ACOUSTIC CHARACTERIZATION OF AN ENGINE EXHAUST SOURCE – A REVIEW

M. L. Munjal

Facility for Research in Technical Acoustics, Department of Mechanical Engineering, Indian Institute of Science, Bangalore 560 012, India

Abstract

For an engine running at a constant speed (RPM), the exhaust process and intake process are periodic. This enables use of the frequency-domain analysis of the essentially linear exhaust system consisting of the exhaust runners, manifold, exhaust pipe, the muffler proper and the tail pipe. Over the last forty years, transfer matrices have been derived for use with aeroacoustic state variables as well as the classical state variables of acoustic pressure and acoustic volume/mass velocity. This frequency-domain analysis, however, requires prior knowledge of the load-independent source characteristics p_s and Z_s, corresponding to the open-circuit voltage and internal impedance in an analogous electrical system (as per Thevenin theorem). Several methods have been suggested for prediction or measurement of the source characteristics over the years, but with little success.

Alternatively, time-domain analysis of the exhaust system, making use of the method of characteristics, does not require prior knowledge of the source characteristics. On the other hand, it can, and indeed has been harnessed to evaluate the source characteristics for use with linear frequency-domain analysis. Unfortunately, the source characteristics so obtained are not load-independent because of the inherent non-linearity and time dependence of the piston motion, exhaust valve/port opening and high blow-down pressure in the cylinder. Besides, the time-domain analysis of complex muffler elements is very cumbersome and error-prone. Therefore, hybrid approach has been mooted where the time-domain analysis of the exhaust source is combined with the frequency-domain analysis of the exhaust muffler making use of the discrete Fourier transform pair. Here again, there are several difficulties and challenges.

This paper reviews all these developments and presents the state of the art for estimating unmuffled exhaust noise and insertion loss of commercial mufflers.

Nomenclature

A	Forward wave acoustic variable
B	Reflected wave acoustic variable
c	Sound speed
k	Wave number
p	Acoustic pressure
P	Forward wave Riemann variable
p_o	Ambient pressure
p_s	Source pressure/strength
Q	Reflected wave Riemann variable
v	Mass/volume velocity
Y	Characteristic impedance
Z_L	Load impedance
Z_s	Source impedance
γ	Ratio of the specific heats
ω	Radian frequency

Introduction

Exhaust noise is known to be a predominant component of the automobile noise. Fortunately, over the last few decades, it has been possible to reduce it the level of the other components (the engine body noise, cooling system noise, etc.) by means of a muffler or silencer. For an engine running steadily at a particular speed (RPM), the exhaust pulse is periodic, and therefore the frequency-domain analysis lends itself as a convenient tool. This analysis developed on the lines of the acoustic filter theory through electro-acoustic analogies. The state variables of acoustic pressure p and mass (or volume) velocity v are analogous to the electro-motive force (or voltage) and current, for plane (one dimensional) waves in stationary medium [1]. The analogy can be extended to plane waves in an incompressible moving medium by adopting the convective state variables p_c and v_c, defined as acoustic perturbation on stagnation (or total) pressure and mass flux [2].

The transfer (or transmission) matrix method is ideally suited for analysis of one-dimensional acoustic filters or mufflers because of the feature of cascading [1]. Making use of the equations of mass continuity, momentum balance, and isentropicity, transfer matrices have been derived over the last three decades by the author and his students, among others, for about sixty different elements [1-17]. Most of them are shown in Ref. 18 and their four-pole parameters have been reproduced explicitly in Refs. [1,15]. The product transfer matrix of the entire muffler can be obtained by successive multiplication of the transfer matrices of the constituent elements. Transmission loss of the muffler can be obtained readily from the four-pole parameters of the product transfer matrix. However, insertion loss and noise reduction (or level difference) of the muffler would depend on the acoustic characteristics of the engine exhaust source (p_s and Z_s) and the radiation (or load) impedance at the tail pipe end (see Fig. 1).

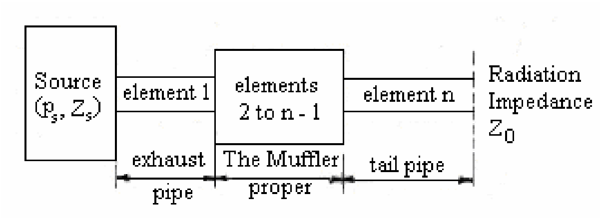

Fig. 1. Schematic of a typical exhaust system (adapted from Ref. [18])

Analogous to Thevenin theorem in electrical network theory, p_s and Z_s correspond to the open-circuit voltage and internal impedance of the source. However, unlike in an electrical source, the zero velocity (corresponding to the requirement of zero current in an open circuit) cannot be obtained in the exhaust system. Nor can the internal impedance be measured directly because of large piston motion and the time-variant valve or port openings. These could be evaluated only indirectly making use of the system response to two, three or four different loads. But then the question arises: do unique (load-independent) source characteristics exist for a non-linear time-variant-geometry source? If not, is there a way we can combine the time-domain simulation if the engine exhaust source with the frequency-domain modeling of the muffler so as to bypass the evaluation of source characteristics? How good are such hybrid approaches? These questions are examined at length in this review paper from the point of view of the muffler designer.

Direct Measurement of Source Impedance

Internal impedance of the engine exhaust source may be measured directly by means of the impedance tube technology with an external source that would produce so strong a sound filed that by comparison the noise produced by the engine at the frequencies of interest would be negligible. An electro-pneumatic sound source would foot the bill. But then, as shown by Prasad and Crocker, even such a source would not be string enough at frequencies of the order of the firing frequency and a couple of harmonics, (up to about 300 Hz). It has been suggested that this problem could be solved by motoring the engine. After all, the source impedance would not be a function of firing; it has to be primarily a function of the geometry of the source. But then, time-variant geometry would preclude the existence of a standing wave pattern, which is a pre-requisite of the impedance tube technology. Could one consider the possibility of a "time-average" source impedance, as a necessary compromise?

As the measured impedance will refer to a particular point in the impedance tube, it can be transferred to another point (say, at the exhaust valve/port exit) by means of simple formulae [19]. Similarly, if one is working with convective state variables [1, 2], the source impedance measured with respect to the classical (stationary medium) state variables can be transformed to the convective sources impedance by means of simple formulae [20].

Approximations for Source Impedance

To a designer, measurement of sources impedance is too much of a hassle. He would like simple expressions in terms of frequency and geometry and perhaps the operating parameters like speed (RPM) of the engine. This, however, has remained a dream! Instead, some researchers have arrived at or offered empirical expressions independent of the geometric parameters. For example, Prasad and Crocker, based on their direct measurements of source impedance of a multi-cylinder inline engines (see Fig. 2), suggested [21]

$$Z_s \approx Y_e \tag{1}$$

where Ye is the characteristic impedance of the exhaust pipe. This implies the suggestion of an anechoic source. Callow and Peat offered a little more realistic expression [22]:

$$Z_s = Y_e e^{j\frac{7\pi}{4}} = Y_e\left(\frac{1-j1}{\sqrt{2}}\right) \tag{2}$$

Note that $|Z_s| = Y_e$ in Eq. (2).

There has also been suggestion of a constant velocity source [23, 24] implying

$$Z_s >> Y_e \text{ or } Z_s \to \infty \tag{3}$$

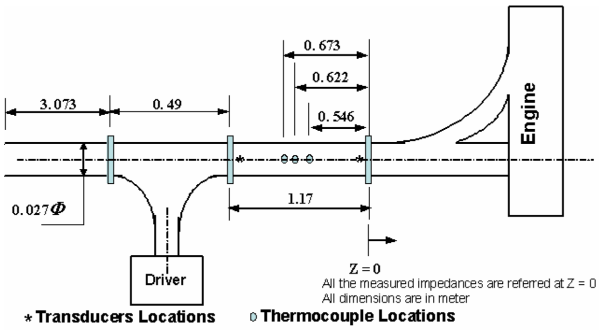

Fig. 2. Diagram of the setup for engine impedance measurement (adopted from Ref. [21])

Interestingly, these three expressions do not give very different values of the insertion loss (IL) for most mufflers at different frequencies, for IL does not have a strong dependence on the source impedance, Z_s. This is indeed fortuitous for the designer.

Incidentally, expression (2) is made up of resistance (damping) and compliance, and therefore is physically more representative of the cylinder-valve combination.

Abom et al. have studied the possibility of prediction of the source impedance of a 4-cylinder engine manifold by using guesswork about the terminal impedances at the valves [25]. Recently, Dokumaci has proposed a new mathematical model based on basic fluid dynamic equations for inviscid one-dimensional flow, encompassing both linear time-invariant, linear time-variant and non-linear one-port source models [26]. This model predicts the source impedance Z_s as well as source strength p_s at a reference plane downstream of the exhaust manifold. Significantly, however, these source characteristics are not unique; these depend on the load (muffler), as has indeed been predicted indirectly by Gupta and Munjal by means of the method of characteristics [27].

Indirect Measurement of Source Characteristics

The difficulties encountered in the direct measurement method and limitations of the proposed approximations tempt one to resort to the indirect methods making use of two, three, four or more different loads, depending on the resources (equipments) on hand. It can be seen from Fig. 1 that for load impedance $Z_{L,i}$ attached to the source to be characterized,

$$p_s = \left(Z_s + Z_{L,i}\right) v_i\,,\ \mathrm{i} = 1, 2 \tag{4}$$

Thus, if complex values of volume velocity v_i or acoustic pressure $p_{Li} = Z_{L,i}\, v_{\mathrm{i}}$ could be measured (retaining the phase information), then two different loads would suffice for indirect evaluation of the source characteristics p_s and Z_s. Eq. (4) can readily be adopted to the evaluation of convective source characteristics $p_{c,s}$ and $Z_{c,s}$ [1, 2, 27]. However, if one has only a sound level meter at his disposal, he can make free-field measurements of the radiated exhaust noise for four different acoustic loads making use of the scheme proposed by Prasad [28] (see Fig. 3), or Munjal [1].

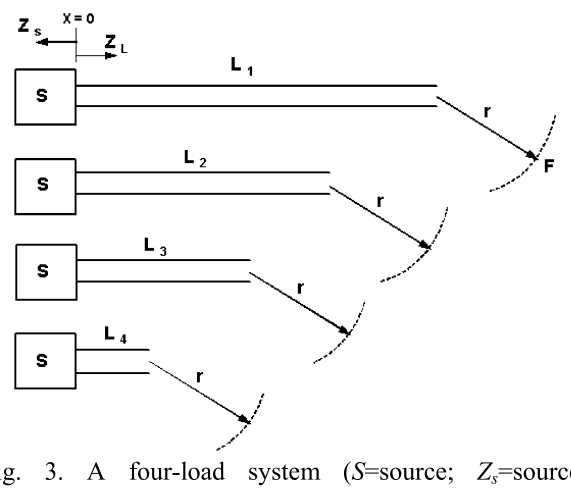

Fig. 3. A four-load system (S=source; Z_s=source impedance; Z_L=load impedance; F=field point) (adapted from Ref. [28])

Doige and Alves, tested the direct as well as indirect methods on a loudspeaker (like Prasad), a low-speed single cylinder engine, and a small motor-driven centrifugal fan, making use of dual-channel analyzer and digital data processing equipment [29]. They observed that for a running engine, the two-load method gave better results when one of the acoustic loads was semi-anechoic.

The indirect method has the advantage that it does not need an extra-powerful secondary source, and that it yields both the source characteristics (p_s and Z_s), not just the source impedance (Z_s). In addition, the four-load method does not need a reference signal, such as is always required when using the two-load method [30].

Simultaneous solution of a set of equations would lead to substantially erroneous results if the coefficients are in error due to experimental inaccuracies. This problem of instability has been studied by several researchers [31-33] and in fact some refinements have also been proposed [34, 35]. Jang and Ih have proposed an error function method based on the linear, time-invariant source model [35] and convective acoustic parameters [20].

However, as indicated earlier, the engine exhaust system is nonlinear and time variant with no unique source characteristics [27]. One of the manifestations of these features is that the use of indirect methods yields negative source resistance (at many frequencies), whereas the measured source resistance by the direct method is positive [21, 34-37]. Of course, there could be measurement errors too inasmuch as in some instances, one measurement method gives positive values of source resistance where another gives negative values. Ih and Peat have investigated the possible causes of source impedance in the measurement of intake and exhaust noise sources, and concluded that the violation of the assumption of a time-variant linear source and the defect in the inherent algorithm of the multi-load method are most probable origins of the negative source resistance [38]. Like Gupta and Munjal [27], they have observed that the resulting source characteristics can nevertheless be used to give good predictions of radiated noise levels from different loads applied to the engine. These source characteristics have however no physical significance for they are not unique for a particular engine; they will vary with varying loads [27, 38].

Time Domain Simulation of an Engine

The method of characteristics, one of the methods for solving the unsteady one-dimensional gas-dynamic equations in the time domain [39-41, 1, 27], is a powerful alternative to the frequency domain method. It has several advantages over the latter; viz.,

a) It can simultaneously predict the exhaust noise, intake noise, and the pressure-time history of the cylinder (see Fig. 4)

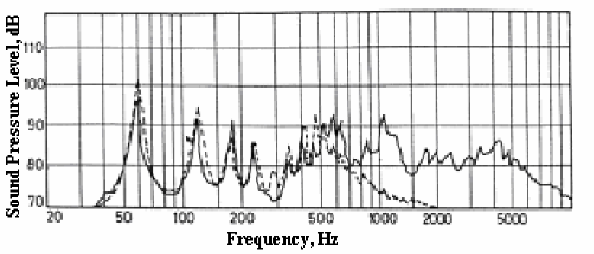

Fig. 4. Comparison of the computed (dashed line) and measured (solid line) exhaust noise in a semi-anechoic room (adapted from Ref. [41])

b) It does not need prior knowledge of the source characteristics, and
c) It can tackle nonlinear and time-variant sources as well as the linear and time-invariant sources

On the other hand, it suffers from certain disadvantages:
i) As of now, it cannot be applied to simulation of nonlinear waves (evaluation of Riemann variables) in complex geometries of commercial automotive mufflers.
ii) It is very cumbersome and time consuming.

In order to incorporate the advantages of the frequency-domain approach (the transfer matrix method) and the time-domain approach (the method of characteristics), efforts have been made by several researchers to develop hybrid approaches.

Hybrid Approaches

The basic idea of hybridization is to couple the time-domain model of the engine exhaust manifold with the frequency-domain model of the exhaust muffler. The idea of hybridization seems to have come from Soedel et al. [42, 43] and then from Jones et al. who wondered if complete nonlinear modeling was necessary [47]. It was implemented by Gupta and Munjal [27] by means of an impulse response convolution technique in conjunction with iteration. In some later works, convolution using the reflection function [44, 45] or scattering matrix [46] has been used. There were some difficulties here. Davies and Harrison assumed that acoustic source is a constant velocity source implying $Z_s \to \infty$ or $Z_s >> Z_L$ [44]. Moreover, the time domain velocity was calculated by scaling the valve lift curve. Payri et al.'s hybrid approach is rather cumbersome because of the repeated use of the Fourier and Inverse Fourier transformations. Besides, their scheme of calculation depends on the initial solution for the time-domain input pressure variables. Therefore, Satyanarayana and Munjal [48] proposed a simple hybrid approach making use of the following interrelationships between wave variables *A(t)* and *B(t)* of linear acoustic theory and non-dimensionalized Riemann variables *P(t)* and *Q(t)* of the method of characteristics [39, 1, 27]:

$$A(t) = \frac{\gamma\, p_o}{\gamma - 1}\{P(t) - 1\} \tag{5}$$

and

$$B(t) = \frac{\gamma\, p_o}{\gamma - 1}\{Q(t) - 1\} \tag{6}$$

or conversely,

$$P(t) = 1 + \frac{\gamma - 1}{\gamma}\frac{A(t)}{p_o} \tag{7}$$

and

$$Q(t) = 1 + \frac{\gamma - 1}{\gamma}\frac{B(t)}{p_o} \tag{8}$$

where γ is the ratio of the specific heats and p_o is the mean pressure in the flow medium. The frequency-domain variables *A(ω)* and *B(ω)* are simple discrete Fourier transforms (DFT) of *A(t)* and *B(t)* in Eqs. (5) to (8). For simplicity, nonlinear propagation in the exhaust pipe was neglected and free expansion was assumed at the end of the exhaust pipe. This latter assumption holds reasonably for free radiation condition and sudden expansion into an expansion chamber, but would not hold for perforated pipe elements like concentric tube resonators, plug mufflers, etc. Recently this assumption has been done away with Hota and Munjal [49] while retaining the associated simplicity of Ref. [48]. They have duly incorporated the reflection of the forward wave at the exhaust valve at each harmonic or frequency (integral multiple of the firing frequency). This reflection coefficient is calculated from the input impedance of the muffler, which in turn is calculated by means of the transfer matrix method [1, 15]. Fig. 5 shows typical results of the computation schemes.

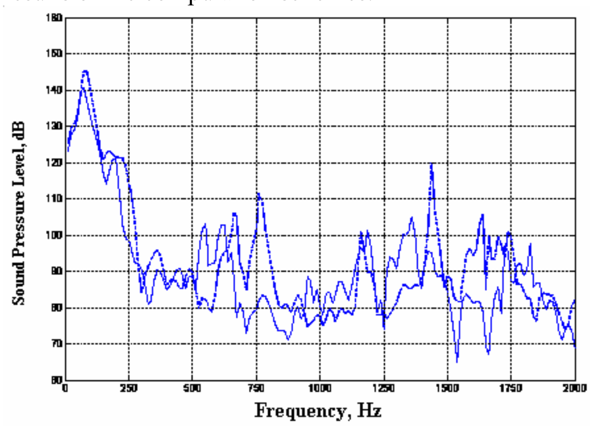

Fig. 5. Sound pressure level at a distance of 1 m from the radiation and of a simple expansion chamber. --- Ref. [48], — present approach (adapted from Ref. [49])

Concluding Remarks

In this review paper, various approaches for measurement or evaluation of the source characteristics of an engine exhaust system have been briefly reviewed, with particular emphasis on their relative implications and limitations. Finally, the hybrid approaches that obviate the need for prior knowledge of the source characteristics, have been reviewed. These approaches combine the advantages of the frequency-domain analysis of mufflers with those of the time-domain analysis of the exhaust manifold source.

Acknowledgements

Funding of the Facility for Research in Technical Acoustics (FRITA) at the Indian Institute of Science by the Department of Science and Technology of the Government of India is gratefully acknowledged. The author wishes to thank Mr. A. Sreenivasa Rao for preparation of this manuscript on computer.

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