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Innovative Acoustic Material Concept Integration into Vehicle Design Process

Denis Blanchet dBVibroAcoustics

Michael Feist-Muench Odenwald-Chemie GmbH

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Abstract

Integration of acoustic material concepts into vehicle design process is an important part of full vehicle design. The ability to assess the acoustic performance of a particular sound package component early in the design process allows designers to test various design concepts before selecting a final solution and long before a design freeze. This paper describes an innovative acoustic material concept which is easily integrated in a design process through the use of vibro-acoustic simulation and a database of intrinsic properties of acoustic materials: *The Biot Parameters*. Biot parameters are widely used in simulation in many industries (and used the most in the automotive industry) to describe the physical interactions between the acoustic waves travelling through foams, fibers or homogeneous metamaterials and the solid and fluid phase of these poro-elastic materials. Therefore, the surface absorption, the insertion loss and the added damping provided by the acoustic treatments on the base plate can all be predicted accurately. Simulation can be performed at component and full vehicle level using Biot parameters since these are the intrinsic properties of the porous material, the same way Young's Modulus is an intrinsic property of steel. Furthermore, Biot parameters can be directly used in FEM (Finite Element Method), BEM (Boundary Element Method) and SEA (Statistical Energy Analysis) thanks to the existence of porous finite elements or the use of TMM (Transfer Matrix Method). This paper introduce a new acoustic material concept which provides a combination of absorption, transmission loss and added damping on the panel it is attached to. It has shown unique vibro-acoustics performance when tested on a German car manufacturer flagship vehicle and provides the ability to reduce the space needed for sound package component compared with classical solutions. It is manufactured by impregnating a fraction of total thickness of a PU foam. This results in two acoustic layers, one light foam and the other a heavy and high damping layer. A description of the Biot parameter measurements of each layer and test results for each sample tested along with standard deviation are provided. Finally, a simulation analysis using TMM is performed to assess the airborne and structureborne acoustic performance of this new unique material.

Introduction

This paper describes how a complex acoustic trim can be modelled using Biot parameters and TMM. The TMM have been widely used in conjunction with geometry/thickness variation mapping algorithms to model complex physical trim parts of automotive vehicles. Furthermore, by having a numerical model of this component, even only in flat uniform shape provides data to an early sound package design phase. The key is the availability of the Biot parameters. All Biot parameters data and the contents of measurement reports presented in this paper are from a new commercial porous material database ***dBPorous*** [1].

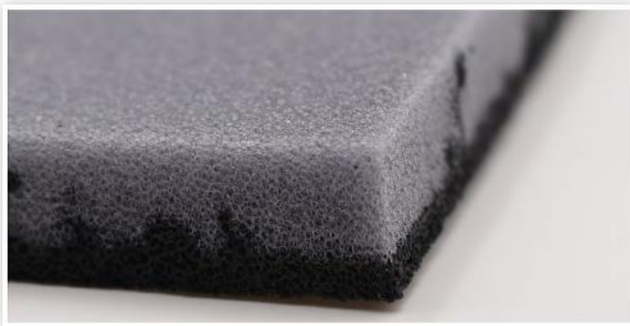
The various sections of this paper will cover a description of the trim material to be modelled, the modelling strategy adopted, followed by an extensive description of the measurements performed to get the Biot parameters of each layer of the trim material. Finally, an analysis of the airborne and structureborne acoustic performance of the trim material is presented.

Description of the New Acoustic Material

O.C.-PREN SC NV is a fine-pored ether-based polyurethane foam. The material composition is characterized by one side laminated with a thin micro-perforated 25 microns PU-foil and by one self-adhesive membrane on the side coated with a special impregnation. This acrylate-based impregnation with mineral fillers is manufactured using a special bath coating process which creates a spring-mass system as one heavy monolayer which entails several properties. A high density of 28 kg/m^3 and thickness of 15 mm are just a few of them. The permanently elastic and soft material is hydrolysis resistant, flame-retardant and emission-free (Figure 1).

Due to its properties, this material is suitable for a number of applications. This includes primarily structural damping, insulation and absorption, thermoacoustic isolation and encapsulation of hydraulic pumps, compressors and servomotors.

FIGURE 1 Innovative material: foam partially impregnated with acrylate-based mineral filler ("foam only" and "impregnated foam" layers shown; self-adhesive membrane and PU-foil not shown).



The O.C.PREN SC NV was designed to be used inside a headliner to absorb the natural frequency of 26 Hz of the headliner, to damp medium frequencies of the big outer skin of the headliner, to reach a significantly better level in case of rain and wind. It is self-adhesive on the outer skin (-40°bis +115°C), sticky overhead as well as flame-retardant, emission-free and odorless in the interior. A stringent requirement on weight of 2–4 kg/m^2 with a max. thickness of 50mm was met with such a solution. Assembled in the headliner of the running G-class this specially impregnated foam ensures sound-dampening, structure-borne noise and thermoacoustic insulation to allow a better speech intelligibility and well-balanced acoustic experience.

Modeling Strategy

Twenty years ago, very few engineers had the chance to use Biot parameters to model acoustic trim inside a vehicle. Existing analytical formulations were implemented in SEA software and work was done to create new FEM elements to represent porous material in a FEM environment. Furthermore, only a handful of acoustic laboratories were able to measure all Biot parameters and had to use expensive sophisticated measurements. Fortunately, today's situation has drastically changed. Biot parameters can be identified accurately and standards are being developed to help get uniform results from the numerous laboratories having the capability to measure Biot parameters.

Today, with a classical impedance tube and airflow resistivity meter measurements, the acoustic porosity and airflow resistivity can be determined experimentally with reasonable accuracy. The remaining acoustic Biot parameters can be calculated using indirect methods [2, 3]. These identified Biot parameters are the intrinsic properties of the poro-elastic material. This means these Biot parameters can be used in simulation to model various thicknesses of the same acoustic materials without the need to repeat any measurements.

To fully characterize foam type materials, extra properties such as Young's modulus, Poisson's ratio and structural damping of the skeleton are needed. These are easily obtained from a quasi-static mechanical test [4]. Table 1 shows the full list of acoustic and elastic parameters needed to fully represent a porous media in a simulation model.

TABLE 1 List of Biot parameters needed to fully represent an acoustic trim in a full vehicle model

σ	Flow resistivity
Φ	Open Porosity
α	Tortuosity
Λ	Viscous Characteristic Length
Λ'	Thermal Characteristic Length
E	Young's Modulus
ν	Poisson's Ratio
η	Damping Loss Factor (DLF)

FIGURE 2 Effect of various Biot parameters on Absorption coefficient

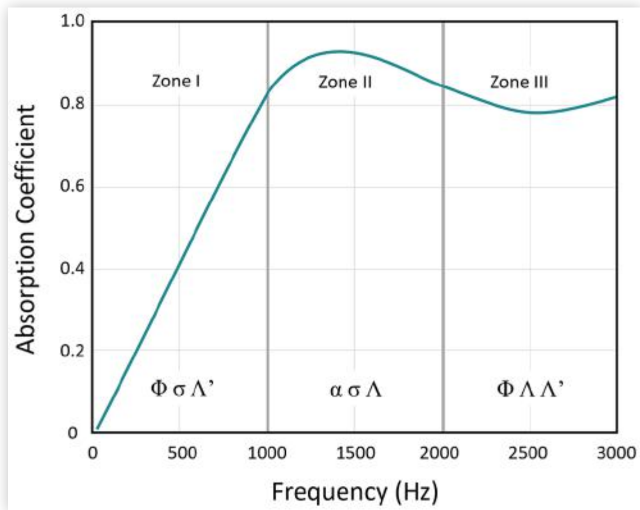


Figure 2 shows which Biot parameters are driving the absorption coefficient response over various frequency ranges [from 5].

The simulation results presented in the last section of this papers will therefore include a Biot parameters representation of the porous media studied. Since the studied trim material has two distinct acoustic layers, one with "*foam only*" and the other with the same foam which is "*impregnated*" (the material is actually a single foam partially impregnated on one side), two sets of Biot parameters are used. Each layers is considered homogeneous in material properties with uniform thickness.

Note that the complex interpenetration at the interface between the two layers is neglected in this study. The PU-foil and the self-adhesive membrane are also neglected for simplicity.

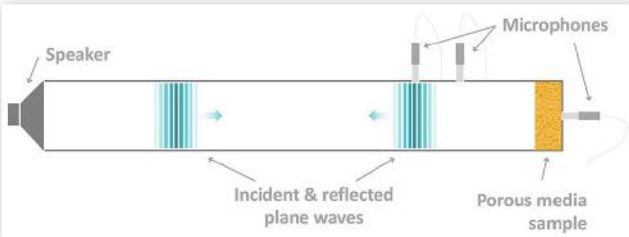
Material Characterization

All measurements results and description of methods used in section "Measurement setup" and "Experimental results" are extracted from ***dBPorous*** database material characterization reports [6].

Measurement Setup

Biot parameters are derived from acoustic measurements performed in a traditional impedance tube and an airflow resistivity meter.

FIGURE 3 Impedance tube schematic representation



Acoustic pressures are measured in front and through the rigid backing behind the porous sample. Samples were cut to fit the 44.44mm impedance tube. The thicknesses of material samples are manually measured using an electronic calipers with a precision of 0.01 mm. for material samples which do not have a perfect flat surface, the thickness precision is 0.1 mm.

Experimental Results

Measurement were conducted on both layers of the "***Layup***"; the "foam only" and the "impregnated foam".

Figure 4 shows on the left the "foam only" sample for the first layer of the ***layup***. On the right, the "impregnated foam" samples after tests. On the left, the visible face is named "*a*". On the right, the visible face is named "*b*". Face "*a*" shows a uniform granular black surface while face "*b*" shows some clear stripes and a more dense surface. The two faces exhibits different properties (i.e. the material is heterogeneous).

The results below apply to face "*a*" which is assumed to be the side facing toward sound incidence. The following table displays the results of the experimental test campaign on both layers of the ***Layup***.

FIGURE 4 Examples of samples used to measure the material parameters. Left: Foam only, Right: Impregnated foam - (Left: face a, right: face b)

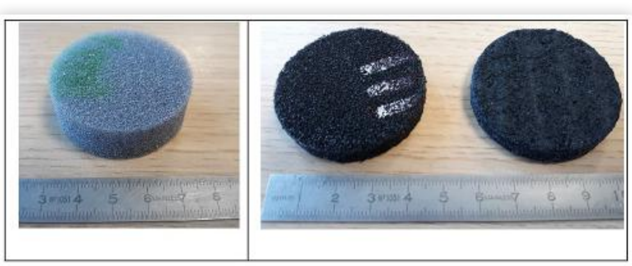


TABLE 2 Measured Biot parameters for both layers: Foam only and foam impregnated.

Material names/Property names	Foam only	Foam Impregnated
Density (kg/m^3)	28	1043
Flow Resistivity (N.s.m^{-4})	7800	174 500
Open Porosity (-)	0.98	0.49
Tortuosity (-)	1.54	1.4
Viscous char. Length (μm)	61	4
Thermal char. Length (μm)	368	238
Young's modulus (x10^3 N.m^{-2})	121.2	11 878
Poisson's ratio (-)	0.49	0.36
Damping Loss Factor (-)	0.12	0.44

The following sections contain the experimental results used to compute the mean value and standard deviation of each Biot parameters. i, ii, iii denotes each sample used in a specific measurement. This should provide the reader with a sense of how these parameters vary from one sample to another.

Acoustic Parameters Results

Flow Resistivity The static air flow resistivity is estimated from the imaginary part of the dynamic mass density measured in an airflow resistivity meter. This measurement follows Standard ISO 9053-1 [7]. σ_x represents the standard deviation of the mean.

TABLE 3 Flow resistivity tests results (Foam only and impregnated foam)

Foam Only			Impregnated Foam		
Sample	σ (δ)	*Thickness*	*Sample*	σ (δ)	*Thickness*
i	7800 (400)	15.1	*i*	168 400 (13 400)	8.0
ii	7900 (400)	15.3	*ii*	203 100 (15 100)	8.1
iii	7700 (400)	15.3	*iii*	152 100 (11 800)	7.9
Mean (σ_x)	**7800** (100)	**15.2** (0.1)	**Mean** (σ_x)	**174 500** (26 000)	**8.0** (0 1)
Units	N.s.m^{-4}	mm	Units	N.s.m^{-4}	mm

Open Porosity The open porosity is estimated from the real part of the dynamic bulk modulus as described in Jaouen et al. [8]. σ_x represents the standard deviation of the mean.

TABLE 4 Open porosity tests results (Foam only and Foam impregnated)

Foam Only		Impregnated Foam	
Sample	*Φ* (δ)	*Sample*	*Φ* (δ)
i	0.98 (0.02)	*i*	0.98 (0.02)
ii	0 98 (0 02)	*ii*	0.98 (0.02)
iii	0.98 (0.02)	*iii*	0.98 (0.02)
Mean (σ_x)	0.98 (0.00)	**Mean** (σ_x)	0.98 (0.00)
Units	(-)	Units	(-)

Tortuosity Estimations of the high frequency limit of the dynamic tortuosity has been realized from measured data of dynamic mass densities and compressibility's.

TABLE 5 Tortuosity tests results (Foam only and Foam impregnated)

Foam Only		Impregnated Foam	
Sample	α_∞ (δ)	***Sample***	α_∞ (δ)
i	1.49 (0.01)	*i*	1.41 (0.39)
ii	1.61 (0.03)	*ii*	1.43 (0.48)
iii	1.52 (0.01)	*iii*	1.36 (0.76)
Mean (σ_x)	1.54 (0.06)	**Mean** (σ_x)	1.40 (0.54)
Units	(-)	Units	(-)

Viscous & Thermal Characteristic Lengths Estimations of the high frequency limit of the characteristic viscous and thermal lengths of the material has been realized from measured data of dynamic mass densities and compressibility's.

TABLE 6 Viscous & thermal characteristic length tests results (Foam only and Foam impregnated)

Foam Only			Impregnated Foam		
Sample	Λ (δ)	Λ'(δ)	***Sample***	Λ (δ)	Λ'(δ)
i	53 (2)	359 (35)	*i*	4.0 (0)	217 (44)
ii	70 (1)	388 (41)	*ii*	4.0 (0)	280 (34)
iii	59 (1)	358 (42)	*iii*	4.0 (0)	219 (16)
Mean (σ_x)	61 (9)	368 (42)	**Mean** (σ_x)	4.0 (0)	238 (48)
Units	μm	μm	Units	μm	μm

The viscous and thermal characteristic length is estimated from their analytical expressions [9, 10] which are deduced from the Johnson-Champoux-Allard (JCA) model [11, 12] or the Johnson-Champoux-Allard-Lafarge (JCAL) model [13]. These models assume a rigid and motionless skeleton.

Validation of the Acoustics Parameters Figure 5, Figure 6 and Figure 7 compares the normal incidence sound absorption coefficient as measured in the impedance tube and as computed using a Johnson-Champoux-Allard-Lafarge model (JCAL) according to the mean values of the parameters characterized above for the samples i) Foam only, ii) Foam impregnated Face "a" and iii) Foam impregnated Face "b".

Measured data are represented as the dispersion envelope obtained over all characterized samples and shown in light gray in the figures.

These results show reasonable accuracy considering the heterogeneous character of some of the samples inherent to the industrial production process of the impregnated layer.

Elastic Parameters

The method used to derive elastic parameters is based on the study of the vibrations of a mass/spring system under a

FIGURE 5 Normal incidence sound absorption coefficient (Foam only)

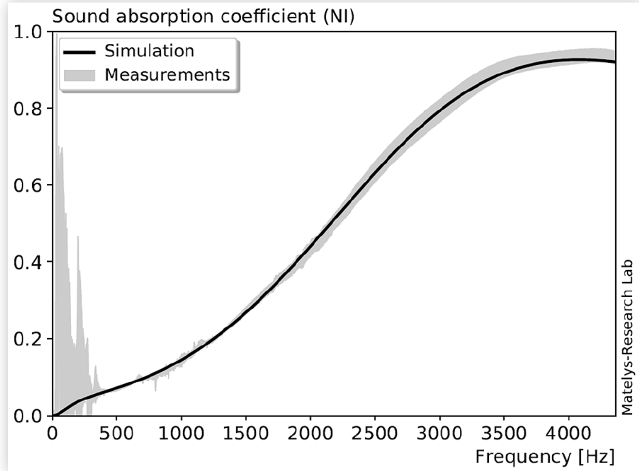


FIGURE 6 Normal incidence sound absorption coefficient (Foam impregnated Face "a")

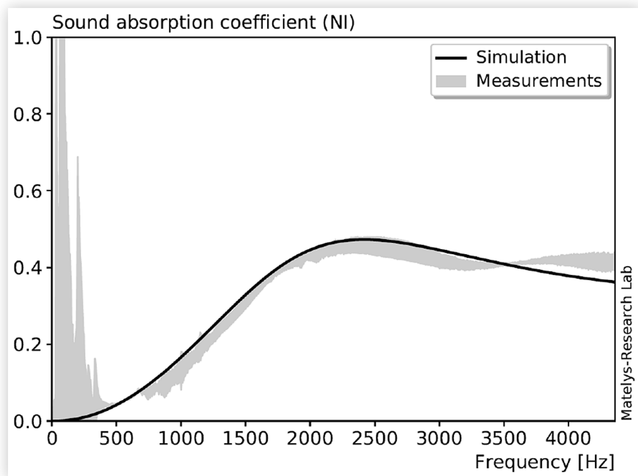


FIGURE 7 Normal incidence sound absorption coefficient (Foam impregnated Face "b")

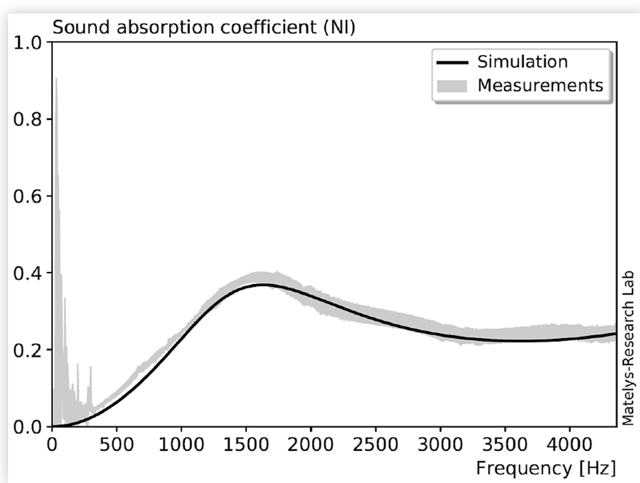
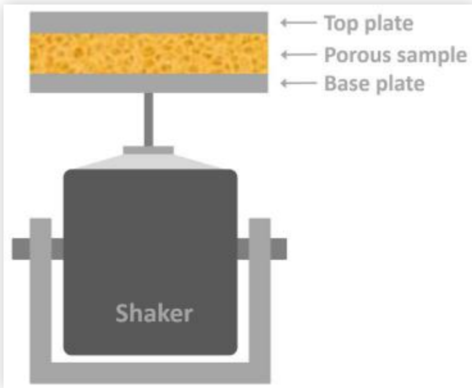


FIGURE 8 Experimental setup to measure elastic parameters



uni-axial compression test. The measured Frequency Response Function (FRF) is defined as the ratio of the displacements of the top rigid mass to the base moving plate for a rectangular parallelepiped or cylindrical (with circular cross section) sample material (see Figure 8). From a practical point of view, an accelerometer is used to determine the base plate displacement and a second one is used to determine the displacement of the top loading mass.

This method can be used for the determination of the Young's modulus, the Poisson coefficient and the structural loss factor of an assumed isotropic material sample when the top loading mass is known. The analysis of the FRF in the vicinity of the resonance of the mass/spring system allows to determine the structural loss factor and the apparent Young's modulus. This latter modulus is linked to the actual Young's modulus of the material by a factor which depends on the shape of the sample and on the Poisson's ratio of the material.

Thus, for a given shape factor, this coefficient only depends on the Poisson's ratio of the material [4]. Therefore, by testing samples having different shape factors, the Poisson's ratio can be estimated. Finally, from this latter value, the actual Young's modulus of the material can be determined.

Elastic Parameter Results Table 7 presents the elastic characterization results for samples of the material under a uniaxial compression test.

TABLE 7 Young's modulus, DLF and Poisson's ratio (Foam only)

Test	E	η	ν	ρ
1	120.0	0.13	0.50	27
2	125.3	0–12	0.48	28
3	118.2	0.11	0.49	28
Mean values (σ_x)	**121.2 (3.7)**	**0.12 (0.01)**	**0.49 (0.01)**	**28 (1)**
Units	**× 10^3 N.m^{-2}**			**kg.m^{-3}**

TABLE 8 Young's modulus, DLF and Poisson's ratio (Foam impregnated)

Test	E	η	ν	ρ
1	11 854.1	0.42	0.34	1 042
2	11 657.7	0.45	0.36	1037
3	12 124.8	0.45	0.39	1 051
Mean values (σ_x)	**11 878.9 (234.5)**	**0.44 (0.02)**	**0.36 (0.03)**	**1 043 (7)**
Unils	**× 10^3 M m^{-2}**			**kg.m^{-3}**

VibroAcoustics Performance

Biot parameters are ideal for the computation of acoustic performance data and for the design of new acoustic trim. Since the Biot parameters represent the intrinsic properties of the material, one can use the same material with various thicknesses without the need to perform any additional experimental measurement. This, of course, assumes that the thickness variation does not involve any compression. If compression is involved, the flow resistivity, porosity and the other parameters will be different. The compressed material therefore constitutes a new and different acoustic material. This is commonly the case with fibers compressed into a particular thickness. Sample from the final compressed material should be measured instead of the initial uncompressed ones.

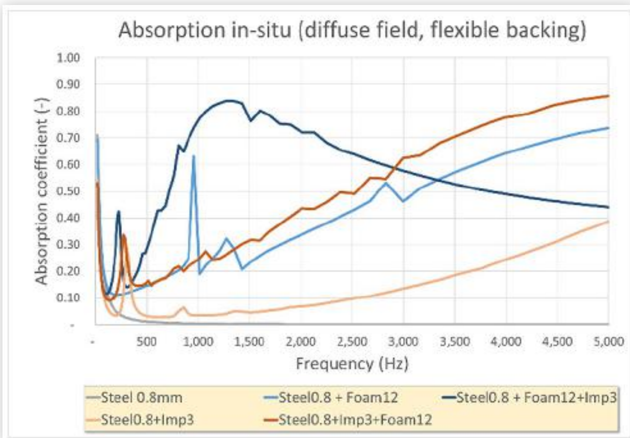
When designing a new acoustic trim material, one should consider the rest of the vehicle acoustic components and the requirements for the actual designed part. Some vehicle might require a higher surface absorption and a lower transmission loss than traditional spring/mass acoustic trim. If the new acoustic trim reduces significantly the need for beading and damping treatments to be applied on the base panel, this also constitutes a great cost and weight reduction and will play a role in its selection.

The following sections presents acoustic results for the 2 layers measured in the previous section stacked in various order on a steel panel of 0.8mm of thickness. The "foam only" layer is 12mm thick (labelled Foam12 in graphs) and the "impregnated foam" layer is 3mm thick (Imp3 in graphs). This section looks at the absorption coefficient of the layers when staked over the steel plate and excited by a diffuse acoustic field. It also looks at the airborne performance of the layers and the order in which they are stacked. Finally, the structureborne performance of the layup is studied using a "rain on the roof" (ROR) type of excitation.

Diffuse Field Absorption

Figure 9 shows the absorption coefficient of various orders of the layers when glued to a 0.8mm steel panel and excited by a diffuse acoustic field. As expected, the steel panel by itself shows no absorption. When the 12mm foam (Foam12) is added to the plate, the absorption increases monotonously with frequency. Additionally, when the 3 mm impregnated foam

FIGURE 9 Absorption coefficient of measured layers in various order under a diffuse field excitation and placed over a 0.8mm flexible steel plate backing



(Imp3) is added on top of the layup, the absorption sharply increases at lower frequency (~ to 1200 Hz) before declining monotonously over frequency, as would be expected of a spring/mass system. If the order of the porous layer is reversed, the mass layer is then glued to the steel panel and the spring/mass system effect disappears. Further analysis and optimization could be done to actually increase the absorption over the full frequency band by using more than one "impregnated" and "foam only" layers and varying thicknesses of all layers.

Airborne Transmission

An important airborne performance criteria of an acoustic trim is its transmission loss (TL).

Transmission Loss Figure 10 shows the TL of a 0.8mm of steel alone, then with a 12mm "foam only" glued to it, then with a 3mm "impregnated foam" added to the layup. A similar exercise is done with the 3mm "impregnated foam" glued to the steel plate and the 12mm foam added on this layup. It is clear from the graph that the spring/mass effect of the "Steel+Foam12+Imp3" provides the best TL over the full frequency range of interest.

Structureborne Transmission

The structureborne contribution to the vehicle interior noise can be quite significant, especially nowadays with the multitude of new electric vehicle designs. Unfortunately, the requirements for structureborne noise reduction are in conflict with those of airborne TL and absorption. This challenge can even be increased if the structural engineers have added beads on the base steel panel which increase the radiated power outside of a few narrowband low frequencies as illustrated in 15. The authors show that when beading is added to a panel, although it reduces the vibration response at a few discreet frequencies, the panel radiation efficiency and power radiated increases on a broad frequency range. Therefore, a solution providing a large amount of damping should be favored.

FIGURE 10 Transmission loss of 0.8mm steel plate with the addition of the measured layers in various orders

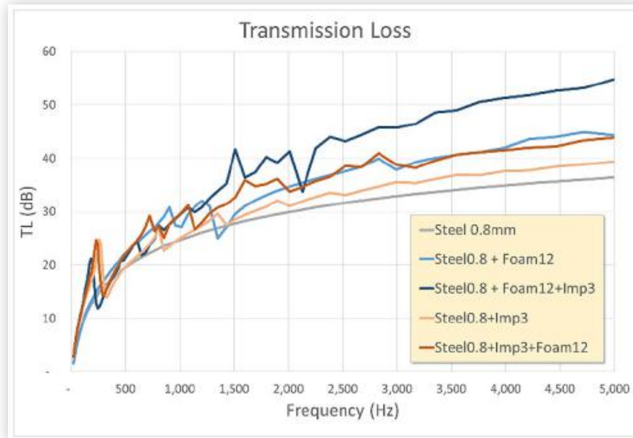
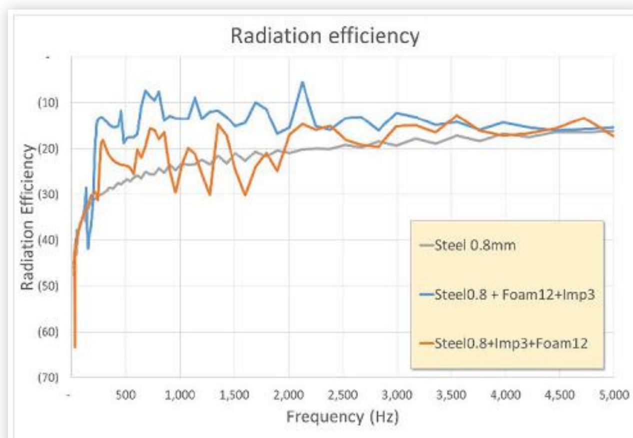


FIGURE 11 Radiation efficiency of 0.8mm steel panel alone and covered with both measured material in various orders



As a reminder, the power radiated by a panel (Π_{rad}) is the product of the quadratic velocity (v^2_{rms}) of the panel and the radiation efficiency (σ). The remaining terms (A: Area, $\rho_o c$: Characteristic impedance of air) are constants when studying variation of a trim on a given panel

$$\prod_{rad} = \sigma A \rho_0 C v^2_{rms} \quad (1)$$

Therefore, the radiated power depends on the amplitude of vibration levels and radiation efficiency of the base panel covered with any trim.

Radiation Efficiency Figure 11 show the radiation efficiency of bare 0.8mm steel panel and the case where Foam12 or Imp3 is glued to the base panel and the remaining foam layer added onto the layup. It is clear that the case where the impregnated layer is in contact with the base panel yields a much lower radiation efficiency compared with the spring/mass system. A full order of magnitude increase in radiation efficiency over low to mid-frequency domain is non-negligible.

Acoustic Intensity (Closely Related to Acoustic Power) Similarly, the case where the impregnated layer is

FIGURE 12 Acoustic intensity of 0.8mm steel panel and covered with both measured material layers in various orders

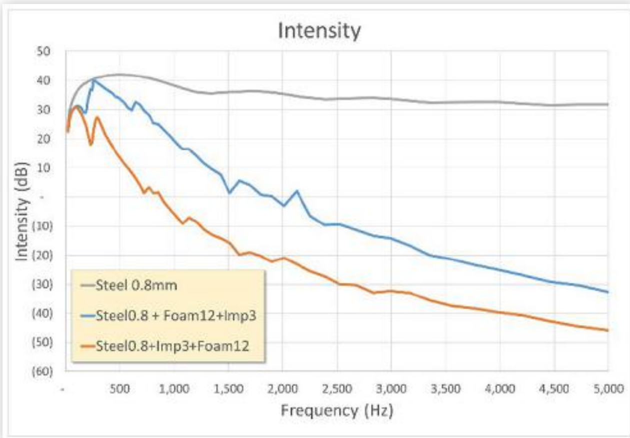
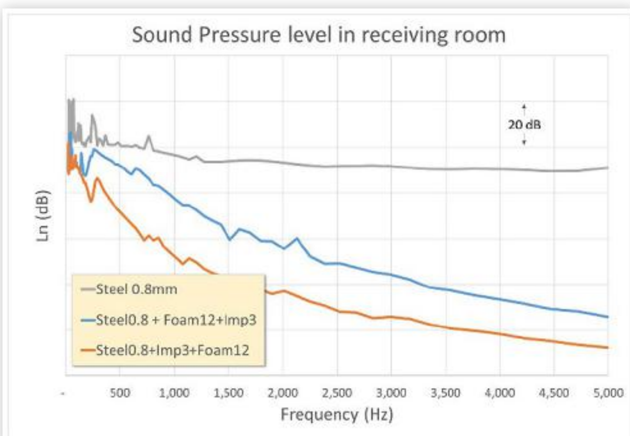


FIGURE 13 SPL of receiving cavity connected to a 0.8mm steel panel alone and covered with both measured material layers in various orders



in contact with the base panel provides a much lower radiated acoustic intensity (related to power with an area term) than the spring/mass system case by a full order of magnitude over the full frequency domain.

Sound Pressure Level in Receiver Cavity Similarly, the sound pressure levels inside an anechoic receiving room connected to the steel panel with the impregnated foam glued to its face yields a much lower SPL, by as much as 20 dB, compared to the spring/mass system.

Conclusion

This study has shown how Biot parameters can be used to predict early in the design process acoustic performance of various acoustic materials and configurations. In particular in the case of O.C.-PREN SC NV, it has been shown that depending on which side the layup is glued to the base panels, results can vary drastically and this can be taken advantage of, when designing an acoustic solution. These Biot parameters can later on be used, in the design process, in a more detailed representation of the trim component in a FEM model where thickness variation of each layer and air gaps can be integrated into the simulation to get even more accurate predictions results.

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Contact Information

The authors invite anyone interested in virtual/physical testing of this new material in your own application or to try the new database of Biot parameters dBPorous to contact them.

Denis Blanchet
denis.blanchet@dbvibroacoustics.com
+420 604 494 511

Michael Feist
Michael.Feist@odenwald-chemie.de
+49 (0) 1727399029

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Definitions/Abbreviations

σ - Flow resistivity

φ - Open Porosity

α - Tortuosity

Λ - Viscous Characteristic Length

Λ' - Thermal Characteristic Length

E - Young's Modulus

ν - Poisson's Ratio

η - Damping Loss Factor (DLF)

ρ - Density

TL - Transmission Loss

SPL - Sound pressure level (SPL)

TMM - Transfer Matrix Method

ROR - Rain on the roof

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