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Critical Design Review

Marine Gear Box

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Summary

The marine gearbox design project was to design a marine gearbox of high quality for inshore fishing boats. The gearbox should be able to reduce the high input speed to a required low output speed. The gearbox is a transmission device capable of changing the angular speed when required.

The aim of the project is to make the output shaft, which is connected to the propeller to rotate at 1300 rpm with tolerance of ±5%. An electronic limiter is used to ensure output speed doesn't exceed 1300 rpm. This produces an output torque of 110.18 Nm. The engine is a twin cylinder diesel which delivers 20.8kW of power at 3600rpm. The maximum torque for the input is developed at 1700rpm and is calculated to be 55.17 Nm. Thus, the gear ratio is found to be 2, which means that the minimum input shaft speed is 2600rpm.

Another requirement was to incorporate a method of disconnecting the engine from the propeller. A cone clutch was designed to achieve this requirement.

To ensure the smooth functioning of the marine gearbox, an oil pump was included in the design. The oil pump is powered by using a gear connected to the output gear. The required speed of the oil pump is 3900 rpm. The oil pump is continuously run to maintain a certain amount of oil in the gearbox. The gearbox with all the parts mustn't exceed 35kg in weight.

Most of the parts in the gearbox assembly including the casing were made using steel since it is corrosion resistant and strong. Some parts were made using material other than steel and is explained later in material selection. Ball bearings and v seals were used to prevent the misalignment of shafts.

The gearbox needs to have a long-life span and failure rate needs to be very low. Also, it should be designed in a way to produce at least 10000 units per year. To meet this requirement, special measures were taken in the design of the gearbox and is explained later in DFMA. The design efficiency of the design is found to be 52%.

Concept Definition

The designed marine gear box uses the power transmitted from the twin cylinder diesel engine to useful output on the propeller shaft. The engine is connected to the input shaft which is attached to the input gear of 21 teeth and spins at 2600rpm. This input gear is linked directly to the output gear which must have 1300rpm. This is done by using reduction gears. Reduction gears are used to increase the torque and decrease the angular speed between of the output shafts. Hence, the gear ratio of 0.5 to increase the torque at the output. Which computes to 42 teeth on the output gear.

An oil pump is included in the gearbox to ensure smooth function by lubricating the system. A gear connected to the oil pump driving gear is also linked at the bottom of the output gear. The oil pump gear is required to drive an oil pump at an angular speed of 3900rpm. So, a gear ratio of 3 is used giving 14 teeth for the oil pump gear.

The output shaft is integrated with a cone clutch which is operated hydraulically, this is so that the propeller only runs when the engine is on and clutch is engaged. It is also to ensure that there is constant oil flow when the engine is on, regardless if the clutch is engaged or disengaged. The outer core of the clutch is connected to a hydraulic actuator, the helmsman will be able to control the engagement and the disengagement of the clutch by means of a leaver or a pedal. When the clutch is engaged the actuating spring pushes the outer core towards the inner core so that the friction lining is in contact with the outer core. The strong friction between this causes the propeller shaft to rotate at the same speed as the inner core.

Bearings are used on either side of the output shaft and on one side of the input shaft. Bearings are used to enable rotation while reducing friction between surface of bearing and the shaft. A ball bearing is used on one side output shaft and roller bearing is used on the other this is to handle stress on the shaft and for stability of the gearbox.

The casing of the gearbox has been designed in a way that it is easy to assemble. The cover of the casing can be removed for the ease of maintenance. It also provides spacing for attachment of bearings and seals. It incorporates enough space around the gears hence, there won't be any overheating of mechanical parts.

Compliance Statement

NO	Requirements	Outcome
1	A gearbox is required to deliver 15 kW at the limited speed.	As shown in the calculations, max power is delivered at a speed of 2600 rpm.
2	The gearbox shall be mounted on the engine bedplate with 4 M10 bolts, spacing to be proposed, and connected to the engine via a spline drive and flexible coupling.	The casing of gearbox can be mounted on the bedplate using the four M10 bolts, where the casing has four M10 holes. See general assembly drawing. There is spline drives in the output and input shafts which can be used to connect the engine for flexible coupling
3	The total weight of the gearbox (less fluids) shall not exceed 35 kg	The mass of the gearbox is found to be 29.4kg.
4	The design should be consistent annual production rate of 10000 units.	Selection of material and manufacturing plan was chosen to have an acceptable efficiency and precision
5	A means of disconnecting the engine and gearing from the propeller shaft by means of an integrated hydraulically operated clutch for starting or testing shall be incorporated.	A cone clutch is used for engaging and disengaging of the engine and gearing from the propeller shaft
6	A small gear pump is required to provide the oil at the required pressure. This needs to run whenever the engine is powered, i.e. driven upstream of the clutch.	An oil pump is used that is connected to the driven gear and runs at a speed of 3900rpm. See general assembly drawing.
7	The gears shall be designed for a life of 10^9 cycles and a failure rate of not greater than 1 in 1000	To ensure a great life span and reliability, the material of each part has been selected based on proper calculations.
8	The gearbox shall have means of filling, draining and monitoring the oil level.	Oil feed union is used for filling oil and the oil pump sucks the oil from the bottom. An oil level indicator is used to check the oil level

9	Provision for oil return from the propeller and pressure regulating valve shall be provided using an M10 connection.	The oil feed union is mounted on top of the casing and it is connected to the oil pump to circulate the oil
10	The gearbox shall remain oil tight at all conditions; all joints shall have suitable sealing.	A gasket is used to seal the casing and the casing lid. A union washer is used to seal the oil union hole.
11	Although not directly in contact with sea water the atmosphere will be salt laden and consideration shall be given for corrosion protection.	The material used for the casing is 201 Annealed Stainless Steel which is corrosive resistant.
12	The gearbox is connected to the engine and propeller transmission via splined shafts which need to be part of the design, Fig 6.	A splined shaft is used at the input and output shafts, so that it can be connected to the engine and propeller

Engineering Calculations

Calculations are required to justify the selection of bearings, seals and fasteners for the chosen design. For reference the given data sheets and Shigley's Mechanical Engineering Design book were used when required. The calculations in the preliminary design review were referenced to complete the stress analysis on the input and output gears and for shaft analysis.

Gear Calculations

W_T = transmitted loads

$$\text{Torque} = T = W_T \cdot \frac{d}{2}$$

$$\text{Power} = T \cdot \omega = W_T \cdot V$$

$$T = \frac{P}{\omega} = W_T \cdot \frac{d}{2}$$

$$\therefore W_T = \frac{2P}{\omega d} \Rightarrow \omega = \frac{2\pi n}{60}$$ and d should be in meters.

$$\therefore W_T = \frac{2P}{2\pi n d} \times \frac{60}{10^{-3}} = \frac{60 \times 10^{-3} P}{\pi n d}$$

- For input gear;

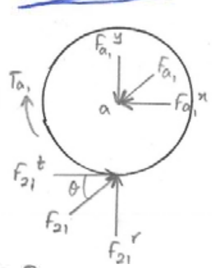
$$W_{T_1} = \frac{60 \times 10^{-3} P}{\pi n d} = \frac{60 \times 10^3 \times 15}{\pi \times 2600 \times 63} = 1.75\,kN$$

- For output gear;

$$W_{T_2} = \frac{60 \times 10^3 P}{\pi n d} = \frac{60 \times 10^3 \times 15}{\pi \times 1300 \times 126} = 1.75\,kN$$

$$\phi = 20^\circ$$

Input Gear



$$F_{21}^b = W_T = 1.75\,kN$$

$$F_{21}^r = F_{21}^t \tan\phi = 0.63\,kN$$

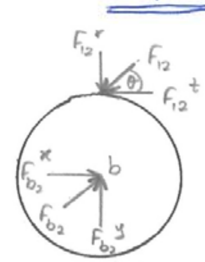
$$\sum F_x = 0$$

$$F_{21}^t = F_{a1}^x = 1.75\,kN$$

$$\sum F_y = 0$$

$$F_{21}^r = F_{a1}^y = 0.63\,kN$$

Output Gear



$$F_{12}^t = W_T = 1.75\,kN$$

$$F_{12}^r = 1.75 \tan 20 = 0.63\,kN$$

$$\sum F_x = 0$$

$$F_{12}^t = F_{b2}^x = 1.75\,KN$$

$$\sum F_y = 0$$

$$F_{12}^r = F_{b2}^y = 0.63\,kN$$

For both input and output gears, reserve factors need to be calculated. For this the forces determined above are used in calculations of the bending and contact stress.

Gear Stress Analysis

- Gear bending stress:

$$\sigma = W_T K_o K_s K_v \cdot \frac{1}{F_m} \cdot \frac{K_H K_B}{Y_J}$$

- Overload Factor (K_o):-

$$K_v = \left(\frac{A+\sqrt{200V}}{A}\right)^B$$

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

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

- $Q_v = 7$ (commercial quality)

$$B = \frac{(12-7)^{2/3}}{4} = 0.731$$

$$A = 50 + 56(1 - 0.731) = 65.1$$

For gears; $V = \frac{d}{2}$, $\omega = \frac{63}{2} \cdot \frac{2\pi \times 10^{-3} \times 2600}{60} = 8.58$

K_v is same for both gears,

$$\therefore K_v = \left(\frac{A+\sqrt{200V}}{A}\right)^B = \left(\frac{65.1+\sqrt{200 \times 8.58}}{65.1}\right)^{0.731}$$

$$k_v = 1.43$$

- Size factor:-

$$K_s = 1$$

- Rim thickness Factor (K_H):-

$$k_H = 1 + C_{mc}(C_{pF} C_{pm} + C_{ma} C_e)$$

→ $C_{mc} = 1$ (uncrowned teeth)

→ $C_e = 1$ (shaft and gear as one part)

→ $C_{ma} = A + BF + CF^2$ (For commercial gears)

$$= 0.127 + 0.622(0.022) + (-0.144)(0.022)^2$$
$$= 0.141$$

→ C_{PF}, as $F < 0.0254$ (m)

$$C_{PF} = \frac{F}{10d} - 0.025$$

However, For both gears $b/10d < 0.05$

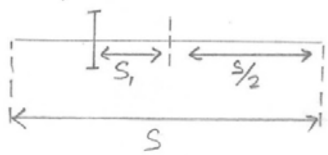
$\therefore$ we set $b/10d = 0.05$

Thus, for both gears

$$C_{PF} = 0.05 - 0.025 = \underline{\underline{0.025}}$$

→ C_{Pm}

Input gear:



$S = 176\,mm$

$S/2 = 88\,mm$

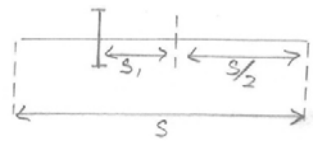
$S_1 = 53\,mm$

$$\frac{S_1}{S} = \frac{53}{176} = 0.301$$

as, $\frac{S_1}{S} > 0.175$

$\therefore C_{Pm} = 1.1$

Output gear:



$S = 176\,mm$

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$S_1 = 53\,mm$

$\therefore \frac{S_1}{S} = 0.301$

as $\frac{S_1}{S} > 0.175$

$\therefore C_{Pm} = 1.1$

$$K_H = 1 + 1(0.025 \times 1.1 + 0.141 \times 1)$$

$$= 1.169$$

Geometry factor (Y_J):- using graph

For input gear → $Y_J = 0.337$

For output gear → $Y_J = 0.385$

• Bending stress (σ):-

→ Input gear $\sigma = 1750 \times 1 \times 1 \times 1.43 \times \frac{1}{3 \times 22} \times \frac{1.169 \times 1}{0.337}$

$\sigma = 131.4\,MPa$

→ Output gear $\sigma = 1750 \times 1 \times 1 \times 1.43 \times \frac{1}{3 \times 22} \times \frac{1.169 \times 1}{0.385}$

$= 115\,MPa$

Bending stress for input and output gears are determined to be 131.4MPa and 115MPa respectively. Subsequently, the allowable bending stress for both gears are calculated. Afterwards, the corresponding values are used to obtain the reserve factors for both gears respectively. However, the bending stress cannot exceed the allowable bending stress as the reserve factor can't be greater than 1. If it is greater than 1, the gears need to be redesigned and these corresponding values are used to obtain the reserve factors for both gears respectively.

Allowable Bending Stress

$$\sigma_{all} = \frac{\sigma_{FP}}{S_F} \cdot \frac{Y_N}{Y_\theta Y_Z}$$

Allowable bending stress (σ_{FP}):-

→ For Grade 2, nitrided through hardened steel

→ Input gear

$\sigma_{FP} = 0.749\,H_B + 110\,MPa$

$\sigma_{FP} = 0.749(400) + 110$

$= 409.6\,MPa$

→ Output gear

$\sigma_{FP} = 0.749\,H_B + 110$

$= 0.749(330) + 110$

$= 357.17\,MPa$

• Life factor (Y_N):-

$N = 10^9$

$Y_N = 1.3558(10^9)^{-0.0178} = 0.938$

• Temperature factor (Y_θ):-

$Y_\theta = 1$

• Reliability factor (Y_Z):- $R = 0.999$

$Y_Z = 1.25$

- Allowable Bending stress (σ_{all}):-

Input gear:

$$\sigma_{all} = \frac{409.6}{1} \times \frac{0.938}{1 \times 1.25}$$

$$= 307.4 \text{ MPa}$$

$\therefore$ Reserve Factor $= \dfrac{307.4}{131.5}$

$$= 2.33$$

Output gear:

$$\sigma_{all} = \frac{357.17}{1} \times \frac{0.938}{1 \times 1.25}$$

$$= 268 \text{ MPa}$$

$\therefore$ Reserve factor $= \dfrac{268}{115.13}$

$$= 2.33$$

As shown above, the allowable bending stress for both gears are larger than the actual bending stress calculated earlier. This in turn results in a reserve factor more than 1. This proves that AISI 4340 is a suitable material for the gears.

Gear Contact Stress:

$$\sigma_c = Z_e \sqrt{W_T K_o K_v' K_s \cdot \frac{K_H}{F_{dp}} \cdot \frac{Z_R}{Z_I}}$$

→ Elastic coefficient (Z_e):- (m = gear ratio, $\nu = 0.3$, $E = 200$ GPa)

$$Z_e = \left\{ 1 \Big/ \left[\pi \left(\frac{1-\nu_P^2}{E_P} + \frac{1-\nu_G^2}{E_G} \right) \right] \right\}^{0.5}$$

$$= \left\{ 1 \Big/ \pi \left(\frac{1-0.09}{200 \times 10^3} + \frac{1-0.09}{200 \times 10^3} \right) \right\}^{0.5}$$

$$= 187$$

→ Geometry factor (Z_I):-

$$Z_I = \frac{\cos\phi \sin\phi}{2} \times \frac{m}{m+1} = \frac{\cos 20 \sin 20}{2} \times \frac{2}{2+1}$$

$$= 0.107$$

→ Surface condition factor (Z_R);-

$$Z_R = 1$$

• Contact Stress (σ_c):-

→ Input gear $\sigma_c = 187\sqrt{1750 \times 1 \times 1.43 \times 1 \times \frac{1.169}{22 \times 63} \times \frac{1}{0.107}}$

$= 830.5\,MPa$

→ Output gear - $\sigma_c = 187\sqrt{1750 \times 1 \times 1.43 \times 1 \times \frac{1.169}{22 \times 126} \times \frac{1}{0.107}}$

$= 587.3\,MPa$

The above calculations for contact stress for gears 1 and 2 show that the stress is 830.5MPa and 587.3MPa respectively. Like the bending stress, the allowable contact stress is calculated. The reserve factors should be greater than 1.

- Allowable Contact Stress

$$\sigma_{c,all} = \frac{\sigma_{HP}}{S_H} \times \frac{Z_N Z_W}{Y_\theta Y_Z}$$

· Life factor (Z_N):-

$N = 10^9$

$Z_N = 1.4488\,(10^9)^{-0.023} = 0.8995$

· Hardness Ratio Factor (Z_W):-

$Z_W = 1 + A\,(m_G - 1)$

$A' = 8.98 \times 10^{-3}\left(\frac{H_{BP}}{H_{BG}}\right) - 8.29 \times 10^{-3}$

$= 8.98 \times 10^{-3}\left(\frac{400}{330}\right) - 8.29 \times 10^{-3} = 0.0259$

$Z_W = 1 + 0.00259\,(2-1) = 1.0026$

Allowable contact stress (σ_{HP}):-

Input gear

$$\sigma_{HP} = 2.41(400) + 237$$

$$= 1201 \text{ MPa}$$

Output gear

$$\sigma_{HP} = 2.41(330) + 237$$

$$= 1032 \text{ MPa}$$

Allowable Contact Stress ($\sigma_{c,all}$):-

Input gear:

$$\sigma_{c,all} = \frac{1201}{1} \times \frac{0.8995 \times 1.0026}{1 \times 1.25}$$

$$= 866.5 \text{ MPa}$$

$$\text{Reserve factor} = \frac{866.5}{830.5}$$

$$= 1.04$$

Output gear:

$$\sigma_{c,all} = \frac{1032}{1} \times \frac{0.8995 \times 1.0026}{1 \times 1.25}$$

$$= 744.6 \text{ MPa}$$

$$\text{Reserve factor} = \frac{744.6}{587.3}$$

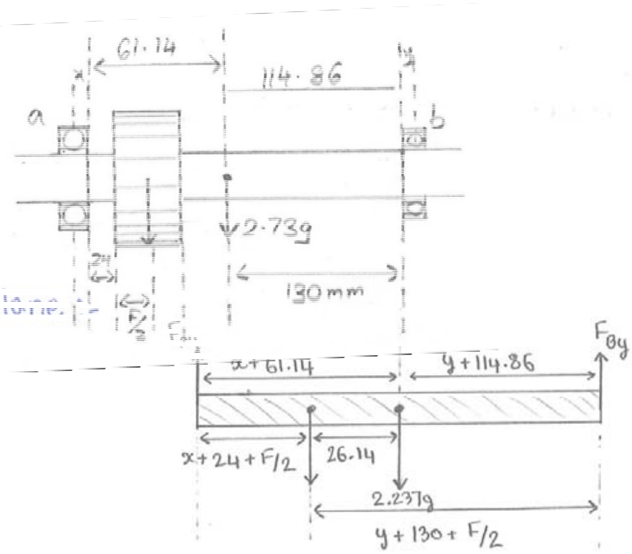
$$= 1.27$$

The allowable contact stress for both gears are larger than the actual contact stress calculated earlier. This is appropriate as the reserve factor must be greater than 1. From calculations it is equal to 1.04 and 1.27 for input and output gears respectively. This proves the material selection.

Bearing Calculations

Input gear :-

- Vertical Plane :-



About A:

$$0.63k(x+11+24)+2.237g(x+61.4)=F_{By}(x+y+130+22+24)$$

$$630(42.5)+2.237\times9.81(68.64)=F_{By}(191)$$

$$\underline{\underline{F_{By}=148.07\text{ N}}}$$

About B:

$$2.237g(y+114.86)+0.63k(y+130+11)=F_{Ay}(x+y+22+24+130)$$

$$2.237\times9.81(122.36)+630(148.5)=F_{Ay}(191)$$

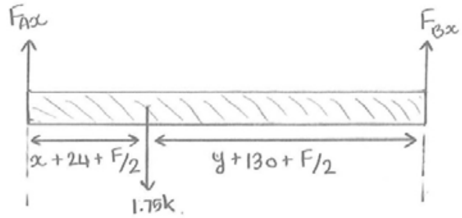
$$\underline{\underline{F_{Ay}=503.9\text{ N}}}$$

Resultant Radial Force on Bearing A

$$F_A=\sqrt{F_{Ax}^2+F_{Ay}^2}$$

$$\underline{\underline{F_A=1451\text{ N}}}$$

· Horizontal Plane :-



About A :

$$1.75k(x+24+11) = F_{Bx}(x+y+176)$$

$$1750(42.5) = F_{Bx}(191)$$

$$F_{Bx} = 389.4N$$

About B :

$$1750(y+130+11) = F_{Ax}(x+y+176)$$

$$1750(48.5) = F_{Ax}(191)$$

$$F_{Ax} = 1360.6N$$

· Resultant Radial Force on Bearing B :

$$F_B = \sqrt{F_{Bx}^2 + F_{By}^2}$$

$$F_B = 416.6N$$

Dynamic load Rating (C)

$$C = A_F P\left(\frac{L}{a_1 L_{10}}\right)^{1/k}$$

P = dynamic bearing load (F_A/F_B)

A_F - Load application factor

$a_1 = 0.21$ (as $R_t = 0.999$)

L - Life time

L_{10} - 10^6 rev

k - exponent of life equation.

For A;

For ball bearing; $k = 3$

$$C = 1.2(1451)\left(\frac{10^9}{0.21\times10^6}\right) = 29.3\ kN$$

But for ball bearing; $25\times52\times15$

$C_{max} = 14.8\ kN$ ∴ Not applicable

For Roller Bearing; $K = \frac{10}{3}$

$$C = 1.2(1451)\left(\frac{10^9}{0.21\times10^6}\right)^{3/10} = 22.1\ kN$$

But for roller bearings; $25\times52\times15$

$C_{max} = 32.5\ kN$ ∴ Hence cylindrical roller bearing is used.

For B;

For ball bearings; $k = 3$

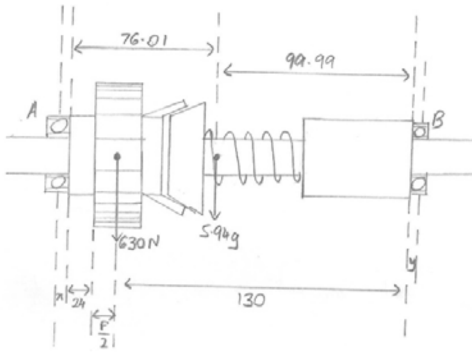
$$C = 1.2(416.6)\left(\frac{10^9}{0.21\times10^6}\right)^{1/3} = 8.41\ kN$$

But for ball bearings; $25\times52\times15$

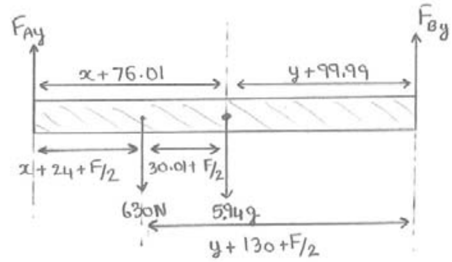
$C_{max} = 14.8\ kN$ ∴ Hence, ball bearing is used

As shown in calculations, A is a cylindrical roller bearing while B is a ball bearing for the input.

Output gear :-



Vertical Pone.



About A:

$$630(x+24+11) + 5.94g(x+76.01) = F_{By}(x+y+22+24+130)$$

$$630(42.5) + 5.94(9.81 \times 83.51) = F_{By}(191)$$

$$\underline{\underline{F_{By} = 165.7\,N}}$$

About B:

$$F_{Ay}(x+y+176) = 5.94g(y+99.99) + 630(y+130+11)$$

$$F_{Ay}(191) = 5.94 \times 9.81(107.49) + 630(148.5)$$

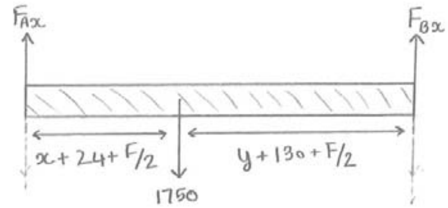
$$\underline{\underline{F_{Ay} = 522.6\,N}}$$

Resulting Radial Force on Bearing A:

$$F_A = \sqrt{F_{Ax}^2 + F_{Bx}^2}$$

$$\underline{\underline{F_A = 1457.5N}}$$

- Horizontal Plane :-



About A :

$$1750\,(x + 24 + 11) = F_{Bx}\,(x + y + 176)$$

$$F_{Bx}\,(191) = 1750\,(42.5)$$

$$F_{Bx} = 389.4\,N$$

About B :

$$1750\,(y + 11 + 130) = F_{Ax}\,(x + y + 176)$$

$$F_{Ax}\,(191) = 1750\,(148.5)$$

$$F_{Ax} = 1360.6\,N$$

Resultant Radial force on Bearing B :

$$F_B = \sqrt{F_{Bx}^2 + F_{By}^2}$$

$$\underline{\underline{F_B = 423.2\,N}}$$

Dynamic Load Rating (C)

$$C = A_F P \left(\frac{L}{a_1 L_{10}}\right)^{1/k}$$

For A;

For Roller bearing ; $k = 10/3$

$$C = 1.2\,(1457.5)\left(\frac{10^9}{0.21\times10^6}\right)^{3/10} = 22.2\,kN$$

For Roller bearing ; 25×52×15

$C_{max} = 32.5\,kN$ ∴ Hence Roller bearing is used.

For B;

For ball bearing ; $k=3$

$$C = 1.2\,(423.2)\left(\frac{10^9}{0.21\times10^6}\right)^{1/3} = 8.54\,kN$$

For ball bearing ; 25×52×15

$C_{max} = 14.8\,kN$ ∴ Hence ball bearing is used.

From the above calculations, it is proved that a cylindrical roller bearing on A, and a ball bearing at B is sufficient for the design.

Shaft Analysis

Although, the shaft is designed with steel, it should be able to handle the shear forces and bending moments on it. If it cannot handle the forces, the shaft could bend or break. Therefore, stress analysis on shaft carrying the highest torque is done to know whether the material chosen can handle the shear forces and prevent the shaft from bending or breaking. The output shaft has the highest torque.

Shaft analysis

$$S_e = k_a k_b k_c k_d k_e k_f S_e'$$

k_a - surface condition modification factor

k_b - sige modification factor

k_c - load modification factor

k_d - temperature modificationfactor

k_e - reliability factor

k_f - missellaneous effects modificationfactor

• $k_a = a S_{ut}^{b}$

cold drawn ; $a = 4.51$, $b = -0.265$, $S_{ut} = 1720$

$$k_a = 4.51(1720)^{-0.265} = 0.626$$

• $k_b = 1.24d^{-0.107}$ $\quad 2.79 \leq d \leq 51mm$

$d = 25mm$

$$\therefore k_b = 1.24(25)^{-0.107} = 0.879$$

• k_c

Load type, torsion

$$k_c = 0.59$$

• k_d

Max oil temperature : 120°C

Temperature	k_d
100	1.02
120	k_d
150	1.025

$$\frac{120-100}{150-100} = \frac{k_d - 1.02}{1.025 - 1.02}$$

$$k_d = 1.022$$

- k_e

Reliability = 99.9%

$\therefore k_e = 0.753$

- k_f

- Shaft is assumed as grooved round bar in torsion

$d = 19mm$ (groove) $r = 3mm$ (fillet) $D = 25mm$

As groove has lowest diameter, it has highest stress concentration.

$\frac{r}{d} = \frac{3}{19} = 0.16 \qquad \frac{D}{d} = \frac{25}{19} = 1.32 \qquad \therefore k_{ts} = 1.38$

$q_s \approx 0.97$

$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.97(1.38 - 1) = 1.37$

Shaft assumed grooved round bar in bending

$\frac{r}{d} = 0.16 \qquad \frac{D}{d} = 1.32 \quad \therefore k_t = 1.7$

$k_f = 1 + q(k_t - 1) = 1 + 0.97(1.7 - 1)$

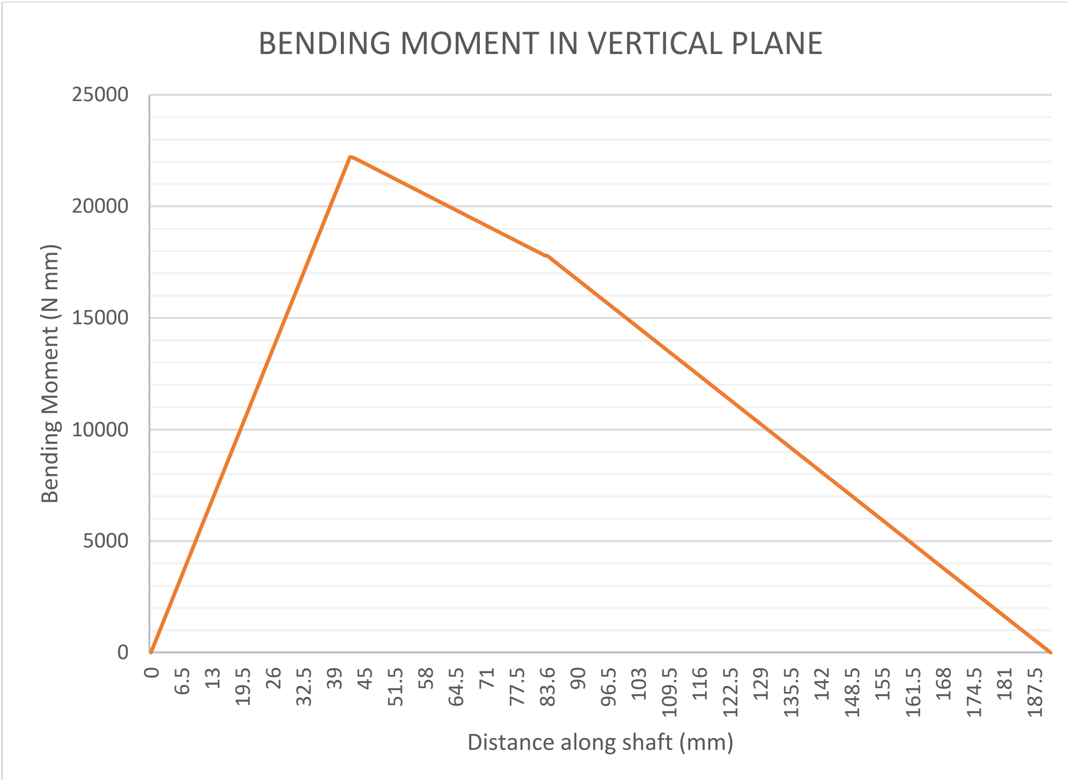
$k_f = 1.68$

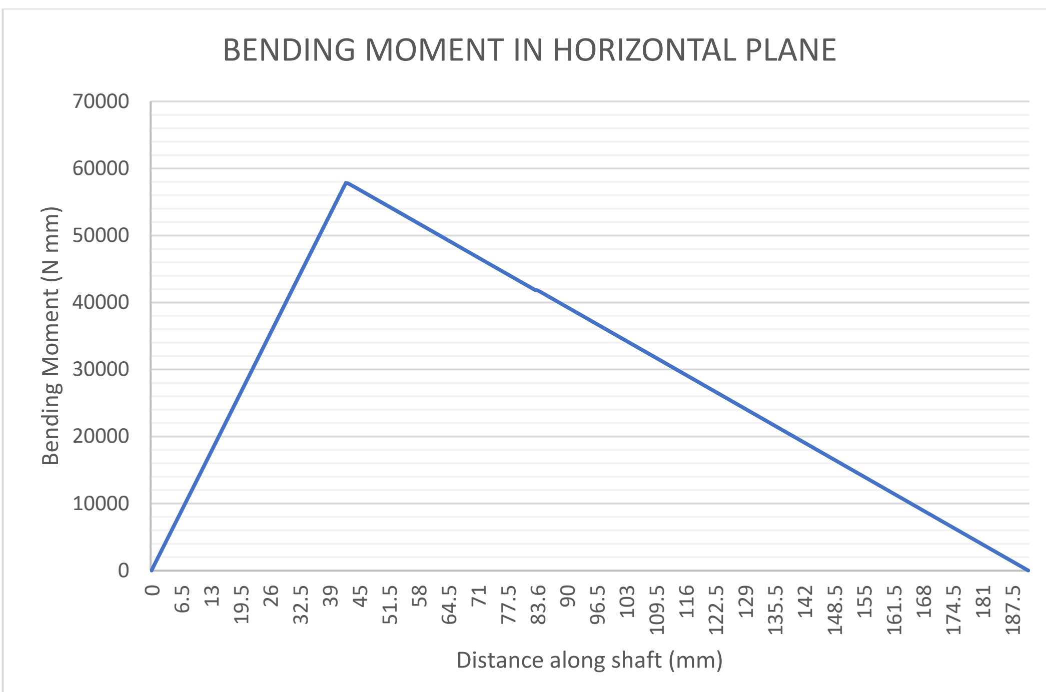
- $S_e' = \frac{1}{2} S_{ut} = 860$

$\rightarrow S_e = 0.626 \times 0.879 \times 0.59 \times 1.022 \times 0.753 \times 1.68 \times 860$

$S_e = 361$

Bending moment shafts are sketched and analyzed to find max bending moment for both horizontal and vertical planes.





Maximum bending moments are determined to be 22.21Nm and 57.826Nm for vertical and horizontal planes respectively.

$$Resultant\ bending\ moment = M$$

$$M = \sqrt{57.826^2 + 22.21^2} = 61.9Nm$$

ASME Calculations

ASME equations

$$d = \left[\frac{32 n_s}{\pi}\sqrt{\left(\frac{k_f m}{S_e}\right)^2 + \frac{3}{4}\left(\frac{k_{fs} T}{S_y}\right)^2}\,\right]^{1/3}$$

n_s = Safty reserve factor ($n_s = 2$)

m = max bending moment on shaft

S_e - Endurance limit stress

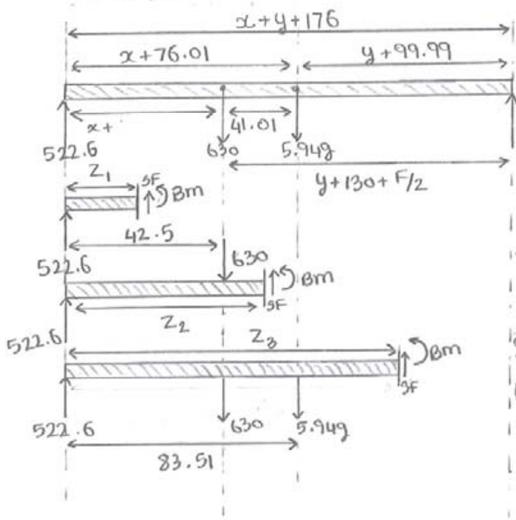
T - Max torque on Shaft

S_y - yield strength of shaft material

Bending Moment and Shear stress:

- Output gear

• Vertical Plane :-



$SF = -522.6\,N$

$Bm = 522.6 \times Z_1$

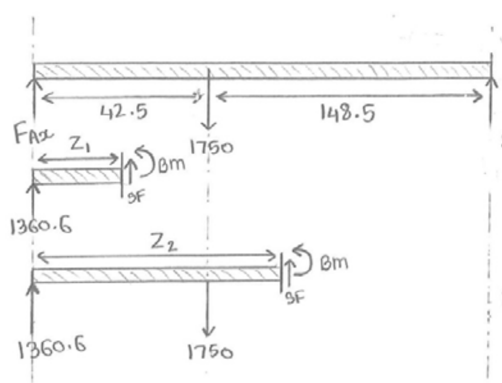
$SF = 630 - 522.6 = 107.4\,N$

$Bm = 522.6 \times Z_2 - 630(Z_2 - 42.5)$

$SF = 630 + 58.3 - 522.6 = 165.7$

$Bm = 522.6(Z_3) - 630(Z_3 - 42.5) - 58.3(Z_3 - 83.51)$

• Horizontal Plane :



$SF = -1360.6\,N$

$Bm = 1360.6(Z_1)$

$SF = 1750 - 1360.6 = 389.4\,N$

$Bm = 1360.6(Z_2) - 1750(Z_2 - 42.5)$

One of the most important things to remember during the design process is that the materials used for each part are suitable to do the work they are supposed to do.

A part whose material must be carefully selected is the shaft. The shaft can be manufactured with aluminum instead of steel to cut cost, but the manufactured shaft should not bend or break. Hence, steel alloy is used for the shaft. However, there are quite a few different steel alloys. The following calculations are done to determine the ideal alloy. For both shafts, the material used is AISI 4340, normalized.

The output shaft is analyzed since it is the shaft with the highest torque and the minimum diameter of shaft possible for the used material is calculated using the ASME equation: -

$$\boxed{d = \left[\frac{32n_s}{\pi} \times \sqrt{\left(\frac{K_f M}{S_e}\right)^2 + \frac{3}{4}\left(\frac{K_{fs} T}{S_y}\right)^2}\,\right]^{1/3}}$$

where;

$n_s = 2$

$M = 61.9$ Nm

$K_f = 1.68$

$K_{fs} = 1.37$

$S_e = 361$ MPa

$T = 110.18$ Nm

$S_y = 1590$ MPa

Substituting the values;

$$d = 0.01827\ m = 18.27mm$$

The diameter used in the design process was 25 which is more than the minimum diameter calculated with ASME. This shows that the material used is appropriate for the requirements of the gearbox.

Material Selections

Gears & Shafts

The normalized steel alloy SAE-AISI 4340 is used to manufacture gears. This material is particularly chosen because it has a very high UTS and high YTS of 1720 MPa & 1590MPa respectively. Thus, the material is very robust and less fragile. Hence it has a low hazard of failure making sure that the gears can withstand high revolution. The material is heated for through hardening procedure. This provides an elongated life span by reducing wear and tear.

Casing

Casing of the gearbox is made using the 201 Annealed Stainless Steel to make it corrosion resistant as the exterior of the gearbox is bare to the sea breeze, which can result in oxidation. Using stainless steel will help the gearbox stay corrosion resistive. The casing needs to have good stiffness and excellent vibration damping properties. Also, the material should not be very expensive.

Gasket

Silicon Rubber is used as the suitable material for the gasket. Rubber has very good flex cracking resistance, tear resistance and impact resistance. It also can act as a very good seal, making sure that the gearbox remains airtight. Also, the rubber is easily accessible in the bazaar.

Others

The clutch cup and clutch cone are made using steel SAE-AISI 4340, since it has high wear and tear resistance, high UTS and YTS. Also, they have high strength and easy to manufacture. These parts can be manufactured using forging procedure. The oil level indicator is a dip stick.