Design of Hydraulic Cylinder for the Operation of Sixty Tons Hydraulic Press

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Abstract:

Hydraulic cylinder is a major and central working component in the design and development for the proper functioning of the sixty tones hydraulic press. The cylinder uses high pressure that produces large forces at precise movement i.e ram movement. It does a non-compressible work (being oil driven) and provides the calculated force that convert fluid power into mechanical motion.

This machine get it power from pressurizes hydraulic fluid, which is oil. The pressure vessel (cylinder or tank) was used to store fluid under pressure over a finite period of time. Therefore, the pressure vessel was designed with great care because the rupture of the vessel may occur if the system pressures exceed certain limits. The material used for the pressure vessel of this work was the thick shell since the ratio of shell diameter (d/t) is less than 10 (factor for construction of wall thickness).

The design was done primarily by what the system will accomplish in terms of actions against resistance loads within specified time limit; hence the motion in it is by the transportation of the desired performance of the system into system hydraulic pressures. The volume of the cylinder was not left out, for this match with the system output characteristics of an available input to the system in order to sustain the operation. The parts components design of this work includes; the design of Flat End Cover, design of Bolts, design of Cylinder Flange, Piston design, design of Static and Dynamic Seals, and design of Pump.

Keywords —**Hydraulic Cylinder, Pressurized Fluid, Pressure, Design, Hydraulic Press.**

INTRODUCTION

Hydraulic cylinder is a machine that generates high pressure hydraulic fluid. Its operation is quite simple, pressurizes fluid (oil) enter the system and acts on a confined piston thereby in turn drives a smaller diameter ram or piston to deliver a given volume of fluid. As a result, the output pressure is intensified and is considerably higher than the input pressure.

Cylinders are design for use with pressured air, hydraulic oil, and fire resistant fluids, in some cases special seals are required. Its body is of strict quality control standard and precision manufacture ensures that all tubes meet rigid standard of straightness, roundness and surface finish. The steel tubing is surface finished to minimize internal friction and prolong seal life, [10].

Deceleration devices or built-in cushions are optional and can be supplied at head end, cap end, or both ends without change in envelope or mounting dimensions. Standard straight or tapered cushions have been used in industrial cylinders over a very broad range of applications. Research has found that both designs have limitations. As a result, there is a new approach in cushioning of industrial hydraulic cylinder and for specific load and velocity conditions have been able to obtain deceleration values that come very close to the ideal. The success lies in a stepped sleave or spear concept where the steps are calculated to approximate theoretical orifice areas. The stepped cushion design shows reduced pressure peaks for most load and speed conditions with comparable reduction of objectionable stopping forces being transmitted to the load and the support structure, [10].

A hydraulic press is a machine that has a bed or a plate in which the metallic material is placed so that it can be crushed, straightened or moulded. The concept of the hydraulic press is based on Pascal's theory, which states that when pressure is applied on fluids in an enclosed system, the pressure throughout the system always remains constant, in simple words, a hydraulic press is a machine that makes use of the pressure exerted on the fluids to crush something, [3].

Since the hydraulic press works on the basis of Pascal's Law, its working is similar to the one of the hydraulic system. A hydraulic press consists of basic components that includes the cylinder, pistons, the hydraulic pipes; etc. The working of this press is very simple. The system comprises of two cylinders; the fluid (usually oil) is poured in the cylinder having a small diameter. This cylinder is known as the slave cylinder.

The piston in this cylinder is pushed so that it compresses the fluid in it that flows through a pipe into the larger cylinder. The larger cylinder is known as the master cylinder. The pressure is exerted on the larger cylinder and the piston in the master cylinder pushes the fluid back to the original cylinder.

The force applied on the fluids by the smaller cylinder results in a larger force when pushed in the master cylinder. The hydraulic press is mostly used for industrial purposes where a large pressure is required for compressing, crushing or punching metals/materials into thin sheets.

Standard cylinders have been designed to meet the wide range of applications. The following types of cylinders provide an overview of what is available.

1.1.1 Single-Acting Cylinder.

The single-acting cylinder is pressurized at one end only, with the opposite end vented to atmosphere through a breather filter (air cylinder) or vented or to a reservoir. The return stroke of the cylinder is accomplished by some external means.

1.1.2 Double-Acting Cylinder.

The most familiar double-acting cylinder is the single rod end. This type of cylinder provides power in both directions with a pressure port at either end. Single rod end cylinders exert greater forces when extending than when retracting, since the piston area on the blind end is larger than the piston area on the rod end (due to the area covered by the piston rod).

1.1.3 Position or Duplex Cylinder.

Duplex cylinders are similar to tandem cylinders in that both are cylinders connected in line, but the pistons of a duplex cylinder are not physically connected; the rod of one cylinder protrudes into the non-rod end of the second, and so forth. A duplex cylinder may be more than two in-line cylinders and the stroke lengths of the individual cylinders may vary. This result in a component that can achieve a number of different fixed stroke lengths depending on which of the cylinders and on which end the cylinders are pressurized.

1.1.4 Spring Return Cylinder.

The spring return cylinder is normally considered a single acting cylinder. The operation of this type of cylinder is the same as a single acting cylinder, except that a spring is used to accomplish the return stroke.

1.1.5 Double Rod End Cylinder.

The double rod end cylinder is used when it is necessary for the cylinder to exert equal force and operate at equal speed in both directions. It also can be used to operate limit valves or switches.

1.1.6 Tandem Cylinder.

A tandem cylinder consists of two cylinders mounted in line with the piston, connected by a common piston rod. The main advantage of this cylinder is the multiplication of force during the entire stroke, without requiring higher operating pressures or large bores, [9].

2. Methodology

2.1. Stresses in Cylinder Shells subject to Internal Pressure:-

In thin shell, we could reasonably assume that the tensile stresses are uniformly distributed over the section of the wall.

In the design of thick cylinders, the following equations are mostly used:

i. Lame's Equation: Assumes that the longitudinal fibers of the cylindrical shells are equally strained.

ii. Bernies Equation: in case of open cylinders (such as pump cylinders, rams, gun barrels) made of ductile material (i,e low carbon steel, brass bronze and aluminum alloys), the allowable stresses theory of failure.

iii. Clavarinos Equation: This equation is also based on the minimum - strain theory of failure, but it is applied to closed-end cylinders (or cylinders fitted with heads) made of ductile material.

iv. Barlow's Equation: This equation is generally used for high-pressure oil and gas pipes.

The use of these equations depends on the type of material and the end construction.

Lame equations has shown that the tangential stress at any radius x is, [1,2,7].

$$\delta t = \frac{P_i\, r_i 2 - P_o\, r_o 2}{r_o 2 - r_i 2} + \frac{r_i 2\, r_0 2}{x 2}\left(\frac{P_i - P_o n}{r_0 2 - r_i 2}\right) \qquad (1)$$

And the radial stress at any radius x is

$$\delta t = \frac{P_i\, r_i 2 - P_o\, r_o 2}{r_o 2 - r_i 2} + \frac{r_i 2\, r_0 2}{x 2}\quad \frac{P_i - P_o}{x r_o 2 - r_i 2} \qquad (2)$$

since we are concerned about internal pressure, (Pi = Po) only, therefore substituting the values of external pressures Po = O

∴ Tangential stress at any radius x

$$\delta r = \frac{P_i\, r_i 2}{r_o 2 - r_i\, 2}\quad 1 + \frac{r_o}{x 2} \qquad (3)$$

$$\delta r = \left(1 + \frac{r_o}{x 2}\right) \qquad (4)$$

We see that the tangential stress is always a tensile stress where the radial stress is a compressive stress.

We also know that the tangential stress is maximum at the inner surface of the shell (i.e. when $x = r_i$) and it is minimum at the outer surface of the shell (i.e. when $x = r_o$).

Substituting the value of $x = r_i$ and $x = r$, in equation (3), we find that the maximum tangential stress at the inner surface of the shell.

$$\delta r \ (max) \quad = \quad \frac{Pi \ (ro2 + ri2)}{ro2 - ri2} \text{ér} \tag{5}$$

And minimum tangential stress at the outer surface of the shell,

$$\delta r \ (max \ ) \quad = \quad \left(\frac{2P \ ri2}{ro2 - ri2} \right) \tag{6}$$

We know that the stress is maximum at the inner surface of the shell and zero at the outer surface of the shell.

Substituting the value of $x = r_i$ and $x = r_o$ in equation (4), we find that maximum radial stress at the inner surface of the shell.

$$\delta t \ (max) \ = \ - \ P \ (compressive) \tag{7}$$

And the minimum radial stress at the Outer surface of the shell,

$$\delta t \ (min) = \ 0 \tag{8}$$

In designing a thick cylindrical shell of ductile material (eg. mild steel), Lame's equation is modified according to maximum shear stress theory.

According to the theory, the maximum shear stress at any point in a strained body is equal to one-half the algebraic difference of the maximum and minimum principal stress at that point.

For a thick cylinder maximum shear stress at the inner surface, and the minimum shear stress at the inner surface as:

Maximum shear stress = $\delta s \ (max)$

$$= \frac{\delta t \ (max) - \delta t \ (min)}{2} \tag{9}$$

From which we obtain:

$$t = r_i \left\{ \sqrt{\frac{\delta s \ - \ 1}{\delta t \ - \ 2P}} \right\} \tag{10}$$

The value of 6s is usually taken as one - half the tensile stress δs. Therefore, the above expression may be written as:

$$t = r_i \left\{ \sqrt{\frac{\delta t \ - \ 1}{\delta t \ - \ 2P}} \right\} \tag{11}$$

From equation (11) above, we can deduce that if the internal pressure P is equal to or greater than the allowable working stress, then no thickness of the cylinder wall will prevent failure. Thus it is impossible to design a cylinder to withstand fluid pressure greater than the allowable working stress for a given material. This difficulty is overcome by using compound cylinders.

2.2. Design of Flat End Cover

The head or bottom of hydraulic cylinder may have flat plates or slightly dished plate. The plate may be cast integrally with the cylinder walls or fixed by means of bolts, rivets or welds.

The design of flat plates forming the heads depends upon the following factors:

a. Type of connection between the head and the cylinder wall (i.e. freely supported or rigidly fixed)
b. Nature of loading (i.e. uniformly distributed or concentrated).

The internal pressure in the cylinder tries to lift the cylinder cover while the bolts try to retain it in its position. But the center of pressures for these two loads does not coincide; hence the cover plate is subjected to bending stresses.

The stress distribution in the cylinder heads and cover plates are of complex nature, therefore, empirical relations based on the work of Grashof and Bach are used in the design of flat plates, [1].

Consider a case with a flat circular plate with uniformly distributed load. The thickness (T) of the plate having a diameter (D) supported at the circumference and subjected to a pressure (P) uniformly distributed over the area is given by:

$$T = \quad KD \ \sqrt{\frac{P}{\delta t}} \tag{12}$$

Where δt = Allowable design stress

D = Diameter of plate

P = Pressure of fluid
K = Coefficient depending upon the method of holding the edge.

The cylinder cover may be secured by means of bolts or studs, though studs are preferred. Bolts are frequently used.

2.3. Design of Bolts:

In order to find the size and number of bolts, the following procedure was adopted:

Let D_i = Diameter of the cylinder
P = Pressure in the cylinder
d_i = Core diameter of the bolt

n = Number of bolts

δ_{tb} = Permissible tensile stress of the bolt material

We know that force (F) acting on the cylinder cover is given by

$$F \quad = \quad \frac{\pi (D_i)^2 P}{4} \qquad (14)$$

This is resisted by n number of bolts provided on the cover:

:.   Resisting force offered by n number of bolts is

$$F \quad = \quad \frac{\pi}{4} (D_c)^2 \; \delta_{tb} \; x \; n \qquad (15)$$

From equations (14) and (15), we have

$$= \quad \frac{\pi (D_i)^2 P}{4} \quad = \quad \frac{\pi (d_c)^2}{4} \; \delta_{tb} \; x \; n \quad (16)$$

The number of bolts, n can be obtained from equation (16), if the size of the bolt is known and vice versa. Usually the size of the bolt can be assumed.

If the value of n as obtained from the above relation is odd or fraction, then the next higher even number is adopted. The bolts are screwed up tightly along with seeds or gasket packing in order to provide a leak proof joint.

Due to the lightening of bolts, sufficient tensile stress is produced in the bolt. This may break the bolt even before any load due to internal pressure acts upon them. Therefore, a bolt less than 16mm diameter is usually not recommended.

The tightness of the joint also depends upon the circumferential pitch of the bolts. The circumferential pitch should be between $20 \sqrt{d_1}$ and $30 \sqrt{d_1}$

Where d, is the diameter of the hole in mm for bolts.

The pitch circle diameter (D_p) in usually taken as:

$$D_p = D + 2t + 3d_1 \qquad (17)$$

Where t = thickness of the cylinder wall.

2.4.   Design of Cylinder Flange

The thickness of the cylinder flange (t_2) may be determined from bending consideration.

Having obtained the cylinder wall thickness from equation (11) in the hydraulic cylinder, the force trying to separate the base plate and the flange has to be resisted by stress produced in the bolts. The force is due to fluid pressure and is given by:

$$F_1 = \quad \frac{\pi D_i^2 P}{4} \qquad (18)$$

Where Di - Internal diameter of the cylinder seal or packing has also to be compressed to make the joint leak proof. The intensity of the pressure should be greater than the pressure of the fluid insider the cylinder. However, for the purpose of calculation, it is assumed that the packaging material is compressed to the same pressure as that of the inside cylinder.

Therefore, the force tending to separate the flanges due to pressure in the packing is given by

$$F_2 \quad = \quad \frac{\pi}{4} [D_1^2 - D^2] \; P \qquad (19)$$

Where D_1 = outside diameter of the packing.

:. Total force acting to separate the two flanges,

$$F = \quad F_1 + F_2$$

$$= \frac{\pi}{4} D^2 . P + \frac{\pi}{4} [D_1^2 - D^2] P$$

$$\Rightarrow F = \frac{\pi}{4} (D_1)^2 p \qquad (20)$$

Since a square flange is fastened by means of four bolts, therefore, the load taken up each bolt, F_b is:

$$F_b \quad = \frac{F}{4} \qquad (21$$

If d_e is the core diameter of the bolts, then

$$F_b \quad = \quad \frac{\pi \; d_c^2}{4} \delta_{tb} \qquad (22)$$

Where δ_{tb} = Allowable tensile strength of the bolt material.

The value of δ_{tb} is usually kept low for initial tightening stress in the bolts. After the core diameter is obtained, then the nominal diameter is chosen form tables.

The thickness of the flange may be obtained by considering the bending of the flange about one of the sections A - A, B - B, C- C. as shown in the diagram. A little consideration will show that the flange is weakest in bending about section A - A. Therefore, the thickness of the flange is calculated by considering the bending of the flange, about A.

The axes of the bolts are arranged at the corners of a square of such as size that the corners of the nut clear the outsider diameter of the cylinder.

:. Minimum length of a diagonal for this square, L is given by:

L = Outsider diameter of the cylinder+ 2 x diameter of the bolt. => $L = D_o + 2d1$ (23)

The side of this square is given as:

$$L_1 = \frac{L}{\sqrt{2}} \qquad (24)$$

The sides of the flange must of sufficient length to accommodate the nuts and bolt head without overhang. Therefore, the length l2 may be kept as

$$L_2 \quad = L_1 + 2d_1 \qquad (25)$$

As mentioned earlier, the thickness of the flange can be found by considering the bending of the flange about section A- A

The bending is brought about due to the force in two bolts.

.: Bending moment due to the force in the two bolts (i.e. due to 2fb) is given by:

$$M_1 = 2Fb \times \frac{L_1}{2} \qquad (26a)$$

Bending moment due to the fluid pressure inside the cylinder,

$$M_2 = 2 F_b \times \text{Centriod of arc from A - A .} \qquad (26b)$$

Since the bending moment M_1 and M_2 are in opposite direction, therefore, net resultant bending moment on the flange about section A - A:

$$M = M_1 – M_2 \qquad (27)$$

The width of the flange at the section A- A

$$B = L_2 - \text{Outside diameter of pipe.} \qquad (28)$$

Let tr = Thickness of the flange in mm

Section modulus,

$$Z = \frac{b_t{}^2}{6} \qquad (29)$$

2.5. PISTON DESIGN

The required piston rod column size necessary to sustain apply load that is the alignment with the centre - line of the cylinder bore is influenced by the strength of the rod material, the force applied to the rod column in compression, the mounting situation of the cylinder itself and the stroke under which the load is to be applied.

As the rod column length is increased the tendency to deflect and buckle under loading is also increased, this is in fact the limiting factor in the design of piston and cylinders, [8].

The procedure to compute piston rod column sizes and cylinder length under the end thrust condition is accomplished using the following:

1. Determine the column length factor from the mounting situation.
2. Calculate the correct length of the rod the relation:
    Corrected rod - Actual stroke x column length factor
3. Determine rod that will be imposed on the end of the column using the piston bore size and the maximum relief valve pressure using the formula:
    Force, $F = \text{Pressure} \times \text{Area}$ \qquad (30)

2.6. DESIGN OF SEALS:

2.6.1. Design of Static Seals

Static seals are designed to fill the space between two attached parts. The method of attachment is not in itself of primary importance to the seal as long as it provides the compressive force necessary to effect a reliable seal between the surfaces of the attaching parts. This is usually expressed in kg per linear mm of seal.

The seal across section, material, seal hardness and the percentage of seal deformation, all affect the compressive force.

2.6.2 Design of Dynamic Seals

Dynamic seals prevent leakage between two surfaces moving relative to each other under condition of varying pressure.

Seal designs have as their objective to prevent leakage under varying condition of pressure and speed as other operating conditions change, and at the same time to extend the life of the seal as long as possible. To accomplish this initial seal must be maintained when the ring is installed by having the design exert a radial force on the ring. As system pressure is increased additional radial as well as longitudinal forces are applied to the ring to increase its sealing capability, [8].

Two factors interact to affect seal design, they are: the coefficient of friction and the operating speed.

The coefficient of friction is largely dependent on the surface finishes of the seal and the metal surface against which it rest.

The initial forces necessary to affect a seal between the lips of the seal and the metal cylinder, rod or housing are caused by a slight deformation of the seal. System pressure increases the sealing force on the lips proportionally to affect a seal directly related to need.

2.7. Design of Platen

The upper and lower platens are subjected to bending stresses during operation. The arrangement is analogous to a simply supported beam carrying a point load at the midpoints of the supports. Hence the design consideration is essentially bending,

2.7.1. Section Modulus

The maximum stress in a beam section is given by

$$\delta = \frac{M. \, Ymaz}{I} \qquad (31a)$$

Where M = maximum bending moment (Nm)
I = moment of inertia (m^4)
Ymaz = maximum reflection of the beam section
But $\quad I/ymaz = Z$ \qquad (31b)
Where Z is the section modulus.
Thus, $\quad \delta = \dfrac{M}{Z}$ \qquad (32)

For a rectangular section, with breath b, and depth d,

$$Z = \frac{\frac{bd^3}{12}}{\frac{d}{2}}$$

$$\rightarrow \quad Z = \frac{bd^2}{6} \qquad (33)$$

.

2.8. Design of Pump

A pump is a device for raising the pressure of a fluid. The parameters involved in the proper design or the pump are:

   i.   Maximum discharge pressure
   ii.   Volumetric discharge

The energy or work equivalent added to the fluid by the pump can be accounted for the total volume of fluid exiting from the pump outlet and the pressure at which it in discharge i.e.

$$\text{Total Energy} = \text{pressure x volume} \qquad (34a)$$
$$= \frac{\text{Force}}{\text{Area}} \text{ x Area x Distance}$$

$$\rightarrow \quad \text{Total Energy} = \text{Force x Distance} \qquad (34b)$$

The total energy volume discharge by the pump per stroke equals the area of the cylinder bore multiplied by the stroke.

In practice, the total fluid energy output from the pump always exceeds the total energy available to the cylinder because of friction head losses within the system.

The determination of the correct pump for a given application essentially involves matching the pump to the required fluid power at the actuator giving consideration to pressure and flow rate.

The piston pump employed here essentially consists of a cylinder barrel and a piston, which reciprocates in the cylinder creating alternate suction and forcing stroke.

2.8.1. Pump Lever Design

A lever is a machine whereby a load is supplied at one end (called an effort) is used to overcome a larger load at the other end.

In this design, the pumping action is achieved by a lever system. Assuming a maximum theoretical effort and since the maximum force to be attained is known, then

$$\text{Pressure} = \frac{\text{Force, F}}{\text{Area, A}} \qquad (35)$$

Since the fluid pressure is constant throughout the system (Pascals's law), then

$$P = \frac{F_1}{A_1} = \frac{F_2}{A_2} = \frac{F_3}{A_3}$$

$$F = \frac{F_1 A_2}{A_1} \qquad (36)$$

$$F_2 = \frac{F_1 D_2{}^2}{A_1} \qquad (37)$$

From the lever system which represents the pumping arrangement, talking moment about the fulcrum O, and using the rule:

  Clockwise moment =   Anti clockwise moment

$$F_1 \text{ X AO} = F_2 \text{ X BO}$$

$$AO = \frac{F1 \text{ X BO}}{F_1} \qquad (38)$$

2.9. Instrumentation

Instrument serves to monitor the operation of the fluid power system by sensing and reading out relevant parameters for example pressure, fluid direction and rate of flow. They vary in sophistication from simple visual device to electronic sensor that uses amplifiers to drive cathode ray tubes and high-speed oscilloscope.

At the pump outlet, pressure is monitored via high pressure gauges to asses condition and the pressure valve of the load resistance. The Bourdon pressure Gauge is most commonly found in use in all pressure ranges because of its simplicity, cost and robust construction, [10].

2.10. Cylinder Barrel Construction

This is a very important component of press. The capacity and efficiency of the machine is dependent on it.

2.10.1 . Material Requirement

A mild steel 25mm cylinder and 15mm thick was brought from the local market with a length of 62Omm. A square plate 20mm thick, 200 x 200mm of steel was also acquired for the flange.

2.10.2 Operational Sequence

One end of the cylinder was held in a three-jawed chuck of the centre lathe and the other end was faced to ensure straightness.

Employing a boring tool of convenient length, the cylinder was bored to the designed size. However, the last cut was taken at high cutting speed and low feed rate. This was to ensure very smooth surface finish, which is an important characteristic of hydraulic cylinders.

At the bottom part of the cylinder, after boring a step was bored to provide for the seal groove to a depth specified in the design drawings.

A whole of diameter 140mm was then cut at the centre of the flange and both of them were welded together and subsequently ground to ensure smooth fitting of the cylinder base on the base plate.

2.11. Piston Construction

One end of the hallow pipe was held in the three-jaw chuck of the lathe and the other end was turned to the required size after facing.

2.12. Pump Construction

This is the heart of the hydraulic system as it provides the motive power to the working fluid.

2.12.1. Material Requirement

i. 30mm mild steel rod X 220mm length.
ii. Two steel balls, 15mm
iii. Two M12 nuts
iv. 20mm mild steel rod X 120mm length.

2.12.2. Operational Sequence

The rod was cut into two lengths of 100mm and 120mm. on the 100mm long rod, a hole of ø20mmlength of rod ø8mm hole was as well drilled right through its whole length and the hole were enlarged to 13mm diameter, 60mm from the front end and40mm from the rear end. The two end of the rod were then chamfered at 40° and the two M12 nut welded to them as indicated in the design drawings.

2.13. Performance Test Procedure:

The performance test started with the priming of the pump. After which the fluid was pumped, initially slowly. This was carried out under no load condition. The machine was left to stand in this position for two hours.

The machine was then subjected to a load of 190kN. This was provided by a compression spring, which has a spring constant of19N/mm, the machine was arranged with the springs between the two platens and then compressed to two hours to observe for leakage. Leakages in the system were indicated by the lower platen failing to position, [5].

3. Results and Discussions

3.1. Design data for the Cylinder and its critical Parts

Note: - Maximum load= 60 tons (600KN), Piston Diameter (Di), =101mm, Length of stroke =190mm, Safety factor = 2.5, Stroke= 4.

3.2. Cylinder design:

i. Piston Area, A_p = $\dfrac{\pi D_i^2}{4}$ (39)

$$= \frac{\pi \times (101 \times 10^{-3})^2}{4}$$

$$= 8.011 \times 10^{-3} \text{m}^2$$

Maximum pressure developed in the cylinder to overcome the load resistance of 300KN is computed from equation (36)

$$\text{Pressure} = \frac{\text{Force}}{\text{Area}}$$

$$= \frac{600,000}{8.011 \times 10^{-3}} = 7.49 \times 10^7 \text{N/m}^2$$

ii. To determine the minimum wall thickness of the cylinder to withstand the maximum internal pressure. From

iii. equation (11) and taking a factor of safety = 2.5

$$t = r_i \left\{ \sqrt{\frac{\delta}{\delta_t - 2P}} - 1 \right\}$$

$$50.5 \times 10^{-3} \left\{ \sqrt{\frac{460 \times 10^6 \times 2.5}{460 \times 10^6 \times 2.5 - 2 \times 7.49 \times 10^7}} - 1 \right\}$$

$$= 0.0195\text{m}$$

∴ a wall thickness of not less than 20mm can be considered adequate for the design.

iv. The maximum external diameter D, for the cylinder is given by;

$$D_o = D_i + 2t$$
$$= 101 + 2(20) = 141\text{mm}.$$

Thus a cylinder of external diameter 150mm and thickness of 14mm made of mild-steel was selected for the design.

3.3. Flat-end - Cover Design:

From equation (12), the minimum thickness, T of the end cover plate is given as:

$$T = KD_i \sqrt{\frac{P}{\delta_t}}$$

Taking K = 0.4

$$= \sqrt{0.4 \times 101 \ 10^{-3} \quad 7.49 \times 10^7/460 \times 10^6]}$$

Thus, a mild-steel plate of 16mm thickness was selected for the design.

3.4. Cylinder Flange Bolt Design

$$\frac{\pi}{4} \times (101 \times 10^{-3}) \times 7.49 \times 10^7 = \frac{\pi}{4}(19 \times 10^{-3})^2 \times 460 \times 10^6 \times n$$

$$\therefore n = 5.108$$

Here n is odd, therefore, the next higher even design.

3.4.1 Design of Cylinder Flange

We have already computed that t = 19mm and D_O = 141mm

The force trying to penetrate the flanges

$$F = \frac{\pi}{4}(D_o)_2 P$$

$$= \frac{\pi}{4}(141 \times 10^{-3})^2 \times 7.49 \times 10^7$$

$$= 1169526.116N$$

Since this force is to be resisted by 6 bolts,

$$F_b = \frac{F}{6}$$

$$F_b = \frac{1169526.116}{6} = 194921.02N$$

Let d_e be the core diameter of the bolts. The force in each bolt is given by;

$$194921.02 = \frac{\pi(de)^2}{4} \sigma_{tb}$$

$$194921.02 = \pi \cdot d_c^2 \cdot \frac{460}{4} \times 10^6$$

$$d_e = 0.0232m = 23mm.$$

The axes of the bolt are arranged at the corners of a square of such size that the corners of the nut clear the outside of the pipe.

Minimum length of diagonal for this square

L = outside diameter of cylinder + 2 core diameter of bolt

$$= 141 + 2(23)$$

$$= 187mm$$

And size of this square

$$L_1 = \frac{L}{\sqrt{2}}$$

$$\frac{187}{\sqrt{2}} = 132.2mm$$

The sides of the flange must be of sufficient length to accommodate the nuts and bolt head without overhang.

∴ The length L_2 should be kept i.e.

$$L_2 = L_1 + 2dc = 32.2 + 2(23) = 178\ 2mm$$

The elevation of the flange was made so, that in order to find the thickness of the flange; I considered the bending of the flange about Section A- A. This will take place due to the force in the bolts.

∴ Bending moment due to the force in two bolts

(i.e. due to .2Flo)

$$M_1 = 2F_b \times \frac{L_1}{2}$$
$$= 2 \times 194921.02 \times (132.2/2) \times 10^{-3}$$
$$= 25768.55\ Nm.$$

The hydraulic fluid pressure acting on one half the flanges

$$= 2 F_b$$
$$= 2 \times 194921.02 = 389842.04N/m^2$$

The centroid of this arc from A -A = 41mm

The bending moment due to fluid pressure

$$M_2 = 2F_b \times 41$$

$$= 2 \times 389842.04N \times 41 \times 10^{-3}$$

$$= 15983.52\ Nm$$

Since bending moment M_1 and M_2 are in opposite directions, therefore, net resultant bending moment on the flange about section A - A:

$$M = M_1 - M_2$$
$$= 25678.55 - 15983.52$$
$$= 9785.03\ Nm$$

Width of flange at A-A:

$$b = L_2 - \text{Outside diameter of cylinder}$$
$$= 178.2 - 141$$
$$= 37.2\ mm$$

Let t, = thickness of the flange in mm

∴ Section modulus,

$$Z = \frac{bt_f^2}{6}$$
$$= 1/6 \times 37.2 \times 10^{-3} \times t_f^2$$
$$= 6.2 \times 10^{-3} \times t_f^2$$

We know that net resultant bending moment (M) is given by:

$$9785.03 = 460 \times 10^{-3} \times 6.2 \times 10^{-3} \times t_f^2$$

$$\therefore t_f = 58.6mm$$

3.5. Piston design

i. The column length factor can be computed as:
   Stroke x 4.0 = 190 x 4.0
   = 760 mm
ii. The corrected length of the rod is therefore given as:
   Actual stroke x column strength factor
   = 190 x 4.0   = 760mm

iii. Determine the appropriate piston rod diameter, locating the next higher value of corrected rod length opposite the maximum load does this.

For a load of 600KN, the piston rod diameter is taken to be 96mm.

Thus, a piston rod of external diameter of 50mm vi/as selected for the design to reduce weight.

3.6.   Design of Static Seal:

The size of the groove into which the seal seats is 4 x 3mm. To form the necessary zero clearance to seal, the static seal was compressed to 15, i.e 30% of its original volume.

3.7.   Design of Platens

To determine the reaction at the supports A and B; Taking moments about A

$$0.3 \times 600,000 = 0.6 \times R_B$$
$$\therefore R_B = 300,000 \text{ N.}$$

Taking moment about B:

$$0.3 \times 600,000$$
$$\therefore R_A = 300,000N.$$

To compute the bending moment M:

Cutting at point C between A and O and summing vertical forces, shear, $V = 300,000N$

Taking moment about C,

At $X = 0$, $M = 0$

At $X = 0.3$, $M = 0.3 \times 300,000$

$$= 90,000Nm$$

Cutting at point D between O and B, and considering the free body DB

Sum of vertical components

$$300,000 - 600,000 = -300,000$$

Taking moment about D,

$$(0.6 - x) \times -300,000 = 0$$

At $x = 0.3$, $M = 0.3x - 300,000 = -90000Nm$

At $x = 0.6$, $M = 0$.

From which we obtain that, the maximum bending moment is given as 90000Nm and it occurs at the center of the platen.

4.1.7.2 Platen thickness Calculations:

To determine the thickness of the platen that will resist the applied load, i use the concept of Section modulus. From equation (33)

$$\sigma = M/2$$
$$Z = \frac{M}{\sigma}$$

But we know that $t = 460 \times 10^6 N/m^2$

$$M = 90000Nm.$$

$$\therefore \quad Z \quad = \quad \frac{90000 \times 2.5}{460 \times 10^6}$$
$$= \quad 4.89 \times 10^{-4} \text{ m}^3$$

From equation (34)

$$Z \quad = \quad \frac{bd^2}{6}$$

$$\therefore \quad d \quad = \quad \sqrt{\frac{Z \times 6}{b}}$$

but b = 600mm

$$\Rightarrow \quad d \quad = \quad \sqrt{\frac{4.86 \times 10^{-4} \times 6}{600 \times 10^{-3}}}$$

$$= \quad 0.0699m \sim 70mm.$$

Therefore, a platen thickness of not less than 70mm is considered sufficient for the design.

3.8.   Design of Pump-lever:

The maximum force to be overcome as the load resistance is given as 600,000N.

From equation 38,

$$F_2 = \frac{F_1 \times d_2^2}{d_1^2}$$

Where F_1 = Effort of operator

F_2 = Load to be overcome

Thus,

$$F_2 = \frac{600,000 \times (20 \times 10^{-3})^2}{(100 \times 10^{-3})^2}$$
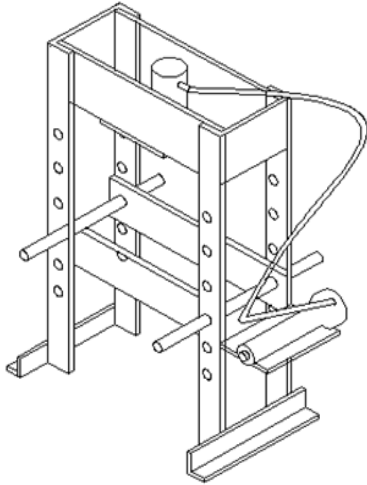$$= 24,000N.$$

From the lever system of the pump, taking moment about the fulcrum O, and using the rule: Clockwise moment = Anticlockwise moment.

$$F_1 \times AO = F_2 \times BO \qquad \qquad .(40)$$

Assuming a maximum theoretical effort of 200N, therefore:

AO = <u>24,000</u> x 10mm
         200

AO = 1200mm

Fig1: shows the hydraulic cylinder mounted on the body of the 60tons hydraulic press.

4. CONCLUSIONS

From this design work, it is evident that hydraulic cylinders which are use for powering of machines and other engineering works can be produce in Nigeria. From the performance test, the force from the pump to the ram pulled bearing from its casing; also, deformation work on metals was successfully done by the hydraulic press whose power came from the hydraulic cylinder. The performance showed that the system performed effectively and efficiently. The machine is made up of simple components that can be assembled with some certain skills especially the fixing of the cylinder seals. This seal aspect needs to be carefully handled in cylinder design work.

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