Article 4

Vibration Control In Membrane Structures Using Fem For Structural Safety

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Key terms used:

Vibration Control;
Elastomer;
DEA;
ANSYS;
Finite Element Analysis

ABSTRACT

This paper deals with active vibration control on the space structures using FEM method for the structural safety. These are light & flexible type structures. They need effective vibration control methods. Various papers are published with respect to the active vibration control. This paper refers a particular study done by Hiruta *et al (2021)* in which the conclusion was to do the experiment in FEA and find out the optimal location for placement of those actuators considered. The actuators used are Dielectric Elastomer Actuator (DEA), these are most effective than Piezoelectric materials. The study includes software numerical analysis using ANSYS and do the finite element analysis on the particular material chosen in the paper by Hiruta *et al (2021),* compare the results and find

The experimental arrangement was reproduced in the numerical prediction being conducted in this paper. It was made possible by comparing the values of the modal analysis and vibration suppression values as referred in the Hiruta *et al.* paper. Thereafter, based on the transient analysis conducted in this report, the optimal location was studied. For this the various locations opted alternately and randomly were centre & diagonal, centre & horizontal and side centre other than the corner placed elastomer as referred in the Hiruta *et al.* paper. It was able to reduce the vibration on an average by 86%, 87% & 32%, respectively. Further it was made clear that Centre & Horizontally placed elastomer patch has much better vibration suppression, i.e, of 87%, with 50% more vibration suppression than when the elastomer patch is placed in corner location or side centre location.

1.INTRODUCTION

1.1 METHODS OF VIBRATION CONTROL

There are different vibration control methods for different machineries and systems & among the various types are the passive vibration control and active vibration control. Passive control involves use of dampeners, springs etc. by using their natural properties to reduce the effect of excitation to the structures/floors of concern. No external energy is input on to this system for any type of control. Dynamic characteristics remains unchanged in this type of control.

Active vibration control used to isolate dynamic excitations on the system. Once it senses the vibrations it will react accordingly to the mathematical model which has been embedded in it. Active vibration involves feedback and feed forward signals for proper working of the controls. Even though the cost is generally higher due to the use of actuators, these have become popular due to the effective suppression of vibrations and effectiveness in control. To avoid resonance the most effective method is the active vibration control compared that to the classic passive control method, in maintaining the level of vibration. For small machines and light weights, the Active Vibration Control is advantageous [1].

Figure 1: Typical Passive Vibration Isolation System (Source: Vibration Engineering Consultants data book, 2019)

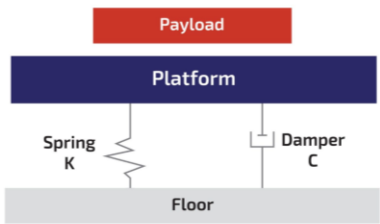
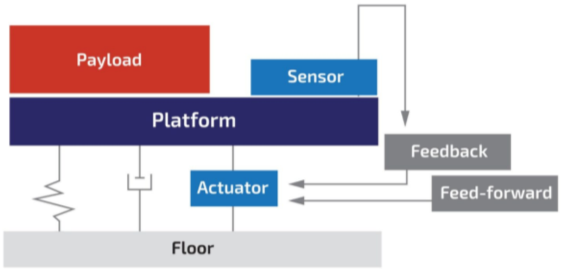


Figure 2: Typical Active Vibration Isolation System (Source: Vibration Engineering Consultants data book, 2019)



1.2 OBJECTIVES OF WORK

Nowadays as optimum control Active vibration is used for flexible structures. The challenge comes in the vibration control process because of complexity of the dynamic system. Some of the techniques used in the control of system vibrations include Piezoelectric materials and Dielectric Elastomers (DEAs). Presently, piezoelectric materials are used in the harvesting of mechanical energy because of its compatibility and compactness. However, its integral limits includes brittleness, aging and depolarization, these confines its development. Hence, DEAs potential in harvesting mechanical is being utilized more and more now. Larger and flexible space structures are made in this age of space technology with modal frequencies and damping ratios being relatively low. The precise requirement is to be met, thus application of active control for suppression of vibrations becomes more significant than ever before. Further Literature review was conducted to know the various studies on this Active Vibration Control behalf. The objective of the project is to perform Finite Element Analysis (FEA) of the experimental work carried out by Hiruta *et al.* (2021) for vibration control in membrane structures using dielectric elastomer actuators [3]. It is proposed to get the similar result using Finite Element Analysis using Ansys software for the experimental analysis done. The results to be obtained include to assess the Effectiveness of vibration control for a membrane structure using DEA in vacuum environments for structural safety and to determine optimal actuator placement effectively.

2. PRESENT INVESTIGATION

2.1 NUMERICAL PROCEDURE

The object is a membrane structure including a 0.05-mm-thick polyimide film (Du Pont-Toray, Kapton 200H) with weights & wires. Tension is applied on the corners of membrane

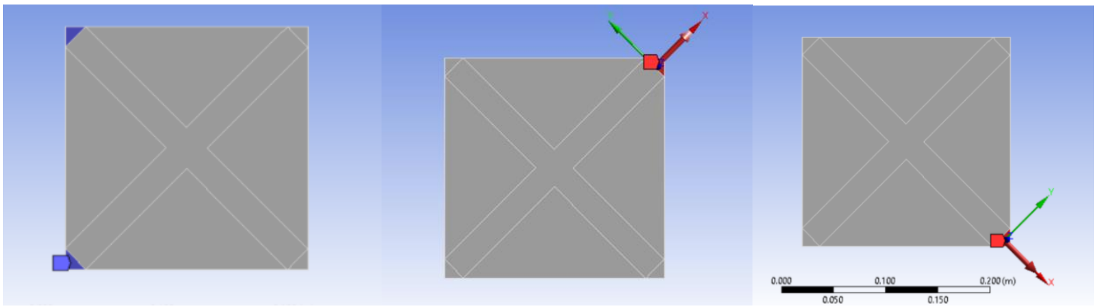
(200 mm × 200 mm) by using weights (each mass: 700 g) combined with wires. The properties of the Polyimide film is as tabulated below.

Physical Property	Value
Young's modulus	2.5 GPa
Mass density	1.42 g/cc
Poisson ratio	0.34 @ 23°C
Tensile strength	231 MPa @ 23°C 139 MPa @ 200°C

Table 1 Physical Properties of Polyimide film (Source: Dupont Kapton Polyimide Film General Specifications, Bulletin GS-96-7". http://www.dupont.com/kapton/general/H-38479-4.pdf)

In ANSYS geometry, tension forces was applied on the two adjacent corners of the film, i.e, of 13.734N each. The other two adjacent corners are made as fixed supports.

Figure 3 Tension applied on the corners (image from ANSYS)

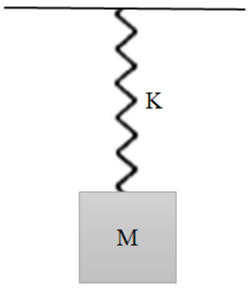


2.2 GOVERNING EQUATION

The governing equation considered is that of a simple Spring Mass System. Basic oscillatory system consists of a mass, massless spring and damper. A system with spring and a mass is capable of free vibrations and no external excitation needed. The parameters and equation are as follows.

Considering a spring mass system with mass 'M' and spring constant 'K'.

Figure 4: Simple Spring Mass System



'A' is the amplitude at which system vibrates

Eigen value problem statement is as follows, in natural case *(taken from Mechanical Vibrations book by S. Graham Kelly);*

$([K] - \omega_i^2 * [M]) * A_i = 0$ ---(1)

Here,

ω_i^2 is Eigen value

A_i is the Eigen vector

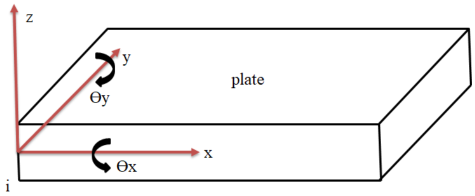
[M] – Mass matrix

[K] – Stiffness matrix

ω_i = angular frequency,

Natural frequency, $f = \omega_i/(2*\pi)$

Figure 5 Degrees of freedom of Membrane



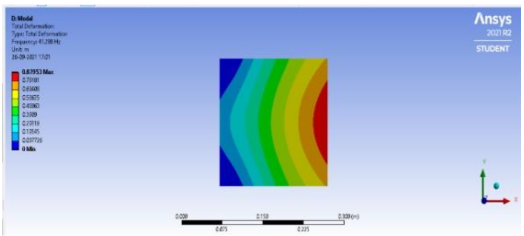
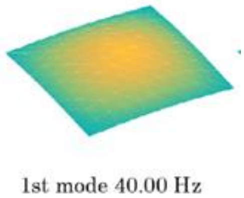
From the above indicative sketch it is clear that, Number of modes = Degrees of freedom = Size of matrix. Degrees of freedom in z direction and Θ_x & Θ_y rotations.

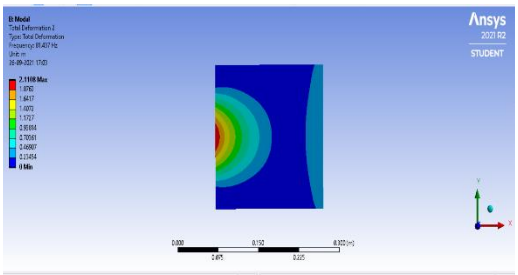
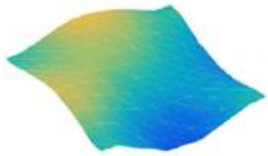
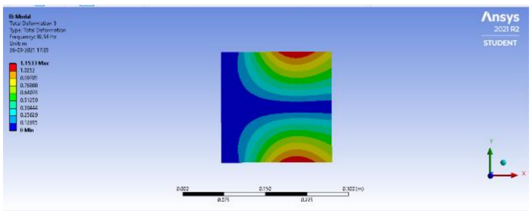
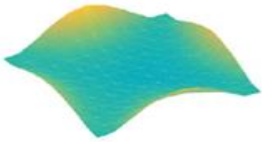
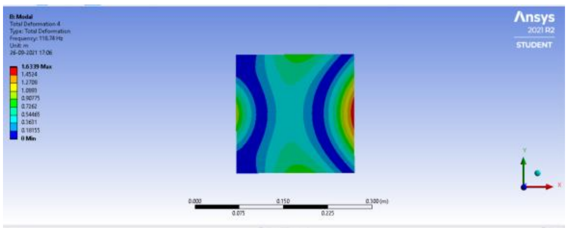
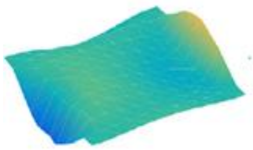
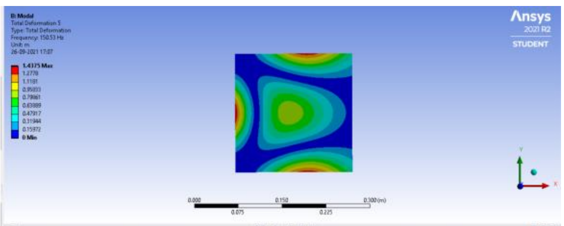
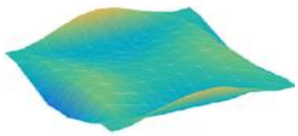
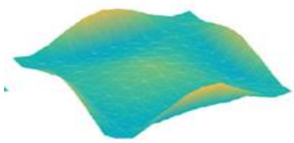
2.3 MODAL ANALYSIS

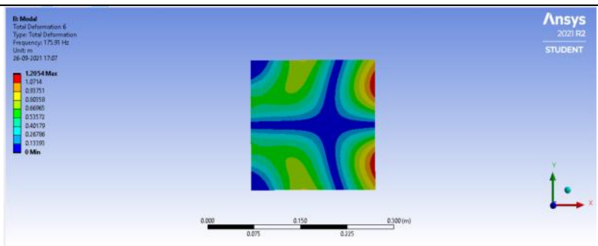
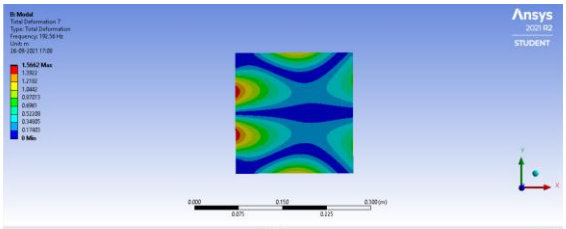
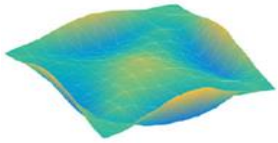
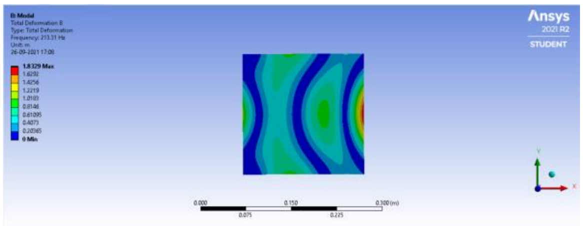
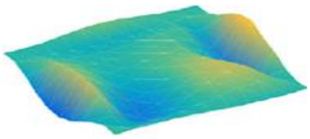
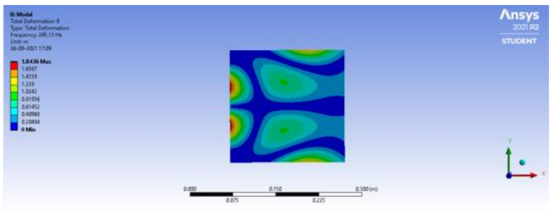
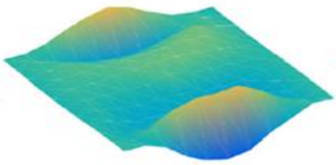
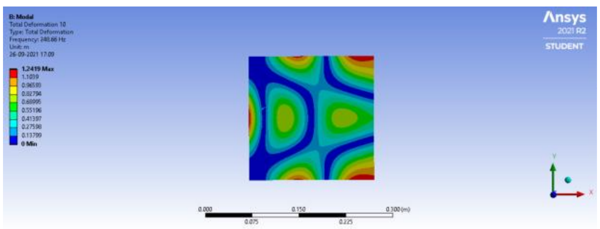
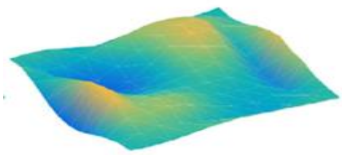
Further based on the geometry created, the modal analysis was done to get the required mode shapes and natural frequencies from the ANSYS software to compare with the experimental results obtained in the paper mentioned [3].

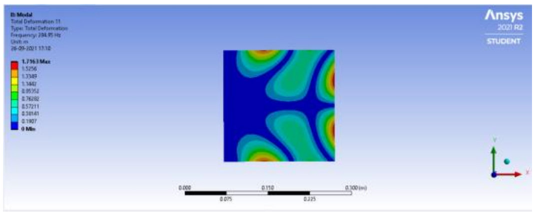
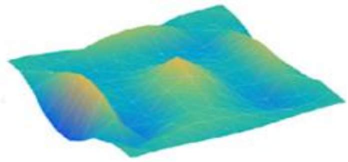
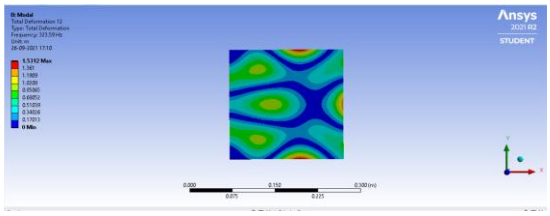
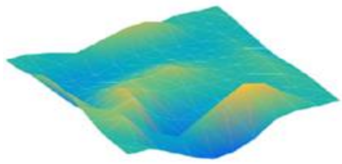
The solution as obtained is presented as below images of ANSYS. The results obtained shows the matching result of the first four modes of natural frequencies to that of the paper. The reasons for the difference in values of other modes may be due to the different restraints considered in the actual experiment considered which will have an impact on the natural modes of vibrations. Such considerations are also not clear in the paper presented by Hiruta *et al.*

Figure 6: Different Modes (Sl. Nos. A to L) from ANSYS Modal Analysis and from referred paper [3]

S. No.	Different Modes from ANSYS Modal Analysis	Different Modes from referred paper [3]
A	Mode 1 	

S. No.	Different Modes from ANSYS Modal Analysis	Different Modes from referred paper [3]
B	Mode 2 	 2nd mode 65.60 Hz
C	Mode 3 	 3rd mode 75.00 Hz
D	Mode 4 	 4th mode 83.13 Hz
E	Mode 5 	 5th mode 93.13 Hz
F	Mode 6	 6th mode 99.38 Hz

S. No.	Different Modes from ANSYS Modal Analysis	Different Modes from referred paper [3]
		
G	Mode 7 	 7th mode 106.9 Hz
H	Mode 8 	 8th mode 116.9 Hz
I	Mode 9 	 9th mode 120.6 Hz
J	Mode 10 	 10th mode 128.1 Hz

S. No.	Different Modes from ANSYS Modal Analysis	Different Modes from referred paper [3]
K	Mode 11 	 11th mode 141.3 Hz
L	Mode 12 	 12th mode 143.8 Hz

2.4 NUMERICAL PROCEDURE FOR VIBRATION CONTROL

An elastomer of acrylic based material is attached to the membrane structure as the Dielectric Actuator for the control/suppression of vibrations occurring on the membrane structure. The material used is of VHB series, 3M, Thickness: 0.5 mm.

Physical Property	Value
Young's modulus	3 MPa
Mass density (as per 3M VHB series product data sheet)	0.710 g/cc

Table 2 Physical Properties of Acrylic material (Romasanta *et al.*, 2015)

During ANSYS transient analysis, 0.650 N force is applied as laser on the central portion of the membrane. According to the report of Hiruta *et al.* (2021), around 37% reduction has happened in the present corner location of acrylic material. Thus the corresponding suppression force considered onto the elastomer patch attached to the membrane structure in the present numerical analysis study. Other than the corner location of elastomer patch, other 3 alternate locations were chosen and analysed for the percentage reduction in vibration. The various locations are as shown in below figures.

Figure 7: Forces applied: Corner placed Elastomer patch

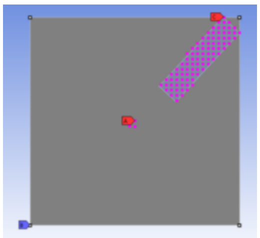


Figure 8: Forces applied: Centre & Diagonally placed Elastomer patch

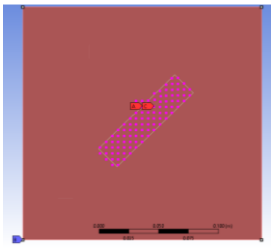


Figure 9: Forces applied: Centre & Horizontally placed Elastomer patch

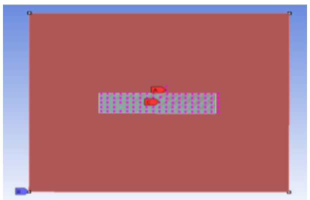
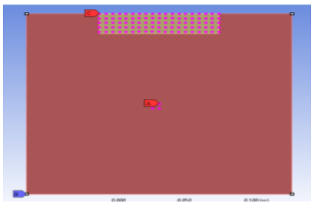


Figure 10: Forces applied: Side centre placed Elastomer patch



2.5 GOVERNING EQUATION FOR TRANSIENT ANALYSIS

With respect to the vibration control, a transient structural dynamic system was analyzed. Time dependent analysis is solved as per the below equation. If the mass M is subjected to a force F(t) acting in the positive x(t) direction as shown in figure below, then the equation of motion, becomes;

$$[M]x''(t) + [C]x'(t) + [k]x(t) = F(t) \quad \text{--}(2)$$

Here,

[M] – Mass matrix

[C] – Damping matrix

[k] – Stiffness matrix

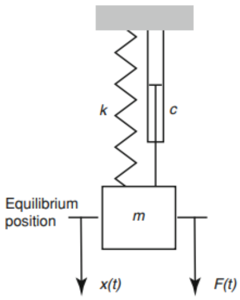
F(t) – Load

x"(t) – Nodal Acceleration

x′(t) – Nodal Speed

x(t) – Nodal Displacement

Figure 11 Forced excitation system



2.6 TRANSIENT ANALYSIS

Transient vibration is one that dies away with time due to energy dissipation. Usually, there is some initial disturbance and following this the system vibrates without any further input. The Finite Element Transient Analysis is done on the system without vibration control and with vibration control. The results of amplitude and deformation for various controls is as shown below.

Figure 12: Amplitude & deformation without control

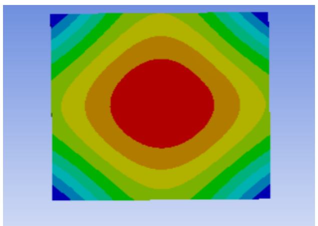


Figure 13: Amplitude & deformation with control of corner placed elastomer patch

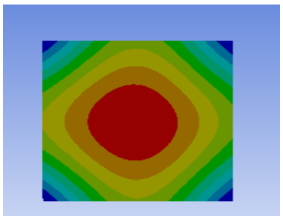
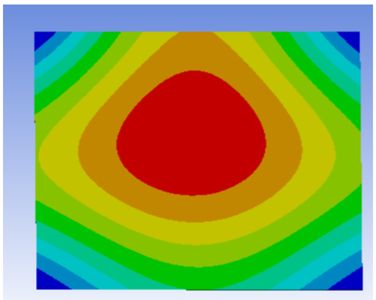


Figure 14: Amplitude & deformation with

Figure 15: Amplitude & deformation with

control of centrally & diagonally placed elastomer patch

control of centrally & horizontally placed elastomer patch



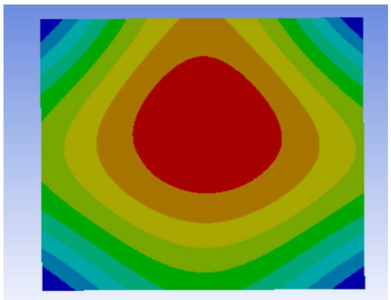
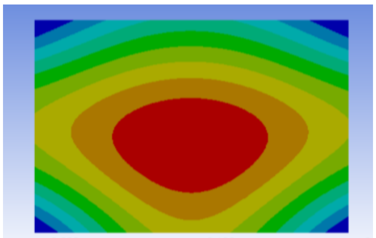


Figure 16: Amplitude & deformation with control of side centre placed elastomer patch



3. RESULTS AND DISCUSSION

Based on the available data from the journal paper presented by Hiruta *et al.* (2021), it was able to achieve nearby values for at least the first 4 modes of natural frequencies. The results so obtained from numerical analysis and value available in the paper is presented as below.

Mode	Numerical Prediction	Experimental Values
1	41.288	42.42
2	81.437	74.67
3	92.54	83.25
4	118.74	84.47

Table 3 Comparison of the Natural Frequency

Once the above tabulated result was obtained, further transient analysis was conducted in order to achieve/reproduce the vibration suppression so studied in the referred paper, i.e, with elastomer placed near to the corner of the membrane structure. Based on the transient analysis performed, it was able to reproduce the vibration suppression for corner placed elastomer patch as referred in the paper. The result of the amplitude from numerical prediction for with and without control for the corner placed elastomer patch is as tabulated below.

Time (s)	Amplitude without control (cm)	Amplitude with control (cm) (As per paper)	Percentage suppression
0.1	1.33	0.83	38%
0.2	0.13	0.08	38%
0.3	1.54	0.96	38%
0.4	0.12	0.08	38%
0.5	1.31	0.81	38%
0.6	0.07	0.05	39%
0.7	1.44	0.89	38%
0.8	0.01	0.004	25%
0.9	1.42	0.88	38%
1	0.02	0.01	38%
1.1	1.38	0.85	38%
1.2	0.03	0.02	38%

Table 4 Amplitude from Numerical Prediction for with and without control (Corner placed elastomer patch)

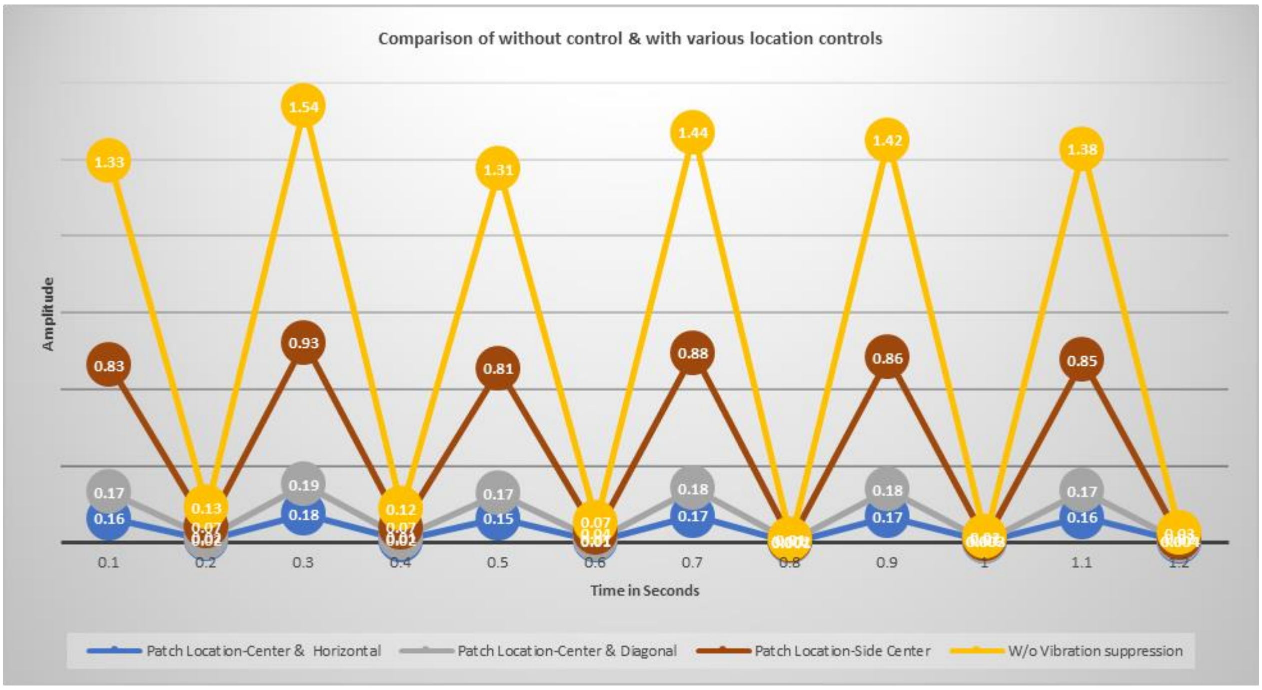
Now further to study the gap left out in the paper of Hiruta et al. by finding out the optimal location for vibration suppression in membrane structures for structural safety. The elastomer was alternately and randomly placed at various locations such as in central & diagonal, center & horizontal and also in side center locations. Thereafter the analysis was carried out with these arrangements. Hence it was able to reduce the vibration on an average by 86%, 87% & 32%, respectively. The tabulated amplitude values from numerical predictions for without and with control for various alternately placed elastomer patch locations is as below.

Time (s)	W/o Vibration suppression	With Vibration Suppression							
		Patch Location-Corner (As per paper)	Percentage suppression	Patch Location-Center & Diagonal	Percentage suppression	Patch Location-Center & Horizontal	Percentage suppression	Patch Location-Side Center	Percentage suppression
0.1	1.33	0.83	38%	0.17	87%	0.16	88%	0.83	38%
0.2	0.13	0.08	38%	0.02	88%	0.02	87%	0.07	45%
0.3	1.54	0.96	38%	0.19	88%	0.18	88%	0.93	40%
0.4	0.12	0.08	38%	0.01	89%	0.02	88%	0.07	45%
0.5	1.31	0.81	38%	0.17	87%	0.15	88%	0.81	38%
0.6	0.07	0.05	39%	0.01	87%	0.01	88%	0.04	46%
0.7	1.44	0.89	38%	0.18	87%	0.17	88%	0.88	39%
0.8	0.01	0.004	25%	0.002	70%	0.001	79%	0.01	-76%
0.9	1.42	0.88	38%	0.18	88%	0.17	88%	0.86	39%
1	0.02	0.01	38%	0.002	90%	0.003	85%	0.01	47%
1.1	1.38	0.85	38%	0.17	87%	0.16	88%	0.85	38%
1.2	0.03	0.02	38%	0.004	88%	0.004	87%	0.02	47%
Average			**37%**		**86%**		**87%**		**32%**

Table 5 Amplitude (in cm) from Numerical Prediction for without control and with control for various alternately placed elastomer patch locations

The Graphical representation of the vibration suppression obtained for all alternate elastomer patch locations placed is as shown below. The graph explicitly shows the difference in the suppression of various patch locations.

Figure 17 Graphical representation of comparison of without control & with various location controls



4. CONCLUSIONS

The modal analysis results obtained from the modelled parameters of the controlled object in vacuum environment is compared with those values in the paper [3]. Based on the available data from the journal paper presented [3], it was able to achieve nearby values for at least the first 4 modes of natural frequencies. The reason for the major variation in the values that of the paper for the modelled geometry is found to be due to the various restrains considered in the actual experimental paper is not clearly indicated hence the difference. Now further transient analysis was conducted to get the active vibration control on the membrane structure to get the reduction/suppression in the vibrations as stipulated in the paper [3] for the structural safety. It was successful in achieving the required results after reproducing the present studied vibration suppression for corner placed elastomer patch as indicated on the paper being referred. The percentage in suppression occurred as per the Hiruta *et al.* paper is around 38% which is being similar to the results obtained from numerical analysis, thus arriving at the conclusion that the similar experimental arrangement has been made in the analysis performed. Based on the above method further numerical analysis was conducted by placing the elastomer patch at alternate locations such as on centre & diagonal, centre & horizontal and side centre.

As part of furtherance of the Hiruta *et al.* paper & based on the Finite Element transient Analysis, the optimal location of elastomer actuator was to be found. Therefore, in ANSYS model, elastomer was placed in central & diagonal, centre & horizontal and also in side centre locations and conducted the transient analysis. Thereafter it was able to reduce the vibration on an average by 86%, 87% & 32%, respectively. From all the alternate and randomly placed locations, it is clear that Centre & Horizontally placed elastomer patch has much better vibration suppression, i.e, of 87%. It is explicit from the findings that 50% more vibration suppression is occurring than when the elastomer patch is placed in corner location or side centre location. However there is only a slight change of approx. 1% in vibration suppression when placed either Centre & Diagonal or Centre & Horizontal. Hence the numerical analysis under this report is concluded by stating the fact that the closer to the disturbance or centred the elastomer is placed, more effective will be the vibration suppression to maintain the structural stability.

5.REFERENCES

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