Machine Design

MCG 3131

Lateral Offset, Non-Reversing

Gear Box Design Project

Group 26

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Abstract

This project report outlines the design and analysis of a lateral offset, non-reversing gearbox engineered to transform 5 horsepower and 4000 RPM input into a 300 RPM output, achieving the desired gear reduction while maintaining power. This design employs a two-stage reduction using three shafts, four helical gears, and six bearings, selected for their effectiveness in offsetting the output shaft and meeting reduction targets efficiently, safely, and feasibly.

Helical gears were chosen for their advantages in load distribution and quieter operation over spur or bevel gears, based on a decision matrix evaluating factors like noise, cost, and efficiency. The gears would be custom made by HPC based on the detailed material specifications outlined in report to ensure durability under operational loads.

The shafts, made from Normalized AISI 1015 and 1095 steel, are designed to withstand transmitted forces with minimal stress, validated through extensive SkyCiv software analysis. Bearings are crucial for handling radial and axial forces, with deep groove bearings selected for their performance and reliability, ensuring the gearbox meets operational and lifespan requirements.

Overall, the report presents a thorough design process for a gearbox, from gear, shaft, and bearing selection to material specification and safety considerations, illustrating a comprehensive approach to machine design.

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List of Acronyms and Glossary

*Table 1**:** Acronyms*

Acronym	Definition
RPM	Revolutions per minute
HP	Horsepower
SF	Safety factor
CAD	Computer-aided design
UBS	Ultimate Bending Stress

Table 2: Glossary

Term	Symbol	Definition
Addendum	a	The radial distance between the pitch circle and the outside circle of the gear.
Angular Velocity	ω	The rate of change of the angular position of a rotating body.
Base circle radius	rb	The radius of the base circle of the gear.
Base pitch	pb	The length of the arc between the starts of the involute curves on two adjacent teeth.
Bending Stress	σ_b	internal stress caused by a bending moment or external load that causes a material to bend
Circular pitch	p	Distance from the point that lies on the pitch circle of a tooth, to the same point on an adjacent tooth.
Contact ratio	CR	The measure of overlapping tooth action which is required for the gears to turn smoothly and in a continuous moving action.
Dedendum	d	The distance between the pitch circle and the base circle of a gear.
Dynamic factor	K_v	Accounts for the additional loads in gear teeth due to velocity and variations in gear meshing conditions.
Gear speed	v	Gear speed, typically measured in revolutions per minute (RPM), is the rate at which a gear rotates within a gear system.
Gear surface width	b	Width of the gear
Geometry factor	I	factor used in gear design that takes into account the effects of tooth geometry on the distribution of stress along the tooth profile.
Helix angle	Ψ	The helix angle of a gear is measured on the cylindrical pitch surface and commonly ranges between 15° and 30°, influencing both the end thrust and smooth operation of helical gears
Speed ratio	v	The ratio of the speed of the driving gear to the speed of the driven gear.
Torque	T	Rotational equivalent of linear force, measuring the tendency of a force to rotate an object about an axis.
Pitch diameter	d	Diameter of the pitch circle on a gear, where the teeth engage with another gear or a rack.
Number of teeth	N	Number of teeth of a given gear

Tooth thickness	t	The width of the gear tooth measured along the pitch circle.
Pressure angle	Φ	The angle between the line of action and the tangent to the pitch circle.
Pitch circle radius	r	The radius of the circle from the gear's center to the pitch point.
Transverse load	W_t	The load acting along the tooth width, perpendicular to the gear axis.
Radial load	W_r	The load applied in the radial direction of the gear, towards the center.
Axial load	W_a	The load that acts parallel to the axis of the gear.
Power	P	The rate of doing work or the rate of transfer of energy.
Gear geometry factor	J	A factor that considers the gear tooth's shape and geometry in strength calculations.
Overload factor	K_o	A factor accounting for variations in applied load beyond the nominal or average load.
Mounting correction factor	K_m	Adjusts the strength calculation for the effects of gear mounting and support rigidity.
Theoretical bending fatigue strength	σ_f	The stress at which a material is expected to fail due to repeated bending loads.
Load factor	K	A factor representing the effect of distribution and magnitude of the load on gear performance.
Gradient factor	C_g	Accounts for the effect of the stress gradient on the fatigue strength of the material.
Reliability factor	K_r	A multiplier that adjusts the bearing life rating to achieve the desired reliability level.
Temperature factor	K_t	adjusts the material strength for the effects of operating temperature.
Mean stress factor	K_{ms}	Consideration of the average stress in a fluctuating load scenario.
Theoretical fatigue strength	σ_n	The stress level at which a material is believed to fail after a certain number of cycles.
Life factor	C_{li}	Reflects the relationship between the actual bearing life and the rated life.
Bending fatigue strength	σ_{bf}	The capacity of a material to withstand repeated bending forces without failure.
Safety factor	S_f	The ratio of the material's breaking strength to the actual load or stress applied.
Spur gear tooth surface stress	σ_s	The stress experienced at the tooth surface of a spur gear.
Surface fatigue strength	σ_{fs}	The stress level a material can endure at its surface for many cycles without failing due to fatigue.

Ultimate strength	S_u	maximum stress a material can withstand before failure
Yield strength	S_y	stress at which a material begins to deform plastically.
Brinell hardness	HB	The Brinell Hardness number (HB) is related to the tensile strength of most steels by the equation Su=500HBSu=500HB

1 Introduction

The objective of this report is to present iterative steps, analysis and calculations that were used in the development of a gearbox given the criteria that was provided for the MCG 3131 course. The report highlights the steps taken in the iterative process to develop the gearbox, the calculations, and design considerations/assumptions made to complete the design. Furthermore, there are also sketches, and CAD models that were developed to create a finalized general appearance of the gearbox. For this report, the design criteria required the input power to be 5 hp, and the output rotation to be laterally offset and non-reversing. Furthermore, an input rotation of 4000 RPM was provided, and is expected to be reduced to 300 RPM on the output shaft.

The report begins by discussing the development of the gears, and how the final gears were selected and calculated, going into detail on the selection process using the decisional matrix, as well as the calculations that helped iterate the design and finalize the selection. Once the gears were selected, it then became possible to work on the shaft analysis, using the forces found in the gear analysis to develop shafts robust enough to sustain the loads in bending and in fatigue. Using SkyCiv [1], an engineering online shaft analysis software, it was possible to determine the design specifications required to support the bending, torsional and axial loads provided by the gears. Lastly, using the shaft analysis calculations, and the geometry of the shaft it was then possible to

determine the required shaft sizes that should be used to support the gear forces adequately without failure.

The configuration of the gearbox used in this report is a simple two stage reducer that uses three shafts, four gears, and six bearings. In this way it is possible to laterally offset the output shaft, whilst also attaining the correct reduction in angular velocity required as per the criteria. The choice to use three shafts and two sets of meshed gears for the reducer was made to lower the concentrated forces on the shafts, by diminishing the size of the ratio required to attain the correct reduction. In theory it is possible to use a single gear reduction to attain the correct ratio, though doing so would prove to give poor factors of safety, whilst increasing the overall cost to manufacture such gears and shafts that could support the loads and would not result in a non-reversing design. Since the design uses three shafts, it was then only a matter of supporting the shafts adequately with each shaft having two bearings straddle mounting the gears.

It is important to note that the design process is iterative and requires multiple calculations and analyses of the gears, shafts, and bearings. Throughout the entire design process the textbook Fundamentals of Machine Component Design by Robert C. Juvinal and Kurt M. Marshek [2], was used to determine the equations, constants, and design parameters required to solve for the factors of safety, forces, stresses, etc.

2 Background and Context

Gears are an essential machine component in the transfer of rotational and linear power in mechanics. They are toothed devices machined to mesh with one another permitting a highly efficient method of transferring rotational energy, whilst modifying the torque and angular velocity. Gears are machine elements designed to fit on shafts, though they can also be fashioned

to displace linearly, such as the rack in a rack and pinion assembly. There exist multiple diverse types of gears, and methods for machining gears that give them versatile utilities in the design of power transmissions; some such examples are spur gears, helical gears, bevel gears, worm gears, racks, etc. Using these gears and the several ways to assemble them makes it is possible to create gear boxes or reducers; gear boxes are devices with two or more shafts that act to transmit an input power on an input shaft to an output shaft. Gear boxes use gears, shafts, and bearings in various unique assemblies to properly satisfy the criteria of a particular problem. Some general assemblies for gear boxes are reducers, differentials, inline shafts, offset shafts, back drivable gear boxes, multipliers, etc. In fact, one of the first know gear assemblies called the Antikythera mechanism dating back to 150 – 100 BC was used as an analog computer to calculate the position of 5 planets, the sun, the moon and even eclipses [3].

3 Design Analysis

The objective of the gearbox design is to facilitate a decrease in rotational speed from an input of 4000 rpm to an output of 300 rpm, effectively increasing the output shaft's torque relative to the input shaft's torque. This design incorporates a non-reversing gearbox, ensuring that the input and output shafts rotate in the same direction. For ideas on how to design this gearbox, several hand sketches were developed, as seen in Appendix II and III, and the best solution was discussed and rated. To accomplish the required speed reduction, the design employs a dual or two-step reduction process, utilizing three parallel shafts and four helical gears. This configuration is known as a lateral offset speed reducer.

3.1 Gear Design

The chosen gearbox design is configured with four helical gears and is specified as a lateral offset speed reducer. In accordance with the design requirements, Gear 1 operates at a speed of 4000 rpm, while Gear 4 delivers an output speed of 300 rpm. The chosen configuration uses a three-shaft design, with two sets of gears acting as reducers. This ensures that the gearbox is non-reversing, with both the input and output rotating in the same direction, adhering to the specified criterion.

Helical gears were chosen instead of spur gears or bevel gears as they help simplify the design whilst still supporting relatively large loads at low speeds. Furthermore, the helical gears gain the benefit of minimizing the sound produced whilst increasing how smooth the gears transition from one tooth to the next. A bevel gear set-up could have been used, but the usage of such gears would only complicate the design of the shafts and bearings. If the bevel gears were used instead, the mounting rigidity could diminish, whilst also increasing the complexity of the design required to mount each shaft properly. Moreover, the usage of strain waves, cycloidal drives and worm gears were generally not considered, as the reduction obtained would be far too great for the design criteria, and the additional complexity required to obtain an offset non-reversing output would be much too complicated compared to the helical gear design.

Note that in the following table, reliability is taken as a measure of how well the gear can support a 5 HP motor, and the equivalent combinations of torque and angular velocity. This encompasses the concepts of backlash and gear inertia.

Table 3: Gear selection decisional matrix.

Type of Gear	Noise	Cost	Reliability	Complexity	Difficulty to Manufacture	Size	Efficiency	Total
Spur	3	4	3	5	4	3	5	27
Herringbone	5	1	5	5	2	4	5	27
Helical	5	3	5	4	3	4	5	29
Spiral Bevel	5	2	5	1	2	4	5	24
Straight tooth bevel	3	3	4	1	3	3	5	22
Worm	5	3	4	1	2	5	1	21

Given that the helical gear offers the best design specifications according to the decisional matrix, the helical gears were selected for the design of this gear box.

3.1.1 Gear Specifications

Input Speed and Output Speed:

$$\omega(\text{rad/s}) = \omega(\text{rpm}) \times 2\pi/60$$

$$\omega 1 = 4000 \text{ rpm or } \omega 1 = 4000 \text{ x } (2\pi/60) = 418.879 \text{ rad/s}$$

$$\omega 4 = 300 \text{ rpm or } \omega 4 = 300 \text{ x } (2\pi/60) = 31.416 \text{ rad/s}$$

Input and output Torque:

$$T = \frac{(63025) \times P(HP)}{\omega(RPM)}$$

Equation 1: Calculate the torque in lb-in given the power in horsepower and the angular velocity in RPM.

In *Equation 1*, the T is the torque in Newton-meters (lb-in), the P is the power in horsepower, and the ω is the angular speed revolutions per minute (RPM).

Example calculation of the Torque on the first pinion and the last gear.

$$T_1 = \frac{63025 \times 5}{4000} = 78.78\ lb \cdot in$$

$$T_4 = \frac{63025 \times 5}{300} = 1050.41\ lb \cdot in$$

Gear Train Value e:

$$e = \frac{n_0}{n_i}$$

Equation 2: Calculation of the gear train value given the input and output RPM of the gearbox.

In *Equation 2*, e stands for the gear train value, n_0 is the output RPM of the shaft, and n_i is the input RPM of the shaft.

Calculation of the gear train value given the input and out RPM of the reducer.

$$e = \frac{300}{4000} = 0.075$$

Number of teeth:

Selecting, N1 = 17 teeth.

We do not want two interacting gears to be a multiple, therefore select N2 = 71 teeth

Gear ratio N1, N2 = 71/17 = 4.176

$$\omega_1 = 4000 \text{ Rpm} \rightarrow \frac{\omega_1}{\omega_2} = 4.176 \rightarrow \omega_2 = 957.85 \text{Rpm}$$

Since gear 2 and 3 are on the same shaft, gear 2 and 3 have the same rpm.

Since the rpm at Gear 4 needs to be 300, the gear ratio needs to be:

$$\frac{\omega_3}{\omega_4} = 957.85/300 = 3.19$$

Selecting, N3 = 16 teeth

$$\frac{N_4}{N_3} = 3.19 \rightarrow N_4 = 51 \text{ for design consideration we will make } N_4 = 49$$

To make sure our value of N_4 is coherent:

$$e = (-1)^k \frac{product\ of\ driving}{product\ of\ dirven} = (-1)^k \frac{17 * 16}{71 * 49} = 0.07818$$

Gear selection:

All the following gears were selected from HPC Catalogue Pg. 666 [4],[5].

1. SH16-17- Custom Non-Standard Technical Request as per Table 4.
2. SH16-71 – Custom Non-Standard Technical Request as per Table 4.
3. SH24-16 – Custom Non-Standard Technical Request as per Table 4.
4. SH24-49 – Custom Non-Standard Technical Request as per Table 4.

Table 4: Properties of the gears.

Gear Number	**1**	**2**	**3**	**4**
HPC catalogue part #	SH16-17	SH16-71	SH24-16	SH24-47
Number of teeth (N)	17.000	71.000	16.000	47.000
Pitch (p) (inches)	0.3142	0.3142	0.3927	0.3927
Pitch diameter (d) (inches)	1.7000	7.1000	2.0000	6.125
Pitch radius (r) (inches)	0.8500	3.5500	1.0000	2.9375
Pressure angle (Φ) (°)	20	20	20	20
Helix angle (Ψ) (°)	20	20	20	20
Maximum bore diameter (inches)	1.0422	6.4422	1.1778	5.3028
Boss diameter (inches)	1.6250	2.000	0.521	0.750
Max Face width (b) (inches)	1.4	1.4	1.750	1.750
Diametral pitch (P)	10	10	8	8
Normal diametral pitch (Pn)	10.6418	10.6418	8.5134	8.5134
Normal circular pitch	0.2952	0.2952	0.3690	0.3690
Addendum (a) (inches)	0.0940	0.0940	0.1175	0.1175

Addendum circular radius (inches)	0.9440	3.6440	1.1175	3.1800
Dedendum (b) (inches)	0.1175	0.1175	0.1468	0.1468
Dedendum Circular Radius (inches)	0.7325	3.4325	0.8532	2.9157
Base Circle Radius (rb) (inches)	0.7987	3.3359	0.9397	2.8778
Base Circle diameter (db) (inches)	1.5974	6.6718	1.8794	5.7556
Transverse circular pitch (Pt) (inches)	0.3142	0.3142	0.3927	0.3927
Axial pitch (Pa) (inches)	0.8631	0.8631	1.0789	1.0789
Normal Circular Pitch (inches)	0.2952	0.2952	0.3690	0.3927
maximum addendum	1.7037	3.659	1.642	3.19568
angular velocity	4000	957.746	957.746	312.7335

Sample calculations for gear 1:

Normal Circular Pitch (Pn):

$$P_n = P \times cos(\Psi)$$

Equation 3: Calculation of the normal diametral pitch given the helix angle and the diametral pitch.

$$P_n = P_1 \times \cos(\Psi) = 0.3142 \times \cos(20) = 0.2952\ teeth/inches$$

Normal Diametral Pitch (P_{n1}):

$$P_{n1} = \frac{P}{\cos(\Psi)}$$

Where P is the diametral pitch and Ψ is the helix angle.

$$P_{n1} = \frac{10}{\cos(20)} = 10.6418$$

Addendum (a):

$$a = 1/P_n$$

Where P_n is the normal diametral pitch in direction of rotation.

$$a = \frac{1}{10.6418} = 0.0940\ inches$$

Addendum Circular Radius:

$$Addendum\ Circular\ radius = a + r$$

Where a is the addendum and r is the pitch radius.

$$radius = 0.094 + 0.85 = 0.944\ in$$

Dedendum (b):

$$b = 1.25 \times a$$

Where a is the addendum

$$b = 1.25 \times 0.0940\ =\ 0.1175\ inches$$

Dedendum Circle Radius:

$$Dedendum\ Circular\ radius = r - b$$

Where b is the dedendum and r is the pitch radius.

$$radius = 0.85 - 0.1175 = 0.7325\ in$$

Base circle radius:

$$Base\ circle\ raidus = \frac{d}{2} \times \cos(\emptyset)$$

Where d is the pitch diameter and $\emptyset$ is the pressure angle.

$$radius = \frac{1.7}{2} \times \cos(20) = 0.7987\ in$$

Transverse circular pitch (P_t):

$$P_t = \frac{P_n}{\cos(\Psi)}$$

Where P_n is the normal circular pitch and Ψ is the helix angle.

$$P_t = \frac{0.2952}{\cos(20)} = 0.3142$$

Axial Pitch (Pa):

$$P_a = \frac{p}{\tan(\Psi)}$$

Where p is the pitch and Ψ is the helix angle.

$$P_a = \frac{0.3142}{\tan(20)} = 0.8631$$

3.1.2 Gear Force Analysis

The design incorporates four helical gears, each with a pressure angle (Φ) of 20° and a helix angle (Ψ) of 20°. During meshing, these gears experience three primary types of forces: Tangential Force, Axial Force, and Radial Force. It is important to note that the calculations for the forces and pitch line velocities for a pair of meshing gears should be identical, as they are in equilibrium with each other. It would not be logical for the forces to not be balanced, or for a gears pitch velocity to be higher than that of the other gear pitch velocities.

1. Pitch Line Velocity (V)

$$V = \pi(d)(n)(\frac{1}{12})$$

In this equation d represents the pitch diameter in inches, n represents the gear rotational speed in rpm.

The pitch line velocity is the speed at which the gear moves at the pitch line, this is an important velocity as it will allow for the calculation of the tangential forces on the gears.

Sample calculation: First pinion and first gear

$$V = \pi(1.7)(4000)\left(\frac{1}{12}\right) = 1780.234\ ft/min$$

2. Tangential Force (F_t)

$$F_t = 33\ 000\frac{W}{V}\ (eq: 16.7)$$

In the equation above, W is the power in hp, V is the pitch line velocity in ft/min and F_t is the tangential force in (lbs). The tangential force is the first force that is calculated and is the force that will push the gears in the appropriate direction of motion. Using the gear equations found in the textbook [2], it is then possible to find the radial and axial forces using the tangential force as a starting point.

Sample calculations: First pinion and first gear

$$F_t = 33\ 000\frac{5}{1780.234} = 92.684\ lbf$$

3. Axial Force (F_a)

$$F_a = F_t \tan(\Psi)\ (eq: 16.9)$$

In the equation above, F_a is the axial force in lb-f, F_t is the tangential force in lbs and Ψ is the helix angle in degrees. The axial force is the force that applies perpendicularly to the plane of rotation of the gear. This force doesn't appear in the

calculations for spur gears, and other straight toothed gears, and hence requires specific design considerations when selecting bearings.

Sample calculations: Gear 1 & 2

$$F_a = 92.684 \tan(20) = 33.73\ lb\ f$$

4. Radial Force (F_r)

$$F_r = F_t \tan(\Phi)\ (eq: 16.8)$$

In the equation above, F_r is the radial force in lbs and Φ is the pressure angle of the gears in degrees. The radial force is the force on the gear that applies straight onto the shaft through the gear. The radial force is mostly used in the calculations pertaining to the shaft analysis and the selection of appropriate bearings.

Sample calculations: First pinion and first gear

$$F_r = 92.684 \tan(20) = 33.73\ lb\ f$$

5. Total Force (F)

$$F = \frac{F_t}{\cos(\Psi)\cos(\Phi)}\ (eq: 16.11)$$

Sample calculations: First pinion and first gear

$$F = \frac{92.684}{\cos(20)\cos(20)} = \sqrt{33.73^2 + 33.73^2 + 92.684^2} = 104.962\ lb\ f$$

Table 5: Forces Acting on Gears

	Pinion Gear 1	**Gear 2**	**Pinon Gear 3**	**Gear 4**
Tangential Force (F_t) (Lb-f)	92.684	92.684	329.030	329.030
Radial Force (F_r) (Lb-f)	33.734	33.734	119.757	119.757
Axial Force (F_a) (Lb-f)	33.734	33.734	119.757	119.757
Total Force (F) (Lb-f)	104.963	104.963	372.618	372.618
Pitch Line Velocity (V) (ft/min)	1780.234	1780.234	501.474	501.474

The values calculated from table 4 will be very important in the selection of the material for each gear as well as in the wear and strength analysis. As seen in the table above, the forces experienced are much higher on gears 3 & 4. This is to be expected as there is a greater torque applied onto the last two gears in the reducer. As the forces on the last two gears will be the largest, they will undergo the largest bending stress and wear. Consequently, the third and fourth gears will need to have higher hardness if they want to have a high factor of safety despite the increased forces. It's also important to mention that the pinions will generally have a higher hardness than the larger accompanying gears, since it is often easier to manufacture a smaller gear, hence having a pinion with a higher hardness. Furthermore, this provides the benefit of increasing the life span of the pinion as it will have to undergo more load cycles than its equivalent paired gear.

3.1.3 Gear Material

Normalized steel was selected as the material for helical gears 1, 2, and 3 due to its more predictable properties, characterized by a uniform and fine-grained structure except for driven gear

2 in which an annealed AISI 1050 material was chosen. Choosing a normalized material for most of the gears offers cost advantages over annealed steel but also results in a harder and stronger metal compared to that produced by the annealing process. It is considered good practice to have similar Brinell hardness between two meshing gears to promote uniform wear between the gears. It was decided however to pick driving gears (pinions) with a higher hardness than the driven gears as the pinion gears have fewer teeth and would wear faster than the driven gears. The material specifications of the gears chosen for our reducer are shown in Table 6.

Table 6: Gear Material Specifications

	Pinion Gear 1	**Gear 2**	**Pinon Gear 3**	**Gear 4**
Material	AISI 1095 Normalized	AISI 1050 Annealed	ASIS 4340 Normalized	AISI 1095 Normalized
Tensile Strength (Su) (ksi)	147	108.5	185.5	185.5
Yield Strength (Sy) (ksi)	72.5	62	125	125
Brinell Hardness (BHN)	293	217	363	363

3.1.4 Gear Wear and Strength Analysis

Contact Ratio (*CR*)

$$CR = \frac{\sqrt{r_{ap}^2 - r_{bp}^2} + \sqrt{r_{ag}^2 - r_{bg}^2} - c\ sin\phi}{Pb} \quad Eq: 15.9(P.448)$$

In the equation above, CR is the contact ratio, r_{ap} & r_{ag} are the addendum radii of the mating pinion and gear, r_{ag}& r_{bg} are the base circle radii of the mating pinion and gear and Φ is the pressure angle of the gears in degrees.

To calculate CR, the values of r_{ap}, r_{ag}, C and Pb must first be calculated for pinion gear 1 and gear 2.

$$r_{ap} = r_p \ + a \ \ (P.447)$$

$$r_{ap} = 0.850 + 0.0940 = 0.9440\ in$$

$$r_{ag} = r_g \ + a \ \ (P.447)$$

$$r_{ag} = 3.550 + 0.0940 = \ 3.644\ in$$

$$C = r_p \ + r_g \quad eq\ 15.1a\ (P.441)$$

$$C = 0.85 + 3.55 = 4.4\ in$$

$$Pb = \frac{\pi}{10} cos(\Phi) \ \ from\ notes$$

$$Pb = \frac{\pi}{10} cos(20) = 0.2952\ in$$

Sample CR calculation: Gear 1 & 2 using above calculated values and $r_{bp\ \&}\ r_{bg}$ for table 3.

$$CR = \frac{\sqrt{(0.944)^2 - (3.644)^2} + \sqrt{(0.799)^2 - (3.336)^2} - (4.4)\ sin(20)}{0.2952} = 1.67$$

Geometric Factor (I)

$$I = \frac{sin\phi cos\phi}{2}\frac{R}{R+1}$$ Eq:15.23 (P.467)

In the equation above, I is the geometric factor, Φ is the pressure angle of the gears in degrees, and R is the ratio of the gear and pinion diameters R = dg/dp.

Sample calculation: Gear 1 & 2

$$I = \frac{sin\phi(20)\cos(20)}{2}\frac{4.176}{4.176+1} = \mathbf{0.129633}$$

Gear Tooth Surface Fatigue Analysis (σ_H)

$$\sigma_H = C_p\sqrt{\frac{F_t}{bd_pI}\left(\frac{cos\psi}{0{,}95\ CR}\right)K_vK_o(0.93K_m)}$$ Eq:16.13 (P.490)

Sample calculation: Gear 1 & 2 using the calculated values indicated below.

$$\sigma_H = 2300\sqrt{\frac{92.684}{(1.4)(1.7)(0.1296)}\left(\frac{\cos(20)}{0.95\ (1.6727)}\right)(1.5409)(1.5)(0.93)(1.3)}$$
$$= \mathbf{51245.23\ psi\ or\ 51.245\ ksi}$$

$$K_v = \frac{78+\sqrt{V}}{78}$$ P. 460 fig 15.24

In the equation above, K_v is the velocity factor using an AGMA quality control class B, high precision shaved & ground, gear V is the pitch line velocity calculated from Table 4.

Sample calculation: Gear 1 & 2

$$K_v = \frac{78 + \sqrt{1780.234}}{78} = \mathbf{1.54\ ft/min}$$

K_o = 1.5 using light shock from power source and a moderate shock from driven machinery obtained from P.461 Table 15.1.

K_m = 1.3 using a face with of 2 inches and accurate mountings, small bearing clearances, minimum deflection, and precision gears obtained from P.461 Table 15.2.

Cp = 2300$\sqrt{psi}$ using steel pinion and steel gear with E = 30,000 ksi obtained from P.468 Table 15.4a.

dp = 1.7, where dp is the pitch diameter in inches obtained from the first pinion gear in Table 3.

Surface endurance strength

$$S_H = S_{fe} C_{Li} C_R \quad \text{Eq:15.25 (P.469)}$$

In the equation above, S_H is the surface endurance strength S_{fe} is the surface fatigue strength for steel using equation found in table 15.5 (pg. 469) $S_{fe} = 0.4(Bhn) - 10ksi$, $C_{Li} = 1$ is the life factor at 99% reliability, and $C_R = 1$ is the reliability factor using a surface fatigue life cycle 10^7 , with a 99% reliability.

Sample calculation: Gear 1 & 2

$$S_H = (0.4(293) - 10ksi)(1)(0.8) = \mathbf{85.76\ ksi}$$

Safety Factor (n)

$$n = \frac{S_H}{\sigma_H}$$

Sample calculation: Gear 1 & 2 using found S_H and σ_H values.

$$n = \frac{85.76}{51.245} = \mathbf{1.6735}$$

Table 7: Properties and Calculated values for gear wear and strength analysis

Gear Number	**1**	**2**	**3**	**4**
Number of teeth (N)	17	71	16	49
Face width (b) (inches)	1.4	1.4	1.7	1.7
Tangential force (Ft) (lb f)	92.684	92.684	329.030	329.030
Radial force (Fr) (lb f)	33.734	33.734	119.757	119.757
Axial force (Fr) (lb f)	33.734	33.734	119.757	119.757
Pitch line velocity (V) (ft/min)	1780.234	1780.234	501.474	501.474
Brinell Harness of Gear (BHN)	293	217	363	363
Contact ratio (CR)	1.6727	1.6727	1.6129	1.6129
Gear tooth surface fatigue (σ_H) in ksi	51.245	51.245	76.663	76.663
Addendum (a)	0.0940	0.0940	0.1175	0.1175
Pitch radius (r_p)	0.85	-	1.000	-
Pitch radius (r_g)	-	3.55	-	3.065
Addendum radius of pinion (r_{ap}) (inches)	0.944	-	0.1175	-
Addendum radius of gear (r_{ag}) (inches)	-	3.644	-	3.18
Base circle radius of pinion (r_{bp}) (inches)	0.7987	-	0.9397	-
Base circle radius of gear (r_{bg}) (inches)	-	3.3359	-	2.8778
Center distance (C) (inches)	4.4000	4.4000	4.0625	4.0625
Base pitch (Pb) (inches)	0.2952	0.2952	0.369	0.369
Pressure angle (Φ) (degrees °)	20	20	20	20
Helix angle (Ψ) (degrees °)	20	20	20	20
Velocity factor (K_v)	1.54	1.54	1.287	1.287
Overload correction factor (K_o)	1.5	1.5	1.5	1.5
Mounting correction factor (K_m)	1.3	1.3	1.3	1.3
Elastic coefficient (Cp)	2300	2300	2300	2300
Geometry factor (I)	0.1296	0.1296	0.1211	0.1211
Diametral pitch (dp) (inches)	10	10	8	8
Surface endurance strength (S_H) (ksi)	85.76	61.44	108.16	108.16

Surface fatigue strength (S_{fe})	107.2	76.8	135.2	135.2
Life factor at 99% reliability (C_{Li})	1	1	1	1
Reliability factor with surface fatigue life cycle 10^7 with a 99% reliability (C_R)	1	1	1	1
Gear wear and strength safety factor (n)	1.67352	1.1989	1.4108	1.4108

Gear Tooth Surface Bending Strength analysis.

Bending stress(σ)

$$\sigma = \frac{F_t P}{bJ} K_v K_O (0.93 K_m) \; eq: 16.12$$

Ft = tangential force

Kv = Dynamic factor ,Fig 15.24 (P.460)

Ko = overload Factor ,Table 15.1 (P.461)

Km = mounting factor ,Table 15.2 (P.461)

J = gear geometry factor ,Fig 16.12 (P.489)

P = Diametral Pitch

B = Face width of gear

Table 8: Values needed to find bending stress.

Gear #	Ft (lb f)	Kv	Ko	Km	J	P (in)	B (in)
1	92.684	1.54	1.5	1.3	0.45	10	1.4
2	92.684	1.54	1.5	1.3	0.57	10	1.4
3	329.030	1.287	1.5	1.3	0.45	8	1.75
4	329.030	1.287	1.5	1.3	0.54	8	1.75

Sample calculation for Gear 1:

$$\sigma = \frac{(92.684)(10)}{(1.4)(0.45)} (1.54)(1.5)(0.93 * 1.3) = 4111.191 \text{ Psi}$$

Safety Factor (n)

$$S_n = S_n' C_L C_G C_S k_r k_t k_{ms} \; eq: 15.18$$

$$SF = \frac{Sn}{\sigma}$$

S_n '= Standard R.R Endurance limite = 0.5 x Su

C_L = Load Factor from table 8.1 (P.223)

C_G = Gradient factor from Table 8.1 (P.223)

C_S from Figure 8.13 (P.221)

Kr = reliability factor from Table 15.3 (P.462)

Kt= temperature factor from eq 15.19 (P.461)

Kms= mean stress factor from eq 15.19 (P.461)

Table 9: Values needed to find safety factor.

Gear #	**S_n '**	**C_L**	**C_G**	**C_S**	**Kr**	**Kt**	**Kms**
1	73 500	1	1	0.7	0.753	1	1.4
2	54 250	1	1	0.75	0.753	1	1.4
3	92 750	1	1	0.66	0.753	1	1.4
4	92 750	1	1	0.66	0.753	1	1.4

Sample calculation for Gear 1:

$$S_n = (30\ 750)(1)(1)(0.7)(0.753)(1)(1.4) = 22691.655$$

$$SF = \frac{54238.59}{4111.19} = 13.192$$

Table 10: Values calculated from strength analysis.

Gear	1	2	3	4
σ	4111.1915	3245.6775	7801.9586	6501.6322
Sn	54238.59	42892.7625	64532.853	64532.853
SF	13.1929	13.2153	8.2713	9.9256

3.2 Shaft Analysis

Drawing from the concepts covered in our course notes, it's understood that components like gears and belts apply axial, transverse, and torsional forces to the shaft. To minimize bending stress and enhance efficiency, a shaft's design should prioritize brevity, rigidity, and lightness. Minimizing gear mesh separation is crucial for limiting shaft deflection and slope. Furthermore, for machined shafts, the allowable torsional twist is restricted to less than 0.08 degrees per foot, as noted in our class materials on shafting. Additionally, any shaft step-ups should not exceed 20% increments and should incorporate a small fillet whenever possible. The design must also ensure that areas prone to stress concentration do not overlap with zones of high stress.

An analysis of all shaft components will be undertaken to assess their attachment methods in relation to thermal expansion and contraction, as well as the use of keys, splines, and setscrews. A force analysis will also be performed to examine the interaction between gears and shaft, the various forces exerted on the shaft, reaction forces, and the bearings required to counterbalance these forces.

3.2.1 Shaft Material Selection

From the gear analysis (table 10, force Analysis), it was determined that high torque will be present in the gear reducer especially between gear 3 and 4, therefore material selection for the shafts needs to be able to sustain these forces.

Table 11: Shaft Material Properties.

	Material	Tensile strength (S_u) (ksi)	Yield Strength (Sy) (ksi)	Hardness (BHN)
Shaft 1	Normalized AISI 1015	61.5	47.0	121
Shaft 2	Normalized AISI 1095	147.0	72.5	293
Shaft 3	Normalized AISI 1095	147.0	72.5	293

Shaft 1 Material (AISI 1015): Normalized AISI 1015 has a lower tensile strength (61.5 ksi) and yield strength (47.0 ksi) compared to AISI 1095. Given the forces on Gear 1 (tangential force 280.113 lb f, radial force 101.953 lb f, axial force 89.665 lb f), this material may be selected for its adequate strength and cost-effectiveness for less demanding applications, where high forces aren't present.

Shaft 2 Material (AISI 1095): This shaft has both Gear 2 and Gear 3 attached, experiencing forces from both gears. AISI 1095, with its high tensile (147.0 ksi) and yield strengths (72.5 ksi), is a robust choice to handle the combined forces from two gears, especially considering the high tangential force from Gear 3 (690.689 lb f). Its higher hardness (293 BHN) also suggests good wear resistance, important for shafts connected to multiple gears.

Shaft 3 Material (AISI 1095): The selection of AISI 1095 for Shaft 3, which is connected to Gear 4 undergoing the same forces as Gear 3, is consistent with Shaft 2's requirements. The high tensile and yield strength will provide durability and reliability under the high tangential load (690.689 lb f) experienced by Gear 4.

3.2.2 Shaft 1

Constructed from Normalized AISI 1015 steel with a machined finish, the input shaft is engineered with multiple stepped sections to facilitate the mounting of components. A square key secures the pinion gear (gear 1) onto the shaft, which is also supported by two fixed-floating bearings. The shaft begins with a ¾ -inch diameter at its start, designated for bearing A installation, secured by putting a step to prevent movement. The diameter transitions from ¾ inch to 1 inch in the first step, enhancing support for both the bearing and the pinion gear. Following this, the shaft narrows back to ¾ inch, where the pinion gear is affixed using a key. Proceeding further along the shaft you will see bearing B which will be mounted to the shaft with an interference fit. All shaft 1 analysis was conducted using SkyCiv [1] and can be seen in figures 1 to 10 and results are found in Table 12.

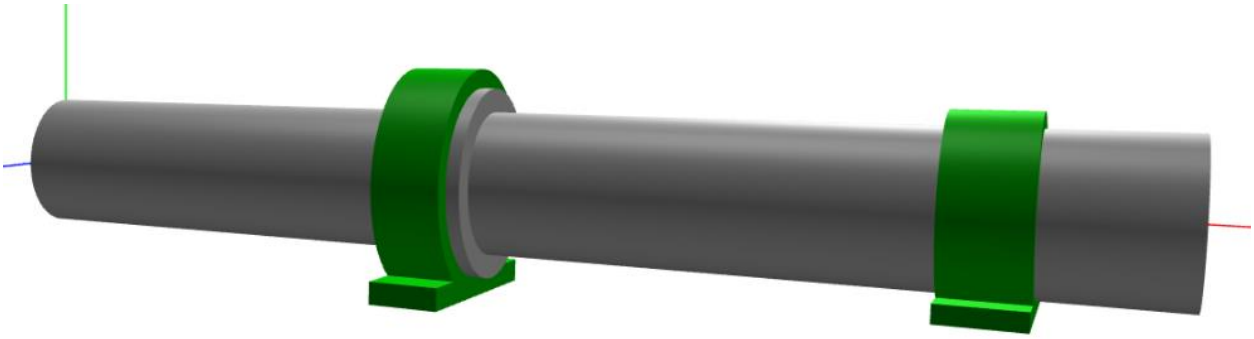


Figure 1. Input Shaft 1

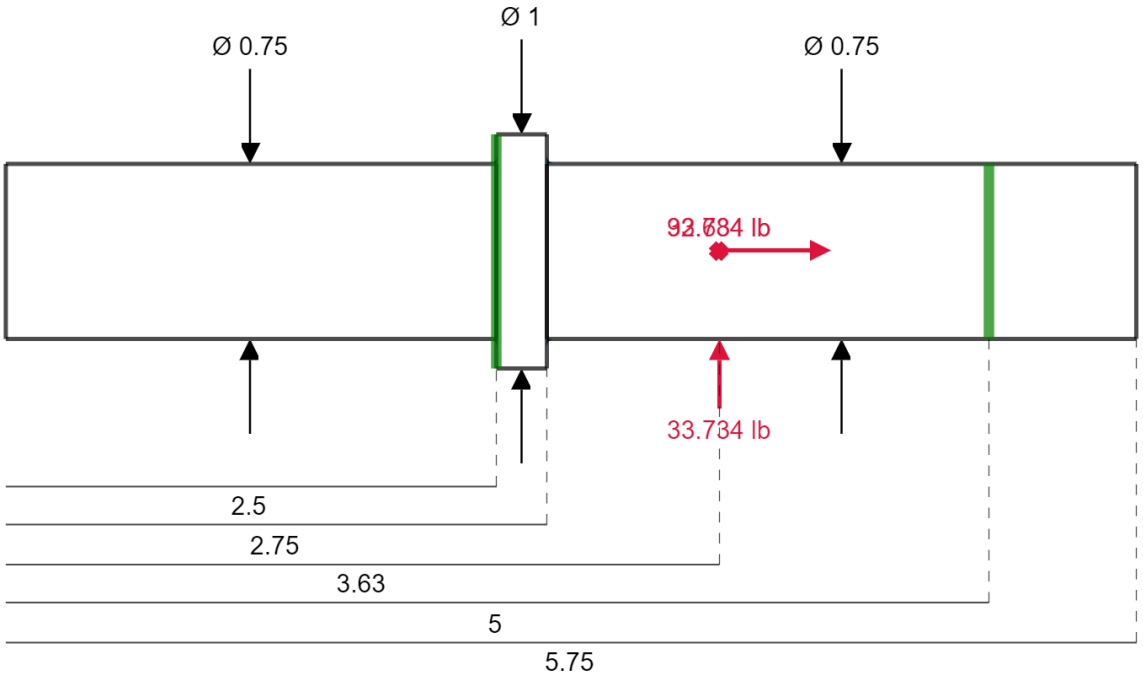


Figure 2. Input Shaft Design

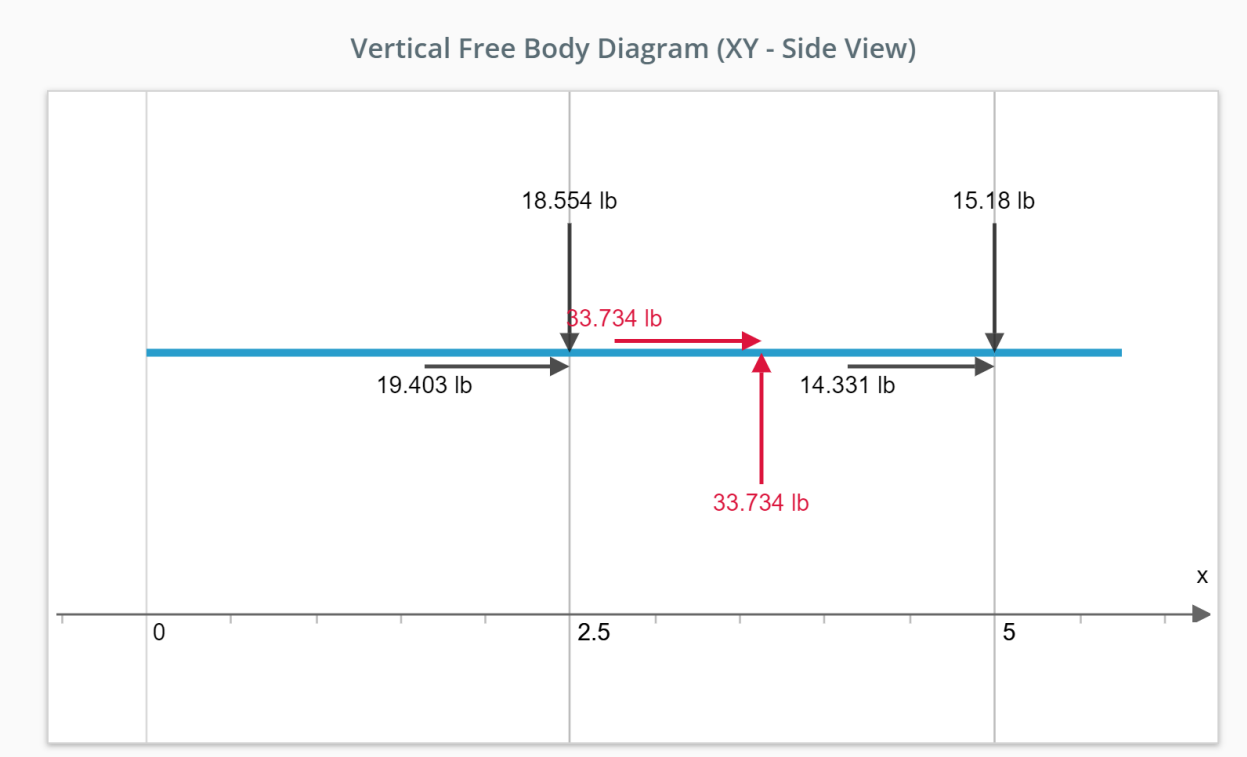


Figure 3. Free Body Diagram in XY Plane

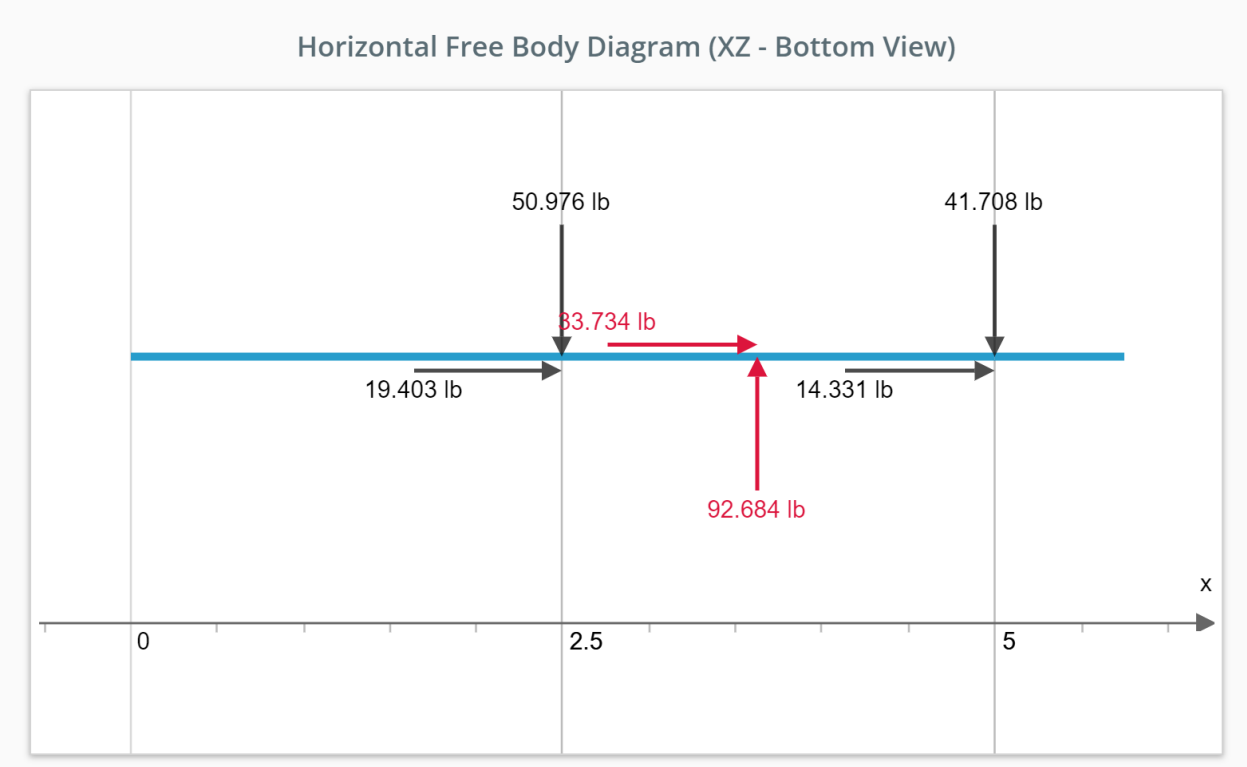


Figure 4. Free Body Diagram in XZ Plane

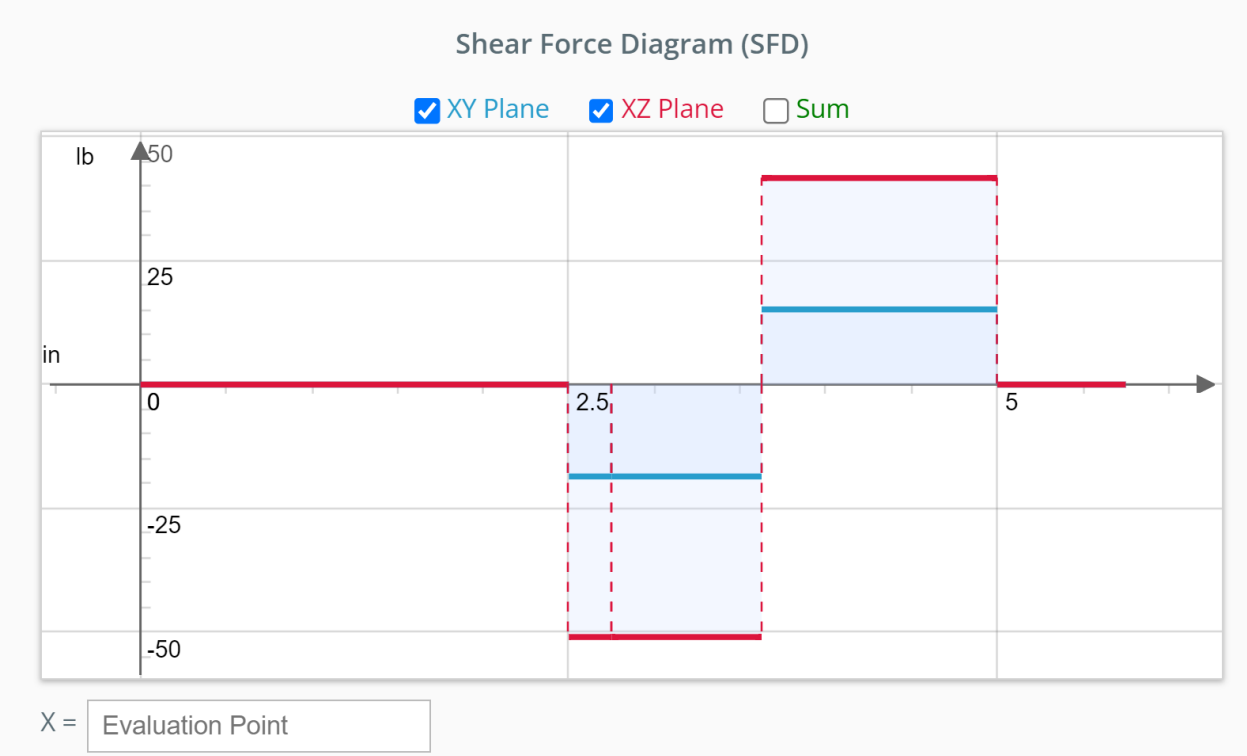


Figure 5. Shear Force Diagram

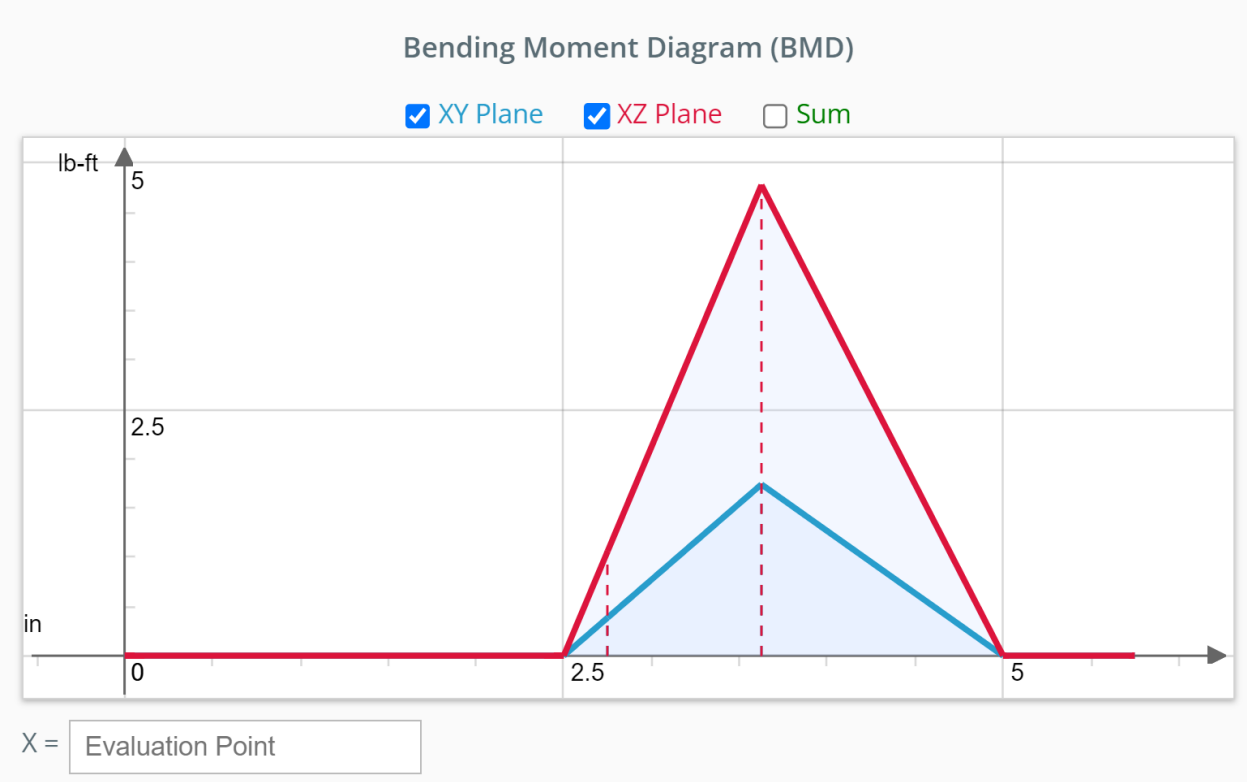


Figure 6. Bending Moment Diagram

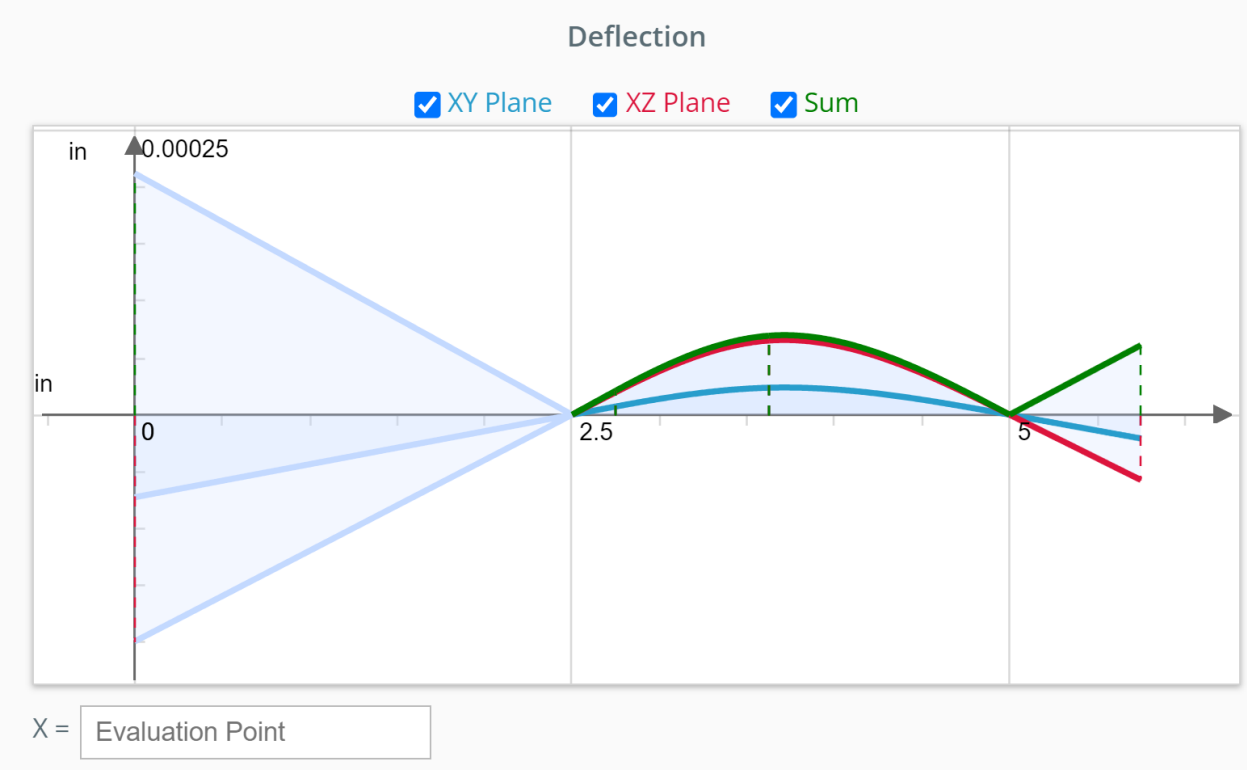


Figure 7. Deflection Diagram

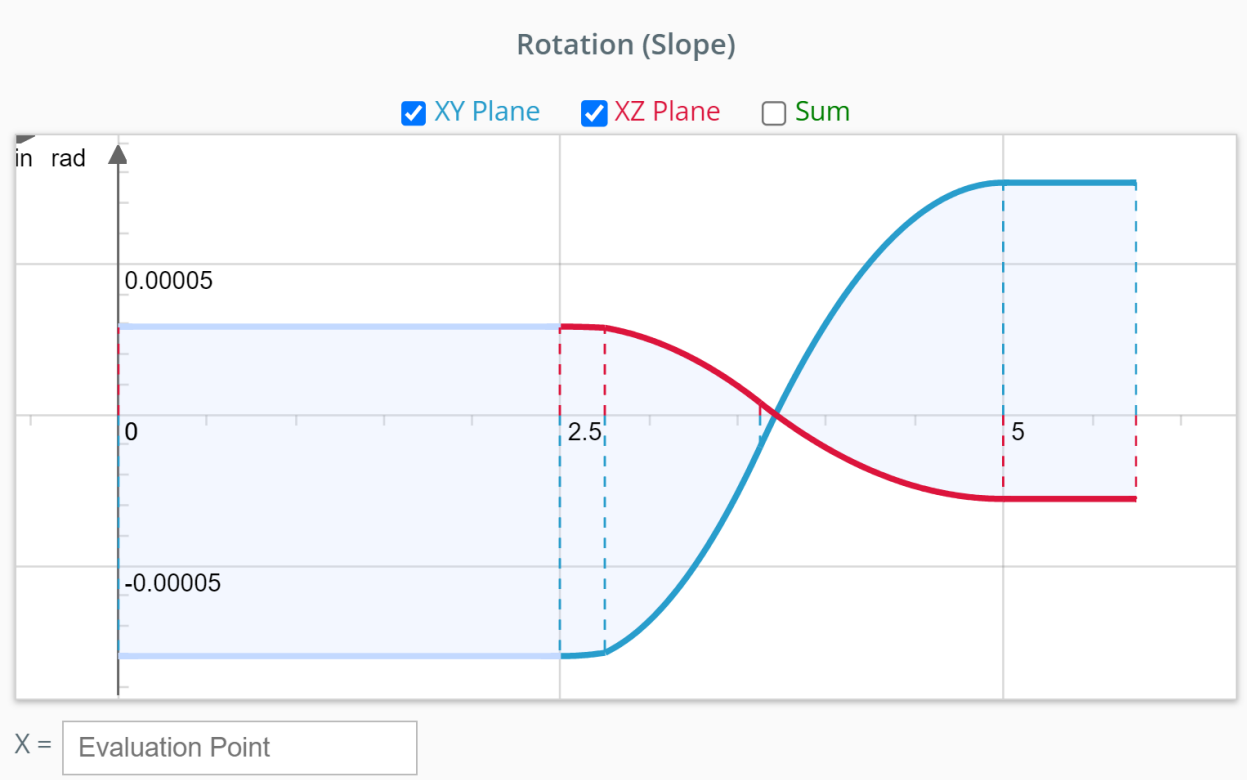


Figure 8. Rotation Diagram

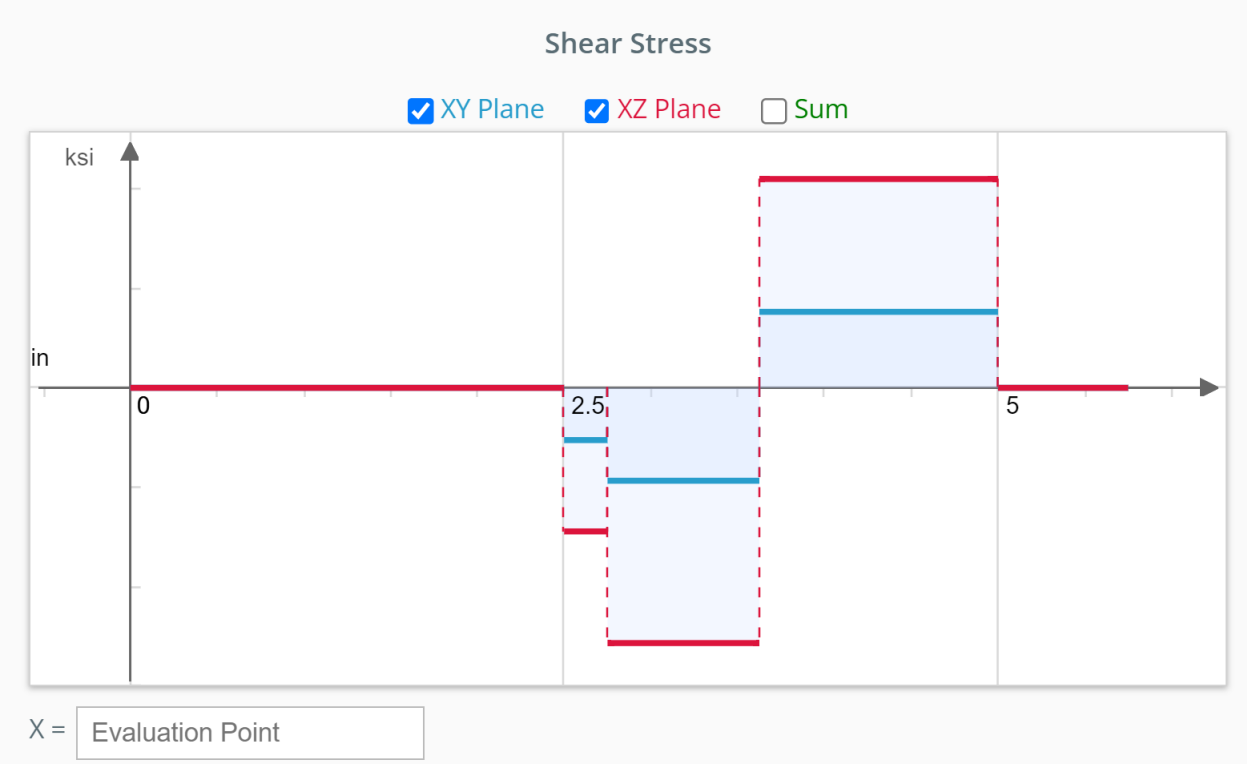


Figure 9. Shear Stress Diagram

Figure 10. Bending Stress Diagram

Table 12: Tabulated Results for shaft 1.

Shaft #1 Results	**Values**	**Unit of Measure**
Shaft Mass	0.76	lb
Critical Speed	38536.441	RPM
Maximum Power	0	HP
Maximum Bending Moment	5.086	lb-f
Maximum Shear Force	54.248	lb
Maximum Deflection	0.0	in
Maximum Rotation	0.0	rad
Maximum Bending Stress	1.474	ksi
Maximum Shear Stress	0.136	ksi
Maximum Torsion Stress	0	ksi
Maximum Axial Stress	0.076	ksi
Von Mises Stress	1.474	ksi

3.2.3 Shaft 2

Made from Normalized AISI 1095 steel, the intermediate shaft stands as the largest component in our assembly with a length of 6.25 inches, accommodating two gears and two bearings. In a manner akin to the input shaft, this shaft employs keys for gear attachment and a fixed-floating arrangement for the bearings. It begins at a diameter of ¾ inch, where bearing C is positioned against a step. To support the bearing and a gear, the shaft's diameter expands from ¾ inches to 1 inch. Subsequently, the diameter is reduced to ¾ inches to accommodate the installation of gear 2. Following the mounting of gear 2, the diameter decreases to ½ inch, aiding in the support of bearing D and for the installation of gear 3. All shaft 2 analysis was conducted using SkyCiv [1] and can be seen in figures 11 to 20 and results are found in Table 13.

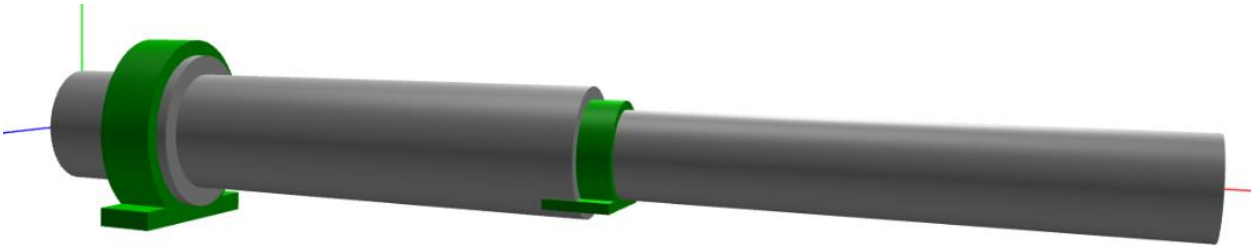


Figure 11. Intermediate Shaft 2

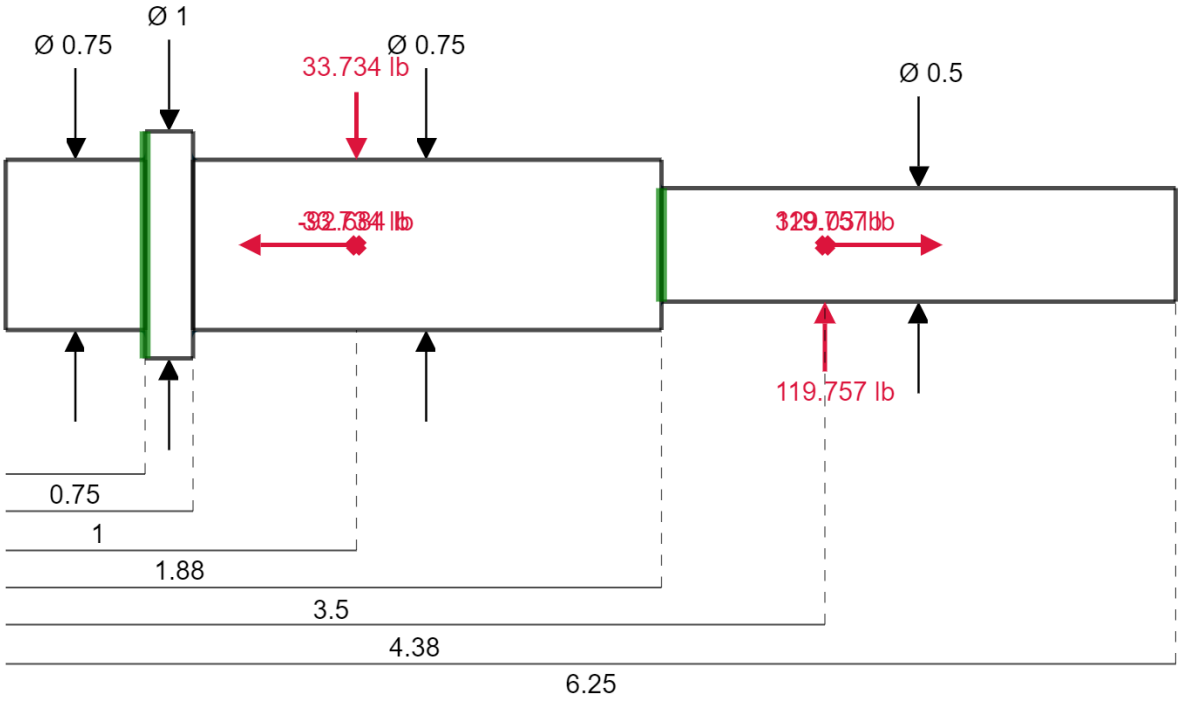


Figure 12. Shaft 2 Design of Shaft 2

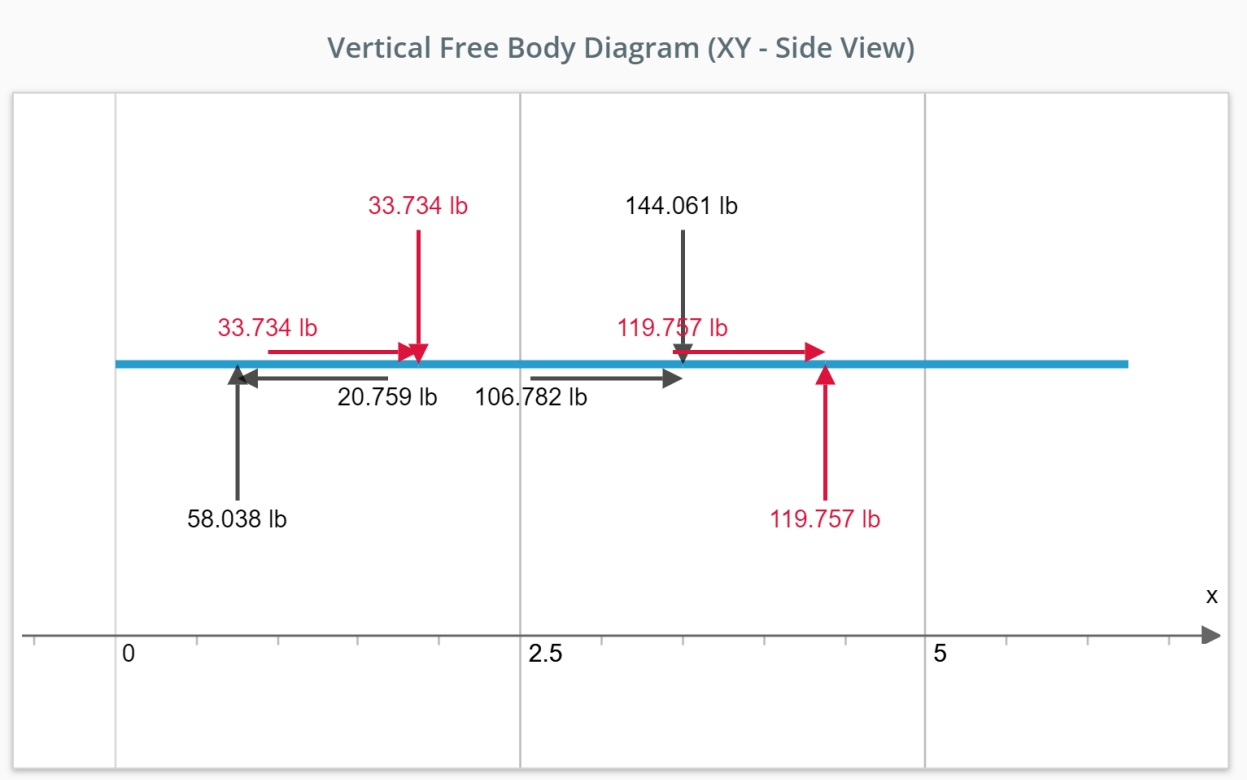


Figure 13. Free Body Diagram in XY Plane of Shaft 2

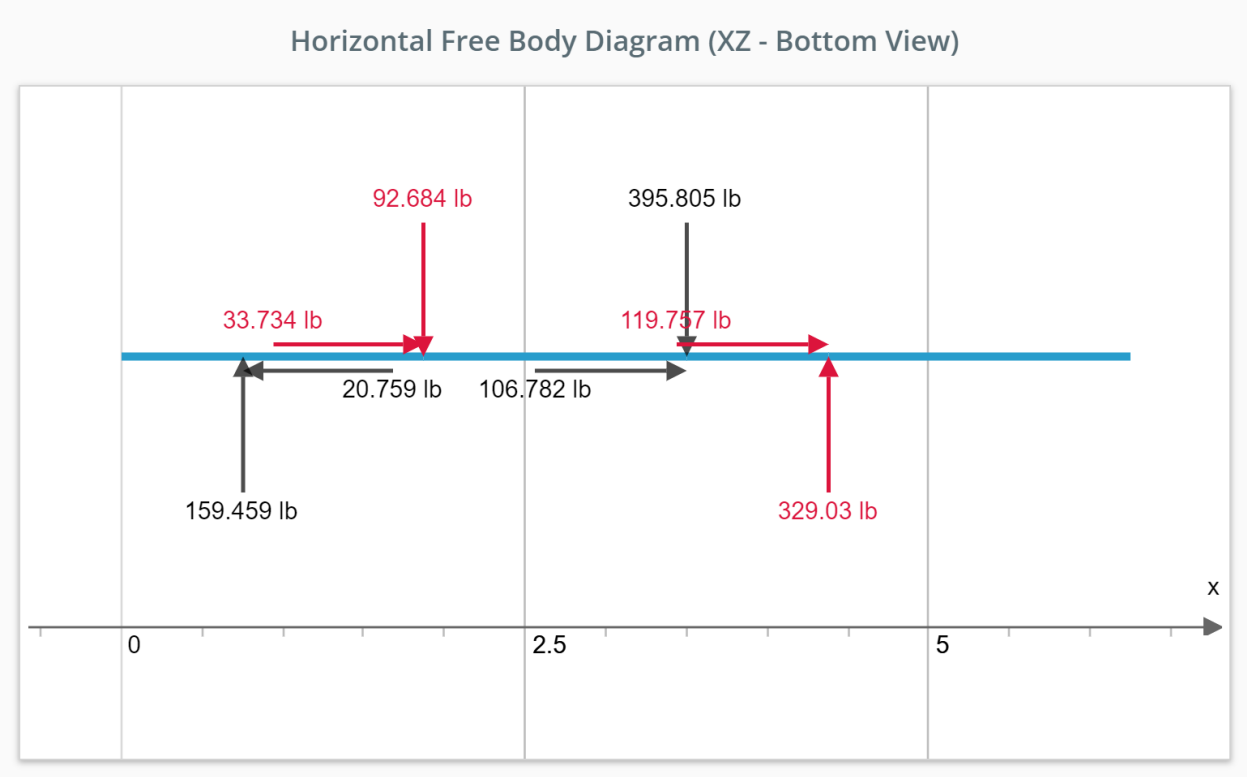


Figure 14. Free Body Diagram in XZ Plane of Shaft 2

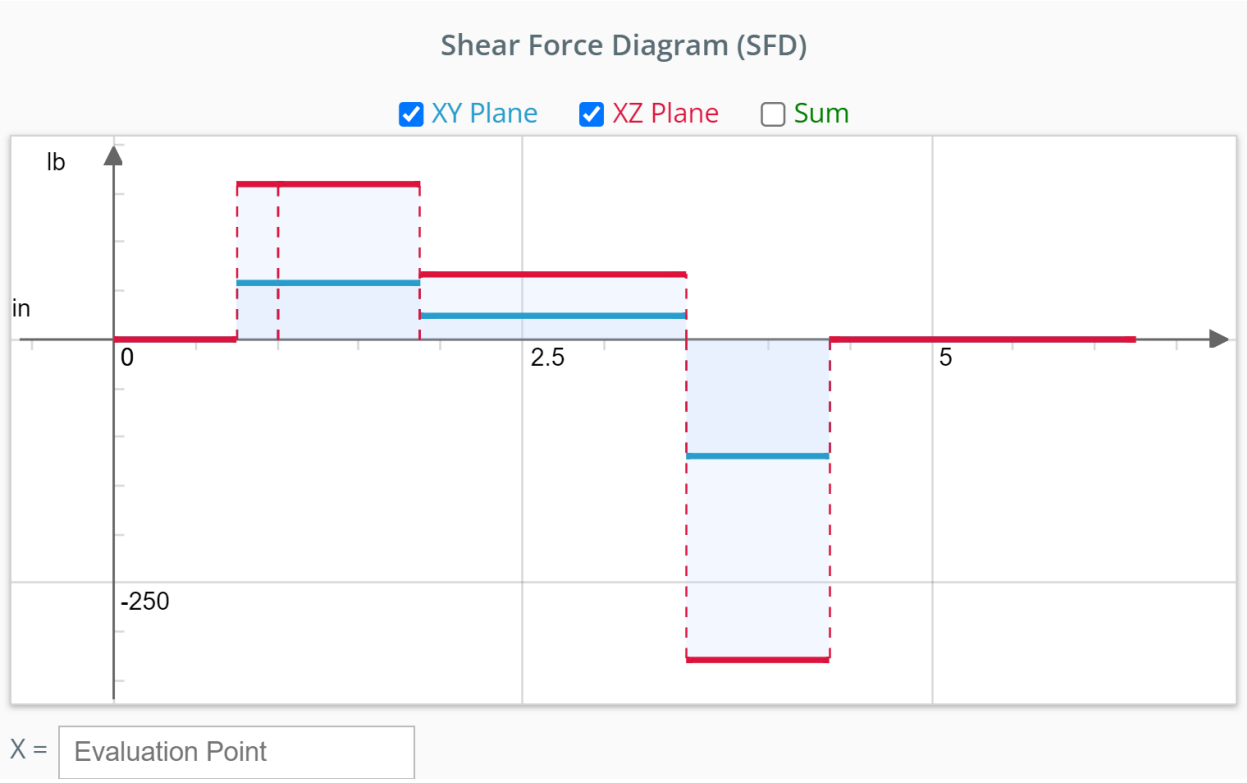


Figure 15. Shear Force Diagram of Shaft 2

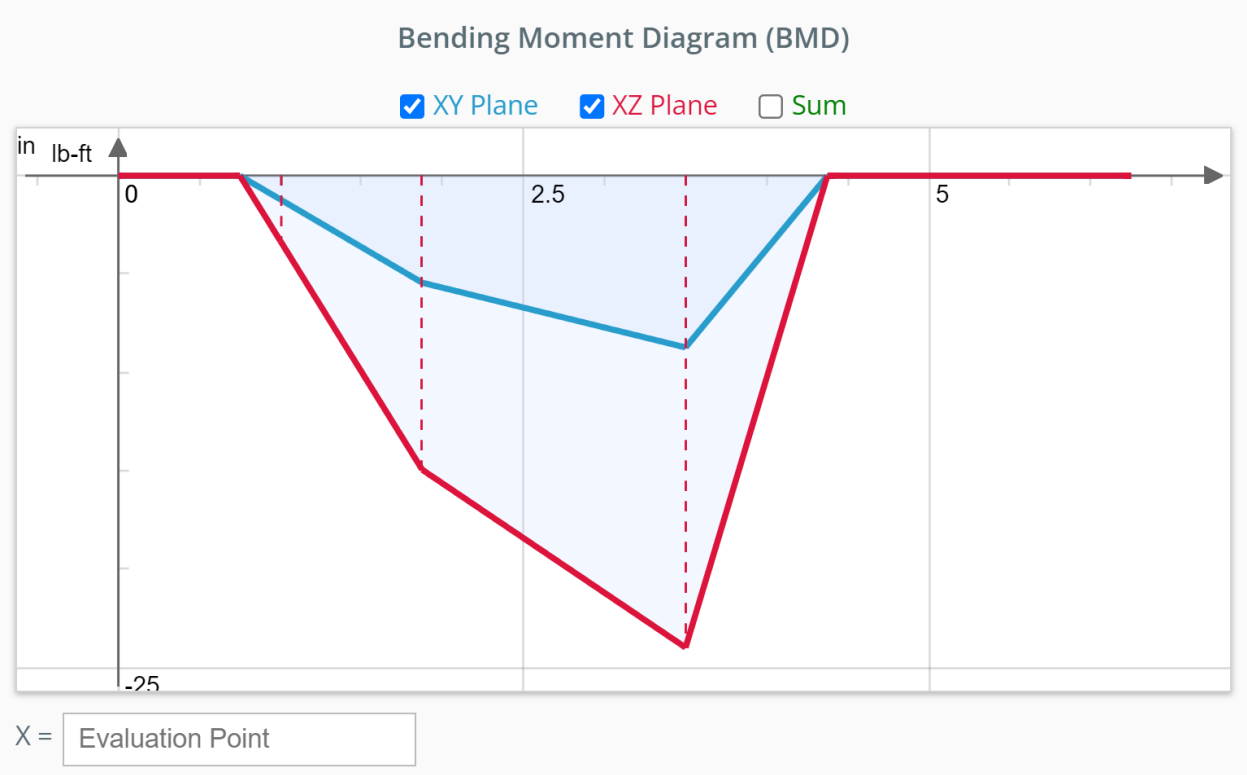


Figure 16. Bending Moment Diagram of Shaft 2

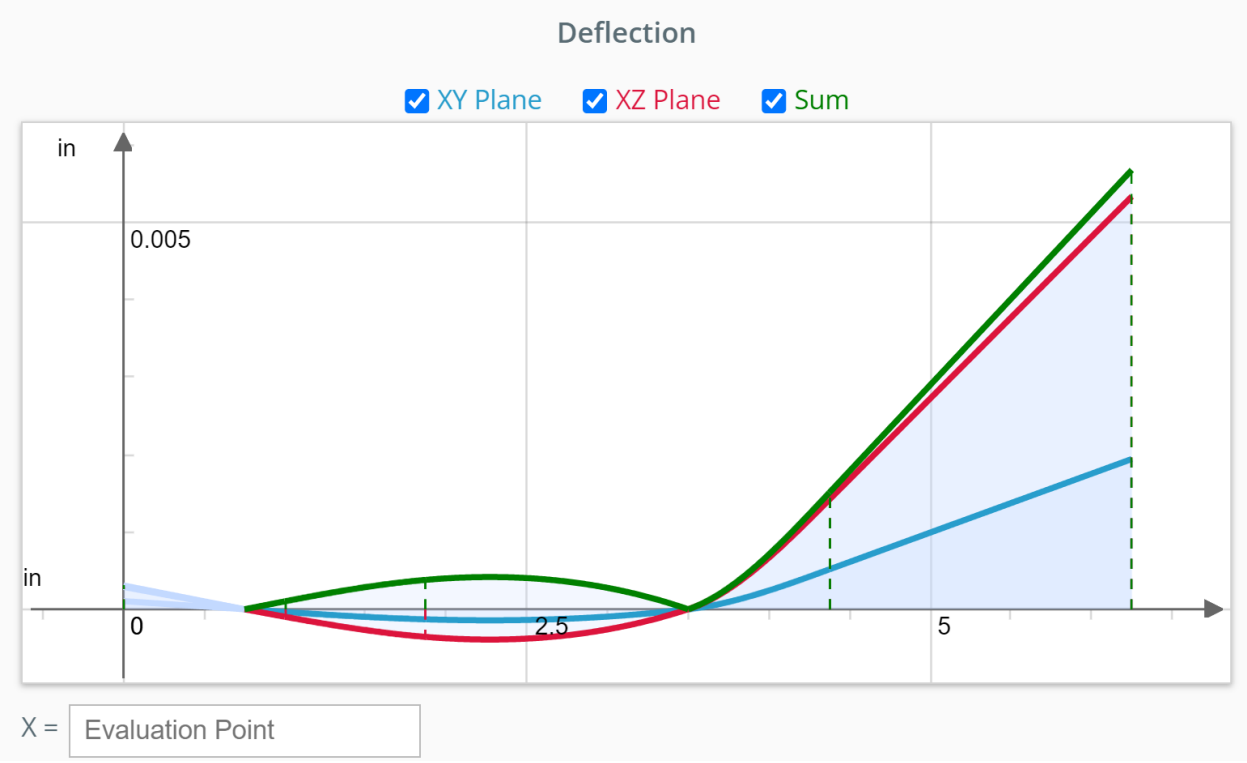


Figure 17. Deflection Diagram of Shaft 2

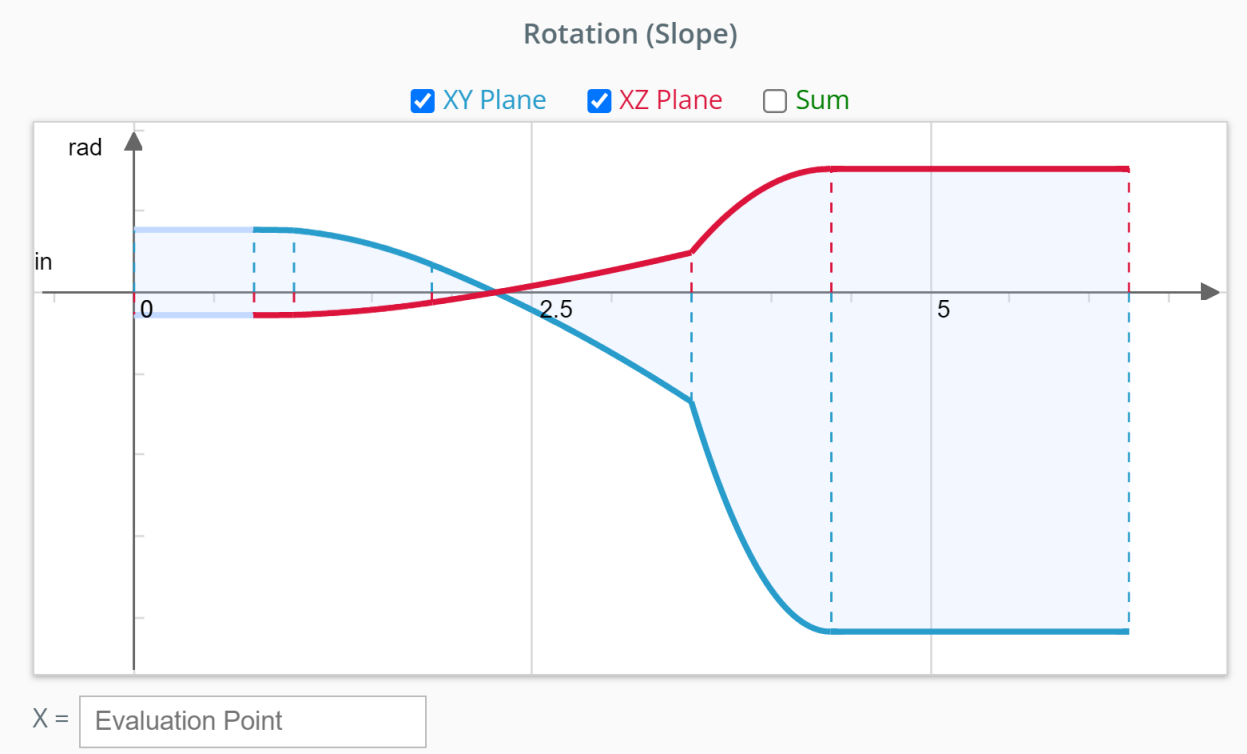


Figure 18. Rotation Diagram of Shaft 2

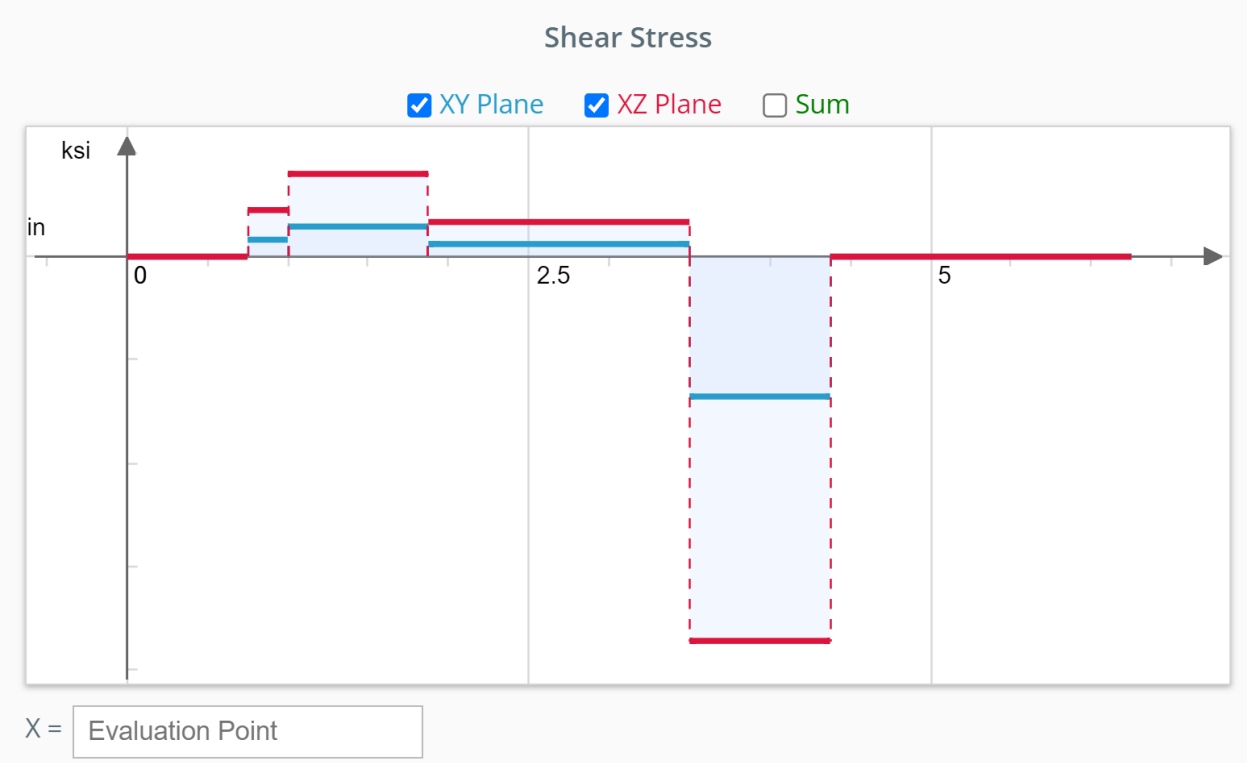


Figure 19. Shear Stress Diagram of Shaft 2

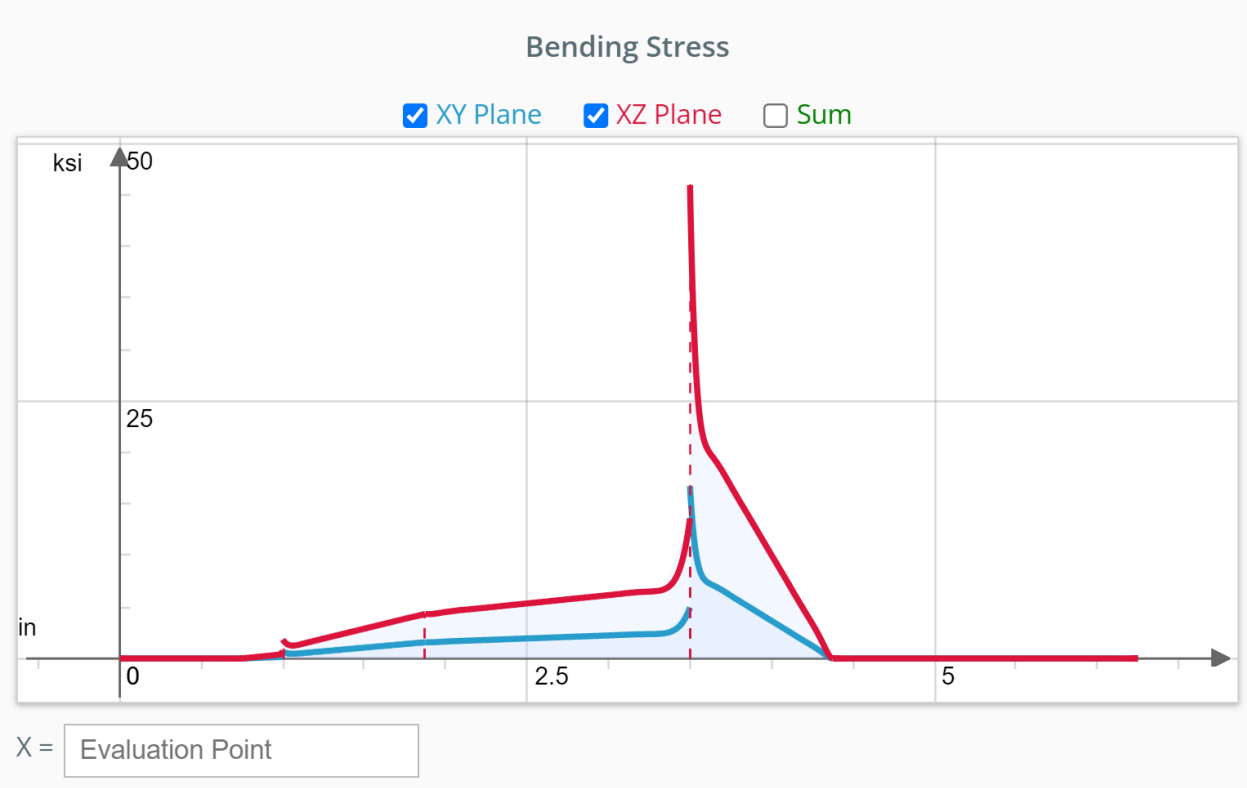


Figure 20. Bending Stress Diagram of Shaft 2

Table 13: Tabulated Results for shaft 2.

Shaft #2 Results	**Values**	**Unit of Measure**
Shaft Mass	0.629	lb
Critical Speed	5808.087	RPM
Maximum Power	0	HP
Maximum Bending Moment	25.532	lb-f
Maximum Shear Force	350.146	lb
Maximum Deflection	0.006	in
Maximum Rotation	0.002	rad
Maximum Bending Stress	48.883	ksi
Maximum Shear Stress	1.981	ksi
Maximum Torsion Stress	0	ksi
Maximum Axial Stress	1.241	ksi
Von Mises Stress	48.883	ksi

3.2.4 Shaft 3

The output shaft, concluding the sequence of shafts in our design, shares its material composition with shaft 2, utilizing Normalized AISI 1095 steel. This shaft accommodates the installation of the final gear (gear 4) through a keyed connection and is bolstered by two bearings. The shaft starts out with a ½ inch diameter where the first bearing is placed. Then there is a diameter increase to ¾ inch and quickly decreases again to ½ inch. The step up and down will help brace the bearing (bearing E) and gear. The diameter of the shaft does not change for the rest of its length. There is a second bearing (bearing F) 2 ½ inches down the shaft that is joined by a key. All shaft 3 analysis was conducted using SkyCiv [1] and can be seen in figures 21 to 30 and results are found in Table 14.

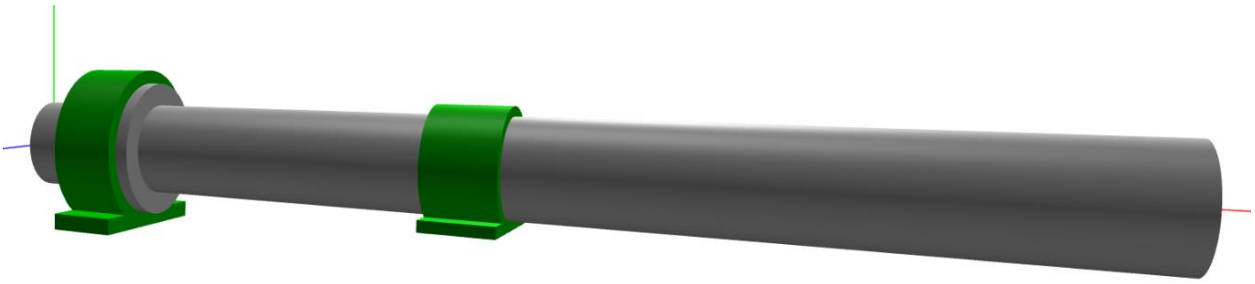


Figure 21. Output Shaft 3

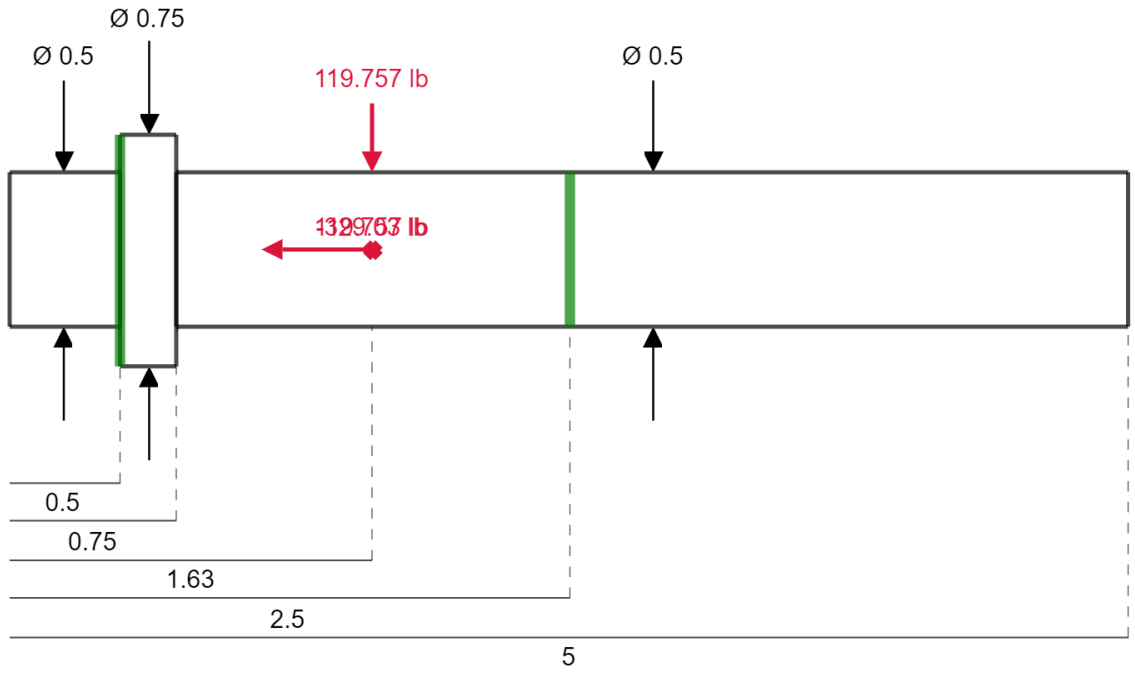


Figure 22. Shaft 3 Design

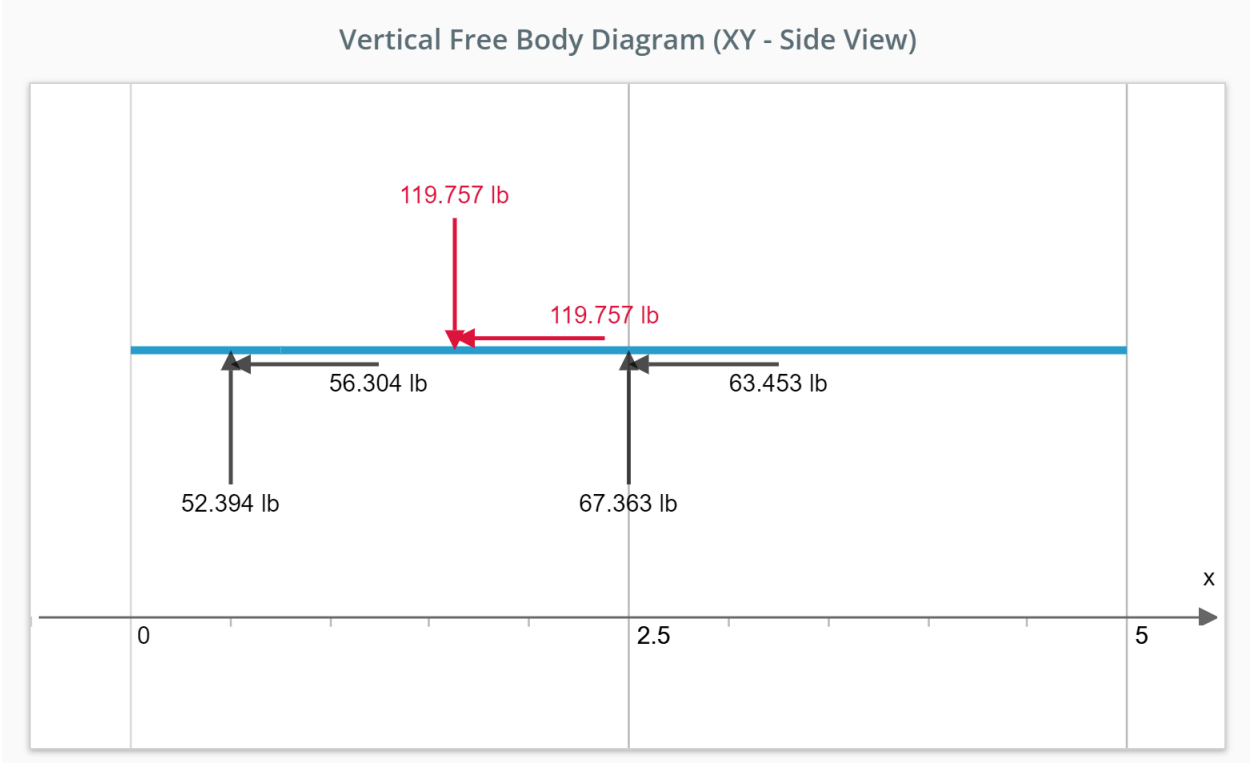


Figure 23. Free Body Diagram XY Plane of Shaft 3

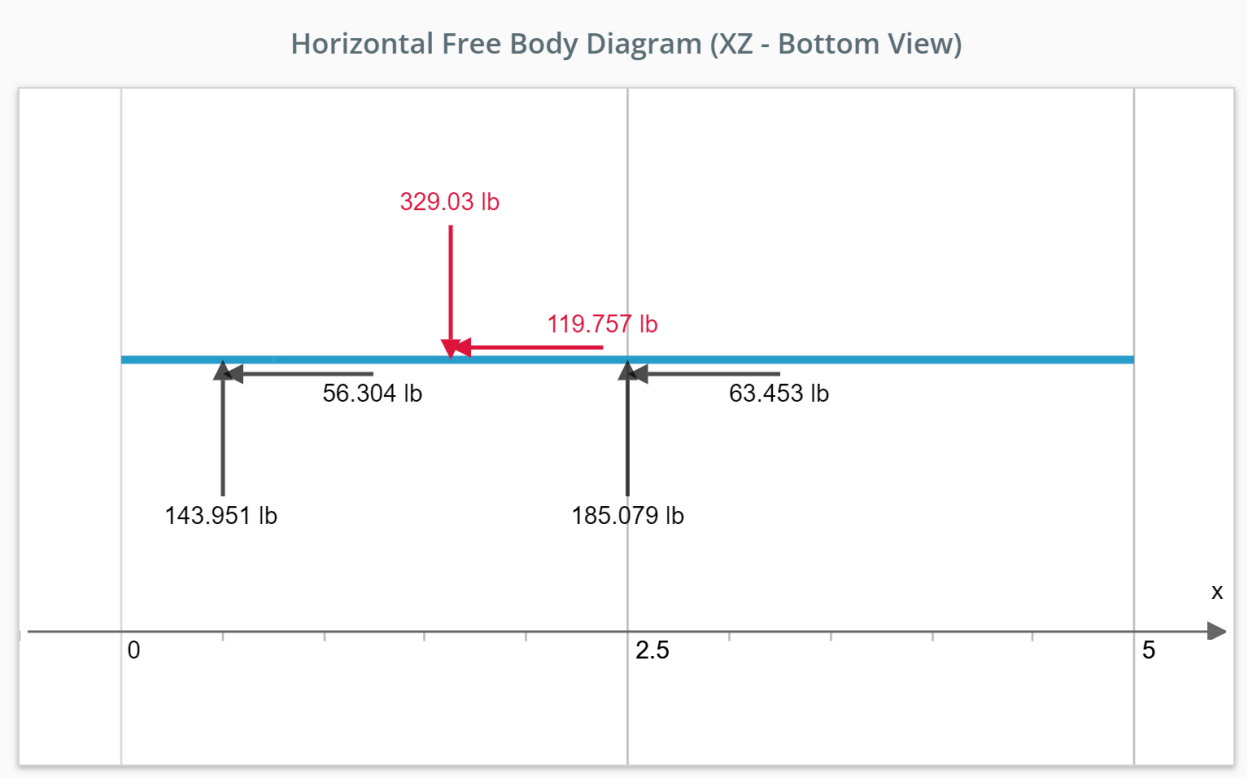


Figure 24. Free Body Diagram XZ Plane of Shaft 3

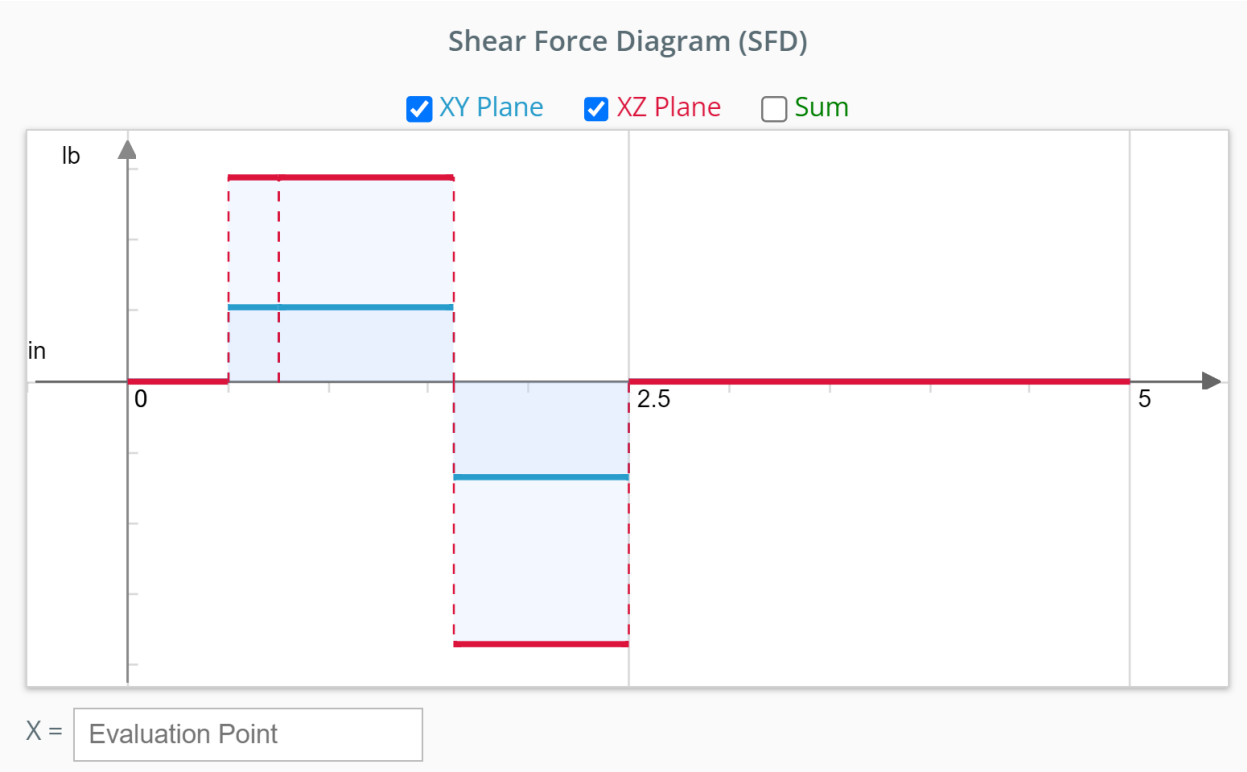


Figure 25. Shear Force Diagram of Shaft 3

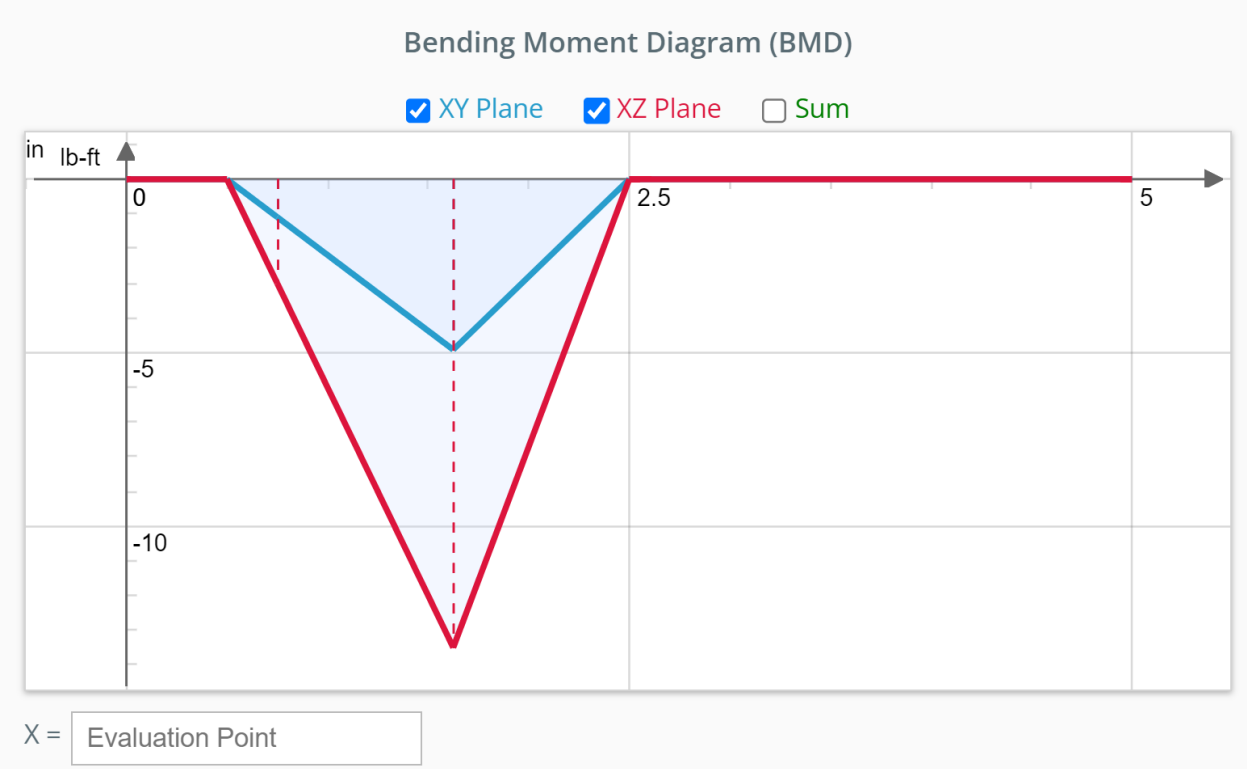


Figure 26. Bending Moment Diagram of Shaft 3

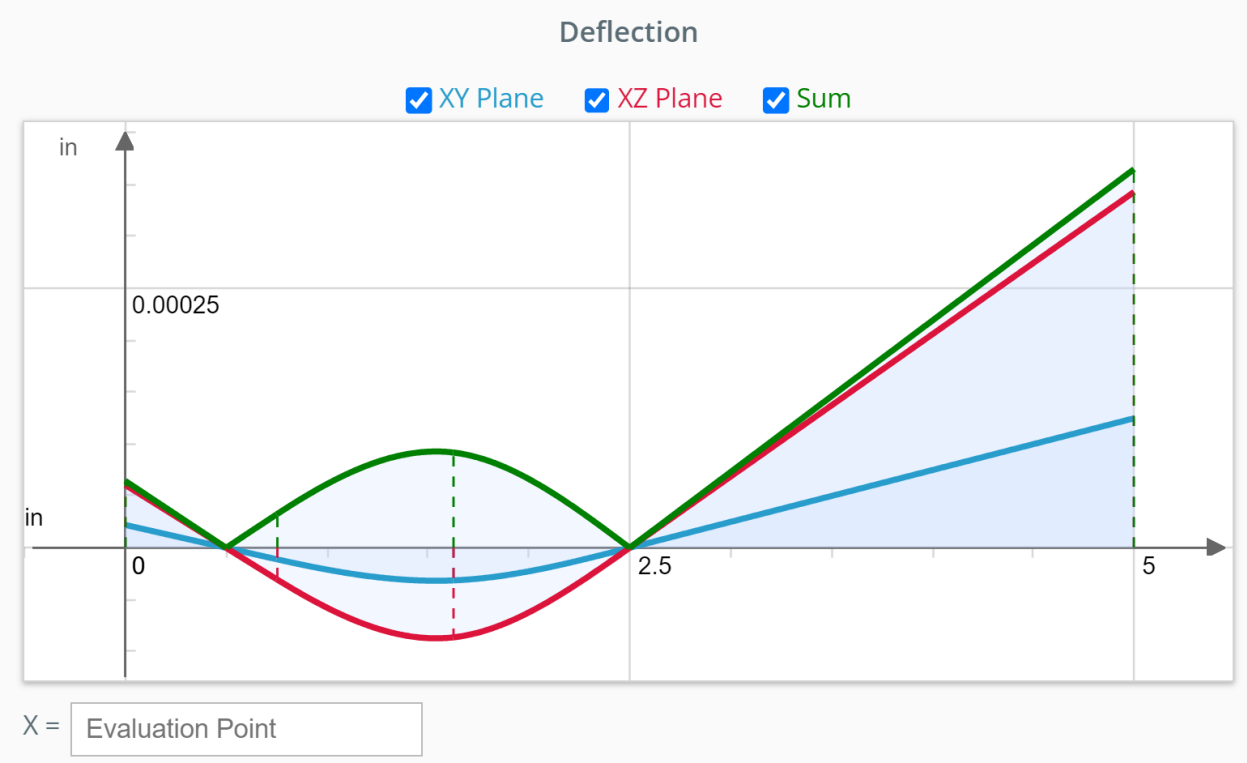


Figure 27. Deflection Diagram of Shaft 3

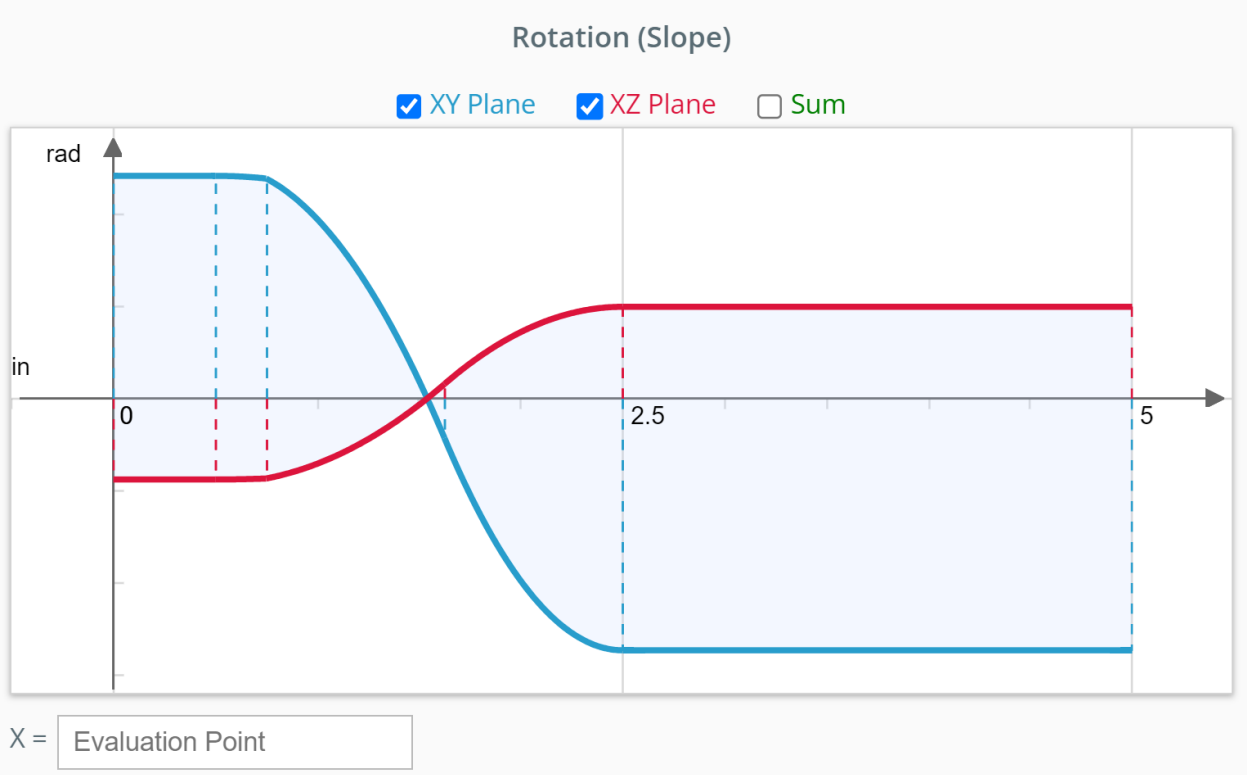


Figure 28. Rotation Diagram of Shaft 3

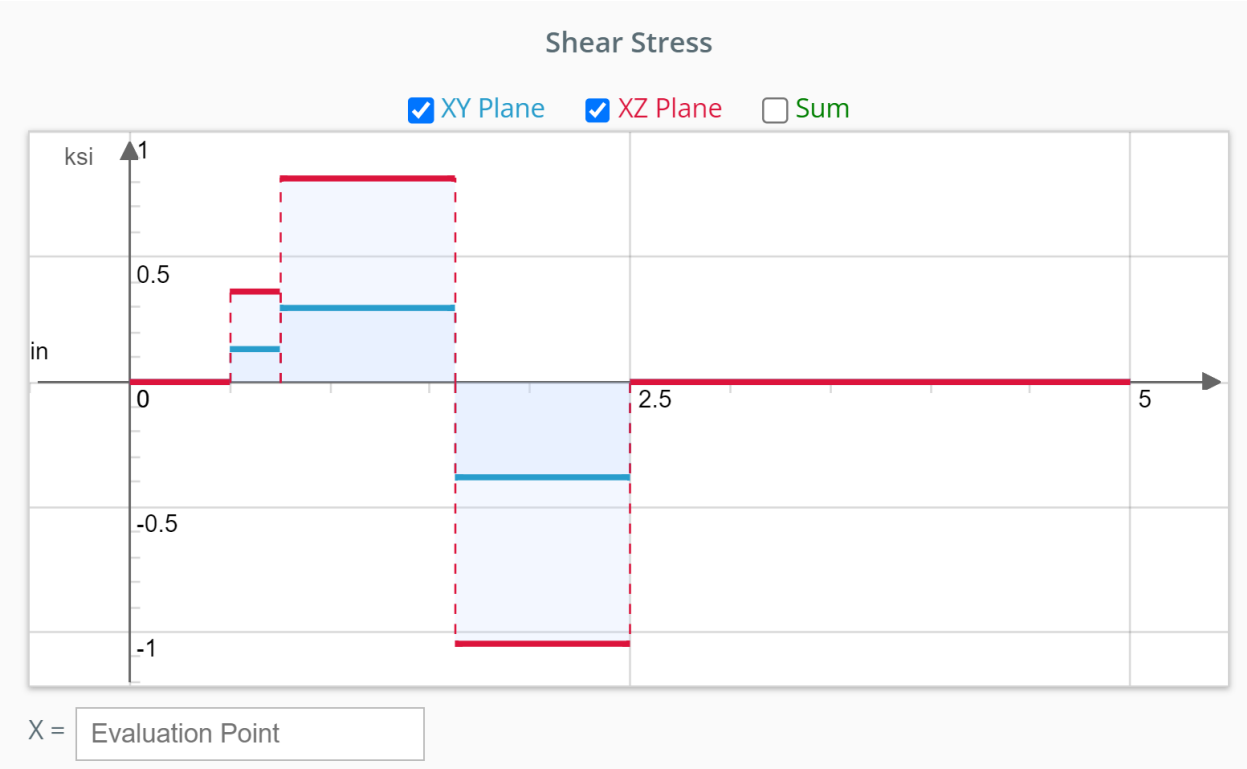


Figure 29. Shear Stress Diagram of Shaft 3

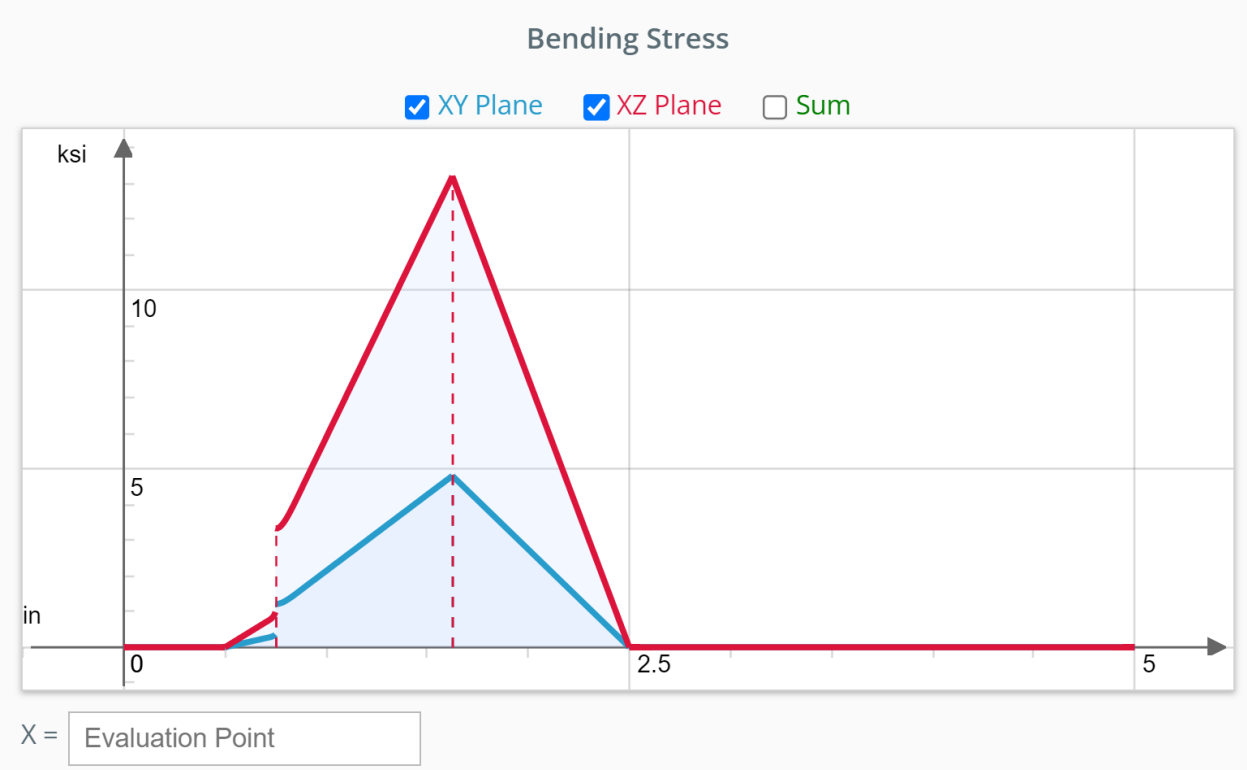


Figure 30. Bending Stress Diagram of Shaft 3

Table 14: Tabulated Results for shaft 3.

Shaft #3 Results	**Values**	**Unit of Measure**
Shaft Mass	1.349	lb
Critical Speed	22492.032	RPM
Maximum Power	0	HP
Maximum Bending Moment	14.361	lb-f
Maximum Shear Force	196.957	lb
Maximum Deflection	0	in
Maximum Rotation	0	rad
Maximum Bending Stress	33.288	ksi
Maximum Shear Stress	1.981	ksi
Maximum Torsion Stress	0	ksi
Maximum Axial Stress	0.66	ksi
Von Mises Stress	33.288	ksi

3.2.5 Safety Factor

To ensure each shaft design and chosen materials meet the requirements, a safety factor for each of the three shafts will be determined using the Distortion Energy equation (Von misses) equation for textbook [2]. Using the Distortion Energy equation to calculate the safety factors, it will provide reliability under the expected operational conditions. Shaft design safety factor results are found in table 14.

$$n = \frac{S_y}{\sigma'}$$

S_y is the material yield strength and σ' is the Endurance stress or also know as the Von Mises stress.

From the material selection S_y is known for all 3 shafts and the σ' was calculated in the software used for the shaft analysis.

Table 15: Shaft Design Safety Factors.

Shaft	**1**	**2**	**3**
S_y (ksi)	47	72.5	72.5
σ' (ksi)	4.453	34.205	9.826
Safety Factor - n	**10.55**	**2.12**	**7.38**

3.2.6 Gear Mounting

Gears are secured to their shafts typically using a spline or a square key and held in place with the use of a set screw. Square keys stand out for their ability to effectively transmit torque

from the shaft to the gear, making them the most common method for such connections. The choice of keys is motivated by their simplicity and cost-effectiveness in manufacturing. For the keys, AISI 1015 Annealed material was selected, with a yield strength (Sy) of 41.3 Ksi, which is intentionally lower than the weakest shaft's yield strength of 47 Ksi. This design choice ensures that, in the event of a failure, the key is the first to fail, most likely through shear. This approach not only manages medium to high torque levels efficiently but also minimizes costs compared to other gear reducer components. In a worst-case scenario, the preference is for the key to fail before any other part, due to its ease of replacement and lower cost relative to more critical components like the shaft or helical gears.

The key dimensions were calculated using the relationship of d/4 for the width and height and the highest length was chosen based on the shear and compressive forces that act on the key shaft, as seen in figure 31. From the literature a good key design tends to use a length close to L = 1.8d, where d is the diameter of the shaft.

Additionally, eq 4.4 a) P.524, [2] was used to help determine the shaft torque capacity.

$$T = \frac{\pi d^3}{16}(0.58S_y)$$

$$T = \frac{\pi \times 0.75^3}{16}(0.58 \times 41.3) = 2.65\ ksi$$

Equation 4.4 b) from textbook [2] for the torque that can be transmitted by compressive forces acting on the sides of the keys was rearranged to find the minimum needed length needed for compressive forces on the key.

$$T = \frac{S_y L d^2}{16}$$

$$L = \frac{2.65\ x\ 16}{41.3\ x\ 0.75^2} = 1.83\ inches$$

Finally, equation 4.4 c) [2], for the torque that can be transmitted by key shear was rearranged to find minimum length needed due to shear forces on the key. All square key dimensions for each shaft are shown in Table 16.

$$T = \frac{0.58 S_y L d^2}{8}$$

$$L = \frac{2.65\ x\ 8}{0.58\ x\ 41.3\ x\ 0.75^2} = \ 1.57\ inches$$

Table 16: Square Key Dimensions for each shaft size.

Gear	**1**	**2**	**3**	**4**
Shaft diameter mounting gear (inch)	0.75	0.75	0.5	0.5
Square key width and height (inch)	**0.1875**	**0.1875**	**0.125**	**0.125**
Compressive force Key length L (inch)	1.57	1.57	0.91	0.91
Shear forces key length L (inch)	1.83	1.83	0.76	0.76
Chosen key length L (inch)	**1.83**	**1.83**	**0.91**	**0.91**

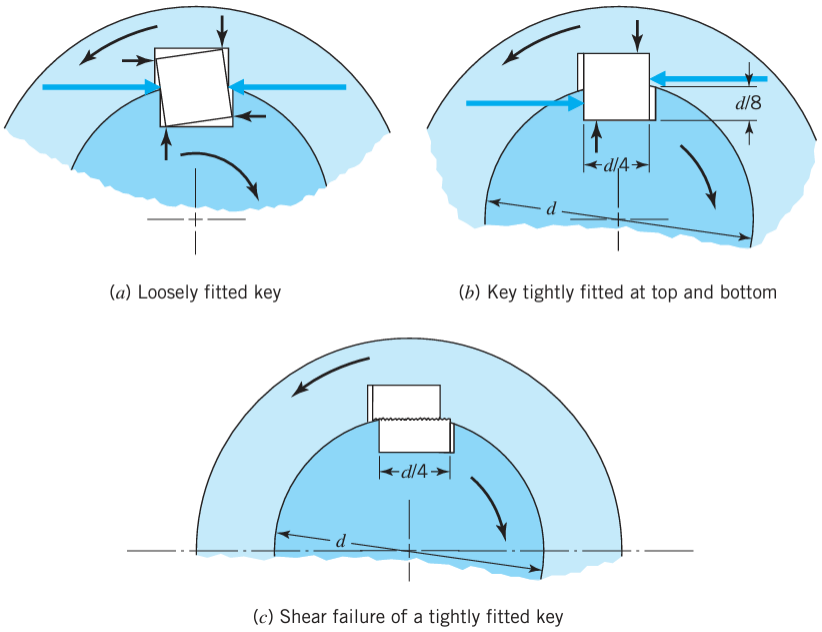


Figure 31. Square Key Loading and Stresses

3.3 Bearings

Bearings serve the purpose of load transmission at their point of contact. When integrated with a rotating shaft, motion occurs between the surfaces in contact. It's crucial to reduce the friction and wear resulting from this motion, achievable through lubrication. Bearings come in two main types: roller element and journal. Given that the reducer is equipped with helical gears, it will experience both axial and radial forces. Journal bearings can support greater loads compared to roller element bearings but demand a consistent lubrication system.

To conserve space and reduce costs, a variety of radial deep groove ball bearings are utilized to accommodate both the radial and axial forces present in the helical gears and will manage the force demands within the gearbox.

3.3.1 Force Analysis and Bearing Requirements

Generally, forces applied on bearing can be radial, axial or a combination of the two. In the context of our gear reducer, the helical gears have a helix angle of 20 deg. This means that the forces caried in the shaft are both axial and radial, therefore the selected bearing needs to reflect accordingly.

From the shaft analysis, figures 3, 13, and 23, shows the axial and radial forces in the bearing, where x is the axial force in lbs. and y is the radial force in lbs. Table 17 shows the radial and axial forces along with the force rating.

Table 17: Bearing Force Rating.

Bearing	Radial forces (F_r) (KN)	Axial Forces (F_a) (KN)	F_a/F_r
1	19.403	18.554	1.045758327
2	14.331	15.18	0.944071146
3	20.759	58.038	0.357679451
4	106.782	144.061	0.741227674
5	55.425	52.394	1.057850136
6	64.332	67.363	0.955004973

Using equation 14.3 and 14.4 the radial and angular force equivalent (Fe) was determined and results for each bearing are found in Table 18.

$\alpha = 0°$ (radial ball bearings)

$$\left.\begin{array}{l} \text{For } 0 < F_t/F_r < 0.35, \quad F_e = F_r \\ \text{For } 0.35 < F_t/F_r < 10, \quad F_e = F_r\left[1 + 1.115\left(\dfrac{F_t}{F_r} - 0.35\right)\right] \\ \text{For } F_t/F_r > 10, \quad F_e = 1.176F_t \end{array}\right\} \quad \textbf{(14.3)}^2$$

$\alpha = 25°$ (angular ball bearings)

$$\left.\begin{array}{l} \text{For } 0 < F_t/F_r < 0.68, \quad F_e = F_r \\ \text{For } 0.68 < F_t/F_r < 10, \quad F_e = F_r\left[1 + 0.870\left(\dfrac{F_t}{F_r} - 0.68\right)\right] \\ \text{For } F_t/F_r > 10, \quad F_e = 0.911F_t \end{array}\right\} \quad \textbf{(14.4)}^2$$

Table 18: Angular Bearing Force Equivalent (Fe).

Bearing	$F_e \ \alpha = 0^\circ$ *angular ball bearing* (Lbs)	$F_e \ \alpha = 25^\circ$ *angular ball bearing* (Lbs)
1	32.947647	24.4580636
2	25.23507	18.667482
3	58.534956	58.038
4	206.90312	151.7348524
5	94.726202	70.3821896
6	112.80477	83.4798892

It will be assumed that the bearings will have a conservative design for light to moderate impact and are expected to last for 15 000 hrs. as it will be expected to run for 8 hours per day. Using this assumption, the bearings will need to be 90% reliable with both radial and angular ability. Bearings will rotate according to their shaft specific rpm. Meaning that bearing 1 and 2 will rotate at the input rpm of 4000. Bearing 3 and 4 will rotate at the rpm of gear 2 and bearing 5 and 6 will rotate at the output rpm.

To convert rpm to rotation cycle of the bearing, the following equation is used.

$$L = rpm\ of\ shaft\ \times\ hours\ of\ use\ \times\ 60min/h$$ (p.433)

Table 19 displays the life cycle for each bearing according to its respective rpm.

Table 19: Bearing Speeds and Life Cycle.

Bearing	Rpm	L life cycle revolution
1, 2	4000	3.6 x 10 ^ 8
3, 4	1440	1.295 x 10^8
5, 6	300	2.7 x 10^7

For bearing 1 and 2 for example

W1 = 4000 rpm for 15 000 hrs.

$L = 4000\,rpm\ \times\ 15\,000h\ \times\ 60min/h\ =$ 3.6 x 10^8 rev

The bore of each bearing is according to their placement on the shaft. For example, bearing 1 and 2 are both on a 0.75-inch diameter shaft, which gives around 19.05 mm (for our selection an imperial bearing was selected to ensure the best fit.) Also needed is the hypothetical dynamics load rating that the chosen bearing will need to be able to sustain. Using equation 14.5b from the textbook [2] C_{req} can be calculated.

$$C_{req}\ = F_e K_a (L/K_r L_R)^{0.3}\ eq14.5b(p.432)$$

Where Fe is 24.458 lbs for bearing 1.

Ka is application factor (table 14.3 P432) of 1.1 for bearing 1.

L is life cycle revolution à 3.6 x 10 ^ 8 for bearing 1.

Kr life adjusted reliability factor figure 14.14 (P.431) for 1 for bearing 1.

LR is = 90 x10 ^ 6 revolution life with 90% reliability.

Example for bearing 1.

$$C_{req}\ RADIAL\ = 24.458 \times 1.1(\ 3.6\ \text{x}\ 10\ \text{^}\ 8/1 \times\ 90\ \text{x}10\ \text{^}\ 6\ )^{0.3}\ =\ 81.36\ Lbs\ for\ bearing\ 1$$

$$C_{req}\ Angular\ = 32.94 \times 1.1(\ 3.6\ \text{x}\ 10\ \text{^}\ 8/1 \times\ 90\ \text{x}10\ \text{^}\ 6\ )^{0.3}\ =\ 109.6\ Lbs\ for\ bearing\ 1$$

Therefore, table 20 summarizes all the needed specifications each bearing needs for our gear box design.

Table 20: Bearing Specifications.

Bearing	Bore (inch)	Life cycle L	C_{req} Radial (Lbs)	C_{req} Angular (Lbs)	SKF Bearing Part #
1	0.75	3.6 x 10 ^ 8	109.6061897	109.6061897	W61903
2	0.75	3.6 x 10 ^ 8	83.94893604	109.6061897	W61903
3	0.75	1.295 x 10^8	143.3232756	109.6061897	W61903
4	0.5	1.295 x 10^8	506.6038459	109.6061897	16101
5	0.5	2.7 x 10^7	0.313930723	109.6061897	61801
6	0.5	2.7 x 10^7	144.8769132	109.6061897	61801

3.3.2 Bearing Selection and Sizing

Shaft 1

For the input shaft, (shaft 1), it is possible to utilize a pair of stainless-steel deep groove ball bearings due to their versatility. This arrangement allows the shaft to adequately support axial or thrust loads from either direction, which can vary based on the rotational direction of the motor and is also rated to support any radial loads. Utilizing deep groove ball bearings in such a configuration ensures that no matter which rotation the motor spins, clockwise or

counterclockwise, the bearings will be capable of handling the thrust generated in either direction. For bearing 1 and 2 an SKF #W61903 bearing, Appendix VI, was chosen for both ends of the shaft as shown in figure 32, which is capably of withstanding the needed C_{req} values as per Table 20. This setup is particularly useful in applications where the direction of the load may change, providing flexibility and resilience in the bearing system.

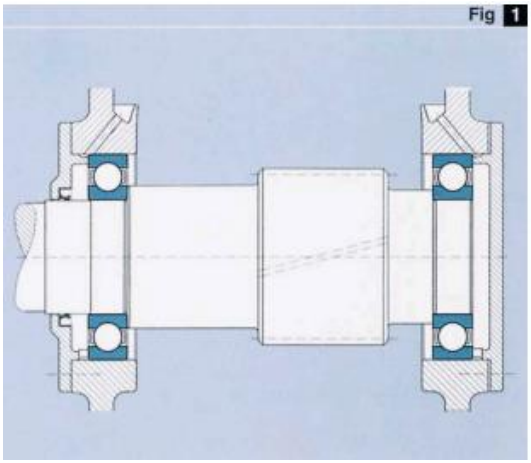


Figure 32. Shaft 1 Deep Groove Ball Bearing

Shaft 2

For the intermediate shaft, the implementation of two spherical roller bearings one either side was considered but was found to be an overdesign so again a set of stainless-steel deep groove ball bearings was found to be a suitable choice due to their versatility. The chosen bearings and arrangement can be seen in figure 33 and the bearings used for bearing 3 was an SKF #W61903, Appendix VI, and for Bearing 4 was a SKF #16101 type, Appendix V, which are both designed to manage higher radial loads and to cope with any misalignment between the shaft and the housing, which is common in applications involving this shaft. These bearings are robust in design, are

capable of radial and axial loads in both directions and require little maintenance with good chemical and corrosion resistance.

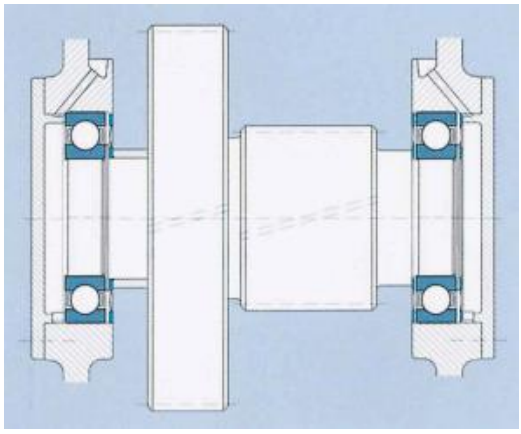


Figure 33. Shaft 2 Deep Groove Ball Bearing

Shaft 3

For the output shaft, employing two spherical roller bearings would be an effective strategy but was an over design, so to cut cost and still provide adequate reliability for the design stainless-steel deep groove ball bearings were again selected. Given that this shaft is likely to transmit high torque and may be subject to external forces based on the mechanisms or loads it drives, an SKF #61801 was specified (Appendix IV). Their design allows for handling not only radial loads but also axial loads in both directions, which is particularly advantageous under high-torque scenarios. Figure 34 provides a picture of the straddle bearing arrangement on the shaft.

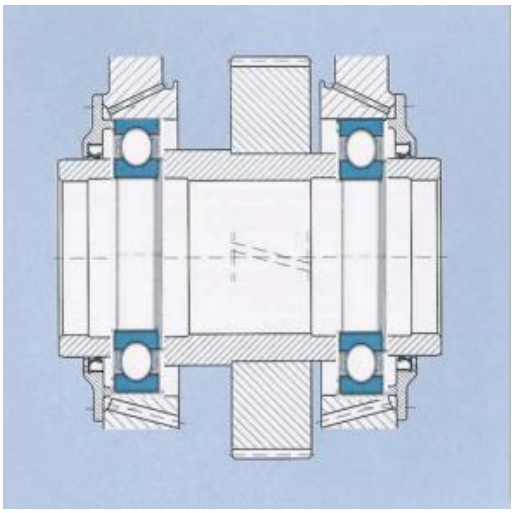


Figure 34. Shaft 3 Deep Groove Ball Bearing

3.3.3 Bearing Mounting, Lubrication, and Maintenance

All bearings for this gear reducer design will be secured to the shaft using a press or interference fit to minimize bearing movement on the shaft. The bearings should not exceed 100˚C under normal operation to prevent degradation of the bearing lubricants, prevent dimensional changes in bearing rings and rolling elements and lastly to prevent any changes in the preloading of the bearings. Adequate lubrication of the bearings and gears should be in the form an oil bath which is essential as approximately 1/3 of the power loss is dissipated through the lubricant and the other sources of power loss is through convection, conduction, and radiation [2].

To prevent damage or debris from entering the bearing rolling elements from exterior sources and to prevent oil from leaking through the opening around the protruding shafts, spring loaded rotary shaft seals made of Buna-N-Rubber with a steel core and spring will be used on the input and output shafts, like those seen in Figure 35 [7].

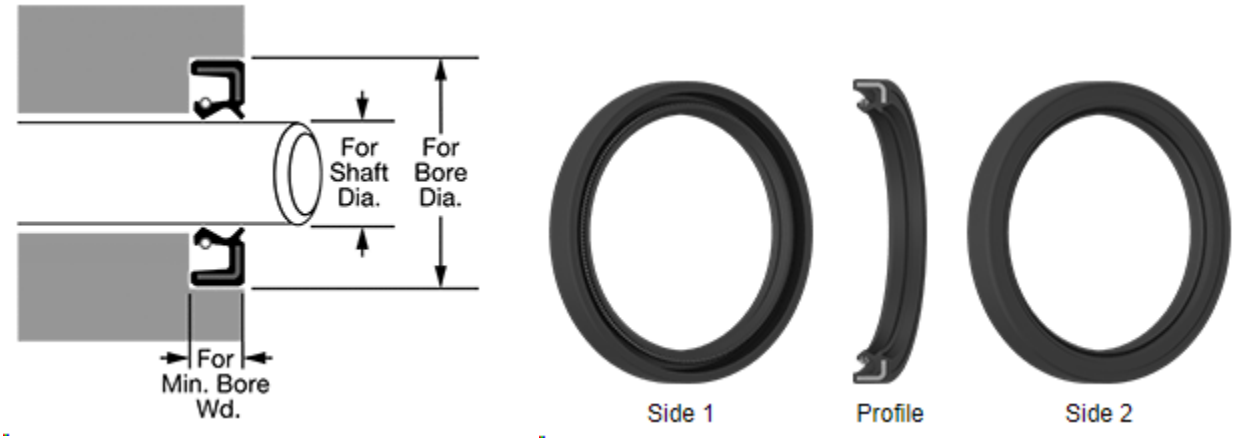
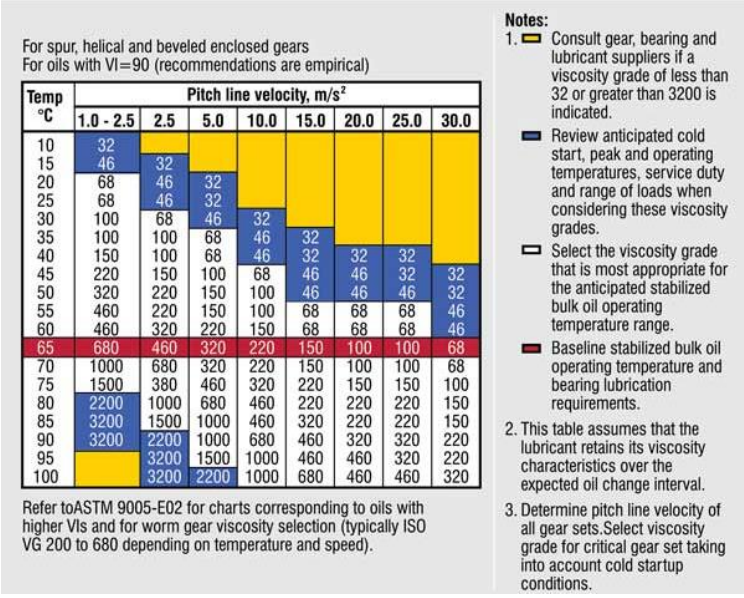


Figure 35. Spring Loaded Rotary Shaft Seals

(Output Shaft Seal Part #1199N61, Input Shaft Seal Part #1199N64)

Gear reducer oil levels should be monitored by use of a site glass located at the bottom side of the gearbox housing and topped up as needed in the event of a low level, which is usually caused by leakage. Oil quality should be sampled during regular maintenance schedules and changed based on lab oil analysis or the number of contaminants, such as metal shavings, found in the oil. Based on the pitch line velocity of the highest gear set of 1780.234 ft/min = 9.04 m/s and a baseline temperature of 65ºC, an oil with a viscosity index of VG = 220, as chosen from the chart in Figure 36 [8], should be used. A few recommended oils would be Super Lube Synthetic Gear Oil ISO 220 or a Mobil SHC 630 Synthetic Bearing and Gear Oil ISO VG 220.



For spur, helical and beveled enclosed gears
For oils with VI=90 (recommendations are empirical)

Temp °C	Pitch line velocity, m/s²							
	1.0 - 2.5	2.5	5.0	10.0	15.0	20.0	25.0	30.0
10	32							
15	46	32						
20	68	46	32					
25	68	46	32					
30	100	68	46	32				
35	100	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	32
50	320	220	150	100	46	46	46	32
55	460	220	150	100	68	68	68	46
60	460	320	220	150	68	68	68	46
65	680	460	320	220	150	100	100	68
70	1000	680	320	220	150	100	100	68
75	1500	380	460	320	220	150	150	100
80	2200	1000	680	460	220	220	220	150
85	3200	1500	1000	460	320	220	220	150
90	3200	2200	1000	680	460	320	320	220
95		3200	1500	1000	460	460	320	220
100		3200	2200	1000	680	460	460	320

Refer toASTM 9005-E02 for charts corresponding to oils with higher VIs and for worm gear viscosity selection (typically ISO VG 200 to 680 depending on temperature and speed).

Notes:

1. Consult gear, bearing and lubricant suppliers if a viscosity grade of less than 32 or greater than 3200 is indicated.
   Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
   Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
   Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.
2. This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval.
3. Determine pitch line velocity of all gear sets.Select viscosity grade for critical gear set taking into account cold startup conditions.

Figure 36. Viscosity Chart based on Pitch line Velocity vs Temperature.

4 Gear Reducer Housing or Casing

The Gearbox casing is an important component of the gearbox design as it is used to hold all shafts in place, needs to dissipate heat, contains all oil within the casing, and prevents debris from entering the bearings and gears. The gearbox needs to be adequately designed to withstand all forces and stresses placed on it without cracking and is the overall structure of the gearbox [8]. For this design, it was decided to use modern technological advances to fabricate the gearbox housing using a 3D printer and SolidWorks software as opposed to conventional sand-casting approaches. Material selection for the gearbox casing was based on a paper by T. N. Nikhil et al. [8] from the book series “Advances in Manufacturing Engineering Select Proceeding of ICFAMMT 2022”. Table 21 discusses the various materials commonly used in 3D printing and their benefits.

Table 21: Gearbox housing materials commonly used in 3D Printing [9].

Materials	Description
Polycarbonate	Polycarbonate is known for its strength and durability. It has very high heat and impact resistance making it an ideal choice for tough environments
Nylon	Nylon is a tough and semi-flexible material that offers high impact and abrasion resistance. It is an ideal choice for printing durable parts
Epoxy with carbon fibre	Carbon fibre filaments contain short fibres that are infused into a PLA or ABS base material to help increase strength and stiffness
Polyamide	With its strength and flexibility, polyamide allows for high levels of detail on a 3D-printed product. The material is especially suited for joining pieces and interlocking parts in a 3D-printed model. Polyamide is used to print everything from fasteners and handles to toy cars and figures
Carbon fibre	Composites such as carbon fibre are used in 3D printers as a top-coat over plastic materials. The purpose is to make the plastic stronger. The combination of carbon fibre over plastic has been used in the 3D printing industry as a fast, convenient alternative to metal

Based on this research, it was found that Polyamide provided the best overall material properties, besides Carbon fiber which is high in cost. Therefore, Polyamide will be selected as the material to be used in this gearbox housing design. The gearbox will be of a split housing design with a bottom and top section which is secured using Class 5 SAE bolts torqued to preset values. Between the 2 halves of the split housing will be a 1/8” thick pre-cut Garlock 5500 compressed inorganic fiber gasket material capable of sealing oil with a continuous max temperature of 290˚C as specified in Appendix I. [10] The base of the gearbox will be a foot mount design for easy mounting purposes.

5 3D- CAD Model and drawing

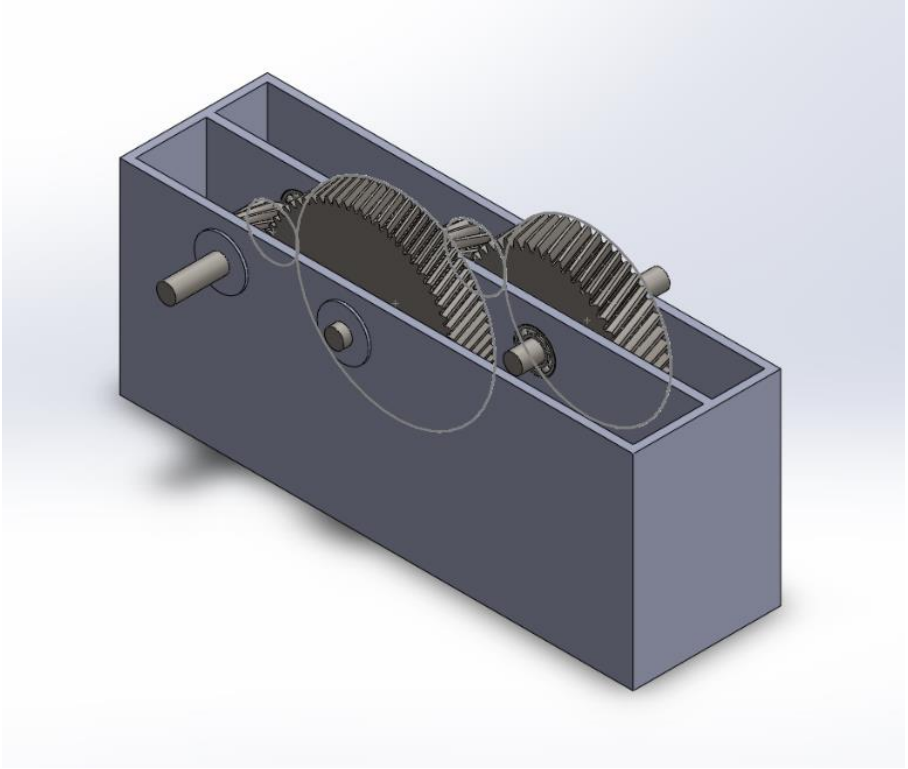


Figure 37: Isometric view of the gearbox with lid removed.

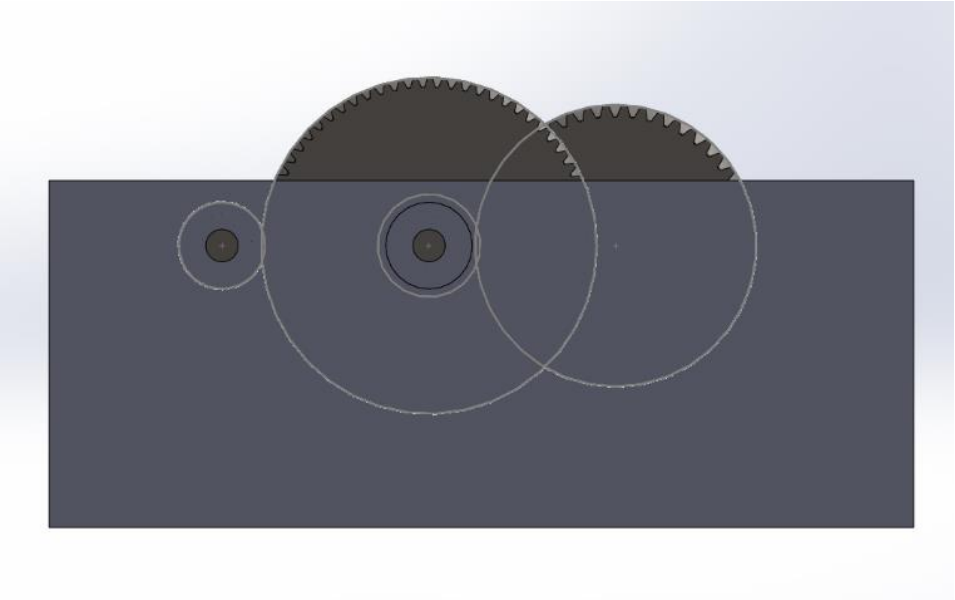


Figure 38: Front view of the gearbox with lid removed.

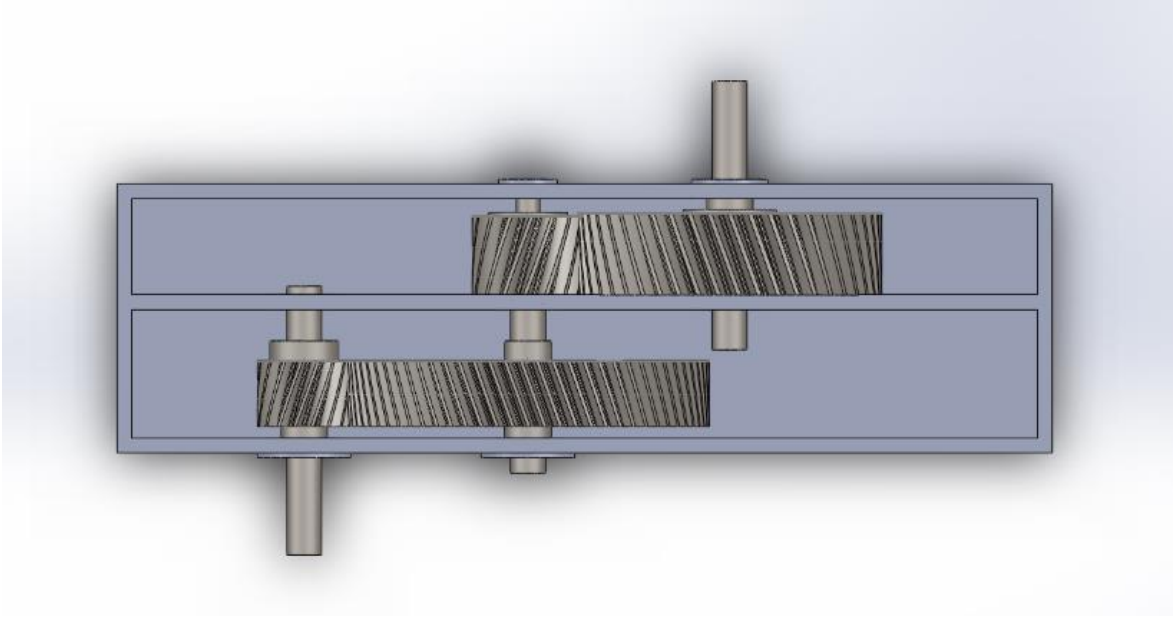
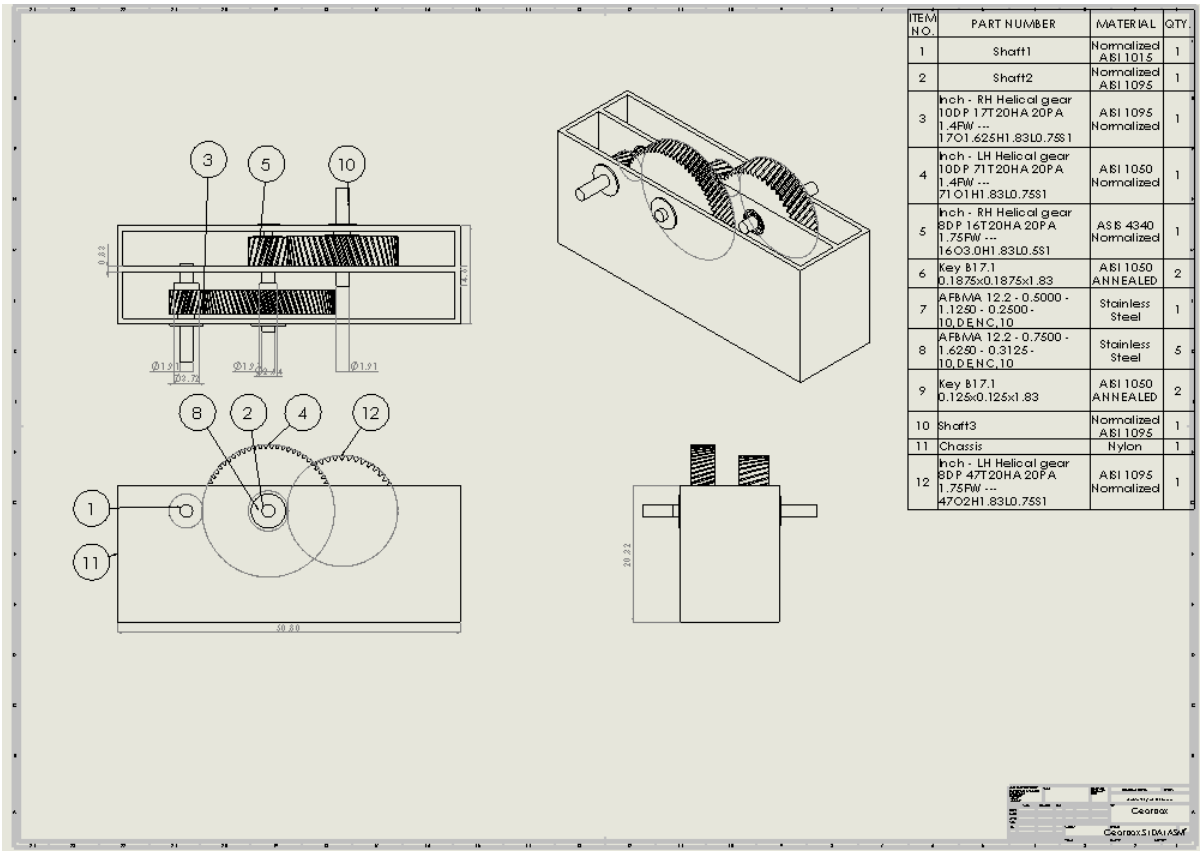


Figure 39: Top view of gearbox with lid removed.



ITEM NO.	PART NUMBER	MATERIAL	QTY.
1	Shaft1	Normalized AISI 1015	1
2	Shaft2	Normalized AISI 1095	1
3	Inch - RH Helical gear 10DP 17T20HA 20PA 1.4FW --- 17O1.625H1.83L0.75S1	AISI 1095 Normalized	1
4	Inch - LH Helical gear 10DP 71T20HA 20PA 1.4FW --- 71O1H1.83L0.75S1	AISI 1050 Normalized	1
5	Inch - RH Helical gear 8DP 16T20HA 20PA 1.75FW --- 16O3.0H1.83L0.5S1	ASIS 4340 Normalized	1
6	Key B17.1 0.1875x0.1875x1.83	AISI 1050 ANNEALED	2
7	AFBMA 12.2 - 0.5000 - 1.1250 - 0.2500 - 10,D,E,N,C,10	Stainless Steel	1
8	AFBMA 12.2 - 0.7500 - 1.6250 - 0.3125 - 10,D,E,N,C,10	Stainless Steel	5
9	Key B17.1 0.125x0.125x1.83	AISI 1050 ANNEALED	2
10	Shaft3	Normalized AISI 1095	1
11	Chassis	Nylon	1
12	Inch - LH Helical gear 8DP 47T20HA 20PA 1.75FW --- 47O2H1.83L0.75S1	AISI 1095 Normalized	1

Figure 40 : CAD drawing with Bill of materials.

6 Conclusion

In conclusion, the design of the gearbox elaborated in the report outlines the steps and iterative process that permitted the finalization of a gearbox reducer that satisfies the given criteria. The designed gearbox is a non-reversing, single output shaft, reducer that takes in 5 horsepower with and angular velocity of 4000 RPM, and outputs a consistent 5 horsepower with an output angular velocity of 300 RPM. The report describes the essential design process and highlights the various equations, assumptions, and design considerations that were made when selecting the components that would make up the gearbox/reducer. More specifically, the designed gearbox could be used in various applications that require low power, low speed reductions and works well for the chosen application of a winch or conveyor. Its important to note that certain engineering software's were used to lessen the workload, and simplify the redundancies associated with the iterative design of the gears, shaft, and bearings. The software's used for the analysis were *Excel* and *SkyCiv*, whilst *SolidWorks* was used to create a 3D-CAD model and generate an appropriate sketch and drawings of the gearbox design. In creating a gearbox that satisfies all the conditions required for a successful design, it is important to note that the design itself may not be perfect, but that it is, all details considered, an appropriate design. Using more complex analysis techniques with different engineering assumptions and design choices could in theory lead to a better design, with more optimized factors of safety. Often when designing the gearing, shafting, and bearing the safety factors they were quite large compared to the optimal regions in which they should find themselves. One such instance is with the bending stress and wear factors of safety for the gear calculations; where given the design choices, it was not possible to further lower the factor of safety in bending into a more appreciable range, without dropping the wear factor of safety below its lower limit. These details considered, and considering the final design that was obtained the

gearbox could be more optimized, but that given the restraints and used engineering assumptions the finalized gearbox serves as a solid design that has a high reliability.

7 Team Member Contributions

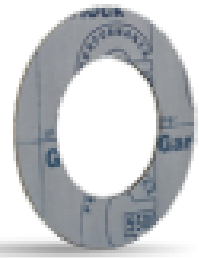
Table 22: Team Member Contributions

Team Member	Student ID	Contributions to Design Project
Christopher Godwin	8951529	Section 2, Section 3, Section 3.1.1, Section 3.1.3, Section 3.1.4, Section 3.2.1, Section 3.2.2, Section 3.2.3, Section 3.2.4, Section 3.2.5, Section 3.2.6, Section 3.3, Section 3.3.1, Section 3.3.2, Section 3.3.3, Section 4, Appendices, References, Formatting, Design Sketches Appendix II & III.
Spiridon Amohnoaie	300214069	Abstracts, List Acronyms, Section 3, Section 3.1.1, Section 3.1.4, Section 3.2.1, Section 3.2.5, Section 3.2.6, Section 3.3, Section 3.3.1, Excel spreadsheet, Formatting
Jacob Picard	300207379	List of Figures, Table of Contents, List of Tables, Section 3, Section 3.11, Section 3.1.2, Section 3.1.3, Section 3.2, Section 3.2.1, Section 3.2.2, Section 3.2.3, Section 3.2.4, Formatting
Antoine Flahaut	300224283	

Appendix I

Styles 5500 and 5507

Compressed Inorganic Fiber Gasket



BENEFITS

Tighter Seal

- Inorganic fiber gasketing offers excellent thermal stability with minimal weight loss
- Reduced creep relaxation and improved torque retention provide optimal sealability

Temperature resistant

- Non-oxidizing fibers withstand a continuous operating temperature of up to 550°F (290°C), and maximum spike of 800°F (425°C)
- Style 5500 has passed the Garlock Fire Test and is ABS Fire Safe Type Approved

Media

Style 5500

- Water, aliphatic hydrocarbons, oils, gasoline, saturated steam[1], inert gases, most refrigerants

Style 5507

- Water, saturated steam[1], mild chemicals and mild alkalies

TYPICAL PHYSICAL PROPERTIES[2]

		5500[4,5]	5507[5]
Color		Gray	Sand
Binder		Nitrile (NBR)	EPDM
Temperature[3]	Maximum Minimum Continuous Max.	800°F (425°C) -100°F (-75°C) 550°F (290°C)	800°F (425°C) -100°F (-75°C) 550°F (290°C)
Pressure[3], **continuous max.**	psig (bar)	1,200 (83)	1,200 (83)
P x T, max. (psig x °F) (bar x °C)	1/32", 1/16" (0.8mm, 1.6mm) 1/8" (3.2mm)	400,000 (14,000) 275,000 (9,000)	400,000 (14,000) 275,000 (9,000)
Sealability (ASTM F37B)[4] **ASTM Fuel A** **Nitrogen**	 ml/hr ml/hr	 0.3 1.0	 0.2 1.0
Creep Relaxation (ASTM F38)	%	25	15
Compressibility Average (ASTM F36)	%	10	10
Recovery (ASTM F36)	%	>50	>50
Tensile Strength across grain (ASTM F152)	psi (N/mm²)	1,500 (10)	1,500 (10)
Density	lbs/ft³ (g/cm³)	100 (1.60)	105 (1.68)
Gas Permeability (DIN 3535 Part 4)[4]	cc/min.	0.05	0.04

NOTES:

1. Based on ANSI RF flanges at our preferred torque. When approaching maximum pressure, continuous operating temperature, minimum temperature or 80% of maximum PxT, consult Garlock Engineering.
2. ASTM F37B Sealability
   ASTM Fuel A (isooctane):
   Gasket load = 500 psi (3.5 N/mm²), Int. pressure = 9.8 psig (0.7 bar)
   Nitrogen:
   Gasket load = 3,000 psi (20.7 N/mm²), Int. pressure = 36 psig (2 bar)
3. DIN 3535 Part 4: Gas Permeability, cc/min. (1/16" thick)
   Nitrogen:
   Gasket load = 4,640 psi (32 N/mm²), Int. pressure = 580 psig (40 bar)
4. Saturated steam service guidelines:
   - For optimal performance, use thinner gaskets when possible.
   - Maximum recommended assembly stress = 4,980 psi.
   - Preferred assembly stress = 6,000 psi to 10,500 psi.
   - Retorque the bolts/studs prior to pressurizing the assembly.
   - If the service is superheated steam, contact Applications Engineering.

[1] For steam service above 150 psig contact Applications Engineering

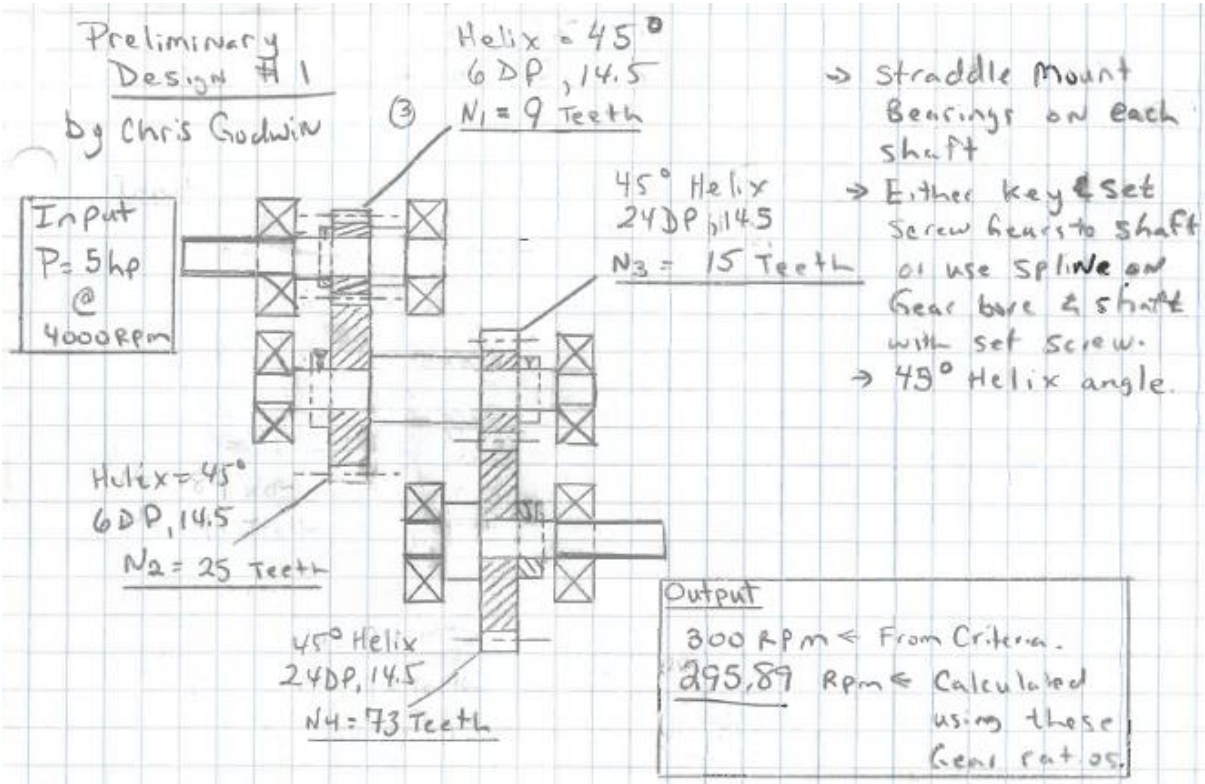
[2] Values do not constitute specification limits

Garlock

Leaders in Sealing Integrity

Appendix II

Gear box Design 1



Gear Options

Spur – Pros: Cheap to Manufacture, No Axial Forces
Cons: Noisy, More Stress on Gears
Uses: Low torque, Low speed Applications

Helical – Pros: Better tooth meshing, Transmit more Power
Less Noise than Spur, less vibration
Cons: Large thrust Forces along shaft (Axial)
higher cost.

Double Helical → Pros: Similar to Helical Benefits but without Axial thrust Forces.

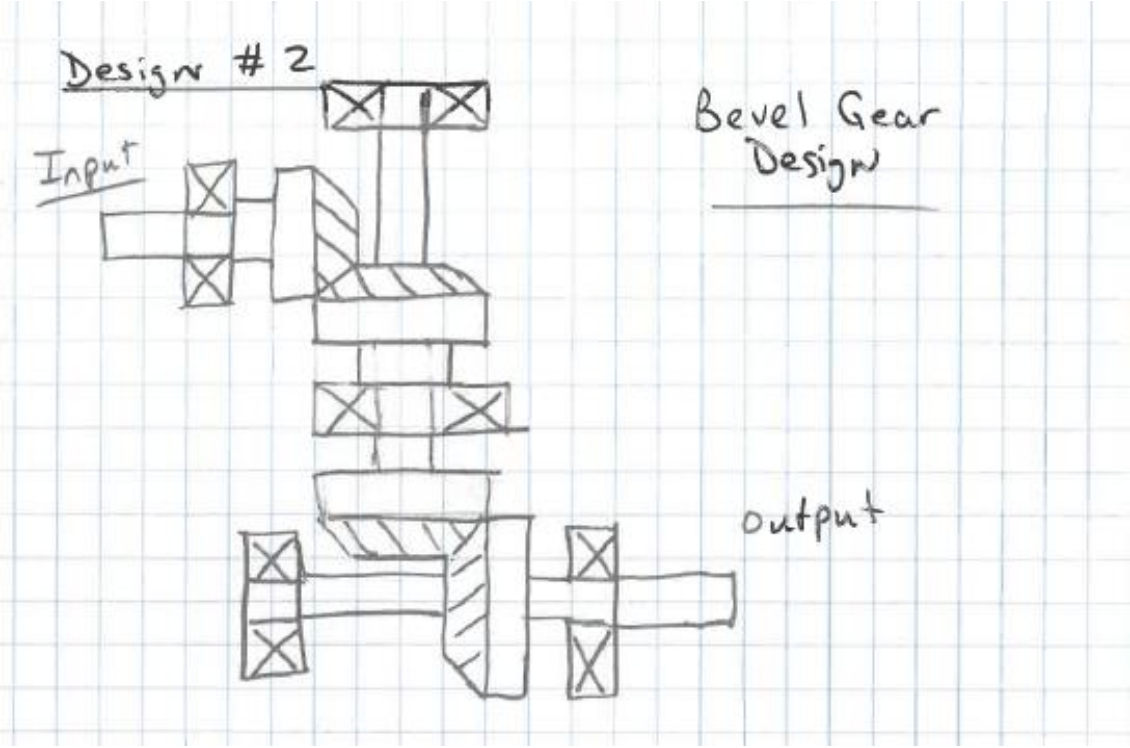
Cons: High Cost / Harder to make.

$T_{Out} = 118.68\ N \cdot m$ (winch)

$T_{In} = 8.9\ N \cdot m$

Appendix III

Gear box Design 2



Appendix IV

Image may differ from product. See technical specification for details.

W 61801

Stainless steel deep groove ball bearing

Stainless steel single row deep groove ball bearings provide greater chemical and corrosion resistance. As with deep groove ball bearings generally, they are particularly versatile, have low friction and are optimized for low noise and low vibration, which enables high rotational speeds. They accommodate radial and axial loads in both directions, are easy to mount, and require less maintenance than many other bearing types.

- Greater chemical and corrosion resistance
- Simple, versatile and robust design
- Low friction and high-speed capability
- Accommodate radial and axial loads in both directions
- Require little maintenance

Overview

Dimensions

Bore diameter	0.472 in
Outside diameter	0.827 in
Width	0.197 in

Performance

Basic dynamic load rating	339 lbf
Basic static load rating	202 lbf
Reference speed	70 000 r/min
Limiting speed	43 000 r/min

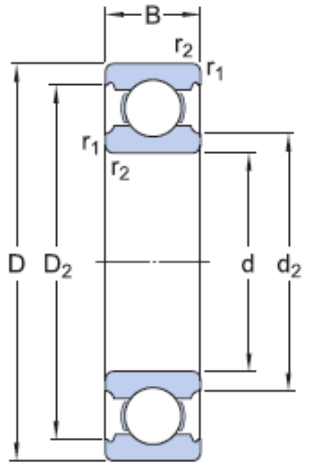
Properties

Filling slots	Without
Number of rows	1
Locating feature, bearing outer ring	None
Bore type	Cylindrical
Cage	Sheet metal
Matched arrangement	No
Radial internal clearance	CN
Tolerance class	Normal
Material, bearing	Stainless steel
Coating	Without
Sealing	Without
Lubricant	None
Relubrication feature	Without

Logistics

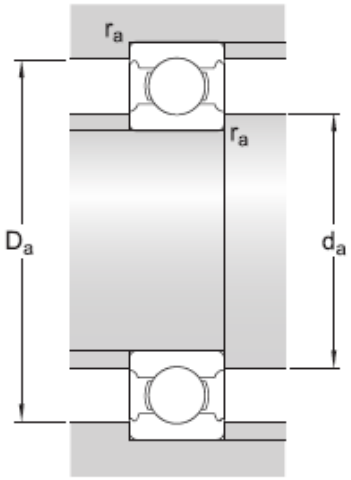
Product net weight	0.01257 lb
eClass code	23-05-08-01
UNSPSC code	31171504

Technical specification



Dimensions

d	0.472 in	Bore diameter
D	0.827 in	Outside diameter
B	0.197 in	Width
d_2	≈ 0.543 in	Recess diameter
D_2	≈ 0.754 in	Recess diameter
$r_{1,2}$	min. 0.012 in	Chamfer dimension



Abutment dimensions

d_a	min. 0.531 in	Diameter of shaft abutment
D_a	max. 0.768 in	Diameter of housing abutment
r_a	max. 0.012 in	Radius of shaft or housing fillet

Calculation data

Basic dynamic load rating	C	339 lbf
Basic static load rating	C_0	202 lbf
Fatigue load limit	P_u	8.8 lbf
Reference speed		70 000 r/min
Limiting speed		43 000 r/min
Minimum load factor	k_r	0.02
Calculation factor	f_0	13.4

Appendix V

Image may differ from product. See technical specification for details.

16101

Deep groove ball bearing

Single row deep groove ball bearings are particularly versatile, have low friction and are optimized for low noise and low vibration, which enables high rotational speeds. They accommodate radial and axial loads in both directions, are easy to mount, and require less maintenance than many other bearing types.

- Simple, versatile and robust design
- Low friction
- High-speed capability
- Accommodate radial and axial loads in both directions
- Require little maintenance

Dimensions

Bore diameter	0.472 in
Outside diameter	1.181 in
Width	0.315 in

Performance

Basic dynamic load rating	1 140 lbf
Basic static load rating	531 lbf
Reference speed	60 000 r/min
Limiting speed	38 000 r/min

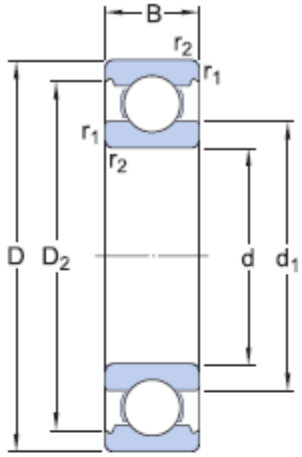
Properties

Filling slots	Without
Number of rows	1
Locating feature, bearing outer ring	None
Bore type	Cylindrical
Cage	Sheet metal
Matched arrangement	No
Radial internal clearance	CN
Tolerance class	Class P6 (P6)
Material, bearing	Bearing steel
Coating	Without
Sealing	Without
Lubricant	None
Relubrication feature	Without

Logistics

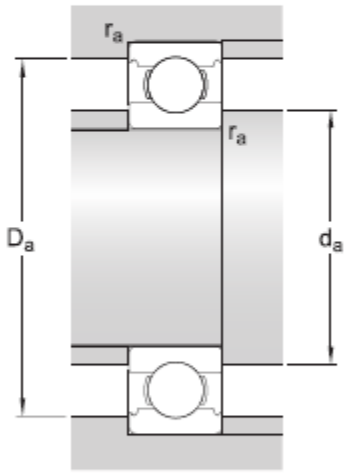
Product net weight	0.05622 lb
eClass code	23-05-08-01
UNSPSC code	31171504

Technical specification



Dimensions

d	0.472 in	Bore diameter
D	1.181 in	Outside diameter
B	0.315 in	Width
d_1	≈ 0.669 in	Shoulder diameter
D_2	≈ 0.9732 in	Recess diameter
$r_{1,2}$	min. 0.012 in	Chamfer dimension



Abutment dimensions

d_a	min. 0.567 in	Diameter of shaft abutment
D_a	max. 1.057 in	Diameter of housing abutment
r_a	max. 0.012 in	Radius of shaft or housing fillet

Calculation data

Basic dynamic load rating	C	1 140 lbf
Basic static load rating	C_0	531 lbf
Fatigue load limit	P_u	22 lbf
Reference speed		60 000 r/min
Limiting speed		36 000 r/min
Minimum load factor	k_r	0.025
Calculation factor	f_0	13

Appendix VI

Image may differ from product. See technical specification for details.

W 61903

Stainless steel deep groove ball bearing

Stainless steel single row deep groove ball bearings provide greater chemical and corrosion resistance. As with deep groove ball bearings generally, they are particularly versatile, have low friction and are optimized for low noise and low vibration, which enables high rotational speeds. They accommodate radial and axial loads in both directions, are easy to mount, and require less maintenance than many other bearing types.

- Greater chemical and corrosion resistance
- Simple, versatile and robust design
- Low friction and high-speed capability
- Accommodate radial and axial loads in both directions
- Require little maintenance

Overview

Dimensions

Bore diameter	0.669 in
Outside diameter	1.181 in
Width	0.276 in

Performance

Basic dynamic load rating	892 lbf
Basic static load rating	573 lbf
Reference speed	50 000 r/min
Limiting speed	32 000 r/min

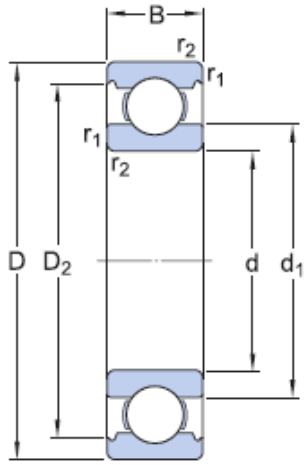
Properties

Filling slots	Without
Number of rows	1
Locating feature, bearing outer ring	None
Bore type	Cylindrical
Cage	Sheet metal
Matched arrangement	No
Radial internal clearance	CN
Tolerance class	Normal
Material, bearing	Stainless steel
Coating	Without
Sealing	Without
Lubricant	None
Relubrication feature	Without

Logistics

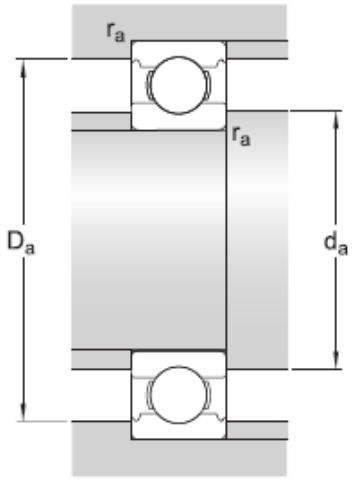
Product net weight	0.03086 lb
eClass code	23-05-08-01
UNSPSC code	31171504

Technical specification



Dimensions

d	0.669 in	Bore diameter
D	1.181 in	Outside diameter
B	0.276 in	Width
d_1	≈ 0.827 in	Shoulder diameter
D_2	≈ 1.094 in	Recess diameter
$r_{1,2}$	min. 0.012 in	Chamfer dimension



Abutment dimensions

d_a	min. 0.748 in	Diameter of shaft abutment
D_a	max. 1.122 in	Diameter of housing abutment
r_a	max. 0.012 in	Radius of shaft or housing fillet

Calculation data

Basic dynamic load rating	C	892 lbf
Basic static load rating	C_0	573 lbf
Fatigue load limit	P_u	24 lbf
Reference speed		50 000 r/min
Limiting speed		32 000 r/min
Minimum load factor	k_r	0.025
Calculation factor	f_0	14.8

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