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Acoustic optimization of an intake system by means of geometry CAD modifications

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Abstract

The intake system of internal combustion engines represents the most prominent noise source at high load and low vehicle speeds because of the de-throttling strategies which are realized in order to maximize the cylinders filling. Therefore, a good acoustic performance of intake systems represents a very important challenge in order to respect the overall noise emission, according to the more strict European standards. In general, the most used method for characterizing the acoustic performances of a system is represented by the Transmission Loss computation. In this study, a 3D numerical analysis by using the Finite Element Method onan intake system for a commercial spark ignition engine has been exploited for the Transmission Loss calculation. The numerical findings of the finite element model have been validated by means of experimental investigations. A very good agreement between numerical and experimental data has been reached. For this reason, an optimization procedure, by implementing different CAD modifications on the system, has been investigated, as well. All the foreseen geometry changes do not modify the overall size of the original system.

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Keywords: Transmission loss, Gas-dynamic noise, Engine, FEM

1. Introduction

In a vehicle various noise sources are involved such as engine noise, aerodynamic noise, gear noise, rolling noise etc. Among these, the most prominent noise source, with particular regard to the low vehicle

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speed, is represented by the gas-dynamic noise emitted by both the intake and the exhaust systems, because they represents the direct transmission path to the external ambient. However, 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 internal combustion engines still represents a very critical aspect. In fact, in this case, the gas-dynamic noise due to the pressure waves created at each inlet valve opening crank angle positions is only attenuated by the throttle. The main difficulty is that the prior task of an intake system is to maximize the cylinders filling and not to attenuate the sound transmission. Moreover, the global size of such system must satisfies as much as possible some compactness requirements.

Therefore, the acoustic project of breathing system, in terms of being made of several acoustic filters, must take into account the above mentioned restrictions. In this paper the acoustic performance of an intake system for a commercial spark ignition engine is optimized in terms of its Transmission Loss by means of a 3D finite element model. Such model does not take into account the structural participation and has been validated by comparing the results with experimental ones [5]. The very good agreement between numerical and experimental data, has allowed the optimization procedure by implementing different CAD modifications on the system, with the aim of improving its acoustic performances within the investigated frequency range.

2. Numerical investigations

In this work, the acoustic performance of the intake system for a commercial spark ignition engine has been optimized, by means of geometrical modifications, in terms of its Transmission Loss without modeling the structural participation, which simply implies that the particle velocity at wall is assumed to be equal to 0 m/s. The CAD model of the system in its original configuration is depicted in Fig. 1.

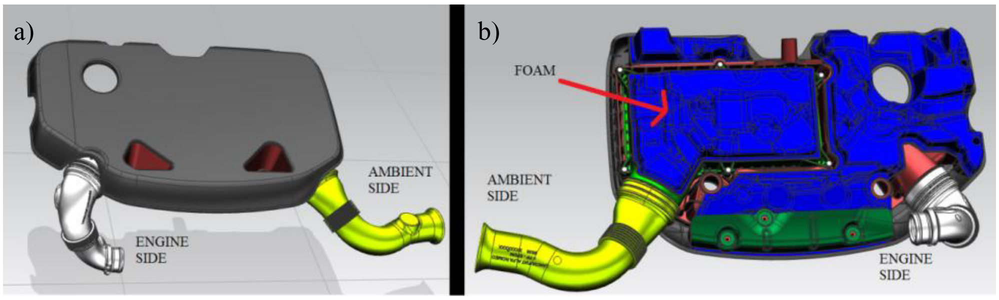

Fig. 1 CAD model of the investigated intake system: a) top view and b) down view

As it is possible to appreciate from Fig. 1 b), a nonstructural material is attached to the intake system itself in his original configuration. Such nonstructural material consists of a foam and it is highlighted in blue within the above figure. This foam acts as a noise control treatment which should reduce the noise emission due to mechanical impacts (valve, injectors, piston slap etc.), thanks to its absorbent properties[6].The corresponding air volume embedded within the intake system is shown in Fig. 2 a), where both the inlet and outlet sections have been modified as during experimental investigation [5].

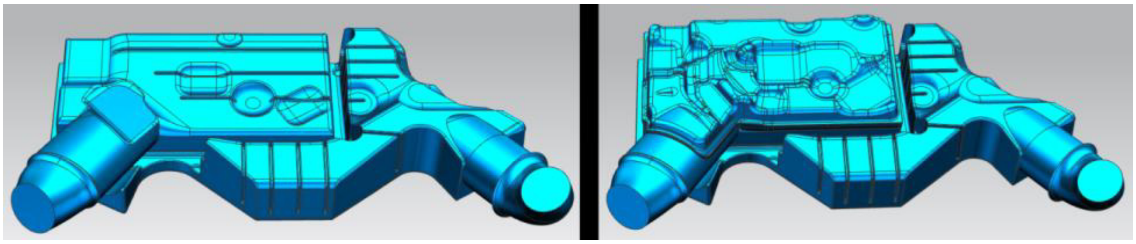

Fig. 2Air volume: a) original configuration, b)first CAD modification (increased area of the main expansion chamber)

Concerning the Transmission Loss of such intake system, it has been deeply discussed in previous work[5]. More precisely, the results of the numerical simulation have shown a fully satisfactory agreement with the experimental findings, demonstrating the potentiality of the numerical 3D finite element model eventually to be used in optimization geometry procedures. In fact, it has been found that, within the frequency range [20; 1000] Hz, the system does not ensure a good acoustic performance (the average acoustic attenuation is equal to 2,45 dB). This is an unacceptable conditions for systems like that, since the most prominent noise sources radiate at multiple of the engine firing frequency. In order to improve its acoustic performances, several geometry modifications have been tested without changing the overall size of the system. The first test has been conducted by using the foam's space useful for enlarging the air volume of the intake system, as clearly shown in Fig. 2 b).In Fig. 3, both the original and the new Transmission Loss trends have been reported as function of frequency, where the black and blue lines refer to the original and modified configuration of the system respectively. Thanks to this first modification, it is expected that the position of the two peaks within the range [20; 600] Hz will not change, but the amplitude of the peaks should be increased.

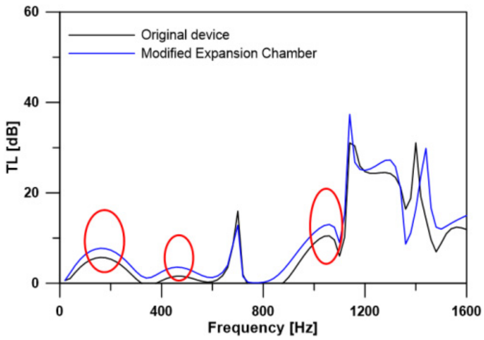

Fig. 3Effect of the first CAD modification on the Transmission Loss

As depicted in Fig. 3, the sound attenuation of the new system is increased (about 2dB) within the considered frequency range and from 800 to 1100 Hz too, as it is highlighted by the red circles. This effect could be easily understood from the analytic expression of the Transmission Loss for a single expansion chamber [7].

$$TL = 10Log\left[1 + \frac{1}{4}\left(\frac{S_c}{S_i} - \frac{S_i}{S_c}\right)^2 \sin^2\left(kl_c\right)\right] \tag{2}$$

In Eq. (2) S_c and S_i are the cross sections of the chamber and that of the inlet/outlet pipes, whilst k is the wave number and l_c in the chamber's length. Even if Eq. (2) represents the analytic expression of the TL in the case of plane wave propagation, it is conceptually clear that increasing the area of the expansion chamber, results in a TL increment as well. However, a TL value below 5 dB within the range [300; 600] Hz does not ensure a good acoustic performance, since within this range the so-called flow noise is expected to be prominent[8][9]. Therefore, one further modification could take into account another part of the foam attached to the system in order to increase the volume of the cavity which is responsible of the peak around 700 Hz. Such cavity acts as an Helmholtz resonator for which the greater the volume the lower the resonant frequency[1]. After this further geometry modification, the air volume embedded within the system should look as depicted in Fig. 4.

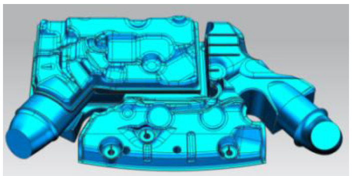

Fig. 4. Second CAD modification: increased volume of the resonator

The corresponding effect on the Transmission Loss profile is depicted in Fig. 5, in which the black and the blue lines refer to the original and modified geometry respectively.

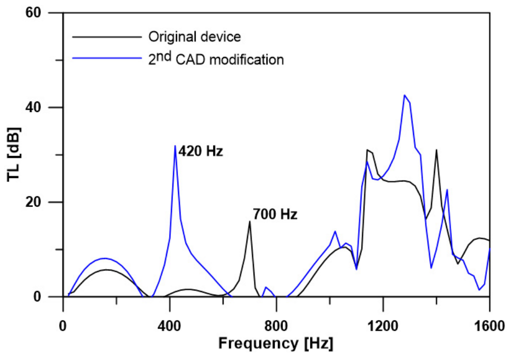

Fig. 5. Effect of the second CAD modification on the Transmission Loss

As expected, the combined effect of the two CAD modifications have improved the TL in the low frequency range, e.g. 32 dB at 420 Hz. More precisely, as highlighted inFig. 5, the peak due to the resonance cavity has moved from 700 to 420 Hz. Such behavior perfectly agrees with the analytic expression of the resonant frequency of an Helmholtz resonator, considering the different volumes highlighted in Fig. 6.In fact, the resonant frequency corresponding to the modification in Fig. 6 b) is given by Eq. (3)

$$f_2 = \frac{a_0}{2\pi}\sqrt{\frac{S}{LV_1}} \cdot \sqrt{\frac{V_1}{V_2}} = f_1 \cdot \sqrt{\frac{V_1}{V_2}} \cong 405Hz \tag{3}$$

where a_0 is the speed of sound, V is the volume of the cavity whilst S,L are the neck's cross section and length respectively.

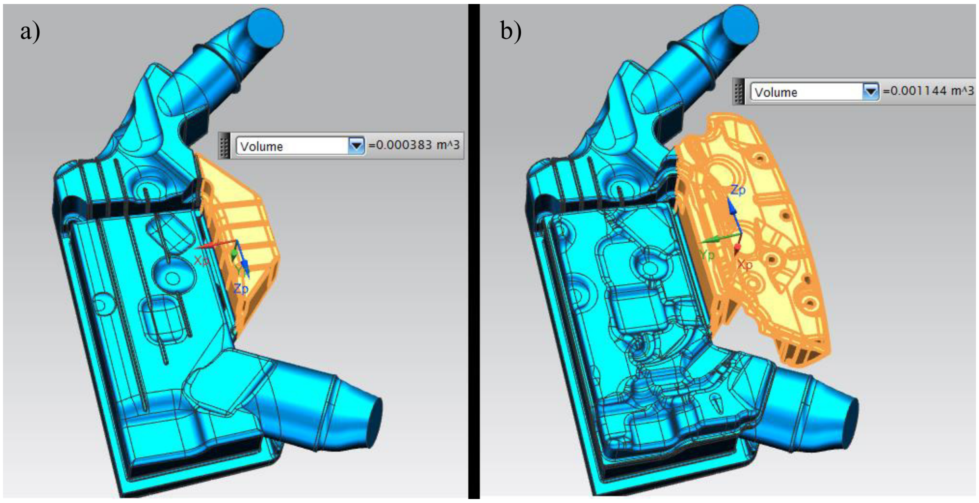

Fig. 6 Comparison between the volumes of the resonator: a) before and b) after the CAD modification

In spite of the fact that the last modification ensure a better acoustic performance at low frequencies, in the frequency range [600; 900] Hz, a very poor attenuation holds so far. Therefore, a further geometrical modification is necessary, which is highlighted in Fig. 7 a). Even in this case, the foam has still been used as the previous modifications to improve the air cavity volume.

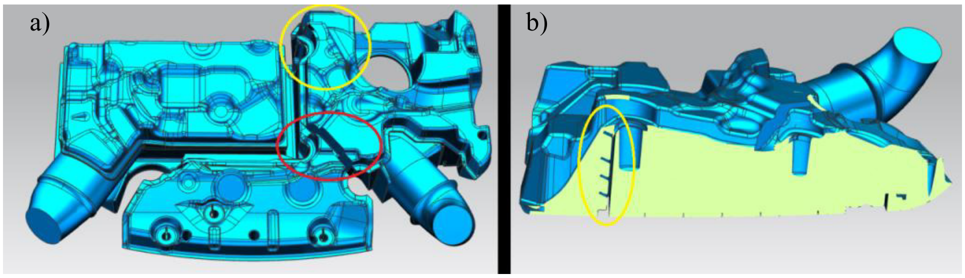

Fig. 7 Third CAD modification: Added volume to the outlet chamber

In this case, the foam has been embedded within the air volume of the system in a way such that it consists of a further volume added to the outlet chamber, without affecting the yellow volume highlighted in Fig. 6(see the red circle in Fig. 7a)). In this way, the main effects of previous changes are preserved. Moreover, the foam has been attached to the air volume such that the cavity highlighted in Fig. 7a) by a yellow circle acts as an additional resonator (see the separation within the yellow circle in Fig. 7b)). The results in term of TL of the above described modification is shown in Fig. 8.

In the above figure, the black, red and blue lines refer to the original device, the second and third modification respectively. As it is possible to point out from the curves, the last geometry change ensure a great increase in the acoustic performance of the intake system within the whole investigated frequency range. In particular, the main gain has been obtained within the range [600; 900] Hz. Here, the peak at 700 Hz is due to the new cavity, as it is possible to appreciate by the pressure distribution in Fig. 9.

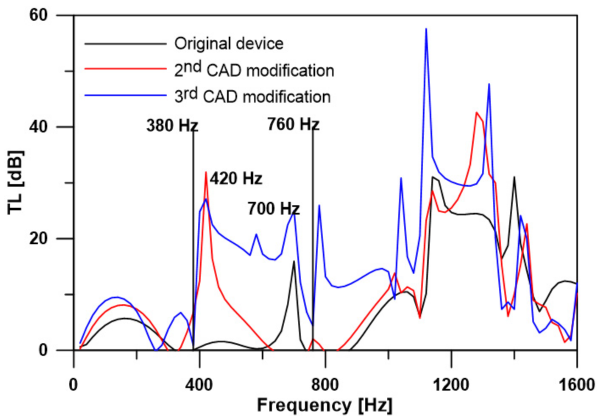

Fig. 8. TL comparison between original and geometry modifications of the system

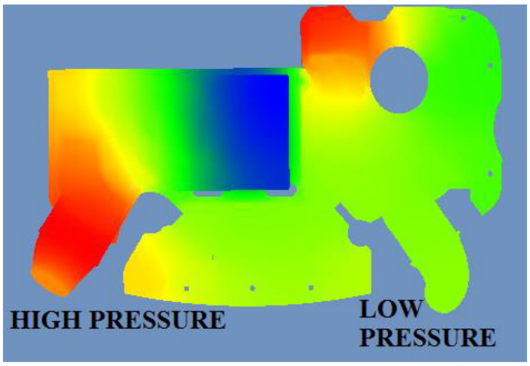

Fig. 9Sound pressure distribution at 700 Hz

Furthermore, thanks to the way in which such last modification has been executed (see Fig. 7a)), the peak at 420 Hz due to the previously modified resonator is not changed. However, there are still some frequencies at which the system does not work very well, e.g. at 380 and 760 Hz as depicted in Fig. 8. Such behavior could be probably explained from the examination of the acoustic modes of the system in this new configuration. In fact, it is well known that, as it happens for the structural modal theory, thanks to the modal expansion theorem[1][7]each pressure distribution inside the system could be represented by a weighted summation of all the possible acoustic modes. For example, this means that the general expression of the sound field in a duct is obtained by the superposition of the modes as expressed by Eq. (4)

$$p'(x,t) = \sum_n \left[\hat{p}_n^+ \psi_n e^{-jk_{1,n}x_1} + \hat{p}_n^- \psi_n e^{+jk_{1,n}x_1} \right] e^{j\omega t} \tag{4}$$

where $\hat{p}_n^{\pm}$ are complex valued amplitude, x_1 is the direction of propagation, $k_{1,n}$is the wave number in the direction of propagation and ψ_n is the nth modes. Thus, for the new geometry configuration of the intake system, mainly two acoustic modes influence the response of the system at the two considered frequencies, as it is depicted in Fig. 10.

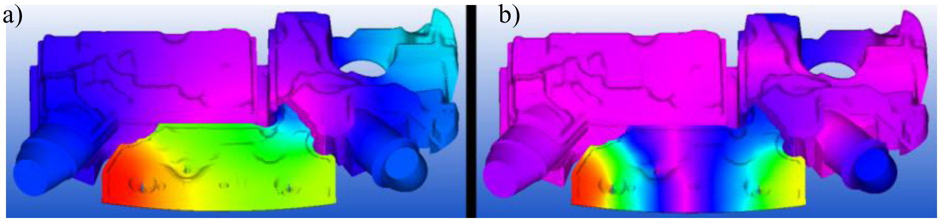

Fig. 10 Acoustic modes at a) 374 Hz and b) 752 Hz

As it is possible to appreciate from the contour plot in Fig. 10, both the sound pressure distributions in correspondence of the two acoustic modes, clearly highlight almost the same magnitude at the inlet and outlet sections. Therefore, the low TL values at both 380 and 760 Hz are justified by the prominent influence of the two above depicted modes.

From the above analysis, it has been demonstrated how the acoustic performance of the investigated intake system may be significantly improved by means of geometry modifications without changing the overall size of the system, which represents a very important restriction. This preliminary study has been performed without taking into account structural participation and presence of mean flow that surely may affect the TL behavior. For this reason, further numerical analyses have to be performed in order to verify that these geometry modifications still improve the acoustic performance of the intake system. Moreover, further experimental validation should be conducted in order to verify that the overall noise level emitted by the engine with and without foam remains unaltered.

Conclusions

In this work, the acoustic performance of an intake system for a commercial internal combustion engine has been tested in several configurations. More precisely, starting from previous study in which a numerical model has been validated by means of experimental investigations, the Transmission Loss in rigid wall condition has been numerically evaluated in order to test the improvements coming from various geometry changes. The main optimization strategies have been focused on the low frequency range [20;1000] Hz in which the original device does not show a good sound attenuation. Thus, it has been demonstrated that opportunely changes in the volume distribution may considerably increase the TL values. The most important aspect is that such CAD modifications do not modify the global size of the system because they just rely on the use of foam material which is attached to the original system in order to attenuate other engine related noise sources. Of course the effects of such modifications on the engine performance must be evaluated as well as the influence of the foam material on the global noise emitted. Moreover, the goodness of the obtained results have to be verified in real working conditions, by considering fluid-structure interaction and the presence of mean flow inside the intake system.

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Biography

D. Siano was born in Naples in 06/01/1969 Italy, and graduated in Aeronautical Engineering at the University of Naples "Federico II", Italy in 1994. Until 2001, she was researcher in acoustic and vibration department at C.I.R.A. (Italian aerospace Research Center). From 2001 until now, she is a Researcher at National Research Council of Italy (CNR) in the field of Acoustic and Vibration in transport field. She is responsible of Acoustic and Vibration Laboratory in her Institution. Expert evaluator within the EU 6th and 7th Framework Research Programme, in Transport-Aeronautics in 2006 and 2007. Project expert evaluator in Ministry Economic Development, Italy.
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