# Design and Analysis of 12 Ton Hydraulic Pressing Machine

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***Abstract:*** **The hydraulic pressing machine used for converting shape of the material to the required form by compressive force of action. In this work hydraulic pressing machine of twelve Ton capacity is Designed and Analysed. The design has to resist the generated force during operation and to calculate design parameters like stress induced and total deformation developed during operation. This pressing machine is made for manufacturing of automotive body buildings and sheet metal applications. The machine is designed for special purpose only, to the load capacity of 12 Ton. Structural analysis becomes a part to identify the product design. The frame and cylinder is modeled using CATIA V5 and analysis by ANSYS software**.

***Index Terms***: **4-Column type hydraulic press, Hydraulic Piston, Hydraulic Cylinder, Modeling, Static Structural Analysis**

## I. INTRODUCTION

The hydraulic pressing machine used for converting shape of the material to the required form by compressive force of action. Upper platen, columns, movable platen, base and cylinder are main design parameters of hydraulic press ranging from kilogram to tons capacity by exerting force [1].

The operation includes deformation of metal specimen to obtain desired shape and size by refining grain structure, pressing machine always uses impact load due to this press machine always experience continuous stresses, hence frequently structural failure of some parts of hydraulic press may occurs, in these some parts experience compressive stress and some parts experience tensile stresses. ANSYS FEM tool is incorporated in the present work to do analysis of design [2]. When material required high magnitude of forces then hydraulic machines are most essential, and effective for forming and crushing operation [3]. Press machine is used for good quality outputs with high accuracy and economical. Ductile materials can be formed in cold working process. The press machine applies force to specimen to transfer shape of non-metallic or metallic materials. Deformation of work piece to required shape and size is done by compressive action [4].

The hydraulic presses have main advantage of positive response for the input pressure when compared to other type of machines. The force and pressure can be controlled accurately and ram travel utilizes entire magnitude force which is available during forward stroke. Press machine used for fitting operations are most valuable machines used in laboratories and workshops whenever high force is required [5].

## II. Literature Review

Mr.Umesh.C.Rajmane et.al. [1] Studied design and development of the 100-ton press forging operation. The proposed work consisting of the modification of existence configuration of the single layered forging press machine into the double-forging press configuration by performing structural analysis and stress distribution analysis on the base platen and on the lower die. It shows a comparative study of the stress distribution in the parts and the modifications of the project. An experimental system will be developed to achieve the established objective, which mainly includes the examination of the tensions in all the main parts of the forging press.

Bhushan.V.Golechha, P.S.Kulkarni et.at. [2] Investigated on the design optimization and structural analysis of ten ton pneumatic press. This work of the project concerned with the design and analysis of the finite elements and the structural optimization of the ten ton pneumatic press. The goal is to reduce the cost and weight of pneumatic press machine without reduction in the quality of production. Using the best possible and available resources in project can affect the cost and weight of the press. One way to do this will be to do optimize to reduce the volume of the material used to build the complete structure of the press machine. Here this is considered as industrial application project consisting of the massive optimization of a pneumatic press. The ANSYS software was used for the analysis of the purposes.

Malachy Sumaila et.al. [3] Carried out design and implementation of thirty ton hydraulic press. A thirty ton hydraulic press was constructed and evaluated by testing with local sourced materials. The design parameters are maximum load of 300 kN and has to with stand load by piston stroke of 150 mm piston diameter is 100 mm. the major component of press consists of piston and cylinder arrangement, hydraulic circuit and frame. Machine was tested with load condition of 10 kN with 2 compressive spring of 9 N/mm constant each arranged between upper platen and lower platen parallel and results were found satisfactory.

Akshay Vaishnav et.al. [4] By optimizing the design of the hydraulic pressure plate through FEA, the hydraulic press operates under continuous load. Due to this continuous loading, tensile as well as compressive stresses are produced in various parts of the machine. These stresses cause permanent deformations in some components of the machine. This work consists on the optimization of a hydraulic press of four pillar and 250 tons capacity, taking into account limitations such as design, weight and cost. The work focuses on the design and optimization of the top platen. The upper is most critical part of hydraulic press machine because it carries cylinder. The project is based on the size optimization method and the results are validated using the finite element method with the appropriate boundary conditions. CAD modelling was done in CREO and the ANSYS software is used for FEA. The attempt was made to limit the total deformation of less than 0.3 mm/m. This is done by changing the height of the plate.

Karel Raz et.al. [5] Studied on hydraulic pressing machine for dynamic behavior for forging operation with 180 SPM speed. Their goal was to study different frame designs of a press machines for forging operation. Evaluation is determination of suitability for each structure for supporting of special purpose operation. FEM modal analysis is made to compare results. It is recommended not to use 2 column press machine for high end operations. Four column press design with bottom drive was recommended.

## III. PROBLEM FORMULATION

Machine which are having complicated geometry are difficult to analyze, Hence Analysis software's are used to compute the results. In this work C-type and 4-Column type hydraulic pressing machine is designed and analysed. Analytical and numerical results are compared to validate the results.

## IV. DESIGNING OF 4-COLUMN TYPE HYDRAULIC PRESS

**Design Pre-requisites:** Hydraulic press system is to be designed according to the requisites of following.

1. 12 ton operating force.
2. 350 bar cylinder pressure.
3. Stroke length 220 mm.
4. Bolster size length 250 mm and width 250 mm.

Table 1: Material selections

| Part | Material | Density (kg/m$^3$) | Young's modulus (GPa) | Tensile strength (MPa) | Poisson's ratio |
|---|---|---|---|---|---|
| Cylinder and pillar | ST-42-S | 7850 | 210 | 412 | 0.3 |
| Piston with rod, Base, Nut and bolts | EN8 | 7850 | 200 | 541 | 0.3 |
| Platens and Bolster | EN24 | 7840 | 207 | 800 | 0.29 |

➢ **Design of Cylinder Body:**

Since we are designing for 12 ton

Capacity force is 12000 kg = 117720 N and

The Hydraulic pressure is 350 bar (350*0.1=35 N/mm$^2$).

**a) Bore internal diameter ($D_i$):**

Formula of pressure

P = F / A ------------------------------------------------------------------------ (1)

Where, P = Pressure in N/mm$^2$

F= Force in N

A = Area in mm$^2$

Therefore,

$35 = 117720 / (\pi / 4 \times D_i^2)$

$D_i = 65.44 = 65$ mm

Since ASTM A106-B Standard Bore from catalogue manual is 63 mm Therefore Bore diameter $D_i$ = 63 mm.

**b) To find the thickness of the press cylinder**

Material for tube: St-42 Structural steel hollow tube

Tensile strength = 42 kgf/mm$^2$

= 412.02 N/mm$^2$

For hydraulic press machine where loading is uncertain in design Application, hence FOS is chosen is 4. FOS chosen is on the higher side because this is the equipment for checking of valves. Normally a FOS of around 2.5 to 3 is chosen since testing equipment should be more accurate and long lasting therefore a FOS of 4 is chosen.

$F_t = \sigma_t = 412.02/4 = 103.00$ N/mm$^2$

c) **Clavarino's equation for closed cylinder**

$t = (D_i / 2) \{\sqrt{[(F_t + (1\text{-}2\mu) P / (F_t - (1\text{-}\mu) P)]} - 1\}$ ---------------------------------- (2)

Where, t = Cylinder thickness

$D_i$ = Inner diameter of the cylinder body = 63mm

P = Working pressure = 35 $N/mm^2$

μ = Poisons' ratio = 0.3

$t = (D_i/2) [\sqrt{\{(F_t + (1\text{-}2\mu) P / (F_t - (1\text{-}\mu) P)\}} -1]$

t = 13.433=13.5 mm

OD of cylinder

$D_0 = D_i + (2 \times t)$ ---------------------------------------------------------------------- (3)

$D_0 = 63 + (2 \times 13.5)$

$D_0$ = 90 mm.

Standard $D_0$ = 100 mm

d) **Calculation For Hoop Stress($F_t$):**

$F_t = \{P(d_o^2+d_i^2)\}/ (d_o^2-d_i^2)$ ---------------------------------------------------------- (4)

$F_t$=102.25 $N/mm^2$.

We got hoop stress equal to allowed working stress ($F_t$ =103 $N/mm^2$).

So design is not secure, we take the standard thickness as 25 mm.

**Hoop Stress($F_t$) at t = 25 mm:**

$F_t$= 61.63 $N/mm^2$ at 25 mm thickness,

The above calculated hoop stress is not exceeding than allowed working stress so Design is safe.

Standard outer diameter of cylinder = 120 mm

- **Piston rod design**

Material: Steel, EN–8

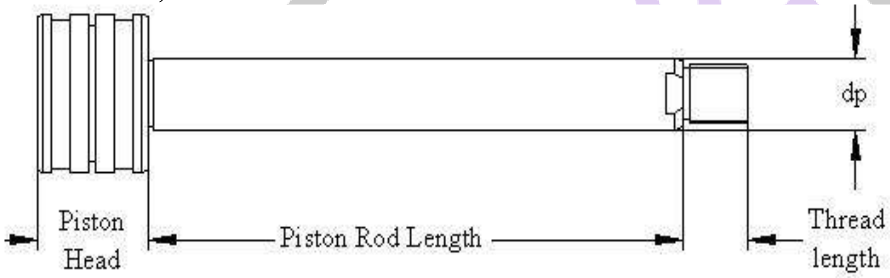


**Figure 1: Piston Rod**

Assumptions made

1) Bar is straight and homogenous material.
2) Rod is continuous and no abrupt changes occur.
3) Line of force acting coincides with the axis of ram/rod.

For pressure of 350 bar

Load F = A * P

$= \{\frac{\pi}{4} \times (63)^2\} \times 35$

= 117720 N

Force = Area x stress

$F = A \times \sigma_t$

For EN-8 material tensile strength is 541.985 N /$mm^2$

For FOS of 4, then

$\sigma_t$ = 541.985 / 4

= 135.5 N /$mm^2$

$117720 = \{\frac{\pi}{4} \times (d_p)^2 \times 135.5\}$

$(d_p)^2 = \frac{(117720 * 4)}{(\pi * 135.5)}$

$d_p$ = 33.26 mm (Minimum diameter)

Standard $d_p$ = 45 mm

- **Design of pillar(column)**

Material: Structural steel St-42

Force acting on each pillar= 117720/4 = 29430 N

Working stress= $F_t = \sigma_t$= 412.02/4 = 103.00 $N/mm^2$

$\sigma_t$ = F / A, F= $\sigma_t$.A
103x(∏/4)xd$^2$ = 29430
d=19.07 (minimum diameter of pillar)
Standard d= 30 mm

- **Stresses acting on the pillar(column)**
  $\sigma_t$ = F / A = 29430/706.85= 41.63 N/mm$^2$
  $\sigma_t$= 41.63 N/mm$^2$
  F= force acting on pillar
  L= length of pillar
  A= C/S Area of pillar
  E= Young's modulus
- **Deformation of pillar**
  δ=(FL)/(AE) ------------------------------------------------------------------------ (5)
  δ= 0.109 mm

➢ **Design of upper platen**

Material: EN-24
Tensile strength= 800 N/mm$^2$
Working stress= $\sigma_t$= 800/4 = N/mm$^2$
To find thickness of platen[19]

$t_1=k_3\sqrt{\frac{a.b.F}{\sigma_t(a^2+b^2)}}$ ------------------------------------------------------------------ (6)

a= length= 420 mm
b= width= 420 mm
K= co-efficient of material= 3 (steel)
F=force acting= 117720 N

$t_1$ = 51.46
Standard thickness is 52 mm

➢ **Bending stresses acting on platen**

$\sigma_b$= (M.y)/I ---------------------------------------------------------------------------- (7)

d= Thickness
b= Width= 420 mm
W=F=force acting= 117720 N
y= Distance from neutral axis

I= (bd$^3$)/12= (420x52$^3$)/12= 4921280 mm$^4$
M= (Wb)/4= (117720x420)/4= 12360600 N-mm
y= b/2=420/2= 210 mm
$\sigma_b$= 52.74 N/mm$^2$

➢ **Design calculation of C-Type Hydraulic Press**

**To find the thickness of each plate**
Material: Structural steel St-42
Force acting on each plate = 117720/2 = 58860 N
FOS: 4
Working stress= $F_t$= $\sigma_t$= 412.02/4 = 103.00 N/mm$^2$
C-Type plates are under to direct and bending stress; maximum stresses are acting at inner fibres.

$\sigma_t= \sigma_d + \sigma_b$

$\sigma_t= \frac{F}{A} + \frac{M*y}{I}$ ----------------------------------------------------------------------- (8)

W=F=force acting= 58860 N
y= Distance from neutral axis = 200 mm
M= Bending moment= 58860*(200+200) = 23544000 N-mm

I=Moment of inertia= $\frac{bh^3}{12}$ mm$^4$

$103 = \frac{58860}{400*b} + \frac{58860(200+200)*200}{\frac{b*400^3}{12}}$

b = 10 mm (minimum thickness of each plate)
Standard thickness of plate 12 mm

➢ **To find the thickness of bolster plate**

Assume the bolster plate as simply support beam

$\sigma_t = \frac{M*y}{I}$ N/mm$^2$

$M = \frac{F.L}{4} = \frac{117720*250}{4} = 7357500$ N/mm$^2$ ------------------------------------- (9)

$y = \frac{t}{2}$ mm

$t = 41.4 \approx 42$ mm

## V. Static structural analysis

ANSYS is the mathematical solution software used to present the static condition analysis. The input loading conditions that can be given in a static condition analysis are moment; force and pressure the output of this analysis are displacement, stress and strain. A static structural analyses is used for determine the result of steady loading conditions by ignore the effects of the inertia and the damping.

- **Analysis of cylinder**

**Stress and deformation generated in cylinder**

61.98 MPa is the Stresses generated inside the cylinder as shown in figure 2 due to the applied pressure. Total deformation of cylinder is 0.0043 mm as shown in the figure 3.

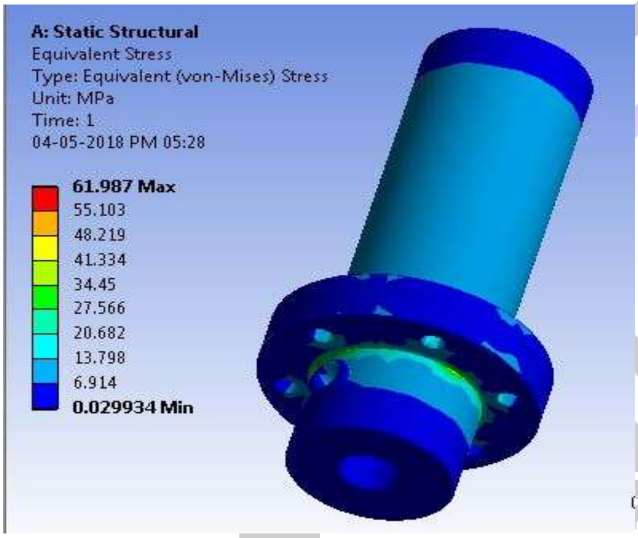


**Figure 2: Stresses in the cylinder**

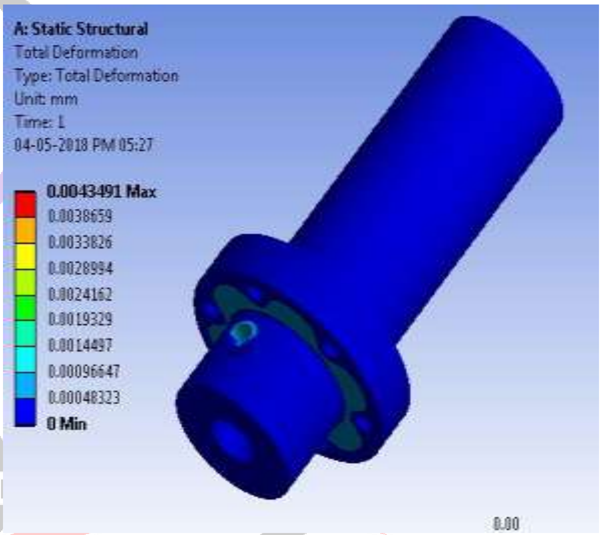


**Figure 3: Total deformation of cylinder**

- **Analysis of piston rod**

**Stresses and Deformation**

Stress of 80.08 MPa is generated in the piston rod due to applied load as shown in figure 4. Total deformation of 0.15 mm is found due to the applied load in the piston rod as shown in the figure 5.

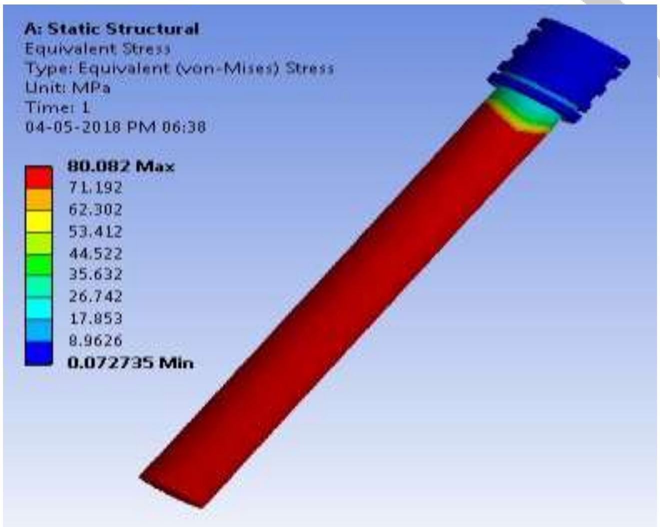


**Figure 4: Stress generated in the piston rod**

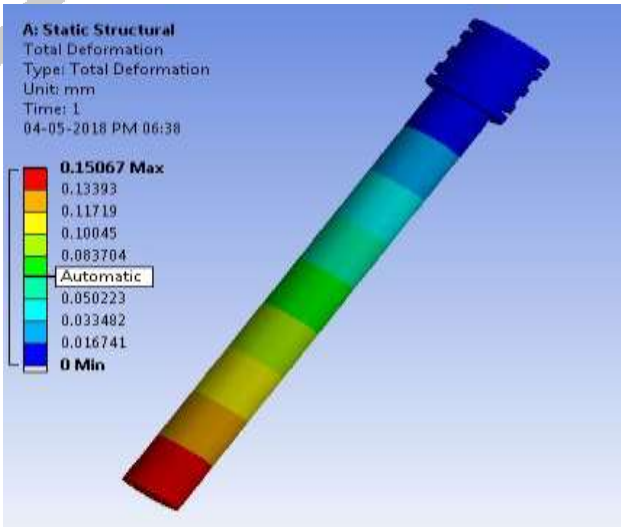


**Figure 5: Total deformation in the piston rod**.

## ➢ Analysis on Movable Platen

### Stress and Deformation in the movable platen

Stress of 57.15 MPa is generated in the movable platen due to applied load as shown in figure 6. Total deformation of 0.09 mm is found due to the applied load on the movable platen as shown in the figure 7.

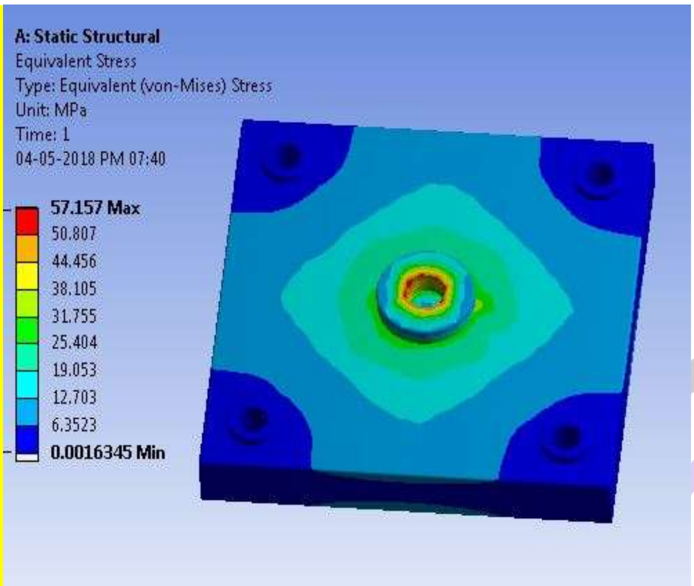


**Figure 6: Stress in the movable platen**

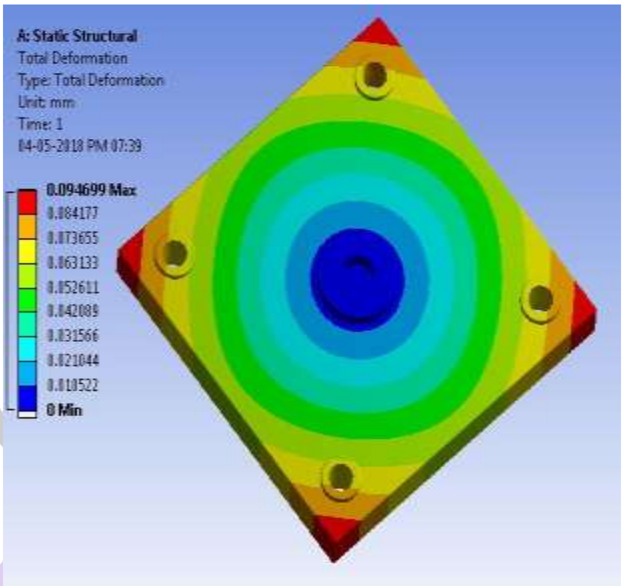


**Figure 7: Deformation in the movable platen**

## ➢ Analysis of Column (Pillar)

### Stress and Deformation

Stress of 43.68 MPa is generated in the column due to applied load as shown in figure 8. Total deformation of 0.052 mm is found due to the applied load on the column as shown in the figure 9.

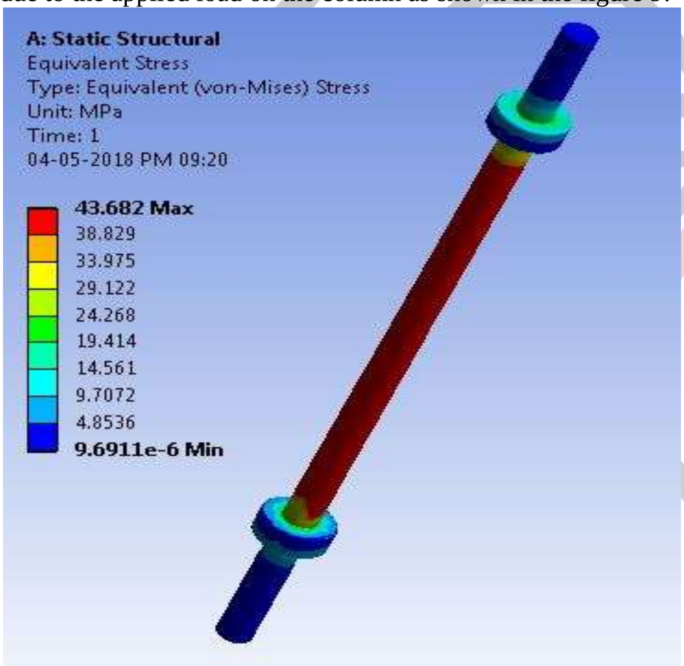


**Figure 8: Stresses in the column**

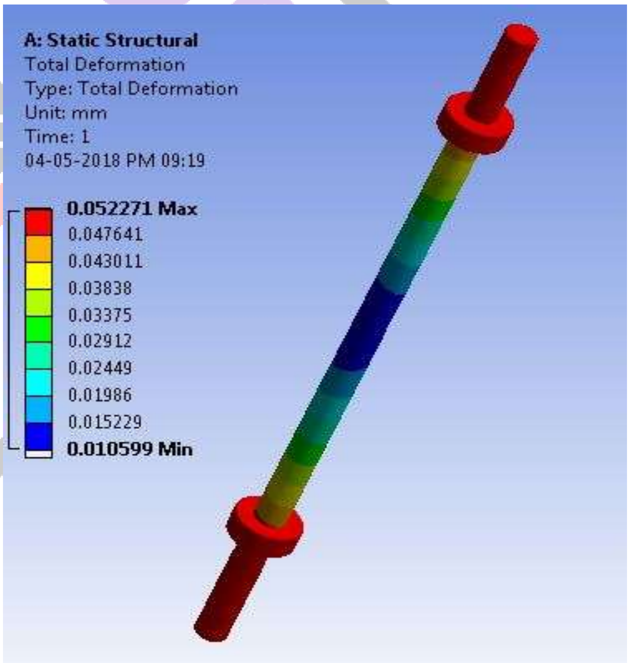


**Figure 9: Total deformation in the column**

## ➢ Analysis of 4-Column Type assembly

### Stress and Deformation

Stress of 127.27 MPa is generated in the Assembly due to applied load as shown in figure 10. Total deformation of 0.07 mm is found due to the applied load on the assembly as shown in the figure 11.

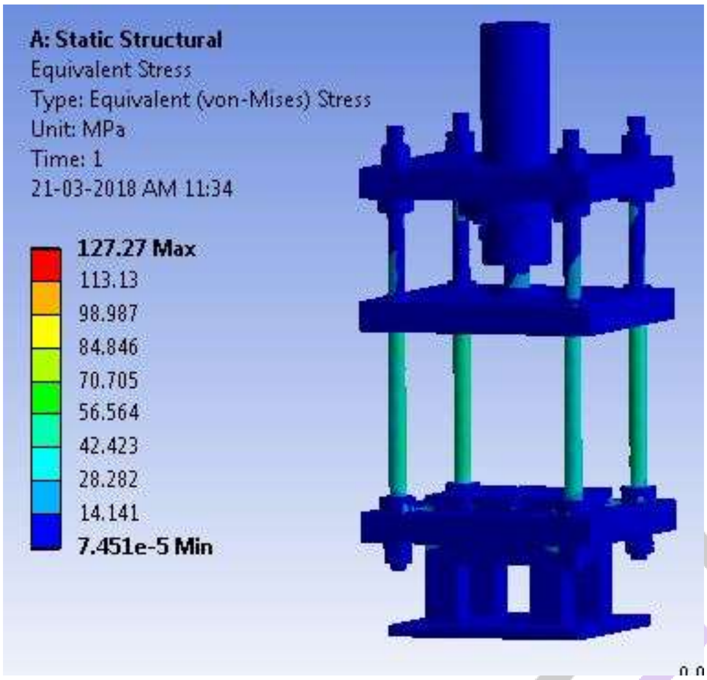


Figure 10: Stress in the Assembly

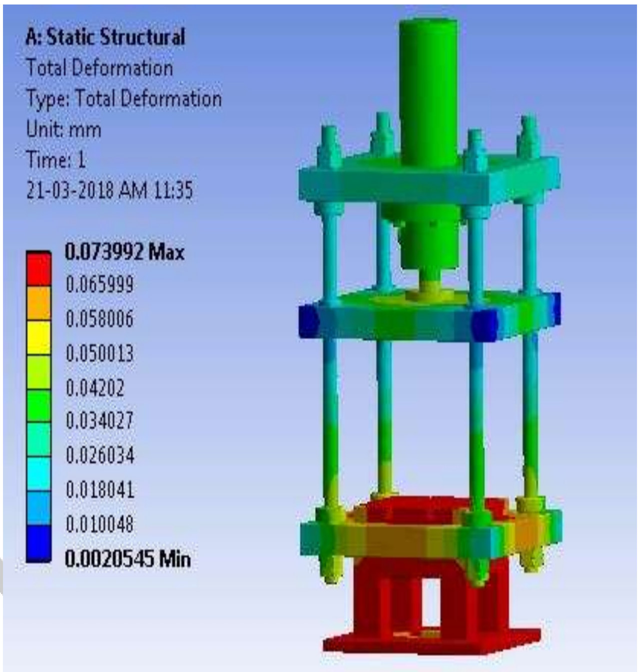


Figure 11: Total deformation in the assembly

- **Analysis of C-Type Hydraulic press assembly**

**Stress and Deformation**

Stress of 124.17 MPa is generated in the Assembly due to applied load as shown in figure 12. Total deformation of 0.92 mm is found due to the applied load on the assembly as shown in the figure 13.

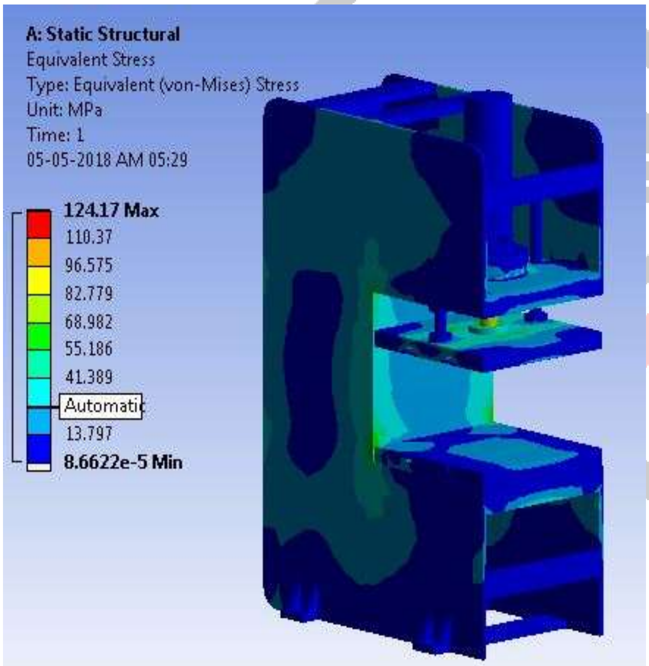


Figure 12: Stress generated in the Assembly

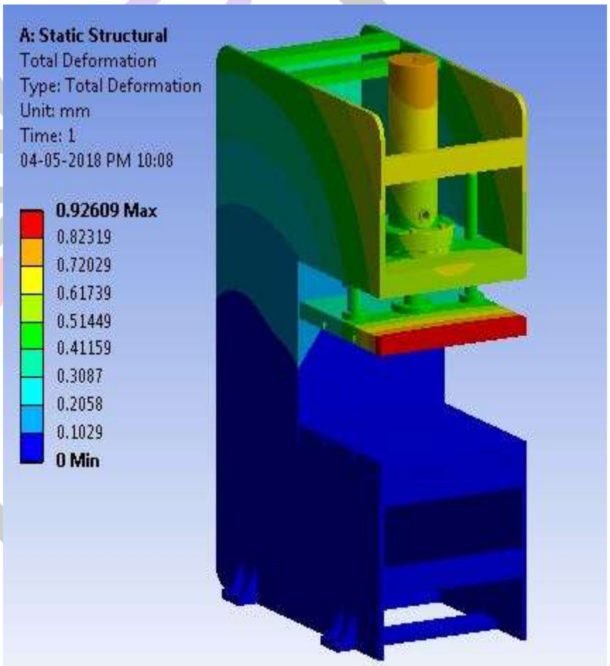


Figure 13: Total deformation of assembly

## VI. Results and Discussion

### 4-Column press versus C-type press Assembly results

Table 2: 4-Column press versus C-type press Assembly results

| Type | Stress (N/mm$^2$) | Total deformation (mm) | Weight (kg) | Estimated cost (Rs.) |
|---|---|---|---|---|
| 4-Column hydraulic press | 127.27 | 0.073 | 329 | 39926 |
| C-type hydraulic press | 124.17 | 0.92 | 445 | 49825 |

In the design of 4-Column hydraulic press and C-type hydraulic press the maximum stress acting on the structure and the maximum deflection regarding static analysis is lesser than the yield stress hence design is safe.

### Analytical versus Numerical comparison

Table 3: Analytical versus Numerical comparison

| S. No. | Part | Theoretical Stress(MPa) | Analysis Stress (MPa) | Error (%) |
|---|---|---|---|---|
| 1 | Cylinder | 61.63 | 61.98 | 0.56 |
| 2 | Piston rod | 74.01 | 80.08 | 8.20 |
| 3 | Movable platen | 52.74 | 57.15 | 8.30 |
| 4 | Column | 41.63 | 43.68 | 4.91 |

Table 3 shows Analytical versus Numerical comparison of different parts of 4-Column hydraulic press in which the errors are within the permissible limit.

## VII. Conclusions

In this work the 4-Column hydraulic press and C-type press is Designed and Analyzed with standards. The maximum stresses induced in the 4-Column hydraulic press and C-type press machine is less than the permissible stress of the material.

Further decrease of material may affect design and gives high deformation and stresses. Which are not acceptable for the machine, as this machine is used for compressive pressing operation in sheet metal industries where the deformation of the press should be permissible. Pressing accuracy plays important role to maintain the close tolerance and to increase productivity.

As analysis shows maximum deformation found is lesser than 1 mm which is tolerable for such hydraulic pressing operations. In this design, the decrease in frame and bed thickness of C-Frame is done so there is a decrease in weight.

It is a multi-purpose machine which is used for performing pressing and forging tasks. By changing the die different operation like blanking, sheet metal and bending etc. can be performed on a hydraulic press machine by varying input to the cylinder which is restricted by output of pump.

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