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Replacement of two wheeler suspension system with compliant mechanism

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ABSTRACT

In this paper, an attempt is made to replace the existing spring-damper suspension system of a two-wheeler by the compliant mechanism. Suspension details of Hero Igniter bike were obtained and analysis of spring was made for static and total deflection. The initial topology of compliant mechanism was obtained through topology optimization in ANSYS. Final topology along with the shape and size of members of compliant mechanism were obtained through trial and error method. Modal and harmonic analysis of obtained compliant mechanism was done to get frequency response curve which subsequently used for calculation of equivalent damping coefficient of the compliant mechanism. Using the equation of force transmissibility, the force transmitted at various frequency ratios was calculated with the help of equivalent damping coefficient obtained for the compliant mechanism. These values of force transmission were then compared with the corresponding values of the spring-damper suspension system. The force transmission values of compliant mechanism were found to be less than the spring damper suspension system leading to the conclusion that the existing suspension system can well be replaced with the complaint mechanism.

Keywords: *Compliant Mechanism, Topology Optimization, Modal Analysis, Harmonic Analysis, Force Transmissibility.*

1. INTRODUCTION

The suspension system is the main part of the bike, where the shock absorber is designed mechanically to handle shock impulse and dissipate kinetic energy. In a vehicle, shock absorbers reduce the effect of traveling over rough ground, leading to improved ride quality and vehicle handling [1]. In this project, efforts are made to check the possibility of replacing the existing spring-damper suspension system with the compliant mechanism. A compliant mechanism is the kind of mechanism which performs its function of force and power transmission through elastic deflection of its members, unlike rigid-body mechanism where in the transmission of required force and power occurs through rotation of links which are connected by hinges. If something bends to do what it is meant to do, then it is compliant. If the flexibility that allows it to bend also helps it to accomplish something useful, then it is a compliant mechanism [2]. Compliant mechanism synthesis method such as topology optimization is used to get the required topology of the isolator. Software such as ANSYS helped in carrying out topology optimization. Size and shape optimization of mechanism has been done through trial and error method to get the required performance characteristics. Subsequent modal and harmonic analysis of the mechanism in ANSYS has provided displacement amplitude at various excitation frequencies which in turn is used to calculate force transmissibility of the isolator or mechanism.

2. DETAILS OF EXISTING SUSPENSION SYSTEM AND ANALYSIS

The spring damper suspension system of Hero Igniter bike is considered as a reference to compare the performance of compliant suspension system. Details of suspension spring and damper of the vehicle were obtained and are as below:

a) No. of Turns of spring: 15
b) Wire diameter: 8 mm
c) Outer Coil diameter: 54 mm
d) Mean Coil diameter: 46 mm
e) The total length of spring: 220 mm
f) Damping coeff. Of damper: 550 N-s/m

Calculation of load acting on spring is done considering weight distribution of bike on front and rear sides as below:

- Gross vehicle weight (obtained from vehicle manual) : 129 kg (Front side weight = 52 kg, Rear side weight = 77 kg)

Hence rear side weight equals 77 kg which is distributed equally on both springs. So each spring carries **39 kg** weight.

- Number of Passengers on the bike : 3
  (Ideally, it should be 2 but considered 3 as worst loading condition)

Assuming average weight of person as 80 kg, total weight of passenger = 80 × 3 = 240 kg.
Using weight distribution criteria of 60:40 along rear and front side respectively as used in gross weight distribution,
Total passenger weight on rear side =
240 × 0.6 = 144 kg

Hence, passenger weight on each spring = 144/2 = **72 kg**.

Total load / weight on spring in static condition = 39 + 72 = 111 kg = 111 ×9.81 = **1100 N**

Solid modeling of spring was done using Solid Works and subsequent finite element analysis is carried out in Ansys to check the deflection of spring under static condition i.e. static deflection and stresses generated in spring.

Figure-1. Spring Solid Model

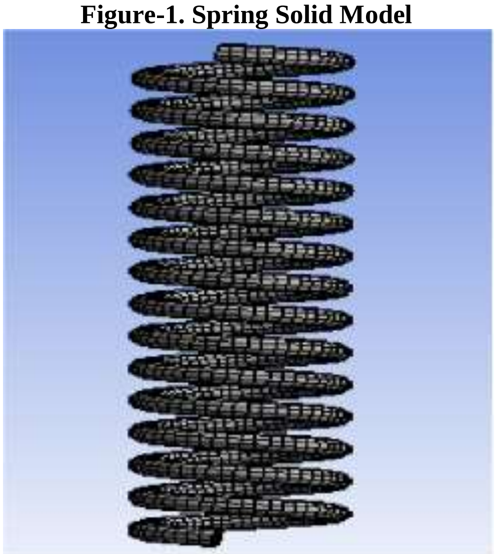

Figure-2. Spring FE model

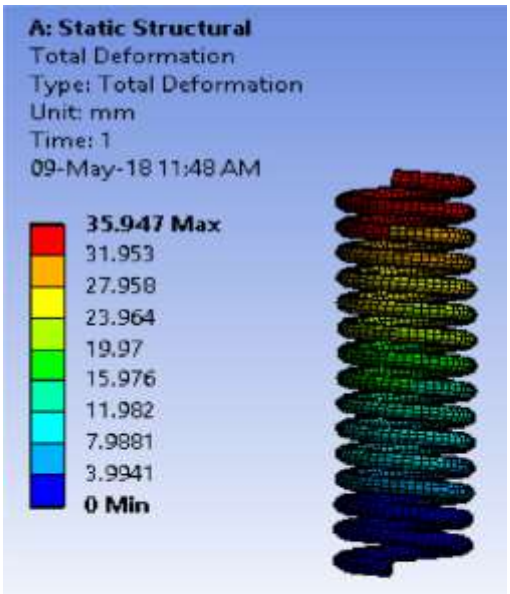

Figure-3: Static Deflection of Spring

Thus the finite element analysis of spring shows that the static deflection of spring is **35.95mm**.
Usually, bike experiences an acceleration of half of 'g' when it travels over a bump on the road [3]. This acceleration adds to the load acting on the suspension spring.
Harmonic load acting on suspension due the bump = mass acting on spring × accn due to bump = 111 × 0.5× 9.81 = 544.45 = **550 N (rounding)**

Hence the total deflection of spring when the vehicle travels over the bump can be calculated with a total load which is simply the addition of static and harmonic loads. So, Total load considering harmonic load is given by, W= 1100+ 550 = **1650 N**

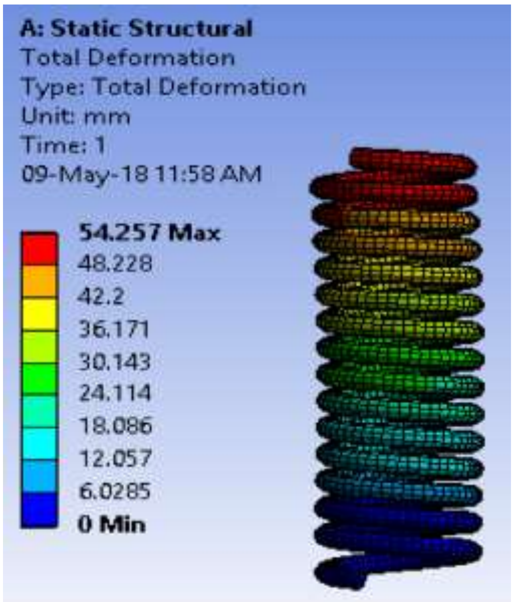

Figure-4: Total Deflection of Spring

Hence total deflection of spring comes out to be **54.26 mm**.

3. TOPOLOGY SYNTHESIS AND SIZE & SHAPE OPTIMIZATION OF COMPLIANT MECHANISM

Design of compliant mechanism consists of two steps:

1) Topology synthesis—which involves generation of a functional design in the form of a feasible topology starting from input/output force/motion specifications, and

2) Size and shape optimization—to meet performance requirements such as maximum stress, motion amplification or force amplification etc [4]

Topology is defined as the pattern of connectivity or spatial sequence of members or elements in a structure. The allowable space for the design in a topology optimization problem is called the design domain. The topology is defined by the distribution of material and void within the design domain [2]. The design domain for topology synthesis is selected as a rectangle with thickness 10 mm as shown in Figure 5. This is in accordance with the space available for spring damper system in the bike.

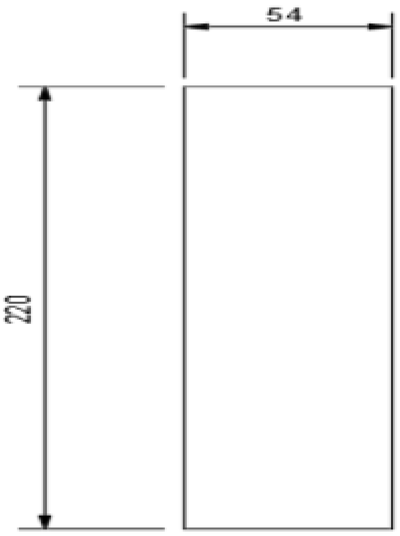

Figure-5. Design Domain for Topology Synthesis

Topology synthesis is carried out in Ansys 18.2 using topology optimization tool with optimization problem as defined below:

Minimize (Mass of Topology)

Subject to

$[K]\{d\} = \{f\}$

Global Von-Mises stress < 250 MPa

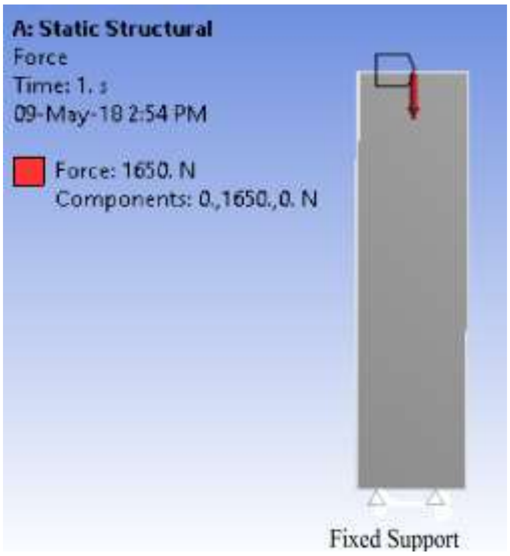

Figure-6. Boundary Conditions for topology optimization

Figure 6 represents the design domain with boundary conditions where lower end of the plate is fixed and force of 1650 N is applied at the other end. Resulting topology after optimization run in Ansys is shown in Figure 7. It seems the topology obtained will not serve our purpose. Hence some modifications in the design domain were needed to check the possibility of getting better topology.

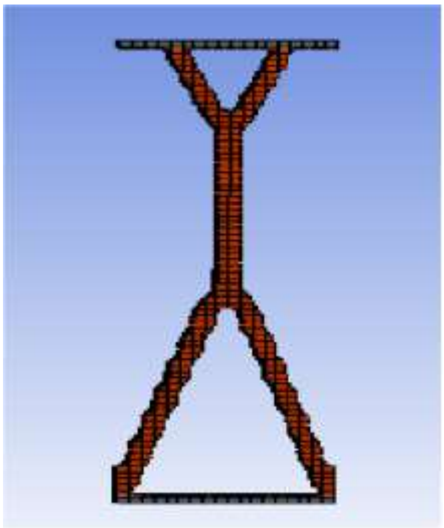

Figure-7. Topology Obtained after optimization

Rectangular plate with elliptical holes in it tried for better topology and topology obtained is as shown in figure 9.

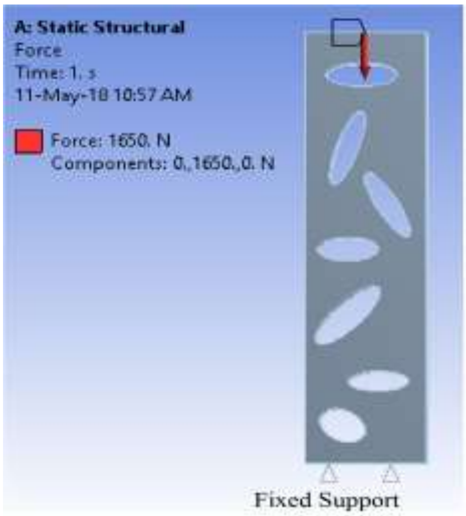

Figure-8. Boundary Conditions for topology optimization of rectangular design domain with elliptical holes

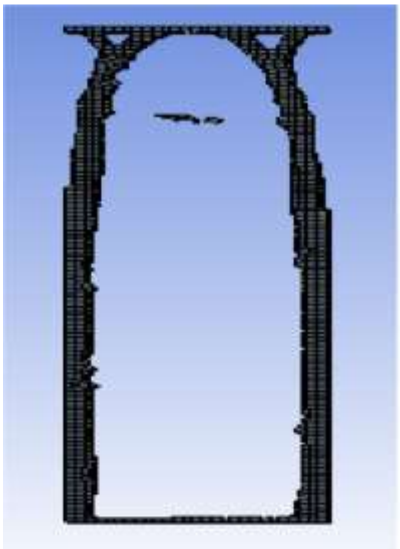

Figure-9. Final Topology Obtained after optimization

Above obtained topology is then selected for further process.After getting topology for the mechanism next step is to determine size and shape of its members. This is done by trial and error method.

3.1 Trial 1: In this step, the very topology is used for static deflection analysis which is obtained after optimization. Width of the topology is considered as 20 mm and member thickness as 1.5mm to begin with. The results are as below in Figure 10 & Figure 11:

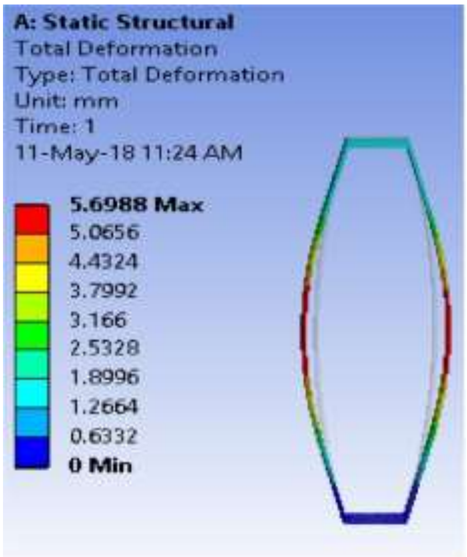

Figure 10. Static deflection for Trial 1

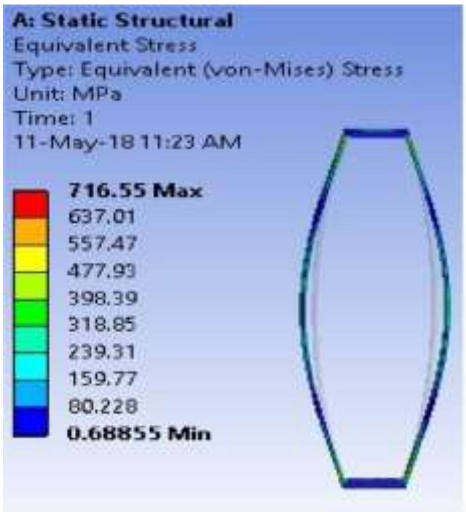

Figure-11. Von-Mises Stress for Trial 1

Above value of Static deflection is very less when compared with the static deflection of spring although Von-Mises stress value is within limits. So we need to proceed with some modification in the topology.

3.2 Trial 2: In this trial two loops are considered instead of one as done in previous step and corresponding values of static deflection and stress are checked.

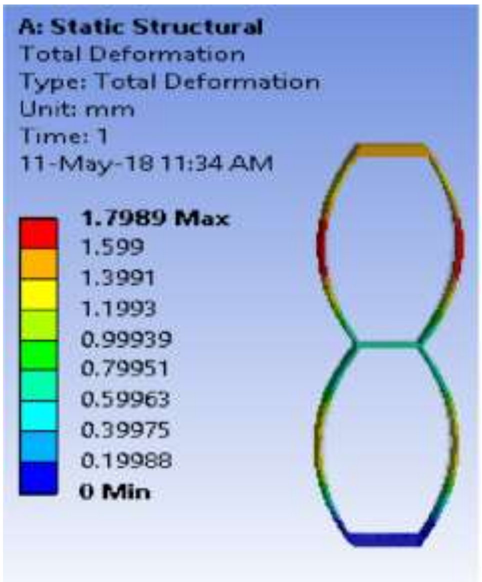

Figure-12. Static deflection for Trial 2

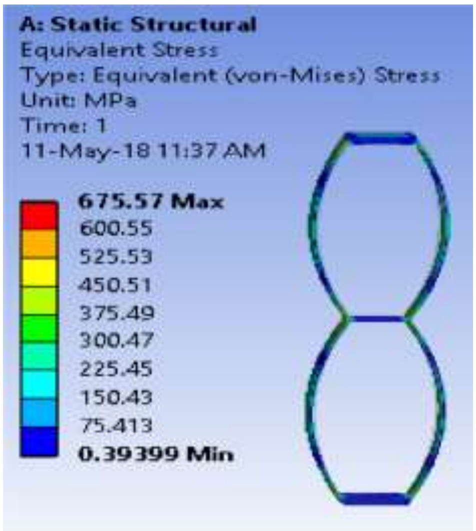

Figure-13. Von-Mises Stress for Trial 2

As can be seen from Figure 12, the deflection value has drastically reduced when compared with initial topology though Von Mises stress has almost remained the same. Hence we moved to next modification or trial in topology.

3.3 Trial 3: In this topology modification, number of loops are increased to four with slight modification in arcs present in each loops. Instead of 'U' shape, 'V' arcs are employed. Deflection and stress results are as below:

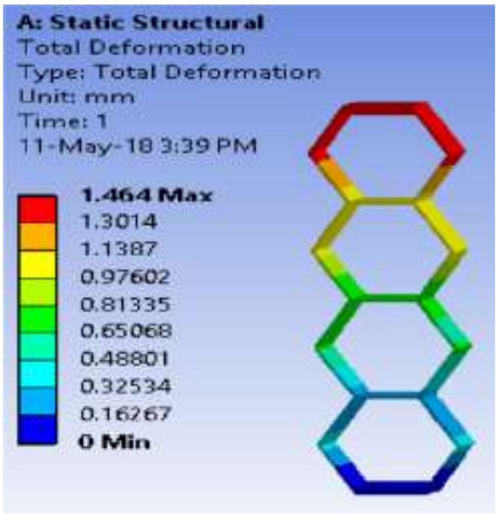

Figure-14. Static deflection for Trial 3

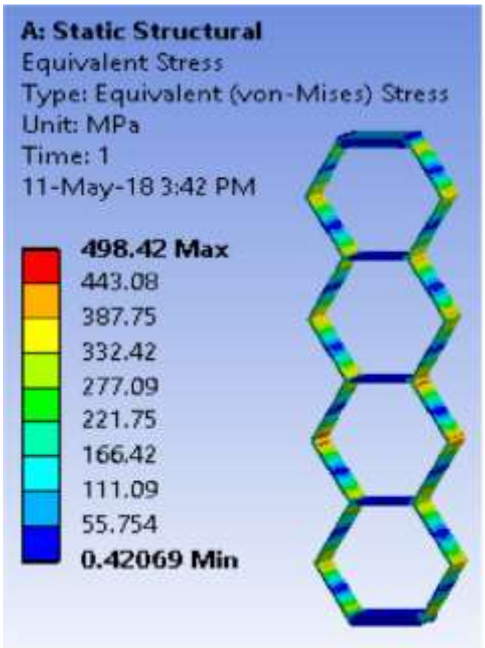

Figure-15. Von-Mises Stress for Trial 3

It seems the idea of only increasing loops only is not working. We will try with loop increment coupled with thickness reduction of members. Till now we were using 1.5mm thickness members. We will reduce the thickness to 1mm in the next modification.

3.4 Trial 4: Since increasing loops only isn't helping with the required deflection, 8 loops are considered in this trial along with thickness reduction of its members to 1mm. Width of topology considered is 20 mm as was used in previous all cases or trials. Since the coil diameter of spring is 55mm, we have liberty to choose the width of topology up to 55mm which is as per the maximum space available with us for the replacement. Hence we started with the 20mm width to begin with. The results obtained are shown in Figure 16 & Figure 17.

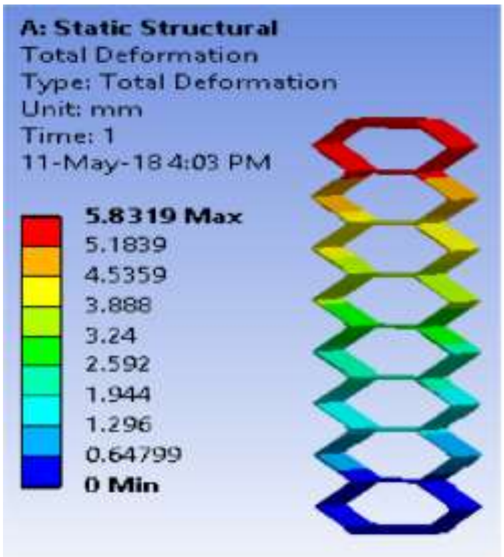

Figure-16. Static deflection for Trial 4

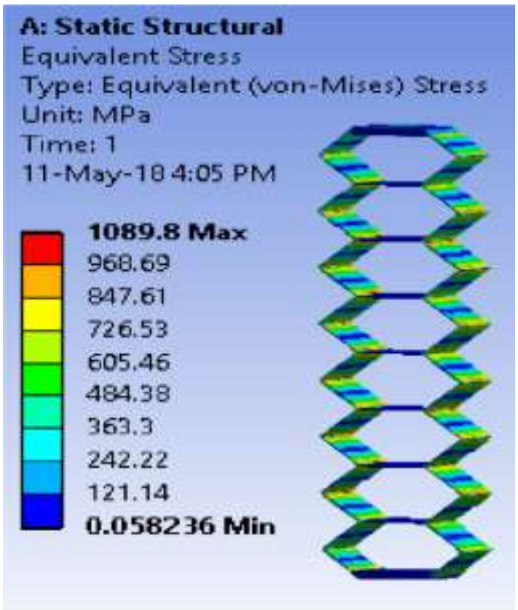

Figure-17. Von-Mises Stress for Trial 4

The value of static deflection has increased slightly compared with trial 3. Stress Value has also increased by considerable value. It is evident that this topology is not yielding the required results. Hence we will do slight modification in the topology itself. In the next modification, we will use arc-shaped members with 16 loops.

3.5 Trial 5*:* This modification was tried with 16 loops and with 1mm thickness of its members. Here in this trial arc-shaped members were tried as stated above. Thickness of the members was kept 1mm. These results are presented below in Figure 18 & Figure 19.

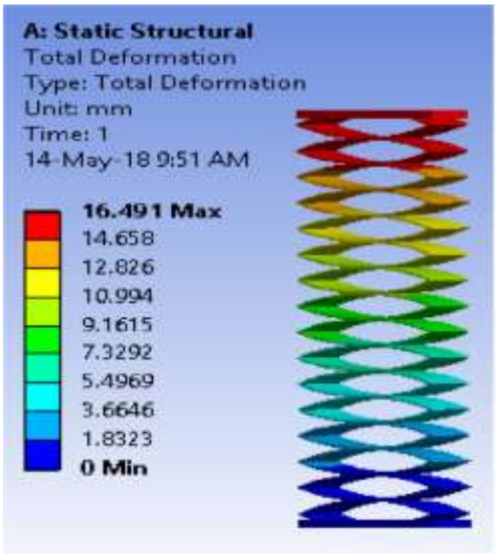

Figure-18. Static deflection for Trial 5

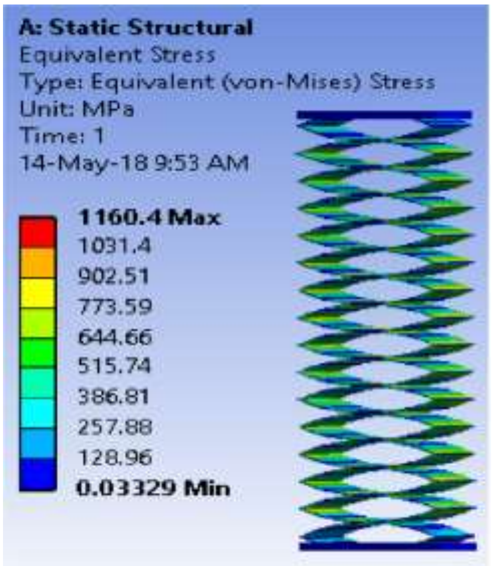

Figure-19. Von-Mises Stress for Trial 5

As evident from the results above, deflection value has seen a great increase from previous value of 5.8mm to 16.49mm with this new topology modification. Stress value though has increased somewhat which needs to be taken care of. So new trial with 20 loops was carried out.

3.6 Trial 6: Here in this trial topology with 20 loops was used. In order to get more deflection value, members with 0.8mm thickness were tried. Topologies with widths of 30mm, 40mm, 45mm, & 50mm were analysed and results for topology with member thickness 0.8mm having width 50mm was finalized. The results obtained are as shown in Figure 20 & Figure 21.

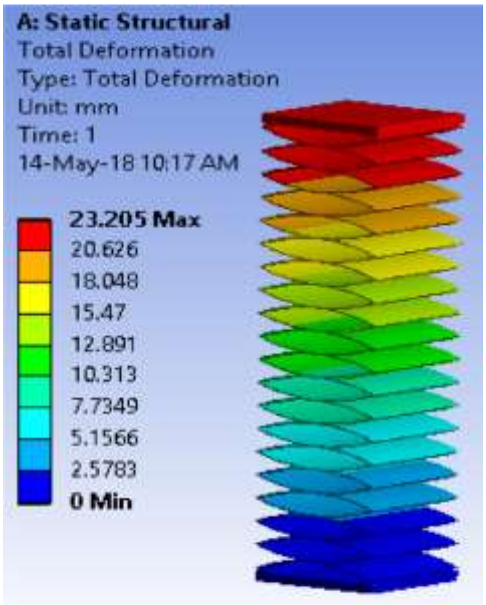

Figure-20. Static deflection for Trial 6

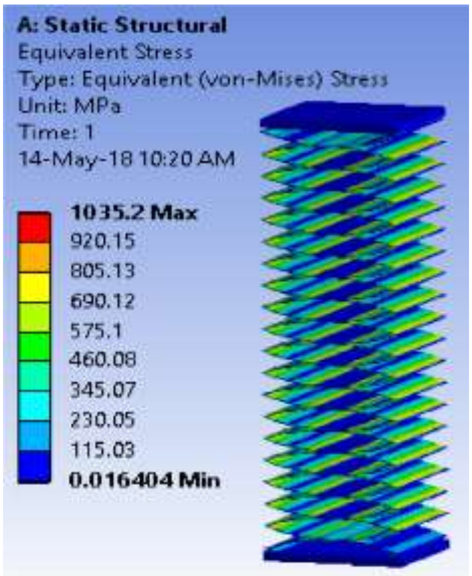

Figure-21. Von-Mises Stress for Trial 6

Hence static deflection comes out to be 23.205 mm whereas Von-Mises stress value is 1035MPa which is within acceptable limits for the material considered i.e. 67SiCr5 (DIN 17221 Spring Steel Grade)

Youngs Modulus E	210 GPa
Ultimate Strength	1700 MPa
Yield Strength	1100 MPa

Total Deflection analysis of the geometry of topology showed total deflection of **34.81 mm**.

4. MODAL, HARMONIC RESPONSE ANALYSIS & FORCE TRANSMISSIBILITY CALCULATIONS

Finalized geometry or topology is then subjected to modal and harmonic response analysis. The harmonic response analysis solves the time-dependent equations of motion for linear structures undergoing steady-state vibration. All loads and displacements vary sinusoidally at the same known frequency [5]. Harmonic force considered here is **550N** as calculated earlier in the paper. Frequency response obtained as a result of the harmonic analysis is as below:

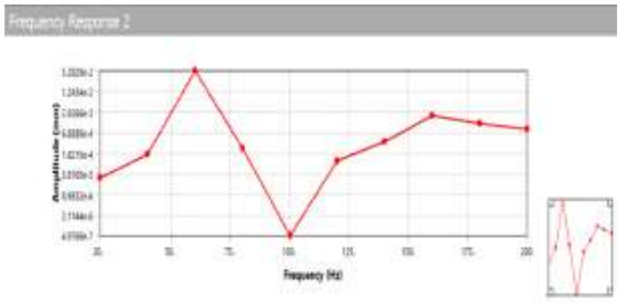

Figure-22. Frequency Response Curve

The next step is to determine equivalent viscous damping from above frequency response curve which will be required for calculation of force transmissibility.

We have, $\frac{X_p}{X_{st}} = \frac{1}{2\zeta}$ .....[A] where,

Xp = Peak amplitude

Xst = Static deflection

ζ = Damping factor

& $\frac{X}{X_{st}} = \frac{1}{\sqrt{[1-({}^{\omega}/_{\omega_n})^2\ ]^2 + [2\zeta({}^{\omega}/_{\omega_n})]^2}}$ ....[B]

We will draw a horizontal line at 0.707Xp cutting response curve at two points, corresponding value at abscissa being ω_1 & ω_2 [6]. Here Xp = 18.57 µm. So, 0.707Xp= 13.13mm. Hence ω_1 & ω_2 are 53 and 66 Hz respectively. ω_p is the peak frequency on frequency response curve. Now we can write,

$$\frac{0.707\mathrm{Xp}}{\mathrm{X_{st}}} = \frac{1}{\sqrt{[1-({}^{\omega}/_{\omega_p})^2\ ]^2 + [2\zeta({}^{\omega}/_{\omega_p})]^2}}$$

From A & B,

$$\frac{0.707}{2\zeta} = \frac{1}{\sqrt{[1-({}^{\omega}/_{\omega_p})^2\ ]^2 + [2\zeta({}^{\omega}/_{\omega_p})]^2}}$$

Finally above equation yields, $\frac{\omega_2-\omega_1}{\omega_p} = 2\zeta$

Putting values in above equation, ζ **= 0.1068**

Also, $\zeta = \frac{C_{eq}}{C_{cc}} = \frac{C_{eq}}{2\sqrt{km}}$

Ceq = 0.1068 × 2$\sqrt{\frac{1100\times1000}{23.21}}$ × 110 = **488 Ns/m**

The equation for force transmissibility is,

$\frac{F_{TR}}{F_0} = \frac{\sqrt{1+[2\zeta({}^{\omega}/_{\omega_n})]^2}}{\sqrt{[1-(\frac{\omega}{\omega_n})^2]^2+[2\zeta({}^{\omega}/_{\omega_n})]^2}}$ ....[C]

Using this equation, the force transmitted F_{TR} is calculated for both Spring damper system and compliant mechanism and the same is tabulated as shown below:

Table 1. Force Transmission at various Frequency Ratios

Sr No	Frequency Ratio	Force Transmitted	
		Spring N	Compliant Mechanism N
1	1.5	454	488
2	2.0	210	197
3	2.5	130	118
4	3.0	92	81
5	3.5	71	61
6	4.0	57	48
7	4.5	48	40
8	5.0	41	34

The plot of force transmitted verses frequency ratio is then plotted for spring as well as a compliant mechanism as shown below:

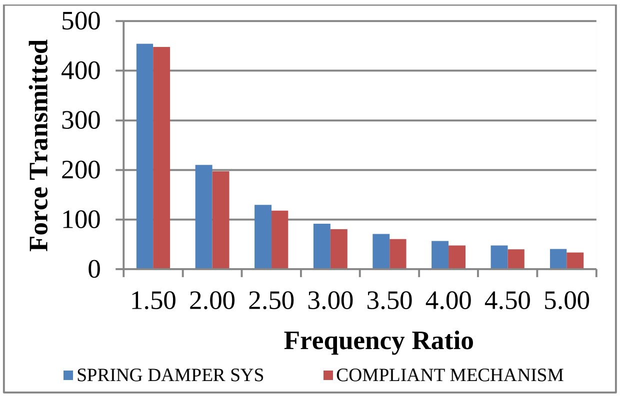

Figure-23. The plot of Force Transmitted Vs. Frequency Ratio

Isolation Efficiency η in percent transmission is related to Transmissibility as
$\eta = 100\ (1\text{-}Tr)\ \%$ where Transmissibility Ratio Tr= (Force transmitted in N / Disturbing force in N)[5].

The plot of isolation efficiency is as below:

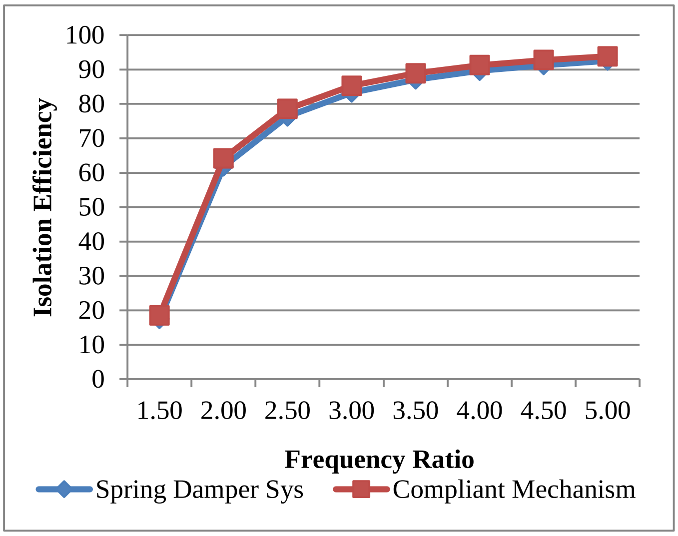

Figure-24. The plot of Isolation efficiency Vs. Frequency Ratio

5. CONCLUSION

In this study, an attempt was made to find a replacement for the existing spring-damper suspension system with the compliant mechanism. Topology optimization method was used for the synthesis of the compliant mechanism. Subsequent modal and harmonic analysis was performed to measure the force transmissibility of compliant mechanism at various frequency ratios and the same then was compared with force transmissibility of spring damper system. The values of force transmission for compliant mechanism were almost the same rather somewhat less when compared with spring damper suspension system as can be seen from figure-23. The plot of isolation efficiency Vs. frequency ratios give a clear idea about the efficiency of compliant mechanism as passive vibration isolation system. Transmission of forces reduced and isolation efficiency increased as the frequency ratio increased. Thus the above study shows that the compliant mechanism can provide an effective vibration isolation from sinusoidal disturbances with known frequency ratios.

6. REFERENCES

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