Modification of the Existing Design of a Car Jack

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Abstract
This paper analyzes the modification of the existing motor screw jack by incorporating an electric motor in the screw in order to make load lifting easier. In this modified design, the power screw is rotated through its connecting gear with the pinion gear when electrical power flows through the cigarette lighter receptacle connected to the motor, plugged to the automobile 12 V battery source to generate power for the prime mover (motor), which transmits its rotating speed to the pinion gear meshing with the bigger gear connected to the power screw to be rotated with required speed reduction and increased torque to drive the power screw. The significance and purpose of this work is to modify the existing car jack in order to make the operation easier, safer and more reliable in order to reduce health risks especially back ache problems associated with doing work in a bent or squatting position for a long period of time. The modified car jack is easy to use by pregnant women or whoever had problem with the vehicle tyres along the road. The designed motorised jack will also save time and requires less human energy to operate. The design when adopted will effectively curb the problems associated with Ergonomics - which is a fundamental concept of design process.

Keywords: jack, power screw, ergonomic, pinion gear, speed reduction, torque.

INTRODUCTION
Doing work in a bent or squatting position for a period of time is not ergonomic to human body. It gives back ache problem in due of time. A mechanical jack is a device which lifts heavy equipment and vehicles so that maintenance can be carried out underneath. [Budynas, and Nisbett, 2008]. Car jacks usually use mechanical advantage to allow human being to lift a vehicle. There are two types of automotive car jacks: Hydraulic and Screw types. Most car jacks that are included with cars are screw types. These two categories also have many subcategories of jacks. Hydraulic jacks have the shape of a bottle, or are built into a trolley (the "floor jack") or the like. By operating the handle, which is a lever (a simple machine), fluid is compressed and routed to an actuating cylinder. This results in lift. Other examples of hydraulic types are hydraulic jack, bottle jack, floor jack, friction jack, and racketing jack.

A jackscrew is a type of jack which is operated by turning a lead screw. It is also known as a screw jack, and is commonly used as car-jacks. In the case of a screw jack, a small force applied in the horizontal plane is used to raise or lower large load. [Khumri and Gupta, 2005]. Of the screw-type mechanisms, there are scissor jacks, common in newer cars, and bumper jacks, common in older cars. A jackscrew's compressive force is obtained through the tension force applied by its lead screw. An Acme thread is most often used, as this thread is very strong and can resist the large loads imposed on most jackscrews while not being dramatically weakened by wear over many rotations. An inherent advantage is that, if the tapered sides of the screw wear, the mating nut automatically comes in closer engagement, instead of allowing backlash to develop [Rajput, 2007]. These types are self-locking, which makes them intrinsically safer than other jack technologies like hydraulic actuators which require continual pressure to remain in a locked position. The correct projection of a screw thread is tedious and takes a considerable time, so threads are shown conventionally on engineering drawings. [Parker and Pickup, 1976]. Most jackscrews are lubricated with grease. Ball screws are a more advanced screw mechanisms that uses a recirculation-ball nut to minimize friction and prolong the life of the screw threads. The thread profile of such screws is semicircular to properly mate with the bearing balls. The disadvantage to this type of screw is that it is not self-locking. [James, M. Gere 2006]. Jackscrews form vital components in equipment. For instance, the failure of a jackscrew on a McDonnell Douglas MD80 due to a lack of grease resulted in the crash of Alaska Airlines Flight 261 off the coast of California in 2000. Due to large numbers of examples of compound stresses met with in engineering practice, the cause of "failure" or permanent set under such conditions has attracted considerable attention [ Rajput, R.K. 2010].

When all the forces that act on a given part are known, their effect with respect to the physical integrity of the part still must be determined [Leonardo Spiegel and George Limbrunner 1995]. It
would therefore be reasonable to suppose that fatigue failure due to lack of allowance does not occur [Raymond and Higgins 1990]. Screws Application is used in the elevation of vehicles or objects. The operation of the screw jack is such that it comprises a handle for driving a bolt element (Lead Screw) manually so as to adjust the height of the Jack to elevate a vehicle or the object. The operation of the jack manually makes it difficult for most women and the elderly to operate since much effort is needed to drive the screw jack which results in low linear speed and time consuming. These presently available jacks further require the operator to remain in prolonged bent or squatting position to operate the jack. Doing work in a bent or squatting position for a period of time is not ergonomic to human body. It will give back ache problem in due of time. Suppose car jacks must be easy to use by pregnant women or whoever had problem with the tyres along the road.

The objective of this paper is therefore to modify the existing design of car jack by incorporating an electric motor into the existing screw jack to make the operation easier, safer faster and more reliable.

**Principle of Operation of the Motorised Screw Jack**

In operation, the jack will lift a load in contact with the load platform when the power screw is rotated through its connecting gear with the pinion gear when electrical power flows through the cigarette lighter receptacle connected to the motor, plugged to the automobile 12V battery source to generate power for the prime mover (motor), which transmits its rotating speed to the pinion gear meshing with the bigger gear connected to the power screw to be rotated with required speed reduction and increased torque to drive the power screw. The power screw rotates within the threaded bore of its connecting members (11 and 16) in the clockwise direction that will cause the connecting members to be drawn along the threaded portion towards each other during a typical load-raising process. During the typical load-raising process, the jack will first be positioned beneath the load to be lifted such that at least a small clearance space will exist between the load platform and the object to be raised. Next, power screw will be turned so that the load platform makes contact with the object and the clearance space is eliminated. As contact is made, load from the object will be increasingly shifted to the load platform and cause forces to be developed in and transmitted through lifting members and connecting members. The force transmitted through the connecting members will be transferred at the threaded bore to the lead Acme threads, there within. A switch button connected to the motor is used to regulate the lifting and lowering process.

**MATERIALS AND METHOD**

The Pictorial view of the designed motorised jack is shown in fig.1
Fig. 1: Pictorial view of the Motorised Screw Jack Design

Table 1. Components Part List

<table>
<thead>
<tr>
<th>Part Number</th>
<th>Description</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Button Switch</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>Embedded Spring slider</td>
<td>1</td>
</tr>
<tr>
<td>3</td>
<td>Base</td>
<td>1</td>
</tr>
<tr>
<td>4</td>
<td>Pinion Gear</td>
<td>1</td>
</tr>
<tr>
<td>5</td>
<td>Gear</td>
<td>1</td>
</tr>
<tr>
<td>8</td>
<td>Power Screw</td>
<td>1</td>
</tr>
<tr>
<td>10</td>
<td>Pin</td>
<td>8</td>
</tr>
<tr>
<td>13</td>
<td>Load Platform</td>
<td>1</td>
</tr>
<tr>
<td>17</td>
<td>Key</td>
<td>2</td>
</tr>
<tr>
<td>18</td>
<td>Motor</td>
<td>1</td>
</tr>
<tr>
<td>19</td>
<td>Cable to Cigarette Lighter Receptacle in an Automobile (12V)</td>
<td>1</td>
</tr>
<tr>
<td>6,9,12,15</td>
<td>Lifting Member</td>
<td>1</td>
</tr>
<tr>
<td>7,11,14,16</td>
<td>Connecting Member</td>
<td>1</td>
</tr>
</tbody>
</table>

Screw Jack Pictorial View

Fig. 2: Pictorial View and Orthographic Projection of the Motorised Screw Jack

Components with their Material Selection
The life span of the jack will depend greatly on the type of materials used for each component to avoid failure. The contact members, connecting members, lifting members, pins and the power screw will all use the High Strength Low-Alloy Steel with 50 ksi (345 N/mm²) minimum yield point, 70 ksi (485 N/mm²) minimum tensile strength and 21% elongation) due to the following reasons:
- Good Machinability
- Good Ductility
- High Strength
- Wear Resistance
- Ease of producing complicated parts
- Economical

Lifting Members
These members are made from simple c-shapes. The web of the lifting member is cut out near the pin connections to allow proper serviceability of the scissor jack at its maximum and minimum heights. Members 12 and 15 have ideal connections to balance the load between the left and right side.

Connecting (Sleeve) Members
The sleeve channels are to open inwards as shown in Figure 4 with an arrow. This is so because the flanges (Lips) are subjected to tension instead of compression. The bending moment from the power screw creates tension on the inner edge of the sleeve and compression on the outside edge. Tension along flanges on the inside prevents the possibility of localized bucking in the flanges from compressive forces.

Fig. 3: Lifting Member
Contact Members (Base and Load Platform)
The members that make contact with ground and the service load are members 3 and 13 respectively. Member 3 has a broader area to provide a stable base for the mechanism while servicing the load. Member 13 has an attached flat plate to one of the connecting members which provide sufficient contact area. Most scissor jacks have ridges which lower the area of contact. This causes stress concentrations which can damage the underside of a car.

Power Screw
The power screw is a single Acme threaded screw with collar at both ends, with one end in contact with Member 11 and the other end having a square key way to enable the transmission of torque from the gears. The collar is assumed to be frictionless and the power screw has been designed to be self-locking.

Gear Reduction System
Gearing system is used to transmit motion from one shaft to another. The spur gear will be used for the speed reduction for this design. Both the pinion and the gear have key ways which allow connection between the motor shaft and the power screw respectively. The gear unit will be housed, to prevent it from excessive exposure to dust and dirt which can bring about wear in the gear teeth, and lubricant breakdown. Keys are used to prevent relative motion between a shaft and the connected member through which torque is being transmitted.

Connecting Pins
The pins are used as fasteners at the various joints of the members. The existence of the jack will depend on the ability of the pin not to fail under sudden shear, tensional and compressive forces. Hence, the design of the pin will check all these effects to avoid failure of the pin.
The Spring and its Supporting Members
Spring is a flexible object that is used to store mechanical energy and is usually made of steel. The spring chosen for this design will be made of alloy steel, for rigidity and elasticity. This device allows controlled application of force or torque; the storing and release of energy can be another purpose. The spring is incorporated to extend the motor and its components along with the movement of the lifting members to avoid the gears meshing from disengaging. The spring is embodied in a sliding member made of mild steel with the inner part of the cover also coated with mild steel coating.

Design Calculations
The main physical parameters of the design are determined through the appropriate calculations and practical considerations with reasonable assumptions. The purpose of the design calculations is the assumptions and predictions of possible stresses or deformations in the major parts and thereby chooses reasonable dimensions for those parts of the machine so it will satisfy the design objectives and also attain its assumed lifespan.

Motor Selection and Gear Design
The drive mechanism of the jack will make use of a motor as its prime mover. The motor required for this particular design is expected to transmit a relatively low speed at high power. The following data have therefore been selected based on the above requirements:

(a) Power of the Prime Mover: Direct Current (D.C) Electric Motor, 2.34 KW/276 RPM
(b) (i) Diameter of gear; \( D_g \): 90 mm
(ii) Diameter of Pinion gear; \( D_p \): 30 mm

The two diameters have been chosen on the basis of achieving a reasonable velocity ratio of 1:3 between the pinion and the gear respectively. The drive transmission of the D.C Motor is via the spindle.

Gear Design
In the design of gears, the following factors were considered:
- The input and output torque of the gears
- Type, dimensions and strength of gears
- Durability
- Wear
- Face width for adequate strength
- Economy of manufacture

Calculation of Gear parameters

\[ D_g : 90 \text{ mm } \quad D_p : 30 \text{ mm} \]

Speed of pinion gear, \( n_p = 276 \text{ rpm} \)

Speed of gear, \( n_g \)

Number of teeth of pinion gear, \( N_p \)

Number of teeth of gear, \( N_g \)

But, \( n_p \times D_p = n_g \times D_g \) \hspace{1cm} (1)

\[ \nu = \frac{(30 \times 276)}{\nu_0} = 92 \nu_{rpm} \]

Design Equation
Since the pitch diameters of the gears are known, the following Lewis equation will be used (Khurmi, 2005).

\[ \frac{1}{m^2} = \frac{s \kappa n^2}{F} \] \hspace{1cm} (2)

where,
- \( s \) = Allowable stress (MN/m²)
- \( F \) = Transmitted force (N)
- \( m \) = Module (m)
- \( \kappa \) = Form factor
- \( k = 4 \), upper limit

Since both gears will be made of the same material, the weaker gear (pinion) will be the weaker gear and governs the design. The number of teeth on either gear for spur gears should not be less than 15. Assuming \( N_p = 16 \) teeth for the pinion gear (Khurmi, 2005).

Torque transmitted by pinion (T),

\[ T = \frac{60 \times \text{Power in KW}}{2 \times n_p} \]

\[ T = \frac{60 \times 2.34}{2 \times 276} = 0.097 \text{ Nm} \]

Transmitted force of the pinion gear,

\[ F = \frac{T}{D_p} \] \hspace{1cm} (3)

\[ F = \frac{0.097 \times 30}{20} = 5397.33 \text{ N} \]

Pitch line velocity of the pinion gear

\[ \nu = \frac{2 \pi n_p D_p}{60} \] \hspace{1cm} (4)
Allowable stress, $S_{\text{allowable}} = \frac{s_0 \times \left( \frac{3}{3+v} \right)}{\varepsilon}$ for $v$ less than 10 m/s
Where, $s_0 =$ endurance strength from (High strength, low carbon steel with $s_0 = 410$ MPa)

$$S_{\text{allowable}} = 410 \times 10^6 \times \left( \frac{3}{3+v} \right) = 318.076 \text{ MN/m}^2$$

Hence, \[ \frac{1}{m^2} = \left( \frac{318.076 \times 10^6 \times 410}{3997.33} \right) = 2326.55 \times 10^3 \] (Allowable)

For, $N_p = 16$ teeth and assuming, $y \approx 0.1$ (Table 4.1, for 20° Stub Involute)

Table 2: Form Factor ($y$), for use in Lewis strength equation

<table>
<thead>
<tr>
<th>Number of Teeth</th>
<th>14$^0$ Full-Depth Involute or Composite</th>
<th>20$^0$ Full-Depth Involute</th>
<th>20$^0$ Stub Involute</th>
</tr>
</thead>
<tbody>
<tr>
<td>14</td>
<td>0.075</td>
<td>0.083</td>
<td>0.108</td>
</tr>
<tr>
<td>15</td>
<td>0.078</td>
<td>0.088</td>
<td>0.111</td>
</tr>
<tr>
<td>16</td>
<td>0.081</td>
<td>0.092</td>
<td>0.115</td>
</tr>
<tr>
<td>17</td>
<td>0.084</td>
<td>0.094</td>
<td>0.117</td>
</tr>
</tbody>
</table>

(Source: Khurmi, 2005)

Design is satisfactory since, $S_{\text{induce}} \ll S_{\text{allowable}}$

Hence, reducing $K$ from $K = 4$, to $k= 4 \times \left( \frac{2326.55 \times 10^3}{3997.33} \right) = 3.87$

Therefore, Face width (b),

$\begin{align*}
& b = \text{m} \times \pi \times 2 \\
& b = 24.3 \text{ mm}
\end{align*}$

i.e. the minimum width that can be used is 24.3 mm but, to increase the life span, reliability and efficiency 25 mm face width will be used.

Checking the tentative design from the standpoint of Strength, Wear and Dynamic Load:

**Allowable Endurance Load, $F_0$**

$$F_0 = s_0 b \pi \varepsilon \times \varepsilon \text{ m}^2 \text{ y}$$

where,

$\varepsilon = \frac{m}{m}$, $s_0 =$ average stress concentration

$$F_0 = 4 \times 10^4 \times \pi \times (0.002)^2 \times 0.111, \ F_0 = 7186.65 \text{ N}$$

**Dynamic Load, $F_d$**

$$F_d = \frac{4N (\varepsilon) + F}{21V (\varepsilon) + F}$$

For $V = 0.867 \text{ m/s}$, an error of 0.05 mm could be tolerated from a noise standpoint.

$$C = 570 \text{ KN/m}$$

$$F_d = \frac{21 \times 0.867 (0.025 \times 570 + 3997.33)}{21 \times 0.867 + \sqrt{(0.025 \times 570 + 3997.33)}} = 6470.97 \text{ N}$$

**Wear Load, $F_w$**

$$F_w = D_p Q b K$$

But, $Q = 2N_p / (N_p + N_b)$

$N_b = \frac{12}{b} = \frac{45}{12} \text{ mm}$

Hence, $Q = \frac{2 \times 45}{30 + 45} = 1.5$

$$K = \left( \frac{2}{S_{\text{max}} \times \text{Sin} \phi} \left( \frac{1}{E_p} + \frac{1}{E_g} \right) \right) / 1.4$$

Assuming a Brinell Hardness Number (BHN) of 250, its corresponding, $S_{\text{max}} = 618 \text{ MN/m}^2$ (Khurmi, 2005).

Where,

$E_p = \text{modulus of elasticity of the pinion material, N/m}^2$

$E_g = \text{modulus of elasticity of the gear material, N/m}^2$

$\phi =$ pressure angle.

$$F_w = (0.03 \times 0.025 \times 1.5 \times 0.88717291 \times 10^6) = 9980.06 \text{ N}$$

Hence, $F_w = (0.03 \times 0.025 \times 1.5 \times 0.88717291 \times 10^6) = 9980.06 \text{ N}$

Design is Satisfactory since, $F_0, F_w > F_d$

**Force and Stress Analysis**

The force analysis consideration is based on the assumption that the screw jack is loaded vertically symmetrical.

Maximum Mass ($m$) = 1000 kg; $F = (1000 \times 9.81) = 9810 \text{ N}$

**Fig. 11. Force and Stress Analysis in Screw Jack members**

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Resolving vertical forces, \( f_y = f_{oy} \)  
And since, \( X_1 = X_2 \), hence, \( f_y = f_{oy} \) 

At maximum raising height of the jack,  
\[
\sigma_{a1} = \frac{f_{ax}}{\cos \theta} \\
\sigma_{a1} = \frac{4905}{\cos 28^\circ} = 5.555256 \text{ KN} \\
f_{a1} = 5.555256 \text{ KN}  
\]

Hence, \( f_{a1} = f_{b1} \).  

Also,  
\[
f_f = f_{a1} X (\sin 28^\circ) + f_{a2} X (\sin 28^\circ)  
\]
\[
= 2 f_{a1} X (\sin 28^\circ) + [2555.256 X (\sin 28^\circ)] 
\]
\[
f_f = f_{b1} = 5216 \text{ N} 
\]

At minimum raising height of the jack,  
\[
\theta = 55.77^\circ  
\]
\[
\sigma_{a1} = \frac{f_{ax}}{\cos \theta} \\
\sigma_{a1} = \frac{4905}{\cos 55.77^\circ} = 8.720 \text{ KN} \\
f_{a1} = f_{a1} X (\sin 55.77^\circ) + f_{a2} X (\sin 55.77^\circ) \\
f_f = f_{b1} = 14419.15 \text{ N} \quad \text{(Tensile force in the Power Screw)}  
\]

Since the maximum loading force will act at the minimum raising height of the jack, the design stresses will be analysed at that point.

**Design of the Power Screw**

The design stress,  
\[
\sigma_d = \frac{\text{Tensile Strength}}{\text{Factor of Safety (FS)}}  
\]

For 35% carbon steel, the tensile strength is 485 MN/m², with yield strength of 345 MN/m². Factor of Safety of 2.5 is chosen, because the material is known under reasonably constant service conditions subjected to loads and stresses that can be determined easily.

Design stress,  
\[
\sigma_d = \frac{485}{2.5} = 242.5 \text{ MN/m}^2  
\]

Applying the design equation,  
\[
\sigma_d = \sigma_{acutal} \quad \Rightarrow \quad \sigma_{acutal} = \frac{F}{A} \\
\sigma_d = \sigma_{acutal} \\
A \geq \frac{F}{\sigma_d} \\
A \geq \frac{14419.15}{(242.5 \times 10^9)} \\
A = 0.05946 \times 10^{-8} \text{ m}^2 \\
A \geq \frac{\pi d^2}{4} \\
d \geq \sqrt{\frac{4A}{\pi}} \\
d \geq 0.0097280 \text{ m}, \quad d \geq 9.7280 \text{ mm}  
\]

For the sake of design convenience, the diameter of the power screw, \( d \) is chosen to be 13 mm, with a pitch of 2.54 mm (Budynas and Nisbett, 2008).

Combined Tensile and Torsional Stress in the Power Screw

Assuming a frictionless collar  
\[
\begin{align*}
\text{Torsional Stress} & = \frac{181}{d^2} \\
& = \frac{181}{2^2} = 45.25 \text{ N/m²} \\
\text{Tensile stress} & = \sigma_x = \frac{4 \times 141419.15}{u(0.018)^2} \\
& = 563.03 \text{ N/m²} \\
\end{align*}
\]

Hence, the combined Tensile and Torsional stress is;  
\[
\sigma^1 = \sqrt{\sigma_x^2 - \sigma_y^2 + \sigma_z^2 + 27\sigma_y^2}  \\
& = \sqrt{(108.63 \times 10^6)^2 - 0 + 0 + 3(563.03)^2}  \\
& = 108.63 \text{ MPa}  
\]

Now,  
\[
\text{Yield Strength} \quad \text{Factor of Safety} \quad \Rightarrow \quad \sigma^1  \\
\frac{345 \times 10^6}{2.5} = 138 \text{ MPa}  
\]

The power screw will not fail under yielding since,  
\[
\frac{\text{Yield Strength}}{\text{Factor of Safety}} > \sigma^1  
\]

Design of Lifting Members

Force in a lifting member, \( F = 8720 \text{ N} \)  

Design stress,  
\[
\sigma_d = \frac{\text{Yield Strength}}{\text{Factor of Safety}}  \\
\sigma_d = \frac{345 \times 10^6}{2.5} = 138 \text{ MPa}  \\
\]

Design stress,  
\[
A \geq F/\sigma_d = \frac{8720}{138 \times 10^3} = 65.166 \times 10^{-3} \text{ mm}^2  
\]

For design consideration an area of 70 mm² will be chosen.
Tensile stress, $\sigma_n \geq \frac{F_n}{A_n} = \frac{8720}{(70 \times 10^2)} = 124.57 \text{ MPa}$

From the maximum distortion energy theorem;
where $\sigma_y = 0$ and $\tau_{xy} = 0$ (Budynas and Nisbett, 2008).

$$\sigma^2 = \sigma_x^2 + \sigma_y^2 - \sigma_x \sigma_y + 3\tau_{xy}^2$$

$$\sigma^2 = \sqrt{(124.57)^2 + 0 - 0 + 0}$$

The lifting members will not fail due to yielding since, $\frac{\text{Field strength}}{\text{Factor of Safety}} > \sigma^1$

RESULTS AND DISCUSSION
The design was focused on all the processes of conception, invention, visualisation, calculation, refinement and specification of details that determine the form of the product. Hence, the said Motorised Screw Jack for Vehicles, specifically the Scissors type has gone under force analysis so that its performance criterion will not fail in any sense. The main physical parameters of the design are determined through the appropriate calculations and practical considerations with reasonable assumptions. Figures 1 and 2 show the Pictorial view and the Orthographic view of the design while figures 3 to 10 show the components. From the force and stress diagram in figure 11, it was discovered that at the maximum raising height of 28° of the horizontal Tensile force in the opposite direction are the same. It is also the same for the minimum raising height of 55.7°. Mild steel is used as the materials for both the keys and gears due to its high strength, toughness, tooth hardness and its economical effects. The compressive spring is used to reduce noise and vibration effect from the motor. The speed at which the designed power screw will run is 92 rpm, which means the gear will complete one and a half revolution in one second or complete one revolution in 0.66 second which is considered safe and reasonable.

CONCLUSION
The existing design was modified by introduction of an electric motor in the power screw, connecting gear with the pinion, the cigarette lighter receptacle connected to the motor and plugged to the automobile 12V battery source to generate power for the prime mover (motor), in order to make load lifting easier. In this modified design, the power screw is rotated through its gear when electrical power flows through it. The main advantages of the modified design over the existing design are that the modified designed motorised jack will save time, be faster and easier to operate and requires less human energy and additional work to operate. There by effectively curb the problems associated with Ergonomics - which is a fundamental concept of design process. The limitation of this design is that it is only applicable to vehicles not weighing above 1000 kg.

RECOMMENDATION
- Further research should be carried out on how to minimise vibration and noise during operation.
- Design applicable to vehicles weighing over 1000 kg should be carried out

REFERENCES


