

# Assessment and Experimental Validation of a 3D Acoustic Model of a Motorcycle Muffler

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## ABSTRACT

The intake and exhaust lines provide the main abatement of the acoustic emissions of an Internal Combustion Engine (ICE). Many different numerical approaches can be used to evaluate the acoustic attenuation, which is commonly expressed by the Transmission Loss.

One-dimensional (1D) and three-dimensional (3D) simulations are conventionally carried out only considering the acoustic domain of the muffler or of the air-box. The walls of the acoustic filter are considered fully rigid and the interaction between the acoustic waves and the structure is consequently negligible. Moreover, the effect of the manufacturing characteristics and the attenuation of the acoustic waves due to the fluid viscous-thermal effects are also commonly disregarded in the numerical analysis of the filters. In addition, the presence of a catalytic converter or a filter cartridge may have an influence on the numerical results. All these aspects, however, can remarkably affect the matching between simulations and experiments both at high frequencies and at medium-low ones.

In the present study, the effect of the aforementioned issues on the prediction capabilities of a 3D FEM model are analysed, with particular attention to the prediction of the acoustic attenuation of a commercial muffler. In detail, simplified models were developed in order to include the manufacturing features into acoustic simulations with almost negligible increase of the computational time.

Numerical models were applied to a commercial motorcycle muffler featuring three chambers and a catalytic converter, and then validated by means of a specific experimental campaign that was carried out on an acoustic test rig. The tests were run at ambient temperature and no-flow conditions, obtaining good agreement between theory and experiments.

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## INTRODUCTION

In the last few years, the exhaust system of motorcycles has become one challenging topic of design for many industrial companies. Noise emission standards contrast with the requests of end-users, who are increasingly interested in the Sound Quality of the engine. Therefore, the new muffler design approach needs to take in to account both the total acoustic energy emitted and the frequency bands of spectrum emission. These two aspects, together, define the characteristic exhaust sound of an engine.

Nowadays the development of a new Internal Combustion Engine is mainly performed with 1D numerical codes that allows the first design and performance prediction of the engine coupled with its intake and exhaust systems, as well its

sound emissions. In particular, the first design fixes the main features of the exhaust system such as volumes, length and diameter of pipes, and consequently, the shape of the muffler. In order to refine the acoustic design of the muffler and better verify its acoustic performance 3D software FEM or BEM are needed. Indeed, the 3D codes allow to take into account the detailed shape of the muffler.

A key aspect of the above design process of a muffler is the accuracy of the 3D models, whose definition has a relevant influence on the acoustic performance. In other words, the higher the accuracy of the model the more accurate is the prediction of the 3D software. However, the increase of details causes an increment of the time required both to build up the model and to solve the calculus. Thus, the prearranged

analysis of the muffler characteristic is a key aspect for simplifying the 3D model without affecting the accuracy of the 3D simulation.

The main aspects that affect the prediction of the acoustic model are the manufacturing characteristics of the silencer, as highlighted by Ferrara et al [1, 2]. Their works show, with a large experimental and numerical campaign, how the manufacturing features can affect the performance of mufflers even on simplified shapes like single chamber and dual chamber mufflers. A particular manufacturing feature, analysed by Siano et al. [3], is the leakage between chambers. The paper presents a numerical analysis on a commercial automotive muffler, performed both with BEM and with FEM codes. This study was focused on the evaluation of the acoustic performance depending on temperature, mean flow and acoustic insulation materials. The numerical and experimental analysis performed by the authors has shown that the leakage areas between inner chambers of the muffler deeply affect the acoustic attenuation characteristics and its shifting in frequency.

Another feature, strongly related to the manufacturing of the mufflers, is the thickness of walls and consequently, acoustic boundary condition of hard or soft walls. Knutsson M. et al. [4] presented a study on the acoustic effects of non-rigid walls in Internal Combustion Engine (ICE) intake systems. The authors had performed an experimental measurement campaign. In particular, they placed the whole intake line in a box filled with sand, in order to reduce the fluid-structure interaction. Their results showed that yielding walls increase the Transmission Loss of the intake system, in the whole frequencies range of investigation, up to 2 dB compared to the test with rigid walls. The authors compared the experimental results with a 3D FEM model. The numerical model had an acoustic attenuation of 2-3 dB less than experimental with external sand loading. The authors asserted that the difference was due to both an inability to establish totally rigid walls using sand loading and to the damping of the sound waves in the fluid. According to their first study, Knutsson M. and Åbom M. [5] [6], analyzed the acoustic attenuation performance of the intercooler. The authors presented a hybrid approach based on the transfer matrix method (TMM) and the 3D linear acoustic finite element method. The symmetric central body of the intercooler was modelled by means of the transfer matrix considering a mean flow and viscothermal effects in narrow ducts. Whereas, the inlet and outlet section had a complex 3D geometry, therefore, they were modelled by means of FEM code. In order to simulate the effects of yielding walls and propagation losses in the inlet and outlet intercooler sections, the damping was applied by putting the imaginary part of the sound speed of the fluid equal to one percent of the real part. Unfortunately, the authors did not explain in detail how they evaluated this quantity.

A third feature to take into account is the viscous thermal damping coefficient, which is not related to the manufacturing. The viscous thermal damping coefficient increases its effect by reducing the equivalent hydraulic diameter of the ducts. Knutsson and Åbom analyzed such phenomena in their work on the propagation of acoustic waves in air cooling system [14]. Their research shows the strong effect of viscous thermal damping on the prediction of acoustic performance of the geometry analyzed.

Finally, a more accurate model leads to higher results accuracy, but with higher simulation computational costs. The last aspect is not compatible with the design timing imposed by the motorcycle market. The present paper introduces a full analysis of the modelling features with the aim to identify the optimum between model simplification and accuracy of numerical prediction. The whole study was performed on a motorcycle muffler produced by Ducati™. The muffler was simulated with a 3D Finite Element software, Virtual.Lab Acoustic®, which uses SYNOISE® as a numerical solver, whereas the experimental tests were performed on the acoustic test rig of the University of Florence [1].

The experimental and numerical analysis were focused on the analysis of the effects due to the leakage areas between inner chambers, the effect of viscous thermal losses and the interaction between fluid and structure. On the acoustic test rig experimental tests had been performed on a real muffler and on a prototype specially built to limit these detrimental losses. By means of these comparisons it was possible to estimate the effects of the manufacturing process on the acoustic performances.

At the same time, the different muffler configurations, tested on the test rig, were modelled with 3D FEM code and the different numerical strategies to improve the match between experimental and numerical results were identified. The effectiveness of the models was evaluated in terms of their complexity level and accuracy of the results.

### 3D FINITE ELEMENT METHOD FOR ACOUSTIC

The 3D Finite Element Method for acoustics is used to solve acoustic wave propagation problems in enclosed volumes. This numerical method was applied with success in many studies to evaluate the Transmission Loss of acoustic filters [3] [15] [5].

The 3D acoustic FEM theory is well known and Desmet and Vandepitte show its complete dissertation in their work [13]. For sake of brevity, in this paragraph the basic concept and equations of the method are only reminded. The 3D FEM approach is based on the tridimensional Helmholtz equation (Eq.1):

$$\nabla^2 p(x, y, z) + k^2 p(x, y, z) = 0$$



where  $p(x,y,z)$  is the acoustic pressure,  $k=\omega/c=2\pi f/c$  is the acoustic wavenumber and  $c$  is the speed of sound. The solution of differential equation (1) needs the definition of Boundary Conditions (BCs) of the domain. The BCs could be of three different types: imposed pressure (Eq. 2)

$$p(x,y,z) = \bar{p} \quad (2)$$

imposed normal velocity (Eq. 3)

$$v_n = \frac{j}{\rho_0 \omega} \frac{\partial p}{\partial n} = \bar{v}_n \quad (3)$$

and imposed normal impedance (Eq. 4):

$$p = \bar{Z} v_n \quad (4)$$

The above interior acoustic problem is then transformed into an equivalently weighted residual formulation. Such mathematical approach allows one to map the fluid domain by means of elements whose length is provided by the maximum frequency of analysis. The frequency defines the wavelength  $\lambda=c/f$  of the acoustic phenomenon, as a consequence, in order to have a good level of accuracy at least 10 elements per acoustic wavelength are required. The acoustic field variables are expressed in terms of shape functions, which are defined within finite elements.

The acoustic pressure or velocity can be evaluated in each node of the model resolving the algebraic system shown in (Eq. 5):

$$([K_a] + j\omega[C_a] - \omega^2[M_a])\{p_i\} = \{F_{ai}\} \quad (5)$$

where the vector  $\{p_i\}$  contains the unknown nodal pressure,  $[K_a]$ ,  $[C_a]$  and  $[M_a]$  are the acoustic stiffness, damping and mass matrices respectively, whereas the vector  $\{F_{ai}\}$  contains the contribution of input pressure and velocity boundary conditions.

The equation (5) shows that the acoustic phenomena are affected only by geometry, the boundary conditions and fluid properties. In particular, the fluid is described by the two state variables density and sound speed, which consequently have a critical role on the acoustic behavior of the geometry.

## EXPERIMENTAL ACOUSTIC TEST RIG

Ferrara et. al [1] have described in detail the design and the components of the acoustic test rig used to evaluate the acoustic properties of mufflers in terms of acoustic impedance and Transmission Loss (TL) [12]. These parameters are evaluated knowing the incident and reflected waves at the inlet

and outlet of the muffler [7] [8]. The decomposition of waves is performed by the Multi Microphone Technique (MMT) [9], that allows, on the aforementioned test rig, the acoustic characterization of muffler prototypes in the frequency range of 200-5000Hz without mean flow.



Figure 1. Acoustic test rig with the new microphone spacing.

The test rig adopted in the present paper is an upgrade of the one described by Ferrara et al. [1]. The actual configuration has a more extended working frequency range and a lower error on the waves decomposition. Such improvement was obtained using four microphones on each measurement section, one more with respect to the earlier configuration and changing the relative spacing between microphones (Figure 1).

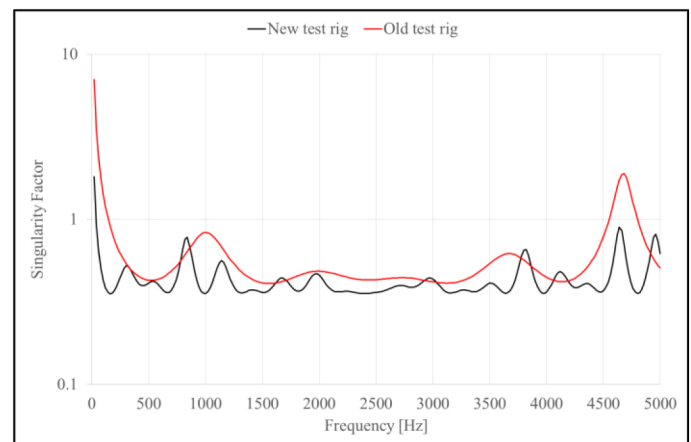


Figure 2. Comparison of the SF between old and the new test rig.

The error associate to the wave decomposition using MMT can be quantified by means of the Singularity Factor (SF), as explained in the study of Ih J. and Jang S. [9]. Figure 2 shows the comparison in terms of the Singularity Factor (SF) between the three and four microphone test rig respectively. The error is considered negligible if the SF is lower than one. The new configuration with four microphones ensures a wider frequency range of measurement, in particular the lower frequency limit decreases from 200 Hz to 30 Hz. The experimental tests are performed setting a planar pressure field generated by eight speakers driven by a waveform generator. The intensity of the acoustic field is usually set to 120 dB and monitored by means of a feedback measurement, whereas its frequency is modified with a step of 20 Hz on the whole range of study.

## MUFFLER GEOMETRICAL DESCRIPTION

The muffler analyzed is a motorcycles silencer made up of three elliptical chambers connected to each other by ducts and a circular opening (Figure 3). The catalytic converter is integrated in the duct-in of the first chamber. Finally, the muffler is equipped with inlet and outlet ducts customized in order to be connected to the test rig.

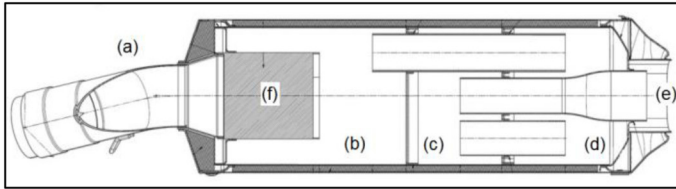


Figure 3. Muffler under test: (a) inlet duct, (b), (c) and (d) chambers, (e) outlet duct, (f) catalytic converter.

This kind of muffler is known as reactive muffler. Its acoustic attenuation is due to the resonator cavities, which reflect the incident acoustic waves. Mean flow effects do not affect such mufflers if the Mach number is lower than 0.05 [12].

## EXPERIMENTAL ANALYSIS

The aim of the experimental campaign was the determination of the manufacturing and geometrical features that mainly affects the acoustic behavior of the muffler. The comparison between the muffler configurations was performed by means of the coherence of the Transmission Loss curves, evaluated using the determination coefficient (Eq. 6):

$$R^2 = 1 - \frac{\sum_{i=1}^n (y_i - \hat{y}_i)^2}{\sum_{i=1}^n (y_i - \bar{y})^2} \quad (6)$$

where  $n$  is the total number of frequencies investigated,  $y_i$  is the data observed,  $\bar{y}$  is the mean of the data observed and  $\hat{y}_i$  is the expected value.

For each test the Transmission Loss was evaluated as function of the Helmholtz number ( $He$ ) which is expressed in the Eq. 7:

$$He = \frac{2\pi f D}{c} \quad (7)$$

where  $c$  is the speed of sound,  $f$  is the frequency and  $D$  is a characteristic geometrical feature of the muffler.

The fluid-structure interaction and the leakages between chambers are identified as the main manufacturing features that affect the acoustic performance of a silencer. In the present work, the listed manufacturing features were analyzed with a set of experimental tests performed on the muffler.

The muffler is made up of two different casings (Figure 3). The internal one is made of a steel plate (with thickness less than 1mm) and defines the fluid domain of exhaust gases, while the external one is made of a steel plate with thickness of 1,5mm and gives the structural consistency and the external appearance. Between the two casings it is placed a glass wool layer to limit the heat transfer.

In order to test the structure interaction of the outer casing, the muffler was put into a sandbox and the resulting Transmission Loss was compared with the one obtained with a test in free air (Figure 4). The comparison of the two TL highlighted that the outer case behaved as a hard wall, the determination coefficient between the two TL trends has a value of 0.98, which means that the two curves are consistent. For a better understanding of the small differences between the two TL, the frequency range of analysis was split in three parts and the determination coefficient was evaluated for each one. The analysis shows that in ranges A and B the  $R^2$  has a value of 0.99, proving the superposition of the two graphs. In range C the coefficient value is 0.97, confirming the differences that can be observed in Figure 4. In conclusion, the outer casing behaves as a hard wall, only few discrepancies from the ideal behavior can be identified at high frequencies.

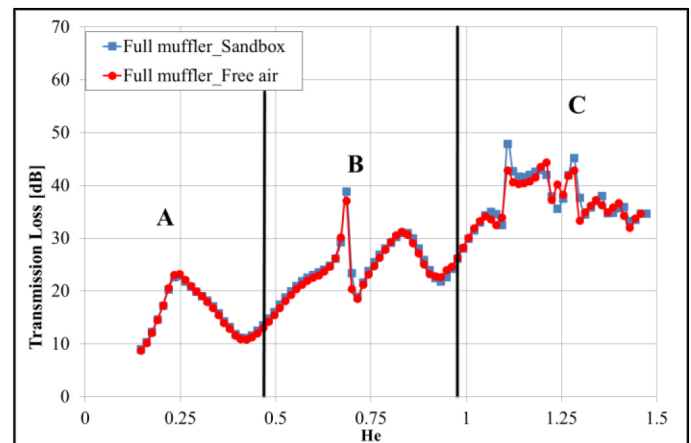


Figure 4. Comparison of the full muffler in free air and in sandbox.

Further steps of experimental campaigns have considered the removal of the outer casing and the glass wool layer, limiting the analysis to the only fluid-dynamic domain. The inner casing is made of a thin steel sheet that could be prone to fluid structure interaction. In order to evaluate these possible interactions the muffler was tested both in free air and in a sandbox. The comparison of the Transmissions Loss was depicted in Figure 5 together with the TL trend of the complete muffler, taken as reference.