DEVELOPMENT OF A FOOT OPERATED HYDRAULIC LIFTER FOR AUTOMOBILE WORKSHOPS

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ABSTRACT

This work looks at the possibility of controlling a hydraulic jack using a foot operated linkage mechanism. This modification greatly reduces stress and allows for comfort during usage in an automobile workshop, and the effort required is also very minimal. The test results showed that this foot operated hydraulic lifter for automobile workshops performed much more efficiently than the hand operated hydraulic lifter. The effort was also greatly reduced from 2050.4N to 136.7N. It also incorporates compression springs which allows for flexing under load to transmit the effect of the load to the floor.

Key words: Controlling, hydraulic jack, foot operated, linkage mechanism, modification.

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INTRODUCTION

A jack is a portable device used for weight lifting. Mechanical jacks, such as car jacks and house jacks, lift heavy equipment and are rated based on their lifting capacity. Hydraulic jacks tend to be stronger and can lift heavier loads higher, and include bottle jacks and floor jacks. Hydraulic jacks depend on force generated by pressure. Essentially, if two cylinders (a large and a small one) are connected and force is applied to one cylinder, equal pressure is generated in both cylinders [1]. However, because one cylinder has a larger area, the force the larger cylinder produces will be higher, although the pressure in the two cylinders will remain the same.Hydraulic jacks depend on this basic principle to lift heavy loads, they use pump plungers to move oil through two cylinders. The plunger is first drawn back, which opens the suction valve ball within and draws oil into the pump chamber. As the plunger is pushed forward, the oil moves through an external discharge check valve into the cylinder chamber, and the suction valve closes, which results in pressure building within the cylinder [1].

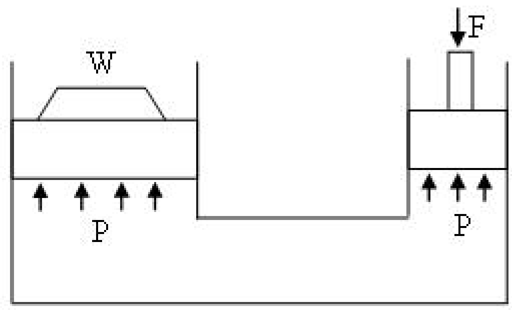


Figure 1: Principle of Hydraulic Jack

Most of the commonly available jacks are hand operated and modifications of such designs to become foot operated will be of great benefit. Hydraulic jacks, automobile brakes and dental chairs work on the basis of Pascal's principle. Basically, the principle states that the pressure in a closed container is the same at all points. A hydraulic jack is simply two cylinders connected as described in Figure 1. This work looks at the possibility of controlling a hydraulic jack using a foot operated linkage mechanism. This modification will greatly reduce stress and allow for comfort during usage and the effort required is also greatly reduced.

DESIGN ANALYSIS AND CALCULATIONS

Determination of the Force on the Plunger

Figure 2 shows a bottle type hydraulic jack.

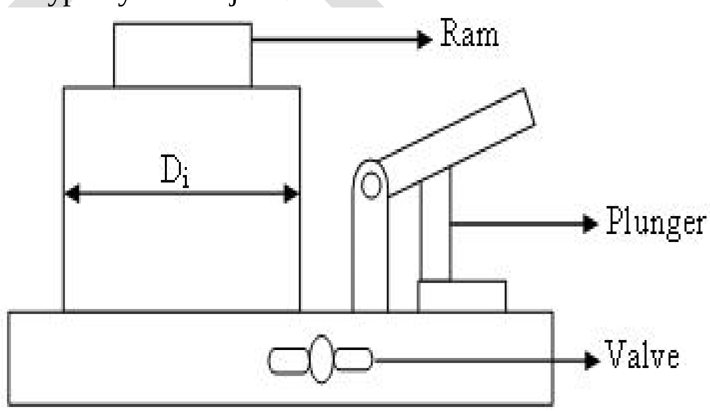


Figure 2: Bottle Type Hydraulic Jack

The mass on the ram M_R may be obtained from

$$M_R = \frac{D_R}{D_i} \times C_{max} \tag{1}$$

Where,
D_R is the diameter of ram = 28.0mm
D_i is the diameter of big cylinder = 62.0mm
C_{max} is the maximum capacity of the hydraulic cylinder = 3000kg

From equation (1), we get

$$M_R = \frac{28}{62} \times 3000kg = 1354.8kg$$

The weight on the ram W_R may be obtained from

$$W_R = M_R g \qquad (2)$$

Where,
g is the acceleration due to gravity = 9.81m/s^2

From equation (2), we get

$$W_R = 1354.8 \times 9.81 = 13290.6N$$

The cross sectional area of the ram A_R is obtained from

$$A_R = \frac{\pi \times {D_R}^2}{4} \qquad (3)$$

From equation (3), we get

$$A_R = \frac{\pi \times 28^2}{4} = 615.8mm^2$$

The cross sectional area of the plunger A_P is obtained from

$$A_P = \frac{\pi \times D_P}{4} \qquad (4)$$

Where,
$D_P$is the diameter of plunger = 11.0mm

From equation (4), we get

$$A_P = \frac{\pi \times 11^2}{4} = 95.0mm^2$$

From Pascal's principle, we can deduce that the pressure of plunger equals that of the ram. The required force on the plunger $F_P$may therefore be obtained as follows:

$$\frac{F_P}{A_P} = \frac{W_R}{A_R} \Rightarrow F_P = \frac{W_R \times A_P}{A_R} = \frac{13290.6 \times 95.0}{615.8} = 2050.4N \qquad (5)$$

It then means that we require a force of 2050.4N on the plunger to lift a weight of 13290.6N on the ram.

Determination of the Forces on the Members of the Mechanism

Figure 3 shows the jack linkage mechanism's free body diagram.

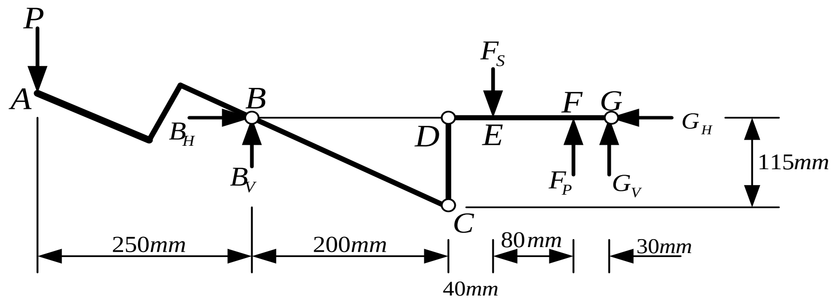


Figure 3: The jack linkage mechanism's free body diagram

Considering member DG in Figure 3 and summing forces vertically, we get

$$F_S = F_P + G_V \tag{6}$$

Taking moment about G, we get

$$110 \times F_S = 30 \times F_P \Rightarrow F_S = \frac{30 \times F_P}{110} = 559.2N \tag{7}$$

From equation (6), we get

$$G_V = F_S - F_P = -1491.2N \tag{8}$$

Considering member AC in Figure 3 and Summing vertical forces, we get

$$\sum F_V = 0 \Rightarrow B_V = P + F_C\,. \tag{9}$$

Where,

$$F_C = \frac{30 \times F_P}{150} = 410.1N \tag{10}$$

FC comes into play when P is disengaged. From equation (9), we get

$$B_V = F_C = 410.1N \tag{11}$$

Considering Figure 3 and taking moment about G, we get

$$600 \times P + 110 \times F_S = 350 \times B_V + 30 \times F_P \tag{12}$$

When P is engaged, F_P=0. From equation (12), we get

$$P = \frac{350 \times B_V - 110 \times F_S}{600} = 136.7N \tag{13}$$

Considering member BC and taking moment about C, we get

$$115 \times B_H + 200 \times B_V = 0 \Rightarrow B_H = -\frac{200 \times B_V}{115} = -713.2N \tag{14}$$

From Figure 3, summing forces horizontally, we get

$$G_H = B_H = -713.2N \tag{15}$$

Determination of the Diameter of the Fulcrum Pin

The fulcrum pin diameter d may be obtained from the following expression [2]:

$$d = \sqrt[3]{\frac{T \times 16}{\pi \times \tau}} = \sqrt[3]{\frac{P \times 250 \times 16}{\pi \times 130}} = 11.0mm \quad (16)$$

Where,
T is the torque

Determination of the Bending Moment of the Wheel Axle

Figure 4 shows the wheel axle load diagram.

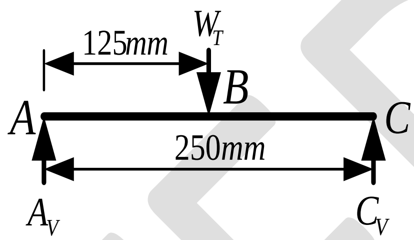


Figure 4: The Wheel Axle Load Diagram

$$W_T = M_T \times g \quad (17)$$

Where,
M_T is the total mass on the axle

From equation (17),

$$M_T = M_R + M_B + M_F = 1354.8 + 8 + 17 = 1379.8kg$$

Where,
M_B is the mass of bottle jack
M_F is the mass of frames

From equation (17), we get

$$W_T = M_T \times g = 13535.8N \quad (18)$$

From Figure 4, summing forces vertically, wet get

$$A_V + C_V = W_T = 13535.8N \quad (19)$$

Taking moment about point B

$$250 \times A_V = 125 \times W_T \Rightarrow A_V = \frac{125 \times W_T}{250} = 6767.9N \quad (20)$$

From equation (19), we get

$$C_V = W_T - A_V = 6767.9N \quad (21)$$

The maximum bending moment occurs at B with a value of

$$M_b = A_V \times 125 = 845987.5Nmm$$

Determination of the Wheel Axle Diameter

From bending equation [3],

$$\frac{M}{I}=\frac{\sigma_b}{y}\Rightarrow \sigma_b=\frac{32\times M_b}{\pi\times D_A^{\;3}}\Rightarrow D_A=\sqrt[3]{\frac{32\times M_b}{\pi\times\sigma_b}} \qquad (22)$$

Where,

D_A is the diameter of the axle

M_b is the bending moment

σ_b is the bending stress

I is the moment of inertia

$y=\frac{D_A}{2}$ is the distance from the neutral axis

From equation (22), we get

$$D_A=\sqrt[3]{\frac{32\times 845987.5}{\pi\times 870}}=21.5mm$$

Determination of the Return Spring Parameters

A spring is defined as an elastic body, whose function is to distort when loaded and to recover its original shape when the load is removed [4]. The return spring parameters are determined as follows:

Return Spring index

In practice spring index varies from 6 to 10. The return spring index may be obtained as follows:

$$C=\frac{D_C}{D_W}=\frac{24}{3}=8$$

Where,

D_C is the mean coil diameter= 24.0mm

D_W is the spring wire diameter = 3.0mm

Number of Turns of Coil for the Return Spring

The number of turns of coil (n) for the return spring may be obtained as follows[5]:

$$n=\frac{G\times D_W\times\delta}{8\times W\times C^3}=\frac{84\times 10^3\times 2.8\times 115}{8\times 559.2\times 8^3}=13$$

Where,

G is the modulus of rigidity of the return spring material = 84kN/mm^2

δ is the maximum deflection of the return spring = 115mm

W is the axial load on the return spring = 559.2N

Number of turns of Coils for square end spring

The number of turns of Coils for square end spring (n') may be obtained as follows:

$$n'=n+2=13+2=15\ turns$$

Free Length of the Return Spring

Free length of spring $L_F$may be obtained as follows [4]:

$$L_F = n \times D_W + (n-1) \times 1mm = 13 \times 3 + (13-1) \times 1 = 51.0mm$$

Return Spring Stiffness Constant

The return spring stiffness constant (k) may be obtained as follows:

$$k = \frac{W}{\delta} = \frac{559.2}{115} = 4.9 \text{ N/mm}$$

Determination of the Compression Spring Parameters

The compression spring parameters are determined as follows:

Compression Spring Index

Spring index (C) may be obtained as follows [4]:

$$C = \frac{D_C}{D_W} = \frac{32}{4} = 8$$

Where,

D_C is the mean coil diameter = 32.0mm

D_W is the wire diameter = 4.0mm

Number of turns of Coilsfor the Compression Spring

The number of turns of coils (n)for the compression spring may be obtained as follows:

$$n = \frac{G \times D_W \times \delta}{8 \times W \times C^3} = \frac{84 \times 10^3 \times 4 \times 40}{8 \times 245.3 \times 8^3} = 14\,turns$$

Where,

G is the modulus of rigidity of the compression spring material = 84kN/mm^2

δ is the maximum deflection of the compression spring = 40mm

W is the axial load on the compression spring = 245.3N

Number of Turns of Coils for Square End Compression Spring

Number of turns of coils for square ends compression spring may be obtained as follows:

$$n' = n + 2 = 14 + 2 = 16\,turns$$

Solid Length of the Compression Spring

Solid length L_S of the compression spring may be obtained as follows:

$$L_S = n' \times D_W = 16 \times 4 = 64mm$$

Compression Spring Stiffness Constant k

Spring stiffness may be obtained as follows [4]:

$$k = \frac{W}{\delta} = \frac{245.3}{40} = 6.1 \text{ N/mm}$$

Wahl's Stress Factor K

Wahl's stress factor may be obtained as follows[6]:

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4\times 8.3-1}{4\times 8.3-4} + \frac{0.615}{8.3} = 1.2$$

Maximum Shear Stress of the Compression Spring

Maximum shear stress of spring may be obtained as follows [6]:

$$\tau_S = \frac{8\times K\times W\times C}{\pi\times D_W^2} = \frac{8\times 1.2\times 245.3\times 8}{\pi\times 4^2} = 374.8\ \text{N/mm}^2$$

Strain Energy Stored in the Spring

Strain energy U stored in the spring may be obtained as follows:

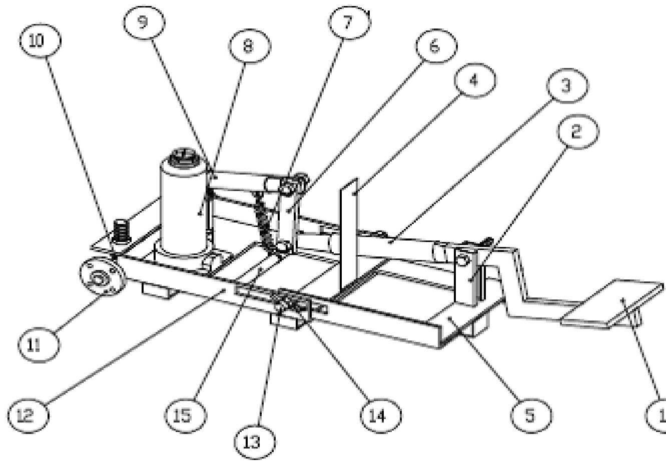
$$U = \frac{1}{2}\times W\times\delta = \frac{1}{2}\times 245.3\times 0.04 = 4.9 Nm$$

TESTING

The hydraulic lifter was tested using different kinds of automobiles. For effective performance evaluation, the following was ensured:

1) The vehicle selected was parked properly
2) The release valve of the hydraulic lifter was securely locked
3) The hydraulic lifter was placed properly under the vehicle
4) In case of vehicles with height, the ram of the hydraulic lifter was unscrewed from the piston up to the point of carriage
5) The foot pedal was used to actuate the pump to raise the ram so as to establish a firm grip with the vehicle
6) Foot pedaling was continued, noting the number of strokes and the time taken to lift the vehicle tyre off the ground

Figure 5 shows the foot operated hydraulic jack. Table1 and Table 2 shows the test results of the performance evaluation of the foot and hand operated hydraulic lifterrespectively.The tablesshow the weight of each vehicle considered, the time taken to lift the vehicle tyre off the ground and the number of strokes. From tables 1 and 2, the number of strokes taken to lift the vehicle tyre off the ground is the same, but the time taken was different. It was also discovered that the stresses associated with the hand operated hydraulic lifter are much more when compared with the foot operated hydraulic lifter. This clearly shows that the foot operated hydraulic lifter is more efficient to use and the effort required is as low as 136.7N (13.9kg).



Part No.	Part Name	Part No.	Part Name
1	Foot Pedal	9	Hydraulic Jack Handle
2	Foot Pedal Support	10	Compression Spring
3	Foot Pedal Bar	11	Metallic Wheel
4	Foot Pedal Guide	12	Compression Spring Frame
5	Foot Pedal Frame	13	Bolt
6	Connecting Bar	14	T-Nut
7	Return Spring	15	Rear Pipe
8	Hydraulic Jack		

Figure 5: Foot Operated Hydraulic Jack

Table 1: Performance evaluation of the foot operated hydraulic lifter

S/N	Vehicle (Salon Car)	Maximum Weight (kg)	Time Taken (Seconds)	Number of Strokes
1.	Toyota corolla	1,385	16.12	24
2.	Mazda	1,450	31.75	28
3.	Honda Accord	1, 660	39.22	40
4.	Toyota Carina E	1, 665	41.28	43
5.	Peugeot 505	1,665	40.14	42

Table 2: Performance evaluation of hand operated hydraulic lifter

S/N	Vehicle (Salon Car)	Maximum Weight (kg)	Time Taken (Seconds)	Number of Strokes
1.	Toyota corolla	1,385	18.48	24
2.	Mazda	1,450	38.14	28
3.	Honda Accord	1, 660	41.25	40
4.	Toyota Carina E	1, 665	43.36	43
5.	Peugeot 505	1,665	42.44	42

CONCLUSION

The existing bottle hydraulic jack is a very common device used in this part of the world used by different categories of people. The most common application is in the raising of an automobile to facilitate the replacement of a flat tyre. But the problems associated with this type of jack include: hand use which is laborious and very stressful in operation, spine and arm aching e.t.c. Some other existing jacks such as floor jacks have so far been made with a rigid frame and solid wheels, without compensation for flexing under load to transmit the effect to the floor. This results in a construction that is cumbersome to use.This work has addressed the above stated shortcomings. The improvements are in terms of lifting effectiveness, easyusage,occupied space, easy transportation, easy maintenance, and operation by foot with an effort as low as 136.7N, which is much more convenient than hand operated. In terms of cost, it is affordable to the common users of hydraulic jacks.

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