Design of an Innovative Acoustic Metamaterial

PAVLOS MAVROMATIDIS[a], ANDREAS KANARACHOS[b]
Electrical Engineering Department[a], Mechanical Engineering Department[b]
Frederick University
7 Y. Frederickou St.
CYPRUS
eng.pam@frederick.ac.cy, eng.ka@frederick.ac.cy http://www.frederick.ac.cy

Abstract: - This report represents the mass dampers based acoustic resonators, inclusion in the light weight structures. Especially 3d printed structures, which are light in weight and thus have poor acoustic properties. With the inclusion of acoustic resonators, the sound vibrations can be attenuated and thus good acoustic properties can be achieved in the light weight structures. Major emphasises is given on the concepts, methods of designing an innovative acoustic resonators in a CAD modeller, their FEA on a FEA solver, their 3d printing manufacturing process and then the experimental modal testing of the structure. In short their correlation to the real structures. CAD modelling of the new light weight plate structure hosting innovative acoustic resonators is done in CATIA V5 6R-2015. It was then analysed in Workbench ANSYS 16.1 for the modal analysis. 3d manufacturing of the part was done in 3d printer (Ultimaker 2) based on Fused Deposition Method (FDM). Material used for manufacturing the part is PLA. Experimental modal testing was done in order to validate the model. The resonance frequency achieved for mode 9 is 1555.4 Hz from the experimental modal testing, Lab test LMS Xpress software. Where the FEA gave 1575.4 Hz resonance frequency for mode 9 thus quite correspondence was seen between the results. Hence a reasonable validation is done of the designed model to real structure. Conclusively the innovative acoustic resonators shows the potential of reducing the noise produced in 3d printed light weight parts. Future recommendations are given, based on the knowledge and skilled learned for reducing the resonance frequency.

Key-Words: - acoustic metamaterial, finite element, modal analysis

1 Introduction

With the advent of additive manufacturing, a 3-D printing structures with good mechanical properties light weight has become possible. Honeycomb structure is the most popular type of it, but one of the problem being associated with these structures are the acoustics, the inability to reduce sound being produced and radiated. To control this drawback and by looking at the compatibility of honeycomb structure with the sound absorbers. A metamaterial based acoustic resonators are installed in the honeycomb panel which acts as a sound absorber for a certain frequency range (Soize & Ohayon, 1998). Similarly on the similar basis, these acoustic resonators can be installed in the two plate structure, improving the acoustic properties of the structure, secondly being 3d printed and due

According to (Kanarachos, 2015) in his lecture 'Multi degree of vibration.' The design of vibration absorbers can be done by tuning the natural frequency of the vibration absorber to the forcing frequency and to avoid the resonance in the system or the vibration at the system natural frequency, the design absorber's natural frequency should be tuned to the natural frequency of the system and thus when the external force will act, vibration absorber will absorb the vibrations occurring at natural frequency of the system, thus not allowing the system to vibrate.

To reduce the settling time, (Vershinan, 2015) in his lecture 'Performance measure of second order system' says, maximum values of amplitude occur when natural frequency is some multiple of cycle time, where the settling time is, when the maximum amplitude is 2% of steady state position which can be represented by the equation(1).

t_(s=4/(ξω_n)) (1)

Here ξ is the damping ratio where, ω_n is the natural frequency of the system. to light weight, the acoustic resonators in the structure can prove to be very effective.

Metamaterial with stop band behaviour, which means to give noise and vibration insulation in a targeted frequency range, works on the principle of including the resonant cells at a scale smaller than the wavelengths which have to be influenced or controlled (Claeys, et al., 2013). According to a research 'Design of a resonant metamaterial based acoustic resonators', published by University of Leuven, (Claeys, et al., 2014), they say stop band behaviour can be achieved by the introduction of any system, which is capable of introducing local resonant behaviour. But the main goal is to find such resonant systems that don't jeopardise the other requirements like structural integrity, light weight, resistance to contaminated environment, fire resistance. The types of resonant systems which fulfil the criteria heavily depends upon the structure's compatibility to which it's going to be added (Claeys, et al., 2014).

So according to equation (1), settling time is inversely proportional to natural frequency and damping ratio, which means more the natural frequency and damping ratio in the system, less will be the settling time and thus more will be the optimized system.

A research done by (Spentzas & Kanarachos, 2002,) on the finite element analysis (FEA) of the mechanisms and robots, presents an applied FEA on a crank slider mechanism and to the four bar linkage mechanism. The main idea behind the FEA is illustrated which decomposes the large displacement of mechanism or robot into successive small series of displacements. The very small displacements allows the FEA method between the two close successive positions. Thus at the end of every small deformation, the position of the deformed element of the mechanisms further provides the initial conditions of the next closest point and hence every member is analysed by giving very accurate results. Similarly another research, (Kanarachos, 2008) on the 2D flexible mechanisms using linear FE techniques gives a fully linear method for kineto-elasto-dynamic analysis for the engineering practises including large displacements and rotation and small elastic displacements and strains, which are based on the standard Euler-Bernoulli finite elements. The method is based on decomposing the motion into small steps of time, so minute that the linear finite method can be applied with in each step.

To cope up with the design requirements, based on an objective for weight minimization and thus overall the cost, methods of FEA and optimization using Evolutionary Algorithms are combined to get an optimum solution (Kanarachos 2013, 2014, 2015).

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2 Problem Formulation

The acoustic design criteria was based on the following parameters.

2.1 Acoustic Design Criteria for Square Plates

The modal or natural frequencies of the square plate can be predicted from a formula

$$\omega_n = \sqrt{\frac{Et^3}{\delta d^4 (1-\vartheta^2)}} \qquad (1)$$

Where,
ω_n = Natural frequency;
E = Young's modulus;
t^3= thickness of the plate;
$\delta = mass\ density$;
$\vartheta = Poisson\ ratio$;
d = side of the plate.
According to the equation (V), natural frequency is proportional to the cube of the thickness of the plate, which means more the thickness, more will be the natural frequency, and to reduce the natural frequency, we need to reduce the thickness, but reducing the thickness is in contradiction to the static structural load scenario, so an optimized selection of plates and acoustic resonators needs to be designed, in order to get the good results for both scenarios.

2.2 Design Criteria for Acoustic resonators

To reduce the vibrations and thus the noise produced in the plate, acoustic resonators were introduced between the plates. The design criteria for acoustic resonators, was based on the fundamental rules of mechanics.
The first fundamental rule of mechanics says, acoustic energy prefers to move and carry with particles free to move in some denser medium. (Dickherber, 2008). According to the acoustic equation.

$$v_a = \sqrt{\frac{c_n}{p_n}} \qquad (2)$$

$$k = \frac{EA}{l} \quad (3)$$

Where,
k = stiffness of the connector;
E = Elastic modulus of the connector;
A = Cross sectional Area of the connector;
l = Length of the connector.

So according to equation (VIII), the stiffness of the rectangular connector is inversely proportional to the length, where directly proportional to the cross sectional area of the connector, which means, greater the length, less would be the stiffness and also from the basic natural frequency equation.

$$\omega_n = \sqrt{\frac{k}{m}} \quad (4)$$

Where,
ω_n = Natural frequency of the connector;
k = Stiffness of the connector;
m =Mass of the connector.

The lesser the stiffness, less will be the natural frequency of the structure. Hence, connectors of lesser stiffness will reduce the natural frequency of the system and so as the noise.

3 Problem Solution

An isosceles shaped triangular resonators were designed in a CAD software CATIA V5-6R2015, on a square lower plate of dimensions 62x62x1 mm, where the upper plate was designed separately of the same dimension of the lower plate and then were assembled in the CAD modeller. The structure was based on the acoustic design criteria. The isometric and top view are shown in Fig 1.
The dimensions of the whole structure are tabulated in Table 1.

Table 1 Final design dimensions.

Parts	**Dimensions (mm)**
Upper square Plate	62x62x1
Lower square Plate	62x62x1
Acoustic resonator 1	15x15x15
Acoustic resonator 2	8x8x8
Lower plate Damper 1	4x4x5
Upper Plate Damper 1	4x5x8
Lower Damper 2	2x10.5x2.5
Upper Plate Damper 2	2x2.5x10.5
Clamp supports	5x5x27
Distance from the edge	6
Distance between the resonators	6
No of Acoustic resonators1	5
No of acoustic resonators2	4
Total Structure	62x62x27
Mass	27 g

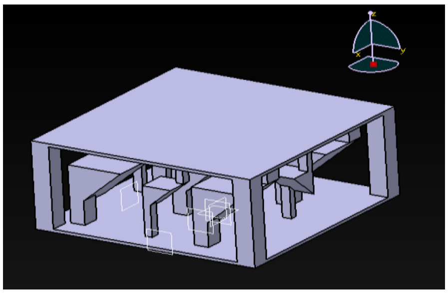

Fig 1. Final design Isometric and Top view, CATIA V5-6R2015.

3.1 FE Modal Analysis

In order to check the natural frequencies or mode shapes for the designed model, modal analysis was

3.1.1 Boundary Conditions

Workbench ANSYS16.1 is the software used for modal analysis. Geometry was imported, which was modified with additional surfaces needed as a boundary condition for the modal analysis. The material selected for the model is PLA.

3.1.2 Discussion of Results

After selecting the mesh size and other boundary conditions for the modal analysis of the designed model, The system was solved for the first 9 natural frequencies. The first nine natural frequencies are tabulated in Table 2. The mode shapes for their respective natural frequencies are created.

The first nine natural frequencies obtained shows the attenuating of the vibrations. Which shows the efficiency of the acoustic resonators. The highest natural or resonance frequency for mode 9 is 1575.4Hz.

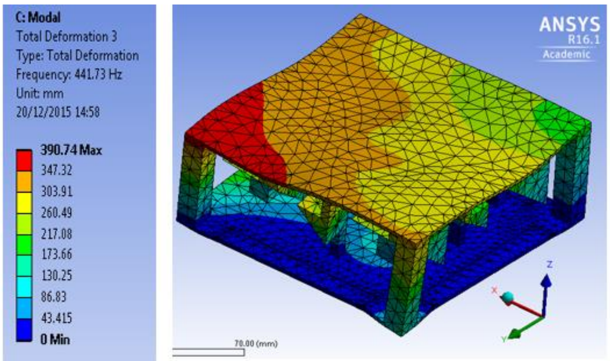

Fig 2. Boundary condition for modal analysis, final design, Workbench ANSYS16.1.

Table 2 Natural frequencies for the final design, modal analysis, Workbench ANSYS16.1

S. No	Mode No	Natural frequency (Hz)
1.	1	369.13
2.	2	441.73
3.	3	506.75
4.	4	639.42
5.	5	850.62
6.	6	1155.4
7.	7	1194.9
8.	8	1336.8
9.	9	1575.4

3.2 Experimental Modal Testing

Experimental modal testing was done in order to validate the model. Impact hammer method was selected for modal testing with the LMS Xpress software, for the analysis of natural frequencies obtained and their respective mode shapes. FFT analyser (LMS SCADAS XS, Impact hammer, accelerometer), wax, tape were the apparatus used.

The FFT spectrum analyser measures the response of the structure, when it is excited by an external force. FFT stands for Fast Fourier Transform. The FFT system consists of data acquisition hardware (LMS SCADAS XS, a portable hardware), the analyser software (LMS Test Xpress software), impact hammer and sensor (accelerometer).

The LMS SCADAS XS collects and records the data from impact hammer (Fig 9) connected to one of its port and response through an accelerometer (sensor) attached to the structure (Fig 8) and connected to the other port of LMS SCADA XS. Then it converts the analog signal into digital signal and sends it to the computer for the analysis. The data acquisition has its own software, like LMS Test Xpress software, which is a sound and vibration analyser with high speed performance and quality.

Fig 3. Light weight plates hosting acoustic resonators.

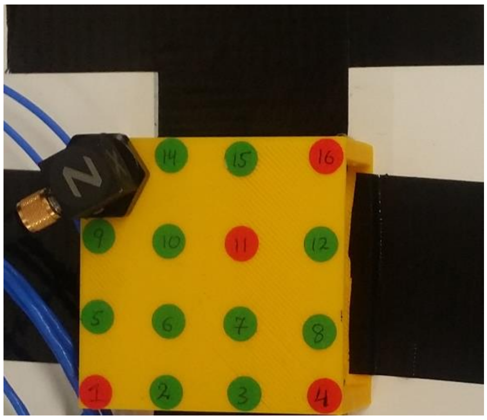

Fig 4. Accelerometer attached to the structure.

The first nine natural frequencies (Table 3) were selected and their mode shapes were plotted. The maximum natural frequency obtained from the modal testing for mode 9 is 1555.409Hz.

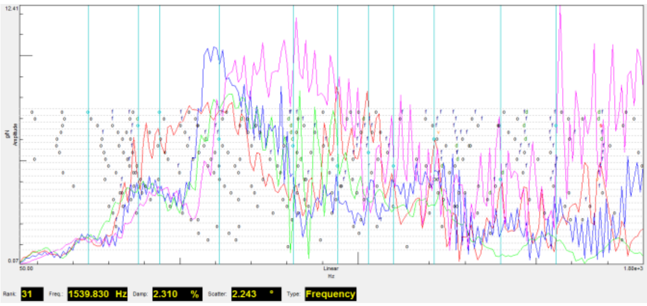

Fig 5. Accelerometer attached to the structure

Table 3 Natural frequencies, modal testing

Modes	Natural Frequency (Hz)
1	382.571
2	443.469
3	610.205
4	817.665
5	942.917
6	1100.565
7	1213.096
8	1399.967
9	1555.409

From the modal testing experiment, the natural frequencies obtained are quite close to the ones obtained from finite element modal analysis on Workbench ANSYS 16.1. This shows that the validation of the model designed.

From the comparison bar chart (Bar chart 1), quite correspondence is observed between the FEA obtained results and experimental test results.

The mode shapes created from the FEA software and lab testing software are also compared Fig 13 and it clearly shows the correspondence between the mode shapes for their respective natural frequencies. This is another method of justification.

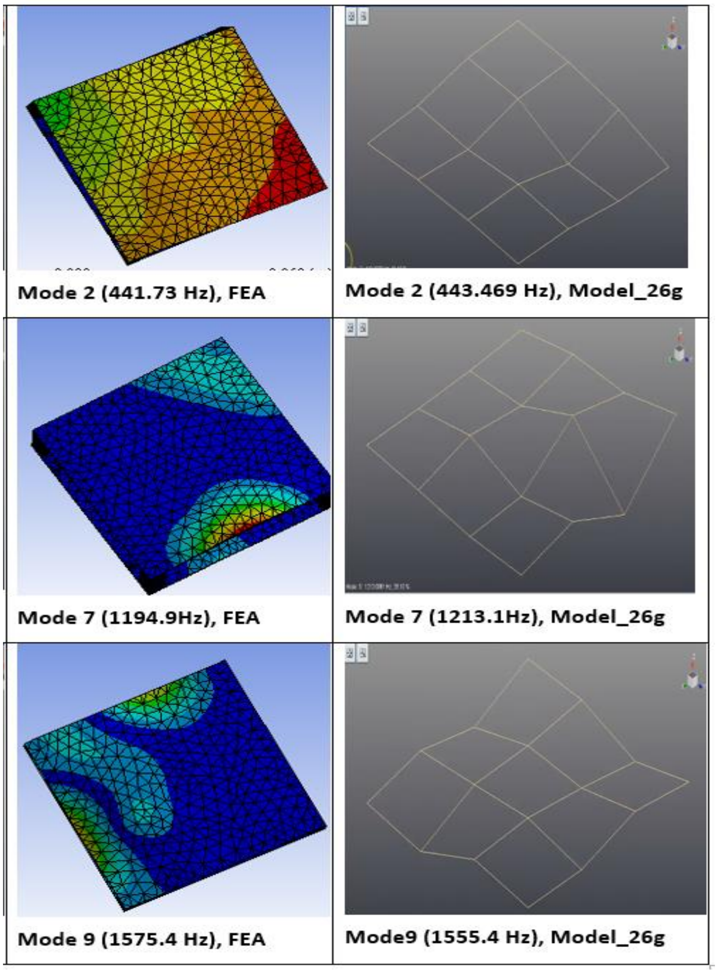

Fig 6. Mode Shapes Comparison.

4 Conclusion

This research shows the potential of the use of innovative acoustic resonators in the light weight structures, especially manufactured from 3d printing. The parts manufactured by FDM additive manufacturing method are quite feasible to be made and experimented. So a major problem being faced as a result of the additive manufacturing, which is the acoustic properties of light weight structures like, honeycomb structure or other light weight structures can be countered by putting acoustic resonators in the structure. Cost and weight must be accounted and so as the feasibility study of the part before manufacturing the model. Processing and manufacturing time are also very important. So the design should be made quite efficient, with low mass, light weight, good acoustic properties in addition to good mechanical properties.

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