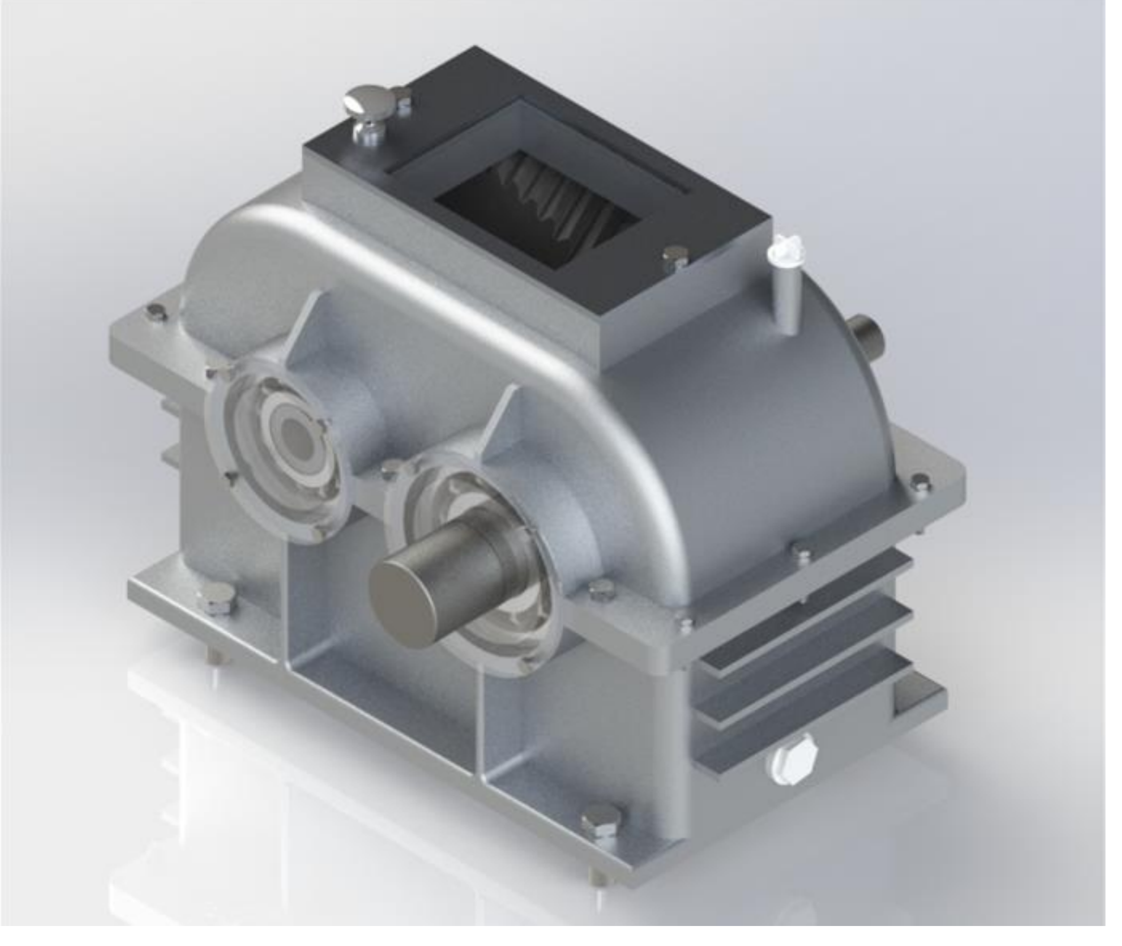
Design of Machines
and
Mechanical Systems

TWO STAGE REVERTED HELICAL GEARBOX



Design Team

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Problem Statement

Provide engineering consultancy to a company which is planning to provide off-the-shelf speed reducers in various capacities and speed ratios to sell to a wide variety of target applications.

Objective

Objective is to provide the most optimized possible gearbox according to the needs of the client.

Client Requirements

Gear Specifications

• Power to be delivered	49 kW
• Power efficiency:	95%
• Steady state input speed:	1000 RPM
• Steady state output speed:	179 RPM
• Type of gears:	Helical
• Gear train type:	Two stage, reverted
• Shocks:	Low to medium shocks
• Gear life:	>12000 hours

Shaft Specifications

• Permissible shaft extension:	100 mm outside gearbox
• Shaft diameter tolerance: Shaft in	+/- 0.025 mm
• line concentricity:	+/- 0.125 mm
• Shaft Alignment:	+/- 0.001 rad

Miscellaneous Specifications

• Lubrication check:	Every 2000 hours
• Change of lubrication:	Every 8000 hours
• Bearing life:	>12000 hours
• Operating temperature:	-5 °C to 65 °C
• Noise:	<85 dB from 1 meter

Note : Summary of final calculations for all selected parts is given in this [Excel Sheet.]

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1. Gears

1.1 Gear Parameters Selection
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1.1 Gear Parameters Selection

1.1.1 No. of Teeth:

Based on the Input and Output RPM requirements and list of choice 1 and choice 2 modules (DDB Table 17.3), an elementary python code was used to iterate through all possible combinations.

Characteristics of an ideal combination of no. of teeth was taken to be as follows:

1) Minimal difference between expected and actual output RPM
2) No. of teeth must be coprime
3) Coaxial Input and Output shaft (Given)
4) Gearbox Size must be minimal. [Based on the assertion in DDB Table 17.5, for compact gearbox size, this implies, speed reduction at both stages must be identical. Combined with Req (3), this means module of all gears must also be identical]

Based on the above requirements, 2 possible pairs were found to be feasible and with the same module at both stages, from all the possible combinations that were iterated through:
i) 19-45, Output RPM = 178.2716
ii) 22-52, Output RPM = 178.9941

Between the two feasible pairs, 22-52 satisfies Req (1) to a greater extent but the 19-45 pair satisfies Req (2) and (4) better. Further, the degree to which Req (1) is better satisfied by the 22-52 pair can be considered fairly negligible compared to the 19-45 pair.

Therefore the gear teeth combination chosen is 19-45

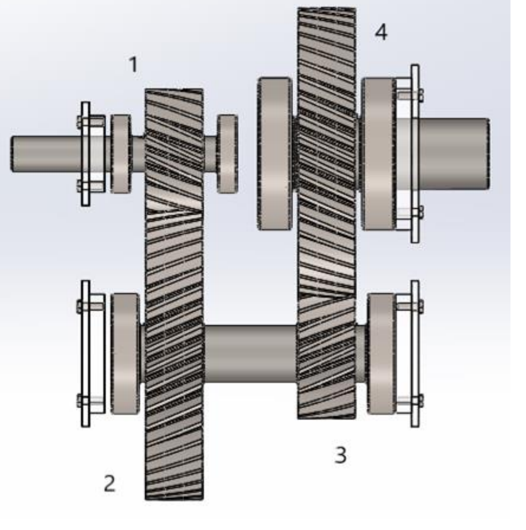
Calculations:

Total Vel Ratio (TVR) = (1000/179) = 5.5866
Therefore, VR at each stage (DDB Table 17.5 Eqn 17.5) = $(TVR)^{0.5}$ = 2.3636

O/P RPM for the 22-52 pair: $1000*(22/52)^2$ = 178.9941
O/P RPM for the 19-45 pair: $1000*(19/45)^2$ = 178.2716

Eqn For ensuring equal Center Distance used in the code:
$m_1(z_1+z_2)=m_2(z_3+z_4)$

Where,
m_1= Module for Gear Pair 1-2
m_2= Module for Pair 3-4
z_i= Teeth on i^{th} Gear



1.1.2 Minimum Module and Helix Angle

Calculations for Minimum Module :

$$m_n \geq 1.15 \cos\beta \sqrt[3]{\frac{[M_t]}{y_v[\sigma_b]\psi_m Z_1}}$$

Where,
β = Helix angle
y_v= Lewis form factor (DDBT 21.26)
$[M_t]$ = Design torque
σ_b = Design bending stress
$\Psi_m = \frac{b}{m} = 10$
Z = Number of virtual teeth

$$[M_t] = \frac{Cs*Mt*FOS}{Cv}$$

- For low shock levels, occasional moderate shock Service factor is: .... DDBT 17.17
  $C_s = 1.5$
- Velocity factor :
  $C_v = \frac{5.6}{5.6+\sqrt{v}}$ (v is the assumed velocity over here that is 10 m/s) .... DDBT 18.8
- Assumed FOS is 2 for calculating minimum module
- $M_t = \frac{60*10^6*Input\ power}{2\pi*RPM}$ ....Input power = $\frac{Required\ power}{Efficiency}$
- $Z = \frac{actual\ teeth}{cos^3\Psi}$

As Minimum required module is dependent on the helix angle along with other assumed constants, iterations were performed using Excel to find both of them.

For deciding a helix angle:

1. Iterations were performed for helix angle ranging from 15 to 25, since the allowable angles range from 8 to 25 (DDBT 21.25)
2. The minimum module corresponding to each angle was obtained and the next common module from recommended series was chosen,
   m_n = 8 mm
3. Further calculations were performed to find effective force corresponding to each angle and the one with the lowest value was chosen.
4. Chosen **Helix angle is 25°**

Note : Since the gear 3 is the critical gear all calculations for safety factors are done giving It more weightage on it.

1.1.3 Calculations for Material

- All previous calculations were done considering arbitrary material properties to see the effect of other parameters on forces.
- The parameters considered for choosing the materials were:
    1. Strength provided
    2. weight of the gear
    3. Cost
- Several different materials, found in the data book and in some online resources for gear design, were analyzed for pair 3-4 containing the highest gear forces. Their detailed calculations can be for Here .
- Out of those materials, 4 were found to have satisfactory FOS and finally, of these, the material that minimized degree of overdesign was selected.

Selected Gear material

Material	0.55 % Carbon Chromium steel
Ultimate tensile strength	850 MPa
Brinell Hardness Number	500 BHN
Density	7.85 g/cm^3

1.1.4 Calculation for FOS of gear for a particular material:

- These calculations were performed for satisfying the condition of strength for material and module.
- Two methods were considered for these:
    1. Velocity Factor Method (Preliminary method)
    2. Buckingham's Incremental Load Method (More detailed method)
- Effective load for both was found and the one with higher value (Buckingham's method) was chosen.

- **Calculations using Buckingham's Incremental Load Method:**

Formulae : …… (Chapter 18 DDB)

For Bending Strength (S_b) :

$S_b = m_n * b * \sigma_b * Y$ (N)

1. Y = Lewis form factor (DDBT 21.26)
2. Gear width (b) = 10*m_n (mm)
3. Permissible bending stress (σ_b)= $\frac{UTS}{3}$ (N/mm^2)

For Wear Strength (S_w) :

$S_w = \frac{b*Q*dp*K}{cos^2\psi}$ (N)

1. Ratio factor (Q) = $\frac{2zg}{zg+zp}$
2. Diameter (d_p) = $\frac{zp*mn}{cos\psi}$ (mm)
3. Load – Stress factor (K) = $0.16(\frac{BHN}{100})^2$ (N/mm^2)

For Effective load (P_{eff}) :

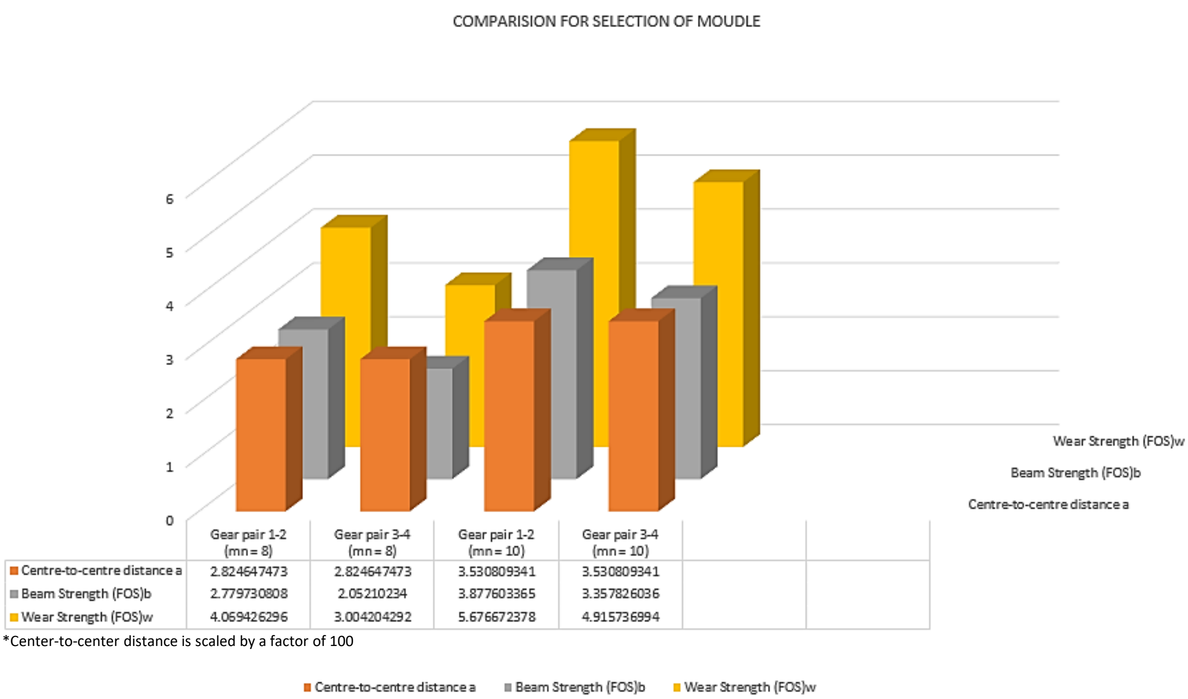
$P_{eff} = (C_s * P_t + P_d)$ (N)

1. C_s = Service Factor ....(DDBT 17.17)
2. Tangential force (P_t) = $\frac{2*Torque\ transmitted\ by\ gear(N-mm)}{d_p}$ (N)
3. Incremental dynamic load (P_d) = $\frac{21v(Cebcos^2\psi+Pt)cos\psi}{21v + \sqrt{(Cebcos^2\psi+Pt)}}$ (N)
4. v = Pitch Line velocity (m/s)
5. C = deformation factor (N/mm^2) ....(DDBT 17.24 and 17.25)
6. Sum of errors between two meshing teeth (e) = $e_p + e_g$ (mm)
   a. e = 8 + 0.63*[Tolerance factor (ϕ)]
   b. $\phi = m_n + 0.25\sqrt{d_p}$

For FOS Bending and Wear :

$(FOS)_b = \frac{S_b}{P_{eff}}$ $(FOS)_w = \frac{S_w}{P_{eff}}$

1.1.5 Module Selection



- Important factors considered in module selection are compared in the above chart.
- As minimum required module is 7, modules **8 and 10 from the recommended series** were tested. Detailed Calculations for the same are provided in this Excel sheet.
- From this it was deemed that m_n = 8 is the best fit.

1.2 Gear Life

Gear Life Estimations will be carried out assuming a reliability of 95% for the required life of 12000 hours. The procedure and theory behind the calculations can be found here

These calculations rely on stress cycle factors Y_N and Z_N which correspond to bending and pitting resistance respectively.

Once the values of these constants are found, they may be converted directly to number of stress cycles (equal to number of rotations) and using the RPM for the gear, the corresponding number of hours can be found

It is worth noting that these calculations were carried out mainly for verification purposes after all design parameters were selected (based on other factors like strength) and did not contribute directly to any of them

Formulae for Z_N and Y_N:

$$Z_N = Z_E \cdot \sqrt{\frac{F_T \cdot K_O \cdot K_V \cdot K_H \cdot K_S \cdot Z_R}{b \cdot d_{W1} \cdot Z_I}} \cdot \frac{S_H \cdot Y_\theta \cdot Y_Z}{\sigma_{H\lim} \cdot Z_W}$$

$$Y_N = \frac{F_T \cdot K_O \cdot K_V \cdot K_H \cdot K_S \cdot K_B \cdot S_F \cdot Y_\theta \cdot Y_Z}{\sigma_{F\lim} \cdot b \cdot m_T \cdot Y_J}$$

Credit: https://gearsolutions.com/features/estimating-gear-fatigue-life/

The procedures for finding K_O K_V K_H K_S K_B Z_R $Y_\theta Y_Z$ Z_W and Z_R can be found in Chapter 14 of *Shigley's mechanical engineering design*

S_F and S_H are chosen from DDB Table 20.55 and 20.34 respectively

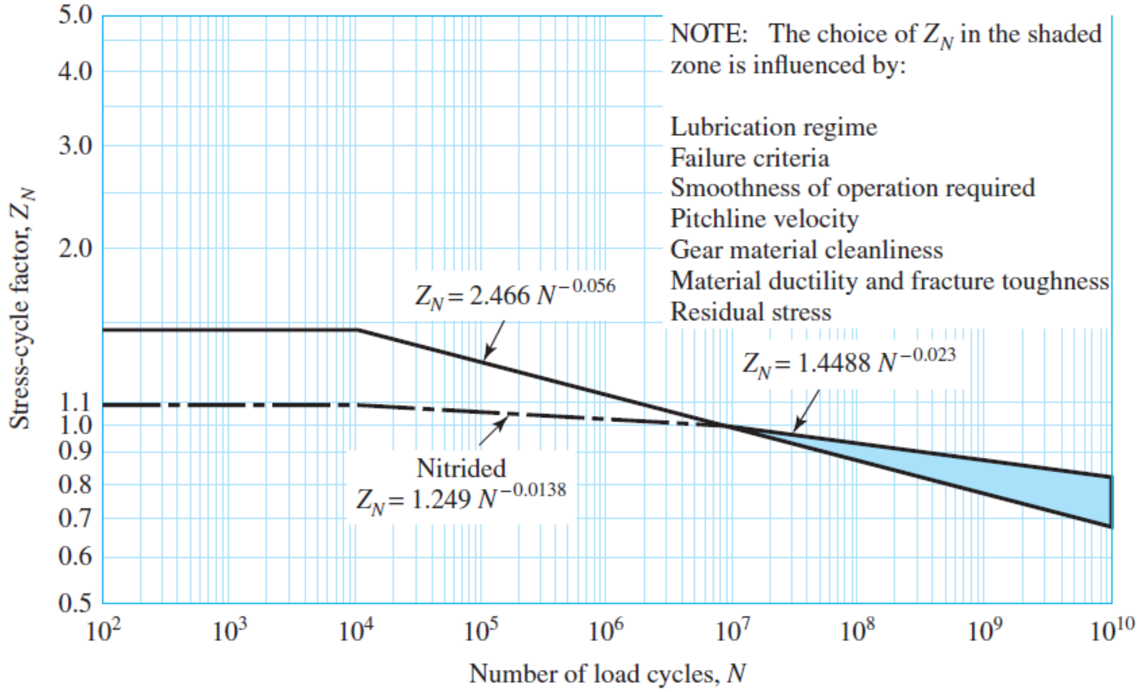
$\sigma_{F\,lim}$ and $\sigma_{H\,lim}$ can be calculated using material hardness and data from DDB 20.50 and 20.17 respectively

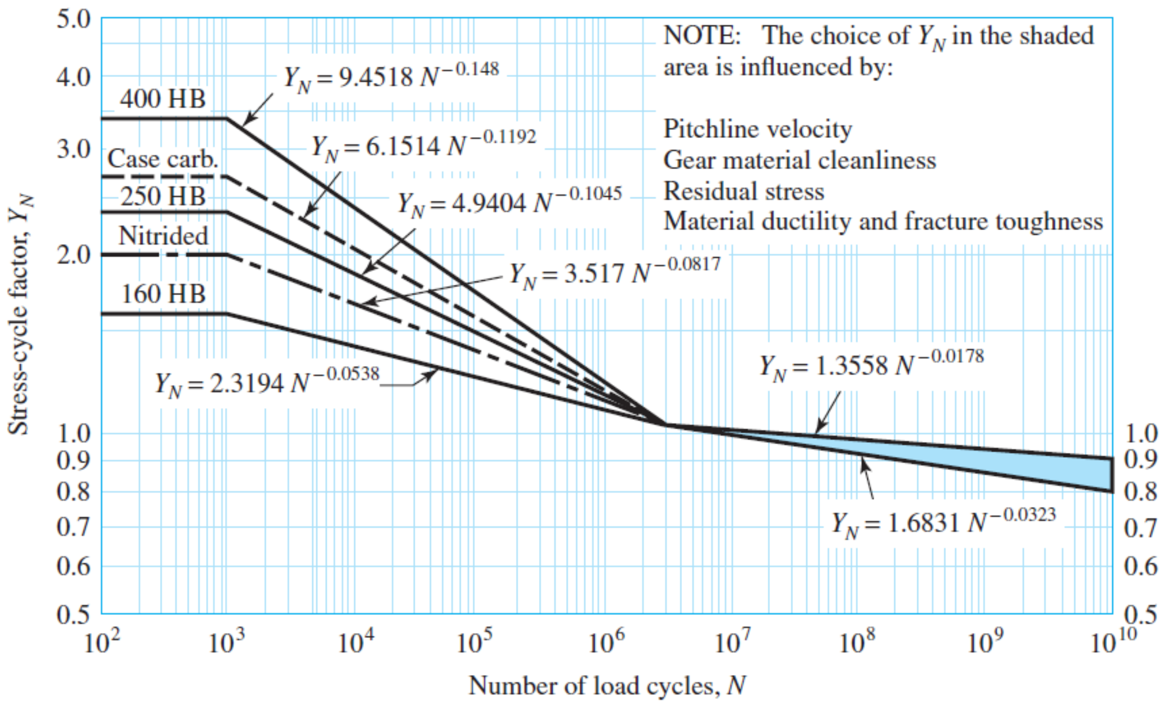
For finding number of gear cycles using Z_N and Y_N multiple formulae exist and an appropriate choice must be made based on the expected range of number of cycles

Since expected life is 12000 hrs, based on RPM, number of cycles are:

1) Gear 1: 1000 RPM * 60 * 12000 hrs = $7.2*10^8$ cycles
2) Gear 2: 422.22 RPM * 60* 12000 hrs = $3.04*10^8$ cycles
3) Gear 3: 422.22 RPM * 60* 12000 hrs = $3.04*10^8$ cycles
4) Gear 4: 178.2716 RPM* 60* 12000 hrs = $1.284*10^8$ cycles

Note that these are all $>10^7$ Hence, a choice of formula can be made using the graphs on the next page





Credit: Shigley's Mechanical Engineering Design

Since all expected values are to the right of the convergence point in the graph, the choice of formulae is narrowed to the equation corresponding to either the lower boundary or the upper boundary of the shaded triangular region.

Lower boundary is selected since it results in a smaller value of number of cycles for a given value of Y_N and Z_N resulting in a more conservative estimate which is preferable.

Formulae for N_L (No of cycles):

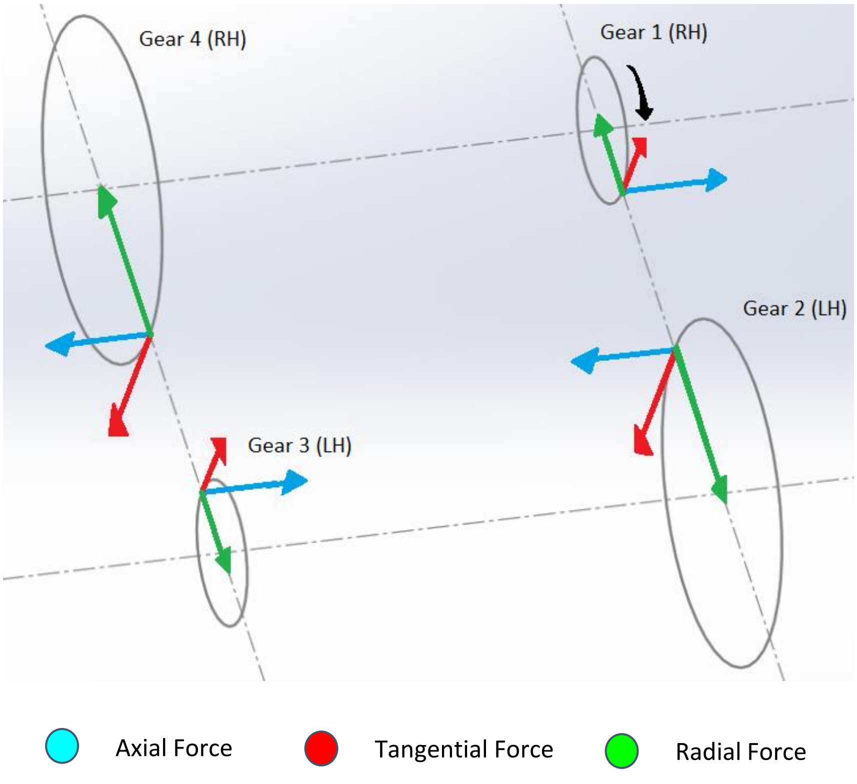
$$Z_N = 2.466\, N^{-0.056}$$

$$Y_N = 1.3558\, N^{-0.0178}$$

Detailed calculations can be found in this Excel spreadsheet. Important final values are tabulated below:

Gear	Z_N	Y_N	N_L(For Z_N)	N_L(For Y_N)	Life For Z_N (hrs)	Life For Y_N (hrs)
1	0.561	0.405	$3.07*10^{11}$	$1.42*10^{19}$	$5.11*10^{6}$	$2.36*10^{14}$
2	0.561	0.319	$3.07*10^{11}$	$2.36*10^{22}$	$1.21*10^{7}$	$9.33*10^{17}$
3	0.808	0.841	$4.52*10^{8}$	$2.15*10^{9}$	17833.58	84895.24
4	0.808	0.662	$4.52*10^{8}$	$3.58*10^{12}$	42237.44	$3.35*10^{8}$

1.3 Gear Forces Analysis



Gear	Tangential Force (N)	Radial Force (N)	Axial Force (N)
1	5873.622	-2358.827	-2738.915
2	-5873.622	2358.827	2738.915
3	13911.209	5586.696	-6486.903
4	-13911.209	-5586.696	6486.903

For Forces : ....(DDBT 18.5)

- Tangential Force (P_t) = $\frac{2*Mt}{d_p}$
- Radial Force (P_r) = P_t*($\frac{tan\alpha_n}{cos\psi}$)
- Axial Force (P_a) = P_t*(tanψ)

1.4 Gear Summary

		Gear 1	Gear 2	Gear 3	Gear 4
Pressure angle		20°			
Helix angle		25°			
Module		8			
Face width		80			
No. of teeth		19	45	19	45
Diameter		167.7134437	397.2160508	167.7134437	397.2160508
Handedness		RH	LH	LH	RH
Life in Hrs.		$5.11*10^{6}$	$1.21*10^{7}$	17833.58	42237.44
Material Properties	Name	0.55 % Carbon Chromium steel			
	UTS	850 MPa			
	BHN	500 BHN			
	Density	7.85 g/cm^3			

Note :

1. Handedness for gears 2 and 3 was taken to be identical to allow the axial forces to be opposite in direction and cancel to some extent, thereby reducing effective load on the bearings (*Ref:https://youtu.be/Y7mhEKuEkUc?t=495*).
2. Final handedness was chosen by analyzing the forces on bearings of Intermediate shaft for both option of handedness for gear 2 and 3.

2. Shaft Design

2.1 Shaft Design

- **Shaft material properties:**

Material	Medium carbon Steel- AISI 1040
Yield Strength (MPa)	420
Ultimate Tensile Strength (MPa)	620
Modulus of Elasticity (MPa)	80000
Modulus of Rigidity (MPa)	200000

- As per the Indian standards, the key material should have minimum tensile strength of 600 MPa and therefore AISI 1040 steel having UTS of 620 MPa was selected as the material for the key.
- The same material as that of the key was chosen for the shaft due to ease of procurement.

- **Shaft Design as per ASME code:**

1) Shear stress equation for hollow shaft:

$$\tau_{max} = \frac{16}{\pi d_o (1 - C^4)} \sqrt{(k_b M_b)^2 + (k_t M_t)^2}$$

where,
τ_{max} = maximum shear stress (N/mm^2)
M_b = bending moment acting on hollow shaft (N-mm)
M_t = torsional moment acting on hollow shaft (N-mm)
C = inside diameter/ outside diameter = d_i / d_o

2) Permissible shear stress calculation:

$0.75 * 0.30\ S_{yt}$ (MPa)	94.5
$0.75 * 0.30\ S_{yt}$ (MPa)	83.7
τ_{max} = Minimum of the above (MPa)	83.7

- The above formulae accounts for the presence of keyways.

3) Combined shock and fatigue factor values selection:

k_b (for bending moment)	1.75
k_t (for torsional moment)	1.25

- The above values are chosen by referring to Design Data Book Table 9.8
- These values are selected from 'load suddenly applied (minor shock)' category as the gearbox is subject to low shock levels and to occasional moderate shocks.

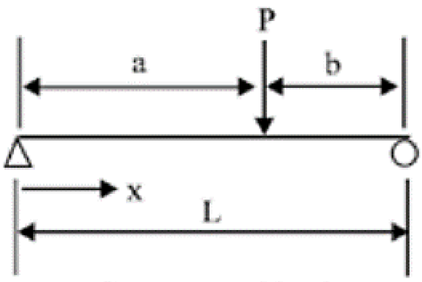
- **Formulae:**

1) Angle of twist/length:

$$\theta = \frac{584 M_t l}{G d_o^4 (1 - C^4)}$$

where,
θ = angle of twist (˚)
l = length of hollow shaft (mm)
G = modulus of rigidity (N/mm^2)

2) Deflection & Slope (simply supported beam):



Concentrated load

$0 \le x \le a$:

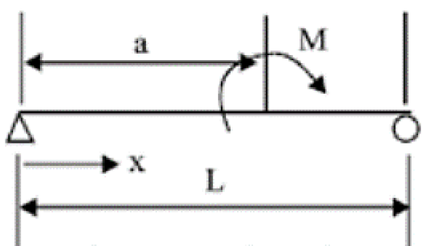
$$\text{Slope} = \frac{Pb}{6EIL}(3x^2 + b^2 - L^2)$$

$$\text{Deflection} = \frac{Pb}{6EIL}(x^3 + b^2x - L^2x)$$

$a \le x \le L$:

$$\text{Slope} = \frac{Pa}{6EIL}[L^2 - a^2 - 3(L-x)^2]$$

$$\text{Deflection} = \frac{Pa(L-x)}{6EIL}(x^2 + a^2 - 2Lx)$$



Concentrated couple

$0 \le x \le a$:

$$\text{Slope} = \frac{M}{6EIL}(-3x^2 + 6aL - 3a^2 - 2L^2)$$

$$\text{Deflection} = \frac{M}{6EIL}(-x^3 + 6aLx - 3a^2x - 2L^2x)$$

$a \le x \le L$:

$$\text{Slope} = \frac{M}{6EIL}(-3x^2 + 6Lx - 3a^2 - 2L^2)$$

$$\text{Deflection} = \frac{M}{6EIL}(-x^3 + 3Lx^2 - 3a^2x - 2L^2x + 3La^2)$$

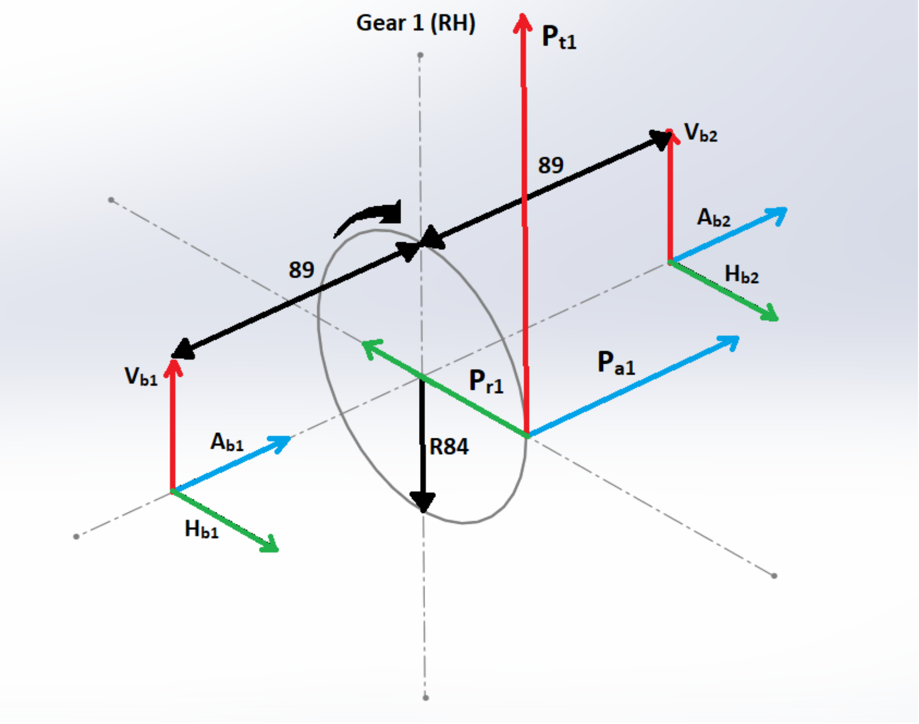
- **Following are the four possible configurations for the design of shafts:**

Configuration	**1**	**2**	**3**	**4**
Input shaft rotation	Clockwise	Anti-clockwise	Clockwise	Anti-clockwise
Gear 1	RH	RH	RH	RH
Gear 2	LH	LH	LH	LH
Gear 3	RH	RH	LH	LH
Gear 4	LH	LH	RH	RH

- For configurations 1 & 2, the axial forces on the intermediate shaft are in the same direction leading to high net force on the bearings whereas the axial forces are in the opposite directions for the configurations 3 & 4.
- Hence, configurations 3 & 4 were considered for the design of shafts.

2.2 Input Shaft Design

- **Design procedure:**



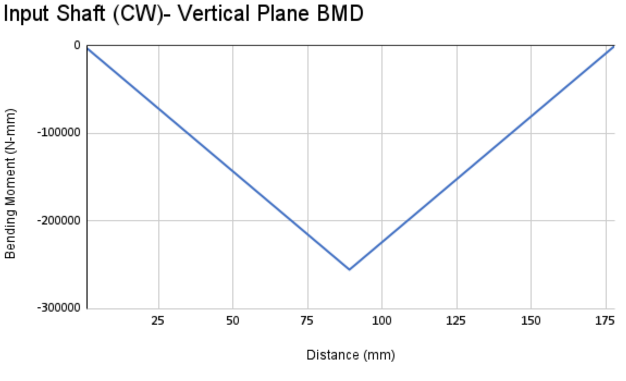
Input Shaft FBD

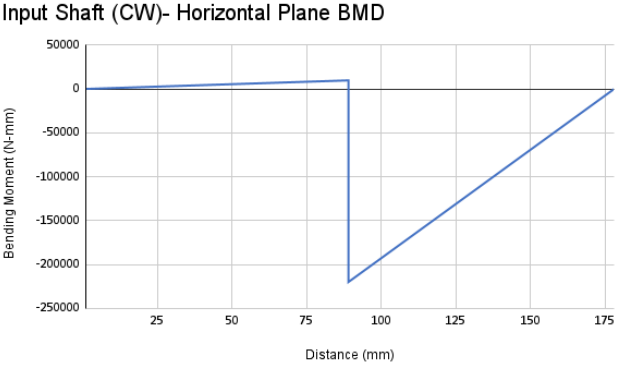
- Initially, by considering the bearing thickness and the space required for the accommodation of the key and the retaining rings; the minimum required length of the shaft was calculated to be 160 mm.
- The force analysis of the shaft gave us the values of moments acting on the shaft and the reactions on the bearings.
- By plugging the values of permissible shear stress, maximum equivalent torsional moment and C = 0.5 into the stress equation; the minimum required diameter of the shaft was calculated to be 37.77 mm
- Then the shaft of this diameter was checked against the limiting values of angle of twist/length, deflection and slope and the shaft satisfied the design constraints.
- After considering the depth of the key and the alignment of all the shafts the input shaft length was increased to 178 mm and minimum diameter was calculated to be 39.23 mm for C = 0.6 Excel.
- The calculations for the final safe design considering the forces on the bearings are provided in this Excel.

- **The final dimensions for the input shaft are as follows:**

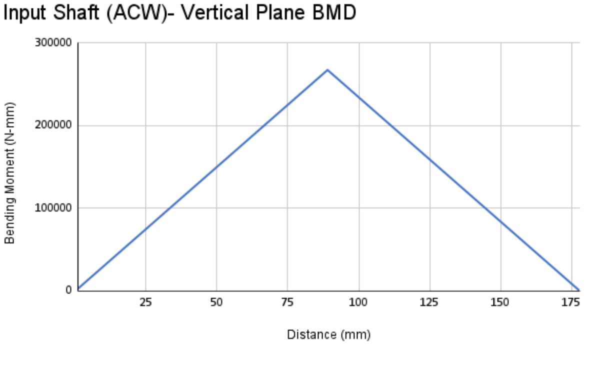
L = length of the shaft (mm)	178
C = diameter ratio	0.6
d_0 = outside diameter (mm)	50
d_i = inside diameter (mm)	30
M_b = bending moment (N-mm)	346171.725
M_t = torsional moment (N-mm)	492542.6659
angle of twist/length < 3˚/m	Pass
concentricity < +/- 0.125 mm	Pass
alignment < +/- 0.001 rad	Pass
shaft diameter tolerance	$\pm$0.025 mm

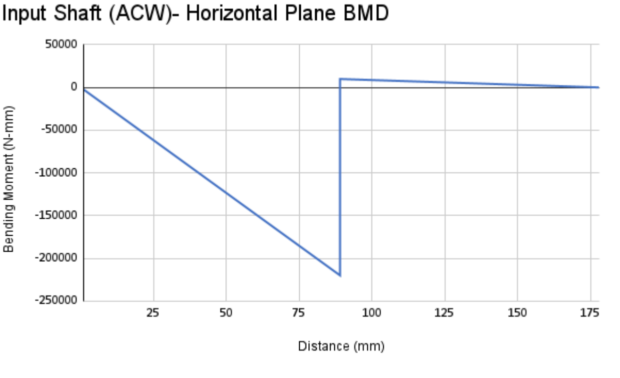
- **BMD for configuration 3:**





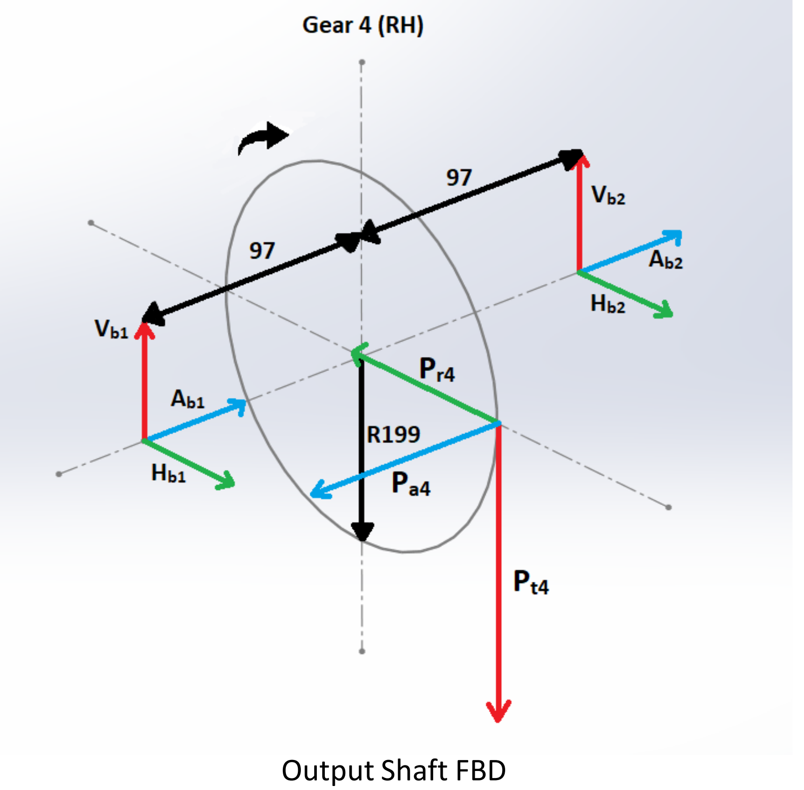
- **BMD for configuration 4:**





2.3 Output Shaft Design

- **Design procedure:**



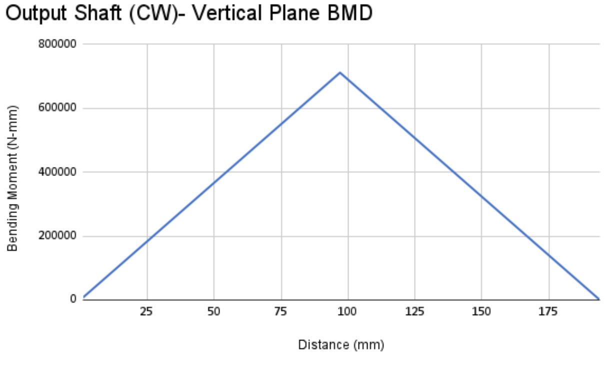
Output Shaft FBD

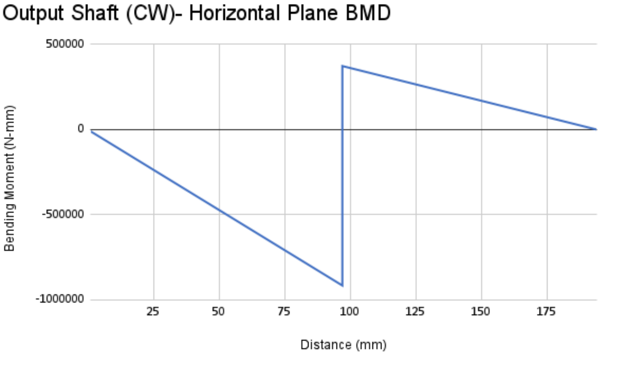
- Initially, by considering the bearing thickness and the space required for the accommodation of the key and the retaining rings; the minimum required length of the shaft was calculated to be 160 mm.
- The force analysis of the shaft gave us the values of moments acting on the shaft and the reactions on the bearings.
- By plugging the values of permissible shear stress, maximum equivalent torsional moment and C = 0.5 into the stress equation; the minimum required diameter of the shaft was calculated to be 63.31 mm
- Then the shaft of this diameter was checked against the limiting values of angle of twist/length, deflection and slope and the shaft satisfied the design constraints.
- After considering the depth of the key and the alignment of all the shafts the input shaft length was increased to 194 mm and minimum diameter was calculated to be 65.42 mm for C = 0.6 Excel.
- The calculations for the final safe design considering the forces on the bearings are provided in this Excel.

- **The final dimensions for the output shaft are as follows:**

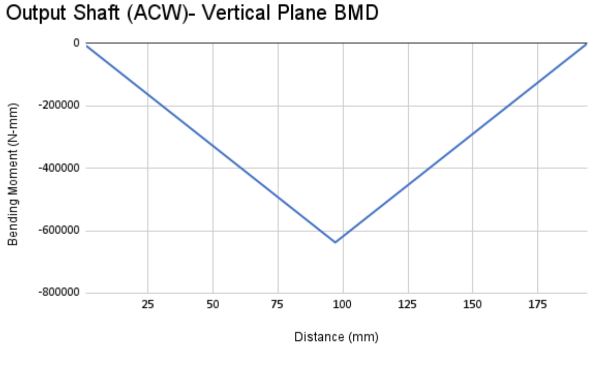
L = length of the shaft (mm)	194
C = diameter ratio	0.6
d_0 = outside diameter (mm)	90
d_i = inside diameter (mm)	54
M_b = bending moment (N-mm)	1159314.109
M_t = torsional moment (N-mm)	2762877.835
angle of twist/length < 3˚/m	Pass
concentricity < +/- 0.125 mm	Pass
alignment < +/- 0.001 rad	Pass
shaft diameter tolerance	$\pm$0.025 mm

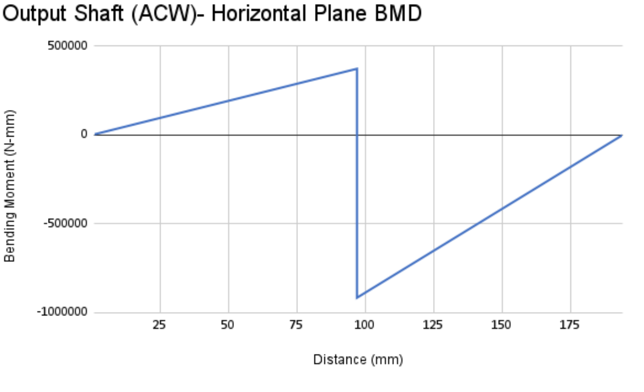
- **BMD for configuration 3:**





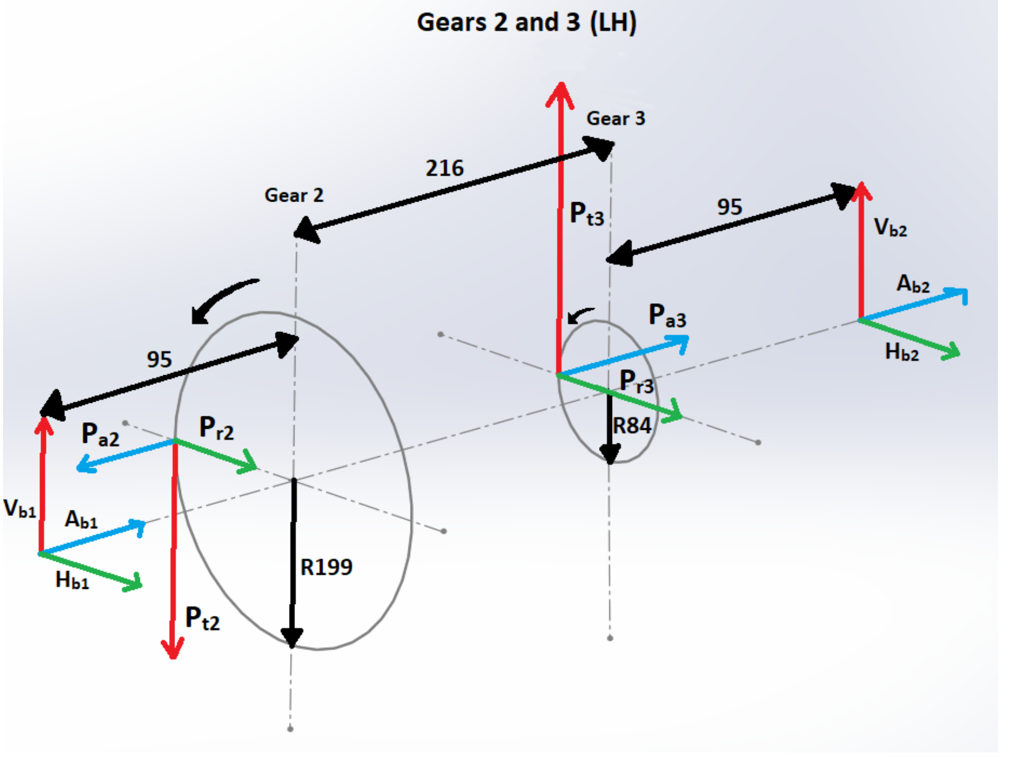
- **BMD for configuration 4:**





2.4 Intermediate Shaft Design

- **Design procedure:**



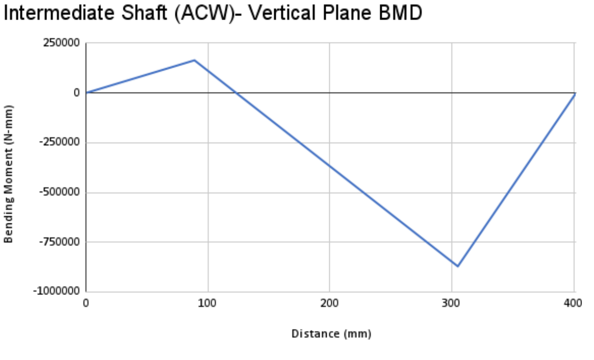
Intermediate Shaft FBD

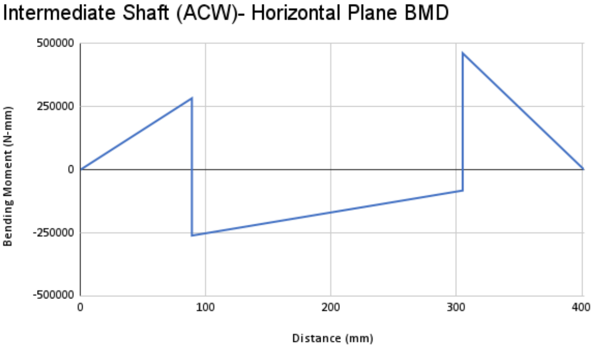
- At this stage the safe lengths of the input and the output shafts were known. Therefore considering the center distance between gear 1 and gear 4, the minimum required length of the shaft was calculated to be 375 mm.
- The force analysis of the shaft gave us the values of moments acting on the shaft and the reactions on the bearings.
- By plugging the values of permissible shear stress, maximum equivalent torsional moment and diameter ratio into the stress equation; the minimum required diameter of the shaft was calculated to be 57.81 mm.
- Then the shaft of this diameter was checked against the limiting values of angle of twist/length, deflection and slope and the shaft did not satisfy the design constraints.
- Therefore, the shaft diameter was increased to a higher value to satisfy the design constraints.
- After considering the depth of the key and the alignment of all the shafts the input shaft length was increased to 402 mm and minimum diameter was calculated to be 59.27 mm for C = 0.6 Excel.
- The calculations for the final safe design considering the forces on the bearings are provided in this Excel.

- **The final dimensions for the intermediate shaft are as follows:**

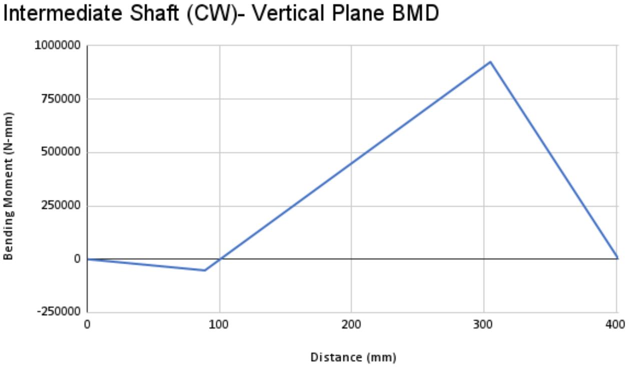
L = length of the shaft (mm)	402
C = diameter ratio	0.6
d_0 = outside diameter (mm)	80
d_i = inside diameter (mm)	48
M_b = bending moment (N-mm)	1484793.461
M_t = torsional moment (N-mm)	1166548.419
angle of twist/length < 3˚/m	Pass
deflection < +/- 0.125 mm	Pass
slope < +/- 0.001 rad	Pass
shaft diameter tolerance	$\pm$0.025 mm

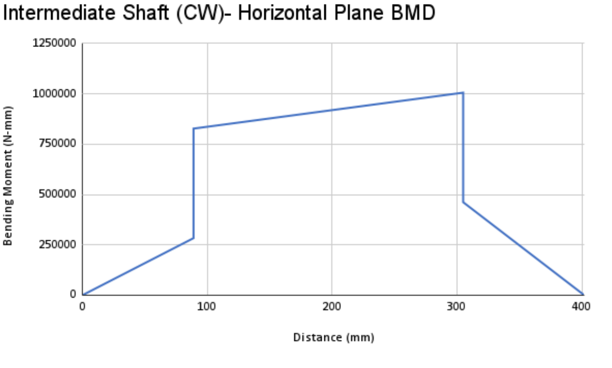
- **BMD for configuration 3:**



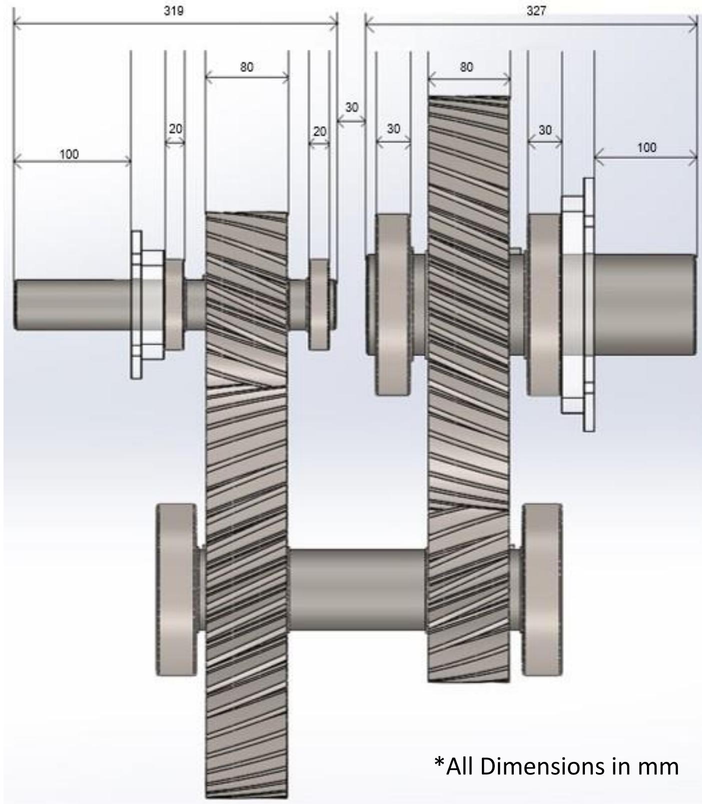


- **BMD for configuration 4:**





2.5 Shaft Layout and Summary



- **Input Shaft:**

L = length of the shaft (mm)	178
C = diameter ratio	0.6
d_0 = outside diameter (mm)	50
d_i = inside diameter (mm)	30

- **Output Shaft:**

L = length of the shaft (mm)	194
C = diameter ratio	0.6
d_0 = outside diameter (mm)	90
d_i = inside diameter (mm)	54

- **Intermediate Shaft:**

L = length of the shaft (mm)	402
C = diameter ratio	0.6
d_0 = outside diameter (mm)	80
d_i = inside diameter (mm)	48

3. Bearings

3.1 Bearing Requirements

Design Conditions:
1) The bearing life must be greater than 12000 h.
2) The bearing must be able to withstand radial and axial loads.
3) The bearing must be economical.
4) The bearing should generate low noise.

Based upon the design conditions, single row deep groove type of ball bearings was selected.

1) DGBB have high load carrying capacity
2) DGBB can take both radial as well as axial loads.
3) DGBB generate less heat and frictional losses because of point contact between the balls and races.
4) DGBB generate less noise due to point contact.

- For each shaft, two bearings of same configuration are used in order to sustain the axial forces in both directions considering the bidirectional operating nature of the gearbox.
- All bearings are provided with shields on both sides, thus protecting them from contaminants, without friction losses and thereby increasing bearing lubrication life.

Reaction forces acting on bearings				
Configuration	Shaft	Radial Load 1 (kN)	Radial Load 2 (kN)	Axial Load (kN)
3	Input	2.870904384	3.78541431	2.738914807
	Intermediate	3.679882536	10.16562927	3.747988683
	Output	12.70481299	8.759812793	6.486903489
4	Input	3.889569944	3.00690619	2.738914807
	Intermediate	3.238697916	10.64982764	3.747988683
	Output	7.61710594	11.49881006	6.486903489

3.2 Selection Methodology

Bearing selection methodology:

1) The minimum outer shaft diameter was calculated.
2) Parameters of bearings having bore size greater than the minimum shaft diameter were then added in the excel sheet.
3) We rounded up the shaft diameter to the closest available Inner diameter for the bearing.
4) Only Bearings of high grades (64xx) could withstand the loads for the closest inner diameter. We observed that bearings with larger bore size could satisfy the load conditions at lower grades(62xx). Lower grades ensured that the weight and cost of the bearing was less compared to higher grades.
5) Based upon the loads acting on the bearings, basic bearing life in hours was calculated using excel.
6) Bearings which sustained the load conditions and had a basic life around 12000 hours were selected for further analysis using SKF Bearing Select tool.
7) The Life modification factor a_{SKF} can range from fractions to 50. Hence the SKF rating life was obtained from the SKF bearing select tool.
8) The SKF rating life for selected bearing grade turned out to be much higher than 12000 hrs because of the high a_{SKF} factor. So, we decided to analyze lower grade bearings with same inner diameter. It was observed that these lower grade bearings have a life close to 12000 hrs.
9) Final selection was done on the basis of bearing mass and cost.

- The **input shaft** has a minimum safe outer diameter of 50mm. Based on the basic rating life, 6310 was selected. After calculating the a_{SKF} factor, even bearing 6210 was sufficient for the required conditions. So we decided to go with bearing **6210-2Z** at both ends of the Input shaft.

- The **intermediate shaft** has a minimum safe outer diameter of 70mm. Bearing of 70mm bore, 6214 was rejected because it just missed the target SKF rating life of 12000 hrs. The higher grade 6314 fulfilled the load conditions but it had a very higher life than required. Hence, we opted to go for the bearing **6216-2Z**, which not only sustained the loads, but also was lighter and cheaper as compared to 6314.

- For **output shaft**, the minimum safe outer diameter came out to be 70mm. According to the basic rating life obtained from Excel calculations, 6316, 6218 and 6220 were feasible. Further calculations of the SKF rating life showed that all of them satisfied the bearing life requirement. Bearing 6316 was discarded owing to its higher price and weight. Out of 6218 and 6220, **6218-2Z** was finalized, based upon its compactness, less weight and less price as compared to 6220.

➢ Basic Bearing Life calculations :

Inputs needed:

- Radial forces (F_r) and axial forces(F_a) acting on the bearing
- Based upon the minimum shaft diameter, a series of bearing is assumed. Obtain the basic Static load carrying capacity (C_o)and the Basic Dynamic load Capacity (C_o)

Calculations:

- X & Y factors are determined from the table below. …….(DDHB table 15.9)
- Equivalent dynamic life= $P = XVF_r + YF_a$
- Convert Required bearing life in hours (L_{10h}) to million revolutions(L_{10})
- Required Dynamic loading $C_{req} = P(L_{10})^{1/3}$
- If C > C_{req}, the bearing series is Safe.
- Calculate the actual basic life in hours = $L_{10h}\left(\frac{C}{C_{req}}\right)^3$

3.3 Bearing Parameters

Shaft	**Input**	**Intermediate**	**Output**
Model	6210-2Z	6216-2Z	6218-2Z
Inner Diameter (mm)	50	80	90
Outer Diameter (mm)	90	140	160
Width (mm)	20	26	30
Mass (kg)	0.47	1.53	2.3
Operating RPM	1000	422.2222222	178.2716
Static load rating, C_0 (kN)	23.2	55.0	73.5
Dynamic load rating, C (kN)	37.1	72.8	101.0
Fatigue load limit, P_U (kN)	0.95	2.2	2.8
Contamination Factor (η_c)	0.48	0.47	0.32
Life modification factor (a_{SKF})*	3.67	2.32	0.56
Basic life (L_{10h})*	3990	9290	22400
SKF rating life (L_{10mh})*	14600	21500	12500
SKF Grease used	MT33	MT33	MT33
Catalogue grease life (hrs)*	15700	17800	20600
Cost (₹)	746	4178	6316

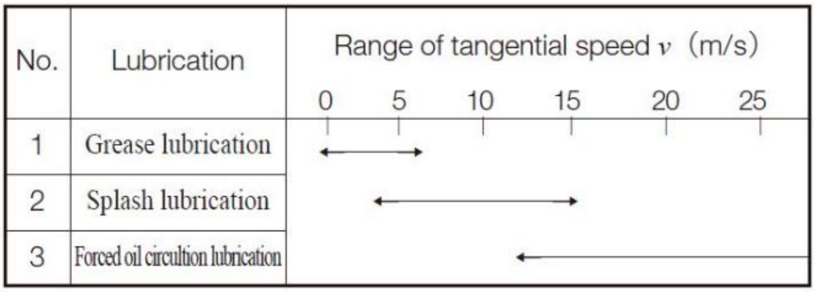
**corresponding value for the bearing subjected to the highest load among both configuration.*

4. Lubrication

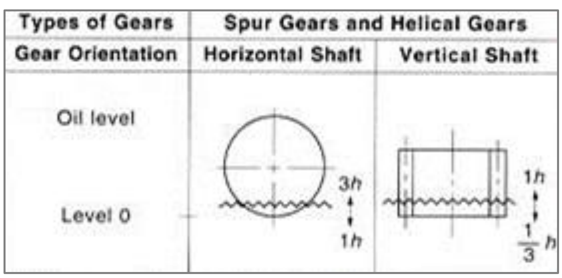
4.1 Lubrication Methods
4.2 Lubricants
4.3 Lubricant Recommendation
4.4 Lubrication life

4.1 Lubrication Methods

➢ **Gear Lubrication:**



Velocity range for chosen gears is 3.7 - 9 m/s
Temperature range is -5 to 65 deg. C
As shown in the figure above there are 3 different gear lubrication methods. Out of the given methods the most appropriate method of gear lubrication is <u>Splash Lubrication</u>. The big gear will be dipped in lubricating oil and it will carry the lubricating oil with it in each rotation to small gear, lubricating the small gear as well.
Maintaining the oil level is a priority in oil splash lubrication. There will be excess agitation loss if the oil level is too high. On the other hand, there will not be effective lubrication or ability to cool the gears if the level is too low. Table below shows guide lines for proper oil level.



Full Depth= h= 2.25*m_n = 2.25*8 = 18mm
Radial clearance between gear face from bottom of housing= 15mm
Minimum oil level= h+15 = 18+15 = 33mm
Minimum oil level= 3*h+15 = 3*18+15 = 69mm

➢ **Bearing Lubrication:**

Grease lubrication was chosen over oil lubrication because:

1) Grease is easily retained in the bearing and housing.
2) No need of complicated sealing arrangements compared with those for oil lubrication.
3) Operation for long periods without paying attention can be achieved.
4) Grease lubrication is cost effective.
5) No need for separate oil feed system or machining of oil galleries. Bearings are too small and don't reach oil level for successful oil splash lubrication.

All bearings are provided with shields on both sides, thus protecting them from contaminants. They are available pre-greased with SKF grease MT33.

4.2 Lubricants

➢ Lube Oil Selection

$$V_{40} = \frac{7000}{\sqrt{V_1}}$$

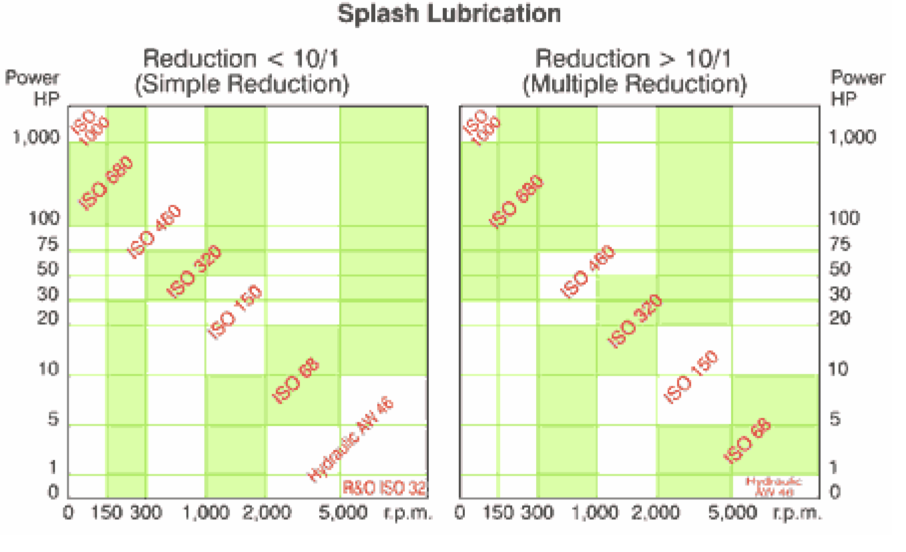
- V40= Kinematic viscosity @40 C, cSt
- V1 = Pitch line velocity of lowest speed gear, feet per minute (fpm)
  = 0.262*(pinion rpm)*(pinion diameter)

According to this formula considering velocity for pair 1-2,

- V1= 10.977 m/s= 2160.82677 fpm, we get,
- V40= 150.5872 cSt. Since we are not using oil cooler and the operating temperature exceeds 50°C, we selected **ISO grade 320 Grade 2** according to guidelines provided

Former R&O Gear Lubes AGMA Grade	Former EP Gear Lubes AGMA Grade	Equivalent ISO Grade	Viscosity Range cSt @ 40°C	Viscosity Range SUS @ 100°F
1		46	41.40-50.60	193-235
2	2EP	68	61.20-74.80	284-347
3	3EP	100	90-110	417-510
4	4EP	150	135-165	626-765
5	5EP	220	198-242	918-1122
6	6EP	320	288-352	1335-1632
7 COMP	7EP	460	414-506	1919-2346
8 COMP	8EP	680	612-748	2837-3467
8A COMP	8A EP	1000	900-1100	4171-5098

This was verified by the following chart:



Power= 49 kW = 65.71 hp. Reduction < 10/1. RPM = 422.22 to 1000 rpm
Thus it is verified that lube oil **ISO VG 320 Grade 2** is suitable for our application.

4.3 Lubricant Recommendation

➢ Gear Lubrication:

JIS Gear Oils			IDEMITSU	COSMO OIL	JAPAN ENERGY	SHOWA SHELL	ENEOS	MOBIL
For Industrial Usage	1	ISO VG 32	Daphne Super Multi Oil 32	NEW Mighty Super 32 Cosmo Allpus 32	JOMO Lathus 32	Shell Tellus Oil C 32	Super Mulpus DX32	Mobil DTE Oil Light
		ISO VG 68	Daphne Super Multi Oil 68	NEW Mighty Super 68 Cosmo Allpus 68	JOMO Lathus 68	Shell Tellus Oil C 68	Super Mulpus DX68	Mobil DTE Oil Heavy Medium
		ISO VG 100	Daphne Super Multi Oil 100	NEW Mighty Super 100 Cosmo Allpus 100	JOMO Lathus 100	Shell Tellus Oil C 100	Super Mulpus DX100	Mobil DTE Oil Heavy
		ISO VG 150	Daphne Super Multi Oil 150	NEW Mighty Super 150	JOMO Lathus 150	Shell Tellus Oil C 150	Super Mulpus DX150	Mobil Vacuoline 528
	2	ISO VG 100	Daphne Super Multi Oil 100	Cosmo Gear SE100 Cosmo ECO Gear EPS100	JOMO Reductus 100	Shell Omala Oil 100	Bonnoc AX M100 Bonnoc AX AX100	Mobil gear 600 XP 100
		ISO VG 150	Daphne Super Multi Oil 150	Cosmo Gear SE150 Cosmo ECO Gear EPS150	JOMO Reductus 150	Shell Omala Oil 150	Bonnoc AX M150 Bonnoc AX AX150	Mobil gear 600 XP 150
		ISO VG 220	Daphne Super Multi Oil 220	Cosmo Gear SE220 Cosmo ECO Gear EPS220	JOMO Reductus 220	Shell Omala Oil 220	Bonnoc AX M220 Bonnoc AX AX220	Mobil gear 600 XP 220
		ISO VG 320	Daphne Super Gear Oil 320	Cosmo GearSE320 Cosmo ECO Gear EPS320	JOMO Reductus 320	Shell Omala Oil 320	Bonnoc AX M320 Bonnoc AX AX320	Mobil gear 600 XP 320
		ISO VG 460	Daphne Super Gear Oil 460	Cosmo Gear SE460 Cosmo ECO Gear EPS460	JOMO Reductus 460	Shell Omala Oil 460	Bonnoc AX M460 Bonnoc AX AX460	Mobil gear 600 XP 460
		ISO VG 680	Daphne Super Gear Oil 680	Cosmo Gear SE680	JOMO Reductus 680	Shell Omala Oil 680	Bonnoc AX M680 Bonnoc AX AX680	Mobil gear 600 XP 680

The above recommendation table was taken from this site .

From the recommendations in the above table, different oils were compared for our required grade and **Shell Omala S2 G 320** was found to be the best fit and was chosen.

- Detailed Technical information
- Safety information is provided

Advantages of the oil:

- Long oil life – Maintenance saving
- Excellent wear and Corrosion protection
- Maintaining system efficiency
- Can be used for highly loaded gears
- Good for Enclosed gear system.

➢ Bearing Lubrication:

Bearings are pre-lubricated with **SKF MT33** grease. The conformity of this grease was cross-checked with the SKF Traffic Light Concept.

4.4 Lubrication life

➢ **Oil Lubrication Life:**

Assuming no contamination effect, a lubricant's life is given by:

$$\log_{10} L = k_1 + \frac{4750}{(T + 273)}$$

Where,
L= Lubricant's life in hours
T= Oil operating temperature in °C
K1= Oil-type factor (selected from the table below)

Oil Type	k_1	Max Temperature for 1,000-hour Life
Uninhibited (used in once-through systems)	-10.64	75°C
Extreme-pressure gear lubricant	-10.31	84°C
Hydraulic	-8.76	99°C
Turbine	-8.45	106°C
Heavily refined, hydrocracked	-8.05	121°C

Considering the use of extreme- pressure gear lubricant, k1= -10.31 and operating temperature= 60°C, we get Lubricant's life as 9000.450842 hours, which satisfies the requirement of minimum life of 8000 hours.
Oil Life= **9000 hours**

➢ **Bearing lubrication life:**

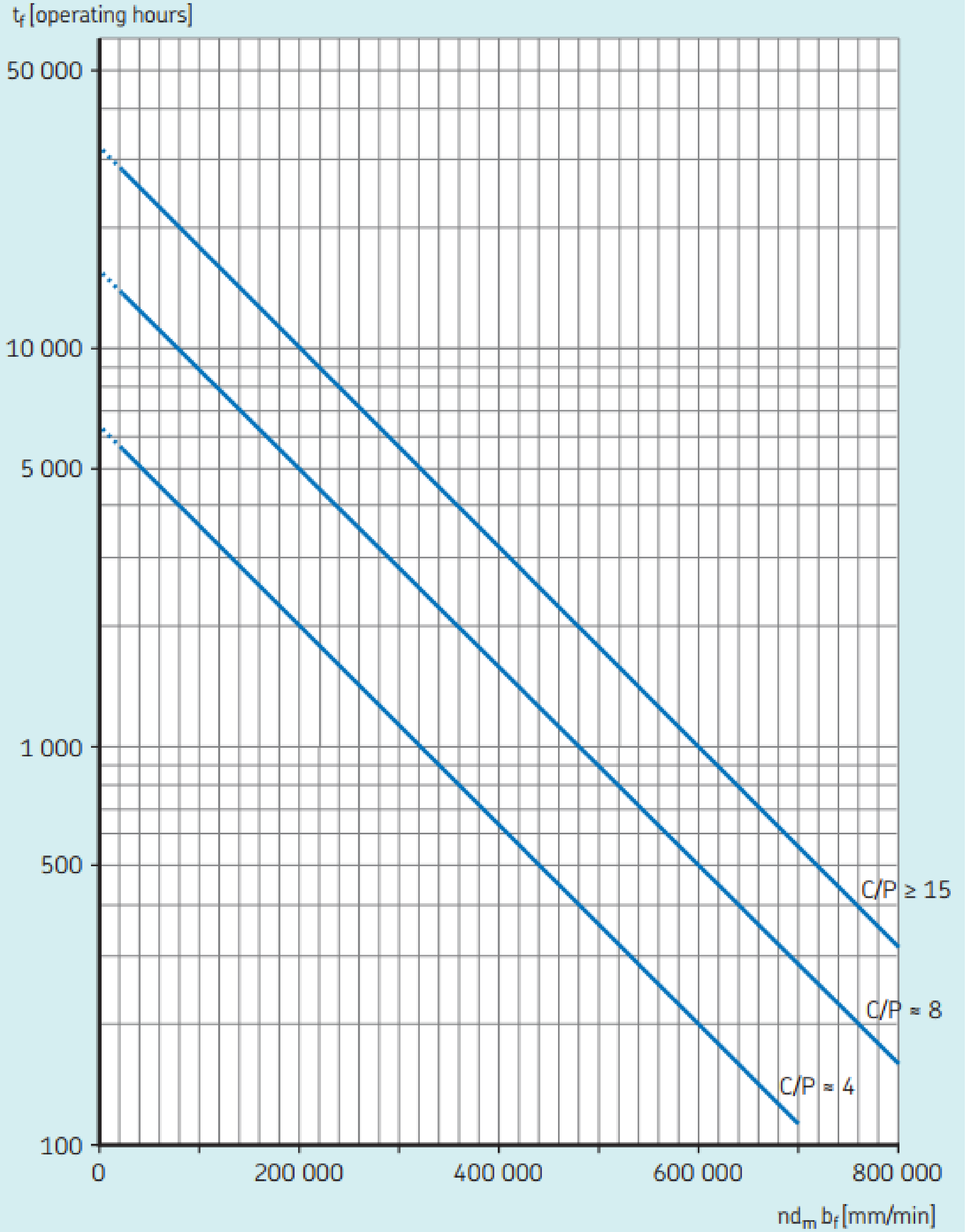
Grease used: SKF MT 33
The grease life was calculated with the help of chart and with the 'SKF Bearing Select' application.

Bearing	6210-2Z	6216-2Z	6218-2Z
Speed (rpm)	1000	422.22	178.27
D (mm)	90	140	160
d (mm)	50	80	90
Mean Diameter	70	110	125
Speed factor	70000	47100	22300
C/P	6.21	6.18	6.21
Grease Life (hrs.)*	15700	17800	20600

**corresponding value for the bearing subjected to the highest load among both configuration.*

Diagram 2

Relubrication intervals at operating temperatures of 70 °C *(160 °F)*



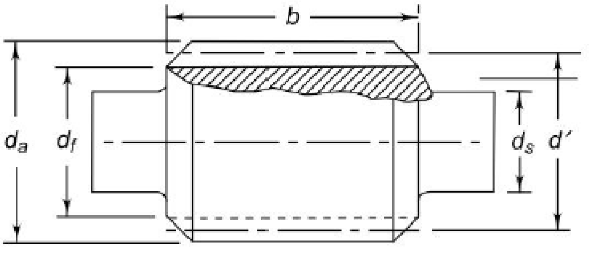
5. Keys & Retaining rings

5.1 Key Selection

Keys were chosen to connect the shaft with gears since the condition for integral gear design is not satisfied over here.

Condition : $d_f < d_s + (d_s/2)$

Where :
d_f is diameter of dedendum circle
d_s is diameter of the shaft



Integral Gear

Taper is used since it satisfies strength requirements and to avoid axial moment in one direction by use of key and in other direction by retaining rings.

Material	Medium-carbon steel (AISI 1040)
Heat-Treatment	Tempered
Ultimate tensile strength	620 MPa
Yield tensile strength	420 MPa

- **Keys selected:**

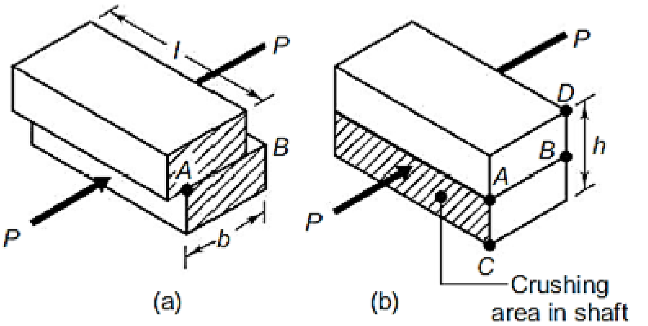
FOS of 3 is considered for keys, since the required length for all keys were coming out to be less than the minimum length according to Design Data book table 9.13 minimum length was chosen.

Shaft	Shaft Diameter (mm)	Key Breadth (mm)	Key Height (mm)	Min length (mm)
Input	50	14	9	40
Intermediate	80	22	14	63
Output	90	25	14	70

Key	Depth (mm)		Keyway Radius (mm)		Chamfer for keys (mm)	
	In Shaft	In Hub	Min.	Max.	Min.	Max.
14*9	5.5	2.9	0.25	0.4	0.4	0.6
22*14	9	4.4	0.4	0.6	0.6	0.8
25*14	9	4.4	0.4	0.6	0.6	0.8

5.2 Key Design Calculations

For key design the required length had to be found out using the provided breadth and height for a particular shaft diameter (from Design Data book table 9.13) and the chosen material.



Failure of Key: (a) Shear Failure
(b) Crushing Failure

Shear stress:

$$\tau = \frac{2M_t}{dbl} \qquad \tau = \left(\frac{0.5*\text{Yield strength}}{3}\right)$$

Compressive stress:

$$\sigma_c = \frac{4M_t}{dhl} \qquad \sigma_c = \left(\frac{0.9*\text{Yield strength}}{3}\right)$$

Shaft	Shaft dia. [d] (mm)	Breadth [b] (mm)	Height [h] (mm)	Torque [M_t] (N-mm)	Min. calculated length [l] (mm)
Input	50	14	9	492542.6659	34.74727802
Intermediate	80	22	14	1166548.419	33.06543139
Output	90	25	14	2762877.835	69.6114345

- Minimum required length based on either compressive or shear stress was selected.

<u>Final Dimensions of Selected Keys are (b*h*l):</u>

Input: 14*9*40
Intermediate: 22*14*63
Output: 25*14*70

❖ Detailed calculations are present in Excel

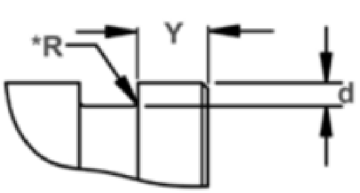
5.3 Retaining Rings

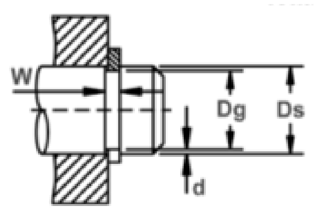
- Parameters considered while choosing retaining ring over creating a shoulder on the shaft were axial force that is applied on the individual shaft and reduction in machining requirements.
- Tapered section - Axially installed retaining rings – Cheap but difficult to install.
  Tapered section - Radially installed retaining rings – Costly but easy to install
- Giving priority to cost over ease of installation **Tapered section - Axially installed retaining rings** were selected.

Retaining rings were chosen from data sheet provided by Rotor Clip.

Material of ring is **Stainless Steel - PH 15-7 Mo** as chosen from the Catalogue. It is an extra strength corrosion-resistant steel, capable of preventing atmospheric oxidation at temperatures up to 900º F. It also offers the following advantages:

1. Minimal distortion due to unique heat-treating process.
2. A minimum of 225,000 psi for high ultimate tensile strength.
3. High creep strength





Parameter for modification of shaft for installation of retaining rings:

Shaft		Input	Intermediate	Output
Shaft Diameter (Ds)		50	80	90
Grove Size	Diameter (Dg)	46.75	76.2	86.15
	Width (W)	2.15	2.65	3.15
	Radius (*R)	2	3	3
Edge Margin (Y)		4.5	5.3	5.3

*All Dimensions in mm

6. Design Considerations

Economic
Environmental and Sustainability
Social and Cultural
Future endeavors
Legal and Ethical
Safety and Health
Manufacturability
Ergonomics

6.1 Considerations

- **Economic**

- Each part's dimension, material and other specifications were chosen to achieve required FOS along with minimal cost.
- For Off-the-shelf items many variations were compared and most economic options were chosen.
- Gears and Shafts were designed considering ease of manufacturability which in turn reduced the cost
- Over design was avoided and in turn unnecessary expenditure was reduced.
- A detailed cost report is provided here.

- **Environmental and Sustainability**

- A biodegradable grease is chosen for bearing lubrication.
- Provisions for avoidance of spillage and any other kind of environmental contact are made.
- Most of the metals used are recyclable.

- **Social and Cultural**

- To support Make in India moment many of the parts were designed so that they can be manufactured indigenously instead of importing them.
- Employment opportunities for local workers can be created in rural areas due to the indigenous nature of manufacturing.
- Training programs to create skilled labor are encouraged as the manufacturing processes require some skill.

- **Legal and Ethical**

- All the practices followed in the construction of the gearbox are in conformity with legal guidelines provided by Govt.
- As specified by Engineering code of Ethics "Hold paramount the safety, health, and welfare of the public", we have designed our gearbox keeping the health and safety of the user and handler in mind.
- All the data provided in this report is verified and true.

- **Safety and Health**

- No toxic materials have been used in the gearbox.
- The design is made such that it avoids any harm due to spillage of fluids.
- The oil is used over here is unlikely to present any significant health or safety hazard when properly used in the recommended application and good standards of personal hygiene are maintained.
- All individual parts are provided with sufficient FOS to avoid any kind suddenly failure and harm occurring due to it.
- All the parts that pose danger to the user are covered inside a hard casing to protect the user in any kind of failure.
- It was ensured that the noise was < 85 dB from 1 m.

- **Manufacturability**

- It is taken care that all materials used in designing are readily available and easy to machine.
- The processes that are used for manufacturing are the standard processes used in industry, to ensure low cost of parts and being able to be manufactured readily.
- Selection of same material for gears, shafts & keys for manufacturing convenience
- Selection of recommended parts from manufacturer's catalogue were chosen to reduce the need for custom made parts.

- **Ergonomics**

- The drainage plug is provided in such a way that no oil is left inside the gearbox while draining.
- A glass window is provided at the top of the gearbox for easy inspection.
- The design ensure ease of dismantling of the gearbox in case on any damage.
- Dip stick is provided for easily checking of oil level.

- **Future endeavors**

- Suggestions for incorporation of IOT are also give to create a better and comfortable monitoring system of the gearbox.

7. IOT and Industry 4.0 Integration

7.1 IOT and Industry 4.0 Integration

IOT refers to the inclusion of sensors and wireless communication in appliances/ machines to allow for seamless data exchange and collection.

Integration of IOT systems and Industry 4.0 technology in an industrial gearbox poses many potential benefits. The objectives of this integration may be broadly categorized as follows:

1) Remote surveillance of gearbox parameters
2) Predictive maintenance
3) Data storage and collection for analysis and improvement of the system over time

To achieve these objectives, relevant data must first be collected using **sensors**, then communicated to a **processing unit** either on a cloud server or local to the machine in the form of microcontrollers like Raspberry Pi, Arduino, etc. Subsequently, said data must be interpreted and transmitted as may be appropriate through **wireless communication** channels. This is carried out through various means including simply the internet for long range transmission and Bluetooth or Wi-Fi for short range.

Some measurable parameters that convey relevant characteristics of the health and life of a gearbox are as follows:

1) Oil Temperature
2) Oil Level
3) Oil Viscosity
4) Noise Characteristics
5) RPM

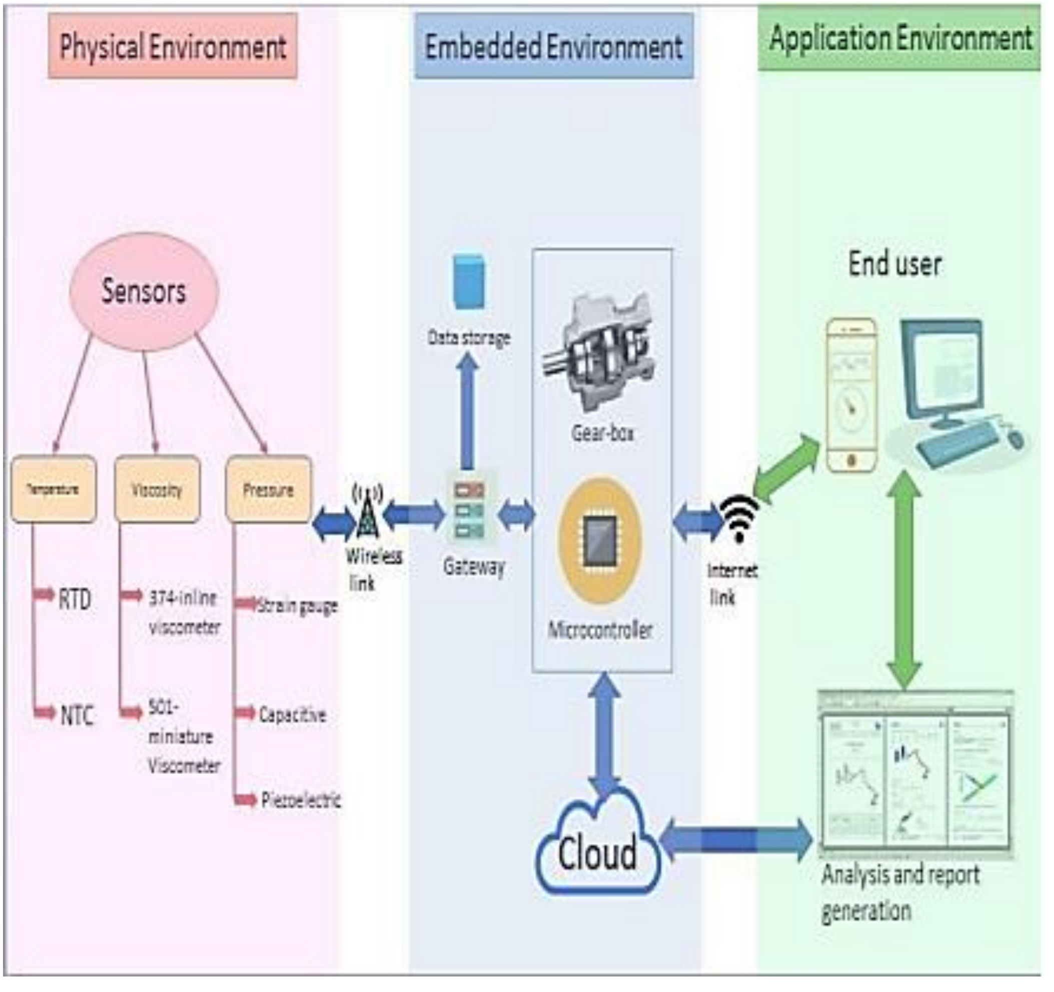
These parameters can be measured with IOT framework compatible sensors.
ex: 374 inline Viscometer, 501-Miniature Viscometer for viscosity; RTDs (Resistance Temperature Detectors) and NTCs (Negative Temperature Coefficient) for temperature; tachometers for RPM

Data from these can be collected over time to develop a knowledge base which may then be analyzed via machine learning techniques to establish functional mapping between sensor values and maintenance requirements

More directly, these sensors can be used to detect values of variables like oil levels and automatically notify relevant personnel to carry out maintenance procedures through the use of wireless communication and data syncing over multiple devices.

Acoustic emission (AE) testing using sound sensors also seems to show promise in predicting signs of failure and cracks in particular.

A block diagram for an IOT enabled gearbox proposed in the open access paper "Harish, S., Sumukh, M. S., Narayan, S. S., & Usha, B. S. IoT Enabled Industrial Gearbox." can be seen on the next page



IC: Harish, S., Sumukh, M. S., Narayan, S. S., & Usha, B. S. IoT Enabled Industrial Gearbox.

8. Dimensions, Sheets & Models

8.1 Dimensions
8.2 All Views of model
8.3 Exploded View with BOM
8.4 General Drawing Assembly
8.5 Intermediate Shaft 2D Sheet
8.6 Gearbox Equipments

8.1 Dimensions

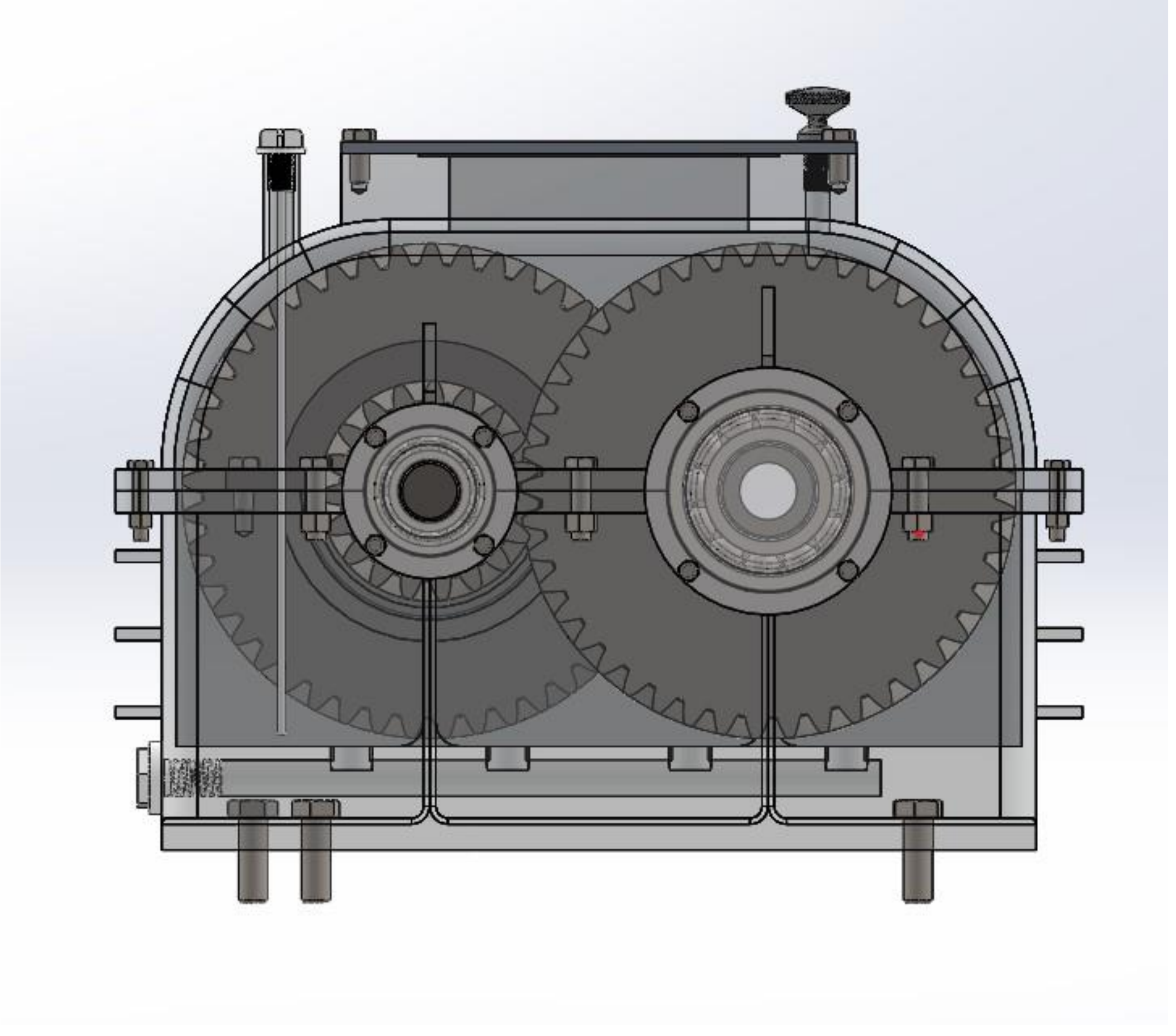
Center Distance between Gears (a)	282.465
Diameter of gear (d_g)	397
Width of the Gearbox (w)	714
Height of Gearbox (h)	300
Diameter of foundation bolts (d_{st})	24
Diameter of bearing bolts (d_1)	16
Diameter of bolts for securing cover and housing (d_2)	12
Diameter of bolts for bearing cap (d_3)	10
Distance of foundation bolt axis from housing wall (C)	32
Distance of bolt (d_2) axis from housing wall (C_1)	21
Width of foundation flange (K)	58
Width of flanges between housing and cover (K_1)	38
Thickness of housing wall (t)	12
Thickness of ribs (t_r)	10
Thickness of cover wall (t_c)	11
Thickness of flanges between housing and cover (t_{f1})	18
Thickness of foundation flanges (t_{f2})	27
Axial clearance between gear side and protruding inner elements of housing (Δ_1)	10
Radial clearance between gear face from bottom of housing (Δ_2)	15
Weight	329.316 Kg

Note:

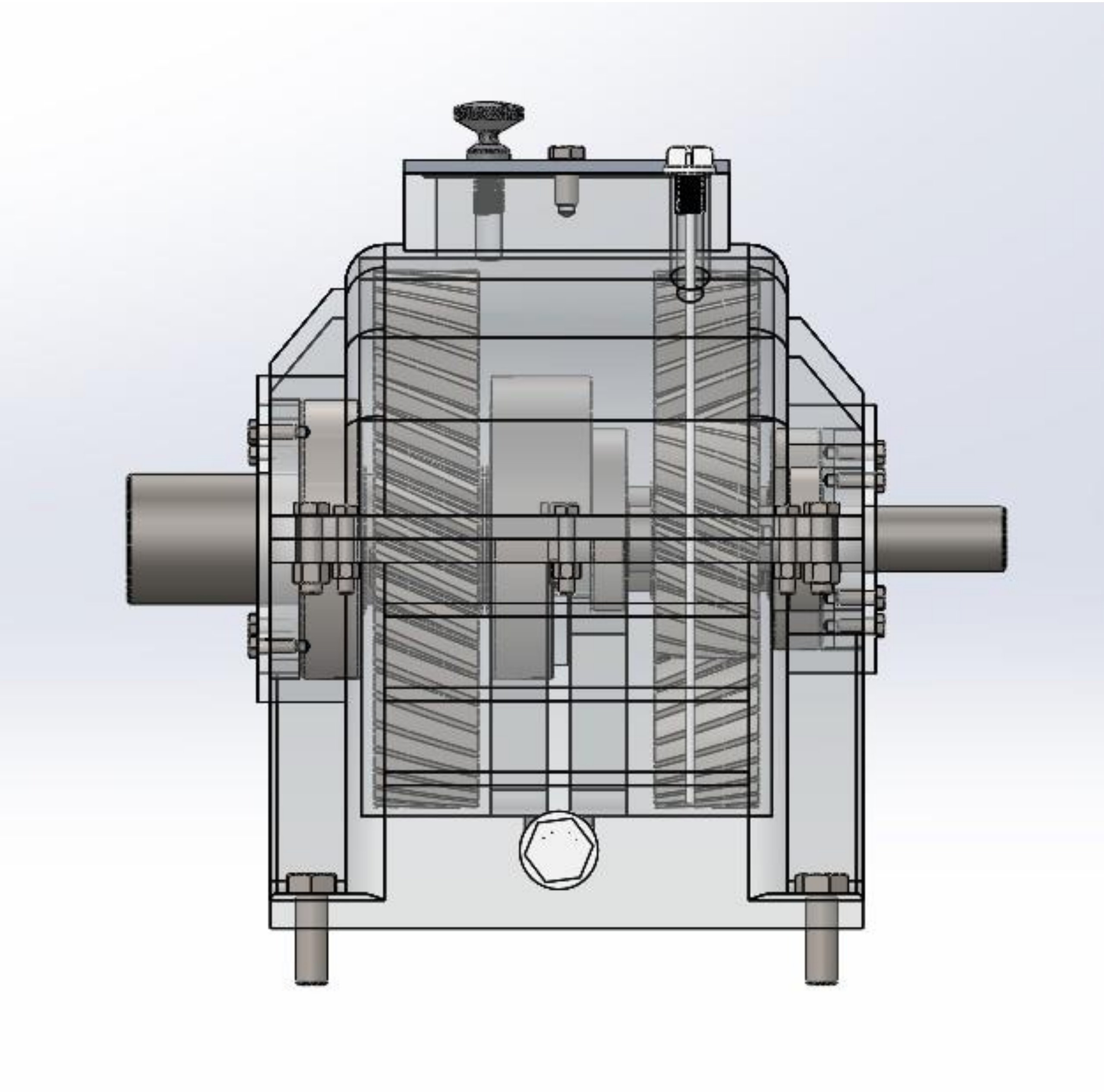
- All Dimension in mm.
- All the Dimensions are taken from DDBT 24.5.
- Materials related to all parts are mentioned in the BOM.

8.2 All Views

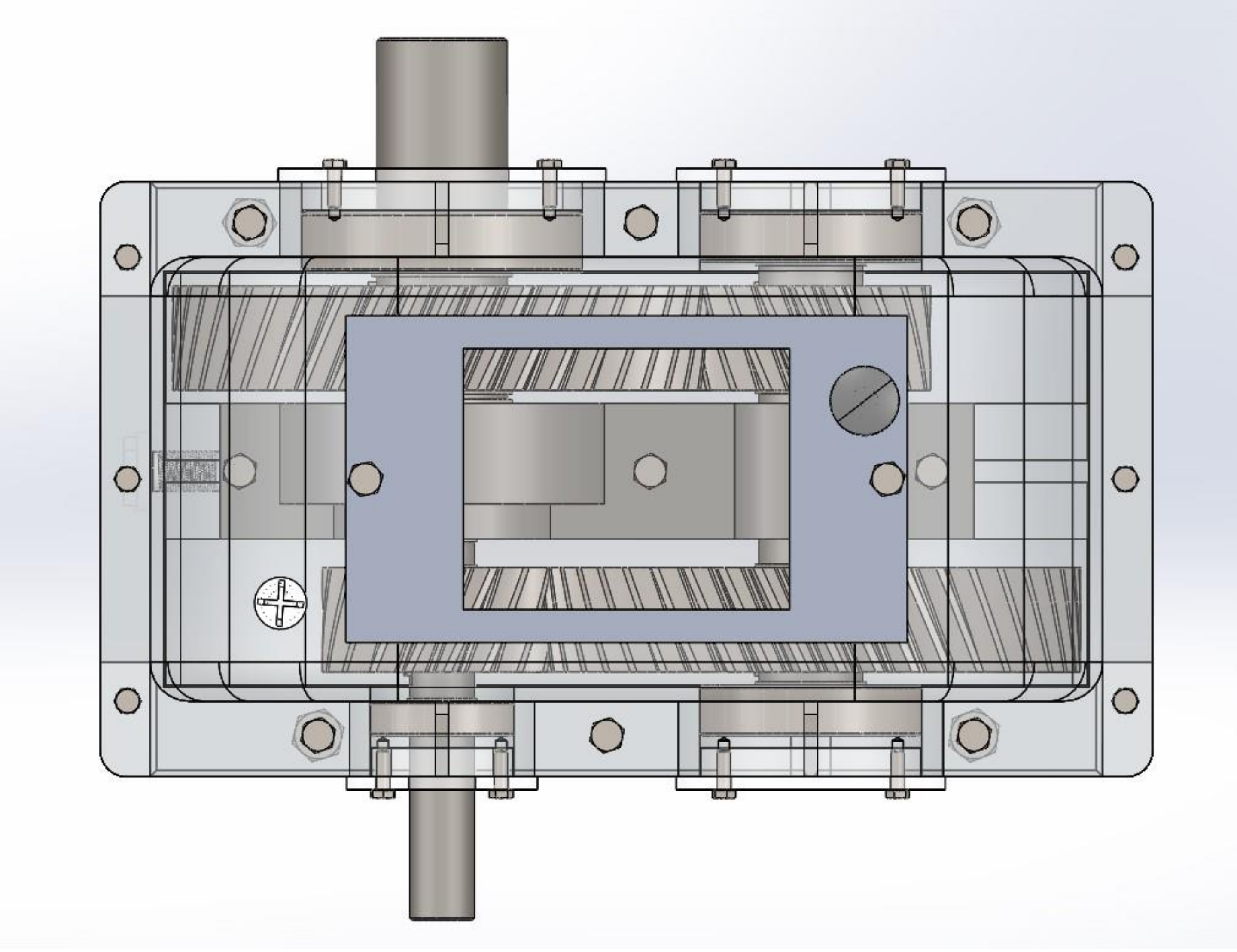
8.2.1 Front View



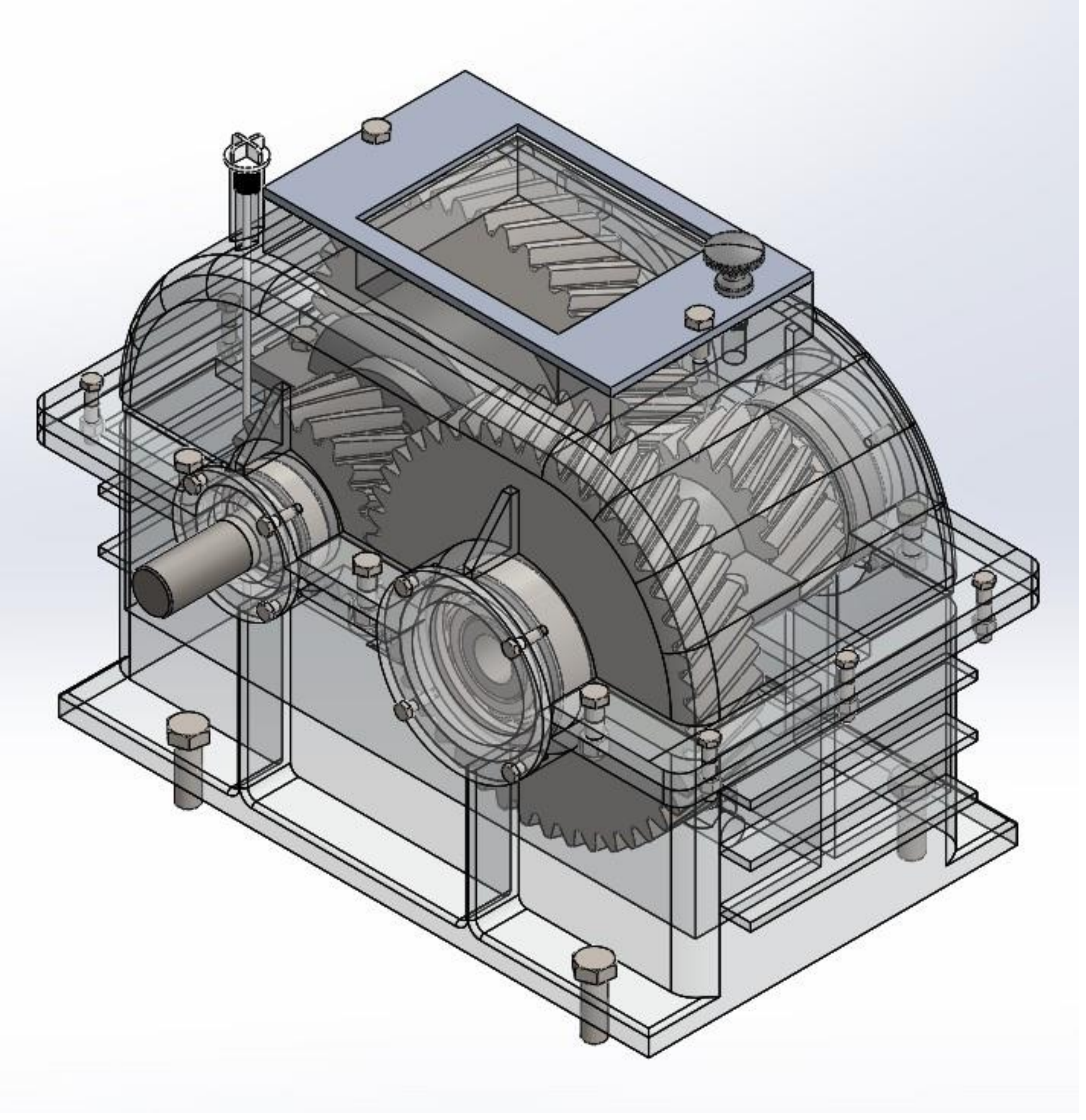
8.2.2 Side View



8.2.3 Top View

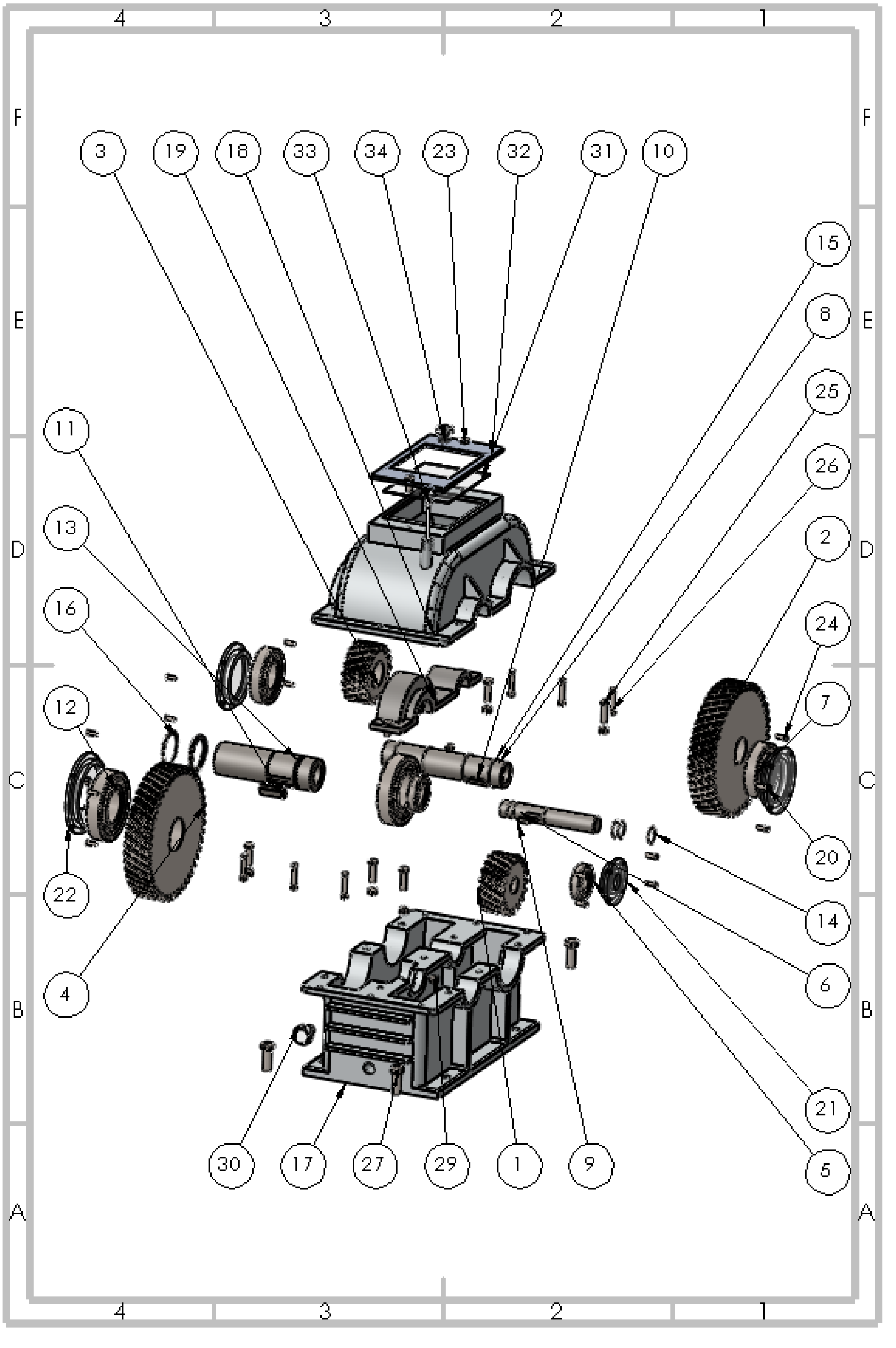


8.2.4 Isometric View



8.3 Exploded View with BOM

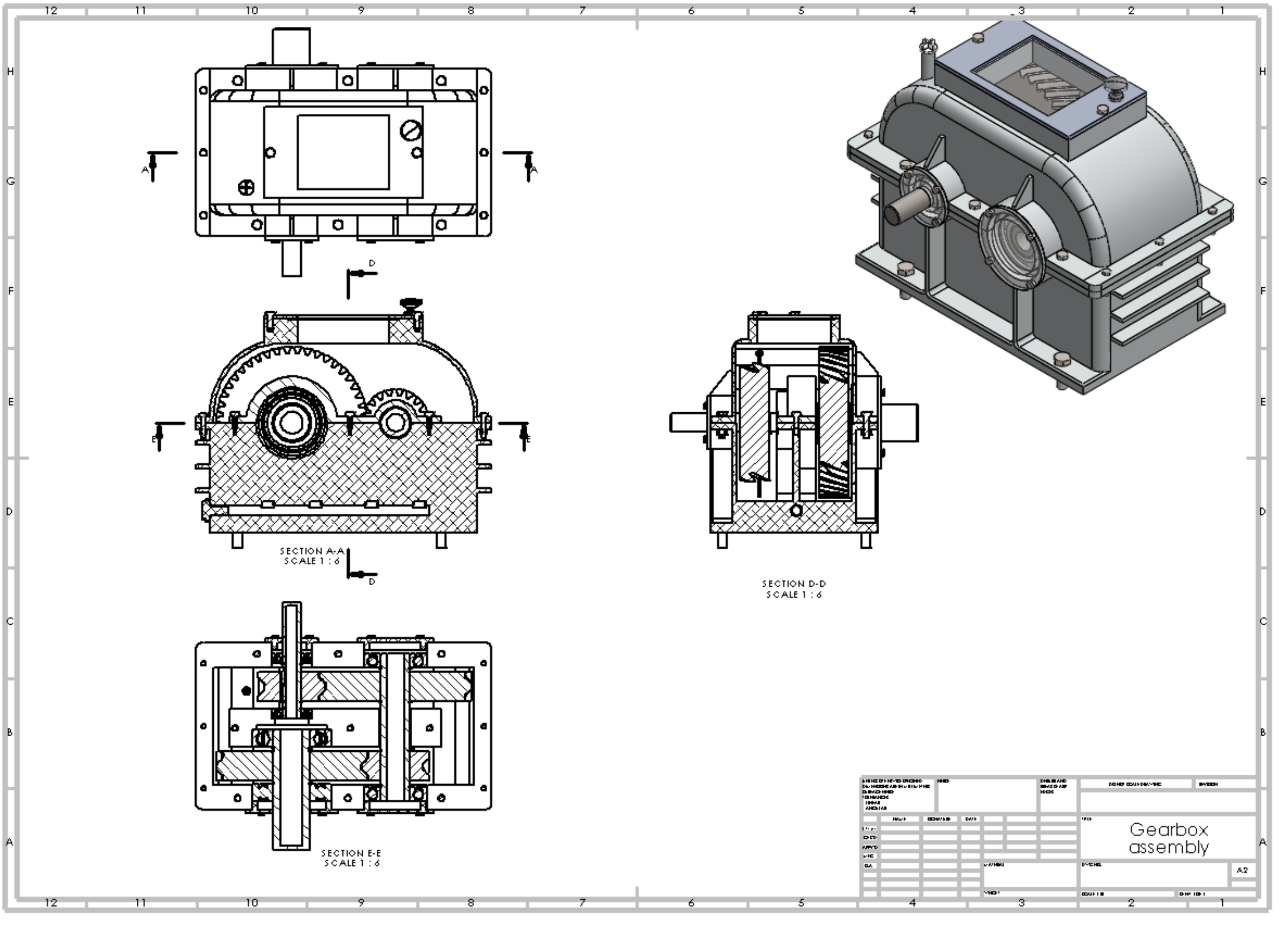
- Exploded view PDF.
- Exploded View animation.



ITEM NO.	PART NUMBER	Material	QTY.
1	Gear 1	0.55% carbon chromium steel	1
2	Gear 2	0.55% carbon chromium steel	1
3	Gear 3	0.55% carbon chromium steel	1
4	Gear 4	0.55% carbon chromium steel	1
5	Input bearing	-	2
6	Input shaft	AISI 1040	1
7	Intermediate bearing	-	2
8	Intermediate shaft	AISI 1040	1
9	Key (Gear 1)	AISI 1040	1
10	Key (Gear 2 and 3)	AISI 1040	2
11	Key (Gear 4)	AISI 1040	1
12	Output bearing	-	2
13	Output shaft	AISI 1040	1
14	Retaining rings (Input)	-	3
15	Retaining rings (Intermediate)	-	4
16	Retaining rings (Output)	-	3
17	Lower casing	AL 6061	1
18	Upper casing	AL 6061	1
19	Bearing housing	AL 6061	1
20	Intermediate Bearing lid	Acrylic	2
21	Input Bearing lid	Acrylic	1
22	Output Bearing lid	Acrylic	1
23	M16 (L35)	-	5
24	M10 (L30)	-	16
25	M16 (L60)	-	6
26	M12 (L60)	-	6
27	M24 (L70)	-	4
28	M12 Nut	-	6
29	M16 Nut	-	6
30	Oil plug	PVC	1
31	Glass	Glass	1
32	Glass cover	AL 6061	1
33	Dip stick	PVC	1
34	Air vent	Polyamide (Nylon 6)	1

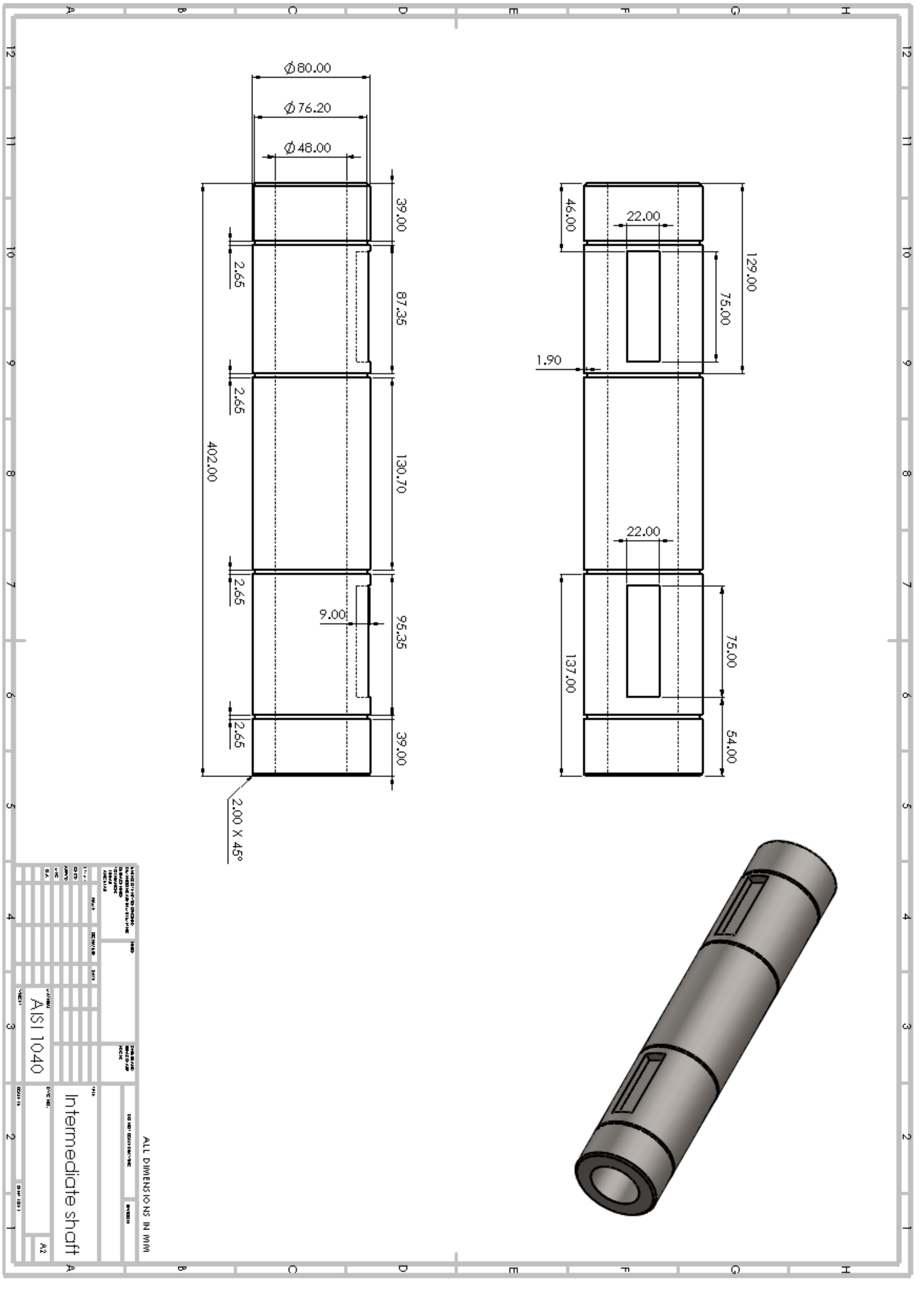
8.4 General Assembly Drawing

- General Assembly Drawing PDF.



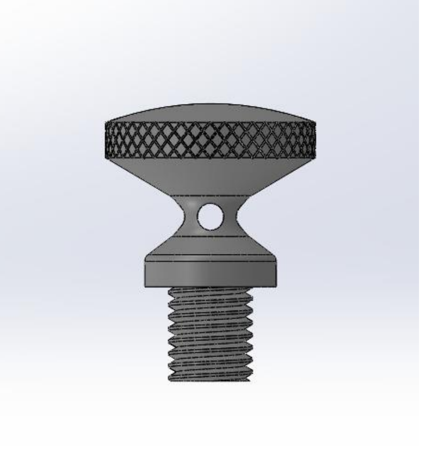
8.5 Intermediate Shaft 2D Sheet

- [Intermediate Shaft PDF]()



8.6 Gearbox Equipments

Breather



- During the operation, the pressure inside the housing increases due to heating of oil and air as a result of frictional heat. This causes the lubricating oil to be ejected outside the housing through seals and flanges in the form of oil mist.
- Breather allows the hot air to escape and cool air to enter into the transmission to prevent overheating issue.
- Breathers are provided in top portion of the cover for communication between the inner space of housing and the surrounding. This keeps the pressure inside the housing under control.

Dip stick

- The level of lubricating oil in gearbox would be measured using a dipstick. The dipstick cap is provided with threads to fit in the upper casing.
- The dipstick has two markings to indicate the prescribed maximum and minimum levels for lubricating oil.
- The same hole would be used for refilling the lubricating oil.

Oil way and Drain Plug

- The Oil path guides the oil in the casing towards the drain plug hole.
- Drain plug is provided at the bottom of the housing and a slope is provided to allow easy flow through oil way.
- Drain plug is provided to drain of used gear oil from the housing without dismantling it. It also has a rubber ring to provide a leak free seal.

Fins

- Fins are provided on the gearbox casing in order to improve the heat transfer rate.