Productivity Improvement of Autoclave Equipment Used For Vulcanization of Rubber Coated Rolls

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Productivity Improvement of Autoclave Equipment Used For Vulcanization of Rubber Coated Rolls

Major Project Report

Submitted in partial fulfillment of the requirements

For the Degree of

Master of Technology in Mechanical Engineering

(Computer Aided Design & Computer Aided Manufacturing)

By

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Declaration

This is to certify that

1. The thesis comprises of my original work towards the degree of Master of Technology in Mechanical Engineering (Computer Aided Design & Computer Aided Manufacturing) at Nirma University and has not been submitted elsewhere for a degree.

2. Due acknowledgment has been made in the text to all other material used.

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This is to certify that the Major Project Report entitled "**Productivity improvement of autoclave equipment used for vulcanization of rubber coated rolls**" submitted by **Mr. Ajay I. Gadariya (14MMCC04)**, towards the partial fulfillment of the requirements for the award of Degree of Master of Technology in Mechanical Engineering (Computer Aided Design & Computer Aided Manufacturing) of Institute of Technology, Nirma University, Ahmedabad is the record of work carried out by him under our supervision and guidance. In our opinion, the submitted work has reached a level required for being accepted for examination. The result embodied in this major project, to the best of our knowledge, has not been submitted to any other University or Institution for award of any degree.

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-Ajay I. Gadariya

Abstract

Rubber coated rolls are primary requirement for industries such as Textiles, Paper mills, Automatic transfer lines for handling dangerous compounds etc. as they are in continuous operation under specific environment. For such applications vast variety of rubber coated rolls are available. One of the main requirements from these industries is the hardness of the rubber coated rolls which depends on the application and working environment. To achieve that specific hardness of the rubber compound various of processes are done on it.

Vulcanization of these rubber coated rolls is one of the main process, which imparts hardness to the soft rubber. The hardness value depends upon the rubber compound and the process parameters. In a rubber manufacturing industries productivity is the major problems due to the long processing time of rubber vulcanization. To vulcanize the rubber autoclave equipment is used which is often found to be under-utilized due to varying size of rubber rolls and lack of proper roll arrangement system on the trolley type structure.

A modular structure has been conceptually designed. Analytical design and FE analysis have been carried out for a safe design with F.O.S. of 2. A model with roll arrangement is developed and it is observed that the space utilization is almost improved by 100%. This will reduce the production cost and thereby the overall productivity of the company can be improved. A program has been developed using Visual Basic software which will provide the roll arrangement on a modular trolley structure to obtain optimum space utilization.

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Nomenclature

ρ	Density of the material (Kg/mm^3)
σ_y	Yield strength of the material (N/mm^2)
σ_u	Ultimate tensile strength of the material (N/mm^2)
τ	Shear stress in the material (N/mm^2)
E	Young's modulus of the material (N/mm^2)
M_b	Maximum bending moment (N-mm)
W	Force due to weight of the rolls (N)
L	Length of the support beam (mm)
$Z_{min.}$	Minimum section modulus (mm^3)
$\sigma_{allowable}$	Allowable stress for the material (N/mm^2)
t_w	Web thickness (mm)
d	Depth of the web (mm)
$\tau_{allowable}$	Allowable shear stress (N/mm^2)
I	Moment of inertia (mm^4)
$\Delta_{max.}$	Maximum deformation (mm)
$\Delta_{max._{allowable}}$	Allowable maximum deformation (mm)
l_e	Effective length of the column (mm)
k	Radius of gyration for column (mm)
λ	Slenderness ratio for column
F	Normal load on the wheel (N)
a	Width of the wheel (mm)
E_{wheel}	Young's modulus of the cast wheel (N/mm^2)
P'	Force per unit width of the cast wheel (N/mm)
σ_c	Contact stress between wheel and rail (N/mm^2)
$\sigma_{c_{allowable}}$	Allowable contact stress (N/mm^2)
C_R	Surface hardness co-efficient
L_c	Length of the bottom channel (mm)
e	Eccentricity (mm)
c	Distance of extreme fibre from neutral axis (mm)
W_{cr}	Critical buckling load (N)
$y_{max.}$	Maximum deformation in column (mm)

Acronyms

C_R	Surface hardness coeffcient
BHN	Brinnel hardness number
FE	Finite element

Chapter 1

Introduction.

1.1 General

Rubber coated rolls (e.g. Ebonite, Neoprene, Butyl, Hypalon, EPDM, Polyurethane, Silicon, Viton etc.) are important component of Textiles industries, Paper mill industries and other machineries as it is in continuous operation. Rubber coated rolls as shown in figure1.1 need to be vulcanized in order to obtain the desired hardness for particular application. These rolls are vulcanized in an autoclave equipment for a specified time period under certain conditions i.e. parameters like pressure and/or temperature depending upon the type of rubber used.

Figure 1.1: Rubber coated roll

The pressure and temperature given during the vulcanization are around 0.344 bar to 1.723 bar and temperature in the range of 398 K to 423 K. Autoclave machine is a closed cylindrical vessel having multiple inlets for stem supply along its length and it is rigidly mounted on saddle supports on foundation as shown in figure1.2. Rolls are arranged on a trolley type structure which is then placed inside an autoclave machine for vulcanization.

Figure 1.2: View of a loaded autoclave

1.2 Process flow of rubber manufacturing processes

Figure 1.3: Process flow of rubber roll manufacturing industry

Figure1.3 shows the process flow of rubber manufacturing procedure in which the initial product which entered in a company is a metal shell with a shafting at each end. Then scrapping operation is done on the metal shell to produce threads on it which provides better grip between rubber sheet and metal shell. The scrapped shell is then heated in an autoclave to remove oil, dirt and other residue from previous rubber coating. The clean metal shell is then undergo a solution applying process in which particular solution is applied on it based on the type of rubber to be used to provide better adhesion. Now the rubber sheet is produced by mixing natural rubber with specific ingredients such as Sulphur, accelerator, wax and other agents to obtain the rubber sheet.

The rubber sheet so produced is then calendared in a calendar machine to produce sheets of uniform thickness and width which is then applied on metal shell one after another to get the final rubber roll diameter. In order to prevent the rubber roll from expanding and/or distorting during vulcanization it is wrapped with cotton and/or nylon tape with certain tension. This wrapped rubber roll is then put inside an autoclave which will harden the rubber as per the required hardness. Then the tape is removed and it is machined on a lathe to achieve the ordered face length and diameter.

The machined rubber roll is then ground on a semi-automatic grinding machine which will provide the final diameter with good surface finish and the camber on the roll if required by the customer. After that each rubber roll has to pass the inspection test which checks for the voids, pin holes, cracks, face length and diameter as per the customer's specification. The finished rubber roll is then packed in a wooden and/or sheet metal box and dispatched to the customer.

1.3 About Lathia Industrial Supplies Co. Pvt. Ltd.

Rubber manufacturing industry is a constantly challenging business as there is a dynamic demand of rubber coated rolls, rubber sleeves, rubber blankets etc. with vast variety of rubber compounds suitable for various applications as well as different environments. It requires lot of research in the field of rubber technology to cope up with such demands and also requires various equipment and machineries to process such rubber compounds to achieve the required rubber mixture. Testing of rubber is itself a huge task as it demands specialized personnel and dedicated instruments to understand the produced rubber's characteristic before it can be put into application. So by considering above facts one case study about the Lathia Industrial Supply Co. Pvt. Ltd has been presented over here as follows.

Lathia Industrial Supplies Co. Pvt. Ltd. has been leader in the business of rubber coated rolls, rubber blankets, leather covering, rubber sleeves and rubber lining etc. since 1953. It is nowadays a big, multi-unit, multi-dimensional company providing best rubber products for industries like Textiles, Paper mills and other rubber driven companies since long time. It has the great research facility available through technical collaboration with Stowe wood-ward, U.S.A. Lathia is a brand name in roll covering and its rubber blankets due to a very long experience in the rubber business. It is a continuously growing company already having plants all over India including Gujarat, Delhi, Vapi and Bombay etc. driven by the directors of the company who are themselves rubber technologist knowing the best and all about the rubber and its applications in wide range of today's dynamic demand.

1.4 Trolley for putting rolls inside autoclave

Figure 1.4: Existing trolley for putting rubber rolls inside autoclave

The autoclave is a cylindrical vessel used for vulcanization of rubber rolls. To slide the trolley inside an autoclave there is a provision of rails inside it on which the loaded trolley slides by pulling it with the help of wire rope and overhead crane. Trolley is fabricated such that it can withstand the total load of the rolls at elevated temperatures as shown in figure1.4. It is properly stiffened at bottom for above purpose.

1.5 Present method of roll arrangement on trolley

Figure 1.5: View of a present method of roll loading on a trolley

Currently rolls are being arranged on a trolley randomly without any prior planning as shown in figure1.5. Such arrangement leads to under-utilization of space inside an autoclave which ultimately lower down the productivity of the equipment. To avoid such situation the existing trolley can be modified to facilitate more number of rolls in single set up so that more number of rolls can be vulcanized at a time which leads to increased productivity of the equipment and thereby increased overall productivity of the company.

1.6 Motivation for the project

Following problems were faced by the company:-

1. Long lead time to process a single rubber roll :- As the rubber roll have to pass through different processes few of which take lot of time such as vulcanization process. If more number of roll can be vulcanized at a time then it is possible to reduce the lead time for rubber roll.

2. Stacking of rubber rolls near the autoclave which are to be vulcanized next :- The rubber rolls which are ready to be vulcanized are stored in front of the autoclave due to which the floor space in the vicinity of the autoclave gets congested which creates difficulty in loading and unloading of rubber rolls on the trolley.

3. Lack of floor space available for putting vulcanized rolls as it is occupied by the wrapped rolls which are to be vulcanized later on :- The rubber rolls which are already vulcanized are also stored in the vicinity of the autoclave as there is no separate floor space available for them. As the floor space is already occupied by the rubber rolls which are to be vulcanized next it creates difficulty in handling of rubber rolls.

4. Non optimal utilization of autoclave equipment :- The space inside autoclave is under-utilized as there is no provision for putting extra rubber rolls on the trolley structure and hence the autoclave is not fully utilized which lower downs the productivity of the autoclave and also the time and energy consumptions are more.

1.7 Problem definition

To circumvent the issues discussed above a dissertation work has been taken for

"Productivity improvement of autoclave equipment used for vulcanization of rubber rolls."

1.8 Objectives

Following objectives have been identified for the dissertation work,

- To understand and analyze the rubber manufacturing processes (especially rubber vulcanization) and the parameters affecting those processes.

- To design a modular structure for supporting rubber coated rolls during vulcanization.

- To improve the productivity of the autoclave equipment by implementing the proposed trolley structure.

- To evaluate the effect of autoclave's improved productivity on the overall productivity of the company.

1.9 Organization of Thesis

The chapters are arranged as follows:

Chapter 1 The first chapter has brief introduction about rubber manufacturing industries and their practises.Why this title is important is justified in this chapter and objectives of project are defines in this chapter.

Chapter 2 This chapter gives idea about literature related to different approaches to design of steel structures and their FE analysis.

Chapter 3 This chapter gives idea about understanding the rubber roll production data and deciding the methodology to be adopted. Based on these things the preparation of conceptual design by selecting the suitable material and to understand the existing practises of the rubber roll manufacturing.

Chapter 4 In this chapter detail analytical design of each and every member of modular trolley structure has been done. The geometric modeling of all the members of trolley strucuture is prepared using Creo3.0 along with it FE analysis in ANSYS14.5. A software has been developed using Visual Basic to plan the roll arrangement on modular trolley to obtain optimum space utilization. Energy calculation is done for the excess steam and Bill of Material is prepared.

Chapter 5 This chapter discusses about conclusion and future scope of the dissertation work presented.

Chapter 2

Literature Review

Proper literature survey is needed for any research or practical work as it helps to understand the various aspects of the project very well and suggests some idea to improve the design of the project. For that reason some of the papers related to design of steel structure are studied to get the help from the good researchers through their valuable work. Some of them are mentioned below,

In this paper Ganping Shu et al.[1] developed the design procedure for stainless steel columns experiencing flexural buckling. From the base strength curve and slenderness conversion formula, the strength curve for any type of S.S. can be prepared by applying validated finite element model. The slenderness conversion formula for any two type of S.S. having same cross section and boundary conditions but different material properties can be derived using material model having strain hardening exponent ‘n’ used and material parameter is selected as base material. Based on the base strength curve and slenderness conversion formula strength curve for any type of S.S. can be generated which avoids the tedious theoretical iterations.

Zhao et al.[2] studied the effects of combined loading on stainless steel sections like square hollow section (SHS) and Rectangular hollow section (RHS). To investigate the behavior of these sections under such loading various test were performed like stub column test , four point bending test, uniaxial bending plus compression test, biaxial bending plus compression test etc. For knowing the bending moment to axial load ratio wide variety of eccentric loading conditions were applied during the testing and same were validated with FE software.

In this paper Kucukler et al.[3] analyzed the steel beams and columns experiencing flexural-torsional buckling by applying stiffness reduction method. Linear buckling analysis was performed by applying developed SRM function and by lowering down the values of modulus of elasticity of the material and its shear moduli. In plane and out of plane analyses of member were carried out as the developed function fully considers the severe effects of imperfections and spread of the material's plasticity thus it avoids iterative theoretical calculations resulting into practical design approach.

In this paper M. Macdonald et al.[4] found out the new empirical design formula that can be used to design a cold formed lipped channel beams when its web is subjected to crippling. To replicate the web crippling loading conditions appropriate series of tests were performed to predict the ultimate strength of the specimen. Test specimens were prepared as per the different loading conditions including the corner radius and web lengths. Finite element model was prepared to compare the test results and were validated with the theoretical model and parametric study was done to arrive at the design procedure for the test specimen.

The objective of this research as stated by the Landesmann et al.[5] to develop the new model for predicting the strength of cold formed steel lipped channel beams at elevated temperatures. The Direct strength method is applied to carry out such design process in which ultimate strength of beam is calculated using direct loading conditions and major axis bending moment forces. As there is no direct rule for designing such beams having such cross sections the prepared numerical model is applied and same is validated using FEA package. Material properties mainly young's modulus and yield strength of the material were altered as per the effect of the elevated temperatures.

Liu et al.[6] tested the stiffened closed-section thin-walled aluminum alloy columns under compression and. it was observed that local buckling happens in every member after certain period of time. No. of axial compression test were performed on the specimen having four stiffened closed section. FEM model was made to simulate the local buckling behavior of the test specimen under compressive axial load and was validated with the test results. Comparison was made between the current codes for design like American code for aluminum design, European code, and Chinese code and DSM to find out the ultimate material strength and it showed that current codes overestimate the design strength for such sections

In this paper Gunalan et al.[7] presented the detailed study of the cold formed SS columns experiencing the flexural-torsional buckling having fixed and hinged end conditions. During this study it was found that the present codes for such design which works on the non-dimensional strength curve for both type of boundary conditions does not account for warping fixity and thereby underestimate the member strength. Hence this study suggested the improved rules for designing such members using appropriate FE model and by validated with test results using different steel grades with varying thickness.

Recent development in the design of cold formed steel sections and structures as studied by Macdonald et al.[8] which are widely popular in today's world due to their low weight and high strength. However there are lots of variety in the types of sections available and also their design is complicated due to effects of slenderness ratio, wall thickness, loading patterns and unpredictable buckling behaviors. These effects are studies in depth here to realize the potential benefits of these sections by taking into account their characteristics and strengths.

This paper involves test methods for austenitic steels columns and beams. As investigated by Zheng et al.[9] cold formed hollow square members and welded H members were tested to obtain the material properties. Test results involving material's strength, modes of failure and response of load-deformations were analyzed and compared with the existing codes. From that it was concluded that these codes underestimate the design strength of the material and for that improved design rules were proposed which improves accuracy of the design and reduces scatter.

The effects of distortional deformation as studied by the Saoula et al.[10] for lateral buckling of box beam elements undergoing combination of bending and axial loading. To do so an analytical model was prepared to check the stability of such beam elements and were discretized by the tangential stiffness matrix and were simulated in Abacus software which shows that the present codes overestimate the design strength for such members.

Chan et al.[11] investigated the hollow S.S. members, the material properties and residual stresses distribution for cold formed S.S. hollow members. No. of tensile coupons were tested to find out the modulus of elasticity, ultimate tensile strength and elongation at the fracture point of the material. These results were also compared with the different standards. Using wire cutting method residual stress was determined both longitudinally and transversely for the hollow S.S members.

Chapter 3

Methodology

For achieving above mentioned objectives proper methodology is to be adopted which will step by step lead to the solution of a problem by considering each aspect of the project. Now to understand the current practices and processes of rubber manufacturing following needs to be understood first.

3.1 Rubber roll production data

Rubber rolls which are coming for vulcanization are of varying size in terms of their face length and diameter. So in order to determine the most common roll size, past six months roll details were analyzed and presented in form of pie charts as shown in fig.3.1 for face length and roll diameter[12].

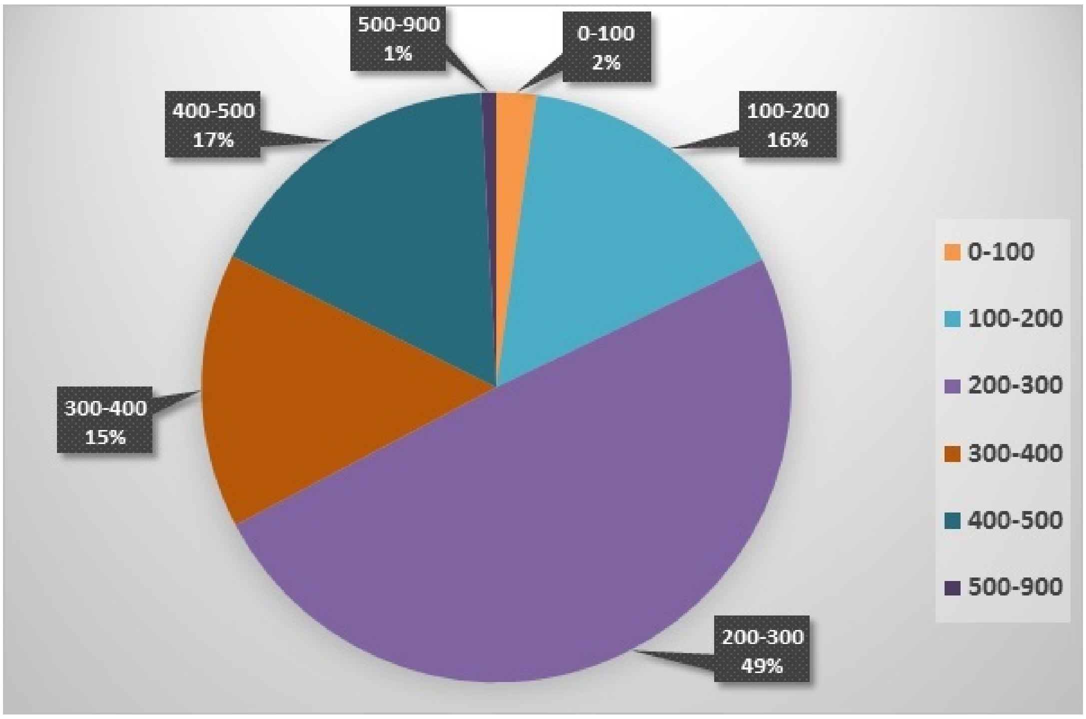

Figure 3.1: Pie chart for rubber roll diameter

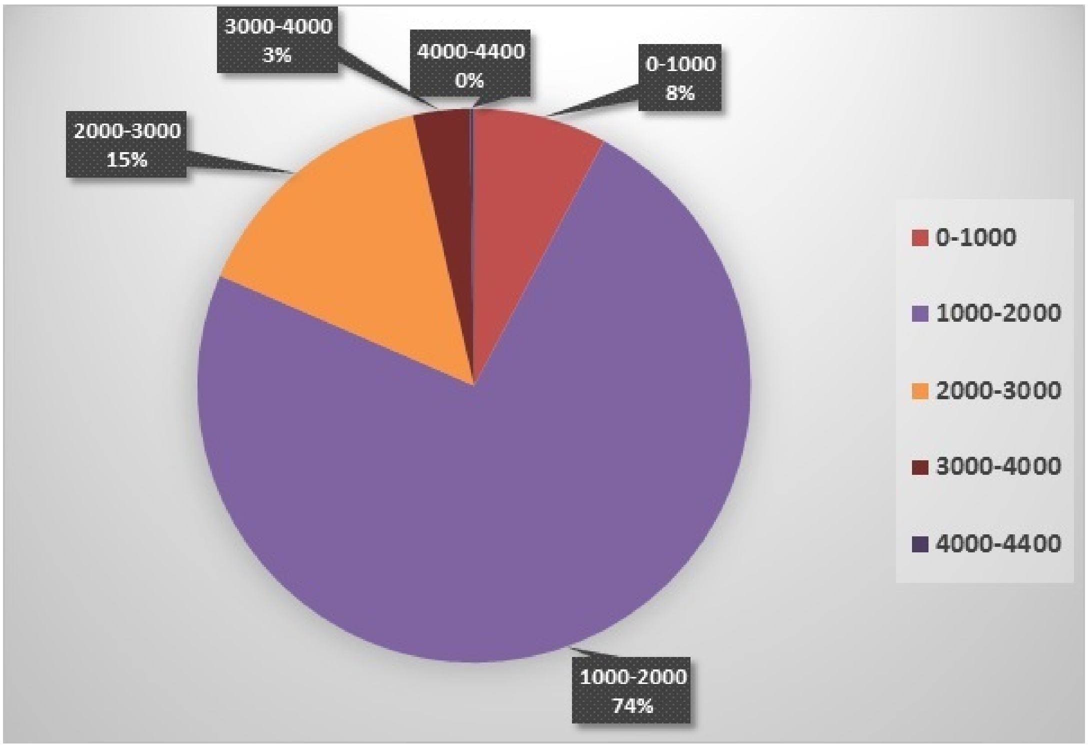

Figure 3.2: Pie chart for rubber roll face length

From the figure3.1 it can be seen that 82% of the rolls are below 400 mm and therefor the support structure is designed as per this diameter. From the figure3.2 it is clear that 97% of the rolls are below 3000 mm face length so the modular structure should be designed as per the weight of the roll having above diameter and face length respectively.

3.2 Specifications of the rubber roll to be vulcanized

From the production planning and material management data of the company the specifications of the rubber rolls that are coming for processing are as follows,

- Length of the rubber roll : 150 mm to 4400 mm.
- Diameter of the rubber roll : 75 mm to 900 mm.
- Weight of the rubber roll : maximum 600 Kg
- Density of the rubber used : 990-1500 kg/m^3

3.2.1 Specification of autoclave equipment

It has been observed that the rubber rolls having above specifications are vulcanized in the autoclave in which hardly half of the space inside the autoclave is utilized for vulcanization and hence the productivity of the equipment is low and hence it is thought to design a modular structure so that maximum space can be utilized. The specifications of the autoclave are as follows,

- Inside diameter = 1600 mm
- Length = 7600 mm
- Pressure = 0.344 bar to 1.723 bar
- Temperature = 150° C to 175° C
- Time taken for vulcanisation = 20 hours to 24 hours based on compounds

The methodology adopted for the maximum space utilization inside the autoclave is mentioned below,

- To understand and analyze the rubber manufacturing processes.
- To observe the current design of the supporting structure for rubber rolls and its capacity.
- To analyze the current space utilization of the autoclave equipment and maximum possibility of space utilization.
- To prepare the conceptual design of modular structure.
- To prepare the analytical design based on the loading conditions.
- Analysis of the design prepared using FEA software.
- Validation of software results with theoretical design.
- To prepare the Costing of the structure designed.
- Fabrication of the structure.
- Implementation and compare it with previous productivity.

3.3 Current utilization of space inside autoclave

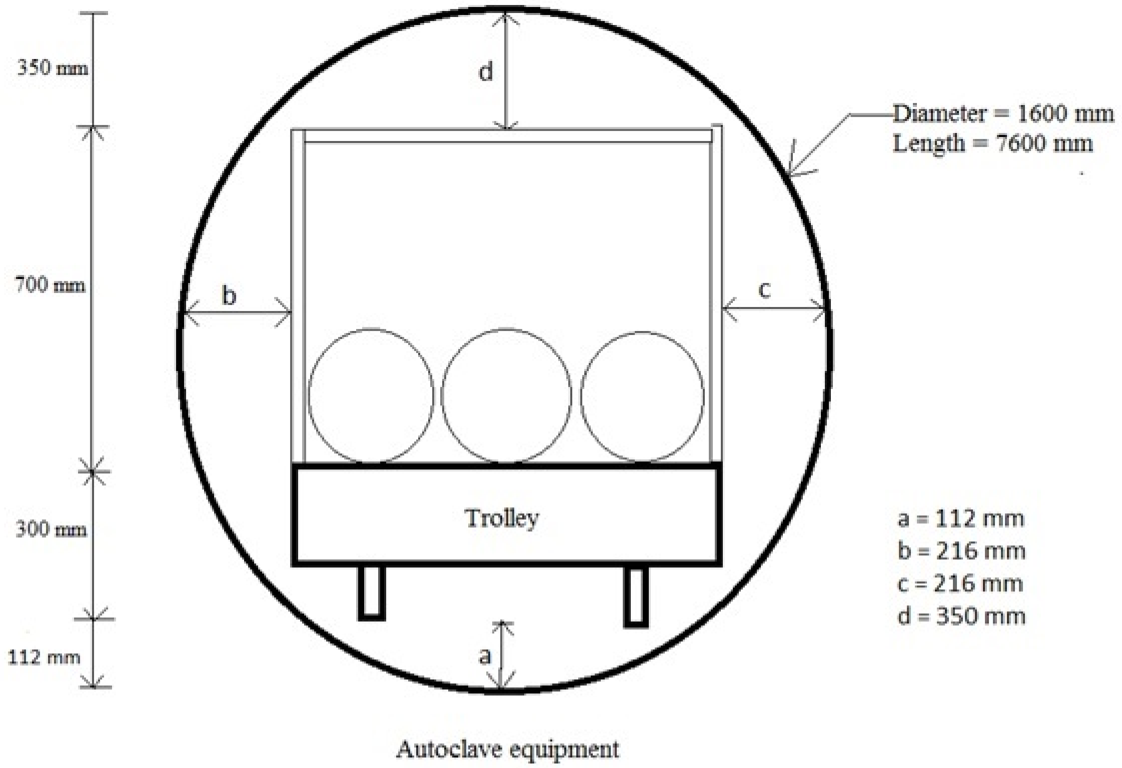

Figure 3.3: Schematic representation of space utilization inside autoclave

The autoclave is closed cylindrical vessel inside which the trolley is placed for vulcanization of rubber rolls. The roll arrangement is shown in schematically in fig.3.3. The space occupied by trolley and the spaces a, b, c and d (fig.3.3) cannot be utilized. Hence the effective volume of autoclave space can be worked out as follows,

Effective volume of autoclave (mm^3) = (Volume of autoclave) – (Volume of trolley) – (space that can't be utilized i.e. a, b, c and d)

= 15272960000 – 2598846720 – 3632452406

= 9041660874 mm^3

Percentage utilization of effective volume can be given as follows,

% utilization = [Volume occupied by rubber rolls / Effective volume of autoclave] x 100

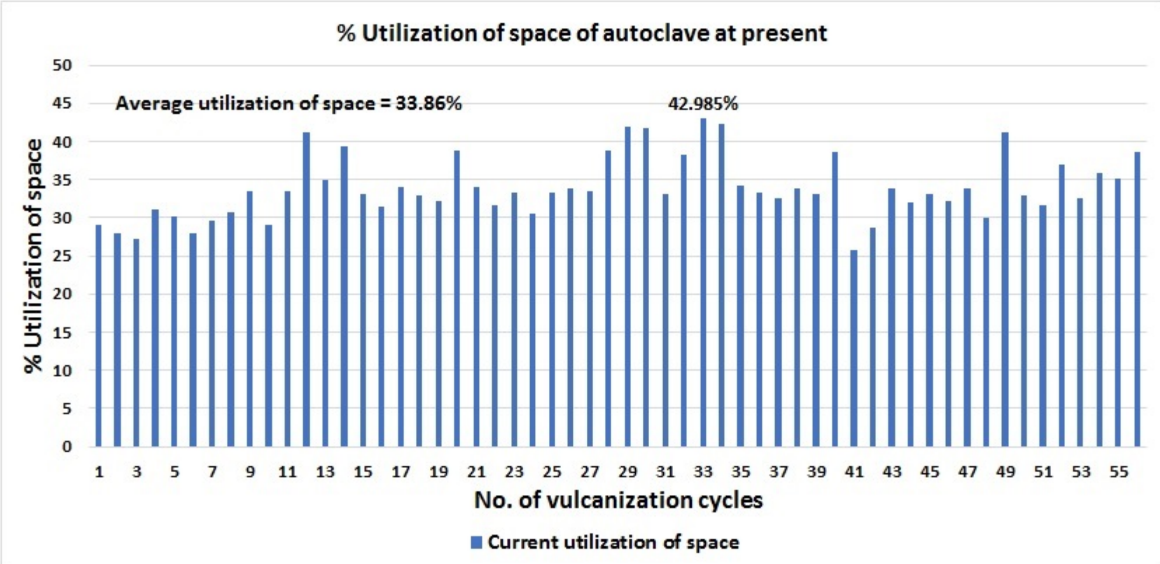

Figure 3.4: Bar chart for current space utilization inside autoclave

Figure3.3 shows the schematic representation of the space utilized in autoclave with modular structure. To check the current space utilization inside autoclave past four months roll production data[12] were analyzed and bar chart is prepared which is shown in figure3.4. From the chart it can be seen that hardly 33.86% of the effective volume of autoclave is being utilized on and average currently which can be improved by providing modular beam column arrangement on the existing trolley structure as shown in figure4.32. Such modular arrangement will lead to the improved productivity of autoclave with lesser time and energy consumption.

3.4 Conceptual design of the trolley structure

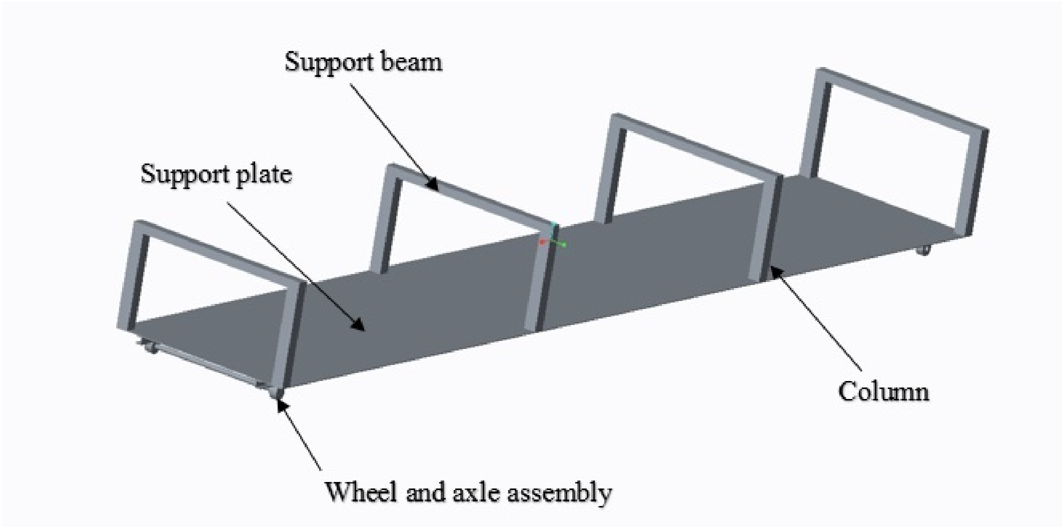

Figure 3.5: Conceptual design of trolley structure with modular arrangement

Based on the rubber roll production data and the scope of the space utilization in the autoclave equipment the conceptual modular structure was prepared in Creo3.0 as shown in fig.3.5. It helps to visualize the proposed trolley structure and provide basis for the force calculations and loading conditions to complete the analytical design.

3.5 Material Properties

The trolley structure with modular beam-column arrangement has been designed using structural steel(Fe410) and its properties are listed in table3.1 below,

Table 3.1: Material (Fe 410) properties[13]

SR. No	Properties	Symbol	Value at room temperature	Value at 150° C
1.	Density	ρ	7850 Kg/m^3	
2.	Tensile yield strength	σ_y	7850 Kg/m^3	235 N/mm^2
3.	Tensile ultimate strength	σ_u	410 N/mm^2	401 N/mm^2
4.	Poisson's ratio	ν	0.3	
5.	Young's Modulus	E	210000 N/mm^2	196000 N/mm^2

3.6 Assumptions

For designing the modular structure following assumptions were made,

- All the joints are rigid.

- All the supports provided are fixed.
- Only static loading is considered.
- There is no external force and/or moment but only self-weight.
- Material is homogenous and isotropic.
- Factor of safety 2 is considered throughout the design.

Chapter 4

Analytical design of proposed trolley structure and its FE prediction

The main structural members of the modular structure are tabulated in table 4.1. These structural elements are designed as per the design manual for designing steel structures[14] according to new IS: 800 for IS: 800-2007 for hot rolled structural members. Design of column is done as suggested by Secant[17]. The detail design calculations are mentioned in subsequent sections.

Table 4.1: Design of main components for trolley structure

SR No.	Component	Criterion	Specification
1.	Support beam	Fixed at both ends and loaded at center by 40000 N	ISMB 150 is selected.
2.	Adjustable column	Fixed at top and secured rigidly at bottom and loaded on top by 20000 N	96X48X4, 80X40X4 rectangular hollow sections are selected.
3.	Plate	Loaded unevenly with maximum load of 192276 N	Plate of 8 mm thickness is selected.
4.	Axle	Loaded at center by 32000 N and fixed at both ends.	Circular rod having 60 mm diameter is selected.
5.	Wheel	Loaded at center by 16000 N	Flanged wheel having 254 mm diameter is selected.
6.	Bottom channel	Loaded unevenly with maximum load of 96138 N at center.	ISMC150 is selected.

4.1 Support beam design

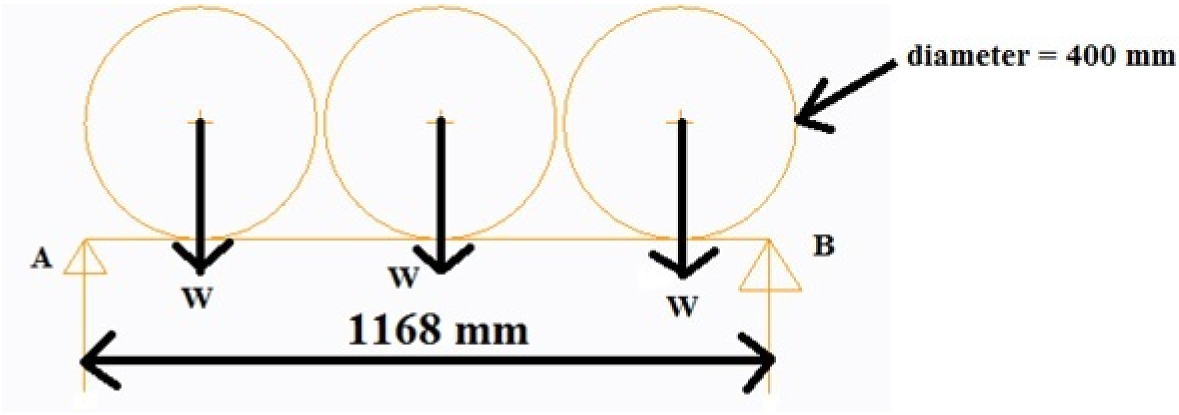

Figure 4.1: Force analysis of support beam

The support beam having a length of 1168 mm is supported on a column at each end. Now from the roll production data for roll having most common size is 400 mm but only three rolls having such diameter can be accommodated on the beam as shown in figure4.1. The maximum weight of the roll is assumed to be 600 kg for the roll having diameter of 400 mm and length of 3000 mm and total six number of rolls can be supported on a middle support beam as shown in figure4.32. Hence the beam has to withstand the load of 40000 N acting at its center which is the worst condition.

The selected structural beam is ISMB150 for supporting upper rubber rolls to carry a load of 40000 N at its center which is the worst condition for a beam having its both ends fixed. For such beam the stress and deformation can be calculated as below,

Maximum bending moment[15], M_b = (W x L)/8

= (20000 x 1200) / 8

= 5840000 N-mm

From this moment finding out the required section modulus for the beam as,

$Z_{min.} = M_b / \sigma_b$

= 5840000 / (0.66 x 235)

= 37563.12 mm^3

The cross section of the support beam (ISMB150) has been selected from the standard manufacturing size catalogue. Section modulus of ISMB150 is found to be 97000 mm^3.

Bending stress in a beam[15],

$\sigma_b = M_b / Z_{min.}$

= 5840000 / 97000

= 60.206 N/mm^2

Which is less than the allowable bending stress (155 N/mm^2) of the material. The allowable bending stress can be found from following formula[13],

$[\sigma_b]_{allowable}$= 0.66 x 235 = 155 N/mm^2

Shear stress in a beam[15],

$\tau = (W/2)/(t_w x d)$

= 20000 / (4.8 x 134.8)

= 30.905 N/mm^2

The shear stress is also less than the allowable shear stress[13] for the material which is,

$\tau_{allowable}$ = 0.5 x σ_y= 0.5 x 235 = 117.5 N/mm^2

Deflection of a beam is given by [15],

$\triangle_{max.} = W \times L^3/(192 \times E \times I)$

= 40000 x 1200^3 / (192 x 201600 x 7260000)

$\triangle_{max.}$ = 0.2268 mm

Allowable deflection[16] can be given as $[\Delta_{max}]_{allowable}$ = L/360 = 1168 / 360 = 3.244 mm

Hence the maximum deflection is less than the allowable deflection which makes the support beam safe in deformation.

4.2 Finite element analysis

The theoretical design have been analyzed using FEA software to predict the actual behavior of the trolley under actual loading conditions. Modelling of main components of the trolley has been done and subsequently FE analysis has been carried out using ANSYS14.5 software. Static loads have been applied. It has been discussed in details through following sections. All the structural elements (support beam, column, axle, support plate etc.) have been modelled and meshed with quadratic hexahedron and wedge elements. The meshing has been done through automatic meshing option which enables flawless meshing without any discontinuity.

4.3 Analysis of support beam

4.3.1 Geometry and meshing

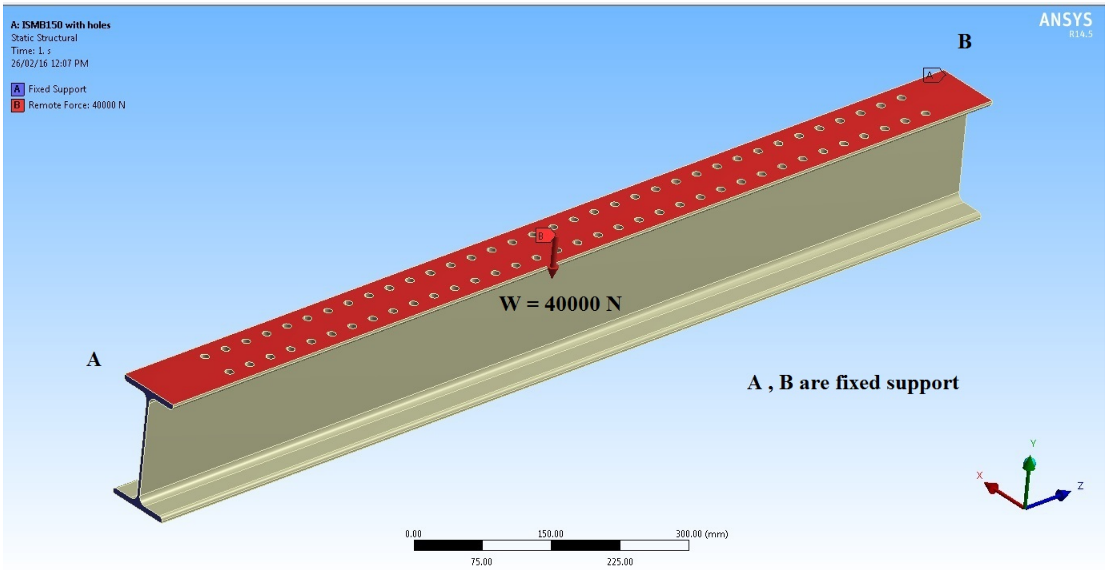

Figure 4.2: Loading and constraints for support beam

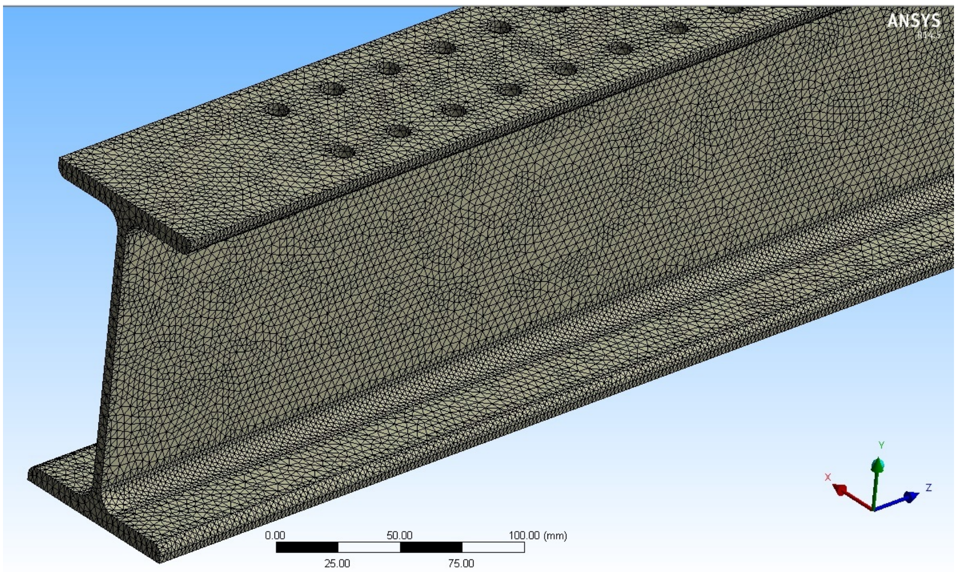

Figure 4.3: Magnified view of meshing

The beam is having a length of 1168 mm and size as per the ISMB150 selected from the manufacturing catalogues as discussed earlier. It is fixed at both ends as shown in fig.4.3.

Total load on the beam is 40000 N as discussed in section 4.1. The load is applied at the center of the beam as shown in figure4.2 to represent the worst load condition and it is analyzed in static structural analysis module of the ANSYS14.5.

4.3.2 Equivalent stress in support beam

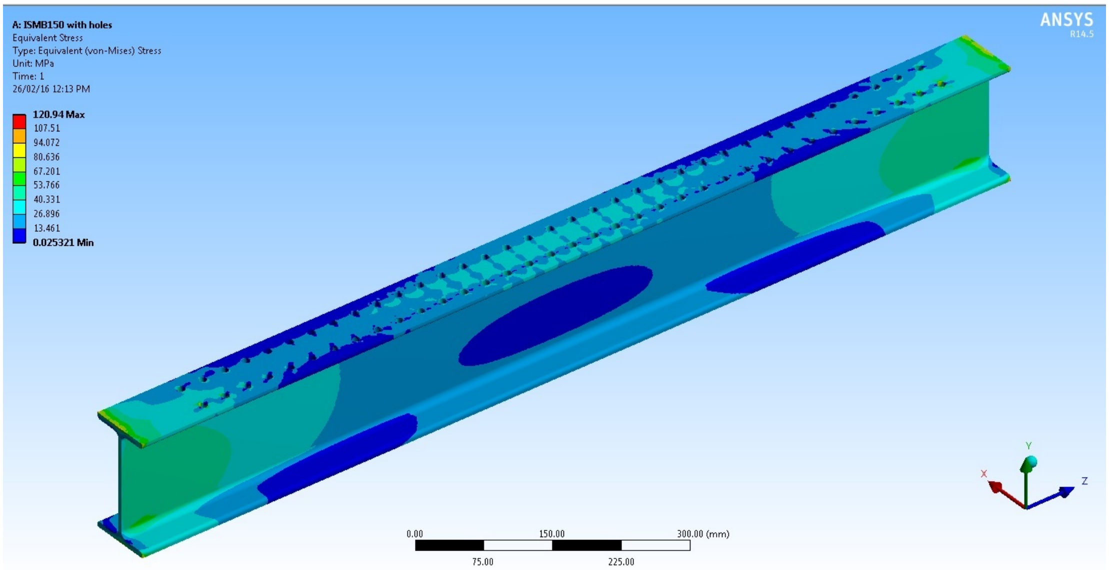

Figure 4.4: Equivalent stress (Von-Mises) for support beam from FE analysis

FE analysis of the support beam has been carried out with load and boundary conditions discussed in previous section. The equivalent Von-Misses stress predicted as 120.94 MPa as shown in figure4.4. The stress is less than the yield stress (235 MPa). Hence the support beam is safe under bending load condition.

4.3.3 Total deformation of support beam

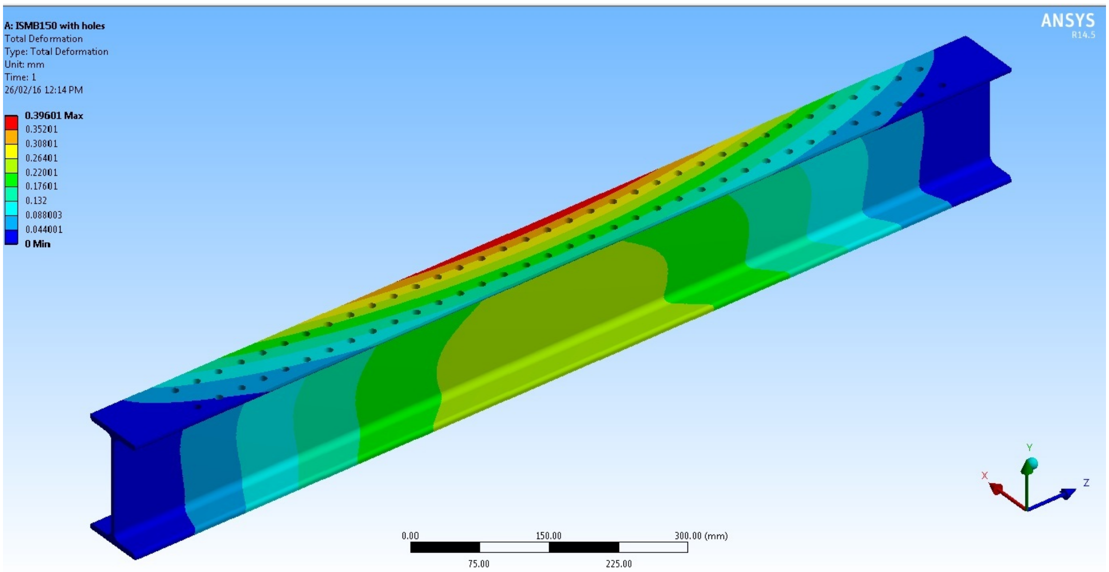

Figure 4.5: Total deformation for support beam from FE analysis

The max. total deformation found from FE analysis in the support beam is 0.3960 mm as shown in figure4.5 which is less than the allowable deformation (3.24 mm). This is in close agreement to analytical deformation (0.2268 mm). Hence the support beam is safe under the loading condition.

4.3.4 Maximum shear stress in support beam

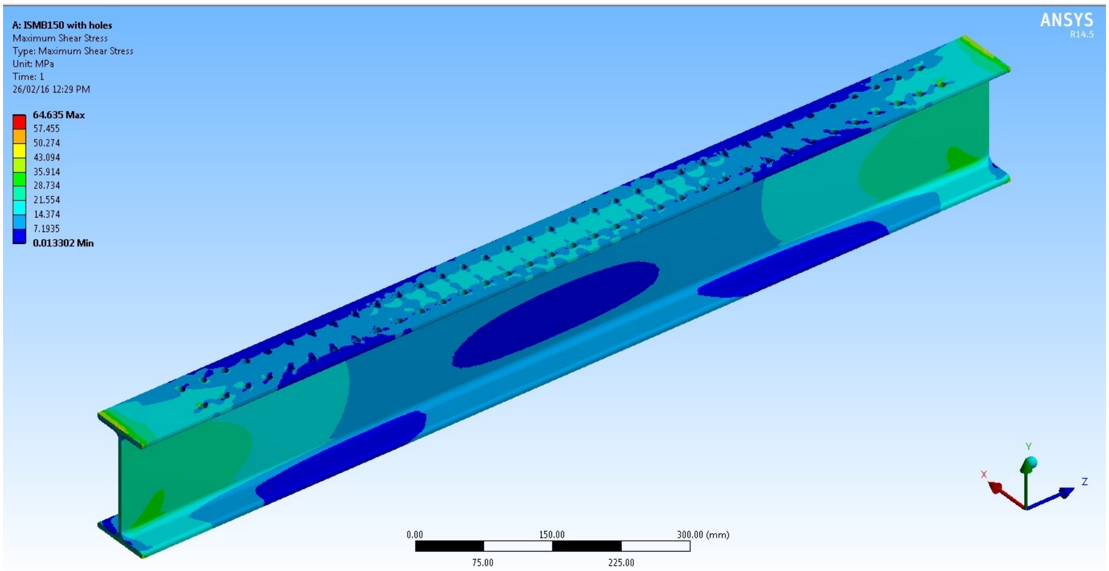

Figure 4.6: Maximum shear stress for support beam from FE analysis

The maximum shear stress obtained through FEA is 64.635 MPa as shown in figure4.6 which is less than the allowable shear stress of the material 117.5 MPa hence the support beam is safe under shear failure.

4.4 Adjustable column design

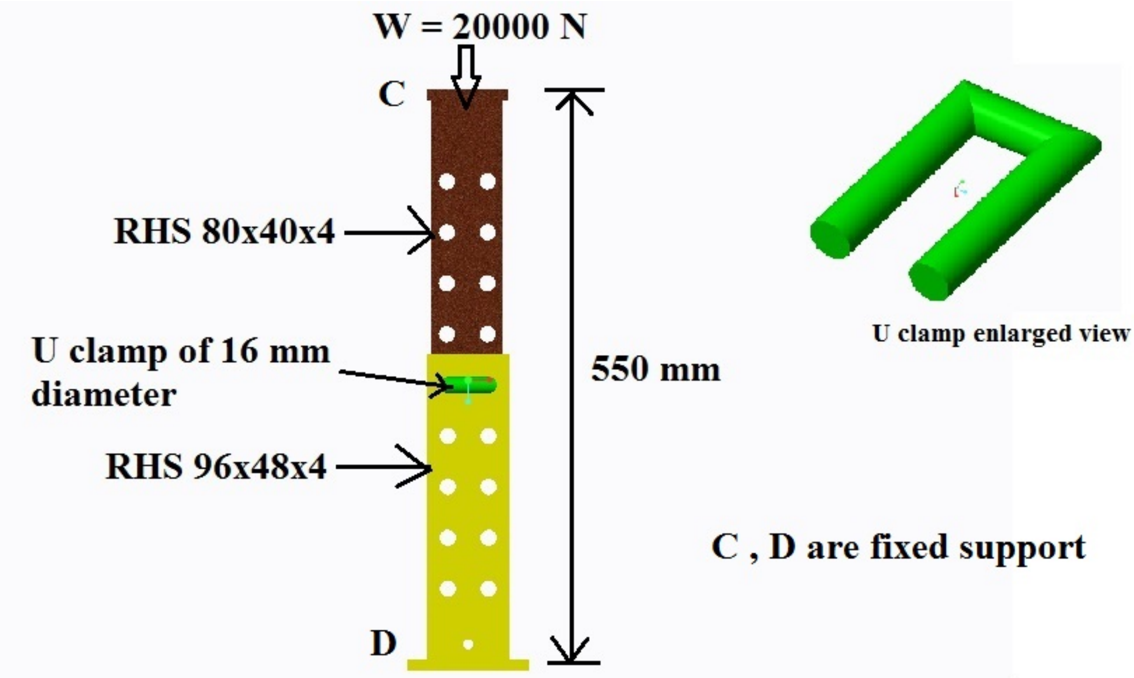

Figure 4.7: Load and constraints given for adjustable column

The total load on the support beam (40000 N) is going to develop the reaction force of 20000 N (compressive) at each column which is fixed at both ends (C and D). Here column is kind of short in length (550 mm) and hence when such columns are subjected to axial compressive forces they failed by crushing and crushing load is due to the combined effects of axial loads and bending stress due to the bending moment caused due to buckling. To support the beam rigidly, RHS 80x40x4[14] mm is selected as it will cover the entire width of the beam and hence there will no chances of toppling of beam during loading and/or process. The load is acting through the centroid of the column and hence it is under direct axial compression. The compressive stress[13] in a column can be worked out as,

$\sigma_{Compressive}$= W / A = 20000 / 855 = 23.39 N/mm^2

which is less than the allowable compressive stress (140 MPa) of the material[13].

Two rectangular hollow sections are connected with the U-clamp which is a rod of mild steel bent in U shape.This U clamp will be in double shear under operating condition and hence calculating the minimum diameter that is required to withstand a max. load of 20000 N,

In double shear τ = F/2A

τ = 0.5 x yield strength = 0.5 x 235 = 117.5 Mpa

$117.5 = 20000/(2 \times 3.14 \times d^2/4)$

$d = 10.41mm$

From above calculation the diameter of U-clamp has been selected as 16 mm

4.5 Adjustable column analysis

4.5.1 Geometry and meshing of a column

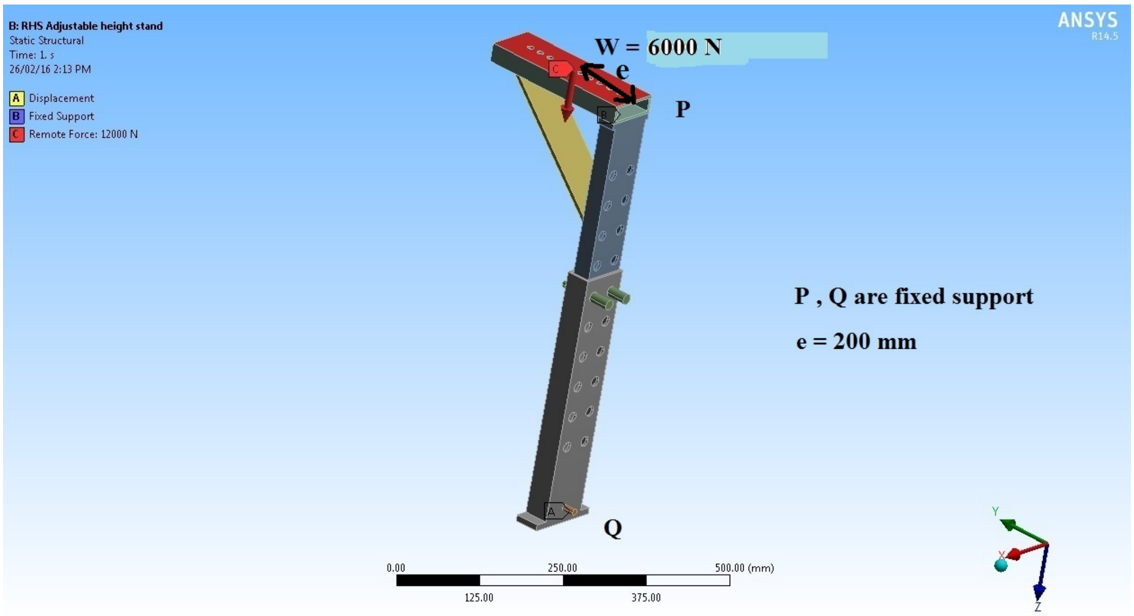

Figure 4.8: Load and constraints given for column

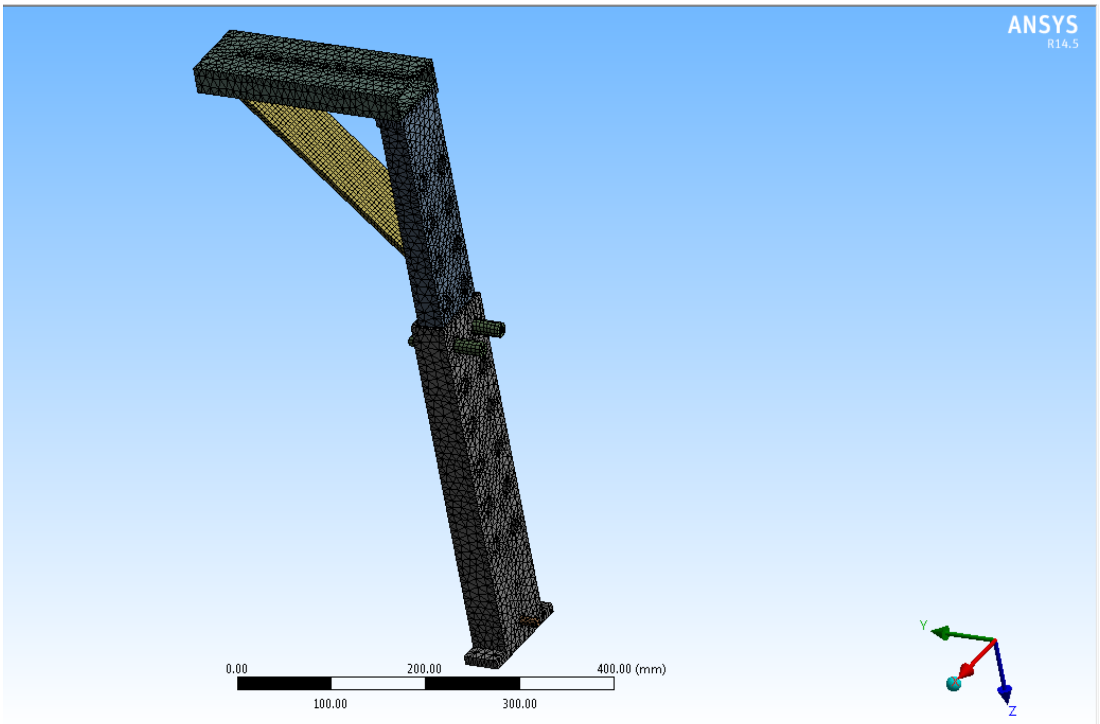

Figure 4.9: Meshing of the adjustable column

The column selected is having size as per RHS 96X48X4 and RHS 80X40X4 manufacturing catalogues[14]. It is fixed at both ends and loaded at top by 6000 N as it has to support one end of rubber roll only. The meshing is done using automatic method which uses the quadratic hexahedron and wedge elements. The meshing and geometry of the column is shown in the figure4.8 and figure4.9 respectively.

The RHS 80x40x4 is the weaker member of the column and hence the stress and deformation can be worked out as follows,

Properties of RHS 80x40x4 are as follows:

- Area A = 855 mm^2

- Moment of inertia about minor axis I_{y-y}= 21.49 x 104 mm^4

- Least radiation of gyration r_{y-y} = 27.5 mm

- Eccentricity e = 200 mm

- Distance of extreme fiber from neutral axis c = 40 mm

- Length of column L = 550 mm

The column with eccentric load can be designed by Secant's formula[17],

$\sigma = \frac{W}{A}\left[1 + \frac{e \times c}{r^2} \times sec\left\{\frac{1}{2} \times \sqrt{\frac{W}{E \times A}} \times \frac{L}{r}\right\}\right]$

$\sigma = \frac{6000}{855}\left[1 + \frac{200 \times 40}{27.5^2} \times sec\left\{\frac{1}{2} \times \sqrt{\frac{6000}{196000 \times 855}} \times \frac{0.65 \times 550}{27.5}\right\}\right]$

$\sigma = 62.6939 N/mm^2 < 140 N/mm^2$(allowable compressive stress)

The maximum deformation[17] of the adjustable column under eccentric load can be gives as,

$y_{max.} = e \times \left[sec\left(\pi \times \sqrt{\frac{W}{W_{cr}}}\right) - 1\right]$

$W_{cr} = \frac{\pi^2 \times E \times I}{L^2}$

Where, W_{cr} = critical buckling load[13], N

$W_{cr} = \frac{\pi^2 \times 196000 \times 21.49 \times 10^4}{(0.65 \times 550)^2}$

= 3249373.922 N

$y_{max.} = 200 \times \left[sec\left\{\pi \times \sqrt{\frac{6000}{3249373.922}}\right\} - 1\right]$

$y_{max.} = 0.00013864$ mm

The max. deformation in the column is well below the permissible value(1.52 mm)[16].

4.5.2 Equivalent stress in adjustable column

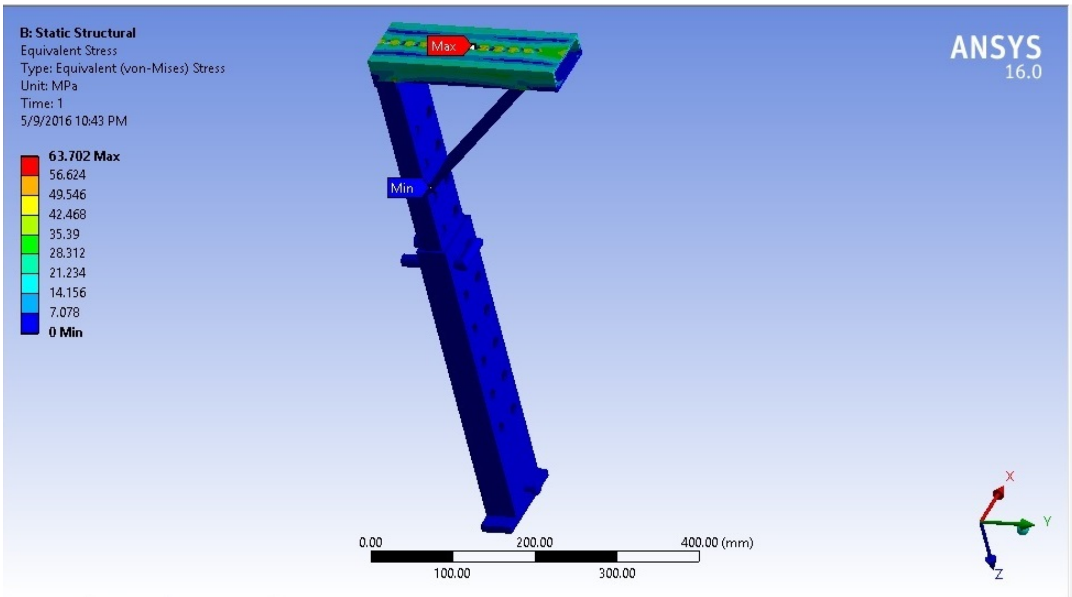

Figure 4.10: Equivalent stress (Von-Mises) for adjustable column from FE analysis

The equivalent stress in a column obtained through FEA is 130.14 MPa which is less than the allowable compressive strength of the material 240 MPa hence it is safe against axial compression failure. The stress distribution can be seen in the figure4.10.

4.5.3 Total deformation of adjustable column

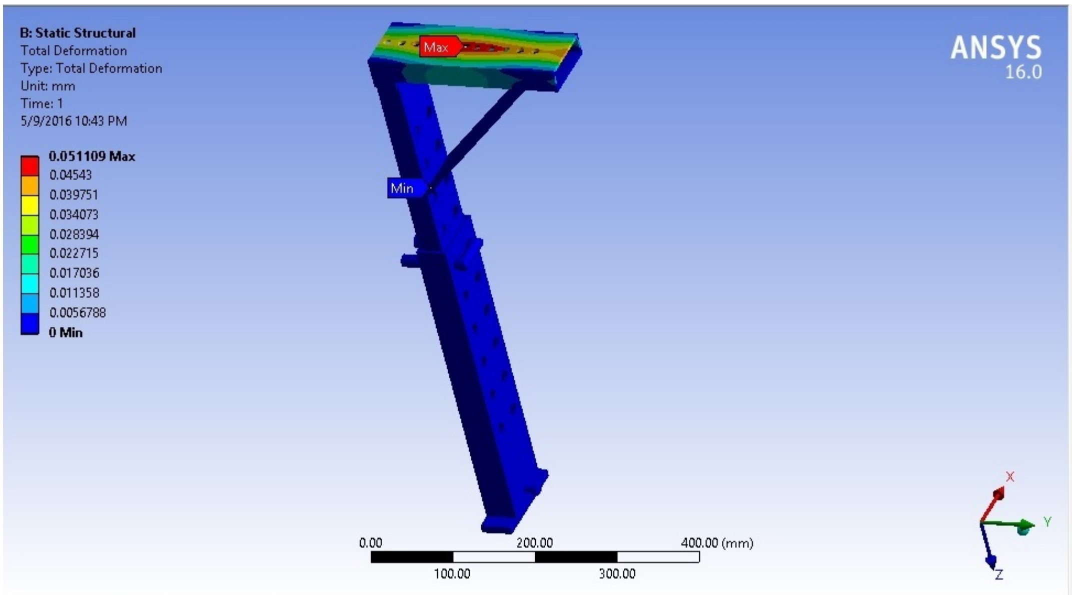

Figure 4.11: Total deformation for adjustable column from FE analysis

The results obtained for total deformation without considering buckling is obtained as

0.10234 mm which is within permissible limit hence it is safe against deformation failure. The deformation pattern can be seen in the figure4.11.

4.5.4 Maximum Shear stress in adjustable column

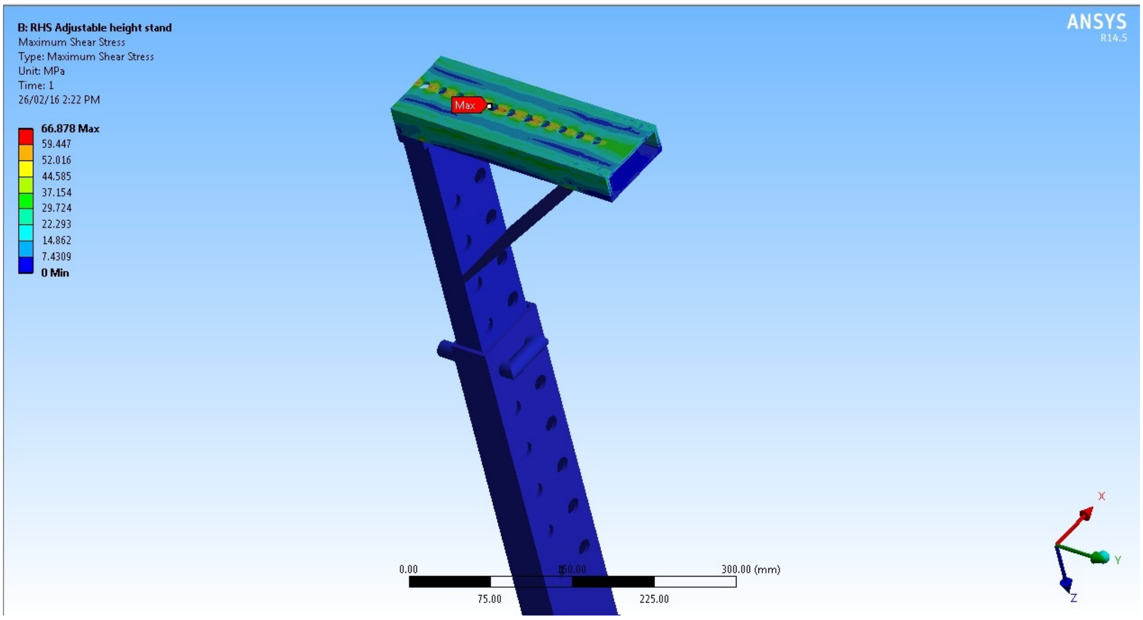

Figure 4.12: Maximum Shear stress for adjustable column from FE analysis

The maximum shear stress within the adjustable column is found to be maximum 66.878 Mpa which is less than the allowable shear stress of the material 117.6 MPa and hence the adjustable column is safe against shear failure. The distribution of the shear stress can be seen in the figure4.12.

4.6 Analysis of axle of trolley structure

Trolley is sliding on the rail by means of wheel and axle assembly. Trolley is having six number of axle equally placed which will take the total load of the rubber rolls including the self-weight of the trolley structure. Considering the worst condition the maximum load that will be bear by each axle is 32000 N acting at its center having its both ends fixed with diameter of 60 mm. Therefor axle will bend under such loading condition which can be calculated as follows,

Total weight of trolley structure including weight of rubber rolls = 20000 Kg

So the force that will be coming on each axle = 20000 x 9.81/6 = 32000 N

Bending stress[18],

$\sigma_b = 32 \times M_b/\pi \times d^3$

Bending moment for axle having fixed ends and load at center,

$M_b = W \times L/8 = 32000 \times 962/8 = 3848000N - mm$

Bending stress $\sigma_b = 32 \times 3848000/\pi \times 60^3 = 181.55 N/mm^2$

Allowable stress $\sigma_y = 235\ N/mm^2$

Hence the bending stress in axle is less than the yield stress of the material therefor it is safe in bending failure.

Shear stress in axle, $\tau = \frac{16 \times T}{\pi \times d^3}$ MPa

Where, T = Torque generated on the axle during beggining of travel, N

T $= \mu \times R_N \times R$

Where, μ = static friction co-efficient between wheel and rail = 0.2

R_N = Force acting on the axle = 32000 N

R= Radius of wheel = 102mm

T =0.2 × 32000 × 102= 652800$N - mm$

$\tau = \frac{16 \times 652800}{\pi \times 60^3} = 15.39 N/mm^2$

which is less than the allowable shear stress (117.5 MPa)[13] of the material and hence the axle is safe in shear failure.

Deformation of axle[15] can be given by,

$\Delta_{max} = W \times L^3/(192 \times E \times I) = 32000 \times 962^3 \times 32/[192 \times 201600 \times \pi \times 60^4]$

$\Delta_{max.} = 1.8173 mm$

Allowable deformation[16] of the axle can be found as,

$[\Delta_{max}]_{allowable} = \frac{L}{360} = \frac{962}{360} = 2.6722 mm$

which is higher than the max. deformation and hence the axle is safe under bending load condition.

4.7 Axle analysis

4.7.1 Geometry and meshing of axle

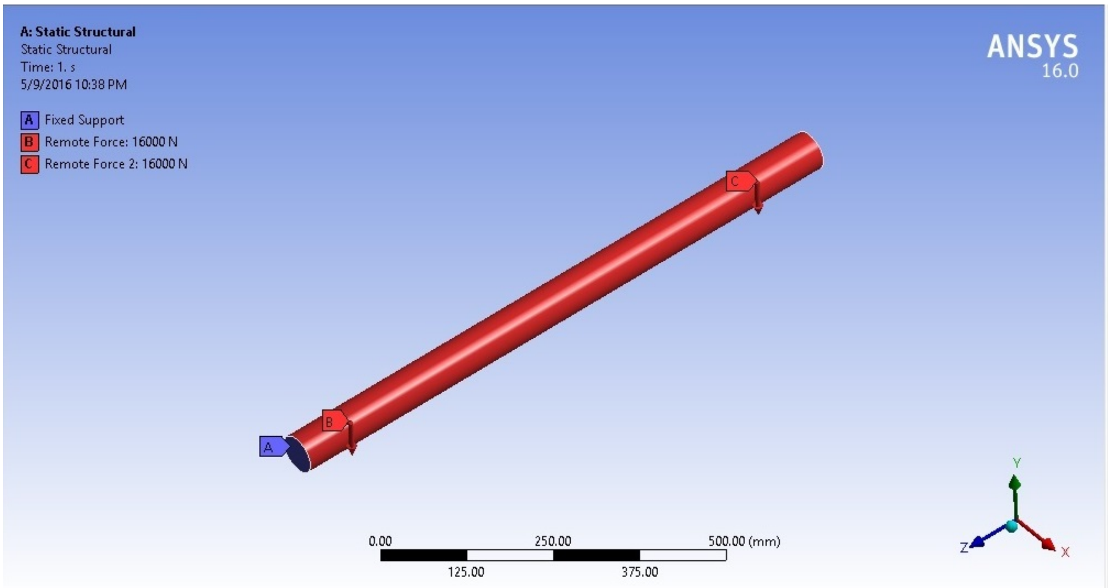

Figure 4.13: Load and constraints given to axle

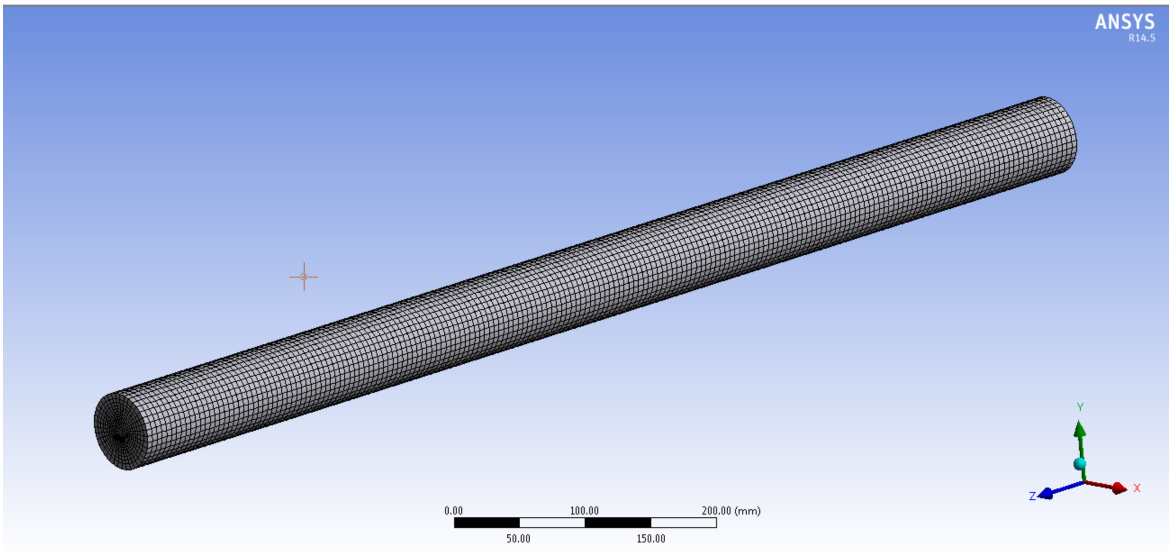

Figure 4.14: Meshing of axle

The geometry of the axle is shown in the figure4.13 with the constraint and loading conditions. The axle is fixed at both ends with load of 16000 N acting on both the ends(100 mm from end). The meshing of the axle is done using quadratic hexahedron elements which can be seen in the figure4.14.

4.7.2 Equivalent stress in axle

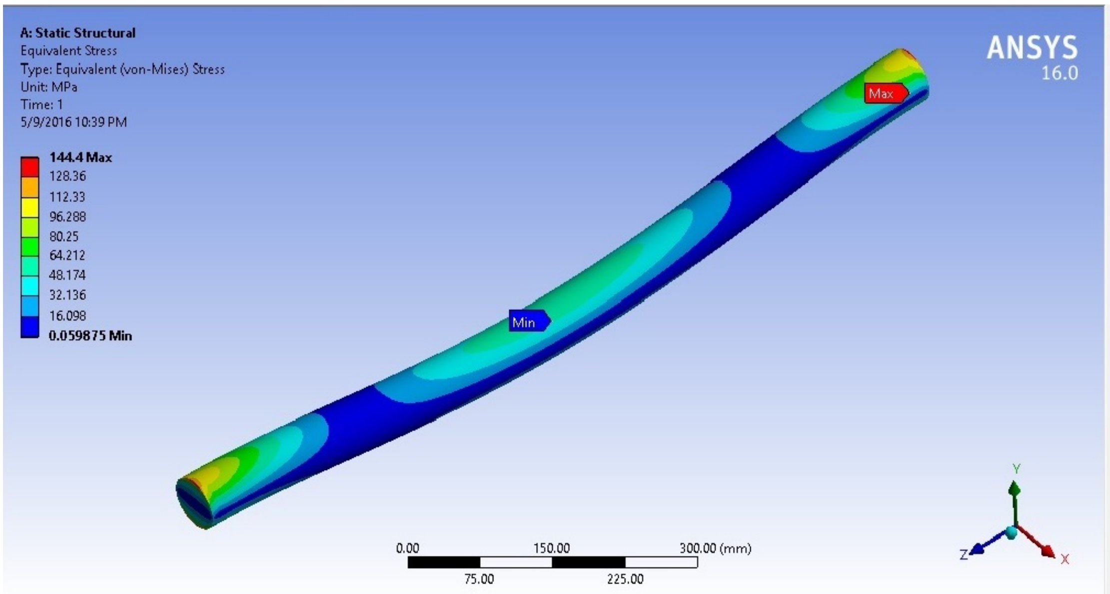

Figure 4.15: Equivalent stress (Von-Mises) for axle from FE analysis

The results for equivalent stress generated in the axle is 144.4 MPa is shown in the figure4.15 which is less than the yield strength of the material 235 MPa hence it is safe against bending failure.

4.7.3 Total deformation of axle

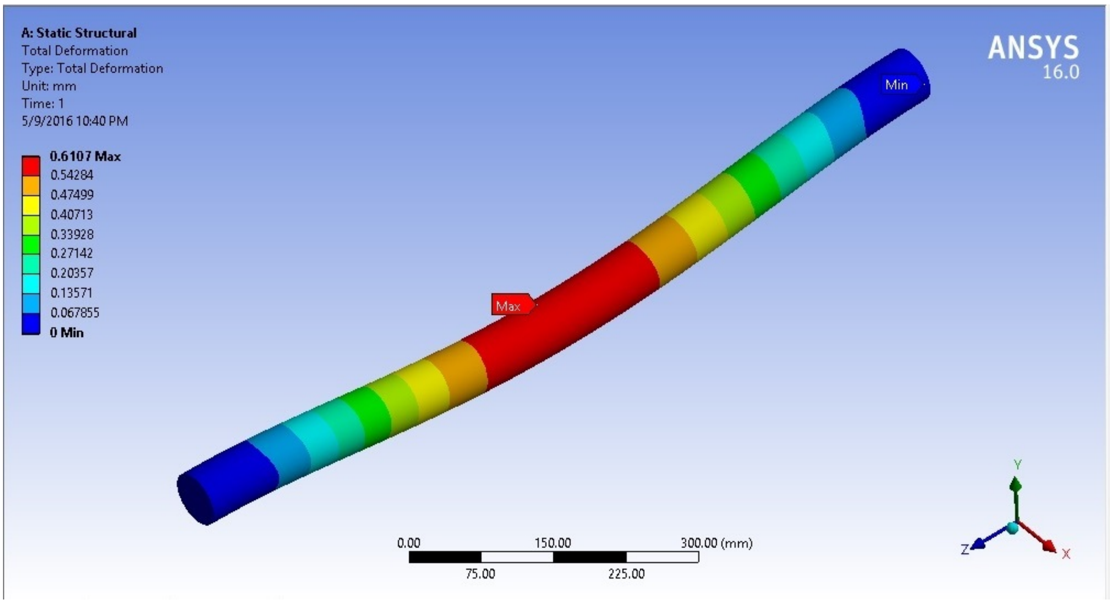

Figure 4.16: Total deformation for axle from FE analysis

The axle will bend under the given load which will cause the deformation in the load direction which is 0.6107 mm which is within the acceptable limit and hence it is safe. The deformation pattern can be seen in the figure4.16.

4.7.4 Maximum shear stress in axle

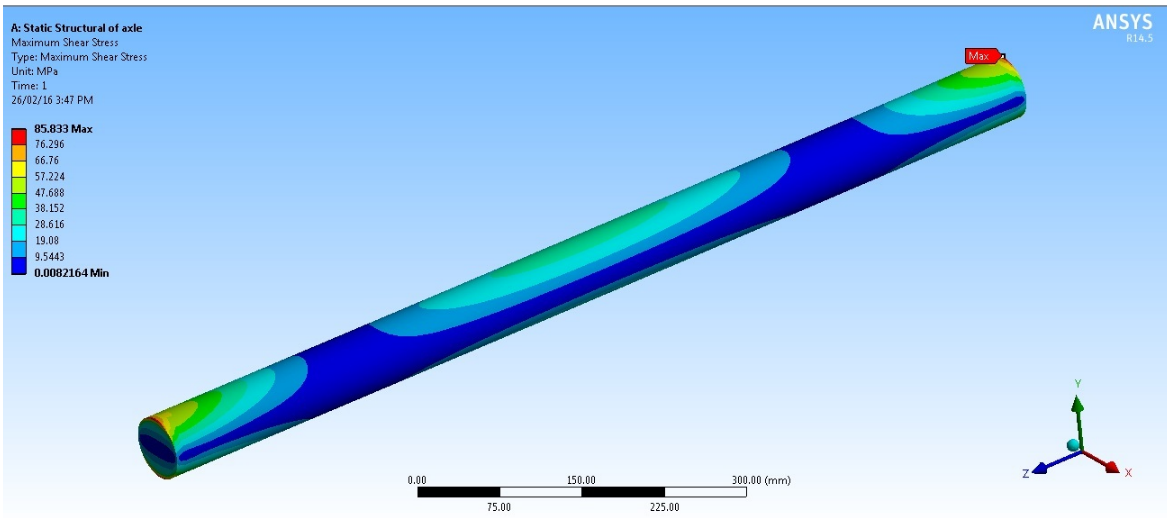

Figure 4.17: Maximum shear stress for axle from FE analysis

The maximum shear stress generated in the axle is 85.833 MPa which is less than the allowable shear stress for the material 117.5 MPa and hence the axle is safe against shear failure as shown in figure4.17.

4.8 Contact stress between wheel and track

As the trolley wheel would slide on rail track inside the autoclave equipment and there is a frictional contact between them hence the contact stress will be generated with type of contact of cylinder to flat surface as shown in figure 4.16.

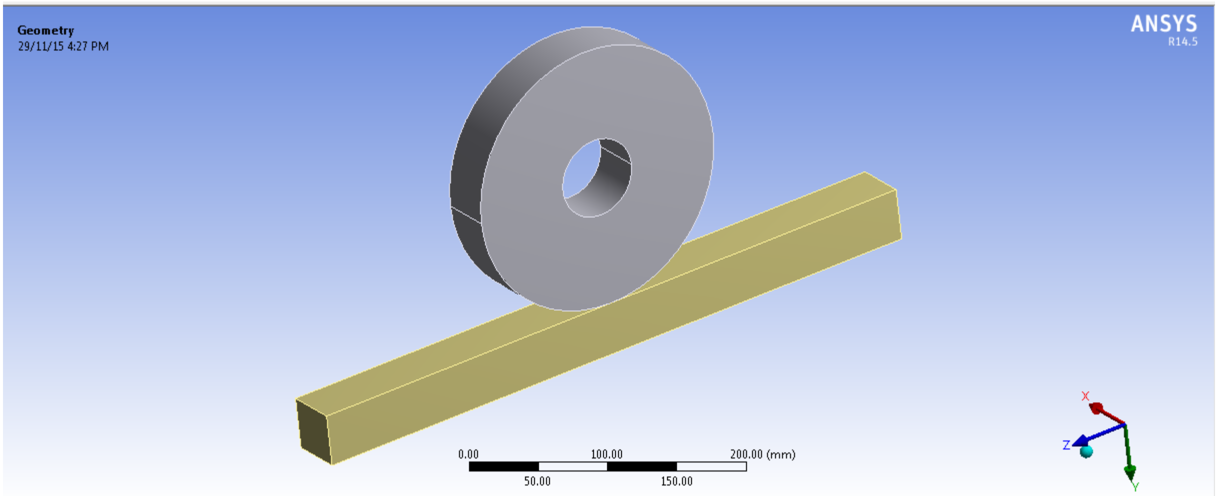

Figure 4.18: Wheel-Rail assembly

As the trolley is having six number of axle at equal distance at the bottom of the trolley having 12 number of total wheels hence the total load will be distributed on 12 wheels. For calculating the contact stress following data will be required:

- Normal load on the wheel F = 16000 N

- Radius of the wheel R = 102 mm

- Width of the wheel a = 40 mm

- Young's modulus of the cast iron wheel Ewheel = 129600 N/mm^2

Now, the contact stress[13] can be calculated as,

σ_c= 0.418 $\times \sqrt{}((P' \times E_{wheel})/R)$

Where, P' = F/a = 16000 / 40 = 400 N/mm

σ_c= 0.418 x √((4000 x 129600)/102)

= 298 N/mm^2

Now, allowable contact stress[13] is given by,

$[\sigma_c]_{allowable}$= C_R x BHN

Where, C_R = Surface hardness co-efficient = 2.3

BHN = Brinnel hardness number = 200

$[\sigma_c]_{allowable}$= 2.3 x 200 = 460 N/mm^2

It can be seen that $\sigma_c < [\sigma_c]_{allowable}$, hence wheel and rail are safe.

4.9 FE analysis for contact stress between wheel and rail

4.9.1 Geometry and meshing wheel and rail

Wheel and rail have been modelled and FE analysis has been carried out for contact stress between the wheel and rail. The CAD model and meshed models have been shown in figure4.19 and figure4.20 respectively.

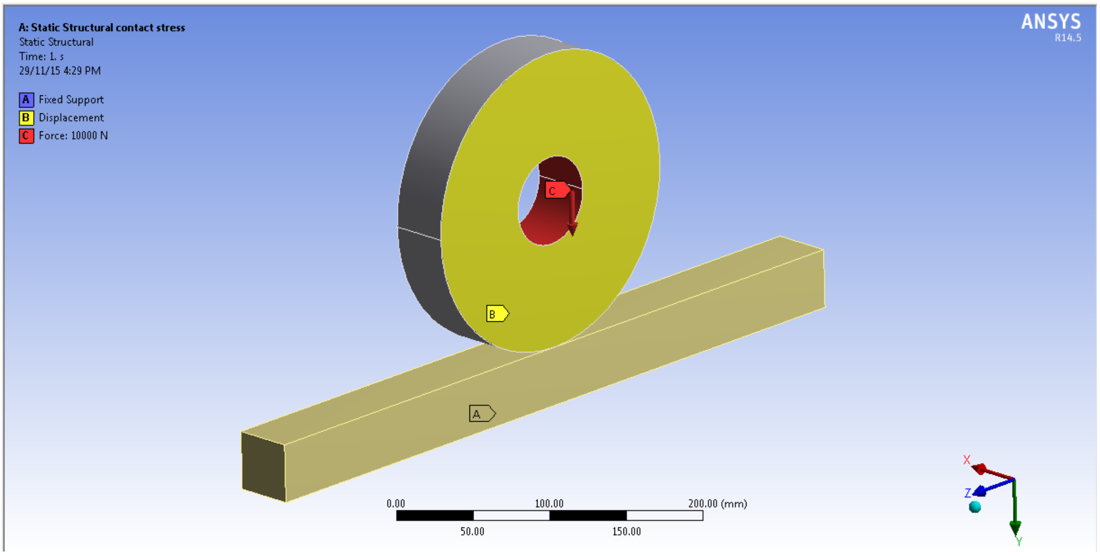

Figure 4.19: Load and constraints given to wheel and rail

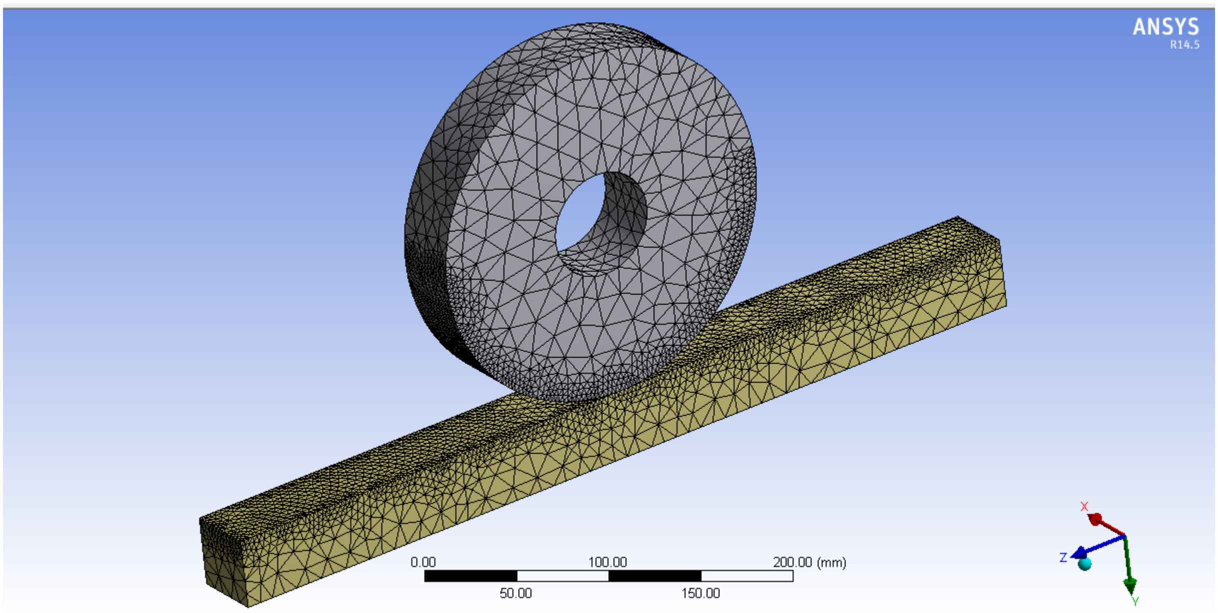

Figure 4.20: Meshing of wheel and rail

The load is given on the circular hole for axle which 16000 N and the wheel is constraint to move in X and Z direction. The contact between two is frictional having frictional co-efficient of 0.7[] as per the materials in contact. The meshing is done using quadratic tetrahedron and triangular elements. The surfaces which are in contact are refined to get the smooth distribution of contact stress.

4.9.2 Equivalent contact stress between wheel and rail

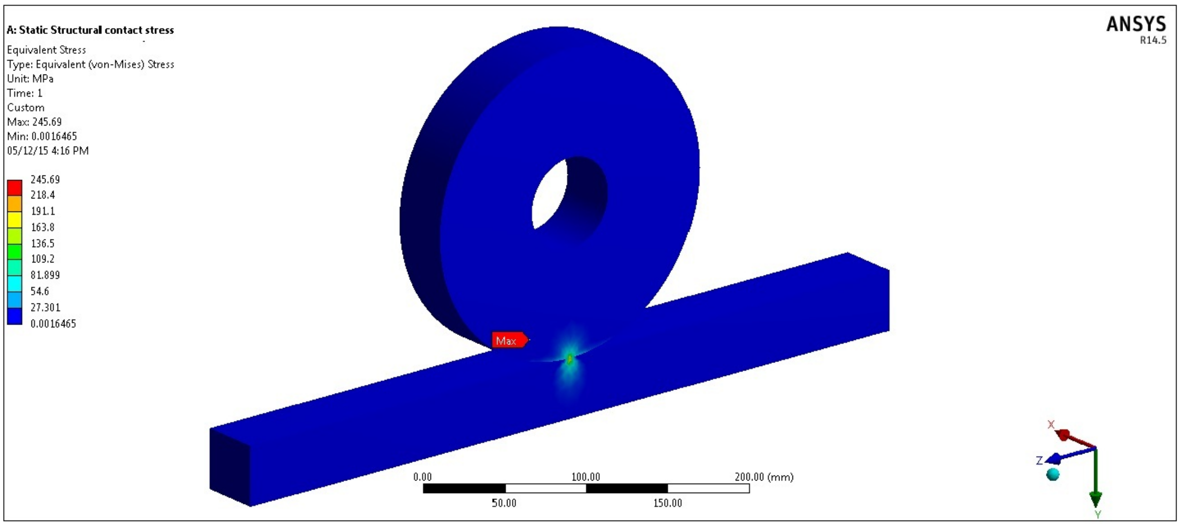

Figure 4.21: Equivalent contact stress of wheel and rail from FE analysis

The equivalent contact stress obtained from FEA is 245.69 MPa as shown in figure4.21 which is less than the allowable contact stress of the material 460 MPa hence it is safe against contact stress failure.

4.10 Bottom channel calculation

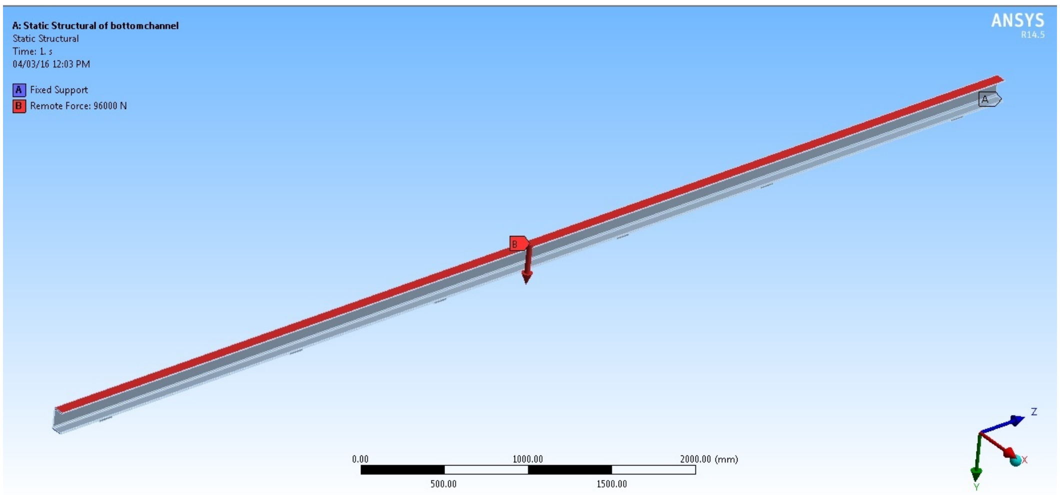

Figure 4.22: Load and constraint given to the bottom channel

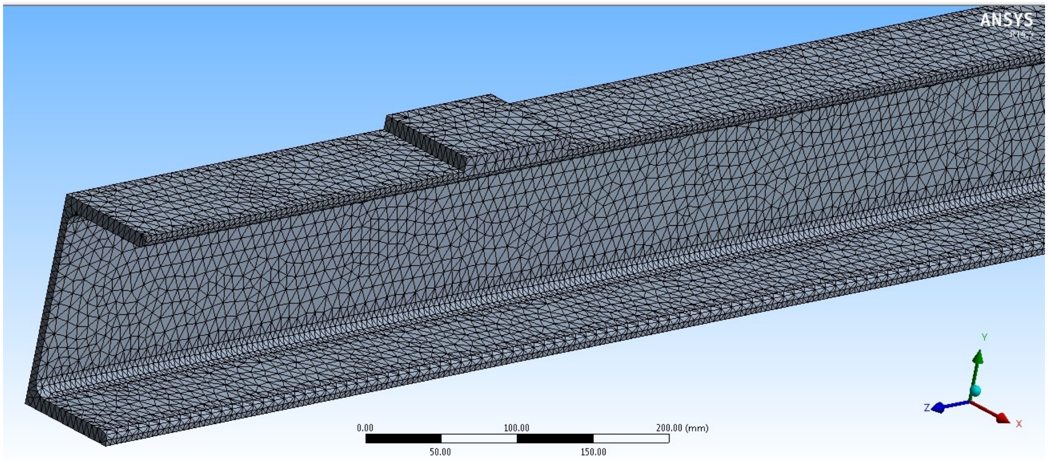

Figure 4.23: Meshing of the bottom channel

The bottom channel is provided to support the plate of trolley at both ends of the plate. The wheel and axle assembly are also to be attached at the bottom of the bottom channel and hence the fixed support for the bottom channel is given as the small plates below the bottom channel to which wheel and axle assembly will be attached. The bottom channel will take half the load of the support plate which is 96138 N as they are two number of bottom channel are provided. To consider the worst condition for it the load is applied at the center of the bottom channel. The load and constraint given can be seen in the figure4.22. The meshing is done using automatic method which uses quadratic hexahedron and quadratic wedge elements as shown in the figure4.23.

Now as it is having more number of fixed supports and overhang also at both ends with load of 96138 N at the center and hence there is no standard formula available for calculating the maximum moment generated, the maximum stress generated will be found out through FE analysis software and then it will be checked that whether it is safe of not. The allowable deflection[16] for this channel can be given as,

$[\Delta_{max.}]_{Allowable}= L_c/360$

$= 7300\ /360$

$[\Delta_{max.}]_{Allowable}= 20.27$ mm

4.10.1 Equivalent stress in bottom channel

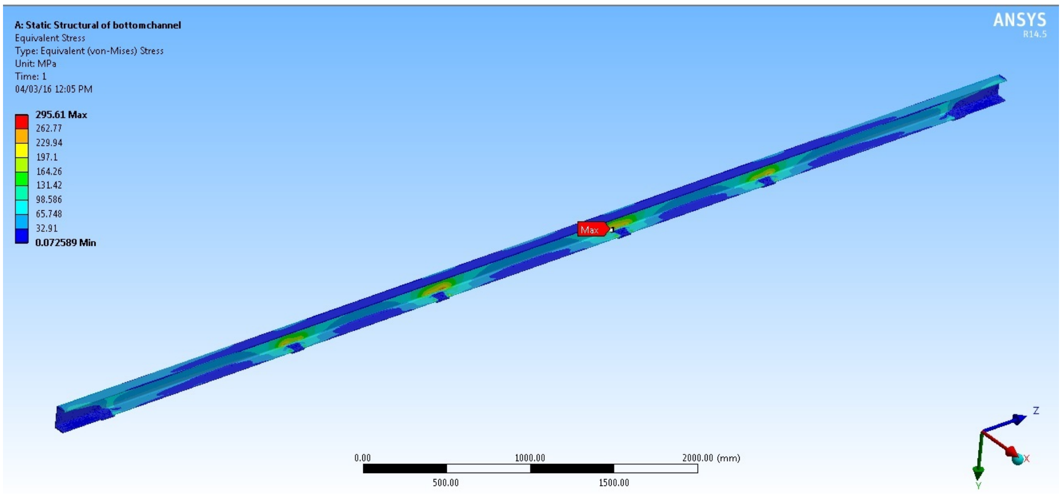

Figure 4.24: Equivalent stress (Von-Mises) of the bottom channel from FE analysis

The equivalent stress for the bottom channel is shown in the figure4.24. It can be seen that the maximum stress is 295.61 MPa which is not less than the material's yield strength 235 MPa but here the maximum stress is taking place at the sharp corner which can be avoided as the welding is to be done at that intersection of bottom channel and small plate below it.

4.10.2 Total deformation of bottom channel

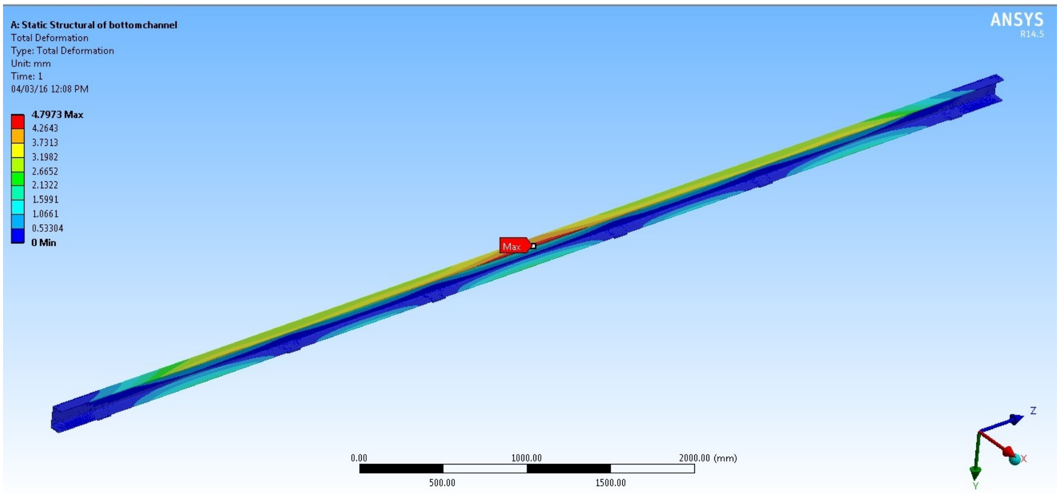

Figure 4.25: Total deformation of the bottom channel from FE analysis

The total deformation of a bottom channel found out through FEA is 4.7973 mm as shown in figure4.25 which is less than the allowable deformation (20.27 mm) and hence it safe against deformation failure.

4.11 FE analysis of plate

The support plate on which rubber rolls are to be placed is stiffened from bottom for providing rigidity and strength by attaching stiffeners like small channels and angles all along the length of the plate. The plate is having 5 mm thickness, 1168 mm width and 7300 mm of length. It is having non-uniformly distributed load but total load of 192276 N including the weight of rolls and it's self-weight. The fixed support is given to the small plate provided for attaching box for supporting axle inside it below the bottom channel. Figure4.26 shows the load and constraints given to the plate. Figure4.27 shows the meshing of the plate by using quadratic hexahedron and quadratic wedge elements using automatic method.

4.11.1 Geometry and meshing of plate

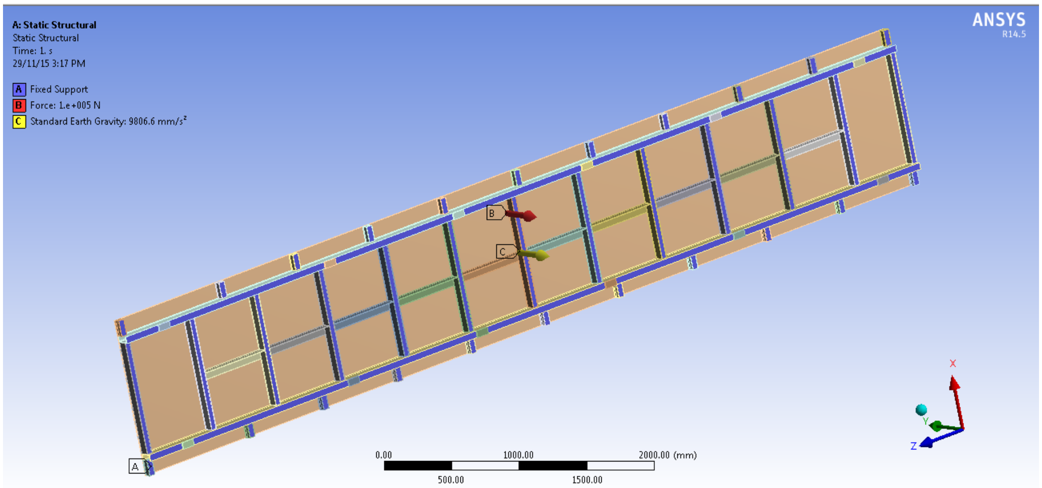

Figure 4.26: Load and constraints given to the plate

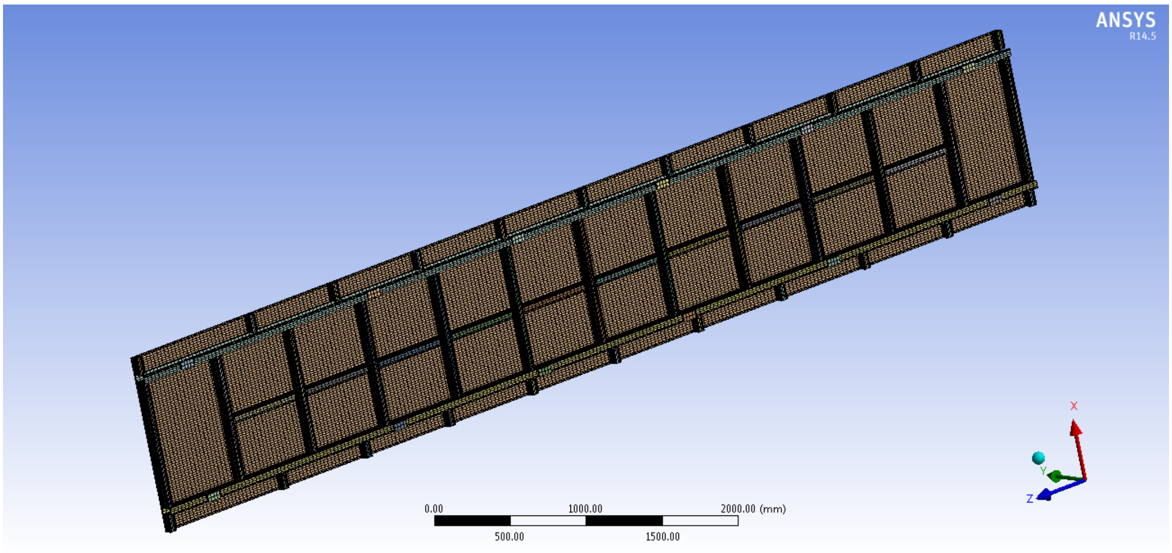

Figure 4.27: Meshing of the plate

4.11.2 Equivalent stress in plate

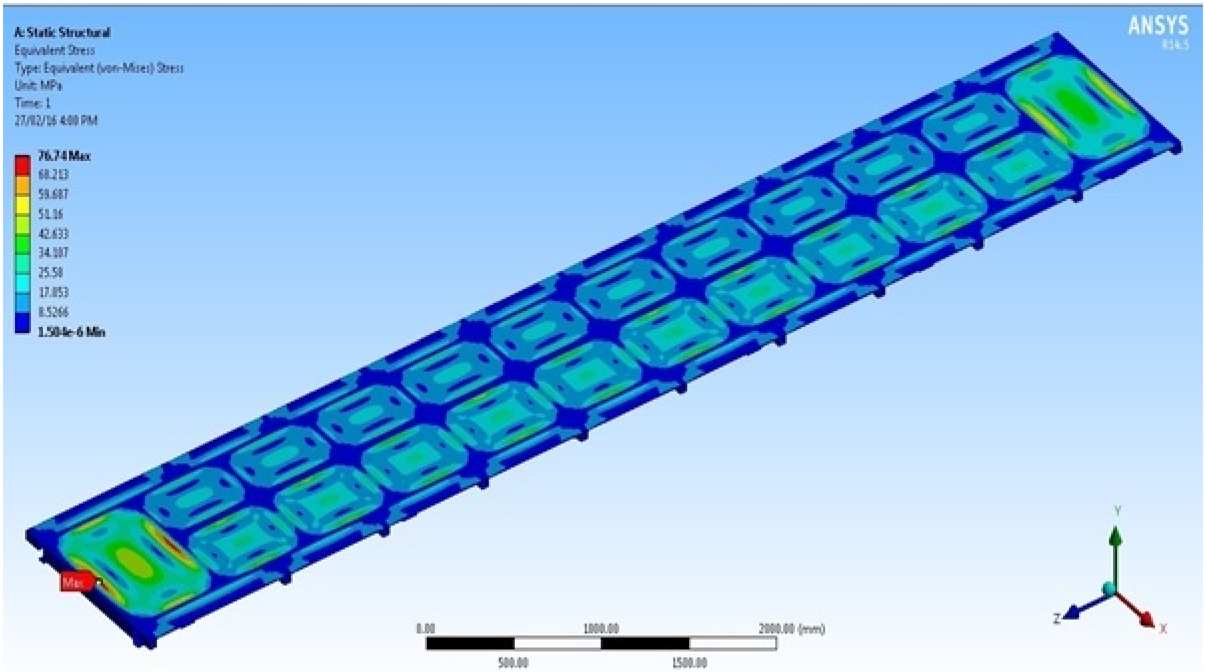

Figure 4.28: Equivalent stress (Von-Mises) of the plate from FE analysis

The equivalent stress results for plate shows that the maximum stress that can be developed in the plate under such loading is 76.74 MPa as shown in figure4.28 which is less than the yield strength of the material 235 MPa and hence the plate is safe.

4.11.3 Total deformation of plate

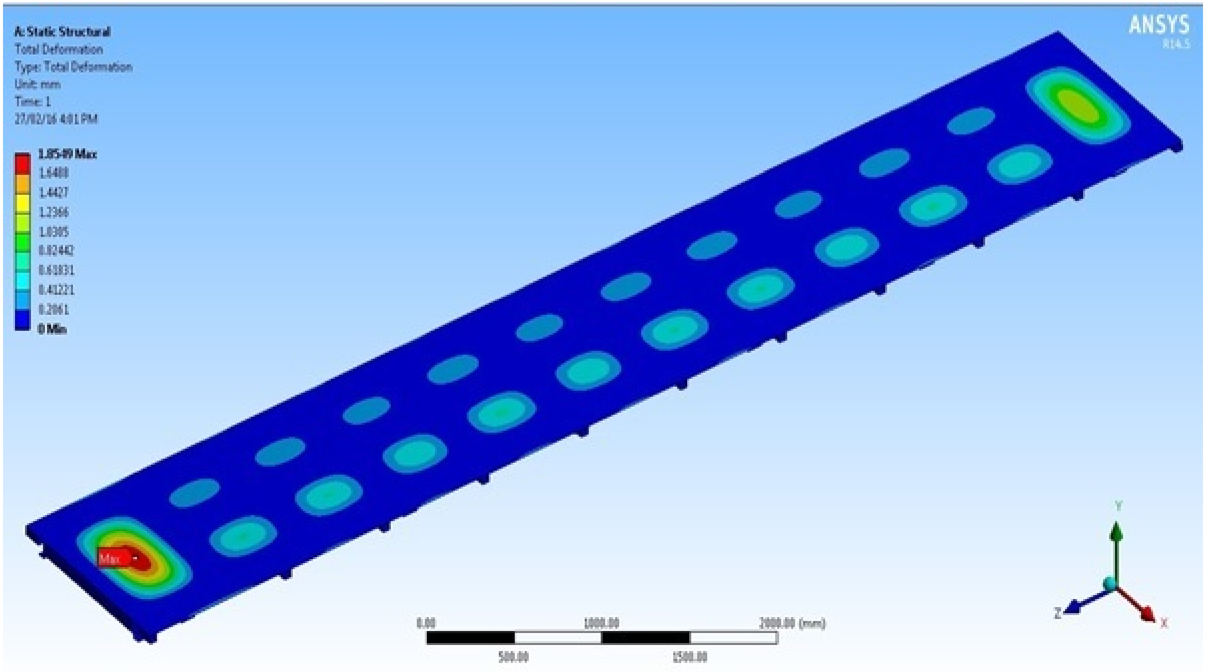

Figure 4.29: Total deformation of the plate from FE analysis

The total deformation of a plate under the loading condition found out as 1.8549 mm as shown in figure4.29 which is within permissible limit(22.46 mm) and hence it is safe against deformation under load.

4.12 Geometric modelling of the trolley structure and its components

The geometric modelling of all the components and final assembly is done in the Creo3.0 software. The dimensions are taken from the design of structural elements from the previous chapter and some from the existing trolley structure. The main components are as follows,

4.12.1 Roll with supporting stand

The rubber coated roll which is to be placed on the trolley will be supported on the fabricated stand as shown in figure4.30. There are variety of rolls coming from different

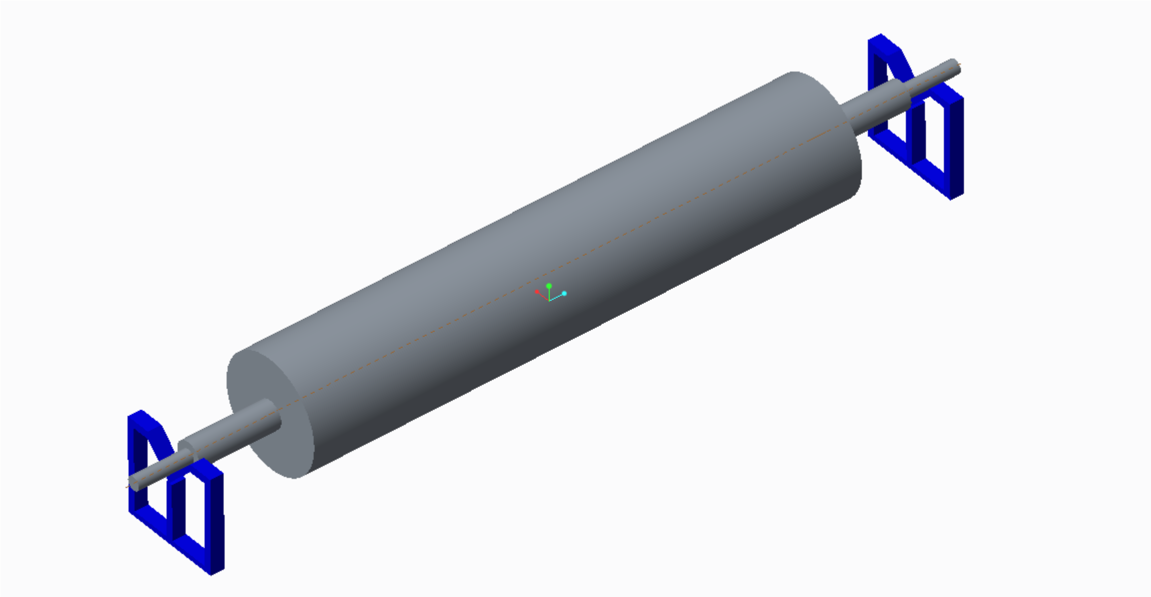

Figure 4.30: Rubber coated roll with stand

companies and for that variety of stands are prepared.

4.12.2 Proposed trolley structure with modular beam column arrangement

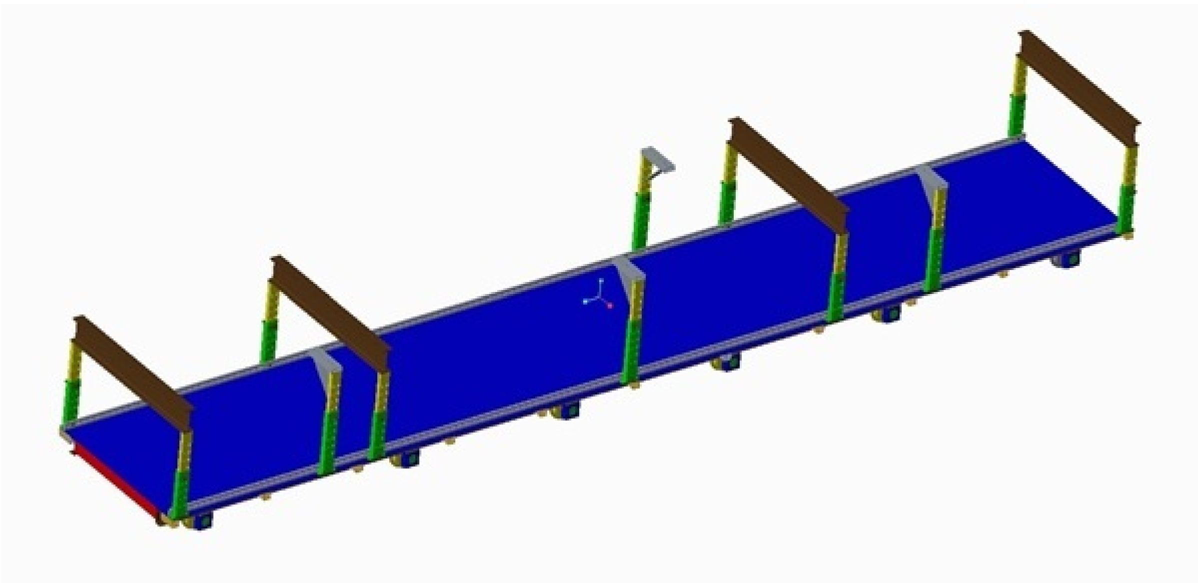

Figure 4.31: Proposed trolley structure with modular beam column arrangement

To utilize the maximum space available inside the autoclave equipment the vertical space above the existing trolley needs to be utilized. For doing so an arrangement is provided on the existing trolley such that rubber rolls can be stacked over the existing loading pattern of rolls by means of the modular arrangement. The modular beam-column arrangement shown in figure4.31 can move all along the length of the trolley providing flexibility for loading the rolls. The shown orientation of the trolley is for the most common roll loading pattern which will cater the need to accommodate the varying roll sizes as discussed in section 3.1.

4.12.3 Space utilization of autoclave equipment with proposed modular structure

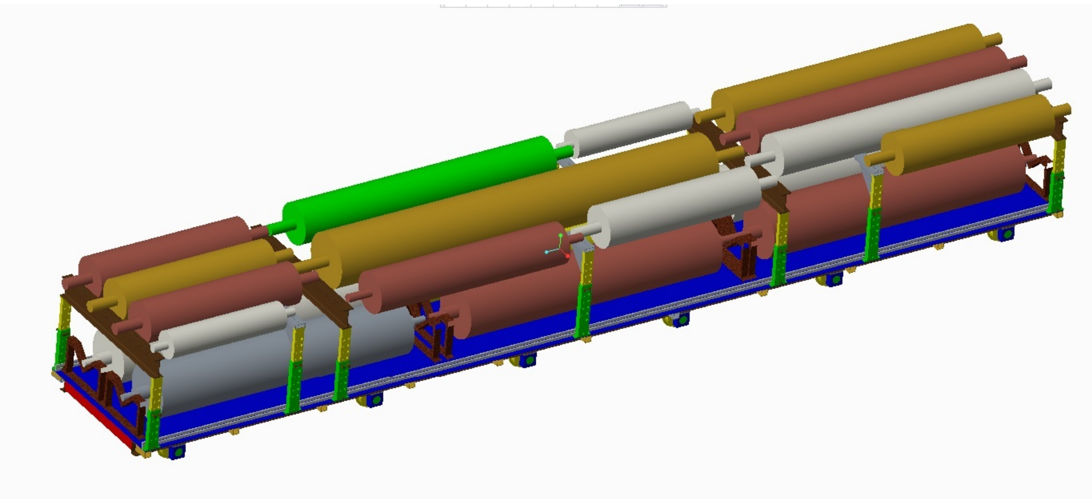

Figure 4.32: Proposed modular structure with roll arrangement

Proposed trolley structure with actual loading pattern of rubber roll is shown in figure4.32 in which it can be seen that the capacity of the autoclave can expected to be almost double than the previous trolley structure in loaded condition as shown in figure4.33 and hence productivity of the autoclave equipment has been improved.

Figure 4.33: Present method of rubber roll arrangement on a trolley

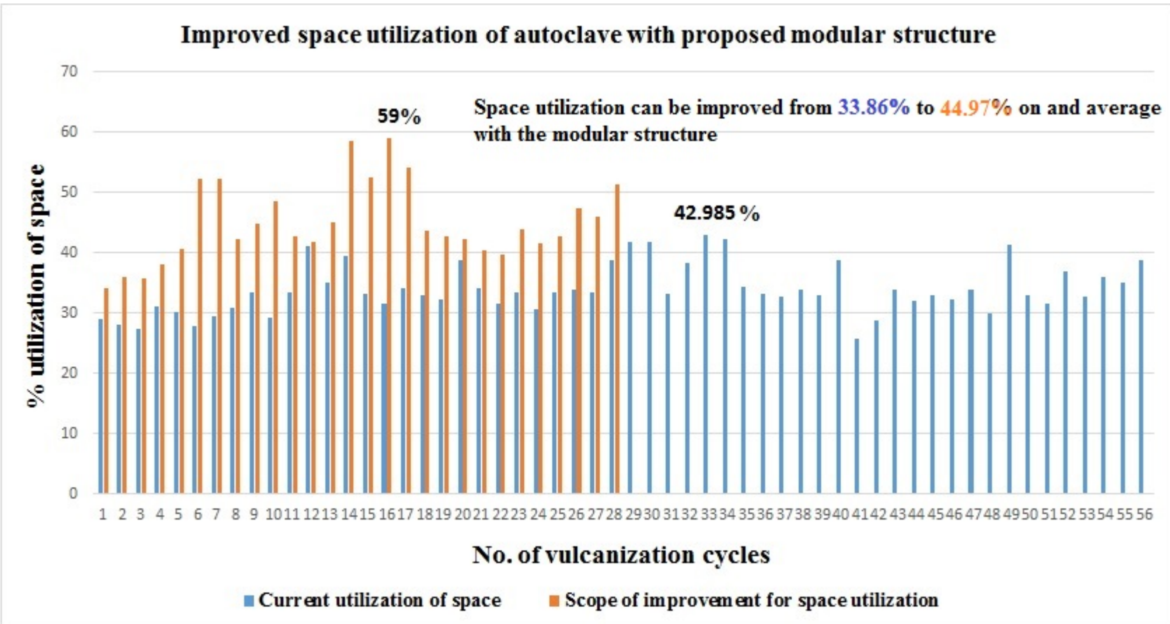

Figure 4.34: Bar chart for comparison of space utilization with modular structure[12]

As shown in figure4.34 the percentage utilization of space is on and average 33.86% presently which can up to 44.97% on and average with the help of proposed modular beam column arrangement. The bar chart shows that the same no. of rolls can be vulcanized within half of the cycles as compared to current method.

4.13 Assembly analysis

4.13.1 Geometry and meshing of proposed trolley structure

The assembly with flexible arrangement is shown in the figure4.35. It can be seen that the loads and constraints given are of same type as that in the individual components of the assembly to predict the behavior of a whole assembly under actual working conditions.

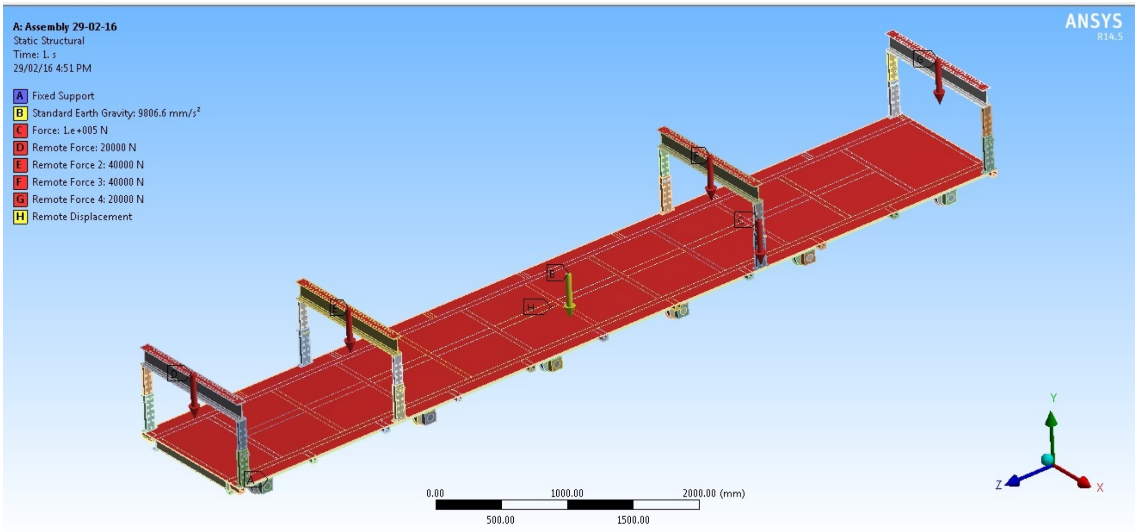

Figure 4.35: Loading and constraints given to the modular trolley structure

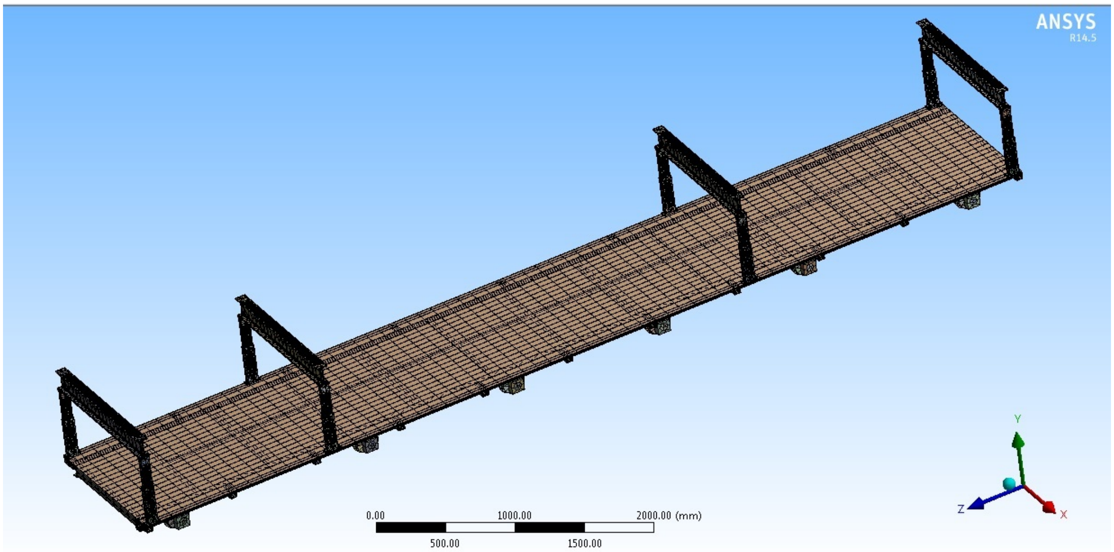

Figure 4.36: Meshing of the modular trolley structure

To consider the self-weight of the assembly the standard earth gravity is also applied. The fixed support is given to the cast wheel as they are ultimately in contact with the rails. The meshing is done using automatic method which uses the quadratic tetrahedron and triangular elements as shown in the figure4.36.

4.13.2 Equivalent stress in trolley structure

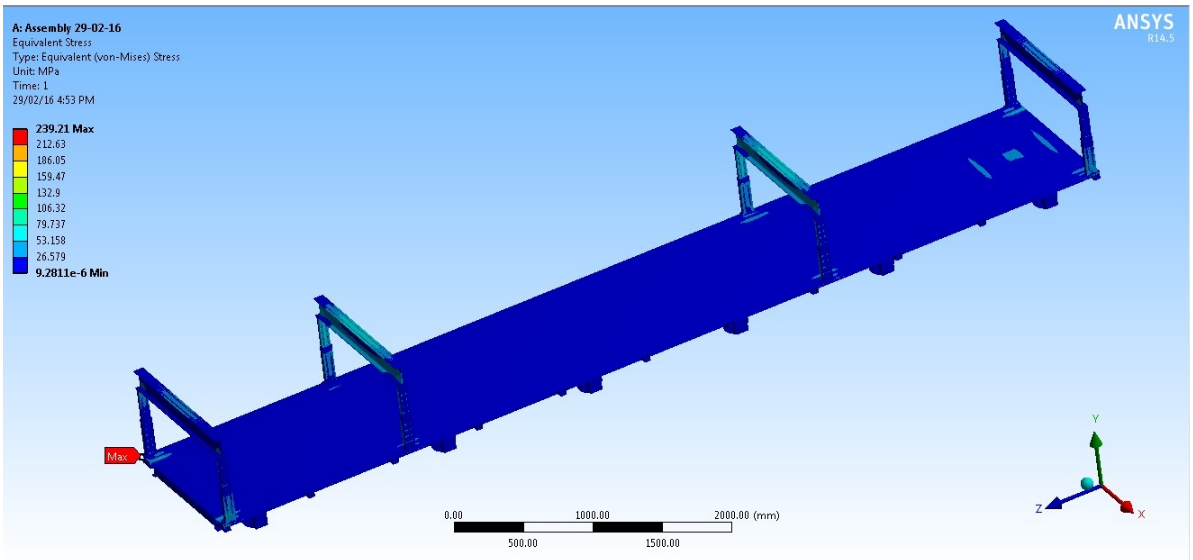

Figure 4.37: Equivalent stress (Von-Mises) of the modular trolley structure

The results for equivalent stress for the assembly found out is 239.21 MPa as shown in figure4.37 which slightly greater than the yield strength of the material 235 MPa but that can be avoided as the maximum stress is occurring at the sharp corner of the intersection of plate and a stiffener which can be taken care of through welding as there will be no sharp corner left once they both are welded together.

4.13.3 Total deformation of modular trolley structure

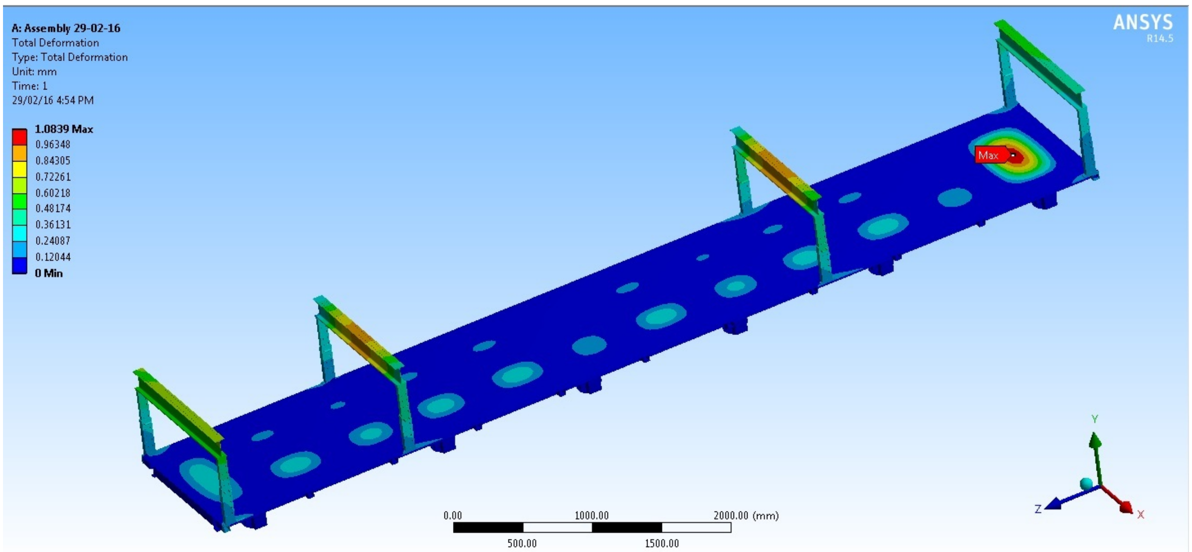

Figure 4.38: Total deformation of the modular trolley structure

The results obtained through FE analysis for total deformation of modular trolley structure is 1.0839 mm as shown in figure4.38 which is within acceptable limit(22.46 mm).

4.14 Calculations for excess steam requirement after implementing proposed structure

In order to vulcanize the rubber coated rolls the autoclave is used in which the steam at specific pressure and temperature is supplied for defined period of time depending upon the type of rubber to be vulcanized. Currently the autoclave is under-utilized in terms of space available inside it and hence the steam is also getting wasted as the autoclave is running full with the steam continuously for nearly 24 hours of operation.

The proposed modular trolley structure has improved space utilization of the autoclave. An attempt has been made to calculate if the existing supply of steam is sufficient for improved capacity or not.

The boiler consumes 400 liters of furnace oil on daily basis and the calorific value of that furnace oil is around 10500 kcal/Kg. The boiler has combined thermal and combustion efficiency of 74% and hence the total heat supplied will be multiplied with the boiler efficiency and also the radiation loss that is 2% for oil fired boiler is to be considered. Now the heat supplied by burning 400 liters of furnace oil per day can be calculated as below,

Heat supplied = mass flow rate of furnace oil/day x calorific value of furnace oil

= 400 x 4187 x 10500 / (24 x 3600)

= 203534.72 kJ/sec

Above generated heat will be absorbed mainly by

1. Autoclave shell

2. Trolley for putting rubber rolls

3. Rubber rolls

Now the heat that will get absorbed can be calculated in the following manner,

- Heat absorbed by autoclave shell:-

$Q = \frac{m \times Cp \times \Delta T}{t} \ \frac{kJ}{s}$

- Mass of autoclave, m= Density x volume = 7930 x 15.27 = 121091 Kg

- Cp = specific heat of material[19]= 0.501 kJ/Kg-K
- ΔT = Temp. difference of 150° C(423 K) and room temp. of 45° C(318 K)

$Q = \frac{121091 \times 0.501 \times (423-318)}{24 \times 3600}\ \frac{kJ}{s}$
$= 73.726\ kJ/s$

- Heat absorbed by trolley:-

$Q = \frac{m \times Cp \times \Delta T}{t}\ \frac{kJ}{s}$

- Mass of trolley = 1200 Kg

- Specific heat of trolley material (Mild Steel)[19] = 0.465 kJ/Kg-K

$Q = \frac{m \times Cp \times \Delta T}{t}\ \frac{kJ}{s}$
$Q = \frac{1200 \times 0.465 \times (423-318)}{24 \times 3600}$
$Q = 0.6781 \frac{kJ}{s}$

- Heat absorbed by rubber roll:-

$Q = Q_{rubber} + Q_{metalshell}$

- Mass of metal shell = 500 Kg max.

- Rubber weight = 100 Kg max.
- Specific heat of rubber = 1.880 kJ/Kg-K

$Q = [(\frac{m \times Cp \times \Delta T}{t}_{rubber}) + (\frac{m \times Cp \times \Delta T}{t}_{metalsheel})]\frac{kJ}{s}$
$Q = \frac{500 \times 0.465 \times (423-318)}{24 \times 3600} + \frac{100 \times 1.880 \times (423-318)}{24 \times 3600}$
$Q = 10.22 \frac{kJ}{s}$ per rubber roll.
The autoclave can accommodate max. 20 no. of rolls on and average and hence
$Q = 10.22 \frac{kJ}{s}$

Now the total heat absorbed can be found out by adding the heat absorbed by above three components which can be given as, Total heat absorbed = 73.726 + 0.6781 + 10.22 = 84.62 $\frac{kJ}{s}$

The boiler is the common source of heat for all the autoclaves inside the company and hence the net heat that is supplied to the autoclave of our concern can be found out by knowing the total volume that is being heated by the boiler and then by dividing the percentage volume of autoclave with that total volume. By doing so it appeared that the autoclave of our interest has 15% of total volume i.e. it is utilizing the 15% of total heat suppliesd by the boiler per day so the net heat supplied can be found out as,

Net heat supplied = 0.15 x 203534.72 = 30530.20 $\frac{kJ}{s}$

As mentioned earlier that heat supplied has to be multiplied with the boiler efficiency and radiation loss to arrive at the actual heat supplied which can be given as,

Actual heat supplied = 0.74 x 30530.20 = 22592.35 $\frac{kJ}{s}$

= 22592.35 – 0.02 x 22592.35 = 22140.50 kJ/s $\frac{kJ}{s}$ >>>> 84.62 $\frac{kJ}{s}$

From the above calculations it is cleared that the autoclave has already enough excess steam energy which can be utilized for increased capacity and hence there is no need of excess heat energy or to change the process parameters i.e. pressure and temperature.

4.15 Software for arrangement of rubber rolls

It is very difficult to decide rubber roll arrangement when there is vast difference in sizes and uncertain order pattern. Hence a software has been developed to suggest the roll arrangement and it works on the concept of filling i.e. it swipes the entire cross-sectional area which is previously defined and within that items that are to be fit are identified and oriented ensuring all the constraints should be satisfied. Generally there are three types of packing and/or orientation algorithm which are 1D packing, 2D packing and 3D packing out of which 2D packing algorithm has been selected here as only length and diameter of rubber roll are to be formulated. Proposed method takes into consideration only the two parameters which are length and diameter of the rubber roll and based on that within the fixed dimensions of the trolley it tries to arrange number of rolls satisfying all the constraints introduced in the program.

Following constraints must be satisfied by the algorithm:-

1. Length of the rubber roll should not be more than the length of the trolley (7300 mm) so that it can be accommodate within the trolley's dimensions.

2. Width/Diameter of the rubber coated rolls should be less than the width of the trolley (1068 mm) so that the rubber roll do not hang outside the trolley's dimensions.

3. Any two rubber rolls should not overlap each other by any means.

4. Two rubber rolls whether arranged side by side or length wise they must not interfere each other.

5. The rubber rolls which are heavier to be placed at the bottom.
6. The rubber rolls must be arranged in such a way that the length of the rubber rolls remains parallel to the length of the trolley.
7. The placement of rubber rolls should be start always from the back of trolley and heavier rolls to be placed first and then swipe across the cross-section of the entire trolley for next rolls.
8. There is a gap of 20 mm, 50 mm and 10 mm between the base of a trolley and lower side rolls, between each upper and lower roll, and side by side(both width and length) between each and every rubber roll respectively.

The trolley's dimension i.e. length , width and maximum possible height up to which rolls can be arranged are defined in the program such that the program takes it as a rectangular box and then tries to put the cylindrical rubber rolls inside it with imposed constraints. The algorithm of software is shown in fig.4.39 as flow chart.

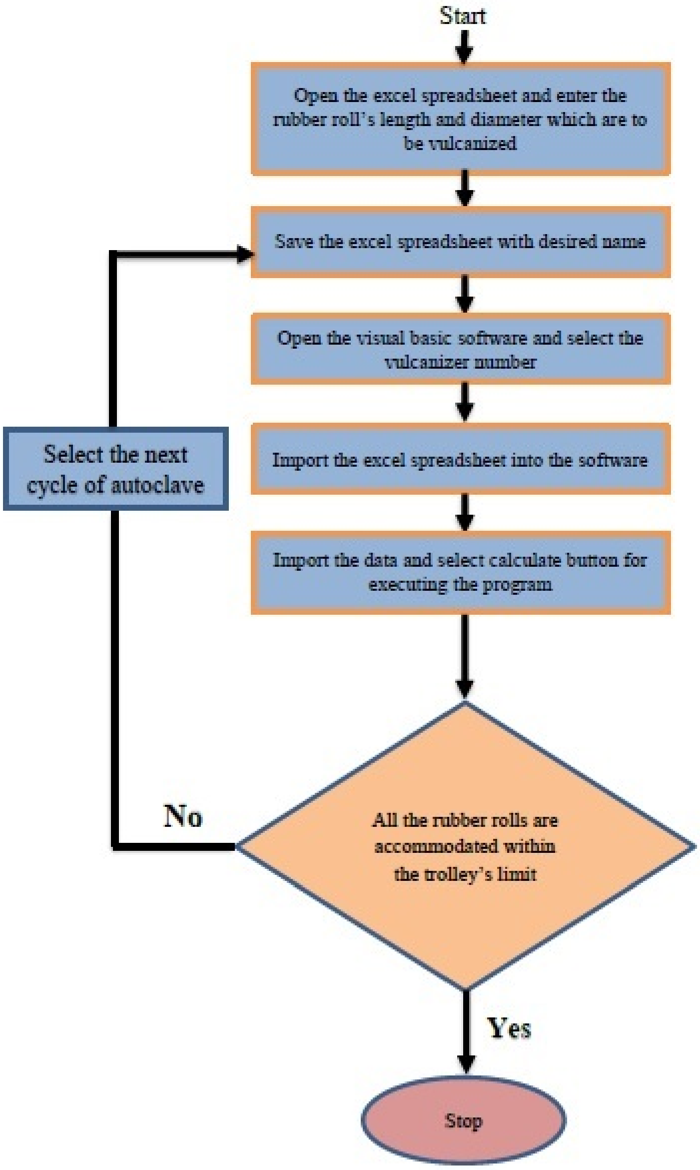

Figure 4.39: Flow chart of the software developed for arrangement of rolls on trolley

The dimensions of the trolley are assumed to be fixed and are assigned in similar manner into a program and various constraints are given as discussed above so program checks for the best possibility of the roll arrangement and present it graphically so that supervisor can see the loading of rolls and may guide the operator accordingly.

As the software developed using VBA has some graphics limitations and also the trolley's dimensions are too big to fit it into a screen the rubber rolls may seem like they are interfering but actually they are not as there is a gap of 10 mm side by side(widthwise and lengthwise) between each and every rolls. The software also shows the percentage utilization of space from the total volume of rubber roll placed on the trolley and the volume of trolley so that for each cycle of operation one could have an idea about the efficiency of autoclave.

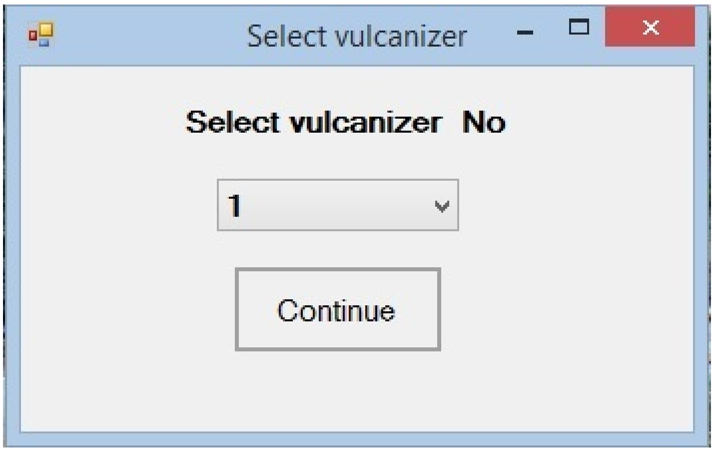

Figure 4.40: Option for selecting autoclave

Sr. No.	Length (mm)	Diameter (mm)
1	1300	254
2	1800	250
3	2000	222
4	1800	253
5	1800	263
6	1810	308
7	1800	253
8	2005	252
9	2450	318
10	2460	310
11	1980	314
12	1960	399
13	1525	266
14	1830	233
15	1830	230
16	2000	223
17	1525	264
18	1525	263
19	1800	249
20	2180	211
21	1815	310
22	2205	252
23	1800	310

Figure 4.41: Excel spreadsheet for inserting rubber roll data

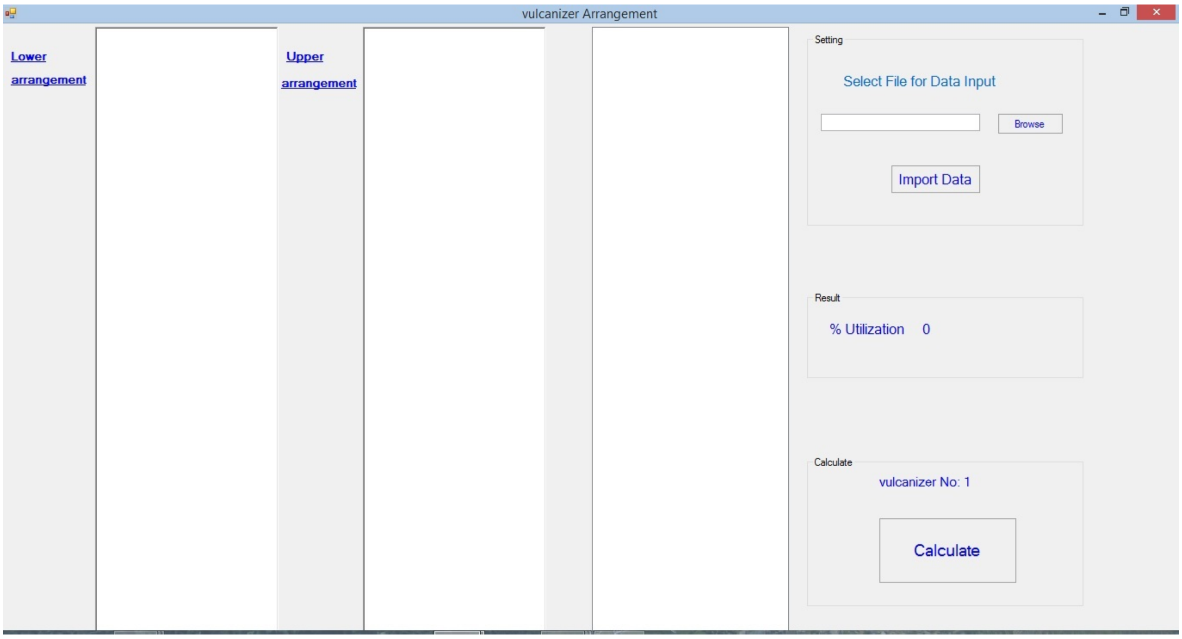

Figure 4.42: General UI of the software

The software developed appears when started as shown in fig.4.40 which facilitates for selecting the autoclave to be used for processing of rubber rolls. The rubber roll's data i.e. the roll diameter and roll length are to be inserted in excel spreadsheet as shown in fig.4.41 and then save the excel file with desired name. Figure4.42 shows the user interface of the software in which there is a provision of importing the excel file previously prepared and then by pressing the calculate button the algorithm checks the excel file for roll data and gives the roll arrangement such that it will satisfied all the constraint and ensures that max. space has been utilized as shown in figure4.43 and figure4.44 respectively.

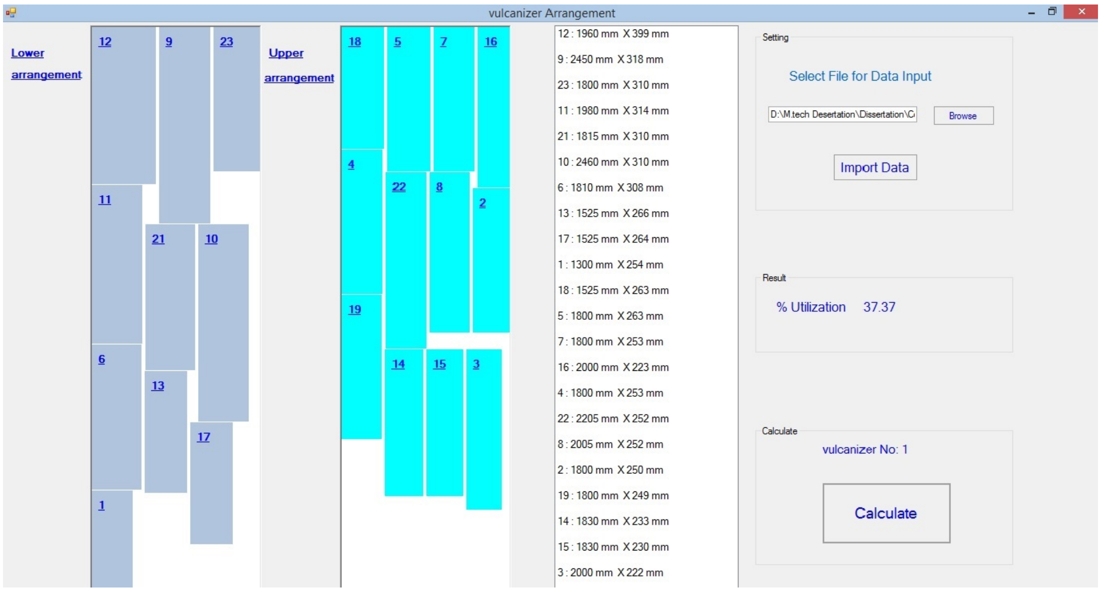

Figure 4.43: Software showing the rubber roll loading arrangement with percentage utilization of space

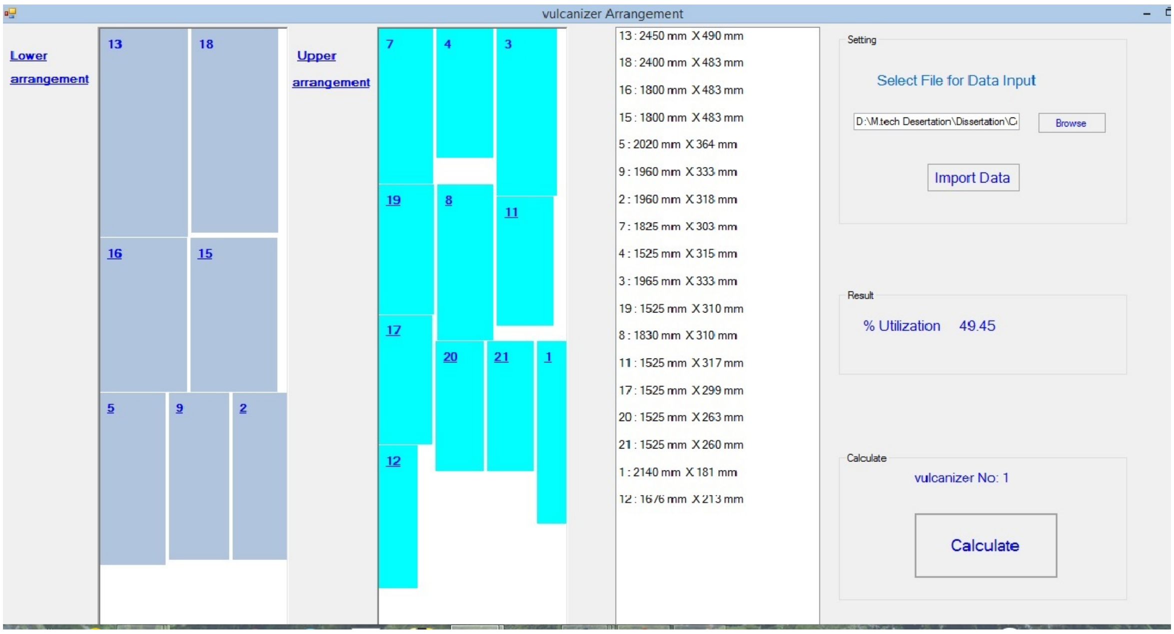

Figure 4.44: Software showing the rubber roll loading arrangement with percentage utilization of space

With the help of the developed software the arrangement of rubber rolls can be visualized easily as shown in figure4.43 and figure4.44 so that there will not be any confusion on the operator's end that which rolls are to be placed where so that maximum rolls can be accommodated within the trolley in a single set up which will save the trial and error method of loading and planning time as well.

4.16 Bill of material for the modular beam column arrangement for trolley structure

The proposed modular beam column arrangement has been conceptually designed and validated using FEA package and it was found to be within safe limits of material's strength. After that in order to fabricate the same its quotation was presented by the company's regular fabricator and based on that bill of material is generated which is as shown in table below.

From the BOM it can be seen that the total cost of proposed structure is nearly 15000 Rupees. The average cost of one vulcanization cycle is around 1100 Rupees hence the total cost of vulcanizing the rubber rolls of four months which took 56 cycles is around 62000 Rupees. The same no. of rubber rolls could be vulcanized with the help of modular beam-column arrangement within nearly the half time as compared to the current method and also the cost of processing in total will be within 31000 Rupees which is almost half and the time to do so will be approximately half also which is a huge benefit for industries.

Table 4.2: Bill of material for modular beam column arrangement

SR No.	Component	Description	Qty.	Price(Rs.)
1.	Support beam with holes	ISMB 150 beam 1.168 m long of structural steel	5	4125
2.	Adjustable column with holes	RHS 96x48x4 mm of 0.4 m long and RHS 81x40x4 mm of 0.4 m long of structural steel	14 each	6600
3.	U clamp	16 mm dia. rod of M.S.	15	300
4.	Guide plate	7300x10x60 mm plate of M.S.	2	3300
5.	Bolts for clamping	M16 bolt corrosion resistant7300x10x60 mm plate of M.S	16	400
			Total	14725

Chapter 5

Results and discussion

5.1 Conclusion

- To improve the productivity of the autoclave equipment a modular structure has been conceptually designed and its finite element analysis has been carried out and it was found that the design is safe.

- A model of modular structure with rubber rolls arrangement has been developed. From the model we can see that the productivity of autoclave equipment has been almost improved by 100%. This will decrease the production cost and will also help to free up the floor space in vicinity to the autoclave.

- Proposed design facilitated to vulcanize the same number of rubber rolls with nearly half of the operating cycles which cut down the fuel cost and time to the half as compared to earlier practice.

- It cost around 1100 Rs. to process one cycle of vulcanization and as discussed earlier with the help of modular beam-column arrangement the same no. of rubber rolls can be vulcanized within 28 cycles as compared to earlier practise which can save around 93000 Rs./year and nearly a month of time which could be utilized for more production.

5.2 Future work

- Fabrication of the proposed modular arrangement and its costing.
- Implementation of proposed trolley structure and from that evaluating the improved productivity of autoclave as compared to previous productivity.
- To evaluate the effect of improved productivity of the proposed trolley structure on the overall productivity of the company.

Bibliography

[1] Gangping Shu , Baofeng Zheng , Lianchun Xin , A new design method for steel columns subjected to flexural buckling, Thin-Walled Structures 83 (2014) 43-51.

[2] Ou Zhao, Barbara Rossi, Leroy Gardner, Ben Young , Behaviour of structural stainless steel cross-sections under combined loading – Part I: Experimental study, Engineering Structures 89 (2015) 236–246.

[3] Merih Kucukler, Leroy Gardner, Lorenzo Macorini, Flexural–torsional buckling assessment of steel beam–columns through a stiffness reduction method, Engineering Structures 101 (2015) 662–676.

[4] M. Macdonald, M. A. Heiyantuduwa, A design rule for web crippling of cold-formed steel lipped channel beams based on nonlinear FEA, Thin-Walled Structures 53(2012) 123-130.

[5] Alexandre Landesmann, Dinar Camotim, Distortional failure and DSM design of cold-formed steel lipped channel beams under elevated temperatures, Thin-Walled Structures 98 (2016) 75–93.

[6] Mei Liu, Lulu Zhang, Peijun Wang, Yicun Chang, Experimental investigation on local buckling behaviors of stiffened closed-section thin-walled aluminum alloy columns under compression, Thin-Walled Structures 94 (2015) 188–198.

[7] Shanmuganathan Gunalan, Mahen Mahendran, Improved design rules for fixed ended cold-formed steel columns subject to flexural–torsional buckling, Thin-Walled Structures 73 (2013) 1–17.

[8] M. Macdonald, M.A. Heiyantuduwa, J. Rhodes, Recent developments in the design of cold-formed steel members and structures, Thin-Walled Structures 46 (2008) 1047–1053.

[9] Baofeng Zheng, Xia Hua, Ganping Shu, Tests of cold-formed and welded stainless steel beam-columns, Journal of Constructional Steel Research 111 (2015) 1–10.

[10] Abdelkader Saoula, Sid Ahmed Meftah, Foudil Mohri, El Mostafa Daya, Lateral buckling of box beam elements under combined axial and bending loads, Journal of Constructional Steel Research 116 (2016) 141–155.

[11] Jia-Lin Ma, Tak-Ming Chan, Ben Young, Material properties and residual stresses of cold-formed high strength steel hollow sections, Journal of Constructional Steel Research 109 (2015) 152–165.

[12] Production data from Lathia's Production planning and material management server.

[13] V. B. Bhandari, Machine design data book, published by McGraw Hill Education Private Limited, Third reprint 2015, U. C. Jindal, Introduction to strength of material, Published by Galgotia Publications Pvt. Ltd. , Revised edition 2007.

[14] Design manual for designing steel structures according to new IS: 800 for IS: 800-2007.

[15] Design manual for designing steel structures according to new IS: 800 for IS: 800-2007, Elastic analysis equation for beams and frames.

[16] Bureau of Indian Standards, Handbook of typified design for structure with steel roof trusses (with or without cranes) based on IS codes.

[17] Ferdinand P. Beer, E. Russel Johnston, Jr. John T. DeWolf, Chapter 10 , Mechanics of Materials, published by Tata McGraw Hill Education Private Limited, Third edition.

[18] Joseph E. Shigley, Mechanical engineering design, Published by Tata McGraw Hill Education Private Limited, Eighth edition in SI units.

[19] Thermal properties of metals, Engineers Edge`http://www.engineersedge.com/properties_of_metals.htm`.