AMERICAN UNIVERSITY OF BEIRUT
MAROUN SEMAAN FACULTY OF ENGINEERING & ARCHITECTURE

MECH 420 – Mechanical Design 1

Steering System - 2023 Supermileage Car

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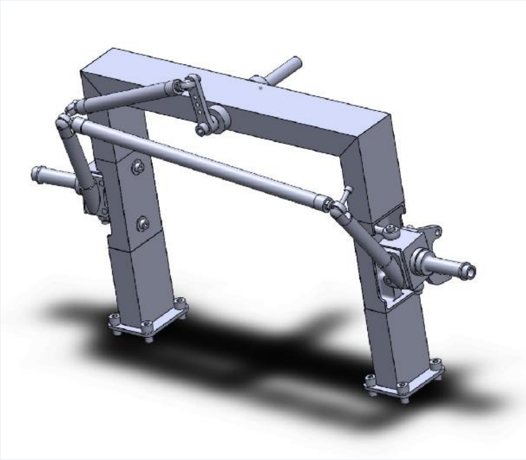
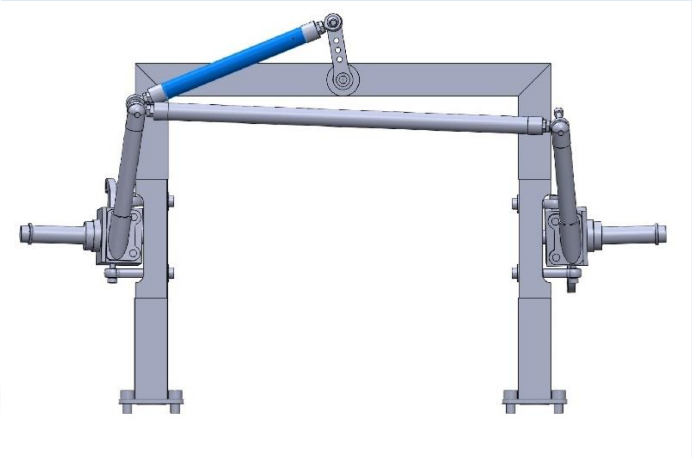
Introduction

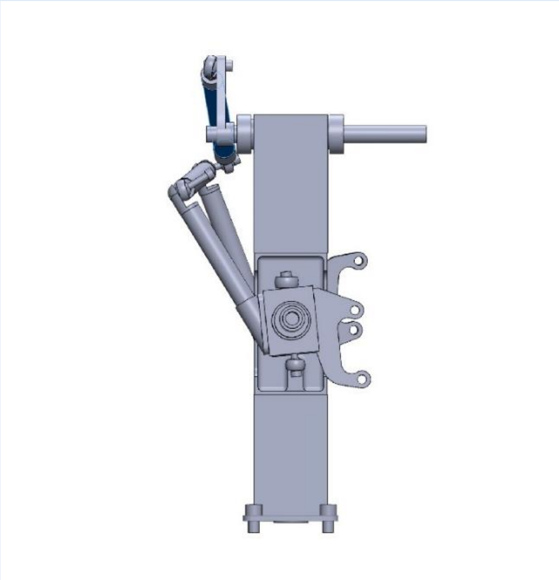
We are working on the final design for the Supermileage project at the American University of Beirut (AUB). Our main goal was to ensure that our design adhered to the supermileage regulations and placed a strong emphasis on being lightweight. Specifically, our attention was directed towards the steering shaft, as it experiences the most stress within the system. This report delves into the details of our approach and considerations in optimizing the steering shaft for improved performance. Find the assembled Design Below.



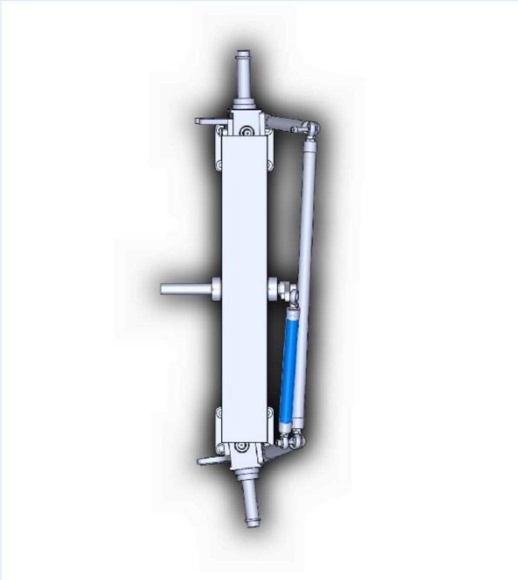
Isometric View



Front View



Side View



Top View

Literature Review

The steering system is a critical component in the design and operation of supermileage cars. Achieving optimal performance in supermileage vehicles not only requires exceptional fuel efficiency but also precise steering control. This literature review aims to explore the existing research on the optimization of steering systems for supermileage cars. The review will discuss various approaches, methodologies, and technologies used to improve steering efficiency while maintaining safety. Supermileage car steering systems prioritize minimal energy consumption and lightweight design to maximize fuel efficiency. The components must be lightweight, ensuring precision and responsiveness for effective navigation in various conditions. The design of supermileage car steering systems emphasizes key factors to enhance overall efficiency. Prioritizing lightweight components, such as aluminum or composite materials, not only reduces the vehicle's weight but also involves optimizing the design to minimize mass without compromising structural integrity. Efficient steering geometry, encompassing caster, camber, and toe settings, is crucial for minimizing rolling resistance and ensuring efficient handling. To further enhance aerodynamics, computer simulations or wind tunnel testing can fine-tune the effects of steering system components. Additionally, the integration of low-friction bearings and joints plays a pivotal role in reducing mechanical losses, while sealed and maintenance-free components are employed to minimize energy-robbing friction within the steering system. This comprehensive approach underscores the commitment to maximizing fuel efficiency in supermileage vehicles through thoughtful material selection, aerodynamic optimization, and the reduction of frictional losses. In conclusion, the optimization of steering systems for supermileage cars is a multifaceted challenge that requires a careful balance between achieving exceptional fuel efficiency and maintaining precise control and safety. This literature review has shed light on the critical requirements of steering systems in supermileage vehicles, emphasizing the need for minimal energy consumption, lightweight design, precision, and safety. As well as showing various optimization paths.

Analysis of Forces on the shaft

I. Assumptions

Material Selected: Aluminum 2024 T3, with S_y=50 kpsi and S_{ut}=68 kpsi

The main shaft is mostly subjected to torque and has the most stress concentration, so it will be the center of our study.

The average torque used for steering is 20 N.m = 180 lb.in on the whole shaft.

The initial diameter assumed for the shaft is:

$$d = 0.5\ in$$

The bearing is assumed to be of the following dimensions (width = 1in, OD = 1.125 in, Bore = 0.5 in).

II. FBD

The following figures are the FBD of the shaft in the x-y and x-z planes.

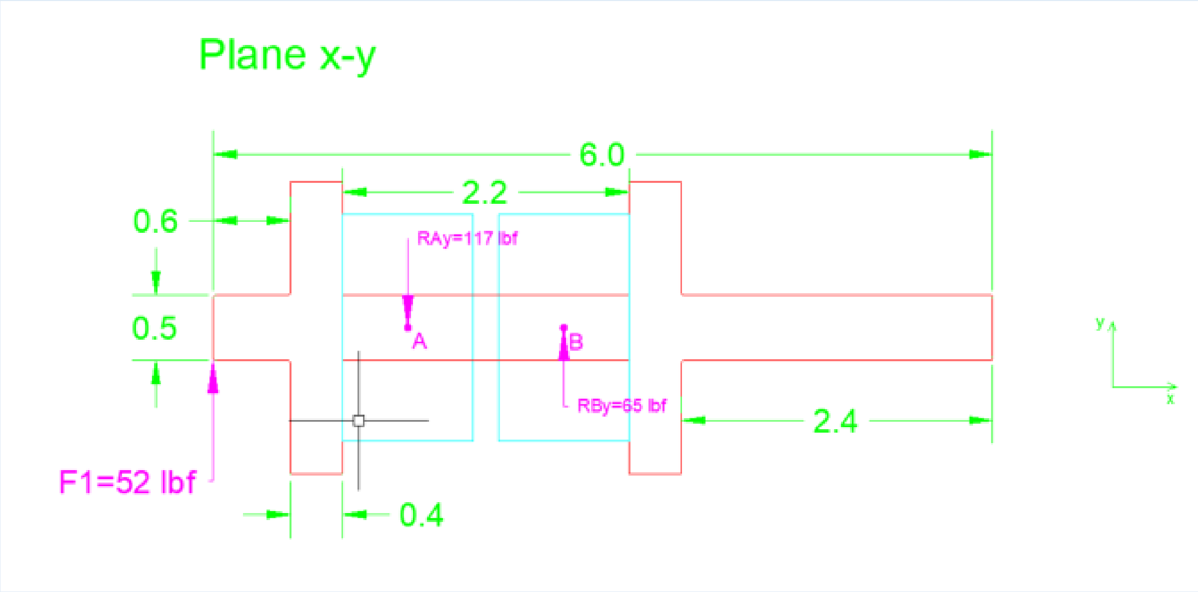


FIGURE 1. FBD OF THE SHAFT IN THE X-Y PLANE

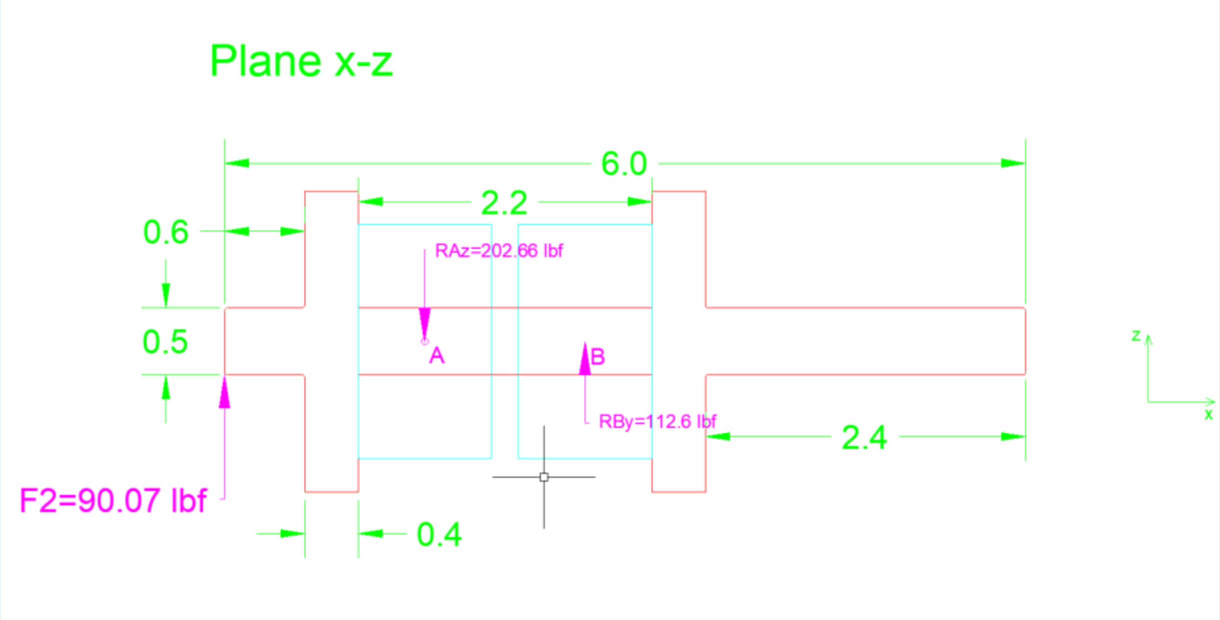


FIGURE 2. FBD OF THE SHAFT IN THE X-Z PLANE

The maximum angle between Part A and the short rod is: $\phi_{max} = 120°$. At this angle, the forces applied on the shaft are maximal, therefore being the most critical position of our shaft.

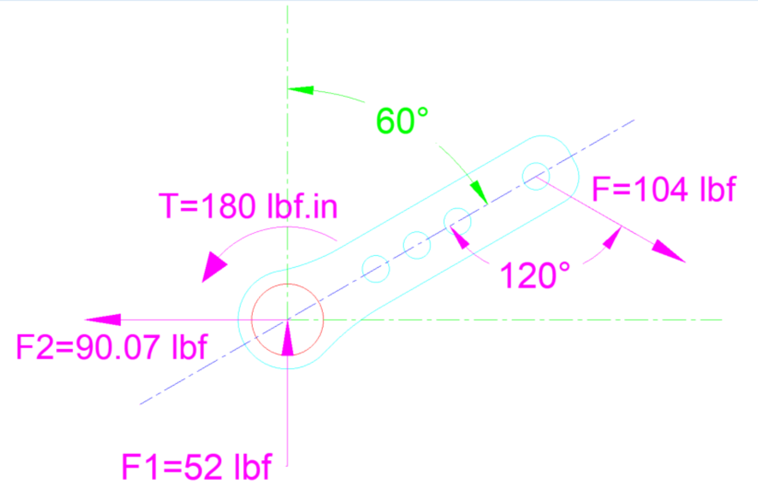


FIGURE 3. FBD OF PART A

We have,

$$2F\cos 30° = T$$

Hence,

$$F = 104\ lbf$$

$$F_1 = F\cos 60° = 52\ lbf$$

$$F_2 = F\cos 30° = 90.07\ lbf$$

Looking at the FBD in figures 7 and 8,

In the x-y plane,

$$\sum M_A = 0$$

$$1.5F_1 = -1.2R_{By}$$

$$R_{By} = 65\ lbf\ \uparrow$$

$$\sum F_y = 0$$

$$R_{Ay} = 117\ lbf\ \downarrow$$

In the x-z plane,

$$\sum M_A = 0$$

$$1.5F_2 = 1.2R_{Bz}$$

$$R_{Bz} = 112.6\ lbf\ \uparrow$$

$$\sum F_z = 0$$

$$R_{Az} = 202.66\ lbf\ \downarrow$$

We can then deduce the shear force and bending moment diagrams in the two different planes as following.

In the x-y plane:

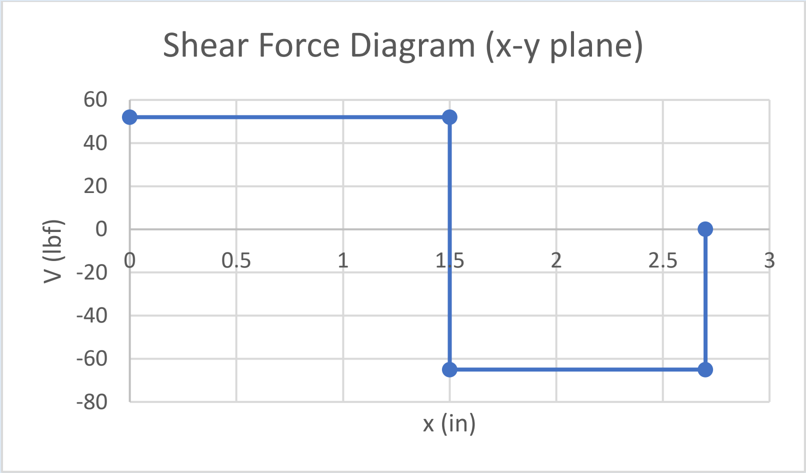


FIGURE 4. SHEAR FORCE DIAGRAM (X-Y PLANE)

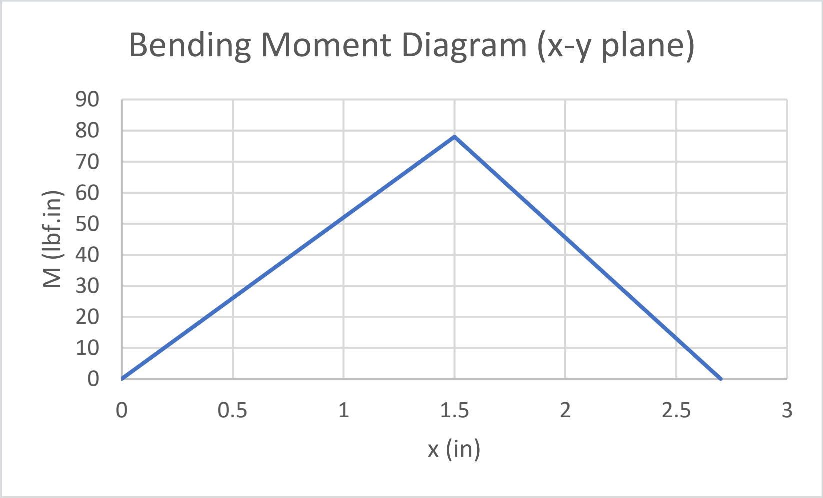


FIGURE 5. BENDING MOMENT DIAGRAM (X-Y PLANE)

In the x-z plane:

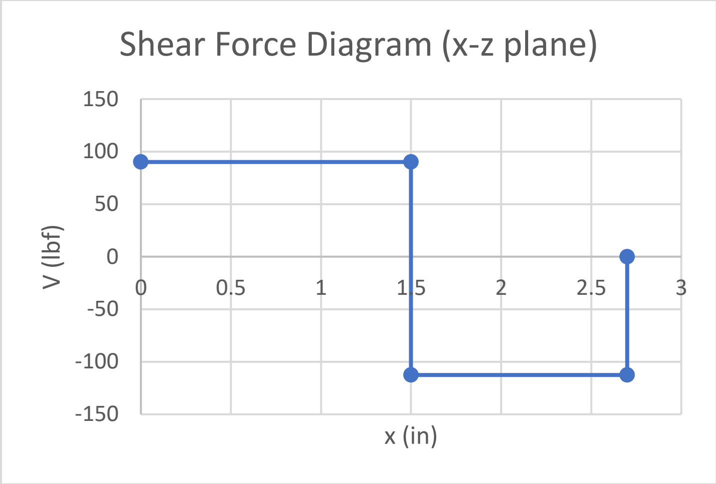


FIGURE 6. SHEAR FORCE DIAGRAM (X-Z PLANE)

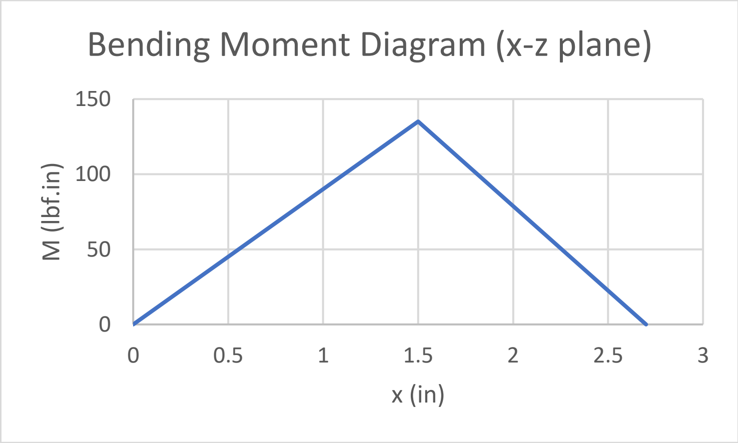


FIGURE 7. BENDING MOMENT DIAGRAM (X-Z PLANE)

Point A is a critical point

At shoulder A,

$$M_{max} = 104\ lbf.in$$

$$M_{min} = -104\ lbf.in$$

:

$$M_m = \frac{M_{min} + M_{max}}{2} = 0$$

$$M_a = \frac{M_{min} - M_{max}}{2} = 104\ lbf.in$$

And,

$$T_{max} = 180\ lbf.in$$

$$T_{min} = -180\ lbf.in$$

Therefore,

$$T_m = \frac{T_{min} + T_{max}}{2} = 0$$

$$T_a = \frac{T_{min} - T_{max}}{2} = 180\ lbf.in$$

Hence,

$$M_m = 0$$

$$M_a = 104\ lbf.in$$

$$T_m = 0$$

$$T_a = 180\ lbf.in$$

Then, assuming the following:

$$r = 0.025\ in$$

$$r/d = \frac{0.025}{0.5} = 0.05$$

$$D/d = \frac{1.125}{0.5} = 2.25$$

Using the following figures,

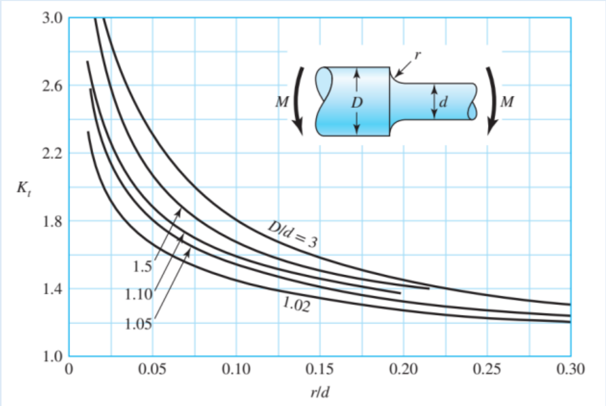


FIGURE 8

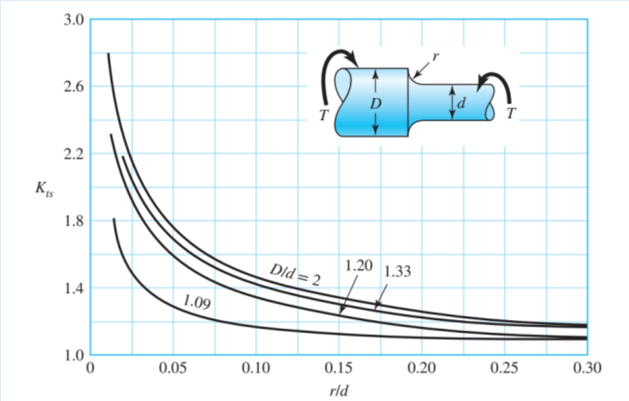


FIGURE 9

We get,

$$K_t = 2.2$$

$$K_{ts} = 1.8$$

With,

$$50 \leq S_{ut} \leq 220\ kpsi$$

From equation 6-35,

$$\sqrt{a}_s = 0.246 - 3.08 \times 10^{-3} \times S_{ut} + 1.51 \times 10^{-5} \times {S_{ut}}^2 - 2.67 \times 10^{-8} \times {S_{ut}}^3 = 0.098$$

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} = 0.203$$

From equation 6-36,

$$\sqrt{a}_s = 0.190 - 2.51 \times 10^{-3} \times S_{ut} + 1.35 \times 10^{-5} \times {S_{ut}}^2 - 2.67 \times 10^{-8} \times {S_{ut}}^3 = 0.073$$

$$q_s = \frac{1}{1 + \frac{\sqrt{a}_s}{\sqrt{r}}} = 0.255$$

Then,

$$K_f = 1 + q(K_t - 1) = 1.2436$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1.204$$

Therefore,

$$A = \sqrt{4(K_f M_a)^2 + 3(K_{fs} T_a)^2} = 455.86$$

$$B = \sqrt{4(K_f M_m)^2 + 3(K_{fs} T_m)^2} = 0$$

Checking for yielding,

$$\sigma'_a = \sqrt{\left(\frac{32 K_f M_a}{\pi d^3}\right)^2 + 3\left(\frac{16 K_{fs} T_a}{\pi d^3}\right)^2} = 18.57\ kpsi$$

$$\sigma'_m = 0$$

Hence,

$$n_y = \frac{S_y}{\sigma'_a} = \frac{50}{18.57} = 2.69 > 1, \text{ no yielding predicted}$$

Checking for fatigue, while assuming the shaft machined,

$$S'_e = 0.5 S_{ut} = 34\ kspi, \text{ since } S_{ut} < 200 \text{ kpsi}$$

$$k_a = aS_{ut}^b = 2.00 \times 68^{-0.217} = 0.8$$

$$k_b = 0.879d^{-0.107} = 0.947, \text{ since } 0.3 < d = 0.5 < 2\text{in}$$

$$k_c = k_d = k_e = 1$$

Hence,

$$S_e = k_a k_b k_c k_d k_e S'_e = 25.76\ kpsi$$

$$n = \frac{\pi d^3}{16}\left(\frac{A}{S_e} + \frac{B}{S_{ut}}\right)^2 = \frac{\pi \times 0.5^3}{16} \times \frac{25.76 \times 10^3}{455.86} = 1.387$$

III. Bearing Selection:

Using the following catalog data for ball bearings:

Dimensions and Load Ratings for Single-Row 02-Series Deep-Groove and Angular-Contact Ball Bearings

Bore, mm	OD, mm	Width, mm	Fillet Radius, mm	Shoulder Diameter, mm d_s	d_H	Load Ratings, kN: Deep Groove C_{10}	C_0	Angular Contact C_{10}	C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

Table 1.

We will choose a deep groove ball bearing, with a=3, x_0=0.02, θ=4.459, b=1.483, $L_R=L_{10}=10^6$, R_D=0.99 & a_f =1.2, and assuming a desired life $L_D=10^8$

We have: F_D =104 lbf and $x_D=L_D/L_R$=100.

So,

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)[1 - R_D]^{1/b}}\right]^{1/a} = 961.2\ lbf = 4.3\ kN$$

We choose: $C_{10} = 5.07\ kN$ with a bore of 10mm < 0.5 in.

Therefore, we will choose a ball bearing with a bore of 15mm, and will increase the diameter of our shaft to match the bore diameter, hence becoming a shaft of diameter d=15 mm.

The new selected bearing is then a deep groove ball bearing with:

$$Bore = 15\ mm = 0.59\ in$$

$$OD = 35\ mm = 1.378\ in$$

$$w = 11\ mm = 0.433\ in$$

Subsequently, our design will need to be modified as follows:

	Old (in)	New (in)
d	0.5	0.59
OD (shoulder)	1.125	1.5
w	1	0.433

The new shaft will look as the following figures:

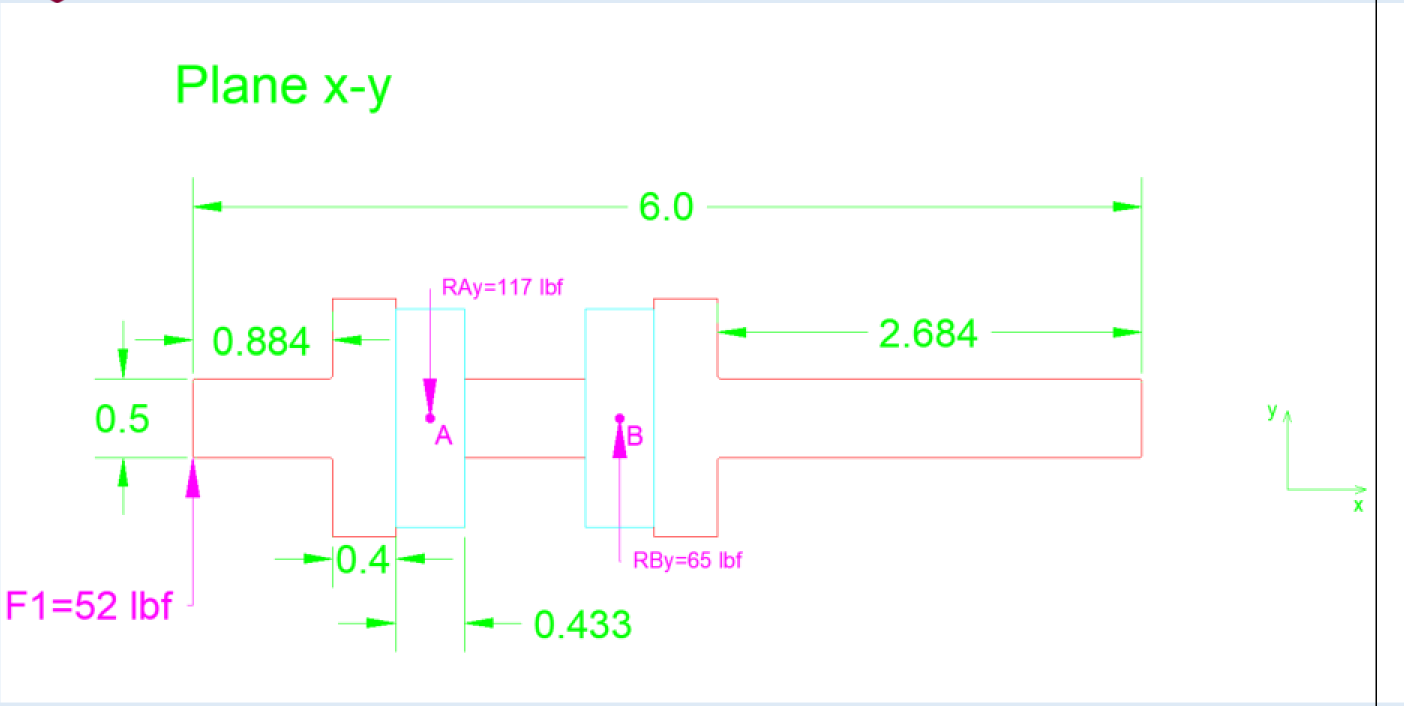


FIGURE 10. UPDATED FBD OF THE SHAFT IN THE X-Y PLANE

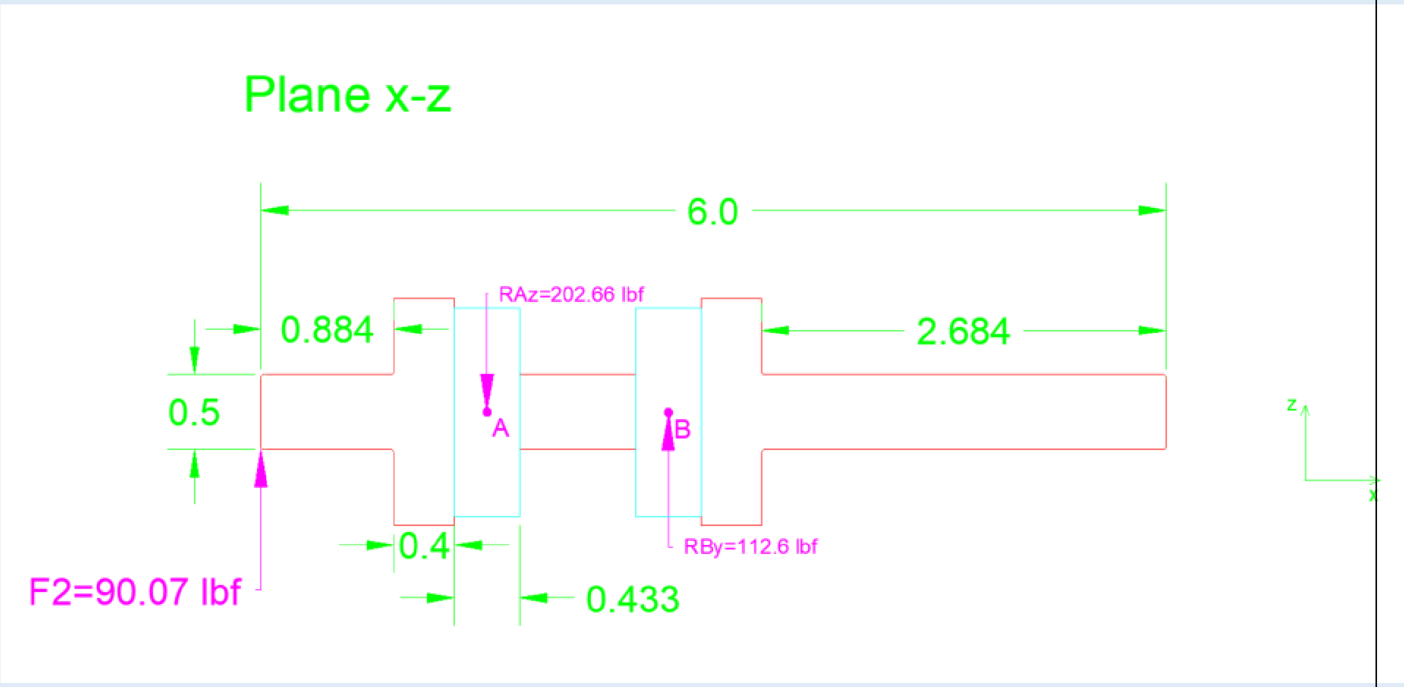


FIGURE 11. UPDATED FBD OF THE SHAFT IN THE X-Z PLANE

The shear and bending moment diagrams will stay the same, but the critical point at the shoulder will be shifted to the right. Therefore, we need to make sure the design is still valid at the shoulder.

At shoulder A:

$$M_{A_{xy}} = 66.742\ lbf.in$$

$$M_{A_{xz}} = 115.6\ lbf.in$$

Hence,

$$M_{max} = 133.488\ lbf.in$$

$$M_{min} = -133.488\ lbf.in$$

$$T_{max} = 180\ lbf.in$$

$$T_{min} = -180\ lbf.in$$

Then,

$$M_m = 0$$

$$M_a = 133.488\ lbf.in$$

$$T_m = 0$$

$$T_a = 180\ lbf.in$$

With,

$$r = 0.025\ in$$

$$r/d = \frac{0.025}{0.59} = 0.0424$$

$$D/d = \frac{1.15}{0.59} = 2.542$$

Using the following figures again,

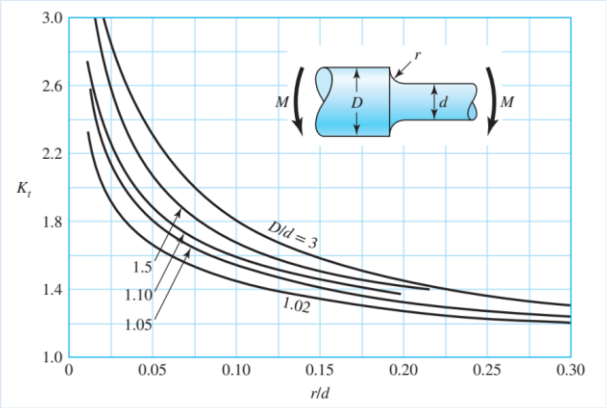


FIGURE 12

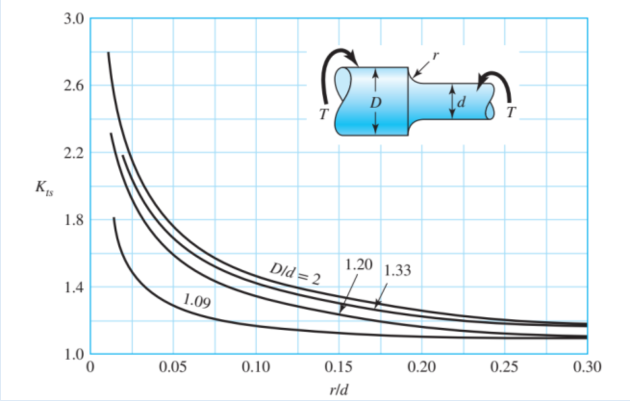


FIGURE 13

We get,

$$K_t = 2.4$$

$$K_{ts} = 1.9$$

With,

$$50 \leq S_{ut} \leq 220\ kpsi$$

From equation 6-35,

$$\sqrt{a}_s = 0.246 - 3.08 \times 10^{-3} \times S_{ut} + 1.51 \times 10^{-5} \times S_{ut}{}^2 - 2.67 \times 10^{-8} \times S_{ut}{}^3 = 0.098$$

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} = 0.203$$

From equation 6-36,

$$\sqrt{a}_s = 0.190 - 2.51 \times 10^{-3} \times S_{ut} + 1.35 \times 10^{-5} \times {S_{ut}}^2 - 2.67 \times 10^{-8} \times {S_{ut}}^3 = 0.073$$

$$q_s = \frac{1}{1 + \frac{\sqrt{a}_s}{\sqrt{r}}} = 0.255$$

Then,

$$K_f = 1 + q(K_t - 1) = 1.2842$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1.2295$$

Therefore,

$$A = \sqrt{4(K_f M_a)^2 + 3(K_{fs} T_a)^2} = 514.28$$

$$B = \sqrt{4(K_f M_m)^2 + 3(K_{fs} T_m)^2} = 0$$

Checking for yielding,

$$\sigma'_a = \sqrt{\left(\frac{32 K_f M_a}{\pi d^3}\right)^2 + 3\left(\frac{16 K_{fs} T_a}{\pi d^3}\right)^2} = 12.7\ kpsi$$

$$\sigma'_m = 0$$

Hence,

$$n_y = \frac{S_y}{\sigma'_a} = \frac{50}{12.7} = 3.94 > 1, \text{ no yielding predicted}$$

Checking for fatigue, while assuming the shaft machined,

$$S'_e = 0.5S_{ut} = 34\ kspi, \text{ since } S_{ut} < 200 \text{ kpsi}$$

$$k_a = aS_{ut}^b = 2.00 \times 68^{-0.217} = 0.8$$

$$k_b = 0.879d^{-0.107} = 0.930, \text{ since } 0.3 < d = 0.5 < 2\text{in}$$

$$k_c = k_d = k_e = 1$$

Hence,

$$S_e = k_a k_b k_c k_d k_e S'_e = 25.3\ kpsi$$

$$n = \frac{\pi d^3}{16}\left(\frac{A}{S_e} + \frac{B}{S_{ut}}\right)^2 = \frac{\pi \times 0.59^3}{16} \times \frac{25.3 \times 10^3}{514.28} = 1.98$$

IV. Clearance

Using the following table (table 2), we will choose the sliding fit (H7/g6), with $d = D = 0.59\ in$.

Type of Fit	Description	Symbol
Clearance	*Loose running fit*: for wide commercial tolerances or allowances on external members (Fit with the largest clearance, Suitable for applications where accuracy is not of the utmost importance)	H11/c11
	Free running fit: not for use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal pressures	H9/d9
	Close running fit: for running on accurate machines and for accurate location at moderate speeds and journal pressures	H8/f7
	Sliding fit: Leaves a small clearance for high accuracy while maintaining ease of assembly. Parts will turn and slide quite freely.	H7/g6
	Locational clearance fit: provides minimal clearance for high accuracy requirements. The assembly does not need any force and the mating parts can turn and slide freely with lubrication.	H7/h6

Table 2.

Now, from the following table (Table 3), we find the tolerance grades:

Basic Sizes	Tolerance Grades					
	IT6	IT7	IT8	IT9	IT 10	IT 11
0 to 0.12	0.0002	0.0004	0.0006	0.0010	0.0016	0.0024
0.12 to 0.24	0.0003	0.0005	0.0007	0.0012	0.0019	0.0030
0.24 to 0.40	0.0004	0.0006	0.0009	0.0014	0.0023	0.0035
0.40 to 0.72	0.0004	0.0007	0.0011	0.0017	0.0028	0.0043
0.72 to 1.20	0.0005	0.0008	0.0013	0.0020	0.0033	0.0051
1.20 to 2.00	0.0006	0.0010	0.0015	0.0024	0.0039	0.0063
2.00 to 3.20	0.0007	0.0012	0.0018	0.0029	0.0047	0.0075
3.20 to 4.80	0.0009	0.0014	0.0021	0.0034	0.0055	0.0087
4.80 to 7.20	0.0010	0.0016	0.0025	0.0039	0.0063	0.0098
7.20 to 10.00	0.0011	0.0018	0.0028	0.0045	0.0073	0.0114
10.00 to 12.60	0.0013	0.0020	0.0032	0.0051	0.0083	0.0126
12.60 to 16.00	0.0014	0.0022	0.0035	0.0055	0.0091	0.0 142

Table 3.

Therefore,

$$IT7 = \Delta D = 0.0007$$

$$IT6 = \Delta d = 0.0004$$

In addition, using the table below, we find the fundamental deviations:

Basic Sizes	Upper-Deviation Letter					Lower-Deviation Letter				
	c	d	f	g	h	k	n	p	s	u
0 to 0.12	− 0.0024	−0.0008	−0.0002	− 0.0001	0	0	+0.0002	+0.0002	+0.0006	+0.0007
0.12 to 0.24	− 0.0028	−0.0012	−0.0004	− 0.0002	0	0	+0.0003	+0.0005	+0.0007	+0.0009
0.24 to 0.40	−0.0031	− 0.0016	−0.0005	− 0.0002	0	0	+0.0004	+0.0006	+0.0009	+0.0011
0.40 to 0.72	− 0.0037	−0.0020	−0.0006	−0.0002	0	0	+0.0005	+0.0007	+0.0011	+0.0013
0.72 to 0.96	−0.0043	−0.0026	−0.0008	−0.0003	0	+0.0001	+0.0006	+0.0009	+0.0014	+0.0016
0.96 to 1.20	−0.0043	−0.0026	−0.0008	−0.0003	0	+0.0001	+0.0006	+0.0009	+0.0014	+ 0.0019
1.20 to 1.60	− 0.0047	−0.0031	−0.0010	− 0.0004	0	+0.0001	+0.0007	+ 0.0010	+ 0.0017	+ 0.0024
1.60 to 2.00	− 0.0051	− 0.0031	−0.0010	− 0.0004	0	+0.0001	+0.0007	+ 0.0010	+ 0.0017	+ 0.0028
2.00 to 2.60	− 0.0055	− 0.0039	−0.0012	− 0.0004	0	+0.0001	+ 0.0008	+ 0.0013	+ 0.0021	+0.0034
2.60 to 3.20	−0.0059	− 0.0039	−0.0012	−0.0004	0	+0.0001	+ 0.0008	+ 0.0013	+ 0.0023	+0.0040
3.20 to 4.00	− 0.0067	− 0.0047	− 0.0014	−0.0005	0	+0.0001	+ 0.0009	+ 0.0015	+ 0.0028	+0.0049
4.00 to 4.80	−0.0071	−0.0047	−0.0014	−0.0005	0	+0.0001	+ 0.0009	+ 0.0015	+ 0.0031	+0.0057
4.80 to 5.60	−0.0079	−0.0057	− 0.0017	− 0.0006	0	+0.0001	+ 0.0011	+ 0.0017	+ 0.0036	+0.0067
5.60 to 6.40	−0.0083	−0.0057	− 0.0017	− 0.0006	0	+0.0001	+ 0.0011	+ 0.0017	+0.0039	+0.0075
6.40 to 7.20	−0.0091	−0.0057	−0.0017	−0.0006	0	+0.0001	+ 0.0011	+ 0.0017	+0.0043	+0.0083
7.20 to 8.00	−0.0094	−0.0067	−0.0020	− 0.0006	0	+0.0002	+0.0012	+0.0020	+0.0048	+0.0093
8.00 to 9.00	−0.0102	−0.0067	−0.0020	− 0.0006	0	+0.0002	+0.0012	+0.0020	+0.0051	+0.0102
9.00 to 10.00	−0.0110	−0.0067	−0.0020	− 0.0006	0	+0.0002	+0.0012	+0.0020	+0.0055	+ 0.0112
10.00 to 11.20	−0.0118	−0.0075	−0.0022	− 0.0007	0	+0.0002	+0.0013	+0.0022	+0.0062	+ 0.0124
11.20 to 12.60	−0.0130	−0.0075	−0.0022	−0.0007	0	+0.0002	+0.0013	+0.0022	+0.0067	+0.0130
12.60 to 14.20	−0.0142	−0.0083	−0.0024	− 0.0007	0	+0.0002	+ 0.0015	+ 0.0024	+0.0075	+0.0154
14.20 to 16.00	− 0.0157	−0.0083	−0.0024	−0.0007	0	+0.0002	+ 0.0015	+ 0.0024	+0.0082	+0.0171

Table 4.

Hence,

$$\delta_F = -0.0002$$

Then,

$$d_{max} = d + \delta_F = 0.59 - 0.0002 = 0.5898\ in$$

$$d_{min} = d + \delta_F - \Delta d = 0.59 - 0.0002 - 0.0004 = 0.5894\ in$$

And,

$$D_{min} = D = 0.59\ in$$

$$D_{max} = D + \Delta D = 0.59 + 0.0007 = 0.5907\ in$$

V. Deflection

We’ll be calculating the angular deflection of the shaft.

From the simulation, we can extract that the displacement of a point on the surface of the shaft is on average $\Delta x = 4 \times 10^{-4}\ inches$ at the bearing.

The perimeter of the shaft is:

$$P = 2\pi r = \pi \times 0.59 = 1.8535\ in$$

Using proportionality,

Distance (in.)	Angle (degrees)
1.8535	360
4×10^{-4}	a

$$a = \frac{4 \times 10^{-4} \times 360}{1.8535} = 0.0777° = 0.00136\ rad$$

Slopes	
Tapered roller	0.0005 to 0.0012 rad
Cylindrical roller	0.0008 to 0.0012 rad
Deep-groove ball	0.001 to 0.003 rad
Spherical ball	0.026 to 0.052 rad
Self-align ball	0.026 to 0.052 rad
Uncrowned spur gear	<0.0005 rad

Table 5.

Our angular deflection falls within the allowable interval from table 5.

Drawings: Assemblies, Subassemblies, and Parts

The Steering Column assembly consists of five parts: two ball bearings, two shoulders, a steerer, a steering column, and a bolt to attach the steerer to the steering column. The Figure 14 below shows this assembly on it owns.

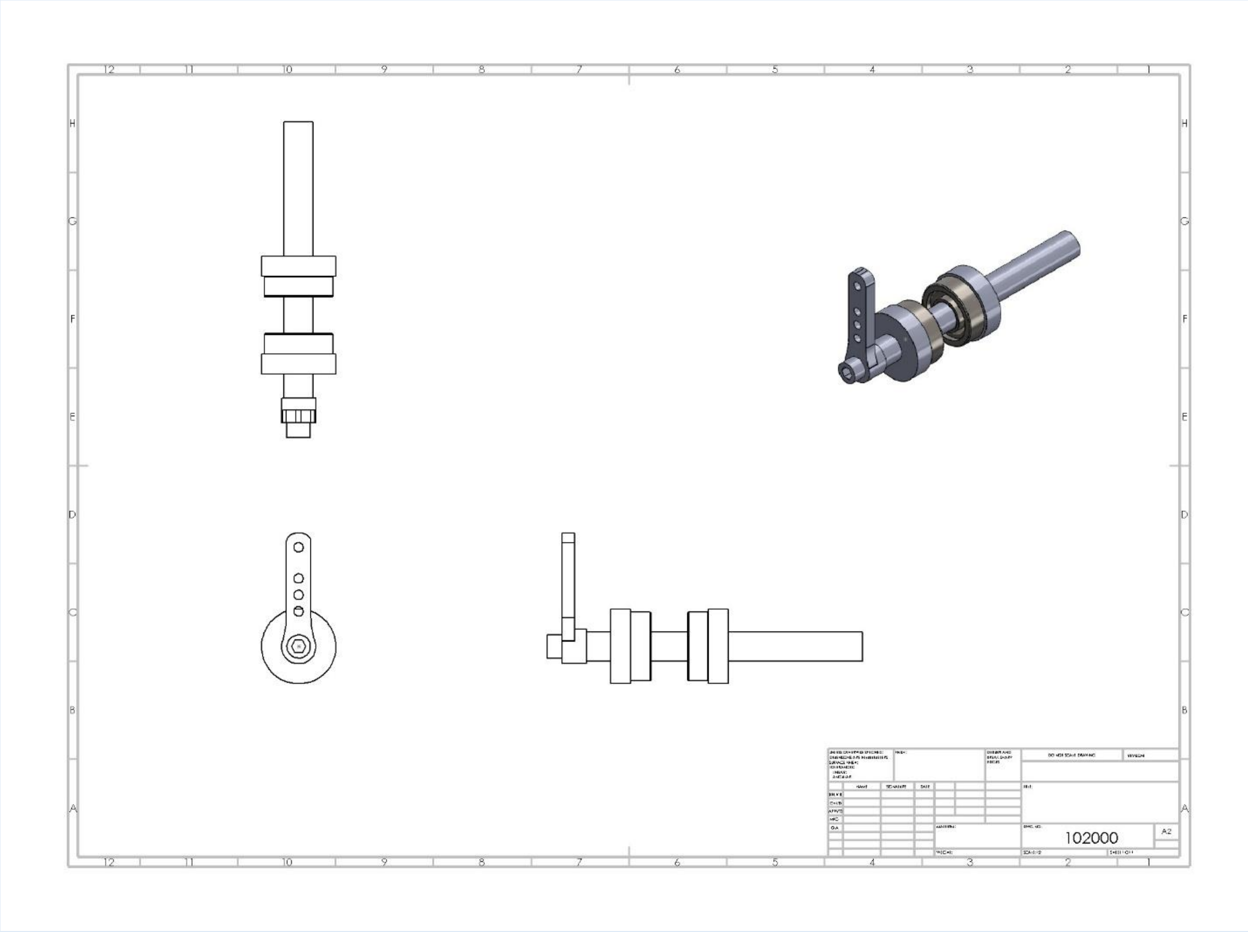


Figure 14. Steering System

The steering column is the component that transfers rotation of the steering wheel to the rotation of the steerer. Since weight is an important specification, this column will be made out of aluminum, but 2024 T3 aluminum. This material was chosen dues to its higher yield and shear strength. This feature can be seen in Figure 16.

The Steerer is a part that will translate the rotational movement of the steering column to the short tie rod as shown in the figure 15 below.

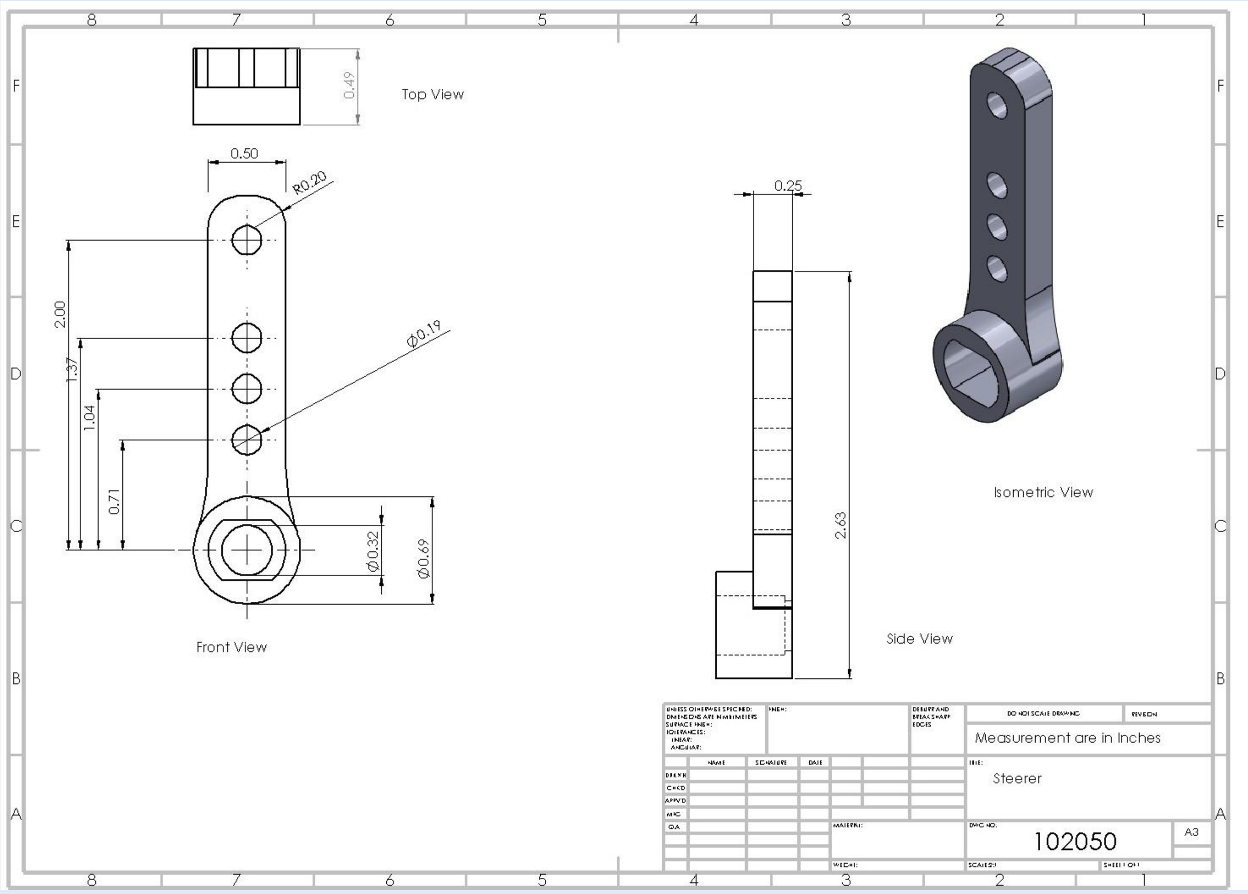


Figure 15. Steerer

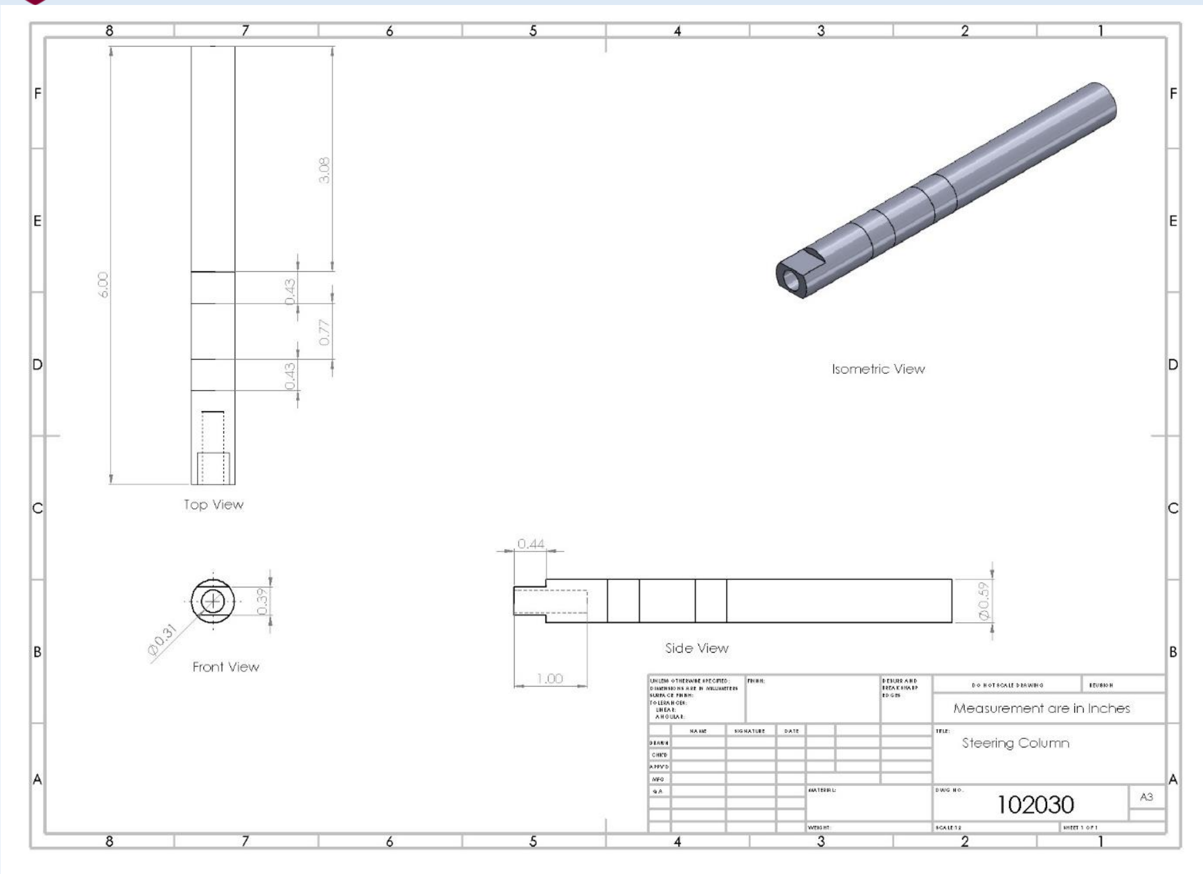


Figure 16. Steering Column

Bill of Materials

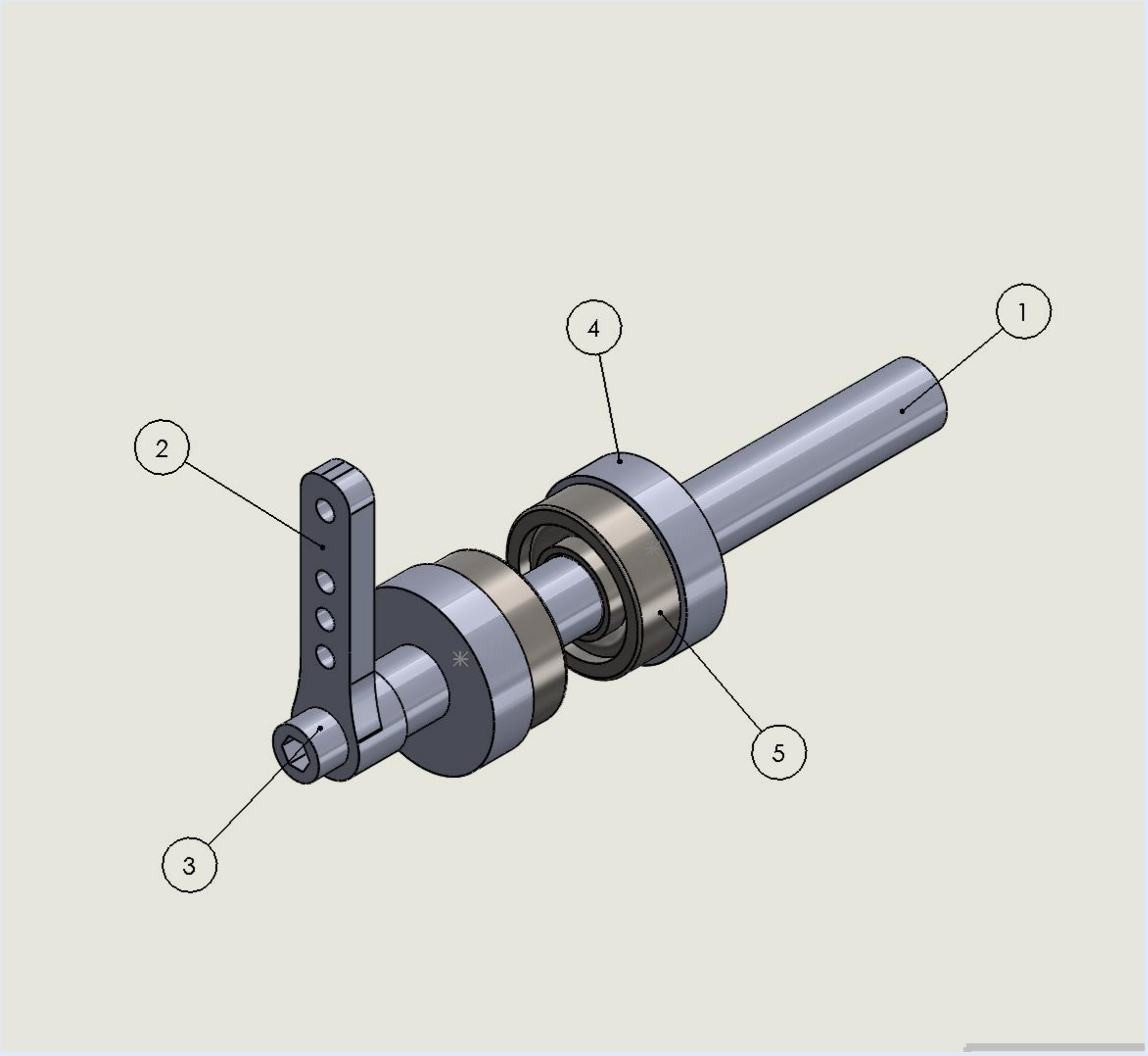
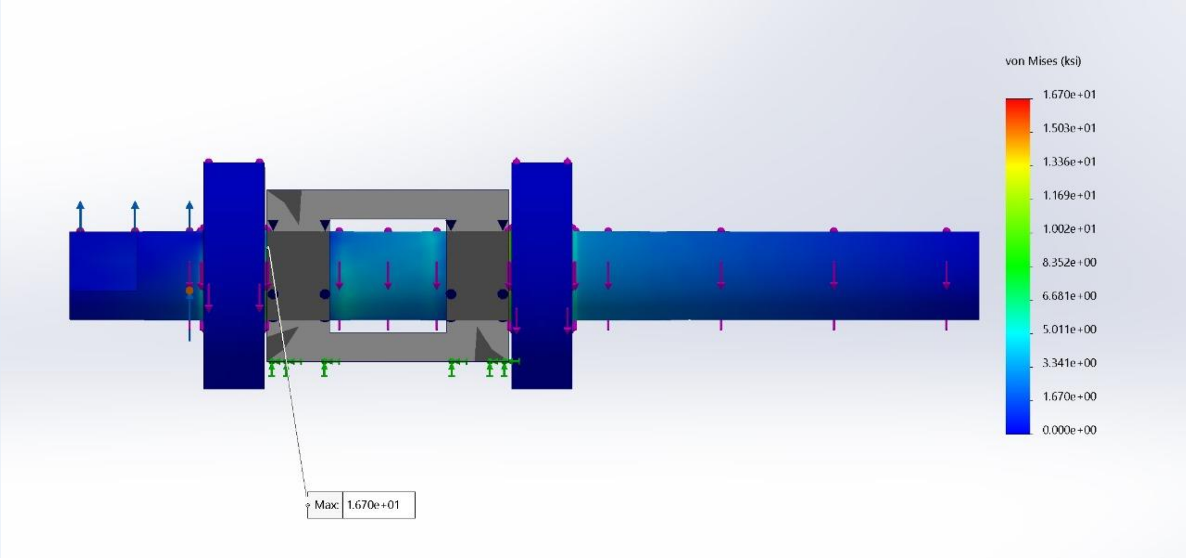
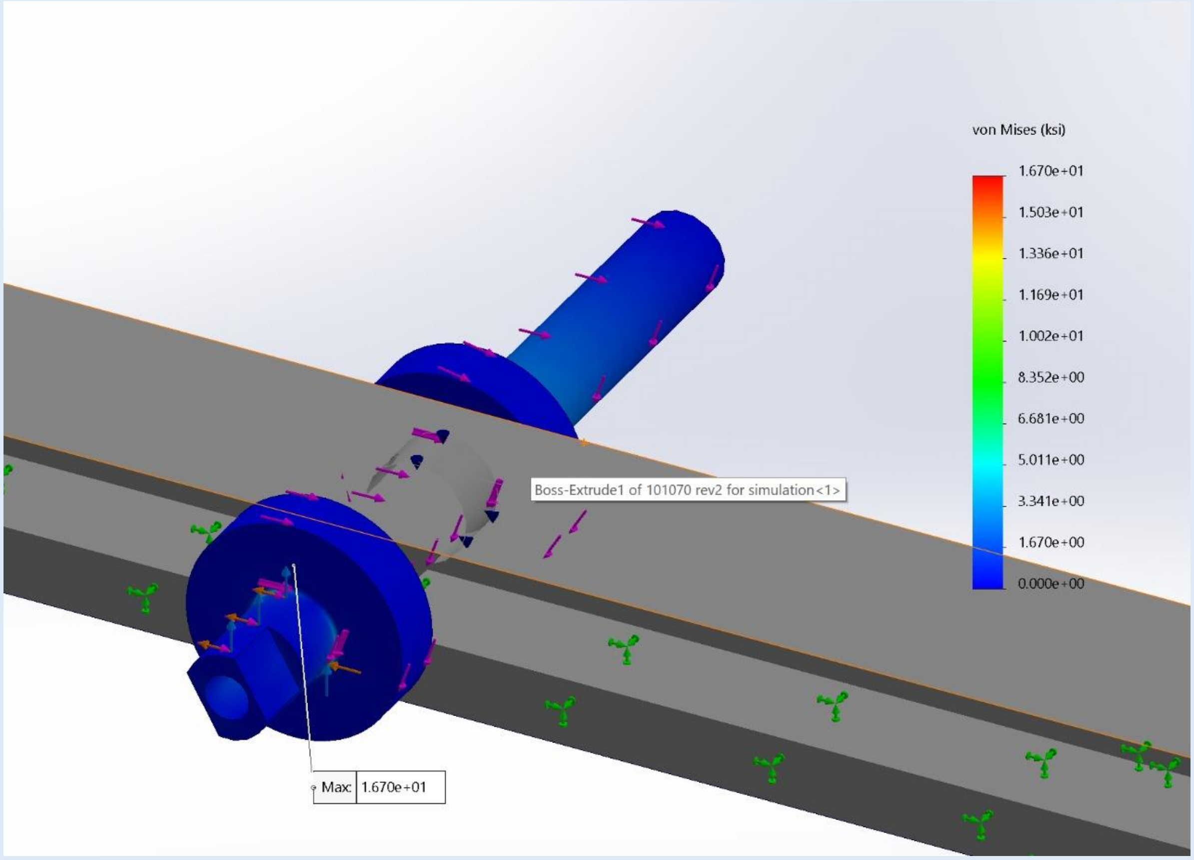


Fig.17. Steering Shaft

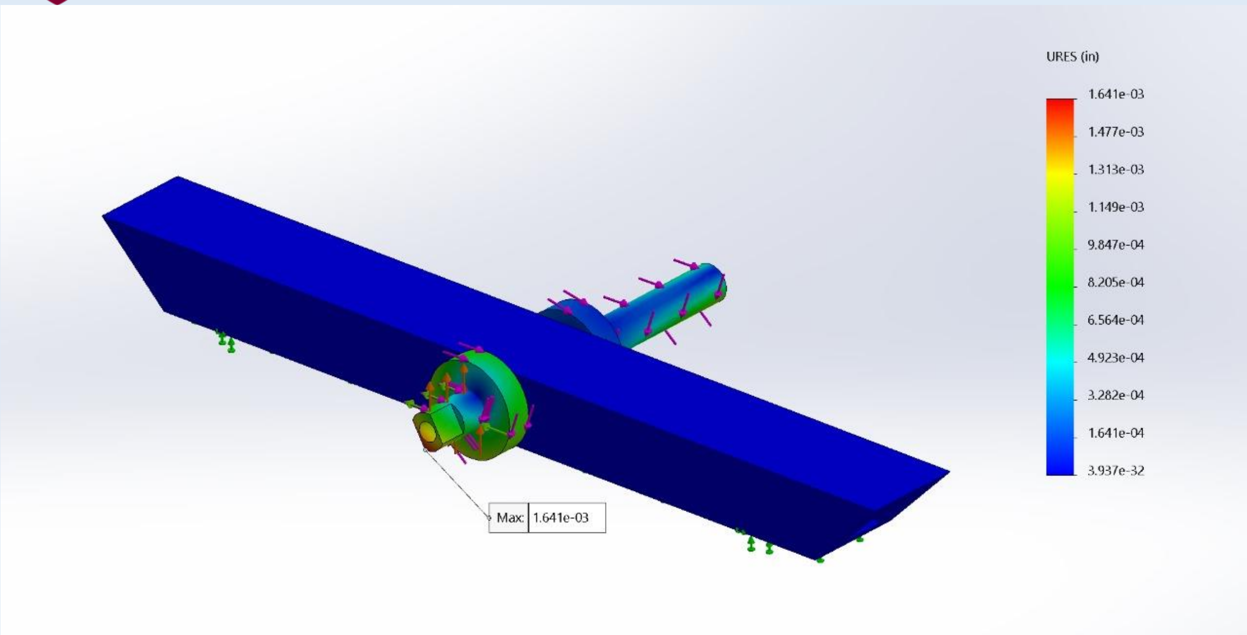
Item No.	Description	Material	Quantity
1	Steering Column	Aluminum 2024-T3	1
2	Steerer	Aluminum 6061-T6	1
3	Socket Head Screw	Black-Oxide Coated Steel	1
4	Clamping Shaft Collar	Aluminum 2024	2
5	Deep Groove Ball Bearing	Steel	2

FEA Analysis





FEA Analysis on SolidWorks of the steering shaft presenting Von Mises, Deflection, and Safety Factors.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1.313e-03
1.149e-03
9.847e-04
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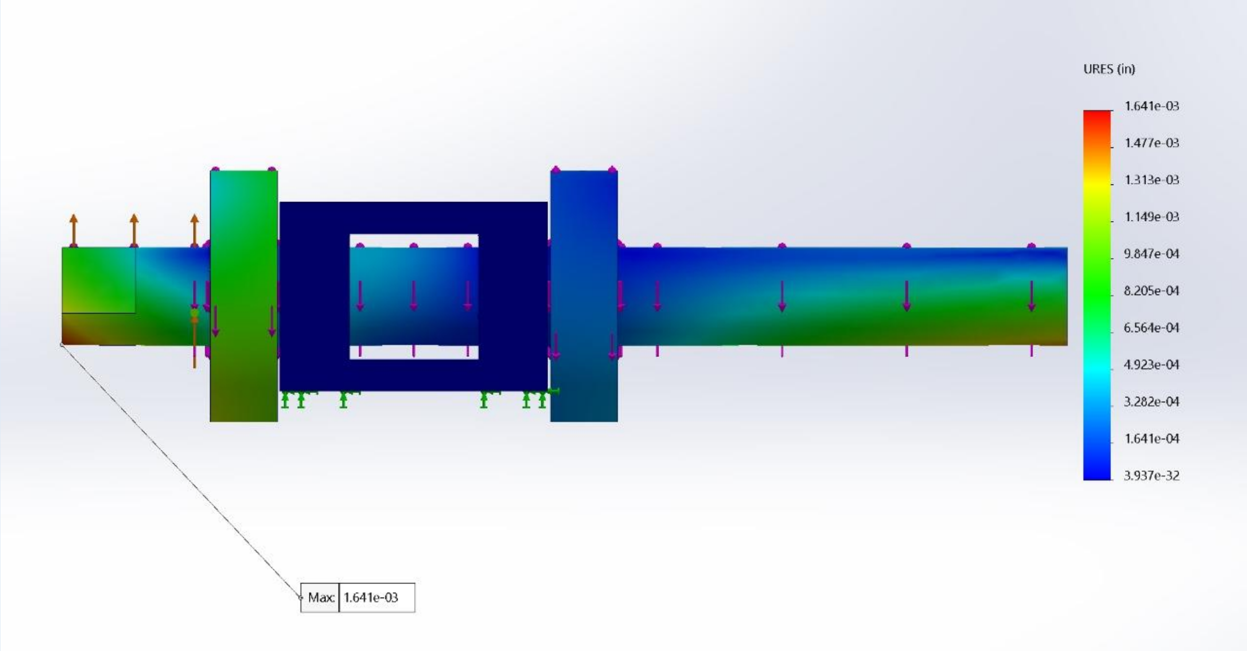
3.937e-32
Max: 1.641e-03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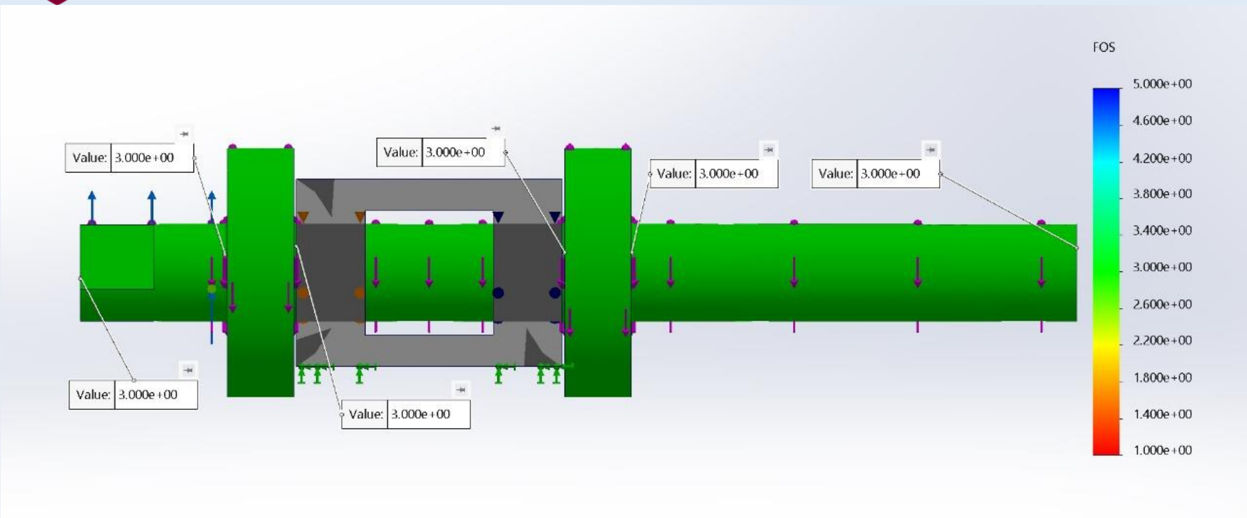
Max: 1.641e-03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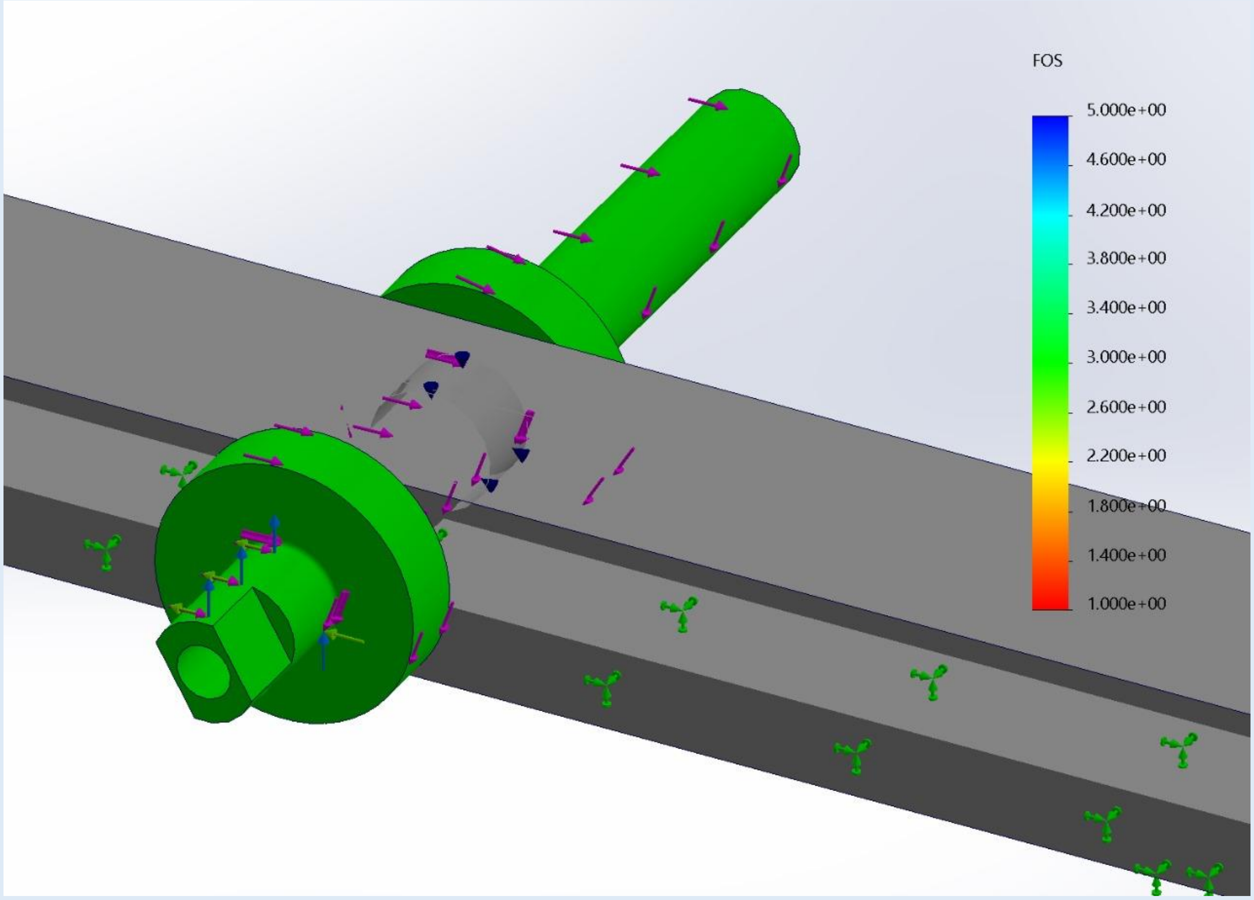
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3.000e+00
2.600e+00
2.200e+00
1.800e+00
1.400e+00
1.000e+00

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Appendix

Tables

Dimensions and Load Ratings for Single-Row 02-Series Deep-Groove and Angular-Contact Ball Bearings

Bore, mm	OD, mm	Width,mm	Fillet Radius,mm	Shoulder Diameter, mm d_S	d_H	Load Ratings, kN Deep Groove C_{10}	Deep Groove C_0	Angular Contact C_{10}	Angular Contact C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

Table 1.

Type of Fit	Description	Symbol
Clearance	*Loose running fit*: for wide commercial tolerances or allowances on external members (Fit with the largest clearance, Suitable for applications where accuracy is not of the utmost importance)	H11/c11
	Free running fit: not for use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal pressures	H9/d9
	Close running fit: for running on accurate machines and for accurate location at moderate speeds and journal pressures	H8/f7
	Sliding fit: Leaves a small clearance for high accuracy while maintaining ease of assembly. Parts will turn and slide quite freely.	H7/g6
	Locational clearance fit: provides minimal clearance for high accuracy requirements. The assembly does not need any force and the mating parts can turn and slide freely with lubrication.	H7/h6

Table 2.

Basic Sizes	Tolerance Grades IT6	IT7	IT8	IT9	IT 10	IT 11
0 to 0.12	0.0002	0.0004	0.0006	0.0010	0.0016	0.0024
0.12 to 0.24	0.0003	0.0005	0.0007	0.0012	0.0019	0.0030
0.24 to 0.40	0.0004	0.0006	0.0009	0.0014	0.0023	0.0035
0.40 to 0.72	0.0004	0.0007	0.0011	0.0017	0.0028	0.0043
0.72 to 1.20	0.0005	0.0008	0.0013	0.0020	0.0033	0.0051
1.20 to 2.00	0.0006	0.0010	0.0015	0.0024	0.0039	0.0063
2.00 to 3.20	0.0007	0.0012	0.0018	0.0029	0.0047	0.0075
3.20 to 4.80	0.0009	0.0014	0.0021	0.0034	0.0055	0.0087
4.80 to 7.20	0.0010	0.0016	0.0025	0.0039	0.0063	0.0098
7.20 to 10.00	0.0011	0.0018	0.0028	0.0045	0.0073	0.0114
10.00 to 12.60	0.0013	0.0020	0.0032	0.0051	0.0083	0.0126
12.60 to 16.00	0.0014	0.0022	0.0035	0.0055	0.0091	0.0 142

Table 3.

	Upper-Deviation Letter					Lower-Deviation Letter				
Basic Sizes	c	d	f	g	h	k	n	p	s	u
0 to 0.12	− 0.0024	−0.0008	−0.0002	− 0.0001	0	0	+0.0002	+0.0002	+0.0006	+0.0007
0.12 to 0.24	− 0.0028	−0.0012	−0.0004	− 0.0002	0	0	+0.0003	+0.0005	+0.0007	+0.0009
0.24 to 0.40	−0.0031	− 0.0016	−0.0005	− 0.0002	0	0	+0.0004	+0.0006	+0.0009	+0.0011
0.40 to 0.72	− 0.0037	−0.0020	−0.0006	−0.0002	0	0	+0.0005	+0.0007	+0.0011	+0.0013
0.72 to 0.96	−0.0043	−0.0026	−0.0008	−0.0003	0	+0.0001	+0.0006	+0.0009	+0.0014	+0.0016
0.96 to 1.20	−0.0043	−0.0026	−0.0008	−0.0003	0	+0.0001	+0.0006	+0.0009	+0.0014	+ 0.0019
1.20 to 1.60	− 0.0047	−0.0031	−0.0010	− 0.0004	0	+0.0001	+0.0007	+ 0.0010	+ 0.0017	+ 0.0024
1.60 to 2.00	− 0.0051	− 0.0031	−0.0010	− 0.0004	0	+0.0001	+0.0007	+ 0.0010	+ 0.0017	+ 0.0028
2.00 to 2.60	− 0.0055	− 0.0039	−0.0012	− 0.0004	0	+0.0001	+ 0.0008	+ 0.0013	+ 0.0021	+0.0034
2.60 to 3.20	−0.0059	− 0.0039	−0.0012	−0.0004	0	+0.0001	+ 0.0008	+ 0.0013	+ 0.0023	+0.0040
3.20 to 4.00	− 0.0067	− 0.0047	− 0.0014	−0.0005	0	+0.0001	+ 0.0009	+ 0.0015	+ 0.0028	+0.0049
4.00 to 4.80	−0.0071	−0.0047	−0.0014	−0.0005	0	+0.0001	+ 0.0009	+ 0.0015	+ 0.0031	+0.0057
4.80 to 5.60	−0.0079	−0.0057	− 0.0017	− 0.0006	0	+0.0001	+ 0.0011	+ 0.0017	+ 0.0036	+0.0067
5.60 to 6.40	−0.0083	−0.0057	− 0.0017	− 0.0006	0	+0.0001	+ 0.0011	+ 0.0017	+0.0039	+0.0075
6.40 to 7.20	−0.0091	−0.0057	−0.0017	−0.0006	0	+0.0001	+ 0.0011	+ 0.0017	+0.0043	+0.0083
7.20 to 8.00	−0.0094	−0.0067	−0.0020	− 0.0006	0	+0.0002	+0.0012	+0.0020	+0.0048	+0.0093
8.00 to 9.00	−0.0102	−0.0067	−0.0020	− 0.0006	0	+0.0002	+0.0012	+0.0020	+0.0051	+0.0102
9.00 to 10.00	−0.0110	−0.0067	−0.0020	− 0.0006	0	+0.0002	+0.0012	+0.0020	+0.0055	+ 0.0112
10.00 to 11.20	−0.0118	−0.0075	−0.0022	− 0.0007	0	+0.0002	+0.0013	+0.0022	+0.0062	+ 0.0124
11.20 to 12.60	−0.0130	−0.0075	−0.0022	−0.0007	0	+0.0002	+0.0013	+0.0022	+0.0067	+0.0130
12.60 to 14.20	−0.0142	−0.0083	−0.0024	− 0.0007	0	+0.0002	+ 0.0015	+ 0.0024	+0.0075	+0.0154
14.20 to 16.00	− 0.0157	−0.0083	−0.0024	−0.0007	0	+0.0002	+ 0.0015	+ 0.0024	+0.0082	+0.0171

Table 4.

Slopes	
Tapered roller	0.0005 to 0.0012 rad
Cylindrical roller	0.0008 to 0.0012 rad
Deep-groove ball	0.001 to 0.003 rad
Spherical ball	0.026 to 0.052 rad
Self-align ball	0.026 to 0.052 rad
Uncrowned spur gear	<0.0005 rad

Table 5.

Graphs

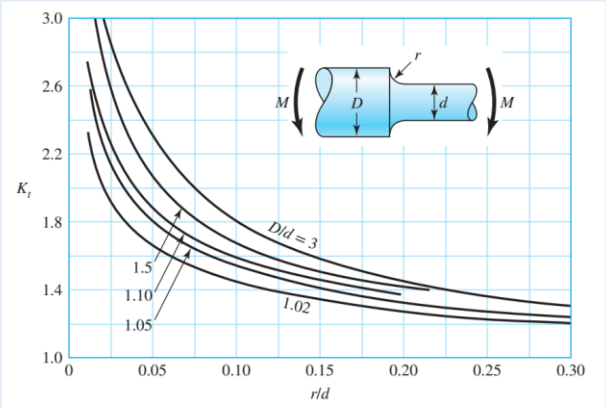

Figure 12.

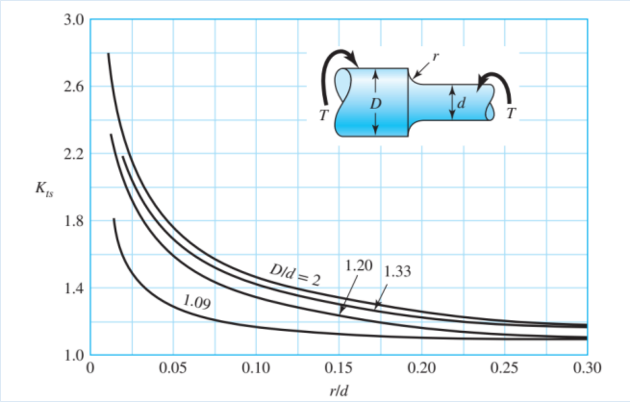

Figure 13.

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1- *https://www.shellecomarathon.com/about/global-rules.html*

2- *https://sites.aub.edu.lb/cogsandcaffeine/2021/02/04/supermileage-car/*

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