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Design Analysis and Testing of a 10-Ton Hydraulic Press

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***Abstract*—This paper presents the design and analysis of a10-ton, H-frame hydraulic press with incorporated force measuring device. Existing presses were investigated to identify the problems and challenges resulting in the use of these machines. The materials were sourced locally for the design. The major parts were designed, and analyzed using Finite Element Analysis. The device was then fabricated with the designed dimensions and tested by using it to compress selected materials. Based on the test results, modulus of elasticity were then estimated and compared with literature values for the tested materials. Also, during the test, the hydraulic cylinder was checked for leakages and frame was checked for other failure modes. No leakages were detected in the hydraulic cylinder and the platens did not show any noticeable deformation. The developed hydraulic press performed satisfactorily and can suitably be used to compress materials and estimate its compressive stress.**

Keywords—Young modulus; Finite Element Analysis; compressive strength

I. Introduction (*Heading 1*)

The hydraulic press is a pressure exerting device that uses hydraulic fluid to transmit and magnify forces or pressure. This device has a wide range of application in the industrial workshop and laboratories. They are used for different operations such as blanking, piercing, bending, shearing, forming and metal testing (1). Their preference over other force magnifying device is due to the large nominal force they can produce, ease with which it can be controlled and the availability of the whole magnitude of force throughout the working stroke (2; 3).

Over the years, some researches into hydraulic presses have been reported. The development of hydraulic press dates back to the late 18th century with Joseph Bramah & William George Armstrong being the two pioneers (4). Although most of the existing presses are electrically powered and not suitable for small workshop, there are a few that have been developed to be operated manually (3; 5). The areas of energy dissipation and sources of carbon emissions in hydraulic presses have been studied and identified with an intention to improve energy efficiency and reduce carbon emissions (6). A methodology for designing and analyzing the performance of the welded areas for a 150-ton press was discussed in the work of (7).

There are some reports that present optimization in the design of hydraulic press. The reduction of the volume via optimization of construction material for the device's H-frame was reported in the work of (8). Also, the optimization of a 250-ton press with four pillars with a focus on the top plate of the device and validated with Finite Element Analysis has been reported (4). The design and optimization of the basic elements of the hydraulic machine using Finite Element Analysis were also reported by (9). Unlike the work of More et al. (8) that focused on just the top plate, he (9) focused on the top plate, movable plate and column design. A review paper on the design and analysis of hydraulic presses was presented by (10).

Despite several reports on developed hydraulic presses, there is a dearth of information on the design of the machines for purposes of mechanical testing. This work focus on the development of a hydraulic press for compression with the capability to measure compressive forces. A hydraulic press with the capability to estimate compressive forces will be suitable for testing compressive strength for different materials.

II. MATERIALS AND METHOD

A characteristic hydraulic press comprises a frame that supports the load of compression; a pump, which is a source of motive power or motion for the fluid; the hydraulic fluid, which presents the medium of transmission of power; hydraulic pipes and connectors, through which the fluid under pressure flows; control devices, like valves that affect the direction of flow and the operating pressure; and the hydraulic actuator or a motor that converts the hydraulic energy produced into useful work (1; 3)

The device will be designed to operate on a maximum load of 10-ton. The frame will be designed from first principles while other core components will be selected based on their suitability for the machine. For this design, the hydraulic power is sourced from a 10-ton bottle jack.

A. *Materials*

The body is conceptualised as an H-frame, with a flat top. It has a support plate for the hydraulic jack, a crank rod and a flat plate for the electric motor. The huge amount of concentrated force and high pressure on which the machine operates are the factors that guided the choice of materials selected. Given the nature of the project, mild steel (IS2062 or AISI 1020) was selected for the H-frame, since it is readily available. Additionally, it is soft and ductile, thus can be easily fabricated for structural applications. So IS2062 (AISI 1020) was selected for all parts of the frame. The properties of the AISI 1020 can be found in the literature.

B. Hydraulic Cylinder Design:

For this component, a standard part (a 10-ton hydraulic jack) was purchased. The bottle jack is capable of producing the required maximum output force of 10 tons.

C. H-frame Design:

The H-frame is the base element of the hydraulic press machine. Its main purpose is to withstand the maximum force generated by the hydraulic system and as such, is subjected to both direct tensile/compressive stress and bending stress. For the H-frame, the following assumptions are made:

1. The load is considered perfectly vertical, and assumed to act at a point.
2. Frame material is homogeneous.
3. The base is fastened to a rock-hard groundwork or base such that all deflections at the base are equal to zero.
4. The frame has a symmetrical cross-sectional area.

Both the vertical and horizontal components are of standard mild steel c-channels (shown in Fig. 1) and flat mild steel plates. This aspect of the design considers the stresses induced the horizontal and vertical beams of the machine

1) Stresses induced in the vertical beams*:*

The design consideration includes four (4) 3 X 5 (ANSI standard inch) c-channel vertical support beams each of length 1524 mm; two in the front and two at the back. The c-channels at the back are separated from those in the front by 200 mm each while, the c-channels on the left are separated from those on the right by 762 mm.

All four (4) posts (or beams) have six (6) evenly spaced holes each of 15 mm radius. The holes accommodate the shafts that support the lower platen (or vertical moveable component).

The stress induced in the four (4) horizontal beams is purely axial stress but is negligible since the same applied force tends to extend and compress the shafts.

2) Stresses induced in horizontal beams:

The horizontal components include three (3) components: the fixed upper platen, the movable platen (supporting the hydraulic bottle jack), and the adjustable lower platen (which carries the workpiece). The arrangement is such that the hydraulic jack has its crown set against the top platen and its base fastened to the moveable platen. As the ram of the hydraulic jack extends, the moveable platen is lowered to meet with the lower platen and hence the workpiece. This explains why both top and lower platens are subjected to pure bending stresses, and hence the deflection at maximum force needs to be known.

The top platen consists of two standard 3 X 5 (ANSI inch) c-channel positioned horizontally and welded to the ends of the top of the vertical beams (one in front and one at the back). The centre of the top platen is crossed with a 150 X 100 mm steel plate of 6mm thickness that serves as a resting place for the crown of the hydraulic jack. This plate, however, is subjected to compressive stress since it is located between the hydraulic jack and the top platen. The stress induced in the plate is calculated as shown below:

Stress $\sigma_{max} = \frac{F_{max}}{A}$ (1)

Where: $\sigma_{max} = maximum\ stress\ induced$

$F_{max} = maximum\ force\ applied$ = 10 tons = 98066.5N

A = plate area; L=150mm; B= 100mm; A = 150*100 = 15000 mm^2; $\sigma_{max} = -6.537\ MPa$

The negative sign indicates compressive stress. This value of stress provides for a safe design given that the material used can withstand up to 420 MPa.

The bending stress and hence deflection of the top and bottom platens are assumed to be equal but in opposite directions, since they are made out of the same material.

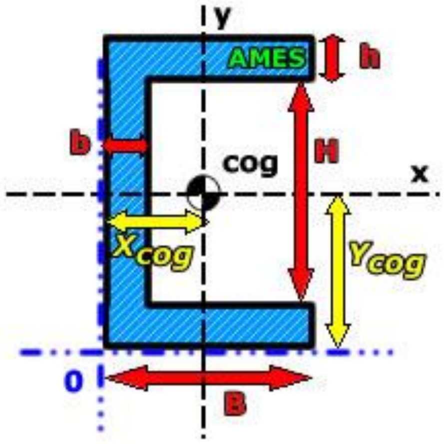

Fig. 1: sectional properties of c-channel beam

H= 76.2 - (2*7) = 62.2mm; h= 7mm; B= 38.05mm; b= 6.55mm;

Y_{cog} = 38.1; X_{cog} = 19.025

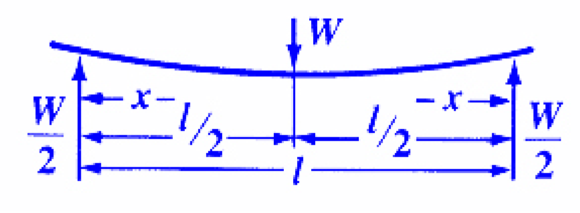

Fig. 2: Deflection for a beam supported at both ends with load at the centre

Considering the deflection to be as shown in Fig. 2, the maximum deflection is given by equation 2.

$$y = \frac{Wl^3}{48EI} \quad (2)$$

Where; W= max. load = 98066.5N

l= length of c-channel beam = 762mm

E= Young's modulus of channel material = 205GPa

I = area moment of inertia of section symmetrical with stress plane = Iyy

The moment of inertia for a c-channel beam is calculated using equation 3.

$$I_{xx} = \frac{H^3b}{12} + 2\left[\frac{h^3B}{12} + \frac{hB(h+H)^2}{4}\right] \quad (3)$$

$$I_{xx} = 77152.688mm^4$$

$$I_{yy} = \frac{b^3H}{12} + bH\left(X_{cog} - \frac{B}{2}\right)^2 + \frac{2B^3h}{12} + 2Bh\left(X_{cog} - \frac{B}{2}\right)^2 (2)$$

$$I_{yy} = 122992.93mm^4$$

Hence, at a maximum load of 10 tons, the maximum possible deflection obtainable is obtained by substituting the values of W, l, E and I into equation 2.

$$y = \frac{Wl^3}{48EI}$$

$$y = \frac{98066.5 * 762^3}{48 * (205 * 10^9) * 122992.93}$$

$$y = 3.585 * 10^{-5}mm$$

The result obtained above shows that the selected material will not show noticeable deformation under a maximum load of 10 tons (98066.5N).

3) Stress-induced in shafts used to support lower platen:

The working condition of the above shafts is depicted by the bending of a beam with ends overhanging, supports and two equal loads applied at symmetrical locations as shown in Fig. 3.

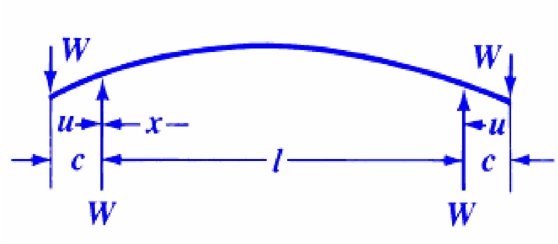

Fig. 3: Structural beam deflection for a beam with ends overhanging, support and two equal loads applied at symmetrical locations.

Where: W= applied load;

l= distance between supports;

c= distance between support and load;

E= modulus of elasticity;

I= moment of inertia,

The deflection at center of beam is computed using equation 4.

$$y = -\frac{Wcl^2}{8EI} \quad (4)$$

W= 49033.25N; c= 60; E= 205GPa; l=150;

d= diameter of circular cross-section = 27mm

$$I = I_{xx} = I_{yy} = \frac{\pi d^4}{64}$$

$$I = \frac{\pi(27)^4}{64}$$

$$I = 26087.04908mm^4$$

$$y = -\frac{49033.25 * 60 * 150^2}{8 * (205 * 10^9) * 26087.04908}$$

$$y = -1.547 * 10^{-6}mm$$

4) Design of tension springs used for supporting moveable paten

Compression springs are required for the sole purpose of retracting the ram of the hydraulic jack when

pressing is complete. It also serves as a guide for the plate to which the base of the hydraulic jack is fastened. A pair of tension springs are required for even distribution of load. The springs are expected to bear the load of the hydraulic jack and base plate at solid length. The design of the spring is shown below with the following assumptions:
Wire material: phosphor-bronze

Winner diameter of spring: ID = 22.5mm

Outer diameter of spring: OD = 29.5mm
Spring wire diameter: d = 3.5mm

Solid length of spring: L_b = 146.5mm

Number of coils: N_b= 40

$$mean\ coil\ diameter\ D = OD - d$$

$$D = 26mm$$

$$spring\ index, c = \frac{D}{d} = \frac{26}{3.5}$$

$$c = 7.429$$

$$Bergsrasser\ factor\ K_b = \frac{4c+2}{4c-3} = 1.187$$

$$number\ of\ active\ turns\ N_a = N_b + \frac{G}{E}$$

The values of G and E correspond to table value of N_b

$$N_a = 40 + \frac{41.4}{103.4} = 40.4\ turns$$

$$spring\ constant\ k = \frac{d^4 G}{8D^3 N_a} = 1094\ N/mm$$

$$free\ length\ of\ spring, L_0 = (2c - 1 + N_b)d$$

$$L_0 = 188.5mm$$

The deflection of spring under service load is given as:

$$Y_{max} = \frac{F_{max} - F_{min}}{k}$$

Where F_{max} = 98066.5N

F_{min} = 160N (initial tension in spring)

$$Y_{max} = 89.5mm$$

$$total\ spring\ length = L_0 + Y_{max}$$

$$L = 278mm$$

D. Computer-Aided Design

On completion of the designs calculations, the frame was drawn in 2D orthographic projection shown in Fig. 4.

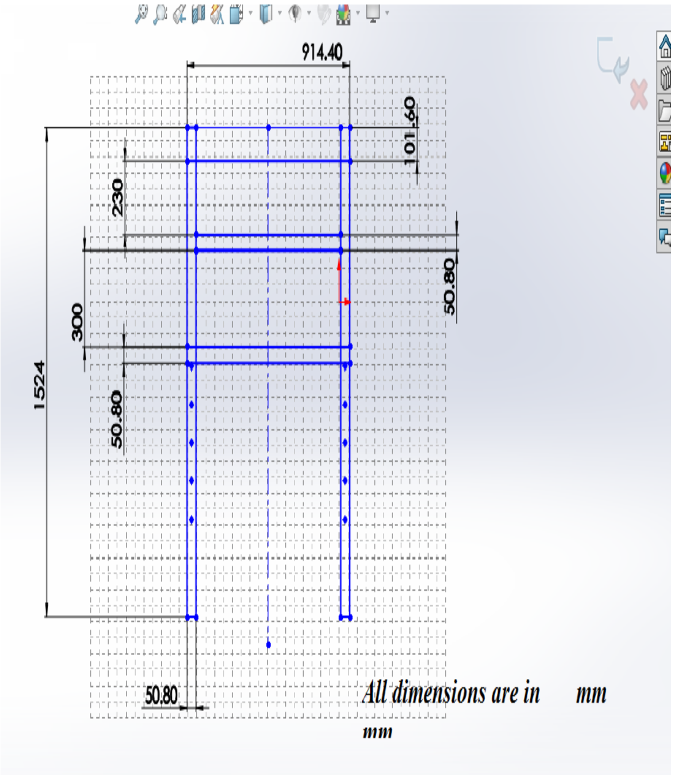

Fig. 4: A 2-d model of the designed frame showing general framework dimensions

E. Finite Element Analysis

The 3D geometric model was produced, and it is shown in Fig. 5. The generated model was then extracted to ANSYS for Finite Element Analysis

2.6 Testing of the device
Operation Procedure

1. Connect the machine to the main power supply and make sure both control switches are in the 'off' position.
2. Turn on the control switch for the pressure gauge and turn on the gauge. Reset the gauge till it reads "0.00" and ensures the unit is set to 'MPa.'
3. Measure and record the dimensions and cross-sectional area of the material to be tested.
4. Close the check valve behind the hydraulic jack and turn on the switch for the electric motor. Turn off the switch for the motor when the hydraulic jack support just touches the material.
5. Turn on the displacement sensor and measure the distance. Hold down the +/- sign on the device till a (–) sign shows at the centre of the display.
6. Turn on the switch for the motor again and allow it to run till completion of compression, then turn off the switch for the motor.
7. During the compression, continually take maximum pressure readings from the gauge.
8. After compression, press the measure button on the displacement sensor again and record the result.
9. Open the check valve behind the hydraulic jack to release the pressure in the hydraulic jack, turn off

the switch for the pressure gauge and turn off the displacement sensor.

10. Disconnect the machine from the mains and remove the compressed material.

The test materials were prepared according to the ASTM standard. Information on the different materials tested and their standard dimension is shown in Table 1. Also, their modulus of elasticity as obtained in the literature presented. Samples of the prepared specimens were mounted on the device, and their maximum pressures, as well as a change in dimension, were recorded. The maximum strain obtained as well as the modulus of elasticity for the different material samples were then determined.

Table 1: a table showing the dimension of different test materials

Test material	Dimension (mm)	Modulus of Elasticity (theoretical) (Gpa)
Aluminum	L=50; b=35; h=70	69 – 75
Mild steel	d=80; l=167	200 – 207
ABS Plastic	d=65; l=142	1.4 – 3.1
Wood	h= 120; l=60; b=80	1.15 – 62

III. RESULTS AND DISCUSSIONS

The stress distribution on the frame is shown in Fig. 6. As observed, the stress is highest in the upper part of the frame, as indicated by red color, being up to 46.82 MN/2m^2 in compression. Some parts of the frame, such as the side beams are, however, undergoing tensile stress. Their values are negative, being bluish, as indicated by the color grid. The values of the stress obtained, for the different section is far less than the yield stress of the material, which is approximately 351.6 MN/m^2. Therefore the frame which is the most critical part of the machine is not likely to fail under maximum loading condition.

To test the device, it was used to compress selected engineering materials. The measured pressure load and change in dimension are shown in Table 2. It is worth noting that the changes in dimensions in the lateral directions are negligible. As shown, no observable change in dimension was found with mild steel and Aluminum materials in the longitudinal directions; thus recorded as 0 MPa. ABS plastic showed the maximum deformation, having a value of 3.1 mm followed by that of wood which was 0.5 mm. The maximum crushing pressure obtained using the device was 97.42 MPa. The crushing pressure for ABS plastic and wood were less than these values, being 56 and 78 MPa respectively. The maximum compressive force being 10 ton designed value is not sufficient to compress mild steel and Aluminum materials, but can effectively compress other materials such as ABS plastics and wood.

The effectiveness of the device was accessed via the computation of modulus of rigidity using experimental results of pressure load and change in dimension after compression. Fig. 8 shows the calculated modulus of rigidity for ABS plastic and wood and their respective average theoretical values. The experimental value of ABS plastic is very close to that in the literature with the error being approximately 17%. The difference in the value between theoretical and experimental values of modulus of elasticity for wood was as high as 40 %. The large range of modulus of rigidity for wood materials which varies with moisture content and type perfectly justifies this disparity. Acceptable values of estimated modulus of rigidities confirm the effectiveness of the machine in measuring forces of compression for materials.

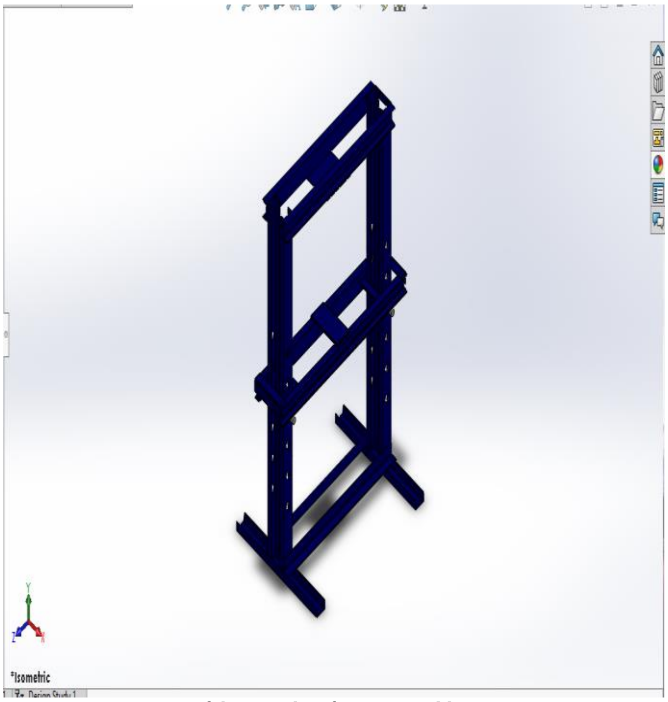

Fig 5: Isometric view of the complete frame assembly

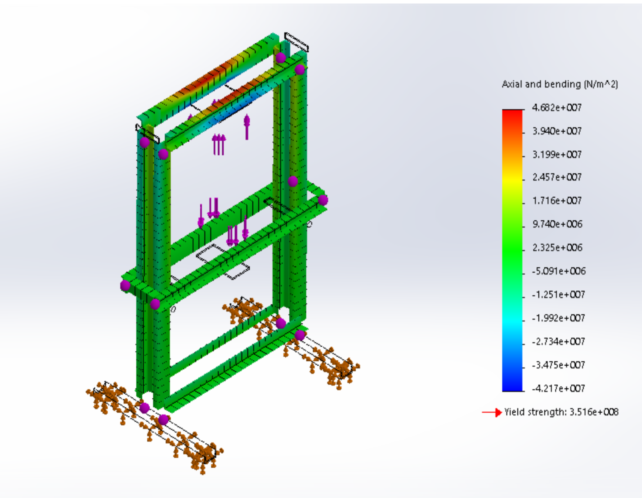

Fig. 6: Stress distribution of the designed hydraulic press frame when subjected to maximum loading condition

Fig. 7: Constructed device

Table 2: a table showing readings obtained with the developed press

Test material	Final pressure reading (MPa)	Change in length (mm)
Aluminum	97.42	0
Mild steel	97.42	0
ABS Plastic	56.179	3.1
Wood	78.49	0.5

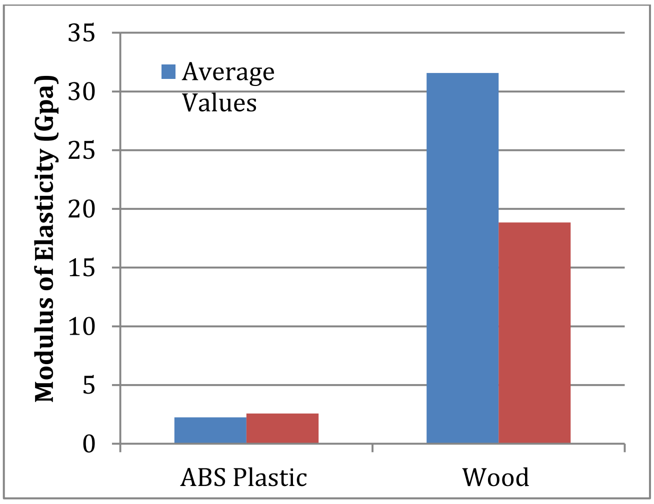

Fig. 8: Estimated and actual modulus of rigidity for ABS plastic and Wood material

IV. CONCLUSION

To conclude, a Hydraulic Press Machine capable of compressing materials under known forces was designed and fabricated. The designed machine had an embedded digital display of compressive pressure. The machine performed satisfactorily, with a capability that allows estimation of the young modulus. The developed machine will be suitably applied in small and medium scale workshops. Additionally, it can effectively be used to ascertain the compressive strength for different low strength materials.

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