Vehicle Cabin Noise Simulation due to High-frequencies Stimulation

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Abstract

In this paper, the acoustic environment in a vehicle cabin under the influence of high-frequencies aerodynamic sources has been studied. Some panels on the windshield, the roof, the doors, the front pillars, and the floor of a vehicle simulated as input source of noise when the car is moving at high speed, i.e. 112 km/h. The status of vehicle cabin in each of these modes has been studied and compared to each other. There are some methods to simulate acoustic behavior of a vehicle cavity such as Finite Elements or Statistical Energy Analysis methods. A brief overview for Statistical Energy Analysis (SAE) is stated. In this study, the statistical energy method is used for determination of acoustic analysis. Auto SEA software is used to simulate and estimate the amount of sound pressure level. In addition, sound pressure formulation presented and used for comparison in vehicle cabin points and with experimental results for validation. Also, considering viscoelastic materials, a common form of material non-binding panel has determined. The result shows that the roof is the most important panel in acoustic analysis under influence of aerodynamic sources. Accordingly, this panel has more effectiveness in optimization to control sound pressure level in a vehicle cabin. In addition, the amount of reduction in sound pressure level (SPL) in the cabin with viscoelastic material is presented as it could diminish the vibration of plates. In addition, the effect of using acoustic glasses is presented. Finally, the SPL effect of passenger position including front and rear is investigated and compared..

Keywords: *Acoustic vibration, Sound pressure level, Vibration, High frequencies, Viscoelastic material***.**

INTRODUCTION

Nowadays, the comfort and safety of automotives has gained more importance than ever. Noise levels are some of the qualitative parameters of the automotive. Experimental noise control methods are time consuming and costly. In the last few years, numerical methods, due to their high speed and accuracy, are very common methods for testing noise control.

In this paper, considering the limited frequency in finite element methods and boundary element methods, the statistical energy analysis (SEA) method is used.

Although the finite element method is an efficient tool for testing the modeling issues related to acoustic vibrations and low frequencies, but this technique is not usable at high frequencies, even with powerful computers. For high frequencies, the compact size finite element should be reduced on account of the small size of the wavelength. Therefore, for high frequencies, this method is costly, time consuming and produces high errors. In high frequencies, the results are also very sensitive to changes in parameters. Therefore, the finite element method is suitable for limited to low frequency range.

In the last few decades, new methods have been developed that express moderate dynamic behavior of the statistical methods, especially in high frequencies. Since the 1960s, the statistical analysis of energy (SEA) described in this method has been an accepted method for the analysis of structural acoustic systems. SEA is a statistical method of the energy complex structure and is divided into several subsystems based on the power balance in each sub-system, which was derived from the basic concepts of statistical mechanics, acoustic rooms, wave propagation and modal analysis. These subsystems include the power input, power dissipation in the system and the power's exchange between subsystems. The overall vibration response of each subsystem is presented in [1]. In 1983, Buchheim showed that the most important source of noise is aerodynamic noise in 100mph velocity [2]. Of course, when high power of engine is needed the noise of engine is an important source of noise in a body vehicle. However, in high velocities the major source of noises is aerodynamic noises [3]. Deye and Lee reviewed the process of noise transmission in the membrane and the formation and transformation of sound from the membrane [4]. Using viscoelastic materials reduces panel vibrations, which reduce the noise component, and this reduction is expressed through the passive method [5]. Noise controlling by means of viscoelastic materials is considered passive control. Passive control is widely used on account of the active way's cost and complexity.

Sound varies in magnitude and frequency and it is normally convenient to give a single number measure of the sound by determining its time-averaged value. The time average of the sound pressure at any point in space, over a sufficiently long time, is zero and is of no interest or use. The time

$$\left(p^2(t)\right)_t = \frac{1}{T}\int_0^T p^2(t)dt \quad (1)$$

where (p2)t denotes a time average.

It is usually convenient to use the square root of the mean square pressure as (2):

$$P_{rms} = \sqrt{\left(p^2(t)\right)_t} = \sqrt{\frac{1}{T}\int_0^T p^2(t)dt} \quad (2)$$

This is known as the root mean square (rms) sound pressure. This result is true for all cases of continuous sound time histories including noise

and pure tones. For the special case of a pure tone sound, which is simple harmonic in time, given by P = $\underline{P}$ cos (wt), the root mean square sound pressure is:

$$P_{rms} = \frac{P}{\sqrt{2}} \quad (3)$$

where P is the sound pressure amplitude.

The range of sound pressure magnitude and sound power of sources experienced in practice is very large. Thus, logarithmic rather than linear measures are often used for sound pressure and sound power. The most common measure of sound is the decibel. Decibels are also used to measure vibration, which can have a similar large range of magnitudes. The decibel represents a relative measurements or ratio. Each quantity in decibels is expressed as a ratio relative to a reference sound pressure, sound power, or sound intensity, or in the case of vibration relative to a reference displacement, velocity, or acceleration. Whenever a quantity is expressed in decibels, the result is known as a level.

The decibel (dB) is the ratio R1 given by

$$log_{10}^{R_1} = 0.1 \quad (4)$$

and

$$10 \times log_{10}^{R_1} = 1\ dB \quad (5)$$

The sound pressure level Lp is given by

$$L_p = 10\ log_{10}\left(\frac{(p^2)_t}{P_{ref}^2}\right) = 10\ log_{10}\left(\frac{p_{rms}^2}{P_{ref}^2}\right) = 20\ log_{10}\left(\frac{p_{rms}}{P_{ref}}\right)dB \quad (6)$$

where Pref is the reference pressure, Pref = 20μPa = 0.00002 N/m2 for air. This reference pressure was originally chosen to correspond to the quietest sound (at 1000 Hz) that the average young person could hear [6].

1. Sound Pressure

With sound waves in a fluid such as air, the sound pressure at any point is the difference between the total pressure and normal atmospheric pressure. The sound pressure fluctuates with time and can be positive or negative with respect to the normal atmospheric pressure.
average of the square of the sound pressure, known as the mean square pressure, however, is not zero. If the sound pressure at any instant t is p(t), then the mean square pressure, (P2(t))t , is the time average of the square of the sound pressure over the time interval T [6].

2. Cubic model

The noise study of a complete model for a vehicle may have some difficulties in modeling and analysis. A more simplified model may be cubic model to be investigated in many researches. This model has some benefits as it can be altered in size and shape to match the desired form. It is common to be considered a hypothetical point near the passenger ear for control the sound pressure level (SPL) as there is not a unique SPL for all containers.

The acoustic cabin accompanies with the noise source and sensor position at point 'A' for noise measurement is illustrated in Figure 1. As shown in this figure, the simulation is accomplished with some data for dimensions and boundary conditions. The required data such as dimensions, boundary

conditions and inputs for modeling and analysis is attained from [7]. In this way, the results of proposed investigation can be compared and validated.

3. Viscoelastic material

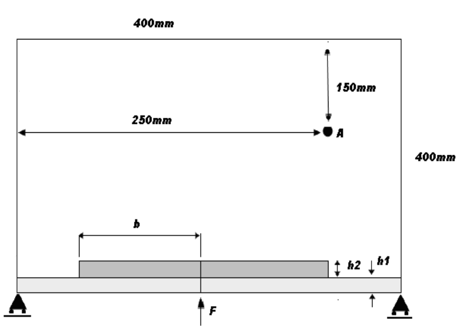

Fig1.. Rectangular acoustic cabin with two close pins

There are some passive ways to damp the vibration of a structure, which in turn reduce the generated noise. Unconstrained viscoelastic material is recently used in automotive industry in many parts of a vehicle such as body panels [8]. Some arrangements are observed in Figure 2 which can be used in plates [9].

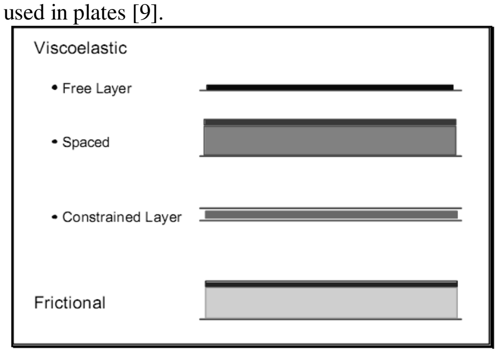

Fig2. Some viscoelastic material arrangements

In the automotive cabin, an unconstrained viscoelastic layer is used which may be considered in the compliance control software solution results. To evaluate viscoelastic material properties in the unconstrained state, noise reduction of an acoustic cabin is used in accordance with [7]. Here, the model has been leading of rigid walls on three sides, and one side of each plate has been leading aluminum. An aluminum plate that is 2 mm thick has pins on the sides and in the middle of it, and the unit force is applied. The viscoelastic material is 200 mm long, and the ISD-110 is 0.889 mm thickness. The specification of ISD-110 is shown in Table 1.

Table1. Specifications of viscoelastic materials used (specifications matching references [7])

Material specification	Poisson's ratio	Elastic modulus(MPA)	Density(kg/m^3)
0.3	1.794E+6	968	1

4. Statistical energy analysis (SEA)

In order to simulate and calculate of SPL, AutoSEA software is used as this software uses statistical energy analysis to process the practical model.

The Sound Pressure Level (SPL) at point A in the case damping page has been determined here in

frequency domain, 0~1000 Hz, and illustrated in Figure 3 for validation. Figure 4 also shows the

experimental test in the same model with similar dimensions and conditions from [7]. It may be observed that there is a close trend between both curves of figures especially in frequencies above 300 Hz. In addition, the max and min of SPL occur in similar frequencies.

Table 2 is also presented in order to compare the simulation accomplished here with the experimental SPL results in some specified frequencies.

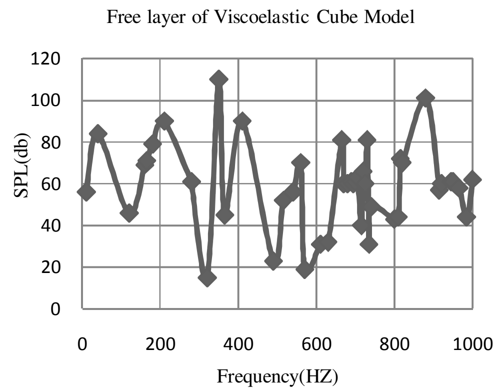

Fig3. Sound pressure level with unconstrained viscoelastic due to dynamic load

As shown in this table, there is a good agreement between the results showing that SAE method in the simulation could be also used for analysis in the acoustic cabin of automotive

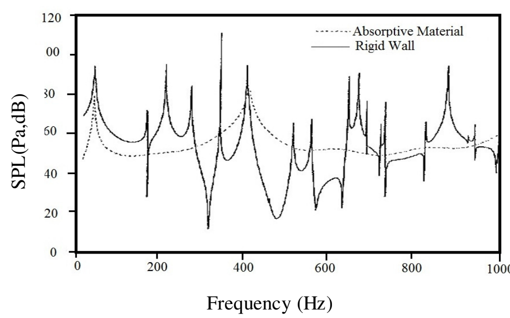

Fig4. Sound pressure level with non-binding viscoelastic due to dynamic load [7]

Table2. SPL of simulation with statistical energy method and experimental results [7] in specified some frequencies

Frequency (Hz)	SPL (Simulation)	SPL [7]
320	15	12
350	110	112
410	90	96
490	23	18
560	70	71
665	81	92
680	60	65
715	40	42
740	49	47
815	72	70
880	101	97
965	58	57

5. Acoustic analysis in vehicle cabin

A more practical model that is studied here is the box form of a sedan in practice, the exact size of a passenger automotive compartment is designed with the customer's opinion in mind. Thus, there is a small classification between standard sizes. The variety of models and sizes of automotives today is stunning, and automotive projects that are in competition with each other for customer satisfaction are presented in this study. The metal housing model that creates the size and thickness of the main components in Figure 5 has been simplified in this model.

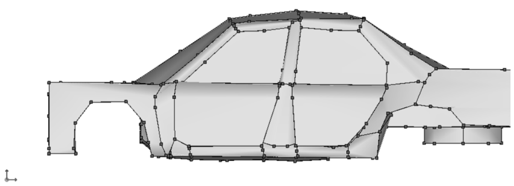

Fig5. Vehicle cabin model

Acoustic properties of air in the cabin, including the amount of medium density $1.21 kg/m^3$ and its speed is equal to $343 m/s$. The cavities are not anechoic and this may lead to some errors in our calculations. For example, it may not able to calculate the real and exact amount of SPL in the cavity. Steel casing properties include $2.1\times10^{11} MPa$ elastic modulus and density mass $7850 \,{}^{Kg}\!/_{m^3}$ and $\nu = 0.3125$. This type of steel is used in the most metal panels, such as the roof, floor, and the A-Pillar. All the glass material with elastic modulus $6.2\times10^{10} MPa$ and density $2300 \,{}^{Kg}\!/_{m^3}$ and $\nu = 0.24$ is considered. Therefore, the glass installation done in accordance with the definitions of the software is

considered on the acoustic performance. To stay consistent with the existing form, the body of the acoustic cavity was created according to Figure 6 and Figure 7.

Power train (engine and gearbox), tires, aerodynamic noises, exhaust, and body vibrations are the main sources in automotives [10].

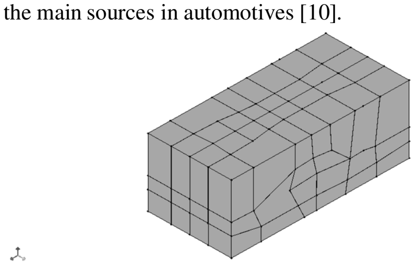

Fig6. Cabin acoustic model - exterior cavity

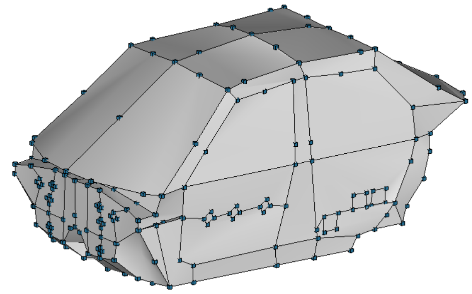

Fig7. Applied cabin acoustic models - interior cavity

The stimulation is applied to the incoming boundary layer to speed on the car panels. Nowadays, improvements in making engine and power transmission systems, especially in luxury vehicles, have reduced the sound. In this research, the boundary layer noise source is considered at 80 mph or 112 km per hour. This excitation is a distributed RMS pressure load that is characteristic of turbulent boundary layer flow. Spatial correlation of pressure is strong in the flow direction. Of this stimulation, the amount of sound pressure level in the area immediately surrounding the driver's head has been designated as the indicator for measuring noise from the perspective of the passenger cabin interior.

Firstly, the effect of windshield glass of the cabin is examined from the viewpoint of internal passenger. It may seem that glasses have lesser effect than metal panels, but it is interesting that if there is problem of sealing in glasses, their effect will not be smaller than metal panels. Velocity excitation is exerted with windshield. Velocity is assumed to be 80 mph or 112 km/h. Figure 8 shows the effect of input on windshield glass of applied model.

The calculated SPL within the range of 100~8000

Hz is illustrated in Figure 9 when the windshield of the car is stimulated in the positions as shown in Figure 8.

The overall amount which is an average of sound pressure level in the acoustic cavity is used to compare the SPL of interior cabin

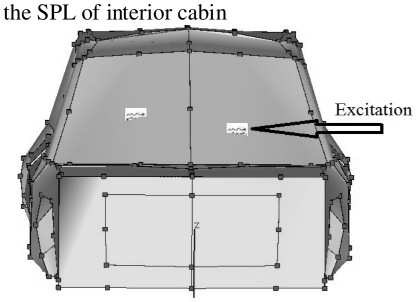

Fig8. Input stimulation on the windshield

. The amount of overall sound pressure in this situation is equal to 74.41dB.

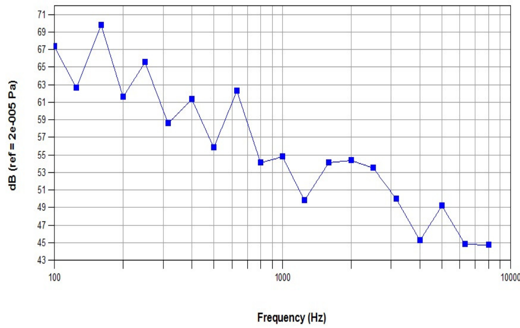

Fig9. SPL of acoustic cabin when windshield is stimulated

In the next step, the plate of car door is excited in high speed that may occur due to not well adjusted of hinges or locks. The driver could hear some noise in this situation.

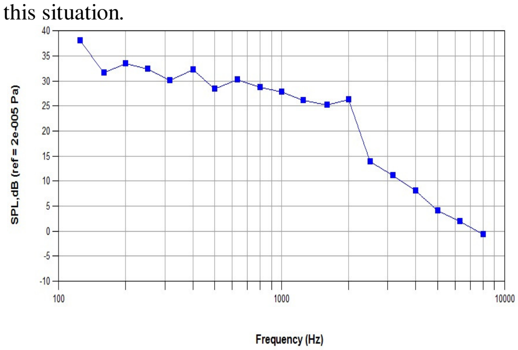

Fig10. SPL of acoustic cabin when doors are stimulated

Figure 10 is drawn to represent SPL near driver's head where the door is stimulated. The overall sound pressure level in this case is 42.7 dB.

In the next step, the panel on the roof is considered as input source of noise. In this case, a sound pressure level that is equal to the overall amount of 63.1 dB has been calculated. Figure 11 shows the status of the acoustic cabin close to the head of the driver when the excitation is located on the roof. When the values are as in the last situation, the effect of excitation on the roof has a considerably larger stimulating effect than it has on the doors.

For the last step, the stimulation rate with the same condition is applied to the front pillars (A-Pillar). Figure 12 illustrates stimulation on the A-Pillar as the source of excitation in the functional model.

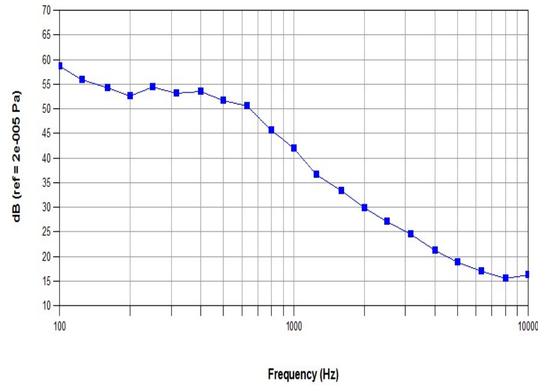

Fig11. SPL of acoustic cabin when roof is stimulated

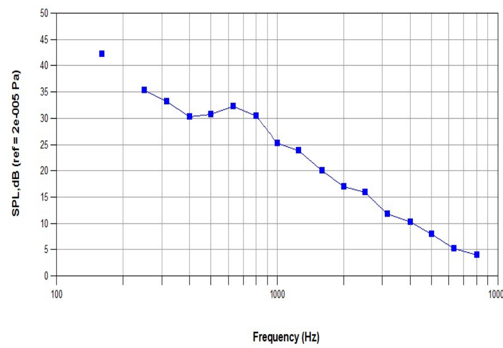

Fig12. SPL of acoustic cabin when A-Pillar is stimulated

The overall sound pressure level is 44.4 dB. When compared with the first case, i.e. roof stimulation, the decibel measurement from the doors was lower, but its effect is more motivated on the doors.

There can be helpful to compare between the different stimulation sources with a standard sound pressure level in the cabin at different frequencies. As is noted, when the stimulation is on the windshield and roof panels, the acoustic cabin has the higher level of sound pressure than doors and A-pillars. Table 3 displays the quantities of SPL for each case separately for comparison

The most influence the level of sound pressure in the driver's compartment near the ears caused by the stimulation of the automotive cabin's roof comparing with other panels except windshield. Thus, it can be concluded that in order to create the most effective noise reduction in the cabin, the roof panel is the most important consideration for using viscoelastic materials. So, to control and optimize the cabin noise caused by aerodynamic noise, it should be focused on the vehicle cabin's roof panel.

Table3. Sound pressure levels in cabin acoustic modes of stimulation

Excitation	SPL(dB)
Windshield	74.41
Doors	42.7
Roof	63.1
A-pillar	44.4

In this stage, in order to show the reduction of sound pressure level caused by using viscoelastc materials on the roof, a comparison chart of sound pressure level in the driver's head area is shown in Figure 13. The specification of ISD-110 that is used to reduce the amount of SPL is shown in Table 1.

The upper curve is the case without viscoelastic and the lower curve is for the case with viscoelastic. Overall value of sound pressure level in the frequency domain between 100 and 8000 Hz in the case without viscoelastic material is 63.1 dB. The amount of pressure with viscoelastic is 58.8 dB.

This study illustrates that the amount of acoustic noise in the cabin compartment near the driver's head using material ISD-110 with a thickness of 1.4 mm on the roof has decreased by about 6.8 percent of the total amount. As it is clear with increasing frequency, the index rate increases and, because of this trend of increasing frequency, sound pressure level is reduced.

In the final stage, an optimization is done by using acoustic glasses on the windshield. Sound damping glass used in this problem is comprised of 3 layers. External layers, tempered glass of the density $2500 Kg/m^3$, modulus of elasticity of $4.85\times10^{10}\ MPa$, Poisson ratio of v=0.499 and the thickness of 0.7 mm.

In this case, by setting input excitation unchanged, overall sound pressure decreases to 54.9 dB.

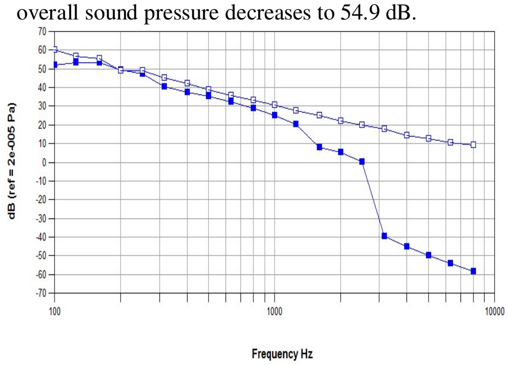

Fig13. Sound pressure levels around the driver's head without viscoelastic and with viscoelastic on the roof

Figure 14 shows that by using such glasses it may be effectively possible to control noise within cabin. In this figure, the curve with less SPL is for status when acoustic glasses have been used.

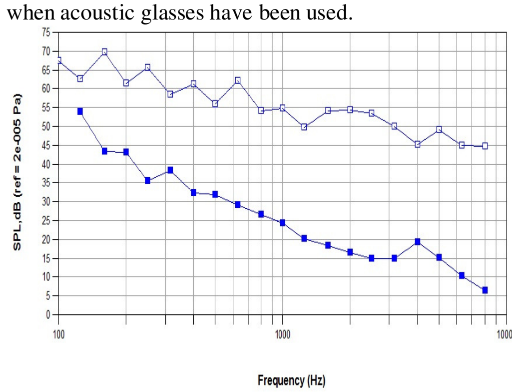

Fig14. Comparison of regular and acoustic glasses

6. . Acoustic behavior in front and rear positions

In the previous section, acoustic behavior is examined in the driver's head area.

In Table 4, a comparison between the level of sound pressure in the area of driver and rear passenger's head is presented up to 8000 Hz frequencies. This table is set in accordance with results of the solution of applied model without viscoelastic materials. It can be seen that the level of sound pressure in the area of rear passenger's head is about 17% higher than the level corresponding to front passenger's one.

Table4. SPL of front and rear passenger's head without viscoelastic

Area	SPL(dB)
Front Passenger's Head	59.92
Rear Passenger's Head	60.02

Using viscoelastic materials of the kind ISD-110 of the thickness of 1.4 mm, there may be determined Table 5 for SPL's as Table 3.

As shown in Table 3 and 4, the SPL of same position in the vehicle cabin can be decreased about 8 percent when viscoelastic material is used.

Table5. SPL of front and rear passenger's head with viscoelastic

Area	SPL(dB)
FrontPassenger's Head	54.78
Rear Passenger's Head	55.00

In addition to reviewing acoustic condition of rear and front passengers in desired intervals, it may be possible to assess level of sound pressure in an area lower than the head of driver. Table 6 shows the amount of SPL corresponding to rear and front passengers' waist.

Table6. SPL of front and rear passenger's waist without viscoelastic

Area	SPL(dB)
Front passenger's waist	50.58
Rear passenger's waist	51.47

The effect of using viscoelastic materials is also shown in Table 7 in the waist position.

Table7. SPL of front and rear passenger's waist with viscoelastic

Area	SPL(dB)
Front passenger's waist	46.17
Rear passenger's waist	47.06

Table 6 and 7 again represent that viscoelastic material can decrease SPL up to about 8 percent in another position of vehicle cabin.

Figure 15 shows the effect of using viscoelastic materials in different conditions of previously mentioned areas. Dark and bright bars correspond to non-viscoelastic and viscoelastic materials respectively.

As shown in this figure, within acoustic cabin, the SPL of passengers' waist is lower than SPL corresponding to the head position. In addition, the amount of SPL in rear passenger's position is higher than in front passenger's position. Moreover, it can be concluded that shorter passengers, compared to taller ones, experience lower amount of SPL.

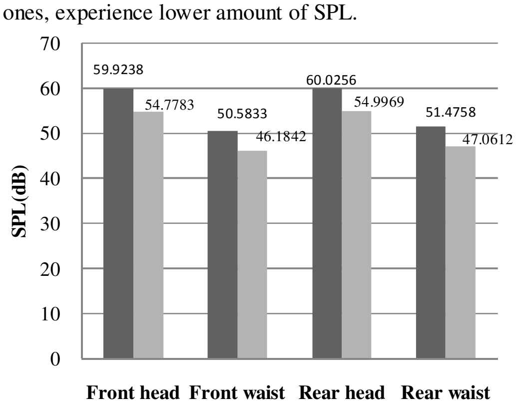

Fig15. Effect of viscoelastic materials in different positions of vehicle cabin

7. Conclusions

Some panels in a car stimulated as input source of noise when it is moving at high speed, i.e. 112 km/h. The vehicle cabin has been studied with statistical energy method for acoustic analysis.

The sound pressure level (SPL) of front and rear position in the vehicle cabin has been calculated when the windshield, the doors, the roof and the A-pillars are excited. It is resulted that the most effective noise as a point of SPL is windshield, then roof panel. In addition, the effect of viscoelastic material in noise reduction has been presented. The results show the amount reduction of SPL using Viscoelastic material with 1.4 mm thickness in the roof panel is about 4.3 dB, i.e. 6.8 per cent reduction.

Moreover, windshield has an important role in aerodynamic noises and it is considerably useful to control this part of automotive to reduce the amount of interior noises.

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