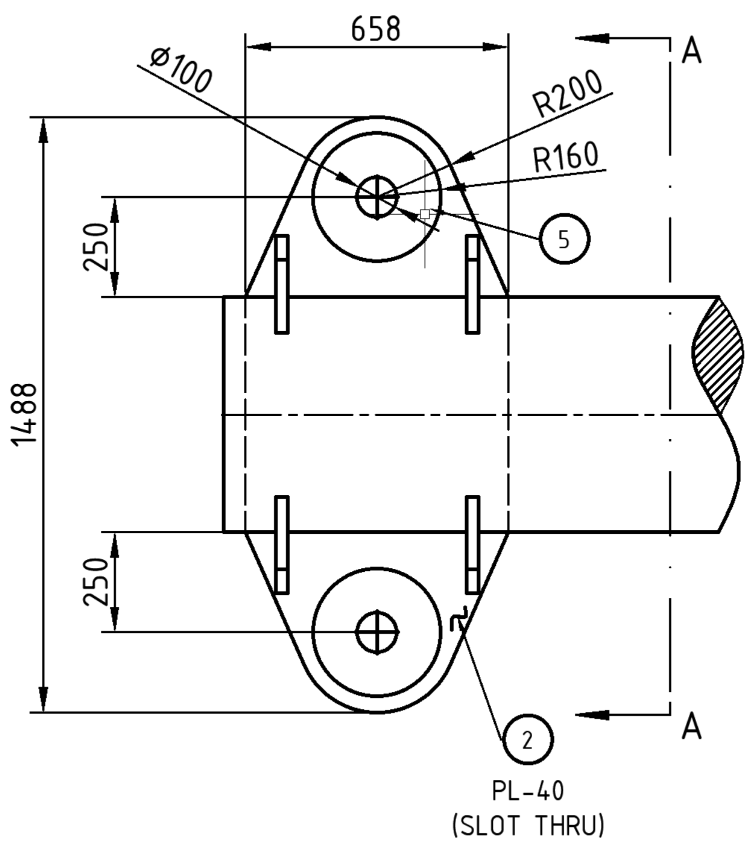
Padeye Design Report

June 30, 2025

1 Introduction

This report summarizes the padeye design checks, including geometric, bearing, and shear stress verifications, using the provided dimensions and material properties. All calculations are shown in plain text and equations for clarity.

Padeye design details



2 Input Data

- Shackle Pin Diameter: $D = 95$ mm
- Shackle Jaw Opening Width: $B = 147$ mm
- Shackle Inside Length: $H = 329$ mm
- Sling Diameter: $D_s = 80$ mm
- Yield Strength of Material: $F_y = 345$ MPa
- Young's Modulus: $E = 210,000$ MPa
- Poisson's Ratio: $\nu = 0.3$
- Padeye Hole Diameter: $d = 100$ mm
- Main Plate Thickness: $t = 40$ mm
- Cheek Plate Thickness: $t_c = 30$ mm
- Provided Main Plate Radius: $R_p = 200$ mm
- Provided Cheek Plate Radius: $R_c = 160$ mm
- Total Plate Thickness: $t_p = 100$ mm
- Padeye Length: $L = 658$ mm
- Number of Stiffeners: $n_{stiff} = 4$
- Distance Between Stiffener Centers: $H_s = 478$ mm
- Sling Load: $S_l = 86$ MT
- Acceleration due to Gravity: $g = 9.81$ m/s^2
- Calculated Sling Load: $SSL = S_l \times g \times 1,000 = 86 \times 9.81 \times 1,000 = 843,660$ N

3 Geometric Checks

3.1 Clearance Between Hole and Pin

Pin Diameter: $d = 95$ mm
Hole Diameter: $D = 100$ mm
Clearance: $C_1 = D - d = 100 - 95 = 5$ mm

3.2 Clearance Between Shackle Jaw and Plate

Shackle Jaw Opening: $B = 147$ mm
Total Plate Thickness: $t_p = 100$ mm
Clearance: $C_2 = \frac{B-t_p}{2} = \frac{147-100}{2} = 23.5$ mm

3.3 Sling Clearance

Sling Clearance: $SC = H - (D_s + R_p - 0.5 \cdot d) = 329 - (80 + 200 - 0.5 \times 100) = 329 - 280 = 49$ mm

3.4 Minimum and Maximum Hole Sizes

Minimum Hole: $\text{Minimum Hole} = \max(D + 3.18,\ 1.05 \cdot D) = \max(95 + 3.18,\ 1.05 \times 95) = \max(98.18,\ 99.75) = 99.75$ mm
Maximum Hole: $\text{Maximum Hole} = D + 9.53 = 95 + 9.53 = 104.53$ mm

3.5 Minimum and Maximum Clearances

Minimum Clearance: $\text{Minimum Clearance} = 0.05 \times t_p = 0.05 \times 100 = 5$ mm
Maximum Clearance: $\text{Maximum Clearance} = \max(0.1 \times t_p,\ 20) = \max(10,\ 20) = 20$ mm

4 Bearing Stress Check

Applied Load: $F = SSL = 843,660$ N
Padeye Hole Diameter: $d = 100$ mm
Main Plate Thickness: $t = 40$ mm
Bearing Stress:

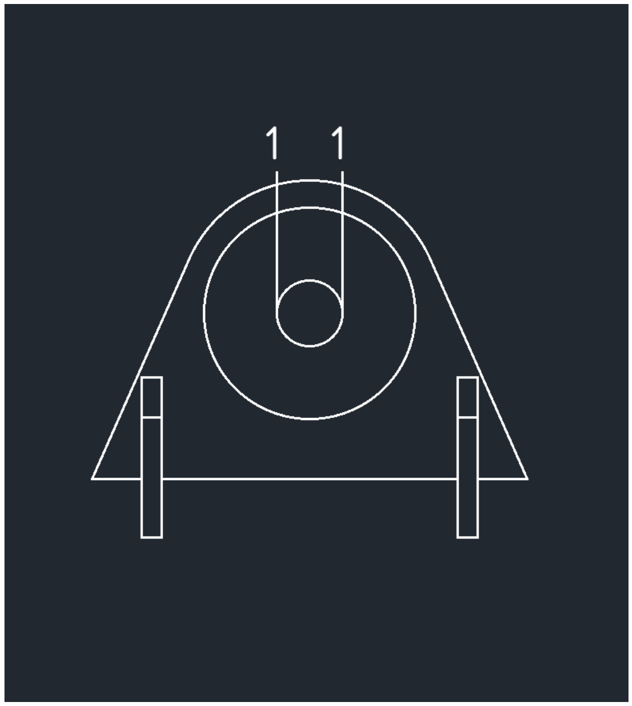
$$\sigma_b = \frac{F}{d \cdot t} = \frac{843,660}{100 \times 40} = 210.92 \text{ MPa}$$

Permissible Bearing Stress:

$$\sigma_{b,\text{perm}} = 0.9 \times F_y = 0.9 \times 345 = 310.5 \text{ MPa}$$

Check: $210.92 \text{ MPa} < 310.5 \text{ MPa} \implies \text{PASS}$

5 Case 1 Check shear failure from pin hole through main/cheek plate



Applied Load: $F = SSL = 843,660$ N
Main Plate Radius: $R_p = 200$ mm
Cheek Plate Radius: $R_c = 160$ mm
Shackle Hole Diameter: $d = 100$ mm
Main Plate Thickness: $t = 40$ mm
Cheek Plate Thickness: $t_c = 30$ mm
Resisting Area:

$$A_{p1} = (R_p - \frac{d}{2}) \times 2t + (R_c - \frac{d}{2}) \times 4t_c = 25,200 \text{ mm}^2$$

Shear Stress:

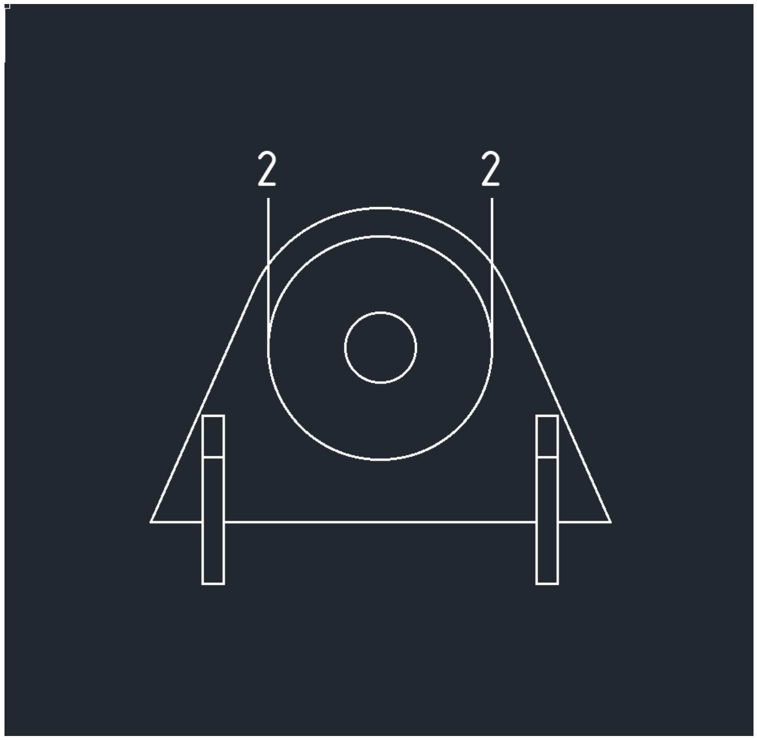
$$\sigma_s = \frac{F}{A_{p1}} = \frac{843,660}{25,200} = 33.49 \text{ MPa}$$

Permissible Shear Stress:

$$\sigma_{s,\text{perm}} = 0.4 \times F_y = 0.4 \times 345 = 138 \text{ MPa}$$

Check: 33.49 MPa $<$ 138 MPa $\implies$ PASS

6 Case 2: Shear Stress Failure in Main Plate



The allowable shear stress is: $0.4 \cdot F_y = 138$ MPa
The resisting area of the plate is calculated as:

$$A_{p2} = (R_p - R_c) \cdot 2t + \pi R_c t$$

Where:

- $R_p = 200$ mm (Provided main plate radius)
- $R_c = 160$ mm (Provided cheek plate radius)
- $t = 40$ mm (Main plate thickness)

Substituting the values:

$$A_{p2} = (200 - 160) \times 2 \times 40 + \pi \times 160 \times 40 = 3,200 + 20,106 = 23,306 \text{ mm}^2$$

The shear stress is:

$$\tau_{v2} = \frac{F}{A_{p2}} = \frac{843,660}{23,306} = 36.22 \text{ MPa}$$

Where:

- $F = 843{,}660$ N (Load on the padeye)
- $A_{p2} = 23{,}306$ mm^2 (Resisting area of the plate)

The check is: 36.22 MPa $<$ 138 MPa $\implies$ OK
Therefore, the main plate is safe against shear stress failure in this case.

7 Summary Table

Check	Calculated Value	Permissible Value	Result
Bearing Stress (σ_b)	210.92 MPa	310.5 MPa	PASS
Shear Stress (σ_s)	33.49 MPa	138 MPa	PASS
Hole Size	100 mm	99.75 – 104.53 mm	PASS
Clearance (C_2)	23.5 mm	5 – 20 mm	FAIL

8 Conclusion

All checks except the shackle jaw clearance (C_2) are within permissible limits. The clearance between the shackle jaw and the plate exceeds the maximum allowed value and should be reviewed in the design.

9 Spreader Bar Critical Stress

The critical buckling load of the spreader bar is calculated as follows:

Step 1: Moment of Inertia

The moment of inertia for a hollow circular section:

$$I = \frac{\pi}{64}(D_o^4 - D_i^4)$$

Where:

- D_o = Outer diameter = 588 mm
- t = Thickness = 25 mm
- $D_i = D_o - 2t = 588 - 2 \times 25 = 538$ mm

So,

$$I = \frac{\pi}{64}(588^4 - (588 - 2 \times 25)^4) = 1.75 \times 10^9 \text{ mm}^4$$

Step 2: Radius of Gyration

$$r = \sqrt{\frac{I}{A}}$$

Where:

$$A = \frac{\pi}{4}(D_o^2 - D_i^2)$$

Step 3: Euler's Critical Buckling Load

The critical buckling load for a pinned-pinned column:

$$P_{cr} = \frac{\pi^2 EI}{(KL)^2}$$

Where:

- $E = 210,000$ MPa
- $K = 1$ (effective length factor)
- $L = 6,288$ mm (length of spreader bar)

Substituting the values:

$$P_{cr} = \frac{\pi^2 \times 210,000 \times 1.75 \times 10^9}{(1 \times 6,288)^2} = 92,018,373.48 \text{ N} = 9,383.26 \text{ Tons}$$

Step 4: Comparison

- Calculated compressive force from rigging: $F_c = 75.01$ Tons
- Critical buckling load: $P_{cr,\text{tons}} = 9,383.26$ Tons

Since $F_c \ll P_{cr,\text{tons}}$, the spreader bar is safe against buckling under the given load.

10 Allowable Axial Compressive Stress (AISC)

The allowable axial compressive stress F_a is calculated according to the AISC code, depending on the slenderness ratio KL/r:

- If $KL/r \leq C_c$:

$$F_a = \left[1 - \frac{(KL/r)^2}{2C_c^2}\right] \frac{F_y}{N_d} \left[1 + \frac{9(KL/r)}{40C_c} - \frac{3(KL/r)^3}{40C_c^4}\right]$$

- If $KL/r > C_c$:

$$F_a = \frac{\pi^2 E}{1.15 N_d (KL/r)^2}$$

Where:

- $F_y = 345$ MPa (yield strength)
- $N_d = 4$ (safety factor)
- $E = 210{,}000$ MPa (Young's modulus)

- $K = 1$ (effective length factor)
- $L =$ 6,288 mm (length)
- $r =$ 303.5 mm (radius of gyration)
- $C_c = \sqrt{\frac{2\pi^2 E}{F_y}}$

Calculate KL/r and C_c: $KL/r = \frac{K \cdot L}{r} = \frac{1 \times 6{,}288}{199.45} = 31.56$
$C_c = \sqrt{\frac{2\pi^2 \times 210{,}000}{345}} = 109.61$
Since $KL/r < C_c$, use the first formula:

$$F_a = 88.03 \text{ MPa} = 8.80 \text{ MT/m}^2$$

Applied Compressive Stress Calculation:
The applied compressive stress is calculated as:

$$F_{\text{app}} = \frac{F}{A_s}$$

Where:

- $F = 735,596.82$ N (compressive force)
- $A_s = 44,217.92$ mm^2 (cross-sectional area)

So,

$$F_{\text{app}} = \frac{735,596.82}{44,217.92} = 16.64 \text{ MPa}$$

The allowable axial compressive stress is $F_a = 88.03$ MPa (8.80 MT/m^2).
The applied compressive stress from the rigging is:

$$F_{\text{app}} = 16.64 \text{ MPa} = 1.66 \text{ MT/m}^2$$

Since $F_{\text{app}} < F_a$, the applied compressive stress is within allowable limits and the spreader bar is safe under the applied compressive load according to AISC.