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Static and Dynamic Analysis of a Two Wheeler Shock Absorber Using Different Materials for Helical Coil Spring

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Abstract – *The Shock Absorber is a part of the suspension system in a vehicle which is designed mechanically to handle shock impulse and dissipate kinetic energy. In a vehicle, it reduces the effect of travelling over rough ground, leading to improved ride quality and increase in comfort due to substantially reduced amplitude of disturbances. The design of spring in suspension system is very important. In this project a shock absorber of a popular model of a two wheeler is selected and a 3D model is created using SolidWorks2014. Structural analysis and modal analysis are done on the shock absorber by varying material for spring viz., two variants in Steel, two variants in Copper Alloys and one Titanium Alloy. The analysis is done by considering the kerb weight of the bike and a combined weight of two riders. Under Static Analyses, Linear Static analysis is done to validate the Stresses, Displacements and Modal analysis is done to determine the different frequencies for number of modes. And under Dynamic Analyses, Frequency Response Analysis is done to determine the displacement due to cyclic loading and Transient Analysis (Shock Analysis) is done to study the performance of the Shock Absorber during shock loading. Comparison is done for five materials to verify best material for spring in Shock absorber. Also, the spring diameter has been varied based on the results to check for scope of improved performance. 3D Modelling is done in SolidWorks 2014, Meshing is done in HyperMesh, OPTISTRUT for solving and Hyperview and Hypergraph for post processing and viewing the results.*

***Key Words*: Linear Static Analysis, Modal Analysis, Frequency Response (Harmonic) Analysis, Transient (Shock) Analysis, Shock Absorber, HyperWorks, SolidWorks, Helical Coil Spring, Optistruct, Redesign, Morphing, ASTM.**

Nomenclature

G	:	*Modulas of Rigidity*
D_m	:	*Mean Diameter of the coil*
d	:	*Diameter of the Spring Wire*
D_o	:	*Max. Outer Diameter of the Spring*
n_t	:	*Total Number of coils*
n	:	*Number of Active coils*
C	:	*Spring Index*
L_f	:	*Free length of the spring*
L_s	:	*Solid Length of the spring*
p	:	*Pitch of the coils*
k	:	*Whal's Stress Factor*
δ	:	*Deflection of spring under load*
τ	:	*Max. Shear Stress in the spring*
K	:	*Spring Stiffness*
ω_n	:	*Natural Frequency*
ω	:	*Frequency*
t	:	*Time Period*

1. INTRODUCTION

Shock absorbers are a critical part of a suspension system, connecting the vehicle to its wheels. The need for dampers arises because of the roll and pitches associated with vehicle maneuvering, and from the roughness of roads. Shock absorbers are devices that smooth out an impulse experienced by a vehicle, and appropriately dissipate or absorb the kinetic energy.

A safe vehicle must be able to stop and maneuver over a wide range of road conditions. Good contact between the tires and the road will able to stop and maneuver quickly. Suspension is the term given to the system of springs, shock absorbers and linkages that connects a vehicle to its wheels. Shock absorber is an important part of automotive suspension system which has an effect on ride characteristics. Shock absorbers are also critical for tire to road contact which to reduce the tendency of a tire to lift off the road. This affects braking, steering, cornering and overall stability.

In this project a shock absorber of Avenger 150CC vehicle will be selected and will be designed. A 3D model will be created by using SolidWorks. Structural analysis and modal analysis are done on the shock absorber by varying material for spring. Further to this, Frequency Response (Harmonic Analysis) and Transient Response (Shock Analysis) will be performed. The analysis will consider the kerb weight of the bike and load of two riders. Structural analysis will be done to validate the strength and modal analysis to determine different frequencies for number of modes. Frequency Response will be done to find out the displacements occurring in the frequency range. Transient analysis will be conducted to simulate the shock loads occurring on the shock absorber and to find the displacements occurring during shock and the time required for the shock absorber to damp the displacements. Comparison will be done among five materials to verify best material for spring in Shock absorber. Also, the wire diameter of the spring will be varied to check for scope of improvement in the performance characteristics of the shock absorber.

1.1 Literature Survey

In an approach involving the shock absorber of 150CC bike by P. Poornamohan[1] (2012), it was designed using parametric software Pro/Engineer. Structural and Modal were performed to validate the strength of the design. Materials used were Spring Steel and Beryllium Copper. It was observed that the Stress values were less for spring Steel compared to Beryllium Copper and hence the wire diameter was reduced by 2mm. It was observed that the stress and displacement values were less for the modified design which also reduced the weight of the spring.

In another approach by Karthik A. S. [2] (2016), a different model of a shock absorber evaluated for Structural analysis for different materials of spring viz., Structural Steel, Copper alloy, Aluminum alloy, Titanium alloy. It was observed that though the overall performance of titanium alloy was good, it is a very costly material to be put into general use.

In a project by Mr. A.C. Basha[3] (2017), a Shock Absorber was selected and analyzed by changing the materials for the springs by Structural analysis alone and the results were presented.

1.2 Formulation of Problem based on surveys and reviews

Based on Literature reviews, the problem in the present investigation is formulated as follows: "To design a model of an existing two wheeler shock absorber and perform Static, Modal, Frequency Response and Transient analysis to ascertain its Structural and Vibrational performance under various loading conditions for different materials for the spring. To optimize the design by varying the diameter of the spring wire and suggesting best suited material for spring to achieve better performance characteristics".
Materials proposed to be used for study will not be generic materials but will be selected from **ASTM standards**. Different materials proposed to be used are:

- Stainless Steel Spring Wire as per *ASTM A 313 (DIN 17221) – AISI 302/304.*
- Cold drawn, high tensile Carbon galvanized Steel as per *ASTM A 228.*
- Phosphor Bronze Grade A Spring Wire as per *ASTM B 159.*
- Beryllium Copper Spring Wire as per *ASTM B 197.*
- Titanium Alloy as per Grade *Ti-5Al-5V-5Mo-3Cr.*

In addition to Structural and Modal analyses, it is proposed to conduct Frequency Response and Transient Analyses also because Modal analysis will give the natural frequencies and corresponding mode shapes only. But to analyze the model completely, the displacements and accelerations related to the corresponding natural frequencies must be evaluated. These results are obtained by performing a Frequency Response analysis for a band of frequencies specified. Graphs between Frequency Vs. Displacements will be plotted for different materials and studied for their suitability.

A true performance of a Shock Absorber can be analyzed under shock loading. This involves a high impulse load applied in a very small time interval. The damping characteristics of shock absorber will be analyzed which will have a direct impact on the ride quality of the two wheeler.

2. Design of Helical Coil Spring based Shock Absorber

SolidWorks is a parametric, feature based, solid modeling, menu driven higher end software. It provides mechanical engineers with an approach to mechanical design with automation based on Built-in intelligence takes the guess work out of 3D design.

2.1 Modeling of Bottom Mount

Draw a circle of 52mm diameter and extrude it by a value of 3.5mm.

On its face, draw a circle of 35mm diameter and extrude it by a value of 36.5mm.

Now, A model of bearing pad is created by drawing two concentric circles of 30mm and 12mm respectively and extruding it to a length of 110mm over the top face. Mounting eye is modeled similarly for the dimensions shown below

This completes the model of the bottom mount and the final model is as shown below:

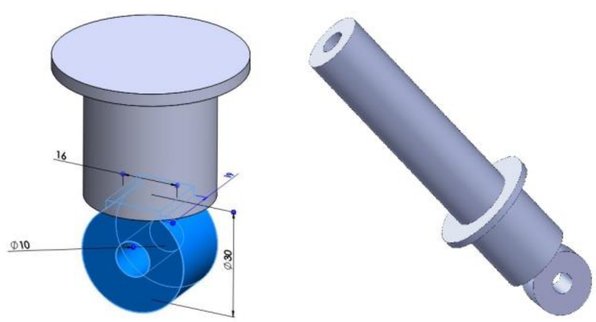

Fig-2.1: Bottom Mount

2.2 Modeling of Top Mount

Draw a circle of 62mm diameter and extrude it by a value of 40.5mm.

Shell Feature is used to create a hollow cup of 2mm wall thickness. Mounting eye is modeled similarly for the dimensions shown below

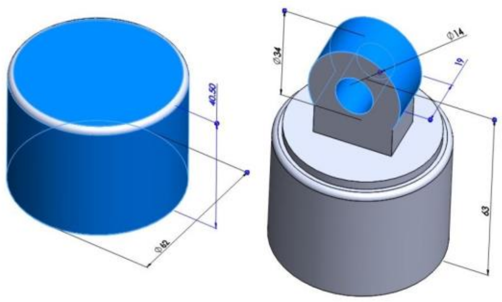

Fig-2.2: Top Mount

2.3 Modeling of Helical Coil Spring

The spring is measured to have a constant pitch and a variable diameters at its ends. The spring has a total of 12 full turns and two half turns which are flattened out to be fixed on to the Top and Bottom Mounts respectively. A helix is constructed with a constant pitch, height of 200mm and the diameters at the ends are 48mm and 44.5mm respectively. The spring wire diameter is measured to be 8mm. A circle of 8mm diameter is swept around the helix to create the basic spring. The ends of the spring are trimmed to become flat to be fixed on to the Top and Bottom Mounts respectively.

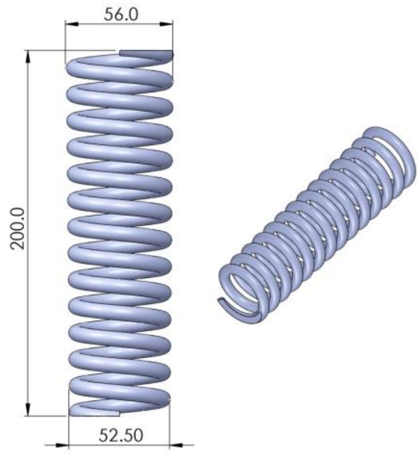

Fig-2.3: Helical Coil Spring

2.4 Assembly of Shock Absorber

Piston Rod for the Dampner is modeled similarly and the shock absorber is assembled in the Assembly module of SolidWorks as shown in the figure below:

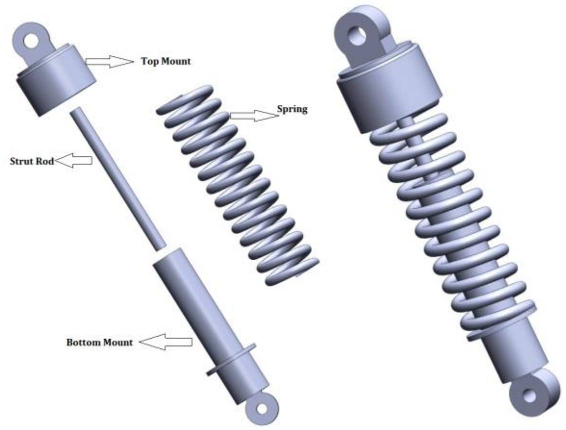

Fig-2.4: Exploded and Assembled views

3. Design Calculations OF Helical Coil Spring of Shock Absorber

3.1 Experimental determination of weight distribution

The weight of the two wheeler will be distributed among the front and rear wheels through the shock absorbers. To accurately determine the weight taken by the rear shock absorbers, the weight under the front and rear wheels is measured. Kerb Weight of the two wheeler from the specification catalogue is 148 kgs.

Fig-3.1: Determination of weight shared by front and rear Wheels

A weighing scale is placed under the rear & front wheels of the two wheeler and the reading is taken as 85.1kgs & 62.5 kgs respectively. Considering two persons are riding on the two wheeler having a weight of 65kgs each totaling to 130 kgs. Now this load has to be divided in the same proportion of the load distribution among the front and the rear suspensions.

Load on rear suspension = 74.95 kgs.

Load on front suspension = 55.04 kgs.

Therefore the total load on the rear suspension= Load due to self weight of the two wheeler + load of two persons shared by the rear suspension

Total load on rear suspension = 85.1 + 74.95 = 160.05 kgs.

This load is shared by two shock absorbers. Therefore the load on each shock absorber will be half of this value = 80.02 kgs. = 784.8 N

3.2 Design calculations for Helical Coil Spring

The material for the spring is taken as Stainless Steel Spring Wire as per *ASTM A 313.*

Modulus of Rigidity (G) = 69000 N/mm^2

Mean Diameter of the coil (D_m) = $\frac{48+44.5}{2}$ = 46.25mm

Diameter of the Spring Wire (d) = 8mm

Max. Outer Diameter of the Spring (D_o) = 56mm

Total Number of coils (n_t) = 14

Number of Active coils (n) = 12

Spring Index (C) = $\frac{D_m}{d} = \frac{46.25}{8}$ = 5.7

Free length of the spring (L_f) = 200mm

Solid Length of the spring (L_s) = d n_t = 8 *x* 14 = 112mm.

Pitch of the coils (p) =

$$\frac{L_f-L_s}{n_t-1}+d=\frac{200-112}{13}+8 = 14.76\text{mm}$$

Whal's Stress Factor (k)

$$k=\frac{4C-1}{4C-4}+\frac{0.615}{C}=1.2674$$

Deflection of spring under load (δ)

$$\delta=\frac{8WC^3n}{Gd}=25.27\text{mm}$$

Max. Shear Stress in the spring (τ)

$$\tau=\frac{k8WD_O}{\pi d^3}=310.6\text{ MPa}$$

Spring Stiffness (K)

$$K=\frac{Gd^4}{8D_m^3n}=29.7579\text{ N/mm}$$

3.3 Experimental determination of the deflection of spring under static load

The deflection of the spring under the self weight of two wheeler and with two persons with a combined weight of 130 kgs sitting on the two wheeler is determined. In all the cases the length is measured from centre to centre of the fixing bolts of the Shock Absorber.

Fig-3.2: Determination of deflection under load

Free length of the Shock Absorber – Vehicle is parked on its centre stand and the rear wheel is lifted off the ground = 285mm

Vehicle is taken off the centre stand, and it is resting on its two wheels – Centre distance of the bolts of the Shock Absorber due to self weight of the bike = 278mm. Deflection of the Shock Absorber = 285-278 = 7mm

Two persons are made to sit on the bike of 65kgs weight each. Centre distance of the bolts of the Shock Absorber = 251mm;
Deflection of the Shock Absorber = 278-251 = 27mm.

4. Linear Static Analysis

When loads are applied to a body, the body deforms and the effect of loads is transmitted throughout the body. The external loads induce internal forces and reactions to render the body into a state of equilibrium. Linear Static analysis calculates displacements, strains, stresses, and reaction forces under the effect of applied loads.

4.1 Meshing the CAD Model in Hypermesh

The complete assembly as a .STEP file is imported into Hypermesh and the different parts of the Shock Absorber that were generated in SolidWorks are arranged into different component collectors.

Taking Spring as an example, once the flat surface of the end of the spring is trimmed with plane, goto 2D page → Automesh → Mesh. Select the surface → Element Size=2 → Mesh Type is quads → Mesh. Once the surface is meshed, use 3D page → Line Drag → Mesh to create a Solid Hexa Mesh for the spring.

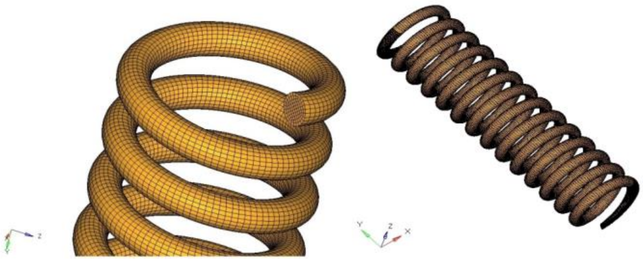

Fig-4.1: Hex Mesh for Spring

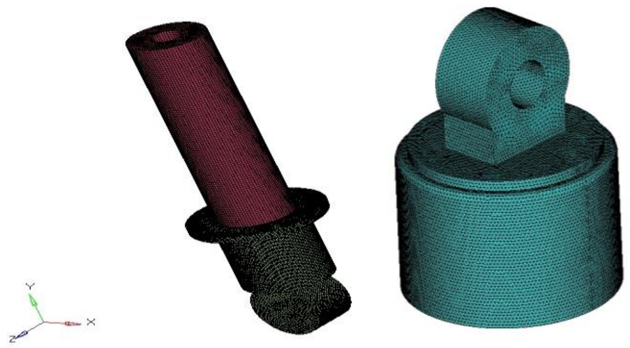

Fig-4.2: Tetra Mesh for Bottom & Top Mounts

Once the meshing of all the components is completed, quality of the mesh is to be checked. Goto Tool page → Check elems → 3d. Warpage, Aspect, Skew etc have to be checked for failed elements. Failed elements have to be adjusted using Geom Page → Node Edit → associate or move node options. All the nodes in the assembly should have connectivity to avoid failures during analysis runs.

Meshing of the Shock Absorber assembly is completed and now the further steps of preprocessing can be proceeded with.

Fig-4.3: Final Mesh Model of Shock Absorber

4.2 Applying Material Properties

As it is proposed to carry out the analyses for five different materials, Material cards for the different materials to be applied to the components have to be created first with the following material properties:

Table 4.1: Material Properties

Material Properties			
Material Specification	**Young's Modulas (GPa)**	**Poisson's Ratio**	**Density (g/cc)**
Stainless Steel Spring Wire as per *ASTM A 313*	193	0.3	7.92
Cold drawn, high tensile Carbon galvanized Steel as per *ASTM A 228*	207	0.31	7.85
Phosphor Bronze Grade A Spring Wire as per *ASTM B 159*	103	0.341	8.86
Beryllium Copper Spring Wire as per *ASTM B 197*	128	0.3	8.26
Titanium Alloy as per Grade *Ti-5Al-5V-5Mo-3Cr*	112	0.3	4.65

4.3 Applying Boundary Conditions & Loads

By the physical assembly of the Shock Absorber to a two wheeler, it can be noted that the top eye is a fixed contact and the bottom eye fixes to the axle of the rear wheel which is where the loads are exerted. The Strut rod is in sliding contact with the bottom mount and the spring is constrained within the top and bottom mounts.

Creating Contact Surfaces is of paramount importance in the modeling of a Shock Absorber as the dampner is simulated for analysis by giving dampning percentage to the contact surfaces of the strut rod and the dashpot of the Bottom Mount.

A Load of 784.8N has to be applied on the nodes of the fixing eye of the bottom mount. For this, the load has to be divided among the nodes of the eye. The number of nodes should be determined and the load has to be divided equally among the nodes and applied at the bottom mount fixing eye. Load Steps for Linear Static Analysis have to be created.

4.4 Solving the setup using OPTISTRUCT solver

Altair® OptiStruct® is an industry proven, modern structural analysis solver for linear and non-linear structural problems under static and dynamic loadings. It is the market-leading solution for structural design and optimization. Based on finite-element and multi-body dynamics technology and through advanced analysis and optimization algorithms, OptiStruct helps designers and engineers rapidly develop innovative, lightweight and structurally efficient designs.

The analyses will be run for all the five materials and respective .h3d files which are the output files will be generated.

4.5 Viewing Results on Hyperview

Case-1: Stainless Steel Spring Wire as per *ASTM A 313*

Open Hyperview window and browse to the location of the .h3d file of the Case-1 Analysis folder. Apply. The results will be loaded in to the model window.

Select Contour → Result Type: Displacement, Magnitude → Selection: Components: All → Apply. Result Type: Element Stresses, von Mises → Selection: Components: All → Apply

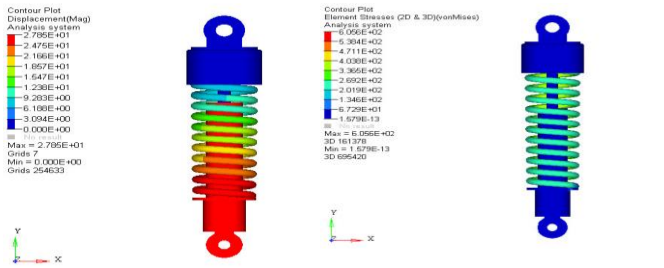

Fig-4.4: Max. Displacement (27.85mm); Stress (605.6MPa)

Case-2: Cold drawn, high tensile Carbon galvanized Steel as per *ASTM A 228*

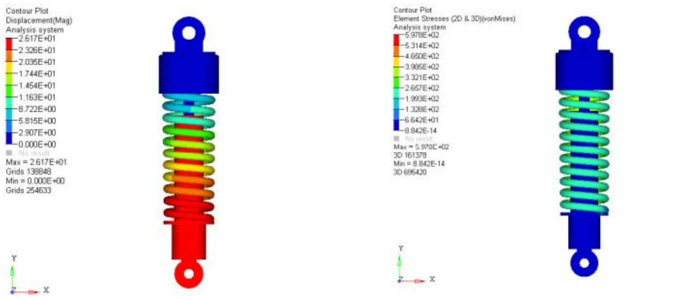

Fig-4.5: Max. Displacement (26.17mm); Stress (597MPa)

Case-3: Phosphor Bronze Grade A Spring Wire as per *ASTM B 159*

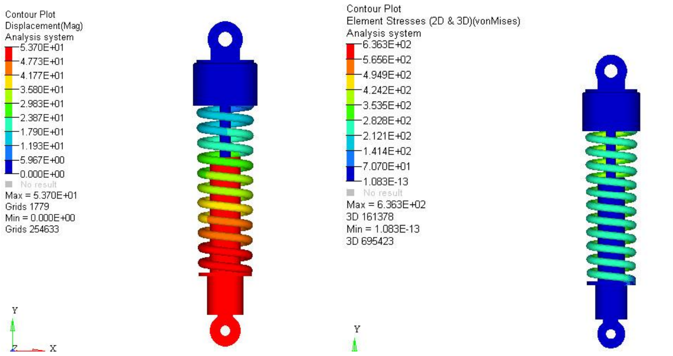

Fig-4.6: Max. Displacement (53.7mm); Stress (636.6MPa)

Case-4: Beryllium Copper Spring Wire as per *ASTM B 197*

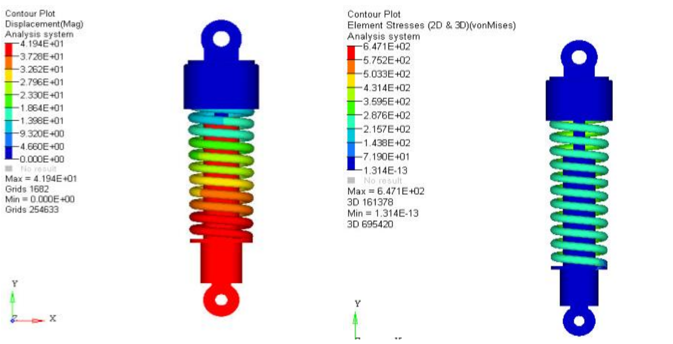

Fig-4.7: Max. Displacement (41.9mm); Stress (647.1MPa)

Case-5: Titanium Alloy as per Grade *Ti-5Al-5V-5Mo-3Cr*

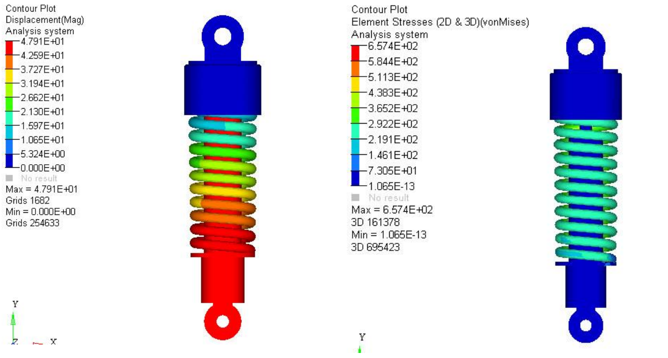

Fig-4.8: Max. Displacement (47.9mm); Stress (657.4MPa)

From the results of Linear Static Analyses it is evident that the displacement and stress values obtained for Copper Alloy based springs and Titanium alloy springs is on the higher side. Similar is the case with the maximum stress values that are obtained.

Spring Steel based springs are showing better results in Linear Static Analysis.

5. Normal Mode / Modal Analysis

If an elastic structure is disturbed from its stationery position by applying a force to have an initial displacement and then the force is removed, the structure will oscillate about its equilibrium position. This type of oscillation by the initial disturbance only is called free vibration. The oscillatory motion occurring at certain frequencies known as natural frequencies and characteristic values follow well defined deformation patterns known as mode shapes. When a structure vibrates with a given natural frequency (ω), that unique shape with arbitrary amplitude corresponding to (ω) is called a mode. In general for a 'n' DoF system, there will be 'n' natural frequencies and modes. The study of such free vibrations is very important in finding the dynamic response of the system. First ten modal frequencies are evaluated for all the materials:

Table 5.1: Results of Modal Analysis (Freq-Hz)

Mode	Case-1	Case-2	Case-3	Case-4	Case-5
Mode 1	25	24.6	16.9	19.3	19.1
Mode 2	77	76.3	50.6	58.7	73.3
Mode 3	77.5	76.7	50.8	59	73.6
Mode 4	107.4	105.8	70.1	81.8	99.1
Mode 5	112.8	111.9	74.4	85.9	107

Mode 6	177.3	175.6	116.5	135.1	168.5
Mode 7	179.1	177.4	117.5	136.4	170.1
Mode 8	202.9	200.1	131.9	154.3	190.8
Mode 9	223.9	222.3	147.7	170.5	212.6
Mode 10	291.7	288.8	190.9	222	276.6

From the results of Modal Analysis it is observed that Case-3 pertaining to Phosphor Bronze material has the first modal frequency below the 20Hz range. Low frequencies, typically less than 20Hz range have a high possibility of getting into resonance with the engine frequencies which are also low frequencies.

Case-4 & Case-5 pertaining to Beryllium copper and titanium alloy are also border line with the low frequency levels. These are also not in the acceptable performance levels. Case-1 & Case-2 of Steel based springs have their first modes beyond the low frequency levels. Though their performance is acceptable, there is scope of improvement by optimizing the spring parameters.

6. Frequency Response (Harmonic) Analysis

Frequency response analysis is used to calculate the response of a structure to steady state oscillatory excitation. Frequency response analysis is used to compute the response of the structure, which is actually transient, in a static frequency domain. The loading is sinusoidal. The loads can be forces, displacements, velocity, and acceleration. The results from a frequency response analysis are displacements, velocities, accelerations, forces, stresses, and strains. OPTISTRUCT supports Direct and Modal frequency response analysis. Results of this analysis for the five materials selected are as follows:

Case-1: Stainless Steel Spring Wire as per *ASTM A 313*

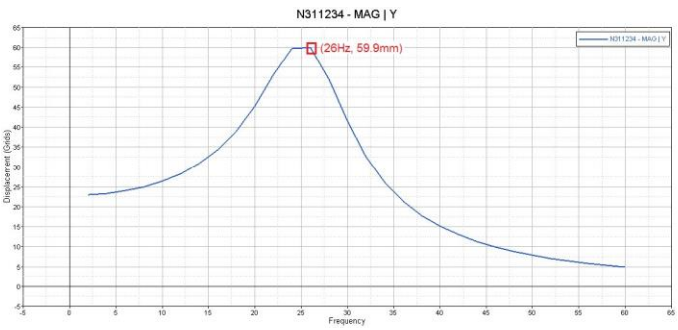

Fig-6.1: Max. Displacement 59.9mm @ 26Hz

Case-2: Cold drawn, high tensile Carbon galvanized Steel as per *ASTM A 228*

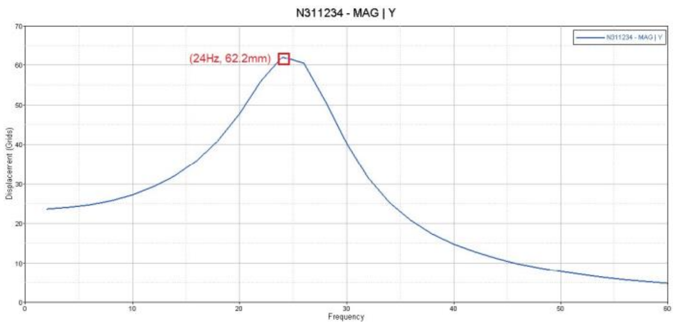

Fig-6.2: Max. Displacement 62.2mm @ 24Hz

Case-3: Phosphor Bronze Grade A Spring Wire as per *ASTM B 159*

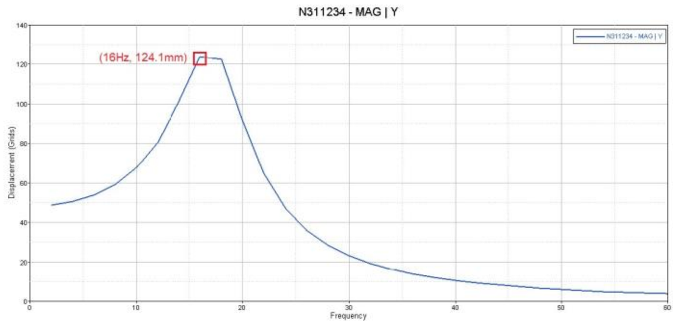

Fig-6.3: Max. Displacement 124.1mm @ 16Hz

Case-4: Beryllium Copper Spring Wire as per *ASTM B 197*

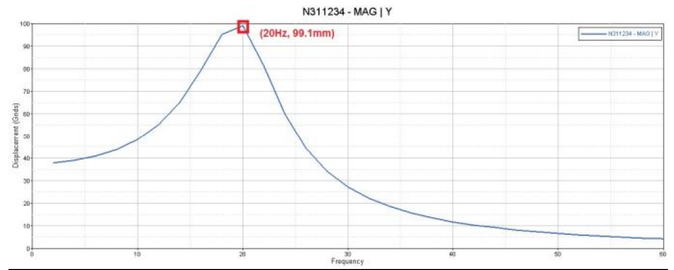

Fig-6.4: Max. Displacement 99.1mm @ 20Hz

Case-5: Titanium Alloy as per Grade *Ti-5Al-5V-5Mo-3Cr*

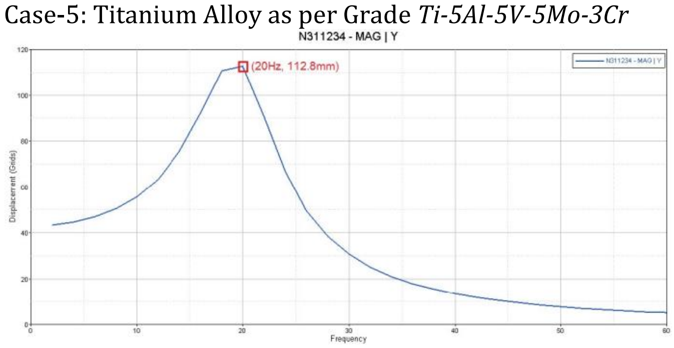

Fig-6.5: Max. Displacement 112.8mm @ 20Hz

All the analyses under Frequency Response Analysis are done by giving a unit load excitation to the bottom eye of the Shock Absorber and a cyclic load is applied. A sinusoidal load is applied in a frequency range of 0Hz to 60Hz.

In the results obtained, it can be observed that the frequency at the first resonance point matches nearest to the results of first modal frequency obtained in modal analysis. This proves the accuracy of frequency response analysis and the corresponding displacements are thus obtained.

From the results above, Copper Alloy based springs and titanium alloy springs have very high deflections almost near to the complete compression of the spring which is not acceptable. Spring Steel based springs are showing considerably better results but there is a scope available for improvement.

7. Transient Response (Shock) Analysis

Transient response analysis is used to calculate the response of a structure to time-dependent loads. Typical applications are structures subject to earthquakes, wind, explosions, or a vehicle going through a pothole.

The loads are time-dependent forces and displacements. Initial conditions define the initial displacement and initial velocities in grid points.

The results of a transient response analysis are displacements, velocities, accelerations, forces, stresses, and strains. The responses are usually time-dependent.

OPTISTRUCT supports Direct and Modal transient response analysis.

7.1 Generation of a Time dependant Loading Curve

A load of 1500N is to be applied as a shock/impulse load for 24 milliseconds. The response of the shock absorber for 100 milliseconds will be studied with an damping of 4% applied to the model. The loading will be a half sine curve calculated as below:

$$F_t = F \sin\left(\frac{2\pi}{T}\right) x\ t$$

The loading curve is generated as a Table in MS Excel first and the same will be imported to Hypermesh as a .csv file which will be used as an input for loading curve in transient response analysis.

Table 7.1: Table for loading curve for Shock/Impact

Time Interval (t) (ms)	Force Applied (F_t) (N)		Time Interval (t) (ms)	Force Applied (F_t) (N)
0	0		0.013	1487.17
0.001	195.79		0.014	1448.89
0.002	388.23		0.015	1385.82
0.003	574.03		0.016	1299.04
0.004	750		0.017	1190.03
0.005	913.14		0.018	1060.66
0.006	1060.66		0.019	913.14
0.007	1190.03		0.02	750
0.008	1299.04		0.021	574.03
0.009	1385.82		0.022	388.23
0.01	1448.89		0.023	195.79
0.011	1487.17		0.024	0
0.012	1500		0.1	0

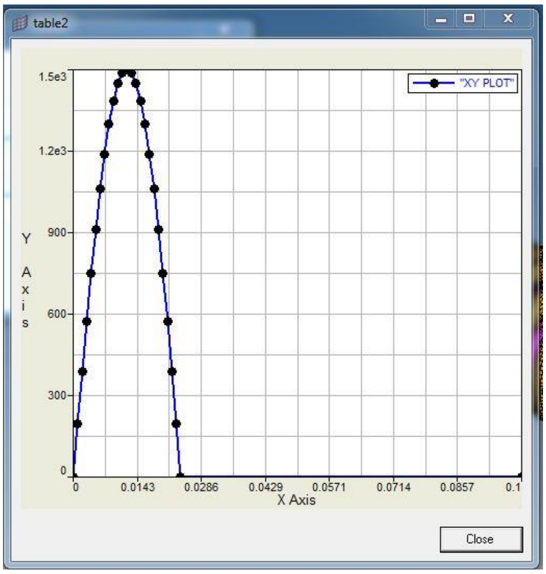

Fig-7.1: Loading Curve imported to Hypermesh

With this loading curve as input along with 4% of damping applied to the model to simulate the action of dampner in the Shock Absorber, Transient Analysis is performed for all the five materials and the results are as per figures below:

Case-1: Stainless Steel Spring Wire as per *ASTM A 313*

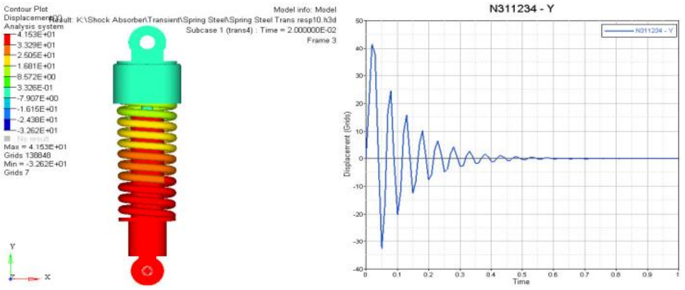

Fig-7.2: Max Displacement: 41.5mm

Case-2: Cold drawn, high tensile Carbon galvanized Steel as per *ASTM A 228*

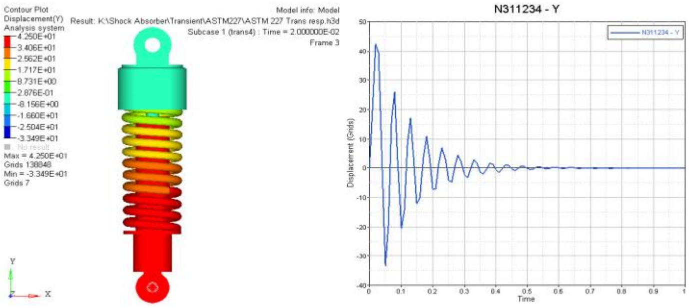

Fig-7.3: Max Displacement: 42.5mm

Case-3: Phosphor Bronze Grade A Spring Wire as per *ASTM B 159*

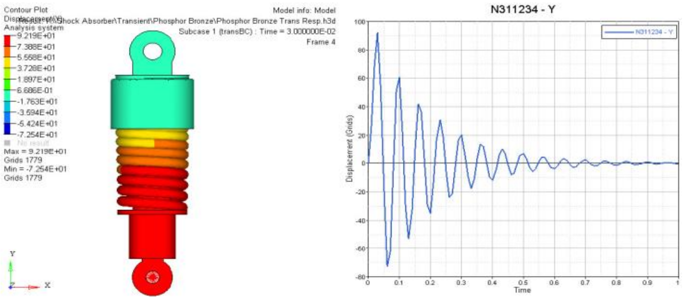

Fig-7.4: Max Displacement: 92.1mm

Case-4: Beryllium Copper Spring Wire as per *ASTM B 197*

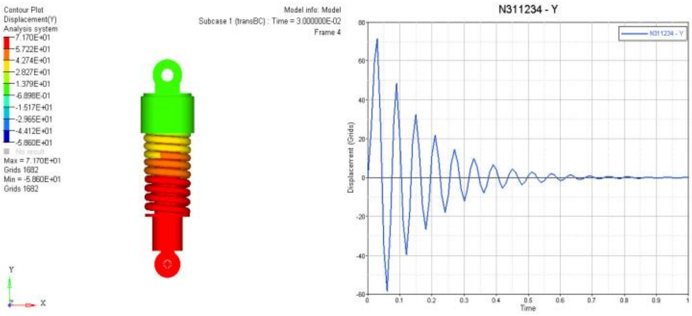

Fig-7.5: Max Displacement: 52.4mm

Case-5: Titanium Alloy as per Grade *Ti-5Al-5V-5Mo-3Cr*

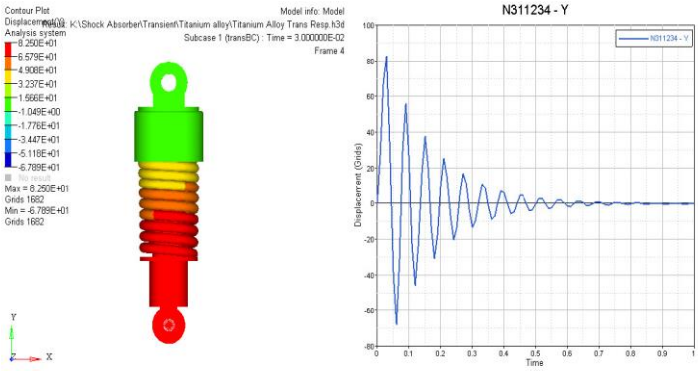

Fig-7.6: Max Displacement: 82.5mm

From the results of the Transient Analysis, again the Copper based and Titanium Alloy springs haven't shown satisfactory results. The deflection ranges are too high in comparison with Spring Steel based springs.

8. Redesign of Spring

The re-design of the spring is targeted at increasing the stiffness of the spring to reduce the deflections in impact loading cases and to shift the resonant frequencies away from the lower frequency ranges of 0-20Hz.

This is achieved by changing the wire diameter of the spring and selecting a different (stiffer) grade of spring steel.

Table 5.1: Results of Modal Analysis (Freq-Hz)

Existing Design Parameters	**Proposed Design Parameters**
Spring Diameter: 8.0mm	Spring Diameter: Next higher Standard Wire Diameter of 9.125mm
Material of the Spring: Stainless Steel Spring Wire as per *ASTM A 313*	Material of the Spring: Cold drawn, high tensile Carbon galvanized Steel as per *ASTM A 228*

8.1 Increasing the Spring Wire Diameter Geometry using Morphing

HyperMorph is a mesh morphing tool that allows you to alter finite element models while keeping mesh distortions to a minimum.

HyperMorph can be used to Change the profile and the dimensions of your mesh, Map an existing mesh onto a new geometry, Create shape variables that can be used for optimization.

Using this module, the wire diameter of the meshed model of the spring is increased to 9.1mm without affecting the rest of the model or the continuity.
Material properties pertaining to ASTM A228 grade of Spring Steel are assigned to the redesigned spring and all the four analyses are repeated.

8.2 Linear Static Analysis

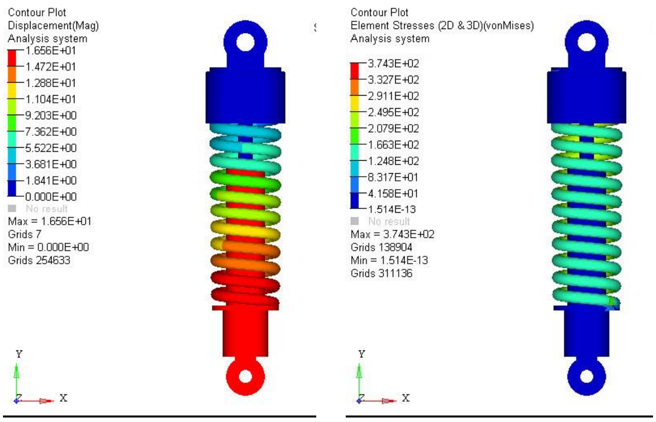

Fig-8.1: Max. Displacement (16.5mm); Stress (374.3MPa)

8.3 Modal Analysis

Table 8.1: Table for loading curve for Shock/Impact

Mode	Frequency		Mode	Frequency
Mode 1	30.1 Hz		Mode 6	195.3 Hz
Mode 2	84.7 Hz		Mode 7	197.4 Hz
Mode 3	85.5 Hz		Mode 8	224.7 Hz
Mode 4	120.2 Hz		Mode 9	247.9 Hz
Mode 5	124.9 Hz		Mode 10	322 Hz

8.4 Frequency Response Analysis

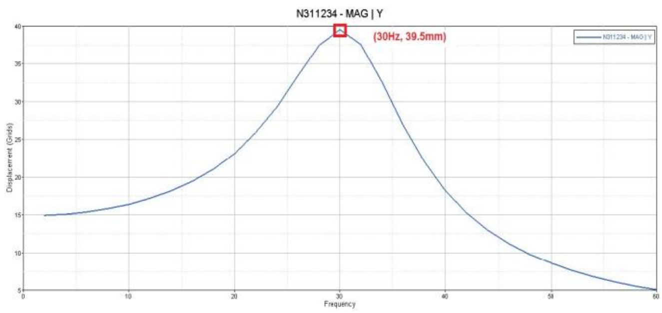

Fig-8.2: Max. Displacement 39.5mm @ 30Hz

8.5 Transient Response Analysis

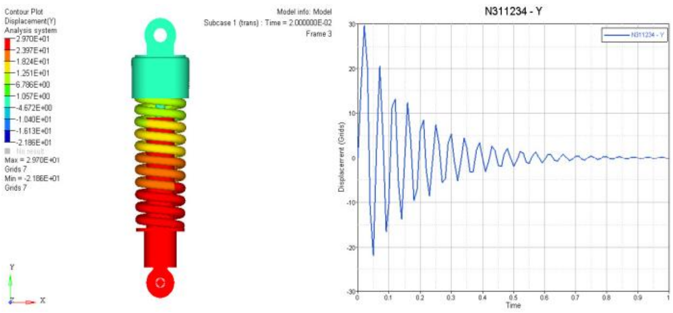

Fig-8.3: Max Displacement: 29.7mm

9. Results and Discussions

In this project, the Shock Absorber used in a 150CC bike was modeled using 3d parametric software – SolidWorks2014. To validate the strength of the design, Structural Analysis and Modal Analysis were performed. To evaluate its dynamic performance characteristics, Frequency Response Analysis and Transient analysis (Shock analysis) were performed. All these different analyses were performed by changing the material of the Spring Wire. The materials that were used were:

Stainless Steel Spring Wire as per ASTM A 313 (DIN 17221) – AISI 302/304.

Cold drawn, high tensile Carbon galvanized Steel as per ASTM A 228.

Phosphor Bronze Grade A Spring Wire as per ASTM B 159.
Beryllium Copper Spring Wire as per ASTM B 197.
Titanium Alloy as per Grade Ti-5Al-5V-5Mo-3Cr.

All the analyses were done and from the detailed results presented in previous sections for different material for different analyses, the maximum values of Displacements, stresses, modal frequencies, displacements for cyclic and transient (impact) loading are consolidated and tabulated and shown in Table 9.1

1. By observing the analysis results of Linear Static Analysis, the stresses observed are less than their respective yield stress values. So from Stress point of view all the materials qualify as suitable materials for Spring.

Table 9.1: Results of all the Analyses of Existing Design

Material	Linear Static		Modal	FRA	TRA
	Disp. (mm)	Stress (MPa)	Freq (Hz)	Freq (Hz) @ Disp. (mm)	Disp. (mm)
Case 1	27.85	605.6	25, 77, 77.5	26 @ 59.9	41.5
Case 2	26.17	597.0	24.6, 76.3, 76.7	24 @ 62.2	42.5
Case 3	53.70	636.3	16.9, 50.6, 50.8	16 @ 124.1	92.1
Case 4	41.94	647.1	19.3, 58.7 59	20 @ 99.1	52.4
Case 5	47.90	657.4	19.1, 73.3, 73.6	20 @ 112.8	82.5

2. The results of displacements from Linear Static Analysis, the existing design of the spring with the above referred material grades is within 50% of the total allowable compression of the spring. Thus the base design with its properties is validated from maximum displacement point of view.
3. The static deflections of Phosphor Bronze, Beryllium Copper, Titanium alloy are all in high ranges. If these materials are employed as springs, there will be very play left in the spring to deflect during shock/impact loads.
4. By comparing the results of Normal Mode analysis, the natural frequencies of Phosphor Bronze, Beryllium Copper, Titanium alloy based Springs are below low frequency ranges of 0-20Hz. This can lead to resonance conditions with the engine operating frequencies.
5. Frequency Response Analysis gives the Resonant Frequency and the associated Displacement at that frequency. From the two grades of Spring Steel of ASTM A313 and ASTM A228, though the natural frequencies are above the 20Hz range, the associated displacements of 59.9mm and 62.2mm are quite high. These values are quite near to the maximum allowable displacement of the Shock Absorber of 70mm.
6. The actual performance of a Shock Absorber can be evaluated from Transient Response analysis as most of the loads that occur during the movement of the two wheeler are Shock/impact loads only. From the results, it is evident that the displacement values of Spring Steel ASTM A313, Spring Steel ASTM A228 and Beryllium Copper are nearest to acceptable range. Beryllium Copper based spring takes beyond 70ms to damp the vibrations whereas both the grades of Spring Steel damp the vibrations within 50ms.
7. These results clearly show that Copper based Alloys and Titanium alloys though have a lighter weight advantage, they do not possess the required properties to achieve acceptable performance characteristics. Commercial viability of using titanium alloy based springs for automobiles is also questionable.
8. From the results it is also evident that Spring Steel based springs are most suitable, yet there is a scope for improvement in the design of the spring to make it stiffer. This will result is lower displacements in static and dynamic conditions as well as the resonant frequencies will be shifted away from low engine frequency values.

9.1 Comparison of Existing Design Vs. Redesigned Spring

With the modified properties all the four analyses viz., Structural, Modal, Frequency Response and Transient Analyses are repeated. The results of the existing design and the modified design are compared and tabulated as below.

Table 9.2: Comparison of results of Existing Design Vs. Redesigned Spring

Parameter	Existing Design	Modified Design	Remarks
Linear Static Analysis Disp. (mm)	27.85 mm	16.56 mm	40% reduction in the Static Displacement
Static Analysis Max. Stress (MPa)	605.6 MPa	374.3 MPa	38% reduction in the Max. Stress
Normal Modes (Hz)	25, 77, 77.5, 107.4, 112.8	30.1, 84.7, 85.5, 120.2, 124.9	No low frequency ranges nearest to 20Hz
Frequency Response	59.9mm @ 26Hz	39.5mm @ 30Hz	34% decrease in the max. displacement. Resonant Frequency has moved to 30Hz
Transient Response Disp. (mm)	41.5 mm	29.7 mm	28% reduction in max. displacement during impact loads.

From the results, it can be noted that there is a considerable reduction in the Static deflection and the max stress values of the modified design by 40% and 38% respectively. The low frequencies observed in the existing design have shifted to 30Hz and above eliminating the low frequency bands which usually tend to resonate with the engine frequencies.

The maximum displacement observed in cyclic loading at 30Hz also has reduced by 34%. Impact loading also has shown an improvement by a 28% reduction in maximum displacement values. In all the parameters considered, the modified design is showing better results. On the down side, there will be a marginal increase in the kerb weight of the vehicle as the weight of the spring is going to increase.

11. Conclusions

In this project, a Shock Absorber of an existing model of a 150 cc Two-wheeler has been designed using Solid Works-2014. To validate the strength of the existing design, Linear Static and Modal Analysis were performed. To validate the dynamic performance of the Shock Absorber, Harmonic (Frequency Response) Analysis and Transient Response Analysis were performed using HyperWorks packages. The material of the Helical Spring was varied for five different materials and the Analyses were performed. From the results, the following conclusions are made

1. It is observed that, Copper based alloys and Titanium alloy based materials for springs are not suitable for Two Wheeler Shock Absorber applications as their spring stiffness is too less for automobile applications.
2. Keeping all the other parameters constant, the wire diameter of the spring is increased to the next standard value and the better performing material of ASTM A 228 is selected for the modified spring.
3. The Static deflection has reduced, which in-turn gives more displacement to be available for the spring to deflect during Shock/Impact loads.
4. The resonant frequencies are moved to 30Hz. There are no frequencies below the low frequency ranges of 0-20Hz.
5. The reduced deflection in Transient Analysis indicate that the modified Spring is stiff compared to the existing design. This will cater to higher payload on the two wheeler and gives sufficient room for deflection of the Shock Absorber when going over bad roads.
6. As per the above study conducted, wire diameter of spring with 9.125mm and material for spring as per ASTM A 228 is found to be a better choice for the spring of the shock absorber as when compared to the existing model.

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