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Research Paper Engineering

Design & Structural Analysis of an Automobile Independent Suspensions type Mac-Pherson Shock Absorber

*Vandana Y. Gajjar ** Nihit Soni, Chauhan Sagar, Shaikh EzazAhmed,Surti Pratik

*Assist Professor, Shri S.V.M. Institute of Technology, Bharuch

**U.G. Students, B.E.-IV Year, Mechanical Engineering SVMIT, Bharuch

ABSTRACT

In this article, the effort is made to design and analyze an automobile Mac-Pherson Absorber model. Mac-Pherson strut suspension most widely used independent suspension system on vehicle, The Mac- Pherson strut is a further development of double wishbone suspension. With this type of suspension, the shock absorber, strut and spindle are a combined unit, which is supported by the coil spring at the upper end and the lower control arm The telescopic shock absorber also serves as a link to control the position of the wheel. Therefore it saves the upper control arm. The main advantage of the McPherson strut is that all the parts providing the suspension and wheel control can be combined into one assembly. Applying various loads and constrains for the structural analysis of the Mac Pearson Shock Absorber to compare the various results with the actual design & variables affecting during the operation to prevent the fatigue failure of the Independent Suspension Shock Absorber. For doing this various assumptions has been made.

Keywords : Automobile, Fe-analysis, Independent, Mac-Pherson Shock-absorber, Suspension

I.Introduction

Mac-Pherson strut suspension most widely used independent suspension system on vehicle, The Mac- Pherson strut is a further development of double wishbone suspension. The upper transverse link is replaced by a pivot point on the wheel house panel, which takes the end of the piston rod and the coil spring.I2 .With this type of suspension, the shock absorber, strut and spindle are a combined unit, which is supported by the coil spring at the upper end and the lower control arm The telescopic shock absorber also serves as a link to control the position of the wheel. Therefore it saves the upper control arm. Besides, since the strut is vertically positioned, the whole suspension is very compact. With this it includes, the vertical movement is constrained by the telescopic pivoted link (damper) and compliance is provided by a coil spring. Lateral constraint is provided by the lower transverse arm and longitudinal constraint is provided by the longitudinal link.I1 The main advantage of the McPherson strut is that all the parts providing the suspension and wheel control can be combined into one assembly.I2.As can be seen in Fig.No.1, I3.

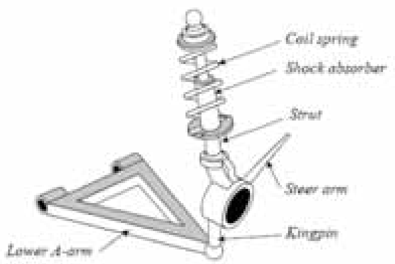

Fig.1 Mac-Pherson Strut Shock Absorber

II. Function of an automobile Mac-Pherson Strut Shock Absorber

A shock absorber is hydraulic piston pump which convert motion energy into heat energy. shock absorber prevents the coil from oscillating after it hits a bump in the road. It prevents this unwanted spring motion by forcing hydraulic fluid through tiny holes in a piston located inside its chamber. This restricted fluid prevents sudden movement and helps isolate the chassis and passenger compartment from the irregular road surface. It provides resistance on both its compression cycle and its extension cycle. Without this dampening effect the vehicle would feel like its floating down the road as shown in Fig. No.2, I3.

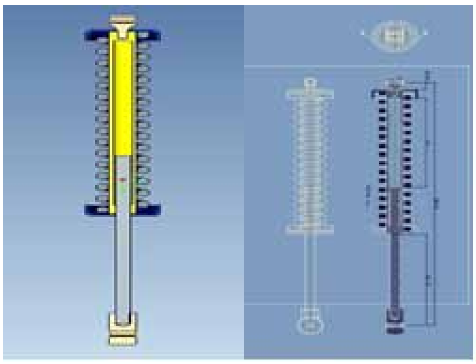

Fig.2.Car Mac person Strut Assembly & Drawing

III.Basic criteria for designing the Macpherson suspension system

3 Design input

The Macpherson system is a consisting of the main two parts which are as :

I. Hydraulic Cylinder
II. Hydraulic Piston and Piston Rod
III. Helical Round Coil Spring

The weight of the Traversal Car is the 1500 Kg. and the weight carrying capacity of the car is the 1500 Kg. So the total weight of the car:

Wc: 3000 x 9.81 : 29430 N

For the design calculation, here we are adding the 25% standard safety on the total weight of the tavera car.

Wc: 36787.50 N

This total weight is carried by the total 4 nos. of suspension system. So the total weight carried by the one suspension system are :

Wc: 9196.875 N

In the design calculation of the suspension system this total load acted on the helical spring as well as the hydraulic cylinder. So here we are design the Hydraulic cylinder and helical spring both for the 9196.875 N.

3.1 Design of Hydraulic Cylinder Assembly
Load Carried by Hydraulic Cylinder: 9196.87 N

Generally cylinders are made from the C50 and C55 materials. The cylinder is manufactured from the casting method and the C50 means cylinder material with 0.50% carbon.

Table. 1 Material Property of the Cylinder C50

Specific Weight (w)	0.0783 gm/cm3
Melting Point	1470 ˚C
Modulus of Elasticity (E)	2.040 * 105 N/mm^2
Modulus of Rigidity (G)	0.890 * 105 N/mm^2
Thermal Conductivity	0.110 cal/s cm ˚C
Coefficient of Linear Expansion	11.10 µm/m˚C
Poisson's Ratio (v)	0.30
Tensile Strength (σt)	660 – 780 N/mm^2
Yield Strength (σy)	380 N/mm^2
Brinel Hardness	241 BHN

From the above data, the Ultimate Tensile Strength for the C50 material is 660 – 780 N/mm^2. For the Design purpose Ultimate Tensile Strength is 660 N/mm^2.

For calculate the dimension of the cylinder, consider the factor of safety by 4 to 6 time of the maximum stress.

Allowable Tensile Stress is 110.00 N/mm^2.

Allowable Yield Stress is 63.33 N/mm^2.

3.2 Design of Cylinder

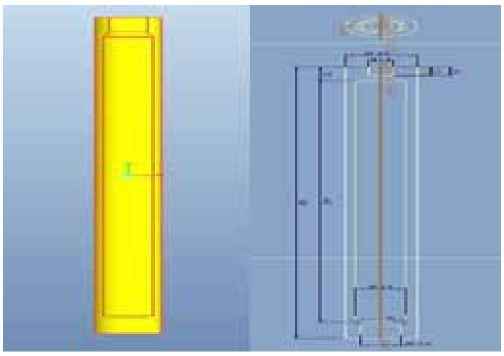

Fig.3 Hydraulic Cylinder Part & Drawing

The load carried by the cylinder assembly is 9196.88 N.

Assuming the inside diameter of the cylinder is 28.00mm.

Now, According to the Simple Pressure relationship, Inside dia. of the cylinder are as under :-

P :- W / A

Now, Inside Diameter of the Cylinder are as under :-

A :- (π / 4) * di^2

A :- 615.75 mm^2

W:- P x A

9196.88:- P x 615.75

P:- 14.936 N/mm^2

So, the require pressure is 15 N/mm2 for the working in the cylinder.

Now, we should find the Thickness of the Cylinder according to two types of the Theory. Which are :-

1.Maximum Principal Stress Theory
2.Maximum Shear Stress Theory
1) Thickness of the Hydraulic Cylinder on the bases of the Maximum Principal Stress Theory t / ri :- { (1 + (P / σt)) / (1 - (P / σt)) }½ - 1

t/ 14 :- {(1 + (15/110.00))/(1 -(15/110.00))}½-1

t :- 2.059 mm

So, According to Maximum Principal Stress theory thickness of the Hydraulic Cylinder is 2.059mm.

2) Thickness of the Hydraulic Cylinder on the bases of the Maximum Shear Stress Theory :-

t / ri :- 1 / { 1 – (2 x (P / τ)) }½ - 1

t / 14 :- 1 / { 1 – (2 x (15 / 63.33)) }½ - 1

t :- 5.297mm

Thickness of the Cylinder is more in Maximum Shear Stress theory. So we should take the t:-5.297mm thickness of the Cylinder for further Designing work. Hence we are taking Thickness t:-6 mm.

Outer Diameter of the Cylinder :- di + (2 x t)

do :- 28.00 + (2 x 6.00)

do :- 40.00mm

Now check the Design of Cylinder for Different types of Stresses generated with the loading condition.

Tangential Stress generated on Inside Surface of the Cylinder:-

σt :- P x ri^2 / ro^2 – ri^2 [1 + (ro2 / ri^2)]

σt :- { 7.5 x 14.00^2 / (20.00^2 – 14.00^2) } x [1 + (20.00^2 / 14.00^2)]

σt :- 43.82 N/mm^2

Generated Stress is less than the Allowable Design Stress. So Design is safe.

Radial Stress generated on Inside Surface of the Cylinder:-

σr :- P x ri^2 / ro^2 – ri^2 [1 - (ro^2 / ri^2)]

σr :- { 7.5 x 14.00^2 / (20.00^2 – 14.00^2) } x [1 - (20.00^2 / 14.00^2)]

σr :- -15.00 N/mm^2

3.3 Design of Piston Rod

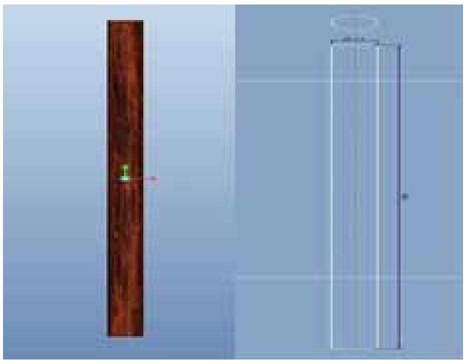

Fig.4 Piston rod Part & Drawing

The stroke of the suspension system is 150mm.

The length of the Piston Rod : Length of the Stroke + 50mm with the assembly requirement of the Hydraulic cylinder assembly.

Lpr : 150 mm

Piston Rod is fixed from the one end and free from the other ends. So the effective length of the cylinder is

L :- l x 2

L :- 300 mm

Load carried by the Piston Rod (Wc) :- 9196.88 N.

Take the Factor of safety 8 for Designing of the Piston Rod.

Buckling or Crippling Load for Piston Rod (Wcr) :- Factor of Safety x W

Wcr :- 73575.04 N

Material of the Piston Rod is Mild Steel. Property for Mild Steel are as under :-

Modulus of Elasticity (E) :- 200 x 103

Crushing Stress (σcr) :- 320 N/mm2

Diameter of the Piston Rod according to Euler's formula:-

Wcr :- π2 x E x I / L2

73575.04 :- π 2 x 200000 x π x d4 / 3002

d :- 6.00 mm

Diameter of the Piston Rod according to Rankin's Formula:-

Wcr :- σcr x A / { 1 + (a x (L / k)2) }

k :- d / 4

Put all these values in equation of the Rankin's Formula :-

73575.04 :- 320 x (π x d2 / 4) / { 1 + ((1 / 7500) x (300 x 4 / d)2) }

d :- 20.62mm

According to Trial and Error method, Diameter of the Piston Rod is 22.00mm.

So, we should consider the Diameter of the Piston Rod which is higher from above two values. So Diameter of the Piston Rod is 22.00mm.

3.4 Design of Piston

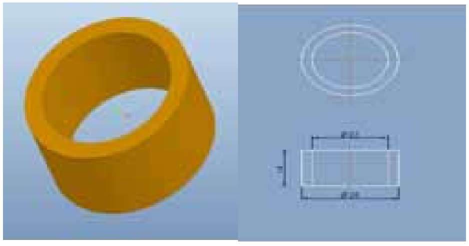

Fig.5 Piston Part & Drawing

Table. 2 Material Property of the Cylinder C60

Specific Weight (w)	0.0783 gm/cm^3
Melting Point	1470 ˚C
Modulus of Elasticity (E)	2.040 * 105 N/mm^2
Modulus of Rigidity (G)	0.890 * 105 N/mm^2
Thermal Conductivity	0.110 cal/s cm ˚C
Coefficient of Linear Expansion	11.10 μm/m˚C
Poisson's Ratio (v)	0.30
Tensile Strength (σt)	660 – 780 N/mm^2
Yield Strength (σy)	380 N/mm^2
Brinel Hardness	241 BHN

From the above data, the Ultimate Tensile Strength for the C60 material is 750 N/mm^2.

For the Structural Member Factor of Safety is 6 to 8 times of the Ultimate Tensile Strength and Yield Strength. So we take the Factor of Safety is 8 for the Load Carrying Member.

Allowable Tensile Stress: - Ultimate Tensile Strength / Factor of Safety

Allowable Tensile Stress is 93.75 N/mm^2.

Allowable Yield Stress: - Ultimate Yield Strength / Factor of Safety

Allowable Yield Stress is 52.50 N/mm^2.

Now, the thickness of the Piston are as under :-

t_h :- Sqrt ((3 * P * di^2) / (16 * σt))

t_h :- 4.0mm

But the thickness 4.0mm is too less by considering the manufacturing consideration as well as the clamping provision for the piston and the piston rod. That's why the thickness of the piston is minimum as to provide the minimum required clamping strength. The minimum diameter of the piston rod is 14.00mm. So the thread provided on the piston rod is M14 size. The minimum clamping strength required to clamp the piston is equal to the thread diameter. So the thickness of the piston is 14.00mm.

3.5 Design of the Flat End Cover

Table. 3. Material Property of the Cylinder C60

Specific Weight (w)	0.0783 gm/cm^3
Melting Point	1470 ˚C
Modulus of Elasticity (E)	2.040 * 105 N/mm^2
Modulus of Rigidity (G)	0.890 * 105 N/mm^2
Thermal Conductivity	0.110 cal/s cm ˚C
Coefficient of Linear Expansion	11.10 μm/m˚C

Poisson's Ratio (v)	0.30
Tensile Strength (σ_t)	660 – 780 N/mm^2
Yield Strength (σy)	380 N/mm^2
Brinel Hardness	241 BHN

From the above data, the Ultimate Tensile Strength for the C50 material is 660 – 780 N/mm^2. For the Design purpose Ultimate Tensile Strength is 660 N/mm^2.

Design Stress for Flat End Cover :- For the Structural Member Factor of Safety is 4 to 6 times of the Ultimate Tensile Strength and Yield Strength. So we take the Factor of Safety is 6 for the Load Carrying Member.

Allowable Tensile Stress :- Ultimate Tensile Strength / Factor of Safety

Allowable Tensile Stress is 110.00 N/mm^2.

Allowable Yield Stress :- Ultimate Yield Strength / Factor of Safety

Allowable Yield Stress is 63.33 N/mm^2.

Load sustain by the Flat End Cover is same as Cylinder :- 9196.88 N

The Force acting on the Flat End Cover :-

W :- di * t_c * σ_t

9196.88 :- 14.00 * tc * 110.00

t_c :- 5.972mm

So, Thickness of the Flat End Cover is 6.00mm

3.6 Design of Helical Round Coil Spring

Fig.5 Helical coil Spring Part & Drawing

In the assembly of the Macpherson suspension system, the spring is fitted on the outside of the Hydraulic cylinder. From the above calculation the outer diameter of the cylinder is 34.00mm. So the inside diameter of the helical coil spring is calculated as under :

Inside diameter of the helical spring Dsi : Outside diameter of the cylinder + minimum clearance required for working of the cylinder

Dsi: 40.00 + 1.00

So the inside dia of the spring Dsi : 41.00mm

The stroke of the helical spring is same as the stroke of the hydraulic cylinder. So the stroke of the helical spring is 150.00mm.

Load acted on the helical spring. W :9196.88N

Inner diameter of the spring Dsi : 41.00mm

Deflection of the spring δ : 150.00mm

The material of the spring is oil tempered wire (basic material of the spring is Carbon Alloy steel). The property of the spring are as under :

Allowable shear stress τ : 525 N/mm^2

Modulus of rigidity G : 80 x 103 N/mm^2

Modulus of Elasticity E : 2.1 x 105 N/mm^2

The spring index of the spring C: 6

From the Spring Index,

C : Dsi + d / d

d : 8.20mm

Outer diameter of the spring Dso:

: Dsi + 2d

: 57.4.00mm

Mean diameter of the Spring Ds:

: Dsi + d

: 49.20mm

Whals factor of the spring Ks

Ks : (4C – 1 / 4C – 4) + (0.615 / C)

: 1.2525

The diameter of the spring according to the Maximum Shear stress calculated as under :

τ : Ks x (8 x W x C) / (π x d^2)

d^2: 1.2525 x (8 x 9196.88 x 6) / (π x 525)

d: 19.00mm

From the new diameter of the spring, the geometrical dimension of the spring are calculated as under :

Outer diameter of the spring Dso : Dsi + 2d

: 41 + (2 x 19) : 79.00mm

Mean diameter of the Spring Ds: Dsi + d

: 41 + 19 : 60.00mm

No. of turns required for working of the spring are calculated are as under :

δ: (8 x W x C^3 x n) / (G x d)

n: (80000 x 19 x 150) / (8 x 9196.88 x 63)

n: 15 Nos.

For support and ground coils, the total no. of coils are as under :

n': n + 2

n': 17 Nos.

Free length of the spring are calculated as under
Lsf: n' x d + δ + 0.15δ
Lsf: 17 x 19 + 150 + 0.15 x 150
Lsf: 495.50mm
Pitch of the coil of the spring calculated as under :
Lsp: Lsf / (n' – 1)
Lsp: 495.50 / (17 – 1)
Lsp: 30.97mm

3.7 Calculation of the Vibration characteristic of the Helical Round Coil Spring :-
Total load carrying capacity of the cylinder :- 9196.88 N

3.3.1 Natural frequency of the Helical Round Coil Spring during the pitching :-
Let the mass moment of inertia of the vehicle (I):-90 Nm^2

I : mk^2 = > k: 0.0101m

Stiffness of the cylinder are calculated as per following calculations:

Movement of the suspension : 0.075m
S_1 : 122625.07 N/m
S_2 : 122625.07 N/m

The wheel base of the car:- 2.685m

from the CG of the car the distance of the front wheel to the CG (l1) : 0.4 x 2.685m
l_1 : 1.074m
l_2 : 0.6 x 2.685
l_2 : 1.611m

The natural frequency of the cylinder for two degree of freedom are calculated as under :

ω2: ½ [C/k^2 + A] ± sqrt {1/4 [C/k^2-A] + B/k^2}

where, A: { S_1 + S_2 } / m

A: {122625.07 + 122625.07) / 9196.88

A: 26.67

B: { $S_1 l_1$ + $S_2 l_2$ } / m

B:{122625.07 x 1.074 + 122625.07 x 1.611) / 9196.8

B: 35.80

C: { $S_1 l_1^2$ + $S_2 l_2^2$ } / m

C: {122625.07 x 1.074 x 1.074 + 122625.07 x 1.611 x 1.611) / 9196.88
C: 49.98

Now substitute the value of the A, B and C in the equation of the natural frequency.

ω2 : ½ [49.98/0.0101^2 + 26.67] ± sqrt {1/4 [49.98/0.0101^2-26.67] + 35.80/0.0101^2}
ω1 : 521.19, ω2 : 468.73

Finding the natural frequency in hobbling up and down of the suspension system :
S: {$(l_1 + l_2)^2$}/{(l_1^2/s_1)+(l_2^2/s^2))
S:{$(1.074+1.611)^2$}/{(1.074^2/122625.07)+(1.611^2/122625.07))

S: 235810.08 N/m

Now the natural frequency of the vehicle is given by :

ωn : sqrt(S/m)
: sqrt(235810.08/9196.88)
: 5.064 Hz

Now fn : ωn / 2π : 0.806
The natural frequency of the suspension is 0.806
Now the damping co-efficient of the cylinder are calculated as under:

Assume the standard value of the ξ : 0.32
Now the critical damping co-efficient :
Cc : 2 x sqrt(S x m)
: 2 x sqrt(235810.08 x 9196.88)
: 93138.97 N/m/s

The damping factor : damping coefficient / critical damping coefficient
C : ξ x Cc
: 0.32 x 93138.94 => C : 29804.47
The damping frequency fd : fn x sqrt(1-ξ2)
fd : 0.806 x sqrt(1-0.322) => fd: 0.764

IV. Methodology used
The modeling of Mac Pherson Strut Shock Absorber is done in Pro-E Wildfire 4.0 and stress analysis is done by Ansys 11.0 By taking various constraints and boundary conditions. The necessary design modifications have also been made to rectify the problems being faced by the designer.

V. Stress Analysis
The following steps are used for problem solving:

5.1. Model Generation
Proper modeling of the parts is very important for getting accurate results of analysis. Creating the parts and its dimensioning scheme are important steps. The components of the Mac-Pherson Strut Shock Absorber were modeled in the part mode of Pro/ENGINEER Wildfire 4.0. The automobile Mac-Pherson Strut Shock Absorber consists of the following main parts.

- Hydraulic Cylinder
- Hydraulic Piston and piston rod
- Helical Coil spring

5.2 Assembly of the Mac-Pherson Shock Absorber.
The assembly of all the components of the Mac Pherson Strut Shock Absorber was done in the assembly mode of Pro/ENGINEER Wildfire 4.0. The placement (or assembly) constraints were used to rigidly bind the components of the Mac Pherson Strut Shock Absorber to their respective positions in the assembly.

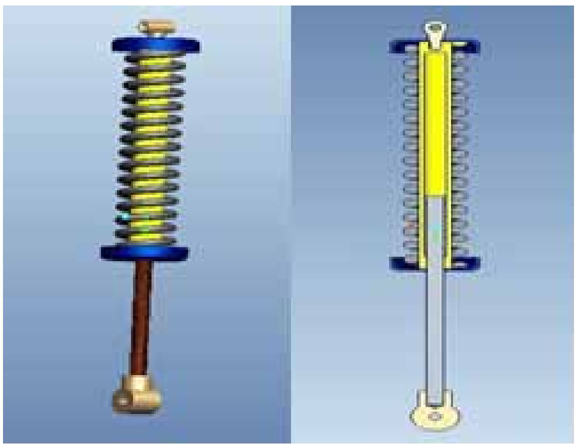

Fig.6 Assembly and front section of the Mac Pherson Strut Shock Absorber.

5.3 Static Structural Analysis
With the wide spread adoption of CAE approach to design, Finite Element (FE) analysis became integrated with the design and analysis procedure. Structural analysis is used to analyze parts and assemblies to find:

• Maximum stresses
• Deformed Shapes (Deformation)

The analysis of a structure during its design process is accomplished by the solution of the partial differential equations that describes the given model.

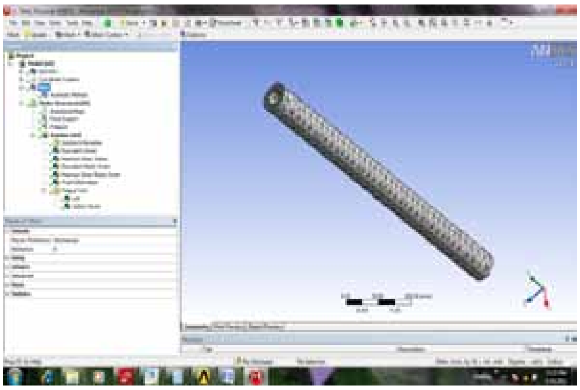

Fig.7 Cylinder Meshing & load application

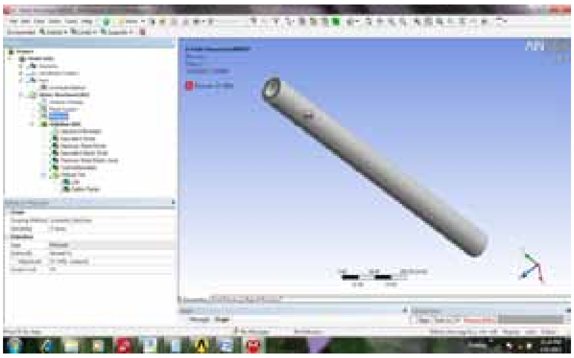

Fig.8 Cylinder Constraints

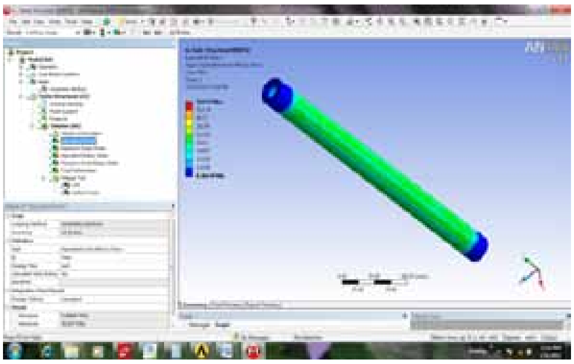

Fig.9 Cylinder Equivalent von Mises Stress analysis

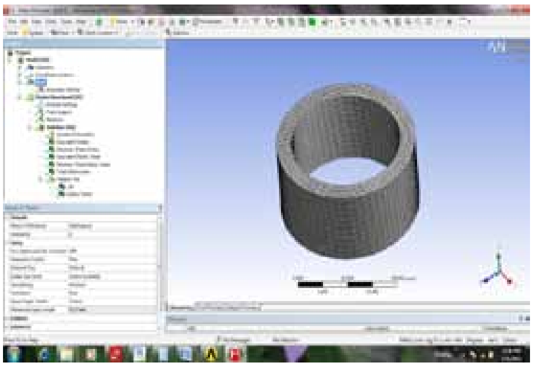

Fig.10 Piston Meshing & load application

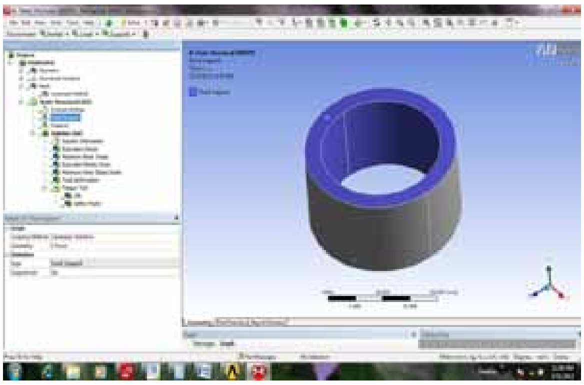

Fig.11 Piston Constraints

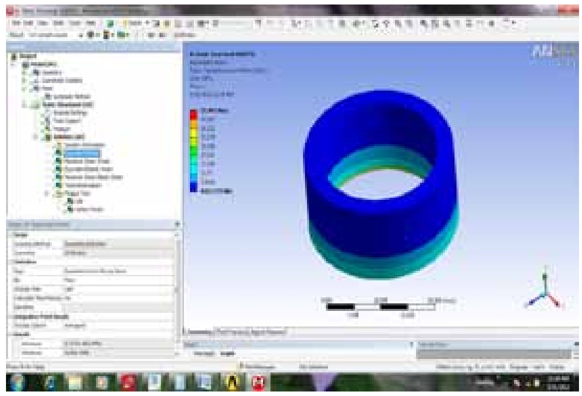

Fig.12 Piston Equivalent von Mises Stress analysis

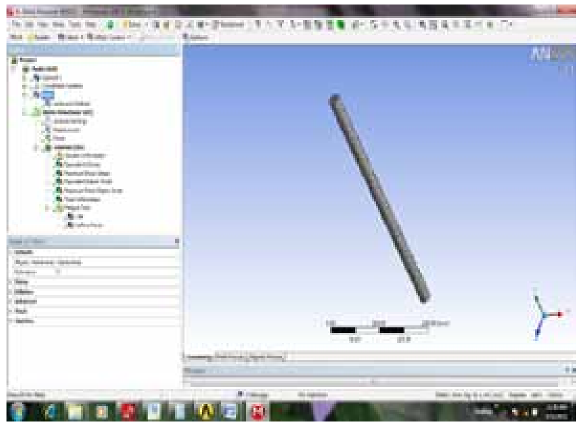

Fig.13 Piston Rod Meshing

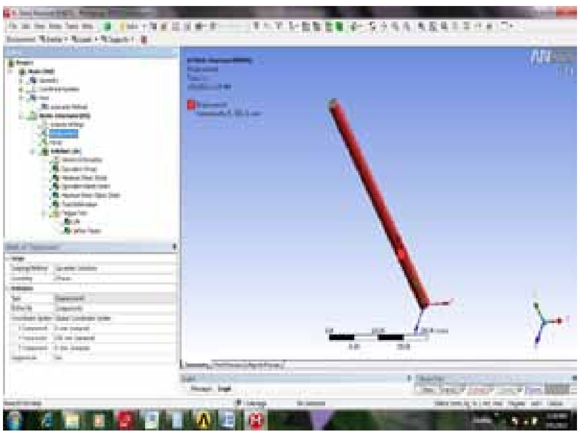

Fig.14 Piston Rod Constraints

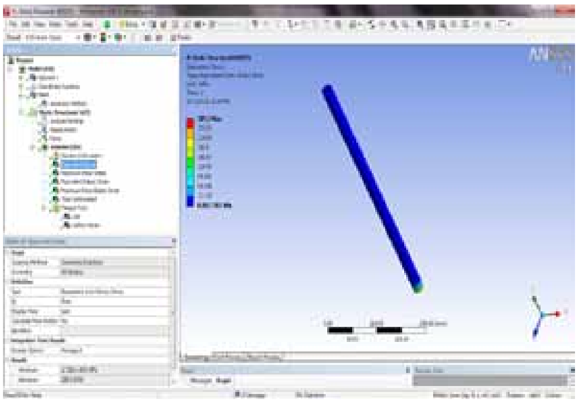

Fig.15 Piston Rod Equivalent von Mises Stress analysis

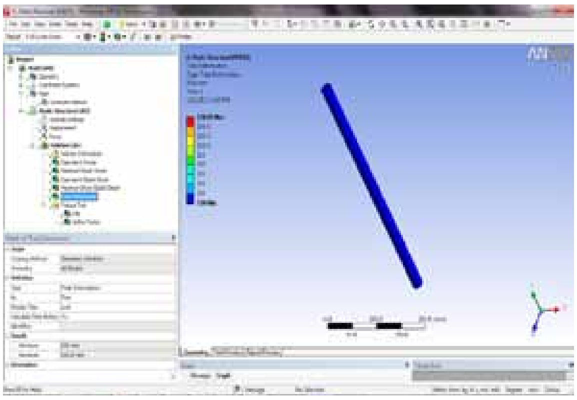

Fig.16 Piston Rod Total Deformation

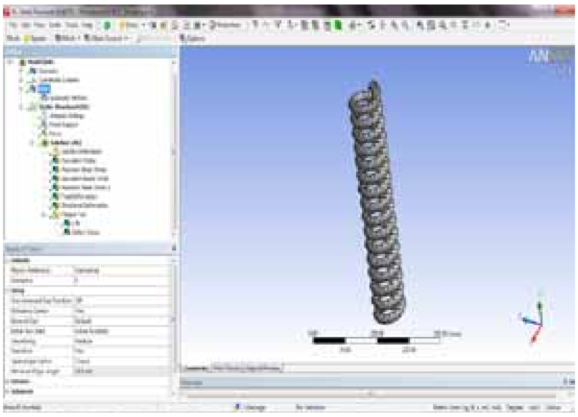

Fig.17 Spring Meshing

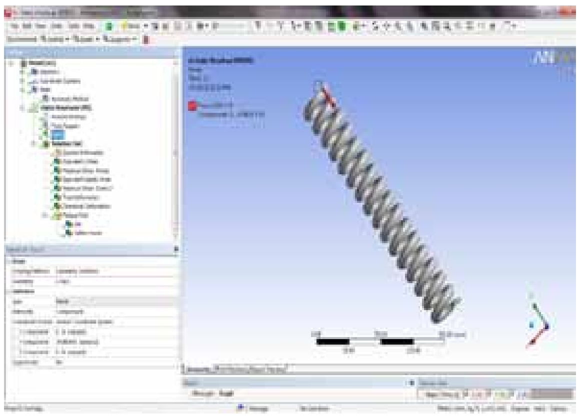

Fig.18 Spring Constraints

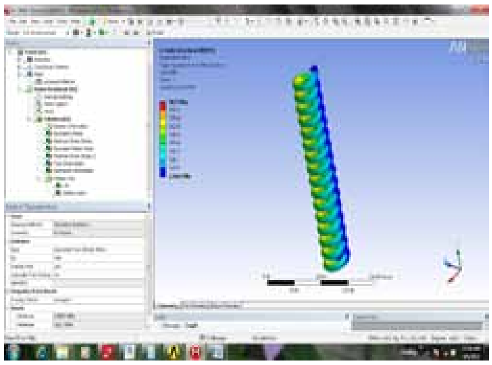

Fig.19 Spring Equivalent von Mises Stress analysis

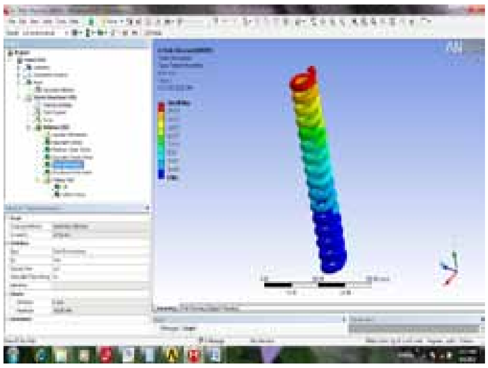

Fig.20 Spring Total Deformation

VI. RESULTS TABLE

Table 4 Cylinder

Type	Equivalent (von-Mises) Stress	Maximum Shear Stress	Equivalent (von-Mises) Elastic Strain	Maximum Shear Elastic Strain	Total Deformation
Minimum	4.6644e+005 Pa	2.6344e + 005 Pa	2.3322e-006 m/m	3.4247e-006 m/m	0. m
Maximum	5.8637e+007 Pa	3.3624e + 007 Pa	2.9318e-004 m/m	4.3712e-004 m/m	3.4516e-006 m

Table 5 Piston rod

Type	Equivalent (von-Mises) Stress	Maximum Shear Stress	Equivalent (von-Mises) Elastic Strain	Maximum Shear Elastic Strain	Total Deformation
Minimum	1738.1 Pa	920.48 Pa	8.6905e-009 m/m	1.1966e-008 m/m	0.15 m
Maximum	2.892e+008 Pa	1.6661e+008 Pa	1.446e-003 m/m	2.1659e-003 m/m	0.15001 m

Table 6 Spring

Type	Equivalent (von-Mises) Stress	Maximum Shear Stress	Equivalent (von-Mises) Elastic Strain	Total Deformation	Directional Deformation
Minimum	2.5663e +006 Pa	1.4635e +006 Pa	1.222e-005 m/m	0. m	-4.891e-002 m

Maximum	1.022e +009 Pa	5.879e +008 Pa	4.8666e-003 m/m	0.16608 m	1.4797e-003 m

Table 7 Piston

Type	Equivalent (von-Mises) Stress	Maximum Shear Stress	Equivalent (von-Mises) Elastic Strain	Maximum Shear Elastic Strain	Total Deformation
Minimum	17178 Pa	9269.3 Pa	8.4206e-008 m/m	1.1814e-007 m/m	0. m
Maximum	5.2903e +007 Pa	2.9675e +007 Pa	2.5933e-004 m/m	3.7822e-004 m/m	3.7939e-007 m

VII. CONCLUSION
The stress analysis of the Mac Pherson Strut Shock Absorber was carried out and it was observed that the stresses induced were found to be well within the allowable/safe limit.

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