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]

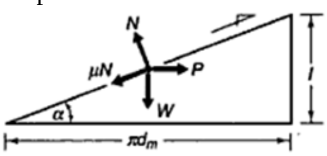

Fig. 2.1

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]

$$P = \mu N \cos \alpha + N \sin \alpha \qquad \text{(a)}$$

For an equilibrium of vertical forces,

$$W = N \cos \alpha - \mu N \sin \alpha \qquad \text{(b)}$$

Dividing expression (a) by (b),

$$P = \frac{W(\mu\cos\alpha + \sin\alpha)}{(\cos\alpha - \mu\sin\alpha)} \quad [5] \qquad (c)$$

The coefficient of friction μ is expressed as,

$$\mu = \tan\theta \qquad (d)$$

Where θ is the friction angle.

Substituting μ = tan θ in Eq. (c),

$$P = \frac{W(\tan\theta + \tan\alpha)}{(1 - \tan\theta\tan\alpha)} \quad [5]$$

Or $P = W\tan(\theta + \alpha)$ (e)

The torque"T" required to raise the load is given by,

$$T = \frac{Pd_m}{2}$$

$$T = \frac{Wd_m}{2}\tan(\theta + \alpha) \quad [5] \qquad (f)$$

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 μN: 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.

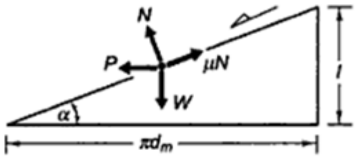

Fig. 2.2

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,

$$P = \mu N\cos\alpha - N\sin\alpha \qquad (a)$$

For an equilibrium of vertical forces,

$$W = N\cos\alpha + \mu N\sin\alpha \qquad (b)$$

Dividing expression (a) by (b),

$$P = \frac{W(\mu\cos\alpha + \sin\alpha)}{(\cos\alpha - \mu\sin\alpha)} \quad [1]$$

Dividing the numerator and denominator of the right hand side by cos α,

$$P = \frac{W(\mu - \tan\alpha)}{(1 + \mu\tan\alpha)} [1] \qquad (c)$$

The coefficient of friction μ is expressed as,

$$\mu = \tan\theta \qquad (d)$$

Substituting μ = tan θ in Eq. (c),

$$P = \frac{W(\tan\theta - \tan\alpha)}{(1 + \tan\theta\tan\alpha)}$$

or $P = W\tan(\theta - \alpha)$ (e)

The torque „T" required to raise the load is given by,

$$T = \frac{Pd_m}{2}$$

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.
θ is inclination of arm at minimum position.

3.2 Design of screw –

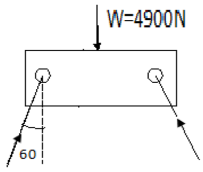

Fig. 3.1.

$\sum F_y$=0

Pcos 60 °+ Pcos 60 ° =4900

P=4900/2cos 60 =4900 N

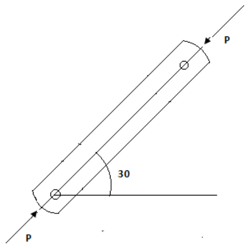

Fig. 3.2

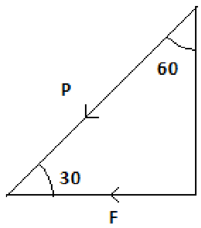

Fig. 3.3

From figure (3.3)

Psin 60=F

F=4900*sin 60 =4900*0.5

F=4243.52N

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

W_1 = 2F = 2*4243.52 =8487.048N

Let material for screw is C50 for which [5]

σ_{yp} =392 N/mm^2 , τ =147 N/mm^2 [2]

Design stress $S_d = \frac{S_e}{nk} = \frac{e_{sz}S_e}{nk} = \frac{392}{2.5*1.6}$ [2]

F.s. ⇒n=2.5 ,e_{s2}=1.5, K=1.6

Now

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

∵ Screw is designed for design stress S_d

∴ Put $\sigma_t = S_d$

$W_1 = \frac{\pi}{4}(d_c)^2 S_d$ or 8487.048 = $\frac{\pi}{4}(d_c)^2 * 98$

d_c =10.5007 mm Say d_c=12mm standard size.

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

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

Nominal dia or outer dia

d_o =14 mm

$d_{mean} = \frac{d_c+d_o}{2} = \frac{12+14}{2}$ = 13mm

d =13mm and Pitch = 3mm By Survey

$\tan\alpha = \frac{p}{\pi d}$ or $\tan\alpha = \frac{3}{\pi*13}$ 0.073

Effort required to rotate screw

P = $W_1 \tan(\alpha+\varphi)$ [5]

$W_1 = \frac{\tan\alpha+\tan\varphi}{1-\tan\alpha\tan\varphi}$, $\tan\varphi$=0.18 [1]

According to max shear stress theory [3]

Max shear stress in simple tension at elastic limit

$= \frac{1}{2}\sigma_t = \frac{1}{2}*392$ = 196 N/mm^2

Shear stress due to torque = 59.62 N/mm^2

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

Length olf screw = 210+ 2*14 =238 mm

$= 8487.048 * \frac{0.073+0.18}{1-0.073*0.18}$ $= 8487.048 * 0.258$ $= 2175.81$ N

Torque $T_o = P * \frac{d}{2}$

$= 2175.81 * \frac{13}{2} = 14142.78$ Nmm or $T_o = 14.142$ Nm

$\eta_{max} = \frac{1-\sin\phi}{1+\sin\phi}$ [2]

$\because \tan\phi = 0.18$ or $\phi = 10.2$

$\eta = \frac{1-\sin 10.2}{1+\sin 10.2}$ or $\eta = 69.9$ %

$\eta = \frac{T_o}{T}$ or $T = \frac{14.142}{0.699}$

T = 20.2313Nm

Shear stress in screw due to torque

$= \frac{16T}{\pi(d_c)^3}$ $= \frac{20231.34*16}{\pi(12)^3}$

$\tau = 59.628$ N/mm^2

We know that

$\sigma_t = \frac{W_1}{\frac{\pi}{4}(d_c)^2} = \frac{8487.048}{\frac{\pi}{4}(12)^2}$ [5]

$\sigma_t = 75.04$ N/mm^2

Max principal stress $\sigma_{t(max)} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{\sigma_t^2 + 4\tau^2}$

$\sigma_{t(max)} = \frac{59.628}{2} + \frac{1}{2}\sqrt{(75.04)^2 + 4*(59.628)^2}$

$= \frac{59.628}{2} + \frac{140.90}{2}$

$\sigma_{t(max)} = 100.264$ N/mm^2 For material 392 N/mm^2

FOS $= \frac{392}{100.26} = 3.9$

Check Bending stress –

$I = \frac{1}{12}\pi d_c * (\frac{p}{2})^3$ and $Sy = \frac{p}{4}$

$\frac{M}{I} = \frac{\sigma_b}{y}$ or $\sigma_b = \frac{M}{I} * y$

$\sigma_b = \frac{\frac{W_1 l}{i}}{\frac{1}{12}\pi d_c (\frac{p}{2})^3} * \frac{p}{4}$ or $\sigma_b = \frac{8487.048*\frac{1.5}{15}}{\frac{1}{12}\pi d_c (\frac{3}{2})^3} * \frac{3}{4}$

$\sigma_b = \frac{848.7048}{10.6028} * \frac{3}{4}$ or $\sigma_b = 60.034$ N/mm^2

Which is less than design stress.

3.3 Design of nut-

Bearing pressure in screw $p_b = 14$ N/mm^2 Table 19.1 Page 450 M& H

No. of Thread $i = \frac{W_1}{\frac{\pi}{4}*[d_o^2 - d_c^2]*p_b}$ or $i = \frac{8487.048*4}{\pi*[14^2-12^2]*14} = 14.843 \approx 15$

Length of nut = i* p = 15*3 = 45 mm [2]

outside dia of nut = $2d_c$ =2*12= 24mm [2]

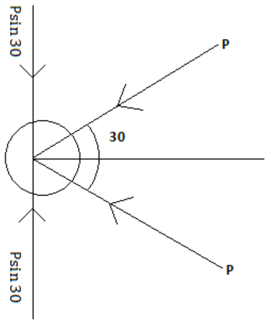

Fig. 5.4

from figure –

$\sum F_y = P\sin 30 - P\sin 30 = 0$

$\sum F_x = P\cos 30 + P \cos 30$

$= 4900*\cos 30 + 4900*\cos 30$ = 8487.048 N

$$R = \sqrt{F_x{}^2 + F_y{}^2} = \sqrt{8487.048^2}$$

=8487.048 N

Check for shear

τ = i* π dc * $(\frac{p}{2})$ * σ_s = 15* π * 12* $\frac{3}{2}$ * 147 = 124.689 * 10^3 N

Or = 124.689 KN

Since calculated value is greater than actual load on nut (8487.048 N)

Hance design is safe.

3.4 Design of pin –

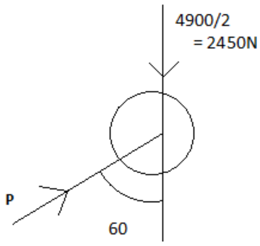

Fig. 5.5

P = 4900N

$\sum F_y$ = P $\cos 60 - 2450$

= 4900*0.5 -2450 =0

$\sum F_x$ = P $\sin 60$

= 4900 $\sin 60$ = 4243.52N

R = $\sqrt{0^2 + 4243.52^2}$ = 4243.52 N

Let dia of pin = d_1

4243.52 = 2 * $\frac{\pi}{4}$ $d_1{}^2 * \tau$ or $\frac{4243.52*2}{\pi}$ = $d_1{}^2$ * 147 or d_1 = 4.2869 mm

Dia of pin head = 1.5 d_1

= 1.5* 4.286 = 6.429 mm

3.5 Design of links:

Material for link is plain carbon steel,C=1.00 to 1.15

$\sigma_{yp=563MPa}$

Now σ_c= $\frac{\sigma_{yp}}{F.S.}$ = $\frac{563}{3.5}$ =160.857

Load o links = F/2

=563/2 =2121.76

Assuming a factor of safety = 3.5 the link must be designed for a buckling load,

W_{cr} = 2121.76*3.5 =7426.16N

Let $t_{1=}$ thickness of the link and

$b_{1=}$ 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

Moment of inertia I = $\frac{1}{12}$ *t_1 *b_1^3 = $\frac{1}{12}$ *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 ,a=1/7500

$W_{cr=}\sigma_c$*A/1+a$(L/K)^2$

7426.16 = $\frac{160.857*3*t_1^2}{1+\frac{1}{7500}*(\frac{160}{0.866t_1})^2}$ or 7426.16 + $\frac{33799.26}{t_1^2}$ - 482.55* t_1^2 = 0

482.55 * t_1^4 - 7426.16 * t_1^2 - 33799.26 = 0

t_1 = 19.063mm t_1=4.366mm

I = $\frac{1}{12}$ * b_1 * $(t_1)^3$ = $\frac{1}{12}$ * 3t_1* t_1^3 = 0.25t_1^4

A = t_1*b_1= $t_1 * 3t_1$ = 3t_1^2

K = $\sqrt{\frac{I}{A}}$ = $\sqrt{\frac{0.25t_1^4}{3t_1^2}}$ = 0.29t_1

Equivalent length L=l/2=160/2=80mm

$W_{cr} = \frac{\sigma_c\ A}{1+a(L/K)^2} \quad = \frac{563*3*(4.366^2)}{1+\frac{1}{7500}*(\frac{80}{0.29*4.366})^2} \quad = \frac{32195.64}{1.53229} = 21011.453N$

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.

$\phi = 20$ involute $T_P = 16$

V.R = T_G / T_P =3:1

σ_{oG}= 60 MPa σ_{oP} = 105 MPa

Pitch line velocity

$$V = \frac{\pi D_p N_p}{60} = \frac{\pi m T_P N_p}{60}$$

$$= \frac{\pi m*16*300}{60} = 251m\ mm/s \quad = 0.251mm/s$$

Service factor $C_s = 0.8$

$$W_T = \frac{p}{v} * C_s = \frac{0.34*10^3}{0.251m} * 0.8 \quad = \frac{9.322*10^3}{m}\ N$$

Velocity factor $C_v = \frac{4.5}{4.5+V}$

$$= \frac{4.5}{4.5+0.251m}$$

$$Y_p = 0.154 - \frac{0.912}{T_p} = 0.154 - \frac{0.912}{16} = 0.097$$

$$Y_G = 0.154 - \frac{0.912}{T_G} = 0.154 - \frac{0.912}{3*16} = 0.135$$

$\sigma_{oP} * Y_P$ = 105* 0.097 =10.185

$\sigma_{oG} * Y_G$ = 60 *0.135 =8.1

Since $\sigma_{oG} * Y_G < \sigma_{oP} * Y_P$

Design tangential tooth load

$$W_T = \sigma_{WG}\ b\pi\ m\ Y_G = \sigma_{OG}\ C_V\ b\pi\ m\ Y_G$$

$$\frac{9.322*10^3}{m} = 60\ (\frac{4.5}{4.5+0.251m}) * 14m*\pi\ m*0.135$$

$$\frac{9.322*10^3}{m} = (\frac{4.5687.0663m^2}{4.5+0.251m})$$

$4.5 + 0.251m = 00737m^3$

m= 4.23 By hit & trial method

Face width

b = 14m = 14* 4.23 = 59.22mm

Pitch diameter of pinion

D_p = m T_p = 4.23*16 = 67.68mm

Pitch dia of gear

D_G = m T_G =203mm

check the gears for wear

we know that the ratio factor

$$Q = \frac{2V.R}{VR+1} = \frac{2*3}{3+1} = 1.5$$

load stress factor $K = \frac{(\sigma_{es})^2 \sin\phi}{1.4}(\frac{1}{E_P} + \frac{1}{E_G})$

$$= \frac{(600)^2 \sin 20}{1.4}(\frac{1}{200*10^3} + \frac{1}{200*10^3}) \quad = 0.44 + 0.88 = 1.32N/mm^2$$

We know that the maximum or limiting load for wear

$W_w = D_p bQK$ =67 *59.22*1.5*1.32 =7856.1252N

Tangential load on the tooth

$$W_T = \frac{9.322*10^3}{m} = \frac{7856.1252}{4.23} = 1857.24 \text{ N}$$

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.

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