ISSN (e): 2250 – 3005 || Vol, 04 || Issue, 7 || July – 2014 ||
International Journal of Computational Engineering Research (IJCER)

# Design, Development and analysis of electrically operated toggle jack using power of car battery

Gaurav Shashikant Udgirkar[1], Manoj Shantinath Patil[1], Rajesh Vijay Patil[1], Nilesh Ramchandra Chavan[1], Prof. Mangesh Panchbhai[2]

*(B.E. Students, Department Of Mechanical Engineering, K. K. Wagh Institute Of Engineering Education And Research)

** (Asst. Professor, Mechanical Engineering, K. K. Wagh Institute Of Engineering Education And Research, Panchavati, Nashik - 422003 (M.H.) India)

**_ABSTRACT_**

*Side road emergency like tire puncher, is a problem commonly observed in cars. Conventional car jacks uses mechanical advantage to allow a human to lift a vehicle by manual force. This paper analyzes the modification of the current toggle jack by incorporating an electric DC motor in the screw in order to make load lifting easier for emergency use with using power of car batter (12 Volts). Gear ratio is used to increase the lifting power. 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 save individual internal energy and reduce health risks especially back ache problems associated with doing work in a bent or squatting position for a long period of time. The car jack is developed using CATIA V5R19 and analyzed using Finite Element Analysis to check safety factor and force acting. Fabrication work has been done using with milling, drilling, grinding, and welding machine. The developed car jack is tested on car. Implementation of design will solve problem associated with ergonomics*

***KEYWORDS:*** *Car battery, CATIA, D.C motor, ergonomics, gear ratio, jack, screw*

## I. INTRODUCTION

Toggle jack is used to jack the car during side road emergency i.e. tire puncher. A mechanical jack is a device used to lift heavy equipment, all or part of a vehicle into the air in order to facilitate vehicle maintenances or breakdown repairs [1]. Changing a flat tire is not a very pleasant experience. Nowadays, a variety of car jacks have been developed for lifting an automobile from a ground surface. Available car jacks, however, are typically manually operated and therefore require substantial laborious physical effort on the part of the user. Such jacks present difficulties for the elderly and handicapped. It further requires 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. A toggle jack is operated by turning a lead screw. In this case of a jack, a small force applied in the horizontal plane is used to raise or lower large load [2]. 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 weakened by wear over many rotations. An inherent advantage is that, if the tapered sides of the screw wear, the mating nut automatically comes into closer engagement, instead of allowing backlash to develop [3]. These type are self locking, which makes them safer than other jack technologies like hydraulic actuators which requires continual pressure to remain in locked positionThe automobile service stations are commonly equipped with large and hi-tech car lift, wherein such lifts are raised and lowered via electrically-powered systems. However, due to their size and high costs of purchasing and maintaining, such lifts are not feasible to be placed in car and owned by car owner. Such electrical-powered portable jacks not only reduce the effort required for lifting an automobile via manually-operated jacks, but also decrease the time needed to repair the automobile. Such a feature can be especially advantageous when it is necessary to repair an automobile on the side of a roadway or under other hazardous conditions [4]. A specified jack purposed to hold up to 1000 kilograms, but tests undertaken by Consumer Affairs has revealed that is fails to work after lifting 250 kilograms and may physically break when it has a weight close to its 1000 kilograms capacity [5]. Tests have proven that the jack has the tendency to buckle under the weight it is subjected to withstand [6]. The purpose of this project is to develop a car jack which is easy to be operated, safe and able lift and lowering the car without involving much physical effort. This paper discussed the design and analysis of modified car jack.

## II. WORKING OF ELECTRICALLY OPERATED JACK

Under working condition the jack will lift a vehicle chassis in contact with the top plate when the power screw is rotated through its connecting gear with the pinion when electrical power applied to the wiper motor when plugged to the 12V battery in car. Motor 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 side member in the clockwise direction that will cause the links to be drawn along the threaded portion towards each other during load-raising process and vice versa. Initially the jack will first be placed below the chassis to be lifted such that at least a small clearance space will exist between the top plate and the vehicle chassis to be raised. Then after power screw will be turned so that the top plate makes contact with the car chassis and the clearance space is eliminated. As contact is made, load of car will be increasingly shifted to the top plate and cause forces to be developed in and transmitted through links and side member. The force transmitted through the side member will be transferred on threads of screw. A switching circuit connected to the motor is used to regulate the lifting and lowering process.

## III. DESIGN AND DEVELOPMENT OF JACK

Figure 3.1 shows modified toggle jack. The main components of required for development of this jack are:

a) Original toggle jack with alteration and replacement of Power Screw and Side members.
b) Power screw.
c) Gear pair.
d) DC wiper motor.
e) Bracket and holder to carry motor.

Table 3.1 includes details of all components required for building the actual model or prototype of electrically operated toggle jack.

**Table 3.1** List of component.

| Component number | Description | Quantity |
|---|---|---|
| 1 | Top plate | 1 |
| 2 | Upper link | 1 |
| 3 | Link | 4 |
| 4 | Power screw | 1 |
| 5 | Side member | 2 |
| 6 | Bottom link | 1 |
| 7 | Stabilizer base | 1 |
| 8 | Solid block | 2 |
| 9 | Pin | 8 |
| 10 | M10 bolt | 5 |
| 11 | Pinion and hub | 1 |
| 12 | Wiper motor | 1 |
| 13 | Bracket assembly | 1 |
| 14 | M15 bolt | 1 |
| 15 | Gear and hub | 1 |
| 16 | Bracket assembly holder | 1 |
| 17 | Screw collar | 1 |

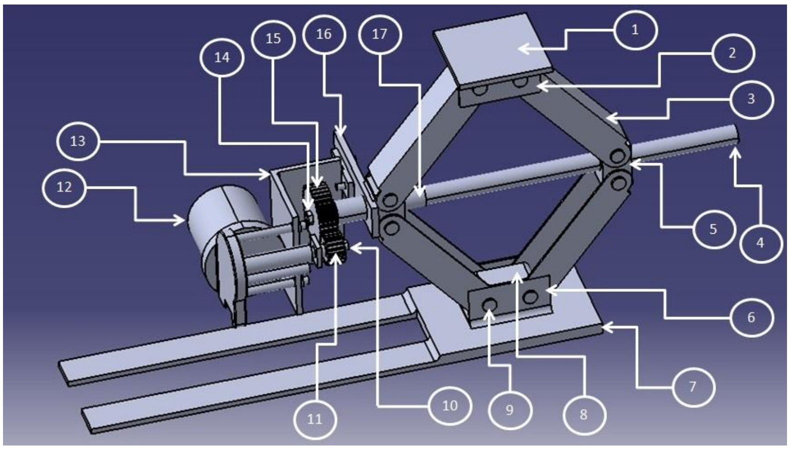


**Figure 3.1:** Modified toggle jack.

Figure 3.2 shows the actual model of electrically operated jack. The prototype is fabricated by making use of the jack and bus wiper motor brought from junk yard.

**Figure 3.2:** Actual model of modified toggle jack.

## IV. MATERIAL SELECTION

Scissor jacks are usually made of materials that are very strong and are suitable for withstanding heavy loads. The two main materials used for making good quality jacks are Steel and Aluminium. When selecting the material suitable for the construction of the Scissor jack one has to consider the properties that will enable it to function with no expected failure and at the same time the weight and ease of machining the product. Therefore the main areas that can be classified in this case are the strength of the material, weight, ease and cost of manufacturing. Aluminium is around one-third the density of steel at 2.72 mg/m cubed compared to steel's 7.85 mg/m cubed. The light weight and low melting point of aluminium makes it easier and more efficient to machine than steel. Aluminium's fatigue performance is half that of steel, which is an advantage steel has over aluminium in car jack life durability. Therefore Steel is the most viable material selected for the manufacture of the car scissor jack [7] Component number 4, 5and 17 will all use the High Strength Low-Alloy Steel (40Ni2Cr1Mo28 / AISI 4340), material is selected on bases of application. Material Property is given in Table 4.1 below:

**Table 4.1** Material property

| | | | |
|---|---|---|---|
| Tensile Strength, Ultimate | 931 MPa | 135000 psi | |
| Tensile Strength, Yield | 834 MPa | 121000 psi | |
| Elongation at Break | 20.2 % | 20.2 % | |
| Modulus of Elasticity | 205 GPa | 29700 ksi | Typical for steel |
| Bulk Modulus | 140 GPa | 20300 ksi | Typical for steel |
| Poisson's Ratio | 0.29 | 0.29 | Calculated |
| Shear Modulus | 80 GPa | 11600 ksi | |

*http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=M434AP*

Component number 7, 13 and 16 will all use the High Alloy steel plates (N8). . Material Property is given in Table 4.2 below:

**Table 4.2** Material property

| `Steel plate/Sheet thickness / mm | σb MPa | σs ≥/MPa | δ Samples from the standard for 50 mm (2 in) | 180 ° of cold bending test | |
|---|---|---|---|---|---|
| | | | | longitudinal | horizontal |
| Hot-rolled/Cold rolling:5 - 150 | 520 | 415 | 16～18 | 2a | 3.5a |

*http://www.steel-grades.com/Steel-grades/High-alloy/denertia-n8.html*

Component number 11 will use the Mild steel/ plain carbon steel (C45/ AISI 1045) and 15 will use the Mild steel/ plain carbon steel (C35 Mn75/IS new: 35C8/ AISI 1035, *http://www.btss.in/technical.php*). Material Property of C45 and C35 Mn75 are given in Table 4.3 below [8]:

**Table 4.3** Material property

| **Parameter** | **Pinion** | **Gear** |
|---|---|---|
| Material | C45 | C35Mn75 |
| Tensile strength, ($\sigma_t$) | 670 $N/mm^2$ | 600 $N/mm^2$ |
| BHN | 229 | 223 |
| Elastic modulus, (*E*) | 210 $GP_a$ | 190 $GP_a$ |
| Izod impact value | 41 $N/mm^2$ | 55 $N/mm^2$ |

## V. DESIGN CALCULATION

We have brought the toggle jack from junk yard which needs little alteration, Parts like power screw and side member are newly designed and replaced. Design of screw and side member are as follows:

**5.1 Assumptions:**

- The ground clearance of the vehicle is assumed to be 165 mm. (Decided after study of various car specifications.)
- When the screw jack carries the maximum load, i.e. when the wheel of the vehicle leaves the ground, the screw jack is assumed to have moved in the vertical axis (linearly) by a distance of 50mm. (Decided after practical observational analysis of conditions.)
- The screw jack supports a quarter of the total vehicle mass, which is approximately 300 kg, i.e. 3000 N of force of car of weight 1200 kg i.e. 12000 N. For safety design weight is taken as 500 kg i.e. 5000 N.

**5.2 Condition derived for initiating design:**

Input parameters are decided by making a study of cars specifications and various loading conditions. Some input are decided by practical analysis of the vehicles lifting condition while jacking during tyre failed condition.

**5.2.1 Input:**

a. Maximum weight of car = 500 *kg* / 5000 *N.*
b. Ground clearance = 165 *mm.*
c. Maximum lift = 50 *mm.*

**5.2.2 Derived:**

a. Max load on jack depending on condition of road:
Conditions:
I. On horizontal road surface.
II. On slop.
It is observed that max load acts on jack when vehicle is on horizontal surface.

b. Angle between the links with horizontal axis (θ) show in figure 5.1 and figure 5.2 after studying jack brought from junk yard:

.

i. Angle in upper position ($\theta_{max}$) :

$$\tan(\theta_{max}) = \frac{120}{85-35/_2}$$

$$\theta_{max} = 78.23^0.$$

ii. Angle in lower position ($\theta_{min}$):

$$\tan(\theta_{min}) = \frac{57.5}{235-35/_2}$$

$$\theta_{min} = 29.89^0$$

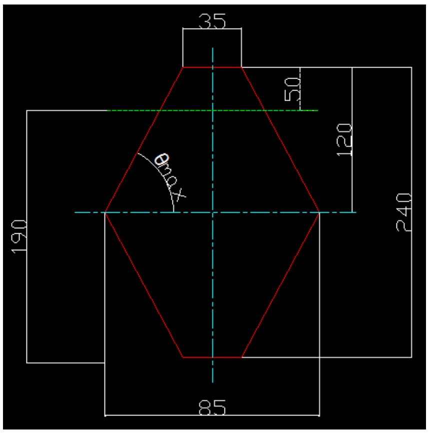


**Figure 5.1:** Upper condition of jack.

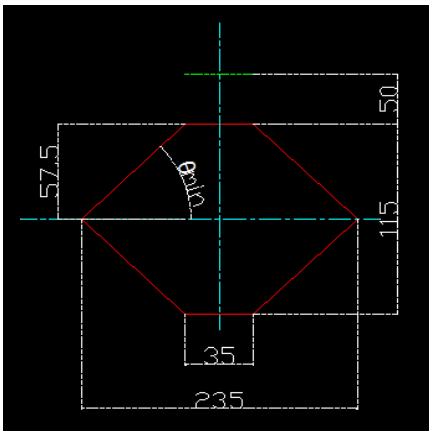


**Figure 5.2:** Lower condition of jack.

### 5.3 Force analysis in toggle jack:

Figure 5.3 shows force distribution in jack body.

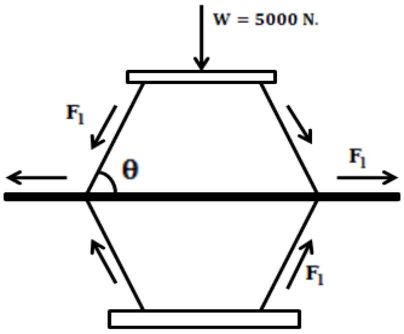


**Figure 5.3:** Force distribution in jack body.

### 5.4 Design of screw:

From figure 5.3

$\sum F_H = 0.$

$$\therefore (F_1 * \cos\theta) - \frac{W}{2} = 0 \text{ or } F_1 = \frac{W * \cos\theta}{2 * \sin\theta} = \frac{W}{\tan\theta}$$

Total axial force in screw (Ws)

$$W_s = 2F_1 = \frac{2 * W}{2 * \tan\theta}.$$

Hence, the axial force (Ws) in a screw is maximum when ($\theta$) is minimum.

$$\therefore W_s = \frac{W}{\tan\theta_{min}} = \frac{5000}{\tan 29.89} = 8689.78\ N.$$

From section 4, we have $\sigma_{yt}$ = 834 $N/mm^2$. and $\tau_s = \frac{\sigma_{yt}}{2} = \frac{834}{2}$ = 417 $N/mm^2$. [2]

Assume, factor of safety ($N_f$) = 3, Service factor (K) = 1.6. [2]

$$\therefore \sigma_{allowable} = \frac{\sigma_{yt}}{K * N_f} = \frac{834}{1.6 * 3} = 173.75\ N/mm^2. \quad (1)$$

$$\therefore \tau_{allowable} = \frac{\sigma_{allowable}}{2} = \frac{173.75}{2} = 86.875 N/mm^2. \quad (2)$$

The direct tensile stress in screw body is given as follow

$$\sigma_t = \frac{W_s}{\pi/4 * d_c^2} \text{ or } 173.75 = \frac{8689.78}{\pi/4 * d_c^2} \quad (3)$$

$d_c = 7.98\ mm$, Say $d_c = 11.5\ mm$ standard size.

Selecting standard screw [2]:

Core diameter ($d_c$) = 11.5 *mm.*
Outer diameter (d) = 16 *mm.*
Mean diameter ($d_m$) = 14 *mm.*
Pitch (p) = 4 *mm.*

Length of screw (l) = 235 + 2*16 = 267 *mm.*

Torque required for overcoming the thread friction ($T_f$) [2]:

For Acme thread, $\beta = 14.5°$.

Helix angle, $(\lambda) = \tan^{-1}\frac{l}{\pi * d_m} = \tan^{-1}\frac{4}{\pi * 14} = 5.19°$.

Coefficient of friction, $(\mu') = \frac{\mu}{\cos\beta} = \frac{0.15}{\cos 14.5} = 0.154$

Friction angle, $(\emptyset) = \tan^{-1}\mu' = \tan^{-1} 0.154 = 8.80°$

Required Torque, $(T_f) = \frac{w * d_m}{2 * \tan\theta_{min}} * \tan(\emptyset + \lambda) = \frac{5000 * 14}{2 * \tan 29.89} * \tan(8.80 + 5.19) = 15.17066\ Nm.$

(4)

Efficiency of threads, $(\eta) = \frac{1 - \sin\emptyset}{1 + \sin\emptyset} = \frac{1 - \sin 8.80}{1 + \sin 8.80} = 73.46\ \%$

Actual torque required, $(T) = \frac{T_f}{\eta} = \frac{15170.66}{0.7346} = 20.65084\ Nm.$ (5)

The direct tensile stress in screw body, $(\sigma_t) = \frac{W_s}{\pi/_4 * d_c^2} = \frac{8689.78}{\pi/_4 * 11.5^2} = 83.66\ N/mm^2$. (6)

Shear stress due to torque, $(\tau_s) = \frac{16 * T}{\pi * dc^3} = \frac{16 * 20650}{\pi * dc^3} = 69.15\ N/mm^2$ (7)

Maximum principle stress theory, $(\sigma) = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{\sigma_t^2 + 4 * \tau_s^2} = \frac{83.66}{2} + \frac{1}{2}\sqrt{83.66^2 + 4 * 69.15^2}$ (8)

$(\sigma) = 123.33\ N/mm^2$

$\sigma = 123.33 < 173.75\ N/mm^2$. .

Hence design is safe.

Maximum shear stress theory, $(\tau) = \sqrt{(\frac{\sigma_t}{2})^2 + \tau_s^2} = \sqrt{(\frac{83.66}{2})^2 + 69.95^2} = 81.50\ N/mm^2$ (9)

$\tau = 81.50 < 86.875\ N/mm^2$

Hence design is safe.

**5.5 Design of Side members:**

$n = \frac{W_s}{\frac{\pi}{4} * [do^2 - dc^2] * P_b} = \frac{8689.78}{\frac{\pi}{4} * [16^2 - 11.5^2] * 15} = 5.96 \approx 6$ [10] [2]

Length of side member, (l) = n * p = 6 * 4 = 24 *mm*.

Height of side member, (h) = 2.5 * $d_o$ = 2.5 * 16 = 40 *mm*.

From figure 5.3 force acting on side member is equal to $W_s$

Check for shear failure of side member threads

$\tau = \frac{W_s}{n * \Pi * d_c * \frac{p}{2}} = \frac{8689.78}{6 * \Pi * 11.5 * \frac{4}{2}} = 20.04\ N/mm^2$ (10)

$\tau = 20.04 < 86.875\ N/mm^2$

Hence design is safe.

**5.6 Motor selection:**

The motor is from the junk yard and it is from used bus wiper motor. From the manufacturer and calculated value for the torque, it supplied 13 Nm torque which is high enough and suitable for the project. Specification of motor obtained from label on motor:

1. Brand name: Lucas TVS
2. Part no: 3SW60
3. Use: Wiper
4. Type: DC motor
5. Motor: Brush
6. Power: 60 Watt
7. Torque: 13 Nm
8. Voltage: 12 V
9. Current: 2A

**5.7 Condition for initiating design of gear pair:**

Input parameters are decided studying and understanding the parameters at output of motor and input required for screw of jack. And certain assumptions are considered for designing of gear pair.

**5.7.1 Initial data:**

a. Input torque ($T_1$) = 13 *Nm.*
b. Input speed ($N_1$) = 50 *rpm* (Obtained by testing motor).
c. Output torque ($T_o$) = 20.65084 *Nm*.
d. Gera ratio (G) = $\frac{\text{output torque}}{\text{input torque}} = \frac{20.650840}{13}$ = 1.58853 ≈ 2
e. Actual Output torque ($T_2$) = 26 *Nm*.
f. Output speed ($N_2$) = $\frac{\text{input speed}}{G} = \frac{50}{2}$ = 25 *rpm.*

**5.7.2 Assumptions:**

a. Number of teeth on pinion ($t_p$) = 18.
b. Number of teeth on pinion ($t_g$) = $t_p * G$ = 36.
c. Factor of safety ($N_f$) = 1.5
d. Assuming 20° full-depth involutes tooth system, $\emptyset = 20^{\circ}$.

**5.8 Design of gear pair:**

Bending stress on pinion, ($\sigma_b$) = $\frac{\sigma_t}{3}$

Lewis form factor (Y):

For 20° full-depth involutes tooth system.

$$Y = 0.154 - \frac{0.192}{t}. \text{ [2]} \tag{11}$$

Flexure endurance stress, $(\sigma_e) = 1.75 * \text{BHN}$ *N/mm²* [2]

Surface endurance stress, $(\sigma_{es}) = 2.8 * \text{BHN} - 70$ *N/mm²* [2]

**Table 5.1** Properties of pinion and gear. [8]

| Parameter | Pinion | Gear |
|---|---|---|
| Material | C45 | C35Mn75 |
| Tensile strength, ($\sigma_t$) | 670 $N/mm^2$ | 600 $N/mm^2$ |
| Tensile strength, ($\sigma_t$) | 670 $N/mm^2$ | 600 $N/mm^2$ |
| Bending strength, ($\sigma_b$) | 223.33 $N/mm^2$ | 200 $N/mm^2$ |
| Lewis form factor, (*Y*) | 0.1433 | 0.1487 |
| BHN | 229 | 223 |
| $\sigma_b * Y$ | 32.00 | 29.74 |
| Flexure endurance stress, ($\sigma_e$) | 400.75 $N/mm^2$ | 390.25 $N/mm^2$ |
| Surface endurance stress, ($\sigma_{es}$) | 571.2 $N/mm^2$ | 554.4 $N/mm^2$ |
| Elastic modulus, (*E*) | 210 $GP_a$ | 190 $GP_a$ |
| Izod impact value | 41 $N/mm^2$ | 55 $N/mm^2$ |

Since $(\sigma_t * Y)_g < (\sigma_t * Y)_p$, therefore gear is weaker. Now using Lewis equation to the gear we have beam strength of the tool as below:

$$W_T = \sigma_{wg} * m * p_c * Y_g = (\sigma_{bg} * K_v) * m * (b * \pi) * Y_g \text{ [2]} \tag{12}$$

Pitch line velocity, (V) = $\frac{\pi * d_p * N_p}{60000} = \frac{\pi * 18\ m * 50}{60000}$ = (0.0471 m) *m/sec*.

Velocity factor, ($K_V$) = $\frac{6}{6+V} = \frac{6}{6+0.0471\ m}$.

Application factor or Service factor, ($K_a$) = 1.0

Beam strength of the tool, $(W_T) = \frac{P}{V} * K_a = \frac{60}{0.0471 * m} * 1.0 = \frac{1273.24}{m}\ N$

Ratio factor, $(Q) = \frac{2 * t_g}{t_g + t_p} = \frac{2 * 36}{36 + 18}$ = 1.333

Load-stress factor for gear, $(K) = \frac{\sigma_{esg}{}^2 * \sin\emptyset}{1.4} * \left(\frac{1}{E_p} + \frac{1}{E_g}\right) = \frac{554.4^2 * \sin 20}{1.4} * \left(\frac{1}{210000} + \frac{1}{190000}\right) = 0.715$

Now,

$$W_T = \frac{P}{V} * K_a = (\sigma_{bg} * K_v) * m * (b * \pi) * Y_g \text{ [2]}$$

(13)

$$\frac{1273.24}{m} = \left(200 * \frac{6}{6 + 0.0471m}\right) * m * (10m * \pi) * 0.1487$$

$$\frac{1273.24}{m} = \left(\frac{5605.8579m^2}{6 + 0.0471\ m}\right)$$

$$6 + 0.0471m = 4.4028m^3$$

$$m = 1.112 \approx 2\ mm$$

Various parameters of gear pair:

Face width, $(b) = 10m = 10 * 2 = 20\ mm$

Pitch diameter of pinion, $(d_p) = m * t_p = 2 * 18 = 36\ mm$

Pitch diameter of pinion, $(d_g) = m * t_g = 2 * 36 = 72\ mm$

Pitch line velocity, (V) = 0.0471m = 0.0942 *mm/sec*

Check the gear for dynamic and wear load,

Wear strength, $(W_w) = d_p * b * Q * K = 36 * 20 * 1.333 * 0.715 = 686.2284\ N$ (14)

Beam strength of the tool, $(W_T) = \frac{1273.24}{m} = \frac{1273.24}{2} = 636.62\ N$ (15)

Since $W_w > W_T$ are design is safe from standpoint of wear

## VI. ANALYSIS

This section shows the details of Finite Element Analysis of this developed prototype. The Finite Element Method is the easy technique to the theoretical method to find out the stress developed in various components of toggle jack. In this paper Finite Element Analysis is carried out in ANSYS Workbench 11 to determine the maximum stress in toggle jack and gear when applied with boundary conditions. Also the deformation is found out for jack and gear pair.

**6.1 Steps in analysis [11]:**

a. Step 1: Import geometry

Figure 6.1 shows CATIA model imported in Ansys.

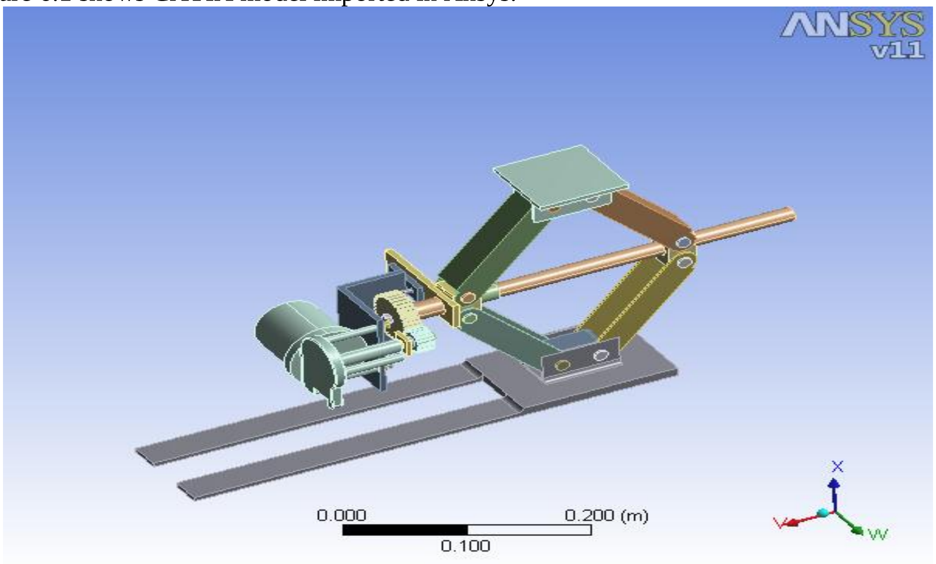


**Figure 6.1:** 3-D geometry of scissor jack.

b. Step 2: Meshing

Figure 6.2 shows the component meshing. Cores meshing of geometry are performed.

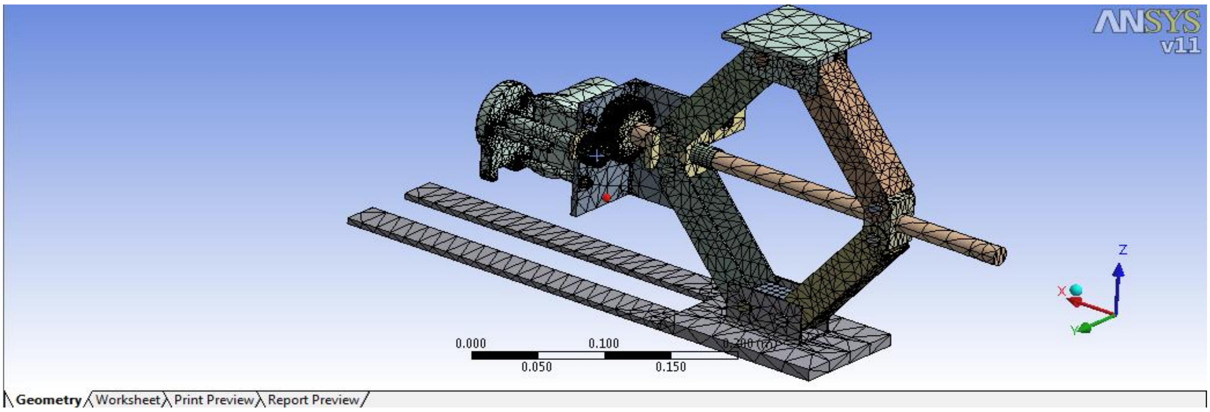


**Figure 6.2:** Coarse mesh of scissor jack.

c. Step 3: Boundary conditions:

Stabilizer base is fixed as per required initial condition. The load of 5000 N is applied on top plate of jack in geometry. Fixed support is applied on inner rim of the pinion. Frictionless support is applied on inner rim of gear to allow its tangential rotation but restrict radial translation. Moment of 13 N-m is applied on outer rim of pinion in clockwise direction as a driving torque. Figure 6.3(a) and 6.3(b) shows applied boundary conditions.

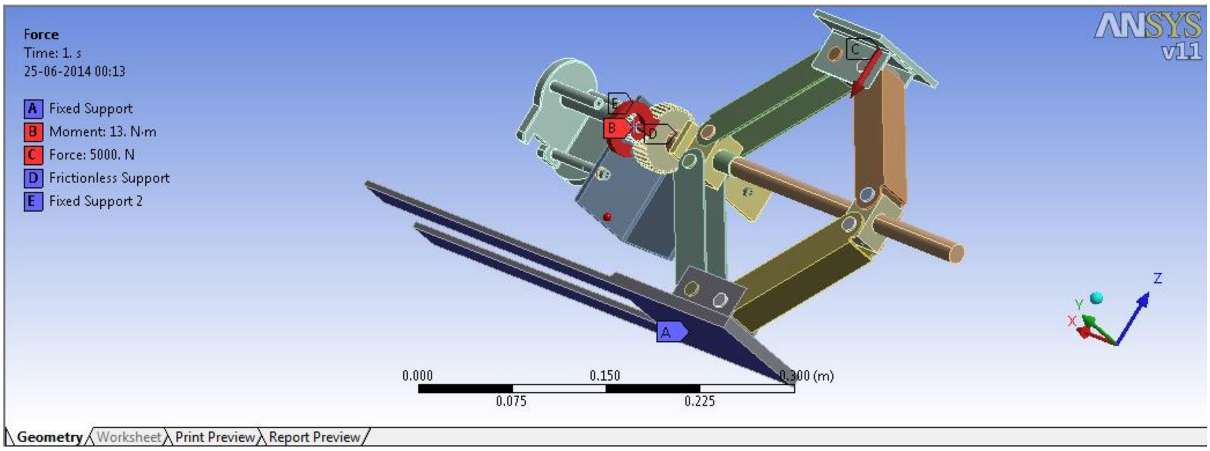


**Figure 6.3(a):** Boundary conditions.

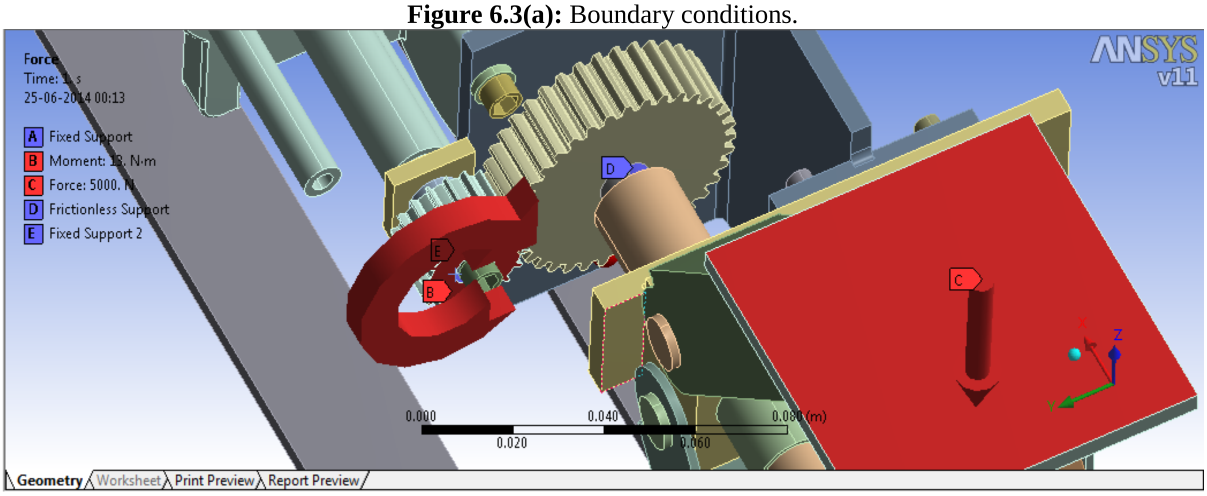


**Figure 6.3(b):** Boundary conditions.

## VII. RESULTS AND DISCUSSION

The design was focused on all the processes of conception, visualization, calculation, refinement and specification of details that determine the form of the product. Hence, the said electrically operated toggle 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. From the force and stress diagram in figure, it was discovered that at the maximum raising height of 240mm of the horizontal Tensile force in the opposite direction are the same. It is also the same for the minimum raising height of 115mm. Mild steel is used as the materials for both gears due to its high strength, toughness, tooth hardness and its economical effects.

Based on the analysis on Finite Element Analysis, it shows that the maximum nodal displacement magnitude on the system car jack is around 0.087974 mm as shown in Figure 7.1 when maximum load (5000 N) applied Furthermore, it observed that maximum Von Misses stress, maximum principle stress, maximum shear stress and shear stress values in safe point because analyzed < σy since tensor stress for all material used. Compression between analyzed and allowable material value of stress are in Table 3.5 below:

**Table 7.1** Stress comparison table

| Parameter | Analytical | Allowable (Jack) | Allowable(Gear) | Safety |
|---|---|---|---|---|
| Von-miss stress | 135.06 *N/mm* | 173.75 *N/mm* | 400 *N/mm* | Safe |
| Max. Principle stress. | 145.33 *N/mm* | 173.75 *N/mm* | 400 *N/mm* | Safe |
| Max shear stress | 71.57 *N/mm* | 86.875 *N/mm* | 200 *N/mm* | Safe |

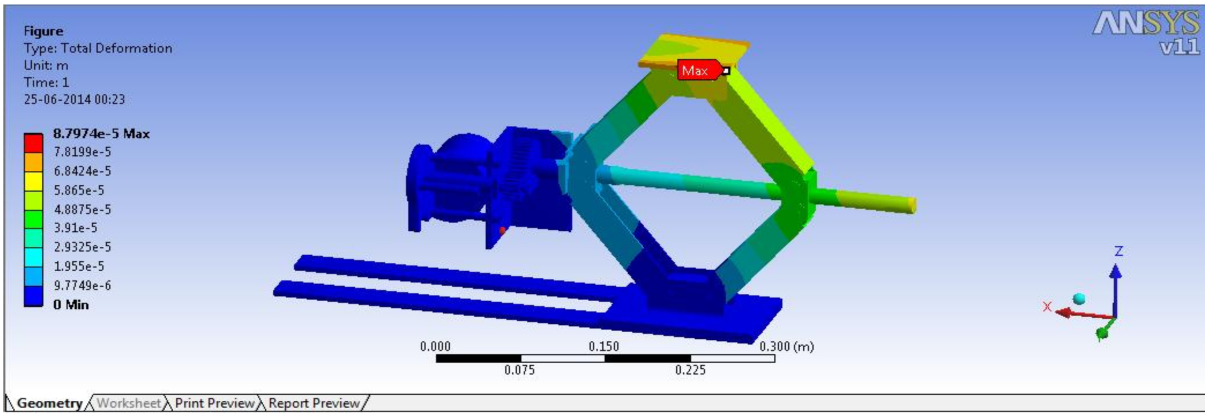


**Figure 7.1:** Deformation pattern for jack.

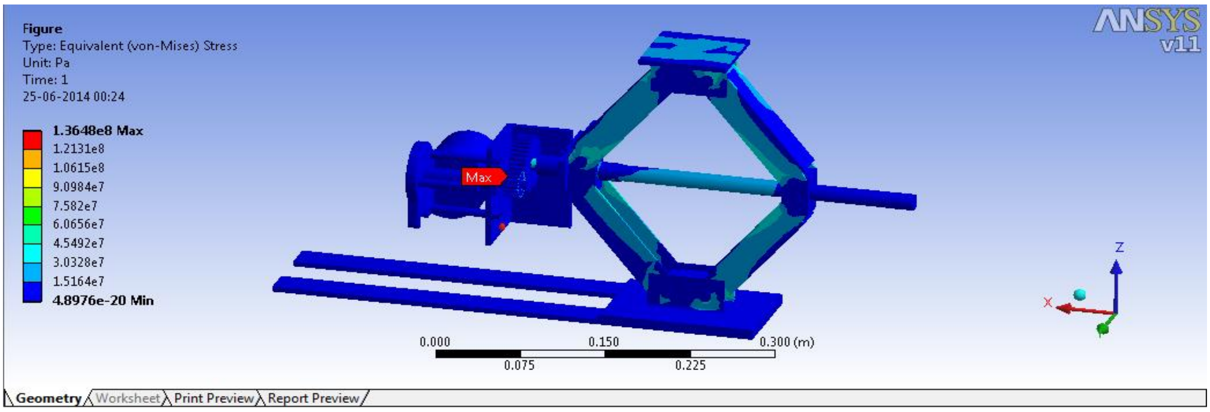


**Figure 7.2:** Von Miss Stress distribution.

## VIII. CONCLUSION

The existing jack was modified by making small alteration and making use of an electric motor to drive power screw, connecting gear with the pinion mounted on the motor shaft. The automobile 12V battery source operates prime mover (motor), to facile load lifting easier. The power screw is rotated through its gear when electrical power flows through it. The advantages of this jack is it 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. Considering all available car jacks in the market, this prototype can be improved by a few modifications on the features and design. The objectives are to design a car jack that is safe, reliable and able to raise and lower the level, to develop a car jack that is powered by internal car power and automated with buttons system.

Based on the testing and results from the analysis, it is considered safe to use Jack car work under certain specifications. Furthermore the torque supplied on the system is more than enough to lift a car weight around 1200 kg. There are certain weak point that can be improved based on gear, motor and design.

## REFERENCES

[1] Budyanas, G.R. and Nisbett, K.J., (2008), “Shigley’s Mechanical Engineering Design”, McGraw-Hill Companies $8_{th}$ Edition, pp 67—410, ISBN: 978 – 007 – 125763 – 3.

[2] Khurmi, R.S. and Gupta, J.K. (2005), “A Teaxtbook of Machine Design”, Eurasia Publication House (P.V.T) Ltd. $14_{th}$ Edition, 1230pp. ISBN: 81 – 219 – 2537 – 1.

[3] Rajput, R.K, (2007), “A Textbook of Manufacturing Technology”, Laxmi Publications, 1st Edition, 899pp. ISBN: 978 – 81 – 318 – 244 – 1.

[4] Lonon, E.M., East Orange, N.J. 1992. “Motorized Jack”, US Patent No. 5085407. Date of patent. Feb 4, 1992.

[5] BJC (Boston Jacks Corporation). 2008. Jacking beams manufacturer ISO 9002 (BS5750) 500. www.boston-ge.com

[6] Faehad Razzaghi, Douglasville, GA, US, “Appratus and Method for an Electric Jack”, US Patent No. US2007/0256526 A1. Date of patent. Nov 8, 2007.

[7] Khanna, O.P, Lal M, (1987), “A Textbook of Material Science and Metallurgy”, Dhanpat Rai, $5_{th}$ Edition.

[8] P.S.G. College of Technology. (2000), “Design Data Book”, Coimbatore Publication, $4_{th}$ Edition.

[9] Bhandari, V.B. (2009), “Design of Machine element”, Tata McGraw-Hill Education.

[10] Prof. Nitinchandra R. Patel, Sanketkumar Dalwadi, Vijay Thakor, Manish Bamaniya, “Design of Toggle Jack Considering Material Selection of Screw – Nut Combination”, International Journal of Innovative Research in Science, Engineering and Technology, (Vol. 2, Issue 5, May 2013) ISSN: 2319-8753.

[11] Vivek Karaveer, Ashish Mogrekar and T. Preman Reynold Joseph, “Modeling and Finite Element Analysis of Spur Gear” International Journal of Current Engineering and Technology (Vol.3, No.5, December 2013) ISSN: 2277 – 4106.