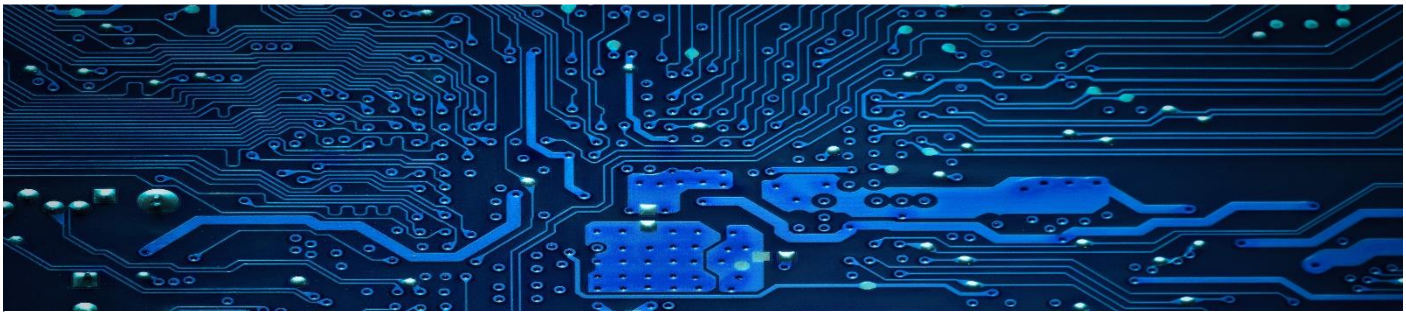
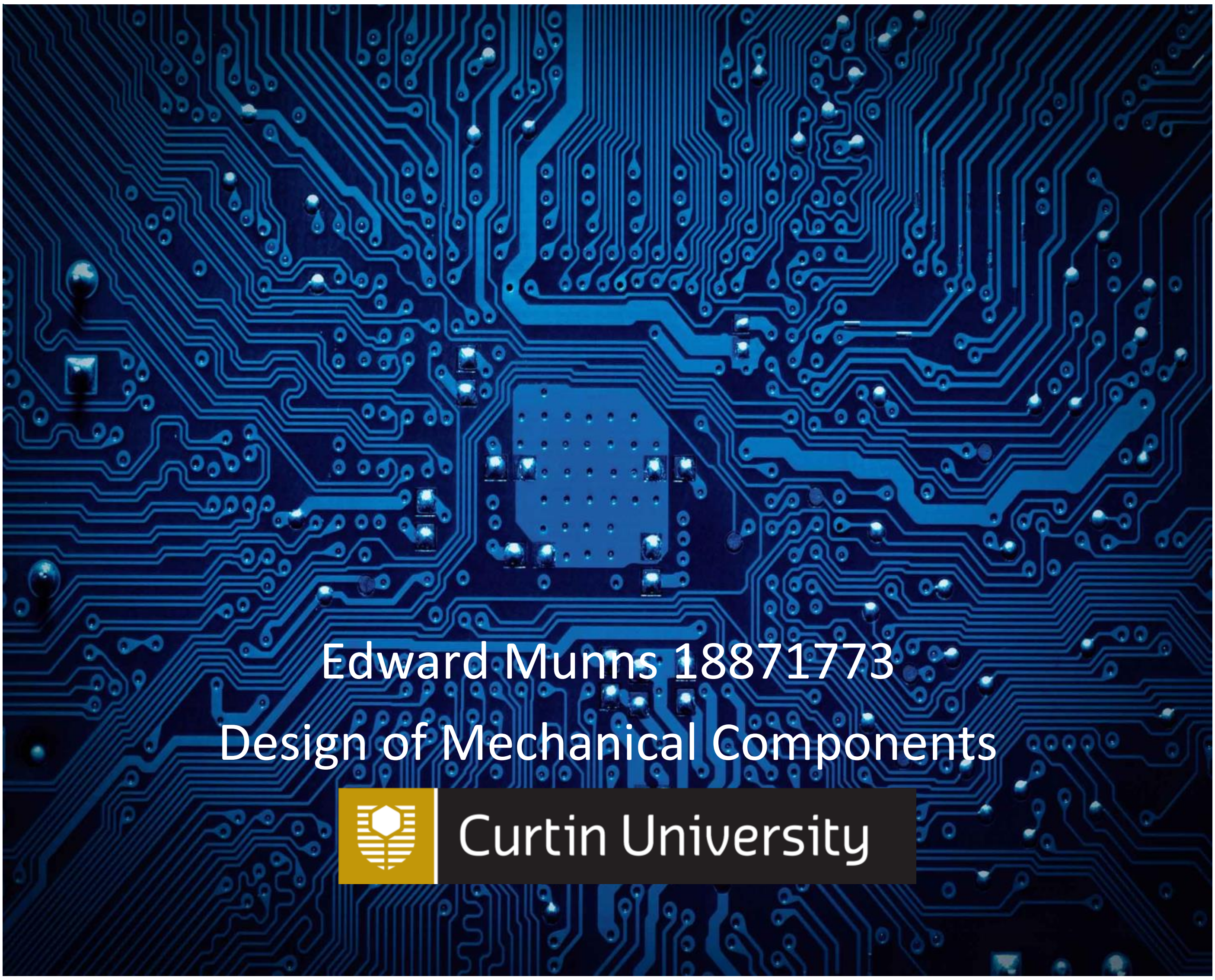


DESIGN OF MECHANICAL COMPONENTS ASSIGNMENT.



Edward Munns 18871773

Design of Mechanical Components

Curtin University

Contents

Table of figures:

1. Introduction:

This report details the design of a crane mechanism intended to lift 4.54kg blocks of aluminium between an outgoing conveyor belt exiting a furnace and placing them onto another belt for further processing. First the design will be explained, then calculation of relevant stresses and safety factors will be documented.

The client requires the following criterion in the design:

- The design must have a safety factor of greater than 2.
- The design should incorporate a selected range of actuators as they are commonly used parts in the organisation.
- The block orientation should not be rotated in the process of moving from the incoming belt to the outgoing belt.
- The block shall be received within 5mm of center line of the incoming belt and placed within 20mm of the centerline of the outgoing belt.
- The walkway between the incoming and outgoing belts shall not be obstructed when the mechanism is not in use.
- The design must be able to complete one cycle in under 30 seconds to keep up with the output of the manufacturing line.

The solution proposed accomplishes this design brief by way of a single double-sided swing arm with the end effector suspended on 20mm flange mounted bearings (McMaster-Carr part number 1483N1) through the center. This arm swings from the incoming belt to the outgoing belt with the end effector acting as a pendulum between the two arms. To accomplish this movement a 1200 Newton actuator of 150mm stroke was selected. The forces involved in this motion were far beyond the safe operating range of this actuator, so a mechanical solution was incorporated to minimize loading. This solution uses two traction (pulling force) gas springs, one on each side arm, to support the actuator through the highest load sections of travel. This solution allows "tunability" of actuator forces. Two gas springs were used to keep the structure as symmetrical as possible and as a form of redundancy should one of the gas springs fail. An attempt has been made in this design to minimize the required material purchases by using the same size steel throughout the design.

The assembly consist of two major components, the arm and the mounting frame. These components are constructed from $50 \times 25 \times 3mm$ rectangular hollow section cold rolled steel welded into 2 sub frames. The mounting assembly is bolted to a mounting plate provided by the client via six M10 bolts. This frame supports the arm assembly via two hinge pins that allow the arm to actuate from one belt to the other. All pins in this design are constructed from 4340 alloy steel. This material was used as forces in some pins are high and require the strength this material can provide.

Calculation of stresses and safety factors were carried out in the assumed position of maximum stress, where the arm is at full extension over the receiving belt in the instant before the payload is delivered. An illustration of this operating condition can be seen in the second render of the design on page 3 of this report.

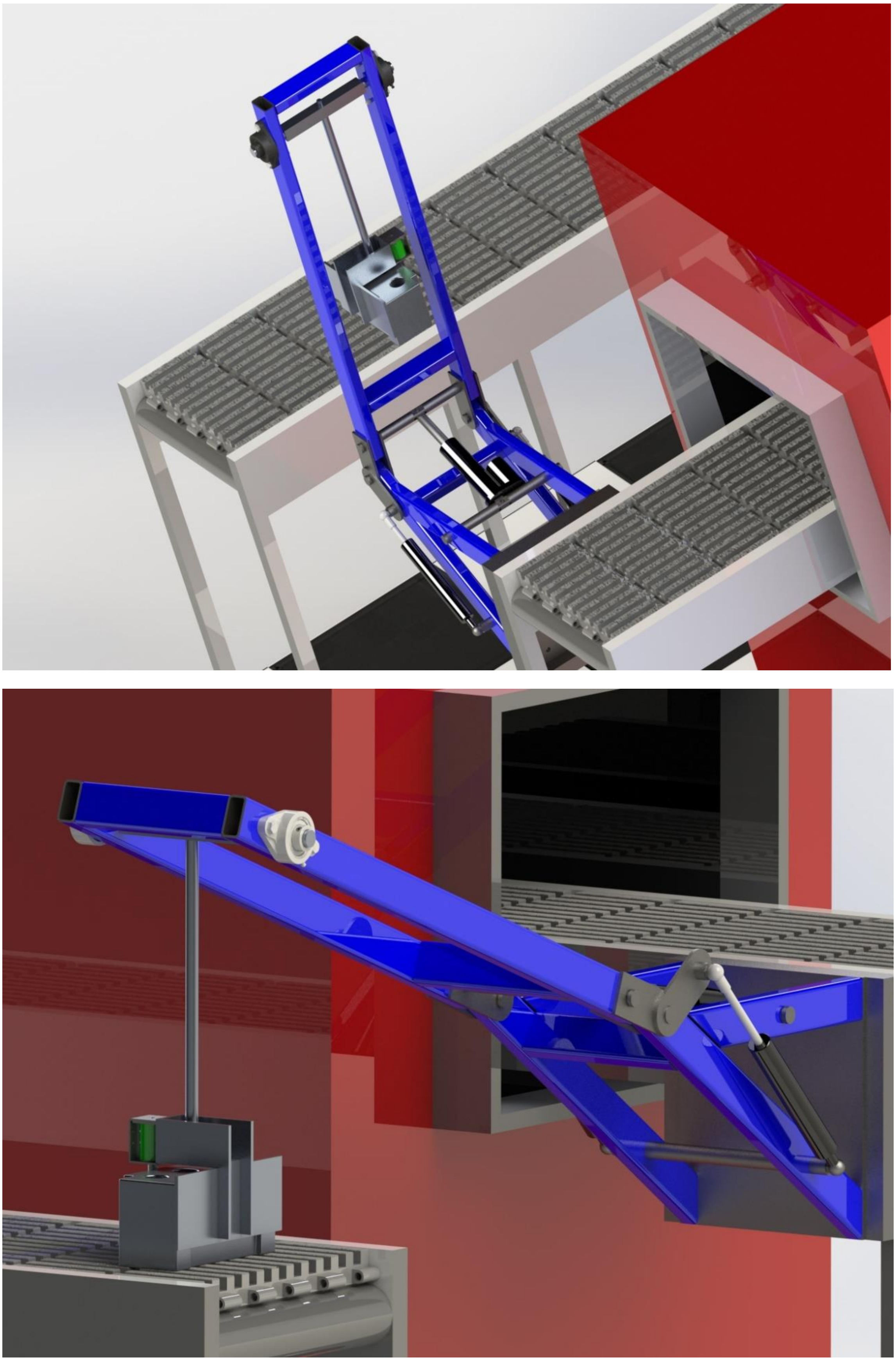


Figure 1: Render of Design.

2. Arm Calculations:

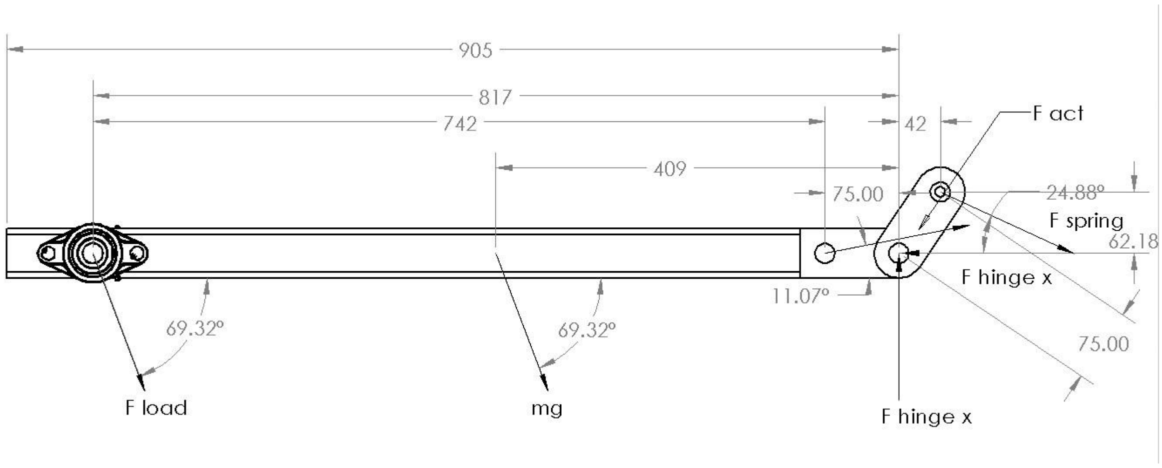


Figure 2: Arm Free Body Diagram.

The above diagram shows the arm assembly in the assumed highest load position. This load occurs at the outgoing belt in the instant before the block is removed from the end effector (as seen in figure 1). As the arm assembly is symmetrical the forces of the load, component mass and hinge can be halved and force calculations can be carried out on a single side. As each side has a gas spring attached and the actuator top shaft will be acted upon from both sides these values will not be halved. The gas spring forces create an opposite moment about the hinge point to the load and weight moments. The actuator chosen for this design has a load rating of 1200 Newtons. This value is assumed to be the static load rating as opposed to the dynamic load rating to add an extra layer of safety. The gas spring pulling force can be tuned to relive the force exerted on the actuator by increasing the moment the springs exert about the hinge. A target actuator force of zero was calculated to approximate the gas spring force required for each side and this value was used in gas spring selection.

2.1 Arm Initial Force calculations:

$F_{load\ x} = \frac{59.32}{2}\cos(69.32) = 10.47\text{N} \rightarrow +$

$F_{load\ y} = \frac{59.32}{2}\sin(69.32) = 27.75\text{N} \downarrow +$

$F_{Gx} = \frac{76.71}{2}\cos(69.32) = 13.54\text{N} \rightarrow +$

$F_{Gy} = \frac{76.71}{2}\sin(69.32) = 35.88\text{N} \downarrow +$

2.2 Arm Gas Spring Force calculations:

Let $F_{actuator} = 0$,

$$\circlearrowright + \sum M_{hinge} = 0, \qquad -F_{load\ y} \times L_{load} - F_{G\ y} \times L_G + M_{required}$$

$$\therefore\ M_{required} = 27.75N \times 0.817m + 35.88N \times 0.409m = 37.35Nm \circlearrowright +$$

$Using\ this\ value\ to\ solve\ for\ F_{spring}$:

$$37.35 = F_{spring} \times \sin(24.88) \times 0.042m + F_{spring} \times \cos(24.88) \times 0.062m$$

$$F_{spring} = \frac{37.35Nm}{\sin(24.88) \times 0.042m + \cos(24.88) \times 0.062m} = 504.30N$$

To prevent the actuator transitioning from tension to compression at this location in arm movement a force of 450N was chosen for gas spring force. These gas springs are available from McMaster-Carr online for $167.50 AUD plus shipping (McMaster-Carr 2019). Using this value all other forces within the arm member can be resolved.

$$F_{spring\ x} = 450 \times \cos(24.88) = 408.24N \rightarrow +$$

$$F_{spring\ y} = 450 \times \sin(24.88) = 189.32N \downarrow +$$

$$M_{spring} = 450 \times \sin(24.88) \times 0.042 + 450 \times \cos(24.88) \times 0.062 = 33.26Nm \circlearrowright +$$

2.3 Arm Final Force Calculations:

$$\circlearrowright + \sum M_{hinge} = 0, \qquad -M_{load} - M_G + M_{actuator} + M_{spring},$$

$$-27.75(0.817) - 35.88(0.409) + F_{actuator} \times \sin(11.07) \times 0.075 + 33.26$$

$F_{actuator} = \frac{22.67Nm + 14.67Nm - 33.26Nm}{\sin(11.07) \times 0.075m} = 283.32N$ per side of actuator shaft.

$$F_{actuator\ y} = 283.32 \sin(11.07) = 54.4N \uparrow +$$

$$F_{actuator\ x} = 283.32 \cos(11.07) = 278.05N \rightarrow +$$

$$\uparrow + \sum F_y = 0, \quad -27.75 - 35.88 + 54.4 - 189.32 + F_{hinge\ y} = 0,$$

$$F_{hinge\ y} = 27.75 + 35.88 - 54.4 + 189.32 = 198.55N \uparrow +$$

$$\leftarrow + \sum F_x = 0, \quad -10.47 - 13.54 - 278.05 - 408.24 + F_{hinge\ x} = 0,$$

$$F_{hinge\ x} = 10.47 + 13.54 + 278.05 + 408.24 = 710.3N \leftarrow +$$

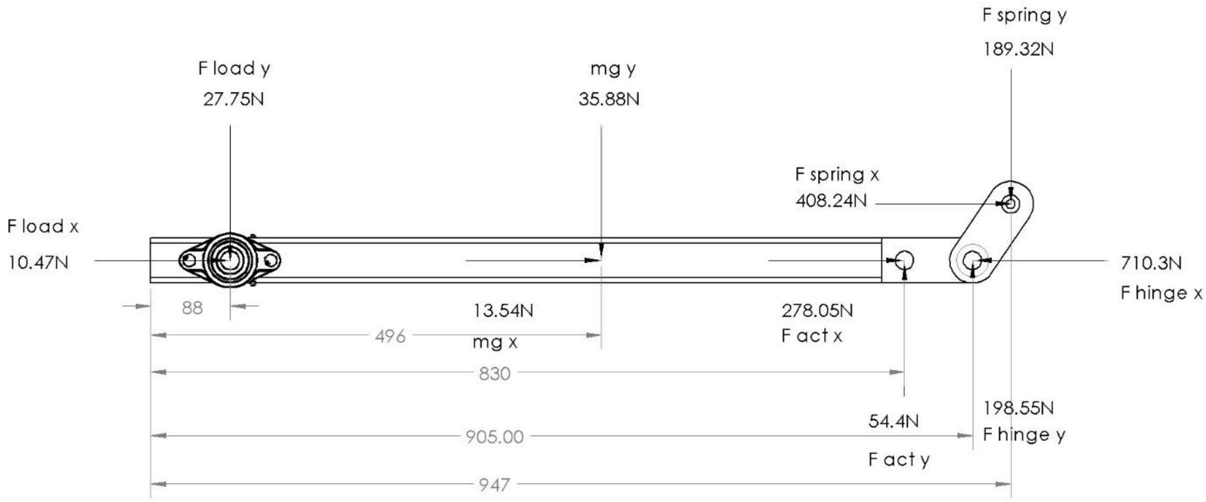
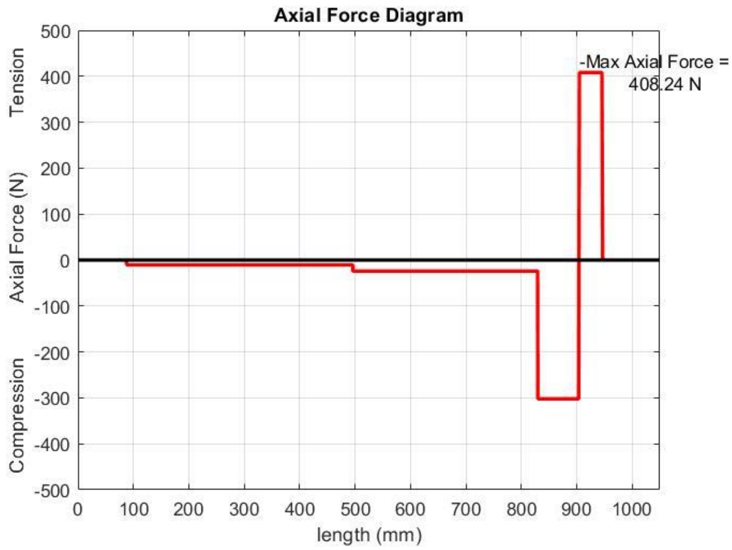
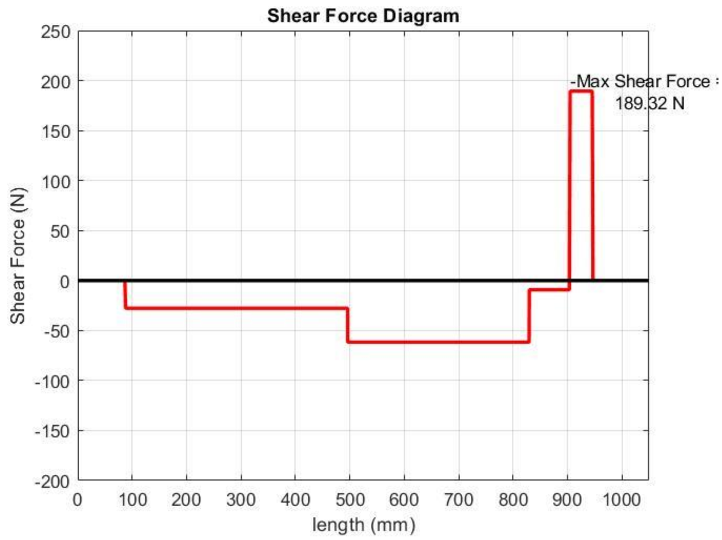


Figure 3: Arm Resolved Free Body Diagram.

2.4 Arm Axial Force, Shear Force and Bending Moment diagrams:





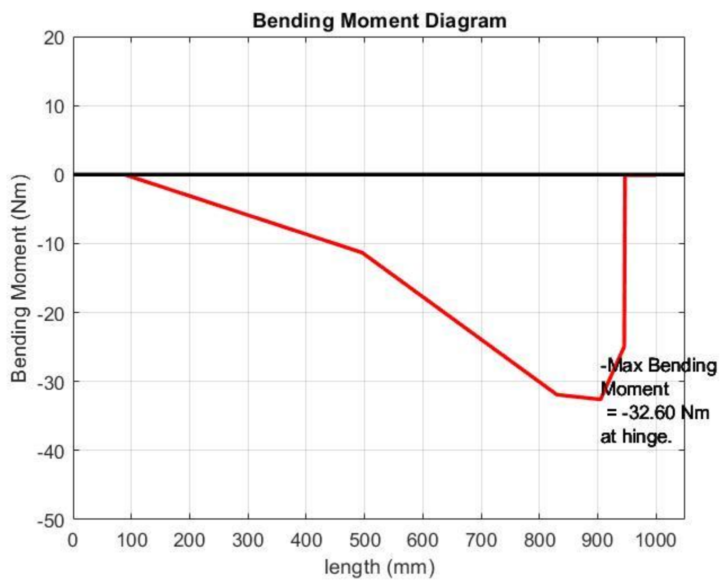


Figure 4: Arm Force diagrams.

2.5 Arm Stress calculations:

Major factors to be evaluated in the arm assembly are:

- The hinge point
- After the hinge point when material changes to rectangular hollow section
- Deflection at the extremity of arm

Hinge Point:

The hinge point must deal with the maximum axial force, shear force and bending moment in the arm assembly. As the hinge consists of a 5× 50mm flat bar with a 20mm hole in the center the raised axial stress factor (K_t) must be found. The chart below was used to find this value:

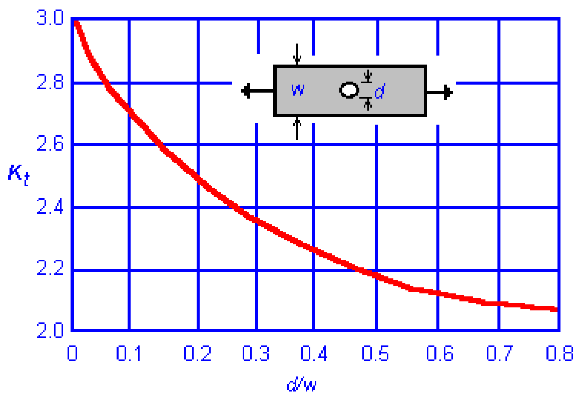


Figure 5: Kt chart for bar with hole in axial loading. (AnalysisChamp 2010)

It is assumed that maximum stress in the hinge area will occur at the bottom of the bar in tension. A small element in this area was taken for the following equations.

$$\sigma_{nominal} = \frac{P}{A} = \frac{408.24}{(50-20)\times 5} = 2.7216Mpa, \quad from\ chart \quad \frac{d}{w} = \frac{20}{50} = 0.4 \ \therefore \ K_t \approx 2.23$$

$$\sigma_{raised} = K_t \times \sigma_{nominal} = 2.23 \times 2.7216 = 6.069Mpa$$

To calculate stress due to bending:

$$\sigma_{nominal} = \frac{Mc}{I}, \qquad I = \frac{bh^3}{12} = \frac{5 \times (50^3 - 20^3)}{12} = 48750mm^4$$

$$\sigma_{nominal} = \frac{32.6 \times 10^3 \times 25}{48750} = 16.72Mpa$$

These two values were summed and next the shear stress was calculated:

$$\tau = \frac{P}{A} = \frac{189.32}{(50-20) \times 5} = 1.2621Mpa$$

Next using equations for maximum stress:

$$\sigma_1, \sigma_2 = \frac{\sigma_1 + \sigma_2}{2} \pm \sqrt{{\tau_{xy}}^2 + (\frac{\sigma_1 - \sigma_2}{2})^2} = \frac{22.79+0}{2} \pm \sqrt{1.2621^2 + (\frac{22.79-0}{2})^2} = 22.85, -0.065$$

$$\tau_{max} = \sqrt{1.2621^2 + (\frac{27.279-0}{2})^2} = 11.46Mpa$$

The hinge component is constructed from C350 cold rolled steel which has a yield stress rating of 350Mpa (Standards Australia 2016). The Australian standard 3990-1992 states that the maximum stress in a beam must not exceed 0.66×material yield stress and shear stress shall not exceed 0.37×yield stress (Standards Australia 2016).

$$\therefore safety\ factor\ for\ maximum\ stress = \frac{0.66 \times 350Mpa}{22.85Mpa} = 10.1$$

Rectangular Hollow Section

To verify the integrity of the rectangular hollow section part of the arm assembly the same values as the hinge calculation can be used with the area and moment of inertia values of this section. If this safety factor exceeds the minimum requirement then there is no need to perform further calculation as the hinge point scenario produces much higher stress. This was done as a fast way to check if further calculation was required. The moment of inertia and area values for this section were taken directly from a steel suppliers data sheet to account for corner radii of the section (Madalia Steel 2019).

$I_{rhs} = \ 0.12 \times 10^6 mm^4, \quad A_{rhs} = 391mm^2$

As these values are higher than the hinge values for area and moment of inertia and loading is lower in this section there is no need for further calculation.

Deflection at the extremity of beam:

Deflection due to bending at the end of the arm assembly must be calculated as the arm spans 905mm from the hinge point and is loaded close to the end creating a high moment situation. It is assumed that axial deformation in this assembly will be minimal and so this will be negated. To calculate deflection at the end of the arm assembly:

$$Deflection(\partial) = \frac{force \times length^3}{3 \times Mod_{elast} \times moment\ of\ inertia} = \frac{Pl^3}{3EI}$$

As the arm is constructed from $50 \times 25 \times 3mm$:

$I_{rhs} = \ 0.12 \times 10^6 mm^4$

$Mod_{elast}\ of\ C350\ cold\ rolled\ steel = 205 \times 10^3 Mpa$ (Madalia Steel 2019)

There are 3 loads present to the left of the hinge point so these values must be summed together. As there is also a load present to the right of the support this distance must also be considered. The distance from the gas spring point to the bearing center = 942mm:

$\therefore\ using\ \frac{P \times l_{total} \times l_{support}{}^2}{3 \times E \times I}:$

$$\partial = \frac{(27.75)(942 \times 817^2) + (35.88)(942 \times 409^2) - (53.6)(942 \times 75^2)}{3(205 \times 10^3)(0.12 \times 10^6)} = 0.309mm$$

3. Mounting Frame:

The mounting frame consists of four welded beams and a cross member. It is assumed forces in the cross member will be negligible as the mounting frame is symmetrical and should therefore be loaded equally from both sides and so should have minimal axial loading when viewed from the axis in figure above. For this reason, a calculation of one side will solve for both sides as in the arm assembly. The entire frame assembly has a mass of approximately 5.5 kgs so component weight in this assembly is assumed to be negligible. As the frame is a rigid assembly the beams are in a fixed/fixed support situation. This makes the assembly statically indeterminate. To calculate resultants the assembly was split into 2 beams, the outrigger beam that provides the hinge mechanism and the strut beam that supports this outrigger.

3.1 Outrigger Beam:

Figure 6: Frame outrigger Free Body Diagram.

$$R_2 = \frac{Pa^2}{l^3}(3b + a), \qquad \frac{90 \times (319^2)}{438^3} \times (3 \times 119 + 319) = 73.68N \downarrow$$

$$M_2 = \frac{Pba^2}{l^2}, \qquad \frac{90 \times 119 \times (319^2)}{438^2} = 5.68Nm \circlearrowleft$$

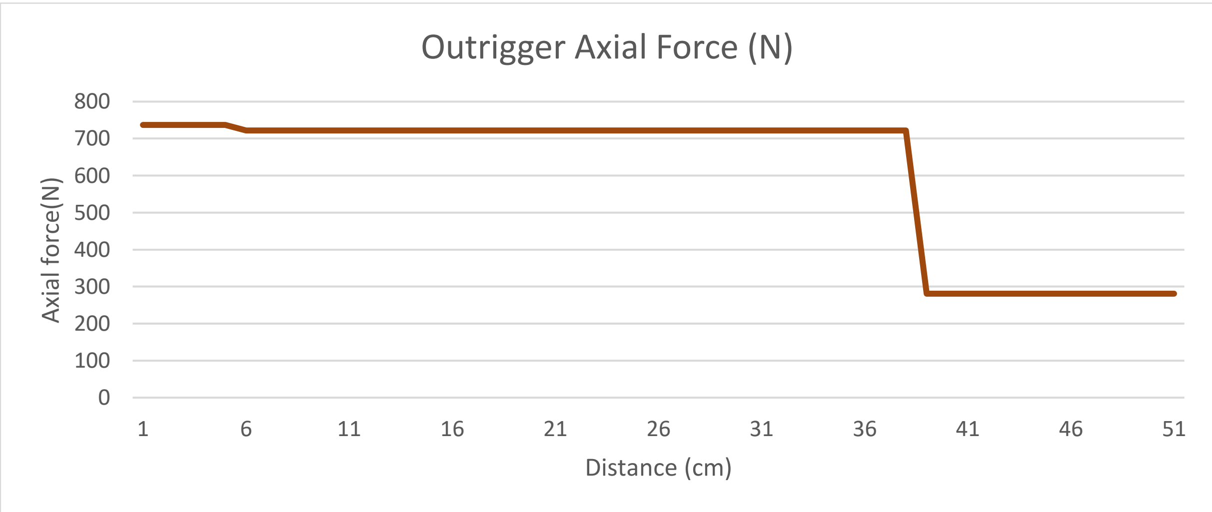
$$\uparrow + \sum y = 0, \qquad -24.43 + F_{strut} \sin(34) + 90 - 73.68,$$

$$F_{strut} = \frac{24.43 - 90 + 73.68}{\sin(34)} = 14.5N$$

$$\therefore F_{strut\,x} = 14.5\cos(34) = 12.02N \rightarrow +, \qquad F_{strut\,y} = 14.5\sin(34) = 8.1N \uparrow +$$

$$\rightarrow + \sum F_x = 0, \qquad 736.94 + 12.02 - 440.9 - F_x, \qquad F_x = 308.06N$$

3.2 Outrigger Axial, Shear and Bending moment diagrams:



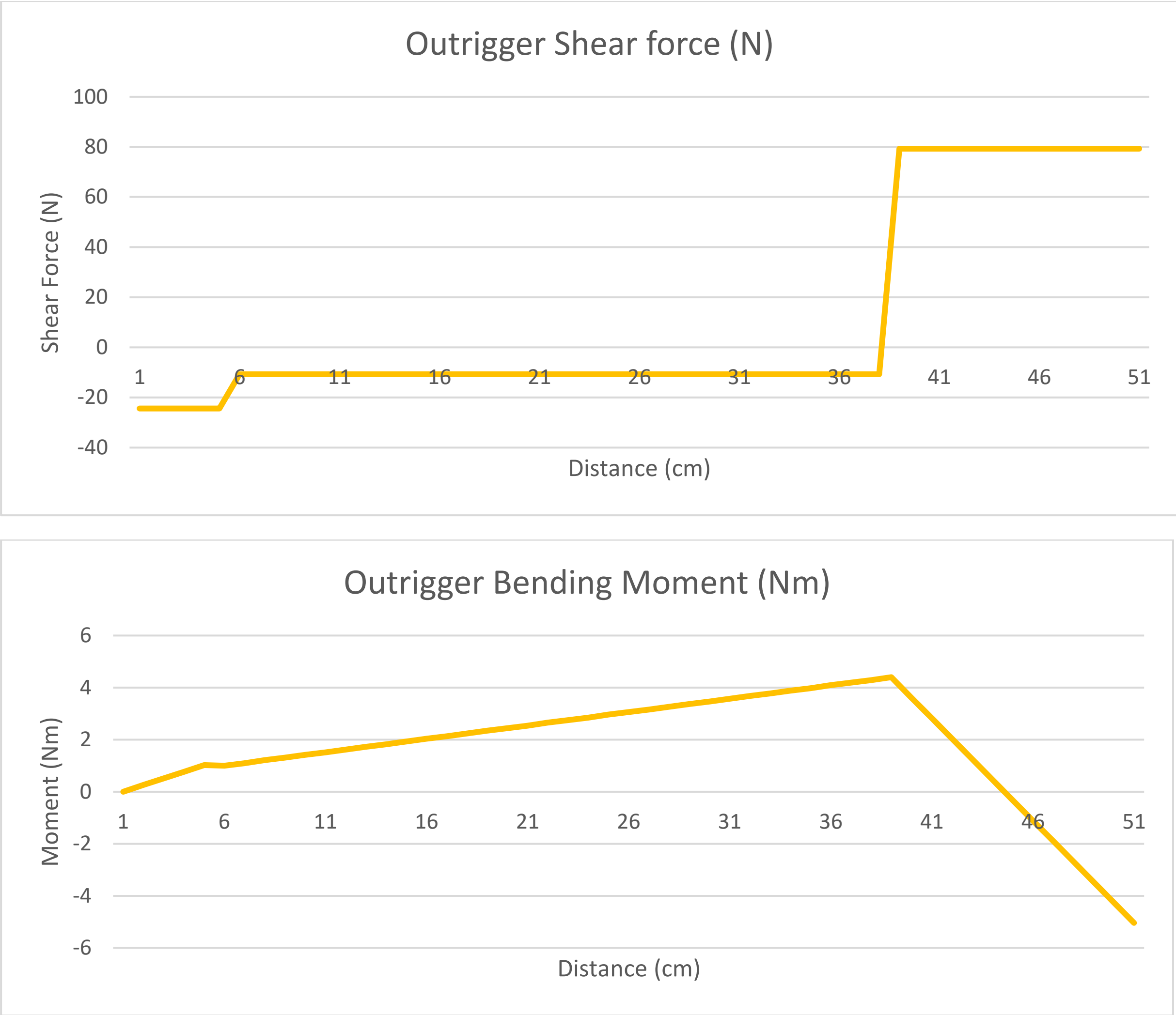


Figure 7: Outrigger Force diagrams.

3.3 Outrigger Stress calculations:

Maximum Axial and shear force in the outrigger occur at the gas strut mounting shaft. The maximum moment occurs at the weld joining the outrigger to the base plate, however, it is assumed that maximum stress will occur the at the gas mounting shaft as the shear and axial loading at this point is much higher. At this point This point is 430mm down centerline from the base plate with a moment of 4.4Nm, an axial force of 722N and a shear force of 80N. As the gas strut is welded on both sides it is assumed that raised stresses due to the 20mm hole will not occur. A point was taken at the top of the outrigger at this point for the following calculations:

$$\sigma_{nominal} = \frac{P}{A} = \frac{722N}{391mm^2} = 1.85Mpa$$

To calculate stress due to bending:

$$\sigma_{nominal} = \frac{Mc}{I}, \qquad I = 0.12 \times 10^6 mm^4$$

$$\sigma_{nominal} = \frac{4.4 \times 10^3 \times 25}{0.12 \times 10^6 mm^4} = 0.91Mpa$$

$$\tau = \frac{P}{A} = \frac{80}{391mm^2} = 0.204Mpa$$

Next using equations for maximum stress:

$$\sigma_1, \sigma_2 = \frac{\sigma_1+\sigma_2}{2} \pm \sqrt{{\tau_{xy}}^2 + (\frac{\sigma_1-\sigma_2}{2})^2} = \frac{2.76+0}{2} \pm \sqrt{0.204^2 + (\frac{2.76-0}{2})^2} = 1.52, -0.01$$

$$\tau_{max} = \sqrt{1.2621^2 + (\frac{27.279-0}{2})^2} = 1.39Mpa$$

$$\therefore safety\ factor\ for\ maximum\ stress = \frac{0.66 \times 350Mpa}{1.52Mpa} = 151.2$$

As this member is subject to large compressive axial loading an investigation into buckling also needs to be performed. As the lowest value for moment of inertia about any axis occurs about the y axis with a value of 0.0367$\times\ 10^6 mm^4$ this value must be used to calculate the radius of gyration (ρ):

$$\rho = \sqrt{\frac{I}{A}} = \sqrt{\frac{0.0367 \times 10^3}{391}} = 9.688mm^2$$

As beam is fixed/fixed $L_e = 0.5 \times l, \quad l = 514mm, \ L_e = 257mm$

Slenderness ratio = $\frac{L_e}{\rho} = \frac{257mm}{9.688mm^2} = 26.52$

$$(\frac{L_e}{\rho})_{tangent} = \sqrt{\frac{2\pi^2 E}{s_y}} = \sqrt{\frac{2\pi^2 \times 205 \times 10^3}{350}} = 127.22$$

As $\frac{L_e}{\rho} < (\frac{L_e}{\rho})_{tangent}\ Johnsons\ equation\ applies$:

$$S_{critical} = S_{material} - \frac{S_{material}^2}{4\pi^2 E} \times (\frac{L_e}{\rho})^2, = 350 - \frac{350^2}{4\times\pi^2\times205\times10^3}(26.52)^2 = 339.35Mpa$$

Critical loading occurs at $S_{critical}\ \times area = 339.35 \times 391 = 132 \times 10^3 N$

$$\therefore saftey\ factor\ for\ column\ buckling = \frac{132 \times 10^3}{736.94} = saftey\ factor\ of\ 179.$$

3.4 Strut Beam:

$$R_1 = \frac{Pb^2}{l^3}(3a+b), \qquad \frac{47.25 \times (150^2)}{350^3} \times (3 \times 195 + 150) = 18.23N \downarrow$$

$$R_2 = \frac{Pa^2}{l^3}(3b+a), \qquad \frac{47.25 \times (195^2)}{350^3} \times (3 \times 150 + 195) = 27.02N \downarrow$$

$$\rightarrow + \sum F_x = 0, \qquad 14.5 - 279.73 + F_x, \;\; F_x = 265.23 \rightarrow +$$

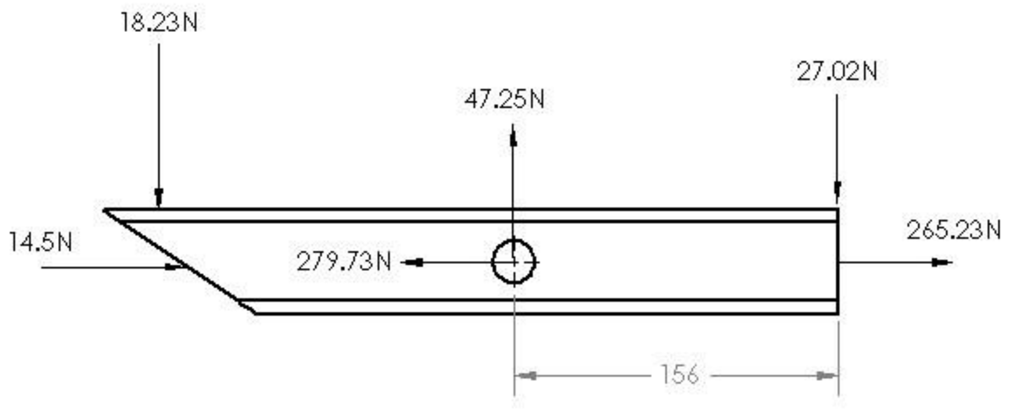
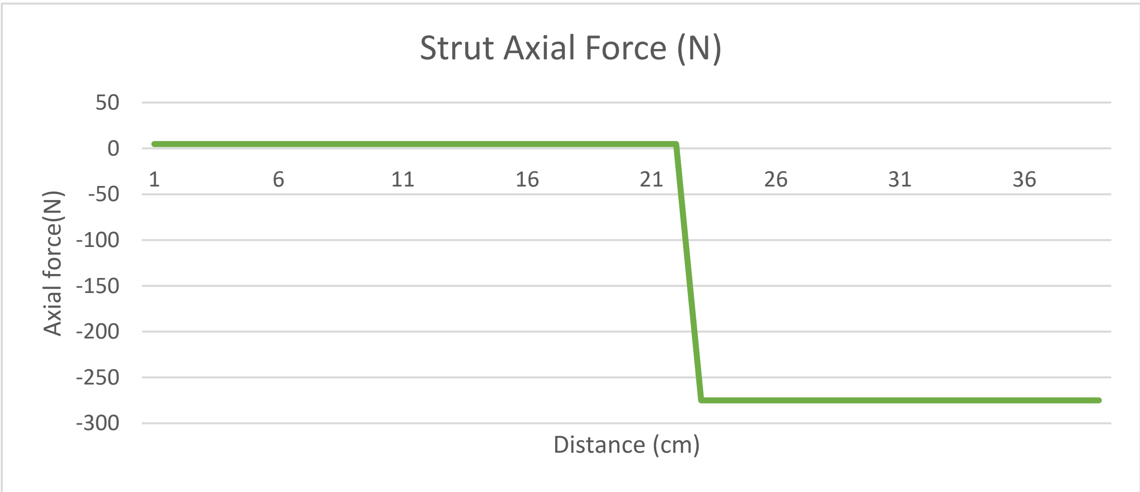
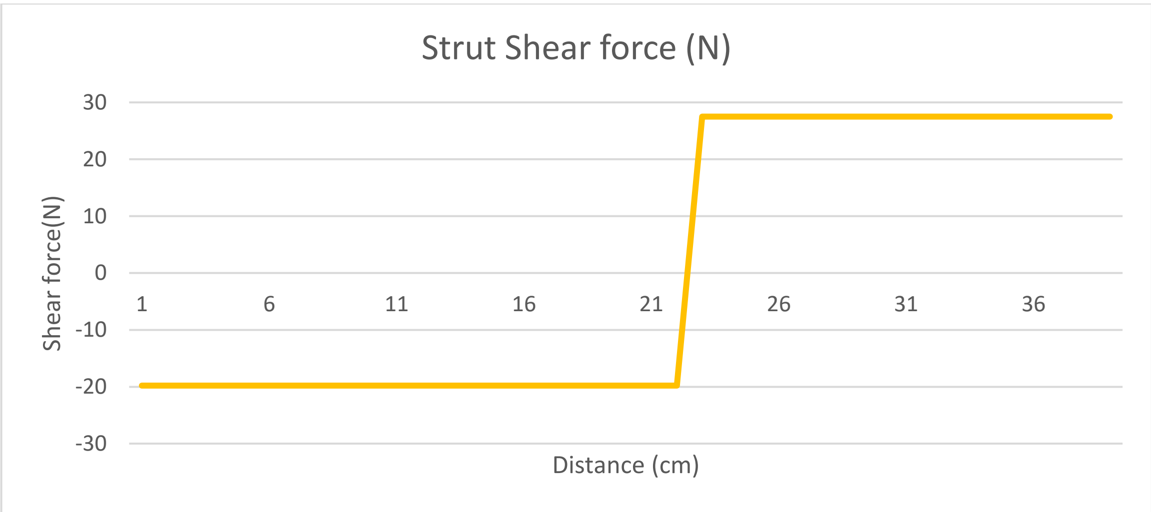


Figure 8: Strut Free body diagram.

3.5 Strut Force Diagrams:





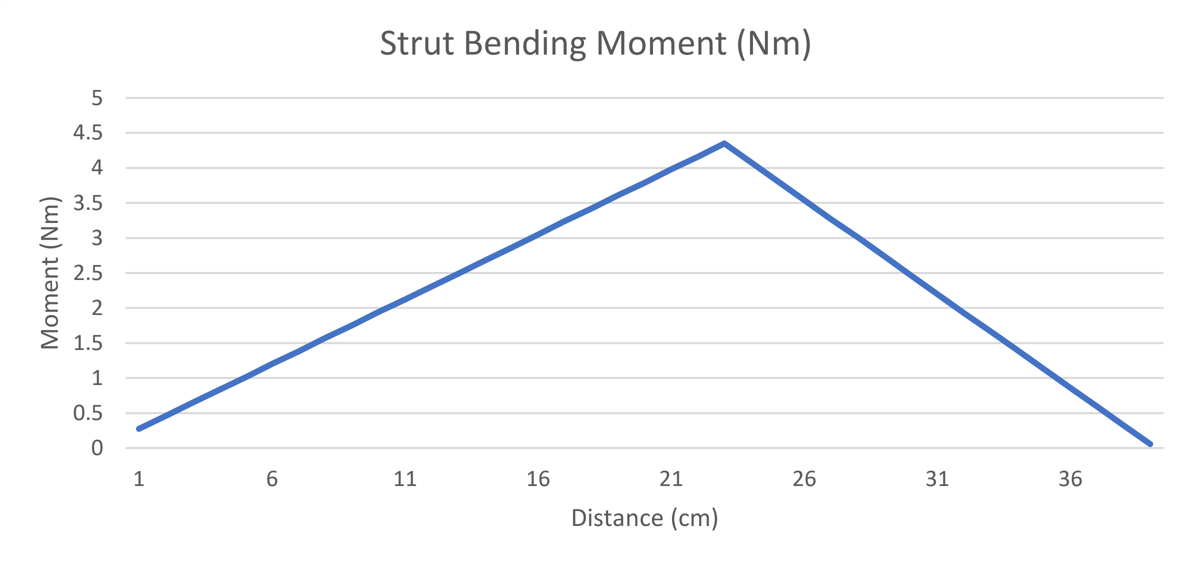


Figure 9: Strut Force diagrams.

3.6 Strut Stress calculations:

Maximum Axial and shear force and bending moment in the strut occur at the actuator mounting shaft. loading at this point is much higher. The actuator mounting shaft passes through a 20mm hole on both sides of this member so raised stresses will occur at this point.

$$\sigma_{nominal} = \frac{P}{A} = \frac{265.23N}{391mm^2} = 0.68Mpa, from\ chart, K_t = 2.23$$

$$\sigma_{raised} = K_t \times \sigma_{nominal} = 2.23 \times 0.68 = 1.52Mpa$$

To calculate stress due to bending:

$$\sigma_{nominal} = \frac{Mc}{I}, \qquad I = 0.12 \times 10^6 mm^4$$

$$\sigma_{nominal} = \frac{4.4 \times 10^3 \times 25}{0.12 \times 10^6 mm^4} = 0.91Mpa$$

$$\tau = \frac{P}{A} = \frac{27.05}{391mm^2} = 0.07Mpa$$

Next using equations for maximum stress:

$$\sigma_1, \sigma_2 = \frac{\sigma_1+\sigma_2}{2} \pm \sqrt{\tau_{xy}{}^2 + (\frac{\sigma_1-\sigma_2}{2})^2} = \frac{2.43+0}{2} \pm \sqrt{0.07^2 + (\frac{2.43-0}{2})^2} = 2.435,0$$

$$\tau_{max} = \sqrt{0.07^2 + (\frac{2.43-0}{2})^2} = 1.22Mpa$$

$$\therefore safety\ factor\ for\ maximum\ stress = \frac{0.66 \times 350Mpa}{2.435Mpa} = 94.87$$

4 Actuator Shaft:

The actuator shaft is machined from 20mm solid round 4340 steel before being normalized. This normalization results in a yield strength of 710Mpa. This extra strength is required in these shafts as they are under high loading conditions. As both the top and bottom actuator shafts are of the same construction, material and loading a single stress and safety factor calculation can be performed for both shafts. The machining performed on these shafts involves an 8mm slot to be cut through the shaft center and a face to be machined 2mm into outer surfaces parallel to this slot. A hole is then drilled through these faces to mount the actuator bolts. This area creates a situation where the weakest point in the shaft is also the highest load point, so stress and safety factor calculations are essential. Note in the following free body diagram the reaction forces are placed 10 mm from the end of the shaft. This shaft is supported closer to the center of the shaft on both sides, but this situation is a "worst case" scenario to calculate worst case safety factor. Weight has been neglected in this calculation as it is assumed it will have little impact on outcome.

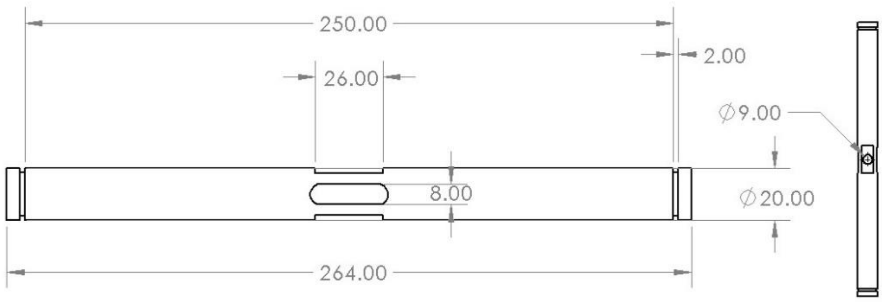


Figure 10: Actuator Shaft Machining.

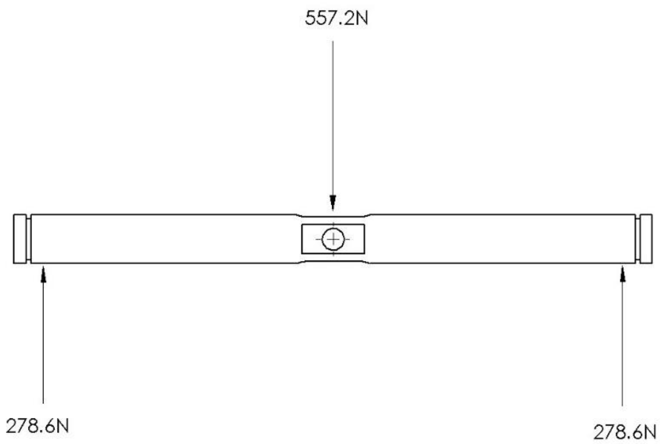


Figure 11: Actuator Shaft Free Body Diagram.

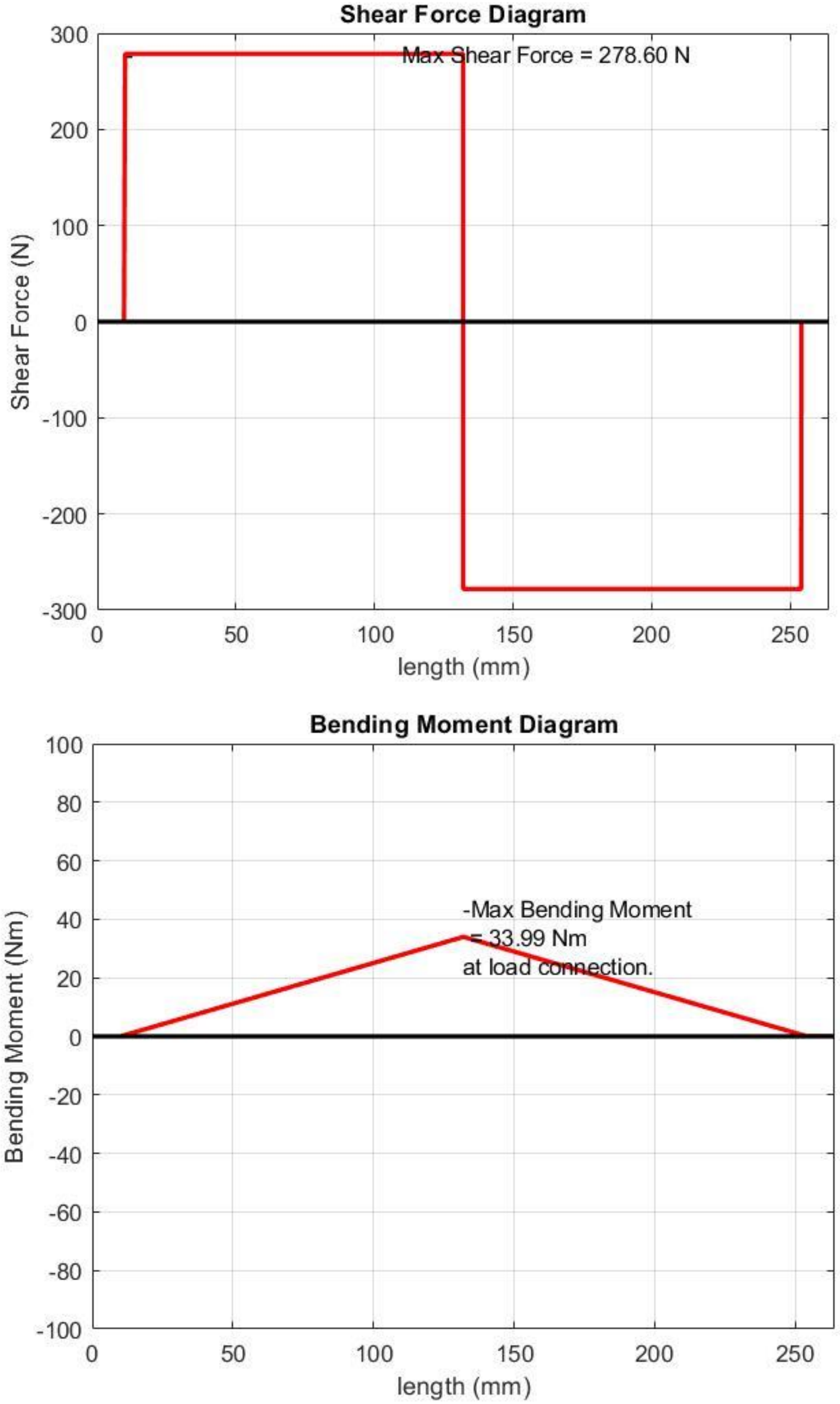


Figure 12: Actuator Shaft Shear and Bending Moment Diagrams.

To calculate stress due to bending:

$$\sigma_{nominal} = \frac{Mc}{I}, \qquad I = \frac{bh^3}{12} = \frac{2 \times 4 \times (18^3 - 9^3)}{12} = 3402mm^4$$

$$\sigma_{nominal} = \frac{39.99 \times 10^3 \times 8}{3402} = 94.04Mpa$$

$$\tau = \frac{P}{A} = \frac{278.6}{2 \times 4 \times (18 - 9)} = 3.87Mpa$$

Next using equations for maximum stress:

$$\sigma_1, \sigma_2 = \frac{\sigma_1+\sigma_2}{2} \pm \sqrt{\tau_{xy}{}^2 + (\frac{\sigma_1-\sigma_2}{2})^2} = \frac{94.04+0}{2} \pm \sqrt{3.87^2 + (\frac{94.04-0}{2})^2} = 94.2, -0.16$$

$$\tau_{max} = \sqrt{3.87^2 + (\frac{94.04 - 0}{2})^2} = 47.18Mpa$$

$$\therefore safety\ factor\ for\ maximum\ stress = \frac{0.66 \times 710Mpa}{94.2Mpa} = 4.97,$$

$$shear\ stress\ saftey\ factor = \frac{0.37 \times 710}{47.18} = 5.56$$

5 Mounting Plate Preparation:

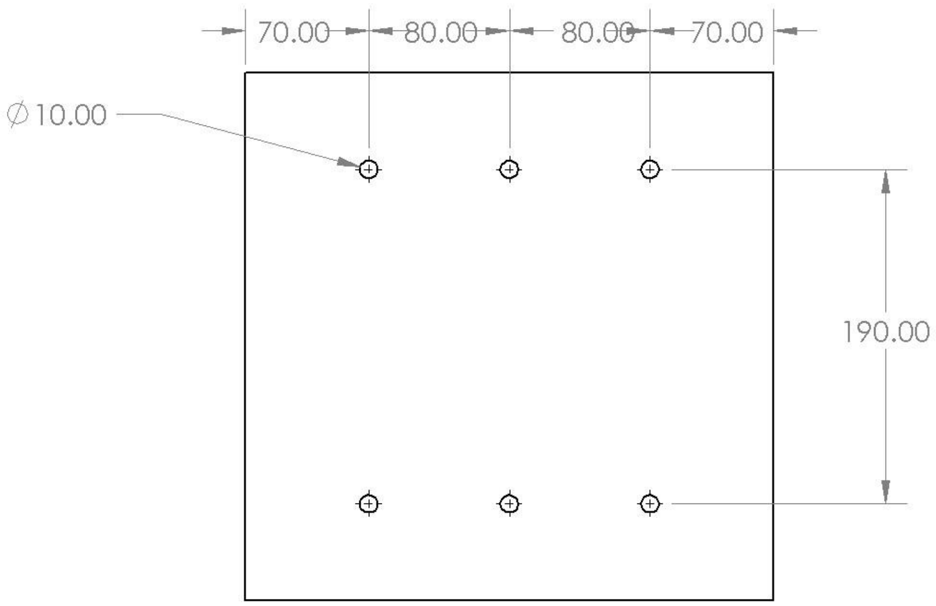


Figure 13: Mounting Plate Preparation.

6 End Effector:

The end effector in this design is constructed from welded 6061 5mm aluminium plate and rod. This material was chosen to minimize weight at the end of the arm assembly and to avoid iron contamination should the block require welding or surface treatment post process. The end effector holds the block by way of a linear solenoid that can be electrically activated upon impact.

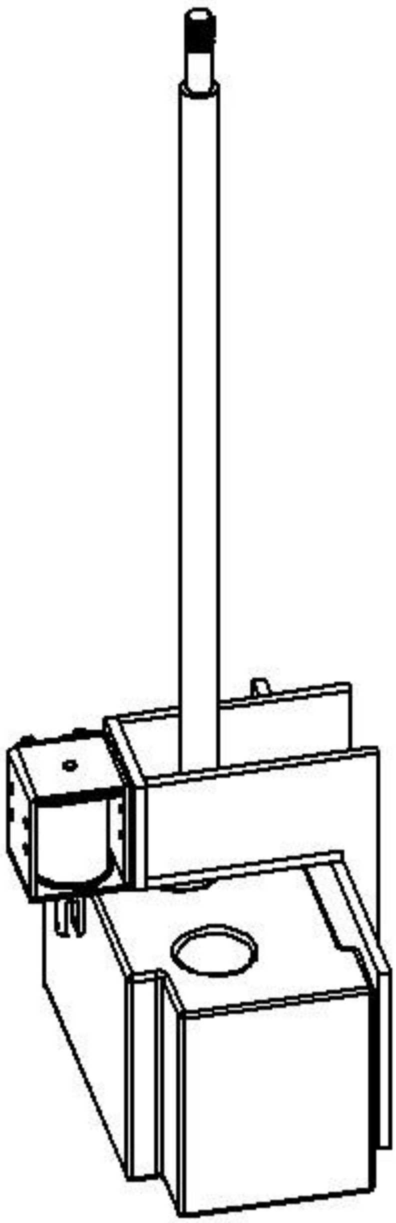


Figure 14: End Effector diagram.

Bibliography

AnalysisChamp. 2010. *Bar in tension or simple compression with a transverse hole.* 6 18. Accessed 05 26, 2019. http://analysischamp.com/stressbook/TABA2501.htm.

Madalia Steel. 2019. *Madalia Steel Stats.* 05 29. Accessed 05 29, 2019.

McMaster-Carr. 2019. *Pulling Gas Spring.* Accessed 05 25, 2019. https://www1.mcmaster.com/9502t3.

Standards Australia. 2016. *AS-3990-1993 Mechanical Equipment-Steelwork.* Standard, Standards Australia. Accessed 05 10, 2019. https://www-saiglobal-com.dbgw.lis.curtin.edu.au.

Standards Australia. 2016. *AS-4100-1998 Steel Structures.* Standard, Standards Australia. Accessed 05 10, 2019. https://www-saiglobal-com.dbgw.lis.curtin.edu.au.