

A Comparison Between External and Internal Resonators Employment to Reduce the Gas-Dynamic Noise of a SI Engine

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ABSTRACT

This paper reports 1D and 3D CFD analyses aiming to improve the gas-dynamic noise emission of a downsized turbocharged VVA engine through the re-design of the intake air-box device, consisting in the introduction of external or internal resonators.

Nowadays, modern spark-ignition (SI) engines show more and more complex architectures that, while improving the brake specific fuel consumption (BSFC), may be responsible for the increased noise radiation at the engine intake mouth. In particular VVA systems allow for the actuation of advanced valve strategies that provide a reduction in the BSFC at part load operations thanks to the intake line de-throttling. In these conditions, due to a less effective attenuation of the pressure waves that travel along the intake system, VVA engines produce higher gas-dynamic noise levels. The worsening of the engine gas-dynamic performance can be compensated with a partial re-design of the air-box device, without significantly penalizing the engine power output. In order to find new design configurations of the air-box device capable of improving the noise levels, different numerical models can be successfully employed.

In the present work, a detailed 1D engine model is firstly developed and validated against the experimental data at full load operations. 1D model is realized within GT-Power™ software and it utilizes proper user routines for the modeling of the turbulence and combustion process and for the handling of different intake valve strategies. The 1D engine model also includes a refined user model of the turbocharger able to better describe the acoustic behavior of the device.

The engine model allows for the prediction of the main overall engine performances and the gas-dynamic noise with good accuracy. It also provides a first estimation of the gas-dynamic noise and gives reliable boundary conditions for the subsequent unsteady 3D CFD analyses, allowing to obtain a more accurate noise prediction.

A proper Helmholtz resonator is designed and virtually installed along the inlet pipe of the air-box device. An additional geometrical configuration of the air-filter box, that includes an internal resonator, obtained through the insertion of inner walls, is considered, too. The effectiveness of the redesigned air-box configurations, are firstly tested in terms of Transmission Loss characteristics, and in terms of gasdynamic noise abatement, as well.

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INTRODUCTION

In the recent years, car manufacturers have heavily invested in the development of engine sub-systems able to improve the engine performance in terms of delivered power/torque, brake specific fuel consumption (BSFC), noxious emissions and

radiated noise. Regarding to the SI engines, the downsized architecture, the turbocharging technique and the employment of flexible valve actuation systems are currently the most promising technologies capable to achieve the above listed issues. Modern valve actuation systems are able to independently define the valve opening, closing and the lift

(VVA - Variable Valve Actuation). The flexibility of a VVA system allows to adjust the engine load with a very limited throttling of the intake ducts, obtaining a significant reduction in the pumping work at low/medium loads. Moreover, a reduction in nitric oxides emissions and an improved combustion stability at idle were experimentally verified [1,2]. The downsized philosophy consists in reducing the engine displacement in order to obtain lower mechanical and pumping losses. In this way, improvements in the BSFC and in the CO₂ emissions are expected. The required power/torque performance for the downsized engine is restored by employing a turbocharging system [3]. In addition, a downsized engine allows for a better vehicle drivability thanks to a higher torque at low engine speed [4].

The engine analyzed in this paper presents the previously described technologies. In fact, it is a small turbocharged twin-cylinders engine equipped with a VVA system, which is able to control the intake valve lift profile [6]. The valve actuation system is provided by an electro-hydraulic device that modulates the valve lift according to the operating conditions. However, VVA engines, while improving the fuel consumption, may involve the worsening of the gas-dynamic noise emission at low loads, due to the low throttling of the intake ducts. The gas-dynamic noise emitted by the intake mouth can be reduced with a proper engine calibration [6]. In this work, new different design configurations for the air-filter box are proposed and numerically analyzed with the aim to improve the gas-dynamic noise emission. The main characteristics of the considered engine are listed in Table 1, while a simple schematization of the engine is depicted in Figure 1.

Table 1. Engine Main Features

Model	2 cylinders, 8 valves, VVA, Turbocharged and Intercooled
Displacement	875 cm ³
Stroke / Bore	86 mm / 80.5 mm
Connecting rod length	136.85 mm
Compression Ratio	9.9
Max Brake Power	64.6 kW @ 5500 rpm
Max Brake Torque	146.1 Nm @ 2500 rpm

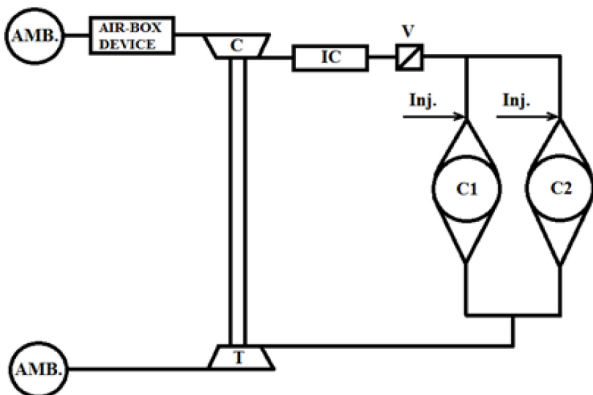


Figure 1. 1D schematization of the engine

First, an experimental campaign is performed on the tested engine in order to measure the overall performance data and the instantaneous pressure cycle in each cylinder. Proper experimental activities are carried out to measure the gas-dynamic noise at the intake mouth of the air-filter box, during an engine run-up test at full load with a variation of the engine speed from 1100 to 6000 rpm. According to standard industrial practices, measurements are carried out by using a microphone placed at 1 cm of distance from the intake mouth.

A 1D engine model is developed within the commercial GT-Power™ software [7], including the schematization of the whole engine. It also presents a virtual microphone to compute the overall sound pressure level (SPL) at the intake mouth. This model is validated both in terms of the gas-dynamic noise, overall output performances (BSFC, torque/power, etc.) and pressure cycles. 3D CFD analyses are also carried out to overcome the limitations related to the 1D model, with particular reference to the schematization of the air-box device. 3D simulations are performed with the FLUENT™ package [8], taking into account the sole air-filter domain. 1D model furnishes the boundary conditions for the 3D CFD unsteady analysis.

The previous numerical and experimental analyses highlight that the maximum radiated noise is reached in a defined frequency range (350-450 Hz). The air-box component is hence re-designed by virtually installing an external resonator and an internal one with the aim to reduce the sound pressure level (SPL) in the frequency range previously identified.

The transmission loss (TL) profiles of the base and redesigned air-box configurations are firstly computed, by using a 3D FEM model, developed within the FEM module of ANSYS™ package [8].

3D CFD analyses of the re-designed intake systems are performed at different engine operating conditions. A comparison between the two proposed solution is finally carried out.

1D/3D MODELS DESCRIPTION

1D Model

As stated above, a 1D schematization of the whole engine is realized within the GT-Power environment. As known, this software is based on a one-dimensional description of the flow inside the intake and the exhaust pipes, while the gas in the single cylinder is treated as a zero-dimensional thermodynamic system.

Turbulence phenomenon and combustion process are simulated using proper sub-models introduced in the code through 'user routines'. In particular, the 'fractal combustion model' [9,10] is utilized to reproduce the combustion process. It represents a phenomenological model sensing the combustion system geometry (head and piston shape, spark

plug position, etc.) and the operating variables such as the spark advance and the air-to-fuel ratio. The turbulence is described by a 0D sub-model which belongs to $K-k$ model category [11]. 1D model also includes a user model of the turbocharging system which furnishes an accurate description of the system performance.

An additional 'user routine' is used to evaluate the actual intake lift profile. It is able to simulate different valve lift strategies such as the LIVO (Late Intake Valve Opening), the EIVC (Early Intake Valve Closure) and the multi-lift strategies. The 1D model of the tested engine also includes a 'virtual microphone', located at 1 cm of distance from the intake mouth. This object has the function to compute the pressure field at the experimental microphone location by treating the intake orifice as a simple pulsating monopole and assuming free field conditions [7,12].

3D Model

The air-box device of an internal combustion engine is usually a complex 3D component, whose shape is often imposed by the space available within the engine bay. The 1D model of the air-box device, formed by a network of 0D and 1D elements, involves relevant geometrical approximations which also affect the prediction of the gas-dynamic noise. The described limitations of the 1D schematization can be overcome by means of a 3D CFD model that is able to simulate in detail the unsteady flow field inside the air-filter box [13] and to perform a more refined acoustic analysis [14]. The CAD model used for the 3D CFD analyses is shown in Figure 2. It is obtained by removing from the constructive CAD drawing, the geometrical details not relevant for the 3D analyses.

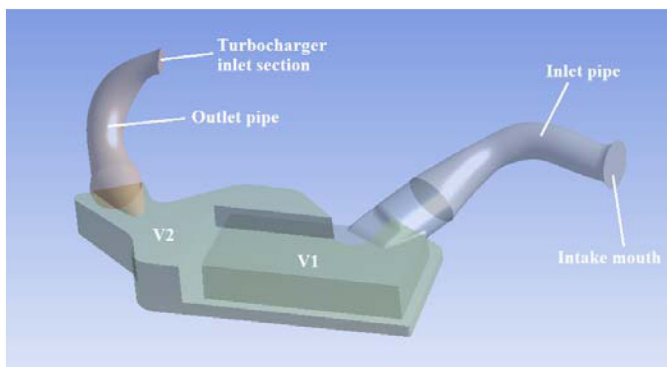


Figure 2. CAD drawing of the original air-box device

As shown in Figure 2, the air box device includes two main volumes, the former (V1) is substantially a parallelepiped volume connected to the upstream pipe and containing a thin filtering element; the latter (V2) is connected to the outlet pipe and it presents a more complex shape. An unstructured mesh is realized for the original device within ANSYS environment, consisting of 65906 nodes and 201992 elements. A preliminary grid invariance analysis is also performed and the corresponding results are not reported here for sake of brevity.

The computational domain includes, in addition to the component geometry, also a spherical shaped volume, centered on the axis of the inlet pipe. The spherical domain allows to compute the pressure field at the virtual microphone, located in the same position as the actual microphone during the experimental analyses. Concerning the boundary conditions, the ambient temperature and pressure are applied all over the spherical surface (pressure-inlet BC), while a time-varying static pressure, derived by the 1D engine model, is imposed in the downstream section (pressure-outlet BC). Referring to the initial conditions, zero velocity, ambient pressure and temperature are specified all over the 3D domain.

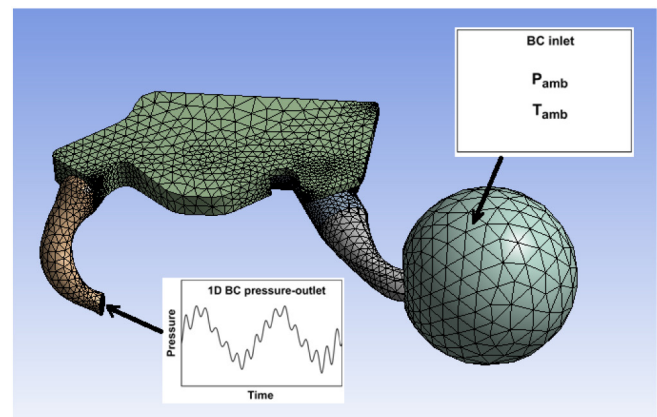


Figure 3. Unstructured mesh of the original air-box device

Concerning the modeling of the flow field within the air-box device, Reynolds averaged Navier-Stokes equations (RANS) are solved, considering the standard $k-\epsilon$ turbulence model and adopting a second order upwind discretization in space and time. Pressure and velocity coupling is based on the SIMPLE algorithm [8]. Ten consecutive engine cycles are computed to ensure an accurate periodic solution of the pressure signal at the virtual microphone. Further details of the 3D model setup are reported in [16]. The post-processing of the time-varying pressure signal at the microphone allows to compute, by means of a discrete Fourier transform (DFT), the root mean square amplitudes of the pressure signal and the SPL in the frequency domain.

TURBOCHARGER MODELING

As known, in the 1D codes the compressor and turbine behavior is usually described following a quasi-steady map-based approach. The latter is proved to be able to properly describe the engine performance both in terms of overall parameters and in-cylinder pressure cycle. On the other hand, the standard methodology does not take into account the dynamic effects inside the turbocharger (wave propagation and the mass and energy storage) and, for this reason, some inaccuracies commonly arise in the modeling of the passive acoustic properties of the compressor and of the turbine. Ultimately, the gas-dynamic engine emission may be estimated by a 1D simulation with a unsatisfactory accuracy.

To overcome this issue, several methodologies have been proposed [17,18]. Most of them includes a detailed 1D schematization of the device [19,20]. However, these approaches, while improving the wave propagation and the noise passive attenuation inside the device, require the definition of several geometrical data, usually unavailable. In addition, the computational effort is significantly increased because of the fine spatial discretization required for the small pipes that schematize the compressor and/or the turbine.

In this work, an enhanced map-based approach is proposed for the description of the turbocharger: it employs the maps measured under steady state operations with the introduction of two 0D capacities placed upstream and downstream the compressor and turbine object. The latter are included to take into account the mass and energy storage inside the device. The volumes of the 0D capacities are evaluated by the geometrical module of the turbocharger model developed by the authors in the recent years [21,22]. In the case of the compressor, the upstream volume coincides with the volume of the inlet duct, while the downstream capacity includes the volume of the wheel, of the diffuser and, mostly, of the volute.

The proposed methodology is implemented in the GT-Power environment as a further user routine. The map is inquired based on the pressure ratio between the upstream and downstream capacities and the rotational speed, while the output parameters are the mass flow rate, the exchanged mechanical work and the enthalpy variation across the device.

FULL LOAD VALIDATION

1D Model

The experimental values of the intake valve closure (IVC), the air-to-fuel ratio (A/F) and the spark-advance are specified as input for the 1D model at full load and for each engine speed. Because of the absence of information about the experimental waste-gate (WG) valve opening, this parameter is evaluated by using a PID controller with the aim of reproducing the experimental pressure level at the firing TDC. Turbulence and combustion sub-models, included in the 1D engine model, are properly tuned by adopting a 0D/3D hierarchical approach [10,11].

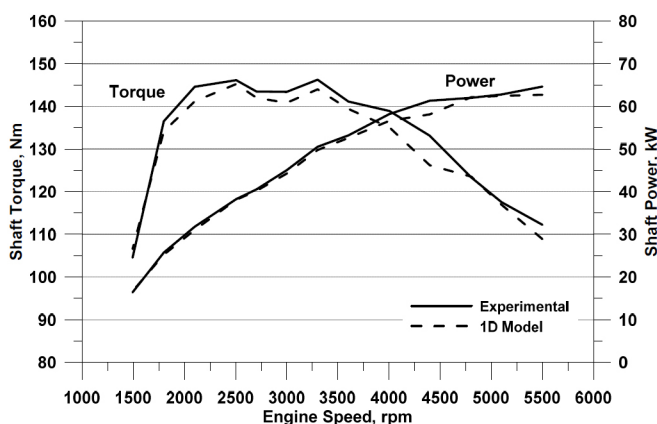


Figure 4. Numerical-Experimental comparison for Shaft Power and Torque

After a tuning of the model constants, they are kept fixed for each analyzed operating condition. 1D results reported in Figure 4, Figure 5, Figure 6 show the numerical-experimental comparison in terms of brake torque, power, BSFC and in-cylinder pressure traces (2100 and 5100 rpm).

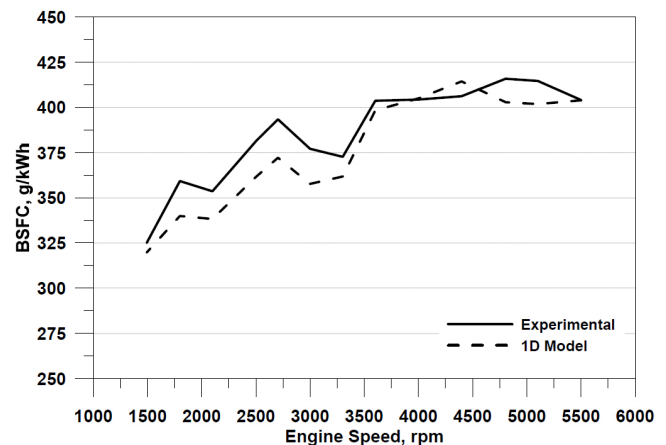


Figure 5. Numerical-Experimental comparison for BSFC

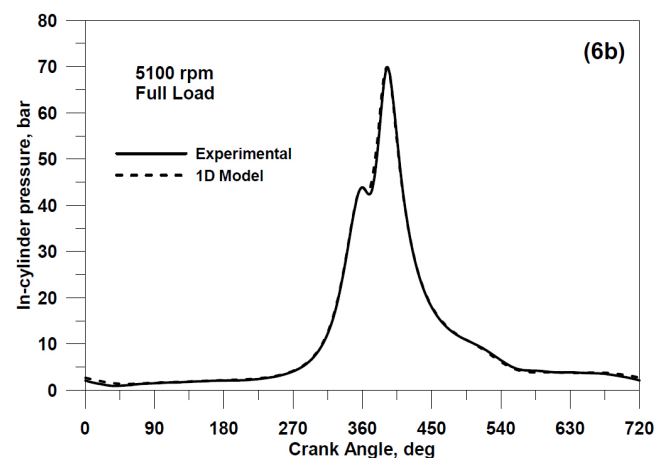
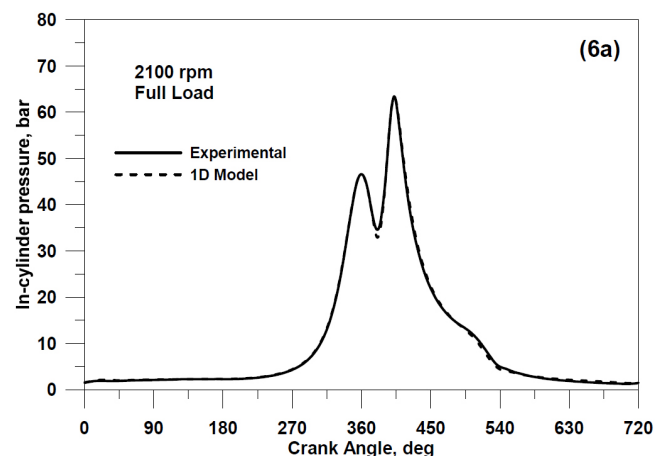


Figure 6. In-cylinder pressure cycles at 2100 rpm (a), 5100 rpm (b), full load

A satisfactory accuracy of the model is obtained for all the above listed quantities. In particular, the pressure cycles prove that a good engine-turbocharger matching is achieved and an accurate description of the combustion process is performed thanks to the "user" sub-models included in the code.