

IC-Engine Exhaust and Intake System Acoustic Source Characterization

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ABSTRACT

The paper gives an overview of techniques used for characterization of IC-engines as acoustic sources of exhaust and intake system noise. Some recent advances regarding nonlinear source models are introduced and discussed. To calculate insertion loss of mufflers or the level of radiated sound information about the engine as an acoustic source is needed. The source model used in the low frequency plane wave range is often the linear time invariant one-port model. The acoustic source data is obtained from experimental tests or from 1-D CFD codes describing the engine gas exchange process. The IC-engine is a high level acoustic source and in most cases not completely linear. It is therefore of interest to have models taking weak non-linearity into account while still maintaining a simple method for interfacing the source model with a linear frequency domain model for the attached exhaust or intake system. The use of source characterization in acoustic design of mufflers is also briefly discussed.

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INTRODUCTION

A muffler or silencer is a device used in a flow duct to prevent sound from reaching the openings of the duct and radiating as far-field sound. Reactive silencers do this by reflecting sound back towards the source while absorptive silencers attenuate sound using absorbing material. They are necessary components in the design of any exhaust or intake system for internal combustion (IC) engines. No car or truck can pass the standard noise tests required by legislation or compete on the market without them. There are three basic requirements for a modern exhaust systems; compact outer geometry, sufficient attenuation and low pressure drop.

The aim of this paper is to discuss source characterization used in acoustical design and analysis of IC-engine exhaust and intake systems. Linear frequency domain prediction codes are frequently used for calculation of low frequency sound transmission in and sound radiation from IC-engine exhaust and intake systems. To calculate insertion loss of mufflers or the level of radiated sound, information about the engine as an acoustic source is needed. The source model often used in the low frequency plane wave range is the linear time invariant one-port model. An overview of previous work in the field is given in [1, 2]. The acoustic source data is obtained from experimental tests or from 1-D CFD codes describing the engine gas exchange process, see e.g. [3, 4, 5].

It is of interest to have models taking weak non-linearity into account while still maintaining a simple method for interfacing the source model with a linear frequency domain model for the

attached exhaust or intake system. Experimental linearity tests have been developed [6]. Some years ago a model which can consider weakly non-linear sources was presented [7] which gave an improvement over the traditional two-load technique for determining source data from experiments. It is however fairly complicated to implement and has not been used a lot. In this paper an alternative technique based on so called polyharmonic distortion modeling, used for nonlinear characterization of microwave systems [8-9] is proposed and tested. Comparisons are made with the results from linear source models and the previously published weakly nonlinear source model.

The linear acoustic one-port source data will depend on the engine operating conditions which means that they should be determined for a range of engine speeds and loads of interest. An interesting attempt to overcome this difficult by identifying a parametric model for how the source data varies with engine parameters has been suggested in [10].

A review paper on muffler design can be found in [11]. In order to assess the success of a new muffler design there is a need for measures to quantify the sound reduction obtained. There are at least three such measures in common use: transmission loss, insertion loss and noise reduction. The transmission loss is defined as the ratio between the sound power incident to the muffler (W_i) and the transmitted sound power (W_t) for the case that there is a reflection free termination on the downstream side and only depends on the properties of the muffler itself. It does not depend on the acoustic properties of the upstream source or the downstream load. Insertion loss is defined as the

difference in sound pressure level at some measurement point in the pipe or outside the opening when comparing the muffler element under test to a reference system. It is common that the reference system is a straight pipe with the same length as the muffler element under test but it could also be a base-line muffler design against which new designs are tested. Insertion loss is obviously easy to measure as it only requires a sound pressure level measurement at the chosen position for the two muffler systems. It does however depend on both upstream acoustic source characteristics and downstream acoustic load characteristics. Sound reduction is defined as the difference in sound pressure level between one point upstream of the muffler and one point downstream. Just as insertion loss sound reduction is easy to measure but difficult to calculate since it depends on source and load properties.

SOURCE MODELS

This section describes the source models used. Figure 1 shows the test geometry discussed. We will assume that the travelling pressure wave amplitudes in the positive coordinate direction p_+ and in the negative coordinate direction p_- can be obtained either from multi-transducer measurements or from numerical simulation as described in [1, 2, 3, 4, 5]. We will use so called multi-load methods [1, 2] to determine the source data. These require that measurements or simulations have been made for a number of different loads characterized by their reflection coefficients (R_L) or normalized impedance (Z_L).

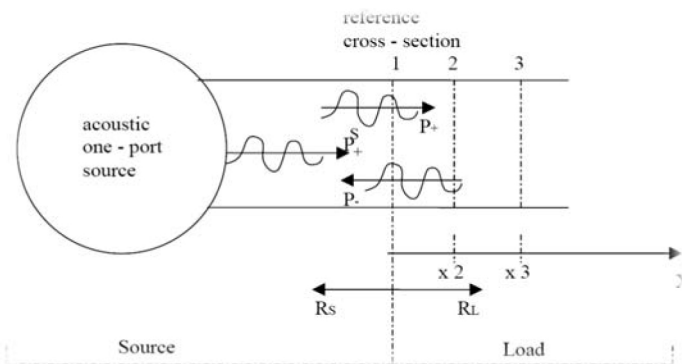


Figure 1. An in-duct source modeled as an acoustic one-port

Linear One-Port Model

In the linear one-port model the source is characterized by two parameters: the source strength and the source reflection coefficient or source impedance. In the literature the source model for one-ports is often expressed in terms of source strength (p_s) and normalized source impedance (Z_s),

$$Z \cdot p = Z_L \cdot p_s - Z_s \cdot p, \quad (1)$$

where p_s is the source pressure, p is acoustic pressure and Z_s is the normalized source impedance.

Nonlinear Frequency Response Function Model

The time domain representation of the source model with nonlinear term was in [7] described by

$$\int z_s(\tau)q(t-\tau)d\tau + \int h_s(\tau)b(t-\tau)d\tau = p_s(t) - p(t) \quad (2)$$

where ($p(t)$) and ($q(t)$) denote the pressure and volume velocity at the source cross section, ($z_s(t)$) is the time domain representation of the source impedance, ($p_s(t)$) is the source strength, ($b(t)$) is the non-linear input and ($h_s(t)$) is the source data coefficient for the non-linear part. When applying this technique in [7] it was assumed that $b(t) = q^3(t)$ which is the first higher order series expansion term obtained for the pressure drop over an orifice. This assumption will also be used here. In the frequency domain Eq. (2) can be formulated as

$$Z \cdot p = Z_L \cdot p_s - Z_s \cdot p - H_s \cdot B = Z_L \cdot p \quad (3)$$

where (H_s) and (B) are the Fourier transforms of ($h_s(t)$) and ($b(t)$). This equation has compared to Eq. (1) a third complex unknown (H_s), which means that now at least three acoustic loads will have to be used in order to solve the equation and to obtain the source data.

Nonlinear Scattering Matrix Model

In this section expressions for the acoustic source data starting from the polyharmonic distortion (PHD) theory [8-9] will be introduced for a source according to Fig. 1. The PHD approach describes nonlinear scattering and transmission properties when a system is excited with discrete tone or harmonic excitation. It has here been adapted to the case that the system also has an active source part. The PHD model is an approximation, which involves linearization using three main assumptions. The first assumption is that the system studied is time-invariant. The second assumption is that the functions describing the system are non-analytic because of the nonlinear system properties. In general, when the sample under test becomes nonlinear, the superposition principle is no longer valid. The third assumption is that there is only one dominant large-signal input component present ($p(f)$) whereas all other harmonic input components are relatively small. In this case, we will be able to use the superposition principle for the relatively small input components. This harmonic superposition assumption is the key to the development of the PHD model. Using these assumptions leads to a source model, where it has in addition been assumed that the nonlinear energy transfer only goes from lower to higher frequencies and not the other way around.