PRINT ISSN 1119-8362
Electronic ISSN 2659-1499

Full-Text Available Online At
https://www.ajol.info/index.php/jasem
https://www.bioline.org.br/ja

J. Appl. Sci. Environ. Manage.
Vol. 28 (11B Supplementary) 3799-3807 Nov. 2024

Design and Fabrication of an Automated Scissors Car Jack to Lift and Lower a Car with Minimal Human Effort

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ABSTRACT: Tyre punctures is almost inevitable in vehicles and changing a flat tyre is not a pleasant experience when it is done manually. With the increasing levels of technology, the effort required in achieving the desired output can be effectively and economically decreased by the implementation of better designs. Hence, the objective of this paper is to design and fabricate an automated scissors car jack to lift and lower a car with minimal human effort using appropriate standard methods including decision matrix. In this research an automated scissors jack was designed and fabricated to lift and lower a car. The power screw of the scissors jack has a torque of 28562N-mm, the top arms have a moment of inertia of 48334 mm4 and 77863.5 mm4, the bottom arms have a moment of inertia of 74250 mm4 and 89842.5 mm4, with bending stress of the top plate given as 5.75 N/mm2. A 12V DC motor was selected to drive the power screw. The fabricated automated scissors jack with a power screw torque of 28562N-mm was tested for performance and the results obtained indicate that it was able to raise and lower a load of 452.92kg in an average lift time of 160 seconds and this performance was satisfactory.

DOI: https://dx.doi.org/10.4314/jasem.v28i11.38

Cite this Article as: OGHOGHORIE, O; OSARETIN, J. O. (2024). Design and Fabrication of an Automated Scissors Car Jack to Lift and Lower a Car with Minimal Human Effort. *J. Appl. Sci. Environ. Manage.* 28 (11B Supplementary) 3799-3807

Dates: Received: 21 September 2024; Revised: 27 October 2024; Accepted: 04 November 2024 Published: 30 November 2024

Keywords: Scissors Jack; DC Motor; Power Screw; Battery; Lifting Arm

Lifting heavy machinery in bent-over positions is not ideal for the human body due to the pain it can induce on the back over a long time; trying to lift a load in a squat position exerts pressure on the spine and can lead to injury. For instance, when the load in question is your car, and you are in a situation where you need to change your tire, the human strength is not enough to lift the car to complete such task. To remedy this cause, a device known as a jack is used. A Jack is a mechanical device used to lift heavy loads or apply great forces. Available jacks present difficulties for the elderly people and women and are especially disadvantageous under adverse weather conditions (Chitins, *et al.*, 2019). During road-side emergency like tire puncher, a jack is required to lift the vehicle. A mechanical jack can lift all or part of a vehicle into the air for repairing breakdowns or vehicle maintenance (Choudhary, *et al.*, 2016). It has been established that the process of lifting a car using an ordinary jack has become undesirable for motorists over the past few years. Motorists have been injured or stuck on the road for long periods of time. The available jacks pose a problem of drudgery and risk of getting injured during jacking process (Madanhire, *et al.*, 2019). A power screw mechanism included in a scissor jack is design to lower the amount of force

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needed to lift the heavy loads. The operation of a scissor jack starts by using a z shaped crank that is mounted to a small hole on the mechanism (Lydia, *et al.,* 2018). The power screw Jack eliminates human effort to operate the jack, through a simple electrical device which can be actuated by a 12 V battery (Babu, *et al.,* 2015). Most of the available scissor jacks are manually operated and this causes difficulties elderly people and most woman to use the manual jacks since it requires power manually. The purpose of this study is to overcome this problem. An automated car jack which has a frame type of design by using electricity from the car will be developed. Operator only needs to press the button from the controller without working in a bent or squatting position for a long period of time to change the tyre. A motorized jack reduces drudgery (hard boring work) during maintenance operations since manual power is not required or is minimized (Asoye, *et al.,* 2015). Motorizing the jack brings about increased timeliness and efficiency in maintenance operations on vehicles as the motor will be lifting the load faster than the manual operation. A motorized jack also reduces the risk of getting injury if malfunctioning of the device occurs during operation of the jack. Single personnel is enough to operate the motorized jack efficiently to lift the load, it does not require a skilled worker (Shaikh, *et al.,* 2015). Therefore, there is a need to design a car jack that is safe, reliable and should be able to raise and lower a height level without the human effort. Hence, the objective of this paper is to design and fabricate automated scissors car jack to lift and lower a car with minimal human effort

MATERIALS AND METHOD

Some of the materials used in this study are: Mild steel, Threaded rod, Hinge, Bolts and nuts, Universal joint coupling, 12V DC gear motor, Flat sheet metal. The list of equipment used, usage and items used upon are shown in Table 1.

Table 1: List of equipment used, usage and items used

S/N	Tools/Equipment	Usage	Items used upon
1.	Welding machine	Welding	Sheet, rods, bars
2.	Drilling machine	Drilling	Drill holes in sheet metals
3.	Cutting disc	Cutting	Metals
4.	Grinding disc	Grinding	Grinding down metal
5.	Mild steel electrode	Welding	Mild steel
6.	Spray Gun	Painting	External and Internal parts of the machine

Methods: In an attempt to automate the scissors jack car lifting mechanism, a step had to be taken to appropriately size the motor that would replace the human effort necessary to turn the power Screw.

Preliminary Design: A car was selected; this step is necessary to determine the weight that needs to be lifted. For example, a design for a motor requirement for a commercial vehicle if used in a truck or trailer vehicle could prove catastrophic and damage the motor. Figure 1 shows a Toyota Camry 2019 model vehicle for which the parameters of weight and its general dimensions were sourced.

Fig. 1: Toyota Camry model with base specifications**,** 2019**.** *Automobiledimensions.com*

The choice for this vehicle stems from the fact that it is a common commercial vehicle owned by Nigerians and should provide for an estimate of the Curb weight vehicles owners of similar sizes may be dealing with (informationngr.com). This vehicle has a mass of 1500kg. For this weight, the car engine and other parts are carried by the front axle and thus, one can say that about 60% of the car weight is distributed to the front axle while 40% by the rear axle (Babu *et al.,* 2015).

Conceptual Designs: Three concepts for the design of the machine were considered, and based on specific criteria, the best or most viable concept was selected. The selection criteria put into consideration was based on the results from the decision matrix table.

Concept 1 of a Scissors Jack: This concept was considered as the manually operated scissors jack. It is scissor jack is operated simply by turning a small crank that is inserted into one end of the scissor jack. This crank is usually "Z" shaped. The end fits into a ring hole mounted on the end of the screw, which is the object of force on the scissor jack. When this crank is turned, the screw turns, and this raises the jack. Figure 2 shows concept 1 of the scissors jack.

Fig. 2: Conceptual design 1 of a scissors jack

Advantage of concept 1: It can be manually operated

Disadvantages of concept 1 [i] It is time consuming [ii] It induces more work on the operator of the machine [iii] It works quite slowly in getting you the maximum height you desire to raise the car to [iv] Its basic use comprises comparatively light-duty vehicles which makes it somewhat inappropriate for heavy duty vehicles

Concept 2 of a scissors jack: This concept is similar to concept 1 only that the scissors jack will not be driven manually but will be driven with the use of a direct drive motor system, eliminating the use of gears. Figure 3 shows concept 2 of the automated scissors jack.

Advantage of Concept 2: It introduces automation with the use of the direct motor which reduces the mechanical effort in operating the scissors jack unlike concept 1 that is manually operated.

Disadvantages of Concept 2: The input speed produced from the direct motor drive would be too high to efficiently lift the weight of the vehicle.

Concept 3 of a Scissors Jack: Here, the scissor jack becomes fully automated due to the introduction of the gear motor as shown in figure 4.

Fig. 3: Conceptual design 2 of a scissors jack

Fig. 4: Conceptual design 3 of a scissors jack

*Advantages of Concept 3: (*i) It is durable and reliable [i] Stress will not be induced on the operator because the scissors jack will be driven by the motor [ii] It will not take time to raise the vehicle to the maximum height

Disadvantage of Concept 3: The overall manufacturing cost of the system increases because elements of this system have to manufacture separately to a high degree of precision.

Evaluation and Selection of Concept Using Decision Matrix: The most viable concept amongst the three concepts considered is selected using a decision matrix table shown in Table 2, putting into consideration key design criteria

Table 2: Decision Matrix Table for the Selection of the Concept

		Concept 1		Concept 2		Concept 3	
Design Specifications	Weighting factor	Rating	Score	Rating	Score	Rating	Score
Cost	2	11	22	10	20	9	18
Weight	2	7	14	8	16	8	16
Size	1	6	6	7	7	8	8
Maintenance	1	6	6	7	7	8	8
Ease of application	2	6	12	10	20	10	20
Reliability	2	6	12	8	16	10	20
TOTAL			72		86		90

**Score=Rating ×Weighting factor*

From Table 2, it is observed that concept three has the highest weighted score based on the criteria considered, hence it was adopted.

Detailed Design: Mass on Front Axle = 60% of 1500 = 900kg

Dividing this weight by 2, a mass of 450kg will be used as a design criterion acting on each individual wheel of the car.

Ground Clearance of Vehicle = 213mm (Toyota Camry model with base specifications, 2024. *Automobiledimensions.com*): Maximum Height of Jack = 381mm: Minimum Height of Jack = 90mm: A 1-ton Car jack is shown in Figure 5

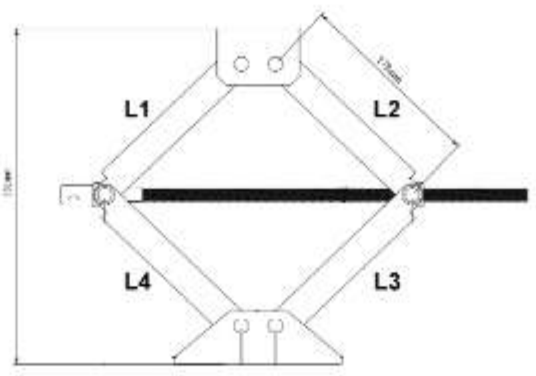


Fig. 5: A 1-ton Car Jack

Length of each arm $L_1 = L_2 = L_3 = L_4 = 178mm$

Maximum lift of the jack $(h_1 + h_2) = 381mm$

θ is the angle made by link with the horizontal when jack is at its lowest position

$\text{Cos}\,\theta = {}^{165}/_{178}$
$\theta = 22^0$

Weight of mechanism = load × acceleration due to gravity

$$W = mg \qquad (2)$$

$W = 450x9.81 = 4414.5 = 4.4KN$
Design of the Power Screw

Length of the Power screw = $W_1 + W_2 + W_3 = 380mm$

$W_1 = W_3 = 165mm, W_2 = 50mm$

The Tension T acting on the power screw is

$$T = {}^{W}/_{2} \times \tan\theta \qquad (3)$$

$$\text{Total tension} = 2 \times T = {}^{W}/_{\tan\theta} \qquad (4)$$

For a power screw under tension, where σt = tensile stress = 124N/mm^2 for mild steel (standard)
Let d_c = Core diameter

$$\text{Load} = {}^{\pi}/_{4} \times d_c = 2 \times \sigma_t \qquad (5)$$

$$2 \times T = {}^{W}/_{\tan\theta} = {}^{\pi}/_{4} \times d^2{}_c \times \sigma_t \quad (6)$$

$2 \times T = {}^{4414.5}/_{\tan 22^0} = 10926.27N$
$$d^2{}_c = {}^{W}/_{\tan\theta} \times 4 / \pi \times \sigma_t \qquad (7)$$

Hence,
$d_C = {}^{4414.5}/_{\tan 22^0} \times 4 / \pi \times 124$

$d_C = 10.6mm$

Since the screw is subjected to torsional shear stress we adopt, $d_C = 11mm$
Taking pitch, p = 3mm

Outer diameter = d_0 =$d_C + p$ = (11+3) =14mm

Mean diameter = d = $d_0 - {}^{p}/_{2} = 14 - {}^{3}/_{2}$=12.5mm

The design of the power screw is shown in Figure 6

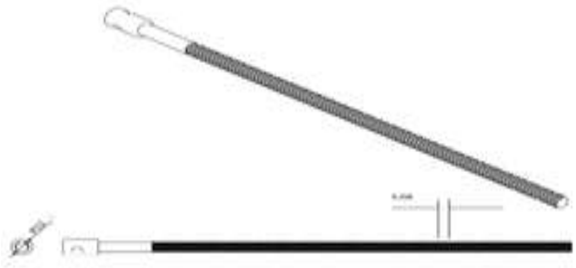


Fig. 6: Power Screw

Check for Self-Locking

$$\text{Tan}\,\alpha = {}^{Lead}\!/_{(\pi \times d)} \quad (8$$

α = helix angle

Lead, $L = 2 \times P$, since the screw has a double start square thread

$\text{Tan}\,\alpha = 2 \times {}^{P}\!/_{\pi} \times d = 2 \times {}^{3}\!/_{\pi} \times 12.5 = 0.153$

Helix angle; α = 8.70

Coefficient of friction $U = \tan\varphi$

$U = \tan\varphi = 0.25$, $\varphi = 14^0$

φ > α hence the screw is self-locking

Effort Required to Support the Load

$$E = 2 \times T tan(\varphi + \alpha) \quad (9)$$

$E = 10926.27(tan\alpha + tan\varphi)/(1 - tan\alpha \times tan\varphi)$

$E = 10926.27(tan8.70 + tan14)/(1 - tan8.70 \times tan14)$

E = 4570.56N

Torque Required to Rotate the Screw

$$\tau = E \times {}^{d}\!/_{2} \quad (10)$$

= 4570.56 × 12.5/2
= 28562.5 N/mm

Shear stress in the screw due to torque

$$\tau = \frac{16 \times T}{\pi \times d^3{}_c} \quad (11)$$

= 16 × 28562.5/ (π × 11^3) = 109.29N/mm^2

But tensile stress $\sigma_t = \frac{2 \times T}{{}^{\pi}\!/_{4} \times dc2}$ = 10926.27/ (π/4) × 11^2 = 114.973N

Maximum principal stress $\sigma_{t\,max} = {}^{\sigma_t}\!/_{2} + \sqrt{(\sigma_t{}^2 + \tau^2)/2}$ (12)

114.973/2 + √ (114.973² + 109.29²)/2 = 136.8 N/mm^2

Maximum shear stress $\tau_{max} = \sqrt{(\sigma_t{}^2 + \tau^2)/2}$ (13)

√ (114.973² + 109.29²)/2 = 79.31N/mm^2

Since the maximum stresses $\sigma_{t\,max}$ and τ_{max} within the safe limits, the design of double start square threaded screw is satisfactory

Design for Nuts: Material Selected Bronze
Design Calculations: Bearing pressure for Bronze (10.5N/mm² – 17.5N/mm²). Khurmi, RS; Gupta, JK, 2005
Let n be the number of threads in contact with the screw assumed that load is Uniformly Distributed over the cross-section area of the nut. Bearing pressure is assumed as 15 N/mm^2

$$\text{Bearing Pressure, } (P_{b)} = \frac{W}{\frac{\pi}{4} \times (d_0{}^2 - dc^2) \times n} \quad (14)$$

$$15 = \frac{10926.27}{\frac{\pi}{4} \times (14^2 - 11^2) \times n}$$

Number of threads, n = 12

Thickness of Nut = n × p = 12 × 3 = 36 mm

Width of Nut, b =1.5 × d_0 = 1.5 × 14 = 21 mm

To control the movement of nuts beyond 330 mm that is (w_1 + w_3) = (165 + 165) = 330 mm, rings of 6 mm thickness are fitted on the screw with the help of set screw

The length of screw portion = 330 + (6×2) + 36 = 378 mm ≈ 380 mm

Total length of screw is 380 mm.

Design for Pins in Nuts: Material selected Mild Steel

Design calculations:

Let d_1= diameter of pins in the nuts. Since Pins are in double shear stress

$${}^{W}\!/_{2} = 2 \times ({}^{\pi}\!/_{4}) \times d^2{}_1 \times \tau \quad (15)$$

τ = Shear stress for mild steel is 200 MPA, taking a factor of safety of 4

We have ${}^{200}\!/_{4} = 50MPa$

Load on pins = $^{W}/_{2}$ =10926.27/2 = 5463.135kg

$5463.135 = 2 \times (^{\pi}/_{4}) \times d^2{}_1 \times 50,$ Hence d_1 = 8.34 mm ≈ 8 mm

Diameter of pins head is taken as 1.5 × d_1 = 12 mm

Design for Top Plate Loading Platform: Material used is Mild Steel
Design calculations

$$\text{Bending Moment, M} = \frac{Force \times Length}{4} \quad (16)$$

F = 4414.5 N and l = w/2 = 50 mm

M =$\frac{4414.5 \times 50}{4}$ = 220725/4 = 55181.25 N-mm

$$\text{Section modulus, Z} = \frac{breadth \times height^2}{6} \quad (17)$$

b = 36 mm, h = 40 mm

Z =$\frac{36 \times 40^2}{6}$ = 9600 mm^3

Bending stress, σ_b= M/Z = 55181.25/9600 = 5.75 N/mm^2

Conclusion The permissible stress for mild steel is 124 N/mm^2 and it is greater than

σ_b = 5.75 N/mm^2. The top plate design is safe.

Design for Top Arm: Material selected Mild Steel
Design calculations
Yield stress, σ_{yt} for mild steel = 248 N/mm^2

Factor of safety $F.S$ = 2.5

Allowable stress, $\sigma_t = {}^{\sigma_{yt}}/_{F.S}$ = 248/2.5 = 99.2 N/mm^2
Width = 33mm, thickness = 4mm, height = 26mm
Cross section area (A) = (33×4) + (26×4) + (33×4) =368 mm^2

$$\text{Moment of Inertia } I_{xx} = \frac{bh^3}{12} \quad (18)$$

$$I_{yy} = \frac{hb^3}{12} \quad (19)$$

From equation (17) and (18) we have, I_{xx}= 48334 mm^4, I_{yy}= 77863.5 mm^4

Radius of Gyration

$$R_x = \sqrt{\frac{I}{A}} \quad (20)$$

$$R_y = \sqrt{\frac{I}{A}} \quad (21)$$

From equation (20) and (21) we have, R_x= 11.46 mm, R_y= 14.54 mm; Rankine's constant (a) =1/7500

Ends are hinged ($L_{eff} = L$)

Critical load, P_{cr} in vertical plane

σ_b= crippling stress = 330 N/mm^2

$$P_{cr} = {}^{(\sigma_c \times A)}\Big/\left(1 + a\left({}^{L}/_{R_y}\right)^2\right) \quad (22)$$

= (330 × 368)/ (1+ (1/7500) × (178/14.54)^2 = 119060.8657 N

P_{cr}in horizontal plane

σ_c= crippling stress = 330 N/mm

$$P_{cr} = {}^{(\sigma_c \times A)}\Big/\left(1 + a\left({}^{L}/_{2} \times R_x\right)^2\right) \quad (23)$$

= (330 × 178 × 33)/ [1+ (1/7500) × (178/2 × 11.46)^2] = 1922956.105 N

Bottom Arm
Material selected Mild Steel

Design calculations
Yield stress, σ_{yt} for mild steel = 248 N/mm^2

Factor of safety (F.S) = 2.5

Allowable stress, $\sigma_t = {}^{\sigma_{yt}}/_{F.S}$ = 248/2.5 = 99.2 N/mm^2

Width = 33mm, thickness = 4mm, height = 30mm

Cross section area (A) = (33×4) + (30×4) + (33×4) =384 mm^2

$$\text{Moment of Inertia } I_{xx} = \frac{bh^3}{12} \quad (24)$$

$$Iyy = \frac{hb^3}{12} \quad (25)$$

From equation (3.24) and (3.25) we have,

I_{xx}= 74250 mm^4, I_{yy}= 89842.5 mm^4

Radius of Gyration

$$R_x = \sqrt{\frac{I}{A}} \quad (26)$$

$$R_y = \sqrt{\frac{I}{A}} \quad (27)$$

From equation (26) and (27) we have,
R_x= 13.9 mm, R_y= 15.2 mm
Rankine's constant (a) =1/7500
Ends are hinged ($L_{eff} = L$)
Critical load, P_{cr} in vertical plane
σ_c= crippling stress = 330 N/mm²

From equ. (22)

P_{cr}= (330 × 384)/ (1+ (1/7500) × (178/15.2)²) = 124444.545 N
P_{cr} in horizontal plane
σ_c= crippling stress = 330 N/mm²
$R_x = 13.9mm$

From Equ. (23)

P_{cr} = (330 × 178 × 33)/ (1+ (1/7500) × (178/2 × 13.9)²) = 1896943.366 N

Bottom Plate (Support): Material used Mild Steel: The size and shape of the bottom plate have been selected to provide the stability to the power Scissor Jack. The size of bottom plate is 148 ×100 × 3 mm.

Table 3 shows the design parameters, designed parts and design values

Table 3: Design parameters, designed parts and design values

S/N	Designed Parts	Design Parameters	Design Value
1	Power Screw	Torque	28562.5N/mm
2	Nuts	Number of threads Thickness of nut Width of nut	12 36mm 21mm
3	Pins	Diameter of pins Diameter of pin head	8mm 12mm
4	Top arms	Moment of inertia, Radius of Gyration Critical load	48334mm⁴, 77863.5mm⁴ 11.46mm, 14.54mm 119060.8657N,1922956.105N
5	Bottom arms	Moment of inertia Radius of Gyration Critical load	74250mm⁴, 89842.5mm⁴ 13.9mm, 15.2mm 124444.545N, 1896943.366N
6	Top plate	Bending stress	5.75 N/mm²

The Automated Jack Assembly: Figure 7 shows the setup of the automated scissors jack with the electric bike brush gear motor. The design is simplistic and gives the benefit of not having heavy parts or complicated assembly pattern of added members. It can easily be fitted to the underside of an axle wheel for lifting purpose.

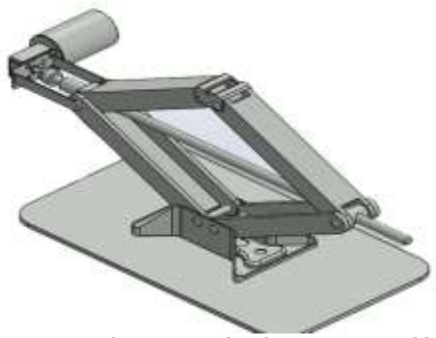


Fig. 7: The Automated Jack Setup in Assembly.

Isometric View and Orthographic Projection: The orthographic projection of the automated scissors jack is shown in Figure 8

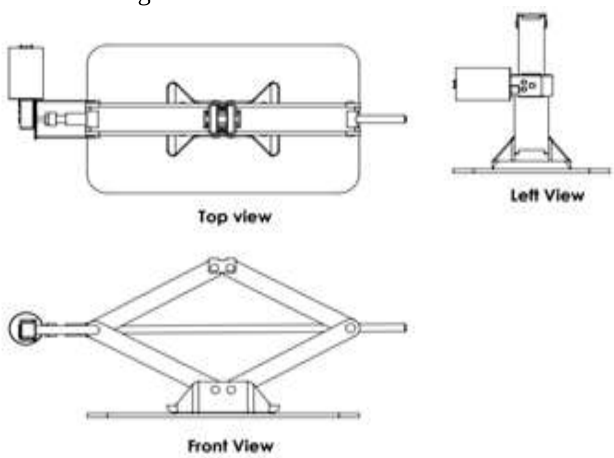


Fig. 8: Orthographic projection

The Isometric view of the automated scissors jack is shown in Figure 9

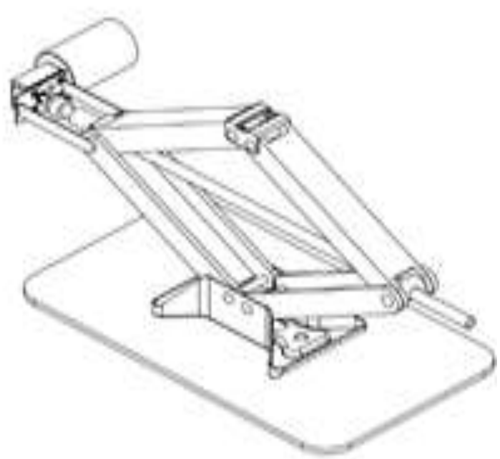
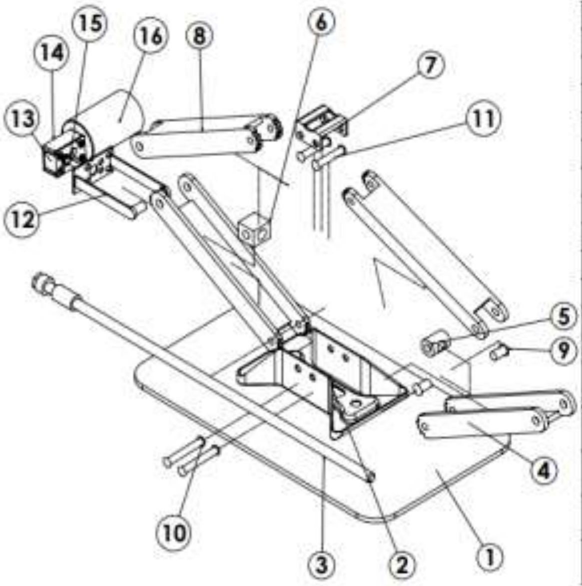


Fig. 9: Isometric view of the automated scissors screw jack

The exploded view of the automated scissors jack assembly is shown in Figure 10



ITEM NO.	PART NUMBER	QTY.
1	Ground Plate	1
2	Base Bracket	1
3	Axle Drive	1
4	Lower Channel	2
5	Cylinder Holder	1
6	Axle Support	1
7	Top Plate	1
8	Upper Channel	2
9	Pin	2
10	Base Pin	2
11	Top Pin	2
12	Brace	1
13	TS-32GZ370-5300, COVER	1
14	TS-32GZ370-5300, GEAR BODY	1
15	TS-32GZ370-5300, SHAFT	1
16	TS-32GZ370-5300, DC MOTOR BODY	1
17	TS-32GZ370-5300, DC MOTOR SHAFT	1

Fig. 10: An exploded view of the automated scissors jack assembly

RESULTS AND DISCUSSION

Table 4 shows the performance evaluation of the fabricated automated jack. In this study, the automated scissor jack was successfully designed and fabricated. The objectives of this design were achieved with the torque of about 28,562.5 N/mm from the power screw to lift and lower the car. This was achieved by the use of 250 watts, 12 V DC motor. The power was gotten from a car battery. An average load of 452.92kg was lifted in an average duration of 160 seconds.

Table 4. Performance evaluation of the fabricated automated jack

Vehicle	Toyota Camry	BMW (5 series)	Mercedes
Initial Jack Height (mm)	90	90	90
Maximum Lift Height(mm)	250	250	250
Mass Lifted (Kg)	1395	1685	2355
Lift time(seconds)	97.5	180	202

The device average Lift time = (97.5 + 180 +202)/3 = 159.83 seconds
Average Mass Lifted = (1395 + 1685 + 2355)/3 = 1811.67 Kg
An average mass lift on one side is approximately = 1811.67/4 = 452.92kg

Working Principle of The Designed Automated Scissor Jack: An existing mechanical screw jack was modified by making use of an electric motor to drive the power screw. Connecting gears were used to increase the torque supply from the motor with the pinion mounted on the shaft of motor.

The project consists of a 12 V D.C motor held horizontal for power supply to the motor. A crocodile clip provided is connected to the battery of the car which sends electricity to the DC motor to drive the scissors jack and raises it. Changes in the polarity of motor when the first crocodile clip is switched from the positive end of the battery to the negative end and the other is switched from the negative end of the battery to the positive which the lowers the jack. Thus, as the motor rotates, the pinion connected to it rotates. The gear on the lead screw meshes with the pinion and as the screw of jack rotates, the jack moves up.

Conclusion: An automated scissors jack was successfully designed and fabricated. The purpose is to reduce the risk of injury when operating the device and to also reduce the amount of human effort required and to save time. Performance evaluations were carried out on the fabricated machine. The results obtained shows that an average torque of 28562.5 N-mm is required to lift the car of 452.92kg to a maximum height of 381 mm in about 160 seconds operating time. The fabricated automated scissor jack will greatly enhance the operating mechanism and reduce the amount of human effort required in carrying out such mechanical work of lifting a car. The automated scissor jack performance was satisfactory.

Declaration of Conflict of Interest: The authors declare no conflicting interest.

Data Availability Statement: Data are available upon request from the first corresponding author or any of the other authors

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