INTERNATIONAL JOURNAL OF INNOVATIVE RESEARCH & DEVELOPMENT

***ISSN: 2278 – 0211***

# Design Of An Automobile Tilting Device Using Hydraulic System

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***Abstract:***

*This paper analyzes the design of an automobile tilting device for tilting of automobile bodies at mechanical workshops, when the need for maintenance arises; especially welding to be performed beneath them is always achieved using an existing traditional tilting technique. The traditional technique involves manual lifting and positioning of the automobile body with one of its sides on the work floor, after which supporting implements such as wooden bars, metal rods, worn-out tyres and pestles are used to keep the tilted bodies at the desired position. This technique has always been resulting to higher worker fatigue and much risk of worker injury, endangering the lives of mechanics. The designing and selection of the automobile tilting device components were based on the application of machine design and hydraulic principles and laws. The height and the degree of tilt of the device are controlled by the position of the control lever of the direction control valve regulating the fluid flow into the tilting and lifting cylinders. This hydraulic automobile tilting device is designed to provide a more effective and efficient technique for tilting and supporting tilted automobile bodies, thereby minimizing and eliminating the risks involved when employing the existing traditional technique. This proposed hydraulic tilting device will therefore provide an effective technique for tilting and supporting class 1 and 2 category automobile bodies thereby eliminating the existing stresses and risks involved with the existing traditional technique.*

**keywords:** *automobile, tilting device, hydraulic, fatigue, fluid flow, lifting cylinder.*

## 1.Introduction

Screws Application is used in the elevation of vehicles or objects. The operation of the screw jack is such that it comprises a handle for driving a bolt element (Lead Screw) manually so as to adjust the height of the Jack to elevate a vehicle or the object. (Akinwonmi and Mohammed, 2012). In the case of a screw jack, a small force applied in the horizontal plane is used to raise or lower large load. [Khurmi and Gupta, 2005].

In olden times, thousand of slaves had to be arranged whenever a heavy load had to be lifted or dragged. Even today, in the absence of a suitable device, many people have to be arranged to lift a motor car so that its tyres can be changed (Khurmi, 2009). Tilting of automobile bodies by mechanics, in the act of performing maintenance, especially welding operations on worn-out components and joints beneath automobiles, has always been risky, time consuming and labour intensive. It has been the customary practice of mechanics in using wood, pestles, worn-out tyres and metal rods to support automobile tilted and positioned by their own efforts. These practices have been yielding worker fatigue and much risk of worker injury as a result of poor support provided to keep the positioned automobile body stationary, thereby endangering the lives of mechanics as positioned bodies sometimes fall back on them during the on-going maintenance processes and welding operation. The requirement of more hands (labour) during positioning or tilting the automobile body on one of its sides is another problem faced, with much time wasted, as mechanics await more hands to tilt the automobile body. There are also the occurrences of side mirrors breaking and scratching of the automobile body side by abrasive materials from the work floor or supporting materials in contact with it after positioning or tilting. The traditional tilting technique is very tedious and requires more hands in order to effect the tilting of the automobile body to the desired position. Also, the lives of mechanics are endangered since the supported tilted automobile body can fall back in case of the supporting material failure. Due to large numbers of examples of compound stresses met with in engineering practice, the cause of “failure” or permanent set under such conditions has attracted considerable attention [Rajput, R.K. 2010]. The application of stress to any material will lead to the production of elastic and / or plastic strain and if the stress is increased progressively, fracture will ultimately occur. [Raymond, 1990].

The idea of applying ergonomic principles which is all about reducing worker fatigue and the risk of worker injury when carrying out maintenance and welding processes beneath automobile at the workshop therefore necessitates the design of this tilting device. The objective of this work is therefore to provide a suitable hydraulic device for effectively tilting and supporting of automobile bodies, minimizing and eliminating wasted time, efforts and the risks involved during and after tilting at mechanical workshops.

## 2.Existing Traditional Tilting Technique

The traditional tilting technique has been the only convenient method employed for tilting automobile bodies at mechanical workshops. This technique involves the manual tilting of automobile by at least five strong men. The automobile is drained of oil and its carburetor removed to prevent oil, water and fuel spillage during and after tilting. Worn-out tyres are used as the main supporting materials on the floor while wooden implements, metal bars and pestles, for example are also used to keep the tilted automobile body at the desired position. In the process of tilting, one person stands at one side of the automobile body to ensure that his side falls correctly on the floor support material while the remaining hands lift the other side, positioning the automobile as shown in Fig. 1.

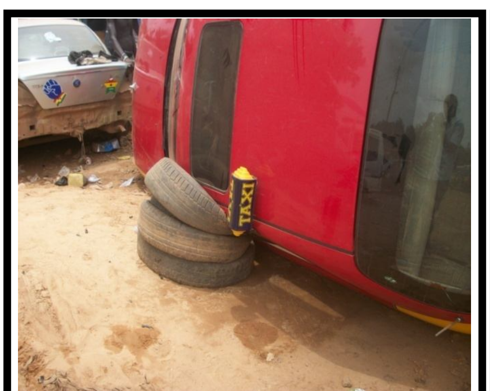


*Figure 1a*

*Figure 1b*

*Figure 1 An Automobile Body Supported With Worn-Out Tyres (A) And A Wooden Bar (B) After Tilting.*

When the desired tilting position or angle is reached, one of the four mechanics effecting the tilting operation quickly place a wooden bar, or any other suitable supporting material to keep the automobile at that position. The base of the automobile chassis is now accessible for the maintenance operation to be performed.

## 3.Materials And Method

Automobile workshops were visited at Tarkwa in Ghana to study the existing ways that vehicles are being lifted in May, 2012 in order to facilitate this design. The design is carried out based on application of machine design principles and theories and application basic hydraulic principles.

### *3.1.Description Of Proposed Tilting Device*

The proposed tilting device mainly consists of;

A main frame structure consisting of a bed on which the vehicle is placed after tilting.

A tilting platform consisting of a laminated faceplate and a fork assembly.

Four folding support steel bars in a criss-cross 'X' pattern for vertically adjusting the tilting platform.

A two position-three way directional control valve mechanism for regulating pump output flow.

Two three position-four way directional control valve mechanism for directing the pump output flow into the tilting and height adjustment mechanism lines respectively.

A pressure relief valve for stepping down the maximum system pressure to that required by the tilting actuator.

Hydraulic actuators consisting three double acting hydraulic cylinders and piston arrangement, two for effecting the tilting motion and one for the height adjusting mechanism.

A screw driven mechanism for providing additional support to the automobile after tilting.

Hydraulic Power Unit (HPU) consisting of an AC motor driving a hydrostatic pump.

A hydraulic reservoir and conduit for fluid storage and flow channels.

### *3.2.Mode Of Operation*

The proposed tilting device as described above is permanently installed at a specific location at the workshop where the automobile will be pushed and positioned on it. The height adjusting mechanism of the device is installed in-ground (within the workshop floor) leveling the top of the fork prongs with the workshop floor, enhancing easy positioning of the automobile onto the tilting platform. The automobile to be tilted is pushed and positioned with its side being parallel to the laminated surface of the tilting platform.

After the automobile has been properly positioned on the platform, the motor is activated by switching on the power supply. The motion of the motor is then transmitted to the pump through couplings thereby resulting to the admission of fluid from the reservoir due to the prevailing suction effect created at the pump inlet. The pump outlet flow is then directed through directional control valves either into the actuation conduit or back into the reservoir according to the position of the valve control lever. The actuation conduit is further branched to the tilting and the lifting cylinders.

During operation, admission of pressurized hydraulic fluid from a pump is directed by a directional control valve into the clearance space at the bottom of the lifting cylinder. The force exerted by the fluid effects the outward movement of the piston, thereby raising the bed (upper components) of the device until the fork prongs are correctly positioned at the bottom of the automobile chassis.

The lever of the directional control valve regulating fluid flow into and out of the lifting cylinder is then turned to the holding position whilst that of the tilting mechanism is tuned to the advance position for fluid admission into the top of the tilting cylinder. The force exerted on the piston by the fluid, causes it to move downward, pulling the tilting platform, together with the automobile body on it gradually onto the bed depending on the desired angle of tilt.

For an 80° to 90° tilt, the lead screw control support structure is operated manually to touch the top of the automobile body in order to prevent turn-over after the tilting mechanism fluid directional control valve is turned to the holding position. After tilting, the lifting cylinder valve lever is turned to the retract position for fluid admission into the top of the cylinder. The fluid then forces the piston downwardly into the cylinder, thereby reducing the height of the automobile with respect to the work floor as the bed is lowered. After positioning, the pump is switched off for the maintenance operation to start.

Lowering of the automobile after the maintenance operation is accomplished by redirecting the fluid back into the lifting cylinder bottom clearance volume. The bed is then lifted, after which the tilting actuator directional control valve lever is turned to the retraction position to direct the fluid into the clearance space at the bottom of the tilting cylinder. The fluid then forces the piston out of the cylinder, pushing the platform with the automobile off the bed gradually onto the work floor.

The height and the degree of tilt of the device are controlled by the position of the control lever of the direction control valve regulating the fluid flow into the tilting and lifting cylinders. Fig. 2 and Fig. 3 illustrate an installed proposed hydraulic tilting device model at the workshop and the nomenclature of the device.

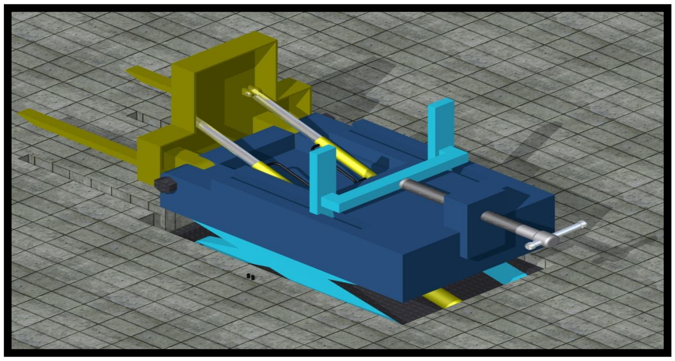

*Figure 2: An Installed Model Of The Proposed Tilting Device*

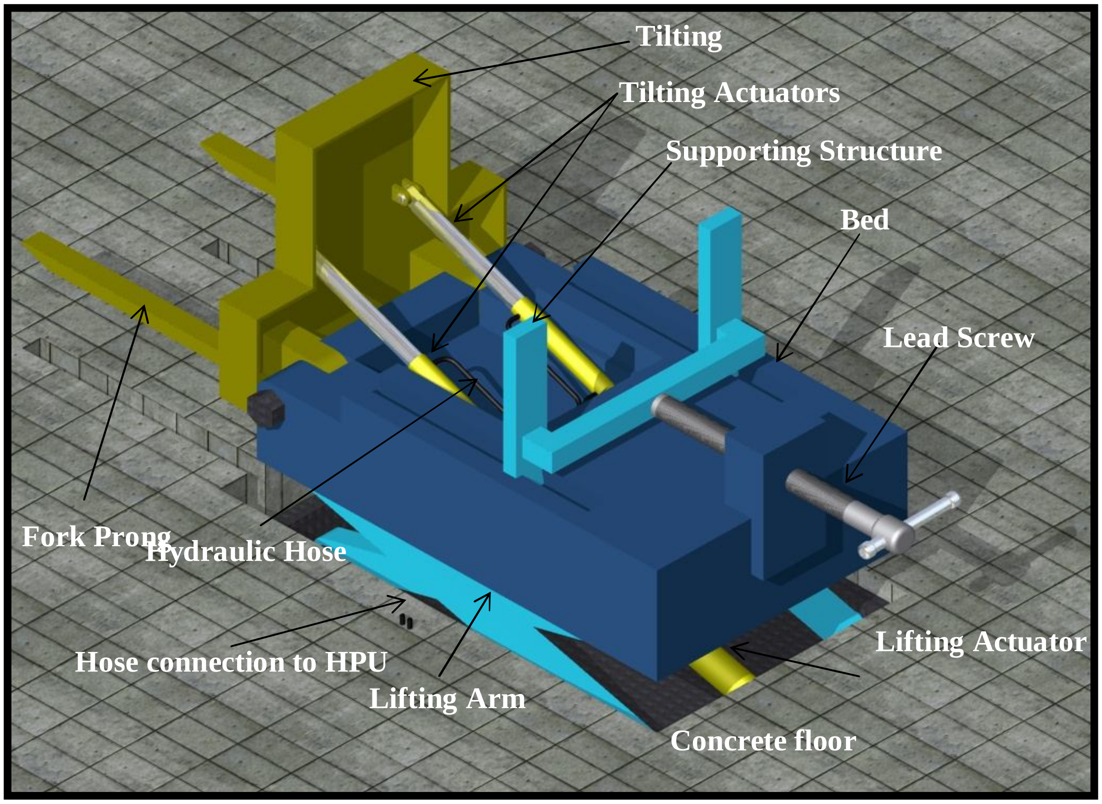


*Figure 3: Nomenclature Of The Proposed Tilting Device*

*3.3.Design Calculations*

The forces and stresses requirement of every component in a system or machine are important factors that determines the geometry, dimensions and the effectiveness of that system's or machine's performance. With regard to this fact, the stresses in the fork prongs and the forces required to effect the lifting and tilting motions have been analyzed as follows:

*3.3.1.Fork Design*

This is the component on which the automobile rests before tilting starts. Considering the maximum designed load capacity of 5000 Kg, the fork must be made of a material capable of withstanding the impacts of this load without undue failure. For this reason, mild steel was selected due to its suitability for machine parts under heavy loading, high tensile strength ranges and its ease of machinability. Assuming that the design load acts uniformly on each of the fork prongs, the free body diagram is as shown by Fig. 4.

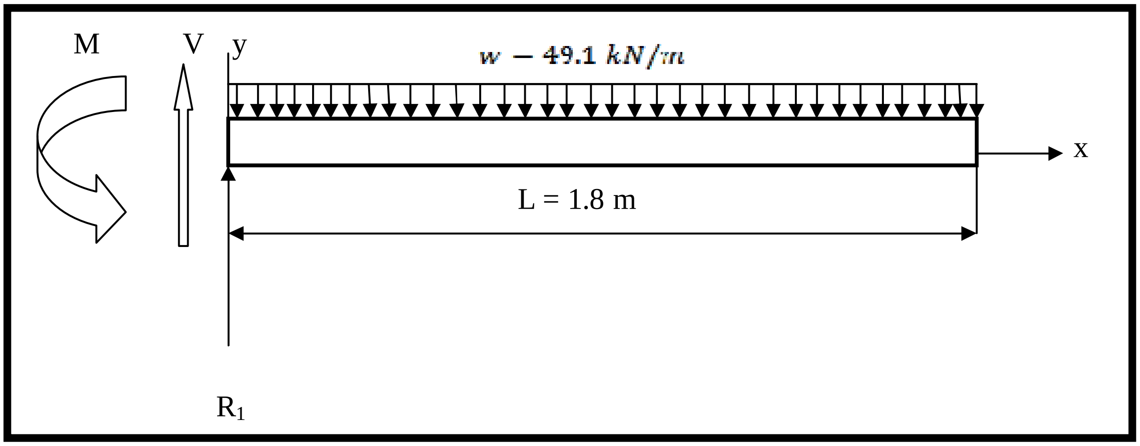


*Figure 4: Free Body Diagram Of The Loaded Fork Prong*

From the above diagram, the reaction $R_1$,

$$R_1 = wl = 49100 \times 1.8 = 88.38\ kN$$

The moment and the shear force equations with respect to the above diagram are given as,

$$V = w(l - x) \quad (1)$$

$$M = (-w/2)(l - x)^2 \quad (2)$$

where, M = the internal bending moment

V = the internal shear force

w = the load or force acting on the fork arm

L = total length of the fork arm

x = the length of the fork prong from the support at any section along the length L

Taking a section at length x = 0.0 m, from equations (4.1) and (4.2), M = 0 Nm, V = 0 N

Taking a section at J-J of length x = 0.45 m,

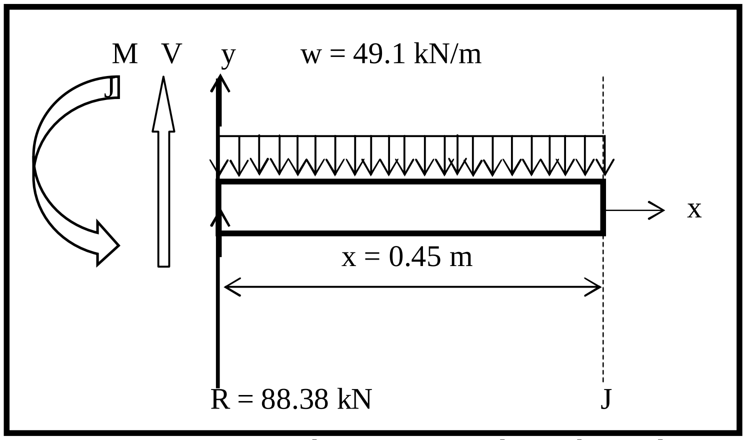


*Figure 5 :A Section At Length X = 0.45 M Along The Fork Prong*

From equation (4.1),

$$V = w(l - x) = 49100(1.8 - 0.45) = 49100(1.35)$$

$$V = 66285\ N = 66.29\ kN$$

From equation (4.2)

$$M = (-w/2)(l - x)^2 = (-49100/2)(1.8 - 0.45)^2$$

$$M = (-24550)(1.35)^2 = -44742.375\ Nm = -44.74\ kNm$$

Taking a section at K-K of length x = 0.90 m,

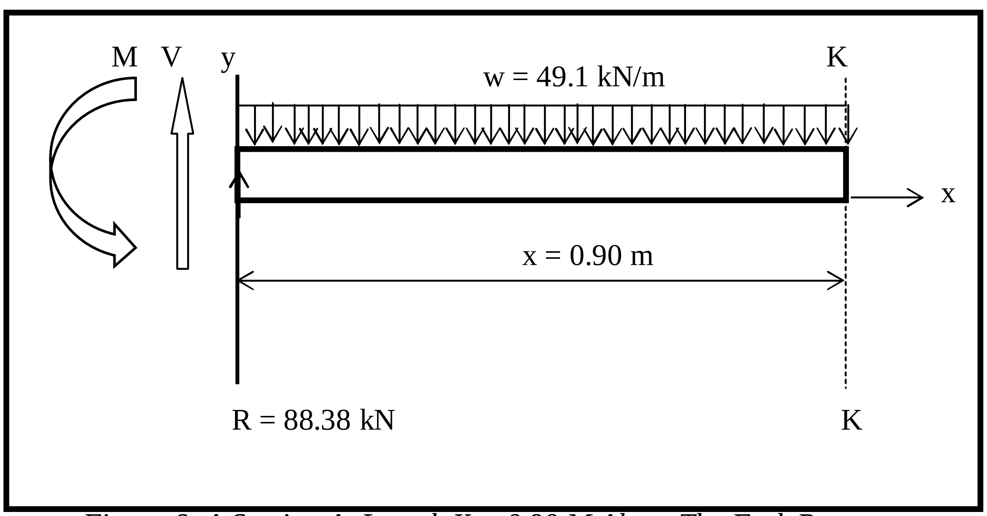


*Figure 6: A Section At Length X = 0.90 M Along The Fork Prong*

From equation (1),

$$V = w(l - x) = 49100(1.8 - 0.90) = 49100(0.90)$$

$$V = 44190\ N = 44.19\ kN$$

From equation (2)

$$M = (-w/2)(l - x)^2 = (-49100/2)(1.8 - 0.90)^2$$
$$M = (-24550)(0.90)^2 = -19885.5\ Nm = -19.89\ kNm$$

Taking a section at L-L of length x = 1.35 m,

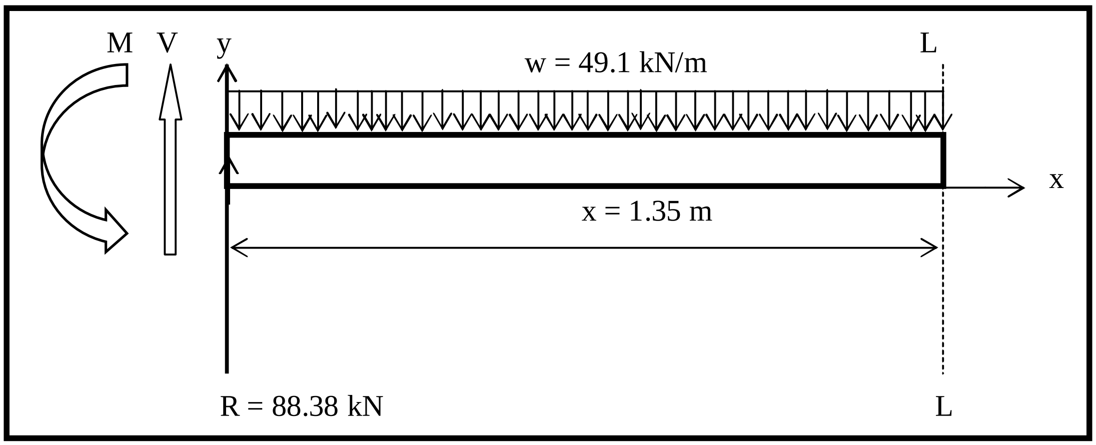


*Figure 7: A Section At Length X = 1.35 M Along The Fork Prong*

From equation (1),

$$V = w(l - x) = 49100(1.8 - 1.35) = 49100(0.45)$$

$$V = 22095\ N = 22.10\ kN$$

From equation (4.2),

$$M = (-w/2)(l - x)^2 = (-49100/2)(1.8 - 1.35)^2$$

$$M = (-24550)(0.45)^2 = -4971.375\ Nm = -4.97\ kNm$$

Taking a section at M-M of length x = 1.80 m,

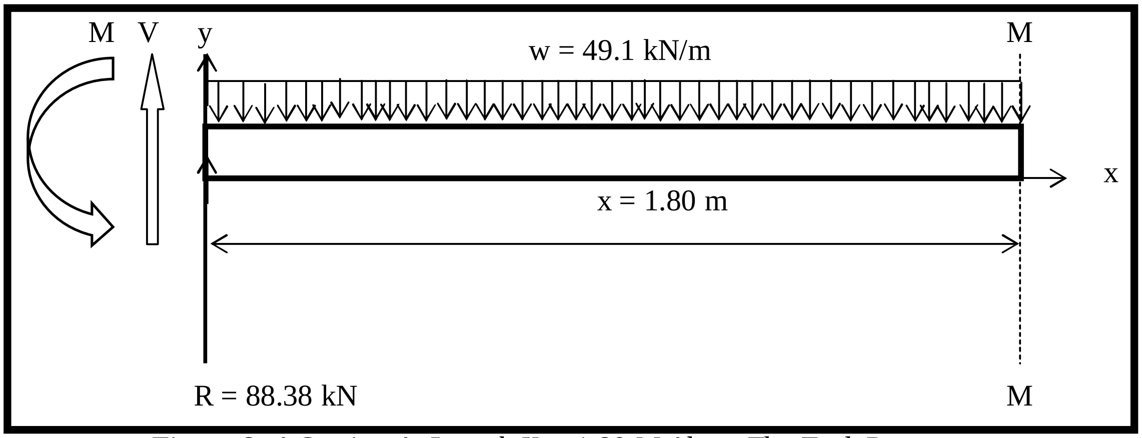


*Figure 8: A Section At Length X = 1.80 M Along The Fork Prong*

From equation (1),

$$V = w(l - x) = 49100(1.8 - 1.8) = 49100(0.0)$$

$$V = 0\ N$$

From equation (4.2)

$$M = (-w/2)(l - x)^2 = (-49100/2)(1.8 - 1.8)^2$$

$$M = (-24550)(0.0)^2 = 0\ Nm$$

Fig. 9 shows the bending moment and shear force diagrams of the load on the fork arm.

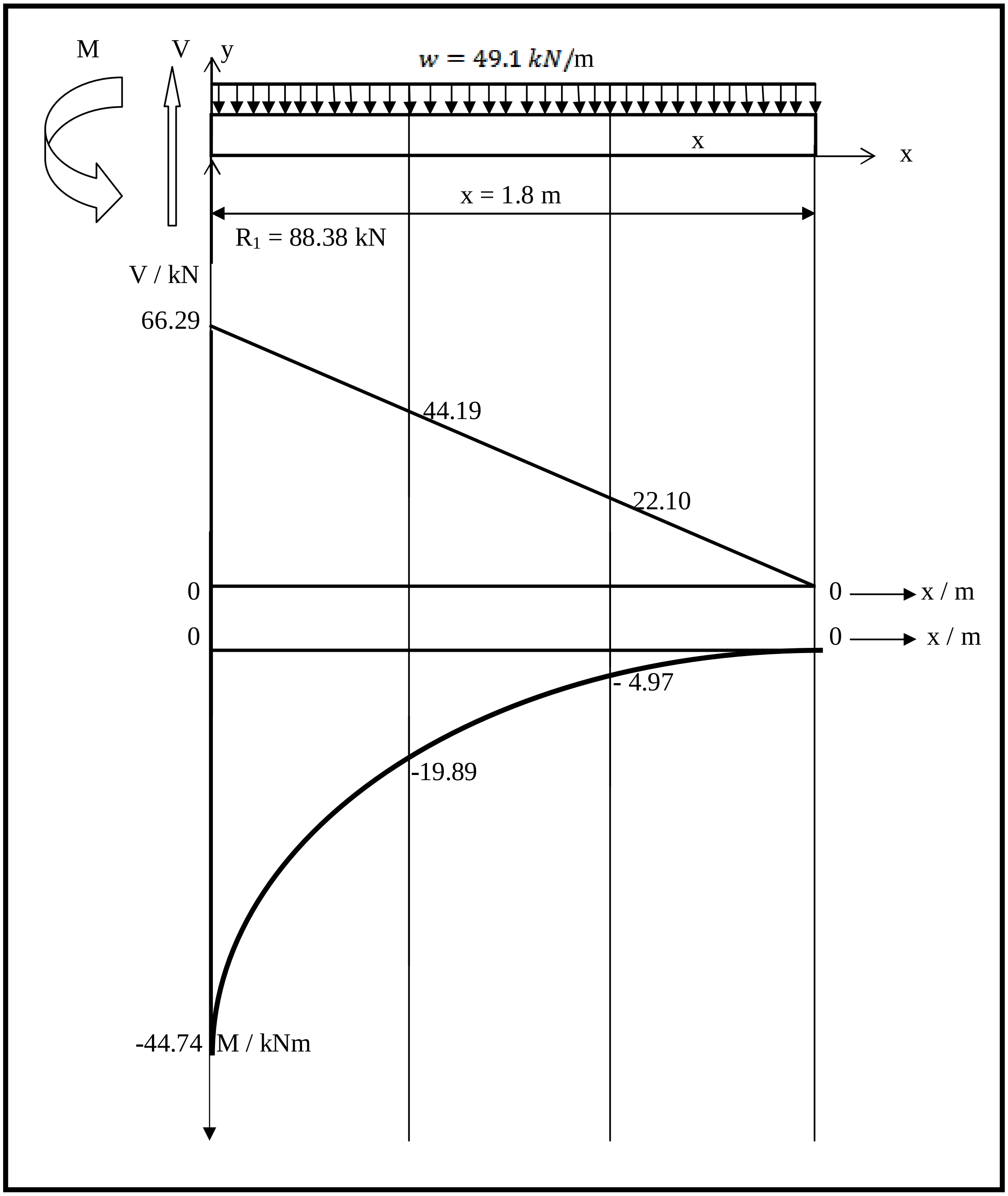


*Figure 9 :Shear Force And Bending Moment Diagrams Of The Loaded Fork Prong*

3.3.1.1.Maximum Fork Bending And Shear Stresses

The maximum shear and bending stresses in the fork arm are given by the formula;

$$\sigma_m = Mc/I \qquad (3)$$

$$\tau_{max} = 3V/2A \qquad (4)$$

where, $\sigma_m$ = maximum bending stress

$\tau_{max}$ = maximum shear stress of the fork arm

I = second moment of area of fork arm cross section

M = maximum diagram bending moment value

V = maximum diagram shear force value

A = cross sectional area of fork arm

Considering the fork arm cross section as shown by Fig. 10, the second moment of area I is given by:

$$I = (b^3h)/12 = (0.15^3 \times 0.1)/12 = 2.813 \times 10^{-5}\ m^4$$

$$c = b/2 = 0.15/2 = 0.075\ m$$

$$A = b \times h = 0.10 \times 0.15 = 0.015\ m^2$$

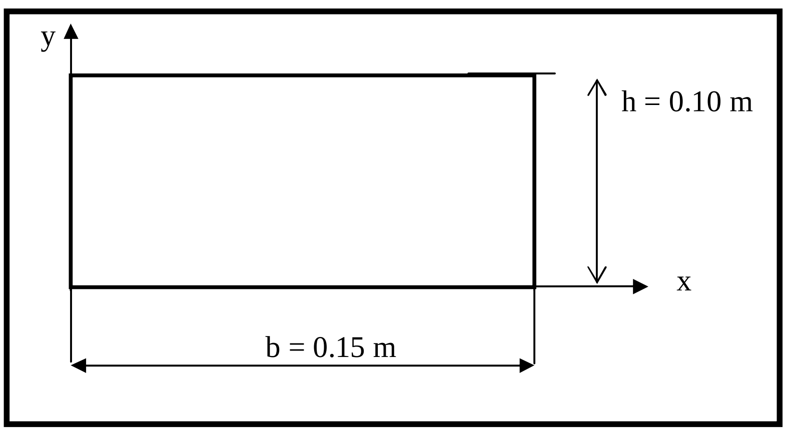


*Figure 10: Cross Section Of The Fork Prong*

therefore,

$$\sigma_m = \left((44742.375) \times 0.075\right)/2.813 \times 10^{-5} = 119291792.6 = 119.29\ MPa$$

$$\tau_{max} = \left(3 \times (66285)\right)/(2 \times 0.015) = 6628500 = 6.63\ MPa$$

From the values of $\sigma_m$ and $\tau_{max}$ obtained, since the ultimate tensile strength ($S_{ut}$) is greater than the bending and the shear stresses in the loaded fork prong, it can confidently be said that the fork prong can withstand the designed load without failing.

*3.3.2.Vertical Height Adjusting Mechanism Analysis*

The analysis of the adjusting mechanism is mainly centered on the needed output cylinder force required to vertically lift and hold the load at the maximum height of 500 mm. The free body diagram of the mechanism at the maximum height is as shown by Fig. 4.8 below.

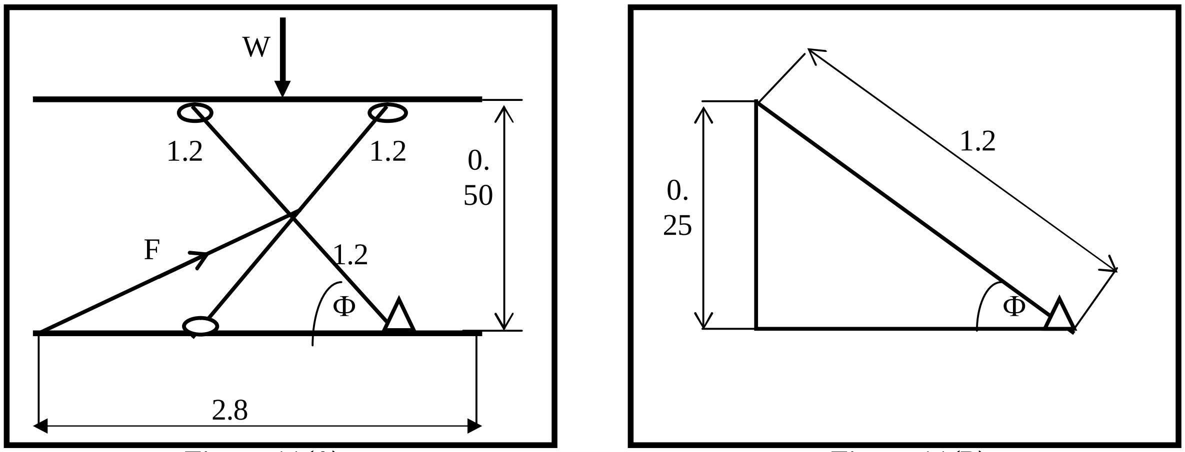


*Figure 11(A)* *Figure 11(B)*

*Figure 11: Free-Body Diagram Of The Vertical Height Adjusting Mechanism (All Dimension In Meters)*

*3.3.3.Actuator Force And Stress Analysis*

The magnitude of the output cylinder force required to vertically lift and hold the load at the maximum height is calculated using the equation below.

$$F - \frac{W + (W_a/2)}{Tan\Phi}$$

where, F = output cylinder force

W = sum of the designed load and the device top main frame weight

$W_a$ = sum of the weight of the lifting arms

Φ = angle between lifting arms and the horizontal axis at the maximum height

From Fig. 11b,

$$\sin\Phi = 0.25/1.2 \Longrightarrow \Phi = \sin^{-1}(5/24) = 12.02^{\circ}$$

For a designed load of 5000 Kg, a top main frame mass of 15200 Kg and a lifting arm mass of 452.74 kg (from the device dimensions, using m = ρV, where ρ = density of steel 7850 kg/$m^3$, V = volume of component.),

$$W - (5000 + 14451.16) \times 9.81 - 190845.31\ N - 190.85\ kN$$

$$W_a = (5 \times 452.74) \times 9.81 = 22206.897\ N = 22.207\ kN$$

Therefore, from the above force equation, the output cylinder force required to vertically lift and hold the load at the maximum height is calculated as;

$$F = \frac{W + (W_a/2)}{tan\Phi} = \frac{(190845.31) + ((22206.897)/2)}{tan(12.02)} = (201948.7585)/0.212921$$
$$= 948468.01\ N$$

$$F = 948.47\ kN$$

For the selected piston material (medium carbon steel) the ultimate strength ($S_{ut}$) is 198 MPa and the modulus of elasticity (E) is 207 GPa (Budynas and Nisbett, 2010).

Using a safety factor (S.F) of 4 (Khurmi, 2009) in other to obtain optimum efficiency since the material is subjected to constant heavy loads and stresses ($\sigma_{ut}$). The allowable ultimate strength is given as;

$$\sigma_{ut} = S_{ut}/S.F = (198 \times 10^6)/4 = 49200000 = 49.2\ MPa$$

The piston working stress ($\sigma_w$) is therefore given as;

$$\sigma_w = F/A_p = 4F/(\pi \times (d_p)^2)\ , \sigma_w = \sigma_{ut} \qquad (5)$$

Where, F = output actuator force

$A_P$ = effective piston area

$d_p$ = piston diameter

From equation (5), the piston diameter is calculated as;

$$(d_p)^2 = 4F/(\pi \times \sigma_W) \Longrightarrow d_p = \sqrt[2]{\{4F/(\pi \times \sigma_W)\}}$$

$$d_p = \sqrt[2]{\{(4 \times 948468.01)/(\pi \times 49200000)\}} = \sqrt[2]{0.02455} = 0.1567\ m$$

therefore, the effective piston area is;

$$A_p = \{\pi \times (d_p)^2\}/4 = \{\pi \times (0.1567)^2\}/4 = 0.05922307417/4 = 0.0193\ m^2$$

The system pressure (P) is also expressed in relation to the output cylinder force and effective piston area as;

$$P = F/A_p = 948468.01/0.0193 = 49143420.21\ Pa = 49.143\ MPa = 491.43\ bar$$

Using a stroke length of 1 m in order to attain the maximum height, the buckling load, $W_B$ is calculated as;

$$W_B = F \times S.F \qquad (6)$$

where, F = cylinder output force

S.F = safety factor

therefore,

$$W_B = 948468.01 \times 4 = 3793872.04 = 3793.872\ kN$$

According to the Euler formula, the Buckling load ($W_B$) can be expressed in terms of the moment of inertia and piston rod length as;

$$W_B = (\pi^2 \times EI)/l^2 \qquad (7)$$

where, $I = \left\{\pi\left(d_{pr}\right)^4\right\}/64$ for a circular cross-section.

I = the moment of inertia of the piston rod

E = the modulus of elasticity of piston rod material

$d_{pr}$ = diameter of piston rod

l = piston rod length

therefore, in terms of the piston rod diameter, the Buckling load is expressed as;

$$W_B = \frac{\pi^2 E}{l^2}\left[\frac{\pi\left(d_{pr}\right)^4}{64}\right] = \frac{\pi^3 E\left(d_{pr}{}^4\right)}{64 l^2} \qquad (8)$$

The piston rod diameter is obtained as;

$$d_{pr}{}^4 = (64 W_B l^2)/(\pi^3 E) = (64 \times 3793872.04 \times 1^2)/(\pi^3 \times 207000000000)$$

$$d_{pr} = \sqrt[4]{(242807810.6/(6.418299 \times 10^{12}))} = \sqrt[4]{(3.783055457 \times 10^{-5})}$$

$$d_{pr} = 0.0784\ m$$

Therefore, the force required to lower the bed is given as;

$$F = P \times \left(A_p - A_{pr}\right) = P \times \{[\pi(d_p{}^2 - d_{pr}{}^2)]/4\}$$

$$F = 49143420.21 \times \{[\pi(0.1567^2 - 0.0784^2)]/4\}$$

$$F = 710509.111 = 710.51\ kN$$

*3.3.4.Cylinder Sizing*

For a stroke length of 1 m, the cylinder height is assumed to be 1.2 m. Assuming a uniform stress distribution in the cylinder, the tangential stress or the hoop stress, $\sigma_t$ is given as;

$$\sigma_t = \left(P d_p\right)/4t \qquad \textbf{(Budynas and Nisbett, 2010)}$$

Accounting for cylinder re-boring, the cylinder thickness is given by;

$$t = \frac{P d_p}{4\sigma_t} + C \qquad (9)$$

where, P = the pressure in the cylinder

t = the cylinder thickness

$d_p$ = diameter of piston

$\sigma_t$ = hoop stress of cylinder material, $\sigma_t$ = 128 MPa

C = allowance for cylinder re-boring

The value of C is calculated by interpolating between a range of values from table 1

.

| **d(mm)** | 75 | 100 | 150 | 200 | 250 | 300 | 350 | 400 | 450 |
|---|---|---|---|---|---|---|---|---|---|
| **C(mm)** | 1.5 | 2.4 | 4.0 | 6.3 | 8.0 | 9.5 | 11.0 | 12.5 | 12.5 |

*Table 1 :Cylinder Diameter And Their Re-Boring Allowances*
*(Source: Aggarwal And Sharma, 2002).*

For a cylinder bore of 156.7 mm, interpolating between 150 mm and 200 mm, the value of C is 4.3082 mm. The cylinder thickness is therefore calculated as shown below.

$$t = \frac{(49143420.21 \times 0.1567)}{4 \times (128 \times 10^6)} + 0.0043082 = 0.0193\ m = 19.35\ mm$$

Also the cylinder outside diameter is calculated from the relation;

$$D_o = 0.1567 + t = 0.1567 + 0.0193 = 0.176\ m = 176\ mm$$

The longitudinal stress in the cylinder is also given as;

$$\sigma_l = PA_p/A_c \qquad (10)$$

where, $A_c = \{\pi(D_o{}^2 - d_p{}^2)\}/4$

$D_o$ = cylinder outside diameter

$A_c$ = area of cylinder

$A_p$ = effective piston area

P = cylinder pressure

therefore,

$$\sigma_l = \frac{(\pi/4) \times Pd_p{}^2}{(\pi/4) \times (D_o{}^2 - d_p{}^2)} = \frac{49143420.21(0.1567)^2}{(0.176^2 - 0.1567^2)}$$

$$\sigma_l = 187928765.8\ Pa = 187.93\ MPa$$

*3.3.5.Tilting Actuator Analysis*

The analysis of the tilting actuator is mainly centered on the needed output cylinder forces, the resulting system pressures and the actuator sizing which must be capable of lifting, lowering, pulling and pushing the load onto and from the device top main frame structure.

*3.3.6.Design Consideration*

The force required to lower the automobile body will be considered first in order to further determined the piston rod diameter to calculate the required force to tilt the automobile onto the bed.

The forces required to move the piston in order to lower the automobile body back to the workshop floor is expressed as;

$$F_{p2} = (m_p + m_a)g = (\rho g V + m_a)g \qquad (11)$$

where, $F_{p2}$ = the force required to lower the automobile body back to the workshop floor

$m_p$ = mass of the tilting platform

$m_a$ = the designed load capacity

V = the volume of the tilting platform

g = acceleration due to gravity

$\rho =$ density of component material

From the device dimensions, using the formula m = ρgV, $m_p$ = 870.3 kg, $m_a$ = 5000 kg, g = 9.81 m $s^{-2}$

Substituting $m_p$ = 870.3 kg, $m_a$ = 5000 kg, g = 9.81 m $s^{-2}$ into equation (14.12)

$$F_{p2} = (870.3 + 5000) \times 9.81 = 57587.64\ N$$

$$F_{p2} = 57.59\ kN$$

For the selected piston material (medium carbon steel), the ultimate tensile strength ($S_{ut}$) is 57 MPa and the modulus of elasticity (E) is 207 GPa (Budinas and Nisbett, 2010). Also, since we are considering steady loading, a factor of safety of 4 is used. (Khurmi and Gupta, 2005).

Therefore, the allowable ultimate tensile strength or the working stress is expressed as;

$$\sigma_{ut} = S_{ut}/S.F = (57 \times 10^6)/4 = 14250000 = 14.25\ MPa$$

The piston working stress ($\sigma_w$) is therefore given as;

$$\sigma_w = F/A_p = 4F/(\pi \times (d_p)^2), \sigma_w = \sigma_{ut} \qquad (12)$$

where, F = output actuator force

$A_P$ = effective piston area

$d_p$ = piston diameter

From equation (12), the piston diameter is calculated as;

$$(d_p)^2 = 4F/(\pi \times \sigma_W) \Rightarrow d_p = \sqrt[2]{\{4F/(\pi \times \sigma_W)\}}$$

$$d_p = \sqrt[2]{\{(4\times 57587.64)/(\pi \times 14250000)\}} = \sqrt[2]{5.145467471 \times 10^{-3}} = 0.0717\ m$$

therefore, the effective piston area is;

$$A_p = \{\pi \times (d_p)^2\}/4 = \{\pi \times (0.0717)^2\}/4 = 0.01615058226/4 = 0.004038\ m^2$$

The system pressure (P) is also expressed in relation to the output cylinder force and effective piston area as;

$$P = F/A_p = 57587.64/0.004038 = 14261426.45 = 14.261\ MPa = 142.61\ bar$$

Using a stroke length of 1 m, the buckling load, $W_B$ is calculated as;

$$W_B = F \times S.F \qquad (13)$$

where, F = cylinder output force

S.F = safety factor

therefore,

$$W_B = 57587.64 \times 4 = 230350.56 = 230.351\ kN$$

According to the Euler formula, the Buckling load ($W_B$) can be expressed in terms of the moment of inertia and piston rod length as;

$$W_B = (\pi^2 \times EI)/l^2 \qquad (14)$$

where, $I - \left\{\pi\left(d_{pr}\right)^4\right\}/64$ for a circular cross-section.

I = the moment of inertia of the piston rod

E = the modulus of elasticity of piston rod material

$d_{pr}$ = diameter of piston rod

l = piston rod length

therefore, in terms of the piston rod diameter, the Buckling load is expressed as;

$$W_B = \frac{\pi^2 E}{l^2}\left[\frac{\pi\left(d_{pr}\right)^4}{64}\right] = \frac{\pi^3 E\left(d_{pr}{}^4\right)}{64 l^2} \qquad (15)$$

The piston rod diameter is obtained as;

$$d_{pr}{}^4 = (64 W_B l^2)/(\pi^3 E) = (64 \times 230350.56 \times 1^2)/(\pi^3 \times 207000000000)$$

$$d_{pr} = \sqrt[4]{(14742435.84/(6418.2993 \times 10^{12}))}$$

$$d_{pr} = \sqrt[4]{(2.296938054 \times 10^{-6})} = 0.0389\ m$$

The forces required to move the piston in order to tilt the load onto the device bed is thus given as,

$$F_{p1} = [\{\pi(d_p{}^2 - d_{pr}{}^2) \times P\}/4] + (m_p \times g) \qquad (16)$$

Where, $F_{p1}$ = the force required to tilt the automobile body onto the device bed

$m_p$ = mass of the tilting platform

$d_p$ = piston diameter

$d_{pr}$ = piston rod diameter

P = pressure in the fluid

g = acceleration due to gravity

therefore,

$$F_{p1} = [\{\pi(0.0717^2 - 0.0389^2) \times 14261426.45\}/4] + (870.3 \times 9.81)$$

$$= 40633.274 + 8537.643 = 49170.92\ N = 49.17\ kN$$

*3.3.7.Cylinder Sizing*

For a stroke length of 0.5 m, from the device dimensions, the cylinder length is assumed to be 0.7 m. From Budynas and Nisbett, 2010, assuming a uniform stress distribution in the cylinder, the tangential stress or the hoop stress, $\sigma_t$ is given as;

$$\sigma_t = (P d_p)/4t \qquad (17)$$

therefore,

$$t = \frac{P d_p}{4\sigma_t} \qquad (18)$$

where, P = the pressure in the cylinder

t = the cylinder thickness

$d_p$ = diameter of piston

$\sigma_t$ = hoop stress of cylinder material, $\sigma_t$ = 87 MPa

$$t = \frac{(14261426.45 \times 0.0717)}{4 \times (87 \times 10^6)} = 0.00294\ m = 2.94\ mm$$

Also the cylinder outside diameter is calculated from the relation;

$$D_o = d_p + t = 0.0717 + 0.00294 = 0.0746\ m = 74.6\ mm$$

The longitudinal stress in the cylinder is also given as;

$$\sigma_l = PA_p/A_c \qquad (19)$$

where, $A_c = \{\pi(D_o{}^2 - d_p{}^2)\}/4$

$D_o$ = cylinder outside diameter

$A_c$ = area of cylinder

$A_p$ = effective piston area

P = cylinder pressure

therefore,

$$\sigma_l = \frac{(\pi/4) \times P d_p{}^2}{(\pi/4) \times (D_o{}^2 - d_p{}^2)} = \frac{14261426.45(0.0717)^2}{(0.0746^2 - 0.0717^2)} = 172806054.2 = 172.81\ MPa$$

**4.Results And Discussion**

The design is carried out based on application of machine design principles and theories and application basic hydraulic principles. The forces and stresses requirement of the component in the device are important factors that determines the geometry, dimensions and the effectiveness of its performance this is evident in the stresses in the fork prongs and the forces required to effect the lifting and tilting motions which have been analyzed From the mode of operation, admission of pressurized hydraulic fluid from a pump is directed by a directional control valve into the clearance space at the bottom of the lifting cylinder. The force exerted by the fluid effects the outward movement of the piston, thereby raising the bed (upper components) of the device until the fork prongs are correctly positioned at the bottom of the automobile chassis. From the Free Body Diagram of the Loaded Fork Prong, Fig.4, $R_1$ was found to be 88.38 k**.** Fig. 9 shows the Shear Force and Bending Moment Diagrams of the Loaded Fork Prong. Maximum bending stress and Maximum shear stress of the Fork arm was found to be 119.29 MPa and 6.63 MPa

**5.Conclusions**

In conclusion, a 5000 kg capacity hydraulic automobile tilting device has been designed and will provide an effective technique for tilting and supporting automobile bodies, thereby eliminating the existing stresses and risks involved.

**6.Recommendations**

Further analysis on the design of the tilting platform should be carried out to enhance adjustability of the interval between fork prongs to suit all wheel base lengths.

Automatic control of the support structure driving mechanism should also be considered.

**7.Reference**


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