

Experimental and Numerical Comparison of the Acoustic Performance of the Air Filter Box of a SI-ICE

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ABSTRACT

In an Internal Combustion Engine, the design of the intake system is a very critical aspect since it affects both the engine power output and noise emissions at the intake side. In particular, downsized VVA engines typically produce higher gas-dynamic noise levels since, due to the intake line de-throttling at part-load, a less effective attenuation of the pressure waves is realized. In this work, the acoustic performance of the intake air filter of a commercial VVA engine is numerically and experimentally analyzed. In particular, a FEM model of the system is realized in order to compute the Transmission Loss (TL) parameter of the base device. The numerical analysis accounts of fluid-structure interaction, which gives the possibility to determine the effect of structure participation on the TL profile. Contemporarily, the experimental tests are performed on an acoustic test bench based on the multi-microphone technique for the evaluation of the acoustic parameters. Numerical outcomes are compared with experimental findings, showing a very good correlation.

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INTRODUCTION

Nowadays, the internal combustion engine technology is continuously evolving due to global marketing and standards requests. In particular, the basic idea is that the average engine should be more cost effective, which means low fuel consumption has to be reached, but with the power output unchanged. This idea has been made practical with the downsizing strategy. Generally speaking, a downsized engine is characterized by a lower piston displacement, a greater compression ratio and a turbocharger group with respect to a traditional one. In this way, a better Brake Specific Fuel Consumption- BSFCs and almost the same power output, are reached [1].

The power output of an internal combustion engine is also directly affected by the design of both intake and exhaust systems, insofar they affect the volumetric efficiency of the engine. Furthermore such systems affect the noise emission of the engine because they represent the transmission line of the combustion noise to the external ambient. While the exhaust system is not more a primary sound source, once a good muffler has been projected, the acoustic design of the intake system of modern downsized spark ignition engine still represents a very critical aspect. In fact, in this case, the gas-dynamic

noise due to the pressure waves created at the IVO crank angle positions is only attenuated by the throttle. Moreover, in modern engine equipped with VVA distribution system, which allows the achievement of greater engine efficiency, the noise emission from the intake opening is greater because of the de-throttling strategies at part-loads, [2]. So, in order to respect the noise emission standards, the project of a breathing system, in terms of adopted geometry and materials, represents a very important requirement, as well.

As it happens for mufflers, the air box of an intake system could be seen as a silencer, although its primary goal is to maximize the air filling of the cylinders. To the aim of characterizing the acoustic performance of a silencer, the three parameter which could be used are named insertion loss, noise reduction (also known as level difference) and Transmission Loss. However, for a practical point of view, the most used parameters are the noise reduction and the Transmission Loss since for the insertion loss it is necessary to know the source impedance, [3]. At the same time, the noise reduction represents just a simple measure which does not take into account any boundary conditions for the outlet of the silencer.

hand, the Transmission Loss is more adopted for characterizing the system acoustic behavior since its calculation assumes the presence of an anechoic termination at the radiation end.

Such an acoustic parameter may be experimentally evaluated by means of a specific instrumentation and, once a good numerical model has been validated, it is possible to optimize the system geometry.

At the current state of art and from a numerical point of view, in order to predict the acoustic behavior of an acoustic filter the classical 1D simulation tools are widely used because of the very good results within the low frequency range in which only plane waves can propagate. Nevertheless, a pure 1D approach presents some additional limitations due to a rather rough schematization of systems that often presents a complex geometry. Therefore, for a more detailed analysis, which involves the middle/high frequency range, a 3D simulation code is the only tool truly useful for designing and optimizing more complex systems. In fact, 3D codes are commonly used to predict the overall system performance, including the gas-dynamic noise emission, through a suitable post-processing of the results [4]. In this paper the acoustic performance of an intake system for a commercial VVA spark ignition engine has been evaluated, both experimentally and numerically, in terms of its Transmission Loss. First, the finite element method (FEM) approach has been used to model the system with and without structural participation. This model has been validated by means of experimental findings coming from a dedicated test rig. The acoustic test rig has been described in detail by the authors in their previous works [5] [6]. Using this facility, it is possible to measure the TL of an acoustic filter and the other acoustic parameters of the geometry under test, e.g. the reflection and transmission coefficients between each couple gate of the filter. In order to evaluate such parameters it is necessary to measure the incident and reflected waves on each port of the intake system. For this reason the Multi Microphone Technique [7] is used to perform the inverse solution of wave equation in hard wall circular duct. The error propagation on the algebraic solution is controlled using an ad hoc number of measurement points placed at specific relative distances between themselves. Usually the acoustic filters like mufflers or intake systems have an asymmetric geometry. This leads to a different acoustic behavior which depends on which port of the acoustic filter is excited by the acoustic source. Therefore, to have a full acoustic characterization of the air intake the Two-Source Method algorithm is used [8] [11] which requires to applied the acoustic pressure field alternatively on both sides of the air intake. The acoustic test rig is made by two measurement sections, which have a fixed diameter in order to guarantee the plane wave propagation until to 4.5kHz. Unfortunately, the diameter of measurement sections is often different with respect to the port diameters of the geometry under test. So, it is necessary to build up customize connections between the facilities and the air intake ports. Adding the connections between acoustic filter and facilities modifies the acoustic domain of geometry under test, if the connections have a strong diameter reductions can have a strong effects on TL measurement. As previously shown by Montenegro et al. [10] the impact of connections on final results can be removed using a post-process procedure based on the Transfer Matrix Method [11]. By means of this procedure it is possible, for simple geometry like link ducts, to express their acoustic behaviour considering the acoustic

state variables at the inlet and outlet of the link itself. In conclusion it is possible to isolate the acoustic properties of the air intake system, as TL, whatever are the duct connections to the test rig.

Transfer Matrix Theory

The Transfer Matrix Method is probably the most known and simple numerical modeling technique used in acoustic [3]. Usually it is used to study the pressure oscillations along pipe systems [13], where the geometry and frequency range of interest limit the field of study to the plane wave theory.

Considering the frequency domain, the transfer matrixes express the connection between the acoustic state variables, acoustic pressure $p(f)$ and acoustic mass velocity $v(f)$, at the downstream and upstream section of the geometry considered.

The state variables are expressed in the well known form, Eq. 1:

$$\begin{aligned} p(f) &= A + B \\ v(f) &= \frac{A - B}{Y} \end{aligned} \quad (1)$$

where f is the frequency, A is the incident wave component, B is the reflected wave component, and $Y = a_0/S$ is the characteristic acoustic impedance of the reference duct evaluated as the ratio of sound speed a_0 and duct area section S .

For a generic acoustic element, the matrix is made up of four elements Eq. 2 which consider the geometry characteristics and the eventually damping effects due to the thermodynamic conditions of the gas.

$$\begin{bmatrix} p \\ v \end{bmatrix}_r = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \begin{bmatrix} p \\ v \end{bmatrix}_{r-1} \quad (2)$$

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

The transfer matrixes of very simple elements like straight or conical ducts have been validated by many works in literature as shown by Munjal [3], Kulik [13] and Abom [9].

Experimental Analysis

The experimental tests have been carried out on an acoustic test rig which is designed with two measurement sections, upstream and downstream the test case, as it is depicted in Figure 1. Each section is made up of a straight duct, 0.700m of length and 0.04m of diameter. Along the duct, four microphones are placed at specific relative distance between each other.

At the inlet of the first measurement section, an acoustic source made of four speakers (Monacor ® SPH-135C) generates an acoustic pressure field that keeps plane in a frequency working range from 40Hz to 4.5kHz. The lower limit is defined by the adopted speakers while the upper limit is imposed by the cut-off frequency of the test tridimensional mode. It is necessary to maintain the plane wave mode

because the acoustic measures are made at the duct wall. The frequency is changed with a step of 20Hz and the intensity of wave incident on the test case is set to 120dB using a wave form generator (Agilent ® 33210A) and an amplicator (Monacor ® PA180). A feedback control is done using the microphone signals. Downstream to the second measurement section, an acoustic termination, designed following the ISO 5136:2003 [14] guidelines, sets acoustic boundary condition.

Using the multi microphones technique [7], it is possible to evaluate the incident and reflect wave on each port of the intake system. This technique reduces the numerical error due to the post-processing of the data (*Singularity Factor*) in the frequency range of interest. The wave decomposition algorithm uses the pressure signals measured by the four microphones (GRAS ¼") flush mounted on the measurement section.

The measurement technique is completed using the Two-Source Method [8] [11]. This method allows to evaluate the acoustic properties of non symmetrical test case. For sake of brevity, a deep description of the test rig and of its softwares is omitted. The authors have shown an exhaustive and complete explanation of the facilities in their previous works [1] [6].

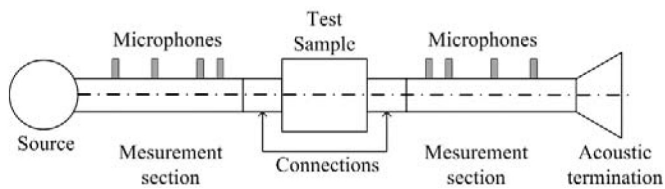


Figure 1. Scheme of the acoustic test rig.

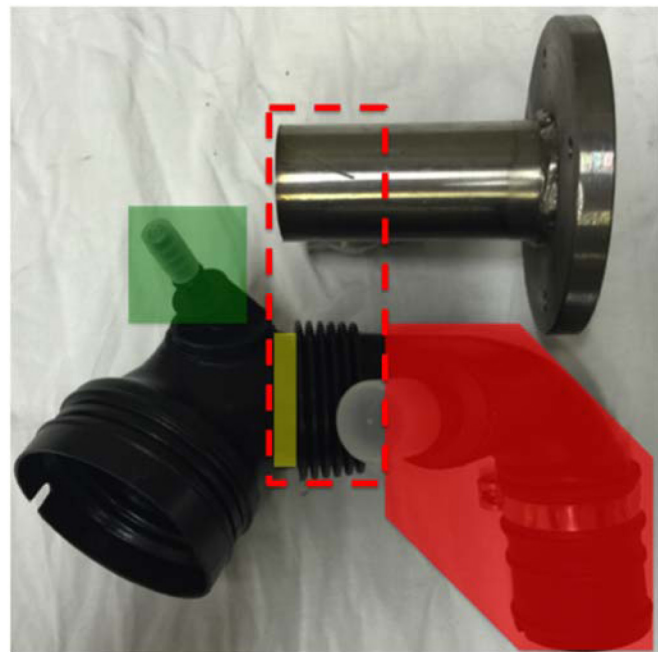


Figure 2. Inlet section of airbox, engine side.

However, it is necessary to highlight the complexity of the air box assembling procedure on the test rig. The airbox has two gates which have different geometric shapes: the gate on the engine side is a straight duct, Figure 2, while the gate that looks towards the ambient

is made up of a conical duct, see Figure 3. Then, considering that the port diameters of the filter are different with respect to that of measurement sections of the test rig, it has been necessary to make two short duct connections which are visible in Figure 2 and Figure 3. The two additional ducts are made of steel and set in place of the segment of the original ducts, which are highlighted in red within the above mentioned figures.

The experimental test allows the evaluation of the *TL* of the whole system under investigation. The contribution of the connections is removed with the transfer matrix approach obtaining the *TL* of the stand-alone intake system. More in detail, the acoustic properties of the airbox can be evaluated between the inlet section depicted in yellow in Figure 2 and the outlet section depicted in yellow in Figure 3, keeping out the contribution of the two steel ducts.



Figure 3. Outlet section of airbox.



Figure 4. The airbox is covered with sand.

In the present work the tests are performed at ambient temperature condition without mean flow. The authors have wanted to focus on the possible interaction between the acoustic excitation and the walls of the air box and how this phenomena affects its acoustic

performance. In order to observe the contribution of the flexible walls, the intake system has been tested in a sand box to eliminate the acoustic wave transmission across the walls (Figure 4).

Experimental Results

In this section the experimental results are shown. The tests have been carried on two different configurations: in free air and in a sand box, both without filter cartridge and mean flow. The acronyms of this two tests are No-Filter_Free-Air and No-Filter_Sand respectively.

As explained in the previously section the additional connections have been indispensable to install the air-box to the test bench. Therefore, the experimental analysis has required the removal of the junctions contribution on the TL results. Such contribution is highlighted in Figure 5, for the rigid wall configuration, where the black and red lines refer to the with and without junctions conditions respectively.

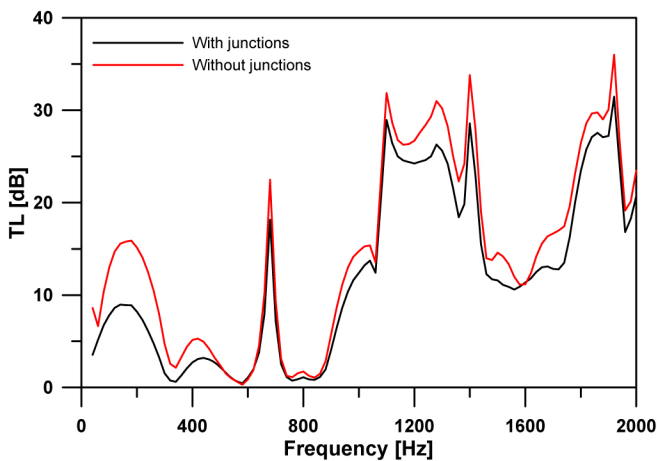


Figure 5. Effect of the junctions on the TL of the airbox.

The TL comparison, with and without the contribution of the junctions, shows that the two connecting elements behave like two horns that amplify the acoustic waves reducing the TL of the airbox itself.

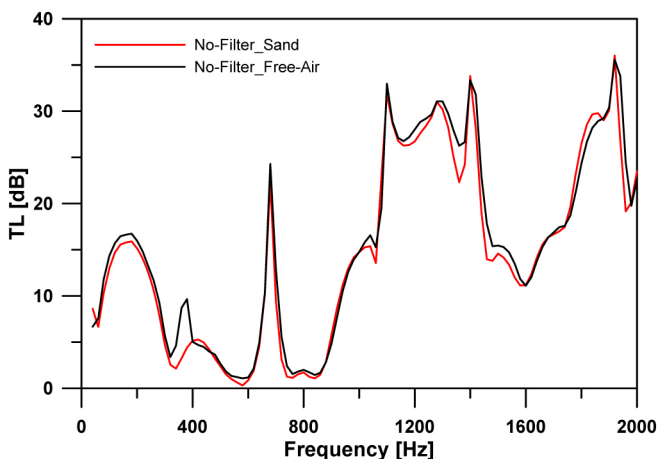


Figure 6. TL trends comparison, intake without filter tested in free air and in sand box.

Moreover, in Figure 6, the effect on the TL profile of the structural participation are depicted. Here, the black line refers to the rigid wall configuration while the red one refers to the fluid-structure interaction.

The comparison of the two tests highlights the effect of the structure on the acoustic response on the air-box. More in particular, the fluid-structure interaction generates a peak of TL at 380 Hz, while on the other frequencies it does not have substantial effect. Such behaviour is supported by the fact that the air box has many ribs that stiffen the whole structure.

CAD Model Modification

The system under investigation is an intake filter box of a commercial VVA internal combustion engine, as reported in Figure 7.



Figure 7. Intake system of the twin air internal combustion engine.

The corresponding CAD model is depicted in Figure 8 in which it is possible to recognize both the original inlet and the outlet sections with their original dimensions.

From the above figures, it is evident that the global system is composed by five subsystems: inlet (green), filter box (white), airbox (red), outlet (yellow), cover (black). In order to reproduce the experimental test set up, both inlet and outlet sections, have been modified. Moreover, the original device presents four layers of foam material: green, white and blue reported in Figure 8 b) that, at this early study, are not modelled in Finite Element Analysis as well as the air filter highlighted in Figure 8 c).

The full assembly is quite complex and it is full of small features as highlighted in Figure 9.

Nowadays, the preparation of the CAD model for FE simulation remains a complex task, and represents an important part of the time required in the simulation work. The preparation of a Finite Element Analysis (FEA) model from a Computer Aided Design (CAD) model is, still, a difficult task since its Boundary Representation (B-REP) is often composed of a large number of faces, some of which may be