AIR INTAKE MANIFOLD RADIATED NOISE INDIRECT ESTIMATION BY MEANS OF PANELS VELOCITIES EVALUATION

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ABSTRACT

During the internal combustion engine operation, the main noise sources are the crank train unbalance forces, the combustion phenomena, the gears coupling, the valves friction and intake and exhaust systems that radiate high noise due to air excitation of some parts resonant frequencies. A four cylinder engine intake manifold was analyzed in order to understand its contribution to the overall engine noise. An specific characteristic of this intake manifold is the presence of a resonator, which contributes to noise radiation from its surface, once the manifold is made of polymer that has lower stiffness when compared to aluminum. The proposal is to demonstrate the possibility of manifold radiated noise estimation by evaluating the panels velocities on its surfaces. Vibro-acoustic simulations were carried out using finite element method with automatically matched layer approach (AML). The sound pressure levels were determined using virtual microphones and panels velocities were calculated at some probes on the manifold surfaces. The results could be compared to NVH measurements in a semi-anechoic test cell. NVH tests and acoustic simulations demand considerable time and costs. Therefore, calculating and measuring only the panels velocities represent a good approach of the noise estimation.

INTRODUCTION

The automotive industry is currently investing time and money to improve the vehicles NVH performance. The new powertrain design methods are starting to consider NVH issues throughout the whole design process. This involves modeling, simulation, evaluation, and optimization techniques into the design process to insure both noise and vibration comfort.

The internal combustion engine is an important noise source and all the powertrain components must be designed looking for low noise radiation to the external environment and to the interior of the vehicle.
One important noise propagation occurs on the engine intake and exhaust systems because the air flow inside flexible components causes the surfaces vibration and consequently noise radiation.

The object of the present study is the air intake manifold and the main objective is to indirectly estimate the noise produced by this component by determining the surfaces vibration velocities, here called as panels velocities, when applying the engine air pressure profile in the frequency domain.

The noise propagated by the manifold is quantified by evaluating the sound pressure level at a specific distance from it, where the noise information is important to be known. Using vibro-acoustic numerical simulation, it is possible to calculate the panel velocities on the component surfaces and calculate the sound pressure levels at once. Further results could be obtained from experimental NVH tests, measuring both sound pressure levels and panel velocities with actual parts installed in the vehicle.

NOISE SOURCES IN INTERNAL COMBUSTION ENGINES

The main noise sources in an internal combustion engine are the crank train unbalance forces, the combustion phenomena, the gears coupling, the valves friction and intake and exhaust systems that radiate high noise due to air excitation of some parts resonant frequencies. Any component assembled on a four cylinder engine should not have resonant frequencies below 250 Hz [1], once this frequency range corresponds to the engine operation speed range (until around 6000 rpm). High vibration amplitudes may occur in this situation, causing possible damage and noise, thus the components must be designed to have highest natural frequencies as possible.

In the case of intake manifolds made of plastic, this requisite is more difficult to be achieved due to the material low stiffness. Hence, additional vibration simulations and validation tests are required to verify the component strength under its resonant frequencies and guarantee its integrity during all the engine desired life.

The noise from the intake and exhaust systems occurs mainly due to the fluid flow inside the ducts and chambers and also due to the excitation of acoustic modes that the fluid passage promotes in these components. The air intake manifold cavity, for example, have some specific acoustic resonant frequencies. When these acoustic modes are excited by the pulsating air inside it, high amplitude pressure is applied on the internal cavity faces. Therefore, the manifold walls will vibrate according to the structural panel modes and high noise can be radiated to the environment and to the passengers inside the vehicle.

AIR INTAKE MANIFOLD

The air intake manifold is a component of the engine control system. Its main function is to provide air-fuel mixture to the engine cylinders. Moreover, intake manifold have an important function on pollutants emission which physical characteristics like regular pressure drop and balanced air distribution among runners for the cylinders, according to Cavaglieri *et al* [2].

The correct amount of fuel is provided by the fuel rail and the injectors. The required amount of air that enters the manifold is controlled by the throttle body valve. Figure 1 shows an air

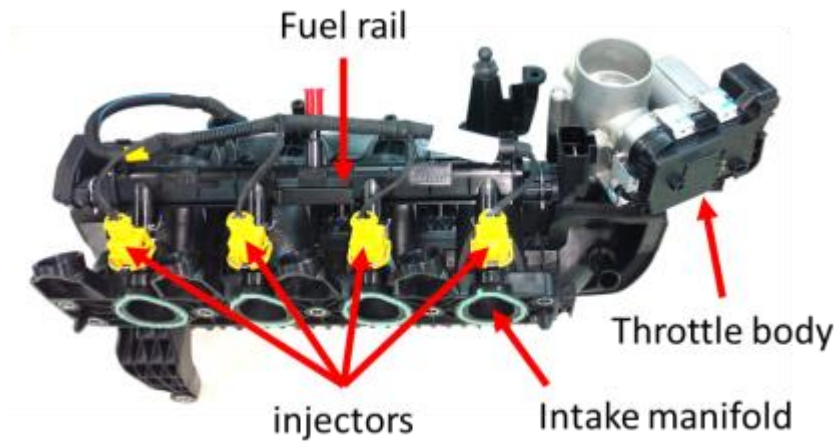intake system of a four cylinder engine, with fuel rail, injectors, intake manifold and throttle body.

Figure 1: Air intake system.

The object of present study is a four cylinder engine air intake manifold. The objective is to compare the radiated noise from two different design options of same application. The first design corresponds to a plastic intake manifold with some of external walls with flat surfaces.

The second design is very similar to the first one, but with implementation of a lot of ribs, aiming the stiffness increasing and trying to reduce the noise radiated by the walls. Basically, only the throttle body position was slightly modified and the ribs were included on the plenum, runners and resonator walls. Plenum is the main intake manifold chamber, the runners are the ducts responsible to distribute the air among the engine cylinders and the resonator is a secondary chamber tuned to improve the manifold volumetric efficiency in a specific frequency. Both air intake manifolds are made of the same material polyamide 6 reinforced with 30% of glass fibers. Figure 2 shows the both intake manifold designs.
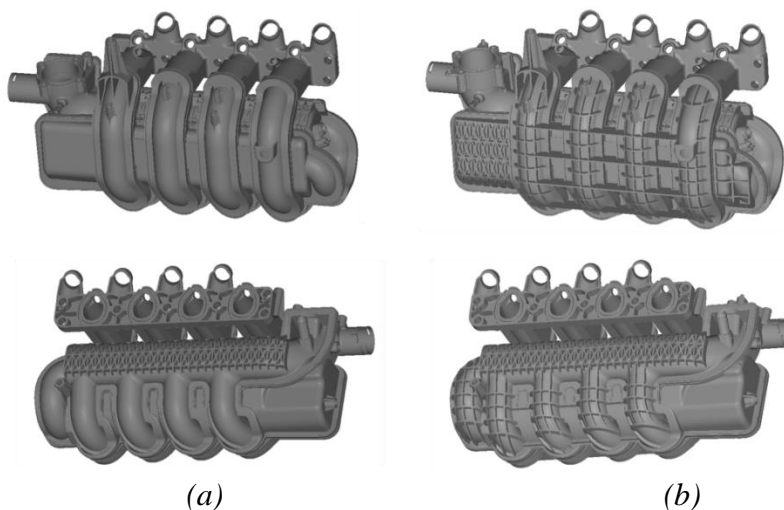

(a)   (b)

Figure 2: Air intake manifold designs. (a) flat walls without ribs. (b) reinforced walls with ribs.

INTAKE MANIFOLD VIBRO-ACOUSTIC SIMULATIONS

The air intake manifold noise radiation can be predicted by conducting vibro-acoustic simulations. Basically, the main idea is to determine the noise level at some specific points placed at a known distance from the manifold.

The vibro-acoustic finite element analysis consists in coupling the structural behavior of the component with the acoustic behavior of the air inside the cavity. Solid meshes have to be generated for both the structural and the fluid models. Moreover, to evaluate the noise radiation to the external environment, a far field mesh must be created to represent the external fluid.

The first calculation step consists of structural modal analysis of the intake manifolds. These simulations pre/post processing and solver were conducted using the software ALTAIR Hyperworks version 14. The finite element meshes used in structural modal analyses are shown in Figure 3. Tetrahedron second order elements were used on both meshes.
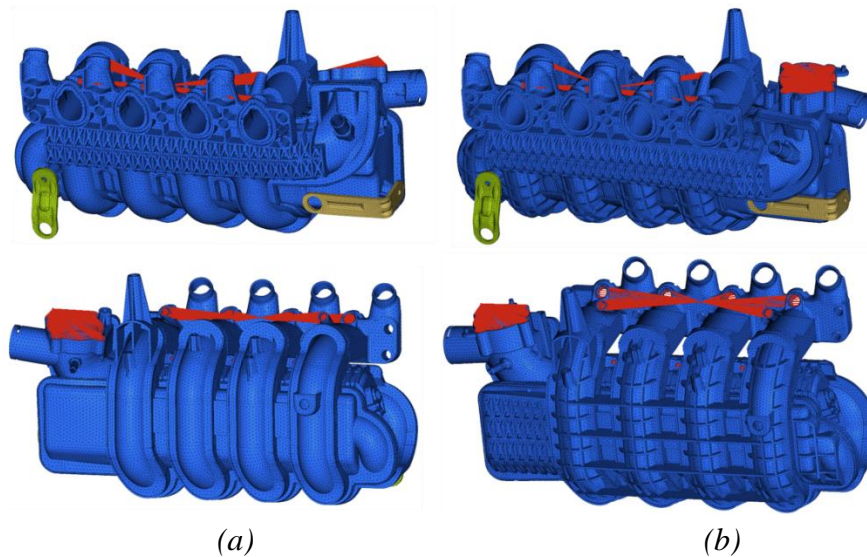

(a)   *(b)*

Figure 3: Structural finite element meshes of air intake manifolds. (a) without ribs. (b) reinforced with ribs.

The boundary conditions considered in the structural modal analyses are illustrated in Figure 4. The throttle body and the fuel rail were considered as concentrated point masses, linked to the mesh with RBE3 elements.
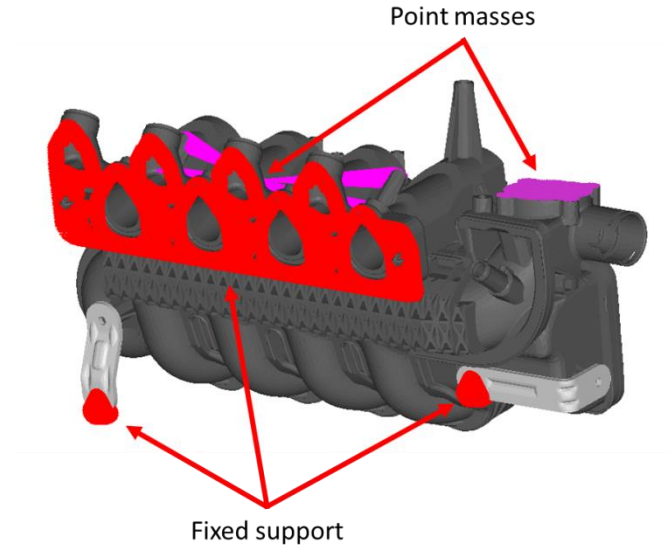
Figure 4: Boundary conditions used in the structural modal analyses.

Table 1 presents the intake manifold material properties considered in structural modal analyses. Once there is no information about the glass fibers alignment, average properties were considered with isotropic elasticity.

Table 1: Intake manifold material properties (polyamide PA 6 GF30).

Material	Density [kg/m^3]	Young Modulus [MPa]	Poisson [-]
PA 6 GF30 @ 80 ºC	1.37e-09	3015	0.4

The vibro-acoustic simulations were performed using the software SIEMENS LMS Virtual.Lab version 13.4. The internal cavity finite element mesh considered in the analyses is shown in Figure 5. This mesh is composed of 328,991 linear tetrahedron elements. As both manifolds have the same internal design, the cavity mesh was shared with both vibro-acoustic simulations.
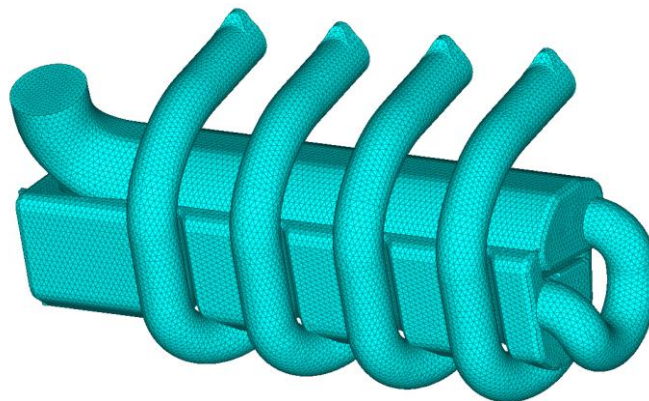

Figure 5: Intake manifolds internal cavity finite element mesh.

The external far field finite element mesh considered in the simulations are shown in Figure 6. Both meshes are composed of liner tetrahedron elements (304,208 elements for the manifold without ribs and 398,545 elements for the manifold with ribs).

The vibro-acoustic analysis were performed regarding the Automatically Matched Layer - AML approach for the external mesh property. This AML method developed by LMS allows to use small and therefore fast FEM models for exterior acoustics and also has some benefits for the analysis: wave absorbing layer modeled inside the solver, absorbing layer optimized for each frequency line, improved absorption function still even faster [3].

(a)   *(b)*
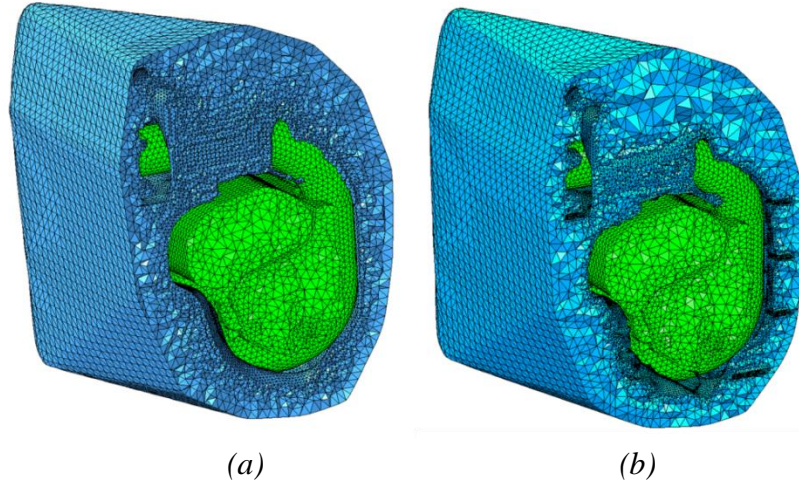
Figure 6: External far field finite element mesh. (a) without ribs. (b) reinforced with ribs.

Table 2 presents the air properties of air considered in the vibro-acoustic simulations.

Table 2: Air acoustic properties.

Fuid type	Density (Real / Imag) [kg/m³]	Sound Velocity (Real / Imag) [m/s]
Air environment @ 25 ºC	1.225 / 0.0	340.0 / 0.00
Air Inside cavity @ 80 ºC	1.000 / 0.0	376.7 / 3.77

Table 3 presents the acoustic boundary conditions considered in the vibro-acoustic simulations. The regions were these boundary conditions were applied are shown in Figure 8. The pressure profile measured at the throttle body valve is shown in Figure 7. These data were used as acoustic boundary conditions on the simulations.

Table 3: Acoustic boundary conditions.

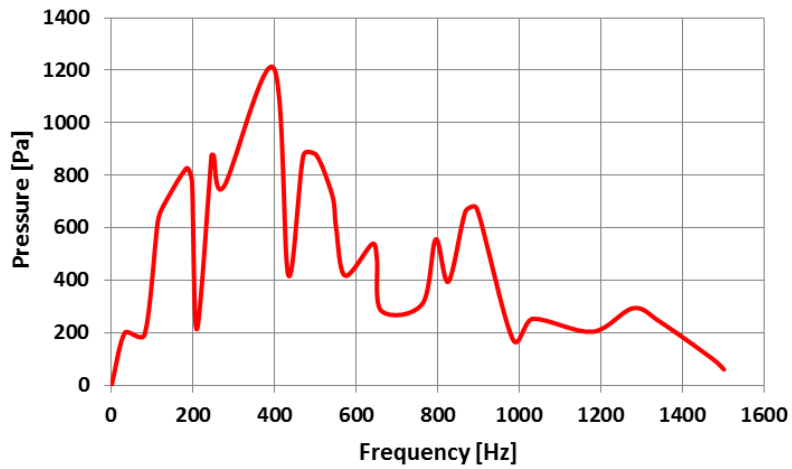| Region	BC type	Admittance (Real / Imag) [m²s/kg]	Pressure [Pa]
Intake manifold outlets	Absorbent panel	416.5 / 0.0	-
Intake manifold throttle inlet	Acoustic pressure	-	Profile of Figure 7
External air envelope	AML	-	-

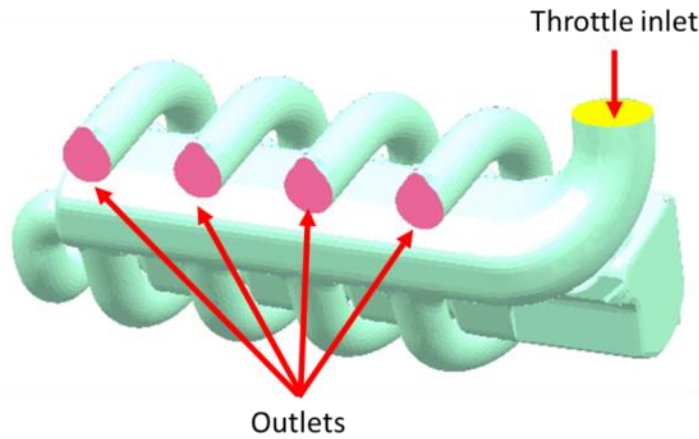
Figure 7: Pressure profile measured on the throttle body.

Figure 8: Acoustic boundary conditions considered in vibro-acoustic simulations.

For sound pressure level (SPL) determination, even in simulations or experimental tests, three microphones were placed at a distance of 1m from the engine, according to Figure 9.
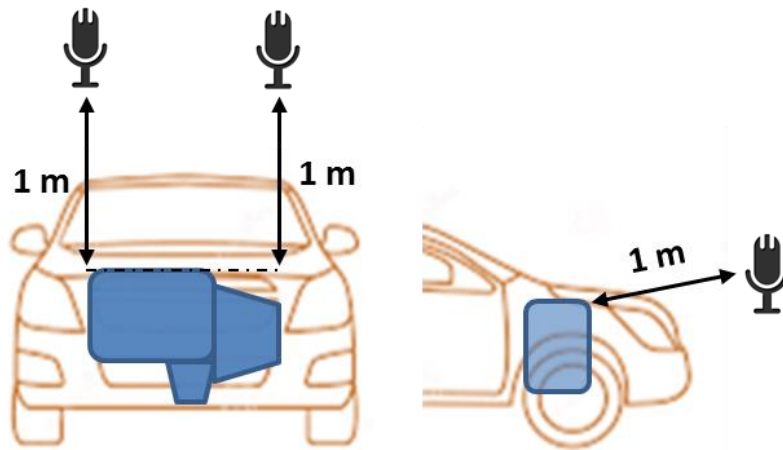
Figure 9: Microphones locations for sound pressure calculation/measurement.

INTAKE MANIFOLD NVH EXPERIMENTAL TESTS

The experimental tests were performed in a semi-anechoic test cell regarding hot idle (750 rpm) and run-up wide open throttle (WOT) conditions. All the tests were conducted with the vehicle placed on a chassis dynamometer with the engine compartment hood removed.

The sound pressure levels were measured by means of ½ inch free field microphones placed outside the vehicle, according to the locations shown in Figure 9. For comparison purposes between the two intake manifold designs, the overall pressure level is evaluated, which refers to the composite sound pressure level that reflects the overall spectrum of acoustical frequencies associated with the measured sound.

During the tests, accelerometers were placed on specific regions for the surface velocities acquirements. Figures 10 and 11 show the locations where the accelerometers were placed on the manifolds. These same points were considered in simulations for panel velocities calculation.
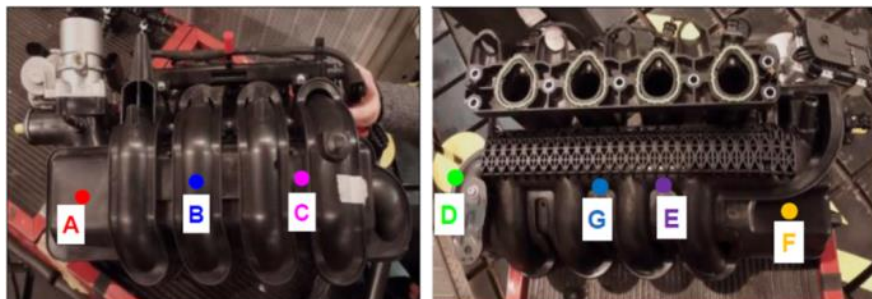

Figure 10: Panel velocities measure points on air intake manifold without ribs.
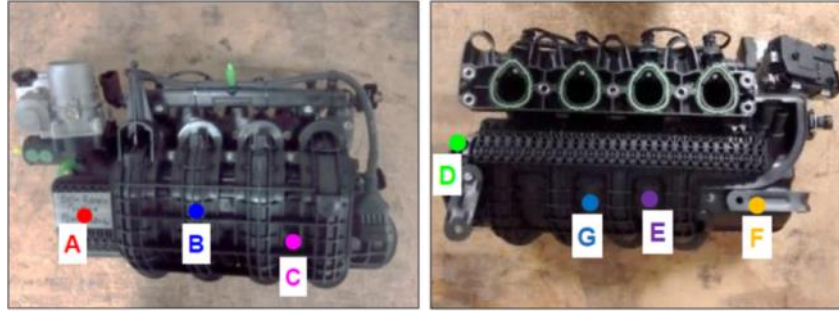
Figure 11: Panel velocities measure points on air intake manifold reinforced with ribs.

RESULTS AND DISCUSSION

The structural vibration modes, the sound pressure levels and the panel velocities results obtained from the numerical simulations are presented below. The experimental sound pressure levels and the panel velocities measured on the actual components are presented in the sequence.

Figure 12 shows the first five vibration modes of both intake manifold models. They correspond to the main important structural modes which can be excited by the engine vibrations. Practically all the modes and frequencies are coincident, with some insignificant differences due to some design adjustments related to the throttle body position.
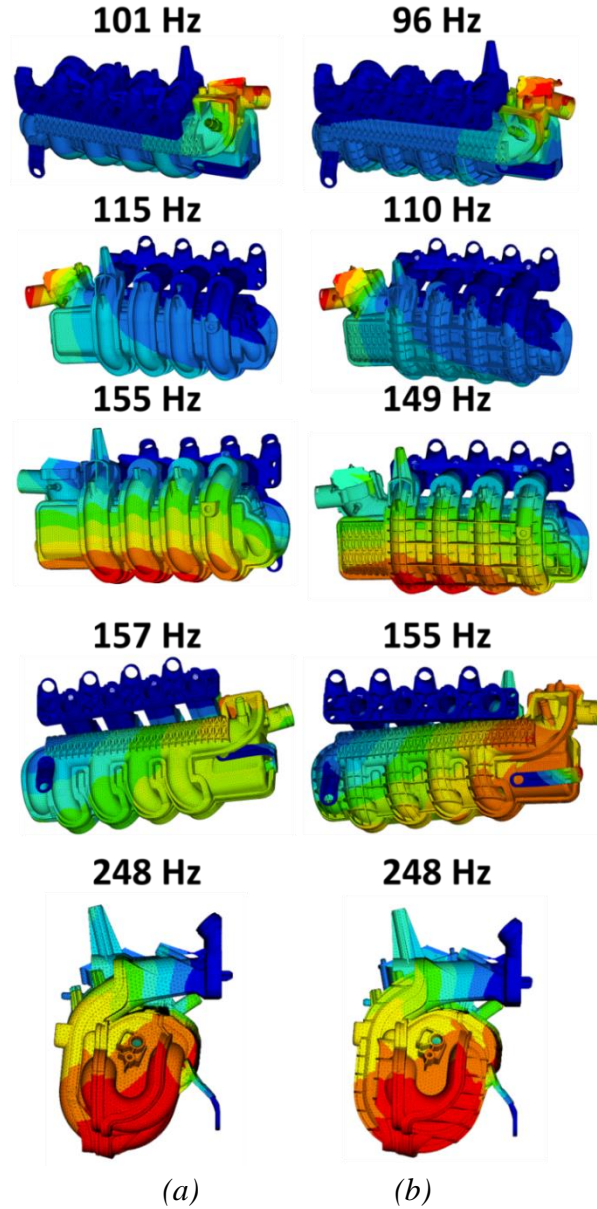
Figure 12: First structural vibration modes of air intake manifolds. (a) without ribs. (b) reinforced with ribs.

The first acoustic cavity modes are shown in Figure 13. The first mode refers to the resonator mode. The sound pressure levels calculated in the vibro-acoustic simulations are shown in Figure 14. They correspond to the average values calculated for all virtual microphones. The overall pressure level decreased from 101.07 dB to 100.96 dB in the ribbed manifold version when compared to the non-ribbed one. Regarding this result, any noise reduction was achieved by improving the manifold stiffness. Substantial noise attenuation was achieved only at around 650 Hz, with a SPL decrease of 9 dB(A).
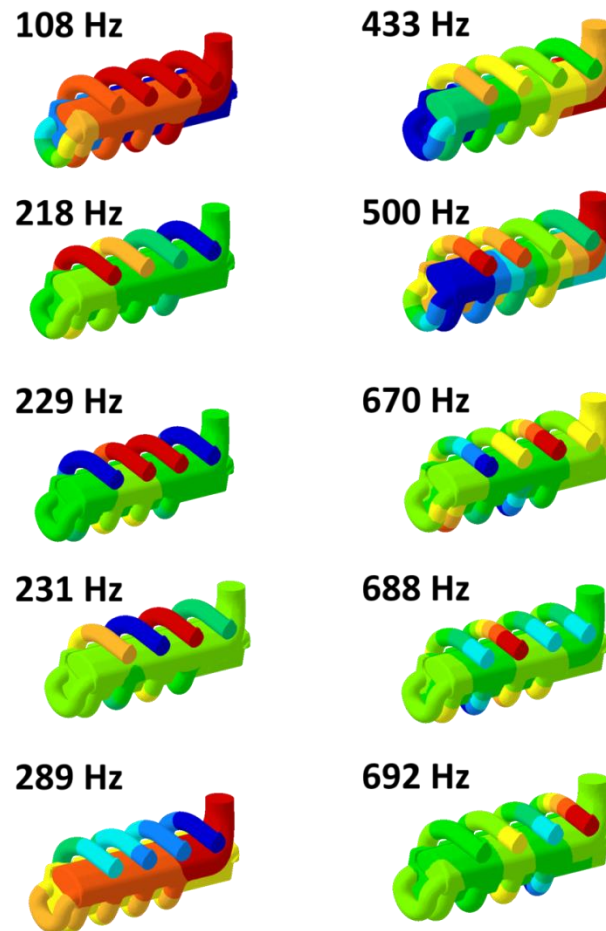
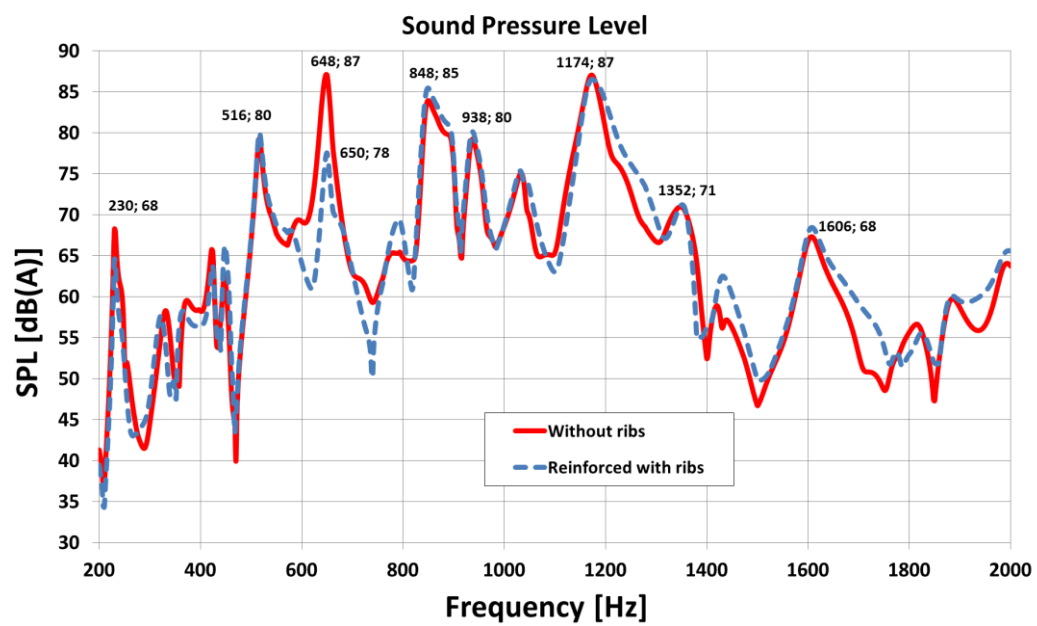*Figure 13: First acoustic vibration modes of air intake manifolds.*

Figure 14: Average sound pressure levels calculated in the vibro-acoustic simulations.

The panel velocities calculated for non-ribbed manifolds are shown in Figure 15. The peak panel velocity occurs at frequencies around 230 Hz, 650 Hz, 850 Hz and 1170 Hz with maximum values of 55 mm/s on point E, 103 mm/s on point A, 81 mm/s on point A, and 118 mm/s on point A.

For the ribbed manifold, the peaks found at same frequencies are 40 mm/s on point E (-27%), 43 mm/s on point A (-58%), 16 mm/s on point A (-80%) and 29 mm/s on point A (-75%). These results for the ribbed manifold are presented in Figure 16.
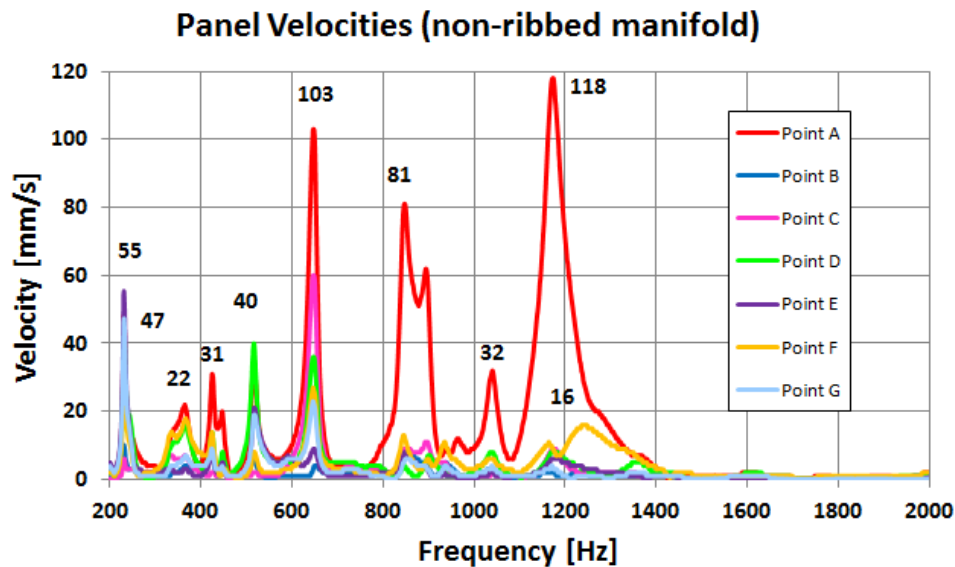

Figure 15: Panel velocities on intake manifold without ribs calculated for the selected points shown in Figure 11.
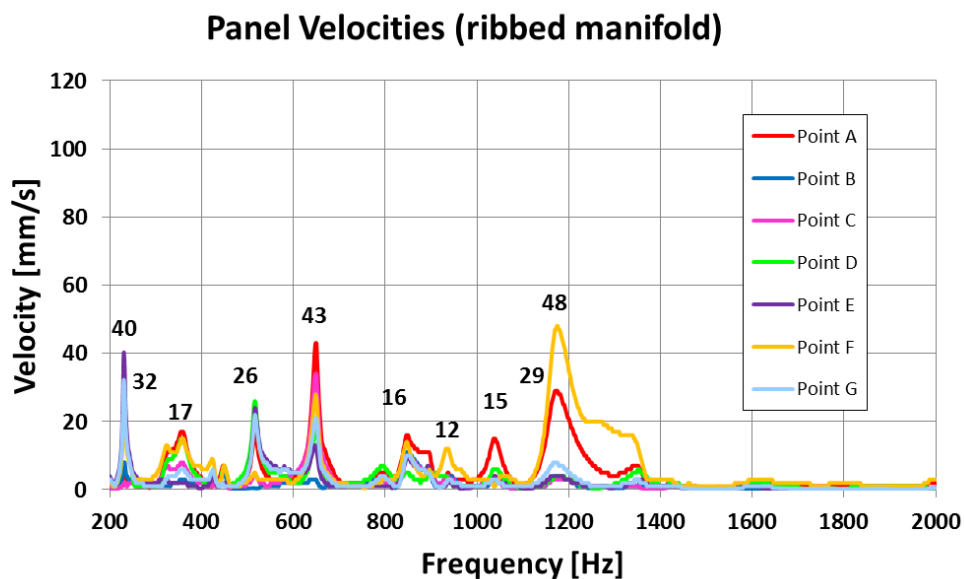

Figure 16: Panel velocities on intake manifold reinforced with ribs calculated for the selected points shown in Figure 12.

Regarding the non-ribbed manifold, the SPL and the panel velocities have the most critical case occurring at a frequency around 650 Hz on point A. Observing the panel modal shape of the non-ribbed manifold for this frequency on Figure 17, there is a high relative displacement on the resonator surface.
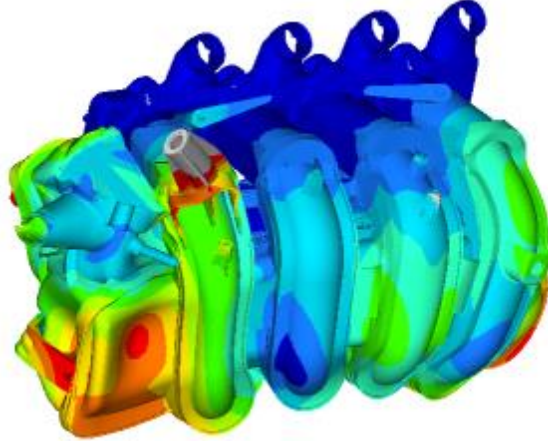

Figure 17: Panel vibration mode of air intake manifold without ribs at 652 Hz.

On Figure 18 is possible to see a vibration amplitude decreasing of 55% at resonator surface on the ribbed manifold when compared to the non-ribbed design. The ribs added to the resonator surface improved that region stiffness and therefore lower panel displacement and velocity was achieved. As a direct consequence, the sound pressure levels have a perceptive reduction.
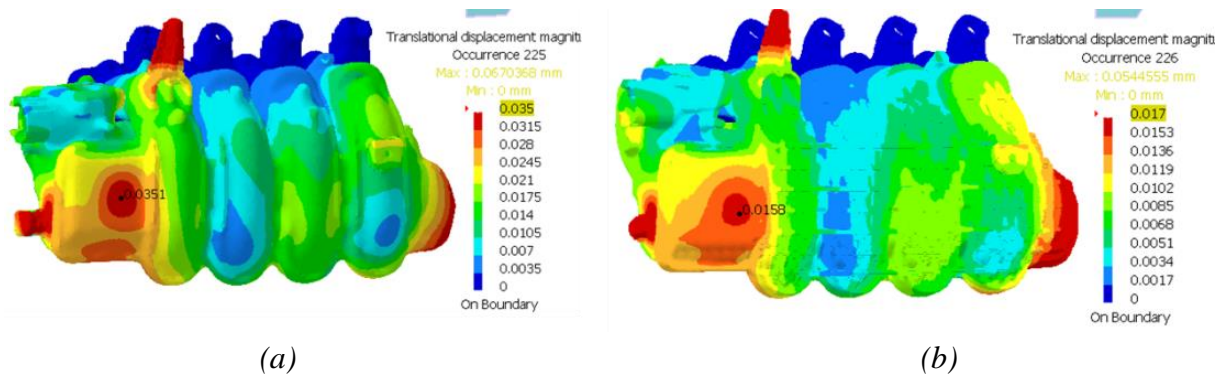

(a)   *(b)*

Figure 18: Translational displacement magnitude at 650 Hz. (a) manifold without ribs. (b) manifold reinforced with ribs.

Figures 19 and 20 show the directivity of the sound field at around 650 Hz for the non-ribbed and ribbed manifold, respectively. In this specific condition, it is possible to see the substantial attenuation of the noise propagation generated by the ribs addition, mainly on the resonator surface.

Basically, it was possible to reduce considerably the noise at around 650 Hz because there is a structural panel mode combined to a excitation pressure peak close to this frequency.

Moreover, there is an acoustic cavity mode at 670 Hz, which contributes to amplify the surface vibration.

If one take a look on the both manifold designs, it will be notice that the resonator wall is the only flat region where ribs were added. The other regions that received the reinforcement correspond to the runners surfaces. In other words, the noise attenuation follows the panel velocities reduction when there is a free surface vibration, that actually corresponds to a panel vibration mode.
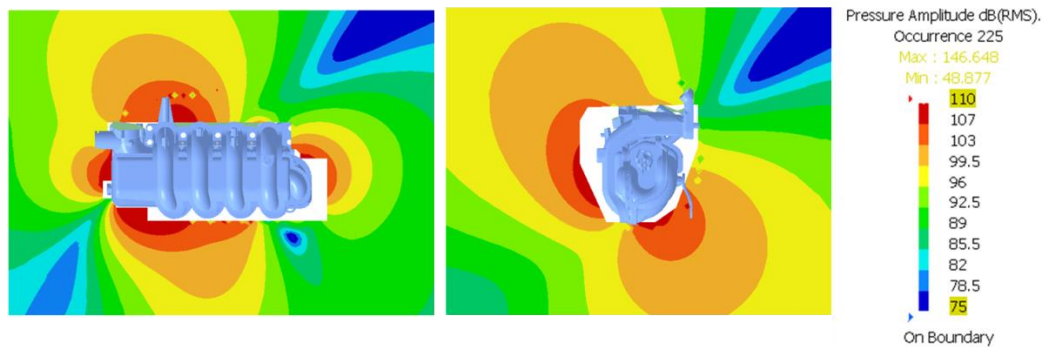

Figure 19: Directivity of the sound field from intake manifold without ribs at 648 Hz.
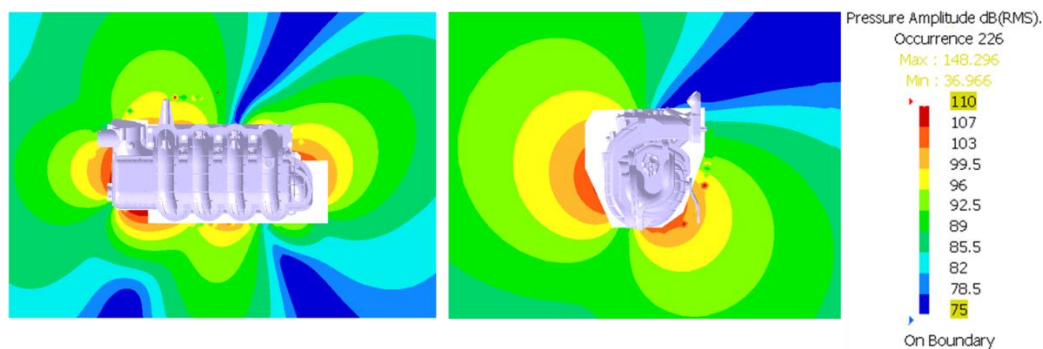

Figure 20: Directivity of the sound field from intake manifold with ribs at 650 Hz.

The sound pressure levels presented in the Figure 21 were acquired from experimental tests in hot idle condition. The overall pressure level decreased 2.6 dB in the ribbed manifold version when compared to the non-ribbed one.

In practice, a decrease of 3 dB(A) is barely perceptible to the human ear, which indicates that the structural improvement, by adding external ribs, was not enough to result in a considerable noise reduction within the whole engine frequency range. Looking at specific frequency bands, below 400 Hz and above 4000 Hz, the sound pressure had more significant attenuation. Depends on the vehicle driving condition, a substantial noise reduction could be achieved.
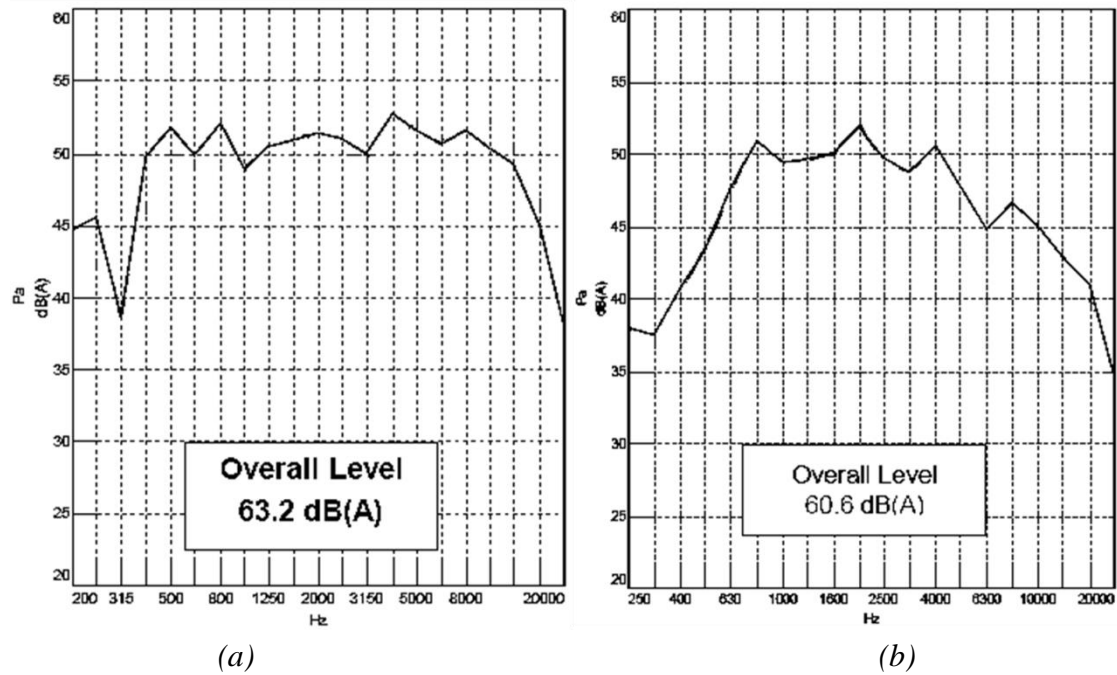
Figure 21: Sound pressure level acquired by the microphones. (a) manifold without ribs. (b) manifold reinforced with ribs.

The experimental panel velocities measured on the non-ribbed manifold are shown in Figure 22. The maximum values were observed at around 850 Hz with amplitude 62 mm/s on point A and at around 1630 Hz with amplitude 29 mm/s.
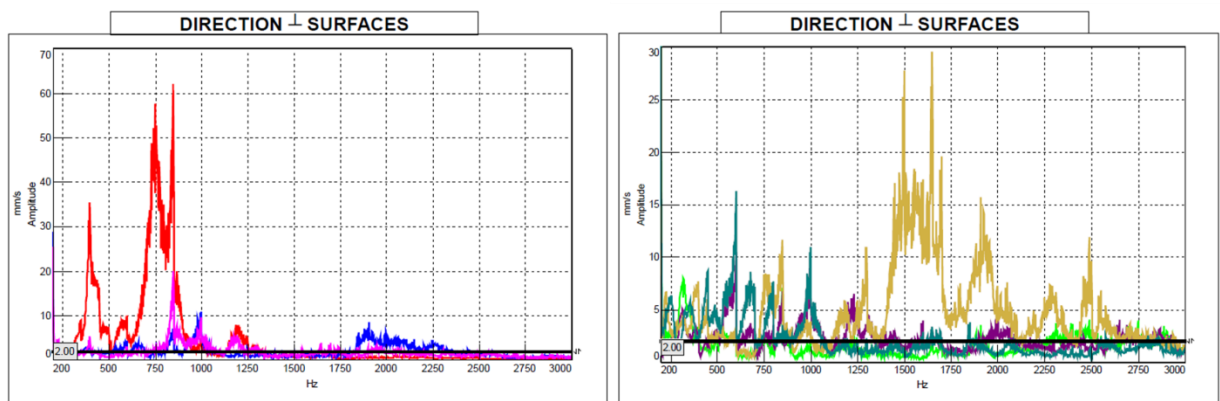

Figure 22: Panel velocities on intake manifold without ribs measured at the selected points shown in Figure 11.

The experimental panel velocities measured on the ribbed manifold are shown in Figure 23. The maximum values were observed at around 1600 Hz with amplitude 6 mm/s on accelerometer A and at around 450 Hz with amplitude 6.5 mm/s. The inclusion of the ribs on the manifold promoted a very important reduction of around 90% in the panel velocity amplitudes.
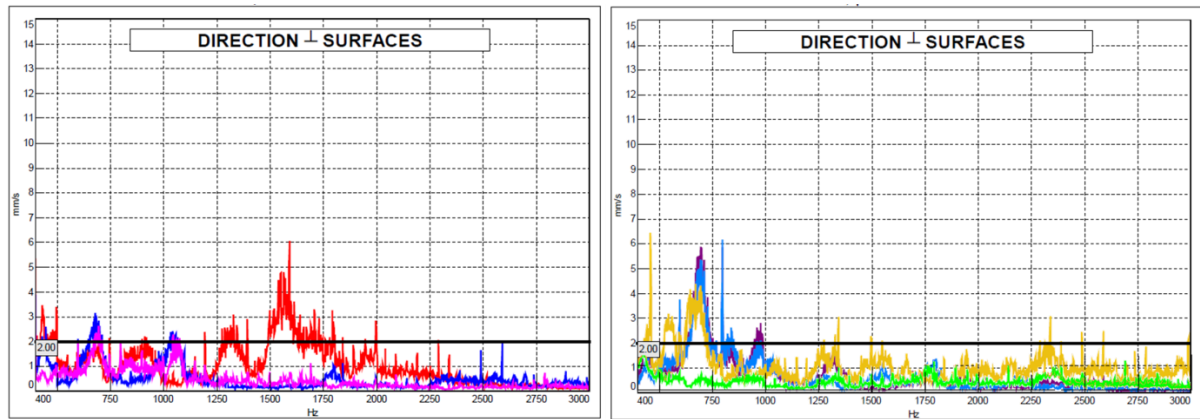
Figure 23: Panel velocities on intake manifold reinforced with ribs measured at the selected points shown in Figure 12.

Although no correlation was observed between simulation and experiments, it was possible to see on both a considerable decreasing in the panel velocities for point A, on the resonator surface. Moreover, the correlation was not an objective of the present study, once the experimental tests are affected by a lot of parameters that were not considered in the simulations. Also, the noise measured on the experiments comes from the whole vehicle operation on the dynamometer, and there is no way to isolate the intake manifold noise contribution.

CONCLUSION

Analyzing the vibro-acoustic simulation results, It could be observe a relation of sound pressure level attenuation with the structural panel velocities on the intake manifold surfaces.

Regarding the global simulations results, it was possible to conclude that the inclusion of ribs on the intake manifold was not too effective for noise reduction, once the most of the ribs were added on the runners surface, which are not the main important for noise radiation. A considerable decreasing of sound pressure levels was achieved for a specific resonator flat region, where could be also observed a great panel velocity attenuation. Once there was a panel mode on this resonator surface, the addition of the ribs improved the stiffness leaded to in a considerable noise reduction.

The suggestions for future studies are to correlate the simulations with the experimental tests, trying to match the resonant frequencies, the vibration amplitudes and sound pressure levels. More realistic boundary conditions should be applied on simulations, including the engine structural vibration, measured damping on whole frequency range and more accurate material properties.

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