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Acoustic Model Reduction for the Design of Acoustic Treatments

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Abstract

Due to constant evolution in both noise regulations and noise comfort standards, noise reduction inside the vehicle remains one of the main issues faced today by the automotive industry. One of the most efficient methods for noise reduction is the introduction of acoustic treatments, made of multilayered trimmed panels. Constraints on these components, such as weight, packaging space and overall sound quality as well as the amount of possible material and geometrical combinations, have led automotive OEMs to use innovative methods, such as numerical acoustic simulation, so as to evaluate noise transmission in a fast and cost-effective way. While the computational cost for performing such analyses is insignificant for a limited number of configurations, the evaluation of multiple design parameter combinations early in the design stage can lead to non-viable computation times in an industrial context. This paper presents a framework for the efficient, almost real-time evaluation of quality indicators, such as the sound transmission loss, using machine learning techniques, with data from a limited amount of vibroacoustic simulations. The method is evaluated on several firewall panels covering a large design space, where the sound transmission loss of the panels can be predicted with good accuracy across the frequency spectrum. Furthermore, the method is applied to the design space covering the properties of individual materials with similar outputs. The resulting models can be further used for optimizing the behavior of the acoustic treatment. The performance of the proposed methodology is demonstrated on an industrial firewall panel application.

Introduction

The introduction of multilayered trimmed panels within vehicle cabins have largely contributed to increase passenger's comfort because of their capability in reducing noise coming from various sources including the engine compartments, the road as well as wind noise [1]. Acoustic trim components in trim package are usually a combination of porous and heavy layers that aim to maximize the effects of absorption, insulation and damping [2]. However, development of these trim packages tends to be quite cumbersome due to the constraints on these components, such as weight, packaging space and the amount of possible material and geometrical combinations. An experiment-based approach, where the product is designed, prototyped and then tested can result in increased amounts of time per design iteration resulting in high costs due to prototyping and testing. In order to reduce this cost, an approach that relies on numerical acoustic simulation has been adopted by original equipment manufacturers (OEMs) and their suppliers, which can be used to rapidly evaluate noise transmission given the properties of the trim component [2]. While the simulation techniques can help reduce the time for each design iteration as well as provide additional insight on local effects [3], the computational costs can still be high when a large number of configurations need to be performed, either for design exploration or optimization.

Another aspect of consideration is the complexity of the supply chain of automotive components. The development of many components is outsourced to suppliers and, in certain cases, suppliers also subcontract parts of their products further down the supply chain. While this approach has accelerated development cycles, it is certainly not frictionless and could be further improved. Often, the requirements, including the necessary indicators to follow as well as the constraints to respect, are passed down the chain and suppliers design components based on their experience. Such an experience is usually available in technical reports, documents, best practices and guidelines (besides living within the brains of engineers), and, while it remains available, does not reach its full potential resulting in situations where the wheel is reinvented. A more robust approach would be to distill this experience (or certain aspects of it) within models that can be easily queried, and which can provide instantaneous answers of the product behavior subjected to certain conditions. These models could additionally be provided further up or down the chain in order to accelerate development of the components and their respective systems.

Model reduction techniques can be utilized for this purpose as they can efficiently model a large variety of systems and can evaluate the performance of previously unseen configurations in real-time [4]. In this article, model reduction techniques are used along with numerical acoustic simulation to construct reduced models with parameters on a vast, multi-dimensional design space for the evaluation of the sound transmission loss of 7 different multilayered trimmed firewall panels. Moreover, further reduced models for one of the panel geometries are created on smaller design spaces aiming to recreate variations in specific materials for the porous layer of the trimmed panel. The performance of the reduced models is evaluated with respect to the simulation results, which take the role of the ground truth.

Methodology

In this section, the methodology used for both data generation and model reduction will be described. First, the process for generating the necessary data will be presented, followed by the reduction techniques and finally the input and evaluation parameters.

Acoustic Simulation

A typical transmission loss evaluation setup is depicted in Figure 1. It typically consists of an upstream reverberant room where a diffuse sound field can be created by an acoustic source, a reverberant or anechoic receiving room where an array of microphones is located, and the panel under evaluation is placed between these rooms such that it separates them [5].

The equivalent vibroacoustic numerical model of the setup shown in Figure 1 is shown in Figure 2.

This numerical model consists of the firewall panel composed of a steel layer, a heavy layer and a porous layer. Besides, an acoustic propagation volume on the receiving side and a non-reflecting boundary condition are defined with a numerical diffuse sound field condition imposed as excitation. The system is solved in the physical coordinates in the frequency domain. The steel layer is modeled as a modal component requiring the normal modes of the layer to

FIGURE 1 Typical transmission loss evaluation setup for multilayered trimmed panels

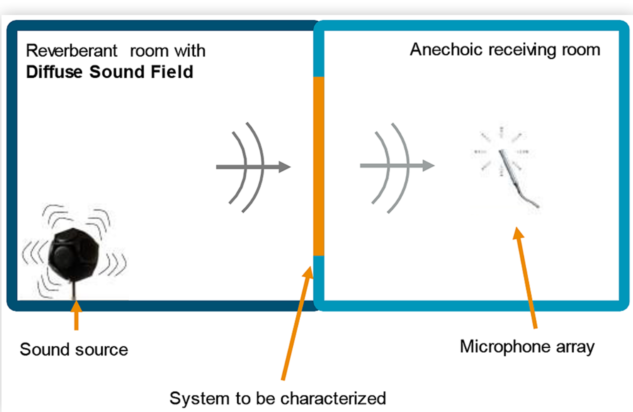
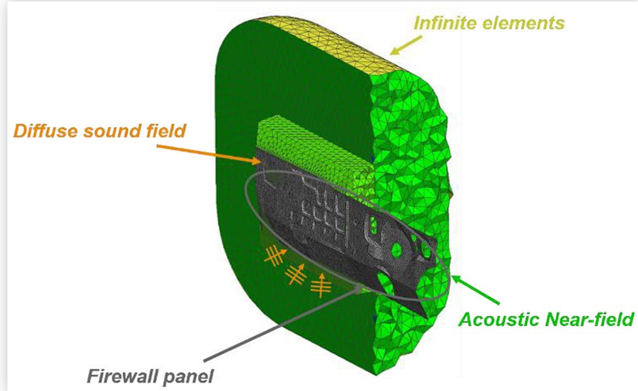


FIGURE 2 Equivalent acoustic model for the evaluation of transmission loss of a multilayered trimmed panel.

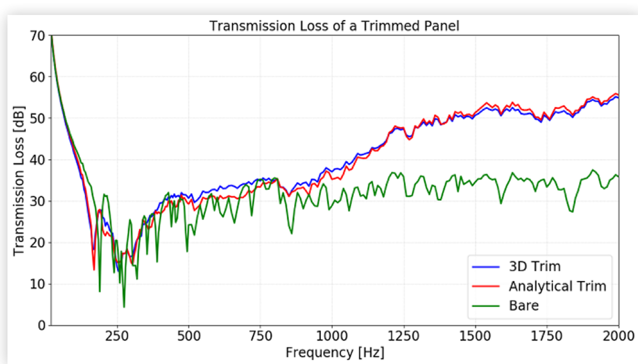


be extracted in advance in order to speed up the whole model calculation. The heavy layer and porous layer are modeled via a locally reacting transfer admittance method, allowing for a meshless definition of the multilayered trim leading to a reduction of computational time by a factor of 3 [6] with respect to the fully meshed trim while at the same time providing an accurate representation. In Figure 3, a comparison is provided between the meshless approach ("Analytical Trim"), the meshed approach ("3D Trim"). The "bare" configuration (without any trims) has been added as a reference.

The three-dimensional acoustic volume corresponding to the receiving room is enclosed by a non-reflecting boundary condition emulating the anechoic property of the receiving room. The anechoic condition is modeled via an infinite element approach [7]. Finally, a numerical diffuse sound field condition [8] is applied directly on the panel to model the diffuse sound field created by the reverberant room. The model is solved for frequencies between 100 Hz and 1250 Hz in the third octave band using the commercial software suite Actran [9].

A benefit of the acoustic simulation is that it does not require complex post-processing of the microphone information to calculate the transmission loss. Instead, at each solution frequency, it is possible to recover the incident power

FIGURE 3 Comparison between the meshless ("Analytical Trim") and meshed approach ("3D Trim") for the transmission loss of trimmed panels.



of the diffuse sound field in the emitting side, as well as the transmitted (or radiated) power on the receiving side. The transmission loss is then calculated using equation 1.

$$TL(dB) = 10log\frac{Incident\ Power}{Transmitted\ Power} \tag{1}$$

A number of different geometries are evaluated for the purposes of this study in order to assess the sensitivity of the reduction method across different models and transmission loss characteristics. The 7 different geometries are shown in Figure 4. The number shown in the figure will be further used to identify each of the geometries.

All calculations are carried out on 4 parallel processes (Intel(R) Xeon(R) CPU E5-2697 v4 @ 2.30GHz). The computation time per frequency is around 32 s and the maximum consumed RAM per process reaches 4 GB leading to a total computational cost around ~4 CPU-hours for the total 241 frequencies (20 frequencies per third octave band).

An example design point for all geometries is shown in Figure 5. All geometries produce results that are roughly on the same level with the same geometric and material parameters. Furthermore, in the frequency range of 160-300 Hz, in the resonance-dominated region of the panels, there are a lot of peaks and troughs which the reduced model will have difficulty capturing as the function becomes quite discontinuous.

As the transmission loss results tend to be "shaky" in the narrow-band, this will be exported in three formats, narrow-band data, third octave band data and octave band data. Third octave and octave band data are commonly used as indicators by OEMs and suppliers when evaluating the transmission loss. Figure 6 shows the third octave band results of the same design point as the one depicted in Figure 5.

The third-octave results are obtained by integrating the narrow-band results over the third-octave bands. The same applies for the octave band results that will be used later.

FIGURE 4 Firewall panel geometries used for the evaluation of transmission loss

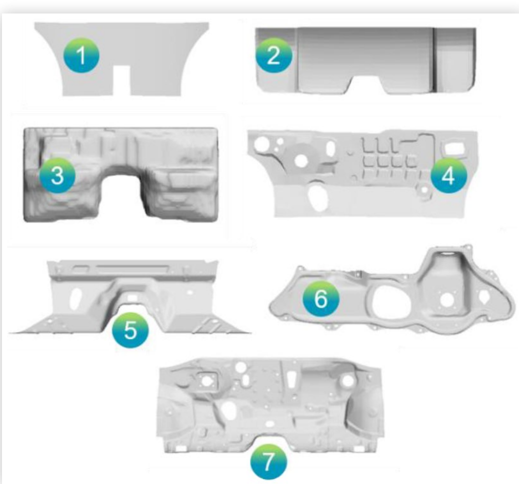
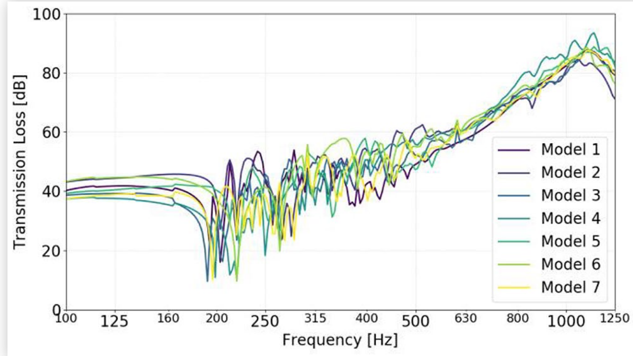


FIGURE 5 Transmission loss curves for all the studied geometries in logarithmic scale.



Reduction Technique

The model reduction technique used in this study has been detailed in reference [4]. A proper orthogonal decomposition (POD) method with an adaptive radial basis function (RBF) kernel is used for decomposing the problem to its proper orthogonal modes and for reconstructing the solution for previously unseen design variable sets. The model reduction and reconstruction results are obtained by the ODYSSEE software package [10]. Moreover, the time required for obtaining results on previously unseen configurations is very small compared to a full acoustic simulation, ranging in the order of a few seconds. Such a numerical performance allows for fast responses to early design queries.

The reduction technique will be applied to all three available data formats (narrow-band, third octave band and octave band) independently. This will allow to consider a trade-off between accuracy and the necessary level of detail in the output of the reduced model.

Design Space

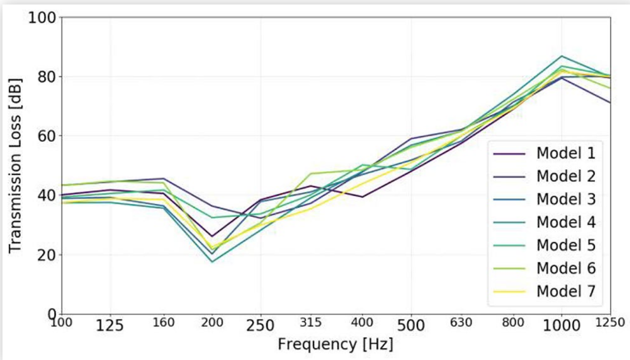
Geometric and material parameters of the trimmed panel (comprised of the heavy and porous layer) will be considered as the design variables. A total of 11 parameters will be used including the thickness of the two layers, the elastic properties (Young's modulus, density and Poisson's ratio) as well as poro-elastic properties of the porous layer, such as the porosity, tortuosity and flow resistivity.

For the generation of the data serving to train the model, the design space defined by geometrical and material parameters is populated by a Latin hypercube sampling technique. This allows to fill the design space in a more intelligent way thus reducing the overall number of required simulations for the data generation process.

The root mean square error (RMSE) is used as the reference metric for comparing with the ground truth, as defined in equation 2.

$$RMSE = \sqrt{\frac{\sum_i^n \left(y_i - \hat{y}_i\right)^2}{n}} \tag{2}$$

FIGURE 6 Transmission loss curves in third octave band for all the studied geometries in logarithmic scale.



Complete Design Space Evaluation In the first part of this study, the complete design space defined by wide limits in the design variable definition will be considered, covering a large range of materials, some of which may not be feasible. This is done in order to assess the performance of the model in larger, less-constraint design space, one, however, that may require a larger number of training samples to properly represent. 150 training samples will be created for each panel geometry, for a total of 1050 samples overall. Each sample of this dataset contains two parts: the set of parameters that will be used as input, and the transmission loss curves as output. Each of the 150-sample datasets for each panel will be split into two parts: A training set that will comprise of 80% of the samples, randomly allocated and a validation set of the remaining 20% of the samples. The training set is used to train the algorithm and fit the reduced model so that it can make predictions based on this information. The validation set is used to evaluate the reduced model's accuracy. The reduced model is given the samples of the validation set (which the model has not seen before) and the predictions are compared with a ground truth. As mentioned before, the simulation results take the role of the ground truth.

Furthermore, a 5-fold cross-validation study will be performed to evaluate the sensitivity of the model on the training data. For the 5-fold cross-validation study, the 150-sample dataset will be split randomly into 5 non-overlapping, randomly allocated validation sets (folds) comprising of 20% of the samples of the full dataset. This way, every sample of the dataset will be considered as unseen once. The remaining samples for each group will be used as the training sets. The cross-validation study will allow us to understand if there is large variability in the predictions made by the model when the data is different as well as to have a better view of the performance of the model.

Table 1 shows the parameters of the first part of the study as well as their range. Overall, a large range of values is considered for each variable as no specific material is targeted for the first stage of this study. The units of the parameters are in SI.

Specific Material Design Space In the second part, the focus is on representing individual materials, which results in a smaller design space. Five different reference materials are created based on common properties of cellulose [11], felt [12], melamine [13], polyimide [14] and polyurethane [15] foams. To reduce the amount of simulations, only the 7th geometry (as shown in Figure 4) will be used. 50 training samples for each of the materials, resulting in 250 new samples overall.

Table 2 shows the reference material properties. As porous materials tend to have very different properties, the range of values for each material property for constructing the design space will be ± 10% while still restricting the quantities to physical values, e.g. the porosity is not allowed to exceed a value of 1. The thickness of the trim will be varied 50% around the reference value.

TABLE 1 Geometric and material parameters of the design space along with their limits. The elastic properties of the porous material refer to the properties of the skeleton (solid phase) of the material.

Quantity	Min	Max
Heavy Layer thickness	0.001	0.006
Heavy Layer density	10000	20000
Heavy Layer Young's	7e7	1.3e8
Heavy Layer Poisson's	0.3	0.4
Porous thickness	0.01	0.08
Porous density	395	2160
Porous Young's	2600	214000
Porous porosity	0.922	0.98
Porous tortuosity	1	1.88
Porous Flow resistivity	2000	135000
Porous Poisson's	0.2	0.4

TABLE 2 Geometric and material parameters of the reference foam materials. The elastic properties refer to the properties of the skeleton (solid phase) of the material.

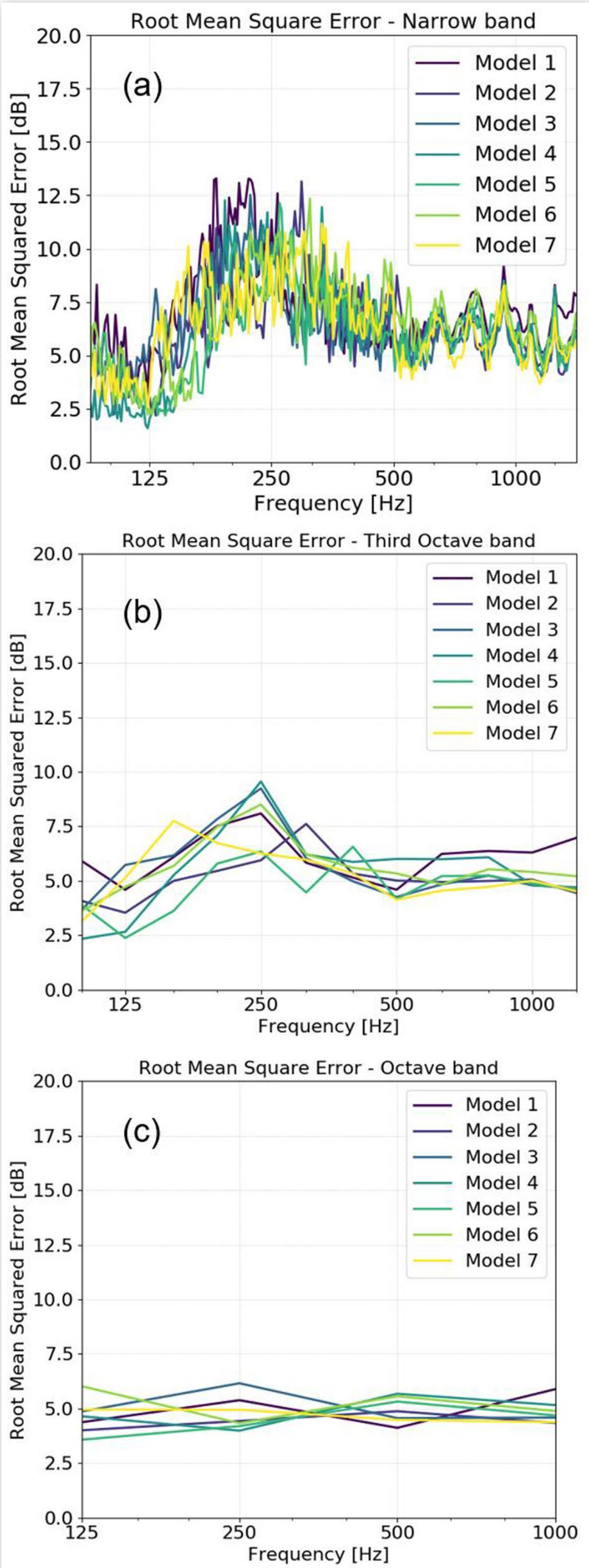
Quantity	Cellulose	Felt	Mel	PI	PU
Thickness	0.04	0.04	0.04	0.04	0.04
Density	2000	1500	750	1400	1500
Young's	10000	35000	120000	800000	100000
Porosity	0.985	0.95	0.98	0.98	0.95
Tortuosity	1.1	1.15	1.1	1.2	1.6
Flow Resistivity	6000	50000	10000	200000	45000
Poisson's ratio	0.1	0.11	0.1	0.23	0.23

Results

Complete Design Space Evaluation

The root mean square error for the validation all models across the frequency spectrum and for three different band definitions are shown in Figure 7. The RMSE for the narrow band results is quite "shaky" over the full frequency spectrum, with

FIGURE 7 Root mean square error for all models for (a) narrow band results, (b) third octave band results, (c) octave band results.



a mean value of around 6 dB. Especially, within the resonance region, the RSME is quite high with respect to other regions. After around 500 Hz, within the stiffness-dominated region, it becomes steadier with some intermittent peaks, showcasing the difficulty that the reduced model has in capturing specific regions of the spectrum. The results for the third octave band and octave band behave better in contrast to the narrow band. In general, all models show similar performance.

The comparison between the best and worst performing samples in the validation set can be performed in order to visualize the sensitivity and provide some insight into the differences within the various regions of the frequency spectrum. In Figure 8, the best and worst samples for the 6th geometry are shown. The true values correspond to the transmission loss evaluated by the acoustic simulation while the predicted values are produced by the reduced model. For the best sample, there are very few discrepancies which start mostly after the resonance region and remain small while the overall trend is followed. On the other hand, for the worst sample, there are large differences across the frequency spectrum. It is likely that these differences will be reduced if more samples are added to the design space in order to cover it in a better way [16].

Finally, in Figure 9, the average root mean square error as well as the envelope can be found for the 5 folds of the cross validation. Overall, the variability of the RMSE with respect to the fold is around 3-4 dB, with certain frequency ranges impacted more than others. Specifically, for the third octave band, high variability is present at 250 and 315 Hz but otherwise the variability is around 1-3 dB, showing little sensitivity to the folds.

FIGURE 8 Comparison between the predicted transmission loss and the true one for Model 6 for the (a) best sample and (b) worst sample.

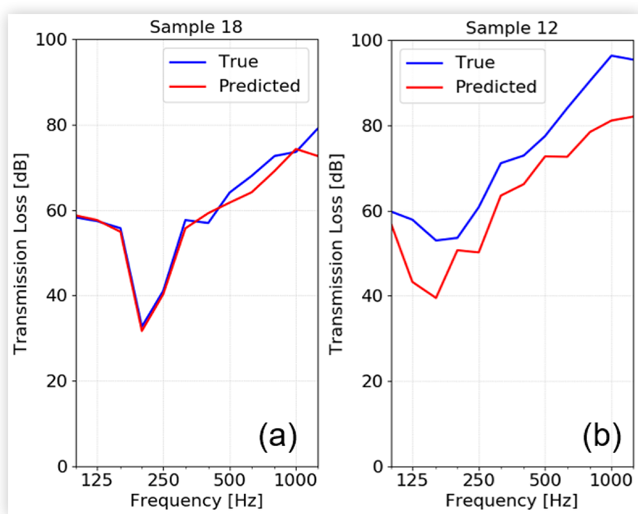
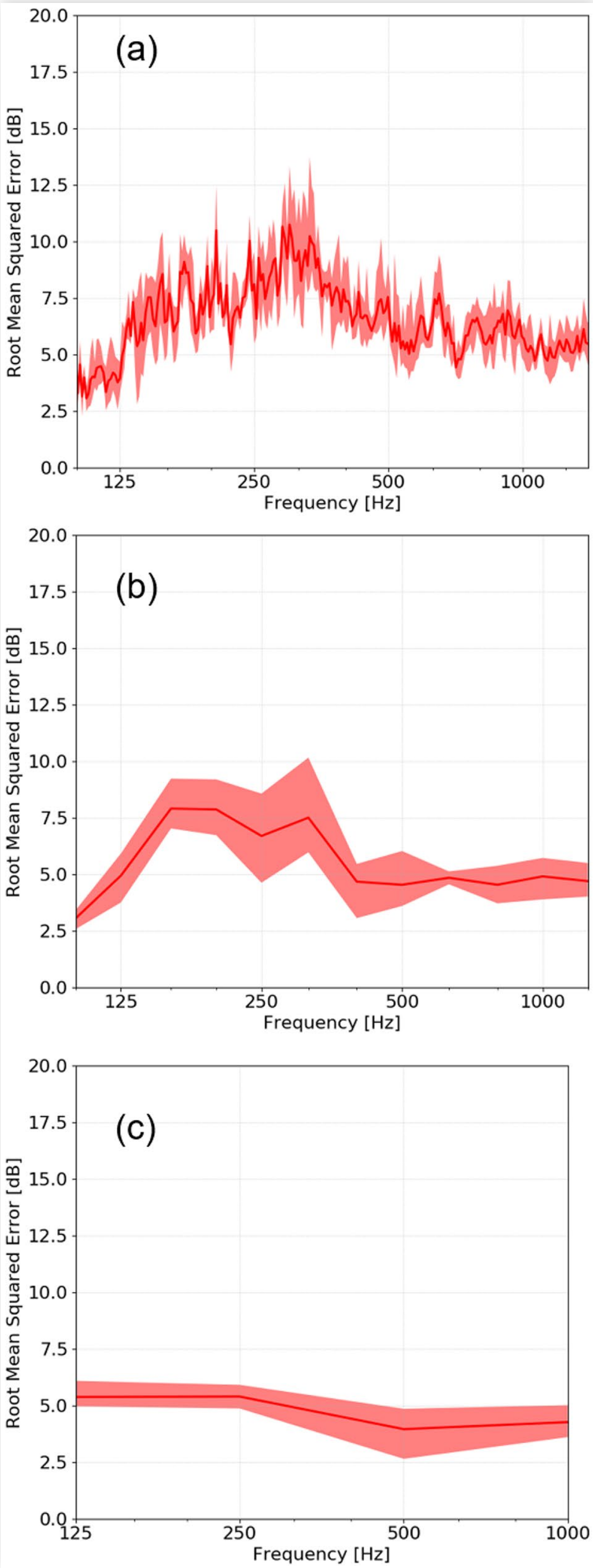


FIGURE 9 Average root mean square error and envelope for 5-fold cross validation for Model 7. (a) Narrow band results, (b) third octave band results, (c) octave band results.



Specific Material Design Space

The results with respect to specific foam materials are presented in this section. Figure 10 shows the root mean square error for all materials.

While the narrow band results show still a large amount of error in prediction, especially for polyurethane form, this error is not translated to the third octave or the octave band results. In third octave band, all materials show a mean RMSE of less than 1 dB except for the cellulose and polyimide foam. For the polyimide foam, the issue seems to be between 500 and 1000 Hz while is more steadily high between 250 and 1000 Hz with a mean of around 2 dB. For octave band results, for all but the cellulose foam, the reduced model produces results within 1 dB of the ground truth.

A further look at some of the individual samples can provide insight on the difference.

From the plots in Figure 11, it is clear that differences are due to relatively bad prediction at specific frequencies, while by looking at the RMSE in Figure 10, predictions tend to be extremely good overall.

Summary

The development of reduced order models for predicting transmission loss of firewall panels is carried out via decomposition, reduction and reconstruction techniques. A set of different multilayered trimmed firewall panels were used where the design space discretization was obtained via a Latin hypercube sampling technique. Geometrical and material parameters were modified for the heavy and porous layer of the trim. Numerical acoustic simulation was used to generate the training and testing data, to be later fed to a reduction algorithm. Once the model was trained, predictions on previously unseen configurations are performed and an error is calculated based on the difference between the predicted and the true (simulated) value. For the larger design space with all models, while accuracy is compromised due to the "shaky" nature of the transmission loss curves in narrow band, in third octave band (which is a better indicator in an industrial context) the prediction is better and much more consistent. The 5-fold cross validation study highlighted the variability of results; however, this variability is limited indicating that prediction is not biased by the training/test split. Moreover, the large error could be an indication that the design space is not covered well enough and the model could be reduced with the addition of more samples. For the second part, a new design space was created based on the properties of common materials and new reduced models were created based on a new dataset. The new reduced models provided very good accuracy on the third octave and octave bands, but the narrow band prediction was still impeded by the resonance region of the transmission loss.

FIGURE 10 Root mean square error for all materials and Model 7 for (a) narrow band results, (b) third octave band results, (c) octave band results.

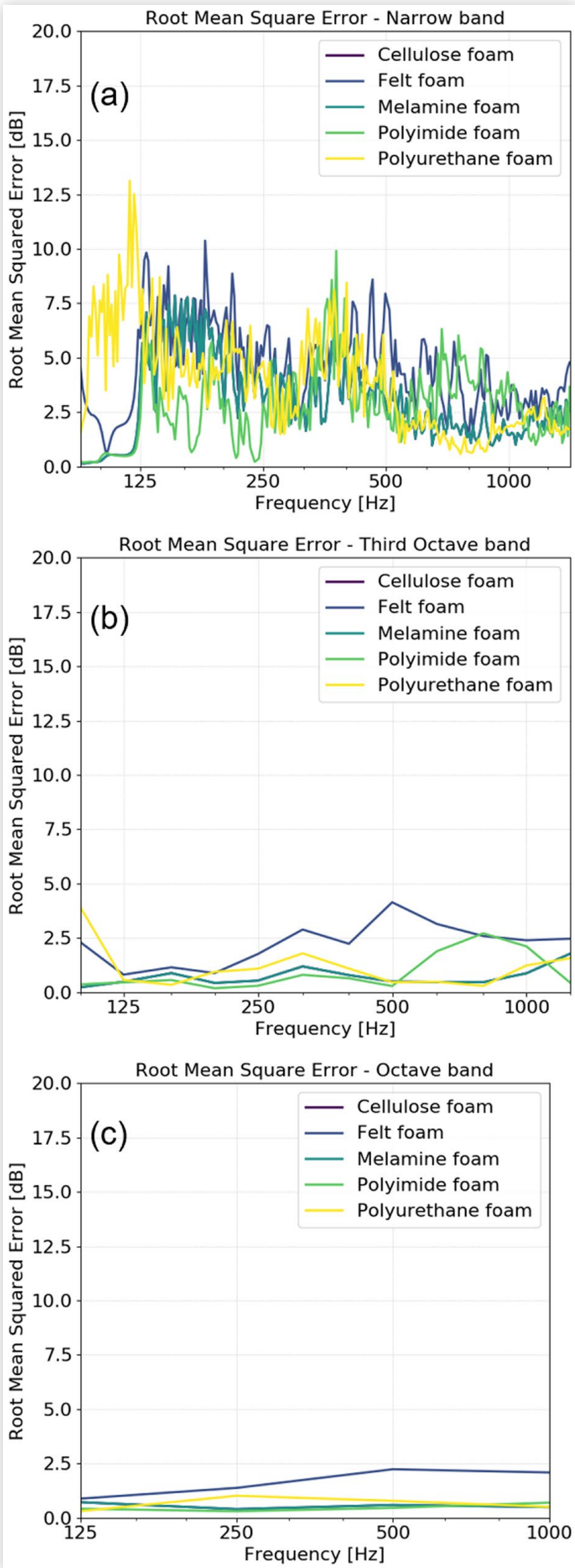
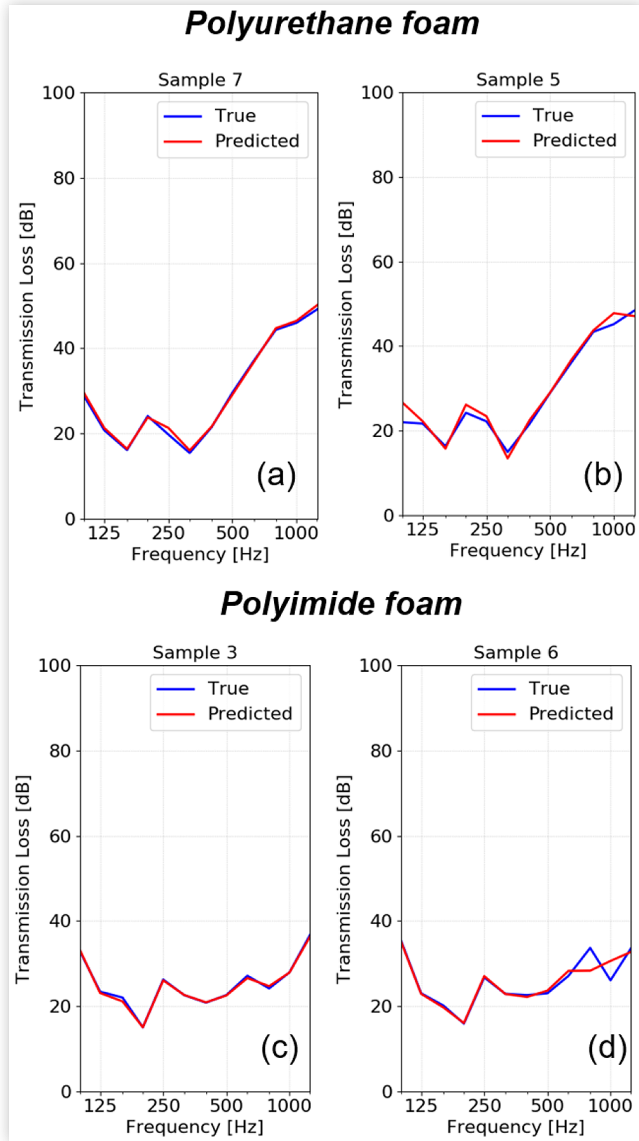


FIGURE 11 Comparison between the predicted transmission loss and the true one polyurethane and polyimide foam in third octave band: (a) best sample for polyurethane foam and (b) worst sample for polyurethane foam, (c) best sample for polyimide foam and (d) worst sample for polyimide foam



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Definitions/Abbreviations

RMSE - Root mean square error

TL - Transmission loss

POD - Proper orthogonal decomposition

RBF - Radial basis function

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