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Design and Fabrication of Shock Absorber Compressor Spring

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ABSTRACT

This report contains the design, fabrication and performance analysis of a shock absorber spring compressor. The detailed data of the different parts was presented. The materials selected for each part, the reason for selecting that material and the design of the different parts of the shock absorber compressor spring was carried out. The manufacturing processes involved was presented. From the design it was established that deflection in the spring decreases with reduction in the stresses applied to the spring. The test result of the fabricated shock absorber compressor spring shows that the tool was very efficient in extracting the spring during the repair process with danger of harming the technician working with the tool.

KEYWORDS: *shock absorber, compressor spring, suspension*

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1. INTRODUCTION

1.1. Background of Study

A spring is an elastic object that stores mechanical energy. Spring can be classified depending on how the load force is applied to them: examples are tension springs, compression or coil springs, helical springs, flat spring, torsion spring etc. Springs are typically made of spring steel. A coil spring, also known as a helical spring, is a mechanical device which is typically used to store energy and subsequently release it, to absorb shock, or to maintain a force between contacting surfaces.

Coil springs are commonly used in vehicle suspension. These springs are compression springs and can differ greatly in strength and in size depending on application. A coil spring suspension can be stiff or soft depending on the vehicle it is used on. Coil spring can be either mounted with a shock absorber or mounted separately. Coil spring suspension is used in high performance cars so that the car can absorb bumps and have a low body roll.

The suspension system is one of the most important components in the automobile, it is responsible for dissipating the kinetic energy and controlling the shock. In automobiles, shock absorber decreases the influence of traveling over the harsh road which leads to improved vehicle control and the quality of the ride.

A coil spring compressing tool is use in compressing suspension springs of cars and trucks when maintenance or replacement of shock absorber is needed. Hence this work designed a shock absorber- spring compressor that will be used to compress the coil spring and remove the shock during maintenance or replacement of shock absorber.

1.2. Literature Review

The automobile suspension system basically defines the following functions: suspends the vehicle on its chassis, stabilizes the vehicle, absorbs shock from the road, offers comfort, and it is an essential part of the vehicle safety architecture.

Various suspension systems, for example strut types suspension systems, utilize shock absorbers surrounded by a coil spring. The access space around the coil spring is usually limited, especially on small cars, and prevents the use of a surrounding type spring compressing tool. While various types of spring compressors have been proposed in the prior inventions, these devices have a tendency to shift laterally when in use, creating the possibility of a potentially dangerous release of the compressed spring.

Most mechanics face dangers when removing shock absorber springs or coil springs with unfit equipment, and this had led to severe injuries. Also, time spent in removal of the coil spring can be exhausting as equipment used are not efficient and also they cannot manage the shock spring to its full capacity.

A coil spring compressing and spreading tool having opposite directed spring engaging hook members disposed on a threaded adapted for use with vehicle suspension strut assemblies was invented by [1]. It is made of two parallel end plates having apertures size to allow insertion of the tubular shock assembly of the strut. The two end plates were joined by threaded support rods which allow the distance between the two end plates to be varied to compress the spring as desired. Pivoted latches allow the quick insertion or removal of the strut assembly from the spring compressor. the short-coming of this device is the spring been compressed easily pull off the two ends of the plates holding the coiled spring.

[2] developed a spring compressing plates with two U-shaped plate opposite to each other. It has threaded shaft which, when turned by an external force, changes the distance between the spring compressing plates so as to compress the strut spring. This requires a great force to operate and hence did not receive a wide acceptance.

A fluid operated piston-cylinder mounted to an adjustable upper platen to provide compression between a push platen and a lower base platen. It has a fluid-operated piston-cylinder which drives a first platen toward a second platen to compress a MacPherson strut mounted between the two platens, the first platen being pivotally mounted for self-adjustment to accommodate many different strut sizes and designs was also discloses by [3]. Another spring compressor which includes a drive mechanism to move a strut clamp assembly toward and away from the front of the frame was designed by [4]. The upper positioning assembly being vertically adjustable independently of the lower positioning assembly but it has a limit as to the size of strut spring it can accommodate for compression.

A spring compressor that has a lower clamping member adapted to be secured to the portion of the cylindrical strut of the MacPherson strut located below the associated helical spring platform was design by [5]. A pair of spaced apart elongate threaded shank members are mounted on the clamping member and extend upwardly where they are threaded and connected to a pair of spaced apart upper spring engaging hooks. The strut compression tools of the prior art are of various types and are widely used. However, these tools suffer from many disadvantages as it requires the use of adapters to accommodate different types of strut assemblies and is bulky and requires a large space around the strut assembly being worked on.

[6] Design a spring compressor. The coil spring compressor tool essentially consists of a main body that houses a screw shaft, along with two built-in jaw bases. The top jaw base is stationary, similar to an anvil on a micrometer. The lower jaw changes distance position as the tool is adjusted, via a large male hex drive at the lower end (this accepts a 1-1/16 in. or 27 mm socket wrench). The draw very back of this invention is that the tool is very heavy and not portable.

An improved coil spring compressor for use in the assembly of automotive suspension systems, was developed by [7]. This coil spring compressor has a movable head plate, provided above the base for movement toward and away from coil spring mounted on a suspension strut. The improvement comprises a pair of spaced guide bars connected to the movable head plate, this prevents sliding movement during compression. Its demerit is that head plate has a limit it can move and hence limits the sizes of coiled spring it work on.

2. Selection Material

Mild steel is selected for the construction of the shock absorber compression spring because of it properties. Its properties include relatively low tensile strength, malleable and surface hardness, which can be increased through carburizing. It is used where ductility or softness are important. The mild steel has a density of $7850kg/m^3$, young modulus of 200GPa and poison ratio of 0.303. it is also affordable as its cost low considering other metals.

3. DESIGN OF THE COMPONENTS.

The different components of the shock absorber compression spring are designed. These parts are;
1. Tripod steering handle, 2. Bevel gear, 3. Power Screw bolt, 4. Screw casing, 5. Base, 6. Bearing, 7. Shaft, 8. gear case cover, 9. screw nut and anchor

The spring compressor was designed based on maximum shear stress theory (MSST), because mild steel is a ductile material. MSST states that maximum allowable stress of the material (design) is proportional to the ratio of yield strength of the material and factor of safety. Hence;

$$\tau_y = \tau_{max} \cdot 2n \qquad (1)$$

For shear stress

$$\sigma_y = \sigma_{max} \cdot n \qquad (2)$$

For tensile strength

Shear strength for mild steel σ_y = $143*10^6$ N/m^2 is τ_y The yield strength in tension of mild steel = $250*10^6 N/m^2$ N Were **n** is the factor of safety = 4. τ_{max} is the maximum allowable stress of the design in shear.

σ_{max} is the maximum allowable stress of the design in tension.

3.1. Force acting on the spring

Force acting through the machine (compression force at the spring) can be calculated using hooks law equation

$$W = kx \ (N) \qquad (3)$$

Where **k** is spring stiffness or constant. **x** is the change of length of the spring, for maximum extension 100mm were used as the maximum compression length = 0.1m

$$k = \frac{w}{x} = \frac{Gd^4}{8D^3} \ (N/m) \qquad (4)$$

Where **G** is the modulus of rigidity of the spring material made of high carbon steel G = 80×10^3 N/m^2. **D** is the diameter of the coil spring = 111mm = 0.111m. **d** is the diameter of the spring material = 13.4mm = 0.0134m

$$k = \frac{Gd^4}{8D^3} = \frac{80\times10^3\times0.0134^4}{8\times0.111^3} = 0.2357N/m$$

$$W = 0.1\times0.2357 = 0.05287N$$

3.2. Design of tripod wheel handle

This is three circular bars moved in circular motion to convert human power to rotational motion of the pinion. The bars are made up of mild steel. The force which is transferred from human power to the handle is proportional to the spring stiffness The input torque **T** through the wheel handle can be calculated as

T = W.D (Nm) (5)

Where W is the axial force compressing the spring (N). D is the the diameter of handle = 500mm = 0.5m

T = 0.02857 $\times$ 0.5 = 0.0114285Nm

Stress **σ** acting on the wheel due to bending moment can be calculated as

$\sigma = \frac{my}{I} (N/m^2)$ (6)

Where **y** is the length of the wheel bars = 235mm = 0.235m

The bending moment **M** acting on the bars = w. y (Nm) (7)

M = 0.02857 * 0.235 = 6.71×10^{-3} Nm

The Moment of inertia **I** can be calculated as = $\frac{\pi d^4}{64} (m^4)$ (8)

d is the diameter of the bars = 18mm = 0.018m

$I = \frac{3.142 \times 0.018^4}{64} = 8.835 \times 10^{-4}\, m^4$

$\sigma = \frac{6.71\times 10^{-3} * 0.235}{8.835 \times 10^{-4}} = 1.78478\ N/m^2$

To calculate the maximum stress τ_{max} we apply the maximum shear stress theory below because the mild steel is a ductile material

$\tau_y = \tau_{max} \cdot 2n$

$\sigma_{max} = \frac{\tau_y}{4n} = \frac{250 \times 10^6}{4 \times 2} = 31250000$

1.78478 < 31250000

Therefore, the material will not fail under the influence of the applied stress

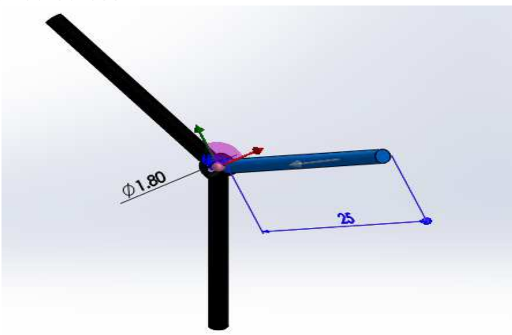


Fig. 1 Tripod Steering handle

3.3. Design of power screw (bolt):

This is a threaded metallic circular bar made of mild steel. It is in mesh with the pinion gear and receive the rotational motion of bevel gear and covert it to linear motion which is used in compressing the spring. The linear distance travelled by the screw bolt is equivalent to the change in length of the compressed spring. Bolt are design on the bases of direct tensile stress and shear stress. Direct tensile stress or compressive stress acting on the bolt due to axial force is given as;

$\sigma = \frac{W}{A_C} (N/m^2)$ (10)

Where A_C is the core area of the bolt = $\frac{\pi}{4} {D_c}^2$. And D_c is the core diameter of the bolt = 27mm =0.027m

$A_C = \frac{\pi}{4} 0.027^2 = 5.72 \times 10^{-4}\, m^2$ $\sigma = \frac{0.02857}{5.72 \times 10^{-4}} = 49.94 N/m^2$

Torsional stress $\tau = \frac{16T}{\pi {D_c}^3} (N/m^2)$ (11)

$\tau = \frac{16 \times 0.0114285}{3.142 \times 0.027^3} = 2957.1 N/m^2$

Since the screw is subjected to both axial and torsional shear stresses, the maximum shear stress will be calculated as given below

Maximum shear stress acting on the bolt $\tau_{max} = \frac{1}{2}\sqrt{\sigma^2 + 4\tau^2} (N/m^2)$ (12)

$\tau_{max} = \frac{1}{2}\sqrt{49.94^2 + 4 \times 2957.1^2} = 2957.2\ N/m^2$

And the maximum tensile stress $\sigma_{max} = \frac{\sigma}{2} + \frac{1}{2}\sqrt{\sigma^2 + 4\tau^2} (N/m^2)$ (13)

$\sigma_{max} = \frac{49.94}{2} + \frac{1}{2}\sqrt{49.94^2 + 4 \times 2957.2^2} = 2982.1\ N/m^2$

$\tau_y = \tau_{max} \cdot 2n$ $\sigma_{max} = \frac{\sigma_y}{n} = \frac{143 \times 10^6}{4} = 35750$

2982.1 < 35750

$\tau_{max} = \frac{\tau_y}{4n} = \frac{250 \times 10^6}{4 \times 2} = 31250000$

2957.2 < 31250000

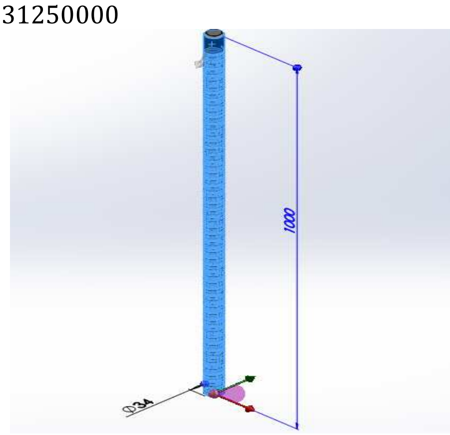


Fig. 2 The power screw bolt

3.4. Design of shaft:

This is the circular bar that transmit the rotational motion of the tripod handle to the rotational motion of the bevel gear, it is also made of mild steel.

The shaft is subjected to a torsional load

Shear stress τ acting on the shaft = $\frac{16T}{\pi d^3}$ (N/m^2) (9)

d is the diameter of the shaft = 30mm = 0.024 m

$\tau = \frac{16 \times 0.0114285}{3.142 \times 0.024^3} = 4210.4 N/m^2$

$\tau_y = \tau_{max} \cdot 2n$

Were **n** = the factor of safety = 4. τ_y is the yield strength of mild steel= 250×10^6 N/m^2

$\sigma_{max} = \frac{\tau_y}{4n} = \frac{250 \times 10^6}{4 \times 2} = 31250000$

4210.4 < 31250000

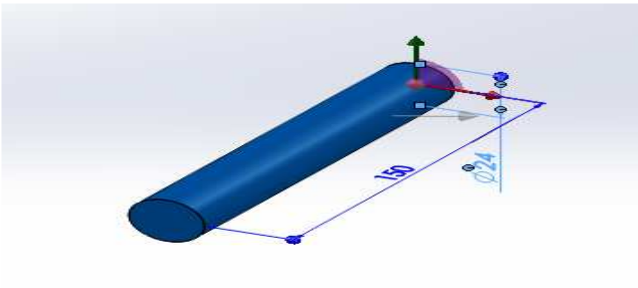


Fig. 3 The shaft

3.5. Design of Srew nut and anchor:

This part moves along the srew bolt, holds and pushes the spring thereby compressing it, it is made up of mild steel.

Average shear stress acting on the nut

$\tau_n = \frac{w}{\pi Dbn} (N/m^2)$ (14)

Where **D** is the major diameter of the power screw = 34mm = 0.034m

n is number of thread in engagement = 8. **b** is the width of the bolt = 5mm = 0.005m

$\tau_n = \frac{0.05287}{3.142 \times 0.005 \times 0.034 * 8} = 12.37 N/m^2$

$\tau_y = \tau_{max} \cdot 2n$

Were **n** = the factor of safety = 4. τ_y is the yield strength of mild steel= $250x10^6$ N/m^2

$\tau_{max} = \frac{\tau_y}{4n} = \frac{250 \times 10^6}{4 \times 2} = 31250000$

12.37 < 31250000

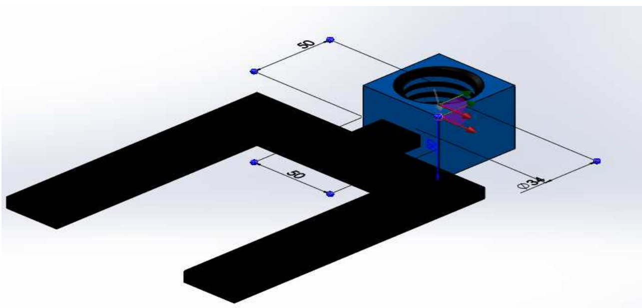


Fig. 4 Screw nut and anchor

3.6. Strength of Gear Teeth

The maximum value of the bending stress **σ** (or the permissible working stress acting on the gear teeth) can be calculated as

$\sigma = \frac{My}{I} (N/m^2)$ (16)

M = Maximum bending moment acting on the teeth = $W_T \times h$ (Nm) (17)

h = Length of the tooth = 9mm = 0.009m

$\boldsymbol{W_T}$ Tangential load acting at the tooth = $\frac{T}{r_p}$ (N) (18)

Where the r_p is the average pitch radius of the gear = 100mm = 0.1m

$W_T = \frac{0.0114285}{0.1} = 0.114285N$

$M = 0.114285 \times 0.009 = 1.02 \times 10^{-3} Nm$

I = Moment of inertia about the center line of the tooth = $\frac{bt^3}{12} (m^4)$ (19)

b = Width of gear face = 15mm = 0.015m. **t** is the thickness of the teeth = 3mm = 0.003m

$I = \frac{0.015 \times 0.003^3}{12} = 3.375 * 10^{-11} m^4$

y is the Half the thickness of the tooth = $\frac{t}{2}$ = 0.0015m

$\sigma = \frac{1.02 \times 10^{-3} \times 0.0015}{3.375 \times 10^{-11}} = 45333.3 N/mm^2$

$\tau_y = \tau_{max} \cdot 2n$

Were **n** = the factor of safety = 4

τ_y is the yield strength of mild steel= $250x10^6$ N/m^2

$\sigma_{max} = \frac{\tau_y}{4n} = \frac{250 \times 10^6}{4 \times 2} = 31250000$

45333.3 < 31250000

3.7. Dynamic Load Rating for Rolling Contact Bearings:

This is a circular ring, it support the srew bollt and enhaces free movement of the bolt. The approximate rating (or service) life **L** of ball or roller bearings is based on the fundamental equation.

$L = (\frac{C}{W})^k$ (revolution) (15)

where **L** = Rating life, **C** = Basic dynamic load rating for a ball bearing with external diameter of 35mm and a bearing number of 300 is 6.3KN = $6.3x10^3$N. **W** = Equivalent dynamic load,**k** = 3, for ball bearings,

$L = (\frac{6.3 \times 10^3}{0.05287})^3 = 1.69 \times 10^{15}$revolution

That is the number of revolution the bearing will run before one of the ball will fracture

Fig. 5 The ball bearing

3.8. Strength of weld at the anchor joint

If σ is the allowable tensile stress for the welded metal (mild steel), then the tensile strength of the joint = 400 $\times 10^6 \mathrm{N}/m^2$. Then the tensile strength of the joint for single fillet weld can be calculated as

P = Throat area × Allowable tensile stress = t × *l* × σ (Ns) (20)

t is throat thickness = *s* × sin 45° = 0.707 *s*(m) (21)

Where s is the Leg or size of weld = 5mm = 0.005m
t = 0.707 $\times$ *0.005* = 3.5 $\times 10^{-3} m$. ***l*** is the Length of weld = 150mm = 0.15m

P = 3.5x10^{-3} x 0.15 x 400 $\times$ = 155400N. $W < P$ the welded joint is safe

3.9. The bevel Gear (driver and driven)

The pinion gear transfers rotational motion of the steering handle to linear motion of the screw bolt. It is made up of mild steel. The extent of rotation of the pinion determines the linear movement of the rack.

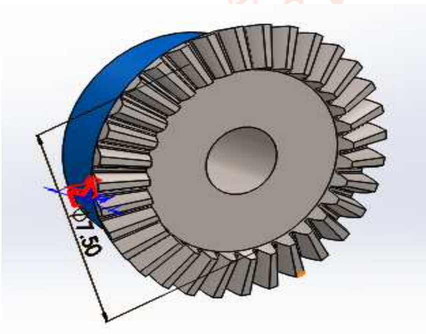


Fig.6 The bevel Gear

3.10. Screw casing

This is a hollow shaft with a square base, it protects and covers the screw bolt.it is made up of mild steel and has the dimensions as shown below.

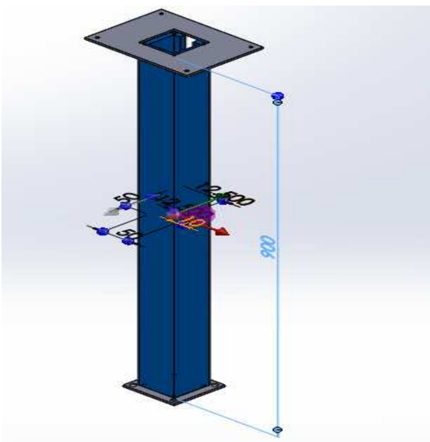


Fig.7 The screw casing

3.11. The base

This is a tripod stand made of angle bares, it holds and support the entire job and transmit all the forces acting on it to the ground. It is made up of mild steel.

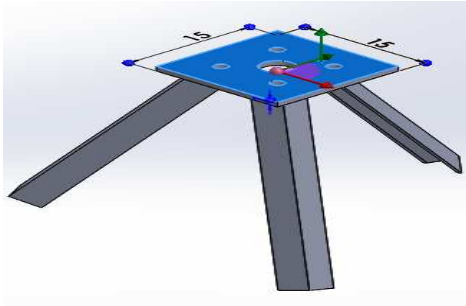


Fig.8 The base

3.12. Gear case cover

This is a square box that covers and protects the bevel gear. It si made up of mild steel.

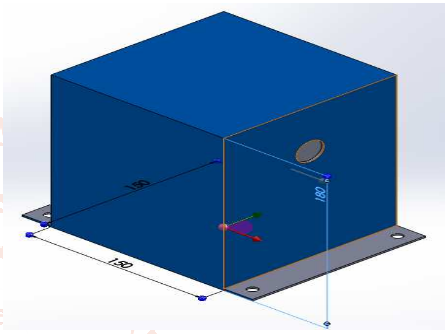
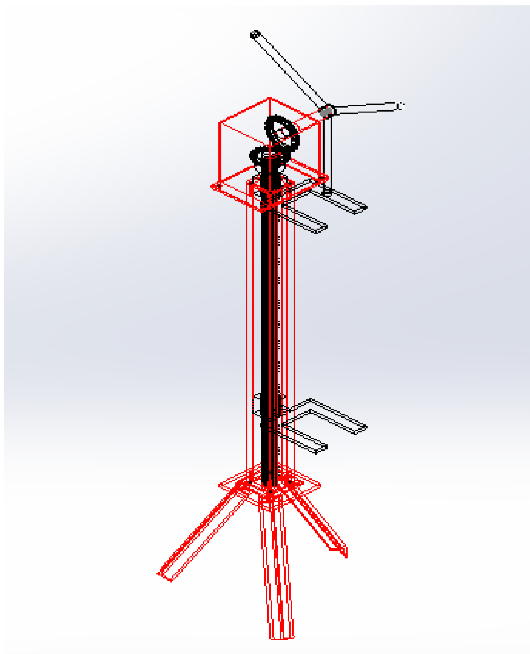


Fig.9 Gear case cover

4. Fabrication

The various designed components were constructed following the following processes; 1. Cutting operation. 2. Bending operation. 3. Threading operation. 4. Machining. 5. Gear cutting. 6. Assembly and Welding 7. Finishing.

After fabricating the various parts, they were coupled to get the shock absorber compression spring shown below.



10 Schematic view of shock spring compressor

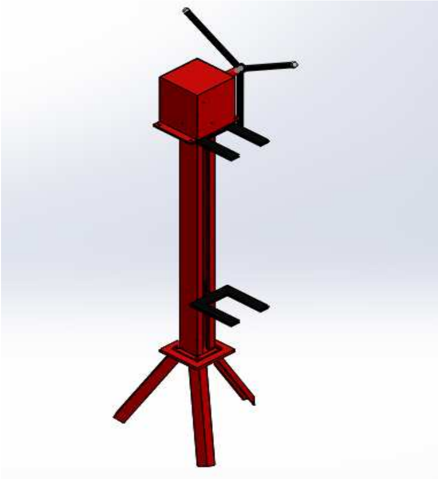


Fig.11 Shock absorber spring compressor

5. TESTING OF THE SPRING COMPRESSOR

Three different shock absorber spring with different young modulus and diameter were used to test the machine, a tale of force and change in length were obtain as shown below

Let L_1 be initial length before compression, L_2 = final length after compression

F = applied force on the spring, E = energy stored in the spring

5.1. Spring for Diameter of sprig material d = 0.00125

Mean diameter of sprig D = 0.1m

Modulus of rigidity of spring material = 80×10^3 N/m^2

$k = \frac{Gd^4}{8D^3} = \frac{80*10^3 \times 0.00125^4}{8*0.1^3} = 0.244$

Table1. Spring for Diameter of sprig material d = 0.00125

L_1(m)	L_2(m)	□L=L_1_L_2 (m)	F = k× □L (N)	E = $\frac{1}{2}$F × □L (Nm)
0.25	0.19	0.06	0.0114	$3.4 * 10^{-4}$
0.25	0.17	0.08	0.0195	$7.8 * 10^{-4}$
0.25	0.15	0.1	0.0244	$1.2 * 10^{-3}$
0.25	0.13	0.12	0.0292	$1.7 * 10^{-3}$
0.25	0.11	0.14	0.0341	$2.3 * 10^{-3}$

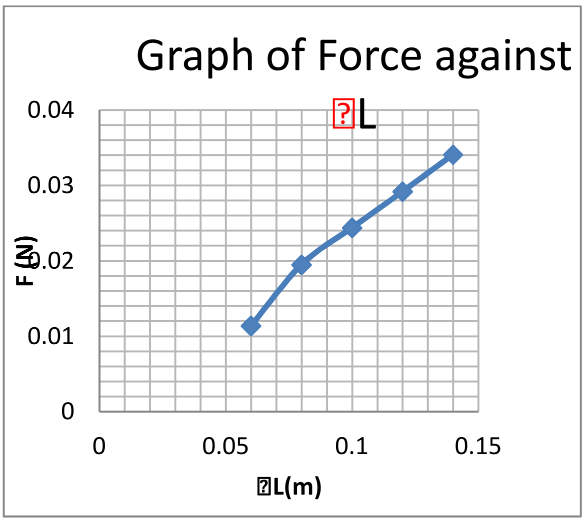


Fig1 graph of force against change in length for sprig Diameter d = 0.00125

5.2. Spring for Diameter of sprig material = 0.013

Modulus of rigidity of sprig material = 80×10^3 N/m^2

Mean diameter of sprig D = 0.115

$k = \frac{Gd^4}{8D^3} = \frac{80\times10^3 \times 0.013^4}{8 \times 0.115^3} = 0.187$

Table2. For Diameter of sprig material = 0.013

L_1(m)	L_2(m)	□L=L_1_L_2 (m)	F = k× □L (N)	E = $\frac{1}{2}$F× □L (Nm)
0.30	0.22	0.08	0.0014	$5.9 * 10^{-4}$
0.30	0.20	0.1	0.0187	$9.3 * 10^{-4}$
0.30	0.18	0.12	0.0224	$1.3 * 10^{-3}$
0.30	0.16	0.14	0.0261	$1.8 * 10^{-3}$
0.30	0.14	0.16	0.0299	$2.3 * 10^{-3}$

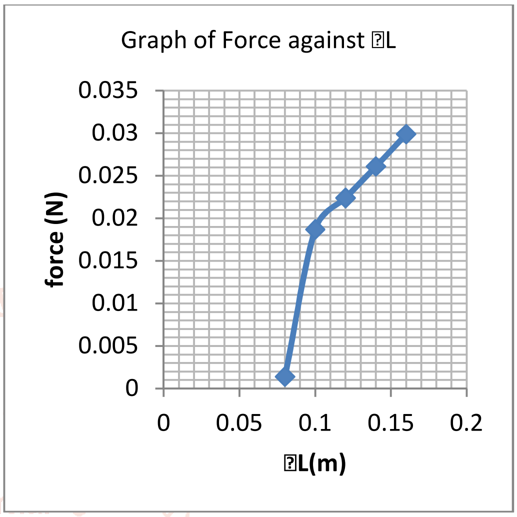


Fig. 2 graph of force against length for sprig Diameter = 0.013

5.3. Spring Diameter of sprig material d = 0.01

Modulus of rigidity of spring material = 80×10^3 N/m^2

Mean diameter of sprig D = 0.105

$k = \frac{Gd^4}{8D^3} = = \frac{80\times10^3 \times 0.01^4}{8\times 0.105^3} = 0.086$

Table3. Spring Diameter of sprig material d = 0.01

L_1(m)	L_2(m)	□L=L_1_L_2 (m)	F = k× □L (N)	E = F×□L (Nm)
0.28	0.18	0.1	0.0086	$4.3 * 10^{-4}$
0.28	0.16	0.12	0.0103	$6.19 * 10^{-4}$
0.28	0.14	0.14	0.0120	$8.4 * 10^{-4}$
0.28	0.12	0.16	0.0137	$1.1 * 10^{-3}$
0.28	0.1	0.18	0.0154	$1.3 * 10^{-3}$

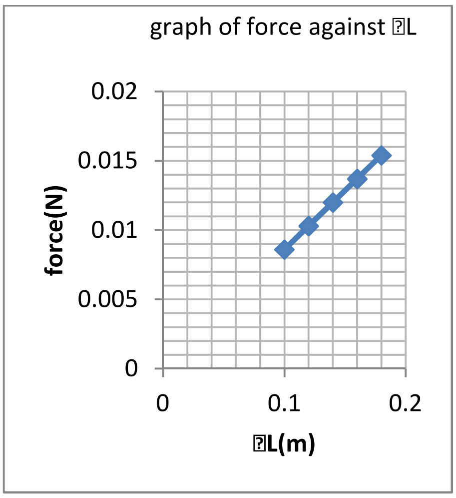


Fig 3 graph of force against change in length for Spring Diameter of sprig material d = 0.01

6. Results and Discussion

The deflection in the spring decreases with reduction in the stresses applied to the spring. The number of turns of spring affects the deflection and stress adversely. As the number of turns increases the deflection decreases and stresses also decreases. This effect is impossible without change in wire diameter as wire diameter has the influence on the deflection and the stresses. Therefore, as the wire diameter increases the deflection and stresses also decreases. This happens due to the better spring index achievement.

7. Summary and Conclusion

This design was carried out in recognition of the present problems experienced by many vehicle owners and auto mechanic shops due to difficulty, accidents and injuries caused during the compression of shock absorber springs. The present work has undertaken the objectives of producing an easy and safe way of removing shock absorber springs. Also, the suitable selection of materials was also taken into cognizance. The various processes need to achieve this work were also highlighted, and this design is meant to solve the problem associated with the compression of coil springs.

From this work, we can simply conclude that the spring compressor is an extremely handy tool for some basic auto repairs. The spring compressor is a fundamental device when it comes to fixing problems related to car suspension. Most people don't realize how dangerous it is to work on the car suspension without using a spring compressor. The purpose of this tool is to extract the coil spring during the repair process. For those who regularly do routine maintenance of their cars at home, learning how to use the spring compressor for common auto repairs is a priceless skill.

To improve the performance of the shock absorber spring compressor I recommend that the strut compressor can be designed to operate automatically using both electrical, electronic and mechanical components like the electric motor, a gear system and a digital display which can display the force applied and other relevant measurement displays. Its control will be automated, thereby reducing the effort in trying to compress the spring.

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