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Design And Fabrication Of An Automated Two-Way Rotational Car Screw Jack

***Wilson, Emmanuel Okon; **Amgbari, Charles Onyinegbaliwene; & ***Etim, Kokoette Idiong; ***Nyoho, Solomon Nathaniel**

*Department of Mechanical Engineering, Akwa Ibom State Polytechnic, Ikot Osurua, PMB 1200. **ORCID Number: 0000-0001-9405-1631**. **Department of Mechanical Engineering, Federal Polytechnic Ekowe, Bayelsa State, Nigeria. **ORCID Number: 0009-0003-1301-0574**. ***Department of Mechanical Engineering, Akwa Ibom State Polytechnic, Ikot Osurua, PMB 1200

Corresponding Author: wilson.okon@akwaibompoly.edu.ng

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Abstract

The research aims to develop an innovative and efficient car-lifting device for vehicle maintenance and emergency applications. The primary objective is to automate the operation of the traditional scissors jack using a two-way rotational mechanism, eliminating the manual effort typically required. This enhancement ensures ease of use, reduced physical strain, and faster operation, making it particularly beneficial for a wide range of users, including those with limited physical strength. The design incorporates a robust and compact structure, ensuring stability and safety during vehicle lifting. A reversible motor with a gear system powers the jack, enabling smooth upward and downward motion. The two-way rotational mechanism allows for effortless adjustment, ensuring precise control during operation. Safety features such as load limiters and anti-slip pads are integrated to enhance reliability and prevent accidents. The fabrication process involves selecting high-strength materials to withstand heavy loads and implementing cost-effective manufacturing techniques to maintain affordability. The prototype undergoes rigorous testing under various load conditions to evaluate its performance, durability, and safety. This automated scissors car jack offers significant advantages over traditional jacks, including reduced time and effort for vehicle lifting, making it a practical solution for both individual users and

commercial automotive services. The project contributes to the development of user-friendly, efficient, and reliable automotive tools, aligning with modern technological advancements and addressing real-world challenges in vehicle maintenance.

Key Words: rotational mechanism, reversible motor, automated, reliability, two-way rotational mechanism

Introduction

The widespread use of traditional scissor jacks has long been accompanied by several significant challenges, making them less than ideal for many users (Kumar and Kumar, 2019). The operation of these devices requires considerable physical strength and endurance, particularly when lifting heavier vehicles (Das and Roy, 2020).

For individuals who lack the necessary physical capability—such as the elderly, people with disabilities, or those with limited upper body strength—traditional scissor jacks are difficult, if not impossible, to use effectively. This limitation highlights a critical accessibility issue in the design of conventional vehicle lifting tools (Khan et al., 2017).

Safety is another significant concern associated with the use of traditional scissor jacks. The manual application of force often leads to inconsistent results, increasing the risk of instability during operation (Arunkumar et al. 2018). Improper use or positioning of the jack can result in accidents, causing injury to the user or damage to the vehicle. Moreover, the lack of integrated safety mechanisms in many traditional designs further compounds these risks, creating a need for enhanced safety features in modern alternatives (Ghosh and Mallick, 2017).

This research problem is rooted in the need to develop an innovative solution that overcomes the physical, operational, and safety-related challenges posed by traditional scissor jacks. By designing and developing a motorized scissor jack, this study aims to provide a tool that is accessible, efficient, and reliable. Such a device would eliminate the need for manual effort, significantly reducing the physical strain on users (Shah and Raval, 2019). The integration of automated systems would also enhance operational efficiency, making it possible to lift vehicles quickly and with minimal user intervention (Sawant et al., 2017).

METHODOLOGY

The methodology imbibed by this research outlines the steps taken to accomplish the aim and objectives of te research.

Materials and Equipment

The design and production of the automated scissors car jack required the selection of appropriate materials and equipment to ensure functionality, durability, and safety. The primary materials and their purposes are listed in the table 1.

Table 1: Materials used and their purposes

Material	Purpose
Mild steel	Used for the structural components due to its strength and weldability.
Stainless steel	Employed for components exposed to environmental elements to prevent corrosion.
Hardened steel	Utilized for power transmission components to withstand mechanical wear.
Bearings	Ensured smooth motion within the mechanical assembly.
Bolts and nuts	Provided firm and secure assembly of various components.
Electric motor	Served as the primary source of power for automation.
Sensors and switches	Integrated for control and safety functions.

The equipment required for the production process is summarized in Table 2.

Table 2: Equipment used and their functions

Equipment	Function
Cutting tools	Used for shaping materials to desired specifications (e.g., saws, grinders).
Welding machines	Utilized for joining steel components.
Drilling machines	Employed for creating precise holes.
Measuring tools	Ensured dimensional accuracy (e.g., vernier calipers, micrometers).
Multimeters and oscilloscopes	Verified functionality of electrical components and connections.

These materials and equipment were selected to balance cost-effectiveness with the need for reliability and durability, ensuring the final product meets safety and performance standards.

Design Considerations and Calculations

The design of the automated scissors car jack was guided by several considerations to ensure optimal performance. The primary factor was the load capacity, which determined the dimensions and materials for the structural components (Akinlabi et al. 2015). The jack was designed to lift a maximum weight of 2,000 kilograms, a value chosen to accommodate a wide range of vehicles.

Load Capacity

The load capacity influenced the selection of the materials and the geometry of the jack. The maximum force exerted by the load is given by:

where:

- is the force (in Newtons),
- is the mass of the load (2,000 kg),
- is the acceleration due to gravity (9.81 m/s^2).

Substituting values:
Thus, the jack must support a force of 19,620 N.

Mechanical Advantage of the Scissors Mechanism
The scissors mechanism provides mechanical advantage by converting rotational input into vertical motion. The mechanical advantage (MA) is calculated using the equation:
where:

- is the length of the scissors arm,
- is the vertical displacement of the load.

For a scissors arm of length 0.5 m and a vertical displacement of 0.1 m:
This implies that the input force is multiplied by a factor of 5.

Torque Calculation
The torque required to operate the jack is determined by the lead screw's properties. The torque is given by:
Where:

- is the load (19,620 N),
- is the thread pitch diameter (0.02 m),
- is the coefficient of friction (0.15),
- is the radius of the lead screw (0.01 m).

Substituting the values:
This torque value guided the selection of the electric motor.

Motor Selection
The motor must provide sufficient power to overcome the torque and lift the load efficiently. The power of the motor is related to the torque and angular velocity by:
where is the angular velocity in radians per second. For a motor operating at 100 RPM:
Substituting:
A motor with a power rating of at least 4.5 kW was selected to ensure sufficient performance.

Stability and Safety
The stability of the jack was ensured by analyzing the distribution of forces during operation. The base dimensions were designed to prevent tipping, even under maximum load. The safety factor was calculated as:

Using a material yield strength of 250 MPa and a maximum stress of 50 MPa:
A safety factor of 5 ensured the jack could handle unexpected stresses.

Design

The design for the jack is discussed in Figure 1. Shows the force diagram for the machine.

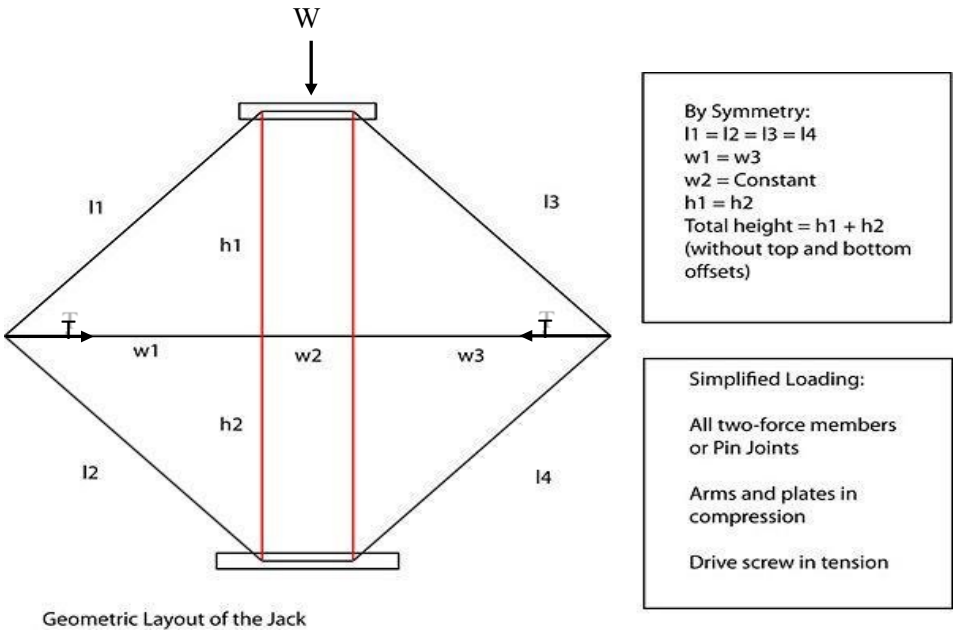


Fig 1 Layout of Scissor Jack

Power Screw

(i) Assumptions

The weight of the car is considered as 1.5 ton. The weight acting on front and rear axle is 60% and 40% of total weight respectively, hence the weight acting on front axle i.e.; 900 kg is considered for designing the jack. A weight of 450 kg acts on each wheel. And the maximum load on screw act when jack is at its lowest position. We assumed the thread on screw be a Double Start Square thread and coefficient of friction between threads is 0.20.

(ii) Design Calculations

Length of each arm = L1 =L2 =L3 =L4 =160mm

Length of the power screw = (w1+w2+w3) = 350mm **w1 = w3 =** 150 mm **w2** = 50 mm

Maximum lift of the jack = **(h1+h2)** = 300 mm

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

cos (β) = (175-25)/160 = 20.36˚

W = (load * g) = (450*10) = 4500 N = 4.5 kN

The tension T acting on the power screw is shown in the above Fig 4.1.

Tension, T = W/2*tan (**β**) Total tension = 2*T = W/tan (**β**)
For a power screw under tension we can take T_t = 124 N/mm^2 for mild steel
Let $\mathbf{d_c}$ be the core diameter of the screw. But load on the screw is
Load = $\mathbf{(\pi/4)* d_c^2 *} T_t$
So,
2*T = W/tan (**β**) = (π/4)* d_c^2 * $\mathbf{T_t}$ 2*T **= 4.5 kN/tan (20.36°) = 12123.44 N** d_c^2 = (W/tan (**β**))*(4/ (π* $\boldsymbol{\beta_t}$))
Hence, $\mathbf{d_c}$ = 11.34 mm
Since the screw is subjected to torsional shear stress we adopt, $\mathbf{d_c}$ = 14 mm
Taking pitch, **P** = 2 mm
Outer diameter, $\mathbf{d_o = d_c + P}$ = (14+2) = 16 mm
Mean diameter, $\mathbf{d = d_o - P/2}$ = 16-2/2 =
15 mm Check for self-locking **tan (β) =**
Lead/π*d; = helix angle
Lead L = 2*P; since the screw has a double start square thread.
tan (β) = 2*p/π*d = 2*2/ π*15 = 0.084
Helix angle; **β = 4.85°**
Coefficient of friction; **μ = tan (β)** = 0.20; friction angle; **β** = 11.3°
Effort required to support the load = **2*T tan (β + β)**
= 12123.44 **(tan (β) + tan (β))/ (1- (tan (β) * tan (β)))**
= 3510.715 N
Torque required to rotate the screw = **effort *d/2** = 3510.715 * 15/2 = 26330.36 Nmm
Shear stress in the screw due to torque **= 16*T/ (π* $\mathbf{d_c^3}$)** = 16*26330.36/ (π*14^3)
= 48.87N/mm^2
But tensile stress T_t **= 2*T/ (π/4) * $\mathbf{d_c^2}$** = 12123.44/ (π/4) * 14^2 = 78.755 N
Maximum principal stress $T_{t\,max}$ **= T t/2 +** $\mathbf{T(T_t^2 + T^2)/2}$ = 102.13 N/mm^2
Maximum shear stress T_{max} = $\mathbf{T(T_t^2 + T^2)/2}$ = 62.76 N/mm^2
Since the maximum stresses $T_{t\,max}$ and T_{max} within the safe limits, the design of double started square threaded screw is satisfactory.

B. Nut

(i) Material Selected Bronze

(ii) Design Calculations

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. Allowable **Bearing pressure** between the threads (P_b) are

Table 3: Nut Parameters

Material		**Safe Bearing pressure (N/mm^2)**	**Rubbing speed at thread pitch diameter**
Screw	Nut		
Steel	Bronze	12.6 - 17.5	Low speed < 2.4 m/min
Steel	C.I	11.2 - 17.5	Low speed < 3 m/min

Bearing pressure is assumed as 15 N/mm^2

P_b = (2*T)/((π/4)*(do^2-dc^2)*n)

15 = (12123.44)/ ((π/4)*(16^2-14^2)*n)

Number of threads, **n** = 10.6 ≈ 11

In order to have good stability let n=11

Thickness of Nut = **n*p** = 11*2=22 mm

Width of Nut **b** =**1.5*do** =1.5*16=24 mm

To control the movement of nuts beyond 300 mm the rings of 8 mm thickness are fitted on the screw with the help of set screw

The length of screw portion = 300 + (8*2) + 22 = 338 mm ≈ 350 mm

Total length of screw is 350 mm.

C. Pins in Nut

(i) Material selected Mild Steel

4.3.2 (ii) Design calculations

Let **d_1** = diameter of pins in the nuts

Since Pins are in double shear stress

Load on pins = **W/2 = 2*(π/4)*d_1^2* =**12123.44/2

Shear stress = 50 MPa for steel

Hence **d_1 =** 8.78 mm ≈ say 10 mm

Diameter of pins head is taken as 1.5***d_1 =** 15 mm and thickness be 4 mm

D. Top Arm

(i) Material selected Mild Steel

4.4.2 (ii) Design calculations

σ_{yt} for mild steel = 248 N/mm^2

Factor of safety (F.S) = 2.5 **σ_t** = **σ_{yt}/F.S**=248/2.5=99.2 N/mm^2 **σ_c** = 1.25***σ_t** = 1.25*99.2 = 124 N/mm^2 **Cross section area (A)** = (40*3) + (24*3) + (40*3) = 312 mm^2

Moment of Inertia I_{xx} = 47376 mm^4, I_{yy}= 51009.38 mm^4

Radius of Gyration R_x= 12.323 mm, R_y= 12.786 mm

Rankine's constant (a) =1/7500

Ends are hinged (L_{eff} = L)

P_{cr} in vertical plane σ_c= crippling stress = 330 N/mm^2

P_{cr} = (σ_c*A)/(1+a*(L/ R_y)^2)= (330*312)/(1+(1/7500)*(160/12.786)^2)

= **100854.26 N** P_{cr} in horizontal plane σ_c= crippling stress = **330 N/mm^2**

P_{cr} = (σ_c*A)/(1+a*(L/2*R_x)^2)= (330*160*40)/(1+(1/7500)*(160/2*12.323)^2)

= 2100198.258 N

Since Buckling load is more than Design load the dimensions of the link safe.

Bottom Arm

(i) Material selected Mild Steel

4.5.2 (ii) Design calculations

σ_{yt} for mild steel = 248 N/mm^2

Factor of safety (F.S) = 2.5 σ_t = σ_{yt}/**F.S**=248/2.5=99.2 N/mm^2 σ_c = 1.25*σ_t = 1.25*99.2 = 124 N/mm^2

Cross section area (A) = (40*3) + (30*3) + (40*3) = 330 mm^2

Moment of Inertia I_{xx} = 72270 mm^4, I_{yy}= 54469.31 mm^4

Radius of Gyration R_x= 14.79 mm, R_y= 12.84 mm

Rankine's constant (a) =1/7500

Ends are hinged (L_{eff} = L)

P_{cr} in vertical plane σ_c= crippling stress = 330 N/mm^2

P_{cr} = (σ_c*A)/(1+a*(L/ R_y)^2)= (330*330)/(1+(1/7500)*(160/12.84)^2)

= **106691.09.09 N** P_{cr} in horizontal plane σ_c= crippling stress = **330 N/mm^2**

P_{cr} = (σ_c*A)/(1+a*(L/2*R_x)^2)= (330*160*40)/(1+(1/7500)*(160/2*14.8)^2)

= 2103804.02 N

Since Buckling load is more than Design load the dimensions of the link safe.

Top Plate (Loading Platform)

(i) Material used Mild Steel

(ii) Design calculations

Moment, M = (p*l)/4 p = 5000 N **l** = 50 mm

M = (5000*50)/4 = 250000/4 = 62500 N-mm **Z = (b*h^2)/6** = (36*40^2)/6 = 9600 mm^3 **b** = 36 mm, **h** = 40 mm **σ_b = M/Z** = 62500/9600 = 6.51 N/mm^2

Conclusion The permissible stress for mild steel is **124 N/mm^2** and it is greater than **σ_b = 6.51 N/mm^2**

The top plate design is safe.

G. Bottom Plate (Support)

(i) Material used Mild Steel

The size and shape of the bottom plate have been selected to provide the stability to the power Scissor Jack.

Fixing the dimensions of bottom plate as 120*70*3 all in mm.

DESIGN DRAWINGS

(i) Power Screw Axle

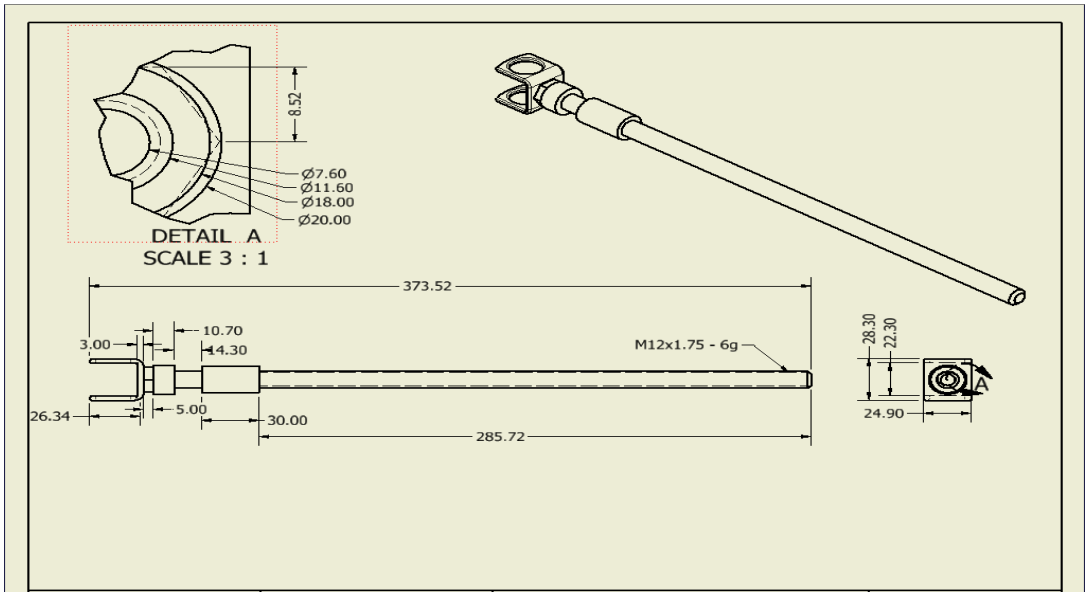


Fig 2 : Power Screw Axle

Axle Support

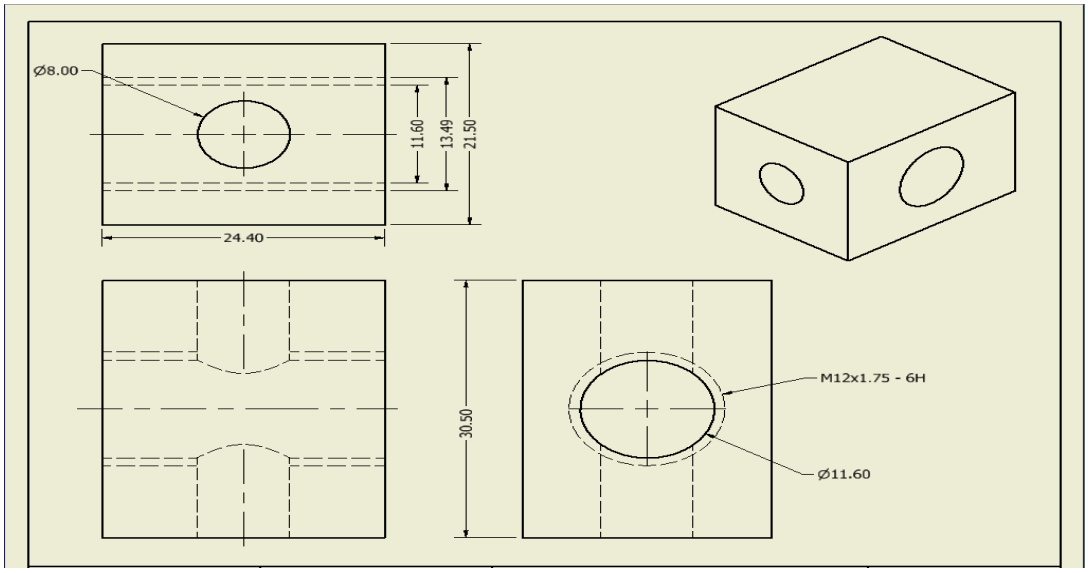


Fig 3: Axle Support

Lower Support

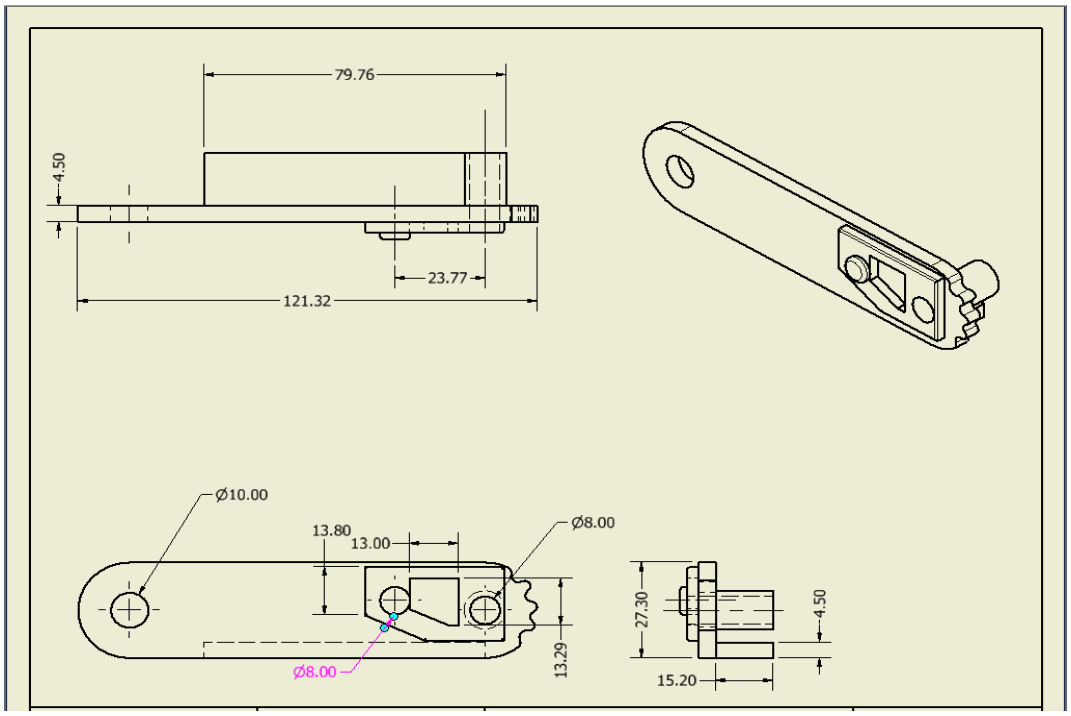


Fig 4: Lower Support

Upper Support

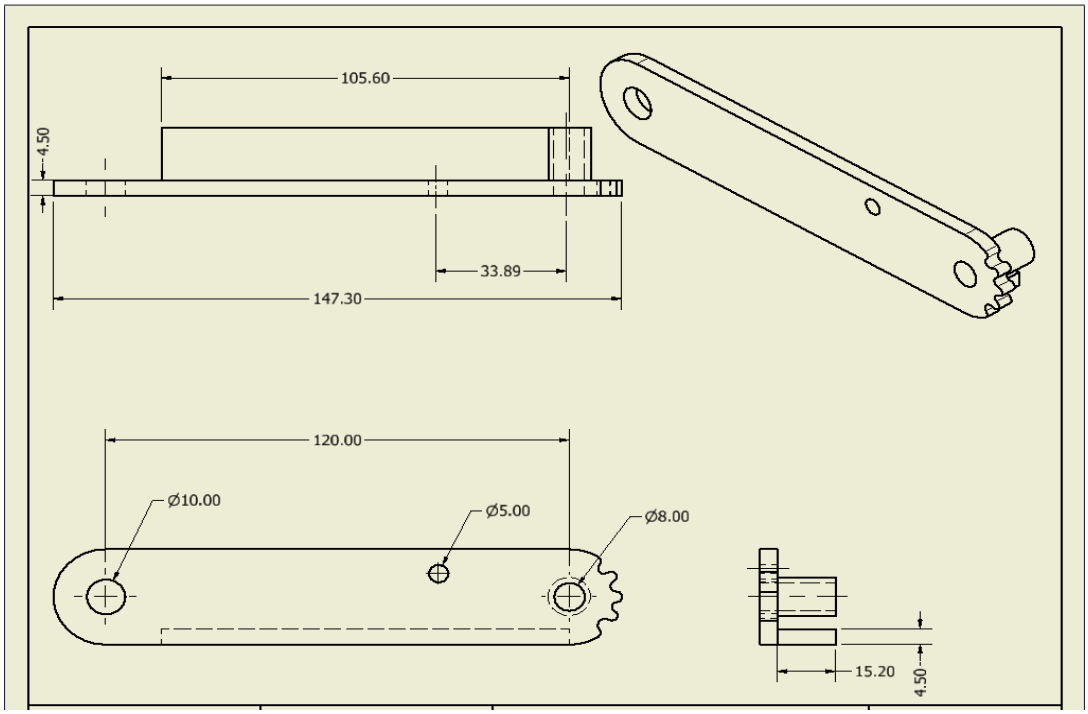


Fig 5: Upper Support

Figure 6 shows the cylinder holder drawing and isometric diagram respectively.

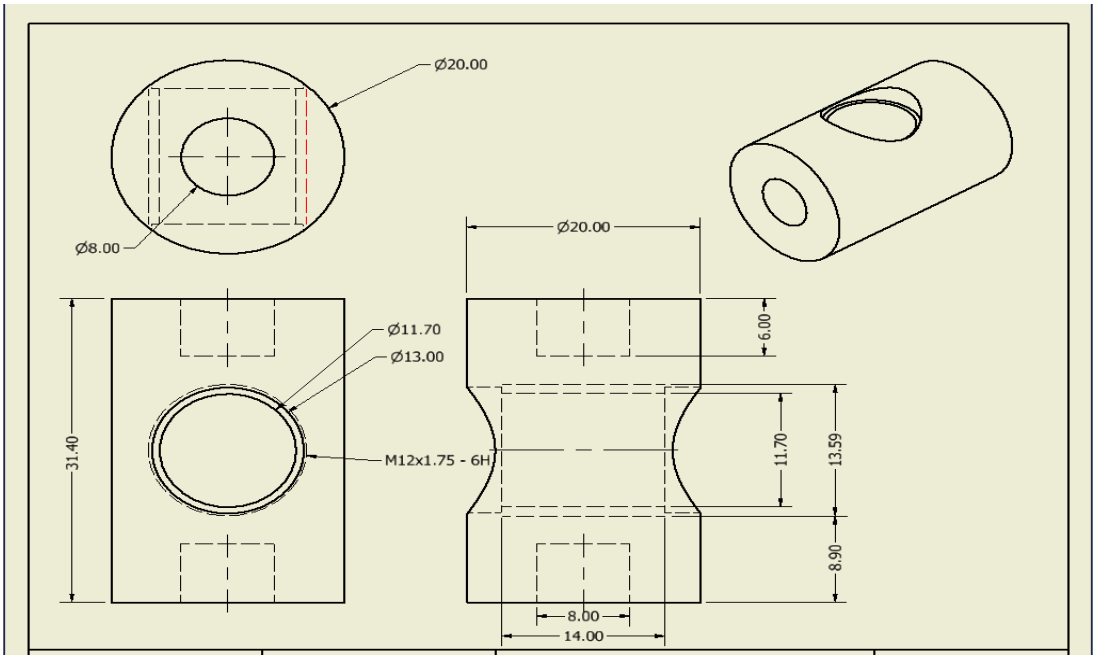


Fig 6: Cylinder Holder

ASSEMBLY DRAWING

Scissor jack

The orthographic view and isometric views of the scissor jack are shown in figure 3.6 A to C. Figure 7 A, B, C and D shows the motor with housing in orthographic view, separated and fixed respectively while figure 3.8 shows the isometric diagram for the coupled device.

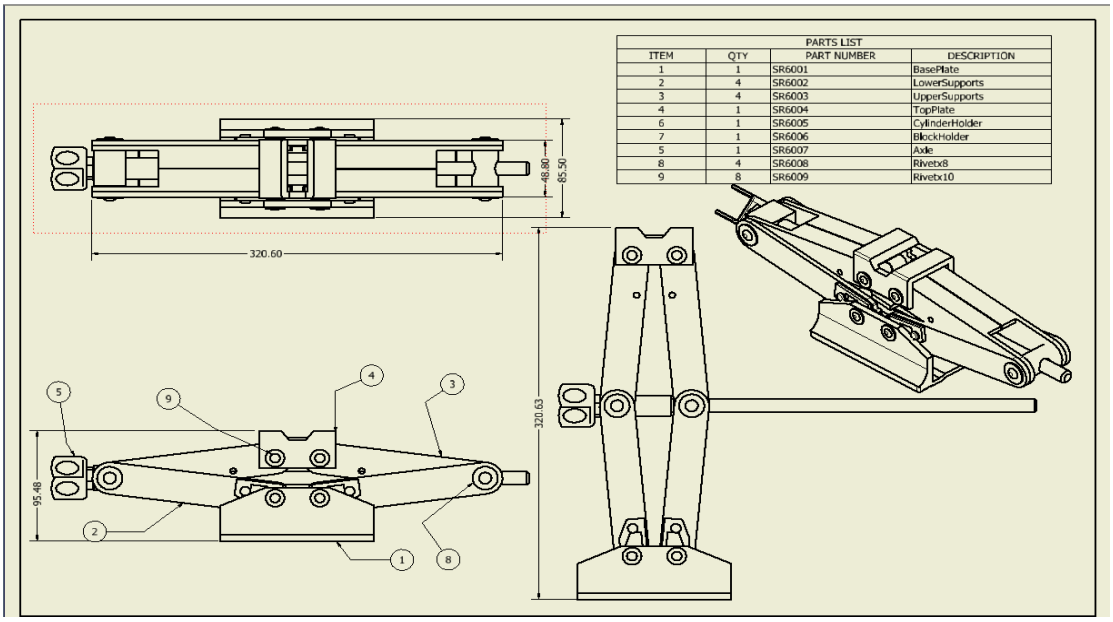


Fig 7A: Part List for Scissor Jack

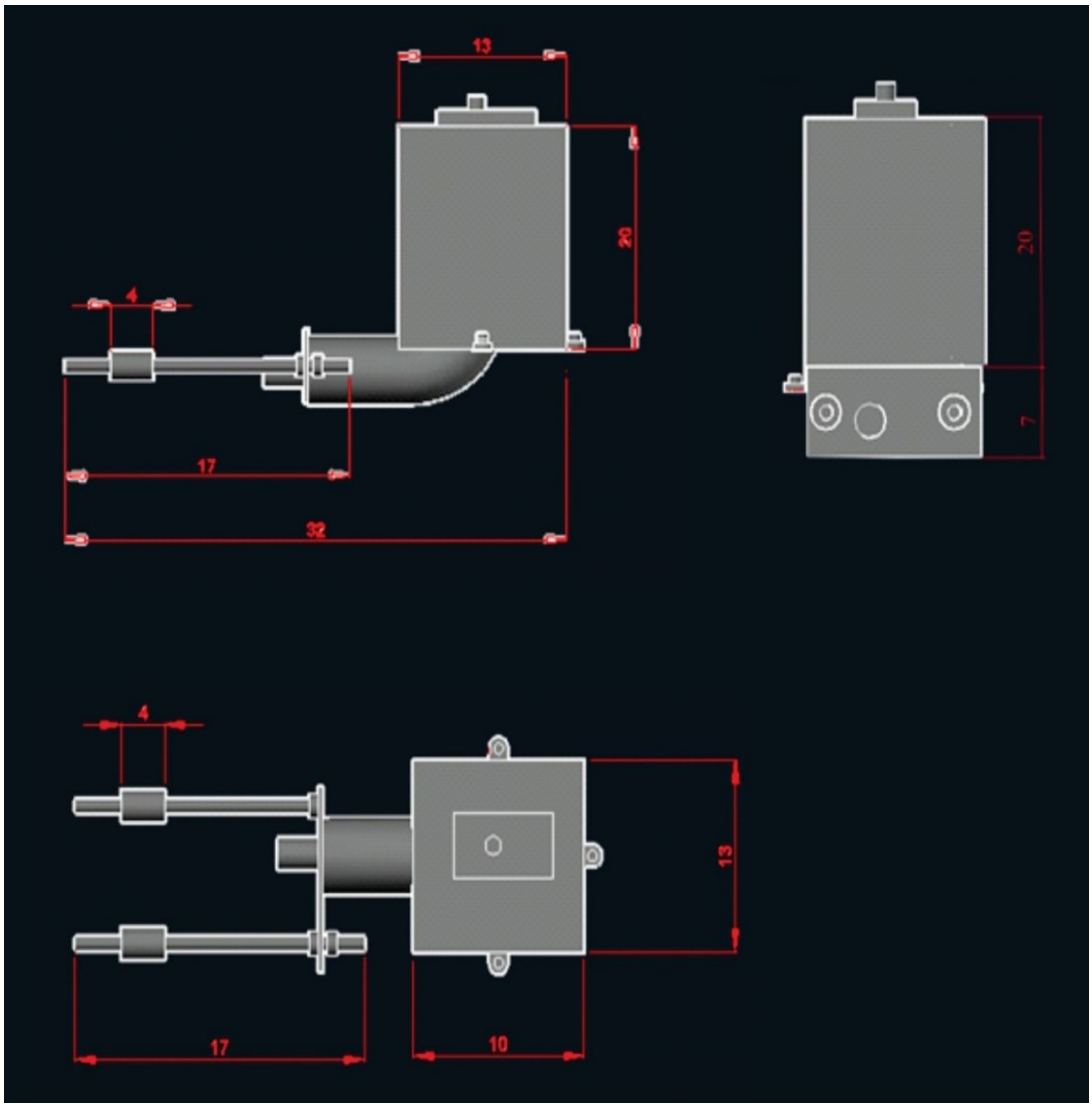


Fig 7B: Orthographic view for two-way motor and its housing

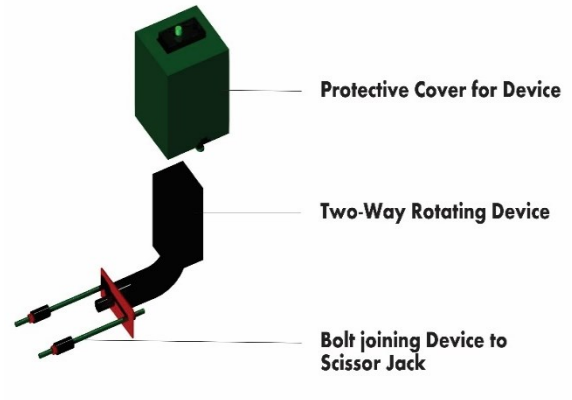


Fig 7C B: Motor with housing separated

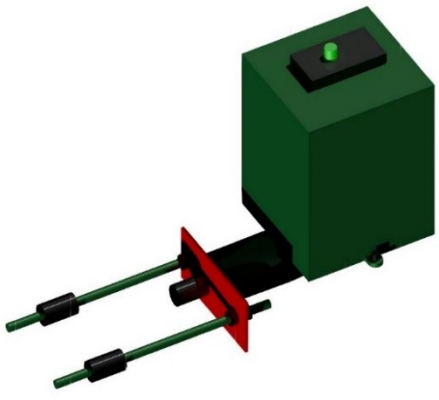


Fig 7D: Motor with housing coupled

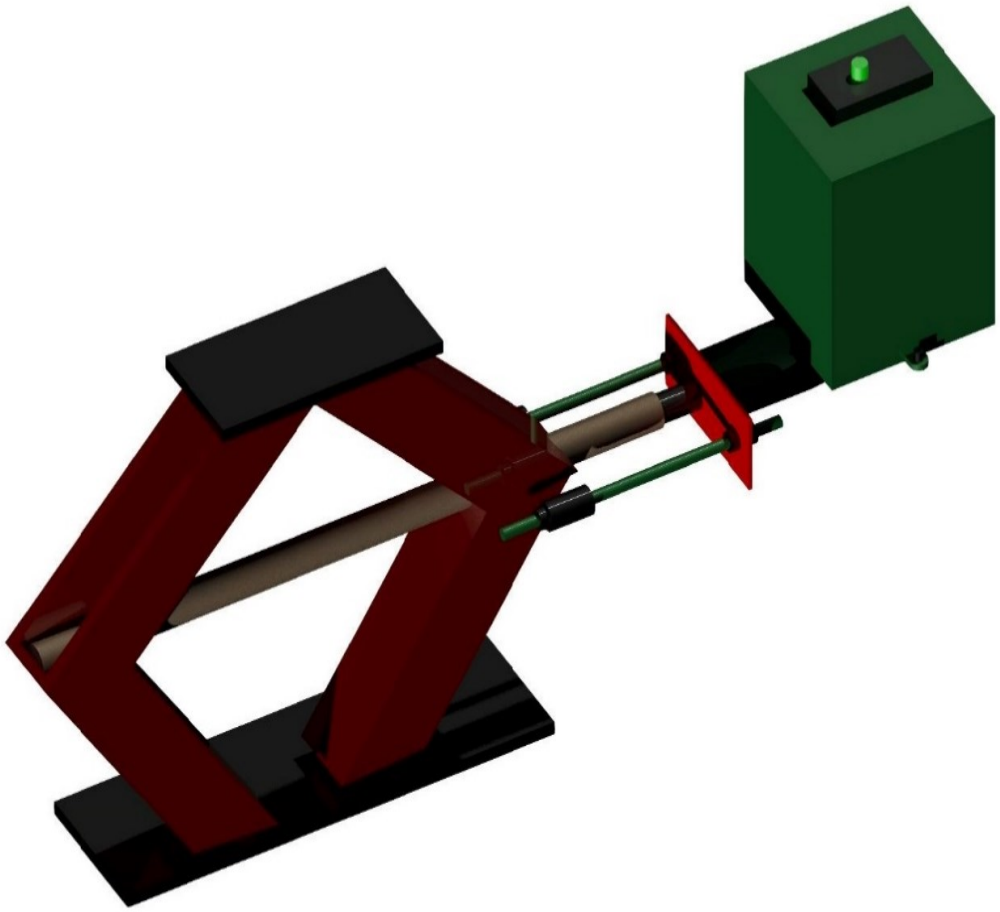


Fig 8: Integration of Jack with motor and housing

RESULTS AND DISCUSSION

The fabrication process started with identification of suitable materials which were used for building and designing the various parts of the prototype.

The following parts were fabricated independently before being coupled. All with specifications from the design. The parts being:

i. Lower and Upper Supports
ii. Power Screw Axle
iii. Cylinder Holder (Trunions)
iv. Jack Top and Base
v. Power Source

PRODUCED AND COUPLED MACHINE

The coupled device is showen in Fig. 9 already painted and poised for performance. The motorized jack was tested on various vehicle types and gave excellent performance.



Fig 9: Two-way rotating automated scissors jack

CONCLUSION

In this project a prototype of power scissor jack which can be operated by a power gun has been designed and fabricated. The jack has been designed to a pay load of 4.5kN. The salient features of the present fabrication are elimination of human effort to operate the jack, through a simple electrical motor which was actuated by a 12 V. All the elements of the jack are fabricated in the machine shop. The assembly of the component can be achieved in 100 minutes. Another feature of the unit is provision of two trunions on both the sides of the jack to ensure jerk free operation. The elements which are useful are readily available commercially for each and early replacement of failed components if required.

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