open to reveal the pumping motion of the pistons. High amplitude pressure pulsations travel upstream through the different components in the air intake system, see Fig.1. The sound waves are finally radiated either through the air intake as orifice noise or through the plastic walls of the intake system as shell noise due to fluid-structural interaction. The amplitude of these resulting intake noise sources is strong enough making them possibly significant contributors to the Pass-by-noise as well as to the sound quality impression of the vehicle. The sound reducing measures available for the intake system are either relatively space consuming or are reducing the efficiency of the engine breathing process which is directly linked to fuel consumption and CO₂ emissions. In order to meet customer demands of attractive design and reduced vehicle weight the space dedicated for powertrain installation is often decreased. New techniques such as homogeneous charge compression ignition (HCCI), where the engine load can be controlled without a throttle, might add further complexity to the situation due to increased intake noise emissions at part load conditions. To meet these requirements, together with intensified noise legislation and the demand for shorter product development cycle times, better methods are needed to enable further acoustic optimization of the air intake system.

Figure 1: Schematic representation of the gas exchange system of a six-cylinder naturally aspirated petrol engine.

The gas dynamics of an IC-engine can essentially be described by a set of non-linear equations for conservation of mass, momentum and energy [1]. In the general case analytical solutions to those equations can not be found and numerical models based on various approximations are necessary. A very powerful and, for IC-engine intake or exhaust ducts, often used simplification is that of considering one dimensional fields only. This assumption basically implies that the variables of pressure, density, velocity and temperature are treated as being constant over the cross-section of the duct under consideration. From this the solution of the coupled non-linear equations will be
greatly simplified. Another convenient simplification is that of assuming small perturbations and perform a linearization of the governing equations. When there is a homogeneous mean flow present the final result will be the convective wave equation and then also 3D effects can be treated without too much difficulty. If only plane waves are considered, the wave equation will transfer to a 1D linear wave problem, which can be efficiently analysed via so called two-port or four-pole methods.

Figure 2: Wave model for the six-cylinder naturally aspirated petrol engine used in the present study [2].

Within the automotive industry the most widely adopted technique for gas exchange studies is to solve the one dimensional coupled set of non-linear equations using the finite volume or finite difference method. This technique is used in several commercial softwares e.g., Ricardo/WAVE, GT-Power and AVL/BOOST, which also provide easy to use graphical user interfaces, see Fig. 2. The main purpose of these codes is for tuning of cycle averaged parameters, such as the torque and power output from engines, but unsteady pressures and flow velocities are additionally provided at positions distributed throughout the intake and exhaust systems. Hence, they can also be used for acoustic studies [2]. The boundary condition usually prescribed at any duct orifice is a fixed pressure corresponding to the ambient conditions. The predicted fluctuating velocities can thereafter, together with the assumption of spherical or hemispherical radiation, be used to calculate the noise that is emitted from the intake or exhaust orifice in a post processing step. The inherent limitation of one dimensional plane wave propagation, however, limits this technique to sufficiently low frequencies where non-plane wave effects are small. Therefore this type of model will first fail in large components such as air cleaners. Further limitations, that might not be important for simulation of engine performance but indeed for acoustics, include difficulties to apply frequency dependent boundary conditions and include effects of vibrating walls. As a result the accuracy of the predictions for intake noise is not fully reliable, neither concerning prediction of absolute sound pressure levels nor resonance frequencies.

Several authors have proposed strategies to improve the predictions of sound based on non-linear 1D gas exchange simulations. Basically there exist three main groups for how this can be done: methods that uses information from non-linear simulations as
input to linear acoustic simulations, hybrid methods where linear information is inter-
changed between the non-linear calculations and the acoustic simulations and the
extension from 1D to a full solution of the 3D non-linear equations. A recent example of
the first group is the work by Shaw, Moenssen and Montgomery [3], where fluctuating
velocities predicted from 1D gas exchange simulations were used as input to linear
boundary element simulations, but without taking into account the frequency
dependence of the boundary impedance at the coupling section. Another example is the
work by Fairbrother, Bodén and Glav [4] who studied the exhaust noise from a turbo-
charged truck engine by using non-linear 1D computational fluid dynamics (CFD) to
extract a linear time invariant source and thereafter coupled that to linear acoustic two-
ports. The predictions of in-duct sound pressure levels shown were of reasonable
agreement to measurements but the free field predictions were not as good. A similar
exhaust noise study was performed on a four-cylinder naturally aspirated diesel engine
from a passenger car by Hota and Munjal in Ref. [5]. Here, the results at free field
appear to be more accurate; however, only three discrete values of engine speed were
reported. Recent examples of hybrid methods include the work by Payri, Desantes and
Torregrosa [6], who also gives a good review of earlier work, and Chiavola [7]. The
method of coupling 1D to 3D non-linear CFD is provided as a built in function in some
of the commercial softwares [8] but is still not very useful for engineering noise
predictions due to extremely long computational times.

The approach used to predict intake orifice noise in the present paper is from the
first group. A linear source, predicted using non-linear CFD, is coupled to 3D acoustic
FE for the transmission and BE for the radiation to free field. The differences compared
to earlier work is that it deals with intake noise instead of exhaust noise and that it
includes response predictions from using 3D linear acoustic FE coupled to a linear time
invariant one-port predicted from non-linear CFD. Important is also that simulations and
measurements are performed for a large number of engine revolution speeds in order to
make the first systematic validation of a complete intake noise model for a wide engine
speed range.

2. Linear time invariant one-port source model

2.1. General

A model that can be used to represent an engine as a source must be able to describe
the power input from the source and how incoming waves are reflected by the source. If
only plane waves are considered at the source cross-section the simplest model that can
be used is the linear time invariant one-port source model [9]. The condition of plane
waves restricts its validity to frequencies below the cut-on frequency for the first non-
planar mode. For a circular duct of radius $a$ the cut-on frequency can be calculated as

$$f_{\text{cut-on}} = \frac{\alpha'_{01} c_0}{2\pi a}$$

(2.1)

where $\alpha'_{01} = 1.841$ is the first root of the Bessel function $J'_0$ and $c_0$ is the isentropic
speed of sound in the gas mixture. In the literature the linear time invariant one-port
source model is often expressed in terms of source strength and source impedance.
Convenient choices of source strength variables, often used in the literature, are pressure
or volume velocity.
The relations between source variables and acoustic pressure \( p' \) and volume velocity \( q' \) can be expressed, with reference to the electric analogy in Fig. 3, as

\[
p_S = p' + Z_0 \zeta_S q' \tag{2.2}
\]

or

\[
q_S = p' \frac{1}{Z_0 \zeta_S} + p' \frac{1}{Z_0 \zeta_L} \tag{2.3}
\]

for the case of a pressure source and a volume velocity source respectively. Here \( \zeta_S \) is the normalized source impedance, \( \zeta_L \) the normalized impedance of the acoustic load, \( Z_0 = \rho_0 c_0 / S \) the characteristic impedance for a propagating plane wave in duct, with cross-sectional area \( S \), filled with gas with the density \( \rho_0 \). For a perfectly linear and time invariant source the relationship between the source pressure and the source velocity is simply

\[
p_S = Z_0 \zeta_S q_S. \tag{2.4}
\]

Figure 3: Equivalent acoustic circuits for linear time invariant one port source model.
(a) Pressure source. (b) Volume velocity source.

Several procedures to extract one-port source data are described in the literature. Basically they can be divided into direct methods, where an external source is required, and indirect or multi-load methods. A short review of these different techniques is given in Ref. [10]. The approach used in the present work is the indirect method. Here the two unknowns of the source are determined via a multi-load procedure. At least two different known acoustic loads are applied to the source where after the acoustic pressure is extracted at the source cross-section. The position of this section is where the linear source is located and it is normally just upstream the throttle, see Fig. 1. For the case of a pressure source the resulting system of equation in matrix form based on Eq. (2.2) becomes

\[
\begin{bmatrix}
\zeta_1 & -p'_1 \\
\zeta_2 & -p'_2
\end{bmatrix}
\begin{bmatrix}
p_S \\
q'_S
\end{bmatrix}
= \begin{bmatrix}
p'_1 \zeta_1 \\
p'_2 \zeta_2
\end{bmatrix}. \tag{2.5}
\]

The corresponding system of equations for a volume velocity source is
\[
\begin{bmatrix}
1 & -p_1'/Z_0 \\
1 & -p_2'/Z_0
\end{bmatrix}
\begin{bmatrix}
q_S \\
1/\xi_S
\end{bmatrix} =
\begin{bmatrix}
p_1'/(\xi_1 Z_0) \\
p_2'/(\xi_2 Z_0)
\end{bmatrix}.
\]

(2.6)

In order to reduce the effect of measurement errors and deviation from source linearity it is possible to use more than two known acoustic loads which results in an over-determined system of equations.

2.2. Source data prediction using non-linear 1D CFD

Most source data in the literature has so far been obtained from measurements. As the speed range of engines for a passenger cars is wide, normally at least 1000 to 6000 rpm for petrol engines, a huge number of measurements are required in order to describe all possible load situations. For instance the investigation by Knutsson and Bodén in Ref. [12], where six different acoustic loads were used, covered engine speeds of 2000 to 3700 rpm in steps of 100 rpm; in total 108 measurements. Except for the drawback that this procedure is expensive and time consuming it also requires physical hardware which normally is not available at early project stages. A procedure to estimate the linear source data virtually is therefore of great interest. Lately, some attempts have been reported were 1D gas exchange simulation softwares have been used to extract source data with relatively good accuracy, see Ref. [4], [11] and [12]. The engine used in the present study is basically the same as in Ref. [12] but here the intake system is properly designed for mass-production, which was not the case in Ref. [12] where a generic system with a very long clean air duct was used to enable the source data extraction. However, the method to extract the source data and the approach to vary the acoustic load with a quarter-wave resonator with variable length is identical to the work in Ref. [12] and is not described here.

The simulation model is based on the commercial software Ricardo/WAVE, see Fig. 2. This non-linear 1D CFD code uses the finite volume approach to solve the compressible gas dynamic equations for mass, energy and momentum. The solver, which is explicit in the time domain, calculates flow quantities at the finite volume boundaries, while scalar properties such as pressure and temperature are predicted at the volume centres. The time step size used in this study is 0.0001 s, which according to the Nyquist theorem yields a maximum frequency of 5000 Hz if the calculated data is periodic. Twenty engine cycles are simulated for each engine speed to ensure that the solution is converged and harmonic conditions are obtained. An average of the five last cycles is used to calculate the source data.

3. Theory for sound transmission through air intake systems

3.1. General

An air intake system consists in principle of a duct, a volume and a duct coupled in series (cascade) before the air enters the inlet manifold, see Fig. 1. Following the air flow the first duct is here referred to as the dirty air duct while the duct between the air cleaner and the inlet manifold is called the clean air duct. The air cleaner box located between these two ducts behaves acoustically as an expansion chamber. The volume of the air cleaner is therefore the most important property for reducing the low frequency noise resulting from engine breathing. Another, from acoustic point of view very important, component located in the intake system is the throttle that is used to control
the load of the engine. At most part load conditions, where the throttle is not fully opened, the flow through the throttle will be choked and no sound waves can propagate from the inlet manifold into the upstream intake system. All results in this study are obtained at full load conditions when the throttle is wide opened (WOT), which is the load case occurring at Pass-by-noise verification.

3.2. Two-port theory

Assuming plane waves at the openings of an acoustic element, the sound propagation can be described using acoustic two-ports [13]. This will be consistent as long as the highest frequency of interest stays well below the cut-on frequency for the first non-planar wave at the openings of the two-port. It is not necessary that the waves are planar within the two-port; however, it might be difficult to describe it analytically if they are not. The frequency domain relationship between the acoustic states at the sections representing the inlet and outlet of a two-port can be written

\[
\begin{bmatrix}
    p_{\text{in}}' \\
    q_{\text{in}}'
\end{bmatrix}
= T_{11} T_{12}
\begin{bmatrix}
    p_{\text{out}}' \\
    q_{\text{out}}'
\end{bmatrix}
\]  

(3.1)

where \( p' \) denotes the acoustic pressure and \( q' \) the acoustic volume velocity for a plane wave. For simple generic geometries, such as circular ducts, it is possible to obtain the complex valued components in the transfer-matrix from analytical expressions. In the low frequency region the air cleaner can be treated as an expansion chamber where the area ratio and the length must be correctly given, see e.g. the book by Munjal [13] for an extensive description. The contribution from higher order modes to the response at higher frequencies, as is described by e.g. Glav in Ref. [14], is not possible to include analytically due to the non regular shape. For a component built up of three sub-components coupled in series, such as an air intake system, an assembled two-port can be calculated from coupling the two-port matrices of the clean air duct (CLEAN-DUCT), the air cleaner (AC) and the dirty air duct (DIRTY-DUCT) in cascade as

\[
T_{\text{INTAKE-SYSTEM}} = T_{\text{CLEAN-DUCT}} T_{\text{AC}} T_{\text{DIRTY-DUCT}}.  
\]  

(3.2)

Here, it is assumed that the two-ports are directed from the source (the engine) towards the air intake orifice. If additional components, such as side-branch resonators are present, their two-ports will just add an extra component in Eq. (3.2).

3.3. Prediction of radiated orifice noise using two-port theory

The procedure to calculate radiated orifice noise, utilizing a linear 1D one-port source coupled to two-ports is well known. For the convenience of the reader the basic steps are given in the following. With reference to Fig. 4, the relation between the acoustic variables at position 1 and at the orifice at position 3, for a system with two-ports coupled in series, is

\[
\begin{bmatrix}
    p_1' \\
    q_1'
\end{bmatrix}
= T_A [T_B] p_{\text{ORIFICE}}'.  
\]  

(3.3)
Insertion of the pressure at position 1, \( p_1' \), given by the one-port source relation in Eq. (2.2), together with an appropriate termination radiation impedance yields

\[
\begin{bmatrix}
\ p_S - Z_0 \zeta_S q_1' \\
\ q_1'
\end{bmatrix} = \begin{bmatrix} [T_A] & [T_B] \end{bmatrix} \begin{bmatrix}
\ Z_0 \zeta_{\text{RAD}} q_{\text{ORIFICE}}' \\
\ q_{\text{ORIFICE}}'
\end{bmatrix}.
\]  
(3.4)

The acoustic volume flow at the orifice can now be solved as

\[
q_{\text{ORIFICE}}' = \frac{p_S}{Z_0 \zeta_{\text{RAD}} T_{11} + T_{12} + Z_0 \zeta_{\text{RAD}} Z_0 \zeta_S T_{21} + Z_0 \zeta_S T_{22}}.
\]  
(3.5)

where \( T_{11}, T_{12}, T_{21} \) and \( T_{22} \) are the components of the two-port resulting from multiplication of \( T_A \) and \( T_B \). Assuming a monopole radiating to free field, the pressure at the distance \( r \) from the orifice can finally be calculated as

\[
p_{\text{FREE-FIELD}}' (r) = \frac{\rho_0 \omega q_{\text{ORIFICE}}'}{4\pi r} e^{-\omega r}.
\]  
(3.6)

Figure 4: Representation of engine intake system via a linear acoustic network.

For some applications, here one is described in Section 3.4, it might be convenient to move the source section along the duct network to a new position. Knowledge about the two-port representing the ducting between the two positions is required. The expressions for equivalent source strength and impedance, moved upstream using the transfer matrix \( T_A \) from position 1 to position 2, as given in Ref. [15] are with reference to Fig. 4

\[
p_{s2} = \frac{p_{s1}}{T_{A,11} + Z_0 \zeta_S T_{A,21}}.
\]  
(3.7)
and

\[
Z_{02_S} = \frac{T_{A,12} + Z_{01_S} T_{A,22}}{T_{A,11} + Z_{01_S} T_{A,21}}
\]

(3.8)

respectively. Here, \(T_{A,11}, T_{A,12}, T_{A,21}\) and \(T_{A,22}\) are the components of the two-port \(T_A\), \(Z_{01_S}\) and \(Z_{02_S}\) are the source impedances evaluated at position 1 and 2 respectively.

### 3.4. Acoustic linear 3D finite elements and boundary elements

For non-regular geometries it might be difficult or impossible to define the two-ports for the intake system components analytically. This is more likely to occur for intake systems than exhaust systems in modern passenger cars due to densely packed engine compartments. Demands for low weight, attractive design and large engines often leads to less available space for powertrain installation. The gas exchange ducting might also suffer from this with complex shapes and large curvatures on ducts as well as air-cleaners with non-regular geometries mainly based on the shape of the volume that is left between the surrounding components. Additionally, the main part of the intake systems available on passenger cars today is made of plastic which might, for bad designs, more easily interact with the sound field inside and as a result radiate shell noise to the surrounding engine compartment. This coupling will certainly affect the sound transmission in the enclosed air and thereby also the orifice noise. For complex geometries it is, however, seldom possible to analytically describe this fluid-structure interaction. The use of 3D linear acoustic finite elements or boundary elements gives a possibility to overcome these difficulties. Modelling interior cavities using acoustic FE or BE is a very mature technique and can be performed by help of most commercial FE or BE softwares. There is a large number of textbooks describing the theory, see e.g. Ref. [16], hence there is no need to describe it in detail here. Several authors have dealt with acoustic FE or BE analyses of gas exchange components, see e.g. Ref. [17] for a recent example.

Concerning extraction of a two-port from FE or BE calculations, a convenient method is the two load method [13]. In principle this method requires at least two solutions from the set-up with two different boundary conditions applied. The two-port parameters can thereafter be found using the calculated pressures and velocities at the inlet and outlet of the two-port for each load. In Ref. [18] an improved method was presented that only requires one solution of the system matrix due to appropriate choice of boundary conditions and in-advance knowledge about the relation between the two-port parameters.

Using the FE or BE method, the radiated orifice noise can be predicted by some different approaches. The first, and most simple, is to couple an indirect BE-model [19], where the orifice is open, direct to the source. Details concerning the procedure of coupling a vibrating piston to a duct with an open end are described in Ref. [3] and [20]. Secondly, the source can be transferred upstream to a new source cross-section using the predicted two-port. The equivalent source at the upstream position can thereafter be coupled to a smaller BE-model representing the remaining duct. In both two cases the BE-model of the radiation can be interchanged to an FE-model. Here, the orifice can be modelled either by using a radiation boundary condition at the opening or by using infinite elements.

In the present work it has been chosen to transfer the source model upstream to the section where the air-cleaner is connected to the dirty air duct, see Fig. 1 and position 2.
in Fig.4. The required two-port, representing the clean air duct and the air cleaner, is obtained from FE calculations as well as from experiments. The radiated orifice noise is thereafter predicted by an indirect BE calculation of the dirty air duct where the orifice is open and the source is described as a vibrating piston with a prescribed admittance.

4. Sound transmission through a mass-produced air intake system

4.1. General

In order to validate the sound transmission through the air cleaner and the clean air duct, used in the final intake orifice noise validation, the assembled two-port is validated separately. The intake system used for validation is originally developed to be fitted in a large size passenger car with a six-cylinder naturally aspirated petrol engine installed. Fig. 5 shows a 3D CAD image of the components in the system. All parts are made of stiff plastic material. The relative movement between the engine and the air cleaner is compensated for by two bellows included in the clean air duct. These bellows are made of the same material as the rest of the duct but the geometry is designed to respond weakly to bending forces. Four side-branch resonators are distributed throughout the clean air duct. The dimensions of these are given in Table 1, numbers with reference to Fig. 5, together with experimentally obtained resonance frequencies. To clean the incoming air, a filter paper is mounted inside the air cleaner. This filter paper consists of a pleated double sheet held in place by a rubber sealing that is also used as a gasket between the two parts of the air cleaner unit. The acoustic effect of the filter paper was found in Ref. [21] to be small below 500 Hz. In order to verify this conclusion for this particular system, measurements were performed with and without the filter paper mounted.

Figure 5: Air cleaner and clean air duct used for validation
### Table 1: Data for the side-branch resonators.

<table>
<thead>
<tr>
<th></th>
<th>Length of neck (mm)</th>
<th>Width of neck (mm)</th>
<th>Volume $\times 10^{-3}$ (mm$^3$)</th>
<th>Resonance frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Quarter wave resonator</td>
<td>142</td>
<td>32</td>
<td>-</td>
<td>570</td>
</tr>
<tr>
<td>2. Helmholtz resonator</td>
<td>36</td>
<td>22</td>
<td>396</td>
<td>390</td>
</tr>
<tr>
<td>3. Helmholtz resonator</td>
<td>34</td>
<td>25</td>
<td>596</td>
<td>300</td>
</tr>
<tr>
<td>4. Helmholtz resonator</td>
<td>16</td>
<td>29</td>
<td>202</td>
<td>430</td>
</tr>
</tbody>
</table>

#### 4.2. Acoustic linear 3D finite element model

All 3D FE-calculations in the present work has been performed using the commercial software LMS/Sysnoise [19]. In this code the convective effect of a superimposed incompressible mean flow can easily be accounted for, although it has been neglected here due to the small Mach numbers present. The effect of flow losses can not be predicted in the linear solution. The FE-model consists mainly of linear tetrahedral elements and linear wedge elements. The chosen element size is 10 mm which yields more than 30 elements per wavelength for analyses performed at frequencies less than 1000 Hz. The total number of elements is about 252 000, built up using 52 000 nodes. The finite element mesh is shown in Fig. 6. When using acoustical finite elements it is well known that the numerical integration of the pressure at a boundary is done exactly. However, the acoustic velocity boundary condition is obtained only in a weak sense since it is of Neumann type. The prediction of the acoustic velocity at a boundary is obtained from extrapolation of the results at the integration points inside the elements to the boundary. To account for the inaccuracy, resulting from the extrapolation, the elements at all openings are extremely short (0.1 mm).

![Figure 6: Acoustic FE-model of air cleaner and clean air duct used for model validation](image-url)
In order to extract the two-port matrix, two solutions (or load-cases) have to be performed with different boundary conditions. For both load-cases the boundary condition used at the flow inlet, see Fig. 6, is unity pressure amplitude. The boundary condition at the flow outlet is, for the two load cases, zero pressure or zero velocity respectively. The predicted acoustic variables at the flow inlet and outlet from load case 1 and 2 is used to calculate the two-port for the system from the relation

\[
\begin{bmatrix}
    p'_{1-LC1} & p'_{1-LC2} \\
    q'_{1-LC1} & q'_{1-LC2}
\end{bmatrix} =
\begin{bmatrix}
    T_{A,11} & T_{A,12} \\
    T_{A,21} & T_{A,22}
\end{bmatrix}
\begin{bmatrix}
    p'_{2-LC1} & p'_{2-LC2} \\
    q'_{2-LC1} & q'_{2-LC2}
\end{bmatrix}.
\]

(4.1)

where the index \( LCn \) denotes a variable taken for load case \( n \).

To include the effect of yielding walls and propagation losses, damping can be applied as a complex speed of sound. Based on the investigation by Knutsson et al. in Ref. [21], an engineering estimate for the damping in intake system components made of plastic is obtained by putting the imaginary part of the speed of sound equal to one percent of the real part. The relevance of this assumption is demonstrated by the results presented in Section 4.4. No further coupling to the structure is included in the simulations meaning that some deviations from the experimental results are expected. Furthermore, no effect of the filter paper is included in the FE-model.

### 4.3. Experimental setup

The validation measurements of the proposed model have been performed using the flow acoustic test facility available at MWL/KTH. All experiments were done at room temperature for different flow speeds as well as without flow. The Mach number in the measurement ducts was varied between 0.05 and 0.1 in steps of 0.025; values chosen as being representative for engine operating conditions. The test ducts used during the experiments consisted of standard steel pipes, with diameters equal to 66 mm, chosen in order to relate to the in- and outlet of the clean air duct and the air cleaner. Eight loudspeakers, equally divided between the up- and downstream side of the rig, were used as acoustic sources, as shown in Fig. 7 and 8. The test rig is terminated at each end by a dissipative silencer and a horn to reduce the effects of standing waves. Fluctuating pressures were measured by using six condenser microphones (Brüel &Kjaer ¼-inch 4938) flush mounted in the wall of the steel pipes. All measurements were performed using random noise excitation. The fluctuating pressures were transformed to the frequency domain with a resolution of 5 Hz and 4000 averages. The two-port matrix for the test object was obtained using the source switching technique as described in Ref. [19]. From the measured two-port data the transmission loss was calculated using the expression [13]

\[
TL = 10 \cdot \log \left\{ \frac{1 + M_{IN}}{1 + M_{OUT}} \right\}^2 \left[ Z_{OUT} \left( T_{11} + \frac{T_{12}}{Z_{OUT}} + \frac{T_{21}}{Z_{IN}} \frac{Z_{IN} T_{22}}{Z_{OUT}} \right) + Z_{IN} \right] \}
\]

(4.2)

and thereafter compared to the transmission loss predicted from theory. The Mach numbers and the acoustic wave impedances in the in- and outlet of the two-port are denoted \( M_{IN}, M_{OUT}, Z_{IN} \) and \( Z_{OUT} \) in Eq. (4.2). To minimize the effects of flow noise at the microphones, source correlation using the loudspeaker voltage signal was performed.
4.4. Experimental validation

As flow loss effects can not be predicted by the linear acoustic model the validation is limited to the case without flow. The filter paper is removed from the air cleaner and all possible leakages have been sealed. The validations are performed in the frequency range between 75 and 1000 Hz. The lower frequency limit is chosen to represent the 3rd engine order at an engine revolution speed of 1500 rpm. Above 1000 Hz the acoustic energy from engine breathing is assumed to be small. The limiting frequency for this validation technique is based on the microphone separation in the two-microphone method or the cut-on frequency of the measurement ducts and is for this experimental set-up about 1600 Hz. All resonators are maintained which can clearly be seen as four distinct peaks in the transmission loss curves in Fig. 9. The transmission loss that has been predicted using FE-models with two different values on the speed of sound is
shown; \( c = c_0(1 + i/200) \) and \( c = c_0(1 + i/100) \), where \( c_0 \) is the isentropic speed of sound and \( i \) is the imaginary number.

The accuracy of the predictions is good except at the frequencies where the side-branch resonators are contributing. At these frequencies the predictions are slightly shifted and the peaks appear at frequencies that are about 30 Hz higher than in the experimental results. This could be a result of yielding walls in the resonators, viscous effects at the entrances of the resonators or simply due to deviations in the CAD geometry used to create the FE-model. The clean air duct is manufactured using a blow-moulding technique where the tolerance is not as fine as can be expected when using injection moulding. Moreover, the speed of sound where the imaginary part is one percent of the real part gives clearly the best predictions.

The acoustic attenuation properties of the system are relatively good except below 100 Hz and at about 800 Hz where the transmission loss is less than 5 dB. Another property worth commenting is the linearity of the curve with respect to frequency. Engine harmonics passing the stop bands due to the resonators will vary rapidly when the engine speed is changed. To create a good sound quality impression of the vehicle a smoother curve might be preferred.

![Figure 9: Transmission loss for the clean air duct and the air cleaner. —, Experimental without paper, without flow; ..., 3D FE without paper, without flow (speed of sound \( c = c_0(1 + i/200) \); ----, 3D FE without paper, without flow (speed of sound \( c = c_0(1 + i/100) \)).](image)

4.5. Experimental results - Effect of flow and filter paper

Since the flow losses and the filter paper are not included in the FE-model it is of interest to quantify the acoustic effect of these on this particular system. Helmholtz resonators are known to be particularly sensitive to flow why it is of interest to see to what extent the three present resonators in this system are affected. Fig. 10 shows the transmission loss for the air intake system with and without filter paper mounted. The effect of flow is also visualized. Measurements were performed for flow speed in the measurement ducts of Mach = 0.05 to Mach = 0.1 in steps of 0.025, however, only
values for the case without flow and the highest flow speed is presented in the following figure. The effect from flow on transmission loss appears to be linearly varying with the flow speed and the results from the other flow speeds will stay in between the curves shown, hence they are not included in the figure. As was concluded in Ref. [21], the filter paper is mainly affecting higher frequencies. Below 500 Hz there is almost no difference at all for measurements taken with and without filter paper, independently of the flow conditions. Above 500 Hz, at frequency bands where the attenuation is low, the filter paper might add as much as 5 to 10 dB damping. The effect of flow is not as pronounced but the flow will give about 1 to 2 dB damping for almost all frequencies except at the peaks caused by the side-branch resonators. The damping at the peaks is reduced with flow except for the peak due to the quarter-wave resonator. Interesting is that the peak at the 300 Hz, due to resonator no. 3, is more affected by flow than the others. Possible explanations for this might be that the shape of the neck is a rectangle with rather large aspect ratio or that it is positioned in the part of the duct just after the 90 degree bend. Flow separation at the inside of the bend might result in locally higher velocities at the mouth of the resonator, hence reducing its efficiency.

![Figure 10](image_url)

Figure 10: Experimentally obtained transmission loss for the clean air duct and the air cleaner.

- , Without paper, without flow; ----, Without paper, with flow ($M = 0.1$); -·-·, With paper, without flow; ····, With paper, with flow ($M = 0.1$).

5. Experimental validation of source + transmission + radiation

5.1. General

The engine used for validation of the proposed linear methodology, where a linear one-port is coupled to linear transmission and radiation, is taken from a passenger car and is mass-produced. It is a naturally aspirated six-cylinder petrol engine with a swept cylinder volume of 3.2 litres and an inlet manifold made of glass-fibre reinforced polyamide. The air intake system is identical to the system that was investigated in section 4 with four side-branch resonators on the clean air duct. The dirty air duct is
short, about 280 mm, and consists of a circular duct that is slightly expanding just before the orifice. The shape of the cross-section is changing with the expansion and the orifice is, as a result, shaped as a rectangle. The CAD geometry representing the dirty air duct is shown in Fig. 12 together with the BE-model used for prediction of radiated noise as was described in Section 3.4. The BE-model consists of 835 four-node quad-elements, 38 triangle-element and 879 nodes. The radiated noise is calculated at a field point, located 100 mm from the midpoint of the orifice. No effects of mirror sources due to reflections at the ground are included in the predictions. The distance to the orifice from the field point is much smaller than the distance to the floor why the contribution from a mirror source is assumed to be negligible compared to the sound from the orifice.

Figure 11: 3D CAD geometry and BE-model representing the dirty air duct.

5.2. Intake orifice noise measurements

Measurement of intake orifice noise was performed in a semi-anechoic engine test cell at Volvo Car Corporation. The engine was installed with the intention to maintain the gas exchange properties that is present in the vehicle installation. However, the exhaust system had to be changed to a much longer system with an extra muffler, intended for use on a truck. The exhaust orifice was not situated inside the test cell. Acoustic pressures were measured by two microphones at a distance of 100 mm from the intake orifice at an angle of 45 degrees above and below the orifice respectively. The positions were chosen in order to make the noise from the intake orifice the strongest contributor, thereby masking the noise due to engine block vibration, while not disturbing the air inflow.

To simulate the behaviour of an engine rpm sweep, a large number of operation points were measured. The engine was held at steady state operating conditions and the pressures were measured at every 100 rpm between 1500 and 4000 rpm at full load (WOT). Finally the predicted and measured order contents of the radiated noise were compared in the frequency domain

5.3. Model validation

The source data used for input to the model validation is shown in Appendix A. Just the three strongest engine harmonics, the 3rd, 6th and 9th engine order, are used in the validation. The experimentally obtained intake orifice noise is compared to the corresponding predictions using 1D CFD plus monopole radiation and to linear source data coupled to two-ports obtained using acoustic finite elements and boundary elements for the radiation. This validation is shown in Fig. 12, 14 and 16 for the 3rd, 6th and 9th engine order respectively. In order to explain some of the discrepancies a
corresponding validation is shown in Fig. 13, 15 and 17, where the two-ports for the transmission are obtained from the loudspeaker measurements described in section 4. The microphones are located very close to the orifice why any deviation from the original position will cause significant errors. This can clearly be seen in the figures where the experimental data is plotted as a filled area where the outer bounds represent the results from the two microphones respectively.

The aim of this study is to verify if more accurate predictions can be obtained from the linear 3D approach compared to 1D CFD simulations with monopole radiation. As can be seen in Fig. 12 the predictions of the 3rd engine order are equally accurate. The deviations mostly stay within 5 dB, which is a good result. The prediction from using the simple monopole radiation model and the linear approach is almost equal to the one obtained from using the more advanced BE model. This conclusion also holds for the 6th and 9th engine order in Fig. 14 and 16, but is of course limited to these lower frequencies where the dirty air duct is much shorter than the wave length. Due to the scattering of the measurement data it is not possible to make a general statement about which of the 1D CFD predictions and the linear predictions that performs the best. The linear predictions at the engine speed between 1500 to 1800 rpm are clearly not so good. This can be explained by the information in Fig. 13. Here, the dotted curve shows the predictions using a measured two-port matrix when there is a small hole in the air cleaner box. This hole is also present in the 1D CFD calculations and in the measurements in the engine test bench. The purpose of the hole is to act as a short-cut for the incoming air if the filter paper is filled with snow. In all FE-calculations the hole is closed, which can explain some of the deviations especially at low frequencies.

When operating at full load conditions the flow speed in the ducts is increasing with the engine speed. At 1500 rpm the Mach number is about 0.025 whereas it is about 0.08 at 4000 rpm. In Fig. 13 the intake orifice noise is shown for predictions using the experimentally obtained two-port without flow as well as with the Mach number equal to 0.1. As can be seen there are only minor differences between the predictions which do not change the overall impression of the curve.

The validation of the 6th engine order in Fig. 14 gives similar information as above. The overall behaviour of the curve is predicted equally well, however, the level is better for the non linear simulations below 2200 rpm. At the peak at 3500 rpm the FE-predictions perform clearly less good compared to those from 1D CFD which can be explained by the frequency shift of the resonance frequencies for the resonators that was observed in Fig. 9. The results from using the experimentally obtained two-port data in Fig. 15 verify this conclusion as the peak is better predicted. If this deviation is due to the effect of yielding walls it can be predicted by a full finite element solution where the elastic walls are included. Why 1D CFD appears to give more accurate predictions at these particular frequencies is not understood as yielding walls are not included in this model.

Finally, the 9th engine order shows better agreement for the linear simulations concerning the overall impression. Using linear 3D simulations the peaks at 2400 and 3500 rpm are found correctly, which not is the case in the predictions using 1D CFD. This is most likely due to the appearance of 3D effects in the air cleaner that is not properly simulated when assuming 1D propagation.
Figure 12: Sound pressure level for 3rd engine order 100 mm from the intake orifice. Experiments in engine test bench; ···, 1D CFD of complete intake system and monopole radiation; ----, Source from 1D CFD, two-port from FEM and BEM radiation; ●●●●, Source from 1D CFD, two-port from FEM and monopole radiation.

Figure 13: Sound pressure level for 3rd engine order 100 mm from the intake orifice. All predictions performed using linear source predicted using 1D CFD, experimentally obtained two-ports and monopole radiation. Experiments in engine test bench; ●●●●, Two-port without flow; ----, Two-port with flow (M = 0.1); ····, Two-port without flow - with leakage; -·-·, Two-port with flow (M = 0.1) - with leakage.
Figure 14: Sound pressure level for 6th engine order 100 mm from the intake orifice. Experiments in engine test bench; ••••, 1D CFD of complete intake system and monopole radiation; ----, Source from 1D CFD, two-port from FEM and BEM radiation; ●●●●, Source from 1D CFD, two-port from FEM and monopole radiation.

Figure 15: Sound pressure level for 6th engine order 100 mm from the intake orifice. All predictions performed using linear source predicted using 1D CFD, experimentally obtained two-ports and monopole radiation. Experiments in engine test bench; ••••, Two-port without flow; ----, Two-port with flow (M = 0.1); ····, Two-port without flow - with leakage; -·-·, Two-port with flow (M = 0.1) - with leakage.
Figure 16: Sound pressure level for 9th engine order 100 mm from the intake orifice. Experiments in engine test bench; ..., 1D CFD of complete intake system and monopole radiation; ----, Source from 1D CFD, two-port from FEM and BEM radiation; ●●●●, Source from 1D CFD, two-port from FEM and monopole radiation.

Figure 17: Sound pressure level for 9th engine order 100 mm from the intake orifice. All predictions performed using linear source predicted using 1D CFD, experimentally obtained two-ports and monopole radiation. Experiments in engine test bench; ●●●●, Two-port without flow; ----, Two-port with flow (M = 0.1); ..., Two-port without flow - with leakage; -··-, Two-port with flow (M = 0.1) - with leakage.
6. Summary and conclusions

This paper describes an entirely virtual methodology to predict intake orifice noise for IC-engines. The procedure is based on coupling a linear time invariant one-port source model, representing the engine, to linear response for the transmission through the intake system. The source data was predicted using non-linear 1D CFD while the transmission was calculated using linear acoustic finite elements in order to include 3D effects in the air cleaner unit. A loudspeaker test rig that can include effect of flow has been used to validate the transmission loss for the air intake system, showing good accuracy. A small shift of the predicted resonance frequencies for the present resonators might be explained by the assumption of rigid walls or from neglecting viscothermal effects in the FE calculation. The radiation from the intake orifice is calculated using boundary elements or just a simple monopole radiation model. Finally the predicted intake noise is validated using experimental data obtained from engine test bench measurements in a semi-anechoic room.

The predictions from the linear methodology are mostly as good as, or for the case of higher frequencies even better, than the predictions from using non-linear 1D CFD for the complete system together with a linear monopole radiation model. The validation is performed for the 3rd, 6th and 9th engine order which includes frequencies between 75 and 600 Hz for the engine speed interval between 1500 and 4000 rpm. For these low frequencies the simple monopole radiation behaves as well as the much more complicated BE-model for the radiation.

Acknowledgements

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Appendix A

Figure A1: Source data for 3\textsuperscript{rd} engine order. Left picture source pressure level (ref 20 \( \mu \)Pa). Right picture normalized source impedance; ---, Real part; ----, Imaginary part.

Figure A2: Source data for 6\textsuperscript{th} engine order. Left picture source pressure level (ref 20 \( \mu \)Pa). Right picture normalized source impedance; ---, Real part; ----, Imaginary part.

Figure A3: Source data for 9\textsuperscript{th} engine order. Left picture source pressure level (ref 20 \( \mu \)Pa). Right picture normalized source impedance; ---, Real part; ----, Imaginary part.
Paper D
Sound propagation in narrow tubes including effects of viscothermal and turbulent damping with application to charge air coolers

Magnus Knutsson\textsuperscript{1,2} and Mats Åbom\textsuperscript{2}

\textsuperscript{1}Volvo Car Corporation, SE-405 31 Göteborg, Sweden
\textsuperscript{2}KTH CICERO, The Marcus Wallenberg Laboratory, Royal Institute of Technology, SE-100 44 Stockholm, Sweden

Abstract

Charge air coolers (CACs) are used on turbo charged internal combustion engines to enhance the overall gas exchange performance. The cooling of the charged air results in higher density and thus volumetric efficiency. Important for petrol engines is also that the knock margin increases with reduced charge air temperature. A property that is still not very well investigated is the sound transmission through a CAC. The losses, due to viscous and thermal boundary layers as well as turbulence, in the narrow cooling tubes results in frequency dependent attenuation of the transmitted sound that is significant and dependent on the flow conditions. Normally, the cross-sections of the cooling tubes are neither circular nor rectangular, why no analytical solution accounting for a superimposed mean flow exists. The cross-dimensions of the connecting tanks, located on each side of the cooling tubes, are large compared to the diameters of the inlet and outlet ducts. Three dimensional effects will therefore be important at frequencies significantly lower than the cut-on frequencies of the inlet/outlet ducts. In this study the two-dimensional finite element solution scheme for sound propagation in narrow tubes, including the effect of viscous and thermal boundary layers, originally derived by Astley and Cummings [\textit{Journal of Sound and Vibration}, 188 (5) (1995) 635-657] is used to extract two-ports to represent the cooling tubes. The approximate solutions for sound propagation, accounting for viscothermal and turbulent boundary layers derived by Dokumaci [\textit{Journal of Sound and Vibration} 182 (1995) 799-808] and Howe [\textit{Journal of the Acoustical Society of America} 98 (3) (1995) 1723-1730], are additionally calculated for corresponding circular cross-sections for comparison and discussion. The two-ports are thereafter combined with numerically obtained multi-ports, representing the connecting tanks, in order to obtain the transmission properties for the charged air when passing the complete CAC. An attractive formalism for representation of the multi-ports based on the admittance relationship between the ports is presented. From this the first linear frequency domain model for CACs, which includes a complete treatment of losses in the cooling tubes and 3D effects in the connecting tanks is extracted in the form of a two-port. The frequency dependent transmission loss is calculated and compared to corresponding experimental data with good agreement.

Part of this paper has been presented on the SAE Noise Vibration and Harshness Conference 2007, May 14-17, St Charles, Illinois, USA and the 14\textsuperscript{th} International Congress on Sound and Vibration, 2007, July 9-12, Cairns, Australia.
1. Introduction

1.1. General

The recent trend of downsizing internal combustion (IC) engines, in order to reduce fuel consumption, while using turbochargers to maintain the engine torque and power imposes additional noise phenomena not created by naturally aspirated engines. Examples of such noise sources are the whining due to unbalance in the turbo-axle and the different aerodynamic phenomena due to the high speed revolution of the turbine and compressor wheels. For the in-duct noise the aerodynamic turbo sources are normally the most important [1] producing high frequency noise [1]-[2] in the kHz range. In contrast the engine breathing represents low frequency noise well below 1 kHz.

Many turbocharged engines are equipped with charge air coolers (CAC), a device used to increase the overall performance of the engine. The cooling of the charged air results in higher density and thus volumetric efficiency. Important for petrol engines is also that the knock margin increases with reduced temperature. The parameters of main interest when designing a CAC are normally the pressure drop and the heat exchange efficiency. However, what seem to have been overseen are the acoustic properties which are still not very well investigated. To the authors’ knowledge the sound attenuation properties are only dealt with in two previous publications [3], [4]. The models in both these references are making use of two-ports (or four-poles) to assemble a complete model for a CAC. However, none of them includes a complete treatment of the losses in the cooling tubes. According to the literature survey in Ref. [3], there are predictive models available describing the thermal efficiency [5], and also models treating flow unsteadiness [6]-[8] in CACs. Still they are only evaluated in terms of heat transfer performance, pressure drop and gas-exchange properties mainly affecting lower frequencies. In Ref. [9] and [10] Knutsson and Åbom have presented some initial parts of the work presented in this paper, which aims to make a complete model of the sound attenuating properties of CACs when there is a mean flow present.

![Figure 1. Schematic representation of a generic air-to-air charge air cooler.](image)

Most CACs consists of two of the most widely used sound attenuation measures, the reactive expansion chamber, with reference to Fig. 1 denoted by inlet/outlet tank, and the dissipative narrow cooling tubes. The assembled component offers thereby possibly underestimated capabilities for broadband noise silencing that could be used
for noise optimization. As the CAC is installed downstream the compressor, see Fig 2, the noise that is radiated from the compressor and travels upstream towards the intake orifice, is of course not affected by the CAC. However, the CAC will attenuate the compressor noise that is transmitted downstream towards the engine and will thereby reduce the amount of break-out noise that radiates through the walls of the ducts situated downstream the CAC. Interesting is also that the low frequency engine breathing noise resulting from the motion of the intake valves as well as the overhearing of exhaust noise through the EGR-system will be reduced by the CAC. In order to take full advantage of this possibility, theoretical CAC models are required to enable optimization.

Figure 2. Schematic representation of a turbocharged diesel engine with charge air cooler.

In this paper a complete CAC linear frequency domain hybrid-model based on coupling multi-ports, representing the inlet/outlet tanks, to resistive two-ports, representing the cooling tubes, is presented. The model is validated for an air-to-air charge air cooler used in passenger cars, see Fig. 3. The cooling tubes in this CAC are equipped with turbulators, as shown in Fig. 4, a folded metal sheet with large area, used to improve the heat exchange efficiency of the cooling tubes. This installation creates narrow, almost triangularly shaped (or isosceles trapezium), internal axial channels with dimensions equivalent to a hydraulic diameter between 2.5 and 3 mm. The propagation of sound in such tiny ducts was found by Kirchhoff [11] to be dissipative because of viscous and thermal effects at the pipe walls. In this work he obtained the solution to the problem, without any flow present, as a complicated, complex transcendental equation which so far has not been solved analytically. In the work by Zwikker and Kosten [12] an approximate solution to the problem was found from a set of simplified equations. When flow is present the situation is somewhat more complicated and no complete theory exists. Several authors [13]-[22] have, however, derived solutions based on simplified equations or numerical calculations. In the present paper the solution for circular ducts by Dokumaci [14], a modified version of the numerical solution scheme for arbitrary cross-sections derived by Astley and Cummings [16] and the solution
accounting for turbulence in circular ducts by Howe [26] are used to model the cooling tubes in order to find the most accurate solution. Two-ports representing a cooling tube are extracted from the three solutions, which has not to the authors’ knowledge previously been done for the two later. The effect of approximating cross-sections that are shaped as isosceles trapeziums with circular geometries, where the hydraulic diameter is equivalent, is studied for cases where laminar or turbulent flow is present.

As the cross-dimensions of the connecting tanks are the largest in the charged air system they will define the upper frequency limit for the two-port technique. In order to further extend the upper frequency limit of the complete CAC model a multi-port technique has been used. The algebra for the multi-ports, based on the admittance relationships between the ports, is derived and presented in an attractive form for easy implementation. Three dimensional acoustic finite elements have been used to establish the admittance relations for each tank, which is represented by a matrix with the dimensions \((N+1) \times (N+1)\); where \(N\) is the number of cooling tubes. The upper frequency limit for this hybrid approach will be defined by the cut-on for the first non-planar mode in the inlet/outlet duct. The highest frequency validated in this particular study is about 1.5 times larger than the cut-on frequency for the tanks while cut-on in the inlet/outlet ducts is outside of the validation frequency band, limited by the experimental set-up. The correspondence between measured and predicted transmission loss is very good in the entire validation band. The suggested technique thereby offers a new possibility to tune the acoustic properties of CACs with respect to frequencies important for breathing noise as well as noise from compressor operation and turbulent flow.

Figure 3. Photograph of charge air cooler used for validation.

Figure 4. Internal geometry of one single cooling tube (geometry of turbulators indicated) in validation CAC.
1.2. Two-port modelling

Assuming a one dimensional (1D) acoustic state throughout the CAC, the sound propagation can be described using acoustic two-ports [23]. This will be consistent as long as the highest frequency of interest stays well below the cut-on frequency for the first non-planar wave. The frequency domain relationship between the acoustic states at sections representing the inlet and outlet of a two-port can be written

\[
\begin{bmatrix}
    p'_{in} \\
    q'_{in}
\end{bmatrix}
= \begin{bmatrix}
    T_{11} & T_{12} \\
    T_{21} & T_{22}
\end{bmatrix}
\begin{bmatrix}
    p'_{out} \\
    q'_{out}
\end{bmatrix}
\]

(1.1)

where \( p' \) denotes the plane wave acoustic pressure and \( q' \) the acoustic volume velocity. For simple generic geometries, such as circular ducts, it is possible to obtain the complex valued components in the transfer-matrix from analytical expressions. For a component built up of several sub-components coupled in series, such as a CAC or an after treatment device, the global two-port can be calculated as the product of the individual two-port matrices. For the case of a CAC this can with reference to Figure 1 be formulated as

\[
T_{\text{CAC}} = T_{\text{DUCT}} T_{\text{IN}} T_{\text{COOL}} T_{\text{OUT}} T_{\text{DUCT}}
\]

(1.2)

where the index DUCT correspond to the inlet/outlet ducts, IN to the inlet tank, COOL to the cooling tubes and OUT to the outlet tank. The two-port matrix \( T_{\text{COOL}} \) for a the complete bundle of cooling tubes are obtained from parallel coupling of \( N \) tubes via the relation

\[
T_{\text{COOL}} = \begin{bmatrix}
    T_{11} & T_{12}/N \\
    N T_{21} & T_{22}
\end{bmatrix}.
\]

(1.3)

Here, the matrix components \( T_{11}, T_{12}, T_{21} \) and \( T_{22} \) represent the data from one single channel. Eq. (1.2) will be valid only as long as the waves are plane at the sections where the two-ports interfere. For the particular case of CACs this will be limited to frequencies well below the cut-on frequency for the section where the cooling tubes enter the inlet or outlet volume. For frequencies above this cut-on frequency, the so called multi-port technique can be used. The following two sections will describe how the two-ports for the cooling tubes can be calculated and how a multi-port technique can be implemented in order to extend the valid frequency range for the complete CAC model.

2. Modelling plane waves in narrow tubes

2.1. General

The shortest dimension of the cross-section of a cooling tube in a CAC is of the order of 1 to 3 mm; hence the sound transmission will be strongly influenced by the viscothermal boundary layers and therefore highly resistive. A typical length of a cooling tube is about 50 cm, and in the case studied here about 70 cm. The ideal choice of model is a therefore a two-port where the viscothermal effects are included. A
number of such models are available in the literature. One early model for sound propagation in circular tubes is the classical Kirchhoff equation [11] from 1868 that includes the effect of viscosity as well as heat conduction. However, no analytical solution has so far been presented to this complicated transcendental equation. Kirchhoff himself was the first to present an approximate solution to his equation, using the restriction of “wide” ducts which is the same as large shear wave numbers. More than fifty years later an approximate solution to a simplified version of Kirchhoff’s equation was found by Zwikker and Kosten [12] for circular geometries. Their solution is only dependent on the shear wave number and is also known as the “low reduced frequency solution” [13] since it is only valid for cases where \( k_0 a << 1 \) and \( k_0 a / s << 1 \). Here \( k_0 = \omega / c_0 \), \( a \) is the duct radius, \( c_0 \) the speed of sound in the fluid, \( s = a (\rho_0 \omega \mu)^{1/2} \) is the shear wave number (also known as the Stokes number), \( \rho_0 \) the mean density, \( \omega \) the angular frequency and \( \mu \) is the dynamic viscosity.

In neither of these early works the effect of mean flow was considered. Inspired by the evolution of catalytic converters, a number of authors have presented improved models accounting for an incompressible mean flow and non-circular geometries [14]-[19]. For practical applications, the most useful is perhaps the work of Dokumaci [14]-[15]. In Ref. [14] he showed that the equations for sound propagation in a thermo-viscous fluid, simplified in the manner of Zwikker and Kosten theory [12], could be solved analytically for a circular pipe with a mean flow profile that is constant over the cross-section. In a later paper [15] Dokumaci extended the model from [14] to rectangular cross-sections by expanding the solution in terms of a double Fourier sine series. Other works starting out from essentially the same equations as used by Dokumaci [14], [15]; include Astley and Cummings [16] and Peat [17], Ih and Park and Kim [18], and Jeong and Ih [19]. At operating conditions CACs as well as catalytic converters experience temperature and pressure gradients. The effect of axial pressure and temperature gradients has been treated by Peat [20], Peat and Kirby [21], and Dokumaci [22].

The present paper aims to establish an efficient acoustic modelling strategy for CACs where the cross-section of the cooling tubes is non regular and an incompressible mean flow is present. The model that is most suitable for this geometry is probably the model by Astley and Cummings [16], which is based on a finite element discretisation of the cross-section of the duct and allows arbitrary geometries. However, the Stokes number for this particular case is larger than for the catalytic converter studied in Ref. [16] why a much denser mesh is required which results in longer calculation times. It is of interest to investigate if an analytical model can produce results equivalent to the numerical with a much smaller computational effort. The model by Dokumaci [14] and a modified version of the model by Astley and Cummings [16] are coded and used for this purpose. Since the shape of the propagating profile of the incompressible mean flow can be arbitrary in Ref. [16] a comparison is performed in order to estimate the effects of choosing a plug flow profile instead of a parabolic. This is indeed interesting since the analytical model in Ref [14] is based on a plug profile which actually makes violence to the assumption of a laminar flow.

All the models mentioned above assume laminar flow and do therefore not take into account any effect of turbulence on the propagation of sound waves. The transition from laminar to turbulent flow is related to the Reynolds number which for a circular pipe is defined as

\[
\text{Re} = \frac{U_s D \rho_0}{\mu}
\]  

(2.1)
where \( U_x \) is the velocity in the axial direction, \( D \) the duct diameter, \( \rho_0 \) the density and \( \mu \) the dynamic viscosity. If the cross-section is non-circular but the aspect ratio is not too large the diameter can be replaced by the hydraulic diameter \( D_h = 4S/C \) where \( S \) is the cross-section area and \( C \) the wetted perimeter. For a circular cross-section transition is known to take place at a Reynolds number just above 2000 [24], but for triangular cross-sections the situation is somewhat more complicated. In the work by Eckert and Irvine [25] it was shown experimentally that transition does not occur simultaneously over the whole of the flow for a channel including a narrow region such as a triangle. For a triangle with an acute angle of 12° more than 20 percent of the height is still laminar when the Reynolds number is 4000 (based on the hydraulic diameter).

The mean flow velocity in a CAC will be defined by the mass flow, temperature, static pressure and the cross-section of the cooling tubes. The area remains of course constant for a particular CAC, but the other three parameters are defined by the operating conditions of the engine and will therefore vary significantly, as will the mean flow. However, in a passenger car the Mach number of the mean flow in the cooling tubes will most likely stay below 0.1 for most driving conditions, except at very high engine revolution speed. The flow regime between a Mach number of 0 and 0.1 will therefore be the most important from the noise point of view. For the particular case of a CAC mounted on a passenger car, the Reynolds number will not be possible to scale just with the mean flow velocity. An increased velocity is the result of the energy added by the compressor wheel which also increases the pressure and the temperature. As the viscosity is dependent of the temperature and the density on both the temperature and the pressure it will follow that the Reynolds number will have to be calculated for each load case separately. Representative data taken from both a diesel and a petrol engine at full load conditions will result in Reynolds numbers exceeding 2000. Due to its wider engine speed range the petrol engine will experience Reynolds numbers as high as 8000.

At those conditions there will most likely be turbulent flow at least to in the major part of the cross-section why a model that takes into account interaction between turbulence and sound waves is required. The model by Howe in Ref. [26] combines the effects of turbulence and viscothermal sub-layers on wave propagations in circular cross-sections. Although it requires the cross-section to be “wide” and the turbulent flow to be fully developed with a constant flow profile, it will provide useful information and understanding concerning the low frequency damping for the cases where the Reynolds number is large. In this paper the model by Howe is applied to create a two-port matrix that includes interaction between turbulence and sound waves. From this two-port transmission loss data is extracted and compared to corresponding data from the models by Dokumaci [14] and Astley and Cummings [16] for cold conditions as well as hot operating conditions.

2.2. Analytical model including viscothermal effects

The fundamental linearized equations for a viscothermal fluid, as formulated by Zwikker and Kosten [12], describing plane wave propagation in the \( x \)-direction, assuming an ideal gas and an incompressible uniform mean flow, are for harmonic time variation \([e^{i\omega t}]\), see Ref. [14], as follows.

Conservation of momentum is

\[
\rho_0 \left( i\omega U_0 \frac{\partial}{\partial x} \right) u'_x = -\frac{\partial p'}{\partial x} + \mu \nabla^2 u'_x, \quad p' = p'(x,t). \tag{2.2}
\]
Conservation of mass is
\[ (i\omega + U_0 \frac{\partial}{\partial x}) \rho' + \rho_0 \nabla \cdot \mathbf{u}' = 0. \] (2.3)

Conservation of energy is
\[ \rho_0 C_p \left( i\omega + U_0 \frac{\partial}{\partial x} \right) T' = \left( i\omega + U_0 \frac{\partial}{\partial x} \right) p' + \kappa_{th} \nabla^2 T'. \] (2.4)

Using the ideal gas law the following equation of state is obtained as
\[ \frac{p'}{p_0} = \frac{\rho'}{\rho_0} + \frac{T'}{T_0}. \] (2.5)

Here \( p_0, \rho_0, U_0 \) and \( T_0 \) are the mean pressure, the mean density, the incompressible velocity in the axial direction and the mean temperature respectively, \( \mathbf{u}' \) is the acoustic velocity that comprises \( u'_x, u'_y \) and \( u'_z \) for each direction, \( p', \rho' \) and \( T' \) are the acoustic pressure, density and temperature respectively, \( C_p \) is the specific heat at constant pressure, \( \kappa_{th} \) the thermal conductivity and \( \nabla^2 \) is the Laplacian over the cross-section of the cooling tube \( S \). The boundary conditions that must be fulfilled at the duct walls are that no slip is allowed and that temperature fluctuations are negligible. In addition rigid walls are assumed. These conditions implies that \( \mathbf{u}' = \mathbf{0} \) and \( T' = 0 \) on the wall \( \partial S \).

### 2.2.1. Cylindrical tubes

Following Dokumaci [14] a propagating wave ansatz is assumed
\[ p' = p_0 p^* \exp(i\omega t - i\Gamma k_0 x), \quad u'_x = p'h(r)/\left( \rho_0 c_0 \right), \quad T' = p' f(r)/\left( \rho_0 C_p \right) \] (2.6)

where \( p^* \) is an arbitrary dimensionless constant, \( k_0 \) is the wave number, \( c_0 = (\gamma p_0 / \rho_0)^{\frac{1}{2}} \) is the adiabatic speed of sound, \( \gamma \) is the ratio of specific heats, \( \Gamma \) is the dimensionless axial propagation constant and \( r \) is the radial co-ordinate. If a circular cross-section is considered the \( \nabla^2 \) - and \( \nabla \cdot \) -operators can be transferred to cylindrical coordinates as
\[ \nabla^2 = \frac{1}{r} \frac{\partial}{\partial r} r \frac{\partial}{\partial r} + \frac{\partial^2}{\partial r^2} \] (2.7)
and
\[ \nabla \cdot \mathbf{u}' = \frac{\partial u'_x}{\partial x} + \frac{\partial u'_r}{\partial r} + \frac{u'_r}{r}. \] (2.8)

Here \( u'_r \) is the acoustic velocity in the radial direction. Substituting Eqs. (2.6) - (2.8) into Eq. (2.2) and (2.4), while dropping \( p' \) from both sides of the equation yields
\[ \rho_0 (i\omega - U_0 i\Gamma k_0) \frac{h}{\rho_0 c_0} = i\Gamma k_0 + \frac{\mu}{\rho_0 c_0} \left( \frac{1}{r} \frac{\partial h}{\partial r} + \frac{\partial^2 h}{\partial r^2} \right) \] (2.9)
and

\[
(i\omega - U_0 i\Gamma k_0) f = (i\omega - U_0 i\Gamma k_0) + \frac{\kappa_h}{\rho_0 C_p} \left( \frac{1}{r} \frac{\partial f}{\partial r} + \frac{\partial^2 f}{\partial r^2} \right). \tag{2.10}
\]

Introducing \( \beta^2 = is^2(1-\Gamma M)/a^2 \) where \( s \) is the Stokes number, \( a \) the radius of the duct, \( M \) the Mach number, and the Prandtl number \( \xi^2 = \mu C_p / \kappa_h \), the Eqs. (2.9) and (2.10) simplifies to

\[
\frac{\partial^2 h}{\partial r^2} + \frac{1}{r} \frac{\partial h}{\partial r} - \beta^2 h = -\frac{i\Gamma s^2}{a^3} \tag{2.11}
\]

\[
\frac{\partial^2 f}{\partial r^2} + \frac{1}{r} \frac{\partial f}{\partial r} - \beta^2 \xi^2 f = -\beta^2 \xi^2 \tag{2.12}
\]

which has the solutions

\[
h(r) = C_H J_0(i\beta r) + \frac{\Gamma}{1-\Gamma M} \tag{2.13}
\]

and

\[
f(r) = C_F J_0(i\beta \xi r) + 1 \tag{2.14}
\]

where \( C_H \) and \( C_F \) are constants and \( J_n \) is a Bessel function of the first type of order \( n \). The boundary conditions are zero acoustic velocity at \( r = 0 \) and zero acoustic velocity and temperature at the duct wall

\[
u'_r(0) = u'_r(a) = u'_t(a) = T'(a) = 0 \tag{2.15}
\]

which yields the solutions

\[
h(r) = \frac{\Gamma}{(1-\Gamma M)} \left[ 1 - \frac{J_0(i\beta r)}{J_0(i\beta a)} \right] \tag{2.16}
\]

and

\[
f(r) = 1 - \frac{J_0(i\beta \xi r)}{J_0(i\beta \xi a)}. \tag{2.17}
\]

Averaging the continuity equation (2.3) over the cross-section and insertion of Eq. (2.5) gives

\[
(i\omega + U_0 \frac{\partial}{\partial x}) \left( \frac{\rho_0 p'}{p_0} - \frac{\rho_0 \langle T' \rangle}{T_0} \right) + \rho_0 \langle \nabla \cdot u' \rangle = 0. \tag{2.18}
\]
Via Gauss’ theorem and by using the boundary conditions in Eq. (2.15) the last term in Eq. (2.18) can be rewritten; this yields

\[
\left( i\omega + U_0 \frac{\partial}{\partial x} \right) \left( \frac{p'}{p_0} - \frac{\langle T' \rangle}{T_0} \right) + \frac{\partial \langle u'_r \rangle}{\partial x} = 0
\]  

(2.19)

Insertion of (2.6) and division by \( p' \) yields

\[
(i\omega - U_0 \delta \Gamma k_0) \left( \frac{1}{p_0} - \frac{\langle f \rangle}{\rho_0 C_p T_0} \right) - \frac{ik_0 \Gamma \langle h \rangle}{\rho_0 c_0} = 0
\]  

(2.20)

which after some simplifications becomes

\[
(1 - \Gamma M) \left[ 1 - \left( 1 - \frac{1}{\gamma} \right) \langle f \rangle \right] - \frac{\Gamma \langle h \rangle}{\gamma} = 0.
\]  

(2.21)

The average of Eq. (2.16) is

\[
\langle h \rangle = \frac{\Gamma}{(1 - \Gamma M)} \frac{1}{\pi a^2} \int_0^a \left[ 1 - \frac{J_0(i\beta r)}{J_0(i\beta a)} \right] \frac{1}{2\pi r} dr = \frac{\Gamma}{(1 - \Gamma M)} \left[ 1 - \frac{2J_1(i\beta a)}{i\beta a J_0(i\beta a)} \right].
\]  

(2.22)

The same operation on Eq. (2.17) yields

\[
\langle f \rangle = \left[ 1 - \frac{2J_1(i\beta \xi a)}{i\beta \xi a J_0(i\beta \xi a)} \right].
\]  

(2.23)

Insertion of Eq. (2.22) and (2.23) in (2.21) yields after some rearrangement

\[
(1 - \Gamma M)^2 \left[ 1 - \left( 1 - \frac{1}{\gamma} \right) G(i\beta \xi a) \right] - \frac{\Gamma^2}{\gamma} G(i\beta a) = 0
\]  

(2.24)

where

\[
G(x) = 1 - \frac{2J_1(x)}{x J_0(x)}.
\]  

(2.25)

Eq. (2.24) can be solved by simple iteration to obtain two roots \( \Gamma_1 \) and \( \Gamma_2 \), representing wave propagation in the positive and negative axial direction respectively. For the special case of \( M = 0 \) Eq. (2.24) simplifies to

\[
\left[ 1 - \left( 1 - \frac{1}{\gamma} \right) G(\xi s \sqrt{-i}) \right] - \frac{\Gamma^2}{\gamma} G(s \sqrt{-i}) = 0
\]  

(2.26)

with the solution
\[
\Gamma^2 = \frac{1 + (\gamma - 1) \frac{2J_1(s\xi \sqrt{-i})}{s\xi \sqrt{-i}J_0(s\xi \sqrt{-i})}}{1 - \frac{2J_1(s\sqrt{-i})}{s\sqrt{-i}J_0(s\sqrt{-i})}}. 
\]
(2.27)

### 2.2.2. Two-port for circular ducts

In order to establish the two-port matrix the wave admittance \( h / (\rho_0 c_0) \) is used. Eq. (2.22) yields after some simplifications

\[
\frac{\langle h \rangle}{\rho_0 c_0} = \frac{\Gamma G(i\beta a)}{\rho_0 c_0 (1 - \Gamma M)} \quad \text{(2.28)}
\]

Finally the two-port of one single channel with the length \( L_p \) can be calculated as

\[
T_p = \begin{bmatrix} 1 & 1 \\ \rho_0 c_0 & \rho_0 c_0 \end{bmatrix} \begin{bmatrix} \exp(-ik_0 \Gamma_1 L_p) & \exp(-ik_0 \Gamma_2 L_p) \\ \frac{S\langle h_1 \rangle}{\rho_0 c_0} \exp(-ik_0 \Gamma_1 L_p) & \frac{S\langle h_2 \rangle}{\rho_0 c_0} \exp(-ik_0 \Gamma_2 L_p) \end{bmatrix}^{-1} 
\]
(2.29)

where the indices 1 and 2 denote the two separate roots of Eq. (2.24). For the case of parallel channels, as in a cooling tube, the two-port \( T_p \) can be calculated using Eq. (2.29) after multiplication of the capillary area \( S \) by the number of channels. Additionally for the case of non circular geometries, where the aspect ratio of the sides of the tube is not too large, the wave damping of a circular section with the same hydraulic diameter should be a good approximation [27].

### 2.3. Numerical model including viscothermal effects

For the case of large aspect ratios, the finite element solution scheme formulated by Astley and Cummings [16] can be used in order to further increase the accuracy of the prediction. This formulation will also allow a parabolic profile of the incompressible mean flow. For the convenience of the reader the derivation of the numerical formulation is partly presented in the section below. The linearized governing equations are the same as was used in Ref. [14] - [15], here given in Eq. (2.2) - (2.5), with two exceptions. First, in order to eliminate the velocity components perpendicular to the axial direction of the duct, an integral form of the continuity equation (2.3) is used in combination with the boundary conditions \( u' = 0 \) and \( T' = 0 \) on the wall \( \partial S \). The continuity equation is, without approximation, reformulated to

\[
\int_S \left( (i\omega + U_i \frac{\partial}{\partial x}) \rho' + \rho_0 \frac{\partial u'_i}{\partial x} \right) dS = 0.
\]
(2.30)

Secondly, in the derivation by Astley and Cummings [16] the fluctuating part of the dissipation function in the energy equation is retained. However, in the following work this term is excluded in order to study the effect of different geometries as well as the shape of incompressible flow profile. It should be pointed out that in the derivation of
Astley and Cummings the acoustic velocities \( u'_y \) and \( u'_z \) in the momentum equation are omitted with reference to the work by Peat [17].

### 2.3.1. Numerical formulation

Following Astley and Cummings [16] a harmonic plane wave type of solution to these equations, where the acoustic variables have been non-dimensionalized using the mean flow quantities and the adiabatic speed of sound, can be obtained from the ansatz:

\[
\begin{align*}
  u'_y &= c_0 u^* (y^*, z^*) \exp \left( i \omega t - i k_x \Gamma x \right), \\
  p' &= p_0 p^* \exp \left( i \omega t - i k_x \Gamma x \right), \\
  T' &= T_0 T^* (y^*, z^*) \exp \left( i \omega t - i k_x \Gamma x \right).
\end{align*}
\]

Here \( y^* \) and \( z^* \) are the in-plane non-dimensional co-ordinates, scaled using half the hydraulic diameter of the cross-section. The cross-sectional area in the non-dimensional \( y^*-z^* \) plane is denoted by \( S^* \) and its boundary by \( \partial S^* \). Substitution of Eqs. (2.31) - (2.33) into the equation of state (2.5) and the integrated continuity equation (2.30) gives

\[
\iint_{S^*} \left[ i \left( 1 - \Gamma M \right) (p^* - T^*) - i \Gamma u^* \right] dydz = 0,
\]

where \( M(y^*, z^*) \) is the Mach number of the incompressible mean flow. The same procedure on the equation for conservation of momentum (2.2) yields

\[
\frac{\gamma}{s^2} \nabla^2_{S^*} u^* - i \gamma \left( 1 - \Gamma M \right) u^* + i \Gamma p^* = 0
\]

where

\[
\nabla^2_{S^*} = \frac{\partial^2}{\partial y^*^2} + \frac{\partial^2}{\partial z^*^2}
\]

and \( s \) is the Stokes’ number. The energy equation (2.4) becomes accordingly

\[
\frac{\gamma}{\gamma - 1} \frac{1}{s^2 \xi^2} \nabla^2_{S^*} T^* - \frac{\gamma}{\gamma - 1} i \left( 1 - \Gamma M \right) T^* + i \left( 1 - \Gamma M \right) p^* = 0
\]

where \( \xi^2 \) is the Prandtl number of the fluid.

The finite element process is based on finding an approximate solution (trial solution) of the velocity and temperature fields on the form:

\[
\begin{align*}
  u^* &= \sum_{i=1}^{n} u_i \phi_i(y, z), \\
  T^* &= \sum_{i=1}^{n} T_i \phi_i(y, z)
\end{align*}
\]

where \( \phi_i \) are the known shape functions, which must be able to satisfy the boundary conditions individually, and \( u_i \) and \( T_i \) are unknown coefficients. The boundary conditions are zero acoustic velocity and temperature at the duct wall.
\[ u^* = T^* = 0 \quad \text{on } \partial S^*. \quad (2.39) \]

The plane wave ansatz for the pressure in Eq.(2.32) gives
\[ p^* = p_1 = \text{constant}. \quad (2.40) \]

Substitution of Eqs. (2.38) and (2.40) into the integral equation (2.34) yields
\[ iS^* \left( 1 - \Gamma \bar{M} \right) p^* - i \left( \phi_m^T - \Gamma \phi^T \right) T^* - i \Gamma \varphi^T u^* = 0, \quad (2.41) \]

where
\[ \phi = \iint_{S^*} \begin{bmatrix} \phi_1 \\ \phi_2 \\ \phi_n \end{bmatrix} dy^* dz^*, \quad \phi_m = \iint_{S^*} \begin{bmatrix} M \phi_1 \\ \ldots \\ M \phi_n \end{bmatrix} dy^* dz^*, \quad (2.42) \]
\[ u^* = \begin{bmatrix} u_1 \\ \ldots \\ u_n \end{bmatrix}, \quad T^* = \begin{bmatrix} T_1 \\ \ldots \\ T_n \end{bmatrix}, \quad p^* = [p_1]. \quad (2.43) \]

The Mach number of the incompressible flow averaged over the cross-section is
\[ \bar{M} = \frac{1}{S^*} \iint_{S^*} M (y^*, z^*) dy^* dz^*. \quad (2.44) \]

The method of weighted residuals (known alternately as the Galerkin procedure) applied to the continuity equation (2.35), while applying the divergence theorem, yields after some rearrangement
\[ \iint_{S^*} \left[ - \frac{\gamma}{s^2} \nabla \phi_j \cdot \nabla u^* - i \gamma \left( 1 - \Gamma M \right) \phi_j \eta^* + i \Gamma \varphi_j \varphi^* \right] dy^* dz^* = 0, \quad j = 1, 2, \ldots, n. \quad (2.45) \]

After substitution of the trial functions for \( p \) and \( u \) Eq. (2.45) becomes
\[ - i \frac{\gamma}{s^2} B u^* - i \gamma \left[ A - \Gamma A_m \right] u^* + i \Gamma \varphi p^* = 0. \quad (2.46) \]

Here \( A, B \) and \( A_m \) are \( n \times n \) matrices, where \( n \) is the number of nodes and the \( j \)-\( k \)th components are given by
\[ [A]_{jk} = \iint_S \phi_j \phi_k dy^* dz^*, \quad [B]_{jk} = \iint_S \nabla \phi_j \cdot \nabla \phi_k dy^* dz^*, \quad (2.47) \]
\[ [A_m]_{jk} = \iint_S M \phi_j \phi_k dy^* dz^*. \quad (2.48) \]
The same procedure can be used for the energy equation (2.37)

$$-\frac{\gamma}{\gamma-1} \frac{1}{s^2 \xi^2} BT^* - i \frac{\gamma}{\gamma-1} [A - \Gamma A_m] T^* + i [\varphi - \Gamma \varphi_m] p^* = 0.$$ \hspace{1cm} (2.49)

Eq. (2.41), (2.46) and (2.49) has to be solved together, which is easiest visualized as the matrix equation

$$\begin{bmatrix}
  iS^* & 0 & i\varphi^T \\
  0 & \gamma B / s^2 + i\gamma A & 0 \\
  -i\varphi & 0 & \gamma / (s^2 \xi^2 + iA)
\end{bmatrix}
\begin{bmatrix}
  iS^* \bar{M} & i\varphi^T & -i(\varphi_m)^T \\
  i\varphi & i\gamma A_m & 0 \\
  -i\varphi_m & 0 & \gamma / (\gamma - 1) A_m
\end{bmatrix}
\begin{bmatrix}
p^* \\
u^* \\
T^*
\end{bmatrix}
= 0.$$ \hspace{1cm} (2.50)

Here 0 indicates a zero matrix of appropriate size. For the case of zero mean flow, \(\bar{M} = 0\), all sub matrices with index \(m\) is zero. The eigenvalue problem in Eq. (2.50) will just contain two non-trivial roots having the same values but opposite sign. As can be observed from the ansatz (2.32) they represent wave propagation in positive and negative axial direction with equal speed and attenuation. For the case of a present mean flow a full set of non-trivial eigenvalues will exist. As discussed in [16] two of those will behave almost as the solutions in the no-flow case. Only those two modes will be retained in the present work.

2.3.2. Two-port calculation

The eigenvectors corresponding to the above mentioned two eigenvalues describe the shape of the profile of the propagating acoustic variables in the axial direction and can be rewritten as

$$\Lambda_j = \begin{bmatrix}
\Lambda_{pj} \\
\Lambda_{uj} \\
\Lambda_{Tj}
\end{bmatrix},$$ \hspace{1cm} (2.51)

where \(\Lambda_{pj}, \Lambda_{uj}, \Lambda_{Tj}\) denotes the eigenvector for the pressure, velocity and temperature respectively, taken for eigenvector \(j\). Since the pressure is assumed to be constant over the cross-section \(\Lambda_p\) is a single value while \(\Lambda_u\) and \(\Lambda_T\) are vectors of size \(n \times 1\). In order to establish a two-port for the duct the dimensional axial acoustic velocity \(u'_x\) has to be averaged over the cross-section. For this purpose the shape functions in equation (2.42) can conveniently be used:

$$\langle u'_x \rangle = \frac{c_0 \Theta^T A_u}{S^*}.$$ \hspace{1cm} (2.52)
The averaged admittance is thereafter calculated using Eq. (2.52) and the pressure from Eq. (2.32) as

$$\frac{\langle h \rangle}{\rho_0 c_0} = \frac{c_0 \Phi^T A_u}{\rho_0 \Lambda_p S^*}. \quad (2.53)$$

Using the definition for speed of sound Eq. (2.53) simplifies to

$$\frac{\langle h \rangle}{\rho_0 c_0} = \frac{\gamma \Phi^T A_u}{\rho_0 c_0 \Lambda_p S^*}. \quad (2.54)$$

Finally, \(\langle h \rangle\) and \(\Gamma\) for the two acoustic modes is substituted into Eq. (2.29), upon maintaining their signs, to establish the desired two-port for the tube-element.

### 2.3.3. Numerical formulation of the mean flow

The establishment of the matrices \(A_m, \phi_m\) requires the mean flow to be known at all positions in the cross-section. The parabolic profile that is present for laminar flow can be obtained using the solution of Poisson’s equation [16] as

$$\frac{\partial^2 M}{\partial y^*^2} + \frac{\partial^2 M}{\partial z^*^2} = 2 \bar{M} \quad (2.55)$$

where \(\bar{M}\) is the cross–sectional averaged Mach number previously defined by Eq. (2.44). The solution scheme of this equation is obtained using the same procedure as was used for the conservation of motion and energy. The Galerkin process, the divergence theorem and the boundary condition \((M = 0)\) on \(\partial S^*\) yields after some rearrangements:

$$\int_{S^*} \left[ \nabla \phi_j \cdot \nabla M + 2 \bar{M} \phi_j \right] dy^* dz^* = 0, \quad j = 1, 2, \ldots, n \quad (2.56)$$

The trial functions for the Mach number is the same that were used for the velocity and temperature

$$M = \sum_{i=1}^{n} M_i \phi_i (y^*, z^*). \quad (2.57)$$

Substitution of Eq. (2.57) into Eq. (2.56) yields

$$BM + 2 \bar{M} \phi = 0 \quad (2.58)$$

where
\[ M = \begin{bmatrix} M_1 \\ - \\ - \\ M_n \end{bmatrix} \] (2.59)

and \( B \) is defined in Eq. (2.47) and \( \phi \) in Eq. (2.42).

### 2.3.4. Calculation of the shape functions

The derivation of the shape functions is straightforward and well documented in several text books, see for instance the book by Zienkiewicz and Taylor [33]. The finite element discretisation in this work is performed using nine-noded isoparametric Lagrangian rectangular elements. The shape functions for these elements are given in Appendix A. Nine-point Gauss-Legendre integration is used to evaluate the element integrals \( A, B, \phi, A_m \) and \( \phi_m \) at the Gauss points. The information about the Mach number of the incompressible mean flow is used when evaluating \( A_m \) and \( \phi_m \) at the integration points. If the same element discretisation is used for the mean flow problem as for the acoustic problem the data is calculated only at the node points and 2D interpolation must be used in order to predict the Mach number at the Gauss points.

### 2.4. Analytical model including effects of turbulence

When a turbulent flow is present the acoustic waves will be attenuated due to transfer of energy to turbulent stresses if the thickness of the acoustic boundary layer, \( \delta_{ac} = \left( \frac{2v}{\omega} \right)^{\frac{1}{2}} \), is larger than that of the viscous sub-layer of the turbulent mean flow boundary layer, \( \delta_v \approx 10v/\left( \tau_w/\rho_0 \right)^{\frac{1}{2}} \), [28]-[29]. Here, \( v \) denotes the kinematic viscosity, \( \rho_0 \) the density of the fluid and \( \tau_w \) the mean wall shear stress. Introducing the friction velocity as \( u_* = \left( \tau_w/\rho_0 \right)^{\frac{1}{2}} \), the viscous sub-layer is \( \delta_v \approx 10v/u_* \). An early effort addressing damping of sound waves in the presence of flow was made by Ingard and Singal [30]. More recent work are Peters et al. [29] and Howe [26] where the latest is the most complete model developed so far [31].

In this work Howe proposed a frequency dependent model for the turbulent boundary layer eddy viscosity controlling the momentum and thermal boundary layers, which are formed from interaction between turbulent boundary layers and sound waves. The model is restricted to cases with fully developed and low Mach number \( (M < 0.1) \) turbulent flow and only treats one-dimensional axial wave propagation. It is strictly valid to situations where the thickness of the turbulent boundary layer is much smaller than the acoustic wavelength so that the layers can be described by an effective acoustic admittance. The model has so far been proven to provide good agreement with experimental data from [29] for Reynolds numbers exceeding \( 10^3 \) which is well above the transition which normally takes place just above a Reynolds number of 2000, based on the hydraulic diameter [24], [25]. These circumstances justify the simplification that the profile of the mean flow is regarded as uniform in the core of the flow.

#### 2.4.1. Circular tubes

The acoustic wave is once again given by the ansatz

\[ p' = p_0 p^* \exp(i \omega t - i \Gamma k_0 x). \] (2.60)
The propagation constant, corrected for turbulent flow, is given by [26] as

\[
\Gamma = \pm \frac{1}{1 \pm M} \pm \frac{2i \rho \omega c_0}{(1 \pm M) D_h k_0} Y_c \left( \pm k_0 \omega \right). \tag{2.61}
\]

Here, \( Y_c \) is the complex conjugate of the impedance of the wall shear layer calculated as

\[
Y(k, \omega) = \frac{e^{-i\pi/4}}{\rho \omega^{3/2}} \left[ k^2 \sqrt{\nu F_A} \left( \frac{i \omega \chi}{\kappa^2 \nu^2}, \delta, \frac{i \omega}{\chi} \right) \right]
\]

\[
+ \frac{\beta \omega^2}{c_p} \sqrt{\chi} F_A \left( \frac{i \omega \chi P^2}{\kappa^2 \nu^4}, \delta, \frac{i \omega}{\chi} \right)
\]

(2.62)

where \( \kappa \approx 0.41 \) is the von Karman constant, \( P_r \) is a turbulence Prandtl number, assumed to be constant and equal to 0.7 for air [26], \( \chi = k_0^2 / (\rho_0 C_p) \) is the thermometric conductivity, which for air at 20°C is about \( 2 \times 10^3 \) m²/s. Moreover, the function \( F_A \) is defined as

\[
F_A(a,b) = \frac{i \left[ H_1^{(1)}(a) \cos(b) - H_0^{(1)}(a) \sin(b) \right]}{H_0^{(1)}(a) \cos(b) - H_1^{(1)}(a) \sin(b)}.
\]

(2.63)

where \( H_n^{(1)} \) is a Hankel function of order \( n \). The friction velocity is calculated from the empirical pipe flow formula

\[
U_0 / \nu_* = 2.44 \ln \left( \nu_* D_h / 2 \nu \right) + 2.0. \tag{2.64}
\]

Finally, the thickness of the viscous sub-layer is calculated using the empirical formula [26]

\[
\frac{\delta u_*}{\nu} = 6.5 \left( 1 + \frac{1.7 (\omega / \omega_*)^3}{1 + (\omega / \omega_*)^3} \right), \quad \omega \nu / u_*^2 \approx 0.01, \quad \omega > 0.
\]

(2.65)

Here, \( \omega_* \) is the critical frequency representing the centre of the range where the principal acoustic–turbulence interaction occur in a close to the wall region where viscous as well as turbulence diffusion is significant.

2.4.2. Two-port for circular tubes

In order to extend Howe’s model to enable extraction of a two-port for a circular duct the wave admittance \( \langle h \rangle / (\rho_0 c_0) \) is required. This can be calculated using the axial momentum conservation equation where the velocity has been averaged over the cross-section,

\[
\frac{D\langle u'_r \rangle}{Dt} + \frac{1}{\rho_0} \frac{\partial p'}{\partial x} = 0
\]

(2.66)

where \( D/Dt \equiv \partial / \partial t + U_0 \partial / \partial x \). Assuming harmonic waves yields the wave admittance as
\[
\frac{\langle u' \rangle}{p'} = \frac{\langle h \rangle}{\rho_0 c_0} = \frac{\Gamma}{\rho_0 c_0 (1 - \Gamma M)}.
\] (2.67)

The two-port for the duct can thereafter be obtained from substituting the expressions for \( \langle h \rangle \) and \( \Gamma \) into Eq. (2.29). As for the two previously described models, there exist two solutions for propagation in opposite directions, in Eq. (2.29) denoted by the indices 1 and 2. Both are required for the two-port extraction and the appropriate sign will follow from the two solutions for \( \Gamma \), obtained from Eq. (2.61).

3. Modelling a complete charge air cooler

3.1. General

The charge air cooler used in this study is taken from a passenger car in series production and is used for diesel as well as petrol engines. It is assembled of several parts made of different materials. The cooler is of brick type with relatively compact dimensions which makes it suitable for densely packed engine compartments. There are ten cooling tubes made of aluminium, each of them divided into 36 channels due to the turbulator installation, as shown in Fig. 4. The air is prevented from flowing between the channels and there is no flow reversal taking place inside the cooler. The cooling tubes are modelled using the two-ports that were extracted using the techniques described in section 2.

3.2. In- and outlet sections

The inlet tank consists of a 90 degree bend and a diverging conical section connecting the inlet duct to the cooling tubes. The walls are made of plastic and are reinforced by some ribs in order to reduce vibrations as well as sound transmission. The total cross-sectional area of the ten cooling tubes is about 50 % larger than that of the inlet duct. This is a good way to compensate for the larger pressure drop present in the narrow cooling tubes. In order to create a good flow for the cold air outside of the cooling tubes, and thereby an efficient heat transfer from the charged air inside the CAC to the cooling air, there is a slit separating each tube from its neighbour. This geometrical separation of the tubes makes the area at the largest section of the inlet tank more than twice to that of the cooling tubes. The geometry of the outlet tank is almost symmetric to that of the inlet, of course with the restriction that the mean flow here is contracted. The acoustics of such conical devices (horns) has been treated by several authors where the most recent publications have addressed the effect of flow. Approximate plane wave models based on series of straight ducts have been discussed by, e.g. Åbom [32] who showed that only a few segments are required to create very good results.

3.2.1. 3D finite element models

In order to extend the valid frequency range and include cross-modes, for such complicated geometries as the tanks, acoustic finite elements can be used. Most commercial finite element softwares include 3D linear acoustic finite elements (FE) and there is a large number of text books describing the theory, see e.g. [33], hence it will not be further described here. In the 3D FE-calculations in the present work
LMS/Sysnoise [34] has been used. In this code the convective effect of a superimposed incompressible mean flow can easily be accounted for, although it has been neglected here due to the small Mach number that remains from the large expansion. Some codes also allow the use of admittance matrices to connect two different volumes using the acoustic velocity and pressure as coupling variables [34]. This procedure requires that the complete model is solved for each frequency for each load case; for models with fine element resolution time consuming as well as expensive. A good method to shorten iteration time, if the properties of the air mixture and the geometry of the tanks are fixed, is to use the multi-port approach that will be described in the next section.

The tanks of the charge air cooler used in this study are very similar in size and shape. The volume of the inlet tank is 853 cm$^3$ and that of the outlet tank is 896 cm$^3$. There is a circular opening in the wall of the outlet tank designed to support a temperature transducer. In all measurements and calculation results presented here this hole is carefully plugged to create a smooth wall. In order to get a fast and simple meshing procedure, linear tetrahedral elements with four nodes are used to model the volumes and linear wedge element with six nodes for the pipes. The use of hexahedral elements with eight nodes would certainly be more efficient but the process of creating the element mesh is still not possible to be fully automated and hence very cumbersome. Parabolic tetrahedral elements would of course also be an efficient choice but for this case it was decided to use the linear element with very high resolution. The chosen element size is 5 mm which yields more than 40 elements per wavelength for analyses performed at a frequency of 1600 Hz at cold conditions and more than 50 for the warmer air present at full load and medium engine speed. The total number of nodes is about 23 000 in each tank, built up using 76 000 elements in the inlet tank and 83 000 in the outlet tank. This finite element mesh is shown in Fig. 5.

Figure 5. Finite element mesh of inlet and outlet tanks (the tanks have been moved together in the image).

The circular pipes, that connect the tanks to the rest of the gas exchange ducting of the engine, have diameters equal to 60 mm. Short parts of the measurement ducts, with the diameter 66 mm, are also included in the model in order to make sure that the waves are plane after the 90 degree bend in each tank. The same technique is also used at the connection to the cooling tubes where acoustical near fields are present due to the sudden contraction and expansion.

When using acoustical finite elements it is well known that the numerical integration of the pressure at a boundary is done exactly. However, the acoustic velocity
boundary condition is obtained only in a weak sense since it is of Neumann type. The prediction of the acoustic velocity at a boundary is obtained from extrapolation of the results at the integration points inside the elements to the boundary. To account for the inaccuracy resulting from the extrapolation the elements at all openings are made extremely short (0.1 mm).

To simulate the effect of yielding walls and propagation losses in the inlet and outlet sections, made of plastic, damping is applied as a complex speed of sound. Based on the investigation by Knutsson et al. in Ref. [35], an engineering estimate for the damping in intake system components made of plastic is obtained by putting the imaginary part of the speed of sound equal to one percent of the real part.

3.2.2. Multi-port mobility matrices

The acoustics of the tank volumes of the CAC can be represented by the admittance matrix relations $A_{\text{in}}$, which relates the acoustic volume velocities $q'$ to the pressures $p'$ at all openings as

$$
\begin{bmatrix}
q'_{\text{in}} \\
q'_{\text{out}}
\end{bmatrix} =
\begin{bmatrix}
A_{11} & A_{12} \\
A_{21} & A_{22}
\end{bmatrix}
\begin{bmatrix}
p'_{\text{in}} \\
p'_{\text{out}}
\end{bmatrix}
$$

(3.1)

where the indices \text{in} and \text{out} represent the in- and outlet of the two-port for the complete CAC, $T_1$ and $T_2$ are the inlet and outlet tank respectively. The vectors $q'_1$ and $p'_1$ are of size $N \times 1$ and represent the acoustic volume velocities and pressures respectively at the inlets to the cooling tubes. Here $N$ is the number of cooling tubes. The vectors $q'_2$ and $p'_2$ represent the corresponding variables at the outlets from the cooling tubes into the outlet tank. The direction of all volume velocities are with reference to Fig. 6.

Figure 6. Definition of acoustic variables at multi-port openings.

The admittance coupling between the inlet and outlet of one cooling tube can be expressed as

$$
\begin{bmatrix}
q'_1 \\
q'_2
\end{bmatrix} =
\begin{bmatrix}
M_{p,11} & M_{p,12} \\
M_{p,21} & M_{p,22}
\end{bmatrix}
\begin{bmatrix}
p'_1 \\
p'_2
\end{bmatrix}
$$

(3.2)

where the indices 1 and 2 represent the inlet and outlet of the cooling tube. The admittance matrix can be expanded in order to include the complete bundle of cooling tubes as
The admittance matrix above can also be written in a more compact form as:

$$\begin{bmatrix} q'_1' & q'_2' \end{bmatrix} = \begin{bmatrix} M_{p,11} & 0 & \cdots & 0 & M_{p,12} & 0 & \cdots & 0 \\ 0 & M_{p,11} & \cdots & 0 & M_{p,12} & \cdots & \cdots & \cdots \\ \vdots & \vdots & \ddots & \vdots & \vdots & \ddots & \ddots & \vdots \\ \vdots & \vdots & \cdots & 0 & \cdots & \cdots & \cdots & 0 \\ 0 & \cdots & 0 & M_{p,11} & 0 & \cdots & 0 & M_{p,12} \\ 0 & M_{p,21} & \cdots & 0 & M_{p,22} & \cdots & \cdots & \cdots \\ \vdots & \vdots & \cdots & 0 & \cdots & \cdots & \cdots & 0 \\ 0 & \cdots & 0 & M_{p,21} & 0 & \cdots & 0 & M_{p,22} \end{bmatrix} \begin{bmatrix} p'_1' \\ p'_2' \end{bmatrix}. \quad (3.3)$$

The admittance matrix above can also be written in a more compact form as:

$$\begin{bmatrix} q'_1' & q'_2' \end{bmatrix} = \begin{bmatrix} EM_{p,11} & EM_{p,12} \\ EM_{p,21} & EM_{p,22} \end{bmatrix} \begin{bmatrix} M_{p,11} & M_{p,12} \\ M_{p,21} & M_{p,22} \end{bmatrix} = \begin{bmatrix} E \end{bmatrix} \begin{bmatrix} A_{11} & a' \\ a^c & A' \end{bmatrix}, \quad (3.5)$$

where \( E \) is a unity matrix of size \( N \times N \). The matrices \( A_{T1} \) and \( A_{T2} \) can be divided into sub-matrices as

$$\begin{bmatrix} A \end{bmatrix} = \begin{bmatrix} A_{11} & a' \\ a^c & A' \end{bmatrix}, \quad (3.5)$$

where \( A' \) is a matrix of order \( N \times N \) where the first row and column in the matrix \( A \) has been excluded, \( a' \) and \( a^c \) are vectors of length \( N \) representing respectively the previously excluded first row and column in \( A \) except the first position which here is denoted \( A_{11} \). Hereafter Eq. (3.3) can be assembled into Eq. (3.1) which results in the following systems of equations

$$\begin{cases} q'_{in} = A_{T1,11} p'_{in} + a'_r p'_1' \\ M_{p,1,r} p'_1 + M_{p,1,p} p'_2 = a'_c p'_{in} + A'_{T1} p'_1' \end{cases} \quad (3.6)$$

and

$$\begin{cases} q'_{out} = A_{T2,11} p'_{out} + a'_r p'_2' \\ M_{p,21} p'_1 + M_{p,22} p'_2 = a'_c p'_{out} + A'_{T2} p'_2' \end{cases} \quad (3.7)$$

The last equations in system (3.6) and (3.7) yields \( p'_1 \) and \( p'_2' \) as

$$p'_1 = -\left[ M_{p,11} - A'_{11} \right]^{-1} M_{p,12} p'_2 + \left[ M_{p,11} - A'_{11} \right]^{-1} a'_1 \quad (3.8)$$

and
respectively. Substitution of Eq. (3.8) into (3.9) yields, after some rearrangements

\[
p'_2 = (E - B_2 B_1)^{-1} B_2 b'_2 p'_1 + (E - B_2 B_1)^{-1} b'_2 p'_o. \tag{3.10}
\]

This expression is used in Eq. (3.8) to obtain \(p'_1\) as

\[
p'_1 = \left(B_1 (E - B_2 B_1)^{-1} B_2 + E\right)b'_1 p'_o + B_1 (E - B_2 B_1)^{-1} b'_2 p'_o. \tag{3.11}
\]

Finally, Eqs. (3.10) and (3.11) is substituted into the first expressions in Eq. (3.6) and (3.7) to obtain the admittance matrix for the complete CAC unit, \(A_{CAC}\). After some simplifications this relation can be expressed as

\[
\begin{bmatrix}
p'_o' \\
q'_o'
\end{bmatrix} = 
\begin{bmatrix}
A_{T1,11} + a'_{t1} (B_1 B_2 B_1 + E)b'_1 & a'_{t1} B_1 B_2 b'_1 \\
a'_{t1} B_2 b'_1 & A_{T1,11} + a'_{out} B_2 b'_2
\end{bmatrix} 
\begin{bmatrix}
p'_o \\
q'_o
\end{bmatrix}. \tag{3.12}
\]

where \(B = (E - B_2 B_1)^{-1}\). The admittance matrix, that is now of the order 2 × 2 since there are only two ports on the component, can easily be transformed to the transfer matrix format as

\[
T_{CAC} = \frac{1}{A_{CAC,21}} \begin{bmatrix}
-A_{CAC,22} & 1 \\
-A_{CAC,12} A_{CAC,21} - A_{CAC,11} A_{CAC,22} & A_{CAC,11}
\end{bmatrix}. \tag{3.13}
\]

To establish the admittance matrices for the tanks in equation (3.1) \(N + 1\) load cases are required. These are composed by using identical FE models but with different boundary conditions. The relation between the admittance matrix and the calculated acoustic volume velocities and pressures at the 11 ports for the inlet tank is

\[
\begin{bmatrix}
q'_{in-LC1} & q'_{in-LC2} & \cdots & q'_{in-LC11} \\
q'_{1-LC1} & q'_{1-LC2} & \cdots & q'_{1-LC11}
\end{bmatrix} = \begin{bmatrix}
A_{T1} & p'_{in-LC1} & p'_{in-LC2} & \cdots & p'_{in-LC11} \\
p'_{1-LC1} & p'_{1-LC2} & \cdots & p'_{1-LC11}
\end{bmatrix} \tag{3.14}
\]

where \(q'_{in-LCn}\) and \(p'_{in-LCn}\) represent the resulting acoustic volume velocity and pressure for load case \(n\) and the vectors \(q'_{1-LCn}\) and \(p'_{1-LCn}\), of size \(N \times 1\), contain the corresponding values at the cooling tube inlets. The admittance matrix is obtained from the product of the matrix containing the volume velocities and the inverted pressure matrix. The corresponding matrix for the outlet tank is obtained accordingly, directions of the acoustic volume velocities with reference to Fig. 6.

### 3.2.3. Compensation for near field effects at multi-port openings

The 3D FE models of the tanks include a short part of the cooling tubes in order to establish a plane wave at the boundary of the multi-port. The FE formulation does not take into account any effects of boundary layers or turbulence in those narrow parts and as a result these will be missing in the final two-port. An approximate method to reintroduce the missing damping is to assume plane waves in the tubular parts of the
tank-model, virtually transfer the position of the multi-port openings back the section
where the tubes enter the tank and use the full length in the 2-ports for the tubes. The
formalism for this will be described below.

The relation between the acoustic variables at position 1 and 1A, where 1A is the
new position for the tube ports, can be described using Eq. (1.1) as

\[
\begin{pmatrix}
    p'_1 \\
    q'_1
\end{pmatrix}
= T_{11} T_{12} \begin{pmatrix}
    T_{11} \\
    T_{12}
\end{pmatrix}
\begin{pmatrix}
    p'_{1,i} \\
    q'_{1,i}
\end{pmatrix}
\] (3.15)

where \( n = 1, 2, \ldots, N \). For a straight duct the transfer matrix is

\[
\begin{pmatrix}
    T_{11} & T_{12} \\
    T_{21} & T_{22}
\end{pmatrix}
= e^{-ikML/(1-M^2)} \begin{pmatrix}
    \cos(kL_s/(1-M^2)) & iZ \sin(kL_s/(1-M^2)) \\
    (i/Z) \sin(kL_s/(1-M^2)) & \cos(kL_s/(1-M^2))
\end{pmatrix}
\] (3.16)

where \( k \) is the wave number, \( M \) the Mach number and \( Z \) the wave impedance; all three
taken for one single tube in the FE model. \( L_s \) is the translation distance which will be
negative for the inlet tank and positive for the outlet. Eq. (3.15) can be rewritten, in
order to represent all the ten cooling tubes, as

\[
\begin{pmatrix}
    p'_1 \\
    q'_1
\end{pmatrix}
= T_{11}E T_{12}E \begin{pmatrix}
    T_{11} \\
    T_{12}
\end{pmatrix}
\begin{pmatrix}
    p'_{1,i} \\
    q'_{1,i}
\end{pmatrix}
\] (3.17)

where \( E \) once again is a unity matrix of size \( N \times N \). Substitution of equation (3.17) into
equation (3.1) yields

\[
\begin{pmatrix}
    p'_{1,i} \\
    q'_{1,i}
\end{pmatrix}
= \begin{pmatrix}
    T_{11} & T_{12} \\
    T_{21} & T_{22}
\end{pmatrix}
\begin{pmatrix}
    T_{11}E \\
    T_{12}E
\end{pmatrix}
\begin{pmatrix}
    p'_{1,i} \\
    q'_{1,i}
\end{pmatrix}
\] (3.18)

where \( T_{ij} = T_{ji}E \). These expressions can thereafter be used to establish an admittance
relation between \( q'_{in}, q'_{1,i}, p'_{in}, \) and \( p'_{1,i} \) on the same form as Eq. (3.1). The expression
for \( q'_{1,i} \) is

\[
q'_{1,i} = \left[ T_{22} - A'T_{12} \right]^{-1} \left[ A'T_{11} - T_{21} \right] p'_{1,i} + \left[ T_{22} - A'T_{12} \right]^{-1} \left[ a'T_{11} + a'T_{12}F \right] p'_{in}
\] (3.19)

which thereafter is used to obtain \( q'_{in} \). After some simplifications this expression
becomes

\[
q'_{in} = \left[ A'T_{11} + a'T_{12}F \right] p'_{in} + \left[ a'T_{11} + a'T_{12}F \right] p'_{1,i}.
\] (3.20)

Finally the admittance matrix for a multi-port where the ports have been translated
becomes
which will be used to represent the inlet tank instead of Eq. (3.1). The procedure to obtain the admittance matrix for the outlet tank is identical.

3.3. Acoustic coupling at cross-section discontinuities

At the outlet from each cooling tube there is an abrupt expansion, see Fig. 7a, which needs a separate two-port for the cases when there is a mean flow present. Flow separation will be present hence entropy is generated due to the dissipation in the turbulent mixing region downstream the flow separation. This interaction between the acoustic field and the flow separation is, however, very complex and complete analytical models are complicated. An example of such a model is the work by Boij and Nilsson [36]. Several authors have suggested models based on simplified velocity fields. The model used here is based on the simple analysis in Ref [37] which has been found to give good results for low Mach numbers and rather low frequencies [38] - [40]. An area contraction is formed at the inlet of the cooling tubes. Depending on the flow situation, a vena contracta can be formed just after the area contraction. This implies some dissipation that can be treated using a similar approach as for the expansion [40] but here the isentropic contraction must be added.

![Figure 7. Mean flow velocity profiles at (a) area expansion (b) area contraction.](image)

### 3.3.1. Area expansion

The model for the area expansion is derived assuming incompressible mean flow and quasi-steady conditions. As the situation is not isentropic the conservation of momentum over the expansion is used as

\[
P_1 S_1 + \rho U^2_1 S_1 = P_2 S_2 + \rho U^2_2 S_2
\]

(3.22)

where \( P_n \) is the static pressure, \( U_n \) the mean axial flow velocity and the indices 1 and 2 are referring to Fig 7a. Conservation of mass yields

\[
S_1 U_1 = S_2 U_2 .
\]

(3.23)
Assuming constant density, \( \rho_1 = \rho_2 = \rho_0 \), and superimposed mean flow the pressure and velocity field can be divided as

\[
P_1 = p_{01} + p_1', \quad U_1 = U_{01} + u_1'
\]  \hspace{1cm} (3.24)

and

\[
P_2 = p_{02} + p_2', \quad U_2 = U_{02} + u_2'.
\]  \hspace{1cm} (3.25)

Together with the open area relation \( m_{\text{EXP}} = S_1 / S_2 \), Eq. (3.22) simplifies to

\[
p_1' = p_2' + 2\rho_0 U_{02} u_2' - 2\rho_0 m_{\text{EXP}} U_{01} u_1'.
\]  \hspace{1cm} (3.26)

where second order components have been neglected and the steady components subtracted. The transfer matrix relation can thereafter be calculated as

\[
\begin{bmatrix}
p_1' \\
a_1'
\end{bmatrix} =
\begin{bmatrix}
1 & 2Z_2 M_{\text{a}2}

\end{bmatrix}
\begin{bmatrix}
1 - \frac{1}{m_{\text{EXP}}}

\end{bmatrix}
\begin{bmatrix}
p_2' \\
a_2'
\end{bmatrix}
\]  \hspace{1cm} (3.27)

where the wave impedance is \( Z_2 = \rho_0 c_0 / S_2 \) and the Mach number is \( M_{\text{a}2} = U_{02} / c_0 \).

### 3.3.2. Area contraction

For the case of an area contraction, as shown in Fig. 7b, the flow is homogenous and without losses at the large section (1) and up to the possible vena contracta (c). The vena contracta might exist for situations where the Reynolds number in the cooling tube is low. The extra losses introduced by the small expansion will be included in the derivation below but will for most cases be very small. Here the conservation of energy is described using Bernoulli’s equation for the mean flow between section 1 and the vena contracta as

\[
P_1 + \frac{1}{2}\rho_0 U_1^2 = P_c + \frac{1}{2}\rho_0 U_c^2.
\]  \hspace{1cm} (3.28)

Between section (c) and (2) a non reversible expansion takes place which can be described using the conservation of momentum in Eq. (3.22). Here the indices has to be changed which yields

\[
P_c S_2 + \rho_0 U_c^2 S_c = P_2 S_2 + \rho_0 U_2^2 S_2.
\]  \hspace{1cm} (3.29)

Using Eq. (3.29) in (3.28) together with conservation of mass as stated in Eq. (3.23) yields

\[
p_1' = p_2' + Z_1 M_{\text{a}2} a_2' \left[ \frac{1}{m_{21}} \left( \frac{1}{m_{c2}} - \frac{2}{m_{c2}} + 2 \right) - 1 \right]
\]  \hspace{1cm} (3.30)

where the area ratios are defined as \( m_{21} = S_2 / S_1 \) and \( m_{c2} = S_c / S_2 \). For the case without losses Eq. (3.30) simplifies to

25
Finally the transfer matrix relation can be calculated as

\[
\begin{bmatrix}
    p'_1 \\
    q'_1
\end{bmatrix} =
\begin{bmatrix}
    1 & Z_1 M_{01} \\
    0 & 1
\end{bmatrix}
\begin{bmatrix}
    \frac{1}{m_{21}^2} - 1 \\
    \frac{1}{m_{21}^2} \left( \frac{1}{m_{c2}^2} - \frac{2}{m_{c2}^2} + 2 \right) - 1
\end{bmatrix}
\begin{bmatrix}
    1 \\
    0
\end{bmatrix}
\begin{bmatrix}
    p'_2 \\
    q'_2
\end{bmatrix}.
\]

(3.32)

The ratio that includes the vena contracta area depends on the shape of the inflow and is approximately between 0.5 and 1 [41] with an expected value between 0.61 and 0.65. More accurate estimates can be obtained either by approximate calculations or from experiments.

4. Measurements

In order to validate the proposed model, measurements have been performed using the flow acoustic test facility available at MWL/KTH. All experiments were done at room temperature for different flow speeds. The Mach number in the main ducts was varied between 0 and 0.1 in steps of 0.025; values chosen as being representative for engine operating conditions. This implies that in the cooling tubes, where the area is expanded by a factor 1.2-1.3 compared to the measurement duct, the Mach number will be as most 0.08. The test ducts used during the experiments consisted of standard steel pipes, with diameters equal to 66 mm, chosen in order to relate to the in- and outlet of the charge air coolers. Eight loudspeakers, equally divided between the up- and downstream side of the rig, were used as acoustic sources, as shown in Fig. 8 and 9. The test rig is terminated at each end by a dissipative silencer and a horn to reduce the effects of standing waves. Fluctuating pressures were measured by using six condenser microphones (Brüel &Kjaer ¼-inch 4938) flush mounted in the wall of the steel pipes. All measurements were performed using random noise excitation. The fluctuating pressures were transformed to the frequency domain with a resolution of 5 Hz and 4000 averages. The two-port matrix for the test object was obtained using the source switching technique as described in [42]. From the measured two-port data the transmission loss for the CAC was calculated using the expression [23]

\[
TL = 10 \cdot \log \left\{ \frac{1 + M_{\text{IN}}}{1 + M_{\text{OUT}}} \right\}^2 \frac{Z_{\text{OUT}}}{4Z_{\text{IN}}} \left| T_{11} + \frac{T_{12}}{Z_{\text{OUT}}} + Z_{\text{IN}}T_{21} + \frac{Z_{\text{IN}}T_{22}}{Z_{\text{OUT}}} \right|^2
\]  

(4.1)

and thereafter compared to the transmission loss predicted from theory. The Mach numbers and the acoustic wave impedances in the in- and outlet of the CAC are denoted \(M_{\text{IN}}, M_{\text{OUT}}, Z_{\text{IN}}\) and \(Z_{\text{OUT}}\) in Eq. (4.1). To minimize the effects of flow noise at the microphones, source correlation using the loudspeaker voltage signal was performed.
The transmission loss obtained from measurements in the direction of flow is shown in Fig. 10. The effect of the mean flow appears to be small at higher frequencies with differences of less than 1 dB. As expected an increased flow speed will result in decreased damping. Interesting is the substantial amount of low frequency damping that clearly increases with increasing Mach number. This effect can, as will be seen in the simulation results, be explained by Howe’s theory as transfer of acoustic energy to the turbulent field due to acoustic boundary layers that are thicker than the viscothermal sub-layers. If the engine breathing noise is considered as the source the transmission loss in the upstream direction is more interesting since the CAC is located on the intake side of the engine. The upstream transfer matrix (\(T_{up}\)) can easily be obtained just by inversion of the downstream transfer matrix (\(T_{down}\)). In order to get the directions for the velocities correct, the signs of the elements at the positions (1, 2) and (2, 1) have to be reversed as
Finally, the transmission loss in the upstream direction can be calculated using Eq. (4.1). The resulting transmission loss is shown in Fig. 10. The effect of the mean flow is more pronounced than in the downstream direction especially at higher frequencies where increasing flow speed corresponds to increasing damping. At low frequencies is the damping due to sound-turbulence interaction about 1 dB larger than in the downstream direction. It is interesting to detect this relatively large amount of turbulent damping for this charge air cooler compared to catalytic converters and particular filters where no effect of turbulence is present.

\[
T_{up} = \frac{1}{T_{down,11} T_{down,22} - T_{down,12} T_{down,21}} \begin{bmatrix} T_{down,22} & T_{down,12} \\ T_{down,21} & T_{down,11} \end{bmatrix}. \tag{4.2}
\]

Figure 10. Measured transmission loss for the complete CAC in the downstream direction. 
- , \( M = 0.0 \); -----, \( M = 0.025 \); ····, \( M = 0.05 \); -··-, \( M = 0.075 \); oooo, \( M = 0.01 \).
Figure 11. Measured transmission loss for the complete CAC in the upstream direction.

$\cdots$, $M = 0.0$; $\cdots$, $M = 0.025$; $\cdots$, $M = 0.05$; $\cdots$, $M = 0.075$; $\cdots$, $M = 0.01$.

5. Model validation at cold conditions

5.1. The case without mean flow

5.1.1. Cooling tubes

The effect of using different cross-sectional shapes to model the cooling tubes is first studied for the case without flow. Two 2D models are used for this purpose; one circular and one with the shape of an isosceles trapezium – in this particular case very similar to a triangle. The meshes in the models are shown in Fig. 12. Symmetry boundary conditions are used to reduce the model size. The circular model consists of a slice with an angle of 22.5 degrees while the isosceles trapezium is just symmetric in one direction and therefore is represented by half the geometry. The mesh, which is biased towards the duct wall in order to resolve the large gradients in the boundary layers, consists of 35 elements and 170 nodes for mesh (a). Corresponding figures for the trapezium geometry in mesh (b) is 270 elements and 1155 nodes. The calculated attenuation and phase speed ratio for the mesh (a) and (b) are shown in Fig. 13. Here, the attenuation and phase speed ratio of a wave are defined as

\[
\text{attenuation} = 8.686 \left| \text{Im} \left( k_0 \Gamma \right) \right| \quad \text{(dB/m)} \tag{5.1}
\]

and

\[
\frac{c}{c_0} = \left| \frac{1}{\text{Re} \left( \Gamma \right)} \right|. \tag{5.2}
\]
The difference between the Zwikker and Kosten solution and the one obtained using the 2D FE solution for a circular cross-section is less than 0.01 dB/m which indicates that the element discretisation has converged. In Fig. 13 it can also be noticed that the difference between the circular solutions and the trapezium solution is very small. There is less than 0.1 dB/m more damping for the trapezium solution. The trend is similar for the phase speed ratio, see Fig. 14.

If the hydraulic diameter is used to calculate the shear wave number, the lower frequency limit at 50 Hz will correspond to a shear wave number of $s = 6$ whereas the upper frequency limit at 1600 Hz corresponds to $s = 35$. The “reduced frequency” for the frequency extremes are $k_0a = 0.0012$ and 0.040 respectively. The requirements that $k_0a < 1$ and $k_0a / s < 1$ are thereby fulfilled and the Zwikker and Kosten approximations are valid [13]. The lower shear wave number is definitely in the same range as the catalytic converters that were studied in [14]-[15]. The upper limit is, however, far exceeding the shear wave number of 10 that was the largest reported. This indicates that the wide duct approximation by Kirchhoff might also be a good choice to simplify the calculations. The axial velocity profile was observed in [13] to consist of an almost flat core and small peaks close to the tube wall is also present for the trapezium geometry; see Fig 15, which emphasizes the need of a biased mesh with fine resolution close to the walls.

Figure 12. 2D meshes used to calculate two-ports for the cooling tubes. (a) circular cross-section; (b) isosceles trapezium. The two shapes have identical hydraulic diameters ($D_h = 2.7254$ mm).
Figure 13. Predicted attenuation for one cooling tube. —, Zwikker & Kosten; ○○ ○○, 2D FE: circular geometry; ΔΔΔΔ, 2D FE: trapezium geometry.

Figure 14. Predicted phase speed ratio for one cooling tube. —, Zwikker & Kosten; ○○ ○○, 2D FE: circular geometry; ΔΔΔΔ, 2D FE: trapezium geometry.
5.1.2. Complete charge air cooler

The complete charge air cooler is modelled using two approaches; completely based on two-ports using the SIDLAB software [43] or based on the multi-port approach to represent the tanks together with two-ports for the cooling tubes as was described in section 3. In order to study the sensitivity to the amount of damping in the 3D FE-models used to obtain the multi-ports two different values on the speed of sound are used; \( c = c_0 \) and \( c = c_0(1 + i/100) \) where \( c_0 \) is the adiabatic speed of sound and \( i \) is the complex number. This artificial damping will compensate the FE-results for the viscothermal losses occurring at the boundaries, damping in the plastic walls and radiation.

In Fig. 16 the transmission loss from these three predictions is shown together with the experimentally obtained curve. It can be observed that the two-port approach provides good results with deviations less than 1 dB up to 1200 Hz where cross modes in the inlet/outlet tanks start to propagate and the response is shifted. The accuracy of the results provided from the multi-port approach with complex speed of sound is very good. The deviation stays within less than 0.5 dB up to 1400 Hz, indeed an impressive result. Above 1400 Hz the deviation is between 0.5 and 1 dB which still is a good result. The predictions made without losses in the tanks are not as good with deviations increasing with increasing frequency. In the following calculations, for the case of superimposed flow, the speed of sound \( c = c_0(1 + i/100) \) is used in the 3D FE-model used to extract the multi-ports.
Figure 16. Transmission loss for complete CAC at $M = 0$. —, Measured; ----, Predicted using multi-port technique and 2D FE trapezium geometry (speed of sound $c = c_0(1 + i/100)$ in 3D FE-model); ····, Predicted using multi-port technique and 2D FE trapezium geometry (speed of sound $c = c_0$ in 3D FE-model); -··-, Predicted using two-port technique and 2D FE trapezium geometry.

5.2. The case with mean flow

5.2.1. Cooling tubes

The study of the dependence on using circular or trapezium shape to model the cooling tubes is here extended to include a small mean flow. The effect of approximating the profile of the mean flow with a constant compared to a parabolic profile is also studied. For this purpose the model by Dokumaci [14] and the modified 2D FE approach [16] is used. The flow speed in the tubes was taken as $M = 0.08$ which corresponds to $M = 0.1$ in the measurement ducts (diameter 0.066m). If the hydraulic diameter is used the Reynolds number becomes $Re = 5000$, which for a circular cross-section indicates that the flow is not laminar but rather turbulent or in the transition zone. For triangular sections, as was discussed in section 2, transition occurs gradually over the cross-section when the Reynolds number is increased. It is therefore not possible to state whether the flow is laminar or turbulent in the entire cross-section for this flow speed. In order to find out if the low frequency damping that was observed in the experiments for this flow speed is related to interaction between the acoustic boundary layers and the turbulence the model by Howe is used. To the authors’ knowledge there has not been published any data using this model for Reynolds numbers below $Re = 10000$. The model requires that the mean flow profile is flat which might not be the case for the less turbulent flow in this particular case.

Figure 17 and 18 shows the attenuation and phase speed ratio in the direction opposite to the mean flow for the predictions. The difference between using the trapezium geometry and the circular one is very small for both the case of flat flow profile and parabolic; although the deviations for the later are slightly more pronounced.
Interesting is that the plug flow profile solution underestimates the damping compared to the corresponding parabolic solution. The effect increases with increasing frequency and can also be seen for the phase speed ratio. Identical observations can also be done for waves propagating with the mean flow, see Fig. 19 and 20. However, here the damping is less than for the case without flow and the parabolic profile solution gives further less damping than the plug flow solution.

The solution obtained using the model by Howe captures the effect of low frequency damping in both directions as expected. However, the high frequency behaviour is based on the “wide” Kirchhoff solution [13] and therefore results in less damping than the Zwikker Kosten solution in the low flow limit.

Figure 17. Predicted attenuation in the upstream direction for one cooling tube at \( M = 0.08 \).

—, Zwikker & Kosten (\( M = 0 \)); ○○○○, Dokumaci (flat mean flow profile); ∆∆∆∆, 2D FE: trapezium geometry (flat mean flow profile); ○○○○, 2D FE: circular geometry (parabolic mean flow profile); ∆∆∆∆, 2D FE: trapezium geometry (parabolic mean flow profile), ---- Howe.
Figure 18. Predicted phase speed ratio in the upstream direction for one cooling tube at $M = 0.08$. —, Zwikker & Kosten (M = 0); ○○○○, Dokumaci (flat mean flow profile); △△△△, 2D FE: trapezium geometry (flat mean flow profile); ○○○○, 2D FE: circular geometry (parabolic mean flow profile); △△△△, 2D FE: trapezium geometry (parabolic mean flow profile), ---- Howe.

Figure 19. Predicted attenuation in the downstream direction for one cooling tube at $M = 0.08$. —, Zwikker & Kosten (M = 0); ○○○○, Dokumaci (flat mean flow profile); △△△△, 2D FE: trapezium geometry (flat mean flow profile); ○○○○, 2D FE: circular geometry (parabolic mean flow profile); △△△△, 2D FE: trapezium geometry (parabolic mean flow profile), ---- Howe.
Figure 20. Predicted phase speed ratio in the downstream direction for one cooling tube at $M = 0.08$. —, Zwikker & Kosten ($M = 0$); ◦◦◦◦, Dokumaci (flat mean flow profile); △△△△, 2D FE: trapezium geometry (flat mean flow profile); ○○○○, 2D FE: circular geometry (parabolic mean flow profile); △△△△, 2D FE: trapezium geometry (parabolic mean flow profile), ---- Howe.

5.2.2. Complete charge air cooler

For the case of a complete CAC and a present mean flow the multi-port approach is used to represent the tanks together with two-ports for the cooling tubes. To represent the cooling tubes it was chosen to use the two-ports obtained in the previous section from the 2D FE model (trapezium geometry) with parabolic mean flow profile and the model by Howe ($M = 0.08$). The models assuming a flat mean flow profile will produce results somewhere in between of these two. The profile of the actual mean flow is not known, as was discussed in section 2.1, and will also change along the axial direction since the zone of unestablished flow might be of significant length.

The coupling elements for cross-sectional area discontinuities that were described in section 3.3 are used with an open area ratio of 0.61 for the vena contracta appearing at the inlet to the cooling tubes. This approximate value can probably be improved but will most likely by larger why this estimate is conservative [41]. The effect on the assembled CAC is, however, very small. The predicted and measured transmission loss for sound propagating through the CAC in the upstream direction is shown in Fig. 21. The high frequency estimate from using 2D FE shows very good agreement to the experimentally obtained one except in the frequency band between 700 and 1000 Hz where some smaller discrepancies can be observed. Possible explanations for those deviations might be the effect of neglecting the dissipation that is included in the original derivation in [16], from neglecting the matching of the near fields when coupling the two-ports to the tanks or from neglecting the in-plane velocity components that was shown in [19] to yield increasing effects when the shear wave number is increased.

Although the model by Howe does not yield as good predictions as the 2D FE solution in the high frequency band, where the deviation is about 1dB, it gives the best estimates in the low frequency band up to 300 Hz. The model does still give an under-
prediction of the measurements data but the effect of extra damping, due to the interaction between the turbulence and the sound field, is captured which can’t be done by the 2D FE solution.

![Graph](image1)

**Figure 21.** Transmission loss in the upstream direction for complete CAC at \( M = 0.1 \) in main duct \( (M = 0.08 \) in cooling tubes). —, Measured; ----, Predicted using multi-port technique and 2D FE (trapezium geometry) with parabolic mean flow profile; --··, Predicted using multi-port technique and Howe’s model.

![Graph](image2)

**Figure 22.** Transmission loss in the downstream direction for complete CAC at \( M = 0.1 \) in main duct \( (M = 0.08 \) in cooling tubes). —, Measured; ----, Predicted using multi-port technique and 2D FE (trapezium geometry) with parabolic mean flow profile; --··, Predicted using multi-port technique and Howe’s model.
Similar observations can be made for the case where the two-port is in the same direction as the mean flow. The predicted attenuation is smaller than the experimentally obtained. The deviation in the high frequency domain is slightly larger for the 2D FE solution than in the upstream direction but stays within 1 dB while the model by Howe shows the same accuracy as before. The low frequency prediction is, however, slightly worsened and gives an underestimate of approximately 2 dB. For low frequencies when the wave-length is much smaller than the length scale of the CAC, the transmission will be defined by the pressure drop. The transmission loss should therefore independent of the direction of the two-port, which is almost the case in the measurements. Why the results from using Howe’s model do not show this behaviour is not completely understood. One possible explanation could be that Howe’s model assumes that the sub-layers are thin which might not be the case for the Reynolds number appearing in this study.

6. Predicted damping for a CAC at operating conditions

To investigate the acoustical properties for a CAC at operating conditions the gas properties is updated to correspond to values taken upstream and downstream a CAC mounted on a running engine. This engine is a 5-cylinder diesel engine and the values in Table 1 were taken in an engine test bench at an engine revolution speed of 4200 rpm where the massflow was 663 kg/h.

<table>
<thead>
<tr>
<th></th>
<th>Upstream CAC</th>
<th>Downstream CAC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature [K]</td>
<td>443</td>
<td>340</td>
</tr>
<tr>
<td>Static pressure [kPa]</td>
<td>253</td>
<td>244</td>
</tr>
<tr>
<td>Reynolds number [-]</td>
<td>4900</td>
<td>5900</td>
</tr>
<tr>
<td>Mach number [-]</td>
<td>0.052</td>
<td>0.047</td>
</tr>
<tr>
<td>Speed of sound [m/s]</td>
<td>442</td>
<td>370</td>
</tr>
<tr>
<td>Density [kg/m³]</td>
<td>1.99</td>
<td>2.50</td>
</tr>
<tr>
<td>Dynamic viscosity ×10⁻⁵ [Pa·s]</td>
<td>2.44</td>
<td>2.01</td>
</tr>
</tbody>
</table>

Table 1. Gas data at operating conditions for 5-cylinder diesel engine

The density is calculated using the law of ideal gases, the dynamic viscosity, the thermal conductivity and the specific heat capacity at constant pressure are only dependent on the temperature at these low pressures and can be taken from standard books [44]. The adiabatic speed of sound is calculated as \( c_0 = \sqrt{\gamma RT} \).

Three different solutions are used are used to extract the two-ports that are used together with the multi-port approach. The geometry is approximated as circular but the shape of the mean flow profile is either flat or parabolic and laminar as well as turbulent flow is considered. The temperature and pressure gradient can be treated by dividing the cooling tubes into several two-ports based on different gas properties coupled in cascade as was shown in Ref. [27] and [38]. The convergence is, however very fast and the difference between using one two-port and ten results in differences less than 0.1 dB. The predicted transmission loss in the upstream direction for the complete CAC at operating conditions is shown in Fig. 23. The gas properties in the tanks are based on the up- and downstream values from Table 1 and the two-ports representing the cooling tubes are calculated using the average values.
Figure 23. Transmission loss in the upstream direction versus frequency for CAC mounted on engine operating at 4200 rpm. —, Predicted using multi-port technique and Dokumaci’s model [14]; ---- Predicted using multi-port technique and Howe’s model [26], ···· Predicted using multi-port technique and 2D FE with circular geometry and parabolic mean flow.

Figure 24. Transmission loss in the downstream direction versus frequency for CAC mounted on engine operating at 4200 rpm. —, Predicted using multi-port technique and Dokumaci’s model [14]; ---- Predicted using multi-port technique and Howe’s model [26], ···· Predicted using multi-port technique and 2D FE with circular geometry and parabolic mean flow.
The difference between the three solutions is very small and stays within 0.5 dB except in the low frequency region where the model by Howe yields about 2 more dB as expected. The transmission loss is generally smaller than for the cold case and the distance between the peaks are larger due to the larger speed of sound at higher temperatures. The low frequency damping is just between 2-3 dB which is smaller than the 4-5 dB that was predicted at cold conditions. Bearing in mind that the difference between predictions and measurements at cold conditions, there is probably one extra dB not predicted that can be added for the hot case as well, thus resulting in 3-4 dB. Concerning transmission loss in the direction of flow the conclusions are similar. The difference between the two directions is less than 1 dB in the range between 50 and 1600 Hz.

7. Summary and conclusions

Sound transmission through charge air coolers has been studied. The frequency range under consideration was low and medium where non-plane waves exist in the inlet/outlet tanks. A new hybrid methodology for calculation of the global acoustic two-port for an automotive intake/exhaust device consisting of volumes with multiple openings coupled to narrow channels has been proposed. An attractive formalism for extraction of multi-ports from numerical 3D finite elements has been derived and presented. The two-ports for the narrow channels include a complete treatment of the losses due to viscous and thermal boundary layers for cases without flow as well as with a superimposed mean flow. The 2D finite element scheme derived by Astley and Cummings [16] and the model by Dokumaci [14] have been used to study the effect of using an isosceles trapezium or a circular cross-section with equivalent hydraulic diameter for the narrow channels. The effect of the shape of the cross-section has been found to be small, but increases with increasing mean flow. Flat profiles as well as parabolic has been used for the superimposed incompressible mean flow and the difference between these has been shown to be of larger importance than the shape of the cross-section.

The hybrid methodology has been applied as the first complete model for charge air coolers. The model has been validated to experimentally obtained transmission loss data at room temperature for a charge air cooler designed for use on a passenger car with very good accuracy. For the case of a superimposed mean flow a considerable amount of low frequency damping due to turbulence has been observed. The model for interaction between sound waves and turbulence proposed by Howe [26] has for the first time been applied to extract two-ports. The use of these two-ports within the hybrid model describes the low frequency attenuation with reasonable accuracy but gives slightly less accurate predictions at higher frequencies.

The proposed models have been used to predict the sound transmission through the charge air cooler at operating conditions. Mass flow, temperature and pressure data for the enclosed air, taken at positions upstream and downstream a CAC mounted on an engine in test bench, was used to calculate the required gas properties. The predicted transmission loss at operating conditions is smaller than those obtained at cold conditions without pressure and temperature gradients. The amount of damping is, however, still significant at all frequencies which gives previously underestimated opportunities to control the final noise spectra of turbo-charged engines.
Acknowledgements

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References


Appendix A

The shape functions used for the isoparametric finite elements with nine nodes in section 2.3 are:

\[
\phi_1^{(e)}(\xi, \eta) = \frac{1}{4} \xi (1-\xi) \eta (1-\eta) \\
\phi_2^{(e)}(\xi, \eta) = -\frac{1}{2} (1+\xi)(1-\xi) \eta (1-\eta) \\
\phi_3^{(e)}(\xi, \eta) = -\frac{1}{4} \xi (1+\xi) \eta (1-\eta) \\
\phi_4^{(e)}(\xi, \eta) = -\frac{1}{2} (1-\xi)(1+\eta) (1-\eta) \\
\phi_5^{(e)}(\xi, \eta) = (1+\xi)(1-\xi) (1+\eta)(1-\eta) \\
\phi_6^{(e)}(\xi, \eta) = \frac{1}{2} \xi (1+\xi)(1+\eta) (1-\eta) \\
\phi_7^{(e)}(\xi, \eta) = -\frac{1}{4} \xi (1-\xi) \eta (1+\eta) \\
\phi_8^{(e)}(\xi, \eta) = \frac{1}{2} (1+\xi)(1-\xi) \eta (1+\eta) \\
\phi_9^{(e)}(\xi, \eta) = \frac{1}{4} \xi (1+\xi) \eta (1+\eta)
\]

The derivatives of the shape functions are:

\[
\frac{\partial \phi_1^{(e)}(\xi, \eta)}{\partial \xi} = \frac{1}{4} ((1-\xi) \eta (1-\eta) - \xi \eta (1-\eta)) \\
\frac{\partial \phi_1^{(e)}(\xi, \eta)}{\partial \eta} = \frac{1}{4} (\xi (1-\xi) (1-\eta) - \xi (1-\xi) \eta) \\
\frac{\partial \phi_2^{(e)}(\xi, \eta)}{\partial \xi} = -\frac{1}{2} ((1-\xi) \eta (1-\eta) - (1+\xi) \eta (1-\eta)) \\
\frac{\partial \phi_2^{(e)}(\xi, \eta)}{\partial \eta} = -\frac{1}{2} ((1+\xi)(1-\xi) (1-\eta) - (1+\xi)(1-\xi) \eta) \\
\frac{\partial \phi_3^{(e)}(\xi, \eta)}{\partial \xi} = -\frac{1}{4} ((1+\xi) \eta (1-\eta) - \xi \eta (1-\eta)) \\
\frac{\partial \phi_3^{(e)}(\xi, \eta)}{\partial \eta} = -\frac{1}{4} (\xi (1+\xi)(1-\eta) - \xi (1+\xi) \eta)
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \xi} = -\frac{1}{2} ((1-\xi)(1+\eta)(1-\eta) - \xi(1+\eta)(1-\eta))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \eta} = -\frac{1}{2} ((\xi(1-\xi)(1-\eta) - \xi(1+\xi)(1+\eta))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \xi} = ((1-\xi)(1+\eta)(1-\eta) - (1+\xi)(1+\eta)(1-\eta))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \eta} = ((1+\xi)(1-\xi)(1-\eta) - (1+\xi)(1+\eta)(1+\eta))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \xi} = \frac{1}{2} ((1+\xi)(1+\eta)(1-\eta) + \xi(1+\eta)(1-\eta))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \eta} = \frac{1}{2} (\xi(1+\xi)(1-\eta) - \xi(1+\xi)(1+\eta))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \xi} = -\frac{1}{4} ((1-\xi)(1+\eta) - \eta(1+\eta))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \eta} = -\frac{1}{4} (\xi(1-\xi)(1+\eta) + \xi(1-\xi))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \xi} = \frac{1}{2} ((1-\xi)(1+\eta) - (1+\xi)(1+\eta))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \eta} = \frac{1}{2} ((1+\xi)(1-\xi)(1+\eta) + (1+\xi)(1-\xi))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \xi} = \frac{1}{4} ((1+\xi)(1+\eta) + \xi(1+\eta))
\]
\[
\frac{\partial \phi^{(c)}(\xi, \eta)}{\partial \eta} = \frac{1}{4} (\xi(1+\xi)(1+\eta) + \xi(1+\xi))
\]