Highly Efficient Motorized Screw Jack

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ABSTRACT:
Power screws are used to convert rotary motion into translatory motion. A screw jack is an example of a power screw in which a small force applied in a horizontal plane is used to raise or lower a large load. The principle on which it works is similar to that of an inclined plane. The mechanical advantage of a screw jack is the ratio of the load applied to the effort applied. The screw jack is operated by turning a lead screw. The effort required to rotate the screw can be eliminated by using a 12V DC motor to rotated screw of jack; which facilitate in easy replacement of tyre. Advantage of this system is that it draws the energy from the battery of vehicle. For torque multiplication; generated by motor two spur gear are used. A small gear is mounted on motor shaft and a large spur gear on power screw of jack. Also we are looking for to increase the efficiency of motorized screw jack by varying helix angle by which energy drawn by motor can be decrease.

Keywords: D.C motor, Power screw, Lifting Arm, Gear & pinion, speed reduction.

I. INTRODUCTION
The motorized screw jack has been developed to cater to the needs of small and medium automobile garages, which are normally man powered with minimum skilled labor. In most of the garages the vehicles are lifted by using screw jack. This needs high man power and skilled labour. In order to avoid all such disadvantages, the motorized jack has been designed in such a way that it can be used to lift the vehicle very smoothly without any impact force. The operation is made simple so that even unskilled labour can use it with ease. The d.c motor is coupled with the screw jack by gear arrangement. The screw jack shafts rotation depends upon the rotation of D.C motor. This is a simple type of automation project. The operation remains to be an essential part of the system although with changing demands on physical input, the degree of mechanization is increased.

II. PRINCIPLE OF WORKING

2.1 Torque Requirement- Lifting Load
The screw is considered as an inclined plane with inclination α. When the load is being raised, following forces act at a point on this inclined plane:

2.1.1 Load W: It always acts in vertically downward direction.

2.1.2 Normal reaction N: It acts perpendicular (normal) to the inclined plane.

2.1.3 Frictional force μN: Frictional force acts opposite to the motion. Since the load is moving up the inclined plane, frictional force acts along the inclined plane in downward direction.[2] 

\[ W = N \cos \alpha - \mu N \sin \alpha \]  

For an equilibrium of horizontal forces.[2]

\[ P = \mu N \cos \alpha + N \sin \alpha \]  

(a)

(b)

Fig. 2.1

2.1.4 Effort P: The effort P acts in a direction perpendicular to the load W. It may act towards right to overcome the friction and raise the load.

For an equilibrium of horizontal forces.[2]
Dividing expression (a) by (b),
\[ P = \frac{W(\mu \cos \alpha + \sin \alpha)}{(\cos \mu - \sin \alpha)} \]  \[ (5) \] (c)
The coefficient of friction \( \mu \) is expressed as,
\[ \mu = \tan \theta \]  \[ (d) \]
Where \( \theta \) is the friction angle.
Substituting \( \mu = \tan \theta \) in Eq. (c),
\[ P = \frac{W(\tan \theta - \tan \alpha)}{(1 - \tan \theta \tan \alpha)} \]  \[ (5) \]
Or
\[ P = \frac{W \tan (\theta + \alpha)}{2} \]  \[ (e) \]
The torque \( T \) required to raise the load is given by,
\[ T = \frac{2P\sin \theta}{2} \]  \[ (f) \]

### 2.2 Torque Requirement Lowering Load

When the load is being lowered, the following forces act at a point on the inclined plane:

#### 2.2.1 Load \( W \): It always acts in vertically downward direction.

#### 2.2.2 Normal reaction \( N \): It acts perpendicular (normal) to the inclined plane.

#### 2.2.3 Frictional force \( \mu N \): Frictional force acts opposite to the motion. Since the load is moving down the inclined plane, frictional force acts along the inclined plane in upward direction.

![Fig. 2.2](image)

#### 2.2.4 Effort \( P \): The effort \( P \) acts in a direction perpendicular to the load \( W \). It should act towards left to overcome the friction and lower the load.

For an equilibrium of horizontal forces,
\[ P = \mu N \cos \alpha - N \sin \alpha \]  \[ (a) \]

For an equilibrium of vertical forces,
\[ W = N \cos \alpha + \mu N \sin \alpha \]  \[ (b) \]

Dividing expression (a) by (b),
\[ P = \frac{W(\mu \cos \alpha + \sin \alpha)}{(\cos \mu - \sin \alpha)} \]  \[ (1) \]

Dividing the numerator and denominator of the right hand side by \( \cos \alpha \),
\[ P = \frac{W(\mu - \tan \alpha)}{(1 - \mu \tan \alpha)} \]  \[ (c) \]
The coefficient of friction \( \mu \) is expressed as,
\[ \mu = \tan \theta \]  \[ (d) \]

Substituting \( \mu = \tan \theta \) in Eq. (c),
\[ P = \frac{W(\tan \theta - \tan \alpha)}{(1 - \tan \theta \tan \alpha)} \]  \[ (e) \]

The torque \( T \) required to raise the load is given by,
\[ T = \frac{P\sin \theta}{2} \]

### III. SYSTEM DESIGN

#### 3.1 Measurement of Angle –
\[ 25 + 160 \sin \theta + 160 \sin \theta + 25 = 210 \]
\[ \sin \theta = \frac{1}{2} \]

Or \( \theta = 30^\circ \) Where, 25mm is height of base and upper support. 160mm is length of arm. 210mm is ground clearance of vehicle. \( \theta \) is inclination of arm at minimum position.
3.2 Design of screw –

\[ \sum F_y = 0 \]

\[ P \cos 60^\circ + P \cos 60^\circ = 4900 \]

\[ P = 4900 / 2 \cos 60^\circ = 4900 \text{ N} \]

From figure (3.3)

\[ P \sin 60^\circ = F \]

\[ F = 4900 \sin 60^\circ = 4900 \times 0.5 \]

\[ F = 2454.52 \text{ N} \]

This is the tension in screw due to one nut. Therefore tension in screw due to both nut.

\[ W_1 = 2F = 2 \times 2454.52 = 4909.04 \text{ N} \]

Let material for screw is C50 for which

\[ \sigma_y = 392 \text{ N/mm}^2, \tau = 147 \text{ N/mm}^2 \] \[ \text{[2]} \]

Design stress \( S_d \)

\[ S_d = \frac{\sigma_y}{n} = \frac{392}{2.5} = 156.8 \text{ N/mm}^2 \]

F.s. \( = n=2.5, \sigma_{y2}=1.5, \tau=1.6 \]

Now

Total tension = \( \frac{\pi}{4} (d_c)^2 \sigma_t \) \[ \text{[2]} \]

\( \bullet \) Screw is designed for design stress \( S_d \)

\( \therefore \) Put \( \sigma_t = S_d \)

\[ W_i = \frac{\pi}{4} (d_c)^2 S_d \] or 8487.048 = \( \frac{\pi}{4} (d_c)^2 \times 98 \]

\[ d_c = 10.5007 \text{ mm} \quad \text{Say} d_c = 12 \text{ mm standard size.} \]

Area of core = \( \frac{\pi}{4} (d_i)^2 = \frac{\pi}{4} (12)^2 = 113 \text{ mm}^2 \) \[ \text{[5]} \]

Corresponding to this area nominal dia can be selected from [5]

Nominal dia or outer dia

\[ d_o = 14 \text{ mm} \]

\[ d_{mean} = \frac{d_o+d_i}{2} = \frac{12+14}{2} = 13 \text{ mm} \]

\[ d = 13 \text{ mm and Pitch = 3 mm By Survey} \]

\[ \tan \alpha = \frac{P}{\pi d} \quad \text{or} \quad \tan \alpha = \frac{\tau}{\pi d} = 0.073 \]

Effort required to rotate screw

\[ P = W_i \tan (\alpha + \phi) \] \[ \text{[5]} \]

\[ W_i = \frac{\tan \alpha - \tan \phi}{1 - \tan \alpha \tan \phi}, \tan \phi = 0.18 \] \[ \text{[1]} \]

According to max shear stress theory \[ \text{[3]} \]

Max shear stress in simple tension at elastic limit

\[ = \frac{1}{2} \sigma_t = \frac{1}{2} \times 392 = 196 \text{ N/mm}^2 \]

Shear stress due to torque = 59.62 N/mm²

Calculated value is less then design Stress . Hence design is Safe

Length of screw = 210+ 2*14 = 238 mm
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\[ T_o = P * \frac{d}{2} = 2175.81 \times \frac{15}{2} = 14142.78 \text{ Nmm} \quad \text{or} \quad T_o = 14.142 \text{ Nm} \]

\[ \eta_{max} = \frac{1}{1 + \tan \phi} \]

\[ \tan \phi = 0.18 \quad \text{or} \quad \phi = 10.2^\circ \]

\[ \eta = \frac{1}{1 + \tan \phi} = \frac{1}{1 + 10.2} = 0.098 \quad \text{or} \quad \eta = 69.9\% \]

\[ T = 20.2313 \text{ Nm} \]

Shear stress in screw due to torque

\[ \tau = \frac{16T}{\pi (d_c)^3} = \frac{20.2313 \times 16}{\pi (12)^3} \]

\[ \tau = 59.628 \text{ N/mm}^2 \]

We know that

\[ \sigma_x = \frac{9T}{4(d_c)^2} = \frac{8487.048}{4(12)^2} \]

\[ \sigma_x = 75.04 \text{ N/mm}^2 \]

Max principal stress \[ \sigma_{(max)} = \frac{5}{2} + \frac{1}{2} \sqrt{\sigma_x^2 + 4\tau^2} \]

\[ \sigma_{(max)} = \frac{5}{2} + \frac{1}{2} \sqrt{(75.04)^2 + 4 \times (59.628)^2} \]

\[ \sigma_{(max)} = 100.264 \text{ N/mm}^2 \]

For material 392 N/mm²

\[ \text{FOS} = \frac{1100}{100.26} = 3.9 \]

Check Bending stress

\[ I = \frac{1}{12} \pi d_c^4 \left( \frac{d_c}{2} \right)^2 \] and \[ S_y = \frac{P}{4} \]

\[ \sigma_b = \frac{M}{I} = \frac{P}{y} \quad \text{or} \quad \sigma_b = \frac{1}{r^2} \frac{1}{4} \left( \frac{d_c}{2} \right)^3 \]

\[ \sigma_b = \frac{548.7468}{106018} \frac{1}{4} \quad \text{or} \quad \sigma_b = 60.034 \text{ N/mm}^2 \]

Which is less than design stress.

3.3 Design of nut-

Bearing pressure in screw \[ p_b = 14 \text{ N/mm}^2 \]

No. of Thread \[ i = \frac{V}{P} \] or \[ i = \frac{8487.048}{4900} = 14.843 \approx 15 \]

Length of nut \[ = i^* \text{P} = 15^*3 = 45 \text{ mm} \]

outside dia of nut \[ = 2d_c = 2^*12 = 24 \text{ mm} \]

from figure –

\[ \sum F_x = P \sin 30 - P \sin 30 = 0 \]

\[ \sum F_y = 4900 \cos 30 + P \cos 30 = 8487.048 \text{ N} \]
R = \sqrt{F_x^2 + F_y^2} = \sqrt{8487.048^2}
= 8487.048 N

Check for shear
\[ \tau = 1^* \pi \frac{d_1^2}{6} * \frac{F_y}{2} * \frac{\sigma_s}{2} = 15^* \pi * 12^* \frac{2}{2} * 147 = 124.689 * 10^3 N \]
Or
= 124.689 KN
Since calculated value is greater than actual load on nut (8487.048 N)
Hence design is safe.

3.4 Design of pin –

\[ P = \frac{4900N}{2} = 2450 \]
\[ \sum F_y = P \cos 60^\circ - 2450 \]
\[ \sum F_x = P \sin 60^\circ = 0 \]
\[ R = \sqrt{0^2 + 4243.52^2} = 4243.52 N \]
\[ \text{Let dia of pin} = d_1 \]
\[ 4243.52 = 2 * \frac{\pi}{4} * d_1^2 * \tau \]
\[ \text{or} \]
\[ \frac{4243.52 \times \tau}{\pi} = d_1^2 * 147 \]
\[ \text{or} \]
\[ d_1 = 4.2869 \text{ mm} \]
\[ \text{Dia of pin head} = 1.5 \times d_1 \]
\[ = 1.5 \times 4.286 \]
\[ = 6.429 \text{ mm} \]

3.5 Design of links:
Material for link is plain carbon steel,C=1.00 to 1.15
\[ \sigma_y = \frac{563}{2.5} = 160.857 \]
Load o links = F/2
\[ = \frac{563}{2} = 2121.76 \]
Assuming a factor of safety = 3.5 the link must be designed for a buckling load,
\[ W_{cr} = 2121.76 \times 3.5 = 7426.16N \]
\[ \text{Let} \]
\[ t_1 = \text{thickness of the link and} \]
\[ b_1 = \text{width of the link} \]
Assuming the width of the link is three times the thickness of the links i.e. b=3t_1
\[ A = t_1^*3t_1 = 3t_1^2 \]
Moments of inertia \[ I = \frac{1}{12}*t_1^*b_1^2 = \frac{1}{12}*t_1^*(3t_1) = 2.25t_1^4 \]
Since for buckling of the link in the vertical plane, the ends are considered as hinged, therefore,
Equivalent length of link, L = 160
Rankine constant \[ K = 1/7500 \]
\[ W_{cr} = \sigma_y^2A^2(1+a(L/K)^2) \]
\[ 7426.16 = \frac{160.857^2 \times 3t_1^2}{1 + \frac{12^2}{160^2} \times \frac{160}{7500}} \]
\[ 7426.16 + \frac{33799.26}{t_1^4} - 482.55 \times t_1^4 = 0 \]
\[ 482.55 \times t_1^4 - 7426.16 \times t_1^4 - 33799.26 = 0 \]
\[ t_1 = 19.063 \text{ mm} \]
\[ t_1 = 4.366 \text{ mm} \]
\[ A = t_1^*b_1 = t_1 \times 3t_1 = 3t_1^2 \]
\[ K = \frac{I}{A} = \frac{2.25t_1^4}{3t_1^2} = 0.29t_1^2 \]
Equivalent length L=l/2=160/2=80mm
Since calculated value is greater than the actual value, so design is safe.

### 3.6 Design of Gear

The allowable static static stress for gear is made of cost iron and pinion of steel are 60MPa & 105 MPa respectively.

\[ \phi = 20 \quad \text{involute} \quad T_P = 16 \]

\[ \text{V.R.} = \frac{T_G}{T_P} = 3:1 \]

\[ \sigma_{G0} = 60 \text{ MPa} \quad \sigma_{OP} = 105 \text{ MPa} \]

Pitch line velocity

\[ V = \frac{\pi D_P N_P}{60} = \frac{\pi D_G N_G}{60} \]

\[ = \frac{\pi \times 16 \times 100}{60} = 251 \text{ mm/s} \]

Service factor \( C_s = 0.8 \)

\[ W_T = \frac{P}{V} + C_s = \frac{0.34 \times 10^3}{0.251} + 0.8 \]

Velocity factor \( C_v = \frac{Y_P}{4.5 + 0.251} \)

\[ = \frac{0.34 \times 10^3}{0.251} + 0.097 \]

\[ \sigma_{G0} \times Y_P = 105 \times 0.097 = 10.185 \]

\[ \sigma_{OP} \times Y_G = 60 \times 0.135 = 8.1 \]

Since \( \sigma_{G0} \times Y_G < \sigma_{OP} \times Y_P \)

Design tangential tooth load

\[ \frac{W_T}{9.212 \times 10^3}{m} = 60 \times \frac{0.34 \times 10^3}{0.251} \times 14 \times \pi \times 0.135 \]

\[ = \frac{4.868 \times 0.0663}{4.5 + 0.251} \]

\[ = 0.00737 \text{ m}^3 \]

\[ m = 4.23 \text{ By hit & trial method} \]

Face width \( b = 14 \times 4.23 = 59.22 \text{ mm} \)

Pitch diameter of pinion \( D_p = m \times T_p = 4.23 \times 16 = 67.68 \text{ mm} \)

Pitch dia of gear \( D_G = m \times T_G = 203 \text{ mm} \)

check the gears for wear

we know that the ratio factor

\[ Q = \frac{2V_R}{R+1} = \frac{2}{3} \]

load stress factor

\[ K = \frac{(\sigma_{G0})^2 \sin \phi}{1.4} \left( \frac{1}{E_P} + \frac{1}{E_G} \right) \]

\[ = \frac{(600)^2 \sin 20}{14} \left( \frac{1}{100 \times 10^3} + \frac{1}{100 \times 10^3} \right) \]

\[ = 0.44 + 0.88 = 1.32 \text{N/mm}^2 \]

We know that the maximum or limiting load for wear

\[ W_w = D_p b K \]

\[ = 67 \times 59.22 \times 1.5 \times 1.32 = 7856.125 \text{N} \]
\[ W_T = \frac{9.312 \times 10^3}{m} \times \frac{785.1252}{4.23} = 1857.24 \text{ N} \]

Since the maximum wear load is much more than tangential load on the tooth, therefore the design is satisfactory from the standpoint of wear.

IV. CONCLUSION

Screw Jacks are the ideal product to push, pull, lift, lower and position loads of anything from a couple of kg to hundreds of tones. The need has long existed for an improved portable jack for automotive vehicles. It is highly desirable that a jack become available that can be operated alternatively from inside the vehicle or from a location of safety off the road on which the vehicle is located. Such a jack should desirably be light enough and be compact enough so that it can be stored in an automobile trunk, can be lifted up and carried by most adults to its position of use, and yet be capable of lifting a wheel of a 400-500 kg vehicle off the ground. Further, it should be stable and easily controllable by a switch so that jacking can be done from a position of safety. It should be easily movable either to a position underneath the axle of the vehicle or some other reinforced support surface designed to be engaged by a jack.

Thus, the product has been developed considering all the above requirements. This particular design of the motorized screw jack will prove to be beneficial in lifting and lowering of loads.

REFERENCES

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