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Vibration Analysis of an Automotive Silencer for Reduced Incidence of Failure

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Abstract— **The Engine exhaust is the major source for contributing noise in automobile. There are many challenges in designing the muffler system as size, engine back pressure, noise level, emission norms and also cost. Sufficient insertion loss at engine frequency and first few harmonics are important. Recent efficient muffler systems are designed considering linear plane wave theory using transfer matrix method. Development of fibrous material strands that is used in hot exhaust system without binders has led to use of combination of muffler and exhaust system. Particulate matter filters and air cleaners also modified acoustically. For all these to happen exhaust muffler designers need simple and fast modeling tool, especially at the initial stages. FEM and Boundary element methods are time consuming. This paper is about plane wave based models such as transfer matrix method that has advantage of building fast prototype for muffler design, find out transmission loss and compared with experimental setup.**

***Index Terms*— Automotive Muffler, Engine Exhaust Noise, Muffler/Silencer, Noise Level, Transmission Of Sound, Transmission loss, Vibration Induced Noise.**

I. INTRODUCTION

Major contribution for creating noise in automotive engines is the exhaust system. To reduce noise Mufflers/ silencers are used. Mufflers are commonly used in a wide variety of applications Mufflers attenuate the noise levels carried by the fluids and radiated to the outside atmosphere by the exhausts. TO comply the environmental legislations mufflers designers have to use high performance and that much reliable techniques. Number of techniques is available to for designing, evaluating and testing of mufflers. Empirical, analytical and numerical techniques have been used and proved reliable under controlled conditions. To design complete muffler system is a very complex task. Particular performance like back pressure, insertion loss, size, reliability, and important parameter cost are to consider. Numerical techniques, such as the Finite Element Method (FEM) and the Boundary Element Method (BEM) have proven to be convenient for complex muffler geometries. Although these methods are applicable to any muffler configuration, when the silencer shape becomes complex, the three-dimensional FEM requires a very large number of elements and nodes. This becomes a long and lengthy hence tedious data preparation. Although high speed computational and storage machines exist, the use of FEM or BEM for muffler design is restricted to trained personnel and is commercially expensive, in particular for preliminary design evaluation. Most muffler manufacturers are small and medium companies with a limited number of resources. For prediction of radiated noise from engine exhaust systems it needs a model of the acoustic behavior of the intake/exhaust system and a model of the engine cycle source parameters. These models are analyzed in the frequency domain or in the time domain. However, the evaluation of the source characteristics remains a challenge. Now recently Sathyanarayana and Munjal (2000) developed a hybrid approach to predict radiated noise that uses both the time domain analysis for the engine and the frequency domain analysis for the mufflers. For the frequency analysis of the muffler they have used the transfer matrix method. Also in other recent studies new algorithms that can reduce the computation time in predicting the performance of mufflers have been reported (Dowling and Peat in 2004). However, it is required a previous identification of sub-systems of two-port acoustic elements before applying the algorithm. In this paper, the fundamentals of the Transfer Matrix Method are summarized and the method is applied to different muffler configurations for the prediction of Transmission Loss. A set of measurements was taken and the results are compared with the numerical predictions.

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II. NEED FOR ANALYSIS

1.1 Introduction

The Automobile silencer under study belongs to a popular 2-Wheeler manufacturer in India with the rated HP of the engine up to @13.5HP. The exhaust gases coming out from engine are at very high speed and temperature. Silencer has to reduce noise, vibrations. While doing so it is subjected to thermal, vibration and fatigue failures which cause cracks. So it is necessary to analyze the vibrations which would further help to pursue future projects to minimize cracks, improving life and efficiency of silencer.

1.2 Aim: -

1. To analyze the heat-flow through the inlet port of the silencer within the exhaust system through the exhaust port open to atmosphere. This might help to minimize the fatigue failure or thermal creep (through redesign) manifested by Heat induced due to the impinging and pulsating flow of hot gases during its passage through the tail end of the Automotive Exhaust System (Silencer)

2. Minimize the incidence of cracks on the end covers or other members of silencer, improving the usable life-span of the silencer

2.3 Objective:-

1. Identify and study using software tools (for simulation/ analysis), the nature and characteristics of vibrations.
2. Evaluate the influence of the vibrations over the design of the silencer
3. Review the existing design and consider improvement for negating the harmful influences of the phenomena

2.4 Industrial relevance:-Every exhaust system of an industrial or automobile system where hot gases discharge from the combustion chamber into the surrounding atmosphere at relatively high velocities has a silencer as an integral part of the system. The Automotive silencer attempts to reduce the audible noise levels in the proximity of the system to acceptable limits for human comfort. While doing so, it has to withstand stresses induced due to heat and other factors such as vibration, fatigue etc.As such, any improvement made to the silencer would directly enhance the function of silencer with marked improvement in its effective life-span.

2.5 Approaches, Techniques :-

[1]. Numerical approach:-

This is global approach, based on the loading definition, the modeling of the constitutive law and of the damage and a failure criterion. This approach is applied on cylinder heads and on exhaust manifolds submitted to transient thermal loading and permits to predict the cracked area as well as the lifetime.

[2]. Computational approach:-

This presents a computational approach for the lifetime assessment of structures. One of the main features of the work is the search for simplicity and robustness in all steps of the modeling, in order to match the proposed method with industrial constraints. The proposed method is composed of a fluid flow, a thermal and a mechanical finite element computation, as well as a final fatigue analysis.

[3]. Experimental set up:-

With the use of experimental set-up we can analyze the fatigue and vibrations for silencer. In lab silencer would be tested to give results required. Of above approaches computational approach will give results more close to practical values through simulation/ analyses. The technique would deploy any of the following software tools: Fluent, Star CCM, CFD++, Patran/ Nastran, ANSYS, MSC fatigue or any compatible CAE software.

2.6 Benefits of using a CAE software:-

It has, intuitive graphical interface with direct access to CAD geometry, advanced meshing, integration with other compatible software for solving. It is optimized for large scale systems, assemblies, dynamics and NVH simulations. It has graphical interface with direct access to CAD geometry, most suitable for fatigue analysis. Here, we shall employ software tools like CFD++, Star or Fluent for analysis limited to the flow of flue gases.

2.7 Methodology:-

1) Creation of Geometry for silencer.
2) Importing the geometry for meshing.
3) Solving for the meshed model with constraints and boundary conditions.
4) Viewing the results during post-processing.
5) Interpretation over the results.
6) Recommendations

The purpose of the exhaust system is simple: to channel the fiercely hot products of fuel combustion away from the engine or generator and the car's occupants and out into the atmosphere. The exhaust system has a secondary purpose- to reduce the amount of noise made. The exhaust gases leave the engine at incredibly high speeds. Moreover, with the opening and shutting of the exhaust valves with each cycle of combustion for each cylinder, the gas pressure alternates from high to low causing a vibration- and hence sound. Silencer has to muffle the vibrations of the exhaust gases, reduce their velocity and thus reduce the amount of noise emitted from the engines. The pulsating flow from each cylinder's exhaust process of an automobile petrol or diesel engine sets up pressure waves in the exhaust system-the exhaust port and the manifold having average pressure levels higher than the atmospheric. This varies with the engine speed and load. At higher speeds and loads the exhaust manifold is at pressures substantially above atmospheric pressure. These pressure waves propagate at speed of the sound relative to the moving exhaust gas, which escapes with a high velocity producing an objectionable exhaust boom or noise. A suitably designed exhaust silencer or muffler accomplishes the muffling of this exhaust noise. Practically, the exhaust gas mass is forced through the pipe after leaving the engine. Its momentum forces the change in the direction of motion, or in the expansion or contraction of the end pipe. This gas produces some resonance in such frequency range that might cause fatigue failure to the exhaust pipe when the resonance exists continuously. Without the consideration of these cases, the development of the exhaust system will be incomplete.

III. MUFFLER DESIGN

3.1 Introduction

There are two main types of mufflers designs namely as absorptive and reactive. Generally automotive mufflers will have both reactive and absorptive properties the reactive or reflective mufflers uses destructive interference, the phenomenon to reduce noise. Meaning of partial interference is when engine fires and blow out noise that time noise is reduced itself partially by designing. It is the theory of wave length in the phase and out of phase. Sound waves in phase that is phase angle between them is 0 degree then the noise will increase whereas if they are out of phase i.e. 180 degrees then sound intensity will reduced automatically. Hence partial reduction of noise can be achieved. . Reflection is occurring where there is a change in geometry or an area discontinuity. In reactive muffler (as shown in Figure 1), commonly series of resonating and expansion chambers that are designed to reduce the sound pressure level at certain frequencies. The inlet and outlet tubes are kept offset to some extent and have perforations that allow sound pulses to scatter out in numerous directions inside a chamber resulting in more destructive interference.

Reactive mufflers are used widely in car exhaust systems where the exhaust gas flow and hence noise emission varies with time scale and also where the power is more. They have the ability to reduce noise at various frequencies due to the numerous chambers and changes in geometry that the exhaust gasses are forced to pass through The down side to reactive mufflers is that larger backpressures are created, however for passenger cars where noise emission and passenger comfort are highly valued reactive mufflers are smooth less vibrant hence ideal and can be seen on most passenger vehicles on our roads today.

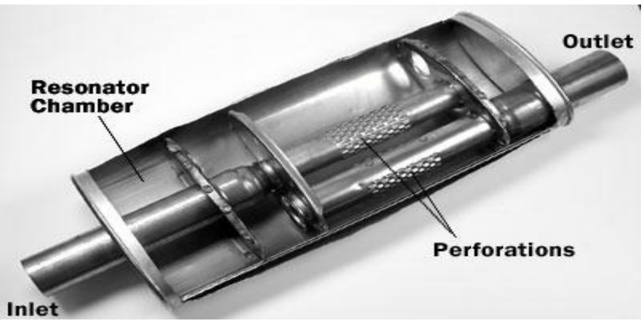

Fig. 3.1 Reactive automotive muffler

An absorptive or dissipative muffler, as shown in Figure 2, it uses the absorption phenomenon where kinetic energy of the firing was absorber in porous material. Sound waves are reduced as their energy is converted into heat energy in the absorptive material. A typical absorptive muffler consists of a straight, circular and perforated pipe with uniform pattern of holes that is encased in a larger steel housing. Between the perforated pipe and the casing is a layer of sound absorptive material like porous material that absorbs some of the pressure pulses.

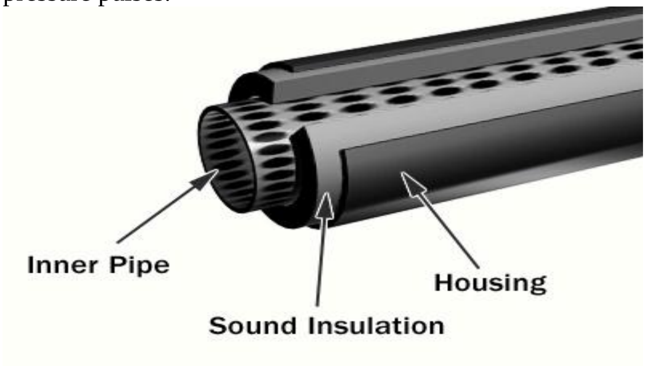

Fig. 3.2 Absorptive automotive muffler

Absorptive mufflers have relatively less backpressure then reactive mufflers; this is because of the porosity in the material and property of material. This material which is used in perforated sheet and steel housing also reduces the emissions produced by fuel burning. There are Nox and Sox which are emitted through engine and they are required to be controlled.

3.3 POSSIBLE MUFFLER DESIGNS

There are many different designs available for muffler and they are used for different purpose as automobile, locomotives, guns, generators etc. Let us see various muffler designs and their key features one by one. Automotive mufflers generally have a circular or elliptical cross section. A circular shaped cross section is very good in a vehicle as it delays the onset of higher order modes Most formulas that are used to predict the transmission loss of a muffler assume plane wave propagation. The properties of the following designs are only valid up to the cut off frequency, where higher order modes occur. Generally for all mufflers maximum transmission loss occurs at odd multiples of a quarter wavelengths.

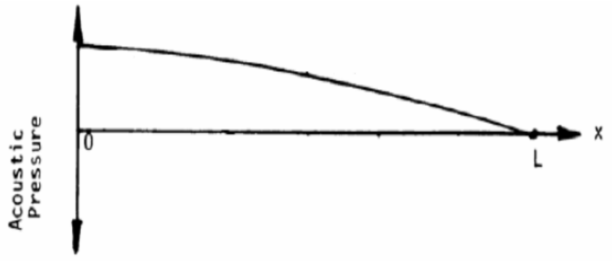

Fig. 3.3 Quarter wavelength

Silencing element that is used generally for intake and exhaust mufflers is the expansion chamber. It consists of an inlet tube, an expansion chamber and an outlet tube as shown in Figure 4. The inlet and outlet tubes may be coaxial known as a concentric expansion chamber or offset known as an offset expansion chamber.

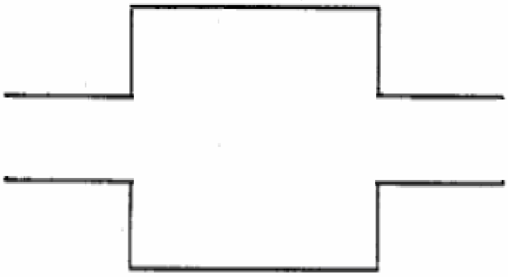

Fig. 3.4 Simple Expansion Chamber

Due to sudden expansion and contraction of the sound wave there is change of phase of waves and they nullify effect of peak sound. Attenuate high frequency sound is reduced to minimum so well as it 'beams' straight through the muffler. Here back pressure and insertion loss can be controlled easily by changing diameter and length of pipe. Change in length cause whirling pattern of sound. Back pressure is maintained by the diameter of pipe and also both inlet, outlet pipe diameters.

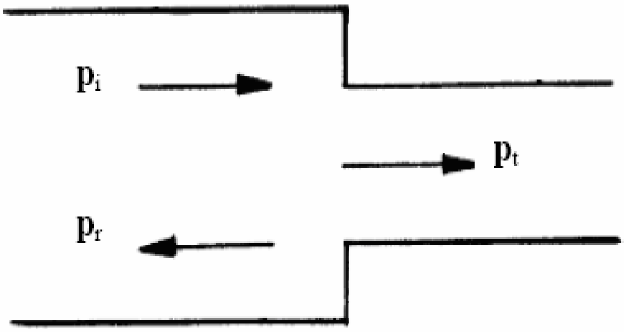

Fig. 3.5 Sound pressure wave propagation.

Irrespective of the features there are some drawbacks also. The length of the chamber should be at least 1.5 times the diameter. Similar to a standard expansion chamber is the extended inlet and outlet expansion chamber, where the inlet and outlet tubes are extended into the expansion chamber as shown in Figure 6. The benefit of such a design is that part of the chamber between the extended pipe and the sidewall acts as a side branch resonator therefore improving the transmission loss. The greater the protrusion into the muffler the greater the transmission loss however the inlet and outlet tubes should maintain a separation space of at least 1.5 times the diameter of the chamber to ensure the decay of evanescent modes.

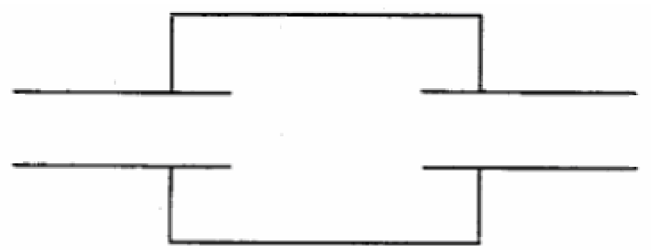

Fig. 3.6 Expansion chamber with an extended inlet and outlet

Noise can be further attenuated by the addition of porous material inside the expansion chamber whilst maintaining the same muffler dimensions. Sound waves loose energy as they travel through a porous medium. The absorptive material (porous material) causes the fluctuating gas particles to convert acoustic energy to heat. The main benefit of a straight through absorptive silencer, as shown in Figure 7, is that insignificant backpressure is generated therefore improving vehicle performance. The perforated tube is used to guide the exhaust flow and avoids the creation of turbulence as is found in an expansion chamber.

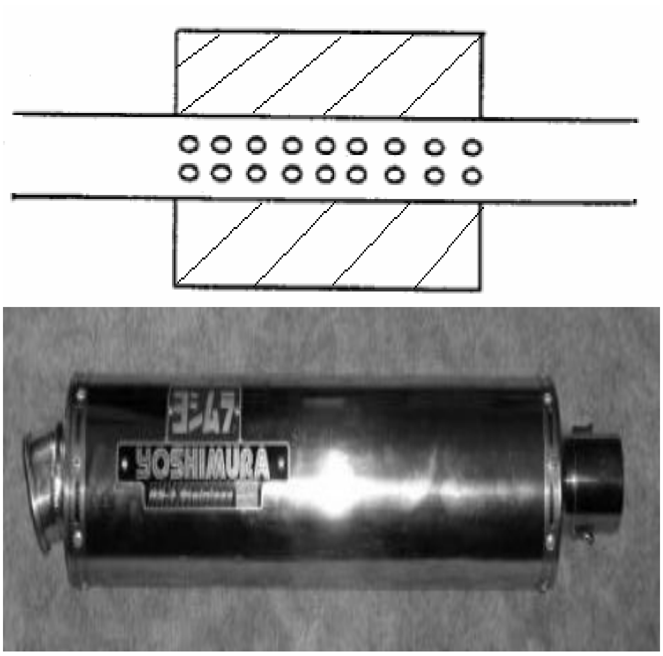

Fig. 3.7 Straight through absorption muffler

The material used to guide the exhaust flow, yet allow sound waves to escape, is usually perforated steel with an open area of approximately 20%. An absorptive silencer produces a more consistent transmission loss (TL) curve as shown in Figure 8. The expansion chamber TL curve is typically domed in shaped and as can be seen the absorptive material not only irons out these humps but also increases transmission loss dramatically especially for the higher frequencies.

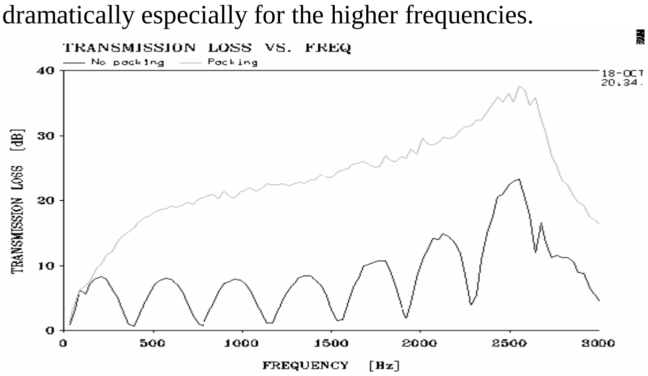

Fig. 3.8 Comparison of a muffler with and without absorptive material

A side branch resonator as seen in Figure 9 is a muffling device used to control pure tones of a constant frequency. It generally takes the form of a short length of pipe whose length is approximately a quarter of the wavelength of the sound frequency to be controlled. A Helmholtz resonator is similar to a side branch resonator the only difference being that there is a backing volume joined to the connecting orifice.

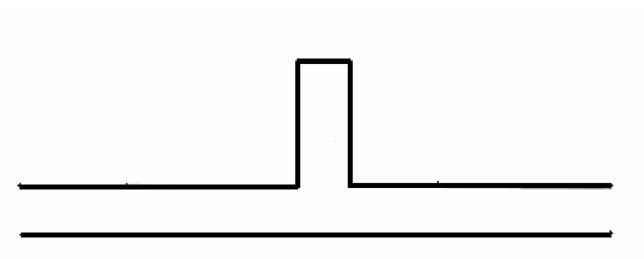

Fig. 3.9 Side branch resonator

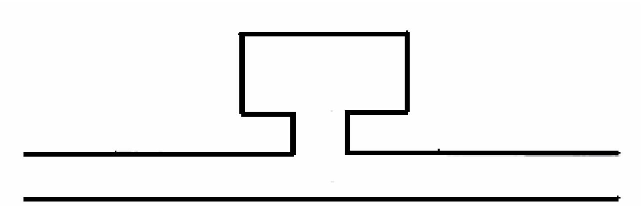

Fig. 3.10 Helmholtz resonator

If it is found that the un muffled exhaust noise spectrum has noticeable peaks a resonating chamber (concentric resonator) Figure 11, to target these specific resonant frequencies. The resonating chamber style of muffler is extremely efficient in providing noise control for a specific frequency band; however the attenuation band is very narrow. This characteristic is not very useful in automotive exhaust system where attenuation is needed over all frequencies. A side branch resonator may be however used in addition to a muffler to treat a particular problem frequency If a broader and improved attenuation spectrum is required multiple resonators should be used. Each chamber is designed to reduce a specific frequency being an odd multiple of a quarter wavelengths apart. Attenuation is increased as the number of chambers increase although the addition of a third chamber only provides a small increase in attenuation. If a tube connects the chambers, the longer the tube the greater the attenuation achieved. This type of muffler is useful when space is limited and low frequency performance is required the volume and shape of the resonating chamber govern its performance capabilities. Generally as the volume of the resonating chamber increases the resonant frequency reduces.

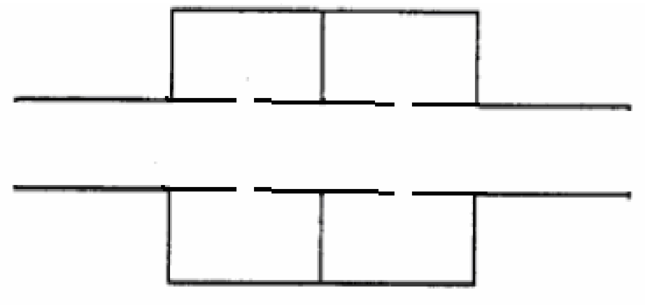

Fig. 3.11 Resonating chamber muffler

IV. MUFFLER MODELING

4.1 Muffler Modeling by Transfer Matrix Method

Mufflers are commonly used in a wide variety of applications. Industrial flow ducts as well as internal combustion engines frequently make use of silencing elements to attenuate the noise levels carried by the fluids and radiated to the outside atmosphere by the exhausts. Restrictive environmental legislation require that silencer designers use high performance and reliable techniques. Various techniques are currently available for the modeling and testing of duct mufflers. Empirical, analytical and numerical techniques have been used and proved reliable under controlled conditions. Design of a complete muffler system is, usually, a very complex task. Each element is selected by considering its particular performance, cost and its interaction effects on the overall system performance and reliability. Numerical techniques, such as the Finite Element Method (FEM) and the Boundary Element Method (BEM) have proven to be convenient for complex muffler geometries. Although these methods are applicable to any muffler configuration, when the silencer shape becomes complex, the three-dimensional FEM requires a very large number of elements and nodes. This results in lengthy and time-consuming data preparation and computation. Although high speed computational and storage machines exist, the use of FEM or BEM for muffler design is restricted to trained personnel and is commercially expensive, in particular for preliminary design evaluation. Most muffler manufacturers are small and medium companies with a limited number of resources. They thereby require fast and low cost methods for preliminary muffler design.

4.2 Theory Plane Wave Propagation

For plane wave propagation in a rigid straight pipe of length *L*, constant cross section *S*, and transporting a turbulent incompressible mean flow of velocity *V* (see Fig. 1), the sound pressure *p* and the volume velocity ν anywhere in the pipe element can be represented as the sum of left and right traveling waves. The plane wave propagation model is valid when the influence of higher order modes can be neglected. Using the impedance analogy, the sound pressure *p* and volume velocity ν at positions 1 (upstream end) and 2 (downstream end) in Fig. 1 (*x*=0 and *x*=*L*, respectively) can be related by

p1=Ap2 + Bv2-------------- (1)

v1 = Cp2 + Dv2------------ (2)

where *A*, *B*, *C*, and *D* are usually called the four-pole constants. They are frequency-dependent complex quantities embodying the acoustical properties of the pipe (Pierce, 1981).

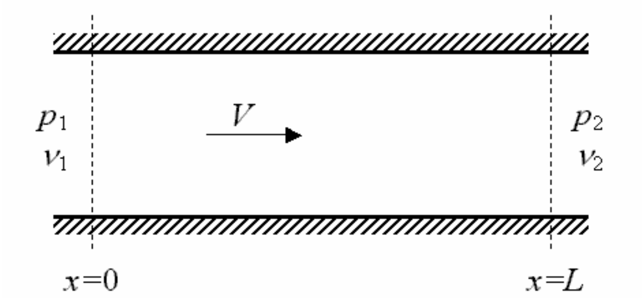

Fig.4.1. Plane wave propagation in a rigid straight pipe transporting a turbulent Incompressible mean flow.

It can be shown (Munjal, 1975) that the four-pole constants for non-viscous medium are

A=exp(-jMk$_c$L)cosK$_c$L--------- (3)
B=j(ρc/S)exp(-jMk$_c$L)sink$_c$L,--(4)
C=j(S/ρc)exp(-jMk$_c$L)sink$_c$L,- (5)
D= exp(-jMk$_c$L)cosk$_c$L -------- (6)

where *M*=*V*/*c* is the mean flow Mach number (*M*<0.2), *c* is the speed of sound (m/s), *kc*=*k*/(1−*M*2) is the convective wave number (rad/m), *k*=ω/*c* is the acoustic wave number (rad/m), ω is the angular frequency (rad/s), ρ is the fluid density (kg/m3), and *j* is the square root of −1. Substitution of *M*=0 in Eqs. (3) to (6) yields the corresponding relationships for stationary medium.

4.3 Transfer Matrix Method (TMM)

It can be observed that Eqs. (1) and (2) can be written in an equivalent matrix form as

q_1=T_1 q_2--------------------- (7)

where **q***i* = [*pi* ν*i*]T is a vector of convective state variables (*i*=1,2) and

$$T_1 = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \text{---------} (8)$$

is the 2×2 transfer matrix, defined with respect to convective state variables. This matrix relates the total sound pressure and volume velocity at two points in a muffler element, such as the straight pipe discussed in the previous section. Thus, a transfer matrix is a frequency-dependent property of the system. Reciprocity principle requires that the transfer matrix determinant be 1. In addition, for a symmetrical muffler *A* and

D must be identical. In practice, there are different elements connected together in a real muffler, as shown in Fig. 2. However, the four-pole constants values of each element are not affected by connections to elements upstream or downstream as long as the system elements can be assumed to be linear and passive. So, each element is characterized by one transfer matrix, which depends on its geometry and flow conditions. Therefore, it is necessary to model each element and then to relate all of them to obtain the overall acoustic property of the muffler.

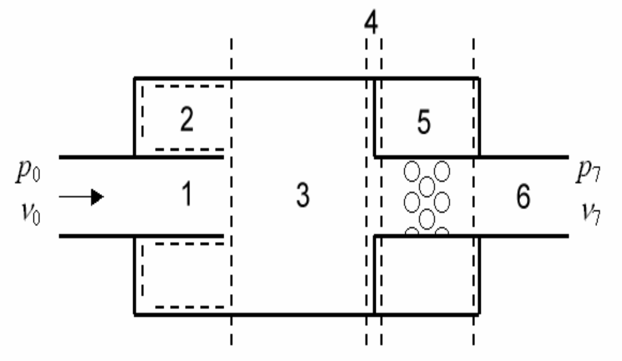

Fig.4.2 Real muffler

If several muffler elements, such as sudden expansions, sudden contractions, extended-tubes and/or perforated tubes are connected together in series, then the overall transfer matrix of the entire system is given by the product of the individual system matrices. For example, the muffler shown in Fig. 2 includes a straight extended tube, sudden expansion and extended inlet, uniform tube, sudden contraction, concentric resonator with perforated tube and a
straight tail pipe. Then, for this particular muffler we can write

$$\mathbf{q}_0 = \mathbf{T}_0\mathbf{q}_0 \text{---------------} \ \mathbf{(9)}$$

Since $\mathbf{q}1=\mathbf{T}1\mathbf{q}2$, Eq. (9) can be written as

$$\mathbf{q}_0 = \mathbf{T}_0\mathbf{T}_1\mathbf{q}_2 \text{------} \ \mathbf{(10)}$$

It is observed that $\mathbf{q}n=\mathbf{T}n\mathbf{q}n+1$, for $0 \le n \le 6$. Consequently, we obtain the final expression

$$\mathbf{q}_0 = \mathbf{T}_0\mathbf{T}_1\mathbf{T}_2\mathbf{T}_3\mathbf{T}_4\mathbf{T}_5\,\mathbf{T}_6\mathbf{q}_7 = \prod_{i=0}^{6} \mathbf{T}_i\mathbf{q}_7 \quad \text{-------}\mathbf{(11)}$$

which relates the convective state variables at two points (inlet and exhaust) in a muffler.

Now, at a point *n*, the vector of convective state variables is related to the vector of classical state variables (for stationary medium) by means of the linear transformation

$$\mathbf{q}n = \mathbf{C}n\ \mathbf{u}n \text{----------} \ (12)$$

where

$\mathbf{u}_n = [\tilde{p}_n \ \ \tilde{v}_n]^{\mathrm{T}}$, $\tilde{p}$ and $\tilde{v}$ are, respectively,

the sound pressure and volume velocity for stationary medium, and $\mathbf{C}n$ is a non singular 2×2 transformation matrix given by

$$\mathbf{C}_n = \begin{bmatrix} 1 & M_n\rho_n c_n / S_n \\ M_n S_n / \rho_n c_n & 1 \end{bmatrix}. \qquad (13)$$

Substituting Eq. (12) into Eq. (7) gives

$$\mathbf{u}_1 = \mathbf{C}_1^{-1}\mathbf{T}_1\mathbf{C}_2\mathbf{u}_2 = \tilde{\mathbf{T}}_1\mathbf{u}_2, \qquad (14)$$

where $\tilde{\mathbf{T}}_1 = \mathbf{C}_1^{-1}\mathbf{T}_1\mathbf{C}_2$ is the transfer matrix for stationary medium.

V. FUTURE SCOPE

1. Transmission Loss From TMM
2. CAD model of Silencer.
3. Verification by Ansys.

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