Design, Analysis and Manufacturing of Hydro-pneumatic Press Machine

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ABSTRACT

A Hydro-pneumatic press is a press machine utilizing both air and oil in its operation and gives higher outlet hydraulic pressure with lower inlet pneumatic pressure. In this project the press is design and manufacture for pressing sleeve bearing into the circular casting part. Casting part is thick cylinder and sleeve bearing is kind of cylindrical bearing. Two actuators are used in the press one is for vertical pressing and other is for horizontal pressing. This paper includes the concept development, design, analysis and manufacturing of press machine. Various parts of the press are modelled by using Pro-E modelling software. Structural analysis has been applied on the parts of press machine by using analyzing software ANSYS.

KEYWORDS: *Hydro-pneumatic, High outlet pressure, Low inlet pressure, Circular casting part, Sleeve bearing, Pro-E, ANSYS.*

I. INTRODUCTION

A system utilizing both air and oil in its operation and gives higher outlet hydraulic pressure with lower inlet pressure is called as hydro-pneumatic system. Hydro-pneumatic systems can give maximum pressure up to 700 bar. No worry of handling oil pumps or tanks and it comes in compact unit. The frame is designed for pressing of four sleeve bearing two are horizontally and two are vertically into circular casting part hence following points are take into considerations[1]. Figure 1.1 shows model hydro-pneumatic press.

- Arrangement for two actuators, one is horizontal and other is vertical
- Use of arrangement on which hitch yoke is placed for assembly and worker can access it in straight comfortable position
- For achieving positional accuracy some sliding arrangement should provided so that yoke can easily placed or lift with the help of hoists and then slide at proper position for pressing
- Yoke should place on the machining surface to achieve dimensional accuracy

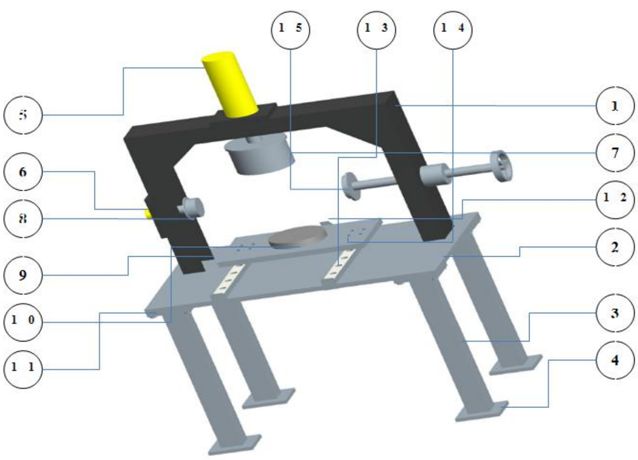

Figure 1.1: Pro-E model of press machine

Table 1.1 includes details of all components required for building the actual model of hydro-pneumatic press.

Table 1.1 List of component.

Component number	Description	Quantity
1	C - Frame	1
2	Base Plate	1
3	Support Column	4
4	Support Plate	8
5	Central Cylinder	1
6	Side Cylinder	1
7	Central Ram	1
8	Side Ram	1
9	Sliding Plate	1
10	Circular Plate	1
11	Bolts	38
12	Stopper	1
13	Rail	2
14	Block Bearing	2
15	Side Support	1

II. WORKING

Hydro-pneumatic system is divided into two main components i.e. hydro-pneumatic pump and cylinder. Main components of hydro-pneumatic pump are air motor, gearbox, eccentric, pump and oil reservoir. Spring operated check valve is provided at inlet port of pump. Connect the pump to the pneumatic connection of compressor. Air motor rotates by air and rotates the shaft of gear box. Reduction gear reduces the speed of outlet shaft on which eccentric cams are mounted. Cams move the pistons of two piston pumps and hydraulic oil enter into cylinders at continuous flow rate and hence smooth stroke is obtained.

Now connect the pump to central cylinder by quick acting coupling and operate the control valve which gives the forward stroke to press first two bearings. The oil enters in the cylinder from pump at controlled rate hence slow forward stroke is achieved. After pressing first bearing, again operate the control valve which releases the pressure on cylinder and return stroke is achieved with help of spring. Similarly connect the pump to the side cylinder and press side bearings. Figure 2.1 shows the circuit diagram of press machine.

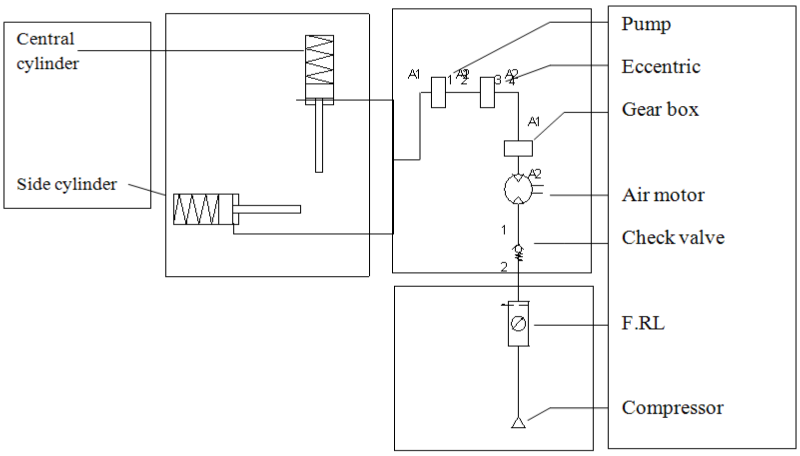

Figure 2.1: Circuit diagram of press machine

III. MATERIAL SELECTION

Material is selected based on properties such as high bending & tensile strength, ease of availability, ease of machining, welding, finishing, cutting etc. and cost factor. Component number 1, 2, 3, 4, 8, 9, 10, 12 and 15 will use the Mild steel/ plain carbon steel (25C8/ AISI 1025). Material Properties of 25C8 are given in Table 3.1 below:

Table 3.1 Material property

Parameter	Details
Material	25C8
Tensile strength, (σ_t)	390 N/mm^2
BHN	170 HB
Elastic modulus, (E)	210 GPa

http://www.btss.in/technical.php

IV. DESIGN CALCULATION

Following are the main components required for design of press and they are designed considering the specification given in the Table 4.1.

a) C – Frame.
b) Base plate
c) Sliding plate.
d) Support column.
e) Side support.

Table 4.1 shows required cylinder specifications of machine.

Parameter	Central Cylinder	Side Cylinder
Press load	23 *KN*	8 *KN*
Stroke length	304 *mm*	54 *mm*

4.1 Design of C-frame

Functional requirement: Length of upper beam = 900 mm and length of side column = 700 mm are taken considering the job size, horizontal stroke and vertical stroke length required for pressing operation.

C-frame design is divided into two main parts as;

a. Design of upper beam b. Design of left and right side column

a. Design of upper beam

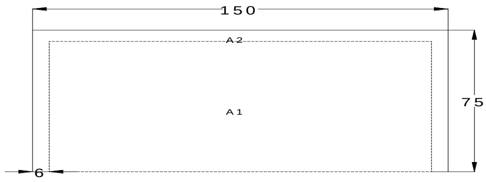

Figure 4.1: Cross section of upper beam & side column

Position of neutral axis, $Y = \frac{A_1 Y_1 - A_2 Y_2}{A_1 - A_2}$ [2]

$A_1 = 75 \times 150 = 11250 \text{ mm}^2$ $A_2 = (150-12) \times (75-6) = 9522 \text{ mm}^2$

$y_1 = 75/2 = 37.5 \text{ mm}$ $y_2 = (75-6)/2 = 34.5 \text{ mm}$

$$Y = \frac{(11250 \times 37.5) - (9522 \times 34.5)}{11250 - 9522} = 54.03 \text{ mm}$$

Moment of Inertia, $I = I_{xx1} - I_{xx2} = (I_{G1} + A_1 h_1^2) - (I_{G2} + A_2 h_2^2)$ [2]

$$I = \frac{150 \times 75^3}{12} + [11250 \times (54.03 - 37.5)^2] - \frac{138 \times 69^3}{12} + [9522 \times (54.03 - 34.5)^2]$$

$I = 940 \times 10^3 \text{ mm}^4$ -----------(1)

Since the beam is subjected to hogging bending moment, compression neutral axis

$y = y_c = 54.0$ mm, $y_t = 75 - 54.03 = 20.97$ mm

Bending moment M from loading diagram

$M_A = R_B \times 900 - 23 \times 400 = 0$

$R_B = 10.22$ KN $R_A + R_B = 23$ $R_A = 12.77$ KN

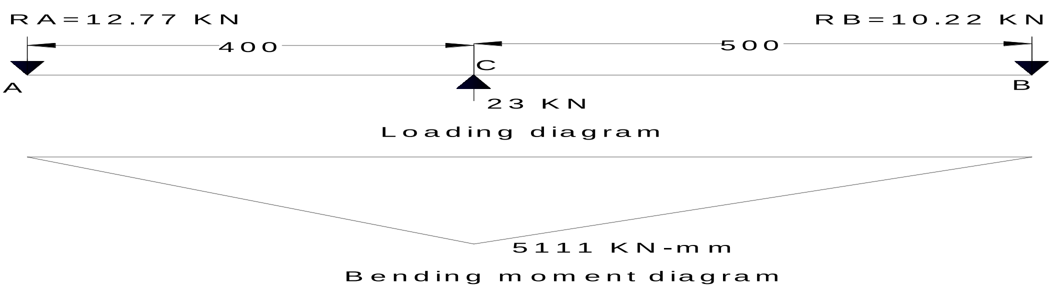

Figure 4.2: Loading & Bending moment diagram of upper beam

Moment at C= $M_C = 10.22 \times 500 = 5111$ KN-m

Compressive stress at C= Max stress $= \frac{M_C Y_C}{I} = \frac{(5111 \times 103 \times 54.03)}{940 \times 10^3}$ = 114.01 N/ mm^2

Material of beam is 25C8, S_{yt} = 390$\frac{N}{mm^2}$ [5]

Max allowable stress = 390 / 1.5 = 260 N /mm^2------ (Assume F.O.S. = 1.5)

Max. allowable stress > Max. compressive stress in beam, Hence design is safe

b. Design of left and right side column

Figure 4.1 shows the dimensions of selected column cross-section.

Consider column AB of length L is fixed at one end and other end is hinged

Effective length, $L_e = L/\sqrt{2}$ = 700 /1.4142 = 494.97 mm [3]

Least moment of inertia I = 940 x 10^3 mm^4 --------------- from (1)

Modulus of elasticity for 25C8, E= 200 x 10^3 N/ mm^2 [5]

Crippling load by Euler's formula[3],$P_c = (\pi^2 EI)/{L_e}^2 = \frac{(\pi^2 \times 200 \times 10^3 \times 940 \times 10^3)}{494.97^2} = 7.573 \times 10^6$ N

Safe load $P_s = P_c/F.O.S. = {(7.573 \times 10^6)}/{3}$-------------- (Assume F.O.S. = 3)

= 2.52 x 10^6 N > 23318.825 N Hence design is safe

4.2 Design of base plate

Functional requirement: Length = 1000 mm and width = 700 mm of base plate is required for easy mounting of all components and easy pressing operation.

Total load acting on base plate = {(7Kg (Upper cylinder) + 2 Kg (Side cylinder) + 23.5 Kg (Sliding plate) + 11.775 Kg (Mounting plates) + 16.120 Kg (Upper ram) + 8.9 Kg (Side ram) + 22.75 Kg (Rails) + 20.22 Kg (Support) + 6.81 Kg (Circular plate) + 93.9 Kg (Hitch yoke) } x 9.81 + 23000 N (Force by cylinder)
= 25089.28 N

From loading diagram shown in Figure 4.3

Moment at C = Max. Moment [4]= $WL^2/8 = [(25.08 \times 10^{-3}) \times 1000^2]/8$ = 3135 KN-mm

Moment of Inertia (I)= $bt^3/12 = 700 \times d^3/12 = 58.33 \times t^3 mm^4$

Using bending formula, $\frac{M}{I} = \frac{\text{bending stress}}{y}$; $\frac{(3135 \times 10^3)}{(58.33 \times t^3)} = \frac{\text{bending stress}}{y}$ [4]

For 25C8, Syt = 390 N / mm^3 [5]

Allowable bending stress = $\frac{S_{yt}}{F.O.S.} = \frac{390}{1.5}$---- (Assume F.O.S =1.5)

= 260 N / mm2

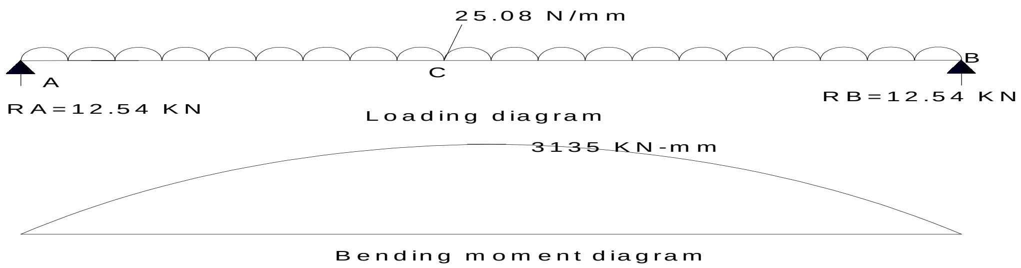

Figure 4.3: Loading & Bending moment diagram of base plate

Therefore $\frac{(53745.92)}{(t^3)} = \frac{130}{t}$

Thickness of plate, t = 20.33 mm

Hence plate with thickness 22 mm is selected for safe design

4.3 Design of sliding plate

Functional requirement: Length = 500 mm and width = 300 mm of sliding plate is restricted for easy rail mounting and considering the yoke size.

Total load acting on base plate = {(6.81 Kg (Circular plate) +93.9 Kg (Hitch yoke)} x 9.81 + 23000 N (Force by cylinder)
= 23927.969 N

From loading diagram shown in Figure 4.4
Moment at C = Max. Moment =11.96 $\times$ 200 – (0.124 $\times$ 96 $\times$96/2) =1820.60 KN-mm
Moment of Inertia, (I) =$bt^3/12$= 300 d^3 /12 = $25t^3$ mm^4

Using bending formula, $\frac{M}{I} = \frac{\text{bending stress}}{y}$; $\frac{1820.60 \times 10^3}{25t^3} = \frac{\text{bending stress}}{y}$ [4]
for 25C8,Syt = 390 N / mm^3 [5]
Allowable bending stress, $\frac{S_{yt}}{F.O.S.} = \frac{390}{1.5}$-------- (Assume F.O.S =1.5)
= 260 N / mm^2

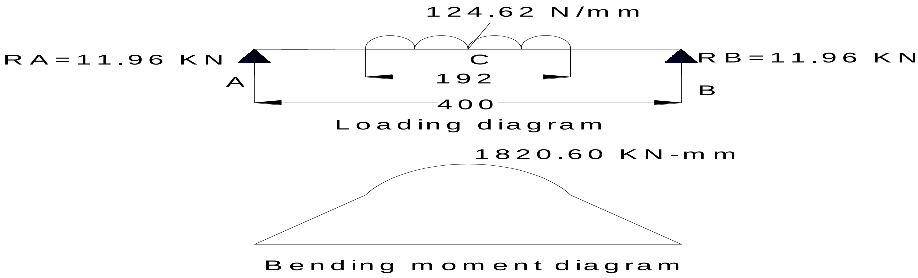

Figure 4.4: Loading & Bending moment diagram of sliding plate

Therefore $\frac{72824}{t^3} = \frac{130}{t}$
Thickness of plate, t = 23.66 mm
Hence plate with thickness 25 mm is selected for safe design

4.4 Design of support column
Functional requirement: Length = 700 mm of support column is required for easy operating and comfort of worker.

Consider column AB of length L is fixed at both ends

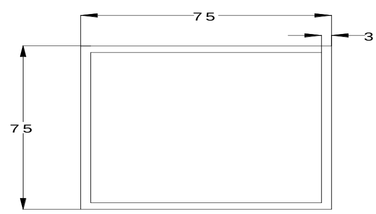

Figure 4.5: Cross section of support column

Effective length [2], $L_e = L/\sqrt{2}$= 700 /1.4142 = 494.97 mm
Least moment of inertia I = $BD^3/12 - bd^3/12 = 75 \times 75^3/12 - 69 \times 69^3/12$
=747.792×10^3 mm^4

Modulus of elasticity for 25C8, E= 200 x 10 ^{3} N/ mm^2 [5]
Crippling load by Euler's formula,$P_c = (\pi^2 EI)/{L_e}^2 = \frac{(\pi^2 \times 200 \times 10^3 \times 747.792 \times 10^3)}{494.97^2}$ [2]
$= 6.024 \times 10^6$ N
Safe load $P_s = P_c/F.O.S. = {(6.024 \times 10^6)}/_3$ -------------- (Assume F.O.S. = 3)
= 2008.31KN> 6.25 KN Hence design is safe

4.5 Design of side support

Functional requirement: Length of rod = 450 mm is required easy access and safe operation.

Max. Pressure = $p_{max} =$ 150 bar = 15 $\frac{N}{mm^2}$

$\text{pressure (p)} = \frac{\text{force}}{\text{area}}$

Force = $15 \times (\frac{\Pi}{4} d^2) = 117.809 \times 10^3$ N

Max. Force = $f_{max} = (117.809 \times 10^3) \times (0.6 \times 9.81 \times 100) = 118.397 \times 10^3$ N

Allowable stress = $\sigma_{allowable} = \frac{s_{ut}}{Nf} = \frac{390}{3} \frac{N}{mm^2} = 130 \frac{N}{mm^2}$ for 25C8 [5]

Maximum stress = $\sigma_{max} = \frac{F_{max}}{area} = 130 \times 10^3 = \frac{118.397 \times 10^3}{\frac{\Pi}{4} d^2}$

Diameter of rod = d =38.07≈ 40mm [4]

V. ANALYSIS

This section shows the details of Finite Element Analysis of this developed model. The Finite Element Method is the easy technique to the theoretical method to find out the stress developed in various components of press. In this paper Finite Element Analysis is carried out in ANSYS Workbench 11 to determine the maximum stress developed in press. Also the deformation is found out for various component of press.

5.1 Steps in analysis:

a. Step 1: Import geometry

Figure 5.1 shows Pro-E model imported in Ansys.

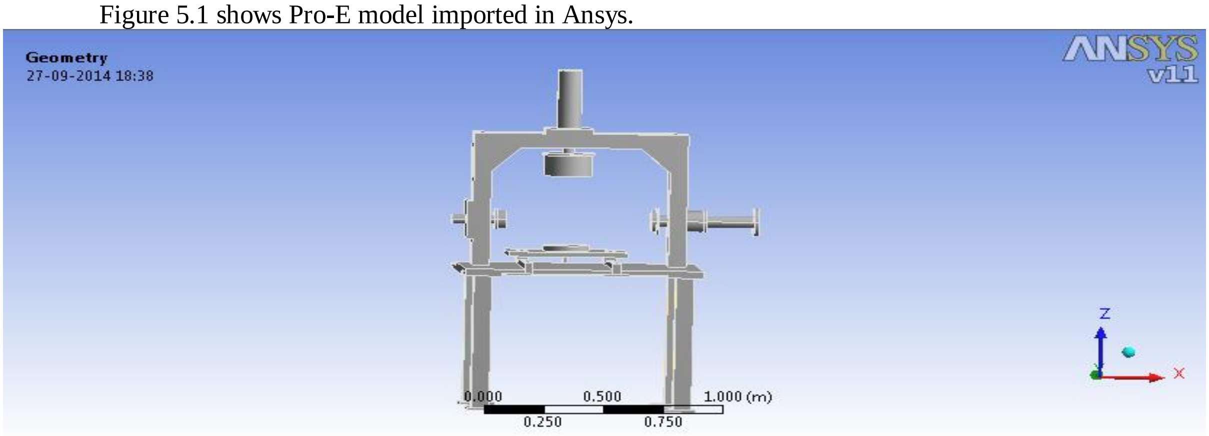

Figure 5.1: 3-D geometry of hydro-pneumatic press machine

b. Step 2: Meshing

Figure 5.2 shows the component meshing. Cores meshing of geometry are performed.

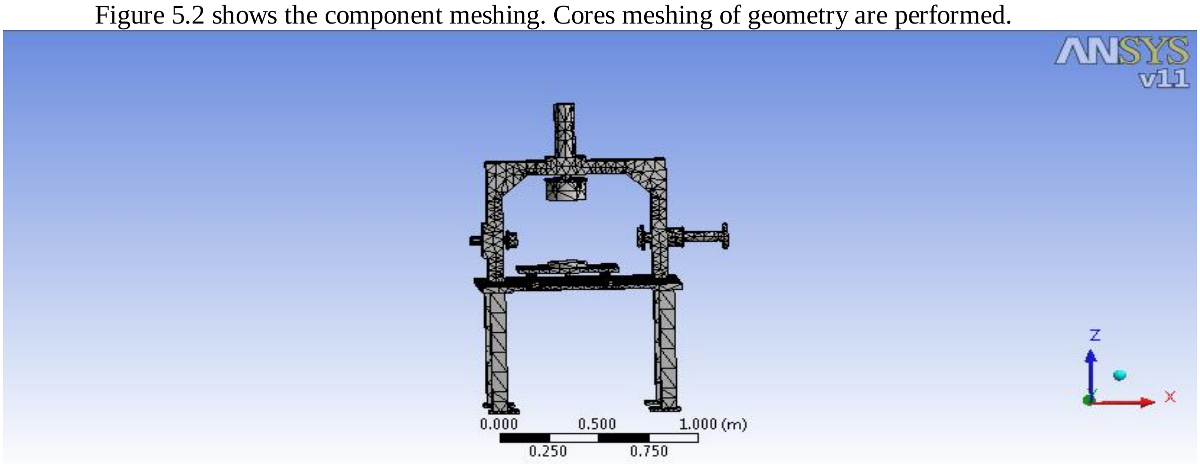

Figure 5.2: Coarse mesh of hydro-pneumatic press machine

c. Step 3: Boundary conditions:

Base columns are fixed as per required initial condition. The load of 23000 N is applied on the central ram and the sliding plate and 8000 N is applied on the side ram and side support which replicate actual working condition in simulation. Figure 5.3 shows the boundary conditions.

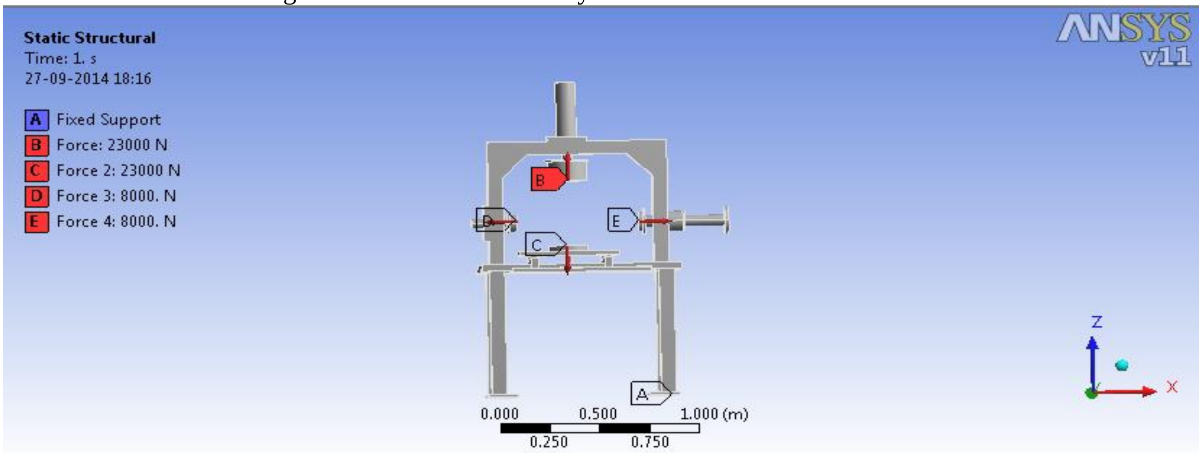

Figure 5.3: Boundary conditions.

VI. RESULTS AND DISCUSSION

The design had main focus on reducing operator fatigue and increase safety, improving the flexibility and makes operation more convenient, and to achieve dimensional and positional accuracy. Components of press are designed to avoid bending failure due to applied load. Mild steel is selected as material based on its properties such as high bending & tensile strength, it compatibility with operation like machining, welding, finishing, cutting etc. and cost as economic factor.

Result of the Finite Element Analysis, it shows that the maximum nodal displacement magnitude on the hydro pneumatic press is 0.00034255 mm as shown in Figure 6.1 when maximum load 23000 N is applied on base plate due to action of actuator. Following result shows that maximum Von Misses stress, maximum principle stress, maximum shear stress values in safe point because analyzed stress < calculated stress. Compression between analyzed and allowable material value of stress are in Table 6.1 below:

Table 6.1 Stress comparison table

Parameter	Analytical	Allowable	Safety
Von-miss stress	108.54 N/mm^2	130 N/mm^2	Safe
Max. Principle stress.	123.09 N/mm^2	130 N/mm^2	Safe
Max shear stress	58.06 N/mm^2	65 N/mm^2	Safe

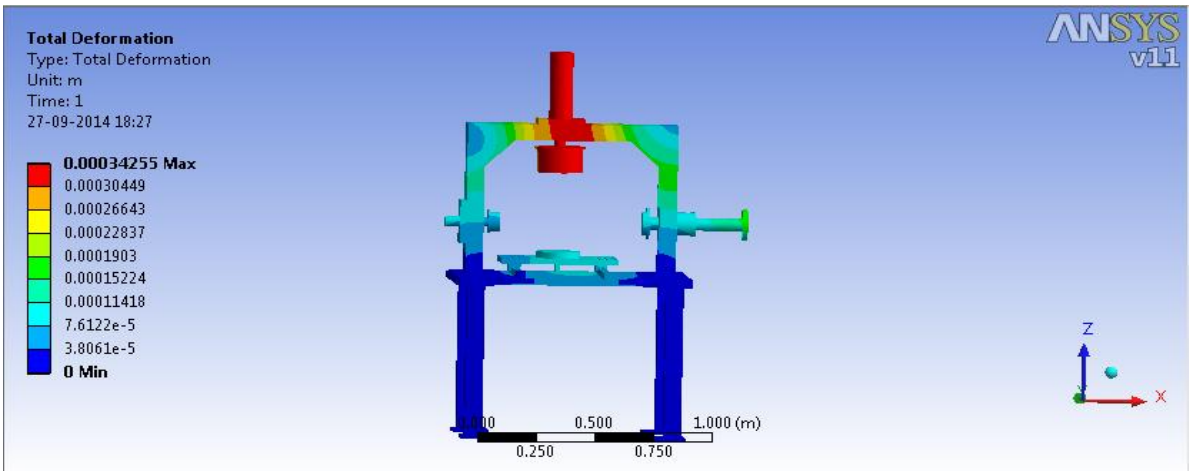

Figure 6.1: Deformation pattern for hydro-pneumatic press machine

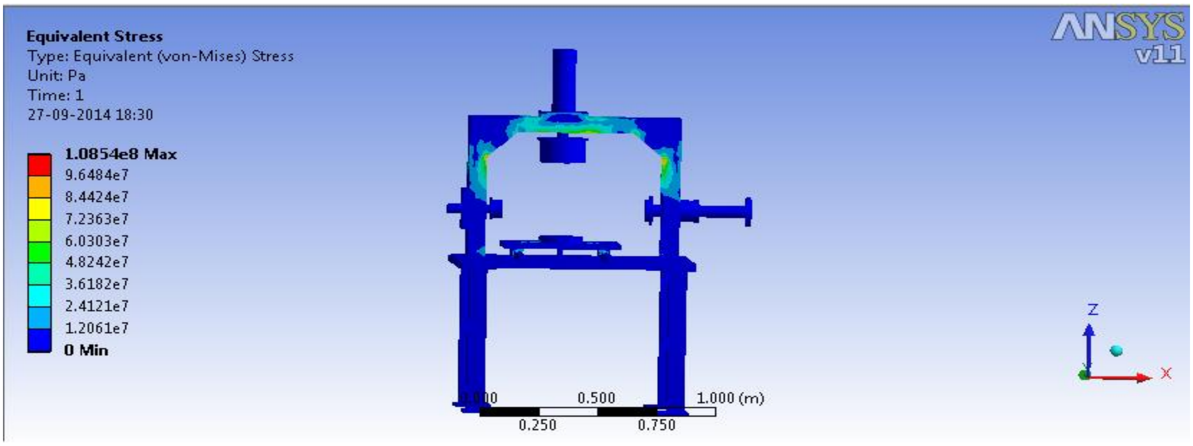

Figure 6.2: Von Miss Stress distribution.

VII. MANUFACTURING PROCEDURE

Plates of size 1000x700x22 mm, 500x300x25 mm, 150x150x16 mm; Hollow columns of size 75x75x700 mm; C-channel of size 150x75x6 mm and Rods of size ϕ192x30 mm, ϕ40x490 mm obtained from the structural steel vendor. All of the above are slightly finished by the hand grinder.

Base plate is manufactured from 1000x700x22 mm plate by drilling 16 holes of ϕ14 at each corner using vertical drilling machine and tapped to ϕ16 mm, same holes are obtained on four plates of size 150x150x16 mm. 18 holes of ϕ12mm are drilled and tapped to ϕ14mm on base plate by keeping 100mm offset from centre on which rails are fitted by bolts. Four plates of size 150x150x16 with holes and four more such a plate without holes are welded on both ends of four hollow columns of size 75x75x700 mm. These columns are then bolted to the base plate by std. bolts of ϕ16 mm dia. as shown in Figure 7.1

Fig 7.1: Stage 1.

Fig 7.2: Stage 2.

The sliding plate i.e. 500x300x25 mm plate is drilled on both sides with four holes of ϕ8mm and tapped to ϕ12mm. Central hole of ϕ8mm is also drilled on this plate on which the rod of ϕ192x30mm is joined by inserting a pin. The whole assembly is then mounted on the block bearing of rails as shown in Figure 7.2

C-channel is cut using the power hacksaw for the length of 500mm, 700mm and 700mm. 45 degree cuts are obtained on C-channel which is then end mill on HMC machine and weld together using arc welding to obtain C-frame. Holes are drilled on C-channel for mounting of hydrodynamic cylinder on which support plate are welded for rigidity. Support of hitch yoke is made from ϕ40x490mm rod which is turn on lathe machine and handle is weld at one end, this is inserted in the bush which mounted on C-frame. Finally the C-frame is welded on base plate using arc welding and cylinders are mounted on the C-frame as shown in Figure 7.3.

Fig 7.1: Actual Hydro – Pneumatic Press machine.

VIII. CONCLUSION

The press was developed after studying the pneumatic system, hydraulic system and hydro – pneumatic system, were it was found that hydro – pneumatic system is more effective the pneumatic and cost efficient than hydraulic system. The system has shown noticeable improvements in various sectors like operation time and cost of operation. It is observed that operation time is reduced from 3 hours to 30 min per assembly and cost of operation is reduced approximately by 90 %. The further advantages of the system has covered the safety of operator and made operation more convenient (reducing fatigue), improved dimensional and positional accuracy of assembly.

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