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Liquid-solid Separation Thickener

Machine Design A 314 Design Project

PD4

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Group 8

Shankar Palamootil - 23687436

Sean La Grange - 23551127

Matthew Prozesky - 23641711

Nico Epler - 23910712

saam vorentoe · masiye phambili · forward together

Department of Mechanical and Mechatronic Engineering
Departement Meganiese en Megatroniese Ingenieurswese
Privaat Sak X1, Private Bag X1, Matieland, 7602
Tel: +27 21 808 4204 | www.eng.sun.ac.za

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Name: Shankar Palamootil	**Student no:** 23687436	**Signature:**
Sean La Grange	23551127	
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Nico Epler	23910712	

Abstract

This report dictates group 8's approach to designing a thickener system. It provides a background into thickeners and their usefulness, the objectives of the report and the motivation as to why this report is worthwhile. Using the calculations in tandem with the engineering specifications, group 8 was able to design a system that satisfied the customer requirements. The various sub-systems and entire assembly are included.

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Not Applicable as we did not use any symbols in the main document.

1 Introduction

1.1 Background (SLG)

Water is a vital resource and is often used in the mining industry for various processes. It is therefore vital that it be recycled and used as much as possible. An example of this is using water as a medium to pump ore out of mines. Once the slurry is out of the mine it needs to be separated so that the ore can be collected. A thickener takes in the slurry mixture and, through a gravitational separation process, separates the mixture into a clearer water and sludge containing a high concentration of the ore sediment. This process is aided by a set of rotating rakes which scrapes the sediment towards an outlet port where it leaves as a sludge. This rake system is connected to a torque tube and motor which allows the rakes to rotate about the tank.

1.2 Objectives

Select and design a suitable drive system, mounting frame, torque tube, and rakes for a thickener. The complete system will be used in a range of mining applications.

1.3 Motivation

This project is to be done so that a robust, standardised design for a thickener can be developed. It will be easily manufacturable and applicable to general mining applications, allowing it to be engineered and operational in a timely manner.

2 Summary of Engineering specifications

Numbers taken from PD1

Table 1: Summary of engineering specifications

Number	Description	Target	Range	Unit	Customer Requirement
	Torque Tube				
1	Maximum torque that the torque tube can withstand	261 * 10^3	(135 to 261) * 10^3	Nm	XII
3	Torque tube length	10.22	< 10	m	III, XV
4	Torque tube diameter	0.46(OD) 0.326(ID)	0.3 to 0.7	m	III, XII, XIII, XIV, XV
	Mounting Frame				
8	Area of mounting frame	6.052	5-7	m^2	XVIII
	Drive System				
9	Power output of motor	3075	1800-4350	W	VIII
10	Overall output speed	0.127	0.127 – 0.143	rev/min	IX
13	Efficiency	89	85 - 95	%	X
	Rakes				
17	Gap between rakes and tank wall	0.05	0.05-0.15	m	XXVI
19	Introducing safety factor	5	3-7	n/a	VII, XXVI
	General				
23	Corrosion resistance	OS3	OS2 – OS4	n/a	I, II

3 Sub-system 1: Drive system (MP)

Concept layout

The drive system is required to supply the power needed for the thickener to function and transfer it to the torque tube. To achieve this, the system in Figure 1, was generated. A gearmotor was vertically mounted, allowing it to be centred close to the middle of the mounting frame, decreasing the moment and deflection that it could cause. The end of the shaft connected to a pinion gear, through two keyways to split the transfer of torque and a shrink fit. This pinion gear transfers the torque and speed delivered by the shaft to a slew ring, scaling it to the required amount through gear ratios. An internal slew ring was chosen to allow the torque tube to be attached to the bottom of the rotating ring, while the outer ring could be supported by the mounting frame. This shielded the inner workings of the slew ring from damaging elements. Through these interfaces, the torque generated by the motor can be transferred to the torque tube, turning the rakes and allowing the thickener to function. Detailed calculations were then done to aid the selection of specific gearmotor combinations, pinion gears and slew rings.

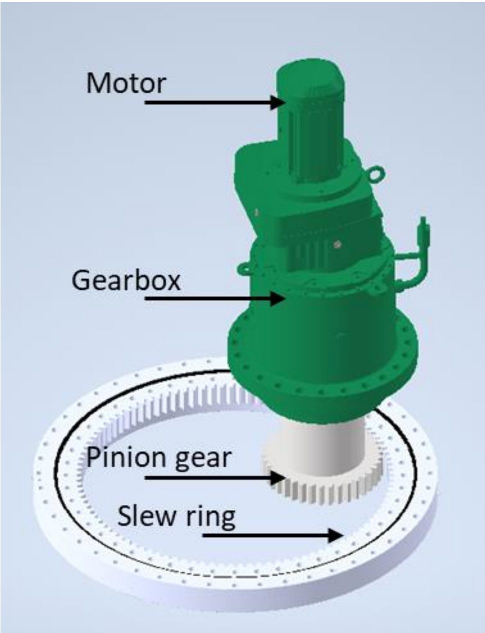

Figure 1:Drive System

Power calculations and force analysis

Table of important calculation results

Variable	Value	Units	Equation
Power needed	4.35	kW	A1.1
Pinion gear pitch diameter	385	mm	A1.2
Pinion gear teeth	38	teeth	A1.3
Module	10.1	n/a	A1.4
Final output speed	0.1276	Rev/min	A1.5
Force of motor on mounting frame	11.03	kN	A1.6
Force of slew ring on mounting frame	193.6	kN	A1.7
Safety factor of axial force	162.7	n/a	A1.8
Safety factor of radial force	5.553	n/a	A1.9
Deflection of shaft	0.1496	mm	A1.10
Safety factor for moment on slew ring	33.36	n/a	A1.11
Safety factor of keyway stress	4.938	n/a	A1.12

Table 2: Calculation results for sub-system 1

The first step in designing the drive system, is to determine the power it must supply. For this system, a total of 4.35 kW of power must be supplied at the highest operating point. A motor with a total power output of 5.5 kW was selected, as the other closest motor produced 4 kW which would not have been sufficient.

Next it was necessary to determine the optimal combination of a gearbox and slew ring to output the desired torque at the required speed. An iterative method was set up, where different slew ring diameters could be combined with 5.5kW gearbox-motors that could output different speeds and torques. Using multiple combinations, it was possible to determine the ideal pitch diameter, 383.2 mm, to achieve the required speed, whereafter the number of teeth required on the pinion was determined to be 37.7. However, teeth are only measured in whole numbers, to adjust this the diameter was rounded up to 385 mm and the number of teeth to 38, allowing the module of 10.1 to match that of the slew ring with a negligible error of 0.02. The change in diameter was observed to have a negligible effect on the output speed of the motor for such a slight increase. From these calculations the XT 16297001 slew ring with an internal gear from Kaydon Bearings was chosen to be combined with the PF052RF127DRN132S4 motor-gearbox from SEW Euro-drive.

With the drive system selected, a force analysis was performed to ensure it could withstand all relevant forces acting on it. The motor and the slew ring would be

mounted at two separate points. The mounting frame would need to support 11.03 kN and 193.6 kN at the motor and slew ring connection respectively. The axial force experienced by the shaft was 162.7 times smaller than the maximum allowable force. The radial forces were calculated to be 5.553 (equation A1.9) times smaller than the maximum allowable force. Deflection of the shaft caused by the radial force was 0.1496 mm, small enough not to affect the shaft integrity and placement. The moment generated by a notional load at the bottom of the torque tube was calculated during the Sub-system 3 calculations and was 33.35 times smaller than the maximum allowable moment on the torque tube.
No standard pinion gears could be found that would fit the requirements needed, therefore would need to be designed and made in the factory. A stress analysis was performed on the pinion gear to determine the stresses it experienced so a material could be selected for it to be made of. The highest stress concentration was found to be in the keyway of the gear, where AISI 4140 steel, a commonly used metal for gears, produced a safety factor of 4.938. This was sufficient to ensure that such an integral part of the drive system would not fail, allowing the gear to be produced using AISI 4140 steel.

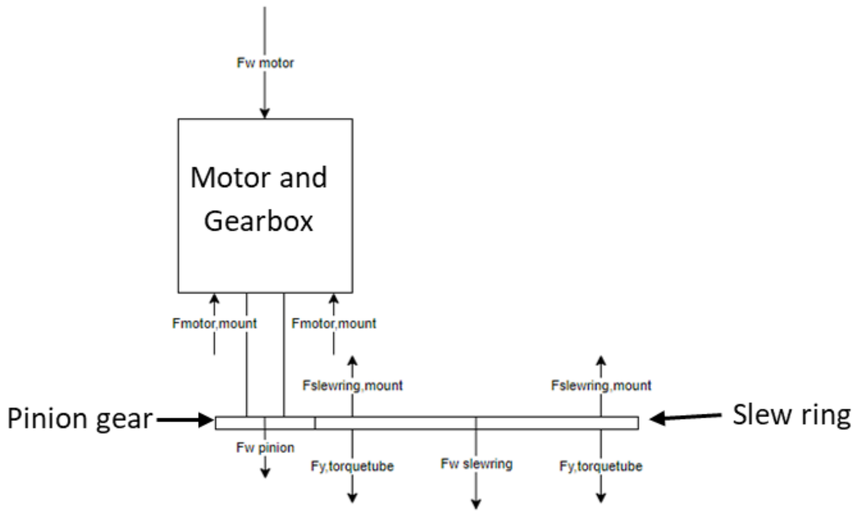

Figure 2: Free body diagram of drive system

<u>Table describing the free body diagram in Figure 2</u>

Force	**Explanation**
Fw motor	The weight of the motor
Fw slewring	The weight of the slew ring
Fw pinion	The weight of the pinion gear
Fmotor,mount	The reaction force from the mounting frame on the motor
Fslewring,mount	The reaction force from the mounting frame on the slew ring.
Fy,torquetube	The force of the torque tube on the slew ring.

4 Sub-system 2: Mounting Frame (SP)

The mounting frame is responsible for securing the motor, torque tube and slew ring to the thickener. It must support the weight of the entire system and allow multiple moving parts to interface with each other without interfering with their motion.

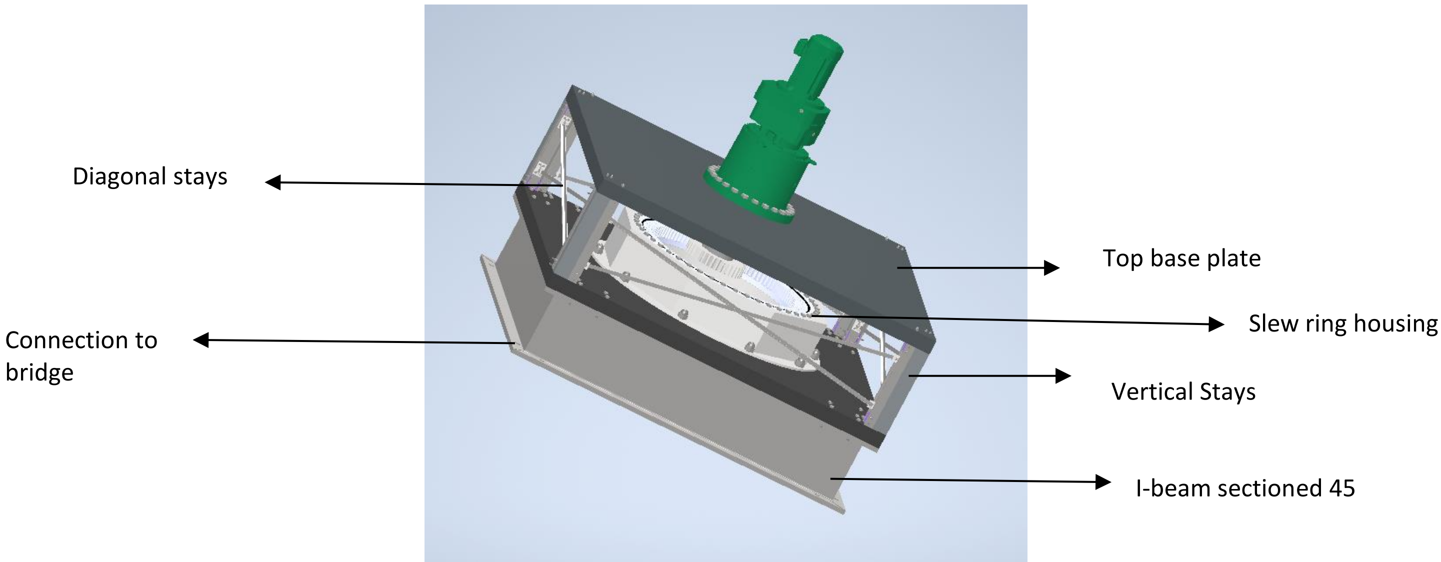

Figure 3: Mounting Frame

The mounting frame design that was chosen is meant to be built from the bottom up. The base consists of 4 standard I-beams sectioned at 45 degrees welded together to form a square. The I-beams are made of 440c stainless steel. It was first decided that steel would be used as the material needed to have a high strength and elastic modulus (A2 ref1.1). It was further required that the beams needed to be corrosion resistant due to the nature of a thickener system's working conditions. This was why stainless steel with a chromium content of 12% was chosen. This was the minimum requirement for corrosion resistance, the least expensive option compared to beams with a higher chromium content and because the beams need to be welded together, a chromium content <18% was required. The thickness of the web is 18 mm and the thickness of the flanges of the I-beam are 32 mm. Using table 9-6, the respective weld sizes should be 6 mm for the web and 8 mm for the flange. On top of the I-beam structure sits the first base plate. The base plate is constructed out of alloyed grey cast iron. The cast iron was chosen mainly due to its machineability and compressive strength capabilities. The machineability factor was incredibly important as most of the bolt holes are machined into the base plate. The grey cast Iron was also alloyed with chromium and nickel to increase strength, hardness, wearability, heat resistance and corrosion resistance. The reason both alloys were added was to ensure that the machinability was not reduced (pg. 75 Shigley's). The top plate is made of the same material. The vertical stays between the two plates are a composite beam consisting of 2 channel beams and a square beam. The composite structure was chosen for its strength in compression and torsion. The channel beams and square beam will be made from 440c stainless steel for the same reasons stated above for the I-beams. For added support against torsion, diagonal stays (made of L-beams) are used to distribute the torque better. To decrease the number of welds, brackets were mostly used to connect the different beams and plates together. The bracket A2 ref1.2 was used to connect the vertical stays to the base plates and the bracket A2 ref1.3 was used to connect the diagonal stays to the composite beams. These stays and the housing for the slew ring will be casted and constructed out of 440c stainless steel.

The system was considered as a rigid system and this assumption is supported by simple deflection calculations (A2 ref1.4). Due to this assumption, none of the internal calculations were required. The calculations that were done include the calculations of the reaction forces and the bolt diameter calculations. The bolt calculations were done first as the reaction forces due to the shear were required for reaction force calculations. There were 16 bolts between the bridge and mounting frame interface. The bolts were spaced in such a way that they did not interfere with the welds on the I-beams. It was previously decided that stainless steel bolts with a property class of 8.8 would be used. This decision was made on the basis that bolts in this property class was used in actual thickener systems. Using the table 8-1 from Shigley's in tandem with the bolt calculations in chapter 8. The maximum force on a bolt due to the torsion and notional load was 11112 kN. It was stated that the notional load only acted on the normal load case but as the cause of the notional load is unknown, it was thought better practise to find the shear when both max torque and the notional load was in effect. Thus, if the bolts could withstand this theoretical case, it could withstand any other configuration of the load. Using this load case, the required diameter of the bolts was 8mm (M8). Practically speaking this value for the diameter is too small even though theoretically it would suffice. In industry the standard size of the bolts started from M16. With this in mind, the size of the bolts chosen was M16. This gave a safety factor of 18.7. This might be quite high but due to the nature of the mounting frame and how much stress it undergoes, having a high safety factor is necessary to ensure that it has a long-life span. The stress on the bolts due to the out of plane bending moment of the notional load was also calculated and a safety factor of 35.5 was achieved. This safety factor was larger than the shear safety factor and thus did not affect the system meaningfully. The bolts that were used were Plain 8.8 M16 Steel Hex bolts that can be bought form a South African company called RS. The accompanying Nuts are RS Pro, Plain Stainless Steel M16b hex nuts (DIN 934).

To find the reaction forces that are transmitted to the bridge, simple force analysis was used. (Refer to A2 ref1.5 for detail calculations). The weight of sub-system 2,3,4 and the slew ring were distributed equally between the four reaction forces because they were located centrally. The drive system was not located centrally and thus exhibited different forces on each of the reaction forces. In the x direction, (axis as per simplified free body of mounting frame on second page of Appendix A2) the reaction force was 30,5kN. This was due to the T_{max} and the notional load. In the y-direction the reaction force 30.8 kN, also caused by T_{max} and the notional load. In the z- Direction, the total reaction force was 258 kN (directions can be found in calculations).

As stated above the mounting frame has been designed to be built from the bottom up. The vertical stays have also been designed modularly and thus the only work that that needs to be done on site is the bolting to the bottom base. A gap of at least 2 cm is also kept between each bolt to allow easy assembly. If a crane is available, the whole mounting frame can be assembled away from site and then using the shackles on the bottom base plate it can be lowered onto the thickener system.

5 Sub-system 3: Torque Tube (SLG)

The torque tube is responsible for transmitting torque from the slew ring, which is connected to the motor, to the rakes. The torque tube is also responsible for supporting the rakes and allowing them to rotate about the thickener. The torque tube interfaces with the other sub-systems in order to perform these functions. The torque tube is connected to the slew ring by a flange which is welded onto the torque tube and bolted onto the slew ring. The rakes connect to the torque tubes through two different types of clevis castings. The one being at the bottom of the torque tube where the rakes connect (Refer to Figure A3.10) and the second being higher up the torque tube where the vertical stays connect (Refer to Figure A3.11).

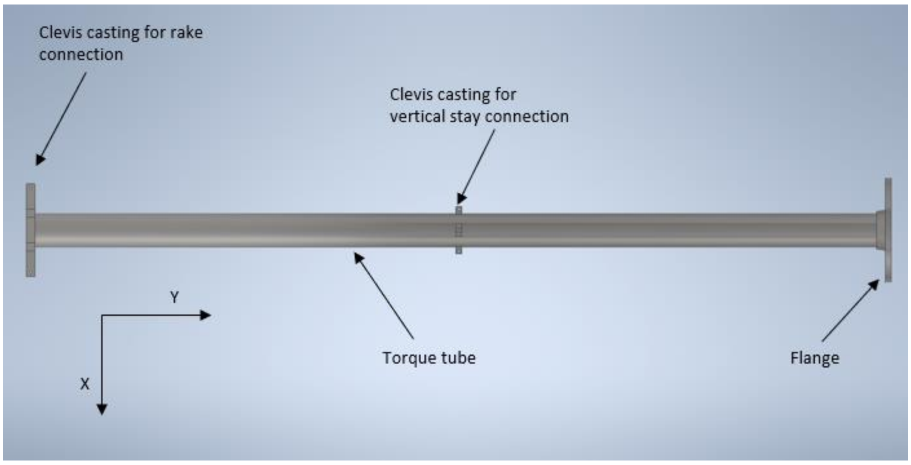

Figure 4: Complete Torque tube

Due to torque tube being submerged into a corrosive environment the material of Stainless steel 316 was selected. This material also has a high modulus of elasticity as well as high tensile strength which is necessary due to the large normal and shear stresses that the torque tube will experience. A standard pipe of 406x326mm from hollowbar.co.za was chosen for torque tube. A force analysis was then performed on the torque tube to determine whether it would be sufficient for the loads applied according to various failure criteria.

Force analysis of torque tube

A force analysis of the torque tube needs to be conducted to determine whether the torque tube will be sufficient for various methods of failure. When analysing the forces, the likely methods of failure for the torque tube were found to be static failure and fatigue failure. The forces acting on the torque tube for three different scenarios were analysed. These three scenarios are: normal operation, overloaded operation, and when the thickener is empty and not rotating. When viewing the free body diagram below as well as the force and moment diagrams we can find the point on the torque tube that will experience the most stress.

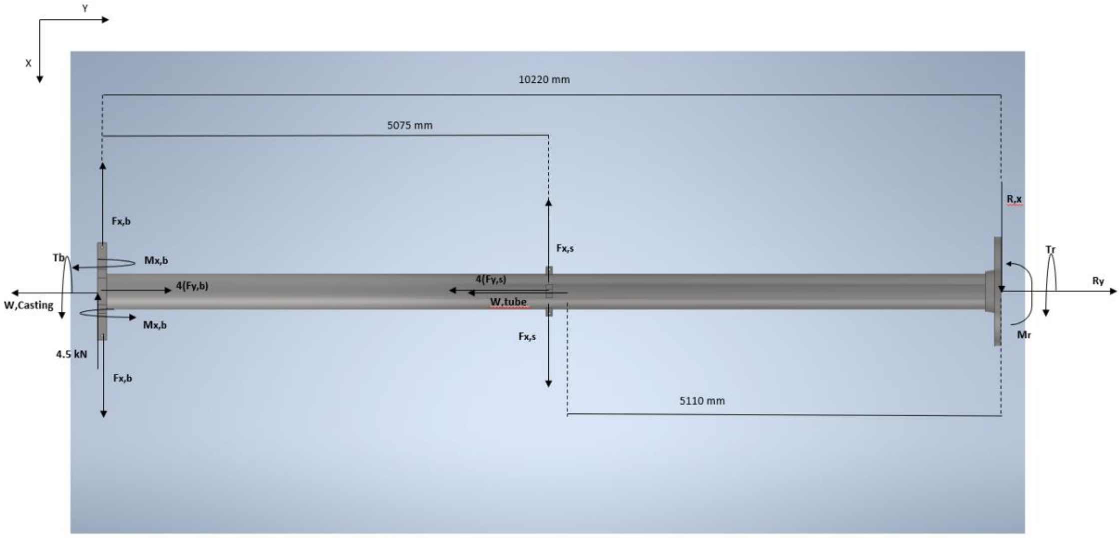

Figure 57: Free Body Diagram of Torque Tube

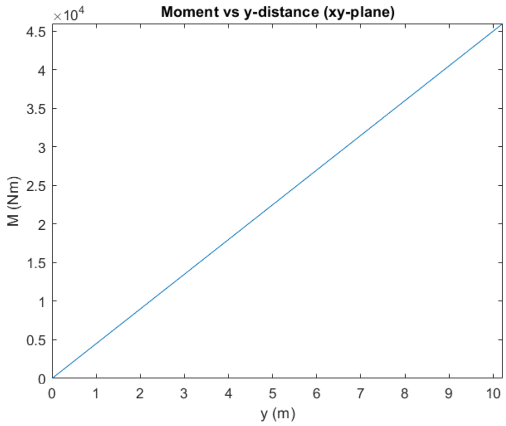

Figure 66: Bending Moment about the z-axis

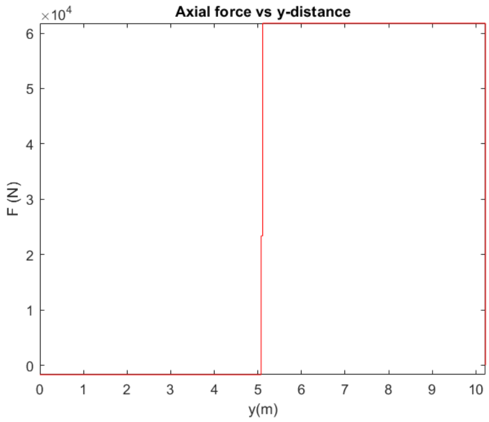

Figure 5: Axial Force in the y direction

As an example, from the force and moment diagrams above we find that the point on the torque tube which experiences the most force and moment is at the top where the shoulder of the flange meets the torque tube. The torque acts uniformly throughout the torque tube and creates a shear stress in the torque tube. The lateral non-rotating notional load acting at the bottom of the torque tube also creates a shear as well as a moment in the torque tube. The shoulder of the flange also causes stress to concentrate in the torque tube where they meet, and therefore this point was chosen as the critical point of the part. The combined stresses caused by these forces are then calculated in order to test whether the torque tube will fail.

When the static forces were analysed for the three different conditions it was found that the overloaded operation created the largest stress in the and therefore had the lowest safety factor. Using the Tresca theory it was found that the safety factor in this case was 3.159.

In order to see if the torque tube fails due to fatigue, we analyse the scenario where the system is in normal operation, as this is the only condition with dynamic loads. These loads being a fluctuating torque, where the torque fluctuates 10% above and below its mean value of 60% of the T_{max}, as well as a fully reversible bending moment and shear force created by the notional load. When applying the von Mises theory in order to calculate the equivalent stress as well as the fatigue criterion, a safety factor of 2.601 was achieved.

The deflection and slope of the torque tube, caused by the notional load, were calculated. A maximum deflection of 0.01065m and a maximum slope of 0.00156251 rad was found. When comparing this to the deflection of the rake system it was found that this amount of deflection of was allowable and did not interfere with the operation of the thickener.

The most critical weld in this sub-system was found to be the weld that connects the flange to the torque tube as it is responsible for withstanding the largest shear stresses. A static as well as a fatigue failure analysis was performed for the weld. This was done using the Tresca theory, as the weld is experiencing mainly shear stress and the Tresca theory will therefore be the most accurate analysis method. When performing the static failure analysis, a safety factor of 2.5399 was found for the weld, once again during the overloaded condition. For the fatigue failure analysis, a safety factor of 1.011 was found. However, for this analysis was done in a conservative manor. The strength of the material of the object was used for the weld during the analysis. However, in reality the weld metal is a mixture of the object material and the actual weld material. The actual weld material has a higher strength than the object material therefore, due to the weld material being a mixture its strength will be higher than the strength used for the analysis. So, in practice the actual safety factor will be higher and therefore the weld will be sufficient for the load application.

Calculations were also performed in order to ensure that the pins used for the clevis castings are thick enough to support the forces applied to them by the rake as well as the vertical stays. A 30mm diameter pin was selected for the rake connection which resulted in a safety factor of 6.216. A 25mm diameter pin was selected for the vertical stay connection which resulted in a safety factor of 2.953.

6 Sub-system 4: Rakes (NE)

Because the rakes are completely submerged in the corrosive sludge, and since the rakes must withstand huge shear forces and bending moments, a material with a large Elastic modulus had to be selected. Due to its strong, corrosion resistant and non-magnetic properties, Stainless steel 316 (AISI 316) was decided on. It has also been decided that 9 blades per rake arm would be used, to reduce the production time and cost of the rakes, by reducing the weld distance needed for attaching the blades, while keeping the deflection of the rake blades to a minimum. The final design of the rake arm welded assembly is shown in figure 3.

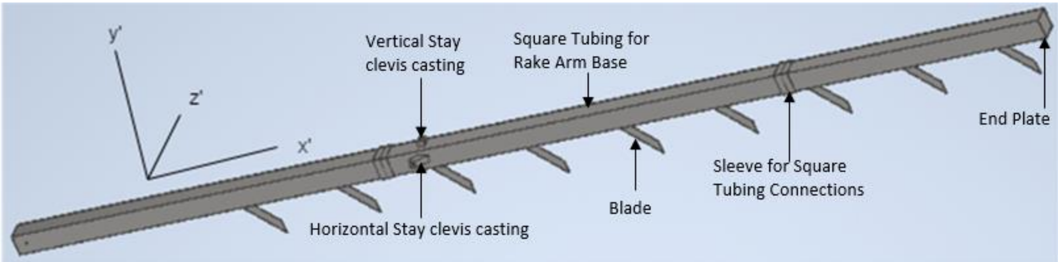

Figure 88: Welded Rake Arm Assembly

By using the given maximum torque and considering the standard and overload operating conditions, force analyses have been done respectively (as seen in appendix A4). It was found that the overload condition is the determining factor, with respect to the blade thickness. Therefore it was found that a blade thickness of 20 mm would be optimal, yielding a high safety factor of 11.0 (Tresca criterion) (as seen in appendix A4, ref A4.1) and having a maximum deflection of only 3mm (as seen in appendix A4, ref A4.2), compared to a 10mm thickness, having a safety factor of only 2.7 (Tresca criterion) and a big deflection of 29mm.

Furthermore, the rake arms were then analysed by looking at their normal operating conditions static and fatigue failure, the overload condition, and their static state, when there is no slurry in the thickener (since in this case, there is no buoyancy force opposing gravity). By comparing the Bending moments and Shear forces of the normal and overload condition (as seen in appendix A4, ref A4.3), it was clear that the normal load under static failure was not the critical condition. For this reason, the normal condition under static failure was not used to calculate safety factors.

By conducting a shear stress and bending moment analysis, the overload condition and the static, empty condition were compared. By setting up Macaulay functions (Example in appendix A4, ref A4.4) and plotting them using MATLAB, the maximum

Bending moment, shear force and deflections (as seen in appendix A4, ref A4.5) could then be deducted. And by constructing normal and shear-stress equations, it was displayed that all Maximum Normal and Shear-stresses were larger for the overload condition, when compared to the empty tank condition (as seen in appendix A4, ref A4.6). Therefore, the safety factor calculations for the rake arm were only conducted for the overload condition and the normal operating condition, using fatigue failure, excluding the empty tank condition. However, the greatest forces in the vertical stays were found to occur when the tank was empty, and thus the stay safety factor was based on the 36.16kN (as seen in appendix A4) axial load under this condition.

By converting the overload conditions plane stresses (as seen in appendix A4, ref A4.7) to principal stresses, the safety factor could be found using the Tresca and Von Mises yielding criterion. Finally, the 300x300mm square tubing, with a thickness of 10mm, chosen earlier, has been confirmed to be a suitable section. Both yield criteria gave rise to a similar factor, being 3.65 (see appendix A4, ref A4.8) and the resulting deflection graphs in the z'-direction and y'-direction are shown in figure 9. and figure 10. respectively. For the stays, it was decided to use Diameter 48mm round tubing with a thickness of 2mm, which yielded a safety factor of 4.66, using the more conservative Tresca yield criterion (see Appendix A4, ref A4.9)

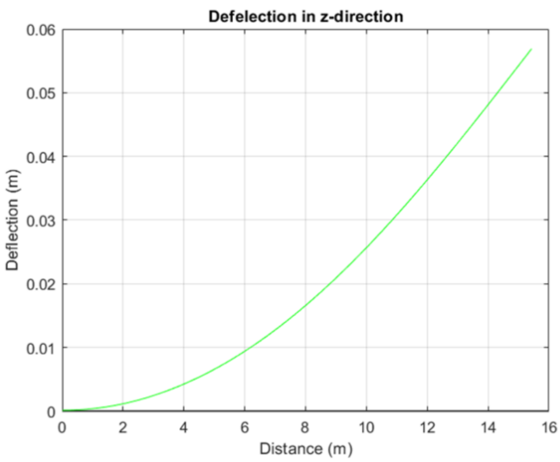

Figure 99: Deflection of Rake Arm in z'-direction

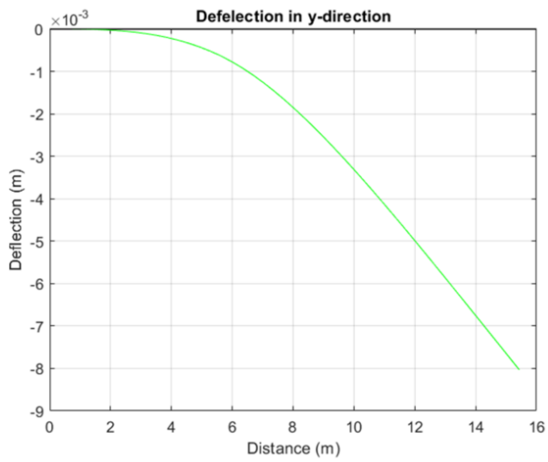

Figure 1010: Deflection of Rake Arm in y'-direction

To ensure that this safety factor was not exceeded by the normal condition under fatigue failure, some more fatigue calculations have been conducted (as seen in appendix A2, ref A4.10). Using the Von Mises failure criterion, the safety factor under fatigue failure, for normal operating conditions was found to be a reliable 4.8 (see appendix A2, ref A4.11).

After fully confirming the dimensions and material for the blades, rake arm tubing and stays, the total deflection of the end of the rake arm, towards the bottom of the tank was investigated. Due to the notional load applied to the Turque tube, a slope was discovered at the end of the torque tube, which together with the maximum deflection of the rake arm was suspected to cause interference with the tank. However, after calculations, the maximum deflection was found to be 46.6mm, while the buffer between the tank and the rake arm was 50mm. Furthermore, this was deemed safe, since in the calculation the absolute maximum deflection was the maximum deflection taking from the overload condition combined with the slope of the torque tube. However, the notional load to the torque tube is only applicable under normal operating conditions and will never superimpose with the overload deflection, indicating that even the calculated value would never be achieved (see appendix A4 for calculations, ref A4.12). With all the dimensions know, the complete rake assembly is shown in figure 11.

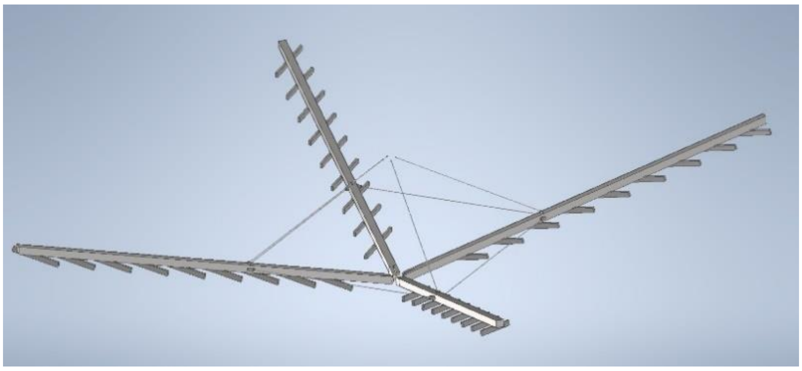

Figure 1111: Complete Rake Assembly

Finally, the welding thicknesses used for the welded assembly of the rake arm were identified. The weld thicknesses for connecting the blades to the rake arm were decided to be given a thickness of 6mm, while the attachment of the endplates and the sleeve (for connecting the square tubing to achieve a long rake arm) were given a thickness of 5mm. Because of the large size of the clevis castings, very thick welds of 12mm needed to be used to attach these to the rake arm (as seen in appendix A4, ref A4.13).

7 Conclusion

The customer requirements were converted to quantifiable values, allowing us to gain a few insights into the most important areas and problems this design faces. After careful consideration, we have generated a design concept that will achieve the objectives of this project. Combing standard parts from respected suppliers for motors and bearings, and our simply designed structures, the concept is easily scalable and applicable to different mining applications. Using basic calculations, the load paths have been considered and suitable designs have been chosen to account for them. These designs have then further been analysed in order to assess whether they will undergo static or fatigue failure.

8 References

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Appendix A Calculations

A.1 Sub-system 1: Drive system (MP)

Keys:
D = diameter
T = torque
n = revolutions per minute
P = power
ω = radians per second
m = mass
F = force
ρ = density
N = safety factor
A = area
V = volume
E = modulus of elasticity
I = second moment of area
L = length
α = angle
g = gravity
σ = stress
M = moment
K = stress concentration factor
τ = shear stress
J = second polar moment

Drive System calculations:

1.Maximum required torque and speed:

Variables(Given in project briefing):

$D := 30\ \mathrm{m}$

$K := 290\ \frac{\mathrm{N}}{\mathrm{m}}$

$\omega := \frac{1}{60}\ \frac{\mathrm{rad}}{\mathrm{s}}$

Calculations:
The maximum K value given and the fastest speed value given were used to calculate the maximum amount of power the drive system would require in an overload state.

$T_{need} := K \cdot D^2 = 261\ \mathrm{kN\ m}$

$P := \omega \cdot T_{need} = 4350\ \mathrm{W}$ A1.1 (maximum power drive system must supply)

Euro-drive supplies 4kW or 5.5kW motor, which are the closest to the range we will need as per the power required. A motor form the 5.5 kW range must be selected so as to ensure evn when in overload, it will provide enough power to the system.

2.Prelimanry motor selection:

Set vairables:

$T_{need} = 261\ \mathrm{kN\ m}$

$n_{minmum} := 0{,}127\ \frac{\mathrm{rev}}{\mathrm{min}}$

Adjustable variables (Provided by SEW-Eurodrive catalogue and Kaydon bearings website):

$D_{motor} := 210\ \mathrm{mm}$ PF052RF127DRN132S4 gearmotor

$T_{motor} := 124060\ \mathrm{N\ m}$

$n_{motor} := 0{,}35\ \frac{\mathrm{rev}}{\mathrm{min}}$

$D_{slewring,pitch} := 1056\ \mathrm{mm}$ XT 16297001 internal gear slewring

$Teeth_{slewring} := 104$

Calculations:

There are multiple output torque and speed combinatations for the 5.5 kW motors when combined with a gearbox. The following calculations, using gear ratios, were done to determine what combination of motor and gearbox combined with a slew ring would produce the required speed and torque needed for the system to operate.

$$D_{maxT} := \frac{T_{motor}}{T_{need}} \cdot D_{slewring,pitch} = 0,5019 \text{ m}$$

Calculation done to determine the minimum possible pitch diameter needed on pinion gear to achieve both the Torque needed and the minimum speed required.

$$D_{maxn} := \frac{n_{minmum}}{n_{motor}} \cdot D_{slewring,pitch} = 0,3832 \text{ m}$$

A1.2

Diameter needed by speed is smaller, therefore is the limiting factor

Calculate maximum torque system could output with this diameter

$$T_{output} := \frac{D_{slewring,pitch}}{D_{maxn}} \cdot T_{motor} = 341,8976 \text{ kN m}$$

Toutput > Tneed , therefore can supply enough torque

Calculate amount of teeth gear would need

$$Teeth_{pinion} := \frac{T_{motor}}{T_{output}} \cdot Teeth_{slewring} = 37,7371$$

A1.3

Impossible to not have full number of teeth therefore round up teeth and round diameter up.

$$D_{pinion} := 385 \text{ mm}$$

$$Teeth_{pinion,final} := 38$$

Ensure that module of slewring and pinion gear are the same with rounded values

$$Mo_{slewring} := \frac{D_{slewring,pitch}}{Teeth_{slewring}} = 10,1538 \text{ mm}$$

$$Mo_{pinion} := \frac{D_{pinion}}{Teeth_{pinion,final}} = 10,1316 \text{ mm}$$

A1.4

Calculate the maximum output torque the motor will deliver for overload condition at 261 kNm of torque.

$$T_{motor,output} := \frac{D_{pinion}}{D_{slewring,pitch}} \cdot T_{need} = 95,1562 \text{ kN m}$$

$$n_{outout} := \frac{D_{pinion}}{D_{slewring,pitch}} \cdot n_{motor} = 0,1276 \frac{\text{rev}}{\text{min}}$$

A1.5

Therefore a PF052RF127DRN132S4 gearmotor will provide enough torque and the correct speed

3.Force calculations for drive system:

Set vairables:

$$g_a := 9,81 \frac{\text{m}}{\text{s}^2}$$

$$m_{motor} := 1020 \text{ kg}$$

$$m_{slewring} := 495 \text{ kg}$$

$Fmax_{radial} := 333000\ N$ (Given in catalouge)

$V_{pinion} := 0,01337\ m^3$

$\alpha_{pinion} := 20\ deg$

$Mmax_{slewring} := 1534\ kN$

$E_{shaft} := 180\ GPa$

$I_{shaft} := 0,00009546\ m^4$

$L_{shaft} := 350\ mm$

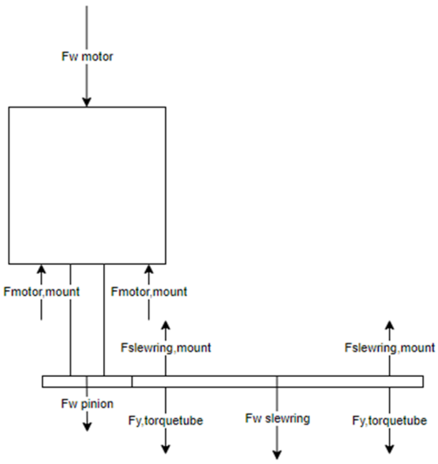
Adjustable variables:

$F_{y,torquetube} := 188700\ N$

$\rho_{pinion} := 7,8\ \frac{g}{cm^3}$

Calculations:

Calculations are done to ensure to determine the forces on the mounting fram at the motor connection point and the slewring connection point. These values can then be given to the relevant memeber to ensure the mounting frame can support the drive system.



$F_{motor,mount} := m_{motor} \cdot g_a + \left(\rho_{pinion} \cdot V_{pinion}\right) \cdot g_a = 11,0292\ kN$ A1.6

$F_{slewring,mount} := F_{y,torquetube} + m_{slewring} \cdot g_a = 193,556\ kN$ A1.7

The weight of the pinion gear creates an axial force on the shaft and must be calculated to ensure the shaft can support it.

$F_{wpinion} := \left(\rho_{pinion} \cdot V_{pinion}\right) \cdot g_a = 1023,0457\ N$

The Euro-drive catalogue stated that all shafts could hold half the radial capacity of the shaft in the axial direction.

$Fmax_{axial} := Fmax_{radial} \cdot 0,5 = 166,5\ kN$

$N_{axial} := \frac{Fmax_{axial}}{F_{wpinion}} = 162,7493$ A1.8

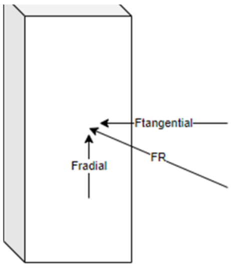
Therefore, shaft has sufficient strength to support pinion gear. (Faxial < Fmaxaxial)

The forces experienced by the gear teeth are determined to compare if the shaft of the motor is capable of supporting them and for later use in determining the stress on the gear. Three teeth are always in contact and thus share the load meaning all the forces can be divided by 3. As they are all calculated from the tangential force, only this force has been split between the 3 teeth.

$$F_{tangential} := 2 \cdot \frac{T_{motor,output}}{3 \cdot D_{pinion}} = 164,7727 \text{ kN}$$

$$F_R := \frac{F_{tangential}}{\cos\left(\alpha_{pinion}\right)} = 175,3475 \text{ kN}$$

$$F_{radial} := F_{tangential} \cdot \tan\left(\alpha_{pinion}\right) = 59,9724 \text{ kN}$$



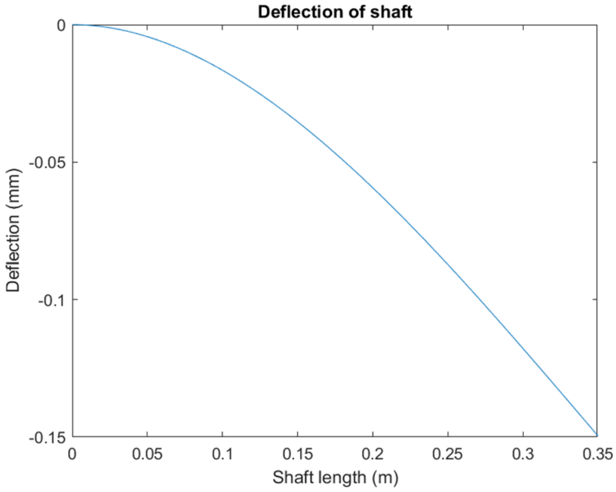
$$N_{radial} := \frac{Fmax_{radial}}{F_{radial}} = 5,5526$$ A1.9

Therefore, shaft has sufficient strength to support radial force on pinion gear teeth. (Fradial < Fmaxradial)

The radial force will also create deflection on the beam and this deflection must be calculated to ensure it is suffieciently small. It will be caused by all the radial forces on the 3 seperate teeth, therefore the radial force must be multiplied by 3.

$$v_{shaft} := -\frac{3 \cdot F_{radial} \cdot L_{shaft}^{3}}{3 \cdot E_{shaft} \cdot I_{shaft}} = -0,1496 \text{ mm}$$ A1.10

Graph modelled by equation:y=(Fx^2)/(6EI) (x-3l)1000 Function obtained from Table A-9 in Shigleys Mechaical Engineering Design



This deflection is small enough to not disrupt the load path of the torque.

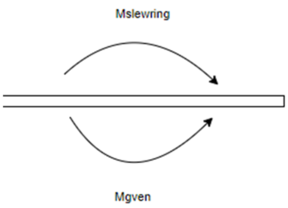
Moment calculations

This value was given after it had been calculated in Sub-System 3's calculations and is caused by the 4.5kN notional load acting on the bottom of the torque tube. It is necassary to ensure that the rated moment the slewring can handle is large enough. The supplier does not specify any specific radial loads that the slewring can handle, so it is assumed these have been incoporated into the maximum moment the slewring may resist.

$$M_{given} := 45990 \text{ N}$$

$$M_{slewring} := M_{given} = 45990 \text{ N}$$

$$N_{slewring} := \frac{Mmax_{slewring}}{M_{given}} = 33,3551$$ A1.11



The slewring can adequately handle the moment applied to it.

4.Stress calculations for drive system:

Set vairables:

$A_{teeth} := 1748\ mm^2$

$A_{kewyway} := 4200\ mm^2$

$K_{t,keyway} := 2,14$

$K_{t,pinion} := 2$

$J_{pinion} := 0,00050344\ m^4$

$D_{pinion,collar} := 290\ mm$

Adjustable variables:

$\sigma_{4140} := 1140\ MPa$ AISI 4140 steel, Q&T at 425 degrees Celsuis

$\tau_{4140} := 684\ MPa$

Caculations:

The stress experienced by the gear teeth are determined to allow a material for the pinion gear to be chosen that will be suffieciently strong enough. They are calculated through the use of the force normal to the teeth, acting across each tooth area.

$$\sigma_{teeth} := \frac{F_R}{A_{teeth}} = 100,3132\ MPa$$

$$N_{teeth} := \frac{\sigma_{4140}}{\sigma_{teeth}} = 11,3644$$

The critical stresses in the keyway must be analysed to ensure the material can withstand them as well. The chosen motor has two keyways on its shaft, allowing the force to be shared between them. This force is calculated using the torque output the motor will be producing at the overload condition of 261kNm acting on the rakes.

$$F_{keyway} := \frac{T_{motor,output}}{2 \cdot \left(\frac{D_{motor}}{2}\right)} = 453,125\ kN$$

$$\sigma_{keyway} := \frac{F_{keyway}}{A_{kewyway}} \cdot K_{t,keyway} = 230,878\ MPa$$

$$N_{keyway} := \frac{\sigma_{4140}}{\sigma_{keyway}} = 4,9377$$ A1.12

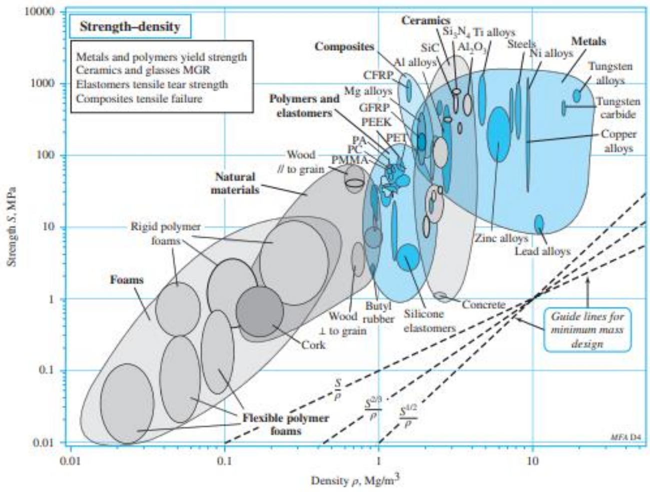
When the gear changes diameter to the supporting collar, there will be a stress concentration. This concentration is calculated to ensure the chosen material may withstand it.

$$\tau_{pinion} := K_{t,pinion} \cdot \frac{T_{motor,output} \cdot D_{pinion,collar}}{J_{pinion}} = 109,627\ MPa$$

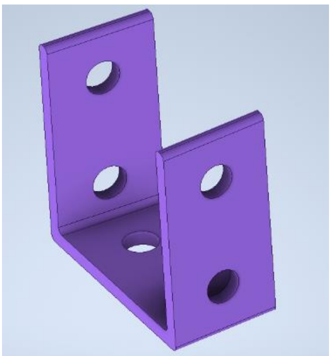
$$N_{pinion} := \frac{\tau_{4140}}{\tau_{pinion}} = 6,2393$$

The gear is an integral part of the system and must therefore have high safety factors to ensure it will be last to fail. All safety factors generated with AISI 4140 steel are higher than 4, meaning AISI 4140 is the most suitable material for the pinion gear.

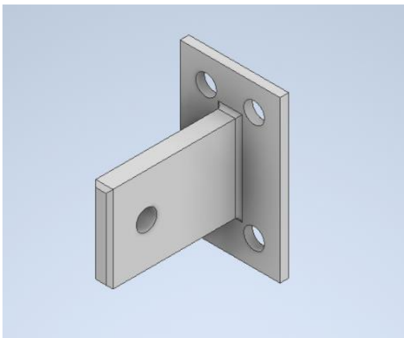
A.2 Sub-system 2: Mounting Frame (SP)



A.2 ref1.1



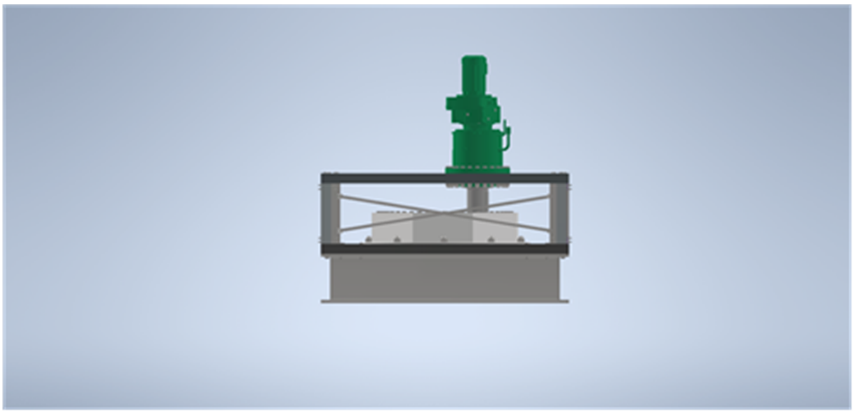
A.2 ref1.2



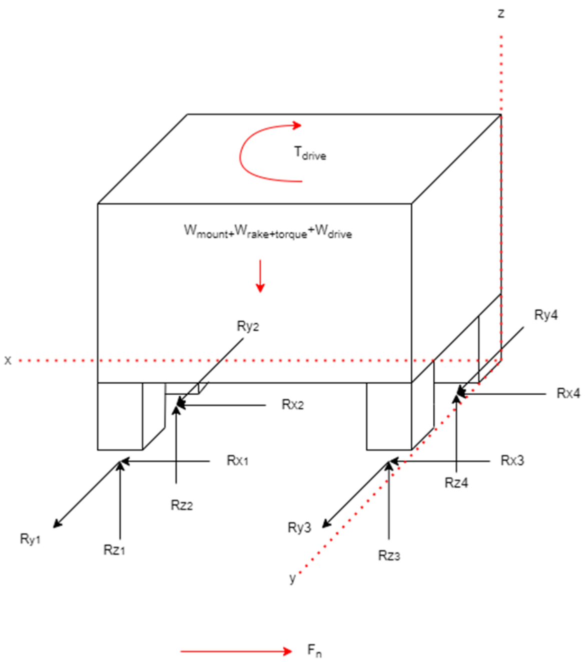
A.2 ref1.3

Calculations for the Mounting Frame

The mounting frame is assumed to be a rigid system and thus all the forces are transmitted to the bolt connection between the mounting frame and the bridge. (For calculations proving this please go to bottom of Doc.)



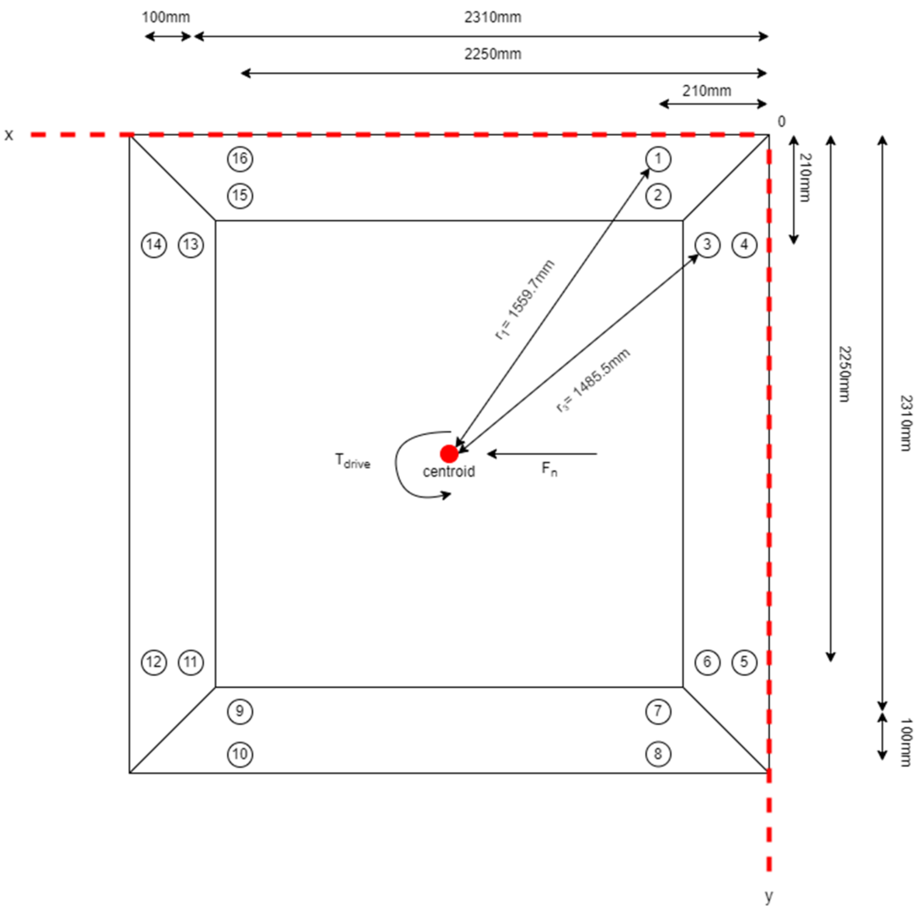
This fact the mounting frame can be seen as a rigid body allows the simplification of the mounting frame assembly above into the simple free body diagram below



Bolt calculations

Shear force bolt calculations:

(The circles with the numbers are the bolt locations on the the 4 I-beam base)



$y_1 := 50\ mm$ $\quad y_9 := 2310\ mm$ $\quad x_1 := 210\ mm$ $\quad x_9 := 2250\ mm$

$y_2 := 150\ mm$ $\quad y_{10} := 2410\ mm$ $\quad x_2 := 210\ mm$ $\quad x_{10} := 2250\ mm$

$y_3 := 210\ mm$ $\quad y_{11} := 2250\ mm$ $\quad x_3 := 150\ mm$ $\quad x_{11} := 2310\ mm$

$y_4 := 210\ mm$ $\quad y_{12} := 2250\ mm$ $\quad x_4 := 50\ mm$ $\quad x_{12} := 2410\ mm$

$y_5 := 2250\ mm$ $\quad y_{13} := 210\ mm$ $\quad x_5 := 50\ mm$ $\quad x_{13} := 2310\ mm$

$y_6 := 2250\ mm$ $\quad y_{14} := 210\ mm$ $\quad x_6 := 150\ mm$ $\quad x_{14} := 2410\ mm$

$y_7 := 2310\ mm$ $\quad y_{15} := 150\ mm$ $\quad x_7 := 210\ mm$ $\quad x_{15} := 2250\ mm$

$y_8 := 2410\ mm$ $\quad y_{16} := 50\ mm$ $\quad x_8 := 210\ mm$ $\quad x_{16} := 2250\ mm$

$$y_{sum1} := y_1 + y_2 + y_3 + y_4 + y_5 + y_6 + y_7 + y_8$$
$$y_{sum2} := y_9 + y_{10} + y_{11} + y_{12} + y_{13} + y_{14} + y_{15} + y_{16}$$

$$y_{bar} := \frac{y_{sum1} + y_{sum2}}{16} = 1{,}23 \text{ m}$$

$$x_{sum1} := x_1 + x_2 + x_3 + x_4 + x_5 + x_6 + x_7 + x_8$$
$$x_{sum2} := x_9 + x_{10} + x_{11} + x_{12} + x_{13} + x_{14} + x_{15} + x_{16}$$

$$x_{bar} := \frac{x_{sum1} + x_{sum2}}{16} = 1{,}23 \text{ m}$$

$$r_1 := \sqrt{\left(y_{bar} - y_1\right)^2 + \left(x_{bar} - x_1\right)^2} = 1{,}5597 \text{ m}$$
$$r_3 := \sqrt{\left(y_{bar} - y_3\right)^2 + \left(x_{bar} - x_3\right)^2} = 1{,}4855 \text{ m}$$

Assumption:The toque is being applied by the slew ring and thus it is in the center of the assembly.

Because the centroid is concentric the distance between bolt 1 and the centroid is the same for bolts: 4,5,8,10,12,14,16

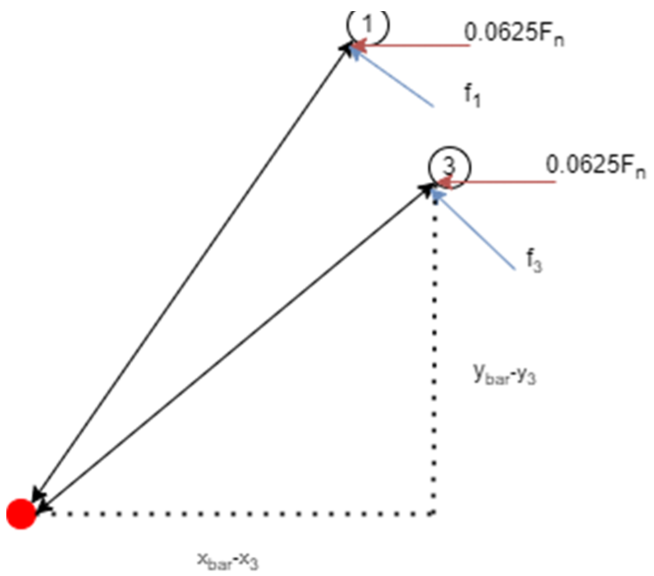
Because the centroid is concentric the distance between bolt 3 and the centroid is the same for bolts: 2,6,7,9,11,13,15

$$\tau := \frac{(T \cdot r)}{J} \qquad f_n := 4{,}5 \text{ kN}$$

$T_{drive} := 260 \text{ kN m}$ Assumption: When the torque is at T_{max} that will be the point when the bolts experience the maximum stress.

$$r_{squared} := r_1^{\,2} \cdot 8 + r_3^{\,2} \cdot 8 = 37{,}1168 \text{ m}^2$$

$$f_1 := \frac{T_{drive} \cdot r_1}{r_{squared}} = 10925{,}8699 \text{ N} \qquad f_3 := \frac{T_{drive} \cdot r_3}{r_{squared}} = 10406{,}0117 \text{ N}$$



$0,0625 \cdot F_n$ is the shear force caused by the notional load.

$$\theta_1 := \operatorname{atan}\left(\frac{y_{bar} - y_1}{x_{bar} - x_1}\right) = 49,1596 \text{ deg}$$

$$\theta_3 := \operatorname{atan}\left(\frac{y_{bar} - y_3}{x_{bar} - x_3}\right) = 43,3634 \text{ deg}$$

$$f_{x1} := f_1 \cdot \cos\left(\theta_1\right) + \frac{1}{16} \cdot f_n = 7426,2625 \text{ N}$$

$$f_{y1} := f_1 \cdot \sin\left(\theta_1\right) = 8265,7988 \text{ N}$$

$$f_{net1} := \sqrt{f_{x1}^{\ 2} + f_{y1}^{\ 2}} = 11111,8317 \text{ N}$$

$$f_{x3} := f_3 \cdot \cos\left(\theta_3\right) + \frac{1}{16} \cdot f_n = 7846,5574 \text{ N}$$

$$f_{y3} := f_3 \cdot \sin\left(\theta_3\right) = 7145,0125 \text{ N}$$

$$f_{net3} := \sqrt{f_{x3}^{\ 2} + f_{y3}^{\ 2}} = 10612,2413 \text{ N}$$

The value of f_{net1} is greater than f_{net3} , thus it will be the force that will most likely cause the the bolts to yield.

For the bolts, the property class that will be used is the 8.8:

$$\sigma_{yieldbolts} := 660\ \mathrm{MPa}$$

$$D_{bolts} := \sqrt{\frac{f_{net1}}{\sigma_{yieldbolts} \cdot \left(\frac{\pi}{4}\right)}} = 4,6299\ \mathrm{mm}$$

$$D_{actual} := 20\ \mathrm{mm}$$

$$\sigma_{actual\_shear} := \frac{f_{net1}}{\left(\frac{\pi}{4} \cdot D_{actual}^{\ 2}\right)} = 35,3701\ \mathrm{MPa}$$

$$n_{shear} := \frac{\sigma_{yieldbolts}}{\sigma_{actual\_shear}} = 18,6599$$

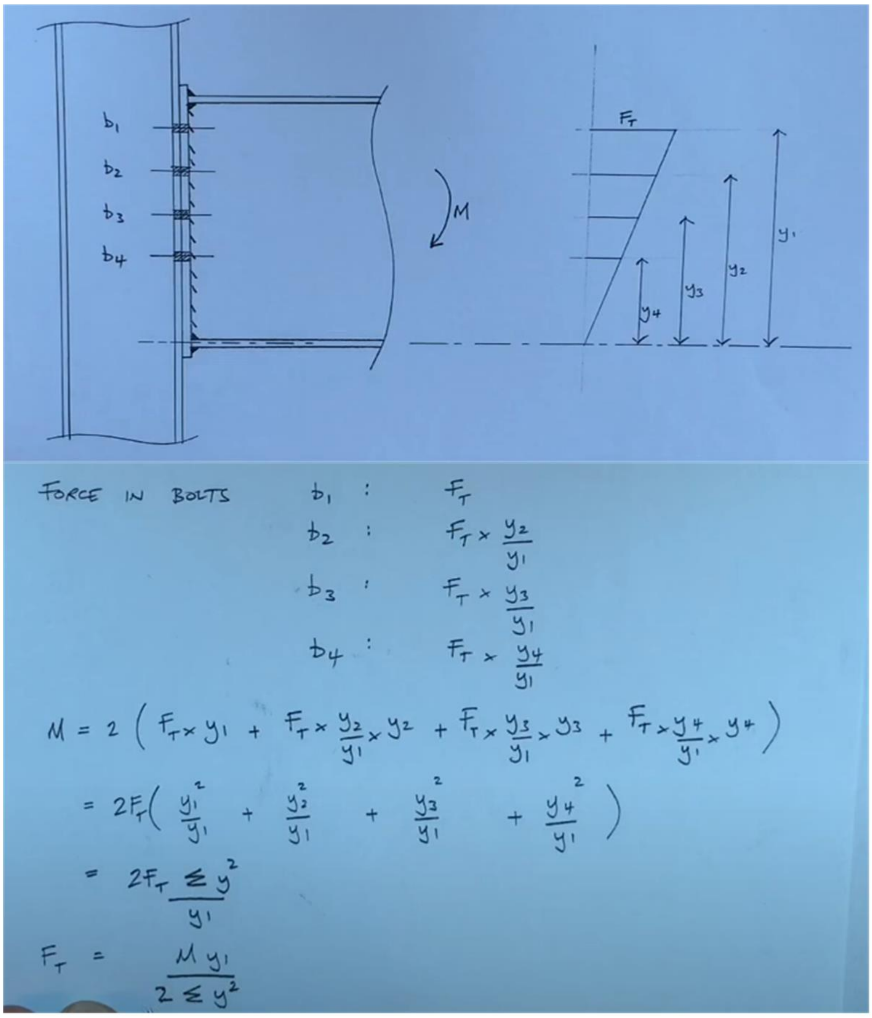
The safety factor is considerably large but this ensures that the system will not fail. In real scenarios, bolts of atleast M16 are used and thus using bolts of lower than M16 to achieve a lower safety factor is not advised.

Bending moment bolt calculations

$l_{nfm} := 10,22\ \mathrm{m}$ This is the distance of the notional load load from the base of the mounting frame

$M_n := l_{nfm} \cdot f_n = 45,99\ \mathrm{kN\ m}$ This is the moment caused by the notional load

According to Mike Bather this formula can be used to calculate the force on the bolts due to an out of plane bending moment.



$$F_t := \frac{M_n \cdot \left(x_{bar} - x_4\right)}{\left(2 \cdot \left(x_{bar} - x_4\right)^2 + 2 \cdot \left(x_{bar} - x_3\right)^2 + 4 \cdot \left(x_{bar} - x_1\right)^2\right)} = 5848{,}3705 \text{ N}$$

The maximum force on a bolt would be F_t and it will exert itself on the bolts furthest from the centroid (bolt:4,5)

$$D_{bending} := \sqrt{\frac{F_t}{\left(\frac{\pi}{4} \cdot \sigma_{yieldbolts}\right)}} = 3{,}3589 \text{ mm}$$

$$\sigma_{actual\_bending} := \frac{F_t}{\left(\frac{\pi}{4} \cdot D_{actual}^2\right)} = 1{,}8616 \cdot 10^7 \text{ Pa}$$

$$n_{bending} := \frac{\sigma_{yieldbolts}}{\sigma_{actual\_bending}} = 35{,}4535$$

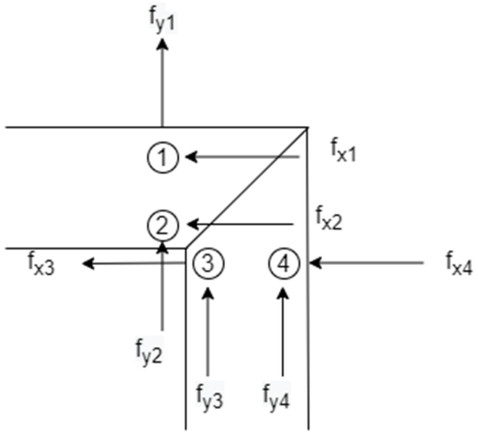
For each pair of 4 bolts there is either a force in the postive z or negative z axis depending on the line of action of the notional load.

Assuming notional load is acting in the direction of the negative x- axis as shown in the first free body diagam.

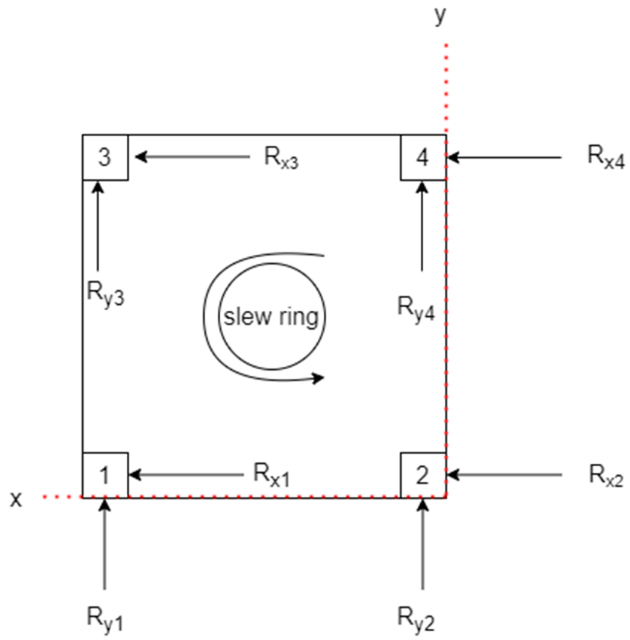
R_z3 and R_z4 will experience a force in the positve z axis

$$F_z := F_t + F_t \cdot \frac{x_{bar} - x_3}{x_{bar} - x_4} + 2 \cdot F_t \cdot \frac{x_{bar} - x_1}{x_{bar} - x_4} = 21311{,}8588 \text{ N}$$

Reaction force calculations: *(A.2 ref1.5)*



The forces on one corner of the mounting frame is the sum of the forces caused by the notional load and the torque.



$$R_{x1} := -\left(2 \cdot f_{x1} + 2 \cdot f_{x3}\right) = -30545{,}6397 \text{ N}$$

$$R_{y1} := -\left(2 \cdot f_{y1} + 2 \cdot f_{y3}\right) = -30821{,}6226 \text{ N}$$

(f_x1 is equal to f_x4 and f_x3 is equal to f_x2)

As the corners are the equidistant from the centre and the forces acting on them in the x and y axises are the same, the reaction forces will be the same too.

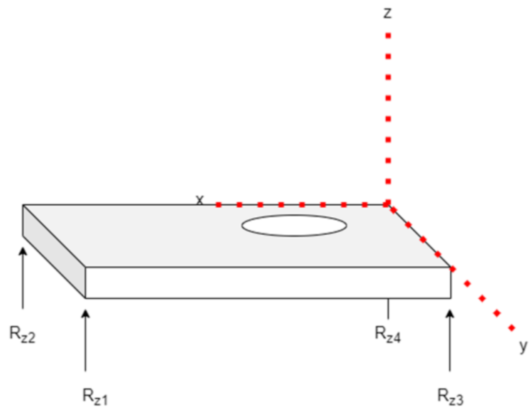
$$R_{x2} := R_{x1} = -30545{,}6397\ \mathrm{N}$$
$$R_{x3} := R_{x1} = -30545{,}6397\ \mathrm{N}$$
$$R_{x4} := R_{x1} = -30545{,}6397\ \mathrm{N}$$

$$R_{y2} := R_{y1} = -30821{,}6226\ \mathrm{N}$$
$$R_{y3} := R_{y1} = -30821{,}6226\ \mathrm{N}$$
$$R_{y4} := R_{y1} = -30821{,}6226\ \mathrm{N}$$

The total reaction force in the x- direction:

$$R_{xtotal} := R_{x1} + R_{x2} + R_{x3} + R_{x4} = -122{,}1826\ \mathrm{kN}$$

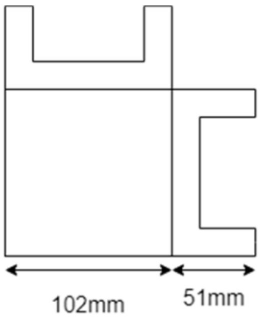
$$R_{ytotal} := R_{y1} + R_{y2} + R_{y3} + R_{y4} = -123{,}2865\ \mathrm{kN}$$



Due to motor not being in the center of the mounting frame, the reaction forces due to the motor at each connection to the bridge will be different.

Axis x-z(moment taken at the drive system)

First the centroid of the vertical stays must be calculated to find the axis of the drive system weight



$y_{c1} := 102\ \text{mm} + 15,1\ \text{mm} = 0,1171\ \text{m}$

$y_{sq} := \dfrac{102}{2}\ \text{mm} = 0,051\ \text{m}$

$y_{c2} := y_{sq} = 0,051\ \text{m}$

$x_{c1} := y_{sq} = 0,051\ \text{m}$

$x_{sq} := x_{c1} = 0,051\ \text{m}$

$x_{c2} := y_{c1} = 0,1171\ \text{m}$

$a_{channel} := 13,28\ \text{cm}^2$

$a_{square} := 102\ \text{mm} \cdot 102\ \text{mm} = 0,0104\ \text{m}^2$

$g := 9,81\ \dfrac{\text{m}}{\text{s}^2}$

$y_{sum\_comp} := y_{c1} \cdot a_{channel} + y_{c2} \cdot a_{channel} + y_{sq} \cdot a_{square}$

$x_{sum\_comp} := x_{c1} \cdot a_{channel} + x_{c2} \cdot a_{channel} + x_{sq} \cdot a_{square}$

$x_{bar\_comp} := \dfrac{x_{sum\_comp}}{2 \cdot a_{channel} + a_{square}} = 0,0577\ \text{m}$

$y_{bar\_comp} := \dfrac{y_{sum\_comp}}{2 \cdot a_{channel} + a_{square}} = 0,0577\ \text{m}$

$x_{motor} := 2460\ \text{mm} - 1570\ \text{mm} = 0,89\ \text{m}$

$y_{motor} := 2460\ \text{mm} - 1237\ \text{mm} = 1,223\ \text{m}$

$F_{motor,mount} := 11,0292\ \text{kN}$ (These values were aquired from the Drive sytem calculations)

$F_{slewring,mount} := 193,556\ \text{kN}$ (The slew ring includs the weight of the toque tube, rakes and slew ring)

$$m_{mountin,top\_plate} := 4153{,}181\ \mathrm{kg}$$ (Inventor iproperties)

$$w_{mounting,top\_plate} := m_{mountin,top\_plate} \cdot g = 40742{,}7056\ \mathrm{N}$$

Take the moment about R_z4 to find the value of R_z2:

$$R_{z2} := \frac{F_{motor,mount} \cdot x_{motor}}{2460\ \mathrm{mm} - 2 \cdot x_{bar\_comp}} = 4186{,}7128\ \mathrm{N}$$

$$R_{z4} \cdot \left(x_{motor} - x_{bar\_comp}\right) - R_{z2}\left(2460\ \mathrm{mm} - x_{bar\_comp} - x_{motor}\right)$$

$$R_{z4} := R_{z2} \cdot \frac{\left(2460\ \mathrm{mm} - x_{bar\_comp} - x_{motor}\right)}{\left(x_{motor} - x_{bar\_comp}\right)} = 7607{,}3997\ \mathrm{N}$$

Axis y-z (Moment taken at the drive system)

$$R_{z3} \cdot \left(2460\ \mathrm{mm} - y_{bar\_comp} - y_{motor}\right) - R_{z4}\left(y_{motor} - y_{bar\_comp}\right)$$

$$R_{z3} := \frac{R_{z4} \cdot \left(y_{motor} - y_{bar\_comp}\right)}{\left(2460\ \mathrm{mm} - y_{bar\_comp} - y_{motor}\right)} = 7517{,}0872\ \mathrm{N}$$

$$R_{z1} := R_{z3} \cdot \frac{\left(x_{motor} - x_{bar\_comp}\right)}{\left(2460\ \mathrm{mm} - x_{bar\_comp} - x_{motor}\right)} = 4137{,}0095\ \mathrm{N}$$

$$R_{z1\_actual} := \left|-R_{z1} - \frac{F_{slewring,mount}}{4} - \frac{w_{mounting,top\_plate}}{4} - F_z\right| = -84{,}0235\ \mathrm{kN}$$

$$R_{z2\_actual} := \left(-R_{z2} - \frac{F_{slewring,mount}}{4} - \frac{w_{mounting,top\_plate}}{4} - F_z \right) = -84{,}0732 \text{ kN}$$

$$R_{z3\_actual} := \left(-R_{z3} - \frac{F_{slewring,mount}}{4} - \frac{w_{mounting,top\_plate}}{4} + F_z \right) = -44{,}7799 \text{ kN}$$

$$R_{z4\_actual} := \left(-R_{z4} - \frac{F_{slewring,mount}}{4} - \frac{w_{mounting,top\_plate}}{4} + F_z \right) = -44{,}8702 \text{ kN}$$

These actual forces are the forces exerted on the bridge in the Z direction

$$R_{ztotal} := R_{z1\_actual} + R_{z2\_actual} + R_{z3\_actual} + R_{z4\_actual} = -257{,}7469 \text{ kN}$$

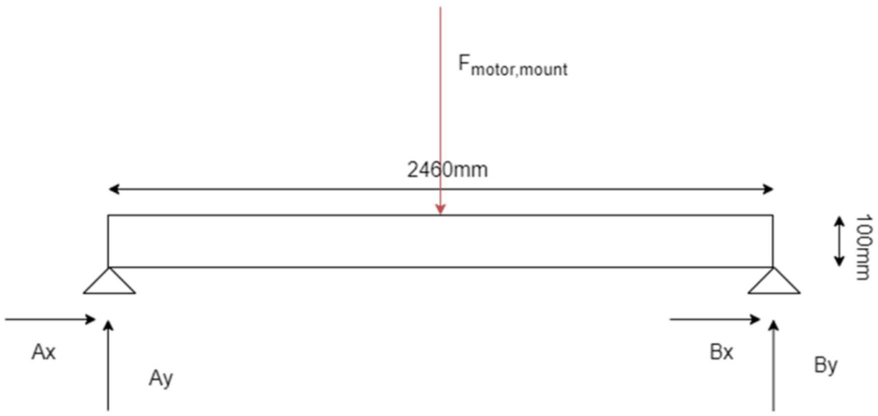
Deflection calculations *(A.2 ref1.4)*

These purpose of these calculations are to prove that the system can be simplified into a rigid system:

Deflection of top plate holding motor:

$F_{motor,mount} := 10,6936\ \text{kN}$ (From the Drive system calculations)

The top plate can be simplifed into a simply supported beam the forces acting on the beam can be super positioned:



As there is no horizontal force, the values of Ax,Bx will be 0. As the $F_{motor,mount}$ is acting in the middle the values of Ay and By will be equal:

$$Ay := \frac{F_{motor,mount}}{2} = 5346,8\ \text{N}$$

$$By := Ay = 5346,8\ \text{N}$$

$$l_{top\_plate} := 2,46\ \text{m}$$

$E_{top\_plate} := 90\ \text{GPa}$ The range for cast iron is between $67 - 96$ Gpa

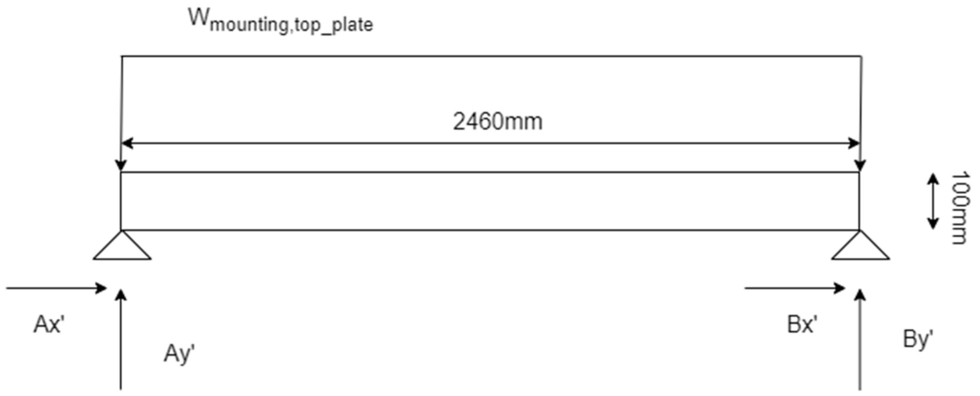
$$b_{top\_plate} := l_{top\_plate}$$

$$h_{top\_plate} := 0,1\ \text{m}$$

$$I_{top\_plate} := \frac{1}{12} \cdot b_{top\_plate} \cdot h_{top\_plate}^{3} = 0,0002\ \text{m}^{4}$$

$$y_{max,top\_plate} := -\frac{F_{motor,mount} \cdot l_{top\_plate}^{3}}{48 \cdot E_{top\_plate} \cdot I_{top\_plate}} = -0,1798\ \text{mm}$$ (This deflection is negligible)

In the y-z axis the $F_{motor,mount}$ is not in the centre but because the deflection is so small, it is assumed that it will not change much due to the offset.



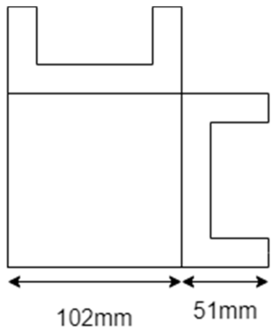
$m_{mountin,top\_plate} := 4153,181 \text{ kg}$ (Inventor iproperties)

$$w_{mounting,top\_plate} := m_{mountin,top\_plate} \cdot \frac{g}{l_{top\_plate}} = 16562,0755 \text{ m Pa}$$

$$y_{max,top\_plate'} := \frac{(-5) \cdot w_{mounting,top\_plate} \cdot l_{top\_plate}^{\ 4}}{384 \cdot E_{top\_plate} \cdot I_{top\_plate}} = -0,4281 \text{ mm}$$ (Eq, from table A-9)

$$y_{max,total} := y_{max,top\_plate} + y_{max,top\_plate'} = -0,6078 \text{ mm}$$ (this is less than 1mm)

Buckling of vertical stays:



The properties of the c-channel can be obtained from table A-7 in Shigley's

Centroid of composite beam:

$$y_{c1} := 102\ \mathrm{mm} + 15{,}1\ \mathrm{mm} = 0{,}1171\ \mathrm{m}$$

$$y_{sq} := \frac{102}{2}\ \mathrm{mm} = 0{,}051\ \mathrm{m}$$

$$y_{c2} := y_{sq} = 0{,}051\ \mathrm{m}$$

$$x_{c1} := y_{sq} = 0{,}051\ \mathrm{m}$$

$$x_{sq} := x_{c1} = 0{,}051\ \mathrm{m}$$

$$x_{c2} := y_{c1} = 0{,}1171\ \mathrm{m}$$

$$a_{channel} := 13{,}28\ \mathrm{cm}^2$$

$$a_{square} := 102\ \mathrm{mm} \cdot 102\ \mathrm{mm} = 0{,}0104\ \mathrm{m}^2$$

$$y_{sum\_comp} := y_{c1} \cdot a_{channel} + y_{c2} \cdot a_{channel} + y_{sq} \cdot a_{square}$$

$$x_{sum\_comp} := x_{c1} \cdot a_{channel} + x_{c2} \cdot a_{channel} + x_{sq} \cdot a_{square}$$

$$x_{bar\_comp} := \frac{x_{sum\_comp}}{2 \cdot a_{channel} + a_{square}} = 0{,}0577\ \mathrm{m}$$

$$y_{bar\_comp} := \frac{y_{sum\_comp}}{2 \cdot a_{channel} + a_{square}} = 0{,}0577\ \mathrm{m}$$

$$x_{motor} := 1570\ \mathrm{mm}$$

$$y_{motor} := 1237\ \mathrm{mm}$$

The torque on the furthest beam only needs to be calculated as it is the composite beam that experience the greatest torque force.

$$r_{comp1,motor} := \sqrt{\left(x_{motor} - x_{bar\_comp}\right)^2 + \left(y_{motor} - y_{bar\_comp}\right)^2} = 1{,}9177\ \mathrm{m}$$

$$x2_{comp} := 2460\ \mathrm{mm} - x_{bar\_comp} = 2{,}4023\ \mathrm{m}$$

$$y2_{comp} := y_{bar\_comp} = 0{,}0577\ \mathrm{m}$$

$$x3_{comp} := x_{bar\_comp} = 0{,}0577\ \mathrm{m}$$

$$y3_{comp} := x2_{comp} = 2{,}4023\ \mathrm{m}$$

$$y4_{comp} := y3_{comp} = 2{,}4023\ \mathrm{m}$$

$$x4_{comp} := x2_{comp} = 2{,}4023\ \mathrm{m}$$

$$r_{comp2,motor} := \sqrt{\left(x2_{comp} - x_{motor}\right)^2 + \left(y_{motor} - y2_{comp}\right)^2} = 1{,}4434\ \mathrm{m}$$

$$r_{comp3,motor} := \sqrt{\left(x_{motor} - x3_{comp}\right)^2 + \left(y_{motor} - y3_{comp}\right)^2} = 1,9092 \text{ m}$$

$$r_{comp4,motor} := \sqrt{\left(x4_{comp} - x_{motor}\right)^2 + \left(y4_{comp} - y_{motor}\right)^2} = 1,432 \text{ m}$$

$$r_{squared\_comp} := \left(r_{comp1,motor}\right)^2 + \left(r_{comp2,motor}\right)^2 + \left(r_{comp3,motor}\right)^2 + \left(r_{comp4,motor}\right)^2$$

$$T_{motor,mount} := 95 \text{ kN m}$$

$$f_{motor,mount} := \frac{T_{motor,mount} \cdot r_{comp1,motor}}{r_{squared\_comp}} = 15902,2704 \text{ N}$$

$\sigma_{yield\_SS440C} := 450 \text{ MPa}$ (The yield strength of Stainless Steel 440C is $450 - 1900$Mpa)

$$A_{required} := \frac{f_{motor,mount}}{\sigma_{yield\_SS440C}} = 35,3384 \text{ mm}^2$$

$$A_{actual} := 2 \cdot a_{channel} + a_{square} = 13060 \text{ mm}^2$$

$$n := \frac{A_{actual}}{A_{required}} = 369,5699$$

With these calculations it is sufficiently proven that the beams will not yield because of the toque caused by the motor.

A.3 Sub-system 3: Torque Tube (SLG)

Torque Tube Calculations

406 x 326, stainless steel 316, tube (from hollowbar.co.za)

$T_{max} := 261\ \mathrm{kN\ m}$ $\quad S_y := 290\ \mathrm{MPa}$ $\quad S_u := 580\ \mathrm{MPa}$ $\quad \rho := 8\ \frac{\mathrm{Mg}}{\mathrm{m}^3}$ $\quad E := 193\ \mathrm{GPa}$

$d_o := 406\ \mathrm{mm}$ $\quad L_{tube} := 10{,}22\ \mathrm{m}$

$d_i := 326\ \mathrm{mm}$

$$c_o := \frac{d_o}{2}$$

$c_o = 0{,}203\ \mathrm{m}$

$$c_i := \frac{d_i}{2}$$

$c_i = 0{,}163\ \mathrm{m}$

$$W_{tube} := \rho \cdot \left(\pi \cdot c_o^{\ 2} \cdot 10{,}37\ \mathrm{m} - \pi \cdot c_i^{\ 2} \cdot L_{tube}\right) \cdot 9{,}81\ \frac{\mathrm{m}}{\mathrm{s}^2}$$

$W_{tube} = 38410\ \mathrm{N}$

Weight of casting that hold up rakes:

$$W_{b,casting} := \rho \cdot \left[\left(0{,}5\ \mathrm{m} \cdot 0{,}5\ \mathrm{m} \cdot 0{,}1\ \mathrm{m}\right) - \left[\pi \cdot \left(\frac{d_o}{2}\right)^2 \cdot 0{,}1\ \mathrm{m}\right] + \left(0{,}1\ \mathrm{m} \cdot 0{,}1\ \mathrm{m} \cdot 0{,}3\ \mathrm{m} \cdot 8\right)\right] \cdot 9{,}81\ \frac{\mathrm{m}}{\mathrm{s}^2} = 2830\ \mathrm{N}$$

Normal Conditions

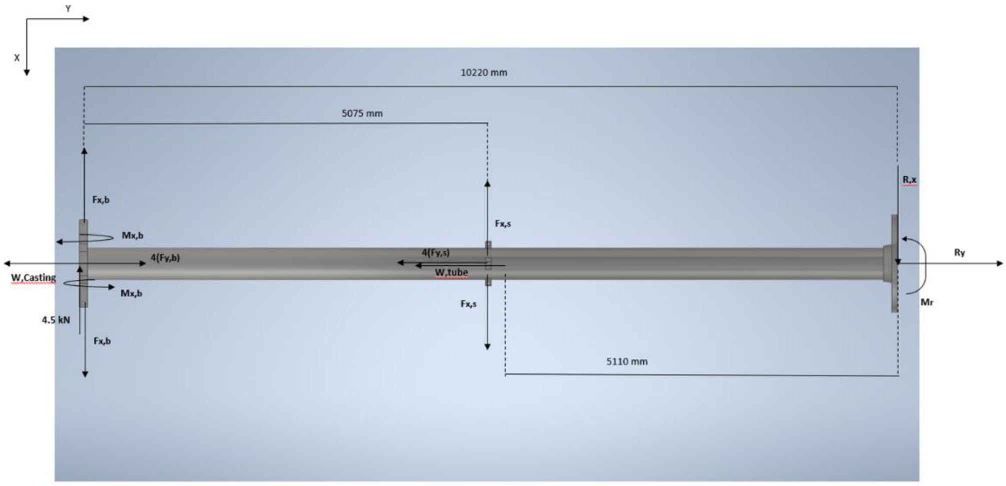


Figure A3.1: Free body diagram illustrating forces acting on torque tube during normal conditions(all except torque)

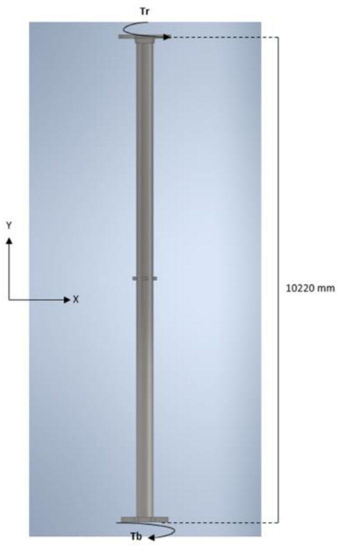


Figure A3.2: Free body diagram illustrating torque acting on torque tube during normal conditions

Torque Calculations

$T_{normal} := \left(0,6 \cdot T_{max}\right) \cdot 1,1 = 1,723 \cdot 10^{5}\ \mathrm{J}$

$J := \frac{\pi}{2} \cdot \left(c_o^{\ 4} - c_i^{\ 4}\right)$

$J = 0,001559\ \mathrm{m}^{4}$

$\tau_{torque,normal} := \left(\frac{T_{normal} \cdot c_o}{J}\right)$

$\tau_{torque,normal} = 2,244 \cdot 10^{7}\ \mathrm{Pa}$

Forces on the Tube

The forces acting along the x- and z-axis at the rake and vertical stay connections will cancel each other out therefore, they will not contribute to the shear and moment experienced by the torque tube.

The moments acting about the x- and z-axis at the rake and vertical stay connections will also cancel each other out therefore they will not contribute to the moment experienced by the torque tube.

Shear Calculations

Therefore the only shear force that the torque tube will experience is due to the torque and the lateral non-rotating notional load:

$Q := \left(\frac{1}{2} \cdot \pi \cdot c_o^{\ 2} \cdot \frac{4 \cdot c_o}{3 \cdot \pi}\right) - \left(\frac{1}{2} \cdot \pi \cdot c_i^{\ 2} \cdot \frac{4 \cdot c_i}{3 \cdot \pi}\right)$

$Q = 0,002690\ \mathrm{m}^{3}$

$I := \frac{\pi}{64} \cdot \left(d_o^{\ 4} - d_i^{\ 4}\right)$

$I = 0,0007793\ \mathrm{m}^{4}$

$\tau_{shear} := \frac{4,5\ \mathrm{kN} \cdot Q}{I \cdot \left(d_o - d_i\right)}$

$\tau_{shear} = 1,941 \cdot 10^{5}\ \mathrm{Pa}$

$\tau_{max,normal} := \tau_{shear} + \tau_{torque,normal}$

$\tau_{max,normal} = 2,263 \cdot 10^{7}\ \mathrm{Pa}$

Axial Force calculations

From rake connection until the stay connection

$F_{y,b,normal} := \left(1131,1804\ \mathrm{N} \cdot 4\right) - W_{b,casting}$

$F_{y,b,normal} = 1695\ \mathrm{N}$

Therefore the bottom section of the tube is in compression

$$\sigma_{axial,b,normal} := \frac{F'_{y,b,normal}}{\pi \cdot c_o^2 - \pi \cdot c_i^2}$$

$$\sigma_{axial,b,normal} = 36858{,}2525 \text{ Pa}$$ **Compressive**

From stay connection until the center of mass

$$F_{y,s,normal} := -6268{,}4514 \text{ N} \cdot 4$$

$$F_{y,s,normal} = -25070 \text{ N}$$

$$F_{y,m,normal} := F_{y,b,normal} + F_{y,s,normal}$$

$$F_{y,m,normal} = -23380 \text{ N}$$

Therefore this section of the tube is in tension

$$\sigma_{axial,m,normal} := \frac{-F_{y,m,normal}}{\pi \cdot c_o^2 - \pi \cdot c_i^2}$$

$$\sigma_{axial,m,normal} = 5{,}0831 \cdot 10^5 \text{ Pa}$$ **Tensile**

From the center of mass until the top of the tube

$$F_{y,t,normal} := F_{y,m,normal} - W_{tube}$$

$$F_{y,t,normal} = -61790 \text{ N}$$

Therefore this force applies more tension to the tube

$$\sigma_{axial,t,normal} := \frac{-\left(F_{y,t,normal}\right)}{\pi \cdot c_o^2 - \pi \cdot c_i^2}$$

$$\sigma_{axial,t,normal} = 1{,}344 \cdot 10^6 \text{ Pa}$$ **Tensile**

Stress created due to moment of 4.5 KN force at bottom

$$\sigma_M := \frac{\left(4{,}5 \text{ kN} \cdot L_{tube}\right) \cdot c_o}{I}$$

$$\sigma_M = 1{,}198 \cdot 10^7 \text{ Pa}$$

Maximum Stress in torque tube

$$\sigma_{axial,max,normal} := \sigma_{axial,t,normal} + \sigma_M$$

$$\sigma_{axial,max,normal} = 1{,}332 \cdot 10^7 \text{ Pa}$$ **Tensile**

Reaction forces acting on top of pipe

Vertical Force

$$R_{y,normal} := -F_{y,t,normal}$$

$$R_{y,normal} = 61790 \text{ N}$$

Horizontal force

$$R_{x,normal} := 4{,}5 \text{ kN}$$

Moment about horizontal axis

$M_{R,normal} := 4,5\ \text{kN} \cdot L_{tube}$

$M_{R,normal} = 45990\ \text{N m}$

Deflection and slope at bottom of torque tube

$$v_{max} := \frac{4,5\ \text{kN} \cdot L_{tube}^{\ 3}}{3 \cdot E \cdot I}$$

$v_{max} = 0,01065\ \text{m}$

y = 0 at the bottom of the torque tube and the the positive y- axis acts along the torque tube

M(y) = 4.5 kN*y

$$\theta\ (y) = \frac{\left(\frac{4,5\ \text{kN}}{2}\right) \cdot y^2 - 235008,9}{E \cdot I}$$

$$v\ (y) := \frac{\left(\frac{4,5\ \text{kN}}{6}\right) \cdot y^3 - 235008,9 \cdot y + 1601193,972}{E \cdot I}$$

From these equations we can determine the maximum slope of the torque tube, as well as the maximum deflection which has already been calculated.

θ,max = 0.00156251 rad

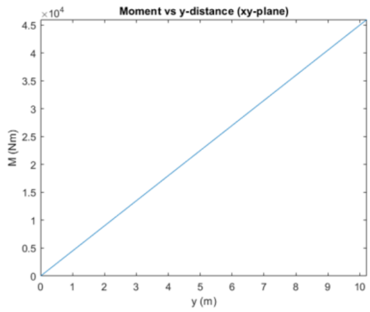


Figure A3.3: Moment vs y-distance graph

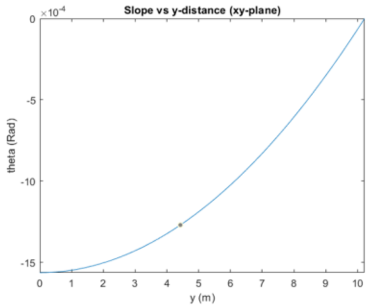


Figure A3.4: Moment vs y-distance graph

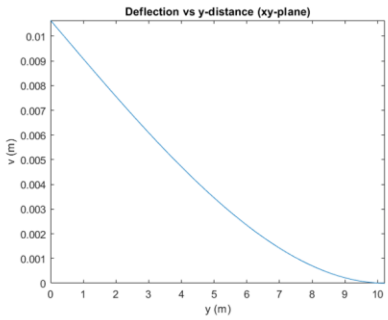


Figure A3.5: Deflection vs y-distance graph

Overloaded Conditions

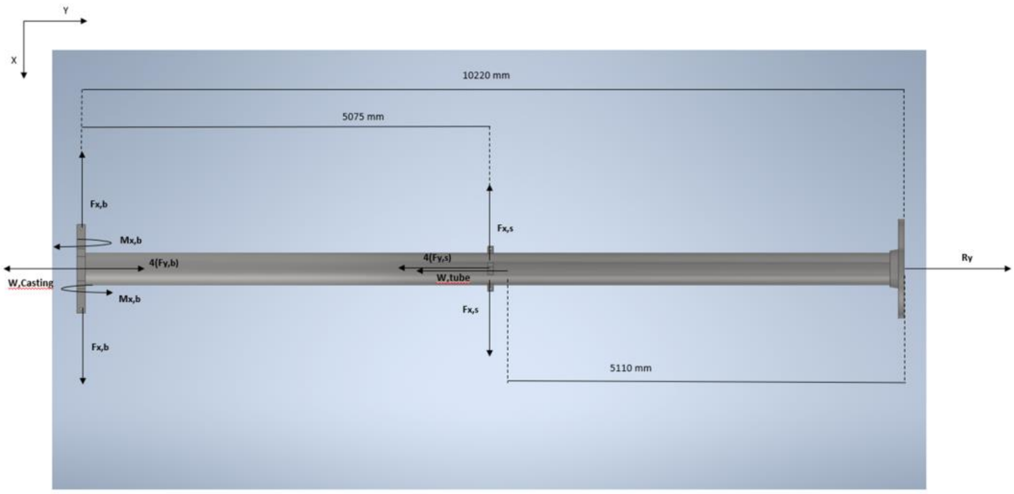


Figure A3.6: Free body diagram illustrating forces acting on torque tube during overloaded conditions(all except torque)

Free body diagram for torque acting on torque tube during overloaded condition is the same as in Figure A3.2

Torque Calculations

$$J := \frac{\pi}{2} \cdot \left(c_o^{\ 4} - c_i^{\ 4} \right)$$

$$J = 0,001559\ m^4$$

$$\tau_{torque,overload} := \left(\frac{T_{max} \cdot c_o}{J} \right)$$

$$\tau_{torque,overload} = 3,399 \cdot 10^7\ Pa$$

Forces on the Tube

The forces acting along the x- and z-axis at the rake and vertical stay connections will cancel each other out therefore, they will not contribute to the shear and moment experienced by the torque tube.

The moments acting about the x- and z-axis at the rake and vertical stay connections will also cancel each other out therefore they will not contribute to the moment experienced by the torque tube.

Shear Calculations

During overloaded conditions the torque tube only experiences shear due to the torque, because the lateral non-rotating notional load is not present during overloaded conditions

$\tau_{max,overload} := \tau_{torque,overload}$

$\tau_{max,overload} = 3,399 \cdot 10^{7}\ \text{Pa}$

Axial Force calculations

From rake connection until the stay connection

$F_{y,b,overload} := (2060,4421\ \text{N} \cdot 4) - W_{b,casting}$

$F_{y,b,overload} = 5412\ \text{N}$

Therefore the bottom section of the tube is in compression

$$\sigma_{axial,b,overload} := \frac{F_{y,b,overload}}{\pi \cdot c_o^{\ 2} - \pi \cdot c_i^{\ 2}}$$

$\sigma_{axial,b,overload} = 1,1768 \cdot 10^{5}\ \text{Pa}$ **Compressive**

From stay connection until the center of mass

$F_{y,s,overload} := -6268,4514\ \text{N} \cdot 4$

$F_{y,s,overload} = -25070\ \text{N}$

$F_{y,m,overload} := F_{y,b,overload} + F_{y,s,overload}$

$F_{y,m,overload} = -19660\ \text{N}$

Therefore this section of the tube is in tension

$$\sigma_{axial,m,overload} := \frac{-F_{y,m,overload}}{\pi \cdot c_o^{\ 2} - \pi \cdot c_i^{\ 2}}$$

$\sigma_{axial,m,overload} = 4,2749 \cdot 10^{5}\ \text{Pa}$ **Tensile**

From the center of mass until the top of the tube

$F_{y,t,overload} := F_{y,m,overload} - W_{tube}$

$F_{y,t,overload} = -58070\ \text{N}$

Therefore this force applies more tension to the tube

$$\sigma_{axial,t,overload} := \frac{-\left(F_{y,t,overload}\right)}{\pi \cdot c_o^{\ 2} - \pi \cdot c_i^{\ 2}}$$

$\sigma_{axial,t,overload} = 1,263 \cdot 10^{6}\ \text{Pa}$ **Tensile**

$\sigma_{axial,max,overload} := \sigma_{axial,t,overload}$

During overloaded conditions there is no stress experienced due to moment due to the absence of the lateral non -rotating notional load

Reaction forces acting on top of pipe

Vertical Force

$R_{y,overload} := -F_{y,t,overload}$

$R_{y,overload} = 58070\ \text{N}$

During overloaded conditions there is no reaction force or reaction moment about the horizontal axis due to the absence of the lateral non-rotating notional load. The bottom of the torque tube will therefore also not be deflected.

When Thickener is empty and not running

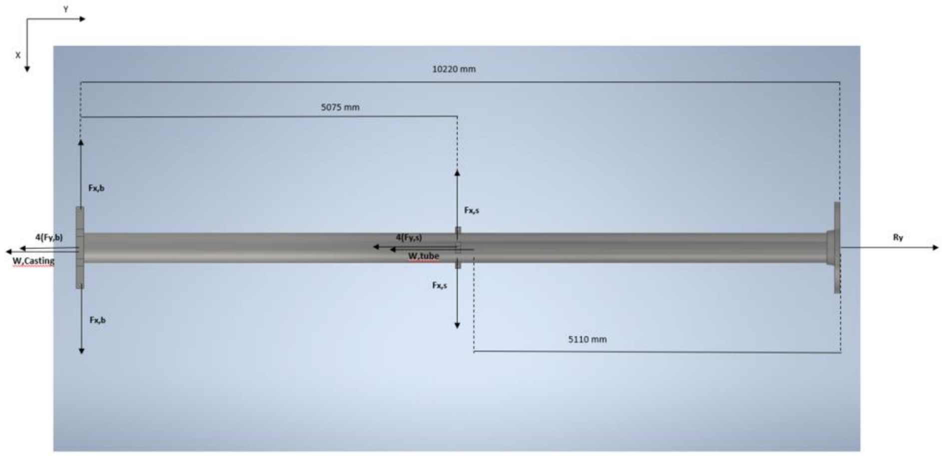


Figure A3.7: Free body diagram illustrating forces acting on torque tube when the thickener is empty and not running (the torque tube does not experience torque during this condition)

Forces on the Tube

The forces acting along the x- and z-axis at the rake and vertical stay connections will cancel each other out therefore, they will not contribute to the shear and moment experienced by the torque tube.

The moments acting about the x- and z-axis at the rake and vertical stay connections will also cancel each other out therefore they will not contribute to the moment experienced by the torque tube.

Shear Calculations

When the thickener is empty and not running the torque tube will not experience torsion and the lateral non-rotating notional load. Therefore, the torque tube will not experience a shear force.

$\tau_{max,empty} := 0\ \text{Pa}$

Axial Force calculations

From rake connection until the stay connection

$F_{y,b,empty} := ((-19383,2903)\ \text{N} \cdot 4) - W_{b,casting}$

$F_{y,b,empty} = -80360\ \text{N}$

Therefore the bottom section of the tube is in tension

$$\sigma_{axial,b,empty} := \frac{-F_{y,b,empty}}{\pi \cdot c_o^{\,2} - \pi \cdot c_i^{\,2}}$$

$\sigma_{axial,b,empty} = 1,7473 \cdot 10^{6}\ \text{Pa}$ **Tensile**

From stay connection until the center of mass

$F_{y,s,empty} := -18187,5023\ \text{N} \cdot 4$

$F_{y,s,empty} = -72750\ \text{N}$

$F_{y,m,empty} := F_{y,b,empty} + F_{y,s,empty}$

$F_{y,m,empty} = -1,531 \cdot 10^{5}\ \mathrm{N}$

Therefore this section of the tube is in tension

$$\sigma_{axial,m,empty} := \frac{-F_{y,m,empty}}{\pi \cdot c_o^{2} - \pi \cdot c_i^{2}}$$

$\sigma_{axial,m,empty} = 3,329 \cdot 10^{6}\ \mathrm{Pa}$ **Tensile**

From the center of mass until the top of the tube

$F_{y,t,empty} := F_{y,m,empty} - W_{tube}$

$F_{y,t,empty} = -1,915 \cdot 10^{5}\ \mathrm{N}$

Therefore this force applies more tension to the tube

$$\sigma_{axial,t,empty} := \frac{-\left(F_{y,t,empty}\right)}{\pi \cdot c_o^{2} - \pi \cdot c_i^{2}}$$

$\sigma_{axial,t,empty} = 4,164 \cdot 10^{6}\ \mathrm{Pa}$ **Tensile**

While the torque tube is empty and not running it will not experience a bending moment as the lateral non-rotating notional load is not present.

Maximum Stress in torque tube

$\sigma_{axial,max,empty} := \sigma_{axial,t,empty}$

$\sigma_{axial,max,empty} = 4,164 \cdot 10^{6}\ \mathrm{Pa}$ **Tensile**

Reaction forces acting on top of pipe

Vertical Force

$R_{y,empty} := -F_{y,t,empty}$

$R_{y,empty} = 1,915 \cdot 10^{5}\ \mathrm{N}$

While the thickener is empty there is no reaction force or reaction moment about the horizontal axis due to the absence of the lateral non-rotating notional load. The bottom of the torque tube will therefore also not be deflected.

Failure conditions

Modes of failure for torque tube

Static failure

The area that is most likely to undergo fatigue failure is where the flange at the top of the toruque tube connects to the torque tube.

$D := 456 \quad d := 406 \quad r := 25$

$\frac{D}{d} = 1,123$

$\frac{r}{d} = 0,06158$

Therefore estimates of the stress concentration factors are:

$k_{t,axial} := 1,85$

$k_{t,bending} := 1,85$

$k_{ts,torque} := 1,35$

Therefore:

$k_t := 1,85$

$k_{ts} := 1,35$

Normal conditions

Tresca Theory

In x-y plane at top of torque tube

$$\sigma_{1,normal} := \frac{0 + k_t \cdot \sigma_{axial,max,normal}}{2} + \sqrt{\left(\frac{0 - k_t \cdot \sigma_{axial,max,normal}}{2}\right)^2 + \left(k_{ts} \cdot \tau_{max,normal}\right)^2} = 4,527 \cdot 10^7 \text{ Pa}$$

$$\sigma_{2,normal} := \frac{0 + k_t \cdot \sigma_{axial,max,normal}}{2} - \sqrt{\left(\frac{0 - k_t \cdot \sigma_{axial,max,normal}}{2}\right)^2 + \left(k_{ts} \cdot \tau_{max,normal}\right)^2} = -2,062 \cdot 10^7 \text{ Pa}$$

$$n_{tresca,normal} := \frac{S_y}{\sigma_{1,normal} - \sigma_{2,normal}} = 4,402$$

Distortion energy/von Mises Theory

$$\sigma'_{normal} := \left(\sigma_{1,normal}{}^2 - \sigma_{1,normal} \cdot \sigma_{2,normal} + \sigma_{2,normal}{}^2\right)^{\frac{1}{2}} = 5,837 \cdot 10^7 \text{ Pa}$$

$$n_{DE,normal} := \frac{S_y}{\sigma'_{normal}} = 4,968$$

Overloaded conditions

Tresca Theory

In x-y plane at top of torque tube

$$\sigma_{1,overload} := \frac{0+k_t \cdot \sigma_{axial,max,overload}}{2} + \sqrt{\left(\frac{0-k_t \cdot \sigma_{axial,max,overload}}{2}\right)^2 + \left(k_{ts} \cdot \tau_{max,overload}\right)^2} = 4,707 \cdot 10^7 \text{ Pa}$$

$$\sigma_{2,overload} := \frac{0+k_t \cdot \sigma_{axial,max,overload}}{2} - \sqrt{\left(\frac{0-k_t \cdot \sigma_{axial,max,overload}}{2}\right)^2 + \left(k_{ts} \cdot \tau_{max,overload}\right)^2} = -4,474 \cdot 10^7 \text{ Pa}$$

$$n_{tresca,overload} := \frac{S_y}{\sigma_{1,overload} - \sigma_{2,overload}} = 3,159$$

Distortion energy/von Mises Theory

$$\sigma'_{overload} := \left(\sigma_{1,overload}{}^2 - \sigma_{1,overload} \cdot \sigma_{2,overload} + \sigma_{2,overload}{}^2\right)^{\frac{1}{2}} = 7,952 \cdot 10^7 \text{ Pa}$$

$$n_{DE,overload} := \frac{S_y}{\sigma'_{overload}} = 3,647$$

When thickener is empty and not running

Tresca Theory

In x-y plane at top of torque tube

$$\sigma_{1,empty} := \frac{0+k_t \cdot \sigma_{axial,max,empty}}{2} + \sqrt{\left(\frac{0-k_t \cdot \sigma_{axial,max,empty}}{2}\right)^2 + \left(k_{ts} \cdot \tau_{max,empty}\right)^2} = 7,704 \cdot 10^6 \text{ Pa}$$

$$\sigma_{2,empty} := \frac{0+k_t \cdot \sigma_{axial,max,empty}}{2} - \sqrt{\left(\frac{0-k_t \cdot \sigma_{axial,max,empty}}{2}\right)^2 + \left(k_{ts} \cdot \tau_{max,empty}\right)^2} = 0$$

$$n_{tresca,empty} := \frac{S_y}{\sigma_{1,empty} - \sigma_{2,empty}} = 37,64$$

Distortion energy/von Mises Theory

$$\sigma'_{empty} := \left(\sigma_{1,empty}{}^2 - \sigma_{1,empty} \cdot \sigma_{2,empty} + \sigma_{2,empty}{}^2\right)^{\frac{1}{2}} = 7,704 \cdot 10^6 \text{ Pa}$$

$$n_{DE,empty} := \frac{S_y}{\sigma'_{empty}} = 37,64$$

Fatigue failure Constant values from textbook

$$S'_e := 0{,}5 \cdot S_u = 2{,}9 \cdot 10^{8}\ \text{Pa}$$

$$k_a := 54{,}9 \cdot 580^{-0{,}758} = 0{,}4415$$

$$k_b := 1{,}51 \cdot 406^{-0{,}157} = 0{,}5881$$

$$k_c := 1$$

$$k_d := 0{,}99 + 5{,}9 \cdot \left(10^{-4}\right) \cdot 20 - 2{,}1 \cdot \left(10^{-6}\right) \cdot 20^{2} = 1{,}001$$

Room temperature(20°C) assumed for kd value.

$$k_e := 1$$

$$S_e := k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot S'_e = 7{,}536 \cdot 10^{7}\ \text{Pa}$$

Shigley fig. $6-20$ and $6-21$ dont go up to 25mm, therefore in order for a conservative analysis to be done:

$$q := 1$$

Therefore:

$$k_f := 1 + 1 \cdot (1{,}85 - 1) = 1{,}85$$

$$k_{fs} := 1 + 1 \cdot (1{,}35 - 1) = 1{,}35$$

Dynamic loads are only present during normal operation therefore, fatigue failure is only assessed for normal operation.

The torque applied to the tube is a fluctuating force. The lateral non-rotating notional load applied to the tube creates a completely reversible shear stress.

$$\tau_{max} := \tau_{torque,normal} = 2{,}244 \cdot 10^{7}\ \text{Pa}$$

$$\tau_{min} := \left(\frac{\left(T_{max} \cdot 0{,}6 \cdot 0{,}9\right) \cdot c_o}{J}\right) = 1{,}836 \cdot 10^{7}\ \text{Pa}$$

$$\tau_{xy,m} := k_{fs} \cdot \left(\frac{\tau_{max} + \tau_{min}}{2}\right) = 2{,}753 \cdot 10^{7}\ \text{Pa}$$

$$\tau_{xy,a} := k_{fs} \cdot \left(\left|\frac{\tau_{max} - \tau_{min}}{2}\right| + \tau_{shear}\right) = 3{,}015 \cdot 10^{6}\ \text{Pa}$$

The lateral non-rotating notional load applied to the tube creates a completely reversible bending moment. Lastly, there is a constant axial force acting.

$$\sigma_{y,m} := k_f \cdot \sigma_{axial,t,normal} = 2{,}486 \cdot 10^{6}\ \text{Pa}$$

$$\sigma_{y,a} := k_f \cdot \frac{\left(4{,}5\ \text{kN} \cdot L_{tube}\right) \cdot c_o}{I} = 2{,}216 \cdot 10^{7}\ \text{Pa}$$

$$\sigma'_a := \left(\sigma_{y,a}^{\ 2} + 3 \cdot \tau_{xy,a}^{\ 2}\right)^{\frac{1}{2}} = 2{,}277 \cdot 10^{7}\ \text{Pa}$$

$$\sigma'_m := \left(\sigma_{y,m}^{\ 2} + 3 \cdot \tau_{xy,m}^{\ 2}\right)^{\frac{1}{2}} = 4{,}776 \cdot 10^{7}\ \text{Pa}$$

Therefore the fatigue factor of safety is:

$$n_f := \left(\frac{\sigma'_a}{S_e} + \frac{\sigma'_m}{S_u}\right)^{-1} = 2{,}601$$

Modes of failure for welding

The weld metal is a mix of the of the object material as well as the actual weld material. Therefore, a weld material must be chosen that has a higher allowable shear than the object material.

According to the Tresca theory:

$$S_{sy} := \frac{S_y}{2} = 1,45 \cdot 10^8 \text{ Pa} \qquad S_{su} := \frac{S_u}{2} = 2,9 \cdot 10^8 \text{ Pa}$$

Therefore a weld meterial of code E80 to E120 should be used.

The weld will therefore fail if the shear stress in the weld exceeds the allowed shear stress of the torque tube material.

Torque tube is the thickest between torque tube and flange with a thickness of 40mm

Therefore, minumum weld size is 10mm. Weld size can not be more than thickness of the thinner part which is the flange (25mm thickness). Weld size can therefore be 10 - 25mm.

$$h_{weld} := 25 \text{ mm}$$

$$A_w := 1,414 \cdot \pi \cdot h_{weld} \cdot c_o = 0,0225 \text{ m}^2$$

$$I_{u,weld} := \pi \cdot \left(c_o\right)^3 = 0,0263 \text{ m}^3 \qquad I_{weld} := 0,707 \cdot h_{weld} \cdot I_{u,weld} = 0,0005 \text{ m}^4$$

$$J_{u,weld} := 2 \cdot \pi \cdot \left(c_o\right)^3 = 0,0526 \text{ m}^3 \qquad J_{weld} := 0,707 \cdot h_{weld} \cdot J_{u,weld} = 0,0009 \text{ m}^4$$

Static failure

Normal conditions:

$$\tau'_{axial,normal} := \frac{-F_{y,t,normal}}{A_w} = 2,7409 \cdot 10^6 \text{ Pa}$$

$$\tau''_{moment,normal} := \frac{c_o \cdot \left(4,5 \text{ kN} \cdot L_{tube}\right)}{I_{weld}} = 2,0098 \cdot 10^7 \text{ Pa}$$

$$\tau''_{torque,normal} := \frac{T_{normal} \cdot c_o}{J_{weld}} = 3,764 \cdot 10^7 \text{ Pa}$$

$$\tau_{weld,normal} := \sqrt{\left(\tau'_{axial,normal}\right)^2 + \left(\tau''_{moment,normal}\right)^2 + \left(\tau''_{torque,normal}\right)^2} = 4,2758 \cdot 10^7 \text{ Pa}$$

$$n_{tresca,weld,normal} := \frac{S_{sy}}{\tau_{weld,normal}} = 3,3912$$

Overloaded conditions:

$$\tau'_{axial,overload} := \frac{-F_{y,t,overload}}{A_w} = 2,576 \cdot 10^6 \text{ Pa}$$

$$\tau''_{torque,overload} := \frac{T_{max} \cdot c_o}{J_{weld}} = 5,7031 \cdot 10^7 \text{ Pa}$$

$$\tau_{weld,overload} := \sqrt{\left(\tau'_{axial,overload}\right)^2 + \left(\tau''_{torque,overload}\right)^2} = 5,7089 \cdot 10^7 \text{ Pa}$$

$$n_{tresca,weld,overload} := \frac{S_{sy}}{\tau_{weld,overload}} = 2,5399$$

When thickener is empty and not running:

$$\tau'_{axial,empty} := \frac{-F_{y,t,empty}}{A_w} = 8,4956 \cdot 10^{6}\ \text{Pa}$$

$$\tau_{weld,empty} := \tau'_{axial,empty} = 8,4956 \cdot 10^{6}\ \text{Pa}$$

$$n_{tresca,weld,empty} := \frac{S_{sy}}{\tau_{weld,empty}} = 17,0677$$

Fatigue Failure:

The the Sse value can be calculated from the Se value previousy calculated. This is because as stated before the stregth of the torque tube material will be the limiting factor.

According to the Tresca theory:

$$S_{se} := \frac{S_e}{2} = 3,7681 \cdot 10^{7}\ \text{Pa}$$

Transverse fillet weld is used therefore:

$$k_{fs,w} := 1,5$$

Once again only normal operating conditions will be analysed for fatigue failure because it is the only condition in which dynamic loads are present.

The axial force due to the vertical forces applied by the rakes as well as the weight of the torque tube will be constant. The bending moment created by the lateral non-rotating notional load is completely reversible and the torque is a fluctuating load.

$$\tau'_{m,axial} := k_{fs,w} \cdot \frac{-F_{y,t,normal}}{A_w} = 4,1114 \cdot 10^{6}\ \text{Pa}$$

$$\tau''_{a,moment} := k_{fs,w} \cdot \frac{c_o \cdot \left(4,5\ \text{kN} \cdot L_{tube}\right)}{I_{weld}} = 3,0148 \cdot 10^{7}\ \text{Pa}$$

$$\tau''_{max,torque,weld} := \frac{T_{normal} \cdot c_o}{J_{weld}} = 3,764 \cdot 10^{7}\ \text{Pa}$$

$$\tau''_{min,torque,weld} := \frac{T_{max} \cdot 0,6 \cdot 0,9 \cdot c_o}{J_{weld}} = 3,0797 \cdot 10^{7}\ \text{Pa}$$

$$\tau''_{m,torque} := k_{fs,w} \cdot \left(\frac{\tau''_{max,torque,weld} + \tau''_{min,torque,weld}}{2}\right) = 5,133 \cdot 10^{7}\ \text{Pa}$$

$$\tau''_{a,torque} := k_{fs,w} \cdot \left(\left|\frac{\tau''_{max,torque,weld} - \tau''_{min,torque,weld}}{2}\right|\right) = 5,133 \cdot 10^{6}\ \text{Pa}$$

$$\tau_{m,weld} := \sqrt{\left(\tau'_{m,axial}\right)^2 + \left(\tau''_{m,torque}\right)^2} = 5,1492 \cdot 10^{7}\ \text{Pa}$$

$$\tau_{a,weld} := \sqrt{\left(\tau''_{a,moment}\right)^2 + \left(\tau''_{a,torque}\right)^2} = 3,0581 \cdot 10^{7}\ \text{Pa}$$

$$n_{weld,fatigue} := \left(\frac{\tau_{a,weld}}{S_{se}} + \frac{\tau_{m,weld}}{S_{su}}\right)^{-1} = 1,011$$

The safety facor is 1, this means that the stress experienced is less than the endurance strength, therefore the weld will have an infinite life.

Also as stated before, the strength of the material of the object was used for the weld during the analysis. However, in reality the weld metal is a mixture of the object material and the actual weld material. The actual weld material will have a higher strength than the object material. Therefore, due to the weld material being a mixture its strength will be higher than the object material due to it being mixed with the stronger weld material. The design factor will therefore be slightly higher in reality.

Calculations for connections between torque tube and rakes

Pin Connection at rake connection

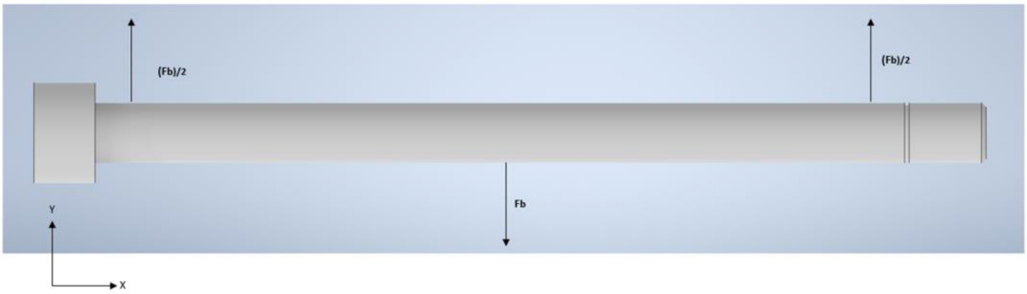


Figure A3.8: Free body diagram illustrating forces analysed to calculate shear in the pin where the rake connects to the torque tube

The largest force experienced by the pin at the rake connection is while the thickener is empty and not running

$$F_{b,pin,empty} := \sqrt{(-19383{,}2903\ \mathrm{N})^2 + ((-15360{,}6697)\ \mathrm{N})^2} = 24730\ \mathrm{N}$$

$$V_{b,pin,empty} := \frac{1}{2} \cdot F_{b,pin,empty}$$

$$V_{b,pin,empty} = 12370\ \mathrm{N}$$

$$d_{pin,b} := 30\ \mathrm{mm}$$

$$c_{pin,b} := \frac{d_{pin,b}}{2}$$

$$c_{pin,b} = 0{,}015\ \mathrm{m}$$

Therefore the maximum shear experienced in the pin will be in the y direction

$$V_{max,pin,b,overload} := V_{b,pin,empty} = 12370\ \mathrm{N}$$

$$Q_{pin,b} := \frac{1}{2} \cdot \pi \cdot c_{pin,b}^{\ 2} \cdot \frac{4 \cdot c_{pin,b}}{3 \cdot \pi}$$

$$Q_{pin,b} = 2{,}25 \cdot 10^{-6}\ \mathrm{m}^3$$

$$I_{pin,b} := \frac{\pi}{64} \cdot d_{pin,b}^{\ 4}$$

$$I_{pin,b} = 3{,}976 \cdot 10^{-8}\ \mathrm{m}^4$$

$$\tau_{pin,b} := \frac{V_{max,pin,b,overload} \cdot Q_{pin,b}}{I_{pin,b} \cdot d_{pin,b}}$$

$$\tau_{pin,b} = 2{,}333 \cdot 10^{7}\ \mathrm{Pa}$$

Therefore desgin factor is:

$$n_d := \frac{\frac{S_y}{2}}{\tau_{pin,b}} = 6{,}216$$

Pin Connection at stay connection

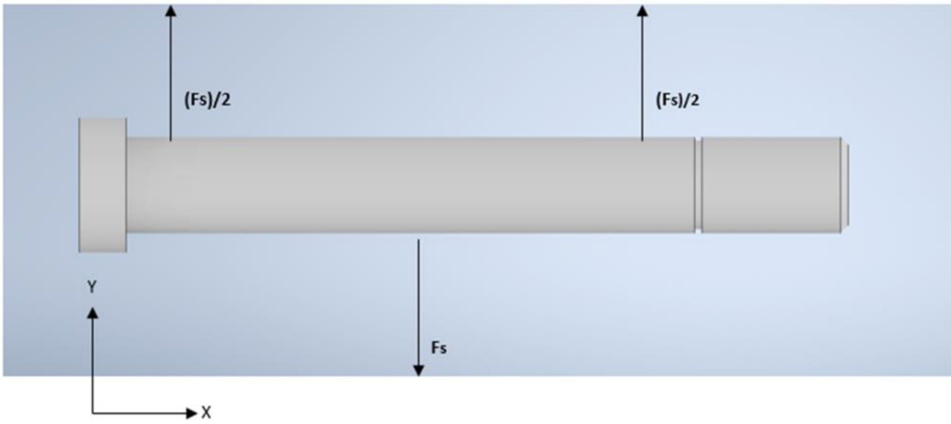


Figure A3.9: Free body diagram illustrating forces analysed to calculate shear in the pin where the vertical stay connects to the torque tube

The largest force experienced by the pin at the stay connection is while the thickener is empty and not running

$$F_{s,pin,empty} := \sqrt{(31247{,}7612\ \mathrm{N})^2 + ((-18187{,}5023)\ \mathrm{N})^2} = 36160\ \mathrm{N}$$

$$d_{pin,s} := 25\ \mathrm{mm}$$

$$c_{pin,s} := \frac{d_{pin,s}}{2}$$

$$c_{pin,s} = 0{,}0125\ \mathrm{m}$$

$$V_{s,pin,empty} := \frac{1}{2} \cdot F_{s,pin,empty}$$

$$V_{s,pin,empty} = 18080\ \mathrm{N}$$

Therefore the maximum shear experienced in the pin will be in the x direction

$$V_{max,pin,s,overload} := V_{s,pin,empty} = 18080\ \mathrm{N}$$

$$Q_{pin,s} := \frac{1}{2} \cdot \pi \cdot c_{pin,s}^{\ 2} \cdot \frac{4 \cdot c_{pin,s}}{3 \cdot \pi}$$

$$Q_{pin,s} = 1{,}302 \cdot 10^{-6}\ \mathrm{m}^3$$

$$I_{pin,s} := \frac{\pi}{64} \cdot d_{pin,s}^{\ 4}$$

$$I_{pin,s} = 1{,}918 \cdot 10^{-8}\ \mathrm{m}^4$$

$$\tau_{pin,s} := \frac{V_{max,pin,s,overload} \cdot Q_{pin,s}}{I_{pin,s} \cdot d_{pin,s}}$$

$$\tau_{pin,s} = 4{,}91 \cdot 10^{7}\ \mathrm{Pa}$$

Therefore desgin factor is:

$$n_d := \frac{\frac{S_y}{2}}{\tau_{pin,s}} = 2{,}953$$

Casting for pin connection at bottom of torque tube

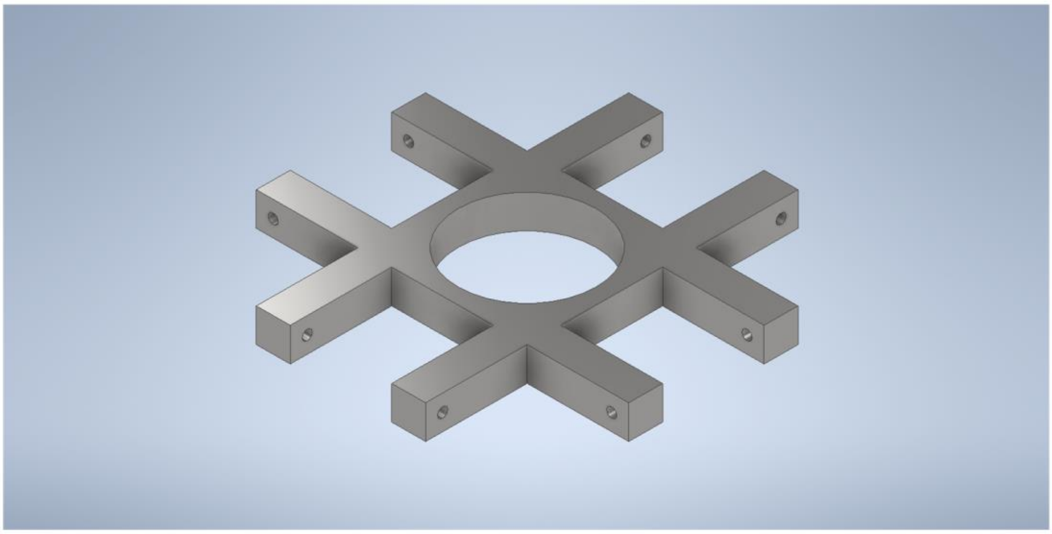


Figure A3.10: Casting for connection of the rakes to the bottom of the torque tube.

Stress concentration factor at hole for pin:

$k_{t,b,casting,axial} := 4$

$A_{b,casting} := (0,1\ \mathrm{m} - 0,03\ \mathrm{m}) \cdot 0,1\ \mathrm{m} = 0,007\ \mathrm{m}^2$

Normal Conditions

Reaction forces are divided by 2 because, the force is split evenly between both parts of the casting.

$$\sigma_{b,casting,axial,normal} := k_{t,b,casting,axial} \cdot \frac{\left(\frac{5528,8293\ \mathrm{N}}{2}\right)}{A_{b,casting}}$$

$\sigma_{b,casting,axial,normal} = 1,58 \cdot 10^6\ \mathrm{Pa}$

$\sigma_{b,casting,max,normal} := \sigma_{b,casting,axial,normal} = 1,58 \cdot 10^6\ \mathrm{Pa}$

$$n_{b,casting,normal} := \frac{S_y}{\sigma_{b,casting,max,normal}} = 183,6$$

Overloaded Conditions

Reaction forces are divided by 2 because, the force is split evenly between both parts of the casting.

$$\sigma_{b,casting,axial,overload} := k_{t,b,casting,axial} \cdot \frac{\left(\frac{1938,4381\ \mathrm{N}}{2}\right)}{A_{b,casting}}$$

$\sigma_{b,casting,axial,overload} = 5,538 \cdot 10^5\ \mathrm{Pa}$

$$\sigma_{b,casting,max,overload} := \sigma_{b,casting,axial,overload} = 5{,}538 \cdot 10^{5}\ \mathrm{Pa}$$

$$n_{b,casting,overload} := \frac{S_y}{\sigma_{b,casting,max,overload}} = 523{,}6$$

When thickener is empty and not running

Reaction forces are divided by 2 because, the force is split evenly between both parts of the casting.

$$\sigma_{b,casting,axial,empty} := k_{t,b,casting,axial} \cdot \frac{\left(\dfrac{15360{,}6697\ \mathrm{N}}{2}\right)}{A_{b,casting}}$$

$$\sigma_{b,casting,axial,empty} = 4{,}389 \cdot 10^{6}\ \mathrm{Pa}$$

$$\sigma_{b,casting,max,empty} := \sigma_{b,casting,axial,empty} = 4{,}389 \cdot 10^{6}\ \mathrm{Pa}$$

$$n_{b,casting,empty} := \frac{S_y}{\sigma_{b,casting,max,empty}} = 66{,}08$$

Casting for pin connection where vertical stays connect of torque tube

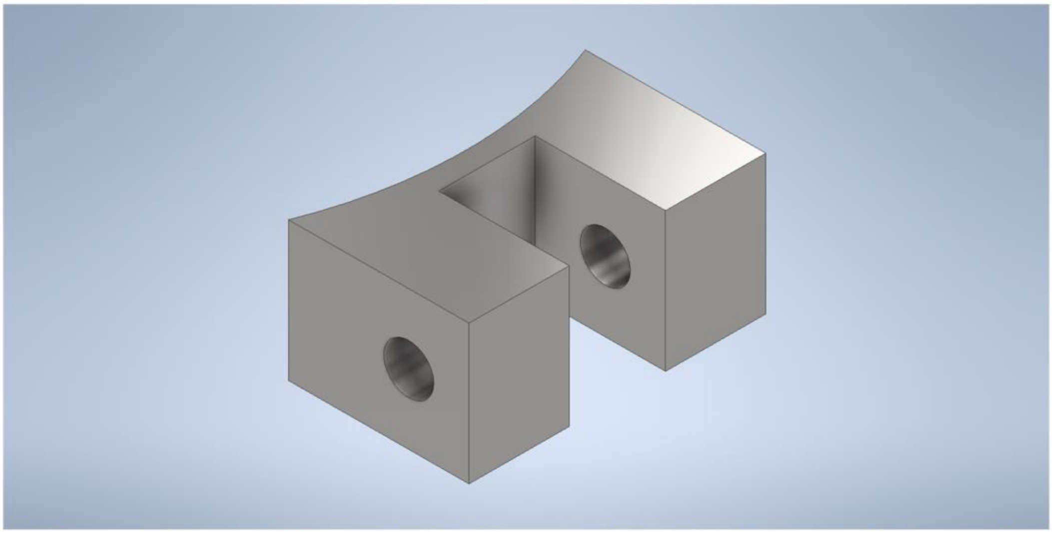


Figure A3.11: Casting for connection of the vertical stays to the torque tube

Stress concentration factor at hole for pin:

$k_{t,s,casting,axial} := 4$

$A_{s,casting} := (0,07\ \mathrm{m} - 0,025\ \mathrm{m}) \cdot 0,05\ \mathrm{m} = 0,00225\ \mathrm{m}^2$

Normal Conditions

Reaction forces are divided by 2 because, the force is split evenly between both parts of the casting.

$$\sigma_{s,casting,axial,normal} := k_{t,s,casting,axial} \cdot \frac{\left(\frac{10603,3595\ \mathrm{N}}{2}\right)}{A_{s,casting}}$$

$\sigma_{s,casting,axial,normal} = 9,425 \cdot 10^6\ \mathrm{Pa}$

$\sigma_{s,casting,max,normal} := \sigma_{s,casting,axial,normal} = 9,425 \cdot 10^6\ \mathrm{Pa}$

$$n_{s,casting,normal} := \frac{S_y}{\sigma_{s,casting,max,normal}} = 30,77$$

Overloaded Conditions

Reaction forces are divided by 2 because, the force is split evenly between both parts of the casting.

$$\sigma_{s,casting,axial,overload} := k_{t,s,casting,axial} \cdot \frac{\left(\frac{10603,3595\ \mathrm{N}}{2}\right)}{A_{s,casting}}$$

$\sigma_{s,casting,axial,overload} = 9,425 \cdot 10^6\ \mathrm{Pa}$

$$\sigma_{s,casting,max,overload} := \sigma_{s,casting,axial,overload} = 9,425 \cdot 10^{6}\ \text{Pa}$$

$$n_{s,casting,overload} := \frac{S_y}{\sigma_{s,casting,max,overload}} = 30,77$$

When thickener is empty and not running

Reaction forces are divided by 2 because, the force is split evenly between both parts of the casting.

$$\sigma_{s,casting,axial,empty} := k_{t,s,casting,axial} \cdot \frac{\left(\dfrac{31247,7612\ \text{N}}{2}\right)}{A_{s,casting}}$$

$$\sigma_{s,casting,axial,empty} = 2,778 \cdot 10^{7}\ \text{Pa}$$

$$\sigma_{s,casting,max,empty} := \sigma_{s,casting,axial,empty} = 2,778 \cdot 10^{7}\ \text{Pa}$$

$$n_{s,casting,empty} := \frac{S_y}{\sigma_{s,casting,max,empty}} = 10,44$$

Bolt calculations for slew ring:

The distance of each bolt from the centre of the of the slew ring is 574.6 mm. This is given by the slew ring catalogue.

Each bolt is the same distance from the centre of the slew ring and there for the r is the same

$\tau := \frac{T \cdot r}{J}$ where J is $\int r^2 \, dA$

The integral of dA is the area of each bolt and therefore the calculation can be simplified

$$\tau := \frac{T \cdot r}{r^2 \cdot A}$$

$r := 574{,}6 \ \text{mm}$

For Overload conditions:

$T_{full} := 260 \ \text{kN m}$

$A := \frac{\pi}{4} \cdot (23{,}8 \ \text{mm})^2 = 0{,}0004 \ \text{m}^2$

$$F_{full} := \frac{T_{full} \cdot r}{48 \cdot r^2} = 9426{,}8477 \ \text{N}$$

$$\sigma_{on\_each\_bolt} := \frac{F_{full}}{A} = 2{,}119 \cdot 10^7 \ \text{Pa}$$

The bolts used will be from the property class 8.8. Thus the yield stress is 660 Mpa

$\sigma_{yield} := 660 \ \text{MPa}$

$$n := \frac{\sigma_{yield}}{\sigma_{on\_each\_bolt}} = 31{,}1474$$

For normal conditions

$T_{normal} := T_{full} \cdot 0{,}6$

$$F_{normal} := \frac{T_{normal} \cdot r}{48 \cdot r^2} = 5656{,}1086 \ \text{N}$$

$$\sigma_{on\_each\_bolt,normal} := \frac{F_{normal}}{A} = 1{,}2714 \cdot 10^7 \ \text{Pa}$$

The bolts used will be from the property class 8.8. Thus the yield stress is 660 Mpa

$\sigma_{yield} := 660 \ \text{MPa}$

$$n := \frac{\sigma_{yield}}{\sigma_{on\_each\_bolt,normal}} = 51{,}9123$$

A.4 Sub-system 4: Rakes (NE)

RAKE ARM CALCULATIONS (Overload Condition):

Torqu Calculations

$T_{max} := K \cdot D^2$ Max Torque

$K := 290 \frac{N}{m}$

$D := 30\ m$

$T_{max} = 2.61 \cdot 10^5\ N\ m$ Total Maximum torque

End Plates

$V_{EP} := 0.3\ m \cdot 0.3\ m \cdot 0.01\ m$

$V_{EP} = 0.0009\ m^3$

Rakes Dimensions:

$l_R := 15.43\ m$ length of rake

$b_{Rout} := 0.3\ m$ bredth rake

$h_{Rout} := 0.3\ m$ height of rake

$t_R := 0.01\ m$ thickness of rake

$b_{Rin} := b_{Rout} - (2 \cdot t_R)$

$b_{Rin} = 0.28\ m$

$h_{Rin} := h_{Rout} - (2 \cdot t_R)$

$h_{Rin} = 0.28\ m$

$V_R := l_R \cdot b_{Rout} \cdot h_{Rout} - l_R \cdot b_{Rin} \cdot h_{Rin}$

$V_R = 0.179\ m^3$ Volume of Rake tube tubing, used for mass calculation

$V_{Rfloat} := l_R \cdot b_{Rout} \cdot h_{Rout}$

$V_{Rfloat} = 1.3887\ m^3$ Volume of total rake tube, for buoyancy calculation

Blade Dimensions

$l_B := 2\ m$

$h_B := 0.2\ m$

$t_B := 0.02\ m$

$V_B := l_B \cdot h_B \cdot t_B$

$V_B = 0.008\ m^3$ Volume of Blades, used for mass and buoyancy calculations

Vertical stay Dimensions

$l_V := 6.463\ m$

$r_{Vout} := \frac{0.048}{2}\ m$

$r_{Vin} := 0.022\ m$ thickness is 2 mm

$V_V := l_V \cdot \left(\pi \cdot \left(r_{Vout}^2 - r_{Vin}^2\right)\right)$

$V_V = 0.0019\ m^3$ Vertical Stay is barely submerged, and is therefore not included in buoyancy force

Horizontal stay dimensions

$l_H := 7.751\ m$

$r_{Hout} := \frac{0.048}{2}\ m$

$r_{Hin} := 0.022\ m$ thickness is 2 mm

$V_H := l_H \cdot \left(\pi \cdot \left(r_{Hout}^2 - r_{Hin}^2\right)\right)$

$V_H = 0.0022\ m^3$

$V_{Hfloat} := \pi \cdot r_{Hout}^2 \cdot l_H$

$V_{Hfloat} = 0.014\ m^3$ Vertical Stay is fully submerged and therefore also contributes to floatation

Constants

$E := 193\ GPa$ Elastic dulus of material

$\sigma_{yield} := 290\ MPa$ Yield strength of material

$\sigma_{maxtensile} := 580\ MPa$ Ultimate ength of material

$p := 8000 \frac{kg}{m^3}$ Properties of AISI 316

$g := 9.81 \frac{m}{s^2}$ Gravitational force

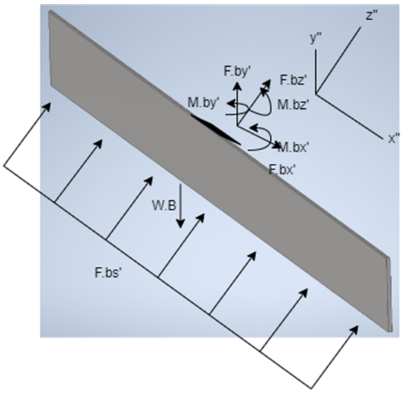
$V_{buoyancy} := V_{Rfloat} + V_B + V_{Hfloat}$

$V_{buoyancy} = 1.4107\ m^3$ Total volume causing buoyancy foorce

$p_{water} := 997 \frac{kg}{m^3}$ Water density, since sludge density is not known. Water will have a lower density, resulting in lower buoyancy force and therefore overstated the forces caused in the rake arm, and may therefore be used for the calculations.

$v := 0.27$

BLADES:



$$T_t := \frac{T_{max}}{4}$$

$T_t = 65250\ \mathrm{N\,m}$ Maximum torque per rake arm

$$w_{bz} := \frac{T_t}{\left(\frac{2 \cdot l_R}{3}\right)^2}$$

$w_{bz} = 616.6393\ \frac{\mathrm{N}}{\mathrm{m}}$ Distributed load of sludge on outer $\frac{2}{3}$ of rake arm

$$F_{bz} := w_{bz} \cdot \frac{2}{3} \cdot l_R \cdot \frac{1}{6} \qquad F_{bx} := \frac{F_{bz}}{\cos(45\ \mathrm{deg})} \cdot \sin(45\ \mathrm{deg}) \qquad F_{by} := p \cdot g \cdot V_B$$

$$F_{bz} = 1057.1938\ \mathrm{N} \qquad F_{bx} = 1057.1938\ \mathrm{N} \qquad F_{by} = 627.84\ \mathrm{N}$$

Converting the forces to forces in a new coordinate system alligned with the blade:

$$F_{bx'} := \left(F_{bz} \cdot \sin(45\ \mathrm{deg}) - F_{bx} \cdot \cos(45\ \mathrm{deg})\right) \qquad F_{by'} := F_{by} \cdot \cos(15\ \mathrm{deg})$$

$$F_{bz'} := \left(F_{bz} \cdot \cos(45\ \mathrm{deg}) + F_{bx} \cdot \sin(45\ \mathrm{deg})\right) - p \cdot g \cdot V_B \cdot \sin(15\ \mathrm{deg})$$

The reaction forces at the top of each blade, in the blades coordinate system are:

$$F_{bz'} = 1332.6008\ \mathrm{N} \qquad F_{bx'} = 0\ \mathrm{N} \qquad F_{by'} = 606.4469\ \mathrm{N}$$

$$M_{bx'} := F_{bz'} \cdot \frac{h_B}{2}$$

$$M_{bz'} := 0\ \mathrm{N\,m} \qquad M_{bx'} = 133.2601\ \mathrm{N\,m} \qquad M_{by'} := 0\ \mathrm{N\,m}$$

Finding the maximum bending moments and shear forces:

<u>x'-z' plane:</u>

$$w_{z'1} := \frac{F_{bz'}}{l_B}$$

$$w_{z'1} = 666.3004\ \frac{\mathrm{N}}{\mathrm{m}} \qquad BM_{y'}(x') := \frac{w_{z'1} \cdot x'^2}{2}$$

$$SF_{z'1}(x') := w_{z'1} \cdot x'$$

$$SF_{z'1}(1\ \mathrm{m}) = 666.3004\ \mathrm{N} \qquad BM_{y'}(1\ \mathrm{m}) = 333.1502\ \mathrm{N\,m} \qquad SF_{x'} := 0\ \mathrm{N}$$

<u>y'-z' plane:</u>

$$w_{z'2} := \frac{F_{bz'}}{h_B}$$

$$w_{z'2} = 6663.0042\ \frac{\mathrm{N}}{\mathrm{m}}$$

$$SF_{z'2}(y') := w_{z'2} \cdot y' \qquad BM_{x'}(y') := \frac{w_{z'2} \cdot y'^2}{2}$$

$$SF_{z'2}(0.2\ \mathrm{m}) = 1332.6008\ \mathrm{N} \qquad BM_{x'}(0.2\ \mathrm{m}) = 133.2601\ \mathrm{N\,m}$$

$$SF_{z'max} := \begin{bmatrix} \left(SF_{z'2}(0.2\ \mathrm{m})\right) \\ SF_{z'1}(1\ \mathrm{m}) \end{bmatrix}$$

$$SF_{z'} := \max\left(SF_{z'max}\right)$$

$$SF_{z'} = 1332.6008\ \mathrm{N}$$

x'-y' plane:

$$w_{y'} := \frac{F_{by'}}{l_B}$$

$$w_{y'} = 303.2234\ \frac{\mathrm{N}}{\mathrm{m}}$$

$$SF_{y'}(x') := w_{y'} \cdot x' \qquad BM_{z'}(x') := \frac{w_{y'} \cdot x'^2}{2}$$

$$SF_{y'}(1\ \mathrm{m}) = 303.2234\ \mathrm{N} \qquad BM_{z'}(1\ \mathrm{m}) = 151.6117\ \mathrm{N\ m}$$

Calculating the Normal Stresses and Shear Stresses

$$I_{y'y'} := \frac{h_B \cdot t_B^{\,3}}{12} \qquad I_{x'x'} := \frac{l_B \cdot t_B^{\,3}}{12} \qquad I_{z'z'} := \frac{t_B \cdot h_B^{\,3}}{12}$$

$$z'bar := \frac{t_B}{2} \qquad y'bar := \frac{h_B}{2}$$

$$\sigma_{z'} := 0 \qquad \sigma_{y'} := \frac{F_{by'}}{l_B \cdot t_B} + \frac{BM_{x'}(0.2\ \mathrm{m}) \cdot z'bar}{I_{x'x'}} \qquad \sigma_{x'} := \frac{BM_{y'}(1\ \mathrm{m}) \cdot z'bar}{I_{y'y'}} + \frac{BM_{z'}(1\ \mathrm{m}) \cdot y'bar}{I_{z'z'}}$$

$$\sigma_{z'} = 0\ \mathrm{MPa} \qquad \sigma_{y'} = 1.0146\ \mathrm{MPa} \qquad \sigma_{x'} = 26.1234\ \mathrm{MPa}$$

Tension

Tension or Compression

$$Q_{y'z'} := \frac{t_B}{2} \cdot l_B \cdot \frac{t_B}{4} \qquad Q_{x'y'} := t_B \cdot \frac{h_B}{2} \cdot \frac{h_B}{4} \qquad Q_{x'z'} := \frac{t_B}{2} \cdot h_B \cdot \frac{t_B}{4}$$

$$\tau_{y'z'} := \frac{SF_{z'2}(0.2\ \mathrm{m}) \cdot Q_{y'z'}}{I_{x'x'} \cdot l_B} \qquad \tau_{x'y'} := \frac{SF_{y'}(1\ \mathrm{m}) \cdot Q_{x'y'}}{I_{z'z'} \cdot t_B} \qquad \tau_{x'z'} := \frac{SF_{z'1}(1\ \mathrm{m}) \cdot Q_{x'z'}}{I_{y'y'} \cdot h_B}$$

$$\tau_{y'z'} = 0.05\ \mathrm{MPa} \qquad \tau_{x'y'} = 0.1137\ \mathrm{MPa} \qquad \tau_{x'z'} = 0.2499\ \mathrm{MPa}$$

Maximum stresses in the blades:

Plane Stresses

$$\sigma_{x'} = 2.6123 \cdot 10^{7}\ \mathrm{Pa} \qquad \tau_{x'z'} = 2.4986 \cdot 10^{5}\ \mathrm{Pa}$$

$$\sigma_{y'} = 1.0146 \cdot 10^{6}\ \mathrm{Pa} \qquad \tau_{x'y'} = 1.1371 \cdot 10^{5}\ \mathrm{Pa}$$

$$\sigma_{z'} = 0\ \mathrm{Pa} \qquad \tau_{y'z'} = 49972.5311\ \mathrm{Pa}$$

Finding the 3 Principle stresses for the blades:

$$aa' := 1$$

$$bb' := -\left(\sigma_{x'} + \sigma_{y'} + \sigma_{z'}\right) = -2.7138 \cdot 10^{7}\ \mathrm{Pa}$$

$$cc' := \left[\left(\sigma_{x'} \cdot \sigma_{y'}\right) + \left(\sigma_{x'} \cdot \sigma_{z'}\right) + \left(\sigma_{y'} \cdot \sigma_{z'}\right) - \left(\tau_{x'y'}\right)^2 - \left(\tau_{y'z'}\right)^2 - \left(\tau_{x'z'}\right)^2\right] = 2.6427 \cdot 10^{13}\ \mathrm{Pa}^2$$

$$da' := \left(\sigma_{x'} \cdot \sigma_{y'} \cdot \sigma_{z'}\right) + \left(2 \cdot \tau_{x'y'} \cdot \tau_{y'z'} \cdot \tau_{x'z'}\right) = 2.8396 \cdot 10^{15}\ \mathrm{Pa}^3$$

$$dd' := -\left[da' - \left(\sigma_{x'} \cdot \tau_{y'z'}^{\,2}\right) - \left(\sigma_{y'} \cdot \tau_{x'z'}^{\,2}\right) - \left(\sigma_{z'} \cdot \tau_{x'y'}^{\,2}\right)\right]$$

$$dd' = 1.2574 \cdot 10^{17}\ \mathrm{Pa}^3$$

$$\sigma_{1'2'3'} := \mathrm{polyroots}\left(\begin{bmatrix} dd' \\ cc' \\ bb' \\ aa' \end{bmatrix}\right) = \begin{bmatrix} -4734.9727 \\ 1.0164 \cdot 10^{6} \\ 2.6126 \cdot 10^{7} \end{bmatrix}$$

Finding the Principal Stresses using Shigley's equation $(3-15)$ and solving for its roots using polyroots function

$$\sigma_{1'} := \mathrm{row}\left(\sigma_{1'2'3'}, 3\right)\ \mathrm{Pa} = \left[2.6126 \cdot 10^{7}\right]\ \mathrm{Pa}$$

$$\sigma_{2'} := \mathrm{row}\left(\sigma_{1'2'3'}, 2\right)\ \mathrm{Pa} = \left[1.0164 \cdot 10^{6}\right]\ \mathrm{Pa}$$

$$\sigma_{3'} := \mathrm{row}\left(\sigma_{1'2'3'}, 1\right)\ \mathrm{Pa} = \left[-4734.9727\right]\ \mathrm{Pa}$$

Calculating the safety factor:

By Maximum-Shear-Stress (Tresca) theory: ref A4.1

$$n_{MSS'} := \frac{\sigma_{yield}}{\sigma_{1'} - \sigma_{3'}} = [\,11.0979\,]$$

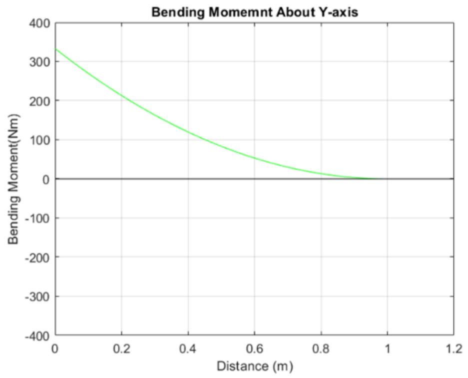
By Distorsion-Energy (Von Mises) theory:

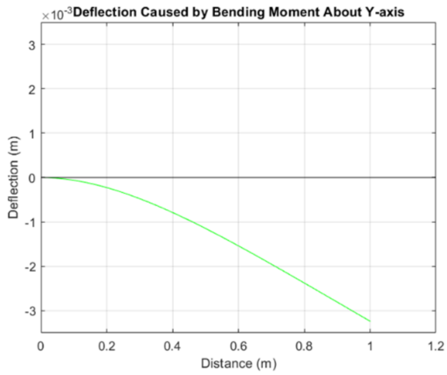
$$\sigma_{DE'} := \left(\frac{(\sigma_{1'} - \sigma_{2'})^2 + (\sigma_{2'} - \sigma_{3'})^2 + (\sigma_{3'} - \sigma_{1'})^2}{2} \right)^{\frac{1}{2}}$$

$$n_{DE'} := \frac{\sigma_{yield}}{\sigma_{DE'}} = 11.3124$$

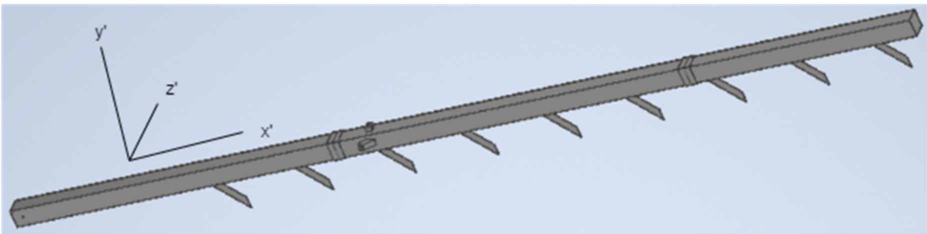
Bending Moment and Deflection graphs, where x(0) is the centre of the blade, where it is connected to the rake arm and 1m is the outer most side of the blade

ref A4.2





Rake Arm:



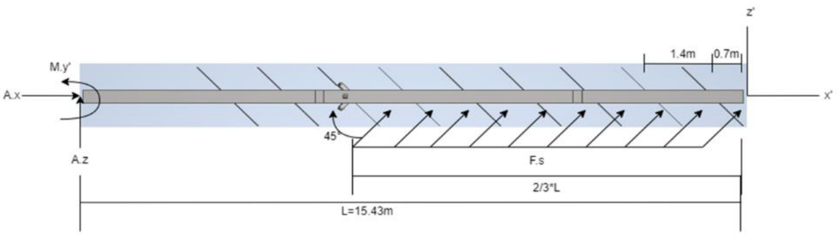
$$a := \frac{2\ \mathrm{m} \cdot \cos(45\ \mathrm{deg})}{2} \qquad b := 2\ \mathrm{m} \cdot \cos(45\ \mathrm{deg}) \qquad t := \frac{t_B}{\cos(45\ \mathrm{deg})}$$

varibles for placement of blade point loads along x'-axis of rake arm

$$c := (l_R - a) + (l_R - a - (b+t)) + (l_R - a - 2 \cdot (b+t)) + (l_R - a - 3 \cdot (b+t)) + (l_R - a - 4 \cdot (b+t)) + \\ + (l_R - a - 5 \cdot (b+t)) + (l_R - a - 6 \cdot (b+t)) + (l_R - a - 7 \cdot (b+t)) + (l_R - a - 8 \cdot (b+t))$$

$$c = 80.5761\ \mathrm{m}$$

Calculations for the x-z axis

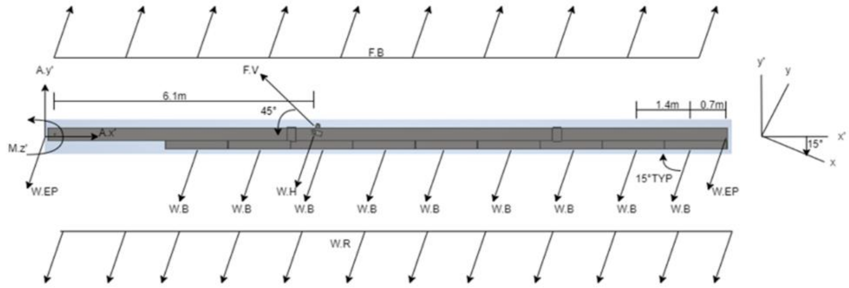


$$M_y := \frac{T_{max}}{4} = 65250 \text{ N m}$$

$$F_s := \frac{M_y}{\left(\frac{2}{3} \cdot l_R\right) \cdot \left(\frac{2}{3} \cdot l_R \cdot \sin(75 \text{ deg})\right) \cdot \cos(45 \text{ deg})} = 902.8226 \ \frac{\text{N}}{\text{m}}$$

Sludge force actingt in the x',z' direction

Calculations for the x-y axis



$M_{z'} := 0 \text{ N m}$ Since a pin support is used

$F_{buoyancy} := p_{water} \cdot g \cdot V_{buoyancy}$ Buoyancy force induced by submerging the rake

$$F_{buoyancy} = 13.7977 \text{ kN}$$

Moment about z-axis to find force in vertical stay

$$F_{va} := \left(V_H \cdot p \cdot g \cdot \cos(15 \text{ deg}) \cdot 6100 \text{ mm}\right) + \left(V_R \cdot p \cdot g \cdot \cos(15 \text{ deg}) \cdot \frac{l_R}{2}\right) + \left(V_B \cdot p \cdot g \cdot \cos(15 \text{ deg}) \cdot c\right) - F_{buoyancy} \cdot \cos(15 \text{ deg}) \cdot \frac{l_R}{2} + V_{EP} \cdot p \cdot g \cdot \cos(15 \text{ deg}) \cdot l_R$$

$$F_v := \frac{F_{va}}{6100 \text{ mm} \cdot \sin(45 \text{ deg})} = 12.2437 \text{ kN}$$

Reaction Forces resulting:

$$A_{y'} := V_H \cdot p \cdot g \cdot \cos(15 \text{ deg}) + V_R \cdot p \cdot g \cdot \cos(15 \text{ deg}) + 9 \cdot V_B \cdot p \cdot g \cdot \cos(15 \text{ deg}) - F_{buoyancy} \cdot \cos(15 \text{ deg}) + 2 \cdot V_{EP} \cdot p \cdot g \cdot \cos(15 \text{ deg}) - F_v \cdot \sin(45 \text{ deg})$$

$$A_{y'} = -2652.5295 \text{ N}$$

x' and y' -reaction forces at torque tube side

$$A_{x'} := V_H \cdot p \cdot g \cdot \sin(15 \text{ deg}) + V_R \cdot p \cdot g \cdot \sin(15 \text{ deg}) + 9 \cdot V_B \cdot p \cdot g \cdot \sin(15 \text{ deg}) - F_{buoyancy} \cdot \sin(15 \text{ deg}) + 2 \cdot V_{EP} \cdot p \cdot g \cdot \sin(15 \text{ deg}) + F_v \cdot \cos(45 \text{ deg}) - F_s \cdot \sin(45 \text{ deg}) \cdot \left(\frac{2}{3} \cdot l_R\right)$$

$$A_{x'} = 3699.7371 \text{ N}$$

Reaction Forces resulting (converted to turque tube axis system):

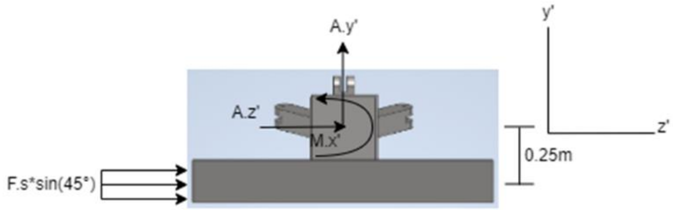
$A_x := \left(A_{x'} \cdot \cos(15 \text{ deg}) - A_{y'} \cdot \sin(15 \text{ deg})\right)$ x and y-reaction forces at torque tube side

$$A_x = 4260.1968 \text{ N}$$

$$A_y := \left(A_{y'} \cdot \cos(15 \text{ deg}) + A_x \cdot \sin(15 \text{ deg})\right)$$

$$A_y = -1459.5267 \text{ N}$$

Calculations for the y-z axis



$$M_{x'} := F_s \cdot \sin(45\ \mathrm{deg}) \cdot \left(\frac{2}{3} \cdot l_R\right) \cdot \left(\frac{h_{Rout}}{2} + \frac{h_B}{2}\right) = 1641.7313\ \mathrm{N\ m}$$

$$A_{z'} := F_s \cdot \cos(45\ \mathrm{deg}) \cdot \frac{2}{3} \cdot l_R = 6566.9252\ \mathrm{N}$$

Reaction froces in x', y' and z' axis (alligned with rake arm):

$A_{x'} = 3699.7371\ \mathrm{N}$

$A_{y'} = -2652.5295\ \mathrm{N}$

$A_{z'} = 6566.9252\ \mathrm{N}$

$M_{x'} = 1641.7313\ \mathrm{N\ m}$

$$M_{y'} := \left(\frac{M_y - M_{x'} \cdot \sin(15\ \mathrm{deg})}{\cos(15\ \mathrm{deg})}\right) = 67111.8702\ \mathrm{N\ m}$$

$M_{z'} = 0\ \mathrm{N\ m}$

These are the reaction forces at the bottom of the torque tube (in axis system same as that of torque tube): ref A4.3

$A_x = 4260.1968\ \mathrm{N}$

$M_x := M_{x'} \cdot \cos(15\ \mathrm{deg}) - M_{y'} \cdot \sin(15\ \mathrm{deg}) = -15784.0395\ \mathrm{N\ m}$

$A_y = -1459.5267\ \mathrm{N}$

$M_y = 65250\ \mathrm{N\ m}$

$A_z := A_{z'} = 6566.9252\ \mathrm{N}$

$M_z := M_{z'} = 0\ \mathrm{N\ m}$

These are the reactiion forces at the turque tube, vertical stay connection point(in axis system same as that of torque tube):

$F_{tx} := (-F_v) \cdot \cos(30\ \mathrm{deg}) = -10603.3595\ \mathrm{N}$

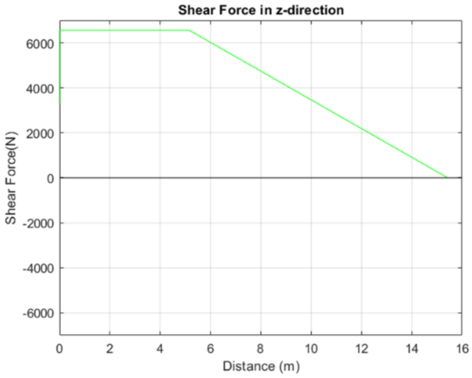
$F_{ty} := F_v \cdot \sin(30\ \mathrm{deg}) + V_v \cdot p \cdot g = 6268.4514\ \mathrm{N}$

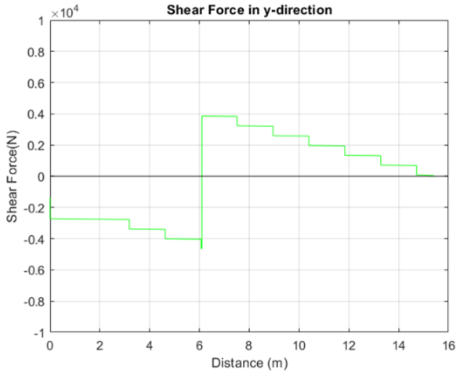
$$F_{vmag} := \sqrt{F_{tx}^{\ 2} + F_{ty}^{\ 2}}$$

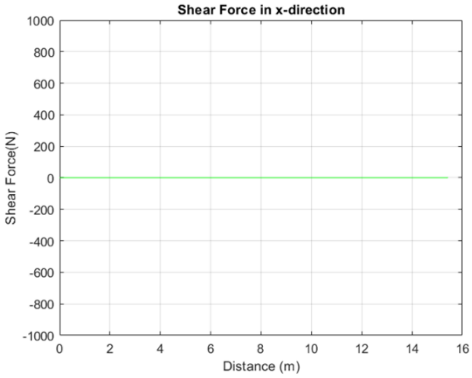
$F_{vmag} = 12317.6587\ \mathrm{N}$

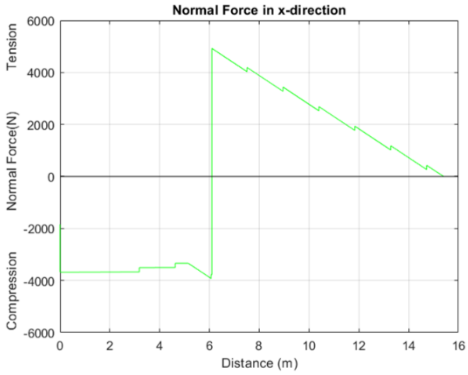
The Shear Force, Normal Force and bending moment graphs have been plotted and analysed using Macaulay functions. AN example of the MATLAB function used is also demonstrated:

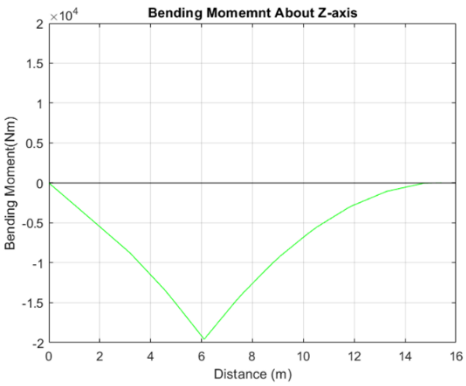
ref A4.5

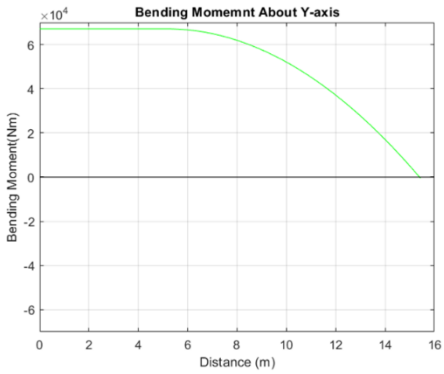


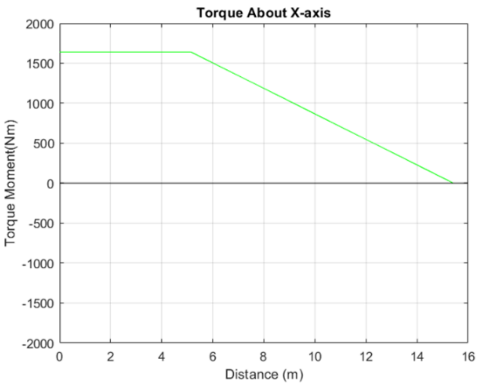


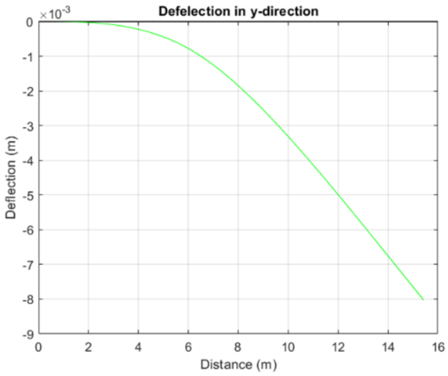


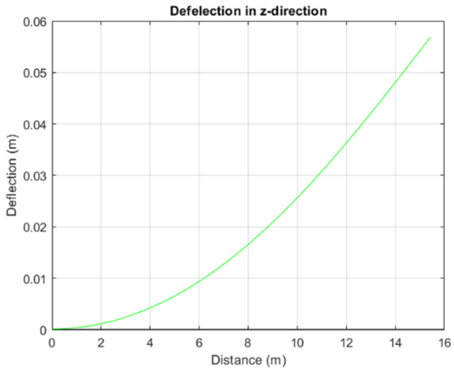












Example MATLAB function and macaulay equation used for graph plotting of shear forces in the y-direction: ref A4.4

The "heaviside" functions enables a specific function only once a specific x-value has been reached. The large number of Vb parts of the euqation are aresult of modelling the blades as individual point loads

```
m= -Ay*(x)^0.*heaviside(x)  -((Vr*p*g*cosd(15)/lr))*x^1.*heaviside(x)  -Vh*p*g*cosd(15)*(x-6.1).^0.*heaviside(x-6.1)
-Vep*p*g*cosd(15)*(x-15.43).^0.*heaviside(x-15.43)  -Vep*p*g*cosd(15)*(x).^0.*heaviside(x)
+(((Fbuoy*cosd(15))/lr))*(x)^1.*heaviside(x)  +Fv*sind(45)*(x-6.1).^0.*heaviside(x-6.1)
-Vb*p*g*cosd(15)*(x-(lr-a)).^0.*heaviside(x-(lr-a))  -Vb*p*g*cosd(15)*(x-(lr-a-b-t)).^0.*heaviside(x-(lr-a-b-t))
-Vb*p*g*cosd(15)*(x-(lr-a-2*b-2*t)).^0.*heaviside(x-(lr-a-2*b-2*t))
-Vb*p*g*cosd(15)*(x-(lr-a-3*b-3*t)).^0.*heaviside(x-(lr-a-3*b-3*t))
-Vb*p*g*cosd(15)*(x-(lr-a-4*b-4*t)).^0.*heaviside(x-(lr-a-4*b-4*t))
-Vb*p*g*cosd(15)*(x-(lr-a-5*b-5*t)).^0.*heaviside(x-(lr-a-5*b-5*t))
-Vb*p*g*cosd(15)*(x-(lr-a-6*b-6*t)).^0.*heaviside(x-(lr-a-6*b-6*t))
-Vb*p*g*cosd(15)*(x-(lr-a-7*b-7*t)).^0.*heaviside(x-(lr-a-7*b-7*t))
-Vb*p*g*cosd(15)*(x-(lr-a-8*b-8*t)).^0.*heaviside(x-(lr-a-8*b-8*t));
fplot(m,[0,lr],'g')
hold on

plot(linspace(0,16),(0*linspace(0,16)),'k')
axis([0 16 -10000 10000])
grid on
title('Shear Force in y-direction')
xlabel('Distance (m)')
ylabel('Shear Force(N)')
hold off;
```

This function could simply be integrated using the following function, to obtain bending moment, slope and deflection equations and graphs:

$\boxed{n := int\ (m)}$

Due to the complexity and length of these functions, they were not all included in the report. However, they all have a very similar structure.

From these diagrams it can be deducted that the critical point, undergoing mayimum combined normal and shear stress, will be at 6.1 m. Therefore, the following values have been deducted from the graphs:

$SF_{x6.1} := 0\ N$

$SF_{y6.1} := 4636\ N$

$SF_{z6.1} := 5957\ N$

$NF_{x6.1} := 4924\ N$ Tension

$BM_{x6.1} := 1490\ N\ m$

$BM_{y6.1} := 66530\ N\ m$

$BM_{z6.1} := 19571\ N\ m$

Maximum flection in y'--direction: 8mm
Maximum flection in z'-direction: 57mm

<u>When no sludge is inside the tank, and the thickener is not running (steady):</u>

For this case, the buoyancy force and all forces caused by the dynamics(movement) of the system have been removed from the calculations.

$$F_{va1} := \left(V_H \cdot p \cdot g \cdot \cos(15\ deg) \cdot 6100\ mm\right) + \left(V_R \cdot p \cdot g \cdot \cos(15\ deg) \cdot \frac{l_R}{2}\right) + \\ + \left(V_B \cdot p \cdot g \cdot \cos(15\ deg) \cdot c\right) + V_{EP} \cdot p \cdot g \cdot \cos(15\ deg) \cdot l_R$$

$$F_{v1} := \frac{F_{va1}}{6100\ mm \cdot \sin(45\ deg)} = 36.0818\ kN$$

$$A_{y1'} := V_H \cdot p \cdot g \cdot \cos(15\ deg) + V_R \cdot p \cdot g \cdot \cos(15\ deg) + 9 \cdot V_B \cdot p \cdot g \cdot \cos(15\ deg) + \\ + 2 \cdot V_{EP} \cdot p \cdot g \cdot \cos(15\ deg) - F_{v1} \cdot \sin(45\ deg)$$

$$A_{y1'} = -6181.0551\ N$$

$$A_{x1'} := V_H \cdot p \cdot g \cdot \sin(15\ deg) + V_R \cdot p \cdot g \cdot \sin(15\ deg) + 9 \cdot V_B \cdot p \cdot g \cdot \sin(15\ deg) + \\ + 2 \cdot V_{EP} \cdot p \cdot g \cdot \sin(15\ deg) + F_{v1} \cdot \cos(45\ deg) - F_s \cdot \sin(45\ deg) \cdot \left(\frac{2}{3} \cdot l_R\right)$$

$$A_{x1'} = 24126.9289\ N$$

<u>These are the reaction forces at the bottom of the torque tube, when the thickener is empty and not running</u>

$M_{y1} := 0\ N\ m$

$M_{z1} := 0\ N\ m$

$M_{x1} := 0\ N\ m$

$A_{y1} := A_{y1'} \cdot \sin(15\ deg) + A_{x1'} \cdot \cos(15\ deg) = 21705.049\ N$

$A_{x1} := A_{x1'} \cdot \cos(15\ deg) - A_{x1'} \cdot \sin(15\ deg) = 17060.3151\ N$

$A_{z1} := 0\ N$

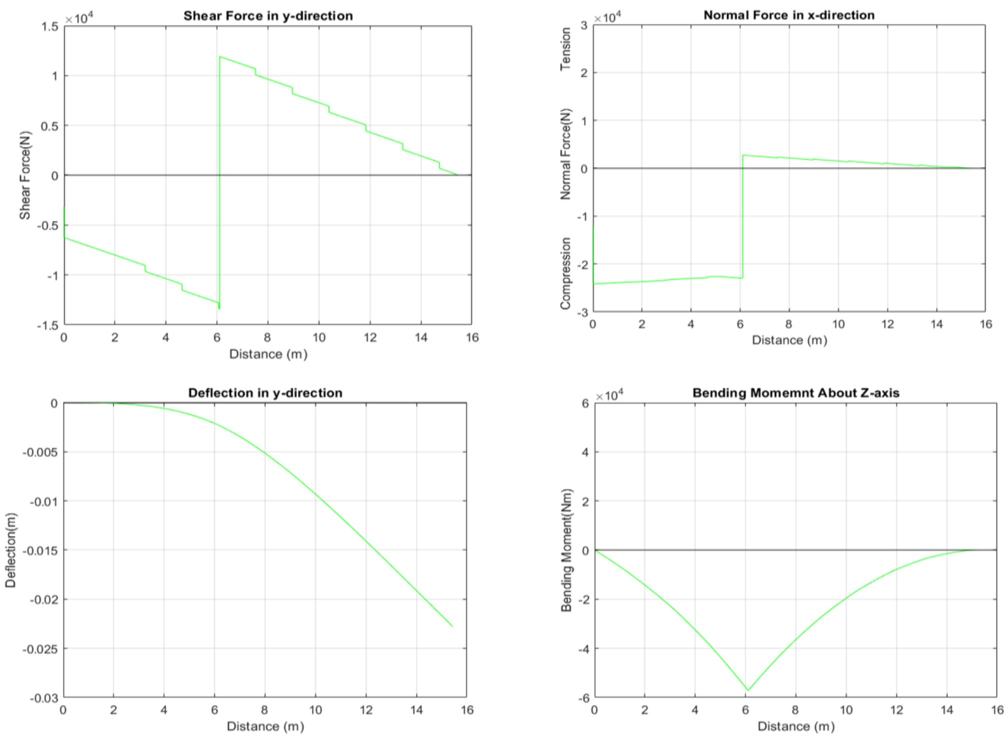
<u>These are the reactiion forces at the turque tube, vertical stay connection point, when the thickener is empty and not running</u>

$$F_{tx1} := \left(-F_{v1}\right) \cdot \cos(30\ deg) = -31247.7612\ N$$

$$F_{ty1} := F_{v1} \cdot \sin(30\ deg) + V_V \cdot p \cdot g = 18187.5023\ N$$

$$F_{vmag1} := \sqrt{F_{tx1}^{\ 2} + F_{ty1}^{\ 2}}$$

$$F_{vmag1} = 36155.3291\ N$$



It can be seen that if the tank is empty, it results in much larger maximum shear forces in the y-direction as well as Normal forces in the x-direction. The critical point is however still identified to be at 6.1m, therefore the following data was deducted from the graphs:

$SF_{y6.1empty} := 13433\ \mathrm{N}$

Maximum deflection in y'-direction is 23mm downwards

$NF_{x6.1empty} := 24109\ \mathrm{N}$ compression

$BM_{z6.1empty} := 57165\ \mathrm{N\ m}$

Because of these two conditions(large forces in empty tank and overload condition) both have to be analysed for failure. The Tresca (maximum-Shear_stress) theory has been decided to be used, since this is a bit more conservative than the Von Mises (Distortion-Energy) theory and therefore will result in lower safety factors, ensuring a more durable system.

Cross section of rake arm:

$l_R := 15.43\ \mathrm{m}$

$b_{Rout} := 0.3\ \mathrm{m}$

$h_{Rout} := 0.3\ \mathrm{m}$

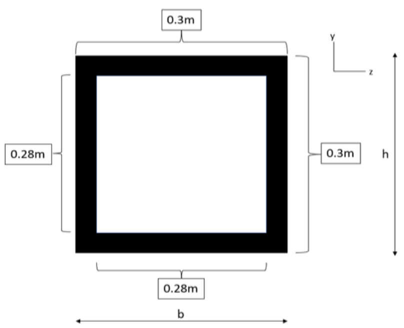
$t_R := 0.01\ \mathrm{m}$

$b_{Rin} := b_{Rout} - (2 \cdot t_R)$

$b_{Rin} = 0.28\ \mathrm{m}$

$h_{Rin} := h_{Rout} - (2 \cdot t_R)$

$h_{Rin} = 0.28\ \mathrm{m}$



$$I_{yy} := \frac{b_{Rout} \cdot h_{Rout}^{\ 3}}{12} - \frac{b_{Rin} \cdot h_{Rin}^{\ 3}}{12} = 0.0002\ \mathrm{m}^4$$

$$I_{zz} := I_{yy} = 0.0002\ \mathrm{m}^4$$

$$Area_x := \left(b_{Rout} \cdot h_{Rout}\right) - \left(b_{Rin} - h_{Rin}\right) = 0.09\ \mathrm{m}^2$$

$$ybar := \frac{b_{Rout}}{2} = 0.15\ \mathrm{m}$$

$$zbar := ybar = 0.15\ \mathrm{m}$$

Plane Stresses:

Note: Overload condition on left of page and empty tank condition on right of page, for easy, direct comparison of stresses.

ref A4.6

Overload Condition:

Normal Stresses due to axial Force:

$$\sigma_{xnormal\_overload} := \frac{NF_{x6.1}}{Area_x} = 54711.1111\ \mathrm{Pa}$$

Tension

Bending Stresses:

$$\sigma_{xbending} := \frac{BM_{z6.1} \cdot ybar}{I_{zz}} + \frac{BM_{y6.1} \cdot zbar}{I_{yy}} = 7.9338 \cdot 10^7\ \mathrm{Pa}$$

$$\sigma_{ybending} := 0\ \mathrm{Pa}$$

$$\sigma_{zbending} := 0\ \mathrm{Pa}$$

Summed Normal Stresses:

$$\sigma_{x1} := \sigma_{xbending} + \sigma_{xnormal\_overload} = 7.9393 \cdot 10^7\ \mathrm{Pa}$$

$$\sigma_{y1} := 0\ \mathrm{Pa}$$

$$\sigma_{z1} := 0\ \mathrm{Pa}$$

Shear Stresses:

$$A_m := (0.29\ \mathrm{m})^2 = 0.0841\ \mathrm{m}^2$$ Area enclosed by mean perimeter

Shear stresses caused by shear forces and Torque about x'-axis:

$$\tau_{xy1} := \frac{3}{2} \cdot \frac{SF_{y6.1}}{Area_x} + \frac{BM_{x6.1}}{2 \cdot A_m \cdot t_R} = 9.6312 \cdot 10^5\ \mathrm{Pa}$$

$$\tau_{xz1} := \frac{3}{2} \cdot \frac{SF_{z6.1}}{Area_x} + \frac{BM_{x6.1}}{2 \cdot A_m \cdot t_R} = 9.8513 \cdot 10^5\ \mathrm{Pa}$$

$$\tau_{yz1} := 0\ \mathrm{Pa}$$

Empty tank:

$$\sigma_{xnormal\_empty} := \frac{NF_{x6.1empty}}{Area_x} = 2.6788 \cdot 10^5\ \mathrm{Pa}$$

Compression

$$\sigma_{xbending\_empty} := \frac{BM_{z6.1empty} \cdot ybar}{I_{zz}} = 5.2675 \cdot 10^7\ \mathrm{Pa}$$

$$\sigma_{x2} := \sigma_{xnormal\_empty} + \sigma_{xbending\_empty} = 5.2943 \cdot 10^7\ \mathrm{Pa}$$

$$\sigma_{y2} := 0\ \mathrm{Pa}$$

$$\sigma_{z2} := 0\ \mathrm{Pa}$$

$$\tau_{xy2} := \frac{3}{2} \cdot \frac{SF_{y6.1empty}}{Area_x} = 2.2388 \cdot 10^5\ \mathrm{Pa}$$

$$\tau_{xz2} := 0\ \mathrm{Pa}$$

$$\tau_{yz2} := 0\ \mathrm{Pa}$$

From this analysis it can be obsrved that for the overload condition, all plane strains are higher than that of the empty tank condition. Therefore, the critical condition is the overload situation and thus the one that will be used for failure and safety factor calculations.

Critical condition Plane stresses:

$\sigma_{x1} = 7.9393 \cdot 10^{7}\ \mathrm{Pa}$

$\tau_{xy1} = 9.6312 \cdot 10^{5}\ \mathrm{Pa}$

$\sigma_{y1} := 0\ \mathrm{Pa}$

$\tau_{xz1} = 9.8513 \cdot 10^{5}\ \mathrm{Pa}$

$\sigma_{z1} := 0\ \mathrm{Pa}$

$\tau_{yz1} := 0\ \mathrm{Pa}$

Finding Principal stresses:

ref A4.7

$aa := 1$

$bb := -\left(\sigma_{x1} + \sigma_{y1} + \sigma_{z1}\right) = -7.9393 \cdot 10^{7}\ \mathrm{Pa}$

$cc := \left[\left(\sigma_{x1} \cdot \sigma_{y1}\right) + \left(\sigma_{x1} \cdot \sigma_{z1}\right) + \left(\sigma_{y1} \cdot \sigma_{z1}\right) - \left(\tau_{xy1}\right)^2 - \left(\tau_{yz1}\right)^2 - \left(\tau_{xz1}\right)^2\right] = -1.8981 \cdot 10^{12}\ \mathrm{Pa}^2$

$dd := -\left[\left(\sigma_{x1} \cdot \sigma_{y1} \cdot \sigma_{z1}\right) + \left(2 \cdot \tau_{xy1} \cdot \tau_{yz1} \cdot \tau_{xz1}\right) - \left(\sigma_{x1} \cdot \tau_{yz1}^{\ 2}\right) - \left(\sigma_{y1} \cdot \tau_{xz1}^{\ 2}\right) - \left(\sigma_{z1} \cdot \tau_{xy1}^{\ 2}\right)\right] = 0$

$$\sigma_{123} := \mathrm{polyroots}\left(\begin{bmatrix} dd \\ cc \\ bb \\ aa \end{bmatrix}\right) = \begin{bmatrix} 4.1 \cdot 10^{-14} \\ -23900.3479 \\ 7.9416 \cdot 10^{7} \end{bmatrix}$$

Finding the Principal Stresses using Shicley's equation (3 − 15) and solving for its roots using polyroots function

$\sigma_1 := \mathrm{row}\left(\sigma_{123}, 3\right)\ \mathrm{Pa} = \left[7.9416 \cdot 10^{7}\right]\ \mathrm{Pa}$

$\sigma_2 := \mathrm{row}\left(\sigma_{123}, 2\right)\ \mathrm{Pa} = \left[-23900.3479\right]\ \mathrm{Pa}$

$\sigma_3 := \mathrm{row}\left(\sigma_{123}, 1\right)\ \mathrm{Pa} = \left[4.1 \cdot 10^{-14}\right]\ \mathrm{Pa}$

By Maximum-Shear-Stress (Tresca) theory:

ref A4.8

$n_{MSS} := \dfrac{\sigma_{yield}}{\sigma_1 - \sigma_3} = \left[3.6516\right]$

By Distorsion-Energy (Von Mises) theory:

$$\sigma_{DE} := \left(\frac{\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2}{2}\right)^{\frac{1}{2}}$$

$n_{DE} := \dfrac{\sigma_{yield}}{\sigma_{DE}} = 3.6511$

Stays:

Safety factor using tresca yield criterion:

ref A4.9

$$\sigma_S := \frac{F_{vmag1}}{\pi \cdot \left(r_{Vout}^{\ 2} - r_{Vin}^{\ 2}\right)}$$

$\sigma_S = 125.0935\ \mathrm{MPa}$

$n_2 := \dfrac{\sigma_{maxtensile}}{\sigma_S}$

$n_2 = 4.6365$

Absolute maximum deflection of rake arm in the y'direction

ref A4.12

Due to the notional load experienced by the torque tube experienced, a slope of 0.00156251 radians was discvered at the end of the torque tube (frum torque tube calculations)

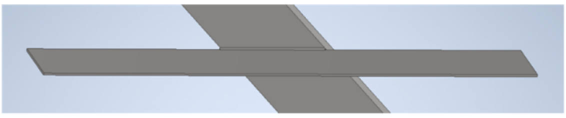
maximum defllection of the rake arm in the y'direction is 8.03mm from overload deflection graph

The total deflection at the end of the rake arm was thus found to be:

$$defl := \left(l_R \cdot \sin(15\ \mathrm{deg})\right) - l_R \cdot \sin\left(15\ \mathrm{deg} - \left(0.00156251 \cdot \frac{360}{\pi}\ \mathrm{deg}\right)\right) = 46.5955\ \mathrm{mm}$$

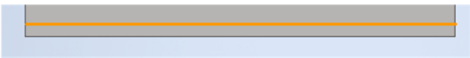
Weld Thicknesses:

ref A4.13



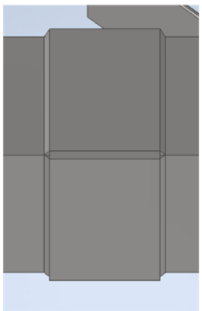
Weld thickness is 6mm

For the welds between the 20mm thick blades and the 10mm thick square tubing, welds of thickness 6mm will be used, as per table $(9-6)$ in Shigley's



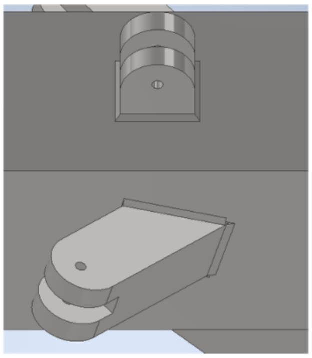
Weld thickness is 5mm

For the welds between the 0mm thick endplates and the 10mm thick square tubing, welds of thickness 6mm will be used, as per table $(9-6)$ in Shigley's



Weld thickness is 5mm

For the welds surrounding and connecting the sleeve at its two ends, welds of thickness 5mm should be used as per Shigley's table $(9-6)$



Weld thickness is 12mm

Since the clevis castings all have the same base dimensions, similar welds will be used. Since the clevis castings are so wide, a large weld willl have so be used, corresponding to a thickness of 12 mm as per Shigley's table $(9-6)$

RAKE ARM CALCULATIONS(Normal Working Condition):

Torqu Calculations

$T_{max} := K \cdot D^2$ Max Torque

$K := 290\ \frac{N}{m}$

$D := 30\ m$

$T_{max} = 2.61 \cdot 10^5\ N\ m$ Total Maximum torque

End Plates

$V_{EP} := 0.3\ m \cdot 0.3\ m \cdot 0.01\ m$

$V_{EP} = 0.0009\ m^3$

Blade Dimensions

$l_B := 2\ m$

$h_B := 0.2\ m$

$t_B := 0.02\ m$

$V_B := l_B \cdot h_B \cdot t_B$

$V_B = 0.008\ m^3$ Volume of Blades, used for mass and buoyancy calculations

Horizontal stay dimensions

$l_H := 7.751\ m$

$r_{Hout} := \frac{0.048}{2}\ m$

$r_{Hin} := 0.022\ m$

$V_H := l_H \cdot \left(\pi \cdot \left(r_{Hout}^2 - r_{Hin}^2\right)\right)$

$V_H = 0.0022\ m^3$

$V_{Hfloat} := \pi \cdot r_{Hout}^2 \cdot l_H$

$V_{Hfloat} = 0.014\ m^3$ Vertical Stay is fully submerged and therefore also contributes to floatation

Rakes Dimensions:

$l_R := 15.43\ m$ length of rake

$b_{Rout} := 0.3\ m$ bredth rake

$h_{Rout} := 0.3\ m$ height of rake

$t_R := 0.01\ m$ thickness of rake

$b_{Rin} := b_{Rout} - \left(2 \cdot t_R\right)$

$b_{Rin} = 0.28\ m$

$h_{Rin} := h_{Rout} - \left(2 \cdot t_R\right)$

$h_{Rin} = 0.28\ m$

$V_R := l_R \cdot b_{Rout} \cdot h_{Rout} - l_R \cdot b_{Rin} \cdot h_{Rin}$

$V_R = 0.179\ m^3$ Volume of Rake tube tubing, used for mass calculation

$V_{Rfloat} := l_R \cdot b_{Rout} \cdot h_{Rout}$

$V_{Rfloat} = 1.3887\ m^3$ Volume of total rake tube, for buoyancy calculation

Vertical stay Dimensions

$l_V := 6.463\ m$

$r_{Vout} := \frac{0.048}{2}\ m$

$r_{Vin} := 0.022\ m$

$V_V := l_V \cdot \left(\pi \cdot \left(r_{Vout}^2 - r_{Vin}^2\right)\right)$

$V_V = 0.0019\ m^3$ Vertical Stay is barely submerged, and is therefore not included in buoyancy force

Constants

$E := 193\ GPa$ Elastic dulus of material

$\sigma_{yield} := 290\ MPa$ Yield strength of material

$S_{ut} := 580\ MPa$ Ultimate ength of material

$p := 8000\ \frac{kg}{m^3}$ Properties of AISI 316

$g := 9.81\ \frac{m}{s^2}$ Gravitational force

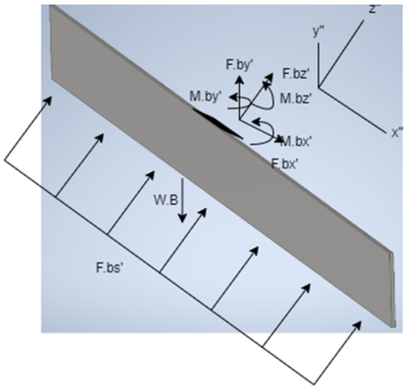
$V_{buoyancy} := V_{Rfloat} + V_B + V_{Hfloat}$

$V_{buoyancy} = 1.4107\ m^3$ Total volume causing buoyancy foorce

$p_{water} := 997\ \frac{kg}{m^3}$ Water density, since sludge density is not known. Water will have a lower density, resulting in lower buoyancy force and therefore overstated the forces caused in the rake arm, and may therefore be used for the calculations.

$v := 0.27$

BLADES:



$$T_t := \frac{T_{max}}{4} \cdot 0.6$$

$T_t = 39150\ \mathrm{N\ m}$ Maximum torque per rake arm

$$w_{bz} := \frac{T_t}{l_R \cdot \frac{l_R}{2}}$$

$w_{bz} = 328.8743\ \frac{\mathrm{N}}{\mathrm{m}}$ Distributed load of sludge on complete rake arm

$$F_{bz} := w_{bz} \cdot l_R \cdot \frac{1}{9}$$

$F_{bz} = 563.8367\ \mathrm{N}$

$$F_{bx} := \frac{F_{bz}}{\cos(45\ \mathrm{deg})} \cdot \sin(45\ \mathrm{deg})$$

$F_{bx} = 563.8367\ \mathrm{N}$

$$F_{by} := p \cdot g \cdot V_B$$

$F_{by} = 627.84\ \mathrm{N}$

$$F_{by'} := F_{by} \cdot \cos(15\ \mathrm{deg})$$

Converting the forces to forces in a new coordinate system alligned with the blade:

$$F_{bx'} := \left(F_{bz} \cdot \sin(45\ \mathrm{deg}) - F_{bx} \cdot \cos(45\ \mathrm{deg})\right)$$

$$F_{bz'} := \left(F_{bz} \cdot \cos(45\ \mathrm{deg}) + F_{bx} \cdot \sin(45\ \mathrm{deg})\right) - p \cdot g \cdot V_B \cdot \sin(15\ \mathrm{deg})$$

The reaction forces at the top of each blade, in the blades coordinate system are:

$F_{bz'} = 634.8885\ \mathrm{N}$ $F_{bx'} = 0\ \mathrm{N}$ $F_{by'} = 606.4469\ \mathrm{N}$

$$M_{bx'} := F_{bz'} \cdot \frac{h_B}{2}$$

$M_{bz'} := 0\ \mathrm{N\ m}$ $M_{bx'} = 63.4889\ \mathrm{N\ m}$ $M_{by'} := 0\ \mathrm{N\ m}$

Finding the maximum bending moments and shear forces:

x'-z' plane:

$$w_{z'1} := \frac{F_{bz'}}{l_B}$$

$w_{z'1} = 317.4443\ \frac{\mathrm{N}}{\mathrm{m}}$

$SF_{z'1}(x') := w_{z'1} \cdot x'$

$SF_{z'1}(1\ \mathrm{m}) = 317.4443\ \mathrm{N}$

$$BM_{y'}(x') := \frac{w_{z'1} \cdot x'^2}{2}$$

$BM_{y'}(1\ \mathrm{m}) = 158.7221\ \mathrm{N\ m}$ $SF_{x'} := 0\ \mathrm{N}$

y'-z' plane:

$$w_{z'2} := \frac{F_{bz'}}{h_B}$$

$w_{z'2} = 3174.4427\ \frac{\mathrm{N}}{\mathrm{m}}$

$SF_{z'2}(y') := w_{z'2} \cdot y'$

$SF_{z'2}(0.2\ \mathrm{m}) = 634.8885\ \mathrm{N}$

$$BM_{x'}(y') := \frac{w_{z'2} \cdot y'^2}{2}$$

$BM_{x'}(0.2\ \mathrm{m}) = 63.4889\ \mathrm{N\ m}$

$$SF_{z'max} := \begin{bmatrix} \left(SF_{z'2}(0.2\ \mathrm{m})\right) \\ SF_{z'1}(1\ \mathrm{m}) \end{bmatrix}$$

$SF_{z'} := \max\left(SF_{z'max}\right)$

$SF_{z'} = 634.8885\ \mathrm{N}$

x'-y' plane:

$$w_{y'} := \frac{F_{by'}}{l_B}$$

$$w_{y'} = 303.2234\ \frac{\mathrm{N}}{\mathrm{m}}$$

$$SF_{y'}(x') := w_{y'} \cdot x' \qquad BM_{z'}(x') := \frac{w_{y'} \cdot x'^2}{2}$$

$$SF_{y'}(1\ \mathrm{m}) = 303.2234\ \mathrm{N} \qquad BM_{z'}(1\ \mathrm{m}) = 151.6117\ \mathrm{N\ m}$$

Calculating the Normal Stresses and Shear Stresses

$$I_{y'y'} := \frac{h_B \cdot t_B^3}{12} \qquad I_{x'x'} := \frac{l_B \cdot t_B^3}{12} \qquad I_{z'z'} := \frac{t_B \cdot h_B^3}{12}$$

$$z'bar := \frac{t_B}{2} \qquad y'bar := \frac{h_B}{2}$$

$$\sigma_{z'} := 0 \qquad \sigma_{y'} := \frac{F_{by'}}{l_B \cdot t_B} + \frac{BM_{x'}(0.2\ \mathrm{m}) \cdot z'bar}{I_{x'x'}} \qquad \sigma_{x'} := \frac{BM_{y'}(1\ \mathrm{m}) \cdot z'bar}{I_{y'y'}} + \frac{BM_{z'}(1\ \mathrm{m}) \cdot y'bar}{I_{z'z'}}$$

$$\sigma_{z'} = 0\ \mathrm{MPa} \qquad \sigma_{y'} = 0.4913\ \mathrm{MPa} \qquad \sigma_{x'} = 13.0412\ \mathrm{MPa}$$

Compression or Tension — Tension — Tension or Compression

$$Q_{y'z'} := \frac{t_B}{2} \cdot l_B \cdot \frac{t_B}{4} \qquad Q_{x'y'} := t_B \cdot \frac{h_B}{2} \cdot \frac{h_B}{4} \qquad Q_{x'z'} := \frac{t_B}{2} \cdot h_B \cdot \frac{t_B}{4}$$

$$\tau_{y'z'} := \frac{SF_{z'2}(0.2\ \mathrm{m}) \cdot Q_{y'z'}}{I_{x'x'} \cdot l_B} \qquad \tau_{x'y'} := \frac{SF_{y'}(1\ \mathrm{m}) \cdot Q_{x'y'}}{I_{z'z'} \cdot t_B} \qquad \tau_{x'z'} := \frac{SF_{z'1}(1\ \mathrm{m}) \cdot Q_{x'z'}}{I_{y'y'} \cdot h_B}$$

$$\tau_{y'z'} = 0.0238\ \mathrm{MPa} \qquad \tau_{x'y'} = 0.1137\ \mathrm{MPa} \qquad \tau_{x'z'} = 0.119\ \mathrm{MPa}$$

Maximum stresses in the blades:

Plane Stresses

$$\sigma_{x'} = 1.3041 \cdot 10^7\ \mathrm{Pa} \qquad \tau_{x'z'} = 1.1904 \cdot 10^5\ \mathrm{Pa}$$

$$\sigma_{y'} = 4.9133 \cdot 10^5\ \mathrm{Pa} \qquad \tau_{x'y'} = 1.1371 \cdot 10^5\ \mathrm{Pa}$$

$$\sigma_{z'} = 0\ \mathrm{Pa} \qquad \tau_{y'z'} = 23808.32\ \mathrm{Pa}$$

Finding the 3 Principle stresses for the blades:

$$aa := 1$$

$$bb := -\left(\sigma_{x'} + \sigma_{y'} + \sigma_{z'}\right) = -1.3533 \cdot 10^7\ \mathrm{Pa}$$

$$cc := \left[\left(\sigma_{x'} \cdot \sigma_{y'}\right) + \left(\sigma_{x'} \cdot \sigma_{z'}\right) + \left(\sigma_{y'} \cdot \sigma_{z'}\right) - \left(\tau_{x'y'}\right)^2 - \left(\tau_{y'z'}\right)^2 - \left(\tau_{x'z'}\right)^2\right] = 6.3799 \cdot 10^{12}\ \mathrm{Pa}^2$$

$$da := \left(\sigma_{x'} \cdot \sigma_{y'} \cdot \sigma_{z'}\right) + \left(2 \cdot \tau_{x'y'} \cdot \tau_{y'z'} \cdot \tau_{x'z'}\right) = 6.4454 \cdot 10^{14}\ \mathrm{Pa}^3$$

$$dd := -\left(da - \left(\sigma_{x'} \cdot \tau_{y'z'}^2\right) - \left(\sigma_{y'} \cdot \tau_{x'z'}^2\right) - \left(\sigma_{z'} \cdot \tau_{x'y'}^2\right)\right)$$

$$dd = 1.371 \cdot 10^{16}\ \mathrm{Pa}^3$$

Finding the Principal Stresses using Shigley's equation $(3-15)$ and solving for its roots using polyroots function

$$\sigma_{1'2'3'} := \mathrm{polyroots}\left(\begin{bmatrix} dd \\ cc \\ bb \\ aa \end{bmatrix}\right) = \begin{bmatrix} -2139.2831 \\ 4.9135 \cdot 10^5 \\ 1.3043 \cdot 10^7 \end{bmatrix}$$

$$\sigma_{1'} := \mathrm{row}\left(\sigma_{1'2'3'}, 3\right)\ \mathrm{Pa} = \left[1.3043 \cdot 10^7\right]\ \mathrm{Pa}$$

$$\sigma_{2'} := \mathrm{row}\left(\sigma_{1'2'3'}, 2\right)\ \mathrm{Pa} = \left[4.9135 \cdot 10^5\right]\ \mathrm{Pa}$$

$$\sigma_{3'} := \mathrm{row}\left(\sigma_{1'2'3'}, 1\right)\ \mathrm{Pa} = \left[-2139.2831\right]\ \mathrm{Pa}$$

By Maximum-Shear-Stress theory:

$$n_{MSS'} := \frac{\sigma_{yield}}{\sigma_{1'} - \sigma_{3'}} = [\,22.2299\,]$$

By Distorsion-Energy theory:

$$\sigma_{DE'} := \left(\frac{\left(\sigma_{1'} - \sigma_{2'}\right)^2 + \left(\sigma_{2'} - \sigma_{3'}\right)^2 + \left(\sigma_{3'} - \sigma_{1'}\right)^2}{2}\right)^{\frac{1}{2}}$$

$$n_{DE'} := \frac{\sigma_{yield}}{\sigma_{DE'}} = 22.6458$$

Rake Arm:

This section used the same FBDs as the ones used for the overload condition, with slight alterations. Since the Final values did not determine the safety factor, the specific FBDs were not included

$$a := \frac{2\ \mathrm{m} \cdot \cos(45\ \mathrm{deg})}{2} \qquad b := 2\ \mathrm{m} \cdot \cos(45\ \mathrm{deg}) \qquad t := \frac{t_B}{\cos(45\ \mathrm{deg})}$$

varibles for placement of blade point loads along x'-axis of rake arm

$$c := \left(l_R - a\right) + \left(l_R - a - (b+t)\right) + \left(l_R - a - 2\cdot(b+t)\right) + \left(l_R - a - 3\cdot(b+t)\right) + \left(l_R - a - 4\cdot(b+t)\right) + \\ + \left(l_R - a - 5\cdot(b+t)\right) + \left(l_R - a - 6\cdot(b+t)\right) + \left(l_R - a - 7\cdot(b+t)\right) + \left(l_R - a - 8\cdot(b+t)\right)$$

$$c = 80.5761\ \mathrm{m}$$

Calculations for the x-z axis

$$M_y := T_t = 39150\ \mathrm{N\ m}$$

$$F_s := \frac{M_y}{\left(\frac{1}{2}\cdot l_R \cdot \sin(75\ \mathrm{deg})\right)\cdot l_R \cdot \cos(45\ \mathrm{deg})} = 481.5054\ \frac{\mathrm{N}}{\mathrm{m}}$$

Sludge force actingt in the x',z' direction

Calculations for the x-y axis

$M_z := 0\ \mathrm{N\ m}$ Since a pin support is used

$F_{buoyancy} := \rho_{water} \cdot g \cdot V_{buoyancy}$ Buoyancy force induced by submerging the rake

$$F_{buoyancy} = 13.7977\ \mathrm{kN}$$

Moment about z-axis to find force in vertical stay

$$F_{va} := \left(V_H \cdot \rho \cdot g \cdot \cos(15\ \mathrm{deg}) \cdot 6100\ \mathrm{mm}\right) + \left(V_R \cdot \rho \cdot g \cdot \cos(15\ \mathrm{deg}) \cdot \frac{l_R}{2}\right) + \\ + \left(V_B \cdot \rho \cdot g \cdot \cos(15\ \mathrm{deg}) \cdot c\right) - F_{buoyancy} \cdot \cos(15\ \mathrm{deg}) \cdot \frac{l_R}{2} + V_{EP} \cdot \rho \cdot g \cdot \cos(15\ \mathrm{deg}) \cdot l_R$$

$$F_v := \frac{F_{va}}{6100\ \mathrm{mm} \cdot \sin(45\ \mathrm{deg})} = 12.2437\ \mathrm{kN}$$

Reaction Forces resulting:

$$A_{y'} := V_H \cdot \rho \cdot g \cdot \cos(15\ \mathrm{deg}) + V_R \cdot \rho \cdot g \cdot \cos(15\ \mathrm{deg}) + 9 \cdot V_B \cdot \rho \cdot g \cdot \cos(15\ \mathrm{deg}) - F_{buoyancy} \cdot \cos(15\ \mathrm{deg}) + \\ + 2 \cdot V_{EP} \cdot \rho \cdot g \cdot \cos(15\ \mathrm{deg}) - F_v \cdot \sin(45\ \mathrm{deg})$$

x' and y'-reaction forces at torque tube side

$$A_{y'} = -2652.5295\ \mathrm{N}$$

$$A_{x'} := V_H \cdot \rho \cdot g \cdot \sin(15\ \mathrm{deg}) + V_R \cdot \rho \cdot g \cdot \sin(15\ \mathrm{deg}) + 9 \cdot V_B \cdot \rho \cdot g \cdot \sin(15\ \mathrm{deg}) - F_{buoyancy} \cdot \sin(15\ \mathrm{deg}) + \\ + 2 \cdot V_{EP} \cdot \rho \cdot g \cdot \sin(15\ \mathrm{deg}) + F_v \cdot \cos(45\ \mathrm{deg}) - F_s \cdot \sin(45\ \mathrm{deg}) \cdot l_R$$

$$A_{x'} = 5013.1222\ \mathrm{N}$$

Reaction Forces resulting (converted to turque tube axis system):

x and y-reaction forces at torque tube side

$$A_x := \left(A_{x'} \cdot \cos(15\ \mathrm{deg}) - A_{y'} \cdot \sin(15\ \mathrm{deg})\right)$$
$$A_x = 5528.8293\ \mathrm{N}$$

$$A_y := \left(A_{y'} \cdot \cos(15\ \mathrm{deg}) + A_x \cdot \sin(15\ \mathrm{deg})\right)$$
$$A_y = -1131.1804\ \mathrm{N}$$

Calculations for the y-z axis

$$M_{x'} := F_s \cdot \sin(45\ \mathrm{deg}) \cdot l_R \cdot \left(\frac{h_{Rout}}{2} + \frac{h_B}{2}\right) = 1313.385\ \mathrm{N\ m}$$
$$A_{z'} := F_s \cdot \cos(45\ \mathrm{deg}) \cdot l_R = 5253.5402\ \mathrm{N}$$

Reaction froces in x', y' and z' axis (alligned with rake arm):

$$A_{x'} = 5013.1222\ \mathrm{N}$$
$$A_{y'} = -2652.5295\ \mathrm{N}$$
$$A_{z'} = 5253.5402\ \mathrm{N}$$
$$M_{x'} = 1313.385\ \mathrm{N\ m}$$
$$M_{y'} := \left(\frac{M_y - M_{x'} \cdot \sin(15\ \mathrm{deg})}{\cos(15\ \mathrm{deg})}\right) = 40179.142\ \mathrm{N\ m}$$
$$M_{z'} := 0\ \mathrm{N\ m}$$

These are the reaction forces at the bottom of the torque tube (in axis system same as that of torque tube): ref A4.3

$$A_x = 5528.8293\ \mathrm{N} \qquad M_x := M_{x'} \cdot \cos(15\ \mathrm{deg}) - M_{y'} \cdot \sin(15\ \mathrm{deg}) = -9130.4946\ \mathrm{N\ m}$$
$$A_y = -1131.1804\ \mathrm{N} \qquad M_y = 39150\ \mathrm{N\ m}$$
$$A_z := A_{z'} = 5253.5402\ \mathrm{N} \qquad M_z := M_{z'} = 0\ \mathrm{N\ m}$$

These are the reactiion forces at the turque tube, vertical stay connection point

$$F_{tx} := (-F_v) \cdot \cos(30\ \mathrm{deg}) = -10603.3595\ \mathrm{N}$$
$$F_{ty} := F_v \cdot \sin(30\ \mathrm{deg}) + V_V \cdot p \cdot g = 6268.4514\ \mathrm{N}$$
$$F_{vmag} := \sqrt{F_{tx}^{\ 2} + F_{ty}^{\ 2}}$$
$$F_{vmag} = 12317.6587\ \mathrm{N}$$

Fatigue Calculations:

ref A4.10

$S_{e'} := \frac{S_{ut}}{2} = 2.9 \cdot 10^{8}\ \mathrm{Pa}$

$S_{ut} = 5.8 \cdot 10^{8}\ \mathrm{Pa}$ Ultimate strength of AISI 316

$k_a := (54.9) \cdot 580^{-0.758} = 0.4415$

$d_e := 0.808 \cdot \sqrt{h_{Rout} \cdot b_{Rout}} = 242.4\ \mathrm{mm}$

$k_b := (1.51 \cdot 242.4)^{-0.157} = 0.3958$ From eqn $(6-19)$ from the textbook

$k_c := 1$

$k_d := 1$

$k_e := 1$

$S_e := k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot S_{e'} = 5.0678 \cdot 10^{7}\ \mathrm{Pa}$ Endurance strength

By adjusting the toruque by 10% the following bending moments and torque values were obtained at the critical point at a distance of 6.1m.

$BM_{z1} := 19571\ \mathrm{N\,m}$	$BM_{z0.9} := 19571\ \mathrm{N\,m}$	$BM_{z1.1} := 19571\ \mathrm{N\,m}$
$BM_{x1} := 749.2\ \mathrm{N\,m}$	$BM_{x0.9} := 714.8\ \mathrm{N\,m}$	$BM_{x1.1} := 873.6\ \mathrm{N\,m}$
$BM_{y1} := 33844.3\ \mathrm{N\,m}$	$BM_{y0.9} := 30460.3\ \mathrm{N\,m}$	$BM_{y1.1} := 37228.9\ \mathrm{N\,m}$
$SF_{y1} := 4635.6\ \mathrm{N}$	$SF_{y0.9} := 4635.6\ \mathrm{N}$	$SF_{y1.1} := 4635.6\ \mathrm{N}$
$SF_{x1} := 0\ \mathrm{N}$	$SF_{x0.9} := 0\ \mathrm{N}$	$SF_{x1.1} := 0\ \mathrm{N}$
$SF_{z1} := 3176.6\ \mathrm{N}$	$SF_{z0.9} := 2858.9\ \mathrm{N}$	$SF_{z1.1} := 3494.4\ \mathrm{N}$
$NF_{x1} := 6710.33\ \mathrm{N}$	$NF_{x0.9} := 7027.5\ \mathrm{N}$	$NF_{x1.1} := 6391.6\ \mathrm{N}$

This yield mean and amplitude moments and shear forces of:

$BM_{zm} := BM_{z1} = 19571\ \mathrm{N\,m}$ not fluctuating, therefore mean only

$BM_{xm} := BM_{x1} = 749.2\ \mathrm{N\,m}$ $BM_{xa} := \frac{(BM_{x1.1} - BM_{x0.9})}{2} = 79.4\ \mathrm{N\,m}$

$BM_{ym} := BM_{y1} = 33844.3\ \mathrm{N\,m}$ $BM_{ya} := \frac{BM_{y1.1} - BM_{y0.9}}{1} = 6768.6\ \mathrm{N\,m}$

$SF_{ym} := SF_{y1} = 4635.6\ \mathrm{N}$ not fluctuating, therefore mean only

$SF_{xm} := SF_{x1} = 0\ \mathrm{N}$ not fluctuating, therefore mean only

$SF_{zm} := SF_{z1} = 3176.6\ \mathrm{N}$ $SF_{za} := \frac{SF_{z1.1} - SF_{z0.9}}{2} = 317.75\ \mathrm{N}$

$NF_{xm} := NF_{x1} = 6710.33\ \mathrm{N}$ $NF_{xa} := \frac{NF_{x1.1} - NF_{x0.9}}{2} = -317.95\ \mathrm{N}$

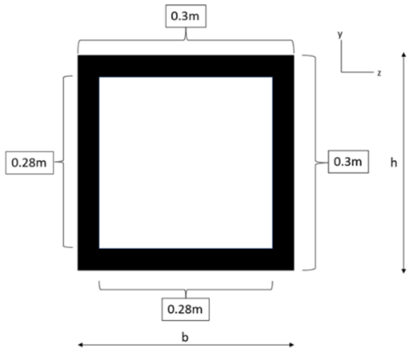
$$I_{yy} := \frac{b_{Rout} \cdot h_{Rout}^{\;3}}{12} - \frac{b_{Rin} \cdot h_{Rin}^{\;3}}{12} = 0.0002\ m^4$$

$$I_{zz} := I_{yy} = 0.0002\ m^4$$

$$Area_x := \left(b_{Rout} \cdot h_{Rout}\right) - \left(b_{Rin} - h_{Rin}\right) = 0.09\ m^2$$

$$ybar := \frac{b_{Rout}}{2} = 0.15\ m$$

$$zbar := ybar = 0.15\ m$$



Plane Stresses:

Mean Values:

Normal Stresses due to axial Force:

$$\sigma_{xm\_normal} := \frac{NF_{xm}}{Area_x} = 74559.2222\ Pa$$

Tension

Bending Stresses:

$$\sigma_{xm\_bending} := \frac{BM_{zm} \cdot ybar}{I_{zz}} + \frac{BM_{ym} \cdot zbar}{I_{yy}} = 4.922 \cdot 10^7\ Pa$$

$$\sigma_{ym} := 0\ Pa$$

$$\sigma_{zm} := 0\ Pa$$

Summed Normal Stresses:

$$\sigma_{x1m} := \sigma_{xm\_bending} + \sigma_{xm\_normal} = 4.9294 \cdot 10^7\ Pa$$

$$\sigma_{y1m} := 0\ Pa$$

$$\sigma_{z1m} := 0\ Pa$$

Amplitude values:

$$\sigma_{xa\_normal} := \frac{NF_{xa}}{Area_x} = -3532.7778\ Pa$$

$$\sigma_{xa\_bending} := \frac{BM_{ya} \cdot zbar}{I_{yy}} = 6.2369 \cdot 10^6\ Pa$$

$$\sigma_{ya} := 0\ Pa$$

$$\sigma_{za} := 0\ Pa$$

$$\sigma_{x1a} := \sigma_{xa\_bending} + \sigma_{xa\_normal} = 6.2334 \cdot 10^6\ Pa$$

$$\sigma_{y1a} := 0\ Pa$$

$$\sigma_{z1a} := 0\ Pa$$

Shear Stresses:

$$A_m := (0.29\ m)^2 = 0.0841\ m^2$$ Area enclosed by mean perimeter

Shear stresses caused by shear forces and Torque about x'-axis:

$$\tau_{xy1m} := \frac{3}{2} \cdot \frac{SF_{ym}}{Area_x} + \frac{BM_{xm}}{2 \cdot A_m \cdot t_R} = 5.2268 \cdot 10^5\ Pa$$

$$\tau_{xz1m} := \frac{3}{2} \cdot \frac{SF_{zm}}{Area_x} + \frac{BM_{xm}}{2 \cdot A_m \cdot t_R} = 4.9837 \cdot 10^5\ Pa$$

$$\tau_{yz1m} := 0\ Pa$$

$$\tau_{xy1a} := \frac{BM_{xa}}{2 \cdot A_m \cdot t_R} = 47205.7075\ Pa$$

$$\tau_{xz1a} := \frac{3}{2} \cdot \frac{SF_{za}}{Area_x} + \frac{BM_{xa}}{2 \cdot A_m \cdot t_R} = 52501.5408\ Pa$$

$$\tau_{yz1a} := 0\ Pa$$

$$\sigma'_a := \left(\frac{1}{\sqrt{2}}\right)\cdot\left[\left(\sigma_{x1a}-\sigma_{y1a}\right)^2+\left(\sigma_{y1a}-\sigma_{z1a}\right)^2+\left(\sigma_{z1a}-\sigma_{x1a}\right)^2+6\cdot\left(\tau_{xy1a}\right)^2\cdot\left(\tau_{xz1a}\right)^2\cdot\left(\tau_{yz1a}\right)^2\right]^{\frac{1}{2}}$$

$$\sigma'_a = 6.2334\cdot 10^6\ \mathrm{Pa}$$

$$\sigma'_m := \left(\frac{1}{\sqrt{2}}\right)\cdot\left[\left(\sigma_{x1m}-\sigma_{y1m}\right)^2+\left(\sigma_{y1m}-\sigma_{z1m}\right)^2+\left(\sigma_{z1m}-\sigma_{x1m}\right)^2+6\cdot\left(\tau_{xy1m}\right)^2\cdot\left(\tau_{xz1m}\right)^2\cdot\left(\tau_{yz1m}\right)^2\right]^{\frac{1}{2}}$$

$$\sigma'_m = 4.9294\cdot 10^7\ \mathrm{Pa}$$

Fatigue Safety factor: ref A4.11

$$n_f := \left(\frac{\sigma'_a}{S_e}+\frac{\sigma'_m}{S_{ut}}\right)^{-1} = 4.8079$$

Appendix B Group charter

Team charter

Code of conduct:
All team members have all agreed upon the following:

Teamwork
Treat all members of the team with respect.
Do not tolerate intimidation or harassment.
Treat all members fairly and don’t tolerate discrimination.
Include all team members in important decisions.
Compromising on different ideas.

Commitment
Each team member is expected to be on time for scheduled meetings.
All members need to complete their assigned work by the agreed date.
Each member has a responsibility to the team to perform their tasks to their fullest
ability.
Be present and engaged.

Diversity
Respect the different cultures and languages in the team.
Don’t discriminate against gender.
Decide not to give offence.

Integrity
Honesty amongst each other.
Do not commit plagiarism.

Contact Information:

Name	Email	Cell phone
Sean La Grange	23551127@sun.ac.za	0840800411
Shankar Palamootil	23687436@sun.ac.za	0742217140
Matthew Prozesky	23641711@sun.ac.za	0820485876
Nico Epler	23910712@sun.ac.za	0826537180

Member agreement:

All members agree to be contactable to between working hours

All members agree to attend all group meetings.

In the event of a member missing a meeting notice has to be given at least 1hr prior to the meeting

Meeting minutes must be disseminated at least 1 day after the meeting. The final version of the minuets will be sent out in pdf format to avoid alterations.

All meeting minuets have to be signed. This will take the form of a physical signature or agreement via email that the attached minuets are correct.

All members will wash their hands before a meeting

Appendix C Work Breakdown Structure and Resource Assignment (SLG)

Level 1:

1. Liquid-solid separation thickener	**100**

Level 2:

1. Liquid-solid separation thickener

1.1 Drive System	**20**
1.2 Mounting Frame	**20**
1.3 Torque Tube	**20**
1.4 Rakes	**20**
1.5 Integration	**10**
1.6 Project Management	**10**
	100

Level 3:

1.1 Drive System	
1.1.1 Motor	**5**
1.1.2 Gear Box	**6**
1.1.3 Coupling	**1**
1.1.4 Drive Shaft	**1**
1.1.5 Spur Gear	**2**
1.1.6 Slew Ring	**5**
1.2 Mounting Frame	
1.2.1 Main mounting frame structure	**7**
1.2.2 Connection to bridge	**5**
1.2.3 Connection to slew ring	**4**
1.2.4 Connection to motor	**4**
1.3 Torque Tube	
1.3.1 Tube	**8**
1.3.2 Flange for connection to slew ring	**4**
1.3.3 Connections to vertical stays	**2**
1.3.4 Connections to rakes	**6**
1.4 Rakes	
1.4.1 Rake arms	**6**
1.4.2 Blades	**6**
1.4.3 Connections to torque tube	**4**
1.4.4 Connections to vertical stays	**2**
1.4.5 Connections to horizontal stays	**2**
1.5 Integration	
1.5.1 Design Calculations	**5**
1.5.2 Assembling thickener by combining all sub-systems	**5**
1.6 Project management	**10**
	100

Appendix D Gannt chart and project schedule (SLG)

Task no	Description	Predecessors	Duration
	Sub-System 1		
1	Generate engineering specifications for sub-system 1	None	3 Days
6	Create functional decomposition for sub-system 1	1	4 Days
11	Create concept layout for sub-system 1	6	5 Days
19	Create free body diagrams for sub-system 1	11	5 Days
23	Material selection and sizing for sub-system 1	11	6 Days
27	Do load calculations for sub-system 1	19, 23	7 Days
31	Create force and moment diagrams for sub-system 1	27	7 Days
	Sub-system 2		
2	Generate engineering specifications for sub-system 2	None	3 Days
7	Create functional decomposition for sub-system 2	2	4 Days
12	Create concept layout for sub-system 2	7	5 Days
20	Create free body diagrams for sub-system 2	12	5 Days
24	Material selection and sizing for sub-system 2	12	6 Days
28	Do load calculations for sub-system 2	20, 24	7 Days
32	Create force and moment diagrams for sub-system 2	28	7 Days
	Sub-system 3		
3	Generate engineering specifications for sub-system 3	None	3 Days
8	Create functional decomposition for sub-system 3	3	4 Days
13	Create concept layout for sub-system 3	8	5 Days
21	Create free body diagrams for sub-system 3	13	5 Days
25	Material selection and sizing for sub-system 3	13	6 Days
29	Do load calculations for sub-system 3	21, 25	7 Days
33	Create force and moment diagrams for sub-system 3	29	7 Days
	Sub-system 4		
4	Generate engineering specifications for sub-system 4	None	3 Days
9	Create functional decomposition for sub-system 4	4	4 Days
14	Create concept layout for sub-system 4	9	5 Days
22	Create free body diagrams for sub-system 4	14	5 Days
26	Material selection and sizing for sub-system 4	14	6 Days
30	Do load calculations for sub-system 4	22, 26	7 Days
34	Create force and moment diagrams for sub-system 4	30	7 Days
	General		
5	Compile the different sub-systems and create engineering specifications table	1,2,3,4	4 Days
10	Create functional decomposition of system	6,7,8,9	4 Days
15	Combine concept layout for each sub-system to create the complete system design.	11,12,13,14	2 Days
16	Compile and write report for PD1	1 to 15	2 Days
17	Assess peer PD1	None	1 Day
18	Make corrections to PD1	16	7 Days
35	Compile and write report for PD2	1 to 34	1 Day

Week columns: 20/2/2022 - 26/2/2022 | 27/2/2022 - 5/3/2022 | 6/3/2022 - 12/3/2022 | 13/3/2022 - 19/2/2022 (days S M T W T F S each)

				3/3/2022 - 9/3/2022							10/3/2022 - 16/3/2022							17/3/2022 - 23/3/2022							24/3/2022 - 30/3/2022							1/4/2022 - 7/4/2022						
Task no	**Description**	**Predecessors**	**Duration**	S	M	T	W	T	F	S	S	M	T	W	T	F	S	S	M	T	W	T	F	S	S	M	T	W	T	F	S	S	M	T	W	T	F	S
	Sub-System 1																																					
44	Assess modes of failure for sub-system 1	27,31	5 Days												■	■	■	■	■																			
50	Perform calculations for modes of failure for sub-system 1	44	9 Days															■	■	■	■	■	■	■	■	■												
54	Incoporate feedback from PD1 and PD2 on sub-system 1	18,35	9 Days																				■	■	■	■	■	■	■	■	■							
	Sub-system 2																																					
45	Assess modes of failure for sub-system 2	28,32	3 Days												■	■	■																					
49	Perform calculations for modes of failure for sub-system 2	45	7 Days														■	■	■	■	■	■	■															
53	Incoporate feedback from PD1 and PD2 on sub-system 2	18,35	12 Days																	■	■	■	■	■	■	■	■	■	■	■	■							
	Sub-system 3																																					
46	Assess modes of failure for sub-system 3	29,33	6 Days												■	■	■	■	■	■																		
51	Perform calculations for modes of failure for sub-system 3	46	8 Days															■	■	■	■	■	■	■	■													
55	Redesign attachment to connect rakes to torque tube	29,33	7 Days																						■	■	■	■	■	■	■							
56	Incoporate feedback from PD1 and PD2 on sub-system 3	18,35	7 Days																						■	■	■	■	■	■	■							
	Sub-system 4																																					
47	Assess modes of failure for sub-system 4	30,34	8 Days												■	■	■	■	■	■	■	■																
52	Perform calculations for modes of failure for sub-system 4	47	11 Days																		■	■	■	■	■	■	■	■	■	■	■							
57	Incoporate feedback from PD1 and PD2 on sub-system 4	18,35	5 Days																								■	■	■	■	■							
	Turned Part																																					
35	Critically assess turned part	13	7 Days	■	■	■	■	■	■	■																												
37	Create concept process plan table for turned part	35	7 Days				■	■	■	■	■	■	■																									
38	Create concept sketches for turned part manufacturing processes	38	7 Days				■	■	■	■	■	■	■																									
41	Create formal drawing for turned part	13,38	4 Days								■	■	■	■																								
	Welded part																																					
36	Critically assess welded assembly	12	7 Days	■	■	■	■	■	■	■																												
39	Create concept process plan table for welded assembly	36	7 Days				■	■	■	■	■	■	■																									
40	Create concept sketches for welded assembly manufacturing processes	39	7 Days				■	■	■	■	■	■	■																									
42	Create formal drawings for welded assembly	12,40	5 Days							■	■	■	■	■																								

Appendix E **Minutes of Meetings**

Minutes of the Meeting of Group 8
Machine Design A 314

on 03/03/2022 and 10:00 in Chalkboard

1 Attendance

- Present:
  - Shankar Palamootil
  - Nico Epler
  - Mathew Prozesky
  - Sean La Grange

- Apologies:
  - None

2 Minutes of the previous meeting

Minutes of the meeting of 28/02/2022 and 10:00 are approved, with the following revisions:

- None

3 Action items

- Peer assessed group 13's PD1

General

- From assessing group 13's PD1, the group also realised flaws in their own design.

4 Next meeting

The next meeting is planned for 04/03/2022 at 17:00

Minutes of the Meeting of Group 8
Machine Design A 314

on 04/03/2022 and 10:00 in Chalkboard

1 Attendance

- Present:
  - Shankar Palamootil
  - Nico Epler
  - Mathew Prozesky
  - Sean La Grange

- Apologies:
  - None

2 Minutes of the previous meeting

Minutes of the meeting of 03/03/2022 and 10:00 are approved, with the following revisions:

- None

3 Action items

- Read through the PD2 scope
- Conceptualise what each group member must calculate
- Create mini deadlines to distribute the work evenly over the week

General

- PD 2 requires the group to do force calculations and force, moment diagrams for each sub-system.
- It was decided that each group member would do the calculations and diagrams for their respective sub-system.
- It was decided that by Tuesday,8th March we would each have done free body diagrams of each sub-system.

4 Next meeting

The next meeting is planned for 08/03/2022 at 16:00

Minutes of the Meeting of Group 8
Machine Design A 314

on 08/03/2022 and 10:00 in Chalkboard

1 Attendance

- Present:
  - Shankar Palamootil
  - Nico Epler
  - Mathew Prozesky
  - Sean La Grange

- Apologies:
  - None

2 Minutes of the previous meeting

Minutes of the meeting of 04/03/2022 and 10:00 are approved, with the following revisions:

- Shankar would help Nico with the overload calculations of sub-system 4

3 Action items

- Check each group members free-body diagrams
- Discuss the calculations each group member should do
- Discuss the feedback the lecturers gave at the 15:00-16:00 session

General

- Everyone’s free body diagrams reflected that they understood the load paths of their respective systems.
- It was decided that according to the lecturer’s feedback, changes had to be made to all of the sub-systems.

4 Next meeting

The next meeting is planned for 11/03/2022 at TBH.

Minutes of the Meeting of Group 8
Machine Design A 314

on 14/03/2022 and 10:00 in Chalkboard

1 Attendance

- Present:
  - Shankar Palamootil
  - Nico Epler
  - Mathew Prozesky
  - Sean La Grange

- Apologies:
  - None

2 Minutes of the previous meeting

Minutes of the meeting of 08/03/2022 and 10:00 are approved, with the following revisions:

- None

3 Action items

- Check each group members free-body diagrams
- Discuss the calculations and fix any mistakes.
- Discuss the layout of PD2

General

- It was decided that more information was needed from the lecturer about the layout of PD2.

4 Next meeting

The next meeting is planned for 15/03/2022 at 16:00.

Minutes of the Meeting of Group 8
Machine Design A 314

on 15/03/2022 and 16:00 in Chalkboard

1 Attendance

- Present:
  - Shankar Palamootil
  - Nico Epler
  - Mathew Prozesky
  - Sean La Grange

- Apologies:
  - None

2 Minutes of the previous meeting

Minutes of the meeting of 14/03/2022 and 10:00 are approved, with the following revisions:

- None

3 Action items

- Finalise calculations
- Finalise report
- SEW drive system guest lecturer

General

- It was decided by Matthew that we would use a planetary gear system.
- We decided to use the Max load condition for the rake as it created more stress than the standard load conditions.

4 Next meeting

The next meeting is planned for TBD at TBD