ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

Analysis and Optimization of Automotive Shock Absorbers with Integrated

Mushipatla Avinash Reddy
M.Tech (CAD/CAM Student),
Department of Mechanical Engineering,
Sree Dattha Institute of Engineering and Science,
Ibrahimpatnam, Hyderabad, India.

M. Arun Kumar, M.E, (Ph.D)
Assistant Professor,
Department of Mechanical Engineering,
Sree Dattha Institute of Engineering and Science,
Ibrahimpatnam, Hyderabad, India.

ABSTRACT:
Safety and driving comfort for car's driver are both dependent on vehicle's suspension system. Safety refers to the vehicle's handling and braking capabilities whereas comfort of the occupants of a car correlates to tiredness and ability to travel long distance with minimal annoyance. The need for dampers arises due to the roll and pitches associated with vehicle maneuvering, and from the roughness of roads.

Rapidly increasing power available from the internal combustion engine made higher speeds routine; plus the technical aptitude of the vehicle and component designs, coupled with a general commercial mood favoring development provided an environment that led to invention and development of shock absorbers. FSAE is an international design competition that aims at building a prototype of a Formula style race car for the non-professional weekend autocross racer.

Each team designs, builds and tests a prototype based on a series of rules whose purpose is both to ensure onsite event operations and promote clever problem solving. One of the major cause of failure of the suspension system is the failure of shock absorber spring due to fatigue. Taking this into account, our aim is to select an appropriate material which can be used for the shock absorber spring by analyzing certain commonly used materials. This report follows a detailed design methodology and describes the criteria and factors for selection of the materials.

It encompasses several iterations, and establishes reasoning for the most suitable material. The designed shock absorber is a mono-tube hydraulic type. Hence based on the shock absorber used in the race car our project aims at optimizing the material of the shock absorbers. Designing of the model was done on solid works and analysis was performed on ansys. This study explains the groundwork laid for the selection of the best shock absorber material which can be used in the years to come.

KEYWORDS:
Maneuvering, prototype, disturbances, encompass, suspension, automotive, spring

I. INTRODUCTION:
The automotive industry, like any other modern industry, is constantly striving to improve the performance of its vehicles. As automobile engines become increasingly more powerful and compact, the demands placed on the suspension system have increased. This need for optimization and refinement has placed a greater emphasis on accurately measuring and predicting the performance of shock absorbers.

The ability to absorb shocks due to the unevenness on the road surfaces makes shock absorbers the most important part of suspension geometry. So, the knowledge about different materials and types of shock absorbers has been a subject of study and research from many years from the automotive background. The amount of absorption of shocks and vibrations and the elasticity of the various springs got the attention of many.

II. RELATED WORK:

Kartik A. S. at el (2016), studied the shock absorbers for automobile of capacity 150cc by varying material for spring using NX UNIGRAPHICS 10. They then compare the models by analysing structural and modal analysis of the models of Structural Steel, Titanium alloy, Copper alloy and Aluminium alloy material on ANSYS 14.5. They considered the loads, bike weight with single person and 2 persons for the analysis.M.Shobha (2015), designed different models of shock absorbers by varying material for spring using Pro/E Creo and analyzed using ANSYS. She then compared the models by analysing structural and modal analysis of the models of Spring Steel, Phosphor Bronze and Beryllium Copper material on ANSYS. Aim of the study was to find the best material for the spring in shock absorber.P.G. Chaudhary and P.S. Bajaj (2015), analysed and designed models of shock absorbers for Hero CBZ Extreme 150cc by varying material for spring using FEA approach. They used the catia v5 R20 for the modeling. They then compare the models by analyzing structural and modal analysis of the models of Spring Steel and CFRP material on ANSYS Workbench.

Rahul Tekade and ChinmayPatil (2015), designed a shock absorber to improve the comfort and safety of the passengers of the vehicle and also sustain the vibrations. They performed the structural and modal analysis of the shock absorber of the vehicle. They concluded that for the spring ASTM A228 (high carbon spring wire) will provide optimum results.P. Karunakar et al (2014), performed the comparative design analysis of the two wheeler shock absorber and designed the models of shock absorbers by varying material for spring using Creo. Also, they compared the models by analysing structural and modal analysis of the models of Structural Steel (ATM-A316), Inconel X750 and Nickel 2000 material on ANSYS. They conclude that Inconel X750 is best suited material for the spring of shock absorber.Sourabh G. Harale and M. Elango (2014) demonstration of composite material like a combination of conventional steel and a metal matrix composite of E-Glass fibre/Epoxy reinforced material in helical coil spring suspension. The results showed that there was decrease in the weight and increase in stiffness of the system but it also pointed out limitations like cost and manufacturing of E-Glass fibre, low stiffness of single composite spring.Achyut P. Banginwar et al (2014) investigated different models of shock absorbers by varying material for spring using Pro/Engineer. They also performed the structural analysis to validate the strength and modal analysis to determine the displacements for different frequencies for number of modes.SudarshanMartande et al (2013), have developed new correlated methodologies which will enable engineers in designing components of Shock Absorbers by using FEM based tools like ANSYS. They performed the experiment on a bike with 194kg weight, one person and then with two persons. They compared the FEA results with the analytical solutions and found out the errors to be 15%.

Saurabh Singh (2012), has demonstrated the feasibility of composite material for helical coil spring suspension system design. The author has replaced conventional steel with a mixture of steel and Glass Fibre/Epoxy which results in increased stiffness of the spring. The reason of implementing combination of steel and composite material was the low stiffness of single composite spring, which limits its application to light weight vehicle only.Priyanka Ghate et al (2012), attempted to analyse the failure of Freight Locomotive Suspension spring of primary suspension and redesigning of the spring to improve the durability and also the ride index. The results revealed to use a single non- linear spring.Pinjarla.Poormohan and Lakshmana Kishore T. (2012), designed different models of shock absorbers by varying material for spring. They then compare the models by analysing structural and modal analysis of the models of Spring Steel and Beryllium Copper material. They used the ProEngineer for the modelling and ANSYS for the analysis of the shock absorber.Prince Jerome Christopher J. and Pavendhan R. (2010), designed and performed analysis of Shock Absorber performance by varying diameter of the coil spring.

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By considering bike mass, loads, and number of persons seated on a bike, comparison is done to verify best dimension of spring in the shock absorber. They used ProE and ANSYS for the modelling and analysis respectively.Budan, D. Abdul, T. S. Manjunathan (2010), demonstrated the feasibility of composite coil spring like glass fibre, carbon fibre and their mixture over the conventional metal coil spring. The experimental results show that spring rate of carbon fibre spring is much more than the other materials and its weight is also lower as compared to the other composite materials tested.Results revealed that the spring rate of the carbon fiber spring is 34% more than the glass fiber spring and 45% more than the glass fiber/carbon fiber spring. The weight of the carbon fiber spring is 18% less than the glass fiber spring, 15% less than the Glass fiber/carbon fiber spring and 80% less than the steel spring. Thus, we can see that a lot of research has been done on this topic. The ultimate goal of this study as discussed earlier is material selection based on different conditions and to lay a groundwork which can be used as a standard for the selection of material for the shock absorber used in the race car.

A. FORMULA SAE – BACKGROUND:

The Formula SAE competition is a student design competition that is open to undergraduate and graduate students around the world. The students are to conceive, design, and manufacture an open wheel formula style vehicle that is suited for a weekend autocross-racing car. At the competition, cars are judged at both static and dynamic events. The SAE sanctioning body has numerous rules that are strictly enforced due to the nature of the competition. Maximum Displacement of 610CC per cycle and wheel base of at least 60in are the primary mandatory rules for the competition

III. DESIGN &CALCULATIONS:

Approximate vehicle data taken into consideration are as follows

Table 1: Vehicle Data

Parameters	Units	Front	Rear
Weight of the Car	Kg	190	
Weight of the driver	Kg	70	
Total Weight	Kg	270	
Weight Distribution	Kg	40	60
Sprung Weight	Kg	96	144
Un-Sprung Weight	Kg	12	18
Actual Weight	Kg	108	162
Motion Ratio		1.08	1.25
Suspension Type		Double Unequal Length Push-rod actuated	Double Unequal Length Pull-rod actuated
Suspension Design Travel	Mm	30	30
Tire Radius Stiffness	N/mm	15	15
Frequency	Hz	1.5	1.3

A. SPRING -RATE CALCULATIONS

K_S = Spring Rate

m_S = Sprung Mass

K_{cd} = Dynamic Spring Stiffness

$$K_S = 4.\pi^2.fr^2.mgMR^2$$

f_r = Natural Frequency

MR = Motion Ratio (Wheel / Spring Travel)

K_r = Ride Rate K_W = wheel rate

Table 2: Calculations for stiffness

Stiffness Considering	Front Wheel(N/mm)	Rear Wheel (N/mm)
$K_s=4.\pi^2.fr^2.msMR^2$	$K_{sf}=27.973$	$K_{sr}=28.146$
$K_s=F/X$	$K_{sf}=17.65$	$K_{sr}=26.487$
$K_{cd}=F2-F1/X2-X1$	$K_{fcd}=103.4$	$K_{rcd}=123.4$
$K_w=K_s*MR^2$	$K_{wf}=122.5$	$K_{wr}=195.3$
$K_r=K_w*Kt/Kw+Kt$	$K_{rf}=12.58$	$K_{rr}=18.113$

Critical Damping coefficients for various materials was calculated and displayed in table 3

Table 3: Critical Damping

Formulas	Front	Rear
$\omega_{n(s)} = 1/2\pi\sqrt{2K_R/W_s}$	2.43 Hz	2.38 Hz
$\omega_{n(us)}=1/2\pi\sqrt{(K_S+K_T)/W_{us}}$	20.72 Hz	16.92 Hz
$C_{cr(s)}=2\sqrt{K_R\times W_s}$	3297.8 Ns/m	4845 Ns/m
$C_{cr(us)}=2\sqrt{(K_S+K_T)\times W_{us}}$	3124.15 Ns/m	3826.28 Ns/m

High and low values of damping ratio is displayed in table 4 along with the desired values of damping ratio.

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Table 4: Values for damping ratio

Desired Damping Ratio ($C/C_{critical}$)			
Front Low	4.00	Front High	0.90
Rear Low	3.20	Rear High	0.80
C/R Ratio			
Front Low	1.5	Front High	1.428
Rear Low	1.5	Rear High	1.428

Damping rate and spring damping force was calculated for compression and rebound cases.

Table 5: Damping rate and force

Damping Rate			
Compression, Ns/m ($C_{critical} \times \xi$)			
Front Low	13188	Front High	2811.6
Rear Low	15504	Rear High	3060.8
Rebound, Ns/m ($C_{compression} \times$ C/R Ratio)			
Front Low	19782	Front High	3392.47
Rear Low	23256	Rear High	4346.

On the basis of the forces applied by the suspension system and the body of the vehicle can be tabulated in 2 different manners.

i) The compression force experienced by the spring due to the weight of the vehicle.

ii) The rebound force experienced by the spring due the reaction force generated by itself in order to damp the vibration which is being experienced by the vehicle.

Compression Force (Ccompression×v)

Compressive forces calculated at different velocity of the vehicle was calculated for front and rear springs and values of which is displayed in table number 6.

Table 6: Compressive force

Velocity (mm/s)	Front Forces on both the Springs (N)	Rear Forces on both the Springs (N)
1	13.2	15
2	26.37	31
4	52.75	62
6	79.13	93
8	105.5	124
10	131.9	155
12.5	164.8	193.8
100	281.2	306.1
200	562.3	612.2
300	843.5	918.2
400	1124.6	1224.3
500	1405.8	1530.4
600	1686.9	1836.5
700	1968.1	2142.6
800	2249.3	2448.6
900	2530	2754.7
1000	2811	3060.8

Compressive force

Front Force (Max.) = 1405.8N

Front Force (Min.) = 6.59N

Rear Force (Max.) = 1530.4N

Rear Force (Min.) = 7.75N

Rebound Force

Rebound forces calculated at different velocity of the vehicle was calculated for front and rear springs and values of which is displayed in table number 6.

Table 7: Rebound force

Velocity (mm/s)	Front Forces on both the Springs (N)	Rear Forces on both the Springs (N)
1	19.782	23.26
2	39.564	46.5
4	79.128	93.02
6	118.692	139.54
8	158.26	186.05
10	197.82	232.56
12.5	247.275	290.7
100	339.25	434.63
200	678.49	869.26
300	1017.74	1303.89
400	1356.99	1738.5
500	1696.24	2173.1
600	2035.48	2607.7
700	2374.73	3042.4
800	2713.97	3477
900	3053.22	3911.6
1000	3392.47	4346.3

Rebound Force (Crebound×v)

Front Force (Max.) = 1696.23N

Front Force (Min.) = 123.64N

Rear Force (Max.) = 2173.1N

Rear Force (Min.) = 145.35N

B. MODELLING:

The designing and modelling of the shock absorber was done on Solidworks followed by the analysis on Ansys. Figure 1 shows dimensional model of the spring used in the system.

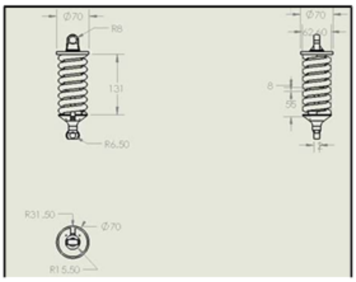

Fig 1: Dimensions of Shock Absorber

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A Peer Reviewed Open Access International Journal

Figure 2 shows the Ansys analysis performed in Ansys 14.5 for four different materials for axial deformation and stress concentration in the spring materials.

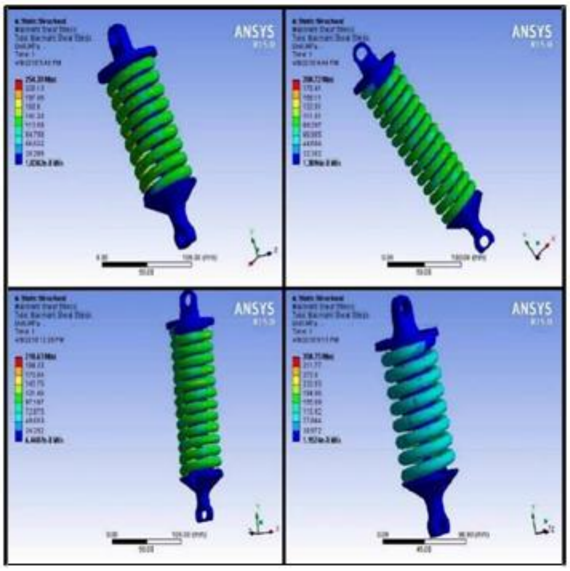

Fig 2: Analysis and testing on Ansys

The analysis and testing of the shock absorber was done with the help of finite elemental analysis (FEA)

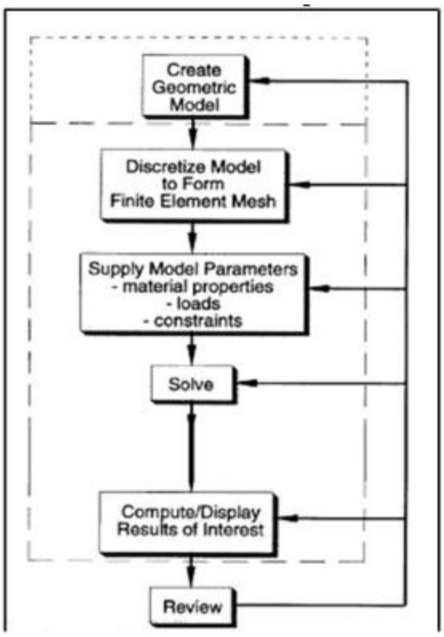

Fig 3: Flow chart for steps inFEA

Figure 3 shows the steps followed for the analysis and problem solving in Finite element analysis implemented in present study.

C .FATIGUE ANALYSIS:

Keeping the outer diameter of the spring constant i.e. D = 50mm, the spring factor can be found out by varying the wire diameter of the spring.

KS = 1+1/2C

Table 8: τm and τa

d (mm)	C (D/d)	Ks	Kw	τ_m (N/mm^2)	τ_a (N/mm^2)
8	6.25	1.08	1.24	22.82	498
10	5	1.1	1.31	11.90	269
10.5	4.76	1.1	1.33	10.28	236
11	4.54	1.11	1.35	9.025	208
12	4.17	1.12	1.38	7.015	164

Taking into consideration various materials for the design of the spring, a number of parameters like the spring constant, various forces acting on the front as well as the rear of the vehicle suspension, minimum length upto which it can hold up the stress, etc.

Design of the spring material is as follows:

Material 1: Chrome-Vanadium wire, ASTM A232

$\sigma_{u/t}$ = 1500 MPa

ρ = 8000 kg/m^3

E = 207 × 10^{3}MPa

G = 79.3 × 10^3 MPa

τ_y = 675 MPa

τ_e = 223.015 MPa

τ_e' = 660 MPa

K = 124 kgf/cm

Table 9: Comparison between Soderberg and Gerber formula

	Soderberg Formula	Gerber Formula
d=10mm	269.37/223.015 + 11.9/675 = 1/n n=0.805	(269.3×n)/223.015+[11.9×n/1500]^2=1 n=0.8278
d=10.5mm	236.24/223.015 + 10.28/675 = 1/n n=0.9306	(236.24× n)/223.015+[10.28×n/1500]^2=1 n=0.9439
d=11mm	208.56/223.015 + 9.025/675 = 1/n N=1.054	(208.56× n)/223.015 [9.025×n/1500]^2=1 N=1.069
d=12mm	164.21/223.015 + 7.015/675 = 1/n N=1.3392	(164.21× n)/223.015+[7.015×n/1500]^2=1 N=1.358

σf = 26.604 N/mm^2.

Number of turns,N = Gd^4/8D^3×K = 13.26 ≈ 13 (plain and ground ends). Solid Length,

$L_S = d \times N_T$ = 156mm.

Free Length,

ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
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Lf = δmax. + δallowance + Ls = 190.5mm.
Pitch,P = 12.8076mm.
But, frequency of applied load is:
fL = V / S = 27.78 Hz.
f should be f > 20 f_L. Hence, SAFE.
Also, δmax. = $8PD^3n/Gd^4$ = 13.439mm

Effect of Bumps on Road:

Every time the vehicle hit a bump or a irregularity on the road, it would bounce up and down over the bump and will continue to induce heavy vibrations. It is therefore necessary that the shock absorber spring of the selected material to keep a check on the deflections and the stresses induced in the spring. Below is calculation for the deflection transmitted by a bump and the stresses induced due to the same is calculated below so as to test the spring material that it sustains the induced stress over a longer cycle of time. Taking a bump of ht=50 mm at 60 KmphTotal spring stiffness= 125*4=500 N/mm Let the length of one cycle of the bump be 1 m.

Base excitation frequency and natural frequency of the bump, Therefore, X=0.436*Y=21.81 mm Thus, a bump of 50 mm is actually transmitted as 21.81 mm to the spring. As seen above in all the four cases of road testing, the maximum force that is transmitted is possible in case of bumps. Thus, we only analyze the stress induced in case of bumps for the already optimized material. Therefore, stress induced in the beryllium copper strip while encountering a bump is calculated as: Since,τmax< [τ]endurancefor Beryllium Copper Spring, it is safe and is ready to used.

IV.COMPARISON AND RESULTS:

The calculations for the axial deformation as well as for the maximum shear stress experienced by various materials of the springs are being compared on the basis of theoretical and Ansys values.

Table 10: Comparison of Theoretical and Ansys Deformation Results

Diameter (mm)	Material	$_{max}$ (mm)		$_{ANSYS}$	
		Compression	Rebound	Compression	Rebound
d=12	Chrome Vanadium	12.09	17.195	12.174	16
	Iconel Alloy-750	12.49	17.7	12.588	16.54
	Carbon Valve Spring	12.09	17.195	11.23	15.129
	Chrome Silicon	12.13	17.2	12.20	15.3
	Stainless Steel	1	18.89	13.812	18.154
	Music Wire	12.09	17.06	11.25	17.168
	Beryllium Copper	12.22	17.37	12.15	15.96
d=11	Chrome Silicon	11.86	16.86	11.845	15.567
	Beryllium Copper	12.98	18.45	12.85	16.82
d=10.5	Beryllium Copper	13.03	18.52	12.588	16.735

Table 10 and Table 11 shows the comparison between the theoretical and Ansys values of shear stress and axial deformation respectively.

Table 11: Comparison of Theoretical and Ansys Shear Stress Results

Diameter (mm)	Material	$_{max}$ (mm)		$_{ANSYS}$	
		Compression	Rebound	Compression	Rebound
d=12	Chrome Vanadium	192.9	274.237	196.99	265.32
	Iconel Alloy-750			200.72	270.07
	Carbon Valve Spring			210.37	265.32
	Chrome Silicon				
	Stainless Steel				
	Music Wire				
	Beryllium Copper			350.75	248.3
d=11	Chrome Silicon	245	348.1	237.17	342.6
	Beryllium Copper			243.54	328.55
d=10.5	Beryllium Copper	277.528	394.525	254.29	398.84

V.CONCLUSION:

We have studied the different types and approaches of shock absorbing materials and their nature under different conditions, and by studying all the types, it is seen that Beryllium Copper is having greater strength and having more shock absorbing capacity than other conventional materials. So, it can be used as an alternative option in future for replacement of conventional suspension with more advantage. As seen from the results table, it is clear that Beryllium Copper (ASTM B197) is found to be the most optimistic material from the above all of the other materials because of the following reasons:

- Evaluating the strength of design, the structural analysis on the shock absorber was carried out in Ansys 14.5 by considering various materials like Stainless Steel, Beryllium Copper, Iconel Alloy, chrome silicon and carbon spring valve.

- Equivalent stress and deformations are noted for different load conditions. The result analysis of these materials is put up in the form of tables for the easy comparison.
- Its elastic properties are also far better than the rest of them.
- It didn't Buckle even after reducing its wire diameter to 10.5mm.
- It could sustain maximum shear stress upto 350.75 MPa.
- It has the least axial deformation of about 12.15mm.
- It takes the least time to return to its mean position once experienced by compression or expansion.

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