Index

1.0. OBJECTIVE

Design a multi-stage Helical Gearbox for a hoist system which runs at 75 RPM with moderate shock loads. The source of power is 42 KW diesel engine runs at 1500 RPM. Do necessary manual calculations to design the helical gearbox with housing, necessary shafts to hold the gears and support bearings so as to have optimum size with adequate factor of safety. Suitable assumptions can be made with proper justification in choosing number of stages, materials used, type of bearings, lubrication requirements etc. Using Fusion 360 software,

1. Model the Gearbox assembly

2. Perform the necessary static/structural simulations to the gearbox housing (check whether the displacement & stress values are within permissible limits) using appropriate boundary conditions and computed loads from manual calculations

3. Also do the modal analysis for the above housing

4. Redesign the assembly in case of any discrepancies found

5. Do topology optimization for the housing

6. Create final assembly drawing and production drawings of parts showing necessary fits, tolerances, surface finish required for manufacturing

1.1. PRELIMINARY CALCULATIONS

The entire design was performed in google sheets and the link to which can be found below:

(https://docs.google.com/spreadsheets/d/1bazfjSViHfB1CyB15RYgQZJT_o-mII6iBlyMpOrTufY/edit?usp=sharing)

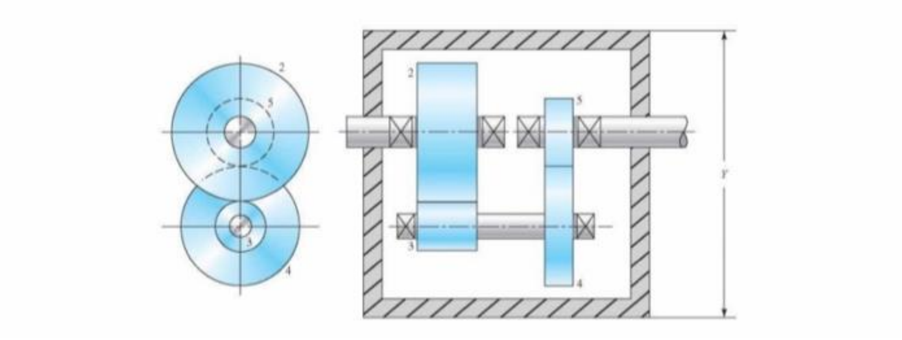
Given:

- Power of Prime Mover: 42 KW
- Input Shaft Speed: 1500 rpm
- Output Shaft Speed: 75 rpm

Calculated Parameters:

- Torque on Input Shaft: 267.3803044 Nm
- Torque on Output shaft: 5347.606088 Nm

1.1.2. LAYOUT OF THE GEARBOX



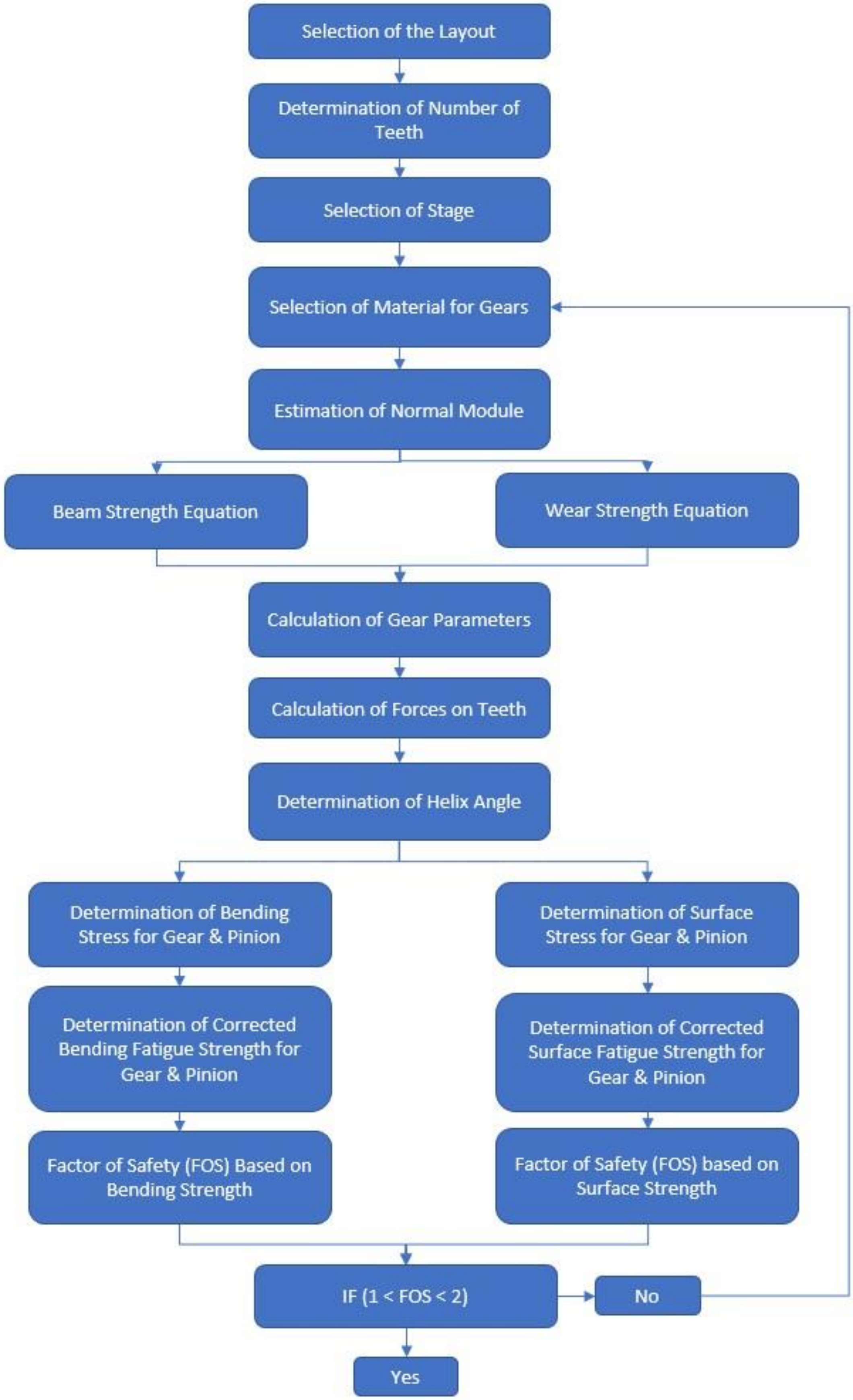
Source: https://www.chegg.com/homework-help/questions-and-answers/ques-1-10-compound-reverted-gear-train-designed-speed-increaser-provide-total-increase-spe-q49319277

Reverted Gear Train has been chosen to reduce the floor space. Helical gears have been chosen to reduce gear box size and decrease noise.

1.1.3. FLOW CHART OF DESIGN PROCEDURE FOR GEARS

A helical gear is uniquely defined by the following parameters:

1. Number of Teeth (*N*)
2. Normal Module (m_n)
3. Normal Pressure Angle (σ_n)
4. Helix Angle (ψ)
5. Face Width (*F*)


Selection of the Layout
Determination of Number of Teeth
Selection of Stage
Selection of Material for Gears
Estimation of Normal Module
Beam Strength Equation
Wear Strength Equation
Calculation of Gear Parameters
Calculation of Forces on Teeth
Determination of Helix Angle
Determination of Bending Stress for Gear & Pinion
Determination of Surface Stress for Gear & Pinion
Determination of Corrected Bending Fatigue Strength for Gear & Pinion
Determination of Corrected Surface Fatigue Strength for Gear & Pinion
Factor of Safety (FOS) Based on Bending Strength
Factor of Safety (FOS) based on Surface Strength
IF (1 < FOS < 2)
No
Yes

1.2. DETERMINATION OF NUMBER OF TEETH

The following methods were tried to determine the number of teeth for the train.

1.2.1 SELFRIDGE ALGORITHM

- Minimum Number of Teeth: 18
- Maximum Number of Teeth: 100
- Error Tolerance: 0.1 % of Train Ratio

The following values were obtained for the teeth:

- Pinion 1: 18
- Gear 1: 78
- Pinion 2: 18
- Gear 2: 83
- Train Ratio ≅ 19.98

Code can be found below:

```
import math

eps1 = 0.02
eps = eps1
counter = 0
R = 20
sol = [[0,0,0,0,0,0,0,0]]
N_min = 14
N_max = 150
s = 0
while(True):
    R_high = R + eps
    R_low = R - eps
    Nh3 = int(N_max**2/R_high)
    Nh4 = int(N_max/math.sqrt(R_high))
    for pinion1 in range(N_min,Nh4+1,1):
        Nhh = min(N_max,int(Nh3/pinion1))
        for pinion2 in range(pinion1,Nhh + 1,1):
            P = int(pinion1 * pinion2 * R_high)
            Q = int(pinion1 * pinion2 * R_low) + 1
            if (P < Q):
                s = s + 1
                continue
```

```
      Nm = max(N_min,int((Q + N_max - 1)/N_max))
      Np = int(math.sqrt(P))
      for K in range(Q,P+1,1):
        for gear1 in range(Nm,Np+1,1):
          if (K % gear1 != 0):
            s = s + 1
            continue
          gear2 = K/gear1
          error = (R - K)/(pinion1 * pinion2)
          if (error > eps):
            s = s + 1
            continue
          else:

sol.append([pinion1,pinion2,gear1,gear2,abs(error),(gear1/pinion1),(gear2/pinion2),((gear1/pinion1)*(gea
r2/pinion2))])
  if (len(sol) == 0):
    eps = eps * 2
    counter = counter + 1
  else:
    print(sol[1])
    print(s)
    break
```

Link to the Python Code: https://drive.google.com/file/d/13c0R413ew8UzXJO3vXvpTVBuI6-lsaex/view?usp=sharing

Material Source: Design of Machinery, New Media Version 2nd Edition - Amazon.com." https://www.amazon.com/Design-Machinery-New-Media-Version/dp/007242351X. Accessed 8 Mar. 2022. Ch.9, Pg. 460

1.2.2. CALCULATION OF NUMBER OF TEETH USING CONVENTIONAL METHOD

Preliminary Velocity Ratio

$$\sqrt[2]{\frac{N_{output}}{N_{input}}} = \sqrt[2]{\frac{1500}{75}} \approx 4.472$$

Where, N_{output} is the output speed in RPM

N_{input} is the input speed from Diesel Engine in RPM

Material Source: https://www.amazon.in/Design-Machine-Elements-third-Used/dp/B009DZ0LKM - Chapter 17 – Page 670 – Section 1.15

Centre Distance Criterion for Gears

$$D_2 + D_3 = D_4 + D_5$$

Where, D_2 is the pitch diameter of pinion 1

D_3 is the pitch diameter of gear 1

D_4 is the pitch diameter of pinion 2

D_5 is the pitch diameter of gear 2

$$m_{t1} \times N_2 + m_{t1} \times N_3 = m_{t2} \times N_4 + m_{t2} \times N_5$$

Where, m_{t1} is the transverse module for first stage

m_{t2} is the transverse module for second stage

$$m_{t1} = m_{n1}/\cos(\psi_1)$$

Where, m_{n1} is the normal module for first stage helical gears

$\cos \psi_1$ Is helix angle for the first stage helical gears

Similarly for the second stage.

Minimum Number of Teeth for Pinion

$$\frac{2}{Sin^2(\alpha)} = 17.097264341 \approx 18$$

Where, α is the pressure angle which is fixed at 20^o

Gear Ratios G_1 and G_2 for first and second stage respectively are given an initial value of 4.472

$$N_4 \times G_2 = N_5$$

$$N_2 = \frac{\cos(\psi_1) \times m_{n2} \times (N_4 + N_5)}{\cos(\psi_2) \times (1 + G_1)}$$

$$N_2 \times G_1 = N_3$$

Or (Whichever gives smaller number of teeth)

$$N_2 \times G_1 = N_3$$

$$N_1 = \frac{\cos(\psi_2) \times m_{n1} \times (N_2 + N_3)}{\cos(\psi_1) \times (1 + G_2)}$$

$$N_4 \times G_2 = N_5$$

The following values were chosen for the number of teeth

$$N_2 = 47,\ N_3 = 165,\ N_4 = 18,\ N_5 = 81$$

$$G_1 = 4.4595,\ G_2 = 4.5,\ G = 20.0676$$

1.3. DESIGN OF FIRST STAGE

1.3.1. Selection of Material for Gears

Material	Gea	Brinell Hardness Number (BHN)	Bending Fatigue Strength (MPa)	Surface Fatigue Strength (MPa)	Density (kg/mm^3)	Modulus of Elasticity (MPa)	Poisson's Ratio
Steel A1 - A5 Through Hardened	Pinion & Gear	400	340	1150	7850	200000	0.3

1.3.2. Estimation of Module Based on Beam Strength

The formula was applied for both the pinion and the gear. Helical Gears requires the following changes in the formula for that is used for spur gears:

1. Virtual number of teeth is used.
    a. $N_n = \frac{N}{\cos^3(\psi)}$,
2. Normal module m_n is used.
3. Corrected Lewis form factor Y' is used.

$$S_b = FOS \times P_{eff}$$

Where, S_b is the beam strength of Gear Teeth

P_{eff} is the effective load on Gear Teeth

FOS (Factor of Safety) = 1.5

S_b = $\sigma_b \times b \times m_n \times Y'$

$$P_{eff} = \frac{C_a \times C_m \times P_t}{C_v}$$

$$P_t = \frac{(P \times 60000)}{\pi \times n \times N' \times m_n}$$

$$m_n^3 = \frac{60 \cdot 10^6 \cdot P \cdot C_a \cdot C_m \cdot FOS}{\pi \cdot N' \cdot n \cdot C_v \cdot \left(\frac{b}{m}\right) \cdot S_{bs} \cdot Y'}$$

1.3.3. Estimation of Module Based on Wear Strength

$$S_w = FOS \times P_{eff}$$

$$FOS = 1.2$$

$$S_w = \frac{\left(d_p \times b \times Q \times K\right)}{\cos^2(\psi)}$$

$$m_n^3 = \frac{60 \cdot 10^6 \cdot P \cdot C_a \cdot C_m \cdot FOS \cdot \cos^2(\psi)}{\pi \cdot N' \cdot n \cdot C_v \cdot \left(\frac{b}{m}\right) \cdot Q \cdot k}$$

Gearing	Module (Beam Strength)	Module (Wear Strength)
Pinion	3.64	3.27
Gear	3.50	1.99

The normal module for the first stage was chosen as 3.50

1.3.4. Determination of Gear Parameters

1. Transverse Pressure Angle, $\phi_t = \tan^{-1}\left(\frac{\tan(\phi_n)}{\cos(\psi)}\right)$
2. Base Helix Angle, $\psi_b = \cos^{-1}\left(\frac{\cos(\phi_n) \cdot \cos(\psi)}{\cos(\phi_t)}\right)$
3. Transverse Module, $m_t = \frac{m_n}{\cos(\psi)}$
4. Transverse Pitch, $p_t = \pi \cdot m_t$
5. Normal Pitch, $p_n = P_t \cdot \cos(\psi)$
6. Axial Pitch, $p_a = \frac{P_n}{\sin(\psi)}$
7. Pitch Diameter, $p_d = N \cdot m_t$
8. Centre Distance, $C = \frac{\left((P_d)_{gear} + (P_d)_{pinion}\right)}{2}$
9. Addendum, $A = m_t \cdot 1$
10. Dedendum, $D = m_t \cdot 1.25$
11. Tooth Depth, $A + D = 2.25 \cdot m_t$
12. Clearance, $D - A = 0.25 \cdot m_t$
13. Outer Diameter of Gearing, $P_d + 2 \cdot A$
14. Pitch Line Velocity, $\frac{(2 \cdot \pi \cdot P_d \cdot N)}{60 \cdot 2000}$

15. Length of Action, $Z = \sqrt{(r_p + a_p)^2 - (r_p \cdot \cos(\phi))^2} + \sqrt{(r_g + a_g)^2 - (r_g \cdot \cos(\phi))^2} - C \cdot \sin(\phi)$
16. Transverse Contact Ratio, $m_P = \frac{Z}{(\pi \cdot m_t \cdot \cos(\phi_t))}$
17. Minimum Value of Face Width, $\frac{(\pi \cdot m_n)}{\sin(\psi)}$
18. Face Width, $8 \cdot m_n < F < 12 \cdot m_n$
    a. First Stage, $F = 12 \cdot m_n$
    b. Second Stage, $F = 10 \cdot m_n$
19. Axial Contact Ratio, $m_F = \frac{F}{P_x}$
20. $n_r = Fractional\ Part\ of\ m_P$
21. $n_a = Fractional\ Part\ of\ m_F$
22. Minimum Length of the Lines of Contact, $L_{min} = \frac{(m_P \cdot F - n_a \cdot n_r \cdot P_x)}{\cos(\psi_b)}\ if\ n_a \leq 1 - n_r L_{min} = \frac{(m_P \cdot F - (1 - n_a) \cdot (1 - n_r) \cdot P_x)}{\cos(\psi_b)}\ if\ n_a > 1 - n_r$
23. Load Sharing Ratio, $m_N = \frac{L_{min}}{F}$
24. Tangential Load,

Stage	Gear	Pinion
1	$F_t = \frac{(T_{idler} \cdot 2000)}{(P_d)_{gear}}$	$F_t = \frac{(T_{input} \cdot 2000)}{(P_d)_{pinion}}$
2	$F_t = \frac{(T_{idler} \cdot 2000)}{(P_d)_{pinion}}$	$F_t = \frac{(T_{output} \cdot 2000)}{(P_d)_{gear}}$

25. Radial Load, $F_r = (F_t \cdot \tan(\phi_n))$
26. Axial Load, $F_a = (F_t \cdot \tan(\psi))$
27. Total Load, $F_T = \frac{F_t}{(\cos(\psi) \cdot \cos(\psi_b))}$
28. Alternating Load, $F_A = \frac{F_t}{2}$
29. Weight of Gears, ($)M = \rho \cdot (P_d + 2 \cdot A) \cdot F$

(Source: https://www.amazon.in/Machine-Design-Robert-L-Norton/dp/0136123708 - Chapter 12 & 13)

Calculated Values Based on the Given Formula

	Pinion 1	Gear 1
Circular Pitch / (Transverse Pitch - Pt) (mm)	11.49798139	11.49798139
Normal Pitch (Pn) (mm)	10.99557429	10.99557429
Axial Pitch (px)	37.60820254	37.60820254
Pitch Diameter (mm)	135.4170825	603.8869894
Centre Distance (mm)	369.6520359	
Addendum (mm)	3.659921148	3.659921148
Dedendum (mm)	4.574901435	4.574901435
Whole Depth (mm)	8.234822582	8.234822582

Clearance (mm)	0.9149802869	0.9149802869
Outer Diameter of Gear (mm)	142.7369248	611.2068317
Pitch Line Velocity (m/s)	10.63563279	10.63563279
Length of Action (mm)	18.81222836	
Transverse Contact Ratio (mP) / Contact Ratio	1.750629484	
Axial Contact Ratio (mF)	1.116777649	
Fractional Part of mP (nr)	0.7506294837	
Fractional Part of mF (na)	0.1167776486	
Minimum Length of the Lines of Contact (Lmin) (mm)	73.04052066	
Load Sharing Ratio (mN)	0.5750232832	
Tangential Load (N) (Wt)	3948.989293	3948.989293
Radial Load (N) (Wr)	1437.314558	
Axial Load(N) (Wa)	1207.327188	
Total Load (N)	4418.402118	
Alternating Load (N)	1974.494647	
Material Denisty (Kg/m3)	7850	7850
Weight of Gears (Kg)	5.275717108	96.73535465

1.3.4. Determination of Helix Angle for Minimum Axial Load

The helix angle (ψ) was determined by automating over all angles from 14° to 25° in steps on 1 degree. The minimum helix angle came out to be 14° for both cases. The code for automating the process in google sheets can be found below. It was implemented in Apps Script add on of google sheets.

```
function HSelect() {
  var app = SpreadsheetApp;
  var ss = app.openByUrl("https://docs.google.com/spreadsheets/d/1bazfjSViHfB1CyB15RYgQZJT_o-mII6iBlyMpOrTufY/edit#gid=183040780");
  var active_sheet = ss.getSheetByName("Helical Practice 2");
  var min_h1 = 0;
  var min_load1 = 1.0;
  var test = 0;
  for(var h1 = 14;h1<26;h1++){
    active_sheet.getRange("B31").clearContent();
```

```
    active_sheet.getRange("B31").setValue(h1);
    test = active_sheet.getRange("B79").getValue();
    Logger.log(h1)
    if(h1 == 14){
      min_load1 = active_sheet.getRange("B79").getValue();
      Logger.log(min_load1)
    }
    var temp = active_sheet.getRange("B79").getValue();
    if(temp <= min_load1){
      min_load1 = temp;
      min_h1 = h1;
    }
  }
  active_sheet.getRange(31,2).clearContent().setValue(min_h1);
  Logger.log(min_h1);
  Logger.log(min_load1);
```

The value of m_F needs to be greater than 1 for this design procedure to be applicable so a helix angle of 17 was chosen.

1.3.5. Bending Stress Calculation

Bending stress is calculated based on AGMA bending stress equation:

$$\sigma_b = \frac{F_t \cdot K_a \cdot K_m \cdot K_s \cdot K_B \cdot K_I}{F \cdot m \cdot J \cdot K_v}$$

Application Factor - K_a

This factor is used when the nature of the loads are not clearly known.

Table 12-17 Application Factors K_a

Driving Machine	Driven Machine: Uniform	Moderate Shock	Heavy Shock
Uniform (Electric motor, turbine)	1.00	1.25	1.75 or higher
Light Shock (Multicylinder engine)	1.25	1.50	2.00 or higher
Medium Shock (Single-cylinder engine)	1.50	1.75	2.25 or higher

Image Source: https://www.amazon.in/Machine-Design-Robert-L-Norton/dp/0136123708 - Chapter 12

- Commercially available 42 KW Diesel Engines have multiple cylinders. (Source: https://resources.kohler.com/power/kohler/enginesUS/pdf/Diesel_KDI_31-55_4kW.pdf)
- Moderate Schock Loading requirement is given in the question.
- A $K_a$value of 1.5 was chosen from the above tabular column.

Load Distribution Factor - K_m

K_m compensates for axial misalignment by increasing the bending stress.

Table 12-16
Load Distribution Factors K_m

Face Width in	(mm)	K_m
<2	(50)	1.6
6	(150)	1.7
9	(250)	1.8
≥20	(500)	2.0

Image Source: https://www.amazon.in/Machine-Design-Robert-L-Norton/dp/0136123708 - Chapter 12

- $F\ (Facewidth)\ =\ 42\ mm$
- A value of 1.7 was chosen based on the selected face width

Size Factor - K_s

K_s factor is chosen to compensate for fatigue loading by increasing the bending stress. A value is 1 is used, 1.25 or 1.5 are used for high fatigue loading cases.

Rim Thickness Factor - K_B

K_B is used to compensate for rim and strokes being used instead of solid disks for gears. It is given a value of 1 for solid disks. K_B was chosen as 1 for the initial stages of the design.

Idler Factor - K_I

K_I is used especially for idler gear as they are subjected to more cycles of stress. A value 1.42 is used for them. This factor is given as 1 for non-idler gears.

Bending Strength Geometry Factor - J

J depends on the following factors:

- Number of teeth on gear - N_g
- Number of teeth on pinion - N_p
- Nature of teeth. (Full Depth – Involute in our case)
- Normal Pressure Angle - ϕ_n
- Helix Angle - ψ

An algorithm exists for calculating this factor based on AGMA 908-B89 (1999) standards titled 'Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth'. But the algorithm was long to implement, an average value of 0.6 was taken for our purpose.

Dynamic Factor - K_v

K_v is chosen based on gear quality index Q_v. Gear quality index was chosen based using the following table with Mining Conveyor being the closest application leading to a value of 7.

Table 12-6
Recommended AGMA Gear Quality Numbers for Various Applications

Application	Q_v
Cement mixer drum drive	3–5
Cement kiln	5–6
Steel mill drives	5–6
Corn picker	5–7
Cranes	5–7
Punch press	5–7
Mining conveyor	5–7
Paper-box making machine	6–8
Gas meter mechanism	7–9
Small power drill	7–9
Clothes washing machine	8–10
Printing press	9–11
Computing mechanism	10–11
Automotive transmission	10–11
Radar antenna drive	10–12
Marine propulsion drive	10–12
Aircraft engine drive	10–13
Gyroscope	12–14

Image Source: https://www.amazon.in/Machine-Design-Robert-L-Norton/dp/0136123708 - Chapter 12

K_v is then calculated based on the following formulae and determined to be 0.6759$K_v = \left(\frac{A}{A+\sqrt{200V_t}}\right)^B$

Where, $A = 50 + 56 \cdot (1 - B)$

$$B = \frac{(12 - Q_v)^{\frac{2}{3}}}{4}$$

Based on AGMA bending stress equation and the following factors, bending stress σ_b was determined as 211.143 MPa.

1.3.6. Surface Stress Calculation

Surface stress was calculated based on AGMA pitting resistance formula which is:

$$\sigma_c = C_p \cdot \sqrt{\frac{F_t \cdot C_a \cdot C_m \cdot C_s \cdot C_f}{F \cdot I \cdot d \cdot C_v}}$$

It is based on Buckingham equation. The various factors are calculated as follows:

Surface Geometry Factor - I

I is calculated from the following formulae:

$$I = \frac{\cos(\phi)}{\left(\frac{1}{\rho_g} + \frac{1}{\rho_p}\right) \cdot P_d \cdot m_N}$$

Where, $\rho_p = \sqrt{\{[(r_p + a_p) + (C - r_g - a_g)] \cdot 0.5\}^2 - (r_p \cdot \cos(\phi_n))^2}$, $\rho_g = C \cdot \sin(\phi) - \rho_p$

ρ_p and ρ_g are the radius of curvature of helical gear and helical pinion teeth respectively.

I was determined as 0.236 for both gear and pinion.

Elastic Coefficient C_p

C_p accounts for difference in the gear and pinion materials. It can be determined by the following formulae:

$$C_p = \sqrt{\frac{1}{\pi \cdot \left[\left(\frac{1 - \nu_p^2}{E_p}\right) + \left(\frac{1 - \nu_g^2}{E_g}\right)\right]}}$$

Where, ν_p and ν_g are the poison's ratio for the pinion and gear material, respectively. E_p and E_g are the modulus of elasticity for the pinion and gear respectively. It was calculated as 187.027 for both the pinion and the gear respectively.

Surface Finish Factor - C_f

This factor accounts for rough surface finishes. It is normally set to 1 for smooth finish and set greater than 1 for rough finishes.

Surface Stress - σ_c

Factors C_a, C_m,C_s & C_v are the same as K_a, K_m, K_s & K_v calculated before. The surface stress was calculated as 696.462 MPa and 329.804 MPa for pinion and gear, respectively.

1.3.7. Corrected Bending Fatigue Strength of the Material

Corrected bending fatigue strength is based on the following formula:

$$S_{fb} = \frac{K_L \cdot S_{fb'}}{K_T \cdot K_R}$$

Where, $S_{fb'}$ (Bending Strength of the Material) can be obtained from the table for the given materials

Life Factor K_L

It must be calculated based on the design life of the material. The assumed design life is:

5 Years	260 Days	8 Hours Shift
Number of Cycles N_c - Pinion	$8\,\frac{hour}{day} \cdot 260\,\frac{day}{year} \cdot 5\,year \cdot n_p \cdot 60\,\frac{\sec ond}{\min ute} = 9.36 \times 10^8\,Cycles$	
Number of Cycles N_c - Gear	$8\,\frac{hour}{day} \cdot 260\,\frac{day}{year} \cdot 5\,year \cdot n_p \cdot 60\,\frac{\sec ond}{\min ute} = 2.098909091 \times 10^8\,Cycles$	

$K_L = 1.3558 \cdot N_c^{-0.0178}$, $K_{Lp} = 0.939$, $K_{Lg} = 0.964$

Temperature Factor K_T

With the assumption of maximum oil temperature of $250^{\circ}F$ for steel, this factor is calculated as 1.145 for both pinion and gear. It is based on the fomula:

$$K_T = \frac{460 + T}{620}$$

Reliability Factor K_R

It is given a value of 1 for 99% Reliabiliy.

Corrected Bending Fatigue Strength

It was calculated as 278.69 MPa and 286.20 MPa for the pinion and gear respectively.

1.3.8. Corrected Surface Fatigue Strength of the Material

It is calculated based on the following formula:

$S_{fc} = \frac{C_L \cdot C_H \cdot S_{fc'}}{C_T \cdot C_R}$

Where S'_{fc} is obtained from the table of the given materials.

Surface Life Factor C_L

It is calculated based on the following formula:

$C_L = 1.448 \cdot N_c^{-0.023}$, it has been obtained as 0.900 and 0.932 for the pinion and gear respectively.

Hardness Ratio Factor C_H

This factor is chosen as 1 as both gear and pinion are of the same material. C_L and C_R are the same as K_L and K_R respectively.

Corrected Surface Fatigue Strength

It was obtained as 904.69 MPa and 936.34 MPa for the pinion and Gear respectively.

1.3.9. Determination of Factor of Safety

Factor of Safety based on bending fatigue strength: $\frac{S_{fb}}{\sigma_b}$

Factor of Safety based on surface fatigue strength: $\left(\frac{S_{fc}}{\sigma_c}\right)^2$

Gearing / FOS	Pinion	Gear
FOS (Bending Fatigue Strength)	1.319	1.355
FOS (Surface Fatigue Strength)	1.687	8.060

1.3.10. Determination of Backlash

The backlash was obtained from PSG Design Data Book based on m_n and pitch line velocity. It was obtained as 0.22 for first stage gears.

1.4. DESIGN OF SECOND STAGE

Following the above design procedure, the following values were obtained.

Material	Gea	Brinell Hardness Number (BHN)	Bending Fatigue Strength (MPa)	Surface Fatigue Strength (MPa)	Density (kg/mm^3)	Modulus of Elasticity (MPa)	Poisson’s Ratio

Steel A1 - A5 Flame Hardened	Pinion	510	345	1250	7850	200000	0.3
Steel A1 - A5 Through Hardened	Gear	240	245	755	7850	200000	0.3

Estimated Normal Modules

Gearing	Module (Beam Strength)	Module (Wear Strength)
Pinion	7.32	6.80
Gear	7.52	6.80

A normal module value of 7 was chosen based on the above table.

Calculation of Gear Parameters

Circular Pitch / (Transverse Pitch - Pt) (mm)	23.4024915	23.4024915
Normal Pitch (Pn) (mm)	21.99114858	21.99114858
Axial Pitch (px)	64.29781697	64.29781697
Pitch Diameter (mm)	134.0863993	603.388797
Centre Distance (mm)	368.7375982	
Addendum (mm)	7.449244407	7.449244407
Dedendum (mm)	9.311555509	9.311555509
Whole Depth (mm)	16.76079992	16.76079992
Clearance (mm)	1.862311102	1.862311102
Outer Diameter of Gear (mm)	148.9848881	618.2872858
Pitch Line Velocity (m/s)	0.5265560589	0.5313429321
Length of Action (mm)	35.48016589	
Transverse Contact Ratio (mP) / Contact Ratio	1.613383938	
Axial Contact Ratio (mF)	1.088683929	
Fractional Part of mP (nr)	0.6133839382	
Fractional Part of mF (na)	0.08868392895	
Minimum Length of the Lines of Contact (Lmin) (mm)	115.8312839	
Load Sharing Ratio (mN)	0.604327239	
Tangential Load (N) (Wt)	17785.12412	17725.24155
Radial Load (N) (Wr)	6473.255794	
Axial Load(N) (Wa)	6473.255794	
Total Load (N)	20296.65615	
Alternating Load (N)	8892.562062	
Material Denisty (Kg/m3)	7850	7850

Weight of Gears (Kg)	9.579481799	164.9826253

Determination of helix angle for minimum axial load

A helix angle of 20^o was chosen based on constraints of Axial Contact Ratio m_F.

Calculation of Bending Fatigue Stress

$$\sigma_b = \frac{F_t \cdot K_a \cdot K_m \cdot K_s \cdot K_B \cdot K_I}{F \cdot m \cdot J \cdot K_v}$$

	Pinion	Gear
σ_b (MPa)	150.22	149.79

Calculation of Surface Fatigue Stress

$$\sigma_c = C_p \cdot \sqrt{\frac{F_t \cdot C_a \cdot C_m \cdot C_s \cdot C_f}{F \cdot I \cdot d \cdot C_v}}$$

	Pinion	Gear
σ_c (MPa)	1015.55	478.039

Calculation of Corrected Bending Fatigue Strength

$$S_{fb} = \frac{K_L \cdot S_{fb'}}{K_T \cdot K_R}$$

	Pinion	Gear
S_{fb} (MPa)	278.69	286.20

Calculation of Corrected Surface Fatigue Strength

$$S_{fc} = \frac{C_L \cdot C_H \cdot S_{fc'}}{C_T \cdot C_R}$$

	Pinion	Gear
(MPa)	1090.59	636.32

Calculation of Factor of Safety

Factor of Safety based on bending fatigue strength: $\frac{S_{fb}}{\sigma_b}$

Factor of Safety based on surface fatigue strength: $\left(\frac{S_{fc}}{\sigma_c}\right)^2$

Gearing / FOS	Pinion	Gear
FOS (Bending Fatigue Strength)	2.03	1.41
FOS (Surface Fatigue Strength)	1.15	1.77

Backlash was determined as 0.2875 from PSG Design Data Book

1.4. FINAL GEAR PARAMETERS

Parameter	Pinion 1	Gear 1	Pinion 2	Gear 2
Profile	Full Depth Involute	Full Depth Involute	Full Depth Involute	Full Depth Involute
Hand of Helix	Right	Left	Left	Right
Normal Pressure Angle (ϕ_n)	20°	20°	20°	20°
Normal Module (m_n) (mm)	3.5	3.5	7	7
Helix Angle (ψ)	17°	17°	20°	20°
Number of Teeth (N)	37	165	18	81
Face Width (F) (mm)	42	42	70	70
	0.22	0.22	0.2875	0.2875

2.0. DESIGN OF SHAFTS

The following material for the shaft was chosen.

Material	SAE AISI 1045 Steel HR
Ultimate Tensile Strength	565 MPa
Yield Strength	310 MPa
Shear Strength	155 MPa
Factor of Safety	2
Loading Condition	Moderate Shock Loads
Combined S & F Bending Factor K_b	2

Combined S & F Twisting Factor K_t	2

2.1. LAYOUT FOR THE SHAFTS

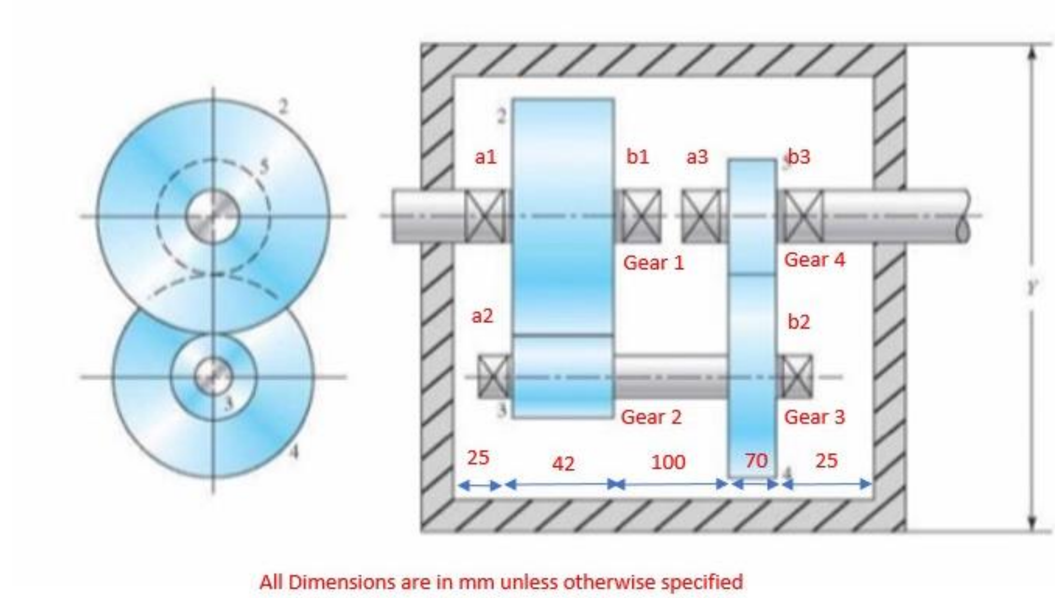


Image Source: https://www.chegg.com/homework-help/questions-and-answers/ques-1-10-compound-reverted-gear-train-designed-speed-increaser-provide-total-increase-spe-q49319277

Sign Convention for Shaft Design

Loads	+ve Direction	-ve Direction
Axial Load	→	←
Horizontal Load	↑	↓
Moment	↺	↻
Vertical Load	↑	↓

2.2. DESIGN OF INPUT SHAFT – DETERMINATION OF DIAMETER OF THE SHAFT

Axial forces

$F_{ag1} = -1207.027\ N$

$$R_{aa1} + F_{ag1} + R_{ab1} = 0$$

Net Strain $\sum \frac{Pi \cdot Li}{Ai \cdot Ei} = 0$

$\Longrightarrow R_{aa1} = 574.986\ N$ and $R_{ab1} = 632.341\ N$

Horizontal forces

$$F_{hg1} = 1437.314\ N$$

Moment at gear 1, $M_{g1} = 81746.362\ Nm$

By moment balance, $R_{hb1} = -1461.204\ Nm$

$$\Longrightarrow R_{ha1} = 23.890\ Nm$$

Vertical forces

Vertical Load at Gear 1, $F_{vg1} = -3948.989\ N$

By Moment balance, $R_{vb1} = 1880.694\ N$

$\Longrightarrow R_{va1} = 2068.295\ N$

Bending Moments

Maximum bending moments in horizontal direction, $M_{1h} = -73242.867\ Nmm$

Maximum bending moments in vertical direction, $M_{1v} = -94269.808\ Nmm$

Resultant Moment, $M_{1r} = \sqrt{M_{1h}^2 + M_{1v}^2}$

$$M_{1r} = 119378.8685\ Nmm$$

Torque on Input Shaft, $M_{t1} = 267.3803044\ Nm$

Equivalent Torque on Shaft, $T_{eq} = \sqrt{(M_{1r} \times K_b)^2 + (M_{t1} \times K_t)^2}$

$$T_{eq} = 585.6399625\ Nm$$

Diameter of Input Shaft, $D_1 = 1000 \times \sqrt[3]{\left(\frac{16 \times T_{eq} \times FOS \times 10^{-6} \times 2}{\pi \times \sigma_y}\right)}$

$$D_1 = 33.76237872\ mm \approx 35\ mm$$

2.3. DESIGN OF INTERMEDIATE SHAFT – DETERMINATION OF DIAMETER OF THE SHAFT

Axial Reaction Forces

Axial Load at Gear 2, $F_{ag2} = 1207.327188\,N$

Axial Load at Gear 3, $F_{ag3} = -6473.255794\,N$

Axial Reaction in Bearing a2, $R_{aa2} = 2134.345831\,N$

Axial Reaction in Bearing b2, $R_{ab2} = 3131.582775\,N$

It is calculated based on; $R_{ab2} + R_{ag2} + R_{aa2} + R_{ag3} = 0$ and $\sum \frac{Pi \cdot Li}{Ai \cdot Ei} = 0$

Horizontal Reaction Forces

Horizontal Force at Gear 2, $F_{hg2} = -1437.314558\,N$

Moment at Gear 2, $M_{g2} = -433987.7807\,Nmm$

Horizontal Load at Gear 3, $F_{hg3} = -6473.255794\,N$

Moment at Gear 3, $M_{g3} = -433987.7807\,Nmm$

Reaction at Bearing b2, $R_{hb2} = 8264.225336\,N$

Reaction at Bearing a2, $R_{ha2} = -353.6549843\,N$

Vertical Forces

Vertical Load at Gear 2, $F_{vg2} = 3948.989293\,N$

Vertical Load at Gear 3. $F_{vg3} = -17785.12412\,N$

Weight of Gear 3, $W_{g3} = -6473.255794\,N$

Taking moment about a2 in vertical direction, $R_{vb2} = 17524.89727\,N$

$$\Longrightarrow R_{va2} = 2123.352212\,N$$

Maximum moment in shaft in horizontal plane, $M_{ih} = 655972.8861\,Nmm$

Maximum Moment in shaft in vertical plane, $M_{iv} = 1012486.161\,Nmm$

Resultant moment in the shaft, $M_R = \sqrt{M_{ih}^2 + M_{iv}^2}$

$$M_R = 1206.411478\ Nm$$

Torque on Intermediate Shaft, $T_i = 1192.371628\ Nm$

Equivalent torque, $T_{eq} = \sqrt{(M_R \times K_b)^2 + (T_i \times K_t)^2}$

$$T_{eq} = 3392.449706\ Nm$$

Diameter of the shaft, $D_2 = 1000 \times \sqrt[3]{\left(\frac{16 \times T_{eq} \times FOS \times 10^{-6} \times 2}{\pi \times \sigma_y}\right)}$

$$D_2 = 60.63554356\ mm \approx 65\ mm$$

2.4. DESIGN OF OUTPUT SHAFT – DETERMINATION OF DIAMETER OF THE SHAFT

Axial Forces

Axial Load at Gear 4, $F_{ag4} = 6473.255794\ N$

Axial Reaction in Bearing a3, $R_{aa3} = -3236.627897\ N$

Axial Reaction in Bearing b3 $R_{ab3} = -3236.627897\ N$

It is calculated based on; $R_{aa3} + R_{ab3} + F_{ag4} = 0$ and $\sum \frac{Pi \cdot Li}{Ai \cdot Ei} = 0$

Horizontal Forces

Horizontal load at gear 4, $F_{hg4} = 6473.255794\ N$

Moment at gear 4, $M_{g4} = 1952945.013\ Nmm$

Taking moment about b3 in the horizontal plane, $R_{ha3} = -16191.55682\ N$

Reaction in bearing b3, $R_{ha3} + R_{hb3} + F_{hg4} = 0$

$$R_{hb3} = 9718.301023\ N$$

Vertical Forces

Weight of Gear 4, $W_{g4} = 164.9826253\ N$

Vertical Load at Gear 4, $F_{vg4} = 17725.24155\ N$

Taking Moment about b3, $R_{va3} = -8566.212054\,N$

$$R_{va3} + W_{g4} + R_{vb3} + F_{vg1} = 0$$

$$\Longrightarrow R_{vb3} = -26456.43623\,N$$

Maximum moment in shaft at Gear 4 in horizontal plane, $M_{h4} = -1250797.764\,Nmm$

The maximum moment in shaft ar Gear 4 in vatical plane, $M_{v4} = -1911477.518\,Nmm$

Equivalent Moment in Shaft, $M_{eq} = \sqrt{M_{h4}^2 + M_{v4}^2}$

$$M_{eq} = 2284.347029\,Nm$$

Torque on output shaft, $T_4 = 5347.606088\,Nm$

Equivalent Torque, $T_{eq} = 11630.15601\,Nm$

Diameter of the Shaft, $D_3 = 1000 \times \sqrt[3]{\left(\frac{16 \times T_{eq} \times FOS \times 10^{-6} \times 2}{\pi \times \sigma_y}\right)}$

$$D_3 = 91.42914308\,mm \approx 95\,mm$$

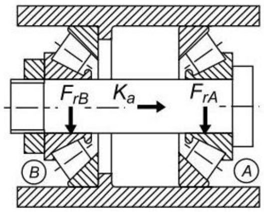
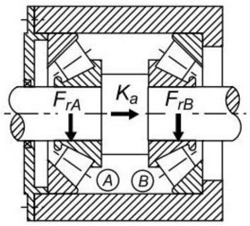
3.0. DESIGN OF BEARINGS

The usage of helical gears warrants the need for bearing taper roller beatings that can withstand axial loads. The bearing manual from https://www.skf.com/in was used as the primary source of information.

(https://www.skf.com/binaries/pub12/Images/0901d196802809de-Rolling-bearings---17000_1-EN_tcm_12-121486.pdf)

The bearings were designed based on the design procedure given in Design of Machine Elements by VB Bhandari. (https://www.amazon.in/Design-Machine-Elements-third-Used/dp/B009DZ0LKM)

The axial loads for the bearings were chosen from the following schematic.

Arrangaments	Load case	Axial loads
 Back to back  Face to face	Case 1(a): $F_{rA} \geq F_{rB}$ $K_a \geq 0$	$F_{aA} = \frac{0.5F_{rA}}{Y}$ $F_{aB} = F_{aA} + K_a$
	Case 1(b): $F_{rA} < F_{rB}$ $K_a \geq 0.5(F_{rB} - F_{rA})$	$F_{aA} = \frac{0.5F_{rA}}{Y}$ $F_{aB} = F_{aA} + K_a$
	Case 1(c): $F_{rA} < F_{rB}$ $K_a < 0.5(F_{rB} - F_{rA})$	$F_{aA} = F_{aB} - K_a$ $F_{aB} = \frac{0.5F_{rB}}{Y}$

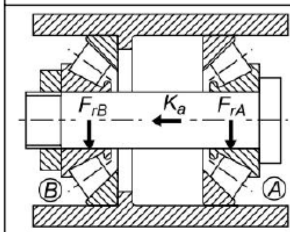
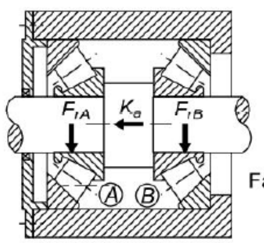
Arrangaments	Load case	Axial loads
 Back to back  Face to face	Case 2 (a): $F_{rA} < F_{rB}$ $K_a \geq 0$	$F_{aA} = F_{aB} + K_a$ $F_{aB} = \frac{0.5F_{rB}}{Y}$
	Case 2 (b): $F_{rA} > F_{rB}$ $K_a \geq 0.5(F_{rA} - F_{rB})$	$F_{aA} = F_{aB} + K_a$ $F_{aB} = \frac{0.5F_{rB}}{Y}$
	Case 2 (c): $F_{rA} > F_{rB}$ $K_a < 0.5(F_{rA} - F_{rB})$	$F_{aA} = \frac{0.5F_{rA}}{Y}$ $F_{aB} = F_{rA} - K_a$

Fig. 15.11 *Axial Loading of Taper Roller Bearings*

3.1. BEARING DESIGN FOR INPUT SHAFT

Calculation of Number of Revolutions (Millions)

Design Life ($L10$)	$8\,\frac{hour}{day} \cdot 260\,\frac{day}{year} \cdot 5\,year = 10400\,hours$
Number of Cycles (Millions of Revolutions)	$\frac{(60\ \cdot N_{input} \cdot L10)}{10^6} = 900$

Single Taper Roller bearing SKF 30207	
Radial Factor Y	1.6
e	0.37
Dynamic Load Capacity, C	63200 N

	Bearing a1	Bearing b1
Radial Load	$F_{ra} = 2068.432832\ N$	$F_{rb} = 2381.623311\ N$
Total Axial Load	$K_a = 1437.314558\ N$	
Condition (Case 2(a)) Back to Back Orientation	$F_{ra} < F_{rb}$	$K_a > 0$
Axial Load F_a	$F_{aA} = F_{aB} + K_a$ $= 646.3852599\ N$	$F_{aB} = \frac{0.5 \cdot F_{rB}}{Y}$ $= 2083.699818\ N$
Axial Load / Radial Load	$\frac{F_{aA}}{F_{rA}} = 0.3125\ < e$	$\frac{F_{aB}}{F_{rB}} = 0.874907383\ > e$
Equivalent Dynamic Load, P	$P = F_{rA}$ $= 2068.432832\ N$	$P = (0.4 \cdot F_{rB}) + (Y \cdot F_{aB})$ $= 4286.569033\ N$
Dynamic Load Capacity $= P \cdot (L_{10})^{0.3} \cdot L_F$	$15918.94108\ N$	$32990.01971\ N$

3.2. BEARING DESIGN FOR INTERMEDIATE SHAFT

Calculation of Number of Revolutions (Millions)

Design Life (L_{10})	$8\ \frac{hour}{day} \cdot 260\ \frac{day}{year} \cdot 5\ year = 10400\ hours$
Number of Cycles (Millions of Revolutions)	$\frac{(60\ \cdot N_{input} \cdot L10)}{10^6} = 201.8181818$

Single Taper Roller bearing SKF 30213	
Radial Factor Y	1.5
e	0.4
Dynamic Load Capacity, C	141000 N

	Bearing a2	Bearing b2
Radial Load	$F_{ra} = 2152.602254\ N$	$F_{rb} = 15198.88476\ N$
Total Axial Load	$K_a = -5265.928606\ N$	

Condition (Case 1(c)) Back to Back Orientation	$F_{ra} < F_{rb}$	$K_a < 0.5 \cdot (F_{rB} - F_{rA})$
Axial Load	$F_{aA} = F_{aB} - K_a$ $= 10332.22353\ N$	$F_{aB} = \frac{0.5 \cdot F_{rB}}{Y}$ $= 5066.29492\ N$
Axial Load / Radial Load	$\frac{F_{aA}}{F_{rA}} = 4.799875827\ > e$	$\frac{F_{aB}}{F_{rB}} = 0.3333333333\ < e$
Equivalent Dynamic Load, P	$P = (0.4 \cdot F_{rA}) + (Y \cdot F_{aA})$ $= 16359.37619\ N$	$P = F_{rB}$ $= 15198.88476\ N$
Dynamic Load Capacity $= P \cdot (L_{10})^{0.3} \cdot L_F$	$= 80399.77365\ N$	$= 74696.42364\ N$

3.3. BEARING DESIGN FOR OUTPUT SHAFT

Calculation of Number of Revolutions (Millions)

Design Life (L_{10})	$8\ \frac{hour}{day} \cdot 260\ \frac{day}{year} \cdot 5\ year = 10400\ hours$
Number of Cycles (Millions of Revolutions)	$\frac{(60\ \cdot N_{input} \cdot L10)}{10^6} = 45$

Single Taper Roller bearing SKF 30219	
Radial Factor Y	1.4
e	0.43
Dynamic Load Capacity, C	266000 N

	Bearing a3	Bearing b3
Radial Load	$F_{ra} = 18317.92841\ N$	$F_{rb} = 28184.89654\ N$
Total Axial Load	$K_a = 6473.255794\ N$	
Condition (Case 1(b)) Back to Back Orientation	$F_{ra} < F_{rb}$	$K_a \geq 0.5 \cdot (F_{rB} - F_{rA})$
Axial Load	$F_{aA} = \frac{0.5 \cdot F_{rA}}{Y}$ $= 6542.117288\ N$	$F_{aB} = F_{aA} + K_a$ $= 13015.37308\ N$
Axial Load / Radial Load	$\frac{F_{aA}}{F_{rA}} = 0.3571428571\ < e$	$\frac{F_{aB}}{F_{rB}} = 0.4617853773\ > e$
Equivalent Dynamic Load, P	$P = F_{rA}$ $= 18317.92841\ N$	$P = (0.4 \cdot F_{rB}) + (Y \cdot F_{aB})$ $= 29495.48093\ N$

Dynamic Load Capacity	$= 57390.51361\ N$	$= 92410.05654\ N$

4.0. DESIGN OF KEY

Rectangular Sunk Keys were chosen as the models for this system. PSG Design date book and Design of Machine Elements by V B Bhandari were used as the references for design.

Table 9.3 *Dimensions of square and rectangular sunk keys (in mm)*

Shaft diameter		*Key size*	*Keyway depth*
Above	*Up to and including*	$b \times h$	
6	8	2 × 2	1.2
8	10	3 × 3	1.8
10	12	4 × 4	2.5
12	17	5 × 5	3.0
17	22	6 × 6	3.5
22	30	8 × 7	4.0
30	38	10 × 8	5.0
38	44	12 × 8	5.0
44	50	14 × 9	5.5
50	58	16 × 10	6.0
58	65	18 × 11	7.0
65	75	20 × 12	7.5
75	85	22 × 14	9.0
85	95	25 × 14	9.0
95	110	28 × 16	10.0
110	130	32 × 18	11.0
130	150	36 × 20	12.0
150	170	40 × 22	13.0
170	200	45 × 25	15.0
200	230	50 × 28	17.0

The cross section for the keys were chosen based on the shaft diameter from the above tabulation.

The minimum length of key values was obtained from PSG Design Data Book for Rectangular Sunk Keys.

The minimum length was also calculated using Shear Stress Formula given below:

$L = \frac{\tau \cdot 2 \cdot FOS}{\sigma_s \cdot b \cdot D}$, Length based on Shear Strength

$L = \frac{\tau \cdot 2 \cdot FOS}{\sigma_y \cdot b \cdot D}$, Length based on Crushing Strength

The following material was chosen as key material

Material	SAE AISI 1045 Steel HR
Ultimate Tensile Strength	565 MPa
Yield Strength	310 MPa
Shear Strength	155 MPa
Factor of Safety	2
Loading Condition	Moderate Shock Loads
Combined S & F Bending Factor	2
Combined S & F Twisting Factor	2

	Input Shaft	Intermate Shaft	Output Shaft
D (Diameter of Shaft mm)	35	65	95
Cross Section (b * h) (mm x mm)	10 x 8	18 x 11	25 x 14
FOS (Factor of Safety)	2	2	2
Minimum Length from PSG Design Data Book (mm)	22	40	70
Minimum Length from Shear Strength (mm)	13.28597786	17.72384434	39.15867158
Minimum Length Based on Yield Strength (mm)	6.642988929	8.861922168	19.57933579
Chosen Length of Key (mm)	22	40	70

5.0. DESIGN OF CASING

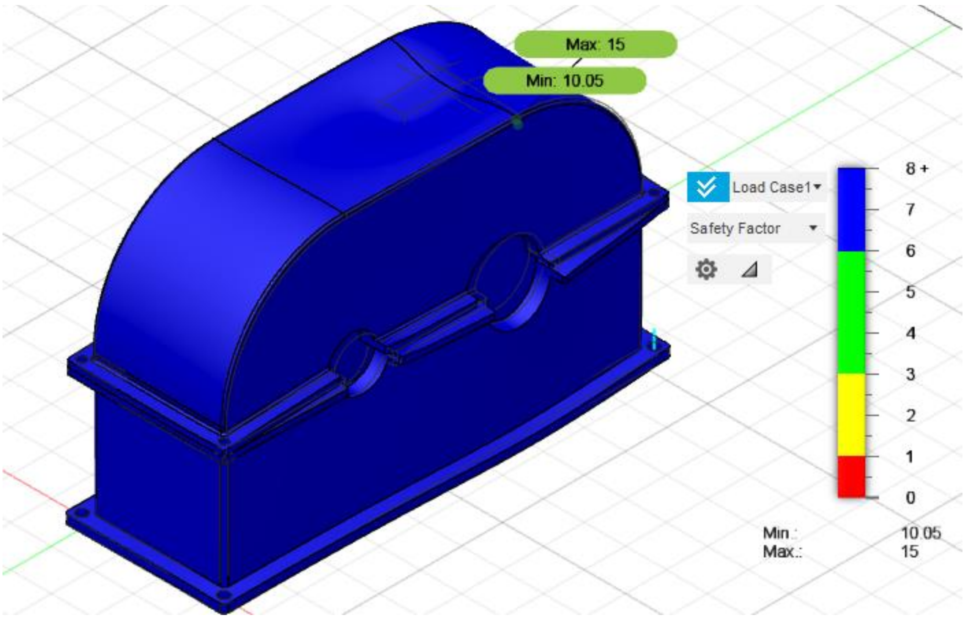
10mm was chosen as the initial thickness of the casing. 10 mm clearance was given between gears and walls. Bolts are used as fasteners.

The above shows the initial casing design. Structural Analysis was performed with loads obtained from shaft design which is shown below.

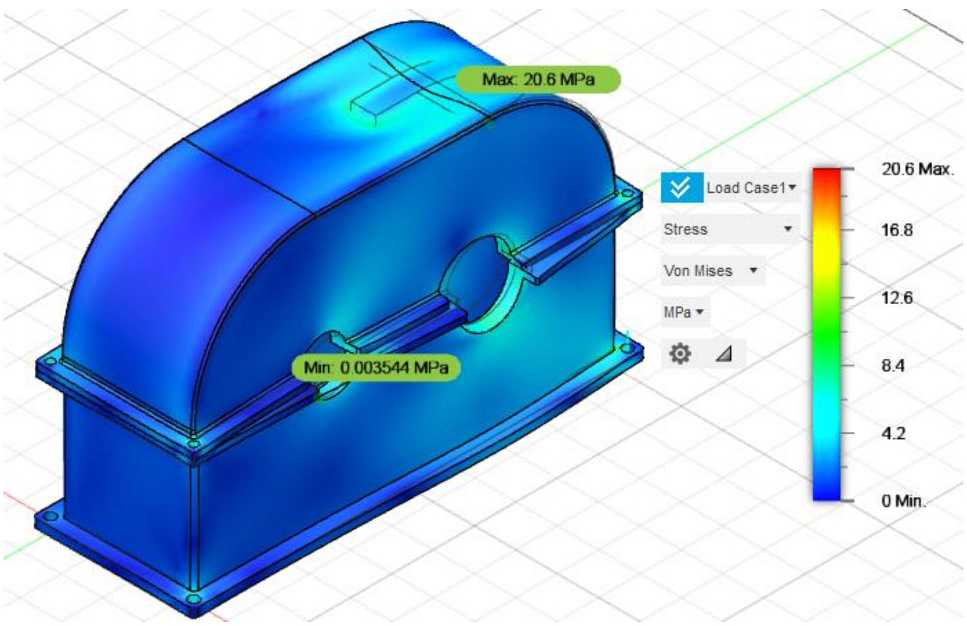
Bearing	Vertical Load (N)	Horizontal Load (N)	Axial Load (N)
A1	2068.294867	23.88976389	574.9859885
B1	1880.694426	-1461.204322	632.3411993
A2	2123.352212	-353.6549843	2134.345831
B2	12524.89727	8264.225336	3131.582775
A3	-8566.212054	-16191.55682	-3236.627897
B3	-26456.43623	9718.301023	-3236.627897

5.1. LINEAR STATIC STRUCTURAL ANALYSIS

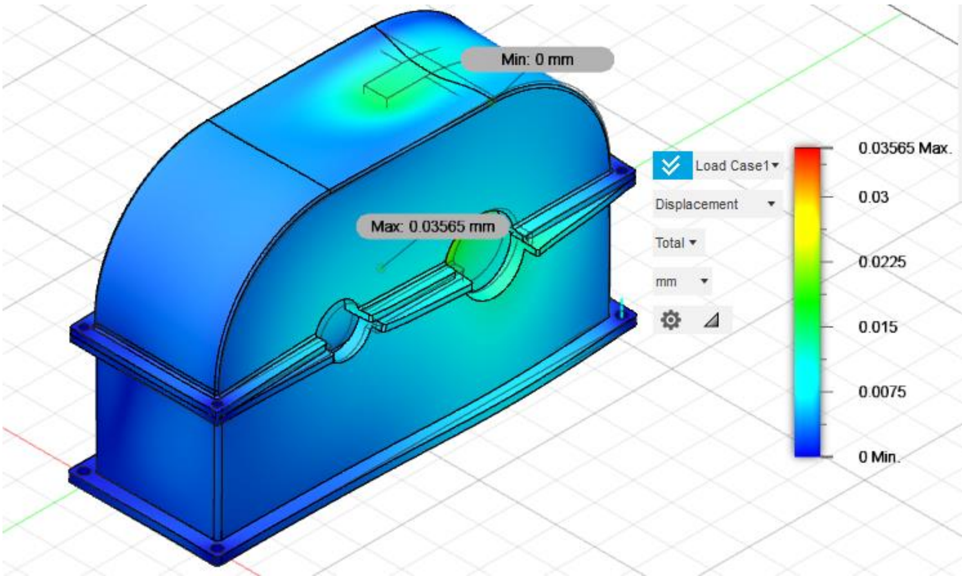
Linear Static Structural Analysis was performed on the casing with the loads given above, standard steel available in Fusion 360 was used as the material.



The minimum factor of safety was determined to be 8. An improved version of the casing was attempted in the next section.



Von Mises Stress is shown above

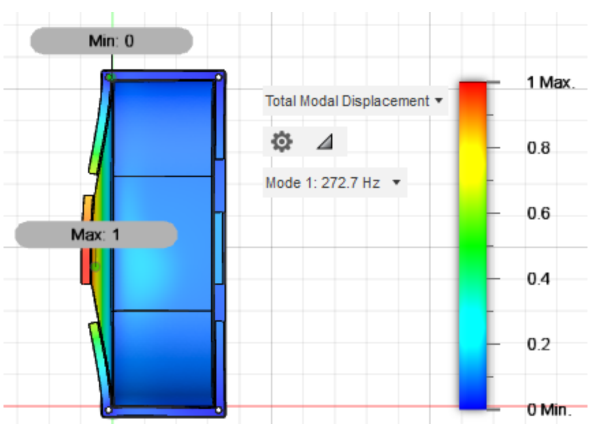


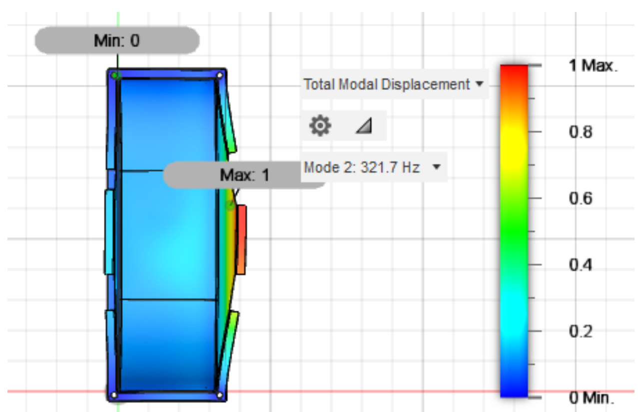
Total displacement can be found above

5.2. MODAL ANALYSIS FOR DETERMINATION OF NATURAL FREQUENCY OF THE CASING

	Input Shaft	Intermediate Shaft	Output Shaft
Speed (Hz)	25		1.25

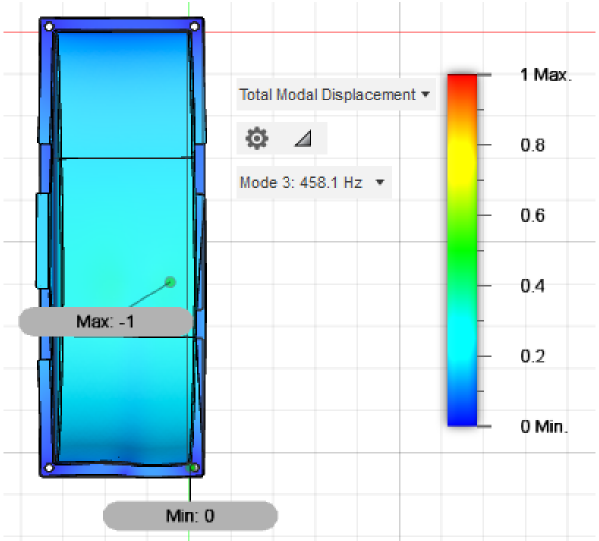
The modal analysis can be found below with the first ten modes. None of the values are near the frequencies of the rotating shafts.



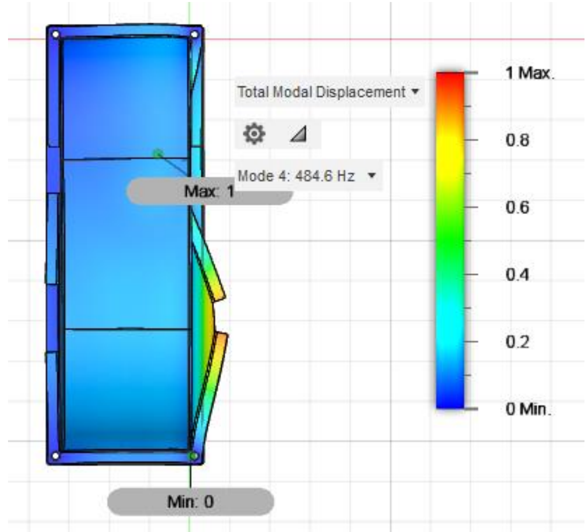


Mode 1: 272.7 Hz

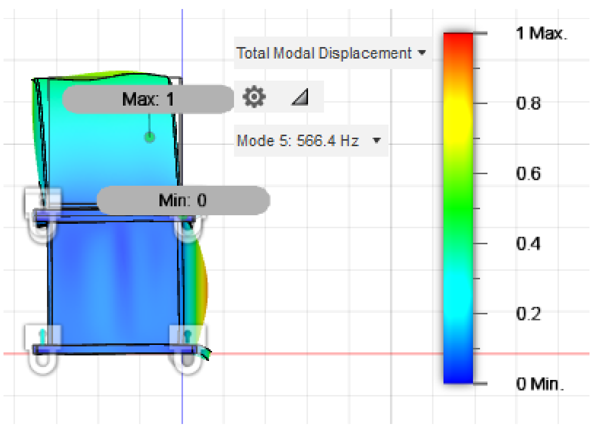
Mode 2: 321.7 Hz



Mode 4: 484.6 Hz

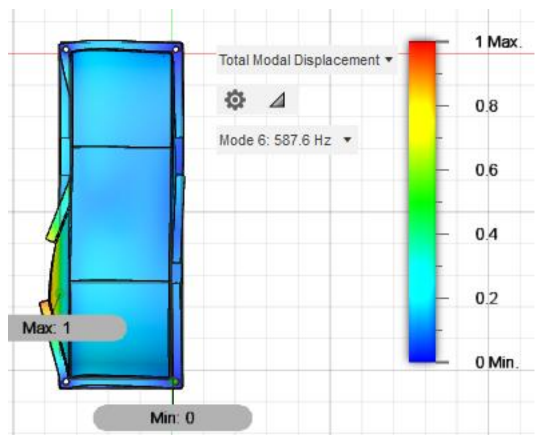


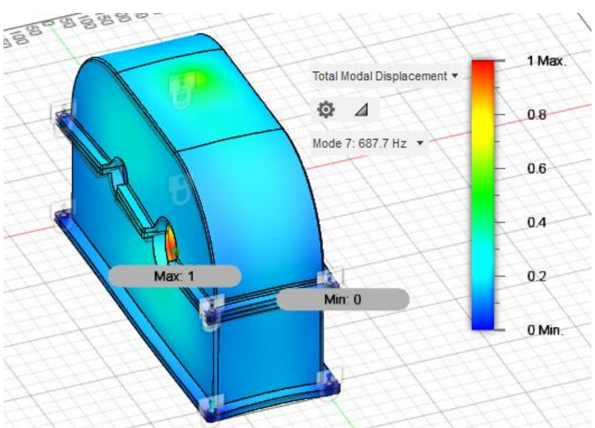
Mode 3: 458.1 Hz



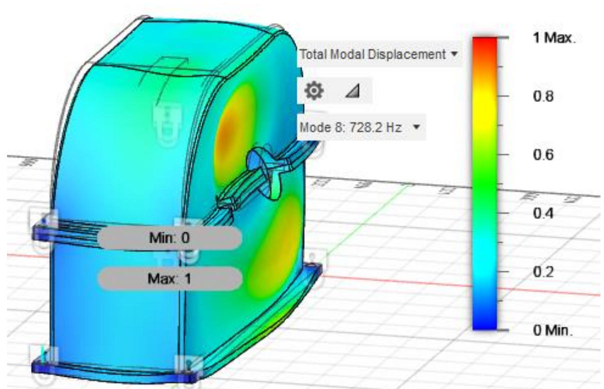
Mode 6: 587.6 Hz

Mode 5: 566.4 Hz

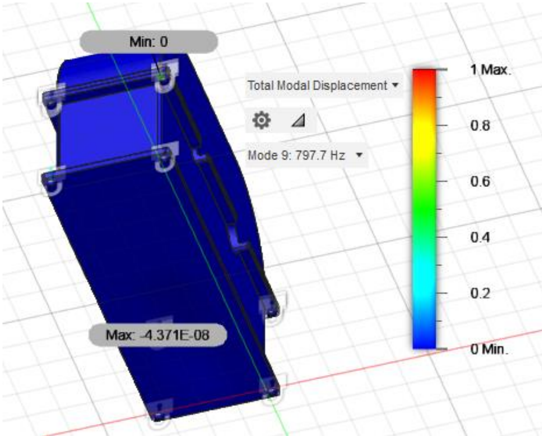


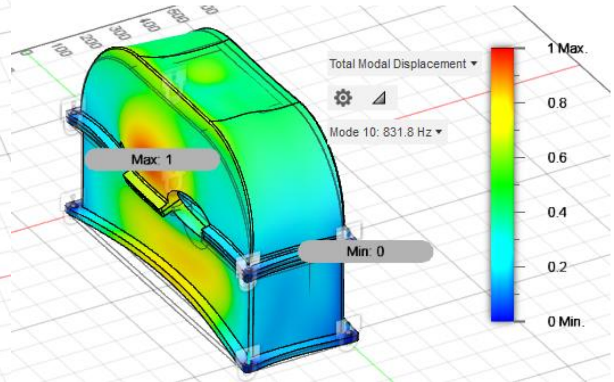


Mode 8: 728.2 Hz



Mode 7: 687.7 Hz



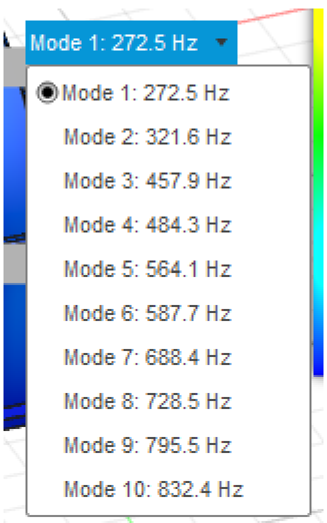


Mode 9: 797.7 Hz

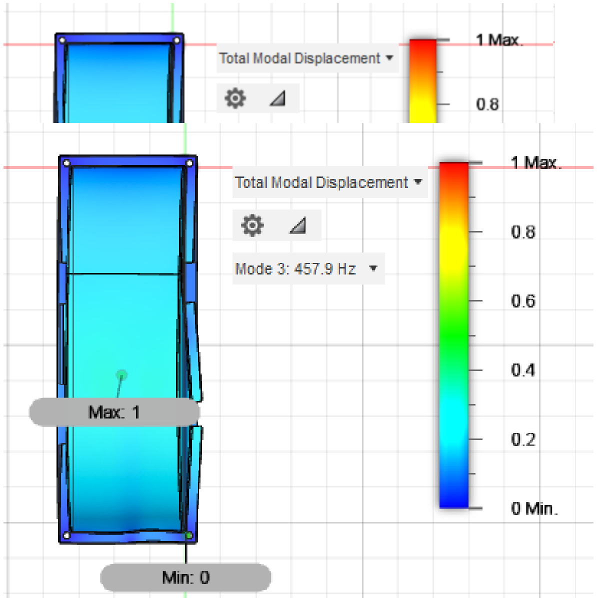
Mode 10: 831.8 Hz

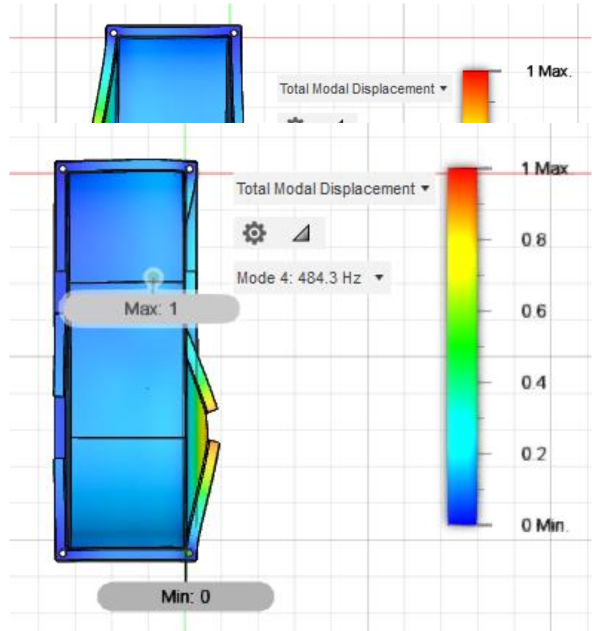
5.3. MODAL ANALYSIS USING BEARING PRELOADS

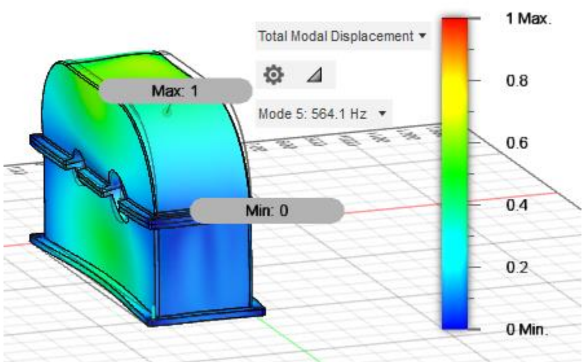
There was not much difference in the values obtained for natural frequencies of the casing:

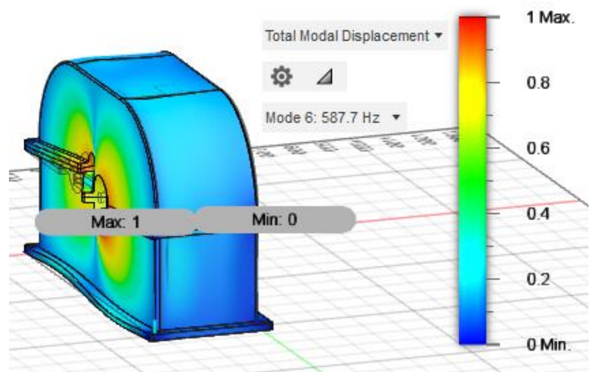


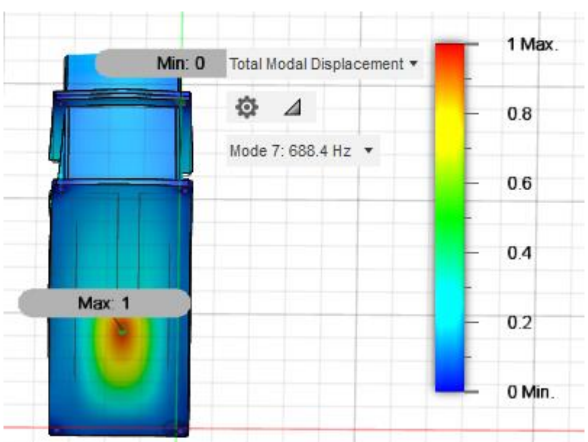
The deformation modes can be found below:

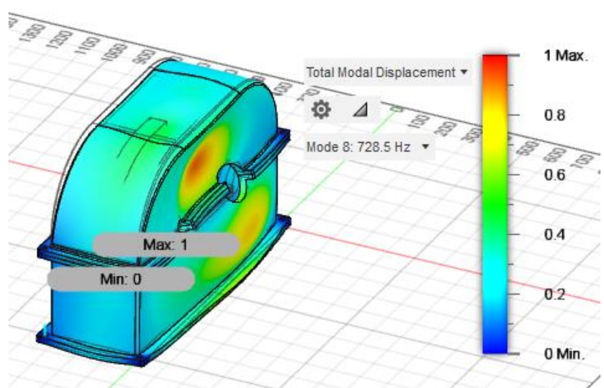

Total Modal Displacement
1 Max.
0.8
Mode 3: 457.9 Hz
0.6
0.4
0.2
0 Min.
Max: 1
Min: 0

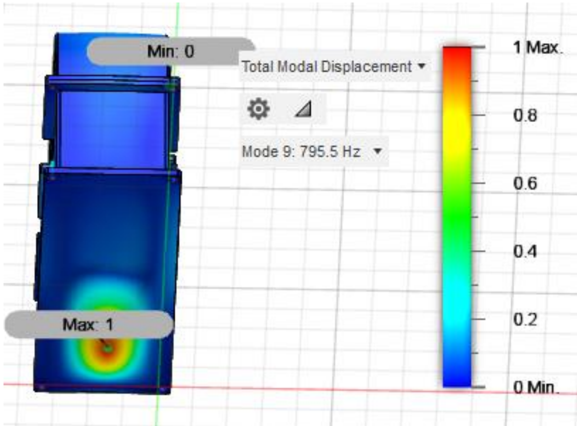

Total Modal Displacement
1 Max.
0.8
Mode 4: 484.3 Hz
0.6
0.4
0.2
0 Min.
Max: 1
Min: 0

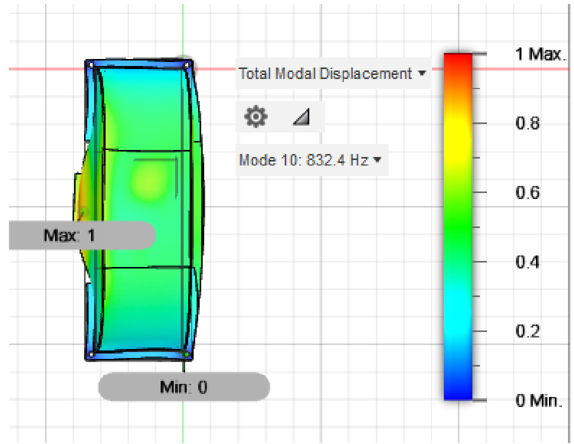

Total Modal Displacement
1 Max.
0.8
Mode 5: 564.1 Hz
0.6
0.4
0.2
0 Min.
Max: 1
Min: 0


Total Modal Displacement
1 Max.
0.8
Mode 6: 587.7 Hz
0.6
0.4
0.2
0 Min.
Max: 1
Min: 0


Min: 0
Total Modal Displacement
1 Max.
0.8
Mode 7: 688.4 Hz
0.6
0.4
0.2
0 Min.
Max: 1


Total Modal Displacement
1 Max.
0.8
Mode 8: 728.5 Hz
0.6
0.4
0.2
0 Min.
Max: 1
Min: 0

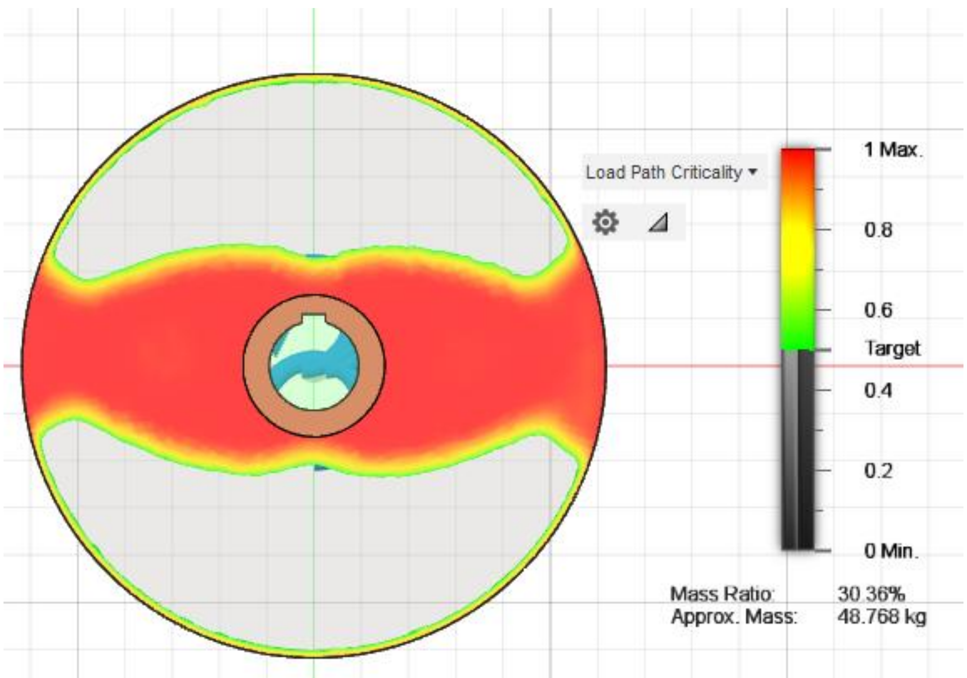


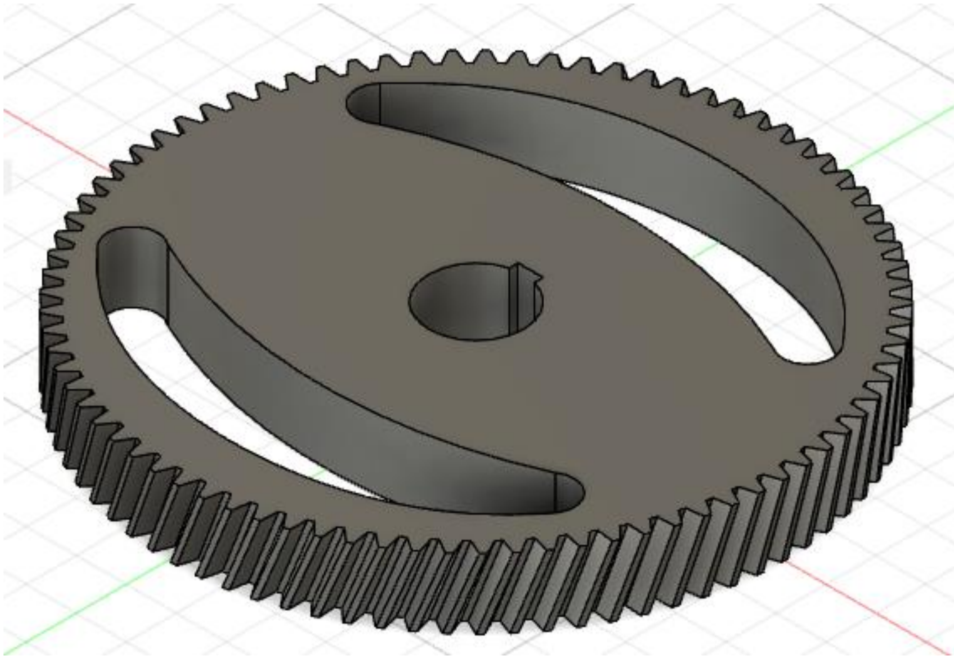


6.0. TOPOLOGY OPTIMIZATION FOR MAZIMUM STIFFNESS AND REDUCED MASS

Topology Optimization was performed on the gear of the second stage to reduce the weight. The teeth portion of the of it was approximated as a cylinder to reduce complexity in giving constraints.

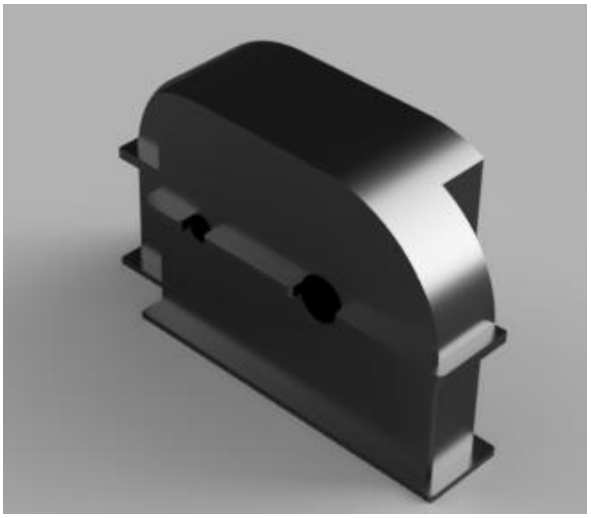
- Torque was given as input in the hole given for shaft and the slots for keys.
- The teeth region was rigidly fixed to optimize the stiffness and weight reduction.



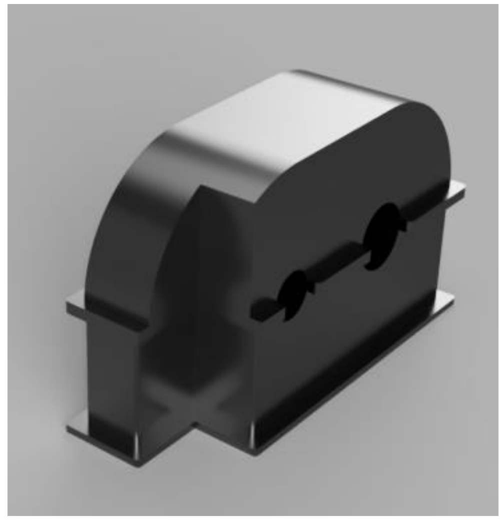


7.0. MODIFIED CASING DESIGN

Considering the high factor of safety that was obtained, a modified casing design was attempted.



7.0.

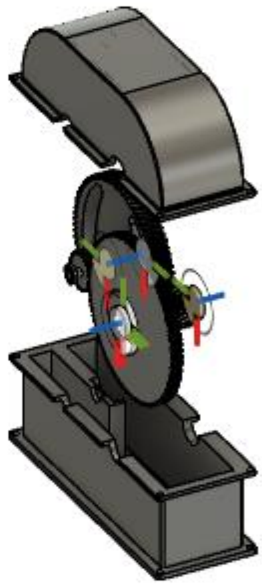


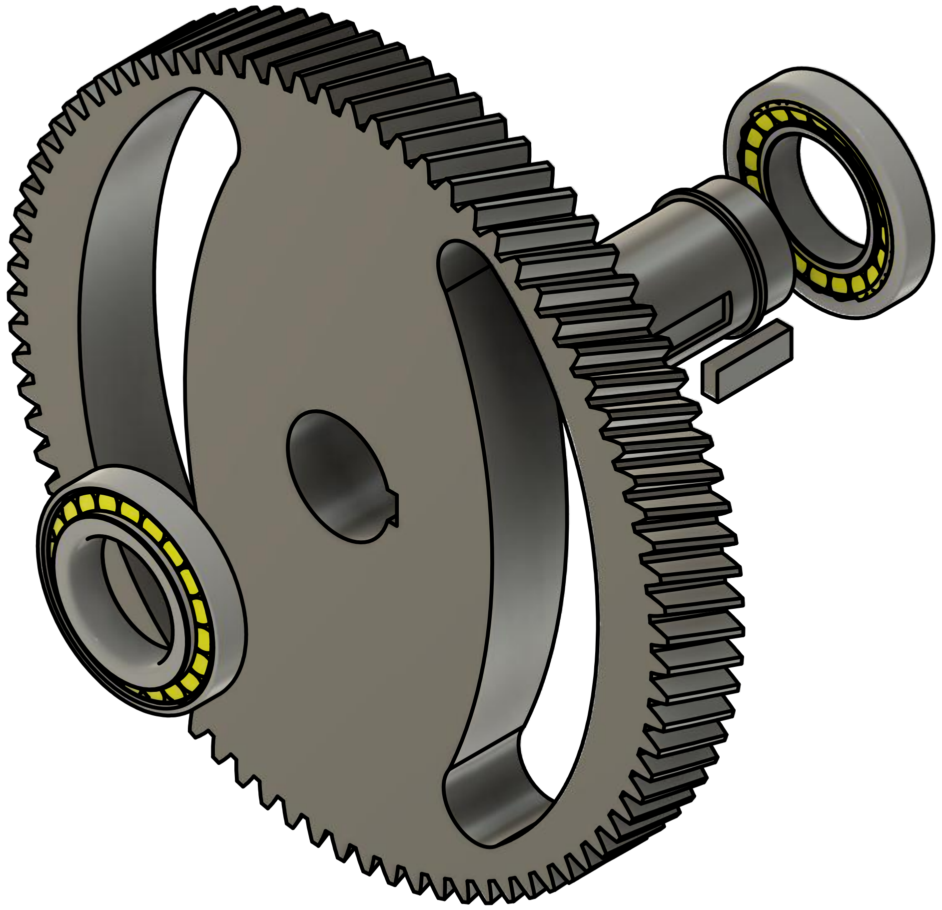
ASSEMBLY DRAWINGS

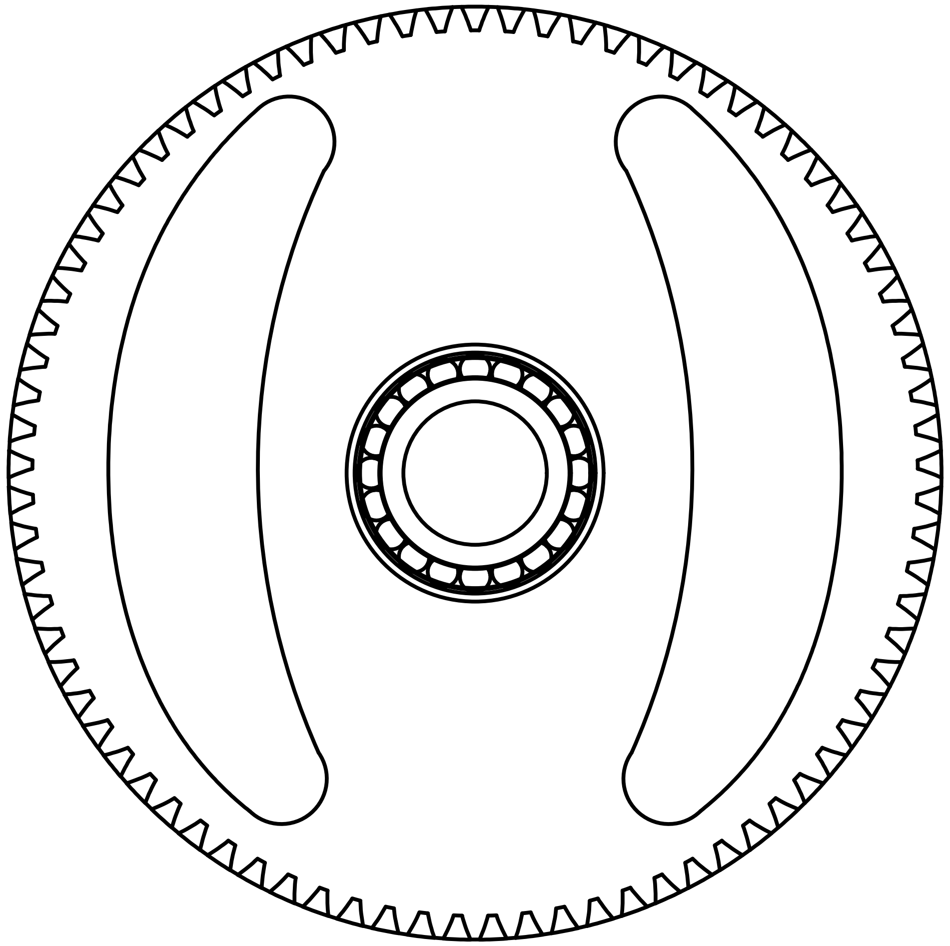
Drawings are divided into three types:

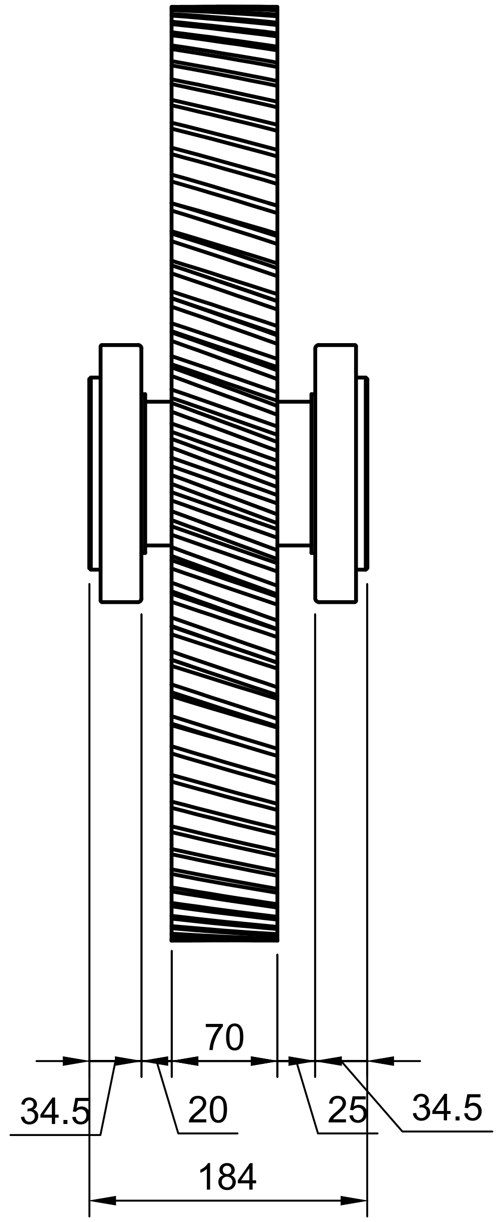
- Design Drawing
  - It refers to the drawing made in the initial stages of the design when the concept was conceived.
- Assembly Drawing
  - It refers to drawing that helps in assembling components and producing assemblies from individual parts.
- Production Drawing
  - It refers to drawing consisting of the all the relevant information for procucing the component from its raw materials.

The drawings can be found below:

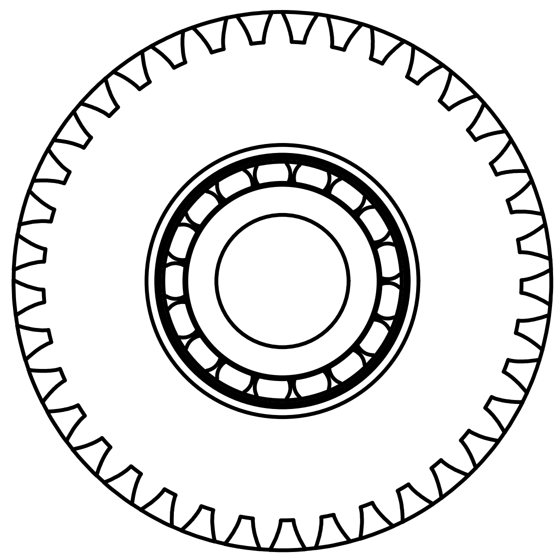


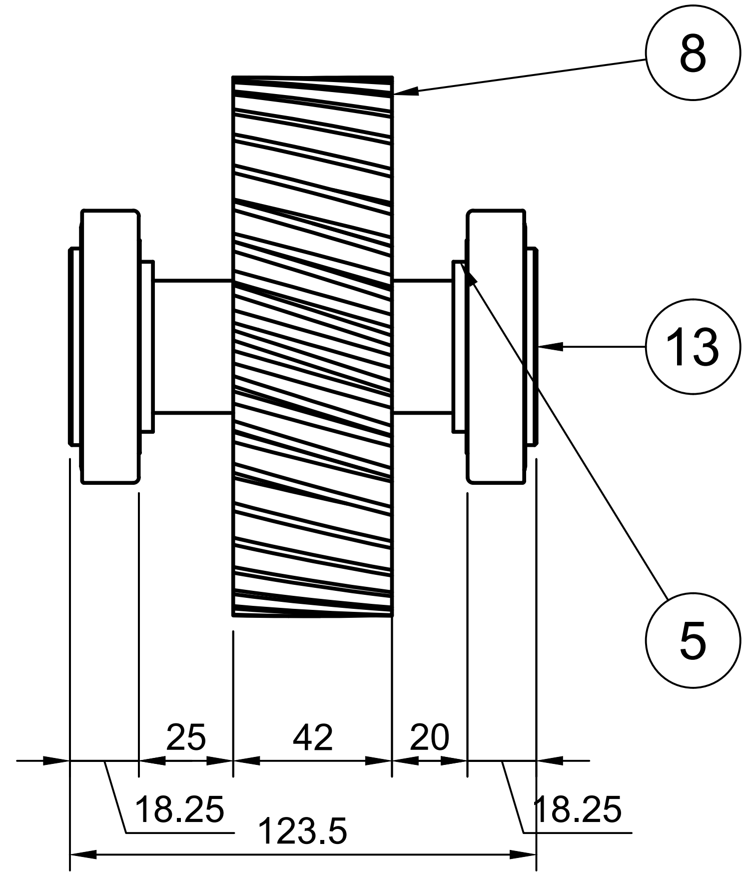


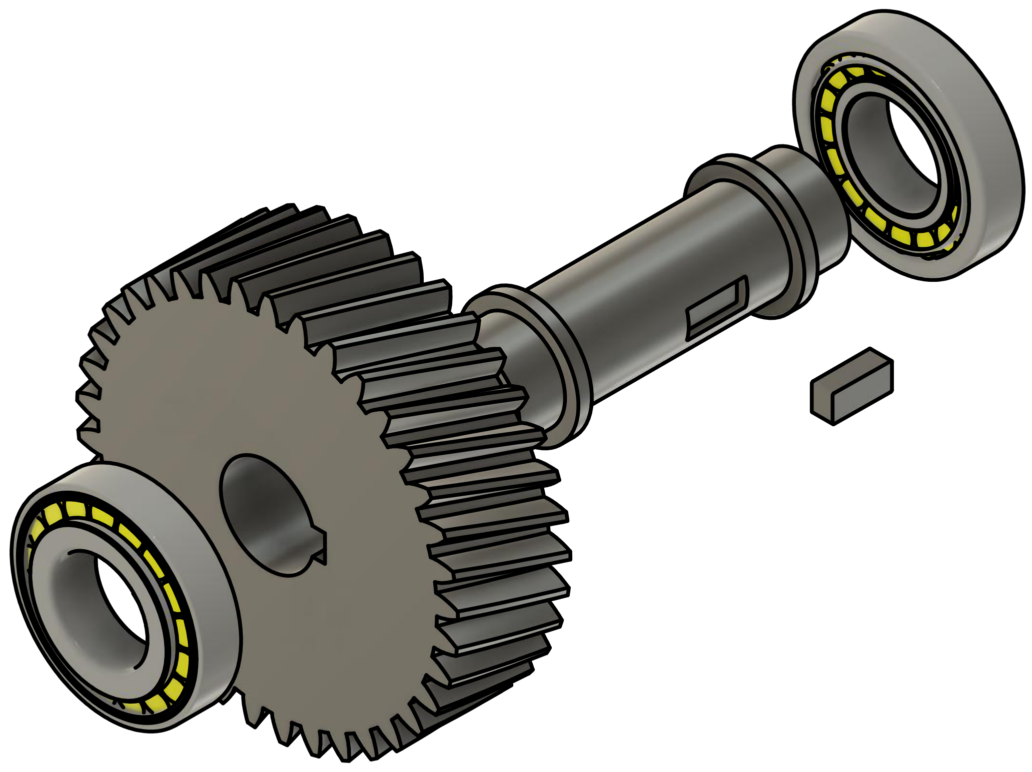




Dept.	Technical reference	Created by	Approved by
		Deepak Somasundaram 22/04/30	
		Document type	Document status
		Title Assembly	DWG No.
			Rev. | Date of issue | Sheet 1/1







Item	Qty	Part Number	Description	Material
13	2	SKF_30207 v1		Steel
12	1	Key_Pinion_1		Steel
8	1	Pinion_1		Steel
5	1	Input shaft v2		Steel

Parts List

Dept.	Technical reference	Created by	Approved by
		Deepak Somasundaram 22/04/30	

Document type

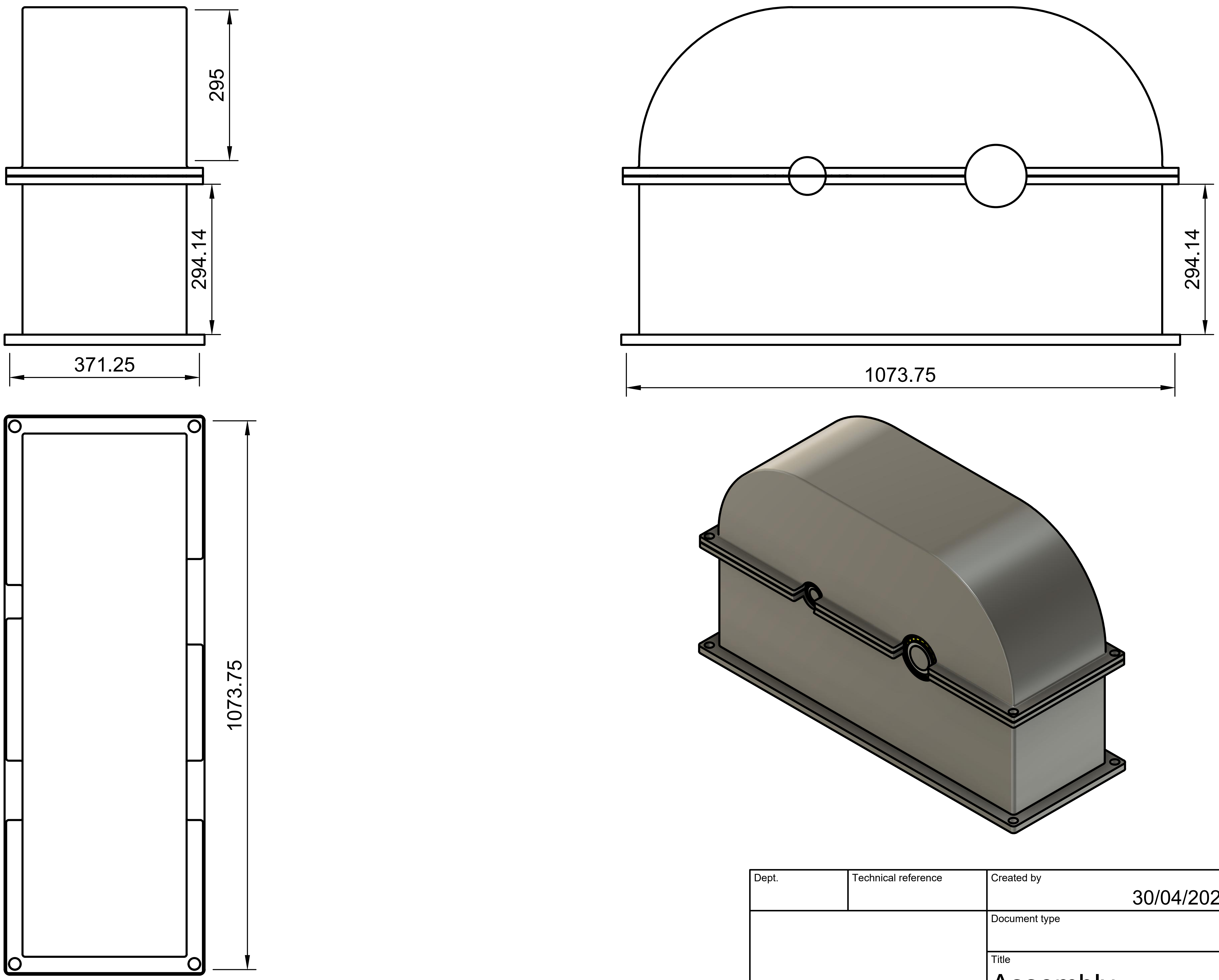
Document status

Title

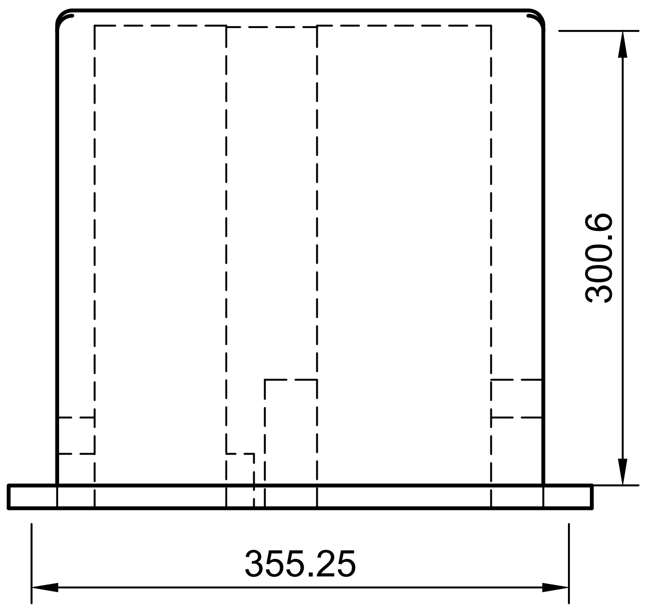
Assembly

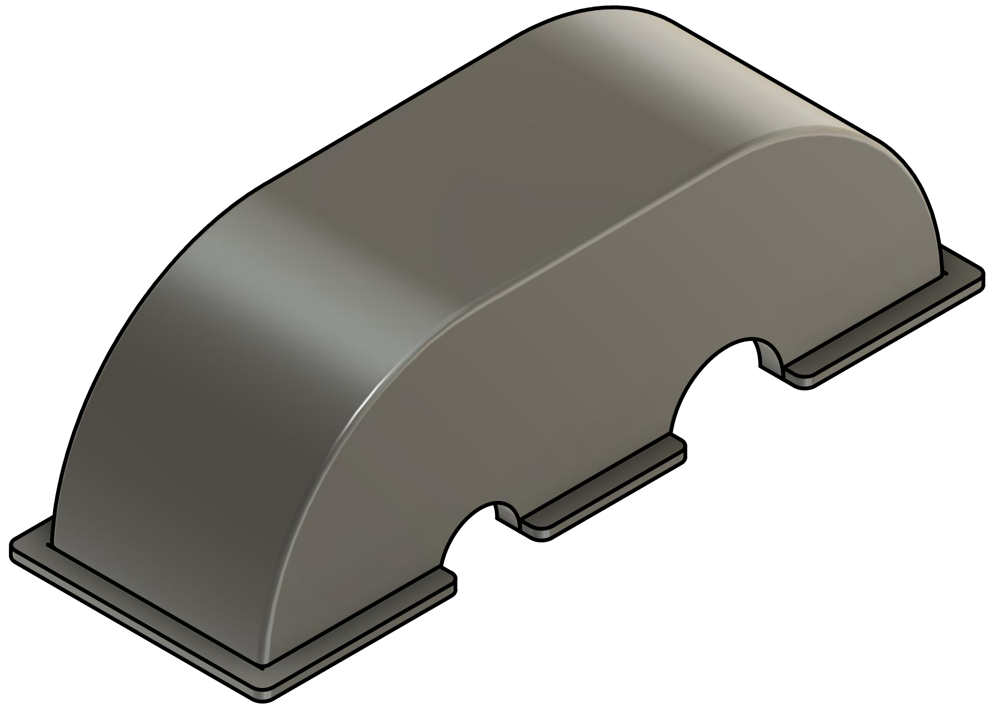
DWG No.

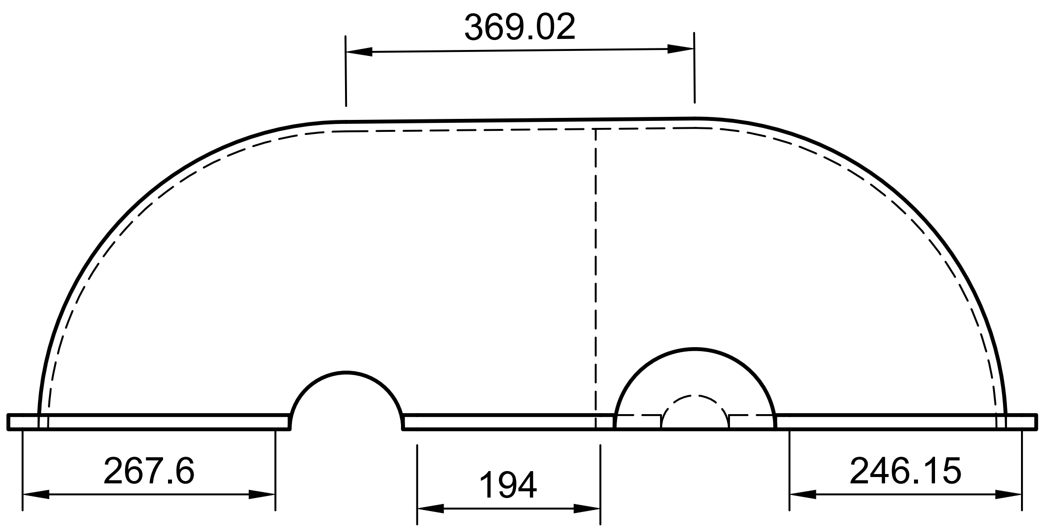
Rev.	Date of issue	Sheet
		1/1

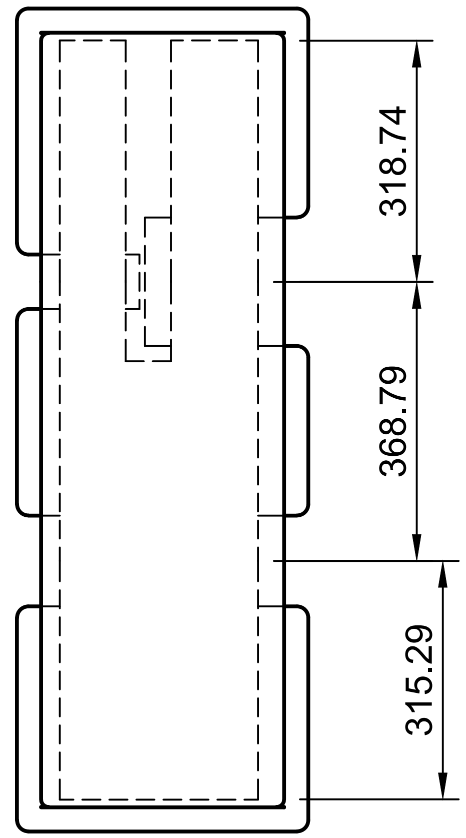


Dept.	Technical reference	Created by	Approved by
		30/04/2022	
		Document type	Document status
		Title	DWG No.
		Assembly	
		Rev.	Date of issue	Sheet 2/2

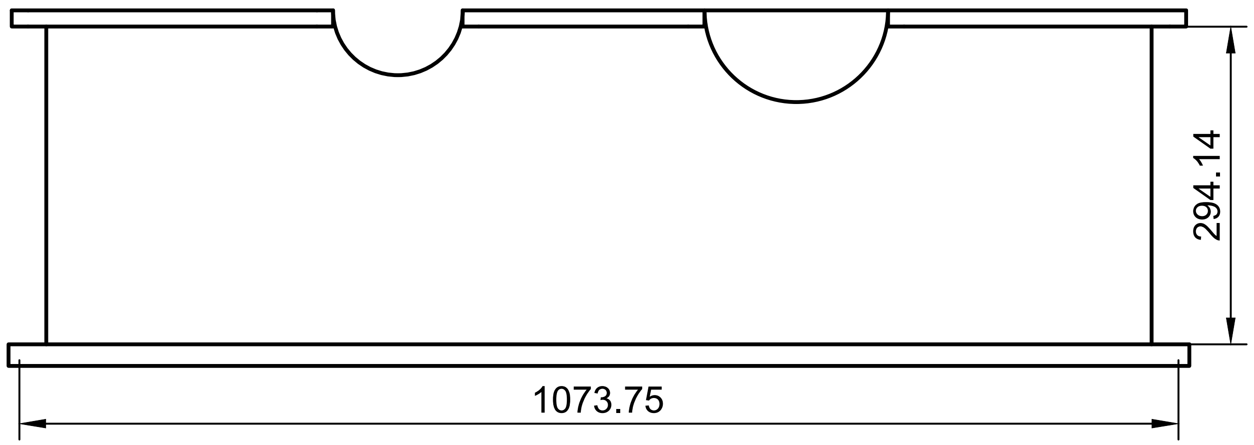


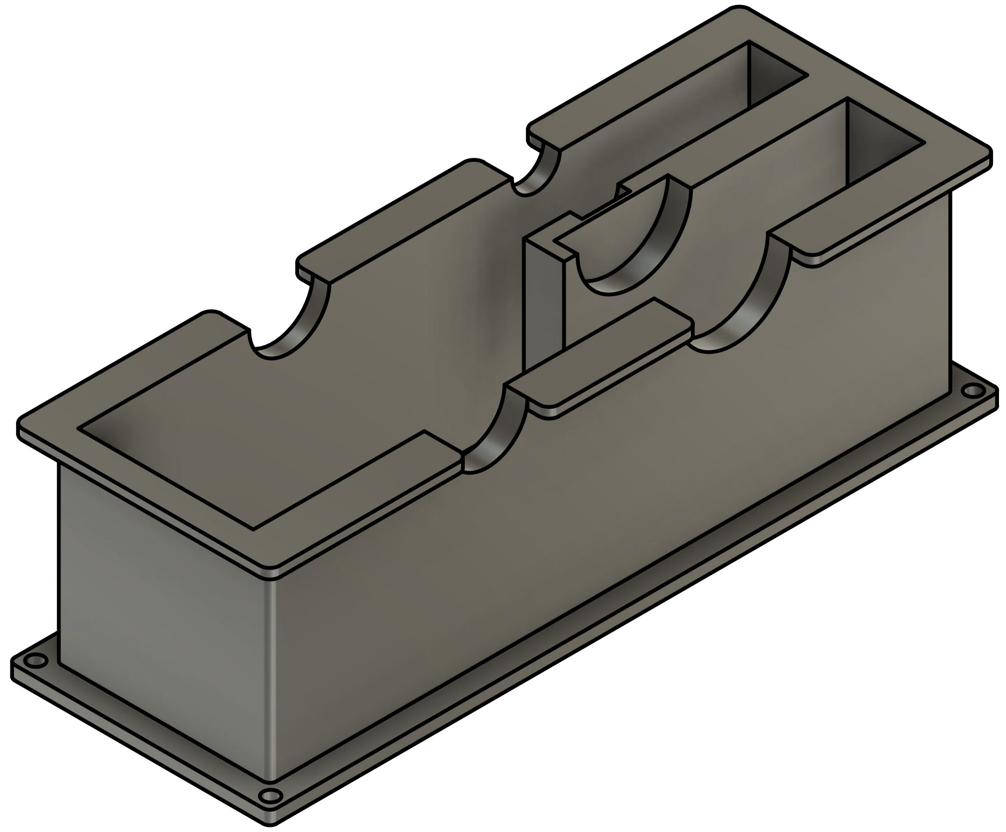


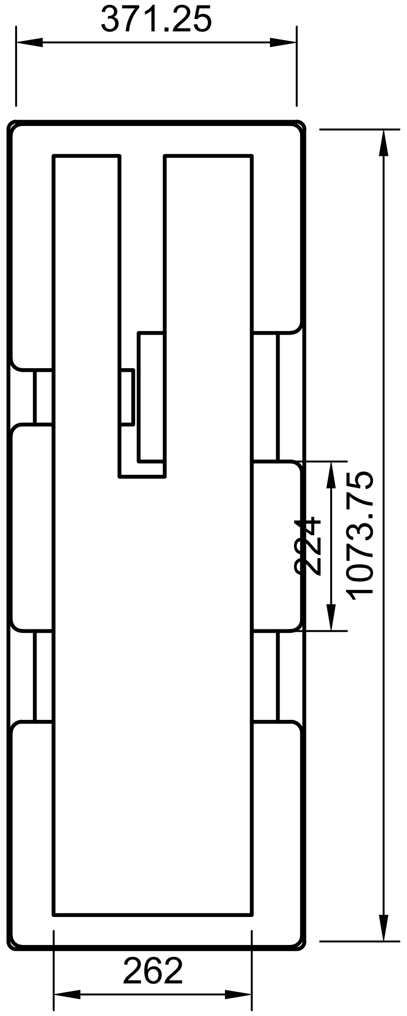


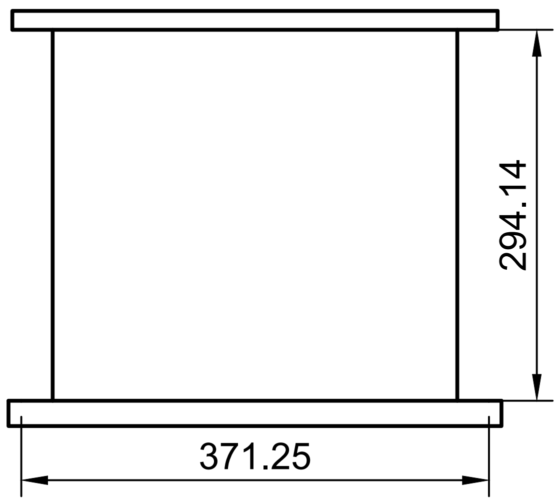


Dept.	Technical reference	Created by	Approved by		
		Divyanshu Gupta 01/05/2022			
		Document type	Document status		
		Title	DWG No.		
		Assembly_Casing_Cover			
			Rev.	Date of issue	Sheet
					1/1









Dept.	Technical reference	Created by Divyanshu Gupta 01/05/2022	Approved by
		Document type	Document status
		Title Casing_Base_V2	DWG No.
			Rev. / Date of issue / Sheet 1/1