

CFD Investigation of the Effect of Fluid-Structure Interaction on the Transmission Loss of ICE Silencers

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ABSTRACT

In the last decades numerical simulations have become reliable tools for the design and the optimization of silencers for internal combustion engines. Different approaches, ranging from simple 1D models to detailed 3D models, are nowadays commonly applied in the engine development process, with the aim to predict the acoustic behavior of intake and exhaust systems. However, the acoustic analysis is usually performed under the hypothesis of infinite stiffness of the silencer walls. This assumption, which can be regarded as reasonable for most of the applications, can lose validity if low wall thickness are considered. This consideration is even more significant if the recent trends in the automotive industry are taken into account: in fact, the increasing attention to the weight of the vehicle has lead to a general reduction of the thickness of the metal sheets, due also to the adoption of high-strength steels, making the vibration of the components a non negligible issue. In this work, a CFD compressible solver capable to describe the fluid-structure interaction problem has been implemented and validated. The approach is based on the adoption of two different domains for the solution of the governing equation for the fluid and for the structure. At each time-step, the pressure field computed on the fluid domain is used as an input for the solution of the displacement of the structure. A mesh motion strategy is adopted, for both the fluid and solid domain, in order to describe the deformation of the walls. The solver has been applied for the simulation of the acoustic behavior of a simple expansion chamber, considering different wall thickness, in order to investigate the effect of the structure vibration on the transmission loss. Results have been validated resorting to experimental measurements available in the literature.

CITATION: Della Torre, A., Montenegro, G., and Onorati, A., "CFD Investigation of the Effect of Fluid-Structure Interaction on the Transmission Loss of ICE Silencers," *SAE Int. J. Passeng. Cars - Mech. Syst.* 9(3):2016, doi:10.4271/2016-01-1815.

INTRODUCTION

The analysis and the optimization of the acoustic behavior of intake and exhaust systems is a fundamental stage of the design of an internal combustion engine. The development process is focused to the control of both the level and quality of the noise, in order to satisfy the legislation limits and provide a pleasant interior and exterior sound. In an internal combustion engine, the gas dynamic noise is originated by the periodic cylinder gas exchange process, which determines highly unsteady flows and pressure wave propagation in the intake and exhaust systems, and it is radiated in the surrounding from the open inlet and outlet ends. In order to control the emitted noise, reactive and dissipative silencers are positioned along the intake and exhaust lines, with the aim to damp or emphasize certain spectral components of the noise according to the expected abatement target. The design and the optimization of the silencing system is a complex operation, which is usually performed by means of concurrent experimental measurements and numerical simulations. In this scenario, acoustic linear or non-linear simulation codes have nowadays become reliable tools, which are commonly used in the industry for the design and the optimization of silencers. These can rely on different approaches, including 1D [1, 2, 3], quasi-3D [4, 5] and fully-3D [6, 7], which are usually applied at different stages of the optimization process on the basis of the balance between computational runtime and level of approximation

which can be accepted [8, 9]. In this way, the simplest approaches are used at the beginning of the optimization process, when several solution needed to be tested and the computational time is the key factor. On the other hand, more complex approaches are applied to progressively refine the most promising solutions in order to define the final optimized concept.

However, the acoustic analysis of the silencer is usually performed under the hypothesis of infinite stiffness of the silencer structure, neglecting the influence of the vibrations of the silencer walls on its acoustic behavior. This assumption can be regarded as valid for many applications, since the thickness of the silencer walls is usually sufficiently high to prevent the excitation of the structure vibrational modes in the frequency range of the gas dynamic noise source (1-3000 Hz). On the other hand, in case of low stiffness of the structure, the interaction between pressure wave propagation and wall vibrations cannot be longer neglected. Recent trends in the automotive industry are raising the interest towards this issue: as a matter of fact, the increasing attention to the weight of the vehicle and the availability of more performing high-strength steels has lead to a general reduction of the thickness of the metal sheets, making the vibration of the components a non negligible issue [10, 9]. One common approach to take into account the structural vibrations during the design phase of the silencer consists of performing two

separate simulations of the pressure wave motion and of the structure dynamics. This is usually done with two different simulation codes, namely an acoustic code (being linear or non-linear) and a structural code (based on Finite Element approach or, less commonly, Finite Volume). At first, the identification of the vibrational modes of the structure is exploited in order to assess whether some resonance frequencies are present in the frequency range of interest. Then, if structural resonances, which can potentially affect the acoustic behavior of the chamber, are found, the information coming from the structural code is used to model the vibrational behavior of the walls during the acoustic simulation. In this way, a prediction of the real acoustic behavior of the silencing device is provided. However, this approach has the drawback of being based on two different simulations which are uncoupled, meaning that the vibration of the wall is not related to the entity of the pressure wave which is exciting that vibrational mode.

In this work, an approach based on the strong coupling between the solution of fluid and structure domain has been adopted. The approach is implemented in the framework of a single simulation code, in which the governing equations for the fluid and for the structure are solved simultaneously on two different regions. Both the regions are discretized on the basis of a finite volume approach and they continuously share the coupling information (interfacial forces and interface displacement) over each time-step. The solver has been validated resorting to a single expansion chamber configuration, comparing the measured transmission loss available for different thickness of the chamber wall.

CASE STUDY

The case study considered in this work consists of a simple cylindrical expansion chamber, having a diameter of 155 mm and a length of 216 mm. Two different configurations are studied, each one with the same geometrical dimensions but different material and thickness of the walls, as reported in Table 1. In particular, the lower wall thickness (2 mm) can be regarded as a reasonable design parameter which can be also found in practical applications, while the higher (7 mm) has been chosen in order to guarantee a sufficiently high stiffness of the structure. These configurations have been experimentally characterized by Ferrara et al. as reported in [10], showing that, in the case of low wall thickness, the transmission loss exhibits significant deviations from the theoretical behavior, as a consequence of the vibrations of the walls excited at specific frequencies.

Table 1. Simple expansion chamber data.

Conf. ID	Material	Wall thickness
A	Common steel	7 mm
B	Stainless steel	2 mm

A comparison of the transmission losses measured for the A and B configurations is reported in Fig. 1. It is evident that the series chamber with the highest wall thickness behaves, as expected by the acoustic theory, as a typical series chamber with transparencies located at $f = a/2L$ exhibiting the dome shaped trend. In the case of

thin wall thickness it can be noticed that the abatement level is decreased between the transparencies. This effect is due to the different stiffness of the chamber walls, which seems to be excited by the pressure oscillations introduced at the inlet. In order to characterize the vibrational modes of the structure and understand their influence on the chamber acoustic behavior, a finite element analysis of configuration B was addressed using the FEM software Abaqus. The analysis showed that, in the frequency range of interest for the acoustic characterization (0-2000 Hz), the structure has around 25 resonance frequencies. In Fig. 2 the most relevant modes are visualized. It can be seen that some modes (for example those at 348 Hz and 796 Hz) are axial-symmetric, while most of them do not exhibit any symmetric behavior. This means that, even if the geometry is axial-symmetric, the problem cannot be reduced to a 2D wedge simplification since its behavior is strongly three-dimensional. On the other hand, it should be considered that, in the range 0-800 Hz, there are three axial-symmetric vibrational modes and, therefore, the 2D wedge simplification could provide reasonable results if the target of the analysis is restricted to the low frequency range.

FSI COMPUTATIONAL MODEL

The prediction of the acoustic response of the silencer, when fluid-structure interaction occurs, is a complex problem which requires a proper modeling of the behavior of the two phases involved, namely fluid and solid, and of their interaction. The computational model developed in this work is based on the adoption of a finite volume (FV) approach for the discretization of the governing equations for both fluid and structure. The overall computational domain is partitioned in two regions, adjoining but not overlapping, where the equations of fluid and solid media are solved. Proper boundary conditions are introduced at the interface between the regions in order to describe the interactions between the phases. In the following section, the set of governing equations will be presented and the solution procedure will be described.

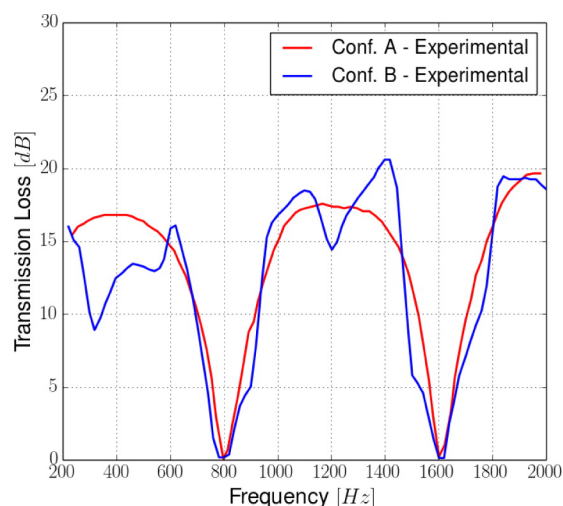


Figure 1. Experimental characterization of the transmission loss of a single expansion chamber for different thickness of the wall: A) 7 mm and B) 2 mm. Taken from [10].

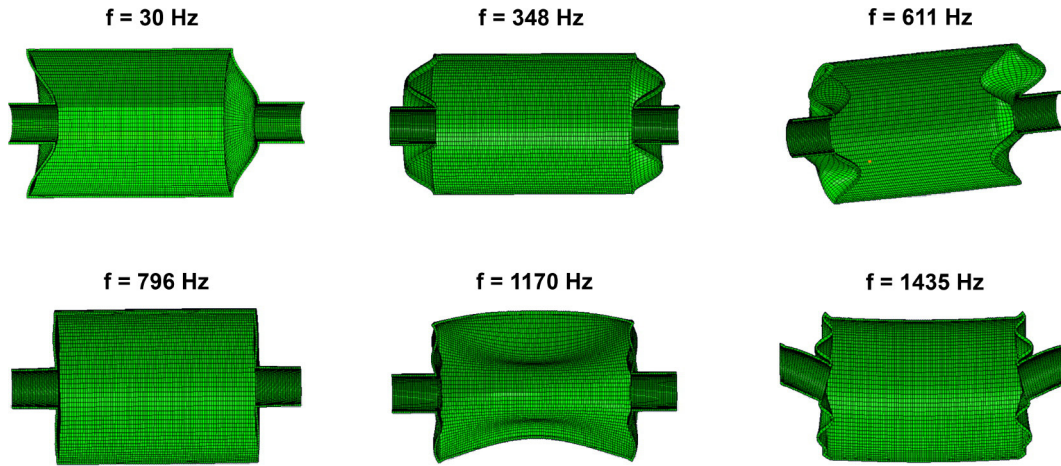


Figure 2. Visualization of some resonance frequencies of the structure and of the corresponding modes.

Governing Equations

In order to describe the acoustic wave motion in the fluid domain, the compressibility of the media must be taken into account. Therefore, the fluid was modeled as compressible, leading to the following set of governing equations [11]:

- Continuity equation, which states the conservation of mass:

$$\frac{\partial \rho_f}{\partial t} + \nabla \cdot (\rho_f \mathbf{v}_f) = 0 \quad (1)$$

- Navier-Stokes equation, derived from the conservation of momentum (linear and angular):

$$\frac{\partial \rho_f \mathbf{v}_f}{\partial t} + \nabla \cdot (\rho_f \mathbf{v}_f \mathbf{v}_f) = \rho_f \mathbf{g} - \nabla \cdot (\sigma_f) \quad (2)$$

- Energy equation:

$$\begin{aligned} \frac{\partial \rho_f e_f}{\partial t} + \nabla \cdot (\rho_f e_f \mathbf{v}_f) &= \rho_f \mathbf{g} \mathbf{v}_f - \nabla \cdot (\sigma_f \mathbf{v}_f) \\ + \nabla \cdot (\lambda_f \nabla T_f) + \rho_f Q_f \end{aligned} \quad (3)$$

The constitutive equation of the media is expressed, under the assumption of Newtonian fluid, as:

$$\sigma_f = \left(p_f + \frac{2}{3} \mu_f \nabla \cdot \mathbf{v}_f \right) \mathbf{I} - \mu_f (\nabla \mathbf{v}_f + \nabla \mathbf{v}_f^T) \quad (4)$$

This set of equations is closed by means of the perfect gas equation of state:

$$\frac{p_f}{\rho_f} = RT_f. \quad (5)$$

Since the propagation of acoustic waves occurs with negligible variations of the temperature field, the heat capacity $c_{v,f}$ of the fluid is assumed constant. In the same way, the temperature dependence of the viscosity μ_f can be neglected. Moreover, since the test condition does not consider the presence of mean flow, the velocity of the fluid under the small amplitude pressure perturbations is very low and therefore the assumption of a laminar flow regime can be assumed as valid. This makes unnecessary the adoption of any turbulence model.

On the other hand, the dynamic behavior of the structure can be described, under the hypothesis of elastic and isothermal media, by considering the linear momentum conservation law in Lagrangian formulation [12]:

$$\frac{D \rho_s \mathbf{v}_s}{Dt} = \nabla \cdot (\sigma_s) + \rho_s \mathbf{F}_{b,s}, \quad (6)$$

where σ_s is the Cauchy stress tensor and $\mathbf{F}_{b,s}$ is the resulting body force. The Cauchy stress tensor σ_s can be expressed as:

$$\sigma_s = \frac{1}{\det \mathbf{F}} \mathbf{F} \cdot \Sigma \cdot \mathbf{F}^T, \quad (7)$$

where $\mathbf{F} = \mathbf{I} + (\nabla \mathbf{u})^T$ is the deformation gradient tensor and Σ is the 2nd Piola-Kirchhoff stress tensor. By using the constitutive equation for a St. Venant-Kirchhoff material, the relation between the 2nd Piola-Kirchhoff stress tensor Σ and the Green-Lagrange strain tensor $\mathbf{E}$ can be given as:

$$\Sigma = 2\mu_s \mathbf{E} + \lambda_s \text{tr}(\mathbf{E}) \mathbf{I}, \quad (8)$$

where μ_s and λ_s are the Lamé's coefficients and $\mathbf{E}$ is defined as:

$$\mathbf{E} = \frac{1}{2} \left[\nabla \mathbf{u} + (\nabla \mathbf{u})^T + \nabla \mathbf{u} \cdot (\nabla \mathbf{u})^T \right]. \quad (9)$$