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Energy Procedia 81 (2015) 742 – 754

Energy Procedia

69th Conference of the Italian Thermal Machines Engineering Association, ATI2014

Sensitivity analysis and correlation Experimental/Numerical FEM-BEM for Noise Reduction assessment of an engine beauty cover

Siano D.[a], M. Viscardi [b], Aiello R.[c]

[a] *Istituto Motori - CNR (National Research Council), Via Marconi 4,Napoli 80125, Italy*
[b] *University of Naples "Federico II", DII (Department of Industrial Engineering),Via Claudio 21, Napoli 80125- Italy a,b*
[c] *PhD. Student University of Naples "Federico II"- Istituto Motori - CNR (National Research Council), Via Marconi 4, Napoli 80125, Italy*

Abstract

In this paper a deep analysis on an internal combustion engine's beauty cover is carried out for identifying the best material for Noise Reduction improvements.
The paper is organized as follows:

- Experimental modal analysis and comparison with the FEM simulation
- Experimental test on Noise Reduction assessment by using a novel experimental set up with a Pressure/Velocity Sound Intensity
- Numerical Boundary Element Analysis for computing the Noise Reduction on the engine cover
- Comparison between the experimental and BEM simulation

The comparison of the experimental and FEM/BEM analysis results show a very good agreement in all investigated frequency range. The obtained results encourage to use the numerical model for further investigations aimed at the improvement of the acoustic performances.
The implemented numerical procedure can be applied successfully not only in automotive field but also in all problems where material acoustic performances is due.

Peer-review under responsibility of the Scientific Committee of ATI 2014

Keywords: FEM, BEM, modal analysis, beauty engine cover, Noise Reduction.

1. INTRODUCTION

In recent years, the acoustic problem has represented a rate of the quality of human life. Aside legal obligation, all the big automotive manufactures are directed to solutions for better passenger's acoustic comfort. The reduction of

1876-6102
Peer-review under responsibility of the Scientific Committee of ATI 2014
doi:10.1016/j.egypro.2015.12.080

vibration and noise in and across several components and modules of the automotive, such as the panels, doors, engine covers, seats, and others, is of primary importance. The NVH performance may be a crucial factor in the purchase decisions of numerous buyers. Engine noise represent an important quote of the overall noise perceived inside and outside the automobile and this the reason why the relative reduction strategies represent an important engineering challenge. The engine cover primarily act as an aesthetic trim on top of the engine, but can also reduce noise and improve pedestrian safety.

Engine beauty cover, if well designed, gives a great contribution to the sound quality of a vehicle.

Nowadays, most of the automotive industry uses engine covers realized with rigid plastic such as nylon for the external skin and polyurethane foam used as sound absorption element.

Noise reduction up to 5 dB is possible from the foam's noise damping properties and the ability of the foam to fit snugly to the contours of the top of the engine block.

Even if deeply used, this configuration may not guarantee the noise reduction requirements in all the circumstances. For this reason, the research is going to study the possibility to use innovative materials to assess innovative cover configuration in order to improve the passenger compartment noise abatement.

Engine sound level is important for obtaining type approval. To this end, the simulation capabilities of the FEM/BEM, numerical implementation through the use of commercial software Virtual Lab (LMS Siemens), [11],[14], proved to be very effective in obtaining higher soundproofing capacity of presented engine beauty cover.

The aim of the presented work is to realize a numerical/experimental acoustic properties evaluation of the engine cover under study in order to identify further improvements by using different materials as passive control.

To this aim, the present study can be summarized as following:

- Experimental modal analysis and comparison with the FEM modal analysis
- Experimental test on Noise Reduction assessment by using a sound intensity probe
- Boundary Element Analysis for engine cover Noise Reduction assessment
- Comparison between experimental and BEM simulation

The compared results between experimental and FEM/BEM analysis show an excellent agreement in all investigated frequency range. The implemented numerical procedure can be applied successfully not only in automotive field but also in all problems where material acoustic performances is due.

In figure 1 is reported the flow chart of the implemented procedure.

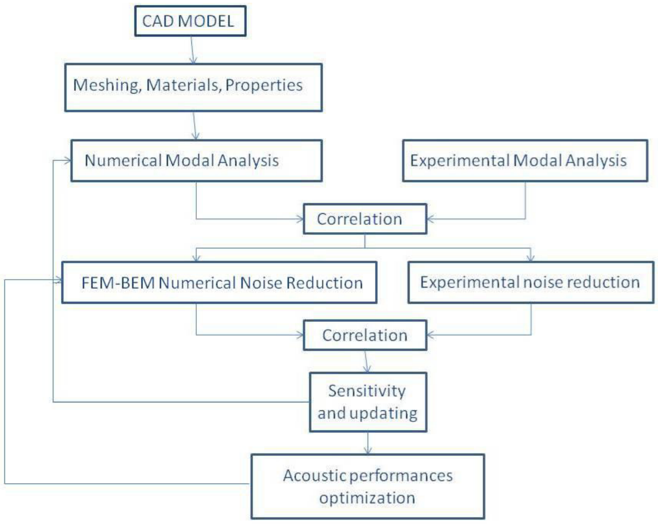


Fig.1 Flow chart of the presented study

Starting from CAD model, a FE mesh is realized. The natural frequencies in terms of mode shapes are compared with the obtained experimental ones.

The very good correlation between them allow to use the numerical model for further analysis aimed to carry out its acoustic performances.

Nomenclature

BEM	Boundary Element Method
CAD	Computer Aided Design
FEM	Finite Element Method
FRF	Frequency Response Function
FFT	Fast Fourier Transform
PSD	Power Spectrum Density
LSCE	Least-Squares Complex Exponential
SPL	Sound Pressure Level
PDE	Partial Derivatives Equation

2. ANALYSIS APPROACH

A FEM/BEM approach is used to predict the noise reduction of engine cover.

As well known, Finite Element Analysis (FEA) is considered as a numerical method to provide solutions to many engineering problems that couldn't be solved manually. In 1943 R. Courant used Ritz method to obtain an approximate solution for some vibration problems [1].

FEA consists of a mathematical model of a material or design that is analysed for specific results, in this application the results are the natural frequencies, and mode shape. In finite element method the structure under study is divided into very tiny elements connected together by nodes .The used commercial software as pre-processor and post processor has been Ansys, [2] . The algorithm used in this solver was Lancsoz Eigensolver Method to calculate the fundamental frequencies and the corresponding mode shapes related to each frequency as this method is considered a powerful tool for extraction of the extreme eigenvalues and the corresponding eigenvectors of vibration problems in engineering. It uses interpolation methods to extract mode shapes and natural frequencies for mechanical structures with complex geometries.

The main aim of FEM is the discretization trough the creation of a mesh, made by finished elements in codified shape (triangular and quadrilateral for 2D domain, hexahedral and tetrahedral for 3D domains, i.e.). On each element, characterized by this elementary shape, the problem's solution is expressed by linear combination of based functions or shape functions.

The basic Hypotheses about the structure concerning modal analysis are:

- The structure is *linear*, that is the answer of the structure for a linear combination of force, applied simultaneously on the structure, equals the sum of the single answers which are evaluated as if every force acting individually;
-The structure is *invariant time*, that is the calculated parameters are constants. The structure obeys to the principle of reciprocity of Maxwell, that is, if a force applied at a point P generates a response in a given point Q, then the same force applied in Q, will be caused the same response in P;
- the structure is *observable*, that is the set of input-output measurements can contain enough informations to describe adequately the behaviour of the entire structure.

In this work, firstly, the engine CAD cover model, reported in figure 2, has been opportunely meshed in order to achieve a good resolution in the investigated frequency range.

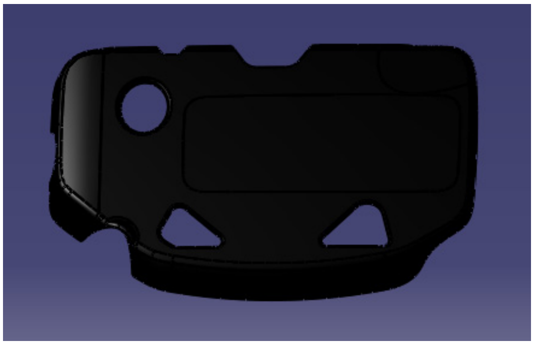


Fig. 2 Cad model of the beauty engine cover

The model consists of 13094 nodes and 12757 elements, including 158 triangular elements and 12599 quadrilateral elements and 4,43 mm of max element size. The aspect ratio has been considered as a goodness' rate of the mesh. The aspect ratio is the ratio of the characteristic quantities of the element shell, in case of a rectangular element, the characteristic quantities, are base and height. The more the aspect ratio tends to unity, the more the mesh will be good. This mesh allows to solve the modal analysis, with 100% of accuracy of the results, up to 2500 Hz.
The used mesh for FE analysis is reported in figure 3.

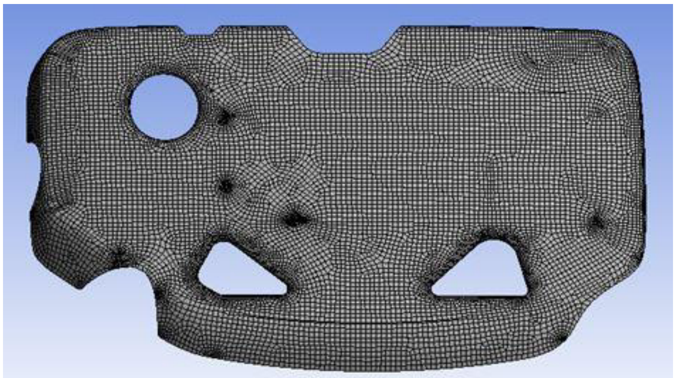


Fig. 3 Cover's mesh

The material properties in terms of Young's modulus (E), Poisson's ratio (ν), mass density (ρ) and 3% structural damping (considered appropriate for this structural component) were considered for simulation and the geometry with no constraints (representing completely free boundary conditions in the vibration) were input, too.
In Table 1 the cover properties are reported:

Material	Nylon PA 6.6
Young's Modulus [N/m²]	$3{,}884 \cdot 10^{9}$
Density [kg/m³]	1360
Poisson's Ratio	0,4

Table 1: Mechanical properties of cover material

The model has been analysed in the frequency range 0-2000 Hz. For sake of brevity only the first 20 mode will be showed.

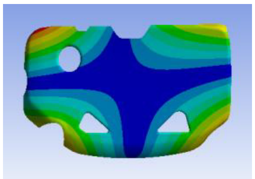


Fig. 4. Mode 1 at 12.79 Hz

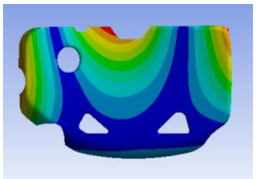


Fig. 4. (b)Mode 2 at 19.56 Hz

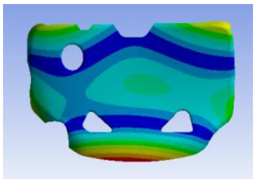


Fig. 4. (c) Mode 3 at 34.06 Hz

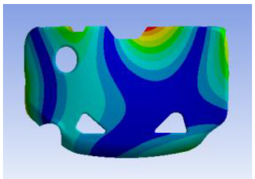


Fig. 4. (d)Mode 4 at 38.3 Hz

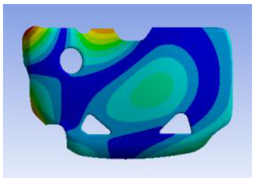


Fig. 4. (e) Mode 5 at 48,0 Hz

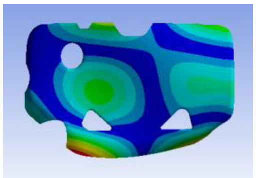


Fig. .4 (f) Mode 6 at 66,0 Hz

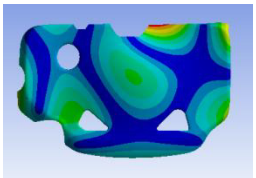


Fig. 4. (g) Mode 7 at 72,7 Hz

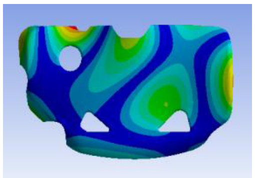


Fig. 4. (h)Mode 8 at 76,8 Hz

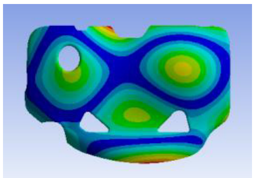


Fig. 4. (i) Mode 9 at 80,9 Hz

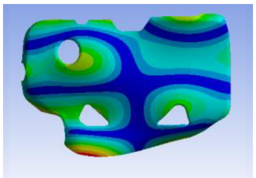


Fig. 4. (j)Mode 10 at 96,9 Hz

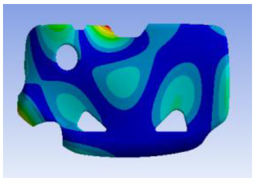


Fig. 4. (k) Mode 11 at 99,5nHz

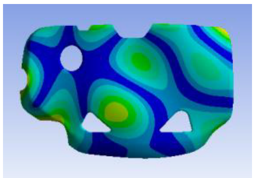


Fig. 4. (l)Mode 12 at 119,7 Hz

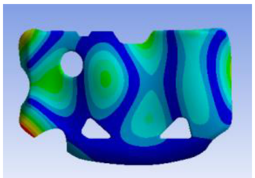


Fig. 4.m) Mode 13 at 127,9 Hz

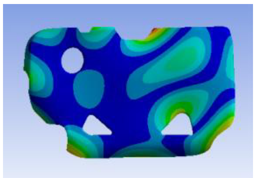


Fig. 4 (n)Mode 14 at 132,569Hz

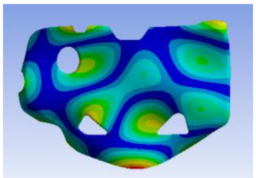


Fig. 4. (o) Mode 15 at 145,3 Hz

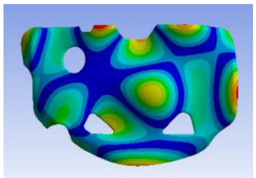


Fig. 4. (p) Mode 16 at 47,2 Hz

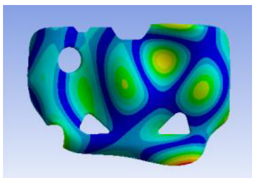


Fig. 4. (q) Mode 17 at 157,7 Hz

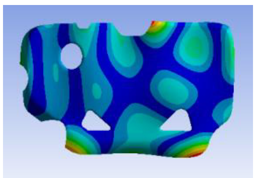


Fig. 4. (r)Mode 18 at 161,9 Hz

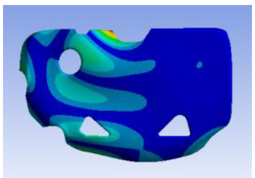


Fig. 4. (s) Mode 19 at 182,8 Hz

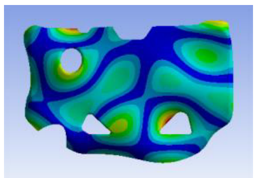


Fig. 4. (t) Mode 20 at 185,1 Hz

3. EXPERIMENTAL MODAL ANALISYS EVALUATION

Modal testing and analysis can be used to achieve a complete dynamic description of the structure.

The method involves exciting the structure and measuring its response in terms of displacement, velocity, or acceleration, the latter being the most common measurement. The latter and the impact force signals are then Fourier transformed into the frequency domain, from which frequency response functions (common called transfer functions) are established. The frequency response function (FRF) is analysed to find the natural frequencies, mode shapes, and the system parameters of equivalent mass, stiffness and damping ratio.

The impact test was developed during the late 1970's, and has become the most popular modal testing method used today. It allows to compute FRF measurements in an FFT (Fast Fourier Transform) analyser. When the output is fixed and FRFs are measured for multiple inputs, this corresponds to measuring elements from a single row of the FRF matrix. This is typical of a roving hammer impact test, which is the most common type of impact test, and which is also the type of test used in the present work.

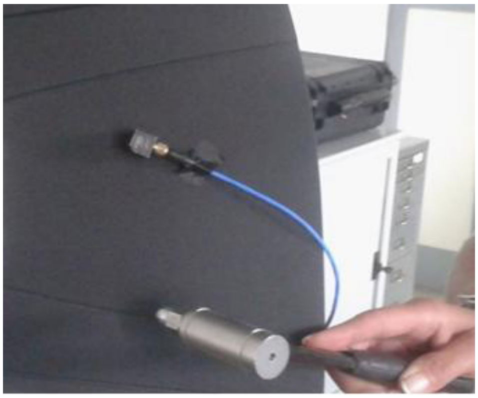
The necessary equipment to perform the impact test (see Figure) is composed by:

1. an impact hammer with a load cell attached to its head to measure the input force;
2. a PCB tri-axial accelerometer to measure the response acceleration at a fixed point (DOF) and directions;
3. a channel FFT analyser to compute FRFs (Multi-channel Scadas III Acquisition LMS Systems);
4. a pre and post-processing modal software for identifying modal parameters and displaying the mode shapes in animation (LMS Test.Lab).

The measurement setup of the used software requires as input data: geometry and orientation of the structure, positions of acquisition points (nodes) with respect to the chosen coordinate system, sensitivities of the tri-axial accelerometer (one for each direction) and of the load cell attached to the hammer, frequency range, trigger point. The last-named one is, usually, set to a small percentage of the peak value of the impulse.

The frequency range to excite is fixed so that a Teflon impact hammer head can be used. For testing the cover structure, 113 acquisition points are chosen for the experimental evaluation (Fig. 5), corresponding to the DOFs at which the structure is excited.

For this purpose the tri-axial accelerometer must be simultaneously sampled together with the force data. The sensitivities are: 0.23 mV/N for the load cell; for the accelerometer 102.5 mV/g in the x direction, 99.7 mV/g in the y direction and 100.8 mV/g in the z direction. The chosen frequency range is fixed up to 2000 Hz.



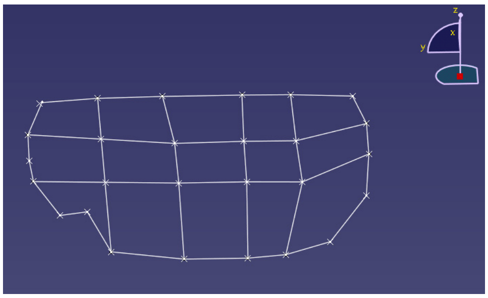


Fig. 5. Test set up and equipment (left); geometry structure (wireframe) used for modal impact test (right)

The experimental impact procedure is clearly reported in figure 6.

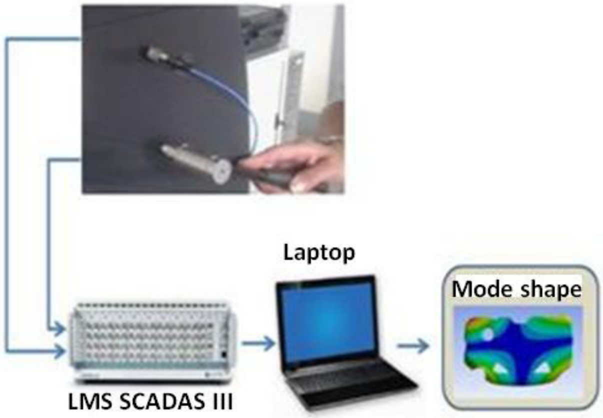


Fig.6 Experimental set-up of data acquisition through roving hammer impact test.

Obviously, it is important to locate the accelerometer in a proper manner, so that its directions coincide with reference ones. FRFs are so computed between each impact DOF and the fixed response DOF.

In order to reduce the error during impacts each DOF is hit 3 times, therefore each FRF is calculated averaging over 3 instantaneous FRFs. The first indication of experimental results is observed on the spectrum in the form of transfer function , phase, Power Spectral density and coherence plots. The coherence plot represents an important parameter that measures the accuracy of test data and a value of more than 95% is regarded as acceptable during the tests even though a rather lower coherence in field testing conditions due to external vibrations is also accepted.

Typical graphs of PSD and coherence are shown in figure 7.

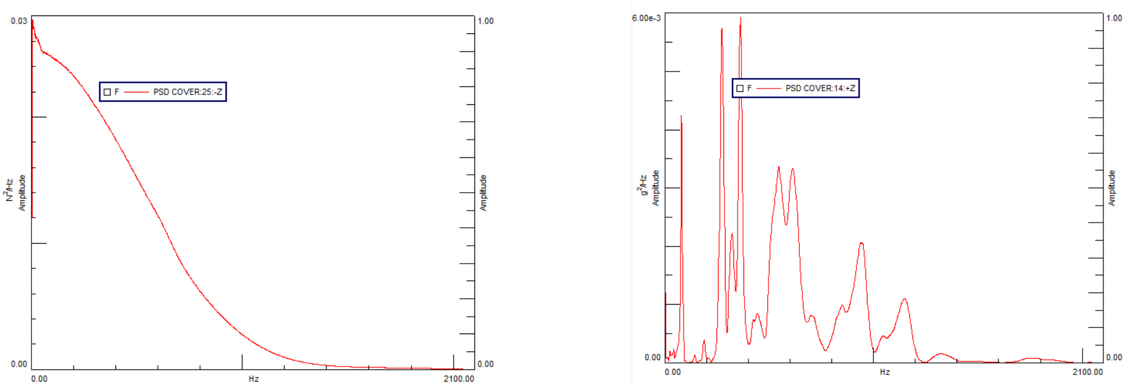


Fig. 7 Power Spectrum Density (PSD) of input force (left) and accelerometer response (right)

Figure 8 shows the coherence graph and the averaged FRF amplitude plot of one of the performed measurements.

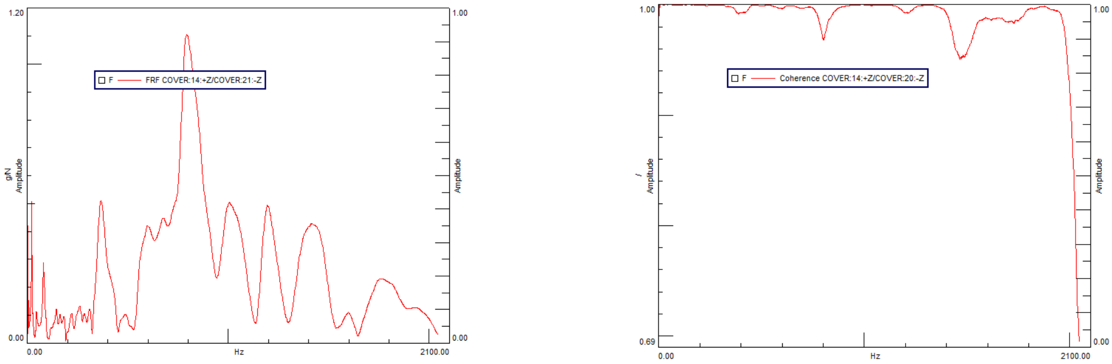


Fig. 8 Averaged amplitude FRF (left) and coherence plot (right)

Finally, testing results analysis, is carried out.

An example of the sum function of all the averaged acquired FRFs along z-axis is reported in Figure 9.

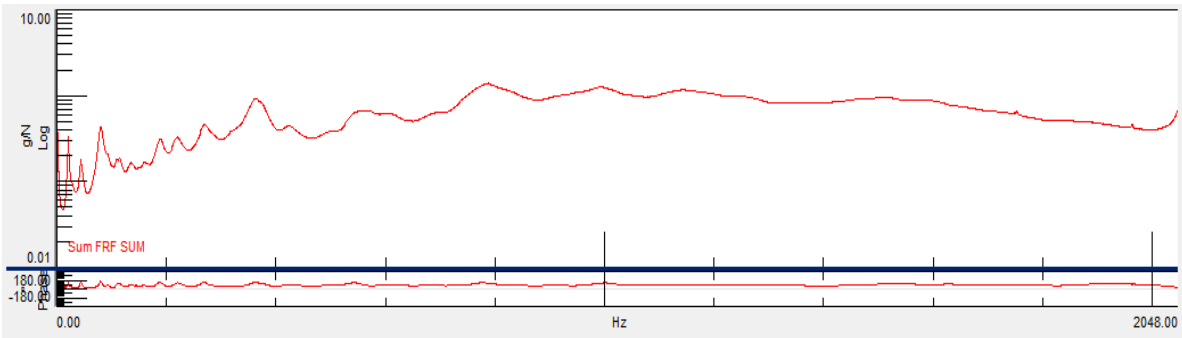


Fig. 9 Contribution along z-axis of FRFs sum

In modal analysis it is possible to use a number of parameter estimation techniques such as LSCE (Least-Squares Complex Exponential) and PolyMax. Estimated poles are calculated and the results of this operation are presented in a so-called stabilization diagram (Figure 10) from which stabilized modes can be picked. Such a diagram shows the evolution of frequency, damping, and mode/participation vectors as the number of modes is increased.

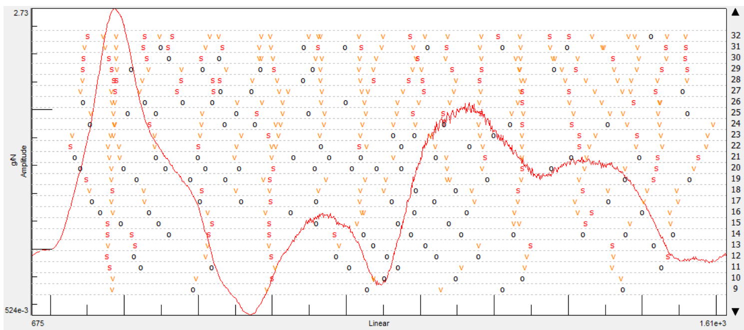


Fig. 10 The stability diagram in a part of frequency range

To reduce the bias on the modal parameters and to allow the capture of all relevant characteristics of the structure,

the identification order is usually chosen quite high. However, the higher the model order is, the higher number of estimated poles will be calculated. But this occurrence could not represent a real physical behaviour of the structure. For this reason, it is necessary to distinguish physical from mathematical poles. There are several pole-selection methods that allow engineers to pursue this aim. Generally speaking, it is possible to state that physical modes can be seen as those modes for which the frequency, damping, and mode/participation vector values do not change significantly and that they surely have to be searched among FRF peaks.
As discussed before, in order to obtain a reliable numerical model the experimental results were compared with the numerical ones and reported in Table 2

Mode	Numerical	Experimental	%
1	12,79	12,55	0,01
2	19,56	20,10	0,02
3	34,06	35,03	0,03
4	38,39	41,86	0,08
5	48,09	53,73	0,1

Table 2: Numerical and Experimental modal analysis comparison for the first 5 modes

The experimental and numerical results are in excellent agreement and this allows the use of the FEM numerical model for the successive acoustic parameter evaluation as previously mentioned.

4. EXPERIMENTAL NOISE REDUCTION INDEX EVALUATION

The most important acoustic quantity is the sound pressure, which is an acoustic first-order quantity. However, sources of sound emit sound power, and sound fields are also energy fields in which potential and kinetic energies are generated, transmitted and dissipated. In spite of the fact that the radiated sound power is a negligible part of the energy conversion of almost any sound source, energy considerations are of enormous practical importance in acoustics. Sound intensity is a measure of the flow of acoustic energy in a sound field. More precisely, the sound intensity "I" is a vector quantity defined as the time average of the net flow of sound energy through a unit area in a direction perpendicular to the area. The dimensions of the sound intensity are energy per unit time per unit area (W/m2).

The introduction of sound intensity measurement systems in the1980s has had a significant influence on noise control engineering. Sound intensity measurements make it possible to determine the sound power of sources without the use of costly special facilities such as anechoic and reverberation rooms. In some cases the environmental conditions must be kept in account to investigate the coupled source-environment behaviour; in other cases the measure must be done in situ on big scale objects. In both these applications the sound intensity method represents an attractive alternative to the conventional techniques.

Specifically, sound intensity analysis allows to determine typical material acoustic properties such as the surface impedance and the absorption coefficients, the Noise Reduction, the Insertion Loss and the Transmission Loss parameters.

To quantify the noise reduction parameter of the system sound intensity measurements according to ISO 9614-2 (intensity scanning) were performed on the system (see figure 11). For this purpose, a sound intensity probe kit (BSWA), intensity microphone pair and a LMS Scadas III multichannel data acquisition system, were used. The sound power of the source is determined by taking the surface integral across the measurement surface of the normal component of the sound intensity vector.

The Noise Reduction is one of the most important parameters for the investigation of sound attenuation. The investigation procedure is clearly reported in the following sketch (see figure 11)

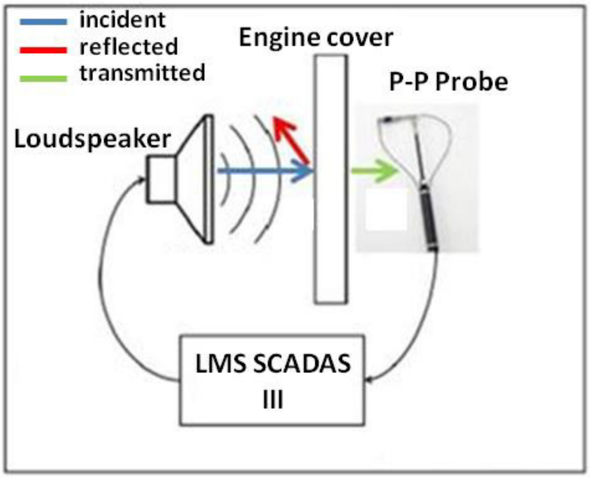


Fig. 11. Sketch of the experimental NR

In figure 11 is reported the used experimental set-up, where a loudspeaker emitting a white noise is placed on the left of the system. Then, through the use of an intensity probe, previously described the sound pressure level upstream and downstream of the cover, is measured. More precisely, the Noise Reduction (NR) can be expressed as following:

$$NR = SPL_i - SPL_t \tag{1}$$

where SPL_i represents the incident sound pressure level on the engine cover (dB) and SPL_t the sound pressure level trasmitted trough beauty engine cover (dB), respectively.

The intensity analysis measurements were carried out in a room acoustically treated. To determinate the Noise Reduction of the engine beauty cover, experiments are carried out trough the use of the pressure-pressure (p-p) sound intensity probe (Fig. 12). In table 3 are reported the used intensity kit.

Sound intensity probe SI512	
Frequency range (1/3 octave)	8,5mm spacer: 250-5000 Hz
	12mm spacer: 160-5000 Hz
	50mm spacer: 63-1250 Hz
Microphones pair MP201	
Diameter	1/2 inch, Prepolarized
Response	Free Field
Combined Sensitivity	45 mV/Pa

Table 3: Sound intensity kit

Fig. 12 Sound intensity probe

Hence, the loudspeaker provides an acoustic excitation (white noise) to the surface of the cover, the probe measures the trasmitted sound intensity for each defined point. Contemporarily, the loudspeaker excites all over the frequency range with a white noise signal.

The results in terms of Noise Reduction is reported and compared with the numerical ones in the next paragraph.

5. BEM NUMERICAL NOISE REDUCTION SIMULATION

Interior acoustic field analysis of structures can be performed using FEM and/or BEM approach. One advantage of BEM is that it uses 2D meshes only at the boundaries; whereas mesh generation in FEM is more complex since it needs 3D mesh throughout the volume. BEM can also be used when there are openings at the structure. However, BEM yields fully populated non-symmetric system of equations; whereas FEM yields symmetric system of equations. Considering the advantages and disadvantages of both methods, BEM is preferred. FEM is used to model the structure itself and it is coupled with BEM to perform vibro-acoustic analysis of engine cover for low to mid frequency ranges, [6], [7], [8].

For the analysis of acoustic domain excited by structure vibration, BEM requires vibration velocity (displacement/acceleration) of the structure in the normal direction as boundary conditions. Solving the Helmholtz integral equation with the boundary conditions, acoustic pressure at a predefined interior (or exterior) point can be obtained.

In order to apply BEM, PDE(partial derivatives equation), which rule the domain , must be reformulated like integral equations of defined functions just on the border of the same domain. Representing the domain's border through a set of surface panels and the border functions like parametric functions on each panel, the integral equation on the border reduce themselves to a linear system of equations and is possible to obtain a numerical solution.

In order to reproduce the same experimental intensity set up the well-known numerical software LMS Virtual.Lab, is used, [11]. In figure 13 the used acoustic mesh, is reported.

The boundary element mesh is simpler and coarser than the structural finite element mesh. Since features like the small ribs have dimensions much less than an acoustic wavelength, they have a negligible effect on the acoustics even though they may significantly impact the structural response. Those features should be removed from the solid model before meshing so that the mesh is coarser and can be analyzed in a timely manner. The boundary condition for the BEM analysis is the vibration on the cover (i.e. the particle velocity). The numerical simulation correspond to an interpolation of the vibration results from the structural finite element model on to the surface of the boundary element mesh.

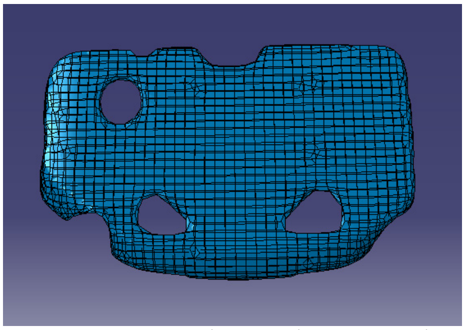
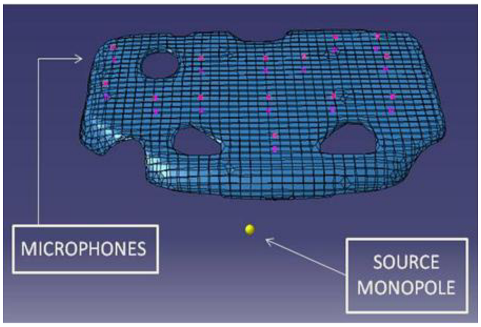



Fig. 13. Engine cover acoustic mesh BEM (left) and numerical set-up of Noise Reduction

In order to faithfully reproduce the experimental set up 113 virtual microphones are placed at a distance of 1 cm over and above the cover. In order to simulate the source used during the experimental campaign, the “source point” better known as “monopole”, is adopted, [12], [13]. The choice of using a monopole source depends on the fact that the used excitation during the experimental measurements is an omni-directionality source.

The numerical sound pressure levels in all microphone positions over and above the cover is carried out in all the investigated frequency range up to 2000 Hz . Then, the numerical Noise Reduction by using the equation 1 is evaluated and compared with the experimental one.

The overall comparison is reported in figure 14.

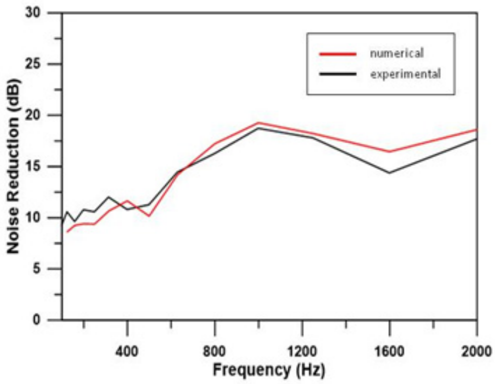


Fig. 14 Experimental and numerical Noise reduction comparison

As Figure 14 indicates, the BEM results compare reasonably well with the experimental results. The little differences some peaks can be attributed to errors in the measure of the damping of the engine cover.

In the curve can be quite well identified the mode's dominated frequency region, the mass dominating region and the coincidence frequency, as expected for homogeneous massive plate type structure.

6. CONCLUSION

In this work an integrated experimental numerical procedure to evaluate the acoustic performances of the engine cover is outlined and implemented. More precisely, two different experimental tests have been presented: the first to investigate the natural frequencies of the system (experimental modal analysis) and the second to calculate the acoustic performances, as well.

Contemporarily, in order to investigate future activities aimed to improve Noise Reduction parameter an integrated FEM/BEM numerical approach has been presented.

The comparison between experimental and numerical analysis are in excellent agreement opening the way to future improvements by the change of the material or the geometry of the structure.

It can be reasonably assumed that the finite element (FEM) and boundary element (BEM) techniques are suitable methods for predicting the low- and mid-frequency acoustic performances of an engine cover, as reported in this paper.

Moreover, the results suggest the potential benefit to use these methods early in the design phase.

The relative quick implementation and the opportunity to manage different aspects of the structure makes, in fact, these tools of great importance in an optimization design process.

Future developments will mainly regard the effects of the foam and of the plastic material modifications in order to assess a more complex parametric design tool.

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