Acoustic Modeling and Testing of Exhaust and Intake System Components

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Abstract

Intake and exhaust orifice noise contributes to interior and exterior vehicle noise. The order noise radiated from the orifice of the intake and exhaust systems is caused by the pressure pulses generated by the periodic charging and discharging process and propagates to the open ends of the duct systems.

The propagation properties of these pulses are influenced by the dimensions and acoustic absorption properties of the different devices in the intake/exhaust line (muffler, turbocharger, catalyst, intercooler, particulate filter, etc.). Additional to this pulse noise, the pulsating flow in the duct system generates flow noise by vortex shedding and turbulence at geometrical discontinuities.

Several turbochargers, catalytic converters, Diesel particulate filters and intercoolers elements were investigated and analyzed by performing two-port acoustic measurements with and without mean flow at both cold conditions (room temperature) and hot conditions (running engine test bed) to investigate these devices as noise reduction elements. These measurements were performed in a frequency range of 0 to 1200 Hz at no flow conditions and at flow speeds: 0.05 and 0.1 Mach.

A new concept for the acoustic modeling of the catalytic converters, Diesel particulate filters and Intercoolers, and a new geometrical model for the turbocharger were developed. The whole test configuration was modeled and simulated by means of 1-D gas dynamics using the software AVL-Boost. The results were validated against measurements. The validation results comprised the acoustic transmission loss, the acoustic transfer function and the pressure drop over the studied test objects. The results illustrate the improvement of simulation quality using the new models compared to the previous AVL-Boost models.
CHAPTER 1

Introduction
Introduction

In this chapter an introduction to the use of the automotive turbocharger and its influence on vehicle development is given as well as introductions to the main design and use of catalytic converters (CATs), Diesel particulate filters (DPFs) and intercoolers.

1.1 Introduction to Turbochargers

There has been an increasing trend to use turbochargers in vehicles in order to increase not only the power but also the fuel efficiency and to lower the exhaust pollutants. Turbocharged engines are not only commonly found in Diesel-powered vehicles but also in gasoline- powered vehicles ranging from the small 3-cylinder Kei cars from Japan to the twin-turbo high-performance vehicles of Germany. A turbocharger consists of a compressor Figure 1-a, which raises the density of air entering the engine and is typically driven by a turbine Figure 1-b, which uses the energy from the exhaust gases.

Rämmal and Åbom [1] mentioned that, due to the presence of the muffler in the exhaust system, noise problems are usually associated with the compressor in the intake side, especially the high frequency noise generated by the turbocharger, and that was referred to as the active acoustic properties of turbochargers. In the low-frequency region, the turbocharger unit also has an influence on the pressure pulses propagating from the engine. This effect is referred to as the passive acoustic properties of turbochargers [1].

Low frequency sound waves that propagate as plane waves in ducts can be described using linear acoustic 2-port models or nonlinear time-domain models. The former is typically used for quick prediction of acoustic properties for small pressure perturbation in ducts and the latter is typically used to model the high amplitude pressure wave propagation in an automotive engine intake and exhaust system.

Figure 1: A Garrett automotive turbocharger a) Compressor side and b) Turbine Side.
1.2 Introduction to catalytic converters, Diesel particulate filters and intercoolers

Due to the emission regulations both catalytic converters and Diesel particulate filters are installed in diesel engine exhaust systems. Usually these two elements are placed in cascade in an expansion chamber. The whole unit is called an after treatment device (ATD), Figure 2.

![An After Treatment Device](image)

**Figure 2: An After Treatment Device [2].**

Catalytic converters are used to reduce the toxicity of emissions by converting nitrogen oxides, carbon monoxide and hydrocarbons to nitrogen, oxygen, carbon dioxide, water and so on, which are less harmful for the environment. Therefore the exhaust pollution is reduced dramatically. The core part of a catalytic converter is usually a ceramic honeycomb, but sometimes stainless steel honeycombs are also used.

Diesel particulate filters (DPFs) are used to remove particulate matter and soot from the exhaust gas of diesel engines by physical filtration. The most common type is a ceramic honeycomb monolith. The structure of a diesel particulate filter is similar to a catalytic converter. The main difference is that channels are blocked at alternate ends for a diesel particulate filter, but not for a catalytic converter. Therefore the exhaust gas must pass through the walls between the channels causing the filtering of soot and particulate matter. As shown in Figure 3 the exhaust gases flow into the inflow channel passes through small holes in the walls to be filtered and then flow out through the outflow chamber.

![Sketch of the structure of a Diesel particulate filter](image)

**Figure 3: Sketch of the structure of a Diesel particulate filter [3].**

Due to the structure of the catalytic converters and Diesel particulate filters, they also improve the acoustic performance of exhaust systems. In earlier studies the pressure drop of after treatment devices was the main concern. Recently acoustic properties have been investigated by a number of authors [4-7] as will be discussed below.
**Introduction**

The intercooler is a heat exchanger device in the intake system used to cool the air pressurized by the turbocharger improving the volumetric efficiency, see Figure 4. This means that more air and fuel can be combusted and the power output of the engine is increased.

![Figure 4: An intercooler [8].](image)

There are air cooled and water cooled intercoolers. Depending on the purpose and space requirement, intercoolers can be dramatically different in shape and design. However, most intercoolers have a configuration with cavities at both inlet and outlet and dissipative narrow tubes in-between. From the acoustic point of view these two components can reduce the transmitted sound [8], but from the performance point of view the main properties of the intercooler design is the pressure drop and heat exchange efficiency.

The common feature of catalytic converters, Diesel particulate filters and intercoolers is narrow tubes. Knowledge about the acoustic damping of these narrow tubes is needed in order to have a good acoustic model. Theoretical models of wave propagation in such narrow tubes with and without mean flow have been investigated by a number of authors [4-7].
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1.3 Turbochargers

1.3.1 Turbocharger Basics

A turbocharger is a type of forced induction used in internal combustion systems, especially in automotive applications. The purpose is to increase the density of air entering the engine to create more power and improve efficiency. In a turbocharger, the compressor is driven by a turbine, which is usually driven by the hot exhaust gases.

Typically, an automotive turbocharger compressor [1] is composed of a centrifugal / radial compressor supported by center housing to a radial turbine. A shaft runs through the center housing, which connects the compressor and turbine wheels together. An image of a typical turbocharger compressor and turbine is shown in Figure 5. In the compressor, air is sucked through the inlet by the compressor rotor, which then compresses it through the ring diffuser and the compressor housing/volute collects the compressed air and directs it towards the engine. Usually, an intercooler at the outlet of the compressor is used to adiabatically cool the compressed air before entering the engine.

An automotive compressor and an automotive turbine are shown in Figure 5 and Figure 6.

Figure 5: Automotive turbo-compressor a) Photo b) Sketch.

Figure 6: Automotive Turbine.
The main parameters that should be considered for both the turbine and compressor are:

1. The compressor / turbine outlet diameter.
2. The Inducer diameter: the diameter where the air enters the rotor,
3. The Exducer diameter: the diameter where the air exits the rotor,
4. The Trim value: a term to relate the area ratio between the inducer and exducer of a compressor/turbine rotor.

\[
\text{Trim} = (\text{Inducer/Exducer})^2 \times 100
\]  

5. The area/radius (A/R) value: is a term used to describe turbocharger compressor / turbine housing size. It is defined as the ratio of the cross-sectional area of the outlet to the radius from the centerline of the centroid of the area as shown in Figure 8 [9, 10]. A larger A/R value would mean a larger housing (larger internal volume), which would allow more flow and vice versa.

Figure 7: Schematic to show the turbocharger rotor parameters.

In terms of performance, a higher trim value will allow more flow than a smaller value.

Figure 8: Schematic to show the way of calculating the A/R Ratio [9, 10].
1.3.2 Acoustic models

The developments in the field of turbocharger acoustics was recently discussed by Rämmal and Åbom [1]. He discussed the acoustic properties of a turbocharger divided into two main parts: the active acoustic properties and the passive acoustic properties.

The active acoustic properties of a turbocharger refers to the sound generation due to the rotating blades of a turbocharger unit, the clearance between the rotating blades and the housing, the RPM of the rotating blades, etc.

Raitor and Neise [11] made an experimental study to characterize the dominant acoustic source mechanisms for sound that propagates upstream of the compressor inlet side and downstream of the outlet side. At subsonic rotor blade tip speed, the noise generated by the secondary flow through the gap between the compressor casing rotor blade tips dominates. At sonic and supersonic rotor tip speed, the dominant noise is a tonal noise due to the rotor at its blade passing frequency and its harmonics. At supersonic rotor tip speed, buzz saw noise is produced in the inlet side due to the shock waves produced by the rotor blades.

The passive acoustic properties of a turbocharger refer to the reflection, transmission and absorption of sound of the unit, which is most important in the low frequency region. This means that the turbocharger unit will damp and interact with the pressure pulses coming from the engine.

Generally, acoustic models of the turbocharger are based on 1D gas dynamics. An attempt to extract the low frequency reflection and transmission properties of a turbo-compressor from a numerical simulation model was presented by Rämmal and Åbom [12]. It was based on a model proposed by Torregrosa, et al. [13] where the compressor/ turbine was divided into two volumes representing the inlet and outlet side of the unit. The simulated results using this method were compared to the experimental results from Rämmal and Åbom [14] and a good agreement was obtained in the low frequency range.

Two experimental studies [15, 16] including comparisons with models based on the geometry of the turbine side of the turbocharger unit, have been presented. The quality of the measurement results was however very poor at higher pressure ratios. Rafael [15] did an experimental investigation on a radial compressor with no flow and at low flow speeds. He found that the compressor was acoustically transparent at low frequencies and that the local maxima were formed at higher frequencies. The amplitude and locations of that local maximum changed when the flow speed was not zero.

Rämmal and Åbom [1] worked on two main models based on the two-port linear acoustic models to simulate the behavior of the compressor side. One model is based on the compressor geometry consisting of pipes, horns and area discontinuities coupled in series or in parallel. The second model was composed of resonators where the length of the resonators was based on the path differences within the compressor.

Peat, et al. [16] did a study on the turbine side where the model system was asymmetric as the reflection and transmission coefficients for the downstream and upstream transmission were not the same. This effect was attributed to the geometry of the turbine and the mean flow. Similar to for the compressor, they found that the transmission loss at low frequencies was low but increased with increasing flow speed. There was also a local maximum (peak) at high frequency. This peak in the transmission loss was attributed to a
Herschel-Quincke tube [17] effect due to the rotor passages, which acts as multiple acoustic paths of different length.

Pischinger [18], discussed a concept of modeling the turbo-unit geometry based on simplifications similar to in [16]. He presented the transmission loss for both the compressor and turbine at zero flow compared to simulation with the GT-Power software.

A 1D gas dynamics model of a simplified geometry model for the turbine was presented by Peat, et al. [16]. The model was divided into three sections: a volute, a rotor and a diffuser. The results of comparisons with experimental data were good in the low frequency range. A linear two-port model was also presented by Peat, et al. [16] which was composed of equivalent duct rotor passages. The volute was modeled as successive divisions which were connected to the rotor passages. This model was good in the high frequency range where it was able to reproduce the high frequency peaks obtained in the experiments. They concluded that a combination between the 1D gas dynamic models and the linear two-port model could give good results over a wide range of frequencies.

1.4 Catalytic converters, Diesel particle filters and intercoolers

Glav, et al. [4] first proposed a simple acoustic model for a catalytic converter using visco-thermal losses from narrow pipe theory while turbulent losses was assumed to be the same as in wide pipe theory with uniform mean flow.

Almost simultaneously Peat [5], Astley and Cummings [6] and Dokumaci [7] proposed models which deal with the convected visco-thermal acoustic equations simplified in the manner of the Zwiker and Kosten theory [19] with consideration of superimposed mean flow. Peat and Astley and Cummings [5, 6] used FEM to solve the equations with parabolic flow profiles. Both circular and rectangular cross sections were investigated by Astley and Cummings [6]. Dokumaci [20] also solved the same set of equations with uniform flow profile for a circular pipe. Later Dokumaci [20] concluded that although for laminar flow the parabolic flow profile is more realistic indeed there is no large difference in the results using a uniform mean flow profile. Dokumaci also proposed a model for both circular and rectangular section pipes with uniform mean flow profile and compared the results with results for a parabolic flow profile.

Howe [21] derived a model considering the viscous and thermal boundary layers. He combined the effects of turbulence and visco-thermal sub-layers on wave propagation in circular cross-section ducts. Although Howe’s model is not for narrow pipes it can still provide useful information.

The main issue of the wave propagation model in narrow pipes is the acoustic energy losses in pipes that cause damping of the acoustic waves. The wave propagation is characterized in the frequency domain by a complex valued wave number. The imaginary part corresponds to the viscous, thermal and other dissipation terms. The wave number without losses is defined as $k_0 = \omega/c_0$, where $c_0$ is the sound speed. For waves propagating in narrow pipes in order to account all sorts of losses, like viscous, thermal, mean flow, this wave number needs to be modified. To find the wave number is therefore the target for the modeling of narrow pipes. During the investigation within this project it was found that the viscous and thermal dissipations on the walls of the narrow tubes can be modeled using absorptive material theory to describe the wall impedances when calculating the targeted wave number.
1.4.1 Acoustic two-port models for catalytic converters and intercoolers

Dokumaci [7] used a scattering matrix formulation which can easily be converted to a transfer matrix formulation. The Fourier transfer form of the pressure and acoustic velocity is given by

\[ \hat{p}(x) = A \exp(iKk_0x) \quad \text{and} \quad \hat{q}(x) = H(r) \hat{p}(x), \]

where \( K \) has two roots, and \( H(r) \) is a function of tube radius \( r \), air density \( \rho_0 \), and air temperature \( T_0 \) [7].

The acoustic two-port for a single tube with length \( L \) can be written as:

\[
\begin{pmatrix}
\hat{p}(0) \\
H(r_1) \hat{p}(0)
\end{pmatrix}
= T
\begin{pmatrix}
\hat{p}(L) \\
H(r_2) \hat{p}(L)
\end{pmatrix},
\]

Therefore the transfer matrix \( T \) is:

\[
T = \begin{pmatrix}
1 & 1 \\
H(r_1) & H(r_2)
\end{pmatrix}
\begin{pmatrix}
\exp(ik_0K_1L) & \exp(ik_0K_2L) \\
H(r_1)\exp(ik_0K_1L) & H(r_2)\exp(ik_0K_2L)
\end{pmatrix}^{-1},
\]

This is for one narrow tube. The transfer matrix of a whole converter or intercooler with \( N \) tubes is:

\[
T' = \begin{pmatrix}
T_{11} & T_{12} \\
NT_{21} & T_{22}
\end{pmatrix},
\]

1.4.2 Acoustic two-port models for Diesel particulate filters

For Diesel particulate filters the channels are blocked at the end. Therefore the model for a DPF should be modified somewhat compared to the model for catalytic converters. The pressure will be different for channel in-flow and out-flow, since the flow needs to go through the filter parts. Allam and Åbom [24] proposed a DPF model based on Dokumaci [20] rectangular tube model with square cross section, \( 2a \times 2a \), to solve the convective equations. And by applying Darcy’s law, the coupling between these channels inflow and outflow is:

\[
p_1 - p_2 = R_w u_w, \quad R_w = \mu t_w / \sigma_w. \]

where:

- \( p_1 \): is the in-flow channel inlet pressure,
- \( p_2 \): is the out-flow channel outlet pressure,
- \( u_w \): is the velocity at the wall,
- \( R_w \): is the wall resistance,
- \( \mu \): is the dynamic viscosity,
- \( t_w \): is the wall thickness and,
- \( \sigma_w \): is the wall permeability.
Accordingly the boundary condition of acoustic velocity at the wall is no longer zero but \( u_w \). The averaged velocity gradient \( \langle \nabla \cdot u' \rangle \) becomes

\[
\langle \nabla \cdot u' \rangle = \frac{1}{4a_j} \int \left( \frac{\partial u'_{ij}}{\partial x} + \frac{\partial u'_{ij}}{\partial y} + \frac{\partial u'_{ij}}{\partial z} \right) = \frac{\partial (u'_{ij})}{\partial x} + \frac{1}{4a_j} \phi u_j n_j ds. \tag{8}
\]

with \( j = 1, 2 \), which denotes the inflow channels and outflow channels. \( C_j \) is the curve around the channel perimeter. With the parameter acoustic wall velocity \( u_w \), the last term of the right hand side of equation (8) becomes

\[
\phi u_j n_j ds = (-1)^{j-1} \phi u_w ds = (-1)^{j-1} 8a_j u_w. \tag{9}
\]

where \( \bar{u}_w \) is the averaged acoustic wall velocity along the perimeter \( C_j \). Replacing the acoustic wall velocity \( u_w \) by Darcy’s law the equation above can be rewritten as

\[
\phi u_j n_j ds = (-1)^{j-1} \frac{8a_j (p_1 - p_2)}{R_w}. \tag{10}
\]

Therefore the averaged term \( \langle \nabla \cdot u' \rangle \) can be written as

\[
\langle \nabla \cdot u' \rangle = -iK k_0 \langle H_j \rangle p_j + (-1)^{j-1} \frac{2}{a_j} \frac{(p_1 - p_2)}{R_w}. \tag{11}
\]

This is used for the mass conservation equations. From the mass equations the general expression for \( K \) can be obtained. Using iteration, by the Newton-Raphson method, the roots of \( K \) can be obtained. There are four roots for \( K \). The roots for the wave propagation in positive and negative directions have the same amplitude but different signs. Compared to the earlier work of Allam and Åbom [2] this model is more accurate since the earlier model approximates the viscous and thermal losses, whereas in the new model, the convective acoustic wave equations are solved.

Each of roots corresponds to a 2-D mode \( e_n \). The Fourier transfer form of sound pressure can be expressed as

\[
\begin{pmatrix}
\hat{p}_1(x) \\
\hat{p}_2(x)
\end{pmatrix} = \sum_{n=1}^{4} \hat{a}_n e^{-ik_n K x e_n}, \tag{12}
\]

and with the relation \( u'_i = p'H(r) \) the acoustic volume velocity can be expressed as

\[
\begin{pmatrix}
\hat{q}_1(x) \\
\hat{q}_2(x)
\end{pmatrix} = \sum_{n=1}^{4} \hat{a}_n H(r) e^{-ik_n K x e_n}, \tag{13}
\]

where \( \hat{q} = u' \cdot S \), and \( S \) is the cross sectional area.

For \( x=0 \) and \( x=L \) the pressure and acoustic velocity can be expressed as
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\[
\begin{pmatrix}
\hat{p}_1(0) \\
\hat{p}_2(0) \\
\hat{q}_1(0) \\
\hat{q}_2(0)
\end{pmatrix} = S \begin{pmatrix}
\hat{p}_1(L) \\
\hat{p}_2(L) \\
\hat{q}_1(L) \\
\hat{q}_2(L)
\end{pmatrix},
\]

(14)

where \( S \) is a four port matrix. Using the boundary condition \( \hat{q}_2(0) = 0 \) and \( \hat{q}_1(L) = 0 \), equation (14) can be simplified to a two-port problem:

\[
\begin{pmatrix}
\hat{p}_1(0) \\
\hat{q}_1(0)
\end{pmatrix} = T \begin{pmatrix}
\hat{p}_2(L) \\
\hat{q}_2(L)
\end{pmatrix},
\]

(15)

where \( T \) is a 2 by 2 matrix as function of \( S_{i,j} \), where \( i,j=1,2,3,4 \). Because there are \( N \) tubes for the whole filter section the entire two-port matrix can be expressed as

\[
T' = \begin{pmatrix}
T_{11} & T_{12}/N \\
NT_{21} & T_{22}
\end{pmatrix}.
\]

(16)

By using the coupling condition of equation (16) the expression of \( K \) can be obtained after some mathematical steps. Therefore there are four roots for \( K \).

1.4.3 Models for Absorptive Materials

1.4.3.1 Formulas for characteristic impedance and wavenumber

Delany and Bazley (1970) [22], gave a normalized empirical formula for the complex impedance \( \tilde{Z} \) and wavenumber \( \tilde{k} \) for both fibrous materials and porous materials, and for different filling densities.

\[
\tilde{Z} = \rho_o c_o \left[ 1 + c_1 \left( \frac{f}{R_f} \right)^{c_2} \right] + i \left[ c_3 \left( \frac{f}{R_f} \right)^{c_4} \right],
\]

(17)

\[
\tilde{k} = \frac{\omega}{c_o} \left[ 1 + c_5 \left( \frac{f}{R_f} \right)^{c_6} \right] + i \left[ c_7 \left( \frac{f}{R_f} \right)^{c_8} \right].
\]

(18)

with:

- \( \rho_o \): Air density at room temperature during the measurement [kg/m\(^3\)].
- \( c_o \): Speed of sound at room temperature during the measurement [m/s].
- \( f \): Frequency [Hz].
- \( R_f \): Material flow resistivity [Pa s/m\(^2\)].
- \( \omega \): Angular frequency \( \omega = 2\pi f \) [rad/s].
- \( c1-c8 \): Material constants shown in table 1 for both porous and fibrous materials as described by Delany and Bazley.
Table 1: c1-c8 Values for the material constants, as described by Delany and Bazley [22].

<table>
<thead>
<tr>
<th></th>
<th>Fibrous Materials</th>
<th>Porous Materials</th>
</tr>
</thead>
<tbody>
<tr>
<td>c1</td>
<td>0.0511</td>
<td>0.0571</td>
</tr>
<tr>
<td>c2</td>
<td>-0.75</td>
<td>-0.754</td>
</tr>
<tr>
<td>c3</td>
<td>-0.0768</td>
<td>-0.087</td>
</tr>
<tr>
<td>c4</td>
<td>-0.73</td>
<td>-0.732</td>
</tr>
<tr>
<td>c5</td>
<td>0.0858</td>
<td>0.0978</td>
</tr>
<tr>
<td>c6</td>
<td>-0.7</td>
<td>-0.7</td>
</tr>
<tr>
<td>c7</td>
<td>-0.1749</td>
<td>-0.189</td>
</tr>
<tr>
<td>c8</td>
<td>-0.59</td>
<td>-0.595</td>
</tr>
</tbody>
</table>

Delany and Bazley [22], set constraints for using equations (17) and (18) and for homogeneous materials with porosity close to one, this constrain is as follows

$$0.01 < \frac{f}{R_f} < 1.0.$$  \hspace{1cm} (19)

Mechel (1992) [23] presented another set of empirical formulas for the characteristic impedance and wave number for glass fiber with

$$\tilde{Z} = \rho_o c_o \left[ 1 + 0.0235 \left( \frac{\rho_o f}{R_f} \right)^{-0.887} \right] + i \left[ -0.0875 \left( \frac{\rho_o f}{R_f} \right)^{-0.770} \right],$$  \hspace{1cm} (20)

$$\tilde{k} = \frac{\omega}{c_o} \left[ 1 + 0.102 \left( \frac{\rho_o f}{R_f} \right)^{-0.705} \right] + i \left[ -0.179 \left( \frac{\rho_o f}{R_f} \right)^{-0.674} \right],$$  \hspace{1cm} (21)

$$\left( \frac{\rho_o f}{R_f} \right) < 0.25.$$  \hspace{1cm} (22)

The negative imaginary part of the wave number particularly ensures the dissipation of sound, the real part greater than unity indicates a lower speed of sound through the absorbing material. When the flow resistivity approaches zero, both characteristic impedance and wave number approach the values of air.

1.4.3.2 Speed of sound and density in absorbing materials

Calculation of the cut-off frequency of the higher order modes depends on the speed of sound through the absorbing material. The complex speed of sound of the absorbing material and the complex density of the absorbing material can be estimated from
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\[ c_a = \frac{c_o}{\left( \frac{k}{k_o} \right)} \]  
\[ \rho_a = \frac{\bar{k} \bar{Z}}{\omega} \]  

with:
- \( c_a \): Complex speed of sound of the absorbing material.
- \( k_o \): Wavenumber of the air at room temperature during measurements.
- \( \rho_a \): Complex density of the absorbing material.

As the speed of sound in the absorbing material is lower than in air; higher order modes can propagate at lower frequencies in the absorbing material compared to in air.

The above model for absorptive material was used to fill in a pipe and solve the problem using the two port theory in the frequency domain, by changing the internal medium parameters such as the speed of sound, wave number and complex density with the values calculated from equations (23) and (24).

1.4.4 Modeling of horn elements

The horn or conical section, which have been used in the modeling of turbocharger as will be discussed in section 4.2 can be modeled with an approximate method as a series of “short straight ducts” as proposed by Åbom [25]. The procedure is shown in Figure 9 [26].

![Diagram of horn approximation](image)

Figure 9: Approximation of a horn using piecewise constant area straight duct [26].

To ensure convergence, typically five or seven pieces are chosen per wavelength for the highest frequency of interest. For the whole section, the total transfer matrix is the product of the individual straight pipe element transfer matrices. The no flow transfer matrix of a straight pipe with length \( l \) is [26]:
\[
\begin{pmatrix}
    p_1 \\
    q_1
\end{pmatrix} =
\begin{pmatrix}
    \cos(k_0 l) & jZ_0 \sin(k_0 l) \\
    j/l Z_0 \sin(k_0 l) & \cos(k_0 l)
\end{pmatrix}
\begin{pmatrix}
    p_2 \\
    q_2
\end{pmatrix},
\]

where:

\[Z_0 = \rho_0 c_0 / S,\]

- \(S\) is the duct cross section area,
- \(L\) is the distance of the straight pipe,
- \(P\) is the pressure at one end \(I\),
- \(q_i\) is the volume velocity one end \(I\),
- \(k_0 = \omega/c_0\) is the wave number.
CHAPTER 2

Two-port Acoustic Measurements
In this chapter a brief description of the measurement procedures for acoustic two port measurement techniques used in the zero mean flow mean flow test cases are presented. The tests are used to determine the transfer matrix of the object under investigation.

### 2.1 Two-microphone wave decomposition

Sound in straight hard-walled ducts propagates as plane waves below the first cut-on frequency, and the sound field can in the frequency domain be expressed as

\[
\hat{p}(x, f) = \hat{p}_+(f)e^{(-i k_x s)} + \hat{p}_-(f)e^{(i k_x s)},
\]

and

\[
q(x, f) = \frac{A}{\rho c} \hat{p}_+(f)e^{(-i k_x s)} - \hat{p}_-(f)e^{(i k_x s)}
\]

where \( p \) is the acoustic pressure, \( q \) is the acoustic volume velocity, \( p_+ \) and \( p_- \) are the travelling wave amplitudes, \( k_x \) is the complex wave number in the positive and negative \( x \)-direction, \( \rho_o \) is the density of air, \( c_o \) is the speed of sound and \( A \) is the duct cross-section area and \( x \) is the position along the \( x \)-axis.

![Figure 10: Measurement configuration for the two-microphone method.](image)

The travelling wave amplitudes can be calculated by using pressure measurements at two microphone positions as shown in Figure 10. The acoustic pressures at position 1 and 2 are

\[
\hat{p}_1(f) = \hat{p}_+(f) + \hat{p}_-(f),
\]

\[
\hat{p}_2 = \hat{p}_+(f)e^{(-i k_x s)} + \hat{p}_-(f)e^{(i k_x s)},
\]

where \( s \) is the microphone spacing.

Rearranging equation (28), the traveling wave amplitudes are obtained as

\[
\hat{p}_+(f) = \frac{\hat{p}_1(f)e^{(i k_x s)} - \hat{p}_2(f)}{e^{(i k_x s)} - e^{(-i k_x s)}},
\]

\[
\hat{p}_-(f) = \frac{-\hat{p}_1(f)e^{(-i k_x s)} + \hat{p}_2(f)}{e^{(-i k_x s)} - e^{(-i k_x s)}}.
\]
According to Allam [27] the following conditions should be fulfilled for successful use of this method:

- The measurements should be in the plane wave region.
- The duct wall must be rigid in order to avoid higher order mode excitation.
- The test object should be placed at a distance at least twice the duct diameter from the nearest microphone in order to avoid the near field effects coming from the higher order mode excitation of non-uniform test objects.
- The plane wave propagation should not be attenuated. As this is not true in practice; neglecting the attenuation between the microphones leads to a lower limit in applicability. Åbom and Boden [28, 29] showed that the two-microphone method has the lowest sensitivity to errors in the input data in a region around $k_s = 0.5\pi(1 - M^2)$ and to avoid large sensitivity to the errors in the input data, the two microphone method should be restricted to the frequency range of

$$0.1\pi(1 - M^2) < k_s < 0.8\pi(1 - M^2),$$

where $M$ is the Mach number.

### 2.2 Acoustical two-port systems

An acoustical two-port system [30], see Figure 11, is a linear input-output system. The acoustic properties of such a system can be determined through linear acoustic theory or by measurements. The acoustical two-port system is often expressed in the transfer matrix form with the acoustic pressure and volume velocity as variables.

![Figure 11: Schematics of an acoustic two port system.](image)

In the frequency domain, the transfer matrix $T$ with acoustic pressure $p$ and volume velocity $q$ for an acoustic two-port system with no internal sources is given by

$$\begin{pmatrix}
\hat{p}_a \\
\hat{q}_a 
\end{pmatrix} = 
\begin{pmatrix}
T_{11} & T_{12} \\
T_{21} & T_{22}
\end{pmatrix}
\begin{pmatrix}
\hat{p}_b \\
\hat{q}_b
\end{pmatrix},$$

where $a$ and $b$ denote the inlet or outlet side of the test object, respectively.

Equation (31) can be solved when the following condition is satisfied

$$\det\begin{pmatrix}
T_{11} & T_{12} \\
T_{21} & T_{22}
\end{pmatrix} \neq 0.$$
Two-port Acoustic Measurements

The acoustical two-port theory assumes that: the sound field is linear so that the analysis can be done in the frequency domain, only plane waves are allowed to propagate at the inlet and outlet section and the system is passive, i.e. no internal sources.

2.3 Experimental determination of the acoustical two-port system

As equation (31) has four unknowns and only two equations two sets of independent data are needed in order to obtain the transfer matrix

$$\begin{bmatrix} 1,1 & 1,2 \\ 2,1 & 2,2 \end{bmatrix} \begin{bmatrix} p_a \\ q_a \end{bmatrix}_1 = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} p_b \\ q_b \end{bmatrix}_2,$$

where subscripts 1 and 2 denote the two test states.

In order to experimentally determine the two-port data (transfer matrix), two independent test states are needed and these test states can be obtained by using either of the following methods: the two-load method and the two-source method [31].

2.3.1 The two-load method

Using the two load method, the two independent test cases needed to get the transfer matrix are obtained by changing the loads at the termination as shown in Figure 12. The two loads should not be similar; otherwise the results will have large errors. In this study, the impedance at the termination was changed by using an open termination (impedance A) and closed termination (impedance B). The two-load technique was in the present study only used for no-flow investigations.

2.3.2 The two-source method

The two-source method uses loudspeakers at both sides of the test object as shown in Figure 13. The first test state is obtained by turning loudspeaker A on and B off and the second test state is obtained by turning loudspeaker B on and A off.
Two-port Acoustic Measurements

![Diagram of measurement configuration for the acoustic two-port system of the two-source method.](image)

**Figure 13:** Measurement configuration for the acoustic two-port system of the two-source method.

### 2.4 Acoustic transmission loss

One way of describing the acoustic performance of a system is through the transmission loss. The transmission loss (TL) is the difference in sound power level between the incident wave entering and the transmitted wave exiting the system, assuming that the termination is anechoic.

\[
TL_d = 10 \log_{10} \frac{W_i}{W_t},
\]

(34)

The incident \((W_i)\) and transmitted \((W_t)\) sound power levels can be written as

\[
W_i = \frac{P_i^2}{z_a}(1 \pm M_a)^2,
\]

(35)

and

\[
W_t = \frac{P_t^2}{z_b}(1 \pm M_b)^2,
\]

(36)

where:

- \(z_a = \rho_a c_o / A_a\) is the characteristic impedance at port (a),
- \(z_b = \rho_b c_o / A_b\) is the characteristic impedance at port (b),
- \(A_a, A_b\) are the cross-sectional areas for port (a) and port (b) respectively,
- \(M_a, M_b\) are the Mach numbers for port (a) and port (b) respectively.

The sign depends on the direction of the sound propagation, positive if it is towards the downstream direction (in the direction of the flow) and negative if it is towards the upstream direction (in the direction against the flow).

From equations (35) and (36), the transmission loss in the downstream direction (sound transmission in the direction of flow) is given by

\[
TL_d = 10 \log_{10} \left( \frac{P_i}{P_t} \right)^2 \frac{z_b (1 + M_a)^2}{z_a (1 + M_b)^2},
\]

(37)

or in terms of the transfer matrix
Two-port Acoustic Measurements

\[ TL_a = 10 \log_{10} \left( T_{11} + \frac{T_{12}}{z_b} + \frac{z_a T_{21}}{z_b} + \frac{z_a T_{22}}{z_a} \right)^2 \frac{z_b (1 + M_a)^2}{z_a (1 + M_a)^2} \] \tag{38}

Similarly, the transmission loss in the upstream direction (sound transmission in the direction against flow) is given by

\[ TL_u = 10 \log_{10} \left( \frac{\tilde{p}_a}{\tilde{p}_b} \right)^2 \frac{z_b (1 - M_a)^2}{z_a (1 - M_a)^2} \] \tag{39}

and in terms of the transfer matrix

\[ TL_u = 10 \log_{10} \left( T_{11} - \frac{T_{12}}{z_b} + \frac{z_a T_{21}}{z_b} + \frac{z_a T_{22}}{z_a} \right)^2 \frac{z_b (1 + M_a)^2}{z_a (1 + M_a)^2} \] \tag{40}

where

\[ \begin{pmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{pmatrix} = \begin{pmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{pmatrix}^{-1} \] \tag{41}

In the zero flow case, the transmission loss is simplified to

\[ TL_u = 10 \log_{10} \left( \frac{\tilde{p}_a}{\tilde{p}_b} \right)^2 \frac{A_a}{A_b} = 10 \log_{10} \left( T_{11} - \frac{T_{12}}{z_b} + \frac{z_a T_{21}}{z_b} + \frac{z_a T_{22}}{z_a} \right)^2 \frac{z_b}{z_a} \] \tag{42}

2.4 Acoustic transfer function

In this work an acoustic transfer function defined as the ratio between the acoustic pressures on either side of the test object (microphones 2 and 3 as seen in Figures 12 and 13). This transfer function was used to test the simulation models geometries and to make sure that the simulation models will have identical geometries as the tested objects.
CHAPTER 3

Acoustic Simulation Models
In this chapter, the development and description of the nonlinear acoustic models of the turbocharger at the compressor and turbine sides, catalytic converter (CAT), the Diesel particulate filter (DPF) and intercooler units are presented. The model development was made within the AVL Boost software environment and involved empirical fits of model parameters to obtain the correct acoustical as well as thermodynamic performance data.

3.1 Simulation in the AVL Boost environment

The AVL Boost environment has two types of modeling solvers, one for the linear acoustic analysis in which the equations of the acoustic frequency domain two port theory is used, while the other solver is for nonlinear acoustic analysis in which the equations for 1-D gas dynamics are used and are solved in the time domain. The nonlinear solver can in a direct way handle the engine acoustic source excitation as will be discussed in section 4.2.

Using AVL Boost nonlinear solver the user needs to make averaging for the produced simulation results to avoid the noisy results as seen in most of the following simulation that can cost more computational time.

3.2 Acoustic Simulation model for Turbochargers

3.2.1 Turbo-Compressor

An equivalent pipe model based on the geometry of the compressor was made using the AVL BOOST nonlinear solver. The compressor model is composed of 5 main parts: inlet-rotor-diffuser-volute-outlet. The inlet and outlet sections are straight/conical pipes. The rotor section is modeled as a constricting pipe neglecting the rotor blades. The Blades do not form a seal and thus cannot be considered as separate paths. In order to represent the asymmetric 3D shape of the diffuser and volute sections into essentially a 1D model, the diffuser was divided into branches that intersect at different points in the volute as shown in Figure 14. The model is similar to the model of Peat, et al. [16] on the turbine side. The BOOST [26] representation of the compressor model is shown in Figure 15.

![Figure 14: Schematics of the equivalent pipes diffuser branching.](image)
In the BOOST model, the measurement points for the sound pressure levels are marked MP1 and MP2 in the figure and are positioned at the same locations as microphones 2 and 3 in the experiment. The acoustic source in the model with a frequency resolution of 10 Hz, is located at system boundary 1 (SB1) and the termination is at the opposite end, system boundary 2 (SB2). The termination type was set to anechoic termination for transmission loss simulations and standard termination (open end) for transfer function simulations. The boundary conditions were set to the ambient conditions.

### 3.2.1.1 Effect of the diffuser discretization

To test the effect of the discretisation of the diffuser, 4 models were made for compressor 1 by dividing the diffuser region into 2, 4, 6 and 12 branches (Figure 16) and the results were then compared to the experimental results. The total volume of the diffuser region was kept constant for all the models and the length, inlet and outlet diameters of the each diffuser branch of the model are given by

\[
L_{diff} = \frac{(d_{diff} - d_{exd})}{2},
\]

\[
D_{diff} = \sqrt{w \times d_{exd} \times 4 / n_{seg}}
\]

\[
D_{diff} = \sqrt{w \times d_{diff} \times 4 / n_{seg}}
\]

where \(d_{diff}\) is the diffuser plate diameter, \(d_{exd}\) rotor exducer diameter, \(w\) is the diffuser width and \(n_{seg}\) is the number of branches.
Figure 16: BOOST representation of the diffuser-volute region, a) 2 diffuser branches, b) 4 diffuser branches, c) 6 diffuser branches and d.) 12 diffuser branches.
Acoustic Simulation Models

A comparison of the transmission loss curves for the 4 models and the measurement are shown in Figure 17 and similarly for the transfer function in Figure 18. The measurements of the transfer function were performed with open end boundary conditions and for zero flow conditions.

As shown by the figures, by increasing the number of branches, the results from the simulations approach those from the measurements. As the difference between the results of models with a diffuser with 6 and 12 branches are negligible and in order to minimize the computation time, the model with 6-branches was used for the remainder of this study.

Figure 17: Comparison between the measured and simulated transmission loss results of the different diffuser branching configuration models for compressor 1.

Figure 18: Comparison between the measured and simulated transfer function results of the different diffuser branching configuration models for compressor 1.
3.2.2 Turbine

The turbine model is geometrically the same as the compressor model with the flow directions reversed. The diffuser area of the compressor is equivalent to the vane-less nozzle or guide vane chamber for a turbine.

3.2.3 Compressor model under flow conditions

The zero-flow model presented in the previous section was modified by adding a turbo-compressor element (TCP 1) in the middle of the rotor segment as shown in Figure 19. This was added as a substitute for the rotor and to generate the necessary mass flow and boost pressure. A PID Controller (PID1) was added to control the boost pressure generated by the turbo-compressor element in order to reach the desired mass flow speed. Additional pipes (pipes 19 – 25) were also added in order to add the restrictions (marked as R1 – R6 in the diagram). The restrictions were added in order to control the flow distribution and the acoustic propagation to the different diffuser branches.

The system temperature and pressure boundary conditions for both System Boundary 1 (SB1) and System Boundary 2 (SB2) were set to the same values as in the experiment. The acoustic source and anechoic termination were alternated between the two system boundaries depending on what transmission loss direction was to be simulated. SB1 was set as an acoustic source and SB2 as an anechoic termination for the downstream transmission loss and SB2 were set as an acoustic source and SB1 as an anechoic termination for the upstream transmission loss.

![Figure 19: BOOST model of a turbocharger compressor with compressible flow.](image)

The flow coefficients in the restrictions were set to distribute the mass flow across the diffuser evenly and were adjusted to give the correct transmission loss curves.

3.2.4 Turbine model under flow conditions

Similar to the compressor model, a turbine element (TC1) and flow restrictions were added into the model as shown in Figure 20. These were added in order to control the pressure drop and mass flow through the system. A PID Controller (PID1) was added to control the flow restrictions which in turn control the mass flow to match the experimental values. A second PID was added to control the temperature drop through the rotor. Flow resistivity was added also in the rotor in order to get the right level of acoustic damping.
Acoustic Simulation Models

The system boundary conditions were again set to the same values in the experiment with the high pressure boundary and the flow coming from SB2. The acoustic source and anechoic termination were alternated between the two system boundaries depending on what transmission loss direction was to be simulated. SB1 was set as an acoustic source and SB2 as an anechoic termination for the downstream transmission loss and SB2 were set as an acoustic source and SB1 as an anechoic termination for the upstream transmission loss.

![BOOST model of a turbocharger turbine with compressible flow.](image)

**Figure 20:** BOOST model of a turbocharger turbine with compressible flow.

3.3 CAT, DPF and Intercooler simulation Models

3.3.1 CAT Simulation Model

![Schematic drawing showing the Original Boost model for catalytic converters.](image)

**Figure 21:** Schematic drawing showing the Original Boost model for catalytic converters.

Figure 21 shows a schematic diagram for the Boost original CAT model, which was built for normal thermodynamic performance applications and did not account for acoustical performance calculations. This model anyway has some influence on the acoustical performance of the complete exhaust system.

The initial work with the original Boost CAT model to get the correct acoustical performance involved varying the following parameters in the model:

1. The inlet/outlet volume dimensions.
2. The Monolith Length and volume.
3. The friction coefficients for the CAT.
4. Open Front Area (OFA).
5. Hydraulic Unit.
6. Geometrical Surface Area (GSA).

Figure 22 shows a schematic diagram for the internal construction of the CAT element, it consists mainly of inlet and outlet collectors (volumes) with conical (horn) shapes and the core part (monolith). The core part of a catalytic converter is commonly a ceramic honeycomb, but sometimes stainless steel honeycombs are also used. The monolith has the shape of narrow parallel tubes and this has an effect on CAT acoustic properties by adding losses.
Acoustic Simulation Models

Figure 22: Schematic drawing showing the internal construction of the catalytic converter.

The currently used CAT monolith element model is based on one single narrow tube. The results are multiplied by the number of tubes in terms of the so called cell density.

For the acoustical wave propagation in ducts, it was assumed that the duct walls were hard walls with no sound absorption. The material forming the CAT monolith narrow tubes, will in reality have a sound absorption which can be represented by means of a flow resistivity.

Figure 23: Example CAT monoliths.

The flow resistivity is defined as the ratio of pressure difference ($\Delta p$) per unit length of the tube and flow velocity, and it is one of the important acoustic properties of absorbing material

$$R_f = \frac{\Delta p}{u \cdot h},$$

(46)

where:

- $R_f$: is the flow resistivity [Pa s/m²],
- $u$: is the mean flow velocity of the air [m/s],
- $h$: is the material thickness [m].

For the newly developed model as discussed below the cross section of a circular CAT was selected as the basic shape. The main cross section of the monolith was divided into two sections, the first section representing the open ended narrow tubes and the second section represents the walls between these tubes.

To simulate this; both linear and nonlinear solvers in AVL Boost have been used. Geometrically the flow path was divided into two parallel branches with equal lengths and different diameters as discussed below see Figure 24.
The reason for having two tubes is to separate the effect of the flow resistivity in PIPE I and the hydraulic diameter effect in PIPE II since both of them are influencing the mean flow velocity in the pipes. Parametric studies were made on the CAT main geometries and the monolith flow resistivity to fit the CAT acoustical properties to the measured data. From these studies the CAT monolith was modeled using the Boost nonlinear solver as seen in Figure 24 with the following considerations:

a) The first path denoted with PIPE I, can be considered as a chamber filled with absorptive material that has a flow resistivity value of 1800 [Ns/m$^4$], which simulates the sound absorption effect of the wall section, to account for the acoustic damping at high frequencies. The diameter of this pipe should be approximately 70% of the main CAT cross sectional diameter. The length of this pipe is the same length of the CAT monolith.

b) The second path, denoted by PIPE II, is a pipe with a diameter approximately half the diameter of the PIPE I, to allow the flow to go through without acoustical damping and to generate the corresponding pressure drop across the monolith, and applying a hydraulic diameter equal to 1.3 [mm] to account for the conversion from non-circular cross section of the CAT narrow tubes to circular cross sections. The length of this pipe is the same length of PIPE I.

It was also possible to use the Boost linear solver, with the same conditions as above, except for the hydraulic diameter which was not included in the current Boost version. The hydraulic diameter is needed to include the effect of the shape of the narrow pipes in the monolith on the pressure drop. This problem caused a slight difference in geometry which may be the cause of differences seen in the transmission loss results.

### 3.3.2 DPF Simulation Model

The DPF element consists mainly of inlet and outlet collectors with different shapes and the core part (monolith), the core part has a shape of narrow parallel tubes which forces the exhaust gases to pass through the tube walls, as shown in Figure 25. This design gives acoustic losses and influences the acoustic performance of the exhaust system.
Acoustic Simulation Models

Parametric studies were done on the DPF main geometries and the DPF core part flow resistivity to fit the DPF acoustical properties to the measured data. From these studies we concluded that the DPF core part can be modeled as seen in Figure 26 with the following considerations:

**a)** The first path denoted with PIPE I, can be considered as a chamber filled with absorptive material that has a flow resistivity value of 1800 [Ns/m^4], which simulates the wall sound absorption effect, to account for the acoustic damping at high frequencies. The diameter of this pipe should be approximately 70% of the main DPF cross sectional diameter. The length of this pipe is the same as the DPF monolith length.

**b)** The second path, denoted by PIPE II, is a pipe with a diameter approximately half the diameter of PIPE I, to allow the flow to pass through without acoustical damping and to generate the corresponding pressure drop across the monolith, and with a hydraulic diameter equal to 1.3 [mm], to account for the conversion from non-circular cross section of the DPF narrow tubes to circular cross sections. The length of this pipe is the same length of PIPE I.

Using the Boost linear solver, the same geometry described above was used except for applying the hydraulic diameter to take the shape of the pipe in the monolith into account, as discussed in section 3.3.1
Acoustic Simulation Models

The reason that both pipes have the same length is that the thickness of the core part narrow tubes walls is very small compared to the main core part length. This thickness can then be neglected and we can simply assume that the model pipes lengths are the same as the main core part length.

3.3.3 Intercooler Simulation Model

An intercooler is a component of inlet system and used to cool engine air, see Figure 27 for a photo of an air cooled intercooler.

![Figure 27: A photo of an intercooler.](image)

Parametric studies were made on the Intercooler main geometries and its flow resistivity to fit the intercooler acoustical properties to the measured data. From these studies we concluded that the intercooler flow passages can be modeled as seen in Figure 28 with the following considerations:

a) The first path denoted with PIPE I, can be considered as a chamber filled with absorptive material that has a flow resistivity value of 400 [Ns/m4], which simulates the wall sound absorption effect, to account for the acoustic damping at high frequencies. The diameter of this pipe should be approximately 70% of the main intercooler cross sectional diameter. The length of this pipe is the same length of the CAT monolith.

![Figure 28: Simulation Model in AVL-Boost software.](image)

b) The second path, denoted by PIPE II, is a pipe with a diameter approximately half the diameter of the PIPE I, to allow the flow to go through it without acoustic damping and to generate the corresponding pressure drop across the intercooler core part and with a hydraulic diameter equal to 4 [mm] to account for the conversion from non-circular cross section of the Intercooler tubes to circular cross sections. The length of this pipe is the same length of PIPE I.
Using the Boost linear solver, the same geometry described above was used except for using the hydraulic diameter as discussed in section 3.3.1.
CHAPTER 4

Results
Results

In this chapter a series of comparisons between the measurements results and the Boost acoustic models simulation results for the turbocharger both compressor and turbine sides, catalytic converter (CAT), the Diesel particulate filter (DPF) and intercooler based on the previously described models are presented.

4.1 Turbocharger results at cold conditions

4.1.1 Zero flow results for Turbo-compressors

Four different turbo-compressors were studied at zero flow conditions to build up the understanding of the acoustical behavior of the combination of geometries (diffuser, rotor, and volute), Figure 29 shows the ViF zero flow test rig used for the measurements of these turbo-compressor units.

![ViF Zero flow test rig for Turbocharger measurements.](image)

Table 1 shows the basic geometries for the four tested turbo-compressors.

<table>
<thead>
<tr>
<th>Compressors</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>A/R</td>
<td>0.43</td>
<td>0.53</td>
<td>0.33</td>
<td>0.42</td>
</tr>
<tr>
<td>Trim</td>
<td>54</td>
<td>43</td>
<td>57</td>
<td>50</td>
</tr>
<tr>
<td>Inlet length (mm)</td>
<td>26</td>
<td>38</td>
<td>36</td>
<td>30</td>
</tr>
<tr>
<td>Inlet diameter (mm)</td>
<td>42</td>
<td>52</td>
<td>40</td>
<td>41</td>
</tr>
<tr>
<td>Rotor inducer (mm)</td>
<td>36</td>
<td>34</td>
<td>33</td>
<td>34.6</td>
</tr>
<tr>
<td>Rotor exducer (mm)</td>
<td>49</td>
<td>52</td>
<td>43.8</td>
<td>49</td>
</tr>
<tr>
<td>Rotor length (mm)</td>
<td>29</td>
<td>31</td>
<td>27</td>
<td>29</td>
</tr>
<tr>
<td>Rotor hub diameter (mm)</td>
<td>12</td>
<td>12</td>
<td>10</td>
<td>12</td>
</tr>
<tr>
<td>Diffuser plate diameter (mm)</td>
<td>89</td>
<td>93</td>
<td>76</td>
<td>83</td>
</tr>
<tr>
<td>Diffuser channel width (mm)</td>
<td>3.6</td>
<td>2.8</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Volute spiral diameter (mm)</td>
<td>95</td>
<td>96</td>
<td>82</td>
<td>91</td>
</tr>
<tr>
<td>Outlet length (mm)</td>
<td>49</td>
<td>54</td>
<td>44</td>
<td>150</td>
</tr>
<tr>
<td>Outlet diameter (mm)</td>
<td>32</td>
<td>43</td>
<td>30</td>
<td>38</td>
</tr>
</tbody>
</table>

Figure 30 to Figure 33 shows comparisons between the measured and simulated results for these configurations in terms of transmission loss and transfer functions.
Results

Figure 30: Comparison between measurements and Boost simulation for Turbo-compressor 1 a) TL b) TF.

Figure 31: a) Comparison between measurements and Boost simulation for Turbo-compressor 2 a) TL b) TF.

Figure 32: Comparison between measurements and Boost simulation for Turbo-compressor 3 a) TL b) TF.

Figure 33: Comparison between measurements and Boost simulation for Turbo-compressor 4 a) TL b) TF.
Results

There seems to be a relationship between the A/R ratio and the transmission loss. The local maxima in the frequency range of 800 – 2000 Hz have higher amplitudes and are shifting to lower frequencies with increasing A/R. The diffuser width also has some effect on the amplitude of the transmission loss. Comparing compressors 1 and 4, which have similar A/R values, the transmission loss of the latter is higher than the former. This could mean that the diffuser width has an effect on the amplitude of the transmission loss. A narrow diffuser width would give more losses.

4.1.2 Turbocharger results with mean flow

Two different turbocharger units were measured in the KTH CICERO turbocharger test rig the first turbocharger unit was with Fixed Geometry Turbine (FGT) and waste gate while the second Turbocharger unit was with Variable Geometry Turbine (VGT).

The measurements were performed at different operating points to cover most of the turbocharger compressor maps, for getting more understanding of the turbocharger compressor and turbine acoustical behavior at different operating conditions.

4.1.2.1 Turbocharger 1

4.1.2.1.1 Specifications

4.1.2.1.1.1 Compressor side

Table 2 shows the main specifications for the compressor side of turbocharger 1 according to the manufacturer data sheets.

<table>
<thead>
<tr>
<th>Spec.</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>A/R</td>
<td>0.53</td>
</tr>
<tr>
<td>D [mm]</td>
<td>96</td>
</tr>
<tr>
<td>Exducer [mm]</td>
<td>52</td>
</tr>
<tr>
<td>Inducer [mm]</td>
<td>34</td>
</tr>
<tr>
<td>Inlet Length [mm]</td>
<td>26</td>
</tr>
<tr>
<td>Inlet Diameter [mm]</td>
<td>44</td>
</tr>
<tr>
<td>Outlet Length [mm]</td>
<td>51</td>
</tr>
<tr>
<td>Outlet Diameter [mm]</td>
<td>43</td>
</tr>
</tbody>
</table>

Figure 34 shows the definitions of these specifications and Figure 35 shows the Turbocharger 1, compressor side operating maps with the operating points used in the measurements.

![Figure 34: Definitions of the specifications parameters of the compressor side.](image-url)
Results

Figure 35: Turbocharger 1, compressor operating map showing the measured operating points.

4.1.2.1.1.2. Turbine side

Table 3 shows the important specifications for the turbine side of turbocharger 1 according to the manufacturer data sheets.

Table 3: Turbocharger 1, Turbine side specifications

<table>
<thead>
<tr>
<th>Spec.</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>A/R</td>
<td>0.47</td>
</tr>
<tr>
<td>D [mm]</td>
<td>49</td>
</tr>
<tr>
<td>Exducer [mm]</td>
<td>36</td>
</tr>
<tr>
<td>Inducer [mm]</td>
<td>44</td>
</tr>
<tr>
<td>Inlet Length [mm]</td>
<td>82</td>
</tr>
<tr>
<td>Inlet Diameter [mm]</td>
<td>40</td>
</tr>
<tr>
<td>Outlet Length [mm]</td>
<td>61</td>
</tr>
<tr>
<td>Outlet Diameter [mm]</td>
<td>56</td>
</tr>
</tbody>
</table>

Figure 36 shows the definitions of these specifications while Figure 35 shows the Turbocharger 1, operating conditions which are related to the same operating conditions of the compressor side. The measurements were performed to achieve the same operating conditions on the map as for the compressor side.

Figure 36: Definitions of the specifications parameters of the turbine side.
Results

4.1.2.1.2 Results and discussion:

4.1.2.1.2.1 Compressor side

Figure 37 shows the measured acoustic transmission loss for Turbocharger 1 (FGT), compressor side for upstream and downstream directions. It shows that the acoustic transmission loss in the downstream direction is not significantly influenced by changing either the mass flow rate or the pressure ratio, while in the upstream direction the acoustic transmission loss is increasing with increasing mass flow rate and pressure ratio. Also the upstream direction transmission loss has a peak at high frequencies which is shifting to lower frequencies with increasing mass flow rates.

![Figure 37: Transmission loss for different operating points a) Upstream, b) Downstream.](image)

Figure 37: Transmission loss for different operating points a) Upstream, b) Downstream.

Figure 38 shows the measured transmission loss for turbocharger 1, compressor side at zero flow condition compared to Boost simulated result. As can be seen the agreement is good.

![Figure 38: Comparison measurements and Boost simulation at zero flow conditions OP1.](image)

Figure 38: Comparison measurements and Boost simulation at zero flow conditions OP1.

Figure 39 to Figure 45 show the measured acoustic transmission loss results for turbocharger 1, compressor side at different operating points compared to Boost simulated result, indicating a good match in the frequency range of interest for both the upstream and downstream case, especially in predicting the peak at high frequencies.
Results

Figure 39: Comparison between measured and simulated transmission loss OP2 a) Upstream, b) Downstream.

Figure 40: Comparison between measured and simulated transmission loss OP3 a) Upstream, b) Downstream.

Figure 41: Comparison between measured and simulated transmission loss OP4 a) Upstream, b) Downstream.

Figure 42: Comparison between measured and simulated transmission loss OP5 a) Upstream, b) Downstream.
Figure 43: Comparison between measured and simulated transmission loss OP6 a) Upstream, b) Downstream.

Figure 44: Comparison between measured and simulated transmission loss OP7 a) Upstream, b) Downstream.

Figure 45: Comparison between measured and simulated transmission loss OP8 a) Upstream, b) Downstream.

4.1.2.1.2 Turbine side:

Figure 46 shows the measured acoustic transmission loss for turbocharger 1, turbine side for the upstream and downstream directions. Similarly to at the compressor side the transmission loss in the downstream direction is not significantly influenced by changing either the mass flow rate or the pressure ratio, while in the upstream direction the transmission loss is increasing with increasing mass flow rate and pressure ratio. It also seems like there is a peak at high frequencies, but since the measurements only go up to 1600 Hz, it cannot be determined if it shifts due to the change in mass flow rate or pressure ratio.
Results

Figure 46: Transmission loss for different operating points a) Upstream, b) Downstream.

Figure 47: Show the measured transmission loss for turbocharger 1, turbine side at zero flow conditions, compared to Boost simulations. The agreement is good except in a region around 1200 and 1400 Hz.

Figure 47: Comparison between measurements and Boost simulations at zero flow conditions OP1.

Figure 48 to Figure 53 show the measured transmission loss for turbocharger 1, turbine side at different operating points compared to Boost simulations.

Figure 48: Comparison between measured and simulated transmission loss for OP2 a) Upstream, b) Downstream.
Results

Figure 49: Comparison between measured and simulated transmission loss for OP3 a) Upstream, b) Downstream.

Figure 50: Comparison between measured and simulated transmission loss for OP4 a) Upstream, b) Downstream.

Figure 51: Comparison between measured and simulated transmission loss for OP5 a) Upstream, b) Downstream.

Figure 52: Comparison between measured and simulated transmission loss for OP6 a) Upstream, b) Downstream.
Results

Figure 53: Comparison between measured and simulated transmission loss for OP7 a) Upstream, b) Downstream.

It can be seen from figures 52 and 53 that the simulation results is becoming noisy and deviates from the measurements for operating points 6 and 7. This could possibly be due to instabilities in the simulation model at high flow rates. It can also be seen that the transmission loss is over-predicted by about 3 dB in the upstream direction while the agreement is good in the downstream direction.

4.1.2.2 Turbocharger 2:

4.1.2.2.1 Specifications:

4.1.2.2.1.1 Compressor side

Table 4 shows the specifications for the compressor side of turbocharger 2 according to the manufacturer data sheets. Figure 34 shows the definitions of these specifications and Figure 54 shows the operating points included in the study.

<table>
<thead>
<tr>
<th>Spec.</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>A/R</td>
<td>0.42</td>
</tr>
<tr>
<td>D [mm]</td>
<td>89</td>
</tr>
<tr>
<td>Exducer [mm]</td>
<td>49</td>
</tr>
<tr>
<td>Inducer [mm]</td>
<td>35</td>
</tr>
<tr>
<td>Inlet Length [mm]</td>
<td>46</td>
</tr>
<tr>
<td>Inlet Diameter [mm]</td>
<td>44</td>
</tr>
<tr>
<td>Outlet Length [mm]</td>
<td>92</td>
</tr>
<tr>
<td>Outlet Diameter [mm]</td>
<td>38</td>
</tr>
</tbody>
</table>
Results

Figure 54: Turbocharger 2, Compressor side operating showing the measured operating points.

4.1.2.2.1.2 Turbine side

Table 5 shows the specifications for the turbine side of turbocharger 2 according to the manufacturer data sheets. Figure 55 shows the definitions of these specifications and Figure 56 shows the turbocharger 2 operating points. The measurements were performed to achieve the same operating conditions on the compressor map as for the compressor side, and at three different Guide vane angles defined as: minimum position (0%), middle position (50%), and maximum position (100%).

Table 5: Turbocharger 2, Turbine side specifications

<table>
<thead>
<tr>
<th>Spec.</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>A/R</td>
<td>0.47</td>
</tr>
<tr>
<td>D [mm]</td>
<td>49</td>
</tr>
<tr>
<td>Ducer [mm]</td>
<td>36</td>
</tr>
<tr>
<td>Inducer [mm]</td>
<td>44</td>
</tr>
<tr>
<td>Inlet Length [mm]</td>
<td>82</td>
</tr>
<tr>
<td>Inlet Diameter [mm]</td>
<td>40</td>
</tr>
<tr>
<td>Outlet Length [mm]</td>
<td>61</td>
</tr>
<tr>
<td>Outlet Diameter [mm]</td>
<td>56</td>
</tr>
</tbody>
</table>

Figure 55: Definitions of the specifications parameters.
4.1.2.2.2 Results and discussion:

4.1.2.2.1 Compressor side

Figure 56 shows the measured operating points for the compressor side of turbocharger 2. The measured operating points are shown in Figure 56. The pressure ratio is on the x-axis, and the corrected mass flow is on the y-axis. The measured points are indicated by green circles.

Figure 57: Transmission loss for different operating points a) Upstream, b) Downstream.

Figure 57 shows the measured transmission loss for the compressor side of turbocharger 2 at different operating points. The transmission loss is plotted against frequency for different operating points. The transmission loss is given in decibels (dB) on the y-axis, and the frequency is given in Hertz (Hz) on the x-axis. The transmission loss is plotted for different operating points (OP0 to OP6).

Figure 58 shows the measured transmission loss for the compressor side at zero flow conditions compared to boost simulations, indicating a good agreement.
Figure 58: Comparison between measurements and Boost simulation at zero flow conditions.

Figure 59 to Figure 64 show the measured transmission loss for turbocharger 2, compressor side at different operating points compared to Boost simulations.

Figure 59 Comparison between measured and simulated transmission loss for OP1 a) Upstream, b) Downstream.

Figure 60 Comparison between measured and simulated transmission loss for OP2 a) Upstream, b) Downstream.

Figure 61: Comparison between measured and simulated transmission loss for OP3 a) Upstream, b) Downstream.
In Figure 62 it can be seen that the upstream simulation results deviates a lot from the measured results. This could again be due to the instabilities of the simulation model at high flow rates. Otherwise there is a good agreement between measurements and simulations.

4.1.2.2.2 Turbine side

Figure 65 shows the measured transmission loss for turbocharger 2, turbine side for the upstream and downstream directions. The acoustic transmission loss in the downstream direction is again not significantly influenced by changing the mass flow rate or the pressure ratio, while in the upstream direction the acoustic transmission loss is increasing with increasing mass flow rate and pressure ratio. This is due to supplied mass flow rate and the change in guide vane position. As shown in Figure 65 (a) the lower the opening percentage
the higher the transmission loss at the same mass flow rate. Also, a peak at high frequencies is formed and is slightly shifted to higher frequencies with increasing mass flow rate.

![Figure 65](image1.png)

**Figure 65:** Comparison between different operating points a) Upstream and b) Downstream.

Figure 66 shows the measured transmission loss for turbocharger 2, turbine side at zero flow condition compared to Boost simulations. As can be seen the agreement is fairly good.

![Figure 66](image2.png)

**Figure 66:** Comparison between measured and simulated transmission loss for OP1 a) Upstream, b) Downstream.

Figure 67 to Figure 75 show the measured transmission loss for turbocharger 2 (VGT), turbine side at different operating points compared to Boost simulations.

![Figure 67](image3.png)

**Figure 67:** Comparison between measured and simulated transmission loss for OP1 and 0% Vane working condition a) Upstream, b) Downstream.
Figure 68: Comparison between measured and simulated transmission loss for OP1 and 50% Vane working condition a) Upstream, b) Downstream.

Figure 69: Comparison between measured and simulated transmission loss for OP1 and 100% Vane working condition a) Upstream, b) Downstream.

Figure 70: Comparison between measured and simulated transmission loss for OP2 and 0% Vane working condition a) Upstream, b) Downstream.
Results

Figure 71: Comparison between measured and simulated transmission loss for OP2 and 50% Vane working condition a) Upstream, b) Downstream.

Figure 72: Comparison between measured and simulated transmission loss for OP2 and 100% Vane working condition a) Upstream, b) Downstream.

Figure 73: Comparison between measured and simulated transmission loss for OP3 and 0% Vane working condition a) Upstream, b) Downstream.
Results

From the above results it is clear that the Boost model has a fairly good agreement with the measurements, even though there are some differences especially for higher flow speeds (OP3). The vane position also has an effect on the transmission loss.

Figure 74: Comparison between measured and simulated transmission loss for OP3 and 50% Vane working condition a) Upstream, b) Downstream.

Figure 75: Comparison between measured and simulated transmission loss for OP3 and 100% Vane working condition a) Upstream, b) Downstream.
4.2 Turbocharger results at hot conditions:

4.2.1 Diesel Engine:

The measurements were performed on a variable geometry turbocharged (VGT) Diesel engine with the following specifications:

1. Cylinders: 5-Cylinder in line.
2. Swept vol.: 2.4lit.
3. Bore: 81mm.
4. Stroke: 93.2mm.
5. Compression: 15.9.
6. Con. rod: 147mm.
7. Common rail injection system

The exhaust system was replaced with long tubes, while the intake system included an air box and a water cooled intercooler.

4.2.1.1 Compressor Side

Table 6 shows the specifications for the compressor side of the Diesel engine according to the manufacturer information. Some data were also determined by investigation of the turbo unit.

<table>
<thead>
<tr>
<th>Spic.</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>GT20V C101(52)</td>
</tr>
<tr>
<td>A/R</td>
<td>0.45</td>
</tr>
<tr>
<td>D [mm]</td>
<td>93</td>
</tr>
<tr>
<td>Exducer [mm]</td>
<td>52</td>
</tr>
<tr>
<td>Inducer [mm]</td>
<td>38.5</td>
</tr>
<tr>
<td>Inlet Length [mm]</td>
<td>50</td>
</tr>
<tr>
<td>Inlet Diameter [mm]</td>
<td>50</td>
</tr>
<tr>
<td>Outlet Length [mm]</td>
<td>42</td>
</tr>
<tr>
<td>Outlet Diameter [mm]</td>
<td>30</td>
</tr>
</tbody>
</table>

Figure 76 shows the Boost model for the intake line from the intake orifice to the outlet of the turbo-compressor. The measured dynamic and static pressure signals from the engine side at the turbo-compressor outlet was considered as the excitation signals for the Boost system.
Results

Figure 76: Diesel Engine Boost Model.

Figure 77 to Figure 84 show the Boost model results for the compressor side. Results are presented for four different engine speeds 1100, 2000, 3000, 4000 RPM and engine full load wide open throttle condition. First the excitation from the engine at the compressor output is shown and then the inlet results are compared to the measured pressure signals in both time and frequency (order) domain.

Figure 77: Engine excitation signal at 1100 Engine RPM.

(a) Measured inlet pressure (Pa) (Pa)
(b) Simulated inlet pressure (Pa)

Figure 78: Comparison between measured and simulated pressure signal at the compressor outlet and at 1100 Engine RPMs a) Time domain, b) Order domain.
Results

Figure 79: Engine excitation signal at 2000 Engine RPM.

Figure 80: Comparison between measured and simulated pressure signal at the compressor outlet and at 2000 Engine RPMs a) Time domain, b) Order domain.

Figure 81: Engine excitation signal at 3000 Engine RPM.

Figure 82: Comparison between measured and simulated pressure signal at the compressor outlet and at 3000 Engine RPMs a) Time domain, b) Order domain.
Results

Figure 83: Engine excitation signal at 4000 Engine RPM.

Figure 84: Comparison between measured and simulated pressure signal at the compressor outlet and at 4000 Engine RPMs a) Time domain, b) Order domain.

From the above results it is clear that the simulation model has a fairly acceptable agreement with the measurements especially at the dominant order (2.5 for a 5 cylinder engines).

4.2.1.2 Turbine Side

Table 7 shows the specifications for the turbine side of the Diesel engine according to the manufacturer information. Some data were also determined by investigation of the turbo unit.

Table 7: Diesel engine turbocharger unit (VGT), turbine side specifications

<table>
<thead>
<tr>
<th>Spic.</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>GT20V NS111(47) VNT</td>
</tr>
<tr>
<td>A/R</td>
<td>0.64</td>
</tr>
<tr>
<td>D [mm]</td>
<td>90</td>
</tr>
<tr>
<td>Exducer [mm]</td>
<td>39.5</td>
</tr>
<tr>
<td>Inducer [mm]</td>
<td>46</td>
</tr>
<tr>
<td>Inlet Length [mm]</td>
<td>40</td>
</tr>
<tr>
<td>Inlet Diameter [mm]</td>
<td>45</td>
</tr>
<tr>
<td>Outlet Length [mm]</td>
<td>50</td>
</tr>
<tr>
<td>Outlet Diameter [mm]</td>
<td>67</td>
</tr>
</tbody>
</table>

Figure 85 shows the Boost model for the exhaust line from the exhaust orifice till the inlet of the turbine. The measured dynamic and static pressure signals from the engine side at the turbo-turbine inlet were also here considered as the excitation signals for the Boost system.
Figure 85: Diesel Engine Boost Model.

Figure 86 to Figure 93 show the Boost model results for the turbine side showing the inlet of the turbine as excitation to the system and the outlet results are compared to the measured pressure signals and the sound pressure level at the outlet of the turbo-turbine. The comparisons are for four different engine speeds 1100, 2000, 3000, 4000 RPM and engine full load with wide open throttle condition.

Figure 86: Engine excitation signal at 1100 Engine RPM.

Figure 87: Comparison between measured and simulated pressure signal at the turbine inlet and at 1100 Engine RPMs
a) Time domain, b) Order domain.
Results

Figure 88: Engine excitation signal at 2000 Engine RPM.

Figure 89: Comparison between measured and simulated pressure signal at the turbine inlet and at 2000 Engine RPMs a) Time domain, b) Order domain.

Figure 90: Engine excitation signal at 3000 Engine RPM.

Figure 91: Comparison between measured and simulated pressure signal at the turbine inlet and at 3000 Engine RPMs a) Time domain, b) Order domain.
Results

Figure 92: Engine excitation signal at 4000 Engine RPM.

Figure 93: Comparison between measured and simulated pressure signal at the turbine inlet and at 4000 Engine RPMs a) Time domain, b) Order domain.

From the above results it can be seen that the agreement between measured and simulated results is fairly good for RPMs over 1000. The bad agreement at 1000 RPM may be due to measurements error at this particular RPM since it was difficult to reach stability during the measurements.
Results

4.3 CAT, DPF and Intercooler results

4.3.1 CAT results

Three catalytic converter (CAT) elements were used for acoustic measurements and comparison with simulation results.

4.3.1.1 CAT 1

Figure 94 shows CAT 1 and a cross section through its monolith, and Figure 95, shows the schematic diagram including dimensions.

Figure 94: a) CAT 1, b) CAT 1 cross section of monolith.

Figure 95: Schematic diagram for CAT 1.

Figure 96 shows CAT 1 connected to the ViF, zero flow test rig, using the two load method to perform the transfer matrix measurements. The test rig includes a loudspeaker for noise generation, the test object, two microphones before and two microphones after the element. Figure 97, shows a schematic diagram of the ViF zero flow test rig including the dimensions.

Figure 96: ViF, zero flow test rig with CAT 1.
Results

Figure 97: Schematic drawing including dimensions of the zero flow test rig.

Figure 98 shows the KTH mean flow test rig with CAT 1, using the two source method to perform the transfer matrix measurements. The test rig includes a set of loudspeakers at the inlet and other set at the outlet, the test object, three microphones before and three microphones after the CAT element. Figure 99, shows a schematic diagram of the KTH zero flow test rig including dimensions.

Figure 98: KTH, Mean Flow test rig construction for CAT_1 (cold condition).

Figure 99: Schematic drawing including dimensions of the KTH mean flow test rig.

Figure 100 shows the transmission loss measurements results without mean flow and with flow up to 0.1 Mach. It is clear that the flow does not have any significant effect in the studied frequency range.

The measurements of the transfer function were performed with open end boundary condition and zero flow.

Figure 100: CAT_1 Transmission Loss Measurements with/without mean flow up to 0.1 Mach.
Results

Figure 101 shows a comparison between the zero flow experimental transmission loss result and Boost simulation results using the model described in section 3.3.1. As can be seen the agreement is good at least up to around 1000 Hz.

![Figure 101: Comparison between transmission loss results for CAT 1 measured and simulated using Boost.](image)

Figure 102 shows a comparison between the zero flow experimental transmission loss result and linear acoustic simulation results using the model described in section 3.3.1, as can be seen the agreement is good also in this case at least up to around 1000 Hz. The variation in the simulated curves at approximately 1100 Hz could be due to the Herschel-Quincke effect [18].

![Figure 102: Comparison between transmission loss results for CAT 1 measured and simulated using the linear acoustic model.](image)

Figure 103 shows a comparison between measured and simulated transfer function, between microphones 3 and 4 without mean flow. Figure 104 shows the corresponding comparison with the linear acoustic model. As expected also these comparisons show a good agreement up to around 1000 Hz.
Figure 103: Comparison between transfer function results for CAT 1 measured and simulated using Boost.

Figure 104: Comparison between transfer function results for CAT 1 measured and simulated using the linear acoustic model.

Figure 105 shows a comparison between measured and simulated pressure drop across the CAT 1 element. The results show that the new two pipe Boost model, according to section 3.3.1, gives acceptable results also from the performance point of view.

Figure 105: CAT 1 pressure drop comparison between measurement and simulation using the new Boost model.
Results

4.3.1.2 CAT 2

Figure 106 shows CAT 2 and a cross section through its monolith, and Figure 107, shows a schematic diagram including dimensions.

![Figure 106: a) CAT 2, b) CAT 2 cross section.](image)

Figure 107: Schematic diagram for CAT 2.

Figure 108 shows the ViF, zero flow test rig with CAT 2. Figure 109, shows the schematic diagram for ViF zero flow test rig including dimensions.

![Figure 108: ViF, Zero Flow test rig with CAT 2.](image)

Figure 109: Schematic drawing including dimensions of the zero flow test rig.

Figure 110 shows the KTH, mean flow test rig construction with CAT 2.

![Figure 110: KTH, mean flow test rig with CAT 2.](image)
Results

Figure 111 shows the transmission loss measurements results without mean flow and with flow up to 0.1 Mach. It is clear that also in this case the flow does not have any significant effect in the studied frequency range.

![Figure 111: CAT 2 Transmission Loss Measurements with/without mean flow up to 0.1 Mach.](image)

Figure 112 a comparison between the zero flow experimental transmission loss result and Boost simulation results using the model described in section 3.3.1. As can be seen the agreement is good at least up to 1000 Hz.

![Figure 112: Comparison between transmission loss results for CAT 2 measured and simulated using Boost.](image)

Figure 113: Comparison between transmission loss results for CAT 2 measured and simulated using the linear acoustic model.

![Figure 113: Comparison between transmission loss results for CAT 2 measured and simulated using the linear acoustic model.](image)
Results

Figure 113 shows a comparison between the zero flow experimental transmission loss result and linear acoustic simulation results using the model described in section 3.3.1 as can be seen the agreement also in this case is good at least up to around 1000 Hz. The variation in the simulated curves at approximately 700 Hz could again be due to the Herschel-Quincke effect [18].

Figure 114 shows a comparison between measured and simulated transfer function, between microphones 3 and 4 without mean flow. Figure 115 shows the corresponding comparison with the linear acoustic model. As expected also these comparisons show fairly good agreement up to around 1000 Hz.

Figure 114: Comparison between transfer function results for CAT 2 measured and simulated using Boost.

Figure 115: CAT 2 linear Transfer Function New Boost Model.

Figure 116 shows a comparison between measured and simulated pressure drop across the CAT 2 element. The results again show that the new two pipe Boost model, according to section 3.3.1, gives acceptable results also from the performance point of view.
4.3.1.3 CAT 3

Figure 117 shows CAT 3 and a cross section through its monolith, and Figure 118, shows the schematic diagram including dimensions.

Figure 118: Schematic diagram for CAT 3.
Results

Figure 119 shows CAT 3 connected to the ViF, zero flow test rig and Figure 120, shows a schematic diagram including the dimensions.

![Figure 119: ViF, zero flow test rig construction for CAT 3 (cold condition).](image)

Figure 119: ViF, zero flow test rig construction for CAT 3 (cold condition).

Figure 120: Schematic drawing including dimensions of the zero flow test rig.

Figure 121 shows the schematic diagram for KTH mean flow test rig with CAT 3.

![Figure 121: Schematic drawing including dimensions of the KTH mean flow test rig.](image)

Figure 121: Schematic drawing including dimensions of the KTH mean flow test rig.

Figure 122 shows the transmission loss measurements results without mean flow and with mean flow up to 0.1 Mach. It is, also for this case, clear that the flow does not have any significant effect in the frequency range of interest.

![Figure 122: CAT 3 transmission loss measurements with/without mean flow up to 0.1 Mach.](image)

Figure 122: CAT 3 transmission loss measurements with/without mean flow up to 0.1 Mach.
Results

Figure 123 shows a comparison between the zero flow experimental transmission loss result and Boost simulation results using the model described in section 3.3.1. As can be seen the agreement is good.

Figure 123: Comparison between transmission loss results for CAT 3 measured and simulated using Boost.

Figure 124 shows a comparison between the zero flow experimental transmission loss result and linear acoustic simulation results using the model described in section 3.3.1 as can be seen the agreement also in this case is good at least up to around 1000 Hz.

Figure 124: Comparison between transmission loss results for CAT 3 measured and simulated using the linear acoustic model.

The peak at approximately 1200 Hz could be formed due to the Herschel-Quincke effect [18].

Figure 125 shows a comparison between measured and simulated transfer function, between microphones 3 and 4. Figure 126 shows the corresponding comparison with the linear acoustic model. As expected also these comparisons show a good agreement up to around 1000 Hz.
Results

Figure 125: Comparison between transfer function results for CAT 3, measured and simulated using Boost.

Figure 126: Comparison between transfer function results for CAT 3 measured and simulated using the linear acoustic model.

Figure 127: CAT 3 Pressure drop comparison between the measurements and the Boost models under cold conditions.
4.3.2 DPF results

Figure 128 shows the photos of the studied DPF units, and Figure 129, shows the KTH flow test rig setup.

![Figure 128: The studied DPF.](image)

Figure 129: a) Flow test rig setup, b) Schematic drawing including dimensions.

![Figure 129: Flow test rig setup, schematic drawing including dimensions.](image)

Figure 130 shows the result of the transmission loss measurements results without mean flow and with flow up to 0.1 Mach. The transmission loss is increasing with the increase in flow speed, in the frequency range up to approximately 300 Hz.

![Figure 130: DPF transmission loss measurements.](image)
Results

Figure 131 shows a comparison between the measured zero flow experimental transmission loss result and Boost simulation results using the model described in section 3.3.2. As can be seen the agreement is good in the complete frequency range of interest it can be noticed that the Boost simulation dose not succeed in modeling the increase at low frequencies seen in the experiment in Figure 130. This could be because the turbulence losses in narrow pipes are not modeled correctly [8].

![Figure 131: Comparison between transmission loss results for DPF measured and simulated using Boost.](image)

Figure 132 shows a comparison between the zero flow experimental transmission loss result and linear acoustic simulation results using the model described in section 3.3.2 as can be seen the agreement also in this case is good at least up to around 500 Hz. The variation in the simulated curves in the range 800 to 1200 Hz could be due to the Herschel-Quincke effect [18], which is due to the inability of applying the correct hydraulic diameter and the use of the dimensions as discussed in section 3.3.2.

![Figure 132: Comparison between transmission loss results for DPF measured and simulated using the linear acoustic model.](image)

Figure 133 shows a comparison between the measured and simulated transfer function, between microphones 3 and 4 without mean flow.
Figure 133: Comparison between transfer function results for DPF measured and simulated using the nonlinear acoustic model.

Figure 134 shows the corresponding comparison with the linear acoustic model. It shows some differences specially at low frequencies which may be due to the difficulty to apply the correct dimensions needed for the linear model.

Figure 135 shows a comparison between measured and simulated pressure drop across the DPF element. The results show that the new two pipe Boost model, according to section 3.3.2, gives acceptable results also from the performance point of view.
4.3.3 Intercooler Results

4.3.3.1 Axial inlet/outlet intercooler

Figure 136 shows a photo of the axial inlet/outlet Intercooler tested and Figure 137 shows this intercooler installed in the KTH flow test rig. Figure 138 shows a schematic diagram for the KTH test rig with the intercooler installed.

Figure 136: Axial inlet/outlet intercooler.

Figure 137: Axial inlet/outlet intercooler installed in the KTH test rig.

Figure 138: Schematic diagram for the axial inlet/outlet intercooler installed in the KTH test rig.

Figure 139 shows the transmission loss measurements results without mean flow and with flow up to 0.1 Mach. It is clear that the flow gives an increase with 3-4 dB for the transmission loss in the studied frequency range.
Results

Figure 139: Axial inlet/outlet Intercooler transmission loss measurements with/without mean flow up to 0.1 Mach.

Figure 140 to Figure 142 show a comparison between the zero and mean flow experimental transmission loss result and Boost simulation results using the model described in section 3.3.3. As can be seen the agreement is very good.

Figure 140: Comparison between transmission loss results for axial inlet/outlet intercooler measured and simulated using Boost at Zero mean Flow.

Figure 141: Comparison between transmission loss results for axial inlet/outlet intercooler measured and simulated using Boost at 0.05 Mach mean Flow.
Results

Figure 142: Comparison between transmission loss results for axial inlet/outlet intercooler measured and simulated using Boost at 0.1 Mach mean flow.

Figure 143 shows a comparison between measured and simulated transfer function, between microphones 3 and 4 without mean flow.

Figure 144 shows a comparison between measured and simulated pressure drop across the axial inlet/outlet intercooler element. The results show that the new two pipe Boost model, according to section 3.3.3, gives acceptable results also from the performance point of view.

Figure 144: pressure drop comparison between measurement and simulation using the new Boost model measurements for the axial inlet/outlet intercooler.
4.3.3.2 Non Axial inlet/outlet intercooler

Figure 145 shows a photo of the non-axial inlet/outlet intercooler studied and Figure 146 shows three tubes which were cut from that intercooler for measuring the influence of the tubes only without the inlet outlet collectors. Both the full intercooler and the three tube configurations were installed in the KTH flow test rig see Figure 147 and Figure 148.

Figure 149 shows a schematic diagram for the KTH test rig including the main dimensions of the test setup.

Figure 145: Non axial inlet/outlet intercooler.

Figure 146: Three tubes from the non-axial inlet/outlet intercooler with added axial inlet/outlet connections.

Figure 147: The non-axial inlet/outlet intercooler in the KTH mean flow test rig.

Figure 148: The three tubes from the non-axial inlet/outlet intercooler in the KTH mean flow test rig.
Figure 149: Schematic drawing for the flow tests on the KTH mean flow test rig.

Figure 150 and Figure 151 show the transmission loss measurements for both the full intercooler and the three tube configuration. It can be seen that for the full intercooler, the transmission loss increases at low frequencies with increasing flow speed. For the three tube configuration the transmission loss increase up to 5 dB with flow for frequencies where there is a minimum for the no flow case. This means that the inlet/outlet collectors have a large influence on the acoustic transmission loss.

Figure 150: Non axial inlet/outlet intercooler transmission loss at three different mean flow speeds 0, 0.05, and 0.1 Mach.

Figure 151: Transmission Loss for the three tube configuration of the non-axial inlet/outlet intercooler at two different mean flow speeds 0 and 0.02 Mach.

Figure 152 and Figure 153 are showing a comparison between the zero and mean flow experimental transmission loss result and Boost simulation results, for the three tube configuration, using the model described in section 3.3.3. As can be seen the agreement is fairly good.
Results

Figure 152: Comparison between the simulated and measured transmission loss for the three tube configuration at Zero mean flow.

Figure 153: Comparison between simulated and measured transmission loss for the three tube configuration at 0.02 Mach mean flow.

Figure 154 shows a comparison between measured and simulated pressure drop across the axial inlet/outlet three tube intercooler element. The results show that the new two pipe Boost model, according to section 3.3.3, gives acceptable results also from the performance point of view.

Figure 154: Comparison between the simulated and the measured pressure drop for the three tube configuration at cold conditions.

Figure 155 to Figure 157 are showing a comparison between the zero and mean flow experimental transmission loss result and Boost simulation results, for the complete intercooler, using the model described in section 3.3.3. As can be seen the agreement is fairly good.
Figure 155: Comparison between simulated and measured transmission loss for Intercooler element using Boost at zero mean flow.

Figure 156: Comparison between simulated and measured transmission loss for Intercooler element using Boost at 0.05 Mach mean flow.

Figure 157: Comparison between simulated and measured transmission loss for Intercooler element using Boost at 0.1 Mach mean flow.

Figure 158 shows the comparison between the measured transfer function and the Boost simulation without mean flow.
Figure 158: Comparison between simulated and measured transfer function for intercooler element using Boost at zero mean flow.

Figure 159 shows a comparison between measured and simulated pressure drop across the axial inlet/outlet intercooler element. The results confirm that the new two pipe Boost model, according to section 3.3.3, gives acceptable results also from the performance point of view.

Figure 159: Comparison between simulated and measured pressure drop for Intercooler element using Boost.
CHAPTER 5

Conclusions
In this chapter conclusion regarding the outcome of experimental studies and simulations are summarized.

5.1 Conclusions for turbocharger studies

Transmission loss measurements under zero-flow conditions (section 4.1.1) showed that the compressor is acoustically transparent in the frequency region below 800 Hz with local maxima in the higher frequency range. A relationship between the area to radius ratio A/R value and the transmission loss was observed, indicating that the transmission loss maxima had higher levels and shifted to lower frequencies with increasing A/R value. A smaller diffuser width also increases the transmission loss.

Results from downstream transmission loss measurements under operating conditions (section 4.1.2) were essentially independent of the flow conditions below 1400 Hz and the upstream transmission loss formed a distinct maximum that shifted with increasing flow velocity and pressure ratio. The damping effect of the compressor on incident sound waves was found to be asymmetric and depends on the propagation direction. The upstream direction exhibited higher transmission loss values than the downstream direction.

A nonlinear time-domain gas dynamic acoustic model based on the compressor geometry (section 4.1.2.1.1) was successfully developed and verified under zero-flow conditions. The model was adapted for flow in operating conditions and the simulated results were in reasonable agreement with the measured results for both downstream and upstream directions. However, there are some discrepancies below 400 Hz for the upstream results.

The acoustic model of the compressor can be applied for the turbine side as the geometrical profiles of the two units are similar but with opposite flow directions. Extra components must be taken into consideration, like the waste gate and possibly also the guide vanes of a variable geometry turbocharger. A more detailed experiment has been made where parameters like the pressure ratio and mass flow were studied in order to individually identify the effect on the acoustical damping and it was found that the flow speed has the largest effect on the transmission loss followed by the pressure ratio.

The new turbo system model was, together with the new models for other components such as the intercooler, used to create a complete model of the automotive intake and exhaust system for both diesel and gasoline engines (section 4.2). With this full model, the pressure waves propagated from the engine through the intake and exhaust system and the sound pressure levels produced were simulated showing a fairly good agreement with measurements.

5.2 Conclusions for CAT, DPF and Intercooler studies

The use of flow resistivity to represent the wall sound absorption for the narrow tubes in catalytic converters (CAT), Diesel particle filters (DPF) and also on intercoolers (section 3.3) was effective, and gave a good match between simulations and measurements.

For simulating both of the CAT and DPF elements, there was a need for separating the calculated flow variables leading to a model using two parallel pipes based on the main CAT cross sectional area (section 3.3). This solution was successfully in the frequency range studied. It could reproduce the transmission loss of the simulated elements with a good agreement.
agreement to the measurement results up to 1000 Hz, for three different catalytic converters and Diesel Particulate filter (sections 4.3.1 and 4.3.2).

In addition to that, the new model has been validated with the measured pressure drop and showing that it can predict the pressure drop across the CAT element with a good accuracy (sections 4.3.1 and 4.3.2).

Regarding the Intercooler element the same procedure as for the CAT and DPF were used, but with lower flow resistivity, because the narrow tubes forming the intercooler channels are bigger than those of the CAT or the DPF. But finally it leads to the two parallel pipe models for the intercoolers, which gave a fairly good agreement with the measured results for two different intercoolers (section 4.3.3).

5.3 Future Work

The next planned future work will be to focus on the improvement of the turbocharger models especially in the rotor part at subsonic and supersonic rotor tip speeds; to be able to simulate the active acoustic behavior of the real turbocharger units.

Work on building analytical models to compute the acoustic properties of turbochargers in both active and passive parts. these models should predict the different noise patterns of the turbocharger unit on both sides (turbine and compressor) such as the tonal noise at subsonic rotor blade tip speeds and the buzz saw noise at supersonic rotor blade tip speeds Also it should include detailed modeling of the aerodynamic characteristics of the rotor, taking into account the effect of the rotor-casing clearance and rotor blade passing frequencies that could influence the acoustic behavior.

To study the vibro-acoustic influence of the turbine side (driving side) on the compressor side and the radiated noise from both sides of the turbocharger unit into the passenger compartment, considering different types of turbochargers (like VGT and FGT) and under different operating engine conditions (Gasoline and Diesel).
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