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Acoustic Trim Modelling: Traditional Spring/Mass System vs Biot Theory

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ABSTRACT

Traditionally in the automotive industry, the acoustic trim has been modelled using a series of spring/mass/dampers added to an FE description of the structure. This simple method has deficiencies in accuracy even in the low frequency domain and could therefore not be used with confidence as a predictive method over 150 Hz. Other approaches literally ignored the effect of trim on the structure and focused at representing the acoustic trim absorption in the interior fluid.

In recent years, new developments have shown that the physics of acoustic trim can be well represented when modelling the porous layers using Biot parameters. In fact, since the Biot theory describes the interaction between the acoustic trim, the structure and the fluid using intrinsic properties of the foams and fibers, this modelling approach can be considered predictive over the full audible frequency range. It has been shown for many years now that for high frequency analysis using Statistical Energy Analysis (SEA), Biot parameters play a critical role in the predictive character of a model and therefore its accuracy. Similarly, combining Biot theory with the use of Finite Element Method (FEM) to model acoustic trim in low frequency has shown a significant improvements in the accuracy of predictions and has pushed the upper limit of FE trim simulation over 400 Hz.

This paper introduces the Biot theory approach and the interaction between trim, structure and fluid. It also presents a comparison between traditional and Biot theory approach on an academic and industrial case. Sensitivity of the response to Biot parameter values is also presented.

1. INTRODUCTION

Over the last few years, a great deal of work concerning the modelling of trim in FEM has been published. The development of a new (u,p) formulation for the representation of trim in FEM using Biot parameters has provided researchers and engineers the opportunity to investigate further the physics involved with full vehicle and component trim modeling [1,2,3]. This paper discusses the existing methods used in the automotive industry to represent an acoustic trim component and compares in particular the Non-Structural Mass (NSM) approach with the Biot Theory formulation. In this study, the acoustic trim is defined as the components that contribute to a better sound quality in the vehicle. For example: carpet, dash insulator, headliner, seats, trunk floor and wheelhouse covers…

2. OVERVIEW OF ACOUSTIC TRIM REPRESENTATION

In traditional FEM modeling, it is common practice to add non-structural mass and increase the structural damping of the base panel or the acoustic damping of cavity in order to represent the acoustic trim. One difficulty encountered with this approach is to represent a complex component both in material properties and shape with a uniform distributed mass and damping. The interaction between the acoustic trim, the base panel and the adjacent fluid is a complex phenomenon difficult to represent with NSM and structural damping alone. "While NSM models are good for describing local impedances they are not particularly well suited for predicting interior noise in a fully trimmed vehicle. The acoustic trim in a vehicle has three effects: (i) it adds acoustic absorption to the cavity and modifies the natural frequencies of the cavity), (ii) it adds mass, stiffness and damping to the body structure and (iii) it isolates or decouples the structure and cavities. Below the first cross-resonances of the sound package the impedance of the sound package is indeed mass-like, therefore the NSM method is adequate. However, above the first cross-resonance, the impedance becomes a complex function of frequency and depends on the detailed internal dynamic behavior of the sound package" [4].

Biot Theory Approach

In the past ten years, the use of Biot parameters to represent acoustic trims porous material has been steadily increasing in all major industry sectors such as automotive, aerospace, train and marine industry [5,6,7,8,9,10,11]. This progress has been greatly due to the fact that affordable techniques to characterize porous material have been developed. Direct measurement of all five Biot parameters is possible and indirect methods have even simplified testing to a simple impedance tube (Kunt's tube) measurement [12]. The five Biot parameters are flow resistivity, porosity, tortuosity, viscous and thermal characteristic length. For a foam material, the structure of the foam is also accounted for using the following properties: Density, Young's modulus, Poisson's ratio and structural damping. An elastic porous material is made up of a solid skeleton portion, or frame, and a fluid portion. The energy transfer of three different wave types within the material controls the vibro-acoustic performance of elastic porous materials. These three types of waves are composite waves in that each wave type is present in both the frame and the fluid. Of these composite wave types, one compression wave and one shear wave have properties predominately influenced by the frame properties. The properties of the other compression wave are strongly influenced by the fluid properties. The behavior of these waves is defined by the physical properties of the fluid and bulk properties of the elastic porous material. The elastic porous model is used for foam materials where the stiffness of the frame is important in vibro-acoustic response of the noise control material. The energy exchange between structural energy and acoustical energy within a foam material typically provides much of the desired energy absorption. The full elastic porous model requires all the fluid properties and the elastic bulk properties. For full details of the theoretical development refer to the documentation provided in [13]. The set of Biot parameters represent the intrinsic properties of the porous material and can therefore be used in any predictive models using either a transfer matrix method (TMM) or a finite element representation of the poro-elastic material (PEM).

Transfer Matrix Method (TMM)

The methodology is based on the representation of plane wave propagation in layered media, in terms of transfer matrices. Each layer is assumed to be constructed of a homogeneous and transversely isotropic material. The elastic and fluid layer models are based on classical plane wave propagation. The model used to represent foam and fiber materials is based on an extension of the Biot theory of porous media to elastic porous acoustic materials as presented by Allard [13,14,15].

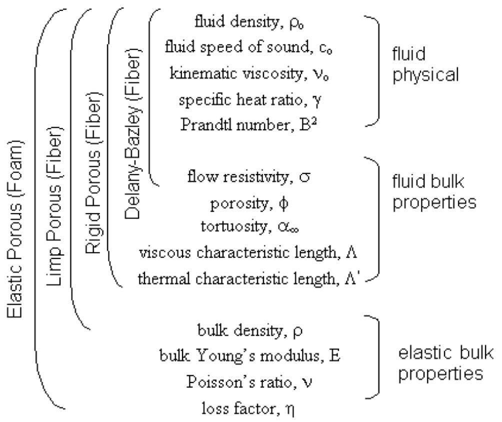

Figure 1. Different theoretical models

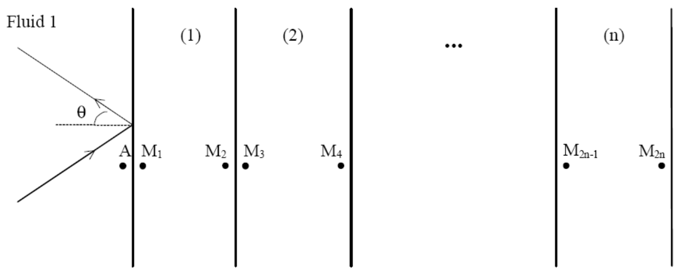

Figure 2. An acoustic plane wave incident on a multi-layer

To compute the global transfer matrix, one has to compute transfer matrices of each layer. If a layer is made of foam or fiber, different theoretical models can be used depending on the number of parameters available (see Figure 1). Then one need to define the continuity conditions between two adjacent layers and finally define the termination conditions on each side of the trim (see Figure 2). The TMM is a 2D approach and assumes that the acoustic trim is flat and infinite in size. For smaller trim, a size correction can be applied. This trim representation has been widely used in the automotive, aeronautical and marine industry where SEA models are routinely built to design sound package of a vehicle, aircraft or ship [5,7,9,16]. It is also possible to apply a TMM trim over a FEM panel. This unique use of the TMM allows a closed full vehicle BIW FEM model to be acoustically trimmed in a very short time [13,17]. It has been shown that TMM has its drawbacks and that for curved and fairly stiff acoustic trim or trim that exhibit a large thickness variation a 3D full FEM description of the trim might be necessary[18,19].

Poro-Elastic Material (PEM)

The propagation of elastic and acoustic harmonic waves, with an $e^{-i\omega t}$ time dependency in porous elastic media is governed by the modified Biot's equations. Refer to [1,3,14] for a complete theoretical derivations of these equations. As described in [12], the FEM trim of a full vehicle analysis can be added to the classical FEM structure/fluid coupled linear system as a trim impedance matrix $\widetilde{Y}$. The dynamic equation of the trimmed vehicle can be written in the following form:

$$\left[\begin{pmatrix} Z_s & C_{sc} \\ C_{sc}^t & A_c \end{pmatrix} + \begin{pmatrix} \widetilde{Y}_{ss} & \widetilde{Y}_{sc} \\ \widetilde{Y}_{sc}^t & \widetilde{Y}_{cc} \end{pmatrix}\right]\begin{bmatrix} U \\ P \end{bmatrix} = \begin{bmatrix} F \\ Q \end{bmatrix} \tag{1}$$

Where Z_s is the mechanical impedance of the master structure (car body in white), A_c is the acoustic admittance of the internal cavity. C_{sc} is the surface coupling operator between the untrimmed master-structure surfaces directly in contact with the internal acoustic cavity. U is the displacement field vector of the master-structure, P the pressure field of the internal cavity; F the external force field applied to the master-structure, and Q represents internal acoustic sources. The matrix Y R'YR is the transferred impedance matrix of the porous component where R is the transfer operator relating the degrees of freedom of the porous component to the degrees of freedom of the master structure and of the internal cavity. Linear system in equation (1) is solved using structural and acoustic normal modes. This has the great advantage of keeping the trimmed linear system to be solve the same size as the initial BIW linear system (No additional DOF).

Material Characterization. In order to benefit from the Biot equations and their software implementation, the five Biot parameters for each porous material are necessary. Each Biot parameter can be measured individually. This approach is less and less used since nowadays the Biot parameters can be derived with confidence from a simple impedance tube test (Figure 3). This simple approach is discussed in [12,15,20]

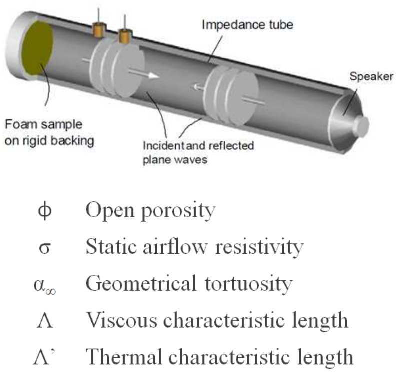

Figure 3: Biot parameters to identify

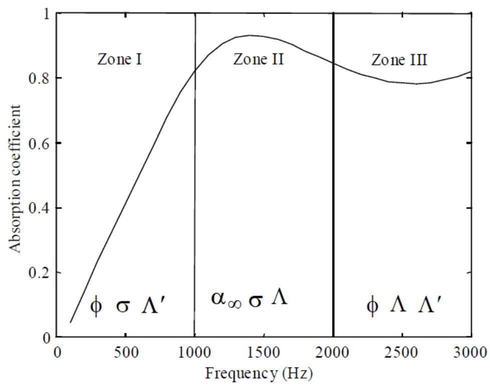

Figure 4: Frequency Zones of a Typical Biot parameter

2. ACADEMIC AND INDUSTRIAL VALIDATION CASES

Description of RTC3 and industrial cases

To illustrate the use of the Biot formulation and parameters in a PEM model, two different example models were built. The first model is called RTC3 (Figure 5). It consists of a steel panel within a frame and a cavity on top. The acoustic trim, an open cell foam and a mass layer, is placed on top of the steel panel (see Table 1 for properties). The panel is excited by an impact hammer and average vibration and SPL are measured. Refer to [21] for model setup details and experimental results reproduced here. The industrial case is a floor panel on which is placed the same acoustic trim as in the RTC3 example. The panel is connected to a vehicle cavity (see Figure 6). The floor panel is excited along its edges with a constrained acceleration level using the large mass method. The damping in the cavity is set to 3% to account for damping that would be coming from other trim not modeled explicitly here. Four microphones are located at the outer ear position of each vehicle occupant.

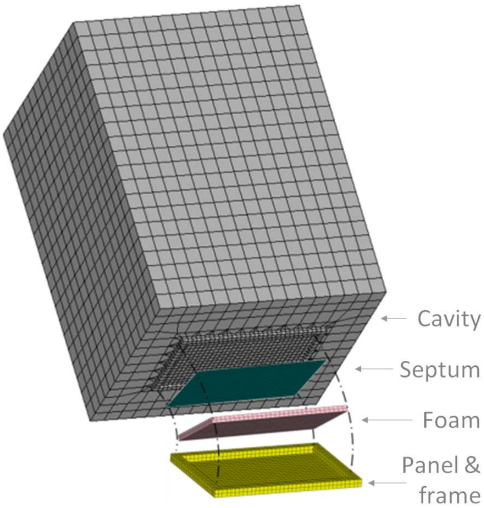

Figure 5. General arrangement of RTC3 case.

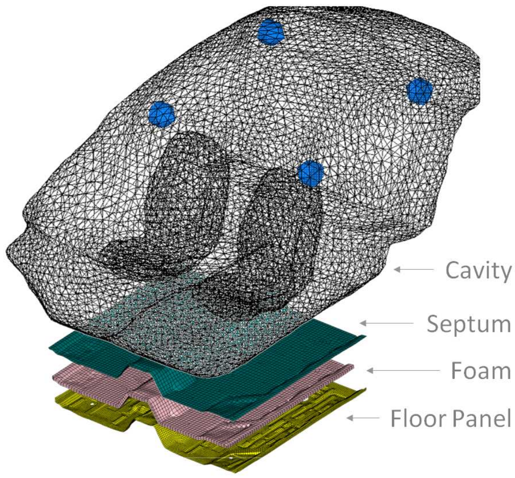

Figure 6. General arrangement of industrial case. Note microphone positions in blue

Table 1: Physical properties of base panel and acoustic trim for both cases

	Thick (mm)	E (Pa)	ρ (kg/m^3)	ν (-)	η (%)	Φ (-)	σ ($N.s/m^4$)	α_∞ (-)	Λ (10^{-6}m)	Λ’ (10^{-6}m)
Panel RTC3	1.05	2E11	7820	0.29	3.0	--	--	--	--	--
Panel Floor	0.8	2E11	7820	0.29	3.0	--	--	--	--	--
Foam	20	4.2E4	62.8	0.3	0.17	0.93	1.9E4	1.7	40	120
Septum	2	--	2500	--	--	--	--	--	--	--

RTC3 Results

The simulation and test data presented here for the Biot case have been done without any tuning (blind simulation results).

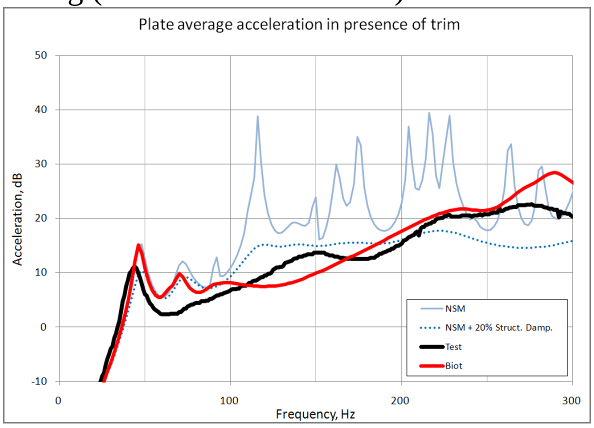

Figure 7: Plate average acceleration

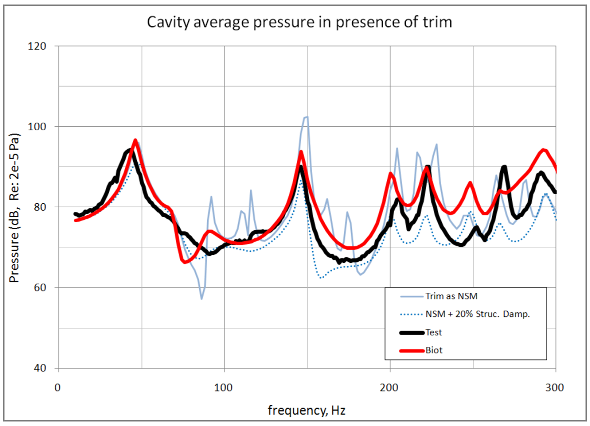

Figure 8: Cavity Average Pressure

The presence of the trim has a dramatic effect on the panel velocity dampening all the peaks except for the first few ones (see Figure 7 and [21]). The Biot formulation predicts this phenomenon very well. The NSM approach alone cannot predict the correct behavior of the plate. A trial and error approach was used to find a structural damping value (20%) to apply to the base panel in order to get the right order of magnitude of the panel velocity. The same phenomena can be observed with the average pressure in the cavity (see Figure 8). Note that NSM by itself is insufficient to represent the physics of the problem.

Industrial case: Floor panel Results

In the following results, no test data are available. Since the Biot approach has been validated in numerous papers, the Biot results will be considered the reference results. The average velocity of the floor panel is similar to the RTC3 case: most of the peaks of the BIW response have been dampened by the presence of the acoustic trim modeled as Biot (see Figure 9). For the NSM approach, a 20% damping on the base panel was applied to get the low frequency NSM curve to agree with Biot. NSM method fails to predict correctly higher than 150 Hz. For the average SPL at the four microphone position, the Biot and NSM agree fairly well up to about 200 Hz where they start diverging (see Figure 10).

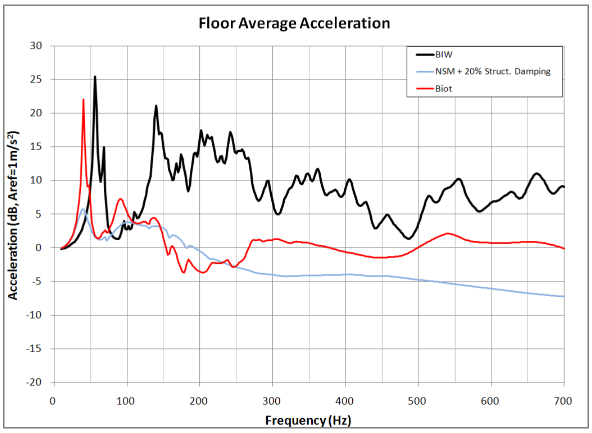

Figure 9. Comparison of floor acceleration between BIW, NSM (+20% Struct. Damp.) and Biot approach

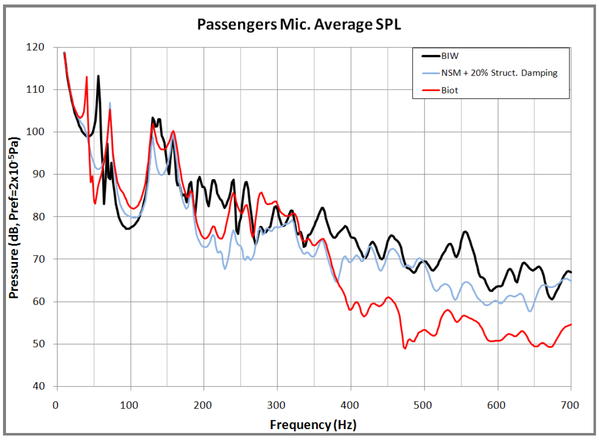

Figure 10. Comparison of average mic. SPL between BIW, NSM (+ 20% Struct. Damp.) and Biot approach

This limiting frequency has been observed in many other full vehicle analyses and it is fair to say that below 150Hz, when the right damping is used, the NSM correlates fairly well with Biot.

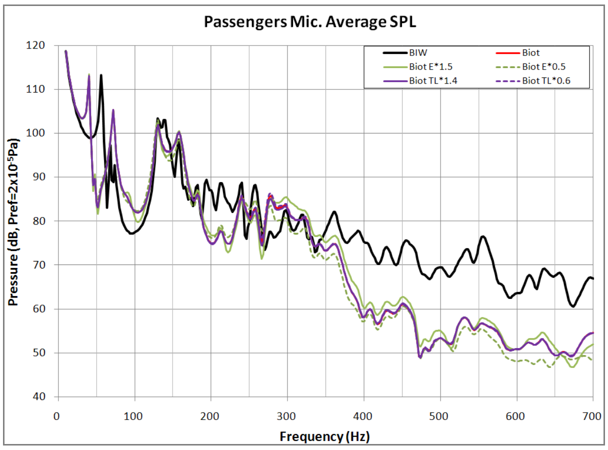

Figure 11. Variation of Young's modulus and thermal characteristic length

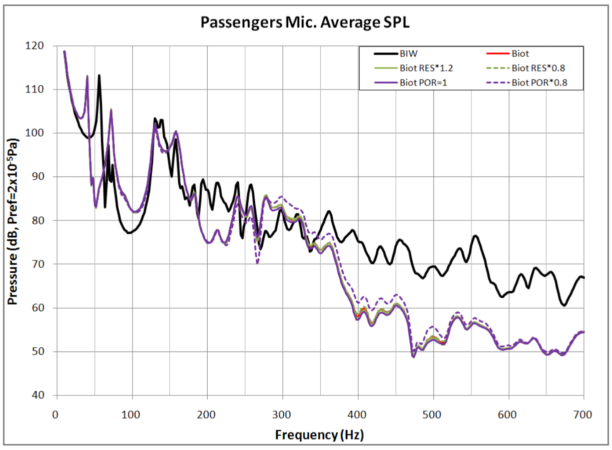

Figure 12. Variation of flow resistivity and porosity

Nevertheless, to find the right damping is not an easy task when one wants to adopt a fully predictive approach. In the case studied here, the top layer is a septum. What if the top layer was absorptive? What would be the damping to apply to the cavity or the structure? What if the porosity of the foam or fiber changes without changing the mass of the acoustic trim? How could the NSM be predictive in such a case? The Biot approach on the other hand is predictive and all these parameters can be varied to allow design studies and material selection that would provide the best performance for the budget available. Also, designing a trim on a wider frequency domain provides a better insight into the effect of a trim not only on the first few peaks of the response, but on a broader frequency range enabling the designer to take more enlightened decisions. According to Figure 4, the Biot parameters that are likely to affect the frequency domain of interest are porosity, resistivity and thermal Characteristic length. The Young's modulus variations are also presented. Figure 11 shows that reducing Young's modulus of the foam can decrease SPL by as much as 5 db and modifying the thermal characteristic length has no influence for this foam. Figure 12 shows that a porosity close to one provides the best results. Flow resistivity has almost no impact on the response. Finally, an acoustic trim with the foam layer of varying thickness and same total mass as the uniform foam layer was used and compared to NSM (see Figure 13 and Figure 14). The NSM still can't follow the Biot curve to frequencies higher than 180 Hz.

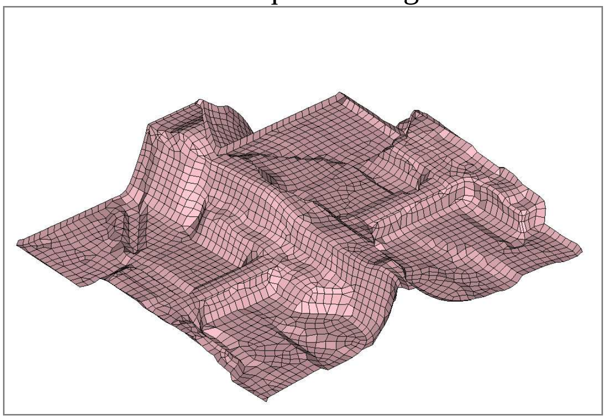

Figure 13. Acoustic trim thickness variation

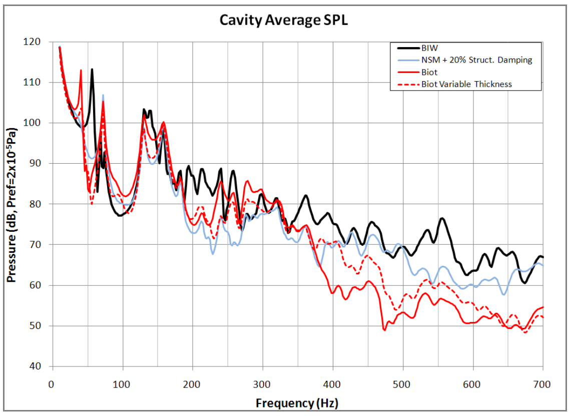

Figure 14. Effect of thickness variation on SPL

4. CONCLUSIONS

This paper has demonstrated that the Biot approach of representing complex acoustic trim has been widely used in high and low frequency domain. It has also demonstrated that the NSM is limited in frequency to below 180 Hz and that over this limiting frequency the Biot approach represents more accurately the interaction between the base panel, the acoustic trim and the cavity connected to it.

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6. ABBREVIATIONS

Symbol	Definition
E	Young’s Modulus
ρ	Density
ν	Poisson’s Ratio
η	Structural Damping
Φ	Open Porosity
σ	Static airflow resistivity
α_∞	Geometrical tortuosity
Λ	Viscous characteristic length
Λ'	Thermal characteristic length