MECH 344 – Machine Element Design

Team 2 – Final Design Project Report

Group 2

Adrien Blanchet (40189935)

Cristian Ayau Castillo (40192155)

Rakeeb Hossain (40211509)

Jessica Meilleur (40212979)

Kayla Yip (40201787)

Section M-MA

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Course instructor: Dr. Amar Sabih

Concordia University

Montreal, QC, Canada

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Gear Data

In this project, the following details are provided by the problem:
$P = 18\ hp$
$n_p = 2100\ rpm$ *from the electric motor*
$n_G = 700\ rpm\ (\pm 5\%)$ *from the general industrial machine. The general industrial machine can run between 665 rpm and 735 rpm.*

Number of teeth in the pinion:
Using $N_p = 18\ teeth$

Velocity Ratio:

$$VR = \frac{n_p}{n_G} = \frac{2100}{700} = 3$$

Number of teeth in the gear from $\boldsymbol{N_G = N_p \times VR}$:
$N_G = (18)(3) = 54\ teeth$

Pitch diameters of pinion and gear:

$$D_p = \frac{N_p}{P_d} = \frac{18}{6} = 3\ in$$

$$D_G = \frac{N_G}{P_d} = \frac{54}{6} = 9\ in$$

Center distance:

$$\frac{D_p + D_G}{2} = \frac{3+9}{2} = 6\ inches$$

Pitch line velocity on pinion and gear:

$$v_t = \frac{D_p}{2} \times n_p = \frac{3\ inches}{2}(2100\ rpm)\left(\frac{2\pi\ rad}{1\ rev}\right)\left(\frac{1\ ft.}{12\ in.}\right) = 1649.34\ ft/min$$

$$v_t = \frac{D_G}{2} \times n_G = \frac{9\ inches}{2}(700\ rpm)\left(\frac{2\pi\ rad}{1\ rev}\right)\left(\frac{1\ ft.}{12\ in.}\right) = 1649.34\ ft/min$$

Design Power:

$$P_{des} = K_o P = (1.40)(18\ hp) = 25.2\ hp$$

Torque on pinion and gear:

$$T_p = \frac{P_{des}}{n_p} = \frac{25.2\ hp}{2100\ rpm}\left(\frac{33000\ \frac{lb * ft}{min}}{1\ hp}\right)\left(\frac{1\ rev}{2\pi\ rad}\right)\left(\frac{12\ in.}{1\ ft.}\right) = 756.3\ lb * in$$

$$T_G = \frac{P_{des}}{n_G} = \frac{25.2\ hp}{700\ rpm}\left(\frac{33000\ \frac{lb*ft}{min}}{1\ hp}\right)\left(\frac{1\ rev}{2\pi\ rad}\right)\left(\frac{12\ in.}{1\ ft.}\right) = 2268.91\ lb*in$$

Bending Stress:

To calculate the bending stress, the following equation is used:

$$s_t = \frac{W_t P_d}{FJ} K_o K_s K_m K_B K_v$$

Where,

- $W_t = \frac{2T_p}{D_p} = \frac{2*(\ 756.3\ lb*in)}{3\ in} = 504.2\ lb$ *(Will be the same for both pinion and gear)*
- Wr = Wt • tan (**ϕ**) = 504.2 tan (20) = 91.76 lb
- $P_d = \frac{N_p}{D_p} = \frac{18\ teeth}{3\ in} = \ 6\ in^{-1}$ *(also in description)*
- $\frac{8}{P_d} < F < \frac{16}{P_d}$, using nominal size: $F = \frac{12}{P_d} = \frac{12}{6\ in^{-1}} = 2\ in$
- $J_p = 0.32$ & $J_G = 0.4$ *(From: FIGURE 9–10)*
- $K_m = 1 + C_{pf} + C_{ma}$
  - $For\ 1.0 \leq F \leq 15.0$, $C_{pf} = \frac{F}{10D_p} - 0.0375 + 0.0125F$
  - $C_{pf} = \frac{2\ in}{10*(3\ in)} - 0.0375 + 0.0125*(2\ in) = 0.05417$
  - $For\ commercial\ enclosed\ gear\ units,\ C_{ma} = 0.127 + 0.0158F - 1.093*10^{-4}F^2$
  - $C_{ma} = 0.127 + 0.0158*(2\ in) - 1.093*10^{-4}(2\ in)^2 = 0.15816$
  - $K_m = 1 + 0.05417 + 0.15816 = 1.21233$

- $K_s = 1$ *(Based on Table 9-2)*
- $K_B = 1$ *(Solid Gear)*
- $K_o = 1.40$ *(Will not be applied as it is already applied in the design power)*
- $K_v = 1.33$ *(Based on Figure 9-16)*

Therefore,

$$s_{t_p} = \frac{(504.2\ lb) * (6\ in^{-1})}{(2\ in) * (0.32)}(1) * (1.21233) * (1)(1.33) = 7.62\ ksi$$

And,

$$s_{t_G} = s_{t_p}\frac{J_p}{J_G} = (7.62\ ksi) * \frac{0.32}{0.4} = 6.1\ ksi$$

Contact Stress Number:

To calculate the contact stress number, the following equation is used:

$$s_{c_p} = C_p\sqrt{\frac{W_t K_O K_s K_m K_v}{F D_P I}}$$

Where,

- $I = 0.100$
- $C_p = 2300$ **(Assuming using Steel)**

Therefore,

$$s_{c_p} = 2300 * \sqrt{\frac{(504.2\ lb) * (1) * (1.21233) * (1.33)}{(2\ in) * (3\ in) * (0.100)}} = 84.66\ ksi$$

Next, we need to determine the material for the pinion and the gear such that the contact and bending stresses fall below the allowable stress for such a given material.

Such that,

$$s_t < s'_{at} = s_{at}\frac{Y_N}{(SF)(K_R)}$$

And,

$$s_c < s'_{ac} = s_{ac}\frac{Z_N}{(SF)(K_R)}$$

Then,

$$s_{at} > s_t\frac{(SF)(K_R)}{Y_N}$$

And ,

$$s_{ac} > s_c\frac{(SF)(K_R)}{Z_N}$$

Where,

- $SF = 1$ *(typical and not specified)*
- $K_R = 1$ *(99% reliability)*
- $Y_N = f(N_c)$
    - $N_c = (60)(L)(n)(q)$
    - $L = 14{,}000\ h;\ n = 2100\ rpm;\ q = 1$
    - $N_c = (60)(14{,}000\ h)(2100\ rpm)(1) = 1.764 * 10^9\ cycles$
    - $Y_N = 1.3558N_c^{-0.0178} = 1.3558 * (1.764 * 10^9\ cycles)^{-0.0178} = 0.928$
- $Z_N = f(N_c)$
    - $Z_N = 1.4488N_c^{-0.023} = 1.4488 * (1.764 * 10^9\ cycles)^{-0.023} = 0.888$

$$s_{at_p} > (7.62\ ksi)\frac{(1)(1)}{0.928}$$
$$s_{at_p} > 8.21\ ksi$$

And,

$$s_{ac_p} > (84.66\ ksi)\frac{(1)(1)}{0.888}$$
$$s_{ac_p} > 95.34\ ksi$$

From this, we can determine that the pinion could be made of a **Grade 1 Through-Hardened steel** with a Brinell hardness greater than or equal to **200 HB**.

Material meeting requirement: **SAE 1040 OQT 400 (262HB),** $s_{ac} = 115\ ksi$

Since the pinion undergoes larger stresses than the gear, it is safe to assume the same material will fulfill the allowable stress requirements for the gear as well.

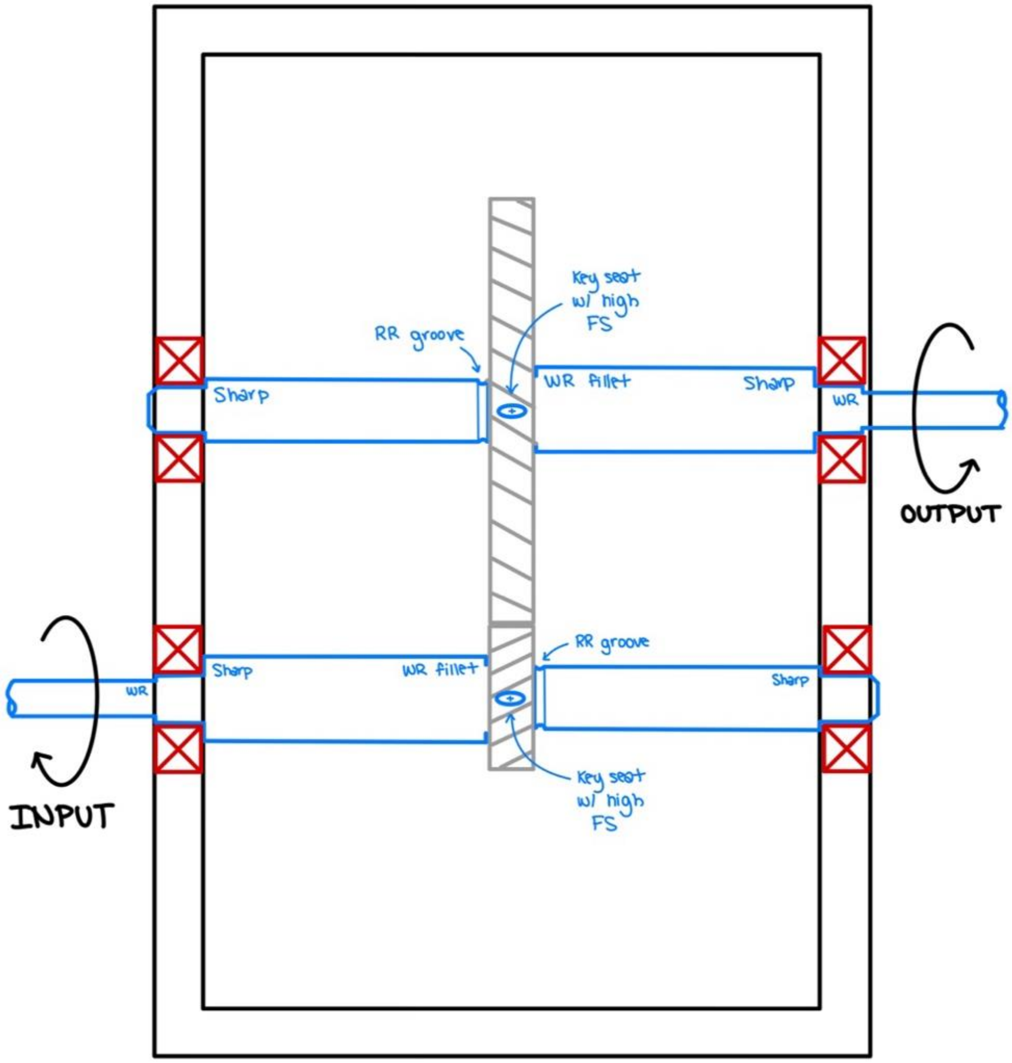


Figure 1: Shaft Design Sketch

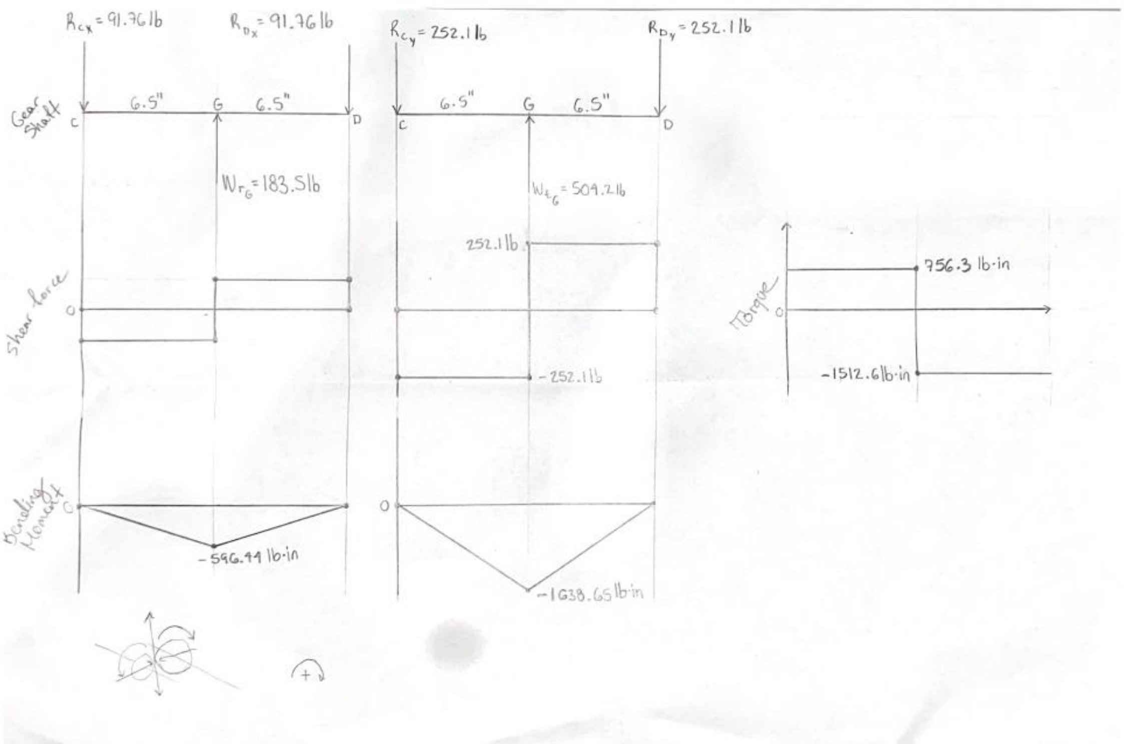


Figure 2: Shear Force and Moment Diagrams in x and y directions for the Gear Shaft.

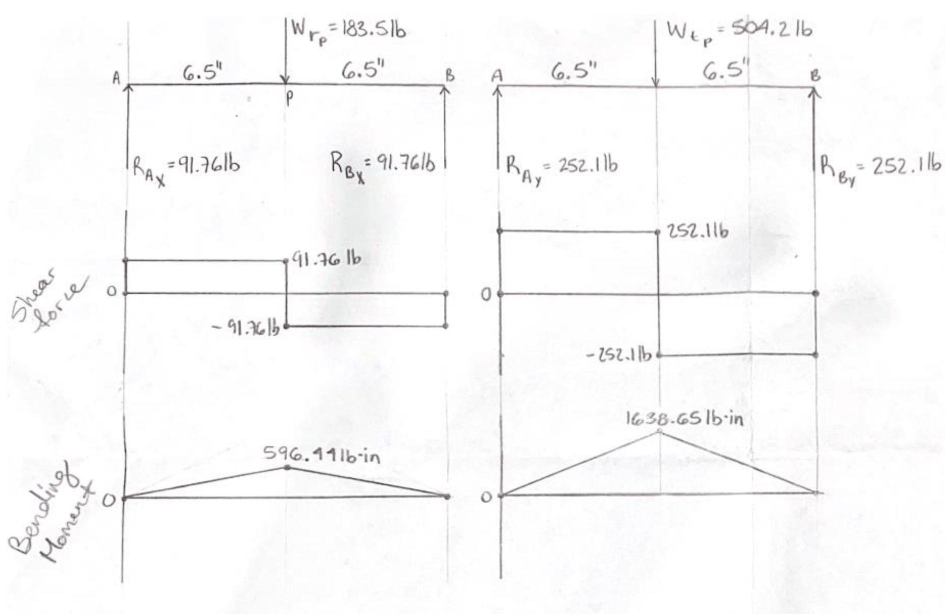


Figure 3: Shear Force and Moment Diagrams in x and y directions for the Pinion Shaft.

Bending moment at the gear is the resultant of the moments in both the x and y planes:

$$Mgear = \sqrt{(Mgear_x)^2 + (Mgear_y)^2}$$

$$Mgear = \sqrt{(596.44)^2 + (-1638.65)^2}$$

$$Mgear = 1\ 743.82\ lb \cdot in$$

To find the diameter of the shaft, the following equation is used:

$$D = \left[\left(\frac{32N}{\pi}\right)\sqrt{\left(\frac{K_t M}{s'_n}\right)^2 + \frac{3}{4}\left(\frac{T}{s_y}\right)^2}\right]^{\frac{1}{3}}$$

Material used is SAE 1040 OQT 400, therefore s_y = 87 000 psi and s_{ut} = 113 000 psi

For D_1:

Bending moment to the left of the gear = 0.

The final diameter must be increased by 6% due to the presence of a retaining ring.

$$D_1 = \left[\left(\frac{32(2.5)}{\pi}\right)\sqrt{\frac{3}{4}\left(\frac{2\ 268.91}{87\ 000}\right)^2}\right]^{\frac{1}{3}}$$

$$D_1 = 0.83\ in * 1.06$$

$$\boldsymbol{D_1 = 0.88\ in}$$

For D_2:

K_t is 2 due to the well-rounded fillet needed at the gear.

s'_n must be found to solve for diameter.

$$s'_n = s_n * c_r * c_s$$

S_n: Figure 5-11

Cr: Table 5-3 (for 99% reliability)

Cs: Assume 0.8

$$s'_n = 42\,500 * 0.81 * 0.8$$

$$s'_n = 27\,540\ psi$$

Now we can use the same equation once again:

$$D = \left[\left(\frac{32N}{\pi}\right)\sqrt{\left(\frac{K_t M}{s'_n}\right)^2 + \frac{3}{4}\left(\frac{T}{s_y}\right)^2}\right]^{\frac{1}{3}}$$

$$D_2 = \left[\left(\frac{32(2.5)}{\pi}\right)\sqrt{\left(\frac{(2)(1\,743.82)}{27\,540}\right)^2 + \frac{3}{4}\left(\frac{2\,268.91}{87\,000}\right)^2}\right]^{\frac{1}{3}}$$

$$\boldsymbol{D_2 = 1.48\ in}$$

Since this shaft receives higher torque than the other, these values will be the smallest diameter the shaft can be. For ease of machining, we will allow both shafts to have identical dimensions but will rotate them as seen in Figure 1.

From Appendix 2, the minimum diameters for the stepped shaft are [1];

$$\boldsymbol{D_1 = 1\ in}$$

$$\boldsymbol{D_2 = 1.6\ in}$$

Bearing Calculation and Selection

No thrust, therefore:

P = VR

Inner race of bearing rotates, V = 1, Radial Load (Wr) = 91.76 lb

P = 1 • 91.76 = 91.76 lb

Ld of pinion = 14,000 x 2100 x 60 = $1.764*10^9$

Ld of gear = 14,000 x 700 x 60 = $0.588*10^9$

C = P (Ld / 10^6)^(1/3) = 91.76 (Ld / 10^6)^(1/3)

Cp = 1108.714 lbf

Cg = 768.739 lbf

D_1 minimum = 0.88 in

Ball bearing 6005 for D_1: C = 2518 lbf, D = 0.9843 in

D_2 minimum = 1.48 in

Ball bearing 6008 for D_2: C = 3777 lbf, D = 1.5748 in

Key Selection

SAE 1018 steel will be selected as the material for the keys in the input and output shaft assemblies. From Table 11-4, s_u = 64 000 psi and s_y = 54 000 psi [1]. The

By using Table 11-1 and selecting a shaft diameter of 1.00 inch, the diameter of the shaft the gear and pinion will sit on as shown in Figure 1, the width and height of the key will be 0.25 in, respectively, resulting in the usage of a square parallel key [1].

The following equation is used to obtain the minimum key length of a square parallel key [1];

$$L_{min} = \frac{4TN}{DWs_y}$$

$T = 2268.91\ lb \cdot in$

Note: the gear's torque will be used because it is the controlling factor. Since the same stepped shaft will be used for the pinion and gear, the design will be governed by the largest applied torque.

$N = 2.5$

Note: the same design factor will be used from the shaft analysis.

$D = 1.00\ in$

$W = 0.25\ in$

$s_y = 54\ 000\ psi$

$$L_{min} = \frac{4(2268.91\ lb \cdot in)(2.5)}{1.00\ in(0.25\ in)(54\ 000\ psi)} = 1.68\ in$$

$$(L_{min})_{basic\ size} = 1.8\ inch \rightarrow selected\ key\ length$$

Retaining Ring Selection

As shown in Figure 1, an external retaining ring will be inserted to fix the position of the pinion and gear on the 1 inch diameter section of the input and output shafts respectively. Using the ARCON Ring catalog for external retaining rings, the selected external retaining ring is part number 1400-100 [2]. This retaining ring corresponds to a shaft diameter of 1 inch.

Lubrication Selection

By using Table 9-17, the recommended lubricant for the designed enclosed gear drive is a rust and oxidation inhibited gear oil with an ISO of 460 [1]. The lubricant is determined using the calculated pitch line velocity, 1649.34 ft/min, and assuming that the operating temperature range of the gearbox is from 26.7°C to 48.9°C.

CAD Design

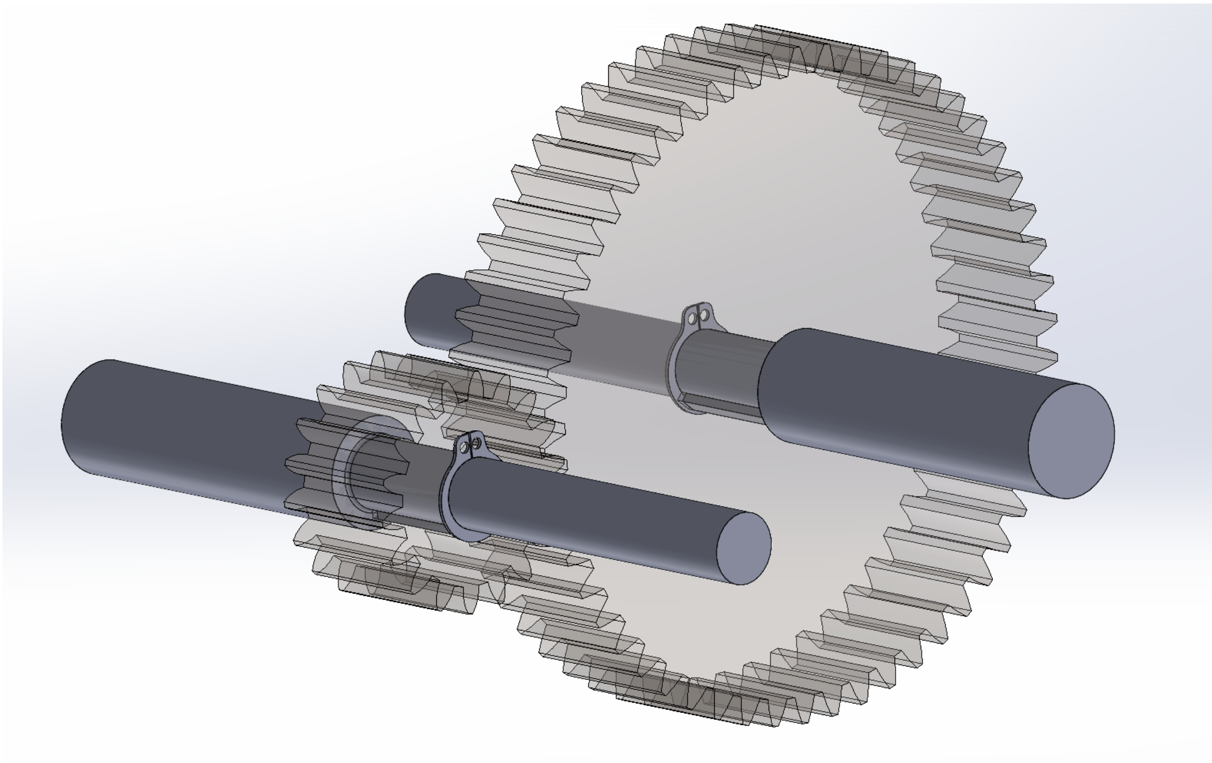


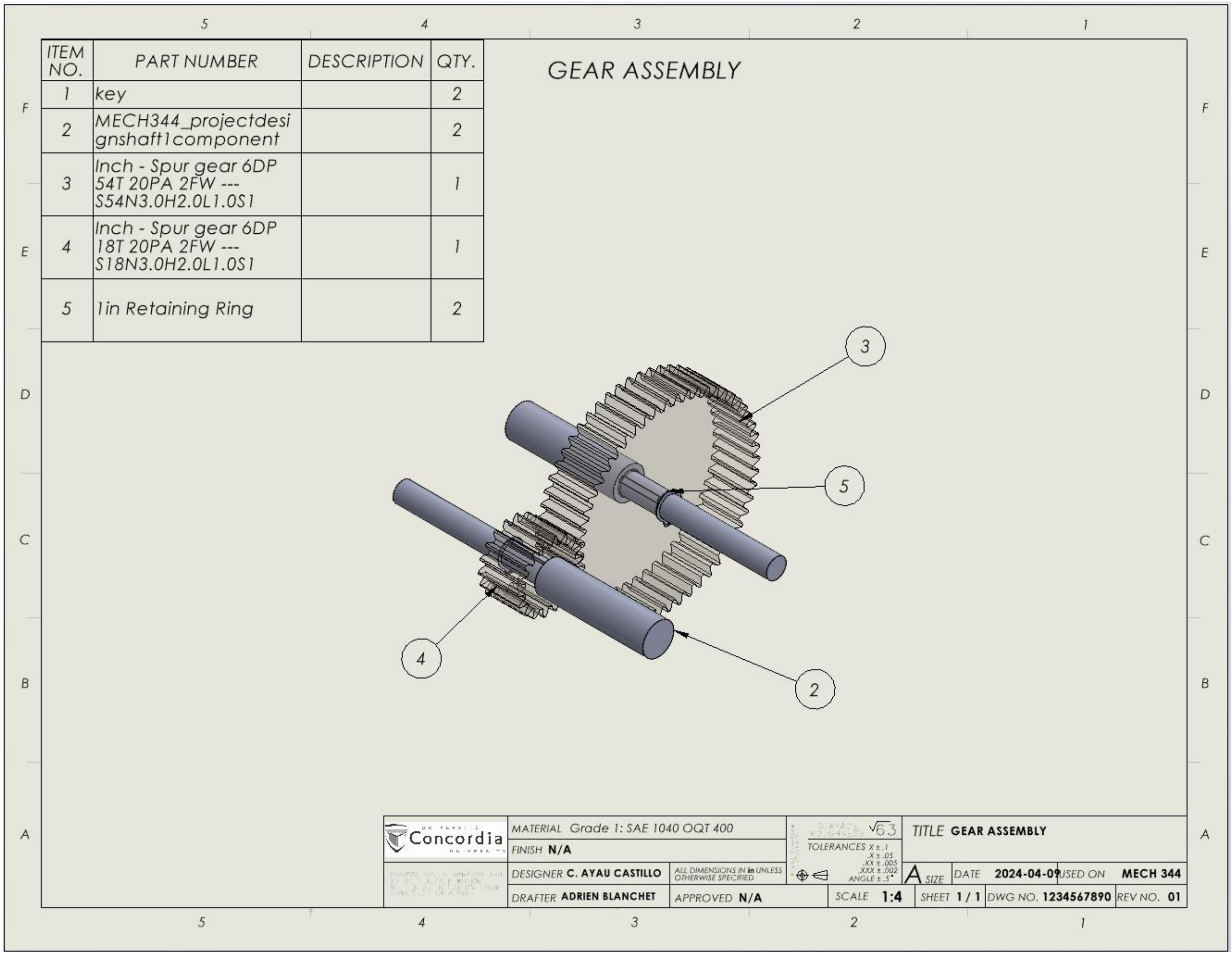
Figure 4: Gear Box Design

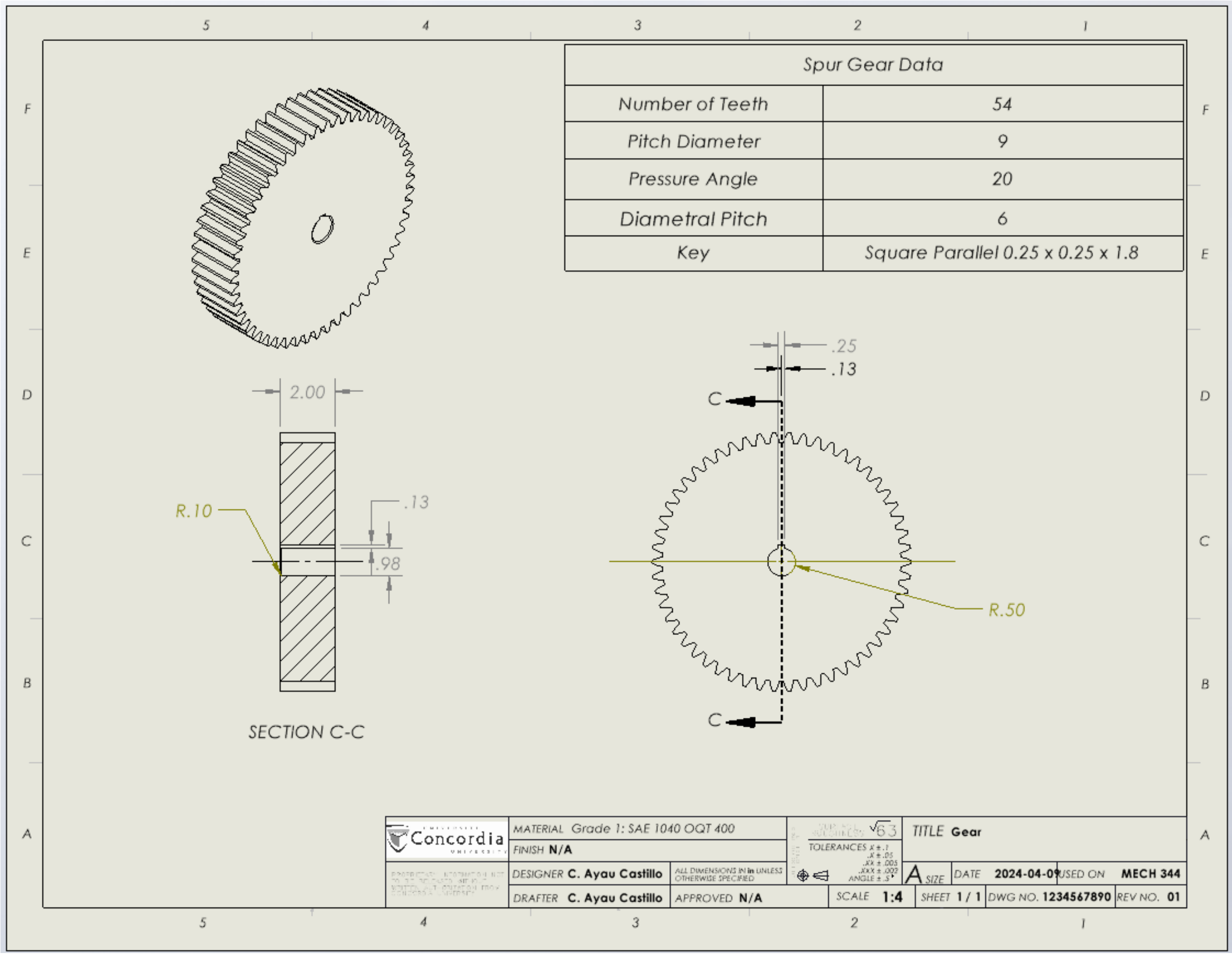
References

[1] Robert L. Mott, Edward M. Vavrek, Jyhwen Wang. *Machine Elements in Mechanical Design, sixth ed*. (2018).

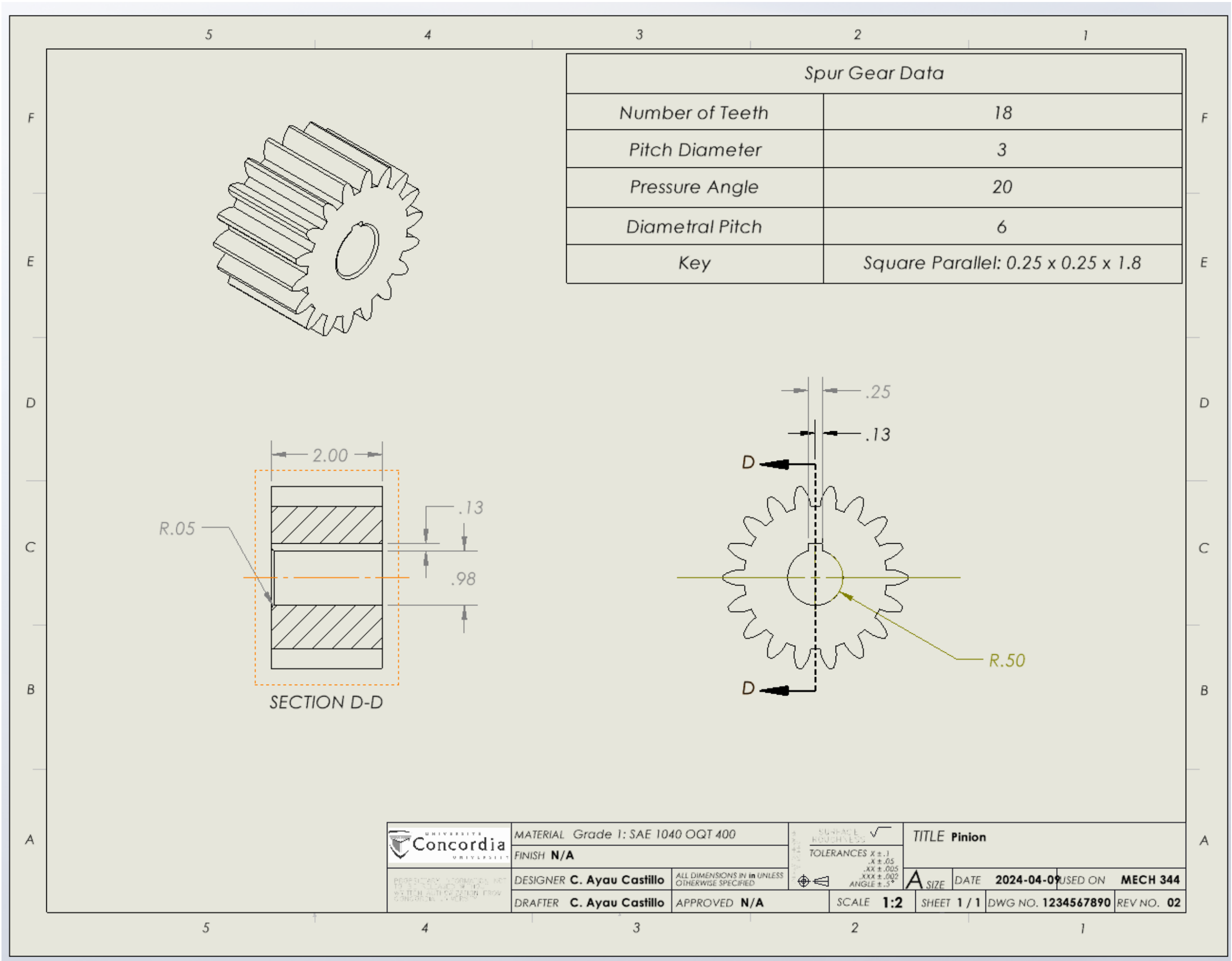
[2] "External Retaining Rings: Standard Retaining Rings: Arcon Ring." *Standard External Retaining Rings Product Category*, www.arconring.com/stamped-retaining-rings/external/. Accessed Apr. 2024.

Appendix

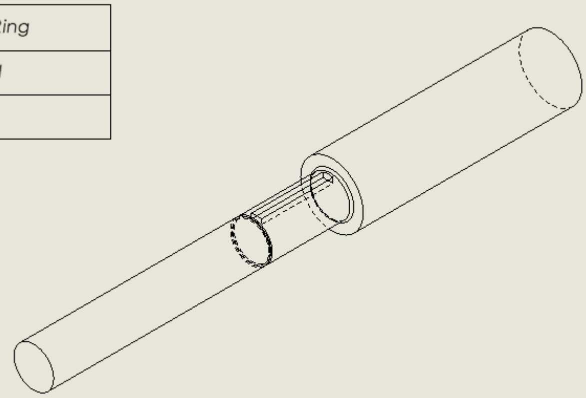

GEAR ASSEMBLY
ITEM NO. | PART NUMBER | DESCRIPTION | QTY.
1 | key | | 2
2 | MECH344_projectdesignshaft1component | | 2
3 | Inch - Spur gear 6DP 54T 20PA 2FW --- S54N3.0H2.0L1.0S1 | | 1
4 | Inch - Spur gear 6DP 18T 20PA 2FW --- S18N3.0H2.0L1.0S1 | | 1
5 | 1in Retaining Ring | | 2
3
5
4
2
Concordia
MATERIAL Grade 1: SAE 1040 OQT 400
FINISH N/A
DESIGNER C. AYAU CASTILLO
ALL DIMENSIONS IN in UNLESS OTHERWISE SPECIFIED
DRAFTER ADRIEN BLANCHET
APPROVED N/A
√63
TOLERANCES X ± .1
.X ± .05
.XX ± .005
.XXX ± .002
ANGLE ± .5°
TITLE GEAR ASSEMBLY
A SIZE
DATE 2024-04-09
USED ON MECH 344
SCALE 1:4
SHEET 1 / 1
DWG NO. 1234567890
REV NO. 01

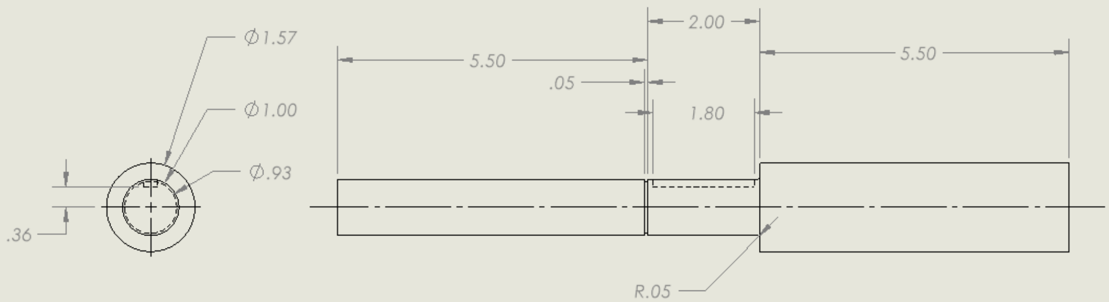


Spur Gear Data
Number of Teeth 54
Pitch Diameter 9
Pressure Angle 20
Diametral Pitch 6
Key Square Parallel 0.25 x 0.25 x 1.8
2.00
R.10
.13
.98
SECTION C-C
.25
.13
C
C
R.50
Concordia
MATERIAL Grade 1: SAE 1040 OQT 400
FINISH N/A
DESIGNER C. Ayau Castillo
DRAFTER C. Ayau Castillo
ALL DIMENSIONS IN in UNLESS OTHERWISE SPECIFIED
APPROVED N/A
TOLERANCES
SCALE 1:4
TITLE Gear
A SIZE
DATE 2024-04-09
USED ON MECH 344
SHEET 1 / 1
DWG NO. 1234567890
REV NO. 01


Spur Gear Data
Number of Teeth 18
Pitch Diameter 3
Pressure Angle 20
Diametral Pitch 6
Key Square Parallel: 0.25 x 0.25 x 1.8
2.00
.13
R.05
.98
SECTION D-D
.25
.13
D
D
R.50
Concordia
MATERIAL Grade 1: SAE 1040 OQT 400
FINISH N/A
DESIGNER C. Ayau Castillo
DRAFTER C. Ayau Castillo
APPROVED N/A
TOLERANCES
TITLE Pinion
SIZE A
DATE 2024-04-09
USED ON MECH 344
SCALE 1:2
SHEET 1 / 1
DWG NO. 1234567890
REV NO. 02

Key	Retaining Ring
Square Parallel: 0.25 x 0.25	External
SAE 1018 steel	ARCON





Concordia University	MATERIAL Grade 1: SAE 1040 OQT 400		$\sqrt{63}$	TITLE **Shaft 1 and 2**
	FINISH **N/A**		TOLERANCES X ± .1 .X ± .05 .XX ± .005 .XXX ± .002 ANGLE ± .5°	
	DESIGNER **C. Ayau Castillo**	ALL DIMENSIONS IN in UNLESS OTHERWISE SPECIFIED		A SIZE · DATE **2024-04-09** · USED ON **MECH 344**
	DRAFTER **C. Ayau Castillo**	APPROVED **N/A**	SCALE **1:2**	SHEET **1 / 1** · DWG NO. **1234567890** · REV NO. **03**