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Experimental Methods for Sound Propagation Studies in Automotive Duct Systems

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Declaration:
Hereby I declare that this doctoral thesis, my original investigation and achievement, submitted for the doctoral degree at Tallinn University of Technology has not been submitted for any other degree or examination

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Abstract

This work deals with experimental methods for sound propagation studies in automotive duct systems.

The use of active one-port models has been tested to characterize an air terminal device as a source of flow generated noise.
In order to provide a prediction of noise generation at different operating points for the device a scaling law was derived and verified. In the experimentally derived scaling law a flow speed dependence of 3 was found for the narrow band spectra, corresponding to a dipole-like behavior of the source in the plane wave range. This technique was found to predict well the in-duct sound produced by an air terminal device.

A new method based on the well-known two-load technique was developed and used to characterize the source data of various piston-engines with non-linear behavior. The method was validated with numerical simulation results, and with experimental data from modified compressor and 6-cylinder turbocharged truck diesel engine. The new non-linear indirect source characterization technique proposed requires one additional acoustic load compared to the two-load technique. Since over-determination is anyway used in many cases the additional data would often be available. The new non-linear multi-load technique gave improved results when the source was slightly non-linear. For cases when the source is linear and time-invariant the new technique gives the same result as the two-load technique. It can therefore be recommended that the new technique is used whenever sufficient data is available.

An overview of the existing research in turbocharger acoustics has been presented. Based on the published work in the field the main noise problems seem to be associated with the turbo-compressor outlet side. The dominating aerodynamic noise generating mechanisms in turbocharger compressors are: tonal noise at blade passing frequencies, “buzz-saw” noise and blade tip clearance noise. Additionally, a description of the novel test-rig for acoustic characterization of automotive turbochargers, designed for the new gas management research centre, KTH CICERO, has been given.

The sound reflection from hot flow duct openings has been investigated experimentally and by using a FEM simulation. The obtained reflection coefficient results for flow temperatures up to 500 ºC have been compared with famous Munt’s theory. It was demonstrated that at low Mach number and Helmholz number cases the results agree well with the Munt’s model. This is a first experimental validation of the theory for hot flow conditions.

Keywords: Acoustic source, one–port, source model, duct termination, reflection coefficient, source strength, IC-engine, turbocharger, flow duct, multi-load method, non-linear, FEM.

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H. Rämmal

Hans Rämmal, June 2007

CONTENTS

INTRODUCTION

A dominating contribution of automobile induced noise is often originated from its duct systems. The sources related to engine gas exchange systems, including inlet and exhaust systems, are the primary sources for exterior noise up to highway speeds while the aero-acoustical noise from ventilation and climate control system pipeline elements is crucial for acoustical quality of the passenger compartment. Investigations on sound propagation studies in automotive duct systems, with the aim to develop noise control techniques, is therefore of global interest.

To understand noise generation mechanisms and for effective application of noise control measures it is essential to characterize noise sources. For characterization of in-duct acoustic sources, it is often necessary to create source models with parameters derived from experiments.

Despite of advanced capabilities in virtual testing and analytical methods experimental techniques are extensively used in acoustic modeling and characterization of noise sources in ducts. For several modern IC-engine inlet and exhaust system components the complete and accurate source characterization is possible only by experiments, primarily due to the complexity of the source mechanisms.

In this thesis experimental techniques for characterizing in-duct aero-acoustic and fluid machine sources are mainly treated. A complete acoustic source model gives information how much sound the source delivers into any receiving system. The source data is defined via the physical quantities by which the source interacts with the receiving system.

In developing source models one is looking for the simplest model that is able to provide acceptable results. For the low frequency (plane wave) region cases studied in this thesis, an acoustic one-port source model is applicable. The one-port model is a linear time-invariant model. More complicated models include in order of increasing complexity: linear time-varying, hybrid and non-linear models.

The aim of the present dissertation is:

- To test the application of active one-port source models for characterization of flow generated noise sources in automobile duct systems,
- To investigate a generation mechanism in air terminal devices and to derive a scaling law for prediction of noise generation from similar devices at chosen operating points,
- To modify the classical two-load method for improved characterization of the non-linear acoustic one-port sources,
- To investigate the influence of parameters controlling the linearity of the IC-engine,
- To investigate the acoustical properties and research done on acoustics of turbochargers together with the description of the dominating sound generation mechanisms and passive acoustic effects of the turbocharger,
- To determine the measurement schema and to develop an experimental setup for acoustic characterization of turbochargers
- To experimentally investigate the plane acoustic wave reflection for hot jet flow from duct opening and to validate the existing theory for hot flow conditions,
- To investigate the sound propagation through open duct termination in hot flow conditions using FEM simulation.

It must be noted here that the experimental methods and measurement techniques developed are intended to be general and therefore applicable for similar noise problems in other vehicles and fluid machines.

LIST OF PUBLICATIONS

The dissertation is based on the following papers, which are referred in the text by their Roman numerals I-V.

I. Hans Rämmal, Mats Åbom, 2007, Characterization of Air Terminal Device Noise Using Acoustic 1-Port Source Models, *Journal of Sound and Vibration*, Vol. 300**,** pp. 727-743.

II. Hans Rämmal, Hans Bodén, 2007, Modified Multi-Load Method for Non-Linear IC-Engine Source Characterization, *Journal of Sound and Vibration,* Vol. 299, pp. 1094-1113.

III. Hans Rämmal, Mats Åbom, 2007, Acoustics of Turbochargers, *Society of Automotive Engineers (SAE) Technical Paper Series*, Paper no. 2007-01-2205.

IV. Hans Rämmal, Jüri Lavrentjev, 2007, Sound reflection at an open end of a circular duct exhausting hot gas, *International Noise Control Engineering Journal (INCE)* (Submitted). Also in *Proceedings of the 35th International Congress on Sound and Vibration (INTER-NOISE 2006)*, Honolulu, USA, 3-6 December, 2006.

V. Hans Rämmal, Jüri Lavrentjev, 2007, Determination of acoustic properties for open duct termination in hot jet conditions using FEM simulation, *Proceedings of the 14th International Congress on Sound and Vibration (ICSV 14)*, Cairns, Australia, 9-12 July, 2007.

Other publications (not included in the thesis)

VI. Hans Rämmal, Jüri Lavrentjev, 2004, A Method for Experimental Determination of Source Impedance in Helium-Air Mixture, *Proceedings of the 33th International Congress on Sound and Vibration (INTER-NOISE 2004)*, Praque, Czech Republic, 22-25 August, 2004, CD-ROM.

VII. Hans Rämmal, Jüri Lavrentjev, 2005, Experimental Determination of Acoustic Reflection Coefficient of Duct Openings in Simulated Hot Conditions, *Proceedings of the 34th International Congress on Sound and Vibration (INTER-NOISE 2005)*, Rio de Janeiro, Brazil, 7-10 August, 2005, CD-ROM.

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

1. 10th International Congress on Sound and Vibration (ICSV 10), Stockholm, Sweden, 7-10 July, 2003.

2. 11th International Congress on Sound and Vibration (ICSV 11), St. Petersburg, Russia, 5-8 July, 2004.

3. 33th International Congress on Sound and Vibration (INTER-NOISE 2004), Prague, Czech Republic, 22-25 August, 2004.

4. 34th International Congress on Sound and Vibration (INTER-NOISE 2005), Rio de Janeiro, Brazil, 7-10 August, 2005.

5. 35th International Congress on Sound and Vibration (INTER-NOISE 2006), Honolulu, USA, 3-6 December, 2006.

6. International Noise and Vibration Conference 2007, Society of Automotive Engineers (SAE), Illinois, USA, 15-17 May, 2007.

7. 14th International Congress on Sound and Vibration (ICSV 14), Cairns, Australia, 9-12 July, 2007.

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1. "Aerodynamic research on Turbocharged Engine Inlet and Exhaust Systems" – ARTEMIS, funded by EC (European Commission), contract number G3RD-CT-2001-0511. (Paper II, IV and VII)
2. "Measurement methods of industrial and mobile noise sources and development of measurement set-up" Grant nr 5622 from the Estonian Science Foundation, 2003-2004 (Paper IV and VI)
3. "Development of measurement methods for non-stationary flow noise" Grant nr 6557 from the Estonian Science Foundation, 2006-2008 (Paper IV and V)

1 ACOUSTIC SOURCE MODELS

1.1 Linear time-invariant source model

If only plane waves are considered in the duct system the simplest model that can be used to describe the source is the linear time-invariant frequency domain one-port model. If there is only one degree of freedom at the interface between the source and the system the one-port source models can be used. For in-duct fluid-borne sound sources this corresponds to cases where there is a plane wave state in the connected duct. In-duct sources normally have at least two openings which means that it further requires that the external acoustic load only can vary at one of the openings, or the openings are acoustically uncoupled from each other so that they can be treated separately.

In the frequency domain an acoustic one-port can be completely described by two complex parameters: the source strength p_+^S and the source reflection coefficient R_S (or alternatively the source impedance). The behavior of the one-port (see Fig. 1) can in the frequency domain, be described by [1]:

$$p_+ = R_S p_- + p_+^S , \tag{1}$$

where p_- and p_+ are traveling acoustic pressure amplitudes, R_S is the source reflection coefficient at cross-section, where *x=0* (see Fig. 1), and p_+^S is the source strength. The source strength p_+^S can be interpreted as the pressure generated by the source-side when the system is reflection free.

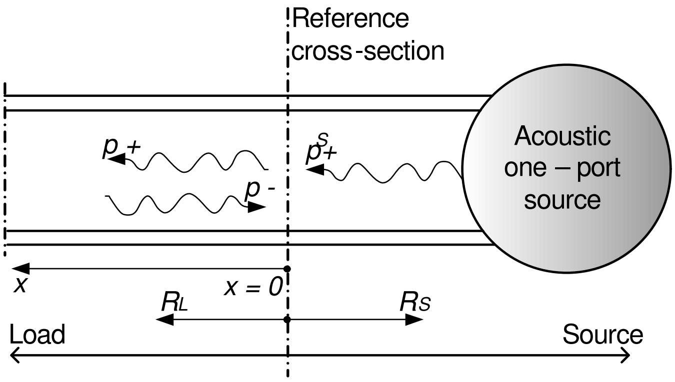


Figure 1. An in-duct source modeled as an acoustic 1-port

In the literature the source model for one-ports is often expressed in terms of source strength p_S and normalized source impedance Z_S

$$p = p_S - Z_0 Z_S q , \tag{2}$$

where p_S is the source pressure, p and q are acoustic pressure and volume velocity, respectively, and Z_0 is the characteristic impedance of the fluid. The source impedance Z_S represents the acoustic impedance seen from the reference cross-section towards the source.

Fig. 2 shows the equivalent acoustic circuit for a linear time invariant source. In this figure p_L and Z_L denote the acoustic load data (the load pressure and the load impedance), while p_S, q_S and Z_S denote the source data respectively. Theoretically the two representations of the source shown in Fig. 2 are equivalent and it is possible to go from one representation to the other by using the relationship $q = p_S / Z_S$. If there are errors in the experimental data or deviations from system linearity it can be expected that the error propagation is different for the two representations leading to different results when source data is extracted using over-determination. Extracting source data using both formulations and comparing the resulting source impedance is a possibility to see if experimental data are in agreement with linear time-invariant source model. It can also be expected that if the source is close to a constant velocity source this model will give smaller errors than if a pressure source model is applied and vice versa.

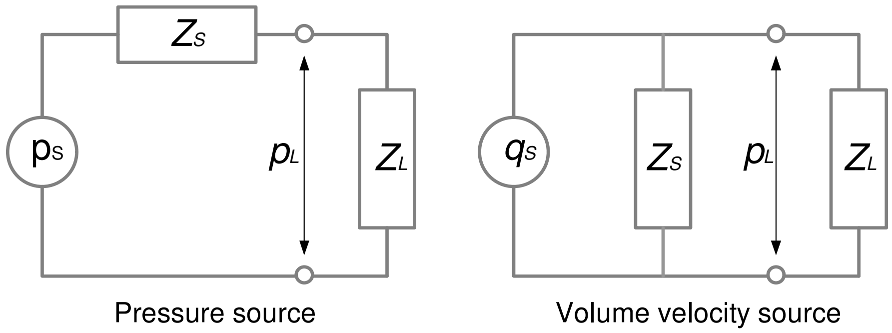


Figure 2. Equivalent acoustic circuits for linear time invariant source

The linear time-invariant equivalent source model will strictly be applicable only in situations where the pressure-fluctuations are small. Several authors however have found the linear time-invariant model to give reasonable results for modeling the systems with relatively large pressure fluctuations, *i.e.*, slightly non-linear systems.

Paper I represents an effort to use 1-port models to characterize flow generated sound which works as long as the source process is unaffected by the acoustic field. This kind of assumption is consistent with the basic assumption used when writing down the source term in Lighthills well-known equation. Therefore it holds in many cases with the exception of situations where strong flow-acoustic feedback occurs, *e.g.*, whistles. In **Paper I** the use of this approach has been tested and proven for a flow constriction mounted at a duct opening (see Fig 3).

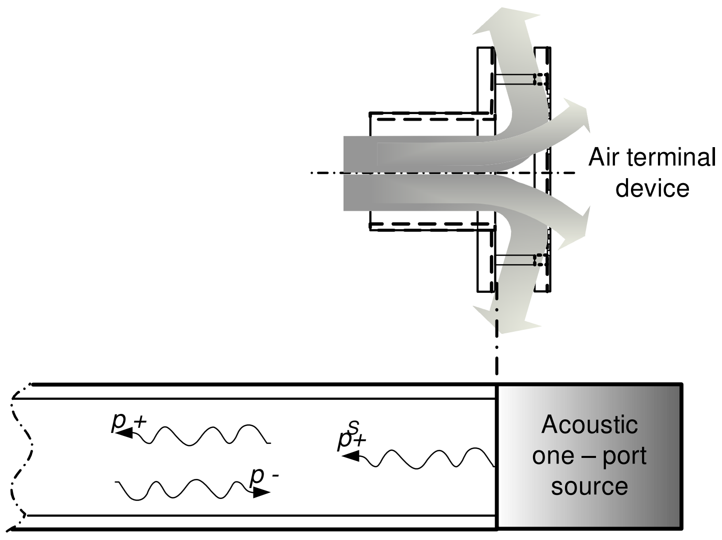


Figure 3. A schematic representation of the air terminal device characterized as acoustic one-port.

In earlier works the approach has also been successfully applied to in-duct elements that can be described as active 2-ports such as bends [2, 3]. The use of 1- and 2-port models is of importance for modeling the low frequency plane wave range in duct systems. This range is of particular interest to study resonance and standing wave effects that significantly can affect the acoustic output.

To enable source data measured in **Paper I** at one operating condition of the air terminal device to be used at other operating points (flow speeds) a scaling law is necessary. As observed in earlier investigations, *e.g.* [4], the passive part of the source data is often weakly dependent on the flow speed. Therefore it is mainly of interest to find a scaling law for the source strength. Based on earlier works, *e.g.*, Nelson & Morfey [5] and Nygård [2], the in-duct plane wave sound power produced by flow separation from a compact source region will scale as

$$\frac{G_{SS}}{\rho c} = \rho U^3 M^{\alpha} F(St)\,. \tag{3}$$

Where G_{SS} is the source strength, U is the mean flow speed, ρ is the density, α depends on the aero-acoustic source type, *F(St)* is a dimensionless source-spectrum depending on a Strouhal number, $St = fd/U$, f is the frequency and d is the duct diameter.

In Fig. 4, the collapse of all the source strength data is shown, based on the scaling law in Eq. (3) with α=*0*, which was found to give the best fit. The scatter (''black region'') around the best fit points is a measure of the standard deviation, which is of the order 3 dB. The α value found will create a flow speed exponent of 4 when integrated over all frequencies or over a frequency band with constant relative band-width, *e.g*., a 1/3-octave band. This value corresponds to a ducted aero-acoustic dipole in the plane wave range [5].

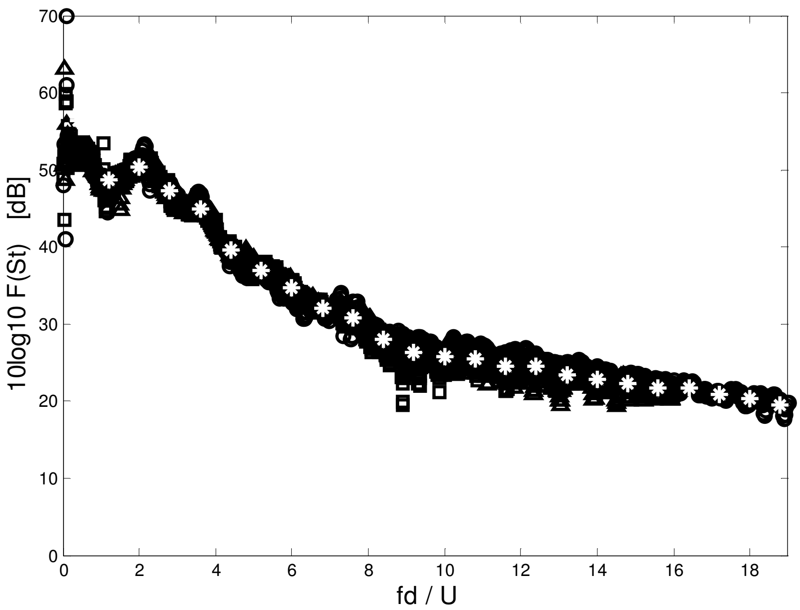


Figure 4. Air terminal device: data collapse for the Strouhal number-dependent source spectrum F in *eg*. (3);
The source strength based on in-duct measurements, U=8.2 m/s (Circles);
The source strength based on in-duct measurements, U=12.3 m/s (triangles);
The source strength based on in-duct measurements, U=16.4 m/s (rectangles);
The best fit source model used in the validation (white stars).

To validate the source model, including the scaling law for the source strength, a modified duct was used in **Paper I**. In Fig. 5, example for the prediction of the pressure cross-spectrum inside the validation duct is shown. As can be seen the agreement with measurements is good for the in-duct case with a deviation of typically less than 2 dB up to about 1000 Hz.

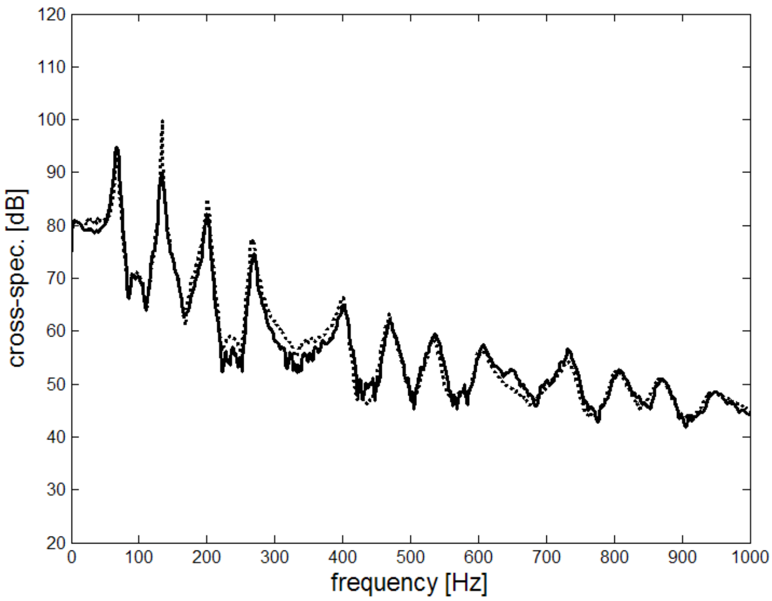


Figure 5. The cross-spectrum of the air terminal device with 7.0 m/s flow velocity in validation duct system; measurements (full-line), prediction (dotted-line).

1.2 Linear time-varying model

When determining the passive acoustic data, *i.e.*, the source impedance Z_S of a fluid machine it is assumed that it does not change with time. If the operating machine is studied it is observed that various parts, such as pistons, valves, or fan blades move. Therefore it can be expected that even the passive acoustic properties should be time-varying. In mathematical terms this means that the source is described by linear differential equations with time-varying coefficients. The time variation in the coefficients is normally caused by the periodic motion of the machine and will therefore be periodic [1].

A frequency domain linear time-varying source model was developed by Wang [6-7] for an internal combustion engine inlet system. By assuming that the variables and the coefficients have periodic time dependence, so that they can be expanded in Fourier series, a frequency domain model for the source can be deduced. Here the source strength is replaced by a vector containing the data for each frequency component, and the source impedance is replaced by a matrix which also describes the coupling between different frequency components which can occur at the source. Bodén [8] presented a measurement method for determining the source data for such a model. The method used is similar to the multi-load methods used for time-invariant one-port sources.

In [6] it was found when comparing the experimental sound pressure levels with analytical ones that time-varying model gave better prediction than time-invariant for automobile 4-stroke engine inlet system.

1.3 Hybrid models

The hybrid linear/non-linear method, where a non-linear time domain model is used for the source and a linear frequency domain model is used for the receiving system, was introduced for in-duct sources in [9].

Generally one can say that the hybrid approach is the attempt to combine the linear and the non-linear techniques. The main idea is to retrieve the advantages of both types of methods. The harmonic balance (HBM) technique [10] is an alternative frequency domain technique with better convergence properties. The main conclusions, after implementing it on simple 1-cylinder “cold” engine model, were that the harmonic balance method was preferable for harmonic steady state simulations where parametric studies are performed. As the HBM method is a steady state method, true transient behavior cannot be modeled correctly. The 1-

cylinder "cold" engine model following the HBM method was used in **Paper II** to validate the new source characterization technique.

The hybrid methods can be divided into a number of main groups. One group is the iterative techniques which can be further subdivided into frequency domain iterative techniques [9, 10] and time domain iterative techniques [11] depending on in which domain the convergence check and the coupling is performed. For applications to IC-engine exhaust systems the frequency domain iterative method was suggested by Jones [12] and tested by Bodén [8] for a modified compressor with unstable results.

Another group is the convolution techniques where the frequency domain boundary condition for impedance is transformed into the time domain. There are works on IC-engines exhaust and inlet systems where the convolution technique using the reflection function [13] or scattering matrix [14] has been used.

1.4 Non-linear model

Many fluid machines such as compressors and IC-engines generate high sound pressure levels or high flow velocities and are therefore considered as high level acoustic sources. The validity of modeling them as linear time-invariant systems may therefore decrease accuracy of the results. It has been noted [15] that a linear time-invariant source model when experimentally determining the source data of high level sources frequently gives unphysical negative source resistance values. For a linear time-invariant passive system the real part of the impedance must be positive since this shows the direction of energy. Energy can only be lost into the system – it cannot be created since the system is passive. For a non-linear system energy can be transferred from one frequency to another. This could at certain frequencies, suggest that the system was no longer passive by giving a negative real part of the impedance. Linear time varying system can also produce this negative resistance as shown by Peat and Ih [16].

Therefore as an alternative to linear techniques, non-linear models can be used to describe the complete system, see, *e.g.*, Jones [12]. Non-linear methods are often used for systems with high sound pressure levels or when clearly non-linear effects are present.

The non-linear time-domain methods are based on numerical simulations of the unsteady flow. The advantage, compared to the linear description, would be that the system is more correctly modeled. The results agree generally well with experimental results but the methods are time consuming. These methods also require a good knowledge of engine modeling, such as the combustion process, mechanics and timing of the valve movement, exact knowledge of the system geometry, temperature and so on.

To improve the described source characterization methods, especially for applications where non-linear effects are expected, such as for IC-engines, a technique was suggested by Jang and Ih [17]. The idea was to include non-linear effects in the direct methods for source impedance determination. This method was suggested without showing any experimental results and without making it clear how the time domain volume velocity $q(t)$ would be obtained from experiments.

In **Paper II** the method has been modified for applications with indirect or multi-load methods. The time domain representation of the source model with non-linear term is described by

$$\int z_S(\tau)q(t-\tau)d\tau + \int h_S(\tau)b(t-\tau)d\tau = p_S(t) - p(t), \tag{4}$$

where $p(t)$ and $q(t)$ denote the pressure and volume velocity at the source cross section, $z_S(t)$ is the time domain representation of the source impedance, $p_S(t)$ is the source strength, b(t) is the non-linear input and $h_S(t)$ is the source data coefficient for the non-linear part. When applying this technique in section 2 it has been assumed that $b(t) = q^3(t)$ which is the first higher order series expansion term obtained for the pressure drop over an orifice. It can be expected that the main type of non-linearity for many applications will be caused by the flow through a constriction characterized by the pressure difference over the constriction being equal to $\Delta p(t) = \left(\frac{\rho_0}{2S^2}\right) \cdot q(t) \cdot |q(t)|$. It is shown in [18] that, under the assumption that $q(t)$ follows a zero mean Gaussian distribution, the third-order polynomial least-squares approximation to $\left(\frac{\rho_0}{2S^2}\right) \cdot q(t) \cdot |q(t)|$ is

$$p(t) = \frac{\rho_0}{2 \cdot S^2} \cdot \left[\left(\sigma_q \cdot \sqrt{2/\pi} \right) \cdot q(t) + \left(\frac{\sqrt{2/\pi}}{3 \cdot \sigma_q} \right) \cdot q^3(t) \right], \tag{5}$$

where σ_q is the standard deviation of $q(t)$. Taking the Fourier transform of (5) gives

$$P(f) = \frac{\rho_0}{2 \cdot S^2} \cdot \left[\left(\sigma_q \cdot \sqrt{2/\pi} \right) \cdot Q(f) + \left(\frac{\sqrt{2/\pi}}{3 \cdot \sigma_q} \right) \cdot Q_3(f) \right], \tag{6}$$

where $P(f)$, $Q(f)$ and $Q_3(f)$ are the Fourier transforms of $p(t)$, $q(t)$ and $q^3(t)$. The original "square-law system with sign" $\left(\frac{\rho_0}{2S^2}\right) \cdot q(t) \cdot |q(t)|$ can therefore to the third order be replaced by a linear system in parallel with a cubic system.

In the frequency domain Eq. (4) can be formulated as

$$P_S Z_0 Z - P Z_0 Z_S - H_S B = PZ \tag{7}$$

where H_S and B are the Fourier transforms of $h_S(t)$ and $b(t)$. This equation has compared to Eq. (2) a third complex unknown H_S, which means that now at least three acoustic loads will have to be used in order to solve the equation and to obtain the source data.

2 SOURCE CHARACTERIZATION METHODS

A number of different methods exist for determining acoustic source data from experiments. An overview of the state of the art of experimental methods for determining the 1-port source data for in-duct fluid-borne sound sources was described in the review papers [1] and [19]. The measurement methods can be divided into direct (with an external source) [20] and indirect or multi-load methods (without an external source) [21].

2.1 Direct methods

The direct methods are two-step methods. First, the passive source data *e.g.*, source reflection coefficient, is determined using the external source and the two-microphone technique [22] (see Fig. 6). Then with the external source off or removed a known acoustic load is applied to the source and the source strength is obtained.

One problem with this method is that in the first step when the external source is used only the signal from this source and not the signal from the source under test must be picked up by the transducers. Increasing the level of the external source can in principle solve the problem. This has been attempted for IC-engines [20], [23] but did not succeed completely and good results could not be obtained in the low frequency region, where the engine produced the highest sound levels.

Another possibility is to use a reference signal correlated with the sound field from the external source, *e.g.*, an electric signal exciting a loudspeaker, but not correlated with the sound field from the machine under test [24] and by signal processing methods extracting the signal from the external source. Still there are sometimes difficulties to find an external source that produces sufficiently high sound levels. There may also be practical problems in mounting the external source in, *e.g.*, hot and "hostile" environments. This makes the indirect methods attractive in many applications.

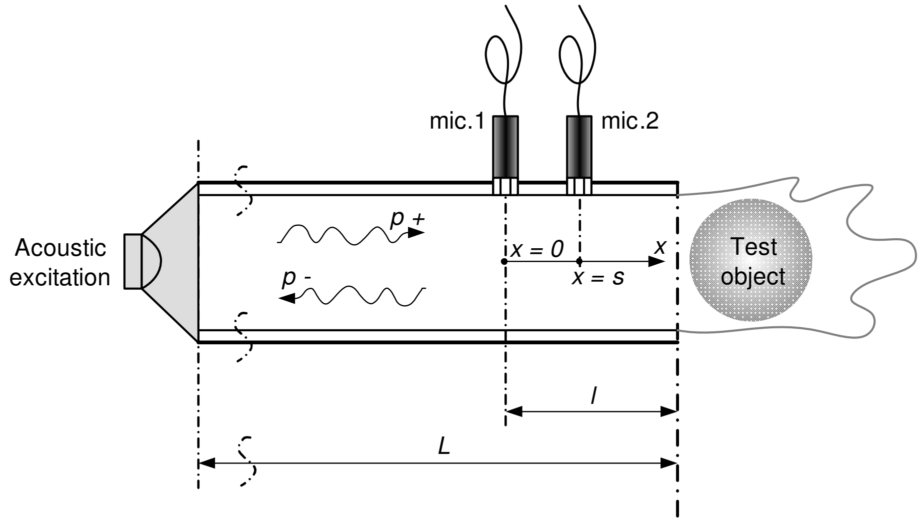


Figure 6. A schematic representation of the two microphone technique used in **Papers I, IV,** and **V** for characterization of passive acoustic data.

In **Paper IV**, experimental investigations of plane acoustic wave reflection at duct openings where a hot jet flows into relatively cold surrounding media have been carried out in order to validate the existing theory on duct openings.

It is believed that the Munt's model [25, 26] for duct terminations is presently the most refined and accurate and is therefore often used for predictions. However it has only partly been validated by the published experimental data and accurate measurements of the reflection coefficient for hot gases or two gases with different acoustic properties are of interest [27]. There have been attempts [28, 29] to experimentally investigate the radiation properties of ducts exhausting hot gases using the potentially less accurate and more time consuming standing wave technique. The temperature ranges studied have typically been rather limited in case of heated air or with less determined chemical consistency of the jet when the burners have been used.

The purpose of the study in **Paper IV** was to precisely measure the acoustic reflection from an open ended pipe exhausting high temperature gas and to validate the Munt's theoretical model for this problem. Hereby the standard two-microphone technique was further developed for experiments in hot environments to determine the reflection properties of the duct opening treated as acoustic one-port. For well determined and homogenous chemical consistency along the duct axis a heated air was used as a testing media inside the duct during the experiments. A dedicated test-rig (see Fig. 7) was built to investigate the acoustic wave reflection in jet temperatures ranging from room temperature to 500 ºC. A number of experiments were performed and the results obtained agree well with the ones calculated by using the Munt's model. An example of the results for three different temperature conditions, showing a good correlation between the theoretical and experimental data, is presented in Fig. 8. Hence it has been demonstrated that the theory can be used with reasonable accuracy to predict the sound reflection from the open duct termination in high jet temperature conditions in case of low Mach and Helmholz numbers.

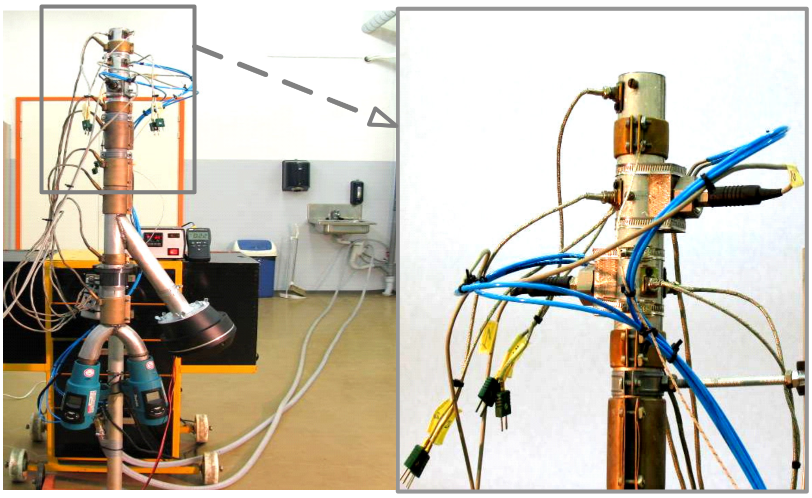


Figure 7. Application of direct method during duct termination characterization experiments in hot flow conditions using two microphone technique. Take notice of the two piezoelectric pressure transducers mounted to the test section via water-cooling jackets

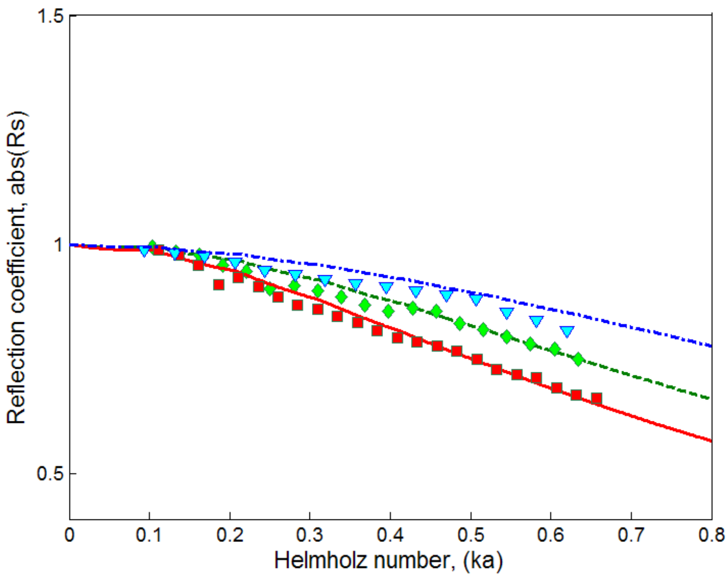


Figure 8. The magnitude of the reflection coefficient for duct open termination; measured at t=21 ºC, (blue triangles), predicted by using Munt's theory (blue dash-dotted line). measured at t=200 ºC, (green diamonds), predicted by using Munt's theory (green dashed line). measured at t=400 ºC, (red squares), predicted by using Munt's theory (red solid line).

In **Paper V** a finite element method (FEM) simulation has been carried out to test the applicability of a simple FEM model for this type of analysis and to investigate the effects of high temperature media on the sound propagation through open duct termination using numerical simulation. In order to simulate the experimental conditions and to obtain the acoustic pressure data, a commercial FEM software COMSOL has been used. A 3-D FEM model of the duct termination exhausting hot flow is shown in Fig. 9. The acoustic pressure reflection coefficient of the duct termination was calculated from the complex pressures simulated at the location of two microphones in the test-duct model.

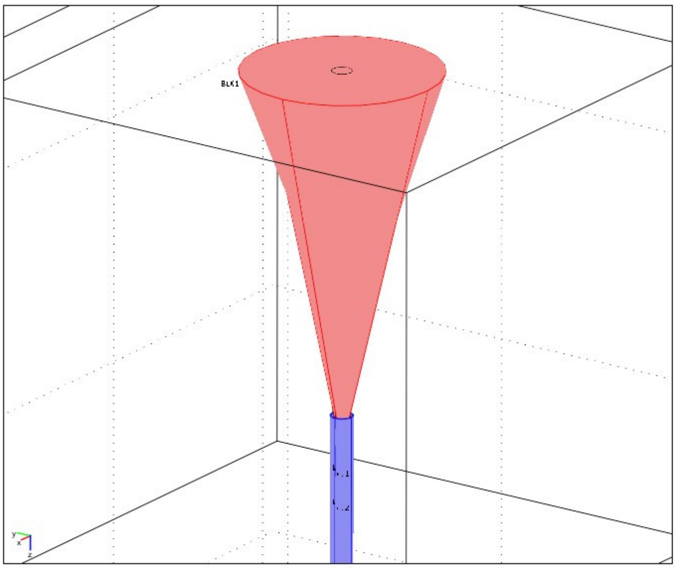


Figure 9. An image of the FEM model (with conical jet) of the duct termination exhausting hot flow, presented in **Paper V**.

An example of the numerically determined reflection coefficient magnitudes for two different jet temperatures is presented in Fig. 10. The corresponding results obtained experimentally and analytically in **Paper IV** are included for comparison. Despite of a relatively simple linear FEM model with neglected flow effects the numerical results were in good agreement with the experimental and theoretical ones.

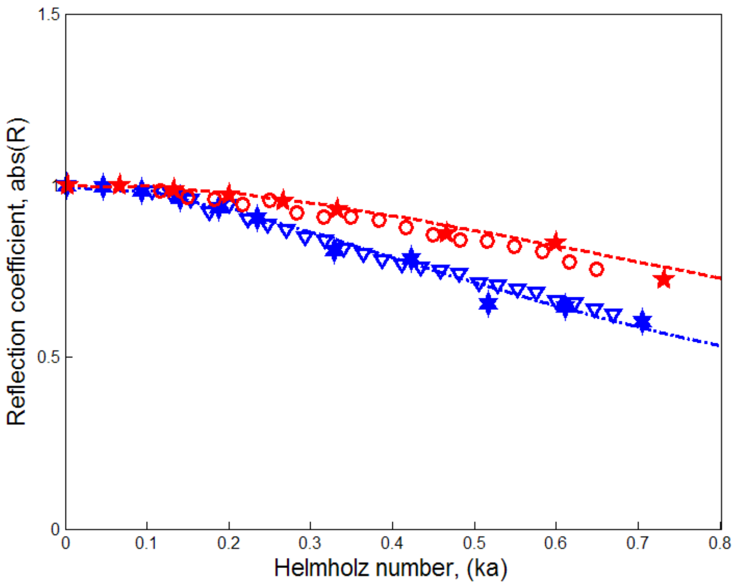


Figure 10. The magnitude of the reflection coefficient for duct open termination presented in Helmholz number domain; Experiments, t = 100 °C (red circles), FEM (cylindrical model), t = 100 °C (red stars), Munt's theory, t = 100 °C (red dashed line), Experiments, t = 500 °C (blue triangles), FEM (cylindrical model), t = 500 °C (blue hexagrams), Munt's theory, t = 500 °C (blue dash-dotted line).

2.2 Indirect methods

When using the indirect methods the two unknowns, the source strength and the source impedance, are determined via a multi-load procedure, *i.e.*, by applying known loads Z and measuring the acoustic pressure at the source receiver interface P. Since there are two unknowns, two loads should be sufficient to obtain the source data P_S and Z_S, which leads to the two-load method [30].

In case of linear time-invariant one-port model the pressure at the source cross-section can be expressed by the equation,

$$P = P_S \frac{Z}{Z_S + Z} \tag{8}$$

or with the unknown source data on the left hand side,

$$P_S \cdot Z - P \cdot Z_S = P \cdot Z \tag{9}$$

Eq. (9) has got two complex unknowns, which means that it can be solved if we have at least two complex equations. If we use n acoustic loads we get

$$\begin{bmatrix} Z_1 & P_1 \\ Z_2 & P_2 \\ \vdots & \vdots \\ Z_n & P_n \end{bmatrix} \cdot \begin{pmatrix} P_S \\ Z_S \end{pmatrix} = \begin{pmatrix} P_1 \cdot Z_1 \\ P_2 \cdot Z_2 \\ \vdots \\ P_n \cdot Z_n \end{pmatrix}, \tag{10}$$

where we have included more acoustic loads than we need, in order to get an over-determined system, which can be useful for improving the measurement results [8,31], and for checking if the source behaves as a linear system [24,15].

Alternatively the one-port model for the volume velocity source (see Fig. 2) can be expressed as:

$$Q - P\frac{1}{Z_0 Z_S} = \frac{P}{Z_0 Z}, \tag{11}$$

or with n loads:

$$\begin{bmatrix} 1 & -P_1 \\ 1 & -P_2 \\ \vdots & \vdots \\ 1 & -P_n \end{bmatrix} \cdot \begin{pmatrix} Q_S \\ 1/(Z_0 Z_S) \end{pmatrix} = \begin{pmatrix} P_1/(Z_0 Z_1) \\ P_2/(Z_0 Z_2) \\ \vdots \\ P_n/(Z_0 Z_n) \end{pmatrix}, \tag{12}$$

where we now solve for Q_S and $1/Z_S$. In order to determine the normalized impedances of the acoustic loads Z_i, used for experiments, a number of pressure transducers are usually mounted in the exhaust pipe (see Fig. 11). In the plane wave range we can use this information to perform wave decomposition and to determine the reflection coefficient looking into the acoustic load, which in turn gives the normalized load impedance.

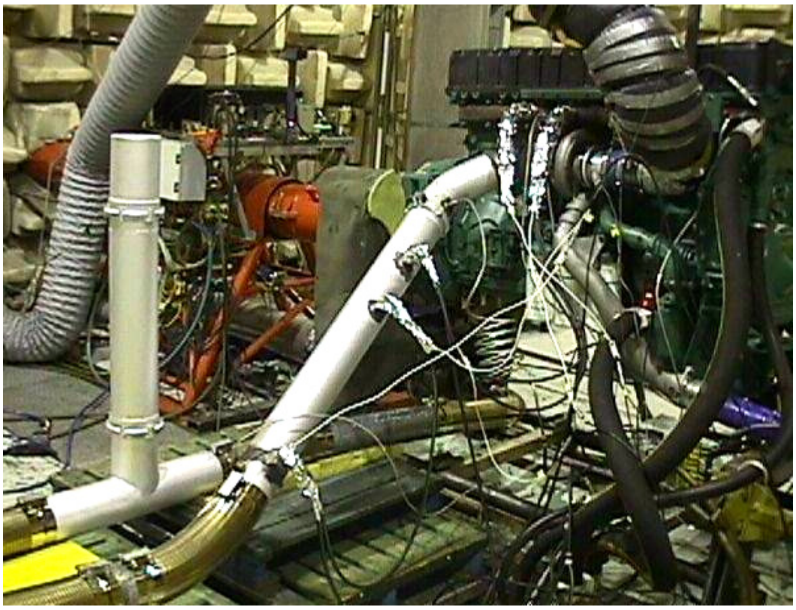


Figure 11. Pressure transducers mounted in an exhaust system of a 6-cylinder turbocharged diesel engine during source characterization experiments (described in **Paper II**) using the indirect approach

To determine the complex load pressures P_i we need a reference signal to ensure that the pressure time histories for the different acoustic loads start at the same point in the engine cycle. The pressure time histories are then Fourier transformed and used to calculate the load pressures and load impedances and subsequently the source data.

For a linear source it can be expected that the source impedance results from both the source formulations (see Fig. 2) should converge towards the same results when a large degree of over-determination is used. Any discrepancy between the results can therefore be an indication of a non-linear source behavior.

As described above the two-load method requires complex pressure measurements and a reference signal unaffected by acoustic load variations, which is related to the sound generating mechanism of the source. For fluid machines with periodic operation cycle the normal solution is to try to obtain a trigger signal for instance giving one pulse per revolution [8]. This procedure can catch harmonic part of the spectrum generated by machine but not the broad band part. It can also be noted that a trig signal can also be used to reduce flow noise disturbances from measured pressure signals.

Although the two-load method is strictly valid only for a linear time-invariant source model it has been reported to give useful results also in situations that are not exactly time-invariant or linear [1,19,8]. By using a number of extra loads, a solution which is the best fit in least squares sense, can be obtained.

As described above the two-load method requires complex pressure measurements and a reference signal unaffected by acoustic load variations, which is related to the sound generating mechanism of the source. For fluid machines with periodic operation cycle the normal solution is to try to obtain a trig signal for each period [8]. This procedure can catch harmonic part of the spectrum generated by machine but not the broad band part. It can also be noted that a trig signal can be used to remove the flow noise disturbances from measured pressure signals. An alternative method, used for flow noise suppression, is to create a “noise-free” acoustic reference signal, by using one or several reference microphones [24].

Although the two-load method is strictly valid for linear time-invariant equivalent source characterization, several authors have reported that it gives useful results also in situations that are not exactly time-invariant or linear, if applied by using a number of extra loads to average out the measurement errors in the least squares sense.

For situations where no suitable source reference is available alternative methods have been developed, where the auto-spectra of the pressures are measured instead of the complex pressures. The first such method was the three-load method [32].

By taking the squared magnitude of the equation (2), describing the one-port source, one gets, after substituting: $q = p / Z_o Z_S$, a real-valued equation with three unknowns, *i.e.*, $G^s = \left| p^s \right|^2$ and the real and imaginary parts of Z_S, $\mathrm{Re}(Z_S)$ and $\mathrm{Im}(Z_S)$. To determine the unknowns measurements using three different loads are needed. The resulting system of equations is non-linear and can have more than one real-valued solution. This method is quite impractical to use and has also been reported to give large measurement errors. A four-load method for evaluation of source impedance in ducts was introduced by Prasad [33]. Following this method, the magnitudes of sound pressures in terms of sound pressure level, and measured with single channel instrumentation, were used to obtain the source data. In the four-load method a fourth measurement is used to eliminate the non-linear term containing $\left| Z_S \right|^2$. This method has also been reported to be very sensitive to errors in the input data and can therefore sometimes give erroneous results. Bodén [34] showed that the four-load method can be formulated as a linear system of equations, if the non-linear term is interpreted as an independent unknown. By analyzing this formulation it was concluded that the main reason for previously reported is the choice of loads. A new improved method for analyzing the same experimental data used for the four-load method was also presented [34]. This method is based on a direct numerical fit of the data to the non-linear model using least squares methods. A comparison between the results, obtained when applying the described measurement methods to various sources, was also made. It was concluded that generally the direct methods give better results than the indirect methods in situations where it is suitable to use them. A further improvement of the technique of [34] has been presented by Jang and Ih [17].

An extension of the conventional two-load method, to characterize the linear time-variant sources, was presented by Boden [8]. This method is called the multiple-load method and it requires 1+2N loads, where N denotes the number of harmonics to be included in the source spectrum.

The multiple load method was derived assuming that the time variance was caused by parametric excitation. In deriving the expressions a second order differential equation was used as an example [19]:

$$A(t) \cdot \ddot{Q}(t) + B(t) \cdot \dot{Q}(t) + C(t) \cdot Q(t) = P_S(t) - P(t)\,, \tag{13}$$

where $Q(t)$ is the volume velocity, $P(t)$ is the pressure at the outlet of the source and $P_S(t)$ is the source pressure. The time variation in coefficients $A(t)$, $B(t)$ and $C(t)$ represent for instance time varying volumes in a cylinder or time varying cross sectional areas in valves, giving a parametric excitation of the system.

For a typical fluid machines all time varying quantities in (13) are periodic and can be expanded in complex Fourier series,

$$Q(t) = \sum_{n=-\infty}^{\infty} q_n \exp(jn\omega_0 t)\,, \qquad P(t) = \sum_{n=-\infty}^{\infty} p_n \exp(jn\omega_0 t)\,, \qquad A(t) = \sum_{n=-\infty}^{\infty} a_n \exp(jn\omega_0 t)$$

$$B(t) = \sum_{n=-\infty}^{\infty} b_n \exp(jn\omega_0 t)\,, \qquad C(t) = \sum_{n=-\infty}^{\infty} c_n \exp(jn\omega_0 t)\,. \tag{14}$$

Inserting (14) in (13) and identifying terms with the same time variation gives, *e.g.*, for the term with time variation ($\exp(jnw_0 t)$),

$$\sum_{k=-\infty}^{\infty}\left(-\omega_0^2 k^2 a_{n-k} + j\omega_0 k b_{n-k} + c_{n-k}\right)\cdot q_k = p_{sn} - p_n \,. \tag{15}$$

The result can also be expressed in matrix form,

$$[Z_S]\cdot(q)+(P)=(P_S), \tag{16}$$

where Z_S is the source impedance matrix, and q, P and P_S are vectors. The matrix and vectors will have infinite dimension, but in practice only a limited number of terms can be included. The number of terms, *i.e.* frequency components included, must be chosen so that sufficiently good description of the main characteristics of the studied source is given.

In **Paper II** a modified version of the two-load method to improve the characterization of the non-linear acoustic 1-port sources has been developed and tested.

The procedure for obtaining (7) is that $p(t)$ and $q(t)$ are first determined from measurements or simulations. The nonlinear function $b(t)$ is then calculated followed by the Fourier transform of these quantities. It should be noted that an anti aliasing filter should be applied on $b(t)$ to avoid aliasing problems caused by the presence of frequency components higher than half the sampling frequency. The load impedance Z is obtained from the ratio of the Fourier transform of $p(t)$ and $q(t)$ ($Z = P/Q$). By using n acoustic loads (7) gives

$$\begin{bmatrix} Z_0 Z_1 & -Z_0 P_1 & -B_1 \\ Z_0 Z_2 & -Z_0 P_2 & -B_2 \\ \vdots & \vdots & \vdots \\ Z_0 Z_n & -Z_0 P_n & -B_n \end{bmatrix} \cdot \begin{pmatrix} P_S \\ Z_S \\ H_S \end{pmatrix} = \begin{pmatrix} P_1 \cdot Z_0 Z_1 \\ P_2 \cdot Z_0 Z_2 \\ \vdots \\ P_n \cdot Z_0 Z_n \end{pmatrix}. \tag{17}$$

A minimum of three loads is required to solve for the source data while over-determination is used to reduce effects of measurement errors and deviations from the model just as for the two-load method. It should be noted that one difference between the method proposed in **Paper II** and the method of Jang and Ih [17] is that in the method proposed in **Paper II** the non-linear term is expressed using boundary conditions seen from the source side towards the load while situation is reverted for method of Jang and Ih [17]. This makes the method of Jang and Ih [17] potentially more relevant for characterization of non-linear source behavior. In their method it is however difficult to see how the particle volume velocity $q(t)$ at the source cross section can be determined from experiments. In the method suggested in **Paper II** this is easier. It is also possible that the extra non-linear term introduced will anyway give an improved result compared to the linear time-invariant model for non-linear sources.

The new non-linear indirect source characterization technique proposed in **Paper II** requires one additional acoustic load compared to the two-load technique. Since over-determination is anyway used in many cases the additional data would often be available. It has been shown that the new technique gives improved results compared to the two-load technique if the source is non-linear or time-varying.

In Fig. 12 an example of the sound pressure prediction in a load duct presented for a simulated highly non-linear 1-cylinder engine model configuration with geometrical parameters similar to those used in diesel powered IC-engines. It can be seen that the non-linear multi-load technique gives a significantly better result for this case.

For cases when the source is linear and time-invariant the new technique gives the same result as the two-load technique. This means that there is no risk for increased errors when the source is linear and time-invariant.

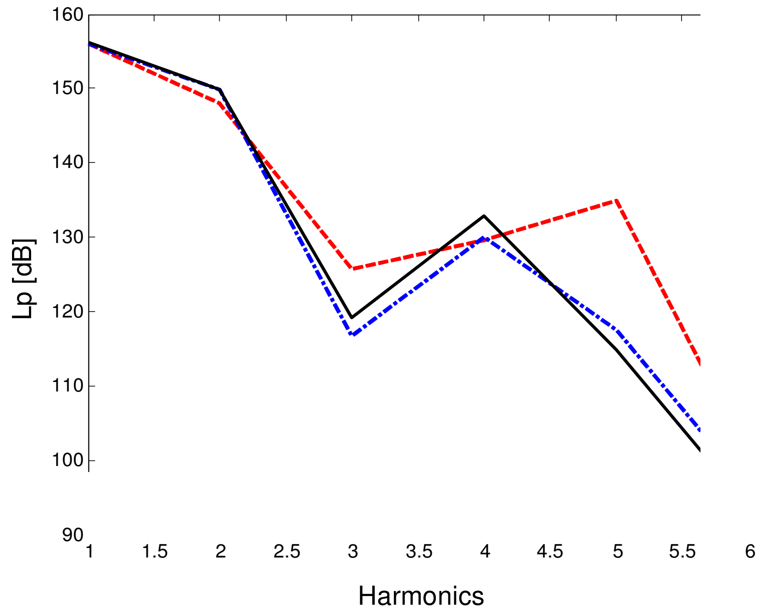


Fig. 12. Sound pressure level in a load duct of 1-cylinder engine model;
New non-linear multi-load technique (blue dash-dotted line),
Two-load technique (red dashed line),
Direct simulation (black full-line).

3 SOUND GENERATION MECHANISMS

According to Lighthills classical theory there are three basic mechanisms that can create sound in a fluid: i) fluctuating volume flow – monopole type of source; ii) fluctuating surface pressures – dipole type of source and iii) free turbulence – quadrupole type of source.

In a rotating machine (see Fig. 13), such as a turbo-compressor treated in **Paper III**, the blades will generate a monopole type of source only when the blades move close to the speed of sound or supersonically. In the supersonic case there will be rotating shock waves attached to the blades. The spectrum of the monopole contribution will in the supersonic case be harmonics of the rotation frequency f_0.

This type of rotating shock wave noise is referred to as "buzz-saw" noise and exists for instance on modern aero-engines. Supersonic tip speeds are also common for modern high speed turbo compressors.

A number of dipole or fluctuating pressure sources can be found in rotating machines. One is the blade pressure. The most important contribution to time varying blade pressures are inflow disturbances. These can be of two types: stationary and non-stationary flow distortions, *e.g.*, turbulence. The stationary distortions will lead to periodic acoustic signals and the turbulence to a broad band signal. The periodic signals will consist of harmonics of the blade passing frequency (BPF). Without inflow disturbances these tones are referred to as rotor-alone tonal noise. A second source of dipole noise on rotating machines is rotor-stator interaction, *e.g.*, fluctuating pressure fields created on outlet guide vanes by the rotor. This noise is also periodic in time and consists of harmonics of the BPF.

For purely axial machines there is a possibility to block the generation of the first rotor-stator harmonic by using the so called Tyler-Sofrin rule, which states that the number of guide vanes at least must equal twice the number of blades. A third dipole mechanism is associated with the leakage flow between the high and low pressure side of a blade, which under certain conditions creates unsteady vortex shedding. This can generate a peak in the sound spectrum, referred to as tip clearance noise (TCN), since the effect is most predominant at the blade tips.

Finally the turbulence in the flow will act as a broad band acoustic source but normally this source is not important unless the flow forms a jet with a Mach-number close to 1. This quadrupole type of source is normally important only at the outlet of jet engines and high pressure valves, but for turbochargers it is of little interest.

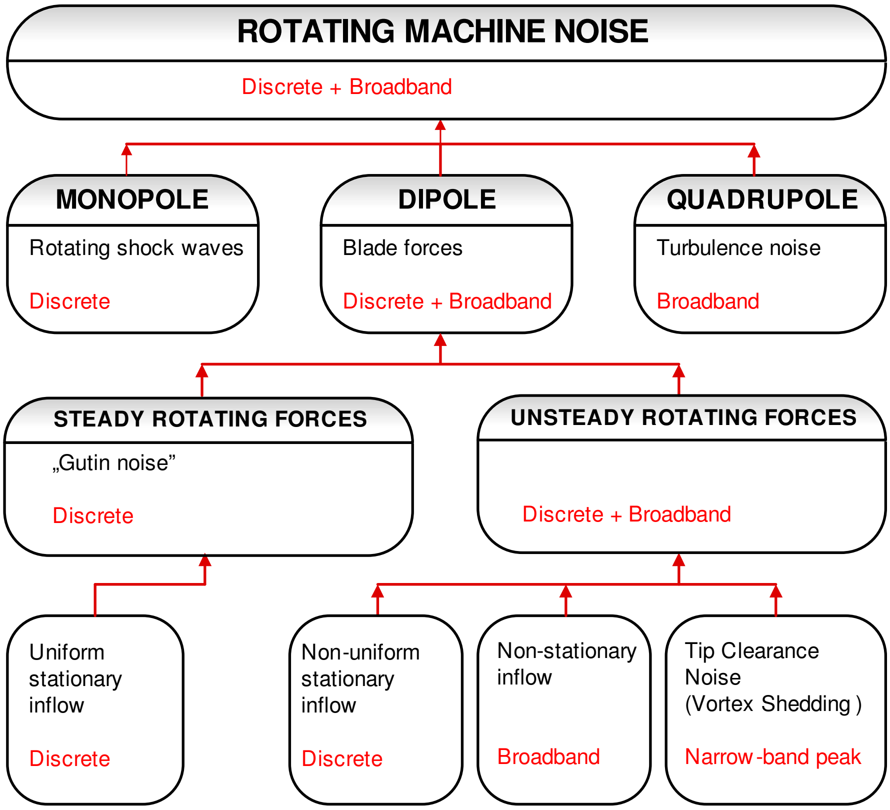


Figure 13. Summary of noise sources for rotating machines. For subsonic rotors only dipole (fluctuating surface pressure) is important. For supersonic rotors rotating shock waves are also important. The quadrupole – noise from free turbulence is normally negligible.

In general the influence of turbocharger noise has been reported (*e.g.*, in [35-37]) to be more important on the intake, *i.e.*, compressor side, mainly because of the relatively good damping characteristics by modern mufflers on the exhaust side. Based on the above discussion the main aerodynamic noise generating mechanisms in centrifugal compressors are: tonal noise at blade passing frequencies, buzz-saw noise and blade tip clearance noise. The flow conditions responsible for the noise generation in turbocharger compressor are illustrated in Fig. 9. A more detailed discussion of the mechanisms is given in **Paper III**, mainly based on a recent work on high performance centrifugal compressors [38].

In general the aerodynamic output of a centrifugal compressor will increase proportional to the RPM raised to the power 3. However, the aero-acoustic power dominated by dipole type of mechanisms can be expected to be proportional to the RPM raised to the power 5 to 6. This implies that unless measures are taken turbochargers will become more and more important noise sources as the pressure ratio and swallowing capacity is increased.

Recently, a new research centre, KTH CICERO, for investigations on IC-engine gas management has been established in Stockholm. The centre is a combined effort between KTH, the Swedish Energy Agency (STEM) and the leading vehicle manufacturers in Sweden and research on pulsed compressible flows and turbocharging are central themes. The centre will have access to special test rigs in particular a unique rig for acoustic characterization of turbochargers described in **Paper III**. A sketch illustrating the layout of this acoustic test-rig is presented in Fig. 15. Besides measurements for active data the rig can also be used to measure passive acoustic data.

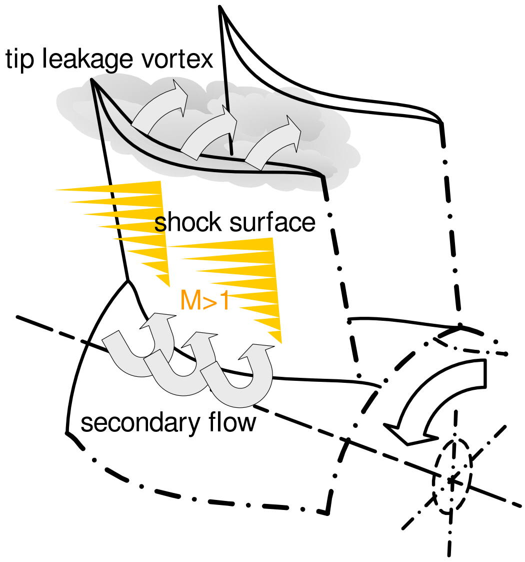
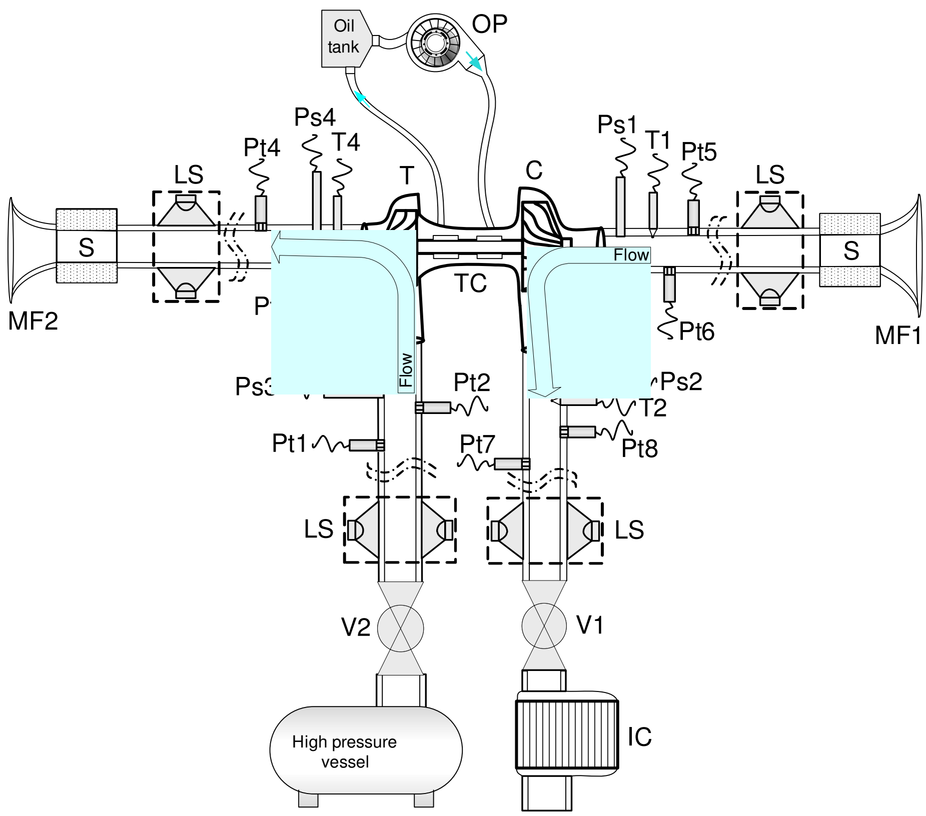


Figure 14. Schematic representation of the flow conditions in turbocharger compressor that are responsible for the noise generation.



Ps – Static pressure probe
Pt – Pressure transducer
V – Control valve / orifice / throttle
T – Temperature sensor (bi-metal type)
LS – Removable loudspeaker unit
MF – Massflow metering system
OS – Optical sensor

T – Turbocharger turbine side
C – Turbocharger compressor side
TC – Turbocharger
OP – Electrically driven oil pump
IC – Intercooler
S – Silencer

Figure 15. The KTH CICERO test-rig designed for acoustic characterization of turbochargers and turbocharger system components.

Of interest in **Paper I** was also the sound generation and the determination of the sound power radiated from the air terminal device. When the 1-port source data had been determined according to the approach described in section 1, the resulting volume flow q in the opening creates a monopole type of source. The volume flow in the opening can be written as: $q = S \cdot (u_+ + u_-)$, where u is the acoustic velocity in the +/- x-direction and S is the cross-sectional area of the duct. It is known that: $u_+ = \frac{p_+}{\rho c}$ and $u_- = \frac{p_-}{\rho c}$, using the equation for source strength $p_+^S = p_+(1 - R_S R_L)$, this implies

$$q = \frac{p_+^S \cdot S(1 - R_L)}{\rho c(1 - R_S R_L)}. \tag{18}$$

For sufficiently low frequencies and small Mach-numbers the radiation resistance of the ATD can be approximated by that of a monopole [39].
This implies that if the acoustic volume flow q at the ATD is known then the radiated power is

$$W_{rad} = \frac{\rho c k^2 |q|^2}{4\pi}. \tag{19}$$

Combining this with equation (18) leads to

$$W_{rad} = \frac{\pi^3 f^2 d^4 |1 - R_L|^2 G_{SS}}{16 \cdot c^3 \rho |1 - R_S R_L|^2}, \tag{20}$$

where $S = \pi d^2 / 4$, G_{SS} is the source strength, k is the acoustic wave-number, R_L is the reflection coefficient of the load, ρ is the density and d is the duct diameter. Expressing this as levels using the normal reference values gives the sound power radiated:

$$L_W = L_S + 10 \cdot \log_{10}\left(\frac{\pi^3 f^2 d^4 |1 - R_L|^2}{16 c^3 \rho |1 - R_S R_L|^2} \right) + 26, \tag{21}$$

where $L_S = 10 \cdot \log_{10}\left(G_{SS} / p_{ref}^2\right)$.
In Fig. 16, an example for the prediction of the radiated sound power (Eq. (21)) is shown in 1/3-octave bands. The predicted curve is calculated by using the same source model for the air terminal device as for the prediction of the in-duct cross-spectrum, described in section 1. It should be noted here that the source model used for predictions was derived from the data measured inside the duct.
For the outside radiation the agreement with measurements is not as good as for the in-duct case (see Fig. 5).

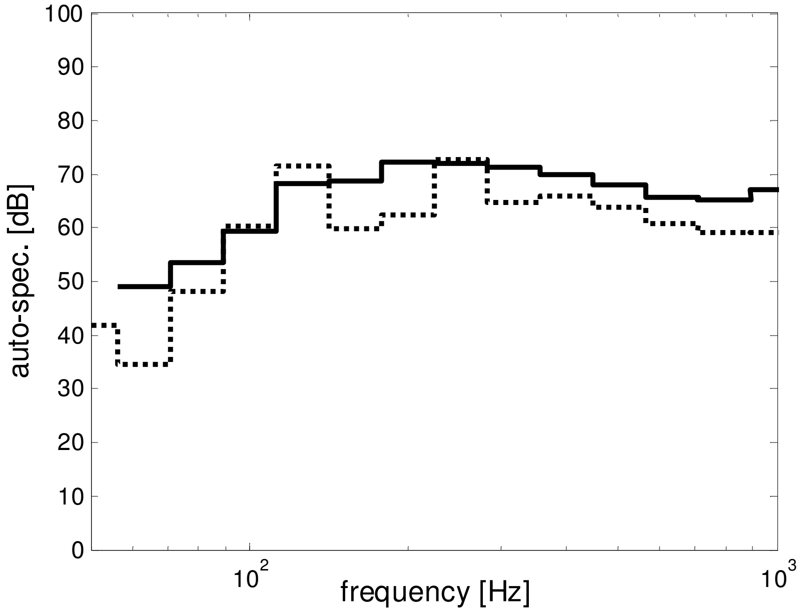


Figure 16. Sound power spectrum (1/3 octave band) in the reverberation room from the air terminal device with 12.0 m/s flow velocity in validation duct system; measurements (full-line), prediction (dotted-line).

In order to investigate the reason for the less good agreement for the radiated sound power the same source characterization procedure was performed with a simplified air terminal device, an orifice plate with a circular hole. This and other tests led to the conclusion that the resulting dipole generated by an obstruction at a duct opening often has non-axial components. These components are created when the obstruction deflects the flow from the axial direction by the fluctuating part of the external force components needed for the deflection. When the source data is determined inside the duct only the axial (along x axis) component of the dipole will excite a plane wave field. The source data determined inside the duct will therefore only contain information about this component. For the outside radiation a 3-D sound field exists and all dipole components, axial and non-axial, can play a role. Also since for the air terminal device studied here the source is located just outside the duct opening, around 1/10 of a wavelength at 1 kHz, the full axial dipole strength will not excite the sound field inside the duct. This effect together with the effect of non-axial dipoles creates a tendency to underestimate the radiated sound based on in-duct (plane wave) source data. It can be noted that the discussion above is in accordance with results presented in the paper by Heller and Widnall [40].

4 CONCLUSIONS

The methods to study sound propagation in automotive duct systems have been treated in this thesis. The thesis is based on five published papers. The techniques for in-duct noise source characterization and modeling together with aero-acoustical sound generation mechanisms have been the central themes.

4.1 The main contributions

The following are the main original results of this thesis:

1. A measurement method to characterize a standard air terminal device as an acoustic one-port source has been tested and validated. A scaling law for air terminal device noise generation is derived and verified for prediction of radiated noise. In the experimentally derived scaling law a flow speed dependence of 3 was found for the narrow band spectra, corresponding to a dipole-like behavior of the source in the plane wave range. The present work represents the first efforts to apply active 1-port models to describe flow generated noise.
2. A new nonlinear source model and a multi-load technique for extracting the source data have been presented. It has been proved that the new source characterization technique gives better results compared to the classical two-load method if the source under test exhibits non-linear or time-varying behavior.
3. The dominating noise generation mechanisms together with the passive acoustic effects of a turbocharger have been systematically analyzed. A novel experimental setup is designed for acoustic characterization of turbochargers.
4. The reflection coefficient magnitude and phase for a circular duct termination have experimentally been determined for hot flow conditions and compared with Munt's theory. Good correlation was found between the results. This result is a first experimental validation of the well-known Munt's theory for hot flow conditions.
5. A technique has been proposed to numerically determine the acoustic passive properties of the duct termination from the complex pressures simulated at the location of two microphones in the test-duct. The reflection coefficient results have been obtained numerically for a circular duct termination exhausting hot gas. The results showed a good agreement with the experimental and theoretical ones. Hence the applicability of FEM to investigate the sound propagation through the open duct termination in hot flow conditions is demonstrated.

4.2 Future research

An interesting continuation of the work presented in the first paper of the thesis would be to try to establish methods to estimate one- and two-port source data directly from CFD calculations. Since the passive part of the source data is often weakly affected by the flow it could be sufficient "only" to estimate the active part (source strength).

In the third paper of the thesis it was concluded that presently there is a lack of both experimental and theoretical acoustic research on high performance centrifugal machines as compared to the axial machines (aero-engines). Also for the IC-engine turbochargers, the effect of pulsed flow on both the performance and the acoustics is a key factor that needs to be addressed.

For the fourth and fifth part of the thesis, where the acoustical properties of flow duct termination have been studied it would be of interest to perform such the high temperature experiments with substantially higher flow velocities, in order to including the Mach number effects. As well as to introduce mean flow into the FEM model applied in the fifth part. Similar investigations with different duct opening geometries that are in practice often used in hot jet conditions (bends, diffusers, oblique cuts) are potentially providing a number of interesting continuations of the research.

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KOKKUVÕTE

EKSPERIMENTAALSED MEETODID HELI LEVIKU UURINGUTEKS AUTO-TEHNILISTES TORUSÜSTEEMIDES

Antud doktoritöös on vaatluse all meetodid heli leviku ja müraallikate uuringuteks autotehnilistes kanalsüsteemides. Töö on koostatud viie publitseeritud artikli baasil.

Töö esimeses osas on välja töötatud mõõtmismeetod voolukanali väljundava akustiliseks kirjeldamiseks kasutades ühe vabadusastmega mudelit.
Väljundava kui müraallika tugevuse iseloomustamiseks loodi eksperimentaalsete andmete põhjal matemaatiline mudel, milles sisalduv kanali voolukiiruse eksponent viitas dipolaarset tüüpi kiirgurile. Antud mudelit testiti ning tulemustest võis järeldada, et kirjeldatud mudelit järgides on võmalik suhteliselt täpselt ennustada väljundava poolt kanalisse kiiratava müra tugevust kõikide voolukiiruste ning voolukanali konfiguratsioonide korral. Kirjeldatud meetod on esmakordne katse kasutada ühe vabadusastmelisi akustilisi mudeleid voolu poolt genereeritud müra modelleerimisel.

Töö teises osas on tavalist kahe-koormuse meetodit modifitseeritud nii, et suurendada selle täpsust mitte-lineaarsete ühe vabadusastmega akustiliste allikate modelleerimisel. Lisaks uuriti sisepõlemismootori lihtsustatud matemaatilise mudeli abil mootori lineaarsust mõjutavaid parameetreid. Uut meetodit katsetati ning võrreldi tavalise kahe-koormuse meetodiga kasutades lihtsustatud mudelil saadud andmeid ning eksperimentaalseid andmeid ühesilindrilise kompressori ja kuuesilindrilise turbolaadimisega diiselmootori gaasivahetussüsteemist. Tulemused tõestavad uue meetodi eeliseid olukordades, kus akustiline allikas oli mittelineaarse või ajas muutuva iseloomuga.

Töö kolmandas osas on kokku võetud ja analüüsitud turbokompressori akustikat. Lisaks järeldustele varem publitseeritud töödest, esitatakse ka autori poolt EL projekti ARTEMIS raames läbi viidud uuringute tulemusi ning hiljuti käivitunud uurimiskeskuses KTH CICERO läbi viidud eksperimentide tulemusi.

Töö neljandas osas on uuritud akustilise laine peegeldust kanali avatud otsal, kust kuum gaasiline keskkond väljub toatemperatuuril keskkonda. Loodud on mõõtestend ning läbi viidud eksperimendid kuni 500ºC voolu korral. Kõrgetemperatuurilise voolava gaasina kasutati kuumutatud õhku. Mõõtmismeetodina rakendati klassikalist kahe-mikrofoni meetodid ning uuritavat väljundava käsitleti ühe vabadusastmelisena. Eksperimentaalselt määratud tulemused on võrreldud analüütilisel teel saadutega, kasutades tuntud Munt'i teoorial põhinevat matemaatilist mudelit. Võrreldud tulemused ühtivad hästi. Kirjeldatud eksperimendid on esmakordne Munt'i teooria kontroll kõrgtemperatuuriliste voolutingimuste korral.

Töö viiendas osas on loodud lõplike elementide meetodil (LEM) baseeruv mudel, et jätkata eelnevas osas kirjeldatud heli leviku uuringuid läbi kanali väljundava, kõrge temperatuuriga gaasijoa korral. Kolmedimensionaalse LEM mudeli abil simuleeriti akustilise tasalaine rõhk rõhuandurite asukohale vastavates ristlõigetes ning nende kaudu arvutati kanali otsa peegeldustegur. Testiti kahte geomeetriliselt erineva gaasijoaga mudelit ning tulemused esitati võrdlusena eksperimentaalsel teel määratutega. Peegeldusteguri amplituudi ja faasi vastavad väärtused ühtivad hästi eksperimentaalsetega ning tulemuste erinevus kahe mudeli puhul on marginaalne. Sooritatud analüüs demonstreeris kirjeldatud simulatsioonimeetodi sobivust samalaadsete probleemide lahendamisel ning kuuma vooluga kanalist kiiratava müra teoreetilisel määramisel.

Töö peamised tulemused on järgmised:

1. Esmakordselt on tõestatud ühe vabadusastmeliste akustiliste mudelite rakendatavus voolu poolt genereeritud müra modelleerimisel. Väljundava kui müraallika iseloomustamiseks on loodud matemaatiline mudel, mida saab rakendada sarnaste avade poolt kiiratava müra ennustamisel mistahes voolukiiruse ja kanali konfiguratsiooni korral,

2. Klassikalist kahe-koormuse meetodit on modifitseeritud, suurendades selle täpsust mitte-lineaarsete ühe vabadusastmega akustiliste allikate iseloomustamisel,
3. Analüüsitud on turbolaaduri müra genereerumise mehanismi ning välja on arendatud uudne katsestend turbolaadurite akustiliseks iseloomustamiseks.
4. Esmakordselt on teostatud tuntud Munt'i teooria eksperimentaalne valideerimine kanali väljundava kõrgtemperatuuriliste voolutingimuste korral,
5. Demonstreeritud on lõplike elementide meetodil põhineva simulatsioonimeetodi rakendatavust heli leviku uuringutel läbi kanali väljundava, kõrge temperatuuriga gaasijoa korral.