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PORTABLE ELECTRIC CEMENT MIXER

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EXECUTIVE SUMMARY

This case study highlights and reviews the design process of an electric cement mixer according to the numerical analysis based on methodologies and fundamentals of machine component design. The resulting sub-systems are analyzed in a modular fashion, followed by a combined integration to form the overall system. The resulting analysis ensures the use of belt-pulley transmission systems, bearings, gear train reduction, bolts, welds, and other mechanical structural calculations required to satisfy safety concerns of the system. The designed system must function using a 240V AC electric motor, with a drum designed to rotate approximately 15~20 rpm at an inclined angle between 45 ~60 degrees from the horizontal plane. The analysis verifies calculations and selects components from catalogues of verified manufacturers as listed and referenced. Overall, the project reproduces a 3D rendered model of the finished system, along with an isometric drawing dimensioned to provide appropriate perspective to those with any concerns.

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CHAPTER 1

INTRODUCTION

1.1 Problem Background

In the age of automation and power tools, the construction industry is typically the largest consumer. Hence, a portable cement mixer is the best time-efficient innovation that allows workers to delegate the laborious task of mixing and pouring cement to a portable machine that is electrically powered. Hence, the quality of the cement mix is ensured, and lead-times are shortened. This task delegation seems simplistic at first, however the mechanisms and considerations in designing a reliable and safe machine, certified to be used sustainably in those conditions, is not a simple endeavour.

To elaborate further, it is pivotal to understand the mechanics behind an electric cement mixer. Firstly, like any power tool or other automated machinery, it requires a power source. Secondly, it requires a steadfast mechanism to convert this power for its own use, and lastly, it needs to be reliable enough to be used in rough conditions. This implies that the design should not just be "workable", instead it needs to satisfy various conditions and safety parameters to ensure its operation would not bring harm to any worker on-site. Hence, in this case study, the various parameters required to ensure its safety are analysed, with systematic methodology and a critical overview supported by engineering evidence.

1.2 Problem Statement

The mixing drum to be analysed in this case-study is required to blend a total mass of 150 kg of cement mixture. While mixing, the barrel drum must be inclined at an acute 45° to 60° angle. When this mixing process is completed, the drum is required to recline back into its horizontal position for pouring out the cement. To better visualize the overall system, a sample figure is shown below.

Figure 1-1 Electric Cement Mixer

As shown in the figure above, the cement mixer follows a very traditional design. However, its design criteria are based on the performance required. The case study sets a plethora of requirements regarding its operation.

1.2.1 Design Requirements

- Drum must rotate between 15 to 20 rpm in clockwise direction, and its dimensions should not exceed 1.0 m in diameter and depth.
- System is powered through a 240V AC motor with appropriate torque and power.
- System must integrate gear reducers and belts with pulleys to maintain the appropriate rotational speed required.
- Overall structure is supported using frames that are welded or bolted together.

Once all design requirements are fulfilled, the system can function as intended. However, aside from fulfilling these parameters, the project must be logically convincing and engineeringly sound. Lastly, the project must also be practically applicable, and hence the design process will be based on actual methodologies utilized to calculate and design power transmission systems in use today.

1.3 Project Objectives

The objectives of this project are:

(a) To determine the power and torque required to rotate the cement drum loaded with 150 kg cement.

(b) To design gear reducers and belt-pulley systems to limit the rotational speed as required by selecting appropriate V-belts and pulleys.

(c) To determine the overall dimensions and isometric schematics of the system.

(d) To calculate and analyse the structure with regards to stress experienced by the supporting frame to determine appropriate material, bolt sizes and weld thickness.

1.4 Project Scope

The scope of this project and case study is defined as the basic assumptions and limitations that are necessary for analysis, as well as the depth and detail of the numerical analyses regarding stress and force calculations. In essence, the scope highlights what the case-study considers a baseline which is required to convince beyond a reasonable doubt, that the system is practically and theoretically feasible. However, with regards to this declaration, it is pivotal to also interpret the scope as a logical limitation due to the depth of analyses required.

Hence, moving forwards, this section will define and describe the scope of this case-study in detail. With regards to all motor selections, only the power and torque requirement is considered. Additionally, V-Belt selection is based upon the same power requirement referenced as the transmission power, with the following catalogues set as reference.

- TECHTOP Australia Motor's Catalogue.
- V-Belt Manufacturer- MITSUBOSHI

Furthermore, throughout this report, material strengths, hardness and bearing sizes, along with bearing numbers are referenced from the following:

- Fundamentals of Machine Component Design, Sixth Edition, Juvinall

Lastly, the assumption of design factor is assumed to be (2.5) to ensure all calculations are within acceptable limits of safety.

CHAPTER 2

BASIS OF ANALYSIS AND METHODOLOGY FOR DESIGN OF SYSTEM AND SUBSYSTEM

2.1 Basis of Safe Design and Selection

This chapter discusses the basis for safe design for the mechanical components such as bolts, motor, belts, and weld selection for the system and subsystem. Based on these guidelines, the methodology will be outlined in the later sections of the report.

Considering the base requirements as proposed in the in the introduction, the best possible way to safely design and select mechanical components for the system is to reverse engineer and back pedal from the output stage. As such, the analysis begins, where it ends. At the barrel of the mixer, where its speed is required to be within 15 ~20 rpm. Hence, to begin selection, the power and torque required must be obtained. As no other parameters are given, except the mass of the cement. An assumption can be made for the material of the barrel, and the total mass of the barrel can be obtained. In this case, it was found to be 215 Kg, as calculated in Appendix 1. Hence, by utilizing the equation (2-1) for the mass moment of inertia, assuming the barrel is a solid cylinder with mass of 215 kg.

$$I = \frac{mR^2}{2}$$

(Equation 2-1)

As calculated, the mass moment of inertia was found to be (8.125 Kgm^2). Hence, through rotational dynamics, and by utilizing the torque in relation to angular acceleration α, the equation below can be used to find the torque required for the barrel to rotate between 15~20 rpm.

$$\alpha = \frac{Torque}{I}$$

(Equation 2-2)

Assuming the angular acceleration is (5 rad/s^2). This assumption can be readily made as a typical cement mixer takes approximately four seconds to accelerate from 0 rpm to its top speed of 20 rpm. Hence, with a reasonable estimate of this acceleration, calculate the torque required for the barrel.

2.1.1 Motor Selection Basis

As previously mentioned, the motor selection depends on the power and torque required at its output. However, estimating the power is complicated and requires further assumptions. For this design, the barrel rotates using a face and spur gear mesh combination, as shown in figure 1-1. Hence, by following a standardized diametral pitch and face width for each gear utilized in this report. As established in (Juvinall & Marshek, 2020), utilizing the equation below.

$$\frac{9}{P} < b < \frac{14}{P}$$

Equation 2-3

Hence, if the diametral pitch is assumed to be 10 (teeth/inch), the face width must be greater than 0.9 inches. In this case, a standard assumption of 1 inch suffices. As the diameter of the barrel is known and defined to be 1.0 m, which equates to 39.37 inches. The number of teeth required for the barrel gear can be calculated to be 394 teeth.

Following this calculation, the spur gear that drives this barrel exerts an equal tangential force as is required for the torque. Hence, the required torque of the barrel can be used to determine the tangential force experienced by both gears. However, as the two gears have meshed, there is an existing gear ratio that equates their rotational speeds.

Although the speed of the barrel is assumed to be 20 rpm, the speed of the spur gear is undetermined until a gear ratio is decided. Hence, assuming a standard 40 teeth spur gear to simplify the geometry factor analysis, the gear and shaft's rotational speed can be determined.

Hence, once the tangential force and rotational speed of the spur gear is determined, by utilizing the standard gear power formula below, the required power the shaft transmits to the barrel can be calculated.

$$W = Ft * V$$

Equation 2-4

In the equation above, tangential force is in newtons, while the velocity is in (m/s). Hence, the transmitted power in (Watts) can be determined. This value can be converted to obtain (hp).

Hence, by finalizing the two essential requirements of the 240 V AC motor, power, and torque, by following any commercial catalogue, determine the electric motor required, considering its maximum rotational speed must support the expected speed of the shaft connecting the belt transmission to the spur gear.

Overall, these steps were outlined and carefully followed to determine the type of motor required. The catalogue in this section refers to the base reference as mentioned in the project scope.

2.1.2 V-Belt Selection

Following the power determined in the previous section, as the large pulley is mounted on the same shaft, and all bearings, gears and pulleys are assumed to be 100% efficient, it is reasonable to conclude that the power transmitted is equivalent. Hence, the transmission power of the pulley is equal to the power used to determine the motor, which is verified regardless, as the motor chosen is the one that drives the belt-pulley transmission.

However, the next challenge presented is the speed ratio. As the large pulley mounted to the shaft rotates at the same speed as the spur gear, the missing speed ratio is required to determine the small pulley speed, as well as the belt type required for this transmission. Hence, for reasonable estimates and simplifying the analysis, a common speed ratio of 3.0 is chosen.

Once speed ratio is determined, using the MITSUBOSHI catalogue, determine the service correction factor and hence the design power required for the belt using the equation shown below.

$$Hd = Ht * Ks$$

Equation 2-5

Utilizing this design power and small pulley speed as determined through the speed ratio, use the catalogue to determine the Belt type required. Hence, referencing Table 1-4 in the catalogue, determine the minimum small pulley datum diameter, and use the speed ratio relation to calculate the large pulley datum diameter.

This is followed by an estimate of the centre distance between the pulleys, followed by determining the belt code and then determining the final centre distance as required by the catalogue. Lastly, the number of belts required for the transmission can be determined by the ratio of design power and corrected power rating per belt.

However, as listed in the catalogue the corrected power rating requires one to estimate the basic and additional power ratings (Hs) and (Ha) respectively through the belt type power rating tables. Once determined, utilize the formula as shown below to calculate the correct rating.

$$Hc = (Hs + Ha) * Kc$$

Equation 2-6

Hence, determine the number of belts required for the system, and utilize the dimensions to model the required enclosure and 3D rendered system.

2.1.3 Gear Train Reducers and Material Selection

Once the belt-pulley system has been determined and properly dimensioned, the next step to reverse engineer the problem is to determine the gear train reducer from the input (electric motor) to the output (small pulley).

In the previous section, we determined the small pulley speed, hence, this speed is effectively the output of the gear train. The gear parameters such as diametral pitch and face width remain the same. Hence, a two-stage reduction can be used. As no parameters are present to determine the stages. For manufacturing and inventory ease, the pinion of the gear train can be set to be the same as the spur gear for the barrel.

Hence, the first stage, connecting the motor input to a secondary shaft, reduces the motor's base speed by a factor determined by the gear ratio, which must be assumed. The second stage, connecting the secondary shaft to the small pulley shaft reduces the speed further once more, providing just enough speed to ensure the barrel spins within 15 ~ 20 rpm. The exact methodology essentially is to creatively utilize the gear equations relations diameter and speed.

$$\frac{\omega p}{wg} = \frac{np}{ng} = -\frac{dg}{dp} = -\frac{Ng}{Np}$$

Equation 2-7

Hence, by shuffling around the diameters, number of teeth and expected rotational speed (n), we can determine the gear ratios required for each stage of reduction.

However, although this criterion would function adequately, to effectively ensure that the chosen gear configuration is viable, a stress analysis must be done. To simplify this process, the bending and surface fatigue of the gear tooth can be tested to ensure that no failure would take place while transmitting this power. Hence, following the formula below, and assuming a design factor of 2.5, the material strength required to sustain the load can be approximated.

$$\eta = \frac{Sn}{\sigma e}$$

Equation 2-8

Where σe represents the equivalent stress caused by the tangential force on the gear tooth. The two variables are as represented by the equations below.

$$Sn = S'n[Cl * Cg * Cs * Kt * Kr * Kms]$$

Equation 2-9

$$\sigma e = \frac{FtP}{bJ} * Kv * Ko * Km$$

Equation 2-10

The first priority to determine the equivalent stress is to determine all the factors affecting the outcome. Namely, the velocity factor Kv, overload correction factor Ko, and mounting correction factor Km. It is noted that to determine the Kv, the surface quality and manufacturability must be assumed. In this case, it was assumed that Qv = 6, signifying that the relevant formula for curve B should be used to calculate the velocity factor.

Additionally, it was assumed that the power source and driven machinery were both uniform, and mounting were accurate with face width of 1 inch. Hence, Ko and Km were determined to be 1.0 and 1.3 respectively. Furthermore, J which is the gear factor, is referenced from figure 15.23, (Juvinall & Marshek, 2020), assuming a pressure angle of 20 degrees and no load sharing for the gears involved.

Once a value for the stress is obtained, calculate the S'n of the material, and hence following the principle that the fatigue strength for a bending load is half of its ultimate strength, calculate the relevant ultimate strength of the material and evaluate a material to be used from the Appendix C-1 (Juvinall & Marshek, 2020).

Lastly, to determine whether the gears are safe from surface fatigue follow a similar principle, except this time utilize two different equations for strength and stress relating to surface fatigue.

$$SH = Sfe * CLi * CR$$

Equation 2-11

$$\sigma H = Cp\sqrt{\frac{Ft}{b * dp * I} * Kv * Ko * Km}$$

Equation 2-12

All variables listed above are the same as the bending fatigue analysis, except the Cp, and I. Cp corresponds to the elastic coefficient of materials, referenced from Table 15.4 a, (Juvinall & Marshek, 2020). Whereas I, commonly referred to as the geometry factor is calculated through the equation shown below.

$$I = \frac{Sin\phi Cos\phi}{2} * \frac{R}{R+1}$$

Equation 2-13

Hence, once the stress is determined, compare to calculate the required strength of material. Overall, evaluate whatever material should be used to sustainably implement the gear train reducer.

2.1.4 Bearing Size and Number Determination

For any system supporting rotating shafts and gears, it is essential to include bearings in the design to support the shaft's radial and thrust load. Hence, for this system, the basis of bearing selection were the standardized procedures, beginning with force analysis, followed by calculation of equivalent load, and the required bearing capacity to sustain the load. Hence, bearings from all series were then listen, and a suitable bearing is selected from the ones available.

All bearing selections are done using (Juvinall & Marshek, 2020) bearing catalogue as a reference. However, to simplify the bearing analysis process, first it is necessary to define the locations of all the bearings involved. Overall, there are four bearings in the structure. The figure below crudely highlights bearings.

Figure 2-1 Bearing Locations

The first two bearings support the weight of the barrel, as well as the shaft rotation for the spur gear. Whereas, bearing 3 supports the angular rotation of the barrel to pour out the cement once process is completed. Lastly, the bearing at the bottom supports the rotation of the barrel about its own axis, as well as supporting most of the weight of the barrel and cement mixture.

The working hour assumption for each bearing is slightly different, as the rotational speed of the bearing 1,2 depends on the shaft's rotation, whereas the bearing on the right only supports a very low rpm, as it is only operated when the cement mixture is done. Lastly, bearing 4 supports the slow rotation of 15~20 rpm of the barrel. Hence, each bearing has a different assumption of working ours.

Following this assumption, the reliability and rated life of each bearing is the standard 90% and 90 million cycles to align with the bearing catalogue. Hence, radial and angular ball bearings are evaluated. Although there isn't significant thrust load for the system, it is far safer to select radial ball bearings as compared to roller bearings due to the safety in case of unexpected thrust load.

As bearing for each series are listed, the bearing basic number is determined, and the best bearing is chosen according to the rated capacity of the bearings. The basis of selection is described briefly for each bearing.

<u>Bearing 1:</u>

Radial force is the only force applied to the first bearing. The forces operating perpendicularly on the bearing on two separate axes, their resultant will be the radial force. The radial and tangential forces generated by the gears' contact are what cause the bearing forces acting on various axes. once the radial force acting on bearing 1 has been obtained. We only need to perform further calculations to determine the rated capacity. By assuming uniform no impact, 90% reliability, and Lr=90x10^6, the rated capacity can be determined. The bearing life may then be calculated using the shaft rpm that is 197 rpm and the bearing's functioning time. Once the functioning time is determined the required bearing capacity can be determined. For bearing 1, an angular ball bearing is considered, hence the formula is as shown below.

$$Creq = FrKa\left(\frac{L}{KrLr}\right)^{0.3}$$

Equation 2-14

Any assumptions made and not listed in the methodology are defined in appendix-2 for the working solution in selecting each bearing.

Bearing 2

Bearing 2 will only be subject to radial force. Also, according to the cement mixer, bearing 1 will be inside Bearing 2. To calculate the radial force, we simply need to assume that the sum of forces on the y axis equals zero. We then need to calculate the weight load of the mixer, which equals 2109.15 N, and calculate how much of this load will be carried by each supporting side. There is nothing missing now. Note that since the radial force pressing on the gears is on the same axis as the radial force from Bearing 2, we will only be using the Fr, not the resulting force from Bearing 1. 1025 N will be the radial force with bearing 2. Next, we will just use the rated capacity assumption from bearing 1 again.

Bearing 3

A radial force only will be applied to bearing number 3. Simply assume that forces on the y axis are equal to zero, as was the case in the calculations for bearing 2, and you will be able to determine the radial force of bearing 3. The supporting force, which acts in opposition to the radial force in this bearing, is the only difference. The radial force of bearing 3 will exactly match the supporting force, which is equal to half of the total weight load which is 1.05458 kN. Also, the equivalent radial force will equal the radial force obtained since there is no thrust force. With one exception of changing the bearing speed to 7.5 rpm from 197 rpm as in bearings 1 and 2 resulting in smaller bearing life. the rated capacity will be obtained in the same way through equation (2-14).

Bearing 4

Bearing 4 is unique, since the bearing can be considered as inclined and thus, there is a thrust force occurring along with a radial force. Analyse the load force and break it down into components. Once, we will use 60 degrees and another time we will be using 45 degrees. The load that's going to be parallel to the bearing axis is going to be the thrust force. While the perpendicular is going to be the radial force. Then we will be having the following results thrust force and radial force equal to 1.827 kN and 1.055 kN respectively. Bear in mind those values obtained when the angle for the bearing was 60 degrees. When it was 45 degrees the value was 1491.34 N for both thrust and radial forces. Then, for the rated capacity calculation. It's going to be completely the same as bearing 3. but there's a main difference that we are going to include the equivalent radial force. In the past 3 bearings calculations. We simply equated the radial force with the equivalent radial force since there was no thrust forces in any of the cases. But here since there is a thrust force acting on the bearing. we will be dividing the radial force from the thrust force to find the ratio. The ratio was equivalent to 1 when the angle was 45 degrees and 1.732 when the angle was 60 degrees. For both cases will we end up using the same equation. Only 45 degrees is considered as it is the critical position. Then, we will substitute the values of the equivalent radial force into the rated capacity equation (2-14). Obtaining, rated capacity values for 45 degrees.

2.1.5 Weld Size Determination

The bolt and weld sizes are determined by first analysing the load acting on the structure. Primarily, the load is due to the total mass of the barrel and cement mixture, as well as the enclosure box accounting for all gears, and belt transmission system.

Referring to the structure, the horizontal structural support is welded to the vertical right-side column around its parameter. The determination of the weld size is governed by a series of formulas. The analysis considers primary loading only – transverse shear. No bending or torsional loads are experienced by the welded shaft. Initially, the formula for transverse shear stress experienced by the weld is given by the equation shown below.

$$\tau_{max} = \tau_v = \frac{V}{A_w}$$

Equation 2-15

The shear force (V) corresponds to the reaction force experienced (Fw) at edge of the horizontal shaft [refer to the Appendix weld analysis solution]. The weld area – given that it covers the perimeter of the rectangular cross-section of the horizontal shaft – can be found by the equation below, which is the product of the perimeter length (L) and the weld thickness (h).

$$A_w = L \times h$$

Equation 2-16

Hence, the shear stress is obtained in terms of h. By employing the formula for safety factor according to the Distortion Energy Theory, the weld thickness (h) can be obtained after assuming the usage of AWS Electrode Number E60 to determine the yield strength of the weld. The following equation will only have the weld thickness (h) as the unknown to be found; ultimately completing the calculation of the weld size.

$$\eta_{DET} = \frac{0.577 S_y}{\tau_{max}}$$

Equation 2-17

2.1.6 Bolt and Structural Dimensions and Requirements

The analysis of the structure to investigate the possibility of critical bending stress is necessary for the durability of the cement mixer. The structure was analysed by utilizing the fundamental third law of Newton.

Referring to the FBD sketch of structure analysis in the appendix, there were two unknowns to be found: F_B – the force at bolts, and F_w – the force at weld. Moment of point O was equated to zero to eliminate the force F_w and help determine the force F_B since it is now reduced to one unknown. Subsequently, the sum of forces in y direction can be taken to find F_w, since the value for F_B is now known.

Looking at the forces, a moment of point x is expected to take into effect. Given all forces are now know, the moment M_R can be calculated.

Furthermore, the shear force and the bending moment diagrams were constructed to determine the maximum moment. Therefore, the most critical bending stress can be determined using the following formula.

$$\sigma_b = \frac{M_{max} \times c}{I_{xx}}$$

Equation 2-18

The moment of inertia about the x axis (I_{xx})was determine using the following formula for hollow beams.

$$I_{xx} = \frac{1}{12}(b_o d_0^3 - b_i d_i^3)$$

Equation 2-19

Upon determining the value of bending stress, it was found to be very minute. It was decided that it was negligible to consider the bending and shear stresses experienced by the beam due to their insignificance compared to the yield strength of the shaft material 1045HR (414MPa).

The two inclined beams supporting the wheel of the structure are attached to the horizontal beam by 4 bolts: two on each side. The bolts are expected to experience the primary transverse shear force F_B and the secondary torsional shear stress due to the moment M_R. Utilizing the following formulas, the transverse shear stress and torsional shear stress are obtained.

$$\tau_v = \frac{V}{A_r}$$

Equation 2-20

$$\tau_T = \frac{T}{A_t}$$

Equation 2-21

Therefore, resultant shear stress can be obtained using Pythagoras theorem since the angle at which the torsional shear stress acts is perpendicular.
Finally, following the Distortion Energy Theory equation for safety factor, the required proof Strength can be determined. Hence, a suitable bolt SAE class is selected.

2.2 Limitation

The limitations in this analysis stem from the obvious fact that too little information is provided for the scenario. Many important relations that impact every factor and calculation are assumed as reasonable assumptions, hence it is likely that the resulting analysis is not entirely accurate or coherent. This is largely due to integration problems as the sub-systems are designed and analyzed in a modular fashion, and hence, when put together, often fail to reproduce expected results.

Secondly, the structural analysis is not completely viable in practice. As many assumptions are made regarding material choice, thickness of structure, and load distribution that might not hold up numerically. It is far more accurate to use techniques such as finite element methods and use CAD software to analyse critical points and modes of failure. However, in case of manual analysis, often, the limitations are our assumptions in load and stress distribution, as well as an inability to accurately define critical points in the structure.

Overall, all limitations in this report are systematic and difficult to tackle. The assumptions listed down normalize the process, however, it is best to consider these results from an engineering perspective and qualitatively consider this an initial iteration in a long process to properly model a working cement mixer.

CHAPTER 3

Analyses of Subsystems

This chapter discusses the results of stress analysis and selection of mechanical component based on numerical calculation of all subsystems included in the appendix 1, appendix 2 and appendix 3. Furthermore, the calculations follow the basic sketch as shown in the figure below.

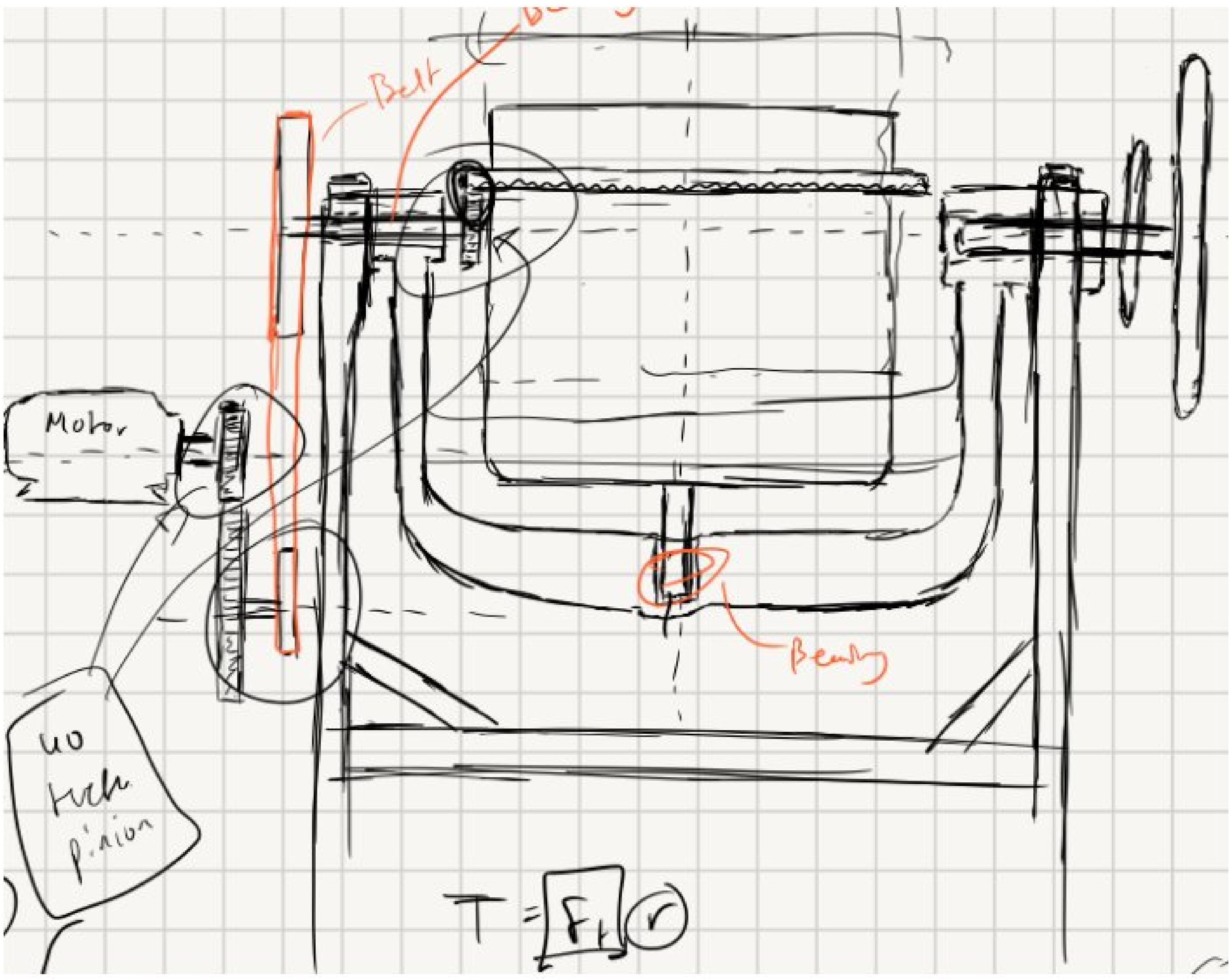

Figure 3-1 Basic Sketch of System

3.1 Motor, Gear Reducer and Belt-And-Pulley System

As shown in appendix 1, following the methodology stated in the previous section, the first step is to calculate the required torque, which is found to be (40.63 Nm). Hence, the following step, by utilizing the preselected gear parameters, a value for the barrels number of teeth (394) and spur gear teeth (40) is found. Hence, the transmitted power from the shaft to the barrel is calculated using the formula previously shown, (0.09 kW). These two parameters are used to select a motor from the TECHTOP Australia Catalogue. Hence, following the catalogue, a TMY series aluminium, single phase, motor is selected matching the required perimeters. It is noted that it has a base speed of 2760 rpm. The results discussed are tabulated in the table below.

Torque (Nm)	40.63
Power (kW)	0.09
Spur Gear Teeth	40
Barrel Face Gear Teeth	394
Motor Speed (rpm)	2760

Table 3-1 Results for Motor Calculations

Hence, following the motor selection, the V-Belt-pulley system is designed with the required power of (0.09 kW) as transmission. By following the MITSUBOSHI catalogue in the methodology as shown, Hence, using a speed ratio of 3.0, a small pulley speed of 591 rpm, and a design power of (0.192 hp), the belt type selected is type A. Following this, minimum small pulley datum diameter is selected from the catalogue to be 3 inches. With a speed ratio of 3, the large pulley datum diameter is found to be 9 inches. Lastly, the centre distance is found to be 27.56 inches with designated length 74.3 inches.

Additionally, the number of belts required for this transmission are calculated and found to be one. This is only possible as the power transmitted is very low and hence, does not require more than one belt to safely transmit.

Results	Value
Transmitted Power (kW)	0.09
Design Power (hp)	0.192
Small Pulley Speed (rpm)	591
Small Pulley Datum Diameter (in)	3
Large Pulley Datum Diameter (in)	9
Centre Distance (in)	27.56
Designated Length (in)	74.3
Number of Belts required	1

Table 3-2 Results for V-Belt Selection

Following the V-Belt system, the next step is to configure the gear train reduction so that a motor input of 2760 rpm can be safely reduced to 591 rpm. Following the design previously presented, the figure below illustrates the gear train for better analysis.

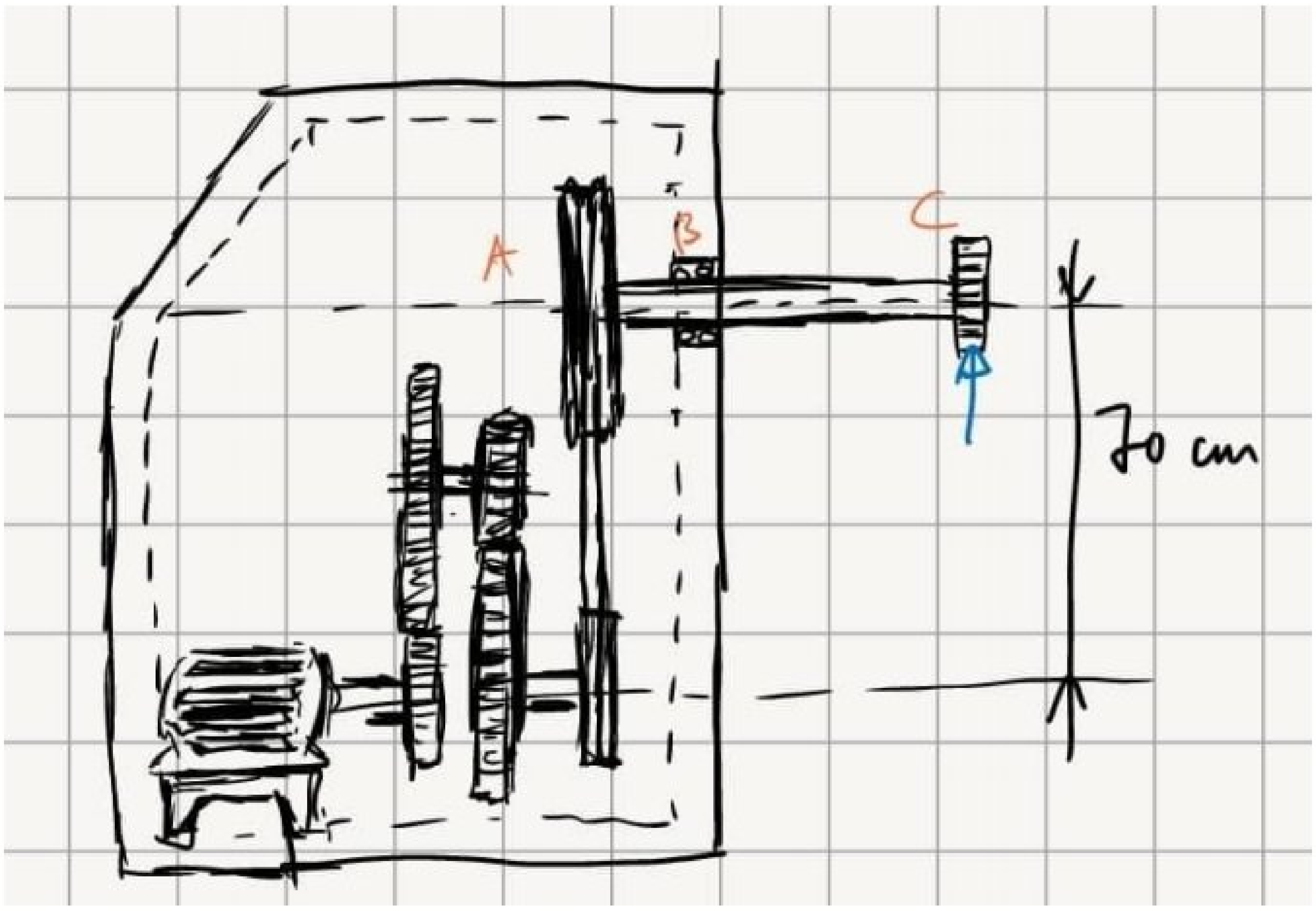

Figure 3-2 Gear Train Basic Sketch

As shown in the figure above, the gear train follows a two-stage reduction. Following the analysis as described in the methodology, the number of teeth for each gear are determined. The pinion is found to have 40 teeth, whereas the next gear is selected to have 100 teeth to have a gear ratio of (2.5). Hence, the speed is reduced to 1104 rpm from the first stage. Following this reduction, the second stage further reduces the speed, with the third gear having 40 teeth, and the final output gear having 75 teeth. The speed is reduced to 588.8 rpm, which is now the input speed of the small pulley. Noticeably, the input speed is lower than 591 rpm, which is previously assumed. However, the speed still manages to transmit enough power and torque to rotate the cement drum within 15~20 rpm. Hence, the rotational speed at the out of the gear train is within acceptable limits.

However, to fully ensure the gear train, as stated, there must be a fatigue analysis. Hence, following the methodology, by determining the minimum stress S'n, required to sustain this power transmission, a suitable material can be selected.

Hence, it is determined that the equivalent stress applied to the gear (107.49 psi) is very low, and hence the ultimate strength required is 0.43 ksi. Referencing Appendix C-1 of (Juvinall & Marshek, 2020), every material stated is more than qualified to be used for gear manufacturing as all have greater strength than required.

For the sake of analysis, an aluminium-bronze alloy is chosen to showcase the safety factor for surface fatigue. The surface fatigue strength is determined to be 73.125 ksi, whereas the Hertz stress is found to be 4988 psi. Hence, the safety factor is an astounding 14.67, which is more than required. However, Aluminium-Bronze alloy is the one of the weakest materials with regards to ultimate strength, offered in the reference material (Juvinall & Marshek, 2020). Hence, for ease of analysis, any material in Appendix C-1 can be chosen as the gears will in no circumstances be subjected to tooth-bending or tooth-surface fatigue. Hence, aluminium-bronze alloy can be used to manufacture the gears.

3.2 Bearing

As noted in the methodology section and shown in appendix-2 the bearing selection is based on the required equivalent load, as well as the estimated life of the bearing to select the appropriate bearings from a catalogue. Bearings not on the same shaft, or at different locations along the system have different life expectancy, and hence different requirements for bearings. The analyzed bearings are all angular ball bearings to simplify the analysis. Hence, following the methodology, using a simple force analysis, the radial load for bearing-1 is found to be 86.475 N, as no thrust force acts on bearing-1, this is assumed as the equivalent load and a bearing capacity is calculated using relevant assumptions as listed in appendix 2.

The bearing capacity required is approximately 0.13 kN. Hence, selecting from angular ball bearings, table 14.2 (Juvinall & Marshek, 2020), three bearings are chosen, one from each series. Namely, the bearings of series LXX, 2XX, and 3XX with force capacities of 1.02, 1.0 and 1.88 kN respectively. Hence, the bearing with basic number 300 and bore size 10 mm is chosen due to its highest force capacity of 1.88 kN. However, in this scenario, where bearings are not limited by force capacity, the bearing can also be chosen due to other circumstances, such as predetermined shaft size, or cost.

Bearing Series	Bore Size (mm)	Rated Capacity (kN)
LXX	10	1.02
2XX	10	1.10
3XX	10	1.88

Table 3-3 Bearing 1 Selection Results

Following bearing 1, bearing 2 and bearing 3 are analysed as shown. As no thrust force is applied on bearing 2, the equivalent load is only the radial load, which is calculated to be 1025 N. Hence, following the rated capacity equation, bearing 2 requires 0.317 kN. Furthermore, a similar analysis on bearing 3 reveals that its rated capacity is 0.326 kN. Hence, since the two loads are very similar, only one bearing is chosen for both, this is to drive down costs and reduce inventory and storage redundancies.

Hence, following the analysis and referencing table 14.2 for a selection, three bearings are chosen as tabulated below.

Bearing Series	Bore Size (mm)	Rated Capacity (kN)
LXX	35	4.75
2XX	35	8.20
3XX	35	11

Table 3-4 Bearing 2, 3 Selection Results

Finally, the best bearing is chosen according to its rated capacity. Hence, the chosen bearing in this scenario for locations 2 and 3 is bearing number 307, with bore 35 mm and rated capacity of 11 kN.

Lastly, bearing 4 is selected, although its rated life is lower than bearing 1 as it does not spin at the same rate as the shaft, instead it supports the rotation of the barrel. Although two configuration 45 and 60 degrees are mentioned, the bearing is only chosen at 45 degrees as analysis proves that it is the most critical angle, with the highest equivalent load. The load is determined by the total weight of the barrel and cement mixture inclined at a 45 degree angle and it is calculated to be 2.334 kN. Hence, the rated capacity is calculated to be 1.7736 kN and three bearings are selected from angular ball bearing table 14.2 as tabulated below.

Bearing Series	Bore Size (mm)	Rated Capacity (kN)
LXX	20	2.20
2XX	17	2.20
3XX	10	1.88

Table 3-5 Bearing 4 Selection Results

Hence and finally, the bearing is chosen according to the best rated capacity for maximum safety. In this scenario, the two rated capacities for LXX and 2XX series are the same. Hence, the decision falls on cost, since LXX bearings are typically cheaper, the selection will be from the extra light series with a bore size of 20 mm and rated capacity of 2.2 kN and bearing basic number of L04.

3.3 Support Structure

The calculation in Appendix 3 was performed according to the aforementioned methodology. The forces F_B and F_W acting on the bolts and weld were found to be 964.25 N and 169.2 N respectively. These forces were utilized in the analysis of their respective receipt components. Upon analysing the structure for bending failure, the bending stress was found to be 17.5 KPa, which is negligible as mentioned before.

Following the methodology (and referring to Appendix 3), the bolt and weld analysis results were summarized in the following table.

Result	Value
Required Sp	125.18 MPa
SAE Class	4.6
Bolt Diameter	5mm
Selected Bolt Sp	225 MPa

Table 3-6 Bolt Calculation Results

Result	Value
AWS Electrode No.	E60
Yield Strength	345 MPa
Weld thickness, h	10.1 mm
Weld Length, L	210 mm
Thickness, t=0.707h	7.14 mm

Table 3-7 Weld Calculation Results

CHAPTER 4

ANALYSIS OF SYSTEM

This chapter integrates all analysis from the subsystem from the previous chapters, and includes the overall system modelled to give a perspective regarding its scale and dimensions.

4.1 Completed Rendered Model

Integrating all subsystems regarding structural requirements, belt-pulley system dimensions, gear configurations and calculations, a completed rendered model of the system is generated as shown in the figure below. The complete isometric drawing with relevant dimensions in an A3 format, is attached in Appendix 4.

Figure 4-1 Completed Rendered Model

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

This chapter summarizes what has been done and highlights results for this project. Furthermore, recommendations regarding further analysis as well as errors or limitations in the current analysis are briefly discussed.

5.1 Research Outcomes

Overall, the system was analyzed in a modular fashion and primarily it relies on reverse engineering from the output to the cement barrel. The output is traced back towards gears, belt-pulley system and then a gair train reduction to its input from an electric motor. Each stage is carefully analyzed and determined through calculations that are attached in the appendix and reference accordingly within this report. The resulting system also includes various dimensioning requirements to withstand the applied load due to the total weight of the system. Hence, a portable electric cement system was successfully designed.

However, although a complete model was produced, as well as dimensioned. There are certain factors that are inhibiting the accuracy of these calculations. Primarily, the analysis of rollers to be used on the system cannot be accurately determined as a finite element analysis is required to determine the stress and load applied to each wheel appropriately. In the instance that this isn't available, the analysis is overcomplicated and not much can be done without major assumptions leading the perspective. Furthermore, many assumptions made had no primary bases and were assumed for ease of analysis and due to the availability of information regarding those assumptions within the reference materials. In the practical application of such a system, the pool of information available is far wider, and hence, the choices are far more appropriate. Instead, the choices made in this case study are very limited and merely approximations.

5.2 Recommendations

The primary recommendations for this project to succeed are the use of software such as SOLIDWORKS and other CAD programs to perform finite element analysis on the systema as a whole and on the sub-assemblies. As it is brought to attention, on numerous sections across this report, that the sub-system analysis can be done manually, however an exact analysis of the overall system is near impossible to approximate due to the uncertainties of load distribution and other factors. Hence, a finite element analysis will reveal critical junctions that were overlooked. These might prove to be limiting factors that need to be improved. Hence, this analysis will fuel more development towards the project. Aside from the use of digital calculation and verification, the project succeeds at every step, however there are still a lot of ambiguities regarding approximations, and it is best to derive or obtain guidelines to base analysis on, as it reduces unknown variables and analysis is simplified.

5.3 Conclusion

That is not to suggest that the analysis written and calculated is wrong by any measure, however, here is a degree of practically missing that can only be introduced if more guidelines and variables are implemented. The factors not considered are cost, availability of materials, manufacturing complexities, overall production, and assembly complexity as well as the practicality as a consumer product. These factors are major influencers in engineering design. Although they do not determine or control any analysis. Overall, they do present guidelines that engineers can derive or an understanding from. Hence, these requirements fuel the design process with ideas. The smaller the selection pool and possibilities the more refined the design. Ambiguity is the biggest limitation as there is no exact answer or justification that trumps the other.

In conclusion, the objective of this case study was to review and design an electric cement mixer and all its mechanical components with a brief methodology and detailed manual mathematical analysis. This objective was achieved, despite there being numerous approximations in the result, the resulting design produced fulfils all client requirements to the best of its ability.

Appendix 1 Mathematical Analysis for Motor, Belt and Gears

No. ____

Date ____

Muhammad Arslan Babur - A20EM3003

Component Design

Major- Project

· Material of barrel : Steel-alloy, $\rho = 7850\ Kg/m^3$
(AISI-4340 steel)

According to Solid-works model, barrel volume for a diameter of 1.0 m and depth of 1.0 m is 8058259 mm^3

Hence, mass of barrel is approximately 65 Kg.

Combined with mass of cement, total mass is :

Total-mass = 65 + 150 = 215 Kg

∴ Barrel must rotate at approximately 20 rpm

Assuming a perfectly cylindrical-shape for the barrel

mass-moment of inertia = $\frac{mr^2}{2}$ (About cylindrical axis)

$$I = \frac{65 \times 0.5^2}{2} = 8.125\ Kgm^2$$

Using rotational dynamics ; $\alpha = \frac{T}{I}$ $\left\{ \begin{array}{l} \alpha = \text{angular acceleration} \\ T = \text{Torque} \end{array} \right.$

$T = I\alpha$ { assuming an angular acceleration of 5 rad/s²

$T = 8.125 \times 5 = 40.63\ Nm$ { Torque required

Muhammad Arslan Babur

A20EM3003

• Torque transmitted to the barrel is approximately 40.63 Nm

Torque = $F_t \times 0.5$ { barrel tangential force

$\boxed{F_t = 81.26\,N}$ ∴

∴ Assuming gear parameters; $P = 10$ teeth/inch

$b = 1$ inch

as according to manufacturing requirement, (standardized)

$$b > \frac{9}{P} \quad \text{and} \quad b < \frac{14}{P}$$

Hence; $b = 1.0$; $1 > \frac{9}{10}$ ∴

Hence, barrel face gear has $P = 10$ teeth/inch

$$P = \frac{N}{39.37\ (\text{inch})}$$

$\boxed{N = 394 \text{ teeth}}$ ∴

Assume a gear ratio with shaft gear and barrel

$$\frac{N_g}{N_p} = 9.85$$

$$\boxed{N_p = \frac{394}{9.85} = 40 \text{ teeth}}$$

No.

Date

Mohammad Arslan Babar

Calculate rotational speed

$$\frac{N_g}{N_p} = \frac{n_g}{n_p}$$

$n_p = 20 \times 9.85 = 197$ rpm (shaft speed)

Gear-power

$\dot{W} = F_t \times v$ $\begin{cases} F_t = 81.26 N \\ v = \frac{\pi d n}{60,000} = \frac{\pi \times 101.6 \times 197}{60000} = 1.048 m/s \end{cases}$

$\dot{W} = 85.16 W = 0.08516 kW$

$\boxed{\dot{W} \approx 0.09 kW}$

utilizing the torque and power requirement.

∴ Using any commercial motor catalogue for 240v AC motors.

∴ In this case, refer to TechTop Australia; Motor catalogue

TM series — Aluminium single phase motors
power rage: 0.06 to 7.3 kW

Select -TMY-series ; power = 0.09 kW = 0.12 hp

Base speed = 2760 rpm

Arsls

Muhammad Arslan Babar

Hence, appropriate motor is selected

Design Belt-pulley system to reverse engineer

Output speed = 197 rpm

∵ Assume speed ratio = 3.0

Input speed = $197 \times 3 = \underline{591}$ rpm

∴ Determine belt-type using Mitsuboshi-catalogue

$H_d = H_t \times k_s$; $H_t = \underline{0.12}$ hp

$k_s = k_e + k_o + k_i$

$k_o = 1.4$ (Assuming gyratory system, runs 8-12 hours a day)

$k_i = 0.0$ (No idler)

$k_e = 0.2$ (Dusty environment)

$H_d = 1.6 \times 0.12 = \underline{0.192}$ hp ; $n = 591$ rpm

Using Fig 1-1 : Belt type A is chosen.

Muhammad Arslan Babar

∵ Hence, choose minimum small pulley datum diameter.

$\boxed{dd = 3 \text{ inches}}$

$$SR = 3 = \frac{Dd}{dd}$$

$\boxed{Dd = 9.0 \text{ inches}}$

∵ Assuming large pulley is in line with cement barrel at a height of 70cm.

∵ Assuming small pulley is in line with bottom of the barrel at height/depth 0 cm.

* Interim centre distance is 70 cm or 27.6 inches

$C' = 27.6$ inches

$Ld' = 2C' + 1.57(Dd + dd)$

$Ld' = 2 \times 27.6 + 1.57(9+3) = 74.04$ inches

∵ Length designation is $75 \sim 79$ inches; $k_L = 1.03$

Belt-code is A73, with $Ld = 74.3$ inches

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Hence, $C = \frac{b + \sqrt{b^2 - 8(Dd - dd)^2}}{8}$

$b = 2Ld - \pi(Dd + dd) = 110.9$

$\boxed{C = 27.56 \text{ inches}}$ ∵

∵ Determining number of belts

$nb = \frac{Hd}{Hc}$ $\begin{cases} Hd = 0.12 \\ H_c = (Hs + Ha) \times k_c \end{cases}$

$k_c = k_\theta \times k_L$ $\begin{cases} k_\theta = \frac{9-3}{27.56} = 0.2177 \end{cases}$

interpolation

$0.2 \longrightarrow 0.97$

$0.3 \longrightarrow 0.96$

$\frac{0.3 - 0.2}{0.2177 - 0.2} = \frac{0.96 - 0.97}{x - 0.97}$

$k_\theta = x = 0.968$

$k_c = 1.03 \times 0.968 = 0.997$

Hs at 591 rpm, 3.0 inches datum diameter, interpolate

$600 \longrightarrow 0.81$; $\frac{600 - 500}{591 - 500} = \frac{0.81 - 0.71}{x - 0.71}$

$500 \longrightarrow 0.71$

Muhammad Aisha Babar

$\boxed{H_s = x = 0.801 \text{ hp}}$

H_a at $SR > 1.57$

$600 \rightarrow 0.17$ } interpolate
$500 \rightarrow 0.14$

$H_a = 0.1673$ hp

$H_c = (0.1673 + 0.801) \times 0.997$

$H_c = 0.9653$ hp

$$n_b = \frac{H_d}{H_c} = \frac{0.12}{0.9653} = 0.124$$

∵ Only one belt is needed for power transmission.

Designing Gear reducers :

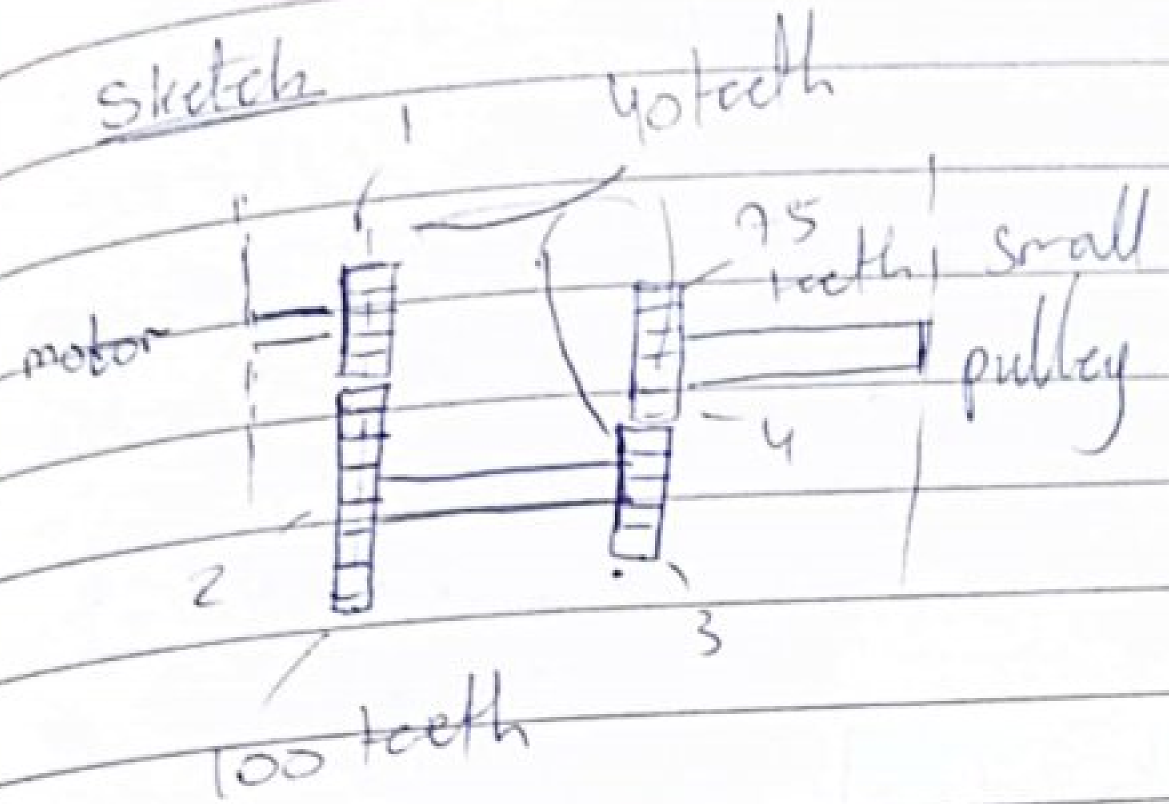
small pulley speed = 591 rpm (targetted output)

motor input = 2760 rpm (full load)

Hence, assuming reduction happens in two stages

Muhammad Aslan Babur

Sketch



gear-1 (pinion)

$$\frac{N_p}{N_g} = \frac{40}{100} = \frac{n_g}{2760}$$

$$\boxed{n_{g_2} = 1104 \text{ rpm}}$$

$$\frac{N_B}{N_g} = \frac{N_3}{N_4} = \frac{40}{75} = \frac{n_4}{1104}$$

$$\boxed{n_4 = 588.8 \text{ rpm}}$$

591 rpm ensures barrel spins at 20 rpm. lower speed ensures barrel spins between 15 ~ 20 rpm.

Hence, gear reduction is acceptable

Muhammad Arslan Babar

Testing gear strength

Bending fatigue

$$\eta = \frac{S_n}{\sigma_b}$$

$$S_n = S'_n \left[C_L C_G C_S k_T k_r k_{ms} \right]$$

∵ Calculation is to determine best material choice for gears to sustain function in given configuration.

$$\sigma_b = \frac{F_t P \, k_v k_o k_m}{bJ} \quad \begin{cases} P = 10 \\ b = 1 \end{cases} \quad F_t = \frac{W}{v} = \frac{0.09 \times 1000}{14.68} = \underline{\underline{6.13N}}$$

∵ Analyze pinion as it is most critical gear.

geometry factor from $\theta = 20°$, no load sharing
$N = 40$ teeth

$$J = 0.28$$

$$K_v = \frac{78 + \sqrt{v}}{78} \quad \{ \; Q_v = 6 \; ; \text{ Curve C}$$

$$V = \frac{\pi d n}{12} = \frac{\pi \times 4 \times 2760}{12} = 2890.27 \text{ ft/min}$$

$$\boxed{k_v = 1.69}$$

Muhammad Arslan Babar

$k_o = 1.0$ (Assume uniform load and power)

$k_m = 1.3$ ($b = 1$, Accurate mountings)

$$\sigma_b = \frac{\cancel{6.13}\ 1.37 \times 10}{1 \times 0.28} \times 1.69 \times 1 \times 1.3$$

~~σ_b = 480.48~~

$$\boxed{\sigma b = 107.496 \text{ psi}}$$

assuming a safety factor for gear = 2.5

$S_n = 2.5 \times 107.496 = 268.74$ psi

$$\boxed{S_n = 0.268 \text{ ksi}}$$

$$S'_n = \frac{S_n}{C_L C_G C_S k_T k_r k_{ms}}$$

$C_L = 1.0$ (Bending)
$C_G = 1.0$ ($P > 5$)
$k_T = 1.0$ ($T < 160°F$)
$k_{ms} = 1.4$ (not idler)
$k_r = 1.0$ (50%)

$C_S = 0.9$ (Assuming range of Hardness = 320 Bhn ~ 120 Bhn)

$\boxed{S'_n = 0.213 \text{ ksi}}$; $S_U = 0.43$ ksi (minimum required)

∴ As such, any commercial material will guarantee that gears does not experience fatigue.

Muhammad Aaslar Babir.

Choose 1010 A, 44 ksi S_u ; Carbon alloy steel for gears, as it is lowest strength refence available

Otherwise, aluminum alloys can be used with $S_u = 12$ ksi

Testing for surface fatigue:

Assuming aluminium alloy is used for gear

$$S_H = S_{fe} C_{Li} \times C_R = 65 \times 0.9 \times 1.25$$

∵ assume 10^8 cycles, Aluminium - Bronze alloy

$$S_H = 73.125 \text{ ksi}$$

$$\sigma_H = C_p \sqrt{\frac{F_t}{b d_p I} k_v k_o k_m}$$

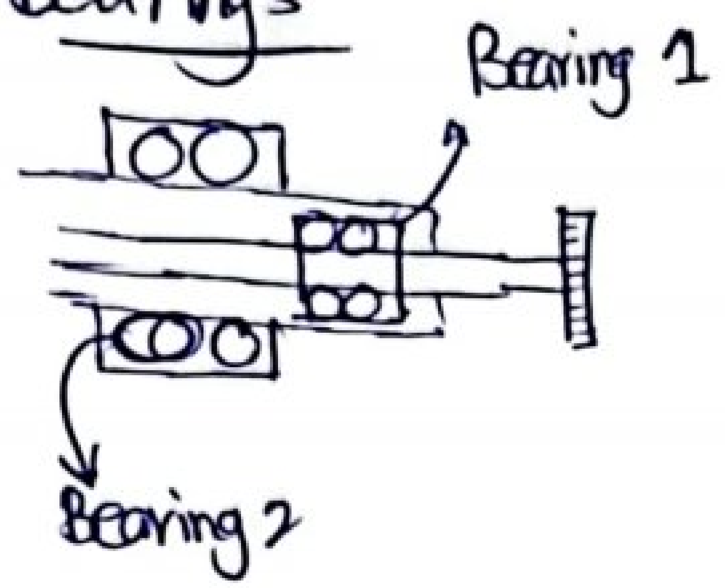
$C_p = 1950 \sqrt{psi}$	$b = 1$ inch	$I = 0.115$	$k_o = 1.0$
$F_t = 1.37$ lb	$d_p = 4$ inch	$k_v = 1.69$	$k_m = 1.3$

$$\sigma_H = 1950 \sqrt{\frac{1.37}{1 \times 4 \times 0.115} \times 1.69 \times 1 \times 1.3} = 4988.05 \text{ psi}$$

$$\eta = \frac{S_H}{\sigma_H} = \frac{73125}{4988.05} = 14.67$$

∵ System is very safe from any sort of fatigue. As power requirement is very low and material properties are very high.

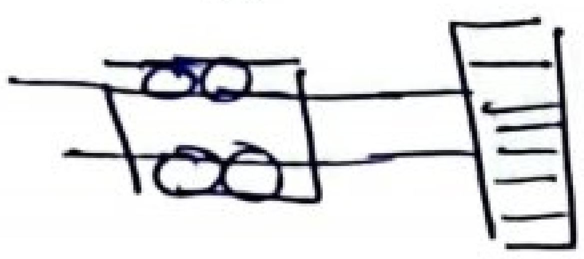
Appendix 2 Bearing Selection

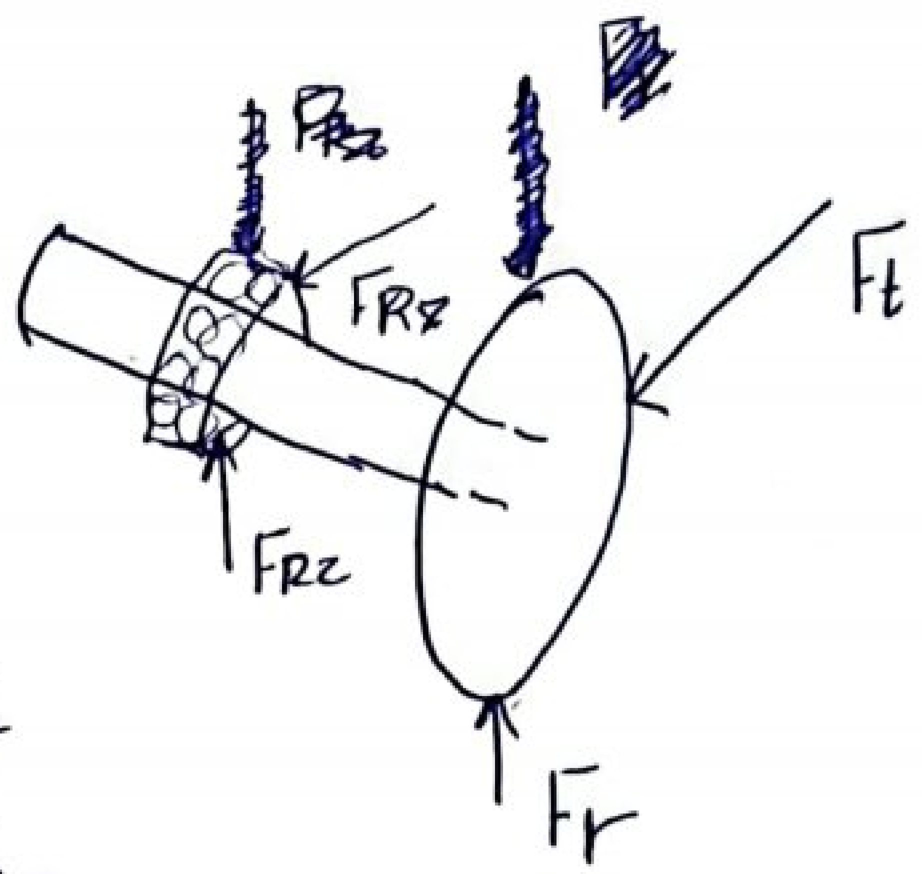


Osama A. S. Shurrab

* So, for bearing 1, 2 and 3 an Angular Bearing would be chosen. in case any thrust Force acting on the bearing that is unknown.

* Bearing 1





$$F_t = 81.26\,N = F_{R1y}$$

$$F_r = F_t \tan(20) = F_{R1z}$$

$$F_r = F_{Rz} = 29.576\,N$$

$$F_{R1} = \sqrt{29.576^2 + 81.26^2} = 86.475\,N$$

- Since, there's no thrust Force acting on the bearing

$$F_{R1} = F_e = 86.48\,N$$

- Reliability 90% (assumed)
- uniform no impact (assumed)
- Bearing working 8 hours For every working day hence, according to Jovinall textbook 30000 hours

table 14.4

(1)

Osama A. S. Shirab

$$C_{req} = k_a F_e \left(\frac{L}{L_R K_r}\right)^{0.3}$$

$$= (1)(86.48)\left(\frac{354.6}{90 \times 1}\right)^{0.3}$$

$$= 130.486 \text{ N} \approx 0.13 \text{ kN}$$

$n = 197$ Rpm

$L_R = 90 \times 10^6$ (assumed)

$k_r = 1$

$k_a = 1$

$L = 30000 \times 60 \times 197$

$= 354.6 \times 10^6$

* Angular bearing As Follows:

100 / 10mm Bore Size / 1.02 Force capacity

200 / 10mm Bore Size / 1.10 Force capacity

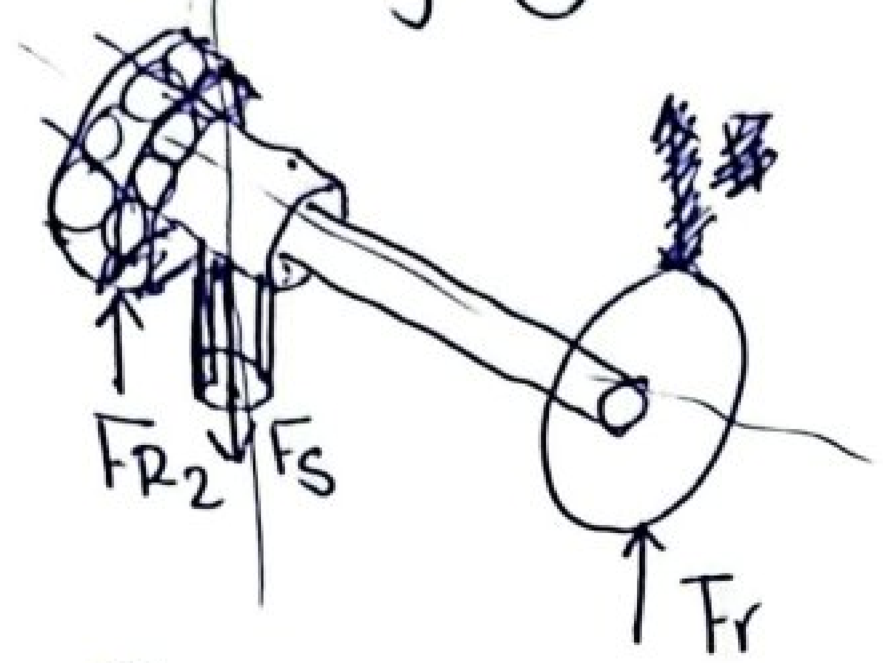
300 / 10mm Bore Size / 1.88 Force capacity

Basic no: 300
Bore size: 10 mm
Force capacity: 1.88

* Selected. Since it provides the highest Force capacity.

Osama A. S. "Shurrab" Case

Bearing 2



$$F_{R_2} = -F_r + F_s$$

$$F_s = \frac{W}{2} = \frac{(215)(9.81)}{2} = 1.054.58$$

$$F_{R_2} = -29.576 + 1054.58 =$$
$$= 1025.004\,N$$

- No thrust Force :-

$$F_e = F_{R_2} = 1025.004N$$

$$C_{req} = (\cancel{k_a})(\cancel{F_e})\left(\frac{L}{L_R\,k_r}\right)^{0.3}$$

$n = 7.5$

$k_a = 1$

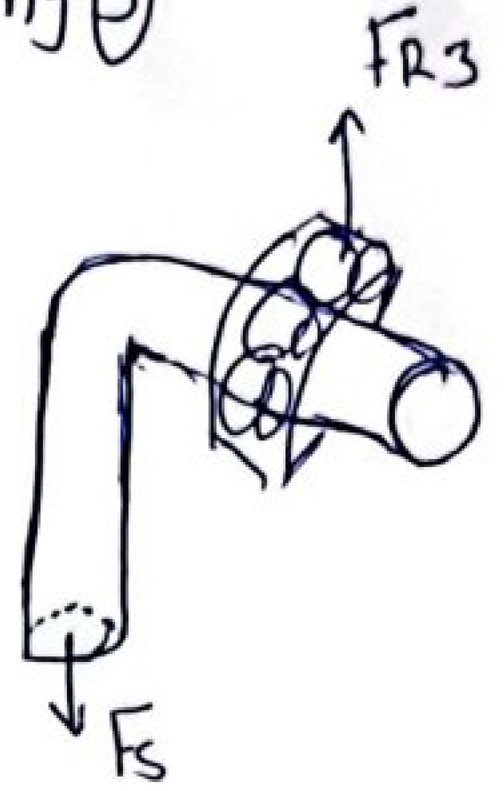
$k_r = 1$

$L_R = 90 \times 10^6$

$L = 4 \times 10^3 \times 7.5 \times 60$
$= 1.8 \times 10^6$

$$C_{req} = (1)(1025.004)\left(\frac{1.8}{90 \times 1}\right)^{0.3}$$
$$= 0.31698\,kN$$

Bearing 3



$$F_s = 1.054.58$$

$$F_{R_3} = F_s = 1054.58\,N$$

- No thrust Force :-

$$F_e = F_{R_3} = 1054.58N$$

$$C_{req} = (k_a)(F_e)\left(\frac{L}{L_R\,k_r}\right)^{0.3}$$

$$C_{req} = (1)(1054.58)\left(\frac{1.8}{90 \times 1}\right)^{0.3}$$
$$= 0.3262\,kN$$

Osama A. S. Shurrab

* Minimum bore size must be 35mm due to design of the barrel support

For both bearing 2 and 3

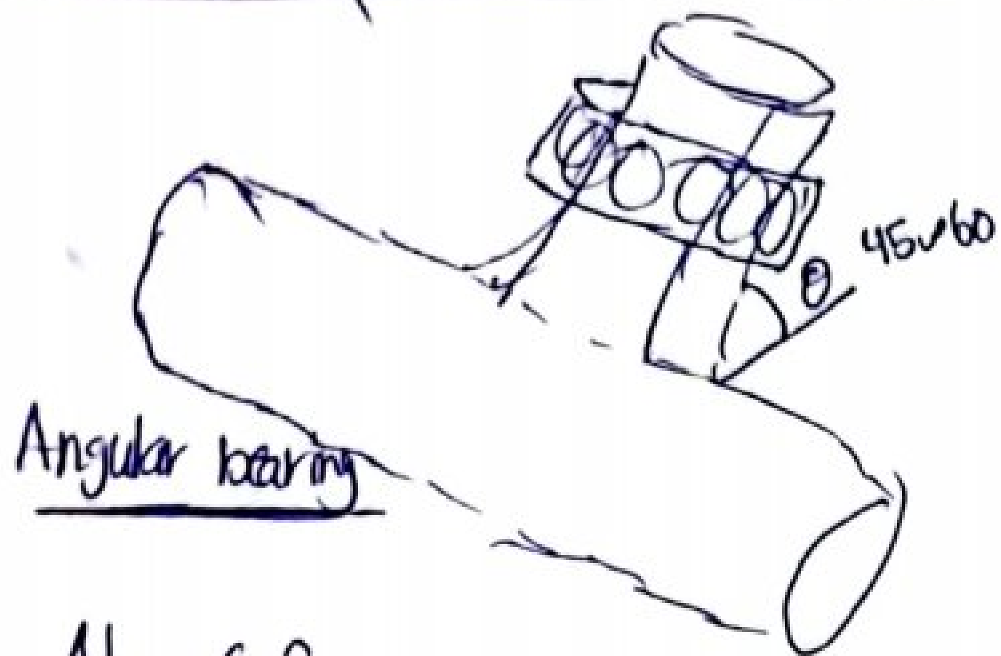
100 / 35 mm Bore / 4.75 kN Force capacity
200 / 35 mm Bore / 8.20 kN Force capacity
300 / 35 mm Bore / 11 kN Force capacity

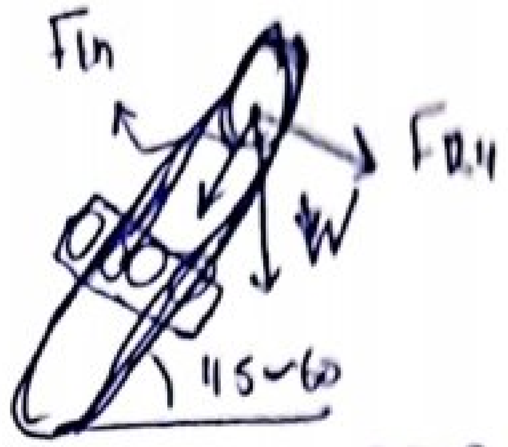
basic No: 307
Bore: 35
OD: 80
Force capacity: 11 kN

This selection was made based on the model the offers higher Force capacity.

Osama A. S. Shurrab

OSE





$W = 215 \times 9.8 = 2109.15\ N$

At 60°	At 45°
$F_{R4} = W \sin(30)$	$F_{R4} = W \sin(45)$
$F_{th} = W \cos(30)$	$F_{th} = W \cos(45)$
$F_{R4} = 2109.15 \sin(30) = 1054.575\ N$	$F_{R4} = 1491.39\ N$
$F_{th} = 2109.15 \cos(30) = 1826.577\ N$	$F_{th} = 1491.39\ N$
$\frac{F_{th}}{F_{R4}} = 1.732$	$\frac{F_{th}}{F_{R4}} = 1$
Angular bearing	
$0.68 < 1.732 < 10$	$0.68 < 1 < 10$
$F_e = F_{R4}\left[1 + 0.870\left(\frac{F_{th}}{F_{R4}} - 0.35\right)\right]$	$F_e = 1491.39\left[1 + 0.870(1 - 0.35)\right]$
$= 1054.575\left[1 + 0.870(1.732 - 0.35)\right]$	$= 2334.77\ N \cong 2.334\ kN$
$= 2.3225\ kN$	

At angle of 45° F_e is the highest so it will be the most critical angle.

Osama A. S. Shurrab

Q5

$$C_{req} = (2334.77)(1)\left(\frac{36}{90 \times 1}\right)^{0.3}$$

$$= 1.7736 kN$$

$L = 30000 \times 20 \times 60$

$= 36 \times 10^6$

$L_{R} = 90 \times 10^6$

$n = 20 rpm$

$k_r = 1$

$k_a = 1$

L00 / 20mm / 2.20 Force capacity

200 / 17mm / 2.20 Force capacity

300 / 10mm / 1.88 Force capacity

basic no :. L04

Bore size :. 20 mm

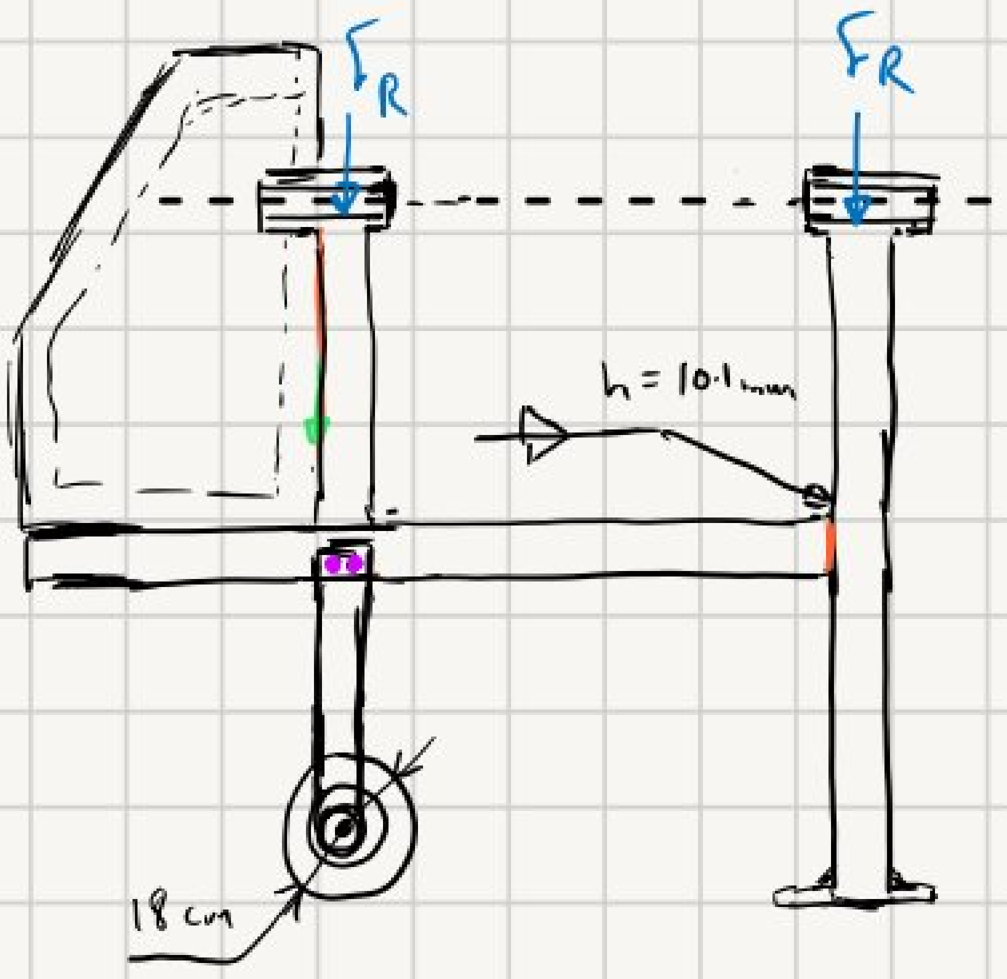
Force capacity :. 2.20 kN

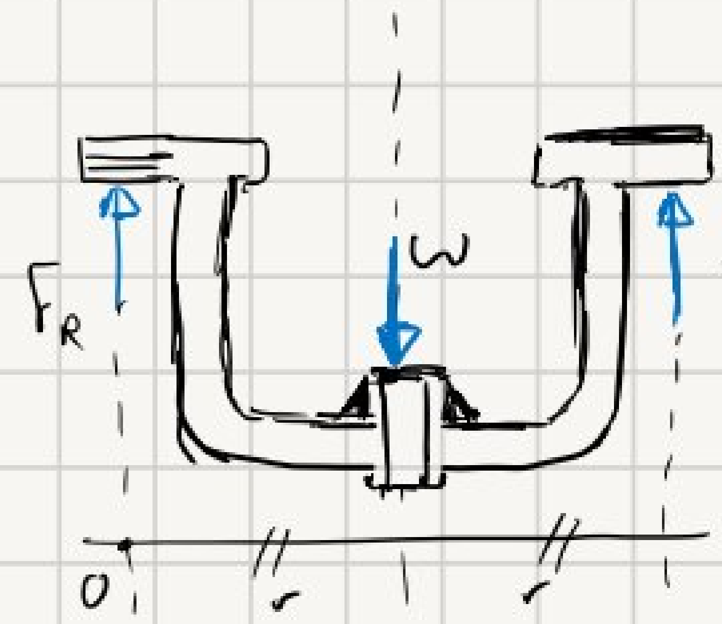
OD :. 42 mm

* We are going to pick the model with the highest Force capacity. So, it's whether model L00 or 200. Since, L00 is cheapest we will take it.

Appendix 3 Structure, Weld and Bolts

Ahmed Mohamedali - A20EM3002 Signature:





$W = (150 + 65) \times 9.81$
$= 2109.15$ N

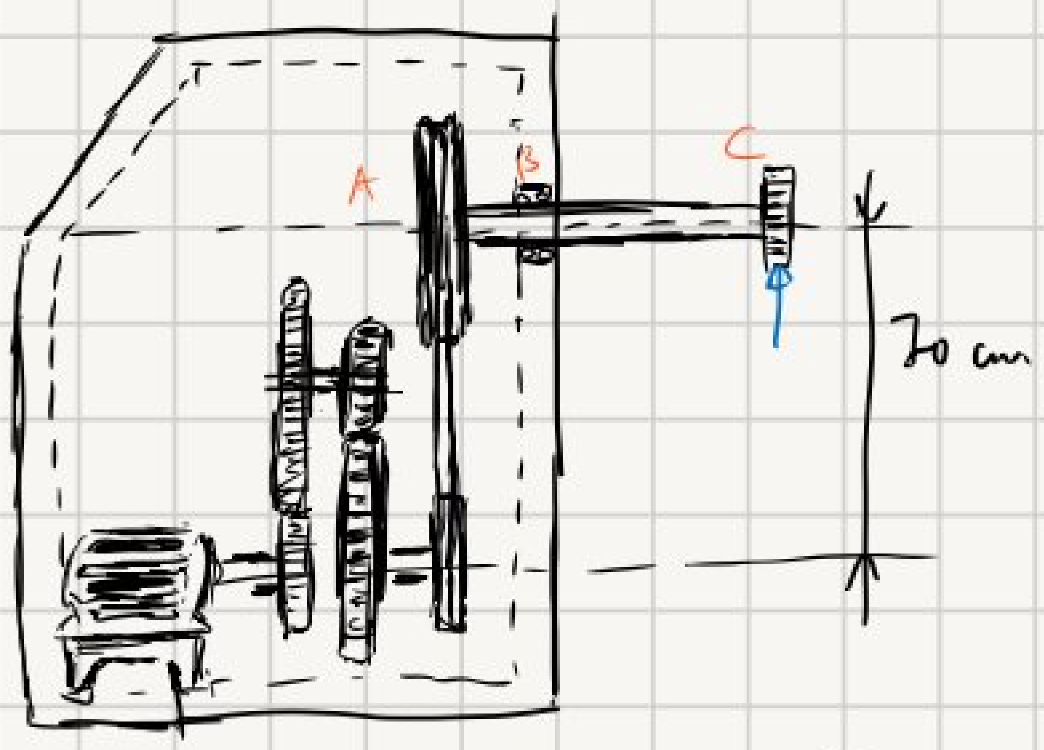
$\circlearrowleft \Sigma M_0 = 0$

$0 = Wr - F_R(2r)$

$F_R = \frac{Wr}{2r}$

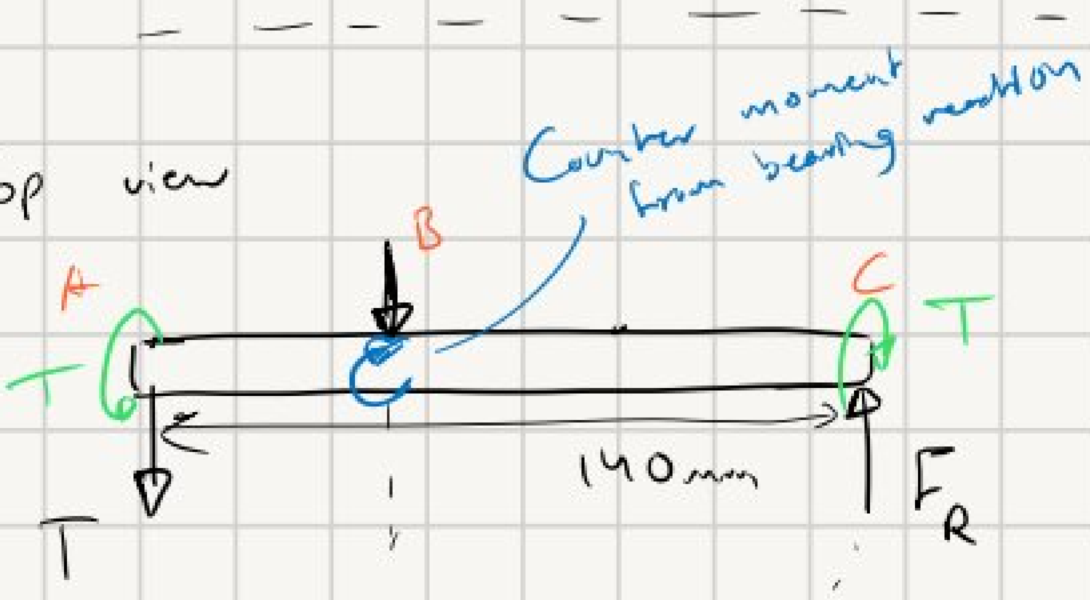
$F_R = \frac{1}{2} W = \frac{1}{2}(2109.15)$

$F_R = 1054.58$ N



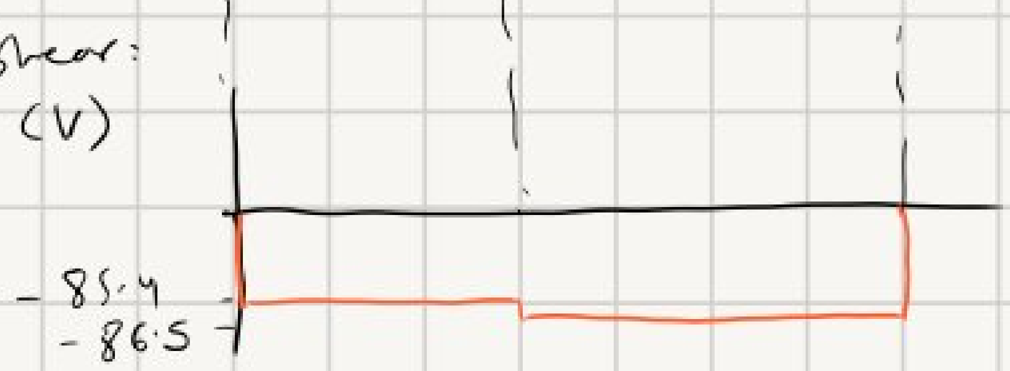
Weight of system housing: [obtained from SolidWorks]:
- → 2 pinions = 2 × (0.2 kg) = 0.4 kg
- → 2 Gears = 75 teeth + 100 teeth = (0.716 + 1.276) = 1.992 ≈ 2 kg
- → 1 belt: $W \times L_d$ = 0.12 kg/m × $\frac{74.3 \text{ inch}}{32.39}$ = 0.238 kg
- → 1 Motor (from Techtop catalogue) = 2.9 kg
- → Casing (approximately) = 2.5 kg

Top view

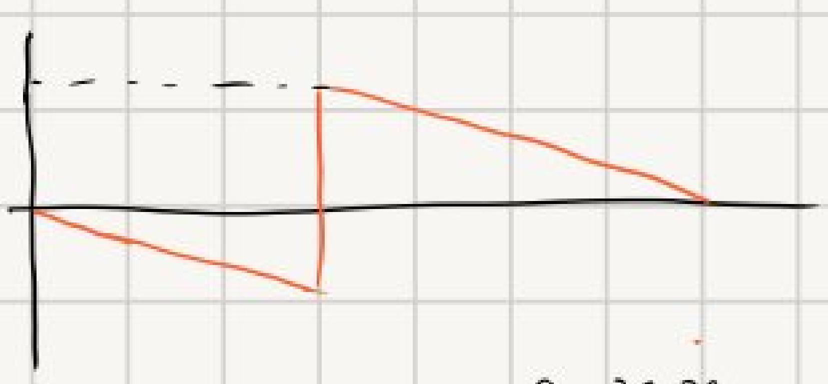


→ $F_R = 86.475$ N referring to bearing calculations.

Shear: (V)

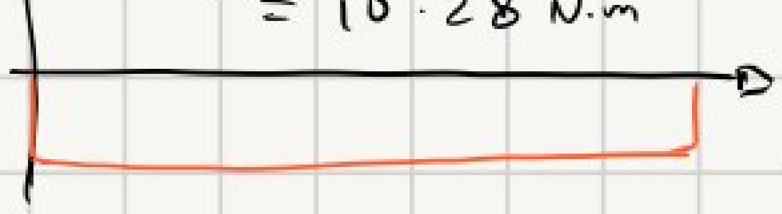


Moment



Torque

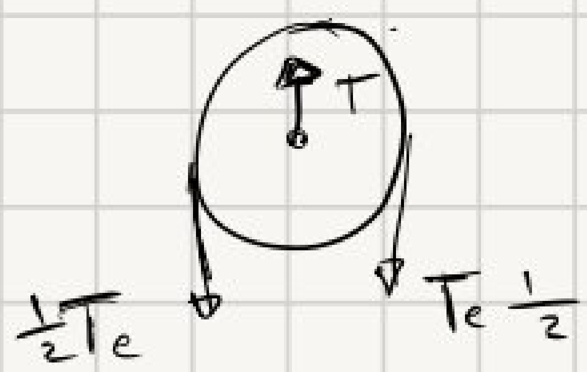
$T_q = 85.4 \times \frac{9 \div 32.39}{2}$
$= 10.28$ N.m



$\tau_{Tq} = \frac{Tc}{J}$

$= \frac{10.28\,(5\times10^{-3})}{\frac{\pi\,(5\times10^{-3})^4}{2}} = 1.23$ MPa

Shaft analysis



$T_e = \frac{33000 \times Hp}{V}$

$V = 1.048$ m/s × 196.9
$= 206.3$ ft/min

$T = T_e = \frac{33000 \times 0.12}{206.3}$

$= 19.2$ lb × 4.448

∴ $T = 85.4$ N

$M_{max} = (86.475)(\frac{1}{2}(140\times10^{-3}))$ ← assumed bearing is in the middle of shaft.

$= 6.053$ Nm

$\sigma_b = \frac{Mc}{I}$

$= \frac{6.053\,(5\times10^{-3})}{\frac{\pi\,(10\times10^{-3})^4}{64}}$

$= 61.66$ MPa

$c = \frac{1}{2} D$ D = bore = 10 mm
$= 5$ mm

$\tau_{shear} = \frac{86.475}{\pi(5\times10^{-3})^2}$
$= 1.1$ MPa

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DET $S.F = \dfrac{S_y}{\sigma'}$ S_y for 1045HR $= 414$ MPa

$S.F = 2.5 = \dfrac{414}{\sigma'_{allow}}$ $\sigma' = \sqrt{\sigma_b^{\,2} + 3(\tau_v + \tau_t)^2}$

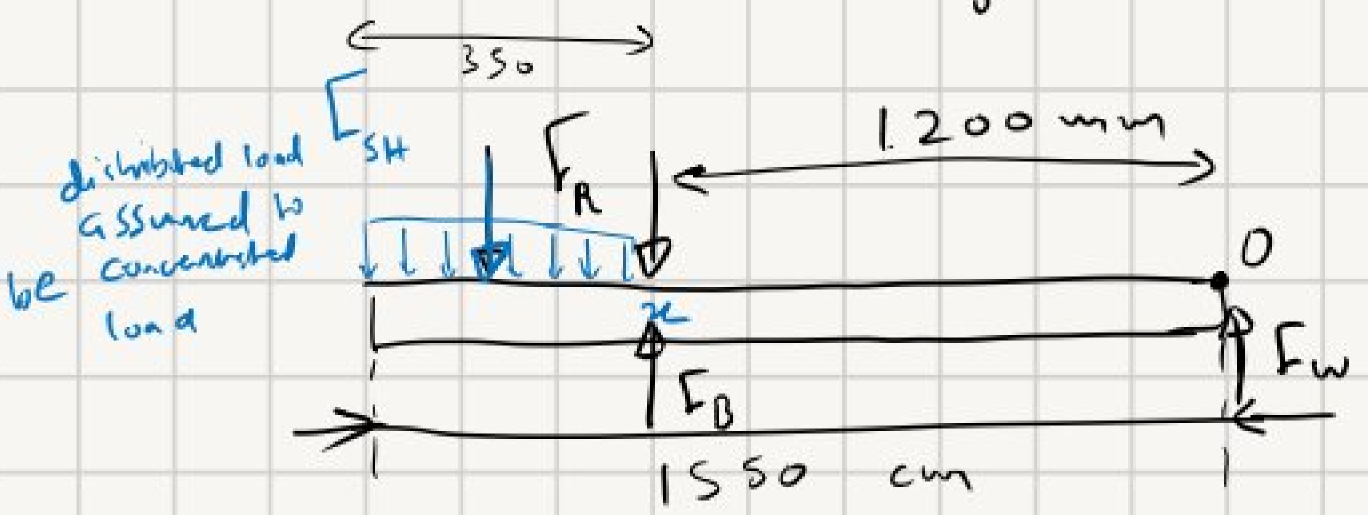
$\sigma'_{allow} = 165.6$ MPa $= \sqrt{61.66^2 + 3(1.1 + 1.23)^2}$

$\sigma' < \sigma'_{allow}$ hence, the 10 mm rod is safe against bending. $= 61.8$ MPa

(at the top)

* Since analyzing the load on the longest 10 mm shaft resulted in "safe"; this allows for the assumption that all other shorter 10 mm shafts will not fail due to bending either.

Resume Structure analysis:



$F_{SH} = 9.81(0.4 + 2 + 0.238 + 2.9 + 2.5)$
$= 78.85$ N

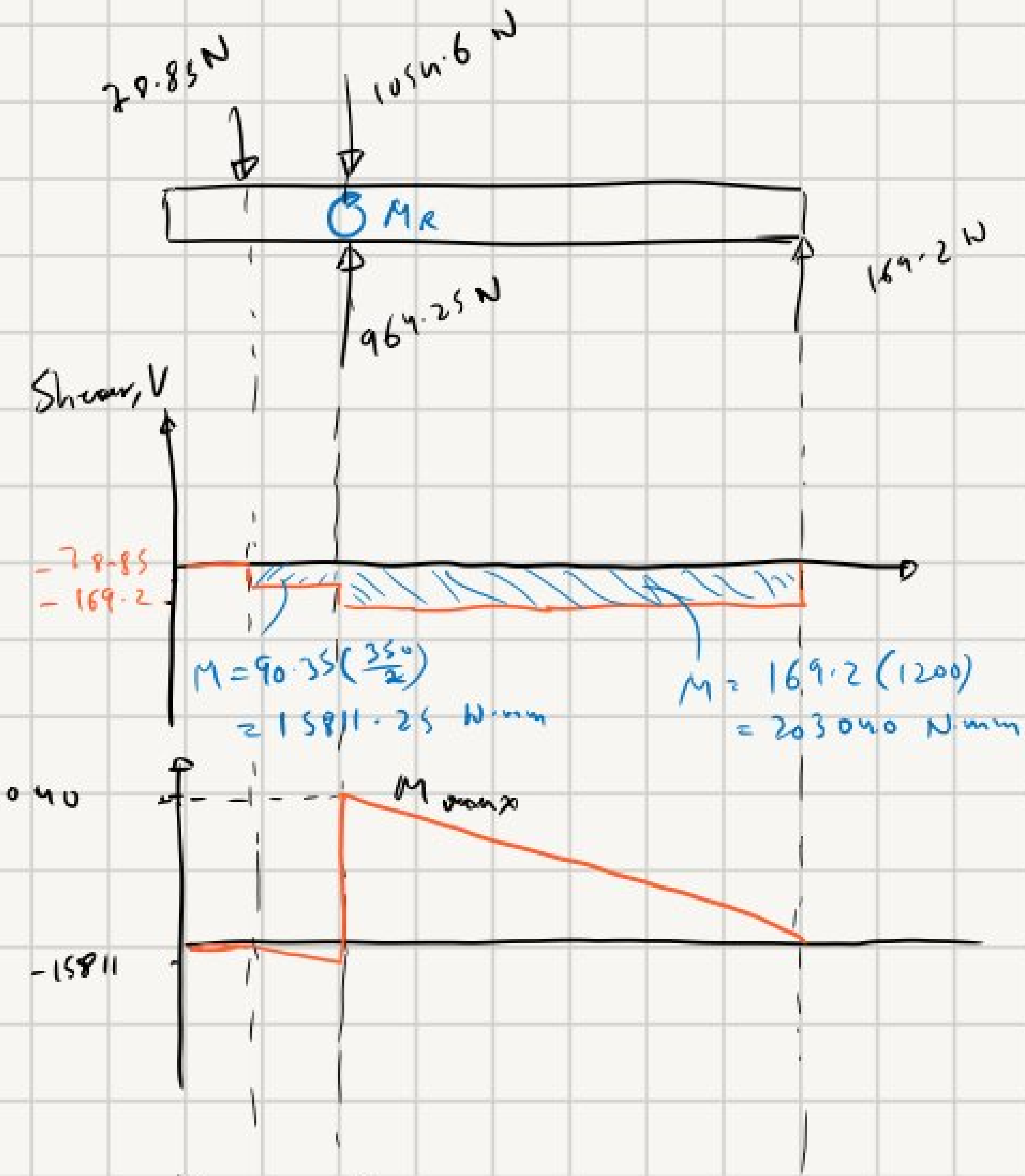
$F_R = \frac{1}{2} W$
$= \frac{1}{2}(2109.15)$
$= 1054.6$ kg

$\circlearrowleft \Sigma M_o = 0$ $F_B = ?$ $F_w = ?$

$F_B(1200) - F_R(1200) + F_{SH}(1200 + \frac{350}{2}) = 0$

$F_B = \dfrac{(1054.6)(1200) - (78.85)(1200 + \frac{350}{2})}{1200}$

$= 964.25$ N $\longrightarrow$ Force at bolts



$\uparrow \Sigma F_y = 0$

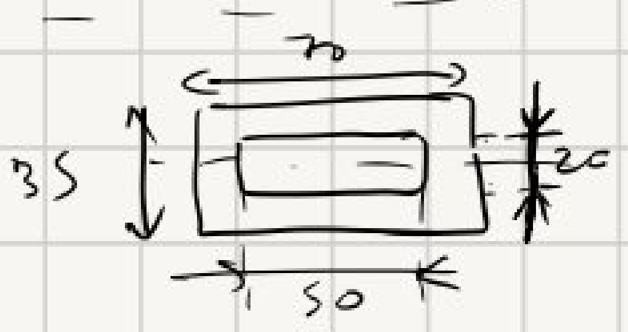
$0 = F_w + F_B - F_{SH} - F_R$

$F_w = -964.25 + 78.85 + 1054.6$
$= 169.2$ N $\longrightarrow$ Force at weld

$\circlearrowleft \Sigma M_x = 0$

$0 = M_R - (78.85)(\frac{350}{2}) - (169.2)(1200)$ $\sigma_b = \dfrac{M_{max}\, c}{I}$

$M_R = 216838.75$ N.mm $I_{xx} = \frac{1}{12}\left((70)(35)^3 - (50)(20)^3\right)$
$= 216.84$ N m $= 216770.8$ mm^4



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Given that the S_y of the shaft material (1045 HR steel) is 414 MPa; it is safe to neglect any possibility of failure due to bending or even perhaps shear stress.

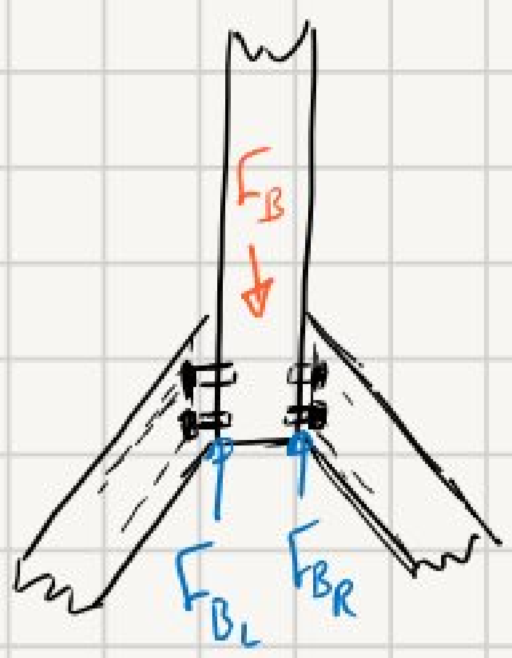
$$\sigma_b = \frac{(216.84)\left(\frac{35}{2}\right)}{216770.8}$$

$$= 0.01751 \text{ MPa}$$

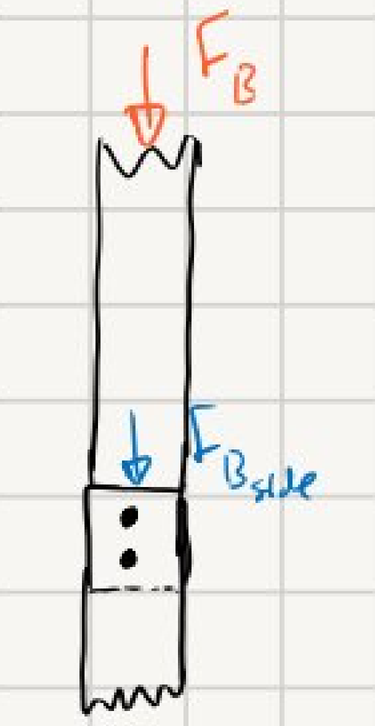
$$= 17.5 \text{ kPa}$$

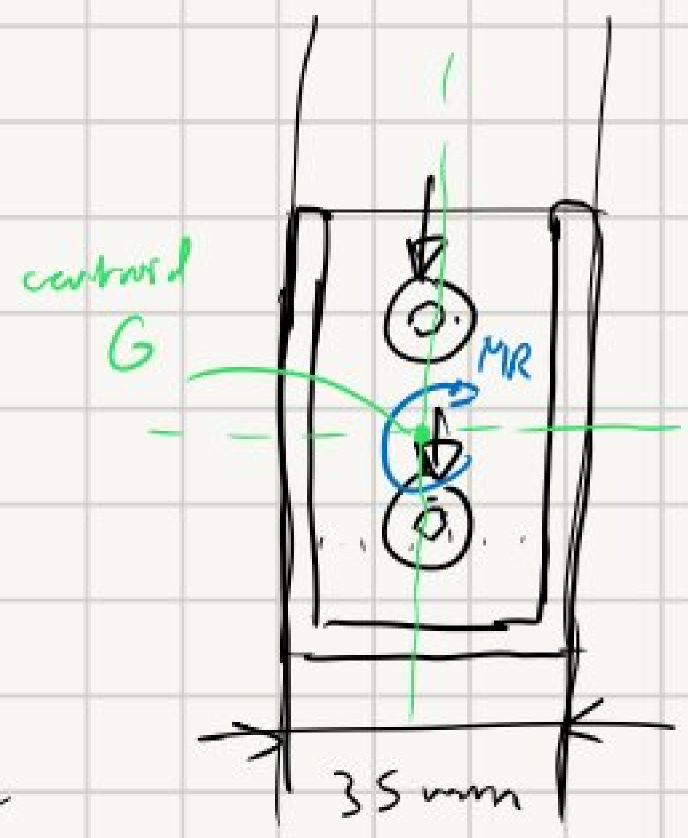
Bolt analysis:

Front view



Side view





Diameter chosen to be 5 mm

Primary load:

Transverse:

$$\tau_v = \frac{V}{A} = \frac{F_B}{A_v}$$

$$\tau_v = \frac{482.125}{\pi(2.5\times10^{-3})^2}$$

$$= 24.55 \text{ MPa}$$

$$F_{B_{side}} = \frac{1}{2} F_B$$

$$= \frac{1}{2}(964.25)$$

$$= 482.125 \text{ N}$$

$$\tau_T = \frac{T}{A_t}$$

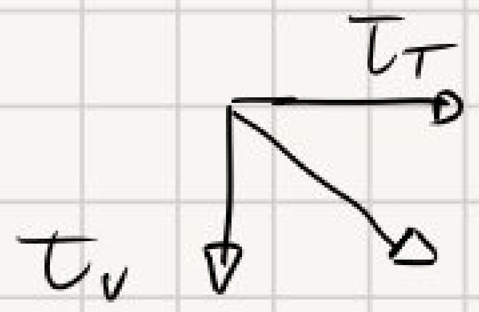
For $d = 5$ mm, from table 10.2:

$A_t = 14.2 \text{ mm}^2$

$T = M_R = 216.84$ N·m

$$\tau_T = \frac{216.84}{14.2}$$

$$= 15.27 \text{ MPa}$$



$$\tau_{resultant} = \sqrt{(15.27)^2 + (24.55)^2}$$

$$= 28.91 \text{ MPa}$$

→ No need to get centroid since (MR) moment is acting at the centre.

$$\therefore \text{ DET S.F} = \frac{S_p}{\sqrt{\sigma_{max}^2 + 3\tau_{max}^2}}$$

$$2.5 = \frac{S_p}{\sqrt{3(28.91)^2}}$$

$$S_p = 125.18 \text{ MPa}$$

→ Hence choose Bolt SAE Class 4.6, $d = 5$ mm since it has S_p of 225 MPa, which cover the stress demand ($125.18 < S_p$).

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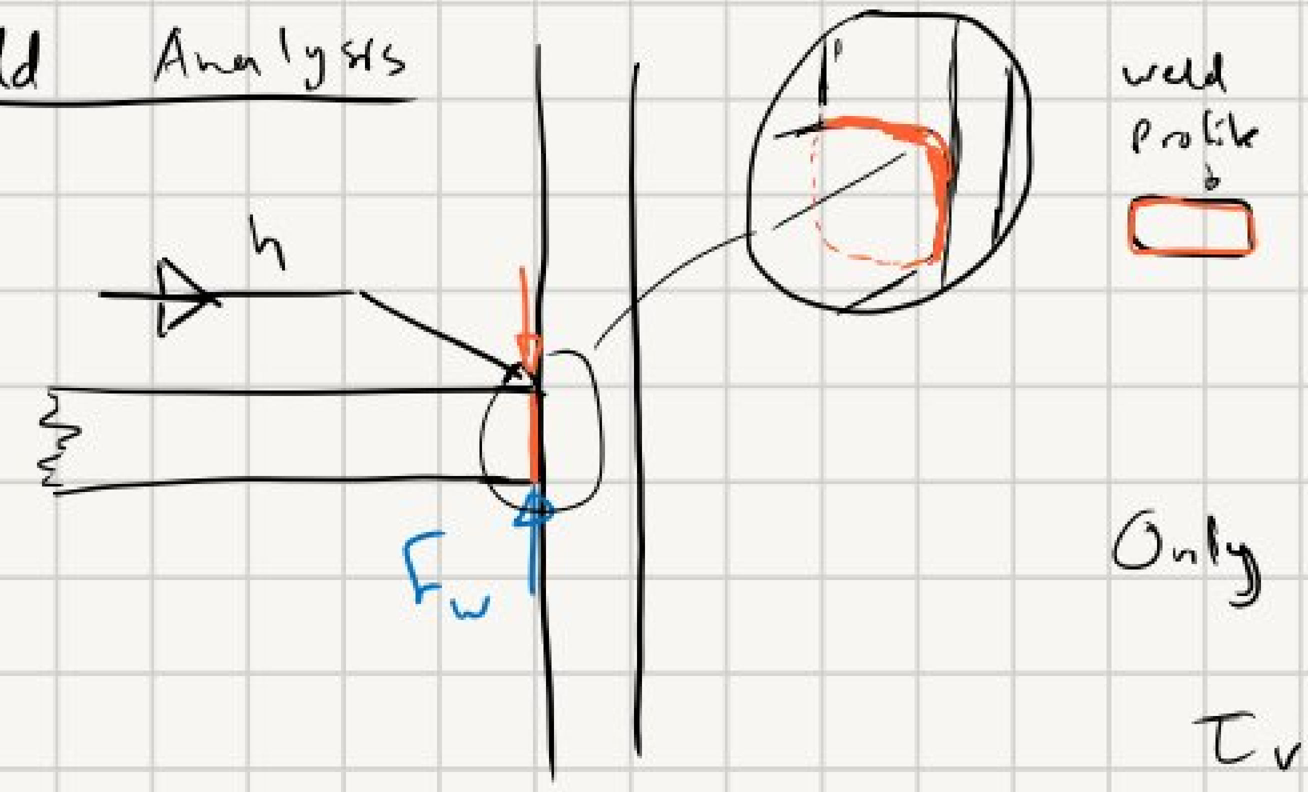
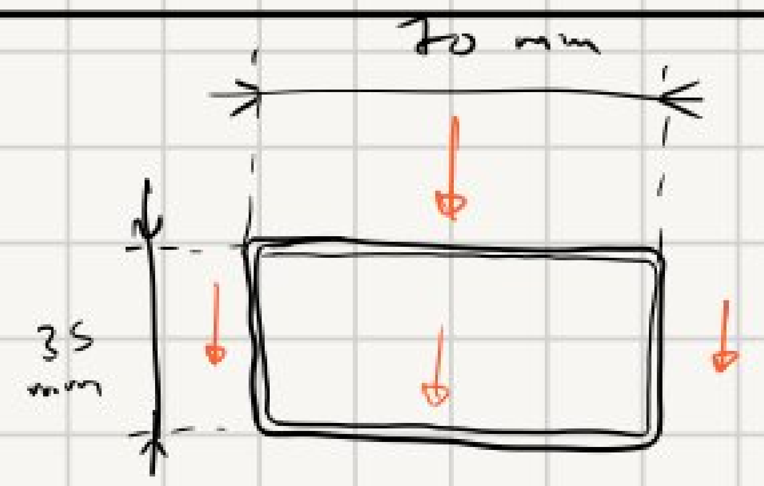
Weld Analysis





$F_w = 169.2$ N

Only Primary Load: Shear

$$\tau_v = \frac{V}{A_w} \qquad A_w = L \times h$$

$\tau_{max} = \tau_{resultant} = \tau_v$ since there is no torsional stress.

$$\begin{aligned} &= L \times h \\ A_w &= (70\times2 + 35\times2)\,h \\ &= 210\,h \quad mm^2 \end{aligned}$$

$$\text{DET}, \quad S.F. = \frac{0.577 S_y}{\tau_{max}}$$

$$\tau_v = \frac{169.2}{210\,h} = 0.8057h \quad MPa$$

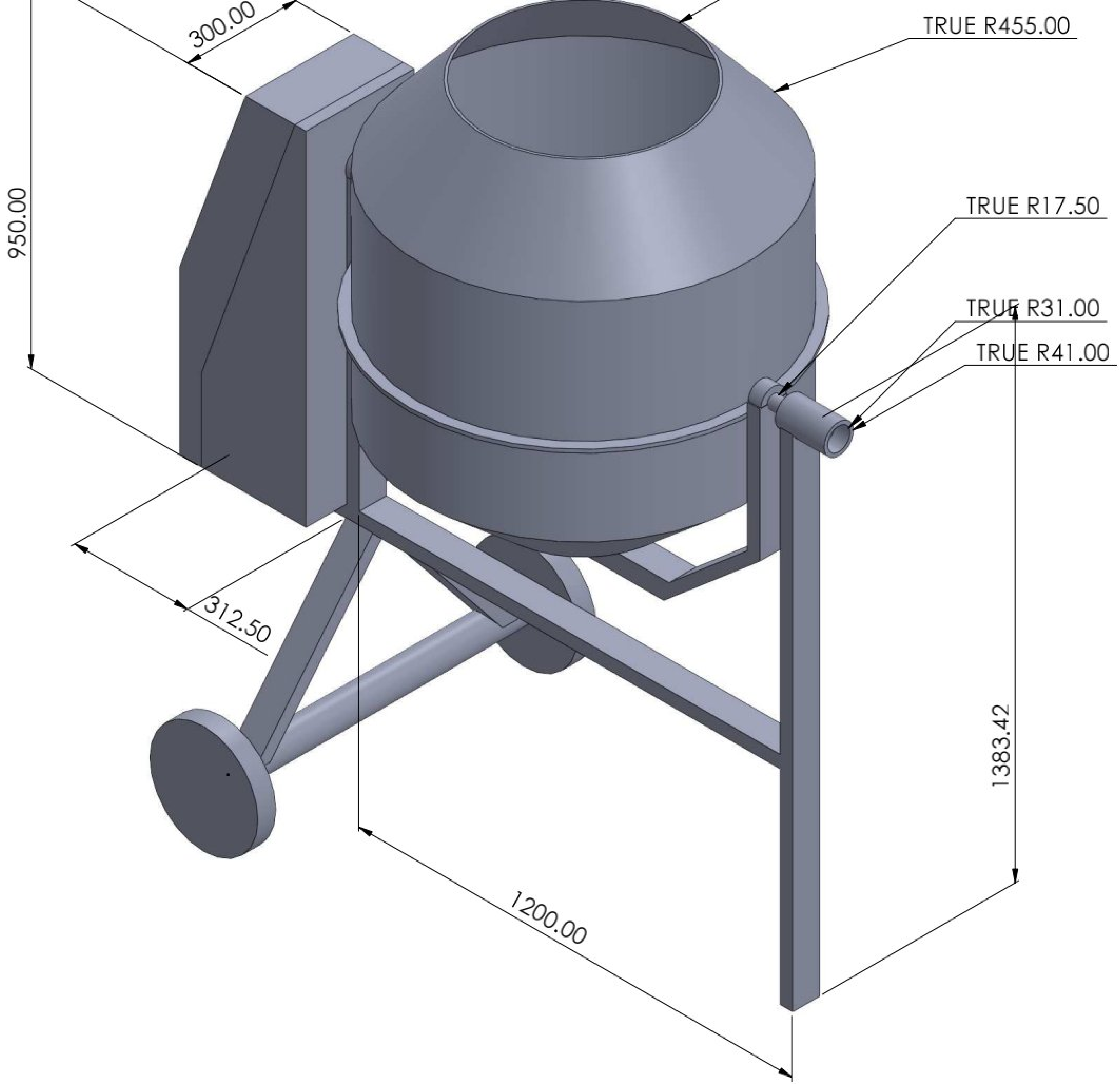
AWS Electrode No. ⇒ E60, hence $S_y = 345$ MPa

$$2.5 = \frac{0.577(345)}{0.8057\,h}$$

Weld thickness, $h = 0.0101$ m

$= 10.1$ mm #

Appendix 4 ISOMETRIC DRAWING OF OVERALL SYSTEM



300.00
TRUE R455.00
950.00
TRUE R17.50
TRUE R31.00
TRUE R41.00
312.50
1383.42
1200.00