The University of Hong Kong

Department of Mechanical Engineering

Power Transmission Unit Design

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1. Introduction

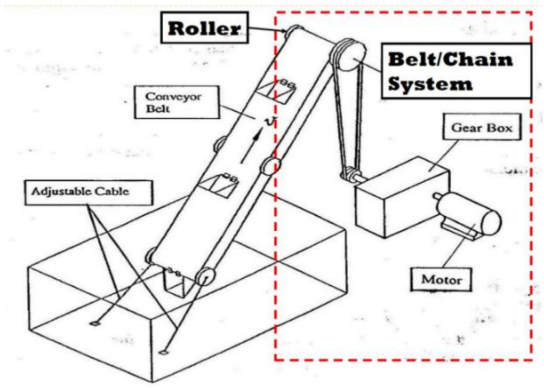
Engineering principles permeate every facet of our lives, making it vital for engineering students to go beyond theoretical learning and engage with practical experiments and real-world applications. This project's objective is to design a power transmission system for either an irrigation system or a mining field, incorporating the following components:

- A motor
- A gearbox
- A chain drive system

In a mining field power transmission system, the motor serves as the main power source, transforming electrical energy into mechanical energy to operate various equipment and machinery. Motors in mining are typically employed to drive conveyor systems, crushers, pumps, fans, and other heavy machinery. These motors must be sturdy and resilient to endure challenging conditions like dust, humidity, and extreme temperatures common in mining environments.

A gearbox is a critical component in the system, enabling the adjustment of the motor’s speed and torque to meet the specific needs of different equipment. The gears within the gearbox transfer power from the motor to driven machinery, such as conveyors, crushers, or hoists. Designed for heavy-duty performance, gearboxes in mining are built to manage substantial loads and ensure consistent power delivery in tough conditions. They often include features like lubrication and cooling systems to enhance efficiency and durability in the harsh mining environment.

In mining operations, chain drive systems are frequently employed to transfer power from the motor to driven equipment across extended distances. These systems link the motor shaft to the shaft of equipment like conveyor belts, crushers, or screens. Chain drives are favored for applications needing smooth and consistent power delivery, whereas chain drives are better suited for heavy-duty tasks requiring high torque. To ensure efficient power transmission and avoid early wear or failure, proper tensioning, alignment, and regular maintenance of these systems are essential.



1.1. <u>Design Requirements</u>

- Speed of conveyor belt v = 1.2m/s
- Diameter of roller Droller = 200mm
- Force to drive conveyor belt F = 6000N
- The rotational speed of the motor is 1440rpm
- Operation life: 8 hours/day for 10 years
- Space: as small as possible
- Cost: as small as possible

2. Design Concerns and Methodology

2.1 Design Concerns

2.1.1 Durability and Lifetime extension

Although the system is designed for 8 hours of daily use over 10 years, we aim to extend its lifespan beyond this period. Certain components are inherently weaker or subject to greater forces, and targeted lifetime extension measures can maintain their structural integrity. System failures are undesirable, dangerous, and can generate non-biodegradable metal waste, contributing to environmental pollution, especially since sustainable metal disposal methods are lacking in many regions. Durable, longer-lasting components reduce waste and support sustainability. Additionally, companies that prioritize durability and lifetime extension in their designs often earn a strong reputation, fostering customer loyalty and positive word-of-mouth. In contrast, products that fail prematurely can harm a company's reputation and erode consumer trust.

2.1.2 Potential market size

To reduce costs, the potential market size for the product must be considered, as component costs, like those for gearbox housing, depend partly on production scale. Thus, analyzing the market size is essential to guide the design of specific components. A detailed market analysis will be included later in the report.

2.1.3 Use

As previously mentioned, the project focuses on designing a power transmission system for either an irrigation system or a mining field. Designing a system to meet both environments' conditions would likely result in reduced efficiency, lifespan, and performance. Thus, it is critical to select the specific application for the system, or in other words, to clearly "identify the target customer."

2.1.4 Efficiency

We recognize potential energy losses in the motor, gearbox, and drive systems. Efficiency, defined as the ratio of input energy to useful energy, is critical since customers pay for all input energy, but the amount converted to useful energy depends on our design. Our goal is to maximize practical efficiency to enhance customer experience. Lower efficiency increases operating costs through higher energy consumption and poses safety risks, as wasted energy often dissipates as heat. Given that most components are metal, heat conducts rapidly throughout the system, creating a significant hazard if it contacts human skin. Additionally, the system must meet the requirements outlined in section 1.1. Higher efficiency ensures these requirements are sustained over a longer period.

2.1.5 Size and cost

The requirements mandate minimizing both size and cost, which are interrelated. However, components must maintain a minimum size or use sufficiently high-quality materials to withstand operational forces. Balancing these requirements is essential when determining size and cost. Reducing cost enhances market competitiveness, as customers typically prefer the lower-priced option when performance and functionality are comparable. Some companies with strict budget limits may automatically exclude products exceeding a certain cost threshold. Increased competitiveness can boost market share and sales volume. Minimizing size can streamline manufacturing, lower production costs, and improve efficiency, ultimately increasing profit margins.

2.2 Methodology

Our system was designed in 5 stages:

Stage 1: Identify our customer

We recognize that the project permits selecting between two design scenarios: irrigation or mining field. While both share similar basic operational requirements, significant design modifications are necessary to address the distinct environmental challenges each presents. We have chosen the mining field scenario, identifying our customer as a mining company.

Stage 2: Acknowledge requirements

The instructors have clearly outlined the design requirements, but these lack a broader context to interpret their significance. For instance, we lack the expertise to determine whether a motor rotational speed of 1440 rpm is considered high. Thisstage is specifically dedicated to researching other systems to gain such insights.

Stage 3: Motor selection

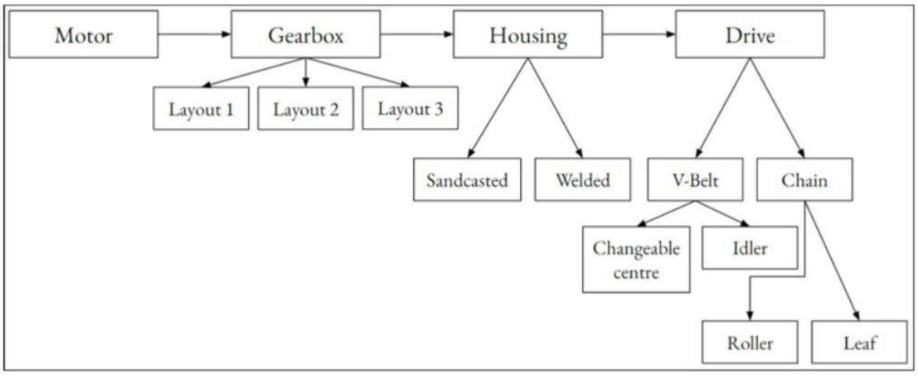
Since the motor will not be designed by us, we will choose from on-the-shelf motors on the market, while taking into account the size and cost limitations in the requirement.

Stage 4: Calculations

The motor is the only source of force in the system, and all data and calculations must stem from the selected motor. Calculations are needed to determine stresses, strains, torques, speeds, and gear properties.

Stage 5: Decision-making

We will carry out a two-tier system analysis to gain a clear understanding of our feasible options. Then, employing a decision matrix with tailored weightings, we will choose the most suitable option for our project. This method will be used for the gearbox, gearbox housing, and drive system.



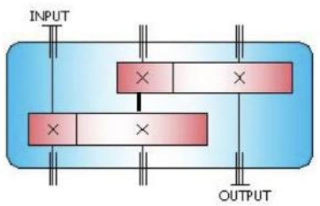
3. Layout of the System

3.1 Introduction

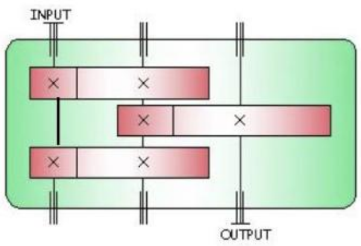
As mentioned in the design scenario and requirements, we are required to design a gearbox which include gears, shafts, key and keyseats, couplings, retaining rings and bearings for the system. In this section, we would be illustrating our thinking process of the choice of system layout.

3.2 Gearbox

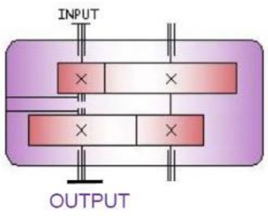
There are three types of reducer layouts given from the design scenarios. A gearbox is for adjusting the rotational speed of the system to the desired speed as there is a difference of rotational speed between the input and output. Here shows the 3 gearbox layouts provided:



REDUCER LAYOUT 1



REDUCER LAYOUT 2



REDUCER LAYOUT 3

Their respective advantages are as follows.

Layout 1	Layout 2	Layout 3
Simple Structure	Heavy loads	Compact size
Assemble with ease	Smooth transmission	Short shaft
Manufacture with ease	Strong endurance	Concentric I/O
Low manufacturing cost	Balanced shaft forces	Small bending moment

Their respective disadvantages are as follows.

Layout 1	Layout 2	Layout 3
Unbalanced shaft moments	Bulky housing design	Small L/w ratio
Large L/w ratio	Strong output shaft	Complex housing design
	High manufacturing cost	Shaft alignment
	Greatest in size	Highest manufacturing cost

Next, we introduced a weighting for each criterion, with score 3 representing the most favourable element and 1 representing the least favourable element. Combining the advantages and disadvantages from the tables above, we concluded a decision matrix:

Criteria	Weighting	Layout 1	Layout 2	Layout 3
Size	1	2	1	3
Structural simplicity	3	3	2	1

Cost of manufacture	3	3	2	1
Ease of assembly & maintenance	3	3	2	1
Housing design (simplicity & size)	2	3	2	1
Durability (balance of moments)	1	1	3	2
Total		**36**	26	16

We have decided that structural simplicity to be the most important criterion to consider, as it directly contributes to costs, including maintenance and manufacture. Size and durability are often inversely related to cost; therefore, they are given the lowest importance to be considered. As a result, Layout 1 scored the highest and is chosen to be the preferred layout. Otherwise, Layout 3 would be the second choice.

4. Chain Drive System

4.1 Introduction

We use chain drive to drive the power transmission system. A chain drive consists of chain and driving/driven sprockets. Parameters of the chain and sprocket like pitch, center distance and size, and type of sprocket and will be decided by using tables and calculations.

Constrains: In common engineering practice, maximum speed ratio for chain drive is 7. In the double gear reduction setup, first gear pair reduction speed ratio is 1.4 times the second gear pair reduction speed ratio. To limit contact stress of the double reduction gear setup, the speed ratio for each gear pair is limited from 3 to 5

4.2 Calculations

The power rating of the chain would be $6000 \times 1.2 \times 1.1 = 7920W$

Service factor

The chain system has a soft start with a medium duty; therefore, the service factor is 1.1.

Efficiency

The efficiency of the system would be

$$\eta = \eta_1(\eta_2)^2(\eta_3)^3(\eta_4)(\eta_5) = 0.99 \times 0.97 \times 0.98 \times 0.89 \times 0.954 = 0.799$$

The input power where the motor at would be $\frac{7920}{0.799} = 9912.4W$

Power at the third shaft would be $9912.4W$

Speed ratio of system

Speed Ratio of System: Ratio between roller's rotational speed and motor's rotational speed.

$$n_{roller} = \frac{60v}{\pi D_{roller}} = \frac{60 \times 1.2}{\pi \times 0.2} = 114.59 \text{ rpm}$$

$$i_{system} = \frac{n_{motor}}{n_{roller}} = \frac{1440}{114.59} = 12.57$$

Speed ratio of chain

The number of teeth for the first pair of gears is 20:60, a ratio of 3. The number of teeth for the second pair of gears is 24:45, a ratio of 1.875. Since the gear in the second set rotates in 218 rev/min, the driving speed is more than 100 rpm. Therefore, the minimum number of teeth has to be 17 teeth. Therefore, the teeth number for driving/driven sprocket can be chosen as 17/38, where the speed ratio would be = 38/17=2.24.

From Table, Teeth number of 17/38 for driving/driven sprocket is chosen

Chain-Table 4: Speed ratios & number of sprocket teeth

Speed Ratio	Number of Teeth		Speed Ratio	Number of Teeth		Speed Ratio	Number of Teeth	
	Driving Sprocket	Driven Sprocket		Driving Sprocket	Driven Sprocket		Driving Sprocket	Driven Sprocket
1,00	13	13	1,35	17	23	3,00	19	57
1,00	15	15	1,40	15	21	3,04	25	76
1,00	17	17	1,46	13	19	3,30	23	76
1.00	19	19	1,47	17	25	3,35	17	57
1,00	21	21	1,52	25	38	3,62	21	76
1.00	23	23	1,53	15	23	3,80	15	57
1,00	25	25	1,61	13	21	3,80	25	95
1,09	21	23	1,65	23	38	4,00	19	76
1,09	23	25	1,67	15	25	4,13	23	95
1,10	19	21	1,77	13	23	4,38	13	57
1,12	17	19	1,81	21	38	4,47	17	76
1,13	15	17	1,92	13	25	4,52	21	95
1,15	13	15	2,00	19	38	5,00	19	95
1,19	21	25	2,23	17	38	5,07	15	76
1,21	19	23	2,28	25	57	5,59	17	95
1,23	17	21	2,48	23	57	5,85	13	76
1,27	15	19	2,53	15	38	6,33	15	95
1,31	13	17	2,71	21	57	7,31	13	95
1,31	19	25	2,92	13	38			

We choose double sprockets to be implemented in our gearbox. The diameter of the sprocket with 17 teeth is Ø56 and that of the sprocket with 38 teeth is Ø100.

T.I.B.

Sprocket for taper bush

Bushes - DIN 8187 - ISO 606

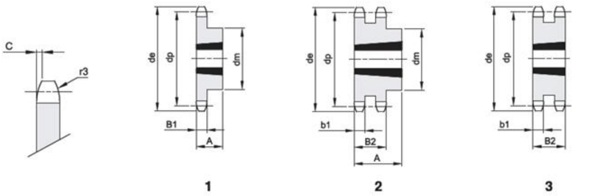
1/2" x 5/16"

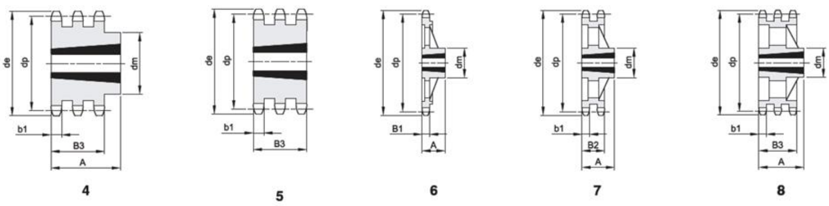
12.7 x 7.75 mm

Roller **8.51 mm**

08B - 1 - 2 - 3

B1	B2	B3	b1	C	r3
7.2	21	34.9	7	1.3	13





Z	de	dp	Simple Sprokets dm	A	Bush	type	KG	Double Sprokets dm	A	Bush	type	KG	Triple Sprokets dm	A	Bush	type	KG
15	66.5	61.80	45	22	1008	1	0.16	46	22	1008	2	0.22		34.9	1008	5	0.36
16	69.9	65.10	50	22	1108	1	0.20	50	22	1108	2	0.22					
17	74.5	69.11	60	25	1210	1	0.22	56	25	1210	2	0.23		34.9	1210	5	0.35
18	78.0	73.14	60	25	1210	1	0.27	60	25	1210	2	0.30					
19	82.5	77.15	63	25	1210	1	0.33	62	25	1210	2	0.38	62	38	1215	4	0.61
20	86.0	81.19	67	25	1610	1	0.27	66	25	1610	2	0.45					
21	90.6	85.21	71	25	1610	1	0.36	70	25	1610	2	0.50	70	38	1615	4	0.65
22	94.1	89.24	71	25	1610	1	0.36	76	25	1610	2	0.55					
23	98.7	93.26	76	25	1610	1	0.50	79	25	1610	2	0.62	70	38	1615	4	0.93
24	102.1	97.29	76	25	1610	1	0.53	84	32	2012	2	0.68					
25	106.7	101.32	76	25	1610	1	0.56	87	32	2012	2	0.72		34.9	2012	5	0.85
26	110.2	105.36	76	25	1610	1	0.60	87	32	2012	2	0.82					
27	114.8	109.39	76	25	1610	1	0.63	87	32	2012	2	0.92		34.9	2012	5	1.18
28	118.3	113.42	90	32	2012	1	0.77	87	32	2012	2	1.10					
30	126.9	121.49	90	32	2012	1	0.91	87	32	2012	2	1.24		34.9	2012	5	1.73
38	159.2	153.79	90	32	2012	1	1.25	100	32	2012	2	2.50		34.9	2012	5	3.53
45	188.6	182.07	111	32	2012	1	1.68										
57*	237.1	230.54	111	32	2012	6	2.78	111	32	2012	7	3.64					
76*	313.9	307.33	111	32	2012	6	3.81	111	32	2012	7	5.09					

Speed Ratio of Gearbox

$$i_{gearbox} = \frac{i_{system}}{i_{chain}} = \frac{12.57}{2.24} = 5.612$$

Chain-Table 1: Service Factors

	TYPES OF PRIME MOVER					
	'Soft' starts			'Heavy' starts		
	Electric motors: A.C. - Star-delta start D.C. - Shunt wound Internal combustion engines with 4 or more cylinders. All prime movers fitted with centrifugal clutches, dry or fluid couplings.			Electric motors: A.C. - Direct-on-line start D.C. - Series and compound wound Internal combustion engines with less than 4 cylinders.		
	Hours per day duty					
TYPES OF DRIVEN MACHINE	10 and under	Over 10 to 16	Over 16	10 and under	Over 10 to 16	Over 16
Light Duty Agitators (uniform density), Belt conveyors (uniformly loaded).	1,0	1,1	1,2	1,1	1,2	1,3
Medium Duty Agitators and mixers (variable density), Belt conveyors (not uniformly loaded), Kilns Laundry machinery, Line-shafts, Machine tools, Printing machinery, Sawmill and woodworking machinery, Screens (rotary).	1,1	1,2	1,3	1,2	1,3	1,4
Heavy Duty Brick machinery, Bucket elevators, Conveyors (heavy duty), Hoists, Quarry plant, Rubber machinery, Screens (vibrating), Textile machinery.	1,3	1,4	1,5	1,5	1,6	1,7

Service Factor:

Indicates the demand for the power coefficient based on specific conditions. For our system, the prime mover is an irrigation/material, and it should be able to operate 8 hours per day with a design life of 10 years. it should also be able to perform medium duty and soft starts for farmer because soft starts save the motor and keep water flowing., so the service factor is 1.1 in our system.

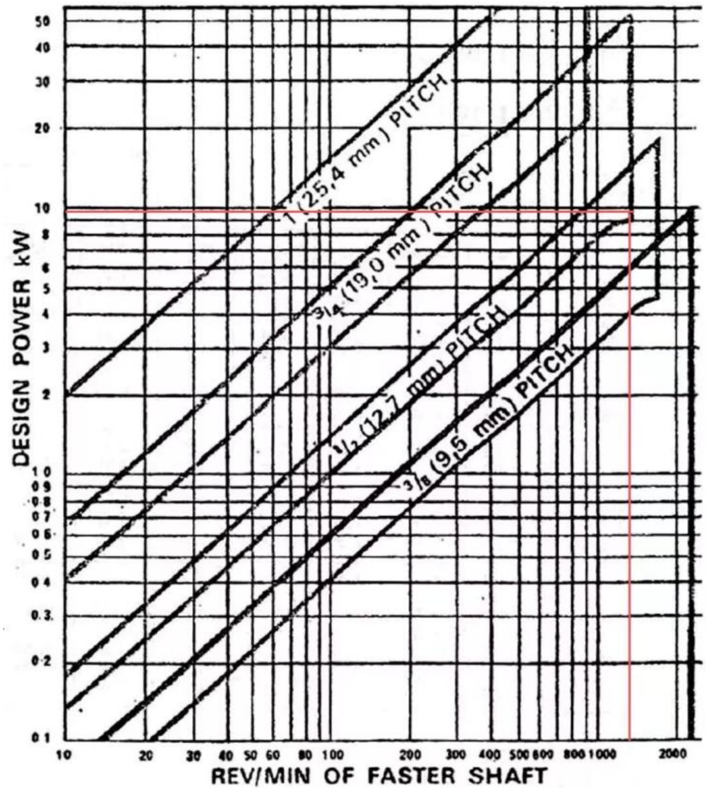
Power

The nominal Power is FV/1000 = (6000)(1.2)/1000 = 7.2kW

Design Power = 7.2 x SF = (7.2)(1.1) = 7.92 kW

Chain Pitch

From graph relating Chain Pitch to rated power, the intersection falls into the region of $\frac{1}{2}'$ pitch (12.7 mm), thus $\frac{1}{2}'$ pitch should be used in our system.



Rated power

3/4 (19mm) pitch

Rev/min faster shaft	Rated power (kW) for small sprocket																					Lubrication
	Number of teeth																					
	13			15			17			19			21			23			25			
	Single	Duplex	Triplex	Single	Duplex	Triplex	Single	Duplex	Triplex	Single	Duplex	Triplex	Single	Duplex	Triplex	Single	Duplex	Triplex	Single	Duplex	Triplex	
10	0,12	0,20	0,30	0,15	0,26	0,38	0,17	0,29	0,43	0,19	0,32	0,48	0,21	0,36	0,53	0,23	0,39	0,58	0,25	0,43	0,63	Type 1
20	0,23	0,39	0,58	0,27	0,46	0,68	0,31	0,53	0,78	0,36	0,61	0,90	0,40	0,68	1,00	0,44	0,75	1,10	0,47	0,80	1,18	
30	0,33	0,56	0,83	0,39	0,66	0,98	0,45	0,77	1,13	0,51	0,87	1,28	0,57	0,97	1,43	0,63	1,07	1,58	0,68	1,16	1,70	
40	0,43	0,73	1,08	0,51	0,87	1,28	0,59	1,00	1,48	0,66	1,12	1,65	0,74	1,26	1,85	0,81	1,38	2,03	0,88	1,50	2,20	
50	0,52	0,88	1,30	0,62	1,05	1,55	0,72	1,22	1,80	0,81	1,38	2,03	0,90	1,53	2,25	0,99	1,68	2,48	1,08	1,84	2,70	
60	0,62	1,05	1,55	0,73	1,24	1,83	0,85	1,45	2,13	0,96	1,63	2,40	1,06	1,80	2,65	1,17	1,99	2,93	1,27	2,16	3,18	
70	0,71	1,21	1,78	0,84	1,43	2,10	0,97	1,65	2,43	1,10	1,87	2,75	1,22	2,07	3,05	1,34	2,28	3,35	1,46	2,48	3,65	
80	0,80	1,36	2,00	0,95	1,62	2,38	1,10	1,87	2,75	1,24	2,11	3,10	1,38	2,35	3,45	1,52	2,58	3,80	1,65	2,81	4,13	
90	0,89	1,51	2,23	1,06	1,80	2,65	1,22	2,07	3,05	1,38	2,35	3,45	1,53	2,60	3,83	1,69	2,87	4,23	1,83	3,11	4,58	
100	0,98	1,67	2,45	1,16	1,97	2,90	1,34	2,28	3,35	1,51	2,57	3,78	1,69	2,87	4,23	1,85	3,15	4,63	2,02	3,43	5,05	
200	1,83	3,11	4,58	2,17	3,69	5,43	2,50	4,25	6,25	2,83	4,81	7,08	3,15	5,36	7,88	3,46	5,88	8,65	3,76	6,39	9,40	
300	2,63	4,47	6,58	3,12	5,30	7,80	3,60	6,12	9,00	4,07	6,92	10,18	4,63	7,70	11,33	4,98	8,47	12,45	5,42	9,21	13,55	Type 2
400	3,41	5,80	8,53	4,04	6,87	10,10	4,67	7,94	11,68	5,27	8,96	13,18	5,87	9,98	14,68	6,45	10,97	16,13	7,02	11,93	17,55	
500	4,17	7,09	10,43	4,94	8,40	12,35	5,70	9,69	14,25	6,45	10,97	16,13	7,18	12,21	17,95	7,89	13,41	19,73	8,58	14,59	21,45	
600	4,91	8,35	12,28	5,82	9,89	14,55	6,72	11,42	16,80	7,60	12,92	19,00	8,46	14,38	21,15	9,29	15,79	23,23	10,11	17,19	25,28	
700	5,64	9,59	14,10	6,69	11,37	16,73	7,72	13,12	19,30	8,73	14,84	21,83	9,71	16,51	24,28	10,68	18,16	26,70	11,62	19,75	29,05	
800	6,37	10,83	15,93	7,54	12,82	18,85	8,71	14,81	21,78	9,84	16,73	24,60	10,95	18,62	27,38	12,04	20,47	30,10	13,10	22,27	32,75	
900	7,08	12,04	17,70	8,39	14,26	20,98	9,68	16,46	24,20	10,94	18,60	27,35	12,18	20,71	30,45	13,39	22,76	33,48	14,57	24,77	36,43	
1000	7,78	13,23	19,45	9,22	15,67	23,05	10,64	18,09	26,60	12,03	20,45	30,08	13,39	22,76	33,48	14,72	25,02	36,80	16.02	27,23	40,05	

The rated power for the smaller sprocket is 4.5866x2.5=11.5kW, which is more than the required rated power. Therefore, the 17/25 teeth are verified.

Chain Length

Chain length can be found using the formula:

$$L = \frac{2C}{P} + \frac{T+t}{2} + \frac{KP}{C}$$

L: length of chain in pitches;

C: Centre distance in mm(30-50 pitches);

P: Pitch of chain in mm;

T: Number of teeth in the large sprocket;

t: Number of teeth in the small sprocket;

K: Factor from Table

The K-Factor can be found on the table:

$T - t = 38 - 17 = 21$

T-t	K
21	11
22	12
23	13
24	15
25	16
26	17
27	18
28	20
29	21
30	23

K-factor = 11

We assume the centre distance C to be 32 pitches, which is 608mm, and the pitch of chain P to be 19mm.

$$L = \frac{2(608)}{19} + \frac{38+17}{2} + \frac{11(19)}{608}$$

$$L = 91.84\, pitches$$

With iteration, we can make the centre distance $C = 612mm$.

Summary of Chain Parameters:

Design Power	11.5 kW
Chain Pitch	19 mm
Speed Ratio of Chain Drive	2.24
Teeth Number of Driving Sprocket	17
Teeth Number of Driven Sprocket	38
Chain Strand	3
Centre Distance	608 mm
Chain Length	92 pitches

5. Gear Design

5.1 Introduction

According to the gearbox design 1 mentioned above, there are a total of 4 gears (2 pairs) to transmit rotary motion. According to the design requirement, they achieve both speed reduction and torque amplification from the input shaft to the output shaft. For a pair of gears to work well, a single pair of gear should have the same module and pressure angle, with the same teeth size for meshing as well as the same force action line which keeps the force on the gears stable when they are rotating.

5.2 Types of Gears

There are several available gear structures and designs, such as spur gears, worm gears, rack and pinion, bevel gears, helical gears, etc. Inside the gearbox, the 3 fixed shafts will be installed in parallel positions, and rotary motion transmission is required. Therefore, both spur gears and helical gears are considered and compared as below.

Pros and Cons of Spur and Helical Gears

Helical Gear

Spur Gear

Gear Set Design Parameters

Gear	1st Pair Pinion	1st Pair Driven Gear	2nd Pair Pinion	2nd Pair Driven Gear
Material	S45C+HRC50-55	S45C+HRC50-55	S45C+HRC50-55	S45C+HRC50-55
Number of Teeth	24	45	20	60
Module	4	4	5	5
Pitch Circle Diameter (mm)	96	180	100	300
Face Width (mm)	44	44	55	55
Pressure Angle	20	20	20	20

Gear Ratio

The number of teeth for the first pair of gears is 20:60, a ratio of 3. It was initially set at 17:45 as the minimum requirement on number of teeth of a gear so that it does not cause interference and torsion of the gear is 17. However, it was later iterated to current ratio as the calculation does not match with the value of speed ratio of the chain.

The number of teeth for the second pair of gears is 24:45, a ratio of 1.875. The module for the first set of gears is 4, and that of the second set of gears is 5. For both pathways, materials satisfying the stress requirements are available in the market.

5.3 Bending Stress and Allowable Bending Strength

Bending Stress

Bending Stress $\sigma_t = \frac{F_t}{mbJ} \cdot \frac{K_a K_s K_m K_b}{K_v}$

b: face width

J: geometry factor

K_a: application factor for bending strength – load condition

K_s: size factor for bending strength – teeth size

K_m: load distribution factor for bending strength – face width

K_b: rim thickness factor – rim design

K_v: dynamic factor – velocity

Pinions will be considered for the analysis as it is subjected to higher bending and contact stress than the gear.

Tangential Force of the pinions

$$F_t = \frac{P}{2\pi \times \frac{v}{60} \times \frac{D}{2}}$$

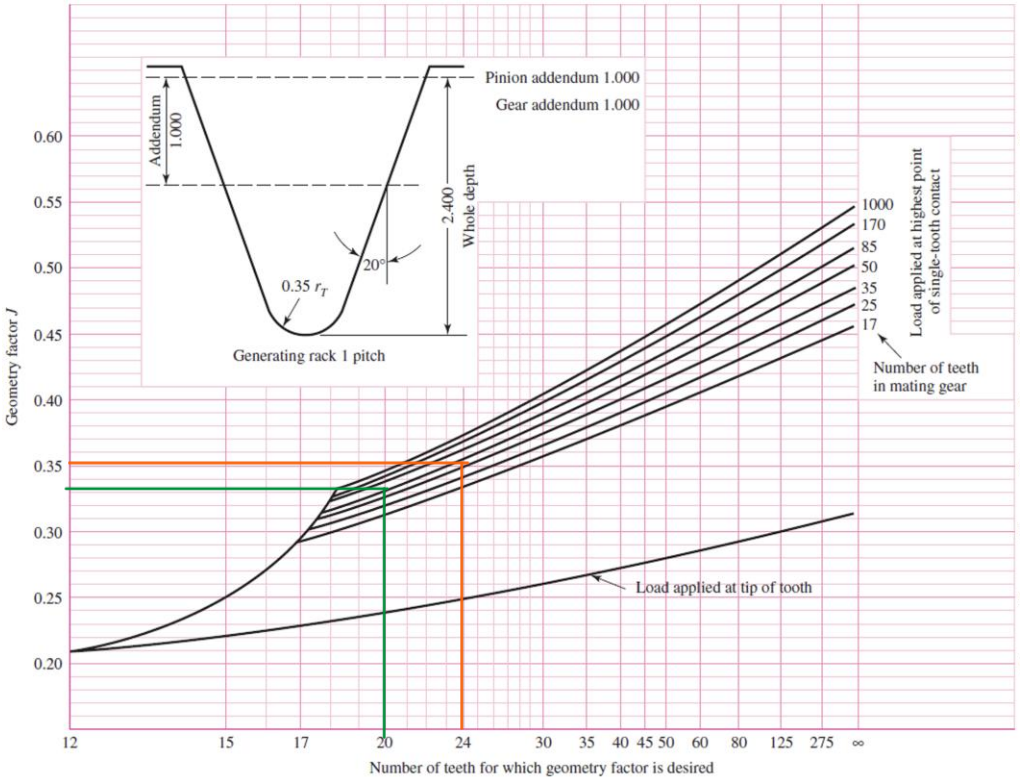
P: Power(W)

v: Rotation Speed of pinion (rpm)

Pinion	**Pinion 1**	**Pinion 2**

P(W)	9912	9327
v(rpm)	1440	218
V(m/s)	2.304	0.363
$\boldsymbol{F_t}$**(N)**	1369	8171
D	96	100

Geometry Factor



Geometry Factor J of pair 1 = 0.352, pair 2 = 0.332

Application Factor (K_a)

	Driven Machinery			
Source of Power	Uniform	Light Shock	Medium Shock	heavy Shock
Uniform	1.0	1.25	1.5	1.75
Light shock	1.25	1.4	1.75	2.0
Medium shock	1.5	1.7	2.0	2.5

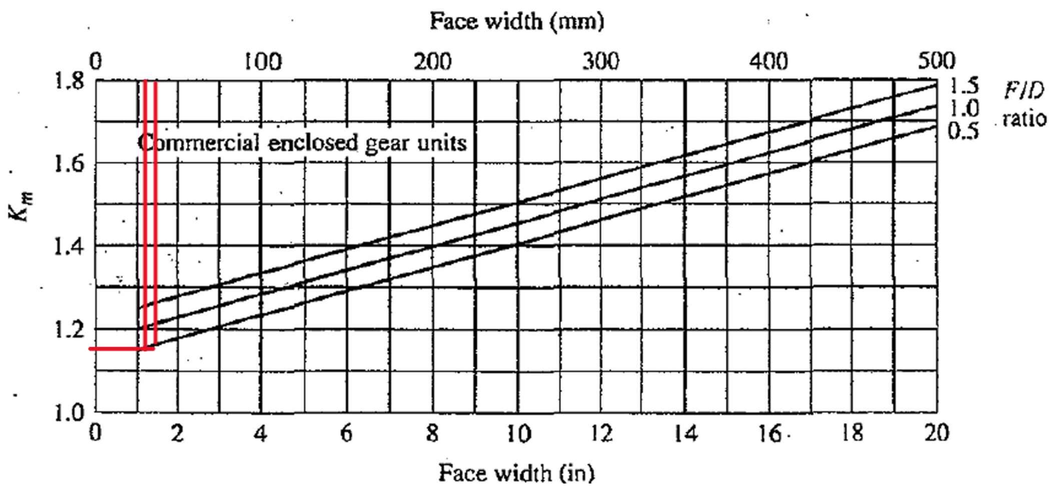
For both pairs of gears, K_a is selected as 1.5.

Size Factor K_s

Diametral Pitch, P_d	*Metric Module*, m	*Size Factor* K_s
≥5	≤5	1.00
4	6	1.05
3	8	1.15
2	12	1.25
1.25	20	1.40

For easier calculation and analysis, both pair of gears are set as $K_s = 1$

Load Distribution Factor

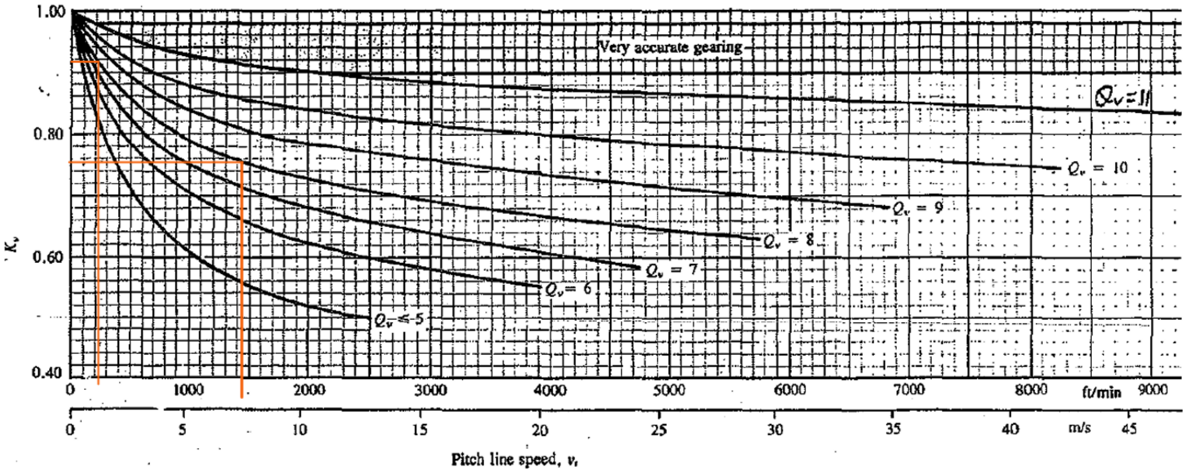


For both pairs, $K_m = 1.15$

Rim thickness factor

For both pairs, $K_b = 1$

Dynamic Factor



Both sets of gears have a quality number Q_v of 8.

For pair 1, $K_v = 0.76$, for pair 2, $K_v = 0.92$

A summary of parameters of two pairs of gear set:

Pairs of Gear Set	Pair 1	Pair 2
F_t Force(N)	1369	8171
m Module(m)	4	5
b Face Width(m)	44	55
J	0.352	0.332
K_a	1.5	1.5
K_b	1	1
K_s	1	1
K_m	1.15	1.15
K_v	0.76	0.92
σ_t(MPa)	50.156	167.81

Allowable bending strength σ_{at}: $\sigma_t \leq \sigma_{at} \times \frac{K_L}{K_R}$

K_L: life adjustment factor

K_R: reliability adjustment factor

Reliability Adjustment Factor

Reliability adjustment factor K_R

Reliability required	K_R
0.90	0.85
0.99	1.00
0.999	1.25
0.9999	1.50

We assume that the gear operates at 99.9% reliability, therefore, $K_R = 1.25$.

Number of load cycles, N	KL
10^7	1.00
10^8	0.92
10^9	0.87
10^{10}	0.8

The number of load cycles of pinion in pair 1:

$1440 \times 60 \times 8 \times (365 \times 10 + 2) = 2524262400 = 2.52 \times 10^9$ cycles

Therefore, the life adjustment factor K_L for pair 1 is 0.87

The number of load cycles of pinion in pair 2:

$218 \times 60 \times 8 \times (365 \times 10 + 2) = 2524262400 = 382145280 = 3.82 \times 10^8$ cycles

Therefore, K_L for pair 2 is 0.92

Parameters for Calculation of Minimum Allowable Bending Strength:

Gear Pair	**Pair 1**	**Pair 2**
$\boldsymbol{\sigma_t}$ **(MPa)**	50.156	167.81
$\boldsymbol{K_L}$	0.87	0.92
$\boldsymbol{K_R}$	1.25	1.25
$\boldsymbol{\sigma_{at}}$ **(MPa)**	72.063	228

Surface Contact Stress and Allowable Contact Strength

Surface contact stress $\sigma_H = C_p \sqrt{\frac{F_t}{bD_1 I} \cdot \frac{K_a K_s K_m}{K_v}}$

C_p: elastic coefficient

D_1: pitch diameter of pinion

I: geometry factor $= \frac{\sin\emptyset \cos\emptyset}{2} \cdot \frac{R}{R+1}$, where R= gear ratio

$\Rightarrow AAA\sqrt{BBBMPa} = CCC\ MPa$

Elastic Coefficient

Values of Elastic Coefficient C_P for Spur Gears, in $\sqrt{\text{MPa}}$

Pinion Material ($\nu = 0.30$ in All Cases)	Gear Material: Steel	Cast Iron	Aluminum Bronze	Tin Bronze
Steel, $E = 207$ GPa	191	166	162	158
Cast iron, $E = 131$ GPa	166	149	149	145
~~Aluminum bronze, $E = 121$ GPa~~	~~162~~	~~149~~	~~145~~	~~141~~
Tin bronze, $E = 110$ GPa	158	145	141	137

Steel is used in both gear pairs, thus $C_p = 191\text{MPa}$

Geometry Factor

$$I = \frac{\sin \emptyset \cos \emptyset}{2} \cdot \frac{R}{R+1}$$

For pair 1,

$$I = \frac{\sin 20^{\circ} \cos 20^{\circ}}{2} \cdot \frac{3}{3+1} = 0.121$$

For pair 2,

$$I = \frac{\sin 20^{\circ} \cos 20^{\circ}}{2} \cdot \frac{1.875}{1.875+1} = 0.105$$

5.4 Surface Contact Stress and Minimum Allowable Bending Strength

Gear Pair	Pair 1	Pair 2
Force (N)	1369	8171
Module (m)	0.004	0.005
Pitch Diameter (m)	0.096	0.1
K_a	1.5	1.5
K_s	1	1
K_m	1.15	1.15
K_v	0.76	0.92
I	0.121	0.105
σ_H**(MPa)**	156	326

Allowable contact strength σ_{ac}: $\sigma_H \le \sigma_{ac} C_{Li} C_R$

C_{Li}: life adjustment factor

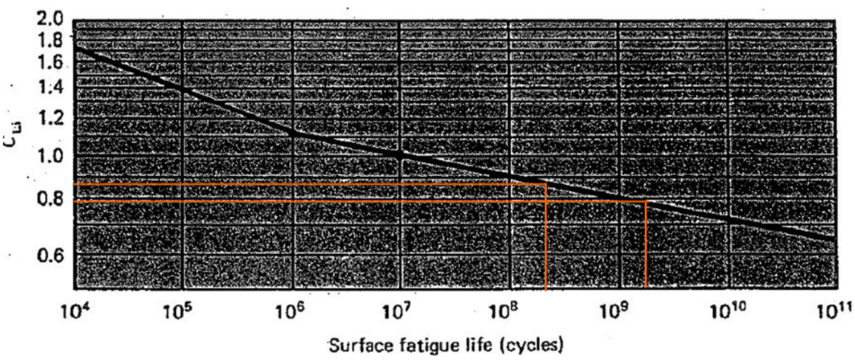
C_R: reliability factor

Reliability Factor C_R

Reliability (%)	C_R
50	1.25
99	1.00
99.9	0.80

The system is designed to have a 99.9% reliability, so C_R for both gear pairs are 0.8.

Life Adjustment Factor C_{Li}



For pair 1, $1440 \times 60 \times 8 \times (365 \times 10 + 2) = 2524262400 = 2.52 \times 10^9$ cycles. Therefore, $C_{Li} \approx 0.79$

For pair 2, $218 \times 60 \times 8 \times (365 \times 10 + 2) = 382145280 = 3.82 \times 10^8$ cycles. Therefore, $C_{Li} \approx 0.87$

Minimum Allowable Contact Strength

Pinion	Pair 1	Pair 2
σ_H (MPa)	156	326
C_{Li}	0.79	0.87
C_R	0.8	0.8
σ_{ac} (MPa)	247	468

Summary of Stress Analysis on two sets of pinions

	Bending Stress σ_t	**Surface Contact Stress σ_H**	**Least Allowable Bending Strength σ_{at}**	**Least Allowable Surface Contact Strength σ_{ac}**
Gear Pair 1	50.156 MPa	156 MPa	72.063 MPa	247 MPa

5.5 Material Selection

Transition Table:

Function	• **Power Transmission**
Objective	• Maximizing the strength of the gear • Minimizing the cost
Constraints	• Geometric constraint: size is specified by the gear box • Functional Constraint: Carry loading force (F) without failure
Free Variable	• Choice of materials

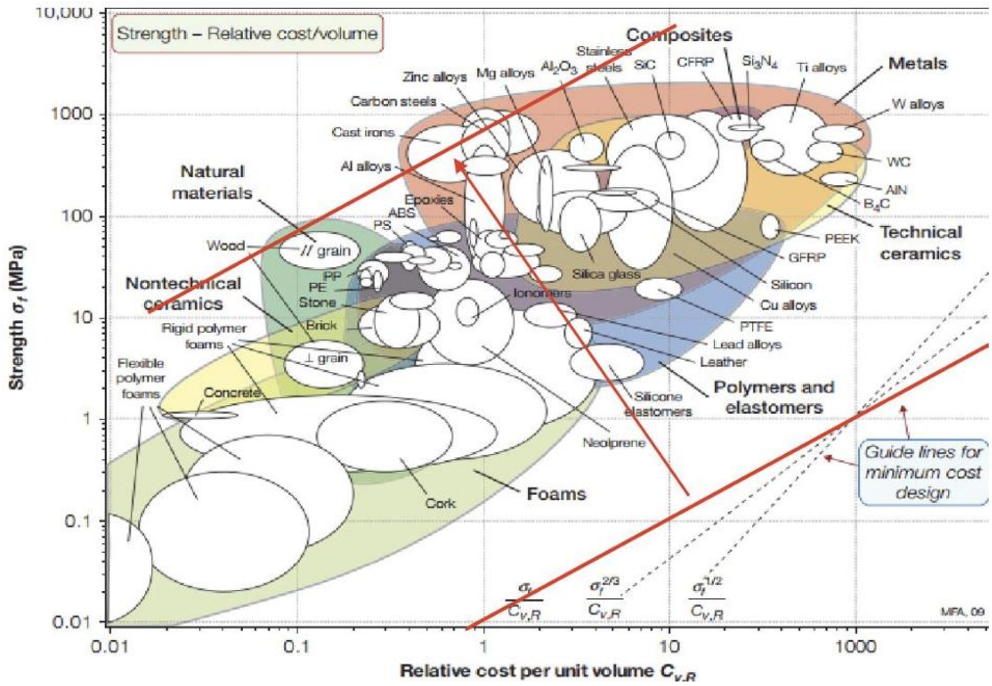
Material Indices

Objective Function	**Constraint Function**
$$\boldsymbol{C = mC_m = AbC_m\rho}$$ **where** $\boldsymbol{C}$**: cost** $\boldsymbol{C_m}$**: material cost per unit mass** $\boldsymbol{A}$**: area** $\boldsymbol{b}$**: face width of the gear** $\boldsymbol{\rho}$**: density**	$$\sigma_f \geq \frac{F^*}{A} \longrightarrow A \geq \frac{F^*}{\sigma_f}$$ where F^* is allowable loading given by safety regulation (failure strength)

Combining the objective function and constraint function, we get $C \geq \frac{F^*}{\sigma_f} bC_m\rho$.

Therefore, we need to maximize the material indices $M_{pl} = \frac{\sigma_f}{C_m\rho}$.

If we consider the stiffness instead of strength, we get $M_{pl} = \frac{E}{C_m\rho}$.



From the above table Strength-Relative cost per unit volume, the material candidates are wood, cast iron and carbon steel. The material strength factor is valued over cost for mining company.

While wood has a relatively low cost, its strength is also relativity low. It also exhibits a very low performance under outdoor condition for mining. Therefore, we focus on cast iron and carbon steel and make a comparison with these two materials.

	Weighting Factor	**Cast Iron**	**Carbon Steel**
Strength	3	1	2
Cost	2	2	1
Corrosion resistance	2	1	2
Melting point	1	1	2
Total score		11	14
Highest score: 3 Lowest score:1			

Therefore, carbon steel is select as a gear material.

A Summary of Gear Sets' Bending and Contact Strength

	Gear set 1	Gear set 2
Allowable bending strength $\boldsymbol{\sigma_{at}}$	$\boldsymbol{\sigma_{at}} > 72.063$ MPa	$\boldsymbol{\sigma_{at}} > 228$ MPa
Allowable surface contact strength $\boldsymbol{\sigma_{ac}}$	$\boldsymbol{\sigma_{ac}} > 247$ MPa	$\boldsymbol{\sigma_{ac}} > 468$MPa

To select the allowable stress numbers for case hardened steel gear materials, according to the table below, we would select the hardness of 50 HRC Grade 1 Flame or Induction Hardened Steel for both allowable bending stress and allowable contact stress for the two gear sets.

	Allowable Bending Stress Number				*Allowable Contact Stress Number*			
Hardness at Surface	*Grade 1 (Ksi)*	*Grade 1 (MPa)*	*Grade 2 (Ksi)*	*Grade 2 (MPa)*	*Grade 1 (Ksi)*	*Grade 1 (MPa)*	*Grade 2 (Ksi)*	*Grade 2 (MPa)*
Flame or Induction Hardened:								
50 HRC	45	310			170	1200	190	1300
54 HRC			55	380	175	1200	195	1300
Carburized and Case Hardened:								
55–64 HRC	55	380			180	1250		
58–64 HRC			65	450			225	1560
Nitrided AISI 4140:								
84.5 15N	34	230	45	310	155	1100	180	1250
Nitrided AISI 4340:								
83.5 15N	36	250	47	325	150	1050	175	1200
Nitrided Nitralloy 135M:[a]								
90.0 15N	38	260	48	330	170	1170	195	1350
Nitrided Nitralloy N:[a]								
90.0 15N	40	280	50	345	195	1340	205	1410

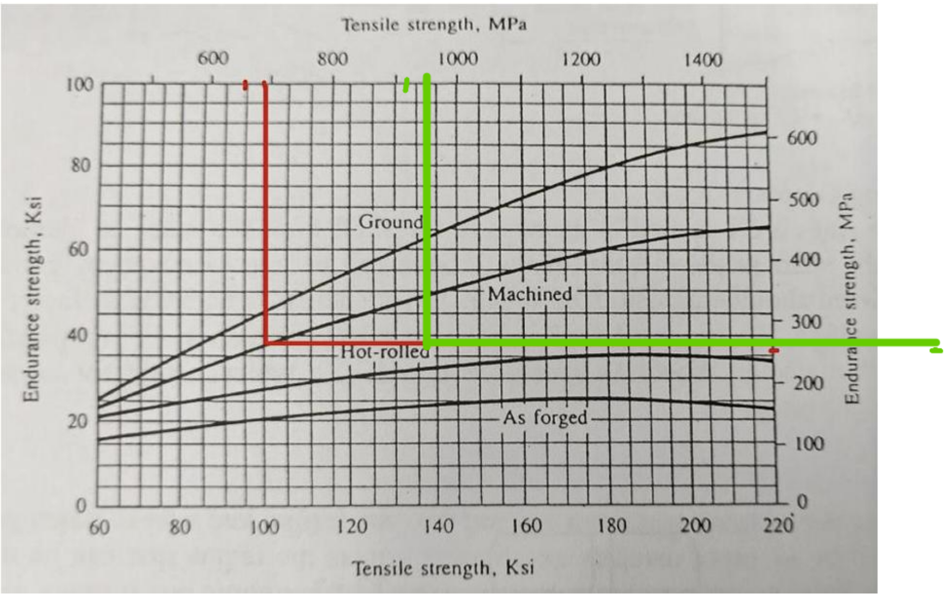
6. Shaft Design

6.1 Introduction

The shaft allows rotary and torque transmission between two pairs of gears, power input and output of the gearbox. With the presence of the bearings, gears, and torque input, they exhibit several combinations of shear forces, torsion, tension or compression & bending moment on shafts. However, as only one material is used for manufacturing our shafts to keep its simplicity, different diameters are designed at different location of the shaft to ensure it has sufficient strength. This section describes the thinking process of determining the minimum diameter at different location of shaft 1, 2 & 3.

6.2 Basic Information

For the material of the shafts, S45C/C45 Carbon Steel, the same material of gear sets, is chosen to be the material of the designed shafts. Referring to literatures, its ultimate tensile strength is 686 MPa and the yield strength is S_y is 490 MPa. From the Tensile Strength – Endurance Strength Table, we conclude that the endurance strength $S_n = 260$ MPa can be obtained.



By the modified endurance strength S'_n Equation:

$S'_n = S_n C_s C_r$, where:

S_n: Endurance strength

C_s: Size factor = $(\frac{D}{7.6})^{-0.068}$ for D < 50 mm, $1.85D^{-0.19}$ for 50mm < D < 250 mm

C_r: Reliability factor

C_s: D is firstly assumed to be 96mm. Substitute into the equation, C_s = 0.7772. Further adjust the value as 0.78 and concluded that D = 94.21 which is acceptable.

C_r: 0.75 as reliability is chosen as 99.9% to make sure reliability is high enough for a harsh mining field.

Therefore, the modified endurance strength = 260 x 0.78 x 0.75 = 152.1 MPa.

We select the safety factor N = 3, as the shaft is under varying loads with uncertainty on dynamic stress or material properties and analysis.

To calculate the designed shear stress, $\tau_d = 0.577\frac{S_y}{N} = 0.577\frac{490}{3} = 94.243$ MPa.

Material	S45C
Ultimate Tensile Strength	686 MPa
Yield Strength S_y	490 MPa
Endurance Strength S_n	260 MPa
Modified Endurance Strength S'_n	152.1MPa
Designed Shear Stress τ_d	94.243 MPa

Shaft Calculations

By $\frac{60P}{2\pi n}$,

Shaft	1	2	3
Revolution	1440	480	218.18
Power Input	9912.4W	9327.2W	8776.7W
Torque	65.73N	197.2N	433.45N

Calculation based on torsional strength

6.3 Length of Shafts

From the previous section of design of gears, the face width of the pinion and the gear in gear pair 1 are 44 mm, and the face width of gear pair 2 are 55 mm.

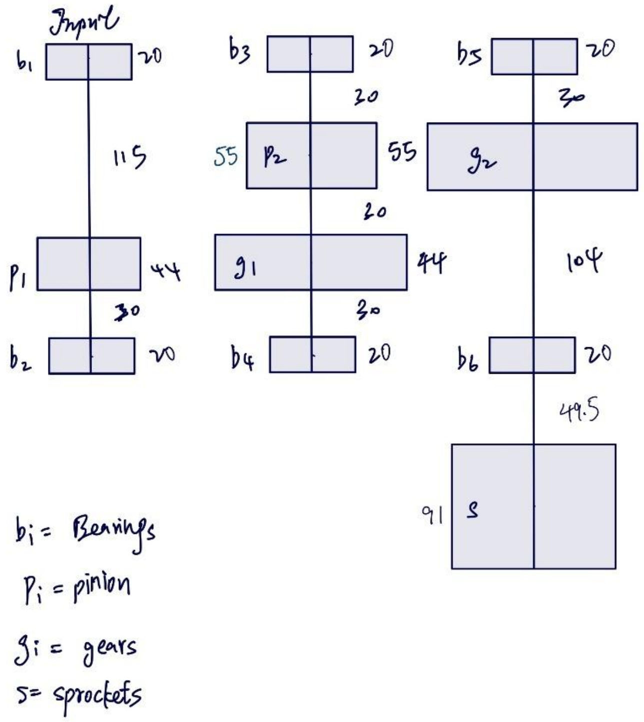
The separation between the components is assumed to be 30 mm.

The width of the bearing is assumed to be 25 mm.

As shaft 2 has the most components, the gear box is designed as a rectangular shape with fillets at the top edges, therefore, shaft 2 also has the minimum required length for shaft 1 & 3. The minimum end to end length of shaft 2 = 20 + 30 + 55 + 30 + 44 + 30 + 20 = 229 mm.

The face width of the sprocket is 91 mm and let the separation between the midpoint of bearing 6 and the sprocket be 100 mm.

The following figure summarizes the location of all components on shaft 1, 2 & 3:



<u>6.4 Force Analysis on Shafts by Gears and Sprockets</u>

With the equation to calculate the tangential force exerted on the gears and sprockets that were previously introduced in the gear section, we could calculate the value of F_t, and $F_r = F_t \times \tan(20°)$, which $20°$ is the pressure angle.

Pinion 1:

F_t= 1369N

$F_r = 1369 \tan\,(20°)$ = 498.275 N

Gear 1:

F_t= 1288.08N

$F_r = 1288.08 \tan\,(20°)$ = 468.823 N

Pinion 2:

F_t= 8171 N

$F_r = 8171\tan(20°) = 2974$ N

Gear 2:

F_t= 7699.884 N

$F_r = 7699.884\tan(20°)$=2802.529 N

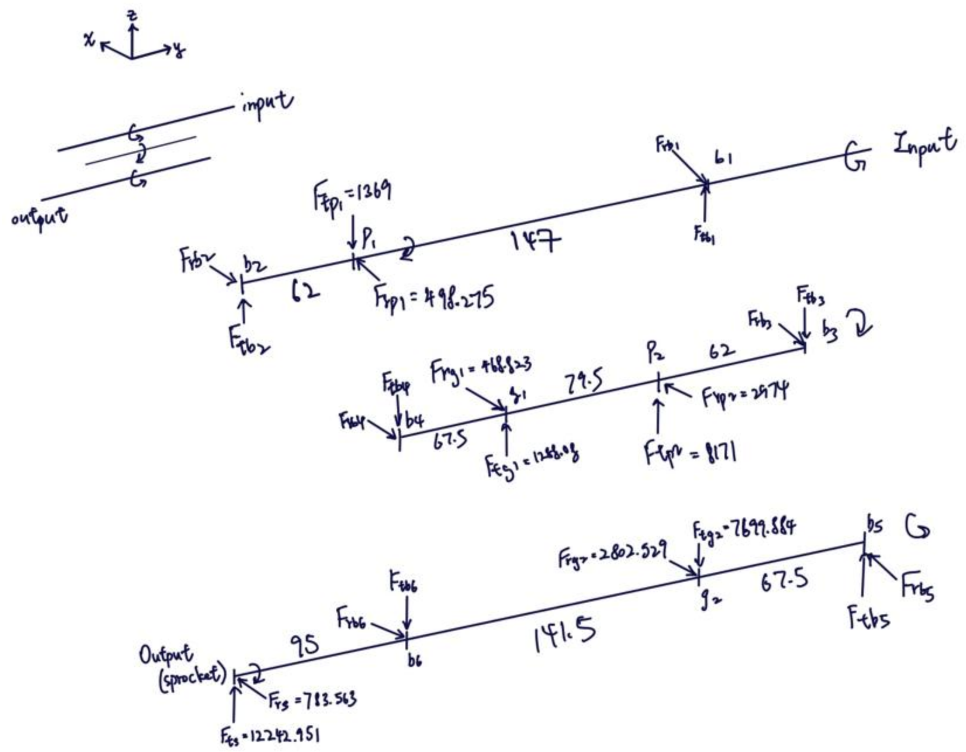
Sprocket:

The force exert on the sprocket is 12268 N 3.662° anticlockwise

F_t = 12268 cos 3.662 = 12242.951 N

F_r = 12268 sin 3.662 = 783.563 N

		Pinion 1	**Gear 1**	**Pinion 2**	**Gear 2**	**Sprocket**
Shaft 1	F_t (N)	1369				
	F_r(N)	498.275				
Shaft 2	F_t(N)		1288.08	8171		
	F_r(N)		468.823	2974		
Shaft 3	F_t(N)				7699.884	12242.951
	F_r(N)				2802.529	783.563



Where

F_{rpi}: Radial force from pinion i

F_{rgi}: Radial force from gear i

F_{rbi}: Radial force from bearing i

F_{tpi}: Tangential force from pinion i

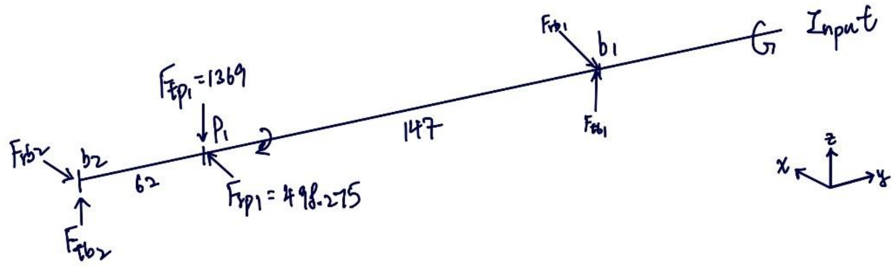
F_{tgi}: Tangential force from gear i

F_{tbi}: Tangential force from bearing i

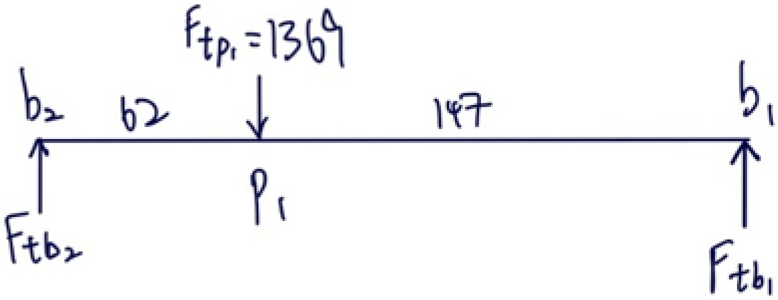
<u>6.5 The Forces on the Shafts by Bearings & Forces Diagram</u>

The forces acting on the bearings can be found by solving equations of moments on the vertical and horizontal components.

Shaft 1 Diagram



Tangential Forces (z-y Plane):



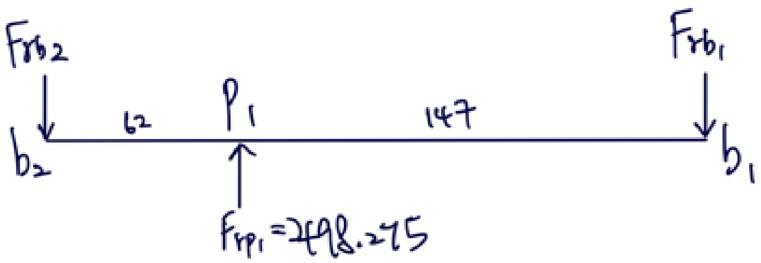
Take moment about b_2:

$1369 \times 0.062 - F_{tb1} \times 0.209 = 0$

$F_{tb1} = 577.401\text{N}$

$F_{tb2} = 791.599\text{N}$

Radial Force (x-y Plane):



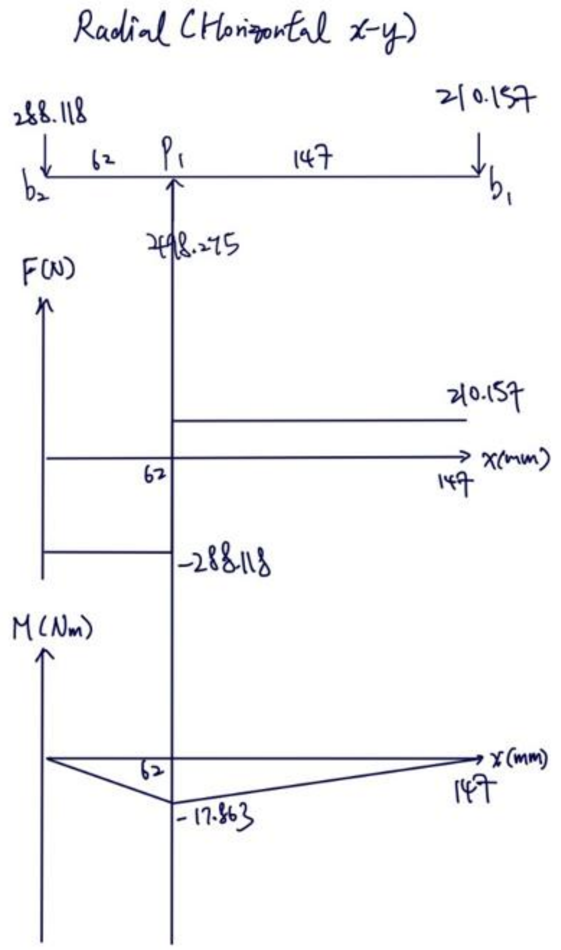
Take moment about b_2:

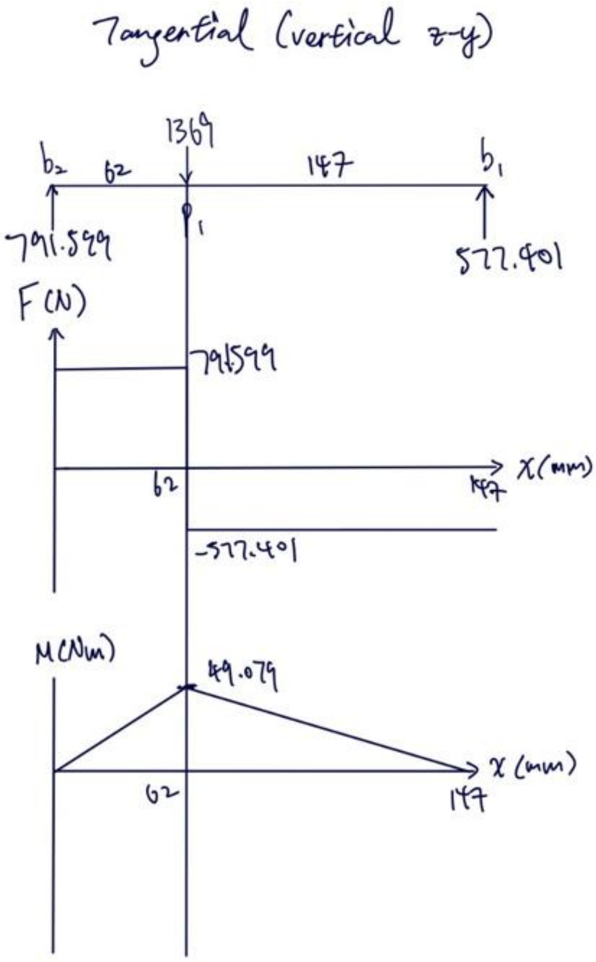
$498.275 \times 0.062 - F_{rb1} \times 0.209 = 0$

$F_{rb1} = 210.157\text{N}$

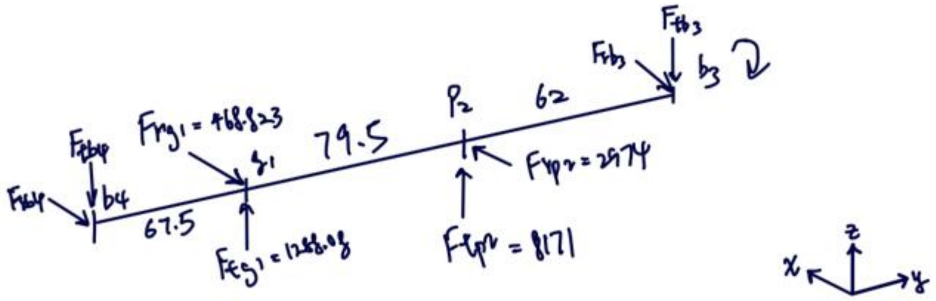
$F_{rb2} = 498.273 - 210.157 = 288.118\text{N}$

The figure below describes the shear force diagram and bending moment diagram for shaft 1.

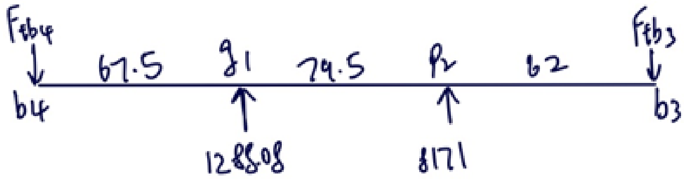




Shaft 2 Diagram



Tangential Force (z-y Plane):

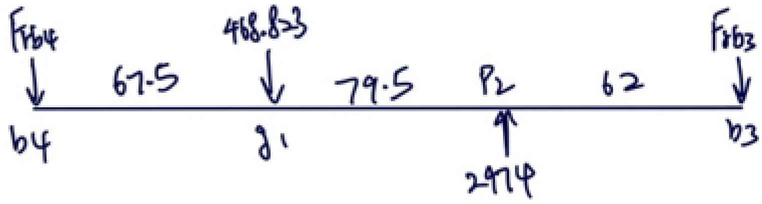


Take moment at F_{tb4}: $1288.08 \times 0.0675 - 8171 \times 0.147 + F_{tb3} \times 0.209 = 0$

$F_{tb3} = 6163.074N$

$F_{tb4} = 3296.006N$

Radial Force (x-y Plane):

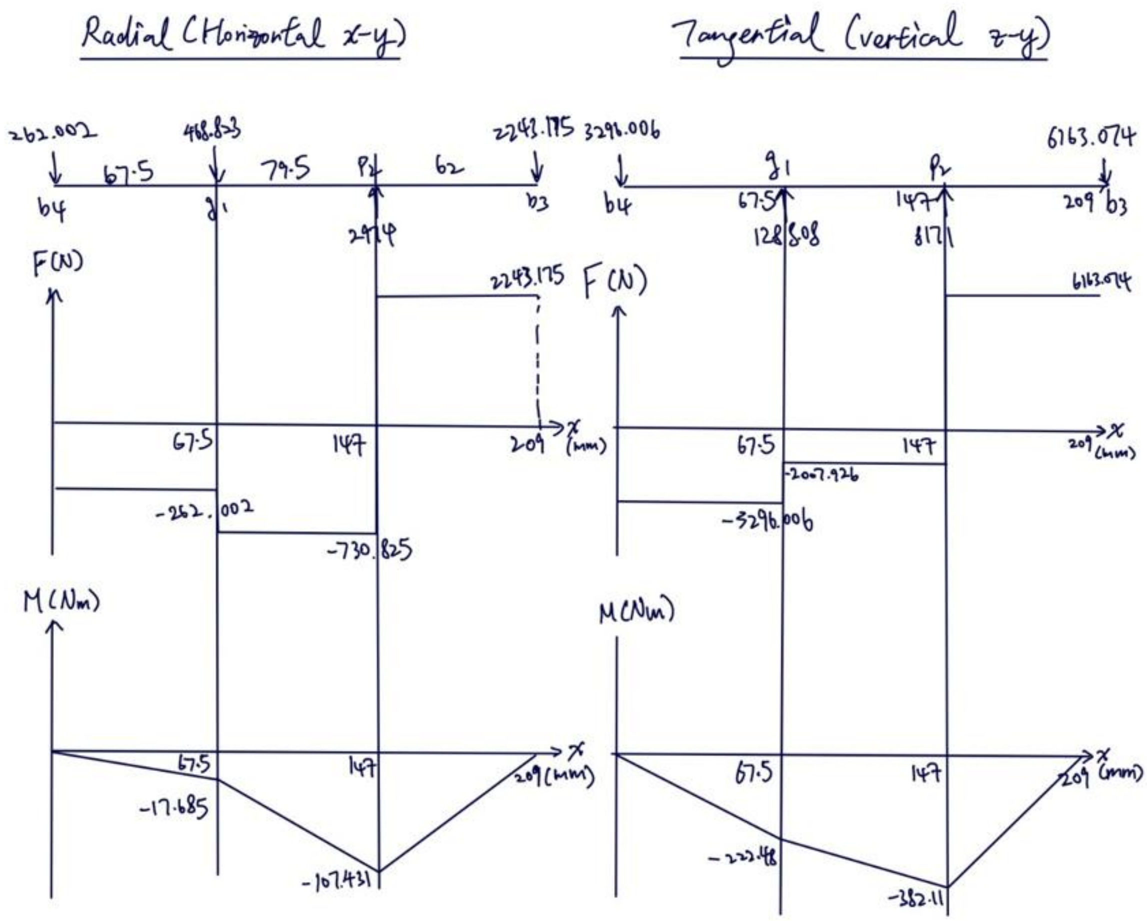


Take moment at F_{rb4}: $468.823 \times 0.0675 - 2974 \times 0.147 + F_{rb3} \times 0.209 = 0$

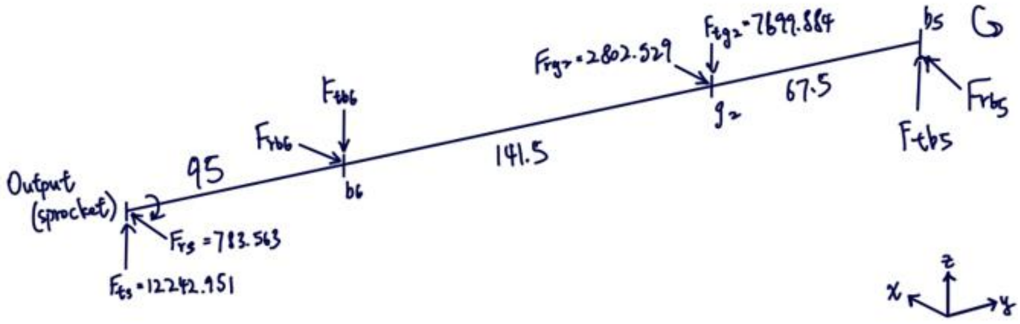
$F_{rb3} = 2243.175N$

$F_{rb4} = 262.002N$

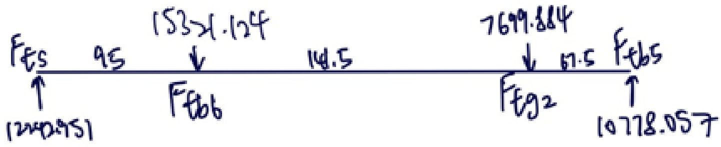
The diagram below illustrates the shear force and bending moment for shaft 2



Shaft 3 Diagram



Tangential Force Analysis (z-y Plane):

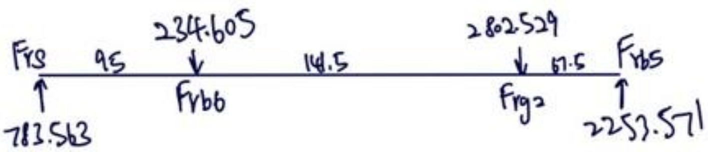


Take moment at F_{tb5}: $7699.884 \times 0.0675 + F_{tb5} \times 0.209 - 12242.951 \times 0.304 = 0$

$F_{tb6} = 15321.124N$

$F_{tb5} = 10778.057N$

Radial Force Analysis (x-y Plane):

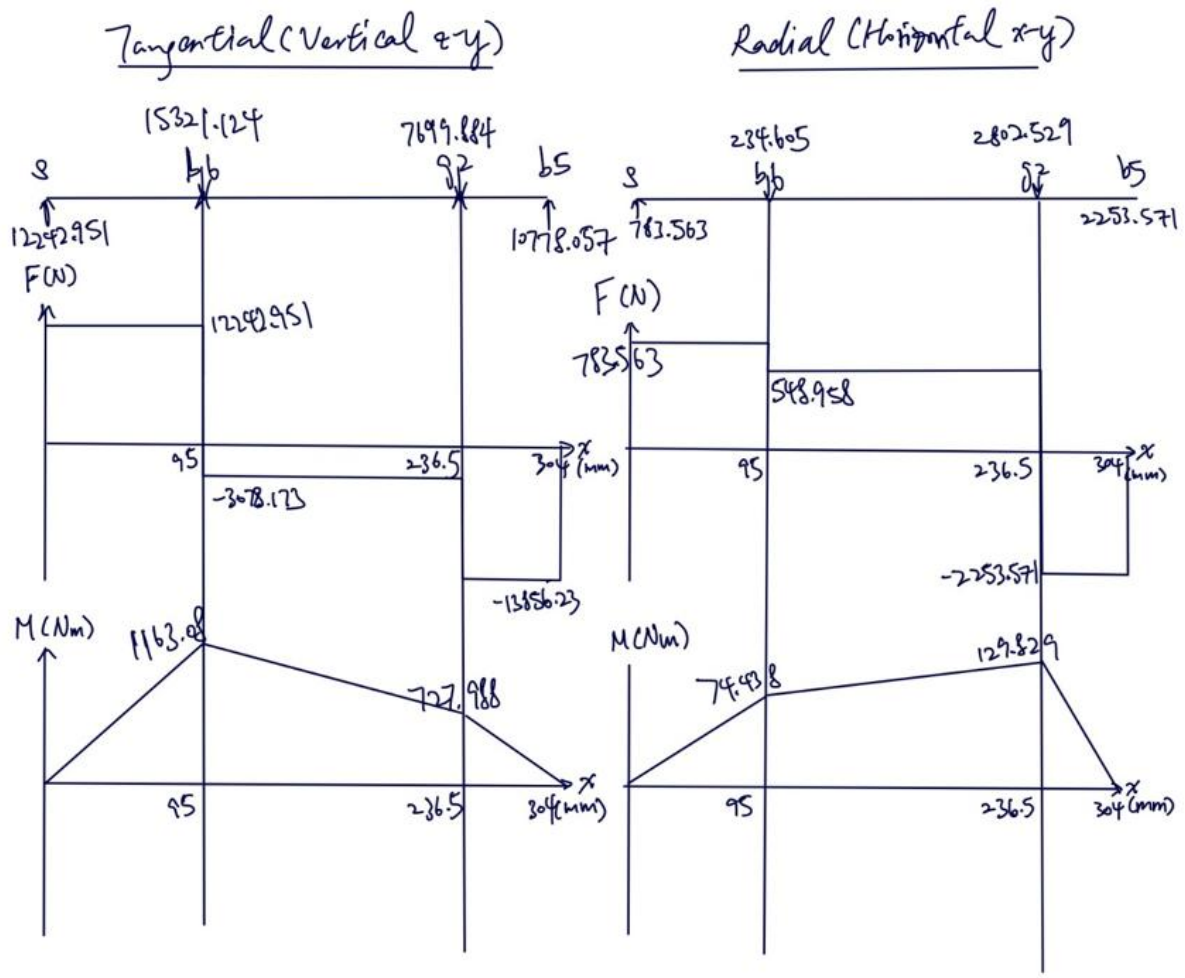


Take moment at F_{rb5}: $2802.529 \times 0.0675 + F_{rb6} \times 0.209 - 783.563 \times 0.304 = 0$

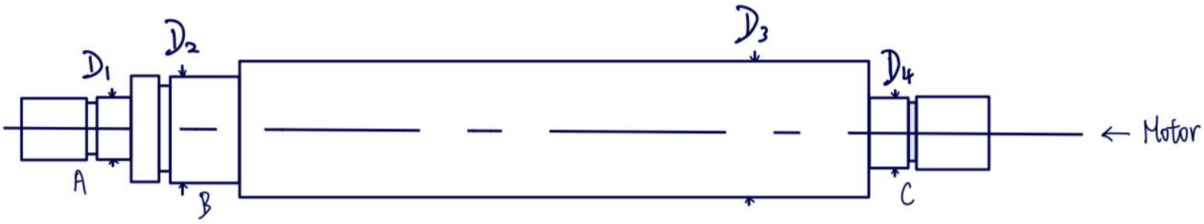
$F_{rb6} = 234.605N$

$F_{rb5} = 2253.571N$

The diagram below illustrates the shear force and bending moment for shaft 3.



Shaft 1



The figure above demonstrates the proposed design for the shaft, where A refers to the location of bearing 1, B refers to the location of pinion 1 and C refers to the location of bearing 2.

For D_1:

On the left side of D_1, components and forces involve retaining ring, no torque, no bending moment nor shear force.

On the right side of D_1, components and forces involve sharp fillet, no torque nor bending moment, involve shear force

Therefore, only the shear force in 2 planes is required to calculate.

Resultant shear forces:

$V_A = \sqrt{577.401^2 + 210.157^2}$

= 614.457N

$\left(\frac{4}{3}\right)\left(\frac{F_S}{A}\right) = \tau_d$

$A = \left(\frac{4}{3}\right)\left(\frac{F_S}{\tau_d}\right)$

$= \left(\frac{4}{3}\right)\left(\frac{614.457}{94.243\times10^6}\right) = 8.693 \times 10^{-6}$m

$A = \pi(\frac{D}{2})^2$

$D = 2\sqrt{\frac{A}{\pi}}$

$= 2\sqrt{\frac{8.693\times10^{-6}}{\pi}}$

$= 0.00333m = 3.33mm$

Therefore, the minimum diameter for $D_1 = 3.33mm$

For D_2:

On the right side of D_2, components and forces involve profile keyseat for pinion 1, well-rounded fillet, with torque and bending moment. Considering that the stress concentration factor for profile keyseat is larger than that of the well-rounded fillet side, it is ensured that the stress concentration from the profile keyseat is enough.

On the left side of D_2, components and forces involve a retaining ring, no torque and has bending moment. The resultant bending moment to the left side: $= \sqrt{17.863^2 + 49.079^2}$

$= 52.229$ Nm

Stress concentration factor for retaining ring $K_t = 3$.

Designed bending stress:

$\sigma_d = \frac{S'_n}{NK_t}$

$= \frac{156}{3\times3}$

$= 17.333\ MPa$

$\sigma_d = \frac{M}{Z}$

$Z = \frac{M}{\sigma_d}$

$= \frac{52.229}{17.333\times10^6} = 3.013 \times 10^{-6}\ m^3$

$Z = \frac{\pi D^3}{32}$

$D = \sqrt[3]{\frac{32Z}{\pi}}$

$= \sqrt[3]{\frac{32\times3.013\times10^{-6}}{\pi}}$

$= 0.0313$ m $= 31.3$ mm

To the right side, the stress concentration factor for profile keyseat $K_t = 2$

Torque = 49.079 Nm

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

$$= \left[\frac{32\times3}{\pi}\sqrt{\left[\frac{2\times52.229}{156\times10^6}\right]^2 + \frac{3}{4}\left[\frac{65.73}{490\times10^6}\right]^2}\right]^{\frac{1}{3}}$$

$= 27.5$ mm $\qquad < 31.3$ mm

Minimum diameter for $D_2 = 31.3$ mm

For D_3:

As D_3 is located between D_2 & D_4, and a step-up is needed to restrict movement of pinion 1 and bearing 2, so $D_3 > D_2$ & D_4.

For D_4:

On the left side of D_4, components and forces involve sharp fillet, no bending moment and involve torque and shear force. On the right side of D_4, components and forces involve retaining ring, no bending moment nor shearing force and involve torque. Therefore, only the torsional shear strength is required to be calculated. Recall the torque calculated is 49.079 Nm.

$$\frac{T}{Z_p} = \tau_d$$

$$Z_p = \frac{T}{\tau_d}$$

$$= \frac{65.73}{94.243 \times 10^6} = 6.975 \times 10^{-7}\ m^3$$

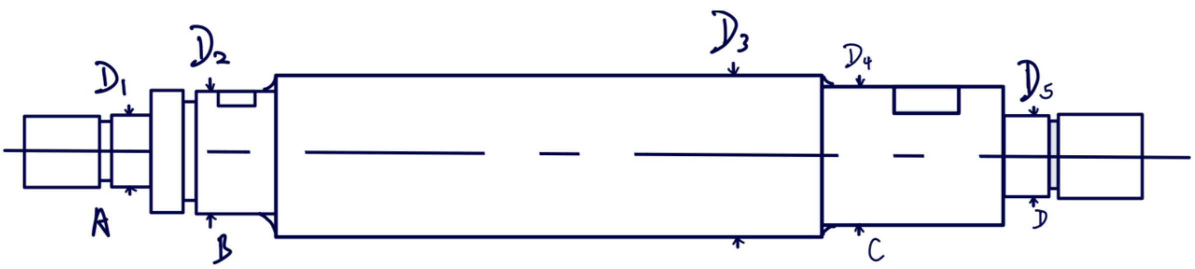
$$Z_p = \frac{\pi D^3}{16}$$

$$D = \sqrt[3]{\frac{16 Z_p}{\pi}}$$

$$= \sqrt[3]{\frac{16 \times 6.975 \times 10^{-7}}{\pi}} = 15.258 \text{ mm}$$

The minimum diameter for $D_4 = 15.258$ mm

<u>Shaft 2</u>



The figure above demonstrates the proposed design for the shaft, where A refers to the location of bearing 3, B refers to the location of gear 1, C refers to the location of pinion 2 and D refers to the location of bearing 4.

Since there is an obvious increase in bending moment at component C (pinion 2) and a continuous increase in torque from shaft 1 to 3, retaining ring will be replaced by spacer to fix the position of pinion 2. This method can achieve a lower stress concentration, therefore, there is no need to be a groove for retaining ring. Moreover, the shaft can be designed with a smaller diameter compared to that of in the design of shaft 1.

For D_1:

On the left side of D_1, components and forces involve a retaining ring, with no torque, no bending moment and no shear force. On the right side of D_1, components and forces involve sharp fillets, with no torque and bending moment, and involve shear force. Therefore, for D_1, only the shear force in 2 planes is required to be calculated.

$V_A = \sqrt{3296.006^2 + 262.002^2}$

$= 3306.403$ N

$\left(\frac{4}{3}\right)\left(\frac{F_S}{A}\right) = \tau_d$

$A = \left(\frac{4}{3}\right)\left(\frac{F_S}{\tau_d}\right)$

$= \left(\frac{4}{3}\right)\left(\frac{3306.403}{94.243\times10^6}\right) = 4.678 \times 10^{-5} m^2$

$A = \pi(\frac{D}{2})^2$

$D = 2\sqrt{\frac{A}{\pi}}$

$= 2\sqrt{\frac{4.678\times10^{-5}}{\pi}} = 7.718$ mm

The minimum diameter of $D_1 = 7.718$ mm

For D_2:

On the left side of D_2, components and forces involve a retaining ring, with no torque, involves bending moment. On the right side of D_1, components and forces involve a profile keyseat for gear 1, well rounded fillet, and involve torque and bending moment. Therefore, for D_2, at the right side, as the stress concentration factor of profile keyseat is higher than that of the rounded fillet, the calculation on the stress concentration from the profile keyseat is suffice. The resultant bending moment would be: $= \sqrt{222.48^2 + 17.685^2}$

$= 223.182$ Nm

Stress concentration factor for the retaining ring $K_t = 3$

The designed bending stress would be:

$\sigma_d = \frac{S'_n}{NK_t}$

$= \frac{156}{3 \times 3}$

$= 17.333$ MPa

$\sigma_d = \frac{m}{Z}$

$Z = \frac{M}{\sigma_d}$

$= \frac{223.182}{17.333 \times 10^6}$

$= 1.28 \times 10^{-5}$ m^3

$Z = \frac{\pi D^3}{32}$

$D = \sqrt[3]{\frac{32Z}{\pi}}$

$= \sqrt[3]{\frac{32 \times 1.28 \times 10^{-5}}{\pi}} = 50.71$ mm

On the right side, the stress concentration factor for keyseat $K_t = 2$

Torque = 442.16 Nm

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

$$= \left[\frac{32\times3}{\pi}\sqrt{\left[\frac{2\times223.182}{156\times10^6}\right]^2+\frac{3}{4}\left[\frac{442.16}{490\times10^6}\right]^2}\right]^{\frac{1}{3}}$$

$= 44.92$ mm $\qquad < 50.71$ mm

Therefore, the minimum diameter for $D_2 = 50.71$ mm

For D_3:

As it is located between D_2 & D_4, a step-up in the diameter value for D_3 is needed to restrict movement of gear 1 and pinion 2, therefore, $D_3 > D_2$ & D_4.

For D_4:

On the left side of D_4, components and forces involve a well-rounded fillet, bending moment and torque. On the right side, it involves a profile keyseat for pinion 2, bending moment and torque. Therefore, for D_4, as the concentration factor of profile keyseat is higher than that of the rounded fillet, consideration on the calculation of the stress concentration from the profile keyseat is suffice. The resultant bending moment would be $= \sqrt{107.431^2 + 382.11^2}$

$= 396.93$ Nm

Stress concentration factor for keyseat $K_t = 2$

Recall the torque = 442.16 Nm

$$D = \left[\frac{32N}{\pi}\sqrt{\left[\frac{K_tM}{S'_n}\right]^2+\frac{3}{4}\left[\frac{T}{S_y}\right]^2}\right]^{\frac{1}{3}}$$

$$= \left[\frac{32\times3}{\pi}\sqrt{\left[\frac{2\times396.93}{156\times10^6}\right]^2+\frac{3}{4}\left[\frac{442.16}{490\times10^6}\right]^2}\right]^{\frac{1}{3}}$$

$= 53.98$ mm

The minimum diameter for $D_4 = 53.98$ mm

For D_5:

On the left side, components and forces involve a sharp fillet, with no torque and no bending moment, involve shear force. On the right side, it involves a retaining ring, no torque, no bending moment and no shear force. Therefore, for D_5, the shear force in 2 planes is required to be calculated only.

$V_D = \sqrt{6163.074^2 + 2243.175^2}$

=6558.606 N

$\left(\frac{4}{3}\right)\left(\frac{F_S}{A}\right) = \tau_d$

$A = \left(\frac{4}{3}\right)\left(\frac{F_S}{\tau_d}\right)$

$= \left(\frac{4}{3}\right)\left(\frac{6558.606}{94.243\times10^6}\right) = 9.48019 \times 10^{-5} m^2$

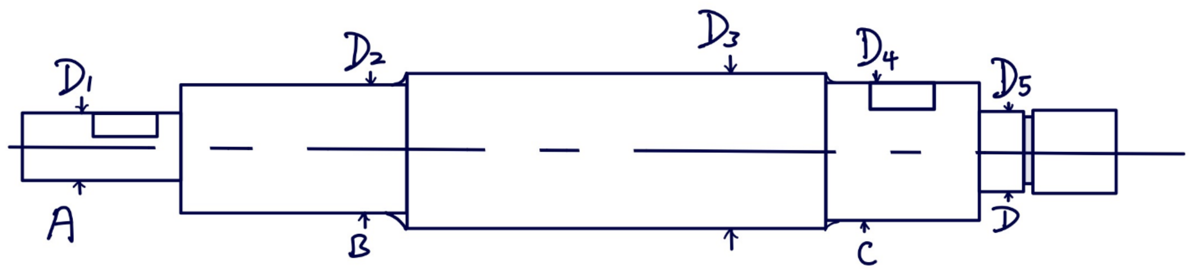
$A = \pi(\frac{D}{2})^2$

$D = 2\sqrt{\frac{A}{\pi}}$

$= 2\sqrt{\frac{9.48019\times10^{-5}}{\pi}} = 10.99$ mm

The minimum diameter of $D_5 = 10.99$ mm

Shaft 3:



The figure above demonstrates the proposed design for the shaft, where A refers to the location of the sprocket, B refers to the location of bearing 5, C refers to the location of gear 2 and D refers to the location of bearing 6.

Since a large torque is exerted on shaft 3 and a large bending moment is induced at component B (bearing 5) and component C (gear 2), two spacers instead of retaining rings are used to fix bearing 5 and gear 2 in their location. This method can reduce stress concentration, thus, grooves for retaining rings will not be needed. Moreover, the shaft can be designed with a smaller diameter than previous shafts and a nut is used to fix the sprocket.

For D_1:

On the left side, it involve no torque, no bending moment and no shear force. On the right side, it involves a profile keyseat for sprocket, a sharp fillet, with no bending moment, but has torque and shear force. Therefore, only the torsional shear strength to the right side of D_1 is required to calculate.

Torque = 1372.123 Nm

$$\frac{T}{Z_p} = \tau_d$$

$$Z_p = \frac{T}{\tau_d}$$

$$= \frac{1372.123}{94.243\times10^6} = 1.488 \times 10^{-5}\ m^3$$

$$Z_p = \frac{\pi D^3}{16}$$

$$D = \sqrt[3]{\frac{16Z_p}{\pi}}$$

$$= \sqrt[3]{\frac{16\times1.488\times10^{-5}}{\pi}} = 42.318 \text{ mm}$$

The minimum diameter for $D_4 = 42.318$ mm

For D_2:

On the left side, it involves torque and bending moment. On the right side, it involves a well-rounded fillet, has torque and bending moment. Therefore, the stress concentration factor from the well-rounded fillet is considered. The resultant bending moment is = $\sqrt{74.438^2 + 1163.08^2} = 1165.460$ Nm

The stress concentration factor for well-rounded fillet $K_t = 1.5$

Torque = 1372.123 Nm

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

$$= \left[\frac{32\times 3}{\pi}\sqrt{\left[\frac{2\times 1163.08}{156\times 10^6}\right]^2 + \frac{3}{4}\left[\frac{1372.123}{490\times 10^6}\right]^2}\right]^{\frac{1}{3}}$$

$= 77.29$ mm

Therefore, the minimum diameter for $D_2 = 77.29$ mm

For D_3:

As it is located between D_2 & D_4, a step-up in the diameter value for D_3 is needed to restrict movement of gear 2 and bearing 5, therefore, $D_3 > D_2$ & D_4.

For D_4:

On the left side of D_4, components and forces involve a well-rounded fillet, bending moment and torque. On the right side, it involves a profile keyseat gear 2, bending moment and torque. Therefore, for D_4, as the concentration factor of profile keyseat is higher than that of the rounded fillet, consideration on the calculation of the stress concentration from the profile keyseat is suffice. The resultant bending moment would be = $\sqrt{727.988^2 + 129.829^2}$

$= 739.474$ Nm

Stress concentration factor for keyseat $K_t = 2$

Recall the torque = 1372.123 Nm

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

$$= \left[\frac{32\times 2}{\pi}\sqrt{\left[\frac{2\times 739.474}{156\times 10^6}\right]^2 + \frac{3}{4}\left[\frac{1372.123}{490\times 10^6}\right]^2}\right]^{\frac{1}{3}} = 58.42 \text{ mm}$$

The minimum diameter for $D_4 = 58.42$ mm

For D_5:

On the left side, components and forces involve a sharp fillet, with no torque and no bending moment, involve shear force. On the right side, it involves a retaining ring, no torque, no bending moment and no shear force. Therefore, for D_5, the shear force in 2 planes is required to be calculated only.

$V_D = \sqrt{2253.571^2 + 13856.23^2}$

$= 14038$ N

$\left(\frac{4}{3}\right)\left(\frac{F_S}{A}\right) = \tau_d$

$A = \left(\frac{4}{3}\right)\left(\frac{F_S}{\tau_d}\right)$

$= \left(\frac{4}{3}\right)\left(\frac{14038}{94.243\times10^6}\right) = 1.986 \times 10^{-4} m^2$

$A = \pi(\frac{D}{2})^2$

$D = 2\sqrt{\frac{A}{\pi}}$

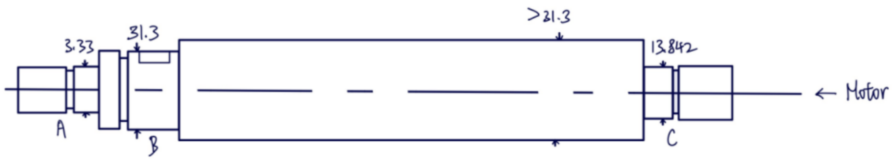
$= 2\sqrt{\frac{1.986\times10^{-4}}{\pi}} = 15.901$ mm

The minimum diameter of $D_5 = 15.901$ mm

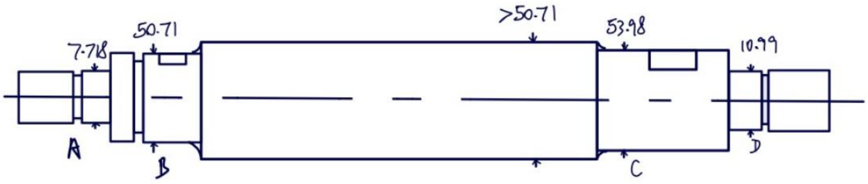
The following summarizes the minimum diameter at different locations of shaft 1, 2 & 3:

(In mm)	D_1	D_2	D_3	D_4	D_5
Shaft 1	3.33	31.3	> 31.3	13.842	
Shaft 2	7.718	50.71	> 50.71	53.98	10.99
Shaft 3	42.318	77.29	> 77.29	58.42	15.901

An overview of all three shafts:

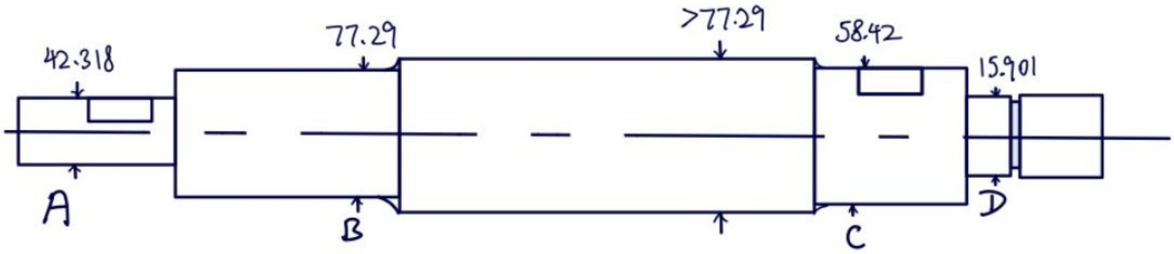


Shaft 1



Shaft 2

Shaft 3



The final value of different locations' diameter of all the three shafts will be adjusted to fit in other components including the gears, bearings and the sprocket. A summary table of the final parameters of all the three shafts will be included in the Engineering Drawing Section.

7. Bearing

7.1 Introduction

A bearing is a device that enables the rotation of a shaft or axle while providing support. It minimizes friction between moving components, ensures stability, and absorbs loads to maintain smooth operation and prevent damage. Various types of bearings are used in mechanical design, as outlined below.

7.2 Types of Bearings

1. Ball Bearing:

Image of ball bearing

A ball bearing is a type of rolling-element bearing that uses small, spherical balls to facilitate smooth rotation between two surfaces, such as a shaft and a housing. The balls, typically made of steel or ceramic, are housed within inner and outer races—concentric rings that guide their movement. A cage or separator keeps the balls evenly spaced, preventing contact and reducing wear. Ball bearings are designed to minimize friction and support both radial and axial loads, making them ideal for high-speed applications like electric motors, automotive wheels, and industrial machinery. Their low friction enhances efficiency and durability, while their compact design allows versatility in mechanical systems. Available in various configurations, such as deep groove, angular contact, and thrust ball bearings, they cater to specific load and speed requirements. Regular lubrication is essential to reduce heat and extend lifespan, ensuring reliable performance in demanding environments.

2. Roller Bearings

:

Image of Roller Bearings

Roller bearings are rolling-element bearings that use cylindrical, tapered, or spherical rollers instead of balls to support and facilitate the rotation of a shaft or axle. These rollers, housed between inner and outer races, distribute loads over a larger contact area, making roller bearings ideal for handling heavy radial loads and moderate axial loads. The rollers are typically guided by a cage to prevent contact and ensure smooth operation. Common types include cylindrical roller bearings, which excel in high-radial-load applications like conveyor systems; tapered roller bearings, used in vehicle hubs for combined loads; and spherical roller bearings, which accommodate misalignment in heavy machinery. Roller bearings offer low friction, high durability, and efficient load distribution, making them suitable for industrial equipment, gearboxes, and wind turbines.

3. Plain Bearings:

Image of Plain Bearings

Plain bearings, also known as sleeve or journal bearings, are simple, non-rolling-element bearings that support a rotating shaft with a smooth, sliding surface. They consist of a cylindrical sleeve, often made of materials like bronze, plastic, or composite polymers, that surrounds the shaft, with no moving parts like balls or rollers. Lubrication, such as oil or grease, is critical to reduce friction and wear between the sliding surfaces. Plain bearings excel in low-speed, high-load applications, such as pumps, hinges, or heavy machinery, due to their ability to handle shock loads and vibrations. They are compact, cost-effective, and operate quietly, making them suitable for space-constrained designs.

Selection of bearing by decision matrix:

Criteria	Weighting(1-5)	Ball Bearing	Roller Bearing	Plain Bearing
Low Cost	5	5	3	3
Less lubrication	5	4	2	2
East to install	3	4	3	2
Heavy load	4	3	4	4
Total score		69	50	47
1: Least suitable. 5: Most suitable				

7.3 Bearing Mounting

Bearing mounting involves installing bearings onto shafts or into housing assemblies, ensuring optimal performance and longevity. There are serval types of bearing mounting: Cold mounting, Hydraulic mounting, Thermal mounting.

1. Cold Mounting:

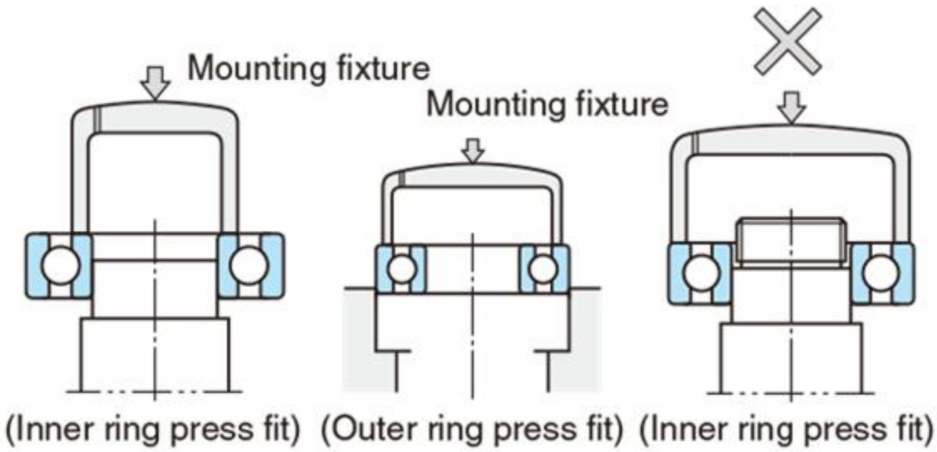


Image for bearing cold mounting

Cold mounting is a technique for installing bearings onto shafts or into housing assemblies by shrinking the bearing through cooling. This facilitates easier fitting. The bearing is cooled using methods like dry ice or liquid nitrogen, causing it to contract. Once cooled, it can be effortlessly placed onto the shaft or into the housing. As the bearing warms up after installation, it expands to its original size, ensuring a tight fit. This method is often used for interference fits or larger bearings.

2. Hydraulic Mounting:

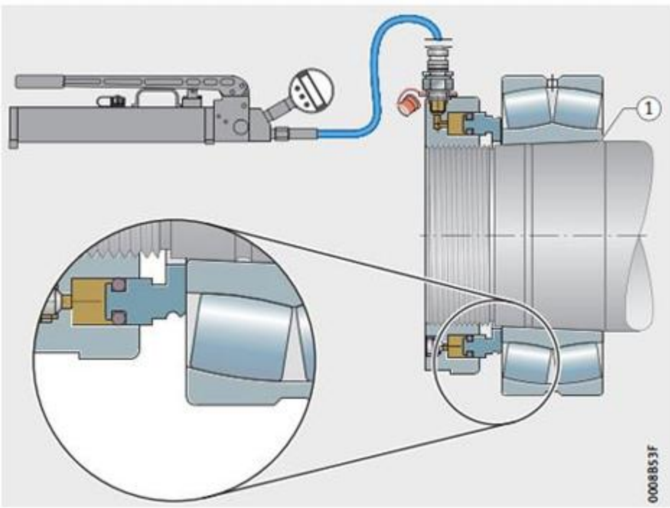


Image for Hydraulic Mounting

Hydraulic tools generate significant force, making them ideal for mounting large bearings with tapered bores. Hydraulic nuts serve as the mounting tool, with pressure applied via oil injectors, hand pumps, or hydraulic units.

3. Thermal Mounting:

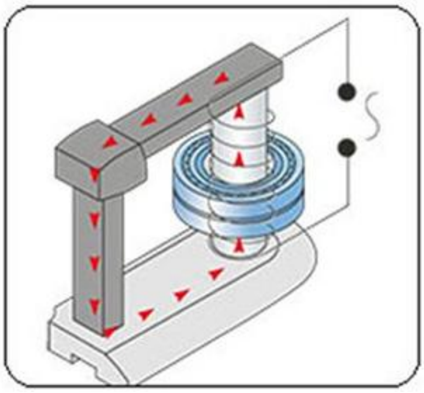


Image for Thermal Mounting

Thermal mounting uses a high-amperage, low-voltage short circuit to heat a bearing, expanding its inner bore diameter for installation. Accurate temperature control and timing are critical for success. However, the need for specialized equipment and careful monitoring can make this method more expensive.

Mounting Selection based on Decision Matrix

Criteria	Weighting(1-5)	Cold Mounting	Hydraulic Mounting	Thermal Mounting
Low Cost	5	5	3	2
Simple to install	4	4	3	2
Not external machine	3	3	2	1
Total score		50	33	21

1:Least suitable 5:Most suitable

8. Key and Keyseat

8.1 Introduction

A shaft key is a small metal piece used to lock the position of a component to the rotating shaft to ensure no slippery or rotation occur between them. The key is inserted into a slot called keyway, which ensured a safe connection for torque transmission between components. Shaft keys are commonly used in mechanical systems for reliable power transmission. There are several types of keys available for mechanical design:

Figure for Different Shaft Key Types

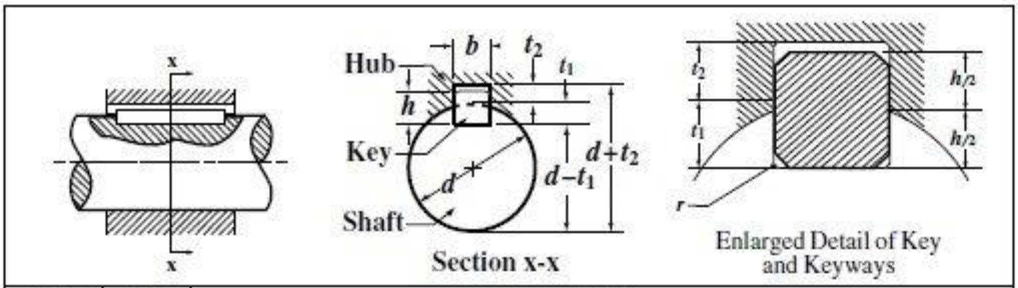
8.2 Decision

Flat keys is the selected type for this project design, as they are the most widely-used type of keys in modern machinery design with a low cost.

8.3 Calculations

The widths and heights are firstly considered for determining the dimensions of the keys and keyseats.

According to the references from this table and calculations,



Shaft: Nominal Diameter d, Over	Shaft: Nominal Diameter d, Up to and Incl	Key: Size, $b \times h$	Keyway Width b: Nominal	Free Fit: Shaft (H9)	Free Fit: Hub (D10)	Normal Fit: Shaft (N9)	Normal Fit: Hub (J_S9)[a]	Close Fit: Shaft and Hub (P9)	Depth Shaft t_1: Nominal	Depth Shaft t_1: Tolerance	Depth Hub t_2: Nominal	Depth Hub t_2: Tolerance	Radius r: Max.	Radius r: Min.
				Tolerances										
Keyways for Square Parallel Keys														
6	8	2 × 2	2	+0.025	+0.060	−0.004	+0.012	−0.006	1.2	+0.1 0	1	+0.1 0	0.16	0.08
8	10	3 × 3	3	0	+0.020	−0.029	−0.012	−0.031	1.8		1.4		0.16	0.08
10	12	4 × 4	4	+0.030 0	+0.078 +0.030	0 −0.030	+0.015 −0.015	−0.012 −0.042	2.5		1.8		0.16	0.08
12	17	5 × 5	5						3		2.3		0.25	0.16
17	22	6 × 6	6						3.5		2.8		0.25	0.16
Keyways for Rectangular Parallel Keys														
22	30	8 × 7	8	+0.036	+0.098	0	+0.018	−0.015	4	+0.2 0	3.3	+0.2 0	0.25	0.16
30	38	10 × 8	10	0	+0.040	−0.036	−0.018	−0.051	5		3.3		0.40	0.25
38	44	12 × 8	12	+0.043	+0.120 +0.050	0 −0.043	+0.021 −0.021	−0.018 −0.061	5		3.3		0.40	0.25
44	50	14 × 9	14						5.5		3.8		0.40	0.25
50	58	16 × 10	16						6		4.3		0.40	0.25
58	65	18 × 11	18						7		4.4		0.40	0.25
65	75	20 × 12	20	+0.052	+0.149 +0.065	0 −0.052	+0.026 −0.026	−0.022 −0.074	7.5		4.9		0.60	0.40
75	85	22 × 14	22						9		5.4		0.60	0.40
85	95	25 × 14	25						9		5.4		0.60	0.40
95	110	28 × 16	28						10		6.4		0.60	0.40
110	130	32 × 18	32	+0.062	+0.180 −0.080	0 −0.062	+0.031 −0.031	−0.026 −0.088	11		7.4		0.60	0.40
130	150	36 × 20	36						12	+0.3 0	8.4	+0.3 0	1.00	0.70
150	170	40 × 22	40						13		9.4		1.00	0.70
170	200	45 × 25	45						15		10.4		1.00	0.70
200	230	50 × 28	50						17		11.4		1.00	0.70
230	260	56 × 32	56	+0.074	+0.220 +0.100	0 −0.074	+0.037 −0.037	−0.032 −0.106	20		12.4		1.60	1.20
260	290	63 × 32	63						20		12.4		1.60	1.20
290	330	70 × 36	70						22		14.4		1.60	1.20
330	380	80 × 40	80						25		15.4		2.50	2.00
380	440	90 × 45	90	+0.087	+0.260	0	+0.043	−0.037	28		17.4		2.50	2.00
440	500	100 × 50	100	0	+0.120	−0.087	−0.043	−0.124	31		19.5		2.50	2.00

[a] Tolerance limits J_S9 are quoted from BS 4500, "ISO Limits and Fits," to three significant figures.

All dimensions in millimeters.

Shaft	**Width (b)**	**Height (h)**	**Manufacturing radius**
1	10	8	0.25-0.4mm
2B	16	10	
2C	16	10	
3	18	11	

To find the lengths of the keyways, the torques T and diameters D of the shaft on them need to be considered.

Starting with finding the shear force F:

$$F = \frac{T}{D}$$

Shaft	Shear force (F) (N)
1	1.568
2B	8.7194
2C	8.1912
3	23.487

Then the length based on the width L_s is

$$L_s = \frac{2FN}{WS_y}$$

And the length based on the width L_b is

$$L_b = \frac{2FN}{HS_y}$$

where safety factor N is taken to be 3. A comparison table can be made as shown:

Shaft	L_s (mm)	L_b (mm)
1	1.92	2.4000
2B	6.67	10.677
2C	6.269	10.030
3	15.978	26.145

The maximum value among these two parameters will be designed length of the keyways. From the table above, it is shown that L_b has a larger value than L_s at every shaft, therefore, L_b will be considered for all keyways design.

As a result, the dimensions of the keyseats are summarized as below:

Shaft	Width (b) (mm)	Height (h) (mm)	Manufacturing radius (r)	L_b (mm)
1	10	8	0.25-0.4mm	2.400
2B	16	10		10.68
2C	16	10		10.03
3	18	11		26.15

9. Shaft Coupling

9.1 Introduction

A shaft coupling is a mechanical component that joins two shafts to transmit torque and rotational motion. It accommodates misalignment, dampens vibrations, and is available in various types to meet diverse application needs. Correct installation and regular maintenance are essential for dependable performance.

9.2.1 Types of Coupling

9.2.2 Cross Type/Universal Couplings:

Image for Cross Type/Universal Coupling

A cross-type or universal coupling, also known as a universal joint (U-joint), is a mechanical device designed to connect two shafts that are not aligned, allowing torque and rotational motion to be transmitted at variable angles. It consists of a cross-shaped component with four arms, each fitted with bearings, connecting two yokes attached to the shafts. This design enables flexibility, accommodating angular misalignment (typically up to 15–30 degrees) and slight axial or lateral offsets. Commonly made from steel or alloys for durability, universal couplings are widely used in automotive drivelines, industrial machinery, and agricultural equipment. They efficiently transfer power while absorbing minor vibrations and shocks. However, they may produce non-uniform motion at higher angles, requiring careful design consideration. Proper lubrication and periodic maintenance are critical to prevent wear and ensure smooth operation, making them reliable for applications demanding flexible power transmission.

9.2.3 Gear coupling:

Image for Gear coupling

A gear coupling is a robust mechanical device designed to connect two shafts, transmitting high torque and rotational motion while accommodating minor misalignments. It consists of two hubs with external gear teeth that mesh with internal teeth on a sleeve or flange, forming a flexible yet strong connection. Typically made from steel or alloy materials, gear couplings are ideal for heavy-duty applications like industrial machinery, steel mills, and power generation systems. They can handle angular misalignment (up to 1–2 degrees), parallel offset, and axial movement, while effectively damping vibrations and shock loads. The coupling's teeth are often lubricated to reduce friction and wear, ensuring longevity. Proper installation, alignment, and regular maintenance, including lubrication, are essential for reliable performance. Gear couplings are valued for their high torque capacity, durability, and ability to maintain efficient power transmission in demanding environments.

9.2.4 Oldham Couplings:

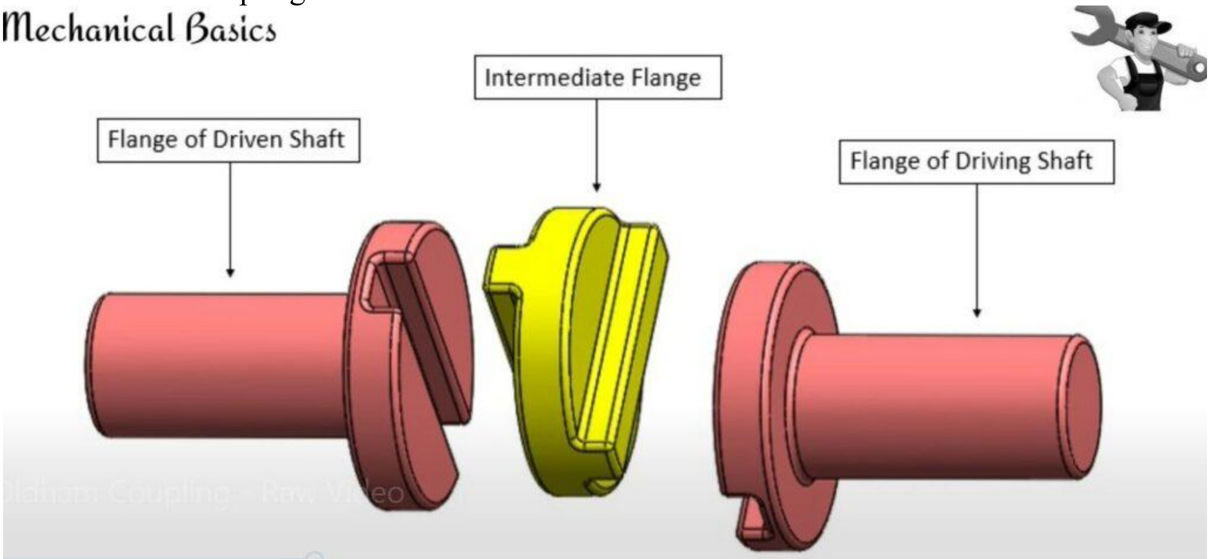


Image for Oldham couplings

An Oldham coupling is a flexible mechanical device used to connect two shafts, transmitting torque and rotational motion while accommodating lateral misalignment. It consists of three components: two hubs, each attached to a shaft, and a central floating disc with perpendicular tongues or slots. The tongues engage with corresponding slots on the hubs, allowing smooth power transfer even when shafts are offset. Made from materials like aluminum, steel, or plastic for the disc, Oldham couplings are used in applications such as servo motors, pumps, and light machinery. They provide low backlash, absorb minor vibrations, and are easy to install. However, they are less suited for high angular misalignment or heavy torque loads. Regular maintenance, including disc inspection for wear, ensures longevity. Oldham couplings are valued for their simplicity, precision, and effectiveness in misalignment compensation.

9.3 Coupling selection based on decision matrix

Criteria	Weighting(1-3)	Universal Coupling	Gear Coupling	Oldham Coupling
Low Cost	3	2	3	2
Misalignment	2	1	2	2
Less Fluctuation	2	2	2	1
Size	3	1	3	3
Total score		15	26	21
1:Least suitable 3: Most suitable				

The above matrix shown the Oldham coupling is suitable for the requirement gearbox as it can accommodate high degrees of misalignment, ease of installation, and cost-effectiveness, and if the torque and speed requirements are moderate. Its ability to absorb vibrations and maintain low backlash ensures reliable operation of the gearbox's components.

10. Retaining Ring

10.1 Introduction

A retaining ring, or snap ring (also known as a circlip), is a small metal ring designed to fasten components on a shaft or inside a housing, preventing axial displacement. It comes in internal and external forms and is made from sturdy materials such as steel. These rings play a key role in maintaining component stability and avoiding unintended movement during operation. They are fitted or removed using specific tools like pliers, and selecting the appropriate size and type is critical for a secure hold.

10.2 Types of retaining ring

10.2.1 External Retaining Ring

Image for External Retaining Ring

External retaining rings, also known as external snap rings or circlips, are metal fasteners designed to secure components on a shaft by fitting into a groove on its outer surface. Typically made from durable materials like carbon or stainless steel, they provide robust resistance to axial movement, ensuring components remain stable during operation. These rings are circular with open ends, featuring lugs or holes for installation and removal using specialized pliers. External retaining rings come in various sizes and styles, such as tapered or constant section designs, to suit different applications, from automotive to industrial machinery. Proper selection of size, material, and groove specifications is crucial for a secure fit and optimal performance. Their ease of installation, cost-effectiveness, and ability to withstand high thrust loads make them essential in assemblies requiring reliable component retention without complex fastening systems.

10.2.2 Internal Retaining Ring

Image for Internal retaining ring

Internal retaining rings, also known as internal snap rings or circlips, are metal fasteners designed to secure components within a housing or bore by fitting into an internal groove. Crafted from durable materials like carbon or stainless steel, these rings prevent axial movement, ensuring components remain stable during operation. Their circular shape with open ends includes lugs or holes for installation and removal using specialized pliers. Available in various sizes and designs, such as tapered or constant section, internal retaining rings cater to diverse applications, including automotive, aerospace, and industrial equipment. Selecting the correct size, material, and groove specification is critical for a secure fit and reliable performance. These rings offer a cost-effective, easy-to-install solution for component retention, capable of handling significant thrust loads without requiring complex fastening systems, making them indispensable in assemblies where stability and precision are paramount.

10.3 Retaining Ring Selection

For ease of installation and compatibility with the gearbox design, external retaining rings have been chosen. A total of seven retaining rings will be employed within the gearbox to secure the positions of pinion 1, gear 1, and bearings 1, 2, 3, 4, and 6. The free diameter, defined as the inner diameter of the retaining ring, must be slightly smaller than the groove diameter to ensure a tight and secure fit within the groove. The parameters for these seven retaining rings are summarized as follows.

	Component to locate	Shaft Diameter (mm)	Retaining Ring		Groove	
			Free Diameter (mm)	Thickness(mm)	Diameter(mm)	Width(mm)
1	Pinion 1	42.318	39.8	1.5	39.8	1.5
2	Gear 1	53.98	50.7	2	50.7	2
3	Bearing 1	31.3	29.4	1.3	31.1	1.3
4	Bearing 2	31.3	29,4	1.3	31.1	1.3
5	Bearing 3	35	32	1.3	34	1.3
6	Bearing 4	77.29	72.65	2	77	2
7	Bearing 6	58.42	54.91	2	58.2	2

11. Motor

11.1 Introduction

An electric motor is a machine that converts electric energy into mechanical energy. It composes of a rotor and stator, and the armature often rounded on the rotor. In our design, a motor is used to receive electric power and drive our gearbox, finally drives the conveyor.

11.2 Requirements

In our system, from the previous calculation, the input power is about 9.9kw. This indicates the minimum power output of the motor, and for longevity reasons, we will select a motor which has a slightly higher power output than 9.9kw. If we select a motor exactly has the same power output, it means the motor will always work at 100% load, which will cause heat generation and premature aging of insulation materials, thus causing safety hazards. Our final selection is 11kw. Another requirement for the motor is rotation speed, which is set by the requirement at 1440rpm. There are many types of electric motors, like DC motor, AC motor, three-phase motor and so on; and we will discuss and compare different motors, find the one that suits our application the most.

11.3 Motor selection

AC vs. DC

AC and DC motors are two primary types of electric motors used to convert electrical energy into mechanical motion, each with distinct characteristics suited to different applications. AC motors, which means powered by alternating current, are widely used in industrial settings due to their simplicity, durability, and low maintenance. AC motors are particularly favored for constant-speed applications like conveyor belts, pumps, and fans. On the other hand, DC motors, which means powered by direct current, provide precise speed and torque control over a wide range. This makes them ideal for variable-speed tasks and applications requiring frequent starts and stops, such as electric vehicles, robotics, or cranes. However, DC motors are typically more expensive and require more maintenance due to brushes and commutators.

In our system, AC motors are generally preferred for their robustness, energy efficiency, and ease of integration with industrial power supplies and gearboxes. Since we do not need precise control of speed and torque, we do not choice DC motors.

Three phase vs. single phase

Single-phase and three-phase power systems are two methods of delivering electrical energy. Single-phase power uses only one alternating voltage cycle and is commonly found in residential and light commercial settings. Single-phase motors are typically smaller, less efficient, and used in low-power applications such as household appliances or small machinery. In contrast, three-phase power delivers electricity in three alternating cycles, providing a constant and balanced power flow. This results in higher efficiency, smoother operation, and the ability to power large motors and equipment with reduced conductor size and energy loss. Three-phase motors are more compact, powerful, and durable, making them ideal for heavy-duty industrial applications like conveyor systems or large pumps. While three-phase systems require more complex infrastructure, their performance benefits make them standard in industrial and high-load environments.

Thus, for our system, three-phase motor is preferred for its efficiency and power, while single-phase is more suitable for simple and small loads.

Synchronous vs Induction

Induction motors and synchronous motors are the two main types of AC motors, each with unique operating principles and advantages. Induction motors, also known as asynchronous motors, are the most common type used in industrial applications. They work by inducing current in the rotor through electromagnetic fields generated by the stator, which means no electrical connection to the rotor is needed. This makes them rugged, cost-effective, and low maintenance. However, induction motors run at a speed slightly less than the synchronous speed, which can lead to minor inefficiencies in precision applications. In contrast, synchronous motors operate at a constant speed that is exactly equal to the supply frequency. They offer high efficiency and are well-suited for applications requiring precise speed control. However, they are more complex, require a separate DC excitation system, and are generally more expensive.

For our conveyor belt systems, induction motors are typically favored for their simplicity and reliability, while synchronous motors may be chosen for specialized, high-efficiency operations.

In conclusion, we will use an AC, three-phase induction motor, which rated at 11kw and 1440rpm.

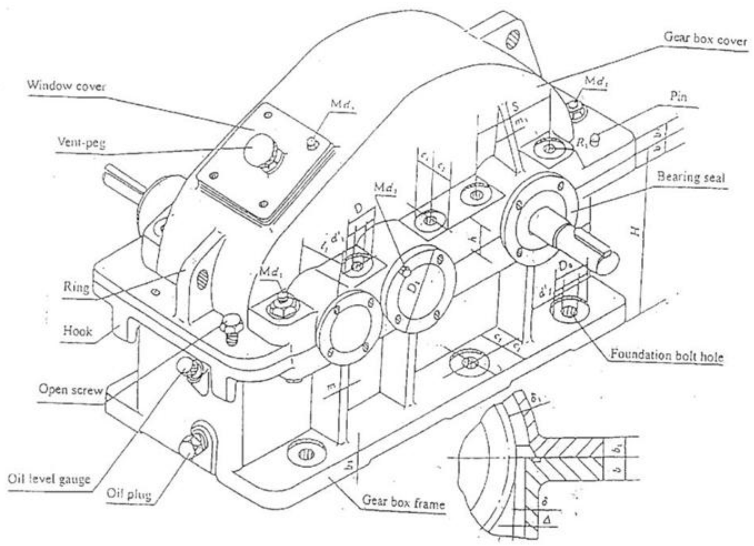
12. Housing Design

12.1 Introduction

The gearbox, designed for a mining operating system, requires a protective housing due to its outdoor use. This housing shields internal components, extending the gearbox's lifespan and improving performance. Key considerations for the housing design include material strength, environmental and weather compatibility, chemical corrosion resistance, dimensional limits, and cost.

12.2 Housing Feature

The basic feature of the housing design is shown below:



12.3 Calculation

i. Wall Thickness of the gear box frame

$\delta = 0.03a2 + \Delta 1 \geq 8$

($\Delta 1$= 1, 3, or 5 for single, double, or triple reducer respectively; and a2 is the center distance of the second pair of gears)

$\Delta 1 = 3$ (for double reducer)

a2 = center distance of the second pair of gears $= \frac{D_{gear3} + D_{gear4}}{2}$

$= \frac{130+400}{2} = 265\text{mm}$

Therefore, $\delta = 0.03a2 + \Delta 1 = 0.03 \times 265 \text{ mm} + 3 = 10.95 \text{ mm}$

$\therefore$ We choose $\delta = 12$ mm.

ii. Wall Thickness of the gearbox cover

δ1= (0.8 ~ 0.85) δ ≥ 8

We assume 0.8 for our calculation,

δ1 = (0.8)(10.95) = 8.76 mm

∴ We choose δ1 = 9 mm.

iii. Housing Thickness

Upper plate of gear box frame thickness b: b = 1.5δ = 1.5(11) = 16.5 mm

Bottom plate of gear box cover thickness b1: b1 = 1.5δ1 = 1.5(9) = 13.5 mm

Bottom plate of gear box cover thickness b2: b2 = 2.5δ = 2.5(11) = 27.5 mm

Rim support of gear box frame thickness m: m ≥ 0.85δ = 0.85(11) = 9.35

∴ We choose m = 10 mm.

Rim support of gear box cover thickness m1: m1 ≥ 0.85δ1 = 0.85(9) = 7.65

∴ We choose m1 = 8 mm.

iv. Diameter of foundation bolts

Number of foundation bolts n = 6

If $a_1 + a_2 =$:	~350	~400	~600	~750 mm
bolt dia. (d_f):	16	20	24	30

where a1, a2 are the center distances of the first and second pair of gears.

$$a_1 + a_2 = \left(\frac{80 + 320}{2}\right) + \left(\frac{130 + 400}{2}\right) = 200 + 265 = 465mm \approx 400mm$$

Therefore, df = 20 mm.

V. Diameter of other bolts

Diameter of gear box cover side bolts d1:

d1= 0.75df = 0.75(20) = 15 mm

∴ We choose d1 = 16 mm.

Diameter of gear box cover back bolt d2:

d2 = (0.5 ~ 0.6)df = (0.5)(20) = 10 mm

Diameter of bearing cover bolt d3:

d3 = (0.4 ~ 0.5)df = (0.5)(20) = 10 mm

Diameter of window cover bolt d4:

d4 = (0.3 ~ 0.4)df = (0.3)(20) = 6 mm

Vi. Diameter of pins

d = (0.7 ~ 0.8) d2

We assume 0.8 for our calculation,

d = (0.8)d2 = (0.8)(10) = 8 mm

Vii. Distance between gear and wall of box or between gears

$\Delta \geq 1.2\delta$ (or 10~15)

$\Delta \geq 1.2(11) = 13.2$ mm

∴ We choose $\Delta = 14$ mm

Viii. Distance between the wall and the edge

Bolt dia.	6	8	10	12	14	16	20	24	30
Bolt hole dia.	7	9	11	13.5	15.5	17.5	22	30	40
D	13	18	22	26	30	33	40	60	85
C_{1min}	12	15	18	20	22	24	28	35	50
C_{2min}	10	12	14	16	18	20	24	30	50

$l_1 = C_1 + C_2 + (5 \sim 8)$

We assume I = C1 + C2 + 5

where df is the diameter of foundation bolts

d1 is the diameter of gear box cover side bolts

d2 is the diameter of gear box cover back bolts

	df	d1	d2
Bolt diameter (mm)	20	16	10
Bolt hole diameter (mm)	22	17.5	11
Spot face D	40	33	22
C1min	28	24	18
C2min	24	20	14
I	57	49	37

Summary	
Parameters	Result
Wall Thickness of the gear box frame δ	12mm
Wall Thickness of the gearbox cover δ1	9mm
Housing Thickness: Upper plate of gear box frame thickness b Bottom plate of gear box cover thickness b1 Bottom plate of gear box cover thickness b2 Rim support of gear box frame thickness m Rim support of gear box cover thickness m1	 16.5mm 13.5mm 27.5mm 10mm 8mm
Number of foundation bolts n	6mm
Diameter of foundation bolts df	20mm
Diameter of gear box cover side bolts d1	16mm
Diameter of gear box cover back bolt d2	10mm
Diameter of bearing cover bolt d3	10mm
Diameter of window cover bolt d4	6mm
Diameter of pins	8mm
Distance between gear and wall of box or between gears Δ	14mm
Bolt hole diameter of df Bolt hole diameter of d1 Bolt hole diameter of d2	22mm 17.5mm 11mm
Spot face D of df Spot face D of d1 Spot face D of d2	40mm 33mm 22mm
C1min of df C1min of d1 C1min of d2	28mm 24mm 18mm
C2min of df C2min of d1 C2min of d2	24mm 20mm 14mm
I of df I of d1 I of d2	57mm 49mm 37mm

12.4 Material selection

Here are the design consideration and selection requirement of the housing material for our gear box:

1. Material strength

2. Compatibility to the operating environment and weather

3. Chemical corrosion resistance

4. Dimensional constraints

5. Cost

We consider and translate the transition table for our material selection of the housing

Function	Housing of the gearbox - Cover and provide sufficient protection to the inner components of the entire gear box
Objective	Minimize cost
Constraints	Sufficient strength without failure (functional constraint) - All dimensions are specified (geometric constraint)
Free Variables	Choice of material

Let mild steel as our reference to the relative cost:

$$C_R = \frac{cost\ per\ kg\ of\ the\ material}{cost\ per\ kg\ of\ the\ mild\ steel\ rod}$$

where CR is the relative cost of the material.

Objective Function

$$C = mC_R = \rho V C_R = \rho A L C_R$$

Where C is the cost of the material

ρ is the density of the material

V is the volume of the material

Constraint Function

$$\sigma_f \geq \frac{F^*}{A} \rightarrow A \geq \frac{F^*}{\sigma_f}$$

where $F*$is allowable loading given by safety regulation (failure strength)

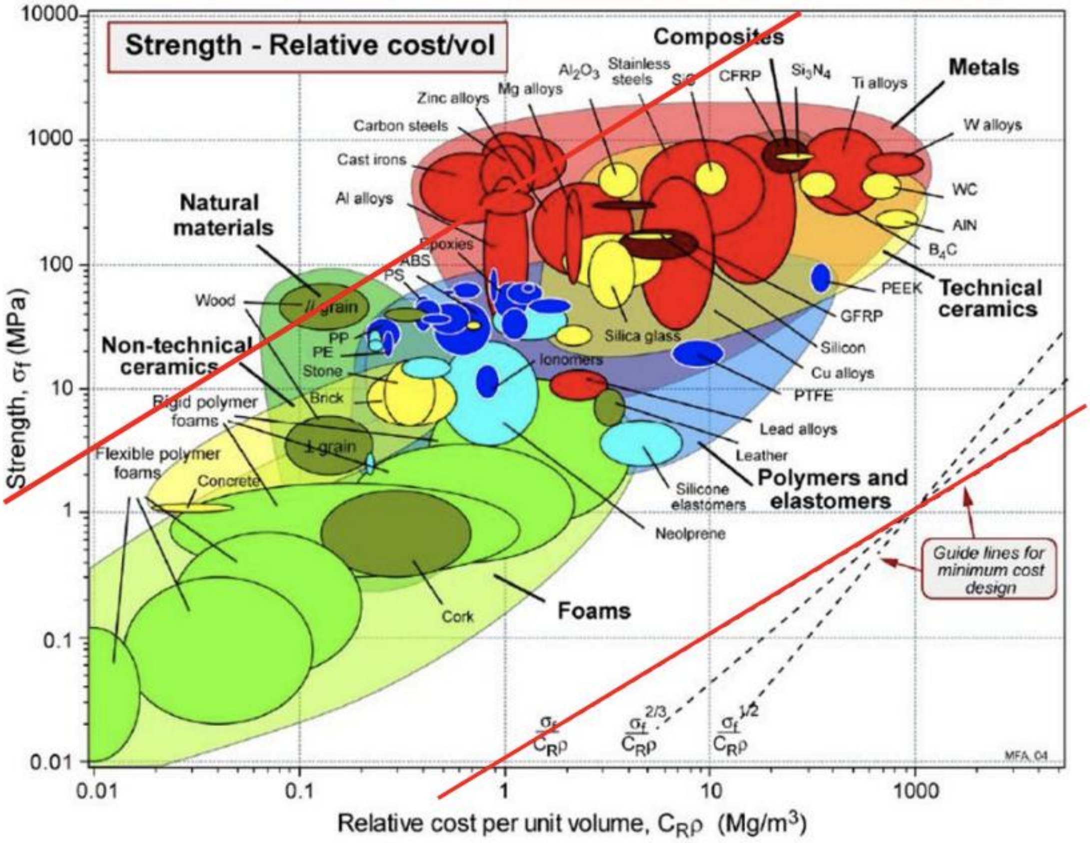
Combining the objective function and constraint function,

$C \geq \frac{F^*}{\sigma_f} \mathrm{L} \rho C_R$

$$C \geq (F^*)(L)(\frac{C_R\rho}{\sigma_f})$$

That is, we need to maximize the material indices

$$M_{p1} = \frac{\sigma_f}{C_R\rho}$$



Decision Matrix

	Weighting Factor	Cast Iron	Carbon Steel	Al Alloy
Strength	x3	2	3	2
Cost	x1	3	1	2
Corrosion resistance	x3	1	3	3
Compatibility (outdoor operation condition)	x2	1	2	2
Total Score		14	23	19

_Final decision: Carbon Steel will be our material for our housing design.

12.5. Casting method for the housing (Sand casting vs die casting)

Casting involves pouring molten metal into a mold to create parts. Compared to milling, milling and machining generate more material waste due to the hollow structure of a gear box. Thus, casting is a more cost-effective approach for producing gear box housings. Several common casting methods are listed below for consideration:

Metal casting can be generally categorized into 2 big groups based on the basic nature of molding design, Expendable mold casting and Non-expendable/Permanent mold casting.

1. Expendable mold casting The mold for the expendable mold casting is created for single or temporary used. The expendable mold is commonly made of sand, plaster and ceramic. It will be destroyed and removed after the casting process is completed and taking the cast product out. (e.g., sand casting, plaster casting, ceramic casting)

2. Non-expendable/Permanent mold casting The mold for the nonexpendable mold casting (also known as permanent mold casting) is created for multiple casting and permanent used. It will not be destroyed when taking the cast product out and can be reused after every production cycle. Therefore, durable materials such as cast iron, steel, or graphite are commonly used as the material of the nonexpendable mold casting mold. (eg. Die casting)

Advantages of die casting

1.Smoother surface finishing

2. Higher production rate

3. Higher dimensional accuracy

4. Able to produce relatively large or small parts

5. Better mechanical properties (E.g. Higher strength and hardnesses of the final product, higher durability)

So we will choose die casting as the casting method for the housing

13. Lubrication

13.1 Introduction

Lubrication indicates that the use of lubricant to reduce friction and wear and tear in a contact between two surfaces. In our designed gearbox, the main source of friction comes from the contact between gears. If the proper lubrication method is not applied, it might cause the production of a large amount of heat; causing early fatigue; shortened the lifespan of the gear box and so on. Furthermore, the bearings are also a source of friction, so they also need proper lubrication. If they are not lubricated properly, they may fall prematurely, causing gears no longer line properly, and might bring disastrous failure to the overall gearbox. Thus, our gearbox requires proper lubrication, and we will discuss the lubrication method which suits our gearbox the best.

13.2 Types of lubrication method

Oil splash

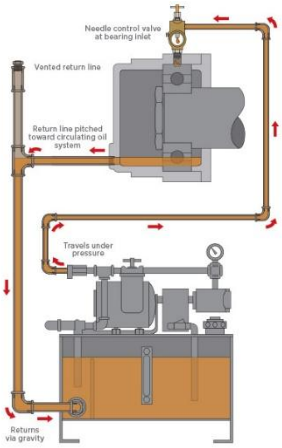
Oil Splash Lubrication is one of the most widely used methods for enclosed gearboxes, particularly in medium-speed, moderate-load applications like conveyor systems. In this method, lower gears are partially submerged in an oil sump. As they rotate, they splash oil onto adjacent gears and bearings, providing lubrication and some degree of cooling. The simplicity of the design makes it cost-effective, easy to maintain, and less prone to failure. However, proper oil level is critical—too low causes inadequate lubrication, while too high can lead to churning losses. It is not ideal for high-speed or heavily loaded systems.

Oil drip

Oil drip lubrication is a controlled lubrication method where oil is supplied drop by drop onto critical components. In this system, oil drops through adjustable drip nozzles to components like gear meshes or bearings. This system ensures a consistent, minimal quantity of oil is delivered directly to the wear surfaces, reducing waste and overheating. It is often gravity-fed from a reservoir and can be manually or automatically regulated. While not suitable for high-speed or fully enclosed gearboxes, oil drip systems are effective for low to moderate speed open gears operating.

Forced circulation (oil pump)

Forced Circulation Lubrication involves actively pumping oil through a closed-loop system that includes filters, coolers, and targeted spray or injection nozzles. This method offers precise control over oil flow, pressure, and temperature, making it ideal for high-load, high-speed gearboxes. The circulating oil not only lubricates but also removes heat and contaminants, extending the life of components. While more complex and expensive than passive systems, it enables real-time monitoring and external conditioning of the lubricant. However, as it also needs extra power to rotate the oil pump, which needs to be counted in while designing; and it also needs more frequent and complicated maintenance than other methods.



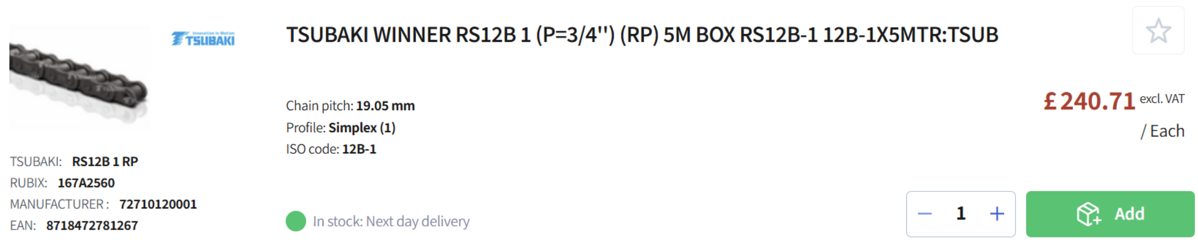
Selection and our considerations

For our design, the conveyor belt system is used for transporting bricks, which indicates it might be used in rough conditions, like in a factory or construction site. In these conditions, durability is one of the most important considerations in the design because during the operation, if it breaks down, the whole working process might be halted, causing thousands of dollars of loss. While being durable, the maintenance process keeps as simple as possible, because complicated processes can cause a significant increase in the labor cost. After all, we need to pin the overall operating cost at a relative low level. After considering all those factors, we decided to choose the oil splash lubrication method, because it can assure proper lubrication for our gearbox while maintaining a low fabrication cost and maintenance cost. The detailed decision matrix is shown below.

Criteria	Oil splash	Oil drip	Forced circulation
Durability (5)	4	4	4
Simplicity (5)	5	4	1
Maintenace (4)	5	3	2
Quietness (1)	3	3	2
Cooling effect (2)	2	3	4
marks	72	61	43

14. Market Research

Chain drive



https://uk.rubix.com/en/bs-gt4-winner-standard-roller-chain/p-G1375006319

Sprocket 1

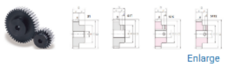
https://uk.rubix.com/en/sprocket-taper-bore-type-simplex/p-G1408006085

Sprocket 2

https://uk.rubix.com/en/sprocket-pilot-bore-simplex/p-G1408012940

Gear:

Gear1



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Documentation

PDF Catalog	View/Download
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KHK SSG4-20J20, Module 4, 20 Tooth, Ground Carbon Steel Spur Gears

Ground Spur Gears (SSG)

The SSG Ground Spur Gears are made from carbon steel, are induction hardened and the teeth are ground finished. Since the heat treatment is applied only to the tooth area, secondary operations can be preformed on the hub and bore. J Series configurations of the SSG Ground Spur Gears are available.

Choose a CAD format

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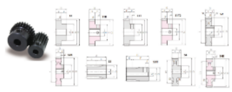
Please note that all J-Series, F-Series, & H-Series products are made-to-order, noncancellable, and non-returnable.

Price: $192.90 *(See below for quantity breaks.)*

Gear2



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Documentation

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KHK SS5-20J50, Module 5, 20 Tooth, Carbon Steel Gears

Spur Gears (SS)

The SS Spur Gears are made from carbon steel and are NOT hardened. However, the SS-H Spur Gears do have the gear teeth hardened. These Spur Gears are economically priced, general usage gears, with a large selection of modules and numbers of teeth. These Spur Gears are easily customizable and J Series configurations of these Spur Gears are also available.

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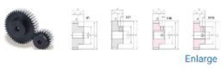
Price: $217.40 *(See below for quantity breaks.)*

Gear3



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Documentation

PDF Catalog	View/Download
Selection Guide / Technical Info.	View/Download
Detailed Drawing	View/Download

KHK SSG4-44J30, Module 4, 44 Tooth, Ground Carbon Steel Spur Gears

Ground Spur Gears (SSG)

The SSG Ground Spur Gears are made from carbon steel, are induction hardened and the teeth are ground finished. Since the heat treatment is applied only to the tooth area, secondary operations can be preformed on the hub and bore. J Series configurations of the SSG Ground Spur Gears are available.

Choose a CAD format

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Please note that all J-Series, F-Series, & H-Series products are made-to-order, noncancellable, and non-returnable.

Price: $470.70 *(See below for quantity breaks.)*

Gear total cost:192.9+2x217.4+470.7=HKD$1098.4

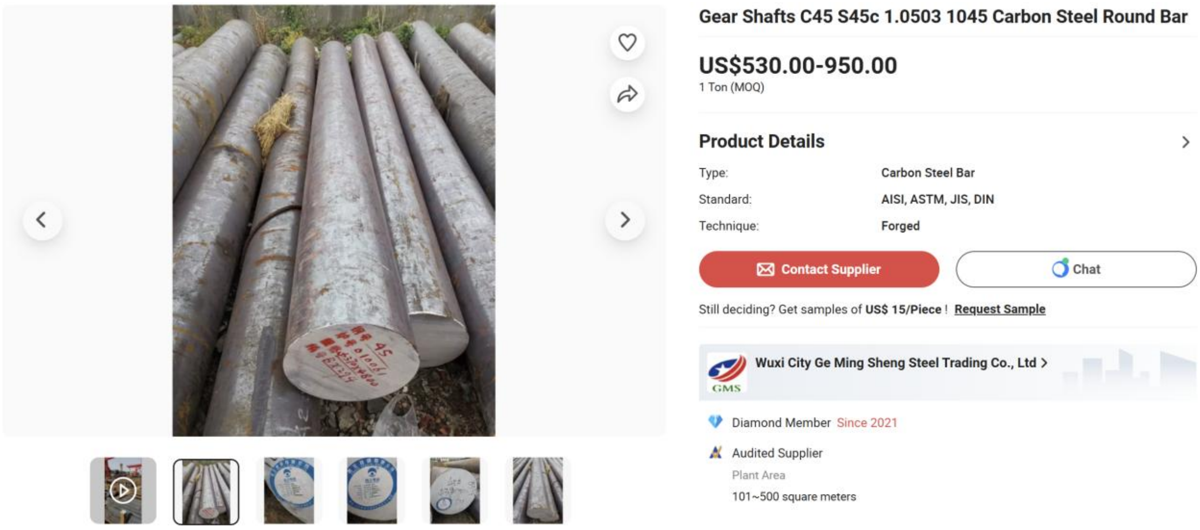
Source:

https://www.khkgears.us/catalog/product/SSG4-20J20/

https://www.khkgears.us/catalog/product/SS5-20J50

https://www.khkgears.us/catalog/product/SSG4-44J30/

Shaft 1, 2 & 3



https://wxgms1.en.made-in-china.com/product/eOwfEdqcnmkn/China-Gear-Shafts-C45-S45c-1-0503-1045-Carbon-Steel-Round-Bar.html

Mass of shaft 1, 2 & 3

Mass properties of shaft1
Configuration: Default
Coordinate system: -- default --

Density = 0.01 grams per cubic millimeter

Mass = 904.14 grams

Volume = 115915.70 cubic millimeters

Surface area = 20419.48 square millimeters

Center of mass: (millimeters)
X = 0.00
Y = 0.00
Z = 104.10

Mass properties of shaft2
Configuration: Default
Coordinate system: -- default --

Density = 0.01 grams per cubic millimeter

Mass = 3055.21 grams

Volume = 391693.83 cubic millimeters

Surface area = 40007.03 square millimeters

Center of mass: (millimeters)
X = 0.00
Y = -0.10
Z = 113.74

Mass properties of shaft3
Configuration: Default
Coordinate system: -- default --

Density = 0.01 grams per cubic millimeter

Mass = 6796.26 grams

Volume = 871314.84 cubic millimeters

Surface area = 64898.57 square millimeters

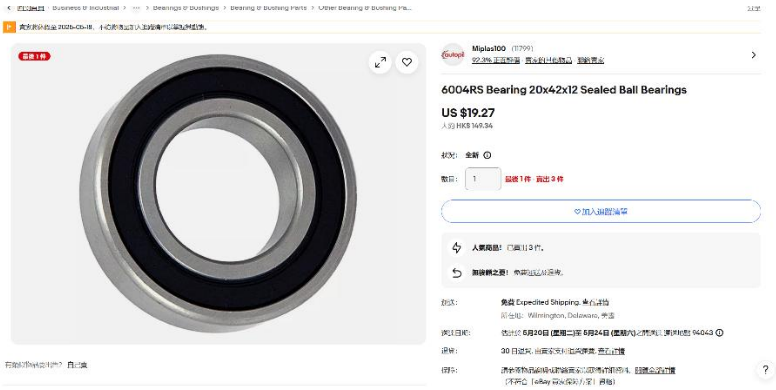
Center of mass: (millimeters)
X = 0.00
Y = -0.10
Z = 107.08

Total mass of all three shafts: 904.14 +3 055.21 + 6796.26 = 10755.61g = 10.75561kg

Price: US$95 = HK$722

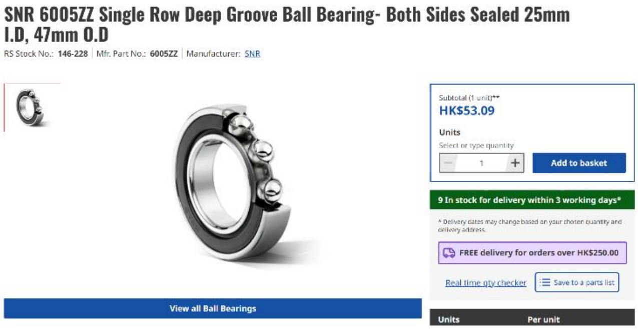
<u>Bearing:</u>

Bearing A

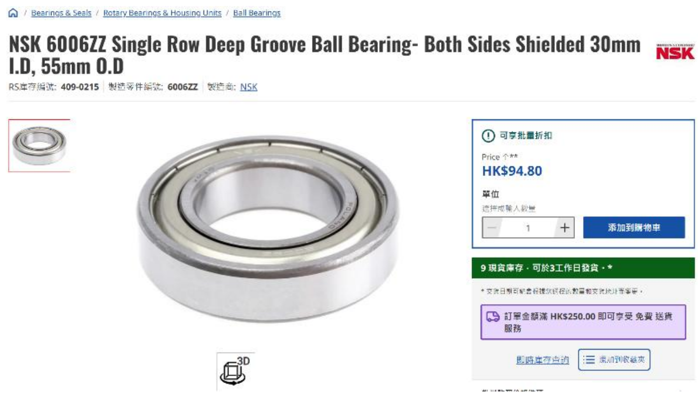


:

Bearing B



Bearing C



Bearing total cost:

150.31x2+53.09x2+94.8x2 = HKD596.39

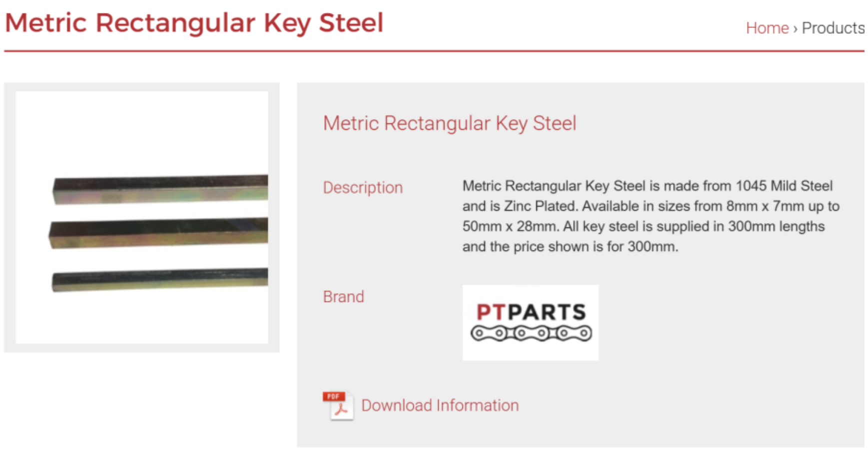
Source:

https://hken.rs-online.com/web/p/ball-bearings/0146228?srsltid=AfmBOopdaeCyQEBCC1iTVZlEdzJxuExjO3LVJGlmqIwhsVo9fiC19zWF

https://www.ebay.com.hk/itm/153740838016

https://hkcn.rs-online.com/web/p/ball-bearings/4090215

Key Steel



KEY10X8R	10mm x 8mm Rect Key Steel 300mm Length	AU$7.39	Each	✔	0
KEY16X10R	16mm x 10mm Rect Key Steel 300mm Length	AU$14.40	Each	✔	0
KEY18X11R	18mm x 11mm Rect Key Steel 300mm Length	AU$16.25	Each	✔	0

Price	7.39+14.4+16.25=AU$38.04 =HK$190.86

Source:

https://www.ptparts.com.au/products/category/ALL/KEY-8X7R--metric-rectangular-key-steel

Spacers:

Price for 1: HK$4.675

https://detail.tmall.com/item.htm?app=chrome&bxsign=scd_hmzgsQgSsLvrbMtqO7L_kFoVgv9ZnUPj_eYJNWTB2qci4XblCRJjUECC8B-0zKXZp5af1KmATKHJKo0piQd_d1JD8ROHtXOcTdRYGIhGSLPxl3bmcsqEwQmf-2Vn_EkwsGYQAvLb3iGFr3X_m0COQ&cpp=1&id=757900639157&price=5.5&shareUniqueId=31458305701&share_crt_v=1&shareurl=true&short_name=h.6MLX6Hb4NTMG1lW&sourceType=item&sp_tk=WXRHRlY1bUlLd3E%3D&spm=a2159r.13376460.0.0&suid=c64e19c3-99ac-4153-8b1e-358bede77949&tbSocialPopKey=shareItem&tk=YtGFV5mIKwq&un=6d0ea3af51b30d0c5eb25bef9ba6fbc9&un_site=0&ut_sk=1.YrneSSiD748DANqt20bdCXsH_21646297_1746458751445.Copy.1&wxsign=tbwPu5wvr9HAZAoAuYHda5ijDQK-vSfT-NpgcBJxp5oZ0Rmm1wkQ777FLVcj1qDujKVKl4tmcGlwNlkjlALepClPsxgU_z5blUgazvHdO02iUzrL6h_syOeenBsORlj8ZsoOAVhwR5a0M9SOVXI3q97ag&skuId=5398085979034

Shaft coupling:

Price:

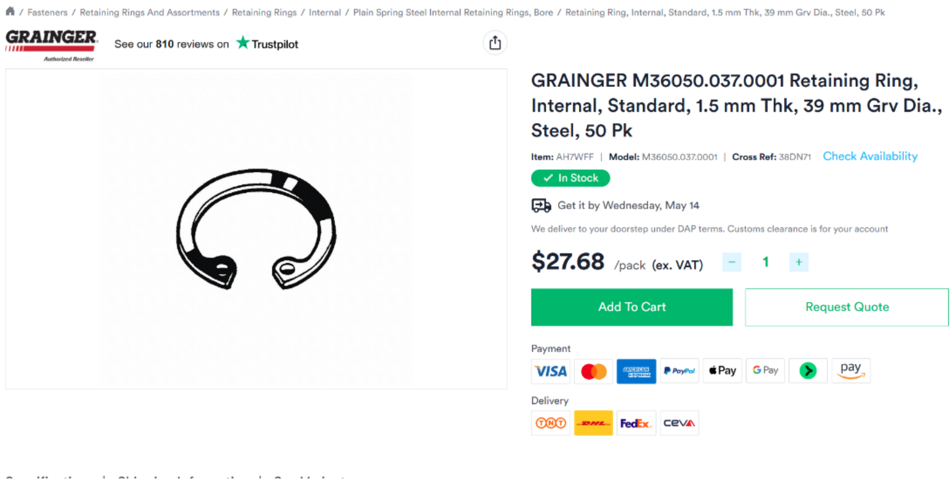
165.54USD = HKD$1291.2

Sources:

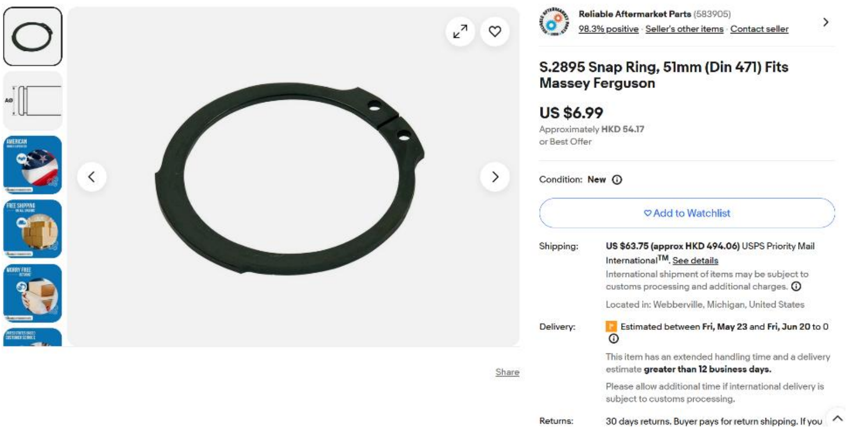
https://www.ruland.com/mbcl41-20-20-a.html

Retaining ring:

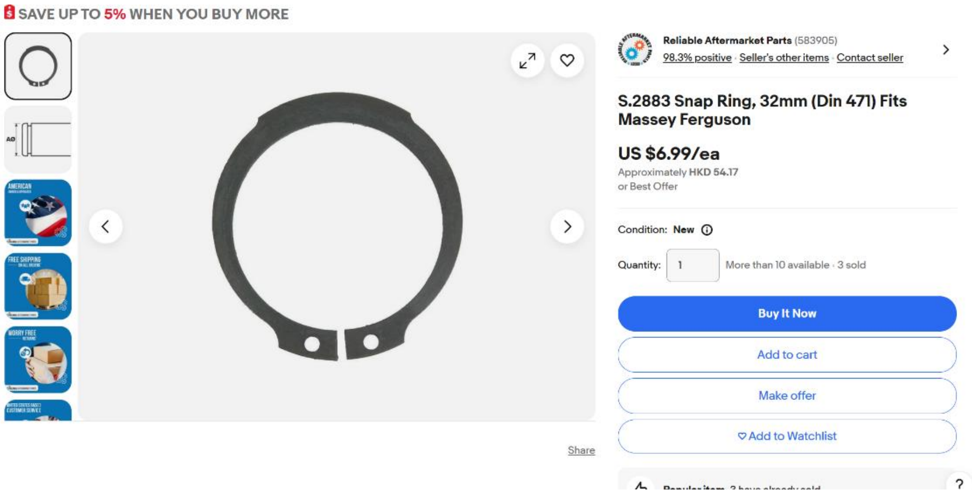
Retaining ring1



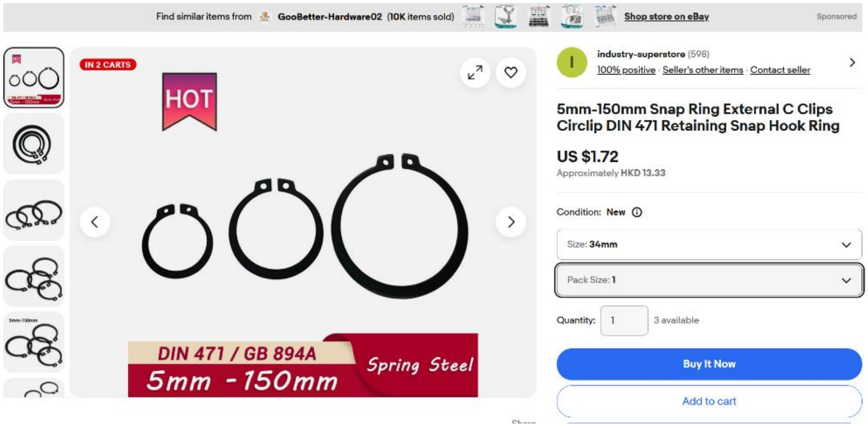
Retaining Ring2



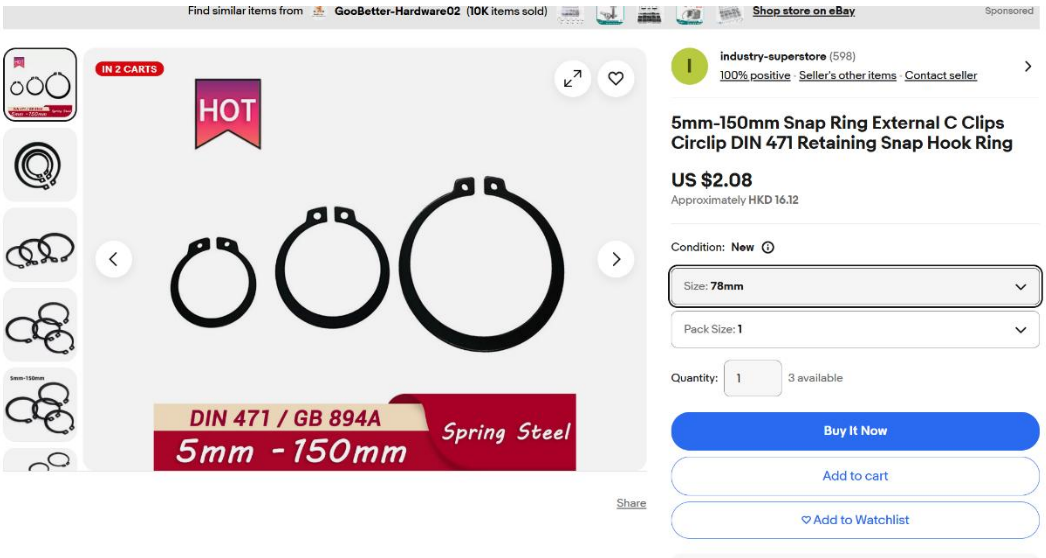
Retaining Ring3 and 4



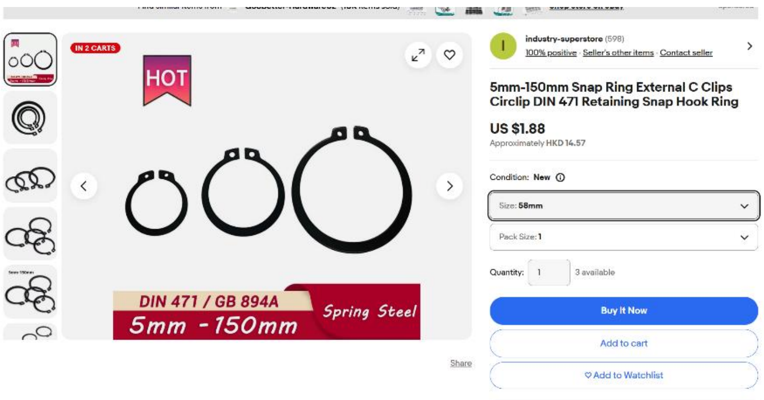
Retaining Ring 5



Retaining Ring 6



Retaining Ring 7



Retaining Ring total cost:

27.68+54.17+54.17x2+13.33+16.12+14.57=HKD$234.21

Sources:

https://www.raptorsupplies.com/pd/grainger/m36050-037-0001

https://www.ebay.com/itm/235511550176?chn=ps&norover=1&mkevt=1&mkrid=711-167022008381 2&mkcid=2&itemid=235511550176&targetid=295607582760&device=c&mktype=pla&googleloc=

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https://www.ebay.com/itm/306135184233?_skw=Snap+Ring%2C+34mm&itmmeta=01JTG2P76NY7TBN6VXV26AEQ9K&hash=item4747143369:g:WSUAAOSwBaxntXw5&itmprp=enc%3AAQAKAAAA8FkggFvd1GGDu0w3yXCmi1cYHNm1NVBtPkGl2trSSJ2dJEN2wOPRAHJ5Xqu8Dmd8pxkdxtfrzdeLeogwdOz%2B2%2BVRQLhuUOxEZ2rZcOjFJemZAdsdz5q2XdMNhUYbPk4%2FxWrQvet%2FV5L0WdPeFZts3%2FgLGhKvninoelHYqhgWMZTwP72JElIqMtkQ3Ebt5xcDsiq6CezLKBI3Vz0%2BNKPxeGwtOr66pzNEFpOcgdLE4WdrlyzqvttAHLD3VFYz7bmjve1VJ8Amd5Q7icM8Ng1uEV3HebQpfHo8oamDY3h6cLvgLmI5EZQh9H%2BSMBbhdNuSYw%3D%3D%7Ctkp%3ABk9SR77z2ILUZQ

Motor:

https://inverterdrive.com/group/Motors-AC/ac-Motor-TEC-T3C160M1-11kw-4pole/

Total Price of the Gear box

Items	Price (HKD)
Gear 1	1350
Gear 2	3290
Chain Drive	2200
Pinion 1	2093
Pinion 2	2093
Shaft 1,2,3	722
Bearing	596.39
Gearbox housing	2900
Sprockets	668.50
Retaining Ring	234.21
Shaft Coupler	1291.2
Motor	6156
Bolts & Nuts	80
Keys	200.21
Total Price	22839.07

15. Installation and Maintenance

15.1. Assemble procedure

Here is a paraphrased version of the gearbox assembly procedure:

1. **Shaft Assembly**: Insert all keys into their respective keyways on the shafts.
2. **Gear Installation**: Mount the gears onto the shafts in order of shaft size.
3. **Securing Components**: Add retaining rings and spacers to hold the gears in place and prevent movement during operation.
4. **Bearing Installation**: Fit the bearings onto the shafts.
5. **Shaft Placement**: Position the assembled shafts into their designated spots in the housing frame, then attach the housing cover using pins.
6. **Labeling**: Apply the gearbox identification label.
7. **Fastening**: Secure the cover to the frame using bolts and nuts.
8. **Sealing**: Install gaskets, O-rings, and dust seals, securing them with endcaps between the covers.
9. **Lubrication Setup**: Attach oiling components for drip lubrication.

15.2. Maintenance of system

To ensure optimal performance and prolong the service life of the gearbox, implementing a thorough and regular maintenance schedule is essential. Various maintenance strategies can be applied from different perspectives to achieve this:

1. Regular Visual Inspection with replacement and cleaning:

   1.1 Mechanical transmission components inspection

   Regular visual inspections of the mechanical transmission components in the gearbox are crucial for promptly identifying physical conditions and potential issues, preventing minor problems from escalating into severe failures. When issues such as worn gears or bearings, loose parts, or component displacement are detected, immediate replacement of damaged or worn parts is necessary before resuming operation. Periodic inspections enable the maintenance team to execute timely replacements, ensuring the components' quality and performance while preventing major breakdowns.

In addition to routine visual checks, regular cleaning and removal of contaminants from gearbox components are vital for sustaining optimal performance. In harsh operating environments, such as those in mining transmission units, contaminants like dust, dirt, and moisture can accumulate inside the gearbox. These contaminants can hinder gear function, accelerate wear, or damage components, ultimately degrading the gearbox's efficiency. Therefore, consistent cleaning and removal of accumulated contaminants are carried out to uphold the performance and efficiency of the gearbox design.

1.2 Alignment checking

Proper alignment of gearbox components plays a vital role in ensuring the efficiency and durability of parts like gears and shafts. When these components or shaft couplings are not aligned correctly, it places extra stress and load on them, resulting in faster wear, particularly on the gears. This wear reduces the gearbox's overall effectiveness and shortens its service life. To avoid breakdowns and safety hazards, it's crucial to regularly inspect and monitor alignment. These steps help catch and fix misalignment early, maintaining the gearbox's performance and extending its operational lifespan.

1.3 Lubricant regular change and inspection

To ensure the gearbox operates efficiently and enjoys a prolonged lifespan, it's vital to conduct routine visual assessments of the lubricants. These inspections should focus on detecting alterations in attributes such as color, viscosity, or the presence of contaminants and oxidation, all of which could compromise lubrication effectiveness and lead to overheating from friction. Should any degradation in lubricant quality be observed, an immediate oil replacement is necessary. The American Gear Manufacturers Association (AGMA) advises that oil should be swapped out after the initial 500 hours of gearbox operation, followed by regular changes every 2500 hours thereafter.

In addition to these checks, maintaining gearbox performance requires the consistent removal of built-up oil and debris. Accumulated oil can attract dust, dirt, and other impurities, particularly in challenging environments, which may impede gear function and diminish overall efficiency. Thus, incorporating regular cleaning into the maintenance routine is essential to prevent these contaminants from affecting the gearbox, ensuring it continues to perform optimally over time.

2. Temperature checking and monitoring:

2.1 Lubrication temperature

To maintain the gearbox's efficiency and avoid overheating, a dedicated sensor is employed to monitor the oil temperature. Regular oversight of the oil is necessary, as it requires periodic replacement based on both the duration of operation and the oil’s temperature. Elevated temperatures accelerate the degradation of the oil’s molecular structure, weakening its ability to safeguard the gearbox components. In our gearbox design, tailored for the demanding conditions of mining operations, we utilize an oil drip lubrication system—a circulating mechanism that aids in cooling the oil. Nevertheless, consistent temperature monitoring remains indispensable to uphold the gearbox’s performance and prevent overheating.

2.2 Mechanical transmission components temperature in the gear box

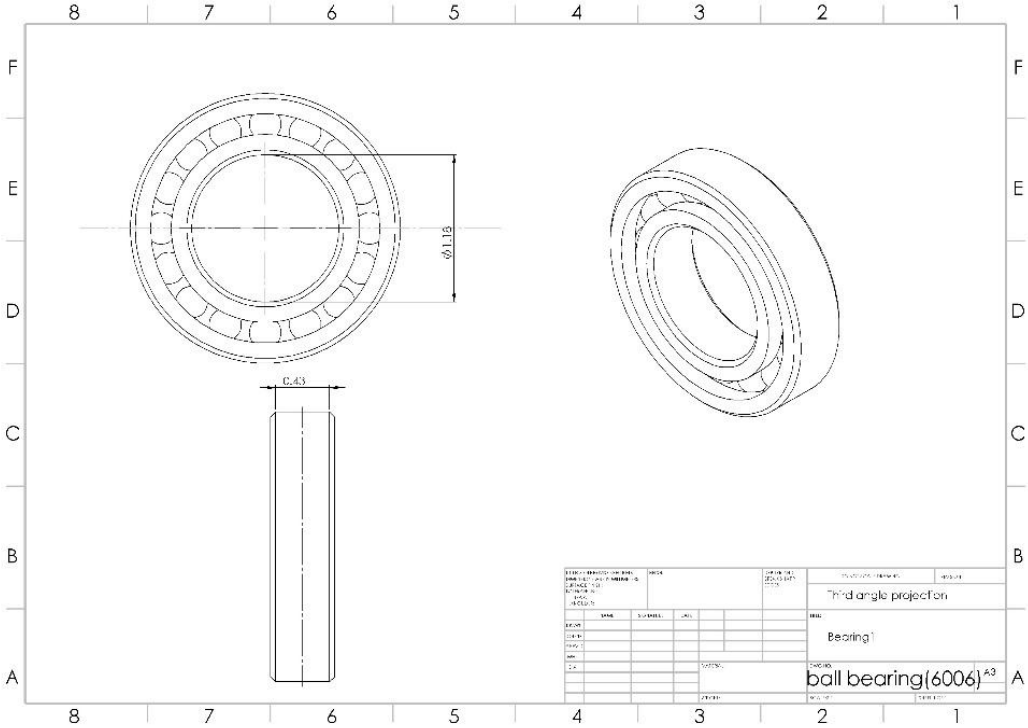
Regular monitoring of the temperatures of mechanical transmission components, such as gears, bearings, and shafts, is vital during gearbox operation. Techniques like infrared thermography cameras can be used to detect hotspots or assess temperature changes by examining thermal images. These unusually high-temperature areas often point to underlying problems, such as excessive friction from poor lubrication or component overloading. As a result, consistent temperature checks enable manufacturing and technical support teams to schedule maintenance or apply corrective measures based on the cause of the heat, helping to avoid safety risks and further damage while keeping the gearbox running smoothly.

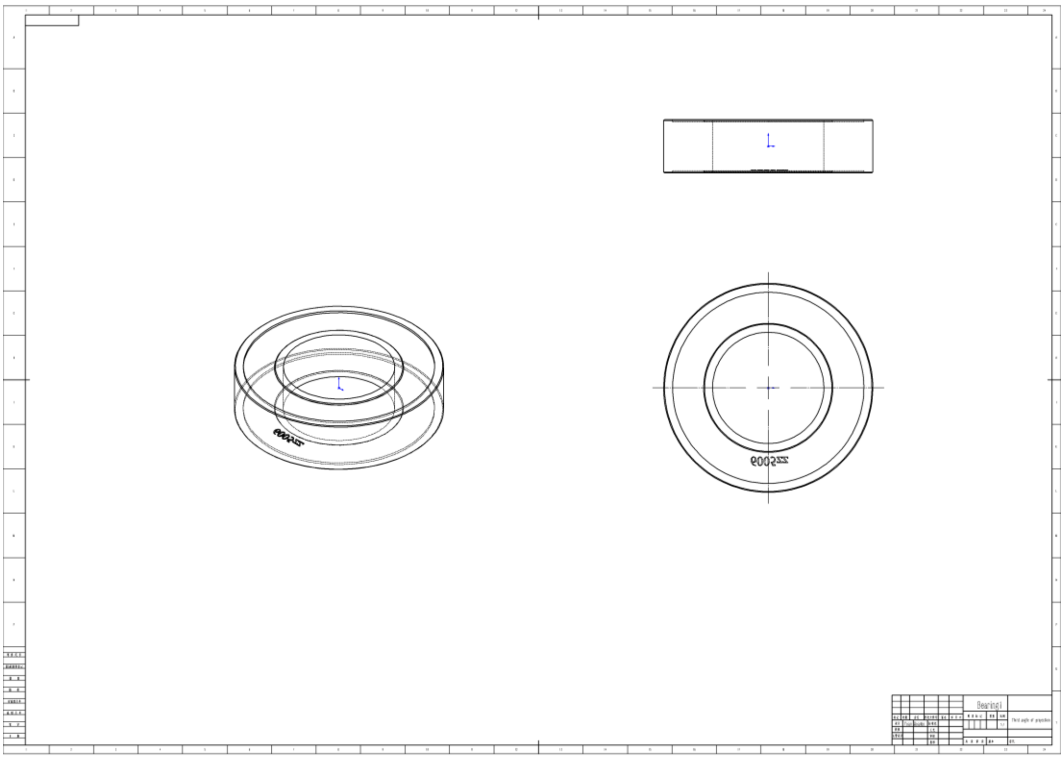
15.3 Vibration Analysis:

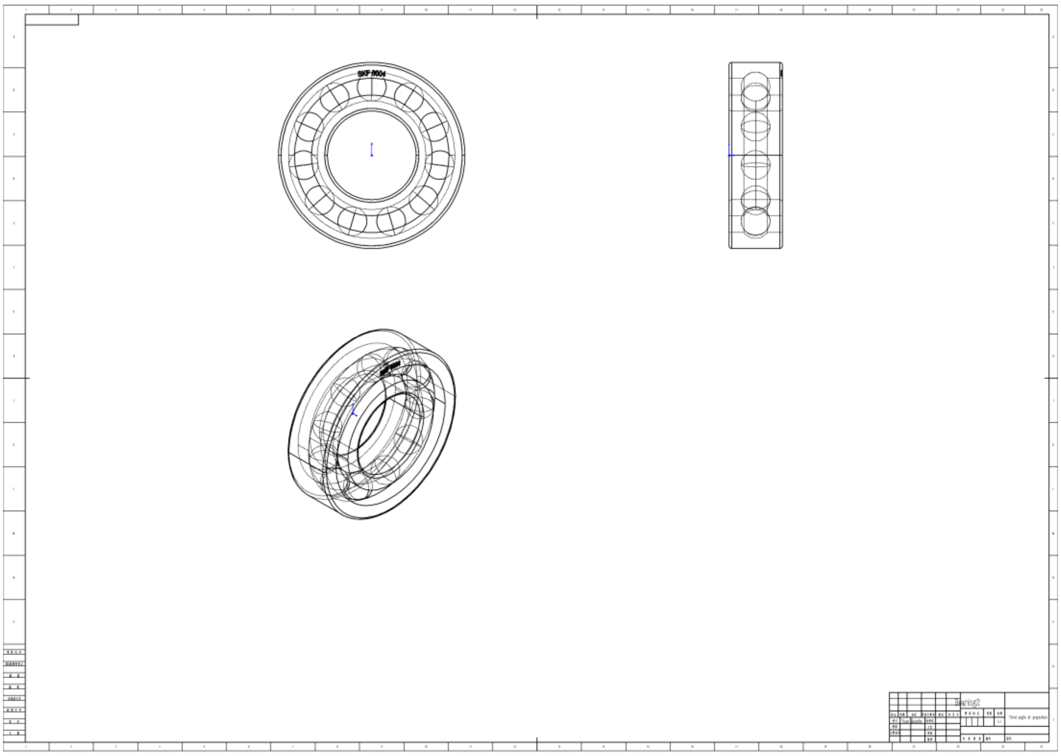
Vibration Analysis stands out as a highly effective predictive maintenance technique for spotting failures or potential problems within a gearbox. As such, it’s critical to keep track of the gearbox’s vibration levels through vibration analysis methods to avoid breakdowns. Each component of the gear system generates a unique vibration signature, which is recorded using a field vibration analyzer. When vibrations become excessive or their patterns shift, it may point to issues like misalignment, gear wear, bearing damage, or other mechanical faults. Upon detecting irregular vibrations, a vibration-monitoring sensor sends out an alert or signal. This allows vibration analysts and engineers from the manufacturer’s technical support team to pinpoint the vibration’s origin and assess potential safety risks by conducting an in-depth analysis of the vibration spectrum. Consequently, this consistent monitoring and evaluation enable them to plan maintenance ahead of time or implement suitable corrective measures, ensuring safety hazards are avoided and the gearbox continues to perform reliably.

16. Engineering Drawing

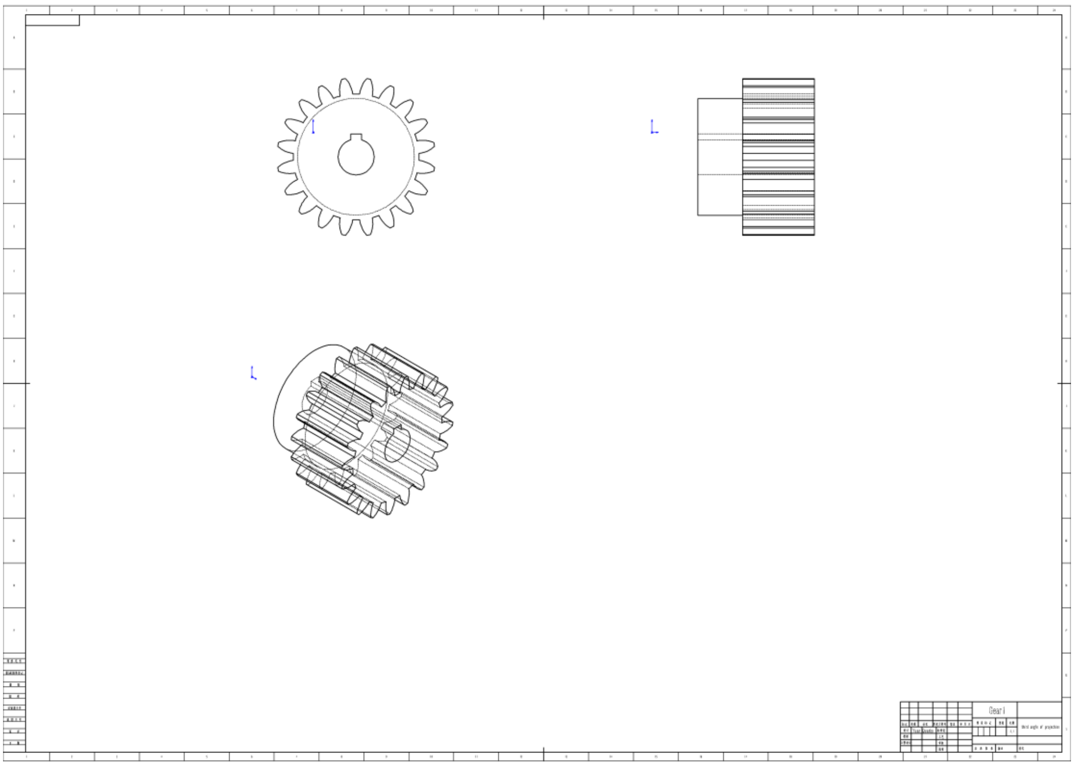
Bearings

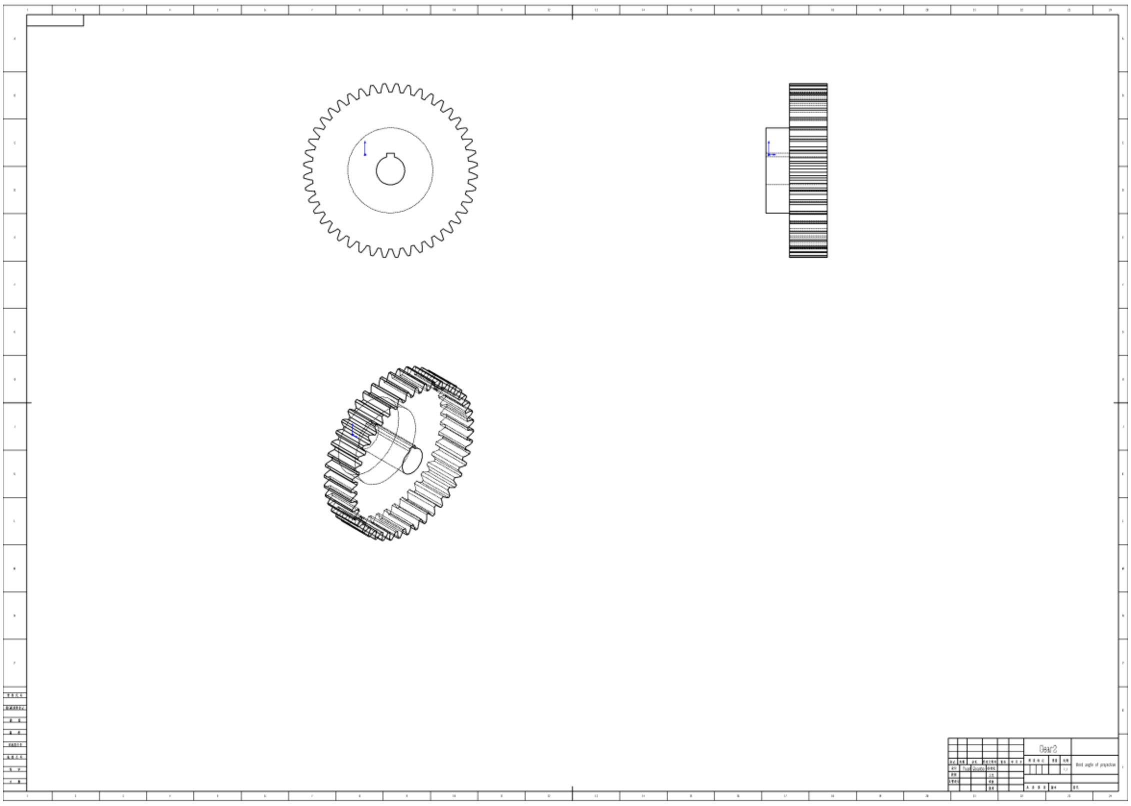


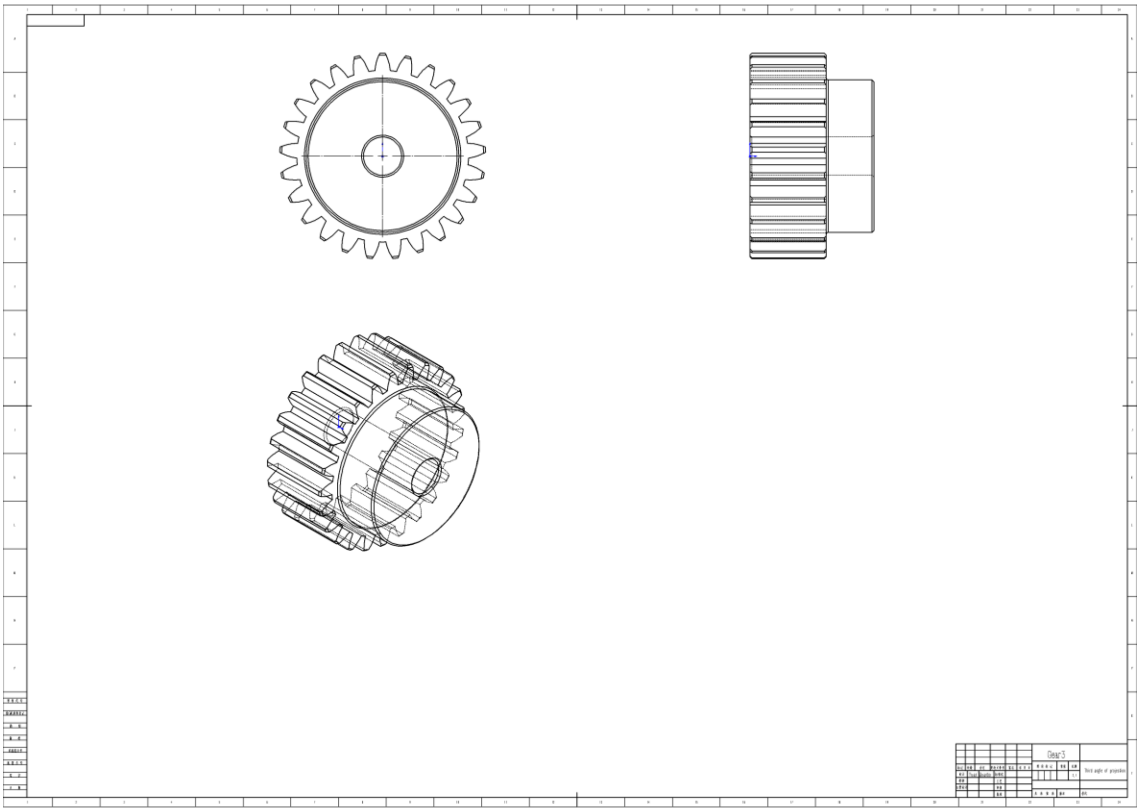


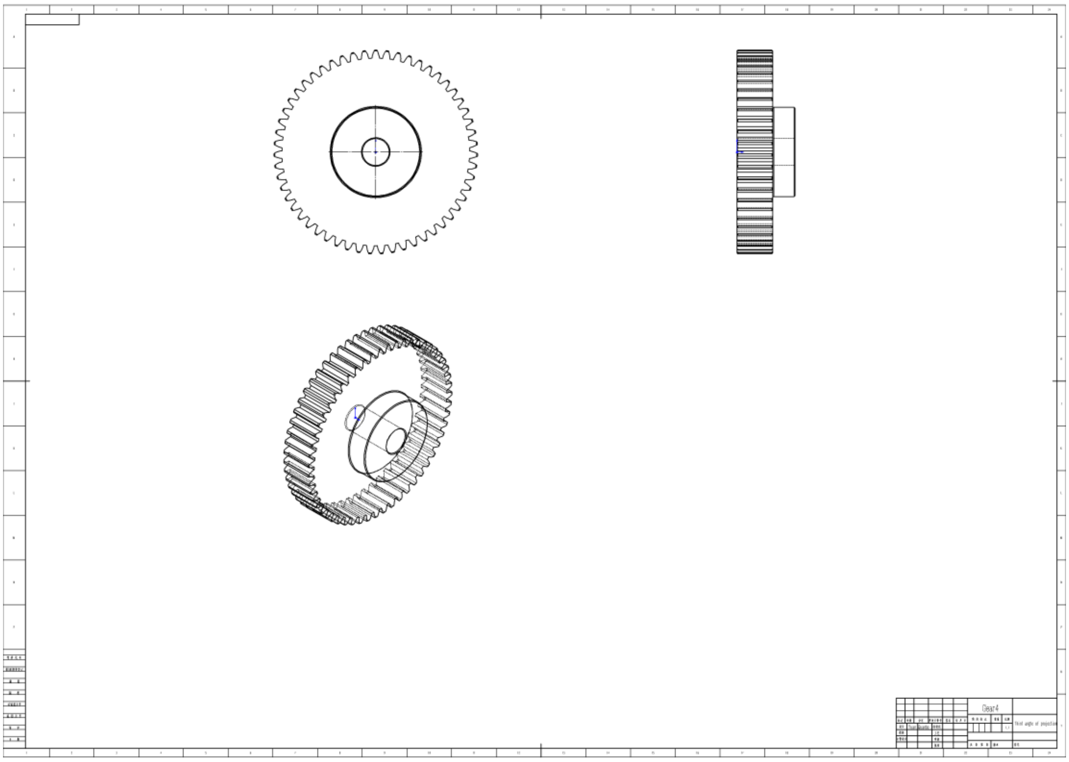


Gears



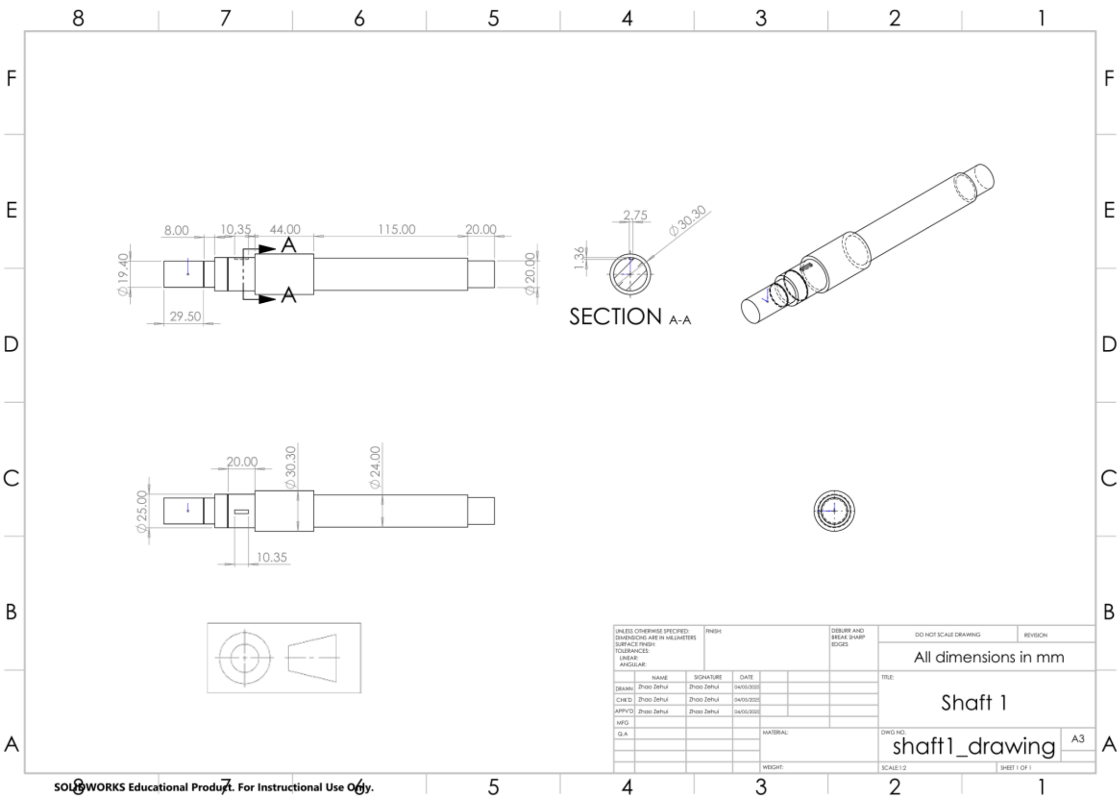

Gear2


Gear3

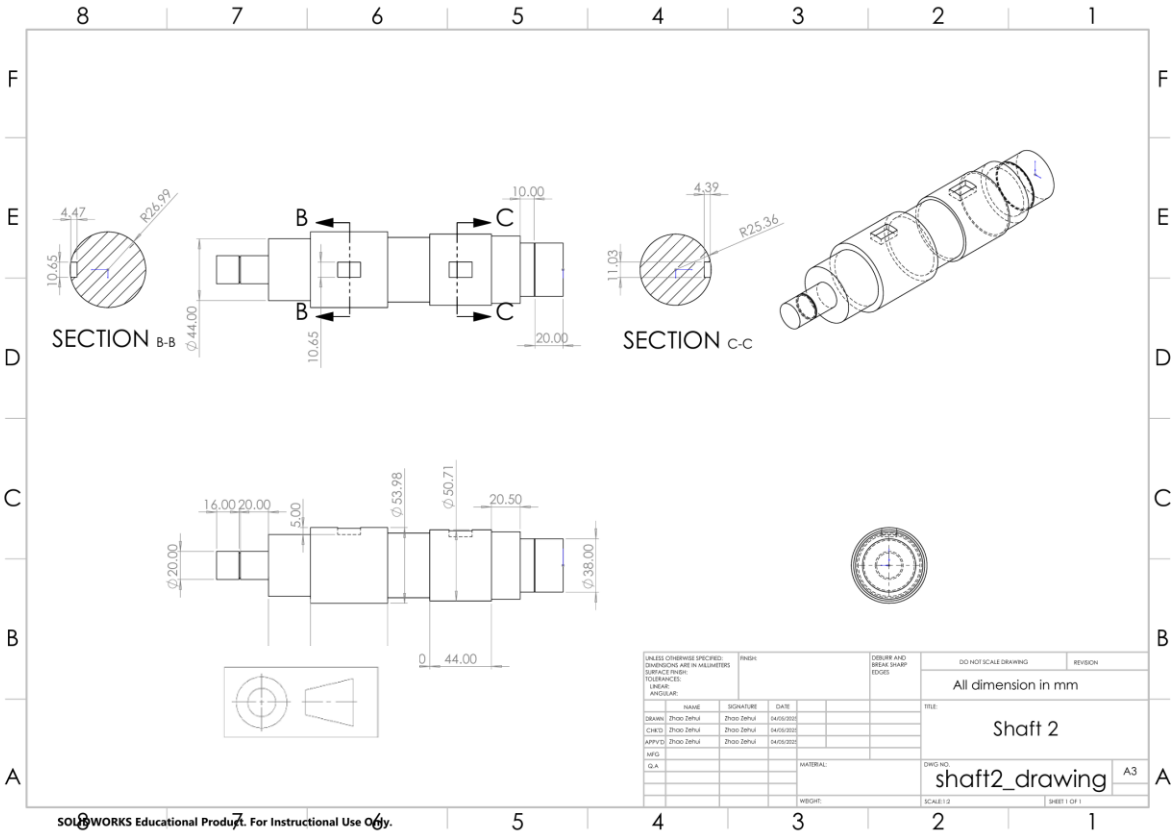


Shafts

Shaft 1



Shaft 2


SECTION B-B
SECTION C-C
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10.65
10.00
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11.03
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16.00 20.00
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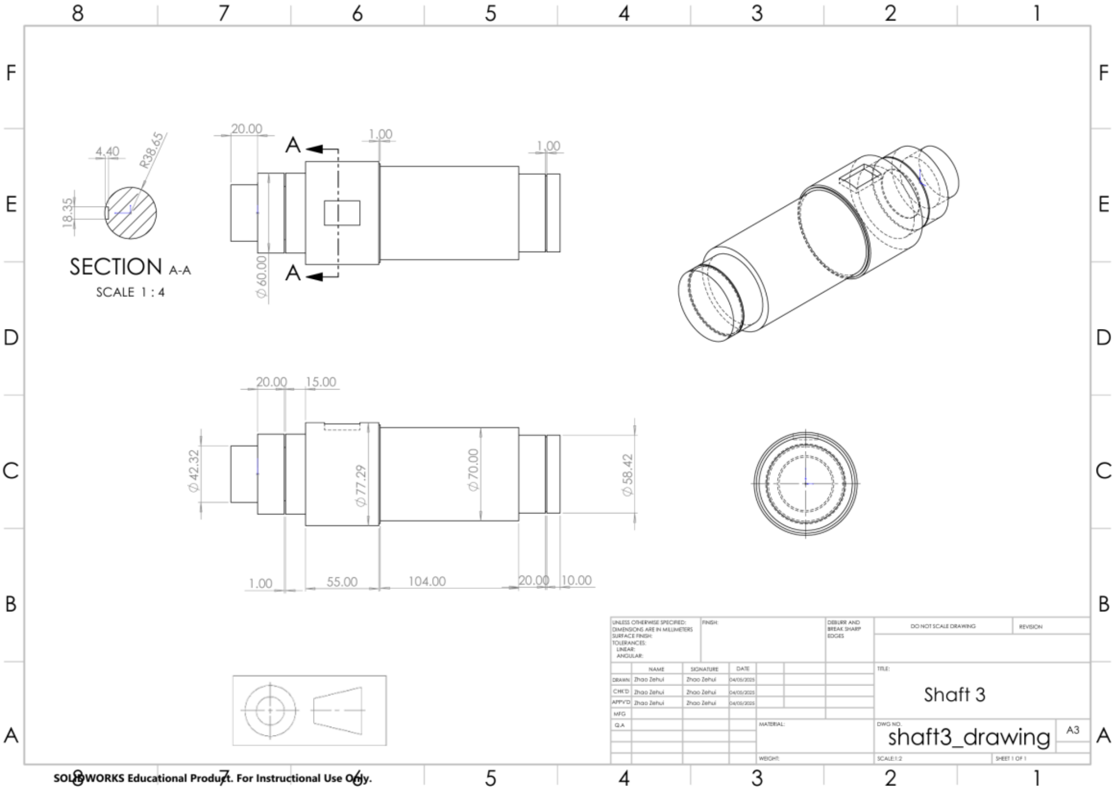
Ø53.98
Ø50.71
20.50
Ø20.00
Ø38.00
0 44.00
All dimension in mm
Shaft 2
shaft2_drawing
A3
DRAWN Zhao Zehui
CHK'D Zhao Zehui
APPV'D Zhao Zehui
SCALE:1:2
SHEET 1 OF 1
SOLIDWORKS Educational Product. For Instructional Use Only.

Shaft 3

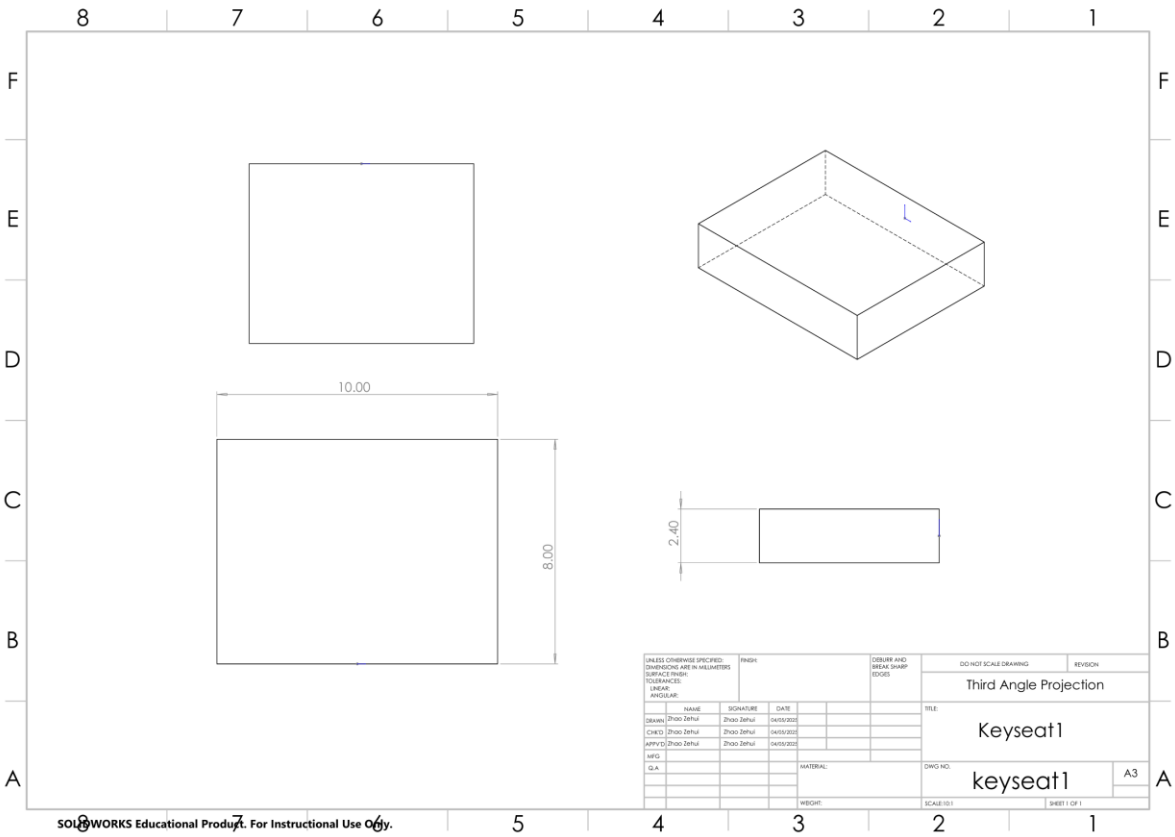


Updated Dimensions for shaft 1, 2 & 3

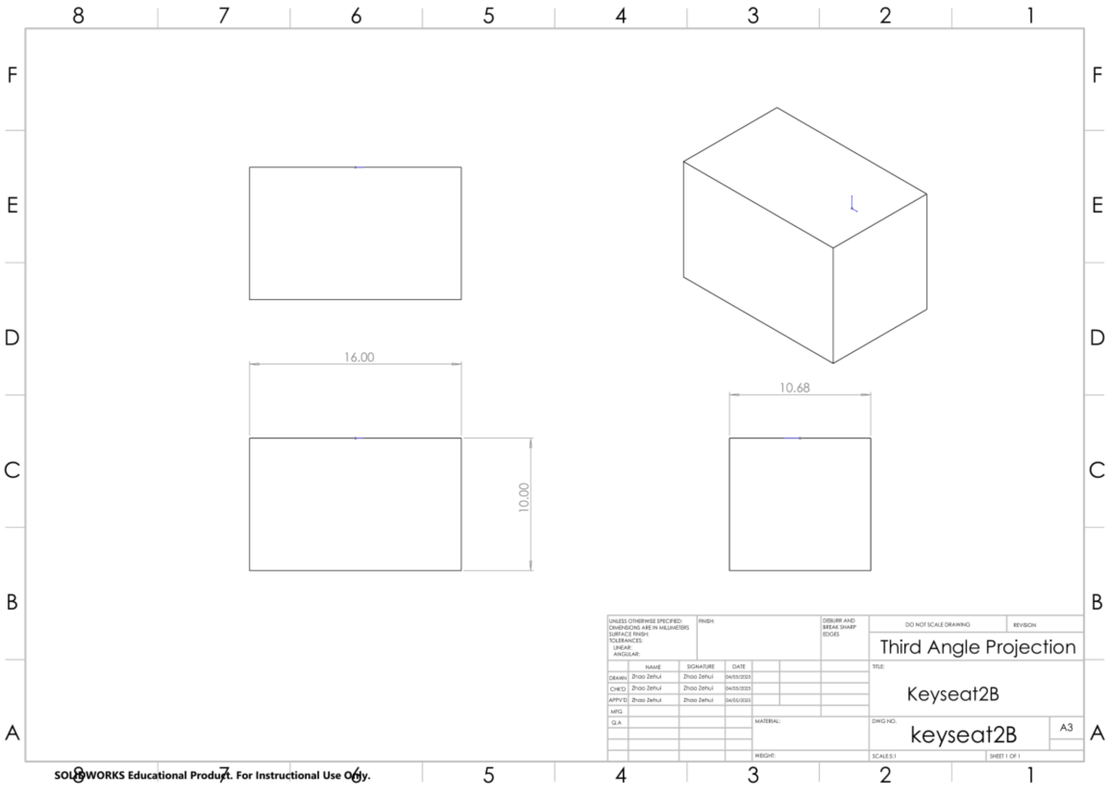
	D_1 (in mm)	D_2 (in mm)	D_3 (in mm)	D_4 (in mm)	D_5 (in mm)
Shaft 1	19.4	25	30.3	20	
Shaft 2	20	44	53.98	50.71	38
Shaft 3	42.32	60	77.29	70	58.42

Keyseats & Spacers

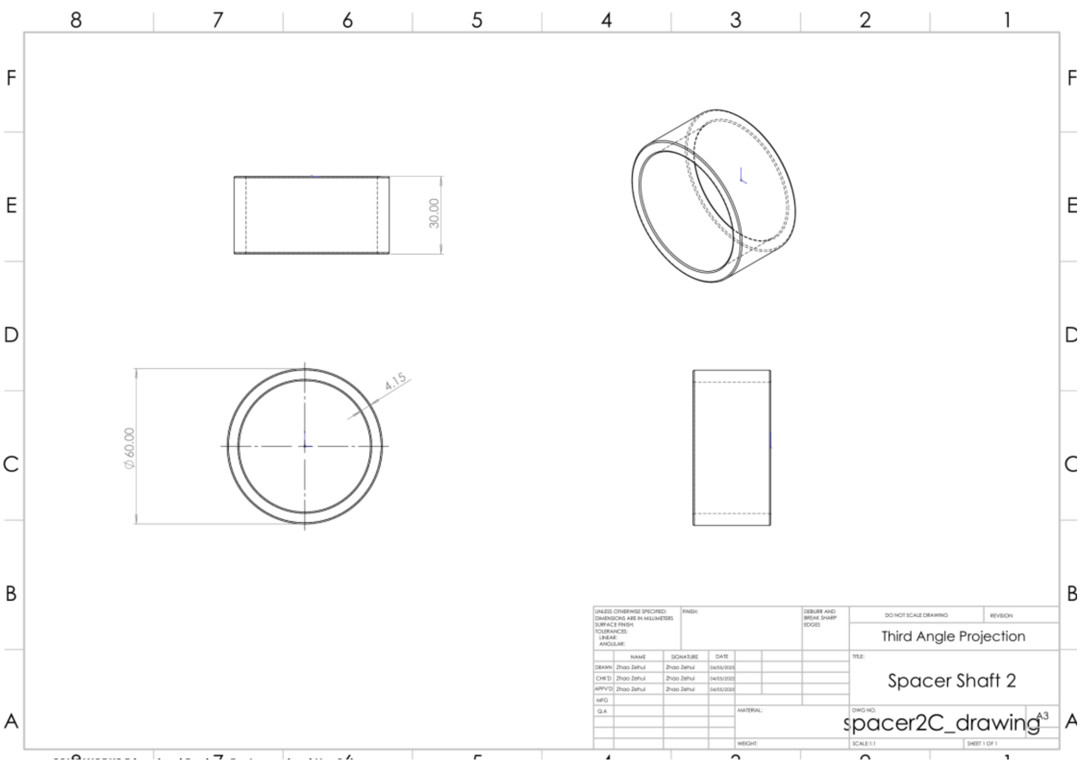
Keyseat 1



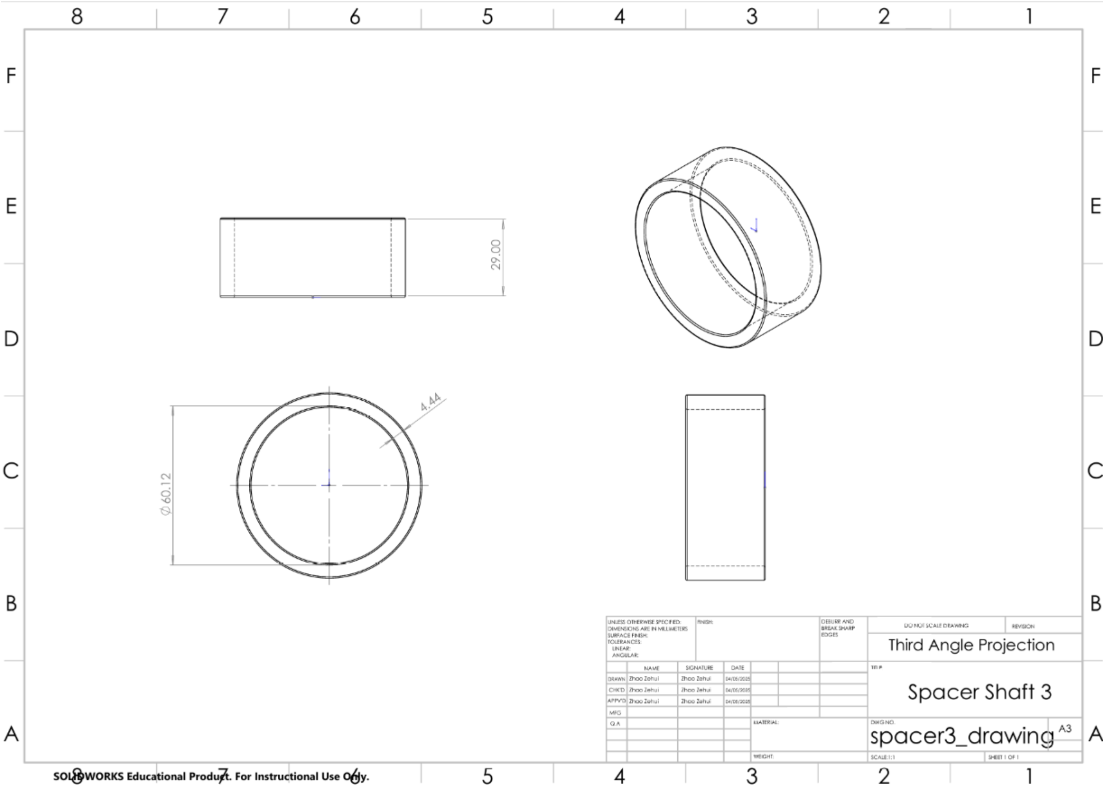
Keyseat 2B



Spacer 2C



Spacer 3



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Ø60.12
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
Third Angle Projection
NAME
SIGNATURE
DATE
DRAWN Zhao Zehui Zhao Zehui 04/05/2025
CHK'D Zhao Zehui Zhao Zehui 04/05/2025
APPV'D Zhao Zehui Zhao Zehui 04/05/2025
MFG
Q.A
MATERIAL:
Spacer Shaft 3
DWG NO.
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A3
WEIGHT:
SCALE:1:1
SHEET 1 OF 1
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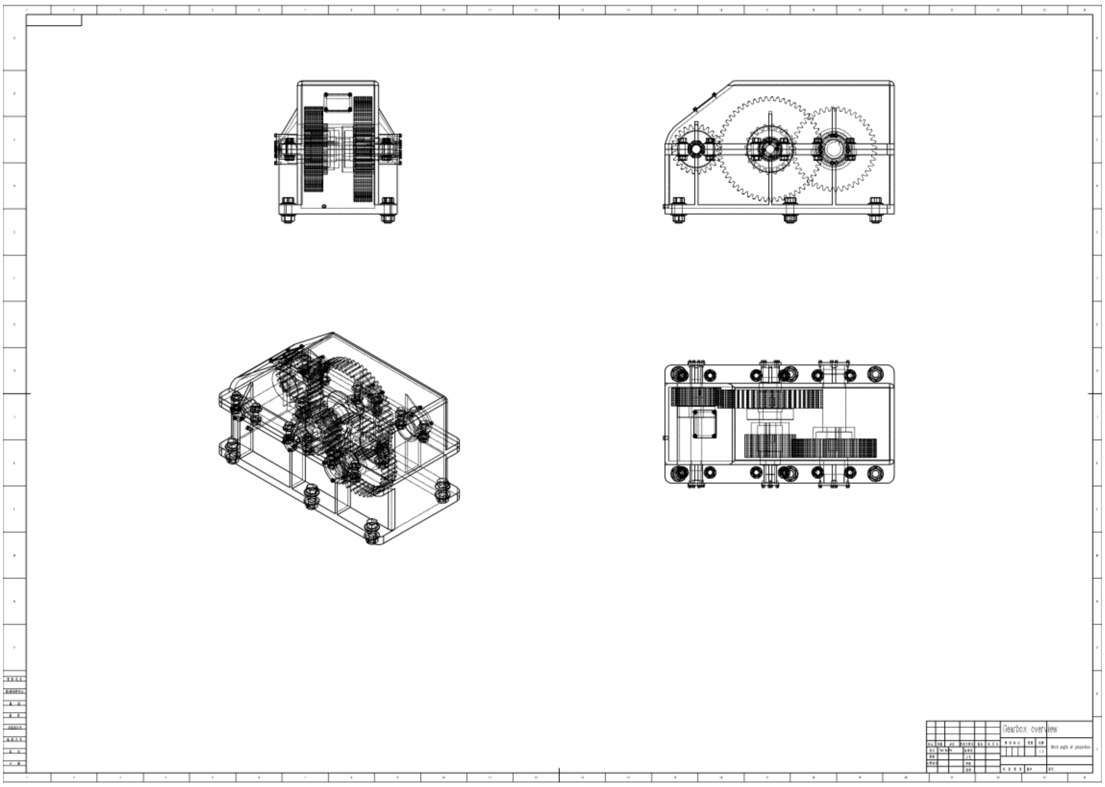
Gearbox



17. Work Distribution

Parts Division	Student Name	Student ID
Design Concerns & Methodology Shaft coupling Retaining ring Housing design Material selection of housing Parameter entry of retaining ring Bearing drawing Gear drawing	Yip Lam	3036068046
Layout of the system Chain drive calculation Chain drive market research Gear design & calculation Shaft design & calculation Shafts model & landscape drawing Shaft market research Bearing calculation Key & Keyseat design & calculation Key & Keyseat model & landscape drawing Key & Keyseat market research	Zhao Zehui	3036064052
Motor selection Market research Lubrication Housing Engineering Drawing Gear landscape drawing Gearbox Assembly	Yuan Quanbo	3036105519