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Performance Analysis of Shock Mounts Using FEA

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Abstract: The shock mounts are designed to withstand harsh environments, shock and vibration. Shock mounts are different elements to connect two units or to install one unit into a system. The design of these shock mounts is a result of the development considering the space, the technical characteristics and requirements (load, frequencies, amplitudes, maximum load, etc.) as well as the surrounding conditions. Finite Element Analysis plays important role in structural safety of machine components. Prior estimation of stress levels prevents catastrophic failure of structure. The earlier techniques of prototype testing using destructive techniques are replaced by computer based simulation. Due to the advances in Finite Element Techniques with fast processing computers, the designer has the flexibility of selecting suitable mechanism and material for the optimum design. Present work is on the shock absorber system used to safeguard the CPU frame of the Aircraft. The system is made of number of components, modeling is done in Catia-V5 and meshing is done in Hypermesh. Theoretical calculations have been carried out for the design. The shock loads will be applied on the system to check the stress condition. Based on the response improvements will be represented for comparison. Various shock mount configuration will be discussed in the project to select the best possible shock absorber system.

I. INTRODUCTION

1.1. Vibration:

Definition: The relative motion of a particle or a body that repeats on its own after a certain time period is called as the vibration. Most vibrations in machines and structures are undesirable because of the increased stresses and energy losses which accompany them. They should therefore be eliminated or reduced as much as possible by appropriate design. The analysis of vibration has become increasingly important in recent years owing to the current trend towards higher speed machines and lighter structures.

There is every reason to expect that this trend will continue and that an even greater need for vibration analysis will develop in the future. A mechanical vibration generally results when a system is displaced from a position of stable equilibrium. The system tends to return to this position under the action of restoring forces (either elastic forces, as in the case of a mass attached to a spring, or gravitational forces, as in the case of a pendulum).

Since the process can be repeated indefinitely, the system keeps moving back and forth across its position of equilibrium. The time interval required for the system to complete a full cycle of motion is called the period of vibration. The number of cycles per unit time defines the frequency, and the maximum displacement of the system from its position of equilibrium is called the amplitude of vibration. In this work first, the finite element models and solution methods needed for the accurate calculation of two dimensional stresses and deflection were determined. Then, the stresses and deflection calculated using ANSYS 11 were compared to the results obtained from existing methods. The purpose of this thesis is to develop a model to study and predict the deflection & stresses, and also modal & spectrum analysis using the ANSYS 11 software package based on numerical method.

1.2 Objectives Of The Research:

The objectives of this thesis are to use a numerical approach to develop theoretical models of the behaviour of shock mount in mesh, to help to predict the effect of stresses and deflection. The main focus of this research as developed here is: To develop and to determine appropriate models of shock mount elements, to calculate stress using ANSYS and compare the results with theoretical. Modal analysis of shock mounts and shock load analysis for stress safety using Ansys. Spectrum and shock analysis for various damper members for minimum displacement response. Comparative study of Shock Mounts by analysis and selecting the optimal one and survey of suitable damping material for the shock mount.

II. LITERATURE SURVEY

Literature survey plays an important part in conducting the research. This gives several technical ideas and steps to be followed to complete the work. Therefore, this provides importance for using various technical papers and books to learn and apply the technical aspects for executing this work.

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III. MATERIAL SELECTION

3.1Material Properties of Steel & Elastomer

TABLE-I MATERIAL PROPERTIES OF STEEL

SL No	Description	Value
01	Density	7800 Kg/m^3
02	Young's Modulus	2x10^5 MPa
03	Yield Strength	250 MPa
04	Poisson's Ratio	0.3

TABLE-II. MATERIAL PROPERTIES OF ELASTOMER

SL No	Description	Value
01	Density	1600 Kg/m^3
02	Young's Modulus	2x10^6 MPa
04	Poisson's Ratio	0.45

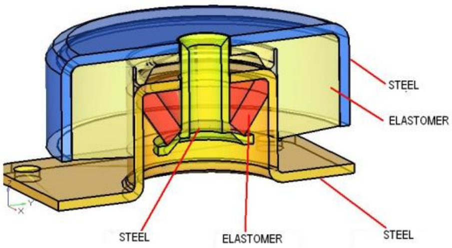

Fig.1 Type of Materials Used In Shockmount.

Elastomers are designed to support heavy equipment and isolate intermittent or continuous vibration. The result is greater accuracy, longer service life, smoother operation and reduced maintenance. ShockMounts are recommended for machines where vertical disturbing frequencies are 600 cpm or higher.

IV. ELEMENT SELECTION & BOUNDARY CONDITIONS

Solid45 is used for 2-D modeling of solid structures. The element is defined 8 nodes having 6 degrees of freedom at each node: translation and rotation in the node x, y, and z directions. The element also has plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities. The element can be used either as a plane element (plane stress or plane strain) or as an axisymmetric element. The element is defined by four nodes having two degrees of freedom at each node: translations in the nodal x and y directions. The element has plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities.

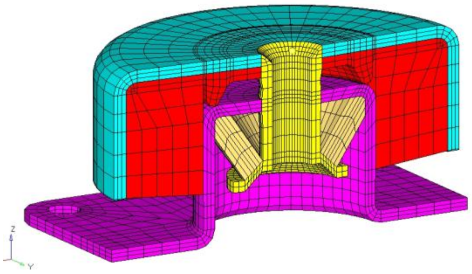

Fig.2 Sectional view of elements in 3- Dimension

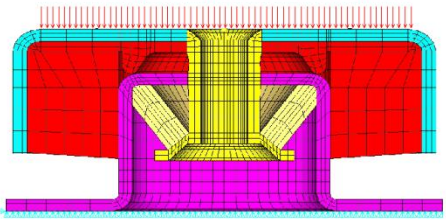

Fig.3 Boundary Conditions Applied for Shockmount

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TABLE-III MESHING INFORMATION

Element Type	Solid 45
Meshing	Coarse
Elements	13948
Nodes	20053
Smart Size	8

V. MODAL ANALYSIS

Shock Mount-1

TABLE IV SHOWING FREQUENCY AND LOAD SET.

***** INDEX OF DATA SETS ON RESULTS FILE *****

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	1.1259	1	1	1
2	4.8629	1	2	2
3	140.42	1	3	3
4	260.75	1	4	4
5	470.69	1	5	5
6	608.23	1	6	6
7	750.70	1	7	7
8	892.45	1	8	8
9	995.66	1	9	9
10	1210.6	1	10	10
11	1356.1	1	11	11
12	1467.3	1	12	12
13	1501.7	1	13	13
14	1623.6	1	14	14
15	1709.8	1	15	15
16	1854.5	1	16	16
17	2008.5	1	17	17
18	2114.9	1	18	18
19	2315.1	1	19	19
20	2867.6	1	20	20
21	0.0000	2	1	21
22	0.0000	3	1	22
23	0.0000	4	1	23
24	0.0000	5	1	24

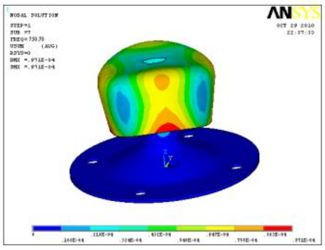

Fig.4 Mode Shape-7

5.2 Shock Mount-2

TABLE V SHOWING FREQUENCY AND LOAD SET.

***** INDEX OF DATA SETS ON RESULTS FILE *****

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	0.8675	1	1	1
2	2.1036	1	2	2
3	7.7300	1	3	3
4	12.458	1	4	4
5	14.867	1	5	5
6	17.064	1	6	6
7	120.62	1	7	7
8	224.55	1	8	8
9	523.98	1	9	9
10	619.22	1	10	10
11	718.54	1	11	11
12	886.12	1	12	12
13	1059.1	1	13	13
14	1156.8	1	14	14
15	1301.5	1	15	15
16	1443.0	1	16	16
17	1541.2	1	17	17
18	1924.6	1	18	18
19	2567.8	1	19	19
20	3185.5	1	20	20
21	0.0000	2	1	21
22	0.0000	3	1	22
23	0.0000	4	1	23
24	0.0000	5	1	24

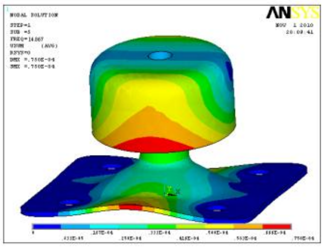

Fig- 5 Mode Shape-5

1.3 Shock Mount-3

TABLE VI SHOWING FREQUENCY AND LOAD SET

***** INDEX OF DATA SETS ON RESULTS FILE *****

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	1.0053	1	1	1
2	2.2619	1	2	2
3	70.232	1	3	3
4	135.85	1	4	4
5	357.61	1	5	5
6	588.45	1	6	6
7	733.67	1	7	7
8	978.95	1	8	8
9	1024.3	1	9	9
10	1315.4	1	10	10
11	1457.1	1	11	11
12	1534.8	1	12	12
13	1601.7	1	13	13
14	1813.3	1	14	14
15	1929.2	1	15	15
16	2010.5	1	16	16
17	2212.9	1	17	17
18	2410.9	1	18	18
19	2765.1	1	19	19
20	3261.7	1	20	20
21	0.0000	2	1	21
22	0.0000	3	1	22
23	0.0000	4	1	23
24	0.0000	5	1	24

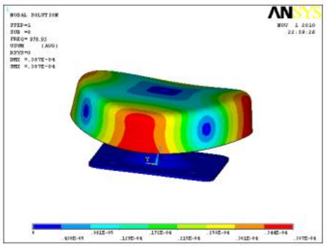

Fig- 6 Mode Shape-8

5.4 Shock Mount-4

TABLE VII SHOWING FREQUENCY AND LOAD SET

***** INDEX OF DATA SETS ON RESULTS FILE *****

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	0.30903	1	1	1
2	3.31228	1	2	2
3	12.3916	1	3	3
4	51.252	1	4	4
5	112.61	1	5	5
6	321.12	1	6	6
7	451.30	1	7	7
8	571.38	1	8	8
9	883.25	1	9	9
10	1204.6	1	10	10
11	1503.9	1	11	11
12	1842.8	1	12	12
13	2211.8	1	13	13
14	2450.6	1	14	14
15	2545.1	1	15	15
16	2765.9	1	16	16
17	2816.5	1	17	17
18	3019.3	1	18	18
19	3249.0	1	19	19
20	3557.4	1	20	20
21	0.0000	2	1	21
22	0.0000	3	1	22
23	0.0000	4	1	23
24	0.0000	5	1	24

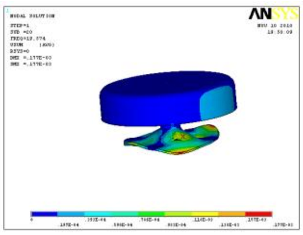

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Fig- 7 Mode Shape- 20

5.5 Shock mount-5

TABLE VIII SHOWING FREQUENCY AND LOAD SET

***** INDEX OF DATA SETS ON RESULTS FILE *****

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	0.78310	1	1	1
2	4.7426	1	2	2
3	177.18	1	3	3
4	276.00	1	4	4
5	442.97	1	5	5
6	595.12	1	6	6
7	783.51	1	7	7
8	852.65	1	8	8
9	1000.6	1	9	9
10	1117.1	1	10	10
11	1271.4	1	11	11
12	1431.1	1	12	12
13	1608.5	1	13	13
14	2113.4	1	14	14
15	0.0000	2	1	15
16	0.0000	3	1	16
17	0.0000	4	1	17
18	0.0000	5	1	18

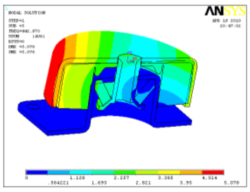


Fig- 8 Mode Shape- 9

VI. SPECTRUM ANALYSIS

6.1 Shock Mount-1

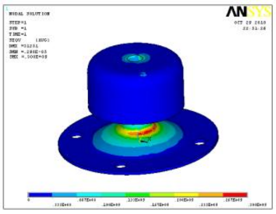
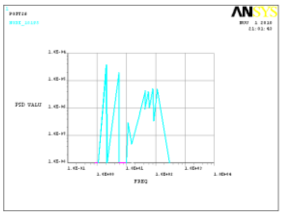


Fig- 9 Stress Due To Vibration



Graph- 1 Vibration Response.

6.2 Shock Mount-2

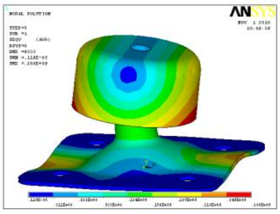
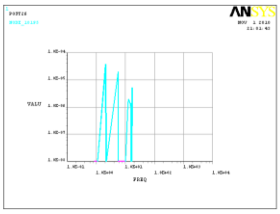


Fig-10 Stress Due To Vibration



Graph- 2 Vibration Response

6.3 Shock Mount-3

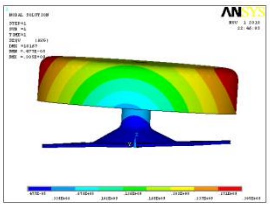
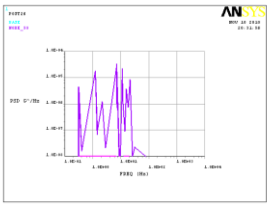


Fig-11 Stress Due To Vibration



Graph- 3 Vibration Response

6.4 Shock Mount-4

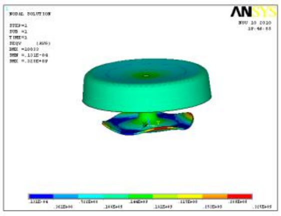
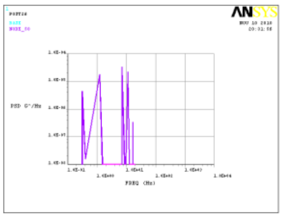


Fig- 12 Stress Due To Vibration



Graph- 4 Vibration Response

VII. IMPROVEMENT ANALYSIS

In order to improve the performance of Shock Mount we have to reduce the stresses, generated due to vibration. This can be obtained by changing the design and material properties. But the major design changes and material properties are not permitted because the structures are designed considering a number of parameters like; space availability,

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calculation, minimum weight, optimum accessibility and so on. Hence only the minor design changes are feasible and permissible [3].

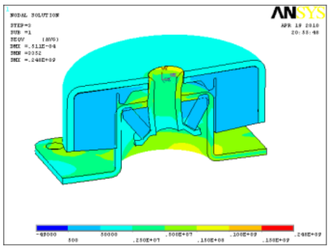


Fig-13 Stress value for 5% damping

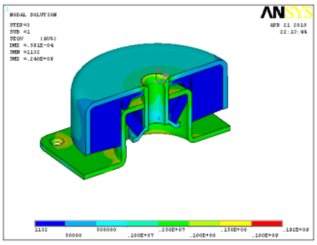


Fig-14 Stress value for 10% damping

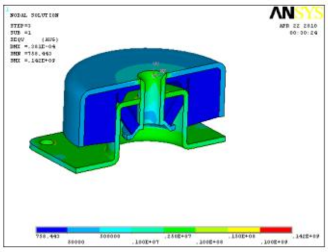
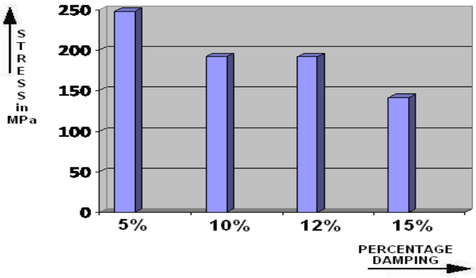


Fig-15 Stress value for 15% damping



7.1 Graphical Representation:

Graph-5 Graph representing the improvement.

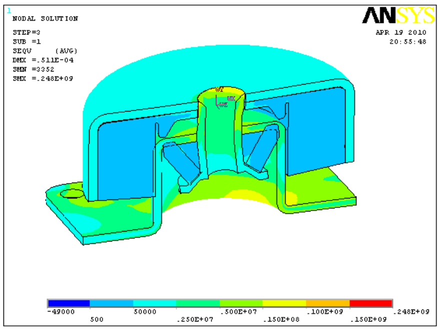
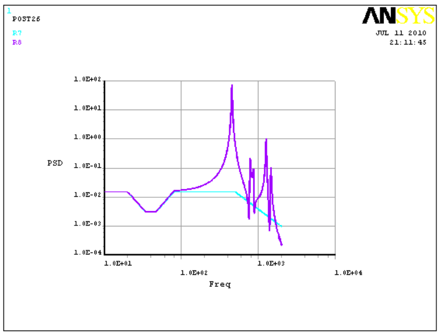
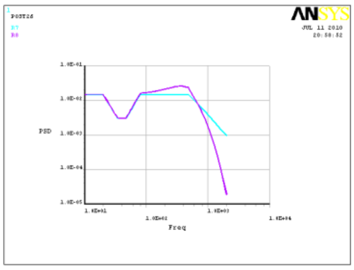


Fig- 16 Stress Due To Vibration

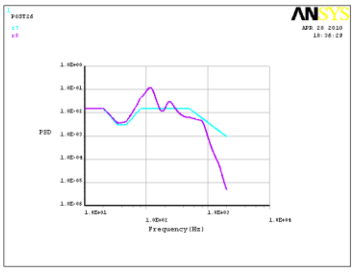
7.2 PSD Response:



Graph-6 Graph for 5% damping



Graph-7 Graph for 10% damping



Graph-8 Graph for 15% damping

The improvement analysis had been carried out in order to increase the performance of Shock Mount. From the above figures we can see that the stresses can be reduced by increasing the damping ability of the elastomer. Fig – 15 to Fig-16 denotes the gradual reduction in the stress value according to the increase in damping value. The graph plotted for percentage damping Vs stress generated tells us clearly about the exact value of damping and the stress. The elastomer with damping ability of 10% is feasible and can be used in the shockmount as the stress value is reduced from 248MPa to 196MPa. Elastomer with more than 10% damping ability can also be used but is not necessary.

VIII. RESULTS AND DICUSSIONS

8.1 calculation of natural frequency:

We know that;

$(EV)_{total} = E_1V_1+E_2V_2+E_3V_3 + \ldots$ ---------------- (1)

$(EV)_{\text{of Shock Mount}} = (E_{steel} \times V_{steel}) + (E_{elastomer} \times V_{elastomer})$

$V_{steel} = 0.207960 \times 10^{-4}$

$V_{elastomer} = 0.333591 \times 10^{-4}$

$E_{steel} = 2 \times 10^{11}$

$E_{elastomer} = 2 \times 10^{9}$

$V_{total} = V_{steel} + V_{elastomer}$

$= 0.207960 \times 10^{-4} + 0.333591 \times 10^{-4}$

$= 0.541551 \times 10^{-4}$

$$E_{total} = \frac{(E_{steel} \times V_{steel}) + (E_{elastomer} \times V_{elastomer})}{V_{total}}$$

$= 77.881 GPa$

$WEIGHT_{total} = (V_{steel} \times \rho_{steel}) + (V_{elastomer} \times \rho_{elastomer})$ --- (2)

$= (0.207960 \times 10^{-4} \times 7800) + (0.333591 \times 10^{-4} \times 1600)$

$= 0.21558 \times 9.81$

$= 2.114872$ N

Where;

ρ=Density

UDL= Weight/Length ----------- (3)

$$= \frac{2.114872}{60 \times 10^{-3}} = 35.24787 \text{ N/m}$$

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Also;

Where;

Deflection $\delta = \frac{WL^4}{8EI}$ -------------- (4)

$I = \frac{\pi D^4}{64}$ ------------ (5) $= 0.51823 \times 10^{-6}\, m^4$

Substituting back in equation (4)

$\delta = \frac{WL^4}{8EI} = 8.4755 \times 10^{-11} m$

And to find natural frequency, we have;

$f_n = (1/2\pi) \times [g/\delta]^{1/2}$

$= (1/2\pi) \times [9.81/8.4755 \times 10^{-11}]^{1/2}$

= 2324 rad/sec.

8.2 Calculation of static analysis

W.K.T

From the hooke's law;

E $= \frac{\sigma}{\epsilon}$ -------- (1) also;

Stress $(\sigma) = \frac{\text{Load (p)}}{\text{Area(a)}}$ ----- (2) $= \frac{100 \times 9.81}{2.5^2}$ =156.96MPa

Deformation (dl) $= \frac{PL}{AE}$ ----------- (3) =1.962 mm.

From the above calculations we can observe that the theoretical static stress results do not exceed the yield strength of the material i.e. 250MPa. Hence the frame structure is safe under static loading.

8.3 Comparision of the results

SL No	ANALYSIS	THEROTICAL	PERCENTAGE ERROR
01	2113 rad/sec	2324/sec	9%

IX. COMPARISION

From the graph it is clear that the shockmount SM-1 up to SM-4 exceed the yield value of the material hence these shock mounts need to be redesigned. The shockmount with 10% damping value as it is already been redesigned is within the limits of the yield value and hence can be implemented successfully.

TABLE-IX COMPARISON OF THE STRESSES

SL NO	SHOCKMOUNT TYPE	STRESS VALUE (MPa)	CONCLUSION
01	5% DAMPING	248	NOT ACCEPTALBE
02	10% DAMPING	192	ACCEPTABLE
03	15% DAMPING	142	ACCEPTABLE
04	DESIGN-1	300	NOT ACCEPTABLE
05	DESIGN-2	280	NOT ACCEPTABLE
06	DESIGN-3	305	NOT ACCEPTABLE
07	DESIGN-4	325	NOT ACCEPTABLE

X. FUTURE SCOPE

10.1 Design-1 Dual cup shockmount:

The Dual Cup Shockmount is a product, which consists of two cylindrical steel cups one fitted into the other aligned concentrically, two elastomers also fitted one inside the other but are separated with a steel cup. All these components are held together with the help of a fastener between the two washers.

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XI. CONCLUSION

1. The problem identified was, vibrations generated in the aircraft were not damped successfully, which further lead in the damage of CPU unit.
2. Literature survey and research surveys disclosed that majority of the vibrations can be damped successfully by proper design of the shockmount.3. Referring the 2-D drawing the solid modeling and FEA modeling using CATIA-V5 and Hypermesh-9 respectively was completed.
3. The analysis result of shockmount, for the present conditions confirmed that the vibrations generated are not damped successfully.
4. Theoretical calculations also justified the analysis values.
5. Accepting the various factors vibrations are reduced step by step by redesigning the damping material suitably.
6. Now the stresses generated due to vibrations were not exceeding the yield value and hence the CPU unit was in safe condition.

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