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Design, Analysis and Optimization of 10 TON Pneumatic Press Machine

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ABSTRACT: Power press working is defined as chip less manufacturing process. We can also say as cold stamping process. The machine used for press working is known as press. This Project work deals with the Design, Finite element analysis and structural optimization of 10 Ton Pneumatic Press Machine. The aim is to reduce the Wight and cost of the Pneumatic press without reducing the quality of the output. Using the best possible resources in design can affect decrease in the weight and cost of the press machine. One way of doing, it will be the optimizing the volume of material utilized for building the complete structure of machine .Here we have consider an industrial application project consisting of mass minimization of a Pneumatic press. For analysis Purpose ANSYS Software has been used.

KEYWORDS: FEA, Structural optimization, Wight reduction,

I. INTRODUCTION

Press Machine are used for producing large quantities of articles quickly, accurately and economically from the cold working of mild steel and other ductile materials. A Press Machine is a machine that supplies force to die used to form, blank or shape metal or non-metallic materials. The Metal forming manufacturing process is almost chip less. Press tools are used to carry out this operation. Deformation of work piece to desired size is done by applying pressure. It consists of bed, frame or bolster plate, Pillar. The ram exert force upon sheet metal or working material through unique tools mounted on the bed or ram. The Energy supplied by a pneumatic cylinder in a pneumatic press is transferred to the ram to provide straight movement. Presses are considered best and most capable way to form a sheet metal into final finished products. Pneumatic presses are commonly used for punching, forging, molding, clinching, blanking, deep drawing and metal forming operation. Pneumatic press is used for producing huge quantities of articles economically, quickly and accurately. The components which are produced range over a very wide field and are used all over industry. By means of particularly designed press tools and combination of operations, most of the sheet parts of any shape are produced. The selection of the proper press and design of die or tool to be mounted on it is very important for any operation to be carried out on the Press Machine.

A. Press Machine working

We know that there are mainly three types of power presses mechanical, hydraulic, and pneumatic. There control systems may be
mechanical or electro-mechanical. Through these three major types of power presses share some common features, the mechanical power press is the most commonly used and researched. In power press two major are stationary bed and a moving ram. Mechanical power press works on the principle of reciprocating motion and the main components for power transmission are the flywheel, and crankshaft, clutch. A motor gives the rotation motion to flywheel and clutch is used for couple the rotation flywheel to the crankshaft. The crankshaft converts the rotary motion of the flywheel to the downward and upward motions of the press ram. A work piece is fed into the lower die, either automatically or manually, and the machine cycle is initiated. On the down stroke, the ram (with an upper die) moves toward the area of operation. When the upper and lower dies press together on the stock material, a re-formed piece is produced. Once the down stroke is completed, the formed work piece is removed and a new work piece fed to the machine and process repeated.

II. OBJECTIVES

The following objective are Considered in the present analysis work for Press Machine,
- To upgrade the design from Mechanical press machine to Pneumatic press machine.
- To study the stress distribution along the Press machine and optimize the product.

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- The stress developed can be understood and suitably reduced by increasing or decreasing the dimensions or by changing design of frame structure. Optimize the volume of material by using structural optimization method.
- Control on human safety factors to reduce accidents.

III. DESIGN CALCULATIONS OF PRESS MACHINE

Following are the assumptions had been made for frame structure.

- The load is considered as a perfectly vertical.
- Frame material is homogeneous and isotropic.
- The base is bolted to a solid foundation so all the deflections of the base plate are zero.
- Frame is having the symmetrical cross section area.

The frame is the base machine element in press. It is designed by the following steps.

1. Function

The main function of the frame is to withstand the force developed by the RAM. Frame is used for mounting and housing the press accessories like ram, die block, motor, flywheel, gears etc.

2. Determination of forces

The capacity of the press machine determines the major forces acting on the Press frame structure.

3. **Material Specifications**

Specification of Material
Designation: S275.
Tensile strength: 370 to 530 MPa.
Density: 7850 kgf/m^3.
Young's Modulus: 2×10^5 N/mm^2.
Poisons Ratio: 0.3.
Factor of Safety: 4.

B. Design and calculation of Existing Mechanical Press Machine Frame

Design of C type mechanical press machine frame has been done in solid works software.

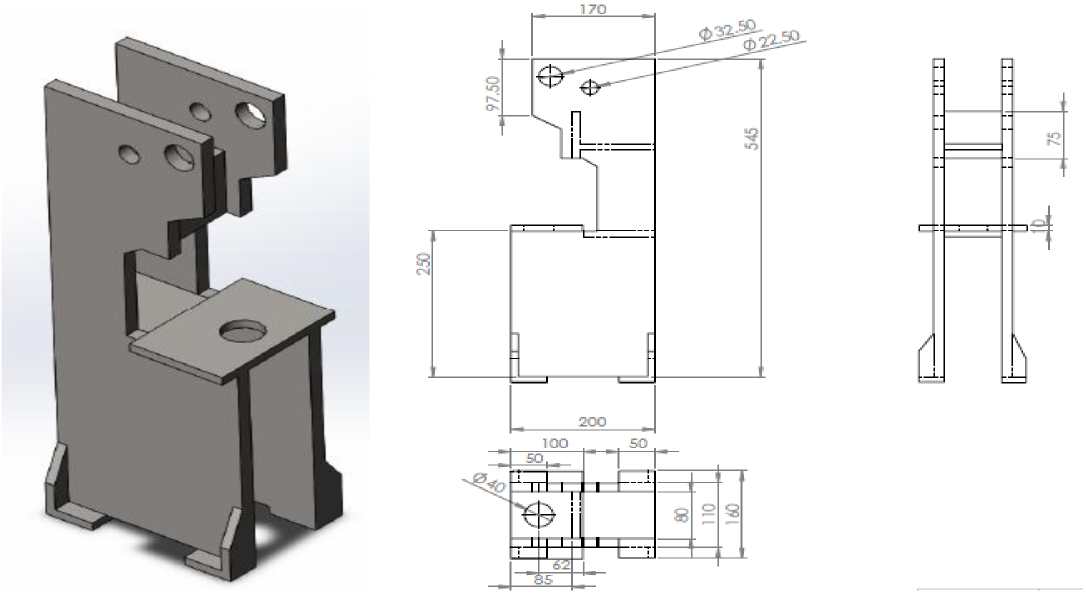

Figure 1: 3D Design of C type mechanical Press Machine Frame

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C. Specification of 10 Ton mechanical Press Machine

Capacity (ton)	10
Diameter of Crankshaft	32.5
Stroke Adjustment	06 -50
Max. Dist Bolster to Ram (SUAU)	195
Side Adjustment	25
Side Face FB X LR	1100 X 2000
Size of Table (FB X LR)	1000 X 1600
Bolster Thickness	100
Flywheel Diameter	700
Stroke per minute	70
Weight Approximate (kg.)	550
Electric Motor (H.P./R.P.M.)	1/1440

Table 1: Specification of 10 ton Mechanical Press Machine

D . Formulas and calculations of Existing Mechanical Press Machine Frame

The frame subjected to direct tensile stress and bending stresses

$$\sigma_{total} = \sigma_{tensile} + \sigma_{bending} \ \mathrm{N/mm^2}$$

$$\sigma = \frac{P}{A} + \frac{M_b Y}{I} \mathrm{N/mm^2}$$

Where
σ = Permissible stress in $\mathrm{N/mm^2}$
P = Applied load/ Force in N
A= Area of the plate section in $\mathrm{mm^2}$
M_b = Bending moment in N. mm
e = Eccentricity in mm
y = Distance from the neutral surface to the extreme fiber in mm
I = Moment of inertia in $\mathrm{mm^4}$
Applied load P =10 tones
P = 10000 × 9.81
P = 98100 N

Cross Section Area of Solid Rectangle = (B X H)
A = 160 X 545 = 87200 $\mathrm{mm^2}$

Distance from the neutral surface to the extreme fiber in mm $y' = \frac{Ay1+Ay2......}{A}$

$$y' = \frac{(87200\ X\ 272.5)}{87200}$$

$$y' = 272.5 \text{ mm}$$

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Moment of inertia for rectangular plate sections $I_{xx} = \frac{bh^3}{12} mm^4$

$$I_{xx} = \frac{545^3 X\ 160}{12}$$

$$I_{XX} = 2158.44 \text{ X } 10^6 \text{ mm}^4$$

Direct Stress $\sigma_{tensile} = \frac{\text{Load}}{\text{A}} \frac{\text{N}}{\text{mm}^2}$

$$\sigma_{tensile} = \frac{98100}{87200} = \mathbf{1.125 N/mm^2}$$

Bending Stress $\sigma_{bending} = \frac{M_b Y}{I}$ N/mm^2

Eccentricity e = 28 mm

Bending moment M = $(Load) \times e$

= 98100 X 272.5

= 26.734 X 10^6 Nmm

Bending Stress

$$\sigma_{bending} = \frac{26.734\ X\ 10^6 \times 272.5}{2158.44 \text{ X } 10^6}$$

$\sigma_{bending}$ = 3.37 N/mm^2

Total Stress σ_{total} = (1.125 +3.37) **N**/mm^2

σ_{total} = 4.5 N/mm^2

Max Allowable Stress **σ = 410/4 = 102 N/mm^2**

Deformation Calculation

Linear elastic deformation is governed by Hooke's law, which states:

$$\sigma = E\,\epsilon$$

Where σ is the applied stress,
E is a material constant called Young's modulus or elastic modulus,
ϵ is the resulting strain.

$$\epsilon = \frac{\sigma}{E}$$

$$= \frac{4.5}{2.1\ x\ 10^5}$$

$$\boldsymbol{\epsilon = 0.021m}$$

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E. Design and calculation of Improved Pneumatic Press Machine Frame

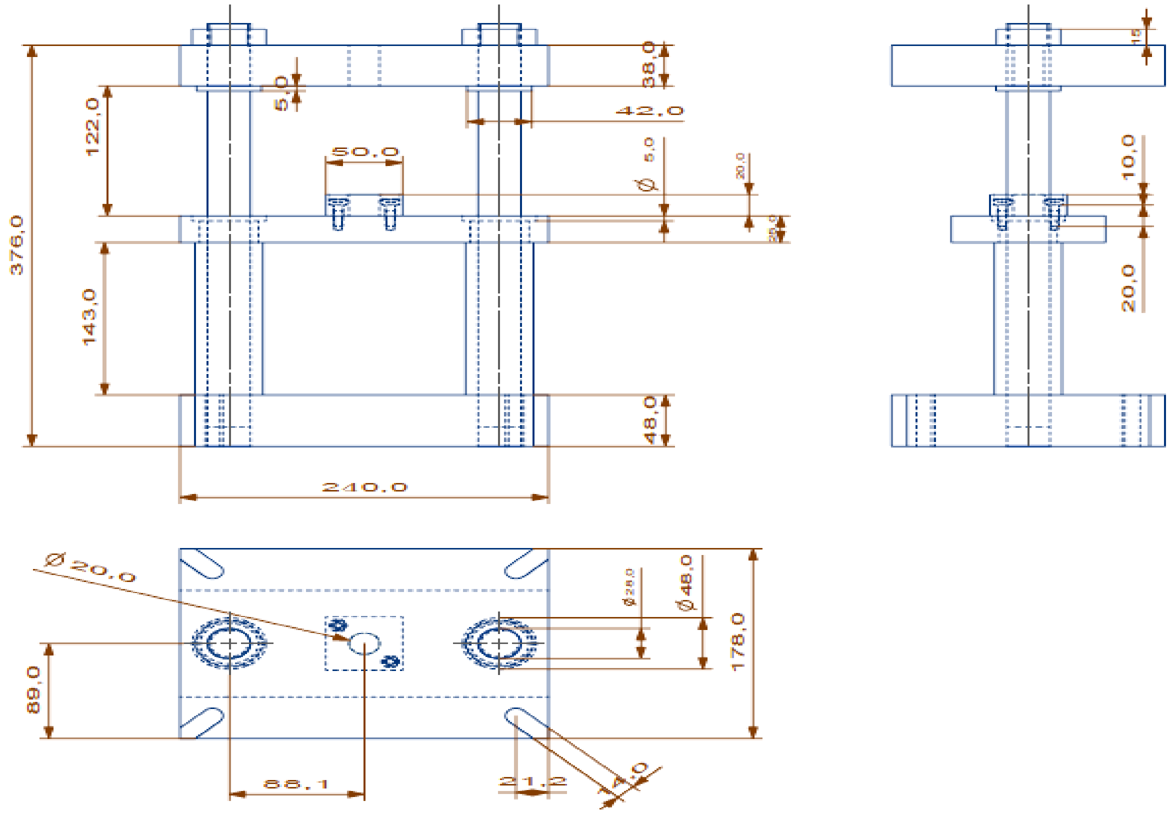

Figure 2: Design of 10 Ton H type Pneumatic Press Machine frame

F. Technical Specification of 10 Ton Pneumatic Press Machine

Capacity (ton)	10
Stroke Adjustment	10 – 40
Max. Dist Bolster to Ram (SUAU)	143
Size of Table (FB X LR)	178 X 240
Opening in the Table Die	265
Bolster Thickness	48
Stroke per minute	65
Weight Approximate (kg.)	75
Approx overall Size FB X LR X HT	178 X 240 X 376
Air Pressure Required (bar)	5

Table 2 : Technical Specification of 10 Ton Pneumatic Press Machine

G. Formulas and calculations of Improved Pneumatic Press machine Frame

The frame subjected to direct tensile stress and bending stresses

$\sigma_{total} = \sigma_{tensile} + \sigma_{bending}$

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$$\sigma = \frac{P}{A} + \frac{M_b Y}{I} \mathrm{N/mm^2}$$

Where
σ = Permissible stress in N/mm^2
P = Applied load/ Force in N
A= Area of the plate section in mm^2
M_b = Bending moment in N. mm
e = Eccentricity in mm
y = Distance from the neutral surface to the extreme fiber in mm
I = Moment of inertia in mm^4
Applied load P =10 tones
P = 10000 × 9.81
P = 98100 N

Cross Section Area of Solid Rectangle = (B x H)
A_1 = 48 X 240 = 11520 mm^2
A_2 = 25 X 240 = 6000 mm^2
A_3 = 38 X 240 = 9120 mm^2
cross sectional Area of hollow cylinder $= \frac{\pi}{4(D^2-d^2)}$

A_4 $= 2\left(\frac{\pi}{4}(48^2-28^2)\right)$ = 2387.61 mm^2

cross sectional Area of Solid cylinder $= \frac{\pi}{4(D^2)}$

A_5 $= 2\left(\frac{\pi}{4}\,28^2\right)$ = 1231.50 mm^2

Total Area A = $A_1+A_2+A_3+A_4+A_5$
=11520+6000+9120+2387.61+1231.50
= 30259.11 mm^2

Distance from the neutral surface to the extreme fiber in mm $y' = \frac{Ay1+Ay2+Ay3+Ay4+Ay5}{A}$

$y_1 = \frac{48}{2} = 24\ mm$

$y_2 = \frac{25}{2} + 191 = 203.5\ mm$

$y_3 = \frac{38}{2} + 338 = 357\ mm$

$y_4 = \frac{143}{2} + 48 = 119.5\ mm$

$y_5 = \frac{356}{2} + 20 = 198\ mm$

$$\text{Total } y' = \frac{(11520\ X\ 24)+(6000\ X 203.5)+(9120 X\ 357)+(2387.61X119.5)+(1231.50X198)}{30259.11}$$

y' = 174.57 mm

Moment of inertia for rectangular plate sections $I_{xx} = \frac{bh^3}{12} mm^4$

$$I_{xx1} = \frac{48^3 X\ 240}{12} = 2.21 \text{ X } 10^6 \text{ mm}^4$$

$$I_{xx2} = \frac{25^3 \text{X } 240}{12} = 0.312 \text{ X } 10^6 \text{ mm}^4$$

$$I_{xx3} = \frac{38^3 X\ 240}{12} = 1.09 \text{ X } 10^6 \text{ mm}^4$$

$$I_{xx4} = 2\left(\frac{\pi\,(48^4-28^4)}{64}\right) = 0.460 \text{ X } 10^6 \text{ mm}^4$$

$$I_{xx5} = 2\left(\frac{\pi\,(28^4)}{64}\right) = 0.0602 \text{ X } 10^6 \text{ mm}^4$$

Total I_{XX} = $(Ixx_1+Ixx_2+Ixx_3+Ixx_4+Ixx_5)$
= (2.21X 10^6 + 0.312 X 10^6 + 1.09 X 10^6+ 0.460 X 10^6 + 0.0602 X 10^6)
I_{XX} = 4.1322 X 10^6 mm^4

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Direct Stress $\sigma_{tensile} = \frac{\text{Load}}{\text{A}} \text{N/mm}^2$

$$\sigma_{tensile} = \frac{98100}{30259.11} = 3.24 \text{ N/mm}^2$$

Bending Stress $\sigma_{bending} = \frac{M_b Y}{I} \text{N/mm}^2$

Eccentricity e = 28 mm

Bending moment $M = \left(\frac{Load}{2}\right) \times e$

$= 49050 \text{ X } 28$

$= 1.3734 \text{ X } 10^6$ Nmm

Bending Stress

$$\sigma_{bending} = \frac{1.3734 \ X \ 10^{6} \times 174.57}{4.1322 \ X \ 10^{6}} \text{ N/mm}^2$$

$$\boldsymbol{\sigma_{bending} = 58.02 \text{ N/mm}^2}$$

Total Stress $\sigma_{total} = (3.24 + 58.02)$ **N**/mm^2

σ_{total} = 61.26 N/mm^2

Max Allowable Stress **σ = 410/4 = 102 N/mm^2**

IV. ANALYSIS

A. Analysis Existing Press Machine frame

More time is required for building finite element model than any part of the analysis. Initial, we state analysis title and job name.

Then defined the material properties, type of element, element real constants, and the geometry of model. The Mechanical press consists of C type frame, bed and slide or ram. Structural steel S275 is used for Frame.

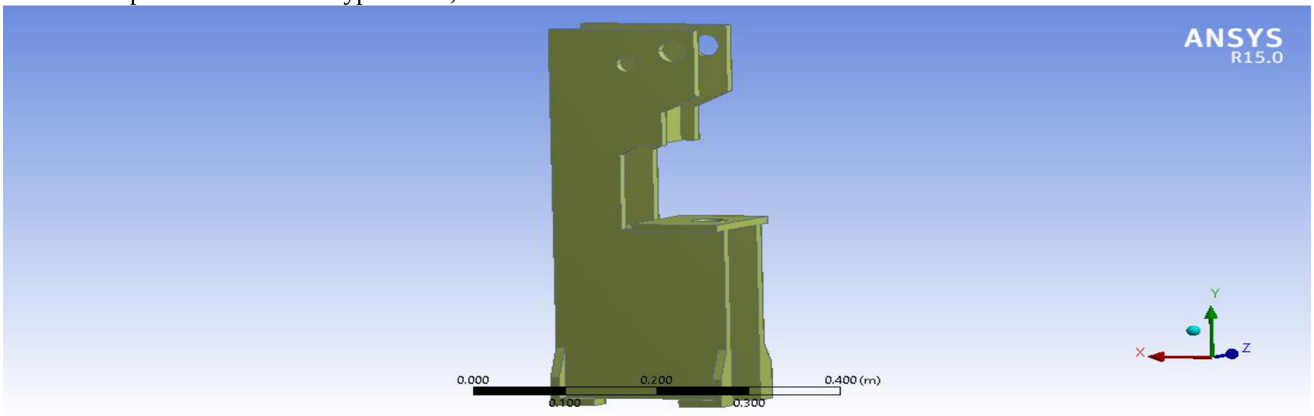

Figure 3: Model of the c type Mechanical press Machine frame

Meshing: - After finishing the modeling of solid machine and set up element attributes and setting meshing controls, we are set to generate the finite element mesh. The element size, element shape and midsize placement of node to be used in meshing of the solid model are established by using meshing controls. For this machine free meshing is used. Meshed view is shown below.

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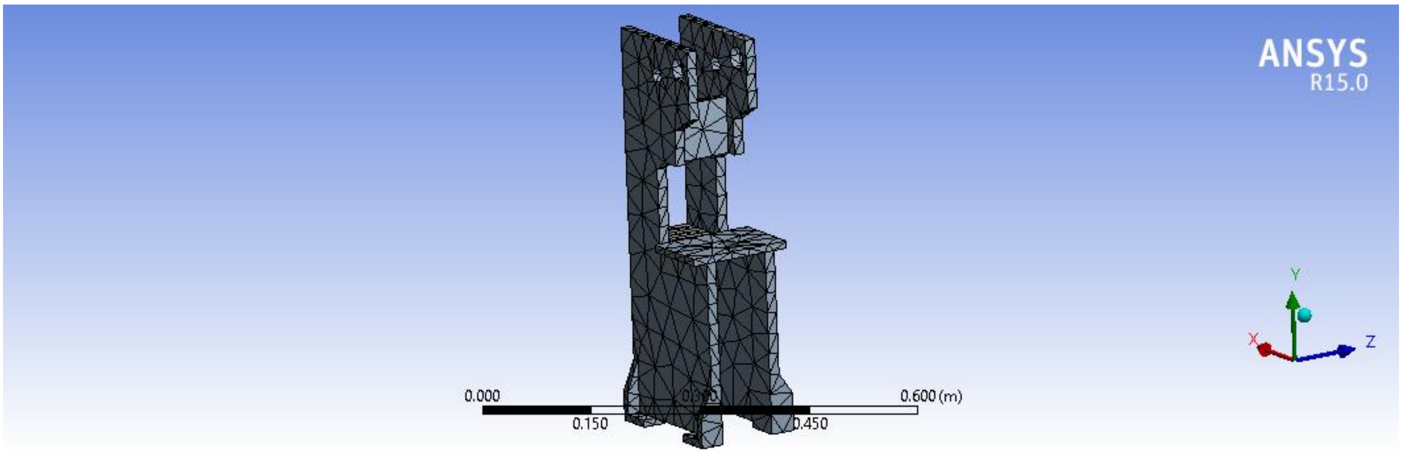

Figure 4: Meshing view of the mechanical press machine frame

Boundary Conditions: - In ANSYS 10 ton load is applied on the Middle supports of frame.

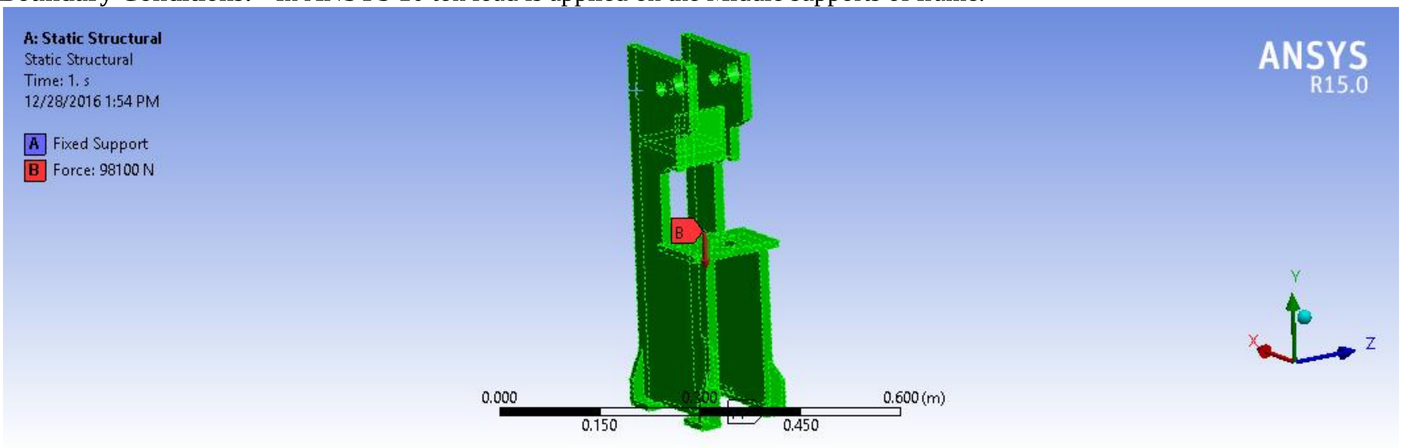

Figure 5: Application of boundary condition on mechanical Press Machine Frame

Analysis:- In this case we can apply most of the loads either on finite element model (on elements and nodes) or on the solid model (on lines, key points areas and faces). Thus on key points or on a node force can be specified. After applying boundary condition, Structural analysis is done to find stress and deformation.

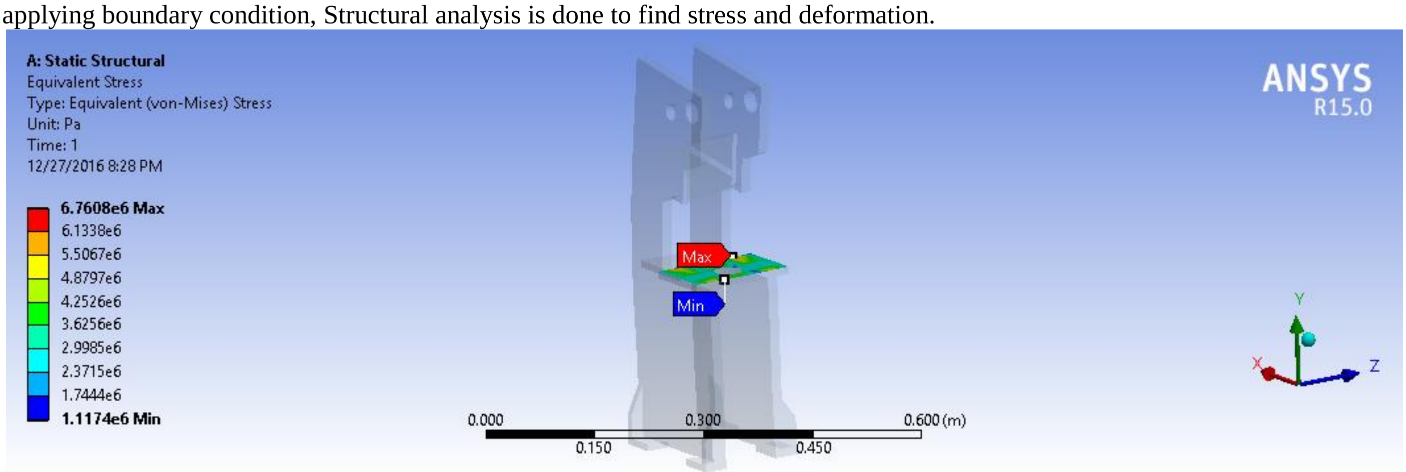

Figure 6: (a)

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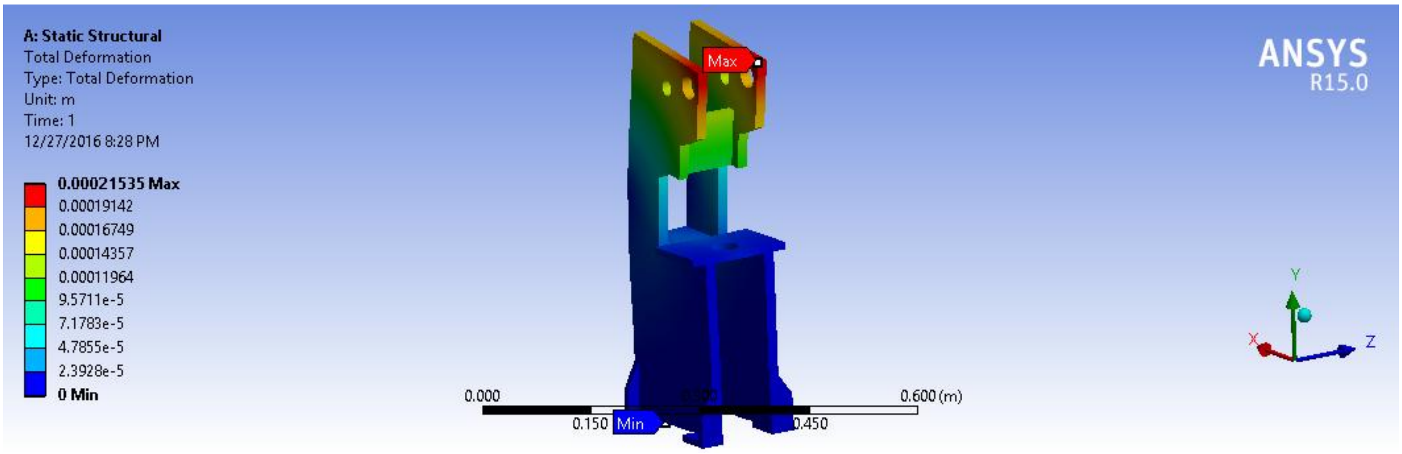

Figure 6: (b)

Figure 6 (a), (b) Maximum stress and deformation on Mechanical Press Machine Frame.

Results Obtained: -

The Result obtained from the analysis of existing mechanical press machine frame are-

1. The Maximum stress induced in the frame= 6.7 MPa
2. The Maximum Displacement of frame = 0.213 M
3. Total weight before optimization= 550 kg

The allowable stress of mild steel or structural steel is 120 MPa and Maximum stress induced in the frame is 6.7 MPa which is very below the allowable stress. Therefore we are able reduce the frame weight, by making some design changes.

H. Analysis Improved Pneumatic Press Machine frame

More time is required for building finite element model than any part of the analysis. Initial, we state analysis title and job name then defined the material properties, type of element, element real constants, and the geometry of model. Structural steel is used for whole assembly.

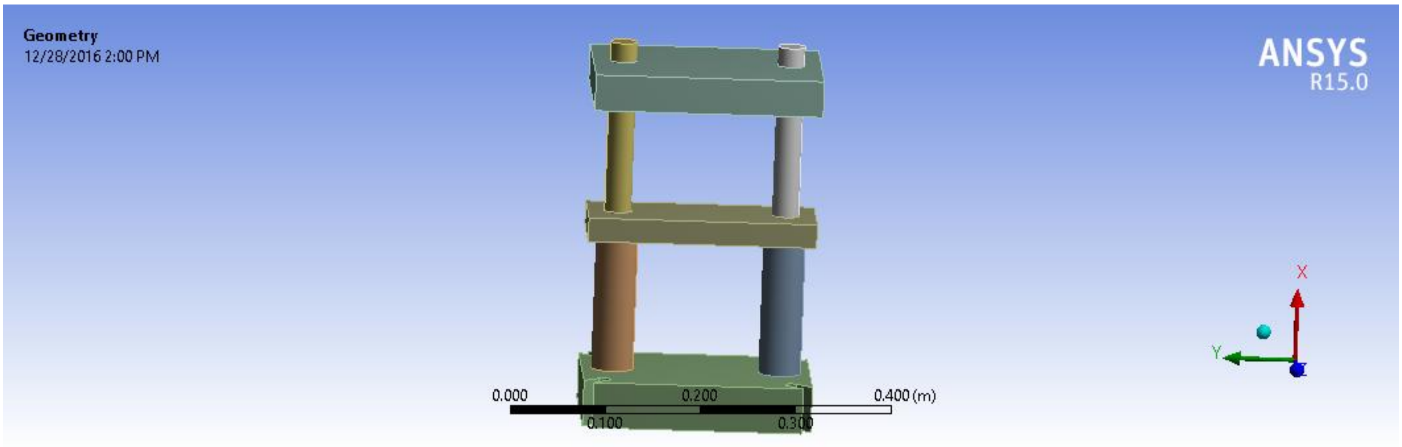

Figure 7: Model of the H type Pneumatic Press Machine Frame

Meshing:- After finishing the modeling of solid machine and set up element attributes and setting meshing controls, we are set to generate the finite element mesh. The element size, element shape and midsize placement of node to be used in meshing of the solid model are established by using meshing controls. For this machine free meshing is used. Meshed view is shown below.

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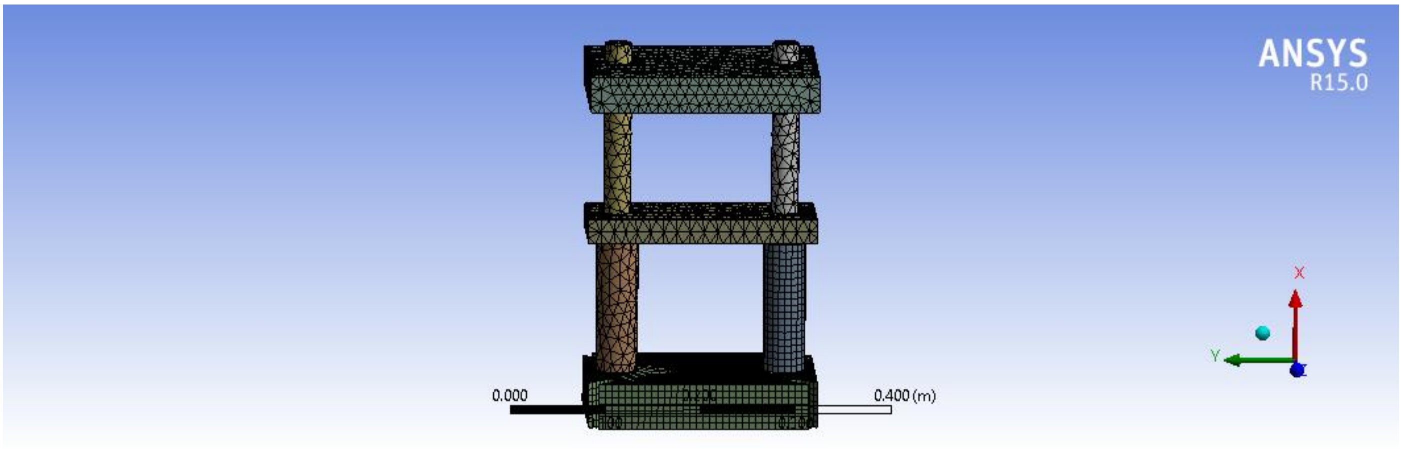

Figure 8: Meshing view of the H type Pneumatic Press machine Frame

Boundary Conditions: - In ansys 10 ton load is applied on the Middle Plate Surface.

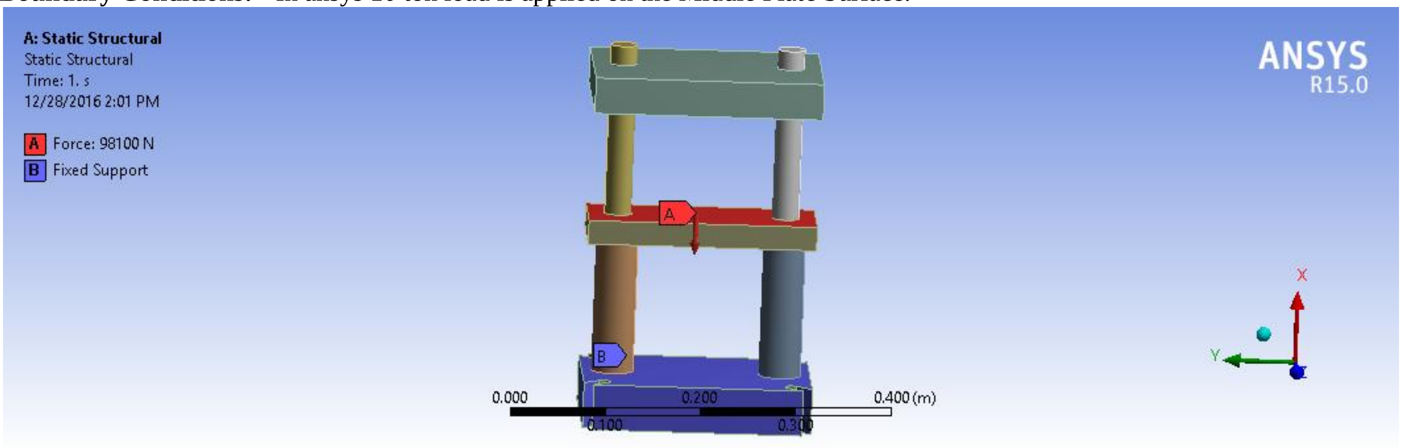

Figure 9: Application of boundary condition on H type Pneumatic Press machine Frame

Analysis:- In this case we can apply most of the loads either on finite element model (on elements and nodes) or on the solid model (on lines, key points areas and faces). Thus on key points or on a node force can be specified. After applying boundary condition, Structural analysis is done to find stress and deformation.

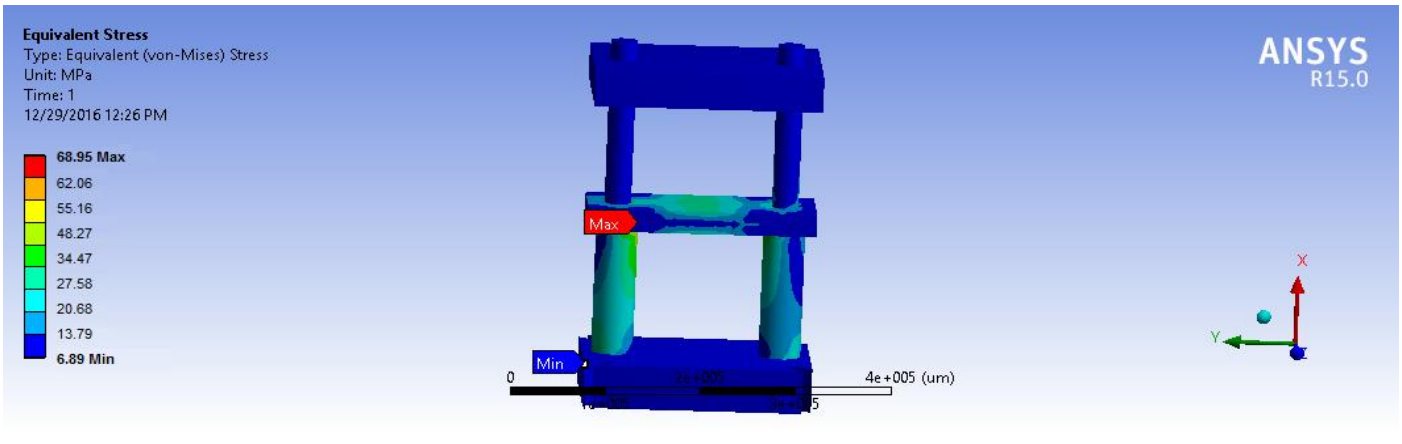

Figure 10: (a)

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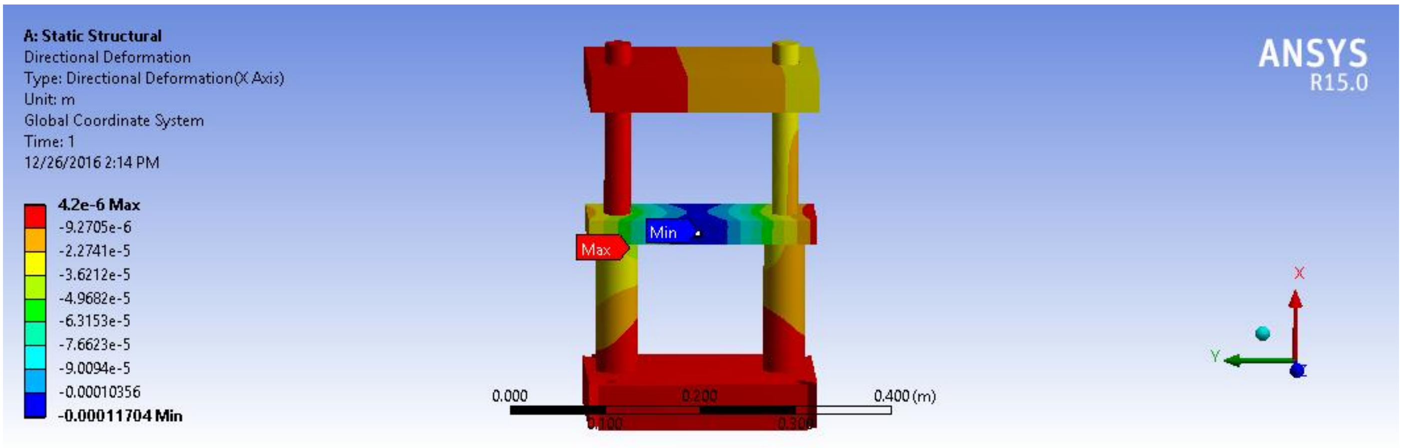

Figure 10: (b)

Figure 10 (a), (b): Maximum stress and deformation of H type Pneumatic Press machine Frame

Results Obtained: -

The Result obtained from the analysis of H type Pneumatic Press machine Frame are-

1. The Maximum stress induced in the frame= 68.95 MPa
2. The Maximum Displacement of frame = 0.0042 m
3. Total weight before optimization= 65 kg

The allowable stress of mild steel or structural steel is 120 MPa and Maximum stress induced in the frame is 68.95 MPa

V. OPTIMIZATION

In this period of hard competition, every industry cracks to have an advantage over their complements by

- Introducing the new products and features to certain range.
- Modifying the previous and making it reliable products.
- By reducing the cost of the component without compromising quality and performance.
- Decreasing the margins of profit.
- By using alternate material for manufacturing.

The main aim of the customer is to get the product of good quality at reasonable price in the market. If the company fulfils the needs of the customer then the company will survive in the market. To survive the company has to modify the product so need to optimize within the available resources. The optimization is done by following three components. Models Created in Solid Works and Imported in Ansys for reducing volume of material the design is changed and Modified as shown in fig. For this machine free meshing is used. Meshed view is shown below.

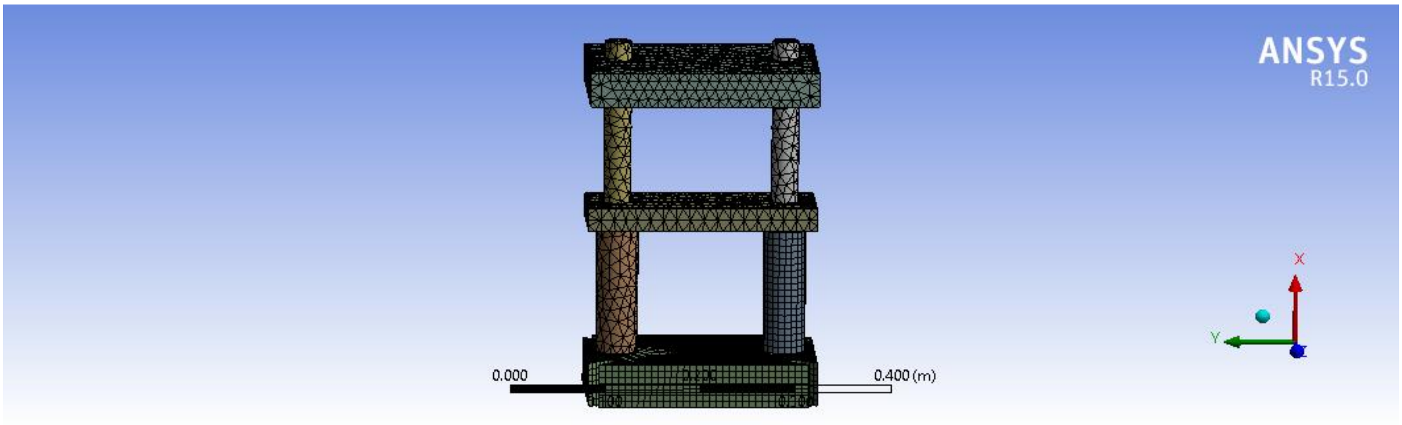

Figure- 11: Meshing view of optimized model of H type Pneumatic Press machine Frame

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Analysis: - After applying boundary condition, Structural analysis is done to find stress and deflection.

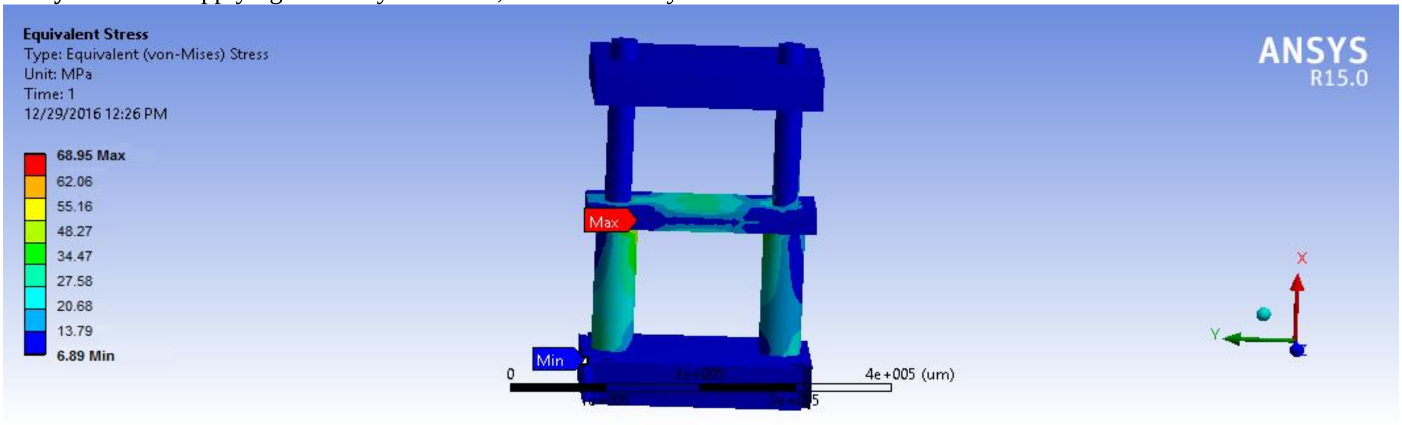

Fig- 12: stresses in structure after optimization of H type Pneumatic Press machine Frame

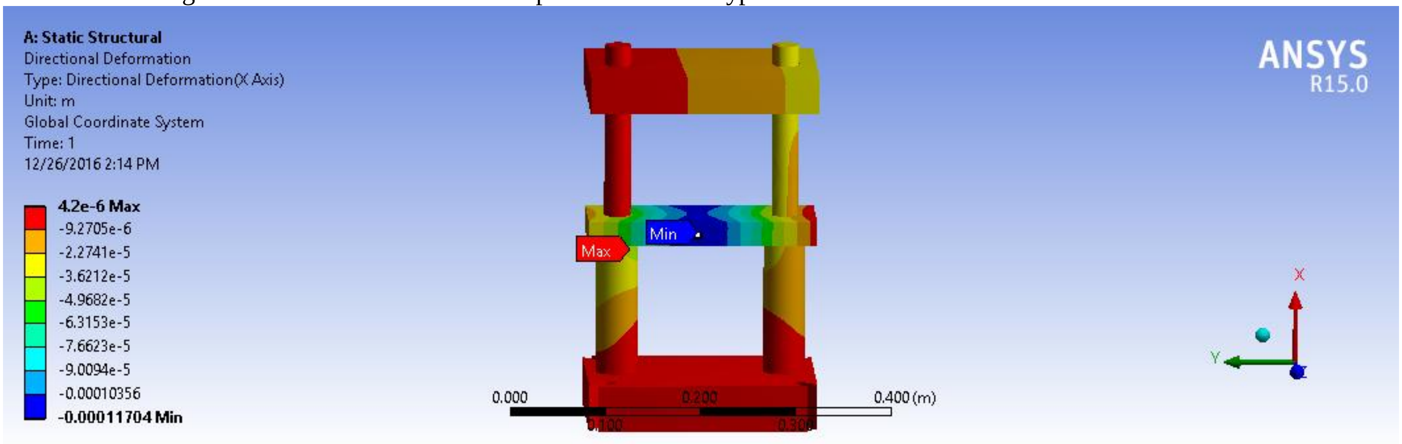

Figure-13: Deflection of Frame after optimization of H type Pneumatic Press machine Frame

Results Obtained after optimization: -

- The Maximum stress induced in the Improved Pneumatic press frame = 66.95 N/mm2
- The Maximum Deformation for 10 ton load on Pneumatic Press machine Frame = 0.0042 m
- Weight of Pneumatic Press machine Frame = 63 kg

Material reduction :-

By Improving design remove material of the frame, weight of the press frame is reduced from 345 kg to 63 kg.
Total Weight reduced = 550 – 63 = **487 kg.**

Comparison between results

Sr. No.	Parameters	Existing Mechanical Press Machine Frame	Improved Pneumatic Press machine Frame
1	Maximum stress in the body.	6.7 N/mm^2	66.95 N/mm^2
2	Maximum deformation of the body.	0.0213 m	0.0042 m
3	Weight of body	550 Kg	63 Kg

Table 3: Comparison between results of Press Machine Frames

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VI. CONCLUSIONS

An attempt was made to analyze and optimize the 10 ton Press Machine using ANSYS software. The project work carried out is successfully designed to meet the requirements as per the constraints. The Press machine frame is carefully designed and cross checked where it does meet the requirements. Press machine is designed using optimum material in order to avoid the excess weight. In this project it has been compared Existing design of press machine Frame with Improved design that have been optimized by using software tool (ANSYS) it has been demonstrate that, under the same loading conditions, constraints, and intended design purpose ANSYS indentifies a lighter design with reduced material cost.

The maximum stress induced in the machine is 66.95 N/mm^2 which is less than the allowable stress of the material. Further reduction of material causes unsafe and gives higher deformations and stresses which are not allowable for the machine, as this machine is used for punching and bending operation in sheet metal industries were the deformation of the press plays important role to maintain the close tolerance between punch and dies. As analysis shows max deformation found is 0.0042 m which is acceptable for such operation. Total weight of the machine reduced after optimization is 63 kg.

In this project, it has been shown a 487 Kg weight reduction of the press machine , which was achieved by changing the design of the frame structure while maintaining structural balance of the press Machine and without affecting on performance. Human Safety factors are controlled by using Hand lever mechanism for operations.

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