Computational Engineering and Physical Modeling 3-3 (2020) 46-58

Contents lists available at **CEPM**

Computational Engineering and Physical Modeling

Journal homepage: www.jcepm.com

Design, Fabrication and Structural Analysis of a 5 Tons Hydraulic Press and Mould Machine for Crucible Production

O.O. Ojo[1,2*], M.O. Olaleke[1], O.A. Dahunsi[1]

1. Mechanical Engineering Department, Federal University of Technology, P. M. B. 704, Akure, Ondo State, Nigeria

2. Industrial and Production Engineering Department, Federal University of Technology, P. M. B. 704, Akure, Ondo State, Nigeria

Corresponding author: *ojooladimeji90@yahoo.com*

https://doi.org/10.22115/CEPM.2020.237362.1113

ARTICLE INFO

Article history:
Received: 29 June 2020
Revised: 17 September 2020
Accepted: 17 September 2020

Keywords:
Design;
Hydraulic Press;
Crucible;
Compaction;
Stress analysis.

ABSTRACT

A hydraulic press machine comprising of the frame, cylinder, hand pump, and pressure gauge was designed, fabricated, and evaluated. The machine is manually operated and the frame was modeled using the Solidwork application software 2018 version. The maximum stress, maximum displacement, maximum strain, and factor of safety of the machine are 97.09 MPa, 0.337696 mm, 0.000361326, and 2.32 respectively. The performance of the developed machine was successfully carried out by using it to mould clay of different sieve sizes (0.6-4.75 mm) into crucible form with an applied maximum pressure of 100 bar. The results showed that the smaller the size of the sieve, the lower the pressure required for compaction, as a result, the hydraulic press designed is effective for clay compaction and the design is safe. The machine which was fabricated with local materials will enhance the production of suitable crucibles, thereby, reducing the over-reliance on the foreign crucible by small and medium scale foundry operators.

1. Introduction

Presses are pressure exerting machine tools and they are widely used in achieving many manufacturing operations in the industries. Presses can be grouped into three main groups

How to cite this article: Ojo OO, Dahunsi OA, Olaleke OM. Design, Fabrication and Structural Analysis of a 5 Tons Hydraulic Press and Mould Machine for Crucible Production. Comput Eng Phys Model 2020;3(3):46-58. https://doi.org/10.22115/cepm.2020.237362.1113

namely hydraulic presses (use hydrostatic pressure), screw presses (use power screws for power transmission), and mechanical presses (use kinematic linkage of elements for power transmission) [1–3]. A cylinder is used to generate compressive force in a hydraulic press [4]. A hydraulic press typically comprises a pump, the fluid (the medium of power transmission via pipes and connectors), and the hydraulic motor [3,5]. Quite some works have been investigated on hydraulic presses (for press works). Raz et al. [6] investigated the dynamic behavior of a hydraulic press used for forging operations. Presently, the innovations on hydraulic presses have aided the development of presses such as large-sized hydraulic presses [7], multi-hydraulic press system [8], double oil hydraulic system [9], cylinder-crown (hemispherical) integrated hydraulic press [10], ultrasonic-assisted presses (for moulding polymers) [11] and other hydraulic presses [12,13] Works on energy-saving in hydraulic presses have been investigated in literature [14–16]. Li et al. [17] reduced the energy consumption (20.6%) and work efficiency (36.1%) of a hydraulic press with double actuators. The use of a combined valve-pump with multiple accumulators was reported to have improved the energy efficiency of a hydraulic blanking press [18]. As compared to other types of presses, hydraulic presses have positive responses to change in (input) pressure, accurate control of force and pressure, and availability of entire force magnitude during the ram travel/working stroke [19]. Yan et al. [20] reported that the high load capacity, power-to-mass ratio, and stiffness are excellent attributes of hydraulic press machines. The application of these salient attributes of a hydraulic press in the production of low-cost crucibles for small and medium scale foundry industries is a research window that is yet to be exploited to reduce accrued production-related costs in these industries.

A lot of works has been done on design and fabrication of presses but no much attention has been given to hydraulic press with permanent mould for foundry crucible production. Researchers have worked on design, optimization, stress analysis, and manufacturing of hydraulic press. Shekar and Rao [4] worked on the design and structural analysis of 1000 tons hydraulic press frame structure. The design was based on the sizing-optimization method and the results were validated by the finite element method (FEM) with the use of appropriate boundary conditions. Hatapakki and Gulhaue [21] worked on design optimization of the C frame of hydraulic press machine via the use of finite element analysis (FEA). The result showed that the equivalent stress of the optimization model was below the design stress, and the deformation of the top frame of the optimized model was less than the desired limit. The net component weight was reduced which shows that the material was optimized. Vaishnav et al. [22] designed and optimized hydraulic press plate (HPP) using finite element analysis (FEA), the simulation was performed on the hydraulic press by taking 250 mm of rib height for three (3) different iteration of all the designs. From the obtained FEA result, it was found that iteration- 3 produced the optimum design and the Von-Mises stress for the design was less than the ultimate tensile stress of the material (the design was reported to be safe against failure).

Sumaila and Ibhadode [5] designed and manufactured a 30-ton hydraulic press. The hydraulic press was tested for leakages and this was first carried out without any load for two hours and later subjected to load of 10 kN provided by two compression spring of constant pressure of 9 N/mm each arranged parallel between the platens. The springs were then compressed axially to a length of 100 mm. The result showed that there was no leakage in the system when the springs were compressed axially to a length of 100 mm and left to stand two. The stress analysis was not discussed but only limited to leakages test. Bethrand et al. [23] developed a manually operated hydraulic press-and-pull machine. Stress analysis of the workpiece support which is the major load-carrying member was done using solid works simulation to identify the maximum shear stress, maximum bending moment, and displacement on the beam when loaded to deformation with a load of 1000N. The result of the stress analysis of the developed machine showed that it performed efficiently but when compared with the electrically operated hydraulic presses, the electrically operated presses are most efficient. Ganesh et al. [24] also designed and analyzed a 12 Ton hydraulic pressing machine. The pressing machine was made for the manufacturing of automobile bodybuilding and sheet metal application. It was designed purposely for the load capacity of 12 Ton. The frame and cylinder were modeled using CATIA V5 and analysis by ANSYS software.

A look at the foundry industries, especially the small and medium scale industries in developing nations, reveals that most of the crucibles used are imported while some make use of ‘used refrigerator compressor container and cut out gas cylinder as their crucible. Based on the literature available, researchers have not exploited the use of the hydraulic press in crucible production, this work, therefore, designs, analyzes, and fabricates a hydraulic press and mould for foundry crucible production for small and medium scale industries.

2. Design methodology

The design methodology adopted for this research includes:

i. Design and analysis of the hydraulic press and mould machine;
ii. Structural simulation using Solid work application 2018; and
iii. Fabrication of the machine and testing

2.1. Design and analysis

The design of the hydraulic press and mould (for crucible production) was carried out with full consideration for ease of operation, maintenance, and repair/replacement in case of breakdown. The design and fabrication of the hydraulic press system/machine were based on these concepts and considerations:

i. The use of standard components/parts for the design;
ii. All parts must be designed and analyzed;
iii. Obtain part, assembly and exploded drawings to ease fabrication process;
iv. Obtain 2D and 3D physical solid modeling of the machine, design analysis, and stress analysis;
v. The hydraulic fluid will be manually pumped (no electrical energy for its operation and this implies that there is no operational cost); and
vi. Serviceability

The conceptual design and its exploded drawings of the hydraulic press and mould machine are shown in Figs.1 and 2 respectively. The design of the components/parts of the hydraulic press and permanent mould is thus carried out.

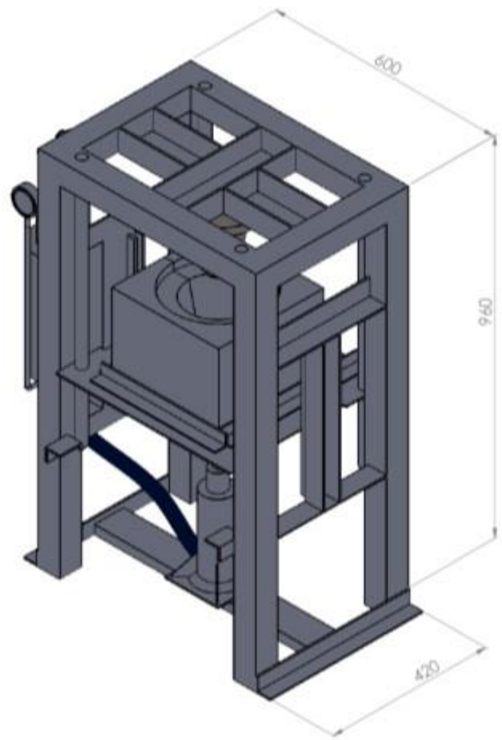

Fig.1. 3D Model of the machine.

S/N	DESCRIPTION	QUANTITY
1	STANCHION/GUIDE WAY	
2	PRESSURE GAUGE	1
3	PRESSURE GAUGE PIPE	1
4	FEMALE MOULD (RIGHT HALF)	1
5	PIVOT PIN	1
6	HYDRAULIC FLUID CRANK	1
7	HYDRAULIC RAM	1
8	HYDRAULIC JACK	1
9	HYDRAULIC SUPPLY RAM	1
10	SUPPLY HOSE	1
11	HYDRAULIC RESERVOIR	1
12	MOULD PLATFORM	1
13	FEMALE MOULD (LEFT HALF)	1
14	FRAME	1
15	MALE MOULD	1
16	PRESSURE RELEASE VALVE	1

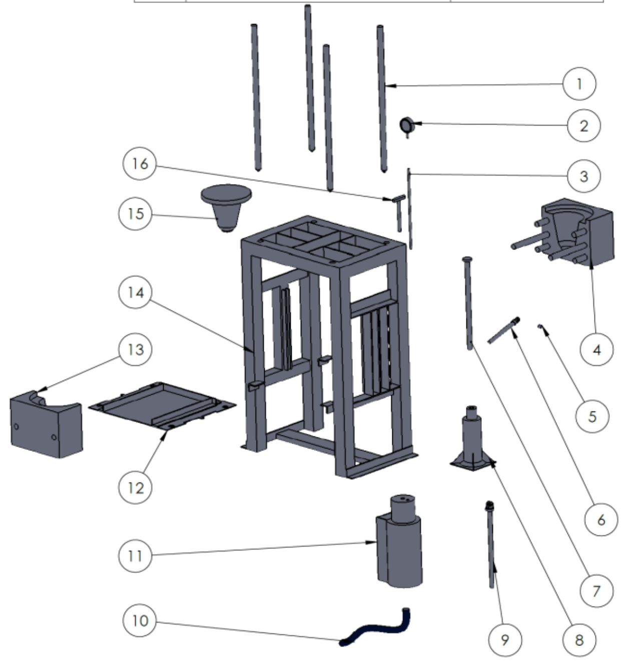

Fig.2. Exploded drawing of the machine.

2.1.1. Frame design

The machine frame (see Fig.2) is the most important part of the machine. It transfers all the forces that are produced during the operation of the machine to the ground. It also provides strength and stability to the machine during operation. The maximum load that the frame will be subjected to is chosen as 50 kN. This load will act on the upper and horizontal frame sections of the frame. The length of the beam used was 600 mm to minimize bending moment and deflection. The beam configuration with attached loads on it is as shown in Fig. 3.

where R_x is the horizontal reaction, R_a is the reaction at point A and R_b is the reaction at point B. Taking moment about A, R_b is 25 kN and R_a is 25 kN (upward forces must be equal to downward forces). Bending Moment for the frame is calculated using equation (1),

$$M_c = \frac{WL}{4} \tag{1}$$

where M_c is the Bending moment, W is the load, L is the distance from both end (W = 50 kN, L = 0.6 m, and M_c = 7500 kNm). The bending stress σ_b is calculated using equation (2).

$$\sigma_b = \frac{M}{I} \times Y \quad (2)$$

Where M is bending moment at a given section, σ_b is bending stress, I is the moment of inertia of the cross-section about a neutral axis, and Y is the distance from the neutral surface to the extreme end. When M = 7500 x 10^6 Nmm^2

$Y = \frac{h}{2} = \frac{1000}{2}$ where h is the height of the frame

Y = 500 mm

$$I = \frac{b}{12} \times h^3, \text{ where b is the length of the frame} \quad (3)$$

When b = 600 mm and h = 500 mm, from equation (3), I = 5x$10^{10}mm^4$and also from equation (2),

$$\sigma_b = \frac{7500e^6}{5e^{10}} \times 500 = 75 \text{ N/mm}^2$$

Therefore, the allowable bending stress for the machine is 75 N/mm² = 75 MPa. Since the yield stress of mild steel is 250 MPa, therefore, the factor of safety for the frame is calculated using equation (4),

$$F = \frac{Y}{A} \quad (4)$$

Where F is the factor of safety, Y is the yield strength of the material and A is allowable stress. Therefore, the factor of safety is 3.33. The shear force and bending moment diagrams of the frames are shown in Fig. 4.

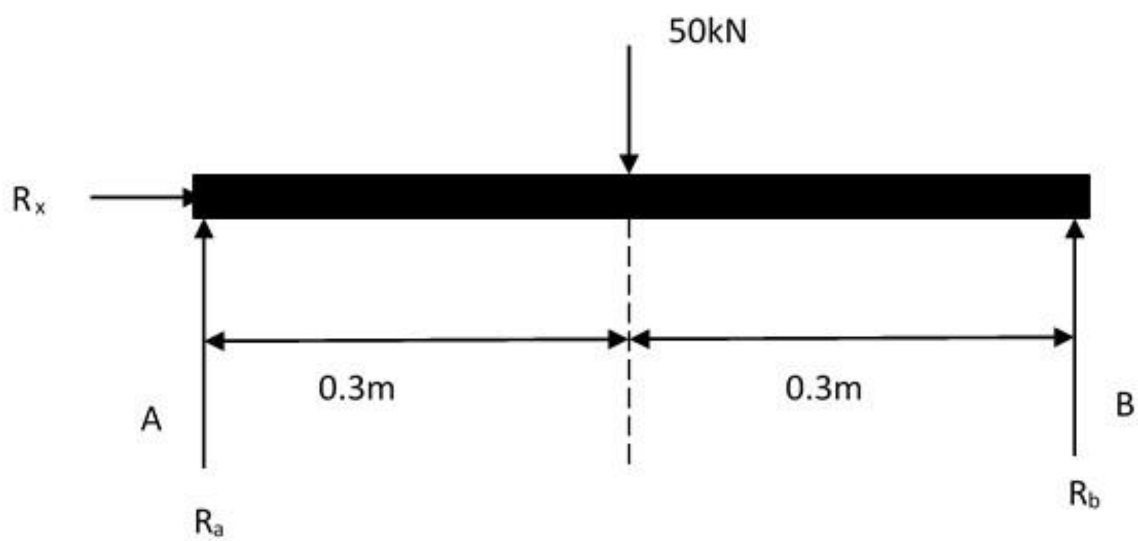

Fig. 3. Beam configurations with attached loads.

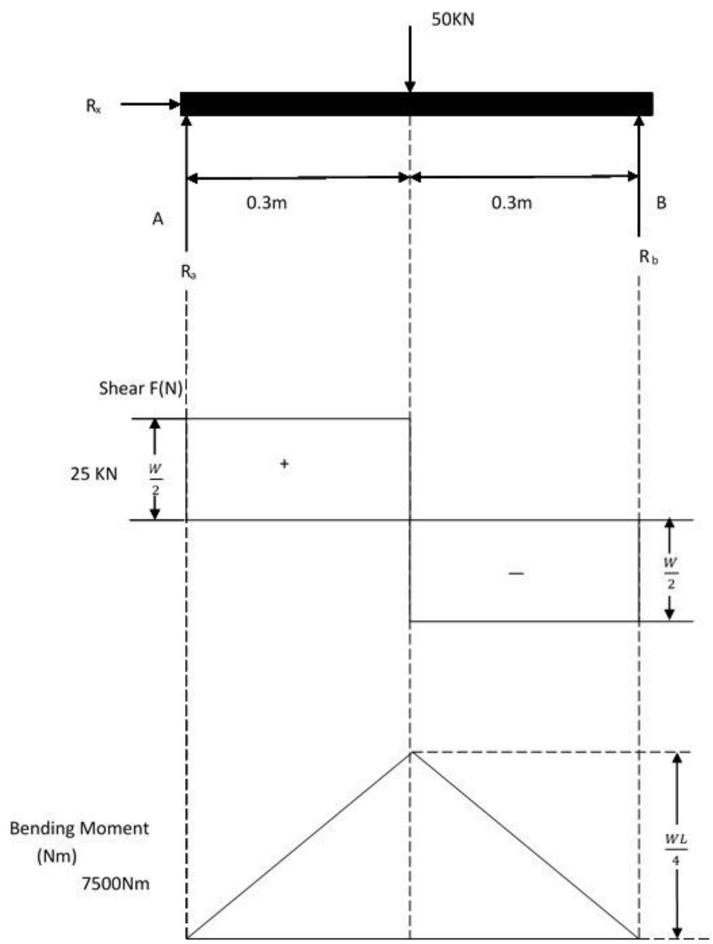

Fig. 4. Loading, shear force and bending moment diagram of the frame.

Hydraulic actuator design: The hydraulic press machine for this work is designed to carry a load of 5 tons (50000N) from the jack under the working pressure of 10 MPa (10N/mm²). Therefore, the inside diameter of the hydraulic cylinder suitable for the work is calculated using equations (5), (6), (7), and (8).

$$P = \frac{F}{A_c} \tag{5}$$

$$A_c = \frac{F}{P} \tag{6}$$

$$A_c = \frac{\pi}{4}(D_i)^2 \tag{7}$$

where P is the working pressure N/mm^2, A_c is the area of the internal cross-section of the cylinder (mm²), F is the load exerted in N, and D_i is the cross-sectional internal diameter of the cylinder. Since working pressure is 100 bar (10 MPa =10N/mm²) and load is 50000 N. Then, D_i = 79.7832838 and D_i $\approx$ 80mm.

$$D_i^2 = \frac{4F}{\pi P} \tag{8}$$

Thickness of the cylinder: The cylinder is made up of mild steel material, therefore the Lame's equation for ductile material as given in equation (9) is used to calculate the thickness [25].

$$t = r_i \left[\frac{\sigma_t}{\sigma_t - 2p} - 1\right] \tag{9}$$

Where t is the thickness of the cylinder in mm, P is the working pressure (N/mm²), σt is the maximum permissible tensile stress (N/mm²) =112 N/mm², and ri is the radius of the inner cylinder (mm). When P = 10 N/mm², σ_t =112 N/mm² and r = 40, from equation (9), t = 4.13 mm, t ≈ 5 mm. Therefore, the external radius of the cylinder is calculated using equation (10)

$$ro = ri + \tag{10}$$

where ro is the external radius of the cylinder (mm), therefore, ro = 40 + 5 = 45 mm

Thickness of the flat end covers: Let t_c be the thickness of the end cover (mm), the force on the end cover is as given in equation (11). Meanwhile, F = Ac × P = 5027.2 × 10 = 50272 N. Then tc is computed to be 5.610 mm ≈ 6 mm.

$$F = D_i \times t_c \times \sigma_t \tag{11}$$

Stress in the cylinder: The stress in the cylinder is calculated using the Lames equation for tangential stress at the inner surface as shown in equation (12)

$$\sigma_t = \mathrm{P} \times \left(\frac{r_{o^2} + r_{i^2}}{r_{o^2} - r_{i^2}}\right) \tag{12}$$

where σ_t is the tangential stress at the inner surface, P is the pressure exerted, r_o is the outside diameter of the cylinder and r_i = inner diameter of the cylinder. (r_o = 80 mm, r_i = 70 mm and P = 10 MPa). Substituting the values in equation (12), the tangential stress is computed to be σ_t = 75.33 N/mm². Therefore, the stress in the inner surface of the cylinder is 75.33 N/mm²

Design of the piston rod: To calculate the diameter of the piston rod, equation (13) is used. Dp is made the subject of the formula in equation (14). Where Dp is the diameter of the piston rod.

$$F = \frac{\pi (D_p)^2}{4} \sigma_t \tag{13}$$

$$(D_p)^2 = \frac{4F}{\pi \sigma_t} \tag{14}$$

From equation (14), Dp is 23.904577 ≈ 24 mm. Thus, the minimum diameter of the piston that can be used is 24 mm. However, the diameter of the piston used for this work is 35 mm for optimum efficiency.

Crucible mould design: The major parameters required in designing the conical crucible mould include outside diameter (D) of the crucible, the inner diameter of the crucible (d), height/depth of the crucible (H) and volume of the crucible as shown in Fig.6.

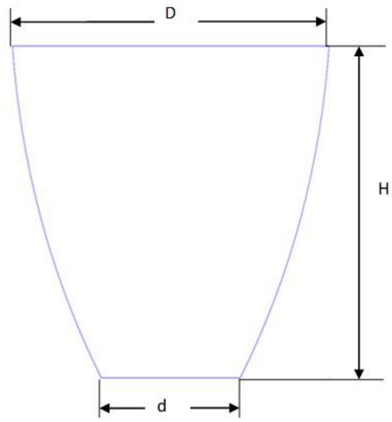

Fig. 6. The Crucible.

For this work, the inner diameter, radius and the height of the crucible as designed is given as 0.1 m, 0.05 m (r $=\frac{D}{2}$ =50 mm) and 0.07 m respectively. To calculate the volume of the crucible V, equation (3.5) is used. The calculated volume of the crucible is 0.00018328 m^3. The relationship between crucible thickness and mould size is given in equation (16). Where, D_d is male mould diameter, D_m is female mould diameter and T_c is crucible thickness

$$V=\frac{1}{3}\pi r^2 h \tag{15}$$

$$D_d = D_m - T_c \tag{16}$$

Therefore, using equation (16) and when D_m = 100 mm and T_c = 12 mm, the male mould diameter (D_d) is estimated to be 88 mm

2.2. Simulation and fabrication of the machine

Based on the governing laws, the simulation of the frame and the mould of the machine was carried. The governing equations in the tensor form include Newton's second law of motion (Eq.17), strain-displacement equation (Eq.18), and the constitutive equation (for elastic materials) or Hooke's law (Eq.19). Where, ∇, σ, F, ρ, $\ddot{u}$, ε, u, T, and C are the nabla operator, Cauchy stress tensor, the applied force, body/mass density, second derivative of the displacement with time, strain tensor, displacement vector, a transpose, and stiffness tensor respectively.

$$\nabla.\sigma + F = \rho\ddot{u} \tag{17}$$

$$\varepsilon = \frac{1}{2}[\nabla u + (\nabla u)^T] \tag{18}$$

$$\sigma = C{:}\,\varepsilon \tag{19}$$

The models of the components (frame and mould) are obtained and assigned the pertinent boundary conditions. The simulation of the machine frame was carried out using the Solid work application 2018 version. A load of 50000 N was applied to the frame while the legs of the frame are constrained to be fixed. The load is also applied to the male and female components of the mould. Mild steel was assigned as the material for the frame and the mould. The properties of the material are shown in Table 1. The components are discretized before the simulation process. The mesh type used for the simulation of the mould (Male and Female) is solid mesh with total elements of 58742 (female) and 43987 (Male) respectively (see Figs.7). The resultant Von-mises, displacement, strain, and factor of safety of the irrespective components are examined. The designed and simulated machine was fabricated.

Table 1
Material Specifications for the Frame.

Properties	Values
Mass Density	7.85 g/cm^3
Yield Strength	207 MPa
Ultimate Tensile Strength	345 MPa
Young's Modulus	220 GPa
Poisson's Ratio	0.275 ul
Shear Modulus	86.2745 GPa

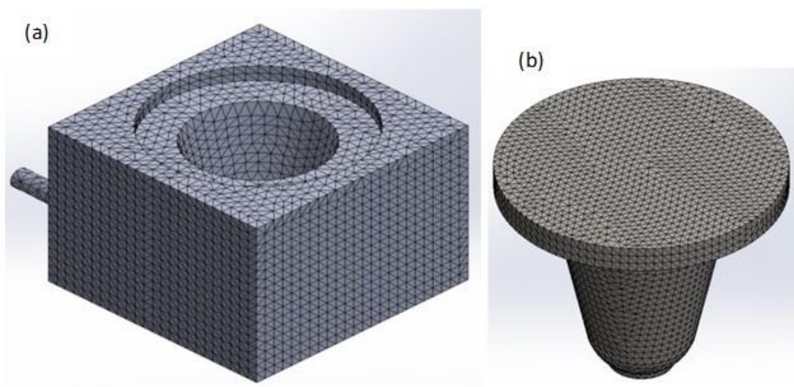

Fig. 7. Mesh of the mould (a) female mould, (b) male mould.

3. Results and discussion

The yield strength of the frame is 207 N/mm² because its material is mild steel. It experiences maximum stress of 97.09 N/mm² (see Fig.8), the maximum displacement of 0.3377 mm, the maximum strain of 0.000361326, and the factor of safety of 2.32 respectively. These results show that the design of the press frame is safe and the machine will not fail under operation since the factor of safety is greater than 1 and the displacement and strain are negligible. The theoretical calculations of the stresses on machine structures show that the bending moment is 7500 kNm. The allowable bending stress on the machine is 75 N/mm². The yield strength of the machine frame is 207 N/mm² because the material used is mild steel. Therefore, the factor of safety gives 2.76, that is, the ratio of yield stress to the allowable stress. The result of the factor of safety shows that the machine will not fail and the design is safe.

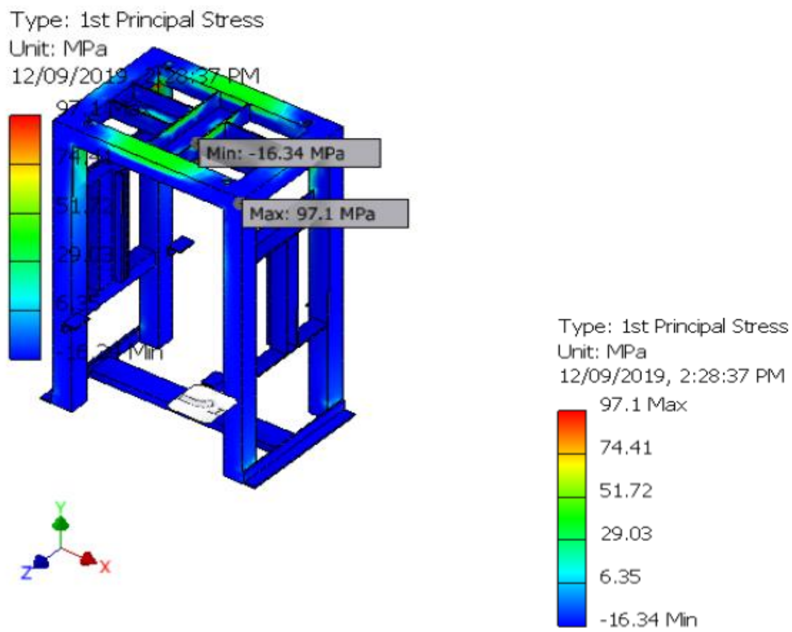

Fig.8. Stress analysis on the Frame.

The inner surface of the mould and the flange area experiences maximum stress of 5.396 ×10^7 N/mm² (see Fig.9), the maximum displacement of 0.007860 mm, the maximum strain of 0.0002012 and factor of safety is 11. The yield strength is 6.204x10^8 N/mm². The results showed that the mould will not fail under operation since the maximum stress experience is less than the yield strength of the material and the factor of safety is greater than 1.

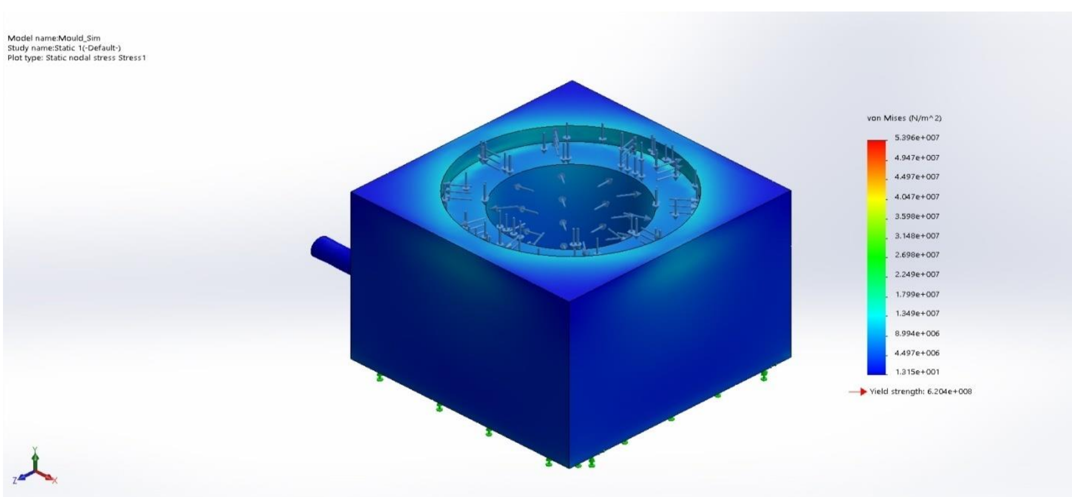

Fig.9. Stress distribution on female mould.

The stress distribution of the male mould is shown in figure 6. The mould is made of alloy steel material with a yield strength of 6.20422× 10^8 N/mm² (see Fig.10). It experienced maximum stress of 6.734×10^7N/mm², the maximum displacement of 0.01352 mm, the maximum strain of 0.0002428, and the factor of safety of 9.2. The results showed that the mould will not fail under operation since the maximum stress experience is less than the yield strength of the material and the factor of safety is greater than 1.

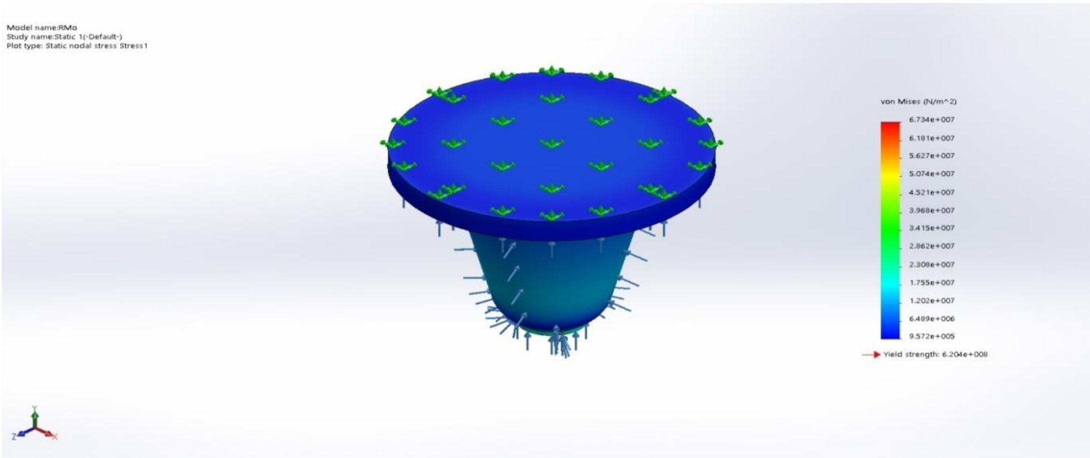

Fig.10. Stress distribution on male mould.

The hydraulic press was subjected to a load of 5 tons to compact clay within the mould, the results show that performance of the machine was satisfactory by successfully compact clay of different sieve sizes into crucible form at the maximum pressure of 100 bar.

The machine was also tested for leakage under the same load but the pressure was increased to 120 bar and left for some minutes and there was no indication of any leakage as the pressure remain constant and did not fall. The hydraulic press and the crucibles produced are shown in Fig. 11.

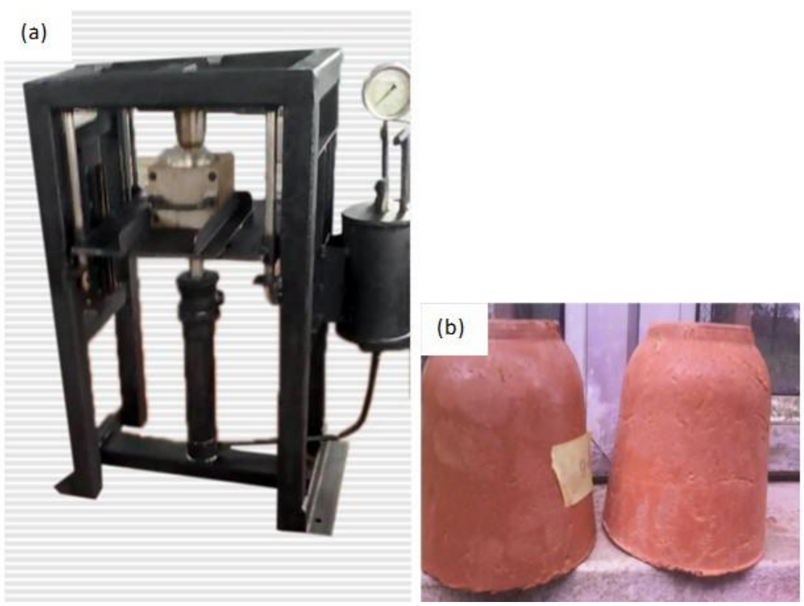

Fig.11. (a) fabricated hydraulic press machine, (b) crucibles.

4. Conclusion

A 5-ton hydraulic press was designed, fabricated, and calibrated. The developed machine was put to test to guarantee its conformability to design objectives and serviceability. The performance of the machine was found to be satisfactory/acceptable at a test load of 50 kN. The theoretically predicted results and those obtained by analytical software have been compared with the design goal and both showed that the design is safe and it will not fail under the designed load. Also, the performance evaluation of the machine affirmed its suitability for compaction of any type of clay into crucible form.

Nomenclature

A_c	Area of the internal cross-section of the cylinder
D_d	Male mould diameter
D_i	Cross-sectional internal diameter of the cylinder
D_m	Female mould diameter
M_c	Bending moment
R_a	Reaction at point A
R_b	Reaction at point B
R_x	Horizontal reaction
T_c	Crucible thickness
σ_b	Bending stress
d	Inner diameter of the crucible
D	Outside diameter of the crucible
Dp	Diameter of the piston rod
F	Load exerted in N
f	Factor of safety
h	Height of the frame
H	Height/depth of the crucible
I	Moment of inertia
L	Distance/length of beam
M	Bending moment
P	Working pressure
ri	Radius of the inner cylinder
ro	External radius of the cylinder
t	Thickness of the cylinder in mm,
V	Volume of the crucible
W	Load
Y	Distance from the neutral surface to the extreme end
σt	Maximum permissible tensile stress

References

[1] Niebel BW, Drapper AB, Wysk R. Modern Manufacturing Process. Eng McGraw-Hill 1989.

[2] DeGarmo EP, Black JT, Kohser RA, Klamecki BE. Materials and process in manufacturing. Prentice Hall Upper Saddle River; 1997.

[3] Sharma PC. A Textbook of Production Engineering. 2005.

[4] Shekar SR, Rao ACUM. Design and Structural Analysis of a 1000 Ton Hydraulic Press Frame Structure. Inter J Res Eng Sci Manag 2018;1:649–52.

[5] Sumaila M. Design and Manufacture of a Thirty-Tonne Hydraulic Press. M. Eng. Project, Production Engineering Department, University of Benin, Benin City, Nigeria (unpublished). 2002.

[6] Raz K, Kubec V, Cechura M. Dynamic behavior of the hydraulic press for free forging. Procedia Eng 2015;100:885–90.

[7] Li L, Huang H, Li X, Liu Z. An improved energy matching method to utilize the potential energy of large-sized hydraulic press at multi-system level. Procedia CIRP 2017;61:547–52.

[8] Li L, Huang H, Zhao F, Liu Z. Operation scheduling of multi-hydraulic press system for energy consumption reduction. J Clean Prod 2017;165:1407–19.

[9] Xu Z, Liu Y, Hua L, Zhao X, Guo W. Energy analysis and optimization of main hydraulic system in 10,000 kN fine blanking press with simulation and experimental methods. Energy Convers Manag 2019;181:143–58.

[10] Zhang W, Wang X, Wang Z, Yuan S. Structural optimization of cylinder-crown integrated hydraulic press with hemispherical hydraulic cylinder. Procedia Eng 2014;81:1663–8.

[11] Janer M, Plantà X, Riera D. Ultrasonic moulding: Current state of the technology. Ultrasonics 2020;102:106038. doi:10.1016/j.ultras.2019.106038.

[12] Raz K, Vaclav K. Using of a Hydraulic Press in Production and Manufacturing of Large Rings. Procedia Eng 2014;69:1064–9. doi:10.1016/j.proeng.2014.03.091.

[13] Li L, Huang H, Liu Z, Li X, Triebe MJ, Zhao F. An energy-saving method to solve the mismatch between installed and demanded power in hydraulic press. J Clean Prod 2016;139:636–45.

[14] Gao M, Li X, Huang H, Liu Z, Li L, Zhou D. Energy-saving Methods for Hydraulic Presses Based on Energy Dissipation Analysis. Procedia CIRP 2016;48:331–5. doi:10.1016/j.procir.2016.03.090.

[15] Zhao K, Liu Z, Yu S, Li X, Huang H, Li B. Analytical energy dissipation in large and medium-sized hydraulic press. J Clean Prod 2015;103:908–15. doi:10.1016/j.jclepro.2014.03.093.

[16] Li L, Huang H, Zhao F, Liu Z. A coordinate method applied to partitioned energy-saving control for grouped hydraulic presses. J Manuf Syst 2016;41:102–10. doi:10.1016/j.jmsy.2016.08.002.

[17] Li L, Huang H, Zhao F, Triebe MJ, Liu Z. Analysis of a novel energy-efficient system with double-actuator for hydraulic press. Mechatronics 2017;47:77–87. doi:10.1016/j.mechatronics.2017.08.012.

[18] Xu Z, Liu Y, Hua L, Zhao X, Wang X. Energy improvement of fineblanking press by valve-pump combined controlled hydraulic system with multiple accumulators. J Clean Prod 2020;257:120505. doi:10.1016/j.jclepro.2020.120505.

[19] S N, T PM, A. G, G M, G V, P. KC. Study of the Design of Compact Hydraulic Press Machine for Rock Drill Component. Inter Conf Emerg Trend Eng Manag Res, Nashik India 2016.

[20] Yan X, Chen B, Zhang D, Wu C, Luo W. An energy-saving method to reduce the installed power of hydraulic press machines. J Clean Prod 2019;233:538–45. doi:10.1016/j.jclepro.2019.06.084.

[21] Hatapakki AB, Gulhane UD. Design Optimization of C Frame of Hydraulic Press Machine. Asian J Converg Technol 2016;2.

[22] Vaishnav A, Lathiya P, Sarvaiya M. Design Optimization of Hydraulic Press Plate using Finite Element Analysis. Int J Eng Res Appl ISSN 2016:2248–9622.

[23] Nwankwojike BN, Nwogu CN, Kalu G. Development of a Manually Operated Hydraulic Press and Pull Machine. FUOYE J Eng Technol 2017;2:73–8.

[24] Ganesh MM, C. G, A.P. P. Design and Analysis of 12 Ton Hydraulic Pressing Machine. Inter J Sci Dev Res,3 2018:109–17.

[25] Khurmi RS, Gupta JK. A textbook of Machine Design. Eurasia Publ 2005.