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Development of an Integrated Toggle Jack for Lifting Automobiles

Musa Nicholas Abodunrin Tosin Oladipo Sarafadeen

Department of Mechanical Engineering, Federal University of Technology, Minna, Niger State, Nigeria.

Abstract

In order to mitigate the problems associated with the use of a single jack and other lifting devices to raise cars completely off the ground to effect repairs, such as changing of tyres, maintenance of suspension system and intricate parts that are on and beneath the chassis, an integrated toggle jack was designed and fabricated for a load of 2.2tonnes. On performance evaluation it was able to raise cars of curb weights of 1.04, 1.24, 1.37, 1.48 and 1.58 tonnes to a height of 200mm in 1.1, 1.5, 1.2, 1.6 and 1.3minutes respectively.
Keywords: Toggle jack, screw, link, curb weight, buckling load

1. Introduction

Screw jacks and hydraulic jacks are used to raise cars for one to effect repairs such as changing or repairing of damaged tyres, repairing or replacing of some parts in the suspension system. Abdulmalik et al(2014) opined that motor vehicle repair and maintenance often require the lifting of the entire vehicle. According to Rana et al,(2012) in most of the garages, these vehicles are lifted by using screw jack. However, when the need arises whereby the car is to be raised completely off the ground, in order to have access to intricate parts of the car that are on and beneath the chassis, the aforementioned jacks are not feasible. Except four jacks are needed and operated differently to raise the car completely off the ground. This perhaps lead to time wasting and cumbersome in operation. However, Rout et al, (2014) reiterated that the need has long existed for an improved portable jack for automobiles.. Cranes can be used to raise cars off the ground but its robustness and cost discourages its use ,most especially when minor repairs are to be carried out. In most stationary automobile workshops, dug pits or raised platforms are put in place to enable repairs to be carried out. It suffices to say that when a car breaks down, and it requires the car to be completely raised off the ground before repairs can be done; it has to be towed to the workshop where the dug pits or raised platform are put in place. This will unavoidably increase the cost of repair of the car.

In order to forestall the problems associated with the use of the aforementioned machines and devices , to raise the cars or create access to the parts that are on and beneath the chassis, for repairs, the concepts of the design and fabrication of an integrated toggle jack for lifting automobiles arose in this research work. Car jacks usually use toggle advantage to allow a human to lift a vehicle by manual force alone(Patel et al, 2013). In this integrated toggle jack, manual force or effort is applied on one pair of the jack and power is transmitted to the other pair of the toggle jack by chain drive.

2. Materials and Method

2.1 Design theory and calculations

The following were considered for the design.

1. The integrated toggle jack and 2. Chain drive

2.2 Design of parts of the toggle jack

The toggle jack consists of the following parts,

1. Threaded screw, 2. Nut, 3. Pins and 4. Links

2.3 *Design of threaded screw*

Power screws are used to convert rotatory motion into translatory motion(Srivastav et al, 2013, Mounika and Priyanka, 2011). Although for the power screws, Udgirkar et al(2014) opined that ACME thread is most often used, it is not as efficient as the square thread because of the addition of friction due to the wedging action (Budynas and Nisbett, 2011). Square threads are mainly used for screw jacks(Lokhade et al, 2012). So square thread was considered for the design. Medium carbon steel (AISII1040) with yield strength of 350Mpa, ultimate strength of 520Mpa and ultimate shear strength of 343Mpa was selected for use.
The line diagram of the integrated toggle jack is depicted in Figure 1

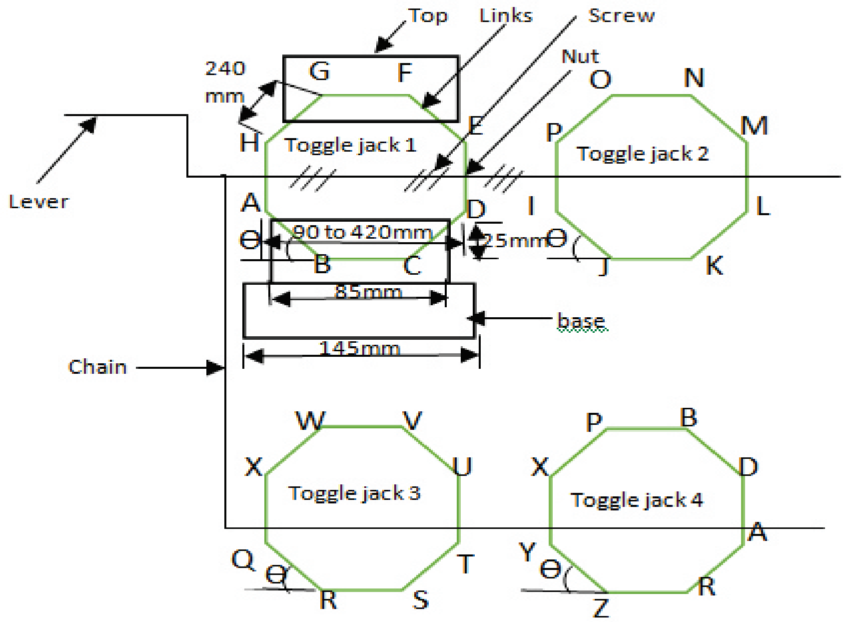

Figure 1. Line diagram of the integrated toggle jack

The four toggle jacks are identical and have the same parts and dimensions. The maximum load on the screws occurs when the four toggle jacks are in bottom most positions. Links AB, IJ, QR, and YZ are identical and their angles of inclinations to the horizontal is Θ, so the position of the link AB which is also the position of links IJ, QR, and YZ in the bottom position is depicted in figure 2

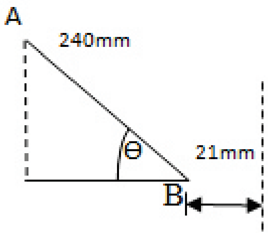

Figure 2. Position of link AB

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The angle of inclination of the links Θ can be found from the geometry of figure 2.2

$$\therefore \text{Cos}\theta = \frac{210-21}{40} = 0.7875$$

$\Theta = 38^{\circ}$

There are eight nuts in the integrated toggle jacks that is, each toggle jack has two nuts. So each nut carries $\frac{1}{8}$th of the total load on the integrated toggle jack. As a result of this , links AB, IJ, QR, and YZ are subjected to tension and the square threaded screws are under pull.(Yadav et al, 2014)

The pull in the screw is given as

$$F = \frac{W}{8\text{Tan}\theta} \qquad (1)$$

Where W is the maximum weight or curb weight of the car that can be lifted by the integrated toggle jack. So, it is taken to be 2.2tonnes.

$$\therefore F = \frac{2.2 \times 100 \times 9.81}{8\text{Tan } 38^{\circ}} = 3453\text{N}$$

The total pull in the square threaded rod as a result of nut and screw assembly in each of the toggle jack is

$$W_{t_1} = 2F \qquad (2)$$

$W_{t_1} = 2 \times 3453 = 6906\text{N}$

Allowable tensile stress, $\sigma_a = \frac{\sigma_u}{n}$ (3)

Where σ_u is the ultimate strength of the medium carbon steel = 520Mpa and n = factor of safety which is taken to be 5.

$\therefore\ \sigma_a = \frac{520}{5}$ = 104Mpa.

Allowable shear stress $\tau_a = \frac{\tau_u}{n}$

Where τ_u is the ultimate shear strength of the medium carbon steel = 343Mpa

$\therefore\ \tau_a = \frac{343}{5} = 68.6\text{Mpa}$

$\sigma_t = \frac{\text{Total pull on the screw}}{\text{Area of the screw}} = \frac{W_{t_1}}{\frac{\pi di^2}{4}}$ (4)

Where di = inner diameter of the screw.

$\therefore\ 104 \times 10^6 = \frac{6906 \times 4}{\pi di^2}$

$di = \sqrt{\frac{6906 \times 4}{104 \times 10^6 \times \pi}} = 0.0092\text{m} = 9.2\text{mm}$

So the inner diameter of each screw is taken to be 12mm in order to amount for the torsional shear stress experienced by the screw.

The outer diameter of the screw,

$do = di + p$ (5)

Where p is the pitch of the thread which is taken to be 4mm.

$\therefore\ do = 12 + 4 = 16\text{mm}$

The mean diameter of each screw,

$dm = do - \frac{p}{2}$ (6)

$\therefore\ dm = 16 - \frac{4}{2} = 14\text{mm.}$

The torque required to rotate the screw is given as

$\therefore\ T = W_{t_1}\text{Tan}(\propto + \emptyset)\frac{dm}{2}$ (7)

The helix angle $\propto = \text{Tan}^{-1}\frac{p}{\pi dm} = \text{Tan}^{-1}\frac{4}{\pi \times 14} = 5.2^\circ$

The friction angle $\emptyset = \text{Tan}^{-1}\mu$

Where μ = coefficient of friction = 0.15

$\therefore\ \emptyset = \text{Tan}^{-1}0.15 = 8.53^\circ$

$\therefore\ T = 6906 \times \text{Tan}\ (5.2^\circ + 8.53^\circ) \times \frac{0.014}{2} = 11.81\text{Nm.}$

This torque will cause shearing stress in addition to tensile stress.

$\because$ The shearing stress $\tau = \frac{16T}{\pi di^3} = \frac{16 \times 11.81}{\pi \times 0.012^3} = 34.8\ \text{Mpa}$

The direct tensile stress in the screw is given as $\sigma T = \frac{4W_{t_1}}{\pi di^2} = \frac{4 \times 6906}{\pi \times 0.012^2} = 61.06\text{Mpa}$

The maximum tensile stress

$$\sigma T_{max} = \frac{\sigma T}{2} + \frac{1}{2}\sqrt{\sigma T^2 + 4\tau^2} \qquad (8)$$

$$= \frac{61.06}{2} + \frac{1}{2}\sqrt{61.06^2 + 4 \times 34.84^2}$$

= 76.8Mpa.

The maximum shear stress

$$T_{max} = \frac{1}{2}\sqrt{\sigma T^2 + 4\tau^2} \qquad (9)$$

$$= \frac{1}{2}\sqrt{61.06^2 + 4 \times 34.84^2}$$

= 46.29Mpa.

Since the allowable tensile stress and shear stress are less than the allowable tensile stress of 104Mpa, and allowable shear stress of 68.6Mpa; the designed square threaded screw is safe.

2.4 Design of nut

Let n_1 = number of threads on the nut in contact with the screw. Based on the assumptions that the load w_{t1} is distributed evenly over the cross sectional area of the nut.

The nut bearing pressure between the thread in line the work of Patel et al (2013), is given as $P_B = \frac{wt1}{\pi d_m \times t_1 \times n_1}$ (10)

For the nut to be safe in tension, the inside diameter, di = 12mm and pitch, P = 4mm. The outside diameter =10+4 = 14mm, mean diameter $= \frac{1}{2}(14 + 10) = 12\text{mm}$. The thread thickness,

$t_1 = 0.5p = 0.5 \times 4 = 2\text{mm}$.

The number of threads in the nut, $n_1 = 5$.

The nut thickness $t_2 = n \times p = 5 \times 4 = 20\text{mm}$.

The width of the nut in line with the work of Khurmi and Gupta (2005) is taken as $b = 1.5d_o$

$\because b = 1.5 \times 16 = 24\text{mm}$,

$\therefore P_B = \frac{6906}{\pi \times 0.014 \times 0.002 \times 5} = 15.7\text{Mpa}$.

In order to forestall the movement of nut beyond 420mm, rings of 7mm thickness are fitted on both sides of the screw with the aid of set screws.

$\therefore$ Length of screw = $205 + t_2 + 2 \times 7 + L_c + L_o$ (11)

Where L_o is the length of the ends of the square threaded rod which is taken to be 35mm

L_c is the width of the car which is taken to be an average of 1500mm.

Length of the screw transmitting power to each of the toggle jack is 420 + 20 + 2 $\times$ 7 +1500 + 35 = 1989mm.

2.5 Determination of the length of lever

It is assumed that a force of 50N is applied by each operator at each end of the rod,

$\therefore$ the required length of the lever is $= \frac{T}{2 \times 50} = \frac{11.81}{2 \times 50} = 118.1\text{mm}$.

2.6 Design of Pin in Nuts

2.6.1 Determination of the diameter, d of the pin

The pins are in double shear

$\therefore$ the load on the pin, $F = 2\tau_a . \frac{\pi d^2}{4}$ (12)

$\therefore$ d = $\sqrt{\frac{4F}{2\pi\tau_s}} = \sqrt{\frac{4\times3453}{2\pi\times68.6\times10^6}}$ = 5.7$\times 10^{-3}$m =6mm

The diameter of the pin head = 1.5d = 1.5 $\times$ 6 = 9mm and thickness 4.5mm. Rings of 4.5mm thickness and 1.8mm split pins are used to keep the pins in the nut in position.

2.7 Design of links

As a result of load, the thickness of the links may buckle in vertical plane where the links are considered as hinged at both ends and in a plane at right angle to vertical plane where the links are considered as fixed at both ends.

The critical buckling and safe load are determined by the use of either Euler's or Johnson's equations. The conditions for use or applications are stated by Hall et al(2002) as follows

If $\frac{L}{K} > \sqrt{\frac{2C\pi E}{\sigma y}}$ use Euler equation but

If $\frac{L}{K} < \sqrt{\frac{2C\pi E}{\sigma y}}$ use Johnson's equation.

Where L is a constant depending on the end conditions

E is modulus of elasticity which is taken as 207Gpa

σy is yield point which is taken as 350Mpa.

The load on each link = $\frac{F}{2} = \frac{3453}{2} = 1726.5N$

The design load for the link taking buckling into consideration and assuming a factor of safety of 2 = 1726.5$\times 2 = 3453N$

2.7.1 Determination of critical buckling load on the links in the vertical plane where both ends are hinged

Let b_L = width of the link, t_L = thickness of the link.

Assuming that $b_L = 3.5t_L$ the cross sectional area of the link A = $3.5t_L^2$

Moment of inertia of the link $I = \frac{1}{12}t_L \times b^3{}_l = \frac{1}{12}t_L \times (3.5t_L)^3 = 3.57t_L^4$.

According to Hall et al (2002), the minimum radius of gyration, k for a rectangular section is given as

$$k = \frac{h\sqrt{3}}{6} \qquad (13)$$

Where h is the small dimension of the rectangle in this case, h = $b_L = 3.5t_L$

$\therefore k = \frac{3.5t_L\sqrt{3}}{6} = 1.01t_L$

On interpolation from different representative data provided by Hall et al(2002), that is between yield point of 415Mpa and 345Mpa and slenderness ratio $\frac{L}{K}$ of 109 and 121, the slenderness ratio of the links $\frac{L}{K}$ was found to be 108.

$\therefore \frac{L}{K} = 108$.

Recall that the length of the link L = 240mm and K = $1.01t_L$

$\therefore \frac{240}{1.01t_L} = 108$

$108 \times 1.01t_L = 240$

$t_L = \frac{240}{108\times1.01} = 2.2mm$ Say 3mm.

$\therefore$the width of the link $b_L = 3.5 \times 3$=10.50mm say 11mm.

The cross sectional area of the link = $3.5 \times 3^2 = 31.50mm^2$

Computing the magnitude of $\sqrt{\frac{2C\pi E}{бy}}$

$$\therefore \sqrt{\frac{2C\pi E}{бy}} = \sqrt{\frac{2\times1\times\pi\times207\times10^{9}}{350\times10^{6}}} = 60.96$$

Since $\frac{L}{K} > \sqrt{\frac{2L\pi E}{бy}}$ i.e (108> 60.96)

Euler's equation is applicable

$\because$ the critical buckling load $F_{CR} = \frac{C\pi^2 EA}{(\frac{L}{K})^2}$ (14)

$$F_{CR} = \frac{1\times\pi^2\times207\times10^{9}\times3.15\times10^{-5}}{108^2} = 5517.38N$$

Since the critical buckling load is more than the designed load, the link is safe for buckling in the vertical plane.

2.7.2 Determination of the critical buckling load on the links in a plane at right angles to the vertical plane with the links fixed at both ends.

The moment of inertia of the cross section of the link

$$I = \frac{1}{12} b_L \times t^3{}_1 = \frac{1}{12} \times 0.011 \times 0.003^3 = 2.48\times 10^{-11}\ m^4.$$

Radius of gyration, $K = \sqrt{\frac{I}{A}} = \sqrt{\frac{2.48\times10^{-11}}{3.15\times10^{-5}}} = 8.87\times10^{-4}m$

Equivalent length of the link

$$L_{eq} = \frac{L}{2} = \frac{240}{2} = 120mm = 0.12m$$

Comparing the magnitude of $\frac{L}{K}$ and $\sqrt{\frac{2C\pi E}{бy}}$

$$\therefore \frac{L}{K} = \frac{L_{eq}}{K} = \frac{0.12}{8.87\times10^{-4}} = 135$$

$$\sqrt{\frac{2C\pi E}{бy}} = \sqrt{\frac{2\times4\times\pi\times207\times10^{9}}{350\times10^{6}}} = 121.9$$

Since $\frac{L}{K} > \sqrt{\frac{2C\pi E}{бy}}$, Euler's equation is applicable.

Since critical buckling load $F_{CR} = \frac{C\pi^2 EA}{\left(\frac{L}{K}\right)^2} = 14124N$

Since the critical buckling load is also more than the design load of 3453N, the link is safe for buckling in a plane perpendicular to the vertical plane.

Therefore, we may take the thickness of the link t_L = 3mm, and the width of the link b_L = 11mm.

2.8 Design of the chain drive

The chain drive actuates a pair of toggle jack from the other pair of the toggle jack where effort of the operator of (50×2) = 100N is applied and transmits torque of 11.81Nm.

The speed N_1 and the diameter d_s of the driving sprockets where effort is applied, are taken to be 30rpm and 80mm respectively. The speed, N_2 diameter d_n, of the driven sprockets are also taken to be, 30rpm and 80mm respectively.

The rated power P_R =TW (15)

$$\therefore P_R = 11.81 \times \frac{2\pi\times30}{60} = 37.1W$$

The velocity ratio of chain drive, $VR = \frac{N_1}{N_2} = \frac{30}{30} = 1.$

The number of teeth on the driving sprocket and driven sprocket for a velocity ratio of 1 is 31 each (Khurmi and Gupta, 2005).

Design power P_D is given as, $P_D = P_R \times K_S$ (16)

Where K_S is the service factor.

$K_S = K_1 \times K_2 \times K_3$ (17)

Where K_1 is the load factor =1.25for variable load with mild shock,
K_2 is the lubricating factor =1 for drop lubrication,
K_3 is the rating factor =1 for maximum of 8hours per day.

$\therefore K_s = 1.25 \times 1 \times 1 = 1.25$

$\therefore P_D = 37.1 \times 1.25 = 46.38W.$

For driving sprocket speed up to 100rpm, chain number 6 with the transmission of power of 250W per strand is selected with the following specifications stated by Khurmi and Gupta(2005) as
Pitch, P_h = 9.525mm,
Roller diameter, d_e = 6.35mm
Minimum width of roller, W_r =5.72mm
Breaking load W_B = 8.9KN
The pitch circle diameter of the sprockets

$$d_p = P_h \operatorname{cosec}\left[\frac{180}{Z_1}\right] = 9.525\operatorname{cosec}\left(\frac{180}{31}\right) = 94mm$$

pitch line velocity of the sprocket

$$V_1 = \frac{\pi d_p N_1}{60} = \frac{\pi \times 0.094 \times 30}{60} = 0.148m/s$$

Load on the chain, $W_n = \frac{P_R}{V_1} = \frac{37.1}{0.148} = 250.68N.$

The center distance between the sprockets is taken to be 35 times the pitch

$\therefore C = 35p = 35 \times 9.525 = 333.38mm$

In order to cater for the initial sag, the center distance is reduced by 5mm,

$\therefore$ The center distance is = 333.38 – 5 = 328.38 = 0.33m

The number of links of chain $N_L = \frac{Z_1+Z_2}{2} + \frac{2C}{P_h} + \left[\frac{Z_2-Z_1}{2\pi}\right]^2 \frac{P_h}{C}$ (18)

Where Z_1 and Z_2 are the number of teeth in the driving and driven sprocket.
In this case, $Z_1 = Z_2 = 31$ and C = center distance

$$\therefore N_L = \frac{31+31}{2} + \frac{2\times 328.38}{9.52} + \left(\frac{31-31}{2\pi}\right)^2 . \frac{9.525}{328.38} = 99.95mm \text{ Say } 100 \text{ mm}$$

$\therefore$Length of chain, $L = N_L . P_h = 100 \times 9.525 = 952.5mm = 0.95m.$

Based on the design theory and calculations materials were selected and used to fabricate the integrated toggle jack for lifting automobiles. The components and assembly drawings are shown in Figures 3 and 4 respectively. It should be noted that all dimensions in Figure 3 are in millimeters.

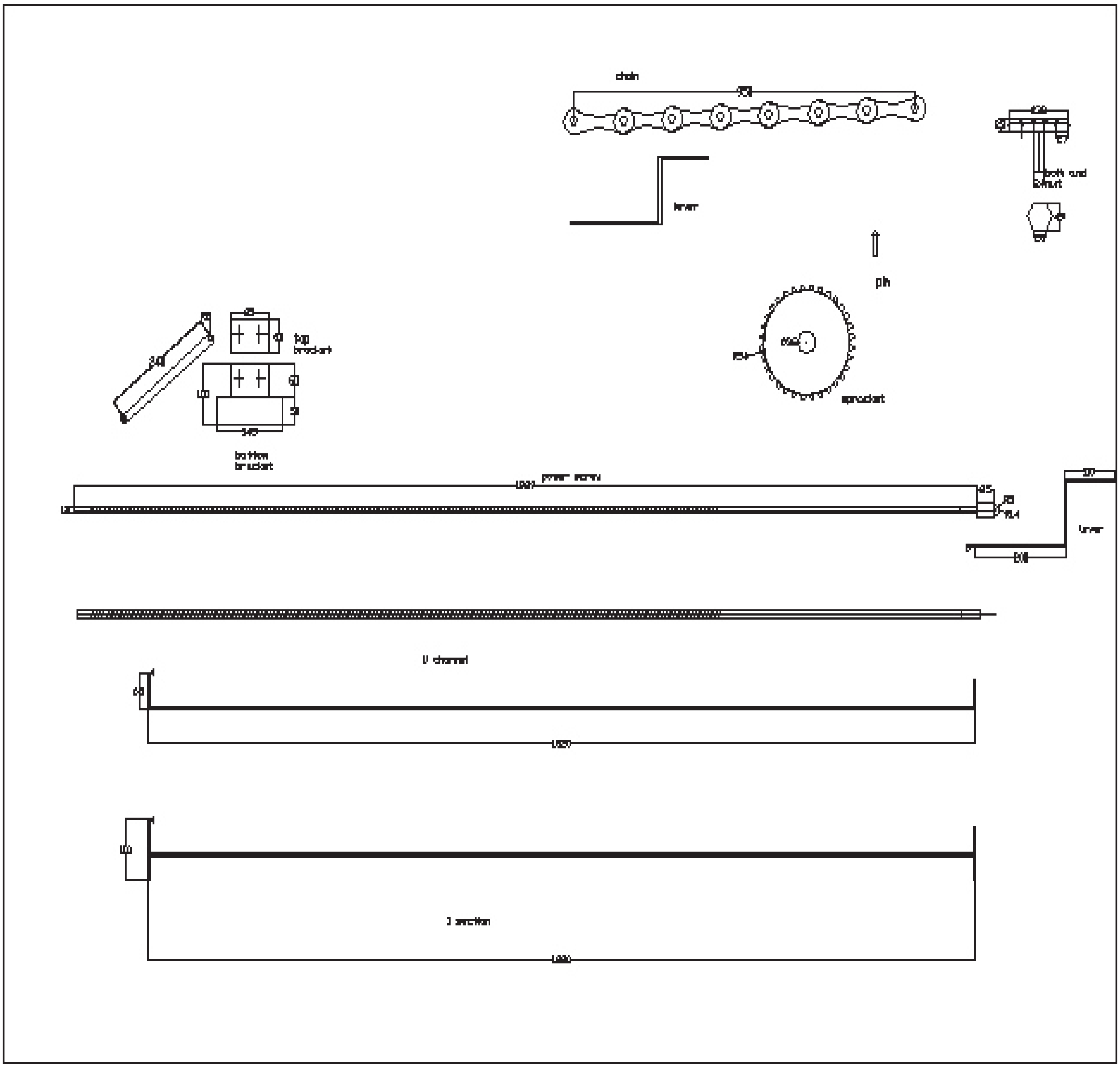

Figure 3. Components drawing of the integrated Toggle jack

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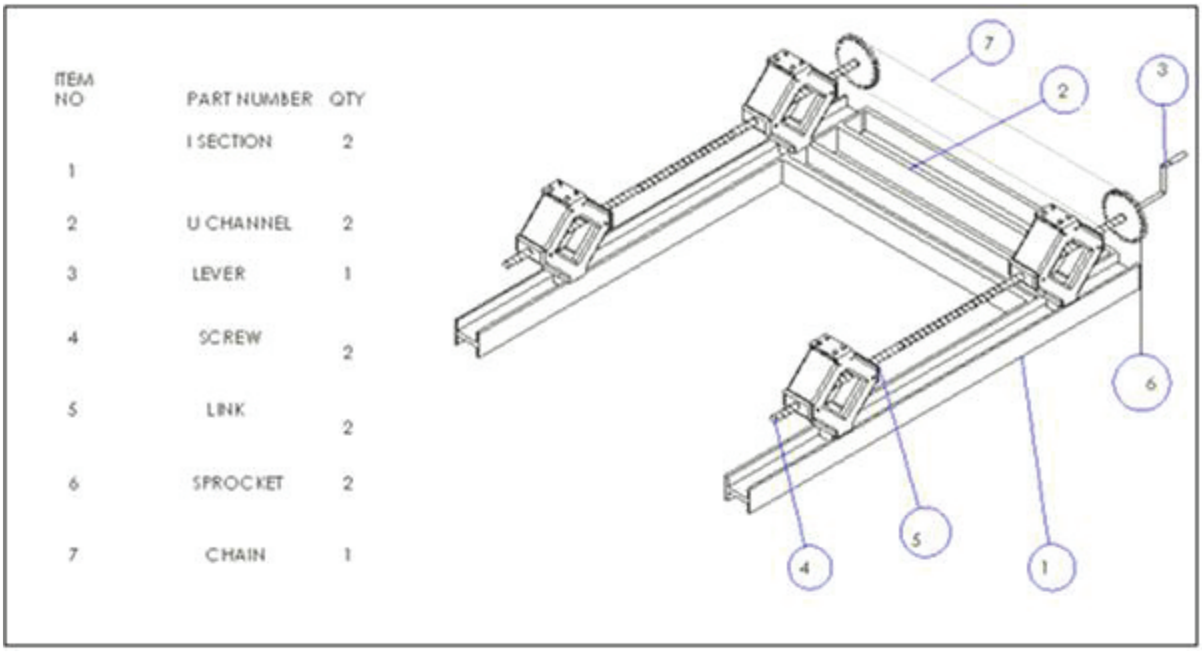

Figure 4. Assembly drawing of the integrated Toggle jack

2.9 Performance Evaluation

The following were used to carry out the performance evaluation,

1. Stop watch,
2. A meter rule,
3. Toyota Yaris2007 model with curb weight of 1.04tonne, Honda civic 2003 model with curb weight of 1.24tonne, Mercedes Benz 190, 1987 model with curb weight of 1.37tonne, Toyota Camry 2004 model with curb weight of 1.48 and Toyota Avalon 2006 model with curb weight of 1.58tonne

The design and fabricated integrated toggle jack was fitted with the chassis of the car; effort was applied by turning the lever steadily, the stop watch was switched on immediately the turning of the lever commenced. The number of turns was noted and when the cars were individually raised to a height of 200mm, the stop watch was switched off and the time taken was noted.

3.0 Results and Discussion

The result of the performance evaluation of the designed and fabricated integrated toggle jack are shown in Table 1.

Table 1. Time taken to lift cars of various curb weights by the integrated toggle jack

Type of car	Model	Curb weight(Tonnes)	Time taken(Minutes)	Number of turns of the lever
Toyota Yaris	2007	1.04	1.1	31
Honda Civic	2003	1.24	1.5	31
Mercedes Benz190	1987	1.37	1.2	31
Toyota Camry	2004	1.48	1.6	31
Toyota Avalon	2006	1.58	1.3	31

It can be seen from table 1 that the time taken by the operator to raise or lift the car to a height of 200mm vary from one car curb weight to another. It took 1.6 minutes to raise Toyota Camry 2004 model of curb weight of 1.48tonne and took the least time of 1.1minutes to lift Toyota Yaris 2007 model of curb weight 1.04tonne. It is evident in table 1 that the time taken to raise the car is independent of the curb weight of the cars. However, the effort applied on the lever has influence on the time taken to lift the cars

4.0 Conclusion

The use of dug pits or permanent raised platform for cars in order to effect repairs can be replaced with the use of an integrated toggle jack. Its compatibility, ease of dismantling, and assembly make it possible to be moved from one place to another for use. Though it is designed for cars, it can be adopted for use in raising or lifting other loads not exceeding the designed load capacity of the integrated toggle jack.

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