MSE3380

Mechanical Components Design for Mechatronic Systems

Final Design Report

Group 5

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Introduction

In the previous report, the gear specifications were determined, corresponding calculations and CAD drawings were also completed. In this final report, the gearbox design will be finalized, and results will be verified. a complete analysis of shaft and bearing, calculations of shaft fatigue safety factor, shaft material endurance limit and bearing life will be conducted.

1. Summary of The Analysis from The Phase 1 Report

Performance Specifications

In the interim report, the performance specification was firstly determined, including the important specifications of weight, size and efficiency; constrains of gears, motor, lifetime and output; and associated assumptions.

Gear Ratios and Reduction Stages

The gear ratio and reduction stages were specified, after a series of analysis including DC motor no-load speed, DC motor stall torque and DC motor peak power, the required torque output of DC motor was determined to be 3.717N.m, and the required speed output was determined to be 9426rpm. Based on the calculated torque output, the gear ratio was calculated to be 1:16.4, in order to implement this gear ratio, the two reduction stages were determined to be 1:4 for both stages.

Torque Within the Shafts

The torque within the shaft was estimated, the shaft that connects to the DC motor will experience least torque, which is the output of DC motor of 3.717N.m. And the intermediate shaft will experience a torque of 14.105N.m, the final shaft will experience most torque which is calculated to be 53.525N.m.

Forces on The Pinions and Gears and Their Dimensions

The forces on the pinions and gears and their dimensions was determined, the forces between DC motor pinion and intermediate gear is calculated to be 2163.12N, the forces between intermediate pinion and output gear is calculated to be 2052.14N. The gears were carefully selected from the Boston gears catalog, the two pinions have the same dimensions but different direction, with a transverse diametral pitch of 16 inch, 12 teeth, a pitch diameter of 0.75 inch, a bore diameter of 0.375 inch, a keyway dimension of 1/16x1/32, A style, and the catalog number is H1612R and H1612L respectively. The two gears also have the same dimensions but different direction, with a transverse diametral pitch of 16 inch, 48 teeth, a pitch diameter of 3 inches, a bore diameter of 0.5 inch, a keyway dimension of 1/8x1/16, a style, and the catalog number is H1648R and H1648L respectively.

Contact and Bending Stress, Contact and Bending Safety for the Gears

The contact stress, bending stress, contact safety factor and bending safety factor of the gears was calculated using MATLAB script, and the results is summarized below.

	Pinion 1	Gear 2	Pinion 3	Gear 4
Bending Stress	42209	40015	34607	33254
Contact Stress	167743	94295	151888	85960

Bending FOS	1.0	1.10	1.27	1.40
Contact FOS	0.75	1.38	1.1	1.55

Additionally, the CAD model of the entire gearbox was completed, and the 2D assembly drawing of the entire gearbox with cross-section was also drawn using Solidworks.

Updates

Due to fact that the calculated pinion 1 contact FOS is less than 1(Pinion 1 cannot withstand such high pressure), all 4 gears' transverse diametral pitch are changed to be 8 so that the gear can withstand such a pressure and they can fit together without interference.

The specifications for newly updated gears can be found in appendix A.

2. Free Body Diagrams for Shafts

Included in Section 3: Calculation of All Torques and Forces Acting on Shafts

3. Calculation of All Torques and Forces Acting on Shafts

Firstly, the reaction forces in each shaft was determined, each shaft is supported by two bearings that treated as a fixed support, since the helical gears are used, the forces between the gear and pinion is three-dimensional, that is, force is acting on x, y and z axis. Detailed calculations for reaction forces are demonstrated below.

Shaft 1 (Input Shaft):

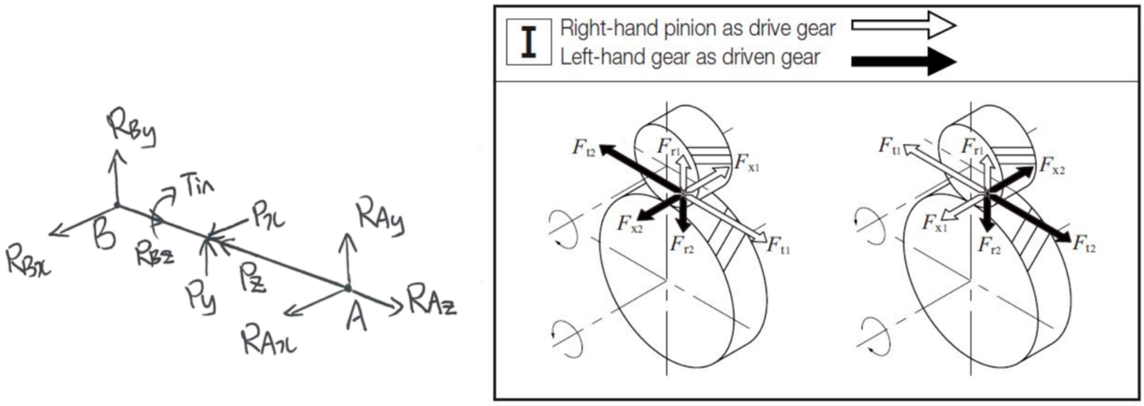


Figure 1. Free Body Diagram for input shaft

The forces between pinion 1 and gear 2 is calculated to be:

$$P_z = -W_{t12} = \frac{T}{r} = \frac{14.105N.m}{0.01905m} = -740.42N$$

$$P_y = -W_{r12} = W_{t12}\tan\phi_t = 740.42 * \tan 20.09 = -270.81N$$

$$P_x = -W_{a12} = W_{t12}\tan\psi = 740.42 * \tan 45 = -740.42N$$

$$W = \frac{W_t}{\cos\psi\cos\phi_n} = \frac{740.42}{\cos 14.5 \cos 45} = 1081.56N$$

From Free Body Diagram, the following equations can be listed:

$$\Sigma M_{Bx} = P_x(1) + R_{Ax}(2) = 0$$

$$\Sigma M_{By} = P_y(1) + R_{Ay}(2) = 0$$

$$\Sigma F_z = R_{Az} + R_{Bz} - P_z = 0 \quad R_{Bz} = R_{Az}\text{(Assumption)}$$

$$\Sigma F_x = R_{Ax} + R_{Bx} + P_x = 0$$

$$\Sigma F_y = R_{Ay} + R_{By} + P_y = 0$$

Solving equations above, the results are:

$$R_{Ax} = 370.21N \qquad R_{Ay} = 135.405N \qquad R_{Az} = -370.21N$$

$$R_{Bx} = 370.21N \qquad R_{By} = 135.405N \qquad R_{Bz} = -370.21N$$

Shaft 2 (Intermediate Shaft):

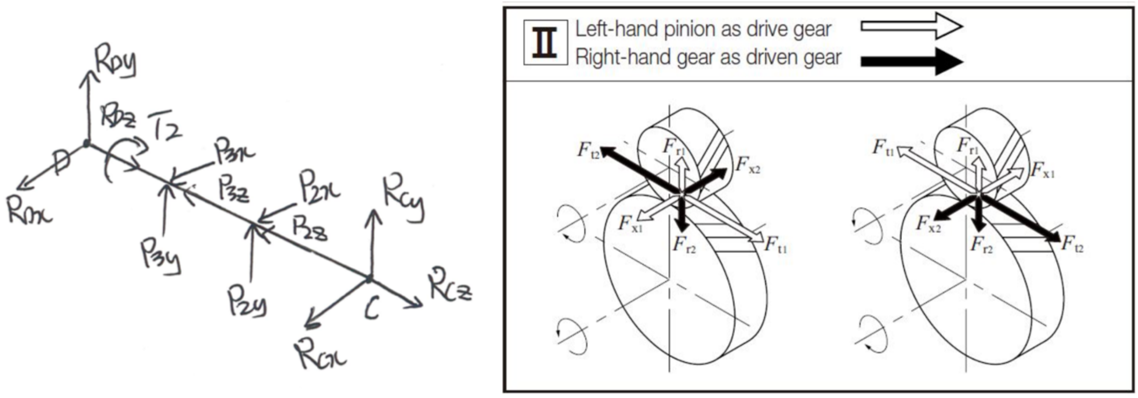


Figure 2. Free Body Diagram for intermediate shaft

The distance between point D and point 3 is determined to be 1/6 of the total shaft length, distance between point 3 and shaft midpoint is determined to be 2/6 of the total shaft length. The shaft is symmetrical.

The forces between pinion 3 and gear 4 and pinion 1 and gear 2 can be calculated based on previous given formulas:

$$P_{2z} = W_{t12} = \frac{T}{r} = \frac{14.105N.m}{0.01905m} = 740.42N\text{e}$$

$$P_{2y} = W_{r12} = W_{t12} \tan \phi_t = 740.42 * \tan 20.09 = 270.81N$$

$$P_{2x} = W_{a12} = W_{t12} \tan \psi = 740.42 * \tan 45 = 740.42N$$

$$P_{3z} = W_{t34} = \frac{T}{r} = \frac{53.525N.m}{0.0762m} = 702.43N$$

$$P_{3y} = W_{r34} = W_{t34} \tan \phi_t = 1404.86 * \tan 20.09 = 256.9N$$

$$P_{3x} = W_{a34} = W_{t34} \tan \psi = 1404.86 * \tan 45 = 702.43N$$

$$W = \frac{W_t}{\cos \psi \cos \phi_n} = \frac{702.43}{\cos 14.5 \cos 45} = 1026.07N$$

From Free Body Diagram, the following equations can be listed:

$$\Sigma M_{Dy} = P_{3y}(1) + P_{2y}(5) + R_{Cy}(6) = 0$$

$$\Sigma M_{Dx} = P_{3x}(1) + P_{2x}(5) + R_{Cx}(6) = 0$$

$$\Sigma F_z = R_{Dz} + R_{Cz} + P_{3z} - P_{2z} = 0 \;\; R_{Cz} = R_{Dz}\text{(Assumption)}$$

$$\Sigma F_y = R_{Cy} + R_{Dy} + P_{2y} + P_{3y} = 0$$

$$\Sigma F_x = R_{Cx} + R_{Dx} + P_{2x} + P_{3x} = 0$$

Solving equations above, the results are:

$$R_{Cx} = -734.09N \qquad R_{Cy} = -268.492N \qquad R_{Cz} = -19N$$

$$R_{Dx} = -708.76N \qquad R_{Dy} = -259.218N \qquad R_{Dz} = -19N$$

Shaft 3 (Output Shaft):

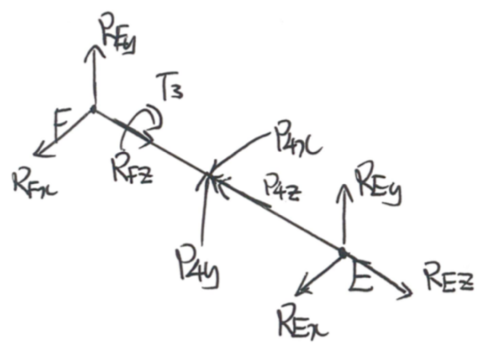


Figure 3. Free Body Diagram for output shaft

The forces between pinion 3 and gear 4 can be calculated based on previous given formulas:

$$P_{4z} = -W_{t34} = \frac{T}{r} = \frac{53.525N.m}{0.0762m} = -702.43N$$

$$P_{4y} = -W_{r34} = W_{t34}\tan\phi_t = 1404.86 * \tan 20.09 = -256.9N$$

$$P_{4x} = -W_{a34} = W_{t34}\tan\psi = 1404.86 * \tan 45 = -702.43N$$

From Free Body Diagram, the following equations can be listed:

$$\Sigma M_{Fx} = P_{4x}(1) + R_{Ex}(2) = 0$$

$$\Sigma M_{Fy} = P_{4y}(1) + R_{Ey}(2) = 0$$

$$\Sigma F_z = R_{Fz} + R_{Ez} - P_{4z} = 0 \;\; R_{Fz} = R_{Ez}\text{(Assumption)}$$

$$\Sigma F_x = R_{Fx} + R_{Ex} + P_{4x} = 0$$

$$\Sigma F_y = R_{Fy} + R_{Fy} + P_{4y} = 0$$

Solving equations above, the results are:

$$R_{Ex} = 351.22N \qquad R_{Ey} = 128.46N \qquad R_{Ez} = -351.22N$$

$$R_{Fx} = 351.22N \qquad R_{Fy} = 128.46N \qquad R_{Fz} = -351.22N$$

4. Shear, Moment and Torque Diagrams for Shafts

Pictures on next page.

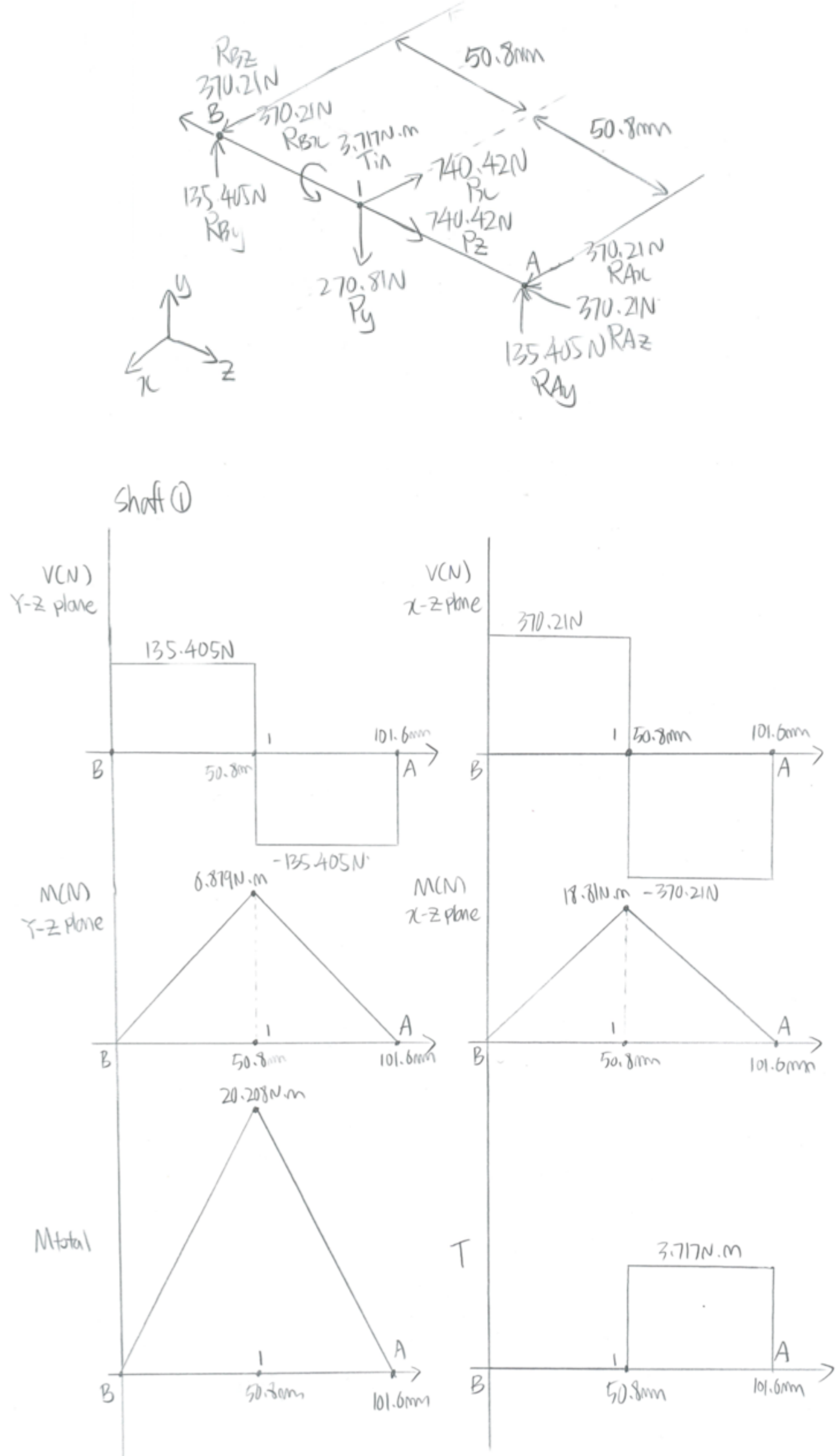


Figure 5. Shear, moment and torque diagrams for input shaft

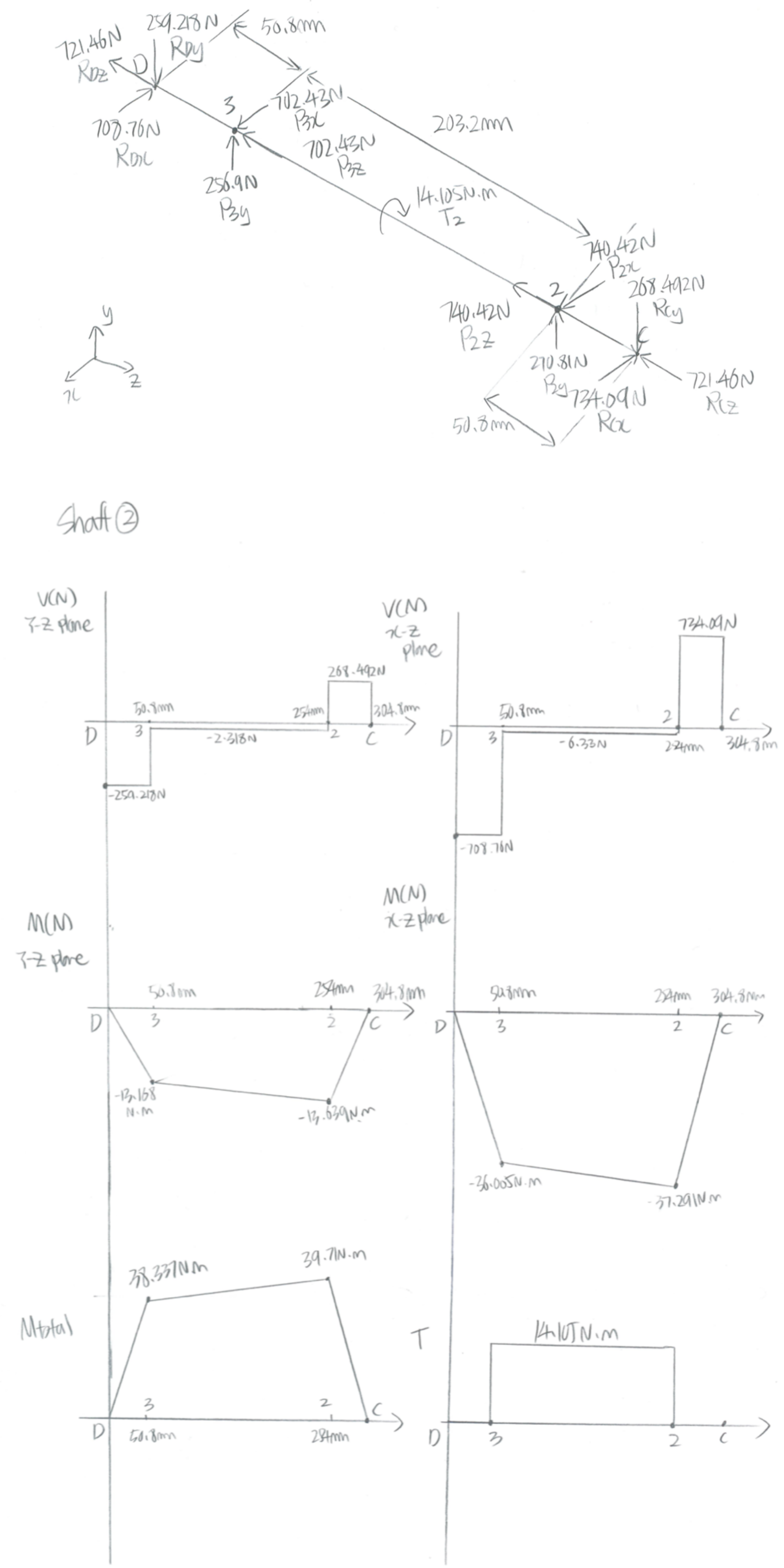
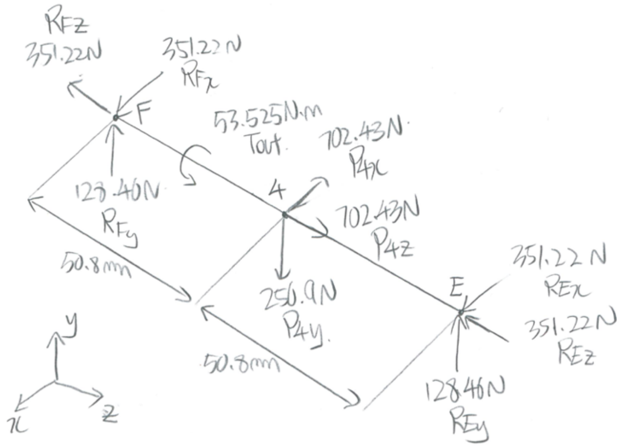


Figure 6. Shear, moment and torque diagrams for intermediate shaft



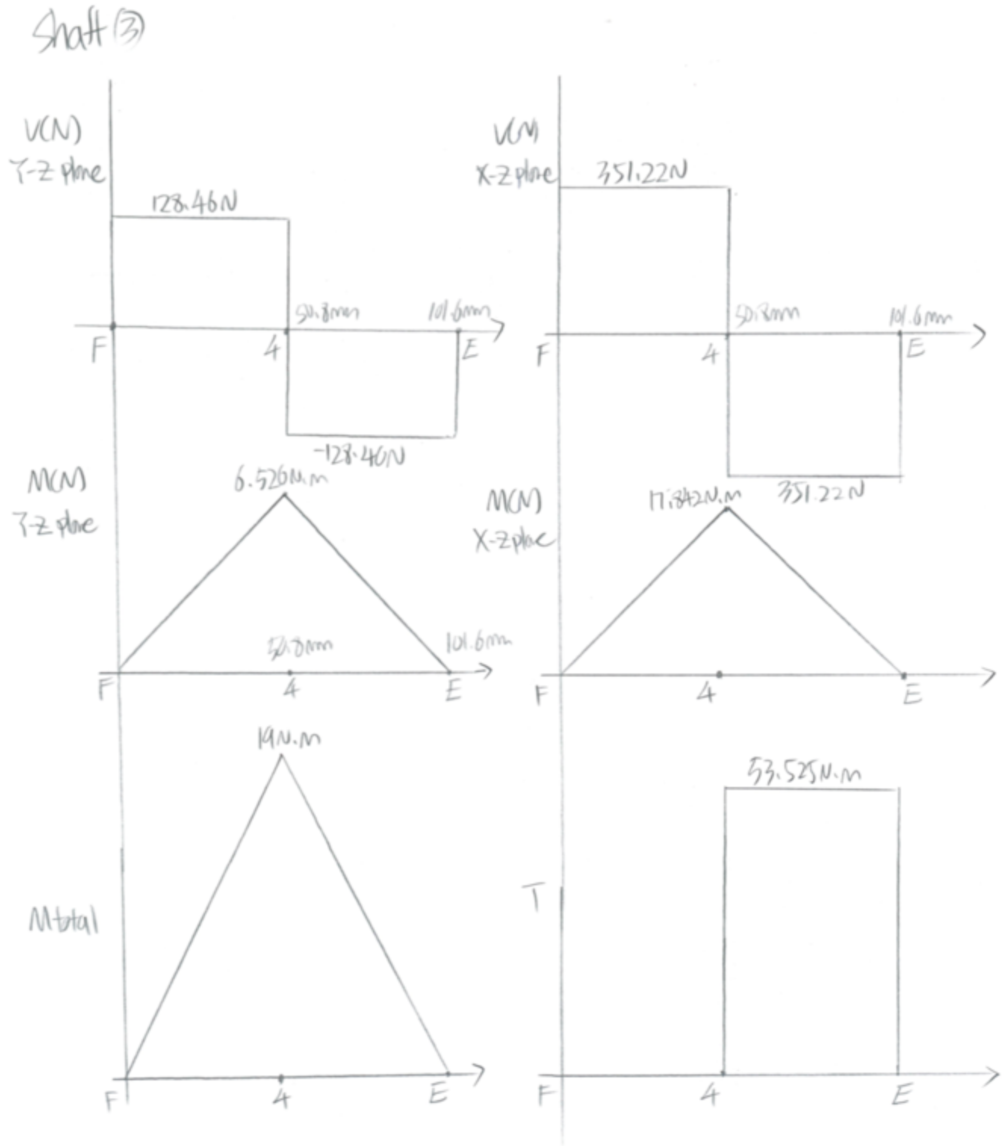


Figure 7. Shear, moment and torque diagrams for output shaft

5. Fully Corrected Endurance Limit for Shaft Material

Include in Section 6. Shaft Fatigue Safety Factors

6. Shaft Fatigue Safety Factors

To obtain the endurance limit for shaft material, the shaft material needs to be determined, the yield strength and tensile strength of the material needs to be sufficient in order to prevent failure. In the case of transmission shaft, the material with high torsional rigidity and fatigue resistance is preferred, which are steel and aluminum. Steel is extremely durable, inexpensive, but heavy. On the other hand, aluminum is less durable but significantly more lightweight than steel. In this application, lightweight is not a major requirement, but high strength is preferred, as a result, steel is chosen for this application. For initial test, AISI 1060 was implemented to calculate the endurance limit.

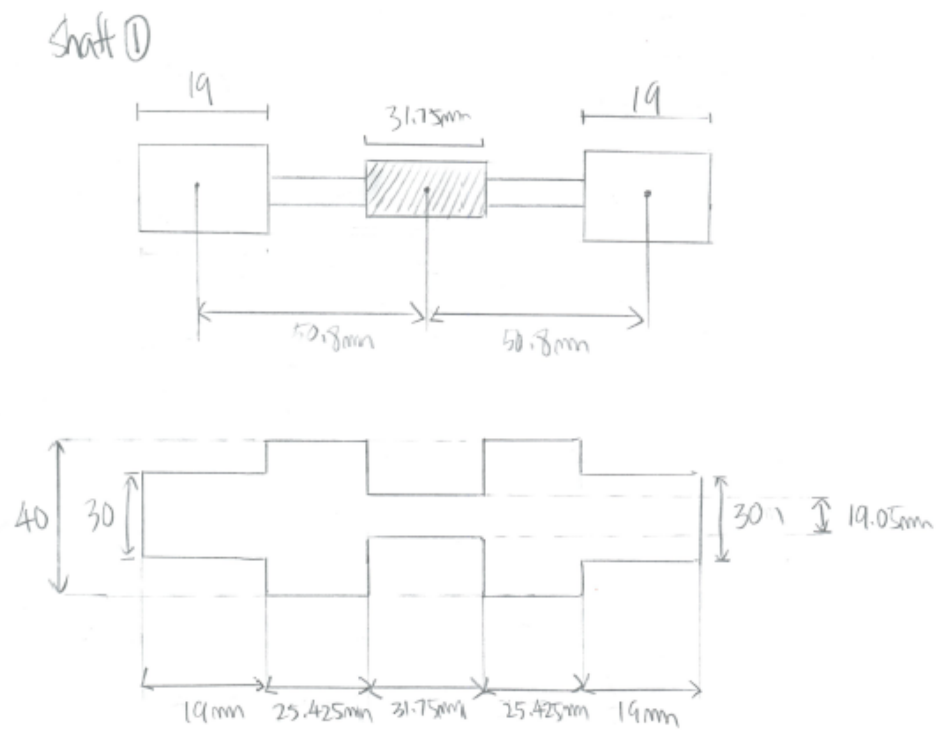


Figure 8. Dimensions for input shaft

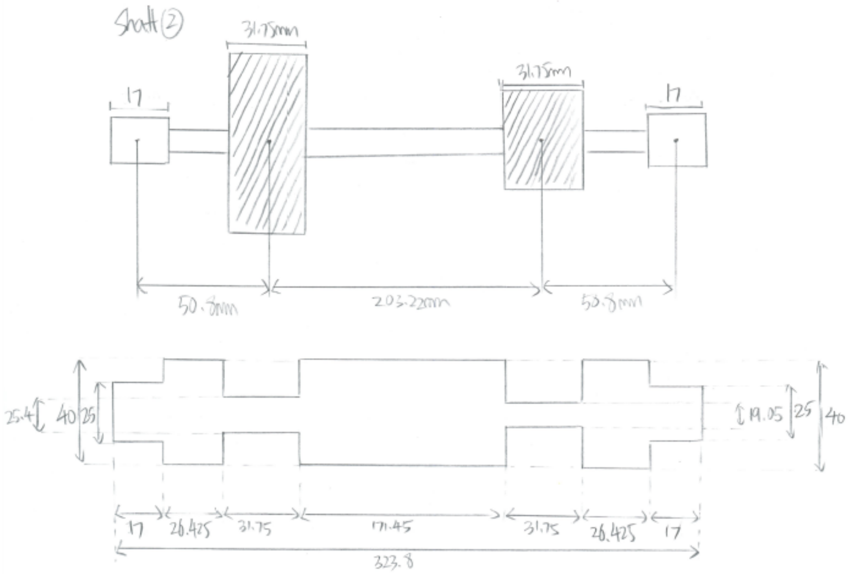


Figure 9. Dimensions for intermediate shaft

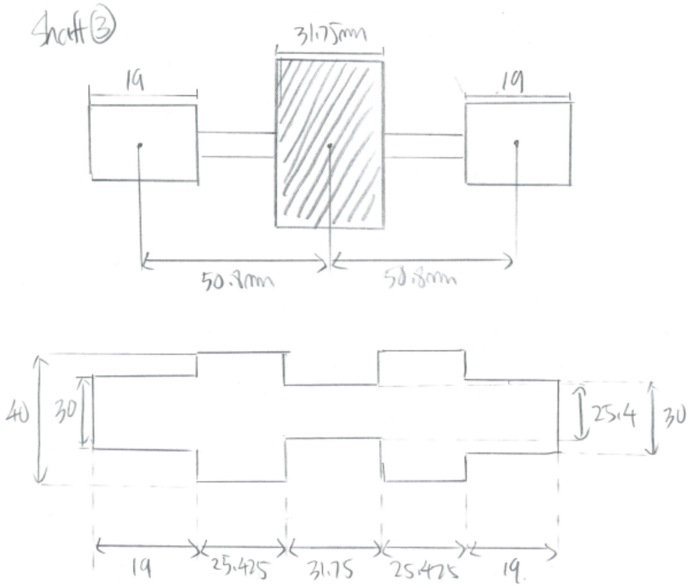


Figure 10. Dimensions for output shaft

In order to determine endurance limit, the following analysis is performed.

1. The rotary-beam test specimen endurance limit of AISI is determined to be 340 MPa.

   Using the equation

$$s_e' = \begin{cases} 0.5S_{ut} & S_{ut} \leq 200kpsi\ (1400\ MPa) \\ 100\ kpsi & S_{ut} > 200kpsi \\ 700\ MPa & S_{ut} > 1400\ Mpa \end{cases}$$

As the Sut, the tensile strength of AISI 1060 is 680MPa, which is smaller than 1400 MPa, the rotary-beam test specimen endurance is half of the tensile strength which is 340 MPa.

2. The corrected endurance limit at the critical location of a machine part in the geometry and condition of use is given as:

$$S_e = k_a k_b k_c k_d k_e k_f S_e'$$

The surface condition modification factor is given as $k_a = aS_{ut}^b$, where a and b are found using the following table:

	Factor *a*		Exponent
Surface Finish	**S_{ut}, kpsi**	**S_{ut}, MPa**	***b***
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

Solving the equation using the parameters of Machined or cold-drawn assuming the shaft should be machined, ka = 0.8009.

3. The size factor is evaluated using the equations

$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \le d \le 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < 254 \text{ mm} \end{cases}$$

The size factor is determined to be 0.9046 for shaft 1 and 2, and 0.8772 for shaft 3.

4. The Loading Factor Kc is determined to be 1 as the load factor is bending.
5. The temperature factor Kd is determined to be 1 as the electric drill should be used in a room temperature condition.
6. The reliability factor, Ke was determined using the following table

Reliability, %	Transformation Variate z_a	Reliability Factor k_e
50	0	1.000
90	1.288	0.897
95	1.645	0.868
99	2.326	0.814
99.9	3.091	0.753
99.99	3.719	0.702
99.999	4.265	0.659
99.9999	4.753	0.620

With 99% reliability, Ke was determined to be 0.814.

7. Miscellaneous-effects factor kf is assumed to be 1.

Then, fatigue stress-concentration factor is solved using the equation:

$$K_f = 1 + \frac{K_t - 1}{1 + \sqrt{a/r}}$$

Where Kt is the stress-concentration factor, a is the $\sqrt{a}$ is the Neuber constant and r is the fillet radius. Please see MATLAB files for calculations.

The stress is calculated using $\sigma_{rev} = \frac{Mc}{I}$.M is the moment derived from moment diagram where stress concentration is located at the fillet transition location. c is the radius of the smaller part of the shaft, and I is solved from $I = \frac{\pi d^4}{64}$. Refer to MATLAB for calculations

The fatigue safety factors for shafts is solved as

$$n_f = \frac{Endurance\ limit}{Maximum\ reverseal\ stresses}$$

$$n_f = \frac{S_e}{K_f \sigma_{rev}}$$

The results obtained are listed below:

	Se	Kf	σrev	nf
Shaft 1	200.5MPa	1.4811	26.04MPa	5.20
Shaft 2	200.5MPa	1.4811	58.17MPa	2.33
Shaft 3	104.4MPa	1.6077	24.5MPa	4.93

All three shafts have a fatigue safety factors greater than 1 with a reliability of 99%. This shows that the shafts should have an infinite life. This result is reasonable as the used material for the shaft is steel which has a high tensile strength.

7. Bearing specification (manufacturer/model number) configuration and life calculation. Specify a recommended lubrication strategy.

Firstly, the bearing type needs to be determined, in this application, helical gears were used, as a result, there are thrust load and radial load exist. Bearings are manufactured to take pure radial loads, pure thrust load or the combination of two, ball bearings are considered to be the most suitable type for this application due to the fact that ball bearing can withstand both thrust load and radial load in a considerable amount. Specifically, single-row deep groove ball bearing was decided to be implement, since single-row deep groove ball bearing is excellent for high-speed application, has good radial load capacity and satisfactory thrust load capacity in both directions, and low torque capacity at start-up and running speed. All bearings are selected from The Timken Company.

To further specify the bearing specification, the dynamic load and static load needs to be determined, following procedures were used to determine these two values. Assuming:

- Continuous one-shift operation at 9426 rpm (8 hours day, 5-day week), life=30kh
- Commercial gear, load factor = 1.2
- 90% reliability
- a=3 for ball bearing

Shaft 1:

Estimate life for 8-hour day:

$$L_D = 9426rpm \ \times 30kh \times \frac{60m}{h} = 1.6967 \times 10^{10}rev$$

The equivalent load:

$$F_e = X_i V F_r + Y_i F_a \ , V = 1$$

Radial load = 394.2N Axial load = 370.12N

Since

$$\frac{F_a}{(VF_r)} = \frac{370.21}{394.2} = 0.939 > e \ \ e_{max} = 0.44$$

Thus

$$F_e = X_2(394.2) + Y_2(370.21)$$

First guess: $X_2 = 0.56 \ Y_2 = 1.63$

$$F_e = 0.56(394.2) + 1.63(370.21) = 824.19N$$

Weibull parameters

$$x_0 = 0.02 \ \theta = 4.459 \ b = 1.483$$

From Eq. (11–10):

$$C_{10} = 1.2(824.19)\left[\frac{\left(\frac{1.6967 \times 10^{10}}{10^6}\right)}{0.02 + (4.459 - 0.02)(1 - 0.90)^{\frac{1}{1.483}}}\right]^{\frac{1}{3}} = 25414.23N = 25.41kN$$

From T 11–2 (DG) bearing 02-40 mm has

$$C_{10} = 30.7kN \ C_0 = 16.6kN$$

$$\frac{F_a}{C_0} = 0.0223, e \approx 0.21, \ Y_2 = 2.15$$

$$F_e = 0.56(394.2) + 2.15(370.21) = 1016.7N$$

$$C_{10} = 1.2(1016.7)\left[\frac{1.6967 \times 10^{10}}{10^6}\right]^{\frac{1}{3}} = 31.35kN$$

From T 11–2 (DG) bearing 02-45 mm has

$$C_{10} = 33.2kN\ C_0 = 18.6kN$$

$$\frac{F_a}{C_0} = 0.0199, e \approx 0.2069,\ Y_2 = 2.1736$$

$$F_e = 0.56(394.2) + 2.1736(370.21) = 1025.43N$$

$$C_{10} = 1.2(1025.43)\left[\frac{1.6967 \times 10^{10}}{10^6}\right]^{\frac{1}{3}} = 31.62kN$$

Selecting from The Timken Company ball bearings catalog, two 30mm bore single-row deep groove ball bearing will be used in both ends of input shaft. The bearing has a outside diameter of 72mm, a width of 19mm, a fillet radius of 1.0mm, a weight of 0.354kg, a static load rating of 15600N and an extended dynamic load rating of 33900N.

Bearing life:

$$\mathcal{L}_{10} = \frac{60\mathcal{L}_R n_R}{60n_D}(\frac{C_{10}}{F_e})^a = \frac{10^6}{60(9426rpm)}\left(\frac{33900}{1025.43}\right)^3 = 63885.5h$$

Shaft 2:

Point C: Radial load = 781.65N Axial load = 19N

$$\frac{F_a}{(VF_r)} = \frac{19N}{781.65N} = 0.02431 < e\ \ e_{min} = 0.19$$

$$F_e = 1(781.65) + 0(19) = 781.65N$$

$$C_{10} = 1.2(781.65)\left[\frac{1.6967 \times 10^{10}}{10^6}\right]^{\frac{1}{3}} = 24.10kN$$

Point D: Radial load = 754.68N Axial load = 19N

$$\frac{F_a}{(VF_r)} = \frac{19N}{754.68N} = 0.02517 < e\ \ e_{min} = 0.19$$

$$F_e = 1(754.68) + 0(19) = 754.68N$$

$$C_{10} = 1.2(754.68)\left[\frac{1.6967 \times 10^{10}}{10^6}\right]^{\frac{1}{3}} = 23.27kN$$

Since the e is minimum for both bearings, no iteration needs to be performed, Selecting from The Timken Company ball bearings catalog, two 25mm bore single-row deep groove ball bearing will be used in point C and D of intermediate shaft. The bearing has a outside diameter of 62mm, a width of 17mm, a fillet radius of 1.0mm, a weight of 0.236kg, a static load rating of 12200N and an extended dynamic load rating of 26600N.

Bearing life at Point C:

$$\mathcal{L}_{10} = \frac{60\mathcal{L}_R n_R}{60 n_D}(\frac{C_{10}}{F_e})^a = \frac{10^6}{60(2356.5rpm)}\left(\frac{26600}{781.65}\right)^3 = 278733.5h$$

Bearing life at Point D:

$$\mathcal{L}_{10} = \frac{60\mathcal{L}_R n_R}{60 n_D}(\frac{C_{10}}{F_e})^a = \frac{10^6}{60(2356.5rpm)}\left(\frac{26600}{754.68}\right)^3 = 309697.5h$$

Shaft 3:

Radial load = 373.98N Axial load = 351.22N

$$\frac{F_a}{(VF_r)} = \frac{351.22\text{N}}{373.98\text{N}} = 0.939 > e \;\; e_{max} = 0.44$$

$$F_e = X_2(373.98) + Y_2(351.22)$$

First guess: $X_2 = 0.56 \; Y_2 = 1.63$

$$F_e = 0.56(373.98) + 1.63(351.22) = 781.92N$$

Weibull parameters

$$x_0 = 0.02 \; \theta = 4.459 \; b = 1.483$$

From Eq. (11–10):

$$C_{10} = 1.2(781.92)\left[\frac{1.6967 \times 10^{10}}{10^6}\right]^{\frac{1}{3}} = 24.11kN$$

From T 11–2 (DG) bearing 02-35 mm has

$$C_{10} = 25.5kN\ C_0 = 13.7kN$$

$$\frac{F_a}{C_0} = 0.0256, e \approx 0.215,\ Y_2 = 2.07$$

$$F_e = 0.56(373.98) + 2.07(351.22) = 936.45N$$

$$C_{10} = 1.2(936.45)\left[\frac{1.6967 \times 10^{10}}{10^6}\right]^{\frac{1}{3}} = 28.88kN$$

From T 11–2 (DG) bearing 02-40 mm has

$$C_{10} = 30.7kN\ C_0 = 16.6kN$$

$$\frac{F_a}{C_0} = 0.0212, e \approx 0.21,\ Y_2 = 2.15$$

$$F_e = 0.56(373.98) + 2.15(351.22) = 964.55N$$

$$C_{10} = 1.2(964.55)\left[\frac{1.6967 \times 10^{10}}{10^6}\right]^{\frac{1}{3}} = 29.74kN$$

Selecting from The Timken Company ball bearings catalog, two 30mm bore single-row deep groove ball bearing will be used in both ends of input shaft. The bearing has a outside diameter of 72mm, a width of 19mm, a fillet radius of 1.0mm, a weight of 0.354kg, a static load rating of 15600N and an extended dynamic load rating of 33900N.

Bearing life:

$$\mathcal{L}_{10} = \frac{60\mathcal{L}_R n_R}{60n_D}(\frac{C_{10}}{F_e})^a = \frac{10^6}{60(589rpm)}\left(\frac{33900}{964.55}\right)^3 = 1228452h$$

For this application, oil is a preferred lubricant because the rotational speed is very high, at high speed, non-viscous lubricant is required to achieve less start up and running torque and have higher speed capability, and grease is not suitable for this application because it generates much higher amount of rolling resistance than oils. However, oil have much high evaporation rate than grease, thus the bearing needs to be maintained frequently.

The bearings specifications are summarized in Appendix A, note that all bearings are single-row deep groove ball bearing that selected from The Timken Company.

Conclusion

By using the appropriate specifications and criteria, the appropriate shafts, bearing and gears were identified. Using AISI 1060 and the parameters according to the graphs shown in the previous sections, the factor of safety of all shafts are above one, which indicate that the shafts should have an infinite life. Thus, the failure of shafts would not be likely to occur. The bearings are found to have a minimum bearing life of 63886 hours, which is much larger than the desired life of 4000 hours according to the requirements of the project descriptions. By using these shafts and bearing, these would satisfy the task proposed by the project guideline.

References

R. G. Budynas and J. K. Nisbett. *Shigley's Mechanical Engineering Design,* New York, NY: McGraw-Hill Education, 2015

Timken, "Deep Goove Ball Bearings," Timken Deep Groove Ball Bearing Catalog, 2017

Appendix A. Specification Sheets

<u>Important Note: In the submitted CAD drawings, the bearing 306K are replaced with 305K since the 306K CAD drawing cannot be found.</u>

	Bore (mm)	Outside Diameter (mm)	Width (mm)	Fillet Radius (mm)	Weight (kg)	Static Load Rating (N)	Extended Dynamic Load Rating(N)	Catalog Number
Bearing A	30	72	19	1	0.365	15600	33900	306K
Bearing B	30	72	19	1	0.365	15600	33900	306K
Bearing C	25	62	17	1	0.236	12200	26600	305K
Bearing D	25	62	17	1	0.236	12200	26600	305K
Bearing E	30	72	19	1	0.365	15600	33900	306K
Bearing F	30	72	19	1	0.365	15600	33900	306K

Table 1. Bearing specifications

	Transverse Diametral Pitch	Number of Teeth	Pitch Diameter	Bore	Keyway	Style	Hub Dia.	Hub Proj.	Overall length	Direction	Catalog Number
Pinion 1	8	12	1.5	0.750	3/16x3/32	A	1.25	0.50	1.25	Right Hand	HS812R

Gear 2	8	48	6.0	1.0	1/4x1/8	B	2.25	0.50	1.25	Left Hand	HS848L
Pinion 3	8	12	1.6	0.750	3/16x3/32	A	1.25	0.50	1.25	Left Hand	HS812L
Gear 4	8	48	6.0	1.0	1/4x1/8	B	2.25	0.50	1.25	Right Hand	HS848R

Table 2. Updated Gear specifications

Component	Material
Bearing A	Steel
Bearing B	Steel
Bearing C	Steel
Bearing D	Steel
Bearing E	Steel
Bearing F	Steel
Pinion 1	Steel–Hardened
Gear 2	Steel–Hardened
Pinion 3	Steel–Hardened
Gear 4	Steel–Hardened
Input Shaft	AISI 1060
Intermediate Shaft	AISI 1060
Output Shaft	AISI 1060
Gearbox Housing	AISI 1060

Table 3. Components material specifications

Appendix B. Detailed and Assembly Drawings

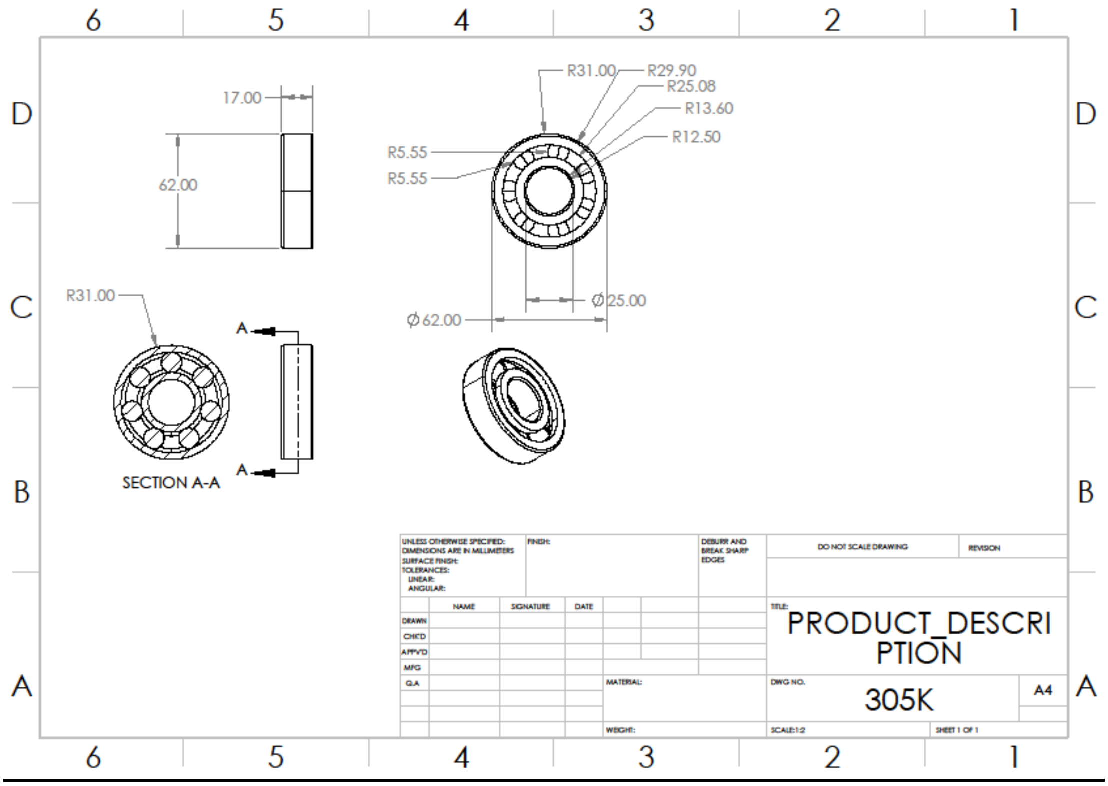


Figure 1.Bearing 305K 2D Drawing

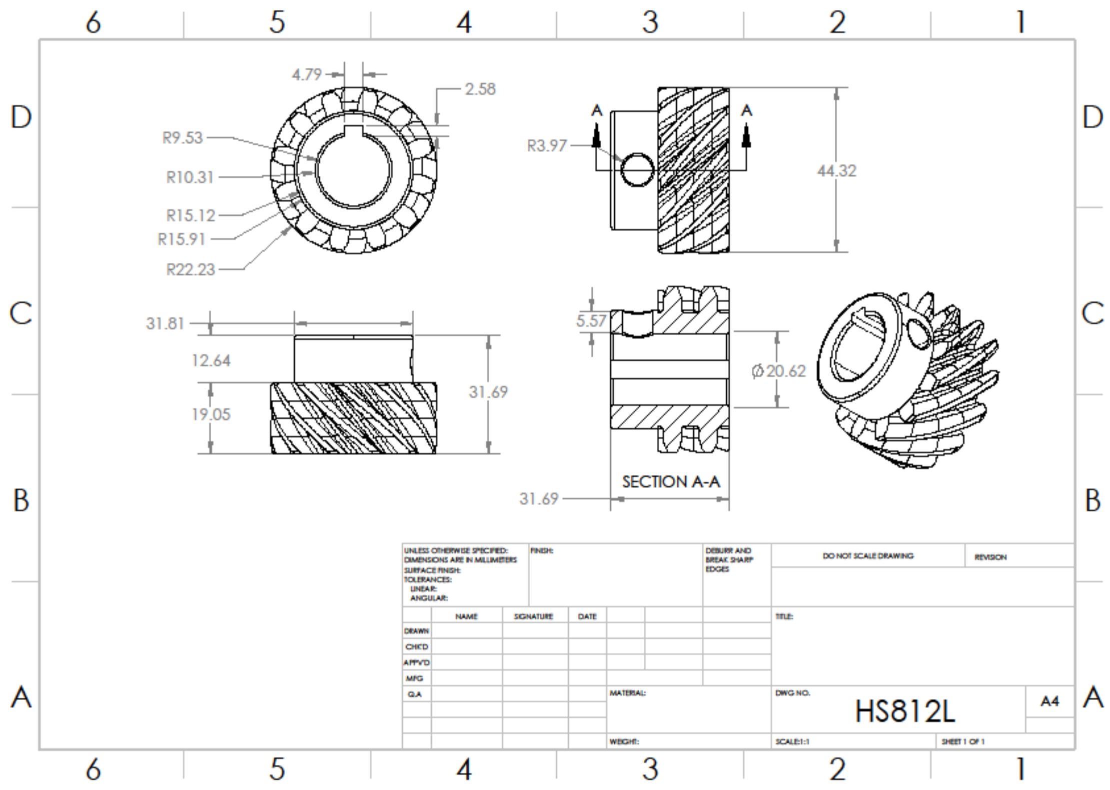


Figure 2.Gear HS812L 2D Drawing

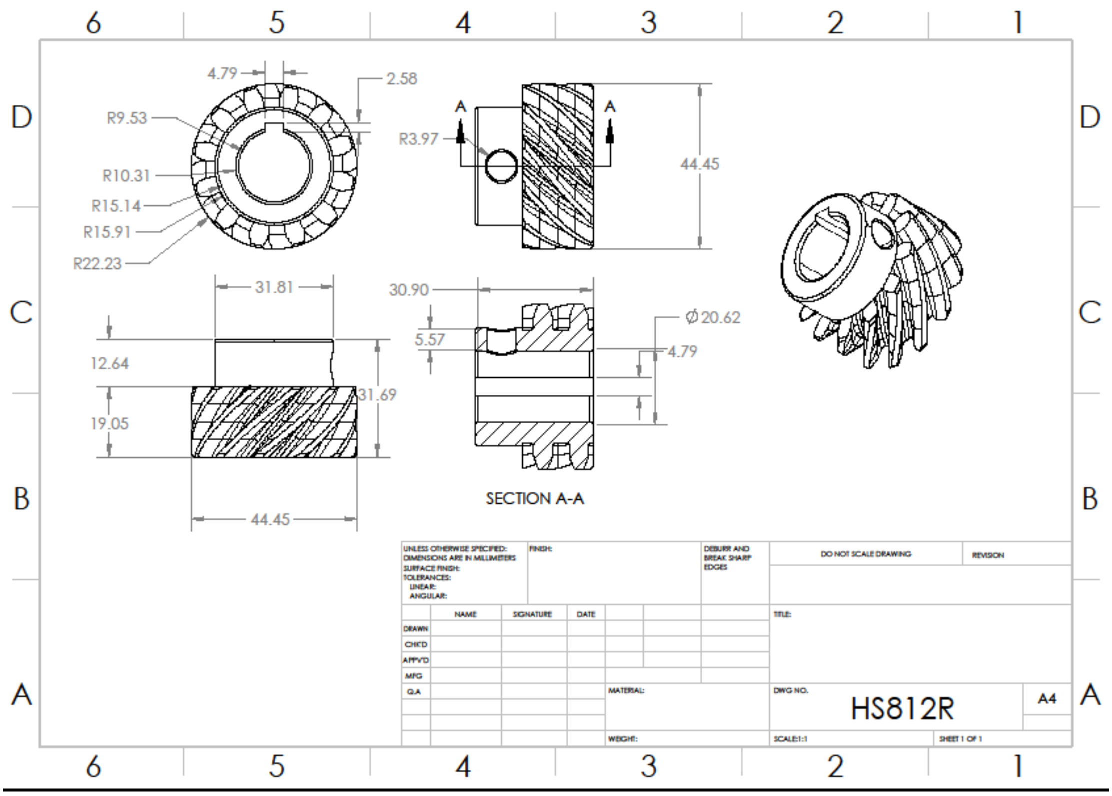


Figure 3.Gear HS812R 2D Drawing

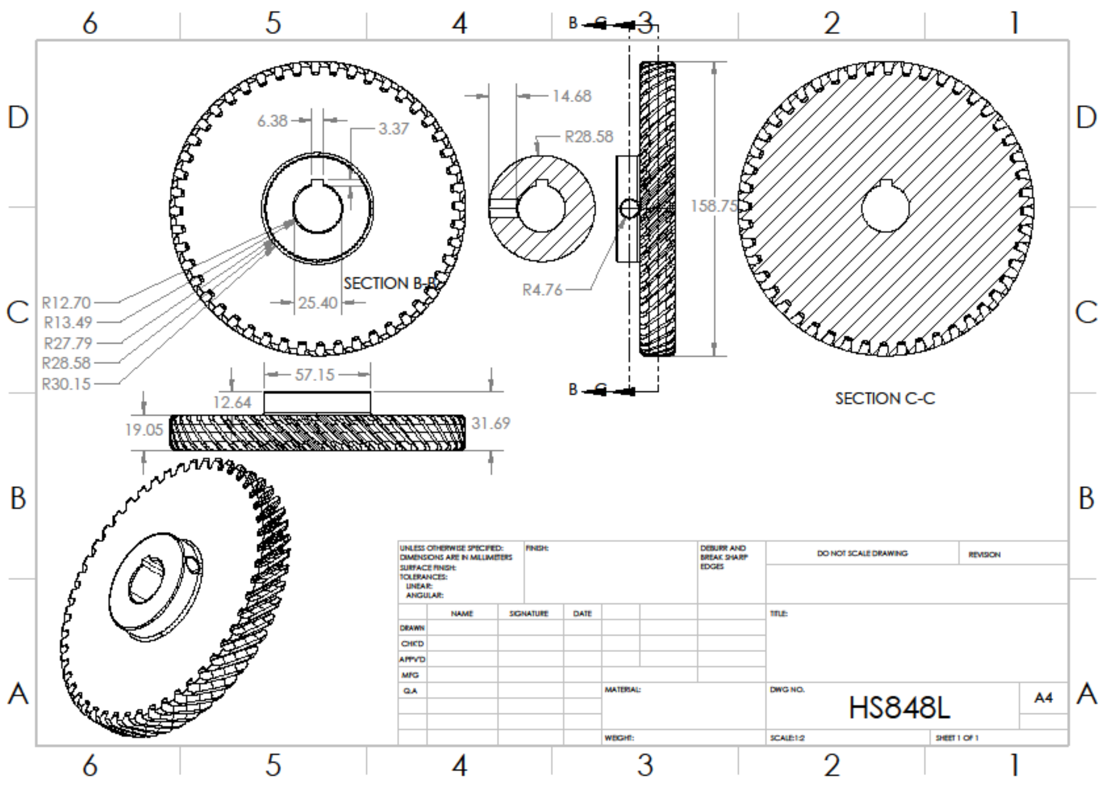


Figure 4.Gear HS848L 2D Drawing

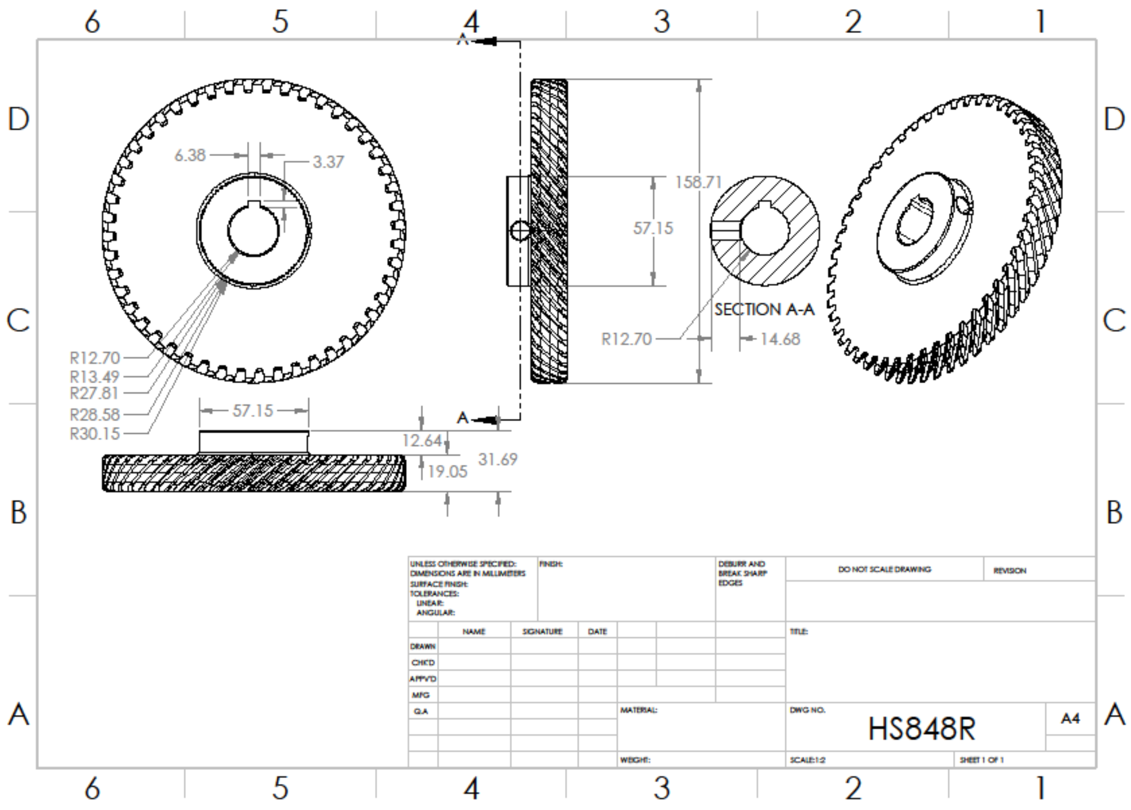


Figure 5.Gear HS848R 2D Drawing

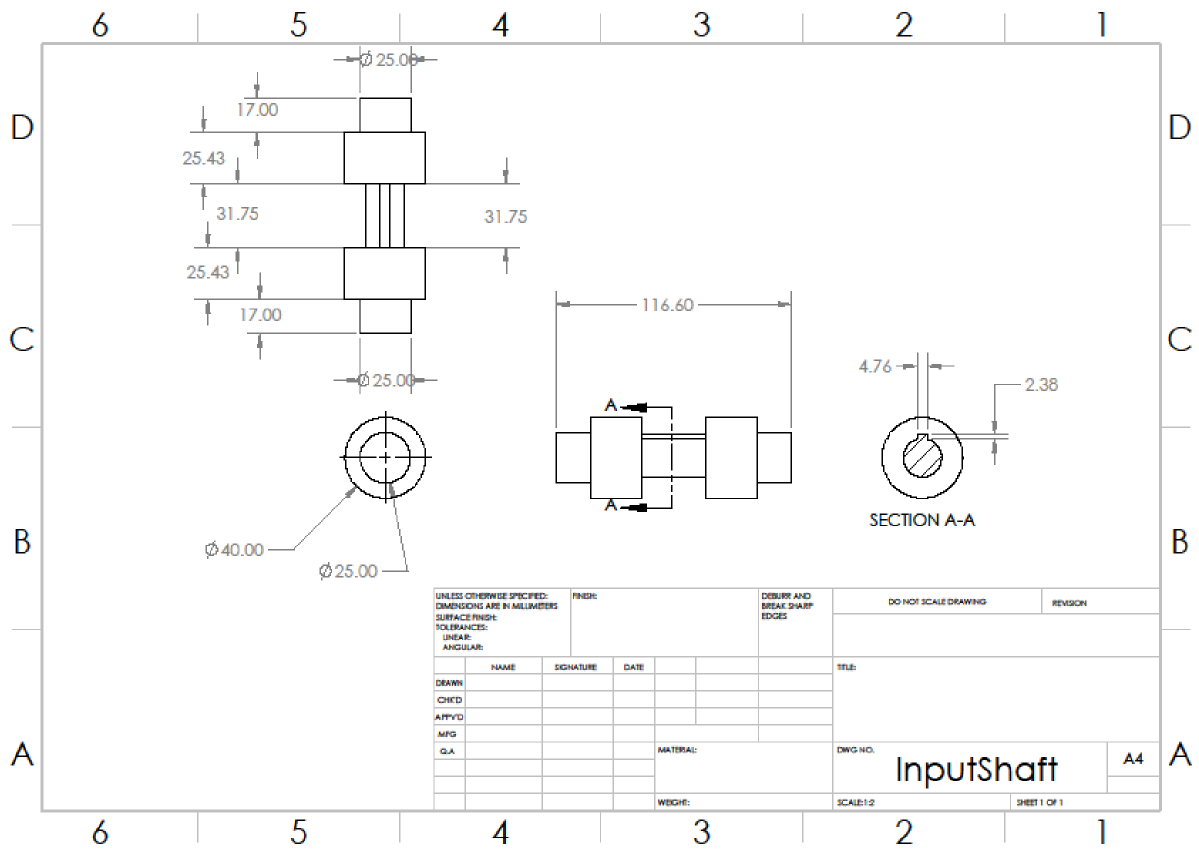


Figure 6.Input Shaft 2D Drawing

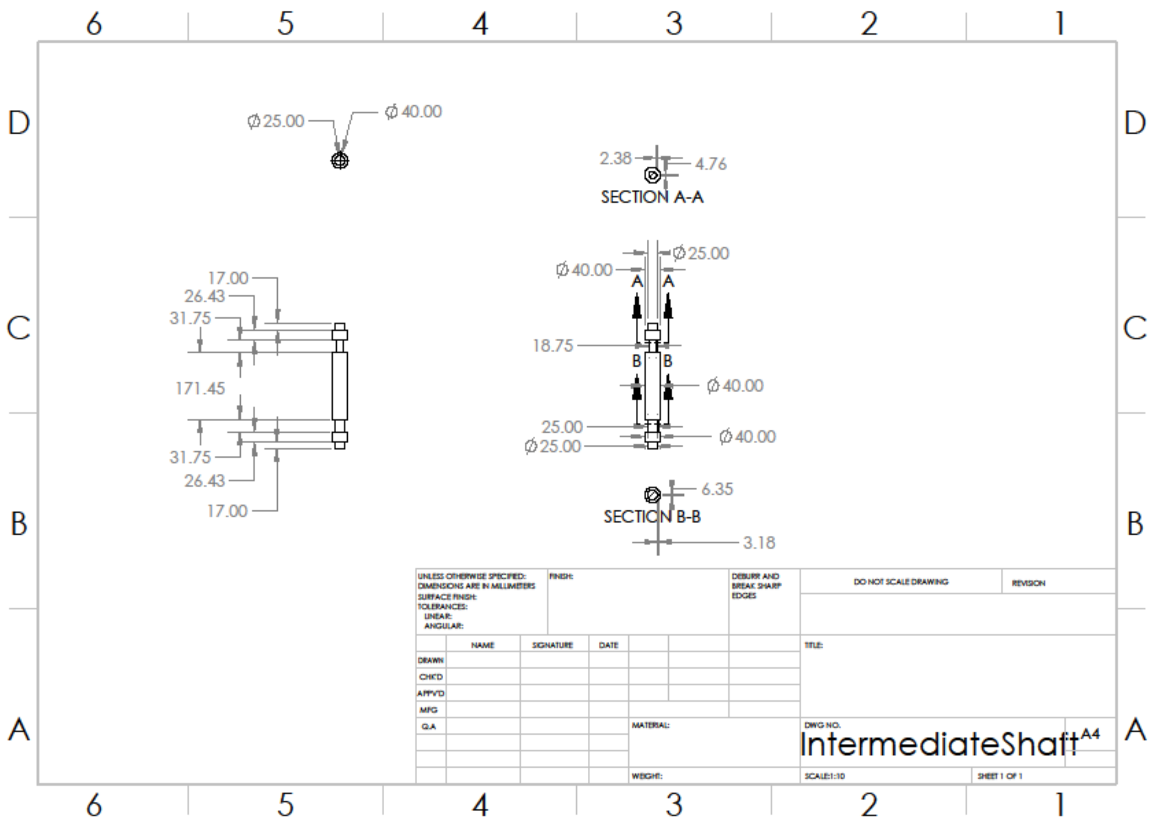


Figure 7.Intermediate Shaft 2D Drawing

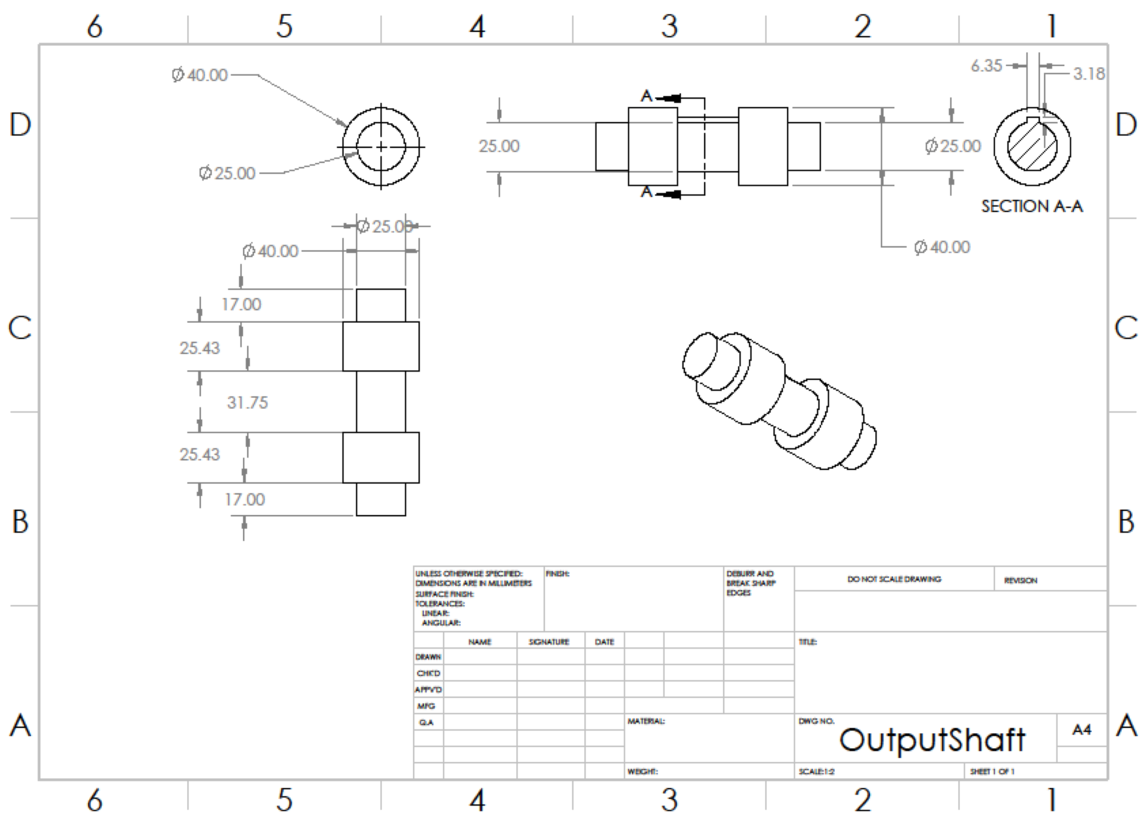


Figure 8.Output Shaft 2D Drawing

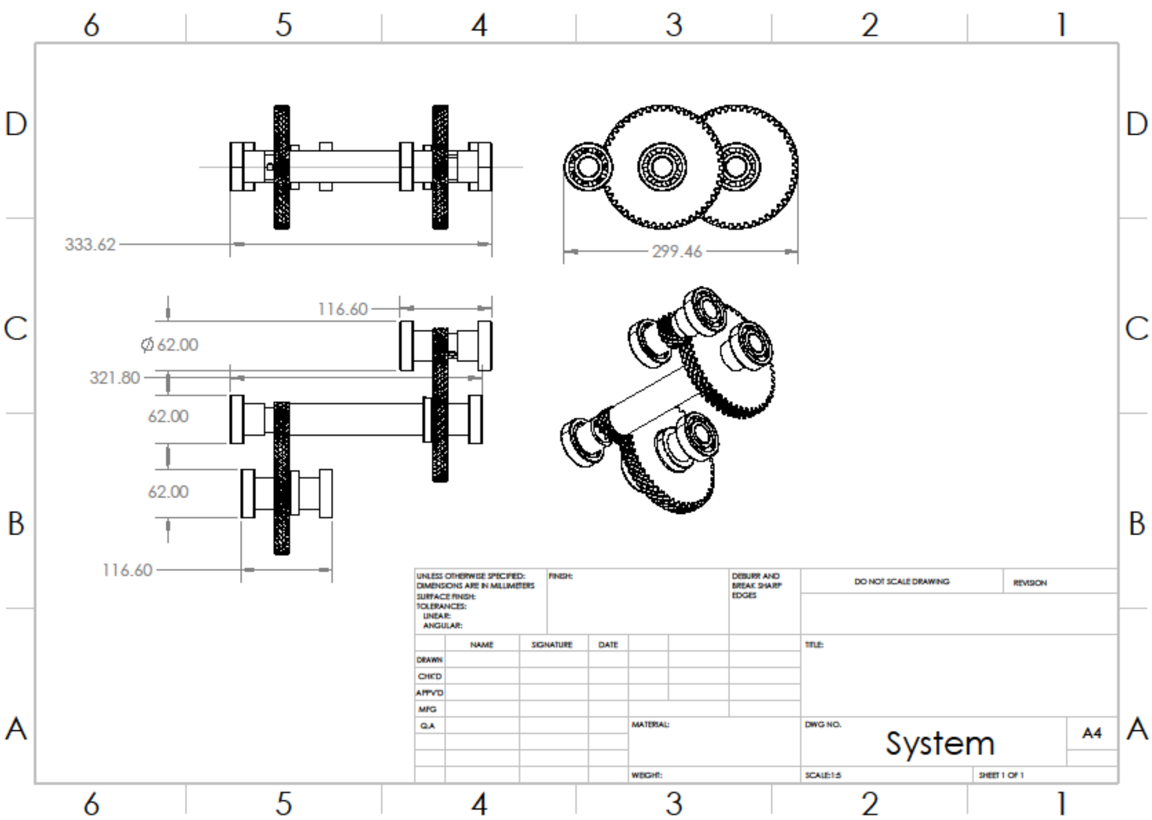


Figure 9.Gear System 2D Drawing

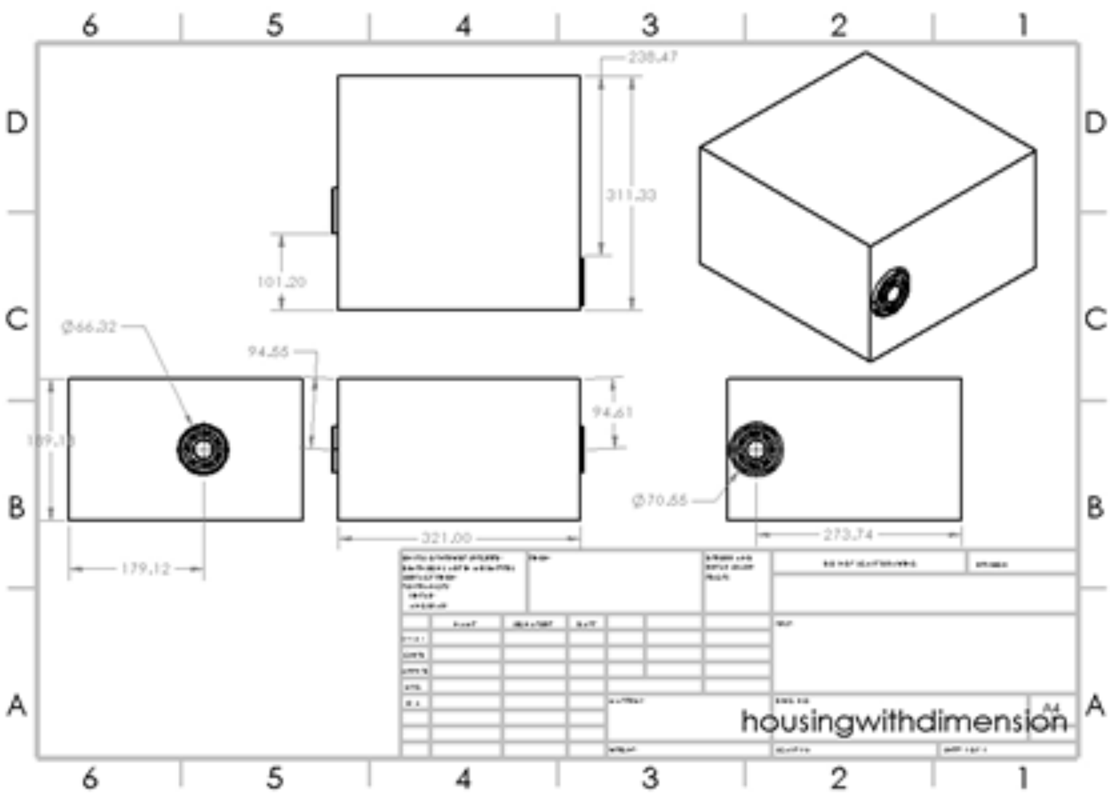


Figure 10.Gear System housing 2D Drawing

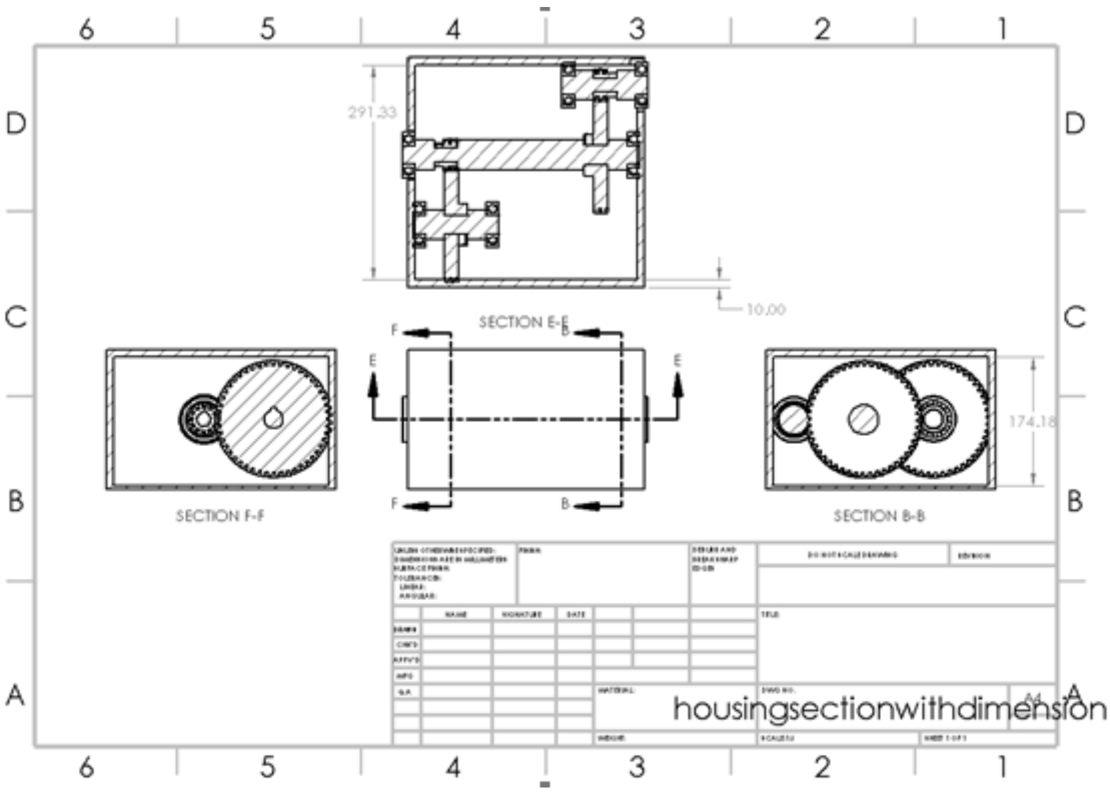


Figure 11. Gear System housing sectional view 2D Drawing