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Development of an Automatic Scissors Screw Car Jack

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***Abstract**: In this paper, an existing manual scissors screw car jack was converted to an automatic scissors screw car jack by introducing a sleeve coupling between the manual screw jack and a 12V DC electric motor, which is powered through a direct current energy flow directly from the car battery. The DC electric motor provides a turning effect to automatically power the screw jack for the purpose of lifting the desired load. A screw jack (manual or automatic) is a simple machine used in lifting heavy loads. It becomes very useful in vehicles when changing a punctured tire. It is mandatory for each vehicle plying an approved road to carry a jack for fixing a punctured tire. However, the operation of most car jacks is done manually and also requires prolonged bending or squatting positions when using the jack, making it difficult for most women, the disabled, the elderly, etc., and above all, not ergonomically suitable for the human body due to crouching and squatting positions when in use, which may result in health complications. The automatic scissors car jack developed was successfully tested on five different types of vehicles, with different weights. Five different tests measurements were carried out on each vehicle. The time taken to lift the vehicles 0.03 m off the ground was recorded. They ranged from 74 s to 90 s depending on the weight of the vehicle involved. The heavier the vehicle, the longer the time taken to lift the vehicle off the ground to the predetermined height of 0.03 m.*

***Keywords:** Automatic Scissors Jack, Electric Motor, Manual Jack, Screw Jack, Sleeve Coupling, Power Screw.*

1. INTRODUCTION

Lifting heavy objects to perform maintenance over time is injurious to human health. This has led to the introduction of a simple machine known as jack in vehicles to carry out the purpose of lifting the vehicle when performing emergency activities like changing a tire [1]. A mechanical jack is a device used to lift heavy equipment and vehicles for maintenance work underneath. Car jacks, as they are popularly called, operates based on mechanical advantage to allow human beings to lift their vehicles. Due to the mode of operation of car jacks, they are generally made from metals for strength under load. Basically, automotive car jacks come in two different types: hydraulic and screw types. The hydraulic type operates on the principle of incompressible fluid to lift the vehicle by operating the jack handle, called a lever (a simple machine), and causing the fluid to perform the work through an actuating cylinder.

The screw jack, commonly referred to as a Screw Jack, utilizes the power from its gears and a turning lead screw to lift the vehicle. This is accomplished by applying a small force to the crank handle, which turns the lead screw, causing the two ends of the jack to close together and lift the load. As stated in [2], a small force applied in the horizontal plane can raise or lower a large load. Screw jacks are commonly used in vehicles nowadays.

Motion transmission in an automatic screw jack can be achieved through a gearing system (Figure 1) or direct drive (Figure 2). In the gearing system, a reduction gearing system is included in the design of the automatic screw jack to enable a seamless and easy transfer of motion from the primary mover to the jack. Therefore, the motion between the shafts is transferred by the gearing system. In direct drive, however, the two shafts are connected via a rigid coupling for the transfer of motion.

Azinee [5] investigated the design of scissors car jacks for sedan model of vehicles in Malaysia using Finite Element Analysis (FEA) software on different types of structural model of the scissors car jacks then proposed a new design for the upper and the lower arm of the jack using topology optimization technique. The outcome of their investigation brought about 35% reduction in weight of the new design. Ekhalak [6] investigated the disadvantages in commercially available car jacks which included high operating energy and unsafe for uneven surfaces. Their modified design which included introduction of gas spring and metal plate resulted in easy operation and great stability of the jack. Patel, [7] studied the implementation of an automatic fitted hydraulic jack system in a four-wheel vehicle. This work described automatic powering the fitted jack by the vehicle itself using press button when the need for tire replacement arises. Rana [8] instigated the Integration of Automated jacks for 4-wheel vehicles. The jack they proposed that was utilized on both sides of a 4-wheel vehicle could easily be operated with a button placed at the dashboard of the vehicle.

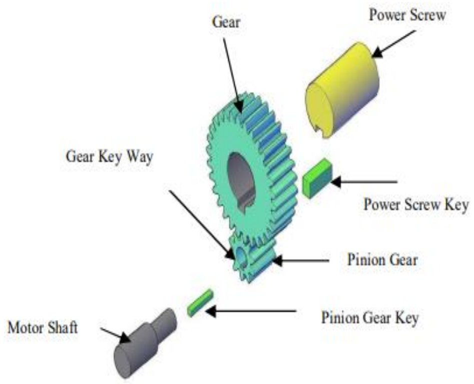

Figure 1: Gearing system [3]

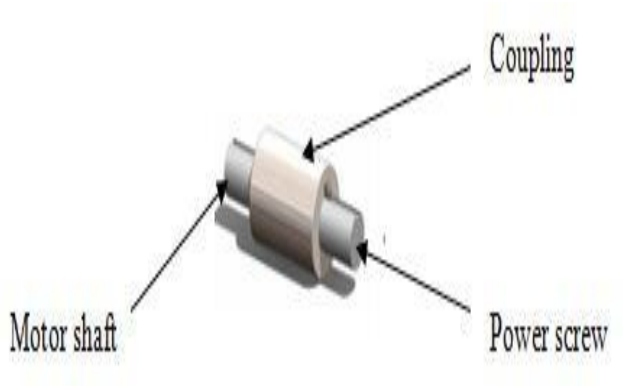

Figure 2: Coupling system [4]

Furthermore, Agu and Igwe [9] investigated remote controlled jack using locally made materials. Their conclusion was that the infrared constructed hydraulic jack powered by DC motor was affordable, simple to maintain and performed well when tested on selected brand of vehicles. Dhamak [10] studied failures in jack and they proposed a unique design standard for scissors jack to lift low and heavy loads under different conditions. The modified jack had a reduction in weight and good strength. Durvesh [11] design a novel motorized hydraulic jack system using wiper motor to control the lever of the hydraulic jack. They also concluded that the combination is simple and effective to lift vehicles. Sukale [12] looked at the total deformation and von-mises stress values of the scissor jack to assess the safety and life prediction of the jack. A detailed structural analysis of the jack design they proposed which was done using ANSYS software helped in predicting the part of the jack which would fail under certain load condition. Based on all the reviews above, potential advantages of screw jack over hydraulic jack were observed and these are, ability to do self-lock unlike hydraulic jack which do not have this feature, ability to slide under any brand of vehicle because it is collapsible unlike the hydraulic jack that is not collapsible and lastly, screw jack require little maintenance. Additionally, all the reviewed had the jack designed with a gearing system and capable to lift 400 kg or 4000 N vehicle. This present work however, is targeted at converting manually operated screw jack to an automatic jack with a direct drive that is capable to lift 700 kg or 7000 N of load. The objective of this present work is to automate the operation of an existing screw jack, to carry out design and analysis using FEA, fabricate the proposed design, test the system.

2. MATERIALS AND METHODS

In this section, the materials used for the development of scissors car jack and the methodology employed will be discussed.

2.1 Materials

In this sub-section, the materials used for the development of scissors car jack will be discussed.

2.1.1 Screw jack

In order to avoid failure and also to increase the lifespan of the car jack, material with high strength was considered for the contact members, lifting members, connecting members, pins and the power screw. In this case, low alloy steel was chosen for its high strength, good machinability, good ductility, wear resistance and economical. However, in most of the standard jacks, the material used are always referred to as “Heavy Duty Steel” [13]. Furthermore, the American Iron and Steel Institute (AISI) also developed a classification system for different types of iron and steel alloys from where Nickel-Chromium, Molybdenum steel alloy may be the probable used material for the construction of the scissor jack. This particular alloy has a classification of AISI 4340 in engineering steel industry [13]. Therefore, High Strength Low-Alloy Steel (40Ni2Cr1Mo28/AISI 4340) [13] was used in the design.

2.1.2 Material for the coupling

Additionally, the following were considered before concluding on the type of coupling to use. The input and output torque, the size of the coupling, the durability, safety and the economy of manufacture of the coupling. Sleeve coupling met the requirement for its simplicity, rigidity and ease of manufacturing and the chosen material for its construction was cast iron because of its good wearing properties excellent machinability and ease of producing complicated shapes. More so, bolts were used to connect the motor shaft and the power screw respectively. This was done to stop the shafts from moving relative to the linked element that transfers the torque. Casing was introduced in order to prevent excessive dust and dirt exposure.

2.1.3 Procured Components based on the design calculations

Table 1 below shows all the component parts used for the construction of the automatic jack.

Table 1: The component parts acquired the construction of the automatic jack

S/N	Item	Name of parts	Specifications
1		Mal Jack	KJ2 Series Working Load: 700Kg
2		Windshield Motor	Voltage: 12 V Torque: 12 Nm Rpm: 50 rpm
3		Power Window Switch	Voltage: 12 V Pins: 4
4		Sleeve	Length:50mm 2 x M3 Allen screw

2.2 Methods

In this sub-section, the methodology employed for the development of scissors car jack will be discussed.

2.2.1 Design procedure

After a brief study of the scissors jack, it was observed that the jack can be made to work automatically with the aid of a 12V DC motor connection via gearing system or coupling thereby, eliminating manual operation causing the inconveniency. The first step considered under the design procedure was the load the jack will be subjected. Then necessary calculations and material selection were done for this load in order to choose the appropriate electric motor, the coupling for the transmission of the torque from the motor to the jack. Procurement of necessary components, assembly and balancing on roads were done while giving consideration to torsion, bending and shear forces acting on strategic point of the members. It is important to note that iterative comparison of stress values obtained, with those required for safe operation (yield stress) was made and the cycle repeated when necessary.

2.2.2 Design consideration

Automobiles and automobile workshops are provided with jacks to aid in lifting the vehicles in case of maintenance or repair beneath the vehicle. It is however, very important to design the jack for safe operation. Presently, we are in the era of automation, where machines like car jack require little or no human involvement during operation. An automatic car jack operates by means of electrical supply (preferably DC) to control all the actions required of the jack. For instance, to lift a load (e.g. a vehicle), a scissors jack is first placed underneath the vehicle for it to have a contact with the load platform, then a DC supply is activated to power the prime mover which in turn transmit its rotating speed to the power screw of the jack whose rotation within a threaded bore of its connecting members causes closure of these members resulting in raising the load from the floor. Technically, as contact is made with the jack platform, object load is increasingly transferred to load platform which causes forces to be developed in and transferred through the lifting members and connecting members. The transferred force through the connecting members goes to the threaded bore and finally to the thread. The forces acting on a scissors jack are shown in Figure 3.

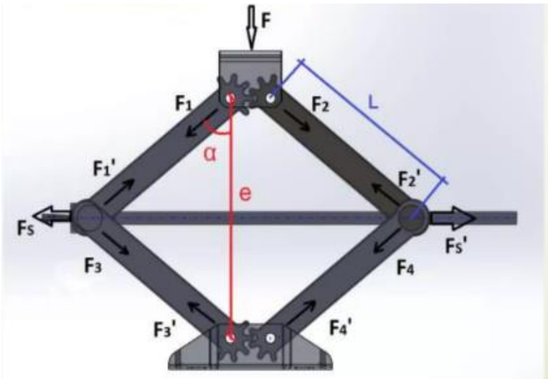

Figure 3: Force distribution on a scissors jack

Due to the load the scissors jack will be subjected to, more consideration was given to the material for the components, condition under which the jack will be subjected, power consumption of the jack from the battery, safety of the user and economy of production.

2.2.3 Scissors jack design

For the design of the proposed scissors jack, the following assumptions were made; The clearance between the ground and the car chassis is assumed to be 165 mm after observing various car's specifications. The scissor jack is carrying maximum load when the car is lifted. The jack is assumed to move in vertical direction only, by 50 mm. The scissors jack only support a quarter of the car mass which is between 300kg (3000N) and 1000kg (10000N) [15]. The power screw thread is acme thread. Moreover, some conditions were also taken for initiating design and these include the use of input parameters which were taken from studying cars specifications and different loading conditions this include inputs from practical analysis of car lifting conditions and by failed condition. Further consideration shows that the maximum load on the acme threaded screw occurs when the jack is in the bottom position [2] and the design was built around this position. This is shown in the figure 4

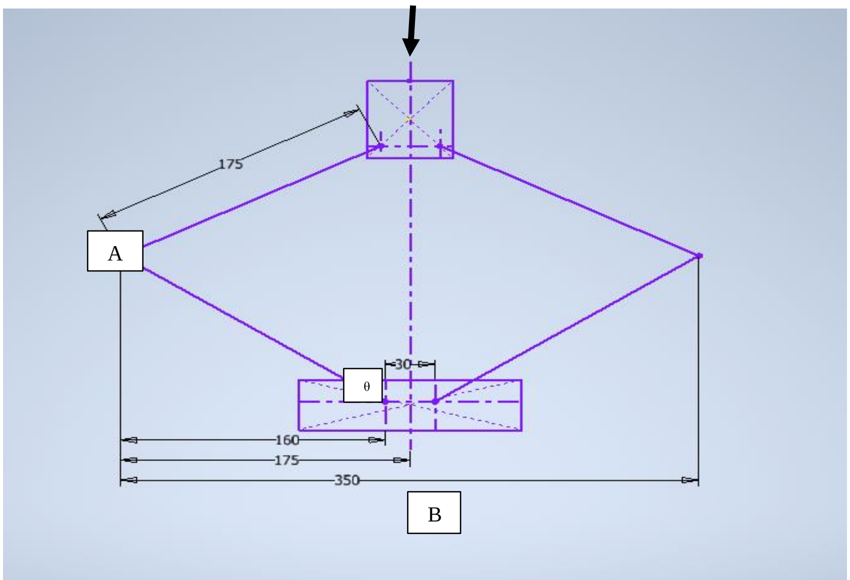

Figure 4: Loaded Scissors Jack

The position of the link AB at an angle θ to the horizontal in the bottom position is represented in the line diagrams below

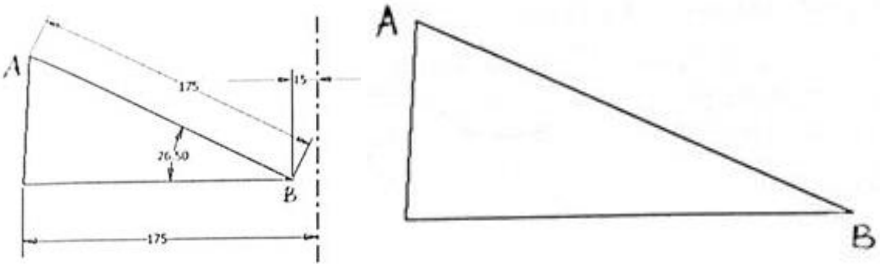

Figure 5: Geometry of the triangle formed.

2.2.4 Design calculations for the threaded screw (Acme) of the jack

The design calculation for the threaded screw of the jack is given in this section.

The angle of inclination (θ) of the link AB with the horizontal using the geometry in Figure 4 above was found by using Equation (1),

$$\text{Cos } \theta = \frac{Adjacent}{Hypothenus} \tag{1}$$

$$\text{Cos } \theta = \frac{160}{175}$$

$$= 0.9143;$$

$$:. \ \theta = \cos^{-1} 0.9143$$

$$= 26.5^{0}$$

Since each member of the part of the scissors jack carries half of the total load on the jack, therefore, member AB is subjected to tension while the threaded screw is under pull. The magnitude of this pulls on the threaded screw was found using Equation (2)

$$\text{F} = \frac{W}{2 \tan \theta} \tag{2}$$

Where F = Tension (N),
W = 700kg
= 7000N and
$\theta = 26.5^{0}$

$$\text{F} = \frac{7000}{2 \tan 26.5}$$

$$= 7916.76 \text{ N}$$

Also, since similar pull acts on the other side (second nut), hence, total tensile pull on the threaded rod was found with Equation (3) below

$$W_1 = 2F \tag{3}$$

$$W_1 = 2 \text{ X } 7916.76$$
$$= 15833.52 \text{ N}$$

Since high strength low steel was chosen as material for the scissors jack, its properties are as shown in the Table 2.
Shear stress (τ_s) was calculated using equation (4) when yield tensile strength (δ_{yt}) = 834MPa as in the Table 2

$$\tau_s = \frac{(\delta yt)}{2} \tag{4}$$

$$\tau_s = \frac{834}{2}$$
$$= 417 \text{ N/mm}^2$$

Table 2: Material Properties of High Strength Low-Alloy Steel [13]

Quantity/Description	SI Units	Imperial Units
Ultimate Tensile Strength	931 MPa	13 500 psi
Yield Tensile Strength	834 MPa	121 000 psi
Elongation at Break	20.2 %	20.2 %
Modulus of Elasticity	205 GPa	29 700 ksi
Poisson's Ratio	0.29	0.29
Shear Modulus	80 GPa	11 600 psi
Bulk Modulus	140 GPa	11 600 ksi

Hence, shear stress is 417 N/mm^2

The allowable stress ($\boldsymbol{\delta}$allowable) was calculated using equation 5 ,

Assuming factor of safety (N) = 3 and service factor (k) = 1.6 [2]

$$\delta\text{allowable} = \frac{(\delta\text{yt})}{K \text{ x } N} \tag{5}$$

Therefore,

δallowable = 173.75 N/mm^2.

Allowable stress is 173.75 N/mm2

The allowable shear stress ($\boldsymbol{\tau}_{\text{allowable}}$) was also calculated using Equation (6)

$$\tau\text{allowable} = \frac{(\delta\text{allowable})}{2} \tag{6}$$

τallowable = $\frac{173.75}{2}$ = 86.875 N/mm2.

Allowable shear stress is 86.875 N/mm2

The core diameter (d_c) of the screw thread was calculated as shown in Equation (7) below

$$\text{Since } \delta\text{allowable} = \frac{\text{W1}}{A} \tag{7}$$

Substituting W_1 (total pull on the threaded rod) and

A (the area of the rod) = $\frac{\pi d_c^2}{4}$ in Equation (7)

:. d_c = 10.8mm is the calculated core diameter of the screw rod

However, since the screw will undergo torsional shear stress, therefore it is best to consider a higher core diameter [2]. Hence, selection from standard screw was considered for the rod with higher core diameter and other parameters as follows.

Core diameter (dc) = 12mm
Outer diameter (do) = 16mm
Mean diameter (dm) = 14mm
Pitch (p) = 3mm

Length of the screw, L = length of jack at minimum position (Figure 4) + (2 X do)

= 350 + (2 X 16)

= 382 mm

To check for principal stresses on the screw rod

Since $\tan\alpha = \frac{p}{\pi dm}$

Where α = helix angle,
p = the pitch and dm is mean diameter

$$\text{Tan}\,\alpha = \frac{3}{\pi \text{ x } 14}$$

$$= 0.0682$$

$$\alpha = 4.336^{o}$$

The effort (E) required to rotate the screw rod was calculated using equation (8) below

$$E = W_1 \tan(\alpha + \emptyset) \tag{8}$$

$$= W_1 \left(\frac{\tan\alpha + \tan\emptyset}{1 - \tan\alpha\tan\emptyset}\right)$$

Where $W_1$and $\tan\alpha$ have been calculated

$\emptyset$ = 0.15 (coefficient of friction for Acme thread between the screw and nut = 0.15) [2]

∴ E = 15833.52 x tan (4.336 + 0.15) = 1242.23 N.

The torque (T) required to rotate the screw under the design load was calculated using equation (9) below

$$T = E \text{ x } \frac{dm}{2} \tag{9}$$

Where E = 1242.23 N and d_m

= 14 mm

Therefore, $T = 1242.23 \text{ x } \frac{14}{2}$

= 8695.61 Nmm

= 8.69561 Nm

The shear stress (τ) in the screw due to the torque was calculated using the equation (10) below

$$\tau = \frac{16\,T}{\pi\,dc^3} \tag{10}$$

$$\therefore\ \tau = \frac{16 \text{ x } 8695.61}{\pi \text{ x } 12^3}$$

$$= 2.136 \text{ N/mm}^2$$

The direct tensile stress (δ_t) on the screw was calculated using equation (11) below

$$\delta_t = \frac{W1}{\frac{\pi\,dc^2}{4}} \tag{11}$$

$$\delta_t = \frac{15833.52}{0.7854 \text{ x } 12^2}$$

$$= 139.9998 \text{ N/mm}^2$$

$= 140\ N/mm^2$

The maximum principal tensile stress ($\delta_{t(max)}$) on the screw was calculated using equation (12)

$$\delta_{t(max)} = \frac{\delta t}{2} + \frac{1}{2}\sqrt{[(\delta_t)^2 + 4(\tau)^2]} \quad (12)$$

$$\delta_{t(max)} = \frac{140}{2} + \frac{1}{2}\sqrt{(140)^2 + 4(2.136)^2}$$

$$= 140.03 N/mm^2$$

The maximum shear stress (τ_{max}) was calculated with equation (13)

$$\tau_{max} = \frac{1}{2}\sqrt{[(\delta_t)^2 + 4(\tau)^2]} \quad (13)$$

$$\tau_{max} = \frac{1}{2}\sqrt{[(140)^2 + 4(2.136)^2]}$$

$$= 70.03\ N/mm^2$$

In order to determine if the design was safe, solutions from the calculated values for both tensile and shear stresses (that is allowable and maximum stresses) for the screw were compared. From Equations (5) and (12), the allowable stress (δallowable) was 173.75 N/mm2 and the maximum principal tensile stress (δt(max)) on the screw body was found to be 140.03 N/mm^2. Therefore, since δt(max) < δallowable the design for the acme threaded screw body is safe. More so, comparing the shear stresses (allowable and maximum) from both Equations (6) and (13), the allowable shear stress (τallowable) was 86.875 N/mm^2 and maximum shear stress was (τ_{max}) 70.03 N/mm^2 for the screw body. Since the value of $\tau_{max} < \tau_{allowable}$ the design is safe.

2.2.5 Design calculation for the nuts of the scissors jack.

Firstly, the thickness of the nut (t) was calculated following these steps. Let n = number of threads in contact with the threaded rod and also assuming the load W1 (15833.52N) is distributed uniformly over the cross-sectional area of the nut. Therefore, the bearing pressure (P_b) between the threads can be found using Equation (14);

$$P_b = \frac{W1}{\frac{\pi}{4}(do^2 - dc^2)\ x\ n} \quad (14)$$

Where W_1, d_o and d_c remain as calculated earlier,

n = no of threads in contact with the nut and

P_b = 17.5 for jack nut at low speed < 2.4m/min [3].

$$n = \frac{15833.52}{\frac{\pi}{4}(16^2 - 12^2)\ x\ 17.5}$$

$$= 10.29 \text{ (approximately 10)}$$

To find the thickness of the nut (t), Equation (15) was used

$$t = n\ x\ p$$

$$= 10\ x\ 3$$

$$=30\ mm \quad (15)$$

Where n = no of threads in contact with the nut;

p = the pitch

From the above, ten (10) threads were provided for in the nut for good stability and strength and also to prevent rocking of the screw inside the nut.

Secondly, the width (b) of the nut which was taken as 1.5do [2] was also determined following these steps.

$$b = 1.5do \quad (16a)$$

where do = 16 mm (as stated earlier)

:. b = 1.5 x 16

= 24 mm

However, in order to control the movement of the nut beyond the 350mm (maximum distance between the center lines of the nuts when the jack is at the bottom position) rings of 8mm thickness were considered and fitted on the screw with the aid of a set screw (16a) [2].

Thirdly, the length of the screwed portion (L) of the rod was calculated as follows

L =350 + t + 2 x ring thickness (16b)

Where 350 mm is distance between centrelines between the nuts when jack is at bottom position and
t = nut thickness.

:. L = 350 + 18 + 2 x 8

= 384 mm

Lastly, since the scissor jack will be operated via a prime mover, therefore provision for the end's attachment was considered. The ends were extended by addition 15 mm and

the total length of the rod = 384 + 2 x 15

= 414 mm

2.2.6 Design calculation of the pins in the scissors jack.

To design for the pins these steps were taken. Firstly, let d_1 = diameter of pin in the nut and since the pins are in double shear, therefore,

load F = 7916.76 N (from $W_1 = 2F$, equation 3).

:. $7916.76 = 2 \times \frac{\pi}{4} \times (d_1)^2 \times \tau_s$

Where $\tau_s = 417$ N/mm^2 for the chosen material in Equation 4

$(d_1)^2 = \frac{7916.76 \times 4}{\pi \times 2 \times 417}$

$= 12.09$ mm^2

$d_1 = \sqrt{12.09}$

= 3.48 mm the considered diameter was 8mm

The diameter of pin head was taken as 1.5 d_1 = 12 mm and thickness 4 mm. The pins in the nuts were kept in position by separate rings of 4 mm thick and 1.5 mm split pins passing through the rings and pins.

2.2.7 Design calculation for the connecting members (links) of the scissors jack.

Due to the load, the links may buckle in two planes at right angles to each other [2]. The links are said to have hinges at both ends when there is buckling in the vertical plane (which coincides with the links' plane). On the other hand, links are considered fixed at both ends when buckling takes place in a plane perpendicular to the vertical one.

The load on the link (f_1) $F_1 = \frac{F}{2}$ (17a)

$F_1 = \frac{7916.76}{2}$

$= 3958.38$ N

The links were designed for buckling load (W_{cr}) with the equation below and assuming a factor of safety of 5. Therefore,

$W_{cr} = F_1 \times 5 = 3958.38 \times 5$

$= 19\,791.9$ N (17b)

More so, let t_1 = thickness of link and b_1= width of the link, it was assumed that the width is

$b_1 = 3 \times t_1$ (18)

Therefore, the cross-sectional area of the link (A)

$A = b_1 \times t_1 = 3t_1 \times t_1$ (19)

$A = 3t_1^{\,2}$

The moment of inertial of the cross-sectional area of the link (I)

$I = \frac{1}{12} \times t_1\,(b_1)^3$ (20)

Therefore $I = 2.25\,t_1^{\,4}$ (since $b_1 = 3t_1$ Equation 18)

Radius of gyration (k) = $\sqrt{\frac{I}{A}}$ (21)

$k = \sqrt{\frac{2.25\,t_1^4}{3t_1^2}}$

$= 0.866t_1$

Since for buckling of the link in the vertical plane, the ends were considered as hinged, therefore equivalent length of the link (L) = 175mm (from measurement) and using Rankine's constant,

$a = \frac{1}{7500}$ and Rankine's formula [2],

Buckling load, $W_{cr} = \frac{(\delta yt) \text{ x A}}{1+a\left(\frac{L}{K}\right)2}$ (22)

Where Wcr = critical load,
σy = Yield stress,
L = Length of link,
k = radius of gyration

:. $19791.9 = \frac{834 \text{ x } 3t_1^2}{1 + \frac{1}{7500} \text{x} \left(\frac{175}{0.866t1}\right)2}$

Solving,

$\frac{19791.9}{2502} = \frac{t_1^4}{t_1^2 + 3.874}$

Rearranging, solving for t_1 using quadratic equation, neglective part of the solution,

$t_1 = 3.280$ mm the considered thickness was 3 mm
and

$b_1 = 3t_1 = 3 \times 3$

= 9 mm was chosen as the width of the link.

Now let us consider the buckling of the link in a plane perpendicular to the vertical plane. The Moment of inertia of the cross-section of the link, cross sectional area and radius of gyration were calculated as shown below: -

let t_1 = thickness of link and b_1= width of the link, it was assumed that the width is

$b_1 = 3 \text{ x } t_1$ (same as Equation18)

Therefore, the cross-sectional area of the link (A)

$A = b_1 \text{ x } t_1 = 3t_1 \text{ x } t_1$ (same as Equation 19)

$A = 3t_1^2$

The moment of inertial of the cross-sectional area of the link (I)

$I = \frac{1}{12} \text{ x } t_1 (b_1)^3$ (same as Equation 20)

:. $I = 2.25t_1^4$ (since $b_1 = 3t_1$ Equation 18)

Radius of gyration (k) = $\sqrt{\frac{I}{A}}$ (same as Equation 21)

$k = \sqrt{\frac{2.25 \text{ X } t_1^4}{3 \text{ X } t_1^2}}$

$= 0.866t_1$

Since for buckling of the link in a plane perpendicular to the vertical plane, the ends are considered as fixed, therefore equivalent length of the link (L_e)

$L_e = L/2$

$= 175/2$

$= 87.5$ mm

Again, according to Rankine's formula, buckling load is calculated from Equation (23) and it is given as Buckling load (W_{cr})

$$W_{cr} = \frac{(\delta yt) \text{ x A}}{1 + a\left(\frac{L}{K}\right)2} \quad (23)$$

Using the values $t_1^2 = 3^2$ mm = 9 mm and $L_e = L/2 = 87.5$ mm

$$W_{cr} = \frac{834 \text{ x } 27}{1 + \frac{1}{7500}\left(\frac{87.5}{0.866 \text{ x } 9}\right)2}$$

$= 22.142$ kN

Since this buckling load is more than the calculated value (that is 19. 79 kN) from equation 17b, therefore the link is safe for buckling in a plane perpendicular to the vertical plane. Therefore, we may take thickness (t_1) as,

t_1= 3 mm; and width (b_1) = 9 mm for the links.

2.2.8 Design calculation and choice for the prime mover (motor) for the scissors jack.

Since the scissors jack was considered to be automatic, it requires an actuator to supply the needed torque. From equation 9 the required torque (T) for this jack was calculated as 8695.61 Nmm or 8.69561 Nm. A low-speed high power DC motor that can supply a rotation of 50 rpm to the power screw was considered. The power (P) rating was calculated using equation (24)

$$\text{Using } P = T\omega \quad \left(\omega = \frac{2\pi N}{60}\right) \tag{24}$$

Where T is torque = 8.69561 Nm and N = 50 rpm

$$P = 8.69561 \text{ x } \frac{2\pi \text{ x } 50}{60}$$

$$= 45.53 \text{ W}$$

Therefore, power of the DC motor considered for use is 46 W at 50 rpm of the spindle.

2.2.9 Design consideration and calculation for the coupling

Since the coupling is accommodating two rotating shafts (that is the motor shaft and the extended end of the power screw) therefore, the following were considered

i. The input electrical power from the battery
ii. The output mechanical power from the motor
iii. The load the coupling will be subjected.

After careful research, a hollow sleeve coupling made from cast iron was chosen due to its rigidity and easy to manufacture. The parameters for the sleeve coupling include, inside diameter which is equal to the diameter of the shafts (d), the outer diameter of sleeve and the length of the sleeve. These three parameters were chosen on the basis to realize a reasonable transmission of torque from the rotation of the motor shaft to the power screw.

2.2.10 Design calculation for the sleeve

The outer diameter of the sleeve (D) = 2d + 13 mm [2] (25)

D = 2 (13) + 13

= 29 mm say 30 mm (where shaft diameter is d = 13mm)

The length of the sleeve (L) = 3.5d

L = 3.5 (13)

= 45.5 mm say 50 mm

To check for the induced shear stress (τ_c) in the sleeve which is made from cast iron and considered to be hollow shaft as against the permissible shear stress for the material of the sleeve with safe value of shear stress taken as 14 MPa = 14 N/mm^2 [2]. Using torque transmitted (T) by hollow section equation [2],

$$T = \frac{\pi}{16} \text{ x } \tau c \text{ x } \left(\frac{D^4 - d^4}{D}\right) \tag{26}$$

Where τ_c= the induced shear stress

T = the torque transmitted = 8695.61 Nmm = 8.69561Nm
D = the outer diameter of the sleeve = 50mm
d = the inside diameter of the shaft =13mm

$$8695.61 = \frac{\pi}{16} \text{ x } \tau c \text{ x } \left(\frac{50^4 - 13^4}{50}\right)$$

$$\tau_c = 8695.61 \text{ x } \frac{16}{\pi \text{ x } 124428.78}$$

$$= 0.35592 \text{ N/mm}^2.$$

Since the induced shear stress (τ_c) = 0.35592 N/mm^2 in the sleeve (cast iron) is less than the permissible shear stress of 14 N/mm^2, therefore the design of sleeve is safe.

2.3 Automatic Jack Construction

The ring end of the manual jack was cut off with a saw blade leaving a small extended end of the power screw. U-shape metal bracket was fabricated for the jack to offer protection and support for the motor. One end of this bracket was welded to the jack pin while the other end had the motor attached by allen screws thereby creating a space for the coupling connection of the small extension and the motor shaft. The jack small extension was connected to the motor shaft via the sleeve and two M6 bolts. Figure 6 below is the pictorial view of the assembled automatic scissors jack.

Figure 6: Automatic jack assembled together

3. TEST, RESULTS AND DISCUSSIONS

3.1 Test Procedure of the Constructed Automatic Jack

i. The Jack was placed on a flat hard surface (concrete floor, hard metal sheet or plank) to avoid the jack sinking while lifting up the vehicle
ii. Connect the red and black wires to a +12 v and – 12 v terminals of a DC supply. (This could be the car battery or other storage devices and ensure it is tightened properly).
iii. The up-control button was pressed continuously with no vehicle on the jack (no load condition). The voltage and the current flow to the motor were measured and recorded with a multi-meter during the no load process.
iv. The down-control button was pressed continuously with no vehicle on the jack. The voltage and the current flow to the motor were measured and recorded with a multi-meter during the no load process.
v. The jack was slid beneath the vehicle at the exact spot meant for fixing the jack for the purpose of lifting the car. The up-control button was pressed continuously with the vehicle on the jack to start lifting (the voltage and the current flow to the motor was measured and recorded with a multi-meter during the lifting process) until a desired height is attained (causing a clearance between the tire and floor). The clearance between the tire and the floor was measured and recorded with a measuring tape.
vi. The down control button was pressed continuously with the vehicle on the jack to lower the vehicle (the voltage and the current flow to the motor were measured and recorded with a multi-meter in the process) until the tire touches floor.
vii. Slide the jack out and disconnect the wires from the DC source.

The pictures in Figure 7 shows the procedure for lifting a care using the automatic jack.

Figure 7: Testing of the automatic jack

3.2 Results

The result of the test presented here in Table 3 and Table 4 are the responses recorded using the Multi-meter and the measuring tape readings with respect to the jack's movement.

Table 3: Test results of the automatic jack on vehicle battery

S/N	Control button	Loading condition	Jack response	Voltage readings	Current readings
1	Press up (continuously)	No load	Upward	13.85V	4.05A
2	Press down (continuously)	No load	Downward	13.60V	3.96A
3	Press up (continuously)	Loaded	Upward	12.75V	5.51A
4	Press down (continuously)	Loaded	Downward	12.74V	4.80A

Table 4: Test results of the automatic jack on different vehicles

S/N	Vehicle Type	Vehicle weight (kg)	Tire clearance from the ground (mm)	Average time taken to attain the tire clearance (minutes)
1	Toyota Camry	1995	30	1.30
2	Toyota Corolla	1670	30	1.20
3	Toyota Rav4	2130	30	1.35
4	Suzuki Every	780	30	1.06
5	Suzuki Ignis	990	30	1.14

3.3 Discussion

Table 3 above presents the results from the constructed automatic jack. From the result, it could be seen that the jack responded by moving up and down when the control button was pressed indicating good operation conditions. Additionally, with no load when moving up the jack has voltage and current flow readings of 14.07 Volts and 4.05 amps respectively, and moving down as 14.05 Volts and 3.96 Amps respectively. When the jack was loaded and moving up, this jack has voltage and current flow readings as 12.64 Volts and 5.51 Amps respectively. Also, when, moving down and loaded, the readings are 12.64 Volts and 4.80 Amps respectively. Furthermore, Table 4 shows the results of the automatic jack when used to lift vehicles of different weights. Toyota Camry weighing 1995 kg was lifted from the ground with a tire clearance of 30 mm with average time of 90s (1min 30s), Toyota Corolla weighing 1670 kg was lifted with a tire clearance of 30 mm with average time of 80s (1min 20s), Toyota Rav4 weighing 2130 kg was lifted with a tire clearance of 30 mm from the ground in average time of 95 s (1 min 35 s), Suzuki Every weighing 780 kg was lifted from the ground with a tire clearance of 30 mm in average time of 66 secs (1 min 6 secs) and Suzuki Ignis weighing 990 kg was lifted off the ground with a tire clearance of 30 mm in average time of 74 secs (1 mins 14 secs).

4. CONCLUSION

An automatic jack was developed. The existing manual jack was converted into automatic jack with the introduction of a 12V DC electric motor to the power screw via a sleeve coupling in order to transfer the motion of the motor to the power screw. Various types of vehicles with different weight were tested with the automatic jack that was constructed. The advantage of this modification over the manual type is to make lifting of the vehicle easier and faster without human energy operating assistance thereby reducing problem related to ergonomics. It is also easier for the elites to operate it.

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