Finite Elements Simulations of Noise Damping in a Muffler

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Abstract: - This study represents an investigation of the COMSOL Finite Elements (FE) capabilities of simulating noise reduction techniques. This general purpose FE code has an acoustics module, from which the present study investigates one type of muffler used in the automotive industry. A comparison is made between a reference design model and an improved model with a number of holes in the two ducts of the muffler. The results indicate that such a design tool can be successfully used in other applications concerning noise reduction such as exhaust nozzles.

Keywords: muffler design, noise damping

1. INTRODUCTION

The theoretical studies, numerical simulations and experiments concerning noise produced by combustion engines represent an important preoccupation for acousticians over the last four decades. The results are visible (or hearable) in modern cars for which the exhaust noise has been reduced below the level of the wheels noise. The numerical and experimental studies are continued for improved acoustical performance and economic efficiency.

In ref [1] is investigated the exhaust manifold system with Perforated Manifold, Muffler and Catalyst in order to improve sound suppression while reducing engine pumping losses and exhaust emissions. One dimensional predictions from acoustic theory are used to configure the initial design. Experiments with the fabricated hardware are then conducted in an engine dynamometer facility and the results were compared to the existing production system as a benchmark engine experiments show that the PMMC concept provides enhanced upstream sound suppression reducing the need for restrictive downstream silencers.

The work [2] describes the development and validation of procedures for measuring the acoustic properties of an individual component of an exhaust system on a test bed under engine acceleration representing realistic operating conditions. Collection of such data requires reliable induct pressure measurements where the contamination of the acoustic pressure field by shear layer turbulence and other hydrodynamic disturbances make this a formidable challenge.

An experimental study is described in [3] to explore the dominant sound generation mechanisms of the spectral components governing the overall noise level of centrifugal compressors. At the design speed with supersonic flow conditions in the rotor blade channels, blade tone noise and buzz-saw noise are the main contributors. On the inlet, rotor-alone noise is the main source while rotor–stator interaction noise dominates on the outlet side in case of vaned outlet diffusers. Over a large range of rotor speeds with subsonic flow conditions, radial compressor noise is dominated by tip clearance noise which is produced by the secondary flow through the gap between rotor blade tips and the casing wall which in turn gives rise to the rotating instability phenomena observed earlier in axial-flow machines.

In ref. [4] is investigated the aerodynamic sound generated by a partially ducted centrifugal pump rotor. The primary objective of the method was to determine the spectral characteristics of the sound source by isolating the effects of acoustic phenomena such as duct resonances or sound reflections. Pump-radiated sound pressure spectra were measured for different impeller rotational speeds, keeping the operating condition constant. The spectra, assumed to be expressed as the product of a source spectral distribution function and an acoustic frequency response function, were then decomposed into a product form following a computer implemented algorithm.

Liner non-uniformities [5], such as distributed impedances, may have a direct influence on the performance of turbofan engine liners. A relevant problem to study these effects is that of sound propagation in a hardwalled duct of circular cross-section, fitted with a region of non-uniform liner. Given the complex modal input amplitudes at one end of the hard-walled duct, the problem is to
compute the complex modal output amplitudes at the other end. In the present paper, a Multi-Modal Propagation Method (MMPM) is proposed to solve this problem in the absence of mean flow. For simplicity, the liner impedance is set piecewise constant along the duct, while being arbitrarily variable along the circumference of each segment. The principle of the method is to expand the sound pressure and axial velocity into double infinite series using the rigid duct modal basis, and then to follow the projection coefficients evolution along the duct axis.

The scattering of plane acoustic waves at area expansions in flow ducts is analyzed in [6] using the vortex sheet model where the flow at the expansion is modeled as a jet, with a vortex sheet emanating from the edge. Of particular interest is the influence of the flow field on acoustic scattering and absorption. First, it is demonstrated that the stability properties of the shear layer can be simulated by modifying the edge condition within the vortex sheet model. Second, it is demonstrated that the acoustic transmission through the jet expansion region can be determined by neglecting the scattering there. This result supports the assumption that the main part of the scattering occurs at the area expansion at least for the low-frequency range.

The present model describes the pressure wave propagation in a muffler of a combustion engine. The approach is general for the analysis of damping of acoustic pressure waves. The purpose is to emphasize the research potential of FE simulations using a highly versatile computer code [7]. The 3D model, consisting of several separate sections and pipes divided by thin perfectly rigid walls is investigated in two configurations: (a) continuous pipes and (b) perforated pipes. The analysis gives the transmission loss in the frequency range 100 Hz–1000 Hz for the two configurations.

2. GEOMETRY OF THE MODEL

The model geometry consists of three separate resonator chambers divided by thin walls. The inlet and the outlet correspond to the connection in the direction of the engine and of free air, respectively.

3. EQUATIONS OF THE HARMONIC WAVES PROPAGATION

The problem is solved in the frequency domain using the time-harmonic Acoustics application mode. The model equation is a slightly modified Helmholtz’s equation for the acoustic pressure, \( p \):

\[
\nabla \cdot \left( \frac{-\nabla p}{\rho} \right) - \frac{p \omega^2}{\rho c_s^2} = 0
\]

where \( \rho \) is the density, \( c_s \) is the speed of sound, and \( \omega \) is the angular frequency. The density needs to be included in the equation in cases where variations in density in different materials exist. The model assumes that in the low-frequency range, reactive damping prevails. Resistive damping is therefore not included. The following equation defines the damping of the acoustic energy, \( d_w \):

\[
d_w = 10 \log \left( \frac{w_o}{w_i} \right)
\]

Here, \( w_o \) and \( w_i \) denote the acoustic energy at the outlet and inlet, respectively. The acoustic energy is calculated using the following equations:

\[
w_o = \int_{\partial \Omega} \frac{|p|^2}{2 \rho c_s} dA; \quad w_i = \int_{\partial \Omega} \frac{|p|^2}{2 \rho c_s} dA
\]
4. BOUNDARY CONDITIONS

The boundary conditions are of four different types. At all the solid boundaries, which include the outer walls of the muffler, the dividing walls between the resonator chambers, and the walls of the pipes, sound hard (wall) boundary conditions are used:

\[ \left( -\frac{\nabla p}{\rho} \right) \cdot \mathbf{n} = 0 \]  \hspace{1cm} (4)

At the inlet boundary is a combination of incoming and outgoing plane waves:

\[ \left( -\frac{\nabla p}{\rho} \right) \cdot \mathbf{n} = \frac{i\omega}{\rho c_s^2} p - \frac{2i\omega}{\rho c_s^2} p_0 \]  \hspace{1cm} (5)

In this equation \( p_0 \) denotes the applied outer pressure and \( i \) the imaginary unit. At the outlet boundary, an outgoing plane wave is set:

\[ \left( -\frac{\nabla p}{\rho} \right) \cdot \mathbf{n} = \frac{i\omega}{\rho c_s^2} p \]  \hspace{1cm} (6)

The perforations impose in the muffler sections the continuity of pressure between the pipe and the muffler chambers.

5. NUMERICAL RESULTS

Two FE analyses were performed for inlet plane waves of frequencies ranging from 100 to 1000 Hz in steps of 100 Hz as parameter. The acoustic pressure is retrieved for each value of the frequency. For information, two such results are plotted. On Fig. 2 is the isobar plot of acoustic pressure for the initial geometry at 240 Hz, chosen because it corresponds to a muffler resonance. For comparison on Fig. 3 is the corresponding plot for the perforated pipe, at 270 Hz, which corresponds to a resonance in this case. To be noted that 240 Hz which in the first case was a resonance, in this case does not correspond to a resonance.

Fig. 4 shows the result of a parametric frequency study of the initial geometry. This plot reveals that the damping is better at higher frequencies, with the exception of several deep dips throughout the frequency range. The dips correspond to the resonance frequencies for different parts of the muffler system.

Figure 2 Acoustic pressure levels at 240 Hz, initial geometry

Figure 3 Acoustic pressure levels at 270 Hz, perforated pipe

Figure 4 The damping (dB) in the non-perforated muffler as a function of the frequency (Hz).

Figure 5 The damping (dB) in the perforated muffler as a function of the frequency (Hz).
The acoustic damping of the perforated pipe is shown on Fig. 5. The number, size and location of the perforations were arbitrary. As a consequence, the damping is not necessarily better for wider frequencies. Even so, it is apparent that a better damping is obtained in the low frequency range (<150 Hz)

6. CONCLUSIONS

The FE simulation has a wide application potential. Starting from an initial geometry, it is possible to extend the study to a variety of geometrical modifications of the initial geometry. In this case a series of perforations has been introduced for the first pipe. The parametric study is of upmost importance in designing a muffler for a specific application. From the source frequency analysis, can be obtained a series of useful results in acoustic design.

REFERENCES