Practical Difficulties in Simulation of Vehicle Sound Package Materials

Paresh Shravage
Alfa Acoustics, India

ABSTRACT

In today's growing market, customers demand for better vehicle mileage and noise comfort inside the passenger compartment. This overgrowing demand could be achieved by optimization of sound package treatments thereby reducing the weight of the acoustic sound package treatments applied inside a vehicle. Inside a passenger vehicle, the major sound package treatments applied are seats, carpet, headliner, dash insulator and package trays. The configuration of this sound package treatments changes depending upon type of a vehicle either it is diesel, gasoline or an electric vehicle. This paper discusses these different types of major sound package treatments used inside a passenger vehicles along with different configurations required for diesel and gasoline vehicles. The acoustic comfort of these sound package treatments is measured in terms of sound absorption or sound transmission loss. Nowadays, OEMs use CAE techniques to simulate these sound package treatments using their intrinsic material properties, but there are some practical difficulties regarding modelling of the multilayer sound package treatments used inside vehicles. This paper discusses difficulties faced by CAE engineers during simulation. This paper also discusses different methodologies for characterization of various layers of sound package treatment along with the characterization results.

INTRODUCTION

Sound package design, simulation and optimization is becoming important in a passenger vehicles as they provide quieter and greater comfort during ride conditions. Headliners, Dash Insulators, Seats, Carpet are all designed with this purpose in mind. The design of automobile sound package materials is especially important due to their notable size and role in noise reduction inside passenger compartment. Dash insulator 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. Similarly Underbody carpet plays a significant role in tire/road noise reduction. For the years, design and optimization of sound package treatments was a challenge for NVH engineers as they are multilayered structures. The acoustic performance of sound package treatments is measured in terms of sound absorption and sound transmission loss or sometimes in sound insertion loss. The sound absorption coefficient can be measured using two microphone impedance tube method based on ASTM E1050 or in a large reverberation chamber as per ASTM C423 / ISO 354 [1, 2, 3]. Recently SAE International has come up with a new standard SAE J2883 for sound absorption measurements in a small reverberation chamber [4]. While the sound transmission loss of a sound package materials is measured as per SAE J1400 in Reverberation suite with coupled anechoic termination facility [5]. Sometimes it was also measured as per ASTM E90 / ISO 10140 in a two reverberation chamber facility or as per ASTM E2249 in coupled reverberation-anechoic chamber facility for detecting weak paths inside the treatments [6, 7, 8]. The performance of these sound package treatments can also be predicted using simulation techniques which require intrinsic physical parameters of sound package materials. There are very few software's commercially available based on FEA and SEA techniques which can predict an acoustic performance of sound package treatments starting from material (flat sample testing) to component level (vehicle buck testing). This paper explains the methodology of sound package characterization and simulation which will result into bridging gap between materials to component level simulation. In this study, a software package based on transfer matrix method and developed in-house is used to predict multilayered performance of sound package treatments. In first section, theory behind modeling of sound package treatments 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, different sound package treatments are modeled and results are validated with measured sound absorption and sound transmission loss measured in Test facility.

SOUND PACKAGE TREATMENTS

Today's modern vehicles are recognized by their NVH comfort, which is governed by sound package treatments applied inside a passenger vehicle. A typical vehicle consists of a dash insulator which separates engine and passenger compartment and at the same time it serves as a passage for steering wheel, brake and clutch components etc. through grommets. Depending upon vehicle type, the dash includes a heavy layer which helps in improving sound transmission properties of dash insulator in diesel vehicles. A vehicle carpet is also plays a similar role, which reduces tire / road noise entering inside the passenger compartment. Figure 1 shows typical internal structures of various sound package treatments.

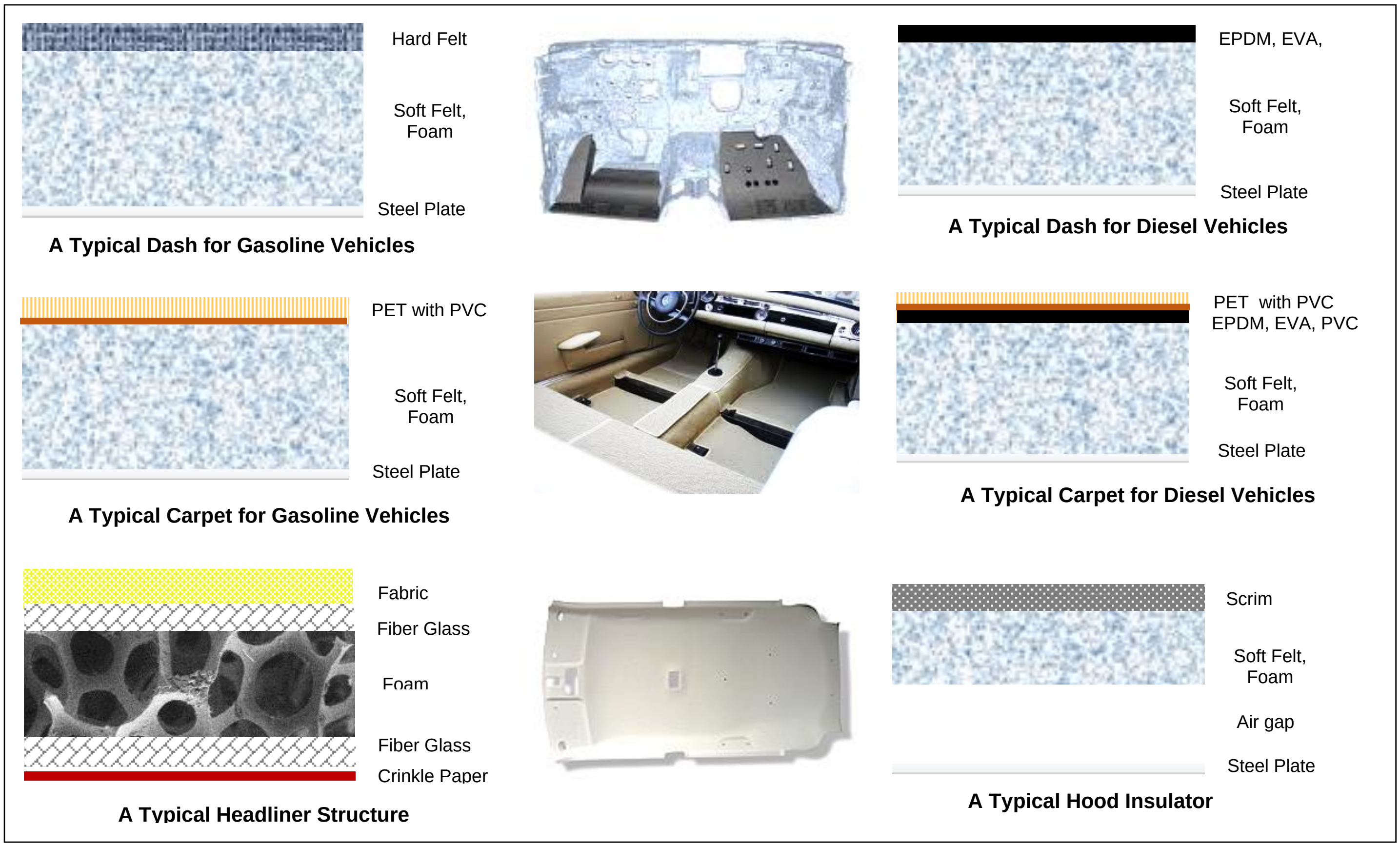


Figure 1. Typical Sound Package Treatments with Internal Structure

The figure below shows volume percentage of sound package materials used inside a passenger vehicle for noise control.

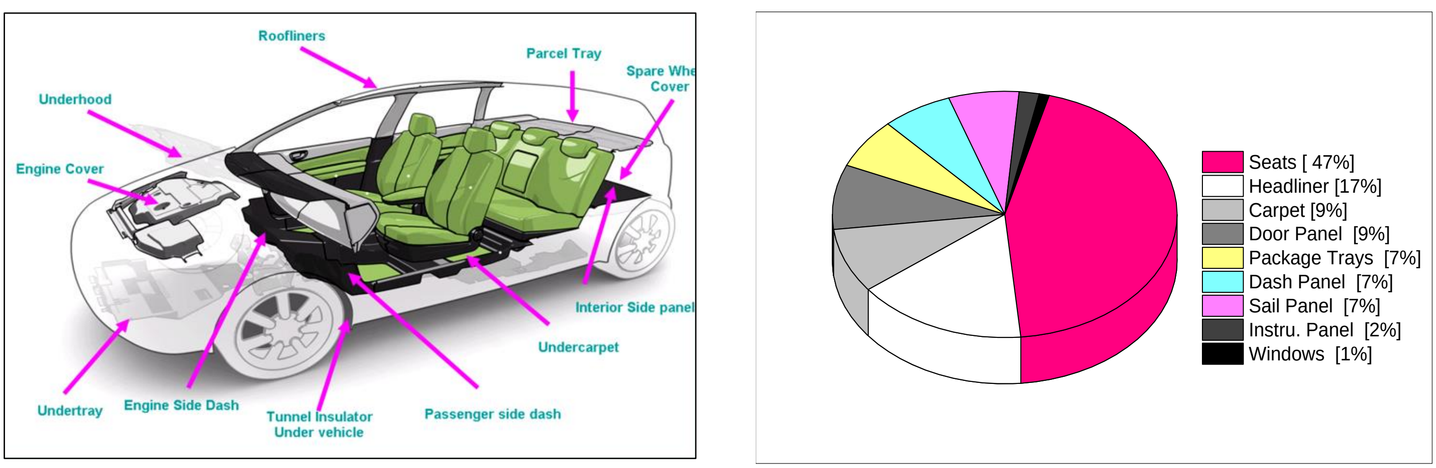


Figure 2. Typical Sound Package Treatments with Percentage of Sound Package Treatments in a Vehicle

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 [9]. 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

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

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

where $\mathbf{u}$ is the vector solid displacement field and $\mathbf{U}$ is the vector fluid displacement field. $\theta^s=\nabla.\mathbf{u}$ and $\theta^f=\nabla.\mathbf{U}$ are volumetric deformations in the phases. $N=E/\left[2(1+\nu)\right]$ 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; $Q=K_c(\omega)(1-\varphi)$ 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

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

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 by (1).

$$k_c=\omega\sqrt{\frac{\rho_c}{K_c}} \quad (4)$$

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 follows.

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

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

The characteristic impedance Z_c and propagation constant k_c 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 [10]. The general representation for a Transfer matrix of a single layer acoustic system (Fig.1) is

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

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

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

where z_c and k_c are characteristic impedance and complex wave number respectively. The total impedance Z_s is given by

$$Z_s = j z_c \coth\left(k_c.d\right) \tag{11}$$

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

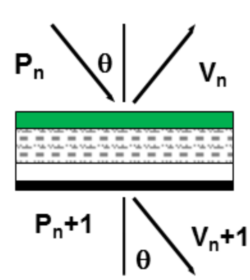


Figure 3: Schematic of transfer matrix

Similarly for films and foils, the transfer matrix is represented as

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

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.

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

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 P_{n+1} and V_{n+1} for the right end plate, as follows

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

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

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

$$\frac{Ce^{-jkd} - De^{jkd}}{\rho_0 c_0} = V_{n+1} = T_{21}P_{n+1} + T_{22}V_{n+1} \tag{17}$$

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

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

The normal incidence sound absorption α for an absorbing material with rigid backing is given by

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

Similarly, the transmission coefficient $T = C/A$ can be calculated and is expressed as [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 as

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

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 as follows [11].

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

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

Where θ is the limiting angle varying between $\theta_{\min}$ and $\theta_{\max}$ with the limiting angle is between 70^0 to 85^0.

Non-acoustic Parameters-

In this section, experimental techniques used to evaluate intrinsic physical parameters and non-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 [12]. Flow resistivity can be measured using flow resistivity test rig based on standard ASTM C522 [13]. Tortuosity and characteristic lengths are measured using ultrasonic techniques or inverted using optimization technique based on Genetic Algorithm [14]. This technique requires prior measurement of sound absorption coefficient with surface impedance in two microphone impedance tube. Then this experimental data with porosity and flow resistivity was used to fit a mathematical model. The global solution of this optimization problem will give tortuosity and characteristic lengths. The effect of intrinsic physical parameters is discussed below in detail.

Porosity

It is the ratio of the fluid volume within the porous material to the total volume of material, on a unit-volume basis. Since the porosity quantifies the relative volume occupied is a key parameter in theories of sound propagation in porous materials [15]. However, the porosity of typical acoustic materials such as foams and glass fibers is normally very high, i.e., greater than 0.90, and often greater than 0.98. Since the porosity is so large in most noise control materials, and because the variations in porosity tend to fall into a very narrow range, variations in porosity are not to be very important when distinguishing between noise control materials. However, it should be remembered that much of the relative motion of the solid frame and the interstitial fluid, and that for this process to work, there must be continuous paths through the material.

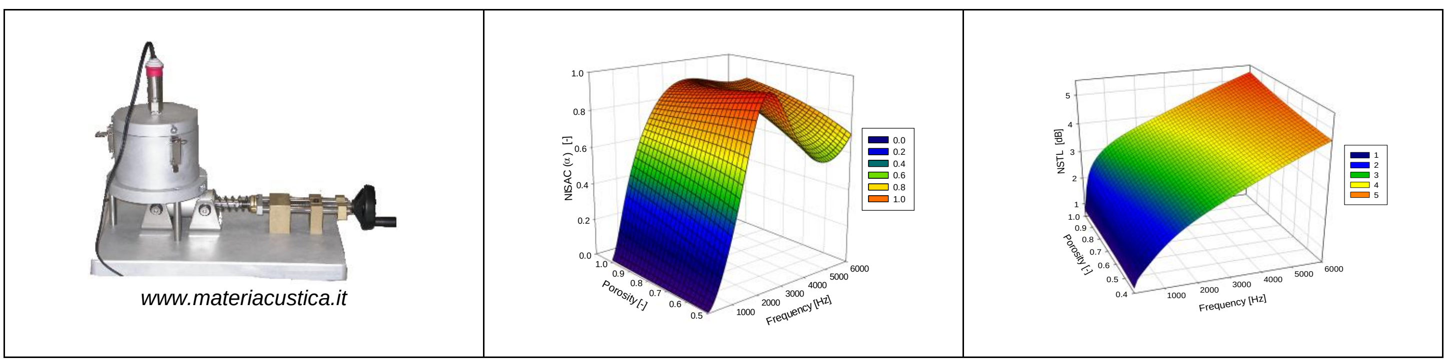


Figure 4: Test rig for porosity measurement (ENDIF, Italy) and Effect of porosity on Sound Absorption and Sound Transmissions Loss

The most direct way of determining the porosity of a porous material is to measure the volume of air contained within the material. This method may be achieved using the apparatus developed by Champoux [16] shown in the figure 4. When the temperature of a rigid chamber containing a test sample is held very constant, a measurement of the change in air pressure that accompanies a known change in volume allows the volume of fluid within the sample to be determined if the change in air pressure accompanying the same volume change in a rigid, empty chamber of the same total volume is known. The figure below shows a porosity rig and effect of porosity on sound absorption coefficient.

Flow Resistivity

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 through and perpendicular to the two faces of the layer [17]. The flow resistivity is then the specific flow resistance per unit material thickness with SI units of MKS rayls/m. The flow resistivity values of useful noise control materials vary widely, but typically fall in the range 10^3 rayls/m to 10^7 rayls/m. 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 larger fibers. 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. The flow resistivity is thus usually taken to be a measure of the viscous coupling between the fluid and solid phases of the porous material, and so is a measure of the potential for viscous dissipation of sound.

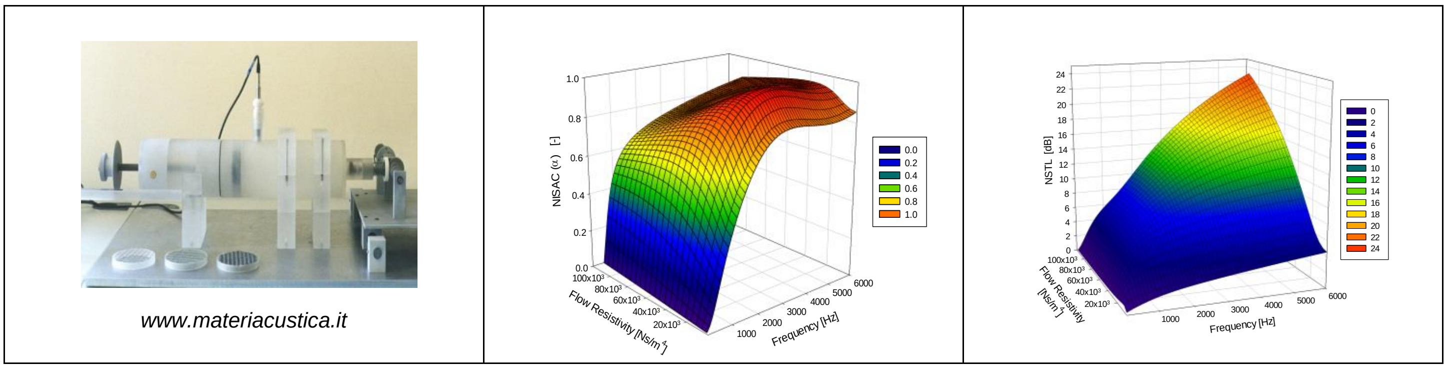


Figure 5: Test rig for Flow resistivity measurement (ENDIF, Italy) and Effect of flow resistivity on Sound Absorption and Sound Transmissions Loss

Tortuosity

It is sometimes referred as structure factor and defined as defined as the ratio of actual path length through the material to the linear path length. It is a measure of deviation between the actual fluid flow path through the material and straight- line flow through the material. It results from inertial coupling between solid and fluid phases. The range of tortuosity is from 1 (low density fibrous material) to values of 10 (partially reticulated foams with any closed cells). Brown had developed an electrical conductivity technique to measure pore tortuosity. The voltage difference arising from passing a high voltage through a fluid-saturated (electrically conductive) sample is measured. This method is obtained from a temporal conductivity of fluid and fluid filled samples, a simple relation may be established to calculate the tortuosity when porosity is known. This method cannot be used when material frame is conducting [18].

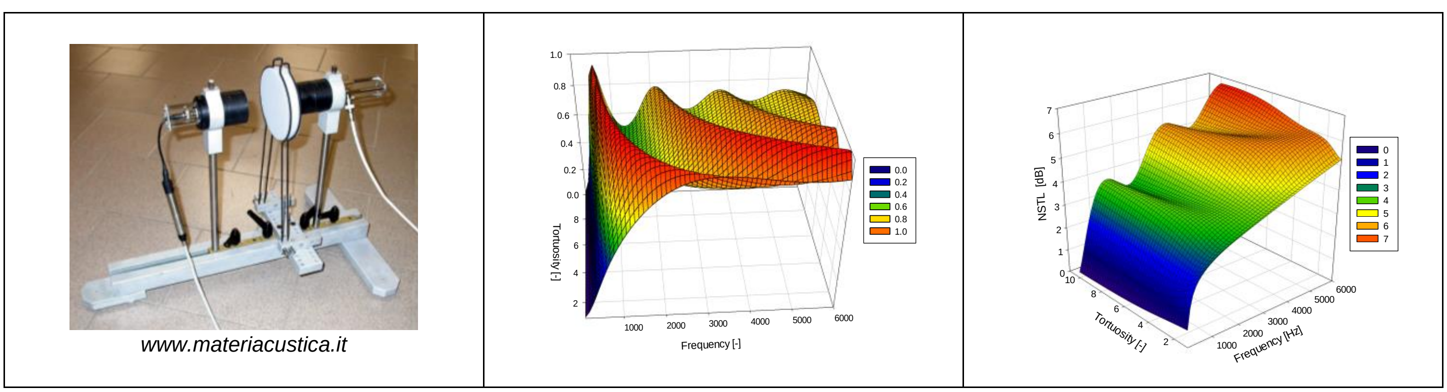
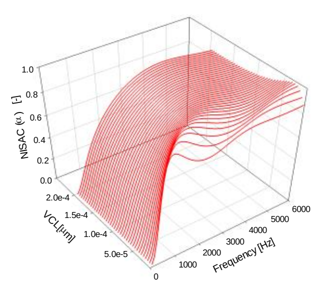


Figure 6: Test rig for Tortuosity measurement (ENDIF, Italy) and Effect of tortuosity on Sound Absorption and Sound Transmissions Loss

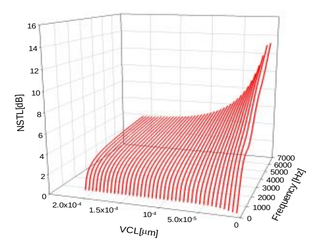
Recently Ultrasonic reflectivity method is proposed for measuring tortuosity of porous materials having a rigid frame. This method is based on measurement of reflected wave by the first interface of a slab of rigid porous material. Tortuosity is a geometrical parameter which intervenes in the description of the inertial effects between the fluids filled the porous material and its structure at high frequency range. It is generally easy to evaluate the tortuosity from model of the direct and inverse scattering problems for the propagation of transient ultrasonic waves in a homogeneous isotropic slab of porous material having a rigid frame [19].

Characteristic Lengths

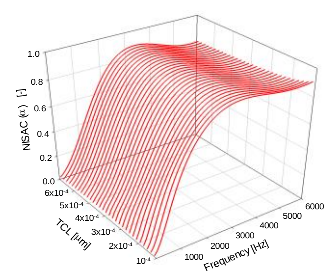
The concept of viscous characteristic length is used to describe the acoustical behavior of fluid-saturated porous media in the high-frequency regime. A method to determine this parameter consists of measuring the wave attenuation in the high-frequency limit. This method has already been used for porous materials saturated by super fluid He. It is tested in the case of air-filled absorbent materials in a frequency range of 50–600 kHz. The thermal characteristic length is assumed to be known or measured independently [20]. Recently inverse characterization is becoming popular for predicting physical parameters using optimization techniques. In this paper characteristic lengths are predicted using genetic algorithm optimization. The effect of characteristics lengths on sound absorption is shown in figure 7.



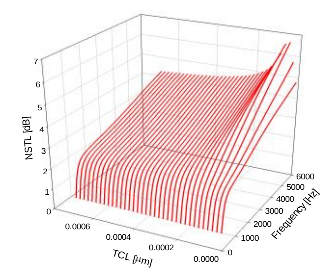
Effect of VCL on Absorption



Effect of VCL on Transmission Loss



Effect of TCL on Absorption



Effect of TCL on Transmission Loss

Figure 7: Effect of Characteristic lengths on Sound Absorption and Sound Transmissions Loss

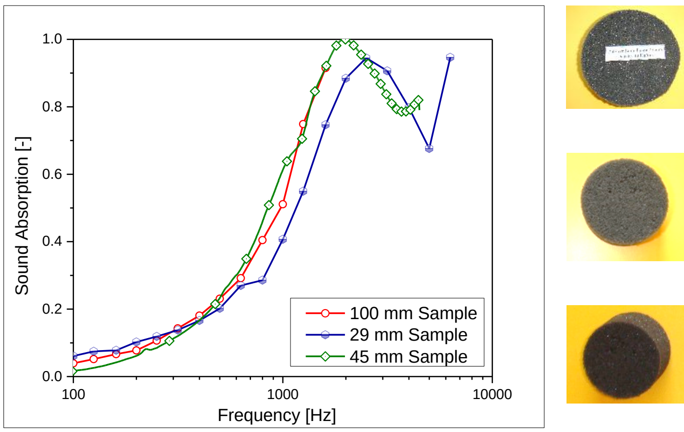
The acoustic absorption of the sound package materials is measured using a two microphone impedance tube in accordance with ASTM E1050. This method is rapid and requires only a small size sample of the material. The test uses an impedance tube with a sound source connected to one end and the test sample 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.

Two Microphone Impedance Tube with 100 mm and 29 mm Diameters

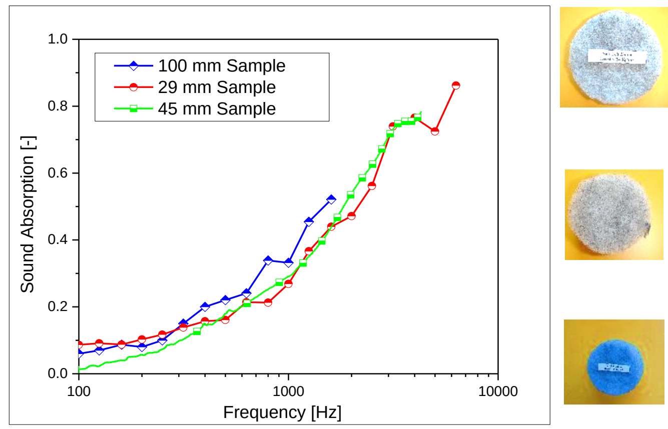
Two Microphone Impedance Tube with 45 mm Diameter

Figure 8: Impedance Tubes

The selection of correct impedance tube is very important during characterization of sound package materials for accurate measurement of sound absorption and surface properties. Generally commercial impedance tubes are designed and fabricated as per standards, so they require two samples of different diameters and this is one of the major concern in estimation of intrinsic physical parameters. All sound package materials are considered to be homogenous at macroscopic scale, but they are not homogenous at microscopic scale. Hence when measurements are conducted on two different diameter samples cut from same sheet, then one may observe results as shown in figure 9 for PU Foam and PET Felt samples of 100 mm and 29 mm diameters. This shift is more prominent for actual sound package materials and if these results are used for evaluation of inverse parameters then obviously this will lead to wrong set of physical parameters.



(9a)

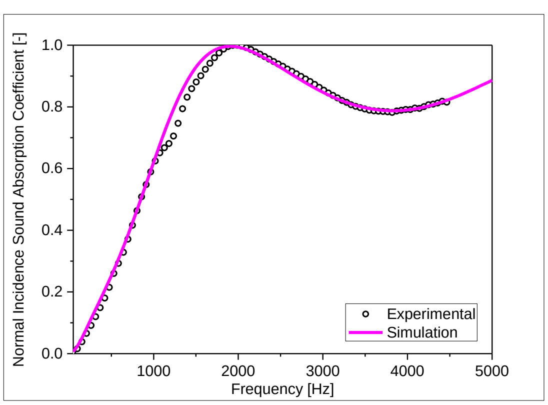


(9b)

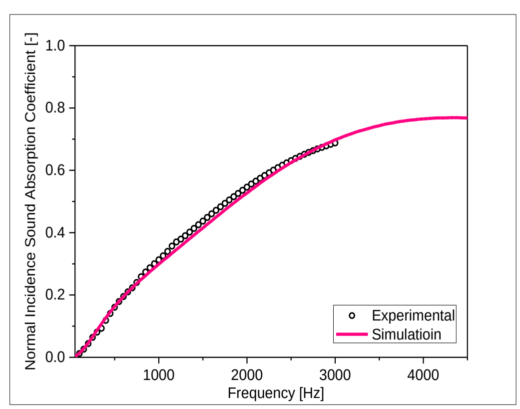
Figure 9: Comparison of standard and Customized Impedance Tube (9a): PU Foam, (9b): PET Felt

	Intrinsic Physical Parameters		
	PET Felt 25 mm 24 Kg/m^3	**PU Foam 25 mm 40 Kg/m^3**	**Units**
Porosity [ϕ]	0.98	0.986	-
Airflow Resistivity [ρ]	6634	23367	Ns/m^4
Tortuosity [α_∞]	1.06	1.7	-
Viscous Length [Λ]	147	43	µm
Thermal Length [Λ']	203	258	µm

Table 1: Intrinsic Parameters of Sound Package Materials



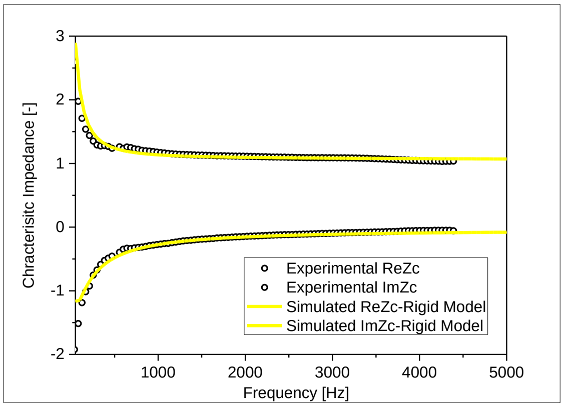
(10a)



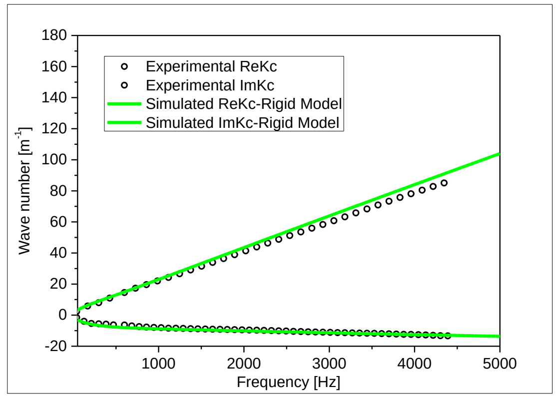
(10b)

Figure 10: Comparison of Sound Absorption coefficients with predicted Sound Absorption (10a): PU Foam, (10b): PET Felt

To overcome this problem, a customized impedance tube of 45 mm diameter is used in this study. The frequency range of this tube is 100 Hz – 4400 Hz and specialty is, it requires only one sample of 44.5 mm diameter. Figure 9 shows test results for PU Foam and PET Felt samples. Table 1 gives inverse physical parameters estimated using test data from impedance tube of 45 mm diameter. These physical parameters are then used to predict sound absorption, surface properties and sound transmission loss of samples. The comparison of sound absorption for PU Foam and PET Felt is shown in figure 10, while figure 11 shows comparison for characteristic impedance and complex wave number for PET Felt.



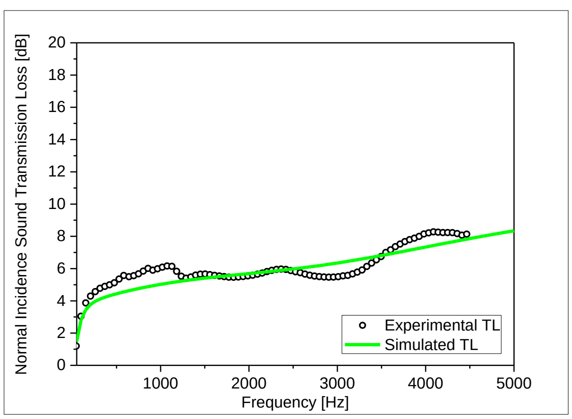
(11a)



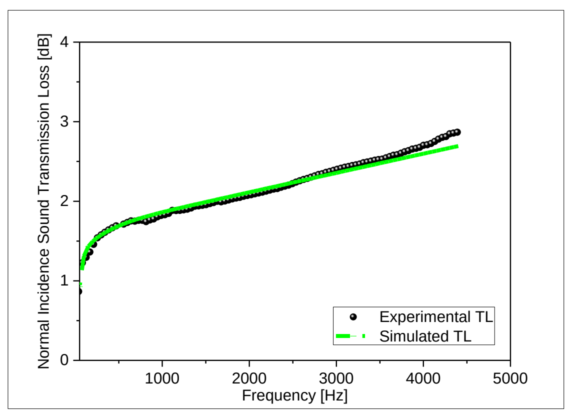
(11b)

Figure 11: Comparison of Measurements with Simulation for PET Felt
(11a): Characteristic Impedance, (11b): Complex Wave Number

The sound transmission loss comparison is shown in figure 12 for PU Foam and PET Felt samples. From this data, it is clear that the physical parameters estimated using customized tube are more accurate than those estimated using standard impedance tubes.



(12a)



(12b)

Figure 12: Comparison of Measurements with Simulation for Sound Transmission Loss
(12a): PU Foam, (12b): PET Felt

CONCLUSION

From this study, it is observed that there is significant difference in sound absorption coefficient and surface properties measured in different diameter impedance tubes. So it becomes a necessity for a NVH engineer to take care about measurements when these test results are used as input to software's for estimating non-acoustic physical parameters. This paper gives guidelines and precautions while conducting absorption measurements along with details about specialized test rigs used to measure intrinsic physical parameters. This paper also presents validation of simulated results using test results.

ACKNOWLEDGMENTS

The authors are also thankful to ENDIF, Italy for providing required test rig facility for the measurement of physical parameters.

REFERENCES

1. ASTM E1050, "Standard Test Method for Impedance and Absorption of Acoustical Materials Using a Tube, Two Microphones and a Digital Frequency Analysis System", 2012
2. ASTM C423, "Standard Test Method for Sound Absorption and Sound Absorption Coefficients by the Reverberation Room Method", 2007.
3. ISO 354, "Acoustics-Measurement of sound absorption in a reverberation room. Switzerland: International Organization for Standardization", 1988.
4. SAE J2883, " Laboratory Measurement of Random Incidence Sound Absorption Tests Using a Small Reverberation Room", SAE Standard Committee
5. SAE J 1400, "Laboratory Measurement of the Airborne Sound Barrier Performance of Automotive Materials and Assemblies, Recommended Practice", 2010.
6. ISO 10140-2, "Laboratory measurement of airborne sound insulation of building elements", 2010.
7. ASTM E90, "Standard Test Method for Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions and Elements", 2004
8. ASTM E2249, "Standard Test Method for Laboratory Measurement of Airborne Transmission Loss of Building Partitions and Elements Using Sound Intensity", 2008
9. 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.
10. Song B.H. and Bolton S.J., "A transfer matrix approach for estimating the characteristic impedance and wave numbers of limp and rigid porous materials", Journal of Acoustical Society of America, 107, 1131-1151, 2000.
11. E.T. Paris, "Oblique reflection of sound," Nature 9, 126, 1930
12. 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.
13. ASTM-C522, "Acoustics-Materials for acoustical applications-Determination of airflow resistance", 1991
14. Paresh Shravage, Paolo Bonfiglio, Francesco Pompoli, "Hybrid Inversion technique for predicting geometrical parameters of Porous Materials", Proc. of Acou. 08, June 29- July 04, Paris France, 2008
15. 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.
16. 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.
17. 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.
18. Brown R., "Connection between the formation factor for electrical resistivity and fluid-solid coupling factors in Biot's equations for acoustic waves in fluid-filled porous media", Geophysics, 45, 1269-1275, 1980
19. Allard J. F., Castagnède B., Henry M. and Lauriks W., "Evaluation of the tortuosity in acoustic porous materials saturated by air", Review of Scientific Instruments **65** pp. 7654-755, 1994.
20. Leclaire P., Kelders L., Lauriks W., Melon M., Brown N. and Castagnède B., "Determination of the viscous and thermal characteristic lengths of plastic foams by ultrasonic measurements in helium and air", Journal of Applied Physics, 80, 2009-2012, 1996

CONTACT

Dr. Paresh Shravage is Founder and Technical Head of Alfa Acoustics-a private R&D and consulting company. He has done his doctorate in Acoustic Materials from University of Pune, India. During his Doctorate, he has also worked as a Marie-Curie Fellow at School of Acoustics, University of Ferrara, Italy under European Doctorate in Sound and Vibration (EDSVS) programme. He is having almost 10 years of research and industrial experience in Acoustic materials, Noise and Vibration. He is also consulting as Principal Scientist, NVH for L&L Products Pvt. Ltd.

He has published several research papers on acoustic materials in national / international conferences and journals. He is a recipient of Sir C.V. Raman award from the Acoustical society of India for his work on Reverberation chamber. During his service at ARAI, he was recognized by Innovation Awards several times.

He is a member of Acoustics Material Standard Committee, SAE International, USA and several other Institutes. His main area of specialization is characterization, simulation and Testing of acoustic materials for vehicle as well as building acoustics applications.

Email- alfaacoustics@gmail.com ; info@alfaacoustics.com
www.alfaacoustics.com