ME 424

MACHINE DESIGN II

GEAR BOX DESIGN

Ömer Faruk Seven

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Course Instructor | Prof. Nuri Ersoy

Abstract

The report starts with an introduction where the design objectives and requirements are discussed. Then, the assumptions made during the design calculations are listed. The introduction is followed by the design configuration part where the overall system configuration is provided. In addition, 3D CAD model of the gearbox and the technical drawings of each component are provided. Subsequently, hand calculations for the verification of the selected design parameters are carried out in the following section – ending up with a table that compares results acquired from both hand calculations and Kisssoft computer software. In the conclusion part, key insights derived from this project are discussed

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1. Introduction

This report aims to elaborate on the procedure of gear box design. The design is carried out in a computer environment, utilizing software called Kisssoft. The verification of the design is also provided via set of hand calculations. Results acquired from both methods are subjected to a comparison in the end.

The objective is to design a gearbox that can transmit power from an input shaft to a parallel output shaft while substantially reducing rotational speed. Both design criteria and assumptions made in creating engineering models are listed below.

Design Criteria:

- The system should be capable of transmitting 8 kW of power without mechanically failing.
- Helical stage and worm stage ratios are 4 and 37, respectively.
- The system is expected to work at least 10,000 h.
- The system is expected to be 95% reliable.
- The system should be as light and as compact as possible.
- Maximum temperature can be up to 50°C.
- Factory of safety is between 1.5-3.

Worm Gear Assumptions:

- The worm and gear are mounted and aligned to mesh properly at mutually perpendicular axis.
- All the tooth load is transmitted at the pitch point and in the midplane of the gears.
- Churning of lubricating oil due to heating is neglected.
- The worm gear is produced by hobbing with shaping cutters.
- Bending fatigue of the worm gear is taken as Buckingham's proposal of 24 ksi.
- Ambient temperature is below 100°F (35°C). [More conservative temperature value is taken rather than 50°C.]
- A steady-state temperature of 100°F above temperature is acceptable. (i.e., oil temperature can be 200°F at maximum.)

Bevel Gear Assumptions:

- The gears are mounted to mesh along their pitch cones.
- All the tooth load is transmitted at the pitch point midway along the tooth face.
- Friction losses are neglected.
- Both source of power and load are uniform.
- The gears are manufactured by hobbing method.
- Selected metal for the gears are fine-grounded and commercailly polished.
- Temperature factor (k_T) is equal to 1. (Temperature is below 100°F)

Shaft and Bearing Assumptions:

- Forces are applied at the mid-width of each component and represented accordingly.
- L_{10} is taken as 10^6.
- Application factor (k_A) is equal to 1.1.
- In Shaft-1, axial load is accommodated only by Bearing-1.
- In Shaft-2, axial load is distributed equally between two bearings.
- In Shaft-3, axial load is distributed equally between two bearings.

2. Design Configuration

All figures provided in this part are generated using Kisssys, which is also used to model 3D CAD of the gearbox. Two-dimensional configuration of the gearbox design and corresponding transmission relationships are shown in Figure 1.

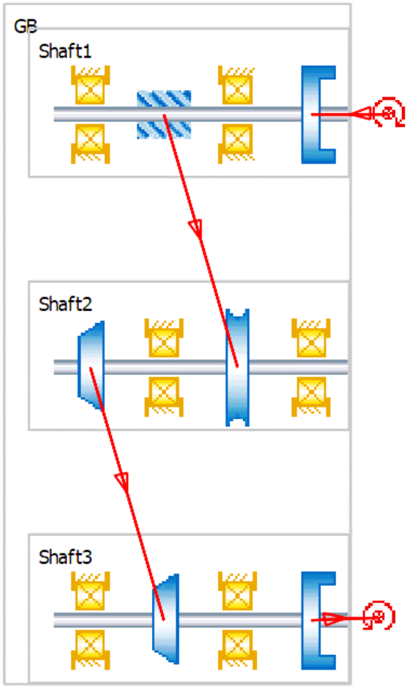


Figure 1: Gearbox 2D Configuration

In Figure 2 below, two dimensional detailed diagram for shaft 1 is provided. The components shown in the diagram are bearing 1, worm, bearing 2, and input coupling from left to right.

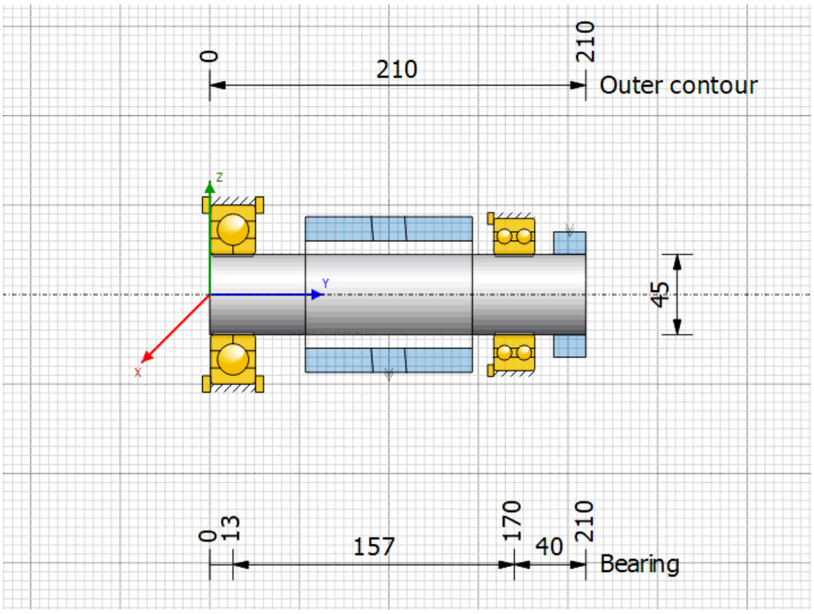


Figure 2: Detailed 2D Diagram of Shaft 1

In Figure 3 below, two dimensional detailed diagram for shaft 2 is provided. The components shown in the diagram are bevel pinion, bearing 3, worm gear, and bearing 4 from left to right.

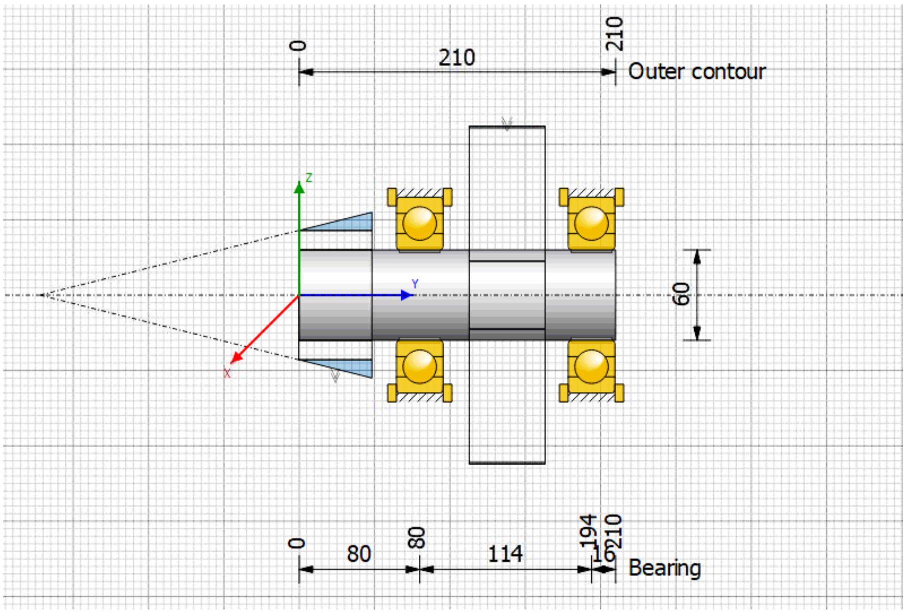


Figure 3: Detailed 2D Diagram of Shaft 2

In Figure 3 below, two dimensional detailed diagram for shaft 2 is provided. The components shown in the diagram are bearing 5, bevel gear, bearing 6, and output coupling from left to right.

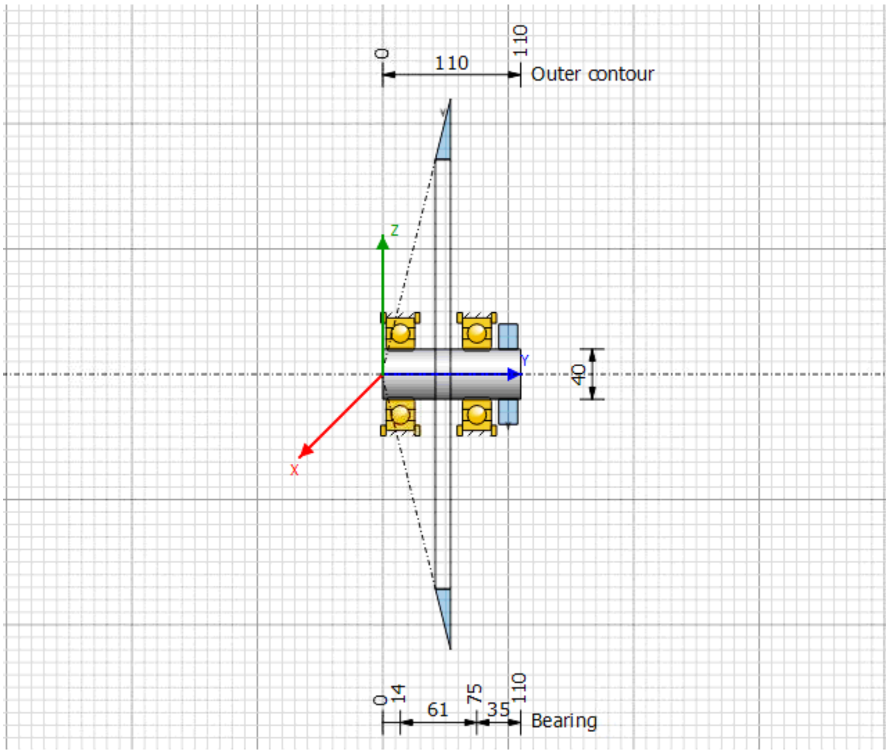


Figure 4: Detailed 2D Diagram of Shaft 3

Three dimensional CAD model of the gearbox is shown in Figure 5 below. 3D design modelled in Kissoft was then transferred to Solidworks to adjust texture in line with the material and to create technical drawings.

Figure 5: 3D CAD Model of the Gearbox

Another illustration of the CAD model from different view is provided in Figure 6.

Figure 6: 3D CAD Model (Different View)

In Figure 7, technical drawing of the worm gear is provided.

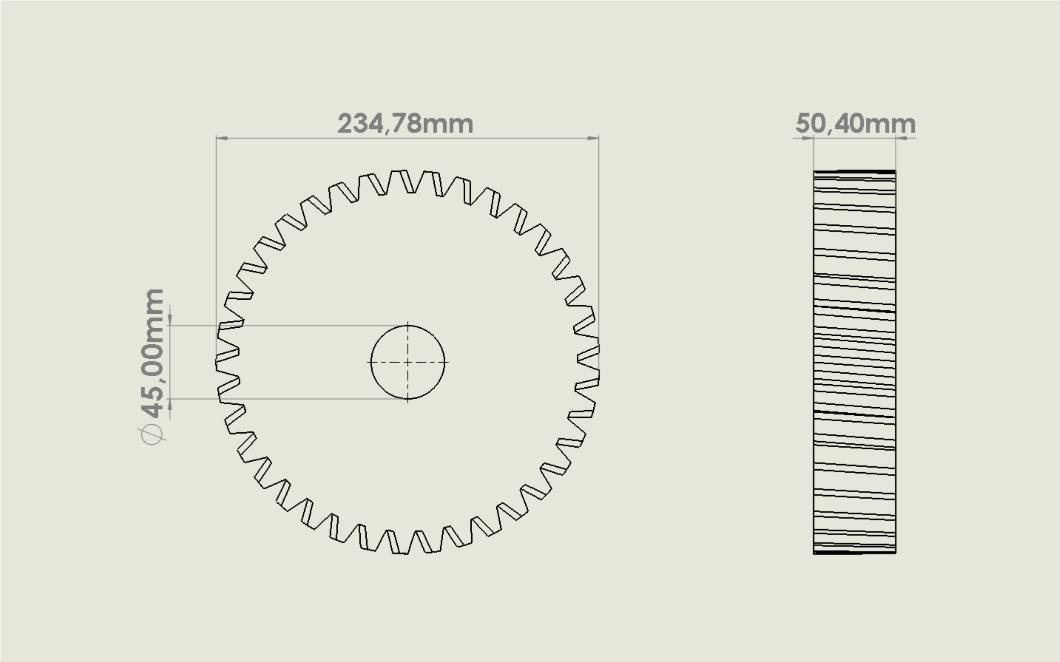


Figure 7: Technical Drawing of the Worm Gear

Technical drawing of the bevel pinion is shown in Figure 8 below.

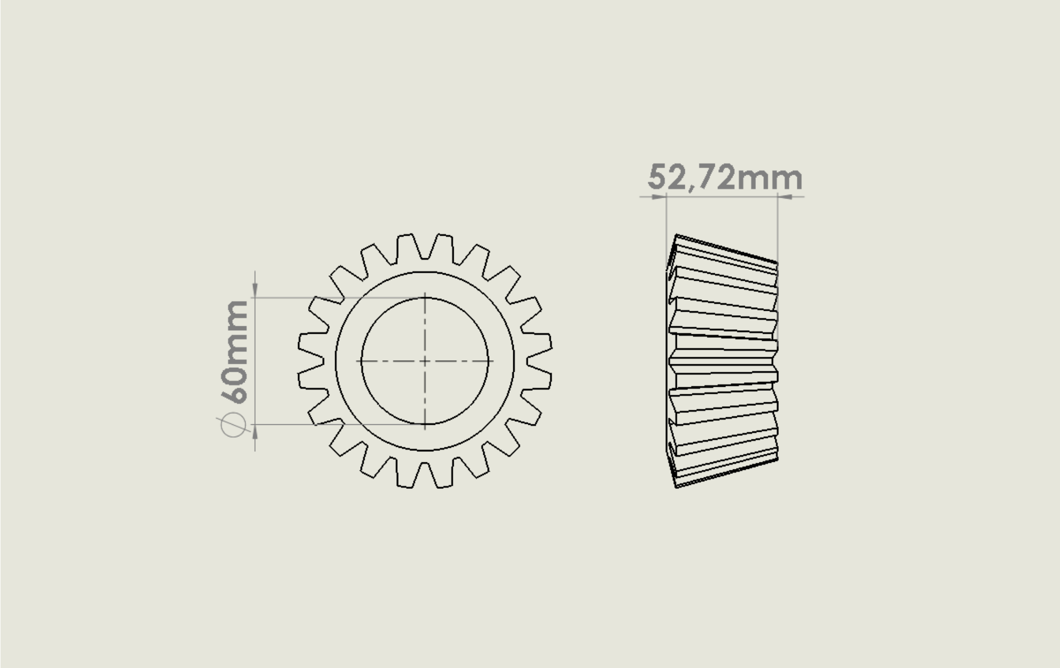


Figure 8: Technical Drawing of the Worm

Technical drawing of the bevel gear is given in Figure 9.

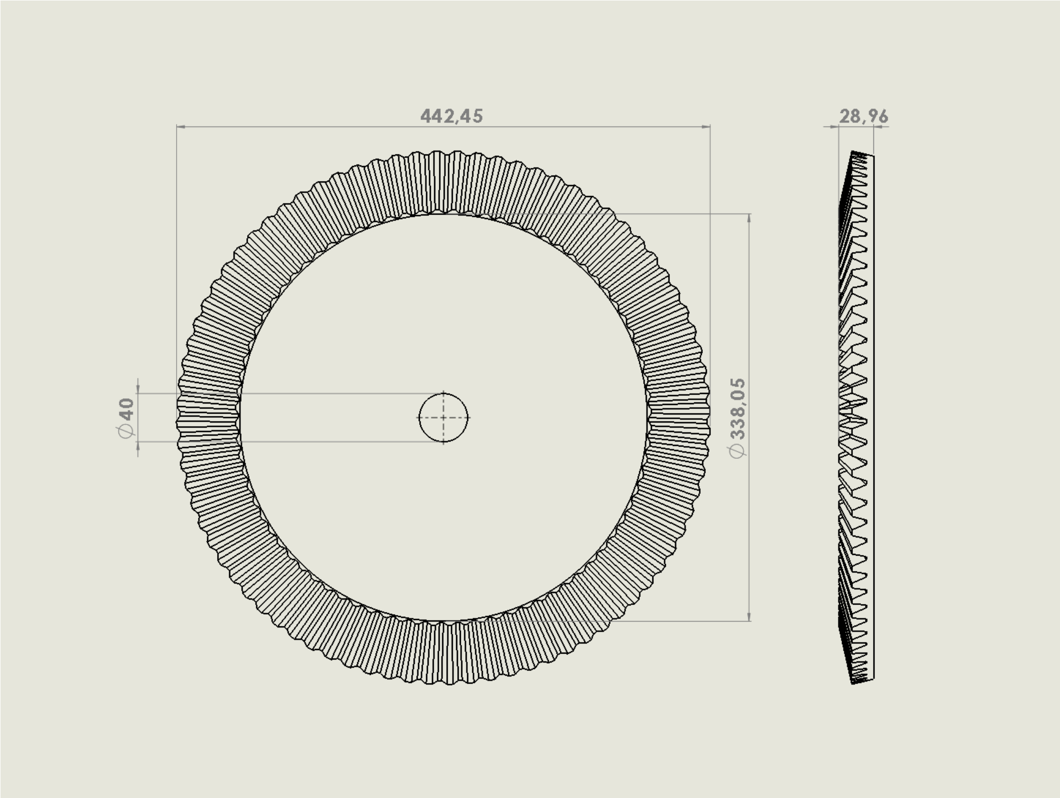


Figure 9: Technical Drawing of the Bevel Gear

In Figure 10, technical drawing of the worm is provided.

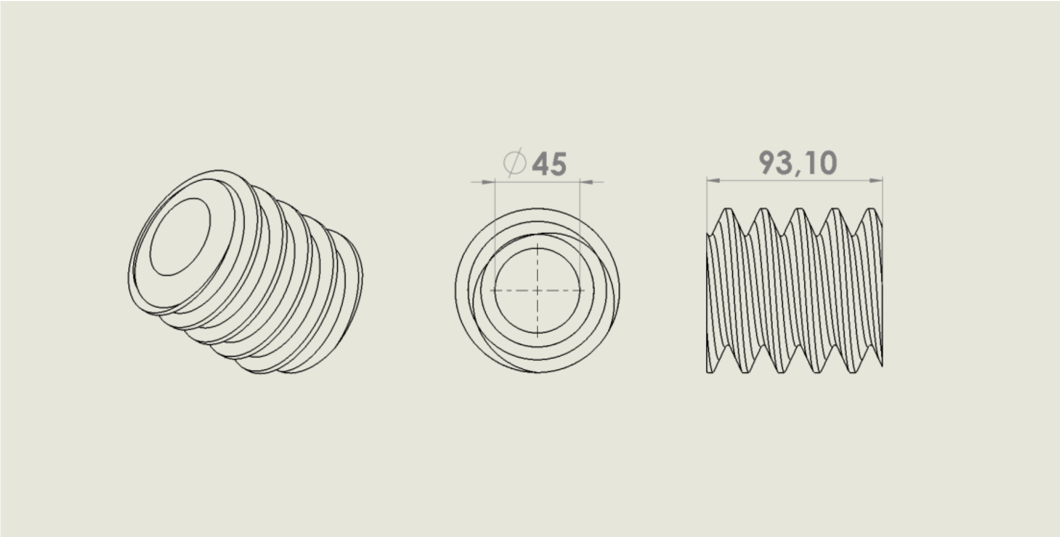


Figure 10: Technical Drawing of the Worm

Design calculations are given in the Appendix. For detailed analysis, please refer to page 11-18.

3. Conclusion

Results obtained from hand calculations were converted into metric system and provided on the right-hand side of Table 1 for the worm gear set and Table 2 for the bevel gear set. In contrast, the results obtained from Kisssoft were provided on the left-hand side of the tables. A comparison was made between results obtained from both methods by showing the percent discrepancy of each variable.

The results obtained for the worm gear set is provided in Table 1. As it is seen, the highest discrepancy is observed in forces and mechanical efficiency, which is basically due to lubrication and heating effect. For the hand calculation part, the effect of oil churning due to overheating was neglected, which is the governing factor for the discrepancy.

Table 1: Results Obtained for the Worm Gear Set

	KissSoft		***Hand Calculations***		***Discrepancy (%)***
Worm Stage	**Worm**	**Gear**	**Worm**	**Gear**	
Number of Teeth	1	37	1	37	0
Module (mm)	6.032	6.032	6.032	6.032	0
Worm diameter (mm)	75	223.2	75	223.2	0
Pressure angle (°)	20	20	20	20	0
Helix angle (°)	85.402	4.598	85.404	4.596	0
Face width (mm)	48.86	48.86	48.86	48.86	0
Center distance (mm)	149.05	149.05	149.05	149.05	0
Material	SAE 8617 Steel	Bronze	SAE 8617 Steel	Bronze	N/A
Axial Force (N)	4547.3	1667.8	5673.4	2074.9	24.7
Radial Force (N)	1667.8	565.9	2074.9	565.6	18.2
Transverse Force (N)	565.9	4547.3	565.6	5673.4	22.0
Sliding velocity (m/s)	14.18	14.18	14.18	14.18	0.0
Friction Force (N)	N/A	N/A	109.2	109.2	N/A
Coefficient of friction	0.016	0.016	0.018	0.018	12.5
Efficiency (%)	64.63	64.63	80.6	80.6	24.7
Dynamic load (kN)	N/A	N/A	N/A	6.73	N/A
Strength capacity (kN)	N/A	N/A	N/A	19.15	N/A
Safety factor for Bending Fatigue	N/A	9.247	N/A	2.85	N/A
Wear Factor (kPa)	N/A	N/A	N/A	689.5	N/A
Wear Capacity (kN)	N/A	N/A	N/A	8.85	N/A
Safety factor for Surface Fatigue	N/A	1.415	N/A	1.32	N/A
Thermal Capacity (kW)	N/A	N/A	1.35	1.35	N/A

The results obtained for the bevel gear set is provided in Table 2. As it is seen, the highest discrepancy is observed in forces and tooth bending stress/strength. The discrepancy between forces stem from the difference between mechanical efficiencies calculated via Kissoft and the hand method. The power transmitted from the input shaft to the consecutive shafts is higher in the hand calculation method, which is basically due to the fact that the heating effect and its impact on lubrication are neglected – leading to forces of higher magnitude in the hand calculation method. Other parameters are at a comparable level.

Table 2: Results Obtained for the Bevel Gear Set

	KissSoft		***Hand Calculations***		***Discrepancy (%)***
Bevel Stage	**Pinion**	**Gear**	**Pinion**	**Gear**	
Number of Teeth	20	80	20	80	0
Module (mm)	5.5	5.5	5.5	5.5	0
Pressure angle (°)	20	20	20	20	0
Pitch cone angle (°)	14.04	75.96	14.04	75.96	0
Face width (mm)	50	50	50	50	0
Material	18CrNiMo7-6 Steel	18CrNiMo7-6 Steel	18CrNiMo7-6 Steel	18CrNiMo7-6 Steel	N/A
Axial Force (kN)	0.93	3.71	1.14	4.57	23.1
Radial Force (kN)	3.71	0.93	4.57	1.14	23.1
Transverse Force (kN)	11.19	11.19	12.93	12.93	15.5
Tooth Bending Stress (MPa)	341.3	400.5	211.2	234.5	39.9
Bending Strength (MPa)	963.7	963.4	545.4	545.4	43.4
Safety factor for Bending Fatigue	2.82	2.41	2.58	2.33	6.1
Surface Stress (MPa)	890.2	890.2	890.8	890.8	0.1
Hardness (HB)	642	642	642	642	0.0
Surface Strength (MPa)	1671.3	1840.1	1778	1853	3.4
Safety factor for Surface Fatigue	1.88	2.07	2	2.1	3.8

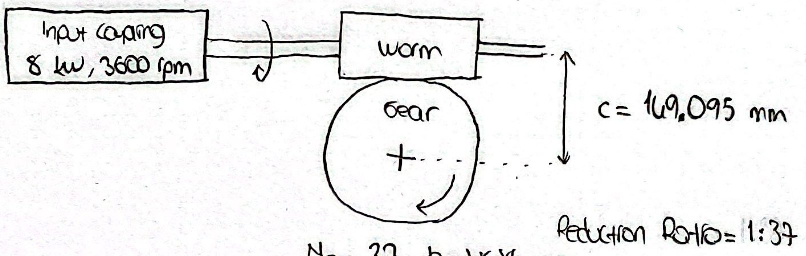
All in all, the gearbox design generated in Kisssoft is validated by the traditional method of hand calculations. It is concluded that the knowledge of computer software is highly important for an engineer to efficiently manage time and effort. Considering the design outputs being reliable and validated, it is very convenient to utilize Kisssoft in this regard for similar applications.

Appendix – Design Calculations

– Worm Gear Design –

– Force Analysis –

Worm: $N_w = 1$, RH, $m = 6.0322$ mm, $\phi_n = 20^\circ$



$N_g = 37$ $b = 48.86$ mm

Material Selection

- Worm: SAE 8617 steel hardened (613 HB)
- Gear: Bronze

Assumptions

- The worm and gear are mounted and aligned to mesh properly at mutually perpendicular axis.
- All the tooth load is transmitted at the pitch point and in the midplane of the gears.
- Gear is hobbed via shaping cutters.
- Bending fatigue strength of gear is taken as Buckingham's proposal of 24 ksi.
- Ambient temperature is below 100 °F (35°C).
- A steady-state temperature of 100°F above ambient temperature is acceptable. (i.e., oil temp. can be 200°F at max)
- Power losses due to bearing and seal friction, and churning of lubricating oil is neglected.

(Converting all design parameters to carry out calculations

$c = 5.87$ inch $p = \pi m = 0.746$ inch $\dot{W} = 10.7282$ hp $b = 1.924$ in)

$d_g = N_g p / \pi$ (Eq. 15.2) $\rightarrow$ $d_g = 8.786$ inch

$d_g + d_w = 2c$ $\rightarrow$ $d_w = 2.954$ inch

$\lambda = \tan^{-1}(L/\pi d_w)$ (Eq. 10.1) $\rightarrow$ $\lambda = 4.596^\circ$

$V_w = \dfrac{\pi d_w n_w}{12}$ $\rightarrow$ $V_w = 2784.08$ ft/min

$F_{wt} = F_{ga} = \dot{W}(33\,000)/V_w$ $\rightarrow$ $F_{wt} = F_{ga} = 127.162$ lb

$V_s = V_w / \cos\lambda$ (Eq. 16.35) $\rightarrow$ $V_s = 2793.06$ ft/min $\rightarrow$ $f = 0.018$ (Fig. 16.23)

$\dfrac{F_{gt}}{F_{wt}} = \dfrac{\cos\phi_n \cos\lambda - f\sin\lambda}{\cos\phi_n \sin\lambda + f\cos\lambda}$ (Eq. 16.31) $\rightarrow$ $\dfrac{F_{gt}}{F_{wt}} = 10.03$ $F_{gt} = 1275.435$ lb $= F_{wa}$

$F_{gr} = F_{wr} = F_{gt} \dfrac{\sin\phi_n}{\cos\phi_n \cos\lambda - f\sin\lambda} = F_{wt} \dfrac{\sin\phi_n}{\cos\phi_n \sin\lambda + f\cos\lambda}$ (Eq. 16.32) $\rightarrow$ $F_{gr} = F_{wr} = 466.456$ lb

$F_n = F_{gr} / \sin\phi_n$ $\rightarrow$ $F_n = 1368.83$ lb $\rightarrow$ Friction Force $= f F_n = 24.549$ lb

$e = \dfrac{\cos\phi_n - f\tan\lambda}{\cos\phi_n + f\cot\lambda}$ (Eq. 16.34) $\rightarrow$ $e = 0.806 = 80.6\ \%$ (total efficiency is lower in reality due to churning of lubricant)

So transmitted power is $\dot{W}_{out} = \dot{W}_{in} e = 8.647$ hp $= 6.448$ kW

If $f \geq \cos\phi_n \tan\lambda$ (Eq. 16.36) $\rightarrow$ $0.018 \ngeq 0.0755$ worm gear-set is not self-locking!

– Bending and Surface Fatigue –

$F_d = F_{gt} K_v$ (Eq. 16.39) where $K_v = \frac{1200 + V_g}{1200}$ (Figure 15.24)

$V_g = \tan\lambda \, V_w$ (Eq. 16.33) → $V_g = 223.806$ ft/min so $K_v = 1.1865$

$F_d = 1513.9$ lb

$F_s = S_n b p y$ (Eq. 16.40) where $S_n = 24$ ksi (proposed by Buckingham, for bronze worm-gears)

$y = 0.125$ (Table 16.2)

→ $F_s = 4305.9$ lb

$F_w = d_g b K_w$ (Eq. 16.41) where $K_w = 100$ lb/in² (Table 16.3)

→ $F_w = 1990.43$ lb

Hence, both F_s & $F_w \geq F_d$

$SF_{bending} = F_s / F_d = 2.85$

$SF_{surface\ fat.} = F_w / F_d = 1.32$

– Thermal Capacity –

$H = CA(t_o - t_a)$ (Eq. 13.13) where $A = 0.3 c^{1.7}$ (Eq. 16.42) $= 6.099$ ft²

$C = 97.14$ (Figure 16.25) (Interpolation)

$H = \dot{W}_{in} - \dot{W}_{out} = 2.0812$ hp $= 67740$ ft·lb/min

$t_a = 100°F$

→ $t_o = 214.7°F$ $t_o > 200°F$

t_o is above acceptable level, which is 200°F. This will lead to churning of lubrication oil. Special cooling provisions are required. Active or passive cooling systems (e.g. fans, fins) would be desired to keep lubrication below 200°F.

Thermal capacity of the current system without cooling application:

(Take $t_o = 200°F$ and $t_a = 100°F$) $H = 1.814$ hp

— Bevel Gear Design —
— Force Analysis —

Material selection

- 18CrNiMo7-6 steel (hardened) for both gear
  622 HB, 1200 MPa, fine-grained, commercially polished

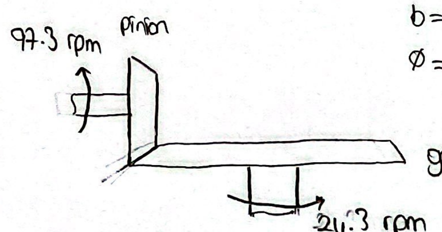
Assumptions

- All the tooth loads are transmitted at the pitch point midway along the tooth face.
- Friction losses can be neglected.
- System is expected to operate 10 000 hours.
- Gears are manufactured by hobbing.
- Reliability is 95 %
- $k_t = 1$, since to < 160°F.
- uniform power source and uniform load. ($k_o = 1$)

$\dot{W}_p = \dot{W}_{out,worm} = 8.6247$ hp $N_p = 20$, $d_p = 110$ mm

$b = 50$ mm

$\phi = 20°$



Reduction Ratio = 1:4

$N_g = 80$, $d_g = 440$ mm

Converting units into imperial system
$d_p = 4.33071$ in $d_g = 17.3228$ in $b = 1.9685$ in

$P = \dfrac{N}{d}$ (Eq. 15.3) → $P = 4.6182$ teeth/inch

$\gamma_g = \tan^{-1}\left(\dfrac{N_g}{N_p}\right)$ (Eq. 16.16) → $\gamma_g = 75.964°$

$\gamma_p = 90 - \gamma_g = 14.036°$

$d_{av} = d - b\sin\gamma$ (Eq. 16.18) → $d_{p,av} = 3.8533$ in $d_{g,av} = 15.413$ in

$V_{av} = \pi d_{av} n / 12$ (Eq. 16.19a) → $V_{av} = 98.155$ ft/min

$F_t = 33\,000\,\dot{W} / V_{av}$ (Eq. 16.20a) → $F_t = 2907.147$ lb

$F_a = F_t \tan\phi \sin\gamma$ (Eq. 16.22) → $F_a = 256.626$ lb (for the pinion) = F_r (for the gear)

$F_r = F_t \tan\phi \cos\gamma$ (Eq. 16.23) → $F_r = 1026.523$ lb (for the pinion) = F_a (for the gear)

– Tooth Bending –

Pinion :

$N = (10\,000\ h)\ n_p\ 60 = 5.84 \times 10^7$ cycle (since $\geq 10^6$, assumed as infinite life cycle)

$S_n = S_n' C_L C_G C_s k_r k_t k_{ms}$ (Eq. 15.18) where $S_n' = 0.5 S_u = 600$ MPa $= 87.0226$ ksi (Table 8.1)

$C_L = 1$ (Table 8.1) $\quad k_r = 0.868$ (Tab 15.3)

$C_G = 0.85$ (for $P < 5$) $\quad k_t = 1$

$C_s = 0.88$ (Fig. 8.13) $\quad k_{ms} = 1.4$ (for input and output gears)

$\rightarrow S_n = 79.1$ ksi

$\sigma = \frac{F_t P}{bJ} K_v K_o K_m$ (Eq. 15.17) where $K_v = 1.082$ (Fig 15.24) $\quad K_m = 1.1$ (Tab. 16.1)

$K_o = 1$ (Tab 15.1) $\quad J = 0.265$ (Fig. 16.13)

$\rightarrow \sigma = 30.63$ ksi

$SF_{bending,\ pinion} = \frac{S_n}{\sigma} = 2.58$

Gear :

$N = (10\,000\ h)\ n_g\ 60 = 1.46 \times 10^7$ cycle ($\geq 10^6 \rightarrow$ infinite life cycle)

$S_n = 79.1$ ksi

$K_v = 1.02 \quad K_o = 1 \quad K_m = 1.1 \quad J = 0.225$

$\rightarrow \sigma = 34.01$ ksi

$SF_{bending,\ gear} = \frac{S_n}{\sigma} = 2.33$

-Surface Fatigue-

Pinion

$S_H = S_{fe} C_{Li} C_R$ (Eq. 15.25) where $S_{fe} = 246.8$ ksi (Tab. 15.5)

$C_{Li} = 0.95$ (Fig. 15.27)

$C_R = 1.1$ (Tab. 15.6)

$\rightarrow S_H = 257.91$ ksi

$\sigma_H = C_p \sqrt{\frac{F_t}{b d_p I} K_v K_o K_m}$ (Eq. 15.24) where $C_p = 2300 \sqrt{psi}$ (Tab. 15.4a)

$I = \frac{\sin\phi \cos\phi}{2} \frac{R}{R+1}$ (Eq. 15.23) $\rightarrow I = 0.1286$

$R \rightarrow d_g/d_p$

$\rightarrow \sigma_H = 129.2$ ksi

$SF_{\text{surface fatigue, pinion}} = \frac{S_H}{\sigma_H} = 2$

Gear

$C_{Li} = 0.99$ (Fig. 15.27) [Other parameters are same with pinion]

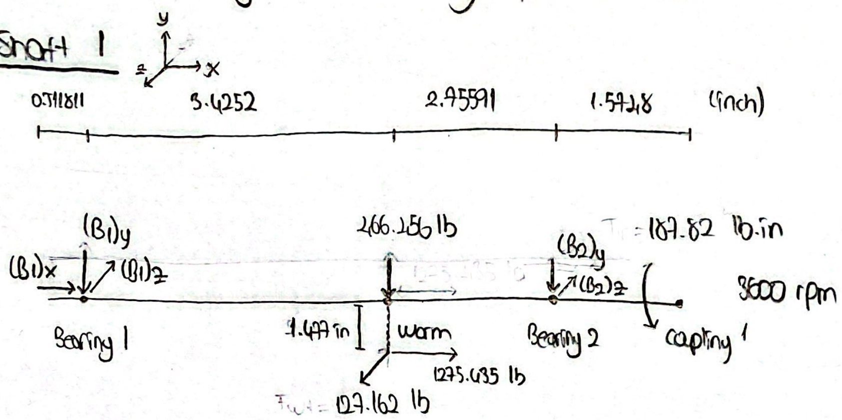
$\rightarrow S_H = 268.76$ ksi

$\sigma_H = 129.2$ ksi

$SF_{\text{surface fatigue, gear}} = \frac{S_H}{\sigma_H} = 2.1$

– Shaft Force Analysis and Bearing Life Calculations –

Shaft 1



$\Sigma F_x = 0 \rightarrow (B_1)_x = 1275.435$ lb

$\Sigma (M_z)_{B1} = 0 \rightarrow (B_2)_y = 246.288$ lb

$\Sigma F_y = 0 \rightarrow (B_1)_y = 512.744$ lb

$\Sigma (M_y)_{B1} = 0 \rightarrow (B_2)_z = 70.465$ lb

$\Sigma F_z = 0 \rightarrow (B_1)_z = 56.697$ lb

$\Sigma M_x = 0 \rightarrow 187.82 - 127.162(1.477) \approx 0$ (System is in equilibrium!)

$(B_1)_R = 515.87$ lb $\quad (B_1)_a = 1275.435$ lb $\quad (B_2)_R = 84.31$ lb $\quad (B_2)_a = 0$ lb

Bearing Selection

- Bearing 1: SKF QJ 309 PHAS Angular Ball Bearing
- Bearing 2: SKF 2209 ATN9 → Radial Ball Bearing

Assumptions

- Forces are applied at the middle of components
- L_{10} is taken as 10^6 (ISO standard)
  *Results are comparable with SKF design life provided in the SKF calculator.
- Reliability is %95
- Application is gearing. $K_a = 1.1$
- Axial load is accomodated only by bearing 1.

Material Selection

- Shaft 1: AISI 1040 steel, 45 mm 163 HB, 565 MPa(S_u), 310 MPa(S_y)

Bearing 1

$K_r = 0.65$ (Fig. 14.13) $L_{10} = 10^6$ (ISO standard) $C = 22481$ lb (SKF catalogue) $K_a = 1.1$ (Tab. 14.3)

$F_e = 1320.3$ lb (Eq. 14.4²)

$L = K_r L_{10} \left(\frac{C}{F_e K_a}\right)^3$ (Eq. 14.5a) $\longrightarrow L = 2.41 \times 10^9$ cycle $= 11161$ h

$L > 10^4$ h design is valid!

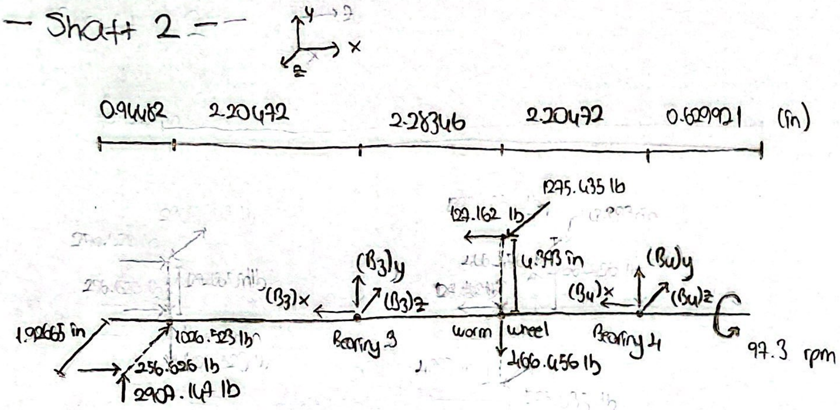
Bearing 2

$C = 8768$ lb (SKF Catalogue) $F_e = 84.31$ lb (Eq. 14.3²) [Other parameters are same with Bearing 1]

$\rightarrow L = 5.5 \times 10^{11}$ cycle $= 2.5 \times 10^6$ h

$L > 10^4$ h design is valid!

– Shaft 2 –



Bearing Selection

- Bearing 1 & 2 : SKF 312 Deep Groove Ball Bearing

Assumptions

- Forces are applied at the mid-width of each component and represented accordingly.
- Axial force is distributed equally between two bearings.
- L_{10} is taken as 10^6. (ISO standard)
- Reliability is 95%.
- Application is gearing. $K_a = 1.1$

Material Selection:

- Shaft 2 : AISI 1025 steel, 60 mm 163 HB, 565 MPa (S_u), 310 MPa (S_y)

$\Sigma F_x = 0 \rightarrow (B_3)_x = (B_4)_x = 64.732$ lb

$\Sigma (M_z)_{B_3} = 0 \rightarrow (B_4)_y = 312.854$ lb

$\Sigma F_y = 0 \rightarrow (B_3)_y = 353.602$ lb

$\Sigma (M_y)_{B_3} = 0 \rightarrow (B_4)_z = 1043$ lb

$\Sigma F_z = 0 \rightarrow (B_3)_z = 794.09$ lb

$\Sigma (M_x) = 0 \rightarrow (2909.14)(1.92665) - (1275.435)(4.393) \approx 0$ (System is in equilibrium!)

$(B_3)_R = 869.26$ lb $(B_3)_a = 64.732$ $(B_4)_R = 1069.09$ lb $(B_4)_a = 67.732$ lb

Bearing 3

$K_r = 0.65$ (Fig. 14.13) $L_{10} = 10^6$ (ISO standard) $C = 20\,525$ lb (SKF Catalogue) $K_a = 1.1$ (Tab 14.3)

$F_e = 869.26$ lb (Eq. 14.3²)

$L = K_r L_{10} \left(\frac{C}{F_e K_a}\right)^3$ (Eq. 14.5a) $\rightarrow L = 6.43 \times 10^9$ cycle $= 1.1 \times 10^6$ h

$L > 10^4$ h design is valid!

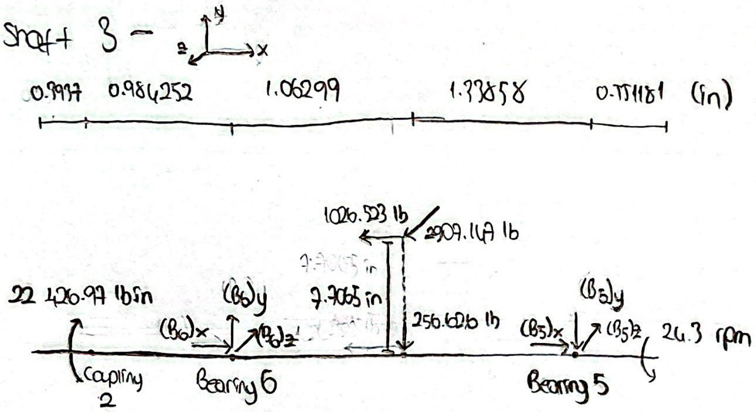
Bearing 4

$F_e = 1069.09$ lb (Eq. 14.3²) [Other parameters are same with Bearing 3]

$\rightarrow L = 3.66 \times 10^9$ cycle $= 6.3 \times 10^5$ h

$L > 10^4$ h design is valid!

– Shaft 3 –



$\Sigma F_x = 0 \rightarrow (B_5)_x = (B_6)_x = 513.26$ lb

$\Sigma (M_z)_{B6} = 0 \rightarrow (B_5)_y = 3180.46$ lb

$\Sigma F_y = 0 \rightarrow (B_6)_y = 3437.09$ lb

$\Sigma (M_y)_{B6} = 0 \rightarrow (B_5)_z = 1286.77$ lb

$\Sigma F_z = 0 \rightarrow (B_6)_z = 1620.38$ lb

$\Sigma (M_x) = 0 \rightarrow (2909.147)(7.7065) - 22426.97 \approx 0$ (system is in equilibrium!)

$(B_5)_R = 3430.9$ lb $(B_5)_a = 513.26$ lb $(B_6)_R = 3800$ lb $(B_6)_a = 513.26$ lb

Bearing 5

$K_r = 0.65$ (Fig. 14.13) $L_{10} = 10^6$ (ISO std.) $C = 10274$ (SKF catalogue) $K_a = 1.1$ (Tab. 14.3)

$Fe = 3430.9$ lb

$$L = K_r L_{10} \left(\frac{C}{Fe\, K_A}\right)^{3.3} \text{ (Eq. 14.5a)} \rightarrow L = 1.77 \times 10^7 \text{ cycle} = 12\,146 \text{ h}$$

$L > 10^4$ h design is valid!

Bearing 6

$Fe = 3800$ lb [Other parameters are same with Bearing 5]

$\rightarrow L = 1.41 \times 10^7$ cycle $= 9670$ h

$L \approx 10^4$ h design is valid!

Bearing Selection:

- Bearing 5 & 6: SKF 308-2NR Deep Groove Ball Bearing

Assumptions:

- Forces are applied at the mid-width of each component, and represented accordingly.
- Axial force is accomodated equally between two bearings.
- Reliability is 95 %.
- Application is gearing. $K_a = 1.1$

Material Selection:

- Shaft 3: AISI 1045 steel, 40 mm 163 HB, 565 MPa (S_u), 310 MPa (S_y)