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Acoustical identification of passenger cabin of vehicles

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Abstract

Sound in the interior cabin of cars has been of the great importance nowadays. The automotive industry has long been fiercely competitive with different vendors attempting to produce vehicles with better performance, handling and fuel efficiency. The competition of the automotive industries today is directed on the reduction of noise in the cars cabin. This paper proposes a new method for noise studies in the cabin, which frequencies of excitation are lower than 200 Hz. The topics, which are covered in the paper, include the computation of the structural and acoustic mode shapes, resonant frequencies of the passenger compartment, coupling between structure and acoustic effects, and forced vibration analysis. In this research work noises, generated by cabin's panels, will be identified and then those, which are main sources of noise generators will be distinguished. The acoustical system of passenger cars cavity and vibration properties of panels and coupling between panels and air are under consideration here. A finite element method is proposed here to make a perfect model for the car cavity and then analyze compartments particularly. Panels, seats, glasses, trims, and the interior air are modeled considering various type of material for each one. Noise generator is considered here to be a loudspeaker located at the front right hand side of the car cabin. Changing frequencies of the speaker in a reasonable range making resultant pressure distributions at the driver ears position. The main goal of the article is analyzing of the acoustical system of car using proposed FEM and identification of the system. Using the proper identification method, transfer function of acoustical system at low frequencies can be concluded. This transfer function is of very high importance for noise control in vehicles.

1 Introduction

Reduction of the noise effects come from the automobile engine and power train, blowing wind, road interaction to the tires, and exhausted gases radiated into the atmosphere are important parameters for passenger conveniences. Making accurate and useful tools for modeling and analysis of automotive interior noise in order to improve design features are of the great importance, which car companies have to consider to be able to compete to others and distribute their own products for their costumers. Cars chassis and the other elements of the body can be said as of the main significant ways for noise transfer to the interior space of the car cavity.

At the low frequency range under 200 Hz, car interior noise is determined by system model specifications. These are acoustic resonance, body vibration modes, and structural-acoustic coupling effects, as well as the properties of noise and vibration sources. If a good design for low frequencies is not considered, several noise problems like booming will occur. As a result modification of the body structure is difficult to do when cars are manufactured. To avoid these problems, detailed analysis on the body compartment is necessary while design process is progressing. Finite element analysis have been performed using commercial programs such as MSC/NASTRAN, ANSYS, ABAQUS, in which vibration responses of the car body are first analyzed and then pressure responses will be calculated [1-3]. For structural modification, noise contribution of boundary panels or acoustic sensitivity of the design variables were introduced [4-6]. In reality, however, body structure has tens of thousands or hundreds of thousands degree of freedom when finite element model considered. Furthermore, various damping element render the acoustic response complicated. This situation has made design engineering feel the limitation of numerical analysis for vehicle noise problem.

In 1966, Gladwel and Zimmermann [7] developed an energy formulation for the structural acoustic theory, and that paper set the stage for application of finite element methods to acoustic cavity analysis. Early contributions to the development of the finite element approach were made by Craggs [8-10]. Shuku and Ishihara [11] proposed a practical method for analyzing the acoustics of the automobile passenger compartment.

By using scaled models of automobile compartments, finite element predictions were verified experimentally by Petyt et al. [12], and also by Richards and Jha [13]. Craggs [9] presents verification of his finite element prediction of acoustic resonance in a passenger compartment, while Nefske and Howel [14] present an experimental verification of the finite element method for predicting noise reduction in the automobile .A diagnostic technique based on the finite element method has been developed for automobile interior noise reduction [14], and it has been applied successfully to diagnose and reduce interior boom noise generated by road input and engine vibration .A comprehensive review of these can be found in the survey by Dowel[15].

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2 Governing equation and finite element formulation

For this analysis Helmholtz equation is used to describe acoustic wave in a cavity as follows [16]:

$$\nabla^2 p_0 + (\omega / c)^2 p_0 = 0 \tag{1}$$

In which c is the speed of sound in the media, p_0 is the acoustic pressure, and ω is the frequency.

Acoustic wave will affect on the surrounded structure as well and vice versa. As a result governing equations for coupling effects between the acoustic media and structure is as follows:

$$[G]\{\ddot{p}\} + [H]\{p\} = -(\rho c)^2 [A]^T \{\ddot{u}\} \tag{2}$$

$$[M]\{u\} + [C]\{\dot{u}\} + [K]\{u\} = [A]\{p\} + \{F\} \tag{3}$$

Where the column matrix {p} gives the pressure at the grid points of the finite element mesh, [G] and [H] represents acoustic mass and stiffness matrices, $\{\ddot{u}\}$ is the structural acceleration vector, $[A]^T$ is the transposed matrix for the boundary surface areas of the acoustic cavity model, {u} designates the structural displacement vector, [M], [C], And [K] represent structural mass, damping, and stiffness matrices, {F} is the vector of external forces applied to the structure, and finally [A] is the structural-acoustic coupling matrix for the boundary surface area.

3 Acoustic modeling of a cavity with FEM

It is necessary to model a vehicle cabin to understand acoustical behavior of the system and to identify system. In this article a medium size vehicle, named Pride production of the SAIPA Co. Iran, is modeled with ANSYS software. In this modeling it is not necessary to model all of the details of the cavity, like steering wheel, dashboard, and curtain. But modeling of seats, trims, glasses, and air gap between trims and panels are important .The geometry of the cavity is shown in Fig. 1a. Shell elements, made of steel, are in use for panels, which are shown in the Fig. 1b. In this model for achievement to a reasonable reality, stiffeners are in use too, making frequencies close to the vibrating frequencies come from the experimental analysis of modal. The whole body is fixed at the tires position to omit zero frequencies.

Air of the passenger cavity and engine cavity is meshed with Fluid30 elements (Fig. 2a). Each element has one degree of freedom normally, but close to the boundary of the cavity three additional degrees of freedom will be added to each element (displacement in three perpendicular directions). Seats are modeled with foam material (Fig. 2b). Trunk cavity is separated from passenger cavity and there is no air connection between them, so it is not necessary to model the trunk. It is necessary to apply absorbing coefficient of boundary of passenger cavity. Absorbing coefficient (MU) is a value between zeros and one. Un-surrounded air absorbs sound energy completely, MU=1. Some kind of materials that have rough surfaces like glass, will not absorb sound energy and reflect the energy

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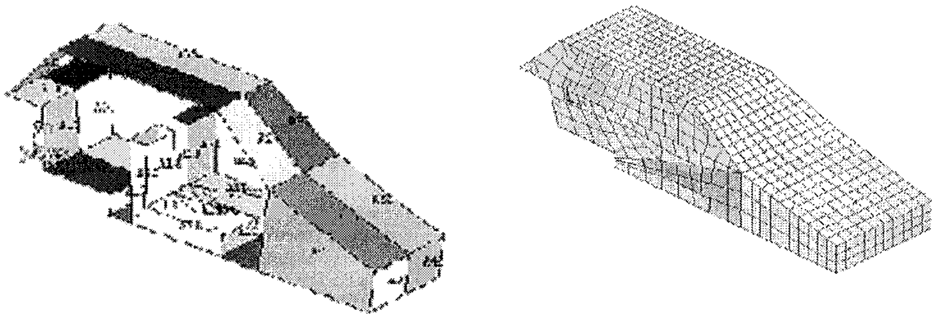
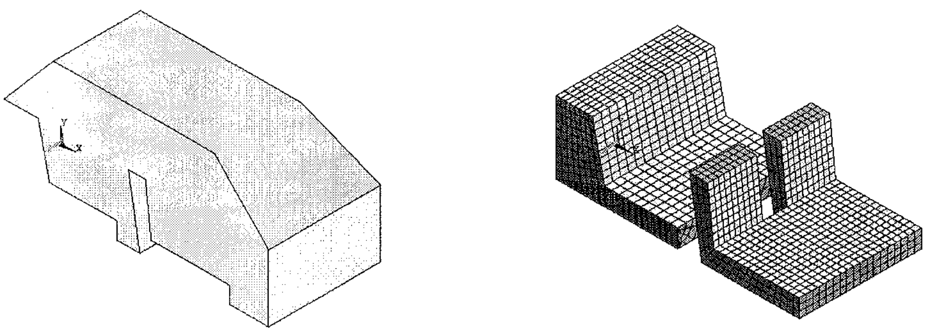
completely, MU=0. For the other material, such as trims, this coefficient is different for various frequencies as shown in the Fig.3.



(a) (b)

Figure 1: Model of Pride vehicle. (a) Geometry of car body. (b) Mesh generation by the shell elements.



(a) (b)

Figure 2: Model Pride vehicle cavity. (a) Geometry of the cavity. (b) Model of seats using foam material.

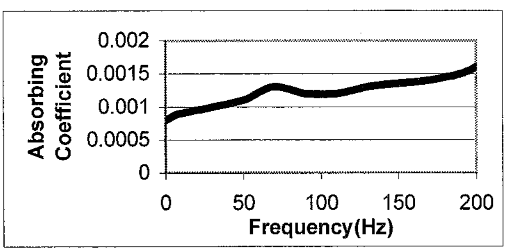


Figure 3: Sound absorbing coefficient of trim for the various freq.

4 Acoustic mode shapes

For the medium size vehicle, modeled in this study, five acoustic mode shapes are derived under 200 Hz. as shown in the Table 1. These mode shapes are shown in the Fig. 4 respectively. First acoustic mode shape is a uniform pressure distribution and has no meaning usually excluded from the results. Nodes of the minimum pressure distribution, describe planes for the second acoustic mode shapes, will be at the driver ears position, and therefore have the minimum noise effects on the driver's ears (Good situation).

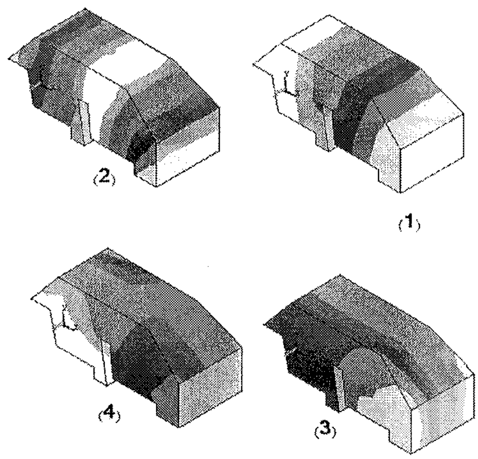


Figure 4: Acoustic mode shapes of passenger cavity indicating various pressures distribution. (1) Second mode f =78 Hz. (2) Third mode f =151.4 Hz. (3) Fourth mode f =188.8 Hz. (4) Fifth mode f =199.2 Hz.

Table 1: Natural frequencies of structure, cavity, and coupling one.

Mode number	Coupling freq.	Structure freq.	Acoustic freq.	Mode number	Coupling freq.	Structure freq.	Acoustic freq.
1	0		0	15	116.70	116.74	
2	38.63	38.62		16	122.41	122.44	
3	52.74	52.77		17	126.09	126.04	
4	61.82	61.86		18	132.12	132.27	
5	62.20	62.21		19	140.3	140.37	
6	73.04	73.06		20	143.5	143.66	
7	81.39		78	21	145.2	145.5	
8	81.34	86.58		22	147.6	147.72	
9	95.82	95.84		23	151.2	151.23	
10	98.83	98.86		24	154.3		151.4
11	101.07	101.01		25	156	155.56	
12	105.69	105.7		26	158.9	158.82	
13	107.93	107.80		27	162.2	162.69	
14	110.95	110.98		28	176.3	175.96	

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5 Structural modes

In this stage the body vehicle together with glasses are analyzed. The effects of air, trim, and seats are ignored. Model consists roof, floor, dash panel, side panel, rear shelf, motor cavity, and glasses. Derived natural frequencies are shown in the Table 1. Body is fixed at the tire positions, so zero frequency is omitted.

6 Coupling analysis

Up to this point, the forced acoustic response has been computed by assuming that the boundary panel motion is unaffected by the cavity sound pressure. However, as in the case of the passenger compartment modes, there are instances where structural-acoustic coupling effects are important, in these cases equations (2) and (3) must be solved simultaneously. In this stage the body and the interior air is modeled together. Because of coupling between structure and acoustic the natural frequencies and mode shapes are changing and the system equations are unsymmetrical. The coupling frequencies are shown in the table 1 too.

7 Noise generated by engine

In this part, noise, generated by engine is measured. In order to model engine noise, a point source of sound (loudspeaker) located at the engine place producing sound at a constant pressure 50 decibel by the frequencies range from zero to 200 Hz. is considered here. Sound at the driver ear position is measured and plotted in a diagram, which is shown in Fig. 6. Sound pressure is increased at some frequencies, which are acoustic frequencies. Loudspeaker in this place cannot excite all of the acoustic modes. At some frequencies the pressure node is at the driver ear position and as a result sound is minimized.

8 Panel noise contributions

A major factor in interior Noise Vibration Harshness (NVH) is the vibration and acoustic behavior of vehicle body panels. When a vehicle travels in a road, its entire panel vibrates. This is due to engine and power train, blowing wind, road interaction to the tires, and exhausted gases radiated into the atmosphere. This makes panels to vibrate causing acoustic pressure in the acoustic cavity of passenger compartment. Thus noise will be heard at the inner and the outer of vehicle. If the input frequency to the acoustic system is the same as natural frequency of one panel, resonance will happened and this panels vibrates violently. Acoustic energy, input to the system, will be consumed in the vibrating panel and a terribly noise will be heard in the vehicle. If all of the panels vibrate together, the sound heard is the resultant vector of individual components. For instance each panel vibrates by 50-decibel pressure and same phase at 40 Hz. Measured pressured and phase are plotted at a polar diagram in Fig. 5. From this panel diagram one can understand the effect of each panel on the noise distributed in the vehicle. Some of them increasing acoustic pressure at the driver

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ear position and some others decreasing. Fig. 5 shows that rear glass and rear shelf have an increasing effects, while floor and front glass have a decreasing effects. Changing stiffness of the decreasing effects panel will change the phase angle of their sound vectors and may increase the resultant vector at the ears position. So if changing is required be better to do at the increasing effects panels. Distances from panels to ears position have no effects on diagram.

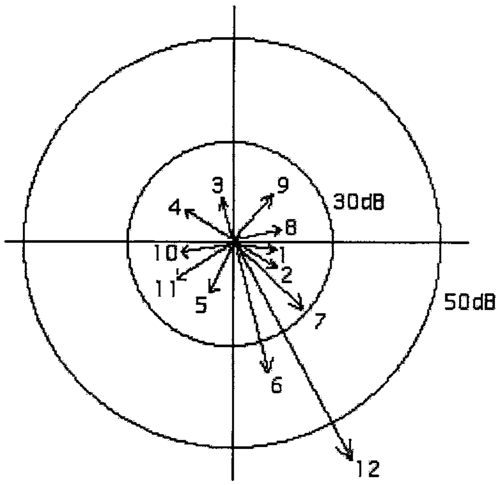

Figure 5: Polar amplitude-phase diagram of panels' noise and resultant noise heard at the driver ear position (40 Hz). 1-front floor .2-middle floor. 3-dash panel. 4-windshield. 5-roof. 6-rear glass. 7- rear shelf. 8-right glass. 9-rear glass. 10-right side. 11-left side. 12-resultant pressure.

9 Identification of the acoustic system

In order to identify the dynamics behavior of a system it is usual to input some states at a point and measure its corresponding output at the other points. In an acoustic system by plotting the responses of the system varying frequency it is possible to identify its acoustic transfer function connecting two points. This function is very useful for control of acoustic systems. Identification is a mathematical approach and there are many ways for identification. It is not possible to say which way is the best one. All of them have its own advantages considerably and depends on the system, which has to be identified. In this article least square method is used for identification. Assume that the points (x_i,y_i), where i=1,2,..,n is plotted as a curve. The purpose of identification is to determine a function y=f(x) that fit the points (x_i,y_i) having proper accuracy. Transfer function of acoustic system often propose by the following [16]:

$$y = \frac{g(a_1, a_2, a_3, ..., x)}{h(a'_1, a'_2, a'_3, ..., x)} = \frac{a_1 + a_2 x + a_3 x^2 + ...}{a'_1 + a'_2 x + a'_3 x^2 + ...} \tag{4}$$

Parameters $a_1, a_2, ...$ and $a'_1, a'_2, ...$ have to be identified as of identification procedure by minimizing following:

$$Error = \sum_{i=1}^{N} [y_i - \frac{g(x_i)}{h(x_i)}]^2 \tag{5}$$

Error depends only to the parameter $a_1, a_2, ...$ and $a'_1, a'_2, ...$ it is necessary to determine these parameters in a way to minimize error. So the partial derivative of error with respect to the each parameter is set to zero. Solving the following system equation will complete identification:

$$\begin{cases} \dfrac{\partial Error}{\partial a_1} = 0 \\ \dfrac{\partial Error}{\partial a_2} = 0 \\ . \\ . \\ . \\ \dfrac{\partial Error}{\partial a'_1} = 0 \\ . \\ . \\ . \end{cases} \tag{6}$$

In this paper the transfer function between engine and ear driver is identified. The input data (x_i, y_i) is taken from curve of Fig. 6. Data is taken at even frequencies 2,4,...,200 Hz., Thus the number of data is 100. With the less numbers of data accuracy of function y=f(x) will diminished. The order of numerator and denominator respectively determine the numbers of zeroes and poles. Orders of numerator and denominator are chosen 49, so there are 50 parameters of $a_1, a_2, ...$ and 50 parameters of $a'_1, a'_2, ...$, to be identified. By solving the system of equation using MAPLE software, parameters will be identified. It is seen that the order of numerator change to 45. Transfer function using ANSYS and MAPLE are shown together in Fig. 6.

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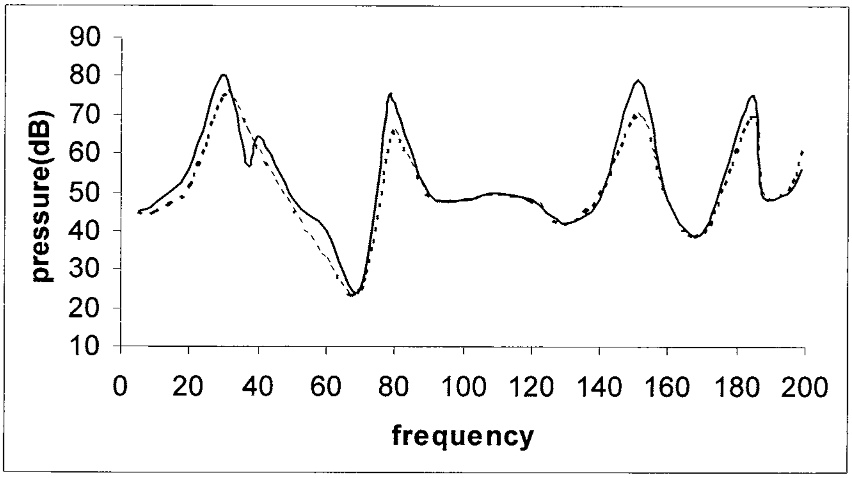

Figure 6: Pressure at the driver ear position induced by engine noise, dashed line is MAPLE transfer function and bold line is ANSYS one.

10 Conclusion

Analytical description of vibration-induced noise in a passenger compartment needs knowledge about acoustic characteristics of the cavity as well as the vehicle structural characteristics and also excitation forces. In this paper the analysis of the acoustic cavity has been considered and a finite element method based on ANSYS software was applied to the acoustic models of a vehicle, named Pride production of an Iranian company, SAIPA.

At the design procedure of a vehicle, finite element analysis can be used to determine cabin resonance due to effects come from environment and vehicle parts. Design variables should be selected in a way to achieve reasonable agreement with the acoustic behavior. In this research work acoustical model of a vehicle is proposed and then by using finite element method modal analysis, acoustic modal analysis, structural modal analysis, and coupling modal analysis have been performed successfully. These analyses are important because of their application for acoustic identification. The results show also influence of the acoustic system on the driver ears to understand these effects. Finally a transfer function showing dynamics behavior of the acoustic system, using results of the acoustic modal analysis, is proposed here. Using methods and results of the paper is a good way for prediction of vehicle interior noise and can be applied to a vehicle design procedure for conformability of the human being when riding.

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References

[1] Nefske, D.J., Wolf, Jr. & Howel, L.J., *Structural-acoustic Finite Element Analysis of The Automotive Passenger Compartment*, J. of Sound & Vibration, Vol 80, pp. 247-266,1982

[2] Sung, S.H., Nefske, D.J., *A coupled Structural-acoustic Finite Element Model For Vehicle Interior Analysis*, ASME Transaction, J. of Vibration, Acoustic, Stress & Reliability, 1984

[3] Yashiro, H, Suzuki, K., Kajio, Y. Hagiwaraand, I., *An Application of Structural-acoustic Analysis to Car Body Structure*, SAE 850961, 1985

[4] Lee, J.M., Sung, M.H., Kang Y.J., Kim S.H., Kim,J.H., *An Effective Method for Interior Reduction of a Passenger Car,* Proceedings of the 6th Intnl. Pacific Conference on Automotive Engineering, pp. 553-560

[5] Hagiward I., Ma, Z.D., *Sensitivity Calculation Method for conducting Modal Freq. Response Analysis of Coupled Acoustic-structure System,* JSME International J. Series III,Vol 35,pp.14-21

[6] Choi K.K.,Shim I. & Wang, S., *Design Sensitivity Analysis of Structure-induced noise & Vib., ASME Transaction, J. of Vib. & Acoustic*, Vol 119, pp. 173-179

[7] Gladwel, G.M.L., & Zimmermann, G., *An Energy & Complimentary Energy Formulation of Acoustic & Structural Vib. Problems*, J. of Sound & Vib.,1966

[8] Craggs A., *The Transaction Response of a Coupled Plate-Acoustic System Using Plate & Acoustic FEM*, J. of Sound & Vib. Vol 15, pp. 509-528,1971

[9] Craggs A., *The Use of Simple Three Dimensional Acoustic Finite Elements for Determining Natural Modes & Freq. Of Complex Shape Enclse*, J. of Sound & Vib. Vol 23, pp. 331-339, 1972

[10] Crags A., *An Acoustic FEM For Studying Boundary Flexibility & Sound Transmission Between Irregular Enclosure*, J. of Sound & Vib. Vol 30, pp. 343-357, 1973

[11] Shuku T. & Ishihara, K., *The Analysis of The Acoustic Field in Irregularly Shaped Rooms by The FEM*, J. of Sound & Vib., Vol 29, pp. 67-76, 1973

[12] Petyt, J.L, Koopman, G.H., *A FEM Method for Determining The Acoustic Modes of Irregularly Shaped Cavity*, J. of Sound & Vib., Vol 45, pp. 495-502, 1976

[13] Richards T.L. & Jha, S.K., *FEM Method for Studying Acoustic Characteristic Inside a Car Cavity*, J. of Sound & Vib. Vol 63, pp. 61-72, 1979

[14] Nefske, D.J. & Howel, L.J., *Automotive Noise Reduction Using FEM,* Transaction of the Society of Engineers, Vol 87, pp. 1726-1737,1978

[15] Dowel, E.H., *Master Plan for Prediction of Vehicle Interior Noise*, American Institute of Aeronautics J.,Vol 18, pp. 353-366, 1980

[16] Harris, C.M., *Handbook of Acoustic Measurement & Noise Control*, 3rd edition, McGraw Hill, 2000