Quick Lifiting Screw Jack Using Spur Gear Arrangement

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Abstract - With the increasing levels of technology, the efforts being put to produce any kind of work has been continuously decreasing. The efforts required in achieving the desired output can be effectively and economically be decreased by the implementation of better designs. 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 height of the jack is adjusted by turning a lead screw and this adjustment can be done either manually or by integrating an electric motor. In this project, an electric motor will be integrated with the screw jack and the electricity needed for the operation will be taken from the battery of the vehicle and thereby the mechanical advantage will be increased.

Keywords - spur gear, power screws, screw jack and translatory motion.

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1. INTRODUCTION

Screw type mechanical jacks were very commonly referred in jeeps and trucks at World War II vintage. For ex., the World War II jeeps (Ford GPW and Willys MB) were introduced with the Jack, Screw type, Capacity 1 1/2 ton, Ordnance part number 41-J-66. In that days, the 41-J-66 jack was carried in the jeep's tool box. Screw type jack's preferred continued for small capacity use due to minimum cost of production for raise or lower the load. It had negligible maintenance. The concept of using a screw as a machine was first demonstrated by Archimedes in 200BC with his device used for pumping water. There is also evidence that screws were preferred in the Ancient Roman world. But, In the late 1400s, the Leonardo da Vinci, who first displayed the method of use of a screw jack for lifting the loads. Its design used a threaded worm gear, supported on bearings, which is rotated by the turning of a worm shaft to drive a lifting screw to move the load instantly recognisable as the principle used today.

Thomas J. Prather (2009) in this, there was a introduction about vehicle lift system. a drive assembly was mechanically coupled to the piston. the drive assembly was operated in first direction to raise an upper end of the piston with respect to the housing. the drive assembly was operated in a second direction to lower the upper end of the piston with respect to the housing. the drive assembly was coupled to the power supply port which is removable to supply electrical power to the drive assembly. Farhad Razzaghi (2007) in this, electrically powered jack shown for normally raising and lowering of automobile from ground surface. the mechanism may be used in joining with a typical portable car jack, during which the mechanism constitute a power drill, a rod, and a numerous jack adapters. Manoj Patil (2014) in this general article, screw jack is to developed to overcome the human effort. it is actually difficult job to operate for pregnant women and old person. changing the tyre is not a pleasant experience. especially women can‟t apply more force to operate. for that, electric operated car jack is introduced lokhande tarachand (2012) this paper referred to optimise the efficiency of square threaded mechanical screw jack by varying different helix angle.

2. Design of a screw jack

2.1. Loads and stresses in screw

The load on the screw is the load which is to be lifted W, twisting moment M, between the screw threads and force F at the handle to rotate the screw. The load W is compressive in nature and induces the compressive stress in the screw. It may also lead the screw to buckle. The load F produces bending and it is maximum, when the screw is at its maximum lift.

The screw also experiences twisting moment due to F. The shear stress is also induced in the screw due to the twisting moment between the threads of screw and nut.

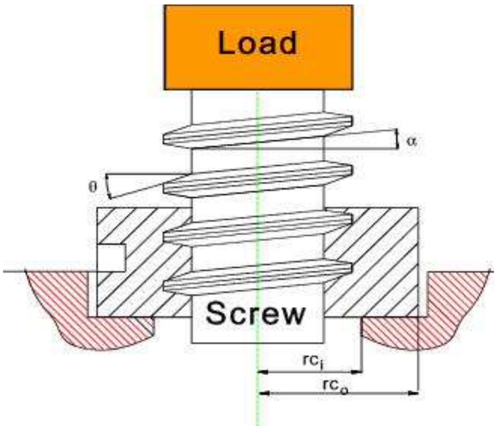

Figure 1 Cross-section of Screw jack

Step I Problem Specification

It is required to design a screw jack for supporting the machine parts during their repair and maintenance. It should be a general purpose jack with a load carrying capacity of 50KN and a maximum lifting height of 0.3m. The jack is to be operated by means of a D.C motor.

Step II Selection of Materials

- The frame of the screw jack has complex shape. It is subjected to compressive stress. Grey cast iron is selected as the material for the frame.
- Cast iron is cheap and it can be given any complex shape without involving costly machining operations.
- Cast iron has higher compressive strength compared with steel. Therefore, it is technically and economically advantageous to use cast iron for the frame.
- The screw is subjected to torsional moment, compressive force and bending moment. From strength consideration, EN8 is selected as material for screw.
- There is a relative motion between the screw and the nut, which results in friction. The friction causes wear at the contacting surfaces.
- When the same material is used for these two components, the surfaces of both components get worn out, requiring replacement. This is undesirable.
- The size and shape of the screw make it costly compared with the nut. The material used for the nut is stainless steel.

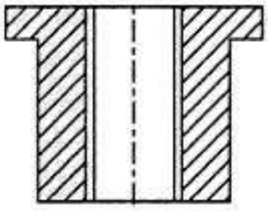

Figure 2 Cross-section of Nut

Step III Design of Screw

The screw jack is an intermittently used device and wear of threads is not an important consideration. Therefore, instead of trapezoidal threads, the screw is provided with square threads. Square threads have higher efficiency and provision can be made for self-locking arrangement. When the condition of self-locking is fulfilled, the load itself will not turn the screw and descend down, unless an effort in the reverse direction is applied.

3. DESIGN CALCULATIONS

3.1 **Design calculations to check the safety of Lead Screw**

Maximum Load to be lifted = 5 Ton

= 50 x 10^3 N

= 50KN

For a 5 Ton capacity screw jack, the suitable screw is the one whose nominal (major) diameter is 36mm.

Corresponding to the nominal diameter = 36mm

The pitch (p) selected is 6mm.

The core diameter (d_c) = 30mm

The mean diameter (d_m) = 33mm

Material for Lead Screw= EN8 material

The ultimate Stress = 450N/mm^2

Yield stress = 230N/mm^2 respectively.
The compressive stresses induced in lead screw due to load = 50KN
Given by

$$F_c = \frac{W}{\frac{\pi}{4}dc^2}$$
= (50 x 103 x 4)/ (π x 302)
= 70.73N/mm^2

Factor of Safety = 230/70.73= 3.25
Hence, Lead screw will bear = 50KN easily (From the Design Hand Book)
The helix angle of screw = tanα $= \frac{p}{\pi dm}$
= 6/ (πx33)
= 0.057

Therefore, α= 3.31°
Assuming coefficient of friction (θ) between screw and nut,
μ = tanθ = 0.14
θ = tan-1(0.14) = 7.96°
α < θ, hence it is a self-locking screw.
The turning moment required to rotate screw under design load is given by
T = W (dm/2) tan (α+θ)
= (50x103) (33/2) tan (3.31°+7.96°)
=164.40KN.mm
The shear stress due to torque,
Ft = 16T/ (πdc^3)
= (16x164.40x10^3)/π(30)^3
= 31.01N/mm^2
Direct stress is given by
Fs = ½√ (Fc^2 + 4Ft^2)
= ½√70.732 + 4(31.01)^2
= 47.03N/mm^2
The lead screw material has 115N/mm^2 shear strength.
Safety factor = 115/47.03
= 2.44

3. 2 Design calculations to check the safety of nut

The material of the nut used is stainless steel.
The yield stress in tension and compression are 216 N/mm^2 and 294N/mm^2 respectively.
Shear stress = 186N/mm^2
Bearing pressure between lead screw material and nut material is P_b =15N/mm^2
n = Number of threads in contact with the screwed spindle.
H = height of nut = n x p
t = thickness of screw = p/2 = 6/2 = 3mm
The number of internal thread (n) in nut for the load 50KN is given by

$$n = \frac{4w}{\pi(d^2 - d_c^2)(P_b)}$$
= (4 x 50 x 10^3)/ (π (36^2-30^2) (15))
≈ 11
H = n x p
= 11x6 = 66mm
The outer diameter of the nut, D1 = 54mm
The inner diameter of the nut, D0 =36mm
The tensile stresses induced in the nut is given by

$$Ft = \frac{4W}{\pi(D_1^2 - D_0^2)}$$
= (4x50x10^3)/π (54^2- 36^2)
=39.29 N/mm^2 which is less than 216 N/mm^2
Safety factor =216/39.29
= 5.49
Design calculations to check the buckling of screw
The maximum length of the screw above the nut when lifting the load is 100mm.
Radius of gyration (K) = ¼ d_c = ¼ x 30 = 7.5mm
Area = π/4dc^2
= π/4(30)^2 =706.85mm^2
L/K=slenderness ratio=100/7.5=13.33 56

Slenderness ratio is less than 30; therefore there is no effect of buckling and such components

3.3 Design considerations for a gear drive

Beam strength of gear teeth –Lewis Equation
Consider each tooth as a cantilever beam loaded by a normal load (WN). It is resolved into two components i.e., tangential component (WT) and radial component (WR) acting perpendicular and parallel to the centre line of the tooth respectively. The tangential component (WT) induces a bending stress which tends to break the tooth. The radial component (WR) induces a compressive stress of relatively small magnitude; therefore its effect on the tooth may be neglected. Hence, the bending stress is used as the basis for design The maximum value of the bending stress (or the permissible working stress), at the section BC is given by

Fw = M.y / I

Where
M = Maximum bending moment at the critical section, BC = WT x h,
WT = Tangential load acting at the tooth,
h = Length of the tooth = 5mm
y = Half of the thickness of the tooth (t) at critical section BC = t/2 = 4/2 = 2mm
I = moment of inertia about the centre line of the tooth = b.t2/12 =33.33
b = width of gear face = 15mm

Bearing strength of teeth WT = Fw x b x pc x y
= Fw.b.πm.y

The quantity y is known as Lewis form factor or tooth form factor and WT is called the beam strength of the tooth. The value of y in terms of the number of teeth may be expressed as follows:

y = 0.154 – 0.192/T, for 20° full depth involute system.

Permissible working stresses for gear teeth in the Lewis equation. The permissible working stress (FW) in the Lewis equation depends upon the material for which an allowable static stress (Fo) may be determined. According to the Barth formula, the permissible working stress is given by,

FW = F_o x C_v

Where F_o = allowable static stress, for cast steel heat treated- 196N/mm2
C_v = velocity factor.

The value of the velocity factor for very accurately cut and ground metallic gears operating at velocities upto 20m/s is given by,

Cv = 6/(6+v)

Where v = pitch line velocity in m/s = πDN/100 (D= Pitch circle diameter = 156mm, Speed = 60)

v =294.053m/min
= 4.900m/sec

Cv = 0.656
FW = 196 x 0.656 = 128.576 N/mm2
WT = 128.576 x 25 x 16/30
= 1714.266N = 174.806 kgf

Dynamic Tooth Load
Dynamic tooth load is given by

WD = WT + WI

Where WD = Total dynamic load,
WT = Steady transmitted load in Newton,
WI = Incremental load due to dynamic action.

For average conditions, the dynamic load is determined by using the following Buckingham equation, i.e.

$W_D = \boldsymbol{W_T + W_I + \frac{21v(b.C+W_T)}{21v+\sqrt{b.C+W_T}}}$

Where W_D = Total dynamic load in Newton,
W_T = Steady transmitted load in Newton = 174.806
V = Pitch line velocity in m/s = 3.141 m/s
b = Face width of gears in mm = 15mm
C = A deformation or dynamic factor in N/mm.

A deformation factor (C) depends upon the error in action between teeth, the class of out of the gears, the tooth form and the material of the gears. The value of C in N/mm may be determined by using the following relation:

$$C = \frac{\boldsymbol{K.e}}{\frac{1}{E_P}+\frac{1}{E_G}}$$

Where K = A factor depending upon the form of the teeth.
= 0.111 for 20° full depth involute system.
EP = Young's modulus for the material of the pinion in N/mm^2 = 2×10^3 N/mm^2
EG = Young's modulus for the material of the gear in N/mm^2 = 2×10^3 N/mm^2
e = Tooth error action in mm = 0.0700 for 3.141m/s

The maximum allowable tooth error in action (e) depends upon the pitch line velocity (v) and the class of cut of the gears

$$C = \frac{0.111 \ x \ 0.0700}{\frac{1}{2 \ x \ 10^3}+\frac{1}{2 \ x \ 10^3}}$$

$$C = 7.77$$

$$WD = 174.806 + W_I + \frac{21 \ x \ 3.141.(21 \ x \ 7.77+174.806)}{21 \ x \ 3.141+\sqrt{25} \ x \ 7.77+174.80}$$

$$= 2862.456N = 291.889 \text{ kgf}$$

Wear Tooth Load

The maximum load that gear teeth can carry, without premature wear, depends upon the radii of curvature of the tooth profiles and on the elasticity and surface fatigue limits of the materials. The maximum of the limiting load for satisfactory wear of gear teeth is obtained by using the following Buckingham equation, i.e.

$$Ww = Dp.b.Q.K$$

Where Ww = Maximum or limiting load for wear in Newton,

Dp = Pitch circle diameter of the pinion in mm = 45mm

b = Face width of the pinion in mm = 16mm

Q = Ratio factor = 1.56

$= \frac{2 \ x \ V.R}{V.R+1} = \frac{2T_G}{T_G+T_P}$, for external gears.

V.R = Velocity ratio = T_G/T_P

K = Load stress factor in N/mm^2.

The load stress factor depends upon the maximum fatigue limit of compressive stress, the pressure angle and the modulus of elasticity of the materials of the gears. According to Buckingham, the load stress factor is given by the following relation:

$$K = \frac{(f_{es})^2 sin\emptyset}{1.4}\left(\frac{1}{E_P}+\frac{1}{E_G}\right)$$

Where f_{es} = Surface endurance limit in N/mm² =630N/mm^2

∅= Pressure angle= 20°

EP = Young's modulus for the material of the pinion in N/mm^2 = ²N/mm^2

EG = Young's modulus for the material of the gear in N/mm^2 = N/mm^2

$$K = \frac{(630)^2 sin20}{1.4}\left(\frac{1}{2 \ x \ 10^2}+\frac{1}{2 \ x \ 10^2}\right)$$

$$= 96.9627$$

Wear Tooth Load

Ww = Dp.b.Q.K

= 48 x 16 x 1.56 x 96.9627

Ww = 116,169.907N

= 11846 kgf

Static Tooth Load

The static tooth load is obtained by lewis formula by substituting flexural endurance limit or elastic limit stress (fs) in place of permissible working stress (fw).

WS = fe.b.pc.y

= fe.b.πm.y

= 84 x 16 x π x 1.92 x (0.152-0.192/25)

= 3854.9N

= 393.09 kgf

For safety, against tooth breakage, the static tooth load (Ws) should be greater than the dynamic load (WD).Buckingham suggests the following relationship between Ws and WD.

For steady loads, Ws≥1.25WD

For shock loads, Ws≥1.5WD

3. 4 Design of Spur Gear In CATIA V5R20

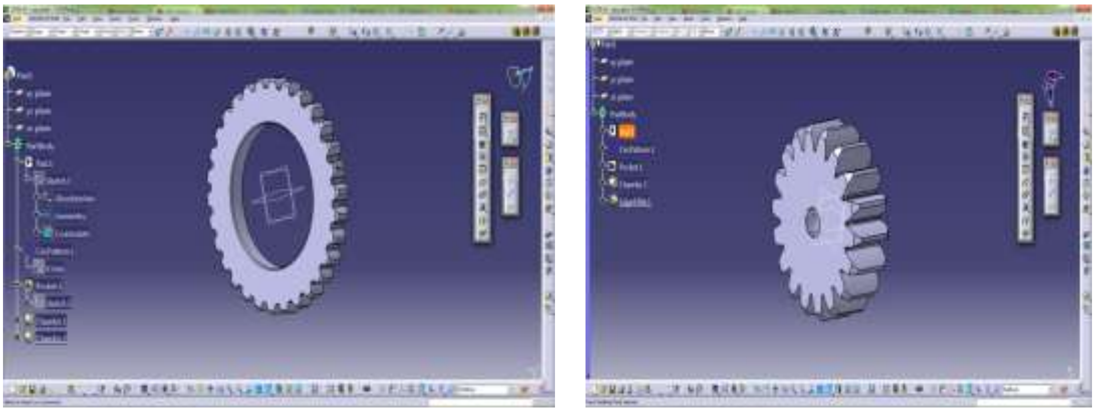

Figure 3 Gear Model

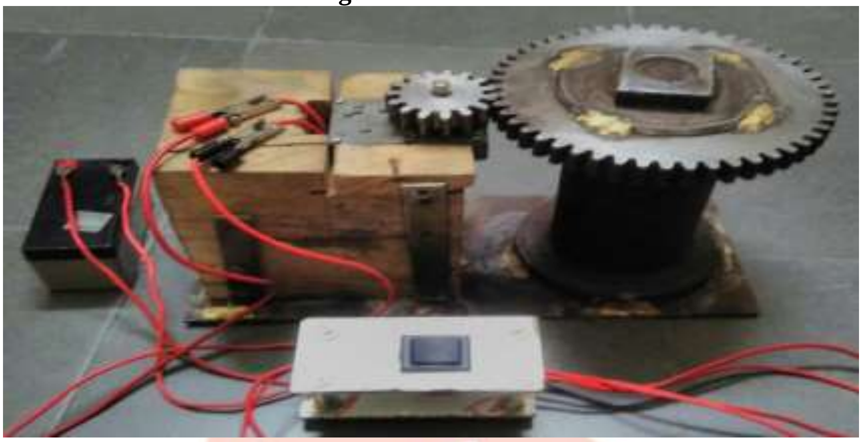

Figure 4 motorized screw jack

4. CONCLUSIONS

- 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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