Sound Package Materials for Vehicle interior NVH refinement

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ABSTRACT

Automotive car manufacturers are facing increasing expectations from customers for driving comfort as well as requirements in weight and cost reduction. These requirements make testing, modeling and optimization of sound package design an emerging challenge to NVH Engineers. The sound package materials like dash panel, carpet assembly, seats, headliner, package tray, hood liner and melt sheet can be used as absorbers, barrier and as damping treatments to reduce vibrations. In order to design and optimize sound package the characterization in terms of sound absorption, transmission loss and damping of materials is essential. The design of sound package materials is very crucial as it requires prior measurement of macroscopic physical and mechanical parameters of porous foams/fibrous materials using specialized test rigs. The simulated results using macroscopic parameters are also validated with experimental sound absorption and transmission loss. This paper presents different techniques available for characterization of sound package materials and also methodology for development of vehicle NVH refinement with case study.

INTRODUCTION

Sound package materials are wieldy used for noise control treatments in Transport industry. These sound package materials consists of poroelastic foams/fibers, films, foils etc. The poroelastic materials are formed from a continuous solid elastic material with open spaces or pores [1]. The pores are saturated with a fluid. The most well known examples are probably sound absorbing foams or fibers. Sound absorbing foams are very soft materials that are characterized with a very high porosity. Viscous and thermal interactions between fluid and frame occur due to the acoustic wave propagation through the porous material. These interactions are the basis of the sound absorbing properties of the sound package materials.

Sound Absorption- Sound absorption is defined as the ratio of reflected sound energy to incident sound energy. It reduces the sound build-up (reverberation) in a space due to reflections from hard surfaces. The schematic of sound absorption mechanism is shown in figure 1.

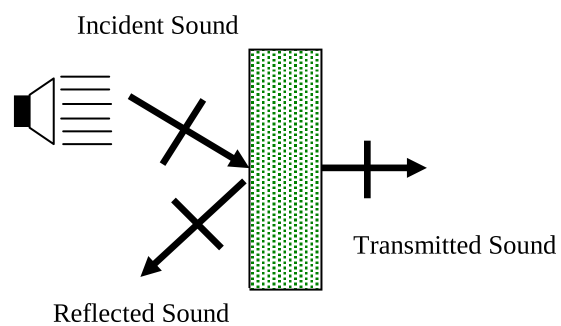

Figure 1. Sound absorption and Transmission Mechanism

When a sound wave strikes an acoustical material the sound wave causes the fibers or particles of the absorbing material to vibrate. This vibration causes tiny amounts of heat due to the friction and thus sound absorption is accomplished by way of energy to heat conversion. The more fibrous a material is the better the absorption; conversely denser materials are less absorptive. The sound absorbing characteristics of acoustical materials vary significantly with frequency. In general low frequency sounds are very difficult to absorb because of their long wavelength [2]. On the other hand, we are less susceptible to low frequency sounds, which can be to our benefit in many cases. The sound absorption coefficient (α) is given as follows. Depending upon applications, sound absorbers can be used as alone, with films, foils, scrims to increase the absorption.

$$\alpha = \frac{I_{reflected}}{I_{incidentt}} \qquad (1)$$

$\alpha = 1$ (If material is absorbing)

$\alpha = 0$ (If material is reflecting)

Sound Transmission loss – It is defined as the ratio of transmitted sound energy to the incident sound energy. The SI unit of sound transmission loss is dB. It blocks the sound propagation through the material from one side to other. The performance of single wall barrier of finite size is shown in figure 2 and it is governed by three regions.

1. Stiffness and Resonance controlled Region
2. Mass Region
3. Wave coincidence Region

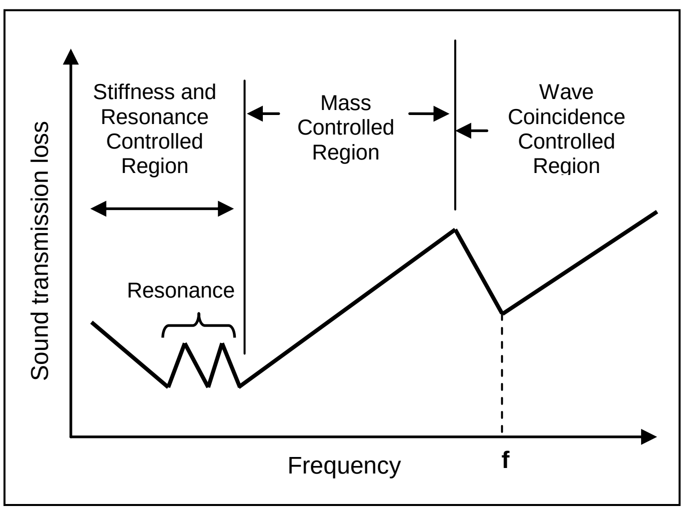

Figure 2. Sound Transmission Loss of single wall

When a sound wave impinges on an acoustical barrier some part of the sound wave will be reflected and some part of the sound wave will pass through the acoustic material. This pass out sound is transmitted sound and it depends upon material properties. Generally thicker walls will have less transmitted sound than thinner walls. The sound transmission properties of acoustical barrier materials vary significantly with frequency.

Sound transmission loss is given as

$$\text{STL} = 10\log\left(\left(\frac{1}{\tau}\right)\right) \quad \text{dB} \qquad (2)$$

$$\tau = \frac{Transmitted\ sound\ energy}{Incident\ sound\ energy}$$

Damping Materials

Damping refers to the extraction of mechanical energy from a vibrating system usually by conversion into heat. Damping serves to control the steady state resonant vibrations and to attenuate traveling waves in the structure. There are two types of damping: first one is material damping and system damping. Material damping is the damping inherent in the material while system or structural damping includes the damping at the supports, boundaries, joints, and interfaces etc. in addition to material damping. Viscoelastic materials have been used to enhance the damping in a structure in three different ways: free-layer damping treatment, constrained-layer or sandwich-layer damping treatment and tuned viscoelastic damper.

$$\eta = \frac{Energy\ lost\ per\ unit\ of\ time}{Energy\ stored\ in\ the\ vibrating\ systemm} \qquad (3)$$

MEASUREMENT TECHNIQUES

The acoustic performance of sound package materials is measured in terms of sound absorption and sound transmission loss. It can also be predicted using Biot's theory, which was originally developed for wave propagation in granular porous media and subsequently adapted for wave propagation in elastic porous sound absorbing materials [3]. It relates the absorption coefficient and transmission loss of sound package material with its intrinsic and geometric parameters. According to Biot theory, elastic materials have two phases which support two longitudinal waves and one rotational wave.

The sound absorption and sound transmission loss can be measured experimentally using impedance tubes and reverberation suites as per ISO and ASTM standards. The experimental methods are discussed in details in later section. Normal incidence sound absorption coefficient is measured as per ISO-10534–II and ASTM E 1050. These standards are based on two microphone tube method for foam, fibers - headliner, carpet, engine hood insulation, etc. Figure 3 shows two microphone impedance tube set up for the measurement of normal incidence sound absorption coefficient. The tube consists of a loudspeaker fixed at one end of the tube and used to excite the sample placed at other end of the tube. Random incidence sound absorption coefficient of large sample size of materials can be measured using Reverberation suites as per standards ISO-354 and ASTM C 423. In Reverberation room an Omni-directional source is placed which is used to create diffused field inside the reverberation room as shown in figure 4. A large sample of size 6 m^2 is used to measure random incidence sound absorption coefficient.

Figure 3. Two Microphone Impedance Tube

Figure 4. Reverberation Room

Sound transmission loss of acoustic materials can be measured using four microphone tube. It is based on standard ASTM E 2611 which is based on transfer matrix methods. The tube consists of two microphone tube at upstream with one more extended tube at downstream as shown figure 5.

Random incidence sound transmission loss of acoustic barriers is measured using Reverberation suite as per ISO 140-3 and ASTM E 90 or by sound intensity method described in standards SAE J 1400. The Reverberation suite consists of two adjacent reverberation rooms with an opening in adjacent wall for mounting the samples. Figure 6 shows Reverberation suite with glass mounted on adjacent wall for transmission loss measurement.

Damping loss factor of viscoelastic materials is measured using Oberst bar method. This test rig is based upon ASTM E-756 and SAE-1637 test standard using 3dB half power technique.

Figure 5. Four microphone Tube

Figure 6. Reverberation Suite for Transmission loss Measurement

SIMULATION TECHNIQUES

Simulation of sound package materials requires five intrinsic parameters to predict sound absorption and sound transmission loss. These intrinsic parameters are porosity, flow resistivity, tortuosity, and characteristic lengths [4, 5, 6]. Out of these five physical parameters, flow resistivity was measured using flow resistivity rig based on ASTM C-522 as shown in figure 7. While other four parameters are inverted using well known optimization techniques based on Genetic Algorithm. The physical parameters used for simulation are depicted in table 1. These physical intrinsic parameters play a vital role in sound absorption and transmission loss phenomenon. In this study, the model used for simulation is rigid frame model by Johnson- Champoux-Allard [7]. Out of these five parameters, flow resistivity is most important parameter which governs sound absorption and transmission loss of sound package materials.

Figure 7. Flow Resistivity Rig

Table 1. Measured Intrinsic Physical Parameters

	Physical Parameters			
	Units	**PU Foam-25 mm**	**Melamine Foam-20 mm**	**Cotton Shody-25 mm**
Density	[Kg/m^3]	40	8.8	90
Porosity	[-]	0.986	0.99	0.92
Flow Resistivity	[N.s/m^4]	23367	10900	25000
Tortuosity	[-]	1.7	1.01	1.6
VCL	[μm]	43	75	72
TCL	[μm]	258	160	158

Sound Absorption- The sound absorption coefficient of 25 mm thick polyurethane foam is measured using two microphone impedance tube. The absorption coefficient is also simulated using mathematical model and results are compared. The figure 8 shows that experimental result matches with mathematically simulated sound absorption coefficient. A parametric study to check effect of flow resistivity on sound absorption is also carried out with variation in flow resistivity values from 1000- 150000 Ns/m^4. The result of this simulation is depicted in figure 9. Form this result, it is clear that sound absorption coefficient increases with increase in flow resistivity of sound package material. The specific flow resistance of any layer of porous material is defined as the ratio of the air pressure differential measured between the two sides of the layer to the steady state air velocity. The flow resistivity is then the specific flow resistance per unit material thickness with unit Ns/m^4. The flow resistivities of useful noise control materials vary widely, but typically fall in the range 10^3 - 10^7 Ns/m^4.

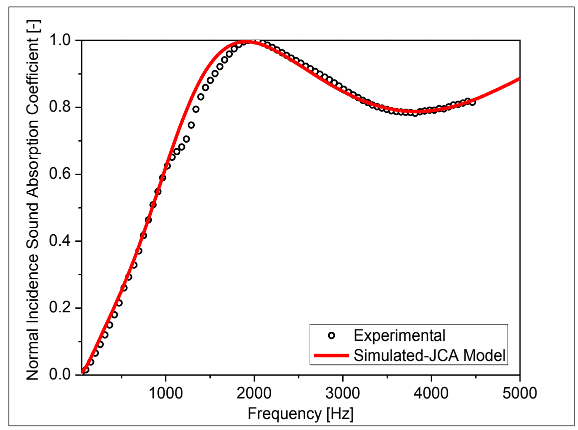

Figure 8. Comparison of Sound Absorption coefficients with predicted sound absorption for Polyurethane Foam at Normal Incidence

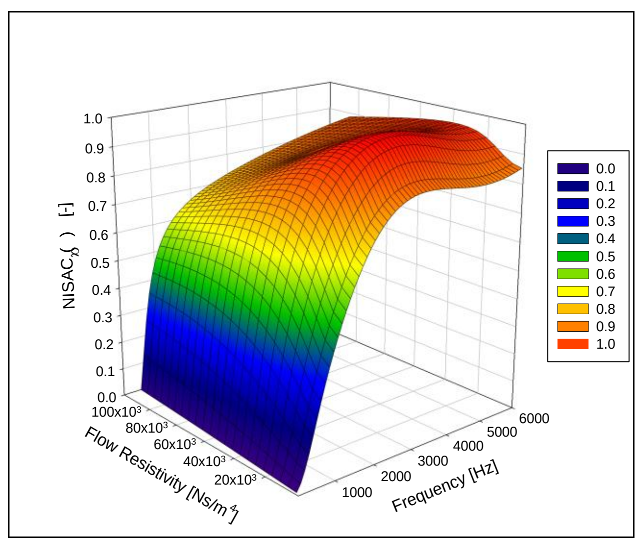

Figure 9. Effect of flow resistivity on sound absorption

The flow resistivity depends on the porosity of a material as well as its tortuosity, but for high porosity, low tortuosity fibrous materials, the flow resistivity is approximately inversely proportional to fiber radius squared at a constant bulk density i.e., a large number of small fiber diameters results in a higher flow resistance than does a small number of fiber with large diameter. At microscopic level, the flow resistance results from the formation of a viscous boundary layer as fluid flows over each fiber, and the amount of shearing in that boundary layer increases as the fiber radius decreases. Similarly effect of porosity and tortuosity can be simulated.

Sound transmission loss- Porous materials of different typologies are tested for normal incidence sound transmission loss in four microphone tube. Physical parameters are measured using specialized test rigs as discussed above. The physical parameters for validation are depicted in table 1. These measured parameters are fed to mathematical model to simulate sound transmission loss. The simulated results are compared with measured normal incidence sound transmission loss measured using four microphone tube. The comparison is shown in figure 10 for PU foam. From these figures it is clear that there is good correlation between measured and simulated sound transmission loss using intrinsic physical parameters of PU Foam. The random incidence sound transmission loss of steel plate with and without some of the automobile sound package material like dash panel is measured in a Reverberation suite. The steel plate used for measurement of sound transmission loss is 0.8 mm thickness which simulates the effect of bare firewall or body panel of the vehicle. The result for sound transmission loss of steel plate is shown in figure 11.

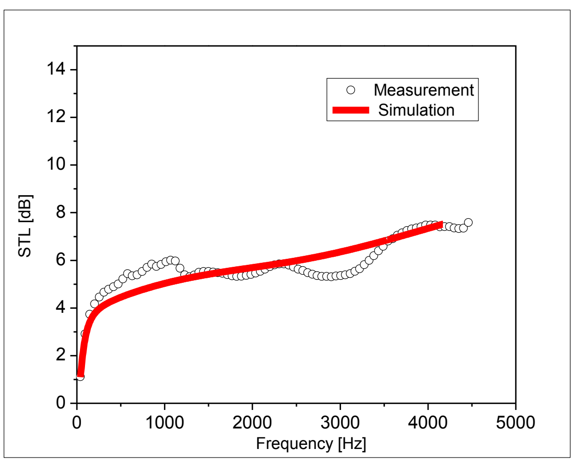

Figure 10. Comparison of Sound Transmission loss with predicted sound absorption for PU Foam at Normal Incidence

The physical and mechanical parameters used for steel plate simulation are given in table 2. The steel plate was modeled in LMS Sysnoise. The comparison of simulated result with experimental results is depicted in figure 11. Similarly sound transmission loss of PU Foam with steel plate is simulated using physical parameters of PU foam and steel plate. The result is compared with experimental values. The correlation is given in figure 12.

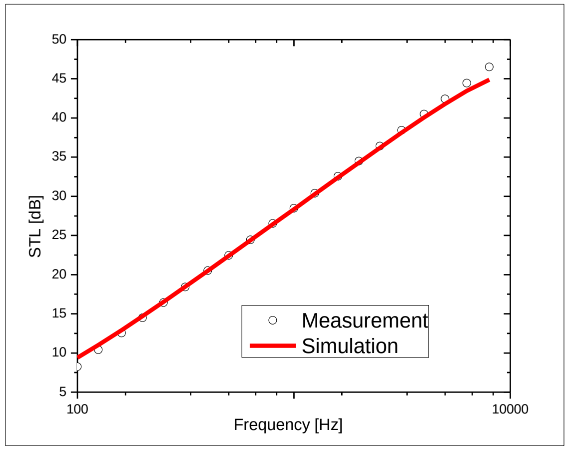

Figure 11. Comparison of Sound Transmission loss with simulation for 0.8 mm Steel Plate at Random Incidence

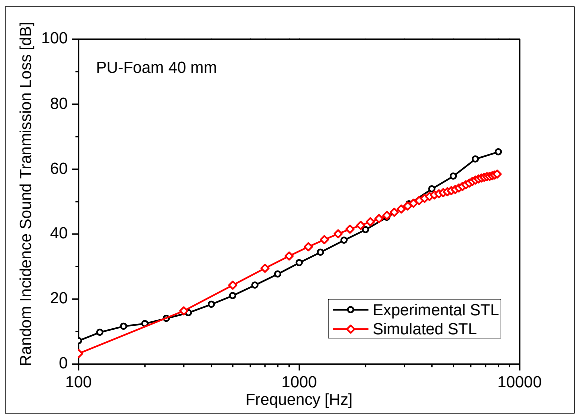

Figure 12. Comparison experimental and simulated Sound Transmission loss of PU Foam with Steel Plate at Random Incidence

Table 2. Physical and Mechanical Parameters

Steel Plate			
Density	*Young's Modulus*	*Poisson ratio*	*Loss factor*
[Kg/m^3]	[N/m^2]	[-]	[-]
7800	207x10^{11}	0.25	0.003

DESIGN OF AUTOMOTIVE DASH

The acoustical performance of dash insulator and carpet contribute significantly in the vehicle interior noise. A study has been carried out to get noise reduction by use of a typical "sandwich insulator" (for dash insulator and carpet), which consists of a steel panel, porous de-coupler and heavy layer. 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 automotive dash used in this study consists of three layers, compressed felt of 10 mm thickness followed by 2 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 3.

Table 3. Measured Intrinsic Physical Parameters

	Units	Compressed Felt	Nonwoven Felt	PE Film
Thickness	[mm]	10	20	28 μm
Porosity	[-]	0.95	0.97	
Flow Resistivity	[N.s/m^4]	36000	15000	Density 1400 Kg/m^3
Tortuosity	[-]	1.7	1.1	
VCL	[μm]	16	120	
TCL	[μm]	180	120	

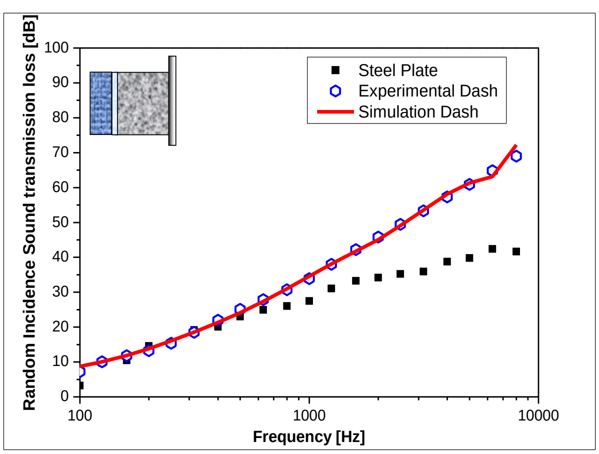

Figure 13. Comparison of experimental and simulated sound transmission loss for a typical dash Insulator

The automotive dash was tested with 0.8 mm steel plate inside the Reverberation chamber and with anechoic termination as per SAE J1400. The experimental results correlated with simulation for dash which shows very good correlation as shown in figure 13.

CONCLUSION

This paper presents a detail discussion on sound absorption and transmission loss mechanism of sound package materials. It gives a brief overview of experimental techniques for measurement of sound absorption, transmission loss and vibration damping. It also compares the experimental results with simulation using mathematical modeling. 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 sound absorption and sound transmission of sound package materials. This provides quick solution for design and optimization of acoustic systems using the physical parameters instead of going through the difficult process of manufacturing and testing every time.

ACKNOWLEDGMENTS

The authors are thankful to Director, ARAI for his permission and support to complete this work. The authors are also thankful to NVH Dept. for their help during the work.

REFERENCES:

1. M.A. Biot, 1956, 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.
2. J. F. Allard, Y. Champoux, 1992, New empirical equations for sound propagation in rigid frame fibrous materials, J. of Acoust. Soc. of Am., 91(6), 3346-3353.
3. Scot R.A., "The absorption of sound in a homogeneous porous medium", Proceedings of Physical Society, 58**,** 165-183, 1946.
4. Y. Champoux, M.R. Stinson, G.A. Daigle, 1991, Air-based system for the measurement of porosity, J. of Acoust. Soc. of Am., 89, pp. 910-916.
5. R.L.Bolt & R.Brown, "The measurement of flow resistance of porous acoustic materials", J. of Acoust. Soc. of Am., Vol.-13, 4, 337-342, 1942.

6. Michael R. Stinson and Gilles A. Daigle, “Electronic system for the measurement of flow resistance”, J. of Acoust. Soc. of Am., 83, 2422-2428, 1988.
7. Y. Champoux, Michael R. Stinson and Gilles A. Daigle, “Acoustical models for sound propagation in rigid frame porous materials and influence of shape factors”, J. of Acoust. Soc. of Am., 92, 1120-1131, 1992.