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Research on Linear Acoustic Modelling and Testing of Exhaust Mufflers

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Abstract: In an Automobile Intake and Exhaust system, noise do a tremendous contribution to the interior and exterior noise. There have been number of Linear Acoustic Tools developed by Industries and Institutions that can predict intake and exhaust acoustic properties of the system. This project through measurements validates and discusses the proper modelling of these systems using BOOST-SID and discusses the ideas of how to properly convert a geometrical model of an exhaust muffler to an acoustic model. The various elements employed with their properties are discussed as well. There are many parameters that explain the performance of a muffler, when acoustic properties are considered. The Transmission Loss can be useful to check the validity of a mathematical model but when it is desired to predict the actual acoustic behaviour of the component after installation in system and subjected to operating conditions, it's when other properties like Attenuation, Insertion loss etc. need to be determined.

Zero flow and Mean flow [M=0.12] measurements of these properties were carried out for mufflers ranging from simple expansion chambers to complex geometry using two approaches 1] Two source location technique 2] Two Load technique. In both the techniques the measured losses in transmission were compared to those obtained from BOOST-SID models. The comparison of acoustic properties with simulated model was well carried out in all the cases.

I. OBJECTIVES

1) *Minimise Rolling Noise:* The noise that arises from tyre road interaction is termed as rolling noise. The models like Linear Acoustic models and Non-Linear models are designed and modified in order to minimise rolling noise.
2) *Minimise Propulsion Noise:* The engine noise, exhaust system noise, intake noise are the components of propulsion noise. Controlling these noise sources is the subject of stringent road noise regulations. It contributes to globally emitted engine noise, which is year by year. Mufflers play an important role in reducing exhaust and intake system noise. The traditional built and test procedure is assisted by numerical simulation models which help to predict the performance of different muffling systems in short time. There are one-dimensional models such as linear acoustic models and non-linear acoustic models which can keep the level of vehicle noise in check.

II. INTRODUCTION

Road traffic noise is caused by the combination of *rolling noise* (arising from tyre road interaction) and propulsion noise (comprising engine noise, exhaust system and intake noise). Controlling these noise sources, which contributes to the globally emitted engine noise is the subject of stringent road noise regulations which is being updated year by year.

Noise is therefore studied, regulated and monitored by many countries, authorities, and establishments due to the negative effects. Noise from the transportation sector, and more specifically road vehicles with internal combustion engines is something people interact within a day-to-day basis making it an important area for noise control.

Manufacturers of all kinds of road vehicles strive to mitigate as much noise as possible to produce silent vehicles both due to legislation and competition. Knowledge of the acoustic source characteristics of internal combustion engines (IC engines) is of great importance when designing the exhaust duct system and its components to withstand the resulting dynamic loads and to reduce the exhaust noise emission.

The goal of the present review is to show numerically and experimentally investigate the variety speed IC-engine acoustic source characteristics, not only in the plane wave range but also in the high frequency range define the wave equation one must first look at one – dimensional the linear conservation equation of continuity which relates density and particle velocity up in the medium. The decomposed definition of density have been inserted and higher-order terms are neglected

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The *propulsion noise* comprises of combustion, mechanical noise and the noise radiated from the open terminations of the intake and exhaust systems which is caused by

1) The pressure pulses generated by the periodic charging and discharging process, which propagates to the open ends of the duct system (Pulse noise), and
2) The mean flow in the duct system, which generates significant turbulence and vortex shedding at geometrical discontinuities (Flow generated noise).

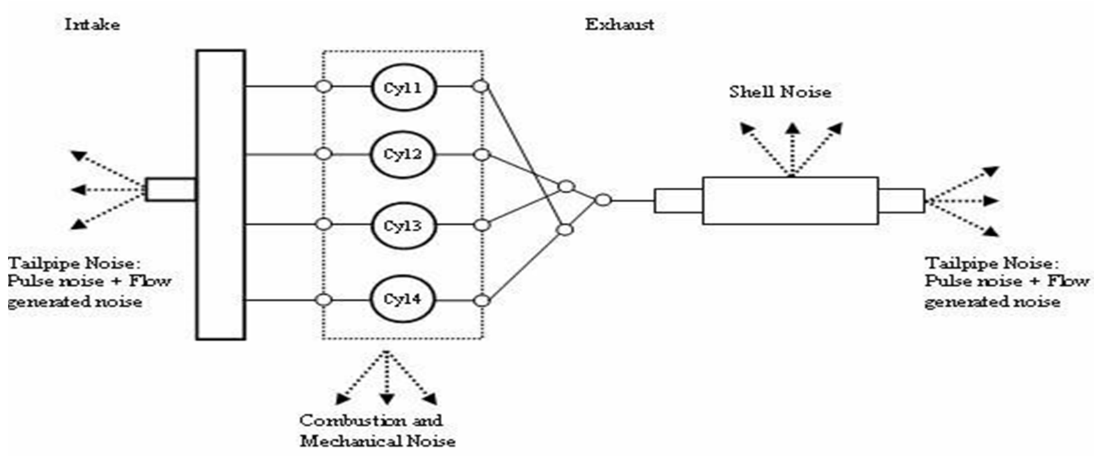
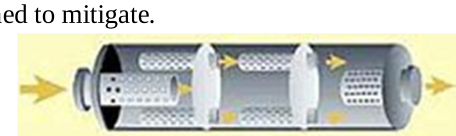


Figure 1-1 Schematic of Engine Noise Sources.

A muffler or silencer is a device for reducing the noise emitted by the exhaust of an internal combustion engine especially a noise-deadening device forming part of the exhaust system of an automobile. Mufflers are installed within the exhaust system of most internal combustion engines. The muffler is engineered as an acoustic device to reduce the loudness of the sound pressure created by the engine by acoustic quieting. The noise of the burning-hot exhaust gas exiting the engine at high speed is abated by a series of passages and chambers lined with roving fiberglass insulation and/or resonating chambers harmonically tuned to cause destructive interference, wherein opposite sound waves cancel each other out.

An unavoidable side effect of this noise reduction is restriction of the exhaust gas flow, which creates back pressure, which can decrease engine efficiency. This is because the engine exhaust must share the same complex exit pathway built inside the muffler as the sound pressure that the muffler is designed to mitigate.



A cutaway muffler showing the interior pipes and chambers which reduces Horsepower

Some aftermarket mufflers claim to increase engine output and/or reduce fuel consumption by slightly reduced back pressure. This usually entails less noise reduction (i.e., more noise).

On May 18, 1905, the state of Oregon passed a law that required vehicles to have "a light, a muffler, and efficient brakes".

The legality of altering a motor vehicle's original equipment exhaust system varies by jurisdiction; in many developed countries such as the United States, Canada, and Australia, such modifications are highly regulated or strictly prohibited.

Aftermarket mufflers usually alter the way a vehicle performs, due to back-pressure reduction.

Mufflers play an important role in reducing the exhaust and intake system noise and as a result, a lot of research is done to designing these systems effectively. The traditional “build & test” procedure which is time consuming and expensive, can nowadays be assisted by numerical simulation models which are able to predict the performance of several different muffling systems in a short time. A number of numerical codes have been developed in the past few decades, based on distinct assumptions.

Considering only one-dimensional models, the two types of simulation models may be distinguished as

a) *Linear Acoustic Models:* This is based on the hypothesis of small pressure perturbations within the ducts, and
b) *Non-linear Gas Dynamics Models:* This describes the propagation of finite amplitude wave motion in the ducts.

Linear acoustic models are frequency domain techniques which for instance use the four pole transfer matrix method to calculate the transmission loss of mufflers. This approach is very fast but the predicted results may be unreliable because the propagating pressure perturbations generally have finite amplitude in an exhaust system.

On the other hand, non-linear gas dynamic models are able to simulate the full wave motion in the whole engine intake and exhaust system and are based on time domain techniques. This simulation follow the gas flow from valves to open terminations and so is suited to deal with finite amplitude wave propagation in high velocity unsteady flows. The excitation source can be modelled by means of appropriate boundary conditions for the flow in these simulations.

AVL BOOST is a 1D- gas dynamic tool which predicts engine cycle and gas exchange simulation of the entire engine. It also incorporates the linear acoustic prediction tool SID (Sound in Ducts) so it is possible to simulate both the non-linear and linear acoustic behaviour of the system.

AVL BOOST is engine cycle and gas exchange simulation software that enables you to build a model of the entire engine by selecting elements from a toolbox and connecting them by pipe elements. These elements include cylinders, air cleaners, catalysts, intercoolers, turbochargers, advanced junction models, and many more. Within the pipes, one-dimensional gas dynamics are considered. An adapted Godunov - Scheme ensures a high accuracy of the solution. BOOST can simulate steady-state operating points as well as engine transients.

BOOST is used in both the automotive and nonautomotive industries for accurate engine simulation. It has been used for modelling a wide range of engine speeds and sizes, including two-stroke and four-stroke engines. In the BOOST engine model, numerical outputs (sensors) can provide values, such as pressure, temperature, air/fuel ratio, and many more, at measuring points in the simulation model. These can be used as input to a Simulink model of the engine control unit (ECU). Signals can be passed into BOOST (actuators) to control parameters such as throttle position or vane position for a variable geometry turbocharger.

III. LITERATURE REVIEW

A. *Experimental Investigation for the effect of Inlet & Outlet Pipe Lengths on Noise Attenuation in a Muffler*

Muna S. Kassim, Muthana K. Al-Doory, Ehsan Sabah M. Al-Ameen

Journal of Engineering and Sustainable Development (JEASD), 2012, Volume 16, Issue 2, Pages 1-15

With the increase use of large industrial machinery (such as huge generators) and the increase in public awareness and concern for noise control the desire to be able to properly design a silencer for specific application is increasing. A test rig is built for a reactive muffler. The object is to minimize the noise level using different pipe lengths for inlet and outlet (discharge) tubes are studied. The major conclusion was that the taller outlets pipe the higher noise attenuation will be obtained.

B. *Design and Analysis of an Expansion Chamber Mufflers*

Bhat, Chandrasekhar and Sharma, SS and Jagannath, K and Mohan, N S and Sathisha, S G (2010) *Design and Analysis of an Expansion Chamber Mufflers.* World Journal of Engineering, 7 (3). pp. 117-118. ISSN 1708-5284

Internal combustion engines are fitted with exhaust muffler to attenuate the pressure pulsation generated during combustion process. Effective prediction of sound pressure loss of hot gases as it flows through the muffler greatly helps in the design of mufflers. An attempt has been made here to predict the transmission loss through modal analysis, followed by acoustic analysis using finite element analysis technique for three different configurations of mufflers under different fixing conditions. It was found that three-chamber muffler provides higher attenuation of sound pressure compare to one and two chamber mufflers. And, fixing the muffler at the center enhances sound pressure attenuation.

C. *Design and Optimization of Exhaust Muffler in Automobiles 1M. Rajasekhar Reddy & 2 K. Madhava Reddy international journal of Automobile Engineering Research and Development (IJAuRD) ISSN 2277-4785 Vol.2, Issue 2 Sep 2012 11-21*

The present work aims at improve the Frequency of NSD (Nash Shell Damper) muffler by controlling the noise level of a diesel engine by developing an exhaust muffler for the same, since exhaust noise is the single largest contributor to the overall noise from the engine. The TATA INDICA TURBOMAX TDI BSIV four-cylinder diesel engine car was considered for test purposes. In this study Muffler dimensions are measured through the Benchmarking, to create CAD models. The CAD models are created in CATIA V5 R19, later these CAD models of muffler are exported to HYPER MESH for pre-processing work. Free analysis is carried out on this muffler by FEA Method using NASTRAN Software. The stress and stiffness of the model is studied from the results obtained from analysis to verify the success of the design

D. SABRY ALLAM, Acoustic Modelling and testing of advanced Exhaust system components for automotive engines, 2004 Doctoral thesis, KTH Sweden

The first part of this thesis considers the modelling of sound transmission and attenuation for traps that consist of narrow channels separated by porous walls. This work has resulted in two new models an approximate 1-D model and a more complete model based on the governing equations for a visco-thermal fluid. Both models are expressed as acoustic 2-ports which makes them suitable for implementation in acoustic software for exhaust systems analysis. The models have been validated by experiments on clean filters at room temperature with flow and the agreement is good. In addition the developed filter models have been used to set up a model for a complete After Treatment Device (ATD) for a passenger car. The unit consisted of a chamber which contained both a diesel trap and a Catalytic Converter (CC). This complete model was also validated by experiments at room temperature. The second part of the thesis focuses techniques for plane.

E. M. L MUNJAL, 1987 Acoustics of Ducts and Mufflers, 1987, New York; Wiley interscience

A complete presentation and analysis of the major topics in sound suppression and noise control for the analysis and design of acoustical mufflers, air conditioning and ventilation duct work. Both fundamentals and the latest technology are discussed, with an emphasis on applications. Deals with reactive mufflers, dissipative silencers, the frequency-domain approach, and the time-domain approach.

F. H. BODEN AND M. ABOM, influence of errors on the two-microphone method for measuring acoustic properties in Ducts, 1986 Journal of the acoustical society of America 72[2], 541-549

Using the two-microphone method, acoustic properties in ducts, as, for example, reflection coefficient and acoustic impedance, can be calculated from a transfer function measurement between two microphones. In this paper, a systematic investigation of the various measurement errors that can occur and their effect on the calculated quantities is made.

The input data for the calculations are the measured transfer function, the microphone separation, and the distance between one microphone and the sample. First, errors in the estimate of the transfer function are treated. Conclusions concerning the most favourable measurement configuration to avoid these errors are drawn. Next, the length measurement errors are treated. Measurements were made to study the question of microphone interference. The influence of errors on the calculated quantities has been investigated by numerical simulation.

G. P.O.A.L DAVIES, practical flow duct acoustics, 1988 journal of sound and vibration 124[1], 91-115

Predictions of plane acoustic wave propagation through acoustically reactive flow duct systems are of practical relevance in design studies for industrial, transportation and environmental noise application. Acoustic conditions throughout the flow duct are described by the complex values of the incident and reflected wave amplitudes.

H. Z. TAO and A.F SEYBERT, a review of current techniques for measuring muffler transmission loss, university of Kentucky, USA

The most common approach for measuring the transmission loss of a muffler is to determine the incident power by decomposition theory and the transmitted power by the Plane wave approximation assuming an anechoic termination. Unfortunately, it is difficult to construct a fully anechoic termination.

Thus, two alternative measurement approaches are considered, which do not require an anechoic termination: the two load method and the two source method. Both methods are demonstrated on two muffler types: (1) a simple expansion chamber and (2) a double expansion chamber with an internal connecting tube. For both cases, the measured transmission losses were compared to those obtained from the boundary element method. The measured transmission losses compared well for both cases demonstrating that transmission losses can be determined reliably without an anechoic termination.

I. Ragnar Glav, ON Acoustic Modelling of Silencers, 1994

This thesis considers modelling of the sound transmission properties of internal combustion engine exhaust systems. The work which is presented in 5 separate papers consists of 3 parts: the overall approach, the catalytic converter and the dissipative silencer.

The 4-pole method as an efficient approach to manage the analysis of exhaust systems and silencers is discussed in paper I. Not only does this building-block method most easily include the behaviour of the tail pipe and the engine in the model but it also enables a synthesis of theoretical and empirical analysis. Based on this method a computer program for analysis and design of exhaust systems has been developed. This is also described and a practical example is given.

J. R.M. MUNT, ACOUSTIC Transmission Properties of a Jet Pipe with Subsonic Jet Flow, 1990

A theoretical model is presented for the transmission of acoustic waves out of a semi-infinite circular jet pipe in the presence of subsonic flow out of the pipe. The jet exhaust is assumed to be separated from the ambient, stagnant or co-flowing, fluid by an unstable cylindrical vortex layer. A solution satisfying the full Kutta condition and causality is derived from which the pressure reflection coefficient is calculated. The results show that, for a mismatch in flow between the jet and ambient fluid, the plane wave reflection coefficient achieves values above unity at low frequencies in accordance with the experimental observations. It is also established that at high subsonic Mach numbers the cylindrical jet of fluid outside the pipe may act as a waveguide within certain frequency bands and when this occurs the reflection coefficient decreases significantly with frequency.

K. Y.P. SOH, E.W.T. YAP AND B.H.L GAN, Industrial Resonator Muffler Design

The paper consists of a thorough numerical investigation of the resonance frequency and the transmission loss due to two different frequency 1-DOF Helmholtz resonators attached to a duct for attenuating low frequency noise. The optimum relative spacing and orientation between the two resonators was found out numerically using finite element method. The investigation was further carried out by changing the geometry of the cavity and neck at the optimum relative spacing. Finally, an optimized system of two different 1-DOF Helmholtz resonators has been demonstrated here on the basis of several investigations of relative spacing, orientation and geometry. A significant improvement in overall noise attenuation was observed while compared to a single 1-DOF Helmholtz resonator of a published study.

L. A. J Torregrossa ET. AL, Numerical Estimation of End Corrections in Extended-Duct and Perforated-Duct Mufflers

This paper is concerned with the synthesis of control laws for semi active suspension systems employing artificial intelligence. A review is made of a simple 2-degree of freedom, quarter car model with passive, active and semi active control. An active linear quadratic Gaussian controller and a semi-active balance logic derived controller are then used to develop two artificial intelligence based controllers a semi-active neuro-controller and a semi-active fuzzy logic controller. This paper focuses on the advancement of balance semi-active logic using variable dry friction and the development of fuzzy semi active controllers. The concerns in view are twofold; the reduction in cost of the control system and the anti-chattering nature of the logic. The development is from an engineering perspective and attempts to reduce the well-known schism between theoreticians and users of feedback control.

IV. MUFFLER AND PROPERTIES

A. Muffler

A muffler is a device used to reduce the sound from systems containing a noise source connecting to a pipe or duct system such as combustion engines, compressors, air conditioning systems etc. In internal combustion engines, mufflers are connected along the exhaust pipe as a part of the exhaust system. There are two main types of mufflers, reactive and dissipative.

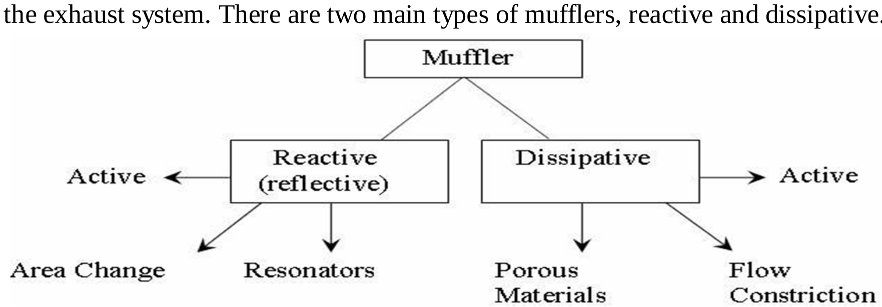


Figure 2-1 Types of Muffler

1) *Reactive:* Reactive mufflers are usually composed of several chambers of different volumes and shapes connected together with pipes, and tend to reflect the sound energy back to the source, they are essentially sound filters and are mostly useful when the noise source to be reduce contains pure tones at fixed frequencies or when there is a hot, dirty, high speed gas flow. Reactive muffler for such purpose can be made quite inexpensively and require little maintenance.
2) *Dissipative:* Dissipative mufflers are usually composed of ducts or chambers which are lined with acoustic absorbing materials that absorb the acoustic energy and turn it into heat. These types of mufflers are useful when the source produces noise in a broad frequency band and are particularly effective at high frequencies, but special precautions must be taken if the gas stream has a high speed and temperature and if it contains particles or is corrosive. Some mufflers are a combination of reactive and dissipative types. Selection of these mufflers will depend upon the noise source and several environmental factors.

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B. Acoustic properties of Mufflers

There are several parameters which describe the acoustical performance of a muffler. These include noise Reduction (NR), Insertion Loss (IL), Attenuation (ATT), and the Transmission Loss (TL). Noise Reduction is the sound pressure level difference across the muffler. It is an easily measurable parameter but difficult to calculate and a property which is not reliable for muffler design since it depends on the termination and the muffler. The Insertion loss is the sound pressure level difference at a point usually outside the system, without and with the muffler present. Insertion loss is not only dependent on the muffler but also on the source impedance and the radiation impedance. Because of this insertion loss is easy to measure and difficult to calculate, however insertion loss is the most relevant measure to describe the muffler performance. Transmission loss is the difference in sound power between the incident wave entering and the transmitted wave exiting the muffler when the muffler termination is anechoic (no reflecting waves present in the muffler).TL is a property fully dependent on the muffler only. Since it is difficult to realize a fully anechoic termination (at low frequencies) TL is difficult to measure but easy to calculate. Attenuation is the difference in the sound power incident and the

$$R_0 = 1 + 0.0133ka - 0.59079(ka)^2 + 0.33576(ka)^3 - 0.6432(ka)^4 \quad \textbf{(2.2)}$$

$$in\,the\,useful\,range\; 0 < ka < 1.5$$

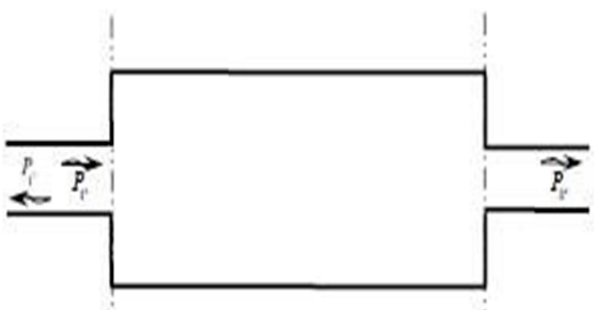
$$Where\; R_0\; corresponds\; to\; the\; zero\; flow\; Mach\; number\; M$$

transmitted through the muffler but the termination need not be anechoic. The acoustic properties measured to validate the models in this thesis are Transmission Loss (TL), Attenuation (ATT) and Reflection Coefficient (REF).

1) Why Transmission Loss (TL)?

a) It is a property of the muffler alone and It is independent of the source (its position and strength)

b) It is easy to predict but difficult to measure since it is very difficult to achieve an anechoic termination



$$TL = 20\log_{10}\left(\frac{P_{1^+}}{P_{2^+}}\right)_{P_{2^-}=0}$$

Figure 2-2 Why Transmission Loss?

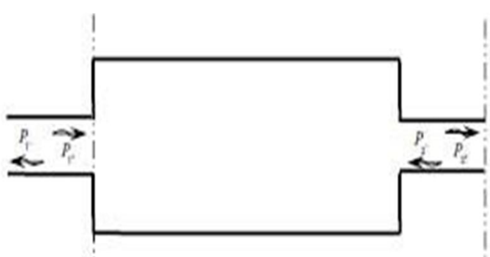
It is a property independent of the inlet and outlet pipe length and solely dependent on the geometry of the muffler itself.

2) Why Attenuation (ATT)?

It is a property dependent on the muffler and also the termination, therefore attenuation predicts the actual behaviour of the muffler after it is installed in a system

a) Helps to verify the acoustic length of the muffler

b) Attenuation is a property dependent on the outlet pipe length so it is helpful to validate the models and their lengths.



$$ATT = 20\log_{10}\left(\frac{P_{1^+}}{P_{2^+}}\right)$$

Figure 2-3 Why Attenuation?

C. Wave Reflection in Flow Ducts

All duct systems consist of sections of uniform duct separated by area and other discontinuities where some of the incident wave energy is reflected, some energy is dissipated, while the remainder is transmitted to the adjacent sections. These area discontinuities can be classified according to their observed behaviour and they normally include terminations of length of uniform duct and sudden expansion or contractions in duct cross-section such as those found at the ends of the expansion chambers and side branches. It should be realized that the one- dimensional models for simulation of plane wave motion in ducts does not take account of the three-dimensional waves arising at these discontinuities.

Evaluation of the pressure reflection and transmission coefficients for each discontinuity involves satisfying the boundary conditions associated with it, conservation of mass, conservation of energy and momentum flux across the discontinuity. The reflective property is expressed by a pressure reflection coefficient R, defined in terms of the component pressure wave amplitude as

R=P-/P+

Since the wave amplitudes p^- & p^+ are complex valued, the reflection coefficient R is also normally, a complex quantity.

A close approximation to the calculated reflection coefficient for an unflanged pipe of radius a is expressed by [Ref-4 Davies (1988)]

A close approximation for R on R0 for an outflow Mach number into a duct is expressed in [Ref10 Munt (1990).]

V. MEASUREMENT OF ACOUSTIC PROPERTIES

The standard technique today for measuring the acoustic plane wave properties in ducts, such as absorption coefficient, reflection coefficient and impedance is the Two Microphone Method (TMM) (Ref-7 Bodén & Å) bom (1984).The sound pressure is decomposed into its incident and reflected waves so that the input sound power can be calculated. Transmission Loss can in principle be determined from measurement of the incident and transmitted power using two microphone method on the upstream and downstream side of the test object provided that a fully anechoic termination can be implemented on the outlet side, which is practically very difficult in low frequency region and especially with flow. Instead the two load technique has been used where the sufficient information for determining the two-port matrix is obtained from two sets of measurements with different loads on the outlet side and for the mean flow measurements carried out at MWL, KTH the two-source location technique was employed by placing the source on the upstream and downstream side of the test object.

A. Two-Microphone Wave Decomposition

The sound field in a straight hard walled duct below the first cut-on frequency will consist only of plane propagating waves .The sound field can be written as [Ref-2 Munjal 1987]

$$p(x,t) = p_+(t - x/c) + p_-(t + x/c) \quad \textbf{(3.1)}$$

Where p=acoustic pressure c=Speed of sound x=spatial coordinate along the duct axis.

The idea behind the two-microphone wave decomposition is that in the low frequency region the sound field can be completely determined by simultaneous pressure measurements at two axial positions along the duct. In the frequency domain, the sound field can be written as

$$\widehat{p}(x,f) = \widehat{p}_+(f)\exp(-ik_+x) + \widehat{p}_-(f)\exp(ik_-x) \quad \textbf{(3.2)}$$

$$\widehat{u}(x,f) = \frac{1}{\rho c}\left[\widehat{p}_+(f)\exp(-ik_+x) - \widehat{p}_-(f)\exp(ik_-x)\right] \quad \textbf{(3.3)}$$

Where, p = Fourier transform of the acoustic pressure, u = Fourier transform of particle velocity averaged over the duct cross-section, x = Length coordinate along the duct axis, f = Frequency,

k± = Complex wave number for waves propagating in the positive or negative x direction, ρ= Density and c = Speed of sound.

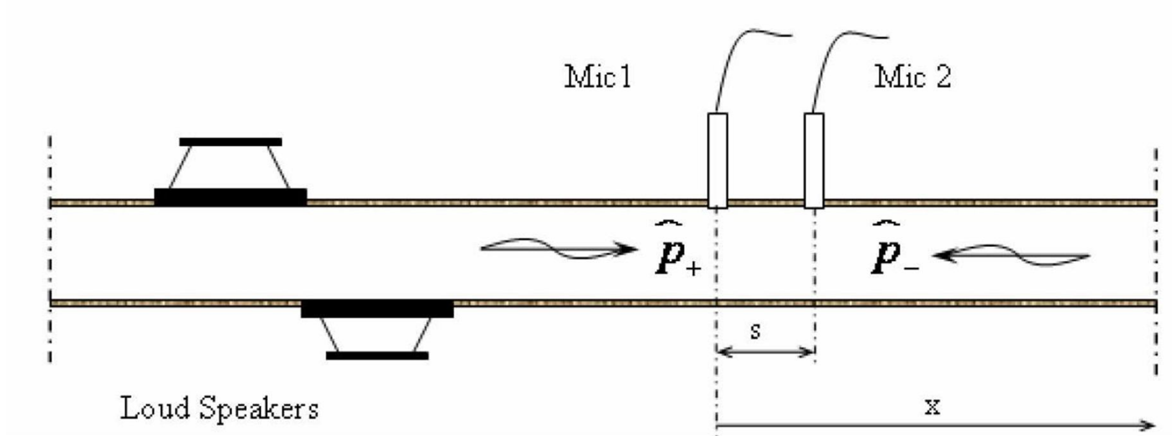


Figure 3-1 Measurement Configuration of Two-Microphone Method

The complex wave numbers can be calculated using the results from Howe [Ref- 18 Howe (1995)] or measured [Ref-1 Allam (2004)] if two microphones on either side are available. When the complex wave numbers are known the incident (p□) and reflected (p□) wave amplitude can be calculated using pressure measurements at two Microphone positions.

$$\hat{p}_1(x,f) = \hat{p}_+(f) + \hat{p}_-(f) \qquad \textbf{(3.4)}$$

$$\hat{p}_2(x,f) = \hat{p}_+(f)\exp(-ik_+s) + \hat{p}_-(f)\exp(ik_-s) \qquad \textbf{(3.5)}$$

Where, s represents the microphone separation as shown in figure, using the above equations $\hat{p}_+$ and $\hat{p}_-$ can be expressed by

$$\hat{p}_+(f) = \frac{\hat{p}_1(f)\exp(ik_-s) - \hat{p}_2(f)}{\exp(ik_-s) - \exp(-ik_+s)} \qquad \textbf{(3.6)}$$

And

$$\hat{p}_-(f) = \frac{-\hat{p}_1(f)\exp(-ik_+s) + \hat{p}_2(f)}{\exp(ik_-s) - \exp(-ik_+s)} \qquad \textbf{(3.7)}$$

According to [Ref-1 Allam (2004)] the following conditions should be fulfilled for successful use of the method
The measurement must take place in the plane wave region.
The duct wall must be rigid in order to avoid higher order mode excitation.
The test object should not be placed closer than 1-2 duct diameters the nearest microphone. This is due to the fact that spatially non uniform test objects could excite higher order modes and therefore create near field effects at the microphones
The propagation of the plane wave must be attenuated. However in practice it is not true even for no flow case because of various mechanisms, mainly associated with viscosity, heat conduction, will cause deviations from the ideal behaviour.
Bodén and Åbom [Ref 3 Bodén & Ref 8 Bodén] showed that the two microphone method has the lowest sensitivity to errors in the input data in a region around KS □ □ (1□ M 2) 2.
Åbom and Bodén [Ref 8 Åbom] stated that to avoid large sensitivity to errors in the input data, the two microphone method should be restricted to the frequency range.

$$\mathbf{0.1\pi\ (1\text{-}M^2) < KS < 0.8\ \pi\ (1\text{-}M^2)} \qquad \textbf{(3.8)}$$

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B. Acoustical one-ports

Reflection co-efficient at the open end (x=0) of an unflanged pipe is given by Holland and Davies [Ref-19 Holland (2000)] as

$$R_0(f) = \frac{H_{12} - \exp(-ik_+ s)}{\exp(ik_- s) - H_{12}} \tag{3.9}$$

Where, H_{12} = Transfer function between microphone 1 and microphone 2,

$k_+ = (2\pi f/c - i\delta)/(1+M)$

$k_- = (2\pi f/c - i\delta)/(1-M)$

δ = Term representing attenuation.

s = Microphone spacing.

Therefore the Reflection coefficient at an arbitrary cross section along the duct is given by

$$R_L(f) = \frac{H_{12} - \exp(-ik_+ s)}{\exp(ik_- s) - H_{12}} \exp\left[2ikL/(1-M^2)\right] \tag{3.10}$$

Ignoring propagation losses and flow, the two pressure signals are identical when $ks=n\pi$,

Where n is any integer; this method will yield poor results when the distance between the microphone is close to multiples of half an acoustic wavelength. Therefore the spacing between the microphones must be kept to within a half wavelength of the highest frequency of interest.

C. Acoustical Two-ports

A two-port is a linear system with an input and output. Assuming plane wave propagation at the inlet and outlet port, the properties of these acoustical two- ports can be determined from theory or by measurements by assuming two state variables at each port. A number of different choices of state variables are possible. However, some state variables are more convenient to use that other, for example, to measure fluctuating density may not be easy. Two state variables which are frequently used is pressure (p) and volume flow (q). This type of formulation is common when having systems with one preferred of energy transport and it is called the transfer matrix formulation and the relation between the input and output of a time invariant and passive two-port can be written as

$$Y = HX \tag{3.11}$$

Where x, y are state vectors at the input and output as shown in figure H = is a [2x2]-matrix

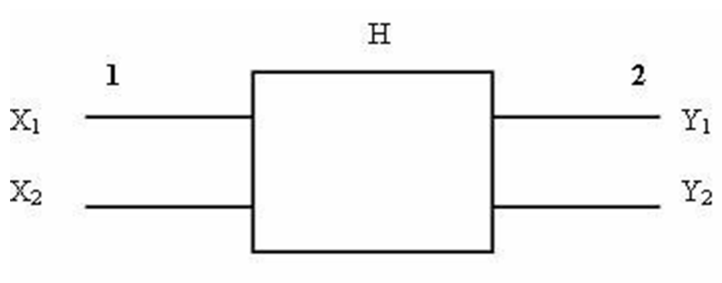


Figure 3-2 Black Box relating two pairs of state Variable x, y

To determine the two-port matrix H from measurements, four unknowns must be determined. To get the four equations needed for complete experimental determination of properties of an acoustical two-port two independent test states (‘and “) must therefore be created. The matrix equation obtained is

$$[Y1\ Y2] = H\ [X1\ \ X2] \tag{3.12}$$

The unknown two-port matrix H can be determined from this equation if and only if

$$\mathbf{Det\ (X) = 0} \tag{3.13}$$

Where, X is the matrix containing the two-port state vectors.

Depending on the coupling of the duct system either the Transfer-matrix (If Cascade) or the mobility-matrix (If Parallel) of the two-port is used

The transfer-matrix form uses the acoustic pressure (p) and the particle velocity

(q) I.e. x = [p q] and y= [p q] .If there is not internal sources inside the two-port element the transfer-matrix could be written in the following form

$$\begin{bmatrix} \hat{p}_a \\ \hat{q}_a \end{bmatrix} = \begin{bmatrix} T_{aa} & T_{ab} \\ T_{ba} & T_{bb} \end{bmatrix} \begin{bmatrix} \hat{p}_b \\ \hat{q}_b \end{bmatrix} \tag{3.14}$$

The transfer matrix can be solved if the below equation is satisfied

$$\det \begin{pmatrix} \widehat{p'}_b & \widehat{p''}_b \\ \widehat{q'}_b & \widehat{q''}_b \end{pmatrix} \neq 0 \tag{3.15}$$

Here "a" and "b" represents two different duct cross-sections. Three basic assumptions concerning the sound field inside the transmission line are made

Only plane waves are allowed to propagate at the inlet and outlet section of the system

The field is assumed to be linear, i.e. the acoustic pressure is typically less than one percent of the static pressure [Ref-20 Åbom (1991)] so that the analysis can be done in the frequency domain.

The two-port system is passive, i.e. no internal sources are allowed. This is a problem concerning flow generated noise. Ab are the characteristic impedances at the duct cross section at "a" and "b".

VI. TEST SET-UP

A. *One Port Measurement*

All one-port experiments were carried out at room temperature in the test setup at the Acoustic Competence Centre (ACC).The setup consists of one loudspeaker as an acoustic source as shown in figure. Fluctuating pressures we measured using three ½ inch condenser microphones thread mounted on the duct wall. The measurements were carried out using burst random signal (with 60% burst time) excitation and with different number of frequency domain averages (20, 50,100 averages).The transfer function between the three microphone positions are measured and used to estimate the transfer matrix components.

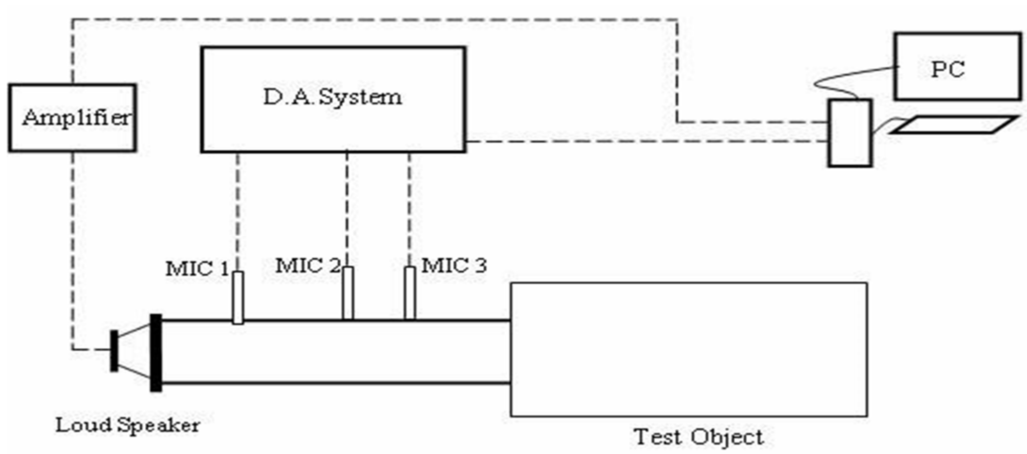


Figure 4-1 Layout of One-Port Test Object

Three microphones were used to in order to cover the frequency range 100-2000 Hz while fulfilling the equation [0.1π(1-M^2)< ks < 0.8 π (1-M^2)].The distance between the microphone 1 and 3 was 15cm giving approximately the frequency range 100900Hz.The distance between microphone 2 and 3 was 5cm giving approximately the frequency range 3502600Hz.

B. Two Port Measurements with Zero Mean Flow

All two port measurements with zero mean flow were carried out in room temperature in the test setup at the Acoustic Competence Center (ACC).The test setup is as shown in Figure 18. The inlet and outlet pipes used during the measurements were made of standard steel with wall thickness 1.5 mm. The inlet and outlet pipe diameter was 51 mm and one loud speaker was used as an excitation source. Fluctuating pressures were measured using six ½ inch condenser microphones thread mounted on the duct walls. The microphone placement and the test setup are as shown in figure below. The measurements were carried out using Burst random (60% burst time) as the excitation signal and with different frequency domain averages. Since the measurements were done with zero flow, the number of averages did not really influence the quality of the result therefore 100 FDA were used during all these measurements. The two- port data was obtained using the two load technique by altering the loads at the termination as described in Section 3.3.2.

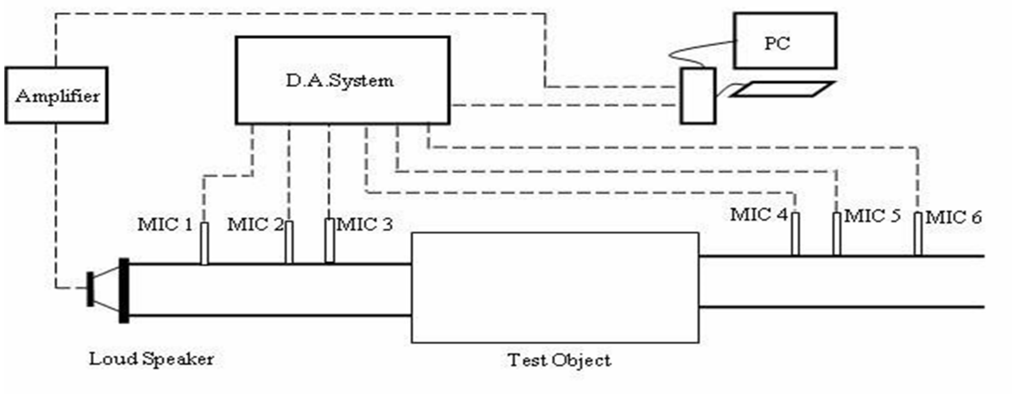


Figure 4-2 Layout of Two-Port Test Object with Zero mean flow

One of the four microphone signal is taken as the reference and the transfer function between the reference and the other three microphones are measure and used to estimate the transfer matrix components.

The same microphone separations described in the previous section for the one- port measurements were used, giving again the frequency range 100-900Hz for 15cm distance and 350-2600Hz for 5cm distance.

C. Two port Measurements with Mean Flow

The two-port measurements with mean flow were carried out at room temperature using the flow acoustic test facility at The Marcus Wallenberg laboratory for Sound and Vibration research at KTH. The muffler configurations were the same as used in ACC but only one inlet and outlet pipe diameter (67mm) was chosen to fit the test rig duct diameter. The loud speakers were divided equally between the upstream and downstream side. The microphone placement and the test setup are as shown in Figure 19. Fluctuating pressures were measured by six ¼ inch condenser microphones flush mounted on the duct wall. The measurements were carried out using stepped sine excitation in the frequency range 100 -1200 Hz using different frequency steps and different Frequency Domain averages. The two-port data was obtained using the source switching technique as described in section 3.3.1, where the measurements were made using the upstream loud speakers on and downstream loud speakers off and vice- versa.

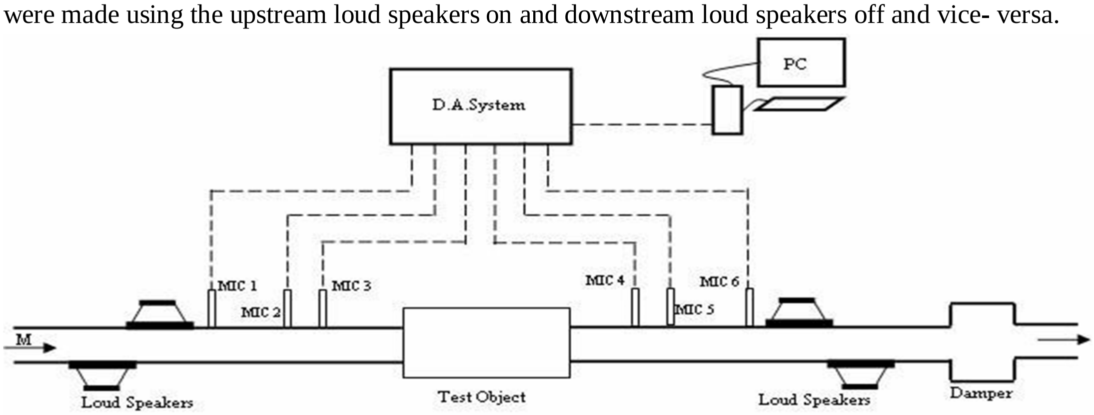


Figure 4-3 Layout of Two-Port Test Object at MWL, KTH

The mean flow velocity was measured with a Pitot tube placed at the centre of the duct. It was assumed that flow the fully developed a boundary layer and the mean velocity = 0.8*Maximum Velocity (measured in the centre of the duct) [Ref-17 Schlichting (1968)]. Once the peak velocity was measured the Pitot tube was removed from the duct before taking the acoustic measurements as it might disturb the flow.

The flow velocity on the upstream side of the test object was measured separately before and after the acoustic measurements and average value was used.

As an additional data, the pressure drop across the test object was also measured for different flow speeds using the Pitot tube. The transfer function between the reference signal and the microphone signal was measured and used to estimate the transfer matrix components.

Two different microphone spacing were used to cover a wide frequency range 100-

1200 Hz while fulfilling the equation $[0.1\pi\,(1-M^2) < ks < 0.8\,\pi\,(1-M^2)]$. The distance between microphone 1 and 2 was 10cm giving approximately the frequency range 170-1300 Hz and the distance between microphone 1 and 3 was 50cm giving approximately the frequency range 40280 Hz.

D. Microphone Calibration

For ordinary sound pressure measurements, only amplitude calibration is enough but for two microphone method we need both amplitude and phase calibration. The fluctuating pressures measured at each position have been corrected using the relative calibration between the microphone channels. Assuming that we have plane waves in a duct the sound pressure amplitude will be constant over the duct cross section and the sound pressure is measured by all microphones would give the same pressure amplitude with zero phase shifts

.However there will in practice be a deviation from this ideal case due to the measuring chain, amplifiers, and cables etc., which introduce amplitude and phase shifts. Relative calibration of the microphone of the microphone measurement chain is therefore needed. In order to calculate the transfer-matrix equation, the transfer function between the microphones and the electrical loud-speaker signal, i.e. Hr1,

Hr2, Hr3, Hr4, Hr5, and Hr6 are needed. It is sufficient to

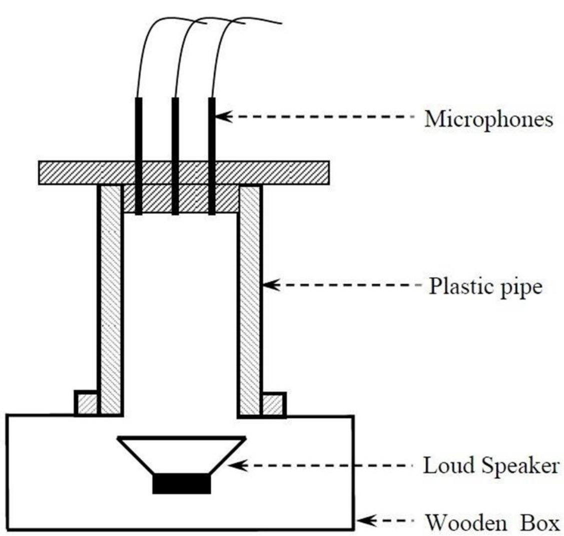


Figure 4-4 Calibration Tube

Measure the transfer function between these microphones and a reference microphone say microphone 1, H12, H13, H14, H15, and H16.

A special calibration tube a shown in figure above was used to measure the transfer functions between the reference microphone and the other microphones. The calibration tube consists of a loudspeaker, a steel pipe, which has the same diameter as the test object and a microphone holder for six ¼ inch microphones. The holder is made-up of a plastic material to avoid possible grounding errors between the microphones. The length of the steel pipe is preferably short to minimize the number of resonance in the pipe.

E. Flow Noise Suppression

An efficient way of suppressing turbulent pressure fluctuations is to use a reference signal, which is uncorrelated with the disturbing noise in the system and linearly related to the acoustic signal in the duct [Ref-9 Åbom (1989)]. A good choice for the reference signal is to use the electric signal driving the external sources as a reference. Deviation from a linear relation between the reference signal and the acoustic signal in the duct can for instance be caused by non-linearity of the loud speakers at high input amplitudes, temperature drift and non-linearity of the loudspeaker connections to the duct at high acoustic amplitudes. One possibility is to put an extra reference microphone close to a loud-speaker or even in the loud speaker box behind the membrane i.e., without contacting the flow. Otherwise one of the measurement microphones can be used as a reference.

The disadvantage of this technique is that one will get a minima's at the reference microphone at certain frequencies or poor signal to noise ratio. To solve this problem one can use the microphone with the highest signal-to-noise ratio as the reference. In this work the electronic signals driving the loudspeakers were used as the reference.

VII. RESULTS AND DISCUSSION

A. Open End Reflection

Figure 5-1 Test Set-up for reflection coefficient measurement at an open end

The reflection coefficient at the outlet cross section of an open pipe end without mean flow was measured as described in Section 4.1.

We can see from these figures that the real and imaginary parts of the measured reflection coefficient have a very good agreement (there is no frequency shift therefore correct acoustic length) and the phase of the reflection coefficient agrees well with the theoretical models which show that the open out flow end corrections used in BOOST SID are correct.

By determining the complex value of the reflection coefficient R at the opening, the phase of R is converted to an end correction as described in Section 2.3.2.

Davies gives the below expression for the reflection coefficient at a pipe opening at zero flow,

$$R_0 = 1 + 0.0133ka - 0.59079(ka)^2 + 0.33576(ka)^3 - 0.6432(ka)^4$$

$$\textit{in the useful range } 0 < ka < 1.5$$

$$\textit{Where } R_0 \textit{ corresponds to the zero flow Mach number } M$$

The and Davies suggests the following Open End corrections which is also used in BOOST SID,

$$\frac{l_0}{a} = 0.6133 - 0.1168(ka)^2 \qquad ka < 0.5$$

$$\frac{l_0}{a} = 0.6393 - 0.1104ka \qquad 0.5 < ka < 2$$

The figure below shows the measured end correction as described in, compared with the theoretical value given in the above equation at zero flow. The predicted end corrections agree well with the experimental value.

For example an expansion chamber of Length, L=500mm and Diameter, D=200mm will have a transmission loss curve as shown in the figure below at M=0 and T=293k where the speed of sound, c=343m/s.The first four maxima calculated from Equation 5.1 for n=1, 3, 5 & 7 matches the predicted curve at 172Hz, 515Hz, 856Hz and 1200Hz respectively.

Transmission Loss of an Expansion Chamber L=500mm & D=200mm at M=0 & T=293k

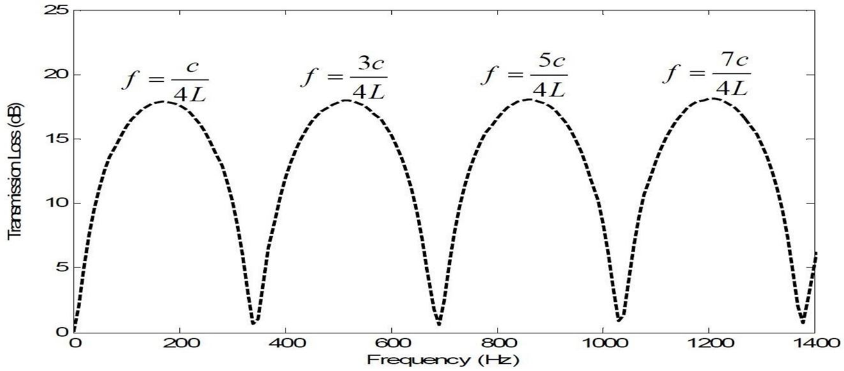


Figure 5-2 TL of a Simple Expansion Chamber

Transmission Loss measurements were performed using the Two-load technique as discussed in the Section 4.2. The test objects used in this measurement are as in the table below,

Muffler ID	Inlet Pipe Dimension in mm		Chamber Dimension in mm		outlet Pipe Dimension in mm		Expansion KARAL End correction in mm	Contraction KARAL End correction in mm
	Dia	Len	Dia	Len	Dia	Len		
Dia 100	51	96	100	500	51	96	8	8
Dia 200	51	96	200	500	51	96	14.7	14.7
Dia 300	51	96	300	500	51	96	17	17

Table 4. Dimensions of the simple expansion chambers

B. Geometric Model to Acoustic Model

Converting the geometric model of a muffler to an acoustic model is an important step in linear acoustic modelling. For example conversion of a diameter 200mm simple expansion chamber as in Table 4 to a BOOST SID model was made as shown below,

1) Geometric model of Diameter 200mm Expansion Chamber

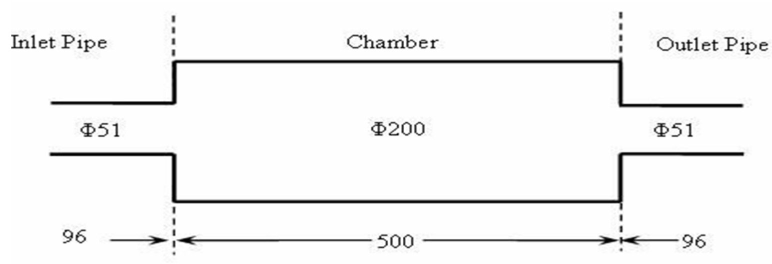


Figure 5-3 Geometric Model

Suitable end corrections are applied to this geometric model. In our case, KARAL end corrections as shown in Table 2 & the values as in Table 4 are applied at the junctions where the inlet and outlet pipes are connected to the chamber and at the open end boundary.

2) Geometric Model with End Corrections Applied

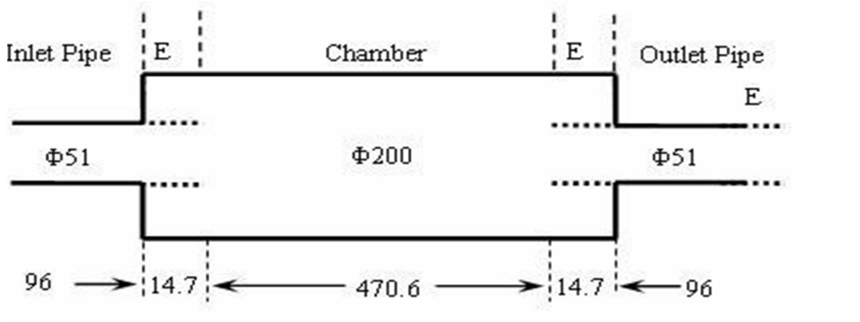


Figure 5-4 Geometric Model with End Corrections

As a result of these end corrections, the length of the inlet and outlet pipes are extended by a length l, equal to the end correction and consequently, the length of the chamber is reduced as shown in figure below. The area between the extended pipe and the chamber forms quarter wave resonators on both ends as shown in figure.

3) Acoustic Model

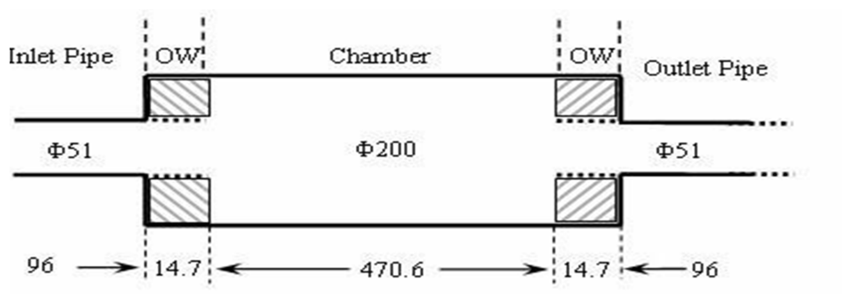


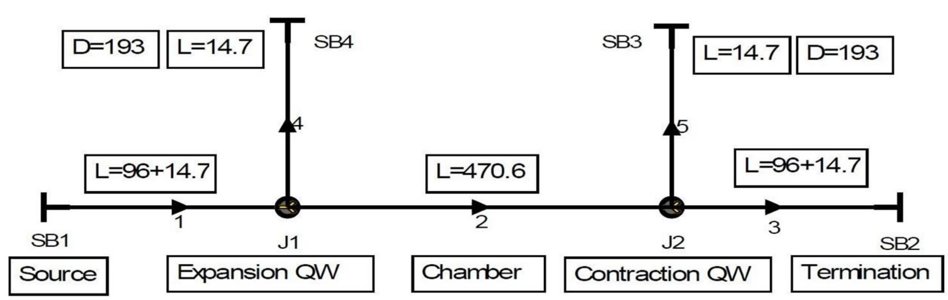
Figure 5-5 Acoustic Model

Acoustic Inlet Pipe length=Geometric Inlet length + End correction.
Acoustic Chamber length =Geometric chamber length – (2 x End correction). Acoustic Outlet Pipe length=Geometric Outlet length + End correction.
Quarter wave Resonators:
Length=14.7mm & Diameter= (2002-512)1/2mm

C. BOOST SID Model

Acoustic modelling of this expansion chamber in BOOST SID can be done in three different ways,

1) *Modelling with Quarter Wave Elements:* The end correction lengths are modelled as separate quarter wave resonators as shown in figure below as quarter wave resonators and Junctions as shown in the figure below,



These models are not useful when there is mean flow inside the muffler because the linear acoustic tool assumes that there is no flow inside the resonator which is not true if the resonator is inside the muffler.

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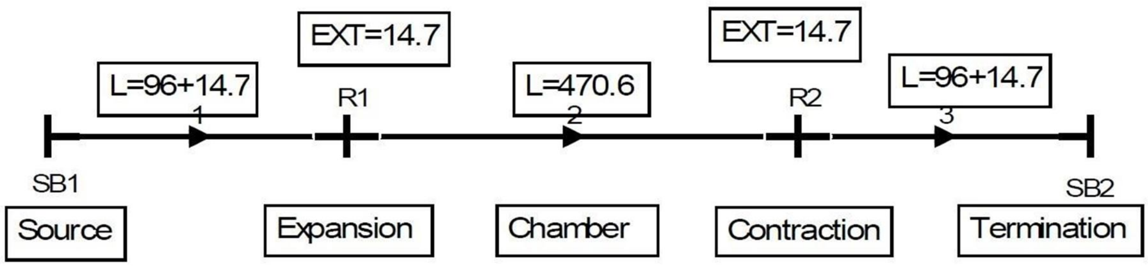
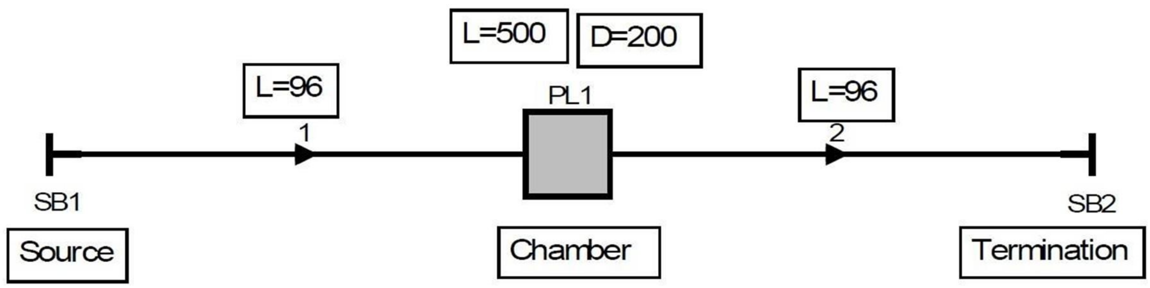
2) *Modelling with Restrictions:* The end correction lengths are modelled as extensions in the restriction elements as shown in figure below. Because of the flow limitation of the Quarter Wave model, this model with restrictions and extensions are used in this thesis.



3) *Modelling with Plenum Chambers:* By modelling the expansion chamber with the plenum element, No end corrections have to be applied to these models since the higher order mode (Both Circumferential & Radial) effects which are excited above the first cut-on frequency are already accounted for.



There are two types of plenum chambers available in BOOST SID

a) *Extended Concentric:* Accounts the Radial Mode effects.
b) *Flush Eccentric:* Accounts the Circumferential Mode effects.

As described in Section 2.4, the cut-off frequency for circumferential and radial higher order modes in circular ducts are given by equation 2.6 and 2.7 respectively. The table below gives the cut-off frequency above which the first circumferential and first radial modes propagates in these three expansion chambers at 293k & M=0.

Muffler Diameter in mm	Cut-off Frequency (Hz) 1st Circumferential Mode	Cut-off Frequency (Hz) 1st Radial Mode
100	2008	4179
200	1004	2089
300	669	1392

Table 5. Cut off frequency of Expansion chambers of different diameter.

Our frequency range of interest is 0 to 2000 Hz so we are not concerned about the higher order modes propagating above 2000Hz. It can be seen from Table 5 that the first circumferential mode is cut on at 1004Hz and 669 Hz for a Diameter 200mm and 300mm expansion chambers respectively and from Figure

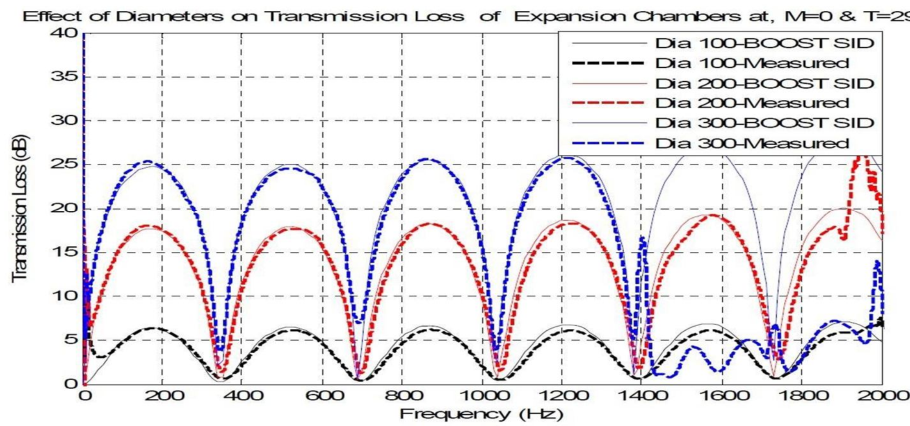


Figure 5-6 Effects of Diameter on Transmission Loss for Expansion Chambers

D. Expansion Chambers with Walls and Extensions

The next configurations tested were mufflers with walls in the middle as shown in the figures below and these mufflers can be made more complex by having extensions in the inlet and outlet side as discussed in the previous chapter.

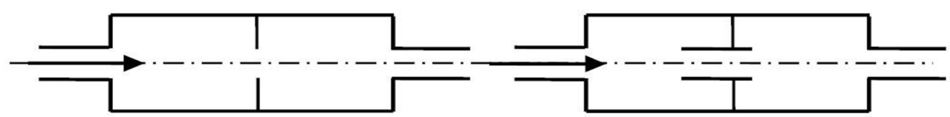


Figure 5-7 Possible Expansion chamber with walls and Extensions

VIII. CONCLUSION

A. A number of muffler configurations starting from simple expansion chambers to complex geometry have been investigated with and without the mean flow effects.

B. When validating the acoustic models created using a linear acoustic tool transmission loss measurement is not sufficient as it is independent of the inlet and outlet pipe lengths which make it difficult to study the effect of the end corrections. Sometimes other properties like attenuation, reflection coefficient has to be measured and compared with the predictions. 1-D wave theory applies for Mufflers having concentric inlet and outlet pipes till the first radial higher order mode is excited as in equation 2.7.

C. For mufflers having eccentric Inlet and Outlet pipes, the circumferential higher order modes are excited above the first cut on frequency as in equation 2.6

D. Reverse Flow Mufflers: The above mentioned condition applies for flow reverse mufflers since the inlet or outlet or both the pipes are flush eccentric to the chamber and the predicted transmission loss result depends on the length of the reverse chamber.

E. Higher the Transmission Loss, the more difficult it becomes to measure. During this thesis some of the mufflers had a very high Transmission Loss (nearly 100dB) and the measurement results was not good.

F. The expansion model used in BOOST SID includes the flow related losses whereas the contraction model does not include these losses which can be seen from the mean flow measurements in Section 5.3.

G. BOOST SID does not include flow inside the Quarter Wave resonators. Since the mufflers investigated in the later part of the thesis is entirely constructed using QW resonators the BOOST SID predictions with and without flow will yield the same TL result which might not agree with the measurements. For example the influence of flow on the flow reversal and Helmholtz muffler are inconclusive from our investigations.

H. When the acoustical openings inside a muffler are too close to each other or close to walls then acoustical length of the elements change therefore measurements were performed to find the end corrections when the open end is close to a wall.

I. Secondary or Internal sources cannot be used in BOOST SID. In a complex muffler when the flow velocities are high then the flow inside the muffler generates a considerable amount of noise which alters the quality of the measurement results.

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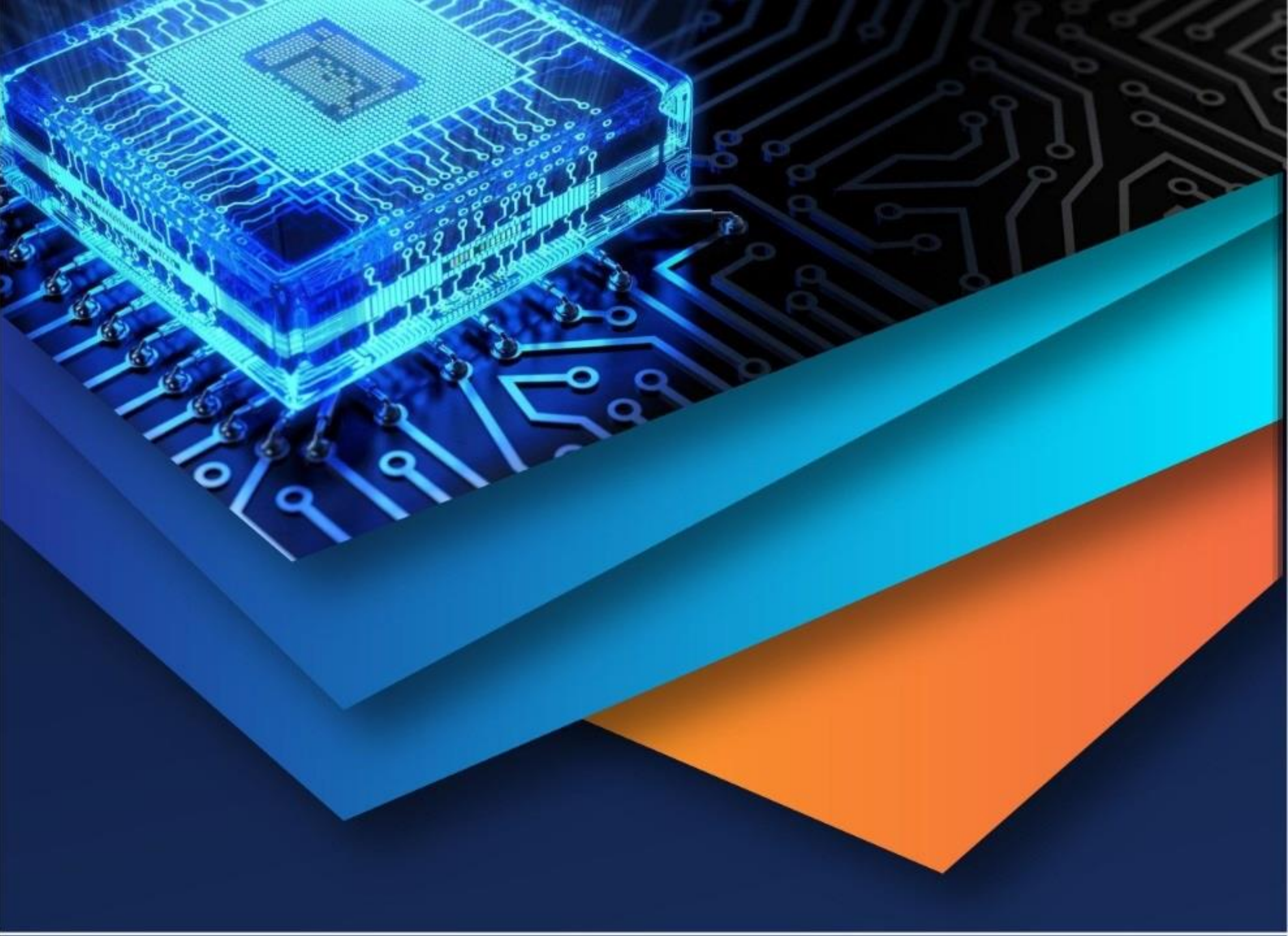

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