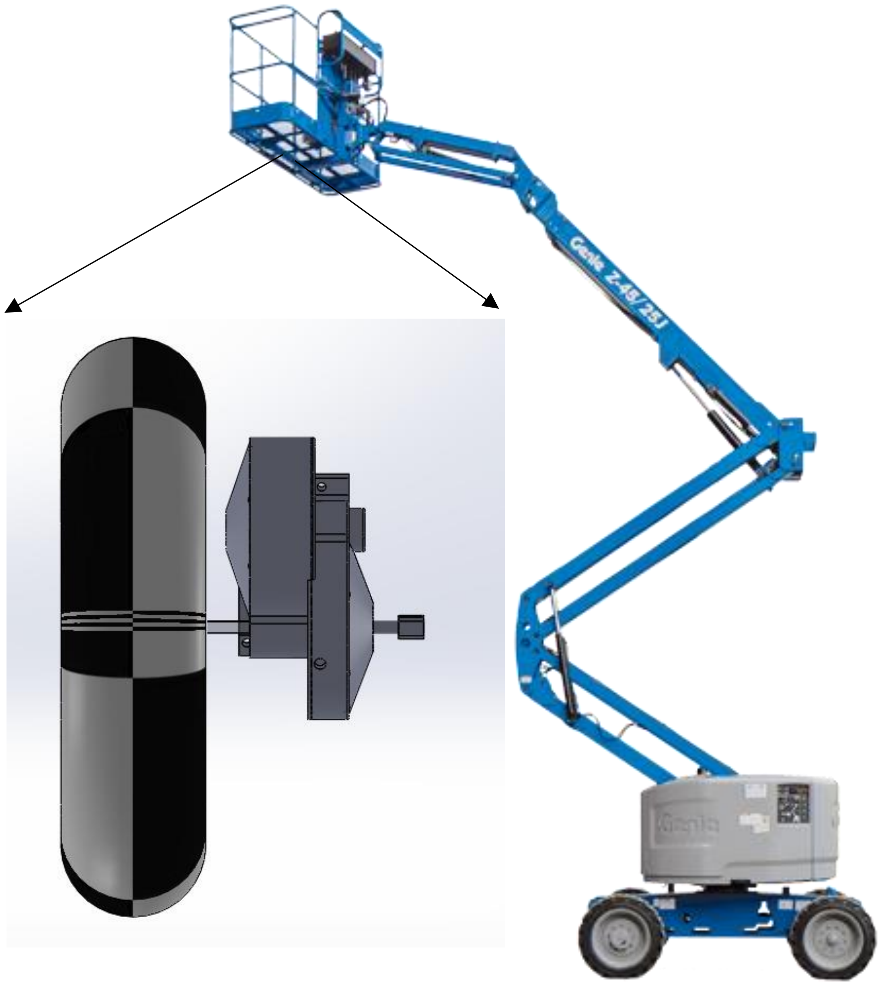

ANAYLYSIS REPORT-CHERRY PICKER GEARBOX

Group 23

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Executive Summary

This in-depth mathematical analysis report focuses on the critical component in the cherry pickers: the coaxially aligned gearbox. Cherry pickers are vital for reaching elevated heights which might have non uniform reach and performing challenging tasks efficiently. The coaxially aligned gearbox plays a pivotal role in transferring power from the motor to the wheels, ensuring optimal energy utilization, safe operations and simple maintenance.

The report delves into various aspects of the gearbox, including gear arrangement, efficiency, load capacity, reliability, and maintenance requirements. The gear calculation analysis establishes the gear ratio, pitch diameters, and face width, ensuring compatibility with the operational requirements. It sheds light on the three types of shafts used, the key in shafts and their suitable dimension aligning with the gears for efficient power transmission.

Bearing calculations have been conducted to determine suitable bearings for different shafts as there are different loads in different shafts. By assessing forces and moments, the report validates the selection of specific bearings for each component.

The analysis accounts for gear sets, considering different scenarios and potential variations in load and operational conditions. Heat treatment and heat treatment factors have been explored to enhance the endurance limit and ensure long-lasting performance.

This report provides valuable insights into the design and functionality of the coaxially aligned gearbox in cherry pickers. By combining theoretical considerations with mathematical data, it contributes to improving cherry picker technology, making elevated tasks safer and more efficient. The results demonstrate that the selected gear and bearing configurations are compatible with the operational requirements, ensuring the gearbox's reliability and safe functionality as per the client’s requirement.

Overall, this report is a substantial resource for engineers, manufacturers, and stakeholders involved in the design and maintenance of cherry pickers, aiding in the continued development of these essential machines for various industries.

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Introduction

In the industry of material handling and aerial work platforms, the cherry picker is an outstanding testament to engineering innovation, making possible for us to reach heights and perform tasks that were once considered challenging or impossible. At the heart of these vehicles and machines lies a crucial component - the gearbox, that is essential for efficiently transferring power from the motor to the wheels, ensuring the safe and effective operation of the cherry picker. Using a suitable gear combination, the speed and torque can be manipulated as required.

This report is a comprehensive mathematical analysis of the gearbox employed in a cherry picker, where the motor and wheel are coaxially aligned. The significance of this simple design cannot be understated; it offers several advantages, including reduced mechanical complexity, enhanced power transmission, and improved space utilization within the machinery.

Our analysis delves into the details of this coaxially aligned gearbox, examining its structural integrity, performance, and overall reliability. We aim to provide valuable insights into the design and functionality of this critical component, shedding light on how it contributes to the operational success and safety of cherry pickers.

Through a combination of theoretical considerations and practical assessments, we will explore various aspects of this gearbox, including:

1. **Gear Arrangement:** An examination of the gear arrangement within the gearbox, which is fundamental to the system's performance.
2. **Efficiency and Power Transmission:** An evaluation of the gearbox's efficiency in transferring power from the motor to the wheel, ensuring optimal energy utilization.
3. **Load calculation and Stress Analysis:** An in-depth look at the gearbox's load-bearing capacity and a stress analysis to determine its structural integrity under real-world operating conditions.
4. **Reliability and Safety:** An assessment of the gearbox's reliability and its role in ensuring the safety of operations.
5. **Maintenance and Serviceability:** A brief discussion of the gearbox's maintenance requirements and serviceability for practical use in line with client demand.

Throughout this report, we will combine theoretical insights with mathematical data to provide a well-rounded mathematical analysis and software simulation of the coaxially aligned gearbox. Our goal is to contribute to the understanding and improvement of this vital component, fostering the continued development of cherry picker technology and, ultimately, the safe and efficient execution of tasks at elevated heights.

Gear Arrangement

In the gearbox system, there are three essential shafts: the input shaft, the intermediate shaft, and the output shaft.

1. **Input Shaft:** The input shaft is where the power is initially introduced into the gearbox. It receives rotational energy, from the motor, and transmits it into the gearbox.

2. **Intermediate Shaft:** The intermediate shaft is a critical component that bridges the input and output shafts. It is positioned parallel to them and serves as a means to transfer power from the input shaft to the output shaft.
3. **Output Shaft:** The output shaft is the final stage of the gearbox, and it is directly connected to the whee. Because of its connection to the wheel, it experiences axial loads, which are forces applied along its axis.

To support the axial load on the output shaft, tapered roller bearings have been utilized. Tapered roller bearings are well-suited for applications where axial loads are present, as they can efficiently handle both axial and radial loads (SKF, 2023).

As for the other two shafts, the input and intermediate shafts, they have roller bearings installed. These roller bearings are chosen for their ability to reduce friction and thereby minimize energy losses within the gearbox. Roller bearings are particularly useful for radial loads and work by rolling elements (cylindrical or needle-shaped rollers) between the inner and outer rings to reduce friction and facilitate smooth rotation.

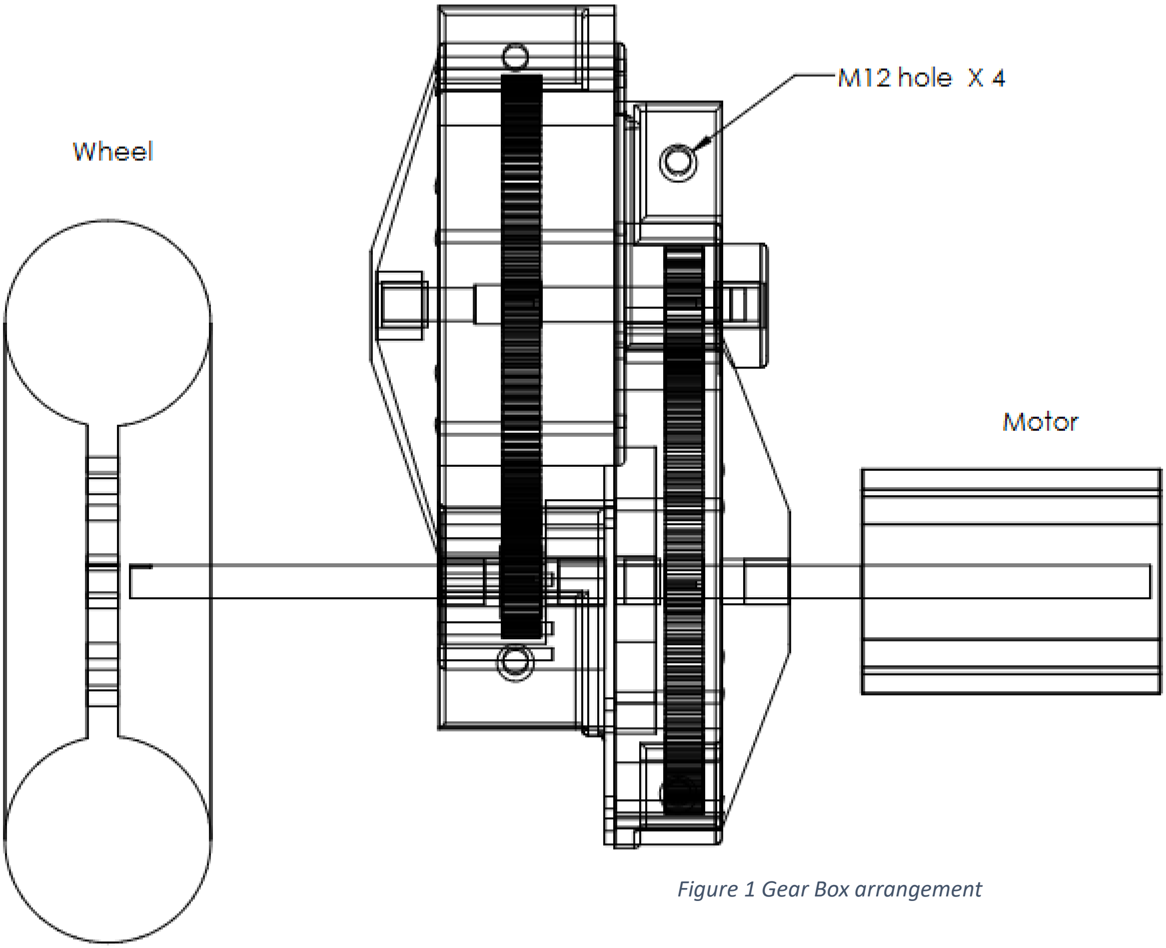


Figure 1 Gear Box arrangement

In addition to the bearings, a specific gear arrangement has been implemented within the gearbox. A small gear, is connected to the first intermediate gear which is size of output gear. This ensures that power is transferred efficiently from the intermediate stage to the output stage. Similarly, the second intermediate gear, with the same size as the input gear, is connected to the output gear, creating a synchronized and optimized gear system that ensures the desired speed and torque are achieved in the output.

This comprehensive arrangement of shafts, bearings, and gears collectively allows for the efficient transmission of power and torque from the input to the output, ultimately driving the wheel with minimal friction and energy loss.

Analysis

Gear Calculation

Our maximum resistive torque that we are facing is 913.7 Nm which includes drag, hill climb and friction forces. Gear ratio to overcome will be following:

$$G.\,R = \frac{913.7\,Nm}{51Nm} = 17.9$$

So, the gear ratio of the gear bon should be more than 17.9

If we keep our gear ratio 18.5 of the whole gear box we will achieve the speed of 31 RPM outer shaft that is connected to wheel. Hence nearly 5km/h speed will obtain with higher.

The gear box has 4 gears with one input, one output and one intermediate shaft. The individual ratio of each matching pairs of gears follows as:

$$18.5 = 4.3 \times 4.3$$

Let us begin with the calculation of the 1st pair of gears.

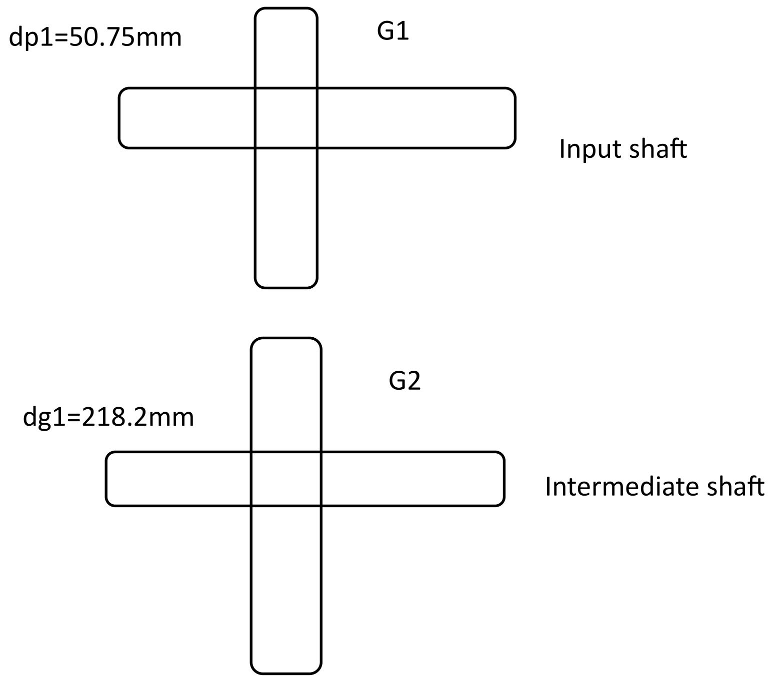


Figure 2 First pair of gear

Material=AISI 9000 series steel

Here,

n1= 570 RPM G. R=4.3

n2=132.5 RPM T1=51 Nm

T2=219.3 Nm

G.R=$\frac{n_1}{n_2}=\frac{w_1}{w_2}=\frac{d_2}{d_1}=\frac{p_2}{p_1}=\frac{N_2}{N_1}=\frac{T_2}{T_1}$

Now calculate Np

Np=$\frac{2k}{(1+2m)Sin^2\theta}\left(m+\left(\sqrt{m^2}(1+2m)Sin^2\theta\right)\right)$

K=1, θ =14.5 for the first set of gears

Np=$\frac{2(1)}{1+2(4.3)Sin^2(14.5)}\left(4.3+\sqrt{4}.3^2+\left(Sin^2(14.5)\right)\right)$

Np=$\frac{2}{9.6\ Sin^2(14.5)}\left(4.3+\left(\sqrt{1}8.49+(9.6)Sin^2(14.5)\right)\right)$

Np=28.8

≈ Np=29

Ng=$\frac{Np^2\ Sin^2\theta-4k^2}{4k-2Np\ Sin^2\theta}$

Ng=$\frac{(29)^2Sin^2\theta(14.5)-4}{4-2(29)Sin^2(14.5)}=\frac{48.72}{0.36}=$ **135.33**

(I)

∴ Np=29 Ng=135.33

Our gear ratio is 4.3

If Np is 29 then Ng=29×4.3=124.7≈125

So, this will work as it's in the limit.

Ne supposes a module of 1.75 mm.

So, Pitch diameter=d_p=mNp

= (1.75) (29)

$D_{pitch}=50.75mm$

$D_{base}=D_{pitch}\,Cos\theta, \theta=14.5°$

$D_{base}=49.13mm$, $D_{Added}=D_{pitch}+2m$

=50.75+2(1.75)

$D_{base} = 49.13mm$, $D_{Added} = 54.25$

$$r_{base} = 24.565mm \quad , \quad r_{Added} = 27.125mm$$

Contact ratio$=\frac{\left(\sqrt{r_{a_1}^2 - r_{b_1}^2}\right)+\left(\sqrt{r_{a_2}^2 - r_{b_2}^2}\right)-C\,Sin\theta}{p\,Cos\theta}$

$$P = m\pi = (1.75)(\pi) = 5.497mm$$

Similarly, we find the values per gear.

$$D_{pitch} \quad = 218.225mm$$

$r_{b_2} = 105.637$, $r_{a_2} = 110.865$, C=134.48

$$= \frac{\left(\sqrt{(27.125)^2 - (24.565)^2}\right) + \left(\sqrt{(110.865)^2 - (105.637)^2}\right) - 134.48\,Sin(14.5)}{5.497\,Cos(14.5)}$$

= 2.159

As C.R> 1.2 so our gear is compatible.

(II)

Now we can find if our face width will be compatible with Np=29, using graphs Luis factor=0.31

$$d_p = mNp = 1.75 \times 29 = 50.75mm$$

Velocity=v$=\frac{\pi dn}{60} = \frac{\pi(50.75\times10^3)}{60}$

Velocity=1.51 m/s

Now we can find power

$$w = \frac{2\pi n}{60} = 59.7\,rad/s \quad when\ n = 570\,RPM$$

P=J.w=(51NM0959.7 rad/s)

Power=30.44w

Tangential Force=Ft=$F_t = \frac{P}{V\,3044.71.51m/s=2016.35}$

The dynamic factor is calculated as

Kv$=\frac{6.1+v}{6.1} = \frac{6.1+1.51}{6.1} = 1.24$

Now face width can be calculated.

b= $\frac{Kv.Ft}{\sigma.m.y}$

Now this allowable stress will be taken against yield strength of our Material.

We can use AISI 9000 series steel.

Here with an average yield strength of 760 MPa.

$\sigma Added = \frac{760}{3} = 253MPa$

Table Strength of 1040MPa.

(III)

Taking a safety factor of 3.

b= $\frac{(1.24)(2016.35)}{(253)(1.75\times10^{-3})(0.31)} = 18.2mm$

So, we can use face width of 18.2mm

$b_{min} = 3\pi m = 16.5mm$

$b_{max} = 27.5mm \ as \ 5\pi m \ = b_{max}$

So, face width is compatible.

Now we will do the stress calculations.

We will find bending stress and Contact stress.

$\sigma B = \frac{Ft}{bmyj}(K_v \ K_a K_s K_H K_B)$

For our gear we can write it as following:

$K_a = 1, \ K_B = 1, \ K_s = 1, K_H = 1, K_v = 1.24$

$Ft = 2016.35N, \ b = 18.2mm, \ \ m = 1.75 \ mm$

$\sigma B = \frac{2016.35}{(18.2\times10^{-3})(1.75\times10^{-3})(0.38)}$

$$\sigma B = 166MPa$$

(IV)

Endurance Strength :

As our material is steel.

Our Skt= 1010

80,$Se' \ = 0.5 \ (1010)$

$$Se' \ = 505 \ MPa$$

Where as $Se = K_a K_b K_c K_d K_e K_f Se'$

Now $K_a = a\, Skt^b$

$$a = 1.58 \quad , \quad b = -0.085$$

$$K_a = (1.58)(1010)^{-0.085}$$

$$K_a = 0.87$$

$$K_b = 1 \quad , K_c = 1, \quad K_d = 1, \; K_e = 1. \;\; K_p = 1.$$

By putting values,

$$S_e = 0.87(505) = 443.39MPa$$

n=$\frac{S_e}{\sigma B} = \frac{443.39MPa}{166MPa}$

n=2.7

(V)

Contract Stress. 1st Gear

$$\sigma H = -C_p \sqrt{\frac{C_v\, F_t}{bdp\; I}}$$

$$C_v = known$$

$$F_t = Force$$

$$b = \; Face\; width$$

$$d_p = Pitch\; diameter$$

$$C_p = \sqrt{\frac{1}{\pi\left(\frac{1 - v_{p^2}}{E_p} + \frac{1 - v_{G^2}}{E_G}\right)}}$$

$$= \sqrt{\frac{1}{\pi\left(\frac{1-0.27^2}{200\times10^3} + \frac{1-0.27^2}{200\times10^3}\right)}}$$

=185.29

(VI)

$$I = \frac{Cos\theta\; Sin\theta}{2} \; \frac{m_G}{m_G + 1}$$

$$= \frac{Cos\; 14.5\; Sin\; 14.5}{2} \; \frac{4.3}{4.3 + 1}$$

= 0.1

$$\sigma H = -964MPa$$

S=2.76(514)-70

S=1348

$$n = \left(\frac{S}{\sigma H}\right)^2 - \left(\frac{1348}{-964}\right)^2$$

Safe and Adequate= 2

Gear calculation of second pairs;

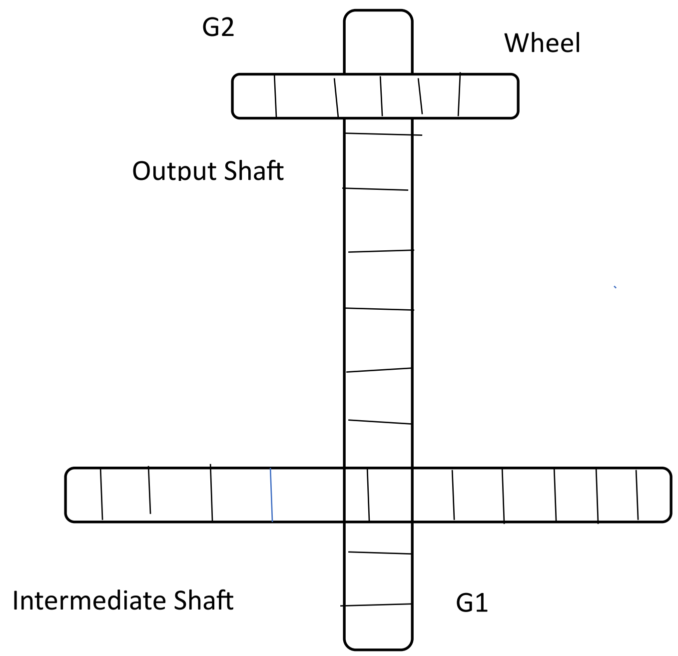


Figure 3 2nd pair gear type

In this case,

$$n_1 = 132.5RPM_1$$

$$n_2 = 30.82RPM$$

$$T_1 = 219.3Nm$$

$$T_2 = 942.99Nm \approx 943Nm$$

G.Ratio=4.3

In this set of gears, we have a pressure angle of 25 as it aligns.

As our pressure angle is 25°minimum teeth should be 17. We have found these values by coaxially align.

We can find the Np and as:

$$N_p = \frac{2(1)}{1+2(4.3)\,Sin^2 25}\left(4.3+\sqrt{4}.3^2(1+2(4.3)Sin^2 25)\right)$$

$$N_p = \frac{2}{9.6\,Sin^2 25}\left(4.3+\left(\sqrt{1}8.49+9.6\,Sin^2 25\right)\right)$$

$$N_p = 10.25 \approx 11$$

But as the condition says minimum teeth with

$$\theta = 25° is\ 12,\ as\ N_p = 12$$

(I)

$$N_g = \frac{(12)^2\,Sin^2(25)-4}{4-2(12)Sin^2(14.5)} = 7.5$$

Whereas if we are chosen as 17 and our ratio is 4.3.

$N_g = 17 \times 4.3 = 73$, so, the design will Successfully operate.

As we have a constraint that states the input and output shaft should coaxially align. For this sole reason we will use a custom module. A module of 1.985 will be used which is remarkably close to standard 3mm.

However, gear design involve precise calculations and esquire specialized manufacturing .

By choosing this module and N_p as 17 we will have the same pitch diameter of the gear on the input shaft.

The calculations follows as:

$$N_p = 17,\ N_c = 73,\ m = 2.985$$

$$d_p = mNp = 2.9853 \times 17 = 50.75mm$$

Similarly, $d_c = (4.3)(50.75) = 218.225mm$

(II)

Velocity=$\frac{\pi dn}{60} = \frac{\pi dnp}{60} = \frac{\pi(50.75\times10^{-3})(132.5)}{60}$

$$V = 0.35m/s$$

Similarly, $w = \frac{2\pi n}{60} = \frac{2\pi(132.5)}{60} = 13.8rad/s$

P=J;w=$(219.3Nm)(13.8rad/s)$

=3044.7w

Now the tangential force can be calculated.

$F_t = p/v = \frac{3044.7}{0.35} = 8697.7N$

Dynamic factor; $K_v = 1.057$

$$\sigma allowable = \frac{760MPa}{3} = 253MPa$$

The face width can be calculated with the help of these.

b=$\frac{K_v.F_t}{\sigma p.m.y}$

Here y=0.3424 from the table lecture notes.

b=$\frac{(1.057)(8697.7)}{(253\times10^6)(2.9853\times10^{-3}m)(0.3424)} = 35.5mm$

$$b_{min} = 3\pi m = 28mm$$

$$b_{max} = 5\pi m = 46.9mm$$

So as our face width is within the limit, it can function successfully.

Now we will calculate the bending stress and endurance limit.

$$\sigma B = \frac{F_b}{bmyj}(K_v\, K_a\, K_s\, K_H\, K_B)$$

For these gears, $K_v = 1.057, K_A = 1, K_B = 1, K_s = 1, K_y = 1$

$$F_t = 8697.7N,\ b = 35.5mm$$

$$d_p = 50.75,\ m = 2.985, y_{i=}$$

$$\sigma b = \frac{8697.7N}{(35.5\times10^{-3})(2.985\times10^{-3})(0.33)}$$

=**226MPa**

Where as $S_e = 443.4Mpa$

$$n = \frac{443.4MPa}{226} = 1.96$$

$$(IV)$$

Contract Stress 2nd gear.

$$\sigma H = -C_p \sqrt{\frac{C_V \ F_t}{bdpI}}$$

$$C_p = \sqrt{\frac{1}{\pi \frac{\left(1-(V_p)^2\right)}{E_p} + \frac{(1-(V_G)^2)}{E_G}}}$$

$$= \sqrt{\frac{1}{\pi\left(\frac{1-(0.27)^2}{200\times 10^3}+\frac{1-(0.27)^2}{200\times 10^3}\right)}}$$

$$C_p = 185.29$$

$$\text{I} = \frac{Cos\theta \ Sin\theta}{2} \ \frac{m_G}{m_G+1}$$

$$\frac{Cos25 \ Sin25}{2} \ \frac{4.3}{4.3+1} = 0.15$$

$$(V)$$

$$\sigma H = -185.29 \sqrt{\frac{(1.057)(8697.7)}{(35.5)(50.75)(0.5)}}$$

= -1080.72

= 2.76(514)-70

=1348

n $=\left(\frac{1348}{-1080}\right)^2 = \mathbf{1.6}$

We can give heat treatment to increase the supply factor above to adequate level. The values of strength taken all average values that can be increased significantly by heat treatment and creating or allowing.

Contract Ratio for 2nd set of gears;

$$D_{pitch} = 50.75mm \quad , D_{pitch} = 218.225mm$$

$$D_{A_1} = 56.72mm \rightarrow r_{A_1} = 28.36 \ , \quad D_{A_2} = 224.1mm \rightarrow r_{A_2} = 112$$

$$D_{B_1} = 45.99 \rightarrow r_{B_1} = 23 \quad , \quad D_{B_2} = 197.8 \rightarrow r_{B_2} = 98.4$$

C.Ratio=134 , $p = \pi m = 9.36$

$$C.R = \frac{\left\{\sqrt{(r_{a^1})^2 - (r_{b^1})^2}\right\} + \left\{\sqrt{(r_{a^2})^2 - (r_{b^2})^2}\right\} - C\,Sin\theta}{pCos\theta}$$

$$C.R = \frac{\left\{\sqrt{(28.36)^2 - (23)^2}\right\} + \left\{\sqrt{(112)^2 - (98.4)^2}\right\} - C\,Sin25}{4.36Cos20}$$

C.R=1.42

As contact ratio$>$ 1.2

The gear will successfully mesh.

Bearing Calculations

We will be using tapered (Bearing Assembly reliability is 95%

$(0.95)^{\frac{1}{6}}$ = Read binary

Reliability of each Bearing = Read = 0.99

L = 43,800 As we have given 5 years.

L_{10} = ?

L = ($5 \times 365 \times 24$) hrs = 43,800 hrs

By using Formula and putting values

0.99 = exp $\left(-\frac{43,800}{6.84\,L_{10}}\right)^{1.17}$

Taking ln (Natural log) on both side

+ 0.01005 = $+\left(\frac{43,800}{6.84\,L_{10}}\right)^{1.17}$

0.0196 = $\frac{43,800}{6.84\,L_{10}}$

$L_{10} = \frac{43,800}{0.134} = 32.655$ hrs

$L_{10} = 32.655$ hrs

This is L_{10} when we have taken L has 5 years of operating time.

(I)

<u>Bearing calculations for the first set of gears</u>

$$F_c = 2016.35 \qquad \theta = 14.5°$$

$$F_t = F\,Cos\theta$$

$$F = \frac{F_t}{Cos\theta} = \frac{2016.35}{Cos(14.5)}$$

$$F = 2082.7N$$

$$F_r = F\ Sin\theta$$

$$F_r = (2082.7)Sin\ 14.5°$$

$$F_r = 521.64N$$

As both forces act radially on bearing.

$$F_r = \left\{\sqrt{(521.64)^2 + (2016.35)^2}\right\}$$

$$F_r = 2082.7N$$

(Π)

<u>Bearing Calculation for Input Shaft</u>

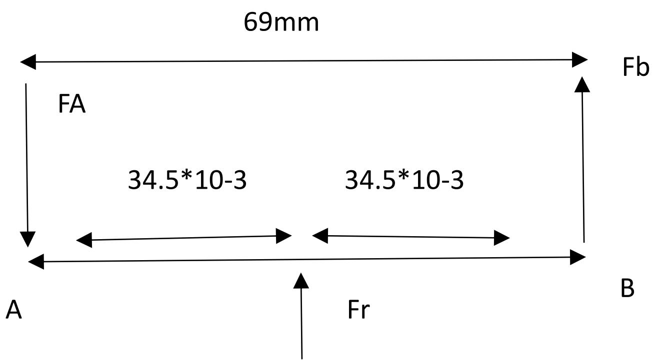


Figure 4 Input shaft bearing calculation

$$+\uparrow \Sigma F_y = 0 = Taking\ summation\ of\ forces.$$

$$F_B + F_R - F_A = 0$$

Here $F_r = 2082.7N$

$$F_B + 2082.7N - F_A = 0$$

$$F_A = F_B + 2082.7 \quad \ldots \ldots (\text{I})$$

Taking Moment at A.

0=$(-F_v)(34.5 \times 10^{-3}) - (69 \times 10^{-3}) + (-F_B)(69 \times 10^{-3}) - 71.853 - 69 \times 10^{-3}\ F_B$

$$= -(2082.7)(34.5 \times 10^{-3}) - (69 \times 10^{-3})F_B$$

$$= -71.853 - 69 \times 10^{-3}\ F_B$$

$$F_B = \frac{-71.853}{69 \times 10^{-3}} = -1041.35N$$

Putting it in equity.

$F_A = F_B + 2082.7 = 1041.35N$

$$F_A = 1041.35N$$

$$(III)$$

So , the assured directions can be collected.

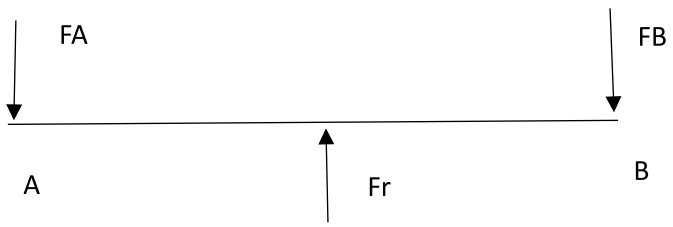


Figure 5 Force diagram for input shaft

The calculations for both bearing are following as:

Bearing A:

$$Fe_A = 0.4\,Fr_A + K_A\left(\frac{0.47\,Fr_B}{K_B}\right)$$

Here $Fr_A = 1041.35N$

$$K_A = K_B = 1.5N\,,\;\; Fr_B = 1041.35N$$

By putting value in above formula.

$$Fe_A = (0.4)(1041.35N) + 1.5\left(\frac{(0.47)(1041.35)}{1.5}\right)$$

$$Fe_A = 416.54 + 489.4$$

$$Fe_A = 905N$$

As in this case Fr_A and Fr_b are equal so,Fe_A and Fe_B will be same It can be observed that Fe_A and Fe_B are smaller than Fr_A and Fr_B . So ,we will use these forces. As $Fr_A = Fr_B = 1041.35N$

Where as $Fe_A = Fe_B = 905N$.

So, we will use Fr_A and Fr_B in Dynamic load rating calculations.

$$(IV)$$

Now, $C_A = Fe_A . a_f \left[\frac{\frac{L_D n_D 60}{L_r n_r 60}}{0.02+4.91\left[\ell n\frac{1}{R}\right]^{\frac{1}{1.4}}}\right]^{\frac{1}{a}}$

Now $a = \frac{10}{3}$, $a_f = 1$, $L_D = 32{,}6554\ hours$

$$R = 0.99\ ,\ n_D = 570RPM\ ,\ Fr_A = 1041.35N$$

Here, $L_R n_R 60 = 1 \times 10^6$

We will do the calculations based on 1×10^6 revolutions L_{10} for the ISO life calculation.

$$C_A = (1041.35)(1)\left[\frac{\frac{(316554)(570)(60)}{1 \times 10^6}}{0.02 + 4.91\left[\ell n\left(\frac{1}{0.99}\right)\right]^{\frac{1}{1.4}}}\right]^{\frac{3}{10}}$$

$$C_A = 1041.35\left[\frac{1116.8}{0.20}\right]^{\frac{3}{10}}$$

$$C_A = 1041.35 \times 13.3 = 13857.3 \approx 13.9KN$$

As $Fr_A = Fr_B\ and\ Fe_A = Fe_B\ So,\ here\ C_A\ will\ be\ similar\ to\ C_B.$

$$(V)$$

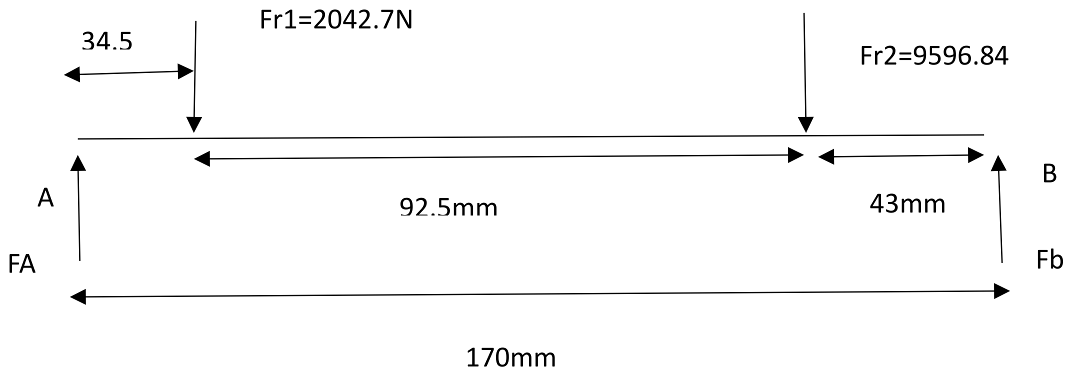
In this Case we will use Tapered roller bearing. We use TIMKEN Type TS tapered Bearing. The specific bearing will be type 21063. Type 21063 will be compatible with our loading conditions.

$$d_{Bore} = 15.875mm$$

Similar bearing will be used for both ends.

$$(VI)$$

Intermediate Shaft:



Now we will find the F_A,F_B (reaction forces) Taking summation of forces.

$+\uparrow \Sigma\ F_A + F_B - 2082.7 - 9596.84$

$$F_A + F_B = 11679.54N\ \ldots\ldots\ldots\ equal$$

Taking moments at A.

$+\curvearrowright \Sigma M_A$=0=$(Fr_1)(34.5) + (9596.84)(127mm) - F_B(170)$

$$(2082.7)(34.5 \times 10^{-3}) + (9596.84)(127 \times 10^{-3}) - F_B(170 \times 10^{-3})$$

$$= 71.85 + 1218.8 - F_B(170 \times 10^{-3})$$

$$Fr_B = \frac{1290.6}{170 \times 10^{-3}} = 7592N$$

$$Fr_B = 7592N \; [Radical\; force\; reaction\; at\; A\,,\; Reaction\; at\; A\,]$$

By using F_B $in\; equal$ $F_A + F_B = 11679.54N$

$$F_A = 4087.5N \; [Radical\; force\,, Radical\; at\; B]$$

Here $Fr_A = 4087.5N$, $Fr_B = 7592N$

By using the values of Fr_A and Fr_B.

Bearing A:

$$Fe_A = 0.4Fr_A + K_A\left(\frac{0.47\; Fr_B}{K_B}\right)$$

$$Fe_A = 0.4(4087.5) + 1.5\left(\frac{0.47(7592N)}{1.5}\right)$$

$$Fe_A = 1635 + 3568.24$$

$$Fe_A = 5203.2N$$

$$Fe_B = 0.4\; Fr_B + K_B\left(\frac{0.47Fr_A}{K_A}\right)$$

$= 0.4\;(7592) + 1.5\left(\frac{0.47(4087.5)}{1.5}\right)$

$= (3036.8) + 1921.125$

= 4957.925N

Now we can get our dynamic load rating based on Fe_A and Fe_B

$$C_A = Fe_A . a_f \left[\frac{\dfrac{L_D n_D 60}{L_r n_r 60}}{0.02 + 4.91\left[\ell n \dfrac{1}{R}\right]^{\frac{1}{1.4}}}\right]^{\frac{1}{4}}$$

In this shaft we use 10×10^6 revolutions L_{10} as ISO method.

Here $a_f = 1$ $a = \frac{10}{3}, n_p = 132.5RPM,\; Fe_A = 5203.2N$

$L_v n_r 60 = 1.0 \times 10^6$, R=0.99, $L_D = 326554$

Putting all the values

$$C_A = (5203.2)(1)\left[\frac{\frac{(326554)(132.5)60}{1\times10^6}}{0.02+4.91\left[\ell n\left(\frac{1}{0.99}\right)\right]^{\frac{1}{1.4}}}\right]^{\frac{1}{\frac{10}{3}}}$$

$$C_A = (5203.2)\left[\frac{28.84}{0.20}\right]^{\frac{3}{10}} = \left(\frac{28.84}{0.20}\right)^{\frac{3}{10}}$$

=(5203.2)(4.4)

=22894N

C_A=22.9KN

$$(III)$$

Similarly, we will be calculating C_B.

But here Fr_B is greater than Fe_S will be use.

$$C_B = Fr_B.1\left[\frac{\frac{L_D n_D 60}{L_r n_r 60}}{0.02+4.91\left[\ell n\frac{1}{F}\right]^{\frac{1}{1.4}}}\right]^9$$

Putting values.

$$(7592).(1)\left[\frac{\frac{(326544)(132.5)60}{1\times10^6}}{0.02+4.91\left[\ell n\frac{1}{0.99}\right]^{\frac{1}{1.4}}}\right]^{\frac{3}{10}}$$

$$(7592)\left(\frac{28.84}{0.20}\right)^{\frac{3}{10}}$$

$$C_B = (7592)(4.4) = 33404$$

For both load ratings we can use TIMKEN 21063 which that load ratings easily d=15.875mm.

$$(IV)$$

$$(1)$$

Bearing Calculation:

$$F_t = 8697.7N$$

$$F_t = F\,cos\theta$$

$$F = \frac{F_t}{Cos\theta} = \frac{8697.7}{Cos25^{\circ}}$$

$$F = 9596.8N$$

$$F_r = F\,Sin\theta$$

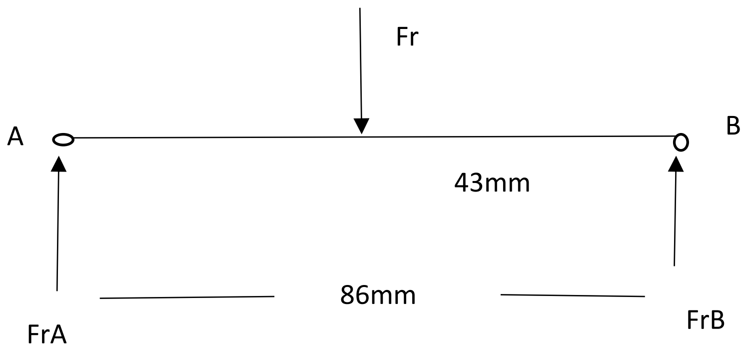
$$F_r = 9596.8\,Sin25^{\circ}$$

= 4055.8N

$$F = \sqrt{(F_r)^2 + (F_t)^2}$$

$$= \sqrt{(4055.8)^2 + (8097.7)^2}$$

=9596.8N



$$(2)$$

$$+\uparrow \Sigma F_y = 0$$

$$Fr_A + Fr_B - Fr = 0$$

$$Fr_A + Fr_B = Fr$$

$$+\curvearrowleft \Sigma A = 0$$

$$Fr_B(86 \times 10^{-3}) - Fr(43 \times b^{-3}) = 0$$

$$Fr_B = \frac{(9596.8)(43 \times 10^{-3})}{(86 \times 10^{-3})}$$

$$Fr_B = 4798.425N$$

$$Fr_A = 9596.85 - 4798.425$$

Bearing A:

$$Fe_A = 0.4Fr_A + K_A\left(\frac{0.47\,Fr_B}{K_A} + T_e\right)$$

=$0.4(4798.425) + 1.5\left(\frac{(0.47)(4798.425)}{1.5} + 0\right)$

=1919.37+2255.25975

=4174.63N

This force (Fe_A) is less than actual radial force (Fr_A) so we will use(Fr_A) =4798.425N.

$$(Fe_B) = 0.4\,Fr_B + K_B\left(\frac{0.47\,Fr_A}{K_B} - T_c\right)$$

$$(Fe_B) = 0.4\,(4798.425) + 1.5\left(\frac{(0.47\,)(4798.425)}{1.5} - 0\right)$$

=1919.37+2255.25975

=4174.63N

This force Fe_B is less than actual radial force Fr_B so we will used Fr_B =4798.425N

$$C_A = Fe_A\left[\frac{\frac{L_D n_D 60}{L_r n_r 60}}{0.02 + 4.91\left[\ell n\,\frac{1}{R}\right]^{\frac{1}{1.4}}}\right]^{\frac{1}{9}}$$

We will use Fr_Ainstead of Fe_A

$n_D = 30.82RPM \;\; Fe_A = 4798.425 \;\; L_D = 336554 \;\; a = \frac{10}{3}$

$$= 4798.425\left[\frac{\frac{(336554)(30.82)(60)}{1 \times 10^6}}{0.02 + 4.91\left[\ell n\left(\frac{1}{0.99}\right)^{\frac{1}{4}}\right]}\right]^{\frac{1}{\frac{10}{3}}}$$

$$(3)$$

$$= 4798.625\left[\frac{622.36}{0.2037}\right]^{\frac{3}{10}}$$

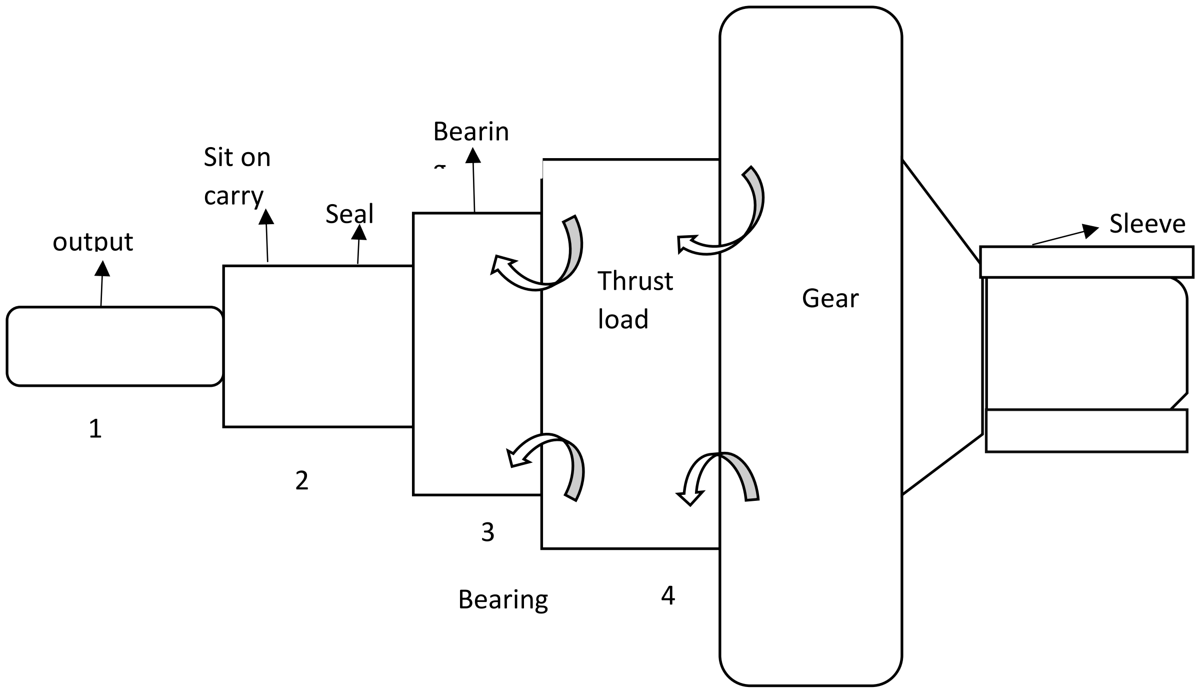
=53288.3N

=53.3KN

Type 21063 TIMKEN type T.S will be used.

Here $C_A = C_B$

So, both have some bearing.



1)Output shoulder connected to wheel (output).there is a change in diameter 1 from 2. as shoulder 1 will sit a casing. The second reason is to save the seal that is connected to 2. If there is no 2nd shoulder wheel come hit the body of gear bon (device body) while with shoulder it sits on the casing.

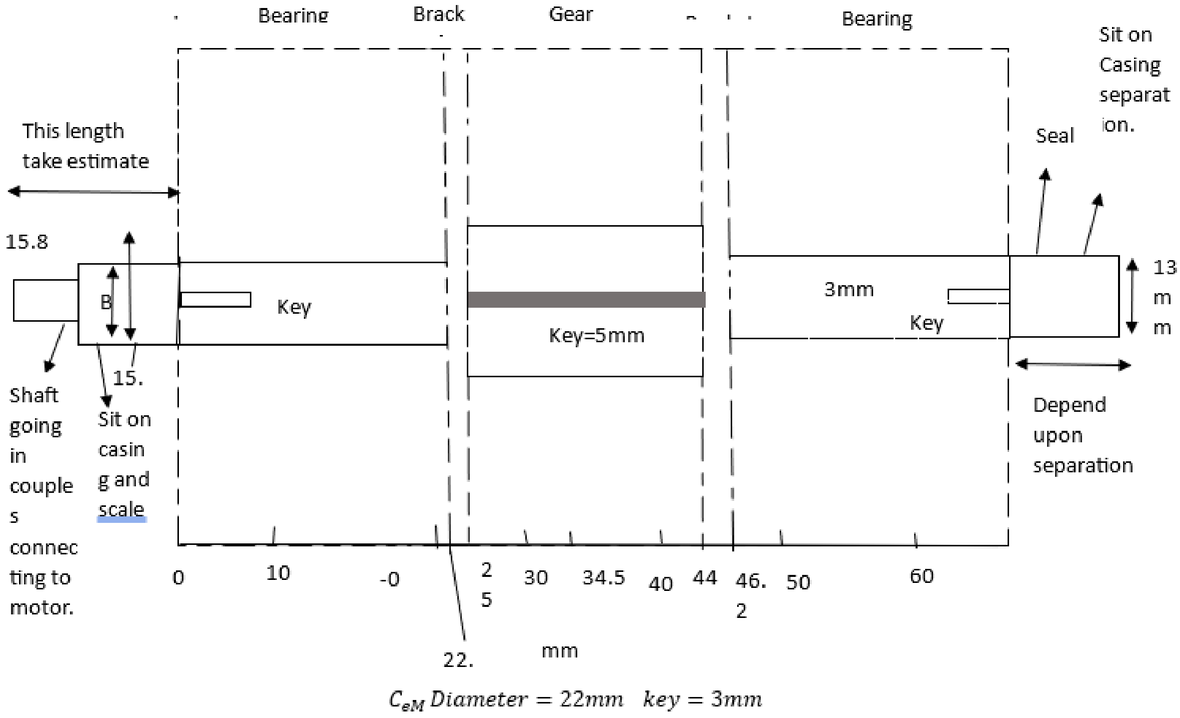


Figure 6 Input Shaft

$C_{eM}\ Diameter = 22mm \quad key = 3mm$

Bearing Bore=15.875mm

Bracket Diameter=18mm

Now we can add the length of the shaft from input side for motor connection.

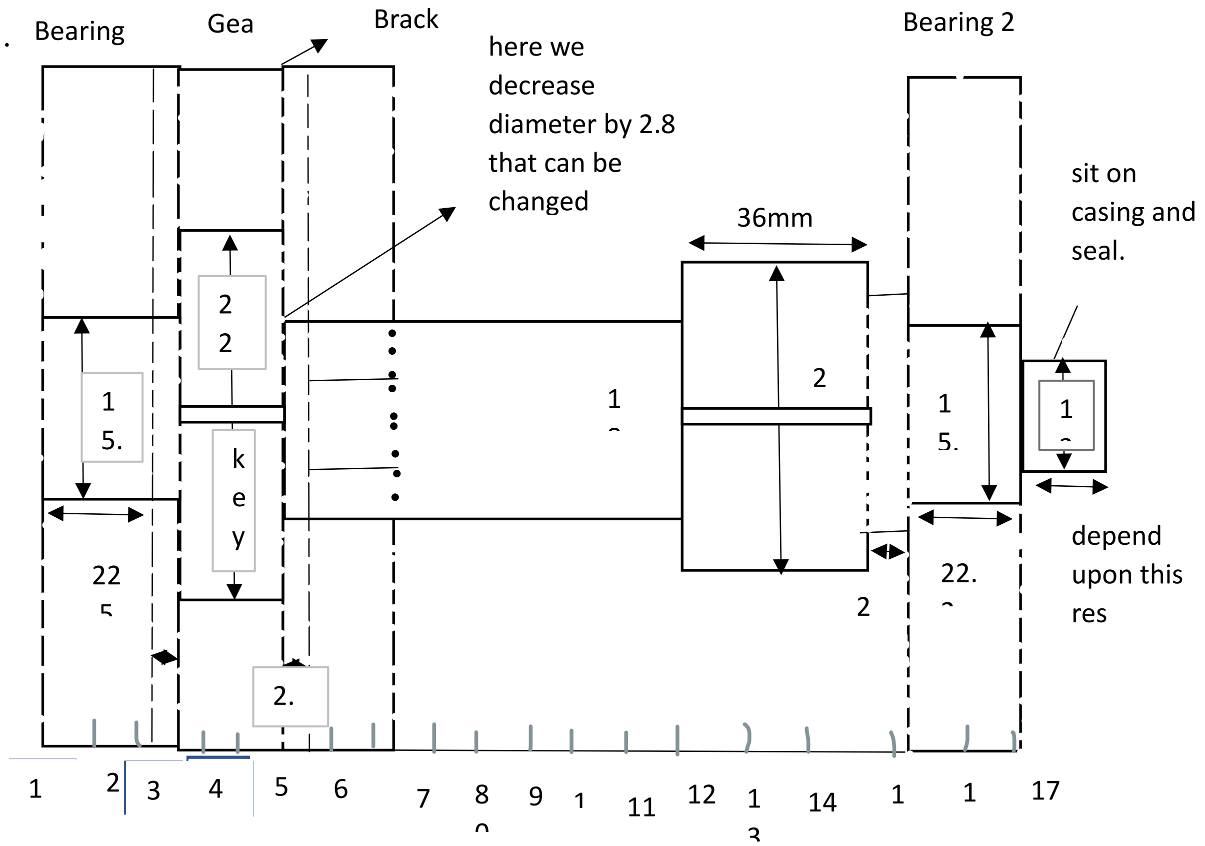


Figure 7 Intermediate Shaft

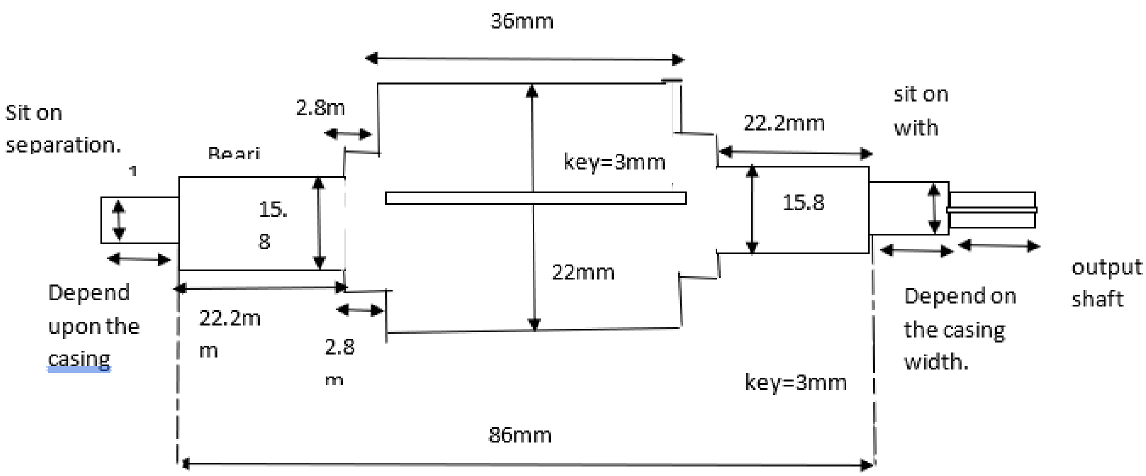


Figure 8 Intermediate Shaft dimensional view -2

Calculation Shaft 1

$$Sut = 1040MPa$$

<u>Sy=760 MPa</u>

<u>Number of Cycles (N)=?</u>

$$+\uparrow \Sigma Fy = 0 \quad R_A + R_B - Fr = 0$$

$$R_1 + R_2 = F_r$$

$$+\curvearrowleft \Sigma M_A = 0$$

$$R_B(69 \times 10^{-3}) - Fr(34.5 \times 10^{-3})$$

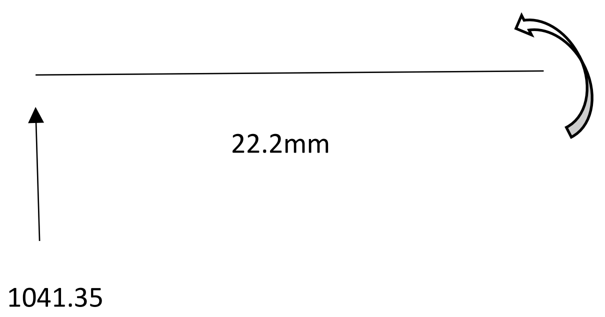
$$R_B = \frac{(2082.7)(34.5 \times 10^{-3})}{(69 \times 10^{-3})}$$

$$R_B = 1041.35N$$

$$R_A = Fr - R_B \Rightarrow 2082.7 - 1041.35$$

$$R_A = 1041.35$$

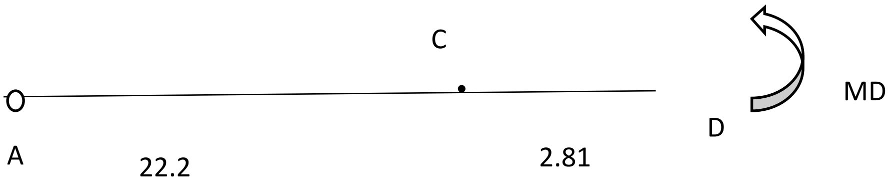
Cut 1



$$+\curvearrowleft \Sigma M_c = 0 \quad M_c - 1041.35(22.2 \times 10^{-3})$$

$$M_c = 231Nm$$

Cut 2



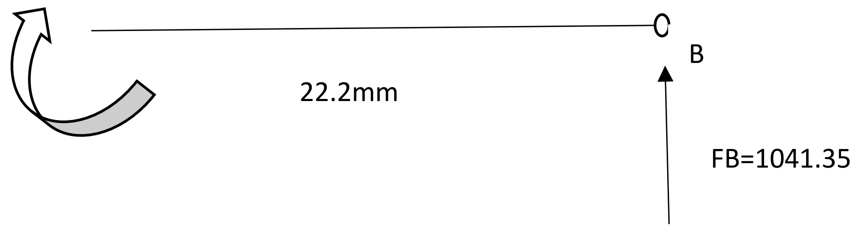
1041.3↿ 5

$$+\circlearrowleft \Sigma M_D = 0$$

$$M_D - 1041.35(22.2 + 2.8) = 0$$

$$M_D - 26.03Nm$$

Cut 3



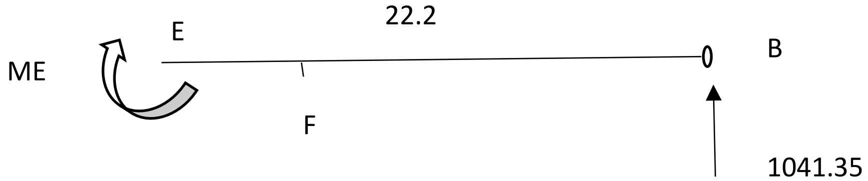
$$+\circlearrowleft \Sigma M_F = 0$$

$$-M_F + 1041.35(22.2) = 0$$

$$-M_F = -23.1$$

$$M_F = 23.1Nm$$

Cut 4



$+\circlearrowleft \Sigma M_E = 0$

$$-M_E + 1041.35(25) = 0$$

$$M_E = 26.04Mp$$

$$M_{E(\max\;)} = 26.04Nm$$

T point C.

$$\sigma c = \frac{M_C Y_C}{I} = \frac{(23.1)\left(\frac{15.875}{2}\right)}{\frac{\pi}{64}(15.875 \times 10^{-3})^4}$$

(4)

$\sigma c = 58.8MPa$. This is also equal to stress at point.

$$\sigma_c = \sigma_F$$

$$\sigma_D = \frac{(26.03)\left(\frac{22 \times 10^{-3}}{2}\right)}{\frac{\pi}{64}(22 \times 10^{-3})^4}$$

$$\sigma_D = 24.9MPa$$

$$\sigma_D = \sigma_E$$

$$K_a = a(S_{ut})^b$$

$a = 1.58$ $b = -0.085$

$$K_a = 1.58(1040)$$

$$K_a = 0.0875$$

$K_b = 1.24\, d^{-0.107}$ $2.79 \leq 15.875 \leq 5$

$$K_b = 1.24\,(15.875)$$

$$K_b = 0.922$$

$$K_c = 1$$

$$K_d = 1$$

$$K_e = 1 - 0.08(1.645)$$

$=1 - 0.1316$

=0.8684

$$K_p = 0.8684$$

$$K_p = 1 + q(K_t - 1)$$

$$r = 3mm$$

$$Sut = 1040MPa = 1.04GPa$$

q=0.85

$$\frac{r}{d} = \frac{3}{15875} = 0.18$$

$$\frac{D}{d} = \frac{22}{15875} = 1.39$$

$$K_t = 1.85$$

$$K_f = 1 + 0.85(1.85 - 1)$$

=1.7225

$$S_c = K_a K_b K_c K_d K_e K_f S_{c^1}$$

= (0.0875)(0.922)(1)(1)(0.8684)(1)(520)

$$S_c = 364MPa$$

$$a = \frac{(f.Sut)^2}{S_c} = \frac{(0.78 \times 1040)^2}{364}$$

$$a = 1807MPa$$

$$b = -\frac{1}{3} \log\left(\frac{f.Sut}{S_c}\right)$$

=b-0116

$$S_f = Fatigal\ Strength\ = K_f.\ \sigma_{max}$$

= (1.7225)(58.8)

=101.283MPa

$$N = \left(\frac{101.283}{180.7}\right)^{\frac{-1}{(-0.116)}}$$

= $6.14 \times 10^{10}\ cycles$

Safety Factors for fatigue here will be n=Yield Strength, Fatigue strength

n=$\frac{760}{101} = 7.5$

The number of cycles is 6.14×10^{10} the client has asked for 5 years operation.

This shaft is stoating at a speed of 570RPM.that speed 2.9×10^8 cycles.

For 5 years of operation, total number of cycles are 1.4×10^9.

N calculated is 6.14×10^{10} that makes the shaft safe.

Intermediate and output shaft

The factors a,b are same for the output shaft as both shafts have same d and all the factor $K_a, K_b, K_c, K_d, K_e, K_p$ will be same .

As we knew th N required are 2.9×10^8

S_f can be calculated.

S_e=364MPa

Where for intermediate shaft with calculated of factors accordingly Se=364

Using the factor, a and b. both have cycles S relater than1×10^9 which depicts operation of 5 years.

Only the maximum changing near the which is for them.

Key design shaft

First Key design for input gear.

i)n shaft=570RPM, P=3044.7N, S_{ykey}=650MPa

Here we are taking our key as lower strength than our shaft.

$$l_{key \leq 1.5\, d\, shaft}$$

$$S_{sy_{key} \leq\, 0.577\left(S_{y_{key}}\right) = (0.577)(650)}$$

=375.05MPa

As we are taking a safety factor of 3.

$$6\ allowable\ = \frac{375}{3} = 125MPa$$

As we are doing a sheet analysis.

$$V = F_t = 2016.35N \Rightarrow \sigma_{all} = \frac{v}{A}$$

$$125 = \frac{2016.35}{t.L}$$

t.L=5.1308

As ou t =3mm

L=5.4mm, So the minimum length should be 5.4mm

Similarly, we do comprehension analysis.

$$\sigma allowable = \frac{S_{y_{key}}}{n} = \frac{650}{3} = 216MPa$$

$$\sigma all = \frac{F\ compression}{A\ compression} = 216MPa = \frac{2016.35N}{\left(\frac{t}{2}\right)(L)} \Rightarrow L = 7mm$$

<u>(1)</u>

So according to the analysis the key length should be mine than 7mm but less than diameter shaft.

ii)Shaft key for input shaft out of casing. The practises will be the same.

Ssy key=375 here just diameter at the end is smaller. But still if fulfills the conditions we can take a key greater than 7mm.

Key design shaft 2

<u>Gear 1 meshing with input gear:</u>

Ssy key=375 MPa⇒ $\sigma allowable = 125MPa$

These values are with safety factor of 3 in tension analysis.

Here $F_t = 2016.35N$ for the 1st gear.

When t=3mm, L=5.4mm

So, the key must have a length of 5.4mm for that gear. whereas L must be less than 33mm.

For compression analysis,$\sigma allowable = 216MPa$

$$\sigma m = 216 = \frac{2016.35}{\left(\frac{t}{2}\right)L} \Rightarrow L = 7mm$$

So, by combining both results, the key must length greater than 7.

Gear 2 meshing with output gear:

Here Ft=8697.7, So, sheet analysis follows as:

$$\sigma allowable \Rightarrow 125 = \frac{8697.7}{t.L}$$

L=23mm, where as $l_{key\ max} = 33mm$

For compression analysis:

$$\sigma allowable = 216MPa$$

216MPa=$\frac{8697.7}{\left(\frac{t}{2}\right)(L)} \Rightarrow L = 27mm$

$$27mm < L_{max}$$

1)So we can use Length greater than 27mm

Key design for Shaft 3(output shaft)

1)Gear key meshing with intermediate shaft tension analysis:

Tension analysis:

$$F_t = V = 8697.7N$$

$$Ssy_{ysey} = 375MPa =, \sigma all = 125MPa$$

$$\sigma allow = 725MPa = \frac{8697.7}{t.L}$$

$$t.L = 69.5 \Rightarrow L = \frac{69.5}{3}$$

L=23.2mm

Compression analysis:

$$\sigma allowable = 216MPa = \frac{8697.7}{\left(\frac{t}{2}\right)(L)}$$

$$L = \frac{8697.7}{(216)\frac{3}{2}} = 27mm$$

This analysis shows the key length must be more than 27 although less than 33mm.

<u>Output shaft</u>

Analysis key on the output shaft where coupling is to be done follow as:

Tension analysis:

$F_t = v = 8697.7N$

Ssy=375MPa, $\sigma allowable = 125MPa$

$$125MPa = \frac{8697.7}{t.L} \Rightarrow L = 23.2mm$$

Compression analysis:

$$\sigma allowable = 216MPa = \frac{8697.7}{\left(\frac{t}{2}\right)(L)}$$

L=27mm

This analysis shows the key length with 3mm t must be greater than 27mm length.

Maximum Gradient of the Cherry Picker

The Loading Condition followed by the constraints offer 913.7 Nm resistive Torque. The calculations are followed as below.

Total Torque = Torque produced by aerodynamic drag + Torque Produced by Gravitational Pull and climbing + Torque Produced by Friction

Total Torque = 396 +0.69 + 516.98

Total torque = 913.7 Nm

Now we have designed the Gearbox with gear ratio of 18.5 that can produce 943.5Nm of Torque. The maximum safe gradient will be in accordance with the maximum torque produced by the gearbox. The angle Gradient can be calculated from our Total Torque equation with respect to the Maximum Torque produced by the Gearbox.

943.5 = ((3822CosØ)0.35)0.4 +(3828SinØ)0.4 + (1.15 SinØ)0.3

Solving the above Equation, the maximum safe gradient is 17 Degree.

The width of the cherry picker keeping the two gearbox, wheels and motors in consideration is 1.5 Meters. If the Clearance while turning is adjusted to 0.5 Meters between both sides the cherry picker will require approximately 2.5 Meters which is sufficient as the spaces between rows is 5 Meters.

Reliability and Safety

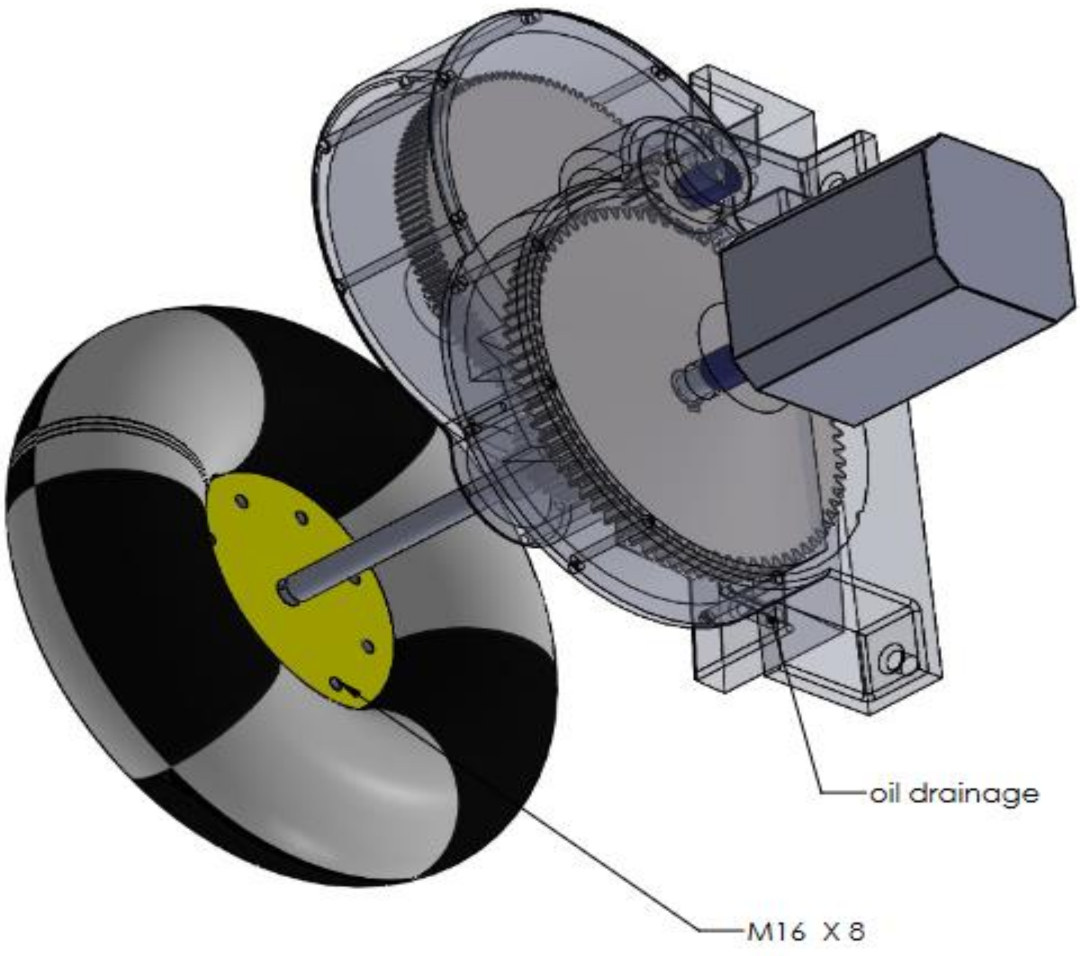
Safety factors in mechanical design are essential for ensuring the reliability and integrity of engineering structure and bodies. They give us a margin of safety that accounts for uncertainties, variations, and unexpected conditions in the real world. The primary purpose of safety factors is to prevent severe failure leading to a catastrophe. According to the website "Engineering Toolbox", the range of 1.3 to 1.5 is for highly reliable material and loading conditions are not severe and weight is important consideration. In the band of 1.5-2, the materials are just reliable and 2-2.5 for ordinary materials (Toolbox, 2010). The higher the factor, the safer for its material properties and more prone to external conditions.

We have taken our safety factor at 3.0 even though we are using stainless steel but it is a high torque gearbox and we aim for a maximum 8-9 years of work life. Instead of having sharp cuts in gear teeth's, we are using circular pitch and filleted radius at the bottom of teeth to prevent fracture. Considering the material and our safety factor of 3 in calculation increases the reliability to maximum.

Maintenance and Serviceability

One of the advantages to making a simple design so that we can easily identify any difference in physical sign such as noise, low speed or structural damage. Repairing any gear or shaft will be easier to disassemble as the housing is simple. There is a hole for refilling gear oil and drain to change the used gear oil for increasing life of gear and reducing friction between the teeth of gear. We can expect a 100$ annual expense for maintenance of gear box and a maximum cost of 500$ for repair of the gear box (My Car, 2023).

Figure 9 Gear Box view with oil drain location



Efficiency and Power Transmission

Spur gears are standard gearbox with basic arrangement. Having a simple arrangement makes it more efficient to transfer power with minimum loss. Our calculation shows the output power is 3Kw which shows the setup is 90% efficient. Secondly the output speed is maximum 5 kmph which makes the gears more reliable (Pty, 2015).

ANSYS Simulation

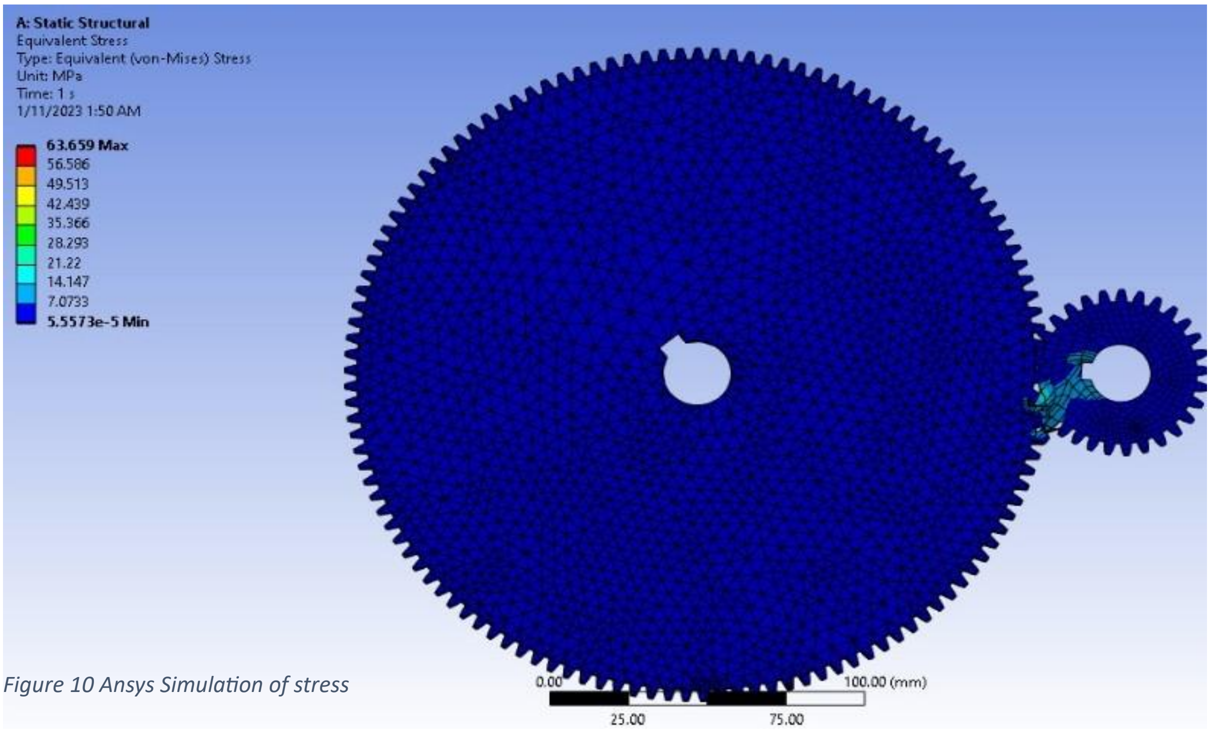


Figure 10 Ansys Simulation of stress

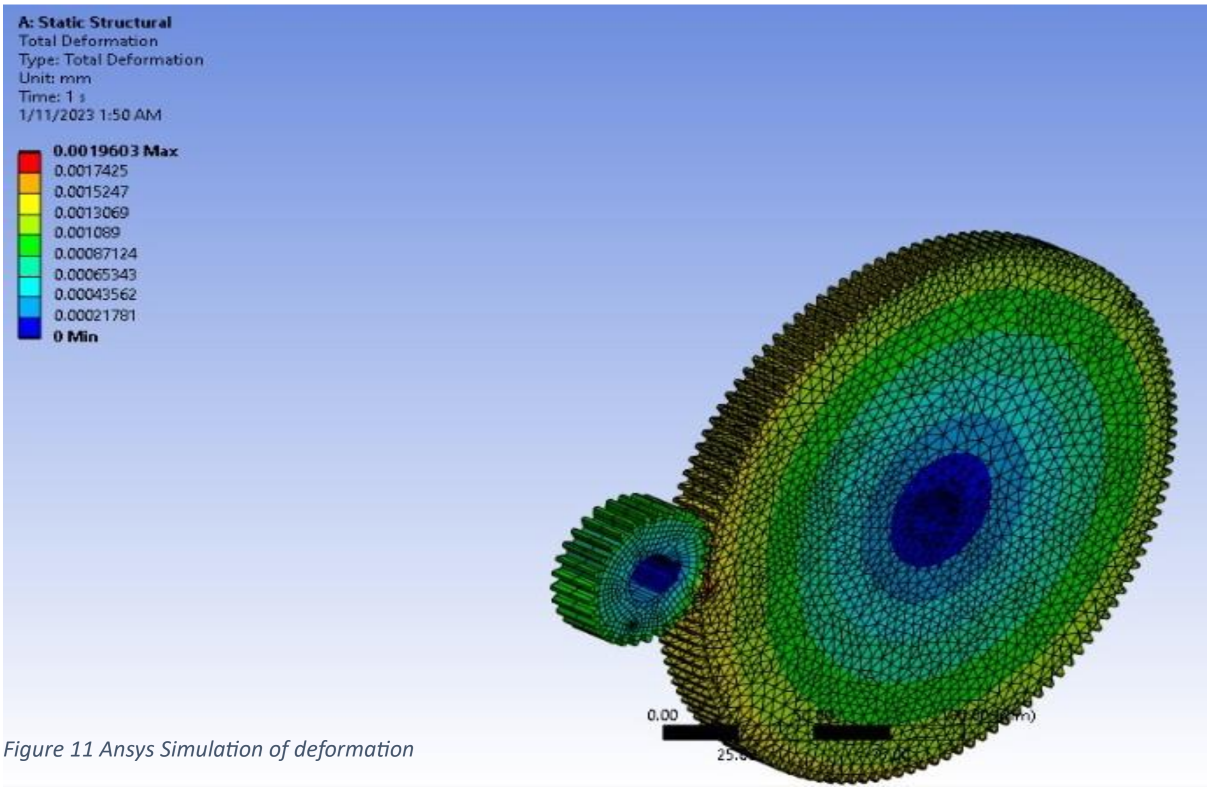


Figure 11 Ansys Simulation of deformation

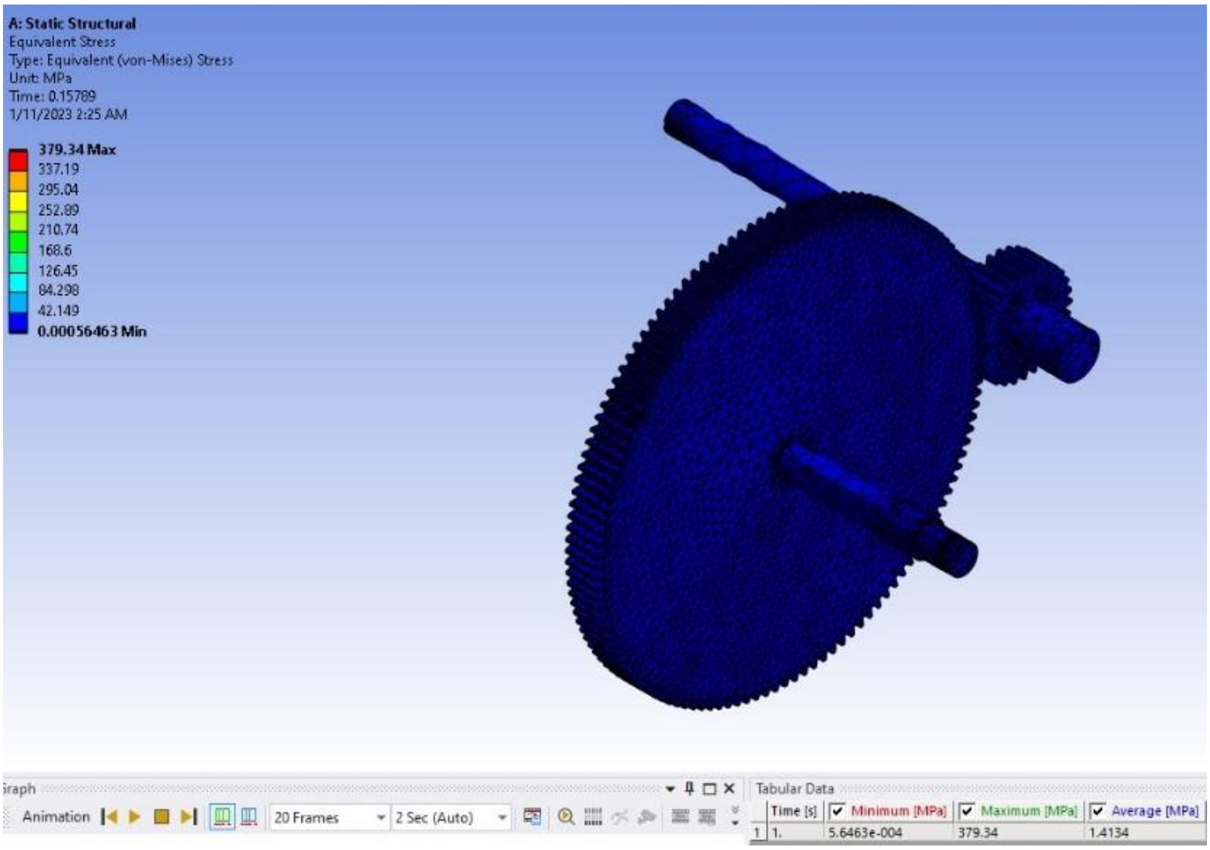


Figure 12 Equivalent stress with all gear and shaft assembly

Conclusion

In summary, the thorough mathematical analysis in this research clarifies the coaxially aligned gearbox, which is a crucial part of cherry pickers. This vital piece of equipment is crucial to maintaining effective and safe operations at high altitudes. By means of meticulous mathematical study, we have acquired significant understanding regarding the design and operation of the gearbox.This study's gear calculations offer a clear grasp of the gear ratio, pitch diameters, and face width—all crucial for the best possible energy transfer and operational efficiency. We have verified the choice of bearings for distinct shafts by performing bearing calculations and evaluating forces and moments, therefore guaranteeing that the gearbox is prepared to meet the needs of diverse operational scenarios.

A layer of robustness and improved dependability is added to the design through consideration of gear sets, considering possible fluctuations in load and operating conditions. Furthermore, the investigation of heat treatment and heat treatment factors raises the gearbox's endurance limit and lengthens its lifespan. The conclusions of this study provide manufacturers, engineers, and other stakeholders with important direction regarding the design and upkeep of cherry pickers. This analysis supports the ongoing advancement of cherry picker technology by guaranteeing compatibility with operational requirements, thereby improving safety, efficiency, and dependability for a range of industries. Our ANSYS simulation shows the stress and deformation is in suitable level.

As cherry pickers are still essential for tasks that need access to high places, our work here highlights how important it is to do precise mathematical analysis to maximise their efficiency. The findings in this report will have a big impact on how cherry picker technology develops going forwards and, in turn, how safe and effective their operations are for the industries that depend on them.

References

My Car. (2023). *Our Gearbox Repair will stop that crunch.* Retrieved from My Car: https://www.mycar.com.au/repairs/gearbox-repair

Pty, M. D. (2015). *Which Gearbox?* Retrieved from Motion Dynamics: https://www.motiondynamics.com.au/gearboxes-which-type.html

SKF. (2023). *Single row tapered roller bearings.* Retrieved from SKF: https://www.skf.com/au/products/rolling-bearings/roller-bearings/tapered-roller-bearings/single-row-tapered-roller-bearings

Toolbox, T. E. (2010). *Factors of Safety - FOS* . Retrieved from The Engineering Toolbox: https://www.engineeringtoolbox.com/factors-safety-fos-d_1624.html