Modelling of IC-Engine Intake Noise

Magnus Knutsson

Doctoral Thesis

Stockholm, Sweden 2009

KTH CICERO

The Marcus Wallenberg Laboratory for Sound and Vibration Research

Postal address
Royal Institute of Technology
MWL / AVE
SE–100 44 Stockholm
Sweden

Visiting address
Teknikringen 8
Stockholm

Contact
+46 31 325 40 64
magnuskn@kth.se
www.cicero.kth.se
www.volvocars.com

Akademisk avhandling som med tillstånd av Kungliga Tekniska Högskolan i Stockholm framläggs till offentlig granskning för avläggande av teknologie doktorsexamen måndagen den 1:e juni 2009, kl. 13.15 i sal F3, Lindstedtsvägen 26, Kungliga Tekniska Högskolan, Stockholm.

TRITA–AVE 2009:16
ISSN 1651–7660

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 CO_2 emissions and the efficiency of IC-engines, as well as new techniques such as homogeneous charge compression ignition (HCCI) might worsen the noise situation. Nonlinear one-dimensional (1D) gas dynamics time-domain simulation software packages 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.

The first part of this thesis treats two different strategies to combine nonlinear and linear modelling of intake systems in order to improve the accuracy of the noise predictions. *Paper A* describes how a linear time-invariant one-port source model can be extracted using nonlinear gas dynamics simulations. Predicted source data for a six-cylinder naturally aspirated engine is validated using experimental data obtained from engine test bench measurements. *Paper B* presents an experimental investigation on the influence of mean flow and filter paper on the acoustics of air intake systems. It also suggests how a linear source, extracted from nonlinear simulations can be coupled to acoustic finite elements describing the intake system and to boundary elements describing the radiation to the surroundings. Simulations and measurements are carried out for a large number of engine revolution speeds in order to make the first systematic validation of an entirely virtual intake noise model that includes 3D effects for a wide engine speed range. In *Paper C* an initial study on a new technique for the use of two-ports in the time domain for automotive gas dynamics applications is presented. Tabulated frequency-domain two-port data representing an air cleaner unit on the impedance form is inversely transformed to the time domain and used as FIR filters in nonlinear time-domain calculations.

The second part of the thesis considers detailed modelling of sound propagation in capillary tubes. Thermoviscous boundary effects and interaction between sound waves and turbulence can, for sufficiently narrow tubes, yield significant attenuation. Several components in the gas exchange system of IC-engines are based on arrays of narrow ducts and might have underestimated silencing capabilities. In particular the sound transmission properties of charge air coolers (CAC) have so far gained interest from very few authors. In *Paper D* a detailed investigation of the acoustic properties of CACs is presented. As a result 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, is proposed. Interesting low frequency damping most likely due to interaction between sound and turbulence is observed in the experimental data. A new numerical model that describes this dissipative effect in narrow tubes is suggested in *Paper E*. Validation is carried out using experimental data from the literature. Finally, in *Paper F* the CAC-model presented in *Paper D* is updated with the new model for interaction between turbulence and acoustic waves proposed in *Paper E*. The updated model is shown to yield improved predictions.

Keywords: IC-engine, intake noise, gas dynamics, linear source data, frequency domain, 2-port, losses, air cleaner unit, filter paper, flow, FEM, BEM, charge air cooler, narrow tube, thermoviscous, turbulence

To Susanne,
Johan, Daniel and David

Acknowledgments

The research in this thesis has been carried out as an industrial postgraduate student project at Volvo Car Corporation (VCC), Göteborg in cooperation with the Marcus Wallenberg Laboratory (MWL) at KTH, Stockholm. I'm very grateful to VCC for this very generous opportunity and to MWL where I have been taken care of more than well.

There are four people who deserve my deepest appreciation due to their contribution to this project. I will here take the opportunity to acknowledge my two supervisors at MWL, *Mats Åbom* and *Hans Bodén,* for their guidance through this indeed very inspiring period of my life. Their all-time-availability for answering my questions has been of utmost importance. I also gratefully acknowledge my industrial supervisor at VCC – *Johan Lennblad* for his never ending encouragement and outstanding supervision when it comes to mathematics and numerical methods. My manager at VCC *Anders Thorell*, who has supported and protected this project from the beginning to the end, also deserves my warmest appreciations.

Colleagues at VCC that are acknowledged include *Pekka Saviharju* who shared my interest in intake noise and introduced me to my academic supervisors, *Johan Stenson* who helped me to initiate the project, *Anders Petrén* who supported the need for answers on the technical questions I have dealt with, *Thomas Johannesson* for discussions about scientific writing and life as an industrial PhD-student, *Fredrik Strömstedt* for qualified MATLAB support and *Börje Grandin* for help with funding issues. A thank is dedicated to *Tomas Hannebäck* for being my mentor. I would also like to thank the rest of my present and former colleagues at the department Engine CAE at VCC for interesting discussions.

My appreciation also goes to my colleagues at MWL. In particular I acknowledge *Susann Boij, Mikael Karlsson, Hans Rämmal* and *Ragnar Glav* for fruitful discussions in terms of acoustics and help with practical issues. Kent Lindgren and Danilo Prelevic are acknowledged for their help during all experiments. A warm thank must be dedicated to *Laszlo Fuchs* at KTH for his yearly review of my work during the entire project period. His friendly advices have been of great value to me. *Henrik Alfredsson* is acknowledged for scientific input but also for incorporating me in KTH CICERO.

I'm grateful to *my parents* who always have encouraged my academic efforts. I also give my appreciation to *Ulla*, my mother-in-law for invaluable help at home.

To my beloved wife *Susanne* who makes my life a gift – Your understanding, patience and support during this period can not be enough acknowledged. I dedicate this thesis to you since I would never have managed to write it without you.

Finally, I also dedicate this thesis to my sons *Johan*, *Daniel* and *David* who will now get their father back – I hope I have managed to inspire your interest into deeper knowledge and that you one day will read this thesis.

Floda in March 2009
Magnus Knutsson

This doctoral thesis consists of an introduction to the research with a summary and the following appended papers:

Paper A

M. Knutsson and H. Bodén, IC-engine acoustic source data from non-linear simulations, *SAE Technical Paper2007–01–2209*, 2007.

Paper B

M. Knutsson, J. Lennblad and H. Bodén, Prediction of IC-engine intake orifice noise using 3D acoustic modelling and linear source data based on non-linear CFD, *Proceedings of the 5th International Styrian Noise, Vibration and Harshness Congress, in cooperation with SAE International*, Graz, Austria, 2008.

Paper C

M. Knutsson, J. Lennblad, H. Bodén and M. Åbom, A study on acoustical time-domain two-ports based on digital filters with application to automotive air intake systems, (2009).

Paper D

M. Knutsson and M. Åbom, Sound propagation in narrow tubes including effects of viscothermal and turbulent damping with application to charge air coolers, *Journal of Sound and Vibration* 320 (2009) 289–321.

Paper E

M. Knutsson and M. Åbom, The effect of turbulence damping on acoustic wave propagation in tubes, *Manuscript submitted to Journal of Sound and Vibration* (2009).

Paper F

M. Knutsson, A note on acoustic wave propagation in charge air coolers, *Manuscript submitted to Journal of Sound and Vibration* (2009).

Division of work between the authors

The formulations of the problems and proposals for the methodologies described in this thesis have been identified in cooperation between Magnus Knutsson and the supervisors Mats Åbom and Hans Bodén. Mats Åbom supervised *Paper D, E* and *F*. Hans Bodén supervised *Paper A* and *B* and participated in the engine test bench measurements described in *Paper A*. *Paper C* was jointly supervised by Hans Bodén and Mats Åbom. The basic WAVE and GT-POWER engine models used in the simulations in *Paper A*, *B* and *C* were designed by Johan Lennblad and refined by Magnus Knutsson. Magnus Knutsson has carried out all the simulations in the thesis, the experiments described in *Paper A*, *B*, *C* and *D* and has been responsible for writing all papers.

The work in this thesis has been carried out with support from

1. VIPP – Volvo Car Corporation Industrial Post Graduate Programme.

2. EMFO – The Swedish Emission Research Programme, Contract 310 10 1726.

3. KTH CICERO – Centre for Internal Combustion Engine Research Opus.

The content of this thesis has been presented at the following conferences

1. 12th International Congress on Sound and Vibration (ICSV12), Lisbon, Portugal, July 11 – 14, 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, July 2 – 6, 2006. “Experimental investigation of the acoustic effect of non-rigid walls in IC-engine intake systems”.

3. SAE Noise and Vibration Conference and Exhibition, St. Charles, Illinois, USA, May 15 – 17, 2007. “IC-engine acoustic source data from non-linear simulations”.

4. SAE Noise and Vibration Conference and Exhibition, St. Charles, Illinois, USA, May 15 – 17, 2007. “Acoustic analysis of charge air coolers”.

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

6. 5th International Styrian Noise Vibration and Harshness Congress, in cooperation with SAE International, Graz, Austria, June 4 – 6, 2008. “Prediction of IC-engine intake orifice noise using 3D acoustic modelling and linear source data based on non-linear CFD”.

7. Annual Conference for the Volvo Car Corporation Industrial Post Graduate Programme, 2004, 2005, 2006, 2007 and 2008. “Virtual acoustic design of intake noise for IC-engines”.

Table of contents

1. Introduction

1.1. Background

Air intake noise is created in an internal combustion (IC) engine when the intake valves open and reveal the pumping motion of the pistons. The resulting high amplitude pressure pulsations travel upstream through the different components in the air intake system. The sound waves are finally emitted either through the air intake as orifice noise or through the walls of the air intake system as shell noise due to fluid-structural interaction. Although the waves will have to pass more obstacles on a turbocharged engine is the basic principle identical as on a naturally aspirated. Schematic representations of the gas exchange system on a naturally aspirated and a turbocharged engine are shown in Fig. 1 and 2 respectively. The amplitude of the resulting intake noise sources can be high enough to make them significant contributors to the Pass-by Noise (Drive-by Noise) from the vehicle. In densely populated areas this is an important environmental issue and is regulated by stringent international standards that car manufacturers have to follow. Intake noise is also known to be an important and appreciated voice in the total sound quality impression of the vehicle.

Noise reducing measures available for air intake systems are either relatively space consuming or are reducing the efficiency of the engine breathing process, which is directly linked to the fuel consumption and hence the amount of CO_2 that is emitted from the vehicle. In order to meet customer demands for attractive design and reduced vehicle weight the amount of 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 further complicate 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 optimisation of air intake systems. This thesis includes six papers that are aiming to improve predictions of intake noise. *Papers A to C* treat two different strategies to combine nonlinear and linear modelling while *Papers D to E* are devoted to detailed modelling of components that include capillary tubes. Since important wave attenuation takes place within the tubes a full system optimization should include the acoustic properties of such devices. An example of such a component, whose acoustic properties has received scarce attention until recently, is the charge air cooler (intercooler). In *Paper D* a complete description of the acoustics of charge air coolers is for the first time given. This model is further improved in *Paper F* utilizing the improved model for turbulence damping that is suggested in *Paper E*.

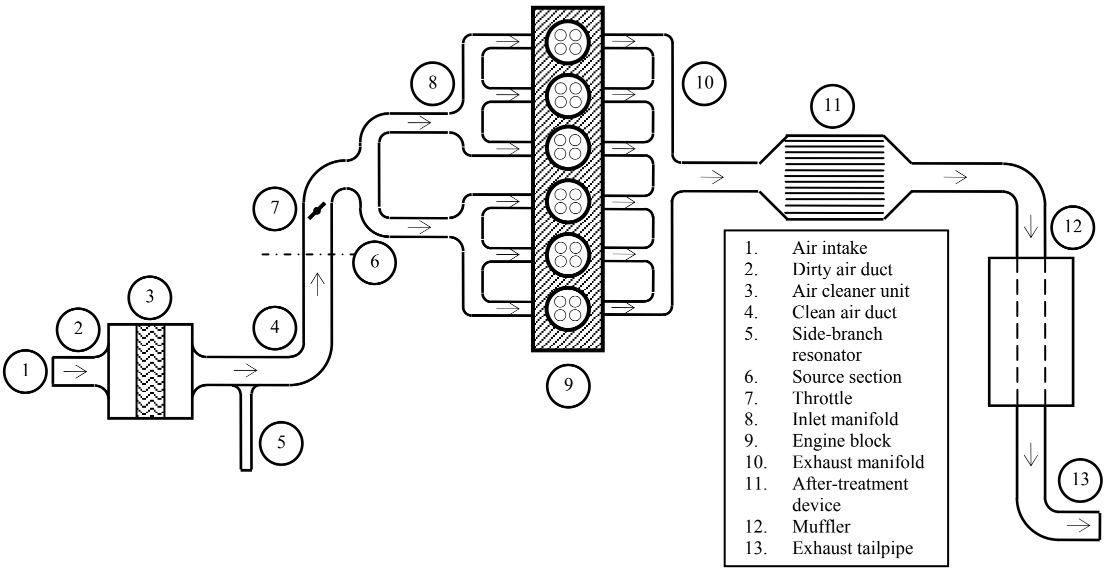

Fig. 1. Schematic representation of the gas exchange system of a naturally aspirated petrol engine.

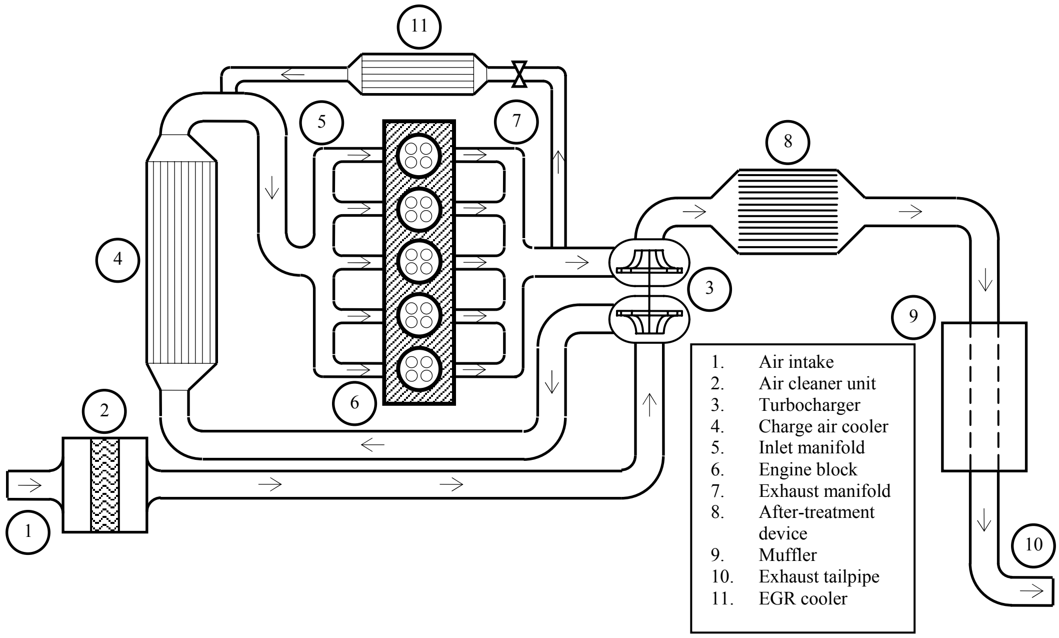

Fig. 2. Schematic representation of the gas exchange system of a turbocharged diesel engine with charge air cooler installation.

1.2. Acoustic modelling of intake systems

The gas dynamics of IC-engines can essentially be described by a set of coupled nonlinear equations for conservation of mass, momentum and energy [1]. In the general case analytical solutions to these equations can not be found and numerical models based on various approximations are necessary. A very powerful and often-used simplification for IC-engine intake or exhaust ducts is to consider one-dimensional (1D) fields only. This assumption 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 nonlinear equations will be greatly simplified. Another possible simplification is to assume 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 3D effects can also be addressed without too much difficulty. If only plane waves are considered, the wave equation will be reduced to a 1D linear wave problem, which can be efficiently analysed via so called two-port (four-pole) methods.

Within the automotive industry the most widely adopted virtual technique for gas dynamics studies is to solve the 1D coupled set of nonlinear equations using the finite volume or finite difference method. This technique is used within several commercial software packages 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 to predict and optimise cycle averaged parameters, such as the torque and power output from IC-engines but unsteady pressures and flow velocities are additionally provided at positions distributed throughout the intake and exhaust systems. As a result, they can also be used for acoustic studies [2], [3]. The boundary condition usually prescribed at any duct orifice is a fixed pressure corresponding to the ambient conditions. The predicted fluctuating velocities can subsequently, 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 1D plane wave propagation however, limits this technique to sufficiently low frequencies where non-plane wave effects are small. Therefore this type of model will fail first in large components such as air cleaners. Further limitations, that might not be as important for simulation of engine performance as for acoustics, include difficulties in applying frequency-dependent damping and boundary conditions as well as effects of non-rigid walls. As a result the accuracy of the resulting intake noise predictions is not completely reliable, nor where absolute sound pressure levels and resonance frequencies predictions are concerned.

Several authors have proposed strategies to improve predictions of sound based on nonlinear 1D gas dynamics simulations. Basically there are three main methods of doing this: methods that use information from nonlinear simulations as input to linear acoustic simulations, hybrid methods where information is inter-changed between simulations carried out in the time domain and the frequency domain and the extension from 1D to a full solution of the 3D nonlinear equations. A recent example of the first group is the work by Shaw *et al*. [4], where fluctuating velocities predicted from 1D gas dynamics 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 *et al*. [5] who studied the exhaust noise from a turbocharged truck engine by using nonlinear 1D gas dynamics simulations to extract a linear time-invariant source and thereafter coupled that to linear acoustic two-ports. The in-duct sound pressure level predictions shown were of reasonable agreement to measurements but the free field predictions were not as good. A similar exhaust noise study on a four-cylinder naturally aspirated diesel engine from a passenger car was reported by Hota and Munjal [6]. 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 *et al*. [7] and Chiavola [8]. Earlier work at MWL/KTH dealing with hybrid methods is described in Refs. [9] and [10]. An example of how nonlinear effects can be included in a one-port source model was suggested by Rämmal and Bodén [11]. The method of coupling 1D nonlinear simulations to 3D nonlinear CFD is provided as a built in function in some commercial software packages [3] but is still not very useful for acoustic optimization due to extremely long computational times. It should be noticed that most of the investigations mentioned here are based on one single interface between two different simulation strategies. To the authors knowledge no interest so far has been directed towards modelling where acoustic two-ports are included in 1D gas dynamics nonlinear simulations as black box models utilizing digital filtering techniques. An initial study of this possibility is presented in *Paper C* in this thesis.

In *Paper B* an investigation is presented where a linear source, which has been extracted from nonlinear 1D gas dynamics simulations, is coupled to linear acoustic transmission in a similar way as was done in Ref. [5] and [6]. The validation of the source data in *Paper A* and of the radiated intake noise at free field in *Paper B* are performed using experimental data for a six-cylinder naturally aspirated engine from a passenger car in an engine noise test cell at Volvo Car Corporation (VCC). Simulations and measurements are carried out 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 acoustic transmission models are taken from experimentally validated 3D acoustic finite element simulations in order to create an entirely virtual engine model.

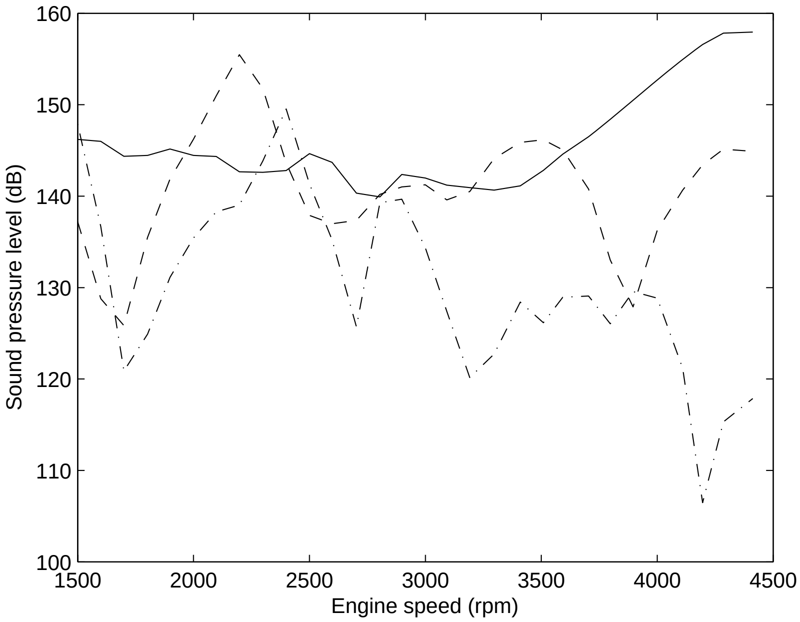

Fig. 3. Example of sound pressure level in the duct just upstream of the throttle on a six-cylinder naturally aspirated petrol engine.

—, 3rd engine order; − − −, 6th engine order; − · − ·, 9th engine order.

An important restriction of linear acoustic simulations is that their use is limited to those cases when the amplitude of the fluctuating pressure is small. Often small amplitude is taken as relative fluctuation amplitude of one per cent of the steady value. This implies that a linear acoustic model is accurate enough up to at least 150 dB (ref. $2 \cdot 10^{-5}$ Pa) for a steady value of 1 bar. The sound pressure level in the duct just upstream the throttle for the engine studied in *Papers A* and *B* is normally smaller than this value, which is why nonlinear effects can be assumed to be small upstream the throttle, see Fig. 3. The pressure amplitude close to the inlet valves is definitely above this limit, hence the choice of a nonlinear technique seems to be more appropriate to use for simulations of this area. In principle there are two effects of nonlinearity on wave propagation; wave steepening (shock forming) and local velocity pulsation amplitudes which are too high e.g. at perforated elements and narrow constrictions. The propagation distance x_s where an initial harmonic wave becomes a shock wave is approximately [12]

$$x_s / \lambda \approx 0.2 \cdot p_0 / \hat{p} \tag{1}$$

where λ is the wave length, p_0 the steady pressure and $\hat{p}$ the amplitude of the pressure pulsations. Considering the highest amplitude for the 3rd engine order at 4400 rpm in Fig. 3 and a steady value of 1 bar the shock forming distance is approximately 14 m. Fortunately this exceeds the length of the intake system on any passenger car. This implies that effects of nonlinear wave propagation are not very important and linear models can be used, at least upstream the inlet manifold. Local nonlinearities appearing at narrow constrictions or perforated elements, which can occur at lower pressure fluctuation levels [13], might be important for intake systems equipped with those types of elements.

1.3. The linear time-invariant one-port source model

A model that can be used to represent an engine as an acoustic 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 [14]. The condition of plane waves restricts its validity to frequencies below the cut-on frequency for the first non-plane mode. For a circular duct with radius *R* the cut-on frequency can be calculated as

$$f_{cut-on} = \frac{\alpha_{10} c_0}{2\pi R} \tag{2}$$

where $\alpha_{10} = 1.841$ is the first zero of the Bessel function J_1' and c_0 is the isentropic speed of sound in the gas mixture. For a duct with radius 40 mm, which is a typical duct dimension in intake systems 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 therefore justified. In the literature the linear time-invariant one-port source model is often expressed in terms of source strength and source impedance. Common choices of source strength variables are acoustic pressure $\hat{p}$ or acoustic volume velocity $\hat{q}$.

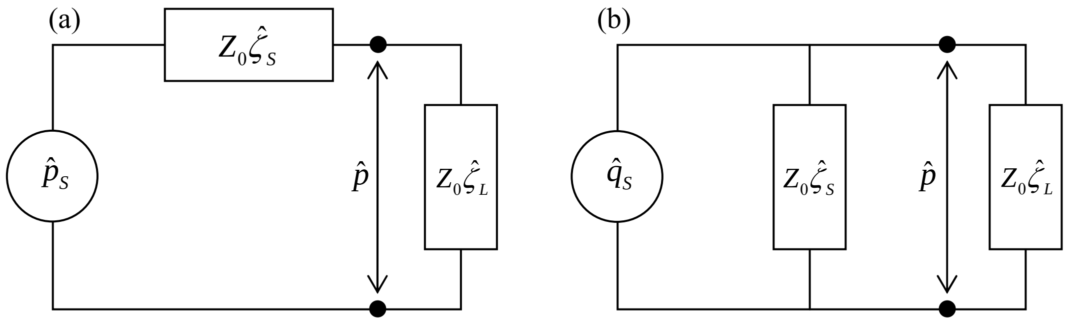

Fig. 4. Equivalent acoustic circuits for a linear and time-invariant one port source model. (a) Pressure source. (b) Volume velocity source.

The relationship between the source variables and acoustic pressure and volume velocity can be expressed, with reference to the electric analogy in Fig. 4, as

$$\hat{p}_S = \hat{p} + Z_0\hat{\zeta}_S\hat{q} \tag{3}$$

or

$$\hat{q}_S = \hat{p}\frac{1}{Z_0\hat{\zeta}_S} + \hat{p}\frac{1}{Z_0\hat{\zeta}_L} \tag{4}$$

in the case of pressure source $\hat{p}_S$ and volume velocity source $\hat{q}_S$ respectively. Here, $\hat{\zeta}_S$ is the normalised source impedance, $\hat{\zeta}_L$ the normalised impedance of the acoustic load, $Z_0 = \rho_0 c_0 / S$ the characteristic impedance for a propagating plane wave in a duct, with cross-sectional area *S*, filled with gas with density ρ_0. For a perfectly linear and time-invariant source the relationship between the source pressure and the source volume velocity is simply

$$\hat{p}_S = \hat{q}_S Z_0 \hat{\zeta}_S \,. \tag{5}$$

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 different techniques is given in Ref. [15]. The approach used in *Paper A* and *B* in the current work is the indirect method and the two unknowns of the source are determined via a multi-load procedure. The position of the source section, see Fig. 1, is where the linear source is located and can for example be located just upstream of the throttle. In the case of a pressure source the resulting system of equations in matrix form, based on Eq. (3), becomes

$$\begin{bmatrix} \hat{\zeta}_1 & -\hat{p}_1 \\ \hat{\zeta}_2 & -\hat{p}_2 \end{bmatrix} \begin{bmatrix} \hat{p}_S \\ \hat{\zeta}_S \end{bmatrix} = \begin{bmatrix} \hat{p}_1 \hat{\zeta}_1 \\ \hat{p}_2 \hat{\zeta}_2 \end{bmatrix}. \tag{6}$$

The corresponding system of equations for a volume velocity source using Eq. (4) is

$$\begin{bmatrix} 1 & -\hat{p}_1 / Z_0 \\ 1 & -\hat{p}_2 / Z_0 \end{bmatrix} \begin{bmatrix} \hat{q}_S \\ 1/\hat{\zeta}_S \end{bmatrix} = \begin{bmatrix} \hat{p}_1 / \left(\hat{\zeta}_1 Z_0 \right) \\ \hat{p}_2 / \left(\hat{\zeta}_2 Z_0 \right) \end{bmatrix}. \tag{7}$$

In order to reduce the effect of measurement errors and deviations from source linearity more than two known acoustic loads can be used which results in an over-determined system of equations. Six different acoustic loads, created by a quarter-wave resonator with variable length, are used in the investigations presented in *Paper A* and *B*.

1.4. Acoustic two-ports

Basically, a two-port describes a linear system with an input and an output port. The relation between two state variables at each port is described by a [2 x 2]-matrix. Depending on how the state variables are chosen the two-port becomes suitable for different problems. In acoustics a common choice is to use the acoustic pressure and the acoustic volume velocity to describe both the input and output state which leads to the transfer-matrix relation

$$\begin{bmatrix} \hat{p}_1 \\ \hat{q}_1 \end{bmatrix} = \begin{bmatrix} \hat{\mathbf{T}} \end{bmatrix} \begin{bmatrix} \hat{p}_2 \\ \hat{q}_2 \end{bmatrix}. \tag{8}$$

Here, the indices 1 and 2 refer to the name of the port with reference to Fig. 5.

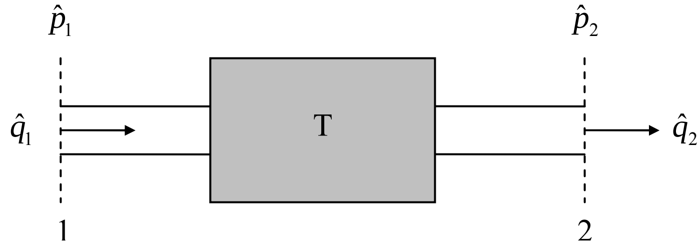

Fig. 5. Acoustic two-port

One advantage of using the transfer-matrix formalism is that a cascade-coupled chain of n elements easily can be combined to yield a representation of a complete system. Mathematically this corresponds to a series of matrix multiplications as

$$\begin{bmatrix} \hat{\mathbf{T}} \end{bmatrix} = \begin{bmatrix} \hat{\mathbf{T}}_1 \end{bmatrix} \cdot \begin{bmatrix} \hat{\mathbf{T}}_2 \end{bmatrix} \cdot \ldots \cdot \begin{bmatrix} \hat{\mathbf{T}}_n \end{bmatrix}. \tag{9}$$

Here, it has been assumed that there is continuity of $\hat{p}$ and $\hat{q}$ at the element interfaces. This is strictly valid only for interconnections in a straight duct where only entirely plane waves exist. If the element interface has to be located at a sudden expansion a special coupling two-port matrix has to be introduced to handle the discontinuity [16], [17]. When the transfer-matrix of a system is known the transmission loss can easily be calculated using the equation [16]

$$TL = 10 \cdot \log_{10} \left[\left(\frac{1+M_1}{1+M_2} \right)^2 \frac{Z_2}{4Z_1} \left| \hat{T}_{11} + \frac{\hat{T}_{12}}{Z_2} + Z_{\mathrm{IN}} \hat{T}_{21} + \frac{Z_1 \hat{T}_{22}}{Z_2} \right|^2 \right] \tag{10}$$

where Z_1 and Z_2 denote the characteristic impedances for propagating waves at the in- and outlet respectively, M_1 and M_2 the corresponding Mach-numbers and $\hat{T}_{11}$, $\hat{T}_{12}$, $\hat{T}_{21}$ and $\hat{T}_{22}$ are the components of the transfer-matrix.

Procedures to obtain two-ports experimentally are well documented, see e.g. Refs. [18] – [21] for descriptions of the two-load method and the source switching technique, and accurate data can be obtained in the entire plane wave range at least for Mach-numbers up to 0.2-0.3 [21]. The upper frequency limit is defined by the cut-on frequency for the first cross-mode of the input or output cross-section or by the measurement set-up itself. The standard technique currently for measuring acoustic plane wave properties in ducts, such as reflection coefficient and impedance is the two-microphone method (TMM) developed by Seybert and Ross [22] and Chung and Blaser [23] – [24]. Here, two microphones are used for plane wave decomposition and the theoretical upper and lower frequency limit is defined by the distance between the microphones [19].

In some situations it can be of interest to include a component in nonlinear time-domain gas dynamics simulations as a black box model. This could be the case if the component is difficult to model and tabulated frequency-domain data in the form of a two-port has been obtained experimentally or numerically. A new method for such implementation, based on inversely transformed two-ports is presented in *Paper C* in this thesis. The obtained impulse response functions on impedance-matrix form are suggested to be used as FIR filters within the time-domain calculations. The authors have not been able to find any published data concerning time-domain implementation of two-port data for acoustic applications based on digital filtering techniques. The work by Payri *et al.* [7] and Chiavola [8] treats coupling between nonlinear 1D simulations and acoustic two-ports in the frequency domain, however, the implementations are based on iteration between time- and frequency domain solutions. There is a lot of work presented that treats acoustic boundary conditions (one-port models) in time-domain simulations where filtering is used, see Ref. [25] and [26] for some recent examples. The majority of the research in this field has been inspired by the evolution of acoustic liners for aircraft engines.

1.5. Sound in narrow tubes

The problem of wave propagation in gas filled ducts has interested scientists for a long time. As was stated in the previous sections, the acoustic state variables can be considered as being constant throughout the cross-section of the duct if the frequency is below the cut-on frequency for the first cross-mode. For real gases where the viscosity can not be neglected this approximation is inappropriate in some situations. The particle velocity just at the duct walls is zero and the transition to the velocity at the centre of the ducts is controlled by the viscosity, the frequency and the radius of the duct. As a result the wave propagation becomes dissipative. The relation between these parameters forms a shear wave number known as the Stokes' number

$$s = R\sqrt{\omega / \nu} \tag{11}$$

where ω is the angular frequency and ν the kinematic viscosity. For large values of the Stokes' number the dissipative effects are very small, but for small values they can become significant. This occurs, for wave propagation in a gas, either at very low frequencies or in narrow ducts. Similarly, energy losses will be present for fluids with nonzero thermal conductivity. The main reason for this is that the temperature in a sound wave also fluctuates around a mean value which can not be followed by the duct wall. The resulting losses can, depending on the material properties of the duct wall, be comparable to those from viscosity [27]. Concerning gas exchange systems in modern IC-engines for passenger cars at least four examples can be given where this attenuation due to boundary effects might be of importance. In the exhaust system are the catalytic converter and the particulate filter, and in the intake of supercharged engines the charge air cooler (CAC), based on arrays of narrow tubes. In the exhaust gas recirculation (EGR) circuit the EGR-cooler might also consist of narrow tubes. Models for thermoviscous sound propagation in narrow ducts are hence of importance for the acoustic optimization of the gas exchange system of IC-engines.

One early model for sound propagation in real gases confined in cylindrical ducts, which includes the effect of viscosity as well as heat conduction, was proposed by Kirchhoff [28] already in 1868. He formulated the solution to the problem, without any mean flow present, as a complicated, complex transcendental equation which has so far not been solved analytically. Kirchhoff himself was the first to present an approximate solution to this equation, using the restriction of "wide" ducts, which is the same as large shear wave numbers. In the work by Zwikker and Kosten [29] more than fifty years later an approximate solution, valid also for narrow ducts, 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. [30]. When flow is present the situation is somewhat more complicated and no complete theory exists. Several authors [31] – [36] have, however, derived solutions based on simplified equations or numerical calculations. For practical applications the most useful are perhaps those proposed by Dokumaci [31] – [32]. In Ref. [31] he showed that the equations for sound propagation in a thermo-viscous fluid, simplified in the manner of Zwikker and Kosten theory [29], could be solved analytically for a circular pipe with a mean flow profile that is constant over the cross-section. In a later paper [32] Dokumaci extended the model from [31] to rectangular cross-sections by expanding the solution in terms of a double Fourier sine series. Other important contributions, starting out from essentially the same equations as was used by Dokumaci but with mean flow that is based on a parabolic mean flow profile, include Astley and Cummings [33], Peat [34], Ih *et al*. [35], and Jeong and Ih [36]. The model in Ref. [33] is based on a finite element discretisation of the tube cross-section and can thus be used for an arbitrary cross-section with an arbitrary shape of the mean flow profile. At operating conditions catalytic converters and particulate filters, but also CACs and EGR-coolers, experience temperature and pressure gradients. The effect of axial pressure or temperature gradients has e.g. been treated by Peat [37], Peat and Kirby [38], and Dokumaci [39].

All the models mentioned above assume laminar flow and therefore do not take into account any effect of turbulence on the propagation of sound waves. When a turbulent mean flow is present in a duct the unsteady transport of momentum, which creates the turbulent shear stress, can be modulated by acoustic waves. This modulation will cause an extra mixing, that is mainly important where the gradients of the acoustic particle velocity are large, which will result in a conversion of acoustic energy into turbulent energy and increased acoustic losses. The effect of attenuation of pulsations through interaction with turbulence has mainly been investigated by the fluid mechanics community. Large amplitude velocity-oscillations in ducted flow was investigated experimentally e.g. by Tardu *et al*. [40]. Examples of recent efforts that use Large-Eddy Simulations (LES) as a tool to study different aspects of the phenomena including acoustic waves are the work by Scotti and Piomelli [41] and by Comte *et al*. [42]. An example of an early attempt to model the effect of turbulence damping of acoustic waves in pipe flow is the work by Ingard and Singhal [43] where a crude model of the dissipation was introduced as a modified wave number. Other attempts to model acoustic wave propagation in turbulent pipe flows were for instance reported by

Ronneberger and Ahrens [44], Mankbadi and Liu [45] and Peters *et al*. [46]. Detailed experimental data was also provided in Refs. [44], [46] and by Allam and Åbom [47]. From these studies it has been concluded that damping is achieved if the thickness of the acoustic boundary layer is larger than the viscous sublayer. The model suggested by Howe [48] includes a frequency-dependent model of the effective turbulent boundary layer viscosity in the form of an acoustic impedance boundary. From solving the resulting inhomogeneous wave equation a complex valued wave number that accounts for the resulting dissipation can be extracted. However, the model requires the cross-section to be wide and the turbulent flow to be fully developed with a constant flow profile. A numerical model that is useful for narrow tubes where the effect of turbulence on the propagating waves is significant is suggested in *Paper E* in this thesis. The model, which is based on a combination of the models by Astley and Cummings [33] and the model by Howe [48], takes into account that the thickness of the acoustic boundary layers for waves propagating upstream and downstream are different.

1.6. Acoustic modelling of special intake system components

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 intake system components that are supposed to have large resistive or reactive properties affecting the transmitted sound are the air cleaner and the charge air cooler. The component with the largest potential for reducing noise is the air cleaner unit, acting as an expansion chamber. Also the charge air cooler has been shown to have previously underestimated silencing capabilities. They therefore deserve to be described carefully and will hence be discussed in the following text. Moreover, the turbocharger is of interest as it will attenuate the in-duct noise but also might act as an active sound source due to the high speed revolution of the compressor wheel. Modelling of the acoustic properties of turbochargers is out of the scope for this thesis. A recent publication that treats experimental as well as theoretical efforts on the subject is the work by Rämmal [49].

1.6.1. Air cleaner units

Two-ports for simple regular elements such as straight ducts, expansion chambers and resonators can often be described analytically see e.g. Ref. [16] and [50]. The amount of available space in the engine compartment of an ordinary passenger car is often severely restricted. This enforces complicated and irregular shapes for some of the components in the intake system, especially the air cleaner unit, which has the largest cross-dimensions. Predictions in the low frequency region are possible by modelling the air cleaner as an expansion chamber where the area ratio and the length must be given correctly, i.e., to preserve the volume. Contribution from higher-order modes to the response at higher frequencies can be included by using more advanced techniques e.g. the null-field approach [51] or the point-matching method [52]. The use of the Finite Element Method (FEM) and the Boundary Element Method (BEM) is widespread within the automotive industry and several software packages can interact with CAD systems without too much effort. This implies that complicated geometries can be modelled without too much difficulty. Several authors have shown that accurate predictions of sound transmission can be achieved; see e.g. the work by Herrin *et al.* [53] 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 [54], Alfredson and Davies [55], Glav [50] and Allam and Åbom [56]. The effect on sound transmission in air intake components due to fluid structural interaction has recently been discussed by Marion and Ye [57] and Alex [58] who stated that the influence of non-rigid walls on the intake orifice noise can be significant. Both works were however, based on generic components that were not optimised for 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 which also reduces the losses for the in-duct sound. Losses due to dissipation in the filter paper have been shown to be of importance in the experimental investigations in *Paper B* and *C*. A suggestion for a filter paper model is given by Ih and Kang in Ref. [59] where a 1D model, originally developed by Allam and Åbom [56] for diesel particulate filters, is applied to extract a two-port. At frequencies where cross-modes appear in the air cleaner and were the filter paper was shown to yield large dissipation, this model will not be sufficient.

1.6.2. Charge air coolers

Many turbocharged IC-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. It is also important for petrol engines 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 [60], [61]. The models in both these references make use of acoustical two-ports to assemble a complete model for a CAC. However, neither of them includes a complete treatment of the losses in the cooling tubes. According to the literature survey in Ref. [60], there are predictive models available describing the thermal efficiency [62], and also models treating flow unsteadiness [63] – [65] in CACs. Still they are only evaluated in terms of heat transfer performance, pressure drop and low frequency gas dynamics. In Ref. [66] and [67] Knutsson and Åbom have presented some initial parts of the work presented in *Paper D* in this thesis, which presents a complete description of the sound attenuating properties of CACs when there is a mean flow present.

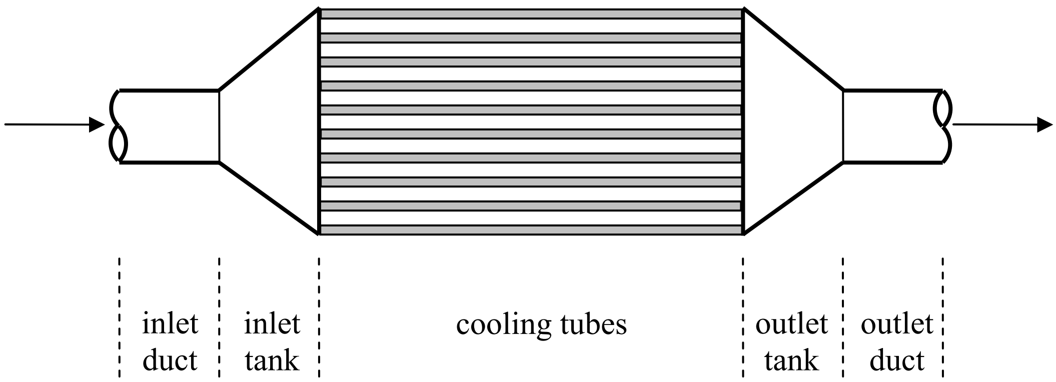

Fig. 6. Schematic representation of a generic air-to-air charge air cooler.

Most CACs consist of two of the most widely used sound attenuation measures: the reactive expansion chamber, denoted in Fig. 6 by inlet/outlet tank, and the dissipative narrow cooling tubes. The assembled component thereby offers possibly underestimated capabilities for broadband noise silencing that could be used for noise optimisation. Of key importance 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 an EGR-short-route-system, will be reduced by the CAC. It thereby provides the possibility of reducing the size of the air cleaner unit as less volume will be required for low frequency damping. A smaller air cleaner unit is easier to position in the engine compartment and as a result it will be easier to design a lay-out of the ducts with low pressure drop which is essential for breathing efficiency. The CAC is installed downstream of the compressor (see Fig. 2) and 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 of the CAC.

Two particular challenges concerning acoustic modelling of CACs have been identified: to correctly address the propagation of sound in narrow cooling tubes with non-circular geometries for different flow speeds and the coupling of several ducts into one volume, where higher order modes are present. In *Paper D* the solution for circular ducts by Dokumaci [31], a modified version of the numerical solution scheme for arbitrary cross-sections derived by Astley and Cummings [33] and the solution accounting for turbulence in circular ducts by Howe [48] are used to model the cooling tubes in order to find the most accurate solution. Two-ports representing the cooling tubes are extracted from the three solutions, which to the authors' knowledge was done for the first time for the latter two. Two-ports have recently also been extracted from Howe's model by Dokumaci [68]. The effect of approximating a cross-section that is shaped as an isosceles trapezium with a circular geometry, where the hydraulic diameter is equivalent, is studied for cases where a laminar or constant flow profile is superimposed. A multi-port approach, based on the admittance relations between the ports, is presented for the coupling of the cooling tubes into the tanks. The multi-ports are established using acoustic finite elements.

The effect of dissipation due to turbulence is shown to be of significant importance for low frequencies. As the model by Howe is valid for wide ducts it underestimates this favourable damping. The numerical model that is suggested in *Paper E*, which is based on the finite element scheme in [33] and the effective turbulent boundary layer viscosity in [48], is aiming to improve predictions for narrow ducts. In the last investigation in the thesis, presented in *Paper F*, is this model used together with the multi-ports suggested in *Paper D* to improve the predictions for the CAC and account for the turbulence damping.

2. Summary of the papers

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

The simplest source model for linear simulation of engine intake noise is the linear time-invariant one-port model. The one-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 an engine with correct settings in the engine control system, which prevents them from being used for early predictions.

The possibility to extract linear acoustic one-port source data for intake systems from nonlinear 1D gas dynamics simulations is tested and validated in *Paper A*. Simulations and measurements are performed for a large number of engine revolution speeds in order to make the first systematic validation of intake acoustic source data prediction using nonlinear gas dynamics simulation for a wide engine speed range. It is concluded that the simulated source data for a six-cylinder naturally aspirated petrol engine are reasonably accurate, as can be seen in Fig. 7. This therefore seems to be a promising technique which could replace expensive, and time consuming experimental source data determination. Noise is most efficiently reduced at the source. This ability to predict source data therefore opens up possibilities for sound optimization of engine parameters affecting source data such as intake valve profiles and timing.

It is also shown from additional simulations 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. 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 three different simulated 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 loads for source data prediction.

Finally the linearity of the source data is studied by means of a linearity coefficient and by comparisons between source impedances simulated using pressure source or velocity source formulations. The 3rd and 6th engine orders show almost no deviation from linearity and good correspondence between impedances while the weaker 5th and 9th order are occasionally more nonlinear 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.

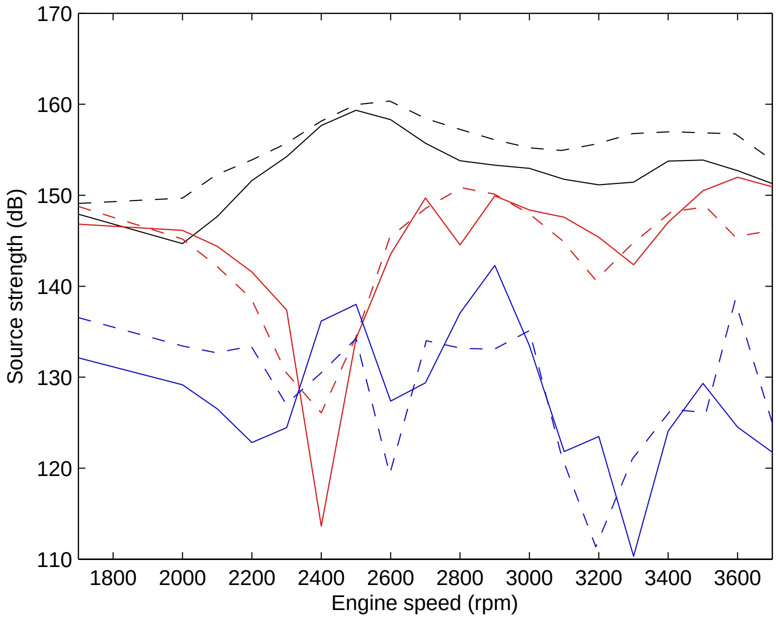

Fig. 7. Source strength obtained from measurements (solid line) and gas dynamics simulations (dashed line). —, 3rd engine order; —, 6th engine order; —, 9th engine order.

2.2. Paper B: Prediction of IC-engine intake orifice noise using 3D acoustic modelling and linear source data based on non-linear CFD.

This paper describes an entirely virtual methodology for predicting intake orifice noise from 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 is predicted using 1D nonlinear gas dynamics simulations while the transmission is calculated using linear acoustic finite elements in order to include 3D effects in the air cleaner unit. A loudspeaker test rig that can include effects of flow is used to validate the simulated transmission loss for the air intake system but also to investigate the effects of flow in this particular case. The accuracy of the predictions is good except for a small shift in the predicted resonance frequencies for the present resonators which might be explained by the approximation of rigid walls or from neglecting viscothermal effects in the FEM calculation. The effect of a low Mach number flow (M = 0.1) is not very large on the experimental data. However, the flow reduces the damping at the resonance frequencies of the Helmholtz resonators. 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. Important is that simulations and measurements are carried out for a large number of engine revolution speeds in order to make the first systematic validation of an entirely virtual intake noise model that includes 3D effects for a wide engine speed range.

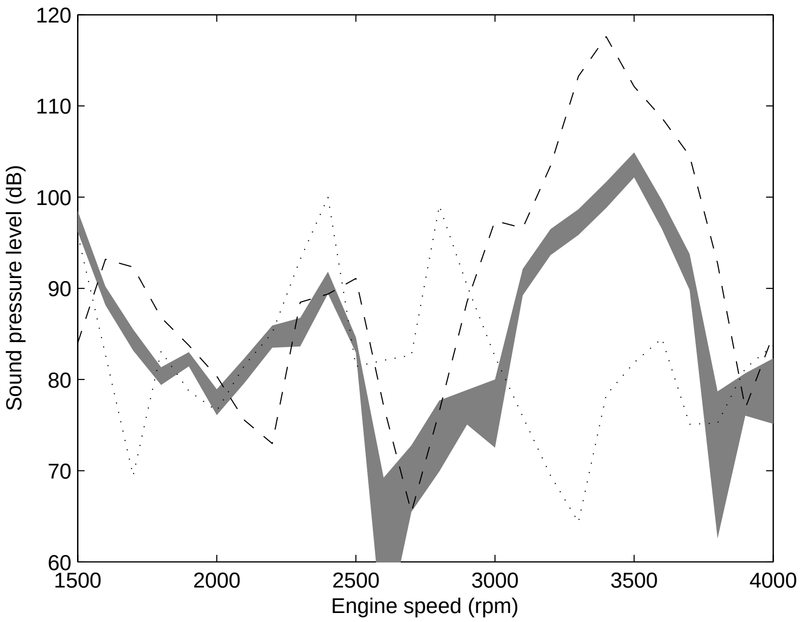

Fig. 8. Sound pressure level for 9th engine order 100 mm from the intake orifice. ▬▬, Experiments in engine test bench; · · ·, 1D nonlinear simulation of complete intake system and monopole radiation; − − −, Linear source data from 1D nonlinear simulation + two-port from FEM and BEM radiation.

It is concluded that the predictions from the proposed linear methodology are mostly as good as, or in the case of higher frequencies even better, than the predictions from using nonlinear 1D gas dynamics simulations for the complete system. This is illustrated in Fig. 8 where sound pressure data for the 9th engine order is shown. It is also concluded that a simple monopole radiation model yields, for the frequencies in the validation (between 75 and 600 Hz), as accurate predictions as the much more complicated BE-model for the radiation.

2.6. Paper C: A study on acoustical time-domain two-ports based on digital filters with application to automotive air intake systems

An initial study on a new technique for the use of two-ports in the time domain for automotive gas dynamics applications is presented in *Paper C*. Experimentally obtained tabulated frequency-domain two-port data on the impedance-matrix form, representing a simplified air cleaner unit, are presented and inversely transformed to the time domain. The resulting impulse response functions are implemented as digital FIR filters in nonlinear gas dynamics simulations. A simple duct-volume-duct system is studied where the acoustic source is a stepped sine fluctuating pressure. The results are validated to corresponding linear frequency-domain results obtained from two-port simulations. Level differences as well as in-duct fluctuating pressures are compared. It can be concluded that the level difference results are very good in the frequency band between 100 and 500 Hz, see Fig. 9, and the predictions of in-duct sound pressure levels are reasonably good showing differences less than 5 dB between the two methods. The results below 100 Hz are less accurate and further studies will have to be carried out before the suggested method can be used for gas dynamics simulations of a complete engine. It also needs to be investigated if the suggested technique can be used in transient simulations. The mean flow or 0 Hz component is also a problem since this part is nonlinear, i.e. proportional to q^2. One solution could be to treat this part separately as an element with resistance.

The experimental two-ports included in the investigation reveal additional interesting information concerning the filter paper. It is concluded that the filter paper under consideration, taken from a passenger car, yields more than 5 dB extra transmission loss at the lowest resonance of the expansion chamber. This can also be observed in the curves for level difference in Fig. 9 at approximately 350 Hz. From a design point of view this is interesting since sound amplification due to resonances is undesirable.

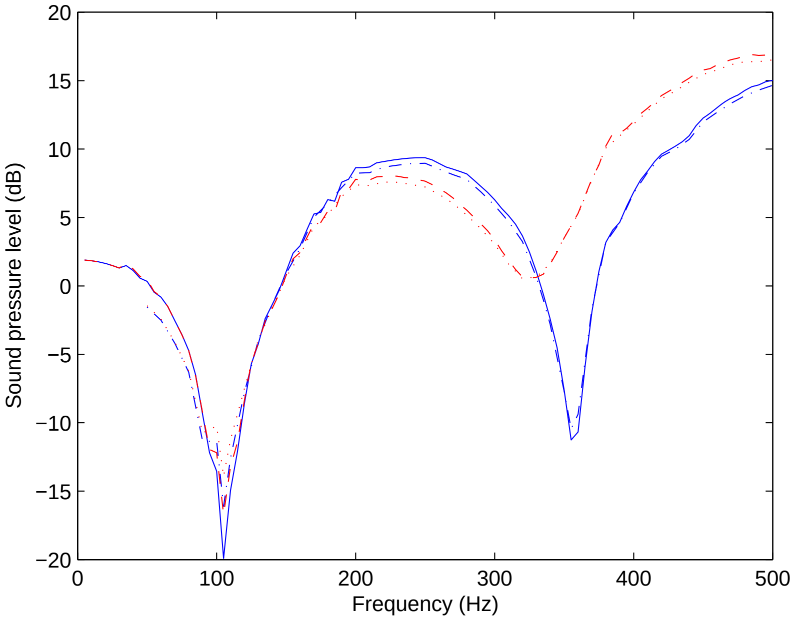

Fig. 9. Level difference for simple air cleaner unit coupled to two straight ducts. —, Without filter paper, frequency response calculation; – · – ·, Without filter paper, time-domain calculation with digital filter; – – –, With filter paper, frequency response calculation; · · ·, With filter paper, time-domain calculation with digital filter.

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

The transmission of in-duct sound through charge air coolers is investigated in *Paper D*. The losses, due to viscous and thermal boundary layers as well as turbulence, in the narrow cooling tubes result 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, which is 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 - see Fig. 6 and 10, 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 [33] 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 [31] and Howe [48], 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. Interesting low frequency damping is observed in the presented experimental data; see Fig. 11. This attenuation increases with increasing mean flow velocity and is stated to depend on interaction between turbulence and the acoustic field.

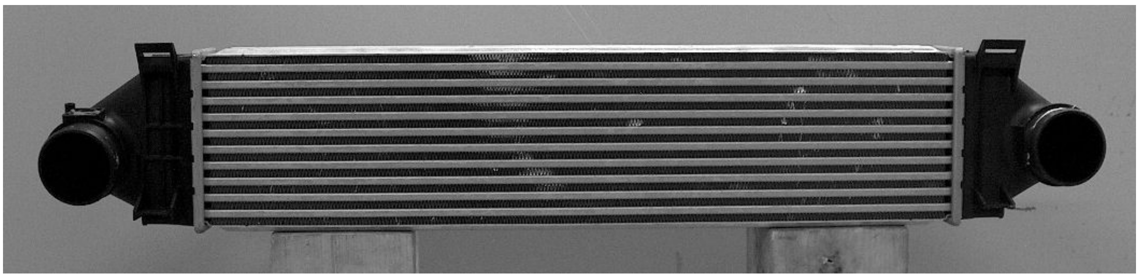

Fig. 10. Charge air cooler used for validation.

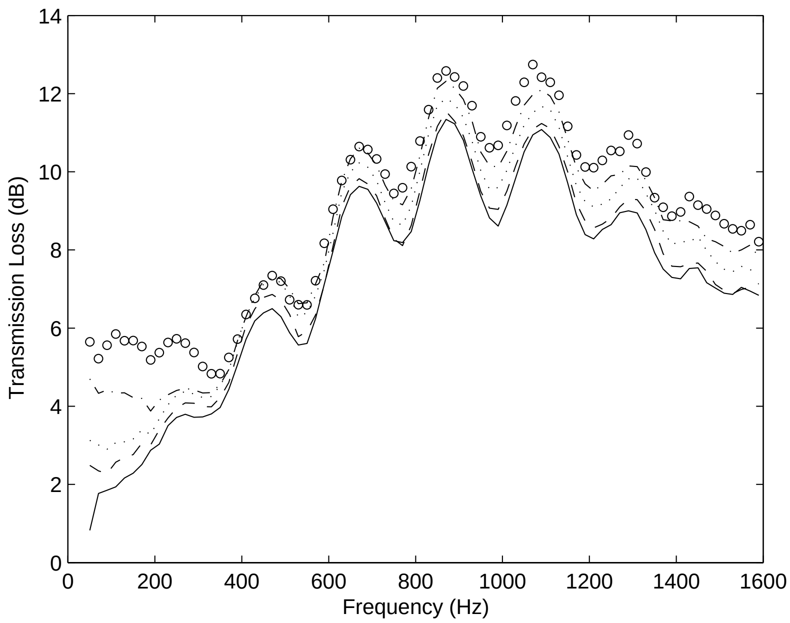

Fig. 11. Transmission loss in the upstream direction for charge air cooler (experimental data). —, $M = 0.0$; − − −, $M = 0.025$; · · ·, $M = 0.05$; − · − ·, $M = 0.075$; ∘ ∘ ∘, $M = 0.1$.

2.4. Paper E: The effect of turbulence damping on acoustic wave propagation in tubes

A numerical model that predicts the damping due to the interaction between turbulence and sound waves propagating in a cylindrical duct is proposed and validated in this paper. It consists of an extension of the finite element scheme proposed by Astley and Cummings [33] in combination with the effective turbulent boundary layer eddy viscosity model proposed by Howe [48]. The formulation, that is expressed in cylindrical coordinates and allows convection of an arbitrary flow profile, is suggested for use also with thick acoustic boundary layers. The distribution of the added turbulence eddy viscosity is controlled by the relation between the thickness of the acoustic boundary layer and the viscous sublayer. It is argued that the thicknesses of the acoustic boundary layers are different for waves propagating upstream or downstream the incompressible mean flow. New empirical formulas controlling the distribution of the added turbulence eddy viscosity are suggested. Three cases from the literature, with different duct diameters and frequencies, are used for validation [44], [47].

It can be concluded that the accuracy is of the same order as the model proposed by Howe [48] for intermediate Stokes numbers (~ 100) and Mach-numbers less than 0.1. However, the proposed model shows better correspondence with the pure thermoviscous predictions obtained from the model proposed by Dokumaci [31] for low Mach numbers. The model should also be more accurate at small Stokes numbers but more detailed experiments are needed to validate this. Fig. 12 shows an example of the validation where the predicted damping is normalised with the damping for a viscothermal fluid in a wide duct given by Kirchhoff, α_0. The damping is here shown as a function of the normalised acoustic boundary layer thickness δ_A^+. An empirical model for engineering purposes has been proposed for the unexplained additional damping observed in all three validation cases for low Mach numbers ($\delta_A^+ < 13$ in Fig. 12) in the upstream direction.

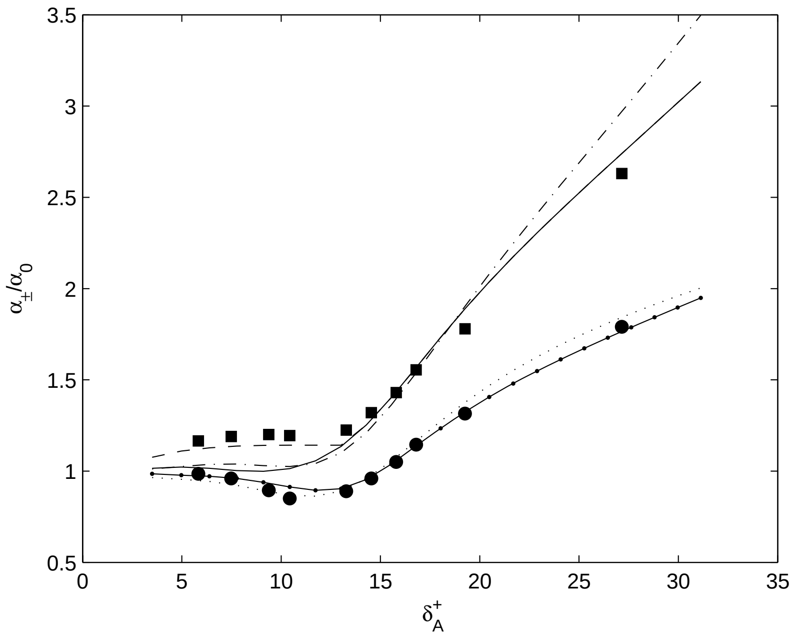

Fig. 12. Relation between damping and normalised acoustic boundary layer for $k_0R = 0.024$. ■ , Experimental, upstream [46]; ● , Experimental, downstream [46]; − · − ·, Howe, upstream [48]; · · ·, Howe, downstream [48]; —, Present model, upstream; •-•-•, Present model, downstream; − − −, Present model, low Mach number correction, upstream.

2.5. Paper F: A note on acoustic wave propagation in charge air coolers

The frequency-dependent numerical model for interaction between turbulence and acoustic waves proposed in *Paper E* is used to improve the model for charge air coolers suggested in *Paper D*. The model is validated using experimental results for the charge air cooler from a passenger car that was presented in *Paper D*. These new results are compared to simulated results where the narrow tubes are described using the models by Howe [48] and Dokumaci [31]. The predictions of transmission loss using the model proposed in *Paper E* show improvements almost over the entire frequency range under consideration; see Fig. 13. For waves propagating upstream the incompressible mean flow it is indicated that the damping at high frequencies is a result of thermoviscous boundary layers *but also* from interaction between sound and turbulence within the acoustic boundary layers. This effect has, to the author's knowledge, not yet been satisfactory explained, but the *ad hoc* model proposed in *Paper E* is shown to yield good estimates of the resulting damping.

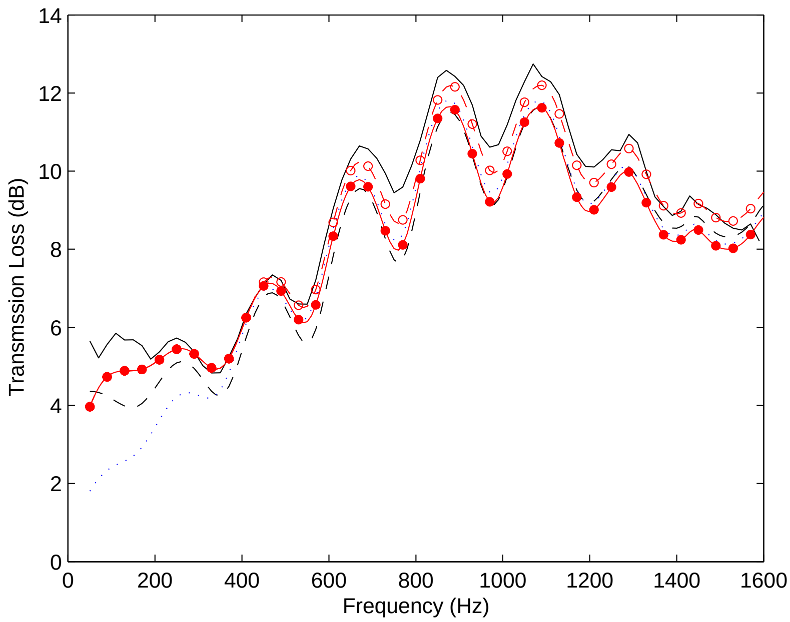

Fig. 13. Transmission loss in the upstream direction for complete CAC at $M = 0.1$ in main duct ($M = 0.08$ in cooling tubes). —, Measured; · · ·, Predicted using multi-port technique and Dokumaci's tube model [31]; – – –, Predicted using multi-port technique and Howe's tube model [48]; ●-●-●, Predicted using multi-port technique and tube model by Knutsson & Åbom; ○-○-○, Predicted using multi-port technique and tube model by Knutsson & Åbom, *ad hoc* model.

3. Recommendations for future research

3.1. Papers A to C

Future research of interest regarding source data extraction includes more research into how to choose acoustic loads for the multi-load method. Using 1D gas dynamics simulations presents us with the opportunity of optimising this process in order to further increase the accuracy of the estimated source data. Furthermore, it is of large interest to verify the linear time-invariant source model on a four-cylinder engine, where the reflections from the expansion at the air cleaner unit will affect the performance of the engine to a larger extent than on a six-cylinder engine. This interest is amplified by the recent focus on downsizing to smaller and more efficient engines in order to reduce CO_2-emissions.

The task of developing an orthotropic model for air filter papers is the most important in order to improve simulation results for sound propagation through air cleaner units. Of interest here is the work by Ih and Kang [59] who used a 1D model to simulate wave propagation through a pleated filter paper. In order to extend the accuracy to higher frequencies 3D 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. One possibility would be to investigate if a Biot type of model can be used.

The proposed concept of representing two-ports in the time domain with help of FIR filters is very promising. However, the problems to achieve good model behaviour for low frequencies (< 100 Hz) will have to be studied further before the suggested method can be used for gas dynamics simulations of a complete engine. It also needs to be investigated how to include a steady flow part and if the suggested technique can be used in transient simulations.

3.2. Papers D to F

The presented model for charge air coolers needs to be validated at operating conditions where the in-duct gases experience a temperature gradient along the cooling tubes.

The development of the numerical scheme proposed in *Paper E* involves some simplifications that can be addressed in order to further improve the accuracy of the predictions. Important are most likely the influence of the neglected radial velocity and the assumption of a constant pressure over the cross-section, which has been shown to be noticeable but small in the purely thermoviscous case [36]. It is not clear in how large extent this will affect this model where the turbulence is included. The turbulence eddy viscosity model that has been used is based on the work by Prandtl and is indeed very simplified. The spatial distribution enforces a discontinuity in the gradient of the total viscosity which is unphysical. The numerical scheme proposed here gives the possibility to use a more advanced turbulence model that better takes into account the properties of the transition between the turbulent region and the viscous sublayer, and also is valid for lower Reynolds numbers. Finally, the model needs to be validated using experimental data for more values on the Stokes number. Particularly at low Stokes numbers, where the acoustic boundary layers are thick, more experimental data is required.

References

[1] D.E. Winterbone, R.J. Pearson, *Theory of engine manifold design, Wave action methods for IC engines*, Professional engineering publishing, London, 2000.

[2] *WAVE, Acoustic Manual*, Ricardo, 2001.

[3] *GT-POWER, User's Manual version 6.2*, Gamma Technologies, 2006.

[4] C.E. Shaw, D.J. Moenssen, W.C. Montgomery, On the use of BEA with engine simulation as an input to predict air induction inlet noise, *SAE Technical Paper 2005–01–2350*, 2005.

[5] R. Fairbrother, H. Bodén, R, Glav, Linear acoustic exhaust system simulation using source data from non linear simulation, *SAE Technical Paper 2005–01–2358*, 2005.

[6] R.N. Hota, M.L. Munjal, Hybrid approach to predict exhaust noise of naturally aspirated engine with commercial mufflers, *SAE Technical Paper 2007–26–039*, 2007.

[7] F. Payri, J.M. Desantes, A.J. Torregrosa, Acoustic boundary conditions for unsteady one-dimensional flow calculations, *Journal of Sound and Vibration* 188 (1) (1995) 85–110.

[8] O. Chiavola, Multi-dimensional CFD-transmission matrix modelling of IC engine intake and exhaust systems, *Journal of Sound and Vibration* 256 (2) (2002) 835–848.

[9] F. Albertson, J. Gilbert, Harmonic balance method used for calculating the steady state oscillations of a simple one-cylinder cold engine, *Journal of Sound and Vibration* 214 (4) (2001) 541–565.

[10] F. Albertson, H. Bodén, J. Gilbert, Comparison of different methods to couple nonlinear source descriptions in the time domain to linear system descriptions in the frequency domain–Application to a simple valveless one-cylinder cold engine, *Journal of Sound and Vibration* 291 (2006) 963–985.

[11] H. Rämmal, H. Bodén, Modified multi-load method for nonlinear source characterisation, *Journal of Sound and Vibration* 299 (2007) 1094–1113.

[12] A.D. Pierce, *Acoustics – An introduction to its physical principles,* The Acoustical Society of America, New York, 1989.

[13] T. Elnady, *Modelling and Characterization of Perforates in Lined Ducts and Mufflers*, Ph.D. Thesis, The Royal Institute of Technology, Stockholm, Sweden, 2004.

[14] H. Bodén, M. Åbom, Modelling of fluid machines as sources of sound in duct and pipe system, *Acta Acustica* 3 (1995) 549–560.

[15] H. Bodén, Characterisation of fluid-borne sound sources and structure borne sound sources, *Proceedings of the 9th International Congress on Sound and Vibration*, Orlando, Florida, 2002.

[16] M.L. Munjal, *Acoustics of ducts and mufflers*, Wiley, New York, 1987.

[17] M. Åbom, *An Introduction to Flow Acoustics*, KTH Aeronautical and Vehicle Engineering, Stockholm, Sweden, 2006.

[18] H. Bodén, M. Åbom, Influence of errors on the two-microphone method for measuring acoustic properties in ducts, *Journal of the Acoustical Society of America* 79 (2) (1986) 541–549.

[19] M. Åbom, H. Bodén, Error analysis of two-microphone measurements in ducts with flow, *Journal of the Acoustical Society of America* 83 (6) (1988) 2429–2438.

[20] M. Åbom, Measurement of the scattering-matrix of acoustical two-ports, *Journal of Mechanical Systems and Signal Processing* 5 (2) (1991) 89–104.

[21] S. Allam, *Acoustic Modelling and Testing of Advanced Exhaust System Components for Automotive Engines*, Ph.D. Thesis, The Royal Institute of Technology, Stockholm,

Sweden, 2004.

[22] A.F. Seybert, D.F. Ross, Experimental determination of acoustic properties using a two-microphone random-excitation technique, *Journal of the Acoustical Society of America* 61 (5) (1977) 1362–1370.

[23] J.Y. Chung, D.A. Blaser, Transfer function method of measuring in-duct acoustic properties. I. Theory, *Journal of the Acoustical Society of America* 68 (3) (1980) 907–913.

[24] J.Y. Chung, D.A. Blaser, Transfer function method of measuring in-duct acoustic properties. II. Experiment, *Journal of the Acoustical Society of America* 68 (3) (1980) 914–921.

[25] Y. Özyörük, L.N. Long, M.G. Jones, Time-domain numerical simulation of a flow-impedance tube, *Journal of Computational Physics* 146 (1998) 29–57.

[26] S.W. Rienstra, Impedance models in time domain including the extended Helmholtz resonator model, *AIAA paper 2006-2686, 12th AIAA/CEAS Aeroacoustics Conference*, Cambridge, Massachusetts, 2006.

[27] S. Temkin, *Elements of Acoustics*, Acoustical Society of America, 2001.

[28] G. Kirchhoff, Uber den einfluss der wärmteleitung in einem gas auf die schallbewegung, *Annalen der Physik und Chemie* 134 (6) (1868) 177–193.

[29] C. Zwikker, C.W. Kosten, *Sound absorbing materials*, Elsevier, Amsterdam, 1949.

[30] H. Tijdeman, On the propagation of sound waves in cylindrical tubes, *Journal of Sound and Vibration* 39 (1) (1975) 1–33.

[31] E. Dokumaci, Sound transmission in narrow pipes with superimposed uniform mean flow and acoustic modelling of automobile catalytic converters, *Journal of Sound and Vibration* 182 (5) (1995) 799–808.

[32] E. Dokumaci, On transmission of sound in circular and rectangular narrow pipes with superimposed uniform mean flow, *Journal of Sound and Vibration* 210 (3) (1998) 375–389.

[33] R.J. Astley, A. Cummings, Wave propagation in catalytic converters: Formulation of the problem and finite element solution scheme, *Journal of Sound and Vibration* 188 (5) (1995) 635–657.

[34] K.S. Peat, A first approximation to the effects of mean flow on sound propagation through cylindrical capillary tubes, *Journal of Sound and Vibration* 175 (4) (1994) 475–489.

[35] J.–G. Ih, C.–M. Park, H.–J. Kim, A model for sound propagation in capillary ducts with mean flow, *Journal of Sound and Vibration* 190 (2) (1996) 163–175.

[36] K.–W. Jeong, J.–G. Ih, A numerical study on the propagation of sound through capillary tubes with mean flow, *Journal of Sound and Vibration* 198 (1) (1996) 67–79.

[37] K.S. Peat, Convected acoustic wave motion along a capillary duct with an axial temperature gradient, *Journal of Sound and Vibration* 203 (5) (1997) 855–866.

[38] K.S. Peat, R. Kirby, Acoustic wave motion along a narrow cylindrical duct in the presence of an axial mean flow and temperature gradient, *Journal of the Acoustical Society of America* 107 (4) (2000) 1859–1867.

[39] E. Dokumaci, An approximate dispersion equation for sound waves in a narrow pipe with ambient gradients, *Journal of Sound and Vibration* 240 (4) (2001) 637–646.

[40] S.F. Tardu, G. Binder, R.F. Blackwelder, Turbulent channel flow with large-amplitude velocity oscillations, *Journal of Fluid Mechanics* 267 (1994) 109–151.

[41] A. Scotti, U. Piomelli, Numerical simulation of pulsating turbulent channel flow, *Physics of Fluids* 13 (5) (2001) 1367–1384.

[42] P. Comte, M. Haberkorn, G. Bouchet, V. Pagneaux, Y. Aurégan, Large-Eddy Simulation of acoustic propagation in a turbulent channel flow, in: E. Lamballais, R. Friedrich,

B.J. Geurts and O. Métais, *Direct and Large-Eddy Simulation* VI, 2006, pp. 521–528.

[43] U. Ingard, V.K. Singhal, Sound attenuation in turbulent pipe flow, *Journal of the Acoustical Society of America* 55 (3) (1974) 535–538.

[44] D. Ronneberger, C.D. Ahrens, Wall shear stress caused by small amplitude perturbations of turbulent boundary-layer flow: an experimental investigation, *Journal of Fluid Mechanics* 83 (3) (1977) 433–464.

[45] R.R. Mankbadi, J.T.C. Liu, Near-wall response in turbulent shear flows subjected to imposed unsteadiness, *Journal of Fluid Mechanics* 238 (1992) 55–71.

[46] M.C.A.M. Peters, A. Hirschberg, A.J. Reijnen, A.P.J. Wijnands, Damping and reflection coefficient measurements for an open pipe at low Mach and low Helmholtz numbers, *Journal of Fluid Mechanics* 256 (1993) 499–534.

[47] S. Allam, M. Åbom, Investigation of damping and radiation using full plane wave decomposition in ducts, *Journal of Sound and Vibration* 292 (2006) 519–534.

[48] M.S. Howe, The damping of sound by wall turbulent shear layers, *Journal of the Acoustical Society of America* 98 (3) (1995) 1723–1730.

[49] H. Rämmal, *Studies of flow duct acoustics with application to turbocharged engines*, Ph.D. Thesis, The Royal Institute of Technology, Stockholm, Sweden, 2009.

[50] R. Glav, *On acoustic modelling of silencers*, Ph.D. Thesis, The Royal Institute of Technology, Stockholm, Sweden, 1994.

[51] R. Glav, The null-field approach to dissipative silencers of arbitrary cross-section, *Journal of Sound and Vibration* 189 (4) (1996) 489–509.

[52] R. Glav, The point-matching method on dissipative silencers of arbitrary cross-section, *Journal of Sound and Vibration* 189 (1) (1996) 123–135.

[53] D.W. Herrin, Z. Tao, J. Liu, A.F. Seybert, Using boundary element analysis to analyze multi-component exhaust systems, *SAE Technical Paper 2007–01–2182*, 2007.

[54] S. Boij, B. Nilsson, Scattering and absorption of sound at flow duct expansions, *Journal of Sound and Vibration* 289 (2006) 577–594.

[55] R.J. Alfredson, P.O.A.L. Davies, Performance of exhaust silencer components, *Journal of Sound and Vibration* 15 (2) (1971) 175–196.

[56] S. Allam, M. Åbom, Sound propagation in an array of narrow porous channels with application to diesel particulate filters, *Journal of Sound and Vibration* 291 (2006) 882–901.

[57] D. Marion, J. Ye, Effects of wall compliance on noise attenuation characteristics of air induction components, *SAE Technical Paper 2005–01–2356*, 2005.

[58] M. Alex, Air intake development – effects of coupled fluid/structure modes, *SAE Technical Paper 2001–01–1431*, 2001.

[59] J.–G. Ih, J.–H. Kang, Acoustic modeling of an air cleaner filter in the engine intake system, *Proceedings of the 13th International Congress on Sound and Vibration*, Vienna, Austria, 2006.

[60] J.M. Desantes, A.J. Torregrosa, A. Broatch, H. Climent, Silencing capabilities of non-silencer elements: An underused potential? *Proceedings of the 4th Styrian Noise, Vibration and Harshness Congress*, Graz, Austria, 2006.

[61] H. Rafael, *A two-port model for a turbo charger compressor and intercooler*, M.Sc. Thesis, The Royal Institute of Technology, Stockholm, Sweden, 2000.

[62] D. Jung, D. Assanis, Numerical modelling of cross flow compact heat exchanger with louvered fins Using thermal resistance concept, *SAE Technical Paper 2006–01–0726*, 2006.

[63] K. Nakonieczny, Numerical modeling of cross-flow plate-fin air-to-air heat exchanger

under unsteady flow conditions, *Numerical Heat Transfer Part A – Applications* 49 (1), (2006) 1–24.

[64] P.A. Bromnick, R.J. Pearson, D.E. Winterbone, Intercooler model for unsteady flows in engine manifolds, *Proceedings of the Institution of Mechanical Engineers Part D. Journal of Automotive Engineering* 212 (1998) 119–132.

[65] J. Galindo, J.R. Serrano, H. Climent, F.J. Arnau, New one-dimensional fluid-dynamic model for automotive intercoolers, *Proceedings of EAEC 8th European Automotive Congress*, Bratislava, Slovakia, 2001.

[66] M. Knutsson, M. Åbom, Acoustic analysis of charge air coolers, *SAE Technical Paper 2007–01–2208*, 2007.

[67] M. Knutsson, M. Åbom, Sound propagation in narrow channels with arbitrary cross sections and superimposed mean flow with application to charge air coolers, *Proceedings of the 14th International Congress on Sound and Vibration*, Cairns, Australia, 2007.

[68] E. Dokumaci, On attenuation of plane sound waves in turbulent mean flow, *Journal of Sound and Vibration* 320 (2009) 1131–1136.