

Fluid Dynamic and Acoustic Optimization Methodology of a Motorbike Intake Airbox Using Multilevel Numerical CFD Models and Experimental Validation Tests

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ABSTRACT

In this work a multilevel CFD analysis have been applied for the design of an intake air-box with improved characteristics of noise reduction and fluid dynamic response. The approaches developed and applied for the optimization process range from the 1D to fully 3D CFD simulation, exploring hybrid approaches based on the integration of a 1D model with quasi-3D and 3D tools. In particular, the quasi-3D strategy is exploited to investigate several configurations, tailoring the best trade-off between noise abatement at frequencies below 1000 Hz and optimization of engine performances.

Once the best configuration has been defined, the 1D-3D approach has been adopted to confirm the prediction carried out by means of the simplified approach, studying also the impact of the new configuration on the engine performances. The calculations have been further validated by an extensive experimental campaign carried out not only on the test bench for engine performance analysis but also on an acoustic test bench developed by the University of Florence. The experimental acoustic characterization is based on the Multi-Microphone Technique, which guarantees a wider reliable frequency range of analysis and lower errors in the evaluation of the acoustic properties of the examined devices (such as mufflers or intake lines, etc.).

Results of the calculations on both the baseline and optimized configurations have been compared to the measured acoustic performance carried with null flow conditions showing an encouraging agreement.

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INTRODUCTION

The control of both the level and quality of internal combustion engine noise is a key aspect of the design process of a new vehicle powertrain system, in order to satisfy the legislation limits and provide a characteristic exterior sound. In particular, the gas dynamic noise is radiated from the engine intake and exhaust duct systems due to the highly unsteady flows in the pipes, originated by the periodic cylinder gas exchange process. The attenuation of gas

dynamic noise relies on suitable reactive and dissipative silencers with simple and complex geometries, designed and optimized to damp or emphasize certain spectral components of the engine noise. Therefore, 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.

The role of simulation codes for the prediction of the acoustic/fluid dynamic behavior of the gas systems is fundamental during the first phase of the investigation to find

out the most promising configurations, which will be finally adopted and successively optimized by further measurements and calculations. In general, 1D and 2D-3D simulation models, with different level of complexity, can be applied to evaluate the attenuation features of simple and complex acoustic filters for both the intake and the exhaust side, in the time and the frequency domain [1, 2, 3]. On one hand, 1D linear/non-linear codes are nowadays widely used to calculate the acoustic performances of mufflers [4, 5]. Certainly, 1D linear acoustic codes are mainly applied for this purpose [6, 7] due to their simplicity, accuracy and very short computational times. However, as the flow speed becomes not negligible and the sound pressure level of the signal reaches values of 160 dB, as it can happen inside an intake or exhaust system of internal combustion engines, the magnitude of the incident pressure perturbation makes the linear approximation not valid anymore [8]. Therefore, in the internal combustion engine field, time-domain simulation tools are preferably applied to better reproduce complex pulsating flows associated with mean velocity, since they directly account for these features in the fundamental equations they are based on [9]. From an industrial point of view, the ability to accurately predict both engine performance and noise level has become a key factor during the prototyping stage.

One dimensional simulation tools are widely used during this phase since they allow the simulation of several engine configurations within a short time. Certain components, however, such as the intake and exhaust manifolds, exhibit a high degree of geometric complexity, which cannot be accurately modelled by 1D codes, unless equivalent 1D models are adopted. The need of achieving good accuracy, along with acceptable computational runtime, has given the spur to the development of simplified geometry based tools, such as quasi-3D and 1D-3D approaches [10, 11, 12, 13, 14].

Quasi-3D models are designed to model the acoustics and the fluid dynamics of both intake and exhaust system components avoiding to resolve the small details of the geometry. Models of components are built using a network, or grid, of quasi-3D cells based primarily on the geometry of the system. The solution procedure is usually explicit and is based on a pseudo staggered grid approach, where the equations of mass and energy are solved at cell centers and the momentum equation at cell connections or boundaries. These models can be used as stand-alone tools or also fully integrated into a 1D code in order to study the behavior of intake and exhaust devices under real engine pulsating flow conditions [15]. Conversely, integrated 1D-3D approaches are adopted for a finer optimization of those details which lay in the sub-grid scale of quasi-3D models [3].

This work shows how the adoption of a multilevel CFD strategy based on the OpenFOAM™ technology, which offers the manufacturer an in-house platform for a cost effective support of the engine design and development phase, can be integrated in an industrial process aiming at the optimization of the acoustic performance of a motorbike air-

box. The underlying idea is to exploit simplified approaches (1D and quasi-3D) for DOE practice and to confine the usage of fully 3D simulations to perform the final validation where possible optimization of details is required. Despite the use of the above mentioned numerical approaches has proved to be predictive in case of simplified configurations [3, 14, 16], in the present work the numerical simulations are further validated by experimental measurements performed on real configurations by means of a dedicated acoustic test rig [17]. This comparison made possible to point out the quality of agreement that can be achieved by numerical approaches with real configurations.

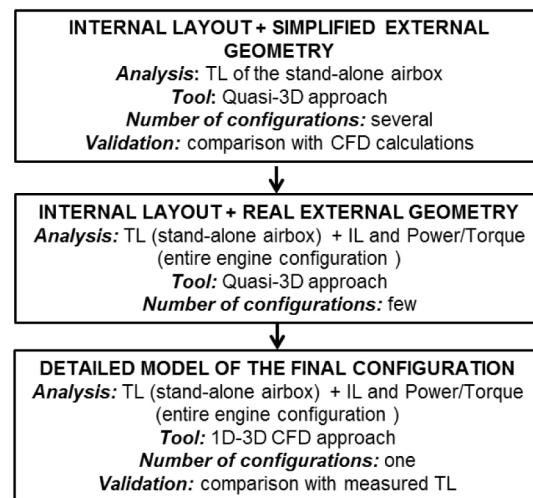


Figure 1. Multi-level analysis for the optimization of the air-box configuration.

The paper begins with the description of the quasi-3D approach developed for the optimization methodology and the experimental test procedure adopted for the acoustic characterization of two air-box configurations, namely the baseline and the optimized. The multilevel cascade process (Figure 1) is illustrated starting from the analysis of simplified geometries, having characteristic dimensions similar to the real geometry. This phase gave the opportunity to achieve the definition of the optimization guidelines and to highlight the agreement between the solution that is provided by models based onto different level of discretization: the fully 3D and the quasi-3D approaches. The agreement reached can be regarded as satisfactory, pointing out that the proposed approach can be adopted not only for the characterization of intake systems, but also for any other component of the whole engine exhibiting a high level of geometric complexity. During the DOE phase, particular care has been taken to verify the consequences of certain solutions on the engine performances, in order to neglect configurations not acceptable in terms of engine efficiency. Different types of intake air-box layouts can be in fact adopted not only to set engine performance by means of internal wave dynamics, but also to control the level and quality of noise radiated from the open ends, in order to

obtain the required response in terms of vehicle performance and sound characteristics.

The numerical models of air-box have been validated by means of experimental data acquired from an acoustic test rig. The main parameters to estimate the acoustic performance of a muffler or air-filter are the Transmission Loss and the Insertion Loss. Such parameters allow ensuring the desired behavior in the frequency range where the engine generates most of the sound pressure.

In order to evaluate the incident and reflected waves on both sides of the air-box the Multi-Microphone Technique [18], which performs the inverse solution of wave equation in hard wall circular duct by controlling the error propagation with ad hoc number of measurement points, has been performed. Since the air-box has an asymmetric geometry, it has a different acoustic behavior depending on the side where the acoustic source is placed. Therefore, the acoustic pressure field was generated alternatively on both sides of the air-box to have a full acoustic characterization resorting to the Two-Source Method algorithm [19, 20].

The measurement sections of the test rig have a diameter suitably defined in order to guarantee a plane wave acoustic field up to 4.5 kHz. However, the measurement section diameter is often different with respect to the inlet and/or the outlet of a specific device. For this reason some connections had to be introduced to properly link the test-sample to the test rig. To take into account the impact of connections between the test rig and the air-box on the acoustic characteristics a data post process analysis was adopted to isolate the acoustic behavior of the connections from the one of the air-box. Such post process procedure is based on the acoustic Transfer Matrix Modelling [21] and allows, for simple geometries, the description of the acoustic behavior by means of the state variables, acoustic pressure and acoustic mass velocity, at the inlet and outlet of the geometry itself. In conclusion it will be demonstrated that it was possible to have a new predictive and robust methodology to quickly test innovative technical solutions in newly developed engines.

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The development, application and experimental validation have been realized within a collaborative framework involving Piaggio S.p.A., Politecnico di Milano and University of Florence.

THEORY

The unsteadiness of the phenomena occurring inside of the exhaust and intake systems has an important role in the fluid-dynamic of internal combustion engines. This means that equations are solved with respect both time and space. Moreover, since the viscosity of the gas, both in the intake and exhaust systems, is very low, it can be neglected without introducing an excessive approximation [22]. This leads to the Euler formulation of the conservation equations:

$$\begin{aligned}\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho U) &= 0, \\ \frac{\partial (\rho U)}{\partial t} + \nabla \cdot (\rho U \otimes U) &= -\nabla p, \\ \frac{\partial (\rho E)}{\partial t} + \nabla \cdot (\rho E U) + \nabla \cdot (p U) &= 0.\end{aligned}\tag{1}$$

This set of equation is then closed by introducing the link between the density and the level of pressure and temperature, as stated by the perfect gas equation of state. These general equations can be solved resorting to different level of approximations. In the fully 3D approach the system is solved as it is, with the addition of terms that may account for viscosity and turbulence. On the other hand, the highest level of approximation is constituted by the 1D formulation, where the momentum equation and the convective terms are referred only to a paramount direction, namely the pipe axis.

An intermediate level of approximation is represented by the quasi-3D approach where, keeping the assumption of inviscid flow, the momentum equation is solved with respect to a paramount direction (like what happens in a 1D model) arbitrarily oriented in space. This makes the model simpler than a fully 3D approach and at the same time removes the main limitation of 1D models in capturing transversal wave motion. Furthermore under the hypothesis of stationary ideal (no viscous) fluid the Euler equation can be solved by the Transfer Matrix Method in the frequency domain. This approach is widely used in the study of acoustic filters, like muffler or air-box, and will be employed in the experimental section for the post-process of the data acquired at the acoustic test rig.

The Quasi-3D Model

The quasi-3D approach applied in this work, named 3Dcell, has been extensively described in previous papers by the authors [12, 23]. In brief, it consists of reproducing a 3D geometry as a network of 0D elements, such as volumes, to which characteristic length in the space are assigned (Figure 2).

The 3Dcell framework is defined by means of two fundamental elements: the cell, which basically defines the

volume, and the port, which stores information about the distance and the flow area between the two neighbour cells. The 3Dcell is a non-linear method, based on the Euler formulation of the Navier-Stokes equations, in which the gas viscosity is neglected. The system of equations is closed by the perfect gas equation of state. The arrangement of the variables on the computational grid refers to a pseudo-staggered scheme, in which intensive properties such as pressure, temperature, density, internal energy and other derived quantities are defined over the cell element, while velocity and momentum are defined on the ports [23]. The continuity and energy conservation equations are consequently solved with respect to the cell element, taking into account the net flux of the conserved variables through the connectors linked to the cell. On the other hand, the momentum equation is solved considering a control volume defined over the connector element. In this framework, despite the velocity in the cell can be arbitrarily oriented in the space (Figure 2), when the solution of the connector momentum is addressed only the projection of the velocities along the connector normal direction is considered, reducing the complexity of the problem to a 1D case [23].

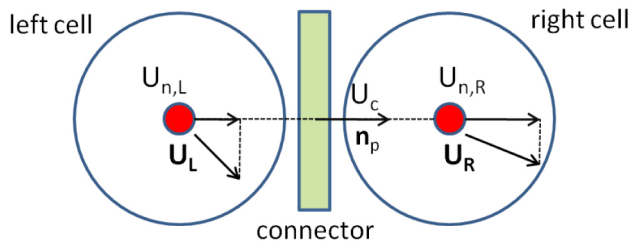


Figure 2. Schematic of the 3D cell element: cells and connectors.

The solution procedure has been optimized in order to achieve the best compromise in terms of the balance between computational burden and accuracy. In particular the numerical method allows the description of the pressure wave propagation with a second order accuracy in both space and time [23]. However, being second order accurate, the solution does not satisfy stability requirements, especially in cases where pressure gradients are high, for instance simulations with mean flow. In order to achieve a stable solution maintaining at the same time a good accuracy in the description of the pressure discontinuities, a suitable stabilization technique has been implemented. This method (DTM) operates on the momentum equation applying a diffusive flux in order to counterbalance the anti-diffusive flux introduced by the truncation error, responsible for the instability of the scheme.

The Coupled 1D-3D CFD Model

The coupled 1D-3D model was developed integrating the 1D code GASDYN [24] with the CFD open-source code OpenFOAM™. The strategy adopted for the coupling of the two codes is described in [3, 16]. It is based on the solution of

multiple local Riemann problems, each one centered on a face constituting the boundary patch which interfaces the 1D domain with the 3D one. This approach allows to locate the 1D-3D interface even in regions close to highly 3D shapes, therefore allowing the treatment of non-uniform fields on the 3D side without compromising the stability of the solution. The CFD code can solve the full set of unsteady Reynolds-Averaged Navier-Stokes equations (URANS), introducing suitable models for the modeling of the turbulence. However, in this work, the Euler set of governing equations was adopted, since at high Reynolds numbers (typical of silencers along the intake and exhaust systems of I.C. engines) the problem is dominated by convection, making viscous phenomena negligible [22]. In order to achieve a good resolution of wave motion inside the muffler, a second order discretization both in space and time was adopted. In particular, linear face cell interpolation was used to determine the face center value of the conserved variables for the calculation of the fluxes while an implicit Crank-Nicholson marching procedure was adopted for the time discretization.

The Transfer Matrix Method

The Transfer Matrix Method is a well-known approach for the study of acoustic filters [21]. The transfer matrixes, evaluated in the frequency domain, are characterized by two state variables: the acoustic pressure $p(f)$ and the acoustic mass velocity $v(f)$, expressed as:

$$\begin{aligned} p(f) &= A + B \\ v(f) &= \frac{A - B}{Y} \end{aligned} \quad (2)$$

where f is the frequency, A is the incident wave component, B is the reflected wave component, and $Y=c/S$ is the characteristic acoustic impedance of the reference duct evaluated as the ratio of sound speed c and duct area section S . By considering an acoustic element, e.g., a duct, the acoustic pressure and mass velocity on the two ends of the duct could be related with a transfer matrix or four poles parameters:

$$\begin{bmatrix} p \\ v \end{bmatrix}_r = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \begin{bmatrix} p \\ v \end{bmatrix}_{r-1} \quad (3)$$

where r is the inlet of the duct and $r-1$ is the outlet of the duct.

The four elements of the matrix come from the Fourier Transform of the solution of the wave equation for the examined geometry, in particular in the literature the transfer matrix of several simple elements such as straight duct, section discontinuities and conical ducts are well known [21, 27].

Experimental Analysis

The Transmission Loss has been chosen as an acoustic parameter suitable for the estimation of the acoustic response

of the investigated air-box and for the comparison of the results provided by the numerical simulations and experimental tests, since it does not depend on the environment surrounding the device [21, 27, 28].

The TL evaluates the attenuation of a generic filter independently by the source and the environmental boundaries condition and is defined by the ratio between the acoustic power level of the incident and transmitted waves:

$$TL = 10 * \log_{10} \left(\frac{P_I}{P_T} \right) \quad (4)$$

The test rig (Figure 3) has been conceived with two measurement sections upstream and downstream to the filter made up of hard wall circular duct of 40 mm of diameter. Four pressure speakers (Monacor® KU616T) generate an acoustic plane wave in a frequency range from 0.2 kHz to 4.5 kHz, they are driven by a wave form generator (Agilent® 33210A) which sets the frequency step and the amplitude of the acoustic field by a feedback control on the microphones. Downstream to the second measurement section an acoustic termination sets acoustic boundary condition. The experimental measurements evaluate the acoustic pressure fields at the inlet and outlet of the air-box; the acoustic pressure waves are divided into incident and reflected components [17]. The measurement technique adopted to evaluate such waves on the two sides of the air-box is the multi-microphone technique [18]. Such technique allows to reduce the numerical error due to the post-processing of the data (*Singularity Factor*) in the frequency range of interest.

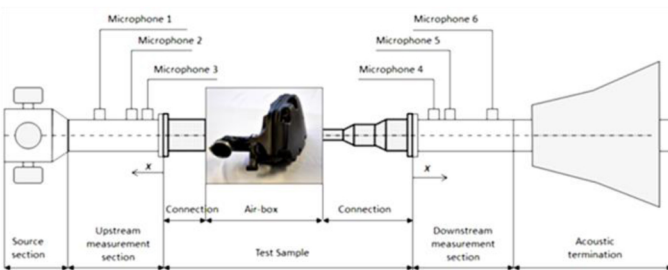


Figure 3. Scheme of the acoustic test rig.

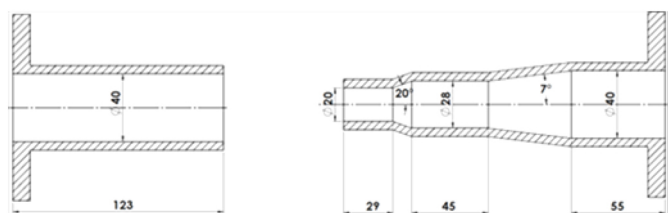


Figure 4. Scheme of the connections between air-box and measurement sections.

The measurements are performed by three pressure microphones (G.R.A.S. 1/4") flush mounted in each of the

two measurement sections (Figure 3). The over mentioned technique is combined with the Two-Source Method which is used to evaluate the acoustic properties of non-symmetrical filters [19, 20]. Since the inlet and outlet diameters of the air-box are respectively 20 mm and 40 mm, they were connected to the test rig measurement sections by *ad hoc* connections composed by the combination of conical and straight hard wall ducts (Figure 4). The shape of the connections has been chosen in order to have a low acoustic pressure drop.

The experimental procedure allows the evaluation of the TL of the whole test configuration, which includes the air-box and the conical connections at the test rig. The contribution of the connections is taken into account with the transfer matrix approach obtaining the TL of the stand-alone air-box. The same test configuration has been simulated by means of the CFD approach, in order to validate the results of the numerical simulations. Tests are performed at environmental temperature condition without mean flow. The acoustic wave field frequency sweep from 0.2 kHz to 2 kHz with a frequency step of 20 Hz. The acoustic pressure level for the tests has been set to 120 dB on the basis of a linearity test (a good trade-off between linearity and accuracy of the measurement) [16].

NUMERICAL OPTIMIZATION

The adoption of CFD approaches for the optimization of specific devices has become a common practice in the latest decades. Many studies have been conducted, resorting to different level of approximations, demonstrating that the main patterns of unsteady flows can be correctly captured by the models available in the literature. In particular, the quasi-3D and the fully 3D approaches, coupled to 1D models by means of a suitable integration strategy, have been comprehensively validated throughout a several test cases of simplified geometries. The reader interested in reviewing the state of the art may be interested in [3, 14, 16, 24].

The optimization activity is performed on a Piaggio® motorcycle engine, whose specifications are summarized in Table 1. The final scope of the optimization is the reduction of the noise emission at the intake side, in particular at low frequency (20-500 Hz), by means of a modified design of the internal layout of the air-box, keeping constant the external shape due to design constraints and preserving the engine performances. The geometry of the air-box is shown in Figure 5. A curved duct brings fresh air from the ambient to the dirty chamber (on the right) of the air-box. Here, air passes through the filter cartridge (removed in the picture) entering the clean chamber, where the inlet section of the intake runner is located (on the bottom side).