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FRITAMUFF: A COMPREHENSIVE PLATFORM FOR PREDICTION OF UN-MUFFLED AND MUFFLED EXHAUST NOISE OF IC ENGINES

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Abstract

Prediction of un-muffled and muffled noise from internal combustion engines requires knowledge of the acoustic attenuation performance of the muffling system along with the engine noise source characteristics. Here, in addition to the conventional cascade type analysis, a novel geometry-based user-friendly scheme has been employed for analyzing multiply-connected elements. It uses the geometry of the muffler along with a numerical algorithm to produce the overall transfer matrix of the element. This scheme has been used in conjunction with the transfer matrix based muffler program, which has a large number of predefined, parameterized elements including the variable area perforated elements, to analyze the entire exhaust system of the automobile. The source characteristics of the engine, as functions of the engine's physical and thermodynamic parameters, are predicted numerically by means of a two load method and incorporated as empirical formulas into the scheme to predict both the un-muffled and muffled noise with any muffler configuration. Thus, the insertion loss of the muffler system is evaluated for different speed orders or frequencies. Though this methodology may not be able to predict the actual noise spectra accurately, yet this is good enough from the practical point of view, as manufacturers are generally interested in the total noise level. A Graphical User Interface (GUI) has been prepared for ready use. This computational platform, FRITAmuff, is applicable for turbocharged as well as the naturally aspirated engines.

1. INTRODUCTION

Prediction of the un-muffled and muffled noise from internal combustion engines is very critical from the production point of view, especially because the pollution control regulations are getting increasingly stringent day by day. Accurate prediction of the noise from an engine in conjunction with the appropriate muffler at the design stage can result in a great amount of saving in time and money. This requires the knowledge of the engine noise source characteristics along with the acoustic attenuation quality of the muffling system. Therefore, the final prediction of un-muffled and muffled noise involves analysis of both the muffler and the engine exhaust system.

The noise-source independent acoustic performance, i.e. the Transmission Loss (TL) characteristic, can be evaluated without any knowledge of the noise source, using only the muffler geometry. However, a more informative and practically usable characterization, i.e., the Insertion Loss (IL) evaluation, calls for a prior knowledge of the Source Strength Level (SSL) and the source impedance (Z_s) in terms of the engine's physical and thermodynamic parameters. 1-D analytical methods are generally followed to evaluate these parameters [1]. Experimental determination of these parameters, as discussed in the literature, has met very limited success.

The present paper discusses a comprehensive platform, FRITAmuff, developed at the Facility for Research In Technical Acoustics (FRITA), IISc, Bangalore, for predicting the un-muffled and muffled noise from IC-engines by combining different types of muffler analysis schemes and engine noise source characterization techniques developed at FRITA over the years.

2. MUFFLER ANALYSIS

2.1. Cascading Type Analysis

In spite of the generality and popularity of the 3-D FEM and BEM methods, analytical techniques have been preferred for many reasons and efforts have been made towards developing schemes based on one- and two-dimensional approximations [2].

1-D analysis has been used extensively for analyzing mufflers based on the cascading method, which uses transfer matrices of simple acoustic elements connected in series to analyze relatively complex mufflers. This method is very fast in analyzing and hence can be used for designing of mufflers by performing a number of geometrical parametric studies in a very short time. In fact, very complex mufflers can be built with judicious combination and a little ingenious engineering assumption. One example of such a muffler which can be analyzed by this method is depicted in Figure 1(a) with its constituent elements (see Figure 1(b)).

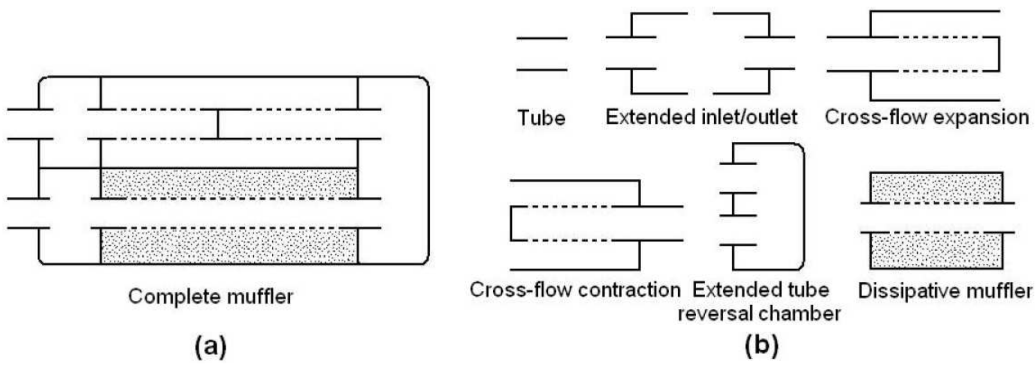


Figure 1. Schematic of a muffler analyzed by the cascading method; (a) the complete muffler (b) constituent components of the muffler.

The cascading type 1-D muffler analysis depends upon the values of the state variables: acoustic pressure p and the acoustic volume velocity v, at the upstream and downstream ends of each component of the system. Simple multiplication of the individual matrices provides the overall transfer matrix connecting the state variables at the upstream point ’u’ and the downstream point ’d’. These can be written for a muffler system by a matrix relation

$$\left\{ \begin{array}{c} p_u \\ v_u \end{array} \right\} = \left[\begin{array}{cc} T_{11} & T_{12} \\ T_{21} & T_{22} \end{array} \right] \left\{ \begin{array}{c} p_d \\ v_u \end{array} \right\} \tag{1}$$

Performance parameters like Transmission Loss (TL) and Insertion Loss (IL) of the muffler can then be evaluated from the above transfer matrix and some other engine source data by using the following relations [1]:

$$TL = 20\ log_{10}\ \left[\left(\frac{Y_1}{Y_n} \right) \left| \frac{(T_{11} + T_{12}/Y_1 + Y_n T_{21} + (Y_n/Y_1)T_{22})}{2} \right| \right], \tag{2}$$

$$IL = 20\ log_{10}\ \left[\left(\frac{R_{0,1}}{R_{0,2}} \right)^{1/2} \left| \frac{Z_s}{Z_s + Z_{0,1}} \right| \left| \frac{v_s}{v_0} \right| \right], \tag{3}$$

where R_0 is the radiation resistance, Z_0 is the radiation impedance, subscripts 1 and 2 denote ‘without’ and ‘with muffler’, respectively, v_s is the source volume velocity, p_s/Z_s, and v_0 is the radiation-end volume velocity downstream of the muffler.

2.2. Multiply-Connected Mufflers

The aforementioned technique of analysis lacks the flexibility of analyzing very complex mufflers, with multiply-connected internal elements which may produce better attenuation with lesser back pressure. The generalized scheme (Volume Synthesis algorithm) developed by the authors [2] uses a simple and novel method of dissecting most of the commercially usable complex mufflers into five simple elemental blocks. Subsequently, on the lines of the FEM and BEM based analysis, it uses the geometric data to extract relevant information about the interconnectivity among various regions of the muffler to construct the system matrix taking into account all the boundary conditions. Then the system matrix is reduced to a row echelon form using Gauss-Jordan elimination with partial pivoting, from which the final transfer matrix for the muffler is evaluated. Emphasis has been laid on minimizing the user’s effort to input the 2-D geometric data for the muffler under investigation and the procedure removes the requirement to deal with the mutual dependencies among the elemental blocks. Thus, the volume synthesis algorithm proposed and used, is more comprehensive in application, requires less pre-computation effort, and is faster in execution. An example of this type of muffler has been shown in Figure 2 and discussed in detail in a subsequent section.

2.3. 1-D Analysis of Generalized Conical and Variable Area Mufflers

Matrizant approach is a useful technique to model complex muffler geometries having multiple cavity resonators, with varying cross sectional areas and perforated tubes using one dimensional wave propagation analysis [3]. Figure 3 shows one such type of muffler in the form of a folded conical concentric tube resonator. Many suchlike mufflers can be designed with multiple interacting perforated ducts to have varied desired performance. This kind of mufflers, currently, cannot be analyzed using the volume synthesis algorithm discussed above.

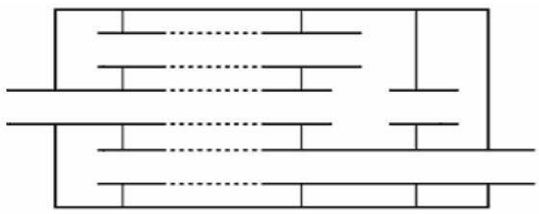


Figure 2. An example of multiply connected muffler

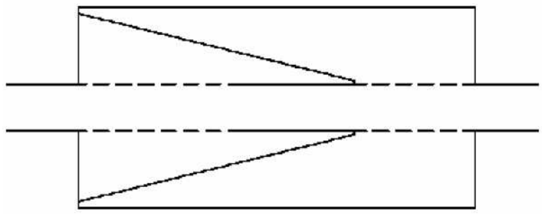


Figure 3. Folded conical concentric tube resonator muffler.

3. ENGINE SOURCE CHARACTERIZATION

In the case of internal combustion engines, one needs to know insertion loss of a muffler as well as the unmuffled exhaust noise. Both these parameters call for prior knowledge of the source characteristics, p_s and Z_s [1], which can be evaluated indirectly by means of the two-load method [1, 4-8]. Unfortunately, however, unique source characteristics don't exist for a highly non-linear and time-variant geometry source like the exhaust system of an I.C. Engine [5]. One of the alternatives is the hybrid approach, combining the time-domain analysis of the nonlinear exhaust source making use of the method of characteristics with the frequency-domain analysis of the linear muffler, by means of the frequency transform pair or the collocation technique [9-13]. Hybrid approaches are cumbersome and prone to instability; convergence of the iteration process is not guaranteed. Turbocharger and emission control devices introduce additional problems. In view of all these difficulties, one may resort to use of the two-load method, improving the accuracy by averaging over several two-load combinations. The exhaust system can be modeled numerically by means of a standard commercial software like AVL-BOOST [14]. Acoustic loads can be in the form of exhaust pipes of different lengths [8, 15].

At a point in the exhaust pipe, just downstream of the exhaust manifold or turbocharger or catalytic converter, as the case may be, one computes the acoustic pressure as a function of time over a steady-state thermodynamic cycle for a pair of acoustic loads, evaluates the discrete Fourier transform of the two arrays to find acoustic pressure in the frequency domain for different speed orders: 0.5, 1.0, 1.5, 2.0....... upto 50 (say). Referring to Figure 4(a), simultaneous solution of the two equations for two different acoustic loads impedances Z_{L1} and Z_{L2} yields [1, 4, 8]

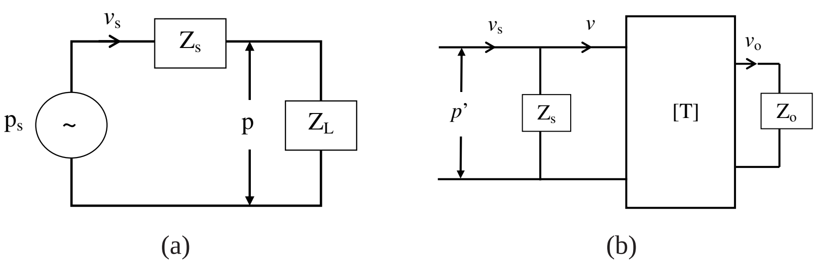


Figure 4. Electrical analogous circuit for (a) an un-muffled and (b) a muffled system.

$$p_s = p_1 p_2 \frac{Z_{L1} - Z_{L2}}{p_2 Z_{L1} - p_1 Z_{L2}} \quad \text{and} \quad Z_s = Z_{L1} Z_{L2} \frac{p_1 - p_2}{p_2 Z_{L1} - p_1 Z_{L2}}. \tag{4}$$

We may define the Source Strength Level (SSL) as

$$SSL \equiv 20\log_{10}|p_s/p_{th}|\,, dB, \quad p_{th} = 2\times 10^{-5}Pa, \tag{5}$$

and source impedance $Z_s(= R_s + jX_s)$ may be normalized with respect to the characteristic impedance of the exhaust pipe, $Y_0 = \rho_0 c_0/S$. Computed values of SSL and Z_s/Y_0 are averaged over nine different pairs of acoustic loads, and plotted against speed orders. Writing the Source Strength Level for a 4-stroke engine with n_{cyl} cylinders, as

$$SSL = A\left(\frac{speed\ order}{n_{cyl}/2}\right)^B, dB, \tag{6}$$

expressions for A and B have been obtained in the form of the least squares fits. It may be noted that 'A' represents the source strength level at the firing frequency of the engine, where it peaks in nearly all cases.

Parametric studies have been performed for the following parameters, varying one at a time keeping other parameters constant at their default (underlined) values:

Turbocharged diesel engines without catalytic converter:

Air fuel ratio, AFR= 18.0, <u>23.7</u>, 29.2. 38.5, 46.0, 58.8
Engine speed in RPM= 1000, 1300, 1600, 2100, 2400, 3000, 3500, <u>4000</u>, 4500
Engine capacity (displacement), V (in liters)= 1.0, 1.5, 2.0, <u>2.5</u>, 3.0, 4.0
Number of cylinders, n_{cyl}=1, 2, 3, <u>4</u>, 6

Naturally aspirated diesel engines without catalytic converter:

Air fuel ratio, AFR= 14.5, <u>17.0</u>, 29.0, 39.6, 47.5, 59.6
Values of RPM, V and n_{cyl} are the same as for the turbocharged engines above.

Figure 5(a) shows typical values of SSL of a 4-cylinder turbocharged compression-ignition engine as a function of speed order for the default values of the air-fuel ratio, engine speed, engine capacity, and the number of cylinders.

Least squares fits have resulted in the following generalized empirical expressions [16]:

Turbocharged engines:

$$A = 115\,(1-0.00182AFR)\left(1+0.459N-0.067N^2\right)(1-0.00405V)\,(1-0.0268n_{cyl}) \tag{7}$$

$$B = -0.189\,(1+0.0142AFR)\,(1-0.1134N)\,(1-0.00832V)\,(1+0.0178n_{cyl}) \tag{8}$$

Naturally aspirated engines:

$$A = 175.4\,(1-0.00197AFR)\left(1+0.111N-0.02N^2\right)(1+0.0022V)\,(1-0.0196n_{cyl}) \tag{9}$$

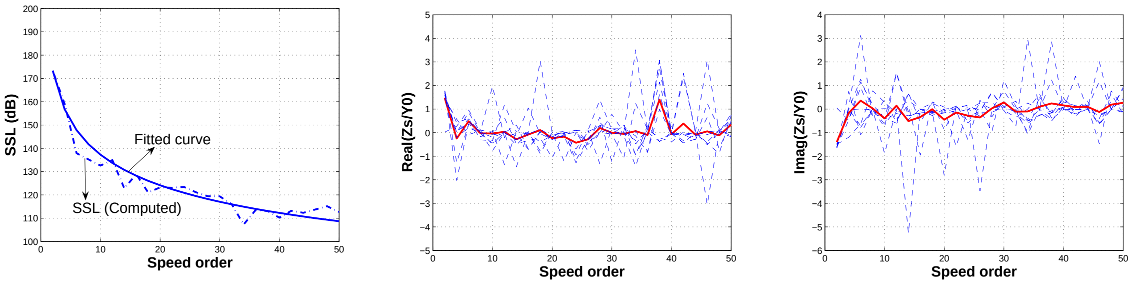
$$B = -0.124\,(1-0.00123AFR)\,(1+0.02245N)\,(1-0.0144V)\,(1+0.124n_{cyl}) \tag{10}$$

However, values of R_s/Y_0 and X_s/Y_0, the real and imaginary components of the normalized source impedance (Z_s/Y_0) show considerable variation for different pairs of acoustic loads (see Figures 5(b) and (c)). In fact, this variation is such that the average values fall around a horizontal line parallel to and nearly touching the speed-order abscissa, thereby suggesting a constant pressure source; i.e., R_s,X_s $<<$ Y_0, or tending to zero. But this situation is not realistic. It is due to the fact that for a nonlinear time-variant-geometry source, unique source characteristics don't exist [5]. Therefore, different pairs of loads yield substantially different values of source impedance, even negative values of R_s/Y_0 [17].

Incidentally, Prasad and Crocker [18] observed that at speed orders of $\gtrsim$10, the exhaust system of a multi-cylinder engine is nearly anechoic: $Z_s \approx Y_0$. Later, Callow and Peat suggested a relatively more realistic formula [19]:

$$Z_s/Y_0 = 0.707 - j0.707 \tag{11}$$

As the un-muffled exhaust sound pressure level and insertion loss of a muffler are known to be weak functions of source impedance, we may as well adopt equation (11) for source impedance.



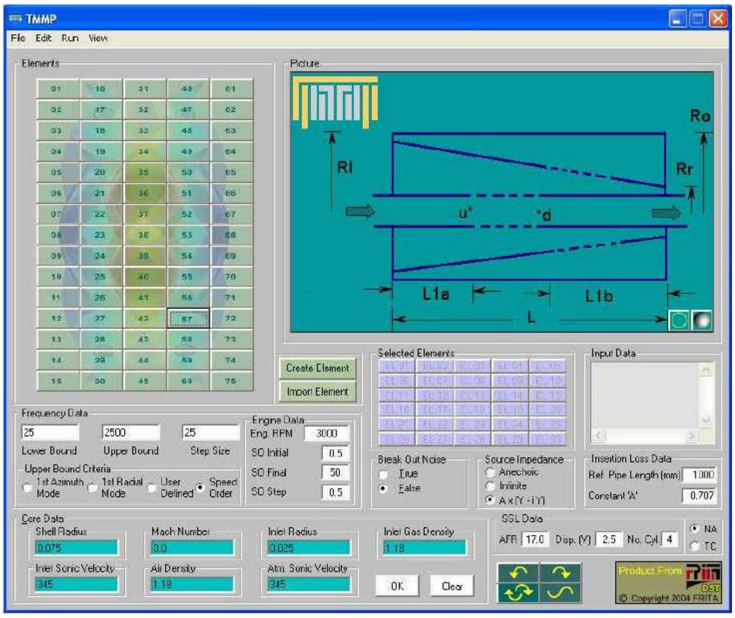
(a) (b) (c)

Figure 5. (a) Source strength level (SSL), (b) real part and (c) imaginary part the of normalized source impedance of a 4-cylinder turbocharged CI-engine. In Figures (b) and (c) dotted lines show results for different pairs of loads and the solid line shows the average over them.

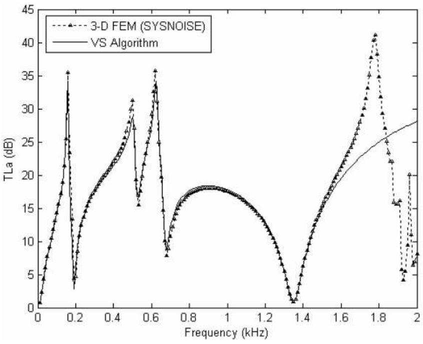
4. COMPREHENSIVE 1-D PLATFORM: FRITAMUFF

All the above mentioned concepts have been put together with some more relevant and necessary features to build a user-friendly platform for analyzing the exhaust system of an IC-engine. Figure 6(a) shows the Graphical User Interface (FRITAmuff) developed for the implementation of the same. The platform provides the user many ways to specify the frequencies of analysis under different situations. The user has the liberty to choose the engine data, such as the speed (RPM), air-fuel ratio (AFR), engine swept volume (V), number of cylinders (n_{cyl}), which are then used to evaluate the SSL by using the developed empirical expressions for the specified engine. The source impedance also can be selected to suit the particular situation. These are then used to predict the un-muffled noise of the engine.

The muffled noise of the engine is then evaluated using the muffler configuration (see Figure 4(b)) prepared using the GUI. In this regard, either pre-parametrized elements can be used to create the muffler or new elements can be created using the volume synthesis algorithm based element creator. Four-pole parameter data from 3-D analysis or experiments conducted on any muffler can also be imported and used in conjunction with the available elements. Figure 6(b) shows the comparison of TL curves between the results obtained from 3-D analysis and those predicted using the present platform for the muffler shown in Figure 2. Correlation between the present analysis and the 3-D analysis results can be seen to be quite well for even such a complex multiply-connected muffler upto the cut-off frequency.



(a) Graphical user interface



(b) Transmission loss performance of the muffler of Figure 2

Figure 6. (a) Graphical User Interface (GUI) for analyzing the complete exhaust system of internal combustion engines and (b) comparison of transmission loss performance evaluated using the FRITAmuff platform and SYSNOISE.

5. CONCLUSIONS

With the capability of combining pre-parameterized muffler elements with the new-element creator module, FRITAmuff provides a wide range of muffler options to be analyzed with different types of IC-engine configurations for the prediction of un-muffled and muffled noise. Using this computational platform, one can analyze a majority of automotive commercial mufflers in conjunction with the engine which is difficult with AVL-BOOST alone. Same methodology can be adopted to characterize the source in the intake system of the engine as well. Though this approach may not be able to predict actual noise spectra precisely, yet this is good enough from the practical point of view, as manufacturers are generally interested in the total noise level.

6. ACKNOWLEDGEMENTS

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