THE GERMAN
UNIVERSITY IN CAIRO

CHAIN-ROLLER CONVEYOR

PRESENTED TO
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Submitted by

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I. Abstract

This report presents a comprehensive analysis of the design process for a Roller Conveyor, comprising key components such as the conveyor chassis, motor, single-stage v-belt, single-stage gearbox, rollers connected by chains, and bearing blocks. The primary objective of this design project is to ensure optimal performance by carefully selecting suitable motor specifications, gears, chains, belts, bearings, and shaft diameters through theoretical force analysis. The Roller Conveyor is intended to convey boxes weighing 25kg. The design parameters include a total length of 6000mm, a linear velocity of 0.5m/s, roller diameters ranging from 100 to 150mm, and a box-to-box distance of 200mm. Notably, all calculations were done by hand in order to select suitable individual components. The conveyor was then modeled on Solidworks to confirm the correct analysis.

II. Introduction

In today's fast-paced manufacturing industry, efficient material handling systems play a crucial role in optimizing production processes. To streamline the movement of medical devices within the XYZ factory, we are tasked with designing a chain-roller conveyor. This report outlines the step-by-step process of designing the conveyor, considering the specified requirements and constraints. The roller conveyor we are designing comprises several key components, including a conveyor chassis, motor, single-stage V-Belt, single-stage gearbox, rollers connected by chains, and bearing blocks. The primary objective is to transport boxes with a weight of 25kg and dimensions measuring 300mm x 300mm x 300mm. The conveyor's total length is approximately 6000mm, with a roller center distance of 250mm. The desired linear velocity is 0.5 m/s, and the roller diameter ranges from 100 to 150mm. Furthermore, the box-to-box distance should be 200mm. To begin the design process, we have the freedom to select an appropriate motor from the CHIARAVALLI Motors Catalogue. We will consider input power and RPM to choose a motor that best suits the requirements of our chain-roller conveyor. Moreover, we need to decide where to attach the motor on the conveyor chassis to ensure optimal performance and ease of maintenance. The subsequent step involves designing a V-Belt Drive to transmit power from the motor to the gearbox. Careful consideration will be given to factors such as power transmission efficiency, belt tension, and pulley sizes to achieve an optimal and reliable drive system. Following the V-Belt Drive design, we will proceed with designing a single-stage gearbox. Stress analysis will be conducted to ensure the gearbox can withstand the loads imposed by the V-Belt Drive. The input of the gearbox will be the V-Belt Drive, while the output will be the driving roller, positioned as the first roller in the conveyor. The chain drive system will be employed to connect each roller to its adjacent counterpart, using sprockets and chains. Throughout the report, we will explore various design considerations, such as load-bearing capacity, material selection, and alignment accuracy, to ensure the chain-roller conveyor operates smoothly, efficiently, and reliably. By following a systematic approach and considering the specifications outlined above, we aim to deliver a comprehensive and well-engineered design for the chain-roller conveyor, meeting the unique requirements of the conveyor belt.

III. Methodology

The following methodology outlines a systematic approach for force analysis and the design of a Chain-Roller Conveyor. The process involves a step-by-step progression, as described below:

1. **Motor Selection:** The initial step involves selecting an appropriate motor from the CHIARAVALLI Motors Catalogue that aligns with the specific design requirements of the conveyor system.
2. **V-Belt Selection:**

   2.1 Preliminary Design: Conducting preliminary calculations to initiate the V-Belt selection process, considering factors such as power transmission requirements and operational conditions.

   2.2 Iterative Trials: Performing iterative trials to determine the optimal V-Belt capable of efficiently transmitting power from the motor to the gearbox.
3. **Gearbox Spur Gears Selection:**

   3.1 Preliminary Design: Doing preliminary calculations to start the selection of suitable spur gears for the gearbox, taking into account design parameters such as torque and speed.

   3.2 Material Selection: Choosing appropriate gear materials to ensure effective power transmission from the gearbox to the chains driving the rollers.

   3.3 Check Surface Durability: Verifying the surface durability of the selected gear materials and dimensions, confirming their suitability based on the specified design requirements.

4. **Chain Design from Output Shaft of Gearbox to Shaft of Roller:**

    4.1 Design Steps: Doing chain selection calculations to select a suitable chain capable of effectively transmitting power from the gearbox output shaft to the roller shaft.

    4.2 Chain Design from Roller to Roller:

    4.2.1 Design Steps: Doing chain selection calculations to select an appropriate chain that ensures efficient power transmission between adjacent rollers.

5. **Chain Design from Roller to Roller:**

    5.1 Design Steps: Doing chain selection calculations to select a suitable chain capable of effectively transmitting power from each roller to the next.

6. **Chain and Belt Forces:**

    6.1 Chain Force: Doing chain force calculations to take into consideration while doing bearing selection.

    6.1 Belt Forces (On Motor shaft and Input Shaft of Gearbox): Calculating belt force on both the motor and input shaft to take into consideration while selecting bearing.

7. **Bearing Selection:**

    7.1 Motor Shaft: Doing force analysis on motor shaft and selecting suitable bearing.

    7.2 Input Shaft: Doing force analysis on motor shaft and selecting suitable bearing.

    7.3 Output Shaft: Doing force analysis on output shaft and selecting suitable bearing.

    7.4 Roller: Doing force analysis on rollers and selecting suitable bearing.

8. **Solidworks Model:** Modeling the Roller-Chain Conveyor on Solidworks to obtain a functional 3D model, 2D drawings of each part and a functional motion study of the conveyor.

IV. Results

1 General Notes

- $$NumberOfBoxes = \frac{6000}{300+200} = \frac{6000}{500} = 12Box$$

- $$TotalForce = 12 * 25 * 9.81 \approx 3000N$$

- $$Power = F * V = 3000 * 0.5 = 1500W$$

- $$ActualPower = \frac{1500}{0.95 * 0.95 * 0.95}$$

  Note: 0.95 is due to the 3 power transmission elements, assuming 0.95 efficiency

- Minimum Roller Diameter = 100mm

$$V = W * R$$

$$0.5 = W * (0.05)$$

$$W = 10rad/s$$

$$W = \frac{2\pi n}{60}$$

$$n = 95.5RPM$$

- Maximum Roller Diameter = 150mm

$$V = W * R$$

$$0.5 = W * (0.075)$$

$$W = 6.66 rad/s$$

$$W = \frac{2\pi n}{60}$$

$$n = 63.66 RPM$$

- Motor Chosen: 2200 W , 1400 RPM
- Transmission Ratio (i) Range :

$$14.66 < i < 22$$

- Taking i = 20
  1st Stage (Belts): i = 2
  2nd Stage (Gearbox): i = 5
  3rd Stage (Chain): i = 2
  Final Stage (Roller to roller):i = 1

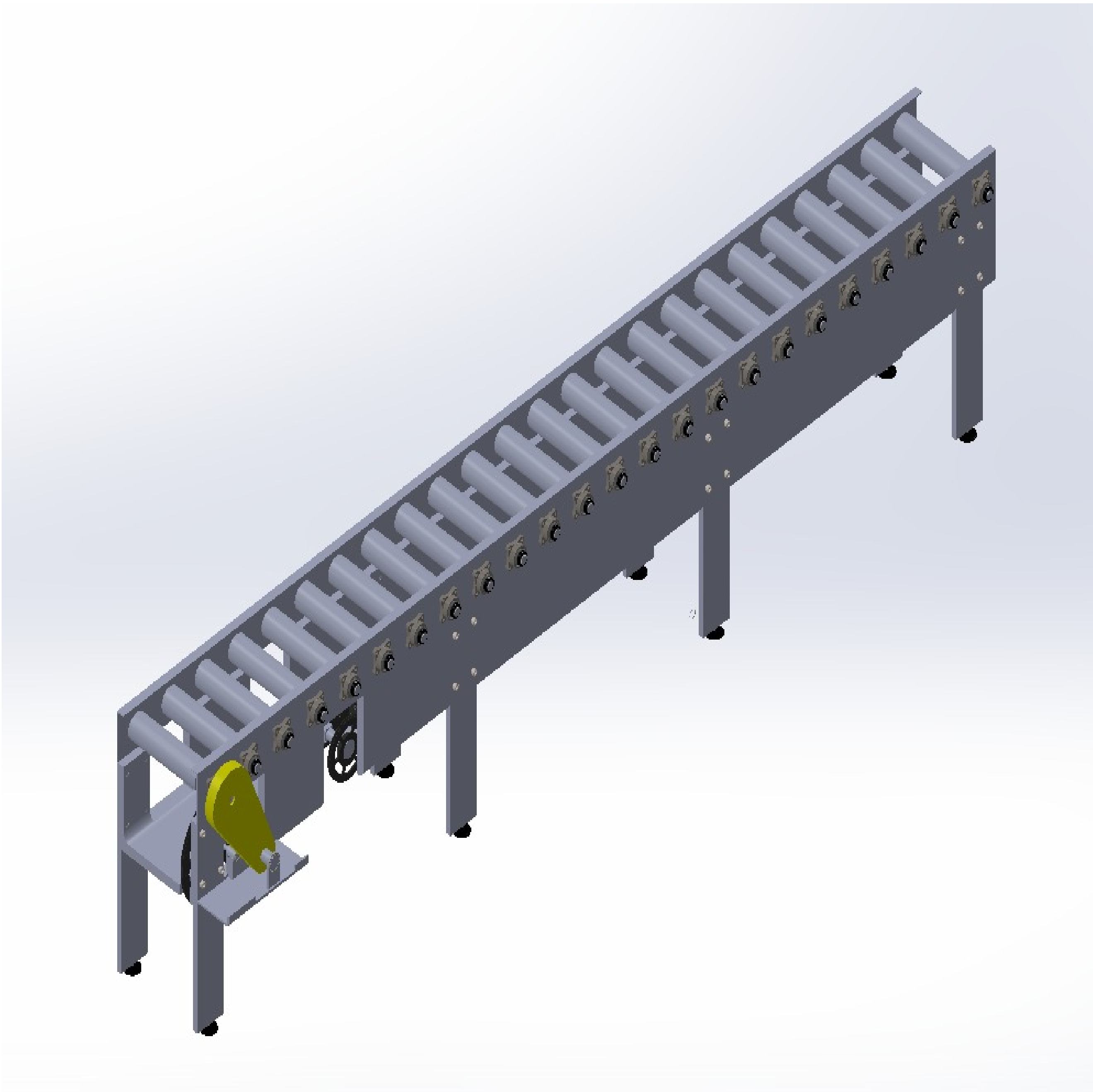


Figure 1: Final Conveyor Design

2 Stage 1: Belts

2.1 Preliminary Design

- Service Factor = 1.3
- $$DesignedPower = H_{\text{des}} = H * SF = 1.3 * 2200 = 2860Watt$$
- Belt Section: Type A
- Transmission ratio: I $_{\text{belt}}$ = 2
- V = 20 m/s (optimum)
- $$d = \frac{60 * V}{\pi n_1} = \frac{60 * 20}{\pi * 1400} = 272.8mm$$
- d $_{\text{tabulated}}$ = 280 mm
- $$D = I * D = 2 * 280 = 560$$

  D $_{\text{tabulated}}$ = 560mm

2.2 1$^{\text{st}}$ trial

1. - $$D < C < 3 * (D + d)$$
     $$560 < C < 2520$$
     Assumption: C = 1000 mm
     Thus, d = 280 mm , D = 560 mm , C = 1000mm , n $_2$ = 700 RPM
   - $$\theta_{\text{s}} = \pi - \frac{D - d}{1000} = \pi - \frac{560 - 280}{1000} = 2.862rad$$
2. - $$L = 2C + \frac{\pi}{2} * (D + d) + \frac{(D - d)^2}{4C} = 3.34m$$
   - L $_{\text{tabulated}}$ = 3287 mm

- L_{in} = 3287 - 36 = 3251 mm

3.
   - $$H_{\text{rating}} = (K_1 * V^{-0.09} - \frac{K_2}{D_{\text{e}}} - K_3 * 10^{-4} * V^2) * V$$
   - K_1 = 0.61 , K_2 = 26.68 , K_3 = 1.04 , D_e = 125
   - H_{rating} = 4.2 HP = 3.145 kW

4.
   - K_L = 1.14
   - K_A = 0.962
   - $$N = \frac{H_{\text{des}}}{K_{\text{L}} * K_{\text{A}} * H_{\text{rat}}} = \frac{2860}{1.14 * 0.962 * 3145} = 0.892 \approx 1$$
   - Conclusion: N = 1 Belt

5.
   - $$4LC' = 8C'^2 + 4C' * \frac{\pi}{2} * (D + d) + (D - d)^2$$
   - $$8C'^2 - 7870 * 12 * C' + 78400 = 0$$
   - C' = 973.7 mm
   - Width of pulley = 2 + f = 2 * 10 = 20 mm
   - $2\beta = 38$

End of Belt Design

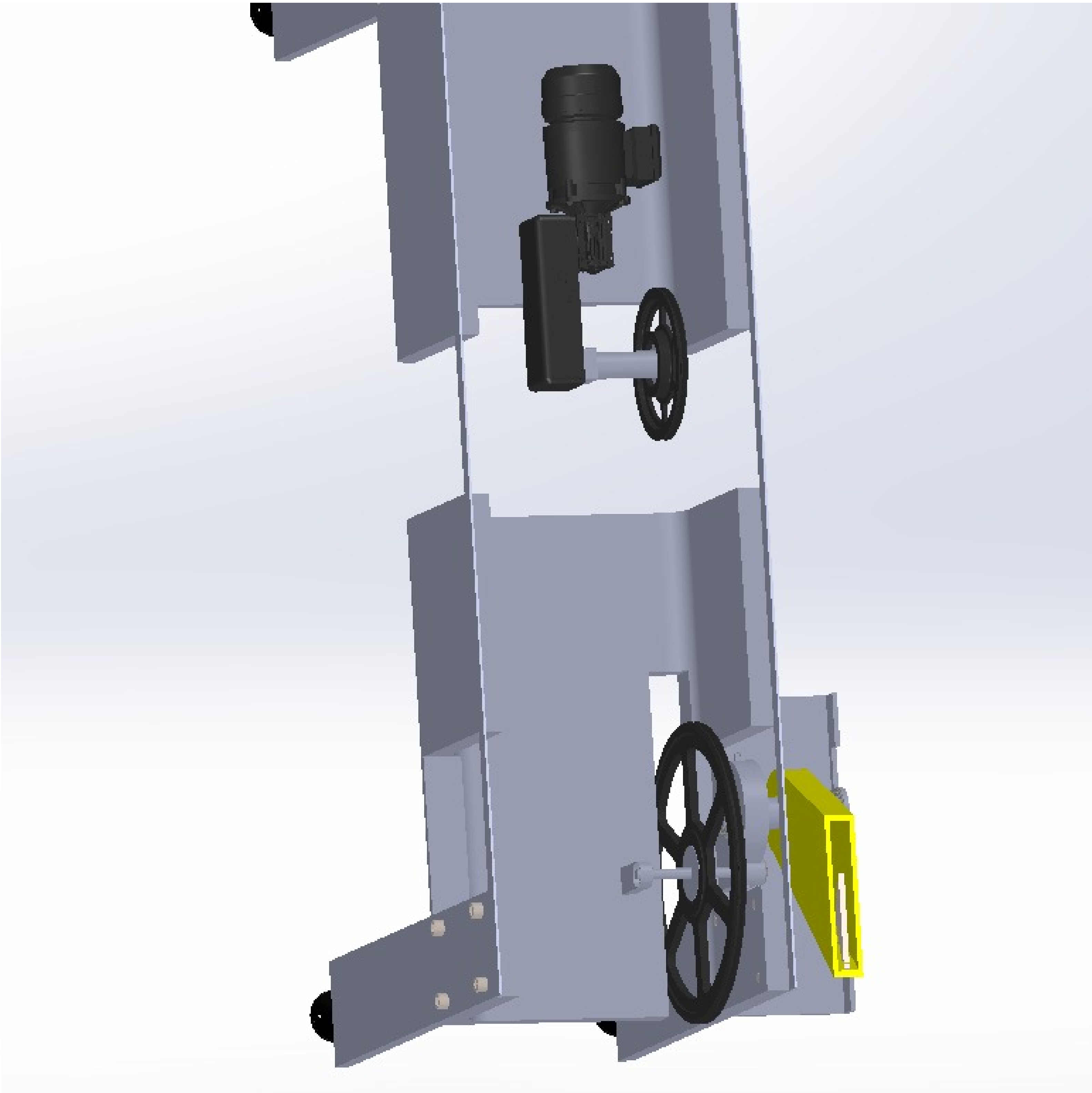


Figure 2: Section of Top View to Further Show Motor and Pulley Alignments.

3 Stage 2: Spur Gears Design

3.1 Preliminary Design

- $H = 2200 * 0.95 = 2090W$ (0.05 loss due to Belt transmission)
- n $_{\text{input}}$ = 700 RPM
- Transmission ratio (i) = 5

1. N $_{\text{pinion}}$ = 18 T
2. module (m) = 2
3. - $d_{\text{pinion}} = mN_1 = 36mm$
   - $N_2 = 5 * 18 = 90T$
   - $D_{\text{gear}} = mN_2 = 180mm$
   - $C = 108mm$
4. $FaceWidth(F) = 4\pi m = 25.1 \approx 25mm$
5. - $$BendingStress\, \sigma = \frac{W_{\text{t}}}{F * K_{\text{v}} * m * Y}$$
   - $$W_{\text{t}} = \frac{H}{V} = \frac{60H}{\pi * n_1 * d_{\text{pinion}}} = \frac{60 * 2090}{\pi * 700 * 36} = 1.58KN$$
   - $V = 1.32m/s$
   - $K_{\text{v}} = \frac{6}{6+V} = 0.820$
   - $Y = 0.29327$
   - $\sigma = 131.4Mpa$
   - $\sigma_{\text{des}} = 4\sigma = 525.6Mpa$

3.2 Material Selection

- Material Chosen: Steel OQT 1000
- $S_{\text{yield}} = 1050Mpa, S_{\text{ult}} = 1160, HB = 341$

6. - $Se^{'} = \frac{S_{\text{ult}}}{2} = 580Mpa$
   - $Ka = 0.66$, $Kb = Kd = Ke = 1$, $Kc = 0.814$ (assuming 0.99 reliability)

- $$Se = Ka * Kb * Kc * Kd * Ke * Se^{'} = 311.6Mpa$$
- $$\sigma_{\mathrm{actual}} = \frac{W_{\mathrm{t}}}{K_{\mathrm{v}} * FmJ}$$
- $Q_{\mathrm{v}} = 6$, $K_{\mathrm{v}} = 0.85$, $J = 0.33$, $W_{\mathrm{t}} = 1.58KN$
- $\sigma_{\mathrm{actual}} = 112.7Mpa$
- $$n_{\mathrm{G}} = \frac{S_{\mathrm{e}}}{\sigma_{\mathrm{actual}}} = \frac{311.6}{112.7} = 2.76$$
- $$n = \frac{n_{\mathrm{G}}}{Ko * Km}$$

  $Ko = 1.25, Km = 1.3$
- $n = 1.7$ Ok Safe

3.3 Check Surface Durability

7.
- $S_{\mathrm{c}} = 2.76HB - 70 = 2.76(341) - 70 = 871.16Mpa$
- $$S_{\mathrm{H}} = \frac{C_{\mathrm{L}} * C_{\mathrm{H}}}{C_{\mathrm{T}} * C_{\mathrm{R}}} * S_{\mathrm{c}}$$

  $C_{\mathrm{L}} = 1.3$, $C_{\mathrm{R}} = 0.8$
- $S_{\mathrm{H}} = 1415.6Mpa$
- $$\sigma_{\mathrm{H}} = C_{\mathrm{p}} * \sqrt{\frac{W_{\mathrm{t}}}{C_{\mathrm{v}}FdI}}$$

  $C_{\mathrm{p}} = 191$, $W_{\mathrm{t}} = 1.58KN$, $C_{\mathrm{v}} = 0.85$, $F = 25mm$, $d = 36mm$, $I = 0.11$
- $\sigma_{\mathrm{H}} = 828.7Mpa$
- $$S_{\mathrm{H}} = \sqrt{Ko * Km * n} * \sigma_{\mathrm{H}}$$

  $n = 1.8$, Safe

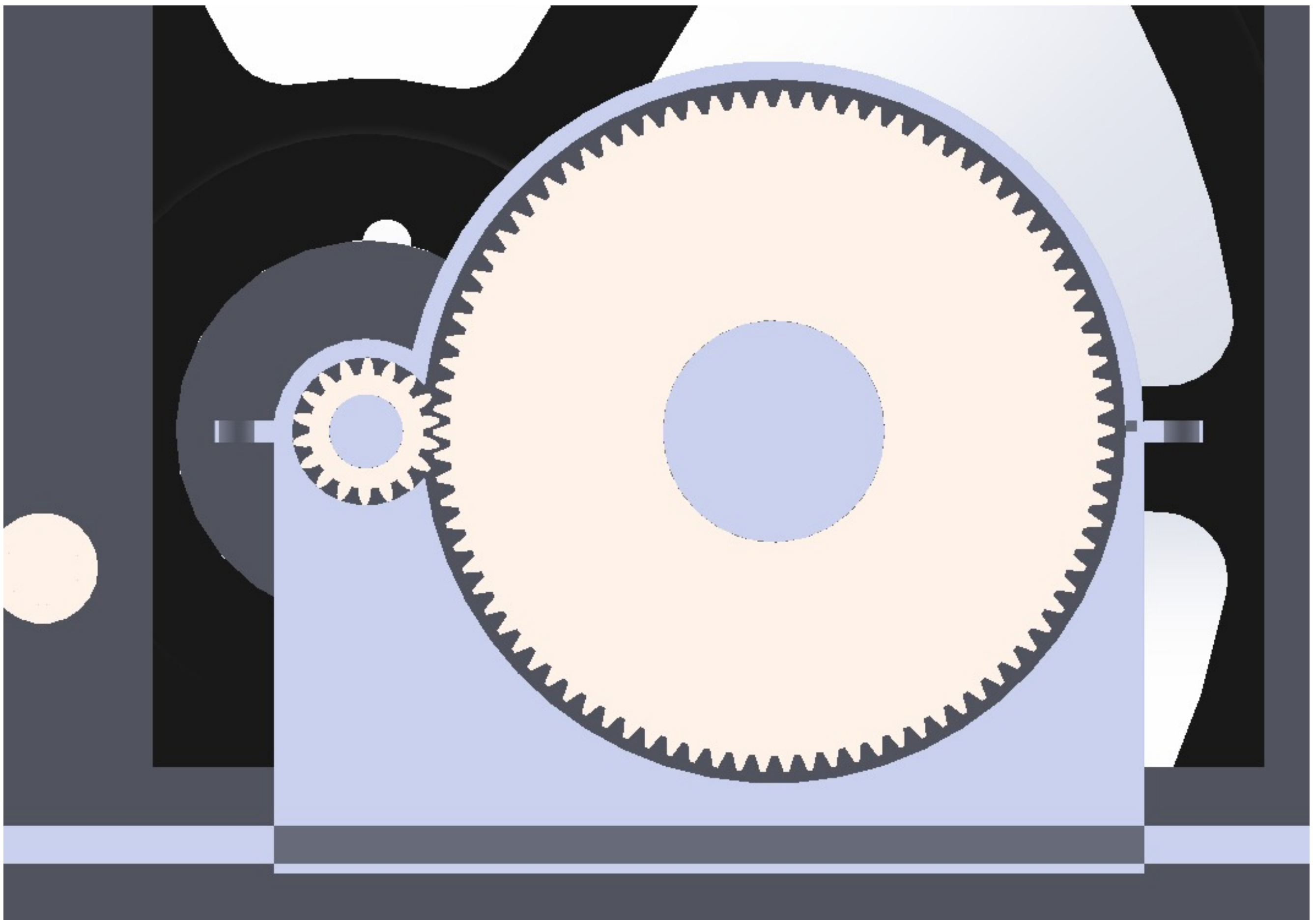


Figure 3: Close Up on Gearbox Showing Gear Reduction and Teeth Ratio

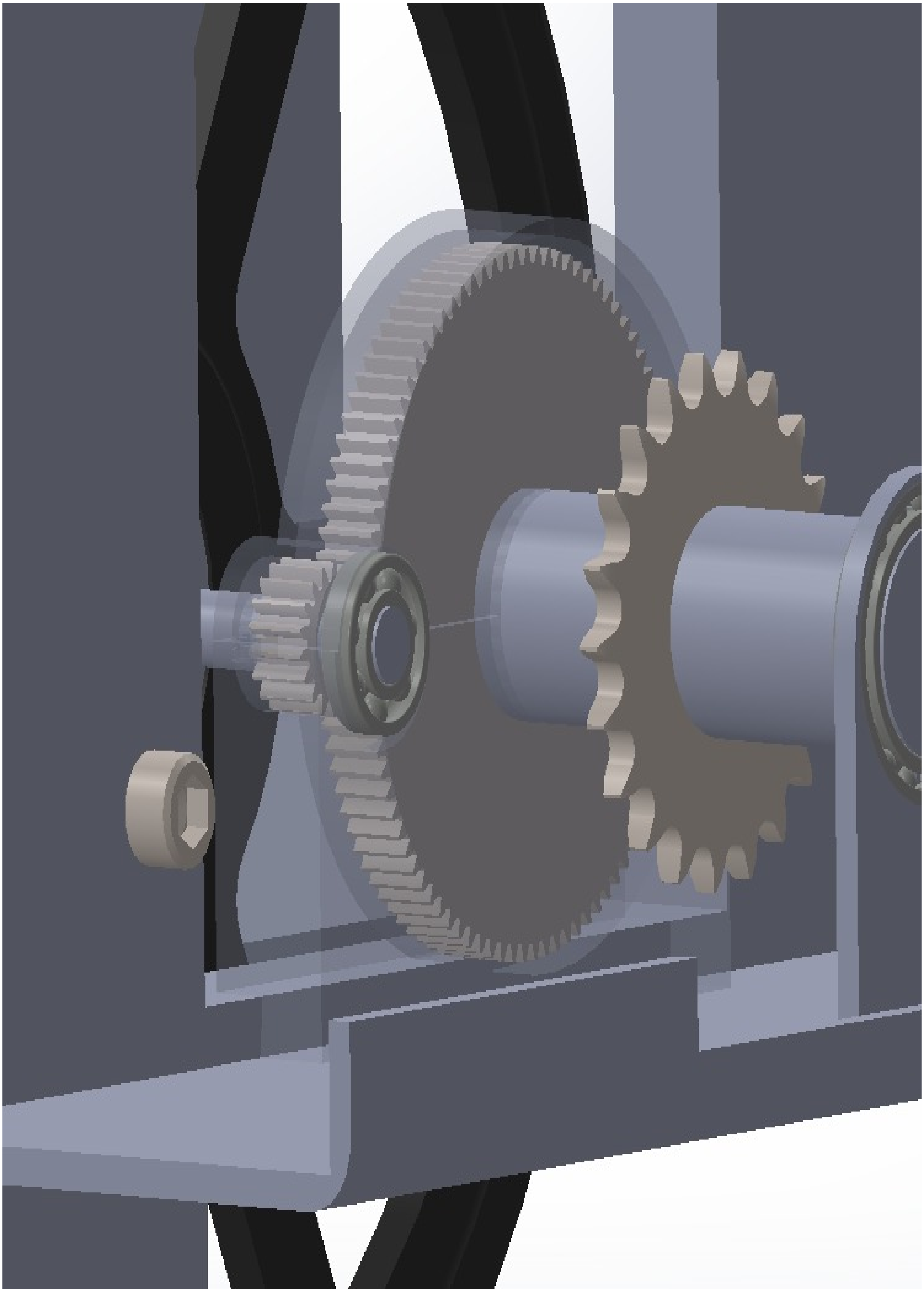


Figure 4: Gearbox (Hidden Under C-Clamp and Sprocket so That It is Away from Machine Edges)

4 Stage 3: Chain Design (from output shaft of gearbox to shaft of roller)

4.0.1 General Notes

- Transmission ratio (i) = 2
- n $_{\text{input}}$ = 140 RPM
- $H = 2090 * 0.95 = 1985.5W$ (0.05 loss due to gear transmission)

4.1 Design Steps

1.
   - $Z_1(driving) = 21T$
   - $Number of strands = 1$, $K_{\text{L}} = 1.0$
   - Electric Motor, Moderate shocks application : $K_{\text{s}} = 1.3$
   - $K_2 = 1.26$ (as Z_1 = 21)
   - $H_{\text{rating}} = \frac{1985.5*1.3}{1*1.26} = 2048.5W \approx 2KW$

2.
   - Trial of choosing 12A as Chain Type
   - By interpolating using equation

$$\frac{3.4 - 1.74}{3.4 - H} = \frac{200 - 100}{200 - 140}$$

   $H = 2.4KW$ for 140 RPM
   - Therefore, 12A is suitable for the desired application

3.
   - Using Table 14.1 , pitch of 12A = 19.05 mm , $p = 19.05mm$
   - $D_1 = \frac{p}{sin(\frac{180}{Z_1})} = \frac{19.05}{sin(\frac{180}{21})} \approx 127.8mm$
   - $Z_2 = Z_1 * \frac{n_1}{n_2} = Z_1 * i = 21 * 2 = 42T$
   - $D_2 = \frac{19.05}{sin(\frac{180}{42})} \approx 255mm$

4.
   - $\alpha = 40p = 40(19.05) = 762mm$
   -
$$L_{\text{n}} = 2(\frac{\alpha}{p}) + (\frac{Z_1 + Z_2}{2}) + (\frac{Z_2 - Z_1}{2\pi})^2(\frac{p}{\alpha}) = 111.8 \approx 112\, links$$

5.
   - Recalculate α
   $L_{\text{n}} - \frac{Z_1+Z_2}{2} = 112 - \frac{21+42}{2} = 80.5$

- $\alpha = \frac{19.05}{4}(80.5 + \sqrt{80.5^2 - 8(\frac{42-21}{2\pi})^2}) = 764.1mm$
  End Of Chain Design

5 Stage 4: Chain Design (roller to roller)

5.0.1 General Notes

- Transmission ratio (i) = 1
- n $_{\text{input}}$ = 70 RPM
- $H = 1985.5 * 0.95 = 1886.225W$ (0.05 loss due to last chain transmission, assume no loss between the chains transmitting the load between rollers)

5.1 Design Steps

1. - $Z_1(driving) = 21T$
   - $Numberofstrands = 1$, $K_{\text{L}} = 1.0$
   - Electric Motor, Moderate shocks application : $K_{\text{s}} = 1.3$
   - $K_2 = 1.26$ (as $Z_1 = 21$)
   - $H_{\text{rating}} = \frac{1886.225*1.3}{1*1.26} = 1950W \approx 1.95KW$
2. - Trial of choosing 16A as Chain Type
   - By interpolating using equation

   $$\frac{4.03 - 2.06}{100 - 50} = \frac{4.03 - H}{100 - 70}$$

   $H = 2.85KW$ for 70 RPM
   - Therefore, 16A is suitable for the desired application
3. - Using Table 14.1 , pitch of 16A = 25.4 mm , $p = 25.4mm$
   - $D_1 = \frac{p}{sin(\frac{180}{Z_1})} = \frac{25.4}{sin(\frac{180}{21})} = 170.4 \approx 170mm$
   - $Z_2 = Z_1 * \frac{n_1}{n_2} = Z_1 * i = 21 * 1 = 21T$
   - $D_2 = D_1 = 170mm$
4. - $\alpha = 40p = 40(25.4) = 1016mm$
   - $$L_{\text{n}} = 2(\frac{\alpha}{p}) + (\frac{Z_1 + Z_2}{2}) + (\frac{Z_2 - Z_1}{2\pi})^2(\frac{p}{\alpha}) = 2(40) + 21 = 101links$$

5.
- Recalculate α
  $L_{\mathrm{n}} - \frac{Z_1+Z_2}{2} = 101 - 21 = 80$
- $\alpha = \frac{25.4}{4}(80 + \sqrt{80^2}) = 1016mm$

End Of Chain Design

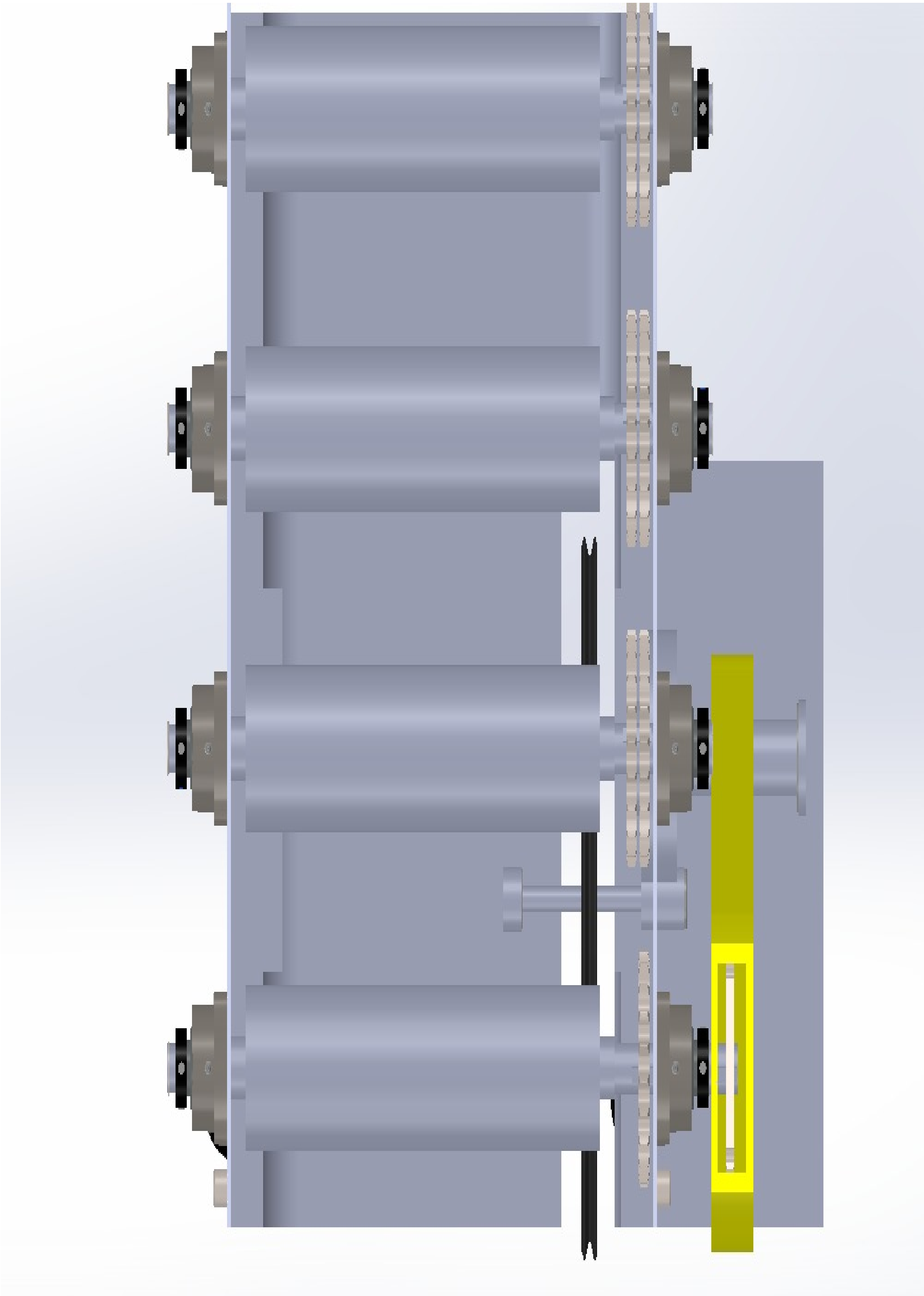


Figure 5: Close Up on Top View Showing Sprocket Trains

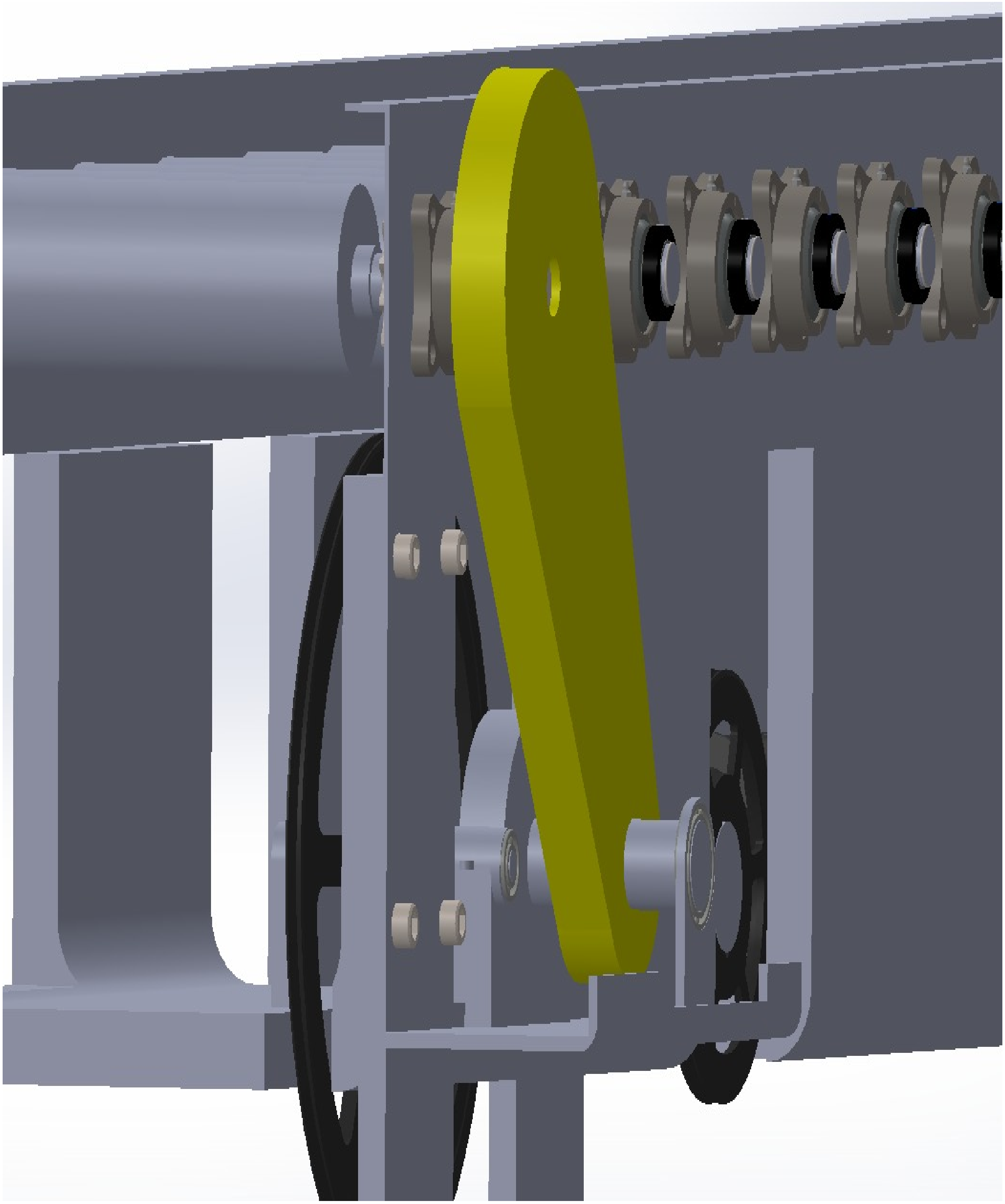


Figure 6: Protective Cover on the Chain Transmitting Power from Gearbox to Roller.

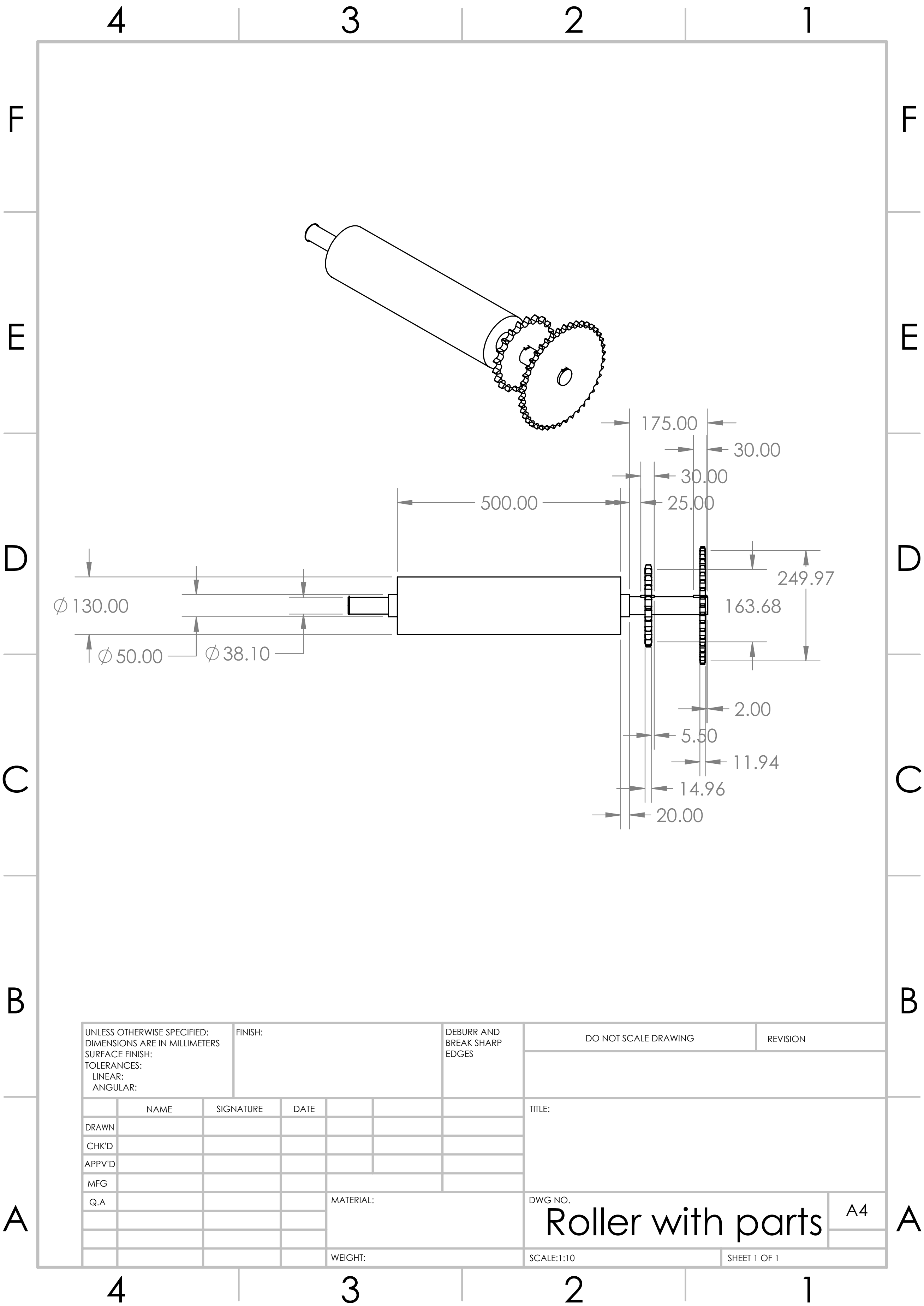


Figure 7: Main Roller Which Receives Power from Chain and Transmits to the Rest of the Conveyor

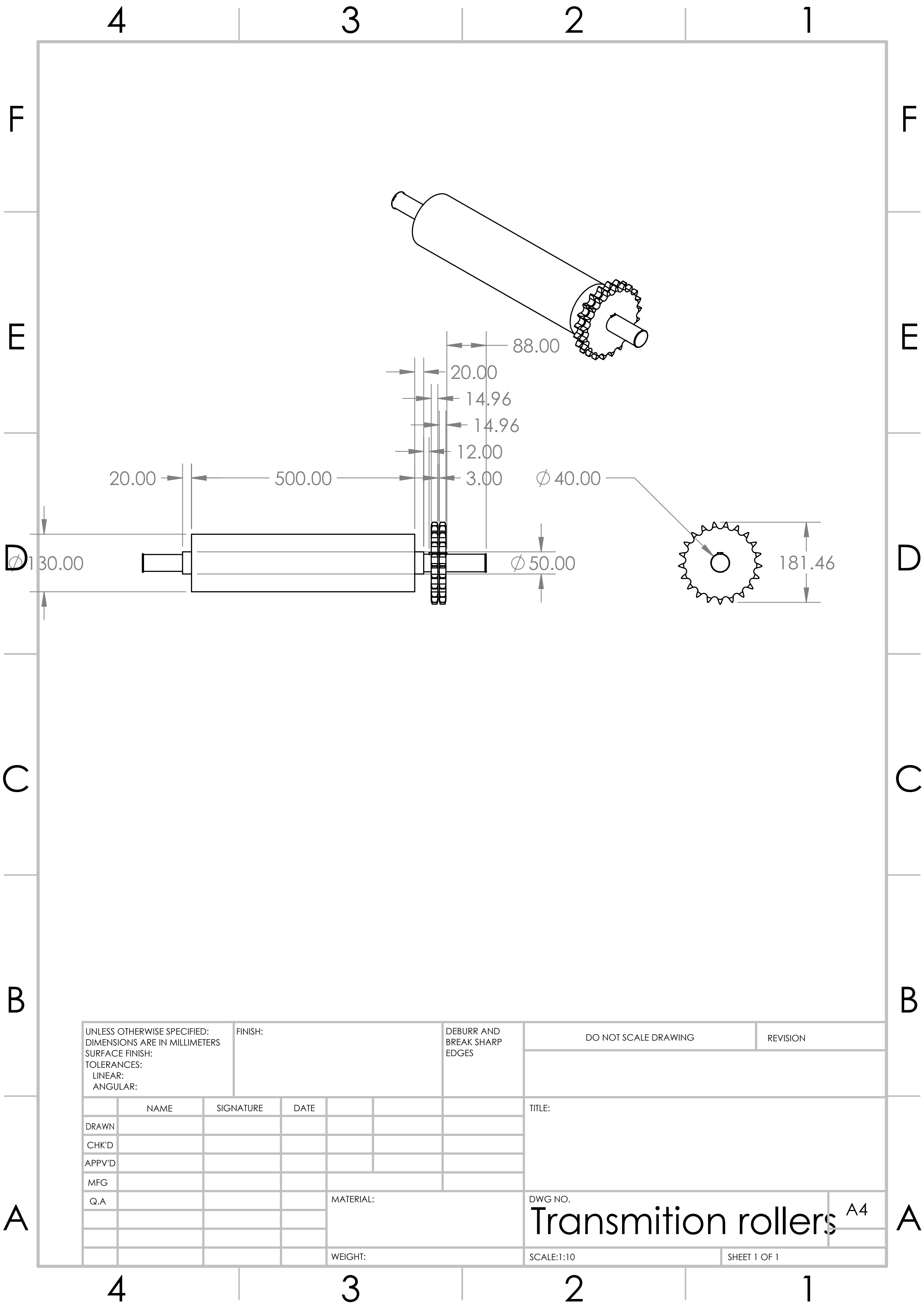


Figure 8: Transmitting roller Which is on Every Shaft Except the Shaft Receiving Power by Means of Chains

6 Analyzing Chain and Belt Forces

6.1 Chain Forces

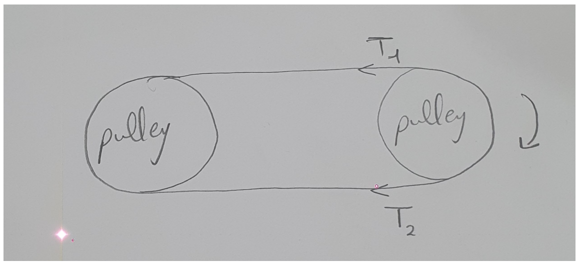
- $$H_{\text{shaft carrying boxes}} = F_{\text{t}} * V$$
- $\text{H}_{\text{shaft carrying boxes}}$ = 1886.225 Watt
- N $_{\text{shaft carrying boxes}}$ = 70 RPM
- $V = W * R = \frac{\pi dn}{60}$

- For large sprocket (D = 255mm) which is used to transmit the power from the gearbox to the rolling shaft carrying the boxes.
  $H = F_{\text{t}} * V$
  $1886.225 = F_{\text{t}} * \frac{\pi * 255 * 10^{-3} * 70}{60}$
  $F_{\text{t}} = 2018 \approx 2KN$

- For other sprocket (D = 170mm) which is used to transmit the power from the roller to the roller on another shaft.
  $H = F_{\text{t}} * V$
  $1886.225 = F_{\text{t}} * \frac{\pi * 170 * 10^{-3} * 70}{60}$
  $F_{\text{t}} = 3027 \approx 3KN$

6.2 Belt Forces (On Motor shaft and Input Shaft of Gearbox)

Important note: The wrap angle was large but θ_s turned out to be 4 degrees which is very small. Accordingly, vertical forces due to belt were considered negligible and only horizontal forces were considered. The diameters of pulleys on Motor shaft and input shaft of gearbox can be taken from previouss results



- n $_{\text{belt}}$ = Variable according to shaft RPM
- $$H = (T_1 - T_2) * W * Radiusofpulley$$
- $T_1 = 2T_2$

6.2.1 Belt Forces for input shaft of gearbox

$n_{\text{input shaft of gearbox}} = 700RPM$

$Radiusofpulley = \frac{280}{2} = 140mm$

$W = \frac{2\pi n}{60} = \frac{2\pi * 700}{60} = 73.3rad/s$

$H = 2090 = T_2 * 73.3$

$T_2 = 203.66N$

$T_1 = 407.3N$

Total Force of Belt $_{\text{in case of Gearbox input shaft}}$=$T_1 + T_2 = 610.98 \approx 611N$

6.2.2 Belt Forces on Motor Shaft

$n_{\text{input shaft of gearbox}} = 1400RPM$

$Radius of pulley = \frac{560}{2} = 280mm$

$W = \frac{2\pi n}{60} = \frac{2\pi * 1400}{60} = 146.6rad/s$

$H = 2200 = T_2 * 146.6$

$T_2 = 53.59N$

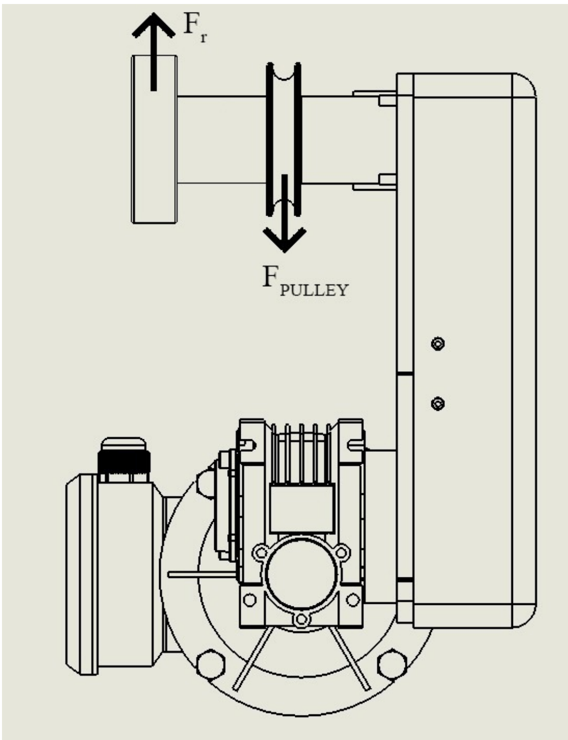
$T_1 = 107.18N$

$TotalForceofBelt_{\text{in case of Motor Shaft}} = T_1 + T_2 = 160.77 \approx 161N$

End Of Belt forces

7 Bearing Selection (4 times for 4 different shafts)

7.1 Bearing Selection for Motor Shaft (only has pulley mounted on it)



- $$F_{\text{pulley due to belt}} = T_1 + T_2 = F_{\text{r}} = 161N$$
- Assuming Diameter of Shaft = 20mm (if motor shaft is not equal to this, a suitable coupler can be used)
- Choosing Bearing with Designation: 61804 as 1^{st} trial
- $C_{\text{ tabulated}}$ = 4.03 KN
- $C_{0\text{tabulated}}$ = 2.32 KN
- $n_{\text{tabulated}}$ = 28 000 RPM
- Driving RPM (n) = 1400 RPM
- L = 25000

- $P = F_{\mathrm{r}} = 161N$
- $P^{'} = K_{\mathrm{sf}} * P = 1.2 * 161 = 193.2 \approx 193N$
- $C_{\mathrm{calculated}} = P^{'}(\frac{60*L*n}{10^6})^{1/3} = 2474 \approx 2.5KN$
- $C_{\mathrm{calculated}} < C_{\mathrm{tabulated}}$, Ok Safe
- $\mathrm{P}_0 = \mathrm{F}_{\mathrm{r}} = 161\mathrm{N}$
- $\mathrm{S}_0 = 1.5$ (Moderate Shocks)
- $C_{0\mathrm{calculated}} = S_0 * P_0 = 241.5 \approx 242$
- $242 < C_{0\mathrm{tabulated}}(2.32KN)$ Ok Safe
- $n < n_{\mathrm{tabulated}}$ Ok Safe

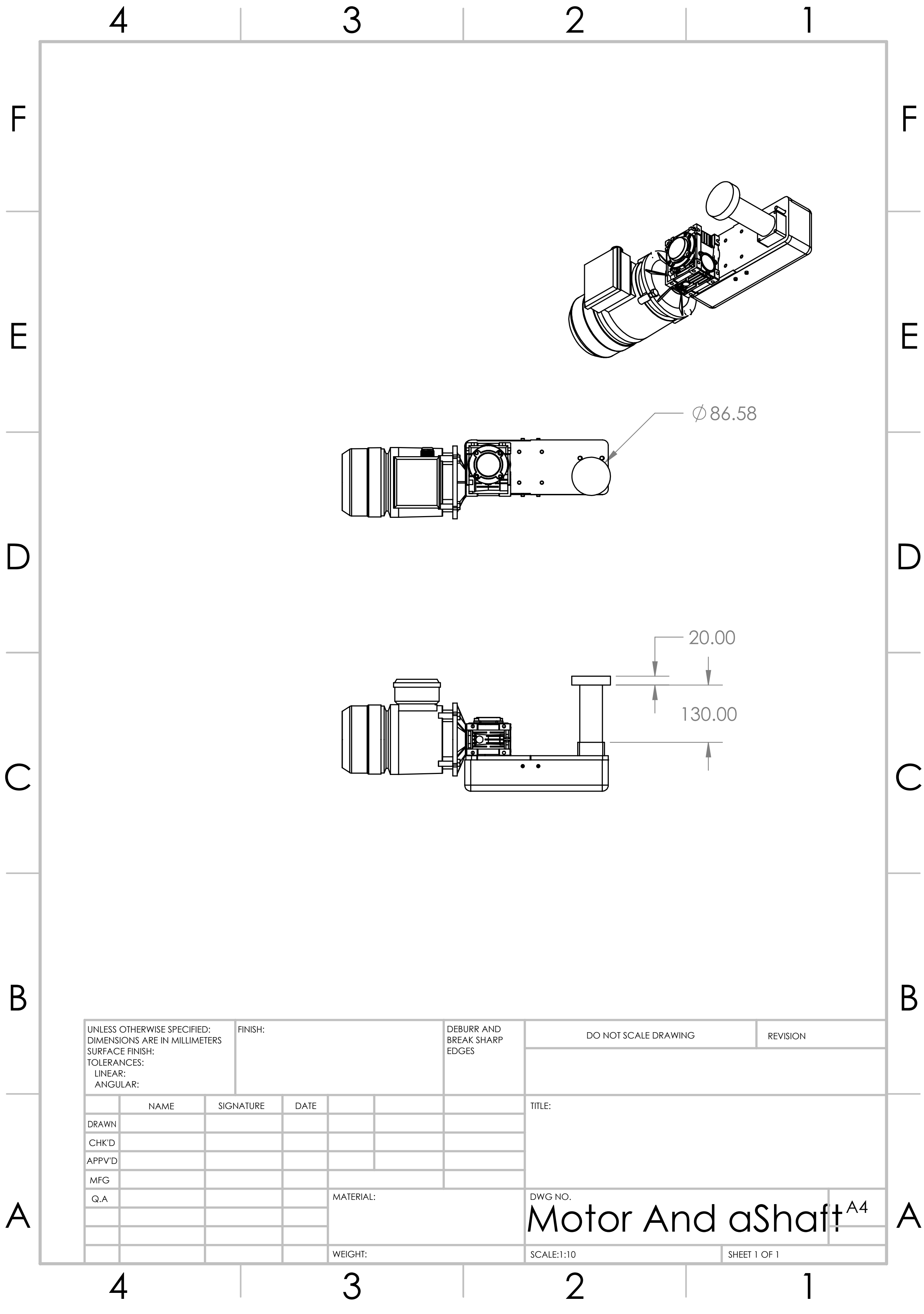
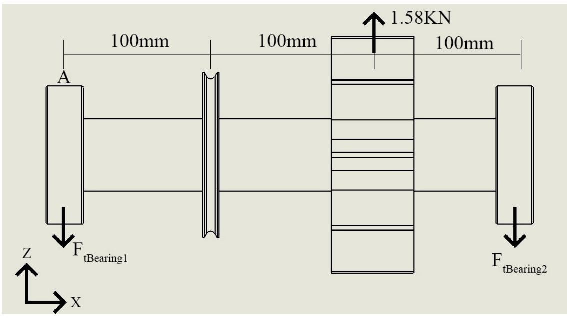


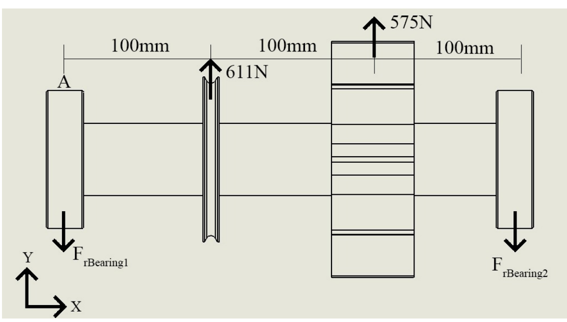
Figure 9: Motor Shaft

7.2 Bearing Selection for Input Shaft Of Gearbox



7.2.1 Force Analysis

- XZ Plane: $F_{\text{tBearing1}} + F_{\text{tBearing2}} = 1.58KN$
- $M_{\text{A}} = 0$ $1.58K(200) = F_{\text{tBearing2}} * 300$, $F_{\text{tBearing2}} = 1.053KN \approx 1KN$
- $F_{\text{tBearing1}} = 527N \approx 530N$



- XY Plane: $F_{\text{rBearing1}} + F_{\text{rBearing2}} = 1186N$
- $M_{\text{A}} = 0$ $611(100) + 575(200) = F_{\text{rBearing2}} * 300$, $F_{\text{rBearing2}} = 587N$
- $F_{\text{rBearing1}} = 599N \approx 600N$
- $TotalForceOnBearing1 = \sqrt{{F_{\text{rBearing1}}}^2 + {F_{\text{tBearing1}}}^2} = 800.5 \approx 800$
- $TotalForceOnBearing2 = \sqrt{{F_{\text{rBearing2}}}^2 + {F_{\text{tBearing2}}}^2} = 1160N$ so Bearing 2 is the most critical

7.2.2 Bearing Selection

- $F_{\text{Bearing}} = F_{\text{r}} = 1.16KN$

- Assuming Diameter of Shaft = 20mm
- Choosing Bearing with Designation: 6204 ETN9 as 1^{st} trial
- $C_{\text{tabulated}} = 15.6$ KN
- $C_{0\text{tabulated}} = 7.65$ KN
- $n_{\text{tabulated}} = 20\ 000$ RPM
- Driving RPM (n) = 700 RPM
- L = 25000
- $P = F_{\text{r}} = 1.16kN$
- $P^{'} = K_{\text{sf}} * P = 1.2 * 1.16 = 1.392 \approx 1.4kN$
- $C_{\text{calculated}} = P^{'}(\frac{60*L*n}{10^6})^{1/3} = 14.23kN$
- $C_{\text{calculated}} < C_{\text{tabulated}}$, Ok Safe
- $P_0 = F_r = 1.16$ kN
- $S_0 = 1.5$ (Moderate Shocks)
- $C_{0\text{calculated}} = S_0 * P_0 = 1.74KN$
- $1.74KN < C_{0\text{tabulated}}(7.65KN)$ Ok Safe
- $n < n_{\text{tabulated}}$ Ok Safe

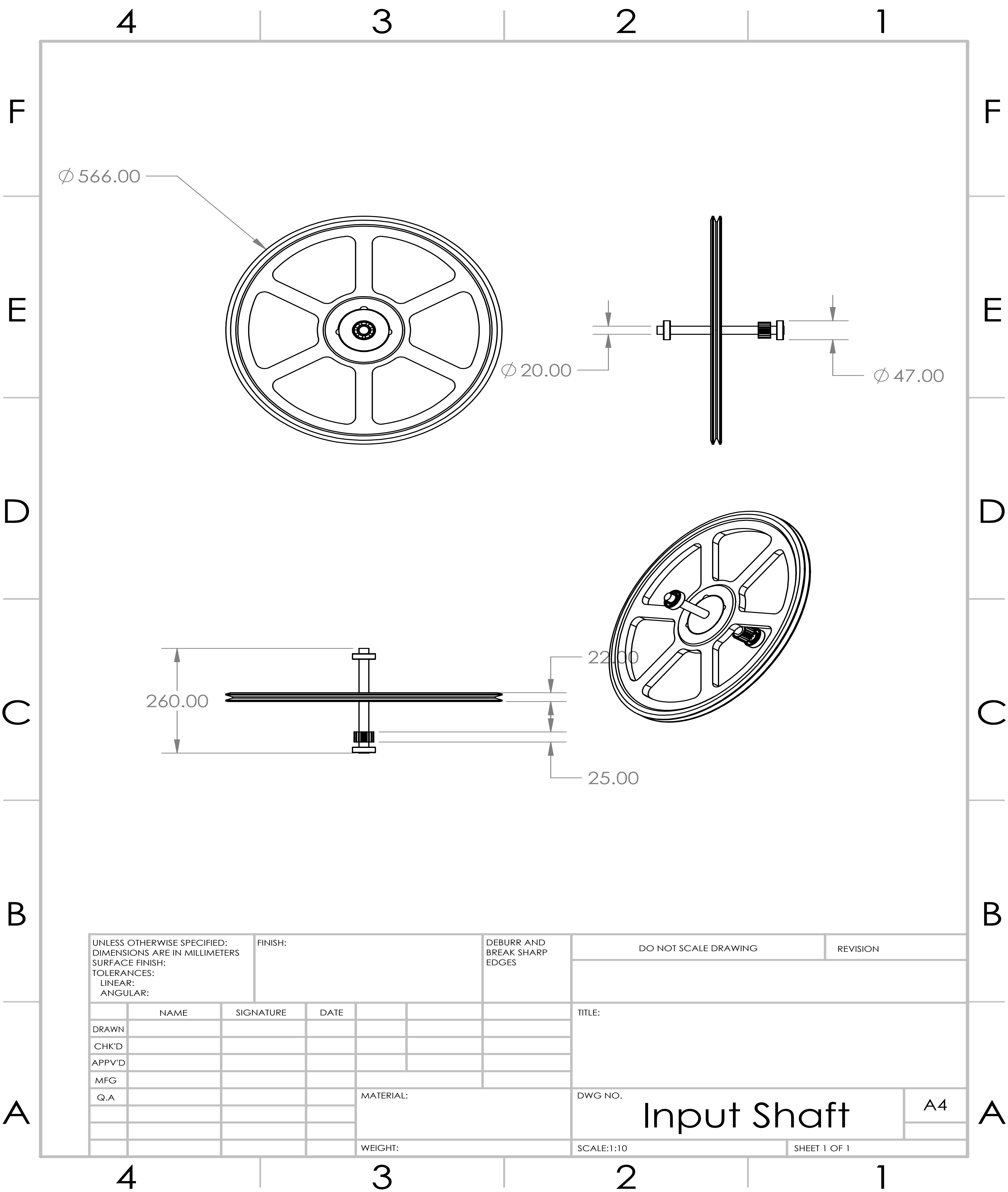
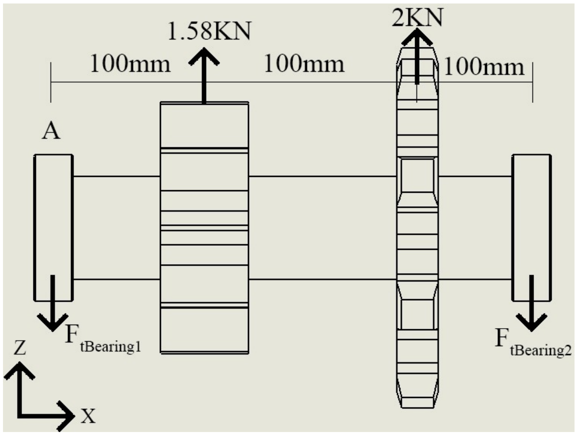


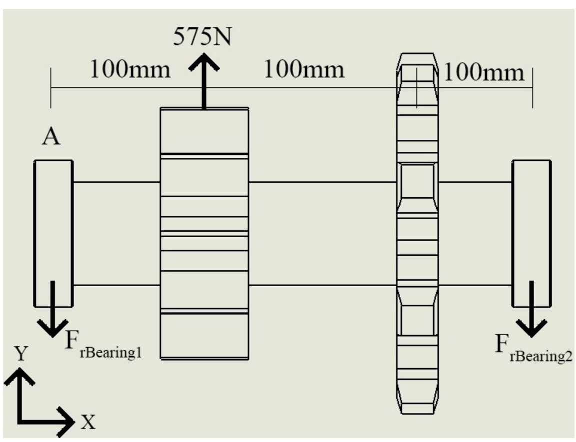
Figure 10: Input Shaft of Gearbox

7.3 Bearing Selection for Output Shaft Of Gearbox



7.3.1 Force Analysis

- XZ Plane: $F_{\text{tBearing1}} + F_{\text{tBearing2}} = 3.58KN$
- $M_{\text{A}} = 0$ $1.58K(100) + 2K(200) = F_{\text{tBearing2}} * 300$, $F_{\text{tBearing2}} = 1.86KN \approx 1.9KN$
- $F_{\text{tBearing1}} = 1.72KN \approx 1.7KN$



- XY Plane: $F_{\text{rBearing1}} + F_{\text{rBearing2}} = 575N$
- $M_{\text{A}} = 0$ $575(100) = F_{\text{rBearing2}} * 300$, $F_{\text{rBearing2}} = 192N$
- $F_{\text{rBearing1}} = 383N \approx 380N$
- $TotalForceOnBearing1 = \sqrt{{F_{\text{rBearing1}}}^2 + {F_{\text{tBearing1}}}^2} = 1.76KN \approx 1.8KN$

- $TotalForceOnBearing2 = \sqrt{F_{\text{rBearing2}}{}^2 + F_{\text{tBearing2}}{}^2} = 1.87KN$ so Bearing 2 is the most critical

7.3.2 Bearing Selection

- $F_{\text{Bearing}} = F_{\text{r}} = 1.87KN$
- Assuming Diameter of Shaft = 60mm
- Choosing Bearing with Designation: 61912 as 1$^{\text{st}}$ trial
- C $_{\text{tabulated}}$ = 16.5 KN
- $C_{0\text{tabulated}}$ = 14.3 KN
- $n_{\text{tabulated}}$ = 10 000 RPM
- Driving RPM (n) = 140 RPM
- L = 25000
- $P = F_{\text{r}} = 1.87kN$
- $P^{'} = K_{\text{sf}} * P = 1.2 * 1.87 = 2.3KN$
- $C_{\text{calculated}} = P^{'}(\frac{60*L*n}{10^6})^{1/3} = 13.7KN$
- $C_{\text{calculated}} < C_{\text{tabulated}}$, Ok Safe
- $P_0 = F_{\text{r}}$ = 1.87 KN
- S_0 = 1.5 (Moderate Shocks)
- $C_{0\text{calculated}} = S_0 * P_0 = 2.8KN$
- $2.8KN < C_{0\text{tabulated}}(14.3KN)$ Ok Safe
- $n < n_{\text{tabulated}}$ Ok Safe

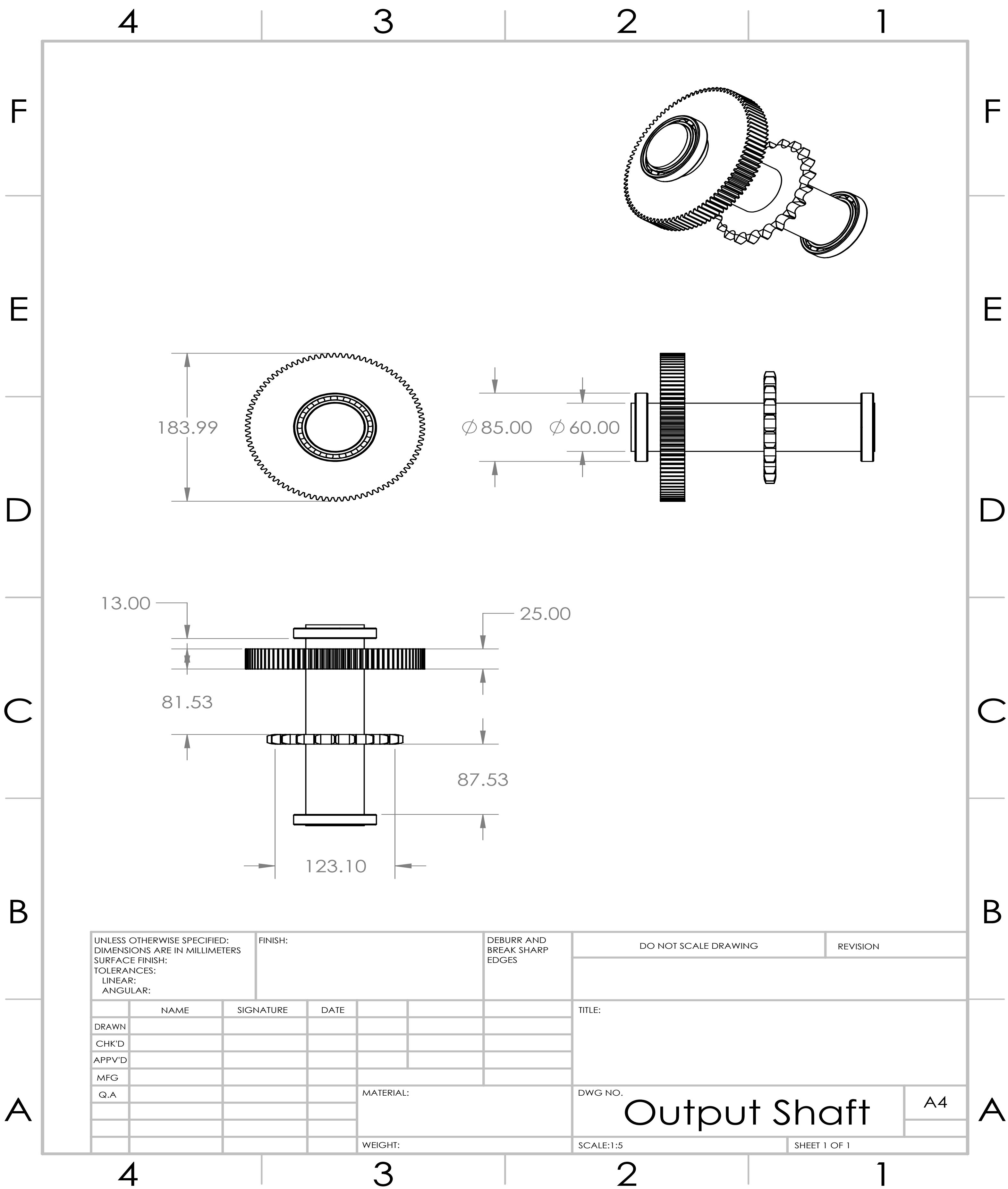
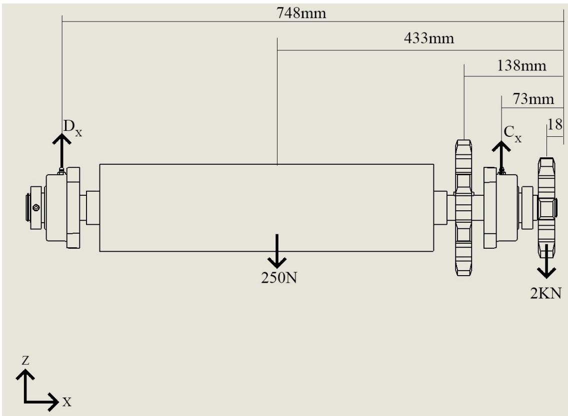


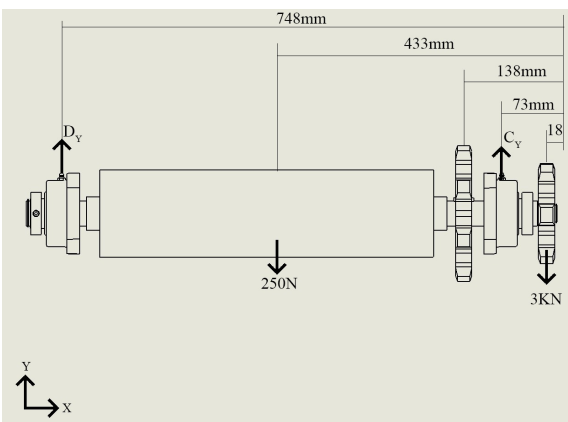
Figure 11: Output Shaft of Gearbox

7.4 Bearing Selection for Rolling Shaft Which is carrying the boxes



7.4.1 Force Analysis

- XZ Plane: $C_{\mathrm{x}} + D_{\mathrm{x}} = 2.25KN$
- $M_{\mathrm{D}} = 0 \; 2K(748-18) - C_{\mathrm{x}}(748-73) + 250(748-433) = 0$
  $C_{\mathrm{x}} = 2.28KN \approx 2.3KN$
- $D_{\mathrm{x}} = -30N(opposite to direction shown on picture)$



- XY Plane: $D_{\mathrm{y}} + C_{\mathrm{y}} = 3.25KN$
- $M_{\mathrm{D}} = 0 \; 3K(748-138) + 250(748-433) - C_{\mathrm{y}}(748-73) = 0$
  $C_{\mathrm{y}} = 2.83KN$
- $D_{\mathrm{y}} = 420N \approx 380N$

- It is shown from the analysis that bearing C is the most critical $TotalForceOnBearingC = \sqrt{C_{\text{x}}{}^2 + C_{\text{y}}{}^2} = 3.62KN \approx 3.6KN$

7.4.2 Bearing Selection

- $C_{\text{Bearing}} = F_{\text{r}} = 3.6KN$
- Assuming Diameter of Shaft = 40mm
- Choosing Bearing with Designation: 6208 as 1$^{\text{st}}$ trial
- C $_{\text{tabulated}}$ = 32.5 KN
- $\text{C}_{\text{0tabulated}}$ = 19 KN
- $\text{n}_{\text{tabulated}}$ = 11 000 RPM
- Driving RPM (n) = 70 RPM
- L = 25000
- $P = F_{\text{r}} = 3.6KN$
- $P^{'} = K_{\text{sf}} * P = 1.2 * 3.6 = 4.32KN$
- $C_{\text{calculated}} = P^{'}(\frac{60*L*n}{10^6})^{1/3} = 20.38KN \approx 20.4KN$
- $C_{\text{calculated}} < C_{\text{tabulated}}$, Ok Safe
- P_0 = F_{r} = 3.6 KN
- S_0 = 1.5 (Moderate Shocks)
- $C_{\text{0calculated}} = S_0 * P_0 = 5.4KN$
- $5.4KN < C_{\text{0tabulated}}(19KN)$ Ok Safe
- $n < n_{\text{tabulated}}$ Ok Safe

End of Bearing Selection Process

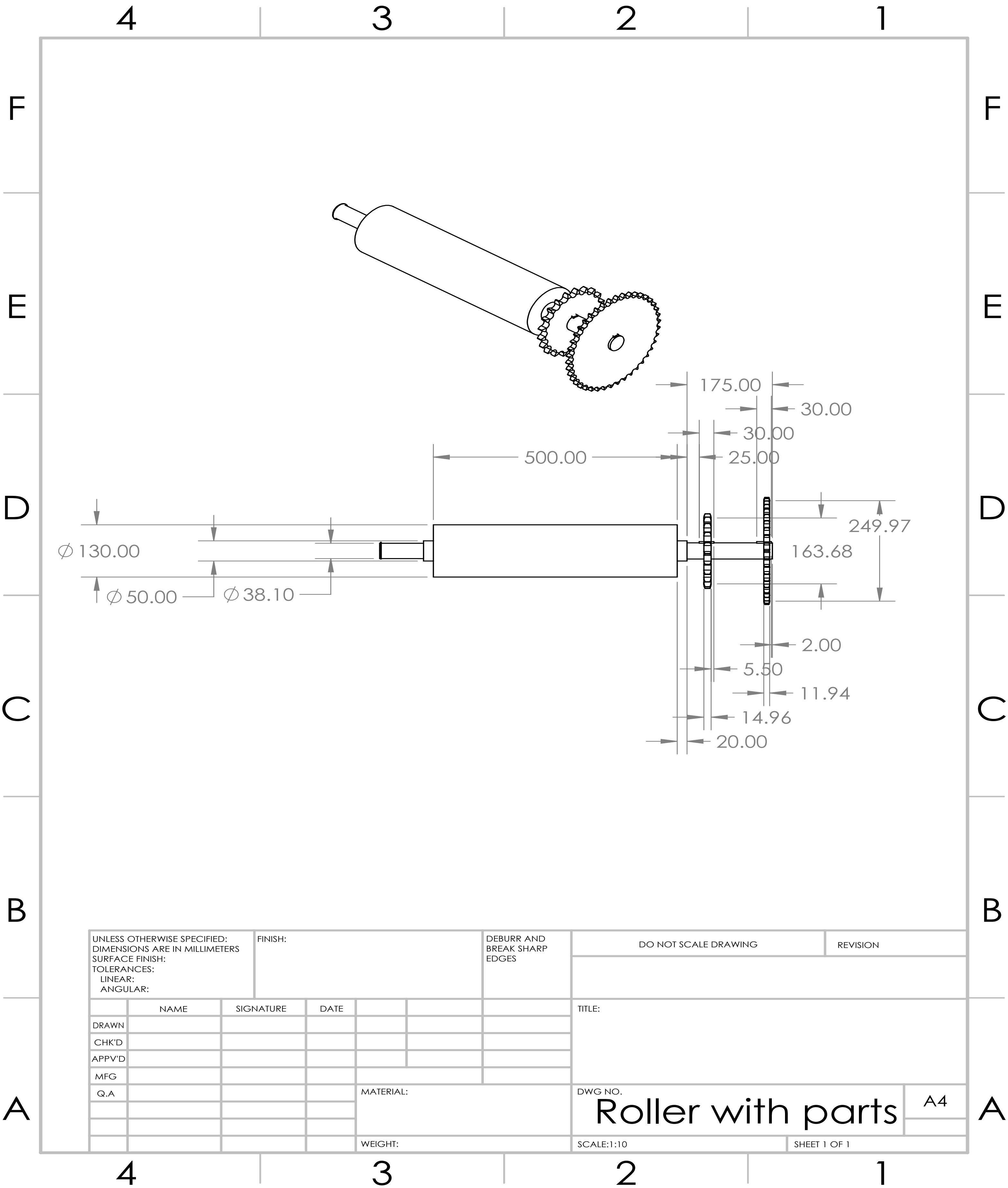


Figure 12: Main Rolling Shaft Which Receives Power

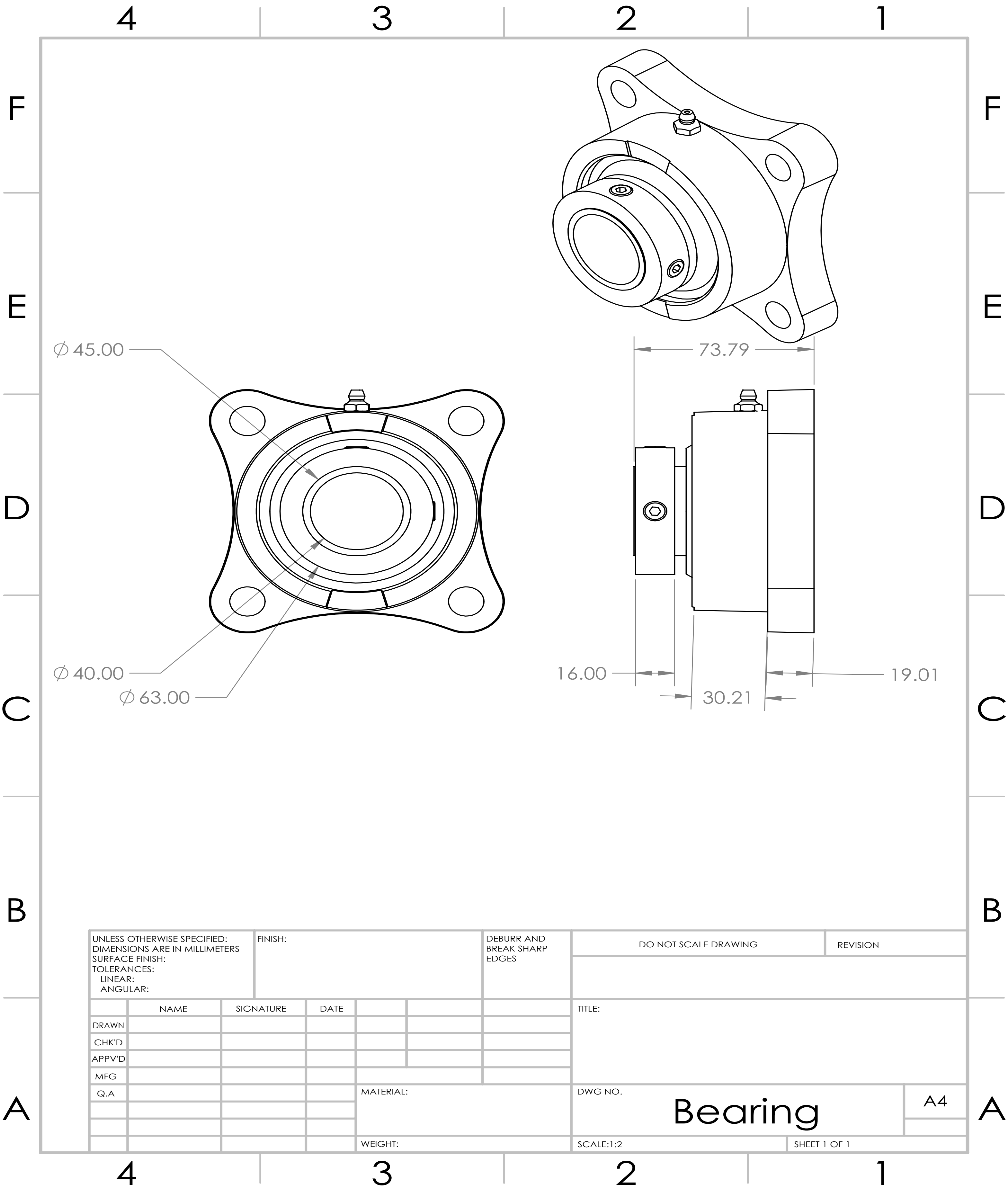


Figure 13: Bearing Mounted on Rollers

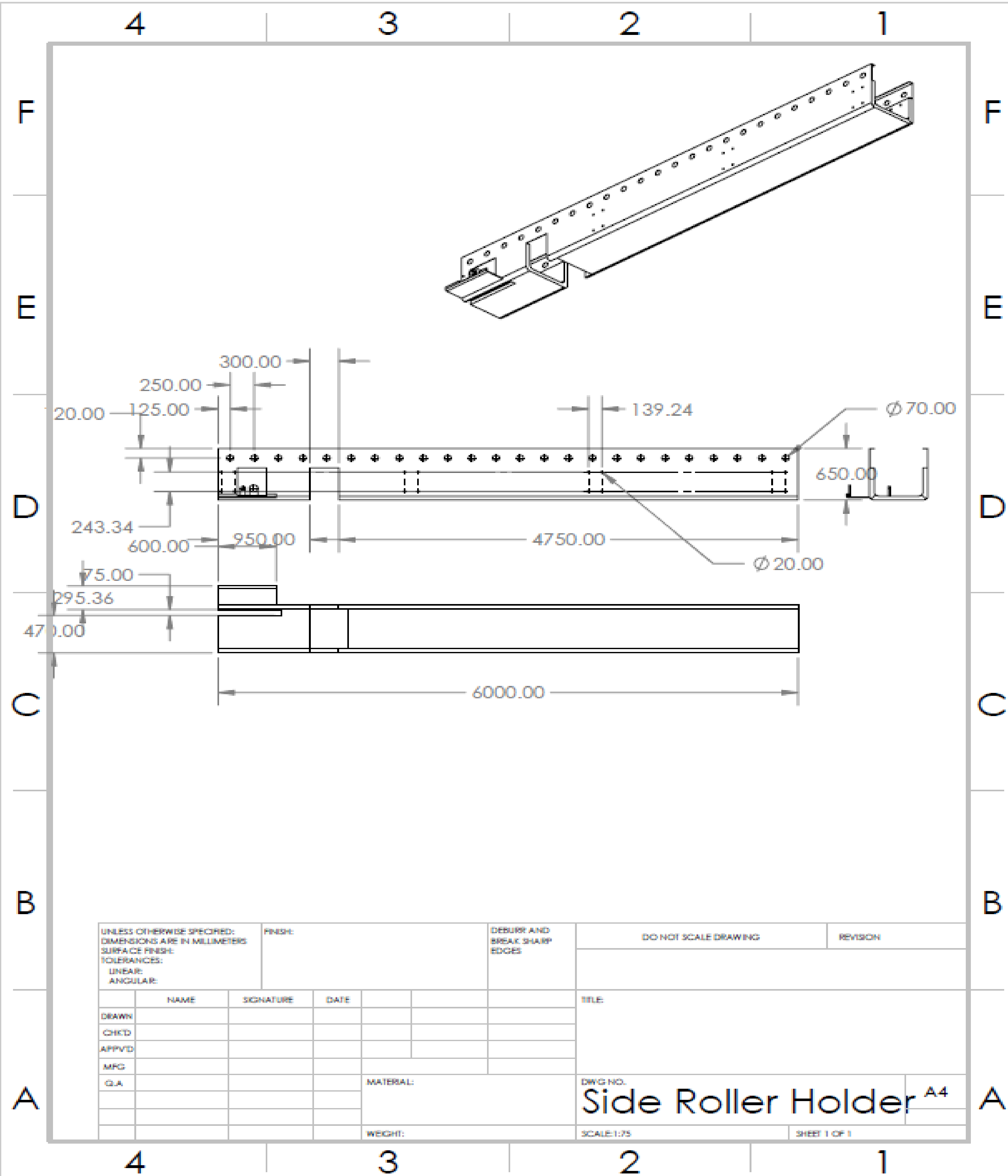


Figure 14: Side of Conveyor Which is Used to Hold Bearings

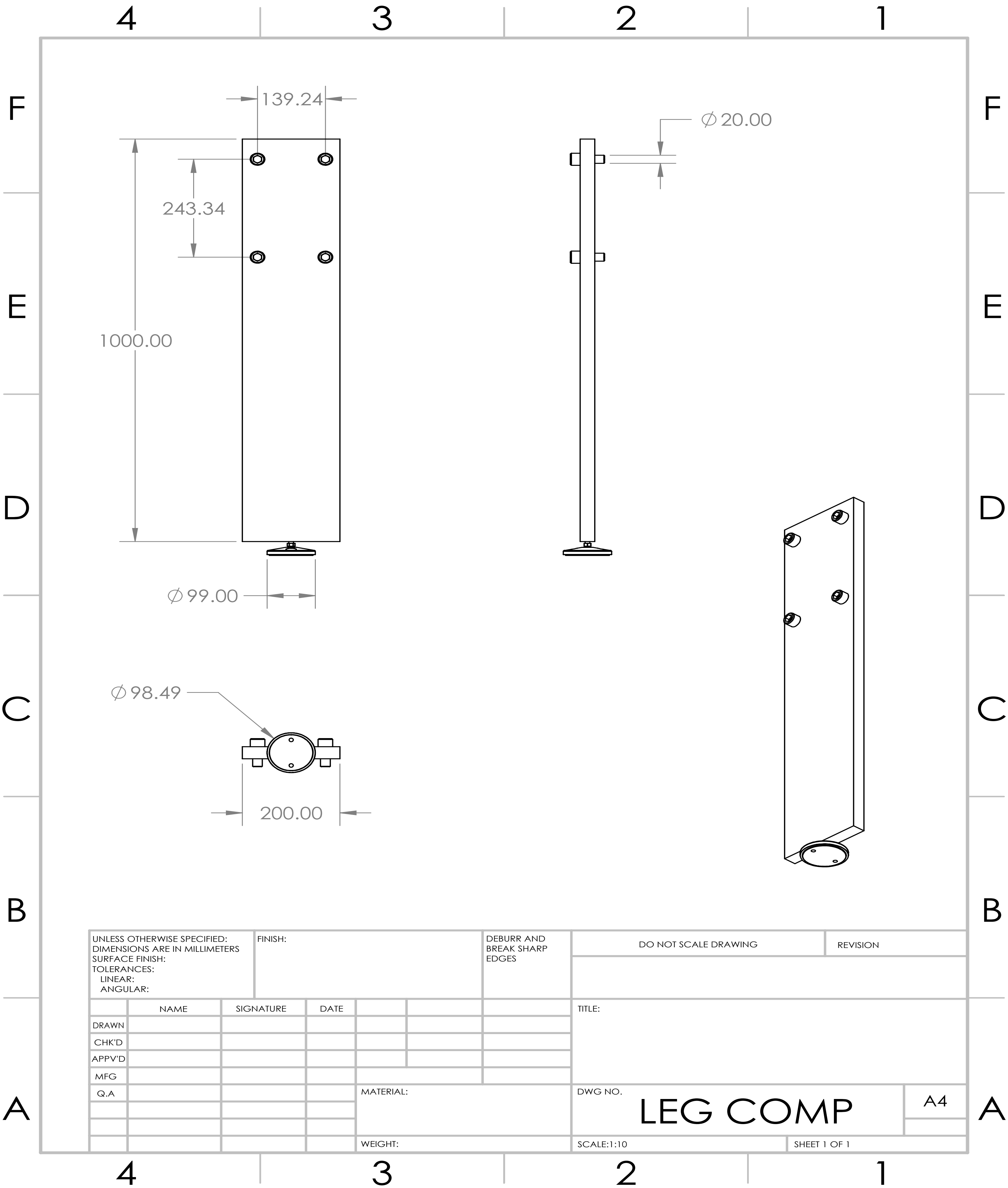


Figure 15: Design of Single Leg, the Conveyor is Held Stable on the Ground by 8 Legs

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Figure 16: QR Code Containing Further Details and a Working Animation of the Conveyor