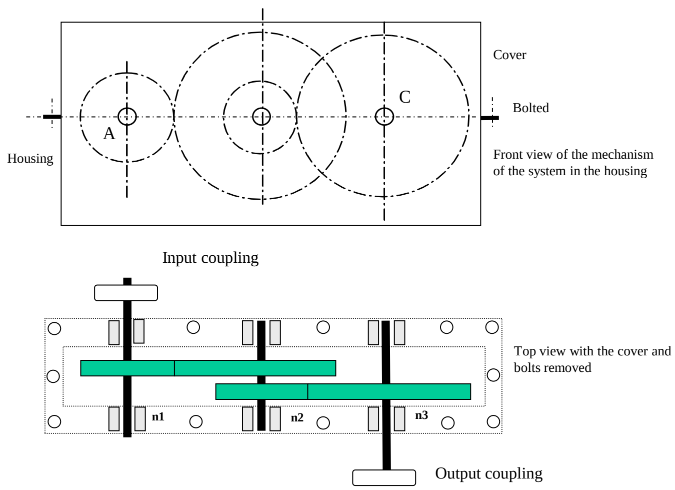
ME 315
THEORY OF MACHINES – DESIGN OF ELEMENTS
Spring 2011

Design Project

The figure on the next page show the mechanism plot of a gear system for transmitting motion and torque from position **A** to **C**. The distance between **A** and **C** should be about 0.3~0.35 m. The power to transmit is in terms of KW. The input is from shaft A by a coupling linked to an electric motor, and the speed of shaft A is 2700 rpm. The output speed is 380 rpm through the coupling on shaft C. The speed-reduction error should be no more than 2%. The overall efficiency is assumed to be 100%. The maximum fatigue stress concentration factors for different loading modes and notches are about 1.7. The life of the reducer is expected to be infinite (at least 100 million cycles for bearings). Each design group should optimize the design for the **highest power** to be transmitted.



Group members

Christina Lee	Shaft system 1
Yoke Peng Leong	Shaft system 2
Shawn Ang	Shaft system 3

Grade

On-time step dues	/5	
Structural design, drawings	/35	
Design analysis	/30	
Report writing	/30	Total____________

Project Description and Requirements

Our mechanical gear system is a two-step gear system with three shafts and four spur gears. The system serves as a speed reducer. The mechanism is designed to fulfil the following requirements:

- The distance between shaft 1 and shaft 3 must be between 0.3~0.35 m
- The input by a coupling is on shaft 1 with a speed of 2700 rpm
- The output speed is 380 rpm through the coupling on shaft 3
- The speed reduction error should be no more than 2%
- Overall efficiency is assumed to be 100%
- The maximum fatigue stress concentration factors for different loading modes and notches about 1.7

Our design is optimized for the highest transmission of power possible.

Task Distribution

All three group members were involved in the initial design of the gears since all the parameters of the four gears are dependent on each other. Teeth number selection and gear diameters were determined by all three members at the beginning of the project. Calculations for the dynamic basic load ratings for the bearings one each shaft were calculated by Shawn Ang and then the bearing selection was done by Christina Lee.

After initial parameters were set, each group member was in charge of the CAD models and drawings for one shaft. For the first set of drawings, Christina Lee was responsible for Shaft 1, Yoke Peng Leong for Shaft 2, and Shawn Ang for Shaft 3.

Since the gear and shaft dimensions and parameters were linked, Shawn Ang and Yoke Peng Leong were primarily in charge of making adjustments to the component design and Christina Lee was in charge of modifying the CAD documents.

Design Considerations & Material Selections

Design Considerations:

The gear system should be able to transmit motion and torque from the input gear A to output gear C.

Gears:

1) Since the system reduces speed, we select a large number for the gear teeth of gears A and C, and a smaller gear teeth number for gears B and D.
2) The gear teeth numbers should not be multiples of each other, or have a common factor among them.
3) The gears on shaft 2 should have adequate spacing between them
4) Stress analysis on the gears should be done to determine the extent of the stresses experience by the gears and their corresponding factors of safety.
5) Only spur gears are considered and hence only pure radial loads will be applied.

Shafts:

1) Shafts should have adequate length to accommodate bearings, gears, spacers and couplings.
2) Shafts should be designed with key ways, shoulders and varying diameters to account for fit with the various components.
3) The diameters of the shafts should be greater than the minimum diameter calculated given the torque and allowable shear stress.
4) For each shaft, proper shear force and bending moments should be calculated and plotted accordingly to account for the maximum torque and bending moment.

Bearings:

1) Proper stress analysis of the bearings should be done.
2) Selection of the bearings will be based upon the minimum bearing diameter
3) By considering pure radial loading and using ball bearings, we calculate C given P and L10.

Material selection:

Use Steel for gears, bearings and shafts
The gears, bearings and shafts will be under a lot of bending and contact stress. Hence, a strong and tough material should be selected.

Young's modulus, $E = 210$ GPa
Allowable Shear Stress, $\tau_{all} = 550$ MPa

Analysis

Gear Analysis

<u>Gear Geometry/Force:</u>

	Shaft A	Shaft B		Shaft C
	Gear 1	Gear 2	Gear 3	Gear 4
N, # of teeth	19	56	29	70
mt, module (mm)	4	4	4	4
n, speed (rpm)	2,700.00	916.07	916.07	379.52
d, pitch diameter (mm)	76	224	116	280
d, pitch diameter (m)	0.076	0.224	0.116	0.28
T, Torque (N*m)	106.10	312.73	312.73	754.85
W_t, Tangential Force (N)	2,792.19	2,792.19	5,391.82	5,391.82
W, total force(N)	2,971.39	2,971.39	5,737.85	5,737.85
W_a, axial force(N)	2,971.39	2,971.39	5,737.85	5,737.85
W_r, Radial Force (N)	1,016.27	1,016.27	1,962.46	1,962.46
dmin of shaft due to torque (m)	0.00994	0.01425	0.01425	0.01912
dmin of shaft due to torque (mm)	9.94	14.25	14.25	19.12

H, Input Power (W)	30000	
φ, Pressure Angle (°)	20	
Distance btw shaft A and C (mm)	348	(300 - 350)
Input/output speed ratio	7.114	
Ideal Input/output speed ratio	7.105	
Speed Error (%)	0.13	(<2%)

Gear Stress Analysis:

	Gears			
	Gear 1	Gear 2	Gear 3	Gear 4
N, # teeth	**19**	**56**	**29**	**70**
mt, module (mm)	**4**	4	**4**	4
n, speed (rpm)	2700	916.07	916.07	379.52
d, pitch diameter (mm)	76	224	116	280
d, pitch diameter (m)	0.076	0.224	0.116	0.280
m_n	4	4	4	4
a	4	4	4	4
b	5	5	5	5
bw	48	48	48	48

Ka, Application Factor	1	1	1	1
Km, Load Distribution Factor	1.2	1.2	1.2	1.2
Ki, Idler Factor	1	1	1	1
Kb, Rim thickness factor	1	1	1	1
Ks, Size Factor	1	1	1	1
vt	10.74	10.74	5.56	5.56
Kv, Dynamic Factor	1.61	1.61	1.44	1.44
Yj, Geometric bending factor	0.37	0.43	0.34	0.4
KE, Elastic factor	477351.635	477351.635	477351.635	477351.635
I, Geometric contact factor	0.75	0.75	0.71	0.71

Bending stress (MPa)	75.76	65.19	142.93	121.49
Contact stress (MPa)	667.77	667.77	731.33	731.33
Allowable bending stress (mpa)	248.20	288.75	248.20	288.75
No of cycles	100000000	33928571.4	33928571.4	14056122.4
Stress cycle factor	0.93	0.96	0.96	0.99
Allowable gear bending stress (mpa)	230.42	277.58	238.60	285.60
Stress cycle factor for contact	0.88	0.93	0.93	0.98
Hardness ratio factor	1.00	1.00	1.00	1.00
Allowable contact stress	866.00	839.50	866.00	839.50
Allowable gear contact stress	764.91	787.78	807.54	822.43
Bending factor of safety	**3.041**	**4.258**	**1.669**	**2.351**
Contact factor of safety	**1.145**	**1.180**	**1.104**	**1.125**

Shaft Analysis

Shaft 1

Dimensions		
a =	0.039	mm
b =	0.092	mm
a+b =	0.131	mm

Forces	Gear 1
Wt (N)	2,792.19
Wr (N)	1,016.27

Sum of moments about B=0	
F_Ay (N) =	713.72
F_Az (N) =	1,960.93

Sum of moments about A=0	
F_By (N) =	302.56
F_Bz (N) =	831.26

Resultant radial forces	
Fr_A (N) =	2,086.78
Fr_B (N) =	884.61

Max Y-plane moment =	27.84	Nm
Max Z_plane moment =	76.48	Nm
Maximum bending moment on shaft =	81.38	Nm
Shaft diameter =	27	mm
Bending stress =	4.21E+07	Pa
Maximum Torque =	106.10	Nm
Torsional shear stress =	2.75E+07	Pa

Using Von Mises Stress		
sigma' m =	4.76E+07	Pa
sigma' a =	7.16E+07	Pa

For infinite life, we need to choose Se

Se' = 0.5 * Sut

Se = k0kfkskrktkm Se'

k0 =	1	
kf =	0.27	
ks =	0.82	
kr =	0.75	
kt =	1	
km =	1	
Se =	8.98E+07	Pa

Kf =	1.7	
Kfs =	1.3	
Sut =	1.08E+09	Pa
Sy =	3.50E+08	Pa

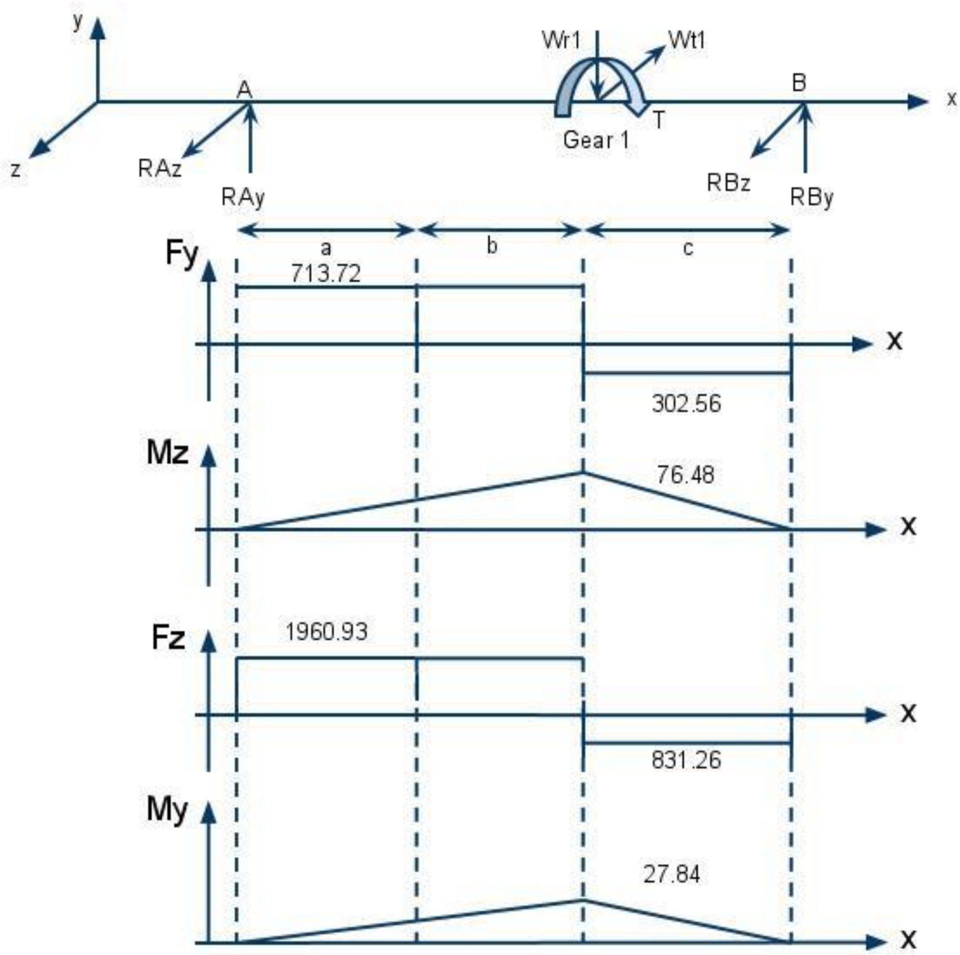
For Goodman Line,

sigma'/Se + sigma'/Sut = 1/ns

safety factor, ns = **1.188**

For Yield Line,

peak von mises stress, sigma' =	6.35E+07	Pa
safety factor, ns =	**5.510**	



Shaft 2

Dimensions	**(mm)**
a=	39
b=	53
c=	39
a+b=	92
a+b+c=	131

Forces	Gear 2	Gear 3
Wt (N)	2,792.19	5,391.82
Wr (N)	1,016.27	1,962.46

Sum of moments about A=0		
F_By (N) =	1,075.66	N
F_Bz (N) =	4,617.88	N
Sum of forces in y = 0		
F_Ay (N) =	-129.48	N
F_Az (N) =	3,566.13	N
Resultant radial forces		
Fr_A (N) =	6,530.67	N
Fr_B (N) =	3,568.48	N

Bending Moment	xy plane	xz plane	Net moment	
Gear 2	39.63	108.90	115.88	Nm
Gear 3	76.54	210.28	223.78	Nm

Shaft diameter =	38	mm
Maximum bending moment on shaft =	223.78	Nm
Maximum torque =	312.73	Nm
Bending stress =	4.15E+07	Pa
Torsional shear stress =	2.90E+07	Pa
Using Von Mises Stress		
sigma' m =	5.03E+07	Pa
sigma' a =	7.06E+07	Pa

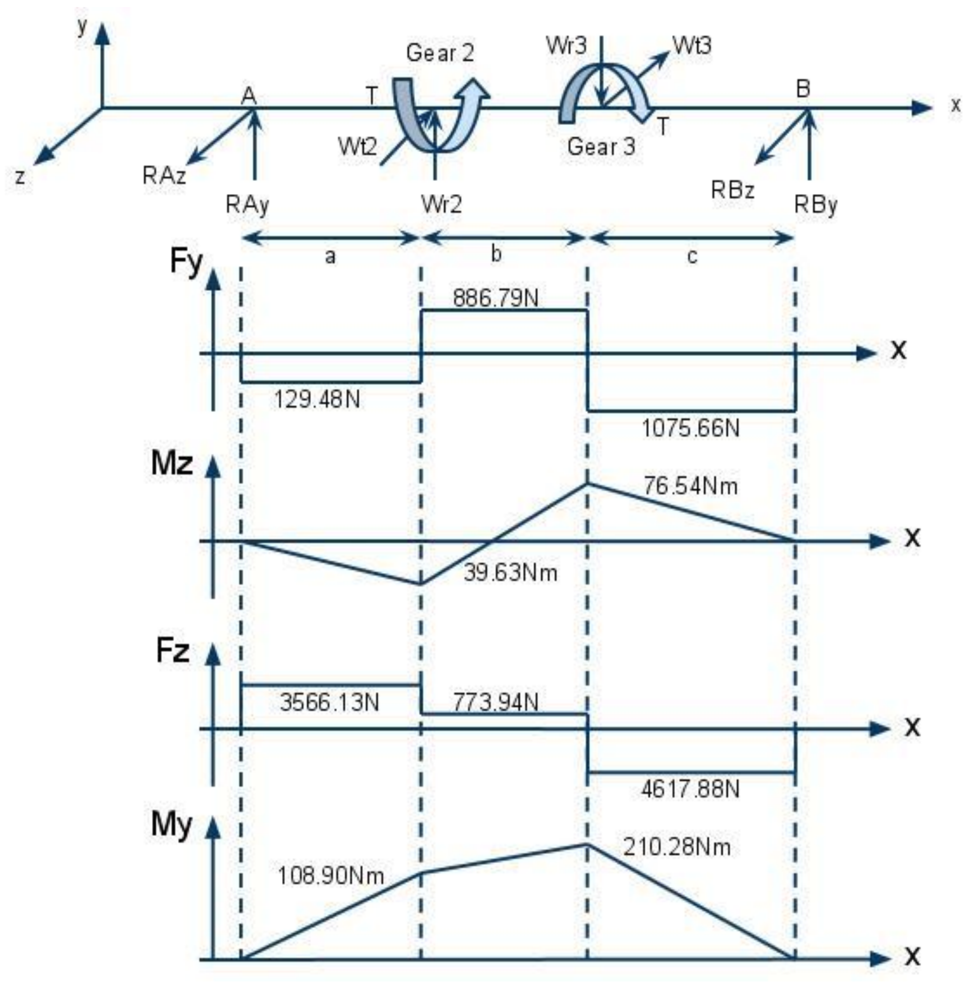
Kf =	1.7	
Kfs =	1.3	
Sut =	1.08E+09	Pa
Sy =	3.50E+08	Pa

For infinite life, we need to choose Se
Se' = 0.5 * Sut
Se = k0kfkskrktkm Se'

k0 =	1.00	
kf =	0.27	
ks =	0.79	
kr =	0.75	
kt =	1.00	
km =	1.00	
Se =	8.64E+07	Pa

For Goodman Line,
sigma' a/Se + sigma' m/Sut = 1/ns
safety factor, ns = **1.158**

For Yield Line,
peak von mises stress, sigma' = 6.52E+07 Pa
safety factor, ns = **5.367**



Shaft 3

Dimensions

a=	0.092
b=	0.029
a+b=	0.121

Forces	Gear 4
Wt (N)	5,391.82
Wr (N)	1,962.46

Sum of moments about B = 0

F_Ay (N) =	470.34
F_Az (N) =	1,292.25

Sum of moments about A = 0

F_By (N) =	1,492.12
F_Bz (N) =	4,099.56

Resultant radial forces

Fr_A (N) =	1,375.19
Fr_B (N) =	4,362.67

Max Y-plane moment =	43.27	Nm
Max Z_plane moment =	118.89	Nm
Maximum bending moment on shaft =	126.52	Nm
Shaft diameter =	33	mm
Bending stress =	3.59E+07	Pa
Maximum Torque =	754.85	Nm
Torsional shear stress =	1.07E+08	Pa

Using Von Mises Stress

sigma' m =	1.85E+08	Pa
sigma' a =	6.10E+07	Pa

For infinite life, we need to choose Se
Se' = 0.5 * Sut
Se = k0kfkskrktkm Se'

k0 =	1	
kf =	0.27	
ks =	0.80	
kr =	0.75	
kt =	1	
km =	1	
Se =	8.78E+07	Pa

Kf =	1.7	
Kfs =	1.3	
Sut =	1.08E+09	Pa
Sy =	3.50E+08	Pa

For Goodman Line,

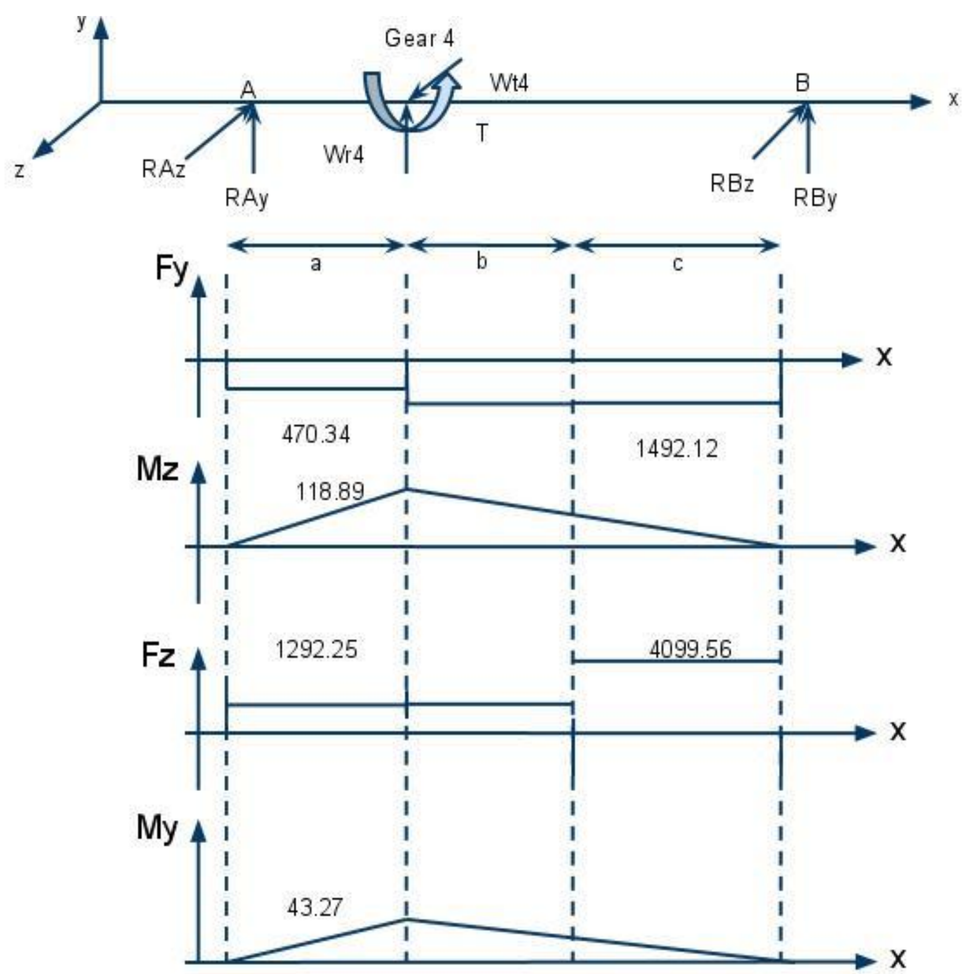
sigma'/Se + sigma'/Sut = 1/ns

safety factor, ns = **1.155**

For Yield Line,

peak von mesis stress, sigma' = 1.89E+08 Pa

safety factor, ns = **1.855**



Bearing Selections

Bearing for shaft 1

Bearing diameter =	25	mm
Torsional shear stress =	3.46E+07	Pa

Using Von Mises Stress

Bearing sigma' m =	5.99E+07	Pa
Bearing k0 =	0.27	
Bearing ks =	0.83	
Bearing kr =	0.75	
Bearing kt =	1	
Bearing km =	1	
Bearing Se =	9.05E+07	Pa

For Goodman Line,

Bearing safety factor, ns =	**18.030**

For Yield Line,

Bearing peak von mises stress =	5.99E+07	Pa
Bearing safety factor, ns =	**5.843**	

Bearing Selection

P = Pr	P = 1738.98	Pure Radial loading choose Fr_A
L10 = 100 mil cycles		
C = PL^(1/m) =	8,071.63	
Bearing dmin =	14 mm	

Select SKF 6305

d=25 mm
OD = 62 mm
bw = 17 mm
C= 22500 N
C0 = 11600 N

Bearing for shaft 2

Bearing diameter	30	mm
Torsional shear stress	5.90E+07	Pa

Using Von Mises Stress

Bearing sigma' m =	1.02E+08	Pa
Bearing k0 =	0.27	
Bearing ks =	0.81	
Bearing kr =	0.75	
Bearing kt =	1	
Bearing km =	1	
Bearing Se =	8.88E+07	Pa

For Goodman Line,

Bearing safety factor, ns = **10.570**

For Yield Line,

Bearing peak von mises stress =	1.02E+08	Pa
Bearing safety factor, ns =	**3.426**	

Bearing Selection

P = Pr	P = 5442.23	Pure Radial loading choose Fr_A
L10 = 100 mil cycles		
C = PL^(1/m) =	25,260.59	
dmin =	20 mm	

Select SKF 6306

d=30 mm
OD = 72 mm
bw = 19 mm
C = 28100 N
C0 = 16000 N

Bearing for shaft 3

Bearing diameter =	30	mm
Torsional shear stress =	1.42E+08	Pa

Using Von Mises Stress

Bearing sigma' m =	2.47E+08	Pa
Bearing k0 =	0.27	
Bearing ks =	0.81	
Bearing kr =	0.75	
Bearing kt =	1	
Bearing km =	1	
Bearing Se =	8.87E+07	Pa

For Goodman Line,

Bearing safety factor, ns =	**4.379**	

For Yield Line,

Bearing peak von mises stress =	2.47E+08	Pa
Bearing safety factor, ns =	**1.419**	

Bearing Selection		
P = Pr	P = 3635.55	Pure Radial loading choose Fr_B
L10 = 100 mil cycles		
C = PL^(1/m) =	16,874.73	
dmin =	26 mm	

Select SKF 6406

d=30 mm
OD = 90 mm
bw =23 mm
C = 43600 N
C0 = 23600 N

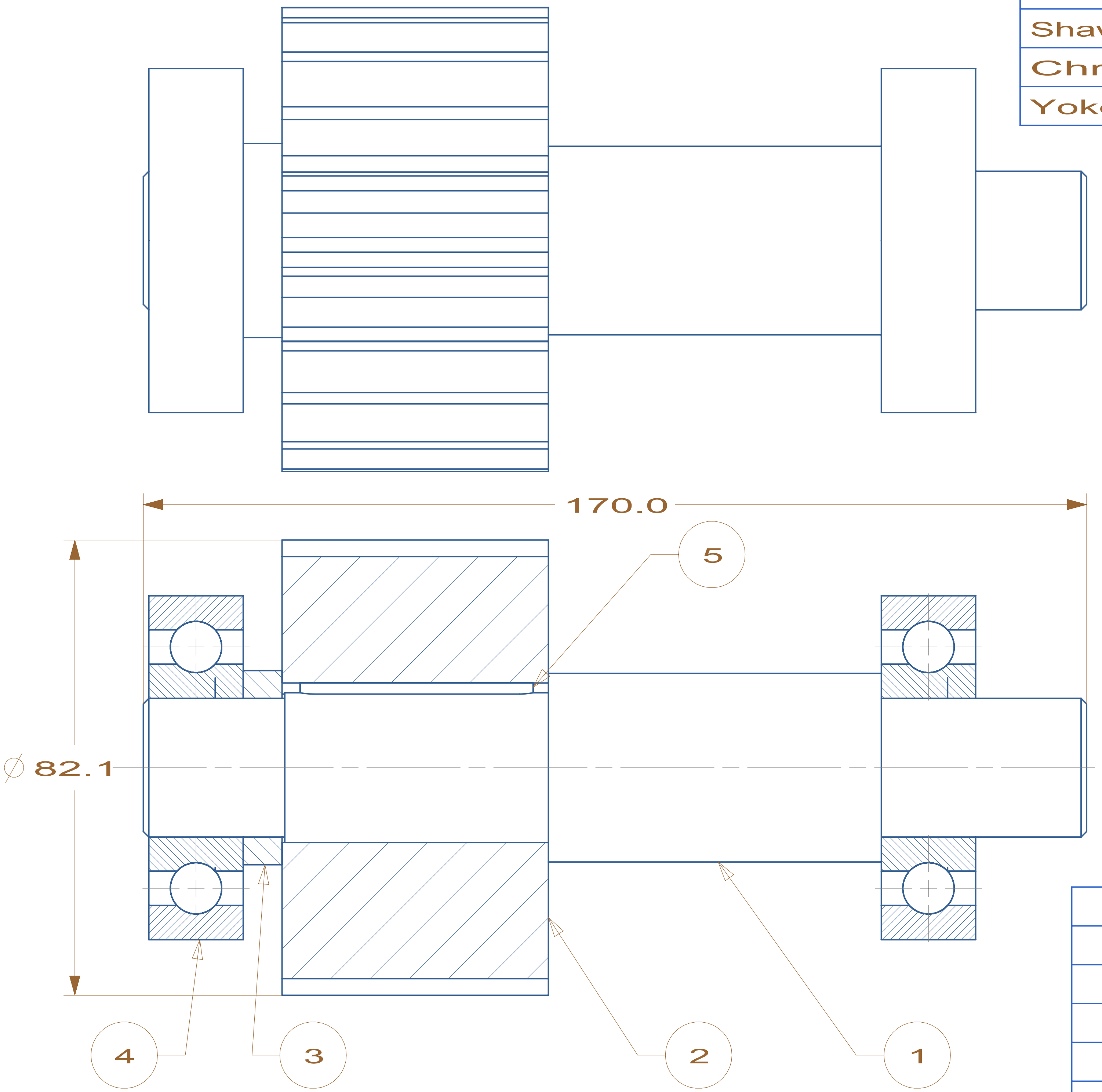
Design Results & Drawings

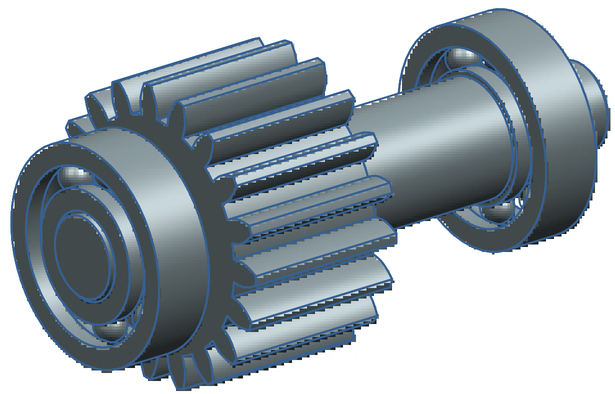
Design results and drawings are following:

Shaft 1 Assembly

1. Shaft 1 Assembly
2. Shaft 1
3. Gear 1

ME 315	Shaft 1
Shawn Ang	Assembly
Christina Lee	
Yoke Peng Leong	





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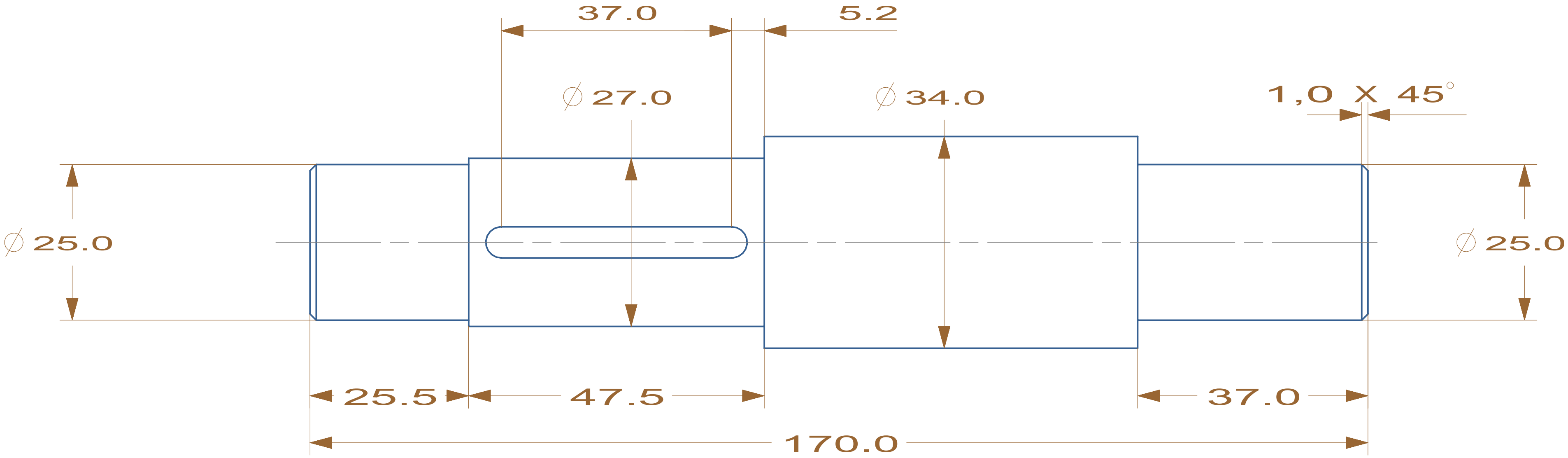
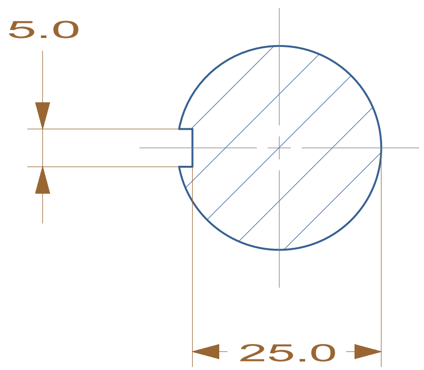
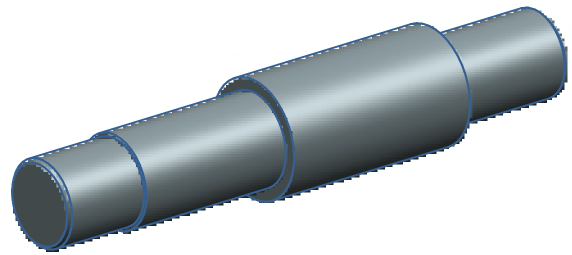
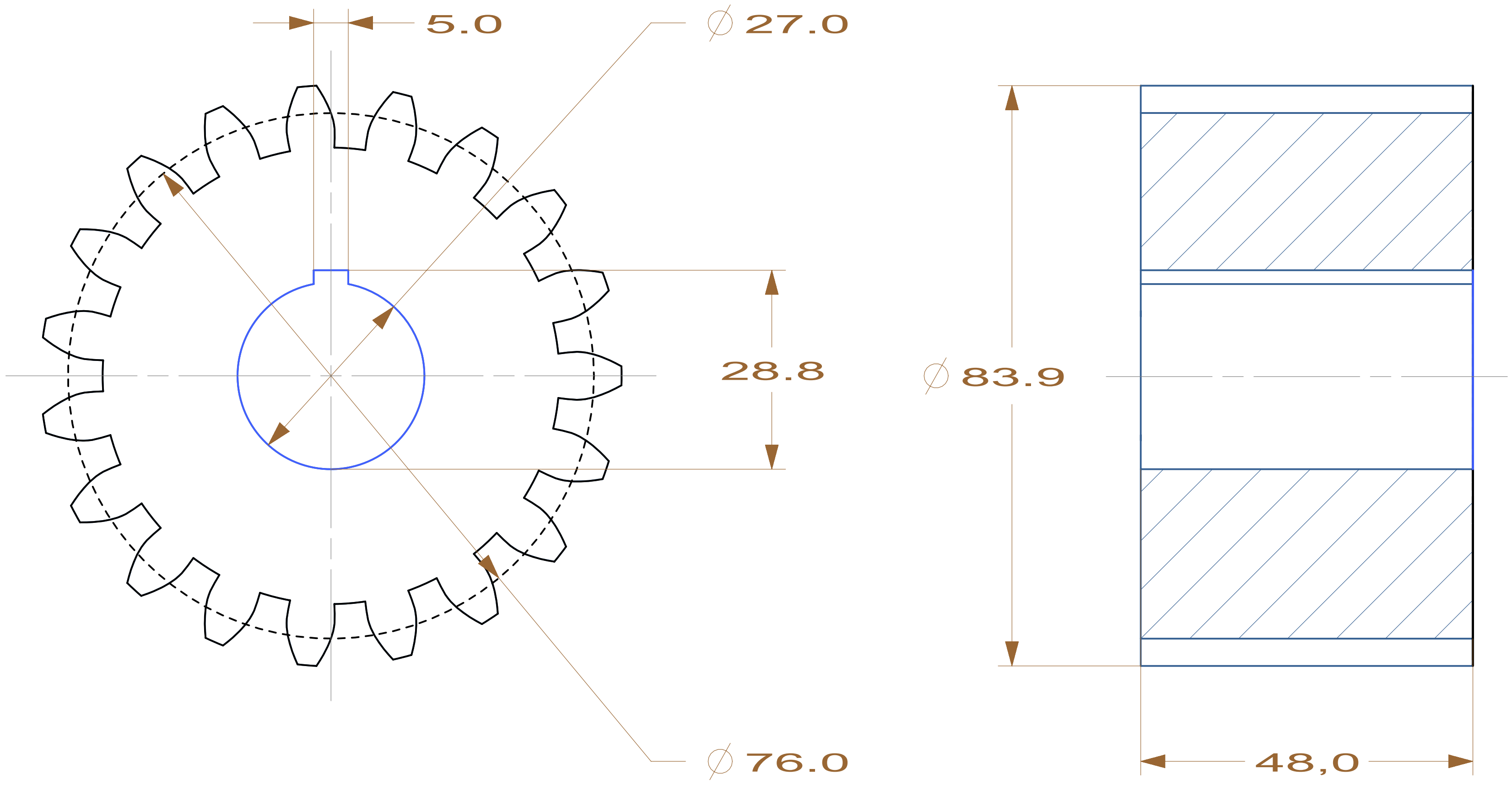
5	KEY	1
4	BEARING1	2
3	SPACER1	1
2	GEAR 1	1
1	SHAFT 1	1







Shaft 1	ME315
	Shawn Ang
	Christina Lee
	Yoke Peng Leong



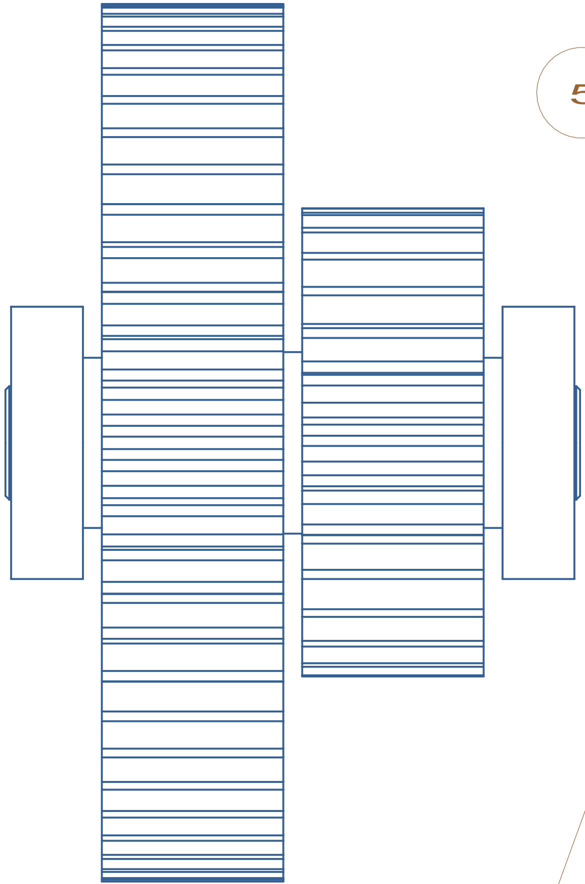
Module, m	4 mm
No. of teeth, N	19
Pressure angle	20 degrees
Addendum, a	4 mm
Dedendum, b	5 mm
Clearance, c	1 mm
Material	Grade 2 Steel

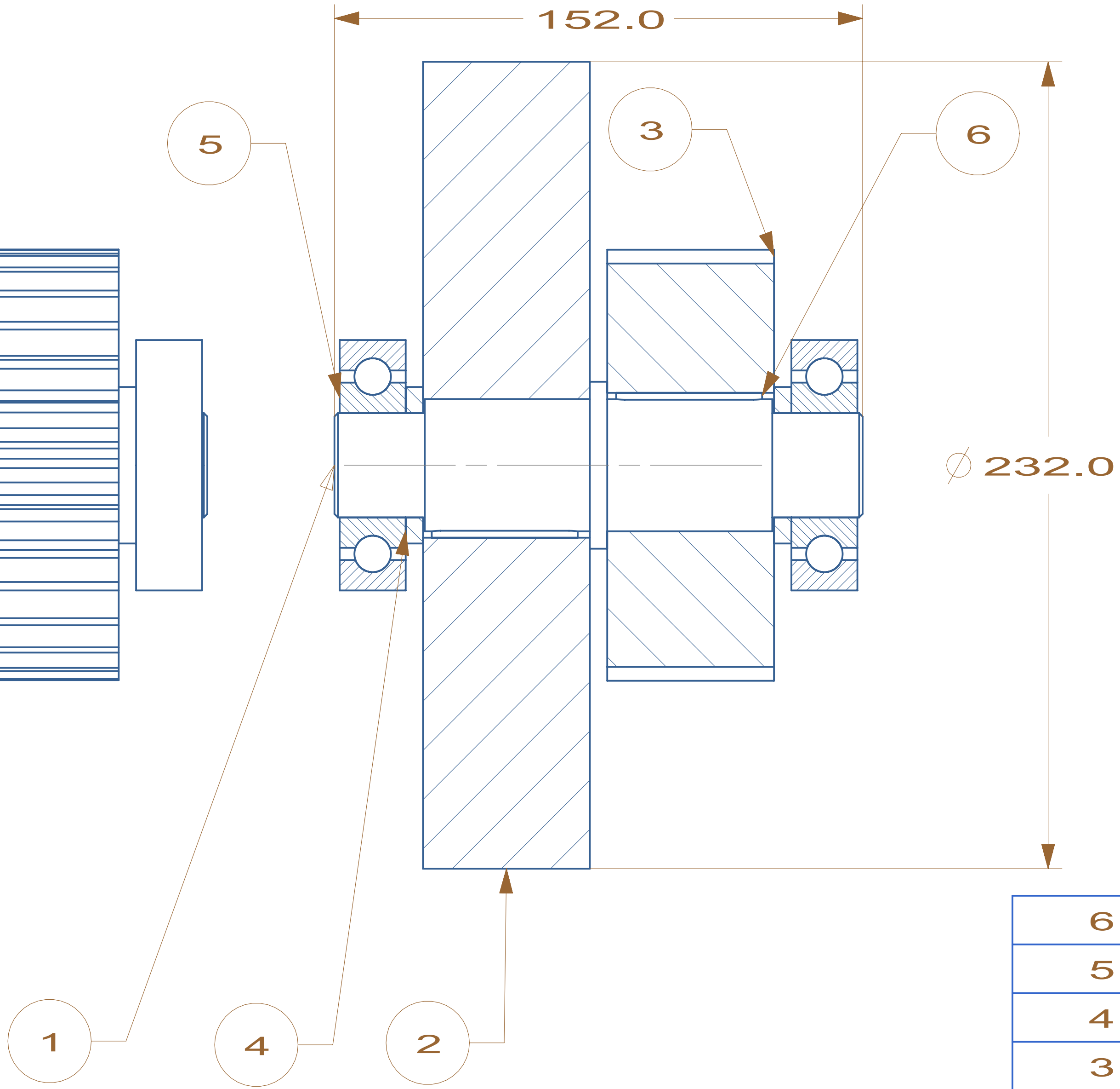
Gear 1	ME315
Input gear of gear set	Shawn Ang
	Christina Lee
	Yoke Peng Leong

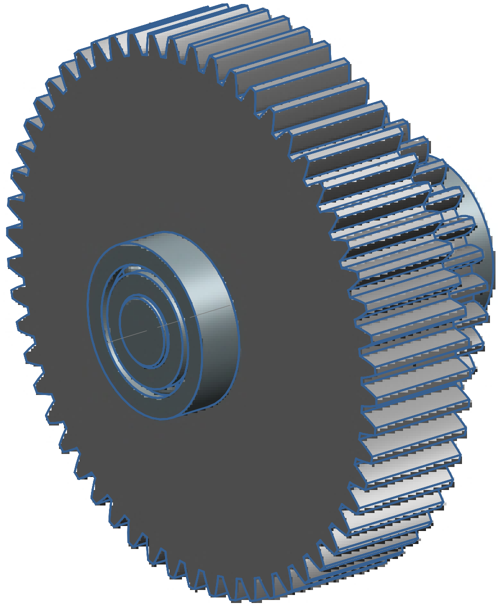
Shaft 2 Assembly

1. Shaft 2 Assembly
2. Shaft 2
3. Gear 2

ME 315	Shaft 2
Shawn Ang	Assembly
Christina Lee	
Yoke Peng Leong	







PC NO	PART NAME	QTY
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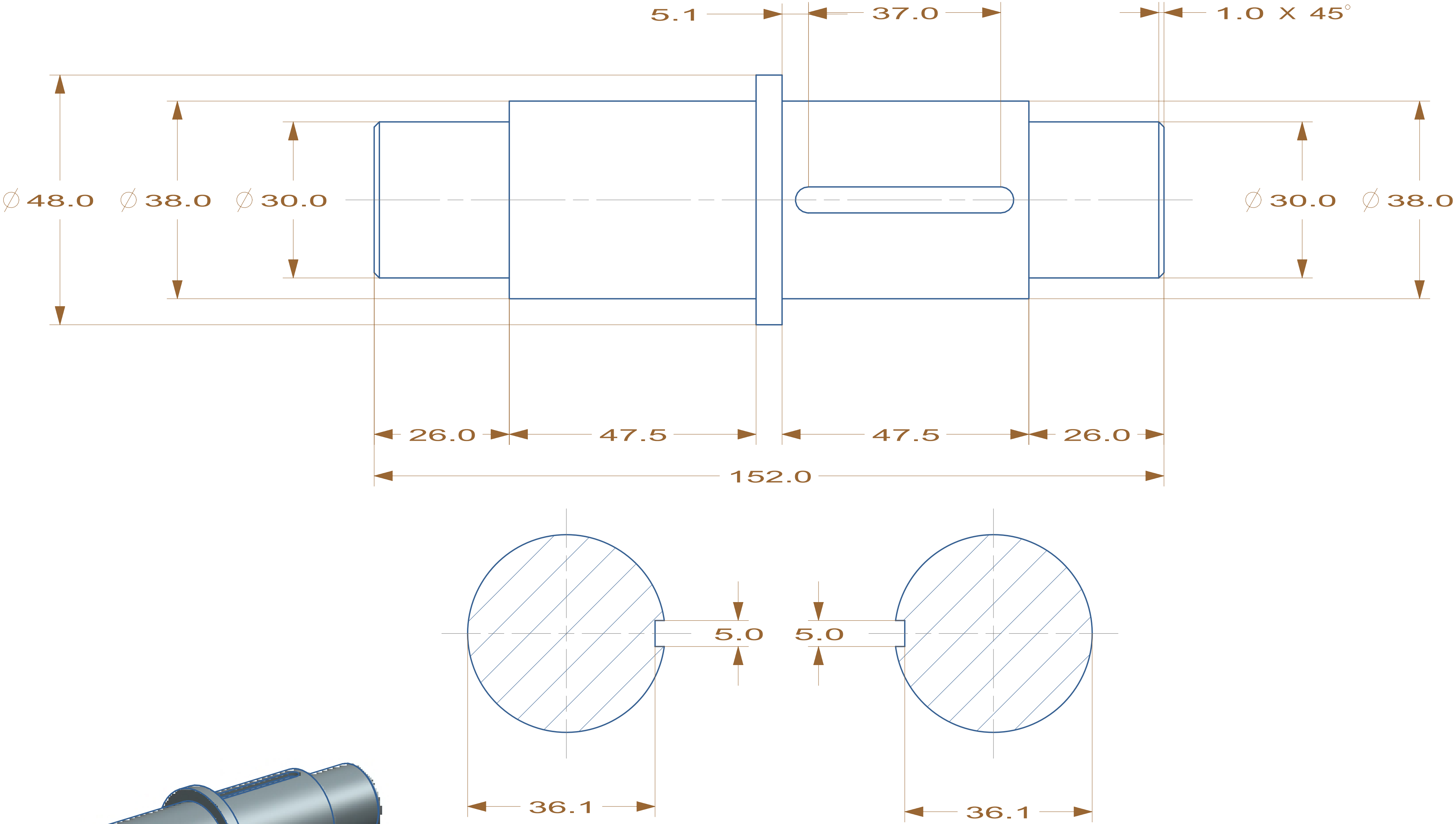
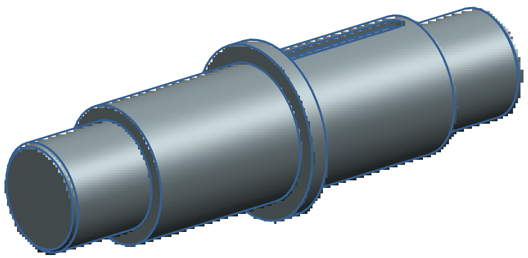
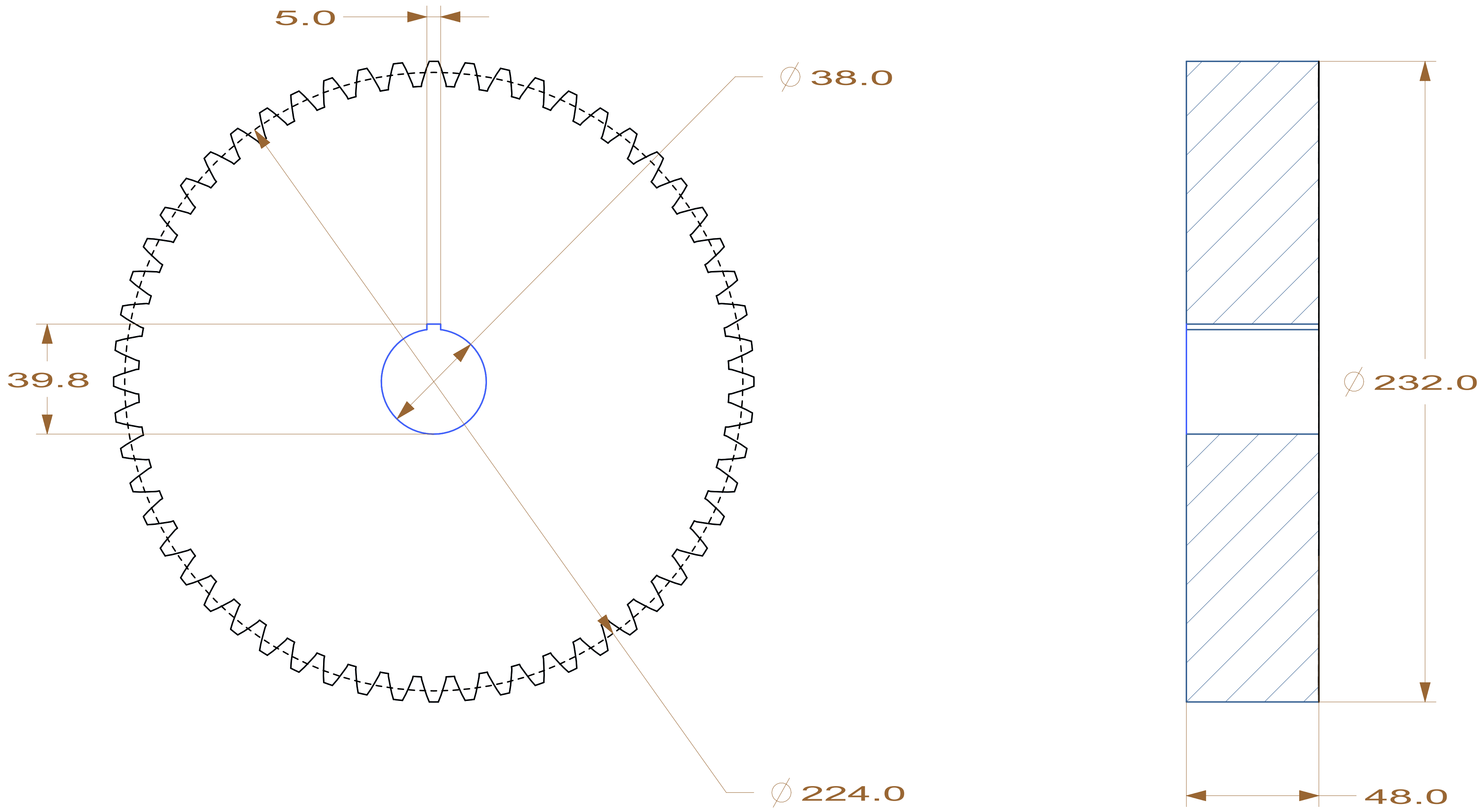
5	BEARING 2	2
4	SPACER 2	2
3	GEAR 3	1
2	GEAR 2	1
1	SHAFT 2	1





Shaft 2	ME315
	Shawn Ang
	Christina Lee
	Yoke Peng Leong



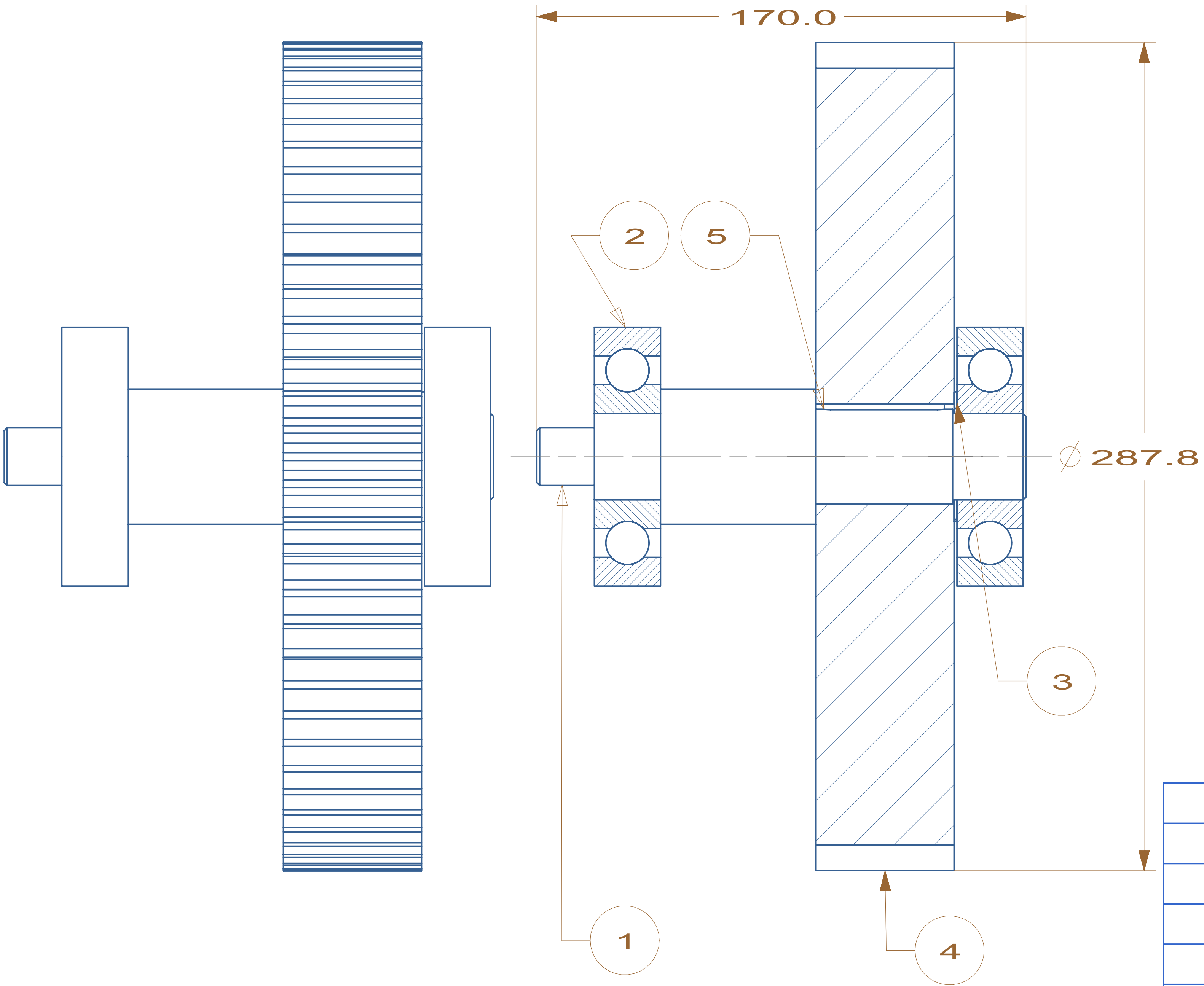
Module, m	4 mm
No. of teeth, N	56
Pressure Angle	20 degrees
Andendum, a	4 mm
Dedendum, b	5 mm
Clearance, c	1 mm
Material	Grade 2 Steel

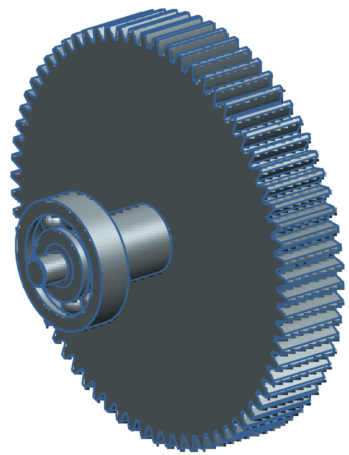
Gear 2	ME315
Output gear of gear set	Shawn Ang
	Christina Lee
	Yoke Peng Leong

Shaft 3 Assembly

1. Shaft 3 Assembly
2. Shaft 3
3. Gear 4

ME 315	Shaft 3
Shawn Ang	Assembly
Christina Lee	
Yoke Peng Leong	





PC NO	PART NAME	QTY
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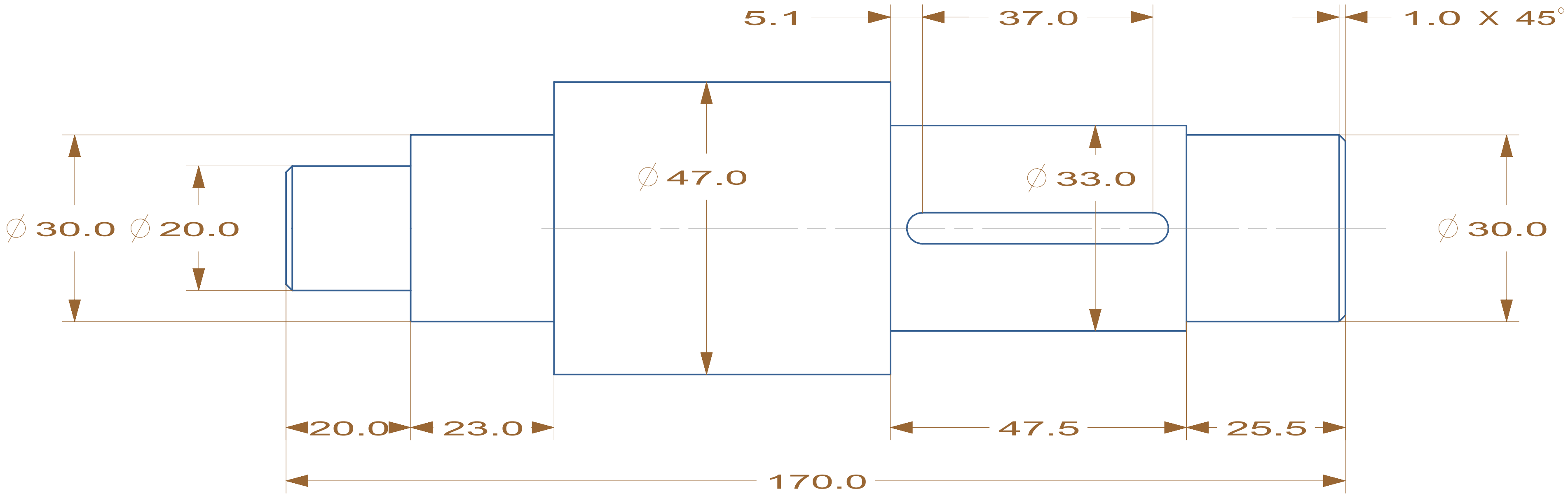
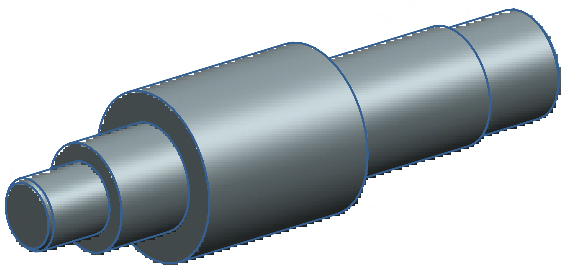
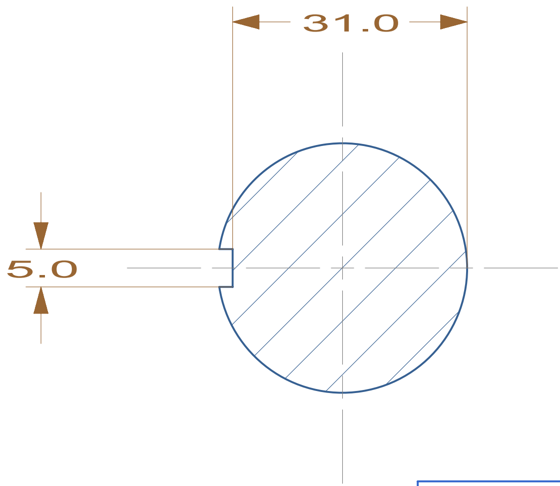
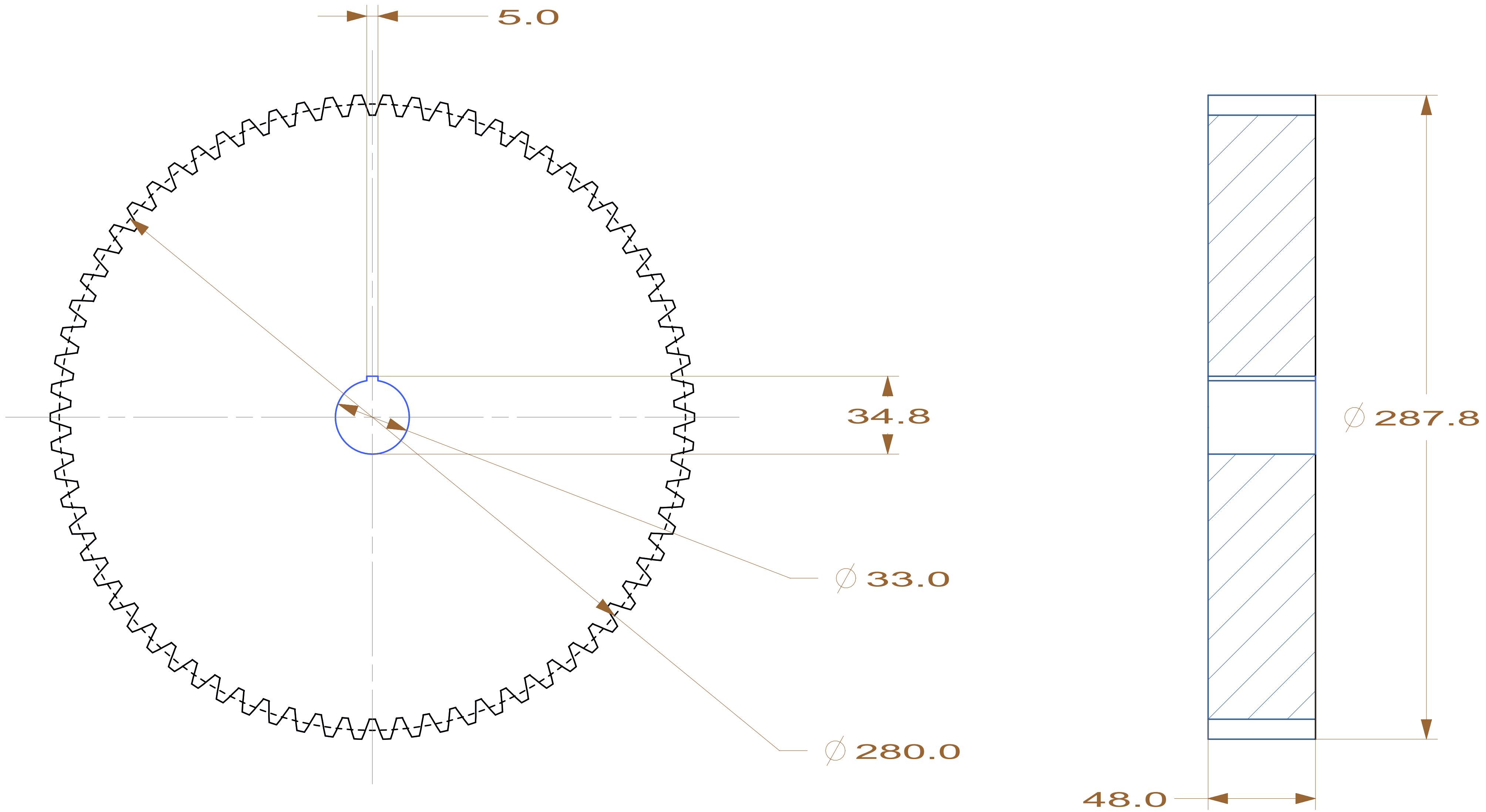
4	GEAR 4	1
3	SPACER 3	1
2	BEARING 3	2
1	SHAFT 3	1







Shaft 3	ME315
	Shawn Ang
	Christina Lee
	Yoke Peng Leong



Module	4 mm
No. of Teeth	70
Pressure Angle	20 degrees
Adendum, a	4 mm
Dedendum, b	5 mm
Clearance, c	1 mm
Material	Grade 2 steel

Gear 4	ME 315
Output gear of gear set	Shawn Ang
	Christina Lee
	Yoke Peng Leong

Conclusions

Our mechanical gear system is a two-step gear system with three shafts and four spur gears. The system serves as a speed reducer. The mechanism designed has fulfilled the following requirements:

- The distance between shaft 1 and shaft 3 must be between 0.3~0.35 m
    - Our design: 3.48 m
- The input by a coupling is on shaft 1 with a speed of 2700 rpm
- The output speed is 380 rpm through the coupling on shaft 3.
- The speed reduction error should be no more than 2%.
    - Our design: 0.13%

Our design is optimized for the highest transmission of power possible at 30kW with a minimum factor of safety of 1.1.