Design of an Improved Hydraulic Accumulator for a Truck Loading Lift

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ABSTRACT

A hydraulic accumulator stores the required amount of energy in the form of pressurised fluid from a pump then expends this energy when it is fully charged within a shorter time in order to do work effectively. This paper presents the design of an Improved Hydraulic Accumulator for a truck loading lift. The truck loading lift was designed to lift a load of 250kg from the ground level up to the height of 1.15m (a mean height for small local truck) by utilizing energy stored up in the hydraulic accumulator of the machine.

Keywords: *Design, Hydraulic Accumulator, Truck loading, Lift*

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1. INTRODUCTION

Man in the past had had to rely on his physical strength to lift and carry load from place to place and up various heights. But with time, he began to lift loads more effectively by making use of items around him and setting them up in such a way that will enable him lift the loads more efficiently. A form of simple mechanisms used in those days and still find it usages till date is that wooden planks from tree trucks were use as inclined planes where heavy loads such as stones could be raised to certain heights by sliding or rolling it through the span of the inclined plane. And in cases where a higher height has to be reached, and there were available horizontal support high and sturdy enough to support the load (such as the branches of a trees), ropes attached to the load were extended over the branch and then an effort is applied at the other end of the rope to lift the load to the greater height. This setup is a simple pulley.There were numerous developments in machine that enable more efficient load lifting. The screw jack which uses hydraulic ram was invented bya French scientist Joseph Michael in the mid-18th century. The hydraulic ram he invented basically involved taking the potential energy of a large volume of water pumped to a certain height to lift a smaller volume of water over a greater height as the system's water reached back its equilibrium position [1].

The first occurrence of any hydraulic system was in ancient Mesopotamia 6000 BC. The Greek also developed more sophisticated hydraulic systems like the Hydraulic wheel use for harnessing the energy of falling water. The Roman Empire utilized some hydraulic networks in the supply of water to the populace of some of her major cities, and so many others. There are many setups and applications of hydraulic in machines for the achievement of various tasks. The Hydraulic jack is a type Hydraulic machine and has undergone many stages of development in many parts the world and hence took many different forms. Despite these changes it retained it fundamental characteristic by using fluid to enable the machine to lift load with less effort.Another form of hydraulic machine is the hydraulic Accumulator [2].

1.1 Hydraulic Accumulator

The hydraulic accumulator is a hydro-Dynamic setup that when connected to a parent hydraulic system can store up excess energy in the form hydraulic pressure from the system. It collects excess energy when the pressure in the system exceeds the normal pressure, and it gives the stored energy back to the system when the system pressure is lower than the normal pressure it is set to operate at. The Hydraulic accumulator accomplishes this by receiving the fluid resulting from the excess pressure into one side of its compartment while compressing gas or a spring at the other side. These two sides are separated by a piston actuator or diaphragm [3].

There are four major types of hydraulic accumulators: the bladder, spring-loaded, piston-type and metal Bellows accumulators. A bladder accumulator consists of pressure vessel with an internal elastomeric bladder with pressurized nitrogen on one side and hydraulic fluid on the other side. It has 3 stage operations with an additional over expanded bag schematic. Nitrogen is charged into the accumulator through a valve installed in the top. The spring loaded accumulator uses the energy stored in springs to create a constant force on the liquid contained in an adjacent ram assembly. The load characteristics of a spring are such that the energy storage depends on the force required to compress a spring. Pressurised liquid enters the ram cylinder and compresses the spring.The pressure on the liquid will rise because of the increased loading required to compress the spring. The Piston-Type accumulator consists of a cylinder assembly, a piston assembly, and two end-cap assemblies. The cylinder assembly houses a piston assembly and incorporates provisions for securing the end-cap assemblies. It contains a free-floating piston with liquid on one side of the piston and pre-charged air or nitrogen on the other side. An increase of liquid volume decreases the gas volume and increases gas pressure, which provides a work potential when the liquid is allowed to discharge. The metal bellows accumulator is similar to a piston accumulator, except that a metal bellows replaces piston and piston seals. Metal bellows accumulators are very reliable, high in response and with long life components, and have a proven service history[2].

1.2 Industrial Lifts

Lift has found it usefulness in industries as it is use for heavy lifting of wares. Fork lift and cranes are the two major application of lift in the industries. Forklifts are basically used to engage, lift and transfer palletized loads in material handling, warehousing, manufacturing, and construction applications either in the form manual drive, motorized drive or fork truck. The lift machines available today are either too expensive or and they operate at a considerably lower efficiency. In small deports factories and other mechanical workshops in developing countries carry loading of products into trucks manually. The workers have to manually do all the lifting and carrying of loads around the factory space. This paper presents the design of an integrated hydraulic accumulator truck loading lift which will application in loading deports or small or medium scale factories where cargo are to be lifted into trucks.

2. DESIGN ANALYSIS AND CALCULATIONS

2.1 Design Consideration

Capacity of hydraulic lift

The capacity of the hydraulic accumulator truck loading lift is the maximum amount of load the machine can carry.The power of the machine to lift load comes from the hydraulic accumulator and hence the power or capacity to do work of the machine comes directly from the hydraulic accumulator. The allowable load design for the lift machine is 250kg (approximate five bags of cement). Hence the springs fitted into the upper part of the hydraulic accumulator fixed to the pistons and which does the work of storing the compression energy must be compressed until it reached 2500N (250Kg) of the force while remain in its range of elastic proportionality.

Lift height specification

Since the machine is aim at local and small scale loading, designing the maximum height to fit the base load entry level of well used trucks locally is essential for the design.Apart from truck applications, this machine is also applicable in various others. The maximum height design for the hydraulic accumulator lift designed is 1150mm.Therefore the machine should be capable of lifting loads from the ground level up the required height. Figure 1 shows the schematic of the hydraulic accumulator for a truck loading lift.

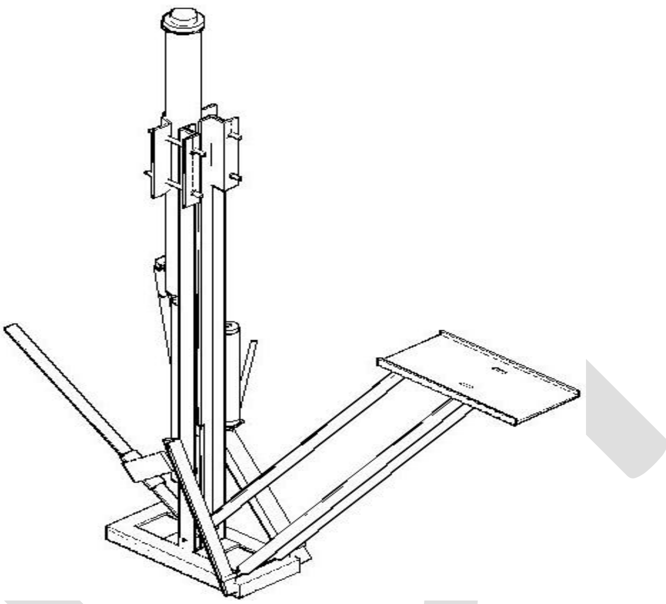

Figure 1. Hydraulic Accumulator for a Truck Loading Lift

2.2 Design Analysis

2.2.1 Pascal's law of hydrostatics

For an enclosed vesselwith two surfaces A1 and A2 as shown in Figure 2, the principle states that the pressure on every surface within the enclosed vessel will be equal. Therefore given that a force F1 acts on the surface A1 then the ratio of the force F1 to the area A1 which represents the Pressure must be equal to the ratio of the force F2 to the area A2 which also represent the uniform Pressure within the same vessel.

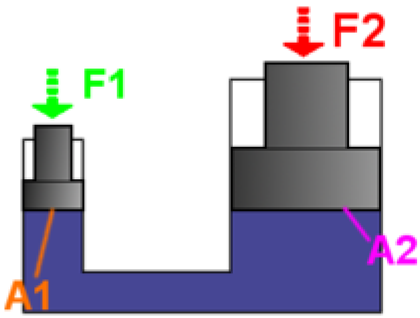

Figure2.Pascal's law of hydrostatics application to press

The hydraulic accumulator lift machine developed in this paper operates based upon the Pascal's law of hydrostatics and is mathematically expressed as:

$$\frac{F1}{A1} = \frac{F2}{A2} \quad (1)$$

The pressure in the Hydraulic accumulator is expressed as the force by the hydraulic pump exerted on the fluid-ward side of the hydraulic ram in the hydraulic cylinder divided by the effective area of the Hydraulic ram:

$$P = \frac{F}{A} \quad (2)$$

Where, P is the pressure in the hydraulic accumulator (Pa); F is the force of hydraulic pump in accumulator cylinder (N); and A is the effective area difference between the ram its shaft (m^2).
The hydraulic accumulator truck loading lift machine is able to do work by relying on this principle to transmit power from the Hydraulic pump to the Hydraulic accumulator and to store up energy in form of hydraulic energy in the hydraulic accumulator. The output work done by the machine can be related to the distance of load moved on the lift platform by the weight of the load and is express as

$$W = L \times F \quad (3)$$

Where, W is the output work done (Nm); L is length travelled by load (m) and F is the weight of the load (N).
The work Input to the hydraulic accumulator is the work done by the hydraulic pump in charging the hydraulic accumulator and is expressed by:

$$W_h = N(L_h \times F_h) \quad (4)$$

Where,W_h is the total work done by the hydraulic pump to charge the hydraulic accumulator (Nm); N is the number of circles or strokes the hydraulic pump makes in charging the accumulator; L_h is the length travelled by the ram of the hydraulic pump in a single stroke (m) and F_h is the force required for a single stroke by the hydraulic pump (N).
Given that the load to be lifted is 2500N and that the internal and external radii are 0.034m and 0.04m respectively, the force required from the hydraulic accumulator to lift the load and the internal pressure of the cylinder calculated from equations (1) to (4) are 4032N and 1.8 MPa

2.2.2 Stress Analysis for Hydraulic Cylinder

Considering the operation of the Hydraulic cylinder, three (3) principal stresses will be developed in the hydraulic accumulator cylinder when its hydraulic compartment is filled with hydraulic fluid. These stresses are the Hoop (tangential/circumferential), Longitudinal and Radial [4]. Figure3 below depicts these stresses.

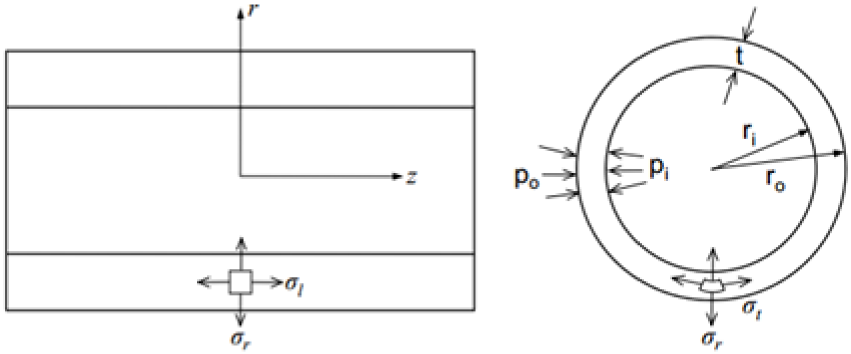

Figure 3. Principal Stresses acting ona cylinder under pressure

Tangential or Circumferential or Hoop Stress σ_h

The circumferential stress, σ_h acting along the circumference of the cylinder which is also known as tangential stress, for a cylinder which has internal pressure arising from pressured fluid in the cylinder is given by the equation below;

$$\sigma_h = \frac{Pd}{2t} \tag{5}$$

Where, σ_h is the circumferential stress (Pa); P is the internal pressure due to presence of pressurized fluid (Pa); t is the thickness of the cylinder wall (m); and d is the internal diameter of cylinder.

For a cylinder with thickness t greater than 1/20 of the internal diameter, the hoop stress is given as;

$$\sigma_h = \frac{P_i r_i^2 - P_0 r_0^2}{r_0^2 - r_i^2} + \frac{r_i^2 r_0^2 (P_i - P_0)}{r^2 (r_0^2 - r_i^2)} \tag{6}$$

Assuming that the external pressure is zero, the above expression reduces to

$$\sigma_h = \frac{r_i^2 P_i}{r_0^2 - r_i^2}\left[1 + \frac{r_o^2}{r_i^2}\right] \tag{7}$$

Where, σ_h is the circumferential stress (Pa); r_i and r_o are the internal and external radii of the cylinder (m); P_i and P_o are the internal pressure and external pressure of the outside (Pa).

Longitudinal Stress σ_L

This stress is as a result of the pressure force acting on the two cover plates (in this case the piston bottom and the lower cover plate) of the cylinder. It is the stress which acts along the length of the cylinder and tends to elongate the cylinder. In this case where the piston is anchored to springs on top which tend to resist the force of the fluid, this stress equals the compressive force store up in the springs of the accumulator.
Thelongitudinal stress is given by;

$$\sigma_L = \frac{Pd}{4\,t} \tag{8}$$

For a cylinder with thickness t greater than 1/20 of the internal diameter, the hoop stress is given as;

$$\sigma_L = \frac{P_i r_i^2 - P_o r_o^2}{r_o^2 - r_i^2} \tag{9}$$

Where the ends are closed, the above equation reduces to;

$$\sigma_L = \frac{P_i r_i^2}{r_o^2 - r_i^2} \tag{10}$$

Radial Stress σ_R

The radial stress is the stress which acts outwards from the inside of the cylinder in the radial direction. For a thick walled cylinder, it is given by;

$$\sigma_R = \frac{P_i r_i^2 - P_0 r_0^2}{r_0^2 - r_i^2} - \frac{r_i^2 r_0^2 (P_i - P_0)}{r^2 (r_0^2 - r_i^2)} \tag{11}$$

Assuming that the external pressure is zero, the above expression reduces to

$$\sigma_R = \frac{r_i^2 P_i}{r_0^2 - r_i^2}\left[1 - \frac{r_o^2}{r_i^2}\right] \tag{12}$$

The projected cylinder wall thickness is 5mm with an internal diameter of 70mm. These two values show that it is a thick wall cylinder and therefore only equations (7), (10) and (12) willbe used.

Given that the internal and external radii are 0.034m and 0.04m respectively, and using the internal pressure obtained from equations (5) to (12), the values of the hoop, longitudinal and radial stresses as calculated from equations (7), (10) and (12) are respectively 8.437MPa, 3.659MPa and -1.12MPa.

2.2.3 Stress Analysis for Hydraulic Ram Shaft

The hydraulic ram shaft is connected to the piston inside the cylinder and extends outward from the lower end of the hydraulic cylinder. By nature of the design, this shaft will experience only a compressive force when it forces are down the rear arm of the linkage.

The below equation applies to the calculation of the stress in any shaft or uniformly cross sectioned bar:

$$E = \frac{\sigma}{\varepsilon} \tag{13}$$

Where,

$$\sigma = \frac{P}{A} \tag{14}$$

And

$$\varepsilon = \frac{e}{L} \tag{15}$$

Given that the material of the ram shaft is of certainYield strength S_y and the length and area are determined. Then to calculate for the maximum-critical load the shaft can carry without failing, the following equations must be considered:

$$S_r = \frac{1}{\sqrt{\frac{I}{A}}} = \frac{1}{K} = \frac{L_e}{r_{min}} \tag{16}$$

Where, E is Modulus of elasticity; $\sigma$is stress; $\varepsilon$is strain; $P$is pressure; S_y is Yield strength of shaft material (GPa); I is second moment of Area (m^4); A is area (m^2); $S_r$is the slenderness ratio;K is radius ofgyration; L_e is effective length constant; r_{min} is minimum radius of column (m).
The transitional slenderness ratio is given as

$$C_c = \sqrt{\frac{2\,\pi^2 E}{S_y}} \tag{17}$$

The slenderness ratio and the transitional slenderness ratio gives an indication if the Ram Shaft is to be consider short or Long and hence which of the below equation

If from $S_r > C_c$, then the column shaft is consider Long and If $C_c > S_r$ then the shaft or column is considered Short.

For a Long shaft, the critical load is given by Euler's equation below:

$$P_{cr} = \frac{\pi^2 EA}{\left(\frac{KL}{r}\right)^2} \tag{18}$$

For a Short shaft or column, we use the J. B. Johnson'sequation which is stated below:

$$P_{cr} = AS_y\left[1 - \frac{\left(\frac{KL}{r}\right)^2}{4\pi^2 E}\right] \tag{19}$$

Where,P_{cr} iscritical Load; L is length of shaft (m); r is radius of shaft (m) [4].

Using a factor of safety, the critical load and the allowable load are related using the equation below;

$$P_a = {}^{Pcr}/_{N} \tag{20}$$

Where, N is factor of safety; and P_a is allowable load (N).

Given that the parameters of the shaft of the hydraulic ram for radius, Length, Yield Strength, area and second moment of area are respectively 0.0225m, 0.81m, 210GPa, 0.00159mm^2 and 2.0129$\times 10^{-7} m^4$. Calculations using equations (13) to (20) yields the following: L_e =1.7; r_{min} = 0.0225m; E =210$\times 10^9$Nm^{-2}; Yield strength of shaft material S_y = 290$\times 10^6$MPa; S_r = 75.6; C_c= 119.557; P_{cr} = 461100N.

2.2.4 Stress Analysis for Stand Frame

The arrangement of the frame is such that a pair of the angle bars holds the upper part of the machine which contains the accumulator to the lower part which includes the Base Stand and the lift linkage. Therefore, the pair of angle bars that connect the upper part and the lower part is under the greatest stress in the frame work as it carries the load of the Hydraulic and hydraulic accumulator during operation.For a centralized load distribution, the following is applicable:

$$S_y = \frac{Pa}{A} \times N \quad (21)$$

Where, S_y is yield strength of angular bar material (MPa); P_a is maximum allowable load (N); A issum ofcross-sectional area through the region of consideration (m^2) and N is factor of safety. Figure 4 shows the schematic of stand frame.

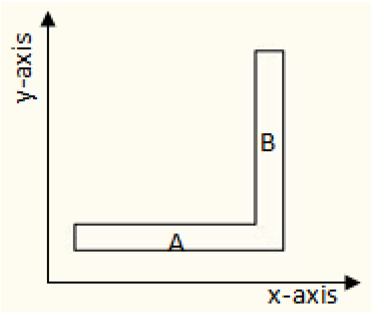

Figure 4. Schematics of stand frame

$$X - axis = \frac{(Area\ (A) \times length\ origin\ to\ center\ of\ A\ on\ x-axis) + (Area\ (B) \times length\ to\ center\ of\ B)}{(Area\ of\ A + Area\ of\ B)} \quad (22)$$

$$Y - axis = \frac{(Area\ (A) \times length\ origin\ to\ center\ of\ A\ on\ y-axis) + (Area\ (B) \times length\ to\ center\ of\ B)}{(Area\ of\ A + Area\ of\ B)} \quad (23)$$

Given that the Sum Cross-sectional area, factor of safety, Yield strength of material and maximum weight are respectively 0.000219m^2, 2, 206MPa and 600N; the centroids on the x-axis and y-axis is 0.001061m and the maximum allowable load is 22557N.

2.2.5 Stress Analysis for Compression Spring

The spring (Figure 5) stores the energy in hydraulic accumulator in the form of compressive elastic or strain energy when the piston in the cylinder moves upward due to the action of the hydraulic pump.The major consideration for the spring is the maximum amount of energy or force it can retain and reproduce within its elastic limits. The maximum weight designed for the lift must not exceed the maximum force of both springs used in the lift machine.

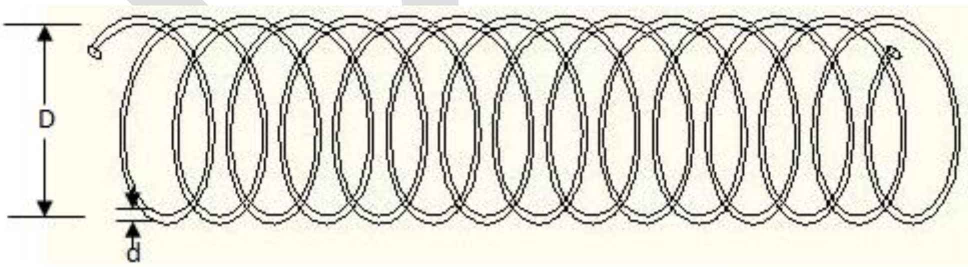

Figure 5.Accumulator compression Spring

The relationship of force produced by spring and the deflection of the spring is given below:

$$F = K(x) \quad (24)$$

Where,F is force applied (N); K is spring constant and x is deflection of spring.
The value for the spring constant for most commercial springs are specified but can be calculated with the following equation:

$$K = \frac{G\, d^4}{(n\ 8\ D^3)} \qquad (25)$$

Where,G is Modulus of rigidity of spring material; d is diameter of spring wire; D is mean diameter of spring coil; n is number of active coils in the spring.
Also given that the air would also be compressed, the below equation shows
The relationship between the pressures of air to change in volume when it is compressed is given as;

$$P = B \times \frac{V_{change}}{V_{initial}} \qquad (26)$$

Where, P is Pressure; B is Bulk modulus; V_{change} is change in volume and $V_{initial}$ is Initial volume.
Given that F= 4032.26N; G= 80$\times 10^9$Pa; d= 0.006m; D= 0.034m; and n=20, the values obtained from calculations using equations (24) to (26) are K = 16486.9N-m^{-1}, B_{air}=1.41 $\times 10^5$, x = 0.245m, V_{change} = 0.000943, V_{change} = 0.0023, P = 31768.1Pa and the force from the compressed air is 122.25N.

2.2.6 Stress Analysis for Lift Linkage

The linkages are by the nature of their application subject to primarily bending moments. The nature of forces acting on the links is shown in Figure 6 below. The lift arm has a single support and hinged to the frame on one end and a load on the other end.

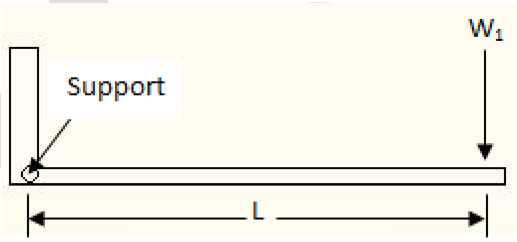

Figure 6.Schematics of forces on links

The Stress due to the Bending moment on the link L in the Figure 7 above may be given by[5]:

$$\sigma_b = {M_b y}/{I} \qquad (27)$$

For a hollow shaft, the second moment of area is given by:

$$I = \frac{\pi[D^4 - d^4]}{64} \qquad (28)$$

Where,σ_b is stress due to bending moment; M_b is bending moment; y is distance from the neutral plane of the bar to outer fibre; I is second moment of area of the shaft of the link; D is outer diameter of the hollowed shaft and d is Inner diameter of the Shaft.
Given that W_1= 2500N; y= 0.009m; D = 0.018m; d= 0.013m, I was calculated from equation (28) as$3.751 \times 10^{-9} m^4$. The Stress due to bending moment of the linkage will be calculated for the rear linkage and for one of the three lift linkage with a 1/3 of the total weight of the

load. Using equation (27), the Stress due to bending moment (σ_b) for rear and 1/3 of the forward linkages are 3.719GPaand2.0994Gpa.

3. MACHINE PERFORMANCE

The overall lift machine efficiency can be expressed as the ratio of work input (work from hydraulic pump) to work output (load lifting), expressed in percentage.

The Efficiency of the lift machine may therefore be expressed as:

$$\eta = \frac{W}{W_h} \times 100 \quad (29)$$

Alternatively;

$$\eta = \frac{MA}{VR} \times 100 \quad (30)$$

Where,

$$MA = \frac{F}{F_h} \quad (31)$$

$$VR = \frac{L_h}{L} \quad (32)$$

Where, η is efficiency of the lift machine;MA is mechanical advantage; W is work output; W_h is work input; VR is velocity ratio; F is load; F_h is effort; L is distance moved by load; L_h is distance moved by effort.

4. DISCUSSION ON DESIGN

The stresses obtained for the Hydraulic Accumulator cylinder design are below the Strength of the material of the cylinder and piston. Therefore the cylinder will not be stressed beyond its capacity. Also, the bending moment of the linkages are below the allowable bending stress of the link shaft material, indicating that the machine is within it limits of operation.

5. CONCLUSIONS

The design and analysis of hydraulic accumulator for application in a truck loading lift carried out. The machine, when fabricated is capable of storing energy in the form of pressurised hydraulic fluid in its hydraulic accumulator, and therefore makes it a highly effective machine in loading trucks.

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