Design & Structural Analysis of Hydraulic and Ferro Fluid Twin Tube Shock Absorber for Two Wheeler

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Abstract*- Stress and strain analysis is very important to predicting and preventing failures in materials when those are exposed to load. This paper aims to model and simulate the stresses and strain analysis of a hydraulic and ferro fluid twin tube shock absorber application of 356 kg designed. Modelling and analysis were performed by using modelling software and analysis software i.e. Solid Work 2014, ANSYS and HYPERMESH10. Initially a 3D modal of shock absorber was created by SolidWorks and meshing is carried out by hypermesh software. Stress and frequencies of both the twin tube shock absorbers were determined by Ansys. The obtained values are compared with analytical values.*

Keywords— *Shock Absorber, Hydraulic fluid, Ferro fluid, Solid Works, HYPERMESH, FEA.*

Introduction

Shock absorbers are very important in automobiles. The shock absorbers absorb maximum loads and provide cushioning effects to passengers and cargos. The amount of cushioning is depends up on the type of the fluids used in shock absorbers. Generally two types of shock absorbers are used one is mono tube shock absorber and second one is twin tube shock absorber. This study attempted to analyse the frequency and stress on hydraulic and ferro fluid shock absorber by using HYPERMESH and FEA analysis. The simulation data is very important because of this information is useful for further design improvements. Stress analysis is very important for to determine fatigue and life of the component. Vibration analysis also very use full for determine frequency, critical damping, under damping, over damping and resonance.

Pinjarla. Poornamohan et.al. (2012)They concluded that spring steel for spring is best and also their modified design was safe. The obtained stress and displacement values were less for modified design[1].
S.Gopinath et.al. (2014) They developed a "magnetic shock absorber" which helps to know how to achieve low cost and minimize the size[2].

Rahul Tekade et.al (2015) They compared the obtained results for both materials and identified the natural frequency is more for ASTM A228 than 67SiCr5.Finally the concluded and suggested as per their analysis using ASTMA228 [high carbon spring wire] for spring is best[3].

G.R. Chavhan et.al (2014) They are analysed the shock absorber by using fem analysis and used three different materials . The concluded the Carbon Fibre has the greater shock absorbing properties but disadvantage is that it was break earlier than Spring Steel and Beryllium Copper[4].

Ammar A.Aldair and et.al , (2011) in their study they reduced the energy consumption resulting for driving the actuators in active suspension, the electromagnetic device has been introduced which is capable of converting most of the vehicle's vibration energy into electrical energy through the rotation of the device and store them in the battery and used to generate appropriate damping forces to improve the riding comfort & road handling[5].

M.D. Rao. Et.al. (2002) They used electrodynamics shakers to obtain the equivalent dynamic properties of shock absorbers for NVH applications. Finally, they concluded some shakers were capable of withstanding static pre-loads which suitable for testing shock absorbers under larger displacements and lower frequencies[6].
Lei Zuo, et al. have worked on a prototype design of Electromagnetic energy harvester for vehicle suspension. In this paper they have designed, characterized and tested a prototype retrofit regenerative shock absorber.

Pradeep Khande, et al. have done an optimization analysis and experimental results of a retrofit regenerative shock absorber for vibration energy harvesting from vehicle suspension. A prototype four phase linear generator was developed and characterized the theoretical and experimental values. Finally his research work is possible to harvest energy from vehicles vibration in a bumpy roads and increases the load carrying capacity.

II. Design Considerations

Spring:

Mean diameter of coil, (D) = 33.3mm

Diameter of wire, (d) = 6.7mm

Total no. of coils, (n) = 6

Height (h)= 99.90mm

Outer diameter of spring coil, D_O = D+d= 40mm

Let, weight of the bike= 131kg

Weight of the three persons = 225kg

Total weight of the bike & persons = 356kg

Consider dynamic loads (w) = 435kg = 4267.35 N

Single shock absorber weight (W) = w/2 = 217.5kg = 2133.67N

Compression spring (δ) = $WD^3n/G.d^4$

Spring index (C) = D/d = 5

Therefore δ = 42.6 mm

Spring rate (K)= W/ δ = 50.08

Pitch of the coil, (P) = $(L_f-L_s/n_1)+ d$

The buckling factor for the hinged end and built in end spring

$W_{cr} = qx\ K_L x L_f$ = 50.08 x 0.1 x 99.99 = 500.74 N

Shock Absorber:

Length of the axial rod = 70mm

Diameter of the plate = 45mm

Thickness of the plate = 3mm

Diameter of top end = 8mm

Diameter of bottom end = 8mm

Diameter of the cylinder = 27mm

Length of the tube =76.93mm

II.2.3D Model

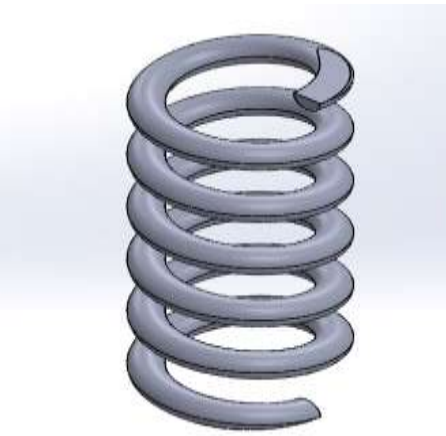

Fig.1. Coil Spring

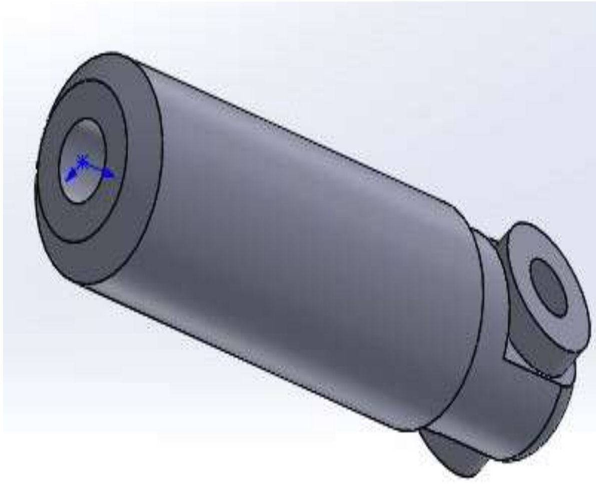

Fig.2.Tube

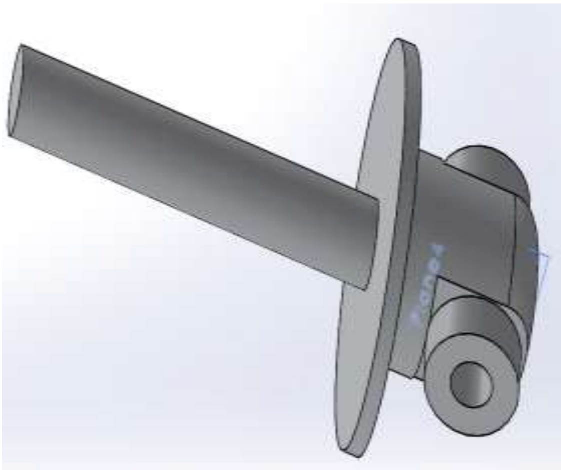

Fig.3. Axial Rod

Fig.4. Shock Absorber

II.3. 2D Model

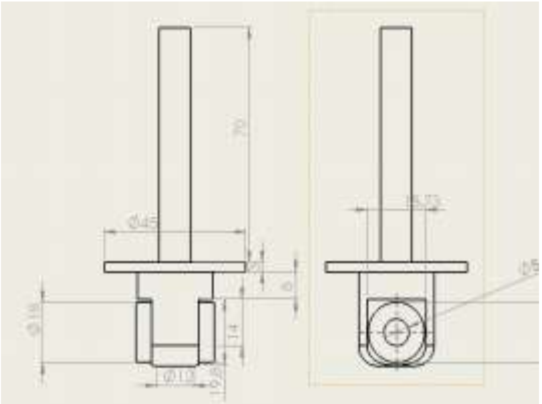

Fig.5. Front& Side view of an axial rod

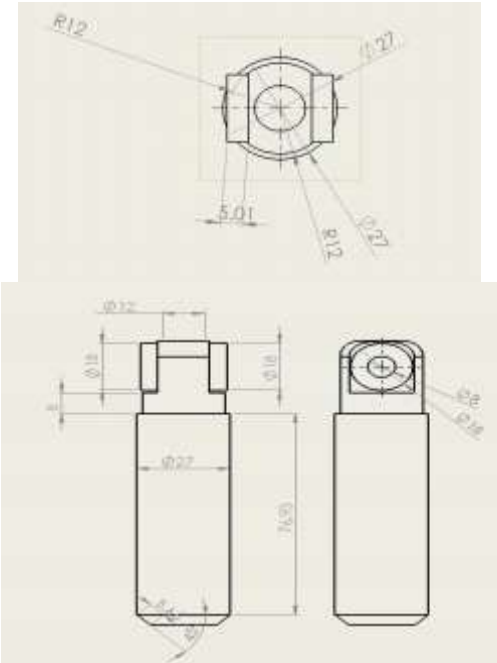

Fig.6. Top & Front view of an cylinder

III. Methodology

The main objective of the study is to analyse the shock obsoebers with using different fluids. Both the obtained values were compared with analytical values.

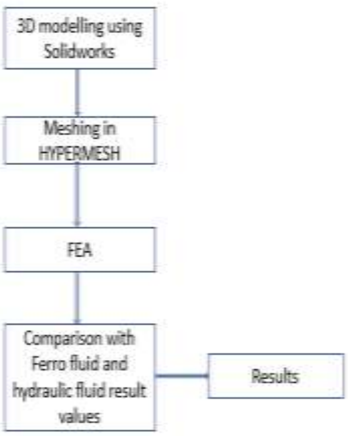

Fig.7. Process flow chart for shock absorber

III.1. Modelling

The 3-D modelling was done by using SolidWorks software.

Fig.8. 3-D model shock absorber

III.2. Meshing

All the components was meshed by using HYPERMESH software

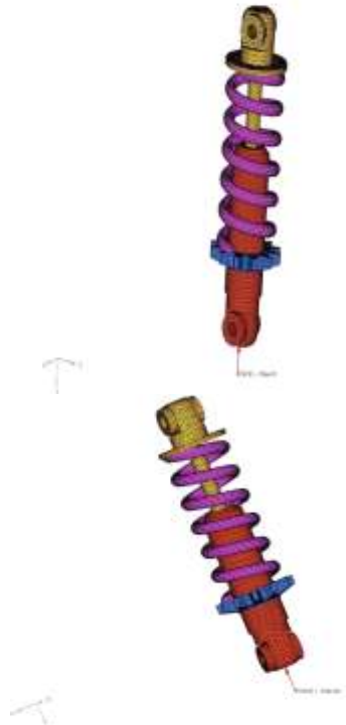

Fig.9.Meshing(Hypermesh) model shock absorber

III.3. FEM analysis

The displacement, frequency, time period, damping and absorption of load is very important for shock absorber. To meet these requirements to perform model and static analysis on hydraulic and ferro fluid shock absorber. The finite element analysis was carried out by using Ansys software. This analysis was performed based on the following assumptions.

The maximum load for both hydraulic and ferro fluid shock absorbers during applications 356kg

III.4. Material and fluids

Steel

Modulus of rigidity (G)	$55x10^3$ N/mm^2
Young's modulus (EX)	$1.965x10^5$ N/mm^2
Poisson's ratio (PRXY)	0.25
Density	$7.86x10^{-6}$ kg/mm^3

Ferro fluid

Density	1.07 g/cm^3
Viscosity	0.27pascal
Shear strength	100kpa-1100kpa

Hydraulic fliuid

Density	0.8 g/ml
Poissionsratio	0.5

III.5. Boundary Conditions

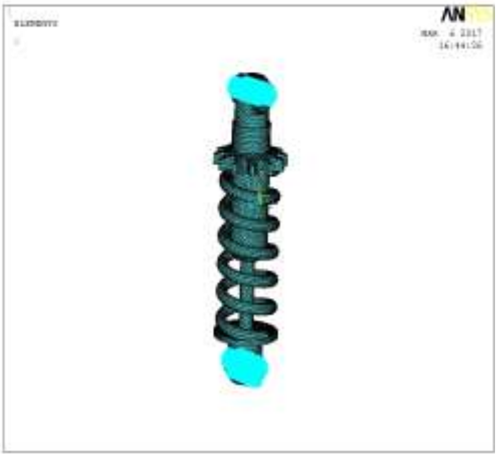

Fig.10. Boundary Conditions

The boundary conditions ware considered at upper and bottom end of the both the shock absorber

III.6. Loading

The force has been acting on shock absorber, with condersing the fluid and without condersing the fluid

IV. Results and Discussions

Fig.27& 33 shows the displacement and frequency distribution on hydraulic and ferro fluid twin tube shock absorber meshing modal. It can be seen that the maximum frequency and displacement of hydraulic fluid values were 19.178 Hz and 4.453m. The maximum frequency and displacement of ferro fluid values were 0.53 Hz and 0.024m.The stiffness of the hydraulic twin tube twin tube shock absorber (783.91 N/m) was much greater than the ferro fluid twin tube shock absorber (775.31N/m).In model analysis observed the damping rations of hydraulic and ferro fluid twin tube shock absorbers were 0.51 and 0.52. These two shock absorbers were belongs to under damped systems because of the damping ratio below the □ = 1. Fig. 26 & 18 shows the stress distribution on hydraulic and ferro fluid twin tube model shock absorber. It can be seen that the maximum Von Misses stress of hydraulic and ferro fluid twin tube shock absorbers were 30.299 and 36.904 KN/m^2.The analytical calculations was calculated by following equations. The obtained analytical values were compared with model analysis values. The theoretical vibration of both the shock absorbers ware provided in the Table 1. The experimental model analysis of both the shock absorbers were provided in the Table 2.The static analysis of both the shock absorbers were provided in reaming tables.

Undamped free vibration:

Stiffness of the spring, $K= (Gxd^4)/(8xnxD^3)$

$= ((55x10e^3) x (6.7)^4/(8x6x(33.3)^3)$

= 62.3 N-m

- Circular frequency of the motion (ω_n) = $\sqrt{k/m}$= $\sqrt{62.3x9.81/356}$

  = 1.31 rad/sec

- Restoring force = W-k(δ+x)
- The frequency of vibration, f_n = $1/2\pi \sqrt{k/m}$Hz = $1/2\pi \sqrt{62.3/356}$

  = 0.02 Hz

- The mass is displaced from its equilibrium position by a distance $x = A \cos\omega_n t + B \sin\omega_n t$

x_1 = (1) cos(1.31x1.22) + (13.33) sin(1.31x1.22)

= 1.37mm

x_2 = (1) cos(1.31x2.30) + (13.33) sin(1.31x2.30) = 1.56mm

x_3 = (1) cos(1.31x3.42) + (13.33) sin(1.31x3.42)= 2mm

Where A = x_o and B = v_o/ ω_n

Energy method :

Kinetic energy, T = ½ m $\dot{x}^2$

T_1 = ½ (356)(17.49)^2 = 54450.21kg-m/sec

T_2 = ½(356)(17.36)^2

= 53643.78 kg-m/sec

T_3 = ½(356)(17.30)^2 = 53273.62 kg-m/sec

Potential energy, V = ½ k $\dot{x}^2$

V_1= ½(62.3)(1.37)^2

= 58.46 N(or)kgm/sec^2

V_2 = ½(62.3)(1.56)^2 =75.80 N(or)kgm/sec^2

V_3 = ½(62.3)(2)^2 =124.6N(or)kgm/sec^2

Rayleigh's method:

Maximum velocity at mean position ,

$\dot{x}$ = ω_nA = (1.31)x(1)

=1.31m/sec

Maximum kinetic energy at mean position = ½ m $\omega_n^2A^2$= ½ (356)(1.31)^2(1)^2= 305 kg m^2/sec^2

Maximum potential energy at extreme position = ½k A^2= ½(62.3)(1)^2= 31.15 kg m^2/sec^2

Hydraulic fluid:

Energy dissipation in viscous damping ΔE= πcωX^2

Amplitude X = 4F/k = (4x356x9.81)/(89.36)= 0.15632 m

Therefore, energy dissipation in viscous damping

$\Delta E = \pi c \omega X^2$

$\Delta E = \pi\ (5.736)(200\pi)(0.15632)^2 = 275.54$ N-m

Power, P = $\Delta E/60$ KW = 275.54/60 = 4.59 KW

Damping ratio, □ = C/C_c

Where, C = Damping coefficient

C_c = Critical damping

Damping coefficient C = Force/Velocity

= (356 x 9.81)/(0.02x1000)= 174.68 NS/m

Critical damping $C_c = 2\sqrt{k/m} = 2(\sqrt{62.3x356})$

= 297 NS/m

Therefore Damping ratio, □ = C/C_c =174.68/297

= 0.58

Therefore, this is the under frequency.

Damped frequency, $\omega_d = \sqrt{1-\varepsilon 2}\,\omega$

= 1.067 rad/sec

Time period of the motion $t_d = 2\pi/\omega_d = 2\pi/1.067$ = 5.85 sec

Ferro fluid:

Energy dissipation in viscous damping

$\Delta E = \pi c \omega X^2$ = 276.67 N-m

Damping ratio, □ = C/C_c

Where, C = Damping coefficient

C_c = Critical damping

Damping coefficient (C)= Force/Velocity

= 0.4818 NS/m

Critical damping $(C_c) = 2\sqrt{k/m}$ =3.6957 NS/m

Therefore Damping ratio (□) = C/C_c = 0.13

Therefore, this is the under frequency.

Damped frequency $(\omega_d) = \sqrt{1-\varepsilon 2}\,\omega$ = 1.29 rad/sec

Time period of the motion $(t_d) = 2\pi/\omega_d$ = 4.87 s

Static Analysis for ferro fluid twin tube shock absorber and weight 356kg using spring steel as a material

Displacement

Direction	Maxi. Stress (MPa)	Mini. Stress (MPa)	Deformation (m)
X	$0.834e^{-4}$	$-0.122e^{-3}$	0.004504
Y	$0.542e^{-7}$	-0.004503	0.004504
Z	$0.320e^{-3}$	$-0.121e^{-3}$	0.004504

Stress

Direction	Maxi. Stress (MPa)	Mini. Stress (MPa)	Deformation (m)
	14.462	-80.169	0.004504
Y	17.741	-80.428	0.004504
Z	13.348	-81.975	0.004504

Strain

Direction	Maxi. Stress (MPa)	Mini. Stress (MPa)	Deformation (m)
Y	0.001154	$2397e^{-8}$	0.004504

Vonmisses stress

Maxi. Stress (MPa)	Mini. Stress (MPa)	Deformation (m)
36.904	$0.408e^{-3}$	0.004504

Fig.11. Static Load applied on tetra meshed model

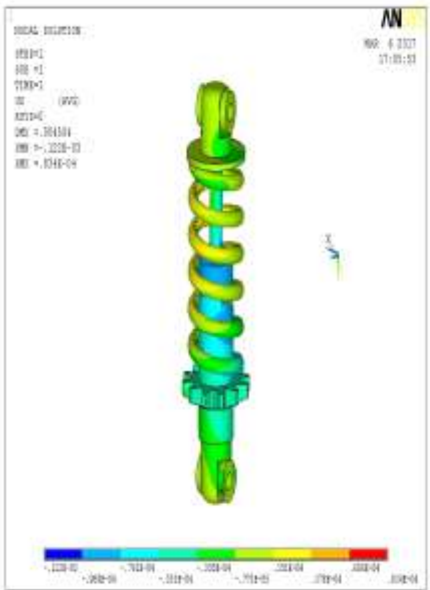

Fig.12. Displacement (u) in x- direction

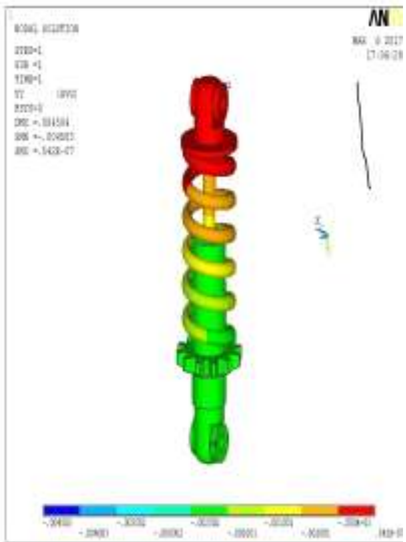

Fig.13. Displacement (u) in y- direction

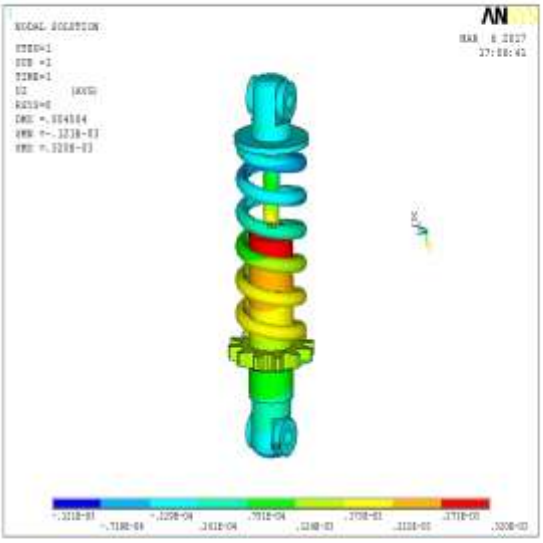

Fig.14. Displacement (u) in z- direction

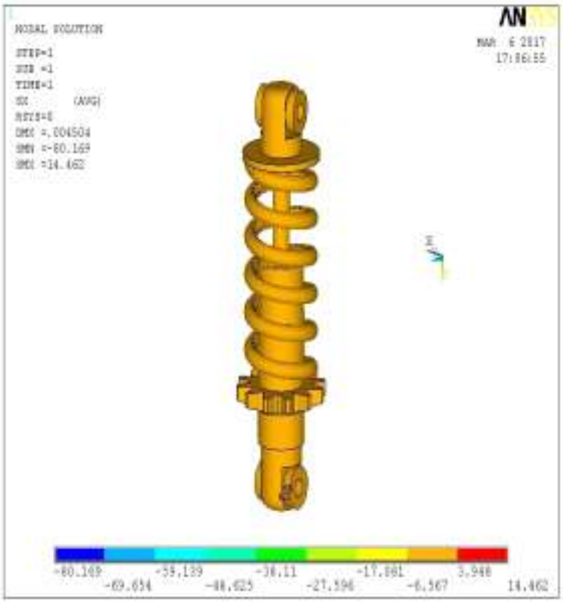

Fig.15.Stress (s) in x- direction

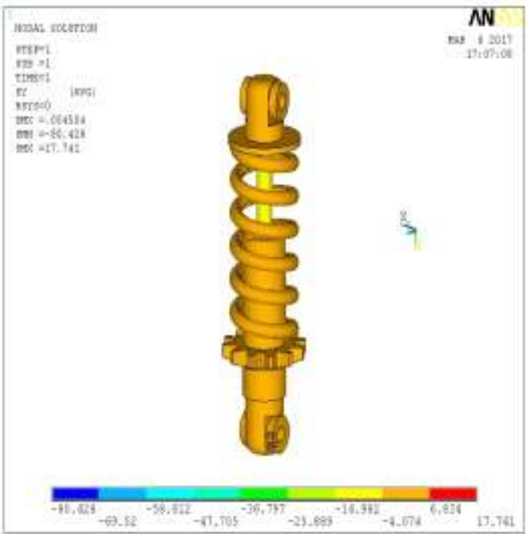

Fig.16.Stress (s) in y- direction

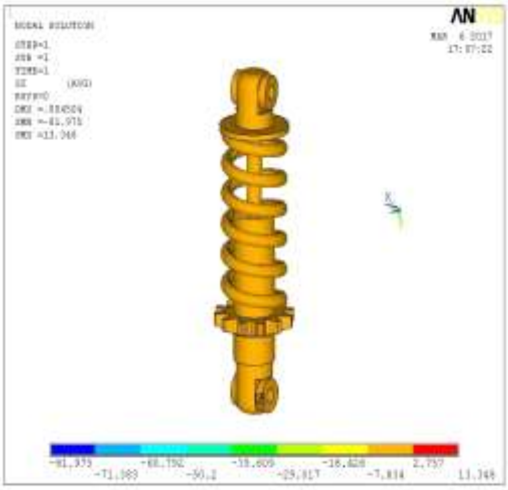

Fig.17. Stress (s) in z- direction

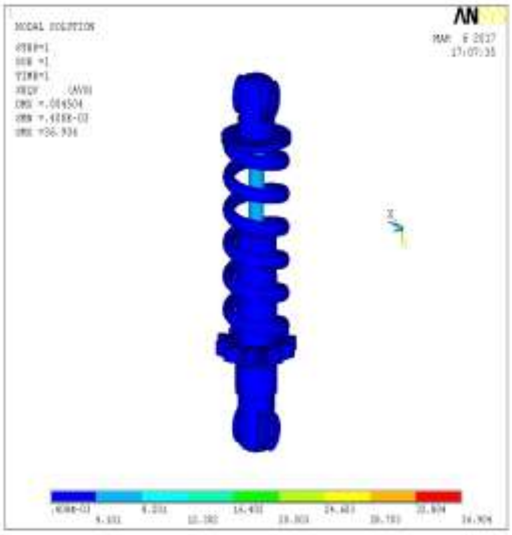

Fig.18. Vonmisses stresses

Static Analysis for hydraulic fluid twin tube shock absorber and weight 356kg using spring steel as a material

Displacement

Direction	Maxi. Stress (MPa)	Mini. Stress (MPa)	Deformation (m)
X	$0.834e^{-4}$	$-0.111e^{-3}$	0.004455
Y	$0.541e^{-7}$	-0.004454	0.004455
Z	$0.319e^{-3}$	$-0.12e^{-3}$	0.004455

Stress

Direction	Maxi. Stress (MPa)	Mini. Stress (MPa)	Deformation (m)
X	14.462	-80.169	0.004455
Y	17.741	-80.428	0.004455
Z	13.349	-81.974	0.004455

Strain

Direction	Maxi. Stress (MPa)	Mini. Stress (MPa)	Deformation (m)
Y	0.001154	$2397e^{-8}$	0.004455

Vonmisses Stress

Maxi. Stress (MPa)	Mini. Stress (MPa)	Deformation (m)
30.299	$0.405e^{-3}$	0.004455

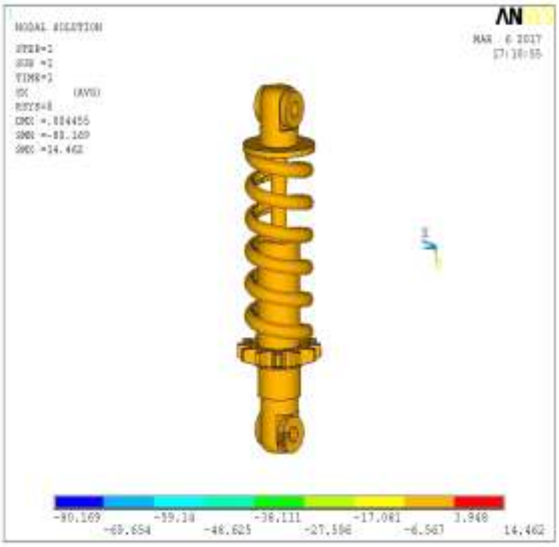

Fig.19. Stress(s) in x- direction

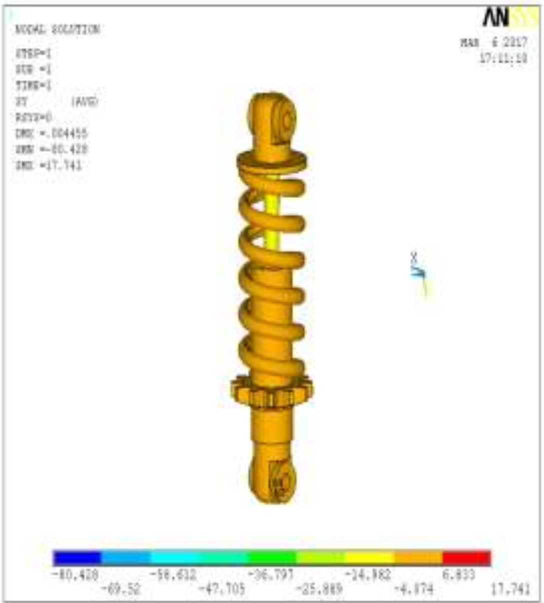

Fig.20. Stress(s) in y- direction

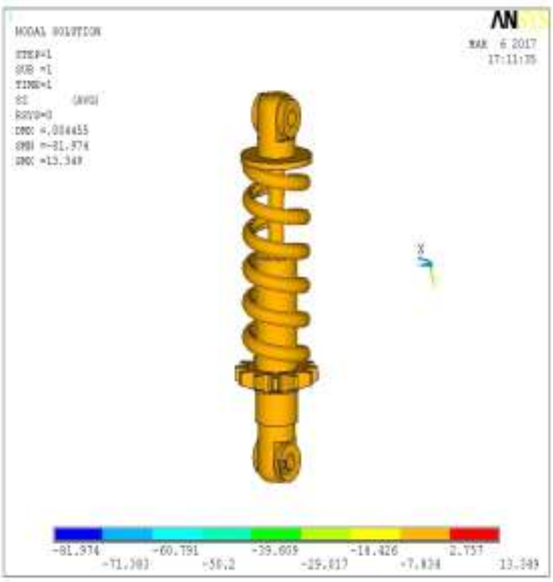

Fig.21. Stress(s) in z- direction

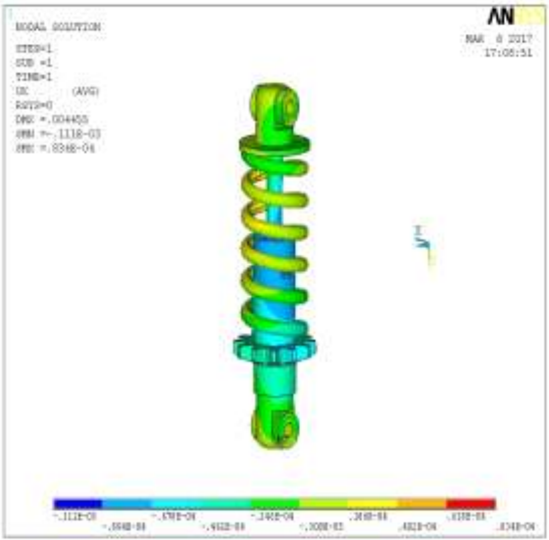

Fig.22. Displacement(u) in x- direction

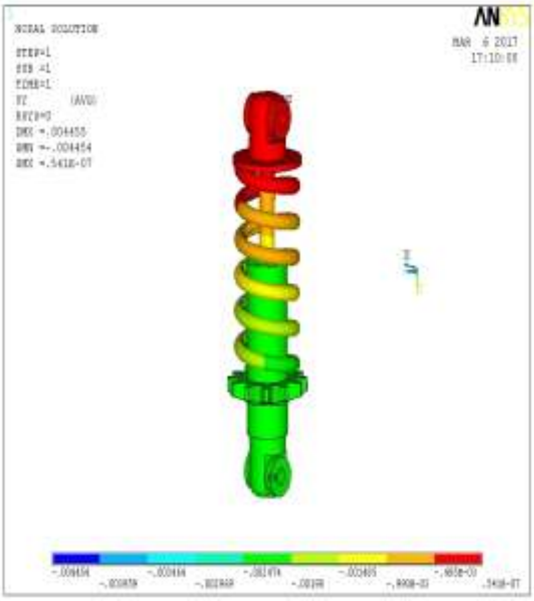

Fig.23. Displacement(u) in y- direction

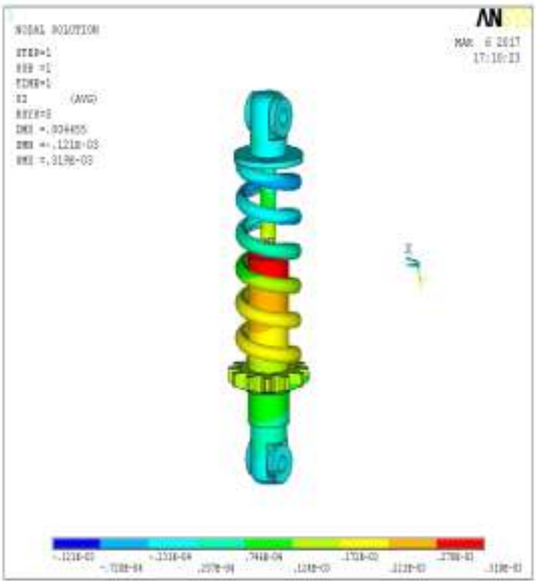

Fig.24. Displacement(u) in z- direction

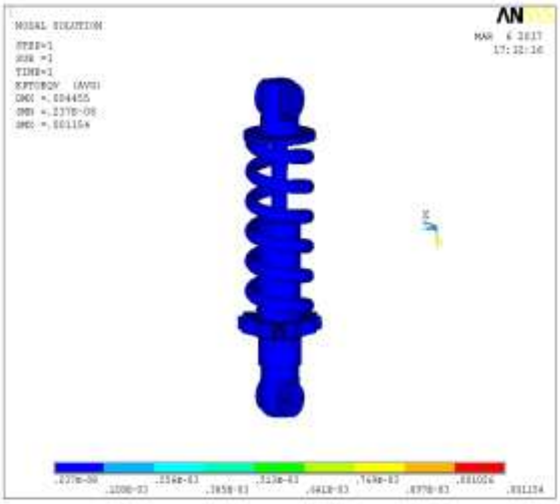

Fig.25.Strain(e) in y- direction

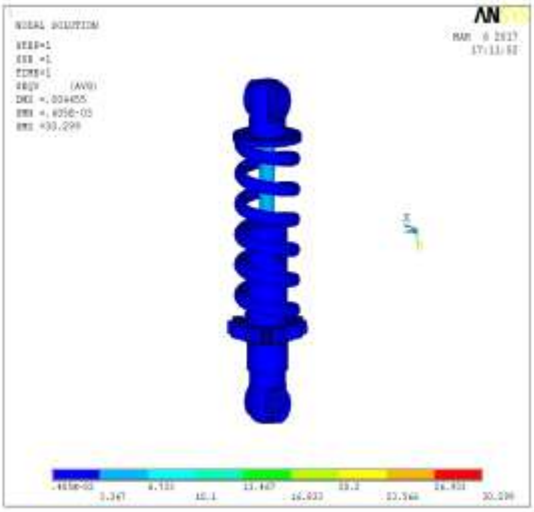

Fig.26 Vonmisses stress

Model analysis for hydraulic and ferro twin tube shock absorber and load 356 kg using spring steel as a material

Hydraulic Fluid

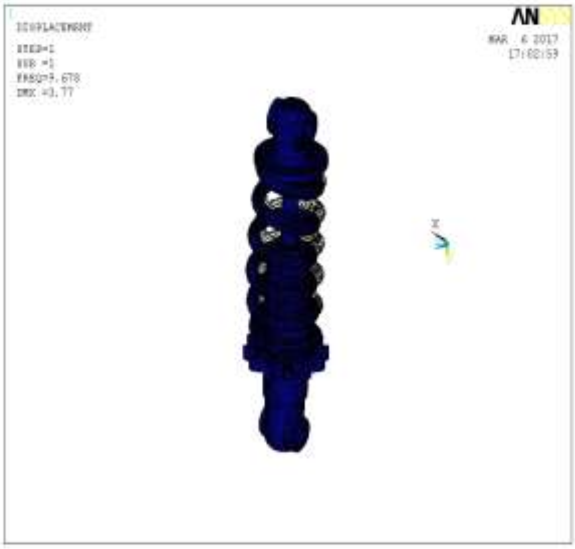

Fig.27. Model 1

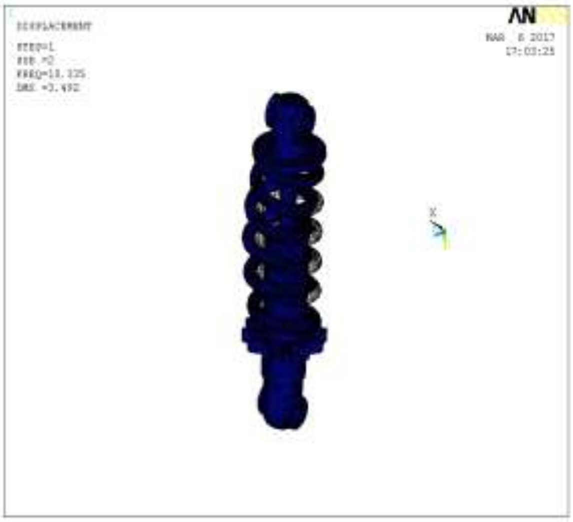

Fig.28. Model 2

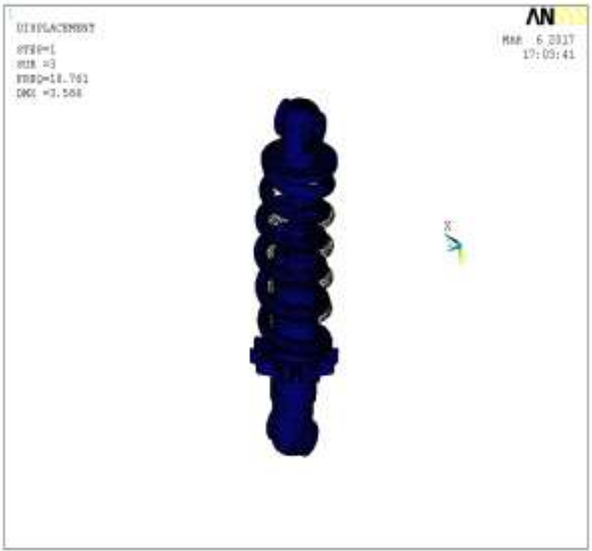

Fig.29. Model 3

Fig.30. Model 4

Fig.31. Model 5

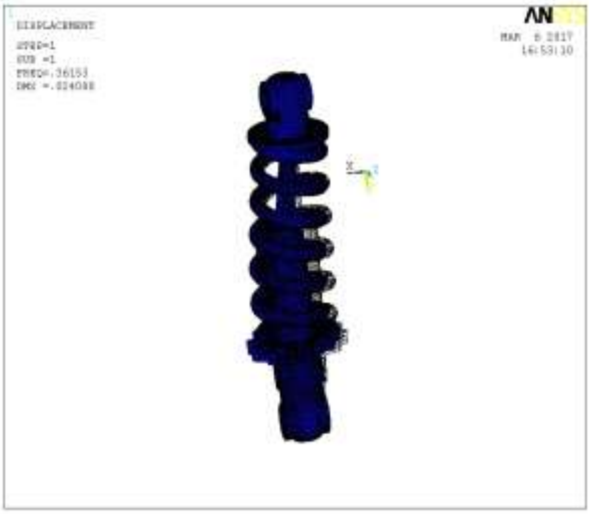

Fig.32. Model 1

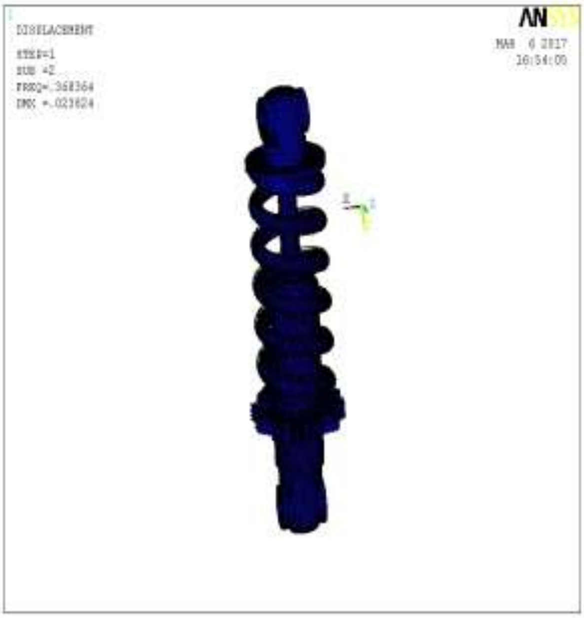

Fig.33. Model 2

Fig.34. Model 3

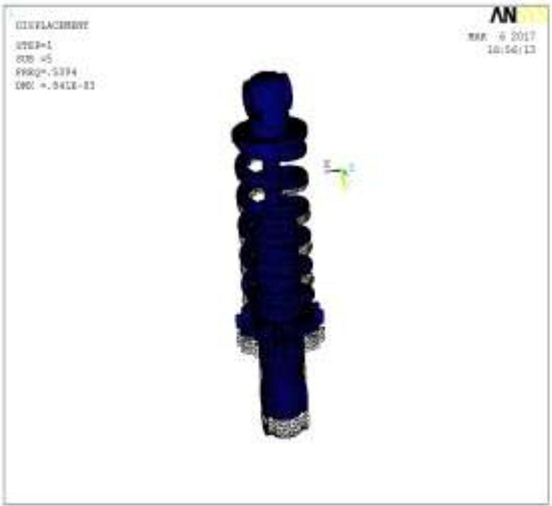

Fig.35. Model 5

Fig.36. Model 5

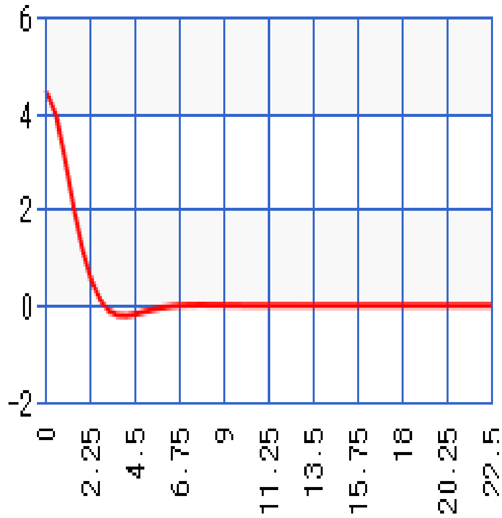

Fig. Displacement – Time period for under damped system (Hydraulic fluid twin tube shock absorber)

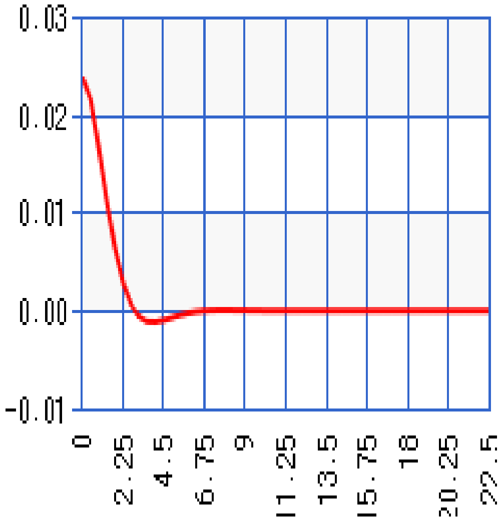

Fig. Displacement – Time period for under damped system (Ferro fluid twin tube shock absorber)

Table.1. Theoretical variational analysis for hydraulic and ferro fluid twin tube shock absorber Calculations:

S. No.	Type of shock absorber	Load (kg)	Stifness (N/mm)	Velocity (m/s)	Frequency (Hz)	Damping ratio	Damping frequency(Hz)	Time period (sec)	Displacement (mm)	Energy Dissipation (N-m)	Amplitude(m)
1	Ferro fluid twin tube shock absorber	356	62.3	0.02	0.20	0.58	1.067	5.85	1.64	276.67	0.15632
2	Hydraulic fluid twin tube shock absorber	356	62.3	0.02	0.20	0.13	1.29	4.87	1.64	275.54	0.15632

Table.2. Model analysis for hydraulic and ferro fluid twin tube shock absorber

S.No.	**Hydraulic fluid twin tube shock absorber**		**Ferro fluid twin tube shock absorber**	
1	Displacement(m)	Frequency(Hz)	Displacement(m)	Frequency(Hz)
2	3.77	9.678	0.024088	0.36153
3	3.492	10.335	0.023824	0.368364
4	3.586	10.761	0.002483	0.433725
5	4.453	11.517	$0.600e^{-4}$	0.494246
6	3.548	19.178	$0.841e^{-3}$	0.5394

Table.3. Variational analysis for hydraulic and ferro fluid twin tube shock absorber

S.No	**Type of fluid**	Material	**Load (N)**	**Stiffness (N/m)**	**Damped frequency (rad/sec)**	**Time Period (S)**	**Damping Ratio**
1	Ferro Fluid	Steel	3492.36	775.3	1.118	5.62	0.52
2	Hydraulic Fluid		3492.36	783.91	1.121	5.60	0.51

Conclusion

- In this paper designed a hydraulic and ferro fluid twin tube shock absorber. The 3D model of shock absorber was designed by using SolidWorks software.The model meshing was done by using HYPERMESH 10 software. The FEA was done by Ansys.
- The modal analysis was successfully carried out to determine displacement and frequencies on a hydraulic and ferro fluid twin tube shock absorber. The structural analysis was also successfully carried out to determine maximum stress and deflection on a hydraulic and ferro fluid twin tube shock absorber. Both the shock absorbers took material as a steel.
- Compared theoretical model values with experimental model analysis values of shock absorbers.
- In this study found out at a 356 kg load the frequency of the ferro fluid shock absorber is less as compared to the hydraulic fluid shock absorber.
- Finally the conclusion is ferro fluid shock absorber is best compare to hydraulic fluid shock absorber.
- This study found out that there is a analytical (2-D) and numerical (3-D) results. The future study will include experimental investigation.

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