TECHNICAL REPORT

ME 4463 Mechanical Systems Design

SE Final Report

For

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Executive Summary

Team SE of ISU was assigned to design a specialized gearbox with single input and double output. The gearbox was required to have a single speed of 20 rpm and variable speeds of 10 rpm and 50 rpm respectively. Similarly, the minimum torque output of the constant speed shaft was 1.75 Nm and that of the variable output shaft was 3.55 Nm. The output shafts were designed to be located as an angle of 90 degrees with respect to each other. Similarly, the environmental condition was provided to be outdoors. The outdoor conditions were assumed to be roofed backyard.

The motor motor selected was a DC gearmotor of 1/17 Hp. The single speed angular velocity was achieved through the double stage gear reduction by using bevel and spur gears. Lubriplate lubrication was chosen. The gears on the constant speed shafts were connected through the press fit.

Similarly, the variable speed power transmission was achieved through multiple staging. The speed reduction for 10 rpm was done through the transmission via intermediate shaft and then to the planetary train via spur gear train and then to the output shaft. A hollow shaft was used to transmit power to the planetary gears. Similarly, reduction to 50 rpm was done through the intermediate shaft to the output shaft using spur gears. Keys were used to transmit the torques from gears to the shaft. Two gears were connected to the output shaft using bearings. Similarly, the dog gears were connected permanently to the output shafts which were connected to a lever which could be used to change the speed of the variable output shaft. Grease lubricants were used. Similarly, the Electro-magnetic Clutch and Brake system was used from Inertia Dynamics. The parts, assembly, drawing files and the cost sheets are provided along with the report in the submission folder.

Table 1: Cost break-down of chosen design.

Component	Lead Designer	Estimated Cost
Engineer A	Erik Bergman	$2052.75
Engineer B	Shishir Khanal	$1576
	Total:	$3628.75

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Abbreviations

MSST = Maximum Shear Stress Theory

FOS = Factor of Safety

N = Number of teeths for a particular gear

S_{ys} = Yield Strength

τ = Torsional Stress

τ_{max} = Maximum Torsional Stress

Tables:

Engineer A:

	Part	Quantity	Price (Each)	Total
Bearings	2342K185	2	14.33	28.66
Bearings	2342K187	1	22.71	22.71
Bearings	234K183	2	19.41	38.82
Bearings	2342K81	1	18.24	18.24
Gear	YP2414	1	80	80
Gear	YP2442	1	80	80
Gear	YA14	1	80	80
Gear	YA70A	1	80	80
Gear	L99Y	1	80	80
	SHAFT 1.1	1	15	15
	SHAFT 2	1	15	15
	SHAFT 0	1	15	15
	SHAFT 3	1	15	15
	BOX	1	1000	1000
Motor	6470K75	1	458.27	458.27
Lube	2289N17	1	26.05	26.05
	TOTAL			2052.75

Engineer B:

	Quantity	Price (Each	Total
Bearings	3	22.71	68.13
Bearings(Custom)	2	40	80
Gear	9	80	720
Gear(Custom)	1	100	100
Dog Gear(Custom)	2	80	160
Shaft	3	40	120
Lubricant	1	30	30
Speed Changing Lever	1	200	200
Electromagnetic Brake-Clutch	1	218.5	218.5
	TOTAL		1576.63

Introduction

Problem Description

Our group is to develop, produce, and sell specialized gearboxes. The buyer is requesting that the new gearbox design has one input shaft and two output shafts. One of the output shafts will run at a constant single speed. The other output shaft will have a mechanical switch that allows the operator to switch speed. In order to activate the switching, the system must come to a stop; hence a brake system is involved in this as well. Your team is assigned to develop this gearbox including the selection of the specific motor. The buyer provided specifications for speed and loading as follows below.

Goals

Following were the design goals for the gearbox design project.

Engineer A	Engineer B
Motor specification and selection	Output shaft 2 (speeds -1)
Output shaft 1 design (one speed)	Output shaft 2 (speeds -2)
Gearbox design (casting)	Speed shift mechanism
Lubrication	Brake system
Bearings, Gears, Sealings, Keys, etc.	Bearings, Gears, Sealings, Keys, etc.

Constraints

1. Group A1 needs to have a 90deg angle between two output shafts in an outdoor environment. Shaft 1 at 20 rpm with a load minimum of 1.75 Nm.Shaft 2 able to reach variable speeds of 10 rpm and 50 rpm with a load minimum of 3.55Nm.

Standards & Codes

- Dimensioning & Tolerancing -Y14.5
- Gear Drawing Standards - Y14.7.1
- Standard Specification for Performance of Manual Transmission Gear Lubricants - ASTM D5760
- Standard Specification for Steel Forgings, Carbon and Alloy, for Pinions, Gears and Shafts for Reduction Gears - ASTM A291/ A291-19

Discussion

Technical Information

1. **Engineer A**

   a. **Motor specification and selection**

Selection of the motor consisted of ease of purchase, lowest rpm that could move the system, horsepower that would meet system requirement, and cost. In doing so, Parallel Shaft DC Gearmotor (6470K75), was chosen. This motor will run at 300 rpm @ 10 in – lbs, power output of 1/17 hp, cost $458.27. More specs can be found in the appendix.

   b. **Output Shaft 1**

Shaft output 1 will need three small shafts to hold the gears and be machined with A36 Steel with a surface finish of 64. The estimated cost is $80. Drawings of small shafts are in the appendix.

   c. **Gearbox design**

The gearbox will be made of Aluminum C355-T61 PMC (SS). It will be molded. Both are common industry materials and practices. Based on searches, cost estimates are $500 – $7500. This cost estimate is based on the size of the molding. Unfortunately time did not allow me to have a completed gearbox. That being said the process for completion would be to integrate Engineer B's parts to the assembly and then form a full case about all the parts. After that creating a flanged edge to the case for installation and access. Lastly creating a legged stand for the gearbox to stand off the ground would be made. This would allow security of the gearbox and parts.

   d. **Lubrication**

The lubrication chosen is Lubriplate, number 5555 grease 2 lb bottle (2289N17). This will be acquired from McMaster-CARR for $26.05. It is a mineral based oil, has Zinc additive, and Calcium Sulfonate thickener.

   e. **Bearings and Gears**

Bearings that have been chosen are all Sealed Bearings purchased from McMaster-CARR. The bearings chosen range from $14.50 to $23.00. They are chosen because of the ease to acquire, lessen any oil that may leak from the gearbox, and to help keep environmental debris out.

Gears will be purchased from Boston Gears. Boston Gears offers many varieties of gears made of many materials and has the option to make any necessary custom gears. For Shaft 1 the gears chosen are as follows;

N_0 – Miter Gear, Number of Teeth 12, Diametral Pitch 16, Ratio 1:1, Steel

N_1 – Spur Gear, Number of Teeth 14, Diametral Pitch 24, Delrin

N_2 – Spur Gear, Number of Teeth 42, Diametral Pitch 24, Delrin

N_3 – Spur Gear, Number of Teeth 14, Diametral Pitch 20, Steel

N_4 – Spur Gear, Number of Teeth 35, Diametral Pitch 20, Steel

The price for the gears are estimated around $80. All the gears chosen are stock gears from Boston and can be made of any material as needed.

There are no seals or keys needed for Shaft 1. However, keys were considered, and it was decided press fit would be easiest. Seals were taken care of by the bearings, but when putting the gearbox together it would be advised to use Loctite with the screws for added security.

2. Engineer B

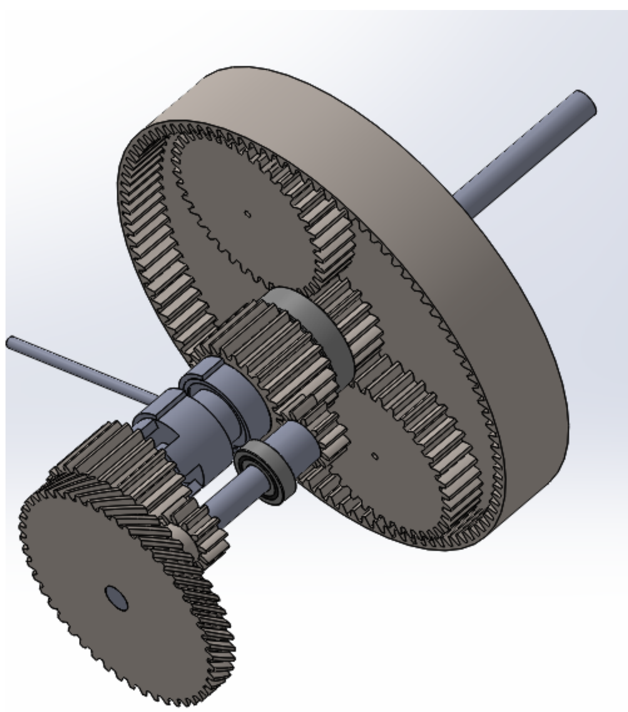

Fig: Assembly of Design of Engineer-B

1. Design components

a. Gear Mesh & Variable Speed

Following were the gear meshes required to obtain the following speed reductions:
The input speed of the motor was 300 rpm. The output speed of 10 rpm required the gear reduction ratio of 1:30. This was accomplished in 3 successive steps. First, Helical gears were connected in the main and the intermediate shafts with the teeths of 16 and 48 respectively. After that, the intermediate shaft was connected to the hollow shaft using spur gears of teeth numbers 12 and 24 respectively. After that the hollow shaft was connected to the planetary gear train of gear ratios 20, 39 and 100 respectively. Hence, in each successive steps following gear reduction ratios were achieved as ⅓,1/2,and ⅕ which was equal to 1:30.
Similarly, for the speed reduction to 50 rpm, a gear reduction ratio of 1:6 was required. This was obtained by connecting another gear train of gear ratio 1:2 directly from intermediate to the output shaft. Hence Helical gear reduction of 1:3 and spur gear reduction of 1:2 was sufficient to achieve the reduction ratio of 1:6.

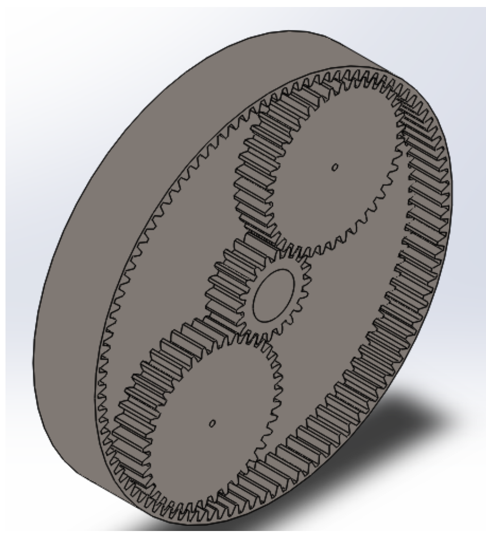

Fig: Assembly of Planetary Gear for ⅕ gear reduction

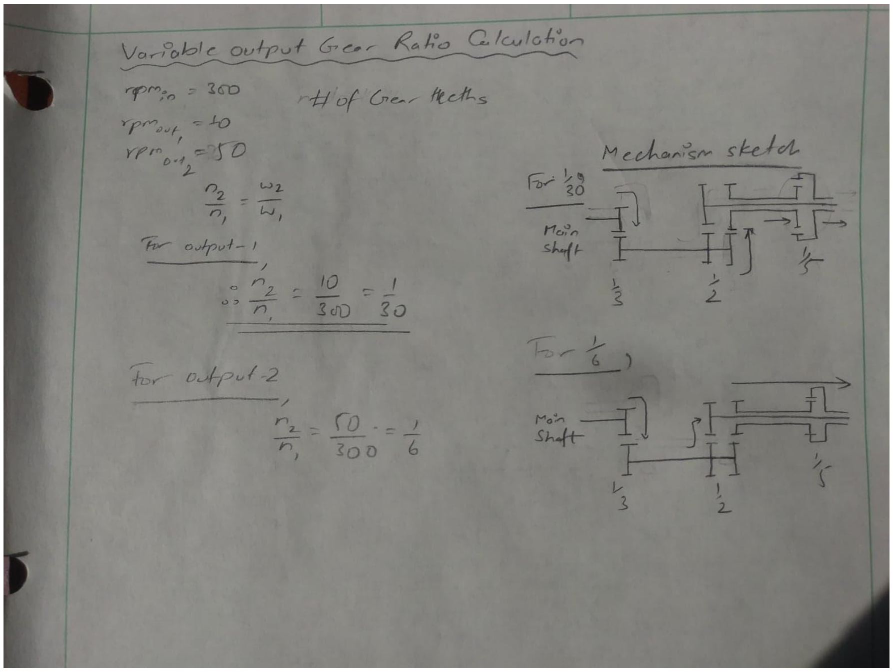

Fig 1: Appendix-IV Mechanism Design

b. Bearings

The bearings were strategically placed to support the shafts. 4 bearings were used from the Mcmaster carr and 2 custom bearings were designed to accommodate the dog gears for the speed shift process. The bearings were selected to accurately position the gears and also to withstand the resultant forces of the shaft.

c. Intermediate Shaft Design

The intermediate shaft was designed to be of length 3.5”. 2 bearings were housed to support the load of 3 gears during rotation. The yield strength of the loads were calculated to be 62.1 MPa. Similarly, AISI 1010, hot rolled bar Steel was chosen as a shaft material for the shaft diameter of 0.5”. FOS was chosen to be 3. The design of the intermediate shaft is provided in Appendix-I.

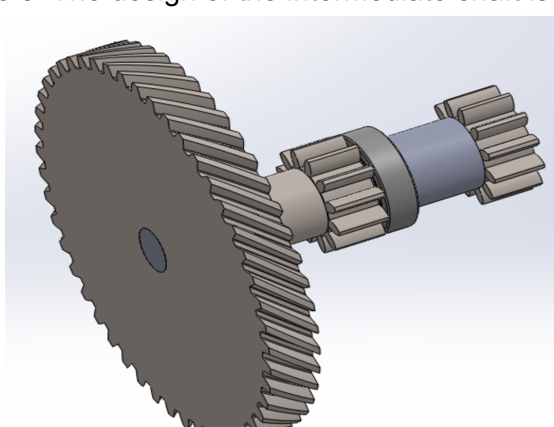

Fig: Intermediate Shaft Assembly

1. Estimation of Loading

The product datasheet had dimensions of the gears and the material information but didn't have the weight information. Hence, the weight of the gears were calculated based on the information of the gear. Sample calculation is provided below:

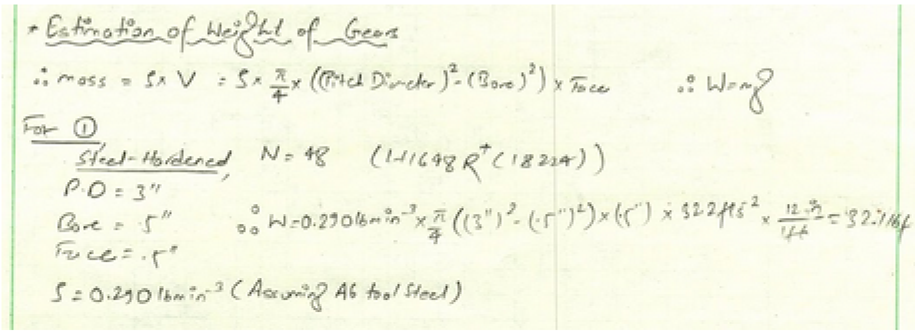

Fig: Sample calculation for the estimation of the Gear1

2. Calculation of Resultant Forces

After the calculation of all the individual loadings, a Force-Moment diagram was created and its static condition of the shaft was used to calculate the resultant forces acting on the bearings.

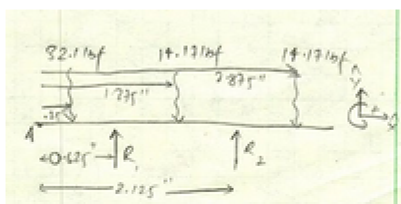

Fig: Force-Moment Diagram for the Intermediate Shaft

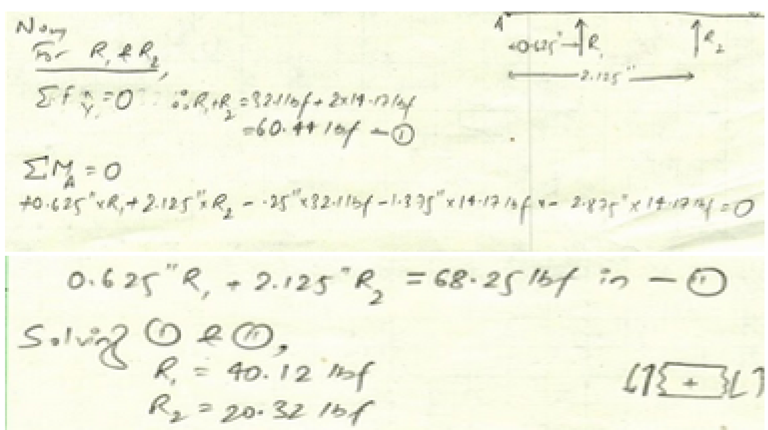

Fig: Calculation of Resultant forces

3. Shear and Moment Diagram

Shear and Moment diagram was created and maximum moment force was noted. Maximum moment force was found to be -9.11 in lbf.

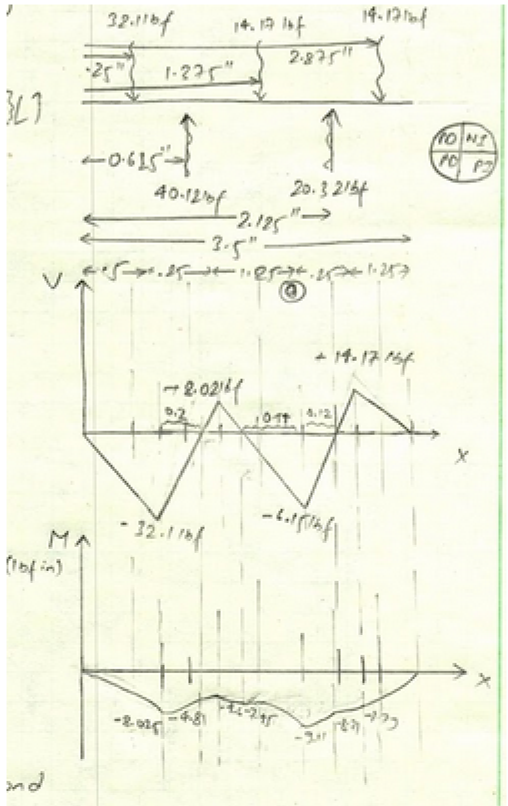

Fig: Shear-Moment Diagram

4. Design for Twisting and Bending

Power transmitted equation was used to calculate the max shear force. After that the shear and moment were used to calculate the equivalent twisting moment. It was found to be 73.86 lbf in.

4. Design for twisting & Bending

Power transmitted by shaft is,

$$P = \frac{2\pi N T}{60} \qquad N = 50\ (max)$$

$T \rightarrow$ Torque
$N \rightarrow$ RPM (speed)
$P \rightarrow$ Power

$$T = 383.6\ in\ lbf/s \times \frac{60}{2\pi \times 50}$$

$$\boxed{\therefore T = 73.3\ in\ lbf}$$

Max bending moment occurs at b which is the starting point of the second bearing support.

$$\therefore M_{b} = -9.11\ lbf\ in$$

Also
Equt Twisting moment,

$$T_{eq} = \sqrt{T^2 + M^2}$$

$$\therefore T_{eq} = \sqrt{(73.3\ in\ lbf)^2 + (9.11\ lbf\ in)^2}$$

$$\boxed{T_{eq} = 73.86\ lbf\ in}$$

Fig: Twisting moment calculation

5. Maximum shear stress calculation

The maximum shear stress on the shaft was calculated using the maximum shear stress theory. Maximum shear stress was found to be 20.7 Mpa.

Shaft will be chosen for ductile material.
Hence, Using Max shear stress theory,

$$\tau_{max} = \frac{16}{\pi d^3} T_{eq}$$

$$= \frac{16}{\pi \times (0.5'')^3} \times 73.86\ lbf\ in$$

$$\boxed{\therefore \tau_{max} = 3003.33\ lbf\ in^{-2}}$$

$$\frac{0.25}{x} = \frac{20.22}{6.15} \qquad x = 0.12$$

$$\tau_{max} = 3003.33\ lbf\ in^{-2} \times \frac{6894.76\ Pa}{1\ lbf\ in^{-2}}$$

$$\boxed{\tau_{max} = 20.7\ MPa}$$

Fig: Max Shear stress Calculation

6. Selection of Material

The maximum shear stress value was adjusted with desired FOS to get the ultimate yield strength requirement. This value was then used to select a material. From Matweb, AISI 1010 steel, Hot rolled was chosen which had yield strength of 180 Mpa.

(c) Selection of shaft Material

$$\tau_{max} = \frac{S_{ys}}{FOS}$$

S_{ys} = Yield Strength
FOS = Factor of Safety = 3 (Assume)

$$20.7\,MPa = \frac{S_{ys}}{3}$$

$$\therefore S_{ys} = 62.1\,MPa$$

The design calculation doesnot incorporate other forces due to the meshed gears acting on the shaft. Hence, larger yield strength is chosen than the value.

Material: AISI 1010 Steel, hot roled bar, 19-32mm round or thickness
[Source: Matweb]

$$S_{ys} = 180\,MPa$$

Fig: Material Selection

d. Hollow shaft Design

The hollow shaft design procedure was the same as the intermediate shaft. The intermediate shaft was designed to be of length 3.4”. 1 bearing was housed to partially support the load of 1 spur gear and a planetary gear train during rotation. The yield strength of the loads were calculated to be 212.4 MPa. Similarly, ASTM A514 steel, Grade Q was chosen as a shaft material for the inner shaft diameter of 0.5” and outer shaft diameter of 1”. FOS was chosen to be 3. The design of the hollow shaft is provided in Appendix-II.

1. Max shear stress theory

The equation of maximum shear stress for the hollow shaft is different from the rod shaft. The calculation is given below.

For Hollow shaft,

$$\tau_{max} = \frac{16}{\pi d_o^3 (1 - C^4)} T_{eq}$$

$$= \frac{16}{\pi \times (1'')^3 (1 - 0.5^4)} \times 5670.2\,in\,lbf$$

$$\tau_{max} = 30803.3\,\cancel{lbf/in^2} \times \frac{6894.76\,Pa}{\cancel{lbf/in^2}}$$

$$\boxed{\therefore \tau_{max} = 212.4\,MPa}$$

Fig: Max shear stress for Hollow shaft

e. Output shaft Design:

The intermediate shaft was designed to be of length 11.95”. 2 bearings were housed to support the load of 3 spur gears and a planetary gear train during rotation. The yield strength of the loads were calculated to be 273.9 MPa. Similarly, ArcelorMittal MarTENsite 1200 cold Rolled Steel was chosen as a shaft material for the shaft diameter of 0.5”. FOS was chosen to be 3.The design of the intermediate shaft is provided in Appendix-III.

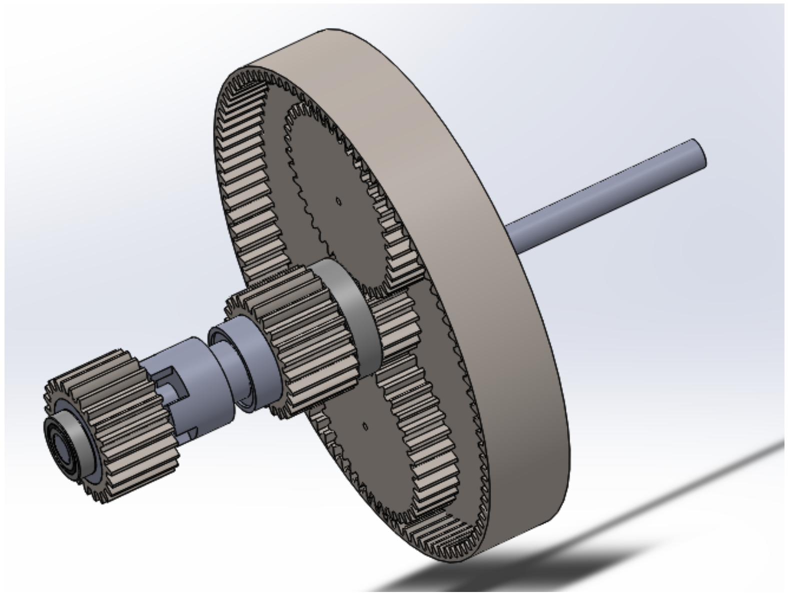

Fig: Output shaft (Hollow + Main Output)

f. Keys

The keys were designed to have a length equal to the length of the respective gear, width of 0.1” and height of 0.2” above the shaft.

g. Speed Shift Mechanism

The spur gears of 24 teeths each connected to the output and hollow shafts using bearings. Both of the gears rotate all the time whenever the motor is turned on. However the hollow or the output shaft only rotates if the dog gear is connected to the respective gear through the bracket in the bearings and the gears. For the 10 rpm output, the power is transmitted via gear connected to the hollow shaft and for the 50 rpm output, the power is transmitted via the gear connected to the output shaft. A lever is connected to the dog gears which still allow the rotation of the gears to limit the mobility of the gears within the distance of 0.4” back and /or forth to facilitate the speed shift.

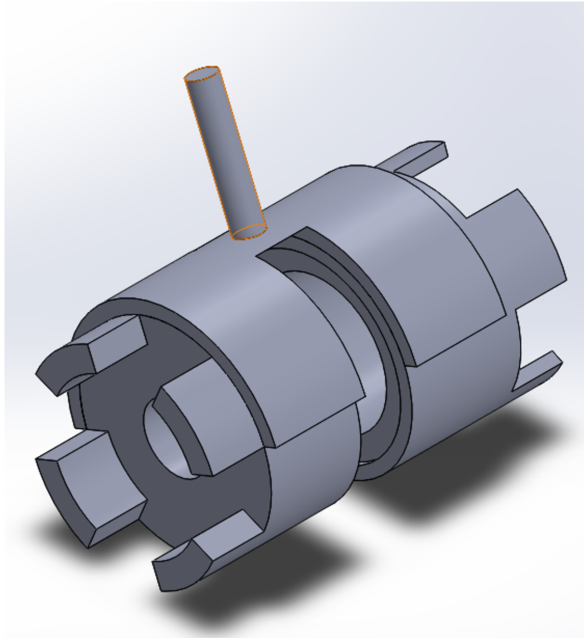

Fig: Speed Shift Lever attached to Dog Gears

h. Clutch/Brakes

The S0B22 clutch Brake system was selected from Inertia Dynamics which is a Shaft mounted clutch/power-on brake system. The maximum torque of the motor is under the torque criteria of the SOB22 torque speed curve. The selection was based on the allowed shaft diameter (0.5”) and permissible power rating (i.e. above 1/17 HP) Appendix VII-IX consists of the torque-speed curves, datasheet and drawing of the clutch brake system.

i. Lubrication

Grease lubrication was selected from super lube for $30. This is because the planetary gear train was tricky to lubricate using other methods. However, the lubrication cycle is shorter with the grease lubrication.

2. Problems encountered

The initial plan was to use a single planetary gear train that would perform both the speed shift . The 1/6 speed shift would be done by gears with teeths 50(sun) and 300(ring) and 1/30 gear shift would be done by gear with teeths 10(sun) and 300(ring). However, the allowed shaft diameter was found to be less than 3 mm for the gear with 10 teeth. Also, it was also found out that the if upgraded to bigger diametral pitch gears the required diameter for the ring gear would be 3 ft which was unreasonably big. Hence, it was decided to drop the idea of planetary gear transmission.

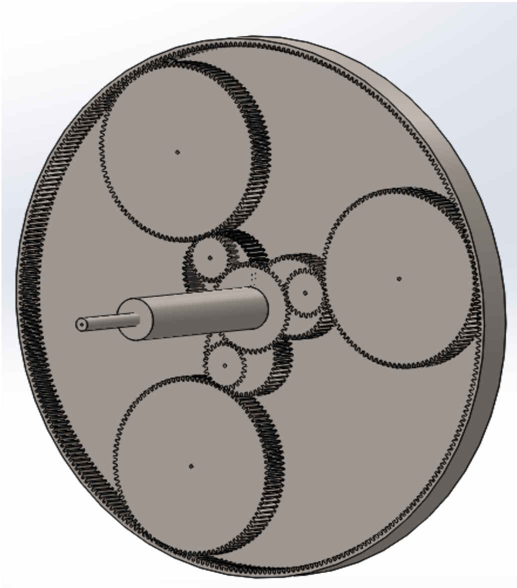

Fig: Single Planetary gear train idea

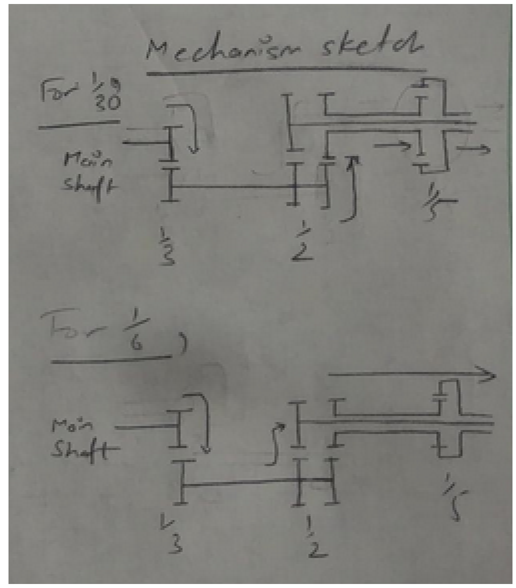

Fig: Modified Mechanism

Conclusions

Hence, a gearbox was designed. The motor was appropriately selected to meet the requirements of the customer without significant increase in the cost. The appropriate speed reductions were achieved by using Bevel, supr and planetary gear trains. The speed shift was done by using the speed switch gear attached to the dog gears . Similarly, the motor of 1/14 HP was selected which would power the gearbox to perform both the constant output and variable output. Shafts were designed appropriately to perform the power transmission between the gears and designed for both the twisting and bending. The total budget of the gearbox was found to be $3628.75

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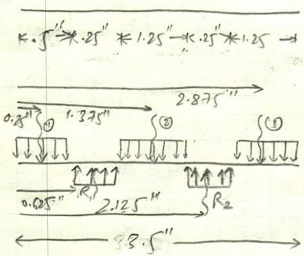
Appendices

I. Design of Intermediate Shaft

Shishir Khanal

1. Design of shaft - A

- Desired Diameter = .5"



- Max available Power;

Power from Motor = $\frac{1}{17}$ HP

$= 388.2 \text{ in lbf s}^{-1}$

Min Power used in constant speed shaft $= 0.35 \text{ Nm} \times \frac{90}{60} \text{ s}^{-1} = 0.525 \text{ Nms}^{-1}$

$= 4.64 \text{ lbf in s}^{-1}$

$\therefore$ Max available Power = $383.6 \text{ in lbf s}^{-1}$

* Estimation of Weight of Gears

$\therefore \text{mass} = \rho \times V = \rho \times \frac{\pi}{4} \times ((\text{Pitch Diameter})^2 - (\text{Bore})^2) \times \text{Face}$ $\quad \therefore W = mg$

For ①

Steel-Hardened, N = 48 (L-H1648 R^+ (18224))

P.D = 3"

Bore = .5" $\quad \therefore W = 0.290 \text{ lbm in}^{-3} \times \frac{\pi}{4}((3")^2 - (.5")^2) \times (.5") \times 32.2 \text{ ft s}^{-2} \times \frac{12 \text{ in}}{1 \text{ ft}} = 32.1 \text{ lbf}$

Face = .5"

$\rho = 0.290 \text{ lbm in}^{-3}$ (Assuming A6 tool Steel)

For ② & ③

Steel, N = 12 (ND12B (01746)) $\quad V = \rho \cdot \left(\frac{\pi}{4}(PD^2 - Hub^2) \times Face + \frac{\pi}{4}(Hub^2 - Bore^2) \times Proj\right)$

PD = 1"

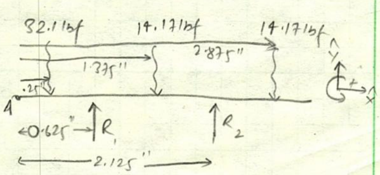
Bore = .5" $\quad \therefore W = 0.289 \text{ lbm in}^{-3} \times \frac{\pi}{4}\left\{(1"^2 - .75"^2) \times .750" + (.75"^2 - .5"^2) \times .5"\right\} \times 32.2 \text{ ft s}^{-2} \times \frac{12 \text{ in}}{1 \text{ ft}} = 14.17 \text{ lbf}$

Face = .758"

$\rho = 0.289 \text{ lbm in}^{-3}$ (Assuming Grade 304 steel)

HUB = .75"

Proj = .5"



Now

For R_1 & R_2,

$\Sigma F_{\hat{y}} = 0 \quad \therefore R_1 + R_2 = 32.1 \text{ lbf} + 2 \times 14.17 \text{ lbf}$

$= 60.44 \text{ lbf}$ — ①

$\Sigma M_A = 0$

$+0.625" \times R_1 + 2.125" \times R_2 - .25" \times 32.1 \text{ lbf} - 1.375" \times 14.17 \text{ lbf} - 2.875" \times 14.17 \text{ lbf} = 0$

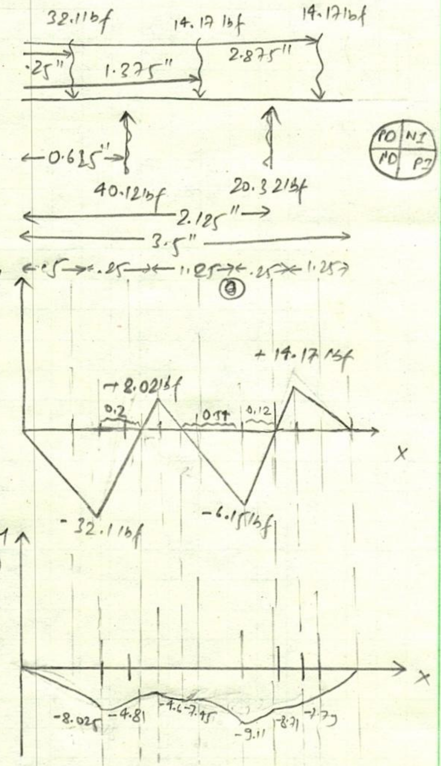
Shishir Khanal 2

$0.625'' R_1 + 2.125'' R_2 = 68.25 \text{ lbf in}$ — (ii)

Solving (i) & (ii),

$R_1 = 40.12$ lbf

$R_2 = 20.32$ lbf



Desired minimum Diameter = .5"

A. Design for twisting & Bending

Power transmitted by shaft is,

$$P = \frac{2\pi N T}{60}$$

N = 50 (max)

T → Torque

N → RPM (speed)

P → Power

$$T = 383.6 \text{ in lbf s}^{-1} \times \frac{60}{2\pi \times 50 \text{ s}^{-1}}$$

$$\boxed{\therefore T = 73.3 \text{ in lbf}}$$

Max bending moment occurs at (a) which is the starting point of the second bearing support.

$$\therefore M_{(a)} = -9.11 \text{ lbf in}$$

Also,

Eqvt Twisting moment,

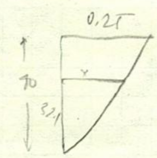
$$T_{eq} = \sqrt{T^2 + M^2}$$

$$\therefore T_{eq} = \sqrt{(73.3 \text{ in lbf})^2 + (9.11 \text{ lbf in})^2}$$

$$\boxed{T_{eq} = 73.86 \text{ lbf in}}$$

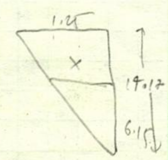
$\frac{0.25}{x} = \frac{40}{32.1}$

$x = 0.2$



$\frac{1.25}{x} = \frac{19.17}{6.15}$

$x = 0.54$



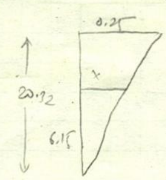
Shaft will be chosen for ductile material. Hence, Using Max shear stress theory,

$$\tau_{max} = \frac{16}{\pi d^3} T_{eq}$$

$$= \frac{16}{\pi \times (0.5'')^3} \times 73.86 \text{ lbf in}$$

$\frac{0.25}{x} = \frac{20.32}{6.15}$

$x = 0.12$



$$\tau_{max} = 3009.33 \text{ lbf in}^{-2} \times \frac{6894.76 \text{ Pa}}{1 \text{ lbf in}^{-2}}$$

$$\boxed{\therefore \tau_{max} = 3009.33 \text{ lbf in}^{-2}}$$

$$\boxed{\tau_{max} = 20.7 \text{ MPa}}$$

Shishir Khanal 3

(B) Selection of shaft Material

$$\tau_{max} = \frac{S_{ys}}{FOS}$$

S_{ys} = Yield Strength

FOS = Factor of Safety = 3 (Assume)

$$20.7\,MPa = \frac{S_{ys}}{3}$$

$$\therefore S_{ys} = 62.1\,MPa$$

The design calculation doesnot incorporate other forces due to the meshed gears acting on the shaft. Hence, larger yield strength is chosen than the value.

Material: AISI 1010 Steel, hot rolled bar, 19-32mm round or thickness

[Source: Matweb]

$$S_{ys} = 180\,MPa$$

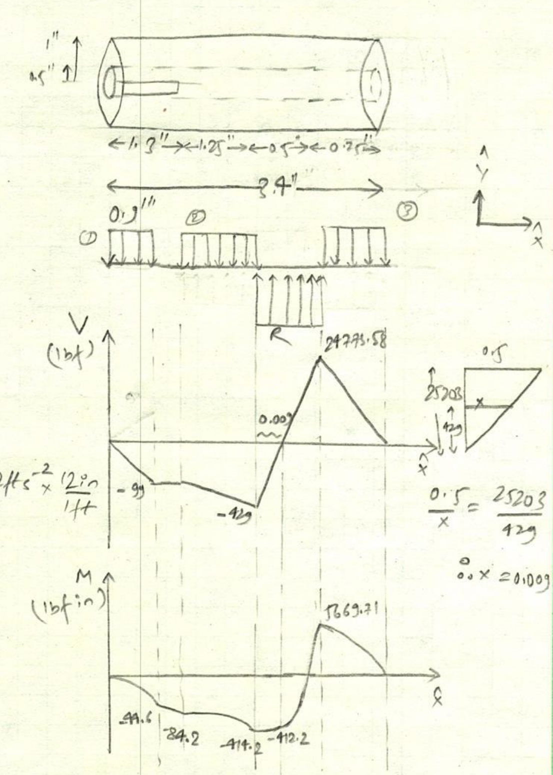
II. Design of Hollow Shaft

2. Design of Shaft B2

① → Dog gear

② N = 24 12-DP (14.5 PA) Gear

③ N = 20 (1-sun gear)
N = 46 (2-planet gears)
N = 100 (1-Ring gear)



For ①,
Face = 0.9"
PD = 1.5"
Bore = 1"
$\rho = 0.290\ lbm\ in^{-3}$ (Assuming A6 tool steel)

$\therefore W = 0.290\ lbm\ in^{-3} \times \frac{\pi}{4}\left((1.5")^2 - (1")^2\right) \times 0.9" \times 32.2\ ft\ s^{-2} \times \frac{12\ in}{1\ ft}$

$= 99\ lbf$

$\frac{0.5}{x} = \frac{25203}{429}$

$\therefore x = 0.009$

For ②,
Face = 1.25"
PD = 2"
Bore = 1"
$\rho = 0.290\ lbm\ in^{-3}$ (A6 tool steel)

$\therefore W = 0.290\ lbm\ in^{-3} \times \frac{\pi}{4}\left((2")^2 - (1")^2\right) \times 1.25" \times 32.2\ ft\ s^{-2} \times \frac{12\ in}{1\ ft}$

$= 330\ lbf$

For ③,

N = 20	N = 46	N = 100
Face = .750"	Face = .750"	Face = .750" × 2 = 1.5"
PD = 1.67"	PD = 3.83"	PD = 9.33"
Bore = 1"	Bore = 1"	Bore = 1"
		OD = 0.75" + 9.33" = 10.08"

$\rho = 0.290\ lbm\ in^{-3}$ (A6 tool steel)

$\therefore W_{20} = 0.290\ lbm\ in^{-3} \times \frac{\pi}{4}\left((1.67")^2 - (1")^2\right) \times 0.75" \times 32.2\ ft\ s^{-2} \times \frac{12\ in}{1\ ft} = 118.08\ lbf$

$\therefore W_{46} = 0.290\ lbm\ in^{-3} \times \frac{\pi}{4}\left((3.83")^2 - (1")^2\right) \times 0.75" \times 32.2\ ft\ s^{-2} \times \frac{12\ in}{1\ ft} = 902.2\ lbf$

$\therefore W_{100} = 0.290\ lbm\ in^{-3} \times \frac{\pi}{4}\underbrace{\left((10.08")^2 - (9.33")^2\right)}_{(OD^2 - PD^2)} \times 1.5" \times 32.2\ ft\ s^{-2} \times \frac{12\ in}{1\ ft} = 22851.1\ lbf$

$\therefore W = 118.08\ lbf + 2 \times 902.2\ lbf + 22851.1\ lbf = 24773.58\ lbf$

$\therefore$ For R, $\Sigma F_y = 0$ $\therefore R = 24773.58\ lbf + 330\ lbf + 99\ lbf = 25202.58\ lbf$

5

(A) Design of Hollow Shaft using Max Shear Stress Theory

$T_{eq} = \sqrt{T^2 + M^2}$

$T = 73.3$ in lbf

$M = 5669.71$ in lbf

$T_{eq} = \sqrt{(5669.71 \text{ in lbf})^2 + (73.3 \text{ in lbf})^2} = 5670.2$ in lbf

For Hollow shaft,

$\tau_{max} = \dfrac{16}{\pi d_o^3 (1 - C^4)} T_{eq}$

d_o = outer diameter = 1"

$C = \dfrac{d_i}{d_o} = \dfrac{0.5''}{1''} = 0.5$

$= \dfrac{16}{\pi \times (1'')^3 (1 - 0.5^4)} \times 5670.2$ in lbf

d_i = inner diameter = 0.5"

$\tau_{max} = 30803.3 \; \cancel{\text{lbf in}^{-2}} \times \dfrac{6894.76 \text{ Pa}}{1 \; \cancel{\text{lbf in}^{-2}}} = 212.4$ MPa

$\boxed{\therefore \tau_{max} = 212.4 \text{ MPa}}$

(B) Selection of Material

$\tau_{max} = \dfrac{S_{ys}}{FOS}$

S_{ys} = Yield Strength

FOS = Factor of Safety = 3 (Assume)

$212.4 \text{ MPa} = \dfrac{S_{ys}}{3}$

$\therefore S_{ys} = 637.2$ MPa

Material : ASTM A514 Steel, Grade Q, plate thickness 19-64mm

$S_{ys} = 690$ MPa

[Source : Matweb]

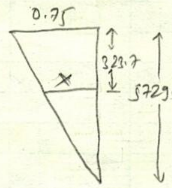
III. Design of Output Shaft

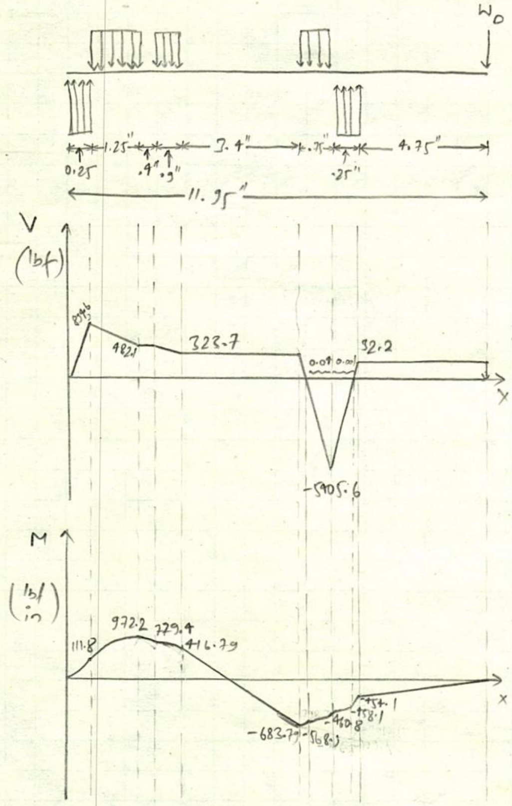
Shishir Khanal

7

$$\frac{0.75}{x} = \frac{5729.3}{323.7}$$

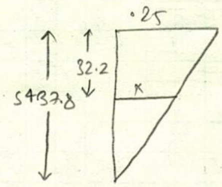
$\therefore x = 0.04$





$$\frac{.25}{x} = \frac{5437.8}{32.2}$$

$\therefore x = 0.001$



Max bending Moment $(M) = 972.2$ lbf in

$T = 73.3$ in lbf

* Design for twisting & bending

$$T_{eq} = \sqrt{T^2 + M^2}$$

$$= \sqrt{(73.3 \text{ in lbf})^2 + (972.2 \text{ lbf in})^2}$$

$$T_{eq} = 975 \text{ lbf in}$$

(A) Using Max shear stress theory,

$$\tau_{max} = \frac{16}{\pi d^3} T_{eq}$$

$$= \frac{16}{\pi \times (.5")^3} \times 975 \text{ lbf in} \quad \therefore \tau_{max} = 39725.1 \text{ lbf in}^{-2} \times \frac{6894.76 \text{ Pa}}{1 \text{ lbf in}^{-2}}$$

$$\boxed{\therefore \tau_{max} = 273.9 \text{ MPa}}$$

(B) Selection of Shaft Material

$$\tau_{max} = \frac{S_{ys}}{FOS}$$

S_{ys} = yield Strength

FOS = Factor of Safety = 3 (Assume)

$$\therefore S_{ys} = 273.9 \text{ MPa} \times 3 = 821.7 \text{ MPa}$$

Material: ArcelorMittal MartINsite® 1200 Cold Rolled Steel

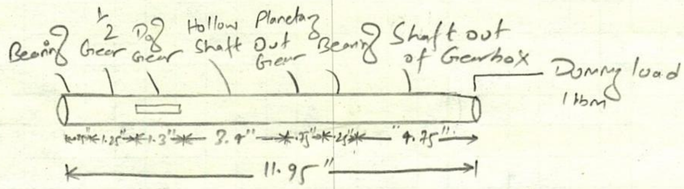
$$S_{ys} = 950 - 1200 \text{ MPa}$$

[Source: Matweb]

Shishir Khanal 6

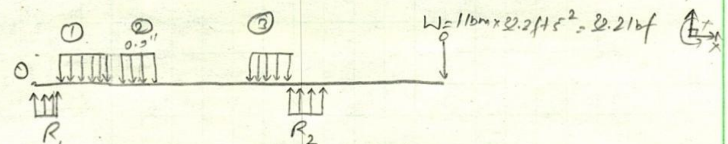
3. Design of Output Shaft

• Shaft Layout



For ①,

Face = 1.25"
PD = 2"
Bore = 0.5"
$S = 0.290\ lbm\ in^{-3}$ (A6 tool steel)



$$\therefore W_1 = 0.290\ lbm\ in^{-3} \times \frac{\pi}{4}\left((2'')^2 - (0.5'')^2\right) \times 1.25'' \times 32.2 ft s^{-2} \times \frac{12 in}{1 ft} = 412.5\ lbf$$

For ②,

Face = 0.9"
PD = 1.5"
Bore = 0.5"
$S = 0.290\ lbm\ in^{-3}$ (A6 tool steel)

$$\therefore W_2 = 0.290\ lbm\ in^{-3} \times \frac{\pi}{4}\left((1.5'')^2 - (0.5'')^2\right) \times 0.9'' \times 32.2 ft s^{-2} \times \frac{12 in}{1 ft} = 158.4\ lbf$$

For ③,

Face = 0.75"
PD = 9.33"
Bore = 0.5"
$S = 0.290\ lbm\ in^{-3}$ (A6 tool steel)

$$\therefore W_3 = 0.290\ lbm\ in^{-3} \times \frac{\pi}{4}\left((9.33'')^2 - (0.5'')^2\right) \times 0.75 \times 32.2 ft s^{-2} \times \frac{12 in}{1 ft} = 5729.3\ lbf$$

Now,
For R's,

$$\Sigma F_y = 0$$

$$R_1 + R_2 = W_1 + W_2 + W_3 + W_D = 412.5 lbf + 158.4 lbf + 5729.3 lbf + 32.2 lbf = 6332.4 lbf \quad —①$$

$$\Sigma M_0 = 0$$

$$R_1 \times \left(\frac{.25''}{2}\right) - W_1 \times \left(.25 + \frac{1.25''}{2}\right) - W_2 \times \left(1.50 + \frac{0.9''}{2}\right) - W_3 \times \left(6.2 + \frac{0.75''}{2}\right) + R_2 \times \left(6.95 + \frac{0.25''}{2}\right) - W_0 \times 11.95'' = 0$$

$$0.125'' R_1 - 360.9 lbf\ in - 308.9 lbf\ in - 37670.1 lbf\ in + 7.1'' R_2 - 384.79 lbf\ in = 0$$

$$0.13'' R_1 + 7.1'' R_2 = 38724.69 \quad —②$$

Solving for R_1 & R_2,

$$\boxed{\therefore R_1 = 894.6 lbf \quad R_2 = 5437.8 lbf}$$

IV. Variable gear mechanism

Variable output Gear Ratio Calculation

$rpm_{in} = 300$ r # of Gear teeths

$rpm_{out_1} = 10$

$rpm_{out_2} = 50$

$$\frac{n_2}{n_1} = \frac{\omega_2}{\omega_1}$$

For output-1,

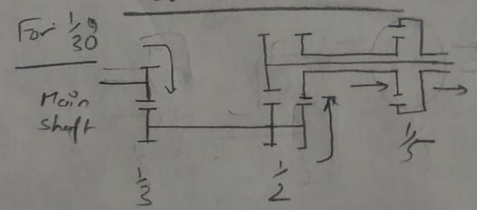
$$\therefore \frac{n_2}{n_1} = \frac{10}{300} = \frac{1}{30}$$

For output-2,

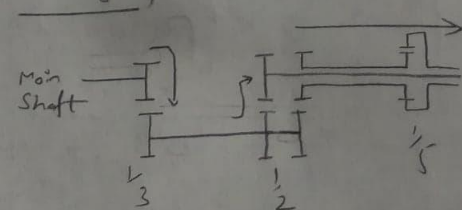
$$\frac{n_2}{n_1} = \frac{50}{300} = \frac{1}{6}$$

Mechanism sketch

For $\frac{1}{30}$



For $\frac{1}{6}$,



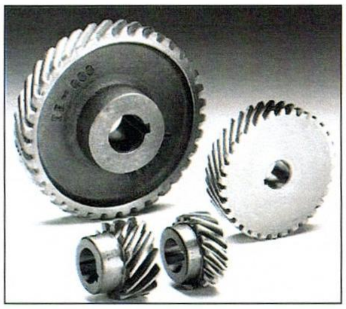
V. Gear Catalogue for Helical Gears

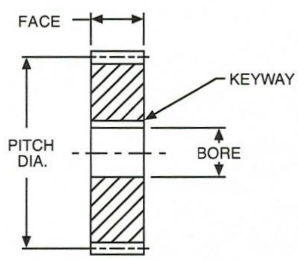
Helical Gears

24 through 10 Transverse Diametral Pitch (Steel – Hardened)

14-1/2° Normal Pressure Angle – 45° Helix Angle

B





These gears are hardened all over, except as noted. Teeth on all steel gears are polished.

ALL DIMENSIONS IN INCHES
ORDER BY CATALOG NUMBER OR ITEM CODE

No. of Teeth	Pitch Dia.	Bore	Keyway	Style See Page 323	RIGHT HAND Catalog Number	RIGHT HAND Item Code	LEFT HAND Catalog Number	LEFT HAND Item Code
24 TRANSVERSE DIAMETRAL PITCH					Face: 8-15 Teeth = .375" 18-72 Teeth = .250"			
8	.333	.1875	*	A	H2408R	18268	H2408L	18270
10	.417	.250	**		H2410R	18272	H2410L	18274
12	.500				H2412R	18276	H2412L	18278
15	.625	.375			H2415R	18280	H2415L	18282
18	.750		1/8 x 1/16		H2418R	18284	H2418L	18286
20	.833	.500			H2420R	18288	H2420L	18290
24	1.000				H2424R	18292	H2424L	18294
30	1.250	.625			H2430R	18296	H2430L	18298
36	1.500				H2436R†	18300	H2436L†	18302
48	2.000				H2448R†	18304	H2448L†	18306
60	2.500				H2460R†	18308	H2460L†	18310
72	3.000				H2472R†	18312	H2472L†	18314
20 TRANSVERSE DIAMETRAL PITCH					Face: 8-15 Teeth = .563" 18-72 Teeth = .375"			
8	.400	.250	**	A	H2008R	18228	H2008L	18230
10	.500	.3125			H2010R	18232	H2010L	18234
12	.600	.375			H2012R	18236	H2012L	18238
15	.750	.4375	1/8 x 1/16		H2015R	18240	H2015L	18242
20	1.000	.500			H2020R	18244	H2020L	18246
25	1.250	.625			H2025R	18248	H2025L	18250
30	1.500	.750	3/16 x 3/32		H2030R†	18252	H2030L†	18254
40	2.000				H2040R†	18256	H2040L†	18258
50	2.500				H2050R†	18260	H2050L†	18262
60	3.000				H2060R†	18264	H2060L†	18266
16 TRANSVERSE DIAMETRAL PITCH					Face = .500"			
12	.750	.375	1/16 x 1/32		H1612R	18200	H1612L	18202
16	1.000				H1616R	18204	H1616L	18206
20	1.250	.500	1/8 x 1/16	A	H1620R	18208	H1620L	18210
24	1.500				H1624R†	18212	H1624L†	18214
32	2.000				H1632R†	18216	H1632L†	18218
40	2.500				H1640R†	18220	H1640L†	18222
48	3.000				H1648R†	18224	H1648L†	18226
12 TRANSVERSE DIAMETRAL PITCH					Face = .750"			
12	1.000	.625	1/8 x 1/16	A	H1212R	18170	H1212L	18168
15	1.250				H1215R	18174	H1215L	18172
18	1.500				H1218R†	18178	H1218L†	18176
24	2.000				H1224R†	18182	H1224L†	18180
30	2.500				H1230R†	18186	H1230L†	18184
36	3.000				H1236R†	18190	H1236L†	18188
10 TRANSVERSE DIAMETRAL PITCH					Face = .875"			
8	.800	.375	1/16 x 1/32	A	H1008R	18130	H1008L	18128
10	1.000	.500	1/8 x 1/16		H1010R	18134	H1010L	18132
12	1.200	.625			H1012R	18138	H1012L	18136
15	1.500	.750	3/16 x 3/32		H1015R†	18142	H1015L†	18140
20	2.000				H1020R†	18146	H1020L†	18144
25	2.500				H1025R†	18148	H1025L†	18150
30	3.000				H1030R†	18154	H1030L†	18152
40	4.000				H1040R†	18158	H1040L†	18156

*1/16" wide x .04" deep slot cut on end of gear for drive pin, not key.
**3/32" wide x .06" deep slot cut on end of gear for drive pin, not key.
†Teeth only hardened.

STANDARD TOLERANCES

DIMENSION		TOLERANCE
BORE	All	±.0005

REFERENCE PAGES

Alterations — 322
Horsepower Ratings — 67
Lubrication — 322
Materials — 323
Selection Procedure — 66

NOTE: Normal Diametral Pitch is equal to the Transverse Diametral Pitch divided by the cosine of the Helix Angle.

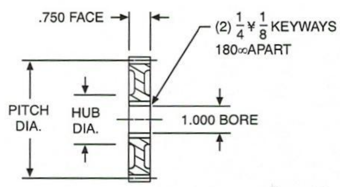
VI. Gear Catalogue for Change Gears (for Planetary Gears)

Change Gears

12 Diametral Pitch (Steel & Cast Iron)

14-1/2° Pressure Angle (will not operate with 20° spurs)

A



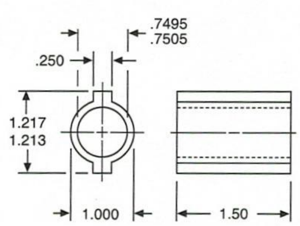
12 D.P.

REFERENCE PAGES

Alterations — 322
Horsepower Ratings — 53
Lubrication — 322
Materials — 323
Selection Procedure — 49

Compound Steel Bushings

These steel bushings have 2 keys, 180° apart and fit bores of GD series change gears with a slip fit. They are used to mount two gears on one shaft (or stud) and drive one from the other.



ORDER BY CATALOG NUMBER OR ITEM CODE

CATALOG NO.	ITEM CODE
GDB12A	18504

ALL DIMENSIONS IN INCHES
ORDER BY CATALOG NUMBER OR ITEM CODE

12 DIAMETRAL PITCH — Outside Dia. = Pitch Dia. +.167"

No. of Teeth	Pitch Dia.	Hub Dia.	Catalog Number	Item Code
STEEL				
20	1.667		GD20	10142
21	1.750		GD21	10144
22	1.833		GD22	10146
23	1.917		GD23	10148
24	2.000		GD24	10150
25	2.083		GD25	10152
26	2.167		GD26	10154
27	2.250		GD27	10156
28	2.333	–	GD28	10158
29	2.417		GD29	10160
30	2.500		GD30	10162
31	2.583		GD31	10164
32	2.667		GD32	10166
33	2.750		GD33	10168
34	2.833		GD34	10170
35	2.917		GD35	10172
36	3.000		GD36	10174
CAST IRON				
37	3.083		GD37B	11118
38	3.167		GD38B	11120
39	3.250		GD39B	11122
40	3.333		GD40B	11124
41	3.417		GD41B	11126
42	3.500		GD42B	11128
43	3.583	1.76	GD43B	11130
44	3.667		GD44B	11132
45	3.750		GD45B	11134
46	3.833		GD46B	11136
47	3.917		GD47B	11138
48	4.000		GD48B	11140
49	4.083		GD49B	11142
50	4.167		GD50B	11144
51	4.250		GD51B	11146
52	4.333		GD52B	11148
53	4.417		GD53B	11150
54	4.500		GD54B	11152
55	4.583		GD55B	11154
56	4.667		GD56B	11156
57	4.750		GD57B	11158
58	4.833		GD58B	11160
59	4.917		GD59B	11162
60	5.000		GD60B	11164
61	5.083	2.19	GD61B	11166
62	5.167		GD62B	11168
63	5.250		GD63A	11170
64	5.333		GD64A	11172
65	5.417		GD65A	11174
66	5.500		GD66A	11176
67	5.583		GD67A	11178
68	5.667		GD68A	11180
69	5.750		GD69A	11182
70	5.833		GD70A	11184
71	5.917		GD71A	11186
72	6.000		GD72A	11188
73	6.083		GD73A	11190
74	6.167		GD74A	11192

No. of Teeth	Pitch Dia.	Hub Dia.	Catalog Number	Item Code
CAST IRON				
75	6.250		GD75A	11194
76	6.333		GD76A	11196
77	6.417		GD77A	11198
78	6.500		GD78A	11200
79	6.583	2.19	GD79A	11202
80	6.667		GD80A	11204
81	6.750		GD81A	11206
82	6.833		GD82A	11208
83	6.917		GD83A	11210
84	7.000		GD84A	11212
85	7.083		GD85A	11214
86	7.167		GD86A	11216
87	7.250		GD87A	11218
88	7.333		GD88A	11220
89	7.417		GD89A	11222
90	7.500		GD90A	11224
91	7.583		GD91A	11226
92	7.667		GD92A	11228
93	7.750		GD93A	11230
94	7.833		GD94A	11232
95	7.917		GD95A	11234
96	8.000		GD96A	11236
97	8.083		GD97A	11238
98	8.167		GD98A	11240
99	8.250		GD99A	11242
100	8.333	2.44	GD100A	11244
101	8.417		GD101A	11246
102	8.500		GD102A	11248
103	8.583		GD103A	11250
104	8.667		GD104A	11252
105	8.750		GD105A	11254
106	8.833		GD106A	11256
107	8.917		GD107A	11258
108	9.000		GD108A	11260
109	9.083		GD109A	11262
110	9.167		GD110A	11264
111	9.250		GD111A	11266
112	9.333		GD112A	11268
113	9.417		GD113A	11270
114	9.500		GD114A	11272
115	9.583		GD115A	11274
116	9.667		GD116A	11276
117	9.750		GD117A	11278
118	9.833		GD118A	11280
119	9.917		GD119A	11282
120	10.000		GD120A	11284

Style See Page 323	
	20 – 60 Teeth – A
	61 – 120 Teeth – C

VII. Gear Catalogue for Spur Gears(for Shaft Power Transmission)

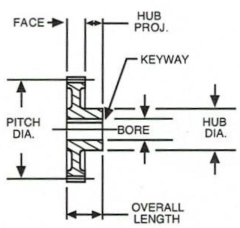
Spur Gears

16 and 12 Diametral Pitch (Steel, Non-Metallic & Cast Iron)

14-1/2° Pressure Angle (will not operate with 20° spurs)

ALL DIMENSIONS IN INCHES
ORDER BY CATALOG NUMBER OR ITEM CODE

No. of Teeth	Pitch Dia.	Bore	Hub Dia.	Hub Proj.	Style See Page 323	Without Keyway or Setscrew Catalog Number	Without Keyway or Setscrew Item Code	With Keyway and Setscrew† Catalog Number	With Keyway and Setscrew† Item Code
16 DIAMETRAL PITCH					Face = .500" Outside Dia. = Pitch Dia. + .125" Overall Length = .500" + Hub Proj.				
STEEL									
32	2.000	.500	1.70	.50	A	NB32	09730	NB32-1/2†	46055
		.625				–	–	NB32-5/8	46056
		.750				–	–	NB32-3/4	46057
		.875				–	–	NB32-7/8	46058
		1.000				–	–	NB32-1	46059
36	2.250	.500	1.95	.50		NB36	09732	–	–
40	2.500		1.69			NB40A	10244	–	–
48	3.000		2.19			NB48A	10246	–	–
NON-METALLIC									
16	1.000	.375	.81	.50	A	QBH16	09014	–	–
20	1.250		1.06			QBH20	09018	–	–
24	1.500	.500	1.31	.50		QBH24	09022	–	–
32	2.000		1.81	.50		QBH32	09024	–	–
40	2.500		–	–		QB40	09000	–	–
48	3.000		–	–		QB48	09002	–	–
64	4.000		–	–		QB64	09006	–	–
CAST IRON									
54	3.375	.500	1.25	.50	B	NB54	10248	–	–
56	3.500		1.25	.50		NB56	10250	–	–
60	3.750		1.38	.62		NB60	10252	–	–
64	4.000	.625	1.38	.62	C	NB64	10254	–	–
72	4.500		1.38			NB72	10256	–	–
80	5.000		1.50			NB80	10258	–	–
84	5.250		1.50			NB84	10260	–	–
96	6.000		1.50		D	NB96	10262	–	–
112	7.000		1.50			NB112	10264	–	–
120	7.500		1.50			NB120	10266	–	–
128	8.000		1.50			NB128	10268	–	–
144	9.000	.625	1.75	.75		NB144	10270	–	–
160	10.000		1.75			NB160B	10272	–	–
192	12.000		2.00			NB192B	10274	–	–
12 DIAMETRAL PITCH					Face = .750" Outside Dia. = Pitch Dia. + .167" Overall Length = .750" + Hub Proj.				
STEEL									
11	1.000*	.500	.75	.50	A	ND11B	09744	ND11B-1/2†	46060
12	1.000		.75			ND12B	09746	ND12B-1/2†	46061
13	1.083		.83			ND13B	09748	ND13B-1/2†	46062
14	1.167		.92			ND14B	09750	ND14B-1/2†	46063
15	1.250	.625	1.00	.50		ND15B	09752	ND15B-5/8	46064
16	1.333		.99			ND16B	09754	ND16B-5/8	46065
18	1.500		1.15			ND18B	09756	ND18B-5/8	46066
20	1.667	.625	1.32	.50		ND20B	09758	ND20B-5/8	46067
		.750				–	–	ND20B-3/4	46068
21	1.750	.625	1.40	.50		ND21B	09760	ND21B-5/8	46069
		.750				–	–	ND21B-3/4	46070
		.875				–	–	ND21B-7/8	46071
22	1.833	.625	1.49	.50		ND22B	09762	ND22B-5/8	46072
		.750				–	–	ND22B-3/4	46073
		.875				–	–	ND22B-7/8	46074
		1.000				–	–	ND22B-1	46075
24	2.000	.625	1.65	.50		ND24B	09764	ND24B-5/8	46076
		.750				–	–	ND24B-3/4	46077
		.875				–	–	ND24B-7/8	46078
		1.000				–	–	ND24B-1	46079
30	2.500	.625	2.15	.62		ND30	09766	–	–
32	2.667		1.92			ND32A	10276	–	–
36	3.000		2.25			ND36A	10278	–	–
40	3.333		2.34			ND40A	10280	–	–
42	3.500		2.50			ND42A	10282	–	–



STANDARD TOLERANCES

DIMENSION		TOLERANCE
BORE	All	±.0005

16 D.P. 12 D.P.

REFERENCE PAGES

Alterations — 322
Horsepower Ratings — 52, 53
Lubrication — 322
Materials — 323
Selection Procedure — 49

*Special Pitch Diameter, used for calculating Center Distance only, not Ratio.

†1/2" bore has one setscrew, no keyway.

5/8" bore and larger have standard keyway at 90° to setscrew. See Page 323.

VIII. Torque-Speed criteria of Clutch-Brake System

Selection Criteria

In addition to the solution steps on previous pages, the dynamic torque required may be calculated.

There are two methods you can use to calculate the dynamic torque required.

$$T_d = \left[\frac{WR^2 \times N \pm T_L}{C \times t}\right] \times S.F.$$

Where:

- WR^2 = Total inertia reflected to the clutch/brake, lb.–in.^{2} (kg.m^2)
- N = Shaft speed at clutch/brake, RPM
- C = Constant, use 3696 for English units and 9.55 for metric units
- t = Desired stopping or acceleration time, seconds
- T_L = Load torque to overcome other than inertia, lb.–in. (N–m)
- S.F. = Service Factor, 1.4 recommended
- T_d = Average dynamic torque, lb.–in. (N–m)

Note: $+T_L$ = engage a clutch or accelerate
$-T_L$ = brake or decelerate

The relationship between the horsepower and speed can also be calculated to determine the dynamic torque required is expressed as:

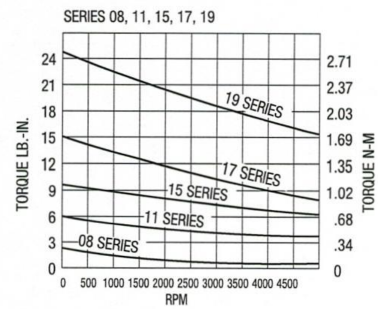
$$T_d = \frac{63{,}025 \times P}{N} \times S.F.$$

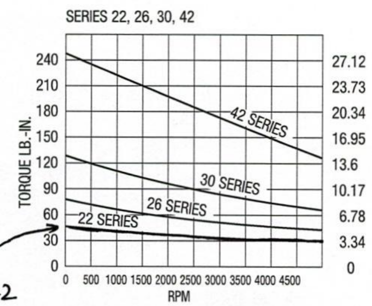
Where:

- T_d = Average dynamic torque, lb.–in.
- P = Horsepower, HP
- N = Shaft Speed
- S.F. = Service Factor
- 63,025 = Constant

Inertia Dynamics clutches and brakes are rated by static torque. The following charts may be used to estimate the dynamic torque.

Dynamic Torque Curve





SoB 22

Torque Data

CLUTCHES: CLUTCH COUPLINGS: POWER ON BRAKES			
SERIES	TYPICAL OUT-OF-BOX TORQUES LB. - IN. (N-M)	RATED STATIC TORQUES LB. - IN. (N-M)	TYPICAL TORQUES AFTER BURNISHING LB. - IN. (N-M)
08	2 (.23)	2.5 (.28)	3 (.34)
11	5 (.56)	6 (.68)	8 (.90)
15	8 (.90)	10 (1.13)	15 (1.69)
17	12 (1.36)	15 (1.70)	20 (2.26)
19	20 (2.26)	25 (2.82)	30 (3.39)
22	40 (4.52)	50 (5.65)	60 (6.78)
26	65 (7.34)	80 (9.04)	90 (10.17)
30	100 (11.30)	125 (14.12)	150 (16.95)
42	225 (25.42)	250 (28.25)	275 (31.07)

Burnishing

Burnishing is a wearing-in or mating process which will ensure the highest possible output torques. Burnishing is accomplished by forcing the brake to slip rotationally when energized. Best results are obtained when the unit is energized at 30-40% of rated voltage and forced to slip for a period of 2-3 minutes at a low speed of 30-200 RPM depending on the unit size. Units in applications with high inertial loads and high speed will usually become burnished in their normal operating mode. Whenever possible, it is desirable to perform the burnishing operation in the final location so the alignment of the burnished faces will not be disturbed. For additional information on burnishing procedures for power-on brakes and clutches ask for burnishing spec. #040-1001.

IX. Datasheet of Clutch Brake System

Electromagnetic Friction Clutches & Brakes

Shaft Mounted Clutch/Power-On Brake – Type SLB & SOB Imperial

Mechanical

MODEL NO.	STATIC TORQUE LB. - IN.	INERTIA LB. - IN.^{2} ROTOR	INERTIA LB. - IN.^{2} ARM & HUB	WEIGHT OZ.
SLB11 SOB11	6	.0011	.0029 .0024	7
SLB17 SOB17	15	.0024	.0360 .0310	22
SLB19 SOB19	25	.026	.0470 .0420	25
SLB22 SOB22	50	.031	.0790 .0700	45
SLB26 SOB26	80	.042	.2920 .3200	60

Electrical

MODEL NO.	90 VDC		24 VDC		12 VDC	
	AMPS	OHMS	AMPS	OHMS	AMPS	OHMS
SLB11 SOB11	.047	1930	.198	121	.447	26.8
SLB17 SOB17	.066	1369	.289	83	.561	21.4
SLB19 SOB19	.074	1213	.322	74.4	.574	20.9
SLB22 SOB22	.079	1140	.322	74.6	.628	19.1
SLB26 SOB26	.088	1024	.350	67.1	.667	18.0

Lead wire is UL recognized style 1213, 1015 or 1429, 22 gage.
Insulation is .050 O.D. on 11 unit; .064 or .095 O.D. on all other units.

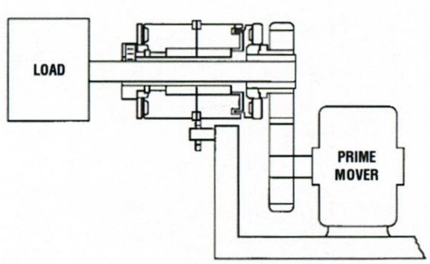
Dimensions

MODEL NO.	A MAX.	B REF.	C NOM.	D MAX.	E NOM.	F *	G MAX.	H **	I NOM.	J MAX.	K MAX.	L MAX.	M MAX.	N MIN.	O ± .5	P MAX.	Q MIN.	R MIN.	S MAX.	KEYWAYS BORE	NOMINAL KEYWAY X	NOMINAL KEYWAY Y
SLB11	2.225	.974	1.229	.051	.094	.410	.700	.506	1/4 5/16	1.160	.700	1.240	.520	.140	12.00	.630	.630	.300	1.050	N.A.	SET SCREWS ONLY	
SOB11	1.970	.974	.983	.051	.094	.094	.700	–	1/4 5/16	1.160	.700	1.240	.520	.140	12.00	.630	.630	.300	1.050	N.A.	SET SCREWS ONLY	
SLB17	2.855	1.245	1.590	.066	.114	.390	1.207	.629	1/4 5/16 3/8	1.780	1.207	1.960	.520	.190	12.00	.990	1.100	.510	1.707	1/4 5/16 3/8	.0625 – .0655 .0625 – .0655 .094 – .097	.285 – .290 .347 – .352 .417 – .427
SOB17	2.608	1.245	1.340	.066	.114	.114	1.207	–	1/4 5/16 3/8	1.780	1.207	1.960	.520	.190	12.00	.990	1.100	.470	1.707	1/4 5/16 3/8	.0625 – .0655 .0625 – .0655 .094 – .097	.285 – .290 .347 – .352 .417 – .427
SLB19	2.993	1.258	1.715	.066	.114	.475	1.207	.756	5/16 3/8	2.000	1.207	1.960	.520	.190	12.00	.990	1.100	.470	1.707	5/16 3/8	.0625 – .0655 .094 – .097	.347 – .352 .417 – .427
SOB19	2.615	1.258	1.337	.066	.114	.114	1.207	–	5/16 3/8	2.000	1.207	1.960	.520	.190	12.00	.990	1.100	.470	1.707	5/16 3/8	.0625 – .0655 .094 – .097	.347 – .352 .417 – .427
SLB22	3.737	1.722	1.995	.093	.115	.450	1.453	.756	3/8 1/2	2.260	1.453	2.340	.580	.190	18.00	1.180	1.136	.480	1.832	3/8 1/2	.094 – .097 .125 – .128	.417 – .427 .560 – .567
SOB22	3.552	1.722	1.810	.093	.115	.115	1.453	–	3/8 1/2	2.260	1.453	2.340	.580	.190	18.00	1.180	1.136	.480	1.832	3/8 1/2	.094 – .097 .125 – .128	.417 – .427 .560 – .567
SLB26	4.050	1.778	2.240	.093	.150	.427	1.610	.999	3/8 1/2 5/8	2.640	1.450	2.650	.645	.190	18.00	1.335	1.730	.480	2.395	3/8 1/2 5/8	.094 – .097 .125 – .128 .1885 – .1905	.417 – .427 .560 – .567 .709 – .716
SOB26	3.677	1.815	1.842	.093	.150	.150	1.450	–	3/8 1/2 5/8	2.640	1.450	2.650	.645	.190	18.00	1.335	1.730	.480	2.395	3/8 1/2 5/8	.094 – .097 .125 – .128 .1885 – .1905	.417 – .427 .560 – .567 .709 – .716

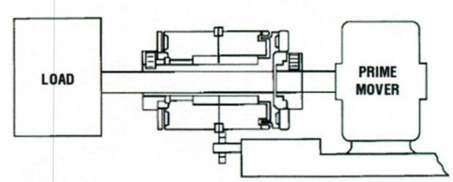
*SLB maximum; SOB nominal.
**SLB hub O.D. ± .002; SOB hub length nominal.

Notes:

1. SLB 26 units have (3)–#8-32 tapped holes on 1.375 in. B.C. in armature hub face instead of knurl.



SLB



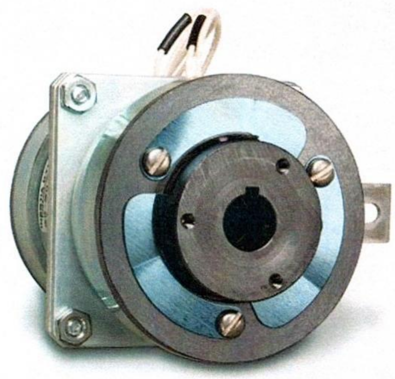
SOB

See page 4 for Ordering Information

X. Drawing of Clutch-Brake System

Electromagnetic Friction Clutches & Brakes

Shaft Mounted Clutch/Power-On Brake – Type SLB & SOB



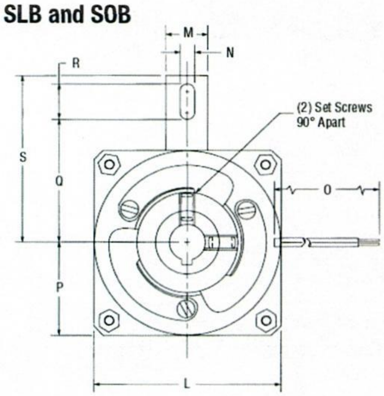
SLB & SOB SERIES POWER-ON BRAKES

Shaft Mounted Clutch Brakes – Type SLB & SOB

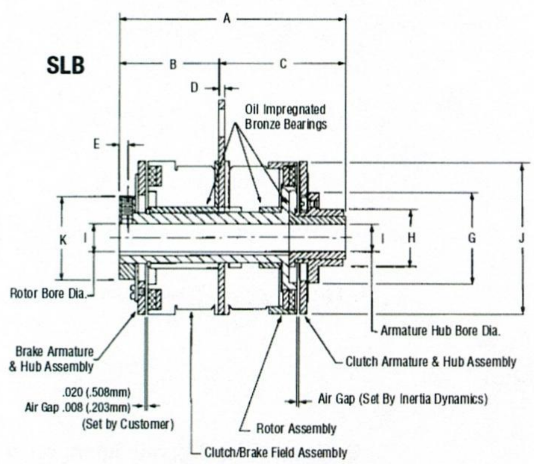
The SLB and SOB series are shaft mounted clutch/power-on brake packages that are used to couple two parallel or in-line shafts. The clutch/brake package combines the features of our model SL or SO with an FB into one unit for easy installation. The clutch armature hub accommodates a pulley, gear, sprocket, etc., to transmit torque to the second shaft. The brake is used to stop or hold the load. The clutch/brake package is shaft mounted and retained by a loose-fitting pin or bracket through the anti-rotation tab.

Customer Shall Maintain:
A loose-fitting pin through the anti-rotation tab to prevent preloading the bearings; initial air gap setting of .008-.020 inches (.203-.508mm) on the brake side. On SOB models concentricity between the shafts within .005 (.127mm) T.I.R.

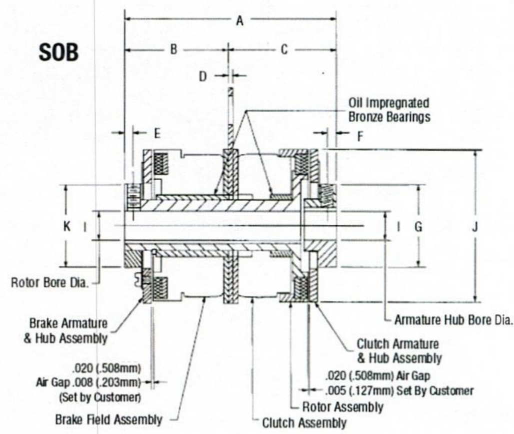
SLB and SOB



SLB

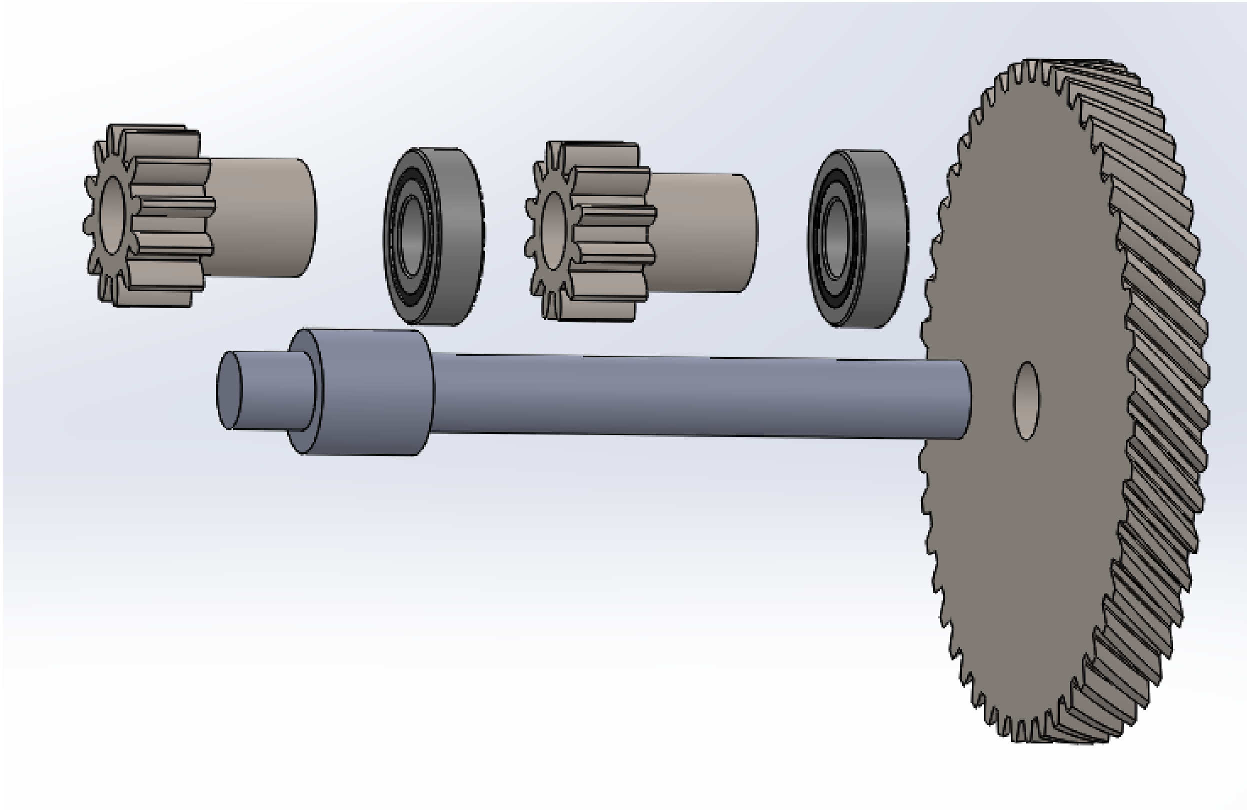


SOB

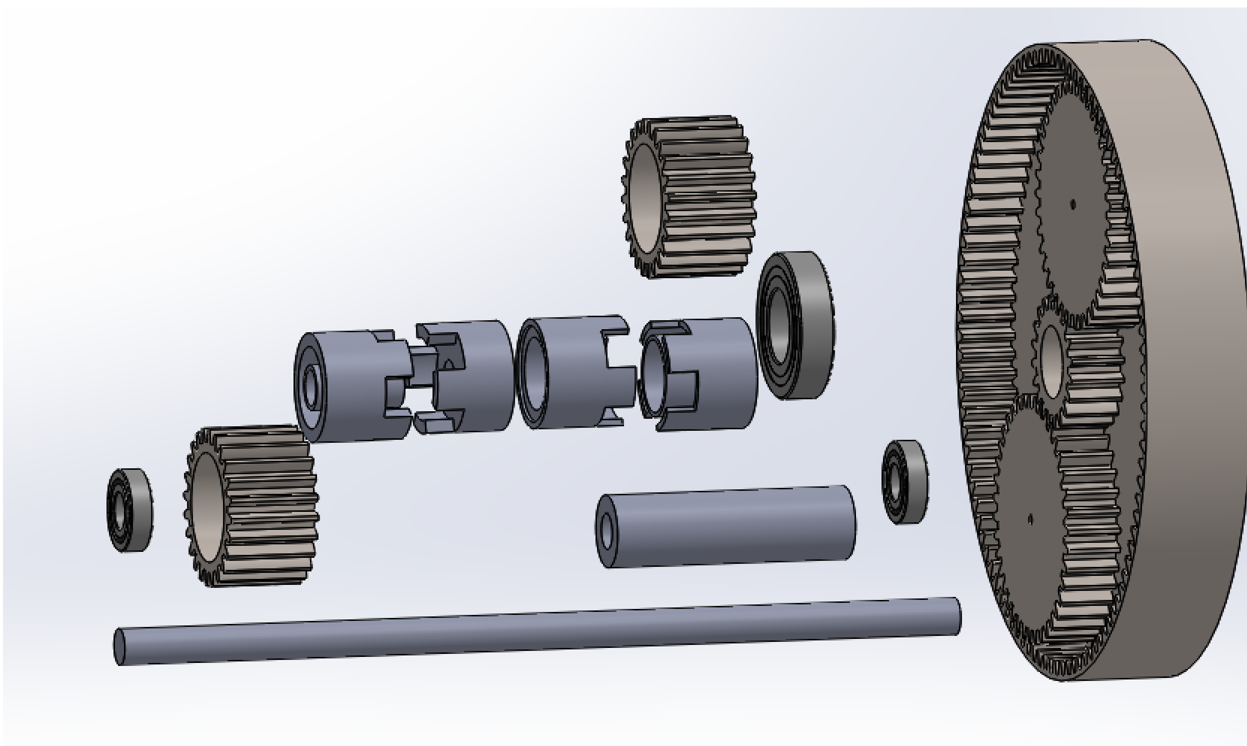


XI. Exploded View of Subsystems

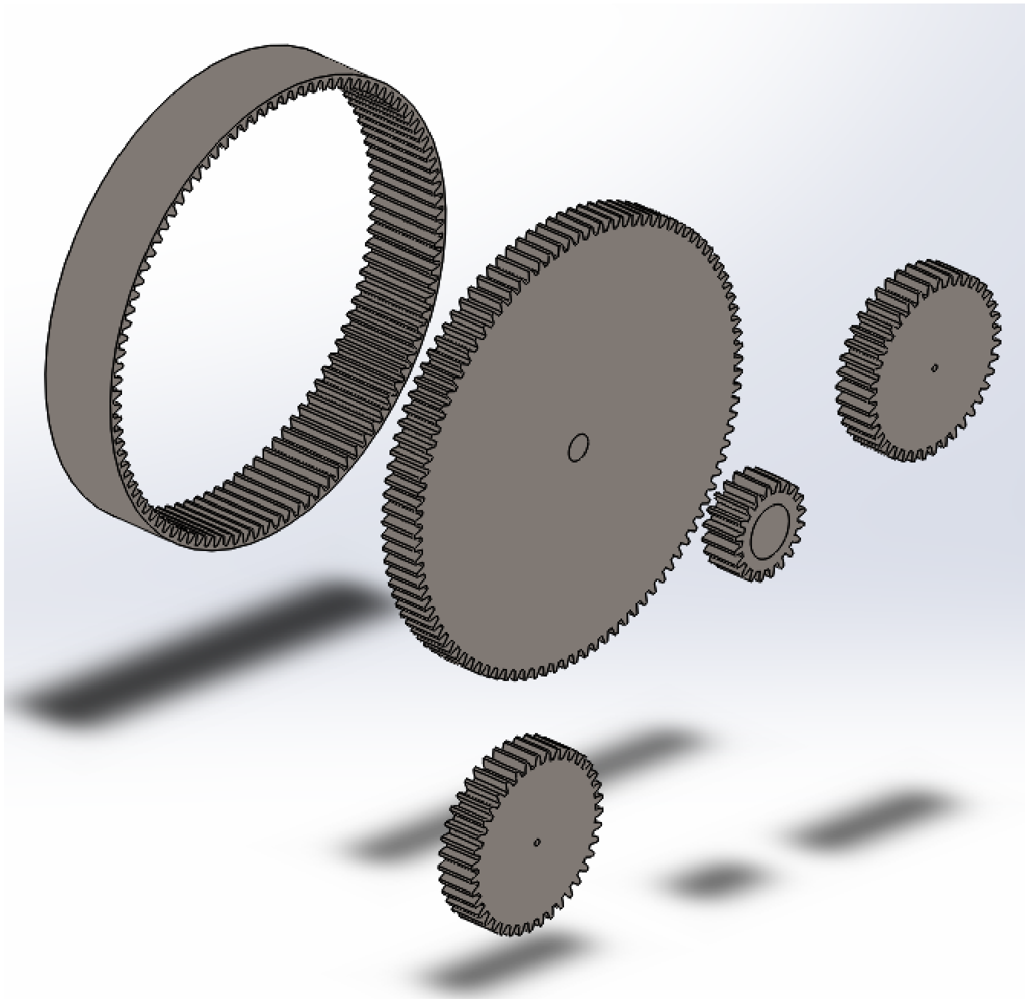
A. Intermediate Shaft



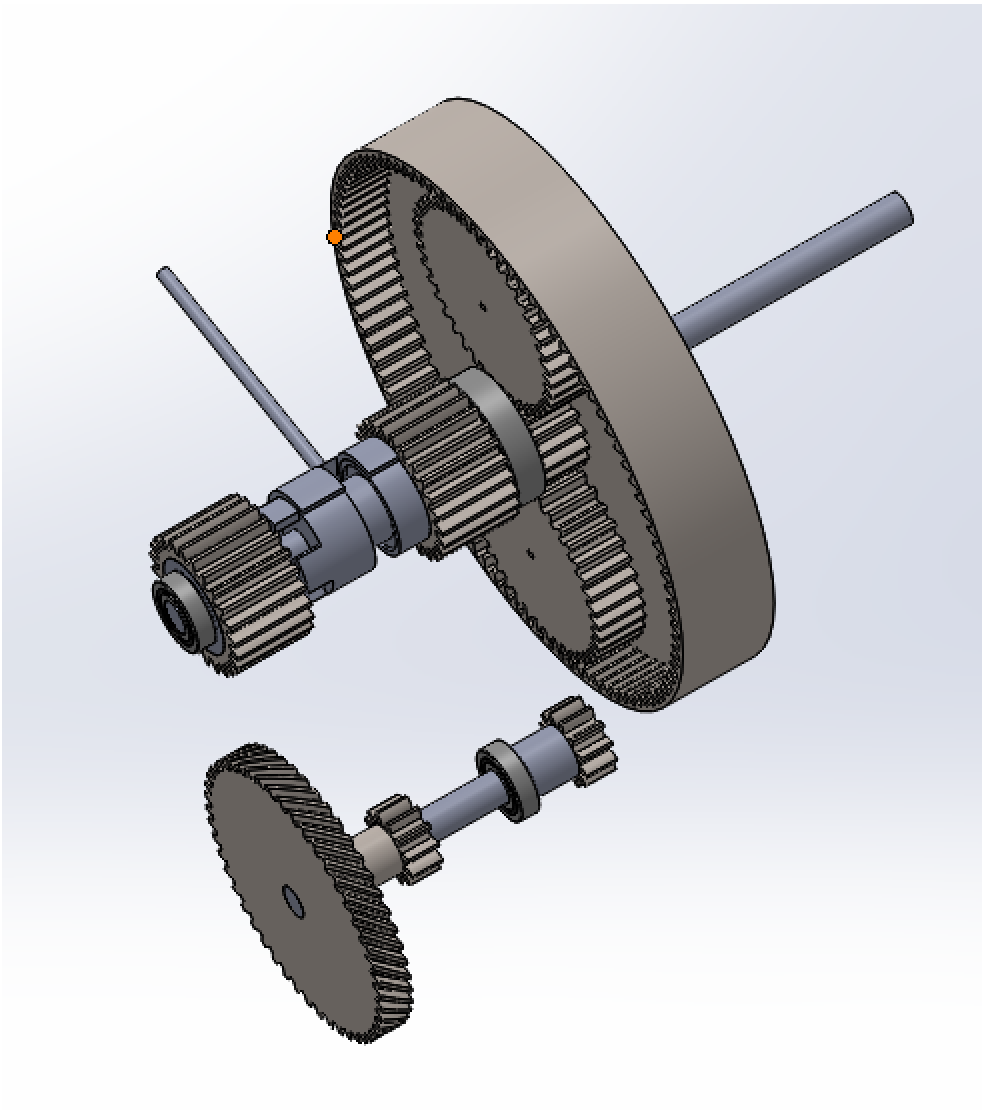
B. Main Output Shaft



C. Exploded view of Planetary Gear

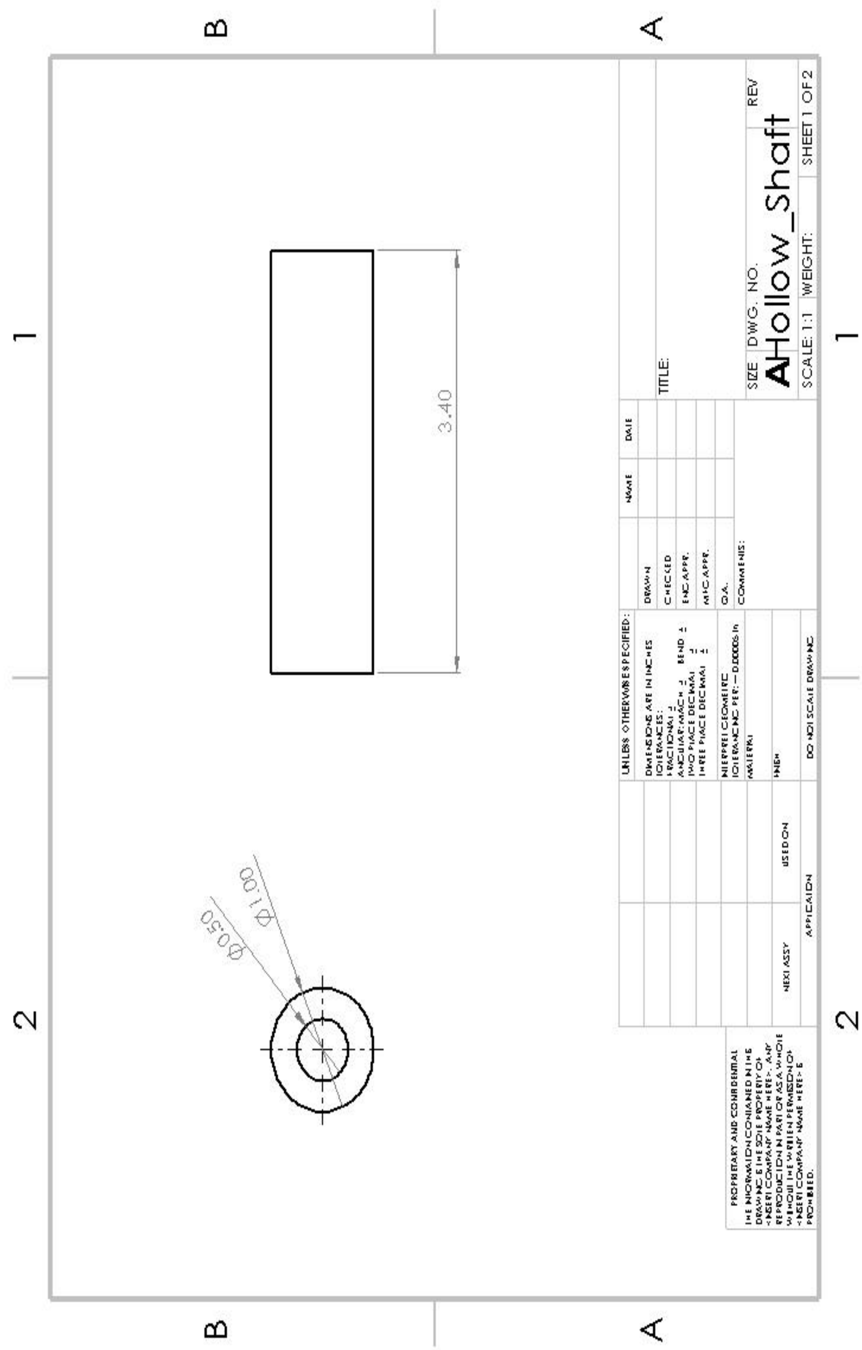


D. Final Assembly

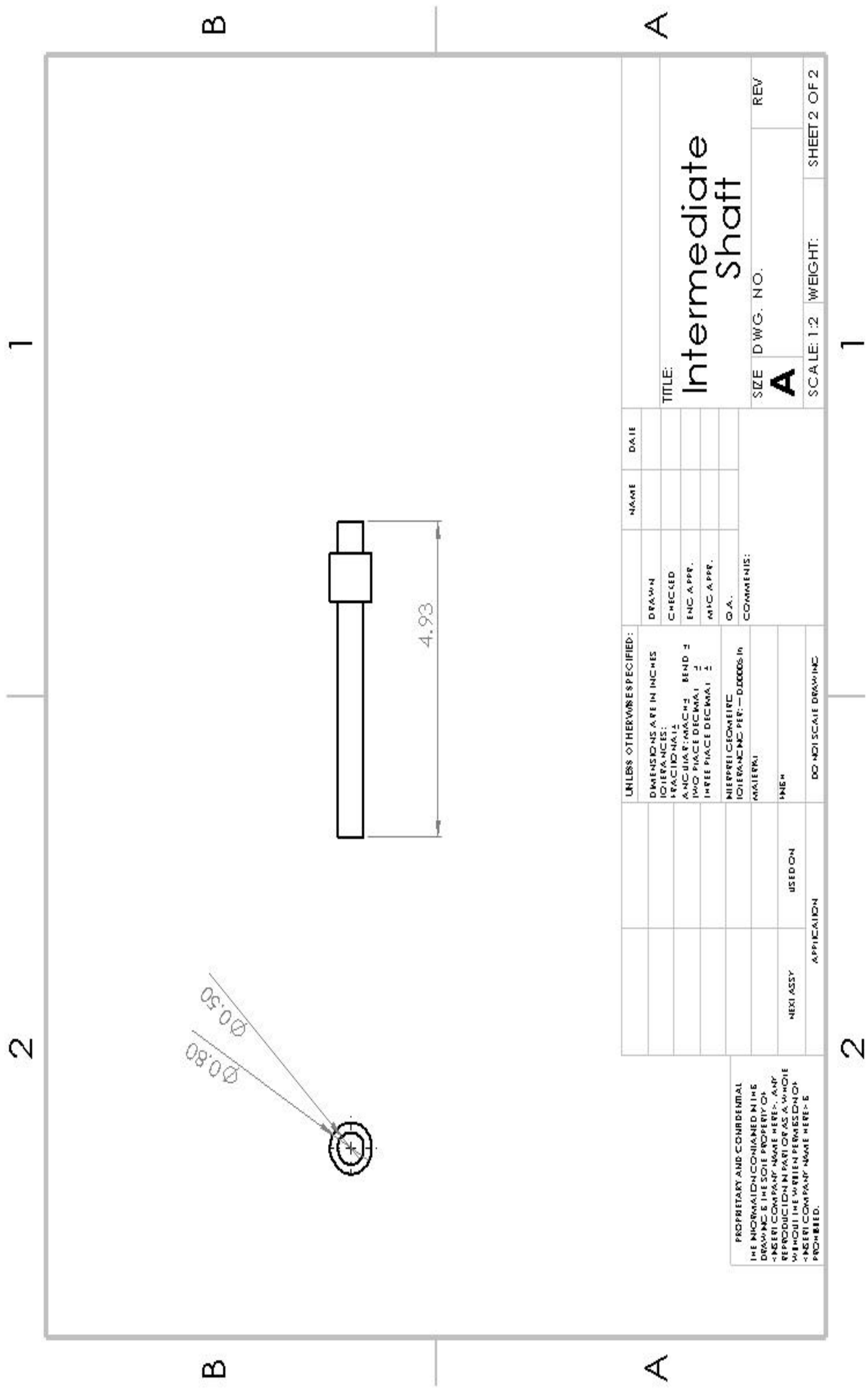


XII: Drawings

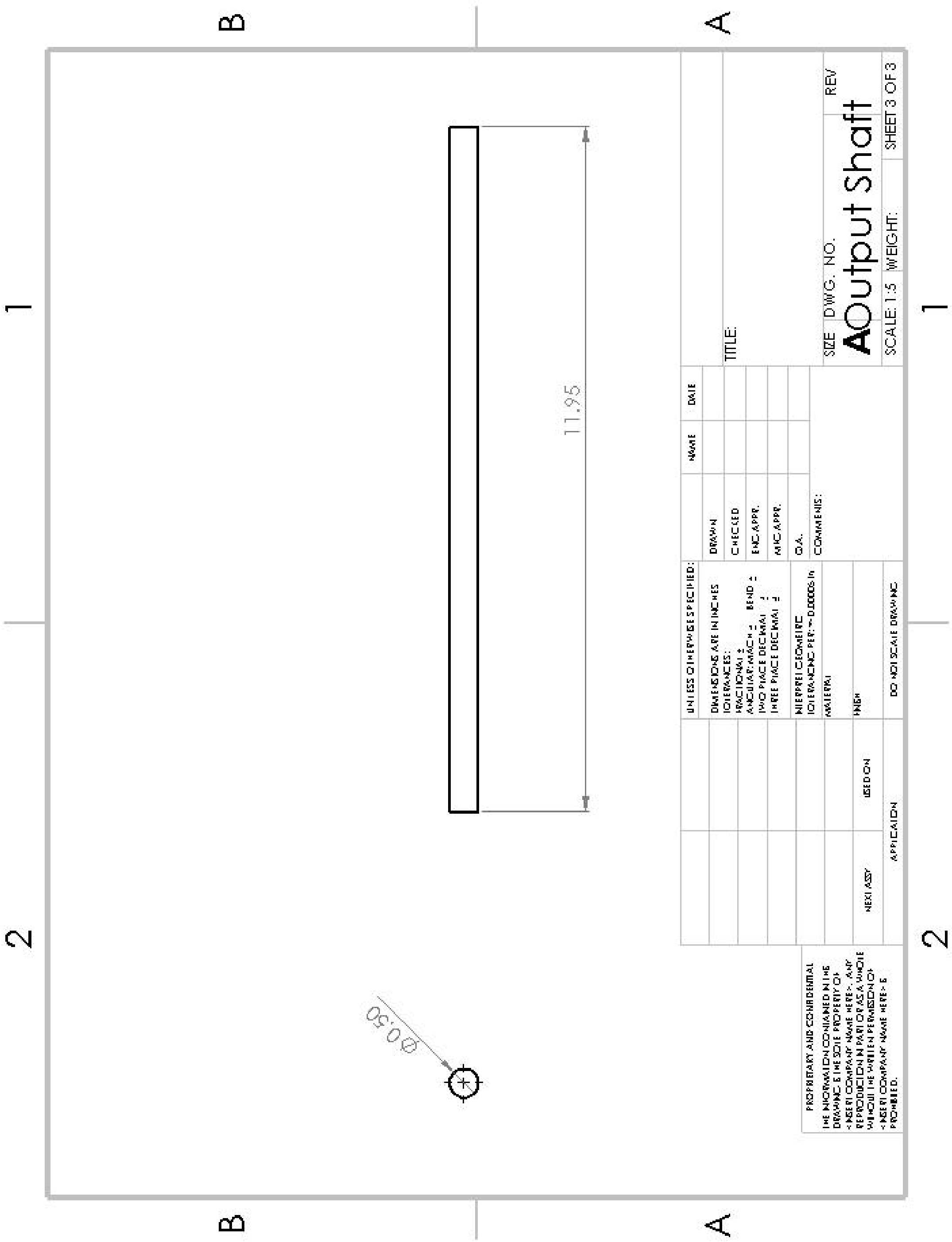
1. Shafts
    a. Hollow Shaft



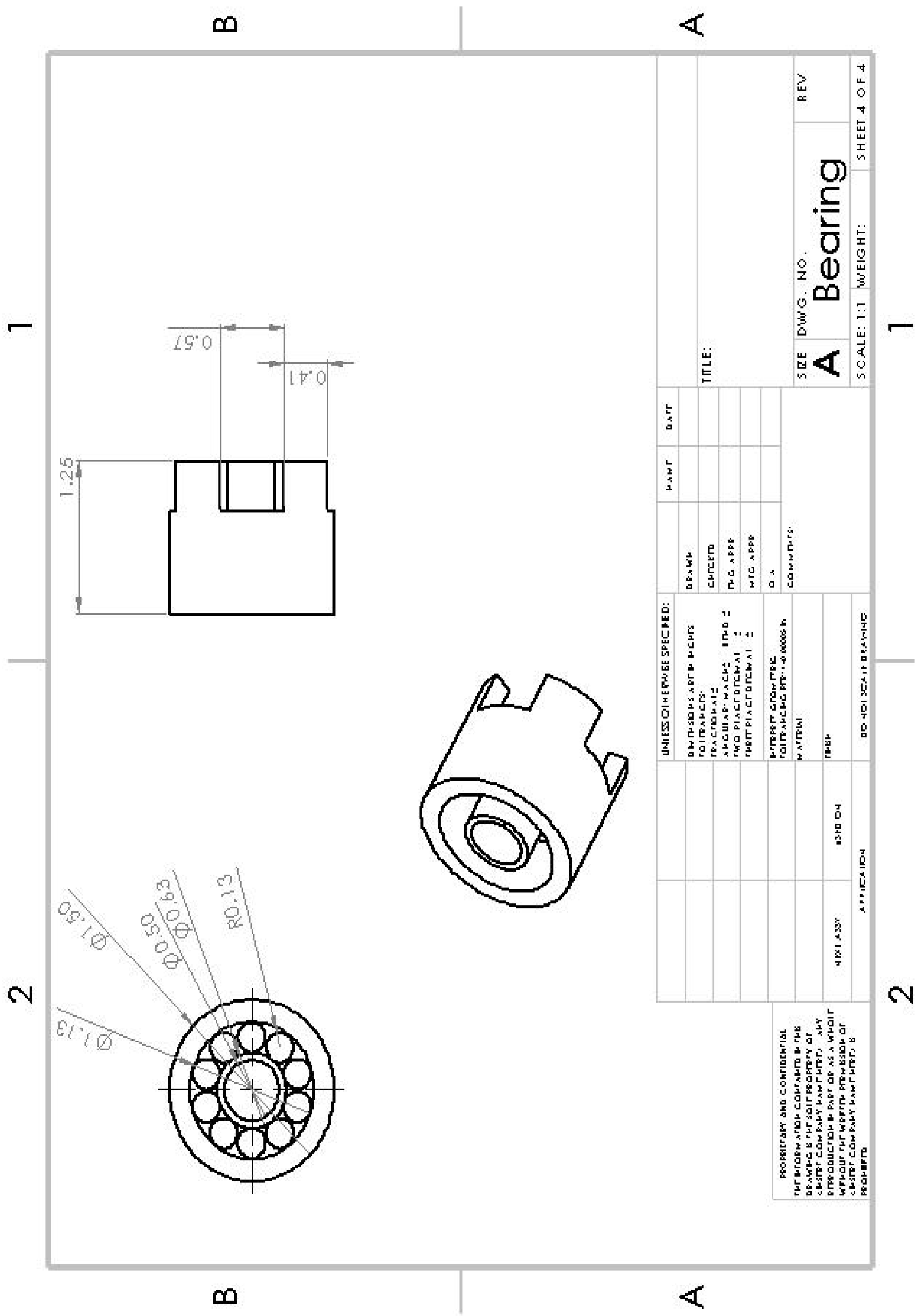
b. Intermediate Shaft



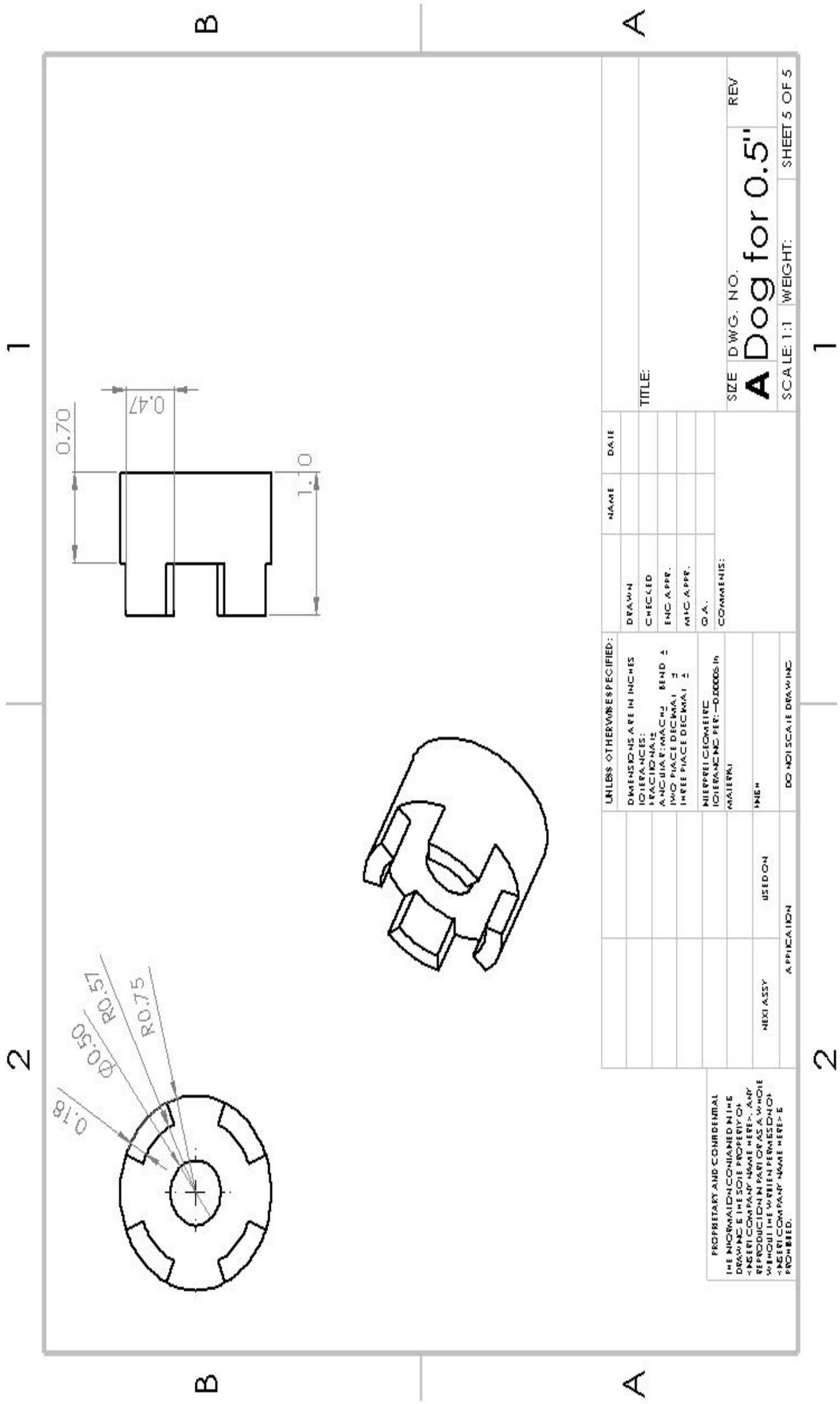
c. Output Shaft



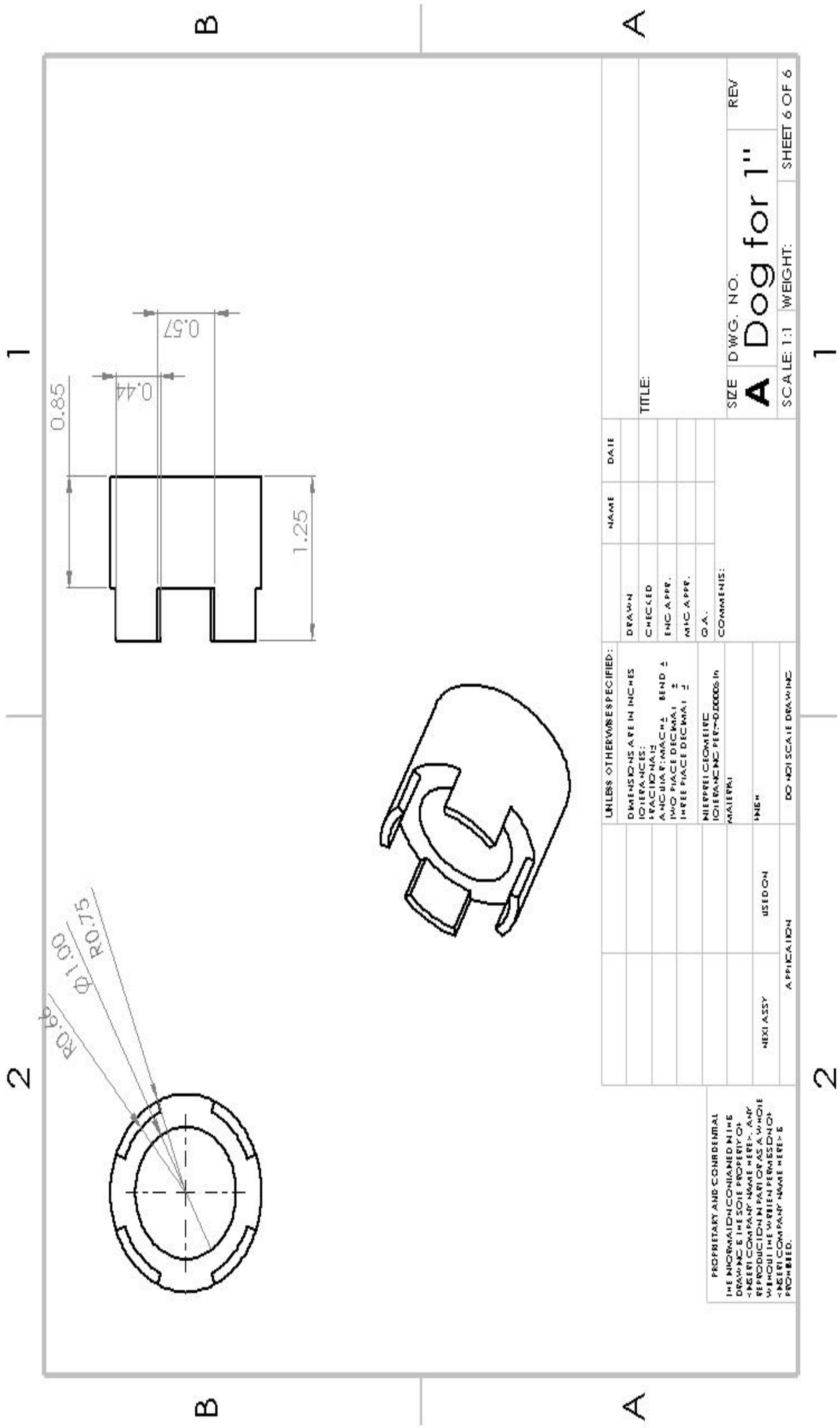
2. Custom Bearing



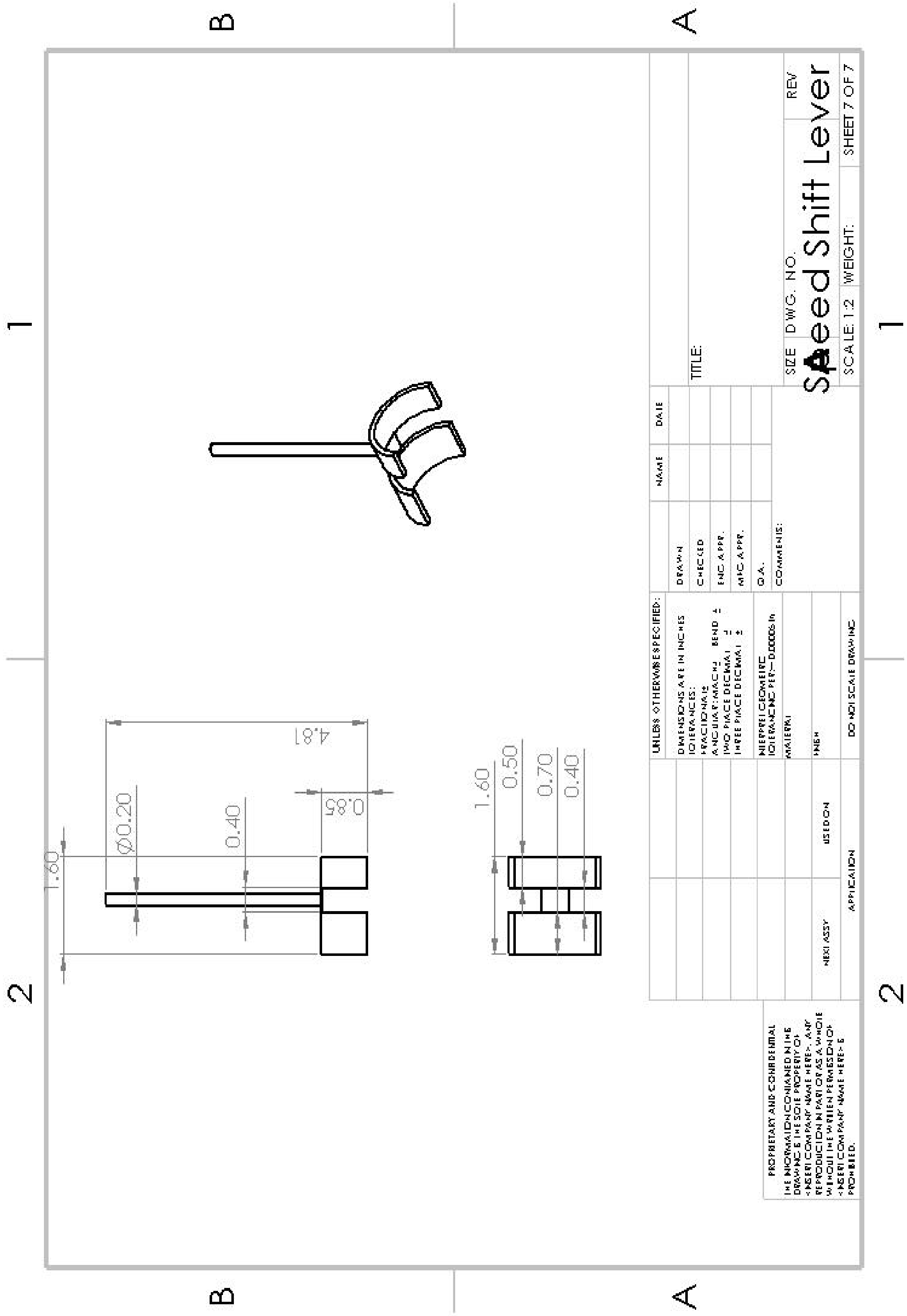
3. Dog Gear
   A. Dog Gear for 0.5 “ shaft



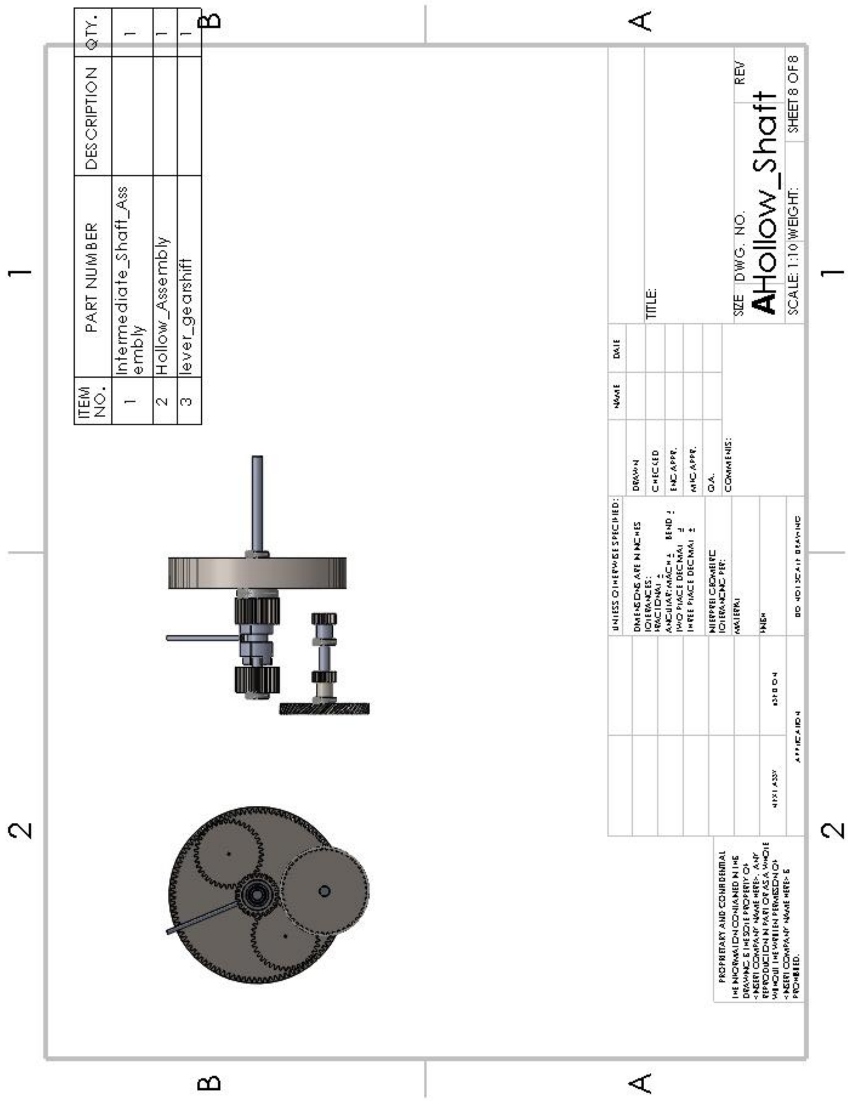
B. Dog Gear for 1” shaft



4. Speed Shift Lever



5. Assembly


ITEM NO.
PART NUMBER
DESCRIPTION
QTY.
1
Intermediate_Shaft_Assembly
1
2
Hollow_Assembly
1
3
lever_gearshift
1
TITLE:
SIZE
DWG. NO.
REV
A
Hollow_Shaft
SCALE: 1:10
WEIGHT:
SHEET 8 OF 8