Investigation and development of a numerical tool for the prediction and influence of natural fibre poroelastic trim behaviour on automotive cabin noise

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Abstract: In recent years, the automotive industry has been under increasing pressure to produce products that are environmentally friendly. Also, due to competition among automotive manufacturers and customer expectation, the automotive industry is under continuous pressure to reduce product lead time. Hence, in the automotive industry, there exists a need to reduce noise levels within the vehicle cabin and reduce product development time. Therefore, this research investigates the effect poroelastic materials have on the internal vehicle cabin noise levels due to the structural vibration induced from road/tyre interaction in the 20-200 Hz frequency range. This was achieved by developing an effective and efficient finite element analysis (FEA) tool that is capable of predicting the change in sound pressure levels (SPLs) due to the use of natural fibre trims in the vehicle cabin. A fully scaled (one fifth-scale) vehicle model using ABAQUS FEA software was developed, employing similarity scaling laws, in order to predict the influence various types of poroelastic trims have on internal vehicle noise. The results indicate that the implemented poroelastic materials reduced the SPL by as much as 2.935 dB. Thereafter, a fully scaled experimental model was developed in order to validate the FEA model developed. Validation of the FEA model was achieved by performing a complete modal and SPL analyses on the experimental model and thereafter, the results were compared. The results indicate that the FEA model developed is capable of accurately predicting the influence that the inserted poroelastic natural fibre trims have on internal cabin noise levels.

Subjects: Composites; Acoustical Engineering; Automotive Technology & Engineering

Keywords: vibration; automotive interior noise; finite element analysis; natural fibres; ABS

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PUBLIC INTEREST STATEMENT
This paper investigates the effect natural fibre poroelastic materials have on vehicle cabin noise levels. A finite element analysis (FEA) software was used to simulate a scaled vehicle in order to quantify the effect a natural fibre porous trim had on internal cabin noise levels. It has been found that the natural fibre porous trim does in fact reduce the internal cabin noise levels. A physical scaled vehicle was manufactured in order to validate the FEA model. A high level of correlation was found between the FEA and physical model.
1. Introduction

In today’s fast-paced world, there is an ever-growing need for quicker product development and testing. This stems from customer demand and the growing competition between automotive manufacturers. Therefore, in the automotive industry, there is a need for computational effective and efficient numerical tools that are able to reduce lead times in the testing and development of new products.

In addition, the need for innovative recyclable and environmentally-friendly materials is of an ever-increasing trend. This is due partly to concerns over rising costs, negative environmental impact and unstable supply of fossil fuels (Sen & Reddy, 2011). The automotive industry in particular is under increasing pressure from governing bodies. These governing bodies, e.g., the European Commission, are pushing for revised legislation and regulations concerning the issue of product recycling, carbon dioxide (CO₂) emissions and the protection of resources (Ghassemieh, 2011; Pickering, 2008). One of the best efforts that can be made in reducing environmental pollution is by implementing the use of natural fibre materials (Dunne, Desai, Sadiku, & Jayaramudu, 2016). By using natural fibre materials, in the production of products, a better recyclable and environmentally-friendly vehicle can be produced. Therefore, in this study natural fibres will be used, in the context of vehicle cabin noise, to produce a sound absorbing material that will act as an internal trim within the scaled model.

Hence, this paper presents the modelling of a scaled-down trimmed vehicle for the prediction of the internal sound pressure levels. More importantly the novelty of this study, provides a FEA model that can predict the influence that a poroelastic natural fibre material has on the internal cabin noise levels. From the literature, it is evident, as presented below, that this phenomenon has not been fully demonstrated.

2. Literature review

Indications from the literature, suggest that (Koopmann, 1980) were most likely the first researchers to have seriously applied modelling techniques to vehicles. The method used to predict the noise inside the vehicle cabin, was based on the aircraft analytical solution. However, simplifying assumptions, for example, the vibration of the body shell, considered to be unaffected by the acoustic volume, were made. Furthermore, they argued that this assumption was realistic, since the generalized stiffness of the surrounding body structure, is considerably greater than that of the enclosed acoustic volume.

In 1982, Nefske, Wolf and Howell were some of the early researchers to use finite elements to predict vehicle internal noise, when they formulated the fully coupled problem, by including a boundary surface acceleration vector in the equations of motion of the air, and boundary pressure vector in the equations of motion of the structure (Lalor & Priebsch, 2007).

(Campbell, Abrishaman, & Stokes, 1993), developed a vehicle FEM model that included trim damping effects. The internal trims damping was applied by using empirical damping values and the vehicle seats were treated as solid boundaries. It should be noted that the internal trims were not modelled and therefore, only an empirical damping value was applied to the surfaces.

In addition, an experimental-numerical method for the determination of the relevant acoustic parameters for various metal panels, covered with typical trim materials, was described by (Tinti, Scaffidi, & Perazzolo, 1997). This was achieved by means of a modal superposition method, used to find the acceleration on the boundary of the panels under investigation.

(Brechlin, Bosmans, Keymeulen, & Dekkers, 2002), presented an experimental study on the effects that trimmed components have on the vibro-acoustic behaviour. It was found that the effect each trimmed component had on the noise levels, was very different from each other and cannot be summarized by one or two general guidelines. There was no numerical modelling presented in their study.

A hybrid FE-SEA method, for the prediction of the interior vehicle noise at the design and development stage, was presented by (Chen, Wang, & Zan, 2011). Reasonably good correlation between the
predicted and experimental results was reported. The sound absorption and insulation effects of the trimmed body were reported to have been taken into account by simply using a damping loss factor. Therefore, no porous sound absorbing materials (trims) were implemented in the model.

A full FE model using NXNastran, considering only the vehicle headliner, was presented by (Cameron, Wennhage, & Göransson, 2010). It was shown that the model presented, correlated well with the experimental data in the frequency range of between 100 – 500 Hz. The interior headliner was modelled as a three dimensional solid using 3D elements and given equivalent homogeneous properties, based on sandwich theory. The acoustic materials that were modelled were structural and acoustic foams. It should, however, be noted that acoustic damping was achieved with surface impedance, applied on the interior side of the headliner.

Therefore, it can be deduced from the above that realistic simulations in the field of acoustic-structural improvement that considers the mass and thickness of trimmed components, together with experimental verification, are limited in number. More importantly, none of the researchers has considered the effects that thickness, mass and fibre type of trimmed components have on the panel vibration or on the interior noise levels.

3. Damping material and numerical model development

3.1. Natural fibre damping material implemented

According to the literature (Saad & Kamal, 2012; Asdrubali, 2006; Azevedoa & Nabuco, 2005), kenaf and sisal natural fibres possess a number of good properties, as mentioned below. Therefore, these fibres were selected as basic constituents for the development of the final composite due to their low cost, low density, good toughness, recyclability, good sound absorption properties, favourable airflow resistivity and biodegradability. Two newly developed poroelastic natural fibre composites with different ratios of kenaf are presented in this study. The matrix employed to bind the fibres was obtained from discarded 3-D printing acrylonitrile butadiene styrene (ABS) scrap. This was done in order to improve the recyclability and environmentally friendly nature of the proposed composite. The material properties of these new composites were obtained by physical testing and can be found in Table 1, and seen in Figure 1. These data are necessary for the FEA model. The poroelastic natural fibre material developed was attached to the interior walls of the vehicle cabin (named as treated or trimmed surfaces) and its effectiveness, in terms of damping, is investigated in this paper.

The Delany-Bazley empirical model was used to predict the sound absorption coefficient and was plotted using a Python script. The Delany-Bazley empirical model was used to predict the sound absorption of the proposed poroelastic natural fibre composites, since it was the best empirical model available (Dunne, Desai, & Sadiku, 2017a).

3.2. Development of numerical model

The ultimate purpose of this study was to develop a predictive tool, by using FEA software that can efficiently and effectively predict the SPLs within a vehicle cabin, treated with the stated natural fibre composite developed. However, a well-developed, scaled-down FEA model has the advantage of predicting accurate resonance frequencies and SPLs at a considerably lower computational cost; this is due to its size and hence, the attractiveness of it. Furthermore, one of the fundamental structural design considerations for a vehicle, is the overall dynamic behaviour in both torsion and bending (Nel, 1997).

| %kenaf/ %sisal | Bulk Density kg/m$^3$ | Permeability m$^2$s$^{-1}$ | Void Ratio | Log Bulk Modulus | Shear Modulus MPa | Tensile Strength kPa | Airflow resistivity Pa.s/m$^2$ |
|----------------|----------------------|----------------------------|------------|------------------|------------------|------------------|--------------------------|
| 60/40          | 77.868               | 0.00405                    | 14.968     | 0.0573           | 9.6836           | 107.353          | 2848.441                 |
| 40/60          | 93.6129              | 0.00325                    | 12.255     | 0.212            | 3.256            | 146.146          | 3688.94                  |
Hence, the torsional and bending stiffness are of primary importance for dynamic similarity in the development of a scaled-down model. Subsequently, a stiff, one-fifth, scaled-down vehicle structure was constructed using similitude theory in order to maintain the approximate relative occurrence of acoustic and structural modes, over the low-frequency domain, that coincide with a typical full-size real vehicle structure. The dimensions of the scaled vehicle were $716 \times 326 \times 232$ mm. The actual in-service conditions in terms of loading (scaled accordingly), contact and damping phenomena, have been considered so that the dominant physics and relevant phenomena can be adequately represented.

The dynamic similarity scaling laws employed, in order to govern the scaled vehicle model behaviour, are as follows (Chakraborty, 2011):

$$K_m = \frac{1}{N} K_p$$  \hspace{1cm} (1)

where $K_m$ is the stiffness of the scaled model, $N$ is the scaling factor $= 5$ and $K_p$ is the stiffness of the full scale vehicle and

$$F_m = \frac{1}{N^2} F_p$$  \hspace{1cm} (2)

where $F_m$ is the force acting on the scaled model and $F_p$ is the force on the full scale vehicle.

(Desai, 2010), reported that the bending stiffness of a typical vehicle is in the range of 1.5 kN/mm. Thus, applying Equation (1), the bending stiffness of the scaled model should be approximately 0.3 kN/mm. (Tebby, Esmailzadeh, & Barari, 2011), observed that the torsional stiffness target for a typical vehicle is in the range of between 10 – 18 kN.m/rad. Thus, applying Equation (1), the torsional stiffness of the scaled model should be approximately 2.8 kN.m/rad. By careful design, the bending and torsional stiffness of the developed scaled model was numerically computed to be 0.212 kN/mm and 2.207 kN.m/rad, respectively. However, it should be noted that due to the nature of vehicle stiffness, many researchers do often, report a range of varying values. Hence, the stiffness's numerically computed are in compliance to the above values. The simulated scaled model in ABAQUS finite element software can be seen in Figure 2.

### 3.3. Modelling approach

The modelling method employed consists of a fully-coupled (body structure-to-interior air volume) vibro-acoustic analysis in order to represent the dominant physics involved, as described in the steps enumerated below. The complete finite element simulation has passed through four steps, which were:
Step 1: Nonlinear static analysis.
Purpose: To include the effects of gravity on the system for improvement of modelling accuracy.

Step 2: Linear perturbation static analysis.
Purpose: To include the computation of residual modes for the purposes of increasing accuracy in subsequent mode-based procedures.

Step 3: Linear perturbation frequency analysis.
Purpose: To extract the coupled acoustic-structural Eigen modes of the system in order for the subsequent structure’s response to the steady-state harmonic excitation to be computed.

Step 4: Steady-state dynamic, mode-based analysis.
Purpose: To determine the vibro-acoustic response of the coupled system that was subjected to a multi-frequency harmonic excitation that acted at an excitation point, which coincided with a front wheel suspension location, described by Equation (8).

The final analysis, which employed modal techniques, was based on frequency response function (FRF) simulations of the fluid-structure interaction between the passenger cabin and its body structure, with respect to the structural excitation.

The vehicle structure and its respective components were modelled as deformable bodies, including the poroelastic fibre developed and the flexible wall panels that enclose the acoustic cavity. Since testing revealed that the natural frequencies for both the structure and acoustic medium span the same range, a coupled approach was adopted as these “vibro-acoustic” modes provide the information necessary to understand the physical phenomena. The discretized equations of motion for a poroelastic material attached to a flexible structure can be written in matrix form as (Mathur, Chin, Simpson, & Lee, 2001):

\[
\begin{pmatrix}
\alpha^2 & \mathbf{M}_{ss} & \mathbf{M}_{sf} & \mathbf{M}_{ff} \\
\mathbf{C}_{ss} & \mathbf{C}_{sf} & \mathbf{C}_{ff} \\
\mathbf{K}_{ss} & \mathbf{K}_{sf} & \mathbf{K}_{ff}
\end{pmatrix}
\begin{pmatrix}
\mathbf{U}_{ss} \\
\mathbf{U}_{sf} \\
\mathbf{U}_{ff}
\end{pmatrix}
+ \begin{pmatrix}
\mathbf{C}_{ss}(\omega) & \mathbf{C}_{sf}(\omega) \\
\mathbf{C}_{sf}(\omega) & \mathbf{C}_{ff}(\omega) \\
\mathbf{K}_{ss}(\omega) & \mathbf{K}_{sf}(\omega)
\end{pmatrix}
\begin{pmatrix}
\mathbf{U}_{ss} \\
\mathbf{U}_{sf} \\
\mathbf{U}_{ff}
\end{pmatrix}
+ \begin{pmatrix}
\mathbf{K}_{ss}(\omega) & \mathbf{K}_{sf}(\omega) \\
\mathbf{K}_{sf}(\omega) & \mathbf{K}_{ff}(\omega)
\end{pmatrix}
\begin{pmatrix}
\mathbf{U}_{ss} \\
\mathbf{U}_{sf} \\
\mathbf{U}_{ff}
\end{pmatrix}
= \begin{pmatrix}
\mathbf{F}_s \\
\mathbf{F}_f
\end{pmatrix}
\]
where $M$, $C$ and $K$ are the equivalent mass, damping and stiffness matrices, respectively and the subscripts, $ss$, $ff$ and $sf$ denote the: solid phase, fluid phase and solid-fluid coupled phases inside the poroelastic material, respectively. $U_{ss}$ is the solid phase displacement and $U_{ff}$ is the fluid phase displacement, respectively. Equation (3) is solved with ABAQUS finite element software by using the modal superposition procedure. Since the acquisition of the actual damping characteristics in complex structures is very difficult, an approximate spectral modal damping scheme is employed, which introduces an energy dissipation term, $\delta W_{diss}$ in the form:

$$\delta W_{diss} = 2 \zeta_i \omega_i \dot{u}_i$$

(4)

where $\zeta_i$ is the $i^{th}$ modal damping ratio, $\omega_i$ is the natural frequency and $\dot{u}_i$ is the velocity. It should be noted that the damping values used in Equation (4) were obtained experimentally and they were post-processed using the Engineering Data Management (EDM) software package.

Finally, the A-weighted SPL response at a particular nodal location $L_p$, is expressed as (Wang et al., 2010):

$$L_p = 20 \log_{10} \frac{p_{rms}}{p_{ref}} + \text{A weighting}$$

(5)

where

$$p_{rms} \approx \frac{p_{max}}{\sqrt{2}}$$

(6)

and $p_{rms}$ is the root mean square pressure, $p_{ref}$ is the standard reference pressure, which is equal to $2 \times 10^{-5}$ Pa and $p_{max}$ is the peak acoustic pressure (Everest & Pohlmann, 2009).

The above descriptions allowed for the simulation of the proposed model. It can be seen from Figures 11–14 that the numerical results closely correspond to the measured results. However, Figures 11–14 only depict the representativeness of the purposed model; they do not give a realistic indication of the effect that a full natural fibre composite trim would have on the internal SPLs. Therefore, an effort was made to simulate the proposed natural fibre composite trim in such a way that it would realistically represent the treated surfaces in a typical vehicle. This will allow for the generation of more useful data in terms of the attenuation in the SPLs due to the application of the poroelastic natural fibre sound absorbing composite developed.

Therefore, it is noted from the literature that up to 20 kg of textiles, carpets and other trim parts are used in the interior of the vehicle (Fung, 2001) and the target thickness for automotive lining is in the range of between 11 – 12 mm (Parikh, Calamari, & Myatt, 2000). Hence, scaling the thickness by a factor of 5 gives an approximate thickness of 2.5 mm. However, the poroelastic natural fibre mass cannot simply be scaled linearly and therefore, needs a scaling law, which can be described as (Ghosh, 2011):

$$m_{scaled} = \frac{m_{actual}}{N^3}$$

(7)

Therefore, taking the mass of fibre, for the carpet and headliner only, in a full vehicle to be approximately 14.5 kg and since the total mass of all textiles, carpets and headliners are approximately 20 kg. and substituting it into Equation (7), a scaled fibre mass of 116 g is computed.

This translates to an internal cabin area coverage of 0.59588 m², which is approximately 66%, for the 60% kenaf composite and 0.4956 m², which is approximately 45%, for the 40% kenaf composite. The area coverage percentage varies due to the density of the composite.

3.4. Mesh design

The structural members and panels of the vehicle frame were modelled in a 3-D modelling space using a deformable type shell extrusion. Surface-to-surface tie constraints were used in order to tie...
all contact surfaces together so that there is no relative motion between them and this method provides better accurate results than the node-to-surface tie constraints.

Meshing considerations in a dynamic structure in FEA is very important and special attention was observed since the number of elements used to model a given structure is a compromise between solution accuracy and cost (Desai, 2010). Although the meshes at the acoustic-structural tied boundary may be nodally non-conforming, mesh refinement, which depends on wave speeds in the two media, affects the accuracy of the solution. The mesh for the medium with the lower wave speed (air) should generally, be more refined and therefore should be the “slave” surface. Since the material properties affect the mesh parameters for wave propagation problems and hence affect the accuracy of the solution, the discretization protocol of the finite element method requires at least six nodes per wavelength. (Donders, 2008), reported that between 6–10 elements per wavelength (structural and acoustic) were used when discretising the parts in order to maintain a reasonable level of accuracy. Mesh convergence studies in the case under investigation, revealed that the use of a mesh with size of 9 mm (for both structure and acoustic media) was ideal with very little loss of accuracy and a substantial saving on the central processing unit (CPU) time. However, this was done cautiously and all structural members were meshed with at least 10 elements per wavelength.

The cabin frame and panels were meshed with first-order, quadratic thin shell, reduced integration elements, of the type S4R with hourglass control in order to reduce the CPU memory requirements. The acoustic domain was discretized by using a three-dimensional, first-order hexahedral, solid brick elements of the type AC3D8 in order to achieve maximum accuracy.

The natural fibre poroelastic composites, as seen in Figure 3, were modelled in a 3D space, by using a solid extrusion technique to form the geometry of the material. The porous material was then assigned properties via the material module in ABAQUS. The properties were assigned as follows: log bulk modulus, shear modulus and tensile strength by using Mechanical–Elastic–Porous Elastic. The permeability and void ratio for the porous material, were assigned using Other–Pore Fluid–Permeability, and the bulk density of the porous material was assigned by using General–Density. The porous material was thereafter discretized by using the first-order Hexahedral, solid brick elements of type C3D8P in order to achieve a high accuracy. This element type allows for the modelling of fully or partially saturated fluid flow, through a deforming porous medium, such as the one under study (Dassault, 2012). All composites in this study were modelled with at least 5 elements per thickness in order to achieve accurate bending behaviour.

The acetoxy silicone material, used to secure the front panel during testing (needed for the cabin accessibility as described below), was discretized by using the first-order hexahedral, solid brick elements of the type C3D8IH. The total system model comprised a total of 313 490 elements.
3.5. Loading and boundary conditions

The typical wheel suspension input forces of a Sedan vehicle, when driving on tarred road surfaces, have a value of \( \sim 100 \text{ N} \) (Nel, 1997). However, the continual changing of the bending and torsional loading on the vehicle structure, due to the irregular surface the vehicle transverses, contributes significantly to the vehicles structural vibration. Therefore, applying Equation (2) in order to preserve the dynamic similarity, gives an input force of \( \sim 4 \text{ N} \). However, it should be noted that the force acting on the wheel is assumed to be harmonic in nature. Therefore, the input forcing function can be expressed as stated by (Rao, 2011):

\[
F_y = 4 \sin \omega t
\]  

(8)

where \( \omega \) is the angular frequency and \( t \) is the time. This loading was applied sinusoidally as a frequency sweep in the low frequency range of between 20 and 200 Hz with an increment of 0.25 Hz.

Note: the dynamics of the system were analysed under free-free boundary conditions.

4. Experimental characterization and setup

In order to initially validate the proposed numerical model (to be used for subsequent analysis), an experimental modal analysis by using the single reference method with shaker excitation was adopted and conducted in order to determine correlation of the resonance frequencies between the numerical and experimental models. The test structure was manufactured and assembled in a similar way to that of an actual Sedan vehicle. All air gaps in the vehicle body were sealed with high-quality acetoxy silicone sealant in order to acoustically seal the interior cavity, thus minimizing acoustic losses. The test vehicle was suspended by using four soft elastic bands in an overhead fashion in order to simulate the free-free boundary conditions used in the dynamic calculations. Subsequently, a steady-state forced response harmonic analysis was carried out in order to measure the vibration of the vehicle cabin panels, in terms of a frequency response function (FRF) and the SPLs within the cabin. All the equipment used was calibrated and operated according to the manufacturers’ specifications and recommendations.

The physical test setup is depicted in Figure 4, below:

The procedure followed for the experimental modal analysis is as follows:

- The first step was the suspension of the scaled vehicle structure by four soft elastic cords. The cords were attached at the top front and back corners close to the nodal points as guided by the numerical simulation results. These elastic cords allow for a close replication of free-free...
boundary conditions, which are needed in a modal analysis for accurate extraction of mode shapes.

- Thereafter, an electromagnetic shaker, Sentek Dynamic exciter, model MS-1000, was attached to the experimental model via a stinger. A stinger was connected to a calibrated PCB force transducer with a sensitivity of 2.248 mV/N, which was attached to the bottom panel of the vehicle, coinciding with the front wheel excitation position. A force transducer connected to the stinger was attached to the vehicle’s base by using good quality cyanoacrylate adhesive, as recommended by (DYTRAN n.d.), which allowed for a rigid bond.

- The next step was the attachment of miniature DeltaTron accelerometers. Special consideration was observed since the mounting method of an accelerometer has an influence on the accelerometer’s frequency response. The accelerometers were to be attached to the right side panel, along its centre line, as depicted in Figure 5. Since the panels to which the accelerometers are to be mounted are very thin (0.8 mm), stud mounting was not practical (DYTRAN n.d.). In addition, the frequency range of excitation is low, hence the accelerometers were attached by using high quality bees wax. The accelerometers were pressed onto the side panel in order to ensure that the thickness of the wax between the side panel and the accelerometer was minimal. This was done in order to reduce any elastic effects that the wax may exert, on the accelerometers, when subjected to excitation. In this study, three accelerometers were used in order to ensure the accuracy of the results and to capture the bending behaviour of the panel more accurately for subsequent analysis.

- Next, the shaker was clamped down by using G-clamps in order to absorb reaction forces, prevent movement of the shaker and to ensure that only normal forces are transmitted to the experimental model.

- The force transducer and accelerometers were then connected to the signal converter and the spectrum analyser Coco-80.

- Since the specially developed poroelastic natural fibre composites were to be attached within the cabin, it was necessary that the front panel be detachable (accessible). Hence, the front panel was attached by using an acetoxy silicone sealant. This allowed for the easy removal of the front panel as well as providing a rigid and acoustically sealed cabin. The poroelastic natural fibre composites were attached to the inner bottom surface of the vehicle cabin by using a very thin layer of ABS, which was painted on the surface, as depicted in Figure 6.

Lastly, a steady-state, forced response, harmonic analysis was conducted; the procedure of which is similar to the above modal analysis with the only difference being the replacement of the accelerometers with microphones, as shown in Figures 7 and 8, in order to capture the SPL’s across the frequency range of interest. In order to minimise the possibility of contaminated data, tribo-electric and sensitivity effects due to cable whip were reduced by using stiff graphite cables. Data cables were separated from power cables in order to reduce electromagnetic interference.
The cables were also checked for sharp bends, cuts or any type of damage since this could inadvertently cause erroneous measurements (Goelzer, Hansen, & Sehrndt, 2001). Relevant precautions were taken in order to minimise mounting resonances. Background noise checks by using the $L_{eq}$ method were also performed before measuring the cabin SPLs in order to ascertain that the background noise had no effect on the measured results. The shaker was driven with a swept sine signal, which was band-limited to the frequency range of interest and coincident with that of the finite element model.
The Coco-80 was configured to capture data at a block size of the time block signals (Block Size/Line) equal to 8192/3600. In addition, appropriate windowing and averaging techniques were selected. Averaging and windowing techniques are necessary in order to achieve the best linear estimate of the system (Agilent Technologies, 2000). The windowing type selected was Hanning and the average number of linear averages taken was set to 32, for the capturing of data. The linear average technique was used, since it improves the signal-to-noise ratio of a measurement (Agilent Technologies, 1994). The frequency range was set at 2300 Hz, since it is important to sample at least 10 times faster than the highest frequency of interest (200 Hz). The frequency range function on the Coco-80 was used to set the sampling rate (sampling frequency), since they are related to each other. The same sampling rate was used for the numerical model.

5. Results and discussion
With a model size of approximately 1.27 million degrees-of-freedom (including contact elements and Lagrange multiplier variables), the model converged at 27.383 minutes of CPU time. The analyses were executed on a 3 GHz i7 Intel acer laptop with 4 GB of RAM, running on a Windows 7 platform. Figure 9, depicts the resonant frequency as a function of the mode numbers for both the experimental and simulated models. The largest difference in the natural frequencies between the numerical model and measured values was ~4.21%. It can be seen that the difference between the experimental and the predicted values is very small, indicating a good modelling accuracy.

Furthermore, it is evident from Figure 10, that the results of the experimental model correlate very well when compared to a full size vehicle. This indeed adds to the validity and builds confidence in the proposed scaled vehicle model.

The experimentally determined damping ratios of the scaled vehicle system varied between 0.022 and 0.263 and were subsequently used as input into Step 4 of the numerical model developed. The correlation between the experimental and predicted A-weighted SPLs at the front driver’s head position and the right rear passenger’s head position, are shown in Figures 11–14.

The data is observed to correlate relatively well over most the frequency range, despite the complexity of the vehicle structure and the poroelastic natural fibre sound-absorbing material employed. The overall results indicate that the numerical model predicts the dominant resonance peaks as well as the general trend of the vibro-acoustic behaviour, thereby, demonstrating the accuracy of the numerical model developed. It should also be noted that the measured and simulated resonance peaks compare very well with that of a full sized vehicle (Zheng, Fang, Tang, Zhan, & Fu, 2015).
Figure 10. Scaled model SPLs versus real vehicle SPLs.

Figure 11. Measured and predicted SPLs at the driver head position for 60% kenaf model.

Figure 12. Measured and predicted SPLs at the rear head position for 60% kenaf model.
The small inaccuracies observed may be attributed to various factors, such as: imperfect boundary conditions (free-free boundary conditions) as well as the microphone connections through the top panel of the structure. Others include: the interference within the measurement equipment, limitations in the physical measurement process, such as the: dynamic interaction between vehicle structure and the shaker (normal forces not perfectly translated through the stinger into the vehicle structure). In addition, are the: inaccuracies in the measurement of the poroelastic natural fibre materials, airflow resistivity, which is used to calculate the permeability used in the numerical model as well as inaccuracies associated with the construction of the experimental model, to name a few.

Depicted in Figure 15–18, are the correlations between the SPLs of the treated and untreated numerical models. From these figures, it can be seen that there is a greater attenuation in SPLs at the front driver head position.

From Table 2, it can be seen that the sound attenuation is greater at the front head position than the rear head positions. In addition, it should be noted that upon observation, certain resonance frequencies are observed to have dropped by as much as 3 dB(A). This is significant since a 3 dB(A) change in SPL is perceivable to the human ear (Brüel, & Kjær, 1984). In addition, it is observed from
Figure 15. Predicted trimmed and untrimmed SPLs, at the driver head position, for 60% kenaf model.

Figure 16. Predicted trimmed and untrimmed SPLs, at the rear head position, for 60% kenaf model.

Figure 17. Predicted trimmed and untrimmed SPLs, at the driver head position, for 40% kenaf model.
Figure 18. Predicted trimmed and untrimmed SPLs, at the rear head position, for 40% kenaf model.

Table 2. Averaged difference between trimmed and untrimmed SPLs over whole frequency range

| Model  | Averaged untrimmed SPLs dB(A) | Averaged trimmed SPLs dB(A) | Average difference dB(A) |
|--------|-------------------------------|-----------------------------|--------------------------|
|        | Front driver head position    |                             |                          |
| 60% Kenaf | 59.93                         | 57.817                      | 2.113                    |
| 40% Kenaf | 59.93                         | 56.995                      | 2.935                    |
|        | Rear passenger head position  |                             |                          |
| 60% Kenaf | 59.176                        | 57.867                      | 1.309                    |
| 40% Kenaf | 59.176                        | 57.152                      | 2.024                    |

Table 2, that the 40% kenaf trim gives the largest reduction in SPLs. This is most likely due to its higher bulk density which has a greater effect on noise attenuation in the low frequency range.

However, greater levels of acoustic damping can be achieved by employing appropriate material distribution optimization algorithms as well as further refinement to the proposed poroelastic natural fibre composites.

The averaged SPLs of the trimmed and untrimmed model were evaluated by (Raichel, 2006):

\[
L_p = 10 \log \left( \frac{1}{n} \sum_{i=1}^{n} 10^{L_i} \right)
\]

(9)

where \(L_p\) is the averaged sound pressure level, \(n\) is the number of SPLs that were taken into consideration and \(L_i\) are the individual SPL points being considered.

For a more comprehensive analysis on the physical capability of the natural fibres ability to reduce noise, (Dunne et al., 2017a; Dunne, Desai, & Sadiku, 2017b) and (Dunne, Desai, & Sadiku, 2017c).

6. Conclusions
In this work, a computational procedure with a numerical model, based on the finite element approach, has been presented in order to predict the acoustic performance as a function of...
a newly developed poroelastic fibre damping material. Achieving a high degree of correlation for complex structures, such as the one under investigation, is rather difficult because simulation models generally represent an ideal discretized world, in contrast to physical experiments, where variability is present.

Despite this, the numerical model developed was successfully validated by actual measurements and was found to be amenable to furnishing reliable frequency response information at the points of interest. Hence, this study proposes both a numerical predictive tool that can be used effectively and efficiently for the prediction of the internal cabin SPLs, as well as offers an alternative recyclable natural fibre sound absorbing product that can be used for the attenuation of the SPLs within the cabin enclosure. The reduction in the noise levels observed is not of a superficial nature, but is perceivable to the human ear, which is advantageous to numerous industries. The insights gained in the noise generation and transmission behaviours of the poroelastic natural fibre trimmed system under investigation, may be materialized into realistic design modifications enabling NVH engineers to move closer towards cabin acoustic refinement in pursuit of their quest for quieter passenger cabins.

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