# Design Development& Investigation of Remote Controlled Screw Jack for Four Wheelers cars

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**Abstract -** Power screws are used to convert rotary motion into translator 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, which is controlled by remote.
In this work, 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. And the motor will be controlled via remote from a short distance.

**Index Terms** - power screw, electric motor, screw jack, battery.

## I. INTRODUCTION

Our survey in the regard in several automobile garages, revealed the facts that mostly some difficult methods were adopted in lifting the vehicles for reconditioning.

Now the project has mainly concentrated on this difficulty, and hence a suitable device has been designed, such that the vehicle can be lifted from the floor land without application of any impact force.

The fabrication part of it has been considered with almost case for its simplicity and economy, such that this can be accommodated as one of the essential tools on automobile garages.

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 labour. 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 shaft's rotation depends upon the rotation of D.C motor. This is a simple type of automation project.

This is an era of automation where it is broadly defined as replacement of manual effort by mechanical power in all degrees of automation. The operation remains to be an essential part of the system although with changing demands on physical input, the degree of mechanization is increased.

Degrees of automation are of two types, viz.

- Full automation
- Semi automation

In semi automation a combination of manual effort and mechanical power is required whereas in full automation human participation is very negligible.

## II. PAGE LAYOUT

### A. *Parts of Remote Controlled Screw Jack*

The main parts of the motorized screw jack are as follows:

- D.C. motor (Permanent Magnet)
- Lead-Acid Battery
- Screw Jack
- Spur Gear drive
- L293D Motor Driver IC
- RF Circuit
- Control cables

### B. *Complete Connection of Motor to IC*

This is the complete connection diagram of L293D IC. 1 IC can run 2 motors at a time but according to our project requirement it'll run only 1 motor. Here 12V VCC voltages are for motor power supply and 5V VCC is for triggering.

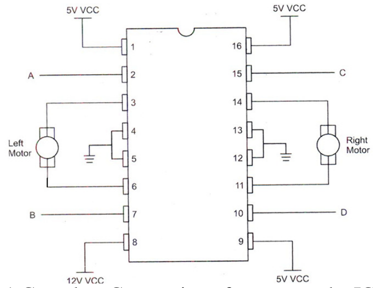

Fig. 1 Complete Connection of motors to the IC

### C. Screw Jack

The screw jack used here is a 4 tonne bottle (cylindrical) jack. It mainly consists of the body, screw, nut and thrust bearings. In this type of a jack, the nut remains stationary while the screw rotates and helps in lifting or lowering of the load.

Capacity: 4 Ton
Closed Height: 160mm
Mechanical Height: 75mm
Total Height: 235mm
Net Weight: 3.000 KG

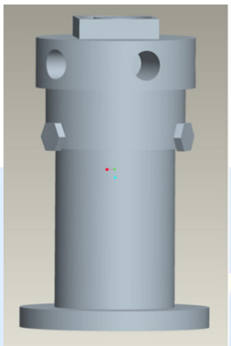
Fig. 2 Design of Screw Jack

### D. Working Principle

The lead-acid battery is used to drive the D.C. motor. The D.C. motor shaft is connected to the spur gear. If power is given to the D.C. motor, it will run so that the spur gear also runs to slow down the speed of the D.C. motor. The screw jack moves the screw upward, so that the vehicle lifts from ground.

The vehicle is lifted by using the lifting platform at the top of the screw jack. The motor draws power supply from the battery. The lifting and uplifting is done by changing the battery supply to the motor.

## III. DESIGN CALCULATIONS

### A. Design calculations to check the safety of lead screw

Maximum Load to be lifted = 4 Ton
= 4 x $10^3$ N
= 40 KN

For a 40 Ton capacity screw jack, the suitable screw is the one whose nominal (major) diameter is 25mm.
Corresponding to the nominal diameter 25 mm, the pitch (p) selected is 6mm.
The core diameter ($d_c$) = 20 mm
The mean diameter ($d_m$) = 23 mm
EN8 material is used for lead screw. The ultimate and yield stresses are 450N/mm$^2$ and 230N/mm$^2$ respectively.
The compressive stresses induced in lead screw due to load of 40KN is given by

$$F_C = \frac{W}{\frac{\pi}{4} {d_C}^2} = \frac{40000 \times 4}{\pi \times 20^2}$$

$$F_C = 127.388\, N/mm^2$$

$$Factor\ of\ Safety = \frac{230}{127.388} = 1.80$$

Hence lead screw will bear 40 KN easily.
The Helix angle of screw = $\tan \alpha = \frac{P}{\pi d_m} = \frac{6}{\pi \times 23} = 0.083$
Therefore, $\alpha = 4.75°$
Assuming coefficient of friction between screw and nut,

$$\mu = \tan \theta = 0.14$$
$$\theta = 0.14 = 7.96°$$

$\alpha < \theta$, Hence it is a self-locking screw.

The turning moment required to rotate the screw design load is given by,

$$T = W\left(\frac{d_m}{2}\right)\tan(\alpha + \theta)$$

$$T = (40000)\left(\frac{23}{2}\right)\tan(4.75°\& + 7.96°)$$

$$T = 103.75\ KN.mm$$

The shear stress due to torque,

$$F_t = \frac{16T}{\pi {d_C}^3}$$

$$F_t = \frac{16}{\pi 20^3}103749$$

$$F_t = 66.04\ N/mm^2$$

Direct Stress is given by,

$$F_s = \frac{1}{2}\sqrt{{F_C}^2 + 4{F_t}^2}$$

$$F_s = \frac{1}{2}\sqrt{127.388^2 + 4(66.04)^2}$$

$$F_s = 91.75\ N/mm^2$$

The lead screw material has 115 $N/mm^2$ shear strength.

$$Factor\ of\ safety = \frac{115}{91.75} = 1.25$$

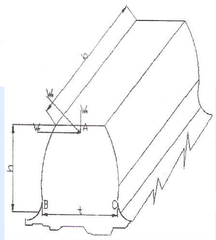

Fig. 1Profile of Gear Teeth

*B. 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 294 $N/mm^2$ respectively.

Shear stress = 186 $N/mm^2$

Bearing pressure between lead screw material and nut material is $P_b = 15\frac{N}{mm^2}$

n= Number of threads in contact with the screwed spindle

H= height of nut = $n \times p$

t=thickness of screw = $p/2 = 3mm$

The number of internal thread (n) in nut for the load 40 KN is given by,

$$n = \frac{4W}{\pi(d^2 - d_c^2)(P_b)}$$

$$= \frac{4 \times 40000}{\pi(25^2 - 20^2)15} \approx 15$$

$H = n \times p$

$$= 15 \times 6$$

$$= 90\ mm$$

The outer diameter of the nut, $D_1 = 80\ mm$

The inner diameter of the nut, $D_2 = 25\ mm$

The tensile stresses induced in the nut is given by,

$$F_t = \frac{4W}{\pi(D_1^2 - D_2^2)}$$

$$= \frac{4 \times 40000}{\pi(80^2 - 25^2)}$$

$= 8.81\ N/mm^2$, which is less than $216\ N/mm^2$

Factor of Safety=216/8.81=24.49

*C. Design calculations to check the buckling of screw*

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The maximum length of the screw above the nut when lifting the load is 75 mm.

Radius of gyration $(K) = \frac{1}{4}d_c = \frac{1}{4}20 = 5\ mm$

Area $= \frac{\pi}{4}d_c^2$

$$= \frac{\pi}{4}(20)^2$$
$$= 314\, mm^2$$

$$Slenderness\ ratio = \frac{L}{K} = \frac{75}{5} = 15$$

Slenderness ratio is less than 30, therefore there is no effect of buckling and such components are designed on the basis of compressive stresses.

*D. 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 ($W_N$). It is resolved into two components i.e., tangential component ($W_T$) and radial component ($W_R$) acting perpendicular and parallel to the centre line of the tooth respectively.

The tangential component ($W_T$) induces a bending stress which tends to break the tooth.

The radial component ($W_R$) 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

$$F_W = M \times \frac{y}{I}$$

Where,

M=Maximum bending moment at the critical section, BC=$W_T \times h$,

$W_T$ =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=$\frac{b \times t^2}{12} = 34.66$

b=width of gear face=26mm

Bearing strength of teeth $W_T = F_W \times b \times p_c \times y = F_W . b . \pi m . y$

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

$y = 0.154 - \frac{0.192}{T}$, For 20° full depth involute system.

**Permissible working stresses for gear teeth in the Lewis equation**

The permissible working stress ($F_W$) in the Lewis equation depends upon the material for which an allowable static stress ($F_O$) may be determined. According to the Barth formula, the permissible working stress is given by,

$F_W = F_O \times C_V$

Where, $F_O$=allowable static stress, for cast steel heat treated- 196 $N/mm^2$

$C_V$ =Velocity Factor.

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

$$C_V = \frac{6}{6+v}$$

Where

v=pitch line velocity in m/s= $\frac{\pi DN}{100}$=3.14 $m/s$

$C_V$=0.656

$F_W = 196 \times 0.656 = 128.576\ N/mm^2$

$$W_T = 128.576 \times \frac{16}{30} = 1714.266N = 101.26\ kgf$$

**Dynamic Tooth Load**

Dynamic tooth load is given by

$W_D = W_T + W_I$

Where,

$W_D$ =Total dynamic load,

$W_T$=Steady transmitted load in Newton,

$W_I$ =Incremental load due to dynamic action

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

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

Where,

$W_D$=Total dynamic load in Newton

$W_T$=Steady transmitted load in Newton=101.26

$V$=Pitch line velocity in m/s=3.141m/s

$b$ =Face width of gears in mm=26mm

$C$ =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:

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

Where

K = A factor depending upon the form of the teeth

=0.111 for 20° full depth involute system.

$E_P$= Young's modulus for the material of the pinion in N/mm$^2$= $2 \times 10^3 \frac{N}{mm^2}$

$E_G$= Young's modulus for the material of the gear in N/mm$^2$ $= 2 \times 10^3 \frac{N}{mm^2}$

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,

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

$$C = 7.77$$

$$W_D = 174.806 + W_I = 101.26 + \frac{21 \times 3.141(26 \times 7.77 + 101.26)}{21 \times 3.141 + \sqrt{26 \times 7.77 + 101.26}}$$

=341 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.

$$W_W = D_P.b.Q.K$$

Where,

$W_W$=Maximum or limiting load for wear in Newton

$D_{P=}$Pitch circle diameter of the pinion in mm=48mm

$b$ =Face width of the pinion in mm= 12 mm

$Q$ =Ratio Factor=1.56

$$= \frac{2 \times V.R}{V.R + 1}$$

$$= \frac{2T_G}{(T_G + T_P)}$$

$$V.R. = Velocity\ Ratio = \frac{T_G}{T_P}$$

$K$ =Load stress factor in $\frac{N}{mm^2}$

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:

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

Where,

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

$\varphi$=Pressure angle=20°

$E_P$= Young's modulus for the material of the pinion in $\frac{N}{mm^2} = 2 \times 10^2 \frac{N}{mm^2}$

$E_G$= Young's modulus for the material of the gear in $\frac{N}{mm^2} = 2 \times 10^2 \frac{N}{mm^2}$

$$K = \frac{(630)^2 \sin 20}{1.4}\left(\frac{1}{2 \times 10^2} + \frac{1}{2 \times 10^2}\right)$$

$$= 96.9627$$

Wear Tooth Load

$W_W = D_P.b.Q.K$

$$= 85.5 \times 12 \times 1.56 \times 96.9627$$

$$= 155194.62N = 15825\ kgf$$

**Static Tooth Load**

The static tooth load is obtained by Lewis formula by substituting flexural endurance limit or elastic limit stress ($f_s$) in place of permissible working stress ($f_w$).

The static tooth load or beam strength of the tooth,

$W_S = f_c.b.p_c.y$

$$= f_c.b.\pi m.y$$

$$= 84 \times 12 \times \pi \times 1.5 \times (0.152 - 0.192/38)$$

=697.65 N

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

For steady loads, $W_S$≥1.25W

For shock loads, $W_D$≥1.5W

## IV. WORKING OF THE PROJECT

### *A. Working of the Remote Controlled Screw Jack*

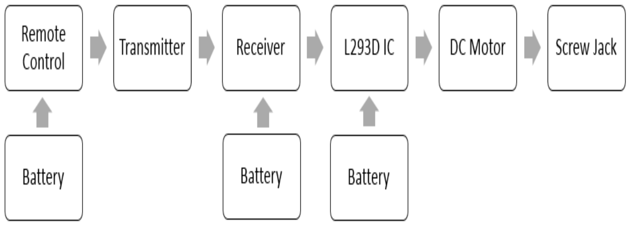


Fig. 4 Block Diagram of the Remote Controlled Screw Jack

### *B. Experimental procedure*

Following is the steps for experiment of remote controlled screw jack:

- Battery gives power to remote control. This are two 1.5v batteries.
- User operate the "Remote Control" and sends signal to receiver by transmitter. Signals are broadcast by antenna.
- Antenna receives signals at receiver side.
- L2933D IC receives the signals and by this trigger voltage it supplies 12V battery power to DC Motor.
- DC Motor receives 12V DC current and runs accordingly. The shaft of motor runs.
- The power is transmitted to Screw Jack with the help of 2 Spur Gear.
- And the Screw Jack goes upward and downward according to user's choice.

## V. READINGS AND GRAPH

TABLE I
EXPERIMENT DATA

| **Kgs.** | 0 | 100 | 200 | 300 | 400 | 500 |
|---|---|---|---|---|---|---|
| **Secs.** | 31.5 | 37.7 | 47.3 | 58.6 | 73.4 | 89.9 |

### *C. Page Layout*

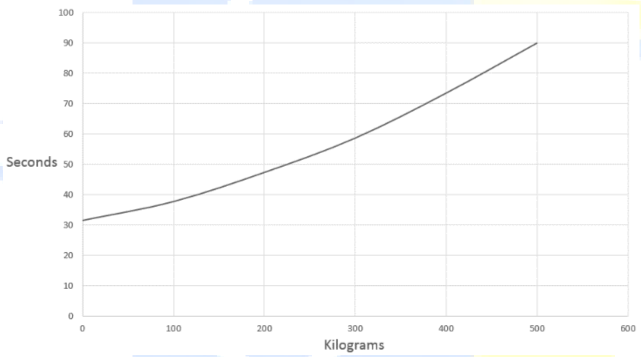


Fig.5Performance Graph

## VI. CONCLUSIONS

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 4,000-5,000 pound 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.

Hence 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

[1] Paolo Gallina, "Vibration in screw jack mechanisms: experimental results" Journal of Sound and Vibration, Volume 282, Issues 3–5, 22 April 2005.

[2] L.P. Laus, H. Simas, D. Martins , "Efficiency of gear trains determined using graph and screw theories" Mechanism and Machine Theory, Volume 52, June 2012

[3] Senthilkumar, R, "Modeling of optimal screw jack design" Frontiers in Automobile and Mechanical Engineering (FAME), ISBN:978-1-4244-9081-3 , Nov2010

[4] A. S. Akinwonmi, A. Mohammed "Modification of the Existing Design of a Car Jack" Journal of Emerging Trends in Engineering and Applied Sciences (JETEAS), ISSN: 2141-7016, 2012

[5] Ray C. Johnson, "Optimum Design of Mechanical Elements", AWilley – Interscience Publication John Wiley & Sons Inc. NewYork, ISBN - 0 – 0471 – 03894 – 6, [pp 1, 2, 3,178 -188, 193 –196.].

[6] V. B. Bhandari 2006, "Introduction to Machine Design", TataMcGraw-Hill Publishing Company Limited, New Delhi, India,ISBN – 0 – 07 – 043449 – 2.

[7] Joseph E Shigley, Charls R Misckhke, Richar G. Budynas, Kith J. Nisbett, "Mechanical Engineering Design", Tata McGrawHillPublishing Company Limited, New Delhi, 2008, ISBN 13:978-0 – 07 – 066861 – 4, ISBN 10: 0 – 07 – 066861 – 2