# Acoustic-Structural Coupling of the Automobile Passenger Compartment

Mariana R. Kruntcheva

*Abstract*— **Renewed interest in reducing interior noise in transportation vehicles has motivated research in low frequency, structural-acoustic analysis. The internal sound field in the enclosed cavity is significantly affected by the acoustic modal characteristics of the cavity, by the dynamic behavior of the surrounding structure, and by the nature of the coupling of these two dynamic systems. The present work is intended to cast more light on the acoustic-structure interaction between the car compartment structure and the enclosed cavity. The system is studied using ANSYS finite element (FE) code. The modeling involved shell finite elements for the structure and three-dimensional (3D) acoustic elements for the cavity. The 3D FE modal analysis produced results visualizing the complex picture of acoustic-structure coupling. It was found that strong coupling between the thin-walled structure and the acoustic enclosure exists in the vicinity of any acoustic resonance. Also it was found that "combined" acoustic-structure modes of vibration exist in the vicinity of an acoustic resonance, which means that the coupled system manifests a new type of energy exchange.**

***Index Terms*—Acoustic-structure interaction, vibration, finite elements.**

## I. Introduction

The automotive industry is involved in a continuous endeavor to improve the noise and vibration characteristics of passenger vehicles. Under road conditions, the noise spectrum inside a passenger vehicle in the low frequency range <400Hz is found to be mainly structure-borne noise (see e.g. [1] ). The vibration energy generated from various sources is transmitted into the compartment cavity through structural connections. Thus vibration characteristics of the cavity and its boundary are very important factors which dominate acoustic response in a vehicle passenger compartment. Unpredictable noise problems can occur when the natural dynamic properties of the car body and compartment system coupled with the enclosed cavity are not well predicted. Among the various vehicle noise problems, structure-borne noise such as booming has been a subject of detailed investigations (see e.g. [2] and [3]).

Many studies have been carried out with an ultimate aim to understand the problem and reduce the boom noise (e.g. [4]). For various reasons, recent research has emphasized the low frequency noise in the range from 20 to 200Hz. It has been found that the internal sound field in the enclosed cavity is significantly affected by the acoustic modal characteristics of the cavity, by the dynamic behavior of the surrounding structure, and by the nature of the coupling of these dynamic systems. In addition, depending upon the relative values of the panel and cavity resonant frequencies, sound transmitted to the interior may be amplified rather than reduced.

More recently, it was proved that the well-designed trim-air gap system can play an important role in improving the acoustic response characteristics of the compartment [5]. Analysis and experiment show that the compartment resonance can be controlled by changing the gap thickness or trim mass. On the other hand, Song *et al*. [6] presented an active vibration control system for structural acoustic coupling of a 3D vehicle cabin model. The structural-acoustic coupling system is analyzed by combining the structural data from modal testing with the acoustic data from the finite element method.

The present study is intended to cast additional light on the acoustic-structure coupling in the gas-structure system discussed by using three-dimensional (3D) finite element (FE) modeling. The thin-walled passenger compartment interacting with the acoustic cavity is modeled employing simplifying assumptions. The external structure is assumed to be elastic, and therefore it is modeled using thin elastic shell elements. The fluid domain is discretized utilizing 3D pressure-formulated, acoustic elements. With respect to the modal analysis of the coupled problem, a special emphasis is placed not only on the changes in the frequency spectra, but also on the respective mode shapes which have rarely been treated in the literature. The present research highlights the very special and unique free-vibration behavior of systems comprising a thin-walled structure and an acoustic cavity, which is direct result of the acoustic-shell coupling.

## II. Procedure

The vehicle geometry selected was a Range Rover 2000 model. In the process of FE modeling the following assumptions were made:

1. The structure of the passenger compartment is modeled taking into consideration some of the curved surfaces. However, due to complexities of

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creating the model, some of the geometry is created by using straight lines.

2. As a qualitative picture of the modal response was aimed, the structure is assumed to be simply supported rather than modeling the accurate vehicle suspension. Moreover, the material properties are assumed to correspond to a lightweight metal without reference to the actual materials present in the cabin.
3. The laminated glass for the windshield consists of two glass layers and is placed between a plastic foil made of polyvinyl butyral. Also, the windshield is indirectly bonded to the vehicle body with the use of rubber seals. In the model, the windshield is assumed to be made of a single layer of glass and is directly connected to the vehicle body.
4. The materials for both metal structure and the windshield are assumed to be linearly elastic, as the modal analysis performed ignores any nonlinearity.
5. The effect of seats and other internal parts in the cabin is not taken into account.

The material properties for the structure are assumed to be Young's modulus $E=70$ *GPa*, Poisson's ratio $\nu=0.29$; and density $\rho=2700\ kg/m^3$; the windshield properties are assumed to be Young's modulus $E=64$ *GPa*, Poisson's ratio $\nu=0.17$; and density $\rho=2400\ kg/m^3$; the air properties are density $\rho_f=1.21\ kg/m^3$, and speed of sound $c=340\ m/s$.

For the system analyzed, FLUID30 (3D element) was used for creating the finite element model of the acoustic cavities. This element has 8 corner nodes with 4 degrees of freedom (DOFs) per node: 3 translational displacements in the nodal x, y, and z directions, and pressure. The translational DOFs are applicable only at nodes which are on the fluid-structure interface. For modeling the thin-walled structures, SHELL63 elements were used. These are 4-node, 3D elastic shells with both bending and membrane capabilities. In this application, the bending version of the elements was used. The SHELL63 element has 6 DOFs at each node: 3 translations in the nodal x, y and z directions and 3 rotations about the nodal x, y and z axes. The choice of this specific shell element was determined by its compatibility with the acoustic element.

The acoustic cavity was discretized utilizing the two versions of the acoustic element: (1) FLUID30 interfacing structure with four DOFs per node, and (2) FLUID30 possessing only one DOF per node, namely the unknown acoustic pressure. The layers of elements external with respect to the cavity were of type (1); the rest of the fluid domain was modeled by the normal type (2). In order to switch on the creation of coupling matrices on the wetted surfaces, a fluid-structure interface (FSI) flag was issued for all shell and fluid elements in contact. After mesh quality studies the final FE model comprised 4505 acoustic elements and 2009 shell elements of approximate size 0.1 m. The passenger compartment model is shown in Fig. 1.

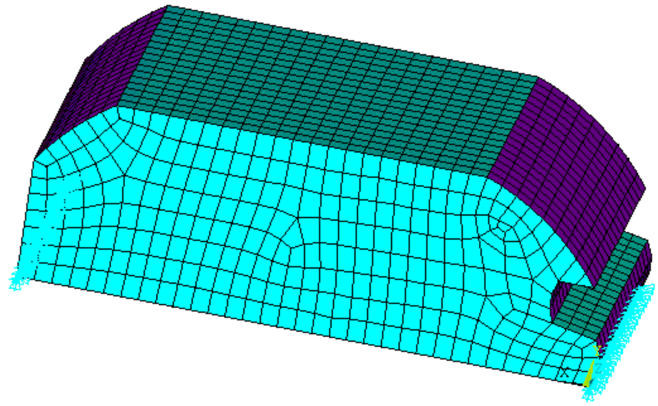

**Figure 1** FE model of vehicle compartment.

## III. Analysis

### A. Acoustic-Structure Coupling

The FE code ANSYS introduces several assumptions for a finite element representation of an acoustic space: (1) the fluid is considered to be compressible and inviscid, but without mean flow velocity; and (2) the mean fluid density and pressure are assumed to be uniform throughout the acoustic field.

Having created a particular structure-acoustic model, the final assembled set of equations takes the following shape:

$$\begin{bmatrix} [M] & [0] \\ [M^{fs}] & [M^{P}] \end{bmatrix} \begin{Bmatrix} \{\ddot{U}\} \\ \{\ddot{P}\} \end{Bmatrix} + \begin{bmatrix} [K] & [K^{fs}] \\ [0] & [K^{P}] \end{bmatrix} \begin{Bmatrix} \{U\} \\ \{P\} \end{Bmatrix} = \begin{Bmatrix} \{0\} \\ \{0\} \end{Bmatrix} \qquad (1)$$

Where

**[**$M$**]**= the assembled structural mass matrix,
**[**$K$**]**=the assembled structural stiffness matrix,
**[**$M^P$**]**= the assembled fluid equivalent "mass" matrix,
**[**$K^P$**]**=the assembled fluid equivalent "stiffness" matrix,
**[**$M^{fs}$**]**= the assembled fluid-structure coupling "mass" matrix,
**[**$K^{fs}$**]**=the assembled coupling "stiffness" matrix,
$\{P\}$ $\{\ddot{P}\}$ = the nodal pressure vector and the vector of its second time derivatives respectively,

$\{U\}$ = the nodal displacement vector

and $\{\ddot{U}\}$ = the nodal acceleration vector.

### B. ANSYS Unsymmetric Solver

In order to solve the set of governing equations presented in Acoustic-Structure Coupling, it was necessary to make use of the Lanczos algorithm implemented in the ANSYS Unsymmetric Solver. A verification procedure was first performed, to make sure the results produced were accurate. A mode-frequency analysis of the elastic shell in vacuo was carried out using both the Reduced and the Unsymmetric

Solvers. With acceptable errors from the engineering point of view, the results were comparable. The comparison above gives sufficient confidence to apply the Unsymmetric Solver in the structure-acoustic modal analysis. It should be noted, however, that the so-called "shift" (the starting frequency or point of discovery) is of paramount importance for achieving meaningful results with the Lanczos algorithm. If some eigenvalues are clustered, the final outcome depends on the first step at which a hidden eigenvalue is expected. Bearing in mind the "strange" behavior of the Unsymmetric Solver, the "shift" value should be kept constant in all applications where any kind of comparison between the solutions is required. This approach was adopted in the present study. In addition, one should bear in mind that the Lanczos algorithm is suitable for extraction of some, but not all, eigenvalues and associated eigenvectors in the frequency ranges of interest. It is assumed that the frequencies extracted are "exact", but some of the mode shapes may be distorted, hence they are difficult to recognize.

## IV. Results & Discussion

Renewed interest in reducing interior noise in transportation vehicles has motivated research in low frequency (20-200Hz) structural-acoustic analysis. Previous testing has shown strong correlation between wall panel motion and the measured noise, and it has been suggested that the noise may be amplified by the passenger compartment cavity resonance (see [4]). Since the passenger compartment of an automobile forms a closed cavity, resonant conditions can develop. These resonances are characterized as acoustic modes of vibration (standing waves) associated with which are specific pressure distributions and natural frequencies. If rigid walls are assumed in the finite element model for the boundary panels, by fixing all the displacement degrees of freedom of the acoustic elements, then the rigid-walled acoustic modes are extracted. The first two acoustic modes corresponding to frequencies $f_a$=57.3 Hz and $f_a$=103.1 Hz are shown in Figs. 2 and 3 respectively. Careful observation of the first mode shows that it has a neutral line of zero pressure around the middle of the cabin length. In this particular model the effect of the seats on the modal analysis is ignored as the study aims at qualitative rather than quantitative results.

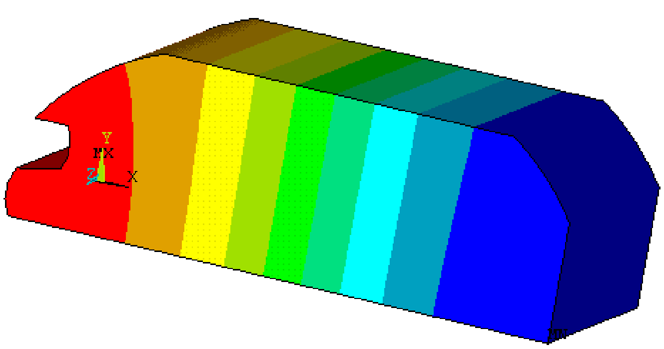


**Figure 2** First rigid-walled acoustic mode of passenger compartment at frequency fa=57.3 Hz.

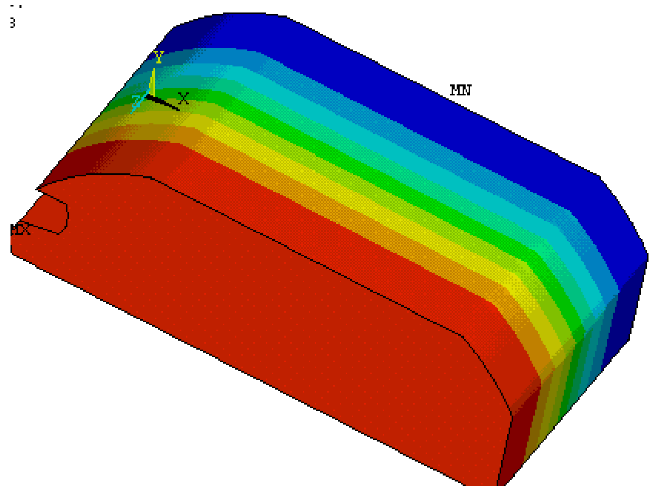


**Figure 3** Second rigid-walled acoustic mode of passenger compartment at frequency fa=103.1.3 Hz.

It has been proven that the seats would change the position of the neutral lines and lower the corresponding acoustic frequencies (see [4[), but would not alter the principle of acoustic-structural coupling.

The vibration patterns receive a great deal of attention in this paper. Three types of coupled modes were identified: 1) coupled "combined" (acoustic–structure) resonances 2) coupled acoustic modes and 3) coupled structural modes. In the regions of strong acoustic-structural coupling (see e.g. [7], and [8]), i.e. when the acoustic and structural natural frequencies are sufficiently close, special "combined" modes of vibration were observed. Normally, any structural mode of the system is characterized by a high amplitudes displacement pattern, i.e. normalized radial amplitudes of order $10^{-4}$, and the corresponding pressure fluctuations affect only the contiguous layer of acoustic elements. In addition, a typical mode of the acoustic cavity is recognizable by its special pressure pattern and very small normalized radial displacements of the shell, of order $10^{-8}$-$10^{-9}$, which clearly follow the pressure wave. The combined modes, however, were seen to possess patterns of relatively high normalized radial displacements for the shell (of order $10^{-5}$ and apparently not related to the internal pressure field), combined with specific pressure distributions indicating acoustic resonances in the annulus. It was found that, when strong acoustic-structural coupling exists, the system responds to free vibration in an unusual manner. The radial structural modes of the shell *in vacuo* and the acoustic pressure modes of the rigid-walled cavity are complemented by a new kind of dynamic behavior, in which the system's vibration involves both the shell and the fluid region. Consequently, these special modes can be called "*combined*" or *acoustic-structure resonances* (modes).

For the particular configuration considered, the first structural mode corresponds to a very low frequency of 6.5 Hz and in the first region of strong acoustic-structure coupling (around the first acoustic resonance at frequency 57.3 Hz) numerous coupled structural and "combined" modes were

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extracted by the Lanczos Solver. For example, for the frequency range of 50 Hz to 60 Hz the solution produced 5 "combined" and 8 structural resonances with a frequency difference in some cases less than 1 Hz. The structural behavior for "combined" mode at frequency 58.26 Hz which is very close to the first acoustic resonance of 57.3 Hz is shown in Fig. 4. It is clear that the elastic structure vibration involves predominantly a panel oscillation of the lower cabin plate.

The next Fig. 5, presents the independent acoustic pressure distribution within the cavity. It can be seen that the pressure distribution does not follow the vibration of the structure and although disturbed, a neutral line at approximately the middle of the cabin length can be observed. The position of the neutral line is very similar to the neutral line position for the first acoustic resonance from Fig.-2

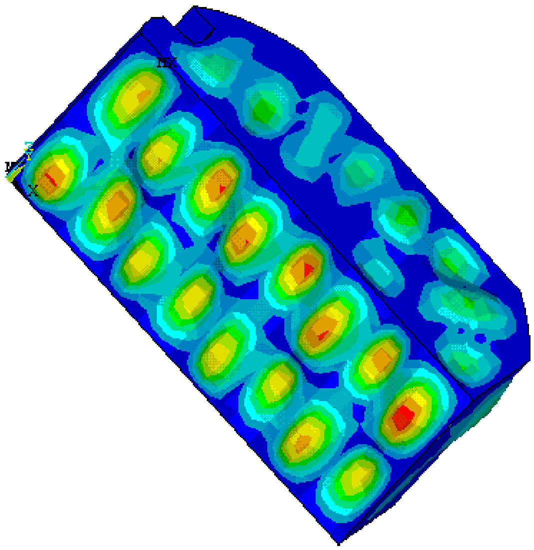

**Figure 4** Structural displacement pattern for coupled "combined" mode at frequency f=58.26 Hz.

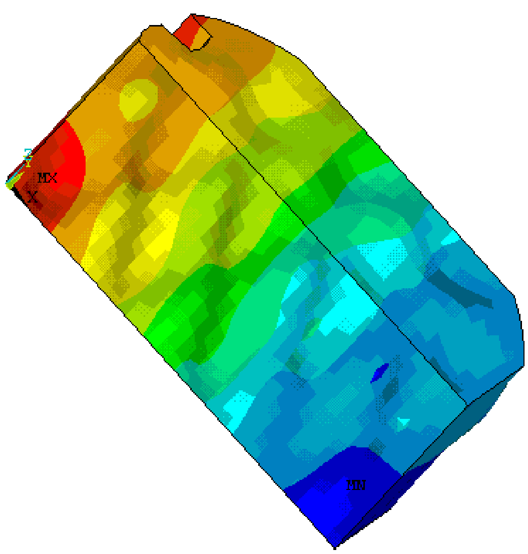

**Figure 5** Pressure pattern for coupled "combined" mode at frequency f=58.26 Hz.

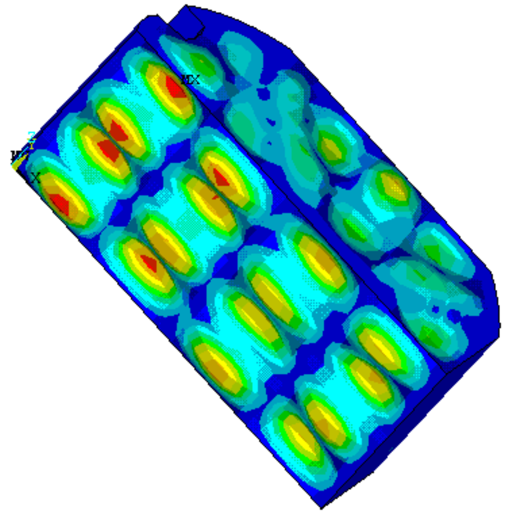

**Figure 6** Structural displacement pattern for coupled structural mode at frequency f=57.33 Hz.

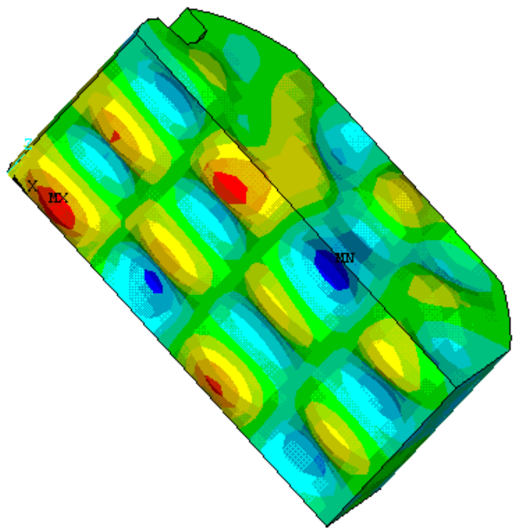

**Figure 7** Pressure pattern for coupled structural mode at frequency f=57.33 Hz.

As discussed earlier, the Unsymmetric Solver does not extract all the possible frequencies and modes in the frequency range of interest and some of the modes can be distorted. This can contribute to the explanation that the pressure patterns for the "*combined*" modes are not exactly matching the rigid-walled acoustic modes.

It is worthy of note however, that in the regions of strong acoustic-structure coupling many coupled structural modes were extracted similar to the mode shown in Fig. 6. It is clear that the corresponding pressure pattern from Fig. 7 follows the vibration of the elastic panels.

The solution of the structure *in vacuo* produced structural modes, most of which were present in the coupled spectrum. One such mode is shown in Fig. 8. The corresponding frequency of 57.89Hz is very close to this of the same coupled structural mode shown in Fig. 6.

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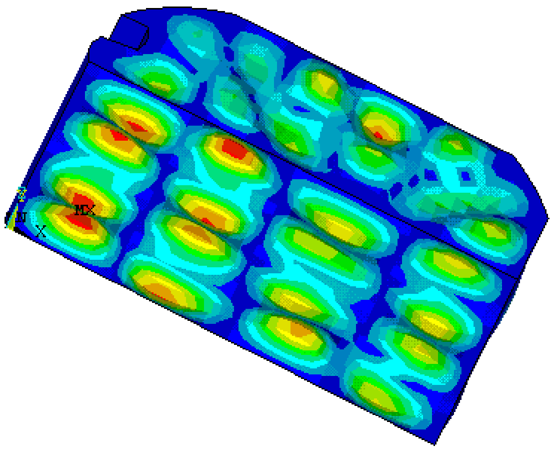


**Figure 8** In-vacuo structural vibration mode of passenger compartment at frequency $f_s$=57.89Hz.

It was observed that the "*combined*" modes are clustered in the zones of strong coupling around the system acoustic modes and the acoustic-structural coupling is dominated by the acoustic resonances.

The 3D FE modeling of the problems considered allows a visualization of the complex interaction between the elastic structures and the acoustic cavities. In addition, the eigenvalue extraction routine produces the frequencies one after another in ascending order. As a direct result, the conclusion drawn by Horácek and Zolotarev [9] that a strong acoustic-structure coupling exists only in the case of wave number coincidence for the structural and cavity modes was invalidated. In the present study, strong coupling was observed in the vicinity of any acoustic resonance. The discrepancy discussed is believed to be a result of the manner in which the previous studies have been carried out, e.g. prerequisites (not always justifiable) on the system's mode shapes have usually been imposed.

As far as the present research is concerned, the emphasis should be on the "*combined*" modes of structure-gas vibration. According to the present study, a group of acoustic-structure modes related to a frequency zone of strong coupling, as defined earlier, involves different structural modes and a single vibration mode of the acoustic cavity. These "*combined*" modes could be excited applying the external forcing on the structure and varying the frequency in a relatively large bandwidth.

Based on the results already published, several comments are in order. Horáček and Zolotarev [10] arrived at a rather general conclusion, as a result of investigating the influence of the acoustic medium on free vibrations of three different mechanical systems. This was that in the regions of strong acoustic-structural coupling, any fluid-mechanical system has two different natural frequencies which are neither purely acoustic nor purely structural, and that the dynamic properties of the solid-fluid configuration cannot be studied separately for the structural and acoustical sub-systems. Moreover, according to these authors, it is not possible to distinguish the acoustic modes from the structural ones under the defined conditions. In the light of the present study however, the aforementioned statement is not completely accurate. The fact that the strongest acoustic-structural coupling exists if the resonances of the structure and the acoustic cavity are close to each other is substantiated by the results presented. However, it is not true that, (1) the coupled system possesses only two natural frequencies which are of some unknown type, and (2) the modal parameters of the sub-systems are not distinguishable. In the course of this research, it was clearly demonstrated that the rigid-walled acoustic modes and the *in-vacuo* structural modes occur unaltered in the spectra of the coupled system. The corresponding mode shapes are neither transformed, nor in any way affected by the fluid-shell interaction. The presence of acoustic medium alters only the natural frequencies, causing the acoustic ones to decrease slightly and the structural ones to diminish at a higher rate. In addition, in the vicinity of an acoustic resonance of any order, the gas-structure configuration behaves in a "*combined*" manner manifesting natural vibration modes which do not exist originally either for the structure *in vacuo* or for the rigid-walled cavity. Essentially, the acoustic influence on the vibrations of the system is found to work in two directions: (1) decreasing the rigid-walled acoustic and *in-vacuo* shell frequencies with no modifications of the associated mode shapes; and (2) causing the appearance of "*combined*" vibration modes.

## V. Conclusion

The 3D finite element modeling used produced results visualizing the complex picture of acoustic-structure coupling. In this study, strong coupling between the structure and the acoustic space was observed in the vicinity of any acoustic resonance.

1. The vibration patterns receive a great deal of attention in this paper. The emphasis should be on the "*combined*" modes of vibration. In the present research, it is found that these vibration modes involve both the structure and the fluid domain. In addition, the "*combined*" mode shapes possess both high-order displacement and high-order pressure amplitudes. In this paper, they are also called *acoustic-structure resonances* to distinguish them from purely acoustic and purely structural vibration responses respectively. In the vicinity of an acoustic resonance, the coupled system manifests a new type of energy exchange. A strong acoustic radiation from the shell resonating in one of the radial structural modes is capable of exciting the pressure standing wave of nearest frequency in the acoustic cavity. Vice versa, when the acoustic annulus is excited to resonance, it can transfer sufficient energy to the contiguous structure so that a radial shell vibration at a close frequency is activated. It is also found that the "*combined*" modes are clustered around a rigid-walled cavity mode, and any acoustic-structure resonance of a given group involves this particular acoustic mode.
2. This study emphasized that *the sub-systems preserve their capability of independent vibration responses* in the frequency ranges of strong acoustic-structure coupling.

## REFERENCES

[1] T. Sakata, H. Morimura, and H. Ide, "Effect of tire cavity resonance on vehicle road noise", *Tire Science and Technology*, vol. 18, 1990, pp. 68-79.

[2] E.H. Dowell., 1980, "Master plan for prediction of vehicle interior noise", *American Institute of Aeronautics and Astronautics Journal,* vol. 18, 1980**,** pp 353-366.

[3] S.H. Kim and J.M. Lee, "A practical method for noise reduction in a vehicle passenger compartment", *ASME Journal of Vibration and Acoustics*, vol. 120(1) 1998, pp. 199-205.

[4] D.J. Nefske, J.A. Wolf Jr, and L.J. Howell, "Structural-acoustic finite element analysis of the automobile passenger compartment: a review of current practice", *Journal of Sound and Vibration***,** vol. 80(2), 1982, pp. 247-266.

[5] S.W. Kang,, J.M. Lee, and S.H. Kim, "Structural- acoustic coupling analysis of the vehicle passenger compartment with the roof, air-gap, and trim boundary," *ASME Journal of Vibration and Acoustics*, vol. 122 (3), 2000, pp 196-202.

[6] C.K. Song, J.K. Hwang., J.M. Lee, and J.K. Hedrick, "Active vibration control for structural- acoustic coupling system of a 3D vehicle cabin model", *Journal of Sound and Vibration*, vol. 267(4), 2003, pp 851-865.

[7] M.R. Kruntcheva and I. Zolotarev, 1996, "Dynamics of cylindrical shells in contact with subsonic flow - finite element modeling", *Proceedings of Inter Noise 96*, Liverpool, UK, vol. 4, 1996, pp 2989-2993.

[8] M.R. Kruntcheva, 1998, "Dynamics of circular cylindrical shells in contact with fluid", Ph.D. Thesis, 1998, The Robert Gordon University, Aberdeen, U.K.

[9] J. Horácek and I. Zolotarev, "Frequency-modal characteristics of cylindrical shell with annular flow of light fluid", *Proceedings of Eight World Congress on the Theory of Machines and Mechanisms*, Prague, Czechoslovakia,1991, pp1503-1506.

[10] J. Horácek and I. Zolotarev, "Influence of the acoustic-structural couplings upon free vibrations of mechanical systems", *Archives of Acoustics*, vol. 18(2), 1993, pp 297-309.