An Hybrid FE/SEA Approach for Engine Cover Noise Assessment

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Abstract: - In this paper a deep numerical analysis on an internal combustion engine's cover is carried out. An Hybrid numerical model approach to evaluate the acoustic performance of engine's cover, has been used. The Hybrid model is composed of two numerical models: FE and SEA models. The model FE patterns the structural system, whilst model SEA patterns the acoustic environment. As main excitation an experimental data has been used for two considered engine conditions. The comparison in terms of sound pressure levels between the experimental and FEM/SEA analysis results shows a very good agreement, in the whole investigated frequency range. The obtained results encourage to use the numerical model for further investigations aimed at the improvement of its 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.

Key-Words: -Hybrid model, FEM, SEA, Engine cover, Transmission Loss, Power Velocity Level

1 Introduction

Vehicle sound quality is one of the important factors used to evaluate vehicle performance. Previously, only the most important automobile manufactures could ensure high performance of sound quality. Moreover, since last decade it is also compulsory by low, so industries must fulfill sound pleasantness requirements. Consequently, nowadays, totally automotive manufacturers invests a lot of effort and money to improve and enhance the vibro-acoustics performance of their products. The NVH (Noise, Vibration, Harshness) performance are coming a crucial factor in the purchase decisions of numerous buyers. The customer demands such requirements to make more pleasant driving and to protect their health. It is well known that a long overexposure to noise and vibration can provoke physics and psychological problems to the human health. In a vehicle there are a lot of source of noise (Fig. 1),

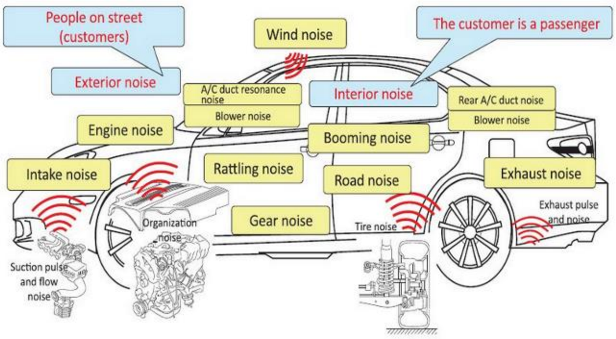

Fig. 1 Sources of noise regarding vehicle

as intake noise, gear noise, rattling noise, exhaust noise, booming noise, but the most important is, certainly, engine noise. Engine noise represents a substantial quote of the overall noise perceived inside and outside the automobile. The engine cover primarily acts as an aesthetic trim on top of the engine, it has become the principal actor of the passive attenuation of the engine noise. Engine cover, if well designed, gives a great contribution to the sound quality of a vehicle. In general the engine covers are composed of rigid plastic such as nylon for the external skin and polyurethane foam used as sound absorption element. Even if deeply used, this configuration may not guarantee the sound attenuation requirements in all the circumstances. The most important issues, in terms of attenuation of sound, of an engine cover are at low frequency range, exactly in correspondence of the engine's combustion frequencies. 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's compartment noise abatement.

The method is based on the hybrid FE-statistical energy analysis (SEA) formulation that allows the coupling of FE and SEA descriptions of various subsystems in a model. By making use of the dynamic properties of poro-elastic materials and acoustic fluids (in terms of wavelength and uncertainty), the method is made particularly fast and appropriate for the prediction of the vibro-acoustic response at mid frequencies.

The aim of the present work is to perform a numerical acoustic properties evaluation of the engine cover by using the hybrid FEM/SEA approach, and to realize a validation of the numerical model in terms of noise behaviour,

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through a comparison with the experimental results with the aim of identifying possible improvements by using different materials or modifying its geometry.

2 Hybrid Model FEM/SEA

In the last 20 years due to the increasing availability of computing power at low cost, and due to the increasing demands of accuracy the typical approach for the solution of engineering problems has been subject to a great evolution. The typical engineering approach is to rely on approximate models based on simplified theories or semi-empirical approaches. The great uncertain in the results are then balanced introducing security factors and relying on the personal experience of the engineer. Nowadays however the need to reduce costs both of economic and computational, and to increase the performances require the capability to predict the behavior of the component that has to be designed with a greater accuracy. The classical engineering approaches are always used, since they constitute the basis for a sound engineering practice, but there is more and more the need to complement them with more sophisticated approaches that can provide more accurate results. The typical approach is therefore to go back to the basic physical equations that drive the phenomenon to be analyzed and to solve them numerically. The increased accuracy, as usual, does not come for free, and an increased effort is required in order to model the problem and analyze the results. New knowledge is, therefore, required to the engineer in order to take full advantages from the application of numerical simulations.

Among the numerical models it is spreading, more and more, the hybrid model. The hybrid model allows to take advantages of points of force each used numerical model. In this work, the hybrid model has been built using a commercial software of ESI GROUP®, VA ONE® ,and it is composed of FEM (Finite Element Model) and SEA (Statistical Energy Analysis) numerical models.

The FE model, now widespread technique to solve various engineering problems, is a mathematical method that it allows to find numerical solutions to the problems described by partial differential equation reducing them to linear algebraic equation, through the discretization of the object. This discretization is said mesh and constituted by nodes and elements.

Whilst regarding SEA model, at first, the name SEA was coined to emphasize the essential feature of the approach: "Statistical" indicates that the dynamical systems under study are presumed to be drawn from statistical populations or ensembles in which the distribution of the parameters is known. "Energy" denotes the primary variable of interest. "Analysis" is used to underscore the fact that SEA is a general framework of methods rather than a particular technique. SEA provides the designer with a method for estimating the response characteristics of such structures to vibratory excitations, from which he can predict the potential for structural fatigue, component failure, and human discomfort caused by noise or excessive vibration levels. SEA is particularly appropriate in applications involving relatively large and lightweight structures. These statistical models are also helpful to mechanical designers who are charged with making environmental and vibratory response estimates at a stage in a project where structural detail is not yet known.

Within VA ONE® the different solution methods to describe the response of a vibro-acoustic system, are patterned from different subsystems. Within VA ONE®, It needs to be worked on each subsystems, and to be joined subsystems through junctions. This junctions represent the interface surfaces that allow the passage of information among several systems. In the following there is a quickly general characterization of each subsystems.

Subsystems with many local modes are typically best represented using Statistical Energy Analysis subsystems. In SEA, the local modes of a subsystem are described statistically and the average response of the subsystem is predicted. It is usually not necessary to provide much detail when modelling a subsystem using SEA (for example, it is often only necessary to know the overall length, width or volume of a subsystem along with an approximate estimate of various properties that govern wave propagation within the subsystem). SEA subsystems are therefore well suited to modeling vibro-acoustic systems at the design stage where detailed information about the properties of a system is often not available.

Whilst, subsystems with very few local modes are often best represented using Finite Element (FE) subsystems. In FE, the local modes of a subsystem are described deterministically and based on detailed information about the local properties and boundary conditions of a subsystem. To get accurate descriptions of the modes and natural frequencies of the FE subsystems in a system it is necessary to describe the properties and boundary conditions of the subsystems in detail. FE subsystems are typically well suited for describing the response of the first few modes of a subsystem, and for

answering very detailed design questions about the local response of a subsystem..
Therefore, a coupled Hybrid model, can then be used to describe the response of a vibro-acoustic system. VA ONE® is the only commercial code available that has a complete implementation of the Hybrid method for rigorously coupling FE, and SEA subsystems together in a single analysis.

3 CAD Model and mesh preparation

The first important step is the simplification of the geometry by removing those parts not necessary for the numerical simulation. For this aim, a powerful software, of the ESI Group®, Virtual Performance Solution® (VPS) has been used to simplify the engine cover CAD model. In particular in the section Visual-Mesh, several operations to repair CAD are available. In the environment Visual-Mesh there are both automatic and manual operations. By automatic operations, the geometry has been checked to verify, the possible presence of the small holes or coincident surfaces or stitch surfaces. Whilst by manual operations, some edges have been repaired, some surfaces have been joined, surfaces which were untrimmed have been recreated, as well. In Figure 1a the red circles point out some edge recognized by the software as broken and in green the presence of holes not possible to close automatic operation. In Figure 1b and 1c, the zoom of that aerea depicted in figure 1a, are reported.

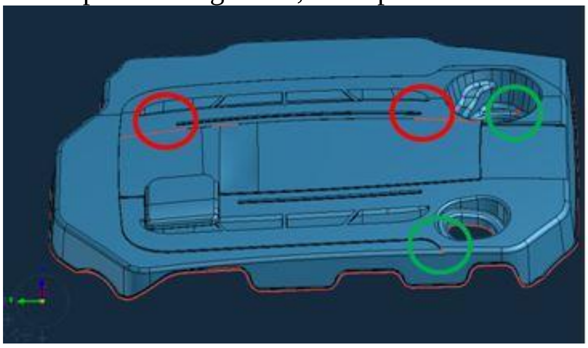

Fig.1a Original Cover CAD

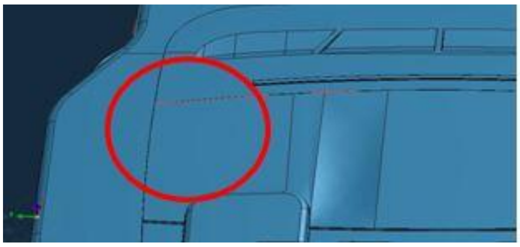

Fig.1b Zoom of the red circle top right

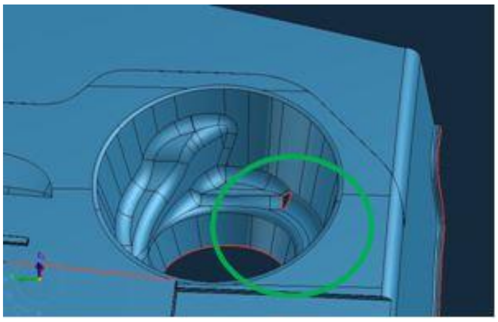

Fig. 1c Zoom of the top green circle

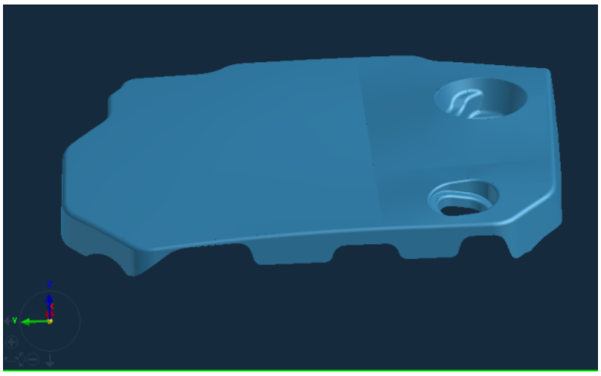

Fig. 1d Final CAD geometry after geometry preparation

In Figure 1d the obtained geometry after cad preparation , is shown.

4 Numerical Models

The purpose of the present work is to characterize, through the employment of the numerical simulation, the acoustic performance of the engine cover by using the TL (Transmission Loss) parameter. Secondly, a numerical frequency response analysis has been implemented by using experimental data in the car chassis for two considered engine conditions and the results have been then compared with the experimental ones.
In the following section, the two numerical models are described, and the obtained results in terms of TL and acoustic noise behaviour of the system, will be discussed.

5 Transmission Loss Numerical Model

In order to set up a reliable numerical model, a FEM/SEA approach, has been implemented. More precisely, the numerical model is composed of two models: FEM and SEA model. FE model represents the plastic structure and the soundproof material of the cover, whilst the SEA model represents the fluid environment. It is well known that, the Transmission Loss (TL) of a generic system is

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defined as the difference between the incident and transmitted power throughout the system:

$$TL = 10 Log_{10}\left(\frac{W_{in}}{W_{out}}\right) \qquad (1)$$

where, W_{in} and W_{out} represent the incident and transmitted acoustic power, respectively.
In the next paragraph a deep discussion of the used methodology, will be presented.

5.1 FEM Model

The FEM technique, now widespread technique to solve several engineering problems, allows to find approximate solutions for problems described to differential equations reducing them to algebraic linear equations, by discretization of the examined object. This discretization is called mesh. The used mesh model consists of 4417 nodes and 4428 elements, including 304 triangular elements and 4124 quadrilateral elements and 3 mm of min element side and 18 mm of max element side. 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 5000 Hz. The used mesh for FE analysis is reported in figure 2.

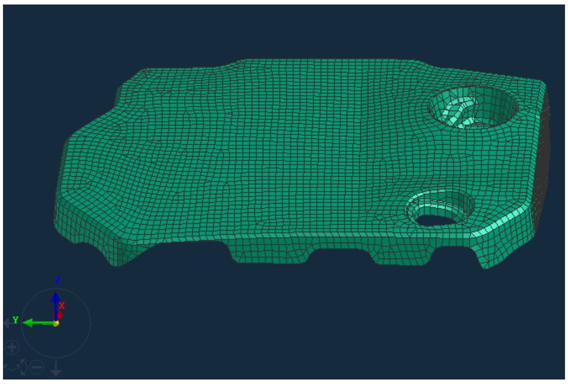

Fig. 2 Cover Mesh

As previously said, the mesh has been obtained through the use of Visual Mesh®.
As first step, a modal analysis has been computed in free boundary condition.
The material properties in terms of Young's modulus (E), Poisson's ratio (ν), mass density (ρ) and η=5% structural damping (considered appropriate for this structural component) have been applied for simulation. In table 1 the used structure material properties, are reported.

Properties		Nylon PA 6.6
Young's Modulus	[N/m²]	$2{,}3\cdot10^{9}$
Density	[kg/m³]	1100
Poisson's Ratio		0,48

Table 1 Nylon PA 6.6 properties

The modal analysis has been performed in the frequency range 0-1000 Hz. In figures 3a, 3b, 3c, 3d, 3e the first five mode shapes, are depicted.

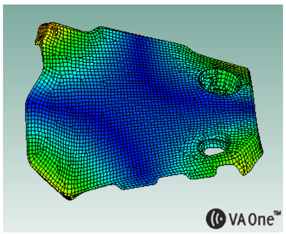

Fig. 3a Mode 1 @ 13,24 Hz

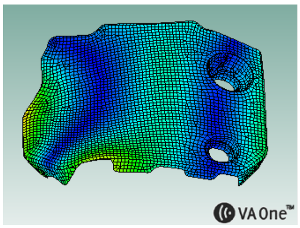

Fig. 3b Mode 2 @ 22,95 Hz

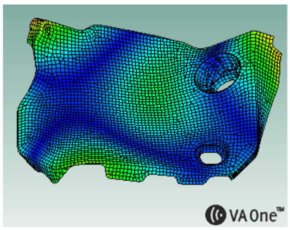

Fig. 3c Mode 3 @ 33,6 Hz

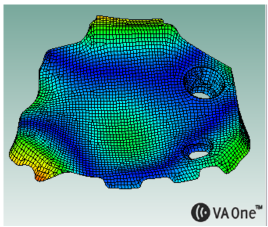

Fig. 3d Mode 4 @ 40,55 Hz

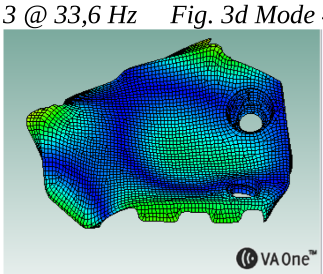

Fig. 3e Mode 5 @ 57,48 Hz

By viewing the above reported pictures, it is possible to note that the first mode shapes present the same behaviour of a typical panel mode shapes.
Once the structure modal analysis has been performed, the implementation of the real poroelastic material, has been realized and following reported.

5.2 PEM Model

With the purpose of realizing a reliable model for successive prediction of potential noise and vibration of the system under test, a layer of Poro-Elastic Material (PEM), with a thickness of 5 cm, has been modelled (Figure 4).

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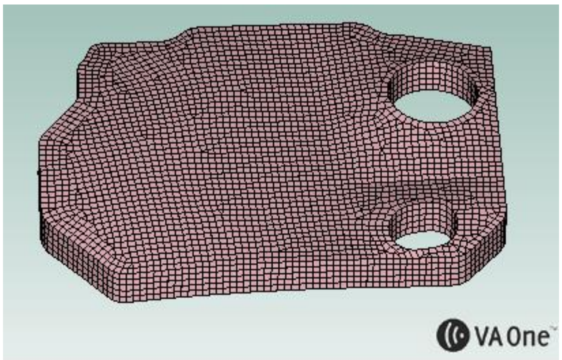

Fig. 4 PEM mesh model

The Biot's theory has been used to pattern the poro-elastic model.

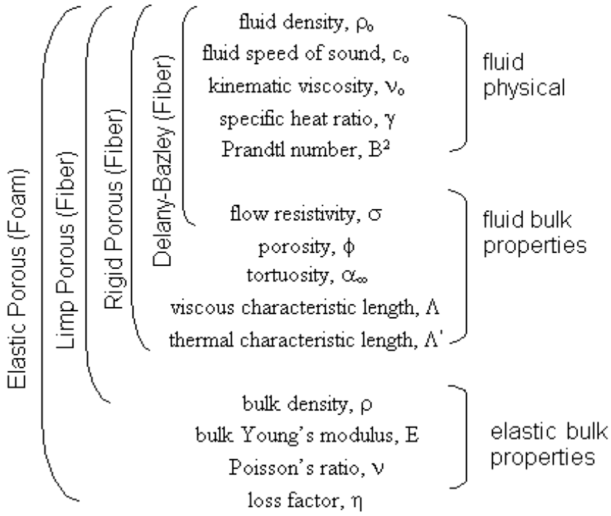

Fig.5 Bulk properties of the elastic porous material

In figure 5, the Bulk properties of the poroelastic material, is reported.
The elastic porous model is used for foam materials where the stiffness of the frame is important in vibro-acoustic response of the noise control material. The energy exchanges between structural energy and acoustical energy within a foam material typically provides much of the desired energy absorption. The full elastic porous model requires all the fluid properties and the elastic bulk properties.
For materials in which the frame waves do not carry a significant amount of energy, such as many fiber-based materials, the frame waves may be disregarded altogether and the noise control layer modeled with a single type of acoustic compression wave. In this case, the fiber model should be used to represent such a material. When the frame of the porous material can be considered limp, the fluid properties and the bulk density are required to predict the vibro-acoustical behavior. Similarly when the frame of the porous material can be considered rigid, only fluid related properties, are required.
In the examined case, a full elastic porous model, has been considered and its propertied are shown in table 2.

Bulk Properties	Polyurethane
Flow resistivity [N·s/m^4]	5000
Porosity	0,96
Totuosity	1,24
Viscous c.l. [m]	0,000105
Thermal c.l. [m]	0,00034
Density [kg/m³]	50
Young's Modulus [Pa]	46500
Poisson's Ratio	0,4
Damping Loss Factor	0,14

Table 2 Polyurethane's bulk properties

The air fluid physical properties (T=20°C), are reported in the table 3.

Fluid Properties	Air
Density [Kg/m³]	1,20
Fluid speed of sound [m/s]	343,5
Kinematic Viscosity [m^2/s]	$1{,}24\times10^{-6}$
Specific heat ratio	1,4
Prandlt number	0,713

Table Air's Physical properties

5.3 SEA Model

Energy based methods are often applied to characterize the vibration behaviour of structures in the high frequency domain. The method consists of decomposing a complete system into subsystems in order to estimate their total mean energy, averaged in time and space. This method is based on the energy equilibrium equation: the sum of the dissipated power in a given subsystem and the

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power exchanged with the other coupled subsystems is equal to the power supplied by the external loads to this subsystem. An important issue is the definition of a proper SEA model. The used numerical FEM/SEA methodology, is reported in figure 5.

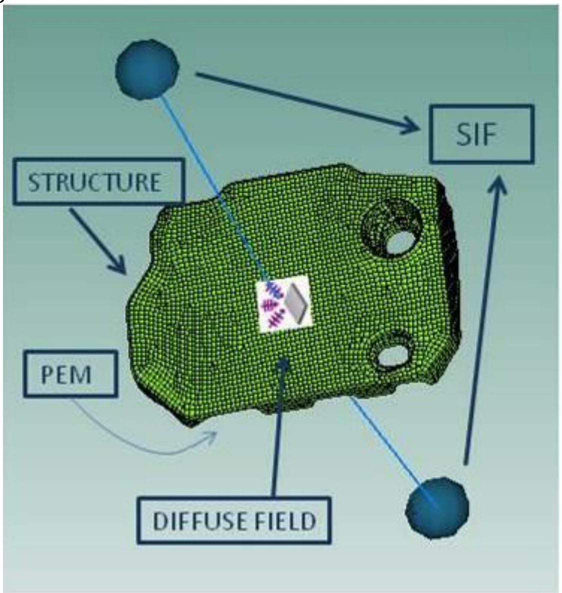

Fig.5 Global numerical implementation

In the Hybrid FE-SEA air-borne TL prediction model (fig. 5), the whole system is composed by the plastic structure (FE subsystem), PEM (Poro Elastic Material) and SIF (SEA Semi Infinite Fluid).
A semi-infinite fluid represents an unbounded exterior acoustic space. The acoustic waves radiated by a subsystem connected to a semi-infinite are not reflected back on the subsystem. The two semi-infinite fluid simulate reverberant and anechoic chamber across the structural system to calculate the Transmission Loss of the system.

A semi-infinite fluid is characterized by the following attributes.

- A fluid through which the acoustic wave propagates.
- A number of radiation loss factors describing the rate at which energy is transferred from each connected SEA subsystem.
- A number of complex impedances that the fluid presents to the modes of each connected FE structural.

6 TL Calculation and Result

In order to calculate TL in the SEA subsystem, an excitation incident power source, has been included. Then, a DAF (Diffuse Acoustic Field) representing the excitation source (100.000 N/m²) has been applied over the surface area (Figure 6).

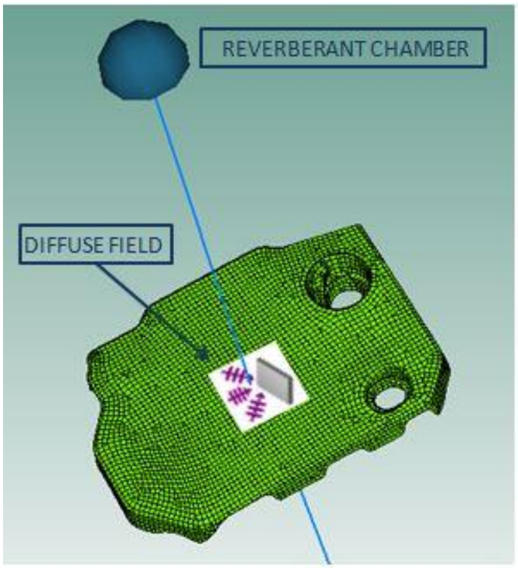

Fig. 6 Reverberant chamber and Diffuse Field

VA ONE® allows to calculate the equivalent in situ Transmission Loss following a rigorous condensation formulation. The implementation to condense a subsystem into equivalent in-situ TL can be so sketched (figure 7):

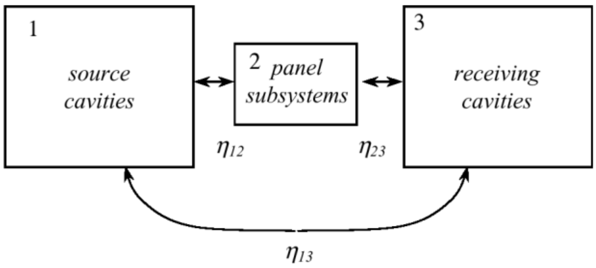

Fig. 7 Sketch of the TL implementation

Where η represent the coupling Loss Factor among several subsystems, so it is an indication of the Energy Loss from reverberant chamber to anechoic chamber, through the system.
The formulation from which is carried out TL is given by:

$$TL = 10Log_{10}\left(\frac{A_c \cdot \omega}{8\pi^2 \cdot c^2 \cdot n \cdot \eta_{tot}}\right) \quad (2)$$

where A_c is the excitation area, n is the modal density and c the sound velocity.
The coupling Loss Factor η is calculated considering the SEA equation 2 in the three subsystems: reverberant and anechoic chamber and structural system.

$$\omega \cdot \begin{bmatrix} h_{11} & h_{12} & h_{13} \\ h_{21} & h_{22} & h_{23} \\ h_{31} & h_{32} & h_{33} \end{bmatrix} \cdot \begin{bmatrix} E_1 \\ E_2 \\ E_3 \end{bmatrix} = \begin{bmatrix} P_{in,1} \\ P_{in,2} \\ P_{in,3} \end{bmatrix} \quad (3)$$

$$\text{where } h_{i,j} = \begin{cases} \eta_i + \sum_{k \neq i} \eta_{i,k} & i = j \\ -\eta_{j,i} & i \neq j \end{cases} \quad (4)$$

where ω is the center frequency of the band (rad/s), E is the cavity subsystem energy and P is the sound power. Ideally, we would like to reduce or condense the panel subsystems from the model. Assuming the panel is not directly excited, the reduced equations are given by:

$$\omega \begin{bmatrix} h_{11} - h_{12} h_{22}^{-1} h_{21} & h_{13} - h_{12} h_{22}^{-1} h_{23} \\ h_{31} - h_{32} h_{22}^{-1} h_{21} & h_{33} - h_{32} h_{22} h_{23} \end{bmatrix} \cdot \begin{bmatrix} E_1 \\ E_3 \end{bmatrix} \cdot \begin{bmatrix} P_{in,1} \\ P_{in,3} \end{bmatrix}$$
(5)

Comparing equation 3 with equation 5, the effective coupling loss factors between the source and receiving chamber in the reduced model are given by:

$$\eta_{rs} = -\left(h_{13} - h_{12} h_{22}^{-1} h_{23}\right) \quad (6)$$

$$\eta_{sr} = -\left(h_{31} - h_{32} h_{22}^{-1} h_{21}\right) \quad (7)$$

where the subscript rs and sr are the coupling Loss Factor from receiving chamber towards source chamber and vice versa, respectively.
Then, the coupling loss factor associated with a user defined transmission loss (TL) between two SEA SIF is given by:

$$\eta_{ij} = \frac{A_c \cdot \omega}{8\pi^2 c_i^2 n_i 10^{TL/10}} \quad (8)$$

Once the implementation of the whole system is completed, a frequency analysis up to 1000 Hz, is then performed. In figure 8 the transmission loss parameter express in dB versus frequency range, is presented.
By looking at the TL curve, some considerations are needed. In the low frequency range (25-200 Hz) the structure behaviour in terms of its dominant natural frequencies, is prevailing with a mean value of about 20 dB. Whilst, at higher frequency, the TL trend is influenced from the mass low increasing the TL value up to 28 dB.

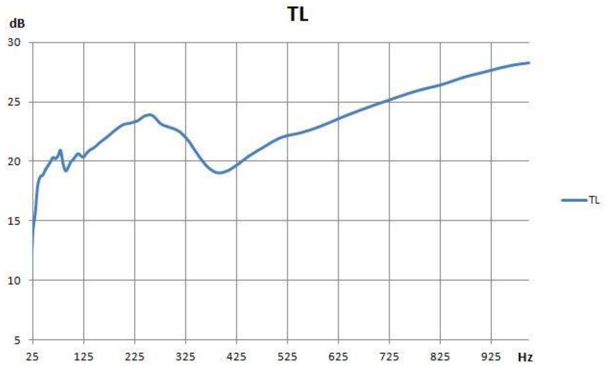

Fig. 8 Transmission Loss (dB) spectrum vs. frequency

7 Conclusion

In this work the acoustic performance of an engine cover, has been evaluated in terms of Transmission Loss that represent the most relevant acoustic global parameter for the acoustic behaviour assessment of soundproofing materials.
This work represent a preliminary study for further planned investigation that will involve the acoustic response under engine operative condition, the comparison between numerical and experimental results, as well as the implementation of innovative materials and different geometrical configurations to improve the engine cover noise reduction.

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