TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

ACTA TECHNICA NAPOCENSIS

Series: Applied Mathematics and Mechanics
Vol. 55, Issue IV, 2012

VIBRO-ACOUSTIC ANALYSIS OF PASSENGER COMPARTMENT

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Abstract: *In the present paper, are presented the acoustics sources for overall interior sound of car and the transfer path for each source. These acoustic and vibration sources can be classified in structure borne and air-borne noise. The structure-borne noise contains the noise coming from the excitation forces and transmitted from the vibrating sources to the vehicle body through the attachment points. The air-borne noise is the radiated noise from the source which is transferred in the air through panels of the car body to passenger compartment. In the second part of this study, to find the natural frequencies and the associated modes shapes for both structure and air cavity, the normal mode analysis of the structure and the air has been performed by using finite element analysis . In order to do the analysis, a prototype has been built to simulate the interior cavity of a medium size automobile.*

Key words: *structure-borne, airborne, finite element analysis, normal mode analysis*

1. INTRODUCTION

Considering the customer demands regarding acoustic performance, the legal regulations on noise emission levels and human exposure to noise, it can be said that the comfort into passenger compartment has become an important commercial target.

In figure 1, it can be observed that the interior noise level in vehicle has been decreased in time [1].

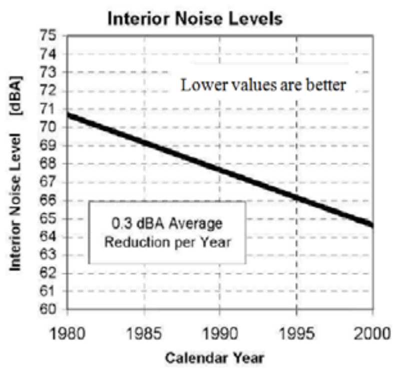

Figure 1.Interior noise trend

Analyzing the car interior noise issues, it can be said that, the driver is a part of a vibro-acoustic system coupled via contact points in the vehicle - steering wheel, seat, floor panel and pedals. It is a coupled system – person and machine [4].

The organs and extremities of humans have certain resonances that could be excited via contact points within a vehicle. However, human responses to vibrations are varied and differ greatly over time and from one person to the other.

Vehicle overall interior noise is determined by a multitude of parameters, and is a sum of the contributions from all the noise sources/transfer path. The main sources which generate noise inside the car are: engine, power train, tyres/road, intake and exhaust system and wind turbulence. Considering the cabin interior noise, the aspects of the noise level in the car, the main sources and transmission paths and the noise level reduction are very important. Figure 2 from [1] shows a vehicle with the next elements: multiple noise sources, multiple receivers (passengers) and multiple transfer paths between the source and the receiver.

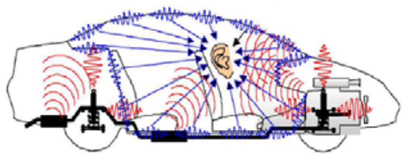

Figure2. Source-Path-Receiver Sound Propagation

The vehicle overall interior noise can be classified in structure-borne and air-borne noise [12]. The structure-borne contains the noise that comes from the excitation forces transmitted from vibrating sources to the vehicle body through the attachment points. The air-borne noise is the radiated noise from the source which is transferred in the air through panels of the car body to passenger compartment.

2. INTERIOR NOISE SOURCES AND TRANSFER PATHS OF VEHICLE

For a passenger car, which is stationary idle, interior noise is about 45 dB (A). When the light commercial vehicle runs with 110 km/h, the interior noise is about 70dB (A), and at maximum acceleration, the interior noise can rise until 78 dB. Taking in account the light commercial vehicles which runs with 130 km/h on noisy road surfaces, the driver may be exposed to continuous level excess of 80dB(A), [2].

2.1 Engine noise

Engine noise is composed of two elements: firstly, combustion noise, which is an important contributor to the diesel engines noise, and secondly, the mechanical noise which dominates the gasoline engines noise [3].

During the combustion period, gas forces in each cylinder are resulted, and also the cylinder pressure increases rapidly. These gas forces are transmitted to the structure of the engine and thus it is created a vibration which is then radiated as noise. One of the indicators used to combustion noise is the spectrum of cylinder pressure. In figure 3, the typical spectra for naturally aspirated engine and direct injection engine (NA-DI) diesel engine at full load, is represented [3]. On the other hand, in gasoline engines, contact surface between crankshaft and bearings through the oil film generates mechanical noise.

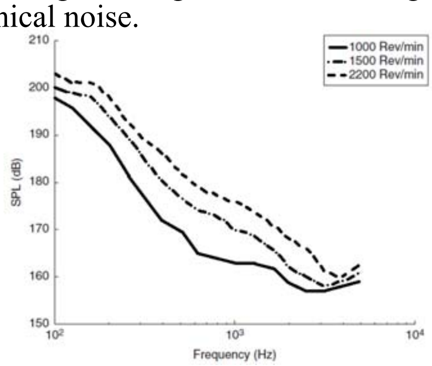

Figure 3. Effect of engine speed on cylinder pressure spectra

2.2 Tyre/Road noise

The tyre / road noise contains interior noise resulting from the contact between the tyres and the road, being transmitted to the interior by both airborne and structure-borne paths.

Vehicle interior road noise is mainly a low-frequency noise problem (<1000 Hz). Contributions are made by structure-borne noise paths through the vehicle suspension (<500 Hz) and direct airborne noise paths from the tyre through the vehicle structure (>500 Hz) [10].

At 250 Hz, the interior road noise level is around 60-65 dB. On the other hand, at frequencies greater than 1000 Hz, levels are typically 15 dB, lower than those at frequencies less than 300 Hz.

The several source generation mechanisms which create energy at the tire-pavement interface and then eventually radiated as sound, are tread impact, air pumping and slip-stick [9].

The resulting impact when the tyre enters in contact with pavement is shown in figure 4 [9]. This impact produces vibration of the tyre carcass and depends on the pavement texture.

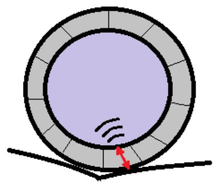

Figure 4. Radial vibrations caused by tread block / pavement impact

During the tyre and pavement contact, the phenomenon that occurs consists on one hand, in the deformation and compression of the tyre grooves and on the other hand, the air entrained in the passages of the tyre is compressed and pumped in and out (see figure 5). This phenomenon generates sound.

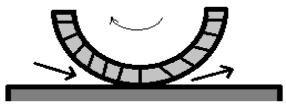

Figure 5. Air pumping at the entrance and exit of the contact patch [7]

Within the contact patch, result the horizontal forces. If their limit is higher, than the limit of friction will slip and stick to the pavement and will generate both noise and vibration (see figure 6).

Figure 6. Slip-stick motion of the tread block of pavement [7]

In figure 7 is represented the adhesion between the tyre blocks and the road during the exit of the contact patch, which generates sound and vibration.

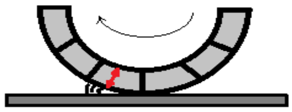

Figure 7.Adhesion between the tread block and pavement at the exit of the contact patch [7]

2.3 Wind (aerodynamic) noise

When the car runs at higher speeds, the wind noise is a very important source of interior noise, being generate in a number of different ways and having the highest dependency on road speed and becomes an issue in passenger cars above 110 km/h.

The sources of wind noise are depicted in figure 8. When the car runs on the road at high speeds the air comes in contact with the body structure, thus creating a turbulent boundary layer. This layer is exposed at pressure fluctuations and it generates noise in vehicle passenger compartment.

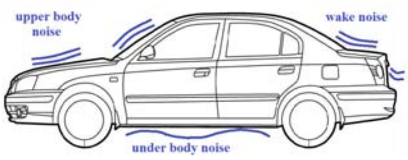

Figure 8. Sources of wind noise

The upper body noise is influenced by the fashion of the day for smooth body forms, while the underbody wind noise is a result of the reflectivity of the road surface.

Weak noise is generated by the meeting between the upper body and underbody airflow, which is transmitted to the interior of the vehicle through the window.

2.4 Intake and exhaust noise

The intake and exhaust noise issues happen at low frequencies, below 500 Hz and having predominantly low–frequency harmonic spectral characteristics.

When the inlet valves are closing, the fast moving of air charge is halted and in this way it results a pressure pulse that generates intake gas noise.

During combustion process, the exhaust valve open and releases the residual pressure. This action generates the exhaust gas noise. Exhaust gases flow through the exhaust system generating noise as well.

3. NORMAL MODE ANALYSIS OF THE STRUCTURE AND THE AIR

The interior acoustics is an important part in the design stage for automakers in order to reduce the sound level in the passenger compartment of automobiles. For this purpose Computer Aided Engineering (CAE) tools are used. In the first step, the 3D geometry of a structure that looks like a passenger compartment is designed. Starting from this 3D geometry, the structure and the air cavity are discretized by using proper finite elements. Then in order to identify the natural frequencies and the associated modes shapes, normal mode analysis of the structure and the air is performed.

3.1 Model creation

The side view of the cavity is shown in figure 9 with its dimensions given in millimeters. As illustrated in figure 9, the cavity is made of pressed sawdust plates having a thickness of 18 millimeters. In order to simulate the vibration of the firewall of the car, the corresponding firewall panel is a thin 1 millimeter thickness steel sheet. The vibration of the firewall was thought to be the most dominant source of noise in the cabin of the car, especially when the engine was running.

The material properties for the structure are assumed to be Young's modulus $E = 3620 MPa$, Poisson's ratio $\nu = 0.25$ and density $\rho = 638 kg/m^3$; the steel sheet properties are assumed to be Young's modulus $E = 210000 MPa$, Poisson's ratio $\nu = 0.3$ and the density $\rho = 7900 kg/m^3$; the air properties are density $\rho = 1.2kg/m^3$ and the speed of sound $c = 343m/s$.

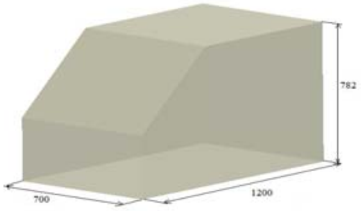

Figure 9. The structure under investigation

The geometry and finite element discretization of the enclosure has been done by using Hyper mesh software. The structure FEM model has approximately 30540 3D CHEXA elements corresponding for pressed sawdust walls and the steel plate has 630 2D CQUAD4 elements. The acoustic FEM model, which represents the interior volume of the structure occupied by the air, is meshed by using 75250 3D CHEXA elements.

3.2 The eigenvalue problem

The usual first step in performing a dynamic analysis is determining the natural frequencies and mode shapes of the structure with damping neglected. These results characterize the basic dynamic behavior of the structure and are an indication of how the structure will respond to dynamic loading. The natural frequencies of a structure or the resonance frequencies are the frequencies at which the structure naturally tends to vibrate if it is subjected to an excitation. The deformed shape of the structure at a specific natural frequency of vibration is termed by the normal mode of vibration or eigenmode. Each mode shape or normal mode is associated with a specific natural frequency.

When doing the normal modes analysis of the structure, the damping is neglected and the external forces are not acting, resulting [6][7], [13]:

$$M\ddot{Q} + KQ = 0 \qquad (1)$$

where Q is the vector of the system generalizes coordinates (system degrees of freedom), M, K are the mass (inertia) and stiffness matrices, which are maybe symmetrical, with (n x n) elements.

The following solution is proposed:

$$Q(t) = u\cos(\omega t - \varphi) \qquad (2)$$

where the ω is the natural frequency of the whole system and u is a constant n-vector of amplitudes.

By plugging the proposed solution in the system of differential equations, results:

$$\left(K - \omega^2 M\right)u = 0 \qquad (3)$$

The set of the homogeneous algebraic equations (3) has the unknown vector u. Considering $\lambda = \omega^2$ as a parameter, we get:

$$Ku = \lambda Mu \qquad (4)$$

known as the eigenvalue problem when trying to determine λ values for which the system (3) has nontrivial solutions.

By solving for λ nontrivial solution, the following characteristic equation (5) is obtained:

$$\det(K - \lambda M) = 0 \qquad (5)$$

where λ_r ($r = 1 \ldots n$) values are the eigenvalues of the system. For each determined eigenvalue λ_r, a vector u_r named eigenvector is satisfying the following equation:

$$Ku_r = \lambda_r Mu_r \qquad (6)$$

The results for the natural frequencies obtained from the simulation for the structure mode shapes are presented in Table 1. The associated mode shapes for the first three natural frequencies are presented in figures 10, 11 and 12.

Table 1. Natural frequencies of the structure

Mod	Frequency	Mod	Frequency
1	2.95 E+01	11	1.15 E+02
2	4.69 E+01	12	1.18 E+02
3	6.03 E+01	13	1.25 E+02
4	7.68 E+01	14	1.28 E+02
5	8.11 E+01	15	1.40 E+02
6	8.72 E+01	16	1.450 E+02
7	9.54 E+01	17	1.70 E+02
8	9.84 E+01	18	1.726 E+02
9	1.04 E+02	19	1.76 E+02
10	1.11 E+02	20	1.80 E+02

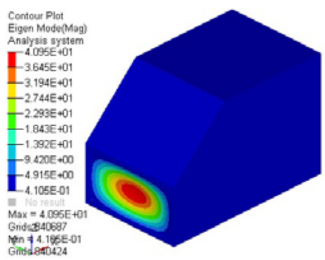

Figure 10. The first mode shape of the structure

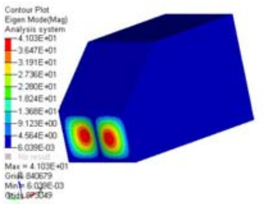

Figure 11. The second mode shape of the structure

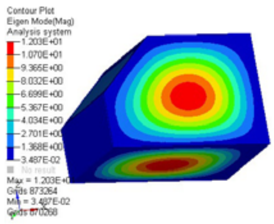

Figure 12. The third mode shape of the structure

For the fluid part, an acoustic normal mode can be interpretated as a sound resonance or natural frequency and the associated mode shape or sound pressure field. For simple regular shapes (l_x, l_y, l_z), the acoustic normal modes can be determined analytically, by solving the three dimensional wave propagation equation [2], [5]:

$$\frac{\partial^2 p}{\partial t^2} = c_0 \left(\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2}\right) \qquad (7)$$

where $p(x, y, z, t)$ is the deviation from the ambient pressure, c_0 is the sound speed in air.

In such enclosures, at the walls level, the waves are reflected and standing pressure waves are generated, thus these standing waves are considered the modes shapes of the air in the cavity. The mode shape is the following type:

$$p(x, y, z) = C_{n_x, n_y, n_z} \cos\left(\frac{\pi n_x}{l_x} x\right) \cos\left(\frac{\pi n_y}{l_y} y\right) \cos\left(\frac{\pi n_z}{l_z} z\right) \qquad (8)$$

where C_{n_x, n_y, n_z} is the modal amplitude of the acoustic mode, identified by de integer numbers n_x, n_y, n_z and x, y, z variables are confined inside the rectangular cavity: $0 \le x \le l_x$, $0 \le y \le l_y$, $0 \le z \le l_z$.

The modal frequencies of the cavity are [5], [9]:

$$f_{n_x, n_y, n_z} = \frac{c_0}{2} \sqrt{\left(\frac{n_x}{l_x}\right)^2 + \left(\frac{n_y}{l_y}\right)^2 + \left(\frac{n_z}{l_z}\right)^2} \qquad (9)$$

For more complicated enclosures only numerical methods implemented by using finite element or boundary element solvers can help in determining the acoustic normal modes.

For a normal mode acoustic analysis and the

Mod	Frequency	Mod	Frequency
1	1.51 E+02	11	4.37 E+02
2	2.26 E+02	12	4.51 E+02
3	2.45 E+02	13	4.63 E+02
4	2.70 E+02	14	4.82 E+02
5	2.87 E+02	15	4.90 E+02
6	3.21 E+02	16	5.01 E+02
7	3.33 E+02	17	5.13 E+02
8	3.65 E+02	18	5.20 E+02
9	3.92 E+02	19	5.17 E+02
10	4.04 E+02	20	5.40 E+02

uncoupled fluid, one can observe the results in the Table 2 and in the figures 13 through 15.

Table 2. Natural frequencies of the air

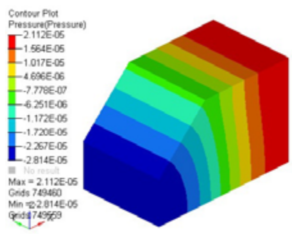

Figure 13. 1st longitudinal eigenmodes (151.03 Hz)

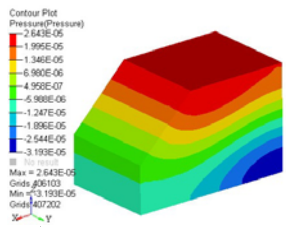

Figure 14. 1st vertical eigenmodes (226 Hz)

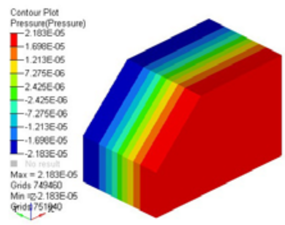

Figure 15. 1st transversal eigenmodes (245 Hz)

4. CONCLUSIONS

In the present study it is shown that interior noise is generated by a series of complex mechanisms and the individual noise excitations are transmitted from their source to the passenger compartment by a complex transfer system through numerous paths.

Taking into account that the passenger compartment acoustic cavity resonances can be either excited by power train or by road input, an acoustic normal modes analysis for an acoustic cavity which resembles with interior passenger compartment was performed.

The resulted vibration and acoustic normal modes (eigenvalues or mode shapes) are presented.

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ANALIZA VIBRO-ACUSTICĂ A HABITATULUI AUTOMOBILULUI

Rezumat: În prezenta lucrare, sunt prezentate sursele acustice pentru sunetul interior total al automobilului precum şi calea de transfer pentru fiecare sursă. Aceste surse acustice pot fi clasificate în zgomotul transmis prin structură şi cel transmis pe calea aerului. Zgomotul transmis prin structură conţine zgomotul produs de forţele de excitaţie şi care este transmis corpului automobilului prin punctele de prindere. Zgomotul transmis pe calea aerului este zgomotul radiat de surse şi transferat în aer către panourile habitaclului. În a doua parte a acestui studiu, sunt determinate frecvenţele naturale şi formele modale asociate atât pentru structură cât şi pentru aerul din cavitate, folosind analiza cu elemente finite. În acest scop, a fost construit un prototip care simulează cavitatea interioară a unui automobil de mărime medie.

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