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A New Optimized Sound Package for the Vehicle Dash Panel

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In this paper, an optimized insulator for sound packaging of the vehicle dash panel is proposed. The combination of the micro perforated panel and porous layers has been selected to insulate the dash panel of a vehicle. The main advantages of the mentioned combination are light weight and various tunable parameters in comparison with other insulators. These provide significant flexibility to achieve an optimal performance for the noise attenuation of the vehicle cabin. Therefore, the parameters of the selected sound package have been optimized in order to achieve suitable sound absorption in a selected frequency range. Furthermore, the Genetic Algorithm (GA) is used to optimize the parameters. It can achieve more reliable and more accurate outcomes compared to the conventional method. Full vehicle SEA (Statistical Energy Analysis) simulations are used to evaluate the optimized sound package. The results indicate that the optimized concept has maximum sound absorption capability. Consequently, the proposed sound package improves the vehicle's engine noise reduction by 5 dB in comparison with un-optimized sample in mid and high frequency ranges.

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1. Introduction

Noise, vibration, and harshness (NVH) have become progressively significant vehicle attributes due to demands for enhancing vehicle refinement [1]. The levels of vehicle NVH are often used as perceived levels of quality by customers, particularly in the premium market [2-3]. Consequently, different techniques are utilized to reduce the level of vehicle NVH. In order to pick up an appropriate technique for NVH refinement, it is necessary to have a qualified appreciation of the process of noise and vibration transfer. This transfer can be regarded in three stages including source, transfer path and receiver.

In Automotive, the main sources of NVH are powertrain, wind, and road excitations. During idle condition and low speeds, the most important source of NVH is the powertrain system [4]. On the other hand, idle NVH is one of the major quality parameters that customer look into while buying the vehicle [5]. Therefore, decreasing the noise level of the automotive interior compartment due to engine source is a critical task especially in idle and low speed condition.

Usually, modifying the characteristics of the source represents the most effective solution. However, this requires planning in the very first stages of the development process and can sometimes create conflicts with the other functional requirements [6]. For this reason, the NVH performance targets can be achieved only by employing additional components along the transmission path or within the receiving environment.

Referring to the transfer path of NVH, there are two different categories for transmission of the powertrain noise in the vehicle including air borne noise and structure borne noise [7]. Commonly in the automotive, the structure-borne noise transmission path dominates at low frequency (<200 Hz) while the air-borne noise transmission path dominates above 500 Hz [8]. It is important to note that, human ears are very sensitive to middle and high frequency sound, and can easily perceive even low levels of sound in these ranges [9]. Therefore, the paper seeks to remedy the airborne noise problem. In the science of acoustical control, there are two kinds of noise treatments to address airborne noise including enhancing sealing and sound packaging [9]. This paper will focus on sound packaging and won't cover the sealing method in automotive applications.

In the sound packaging, the main task is to reduce the level of outside noise that penetrates through the automotive body. Therefore, the body should have good sound insulation. As stated above, the main source of NVH during idle and low speed is engine excitation. Consequently, appropriate sound packaging of the dash panel has significant importance. The conventional structure used in dash insulation is generally the combination of the barrier-absorber system [10]. The barrier is a heavy material such as Ethylene Vinyl Acetate (EVA) and the absorber layer is a porous soft material such as Poly Urethane (PU) foam. In automotive applications, however, heavier materials cannot always be adopted because of concerns over the total weight of the vehicle and fuel consumption [11]. Thus, it would be useful to identify lightweight acoustical treatments that can mitigate vehicle interior noise [12].

In the last decades, different lightweight dash insulators have been suggested for vehicle applications [13, 14]. Dual density insulator is one of the most popular sound packaging tools. The structure of the insulator is including a dense cap layer on the top and a lofted layer on the bottom [15]. Regarding the limited tunable parameters of the above mentioned insulator, it is difficult to achieve optimal performance.

Recently, a Micro-Perforated Plate (MPP) as an acoustical material has been introduced consisting of small holes, generally micro-sized, created in a thin plate [16]. The combination of MPP and absorber layer has been proposed by Parret et al [17]. The main advantages of the mentioned combination are light weight and various tunable parameters. These parameters including perforation size, plate thickness, hole diameter and etc. provide significant flexibility to achieve optimal performance for airborne noise attenuation [18].

Based on the aforementioned history, in this paper, the combination of MPP and porous layers have been used to insulate the dash panel of the vehicle. The sound absorption coefficient has been utilized for exposing the effectiveness of the insulator.

Acoustic insulation optimization is not a common prosses because of the inflexible design parameters. One of the few researches that have been done in this field is reference [19], in which the thickness of roof and floor panels, and glass, have been optimized by RSM.

In this paper, the optimization prosses will be more effective due to different tunable parameters of the combination of MPP and porous layers. Furthermore, the Genetic Algorithm (GA) is used to optimize the parameters of the insulator. It can achieve more reliable and accurate outcomes compared to the conventional method.

For optimization, an appropriate mathematical model of the insulator should be presented. The mathematical model for the acoustical behavior of the sound insulator has been known as an important research topic for several decades. The most important research for mathematical modeling of MPP materials has been conducted by Maa [20]. For the case of porous materials, several researches have been conducted in the literature. A complete theoretical description of wave behavior in porous material was also presented by Johnson-Champoux-Allard (JCA) which is known as the equivalent fluid model [21]. In this paper, the Maa and JCA models are used for modeling the MPP and the porous layers respectively. Two developed models have been connected through using the transfer matrix method. Then, the validity of the developed model is discussed by comparing the absorption coefficient of the presented model with existing experimental data.

In order to compare the performance of an optimized insulator, a vehicle model in VA-One software has been used. This software is based on the statistical energy method in which is an acceptable method for acoustic modeling.

The simulation results indicate that the optimized concept has maximum sound absorption capability. This performance could help with most attenuating the engine noise during different driving conditions.

2. Theoretical background

As stated above, the multilayer sound package selected in this paper consists of a porous material and a micro-perforated plate. A schematic model of the sound package is shown in Figure 1.

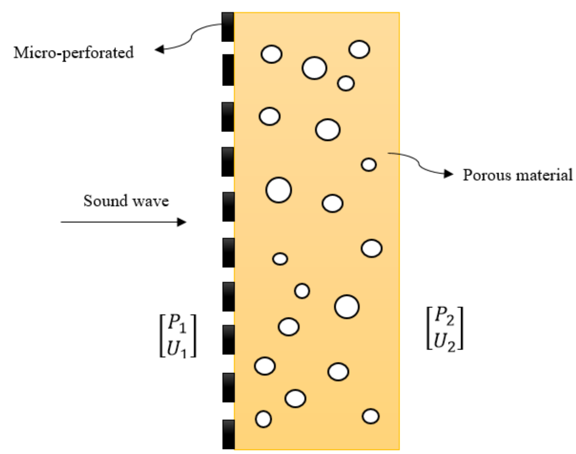

Figure 1: Schematic model for the selected sound package

The transfer matrix method is used to simulate the acoustic behavior of the sound package. The transfer matrix of each layer was separately calculated and then connected to each other to obtain the absorption coefficient. The input acoustic impedance of the MPP layer is calculated by Maa's model and also, the acoustic property of the porous absorber layer is calculated by the Johnson-Champoux-Allard (JCA) model.

2.1. Micro perforated panel

In recent years, the theoretical model for predicting the performance of micro perforated plates has been developed well, which provides an excellent opportunity to design and control the performance of these plates [22]

In order to model these plates, circular holes in MPP are considered as thin tubes. When a sound wave passes through a short thin tube with

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radius r_0 the partial velocity U is a function of the distance r, as shown in Figure 2. The connection between partial velocity in the hole and the applied acoustic pressure on the given thin tube is expressed by the partial air motion, Equation (1), [23].

$$\left(\frac{\partial^2}{\partial r^2}+\frac{1}{r}\frac{\partial}{\partial r}+K_{air}^2\right)U(r)=-\frac{\Delta p}{\mu h} \quad (1)$$

Where $K_{air}^2=-j\rho_0\omega\mu$, $\omega=2\pi f$ is the angular velocity, and f is frequency, μ is the dynamic viscosity of the air, h is the thickness of the thin tube, r is thin tube radial coordinates, Δp is the pressure difference between the front and back of the tube.

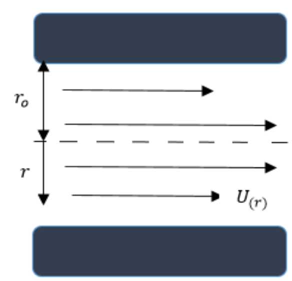

Figure 2: Schematic motion of the partial velocity in a thin tube with radius r_0 .

Equation (1) is a non-homogeneous differential equation and it could be solved by assuming a suitable boundary condition that is introduced by "Maa" [20]. Eventually, after solving Equation (1), the impedance of the MPP will be obtained:

$$Z_t=\frac{0.03675}{r_0^2}\frac{h}{\delta}\left(\sqrt{1+\frac{x^2}{32}}+\frac{2\sqrt{2}}{8}\frac{xr_0}{h}\right)+0.294\times10^{-3}\frac{j\omega h}{\delta}\left(1+\frac{1}{\sqrt{9+\frac{x^2}{2}}}+1.7\frac{r_0}{h}\right) \quad (2)$$

Where x is expressed as $2r_0\sqrt{f/10}$

2.1.1. Porous material

As stated in section 1, the motionless skeleton (equivalent fluid) is used as a suitable model for a porous material. The fundamental wave equation (Helmholtz equation) is used for describing a fluid behavior, [24]. This equation is completely defined by two parameters called complex density and complex bulk modulus. There are a few methods for defining these two parameters. In this paper, the porous layer was modeled as an equivalent fluid by using the Johnson-Champoux-Allard (JCA) model. The JCA model is a semi-phenomenological model describing visco-inertial dissipative effects inside the porous media [25].

In this model, some parameters such as dynamic tortuosity, dynamic permeability plus specific viscosity length and specific thermal length, are defined. The last two parameters are used to express dissipation due to viscosity and heat transfer between the various layers. Based on the results set in [24], in the JCA model, the specific density and specific bulk modulus values are obtained as follows:

Specific density, $\tilde{\rho}(\omega)$:

$$\tilde{\rho}(\omega)=\frac{\alpha_\infty\rho_0}{\phi}\left[1+\frac{\sigma\phi}{j\omega\alpha_\infty\rho_0}\sqrt{1+j\frac{4\alpha_\infty^2\eta\rho_0\omega}{\sigma^2\Lambda^2\phi^2}}\right] \quad (3)$$

Where ϕ is porosity, σ is static flow resistivity of air, α_∞ is tortuosity, Λ is Viscous characteristic length, ρ_0 is the density of the fluid (air) and η is Dynamic viscosity.

Specific bulk modulus, $\widetilde{K}(\omega)$:

$$\widetilde{K}(\omega)=\frac{\frac{\gamma P_0}{\phi}}{\gamma-(\gamma-1)\left[1-j\frac{8\kappa}{\Lambda'^2C_p\rho_0\omega}\sqrt{1+j\frac{\Lambda'^2C_p\rho_0\omega}{16\kappa}}\right]^{-1}} \quad (4)$$

Where γ is the ratio of specific heats, C_p is the specific heat capacity at constant pressure, P_0 is atmospheric pressure (static), κ is the thermal conductivity of air and Λ' is the thermal characteristic length.

After calculating $\tilde{\rho}(\omega)$ and $\widetilde{K}(\omega)$, it is possible to determine equivalent impedance ($Z_{eq\text{-}porous}$) and wave number ($k_{eq\text{-}porous}$) [26]:

$$Z_{eq\text{-}porous}=\sqrt{\tilde{\rho}(\omega)\widetilde{K}(\omega)} \quad (5)$$

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$$\mathrm{k}_{\text{eq-porous}} = \omega\sqrt{\tilde{\rho}(\omega)/\tilde{K}(\omega)} \tag{6}$$

2.1.2. Transfer matrix

The transfer matrix method is a common approach to simulate the acoustic properties of a multilayer media and obtain the absorption coefficient. This method can relate two points of the acoustic field in a media, as can be seen in Figure 1, the following equation can be written by using the transfer matrix method:

$$[\mathrm{F}_1] = \mathrm{T}_{total}[\mathrm{F}_2] \tag{7}$$

Which,

$$[\mathrm{F}_1] = \begin{bmatrix}\mathrm{P}_1\\ \mathrm{U}_1\end{bmatrix} \tag{8}$$

$$[\mathrm{F}_2] = \begin{bmatrix}\mathrm{P}_2\\ \mathrm{U}_2\end{bmatrix} \tag{9}$$

$$[\mathrm{T}_{\text{total}}]=[\mathrm{LP}][\mathrm{MPP}]=\begin{bmatrix}\mathrm{T}_{11} & \mathrm{T}_{12}\\ \mathrm{T}_{21} & \mathrm{T}_{22}\end{bmatrix} \tag{10}$$

Where $[\mathrm{T}_{\text{total}}]$ is the total transfer matrix of the sound package, [LP] is the limp porous material's transfer matrix and [MPP] is the MPP's transfer matrix.

As mentioned in [24], transfer matrix of porous material and MPP can be written respectively as follows:

$$[\mathrm{LP}] = \begin{bmatrix}\cos\left(\mathrm{k}_{\text{eq-porous}}\mathrm{h}\right) & \mathrm{j}Z_{\text{eq-porous}}\sin\left(\mathrm{k}_{\text{eq-porous}}\mathrm{h}\right)\\ \frac{\mathrm{j}}{Z_{\text{eq-porous}}}\sin\left(\mathrm{k}_{\text{eq-porous}}\mathrm{h}\right) & \cos\left(\mathrm{k}_{\text{eq-porous}}\mathrm{h}\right)\end{bmatrix}$$

$$[\mathrm{MPP}] = \begin{bmatrix}1 & \mathrm{Z}_t\\ 0 & 1\end{bmatrix}$$

(11)

The absorption coefficient of a sound package can be calculated by elements of the total transfer matrix, equation 10, and can be defined by [28]:

$$\alpha = 1 - |\lambda|^2$$

(12)

where λ is the reflection coefficient and can be expressed by the total transfer matrix elements as [27]:

$$\lambda=\frac{\mathrm{T}_{11}-\rho_0 c_0\mathrm{T}_{21}}{\mathrm{T}_{11}+\rho_0 c_0\mathrm{T}_{21}}$$

(13)

Also, the reflection coefficient has the following relationship with the normalized acoustic impedance:

$$\lambda=\frac{\frac{Z_s}{\rho_0 c_0}-1}{\frac{Z_s}{\rho_0 c_0}+1}$$

(14)

Eventually, Sound absorption can be shown in terms of $(Z_s/\rho_0 c_0)$ [28]:

$$\alpha = \frac{4\mathrm{Re}(Z_s/\rho_0 c_0)}{[1+\mathrm{Re}(Z_s/\rho_0 c_0)]^2+[Im(Z_s/\rho_0 c_0)]^2}$$

(15)

Where Re and Im represent the real and imaginary parts, respectively.

2.1.3. Model validation

In this section, the above model presented is validated based on the experimental results. The absorption coefficient of the combined layers has been selected as the performance response for validation. In order to validate the proposed model, experimental results have been extracted from reference [26] with properties of MPP listed in Table 1. Referring to Equation (15), the absorption coefficient as the function of frequency is computed.

In Figure 3, the results of the theoretical model and experiment have been compared. According to [26-28], the average error of less than 10% is acceptable performance. As shown in Figure 3, there is a good agreement between the theoretical and experimental results of the paper and the average error percentage is acceptable.

Table 1: properties of modeled layers for validation

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Layer	Thicknes s (mm)	Air flow resistivity ($N.s/m^4$)	Perforation ratio (%)	Hole Diameter (mm)
Absorber	4.5	214000	-	-
MPP	1.07	-	3.5	0.8

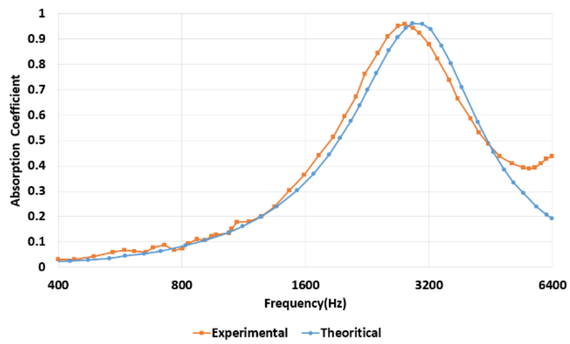

Figure 3: Validation result of the experimental and theoretical model

3. Optimization

Optimization is not a very common scenario for improving insulator performance due to limited tunable parameters. However, some new insulators, such as the combination of MPP and absorber layer provide flexibility to achieve an optimum sound package. It could lead to the sound package with the best possible performance and the least possible weight. In this paper, the absorption coefficient of the insulator has been selected as the performance index. The target is to maximize the absorption coefficient in a selected frequency range to achieve better acoustic performance under certain conditions.

Different mathematical algorithms can be used to optimize the parameters of a problem to achieve a more accurate outcome. Genetic Algorithm (GA) is one of the most powerful methods in the optimization fields. This algorithm is appropriate because of its acceptable accuracy and computational time. Also, the algorithm will not be caught in the local minimum traps and can find the absolute minimum. For the above reasons, GA is used to improve the acoustic performance of the selected sound package. The GA will act as follow to achieve the best performance:

Firstly, this method selects some points in the problem's domain randomly and creates the first generation of the population. In the second step, multiple points of the population are selected so that they have more suitable output values. These points can be known as parents. By performing the fertilization and mutation procedures on the inputs, a new generation of points will be created. After choosing a new generation, the termination conditions are going to be checked. If one of the termination conditions is established, the system identifies the best chromosome as the answer to the problem. Otherwise, the second generation will be returned to the algorithm cycle to resume the procedure. This cycle continues until a termination condition is reached.

As stated, the absorption coefficient is selected as the performance index of the sound package. The maximum absorption coefficient at each frequency is equal to one. It means that all the released energy at that frequency is absorbed. The absorption coefficient is a function of the sound package's physical parameters, and can be written as follows:

$\alpha = f(h.Pr.r_0.D)$
(16)

Which:
h = MPP thickness
Pr = MPP perforation ratio
r_0 = MPP hole diameter
D = Absorber thickness

The mentioned parameters can be defined as the inputs of the function. In other words, the absorption coefficient can be optimized using the above parameters.

For the objective function, the summation of the absorption coefficient over the whole frequency is the first choice. This selection means that the absorption coefficient should be maximized in a wide band of the frequency range. However, from a physical point of view, it is not possible to attain a maximum absorption coefficient over the whole frequency range.

Therefore, in this paper, a critical band of the frequency range is selected and the objective

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function is defined compatible with the selected critical band. The critical band of frequency range could be an area that the human ear is the most sensitive or a frequency band that engine noise is very loud. Therefore, the general form of the objective function can be written as:

$$J = \int_{f_{min}}^{f_1} \alpha_f df + \int_{f_1}^{f_2} W \alpha_f df + \int_{f_2}^{f_{max}} \alpha_f df \quad (17)$$

Where α_f is the absorption coefficient at each frequency, f_{min} (f_1) is the lower limit of whole (critical) frequency band and f_{max} (f_2) is the upper limit of the whole (critical) frequency band. Furthermore, W is the weighting coefficient in the critical band of frequency.

Obviously, when the absorption coefficient is closer to 1, the desired sound package provides better performance, so the maximum value of the J function should be computed. By finding the maximum of J, the absorption coefficient can be optimized for the sound package. In the critical band of frequency, the selected weighting coefficient (W) is used to magnify the effect of the optimization.

In this paper, try and error used to select the weighting coefficient. The chosen value had the best effect in the critical band, while it does not destroy the performance in the rest of the frequencies. The limits for the input parameters are listed in Table 2. Furthermore, the total thickness of the insulator is 15mm.

In order to visualize the effect of the above mentioned method, the selected insulator could be optimized for three different critical frequency bands. These critical frequency bands have been listed in Table 3. The absorption coefficient curve has been shown for three optimized insulators in Figure 4. It is clear that for each insulator, there is a distinct peak compatible with the related critical band.

Table 2: Input values limits

Parameter	Lower limit	Upper limit
t	0.5 mm	2 mm
Pr	1	15
Hd	0.05 mm	3 mm

No.	f_1(Hz)	f_2(Hz)
Insulator 1	3000	3200
Insulator 2	4800	5000
Insulator 3	6000	6200

Table 3: The optimized frequency band

4. Results and discussion

Statistical Energy Analysis (SEA) is one of the standard analytical methods for predicting vehicle acoustic and vibration responses at high frequencies [29].

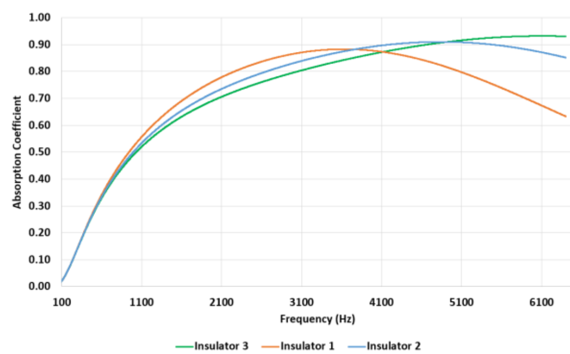

Figure 4: Comparison of insulators 1 to 3

This method has been widely used for different noise and vibration control problems [30]. Finite element and boundary element methods predict vibro-acoustic in the low-mid frequency range up to 500 Hz with good accuracy. However, in the case of the mid-high frequency range, statistical energy analysis can be used to model the NVH even for a complex system such as a vehicle [31]. SEA is an excellent method to model the acoustic spaces and solid material components of the vehicle and their interactions. Consequently, in this paper, the SEA has been utilized to evaluate the performance of the optimized sound package. The interior Sound Pressure Level (SPL) of automotive arising from different sources could be simulated by using the SEA. In this paper,

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VA-one as a vibro-acoustic software developed based on the SEA method is used to simulate the acoustic behavior. By using VA-one, the optimized sound package could be evaluated as the insulator of the dash panel.

Figure 5, shows a 3D model of a vehicle in VA-One software which comprised of roof, floor, dash and engine compartment. According to Figure 6, the interior space of the vehicle is subdivided into acoustic spaces called cavities. The acoustic behavior of automotive is simulated based on the energy balance among in flowing and outflowing energy between the cavities, and the energy loss inside the cavities.

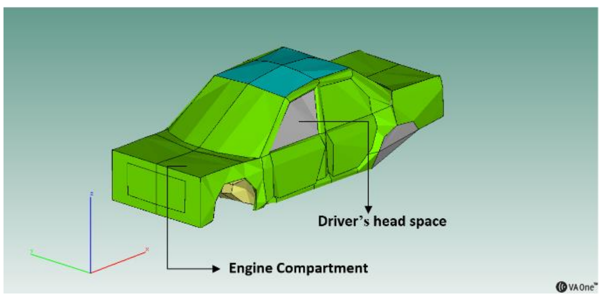

Figure 5: VA-One vehicle model

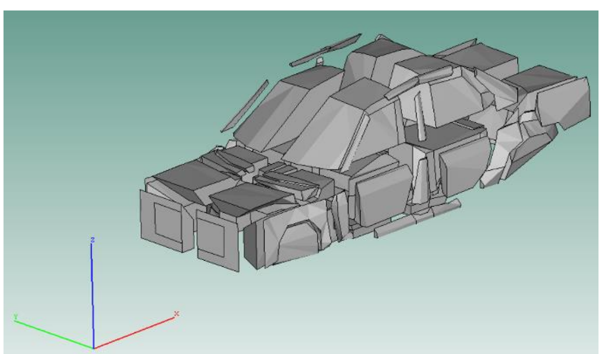

Figure 6: Acoustic cavities (shrink mode) in VA-One vehicle model

For simulating the interior cabin noise, at the first step, the full vehicle model has been built up similarly to what shown in figure 5 and figure 6. At the next step, the optimized sound package has been installed on the dash panel. Finally, a noise source is located at the engine compartment, and the sound pressure level of the interior cabin is measured at the driver's ear to evaluate the effect of the proposed optimization method.

For simulating the engine noise, two separate cases have been considered. Real engine noise in idle condition and white noise could be regarded as the noise sources in the engine compartment, and the effect of the optimized insulator is investigated in the mentioned cases.

5.1- Case 1: Real Engine Noise in Idle Condition

As stated in section 1, idle NVH is one of the major quality parameters that customers look into while buying the vehicle. Consequently, decreasing the noise level of the vehicle's interior compartment due to engine source is a critical task in idle conditions. There are different approaches to achieve this task. In this paper, the main focus is on sound packaging. As stated in the previous section, the critical band of frequency should be defined for the optimization. In order to make a better decision for selecting the critical band, the measured engine noise in a test bench is investigated.

In Figure 7, the third-octave diagram of engine noise has been depicted, which shows the measured sound of the vehicle idle condition in the engine compartment. Obviously, there is a great peak around 3000 to 3200 Hz, and it could have destructive effects on interior cabin noise. For this reason, the Insulator 1 (the absorption performance has been depicted in Figure 4) may be a good choice for dash panel insulation.

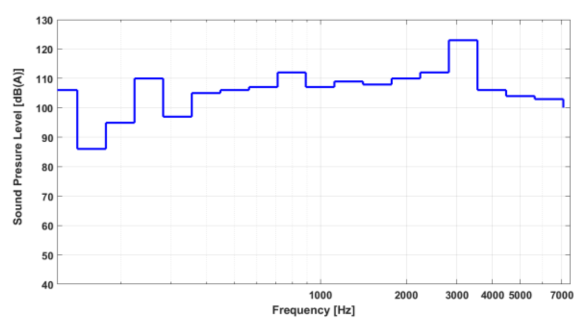

Figure 7: Third-octave diagram of engine noise

in idle condition

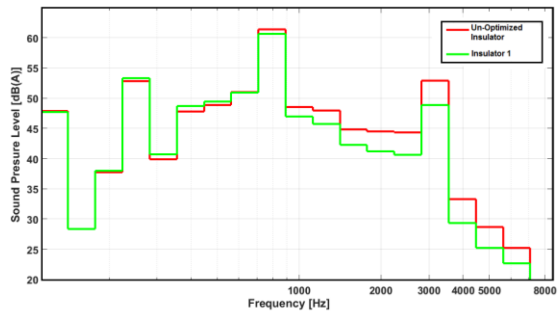

Figure 8: SPL at driver headspace with engine idle noise

In order to evaluate the performance of the insulator, the simulation has been conducted for two insulators, including un-optimized and Insulator 1. The simulation result of the transmitted engine idle noise near the driver's ear has been depicted in Figure 8.

It is clear that Insulator 1 has a better effect on the noise canceling in the mid and high frequency ranges. Obviously, the average sound level is lower than the un-optimized case.

5.2- Case 2: White noise generator placed in the engine compartment

In the previous sub-section, the effectiveness of the proposed optimized sound package has been proved in idle condition. According to Figure 7, there are some separated peaks at different frequencies. On the other hand, white noise is a randomly generated noise signal where the energy is equally distributed across the frequency range. Therefore, in this subsection, in order to have better judgment and a more accurate evaluation of the proposed insulator, the white noise generator has been placed in the engine compartment. The effectiveness of the insulator has been compared with an un-optimized insulator in Figure 9. Clearly, the optimized concept has superior performance in the wide frequency range. This performance could help with most attenuating the engine noise during different driving conditions.

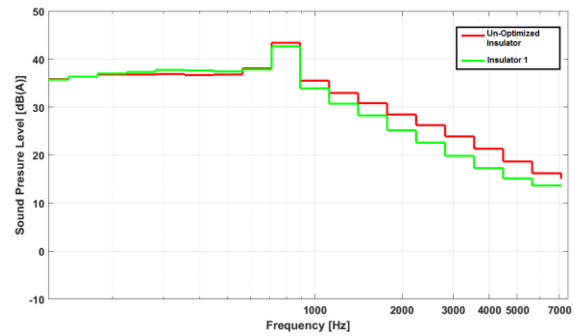

Figure 9: SPL at driver head space with white noise

5. Conclusion

This paper first develops the theoretical model for a multilayer sound package. The transfer matrix method has been used to simulate the acoustic behavior of the sound package. For each layer, the transfer matrix was separately calculated and then connected to obtain the absorption coefficient. It has been shown that there is an acceptable agreement between the presented theoretical model and experimental results. In order to optimize the performance of the proposed insulator, the Genetic Algorithm has been utilized. The optimization target is to increase the absorption coefficient in a critical frequency range. In order to evaluate the effectiveness of the proposed insulator, a full vehicle model has been built up by VA-one software and the proposed optimized sound package has been installed on the dash panel. For simulating the engine noise, two separate sources have been considered as the noise generator in the engine compartment including real engine noise and white noise. The effect of the optimized insulator has been investigated in two mentioned cases. The results show that the optimized concept has higher performance in comparison with an un-optimized insulator during different conditions. Particularly, in mid and high frequency ranges, the vehicle's engine noise reduction by 5 dB in comparison with un-optimized concept.

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