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BASIC DESIGN OF A SAFETY SPRING BACK MECHANISM IN AUTOMOBILE DURING A HEAD ON COLLISION

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ABSTRACT: *The objectives of this research is to initiate the design of a mechanism that will give the vehicle a ball-like behavior and protect both the vehicle and the fragile biological passenger during a head-on collision incident with minimal damage. The component consist of three spring cylinder mechanism behind a light weight non-deformable protective rubber bonded bumper that has a rigid cylindrical thin diameter member, which runs from it to the gear disengagement device designed to withstand an impulsive force of 95000N. Basic scientific theories and equations were used to in the calculations. The severity and impact of head-on collision accidents are such that chances of the people involved, surviving are almost non-existent. The introduction of air power assisted spring cylinder mechanism made the safety spring back mechanism a practical device*

KEYWORDS: safety spring back, helical spring under axial load, maximum deflection, buckling.

INTRODUCTION

Modern Engineering has increased the emphasis on the preservation of materials and energy, placing high value on safe designs. Presently spring back safety mechanism in automobile during a head-on collision forms part of the major safety designs been investigated. Globally there is a need to improve the safety of traffic system for users, and to reduce current inequalities in the risk of incurring road crash injuries. This project is to analyze and simulate the problems of designing of a safety spring back mechanism in automobile during a head-on collision. Among the diverse safety devices, the spring back safety mechanism stands out as the latest safety devices to be introduced in the automobile industry.

Essentially this safety device is designed to augment existing safety devices to yield minimum damage on impact to automobile occupants and materials during a head-on collision. Head-on collision remains one of the worst killers of accidents. The severity and impact of head-on collision accidents are such that chances of the people involved, surviving are almost non-existent. Little wonder why Engineers were mandated to come up with powerful safety device to curb the menace of high death rate through head-on collision during the world Geneva conference. Presently the incessant cases of head-on oriented accidents in Nigeria are subjects of daily discussion and many have lost their lives through it.

Several interest groups including governments have made serious efforts to checkmate and prevent the occurrence of road accidents. These are done through radio, television media, handbills, bill post, speed limits sign, organized seminars and workshop, installing speed control on vehicles etc. The establishment of road safety Corps, use of road signs, drive codes and vehicle inspection officers by the government contributes to the massive efforts towards minimizing accidents rates, yet its rate is still on the increase.

Vehicle manufactures and engineers have tried in several ways to incorporate in the design and fabrication of automobile, devices that can reduce the effect of accidents. Instance of these safety devices include the following, the hanger protector, the rubber bonded bumper, the inclusion of air bags steering wheel but all this can only protect one person or resist a force of low magnitude. Now what happens when one encounters a force of high magnitude on impact? What about the material damage on impact? How can all these safety devices be harmonized to produce a complete safety device? The answers to these questions gave rise to the conception of the basic design of a spring back mechanism.

LITERATURE REVIEW

Early accounts often gave credit to Karl Benz, from Germany, for creating the first true automobile in 1885/1886. The advent of automobile engines alleviated largely the transportation problem of man and improved the economy but the accidents problems associated with it remain unresolved.Ever since the beginning of existence of man on planet earth, man has been relentless in his quest to make himself comfortable. After creating one artifact, he would discover some problems associated with it and then tries to solve it, in most cases he ends up either solving it or creating another out of it. The era of Stone Age brought man into the use of a Stone to make himself a useful implement for fighting, igniting fire and hunting. True, he enjoyed the test of roasted meat from the fire but what about the disastrous effect of it? The means to manage and control the consequences of his invention became a thorn in his flesh.

The era of Civilization introduced man to the world of modern technologies. Mechanized equipment and scientific machines were invented, with these came the invention of automobile as a modern means of transportation. Before this was the era of Antediluvian civilization about 2353 BC when man use cart pulled by horse and caner bicycle as the means of transportation. Carts and chariots were a big advance of legs but they were useless for going across country. These continued in places; like the Roman Empire from the first century BC through the 6th century AD. The Romans were the first to realize that a car is only as good as the road it travels on. Gradually these evolve to the use of coaches with steam engine in England in the 18th century AD to internal combustion engines. From then until date there has been a continuous improvement and modification on these automobile from very simple versions to the more sophisticated ones.

Today there exist in vast types and capacities aircraft for air transportation, ships and boats for sea transportation and automobile for land transportation. Indeed these technological inventions did actually improved man's standard of living but the nature and rate of accidents associated with them have ever since their inventions become the headache of engineers. As it is the modern cars with high safety devices of various capacities and resistances now dominate the automobile industry, still research is currently on going in the field of safety devices in automobile.

Operation Principle

As have been briefly described, the spring back safety mechanism acts as a re-bouncer during direct impact due to a head-on collision of the vehicle with another vehicle or another static or moving body. Its operational principle was derived from the conceptual behavior of a football thrown against a concrete wall. Naturally, when two vehicles travelling in opposite direction collide, they tend to lock together because of deformation. This is so because they are not elastic bodies. From Newton third law both will experience an equal but opposite force on each other but the heavier one will give the less heavy one a greater backward push due to momentum change.

The basic design of a safety spring-back mechanism in vehicles is to convert such vehicles to semi-elastic bodies. The device is made up: non-deformable rubber bumper, curved steel plate, spring base, air-pressured cylinders with piston, air supply valves, helical compression springs, control spring, thin slender rod, control lever, tapered metallic members, the u-channel and bolts, all work simultaneously to spring back the car during collision. During a head-on collision, the bumper receives the impulsive force and transfers it to the spring-cylinder arrangement. The impact force compresses the spring and air in the pneumatic tube. The travel distance of the piston in the pneumatic tube is equal to the distance the spring is been compressed, which is also equal to the travel distance of the slender rod. As the slender rod travels it pushes the clutch fork control level against the clutch release bearing that normally acts on the clutch disc and when this happens the gear disengagement takes place and the clutch fork is locked behind the control spring locking key. The clutch fork remains in this position until the lock release lever is operated upon. The gear engages once again immediately the lock release lever has been operated.

As the piston compresses the air inside the pneumatic tube, a dampening effect takes place, which absorbs a good percentage of the impact force. The spring aids to release the air in the pneumatic tube and returns the bumper to its original position. The car springs back with a controllable speed.
Note all these happen at a very short period, just like the collide impact of two billiards travelling in opposite direction.

METHODOLGY AND ANALYSIS OF DESIGN PROCEDURE AND REQUIREMENTS

When two elastic bodies, travelling in a straight and opposite direction collides, they exert a force on one another, which is equal and opposite in direction. The time interval of this collision is very short, and equal to δt. Both bodies experience an impulsive force, which causes a corresponding change of momentum. But the impulsive of the force is the same in each case and equal to F δt.

The design calculation of this safety mechanism is based on the average weight of a vehicle.

The weight of the empty car = 1302kg [9]

Six people were assume to be inside the and a load in the boot,

The weight of five people in the car = 350kg (70kg each)

The weight of load in the boot = 58kg

The maximum permissible weight of the car with the people and weight in the boot designed as M = 1710kg.

Assuming it takes one second to bring the vehicle car travelling with a velocity of 200km/hr to a halt. It then implies that the acceleration of the car will be

$$Acceleration, a = \frac{Velocity}{time} = \frac{V}{t}$$

$$a = \frac{200 \text{ x } 1000}{60 \ x \ 60} = = 55.5556 \ m/s^2$$

$$Impulsive\ Force, F = mass \text{ x } acceleration$$

$$F = m \text{ x } a$$

$$F = 1710 \text{ x } 55.5556$$

$$F = 95000N$$

Each spring-cylinder takes F_1 load, since there are three spring-cylinder assembly and the force distributed amongst them equally, therefore

$$F_1 1 = \frac{F}{3}$$

$$F_1 = \frac{F}{3} = \frac{95000}{3} = 31666.6667N$$

We assume the ratio of force on the spring F_s, to the force on the cylinder F_c, for the design to be equal to 1:3

$$\frac{F_S}{F_C} = \frac{1}{3}$$

This implies that the force on the spring, F_s

$$F_S = \frac{1}{4} \text{ x } 31666.6667 = 7916.6667N$$

And that on the cylinder F_c

$$F_C = \frac{3}{4} \text{ x } 31666.6667 = 23750N$$

COMPONENTS DESIGN AND CALCULATIONS

Design of the Helical Compression Spring

Mechanical spring is use in machines to exert a force, to provide flexibility, and to store absorbed energy [10]. In general, springs may be classify as either wire spring, flat spring or special shaped spring and there are variations within these divisions. Wire springs include helical springs of round or square shape and are usually design to resist tensile, compressive and torsion loads. The cantilever and the flat spring washers, usually called Belleville springs are included under flat spring.

Spring Material

The spring are manufactured either by hot or cold working process depending upon the size of the material, the spring index, and the properties desired. Generally pre-hardened wire should not be used if $\frac{D}{d} < 4 \; or \; if \; d > \frac{1}{4}$ Numerous spring materials are available to the design Engineer, which includes the followings: alloy steel, corrosion resistant steels, plain carbon steels and non-ferrous materials such as spring brass, beryllium copper, Phosphor bronze and various nickel alloy spring materials. For this work, a chrome silicon wire was selected because of its property as contained in Table 1 shown below;

Table.1 High carbon and alloy spring steel Material

Name of Material	Similar specification	Description
Music wire	UNS G10850 AISI 1085 ASTM A228-51	This is the best, toughest and most widely used of all spring materials for small spring. It has the highest tensile strength and can withstand higher stresses under repeated loading than any other spring material but it is not good to be used at a temperature above 120^0 C or at a subzero temperature, available in diameters of 0.12-3mm
Chrome Silicon	UNS G92540 AISI 9354	This alloy is an excellent material for conditions involving highly stressed spring that require long life, subjected to shock loading, available in diameters of 9.8 -20mm
Hard-drawn wire. 0.6 - 0.70C	UNS G10660 AISI 1066 ASTM A227-47	This is the cheapest general-purpose spring steel and should be use only where life, accuracy and deflection are not too important. It is not good for use above 120^0 C or at subzero temperature, available in diameters of 0.8-12mm.
Oil tempered wire, 0.60-0.70C	UND G10650 AISI 1065 ASTM 229-49	This is a general-purpose spring steel used for the many types of coil springs where the cost of music wire is too excessive and sizes layers that is not available in music wire. It is not good for use at a temperature above 180^0 C and shock/impact loading, available in diameters 0f 3-12mm
Chrome Vanadium	UNS G61500 AISI61500 ASTM 2314-41	This is the most popular alloy spring steel for conditions involving higher stresses than can be used with high carbon steel and for use where fatigue resistance and long endurance are in need. It is also good for shock and impact loading. Can be used for temperatures up to 220^0 C and available in annealed or pre-tempered sizes of 0.8-12mm in diameters.

Spring Index

The spring index is the ratio of the spring mean diameter, D to the spring wire diameter, d.

Spring Index, $C = \frac{D}{d} = \frac{0.1524}{0.02} = 7.62$

Shear Stress Correction Factor

$$K_s = \frac{2C + 1}{2C} = \frac{2 \text{ x } 7.62 + 1}{2 \, x \, 7.62} = 1.066$$

Using the known formula for the tensile strength, the tensile strength of the wire material is calculated

$$S_{ut} = \frac{A}{d_m}$$

$Where\ m = 0.091\ (exponent)$

$A = 1960\ Mpa\ (Spring\ material\ constant)$

d = diameter 0.091m approx. 0.02m

$$S_{ut} = \frac{1960}{(0.02)^{0.091}} = 2798.0943Mpa$$

The maximum permissible torsion stress for static application, S_{sy} is given by,

$S_{sy} = 0.35S_{ut}$ for austenitic stainless steel and non-ferrous alloys

$S_{sy} = 0.35 \text{ x } 2798Mpa = 979.3330Mpa$

The static load corresponding to the yield strength, F_s

$$F_s = \frac{S_{sy}\ \pi\ d^3}{8K_s\ D}$$

Therefore, $F_s = \frac{979.335 \text{ x } 10^6\ \pi \text{ x } 0.02^3}{8 \text{ x } 1.066 \text{ x } 0.1524} = 18938.1735\text{N}.$

Spring Preload

The spring pre load is the minimum force required to compress the spring. The spring pre load is given by

$$F_p = \frac{\pi\ d^3\ \tau_i}{8\ K_s\ D}$$

Where K_s = shear stress correction factor

From table 10.1 in mechanical engineering design by Shigley Edward Joseph, fifth edition, tension stress due to initial tension for steel in helical spring

$\tau_i = 88MPa$

Therefore, $F_p = \frac{\pi \text{ x } 0.02^3 \text{ x } 88 \text{ x } 10^6}{8 \text{ x } 1.066 \text{ x } 0.1524} = 1701.7289\text{N}$

The maximum permissible compression force on the spring is given by

$$F_s' = 7916.6667 - 1701.7289 = 6214.9378\text{N}$$

Since the spring static load corresponding to the yield strength of the wire is greater than the force exerted by the spring by a factor of 2.4, it is within the safety limits.

Spring Constant, K

The spring constant K, is a function of force on the spring F_s' and the solid deflection δ

$\delta = 6inches = 0.1524m$

$F_s' = K\ \delta$

This implies that,

$K = \frac{F_s'}{\delta} = \frac{6214.9378}{0.1524} = 40780.4318N/m$

From spring parameter relations,

$K = \frac{d^4G}{8D^3N}$

$N = \frac{d^4G}{8D^3K}$

$N = \frac{0.02^4 \text{ x } 79.3 \text{ x } 10^9}{8x(0.1524)^3 x\ 40780.4318}$

$N = 10.9874\ \ for\ convenience\ \approx 11turns$

Solid Height of spring

The spring solid height, $L_s = N_t \text{ x } d$

Where N_t is the total number of coils.

$N_t = N + N'$

N' is the number of grounded end = 2

Therefore, $N_t = N + 2$

$= 11 + 2 = 13turns$

Therefore,$L_s = N_t \text{ x } d = 13 \text{ x } 0.02 = 0.26m$

The Spring Rise Angle, θ and $\mathbf{Coil_{pitch}}$ [15]

The rise angle of the spring coil which is the angle between the coil and the base spring is given by

$Coil_{pitch} = \frac{L_F}{N} = \frac{0.508}{11} = 0.0462m$

$\Theta = Tan^{-1}\left(\frac{Coil_{pitch}}{\pi D}\right)$

$\Theta = Tan^{-1}\left(\frac{0.0462}{\pi \text{ x } 0.1524}\right) = 5.54°$

Length of Spring Wire, $\mathbf{L_{WIRE}}$

$L_{WIRE} = \pi D\left(\frac{N}{cos\theta} + 2\right)$

$= \pi \text{ x } 0.1524\left(\frac{N}{cos\ 5.540} + 2\right) \qquad = 6.25m$

Checking For Buckling

Buckling of compression springs is similar to buckling for vertical structural columns. When the free height of the spring (L_{free}) is more than 4~5 times the nominal coil diameter *D*, the spring can buckle under a sufficiently heavy load. Compression spring bucking refers to when the spring deforms under loading action in a non-axial direction. Once buckling sets in the usefulness of the spring in providing projected force is compromised and a dangerous failure condition is imminent. Consequently designing compression springs to operationally display minimum likeliness to buckle is paramount. One quick method for checking for buckling is to compute the deflection to free height ratio (δ/L_{free}) and use the chart below to check if the ratio exceeds the maximum allowable value:

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The maximum allowable spring deflection δ_{MAX} that avoids buckling depends on the free length, the coil diameter, and the spring ends.
Checking for buckling of the spring

$$\frac{L_f}{D} = \frac{20''}{6''} = 3.33$$

$$\frac{\delta_{MAX}}{L_f} = \frac{0.3048}{0.508} = 0.6$$

From the graph of δ_S against L_f, the spring is safe from buckling [13].

Spring Specifications

Spring specifications are as follows
Material, Chrome silicon wire
Ends grounded
Wire diameter, $d = 0.020m$
Outside diameter of spring = $0.1724m$
Mean diameter of spring = $0.1524m$
Total number of coils, $N_t = 13$
Free length of spring $L_f = 0.508m$
Solid length, $L_s = 0.26m$
Spring Index, $C = 7.62$
Spring constant, $K = 40780.4318N/m$
Torsion yield strength, $S_{sy} = 979.335MPa$

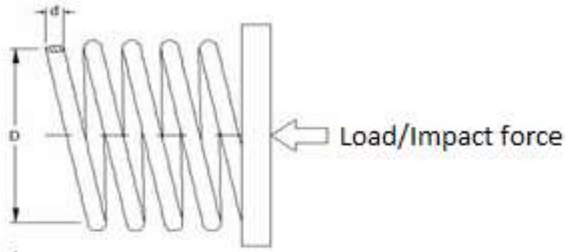

Figure 1 Schematic of the spring

Design of Piston Push Rod

According to secant, the formula for calculating the maximum compressive shear stress δ_c is given as,

$$\delta_c = \frac{P}{A}(1 + \frac{eC_1}{K_g})$$

Where P is the compressive force = 23750N
A is the cross-sectional Area of the cylinder rod = $\frac{\pi d^2}{4}$
E is the eccentricity = 0
K_g is the radius of gyration = $\frac{a}{\sqrt{3}}$
C_1 is the coordinate of line parallel to the x-axis along, which the $normal\ stress\ is\ zero = 1.0$
This implies that,

$$\delta_c = \frac{23750}{\pi \text{ x } (0.04^2} \text{ x } [1 + 0(0)]$$

$\delta_c = 18660917.0775 N/m^2$
$\delta_c = 18.7 MPa$
The material to select should be such that it can withstand the compressive stress of 18.9MPa.

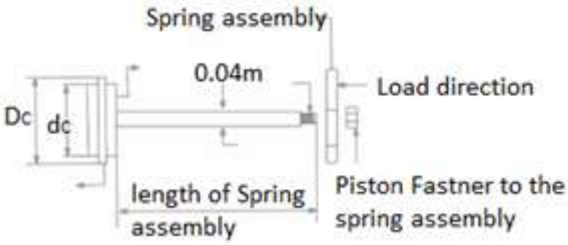

Figure 2. The Piston Assembly

Cylinder with Piston Design
Pneumatic air cylinders are pneumatic linear actuators driven by pressure difference in the cylinder chamber. One side of the piston flange maybe pressurized to provide force and motion with a spring providing return force after pressure is release. On the other hand, both sides can be alternatively pressurized for bi-directional powered motion.

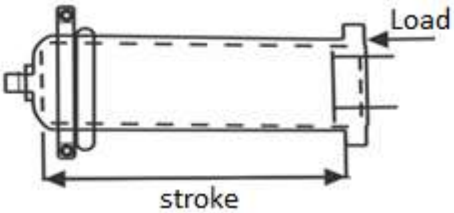

Figure. 3. The Cylinder Assembly

The cylinder can be single or double action. In single action analyzer the cylinder contains pneumatic porting drive for drive in only one direction. Single action cylinders frequently incorporate a return spring to the unpowered position. In double action analyzer, both the sides of the piston can be pressurized for reversible motion. Force ratings can differ somewhat in opposite directions.

Pneumatic systems typically involve a source of compressed air been controlled by valves and causing output devices such as cylinders to operate in a controlled way. The compressed air is typically obtained from a compressor, which is usually driven by an electric motor or an internal combustion engine. Air is routed through pipes to valves, which control the routing of the compressed air. A range of actuators including levers, rollers and solenoids may operate valves. The air is then passed on to cylinders, which convert the energy in the compressed air into linear motion and do useful work. Valves control the switching and routing of the air in a pneumatic system. There are two main types of valves used in pneumatic switching circuits, the 3/2 valves and the 5/2 valves. Important operating specification to consider includes cylinder stroke and operating pressure range. Stroke is the distance between fully extended and fully retracted rod positions while the operating pressure range specifies the full-required range of operating pressure.

Pressure in the Cylinder

Cross sectional area of cylinder, $A_c \ = \frac{\pi d_c^2}{4}$

Where d_c is the internal diameter of cylinder

$d_c = 4.5'' \ = \ 0.1143m$

$A_c \ = \frac{\pi}{4}$ x $(0.1143)^2 = 0.01026m^2$

The maximum pressure the cylinder and the piston assembly, when the piston is three (3) inches or 0.0762m from the base of the cylinder is given as.

$P_{ic} = \frac{F_c}{A_c} \ = 23750/0.01026$

$= 2314814.8148N/m^2 \ = 2.32Mpa$

The pressure P_{2c} when the piston is eleven (11) inches or 0.2794m from the base of the cylinder is given as

$P_{2c} = \frac{V_1}{V_2} \ P_{ic}$

$V_1 = A_c$ x L_1

$=$ 0.01026 x 0.0762

$=$ 7.8181 x $10^{-4}m^3$

$V_2 = A_c$ x L_2

$=$ 0.01026 x 0.2794

$=$ 2.8667 x $10^{-3}m^3$

Therefore $P_{2c} \ = \frac{7.8181 \text{ x } 10^{-4}m^3 \text{ x } (2314814.8148N)}{2.8667 \text{ x } 10^{-3}m^3} \ = 631299.1839N/m^2$

The above pressure values indicates that the air inside the cylinder initially at a pressure of $631299.1839N/m^2$ before impact is compressed to 2314814.8148N/m^2 at impact and secondly, the valve selected should be able to withstand a pressure of 2314814.8148N/m^2

Stress in the Cylinder with Brief Description of the Mechanical Component

Cylinder pressure vessels, gun barrels, hydraulic cylinders and pipes carrying fluids at high pressures develop both circumferential and tangential stresses with values that are dependent upon the radius of element under consideration. The cylinders may fail in one of the two ways depending upon the values of the stresses.

When the wall thickness of a cylinder pressure vessel is about one twentieth or less of its radius, the circumferential stress which result from pressuring the vessel is quite small compared with the tangential stress but if in excess the metal wall will tear along a line parallel to the axis of the cylinder. Under this condition in a closed cylinder, the average longitudinal stress σ_1 is distributed uniformly over the wall thickness is given as,

$\sigma_1 \quad = \frac{Pdi}{4t}$

Where P (2314814.8148N/m^2) is the internal pressure exerted on the wall of the pressure cylinder d_c is the internal diameter of cylinder = 0.1143m

T is the thickness of the cylinder = 0.0076m

Therefore, $\sigma_1 \quad = \frac{2314814.8148 * 0.1143}{4*0.0076}$

$\sigma_1 \ = \ 8703399.12N/m^2$

The rigid body material selected is made of medium plain carbon steel that has minimum yield strength of 500mPa and a medium of elastic of 207GPa
The estimated maximum load to carry, $P = 500N$
Length of the rigid body l = 1.3m.
Assuming that the bar buckling load is greater than the yield load

$$P_{cr} = \frac{\pi^2 EI}{L^2}$$

Using the moment of inertia of a circular bar

$$I = \frac{\pi d^4}{64}$$

$$d = \left(\frac{64 P_{cr} L^2}{\pi^3 * 207 * 10^9}\right)^{1/4} \quad = 0.0108m \ (1..08cm)$$

The specification for the rigid slender body is as follows:

1. Modulus of elasticity = 207Gpa
2. Diameter of rigid body = 0.0108m
3. The estimated max load = 500N
4. Critical load = 800N
5. Minimum yield strength = 500Mpa

Where;
A is the weight of the piston and cylinder plus $a/2$ of the spring weight divided by $9 \ in \ lbf/m$
T is the weight of the bumper plus $4/20$ of the spring weight divided by $4 \ in \ lbf/m$
Kr is the weight of the connection rod plus $7/20$ of the spring weight divided by $7in \ lbf/m$

$$\sum F_v = 0$$

$$R = (9A + 7K_r + 4T) = 0$$

$$R = (9A + 7K_r + 4T) lbf$$

$$\sum M_A = 0$$

$$(9A) * 4.5 + 7K_r * 12.5 + 4T * 18 - M_A = 0$$

$$M_A = (40.5A + 87.5 \, K_r + 72T) \ lbf$$

The bolt at the end of the cantilever and the U-Channel where the bolt is bolted to the chain is made of material that can withstand a force R and a Moment M.

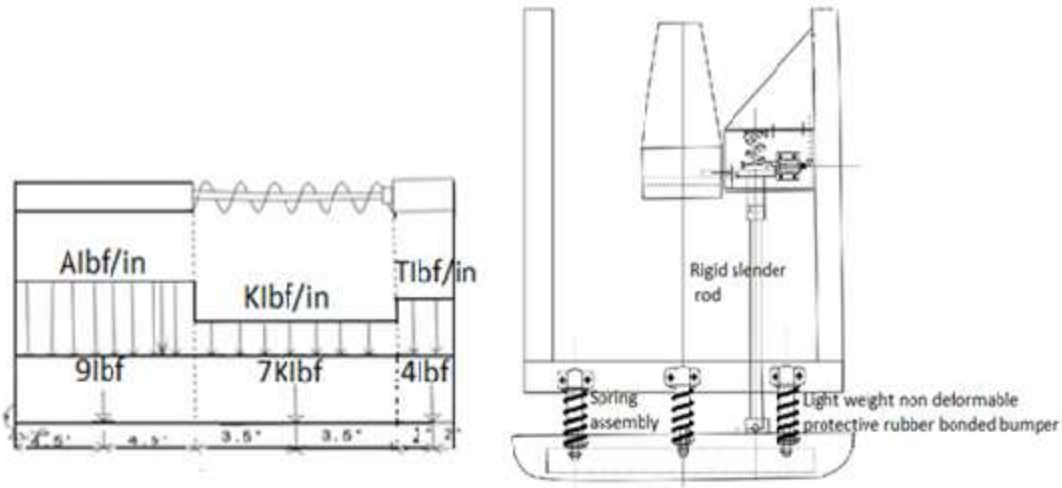

Figure 4. Beam analysis and the schematic of the mechanism

Implication to Research and Practice

This work opens a new line of thought in the world of research. It comes with an enhanced system that will allow minimum damage during accident. There are assumptions made in the course of the work which includes: ability of the driver to re-engage the gear after the impact, the distance behind the moving vehicle is such that the vehicle behind will be able to give way to avoid ripple collisions and the fact that the mechanism will be able to safe guard the radiator/other vital front part of the vehicle. Parts used in the assemble of mechanical systems include: Bearing and Bushing, used to reduce friction and carry loads for rotary or linear motion, Seals, Mechanical Fasteners and Hardware, Springs that extend and retract and return to an unloaded position, high durability hand tools for industrial assembly that are often automated or monitoring capabilities, couplings, collar and mechanical shaft-mounting components. The mechanism components, which are easily sourced, available and the structure makes it a workable device that can be incorporated by automobile manufactures.

Future Research

All the assumptions outlined in the preceding section are still gray areas to research upon in the quest to enhanced safety on the road.

CONCLUSION

The need to further enhance safety on the road and curb the menace of road accidents or at best reduce its high death rate imperatively, initiated the design of safety spring back mechanism, which was carried out in a systematic approach. The component used are reproducible and readily available. The concept prove to record minimal damage during a head on collision with maximum material and personnel safety.

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