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Group 10

MSE3380 Gearbox Design:
PHASE 2

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Introduction

This report aims to offer an in-depth overview of the design of a two-gear speed-reducing gearbox. The gearbox is intended to receive power from a 3-phase, 30 HP AC induction motor, producing an output speed of 180 rpm and output power of 25 HP. Its function is to decrease speed, boost torque, and ensure optimal efficiency. The report will discuss the analysis of the intermediate shaft, specifically focusing on gear design, shaft design, and bearing selection.

Design Concept

The gearbox will include two sets of spur gears and an input, output, and intermediate shaft. All gears will be enclosed within housing and fit within a 1m x 1m x 1.4m box. Both the input and output shafts will have a free end and protrude from the housing on the other. The intermediate shaft will be supported by bearings on either end. A 3-phase, WEG 30 HP General Purpose AC induction motor will apply torque to the input shaft, operating at 93.6% efficiency and spinning at 1800 rpm.

Phase 1 Update Summary

In our phase 1 report, we calculated the torques and loads on the shaft and gears, determined our gear geometry, selected an AC induction motor that met our specifications, and generated CAD models of our initial design. Since then, minor updates have been made to improve the accuracy of our results, specifically concerning our free-body diagram and torque/force calculations. Our free-body diagram failed to include bearing supports and our torque and force calculations were done using an incorrect conversion of horsepower to watts (horsepower was multiplied by 735 instead of 745.7). See **Appendix 1** for a summary of these changes and their results.

The changes increased the accuracy of our theoretical force and torque calculations but did not change our gear ratio. In addition, the motor we selected still meets our specifications.

Free Body Diagram

Using the calculations from Phase 1, we created a free body diagram of our intermediate shaft (see **Appendix 2)**. We placed bearings at both ends of the intermediate shaft and positioned each pair of gear and pinion with equal spacing throughout the shaft. We assumed a distance of 3.9 inches between each component resulting in a total shaft length of 11.8 inches. The shaft was subjected to different X and Y reaction forces on each bearing, as well as contact forces from the input and output shafts. The forces applied to the intermediate shaft generated a torque and bending moment of 216 Nm and 352.2 Nm, respectively.

We then determined the pitch diameters of the gears, opting for 8 inches for the pinion and 24 inches for the gear. The pinion is subjected to tangential and radial forces of 3466 N and 1262 N, while the gear was subjected to much smaller forces of 1156 N and 421 N, respectively. The Free body diagrams for the gears illustrate the direction and contact angle of the tangential and radial forces (see **Appendix 2**).

Gear Design

Contact Stresses Analysis

Upon inspection, it's evident that the pinion will endure the highest contact stress, given that smaller diameters result in higher contact stress levels. While this holds true, we conducted calculations for both the pinion and the gear to ensure that they both meet the constraint of having a factor of safety greater than 1.5. In our final design, we determined the contact stresses on the pinion and gear to be 22,255.4 MPa and 667,766 MPa, respectively (see **Equation 1**). The calculations of these stresses and stress parameters can be found in **Appendix 3.1 and 3.2**.

$$\sigma_c = C_P \sqrt{K_o K_v K_s K_m C_f \frac{W_t}{dFI}} \tag{1}$$

These stress values resulted in a factor of safety of 12.6 for the gear and 4.12 (see **Appendix 3.3**) for the pinion. The variables with the most significant impact on the factor of safety, while also being easily adjustable, were the diameters and face width. After several iterations, we settled on a diameter of 24 inches for the gear and 8 inches for the pinion. These dimensions allowed us to maintain a gear ratio of 3 while reducing the radial force on the shaft. A face width of 2 inches allowed us to achieve a high factor of safety as well while keeping the gears thin enough to fit comfortably on the shaft.

Furthermore, our ability to achieve high safety factors was also influenced by the selection of material and the Quality Factor Qv. Opting for steel for both the pinion and gear provided a high modulus of elasticity. We estimated a Quality Factor Qv of 7, which is a standard value for commercial gearing and provides enough flexibility to achieve our desired safety factors.

Gear and Pinion Design.

Designing the gears heavily relied on determining the contact stresses applied to them. It's important to note that output rpm calculations and motor selections were covered in Phase 1, so our gear ratio of 3 was already established to meet our desired output shaft speed of 180 rpm. After calculating our diameters using constant stress analysis, we needed to determine the diametral pitch and minimum number of teeth per gear to match the gear specifications with SolidWorks toolbox models.

After using our calculated minimum number of teeth for the pinion and gear found in Phase 1 - which were 15 and 45 respectively - we proceeded to solve for our diametral pitch using the minimum number of teeth on the pinion. However, we found that our chosen diametral pitch was not available in the toolbox model. Consequently, we increased the number of teeth on the pinion to 16 to achieve a diametral pitch of 2. Using this diametral pitch and pitch diameter, we determined the number of teeth on the gear to be 48 (refer to **Appendix 1**).

Shaft Design

Shear, Moment, and Torque Diagrams

Using the calculations found in the free body diagrams for both the shaft and gears (see **Appendix 2**), we were able to create shear force, bending moment, and torque diagrams for the intermediate shaft (see **Appendix 4)**. Due to the presence of radial and tangential forces acting on the shaft, the shear and moment diagrams have been separated into the Y-Z and X-Z planes. In the X-Z plane, there is a maximum shear force of 1926 N in the negative X direction with a positive bending moment of 192.6 Nm. In the Y-Z plane, there is a maximum shear force of 981 N in the negative Y direction and a maximum bending moment of 98.12 Nm. In both the X-Z and the Y-Z planes, the maximum shear force and bending moments have a common maximum located at the keyed connection between the shaft and the gear. The max torque, caused by the tangential force, is in between the pinion and gear with a total magnitude of 352.2 Nm.

Bending Stress and Safety Factors

Due to the shear forces and bending moments on the intermediate shaft, bending stress is induced. To determine the static bending stresses and safety factor of the beam, we need to establish the minimum diameter of the shaft and assess whether it meets the specifications of our design. When determining the minimum diameter for the shaft, we must consider the fatigue stress concentrations for a keyway to radially locate the gear, Kf and Kfs, and the maximum bending moment at those locations (see **Appendix 5)**. To determine the Kf and Kfs values, we assumed a standard radius-to-diameter ratio of 0.02 using notch sensitivity factors of q = 0.9 and qs = 0.92 found in figures 6-26 and 6-27 in the textbook. The stress concentration factors, Kt and Kts, were found to be 2.14 and 3 from table 7-1 in the textbook. Kf and Kfs were determined to be 2.026 and 2.84 respectively using **Equations 2 and 3**.

$$K_f = 1 + q(K_t - 1) \tag{2}$$

$$K_{fs} = 1 + q(K_{ts} - 1) \tag{3}$$

For the intermediate shaft, we decided to use the material AISI 4140 quenched and tempered steel due to its high yield strength of 1640 MPa. We calculated the diameter of the shaft using **Equation 4**.

$$d = \left[\frac{16n}{\pi S_y}\left[4\left(K_f M_a\right)^2 + 3(K_{fs}T_m)^2\right]^{1/2}\right]^{1/3} \tag{4}$$

Using the equation provided, for a bending safety factor of 2, the minimum shaft diameter is determined to be 22.9 mm. This is significantly smaller than the diameters calculated in the Goodman's and bearing diameter calculations, which are 34.8 mm and 65 mm, respectively. To determine the maximum stress in the shaft, using the largest minimum diameter of 65 mm, we apply **Equation 5** (7-15 in the textbook):

$$\sigma'_{\max} = \left[\left(\frac{32K_f M_a}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs}T_m}{\pi d^3}\right)^2\right]^{1/2} \tag{5}$$

The resulting maximum bending stress throughout the intermediate shaft is 36 Mpa resulting in a safety factor of 49, which is significantly higher than required.

Endurance Limit

In order to determine the fatigue factor of safety for the shaft we had to determine the endurance limit, given by **Equation 6.**

$$S_e = k_a k_b k_c k_d k_e S'_e \tag{6}$$

The endurance limit depends on a test specimen endurance limit and several modifying factors, including the surface, size, load, temperature, and reliability factors. The factors were determined by making several assumptions and decisions about the shaft and environment, such as its operation at room temperature, a reliability of 99%, and a machined surface finish. A summary of the factor values and their origins can be found in **Appendix 6**.

Fatigue Safety Factor

Now that we've determined the stress in the shaft before yielding, let's shift our focus to fatigue. To make sure our shaft can withstand repeated stress over time, we need to find out its minimum diameter using the DE-Goodman method. We'll use the same stress concentration factors and notch sensitivity values as before in our static analysis (see **Appendix 5**). Applying **Equation 7** (7-7 from the textbook), modified Goodman's equation, we can calculate the minimum diameter for fatigue.

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{S_e} \left[4\left(K_f M_a\right)^2 \right]^{1/2} + \frac{1}{S_{ut}} \left[3\left(K_{fs} T_m\right)^2 \right]^{1/2} \right\} \right)^{1/3} \tag{7}$$

The modified Goodman's equation results in a minimum diameter of 34.9 mm (see **Appendix 5**). Although this provides us with a diameter that is larger than then the minimum diameter calculated through static yielding analysis, it is significantly smaller than the diameter calculated in our bearing calculations of 65 mm. When determining the factor of safety for fatigue we used the largest diameter between the one calculated by **Equation 4, Equation 7**, and the minimum bearing bore diameter. The largest diameter was the bearing bore diameter, at 40 mm, and when substituted into a rearanged Goodman's equation (see **Equation 8)**, resulted in a fatigue safety factor of 12.9.

$$\mathrm{n} = \frac{d^3}{\frac{1}{S_e}[4\left(K_f M_a\right)^2]^{1/2} + \frac{1}{S_{ut}}[3\left(K_{fs} T_m\right)^2]^{1/2}} \cdot \frac{\pi}{16} \tag{8}$$

Deflection Analysis

We leveraged a tool called SkySiv to analyze the deflection of our gearbox shaft. This tool allows us to effectively simplify our model to something simpler than FEA in SolidWorks while still producing accurate results. The process initially began with importing the properties of the accurately modeled shaft into the simulation environment, ensuring that the geometry realistically represented the real-world application, including the locations where bearings support the shaft and where gears apply force. Careful attention was given to defining the material properties of the shaft, which are crucial for predicting its behavior under load.

This simulation is a 2D simulation, so we are required to run simulations for both the x-y and y-z planes for each beam, and then add the displacements. This setup mimics the operational stresses and strains experienced by the shaft during gearbox operation. Running the simulation, the SkySiv software calculated the deflections and stress concentrations throughout the shaft, highlighting areas prone to excessive bending or deformation. As seen in Appendix 8, we intentionally placed our gears as close as possible to the bearing supports to minimize the moment which resulted in a very minimal displacement. The final maximum displacement was calculated to be 0.00013mm and can be proven in **Appendix 8**.

Bearing Design

Life Calculations

To begin selecting our bearings, we set constraints of choosing 02-series, deep-groove ball bearings with a life of 12,000 hours and reliability of 99%. The process of selecting an appropriate bearing involved calculating the dynamic load rating, C_{10}, which was then used to select a bearing from Table 11-2 in the textbook. The first step was defining a multiple rating of life, x (see **Equation 9**), and reliability adjustment, x (see **Equation 10**), since our reliability is not 90%. The multiple rating of life was calculated using the desired life of 12000 hours, speed of the intermediate shaft, and the rated life. We used a rated life of 10^6 revolutions, a value used by many manufacturers. The reliability adjustment was calculated using the Weibull Parameters corresponding to a rated life of 10^6 revolutions. These calculations are shown below.

n_{shaft} = 1800 rpm/m_g = 1800/3 = 600 rpm

Desired Life, L_D = 12000 hours · 60 · 600 rpm

Rated Life = 10^6 rev

$$x_D = \frac{L_D}{L_R} = \frac{12000 \cdot 60 \cdot 600}{10^6} = 432 \tag{9}$$

Weibull Parameters		
Xo	θ	b
0.02	4.459	1.483

$$\begin{aligned} x_B &= x_o + (\theta - x_o)\,(1 - R)^{1/b} \\ &= 0.02 + (4.459 - 0.02)(1 - 0.99)^{1/1.483} \\ &= 0.22 \end{aligned} \tag{10}$$

We then calculated C_{10} for each bearing using these values, the force applied to each bearing (see reaction forces in **Appendix 2**), and a load factor, a, specific to our intended application. Using table 11-5, we determined the load factor to be 1.1, intended for commercial gearing.

$$a_f = 1.1, \quad C_{10} = a_f \cdot F_r \cdot \left[\frac{x_D}{x_B}\right]^{1/3}$$

$$C_{10A} = 1.1 \cdot 799.8 \cdot \left[\frac{432}{0.22}\right]^{1/3} = 11.035 \text{ kN}$$

$$C_{10B} = 1.1 \cdot 2161.46 \cdot \left[\frac{432}{0.22}\right]^{1/3} = 29.822 \text{ kN} \qquad (11)$$

Bearing Selection

Finally, the C_{10} values for each bearing were compared and used to select the appropriate bearing. Bearing B was used for reference using Table 11-2 since it has the greatest C_{10} value. One of our primary goals when selecting the bearing was selecting one with a bore diameter that was a similar value to the minimum diameter determined by Goodman's equation. This would help ensure that our bearings are sized proportionally to our gears. To do this, we used the table to select a bearing with a load rating close to but greater than 29.822 kN. A summary of the bearing dimensions is shown in **Appendix 7**.

CAD

Once the fundamental requirements for the gearbox, including its torque capacity, gear ratios, and dimensions were calculated, we initiated the CAD process by sketching the preliminary layout. This involved determining the spatial arrangement of gears, shafts, and bearings to ensure optimal functionality and efficiency. Utilizing SolidWorks CAD software, we meticulously modeled each component. We started off by using the Toolbox feature in SolidWorks to model our gears, this allowed us to get the exact specification we calculated for in the prior sections. We ensured we were paying close attention to the specifications of gear profiles, bearings, and shafts to minimize friction and wear. The only thing that isn't shown in this CAD model is the tolerancing required for things like a slip fit of the gears or a press fit of the bearings into the case. In this model, everything is designed to be size-on-size, so this would be one of the first things we would investigate before we send the gearbox off for manufacturing. Also, our gears do not have the cutouts inserted into it in the CAD we submitted. This is because the toolbox gears do not allow edits to be made to them. But in the future, we would make new gears that had the keyhole and shaft diameter cut into them if we were to manufacture them.

One of our goals for this project was to make our gearbox as small as realistically possible. Therefore, we employed advanced tools in SolidWorks like offset entities to offset the case a fixed distance away from the large gears. This ensured that our gearbox case would be no larger than it is required to be. On top of this, we took inspiration from many industries that made gearboxes and used common fastening tools, like c-rings, to constrain the gears axial displacement along the shaft.

Throughout this process, we employed simulations to assess the gearbox's performance (ex. motion analysis) which allowed us to verify our calculations we conducted by hand. This approach not only facilitated the visualization of our final assembly but also enabled precise calculations to optimize the gearbox's functionality. The final step involved assembling all individual parts in the CAD assembly, ensuring seamless integration of all our parts, culminating in a detailed and fully functional gearbox design ready to move on to the final stages before manufacturing.

Conclusion

This report provides a comprehensive analysis and design approach for a two-gear speed-reducing gearbox intended to be driven by a 30 HP AC induction motor, delivering an output of

25 HP. The design incorporates two sets of spur gears and includes detailed considerations for gear design, shaft design, and bearing selection, aiming for optimal efficiency, reduced speed, and increased torque. Adjustments from Phase 1 have refined free-body diagram accuracies and torque/force calculations, with the selection of materials, gear ratios, and dimensions being calculated to ensure our gearbox meets specified requirements.

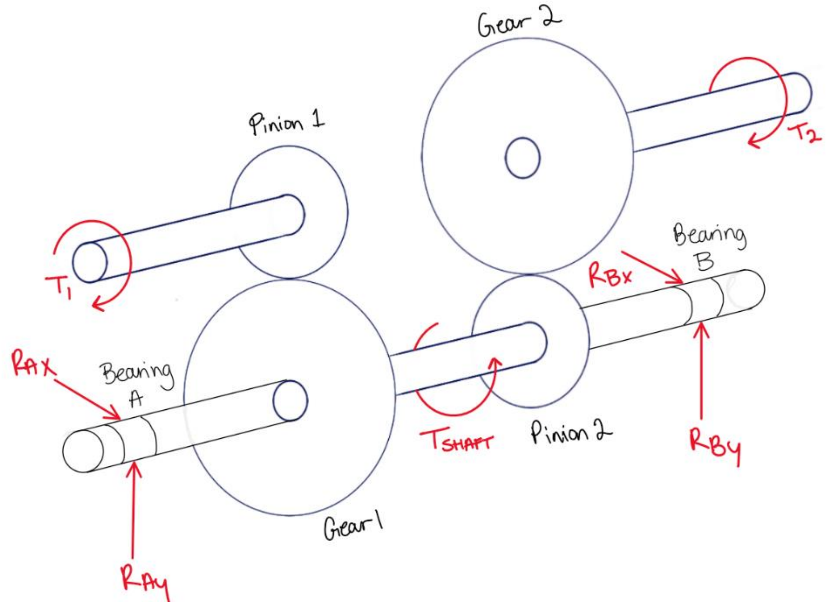
The analysis includes contact stress evaluations, ensuring safety factors greater than 1.5, and detailed considerations of the forces acting on the intermediate shaft resulting in the determination of appropriate gear and pinion sizes, as well as the selection of materials and quality factors to meet our desired safety margins. We also considered shear, moment, and torque diagrams, bending stress considerations, endurance limit calculations, and deflection analysis, resulting in the selection of appropriate bearings to achieve a 12,000-hour lifespan and a reliability of 99%. CAD modeling allowed us to visualize and optimize our gearbox design, incorporating bending simulations to verify our hand calculations.

In conclusion, the gearbox design project successfully achieved its objectives of designing a reliable, efficient, and robust two-gear speed-reducing gearbox.

Appendices

Appendix1 – Phase1 Report Changes

A.1.1 - Revised Free-Body Diagram



A.1.2 - Updated Torque and Force Calculations

$$T_2 = \frac{974.82}{735} \cdot 745.7 = 979.01 \text{ Nm}$$

$$T_1 = \frac{25 \cdot 745.7}{0.9(1800rpm \cdot \left(\frac{2\pi}{60}\right) \cdot 0.936} = 117.4 \text{ Nm}$$

$$m_g = \frac{T_2}{T_1} = \frac{979.01}{117.4} = 8.33 \longrightarrow \sqrt{8.33} = 2.9$$

$$T_{shaft} = T_1 \text{ x } m_g = 117.4 \text{ x } 3 = 352.21$$

$$F_{t1} = \frac{117.4}{0.0381} = 3081.4 \text{ N}$$

$$F_{r1} = F_{t1} \text{ x } \tan(20) = 1121.54 \text{ N}$$

$$F_{t2} = F_{t1} \text{ x mg} = 9244.2 \text{ N}$$

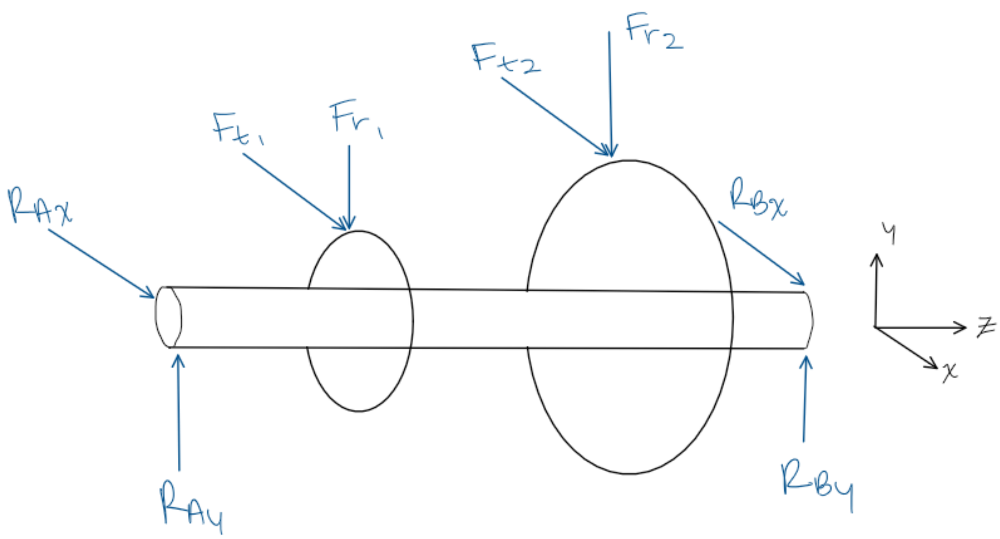
$$F_{r2} = F_{r1} \text{ x mg} = 3364.6 \text{ N}$$

A.1.3 - Pinion and Gear Parameters

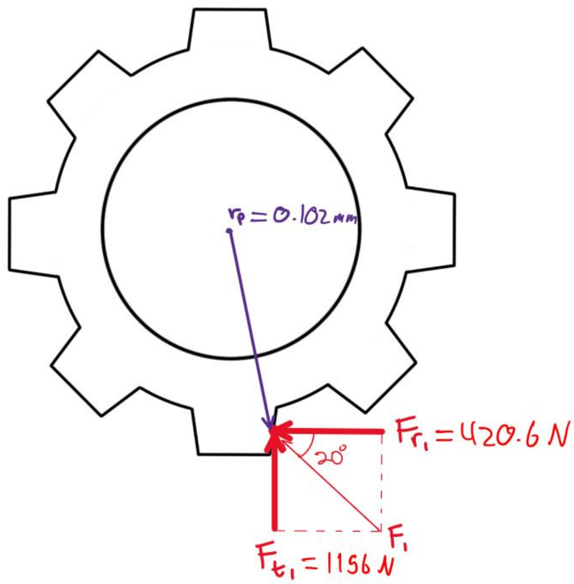
Parameters	Value	Equations
d_p, Pinion Pitch Diameter	8 in	Pre-Selected
d_G, Gear Pitch Diameter	24 in	Pre-Selected
N_{pm} Pinion Minimum Number of Teeth	15	Phase 1 Iteration
N_{Gm} Gear Minimum Number of Teeth	45	Phase 1 Iteration
P, Original Diametral Pitch	1.875 in	$P = \frac{N_{pm}}{d_p} = \frac{15}{8} = 1.875\ in$
N_p Chosen Pinion Number of Teeth	16	
P, New Diametral Pitch	2 in	$P = \frac{N_p}{d_p} = \frac{16}{8} = 2\ in$
N_G Chosen Gear Number of Teeth	48	$N_G = P * d_G = 2 * 24 = 48$

Appendix 2 – Free Body Diagrams

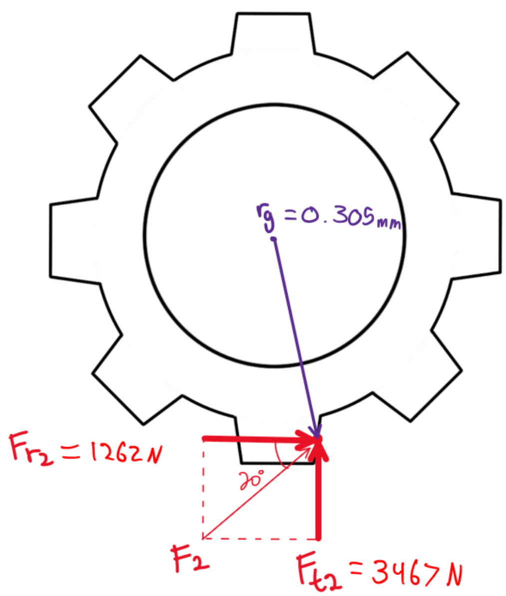
A.2.1 Intermediate Shaft Free Body Diagram



A.2.2 Pinion Freebody Diagram



A.2.3 Gear Freebody Diagram



A.2.4 Updated Forces, Moments and Torques

Loads	Values
Ft1	1155.5(N)

Fr1	420.6(N)
Ft2	3466.4(N)
Fr2	1261.7(N)
Tshaft	352.2 (Nm)
R_{AX}	385.17 (N)
R_{AY}	700.95 (N)
R_{BX}	1925.85 (N)
R_{BY}	981.33 (N)

Appendix 3 - Contact Stress

A.3.1 Km Calculations

Parameter	Gear Value	Pinion Value	Origin
Cpm	1		Based on gear position between bearings.
Ce	1		Non-adjusting gearing.
Cmc	1		Uncrowned teeth.
Cpf	0.0125	0.0375	$\frac{F}{10d} - 0.0375 + 0.0125F$ Where F = 2, dp = 8, and dg = 24
Cma	0.158		$C_{ma} = A + BF + CF^2$ Where F = 2. From 14-9 for commercial gears, A = 0.127, B = 0.0158, C = -0.93e-4
Km	1.171	1.196	$K_m = 1 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right)$

A.3.2 Gear and Pinion Contact Stress & Stress Parameters

Parameter	Gear Value	Pinion Value	Origin
Overload Factor, Ko	1.25		Uniform/constant power
Dynamic Factor, Kv	1.395		$K_v = \left(\frac{A+\sqrt{V}}{A}\right)^B$ $A = 50 + 56(1-B)$ $B = 0.25(12-Q_v)^{2/3}$ Where A = 65.06, B = 0.73, V = 1413.7, and Qv = 7
Size Factor, Ks	1		Not required for regular components, assumed to be 1.
Load Distribution Factor, Km	1.171	1.196	See A.3.1.
Surface Condition, Cf	1		No standards developed, assumed to be 1.
Elastic Coefficient, Cp	2300 psi^0.5		Table 14-8, steel-on-steel contact.
Pitch Diameter, d	24 in	8in	Pre-selected.
Face Width, F	2 in		Pre-selected.
Geometry Factor, I	0.1205		$I = \frac{\cos\phi_t \sin\phi_t}{2m_N} \frac{m_G}{m_G+1}$ Where Φ = 20, Mn = 1, and mg = 3
Transmitted Force, Wt	259.78 lb	779.31 lb	Converted loads from A.2.4 from N to lb.
Contact Stress, σ	22,255.4 MPa	667,766 MPa	See **Equation 1.**

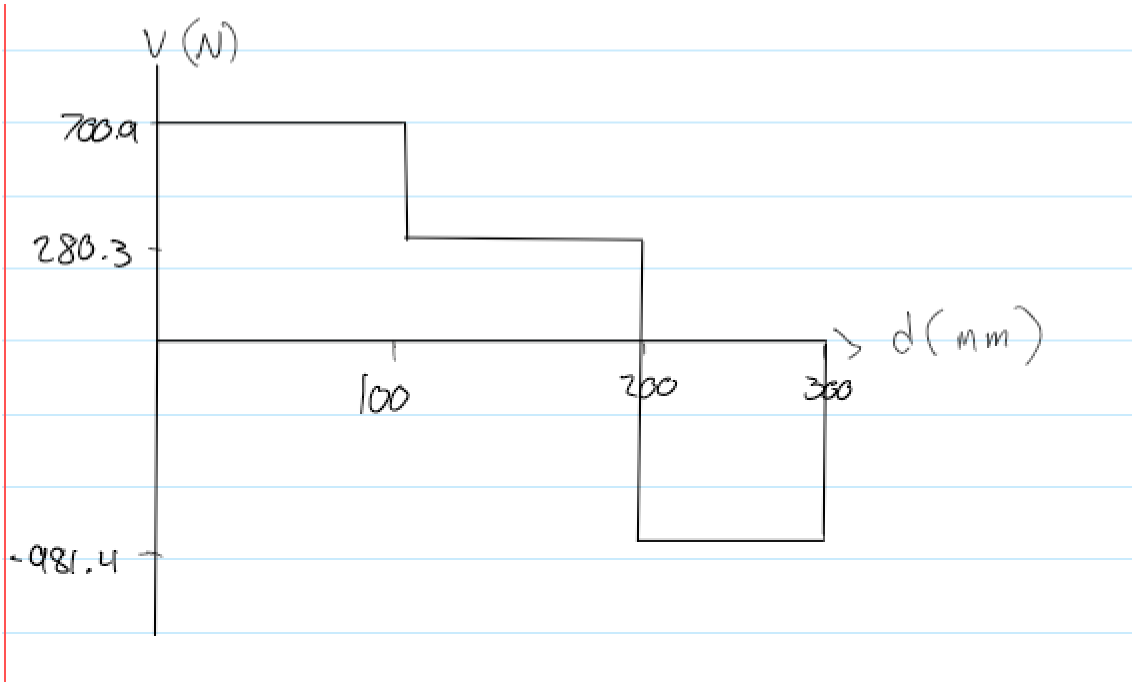
A.3.3 Bending Safety Factor

Parameter	Gear Value	Pinion Value	Origin
Contact Strength, Sc	275000		Table 14-6, carburized and hardened grade 3 steel.
Cycle Factor, Zn	1.025	1	Zn = 1.4488*N^$^{-0.023}$, where N = 10 ^{7} cycles

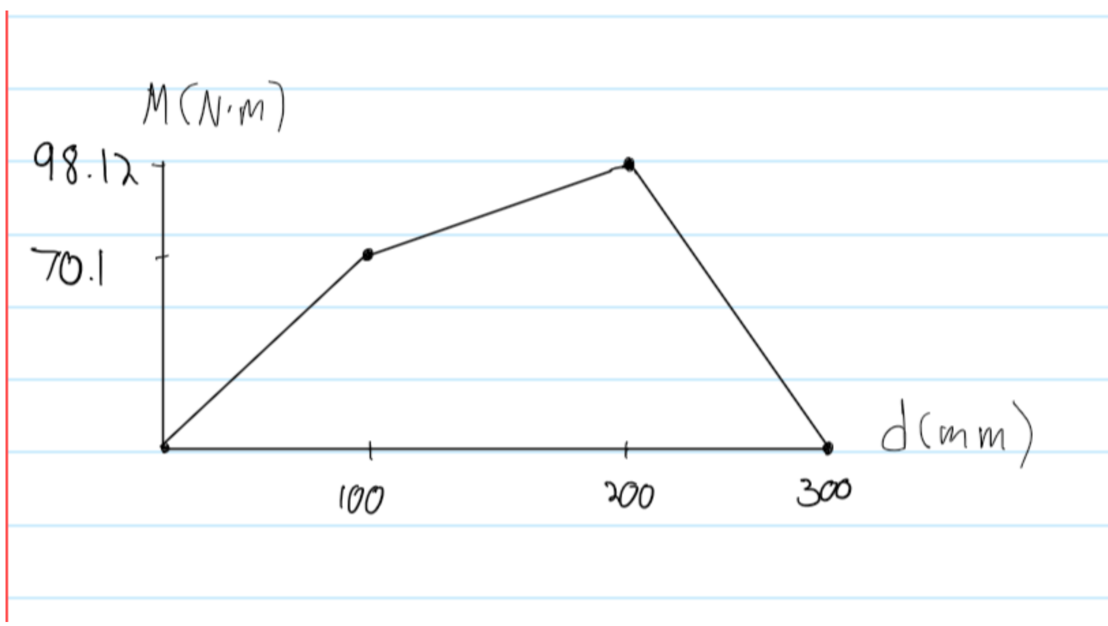
Hardness Ratio, C￼	1		The hardness of the gear and pinion are the same, so CH = 1.
Reliability Factor, K_R	1		Relates to reliability of 99%.
Temp Factor, K_T	1		For temperatures up to 120 deg. C.
Max. Allowable Contact Stress, σ_c	282,042.7 Mpa	275,005.3 MPa	$\sigma_{c,all} = \frac{Z_N C_H}{K_T K_R} S_c$
Safety Factor, S_H	12.67	4.12	$S_H = \frac{\sigma_{c,all}}{\sigma_c}$

Appendix 4 – Shear, Moment, and Torque Diagrams

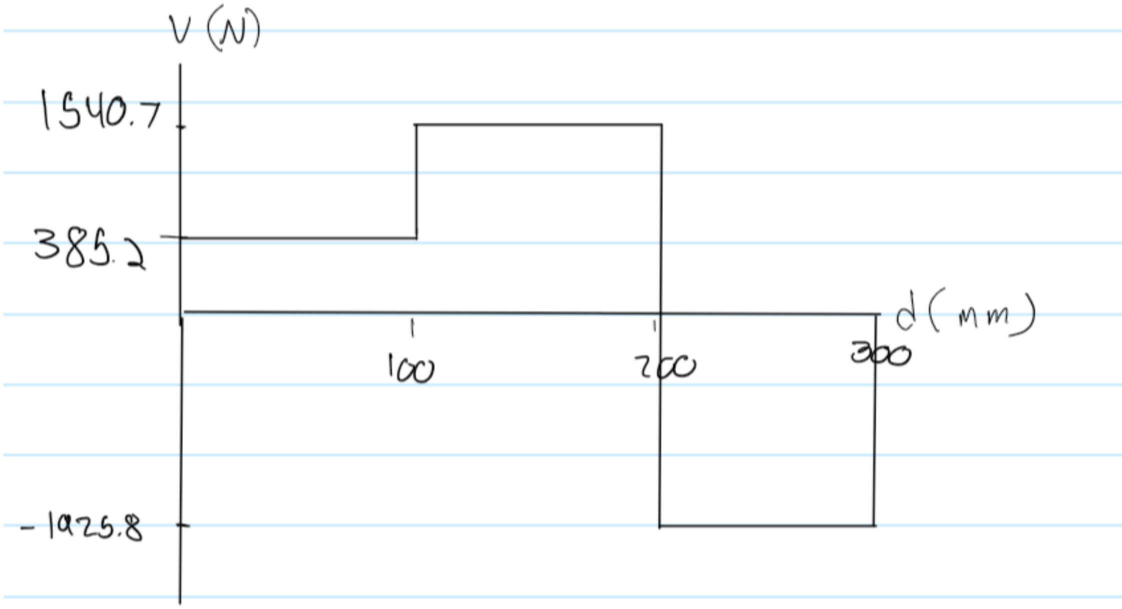
` A.4.1 Shear Force Diagram in the Y- Z Plane



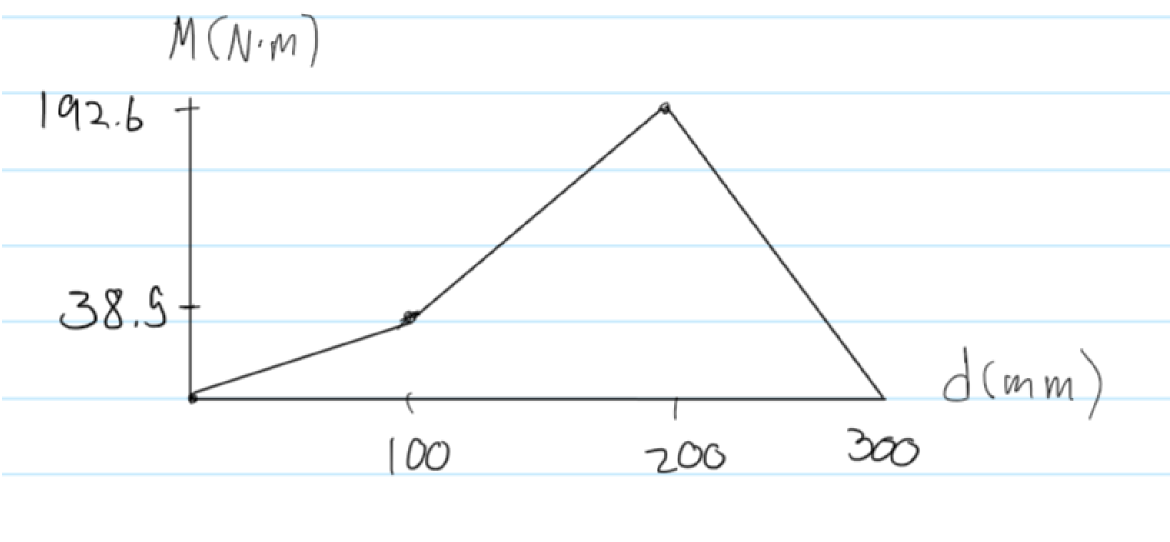
A.4.2 Bending Moment Diagram in the Y-Z Plane



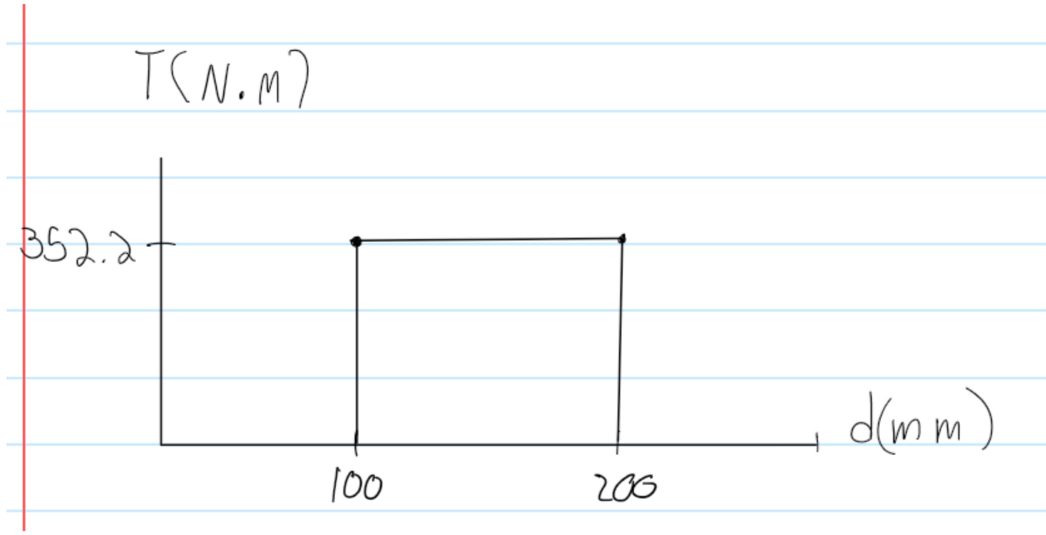
A.4.3 Shear Force Diagram in the X -Z Plane



A.4.4 Bending Moment Diagram in the X –Z Plane



A.4.5 Torque Diagram



Appendix 5 – Bending Stress and Safety Factors

A.5.1 Bending and Fatigue Safety Factor Calculations

Parameter	Values	Origin
Kt	2.14	Table 7-1
Kts	3	Table 7-1
q	0.9	Table 6-26
qs	0.92	Table 6-27
Kf	2.026	See **Equation 2**
Kfs	2.84	See **Equation 3**
Ma (Total Moment)	216 Nm	$\sqrt{M_x^2 + M_y^2}$
Tm (Max Torque)	352.2 Nm	See A.1.1
Sy (Yield Strength)	1640 MPa	Table A-21

d (Minimum Shaft Diameter)	22.9 (mm)	See **Equation 4**
σ (Maximum Bending Stress)	36 MPa	See **Equation 5**
nb (Bending Factor of Safety)	49	σ/Sy
Sut (Ultimate Strength)	1771 MPa	Table A-21
Se (Endurance Limit)	274.7 MPa	See **Equation 6**
nb (Fatigue Factor of Safety)	12.9	See **Equation 8**

Appendix 6 – Endurance Limit

A.6.1 Endurance Limit Calculation

Parameter	Value	Origin
Surface Factor, Ka	0.592	$k_a = aS_{ut}^b$ Where S_{ut} = 1771.1 MPa, a = 3.04, and b = -0.217, corresponding to machined surface.
Size Factor, Kb	0.815	$1.24d^{-0.107}$ Where d = 50.8 mm
Load Factor, Kc	1	K_c = 1 for bending.
Temp Factor, Kd	1	Assuming room temperature.
Reliability Factor, Ke	0.814	Assuming a reliability of 0.99.
Misc. Factor, Kf	1	NA.
Endurance Limit, S_e	274.69 MPa	See **Equation 6**

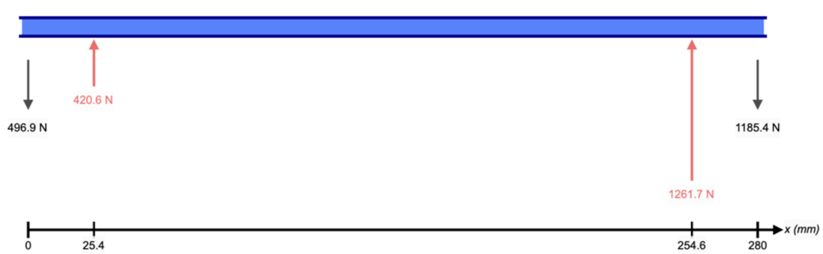
Appendix 7

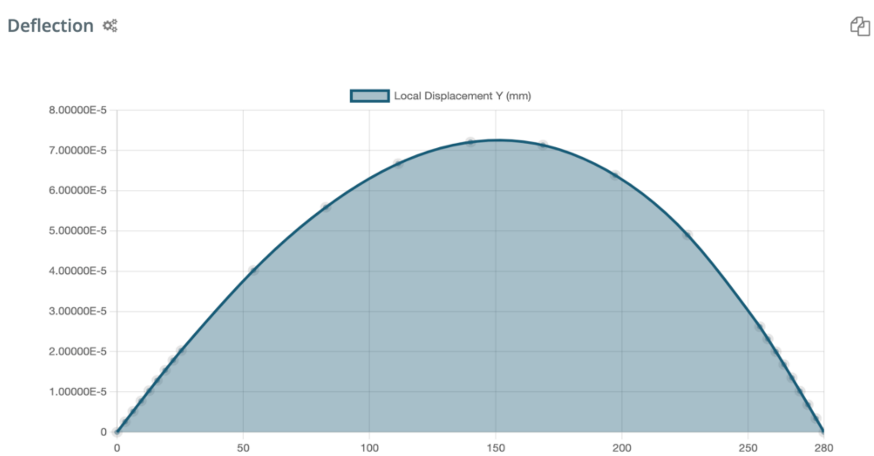
A.7.1 0-2 Series, Deep Groove Ball Bearing Dimensions

Bore (mm)	OD (mm)	Width (mm)	Fillet Rad. (mm)	Shoulder d (mm)	Housing d (mm)	Load Rating, C_{10}
40	80	18	1	46	72	30.7 kN

Appendix 8

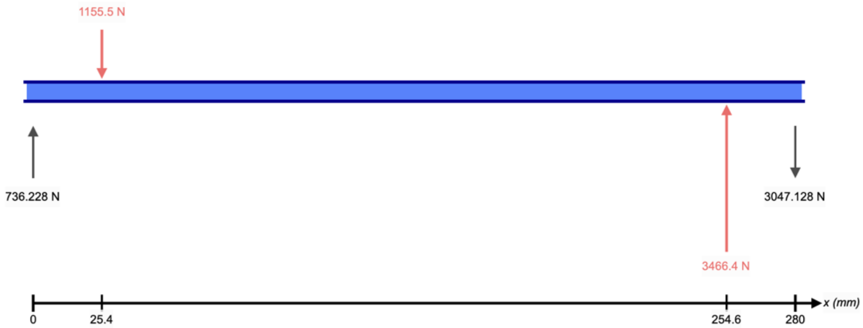
Displacement in the Y direction

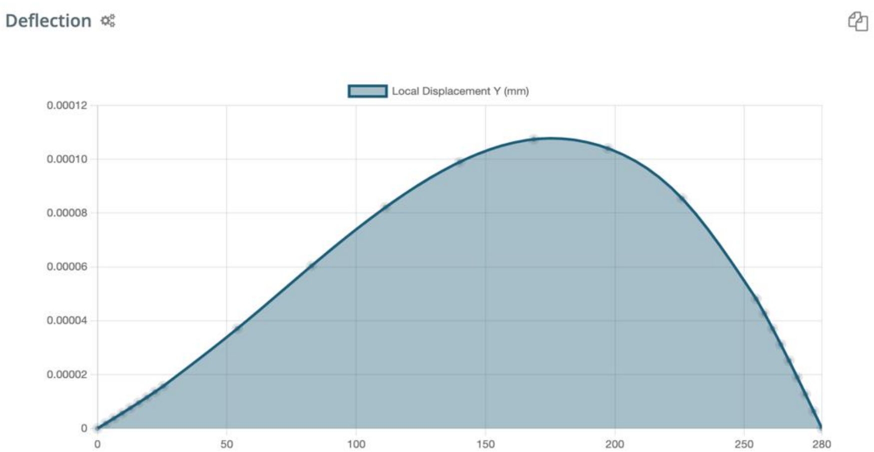




Max Displacement= 0.000075

Displacement in the X direction





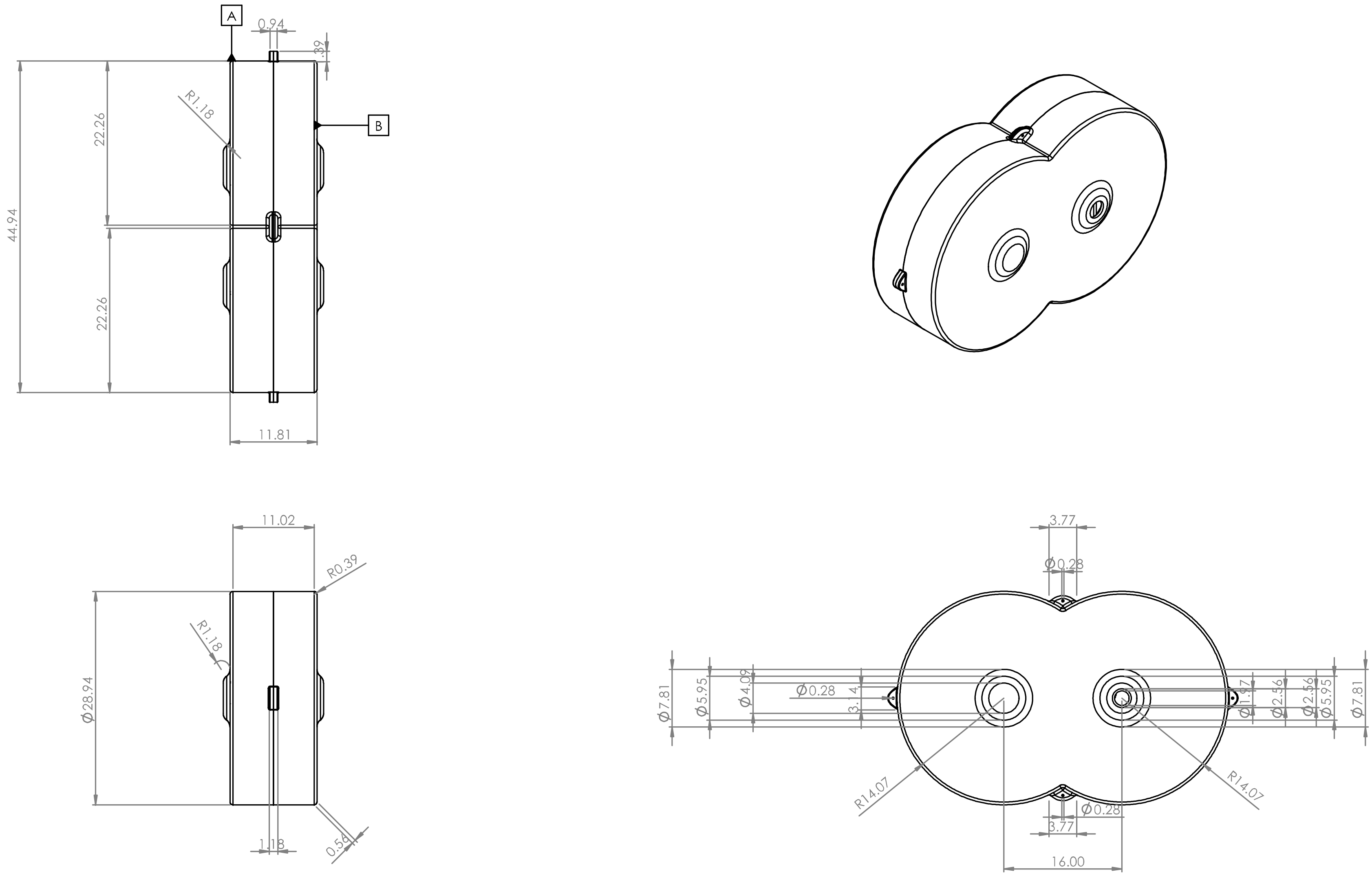
Max displacement = 0.00011 mm

Total displacement = 0.00013 mm

Contribution Table

Name	Contribution
Neil Kirchberger 251210108	Gear, shaft, and bearing analysis and calculations.
Alexander De Rango 251219130	Gear, shaft, and bearing analysis calculations.

Connor Luciani 251207585	CAD modeling, FEA analysis
Anthony Sarauz Aguilar 251217780	Gear, shaft, and bearing analysis and calculations. E-drawings,



	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL: AISI 4140 Q&T REMACHINED

WEIGHT:

DO NOT SCALE DRAWING

REVISION

MSE3380 Group 10

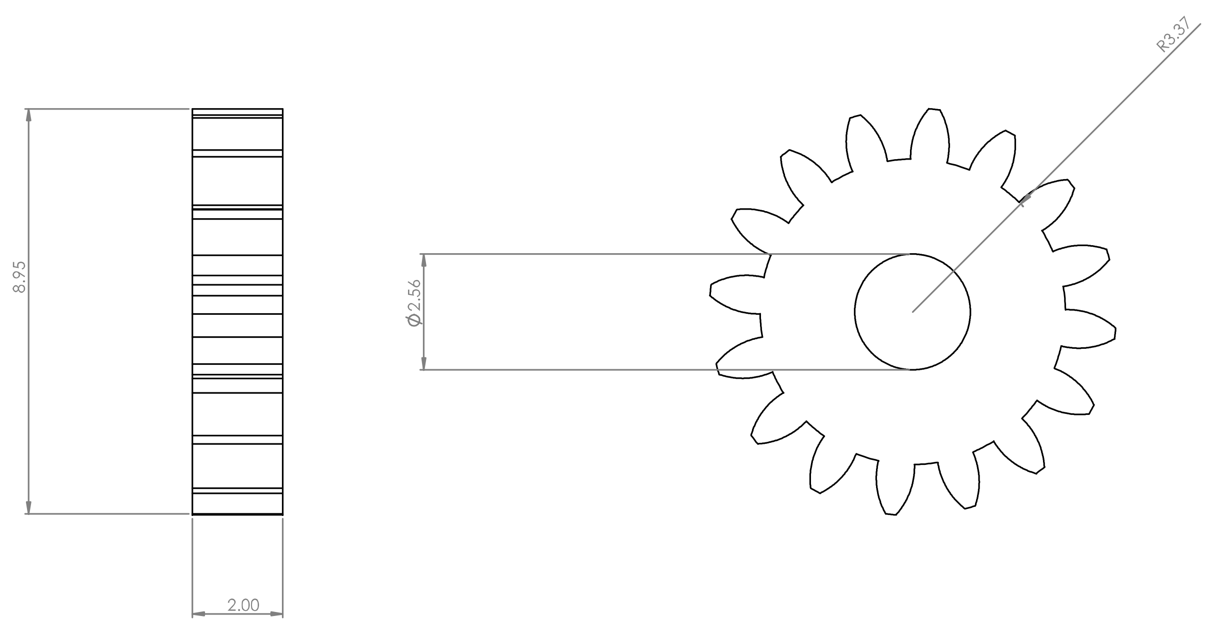
TITLE: Housing V2

DWG NO. V2

A2

SCALE:1:10

SHEET 1 OF 1



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DRAWN				
CHK'D				
APPV'D				
MFG				
Q.A				

DO NOT SCALE DRAWING

REVISION

MSE3380 Group 10

TITLE:

PinionV2

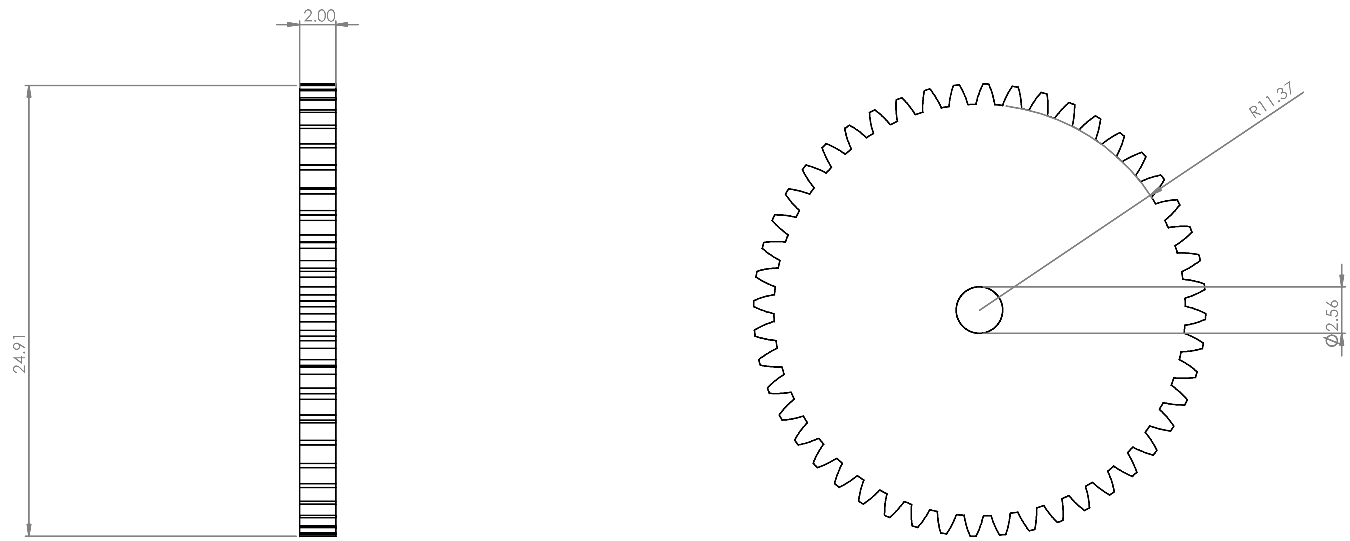
MATERIAL:

AISI 4140 Q&T REMACHINED

DWG NO.

PV2

A2



DO NOT SCALE DRAWING

REVISION

MSE3380 Group 10

TITLE:

Gear V2

NAME	SIGNATURE	DATE
DRAWN		
CHK'D		
APPV'D		
MFG		
Q.A		

MATERIAL:

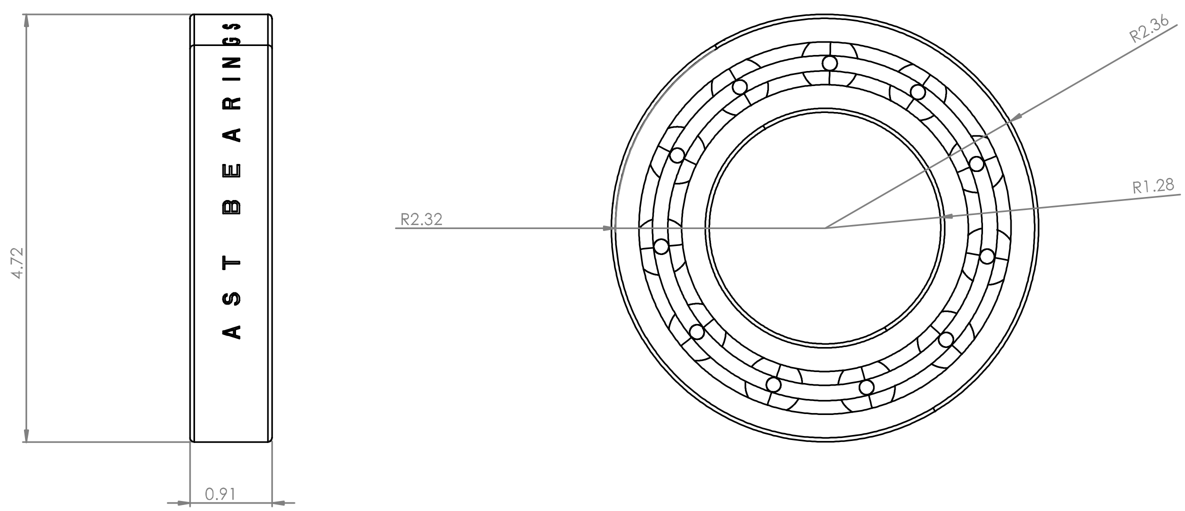
AISI 4140 Q&T REMACHINED

WEIGHT:

DWG NO.

GearV2

A2



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DRAWN				
CHK'D				
APPV'D				
MFG				
Q.A				MATERIAL: AISI 4140 Q&T REMACHINED
				WEIGHT:

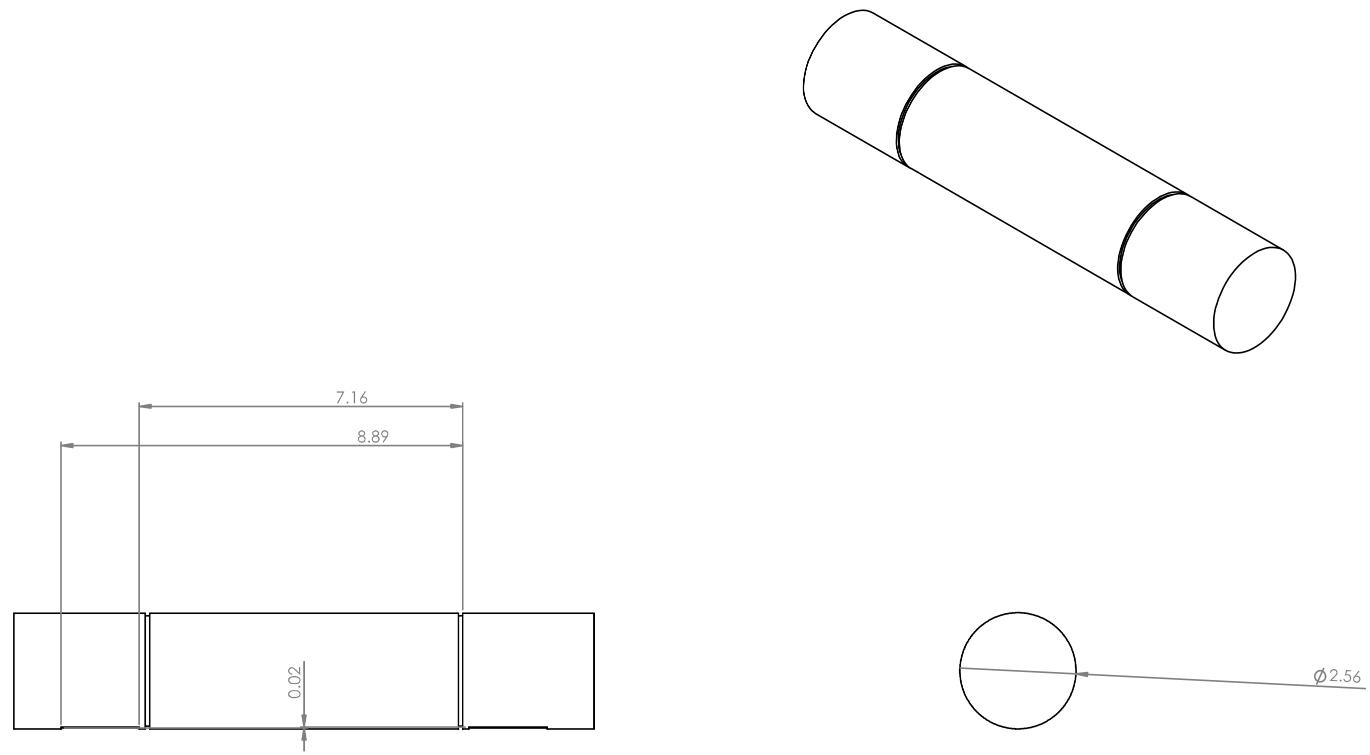
DO NOT SCALE DRAWING | REVISION

MSE3380 Group 10

TITLE: Bearings V2

DWG NO. BV2 | A2

SCALE:1:10 | SHEET 5 OF 5



	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

DO NOT SCALE DRAWING

REVISION

MSE3380 Group 10

TITLE:

Shaft V2

MATERIAL: AISI 4140 Q&T REMACHINED

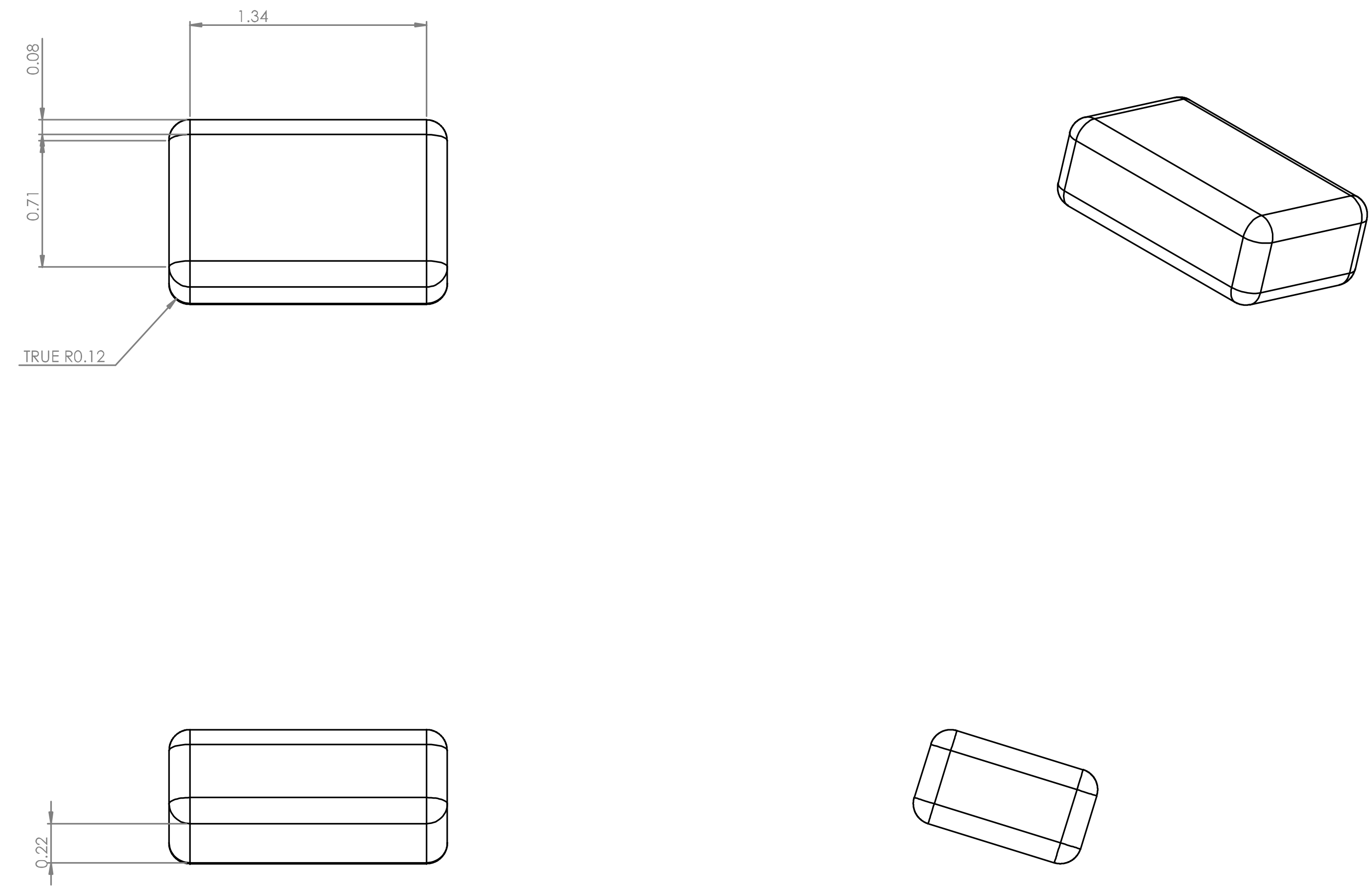
DWG NO. SV2

A2

WEIGHT:

SCALE:1:10

SHEET 4 OF 4



	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

DO NOT SCALE DRAWING

REVISION

MSE3380 Group 10

TITLE: Keys V2

MATERIAL: AISI 4140 Q&T REMACHINED

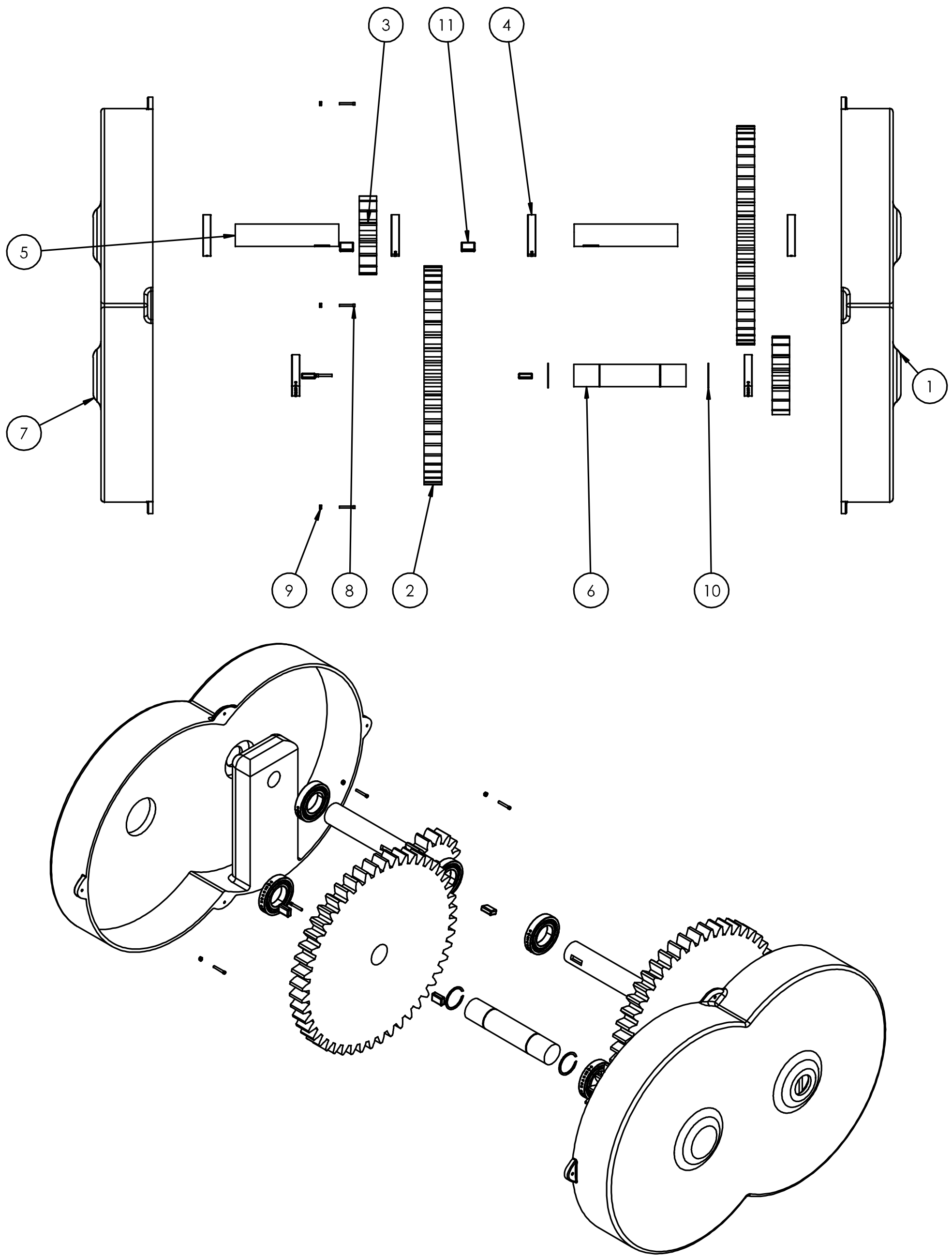
DWG NO. KV2

A2

WEIGHT:

SCALE:1:10

SHEET 2 OF 2



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Gearbox case round	Custom Gearbox Case Half	1
2	Inch - Spur gear 2DP 48T 20PA 2FW --- S48N3.0H2.0L0.5N	48 Teeth, 2 In. Face width, 20 degree pressure angle	2
3	Inch - Spur gear 2DP 16T 20PA 2FW --- S16N3.0H2.0L0.5N	16 Teeth, 2 In. Face width, 20 degree pressure angle	2
4	Bearings	Bearings from ATS	6
5	IN-OUT Shaft	Custom Shaft input-output	2
6	CenterShaft	Custom Center Sheft holding pinion and gear	1
7	MirrorGearbox case round 3.SLDASM	Custom Gearbox Case Half	1
8	Bolt	M6 X 40	5
9	Jam Nut	M6 Jam Nut	4
10	Retaining Rings	Single-Turn Spiral External Retaining Ring	2
11	FinalKey	Custom Key	4

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REVISION
MSE3380 Group 10
NAME | SIGNATURE | DATE
DRAWN
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
WEIGHT:
TITLE:
Exploded View
DWG NO.
EV2
A2
SCALE:1:10
SHEET 6 OF 6