

A Coupled 1D-multiD Nonlinear Simulation of I.C. Engine Silencers with Perforates and Sound-Absorbing Material

G. Montenegro and A. Onorati

Politecnico di Milano

Copyright © 2009 SAE International

ABSTRACT

Nowadays a great attention is paid to the level and quality of noise radiated from the tailpipe end of intake and exhaust systems, to control the gas dynamic noise emitted by the engine as well as the characteristics of the cabin interior sound. The muffler geometry can be optimized consequently, to attenuate or remark certain spectral components of the engine noise, according to the result expected. Evidently the design of complex silencing systems is a time-consuming operation, which must be carried out by means of concurrent experimental measurements and numerical simulations. In particular, 1D and multiD linear/non-linear simulation codes can be applied to predict the silencer behavior in the time and frequency domain.

This paper describes the development of a 1D-multiD integrated approach for the simulation of complex muffler configurations such as reverse chambers with inlet and outlet pipe extensions and perforated silencers with the addition of sound absorbing material. The 1D-multiD integrated approach is exploited to validate the transmission loss prediction of reverse chamber configurations with inlet and outlet extensions. Results have pointed out the capability of capturing transversal resonances at high and mid frequencies. Moreover, a non linear approach is proposed to take into account the presence of the sound absorbing material into the conservation equations of a multidimensional solver. The properties of the sound absorbing material have been taken from correlations adopted in the literature for 1D models. The momentum and energy conservation equations have

been modified to take into account the interaction between the gas and sound absorbing material. Both the 1D and the integrated 1D-multiD approach have been exploited for validation, considering two different geometries: an expansion chamber with an extended outlet pipe, with the sound absorbing material placed between the pipe extension and the canning, and a perforated pipe whose cavity has been completely filled with sound absorptive metallic wool. The results obtained by the fully 1D analysis and the integrated approach are in agreement with the measured muffler performances.

INTRODUCTION

Internal combustion engines are the source of mechanical, combustion and gas dynamic noise. In particular, gas dynamic or pulse noise is related to the unsteady flows in the intake and exhaust systems, induced by the cylinder gas exchange process [1]. The level and quality of noise radiated from the open ends can be controlled by different arrangements of pipe systems and silencers, to achieve the required vehicle exterior and interior sound characteristics by attenuating or enhancing certain spectral components. Today numerical simulation codes are very useful during the design and optimization process of both intake and exhaust manifolds and mufflers, to quickly define the best geometries which can be finally refined experimentally. The attenuation features of simple and complex acoustic filters can be predicted by 1D and 2D-3D fluid dynamic/acoustic simulation codes with different levels of complexity. Nowadays nonlinear 1D and CFD fluid dynamic codes are becoming

rather common for acoustic/fluid dynamic applications [2,3], since they allow the prediction of silencer performances and radiated pulse noise when the duct system is connected to the engine source itself, submitted to finite amplitude pressure waves and significant mean-flow. These conditions are directly accounted for in the fundamental conservation equations of non-linear models. The counterpart is represented by linear acoustic models (1D or multiD), applicable only in the case of small (acoustic) pressure perturbations, which frequently is not acceptable in I.C. engines pipes. Linear acoustic codes are characterized by reduced computational times, generally 2 or 3 orders of magnitudes with respect to non-linear CFD codes. However, the greater computational cost of non-linear 1D and multiD codes is balanced by the wider applicability and validity of the provided results, ranging from the muffler Transmission Loss (TL) including high amplitude wave effects to acoustic and fluid dynamic performances of silencing systems under realistic engine perturbations. Moreover, in recent years the great potential of hybrid 1D-multiD fluid dynamic (non-linear) models has been recognized. Integrated 1D and multiD computational domains can be built to carry out the simulation of the whole engine intake and/or exhaust duct systems including the silencers, in order to dedicate the 3D approach to model the complex shapes and reserve the 1D approach to simulate simple pipe systems; the integrated simulation tool can capture both the non-linear effects dictated by the finite amplitude wave motion with significant mean flow and the complex wave motion in 3D elements, providing a better prediction of the attenuation features of the muffling systems. However, the simulation of silencers with complex internal geometry, involving perforated pipes and sound absorbing material, requires advanced fluid dynamic non-linear models to correctly represent the acoustic behaviour of these elements. Much work has already been done in the field of 1D fluid dynamic codes, achieving reliable results in the case of simple and multi-pass resonators with perforated ducts and with/without sound absorbent material. The available approaches are based mainly on the segmentation of the perforated pipe and the related cavity [4]. But in the case of CFD non-linear codes, based on the solution of the Navier-Stokes (RANS) equations, further improvements are needed to represent the small holes of the perforate in a convenient way, resorting to suitable refined grids. Moreover, different approaches are possible to include the effect of sound absorbing material. This is frequently used in current automotive mufflers due to its efficacy in the abatement of medium-high frequency noise, cost effectiveness and simplicity of installation. The propagation of sound waves in the absorbing material can be modelled on the basis of i) a porous material approach or ii) a flow resistivity approach, both provid-

ing specific terms to be included in the conservation equations [5].

In this paper both a multiD and a 1D non-linear model have been developed and integrated to achieve an hybrid simulation code, applicable to the simulation of I.C. engine silencers with complex shape. In particular, the CFD code OpenFOAM and the 1D code GASDYN have been further developed to cope with the simulation of perforates with/without sound absorbing material, to predict the TL of typical silencer geometries found in the available literature. Flow reversals and perforates with/without sound absorbing material have been investigated, pointing out a good comparison between experimental data and predicted results.

GOVERNING EQUATIONS

1D MODEL

The 1D model adopted for the simulation is a research code developed by the authors (GASDYN) [6, 7]. In the 1D model, the conservation equations of mass, momentum and energy are expressed in a strong conservative form and applied to the simulation of complex systems of pipes, with source terms to account for the friction (f_w) and the heat transfer (q) between the gas and the walls:

$$\frac{\partial(\rho F)}{\partial t} + \frac{\partial(\rho u F)}{\partial x} = 0 \quad (1)$$

$$\frac{\partial(\rho u F)}{\partial t} + \frac{\partial(\rho u^2 F + p F)}{\partial x} = p \frac{\partial F}{\partial x} - \frac{2}{D} \rho u |u| f_w F \quad (2)$$

$$\frac{\partial(\rho e_0 F)}{\partial t} + \frac{\partial(\rho e_0 u F + u p F)}{\partial x} = \rho(q + q_{re}) F, \quad (3)$$

where F represents the pipe cross section. The system of equations is closed with the perfect gas equation of state. The set of equations is then solved by applying explicit Godunov type methods based on the HLLC Riemann solver and on its extension to the second order accuracy (WAF) [8]:

$$W_i^{n+1} = W_i^n + \frac{\Delta t}{\Delta x} \left(F_{i-\frac{1}{2}} - F_{i+\frac{1}{2}} \right) - \Delta t C_i^n \quad (4)$$

where the intercell fluxes $F_{i \pm \frac{1}{2}}$ can be evaluated according to a first order approach (see Eq. A-6 in appendix) or to its second order formulation (Eq. A-13). Moreover, the source term C is discretized according to the stencil proposed by other authors [9].

MULTI-DIMENSIONAL MODEL

The OpenFOAM code was adopted for the multidimensional simulation of the non-linear Euler set of

equations. OpenFOAM [10, 11] is a general purpose finite volume solver, written in C++ object oriented programming language, and provides the user with suitable libraries to discretize partial derivatives operators applied to arbitrarily oriented meshes. The Euler equations have been adopted to describe the unsteady flows also in the multidimensional domain, since at high Reynolds numbers (typical of intake and exhaust systems) the problem is dominated by convection, making viscous phenomena negligible [12]:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{U}) = 0 \quad (5)$$

$$\frac{\partial \rho \vec{U}}{\partial t} + \nabla \cdot (\rho \vec{U} \cdot \vec{U}) = -\nabla p \quad (6)$$

$$\frac{\partial \rho e_0}{\partial t} + \nabla \cdot (\rho e_0 \vec{U}) = -\nabla \cdot (p \cdot \vec{U}) \quad (7)$$

Consequently the HLLC and the WAF Riemann solver have been exploited to calculate the transport term of Eq. 5-7.

Considering the conservation of a generic property ϕ :

$$\frac{\partial \rho \phi}{\partial t} + \nabla \cdot (\rho \phi \vec{U}) = 0, \quad (8)$$

the integral of the transport term, by means of the Gauss theorem, becomes a summation of the volumetric fluxes of the conserved variables through the intercell faces, evaluated at the face center. The HLLC and the WAF Riemann solvers have been exploited to evaluate directly the volumetric flux at the intercell face:

$$\sum_{f=1}^n \rho \phi_f U_f S_f = \sum_{f=1}^n F_f^{HLLC, WAF}. \quad (9)$$

To solve the Riemann problem the velocity vector $\vec{U}_f$ has been decomposed into two components: one normal to the cell face, $\vec{U}_n$, and another laying in the face plane, $\vec{U}_t$. The former is responsible of the net flux across the face of the conserved variables, the latter, giving no contribution to this flux, is advected as a fluid property:

$$\begin{aligned} \vec{U}_n &= \vec{n}(\vec{U}_f \cdot \vec{n}), \\ \vec{U}_t &= \vec{U}_f - \vec{n}(\vec{U}_f \cdot \vec{n}). \end{aligned} \quad (10)$$

This assumption allows to reduce the 3D problem to the solution of an equivalent 1D problem, as described in the Appendix, advecting the tangential component as a fluid property.

1D ABSORPTIVE SILENCERS MODELING

The sound absorbing material dissipates the acoustic energy as heat, due to the flow resistance of material.

To take into account this dissipation, the momentum equation has been modified by including an additional source term which reproduces the effects of the flow resistance operated by a material layer [13]:

$$\frac{\partial (\rho u F)}{\partial t} = -\frac{\partial (\rho u^2 F + p F)}{\partial x} + p \frac{\partial F}{\partial x} \quad (11)$$

$$- \rho F \left(\frac{2}{D} u |u| f_w + u \pi \right) \quad (12)$$

where π expresses the material resistivity. Resistivity is the most important characteristic of a porous material and can be defined by the following equation:

$$\pi = -\frac{1}{u} \frac{\Delta p}{\Delta x}. \quad (13)$$

A correlation to calculate the material resistivity has been proposed by other authors as follows [13]:

$$\pi = \pi_0 + C \pi_0^D |u| \quad (14)$$

$$\pi_0 = \frac{1}{1000} \frac{\eta}{|r_m^2|} A \mu^B \quad (15)$$

where η is the dynamic viscosity of the gas, $|r_m^2|$ is the mean square average fiber radius, μ the ratio between the material packing density and the material density and u is the gas velocity. The A, B, C, D coefficients depend on the type of sound absorptive material used. For example, for rock wool the following values are proposed: A = 4.4, B = 1.59, C = 0.75, D = 0.57 [14]. The energy conservation equation can be modified to take into account the increased surface available for the heat transfer, similarly to what happens in a bundle tube heat exchanger, adding to Eq. 3 the following source term:

$$q_{fiber} = \frac{3\mu}{r_m D} h_{fiber} (T_{fiber} - T_g) \quad (16)$$

where r_m is the fiber mean radius.

3D ABSORPTIVE SILENCER MODELING

Different approaches can be followed to deal with the modeling of flows inside porous or fibrous materials. The most detailed method is based on the microscale analysis of the material itself. This analysis is carried out considering an average cell size which is a fraction of the mean pore size; this means that the material is represented as a wall type boundary condition which forces the flow inside of the pore free space. The main drawback of this approach is the reduction of the cell size to very low values, limiting in this way the numerical analysis to small pieces of material, since the computation time required to simulate a volume of a silencer filled up with the porous material would be not feasible. The approach followed in this work is based onto a macroscale analysis of the sound absorbing

material, considering the treatment of the material effect as a body force. The underlying assumption is that the macrostructure can be modeled once again by the adoption of a parameter which takes into account the flow resistivity ($\vec{\pi}$) without considering the real flow path inside of the fiber pores:

$$\vec{\pi} = -\frac{1}{\vec{U}} \nabla p \quad (17)$$

leading to the following formulation of the momentum conservation equation:

$$\frac{\partial(\rho \vec{U})}{\partial t} + \nabla \cdot (\rho \vec{U} \cdot \vec{U}) = -\nabla p - \vec{\pi} \cdot \vec{U}. \quad (18)$$

For domains containing the sound absorbing material, Eq. 18 is solved only for the region associated with the material, while Eq. 6 is applied everywhere else. This is realized in OpenFOAM creating specific regions inside the computational mesh and adding the resistive term to the solution matrix only for the cells contained within the absorbing material region.

COUPLING METHOD AND BOUNDARY CONDITIONS

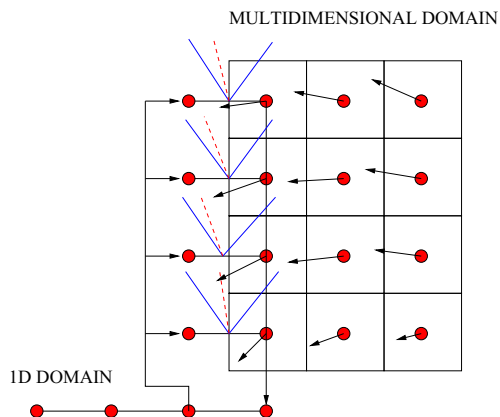


Figure 1: Procedure adopted for the integration the 1D and multiD codes. The solution of the local Riemann problems is pointed out.

A good compromise between accuracy of the acoustic behavior prediction and simulation run-time is the adoption of integrated 1D and 3D simulation tools. This strategy is not something new in the field of internal combustion engine simulations. Several works concerning this topic have been published during the last years [15, 16].

To take into account the effects of eventual variation of the multidimensional domain on the 1D side, the so called "strict coupling" has been proposed. In this procedure the two codes are passing the information forth and back at each time step in such a way that one domain is affected by what happens in the

other one [15, 16]. In order to truncate the 3D domain in regions close to highly 3D shapes, an advanced procedure has been adopted, therefore allowing the treatment of non-uniform fields on the 3D side without compromising the stability of the solution. This approach (see Figure 1) involves the solution of the Riemann problem for every cell which owns the face constituting the boundary surface and is fully described in recent papers [17, 18]. Another important reason why the coupling between the 1D and the CFD models has been developed is the possibility to carry out simulations involving the engine as a pressure pulsations source. This condition at the boundary cannot be easily reproduced recurring to a boundary condition approach. As a matter of fact, imposing the pressure trace, as a function of the time or of the crank angle, could not be enough to reproduce the exact flow condition occurring in the exhaust or intake system, since a mismatch between the measured and calculated mass flow may occur. As shown by the authors, the integrated 1D-multi approach can give encouraging results in this way. However, in the case of acoustic simulations, its adoption may be argued not to be strictly required. This approach was used because the implementation of specific boundary conditions, such as non-reflecting boundaries, can be carried out very easily by a 1D model. In fact, pressure transmissive boundary conditions in CFD codes may be not completely anechoic, they can reflect small pressure waves whose magnitude scales with the jump in pressure across the approaching perturbation. In this work the boundary condition reproducing a non-reflecting impulse perturbation has been adopted, as described in [17].

RESULTS

The integrated 1D-multiD code (Gasdyn-OpenFOAM) has been applied to different configurations of silencers with complex shape, ranging from reactive mufflers with flow reversals to perforates with/without sound-absorbing material. Every configuration has been modeled resorting to the coupled 1D-3D approach and also to the fully 1D approach, to compare the performances and the results of the two simulation tools with experimental data of transmission loss (TL). The expansion chamber with flow reversals is a good test to point out the capability of the simulation technique to capture the main features of the attenuation curve in the frequency domain when the wave motion is significantly multi-dimensional.