مدينة زويل للعلوم والتكنولوجيا

Zewail City of Science and Technology

School of Engineering

Machine Design II Course Project

REE 312

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Contents

Table of contributions

Student Name	Contributions	% Contribution
Marwan Amr	Compiled the report. Aid in calculations	30%
Walid Sherif	Sections 1.1, 1.2, 1.3, 1.4, 1.5, 2.6.1, 2.6.2, 2.6.3. For CAD: • Entire casing • Oil seals • Assembly of parts • Used Autodesk Inventor 2024	39%
Abdelrahman Mohd	Sections 2.1, 2.2, 2.3, 2.4.1, 2.4.2, 2.4.3, 2.5, 2.6.4. For CAD: • Shafts A, B, C • Shaft covers • Assembly drawing sheet • Used SOLIDWORKS 2021	31%

Table 1: Summary of individual contributions by students.

Chapter 1

Gear System Design

Given Parameters

- Power: $P = 55\,\text{kW}$
- Input speed: $n_2 = 5.0\,\text{rpm}$
- Output speed: $n_{\text{out}} = 1000\,\text{rpm}$
- Total gear ratio: 20 (achieved in 2 stages)

1.1 Gear Ratio Calculation

The total gear ratio is the product of the ratios of each stage:

$$\text{Gear ratio} = q_1 \times q_2 \rightarrow q_{\text{eq}} = \sqrt{20} = 4.47 \quad \text{(per stage, assuming equal distribution)}$$

Gear Teeth Selection

The table below explores possible pinion teeth counts and their corresponding gear teeth, approximate ratios, and errors:

Pinion Teeth	Gear Teeth	Approx. Ratio	Total Ratio	Error
10	$4.47 \times 10 = 44.7$	45	$(4.5)^2 = 20.25$	1.23%
11	49.17	49	19.843	0.79%
12	53.64	54	20.25	1.23%
13	58.11	58	19.965	0.477%
14	62.58	63	20.25	1.23%
15	67.05	67	19.951	0.245%
16	71.52	72	20.25	1.23%

Two-Stage Gear Ratio Factorization

Since the total ratio must be exactly 20, equal distribution (4.47×4.47) is not feasible. Alternative factorizations:

$$20 = 4 \times 5 = 6 \times \frac{10}{3} = 8 \times \frac{5}{2}$$

The selected factorization is 4×5 with pinion teeth $N_3 = N_5 = 15$ (for speed increase).

Final Gear Teeth Configuration

- First stage: $N_2 = 60$, $N_3 = 15$ (Ratio $= 4$)
- Second stage: $N_4 = 75$, $N_5 = 15$ (Ratio $= 5$)

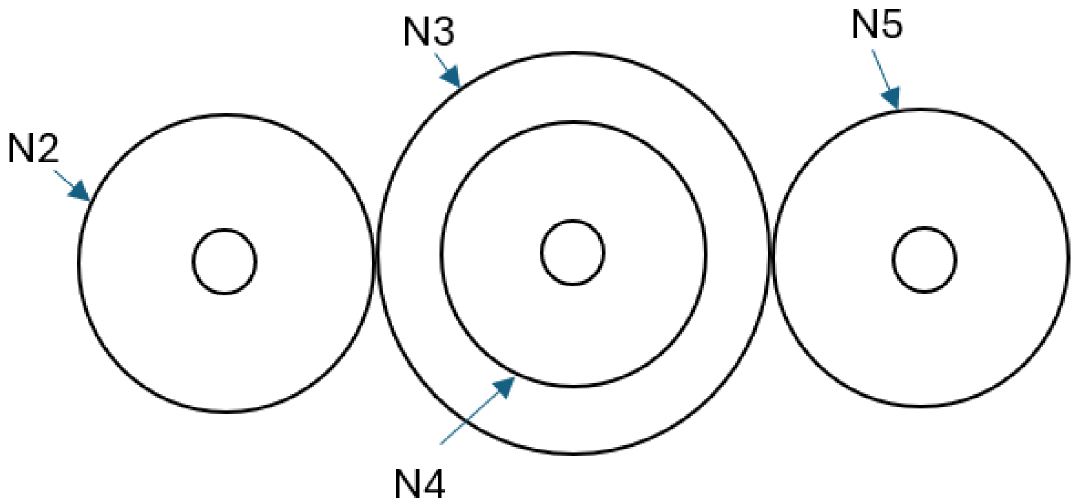

Figure 1.1: This model's speed is increased

1.2 Gear System Force and Torque Calculations

Initial Parameters and Gear Dimensions

- Pressure angle: $\theta = 20°$
- Power equation:

$$P = \frac{W_T \pi d n}{6 \times 10^3} \quad \text{(Equation 1)}$$

- Assumed modules:

$$M_1 = 15, \quad M_2 = 10$$

- Solving the equation:

$$P = T_i n W_i n \qquad 55 = T_i n \times 50 \times \frac{2\pi}{60} \qquad T_i n = 10.5\text{kn m}$$

- Gear diameters and face widths:

$$\begin{aligned} d_2 &= 6 \times 15 = 900\,\text{mm} && \rightarrow \text{Face width} = 90\,\text{mm} \\ d_3 &= 15 \times 15 = 225\,\text{mm} && \rightarrow \text{Face width} = 90\,\text{mm} \\ d_4 &= 75 \times 10 = 750\,\text{mm} && \rightarrow \text{Face width} = 56\,\text{mm} \\ d_5 &= 15 \times 10 = 150\,\text{mm} && \rightarrow \text{Face width} = 56\,\text{mm} \end{aligned}$$

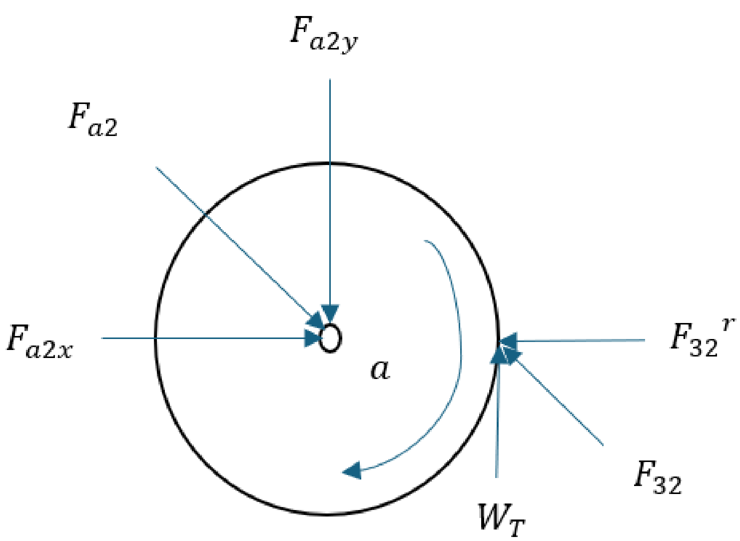

Figure 1.2: Free body diagram of the forces on gear 1

Tangential and Resultant Forces

- Tangential force calculation:

$$W_T = \frac{60000 \times P}{\pi d_2 n_2} = \frac{60000 \times 55}{\pi \times 900 \times 50} = 23.34\,\mathrm{kN}$$

- Radial force:

$$W_R = W_T \tan 20^\circ = 23.34 \times \tan 20^\circ = 8.49\,\mathrm{kN}$$

- Resultant force:

$$F_{32} = \sqrt{(23.34)^2 + (8.49)^2} = 24.83\,\mathrm{kN}$$

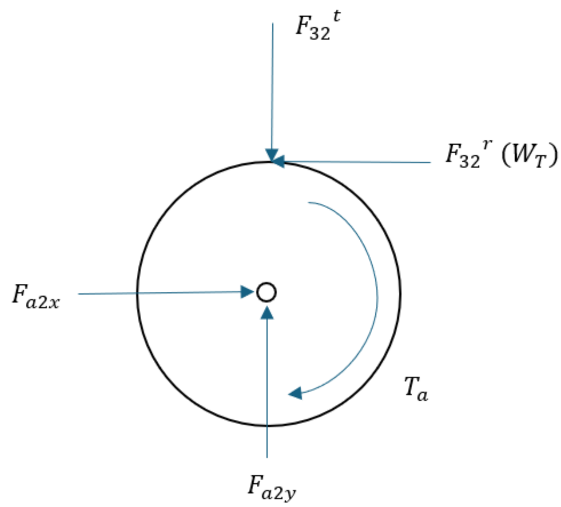

Figure 1.3: N2 GEAR

- Force components:

$$F_{32}^{X} = 23.34\,\mathrm{kN}, \quad F_{32}^{Y} = 8.49\,\mathrm{kN}$$

- Torque on gear 3:

$$T_{b3} = 23.34 \times \frac{225}{2} = 2625.75\,\mathrm{N \cdot m}$$

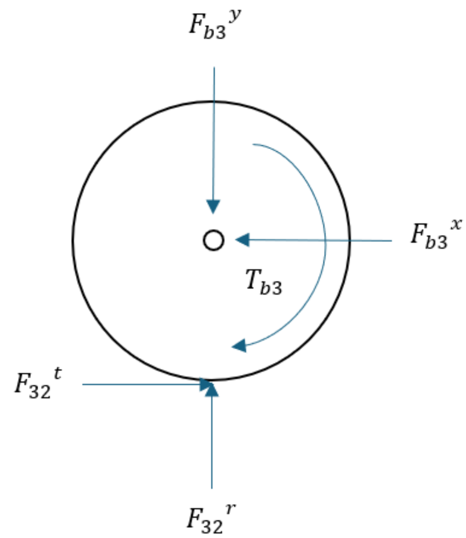

Figure 1.4: N3 GEAR

Forces on Subsequent Gears

- Torque equilibrium:

$$2625.75 = W_T \times \frac{750}{2} \implies W_T = 7.002\,\text{kN}$$

- Radial force on gear 4:

$$F_{54}^{Y} = W_T \tan 20^\circ = 7.002 \times \tan 20^\circ = 2.548\,\text{kN}$$

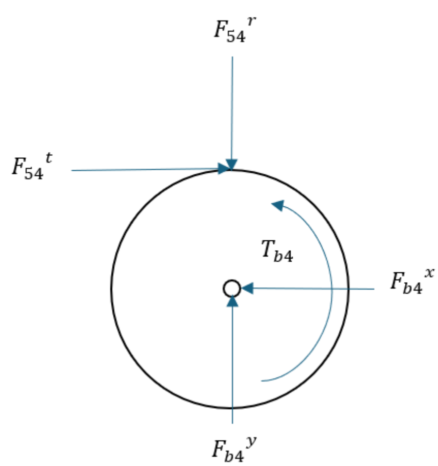

Figure 1.5: N4 GEAR

- Force components:
$$F_{64}^X = 7.002\,\text{kN}, \quad F_{64}^Y = 2.548\,\text{kN}$$
- Forces on gear 5:
$$F_{45}^X = 7.002\,\text{kN}, \quad F_{45}^Y = 2.548\,\text{kN}$$
- Torque on gear 5:
$$T_{C5} = 7.002 \times \frac{150}{2} = 525.15\,\text{N}\cdot\text{m}$$

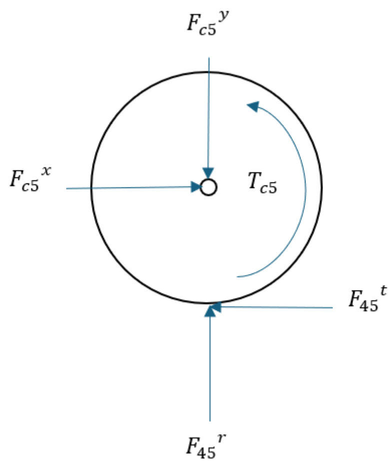

Figure 1.6: N5 GEAR

1.3 1st Stage Gear Analysis

Given Parameters

- Module: $M = 15\,\text{mm}$
- Pressure angle: $\Phi = 20°$
- Face width: $F = 90\,\text{mm}$
- Tangential force: $W_T = 23.34\,\text{kW}$
- Pinion diameter: $d_P = 225\,\text{mm}$
- Gear diameter: $d_G = 900\,\text{mm}$
- Bending factors: $(Y_J)_P = 0.440$, $(Y_J)_G = 0.492$
- Load factors: $K_O = 1.2$, $K_V = 1.054$, $K_S = 1$, $J = 0.09$, $K_R = 1$, $K_T = 1$, $K_H = 1.238$, $q_{mc} = 1$, $C_e = 1$, $C_{pm} = 1$
- Reliability factors: $K_B = 1$, $(Y_N)_P = 0.923$, $(Y_N)_G = 0.965$

Bending Stress Calculations

For the gear:

$$\sigma = W_T K_O K_V K_S \frac{1}{bm_T} \cdot \frac{K_H K_B}{Y_J}$$

$$\sigma = 23.34 \times 10^3 \times 1.2 \times 1.054 \times 1 \times \frac{1}{15 \times 90} \times \frac{1.238 \times 1}{0.492} = 55.023\,\text{MPa}$$

$$(S_F)_{\text{Gear}} = \frac{152 \times 0.965}{1 \times 1 \times 55.023} = 2.66$$

For the pinion:

$$\sigma = 23.34 \times 10^3 \times 1.2 \times 1.054 \times 1 \times \frac{1}{15 \times 90} \times \frac{1.238}{0.440} = 61.52\,\text{MPa}$$

$$(S_F)_{\text{Pinion}} = \frac{152 \times 0.923}{1 \times 1 \times 61.52} = 2.28$$

Contact Stress Calculations

- Life factors: $(Z_N)_P = 0.870$, $(Z_N)_G = 0.940$
- Geometry factors: $Z_W = 1$, $Z_I = 0.097$, $Z_C = 191$
- Pinion diameter: $dw_1 = 225\,\text{mm}$, $Z_R = 1$

$$\sigma_c = Z_E \sqrt{W_T K_O K_V K_S \frac{K_H}{dw_1 b} \frac{Z_R}{Z_T}}$$

$$\sigma_c = 191 \times \sqrt{\frac{23.34 \times 10^3 \times 1.2 \times 1.054 \times 1 \times 1.238}{225 \times 90 \times 0.097}} = 823.86\,\text{MPa}$$

$$S_H = \frac{S_C Z_N Z_W}{Y_\theta Y_Z \sigma_c}$$

For the gear:

$$(S_H)_{\text{Gear}} = \frac{1170 \times 0.940 \times 1}{1 \times 1 \times 823.86} = 1.33$$

For the pinion:

$$(S_H)_{\text{Pinion}} = \frac{1170 \times 0.870}{823.86} = 1.235$$

1.4 2nd Stage Gear Analysis

Given Parameters

- Module: $M = 10\,\text{mm}$
- Pressure angle: $\Phi = 20°$
- Face width: $F = 56\,\text{mm}$

- Tangential force: $W_T = 7.002\,\text{kW}$
- Pinion diameter: $d_P = 150\,\text{mm}$
- Gear diameter: $d_G = 750\,\text{mm}$
- Bending factors: $(Y_J)_P = 0.445$, $(Y_J)_G = 0.491$
- Load factors: $K_O = 1.2$, $K_V = 1.093$, $K_S = 1$, $J = 0.100$, $K_R = 1$, $K_T = 1$, $K_H = 1.201$, $C_m = 1$, $C_e = 1$, $C_p = 1$
- Reliability factors: $K_B = 1$, $(Y_N)_P = 0.876$, $(Y_N)_G = 0.923$

Bending Stress Calculations

For the gear:

$$\sigma = 7.002 \times 10^3 \times 1.2 \times 1.093 \times 1 \times \frac{1}{56 \times 10} \times \frac{1.201 \times 1}{0.491} = 40.11\,\text{MPa}$$

$$(S_F)_{\text{Gear}} = \frac{152 \times 0.923}{1 \times 1 \times 40.11} = 3.498$$

For the pinion:

$$\sigma = 7.002 \times 10^3 \times 1.2 \times 1.093 \times \frac{1}{56 \times 10} \times \frac{1.201}{0.445} = 44.26\,\text{MPa}$$

$$(S_F)_{\text{Pinion}} = \frac{152 \times 0.876}{1 \times 1 \times 44.26} = 3.000$$

Contact Stress Calculations

- Life factors: $(Z_N)_P = 0.795$, $(Z_N)_G = 0.876$
- Geometry factors: $Z_W = 1$, $Z_I = 0.100$, $Z_C = 191$
- Pinion diameter: $dw_1 = 150\,\text{mm}$, $Z_R = 1$

$$\sigma_c = Z_E \sqrt{W_T K_O K_V K_S \frac{K_H}{dw_1 b} \frac{Z_R}{Z_T}}$$

$$\sigma_c = 191 \times \sqrt{\frac{7.002 \times 10^3 \times 1.2 \times 1.093 \times 1 \times 1.201}{150 \times 56 \times 0.100}} = 692.11\,\text{MPa}$$

$$S_H = \frac{S_C Z_N Z_W}{Y_\theta Y_Z \sigma_c}$$

For the gear:

$$(S_H)_{\text{Gear}} = \frac{1170 \times 0.876 \times 1}{1 \times 1 \times 692.11} = 1.47$$

For the pinion:

$$(S_H)_{\text{Pinion}} = \frac{1170 \times 0.795}{1 \times 1 \times 692.11} = 1.34$$

1.5 Autodesk Inventor Analysis of Gear stages.

Below are results of the Autodesk Inventor analysis for both gear stages, showing design compliance.

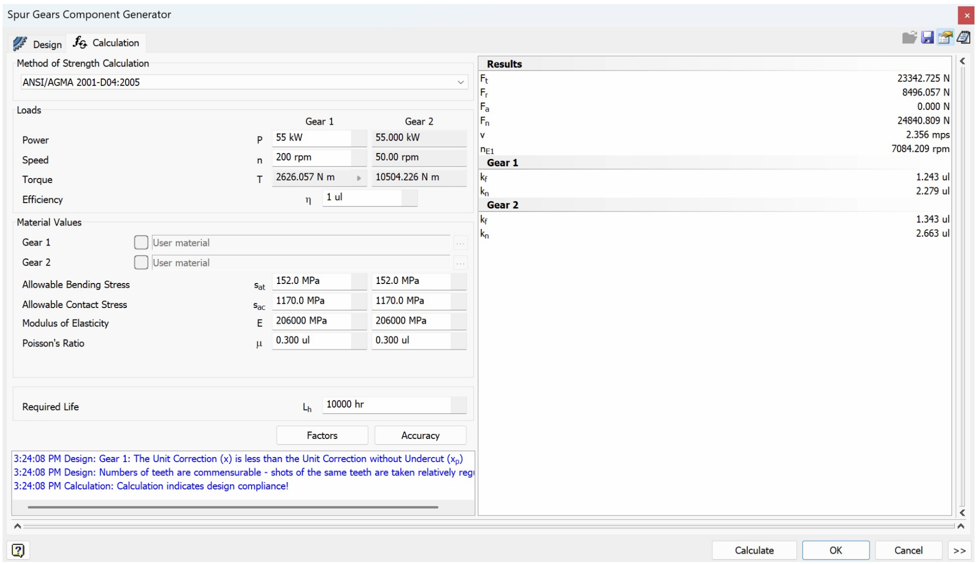

Figure 1.7: First gear stage analysis on Inventor

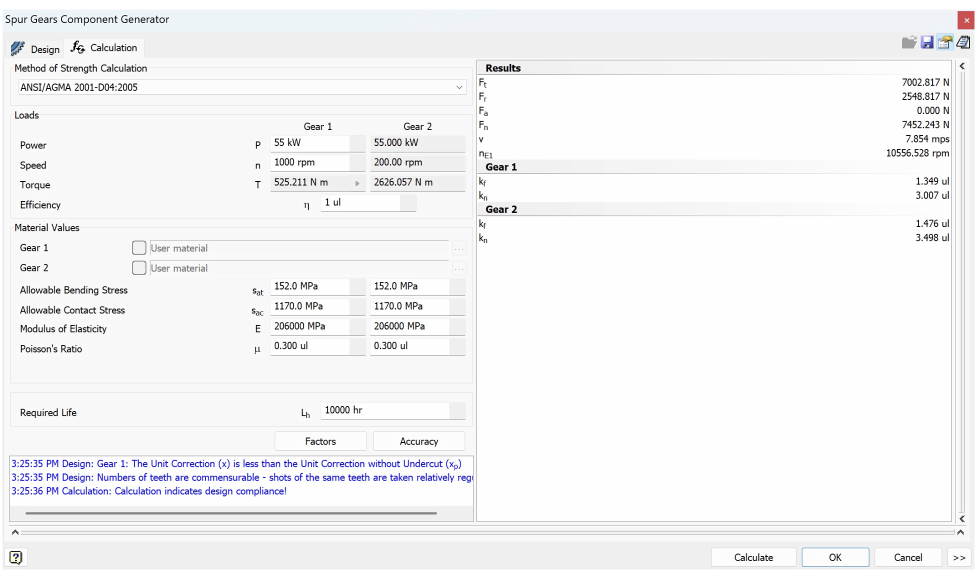

Figure 1.8: Second gear stage analysis on Inventor

Chapter 2

Shaft Design

2.1 Givens & Assumptions

- **Shaft Configuration**:
  - Shaft a: Gear 2 at 150 mm from the left bearing.
  - Shaft b: Gear 3 at 150 mm and Gear 4 at 300 mm from the left bearing.
  - Shaft c : Gear 5 at 300 mm from the left bearing.
- **Bearing-to-Bearing Distance**: 500 mm for all shafts.
- **Forces and Torques**:
  - Gear 2: $F_{2a_x} = 25.01\,\text{kN}$ (tangential), $F_{2a_y} = 9.1029\,\text{kN}$ (radial).
  - Gear 3: $F_{b3_x} = 25.01\,\text{kN}$, $F_{b3_y} = 9.1029\,\text{kN}$, $T_{B3} = 2626.05\,\text{N.m}$
  - Gear 4: $F_{b4_x} = 7.7779\,\text{kN}$, $F_{b4_y} = 2.83\,\text{kN}$
  - Gear 5: $F_{c5_x} = 7.7779\,\text{kN}$, $F_{c5_y} = 2.83\,\text{kN}$, $T_{c5} = 525.00825\,\text{N.m}$
- **Material**: AISI 1045 steel, yield strength $\sigma_y = 350\,\text{MPa}$
- **Target Safety Factor**: ≥ 1.2
- **Allowable Shear Stress**:

$$\tau_{\text{allow}} = 0.30 \times \frac{\sigma_y}{\text{safety factor}} = 0.30 \times \frac{350}{2} = 52.5\,\text{MPa}$$

- **Input Torque**: $T_{\text{in}} = 10.5\,\text{kN.m}$

2.2 Shaft A (input shaft)

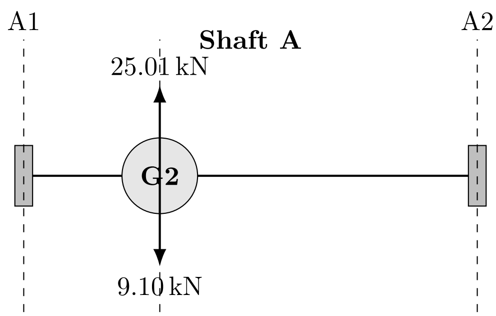

Figure 2.1: Shaft A layout, A1 & A2 are bearings.

- Gear 2 (at 150 mm):

$$F_y = -25{,}010\,\text{N}, \quad F_z = -9{,}102.9\,\text{N}$$

- Bearing reactions:

$$R_{yb} = \frac{25{,}010 \times 150}{500} = 7{,}503\,\text{N}, \quad R_{ya} = 17{,}507\,\text{N}$$

$$R_{zb} = \frac{9{,}102.9 \times 150}{500} = 2{,}730.87\,\text{N}, \quad R_{za} = 6{,}372.03\,\text{N}$$

- Bending moments:

$$M_y = 2{,}626{,}050\,\text{N.mm}, \quad M_z = 955{,}804.5\,\text{N.mm}$$

$$M = \sqrt{2{,}626{,}050^2 + 955{,}804.5^2} \approx 2{,}794{,}583\,\text{N.mm}$$

- Torque: $T = 10{,}500{,}000\,\text{N.mm}$
- Diameter:

$$d = \left(\frac{16}{\pi \times 52.5} \times \sqrt{M^2 + T^2} \right)^{1/3} \approx \boxed{101.7\,\text{mm}}$$

2.3 Shaft B (intermediate shaft)

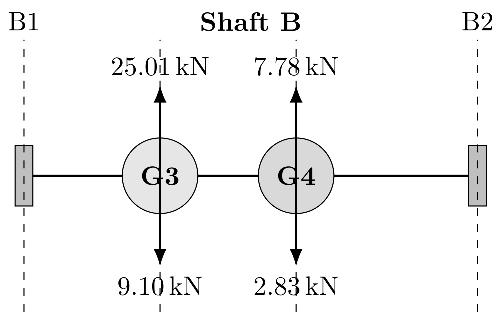

Figure 2.2: Shaft B layout

- Forces at gears 3 and 4:

$$F_{y3} = -25{,}010\,\mathrm{N}, \quad F_{z3} = -9{,}102.9\,\mathrm{N}$$

$$F_{y4} = -7{,}779.9\,\mathrm{N}, \quad F_{z4} = -2{,}830\,\mathrm{N}$$

- Reactions:

$$R_{yb} = 12{,}170.94\,\mathrm{N}, \quad R_{ya} = 20{,}618.96\,\mathrm{N}$$

$$R_{zb} = 4{,}428.87\,\mathrm{N}, \quad R_{za} = 7{,}504.03\,\mathrm{N}$$

- Bending moments at 150 mm:

$$M_y = 3{,}092{,}844\,\mathrm{N.mm}, \quad M_z = 1{,}125{,}604.5\,\mathrm{N.mm}$$

$$M = \sqrt{M_y^2 + M_z^2} \approx 3{,}291{,}135\,\mathrm{N.mm}$$

- Torque: $T = 2{,}626{,}050\,\mathrm{N.mm}$
- Diameter:

$$d = \left(\frac{16}{\pi \times 52.5} \times \sqrt{M^2 + T^2}\right)^{1/3} \approx \boxed{74.2\,\mathrm{mm}}$$

Shaft C (output shaft)

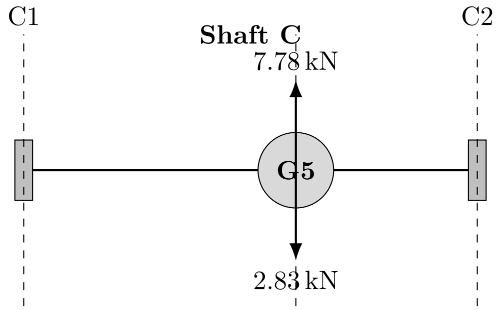

Figure 2.3: Shaft C layout.

- Forces at Gear 5 (x = 300 mm):

$$F_y = -7{,}779.9\,\mathrm{N}, \quad F_z = -2{,}830\,\mathrm{N}$$

- Reactions:

$$R_{yb} = 4{,}667.94\,\mathrm{N}, \quad R_{ya} = 3{,}111.96\,\mathrm{N}$$
$$R_{zb} = 1{,}698\,\mathrm{N}, \quad R_{za} = 1{,}132\,\mathrm{N}$$

- Bending moments:

$$M_y = 933{,}588\,\mathrm{N.mm}, \quad M_z = 339{,}600\,\mathrm{N.mm}$$
$$M = \sqrt{M_y^2 + M_z^2} \approx 993{,}238\,\mathrm{N.mm}$$

- Torque: $T = 525{,}008.25\,\mathrm{N.mm}$
- Diameter:

$$d = \left(\frac{16}{\pi \times 52.5} \times \sqrt{M^2 + T^2}\right)^{1/3} \approx \boxed{47.7\,\mathrm{mm}}$$

2.4 Bearing selection

We already computed radial loads above, we proceed with bearing life calculations directly.

It is worth noting that the standard Inventor bearings library does not have suitable bearings for our design; therefore, we choose from the SKF bearing library.

2.4.1 Shaft A

Life:

$$L_{10,A1} = \left(\frac{127}{18.63}\right)^3 \times 10^6 \approx 316.8 \times 10^6 \text{ rev}$$
$$L_{10h,A1} = \frac{316.8 \times 10^6}{60 \times 50} \approx 105\,600\,\text{h}$$
$$L_{10h,A2} \approx 1\,343\,000\,\text{h}$$

2.4.2 Shaft B

Life Calculation:

$$L_{10h,B1} \approx 13\,292\,\text{h}, \quad L_{10h,B2} \approx 64\,708\,\text{h}$$

2.4.3 Shaft C

Calculating life:

$$L_{10h,C1} \approx 109\,500\,\text{h}, \quad L_{10h,C2} \approx 32\,333\,\text{h}$$

We therefore choose the following bearings:

- Shaft A: 100 mm, SKF 6220-2RS1, $C = 127\,\text{kN}$
- Shaft B: 75 mm, SKF 6315-2RS1, $C = 119\,\text{kN}$
- Shaft C: 50 mm, SKF 6310-2RS1, $C = 62\,\text{kN}$

Target bearing life: $L_{10h} > 10\,000\,\text{h}$ achieved.

Shaft	**Bearing**	**Model**	**ID (mm)**	**Life (h)**
A	A1, A2	SKF 6220-2RS1	100	105600 / 1343000
B	B1, B2	SKF 6315-2RS1	75	13292 / 64708
C	C1, C2	SKF 6310-2RS1	50	109500 / 32333

2.5 Keys

The key width's table in REE302's slides for key design also doesn't have suitable widths that fall within our calculations, therefore, we use:
Key widths per DIN 6885:

- $58 \leq d \leq 65$: 18 mm
- $75 \leq d \leq 85$: 22 mm
- $85 < d \leq 95$: 25 mm

- $95 < d \leq 110$: 28 mm

Shear stress: $\tau = \frac{2T}{d \cdot w \cdot L} \leq \tau_{\text{allow}}$
Thus, $L \geq \frac{2T}{d \cdot w \cdot \tau_{\text{allow}}}$

Shaft a, Gear 2

Given: $T = 10\,500\,000\,\text{N mm}$, $d = 110\,\text{mm}$, $w = 28\,\text{mm}$, $\tau_{\text{allow}} = 60\,\text{N mm}^{-2}$.

$$L \geq \frac{2 \cdot 10500000}{110 \cdot 28 \cdot 60} = \frac{21000000}{184800} \approx 113.6\,\text{mm}$$

Rounded: $L = 115\,\text{mm}$.

Shaft b, Gear 3

Given: $T = 2\,625\,000\,\text{N mm}$, $d = 80\,\text{mm}$, $w = 22\,\text{mm}$.

$$L \geq \frac{2 \cdot 2625000}{80 \cdot 22 \cdot 60} = \frac{5250000}{105600} \approx 49.7\,\text{mm}$$

Rounded: $L = 50\,\text{mm}$.

Shaft b, Gear 4

Given: $T = 2\,625\,000\,\text{N mm}$, $d = 90\,\text{mm}$, $w = 25\,\text{mm}$.

$$L \geq \frac{2 \cdot 2625000}{90 \cdot 25 \cdot 60} = \frac{5250000}{135000} \approx 38.9\,\text{mm}$$

Rounded: $L = 40\,\text{mm}$.

Shaft c, Gear 5

Given: $T = 525\,000\,\text{N mm}$, $d = 60\,\text{mm}$, $w = 18\,\text{mm}$.

$$L \geq \frac{2 \cdot 525000}{60 \cdot 18 \cdot 60} = \frac{1050000}{64800} \approx 16.2\,\text{mm}$$

Minimum practical: $L = 20\,\text{mm}$.

Shaft	Gear	Width (mm)	Height (mm)	Length (mm)
a	2	28	28	115
b	3	22	22	50
b	4	25	25	40
c	5	18	18	20

Table 2.1: Summary of key widths and lengths for all shafts.

2.6 Analysis using Autodesk Inventor

2.6.1 Shaft A

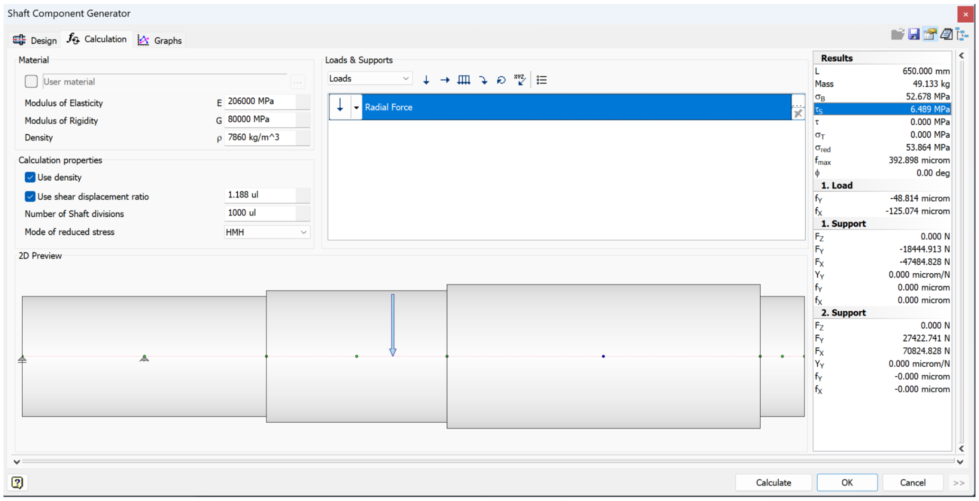

Figure 2.4: Shaft A analysis on Autodesk Inventor

2.6.2 Shaft B

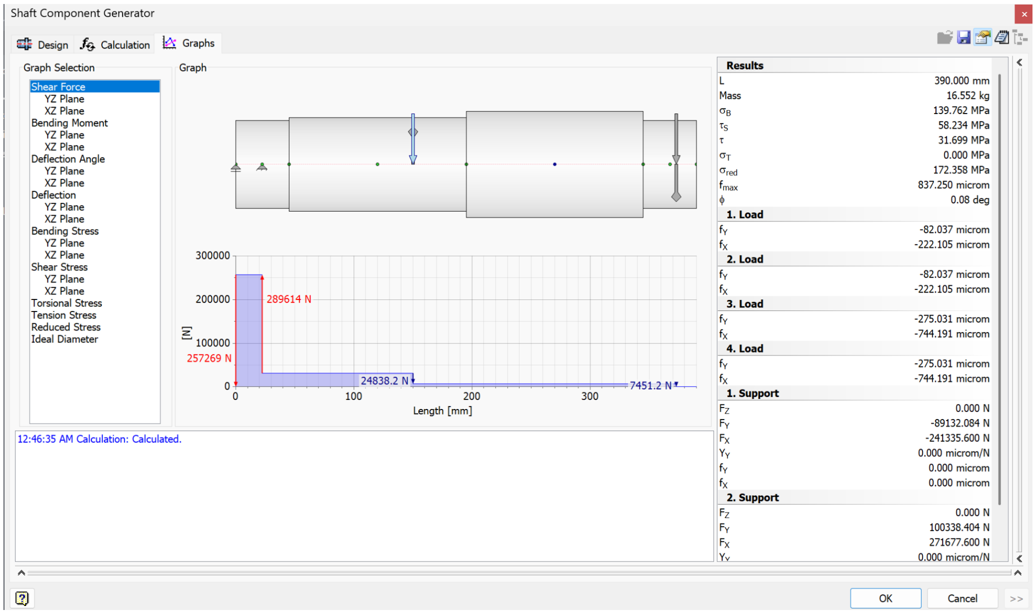

Figure 2.5: Shaft B analysis on Autodesk Inventor

2.6.3 Shaft C

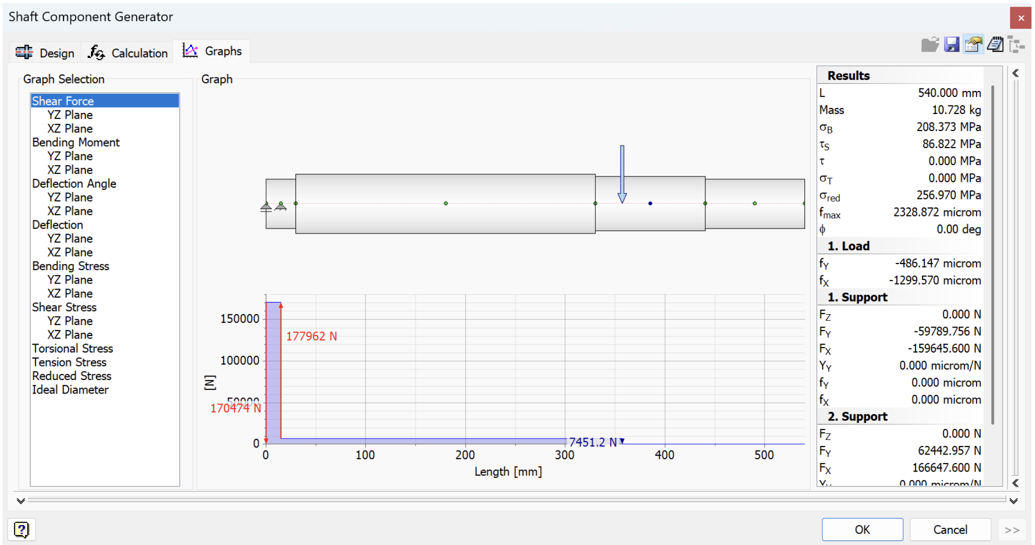

Figure 2.6: Shaft C analyzed on autodesk Inventor

2.6.4 Factors of Safety

Material assumed: AISI 1045 steel.

The factor of safety (n) is calculated using:

$$n = \frac{S_y}{\sigma_{\text{eq}}}, S_y = 310 \text{ MPa}$$

Summary Table

Table 2.2: Equivalent Stress and Factor of Safety for AISI 1045 Steel

Shaft	Equivalent Stress (MPa)	Yield Strength (MPa)	Factor of Safety
A	53.864	310	5.76
B	172.358	310	1.80
C	256.970	310	1.21

Chapter 3

CAD Drawing

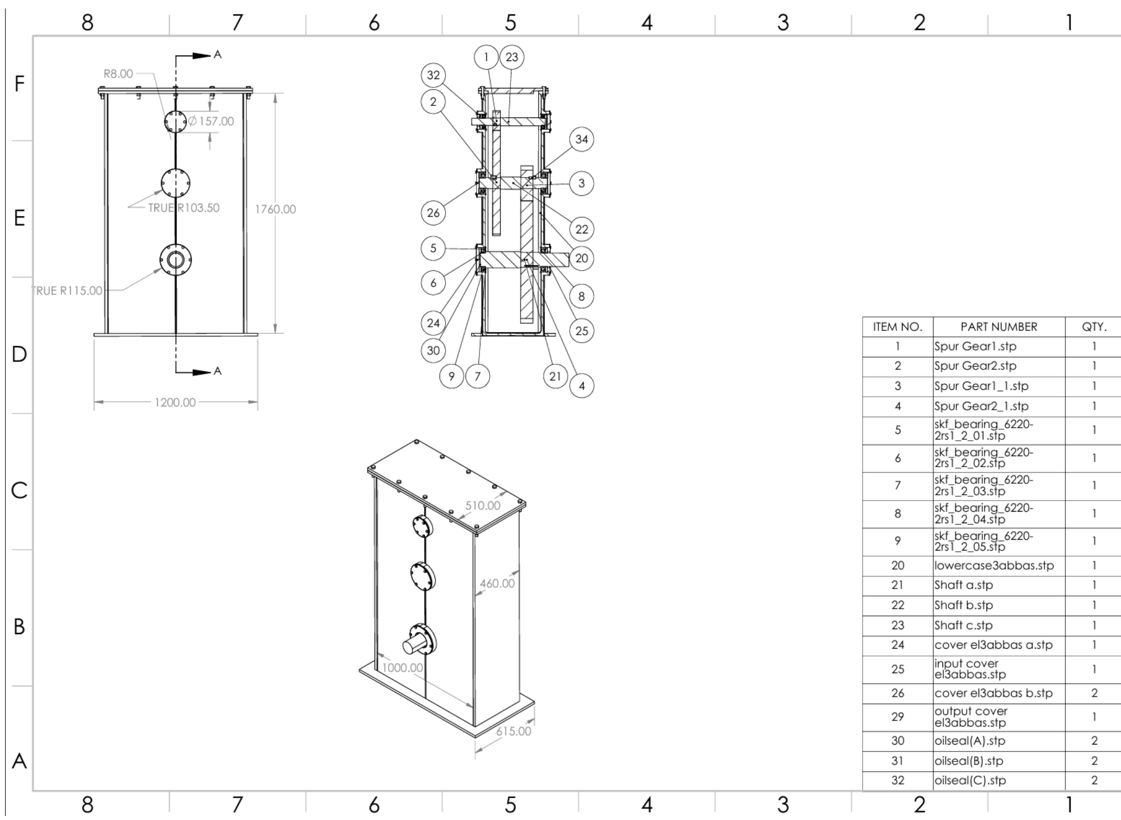

Figure 3.1: Final assembly; front, section side, and isometric views. Displayed on SOLIDWORKS 2021.

3.1 Extra drawing

Right before we were going to submit, Walid informed us that the gearbox should be vertical! And this is how the gearbox looked like that time:

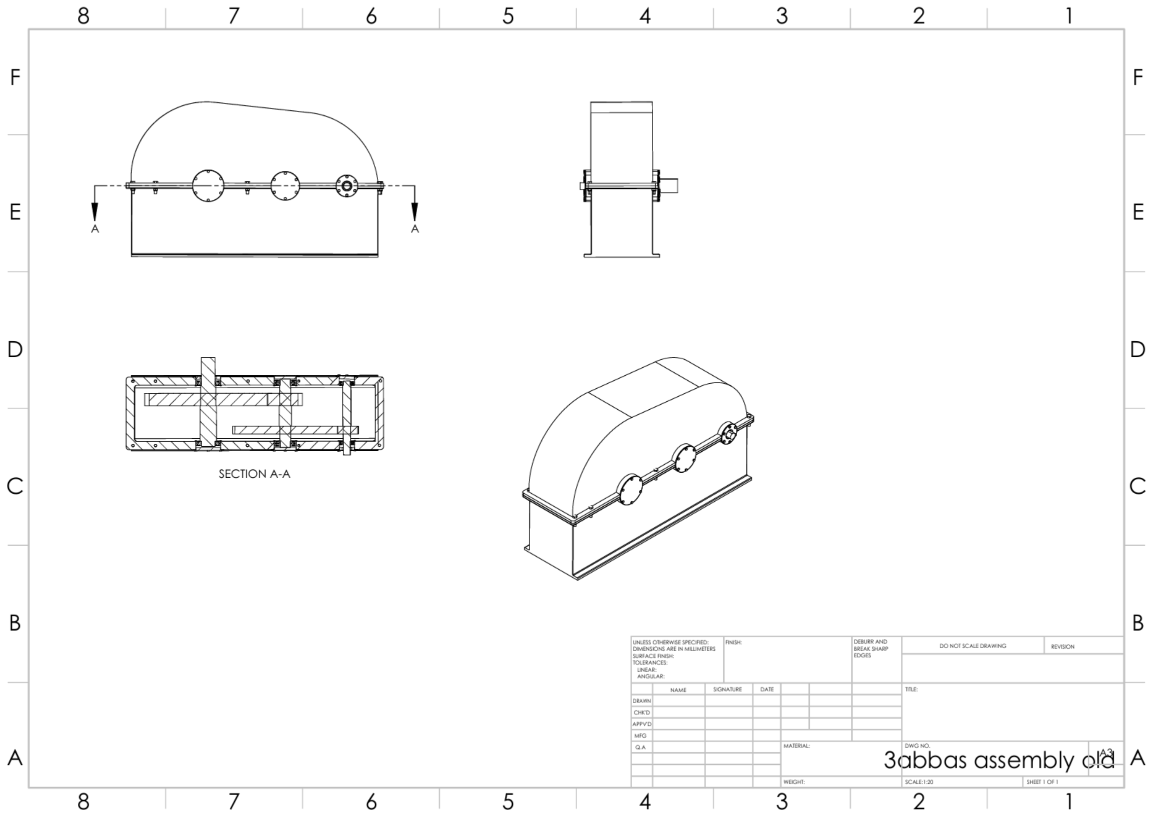

Figure 3.2: Old design (horizontal arrangement)