Transmission Loss Assessment and Optimization of an Intake System for Automotive Application

Siano D. [1], Aiello R. [2], and D'Agostino D. [2]

***Abstract*—** The acoustic project of internal combustion engine's intake systems is a very important task aimed to ensure that a trade-off between noise emissions and perceived sound quality is achieved. In general, the most used parameter for characterizing the performance of an acoustic filter is represented by the so called Transmission Loss which may be numerically evaluated thanks to a three-dimensional FEM code. In this paper, a FEM model of an intake system for a commercial spark ignition engine has been realized in order to simulate its acoustic performance in terms of Transmission Loss, without considering structural participation as well as the presence of mean flow inside the system. After a model validation, thanks to previous experimental analysis, an acoustic optimization procedure of the studied system has been carried out by means of geometry modifications.

***Keywords*—**Transmission Loss assessment, Finite element method, Internal combustion engine, Breathing noise.

INTRODUCTION

NOWADAYS, the modern concept of client perception quality of a vehicle, which represents the prior target behind the production strategies of internal combustion engines, is quite changed with respect to a dozen of years ago. In particular, the concept of engine perceived sound represents a critical aspects that car manufacturers are focusing on, in order to be more competitive. In fact, if the engine noise is something that may be appreciated by customers, on the other hand it strongly reduces the global comfort within the cabin. Such a new project aspect is becoming more and more important as it is testified by the noise emission standards. It may be stated that the acoustic project of a vehicle must satisfy a trade-off between acoustic comfort and perceived sound.

In a vehicle various noise sources are involved which belong to two main categories: structural and airborne noise.

Among these, the most prominent noise source, with particular regard to the low vehicle speed, is represented by the gas-dynamic noise emitted by both the intake and the exhaust systems. Furthermore, such systems directly affect the power output of internal combustion engines because they strongly influence the volumetric efficiency. However, the acoustic design of the intake system of modern internal combustion engine represents a more critical aspect with respect to an exhaust one[7]. In fact, whilst for this latter it is relatively easy to place a so called proper muffler (or dB-killer) at the end of the acoustic transmission path, when dealing with intake systems the gas-dynamic noise due to the pressure waves created at each IVO crank angle positions may only be attenuated by the throttle. That's why modern engines equipped with VVA distribution system sounds stronger.

It is well known that for a system acoustic performance characterization the most used parameters are Insertion Loss, Level Difference and TL (Transmission Loss)[10]. Among these, the Insertion Loss is rarely used since for its calculation the knowledge of the source impedance is required[2]. At the same time, the Level Difference represents just a simple measure of the sound pressure level drop which does not take into account any boundary conditions for the system's outlet. The Transmission Loss is instead the most adopted acoustic performance characterization parameter since it is a measure of the sound power level drop a system ensures, by assuming an anechoic termination at the outlet section[2].

In the presented study, a FEM model of an intake system for a commercial spark ignition engine has been created in order to first evaluate its Transmission Loss. After a model validation procedure by means of experimental data, available from previous studies, an acoustic optimization of the original device has been carried out. All the simulations have been performed by using the commercial 3D code Virtual Performance Solution™, powered by ESI Group.

I. FINITE ELEMENT METHOD IN ACOUSTIC

Today, numerical simulation plays the most important role in studying and solving real life problems. Among the numerical acoustics method, the Finite Element Method (FEM) offers an advantageous combination of modeling flexibility, computation efficiency and results accuracy.

In the classical FEM formulation, the solution of a physical problem relies on the use of a so-called "weak formulation" of

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the governing equation (which represents the so-called "strong form" of the physical phenomenon) through two main steps:

1) The Partial Differential Equation (PDE) is multiplied by a test function and integrated over the whole computational domain
2) An integration by parts makes possible to transfer some of the partial derivate to such test function

In sound propagation problems, the pressure disturbance p' at a generic point *Q(x,y,z)* and at the specified time *t*, is given by the so called wave equation whose homogeneous version is:

$$\left[\frac{1}{a_0^2}\frac{d^2}{dt^2}-\nabla^2\right]p'=0 \quad (1)$$

in which a_0 represents the sound velocity propagating in the undisturbed medium. Equation 1 represents the "strong formulation" of the problem which, if solved, gives the exact solution.

In order to perform such a Finite Element Analysis, the frequency version of the wave equation is generally preferred, which is known as Helmholtz equation:

$$\nabla^2\hat{p}+k^2\hat{p}=0 \quad (2)$$

in which *k* is the spatial frequency or wave number and equals to ω/a_0. By multiplying the Helmholtz equation by a test function u_o and integrating over all the domain, it becomes:

$$\iiint u_0\nabla^2\hat{p}dV+\iiint u_0k^2\hat{p}dV \quad (3)$$

Then, integrating by parts, it is possible to obtain the following weak formulation of the sound propagation PDE , as reported in the following:

$$-\iiint\nabla u_0\nabla\hat{p}dV+\iiint k^2\hat{p}dV+\iint u_0\frac{\partial\hat{p}}{\partial n}dS=0 \quad (4)$$

Then the discretization process of (4) is accomplished through a standard Galerkin formulation[3], which leads to a system of linear equations with the pressure nodal unknowns, whose matrix formulation is represented by (5).

$$\left[K+i\omega C-\omega^2M\right]\{\hat{p}\}=0 \quad (5)$$

In equation 5 K, C and M represents the acoustic stiffness, damping and mass matrices, respectively, whilst $\{\hat{p}\}$ is the unknowns vector.

The above process does not change in presence of acoustic sources and it does not take into account the structural participation, as considered in this early stage.

II. Transmission Loss Analysis

The Transmission Loss represents a measure of the sound power level drop between the upstream and the downstream position of an acoustic filter, when the downstream section is anechoic, as schematized in Fig. 1. Generally, during experiments, both inlet and outlet tubes are such that plane waves propagation is ensured for a wide frequency range.

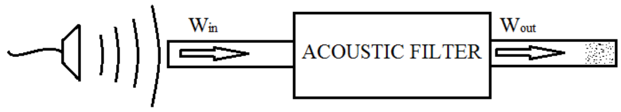

Fig. 1 Transmission Loss definition

In terms of standing wave components of the sound filed, anechoic termination means that there are no back moving waves within the outlet tube, being the reflection coefficient equal to unity. Formalizing, the pressure and particle velocity at the upstream and downstream position may be expressed as:

$$p_u=A_u+B_u \quad (6)$$

$$u_u=\frac{1}{\rho a_0}A_u-B_u \quad (7)$$

$$p_d=A_d \quad (8)$$

$$u_d=\frac{1}{\rho a_0}A_d \quad (9)$$

where *A* and *B* are the amplitudes of the forward and back moving waves, the subscripts *u* and *d* stand for upstream and downstream tubes respectively, whilst *S* and the product ρa_0 stand for the cross section of the upstream and downstream connection tubes and the characteristic impedance of the medium respectively. Mathematically, the Transmission Loss is defined as the ratio of the incident and transmitted power into an anechoic termination, namely

$$TL=10Log\left(\frac{W_{in}}{W_{out}}\right) \quad (10)$$

Such parameter is the most adopted for characterizing the acoustic performance of a system because it only depends upon its geometry and material. By making use of Eq. (6-9), the incident and transmitted acoustic powers may be expressed in the plane wave range as

$$W_{in}=puS=\frac{A_u^2}{\rho a_0}S \quad (11)$$

$$W_{out} = puS = \frac{A_d^2}{\rho a_0} S \tag{12}$$

Thus the expression for the Transmission Loss is

$$TL = 20Log\left(\frac{A_u}{A_d}\right) \tag{13}$$

Its experimental calculation may be performed by calculating A_u and A_d thanks to the Kundt's tube technique[2].

A. Software Implementation

The model developed in Virtual Performance Solution™, in particular within the NVH-Interior Acoustics environment of VPS™[6], follows the definition of TL, i.e. the difference between incident and transmitted power into an anechoic termination. Therefore, the following boundary conditions have been applied to the acoustic mesh:

1) a constant velocity of 1 m/s at the inlet section as well as a 2D absorber acoustic in order to ensure only incident acoustic waves,
2) 2D absorber acoustic at the outlet section in order to simulate the anechoic termination.

The 2D absorber acoustic BC is typically applied in VPS environment in order to simulate a semi-infinite fluid (impedance characteristic given by the product of air density and sound velocity). In Fig. 2, the above described boundary conditions are depicted.

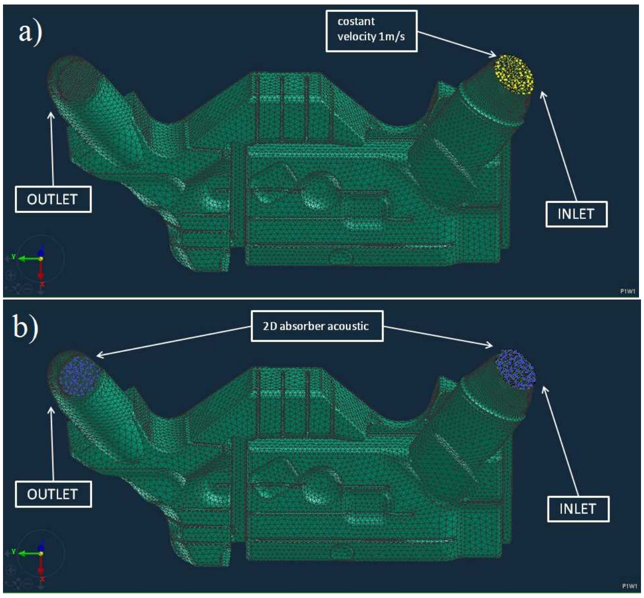

Fig. 2 Boundary conditions applied within the VPS model: a) inlet velocity and b)anechoic termination

In this way, the sound pressure at the outlet termination, may be used to calculate the desired Transmission Loss, as indicated by (14):

$$TL = 20Log\left(\frac{\rho c v_0}{2p}\right) \tag{14}$$

The implementation logic of numerical models in VPS is based on the construction of modules and stages. The module represents the model, namely the acoustic mesh and the boundary conditions, whilst the stage represents the chosen analysis. For the investigated system, one module and two stages, have been created. More precisely, in the first stage acoustic modes of the model are extracted and then used as input to the second stage for frequency response analysis calculation.

Both modules and stages are edited as cards which represent the command lines which will be read from the solver, PAM-NVH®.

III. Intake System Acoustic Analysis

The so-called engine breathing noise is due to both intake and exhaust systems and it is the most prominent vehicle noise source when the engine's load is high and the vehicle itself is still or moving at low speeds. Generally speaking, this kind of noise may be seen as made of two components which are shell noise and tail pipe noise. The first type is due to the structural vibrations exited by the pressure distribution inside the system and may be very disturbing in correspondence of the structural resonant frequencies. On the other hand, the second type of noise (so-called airborne noise), which is radiated by the outlet mouth of such systems, consists of primary noise sources and secondary noise sources. The primary sources are due to the travelling pressure waves caused by the pulsating flow at each IVO crank angle positions, and their spectral components consists of multiple of the firing frequency. The secondary noise sources represent the turbulence and other flow induced aeroacoustics sources, e.g. vortex separation, whose spectral components represent the broadband noise. In Fig. 3, a typical intake noise spectrum (sound pressure level expressed in dB) of a spark ignition engine is depicted as function of the frequency.

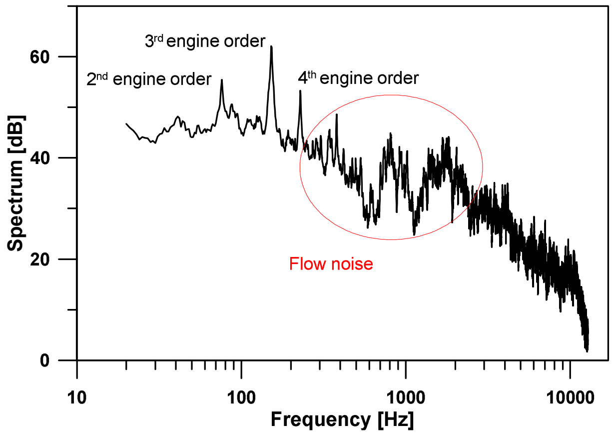

Fig. 3 Tail pipe noise of a four cylinder SI engine at 2280 rpm

As it is possible to appreciate from the above figure, the tonal spectral component (multiple of the firing frequency or engine orders) as well as the broadband noise are clearly visible as main responsible for the tail pipe noise.

If the focus is on sedan car (and not for example on motorbike), there are two main reasons why it is important to control the intake breathing noise. Firstly, as mentioned before, it represents one of the most prominent noise sources at high load and low vehicle speeds[6]. Therefore, reducing intake noise represents a benefit with respect to the overall noise radiated by a vehicle, allowing to respect the emission standards. Besides, a lot of attention must be paid within the low frequency range where the tonal spectral components may excite the acoustic closed space resonant frequencies, reducing the acoustic comfort of passengers. Such coupling effects may become very prominent until 400 Hz, depending on the specific cabin geometry, which affects the resonant frequencies.

From what above, the acoustic project of intake systems is a very complicated task, more than for exhaust systems, because the prior goal must be to maximize the cylinders filling and not to attenuate the sound radiation.

As previously stated, this work deals with the acoustic performance assessment and optimization of an intake system for automotive applications. The considered acoustic performance parameter is the Transmission Loss and the tested geometry is shown in Fig. 4.

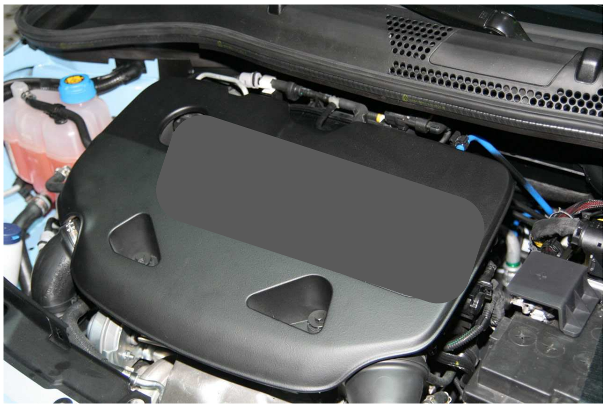

Fig. 4 Intake system under investigation

The global system is composed by five subsystems: inlet (yellow), filter box (green), air-box (red), outlet (white), engine cover (black) as it is highlighted in Fig. 5 a) and b). Within this early study, the presence of the air filter has not be considered as well as the presence of mean flow and the structural participation. This latter hypothesis has been confirmed by previous analysis, at least in the investigated constraint condition[6]. Thus, apart from the above mentioned boundary conditions, the further following conditions have been imposed.

$$\left(\frac{\partial p}{\partial n}\right)_{wall} = 0 \qquad (15)$$

$$\bar{v} = 0 \qquad (16)$$

Thanks to the momentum balance, (15) simply means that the particle velocity at the wall equals zero m/s.

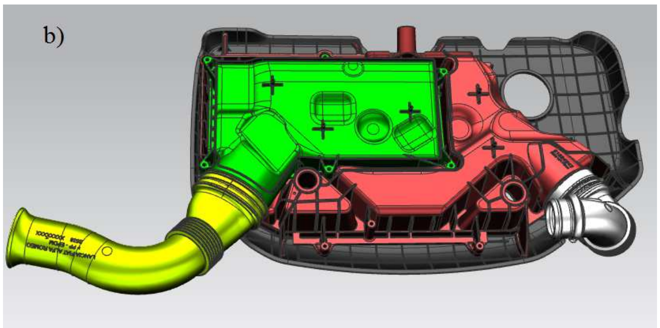

Fig. 5 a) Top and down b) views of the intake system

In Fig. 6 and Fig. 7 the CAD model of the air volume embedded within the system and the corresponding acoustic mesh are shown. It is worth noting that all the acoustic mesh which will be shown allow accurate results until more or less 3000 Hz, choosing six point per wavelength as spatial resolution.

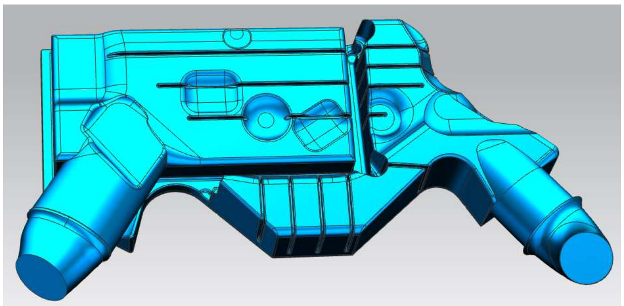

Fig. 6 CAD model of the acoustic domain

Moreover, both the inlet and the outlet ducts have been modified with respect to the original ones in order to reproduce the experimental layout[6].

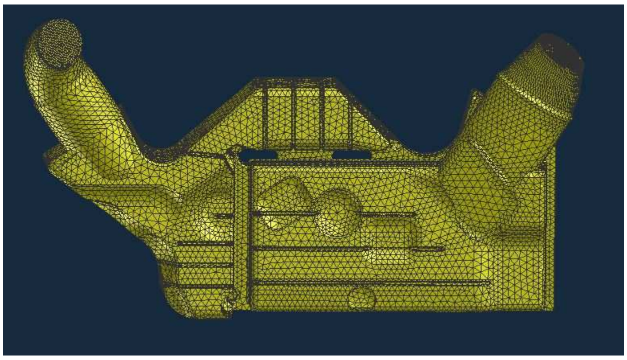

Fig. 7 Air volume's acoustic mesh: 128303 elements (CTETRA 10-noded) and 202931

During all the simulations, the air density and the sound velocity have been set equal to 1,2 kg/m^3 and 340 m/s, respectively, whilst the frequency step has been fixed equal to 5 Hz.

Finally, the experimental/numerical Transmission Loss assessment is depicted in Fig. 8. Here, the TL is expressed in dB and only the region [40; 1000] Hz has been shown because, as it has been highlighted in previous study[7], in this range a perfect agreement has been found between the two analysis.

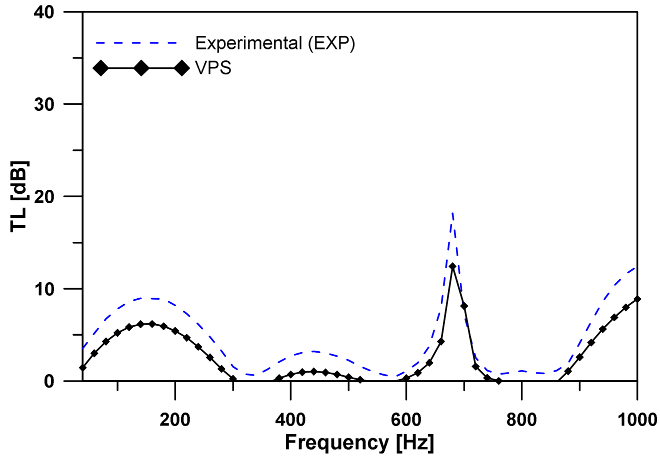

Fig. 8 Transmission loss assessment: model validation

In the graph, the blue dashed line represents the experimental data, whilst the black rhombus line represents VPS outcomes. During the experimental facilities, the system has been inserted within a big box full of sand in order to ensure no structural participation[6].

As it is possible to appreciate by the examination of Fig. 8, the VPS solution globally underestimates the experimental findings but a perfect frequency agreement has been found, meaning that no frequency shift hold[7].

So, the main results coming from the above TL analysis are that the VPS model perfectly reproduces the acoustic performance of the actual system and, secondly, the investigated system does not ensure a good acoustic attenuation within the low frequency range[6]. Such behavior is not acceptable since the exiting frequencies of an internal combustion engine are multiple of the firing frequency. This latter is given by (17):

$$f_c = \frac{n}{60}\frac{z}{\varepsilon} \tag{17}$$

where n is the revolution per minute, z is the number of cylinders and ε represents the number of revolution per thermodynamic cycle. Thus, for example, considering a four cylinders spark ignition engine, the firing frequency at 1000 and 2000 rpm equals 33 and 66 Hz respectively. Besides, in the frequency region where the flow noise sources are prominent, namely from 300 to 600 Hz, no practical attenuation holds. Therefore a geometrical change aimed to improve the noise attenuation up to 600Hz should be performed. For this purpose, it is necessary to associate the various regions of the TL profile to each subcomponents the whole intake system consists of.

Hence, from the theory of the acoustic filters[2], it is obvious that the smooth profile which ranges from 40 up to 600 Hz is due to the main expansion chamber[6]. It is well known that, considering the transmission loss of such element, the amplitude of the peaks depend on the ratio between chamber's cross section and that of both the connecting upstream and downstream ducts[14]. In particular the greater the area ratio, the higher the peaks amplitude. Therefore, in order to increase the TL of the considered system, a CAD modification is required. To the aim of increase the above mentioned area ratio, the inlet duct of the intake system has been modified as depicted in Fig. 9.

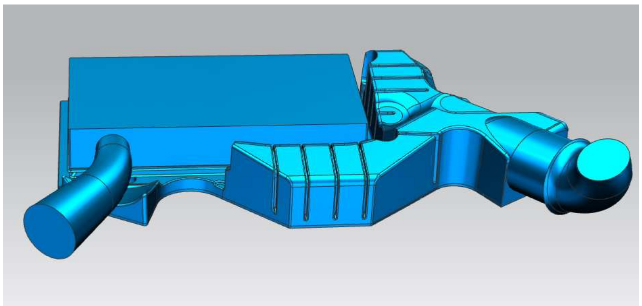

Fig. 9 CAD modification of the inlet duct

As it is possible to appreciate from Fig. 9, the outlet section of the inlet tube has been reduced as much as possible, meaning that a very small cross sectional area would accelerate too much the air flowing into the system with high distributed pressure drops (this last being proportional to the square of the mean flow velocity). The corresponding acoustic mesh is depicted in Fig. 10.

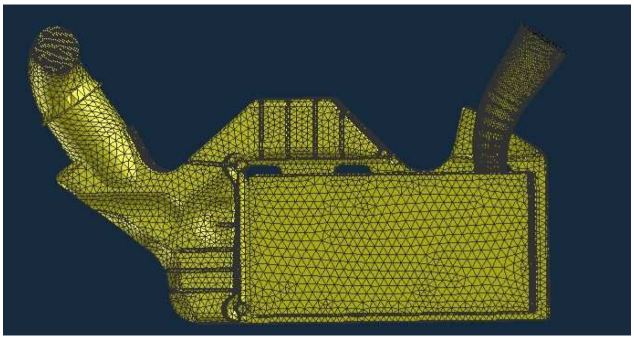

Fig. 10 Air volume's acoustic mesh with modified inlet duct: 101819 elements (CTETRA 10-noded) and 163480 nodes

The effects of the modified inlet duct on the Transmission Loss are shown in Fig. 11. In the picture, the black and blue lines refer to the original and modified geometry respectively.

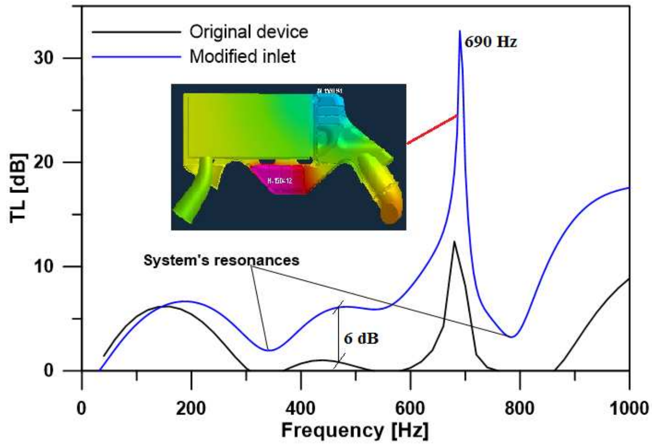

Fig. 11 Transmission loss comparison

Of course, the first peak of the curve is still located at the same frequency, about 690 Hz, because it is due to the resonant cavity, highlighted in the above figure, which has not been modified.

Moreover, the increased TL values around 340 and 800 Hz are explained by changes of the acoustic modes around such frequencies. In fact, considering the original geometry configuration, the acoustic modes at 302 and 866 Hz are such that the expected sound pressure at the outlet is higher than at the inlet[6]. Vice versa, thanks to the inlet modification, the pressure distribution corresponding to the acoustic modes at 307 and 811 Hz are shown in Fig. 12. Here, the hotter the color the higher the sound pressure.

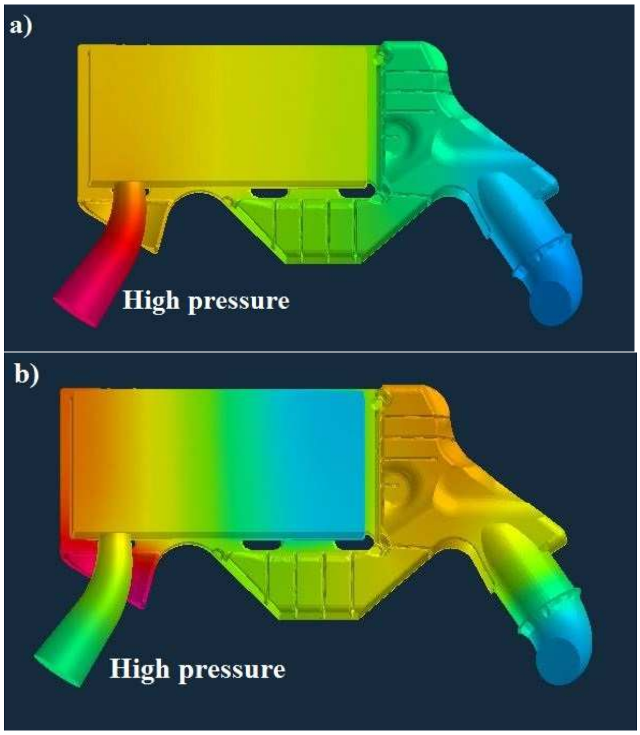

Fig. 12 Cavity modes at: a) 307 Hz and b) 811 Hz

Thus, as expected, increasing the cross sectional area ratio results in an higher amplitude of the TL profile in almost the whole investigated range. More precisely, the main benefits have been achieved within the frequency window [200; 600] Hz, e.g. 6dB at 465 Hz, whilst no sensible improvements have been found before 200 Hz. However, at lower frequencies where a coupling between breathing noise and acoustic cavity modes may appear, i.e. below 400 Hz, the reached TL enhancement could be not satisfactory. For this reason, in order to further increase the TL profile at low frequencies a filter box enlargement has been foreseen. The modified geometry and the corresponding acoustic mesh are depicted in Fig. 13 and Fig. 14 respectively.

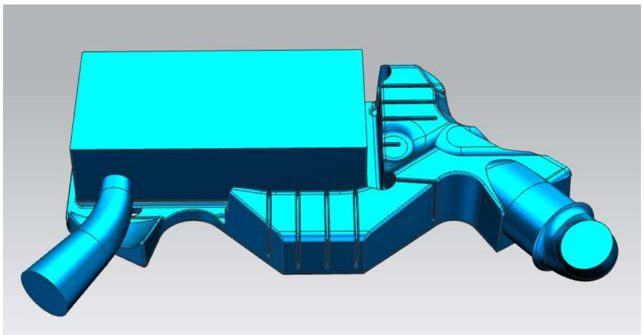

Fig. 13 CAD modification of the filter box

As it is possible to appreciate by the examination of Fig. 13, such increased volume has been achieved through an additional height of the filter box. More precisely, the additional height equals 2 cm.

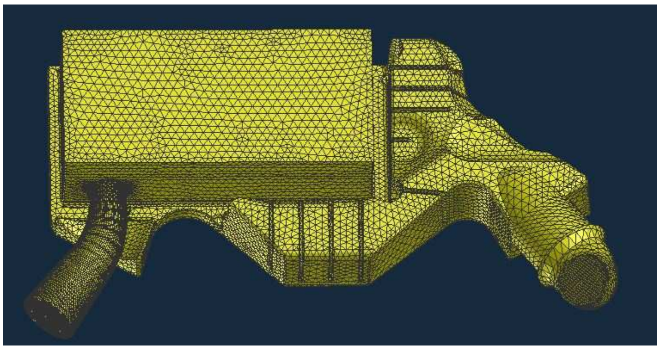

Fig. 14 CAD modification of the filter box: 138480 elements (CTETRA 10-noded) and 218450 nodes

The corresponding Transmission Loss simulation outcomes are shown in Fig. 15 where the red curve refers to the new configuration.

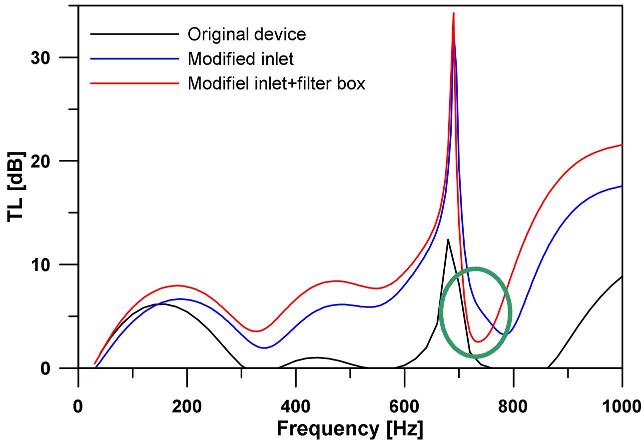

Fig. 15 Transmission loss comparison

As it is highlighted in Fig. 15, the effect of filter box enlargement is analogous to the inlet duct modification, namely the TL profile has been simply shifted in amplitude. Such circumstance is not surprising because this latter modification relies on a cross sectional area increase too. From the TL trends, it comes out that this further geometry change ensure the best acoustic performance for the intake system except around 740 Hz, as putted in evidence by the green circle in the above figure. The low value of sound attenuation at such frequency (about 3 dB), is certain due to the influence that the acoustic mode depicted in Fig. 16 has on the system's response.

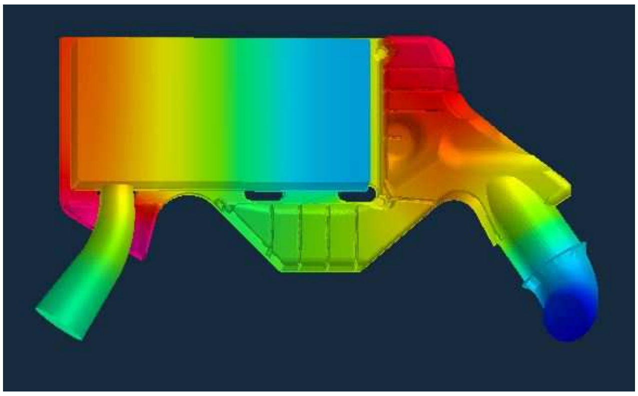

Fig. 16 Acoustic modes at 780 Hz corresponding to the last CAD modification

Nevertheless, this low TL value would not affect the overall acoustic performance since, as previously mentioned, the amplitude of the spectral components corresponding to both the primary and secondary noise sources, which constitute the tail pipe breathing noise, start to rapidly decrease after 600 Hz. In conclusion, within the investigated frequency range, a great improvements have been achieved thanks to the analyzed modified geometry. In fact, the average sound attenuation of the original device equals 2,50 dB whilst it would be equal to 9,70 dB if the two studied CAD modifications would be realized. Obviously, here no attention has been paid on the possibility that the investigated changes are not allowed because of no available space when the intake system is located on the engine head. A more accurate analysis should take into account the degrees of freedom in terms of available space for the geometry modification.

IV. Conclusion

In this work, the acoustic performance of an intake system for automotive applications has been numerically evaluated in terms of its Transmission Loss, in absence mean flow, structural participation and presence of the air filter. For this purpose, a 3D finite element model of the system has been realized in Virtual Performance Solution™ software environment, powered by ESI Group. First of all, the model has been validated by means of a comparison with experimental analysis coming from previous studies. From this evaluation stage, the system has turned out to be not well acoustically projected. Therefore, thanks to the very good agreement between experimental and numerical findings, two geometry modifications have been tested on the numerical model to the aim of performing an acoustic performance enhancement procedure. No limits have been fixed for the degrees of freedom in terms of available space for the geometry changes. The foreseen new configurations ensure, together, a great improvements of the Transmission Loss, being the average gain of about 7 dB within the whole investigated frequency range. Further analysis must be addressed to confirm such results when modeling the real working condition of the intake system, namely with the presence of mean flow inside the system, taking into account the structural participation (in real constraint conditions) and the presence of the air filter.

Acknowledgment

The authors are grateful to ESI Group in the person of G. Pagnozzi for provision of expertise, for software release and technical support in the implementation.

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