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Design and Analysis of Coil Ramp, Coil Car and Coil Mandrel Systems

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ABSTRACT: This research paper presents a comprehensive study on the design and analysis of coil handling equipment, including Coil Ramps, Coil Cars, Coil Mandrel, and Uncoiler. The study delves into the Machine Design considerations, performance requirements, and operational efficiencies of each component. Utilizing CATIA CAD software for design, the paper outlines the iterative design process and integrates ANSYS simulation to analyse structural integrity, Hydraulic Pressures and Body Stress factors. The aim is to optimize the equipment's Mechanical performance, ensuring reliability and long-term efficiency in coil handling operations. This research provides valuable insights and recommendations for enhancing the design and functionality of coil handling systems in Steel industrial applications.

I. INTRODUCTION

The efficient handling of coils is pivotal to the operations at TATA STEEL Ltd, where the seamless management of heavy coils directly influences production efficiency and equipment durability. This research paper delves into the design and analysis of key coil handling systems integral to TATA STEEL's operations, including Coil Ramps, Coil Cars, and Coil Mandrel. These components are meticulously engineered to meet the rigorous demands of steel production, ensuring optimal performance and reliability. Utilizing advanced engineering tools such as CATIA CAD for design and ANSYS for structural analysis, this study includes a detailed V-bed angle analysis to assess stress concentrations across different areas of the coil handling equipment.

Machine design formulation is done for each part of the coil ramp, car and mandrel, with a focus on mechanical integrity and operational efficiency. The Goodman and Soderberg theories are applied to calculate the factor of safety (FOS) for the

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entire body of the equipment, providing a robust evaluation of its durability under cyclic loading conditions. By applying iterative design processes, simulations, and FOS calculations, the paper aims to enhance the mechanical performance and longevity of these systems, offering valuable insights and recommendations to bolster coil handling operations within the steel industry.

Coil ramp is a mechanical device used for placing the coil in stand by position at the steel pipe on-line production. Coil ramp is based on hydraulic piston cylinder mechanism. The couple of coil ramps are arranged to provide desired gap between them to hold on the entire coil depending upon width of the coil. The design consists of a ramp slider, ramp base and rod-shaped bar type piston.

- **Ramp base**

Ramp base is designed by combination of square ribs and trapezoidal ribs. These ribs provide rigid support and endurance to resist bidirectional deformation of base in 2D base plane. The square ribs are extruded between the 2 plates to ensure minimal shear deformation of part. The frequency of this square ribs is increased at the position of side end of base where slider slides over and carries load of Coil on v shaped bed.

The ramp base is bolted at the base plate to lock and fix base plate with ground. In top of the upper plate of base cylinder with hydraulic arrangement is mounted. The cylinder is balanced with thick plates at the cross-section positions of start middle and end. The middle thick plate is fixed with 2 similar thick plates parallel to cylinder piston position. These two plates are welded with the upper base plate of ramp base. The cylinder is surrounded by 4 round bars and passing throughout all square flanged joint plates. These bars resist deformation of the cylinder due to excessive hydraulic pressure.

At the ramp base, on both sides of the slider, inverted L shaped slide locking mechanism is designed to ensure the one directional translation of slider over base. The position of slider entirely locked by bolt screwing at the slider edges and upper portion of base inverted L shaped. This modification can be done when specific position of slider is required for specific width of Coil carriage. This bolts joints provide extra resistance to angular and torsional deformation.

- **Ramp Slider**

The ramp slider consist of a v shaped bed on which coil is settled. This slider has support walls adjacent to the sides of bed providing resistance to shear deformation of bed due to coil load . The bending of this wall due to such force is resisted by trapezoidal ribs along the surface of wall. This causes distribution of stress though this all parts reducing the stress concentration at the contact of the edge sliding.

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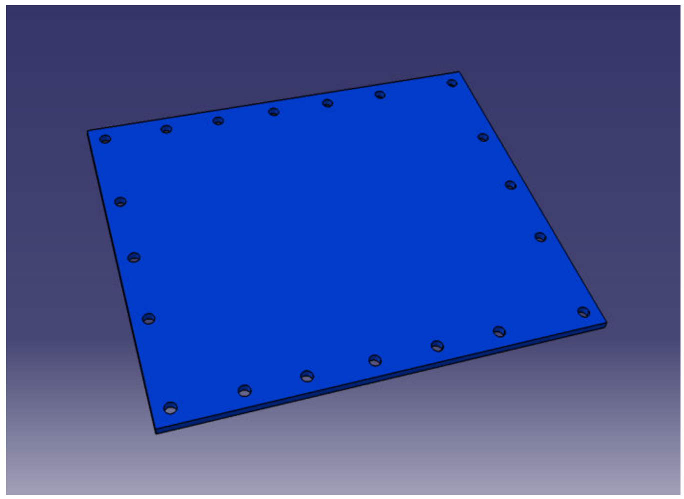
II. MACHINE DESIGN CALCULATIONS AND CATIA MODELS

Coil Ramp Parts Design

AutoCad Bolt Distances Design



Bolted Base Plate

Bolts Design

W=300000 N, $S_{yt} = 400\ N/mm^2$

n=20, $N_{fos} = 2$

1) Primary Shear Force

$F_p = W/n = 300000/20 = 15000N$

2) Secondary Shear Force

$e = 600 + 965 = 1565mm$

$W.e = F_{s1}.l_1 + F_{s2}.l_2 + \cdots + F_{s20}.l_{20}$

$= wl_1^2 + wl_2^2 + \cdots + wl_{20}^2$

$$w = \frac{W.e}{l_1^2 + l_2^2 + \cdots l_{20}^2}$$

$$= \frac{300000 \times 1565}{1151.44^2 + 943^2 + \cdots + 944.8^2}$$

$$w = \frac{300000 \times 1565}{17332379.68} = 27.08N/mm$$

Total load is maximum on the bolts at the right side of central point Equivalent Resultant Secondary Shear force per bolt is calculated by taking resultant of shear force at each bolt by adding sine and cosine components of all of them.

$F_s = 133811.84\ ,\theta = 41.75°$

3) Resultant Shear force

$$F_{S_R} = \sqrt{F_p^2 + F_s^2 + 2F_pF_s\cos\theta} =$$

$$\sqrt{15000^2 + 133811.84^2 + 2 \times 15000 \times 133811.84 \times \cos(41.75)}$$

$F_{S_R} = 145346.30\ N$

4) Bolt Size

$$\tau_{all} = \frac{0.5 \times S_{yt}}{N_f} = \frac{0.5 \times 400}{2.5} = 80N/mm^2$$

$\tau = \frac{F}{A_c} = 80\ , A_c = 1816.82\ mm^2$

$d = 48.09\ mm \approx 50\ mm$

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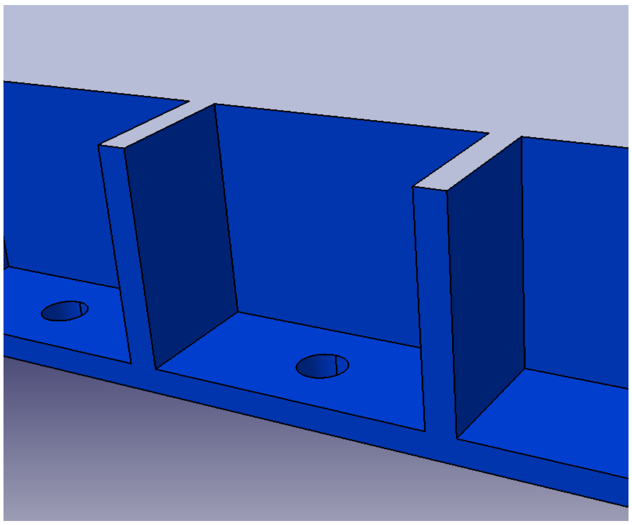


Square Rib Design

Assuming Thickness of rib 60% of the wall thickness.

$t = \frac{60}{100} \times 50 = 30mm$

$\sigma_c = 250\ N/mm^2, \quad \frac{\sigma_c}{N_f} = \frac{F}{A}$

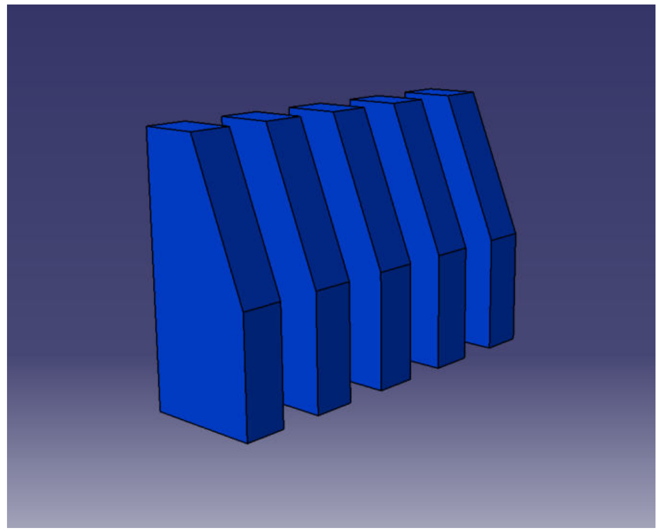
$\frac{300000}{30 \times l} = \frac{280}{4}$

$l = 142.85mm \approx 145\ mm$

$\tau = 0.75 \times \sigma_c, \quad \frac{\tau}{N_f} = \frac{F}{A}$

$\frac{300000}{30 \times h} = \frac{0.75 \times 250}{4.5}$ …. (Shear Surface)

$h = 240mm \approx 250mm$



Trapezoidal Rib Design

Assuming Thickness of rib 60% of the wall thickness

$t = \frac{60}{100} \times 50 = 30mm$

$\tau = 0.75 \times \sigma_c \quad \frac{\tau}{N_f} = \frac{F}{A} = \frac{100000}{b \times t} = \frac{0.75 \times 250}{4}$

$b = 71.11\ mm \approx 80\ mm$

$h = 180$ mm is constrained due to supporting wall height

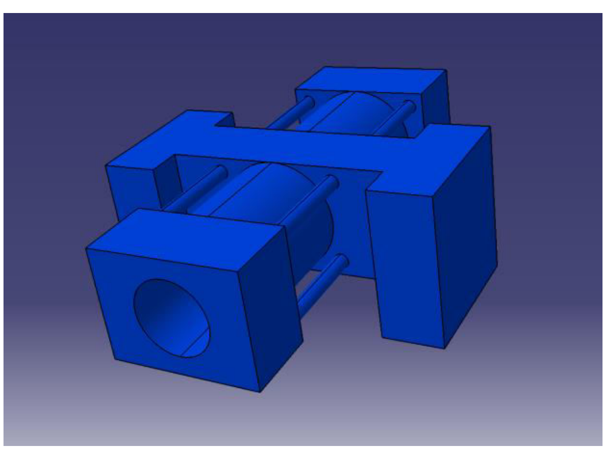
$\sigma_b = 450\ N/mm^2, \quad \sigma_b = \frac{M_{max} \cdot y}{I_z}, \quad y = 90\ mm,$

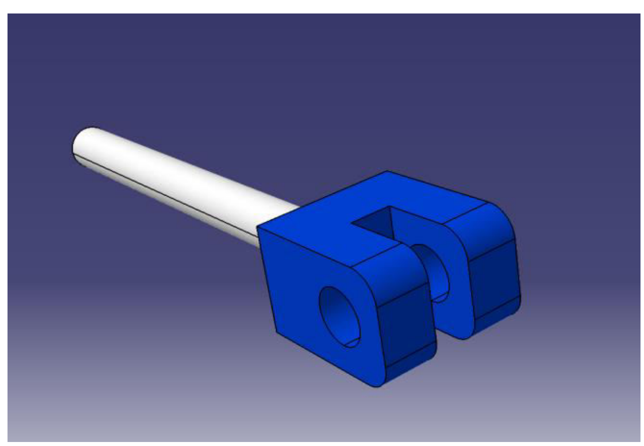
$M_{max} = 27 \times 10^6\ N.mm$

$I_z = \frac{1}{12} \cdot \left(\frac{1}{2} \cdot (a + b) \cdot h\right) \cdot \left(h^2 + \frac{(a - b)^2}{4}\right)$

Substituting value of b in Equation to get value of a

$a = 34.6\ mm \approx 40\ mm$





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Cylinder Design

D_0 − Outside Diameter of Cylinder

D − Inside Diameter of Cylinder

As per ISO 6162-2Standard followed by TATA STEEL,

Considering P − Maximum Pressure in Hydraulic $= 63\ N/mm^2$

As per EN19 Steel Standard of TATA STEEL,

Considering Poisson Ratio $= 1/m = 0.33$

σ_1 − Apparent Longitudinal stress in cylinder $= 50\ N/mm^2$

$$\sigma_1 = \frac{D^2P}{[(D_o)^2 - D^2]} = \frac{D^2 \times 63}{{D_o}^2 - D^2} = 50\ N/mm^2$$

$$50D_0^2 - 13D^2 = 0 \ldots\ldots(1)$$

σ_c − Apparent Circumferential Stress in cylinder $= 125\ N/mm^2$

$$\sigma_c = \frac{Dp}{2t} = \frac{D \times 63}{2 \times 25} = 125\ N/mm^2$$

$D = 99.20\ mm \approx 100mm$

Substituting D in Eq^n 1

$D_0 = 150\ mm$

σ_{nf} − Net Longitudinal stress

$$\sigma_{nf} = \sigma_1 - [\sigma_c/m] = 50.4 - 126 \times 0.33 = 8.82\ N/mm^2$$

σ_{nc} − Net Circumferential stress

$$\sigma_{nc} = \sigma_c - [\sigma_1/m] = 126 - 50.4 \times 0.33 = 109.368\ N/mm^2$$

From PSG 5.137, for D=100mm Clearance C=2.4 mm is selected.

Dry Liner Thickness $(t_1) = 0.035 \times D = 0.035 \times 100 = 3.5\ mm$

$$\text{Cylinder Head Thickness } (t_h) = D\sqrt{\frac{cP}{\sigma_c}}$$

$$= 100 \times \sqrt{\frac{0.1 \times 63}{126}} = 22.36mm \approx 23mm$$

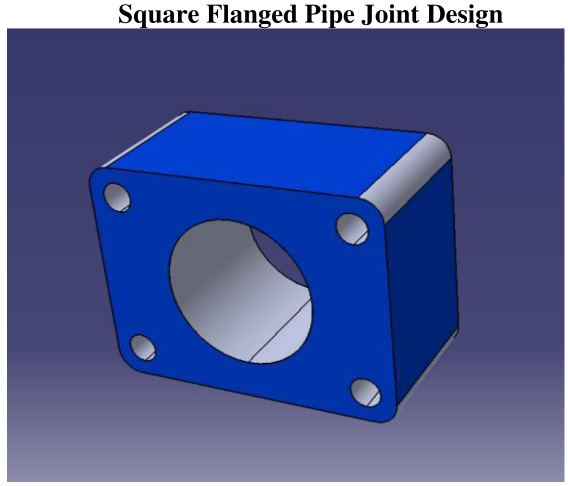
Pitch Circle Diameter $(D_p) = D + 3d = 100 + 3 \times 20 = 160mm$.......[1]

Piston Design

$$t_H = \sqrt{\frac{3pD_0^2}{16\sigma_t}} = \sqrt{\frac{3 \times 63 \times 150^2}{16 \times 380}}$$

$t_H = 27.55\ mm \approx 30\ mm$

Square Flanged Pipe Joint Design



According to Lames Equation,

$\sigma_t = 415\ N/mm^2$ and $P = 63\ N/mm^2$

$$\text{Thickness of Flange (t)} = R\left[1 - \sqrt{\frac{\sigma_t+P}{\sigma_t-P}}\right] =$$

$$50\left[1 - \sqrt{(415+63)(415-63)}\right] = 8.26mm\text{.......[1]}$$

$$\text{Force required to seperate flange (F)} = \frac{\pi}{4}(D_1)^2P = \frac{\pi}{4} \times 121.32^2 \times 63 = 728274.71\ N/mm^2$$

Force on each Bolt $(F_1) = F/n = 728274.71/4 = 182068.67\ N/mm^2$

Min. Length of diagonal for square (L)=$D + 2t + 2d = 100 + 2 \times 8.26 + 2 \times 2.4 = 190mm$

Side of Square flange between centre to centre of bolt holes $(L_1) = L/\sqrt{2} = 190/\sqrt{2} = 134.35mm$

Side of flange for nuts &bolts without overhang $(L_2) = L_1 + 2d = 134.35 + 2 \times 20 = 174.35mm$

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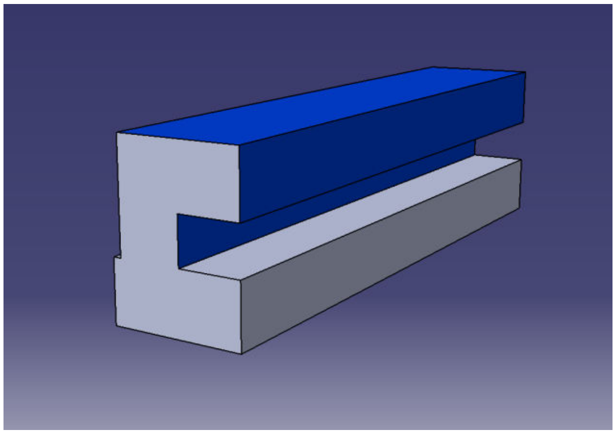
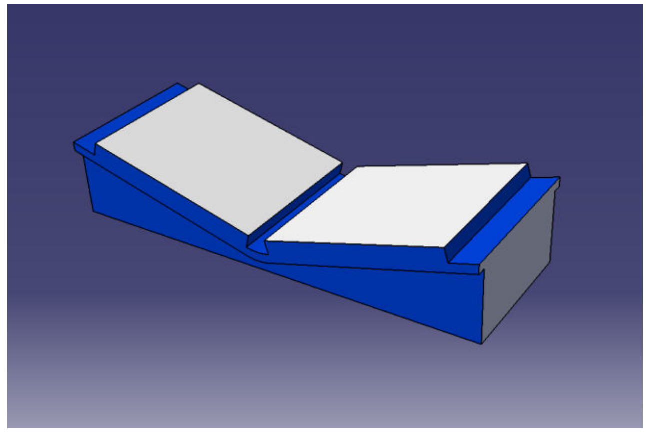


C Section Design

$$\bar{y} = \frac{A_1y_1 + A_2y_2 + A_3y_3}{A_1 + A_2 + A_3}$$
$$= \frac{t \times 160 \times 80 + t \times 60 \times 120 + t \times 160 \times 80}{t \times 160 + t \times 60 + t \times 160}$$
$$= 86.31 \text{ mm}$$
$$I = I_A + I_B + I_C$$
$$I_A = I_C = \frac{t \times 160^3}{12} + t \times 160 \times (86.31 - 80)$$
$$I_B = \frac{60 \times t^3}{12} + t \times 60\ (120 - 86.31)$$
$$\sigma_{max} = 310 \text{ N/mm}^2$$
$$C = 86.31 \text{ mm}$$
$$M_B = 670 \times 300000 \text{ N.mm}$$

Using Equation $\sigma_{max} = \frac{M_B \times C}{I}$, Thickness is

$$t = 78.03 \text{ mm} \approx 80 \text{ mm}$$



V-Bed Design

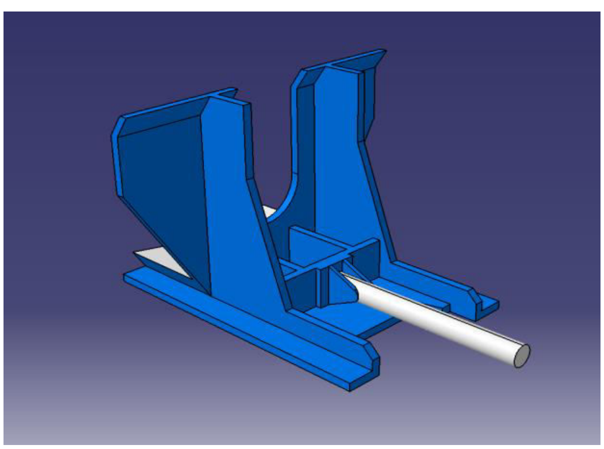
$$\sigma_c = 250 \text{ N/mm}^2, \quad \frac{\sigma_c}{N_f} = \frac{F}{A}$$
$$\frac{300000}{400 \times w} = \frac{0.5 \times 200}{4.5}$$
$$w = 33.75 \text{ mm} \approx 40 \text{ mm}$$

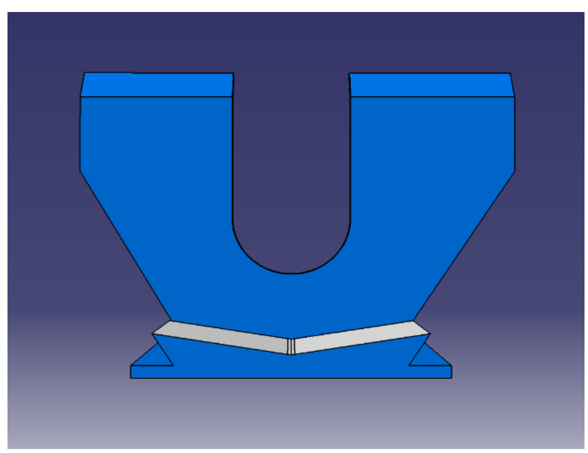
$h = 400\cos\theta = 400\cos 15 = 386$ mm where θ is the angle at which beds are inclined in V position

$$\sigma_b = 450 \text{ N/mm}^2, \quad \sigma_b = \frac{M_{max} \cdot y}{I_z}, \quad y = \frac{b + 2a}{3(a + b)} \text{ mm},$$
$$M_{max} = 50 \times 10^6 \text{ N.mm}, \quad a = 180 \text{ mm}$$
$$I_{zz} = \frac{h}{3}(a^3 + b^3 + a^2b + b^2a)$$

Substituting value of a in Equation to get value of b

$$b = 75 \text{ mm} \approx 80 \text{ mm}$$





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Ramp Wall Design

The instantaneous stress σ in the plate due to the impact

$\sigma = \frac{F}{A} = \frac{mv}{At}$ m = 30000kg, E = 200Gpa, F = 100000N, $\sigma = 2 \times 10^5 N/(mm^2\ sec)$, v = 100mm/s

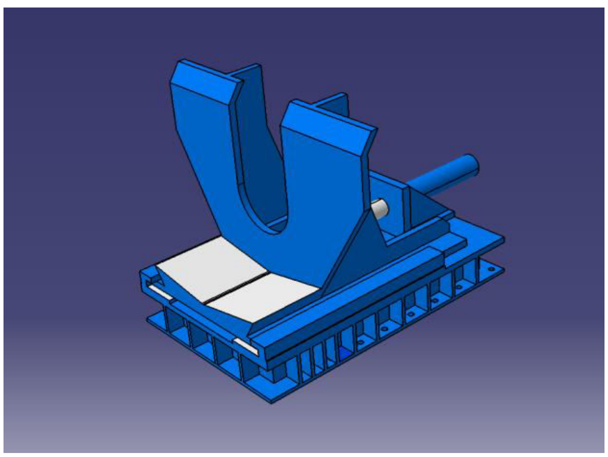
Time duration $t = 2\sqrt{\frac{m\delta}{k}}$, spring stiffness of the plate $k = \frac{AE}{\delta}$

The maximum stress occurs when the plate deforms by δ and the impact force F is maximum. The impact force F is related to the kinetic energy of the impacting object

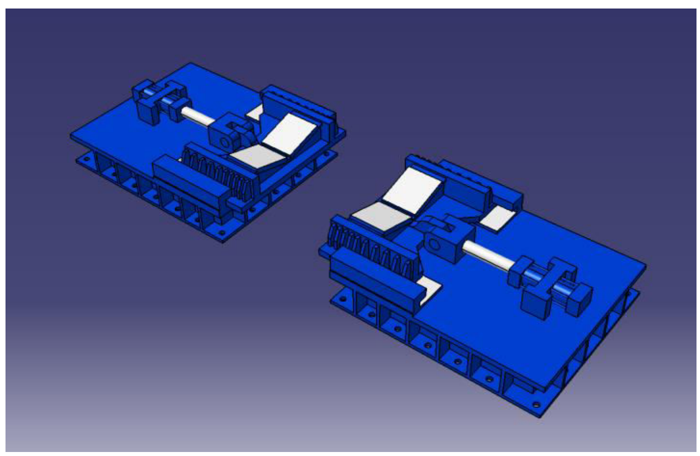
$F = \frac{1}{2}mv^2 = \frac{1}{2}\sigma A\delta$

Substituting this into the stress equation

$\sigma^2 = \frac{\sqrt{AmE}\,v}{4F}$ Width of Plate = 2200 mm Height of plate = 1939.39 mm ≈ 2000mm



Coil Ramp for Low Width Coils



Coil Ramp for High Width Coils

Calculation of FOS of Coil Ramp using Gerber, Goodman, Soderberg Method

Material Selected : 40 Ni 1 Cr 1 Mo 15 Alloy Steel $\sigma_{max} = 7.57 \times 10^7 Pa$ $\sigma_{min} = 2.52 \times 10^7 Pa$

$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = 5.045 \times 10^7\ Pa$ $\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = 2.52 \times 10^7 Pa$ $\sigma_e = 0.5 \times \sigma_{ut} = 0.5 \times 841 \times 10^6\ Pa$

$\sigma_{yt} = 600 \times 10^6 Pa$ $k_f = 0.85$

Gerber Formula : $\frac{1}{F.O.S} = (\frac{\sigma_m}{\sigma_a})^2\ FOS + \frac{\sigma_a \cdot k_f}{\sigma_e}$ $\mathbf{FOS = 11.03}$

Goodman Formula : $\frac{1}{F.O.S} = \frac{\sigma_m}{\sigma_{ut}} + \frac{\sigma_a . K_f}{\sigma_e}$ $\mathbf{FOS = 8.339}$

Soderberg Formula : $\frac{1}{F.O.S} = \frac{\sigma_m}{\sigma_{yt}} + \frac{\sigma_a . K_f}{\sigma_e}$ $\mathbf{FOS = 7.406}$

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$$\mathbf{FOS_{AVG}} = \frac{11.03 + 8.339 + 7.406}{3} = \mathbf{8.925}$$

Coil Car Design

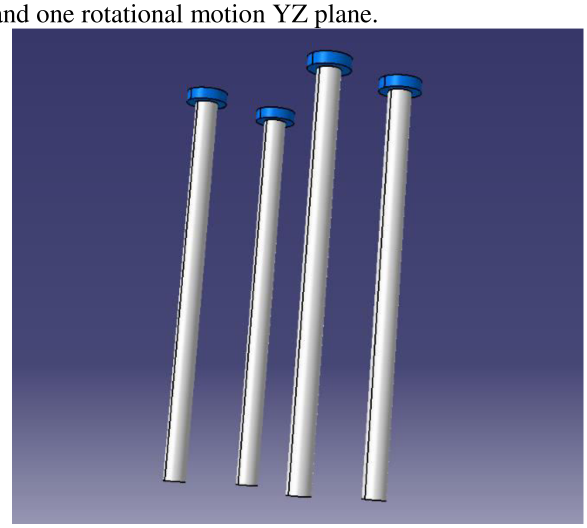
- **Coil base**

It consists of hydraulic cylinder piston mechanism at the bottom centre to provide thrust in the z direction. 4 hollow cylindrical guide bar supports provide support to guide bars of Coil bed which have axial degree of freedom in z direction. The guide bars resist the bending stress at the edge sides of bed thereby stabilize entire body. The trapezoidal ribs are provided for contact support of all guide bars at the top of the base. These guide bar supports provide a static structural stationary support when coil load is not subjected to shaft.

- **Coil bed**

The bed consists of 4 rigid cylindrical solid guide bars at the bottom which are inserted through hollow guide bar supports providing multidirectional support for bed. The bed consists of two rollers at the top which provide rolling motion to coil for adjusting as per opening of Coil sheet part. At the bottom edge of bed shaft arrangement is made to provide coil car transmissibility from one point to another. Cylindrical roller bearings provide more contact over shaft indirectly over bolt width for shaft causing absorption of stress in bearing portion itself.

Therefore, this coil car consists of (LLR-Linear Linear Rotational) degree of freedom, 2 linear transmissions in XZ plane and one rotational motion YZ plane.

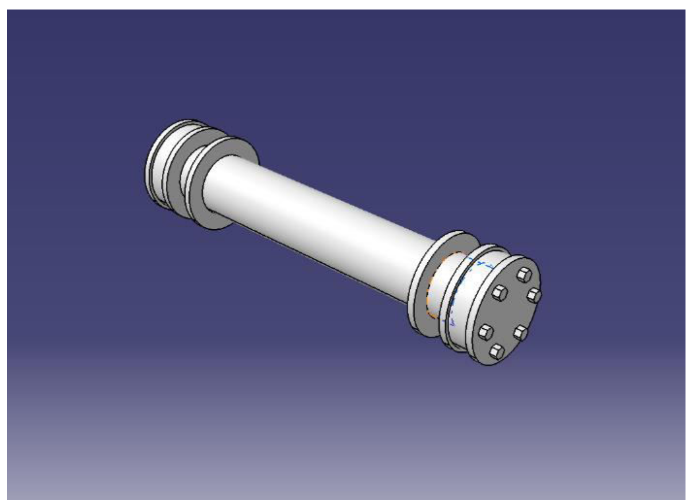


Guide Bar Design

Guide Bar is subjected to bending

$$\sigma_{max} = \frac{32M_b}{\pi d_o^3} = \frac{32 \times 7500}{\pi \times d_i^3} = 400\ \mathrm{N/mm^2}$$

$d_i = 96.32\ \mathrm{mm} = 100\ \mathrm{mm}$



Shaft Design

$$\tau_{max} = \frac{75}{5} = \frac{16M_t}{\pi(d_0^3)}\ (1)$$

$$\frac{\sigma_{bmax}}{N_f} = \frac{80}{5} = \frac{32M_b}{\pi(d_0^3)}\ (2)$$

$$\alpha = \frac{1}{1 - 0.0044 \times \left(\frac{l}{r}\right)} = \frac{1}{1 - 0.0044 \times \left(\frac{604}{75}\right)}$$

$$= 1.0368 \text{ for } \frac{l}{r} < 115$$

Substituting values of α, M_b & M_t from Eqn 1 and 2 in follov

$d_0^3 = \frac{16}{\pi[\tau]}\sqrt{[K_bM_b]^2 + (K_tM_t)^2}$......[3]

Considering values of k_b& k_t as 2 and 1.5 for suddenly appl

$d_o = 148.77\ \mathrm{mm} \approx 150\ \mathrm{mm}$

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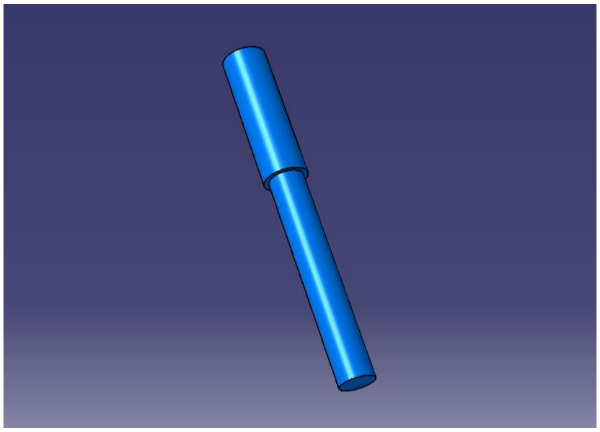


Hollow Cylindrical Guide Bar Support Design

Hollow Cylindrical support is subjected to Bending + Axial Loading

$$\alpha = \frac{1}{1 - 0.0044(\frac{1600}{100})} = 1.163$$

$$\tau = \frac{F}{\frac{\pi}{4} \times (d_o^2 - d_i^2)} = \frac{300000}{\frac{\pi}{4} \times (d_o^2 - 100^2)} \ (3)$$

$$d_o^3 = \frac{16}{\pi[\tau]\left\{1 - \left(\frac{d_i}{d_o}\right)^4\right\}}\left[K_b M_b + \alpha \frac{Pd_o}{8}\left(1 + \frac{d_i^2}{d_o^2}\right)\right]$$

Substituting Eqn 3 of τ in following eqn to get d_o

$$d_o^3 = \frac{16}{\pi \times \tau \times \left(1 - \left(\frac{100}{d_o}\right)^4\right)} \times \left[1.5 \times 75000 + 1.163 \times \frac{30 \times 10^4 d_o}{8}\left(1 + \frac{100^2}{d_o^2}\right)\right] \ ....[1]$$

$d_o = 137.88$ mm $= 140$ mm

$$n_\sigma = \frac{\sigma_{-1}}{K_t \beta_{size} \sigma_b} = 2$$

$\sigma_{-1} = 215 N/(mm^2)$ FOS $= 1.5$

Life of the shaft can be calculated from the following formula

$$N = (\frac{\sigma_a}{a})^{1/b}$$

$$a = \frac{(f S_{ut})^2}{S_e}, \ b = -\frac{1}{3}\log(\frac{f * S_{ut}}{S_e})$$

$$a = \frac{(0.85 \times 400)^2}{215} = 537.674$$

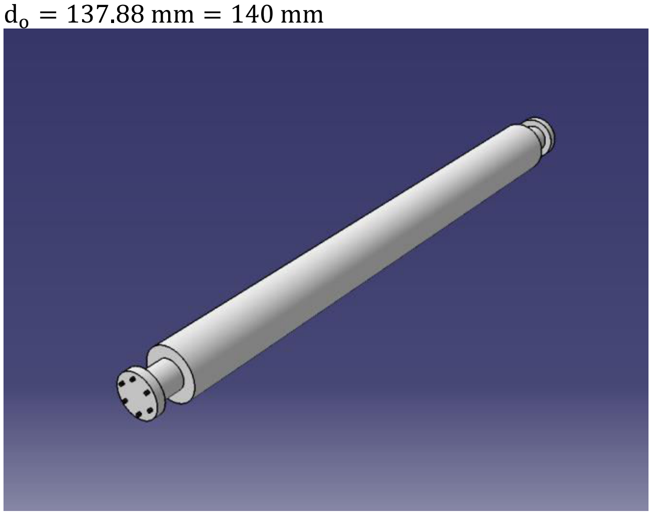
$$b = -\frac{1}{3}\log\left(\frac{0.85 \times 400}{\frac{215}{1.5}}\right) = -0.125$$

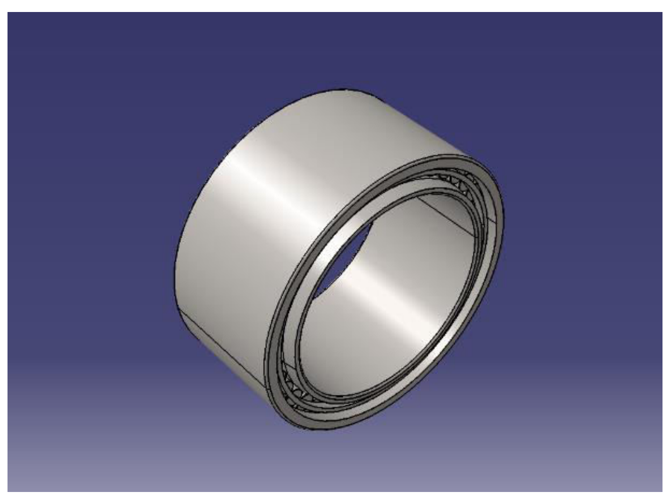
$$N = (\frac{\sigma_a}{a})^{1/b} = (\frac{100}{537.674})^{-1/0.125} = 6.98 \times 10^5$$

Endurance Strength for finite life

$$\sigma_{-1}^1 = \sigma_{-1}\left(\frac{10^6}{N}\right)^{0.09} = 215\left(\frac{10^6}{7 \times 10^5}\right) = 180.45 \ N/mm^2$$

.......[1]





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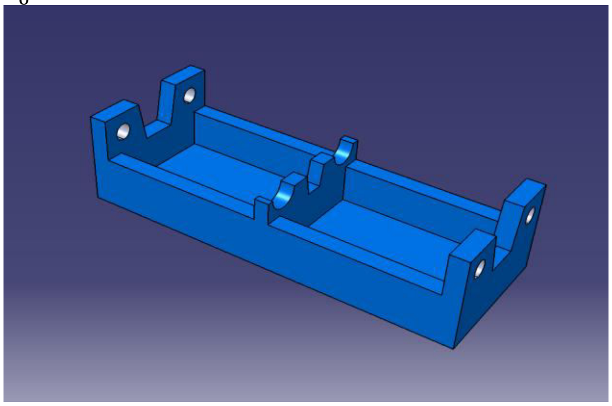
Roller Design

$$\sigma_{bmax} = \frac{32 M_b d_o}{\pi(d_o^4 - d_i^4)}$$

$$600 = \frac{32 \times 30 \times 10^4 \times 1930 \times d_o}{\pi(d_o^4 - 80^2)}$$

$d_o = 215.58\ \text{mm} \approx 220\ \text{mm}$



Car Roller Supporting Bed

Bearing Design

PSG 4.21, Cylindrical Roller Bearing with Inner Diameter 150 mm is selected for which

$d_o = 270\text{mm}$, Width (B) = 73mm,

Static Capacity(C_o) = 45000,

Dynamic Capacity (C) = 43000,

Maximum Permissible Speed (rpm) = 2500

$$C = \left[\frac{L}{L_{10}}\right]^{1/k} P$$

$$43000 = \left[\frac{L}{1}\right]^{\frac{3}{10}} \times 6.6 \times 10^4$$

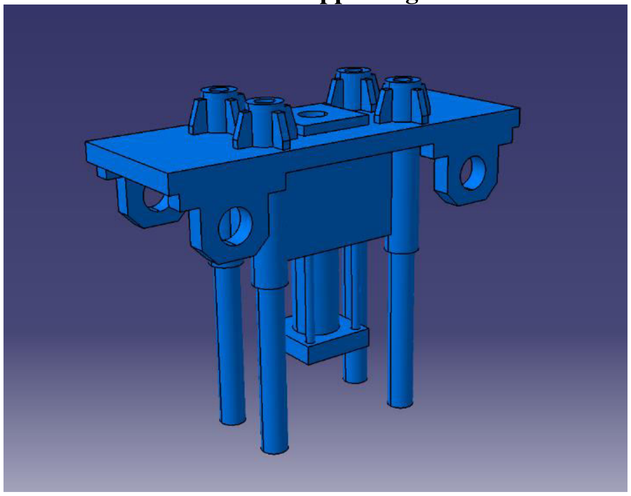
L = 0.234

Probability of Survival:

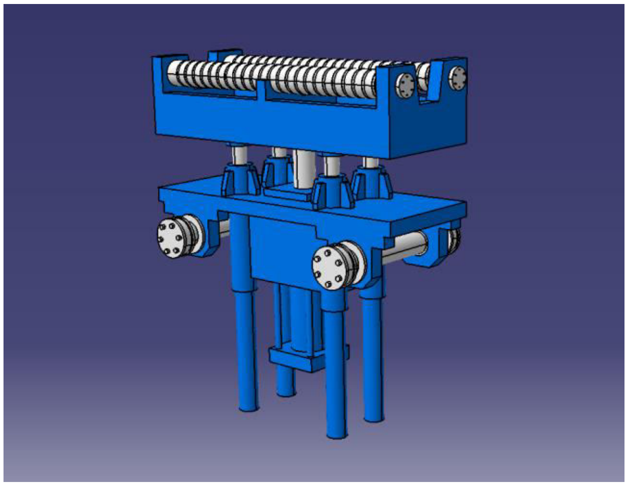
$$\frac{L}{L_{10}} = \left[\frac{\ln\left(\frac{1}{p}\right)}{\ln\left(\frac{1}{p_{10}}\right)}\right]^{\frac{1}{b}} \ldots [1] \qquad b = 1.17, \ln\left(\frac{1}{P_{10}}\right) = 0.1053$$

p = 0.9809

Therefore there is 98.09 % chance of survival for bearing



Coil Car Base



Coil Car Assembly

Calculation of FOS of Coil Car using Goodman, Soderberg Method

Material Selected : 35 Mn 2 Mo 28 Alloy Steel $\sigma_{max} = 3.79 \times 10^7 \text{Pa}$ $\sigma_{min} = 1.26 \times 10^7 \text{Pa}$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = 2.525 \times 10^7\ \text{Pa} \quad \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = 1.265 \times 10^7 \text{Pa}$$

$\sigma_e = 0.5 \times \sigma_{ut} = 0.5 \times 1150 \times 10^6$ Pa $\sigma_{yt} = 800 \times 10^6 \text{Pa}$ $k_f = 0.8$

Goodman Formula : $\frac{1}{F.O.S} = \frac{\sigma_m}{\sigma_{ut}} + \frac{\sigma_a}{\sigma_e}$ **FOS = 18.669**

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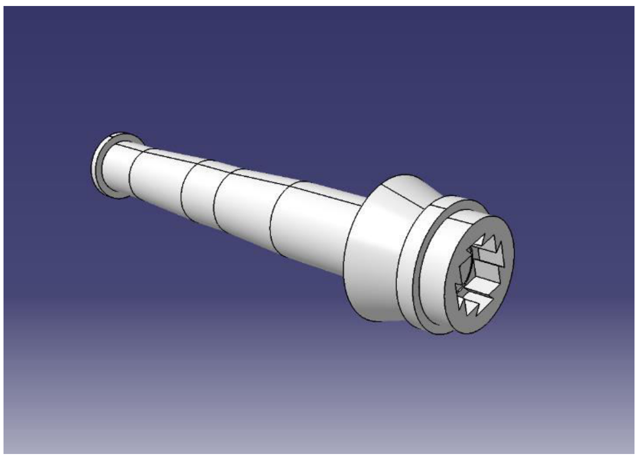
Soderberg Formula : $\frac{1}{F.O.S} = \frac{\sigma_m}{\sigma_{yt}} + \frac{\sigma_a . K_f}{\sigma_e}$ $\mathbf{FOS = 20.34}$

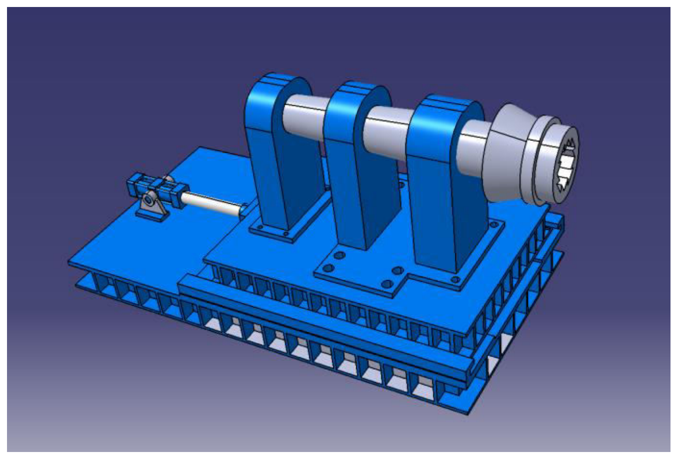
$$\mathbf{FOS_{AVG}} = \frac{18.669 + 20.34}{3} = \mathbf{19.504}$$

Coil Mandrel Design

A tapered mandrel is fitted inside the uncoiler mechanism with tapered side at the motor Operator and big diameter portion at the contact where mandrel is tightly expanded and fitted to other side for supporting the entire coil. The contact surface of mandrel with coil is chamfered and fillet with a specific design integration to reduce the stress concentrated deformation of mandrel and increase in endurance limit at centre.



Mandrel



Mandrel Uncoiler section

Shear Stress from Dead weight of Coil

$$\sigma = \frac{F}{A}$$

$$\text{Area}_{@\text{Failure}} = \frac{\pi}{4} \times (D^2 - d^2) = \frac{\pi}{4} \times (450^2 - 350^2)$$
$$= 62831.85 \text{ mm}^2$$

$$\sigma = \frac{30000 \times 9.81}{62831.85} = 4.683 \text{ N/mm}^2$$

Torsion Stress from Strip Tension

$$T = \frac{2000}{2} * 18000 = 18 \times 10^6 \text{N.mm}$$

$$J = \frac{\pi}{32}(D^4 - d^4) = \frac{\pi}{32}(450^4 - 350^4)$$
$$= 401 \times 10^7 \text{ mm}^4$$

$$c = \frac{450}{2} = 225\text{mm} \quad \tau = \frac{Tc}{J} = \frac{18 \times 10^6 \times 225}{401 \times 10^7} = 1.009 \text{ Mpa}$$

Bending Stress from Coil weight

$$\sigma_b = \frac{M}{I/C} \quad \frac{I}{C} = \frac{\pi d^3}{32} = \frac{\pi \times 450^3}{32} = 8.94 \times 10^6 \text{mm}^3$$

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Evaluating the life of the shaft

$$N = (\frac{\sigma_a}{a})^{1/b}$$

$$a = \frac{(fS_{ut})^2}{S_e}$$

$$a = \frac{(0.75 * 1500)^2}{750} = 2343.75$$

$$b = -\frac{1}{3}\log\left(\frac{f \times S_{ut}}{S_e}\right) = -\frac{1}{3}\log\left(\frac{0.75 \times 1500}{750}\right)$$

$$b = -0.121$$

$$N = (\frac{\sigma_a}{a})^{1/b} = \left(\frac{0.7875 \times 10^9}{2343.75}\right)^{-\frac{1}{0.121}} = 82139.38$$

Endurance Strength for finite life

$$\sigma_{-1}^{1} = \sigma_{-1}\left(\frac{10^6}{\mathrm{N}}\right)^{0.09} = 750\left(\frac{10^6}{8.214 \times 10^5}\right)^{0.09}$$

$$= 763.398\ \mathrm{N/mm^2}$$

…….[1]

Calculation of FOS of Coil Mandrel using Goodman, Soderberg Method

Material Selected : AISI 304 Steel $\sigma_{max} 1.75 \times 10^9 \mathrm{Pa}$ $\sigma_{min} = 1.75 \times 10^8 \mathrm{Pa}$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = 2.0962 \times 10^9\ \mathrm{Pa} \quad \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = 0.7875 \times 10^9 \mathrm{Pa}$$

$$\sigma_e = 0.5 \times \sigma_{ut} = 0.5 \times 1500 \times 10^6\ \mathrm{Pa} \quad \sigma_{yt} = 800 \times 10^6 \mathrm{Pa} \quad k_f = 0.75$$

Goodman Formula : $\frac{1}{\mathrm{F.O.S}} = \frac{\sigma_m}{\sigma_{ut}} + \frac{\sigma_a}{\sigma_e}$ $\mathbf{FOS = 1.25}$

Soderberg Formula : $\frac{1}{\mathrm{F.O.S}} = \frac{\sigma_m}{\sigma_{yt}} + \frac{\sigma_a . K_f}{\sigma_e}$ $\mathbf{FOS = 1.05}$

$$\mathbf{FOS_{AVG}} = \frac{1.25 + 1.05}{3} = \mathbf{1.15}$$

III. ANSYS ANALYSIS AND MATLAB PLOTTING

ANSYS Analysis and MATLAB Plotting were employed to rigorously evaluate the structural integrity and performance of the Coil Ramp, Coil Car, and Coil Mandrel systems. ANSYS simulations provided critical insights into stress distribution and deformation across various components. The results Total Deformation, Directional Deformation Equivalent Stress (Von Mises), Maximum Principal stress are generated in ANSYS. MATLAB plotting illustrated stress concentrations at diverse parameters, supporting the optimization and enhancement of these coil handling systems. Using Cohesive Elements in Mesh Generation, meshing with element size up to 0.01 is generated for accurate results. [4]

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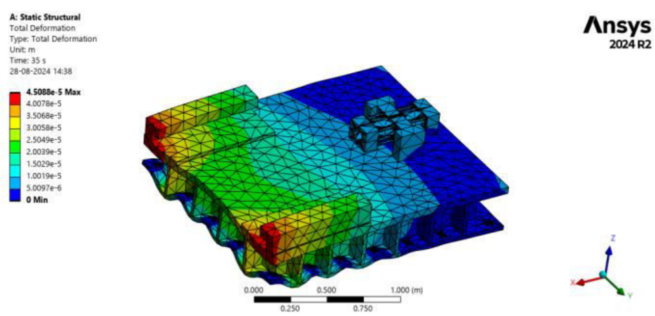
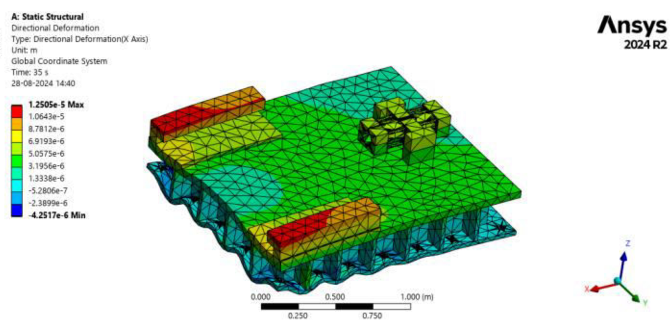
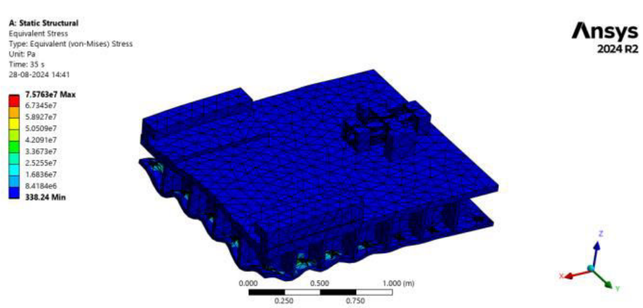
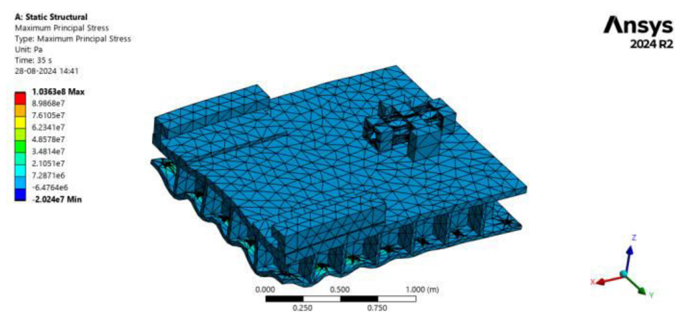
Coil Ramp Base Analysis

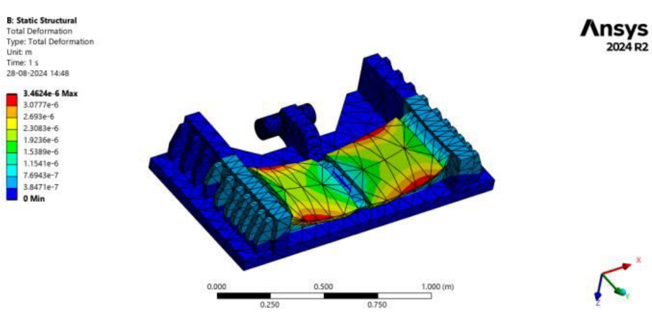


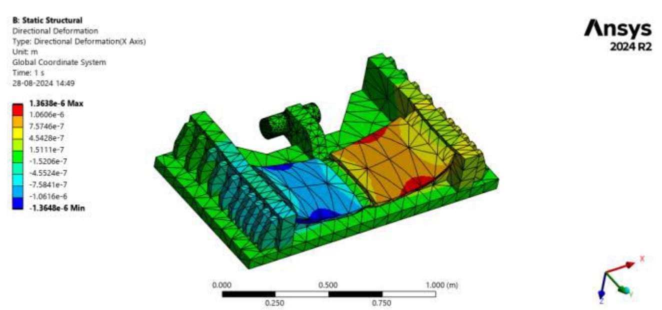


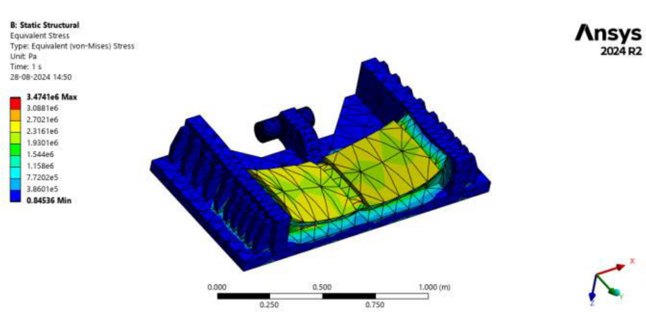


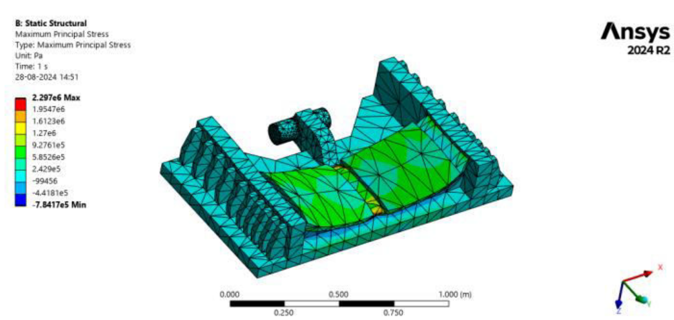


Coil Ramp Slider Analysis









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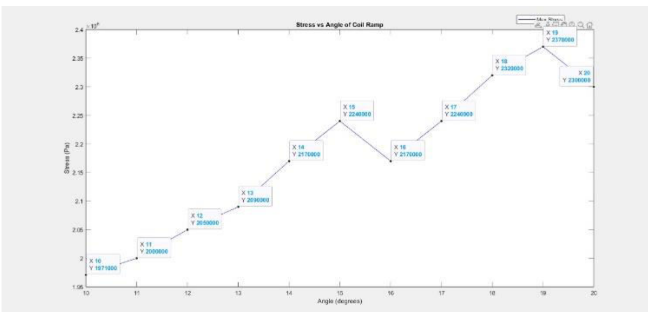
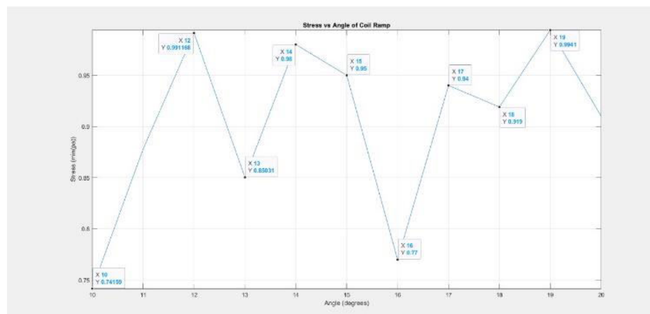
The thickness of the V-bed can be increased to reduce the shear deformation seen in analysis. At lower angle, higher compressive deformation is observed whereas at optimized angle of V-bed compromised values of shear stress and compressive stress are observed which result in long term life of bed.

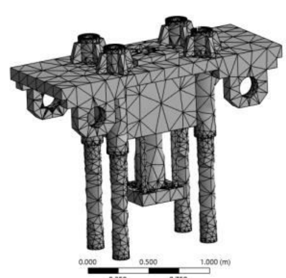
MATLAB : For V-Bed Analysis, from $\mathbf{10° \text{ to } 20°}$ angle range of V shape, maximum and minimum stress is calculated



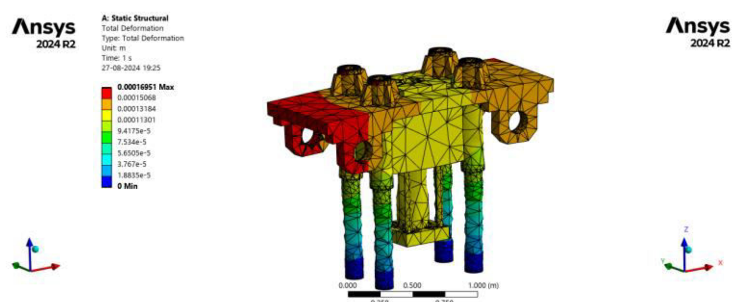


The results were plotted in MATLAB in terms of curves for Maximum and Minimum stresses. It is found that from $\mathbf{16° \text{ to } 16.5°}$ angle range the bed design is most suitable for maximum and minimum stress considerations.

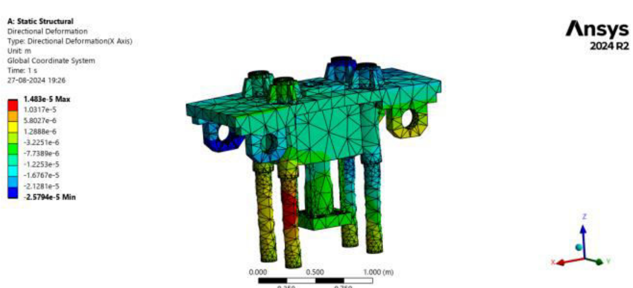
Coil Car Base Analysis



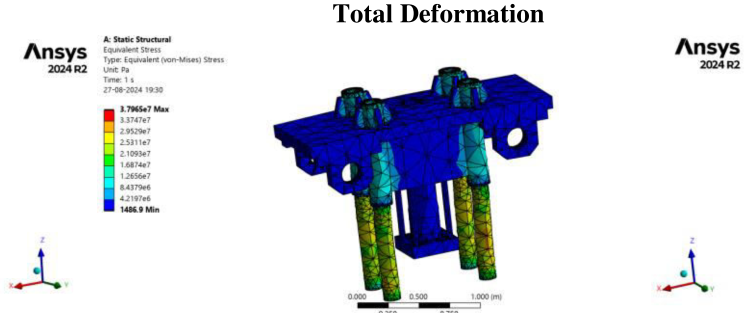
Meshed Car Base



Total Deformation



Directional Deformation



Equivalent Stress (Von Mises)

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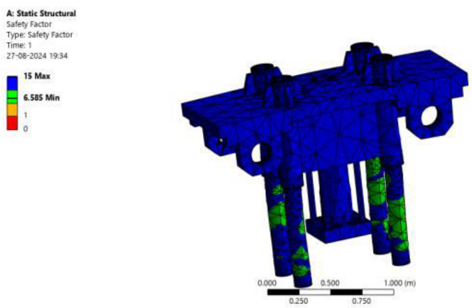
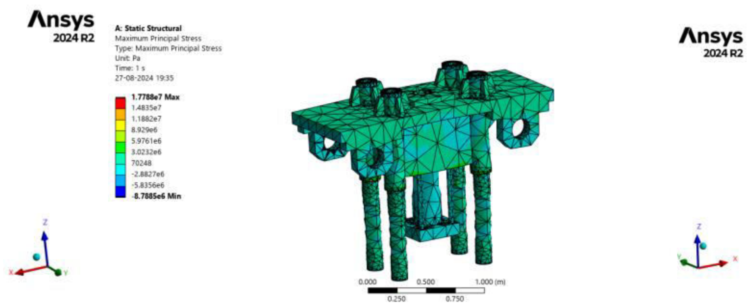
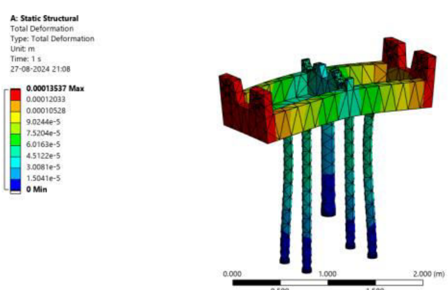


Factor of Safety

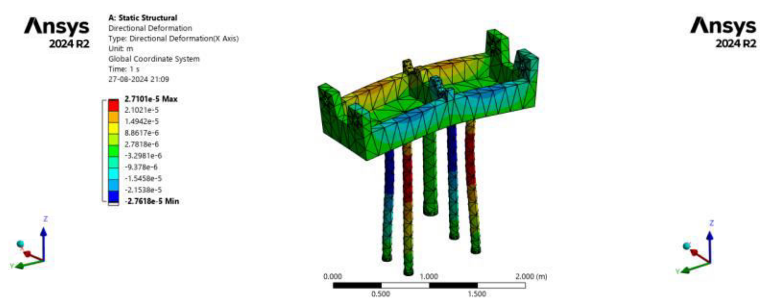


Maximum Principal Stress

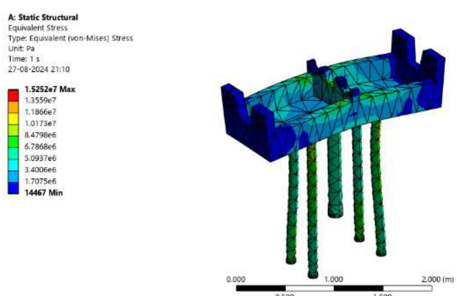
Coil Car Bed Analysis



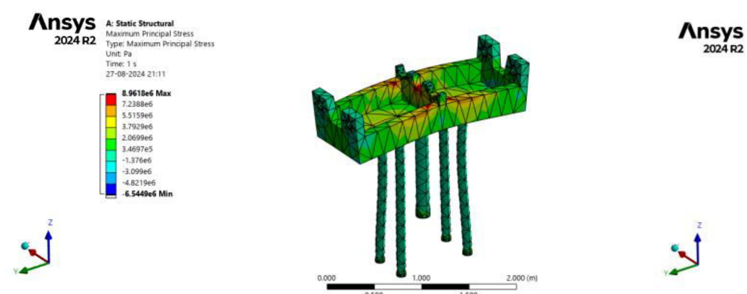
Total Deformation



Directional Deformation



Equivalent Stress (Von Mises)



Maximum Principal Stress

Analysis shows that bending of the bed is maximum at both side end edge. To prevent this additional two guide bars can be supported each at one side just below the roller bolt portion to resist the deformation due to load.

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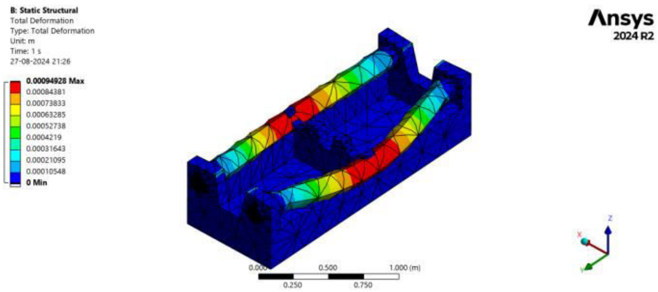
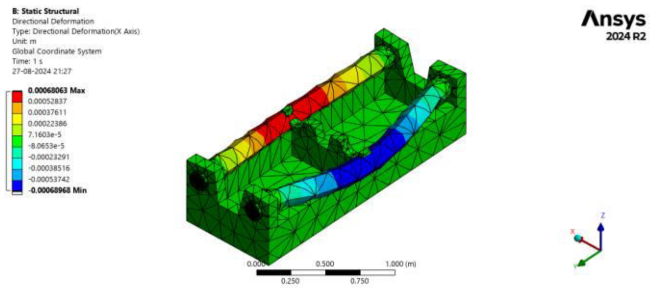
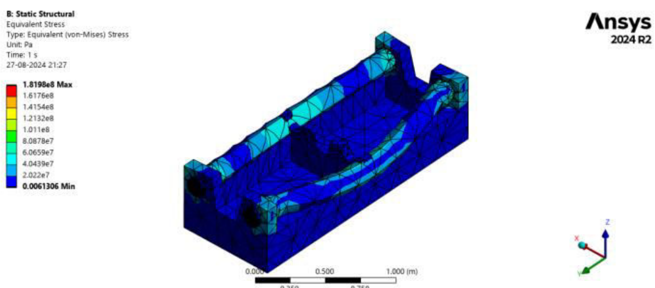
Coil Car Roller Analysis



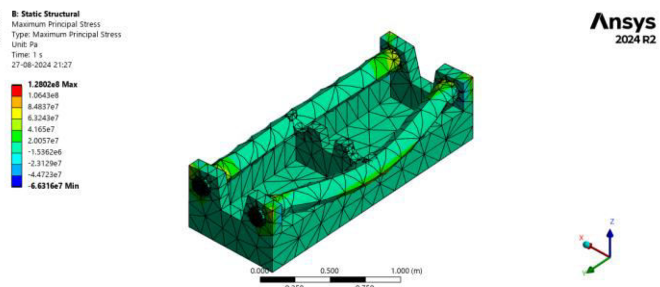
Total Deformation



Directional Deformation



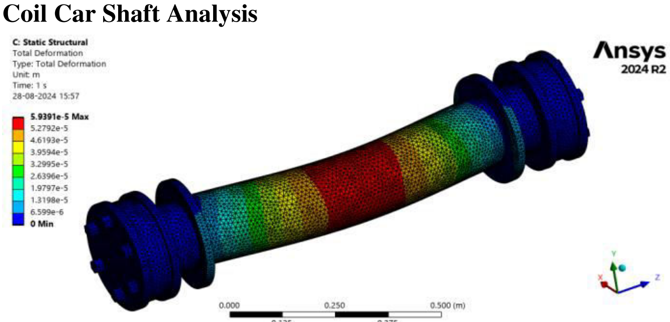
Equivalent Stress (Von Mises)



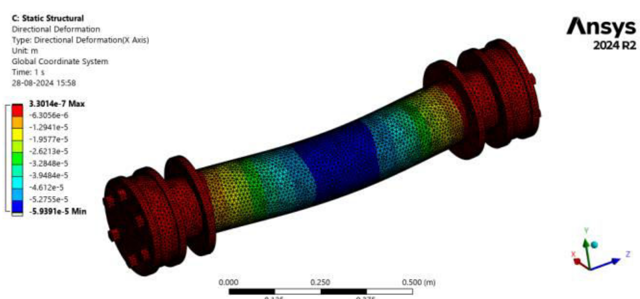
Maximum Principal Stress

After this analysis it can be resulted that the separated distance between rollers should be kept minimum depending upon the diametric size of coil.

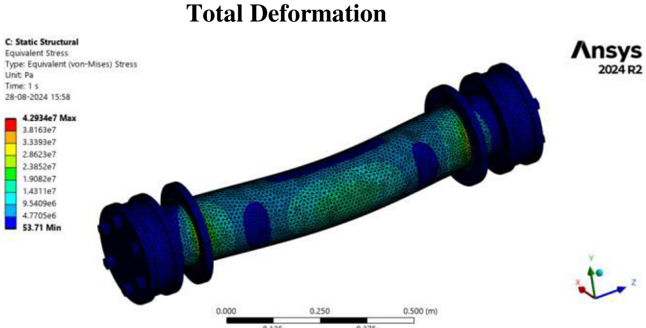
Coil Car Shaft Analysis



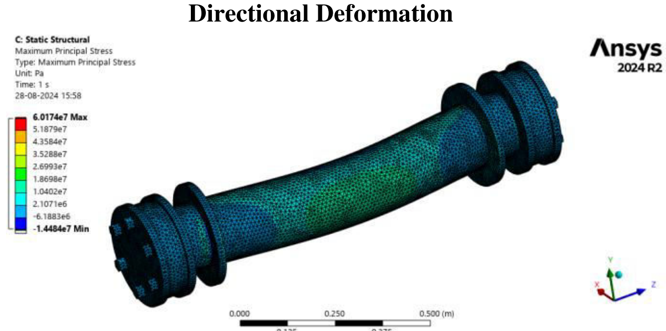
Total Deformation



Directional Deformation



Equivalent Stress (Von Mises)



Maximum Principal Stress

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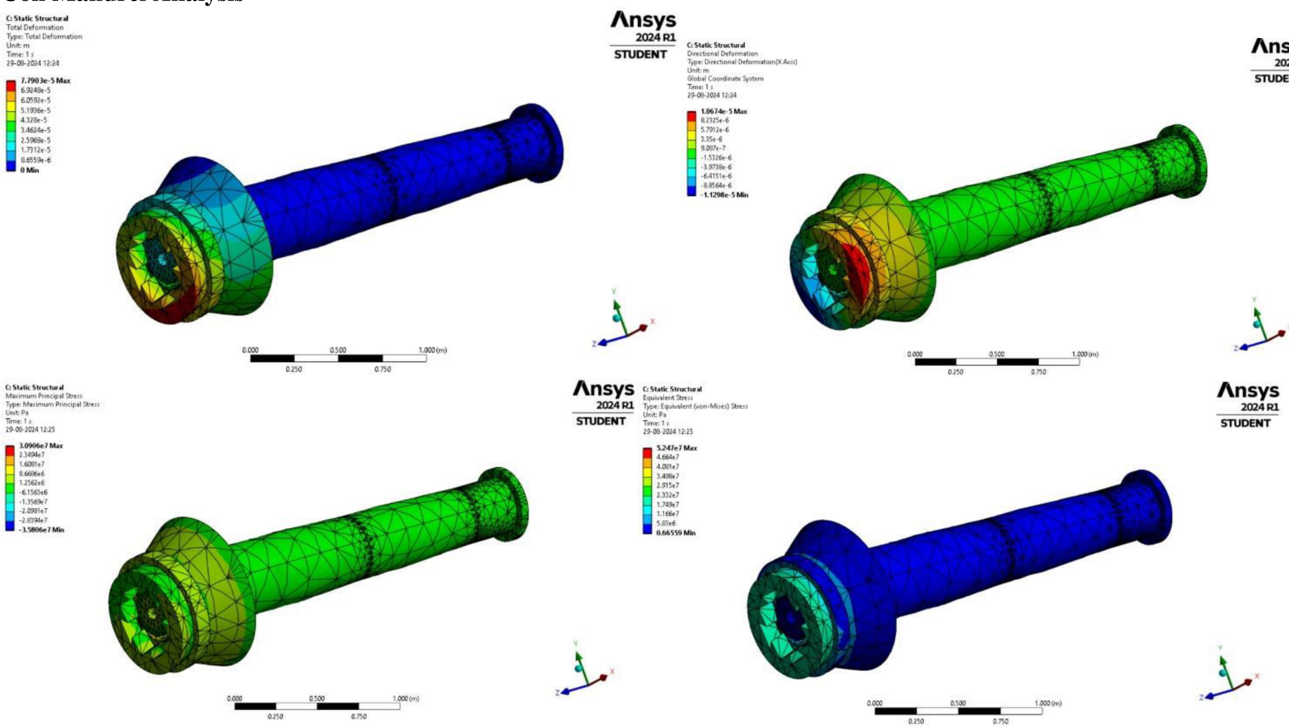
Coil Mandrel Analysis



MATLAB Code and Output:

```
% Define the data
angle = [20, 25, 30, 35, 40, 45];
max_stress = [3.26e7, 4.04e7, 5.25e7, 3.92e7, 2.82e7, 2.81e7];

% Create the plot
plot(angle, max_stress);

% Set the x-axis label
xlabel('Angle (degrees)');

% Set the y-axis label
ylabel('Maximum Stress (Pa)');

% Set the title
title('Relationship between Angle and Maximum Stress');

% Show the grid
grid on;
```

```
% Define the data
angle = [20, 25, 30, 35, 40, 45];
min_stress = [0.544, 0.641, 0.666, 0.808, 0.50825, 0.51045];

% Create the plot
plot(angle, min_stress);

% Set the x-axis label
xlabel('Angle (degrees)');

% Set the y-axis label
ylabel('Minimum Stress');

% Set the title
title('Relationship between Angle and Minimum Stress');

% Show the grid
grid on;
```

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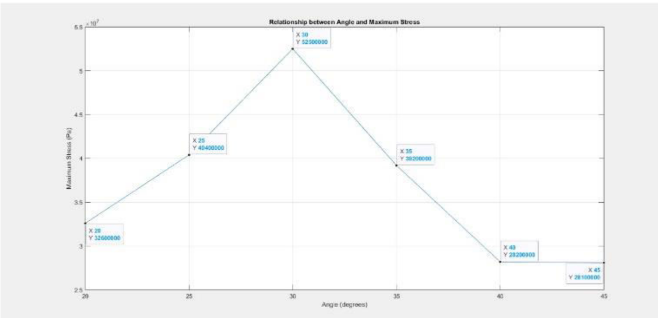
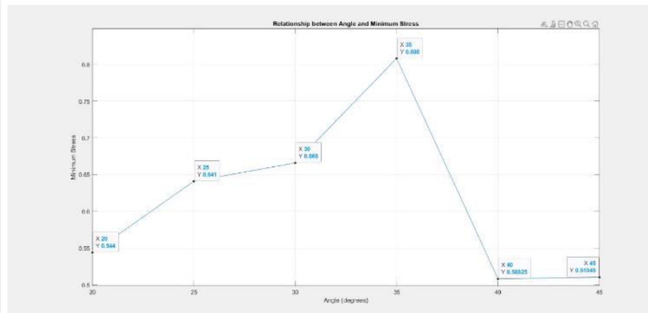
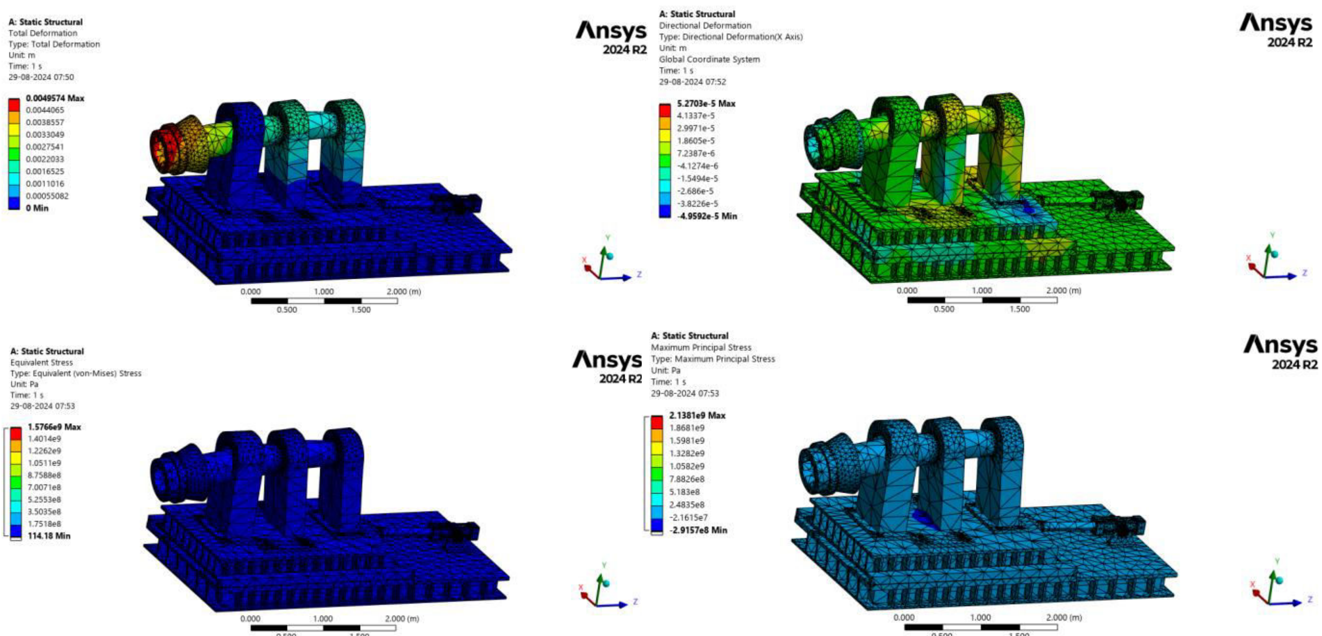




From above MATLAB result it is observed that from **30° to 35° angle** of chamfering the mandrel is most suitable for optimization of absorption of maximum and minimum amount of stress. The definite value for chamfered angle can be chosen as per the internal diameter of coil in which mandrel is to be inserted.

Coil Mandrel Uncoiler Section Analysis



This analysis shows that width of the mandrel supporting components should be higher to resist the bending deformation in mandrel.

IV.CONCLUSION AND FUTURE WORK/SCOPE

In this paper, Design and structural analysis of the Coil Handling Equipment was done for increased stress endurance with modification in their material and dimensional properties. The designs were meticulously reviewed and analysed over time; Machine Design methodology was used to optimize it even further.

Improving mesh parameters and tweaking the analytical models gave us more insightful knowledge about the components. After such intricate and continuous evaluation, we would like to conclude that the suggested design models are viable options while choosing Coil carrying Equipments.

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-Yash Chaudhari

"I am deeply thankful to my parents, (Sudhir Chandra Dwivedi and Seema Sudhir Chandra Dwivedi), for their unwavering support and consistent encouragement during ANSYS of Coil Ramp and throughout the MATLAB Plot process of publishing in this research paper." - Sudhanshu Dwivedi

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