Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 82 (2015) 67 – 74

Energy Procedia

ATI 2015 - 70th Conference of the ATI Engineering Association

Experimental and numerical validation of an automotive subsystem through the employment of FEM/BEM approaches

Daniela Siano[a]*, Massimo Viscardi[b], Roberto Aiello[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*
[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 numerical analysis on an internal combustion engine cover is carried out. A hybrid numerical model has been used to evaluate the acoustic performance of engine cover. The Hybrid model is composed of two numerical models: FE (Finite Element) and BEM (Boundary Element Method) models. The FE model patterns structural system, whilst BEM model patterns fluid environment. The main aim of the present work is to characterize, through the employment of numerical simulation, the acoustic performance of the engine cover by using the NR (Noise Reduction) parameter. The numerical frequency response analysis has been implemented using acoustic impedance experimental data, measured for four different thicknesses of sound proof material forming the cover system. The spectrum of acoustic impedance has been evaluated in near field with a sophisticated probe, pressure-velocity probe (p-v probe).
The analysis of the obtained results have highlighted the weaknesses as regards sound attenuation of the cover system and possible further improvements of cover system acoustic performance.

Peer-review under responsibility of the Scientific Committee of ATI 2015

Keywords: Engine noise, Engine cover, FEM, BEM, Noise Reduction, p-v probe

1. Introduction

Nowadays, 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

* D. Siano:. Tel.: +39-081-7177108; fax: +39-081-2396097
E-mail address: d.siano@ im.cnr.it.

Peer-review under responsibility of the Scientific Committee of ATI 2015
doi:10.1016/j.egypro.2015.11.884

overexposure to noise and vibration can provoke physics and psychological problems to the human health.

The most important noise source, within vehicle, is 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 sound 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 aim of the present work is to perform a numerical acoustic properties evaluation of the engine cover by using the hybrid FEM/BEM (Finite Element Model/ Boundary Element Method) approach, in which experimental data of acoustic impedance have been loaded to simulate the acoustic performance of sound proof material. The acoustic impedance has been carried out with a sophisticated probe, pressure-velocity probe (p-v probe). In the following sections the experimental campaign and implementation of the numerical hybrid model will be presented.

2. Near Field Sound Measurement

Current standard techniques to measure the acoustic material impedance have limitations. The Kundt's method requires a sample to be cut out and put in a fully reflective tube. Many materials including some porous materials are not locally reacting, meaning the impedance depends on the angle of incidence. With the Kundt's tube it is only possible to measure at perpendicular sound incidence. The walls of the tube can influence the behavior especially if the material has a high flow resistivity like some foams or multi-layer materials. Some samples cannot be cut and leakage effects are observed when the tube is put highly porous surfaces (e.g. road asphalt or some aircraft engine liners). With a reverberant room it is possible to measure the diffuse absorption but it requires large and expensive facilities and samples of several square meters. Nowadays, an effective alternative to these current standard techniques is the surface impedance technique, an in situ method to determine the absorption coefficient of acoustic material taking into account not only the first layer of materials but the whole damping structure. Specifically, the acoustic impedance is measured close to the acoustic material and the absorption coefficient is derived from that. The method allows the absorption coefficient to be measured broad banded and also under an angle. This technique makes use of the so-called p-v probe [1], the combination of a pressure microphone and a particle velocity transducer which allows the direct calculation of the acoustic impedance (Z) from the ratio of both magnitudes, pressure (p) and particle velocity (v):

$$Z = \frac{p}{v} \text{ [rayl]} \quad (1)$$

The absorption coefficient (α) is hence determined from the complex reflection coefficient (r), which is strictly correlated to the acoustic impedance (Z):

$$\alpha = 1 - |r|^2 \quad (2)$$

where

$$r = \frac{p_r}{p_i} \quad (3)$$

being p_r and p_i respectively the reflected acoustic pressure and the incident pressure on the material.

The p-v in situ impedance method is used successfully on relatively small samples (> 0.1 m^2) under reverberant conditions (e.g. a car interior) [2]. The small source-sample and probe-sample distance are the main reasons for the relative small sample size requirement and the low influence to background noise and reflections.

In this work Microflown Technologies impedance setup has been employed for measuring the sound absorption properties of the poro-elastic material present below the engine cover lower surface.

Fig. 1 reports the experimental setup with the appropriate instrumentation required for measurements, consisting of an intensity p-v mini probe and a sound source held by an impedance gun, MFDAQ-2 acquisition system, and a laptop equipped with Microflown Impedance software.

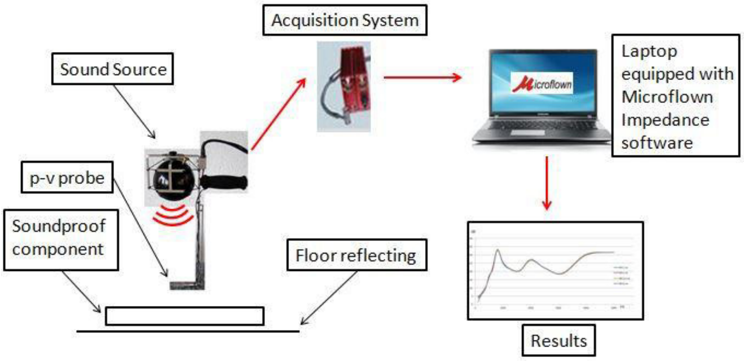
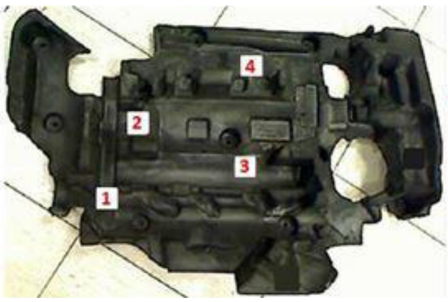


Fig. 1 In situ absorption experimental setup

During the test the loudspeaker located at 23cm from the probe generates a continuous noise signal towards the sample resting on a completely reflecting floor. The sound pressure and acoustic particle velocity are simultaneously measured at the same position right on the surface of the material. The measured signals are sent to the acquisition system connected with the laptop. Finally parameters such as absorption coefficient, intensity and acoustic impedance can be finally extracted by means of Impedance Software.



Measurement point	Soundproof material thickness [mm]
1	20
2	80
3	40
4	55

Fig. 2 The two measurement points for the acoustic characterization of the engine cover poro-elastic material

Table 1: Thicknesses of the material at the measurement points.

Fig. 2 shows a picture of the engine cover poro-elastic material, where the four measurement points chosen for the acoustic material characterization, are indicated. These latter correspond to different material thicknesses, as reported in Table 1.

The sound absorption measurements have been carried out the loudspeaker emitting a white noise signal and positioning the p-v probe very close to the measured surface, in particular at a distance of 5 mm, in order to minimize the influence of background noise and reflections from other objects, as well as avoid that the sound reflected from the sample becomes weaker compared with the incoming sound. The acoustic absorption of porous material is caused by the conversion into heat of the mechanical energy carried out by incident wave through friction phenomena inside the micro cavities. The acoustic incident sound makes the air inside the holes oscillate, dissipating energy because of the viscous friction. For instance in Fig. 3 the measured absorption coefficients, for two different material thicknesses (2 and 8 cm), are reported in the frequency range 100-4000 Hz in 1/3 octave frequency bands.

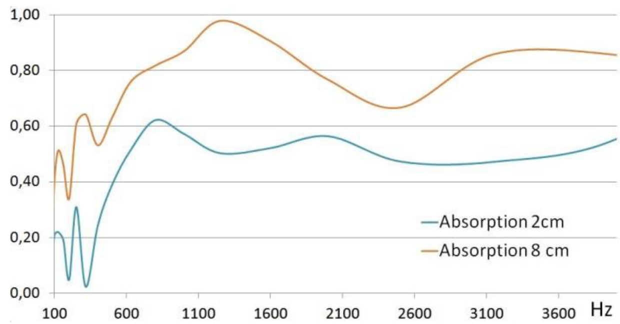


Fig. 3 Absorption coefficient for the two thicknesses of soundproof material.

It is possible to notice that the curves start with a very low value of absorption at low frequencies but they become higher as frequencies increase. As expected the absorption coefficient of 8 cm thick soundproof material assumes higher values than the 2 cm thick one in all the considered frequency range.

3. Noise Reduction Numerical Hybrid Model

Among the numerical models, hybrid model is spreading more and more common [3]. The hybrid model allows to take advantage of points of force of each numerical model used. The connections among several subsystems allow to carry out frequency analysis at mid and high frequencies.

Before building the numerical model, the CAD geometry has been simplified [4]. Those parts not necessary for the numerical simulation, have been removed. For this aim, a powerful software, of the ESI Group^{TM}, Virtual Performance Solution^{TM} (VPS) [5] has been used. To realize the whole numerical model, it has been used a commercial software of ESI GROUP^{TM}, VA ONE^{TM} [6].

The target of the numerical model has been to simulate a airborne transmission of a vibro-acoustic system in order to calculate the Noise Reduction which is one of the most important parameters for the investigation of sound attenuation, together with TL (Transmission Loss) and IL (Insertion Loss).

The numerical model is composed of two models: FEM and BEM model. FE model patterns the structure. Whilst the BEM model patterns the fluid environment which surround the cover system.

It is well known that, when a sound source hits a structural panel, part of sound source incident is reflected and a part is transmitted. Noise Reduction (4) of a generic system is defined as the difference between the Sound Pressure Level incident and the Sound Pressure Level transmitted throughout system.

$$NR = SPL_i - SPL_t \tag{4}$$

Sound Pressure Level is defined as [7]:

$$SPL = 20Log_{10}\left(\frac{P_{mis}}{P_{ref}}\right) \tag{5}$$

where P_{mis} is the pressure measured and P_{ref} is the reference pressure which equals $2 \cdot 10^5$ Pa

The core of the implementation of a numerical model in VA ONE™ is based on the definition of subsystems which represent the different numerical methods. The numerical model developed is based on the interaction of two subsystems which represent FEM and BEM methods. In the next paragraphs a brief discussion of the used methodologies, will be presented [8].

4. FEM

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 used mesh for FE analysis is reported in Fig. 4.

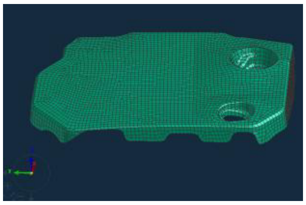


Fig. 4 Cover's mesh

As previously said, the mesh has been obtained through the use of Virtual Performance Solution™ in particular Visual Mesh™ section.

Through mesh, a modal analysis has been computed in free-free boundary condition. The material properties are referred to Nylon PA 6.6 [4].

The modal analysis has been performed in the frequency range 100-4000 Hz. In Fig. 5 (a,b,c,d) first four mode shapes, are depicted.

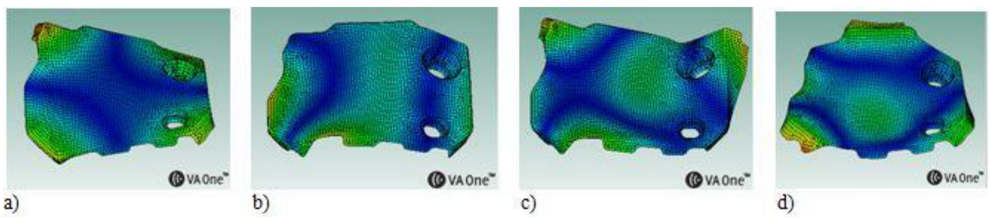


Fig. 5 a)Mode 1 @ 13,24 Hz b) Mode 2 @ 22,95 Hz c) Mode 3 @ 33,6 Hz d) Mode 4 @ 40,55 Hz

By viewing the above reported pictures, it is possible to note that the first mode shapes present the same behavior of a typical panel mode shapes [9].

5. BEM Model

A BEM fluid, which represents an exterior acoustic fluid subsystem, has been inserted in the numerical model. A BEM fluid subsystem has been used to model the radiation, scattering, transmission and response of an acoustic fluid in contact with the FE structural.

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 FEM on to the surface of the boundary element mesh.

6. NR Calculation and Result

The investigation procedure is clearly reported in the following Fig. 6a).

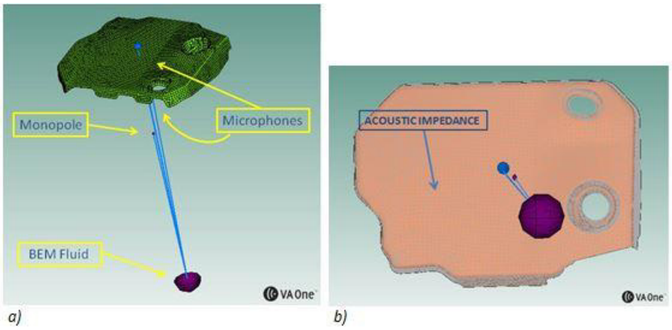


Fig. 6 a) Global numerical implementation b) acoustic impedance underside all the surface cover's skin

In accordance with literature the calculation of the NR has to be realized on modal base. In order to do so the result of the modal analysis of the cover skin has been imported. Fig. 6a) shows the numerical implementation, in which a monopole, immersed in the BEM fluid, functions as a sound source emitting 1 Watt of acoustic power. The monopole is placed 20 cm below the cover system, that is on the side of the sound proof material in correspondence with the center of cover.

Then, two pressure sensors function as the receivers; they are placed upstream and downstream of the cover, 3 cm from cover skin representing the structural component.

In Fig. 6b) the cover is overturned and the specific acoustic impedance is highlighted in orange color. The experimental data of specific acoustic impedance for the four different thicknesses (2,4,5.5,8 cm), have been used to simulate the soundproof characteristic of the poro-elastic material. An area isolator, which allows to model locally reacting acoustic treatments, has been loaded in the VA ONE^{TM} environment, to distribute specific acoustic impedance underside the whole cover skin surface.

The two pressure sensors provided directly the sound pressure level in dB in all frequency range of interest. The difference between the two sound pressure level revealed, allows to evaluate NR. In Fig. 7 is possible to observe NR by using the equation (4) in 1/3 octave bands in frequency range up to 250 Hz for the four thicknesses analyzed. The frequency analysis has been realized at low frequencies because in this range the most important sources are multiples of the firing frequency.

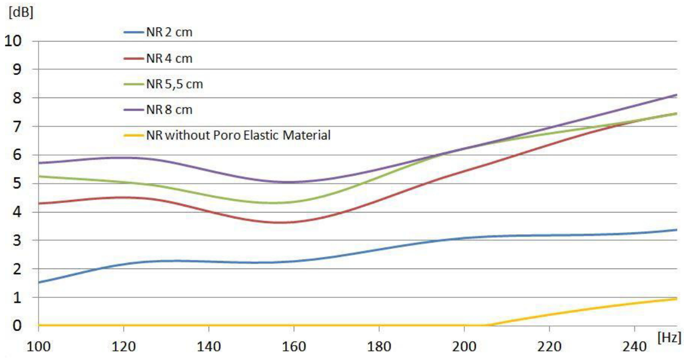


Fig. 7 Enlargement of the Noise Reduction (dB) spectrum vs. frequency

Furthermore in the comparisons of the Fig. 7 NR curve has been added only considering cover skin without sound proof material. As it is possible to observe sound attenuation of the bare cover skin is very low. NR cover skin does not exceeds 1 dB at low frequencies. This occurrence highlights how the contribution of sound attenuation is mainly given by poro-elastic material. Regarding results obtained employing poro-elastic material, as expected the grater the thickness the higher NR. NR related to 2 cm thick (blue line) sound proof material differs from the other trends of NR by 3-4 dB. These results highlighted that after 4 cm thickness there isn't a great improvement of NR, so there isn't a great improvement of the sound attenuation. The analysis have shown that the cover system equipped with this sound proof material, experimentally studied, in the investigated frequency range, doesn't exceed 5 dB.

7. Conclusion

In this work the acoustic performance in terms of NR of an engine cover, has been evaluated. A complex numerical model has been realized, and loaded with experimental results. Through results obtained has been possible to state that cover system shows a less performing behavior , in terms of sound attenuation at low frequencies.

Then, further developments are needed. Future developments could regard the possibility to implement an optimization process through the analysis of different materials and geometrical configurations. Possible alternative materials could be those characterized by absorption coefficient close to unity from low to high frequencies. On the other hand, instead of simply applying much higher thicknesses poro-elastic materials, another solution could consists in employing in addition to a poro-elastic material, also air chamber, functioning as resonators, in order to improve the sound attenuation properties at low frequency range.

References

[1] Microflown™ ebook_impedance

[2] Tijs, Emiel, and Erik Druyvesteyn. *"An intensity method for measuring absorption properties in situ."* Acta Acustica united with Acustica 98.2 (2012): 342-353.

[3] Siano D.,Viscardi M., Aiello R. "*Sensitivity analysis and correlation Experimental/Numerical FEM-BEM for Noise Reduction assessment of an engine beauty cover*", to be published in Energy Procedia 2015.

[4] Siano D., Viscardi M., Aiello R., "*An Hybrid FE/SEA Approach for Engine Cover Noise Assesment*" proceedings of WSEAS conference-Recent Researches in Mechanical and Transportation Systems, pp. 316-323, 27-29 July 2015, Salerno, Italy, ISBN: 978-1-61804-316-0

[5] Virtual Performance Solution^{TM} Reference Manual

[6] VA ONE^{TM} Manual Guide

[7] Leo L.Beranek, Tim J. Mellow, *Acoustics Sound Fields and Transducers*, Academic Press, 2012.

[8] Siano D., Viscardi M., Napolitano P., Panza M.A, "*Numerical and experimental acoustic performance investigations of a high-speed train composite sandwich panel"*, WSEAS Transactions on Applied and Theoretical Mechanics, Volume 9, Issue 1, 2014, Pages 290-300.

[9] Ricci, F., Viscardi, M., "*Dynamic behaviour of metallic and composite plates under in-plane loads"*, Proceedings of the International Modal Analysis Conference - IMAC, Volume 1, 2000, Pages 99-103.

Biography

D. Siano was born in Naples - 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. Referee for some International Journals and session organizers collaborating with SAE conferences. She is author of about 75 Scientific Papers published on International Journals and Conferences Proceedings and editor of two scientific books. She is tutor of several thesis and PhD thesis, as well.