Journal of Manufacturing Engineering, September, 2012, Vol. 7, Issue. 3, pp 164-170

STRESS ANALYSIS OF FRAME STRUCTURE OF THREE ROLLER BENDING MACHINE USING ANSYS

***Shakil A Kagzi [1], Anish H Gandhi [2] and Harit K Raval [3]**

[1,3] Department of Mechanical Engineering, SardarVallabhbhai National Institute of Technology, Surat, Gujarat - 395007, India
[2] Department of Mechanical Engineering, C K Pithawalla Collage of Engineering and Technology, Surat, Gujarat – 395007, India

ABSTRACT

Three-roller plate bending machines are used in metal forming industries for bending of sheets/plates in to shells of constant or varying cross section. Maximum blank thickness, maximum blank width and minimum shell diameter combinations that can be pre-bent for given plate material specifies capacity of three-roller bending machine. Roller stands or frames supporting rollers are key structural elements in roller bending machines. Designing of the frame structure is crucial due to complex force pattern in action during static and continuous roller bending and complexity of its geometry. In reported work, behavior of different frame structure of a three-roller bending machine (i.e. maximum Von Mises stress and maximum deflection) for maximum load under static bending of varying cross-section shell is analyzed using Finite Element Analysis (FEA) in Ansys. Different possible modification in the frame structure has been considered using shape optimization module of Ansys. Alternate frame structure designs are analyzed based on FEA results of maximum Von Mises stress and maximum deflection. Reported work may be important to machine tool design engineer to arrive at a satisfactory design of machine structures.

Keywords: Finite Element Analysis, Three-roller Bending Machine, Frame Structure, Von Mises Stress and Deflection

1. Introduction

A three-roller bending machine belongs to the metal forming machine categories that can roller bend plates into cylinder and cones. Three rollers are supported in roller stand or frames so that the rollers can be adjusted to required position for bending plates. The important applications of this machine are in the manufacturing of cylindrical shells and tanks, fuel tanks for launch vehicles in space application programs, large cylindrical containers etc.

Schematic diagram of three-roller bending process assuming uniform curvature between rollers is shown in Fig. 1. Plate is bent to the desired curvature by adjusting the position of top roller and then fed by two side rollers to cover whole length of the plate to be bent. Desired final curvature can be obtained in one or several passes. Center distance between bottom rollers (a) can be varied. Axes of all the three rollers can be set parallel to each other or at relative inclination to bend the shell of uniform or varying cross-section. Post springback curvature in this case is the function of plate thickness (t), plate width (w), material properties (E, n, K, ν), center distance between bottom rollers (a), top roller position (U), top roller radius (r_t) and bottom roller radius (r_1) [1]. Relationship of top roller position (U) with other operating parameters viz. loaded radius (R), center distance between bottom rollers (a) and bottom roller radius (r_1) considering actual contact point shift can be derived to form given by Eq. (1) [1].

$$R = \frac{U}{2} + \frac{a^2}{8U} - r_1 = \frac{4U^2 - 8r_1U + a^2}{8U} \qquad (1)$$

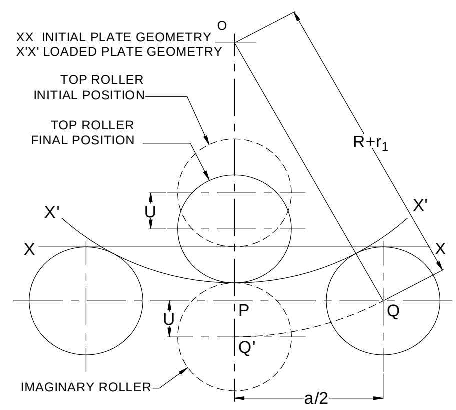


Fig. 1 Schematic Diagram of Three roller Bending Process

**Corresponding Author - E- mail: shakil128@gmail.com*

Fig. 2 shows the six stages of three-roller plate bending operation sequence for fixed bottom roller gap. Top roller load required to bend the plate is the function of various parameters viz. plate thickness, plate width, shell diameter to be roller bend, plate material property, gap between bottom rollers etc. Machine capacity is limited by the factors such as the size of rollers and horsepower of its motor. In the process of continuous roller bending, maximum load is required during the edge pre-bending stage as top roller is at the offset distance from its mid position. So, top roller offset during the pre-bending decides the maximum width and minimum shell diameter combinations that can be roller bend for the particular plate thickness and material property within machine capacity.

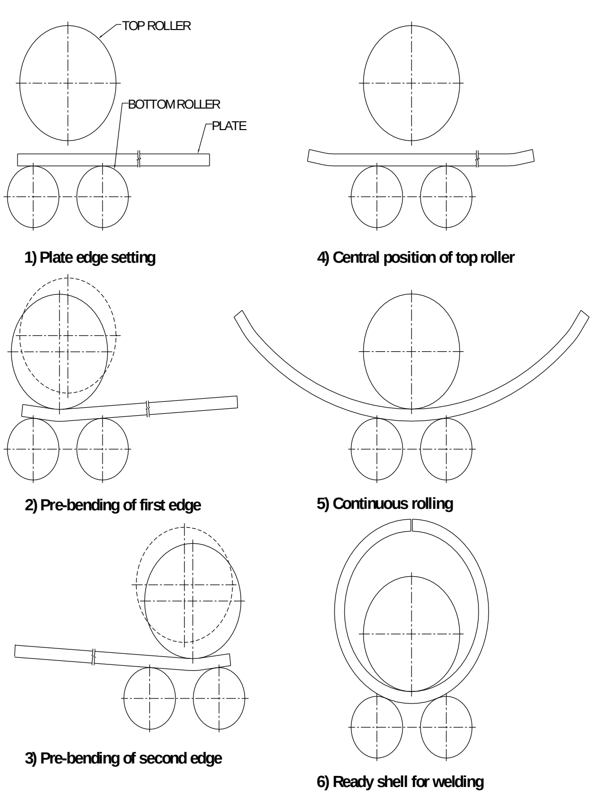


Fig. 2 Roller Bending Sequence for Fixed Bottom Roller Gap

Frames of the machine supporting rollers are key structural component, bearing static and dynamic load during roller bending. Frame geometry should allow the setting of relative positions of the rollers for roller bending of different required shell dimensions. Ramamurti et al. [2] reported parametric study of frame structure using FEA for roller bending of thick plates. They evaluated machine stand performance for different structural modification. Similar, analysis was carried out by Huseyin et al. [5] for machine tool sideways by using two-dimensional sideway model and stress distribution and deformation were determined using finite Element Analysis using Ansys. The method was found to advantageous because of reduced solution times and ease in modeling the structure and to modify it when required for specified deformation and allowable stress distribution. Gandhi and Raval [1] reported an analytical and empirical model to estimate the top roller position as a function of desired radius for three-roller cylindrical bending of plates, considering the contact point shift at the bottom roller plate interfaces and its experimental verification. Geometrical model relating the displacements of rollers in horizontal and vertical planes with the loaded radius of the plates reported by Gandhi and Raval [1] can be used to determine the effect of frame displacements on the loaded radius of the plates. Chudasama and Raval [3] reported mathematical model for the prediction of static load on rollers during bending of plates on three-roller bending machine assuming plain strain condition.

Available literature for designing such a machine frame is scarce. Due to complexity in loading conditions, designing of the frames and optimizing the frame structure is difficult. Attempt is made here to analyze the behavior of hypothetical machine frame using static structural FEA in Ansys Workbench. Applying the shape optimization module, different ribbed constructions of frame structure are considered. Stress and deflection pattern within the different frame structure are studied and optimum frame structure is suggested.

2. Bending Machine Setup

The main working components of the machine are its forged steel wear resistant rollers, which are supported on both sides in the frame. The hypothetical frame structure of bending machine is as shown in Fig. 3. Each end of the rollers is held in the bearing block supported in T-slot provided in frames, which can be adjusted independently to set the roller positions for required product dimensions with the help of nut and lead screw arrangement. Top roller can be moved in the vertical plane in downward direction at the prescribed distance and inclination, keeping the plate on two bottom rollers, for the application of deforming load on the plate. Once the rollers are set in position corresponding to required radius, the drive is given to only the bottom rollers (keeping top roller free to rotate due to friction with plate movement) or all the three rollers depending on the plate thickness to be bent. Drive from motor to roller is given through gearbox and universal joint so that rollers can be rotated at specified speed even with relative roller inclinations. The height of the T slot from the base of frame is maintained based on favorable working conditions. Table 1 shows the specifications of three-roller bending machine under consideration. The T slot in the frame can accommodate maximum angular adjustment for the bottom and top roller, up to 5^0 and 15^0 respectively.

Table 1: Specifications of Three-Roller Bending Machine

Particulars	Qty.	Specifications
Top roller	1	φ 200 mm
Bottom rollers	2	φ160 mm
Motor	2	AC 3φ induction motor, 5 HP
Bottom roller speed	-	6 RPM
Center distance between bottom rollers	-	350-500 mm

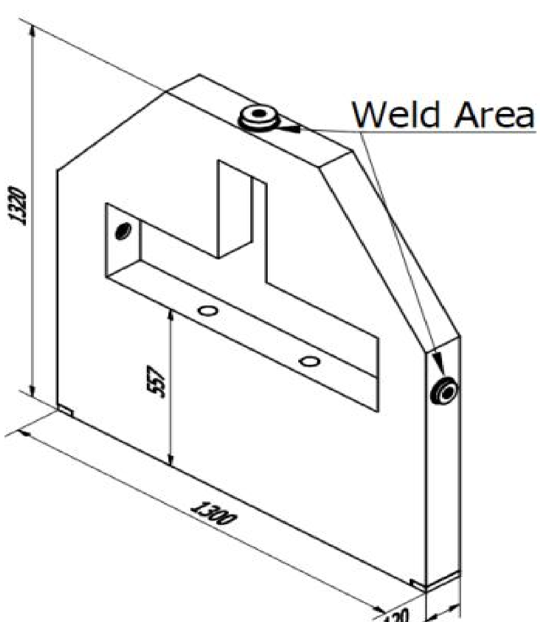


Fig. 3 Frame Structure for Conical Bending Machine

3. Bending Load

During static bending with relative inclination of all the three rollers for bending of varying cross section shells, frame experiences three components of force namely horizontal, vertical and axial acting at roller supports as shown in Fig 4. In case of bending of constant cross section shells, only two components of forces namely horizontal and vertical are in operation. For constant plate thickness, setting up of minimum center distance between bottom rollers (a) and maximum top roller position (U) results into minimum shell diameter and maximum loading on machine frame. Frame structure under consideration was deigned to bend blank of structural steel FE 410 WA [4] with maximum thickness and width of 25 mm and 300 mm respectively to minimum diameter of 360 mm. Maximum top roller load for pre-bending is the main input parameter for designing of the three-roller bending machine, which is calculated, based on the analytical model suggested by Chudasama and Raval [3]. Material property parameters for load prediction are considered as reported in Table 2. Maximum top roller load for pre-bending is calculated as 16 ton (i.e. P), which is considered as equally distributed at its two supports (i.e. (P/2)). Based on maximum top roller load and shell geometry, maximum values of horizontal and vertical load components at the bottom roller supports are considered. At ±5 degree inclination (i.e. maximum inclination of the bottom roller that can be set on the machine) and resultant load of 16 Ton on the top roller, axial components of load on the top and bottom rollers are taken. Table 3 shows the maximum load components considered for the presented analysis.

Table 2: Material Property Parameters

K (MPa)	n	E (GPa)	σ_Y (MPa)	σ_{ut} (MPa)
700	0.3	200	250	460

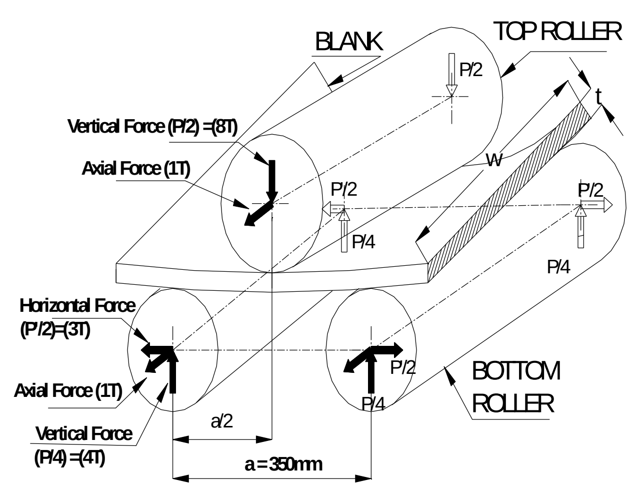


Fig. 4 Forces on Rollers during Bending

Table 3: Maximum Load Components

S. No.	Axis along which it acts	Direction	Load (T)
1	Y-axis (Vertical)	Positive	8 T (P/2)
2	Y-axis (Vertical)	Negative	4 T (P/4)
3	X-axis (Horizontal)	Positive	3 T (P'/2)
4	X-axis (Horizontal)	Negative	3 T (P'/2)
5	Z-axis (Axial)	Positive	1 T

4. Analysis

4.1 Modeling and meshing

Frame geometry as shown in Fig. 3 is modeled in inventor. Frame model is meshed using default tetrahedral mesh with maximum side length limited up to 20 mm. Relevance center was kept as fine to have denser mesh on the surface of application of load. Structural steel material FE 410 WA [4] is considered as a frame material.

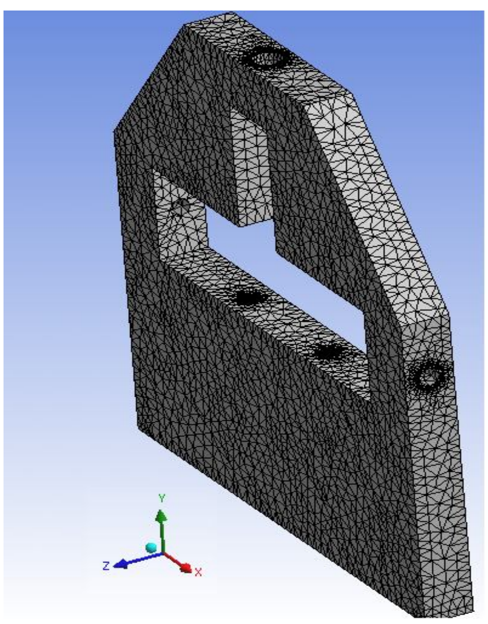


Fig. 5 Meshing of the Frame

4.2 Application of load

The values and the direction of applied forces are given in Table 3. The forces are maximum during the static bending and later on it reduces during continues bending operation. If frame resist these forces without significant deformation and stresses during static bending then it can easily resist the forces during continues deformation.

Static Structural Analysis is used with fixed boundary conditions at location "A" on base area of the frame as shown in Fig. 6. The locations of applied forces are also shown in shown in Fig. 6-7. All the three rollers are supported in bearing block, which are connected to the frame with nut and lead screw arrangement. The nuts are welded to the frame as shown in Fig. 3. Bending load is transferred to the frame through rollers, bearing block, lead screw and nut to the weld area on the frame. The weld areas are highlighted in figure Fig. 6 (locations B, C and F) where the loads are applied in direction perpendicular to the plane. Loads at locations D and E are due to the position of the bottom rollers based on minimum center distance.

The axial forces will act on the inner wall of the nut and is transferred to the frame through its surface in contact with the outer diametric surface of the nut. As the axial load is transferred perpendicular to the axis of the hole through peripheral surface of the nut, the axial force is defined as bearing load on the inner hole of the frame. The bearing load and the area of its application are highlighted in Fig. 7-8. During static bending, if the axial forces on the bottom roller are in positive z-direction than, the axial force on the top roller may be in a positive or negative z direction depending on relative inclination of rollers. The case with axial force of top roller in a negative direction is as shown in Fig. 8. But, the worst case may occur when axial forces on all rollers are in one direction. Hence, the analysis is done based on this worst case as shown in Fig. 7.

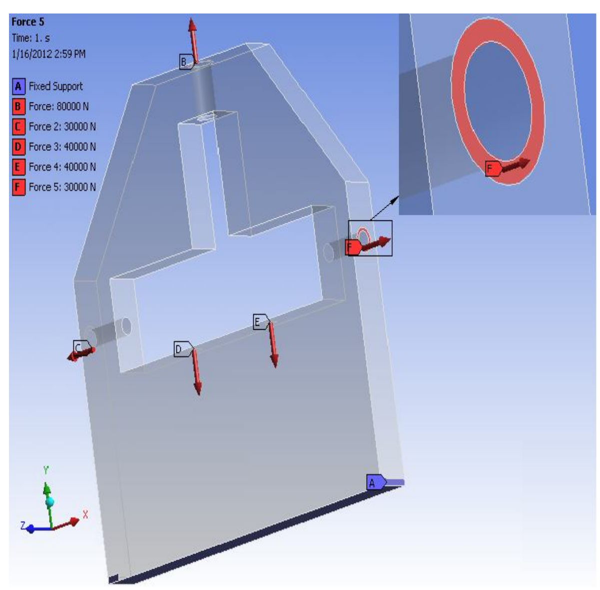


Fig. 6 Applied Fixed Support and Forces in X & Y Directions

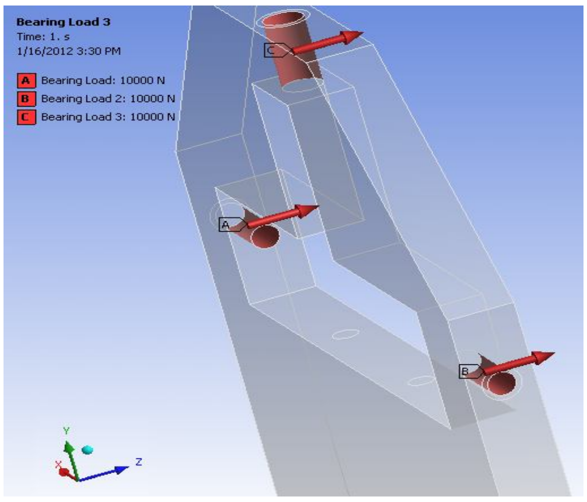


Fig. 7 Applied Axial Bearing Load in Positive Z Directions for all the Tree Rollers

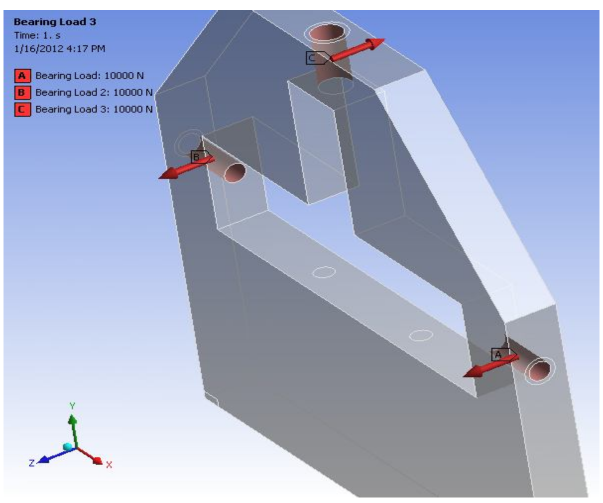


Fig. 8 Applied Axial Bearing Load in Positive z Direction for Bottom Rollers and in Opposite Direction for the Top Roller

4.3 Analysis

FE Simulations are performed to obtain the stress under above-mentioned loading condition at each point on the frame in terms of equivalent Von-Mises stress. The deflection of the frame is also obtained in terms of total and directional deflection, under the above mentioned loading conditions using Static Structural Analysis in Ansys. FEA of the frame structure is performed for different frame geometries as per case 1-5 shown in Fig. 9. Case 1 refers to the analysis of solid geometry, Case 2 refers to shell geometry and Case 3-5 refers to shell geometry with ribbed construction.

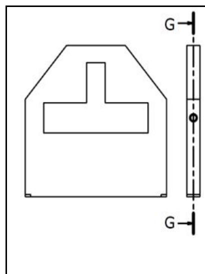
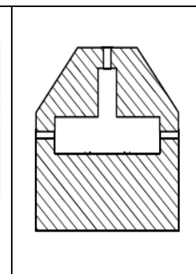
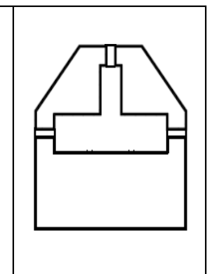
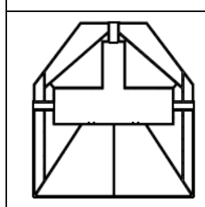
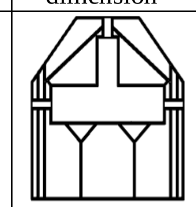
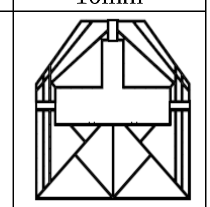
		
Frame Geometry	Case : 1	Case : 2
Section G-G is shown in each case.	Solid Geometry as per the designed dimension	Shell geometry with each wall thickness 10mm
		
Case : 3	Case : 4	Case : 5
Shell Ribbed geometry with wall an rib thickness of 10mm	Shell Ribbed geometry with wall an rib thickness of 10mm	Shell Ribbed geometry with wall an rib thickness of 10mm

Fig. 9 Details of Frame Structures Analyzed in ANSYS Workbench

5. Results

Appendix A shows the results for the equivalent Von-Mises stress, its location and the maximum deformation for alternate frame structures under consideration as per Fig.9. For solid frame geometry as per case 1 in Fig. 9, maximum stress in the frame is generated at the base and it exceeds the yield point. Maximum deformation is observed to be 0.594 mm. Moreover, higher material and fabrication cost is involved in manufacturing of frame geometry as per case 1. As other alternative for reduction in material cost, shell geometry with wall thickness of 10 mm as per case 2 shown in Fig. 9 is considered. For this case the maximum stress generated in the frame material is at the base and observed to be much higher than yield stress, which may be responsible for the failure. Structure is having maximum deformation of 2.537 mm. To minimize the deflection of frame to an acceptable limit and to lower down the stress value below yield strength of the frame material, alternate ribbed constructions are analyzed as per case 3-5 shown in Fig. 9.

For ribbed construction as per Case 3 shown in Fig. 9, deflection is observed to be 1.41 mm, but the maximum stress is observed to be 364 MPa, which is much higher than yield strength of frame material. For frame structure as per case 4, maximum stress and deflection is observed to be 196.7 MPa and 1.287 mm respectively for frame structure shown in case 5 additional ribs provided in comparison to case 4 did not show any significant difference in stress and deflection results. Based on the results of stress and deflection reported in Table 4, frame structure as per case 4 could be preferred for the fabrication. Case 4 was also checked under the reverse loading at the top nut (refer Fig. 8), as may be the case of bending of shell with varying cross section and found to be safe. For this case maximum stress is below yield point and maximum deflection is 0.45 mm.

Table 4: Results of Analysis for Different Cases

Case	Maximum deflection of frame in different direction in mm			Total Deflection in mm	Max. stress (MPa)
	X	Y	Z		
1	0.0247	0.082	0.589	0.595	439.79
2	0.1136	0.284	2.535	2.536	2734.6
3	0.0793	0.234	1.409	1.420	305.36
4	0.0749	0.213	1.275	1.288	196.97
5	0.0756	0.214	1.259	1.268	191.24
4ш	0.0766	0.166	0.427	0.453	197.02

Effect of the frame displacement for the case 4 on the loaded radius of the plate was determined though the geometrical relation between centres distance of the bottom rollers, top roller displacement and loaded radius of the plate, derived by Gandhi [1]. It is assumed that the deflection of the frame from its original position under loading condition in x and y directions, is equal to roller displacement of bottom and top rollers from its original position. It is found that the variation in the loaded radius is less than 0.8 and 0.1 percent, with the variation in x and y directions respectively in case 4. Maximum variation in z-direction or the deflection of the frame in case 4 does not have the significant effect on loaded radius based on assembly consideration.

6. Conclusion

From this structural study it can be concluded that

i. Ribbed structure has a greater economical and functional advantage than solid structures. Providing ribs strengthen the shell structure up to an extent. Further increase in ribbed structure as in case 5, does not affect much in resistance of structure against loading. Hence, in the above study, Case 4 could be preferred for fabrication.

ii. The cases are checked for the worst situation. Case 4 gave fair enough result for the situation of reverse loading on the top roller loading (Refer Fig. 6 and Appendix A).

iii. Maximum Deflection could be minimized to 1.25 mm. This may be due to greater overhanging length with large discontinues area in middle portion of frame structure and smaller support length at both the side. Yet, the structure can resist maximum loading during static bending as stress within the structure could be maintained below yield point.

iv. Study reveals that deflection of the frame for the case 4 does not have significant effect on product of the machine, that is, loaded radius of plate.

v. With the knowledge of the material and model, Different design of the structure can be easily checked using Finite element analysis (FEA), by changing model design, without much experimentation. Hence, the best-suited design can be selected for fabrication. In this way the design can be optimized using FEA.

References

1. *Gandhi A H and Raval H K (2008), "Analytical and Empirical Modeling of Top Roller Position for Three-Roller Cylindrical Bending of Plates and its Experimental Verification", Journal of Material Process Technology, Vol. 197, 268–278.*

2. *Ramamurti V, Ravi, Shankar, Rao V and Sriram N S (1992), "Design Aspect and Parametric Study of 3-Roll Heavy Duty Plate Bending Machines", Journal of Material Process Technology, Vol. 32, 585-598.*

3. *Chudasama M K and Raval H K (2011), "An Approximate Bending force Prediction for Three-Roller Conical Bending Process", International Journal of Materials Forming.*

4. *Hot Rolled Low, Medium and High Tensile Structural Steel (2010), 6th Revision, IS 2062.*

5. *Hoseyin, filiz, Abdullah, Akpolat and Ibrahim Gisizelbey H (1996), "Stiffness of Bolted Members", International Journal of Machine Tools Manufacturing, Vol. 37, 309-318.*

vi. Acknowledgement

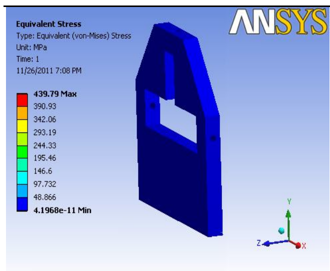
Preseent work is the part of research project funded by Department of Science and Technology, Ministry of Science and Technology, Government of India under SERC-Engineering Science Scheme, sanction letter no. SR/S3/MERC-0026/2007. Authors concede and highly appreciate the financial support provided by Department of Science and Technology, Ministry of Science and Technology, Government of India.

Nomenclature

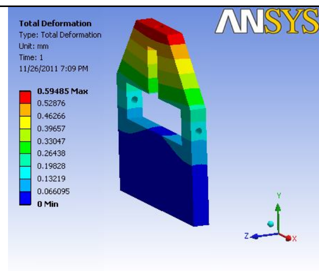
Symbol	Meaning	Unit
R	Radius of Neutral axis of bent Plate	mm
U	Displacement os Top Roller for Bending	mm
a	Center distance of the bottom roller	mm

Appendix

The following table shows the stress distribution and deflection in each case considered in analysis

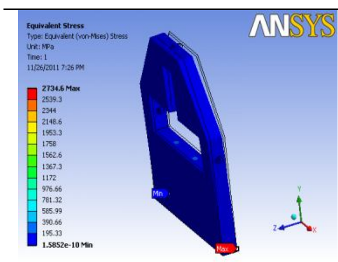


Max. Stress: 439.79Mpa
Location: At base

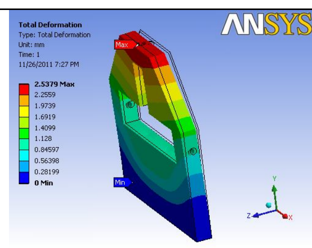


Max. deflection: 0.59485mm
Location: At top nut

Result of Case : 1

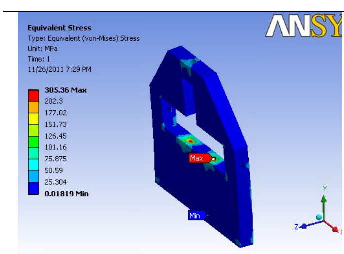


Max. Stress: 2734.6Mpa
Location: At base

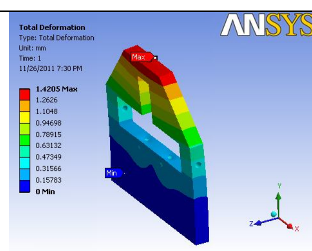


Max. deflection: 2.5379mm
Location: At top nut

Result of Case : 2

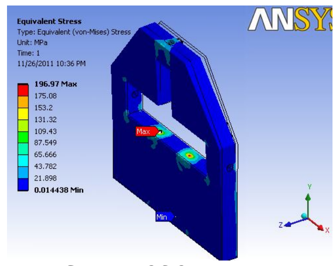


Max. Stress: 305.36Mpa
Location: At mid-slide

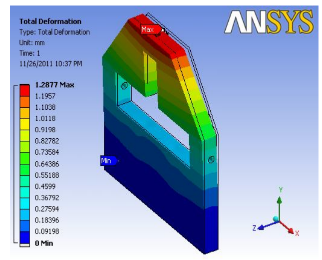


Max. deflection: 1.4205mm
Location: At top nut

Result of Case : 3

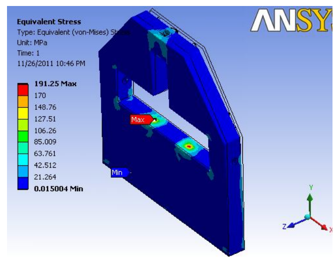


Max. Stress: 196.97Mpa
Location: At mid-slide

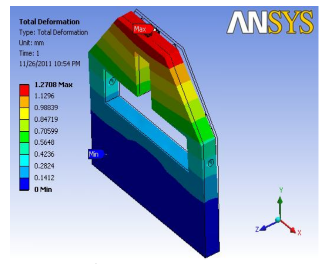


Max. deflection: 1.2877mm
Location: At top nut

Result of Case : 4

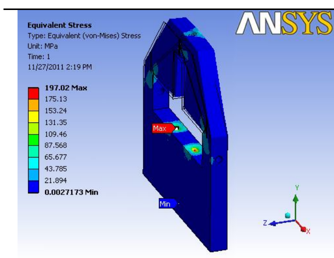


Max. Stress: 191.25Mpa
Location: At mid-slide

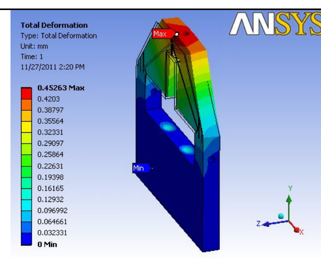


Max. deflection: 1.2708mm
Location: At top nut

Result of Case : 5



Max. Stress: 197.02Mpa
Location: At mid-slide



Max. deflection: 0.4526mm
Location: At top nut

Result of Case : 4 (Reversed Axial Load as per Fig.6)

