A critical analysis of novel acoustic metamaterials applied to automotive silencers.

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Abstract

This paper examines and compares two acoustic metamaterial designs, a microslit tube absorber outer-shell accompanied with a labyrinthine inner core and a narrow axisymmetric conical tube chamber with a poroelastic lamellar structure. The microslit labyrinthine metamaterial manages to reach transmission loss peaks of approximately 50dB for narrow frequency bands. Incredible transmission loss between 12-20dB in the sub 500Hz frequency band is witnessed by the poroelastic lamellar. The poroelastic lamellar metamaterial outperforms the microslit labyrinthine metamaterial in the majority of the octave frequency bands examined. The utility of the U-Net architecture and a generative adversarial network as an optimisation strategy is outlined to further improve such unit cell configurations.

Keywords: Automotive silencers, acoustic metamaterials, microslit absorber, labyrinthine space coiling, poroelastic lamellar, acoustic black hole.

1. Introduction

In 2000, Liu et al. [1] proposed a new structural idea based on new physical concepts. This idea broke the law of mass density that restricts the development of noise reduction materials, and achieves a significant reduction in sound transmission without changing the thickness of the material. This prominent new structure is an acoustic metamaterial, also known as a locally resonant acoustic wave crystal. Acoustic metamaterials create the possibility of opening the acoustic band gap in the audio range using a smaller size system, greatly increasing the sound transmission loss. Acoustic metamaterials can control sound waves in ways that traditional materials cannot. With its zero or even negative refractive index, negative bulk modulus and, negative mass density, it provides new directions and ideas for sound wave control [1].

1.1. Physical basis of the novel metamaterials

Group 5's metamaterial is a porous elastic layered network made of melamine foam. The designed metamaterial has a porous surface, which can produce visco-thermoacoustic loss and provide broadband attenuation. The tapered tubular structure can be expected to attenuate sound waves while allowing exhaust gas to flow due to the porous, slotted inner surface. The interstitial internal structure of the cone is expected to reduce sound appropriately. The vertical direction of the material gap provides better noise reduction performance in certain frequency ranges especially in the range between 450 Hz and 1000 Hz where transmission loss is prevalent. Therefore, the metamaterial structure is vertically oriented from the exhaust flow, designed to conform to the geometry and adaptability of the given structure [2].

(a) Lamellar poroelastic metamaterial. (b) Space-coiling microslit metamaterial.

Figure 1: Novel acoustic metamaterial CAD geometries for comparison.

Group 8 adopted a micro-slit absorber design. The

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micro-slit design uses slits, and the absorber contains three slits and two internal channels formed by micro-slit tubes. This unit cell configuration is arranged in a Poisson geometric pattern to enhance the sound attenuation of the silencer. The proposed micro-slit absorber was expected to have an absorption coefficient value for ranges from 260 up to 700 Hz with an almost Gaussian distribution. The spiral-labyrinthine acoustic metamaterial was intended to attenuate sound within the frequency ranges of 1000 to 2200 Hz.

1.2. Defining the acoustic performance of the metamaterials

To measure the performance of an acoustic metamaterial or automotive silencer we can define it's sound transmission coefficient as being the ratio of sound power incident on the material to sound power transmitted (denoted by τ). A power spectrum analysis of the sound energy lost can be performed by taking the reciprocal of the transmission coefficient and taking the log base 10 as seen in Equation 1, this is the transmission loss [3].

$$TL = 10\log_{10}\left(\frac{W_i}{W_t}\right) = 10\log_{10}\left(\frac{1}{\tau}\right) \quad (1)$$

A two-port sound absorbing device such as an automotive silencer can be characterised by a four-pole transfer matrix T using a lumped parameter model. This equation with four unknowns can be constrained with experimentally measured values of sound pressure (p) and velocity (u) measured on either side of the acoustic material for two different loading conditions (a and b) which will be discussed further in Section 3.

$$\begin{Bmatrix} p_{a,b} \\ u_{a,b} \end{Bmatrix}_{x=0} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{Bmatrix} p_{a,b} \\ u_{a,b} \end{Bmatrix}_{x=d} = T \begin{Bmatrix} p_{a,b} \\ u_{a,b} \end{Bmatrix}_{x=d} \quad (2)$$

2. Optimisation Strategy

To optimize the configurations of the introduced metamaterial unit cells so that sound attenuation can properly be achieved within the necessitated frequency ranges, a data driven topology optimization, i.e., a deep learning model is proposed.

Typically, numerical analyses can prove to consume an abundance of time and be computationally expensive as a result to necessitating many iterations [4]. Especially, due to the complex structures of metamaterials, the design process is hampered as a consequence to the constant requirement of conducting complex numerical calculations when attempting to tune the geometric structure and material properties of the metamaterial with traditional optimisation techniques.

Deep learning models offer instantaneous results for conducting such tedious calculations. Deep learning is a branch of machine learning which attempts to behave like that of the human brain in which it integrates a 'neural network' which is trained and tested to eventually be capable of making accurate conclusions on its own. While deep learning has recently proven to be a powerful tool for applications such as image processing and speech [5], its applicability extends to being utilised in design topology optimization [6]. In recent years, deep learning has been implemented as an optimization method for acoustic metamaterial structures [4] and has been recognized to substantially enhance the effective proprieties by its ability to effectively learn the innate relationship between specific structural parameters and the associated theoretical sound transmission loss.

The output y of the jth node of a layer of in neural network is as follows:

$$y_j = \frac{\sum_{i=1}^{n} f(w_i x_i + b_j)}{n} \quad (3)$$

where x_i, $f(z)$, b_j and n denote the dataset input, the activation function, the bias and the amount of neurons in the layer respectively displayed in Figure 2.

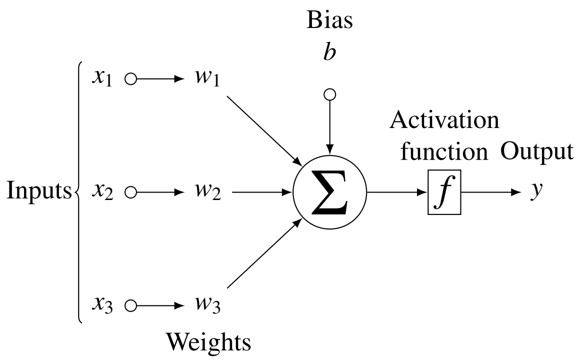


Figure 2: Forward pass in a neural network.

Implementing a convolutional neural network (CNN) model such as the U-Net architecture displayed in Figure 3 permits the utilization of semantic segmentation to determine the appropriate geometric structure to enhance the acoustic metamaterial silencer based off pre-existing data [4]. The proposed CNN architecture employs an encoder which comprises of covenant and pooling layers for feature extraction and a decoder which comprises of transposed convolution to allow for localization [5].

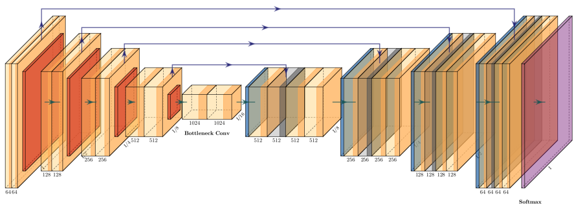


Figure 3: U-net architecture.

A generative adversarial network (GAN) may also be employed into the model as it possesses excellent capabilities of image generation to output an optimal acoustic metamaterial design [7].

Employing the rectified linear unit (ReLU), a non-linear activation function as seen in Equation 4, enhances the model to be more computationally efficient as it only activates neurons which are positive. The implementation and variation of hyper parameters such as batch size, dropout, the number of workers, and a loss function at each step allow for improved model accuracy on training and identifying new data. Furthermore, back propagation is employed to propagate the total loss back into the neural network to subsequently re-valuate the weights developing the overall flexibility of the model.

$$ReLU(x) = max(0, x) \tag{4}$$

The required input design parameters x_i for this model would consist of the 3D geometric structure of the proposed unit cells such as the exterior radius of the micro-slit tube absorber shell and internal dimensions of the labyrinthine core of Group 8's metamaterial configuration. In the case of optimizing Group 5's metamerial configuration, the radius of the conical tube section and the spacing of quasi-periodic ribbed internal structure are defined as the input design parameters. The model is fed images of the various ranges of sizes for the proposed metamaterials, all of which are labelled accordingly to their associated acoustic performance and respective material properties.

The model's goal is to optimize the acoustic performance of the silencer, specifically in the proposed frequency range; thus it is trained and tested to maximize the bulks modulus and shear modulus and minimize the Poisson's ratio. Alongside the acoustic performance, the model must be constrained to minimize monetary cost, weight and take into account manufacturing considerations for the design. The model then generates an optimized design which fulfills the design criteria and adheres to the constraints.

3. Experimental Methodology

To constrain the matrix in Equation 2 values for sound pressure and velocity must be experimentally using an impedance tube method as per ASTM E2611-09 guidelines [8].

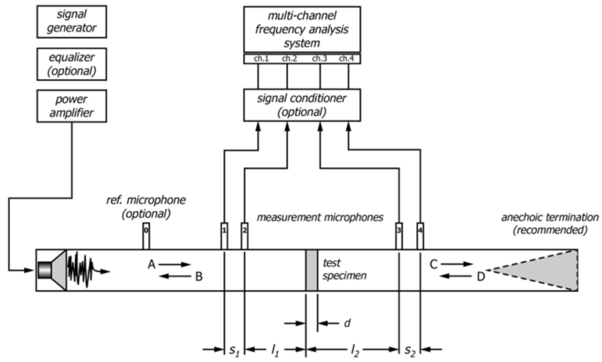


Figure 4: Experimental setup for determination of transfer matrix conducted to ASTM E2611-09 guidelines [8].

A uniform power response sound source with an operating frequency over the range of interest was used to supply the incident sound energy upon the acoustic metamaterial. For load condition *a* a weakly anechoic wedge shaped termination was used. The sound pressure and velocity was measured using four microphones at two locations on each side of the metamaterial. The second load case *b* used either an open or closed termination but the exact same quantities were measured. This can be used in Equation 2 to obtain the four poles of the transfer matrix. The transmission loss can be obtained by substituting these values into Equation 5.

$$TL = 10\log_{10}\left(\frac{1}{4}\left|T_{11} + T_{12}\frac{S}{\rho c} + T_{21}\frac{\rho c}{S}T_{22}\right|^2\right) \tag{5}$$

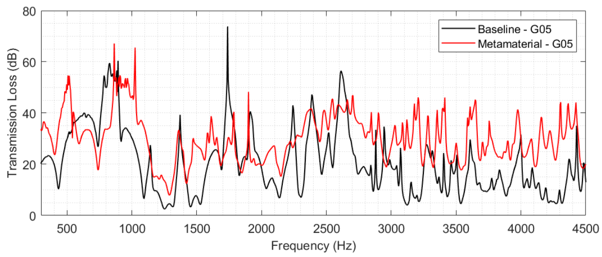
Where S is the area of the duct, ρ is the density of the medium and, c is the speed of sound in the medium. This method was modified to accommodate a large test rig and is derived from the methods used in Deery et al. [9].

This silencers were manufactured using the Original Prusa Mini+ FDM printer. It has a 18x18x18 cm build frame and has a 0.25mm nozzle for high accuracy and a temperature independent probe that doesn't suffer from temperature drift. fused deposition modelling (FDM) was the chosen method of 3-D printing due to its low cost, availability and, relatively high speed.

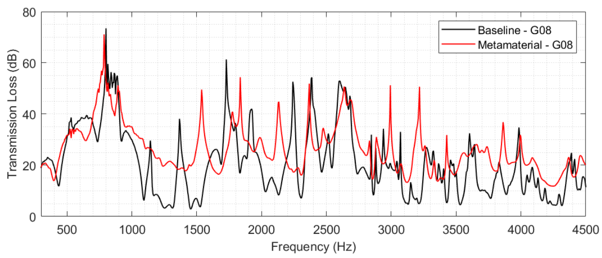
4. Results and Discussion

4.1. Performance compared to baseline

The poroelastic lamellar metamaterials performance as seen in Figure 5(a) shows good broadband performance in the sub 500Hz range and above 2700Hz. In the 900-1700Hz a marked improvement in transmission loss compared to the baseline is also seen. As identified by the group, the perpendicular orientation of the metamaterial structure to the exhaust flow hampers it's attenuating abilities [2] at mid-range frequencies but shows better performance at sub 450Hz and above 1000Hz, this is reflected in the transmission loss data recorded.



(a) Group 5 baseline and metamaterial transmission loss comparison.



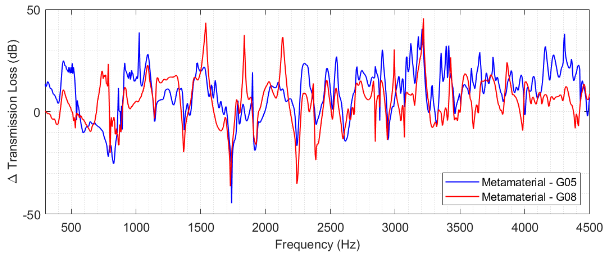
(b) Group 8 baseline and metamaterial transmission loss comparison.

Figure 5: Meta-material transmission loss compared to respective baseline.

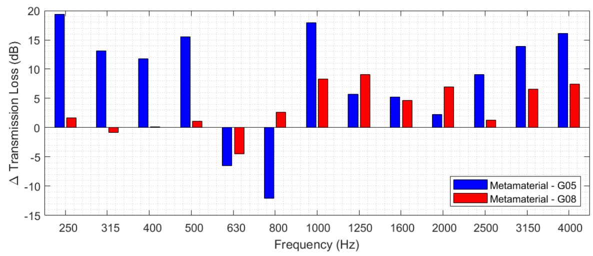
For the microslit labyrinthine metamaterial as seen in Figure 5(b) less of an improvement across the selected frequency spectrum is seen. While it does reach transmission loss peaks of around 50dB these are for narrow frequency bands. Some improvement in transmission loss can be seen in the 900-1600Hz frequency range.

4.2. Direct comparison between metamaterials

Plotting the change in transmission loss provided by each metamaterial compared to their respective baseline provides a direct comparison between the two technologies. Over a continuous frequency band as seen in Figure 6(a) the difference is less clear cut though it can be seen that Group 5's poroeleastic lamellar metamaterial is providing better sound attenuation.



(a) Change in transmission loss (continuous frequency bands).



(b) Change in transmission loss (octave frequency bands).

Figure 6: Change in transmission loss from respective baseline.

Plotting over discrete octave frequency bands (Figure 6(b)) clearly shows that Group 5's poroelsatic metamaterial outperforms Group 8's microslit labyrinthine metamaterial across all but two of the octave frequency bands, the 630Hz and 800Hz bands. Group 5's metamaterial shows great transmission loss improvement between 12-20dB in the challenging sub 500Hz frequency band which shows great promise for this metamaterial. Both metamaterials struggle in the 630Hz and 800Hz bands but again for the poroelastic lamellar metamaterial this was noted by the researchers due to the perpendicular orientation of the metamaterial structure.

From Group 8's research proposal the design did not perhaps have it's intended effect and a number of factors from the original paper [3] could be considered for tuning of the design:

- Varying the angle of the slot gap will have an effect on device performance.
- The gap size of the microslit design was not scaled finely and his a key design parameter as it must be on the same scale as the viscous boundary layer of the flow incident on the metamaterial.

- Wall thickness and the absorptive qualities of the material chosen should be taken into account, both the original microslit design and Group 8's design use ABS, however some additional stiffness has more than likely been added due to the internal structure of the labyrinthine metamaterial combination resulting in a change of the metamaterials' natural frequency of vibration.

Clearly in their current forms without tuning the lamellar poroelastic metamaterial from Group 5 is the best choice of the two for implementation in an automotive silencer.

4.3. *Comparison to current research*

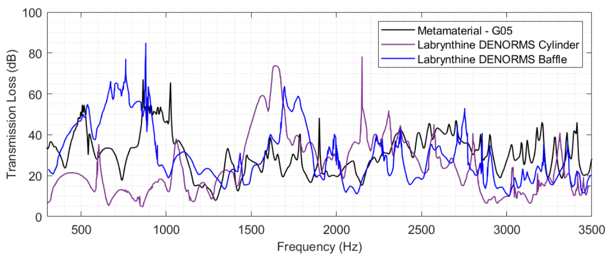


Figure 7: Comparison of lamellar poroelastic metamaterial to cutting edge DENORMS automotive silencer design [9].

Acoustic metamaterials applied to automotive silencers is a relatively unexplored field ripe for innovation. Recently a paper studying the effect of introducing a metamaterial structure, based on research by Design for Noise Reducing Materials and Structures, into acoustic baffles showed great promise in reducing the noise produced by automotive exhaust systems [9].

Data from this study was extracted for both a baffle and cylinder silencer design, both common geometrical arrangements for automotive applications, and plotted their transmission loss data against that of the very successful lamellar poroelastic metamaterial presented in our paper. It should be noted that the frequency range was restricted to 3500Hz in this case and as such data for the lamellar poroelastic metamaterial was only considered in this range.

As seen in Figure 7 the lamellar poroelastic metamaterial performs strongly at 900-1050Hz and shows some potential improvement above roughly 2500Hz. However the DENORMS cylinder metamaterial clearly outperforms it in the 1450-1600Hz and 2050-2300Hz ranges reaching up to 70dB of transmission loss. The DENORMS acoustic baffles outperform it in the sub 900Hz frequencies with an average transmission loss of 60dB.

5. Conclusions

This paper has compared experimental results of two novel metamaterials designed for use in an automotive acoustic silencer. The underlying physics of the two devices was outlined and an optimisation strategy using deep learning techniques for tuning the acoustic metamaterials' parameters was put forth. It was seen from comparison of experimental results that the metamaterial with a lamellar poroelastic structure outperformed a microslit labyrinthine combination in terms of acoustic attenuation in a frequency range of 0-4500Hz. This particular metamaterial was compared against current state-of-the-art research and was found to under perform this DENORMS metamaterial however further research using the deep learning methods to tune both of the metamaterials in this paper could enhance their attenuation capabilities.

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