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Acoustical Design of Vehicle Dash Insulator

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ABSTRACT

The acoustical performance of a vehicle dash panel system is rated by the noise reduction, which is calculated from the sound transmission and absorption characteristics. A typical dash insulator consists of a steel panel (vehicle body panel), a porous decoupler and heavy layer in the form of sandwich construction. The use of dash panel is to block engine noise from entering into the interior cabin. In the present study the transmission loss of dash panel has been evaluated in reverberation chambers and the sound absorption of dash panel has been determined in impedance tube. This paper deals with improving over all sound transmission loss and shifting of the double wall resonance well below the engine firing frequencies by changing the decoupler materials such as felt and foams of different density and thickness and heavy layer mass per unit area. Numerical simulation of dash panel is also carried out for predicting sound absorption and sound transmission loss of different combinations. The simulation requires intrinsic properties of poroelastic and viscoelastic materials of dash panel. These properties are measured using standardized test rigs and inverse techniques. A parametric study is also carried out to check the effect of intrinsic parameters on sound absorption and sound transmission loss. Finally the simulated results will be validated with experimental results.

Keywords : *Noise, STL, Vehicle Dash Insulator, Numerical Simulation*

INTRODUCTION

Sound package design and optimization is becoming important in Modern vehicles as they provide quieter and greater comfort during ride condition. Headliners, molded plastic interior trim, seat fabrics, dash are all designed with this purpose in mind. The design of automobile dash is especially important due to its notable size and its place in between engine and passenger compartment. It blocks the engine noise from entering into passenger compartment and at the same time it serves as a passage for steering wheel, brake and clutch components etc through grommets.

For the years, design and optimization of dash insulator is a challenge to NVH engineers as it is a multilayered structure with pass-through with grommets and if these grommets are not properly designed then, these pass-throughs will affect adversely on acoustic performance of dash insulator [1]. The acoustic performance of dash insulator is measured in terms of sound absorption and sound transmission loss. The sound absorption coefficient can be measured using two microphone impedance tube method based on ASTM 1050 while the sound transmission loss of a dash insulator is measured as per SAE J1400 in Reverberation suite with anechoic termination facility [2,3]. The performance of a dash insulator can also be predicted using simulation techniques which require intrinsic physical parameters of sound package materials used in dash insulator. There are very few software's commercially available based on FEA and SEA techniques. In this study, a software package based on transfer matrix method is used to predict multilayered performance of dash insulator. First theory behind modeling of dash insulator is discussed in detail. Then experimental techniques to measure sound transmission loss and sound absorption are discussed, followed by a discussion on test rigs for intrinsic physical parameters. At the end, four different dash insulators are modeled and results are validated with measured sound absorption and sound transmission loss.

THEORETICAL DASH INSULATOR PERFORMANCE

A typical dash insulator made up of a barrier (heavy layer) of Ethyl Vinyl Acetate (EVA), Poly Vinyl Chloride (PVC) or Thermo Plastic Olefin (TPO) materials. The mass of the barrier is expressed in terms of surface density which in the range of 1.2-7.3 Kg/m^2. The barrier is followed by a decoupler layer to enhance the performance of dash insulator. The decoupler is generally a sound package material like foams and felts or combination of both. The thickness of decoupler varies from 6 mm to 25 mm. It separates barrier from steel firewall. The modeling of this sound package material requires Biot parameters for prediction of sound absorption and sound transmission loss [4].

The Biot parameters are five intrinsic physical parameters like porosity, flow resistivity, tortuosity, characteristic lengths and three mechanical parameters like Young's modulus, Poisson ration and loss factor.

POROELASTIC MATERIAL MODELLING

Biot's theory was originally developed for wave propagation in granular porous media and subsequently adapted for wave propagation in elastic porous sound absorbing materials. It relates the acoustical performance of the materials, typically measured in terms of absorption coefficient and transmission loss, to the intrinsic material and macroscopic intrinsic properties. It presents a powerful framework for the numerical modeling of stress waves propagating in an elastic-porous material as it explicitly accounts for the different wave types that are known to propagate in poroelastic materials.

According to Biot theory, elastic materials have two phases which support two longitudinal waves and one rotational wave. The combination of stress-strain relation and the dynamic equation that describe the motion of solid and fluid phases of elastic material can be written as in equation (1) and (2)

$$N\nabla^2\mathbf{u}+\nabla\left[(A+N)\theta^s+Q\theta^f\right]=\omega^2\left(\rho_{11}\mathbf{u}+\rho_{12}\mathbf{U}\right) \tag{1}$$

$$\nabla\left[Q\theta^s+R\theta^f\right]=-\omega^2\left(\rho_{12}\mathbf{u}+\rho_{22}\mathbf{U}\right) \tag{2}$$

where **u** is the vector solid displacement field and **U** is the vector fluid displacement field. $\theta^s=\nabla.\mathbf{u}$ and $\theta^f=\nabla.\mathbf{U}$ are volumetric deformations in the phases. $N=E/[2(1+\nu)]$ is the shear modulus with ***E*** being the in vacuo Young's modulus of the bulk solid phase and ***ν*** is the Poisson ratio; $A=\nu E/(1+\nu)(1-2\nu)$ is the first Lame' constant; $\phi K_c(\omega)$ is the positive and represents the coupling between the volume change of the solid and that of the fluid with ϕ being the porous material porosity. K_c is the bulk modulus of elasticity of the fluid in the pores that will be presented later. R relates fluid stress and strain and is assumed to equal to $\phi K_c(\omega)$ The parameters ρ_{11}, ρ_{12} and ρ_{22} are mass coefficients that account for the effects of non-uniform relative fluid flow through pores. These coefficients depend on the fluid and solid masses and inertial coupling.

Equivalent Fluid Model

Now considering porous material as a fluid with effective properties may be of interest in some situations and for some kinds of porous materials. Since the porous medium is considered as an equivalent fluid, Helmholtz equation becomes the governing equation. Thus, for an equivalent fluid with effective properties, one can write

$$\nabla^2 p+\omega^2\frac{\rho_c}{K_c}p=0 \tag{3}$$

where ρ_c and K_c are the effective properties of an equivalent fluid and this equation (3) represents the propagation of a single compressional wave through the porous medium. The wave number can be directly related to the effective density ρ_c and the effective fluid bulk modulus K_c in equation (4).

$$k_c=\omega\sqrt{\frac{\rho_c}{K_c}} \tag{4}$$

The fluid effective density ρ_c in the pores is frequency dependent and also depends on five porous material macroscopic properties like porosity (ϕ), flow resistivity (σ), tortuosity (α_∞) and characteristic lengths (Λ) and (Λ'). These parameters are related to Johnson-Champoux-Allard (JCA) model as given equation (5) and (6).

$$\rho_c=\rho_0\alpha_\infty\left[1+\frac{\sigma\phi}{j\omega\rho_0\alpha_\infty}\sqrt{\frac{4j\alpha_\infty^{2}\eta\omega}{\sigma^2\Lambda\,\phi}}\right] \tag{5}$$

$$K_c=\gamma P_0\left[\frac{\gamma-(\gamma-1)}{1+\dfrac{8\eta}{jN\Lambda_{pr}\omega\rho_0}\sqrt{1+j\rho_0\dfrac{\omega N_{pr}\Lambda'}{16\eta}}}\right]^{-1} \tag{6}$$

The characteristic impedance Z_c and propagation constant are predicted using the equation (7) and (8).

$$z_c=\sqrt{\rho_c K_c} \tag{7}$$

$$k_c=j\omega\left[\rho_c/K_c\right] \tag{8}$$

In this section, Transfer Matrix Method (TMM) used to predict acoustic behavior of sound package materials is explained in detail [5,6]. The general representation for a Transfer matrix of a single layer acoustic system (Fig. 1) is given in equation (9).

$$\begin{bmatrix} P_n \\ V_n \end{bmatrix} = \begin{bmatrix} T_1 \end{bmatrix} \begin{bmatrix} P_{n+1} \\ V_{n+1} \end{bmatrix} \tag{9}$$

Where P_n is sound pressure and V_n is sound velocity and T_{11}, T_{12} T_{21} and T_{22} are four pole parameters or transfer matrix elements. For foam and fibrous materials of thickness *d*, the transfer matrix is given as in equation (10).

$$T = \begin{bmatrix} \cos(k_c.d) & j/z_c .\sin(k_c.d) \\ j z_c.\sin(k_c.d) & \cos(k_c.d) \end{bmatrix} \tag{10}$$

where Z_c and K_c are characteristic impedance and complex wave number respectively. The total impedance Z_s is given in equation (11)

$$Z_s = jz_c \coth.(k_c \ d) \tag{11}$$

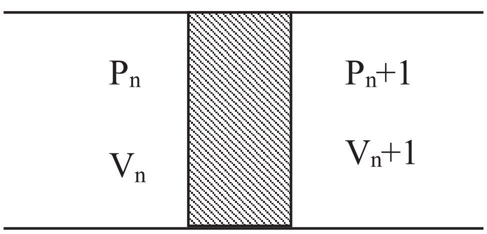


Figure 1. Schematic of Transfer Matrix

For multilayer configuration, the overall Transfer Matrix, *T*, is obtained by multiplying the above matrices for required configuration.

Similarly for films and foils, the transfer matrix is represented equation (12)

$$T_2 = \begin{bmatrix} 1 & Z_f \\ 0 & 1 \end{bmatrix} \tag{12}$$

where Z_f is normalized impedance of the film.

For multilayer configuration, the overall Transfer Matrix, *T*, is obtained by multiplying the above matrices for required configuration and given by equation (13).

$$[T] = [T_1][T_2]\ldots\ldots.[T_n] \tag{13}$$

When the pressure amplitudes for the incident, reflected and transmitted sound waves on the surface are A, B, C and D respectively, the complex amplitudes of the pressure and particle velocity, that is, the state variables, on the surface of the acoustic system can be expressed in terms of matrix elements and the and for the right end plate, as follows in equation (14).

$$A + B = P_1 = T_{11}P_{n+1} + T_{12}V_{n+1} \tag{14}$$

$$(A - B)/\rho_0 c_0 = V_1 = T_{21}P_{n+1} + T_{22}V_{n+1} \tag{15}$$

$$Ce^{-jkd} + De^{jkd} = P_{n+1} = T_{11}P_{n+1} + T_{12}V_{n+1} \tag{16}$$

$$\frac{Ce^{-jkd} - De^{jkd}}{\rho_{00}c} = V_{n+1} = T_{21}P_{n+1} + T_{22}V_{n+1} \tag{17}$$

Since the particle velocity $V_{n+1} = 0$ on a rigid wall, the pressure reflection coefficient $R = B/A$ can be expressed by the transfer matrix elements as as in equation (18).

$$R = \frac{T_{11} - \rho_0 c_0 T_{21}}{T_{11} + \rho_0 c_0 T_{21}} \tag{18}$$

The normal incidence sound absorption for an absorbing material with rigid backing is given by in equation (19).

$$\alpha = 1 - |R|^2 \tag{19}$$

Similarly, the transmission coefficient $T = C/A$ can be calculated and is expressed as in equation (20) [7].

$$T = \frac{2e^{ikd}}{T_{11} + \frac{T_{12}}{\rho_0 c_0} + \rho_0 c_0 T_{21} + T_{22}} \tag{20}$$

Using this equation, normal incidence transmission loss can be predicted by equation (21).

$$STL = 10.\log\frac{1}{|T|^2}\ , \text{dB} \tag{21}$$

The random incidence sound absorption coefficient and transmission loss can be evaluated by considering random incidence and integrating over angles 0^0 to 90^0 using Paris's formula in equation (22 and 23).

$$\alpha_{random} = \int_{\theta_{min}}^{\theta_{max}} \alpha(\theta).\cos\theta.\sin\theta \ d\theta \tag{22}$$

$$RSTL = 10.\log\frac{1}{\int_{\theta_{min}}^{\theta_{max}} |T(\theta)|^2.\cos\theta.\sin\theta \ d\theta} \tag{23}$$

Where θ is the limiting angle varying between θ_{min} and θ_{max}. Generally the limiting angle is between 70^0 to 85^0.

EXPERIMENTAL TECHNIQUES

In this section, experimental techniques used to measure intrinsic physical parameters and acoustic parameters are discussed in detail. As discussed above, the performance of sound package materials can be predicted with prior measurement of five intrinsic physical parameters like porosity, flow resistivity, tortuosity, etc. and three mechanical parameters like Young's modulus, Poisson ratio and loss factor. The experimental measurement of these parameters requires specialized test rigs like; porosity is measured using an air porosity meter based on Boyle's law [8]. Flow resistivity can be measured using flow resistivity test rig based on ASTM C522 standard [9]. In flow resistivity test setup, the apparatus consists of an air compressor, a flow meter and a differential pressure-measuring device. The test sample is mounted in an airtight sample holder which is then fixed into a mounting plate through which the air flows. The volume velocity of airflow through the specimen, and static pressure difference between the faces of the specimen with respect to the atmosphere are the measured quantities. These measurements are then used to calculate the specific airflow resistance and airflow resistivity. A parametric study shows that variation in flow resistivity values has considerable effect on sound absorption and sound transmission loss of sound package materials. Fig. 2 shows effect of flow resistivity on sound absorption coefficient.

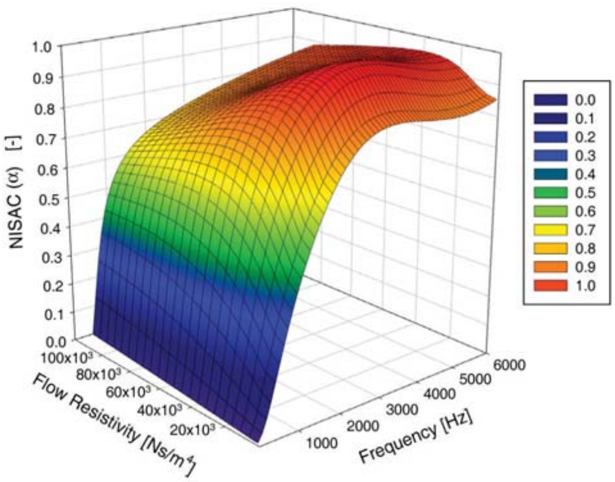


Figure 2. Effect of Flow Resistivity on Sound Absorption Coefficient

In Fig. 3, effect of flow resistivity on sound transmission loss is shown. From these results, it is clear that flow resistivity is one of the most important parameter which governs acoustic behavior of sound package materials. Tortuosity and characteristic lengths are inverted using optimization technique based on Genetic Algorithm [10]. This technique requires prior measurement of sound absorption

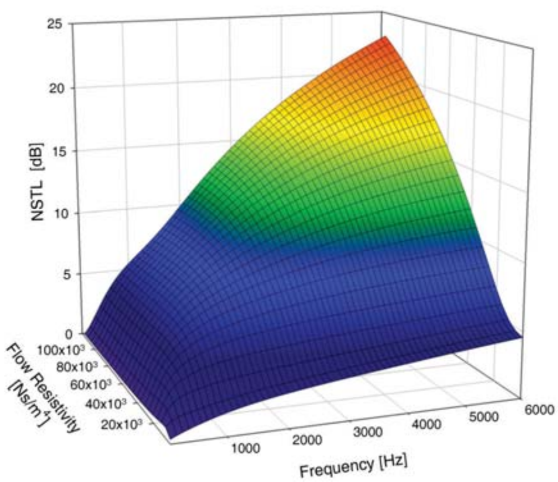


Figure 3. Effect of Flow Resistivity on Sound Transmission Loss

coefficient with surface impedance in two microphone impedance tube. Then this experimental data with porosity and flow resistivity is used to fit a mathematical model. The global solution of this optimization problem will give tortuosity and characteristic lengths. Measurement of mechanical parameters requires quasi-static test set up or laser Vibrometer and transfer functions measured across the sample give mechanical parameters. In this study only five physical intrinsic parameters are considered for simulation as mechanical parameters does not affect acoustic behavior of sound package materials. The barrier materials are viscoelastic in nature. The mechanical parameters of these barrier materials are measured using Oberst bar method as per SAE J1637 [11]. This method is based on 3 dB half power method.

The acoustic absorption of the sound package materials is measured using a two microphone impedance tube in accordance with ASTM E1050. This method is rapid and requires only a small size sample of the material. The test uses an impedance tube with a sound source connected to one end and the test sample mounted within the tube at the other end. The specimen holder is a detachable extension of the tube and makes an airtight fit with the end of the tube opposite the sound source. Random noise is generated by a digital signal analyzer (FFT) and the acoustic pressure at two fixed locations close to the sample is measured using two pressure field microphones. Then applying FFT and using the complex acoustic transfer function from signals of two microphones to compute the normal incidence absorption and reflection coefficient. Similarly Reverberation chamber is used to measure random incidence sound absorption coefficient of large size samples like headliners, package trays etc. This test is based on standards ASTM 423 and ISO 354 [12,13]. In this room an Omni-directional source is placed which is used to create diffused field inside the reverberation room as shown in Fig. 4. A large sample of size $6m^2$ is used to measure random incidence sound absorption coefficient.

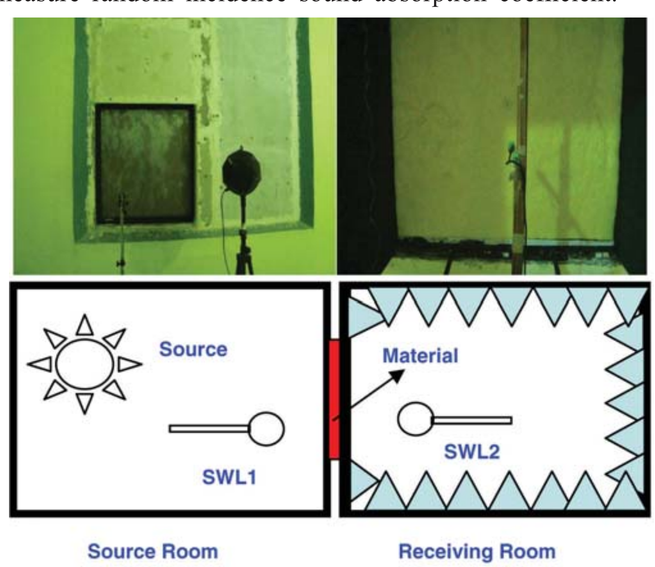


Figure 4. Sound Transmission Loss Measurement Setup-SAE J1400 Method

The sound transmission loss of dash insulator was measured using Reverberation chamber method as per SAE J1400. In small reverberation room source was placed while in large room anechoic wedges were mounted on the walls and transmitted sound was measured using microphone at five different locations. The sound transmission loss was calculated at one third octave band frequency as the difference between the Measured Noise Reduction (MNR) of the testing sample and transmission loss of reference sample which is taken as correction factor CF. The dash insulator was pasted on 0.8 mm steel plate using adhesive spray. The size of steel plate was 1.0 mm X 1.2 mm.

TL= MNR (unknown) - CF (24)

Normal incidence sound transmission loss of acoustic materials can be measured using four microphone tube as per standard ASTM E2611 [14]. It is based on transfer matrix theory of muffler design. The tube consists of two microphone tube at upstream with one more extended tube at downstream. The sample was placed in the middle of the tube and transmission coefficient was calculated using signals from four microphones. Finally transmission loss can be calculated using transfer matrix theory. In all the measurements, data was acquired using B&K PULSE data acquisition system.

RESULTS AND DISCUSSIONS

In this section of this paper, first normal incidence sound absorption coefficient measured in two microphone tube and sound transmission loss measured in four microphone tube are validated with simulated results using intrinsic physical parameters. Two types of materials foam and fibrous were chosen for this exercise. First was a melamine foam and other was a polyester felt. The thickness of both the materials was restricted to 20 mm as this is the maximum thickness used in automobiles. The physical intrinsic parameters used for simulation are given in Table 1.

Table 1. Physical Parameters of Porous Materials

Physical Parameters	Melamine Foam	Polyester Felt	Units
Thickness	20	25	mm
Density	8.8	90	Kg/m^3
Porosity	0.99	0.92	-
Flow Resistivity	10662	25000	Ns/m^4
Tortuosity	1	1.6	-
VCL	112	72	μm
TCL	137	158	μm

The sound absorption coefficient of both these materials was measured in an impedance tube and results for melamine foam are compared with simulation. The Fig. 5 shows comparison experimental and simulated results for melamine foam. Similarly normal incidence sound transmission loss is also compared with simulation and results are depicted in Fig. 6.

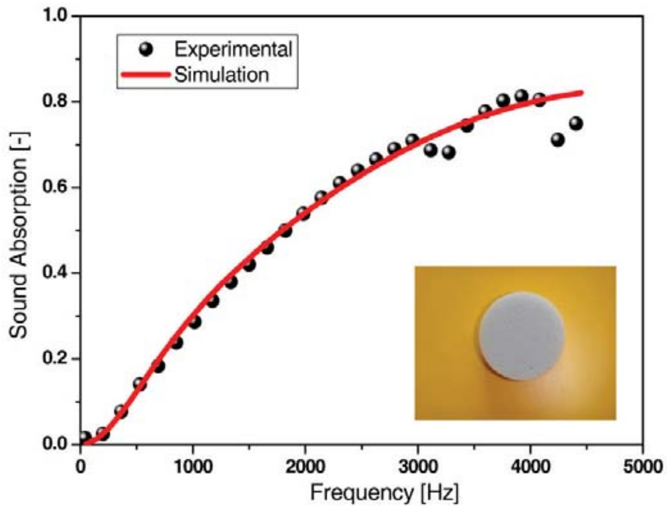


Figure 5. Comparison of Experimental Sound Absorption with Simulation for Melamine

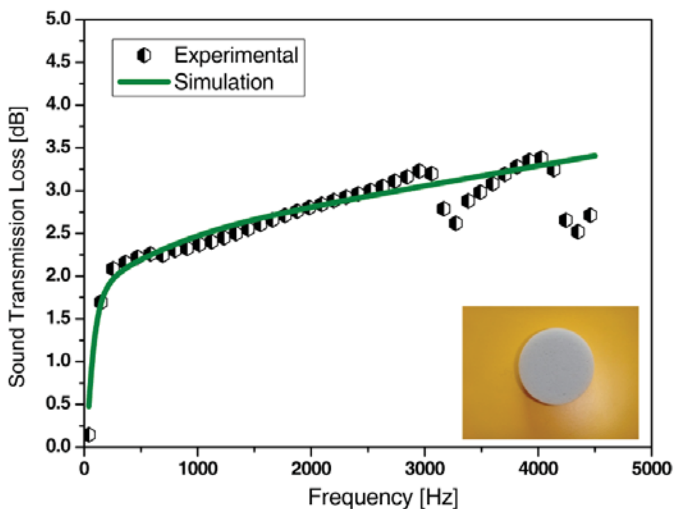
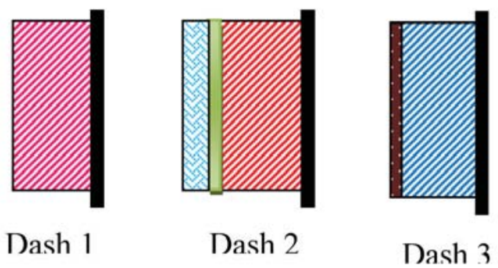


Figure 6. Comparison of Experimental Sound Transmission Loss with Simulation for Melamine

In Fig. 5 and 6, the correlation is very good except at 3300 and 4400 Hz. This is due to frame resonance of the melamine foam due to excitation. From these comparisons, it is clear that physical intrinsic parameters of porous materials can be used to predict sound absorption and sound transmission loss of porous sound absorbing materials.

DESIGN OF AUTOMOTIVE DASH

In this section, three dash insulators of different typologies are considered and their acoustic performance is evaluated in terms of sound transmission loss in Reverberation room and anechoic termination facility as per SAE J1400 with 0.8 mm steel plate. The acoustical performance of dash insulator affects significantly in the vehicle interior noise. In this paper, a study has been carried out to simulate acoustic behavior of dash insulator which can be later used to design new dash insulators for noise reduction. The noise reduction mainly depends upon sound transmission loss of the sandwich insulator. Unfortunately, this sandwich insulator acts as a mass-spring-mass system which resonates between 100 Hz to 500 Hz in the same frequency range as the engine firing frequency. The aim is to tune the system attenuation at resonance, while maintaining performance at middle and higher frequencies. The effectiveness of these products was enhanced to meet the requirements with proper selection of decouplers, heavy layer mass per unit area and use of simulation tool. The schematic configuration of dash insulators used in this study is shown in Fig. 7. The first automotive dash used in this study consists of nonwoven felt of 800 GSM. The sound absorption was measured in two microphone tube and sound transmission loss of dash was tested with steel plate as per SAE J1400. The Mechanical Parameters for Materials are given Table 2.



Dash 1 = 20 mm Material + 0.8 mm Steel Plate
Dash 2 = 10 mm Material + 28μm Film + 20 mm Material + 0.8 mm Steel Plate
Dash 3 = 1 mm Heavy Layer + 20 mm Material + 0.8mm Steel Plate

Figure 7. Different Types of Dash Insulators

Table 2. Mechanical Parameters for Materials

Mechanical Parameters	*Density*	**Young Modulus**	*Poisson ratio*	*Loss factor*
Units	[Kg/m^3]	[N/m^2]	[-]	[-]
Steel	7800	207x10^{11}	0.25	0.003
EVA	2000	-	-	-
PF Film	1400	-	-	-

The physical parameters used for simulation are given in Table 3. The acoustic performance of the dash is also predicted using theoretical models. In Fig. 8, experimental sound absorption for dash 1 is compared with simulation, which shows better correlation with experimental results.

Table 3. Physical Parameters for Dash 1

Physical Parameters	Nonwoven Felt	Units
Porosity	0.99	-
Flow Resistivity	17385.5	Ns/m^4
Tortuosity	1	-
VCL	146.5	μm
TCL	453.5	μm

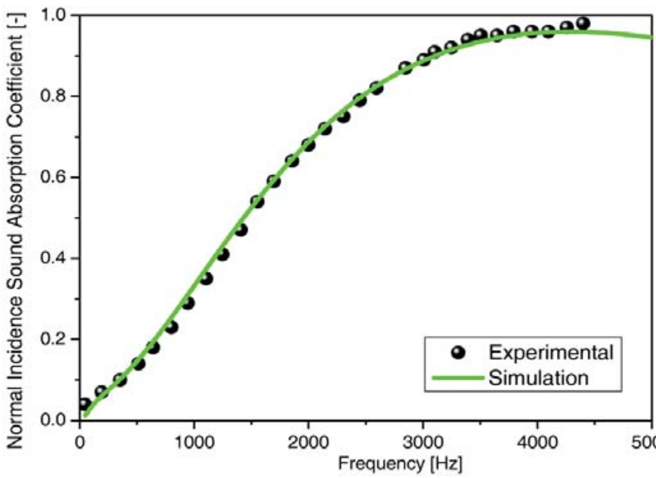


Figure 8. Comparison of Experimental Sound Absorption with Simulation for Dash 1

Fig. 9 shows that, for sound transmission loss there is very good correlation between experimental results and simulated results using mathematical models based on transfer matrix theory.

The second automotive dash used in this study consists three layers, compressed felt of 10 mm thickness followed by 2

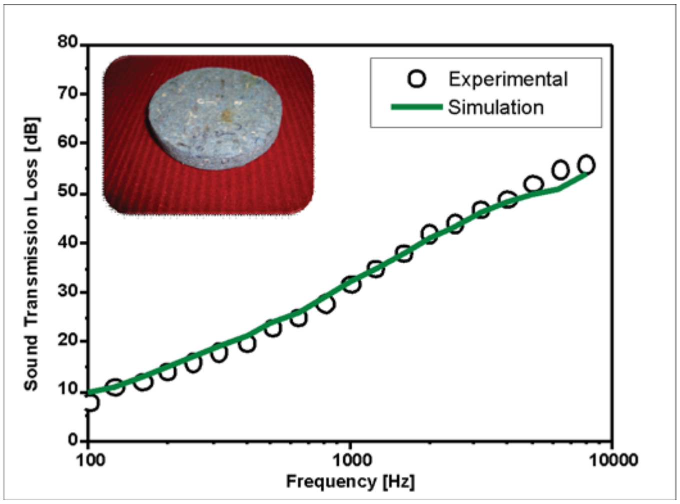


Figure 9. Comparison of Experimental Sound Transmission Loss with Simulation for Dash 1

layers of PVC Films of 28 micron and then nonwoven felt of 20 mm thickness as a decoupler. The physical properties are measured using experimental and inverse techniques and depicted in Table 4. Fig. 10 shows that, there is very good correlation between experimental results and simulated results using mathematical models based on transfer matrix theory. The physical parameters of porous felt and mechanical parameters of heavy layer used for simulation are depicted in Table 5. The acoustic behavior of the dash insulator was tested using SAE J 1400 test setup by gluing the sample with steel plate.

Table 4. Physical Parameters for Dash 2

Physical Parameters	Felt	Nonwoven Felt	Units
Thickness	10	20	mm
Density	-	-	Kg/m^3
Porosity	0.95	0.97	-
Flow Resistivity	36000	15000	Ns/m^4
Tortuosity	1.7	1.1	-
VCL	16	120	
TCL	180	120	µm

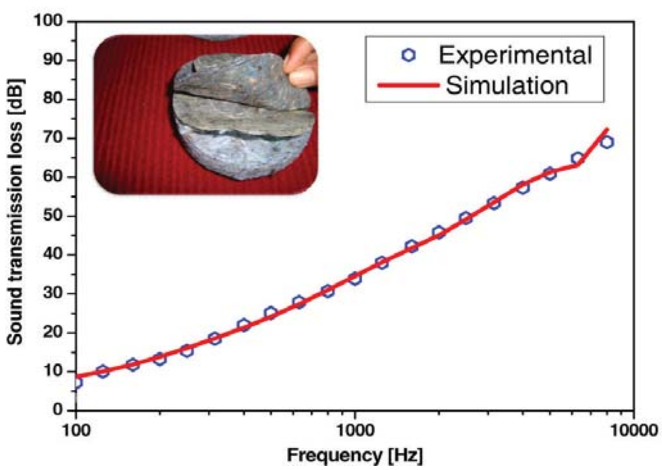


Figure 10. Comparison of Experimental Sound Transmission Loss with Simulation for Dash 2

The test results show that there is droop at 315 Hz and after that there is sharp increase in sound transmission loss of the multilayer. This droop is due to double wall resonance of the heavy layer-felt-steel plate system.

The performance was also predicted using mathematical modeling. The simulated results predicted also match with experimental results for third dash insulator. The correlation is shown in Fig. 11. From these three case studies, it is clear that simulation technique provides an easy to tool predict acoustic performance of the dash insulators at the design stage only. The simulated results compares very well with experimental results also. So this technique can be used to design dash insulators with very good acoustic performance also it can be used to optimize the dash insulator thickness and weight by doing permutations and combinations with different materials and thicknesses.

Table 5. Physical Parameters for Dash 3

Physical Parameters	Felt	Units
Thickness	22	mm
Density	-	Kg/m^3
Porosity	0.99	-
Flow Resistivity	16796	Ns/m^4
Tortuosity	1.16	-
VCL	410	µm
TCL	746	µm

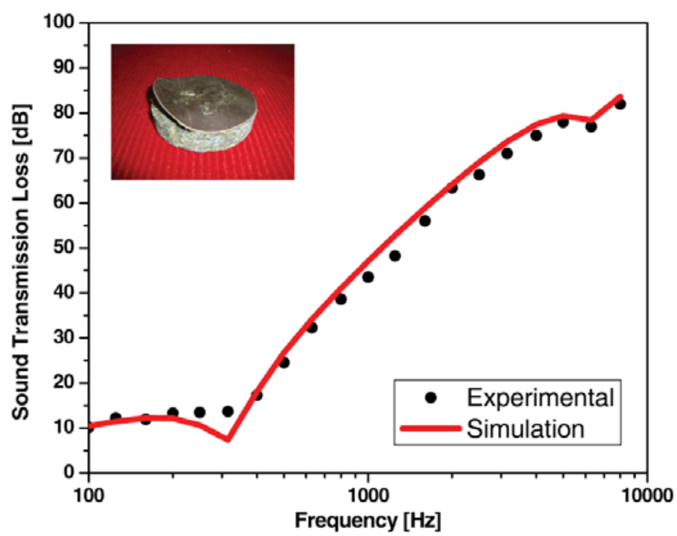


Figure 11. Comparison of Experimental Sound Transmission Loss with Simulation for Dash 3

In Fig. 12, a simulation is carried out to shift resonance droop from 315 Hz to lower frequency by changing mass per unit area of the heavy layer. Likewise different permutations and combinations can be carried out to shift resonance droop to lower frequency and increase over all sound transmission loss by increasing flow resistivity of the decoupler material.

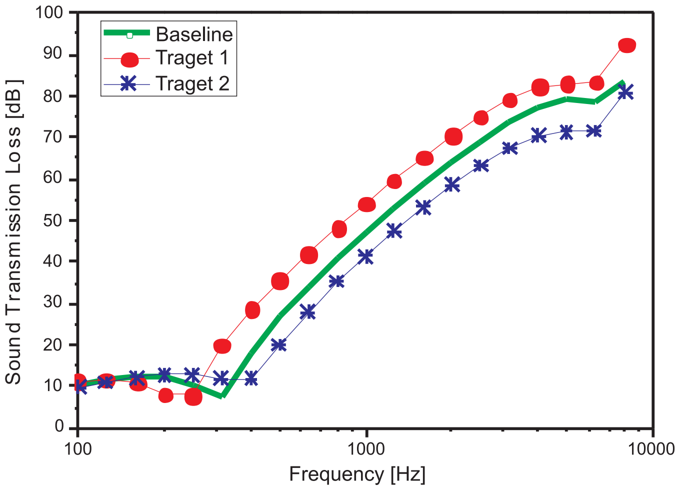


Figure 12. Shift of Resonance Dip at Low and High Frequency

CONCLUSIONS

This paper presents a detail discussion on design and development of dash insulator using simulation techniques for three different dash insulators. It also gives an overview of characterization techniques of sound package materials to find out intrinsic physical parameters required for simulation. It also discusses the details about the experimental evaluation of sound absorption and transmission loss of acoustic materials. From this study, it is clear that intrinsic parameters can be used to predict acoustic behavior of sound package materials. Simulation gives better understanding of a dash insulator in terms of sound absorption and sound transmission. It provides a quick solution to design engineers for optimization of sound package materials like headliner, dash insulators, and package trays as it saves time and cost of manufacturing and testing of materials.

REFERENCES

1. Julio Cordioli, Vincent Cotoni and Phil Shorter, “Numerical Investigation of the Transmission Loss of Seals and Slits for Airborne”, SEA Predictions, SAE Paper No. 2009-01-2205, 2009

2. ASTM E1050 - 08 Standard Test Method for Impedance and Absorption of Acoustical Materials Using A Tube, Two Microphones and A Digital Frequency Analysis System, 2008

3. SAE J1400-Laboratory Measurement of the Airborne Sound Barrier Performance of Flat Materials and Assemblies, SAE Standards, 2010

4. Biot, M A, “Theory of Propagation of Elastic Waves in a Fluid Saturated Porous Solid, I Low Frequency Range, II. High Frequency Range”, J. of Acoust. Soc. of Am., 28,168-191. 1956

5. Allard J F, “Propagation of Sound in Porous Media: Modelling Sound Absorbing Materials”, Elsevier, Ireland, 2010

6. Song B H and Bolton S J, “Effect of Circumferential Edge Constraint on the Acoustical Properties of Glass Fiber Materials”, Journal of Acoustical Society of America, 110, pp. 2902-2916, 2001

7. Lai H Y, Katragdda S and Bolton J S., “Layered Fibrous Treatments for Sound Absorption and Sound Transmission”, SAE Paper No. 972064, 1553–1560, 1997

8. Champoux Y, Stinson M R and Daigle G A, “Air-Based System for the Measurement of Porosity”, J. of Acoust. Soc. of Am., 89, pp. 910-916, 1991

9. ASTM-C522, “Acoustics-Materials for Acoustical Applications-Determination of Airflow Resistance”, 1991.

10. Paresh Shravage, Paolo Bonfiglio and Francesco Pompoli, “Hybrid Inversion Technique for Predicting Geometrical Parameters of Porous Materials”, Proc. of Acou. 08, June 29- July 04, Paris France, 2008

11. SAE J1637 Laboratory Measurement of the Composite Vibration Damping Properties of Materials on a Supporting Steel Bar, 2007

12. ASTM C423 - 09a Standard Test Method for Sound Absorption and Sound Absorption Coefficients by the Reverberation Room Method, 2009

13. ISO 354 – Acoustics - Measurement of sound absorption in a reverberation room, 2003

14. ASTM E2611 - 09 Standard Test Method for Measurement of Normal Incidence Sound Transmission of Acoustical Materials Based on the Transfer Matrix Method, 2009

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