ISSN: 2456-6616

# Replacement of Vehicle Suspension System with Compliant Mechanism

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**Abstract:**
This paper makes an attempt to replace the existing spring damper suspension system of a vehicle by compliant mechanism. Suspension details of TATA Sumo vehicle were obtained, and analysis of spring was made for static and total deflection. 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 compliant mechanism. Using the equation of force transmissibility, force transmitted at various frequency ratios was calculated with the help of equivalent damping coefficient obtained for compliant mechanism. These obtained values of force transmission were then compared with the corresponding values of spring damper suspension system. the force transmission values of compliant mechanism were found to be less than the spring damper suspension system leading to conclusion that the existing suspension system can well be replaced with the complaint mechanism.

**Keywords:** Compliant Mechanism, topology optimization, modal analysis, harmonic analysis

## 1. Introduction

The suspension system is the main part of the vehicle, 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 compliant mechanism. 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 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

Spring damper suspension system of TATA Sumo vehicle is considered as reference to compare the performance of compliant suspension system. Details of suspension spring and damper of the vehicle were obtained and are as below:

1. No. of Turns of spring: 7
2. Wire diameter: 15 mm
3. Coil diameter: 110 mm
4. Total length of Spring: 250 mm
5. Damping coeff. of damper: 2000 N-s/m

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

- Gross vehicle weight (obtained from vehicle manual): 2540 kg (Front axle weight FAW= 1000 kg, Rear axle weight RAW = 1540 kg)

Hence front axle weight equals 1000 kg which is distributed equally on both springs. so, each spring carries **500 kg** weight.

- Seating capacity of vehicle: 9 persons
  Assuming average weight of person as 70 kg, total weight of passenger = 70 × 9 = 630 kg.

Using weight distribution criteria of 60:40 along rear axle and front axle respectively as used in gross weight distribution,

Total passenger weight on front axle = 630 × 0.4 = 252 kg
Hence, passenger weight on each spring = 252/2 = **126 kg**.
Total load / weight on spring in static condition

= 500 + 126 = 626 kg = 626 ×10 = **6260 N**

Solid modeling of spring has 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.

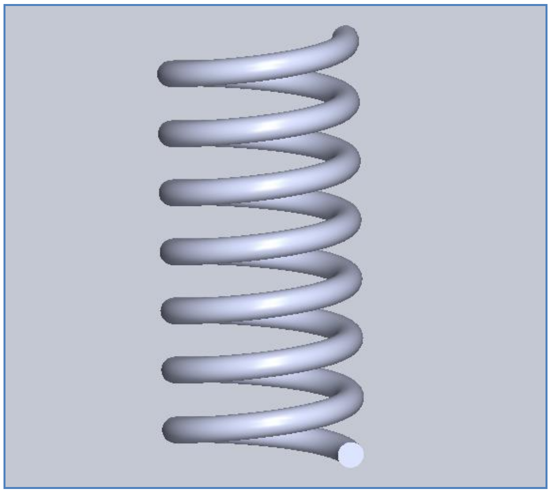

Fig. 1. Spring Solid Model

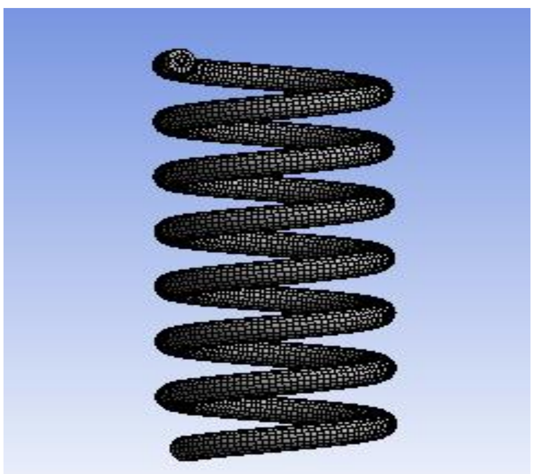

Fig. 2. Spring FE model

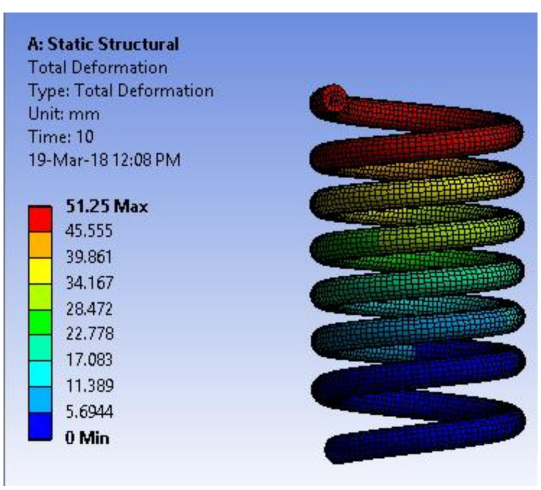


Fig. 3. Static Deflection of Spring

Thus, the finite element analysis of spring shows that the static deflection of spring is **51.25mm**.

Usually vehicle experiences acceleration of half of 'g' when it travels over bump on the road [3]. This acceleration adds to the load acting on the suspension spring. Harmonic load acting on load due the bump = mass acting on spring × accn due to bump = 626 × 0.5× 10 = **3130 N.**

Hence the total deflection of spring when vehicle travels over bump can be calculated with total load which is simply addition of static and harmonic loads. So, Total load considering harmonic load is given by,

$$W = 6260 + 3130 = \mathbf{9390\ N}$$

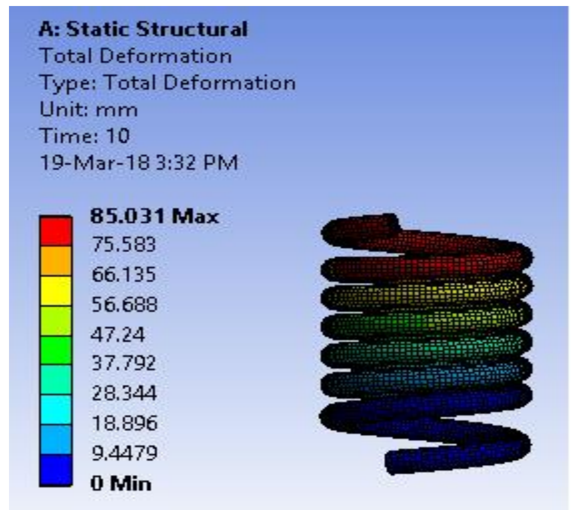


Fig. 4. Total Deflection of Spring.

Hence total deflection of spring comes out to be **85 mm**.

## 3. Topology Synthesis, Size and Shape Optimization of Compliant Mechanism

Design of compliant mechanism consist 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 rectangle with thickness 10 mm as shown in Figure 5.

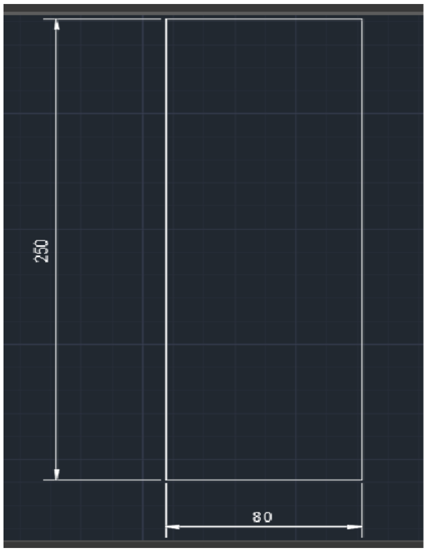


Fig. 5. Design Domain for Topology Synthesis.

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

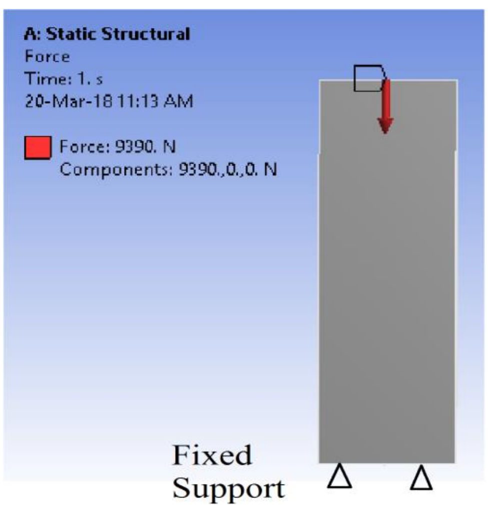


Fig. 6. Boundary Conditions for topology optimization.

Figure 6 represents the design domain with boundary conditions where lower end of the plate is fixed and force of 9390N is applied at the other end. Resulting topology after optimization run in Ansys is shown in Figure 7. Better topology can be found out with some modifications in the design domain.

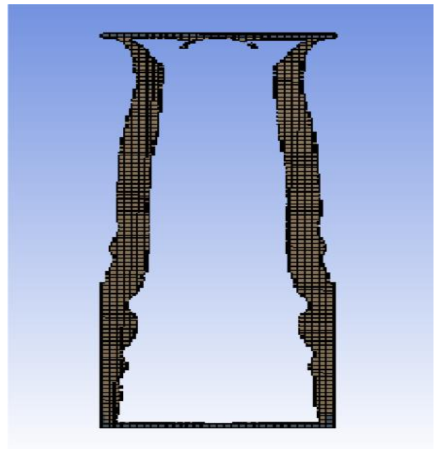

Fig. 7. Topology Obtained after optimization

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

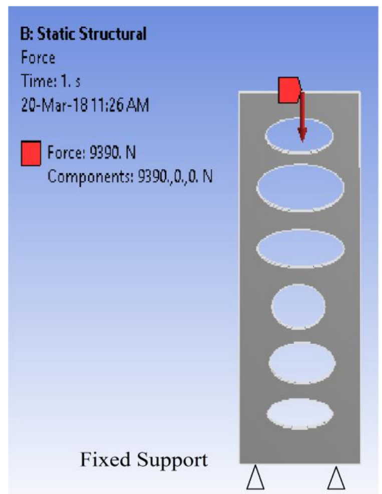


Fig. 8. Boundary Conditions for topology optimization of rectangular design domain with elliptical holes.

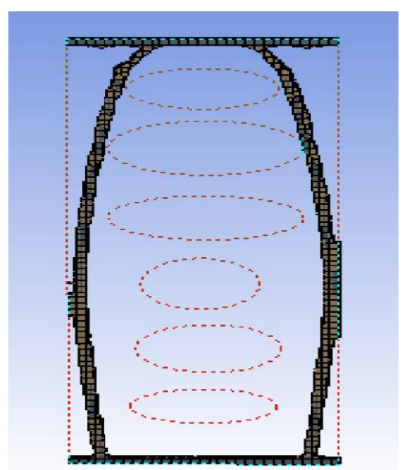

Fig. 9. Final Topology Obtained after optimization

 

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.

1) *Trial 1:* In this step, the very topology is used for static deflection analysis which is obtained after optimization. the results are as below in Figure 10 & Figure 11:

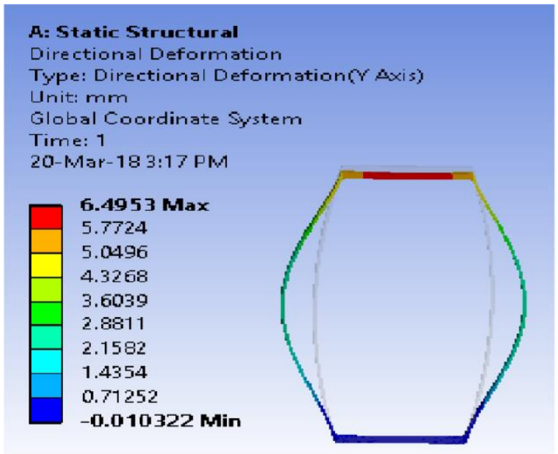


Fig. 10. Static deflection for Trial 1

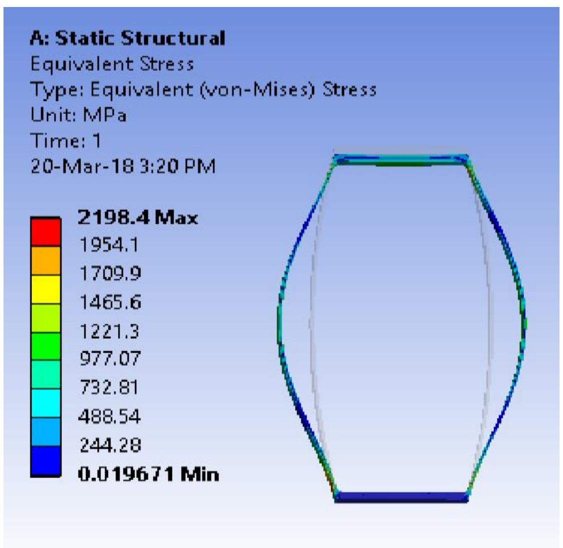


Fig. 11. Von-Mises Stress for Trial 1

Above value of Static deflection is very less when compared with the static deflection of spring and Von-Mises stress value is also very high. so, we need to proceed with some modification in the topology.

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.

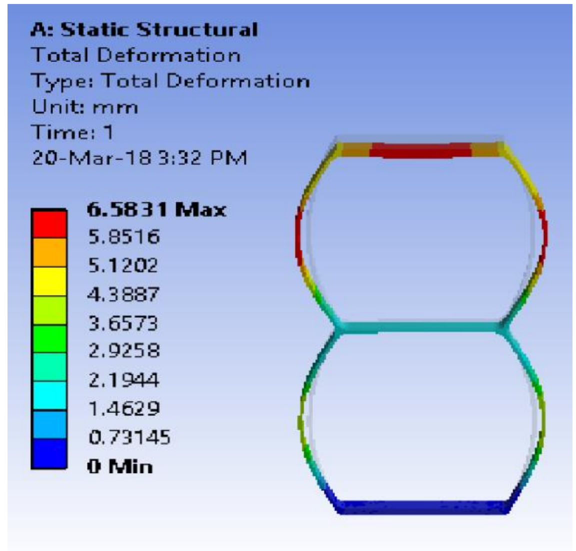


Fig. 12. Static deflection for Trial 2

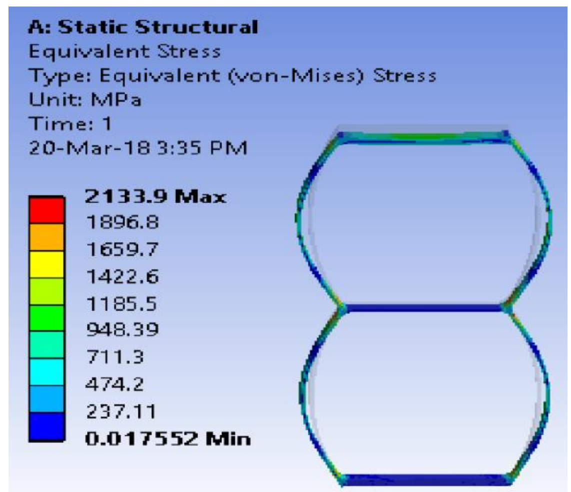


Fig. 13. Von-Mises Stress for Trial 2

Looking at the static deflection and Von-Mises stress values, it is clear that there is no significant change in the values of both when compared with initial topology. The only visible difference is in the lateral deflection of its members as can be clearly seen in the above figures. So, we will move to next modification or trial in topology.

3) *Trial 3:* In this topology modification, number of loops are increased to four with slight modification in arcs present in each loop. Instead of ‘U' shape, 'V' arcs are employed.

As can be seen in deflection figure, deflection has increased to great extent. However, stress has also increased to some extent. It seems increasing loops are helping but stress is something which needs to be addressed. We will move to next modification.

Deflection and stress results are as below:

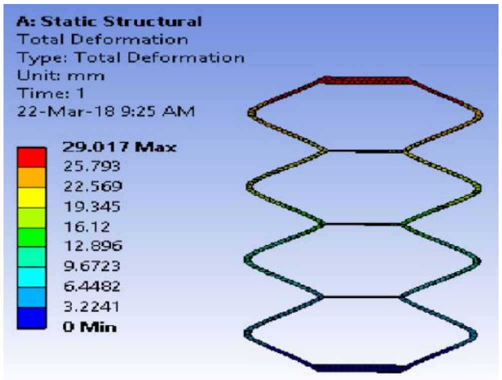


Fig. 14. Static deflection for Trial 3

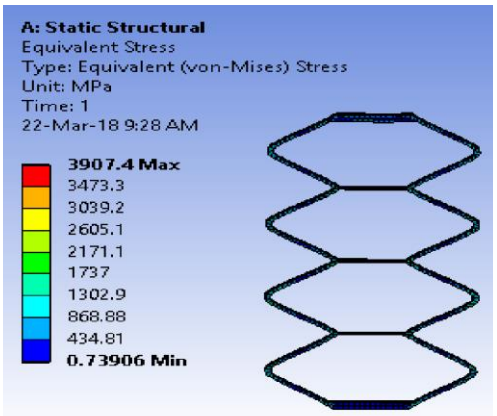


Fig. 15. Von-Mises Stress for Trial 3

4) *Trial 4:* Since increasing loops are helping with the required deflection, 8 loops are considered in this trial. Width of topology considered is 20 mm as was used in previous all cases or trials. Since the coil diameter of spring is 110mm, we have liberty to choose the width of topology up to 110mm which is as per the maximum space available to us for the replacement. hence, we started with the 20mm width to begin with. The results obtained are shown in Figure 16 & Figure 17.

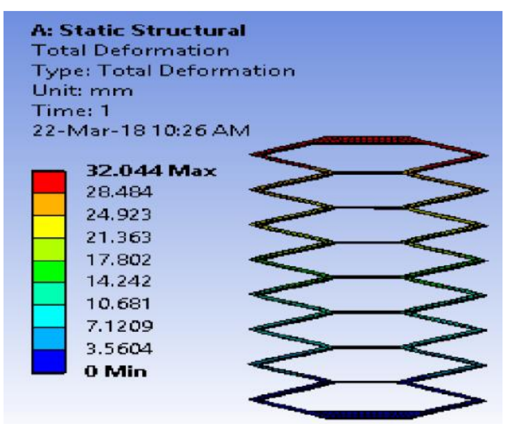


Fig. 16. Static deflection for Trial 4

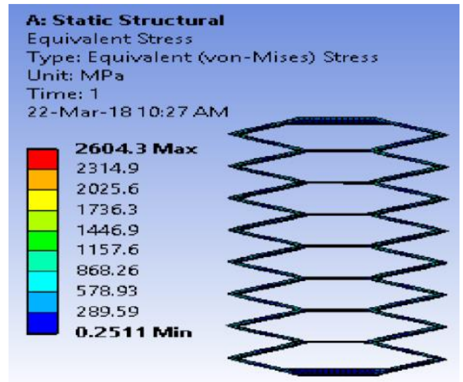


Fig. 17. Von-Mises Stress for Trial 4

The value of static deflection has increased slightly compared with trial 3. Stress Value has reduced by considerable value. The next modification will have 16 loops.

5) *Trial 5:* This modification was tried with 16 loops and with 1.5mm thickness of its members while varying widths like 40mm, 60mm, 70mm, 80mm, 90mm, 100mm and 110mm. However, it is found that 110mm width gives result where stress value is within limit and deflection is also considerable. These results are presented below:

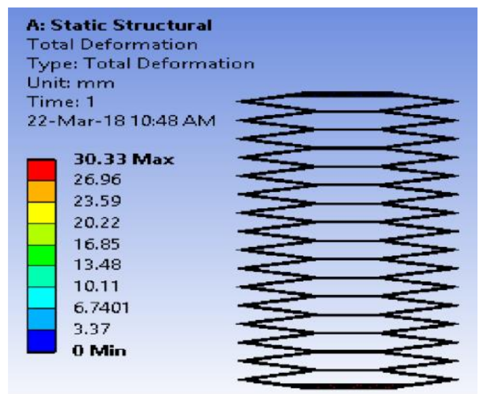


Fig. 18. Static deflection for Trial 5

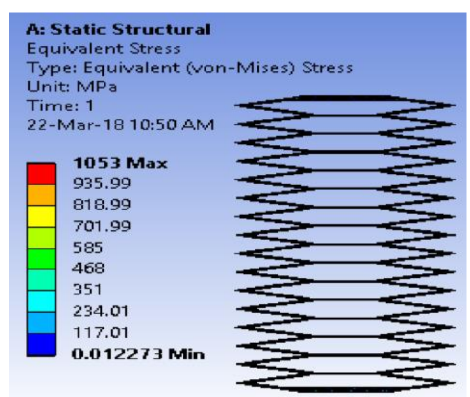


Fig. 19. Von-Mises Stress for Trial 5

Hence static deflection comes out to be 30.33 mm whereas Von-Mises stress value is 1053MPa 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 **45.49 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 **3130N** as calculated earlier in the paper. Frequency response obtained as a result of harmonic analysis is as below:

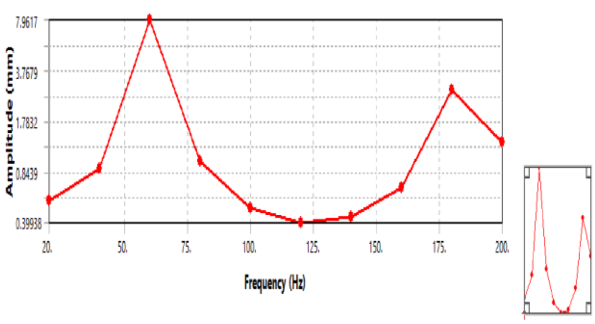


Fig. 20. Frequency Response Curve

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

$\zeta$ = Damping factor

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

We will draw a horizontal line at 0.707Xp cutting response curve at two points, corresponding value at abscissa being $\omega_1$ & $\omega_2$ [6]. Here Xp = 7.9617 mm. So, 0.707Xp= 5.63mm. hence $\omega_1$ & $\omega_2$ are 55 and 65 Hz respectively. $\omega_p$ is the peak frequency on frequency response curve. Now we can write,

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

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, $\zeta$ = **0.083**

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

Ceq = $0.083 \times 2\sqrt{\frac{6260\times1000}{30.33}\times 626}$ =**1886 Ns/m**

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, 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 | 2582 | 2529 |
| 2 | 2.0 | 1194 | 1093 |
| 3 | 2.5 | 738 | 643 |
| 4 | 3.0 | 523 | 436 |
| 5 | 3.5 | 401 | 321 |
| 6 | 4.0 | 325 | 250 |
| 7 | 4.5 | 273 | 201 |
| 8 | 5.0 | 235 | 169 |

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

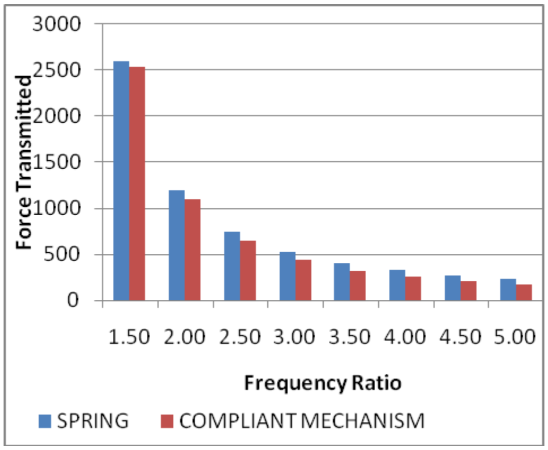


Fig. 21. Plot of Force Transmitted Vs. Frequency Ratio

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:

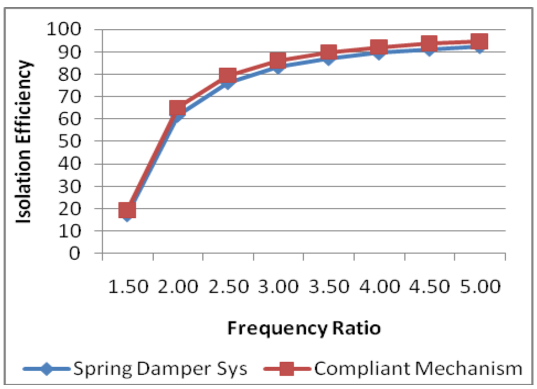


Fig. 22. Plot of Isolation efficiency Vs. Frequency Ratio

## 5. Conclusion

In this study, an attempt was made to find a replacement for the existing spring damper suspension system with compliant mechanism. Topology optimization method was used for the synthesis of compliant mechanism. Subsequent modal and harmonic analysis were 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 21. The plot of isolation efficiency Vs. frequency ratios gives 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.

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