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Finite Element Calculations of Structural-Acoustic Modes of Vehicle Interior for Simplified Models of Motorcars

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Abstract [299] The present paper describes the results of finite element analysis of structural vibration modes, interior acoustic modes, and structural-acoustic modes in some simplified models of road vehicles having different levels of complexity, in particular in the QUArter-Scale Interior Cavity Acoustic Rig (QUASICAR) developed in Loughborough University. All the analysis has been carried out using the original code that had been developed in Patran Command Language (PCL) specifically for the purpose of this research. Resonant frequencies and spatial distributions of structural and acoustic modes have been calculated initially separately and then taking into account structural-acoustic interaction. The results have been compared with the experimental data obtained for QUASICAR. The comparison has demonstrated good agreement between numerical calculations and experimental results. The developed approach is reliable and efficient, and it can be extended to more complex vehicle models, thus assisting in better understanding of vehicle interior noise.

1 INTRODUCTION

Vehicle interior noise is a very important issue for automotive industry [1-4]. The tendency to lighten up a car body structure leads to the reduction of its natural frequencies of vibration and to the rise of interior noise levels. On the other hand, passengers' comfort and market demands stimulate any annoying noise inside the vehicle compartment to be suppressed. These two contradictory trends encourage researchers to develop new efficient methods of analysis of vehicle interior noise that could be used on a design stage. As has been mentioned in [2], the main sources of vehicle interior noise are engine and transmission system, road excitation, and aerodynamic excitation. The resultant noise depends not only on the exciting forces, but also on vibration characteristics of the car body structure and on acoustic properties of the passenger compartment.

Because of the energy exchange between the air and the structure in a vehicle compartment, the dynamic behavior of each of these sub-systems is influenced by the other. In other words, the interaction, or coupling, between air and structure alters their dynamic characteristics, and this determines the complexity of vehicle interior noise analysis. Fluid-structure interaction has always been a major research topic in acoustics [5-8]. The existing analytical solutions for cavities with simple geometries provide a great opportunity for an explicit physical interpretation and understanding of fluid-structure interaction. However, analysis of irregular cavities, such as car compartments, still challenges researchers and requires new investigations. In this case the inability of deriving analytical solutions leads to alternative, either experimental or numerical approaches in treatment of the problem. In this regard, finite element analysis combined with experimental validation represents a very powerful tool, as can be seen in [1,9,10]. Studying fluid-structure interaction by finite element analysis enables many engineering problems to be solved and reveals new areas for further examination of the subject.

The use of simplified and reduced scale vehicle models for theoretical and experimental investigations of structure-borne interior noise has been described in [3,4]. Such models are useful for understanding the physics of the problem and for simulation of the main features of roal vehicles. In particular, the QAUSICAR (QUArter Scale Interior Cavity Acoustic Rig) has been designed in Loughborough University to replicate a 1/4 scale massively simplified model of a passenger car compartment and to verify the analytical approach developed in [3]. Investigations in [4] included separate experimental measurements of acoustic, structural, and structural-acoustic responses due to an external dynamic force imitating the effect of road irregularities.

The aim of the present paper is to present the results of finite element analysis of structural-acoustic phenomena in the above-mentioned reduced-scale model (QUASICAR) and its modifications and to compare the obtained numerical results with the analytical and experimental ones. The analysis reported in the present paper has been carried out using a new code MSC.NASTRAN-Acoustic that has been developed in Patran Command Language (PCL) specifically for the purposes of this research.

2 STRUCTURAL MODES

The first stage of the investigation included finite-element analysis of the basic structural element of QUASICAR which represents a single curved steel plate of 1.2 mm thickness simulating vehicle compartment. The above-mentioned curved plate was attached to massive wooden side walls of QUASICAR implementing simply supported boundary conditions. For more detailed information see [4]. The numerical structural analysis of QUASICAR included determination of the spatial patterns and natural frequencies of free vibrations of the structure. As a result of structural symmetry of the curved plate under consideration, all normal modes are divided into two groups: symmetric and anti-symmetric modes.

3 ACOUSTIC MODES

The acoustic modal characteristics of an arbitrary cavity can be obtained by solving Helmholtz equation. It is well known that normal modes of simple cavities, such as rectangular or cylindrical enclosures, can be derived analytically. However, for arbitrary cavities the only way of solving Helmholtz equation is by using numerical methods. In the FEM, a normal mode analysis of this problem can employ standard structural equation of motion but in this case mass and stiffness matrices are denoted as acoustic mass matrix $[M^a]$ and acoustic stiffness matrix $[K^a]$. More details are available in [10]. It is interesting to make a comparison between numerically derived natural frequencies of the irregular cavity and natural frequencies calculated using the well-known analytical formulae for a rectangular enclosure having the same volume and close linear dimensions: L_x, L_y, L_z (see [4]).

4 COMPARISON BETWEEN ANALYTICAL, NUMERICAL AND EXPERIMENTAL RESULTS

QUASICAR structure can be considered as a combination of simple structures: plates and shells. Table 1 shows natural frequencies of the simply supported rectangular plate (Columns 1, 2) and of the QUASICAR model (Columns 4, 5). The corresponding spatial patterns are shown on Figure 1.

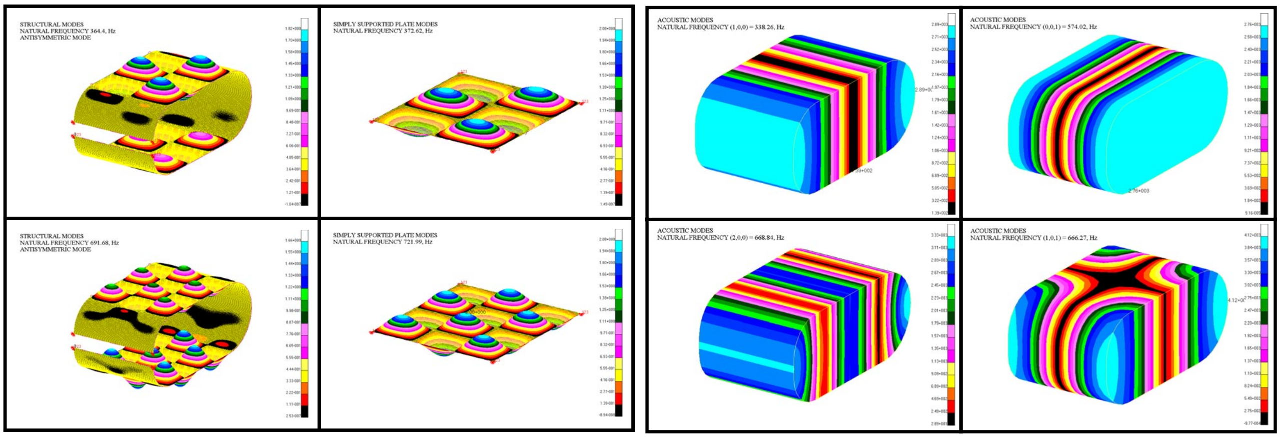

Figure 1: *Structural (left) and acoustic (right) normal modes of the QUASICAR model*

The two flat plates of the QUASICAR have the lowest stiffness and consequently the lowest fundamental frequency in this combination. The analysis of the individual sections shows that the first resonance peak of the two plates is at 67.035 Hz (67.039 Hz– for anti-symmetric mode), the half of a circular shell - 906.04 Hz, and the two quarters of a circular shell – 1271.40 Hz. Bearing in mind the low-frequency range of interest for this research (up to 1000 Hz - corresponding to 250 Hz for a real vehicle) and noticing that resonant frequencies of curved parts are above 900 Hz, it is reasonable to approximate the normal modes of QUASICAR by the normal modes of a simply supported rectangular plate having the dimensions of the QUASICAR flat sides (see Figure 1). Resonance frequencies of the curved plate (a coupled structure) and of the uncoupled flat plates agree well in the frequency range considered. These results demonstrate that in the frequency range below 900 Hz, the predominant influence of the flat plates makes it possible to approximate the modal characteristics of QUASICAR by those for a simply supported flat plate.

Simply supported plate, natural freq., Hz			FE struct. natural freq., Hz	Exp. struct. natural freq., Hz	Acoust. natural freq. of a rectang., Hz		FE acoust. natural freq., Hz	Exp. acoust. natural freq., Hz
1		2	3	4	5		6	7
Analytical		FE						
(1,1)	59.04	59.18	67.035	-	(1,0,0)	345.88	338.26	360.00
-	-	-	67.039	-	(0,1,0)	571.86	574.02	582.00
(1,3)	276.36	277.12	264.51	265.00	(1,1,1)	957.90	985.49	-
-	-	-	264.52	270.00	(2,0,1)	974.39	1017.10	980.00
(1,4)	466.51	467.81	415.13	451.00	(3,1,1)	1369.17	1318.20	-
-	-	-	415.13	-	(0,0,2)	1372.46	1340.90	-
(3,3)	531.40	531.95	521.45	523.00	(1,2,1)	1377.91	1366.60	-
-	-	-	521.45	-	(4,0,0)	1383.53	1403.20	1407.00
(3,4)	721.55	721.99	691.67	700.00	(5,0,1)	1860.58	1593.30	-
-	-	-	691.68	-	(1,3,1)	1879.82	1626.10	1872.00

Table 1: *Natural frequencies of QUASICAR and of a rectangular plate*

The analysis of the experimental data (Table 1, Column 4) shows some disagreement with the numerical results (Table 1, Column 3). First of all, it was difficult to excite all natural frequencies.

The experimental tests covered a frequency spectrum from 231 to 700 Hz. In the low-frequency range, between 230 and 350 Hz, it can be noticed that there is a large number of natural frequencies that do not correspond to those obtained from the numerical and analytical calculations. This can be explained by the presence of symmetric and anti-symmetric natural modes which correspond to different but relatively close natural frequencies. Note that these normal modes were excited by a non-symmetric load (one shaker acting on the bottom plate of QUASICAR). In this way the experimental tests could not simulate the symmetric and anti-symmetric modes in a proper way. In the region between 350 and 700 Hz the measured natural frequencies correspond to one of the groups: symmetric or anti-symmetric natural modes. As a reason for disagreement between experimental and numerical data in the whole range of frequencies one can point out also the differences between the FE model and the real test rig, e.g. the unaccounted influence of masses of the accelerometers, imperfections in the boundary conditions, etc. In spite of these disagreements, the experimental analysis validates to some extent the numerical and analytical results and brings new ideas for further improvements of the experimental tests.

The analysis of the acoustic data (Table 1, Columns 5, 6, 7) shows a good agreement between analytical, numerical and experimental results in the range up to 1000 Hz. Above this range the precision of the numerically determined natural frequencies is deteriorated, which is due to a smaller number of finite elements per wavelength. The differences between measured and numerically calculated acoustic natural frequencies may be partly explained by the unaccounted rectangular gap in the left curved part of QUASICAR.

5 MODIFIED MODELS OF QUASICAR

The initial QUASICAR model has been designed as a massively simplified model of a road vehicle. One of the reasons for such a simplification was the possibility to estimate the interior sound pressure in QUASICAR by approximate analytical formulae. The modified models of QUASICAR have been developed and analysed by means of numerical techniques to eliminate some weaknesses of the original model and to simulate more accurately the main characteristics of road vehicles. Two modified models have been considered: the first (model ***M1***) has a different thickness of the bottom plate, and the second (model ***M2***) employs different boundary conditions.

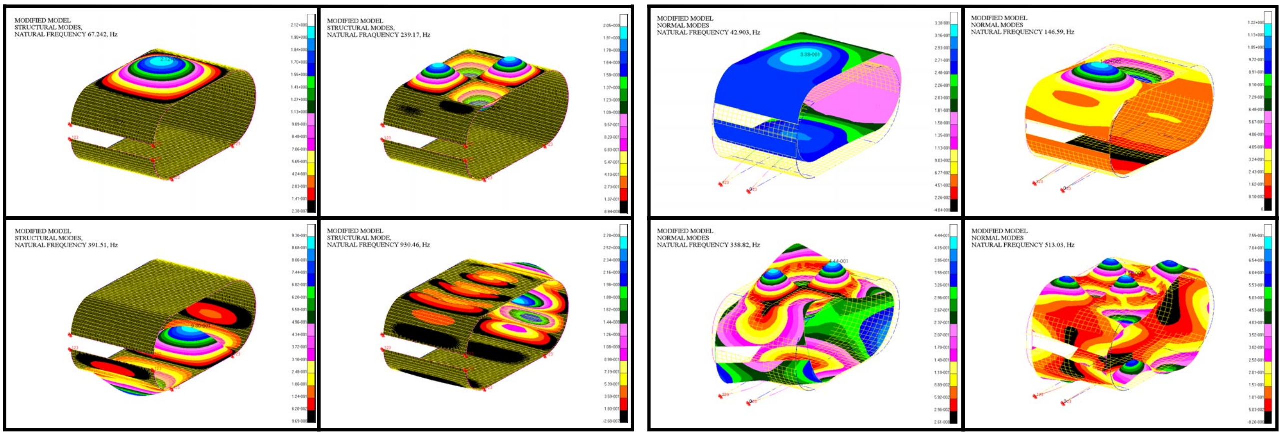

Figure 2: *Structural normal modes of the modified models (**M1**-left and **M2**-right)*

The geometry and the boundary conditions of the model ***M1*** are the same as the original QUASICAR model. The only difference is the dimensions of the bottom plate that was modeled as having 6.0 mm thickness. In this way the symmetry in respect of the bottom and top parts of QUASICAR has been broken, which corresponds more realistically to the case of real road vehicles. Comparing some of the normal modes (Figure 2) and natural frequencies (Table 2, Column 1 and 3) of the modified model ***M1*** with those of QUASICAR, one can notice some interesting facts. First of all, the predominant normal modes belong to one of the main parts of the model: the bottom plate, the top plate or the curved part, and only in certain modes, in the considered frequency range from 0 to 1000 Hz, all three panels are involved. The distinct normal modes are associated with the different stiffness of the panels, while their geometrical forms remain the same. Thereby the simplification of complex structures is possible on the base of the material and geometrical characteristics of their main parts. Secondly, in spite of the change of the model (increase in weight), the fundamental natural frequencies remain the same. They are defined by the top plate, which has the lowest stiffness and was unchanged after the modification. Suppressing the participation of the bottom plate in the formation of normal modes is another important feature demonstrated here. In the frequency range between 0 to 1000 Hz the bottom plate takes part only in the five normal modes, whereas in QUASICAR model the bottom plate plays the same role as the top one. The Modified model ***M1,*** which is closer to real road vehicles, demonstrates some useful ideas for controlling the vibration behavior of the panels and in the same time keeps the calculations simple.

The changes incorporated into the model ***M2*** have been determined with a view of a proper representation of a typical car body construction. The simply supported boundary conditions of QUASICAR model were replaced by beams with a circular cross section of radius $R = 10$ mm. The bottom plate was stiffened by means of two beams in transverse and longitudinal directions, which represents the platform of a car. The only boundary conditions were imposed at the ends of the longitudinal beams: the constraints in X, Y and Z directions were applied at the relevant nodes. These simulated higher stiffness of the bottom part and of the edges of a car body, as well as a fully stiff suspension. Some of the results of the normal modes analysis are shown in Fig. 2 and Table 2, Column 2. The first natural frequencies correspond to displacements of the modified model which are due to longitudinal beams (the lowest stiffness in the model). The first normal mode, which corresponds to the fundamental frequency of QUASICAR model, appears at a higher frequency, 93.019 Hz. The analysis of this model outlines the complexity of structural simplification of a car body. From Figure 2 it can be seen that the spatial patterns of vibrations are a mixture of spatial patterns due to a simply supported rectangular plate and spatial patterns caused by a greater degree of freedom of the model. However, the main location of structural vibrations remains the same - the panel with the lowest stiffness, namely the top plate of the modified model ***M2***.

Model ***M1***, structural natural frequencies, Hz	Model ***M2***, structural natural frequencies, Hz	QUASICAR, structural natural frequencies, Hz	Structural-acoustic interaction models, natural frequencies, Hz		QUASICAR, acoustic natural frequencies, Hz	
1	2	3	4	5	6	
67.242	15.310	67.035 67.039	67.909 76.488	72.313		
265.50	138.92	315.33 315.33	311.98 312.25	270.23		
			338.59	338.09	(1,0,0)	338.26
393.81	174.53	394.69 394.69	386.57 386.83	396.25		
416.25	192.33	415.13 415.13	421.29 421.90	421.58		

Table 2: *Natural frequencies of modified models and structural-acoustic coupling*

6 STRUCTURAL-ACOUSTIC COUPLING IN QUASICAR AND MODIFIED MODEL *M1*

Interaction, or coupling, between an enclosed fluid (air) and a structure means their mutual influence on the dynamic behavior of each other. The fluid acts via its pressure on the structural surface, and in the same time it is influenced by the normal displacements of the structure [11,12]. Thus, fluid pressure on the surface is considered as a disturbing force in the governing equations of motion of the structure, and the normal accelerations of the structural surface enter into the Helmholtz equation via 'flexible wall' boundary conditions. Coupling of these equations leads to a single governing matrix equation for the whole system 'structure-fluid':

$$\begin{bmatrix} [M^s] & [0] \\ [M^{as}] & [M^a] \end{bmatrix} \begin{Bmatrix} \{\ddot{u}\} \\ \{\ddot{p}\} \end{Bmatrix} + \begin{bmatrix} [K^s] & [K^{sa}] \\ [0] & [K^a] \end{bmatrix} \begin{Bmatrix} \{u\} \\ \{p\} \end{Bmatrix} = 0 , \tag{1}$$

where $[M^S]$ and $[K^S]$ are the nxn structural mass and stiffness matrices; $[M^a]$ and $[K^a]$ are the mxm acoustic mass and stiffness matrices; {p} is the vector of the m nodal sound pressure at each fluid point; {u} is the vector of the n structural displacements; $[M^{as}]=\rho c^2[S]^T$, $[K^{sa}]= - [S]$, [S] is a sparse nxm structural-acoustic coupling matrix which elements are determined from the surface area S_{ij} for the boundary grid point corresponding to the structural displacement u_i and the associated sound pressure at that point p_j; ρ is the mass density of air and c is the speed of sound.

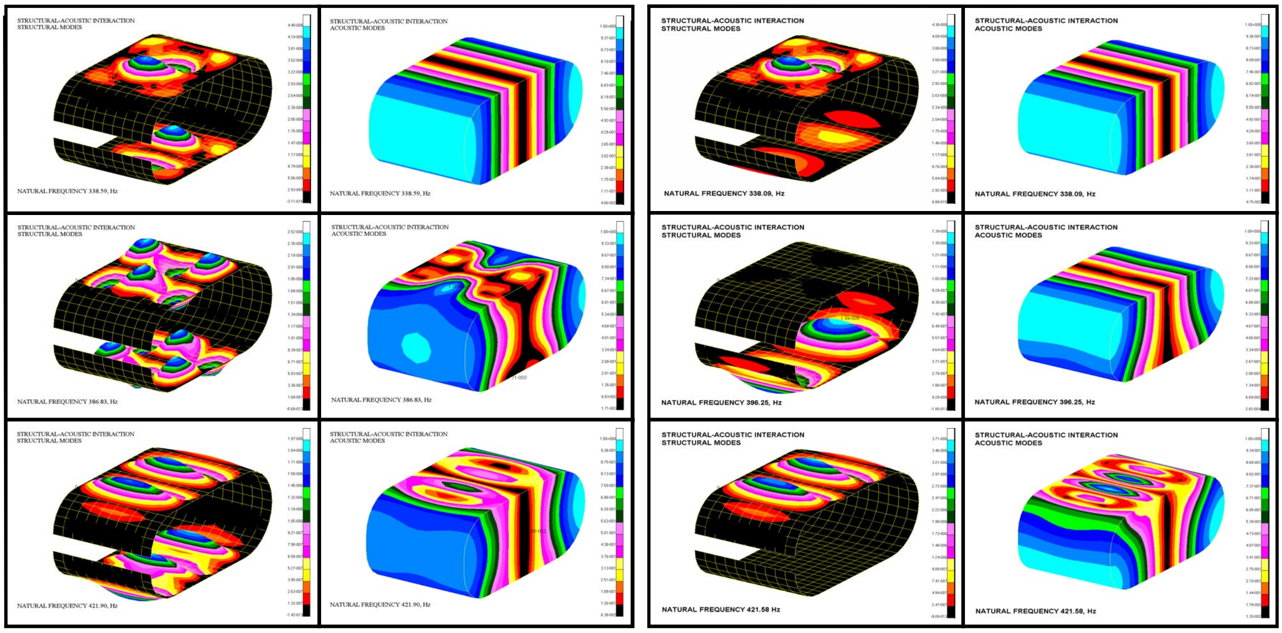

Figure 3: *Structural-acoustic coupling in QUASICAR (left) and in modified model **M1**(right)*

In the structural-acoustic coupling analysis we have considered in detail the coupling of the first rigid-wall acoustic mode at 338.26 Hz with different structural modes 'in vacuo'. The vibration energy of the coupled mode is divided between structural and fluid vibrations. In this respect, from the results shown in Table 2, one can distinguish predominantly "acoustic" and "structural" modes of the coupled QUASICAR model (Column 4) and of the coupled modified model ***M1*** (Column 5). Usually the coupling between acoustic and structural modes depends on their spatial similarity and frequency closeness. The spatial patterns of QUASICAR structure and modified model ***M1*** are two-dimensional; this means that structural modes will correspond best to the acoustic modes in the area of the relevant two-dimensional acoustic spatial patterns. Figure 3 shows the normal modes of the coupled models affected by the first rigid-wall acoustic mode. The acoustic uncoupled mode at 338.26 Hz and with (1,0,0) spatial pattern influences some QUASICAR structural modes with spatial patterns (2,3) at 394.69 Hz, and (4,1) at 415.33 Hz and some structural modes of modified model ***M1*** with spatial patterns (2,1) at 396.25 Hz, and (4,1) at 421.58 Hz. Comparing the coupled QUASICAR modes at 394.69 Hz and at 421.90 Hz, one can notice that the better matching of the structural and acoustic spatial patterns in the latter mode, in spite of its remoteness from the rigid-wall frequency, leads to a more distinctive picture of the coupling rather than for the previous normal mode.

Another interesting point is a great alteration of the fundamental frequency of the coupled models. As was pointed out in [12], this phenomenon is due to a strong coupling of the first structural mode with the zero-order acoustic mode (0,0,0) having zero natural frequency. Usually, the first cavity resonance frequency is above the fundamental structural frequency and coupling effects occur at frequencies above the first acoustic resonance, except for the case when the coupling occurs at frequency lower than the first acoustic peak. In this connection, we recall that QUASICAR model has two groups of natural frequencies: symmetric and anti-symmetric. Finite element analysis shows that only symmetric modes can couple efficiently with the acoustic modes. This might be because in anti-symmetric structural modes the fluid inside the cavity moves as a rigid body and does not exhibit vibration behavior. The influence of fluid on structural vibrations in anti-symmetric modes is also less pronounced than in the case of symmetric modes. This is why natural frequencies of symmetric and anti-symmetric structural modes in a coupled model have greater differences than in the case of the same structural modes in an uncoupled model, particularly for the fundamental modes.

7 FORCED VIBRATIONS OF THE COUPLED MODELS

In the previous section we have considered the vibrations of the coupled QUASICAR model and determined its modal parameters. However, to estimate generation of interior noise due to external or internal disturbing forces is necessary to carry out a forced-vibration analysis. In this case the governing equation of motion can be written as follow:

$$\begin{bmatrix}[M^s] & [0]\\ [M^{as}] & [M^a]\end{bmatrix}\begin{Bmatrix}\{\ddot{u}\}\\ \{\ddot{p}\}\end{Bmatrix}+\begin{bmatrix}[K^s] & [K^{sa}]\\ [0] & [K^a]\end{bmatrix}\begin{Bmatrix}\{u\}\\ \{p\}\end{Bmatrix}=\begin{Bmatrix}\{F^s\}\\ 0\end{Bmatrix}, \tag{2}$$

where $\{F^s\}$ is the vector of the external disturbing forces applied to the structure. Solving Eqn. (2), one can derive the structural displacements and the interior sound pressure due to a certain disturbanse in a coupled structural-acoustic model.

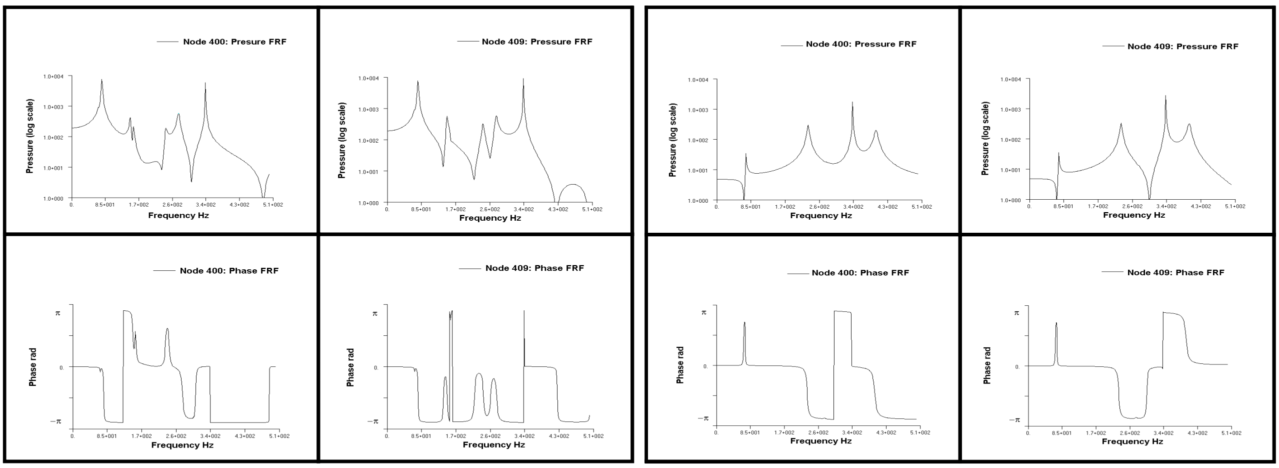

Figure 4: *FRF of QUASICAR model (left) and modified model **M1** (right)*

In the FEM a frequency response analysis was performed in the range from 0 to 500 Hz. A frequency-dependent exciting force was applied to the bottom plates of the QUASICAR and of the modified model ***M1***. The interior sound pressure response was calculated at the driver's ear location (acoustic node 400) and at the rear seat behind the driver's position (acoustic node 409). Only structural damping was included in these computations. Figure 4 illustrates the results for magnitude and phase of the frequency response function (FRF) of the QUASICAR model (left) and of the modified model ***M1*** (right). The solution looks typical for forced dynamic systems with multiple resonances. The resonant peaks correspond to natural frequencies of the coupled models. Phase curves of FRF have regions of positive and negative response corresponding to in phase and out of phase situations between interior sound pressure and the exiting force. Comparing the result for generated interior noise for these two models, one can notice the reduction of the interior sound pressure levels due to the additional thickness of the bottom plate in the modified model ***M1***.

8 CONCLUSIONS

In the present paper we have reported the results of the FEM structural-acoustic analysis of the simplified vehicle model QUASICAR and its modifications. Initially, the structural and acoustic calculations were carried out separately, and then a fully coupled model was studied. In the uncoupled model, the normal modes of the structure and acoustic modes of the enclosure have been calculated. It has been found that in the low frequency range structural vibrations of QUASICAR can be approximated by those of simply supported plates corresponding to the flat parts of the structure. The proposed modified models of QUASICAR have given an additional point of view on understanding the complex structural behaviour of the car body. In the coupled models of QUASICAR and modified model ***M1*** the interaction between structure and air has been studied. It was found that spatial similarity between structural and acoustic normal modes is a prerequisite for a better coupling even if the structural and acoustic natural frequencies do not match well. Finally, the structural-acoustic response due to a harmonic force excitation has been considered in the coupled models of QUASICAR and the modified model ***M1***.

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