Numerical modelling of the Vibro-Acoustic behavior of a closed vehicle with frequency dependent polymer materials

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ABSTRACT

In this paper, the vibro-acoustic response of a closed vehicle (Renault B95) is computed using a prototype tool based on the VPS software developed by ESI Group. This prototype tool takes into account the frequency dependence of the modulus of viscoelastic materials represented by the PVB of the windscreen and joints between the vehicle's body and glass surfaces. In a first step, the accelerations at different points of the windscreen and pressures inside the cabin are computed and compared to those obtained by NASTRAN software. In a second step, these results are compared with experiments obtained by Renault using accelerometers and those obtained by the University of Technology of Compiègne using a 3D laser vibrometer. Comparisons show an acceptable agreement between different numerical and experimental results.

Keywords: Vibro-acoustic, Viscoelastic, Vehicle   I-INCE Classification of Subjects Number(s): 47, 76

1. INTRODUCTION

This work is a part of the project REVA-VERMETAL. The principal aim of this project is to develop a prototype software tool that permits the prediction of the vibro-acoustic behavior of a closed vehicle with glass surfaces. The structure of this vehicle contains viscoelastic materials like the PVB core of the windscreen and the polymer joint between glass surfaces and vehicle's body. The mechanical properties of such kind of material are generally frequency dependent, and necessitate a complex representation of mechanical modules. The complex representation simplifies the analysis of the dynamic behavior of mechanical systems damped by viscoelastic materials. Using this representation in a finite element formulation, leads to systems with complex matrices. In this case, computing the frequency response of a structure using the classical direct method is a hard task especially for large size problems and large frequency band analysis. For this reason, several authors developed new methods to solve complex frequency dependent problems. For example, Poulin and Balmès (1) proposed a pseudo-modal representation using a real modal basis and adding dynamic corrections computed at higher end of the model frequency band. In reference (2), Balmès proposed a method to construct dynamically equivalent models with frequency independent matrices, from reduced models with non-linear frequency dependence. Other authors such as Abdoun et al. (3) proposed an asymptotic numerical method to compute the forced harmonic response of viscoelastic structures.

In the first section of this paper, the developed prototype software tool is presented. It uses an efficient simplified modal approach to compute the dynamic response of structures containing viscoelastic materials. The proposed approach is based on a modal reduction using a real modal basis and takes into account the frequency dependence of the viscoelastic material modulus.

In the second section, the vibro-acoustic response of a closed vehicle (Renault B95) is computed

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using the developed prototype software. Obtained results are compared in a first time to numerical results calculated by NASTRAN and in a second time to experimental results obtained by Renault using accelerometers and the University of Technology of Compiègne using a 3D laser vibrometer.

2. THE PROTOTYPE SOFTWARE TOOL

The prototype software tool called VPS-VERMETAL is based on the VPS (Virtual Performance Solution) software developed by ESI Group (4). VPS-VERMETAL incorporates a simplified modal method that permits to take into account the frequency dependence of viscoelastic materials which are the PVB core of the windscreen and polymer joints.

Since the complex Young modulus of the polymer material is frequency dependent, the equation of motion in the frequency domain can be written as following:

$$[K(\omega) - \omega^2 M]U = F \tag{1}$$

where K and M are respectively the stiffness and mass matrices of the structure, U is the displacement vector and F is the dynamic load vector.

There are two methods to solve the above system (1): the first one is a direct method (or nodal) where the displacement vector is calculated for each frequency step, while the second one is a modal method based on the modal expansion of the displacement field. The direct method has the advantage of being more precise, but it requires a lot of memory and computing time especially for large size problems.

The modal method uses a finite modal expansion of the nodal displacement vector and is hence approximate and less precise especially at high frequencies. However it has the advantage of being very fast because the system to be solved is reduced by a modal projection. The latter method is most suitable for solving large size problems encountered in industrial applications.

To take into account the frequency-dependence of the complex modulus of the viscoelastic material, the complex stiffness matrix of the structure is decomposed in the sum of two matrices K_0 and $\Delta K(\omega)$:

$$K(\omega) = K_0 + \Delta K(\omega) \tag{2}$$

where K_0 is the frequency-independent stiffness matrix calculated with a constant averaged modulus E_{20} of the viscoelastic material, $\Delta K(\omega)$ is the residual stiffness matrix calculated with a residual Young's modulus which the real part is equal to $\Delta E_2(\omega)$ as expressed in equation (3) :

$$\Delta E_2(\omega) = E_2{}'(\omega) - E_{20} \tag{3}$$

The modal basis is obtained by solving the following eigenvalue problem:

$$[K_0 - \omega^2 M]W = 0 \tag{4}$$

System can be solved using standard eigenvalue solvers, leading to a real modal basis composed by eigenvalues and eigenvectors $[\omega_i^2, W_i]$.

The solution of system (1) can be approximated by a modal decomposition:

$$U \approx \sum_{i=1}^{N_{modes}} \alpha_i W_i \tag{5}$$

where $\mathrm{N_{modes}}$ is the number of retained modes and α_i are generalized coordinates.

Considering the decomposition (2), the system (1) can be projected on the modal basis W whose columns are composed by the first N modes including rigid body modes. This projection is used to write the system in the following reduced form:

$$([\omega_i^2] + W^t \Delta K(\omega) W - \omega^2 [I])\alpha = W^t F \tag{6}$$

$[\omega_i^2]$ is a diagonal matrix with dimensions $\mathrm{N_{modes}} * \mathrm{N_{modes}}$ containing eigenvalues of problem (4), $[I]$ is the $\mathrm{N_{modes}} * \mathrm{N_{modes}}$ identity matrix.

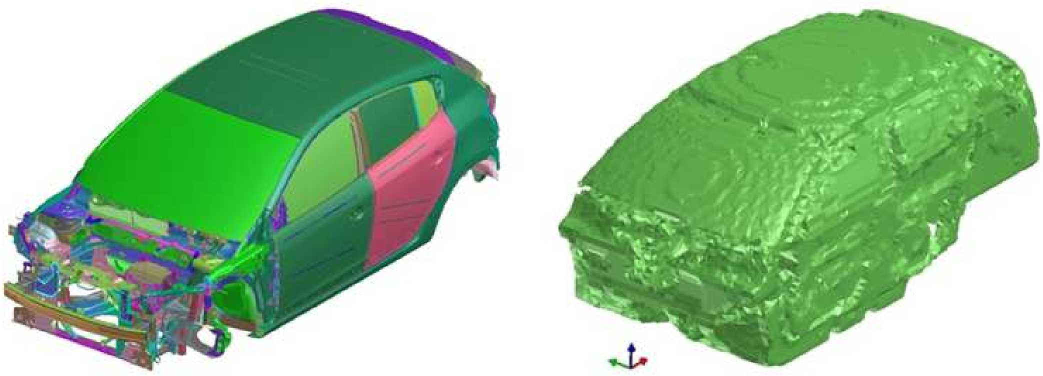
Since the polymer is an isotropic material, the dynamic stiffness matrix can be obtained by the product of the complex modulus that depends on the frequency with a matrix built using a unitary

Young's modulus. At each frequency step, the reduced system (6) is solved by updating the residual stiffness matrix. After determining the generalized coordinates α_i , the displacement field is reconstructed using the modal expansion (5).

This method was validated by Bouayed (5) in the case of a vehicle’s windscreen.

3. VIBRO-ACOUSTIC RESPONSE OF THE “RENAULT B95” VEHICLE

The numerical model of the B95 model contains two domains: the structure and the cavity as shown in Figure 1. The structure is composed of 1752735 nodes and 1559055 elements whereas the cavity is composed of 79241 nodes and 127872 SOLID elements.



(a) (b)

Figure 1 – Finite Element models of the structure (a) and the cavity (b) of the B95 vehicle

The laminated windscreen is modeled by SOLID elements in 3 layers as the polymer joint which represents the interface between the glass surface and the vehicle body (Figure 2). Modelling the windscreen with three layers allows the application of the simplified modal method, presented in the previous section, to take into account the frequency dependence of the modulus of the PVB core.

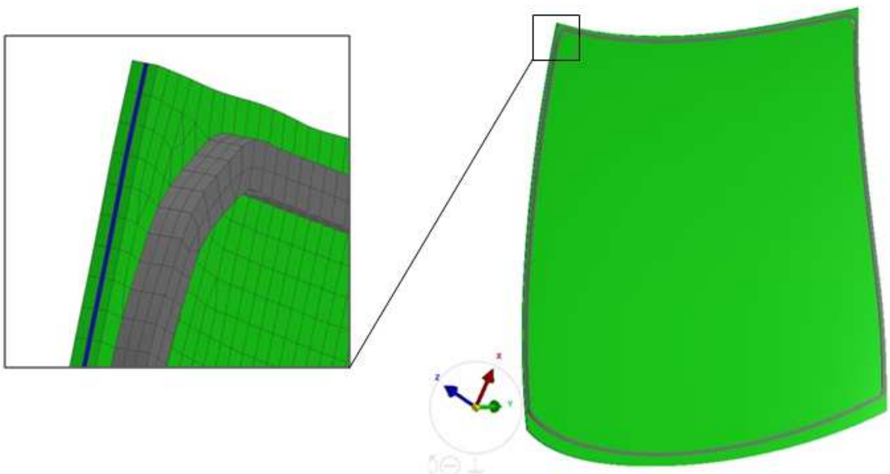
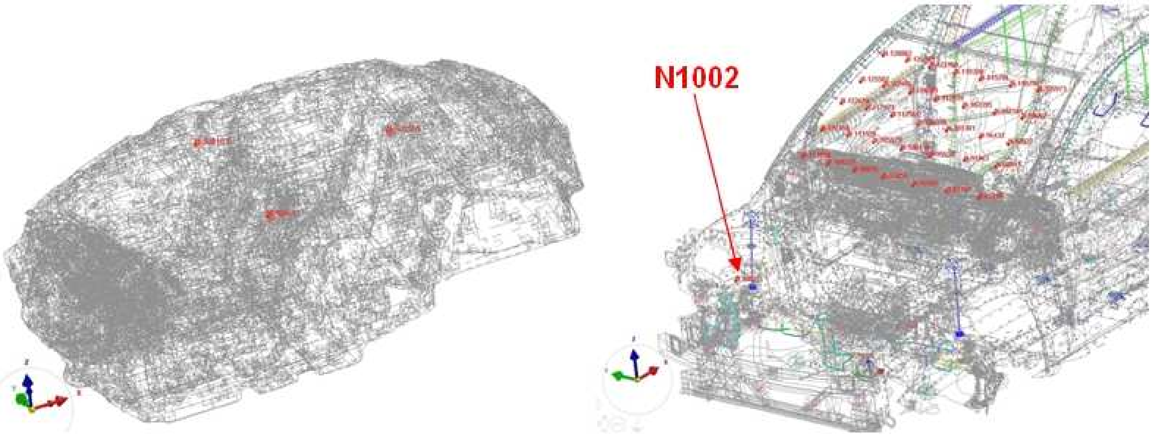


Figure 2 – 3D mesh of the windscreen and the polymer joint

The B95 is excited by a mechanical force at node 1002 on the Z direction as shown on Figure 3. The dynamic response is calculated at the level of different points on the windscreen. The acoustic pressure is computed at three points inside the cavity. The positions of the microphones correspond to the driver’s left ear, the passenger’s right ear and the rear passenger’s ear as shown in Figure 3.



(a) (b)

Figure 3 – Positions of microphones (a) and accelerometers (b)

The method used to compute the vibro-acoustic coupled response is a modal method that requires the computation of the structural and acoustic modal basis.

For a response until the frequency of 400 Hz, the structural modal basis is calculed until 600 Hz and containing 4926 modes including 9 residual modes. The acoustic modal basis contains 150 modes until 616 Hz.

The model of the structure is very huge and had a very big size modal basis which is not very accurate for numerical computations. For this reason, a reduced modal basis, smaller than the complete one, is used. It contains only eigenvectors corresponding to the nodes of the coupling surface between structure and cavity, the nodes of the windscreen and the polymer joints, the excitation nodes and response nodes.

3.1 Numerical validation

The coupled response of the B95 is computed with VPS (including the FEM RAYON solver (6)) using different coupling distances and a reduced modal basis until 600 Hz.

Structural and viscous damping matrices are taken into account and modulus of the PVB core and polymer joints are considered constant and frequency independent.

Figure 4 presents a comparison between dynamic responses computed with NASTRAN and VPS until 200 Hz. According to this figure, it can be seen that the coupling distance hadn't a remarkable influence on the results obtained by VPS and are in a good agreements with those obtained by NASTRAN excepting some differences observed locally.

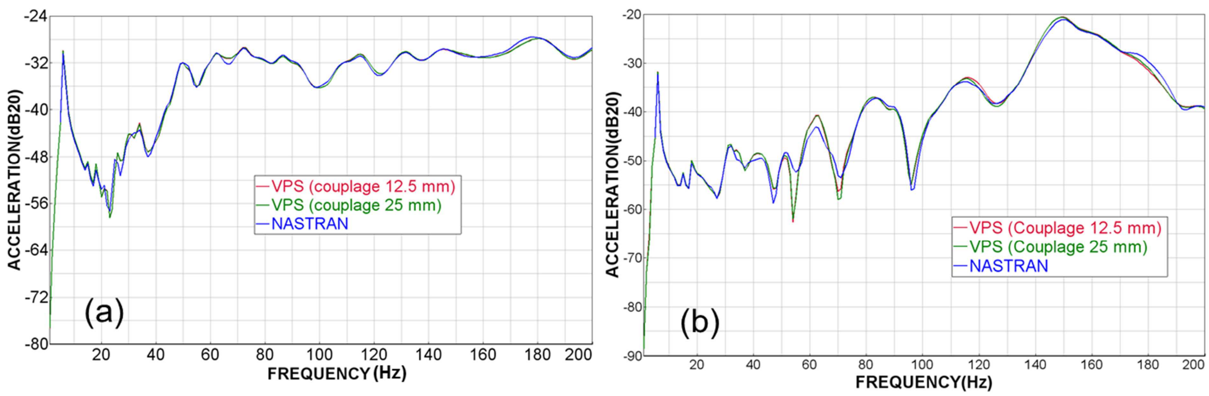


Figure 4 – Accelerations at excitation node 1002 (a) and the windscreen's node 87187 (b)

Figure 5 shows a comparison between acoustic responses of the vehicle obtained by VPS with different coupling distances and those obtained by NASTRAN at two different locations inside the cavity. Some differences can be observed between results of the two softwares. This difference has the same level as the difference observed on VPS results with different coupling distances.

Generally, we have a good accuracy between results obtained by NASTRAN and VPS for a response calculated until 200 Hz. The small difference observed locally can be related to the fact that the model is slightly modified when converted from NASTRAN to VPS format.

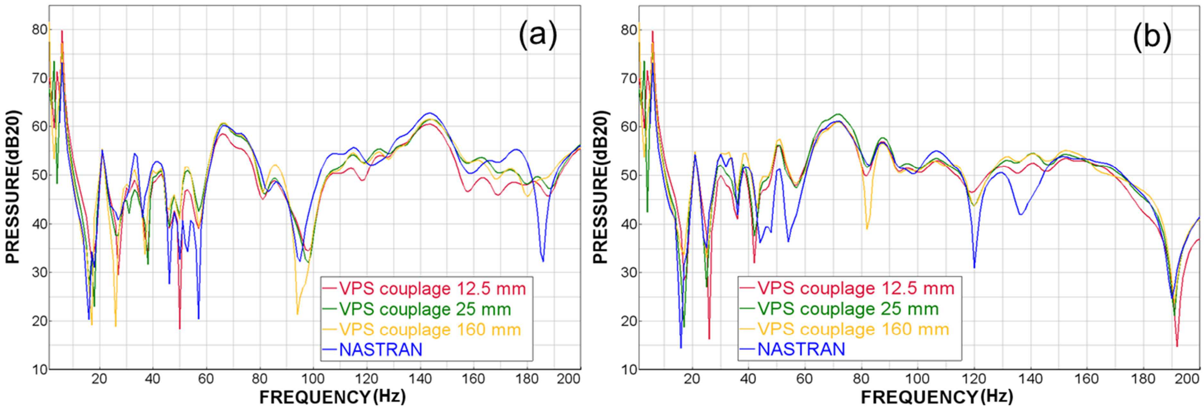
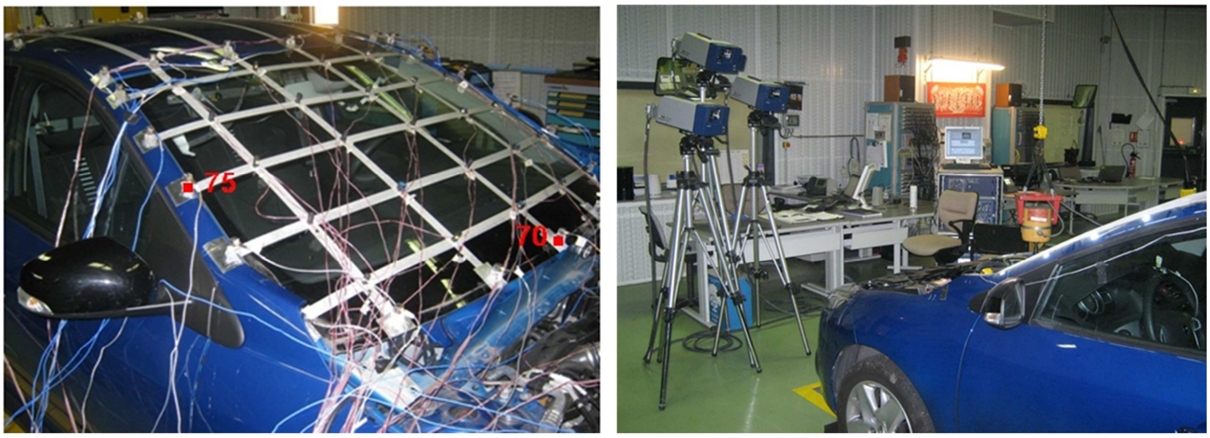


Figure 5 – Computed pressures at front passenger's right ear (a) and rear passenger's ear (b)

3.2 Experimental validation

In this section, numercial results obtained by the tool VPS-VERMETAL will be confronted to measurments condcuted on a B95 vehicle using a laser vibrometer (UTC) and accelerometres (Renault) (Figure 6). For the numercial model, the frequency dependence of the modulus of PVB core and polymer joints are taken into account according to WLF curves (7) shown in Figure 7 and Figure 8 respectively. WLF curves of the PVB are given by the glass manufacturer Saint-Gobain whereas those of the polymer joint are obtained by Roberval Laboratory (UTC) using a Dynamic Mechanical Analysis tester on samples given by Renault.



(a) (b)

Figure 6 – Experimental test - Accelerometers (a) and laser vibrometer (b)

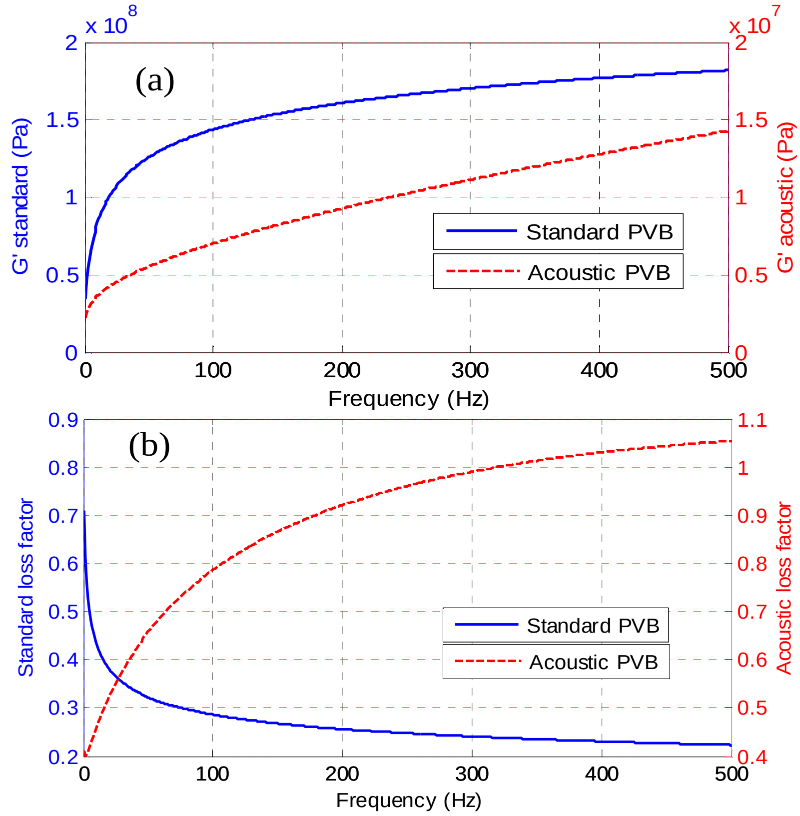


Figure 7 – WLF curves of the PVB

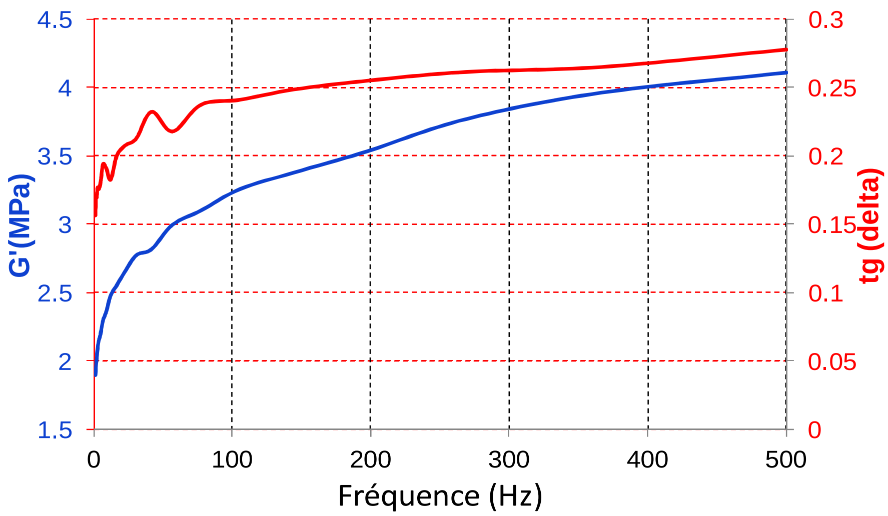


Figure 8 – WLF curves of the polymer joint

Using the WLF curves of the acoustic PVB, Figure 9 shows that until 220 Hz, the acceleration computed by this model at the excitation point is closest to measurments obtained by Renault than those calculated with a standard PVB. Beyond this frequency, the tendency is reversed and a difference of 2 to 3 dB is observed between results obtained by using the two types of PVB.

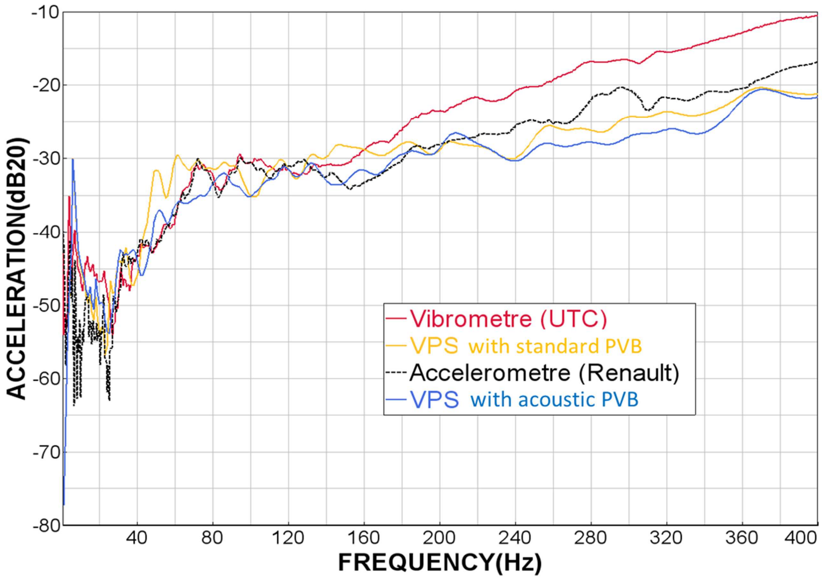


Figure 9 – Acceleration at the excitation point 1002

For the vibratory transfer on the windscreen, the comparison between numerical and experimental results will concern the quadratic mean of accelerations computed at 35 different points of the windscreen as shown in Figure 3.

In Figure 10, the dynamic responses computed by VPS using standard and acoustic windscreens are introduced. In the case of acoustic windscreen, two models was condcuted using a structural modal basis calculted with a constant Young's modulus equal to 42 MPa in the first model and 447 MPa in the second one. The frequency depedence of the PVB and the polymer joint is taken into accout when computing the vibroacoustic coupled response.

According to this figure, it can be seen that numerical responses using an acoustic windscreen are much more damped than those computed with a standard windscreen. This is related to the fact that the acoustic PVB is softer than the standard PVB as shown in Figure 7. In addition, the loss factor of the acoustic PVB, which is between 0.4 and 1.03, is higher than the loss factor of the standard PVB which decreases from 0.7 to 0.23 on the frequency range between 0 and 400 Hz. Generally, responses obtained by a standard windscreen are closest to measurments than those obtained by an acoustic windscreen. A big diffrence is observed between measurments and numercial results obtained by an acoustic windscreen. This diffrence can excceds 10 dB.

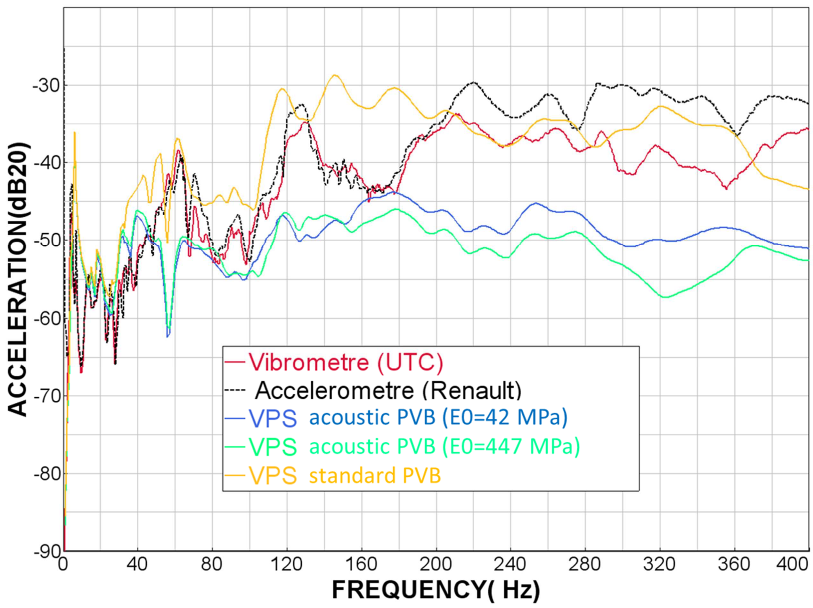


Figure 10 – Quadratic mean of responses at different points of the windscreen (along the normal)

For the noise transfer, numerical reponses considering the two types of windscreen with standard and acoustic PVB are compared to measurments obtained by Renault and UTC. In both models, the coupling distance between the structure and the cavity,   is 25 mm and WLF curves of the PVB and polymer joints are taken into account.

Measurments were conducted on a full trimmed vehicle whereas the numercial model contains only the closed vehicle body (structure) and the cavity.

Figure 11 presents a comparison between calculated and measured pressures at two points inside the cavity. According to this figure, it can be observed that the noise level obtained by an acoustic windscreen is lower than the noise obtained by a standard windscreen. The difference can exceed 20 dB at some frequencies.

In comparison with experiments, responses obtained by VPS-VERMETAL are in acceptable general accuracy except few differences at some frequencies that can be related to the fact that the real vehicle is full trimmed which is not the case of the numercial model.

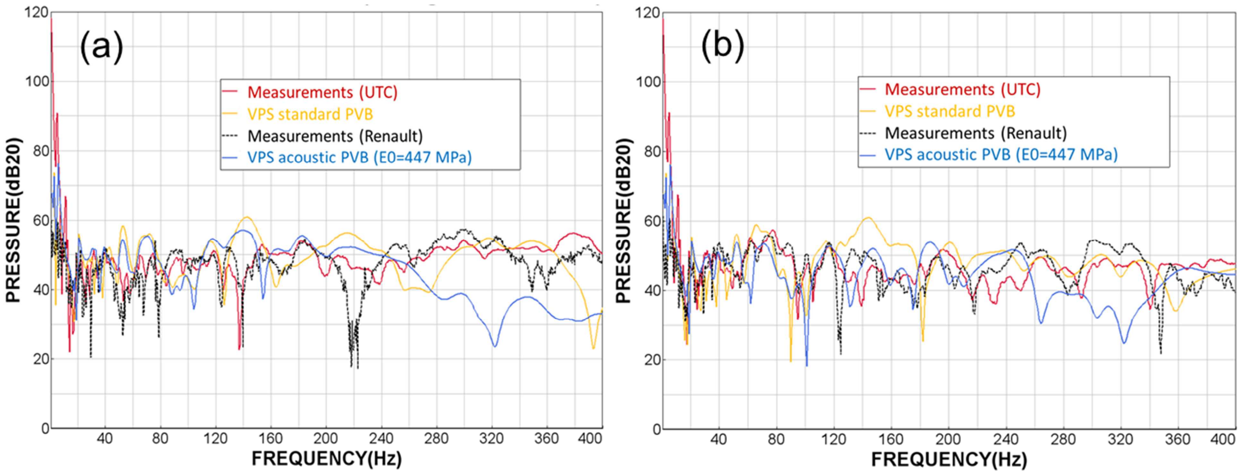


Figure 11 – Numerical vs experimental pressures at driver's left ear (a) and passenger's right ear (b)

4. CONCLUSIONS

In this paper, the vibro-acoustic response of a closed vehicle was calculated with the VPS-VERMETAL prototype software developed by ESI-Group. This software includes a simplified modal method to take into account the frequency dependence, according to WLF laws, of the modulus of viscoelastic materials. In the case of the studied B95 vehicle, viscoelastic components are the PVB core of the windscreen and polymer joints between the body and glass surfaces.

Vibro-acoustic responses obtained by the prototype software were compared in a first step with results obtained by NASTRAN using constant modulus for the PVB and polymer joints. In a second step, the frequency dependence of polymer materials was taken into account and the results of the VPS-VERMETAL code were compared with experimental measurements obtained by laser vibrometer (UTC) and accelerometers (Renault) and carried out on a complete car.

The comparison shows that results obtained by the protoype tool are in a good agreement with both mechanical and acoustic resultats obtained by NASTRAN.

According to numerical results, the noise level obtained by an acoustic windscreen is lower than the noise obtained by a standard windscreen. The difference can exceed 20 dB at some frequencies.

In comparison with experiments, responses obtained by VPS-VERMETAL, using a standard windscreen, are in acceptable general accuracy except few differences at some frequencies. The difference between the measurements and calculations can be explained by the fact that the numerical model does not take into account the internal trims (deck, floor, seats etc…) that are present in the physical vehicle (Mégane III).

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REFERENCES

1. Poulin A-S., Balmès E.,. Pseudo-modal representations of large models with viscoelastic behavior. International modal analysis confrerence. Santa Barbara CA, United States, February 2-5 1998.
2. Balmès E. Model reduction for systems with frequency dependent damping properties. International modal analysis conference. Orlando, USA, February 3-6 1997.
3. Abdoun F., Azrar L., Daya E-M., Pitier-Ferry M.,. Forced harmonic response of viscoelastic structures by an asymptotic numerical method. Computers and Structures. 2009, 87, pp. 91–100.
4. Virtual Performance Solution 2013. Solver Reference Manual.
5. Bouayed K., Hamdi M.A. Finite element analysis of the dynamic behavior of a laminated windscreen with frequency dependent viscoelastic core. Journal of the Acoustical Society of America. 2012, Vol. 132, 2, pp. 757-766.
6. Hamdi M.A., Mebarek L. User's guide for RAYON-Ideas Vibro-Acoustic. 1998.
7. Williams M.L., Landel R.F., Ferry J.D. The temperature dependence of relaxation mechanisms in amorphous polymers and other glass-forming liquids. Journal of the American Chemical Society. July 1955, Vol. 77, pp. 3701-3707.