Target Drive -Machine Design Analysis

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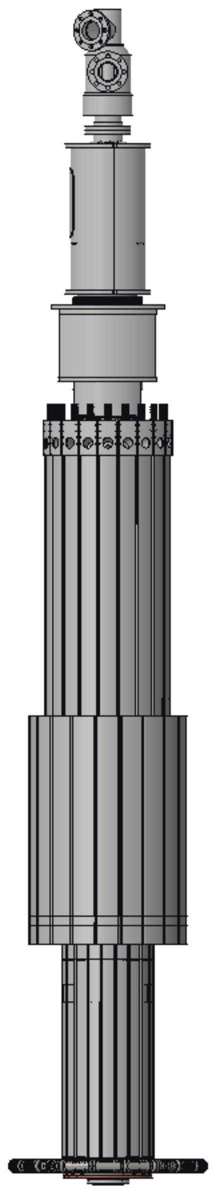


References

[1] Habainy, Jemila. Characterization of mechanical and thermal properties of tungsten for high power spallation target applications. 2018
[2] Defense Metals Information Center. The Engineering Properties of Tungsten and Tungsten Alloys. Report 191, Sept. 1963
[3] Budynas, Nisbett. "Shigley's Mechanical Engineering Design" 8th. Ed. McGraw-Hill
[4] Incropera, Dewitt, Bergma, Lavine. "Introduction to Heat Transfer" 5th Ed. McGraw Hill
[5] Ryerson, "Stock List". 2007 Ed.
[6] Budynas & Young. "Roark's Formulas for Stress and Strain" 7th Ed. McGraw Hill
[7] ASCE 7-10 "Minimum Design Loads for Buildings and Other Structures"
[8] Maloy, Hosemann, Byler, Romero, Dai, Li. "Irradation Effects in Tungsten and Tantalum" Presented at AFCI
[9] White, Frank. "Fluid Mechanics" McGraw-Hill. New York, 5th ed.
[10] Mark's Standard Handbook for Mechanical Engineers, 11th Ed.

Fundemantal Constants

$eV := 1.602\cdot 10^{-19} J$ — Electron Volt definition

$kJ := 1000J$ — Define KiloJoule

$\sigma := 5.6704\cdot 10^{-8}\cdot\frac{W}{m^2\cdot K^4}$ — Steffan Boltzmann constant

$rem := 0.01Sv$ — Definition of Rem

General System Parameters

$P_{beam} := 700kW$ — Beam Power (time averaged)

$E_{beam} := 1.3\cdot 10^{9} eV$ — Proton Energy (per proton)

$P_{pulse} := 2.24\cdot 10^{14}$ — Number of protons per pulse

$f := 15Hz$ — Number of pulses per second

$t_{pulse} := 1\cdot 10^{-6} s$ — Pulse duration

$T_{water} := 35\,°C$ — Inlet temperature of the cooling water

$\alpha := 0.6$ — Amount of beam energy deposited as heat

$Q_{decay} := 8kW$ — Decay heat (total) for all segment blocks, immediately after shutdwon

$e_{cg} := 35kg\cdot m$ — Imbalance of the shaft (assumed) equal to one target foot missing.

Target Parameters

$n_{targets} := 20$	Number of targets in wheel
$\omega_{rot} := \dfrac{f \cdot rev}{n_{targets}} = 45 \cdot rpm$	Rotational speed of the assembly
$P_{target} := P_{pulse} \cdot \dfrac{E_{beam} \cdot f}{n_{targets}} = 34.988 \cdot kW$	Energy deposition per target (Not all energy is heating)
$P_{tgt\_ht} := P_{target} \cdot \alpha = 2.099 \times 10^{4}\,W$	Amount of heat generated in the target (per block)
$E_{pulse} := E_{beam} \cdot P_{pulse} = 46.65 \cdot kJ$	Energy deposition per pulse

Operational Parameters

$t_{op} := 5000 \dfrac{hr}{yr}$	Number of hours "available" per year
$T_{i\_water} := 30\,°C$	Initial inlet water temperature
$V_{flow} := 150 gpm$	Selected flow rate for the target
$t_{start} := 600 s$	Required start-up time (TBD)
$t_{estop} := 1 s$	Time required for an emergency stop
$P_{op} := 75 psi$	Operational pressure of the cooling water
$P_{MAWP} := 125 psi$	Maximum allowable pressure of the cooling water.

Material Properties

Tungsten

$E_{Wi} := 398\text{GPa}$	Modulus of elasticity, Tungsten, irradiated
$\nu_{Wi} := 0.27$	Poisson's ration, tungsten, irradiated
$k_{Wi} := 84\dfrac{W}{m \cdot K}$	Thermal conductivity, irradiated Tungsten
$Cp_{Tungsten} := 133\dfrac{J}{kg \cdot K}$	Specific heat of Tungsten
$\alpha_{Wi} := \dfrac{4.15 \cdot 10^{-6}}{K}$	Coeffeceint of therma expansion, Tungsten
$\rho_W := 19300\dfrac{kg}{m^3}$	Density of pure Tungsten
$Sut_{Wi} := 356\text{MPa}$	Ultimate tensile strength of rolled irradiated tungsten [Haibany]
$Se_W := 157\text{MPa}$	Endurance limit of roleld tungsten (at 1.3E8 cycles, estimate)

Tantalum

$E_{Ta} := 186\text{GPa}$	Modulus of elasticity, (Plansee)
$\nu_{Ta} := 0.35$	Poisson's ratio, Tantalum
$k_{Ta} := 27.5\dfrac{W}{m \cdot K}$	Thernal conductivity, Tantalum (Plansee)
$\alpha_{Ta} := \dfrac{6.4 \cdot 10^{-6}}{K}$	Thermal expansion coeffecient of Tantalum
$Sut_{Ta} := 450\text{MPa}$	Minimum Tensile strength of annealed Tantalum
$Sy_{Ta} := 160\text{MPa}$	Yield Strength of annealed Tantalum
$Se_{Ta} := 230\text{MPa}$	Endurance limit for Tantalum
$\varepsilon_{Ta} := 0.2$	Hemispherical emissivity @ 1460 C (Plansee)

316 Stainless

$Cp_{316} := 468 \frac{J}{kg \cdot K}$ Specific heat of 316L. Ref 4

$\rho_{316} := 8030 \frac{kg}{m^3}$ Density of 316L. ASME B&PVC

$k_{316} := 13.4 \frac{W}{m \cdot K}$ Thermal conductivity of 316L. Ref 4

$E_{316} := 29 \cdot 10^6 psi$ Modulus of elasticity for 316L. Ref. 5, pg 500

$Sy_{316} := 35ksi$ Yield Stress of 316L. Ref 5, pg 500

$Sut_{316} := 85ksi$ Ultimate stress of 316L. Ref 500

$S_{316} := 115MPa$ Allowable stress, 316l, ASME B&PVC (-30 to 150C)

$Spl_{316} := S_{316} \cdot 1.5 = 172.5 \cdot MPa$ ASME Allowable stress, Local plus bending, Section VII, D2, 5.2.2.4

$\alpha_{316} := \frac{17.2 \cdot 10^{-6}}{K}$ Thermal expansion coeffecient

$\nu_{316} := 0.305$ Poisson's ratio for 316L

6061 T6 Aluminum

$Cp_{6061} := 0.896 \frac{J}{gm \cdot K}$ Specific heat of 6061 aluminum

$\rho_{6061} := 2.7 \frac{gm}{cm^3}$ Density of 6061

$k_{6061} := 167 \frac{W}{m \cdot K}$ Thermal conductivity of 316L

$E_{6061} := 68.9GPa$ Elastic Modulus of 6061

$\alpha_{6061} := \frac{23.6}{K}$ Thermal expansion coeffecient

$\nu_{6061} := 0.33$ Poisson's for 6061-T6

General Carbon steel

$E_{carbon\_steel} := 200GPa$ Elastic modulus for carbon steel (bolts, etc.)

$\nu_{carbon\_steel} := 0.3$ Poisson's ratio for general carbon steel

$Sy_{1045} := 310MPa$ Yield strength for 1045 carbon steel (Per ASTM A830)

ASTM A572 GR50

$Sy_{A572\_50} := 50ksi$ Yield strength for A572 Grade 50

$Su_{A572\_50} := 65ksi$ Tensile strenght of A572 Grade 50

4130 Steel

$Sy_{4130} := 1170MPa$ Yield strength, 4130 steel

$Sut_{4130} := 930MPa$ Ultimate strength, 4130 steel

$S_{4130} := 233MPa$ ASME allowable stress, 4130 steel

4340 Steel

$Sy_{4340} := 1170MPa$ Yeild strength, 4340 steel

$Sut_{4340} := 930MPa$ Ultimate strength, 4340 steel

$S_{4340} := 233MPa$ Asme allowable stress, 4340 steel

Inconel 718-bolting per ASTM B637

$Su_{718} := 1275MPa$ Ultimate strength of Inconel per ASTM B637

$Sy_{718} := 1034MPa$ Yield strength of Inconel 718 per ASTM B637

$S_{718} := 36ksi$ Allowable stress for UNS NO7718 bolting, per ASME BPVC SEC II-D, Table 3 (SB-637)

$E_{718} := 195GPa$ Elastic Moduls of UNS NO7718 bolting material (HAYNES 718 datasheet)

Inconel 625 - Bolting

$Su_{625} := 100ksi$ Tensile strength of inconel 625 per AMS 5666

$Sy_{625} := 42 \cdot ksi$ — Yield strength of Inconel 625 per AMS 5666

$S_{625} := 30ksi$ — Allowable stress of UNS 6625 per ASME BPVC SEC II-D, Table 3

$\nu_{625} := 0.31$ — Poisson' ration for Inconel 625

Nitronic 60 - Bolting

$Sy_{N60} := 50ksi$ — Yield Strrength (AMS 5848)

$Su_{N60} := 95ksi$ — Ultimate strength of Nitronic 60 (AMS 5848)

ASME BPVC.II.D.C-2019

Table 3 (Cont'd)
Section III, Classes 2 and 3;* Section VIII, Divisions 1 and 2;† and Section XII
Maximum Allowable Stress Values, *S*, for Bolting Materials
(*See Maximum Temperature Limits for Restrictions on Class)
(†Use with Part 4.16 of Section VIII, Division 2)

Line No.	Nominal Composition	Product Form	Spec. No.	Type/ Grade	Alloy Desig./ UNS No.	Class/ Condition/ Temper	Size/ Thickness, in.
	Nonferrous Materials (Cont'd)						
1	...	Rod	SB–98	...	C66100	O60	All
2	...	Rod	SB–98	...	C66100	H01	All
3	...	Rod	SB–98	...	C66100	H02	≤2
4	99Ni	Bolting	SB–160	...	N02200	Annealed	...
5	99Ni	Bolting	SB–160	...	N02200	Hot fin./ann.	...
6	99Ni	Bolting	SB–160	...	N02200	Cold drawn	...
7	99Ni–Low C	Bolting	SB–160	...	N02201	Hot fin./ann.	...
8	67Ni–30Cu	Bolting	SB–164	...	N04400	Annealed	...
9	67Ni–30Cu	Bolting	SB–164	...	N04400	Hot worked	...
10	67Ni–30Cu	Bolting	SB–164	...	N04400	Hot worked	...
11	67Ni–30Cu	Bolting	SB–164	...	N04400	CD–str. rel.	...
12	67Ni–30Cu	Bolting	SB–164	...	N04400	Cold worked	...
13	67Ni–30Cu	Bolting	SB–164	...	N04400	CD–str. rel.	...
14	67Ni–30Cu–S	Bolting	SB–164	...	N04405	Annealed	...
15	67Ni–30Cu–S	Bolting	SB–164	...	N04405	Annealed	...
16	67Ni–30Cu–S	Bolting	SB–164	...	N04405	Hot worked	...
17	67Ni–30Cu–S	Bolting	SB–164	...	N04405	Hot worked	...
18	67Ni–30Cu–S	Bolting	SB–164	...	N04405	Cold worked	...
19	67Ni–28Cu–3Al	Bolting	SF–468	...	N05500	Ann./aged	1.000–1.500
20	67Ni–28Cu–3Al	Bolting	SF–468	...	N05500	Ann./aged	0.250–0.875
21	47Ni–22Cr–9Mo–18Fe	Bolting	SB–572	...	N06002	Annealed	...
22	47Ni–22Cr–19Fe–6Mo	Bolting	SB–581	...	N06007	Solution ann.	>3/4
23	47Ni–22Cr–19Fe–6Mo	Bolting	SB–581	...	N06007	Solution ann.	<3/4
24	55Ni–21Cr–13.5Mo	Bolting	SB–574	...	N06022	Solution ann.	...
25	40Ni–29Cr–15Fe–5Mo	Bolting	SB–581	...	N06030	Solution ann.	...
26	61Ni–16Mo–16Cr	Bolting	SB–574	...	N06455	Solution ann.	...
27	72Ni–15Cr–8Fe	Bolting	SB–166	...	N06600	Annealed	...
28	72Ni–15Cr–8Fe	Bolting	SB–166	...	N06600	Hot fin.	...
29	72Ni–15Cr–8Fe	Bolting	SB–166	...	N06600	Cold drawn	...
30	72Ni–15Cr–8Fe	Bolting	SB–166	...	N06600	Hot fin.	...
31	72Ni–15Cr–8Fe	Bolting	SB–166	...	N06600	Hot fin.	...

32	60Ni–22Cr–9Mo–3.5Cb	Bolting	SB–446	1	N06625	Annealed	...
33	49Ni–25Cr–18Fe–6Mo	Bolting	SB–581	...	N06975	Solution ann.	...
34	53Ni–19Cr–19Fe–Cb–Mo	Bolting	SB–637	...	N07718	Ann./aged	≤6
35	70Ni–16Cr–7Fe–Ti–Al	Bolting	SB–637	2	N07750	Ann./aged	...

Table 3 (Cont'd)
Section III, Classes 2 and 3;* Section VIII, Divisions 1 and 2;† and Section XII Maximum Allowable Stress Values, *S*, for Bolting Materials (*See Maximum Temperature Limits for Restrictions on Class) (†Use with Part 4.16 of Section VIII, Division 2)

Line	Maximum Allowable Stress, ksi (Multiply by 1000 to Obtain psi), for Metal Temperature, °F, Not Exceeding																
No.	100	150	200	250	300	350	400	450	500	550	600	650	700	750	800	850	900
	Nonferrous Materials (Cont'd)																
1	10.0	10.0	9.9	9.9	9.9	9.8	...	...	...	...	...	...	...	...	...	...	...
2	10.0	10.0	9.9	9.9	9.9	9.8	...	...	...	...	...	...	...	...	...	...	...
3	10.0	10.0	9.9	9.9	9.9	9.8	...	...	...	...	...	...	...	...	...	...	...
4	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	...	...	...	...	...	...
5	10.0	10.0	10.0	10.0	10.0	10.0	10.0	9.8	9.5	8.9	8.3	...	...	...	...	...	...
6	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	...	...	...	...	...	...
7	6.7	6.6	6.4	6.3	6.3	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.0	5.9	*5.8*	*4.8*
8	16.7	...	14.6	...	13.6	...	13.2	...	13.1	...	13.1	13.1	13.1	12.9	12.7	*11.0*	*8.0*
9	16.7	...	14.6	...	13.6	...	13.2	...	13.1	...	13.1	13.1	13.1	12.9	12.7	*11.0*	*8.0*
10	16.7	...	14.6	...	13.6	...	13.2	...	13.1	...	13.1	13.1	13.0	12.9	12.7	*8.5*	*4.0*
11	16.7	...	14.6	...	13.6	...	13.2	...	13.1	...	...	...	...	...	...	...	...
12	16.7	...	14.6	...	13.8	...	13.8	...	13.8	...	...	...	...	...	...	...	...
13	16.7	...	15.0	...	15.0	...	15.0	...	15.0	...	...	...	...	...	...	...	...
14	16.6	...	14.6	...	13.6	...	13.2	...	13.1	...	13.1	13.1	13.1	12.9	12.7	...	...
15	16.6	...	14.6	...	13.6	...	13.2	...	13.1	...	13.1	13.1	13.1	13.0	12.7	*11.0*	*8.0*
16	18.7	...	18.7	...	18.7	...	18.7	...	18.7	...	18.7	18.4	18.0	16.3	14.5	...	...
17	18.7	...	18.7	...	18.7	...	18.7	...	18.7	...	18.7	18.7	18.0	17.2	14.5	*8.5*	*4.0*
18	12.5	...	12.5	...	12.5	...	12.5	...	12.5	...	...	...	...	...	...	...	...
19	21.3	...	21.3	...	21.3	...	21.3	...	21.3	...	...	...	...	...	...	...	...
20	22.5	...	22.5	...	22.5	...	22.5	...	22.5	...	...	...	...	...	...	...	...
21	23.3	...	20.9	...	19.2	...	17.8	...	16.5	...	15.6	15.3	15.0	14.9	14.7	14.6	14.5
22	20.0	...	19.9	...	16.6	...	15.6	...	15.0	...	14.4	14.2	14.0	13.9	13.8	13.7	13.6
23	22.5	...	20.9	...	19.5	...	18.2	...	17.4	...	16.8	16.6	16.4	16.3	16.1	16.0	16.0
24	25.0	...	25.0	...	24.5	...	22.7	...	21.2	...	20.1	19.6	19.2	18.9	18.6	...	...
25	21.3	...	20.0	...	18.3	...	17.2	...	16.4	...	15.8	15.5	15.2	14.9	14.6	...	...
26	25.0	...	24.6	...	23.0	...	21.7	...	20.9	...	20.1	19.8	19.6	19.4	19.1	...	...
27	20.0	...	20.0	...	20.0	...	20.0	...	20.0	...	20.0	19.8	19.6	19.4	19.1	18.7	*16.0*
28	21.2	...	21.2	...	21.2	...	21.2	...	21.2	...	21.2	21.2	21.1	21.0	20.4	20.2	19.5
29	20.0	...	20.0	...	20.0	...	20.0	...	20.0	...	...	...	...	...	...	...	...
30	10.0	...	9.5	...	9.2	...	9.1	...	9.1	...	9.1	9.0	8.9	8.9	8.8	...	...
31	21.2	...	21.2	...	21.2	...	21.2	...	21.2	...	21.2	21.2	21.1	21.0	20.4	20.2	19.5
32	30.0	...	30.0	...	30.0	...	28.2	...	27.0	...	26.4	26.2	26.0	26.0	24.6	24.3	24.0
33	21.3	...	19.6	...	18.4	...	17.6	...	16.5	...	15.6	15.3	15.1	14.9	14.7	...	...
34	37.0	...	36.0	...	35.2	...	34.6	...	34.2	...	33.8	33.7	33.6	33.5	33.3	33.1	32.9
35	28.7	...	28.7	...	28.7	...	28.7	...	28.7	...	28.7	28.7	28.7	28.7	28.7	...	...
36	18.7	...	17.3	...	16.4	...	15.3	...	14.6	...	13.7	13.5	13.2	12.9	12.8	...	...
37	23.8	...	23.8	...	22.4	...	21.5	...	20.5	...	19.4	19.0	18.6	18.3	18.0	...	...
38	18.7	...	18.7	...	17.9	...	17.2	...	16.7	...	16.3	16.1	15.9	15.7	15.5	15.3	15.1
39	16.2	...	15.4	...	14.5	...	13.5	...	12.9	...	12.2	11.9	11.7	11.4	11.1	10.9	10.7
40	21.2	...	21.2	...	20.4	...	19.2	...	18.3	...	17.8	17.6	17.3	17.2	17.1	16.9	16.8

Water - Inlet Cooling Water

$T_{amb} := 32.5\ °C$ Temperature of the cooling fluid

$Cp_{water} := 4179 \frac{J}{kg \cdot K}$ Specific heat of water (at 300K). Ref. 4

$\rho_{water} := 997 \frac{kg}{m^3}$ Density of water. Ref. 4

$\mu_{water} := 855 \cdot 10^{-6} \cdot \frac{N \cdot s}{m^2}$ Viscosity of water at 300K. Ref. 4

$k_{water} := 0.613 \frac{W}{m \cdot K}$ Thermal conductivity of water at 300K. Ref. 4

$Pr_{water} := 5.83$ Prandtl number of water at 300K

$\nu_{water} := 8.01 \cdot 10^{-6}$ Kinematic viscosity of water at 300K. Ref. 4

Water - Saturated Liquid @ 100C

$L_{hv} := 2256000 \frac{J}{kg}$ Latent Heat of vaporization, water

$T_{sat} := 100\ °C$ Saturation temperature of water, 1 atm

Water - Saturated Steam

$\mu_{condensation} := 279 \cdot 10^{-6} \frac{N \cdot s}{m^2}$ Viscosity of condensation at sat. Ref 4.

$Cp_{steam} := 1880 \frac{J}{kg \cdot K}$ Specific heat of steam

$\rho_{steam} := 958.773 \frac{kg}{m^3}$ Density of steam at 1 atm

Air

$CP_{air} := 1.007 \frac{kJ}{kg \cdot K}$ — Specific heat of air at STP

$\mu_{air} := 184.6 \cdot 10^{-7} \cdot \frac{N \cdot s}{m^2}$ — Viscosity of air at STP

$Pr_{air} := 0.707$ — Prandtl number for air at STP

$P_{atm} := 101325Pa$ — Atmospheric pressure

Helium

$k_{He} := 152 \cdot 10^{-3} \cdot \frac{W}{m \cdot K}$ — Thermal conductivity of helium at 300K

$R_{He} := 2.0769 \cdot \frac{kJ}{kg \cdot K}$ — Gas Constant

$\rho_{He} := 0.1557 \frac{kg}{m^3}$ — Density of helium at 35C

Fasteners

$E_{A193\_B8M} := 193GPa$ — Elastic Modulus of ASTM A193

Fastener Geometries

Nominal Major Diameter d mm	Coarse-Pitch Series Pitch p mm	Coarse-Pitch Series Tensile-Stress Area A_t mm^2	Coarse-Pitch Series Minor-Diameter Area A_r mm^2	Fine-Pitch Series Pitch p mm	Fine-Pitch Series Tensile-Stress Area A_t mm^2	Fine-Pitch Series Minor-Diameter Area A_r mm^2
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980
72	6	3460	3280	2	3860	3800
80	6	4340	4140	1.5	4850	4800
90	6	5590	5360	2	6100	6020
100	6	6990	6740	2	7560	7470
110				2	9180	9080

*The equations and data used to develop this table have been obtained from ANSI B1.1-1974 and B18.3.1-1978. The minor diameter was found from the equation $d_r = d - 1.226\ 869p$, and the pitch diameter from $d_p = d - 0.649\ 519p$. The mean of the pitch diameter and the minor diameter was used to compute the tensile-stress area.

Table 8.2.6 Limiting Dimensions of Standard Series Threads for Commercial Screws, Bolts, and Nuts

			External thread (bolt), mm									Internal thread (nut), mm						
					Major diameter		Pitch diameter			Minor diameter			Minor diameter		Pitch diameter			
Nominal size diam, mm	Pitch p, mm	Basic thread designation	Tol. class	Allowance	Max	Min	Max	Min	Tol.	Max*	Min†	Tol. class	Min	Max	Min	Max	Tol.	Major diam, min
1.6	0.35	M1.6	6g	0.019	1.581	1.496	1.354	1.291	0.063	1.151	1.063	6H	1.221	1.321	1.373	1.458	0.085	1.600
1.8	0.35	M1.8	6g	0.019	1.781	1.696	1.554	1.491	0.063	1.351	1.263	6H	1.421	1.521	1.573	1.568	0.085	1.800
2	0.4	M2	6g	0.019	1.981	1.886	1.721	1.654	0.067	1.490	1.394	6H	1.567	1.679	1.740	1.830	0.090	2.000
2.2	0.45	M2.2	6g	0.020	2.180	2.080	1.888	1.817	0.071	1.628	1.525	6H	1.713	1.838	1.908	2.000	0.095	2.200
2.5	0.45	M2.5	6g	0.020	2.480	2.380	2.188	2.117	0.071	1.928	1.825	6H	2.013	2.138	2.208	2.303	0.095	2.500
3	0.5	M3	6g	0.020	2.980	2.874	2.655	2.580	0.075	2.367	2.256	6H	2.459	2.599	2.675	2.775	0.100	3.000
3.5	0.6	M3.5	6g	0.021	3.479	3.354	3.089	3.004	0.085	2.742	2.614	6H	2.850	3.010	3.110	3.222	0.112	3.500
4	0.7	M4	6g	0.022	3.978	3.838	3.523	3.433	0.090	3.119	2.979	6H	3.242	3.422	3.545	3.663	0.118	4.000
4.5	0.75	M4.5	6g	0.022	4.478	4.338	3.991	3.901	0.090	3.558	3.414	6H	3.688	3.878	4.013	4.131	0.118	4.500
5	0.8	M5	6g	0.024	4.976	4.826	4.456	4.361	0.095	3.994	3.841	6H	4.134	4.334	4.480	4.605	0.125	5.000
6	1	M6	6g	0.026	5.974	5.794	5.324	5.212	0.112	4.747	4.563	6H	4.917	5.153	5.350	5.500	0.150	6.000
7	1	M7	6g	0.026	6.974	6.794	6.234	6.212	0.112	5.747	5.563	6H	5.917	6.153	6.350	6.500	0.150	7.000
8	1.25	M8	6g	0.028	7.972	7.760	7.160	7.042	0.118	6.439	6.231	6H	6.647	6.912	7.188	7.348	0.160	8.000
	1	M8 × 1	6g	0.026	7.974	7.794	7.324	7.212	0.112	6.747	6.563	6H	6.918	7.154	7.350	7.500	0.150	8.000
10	1.5	M10	6g	0.032	9.968	9.732	8.994	8.862	0.132	8.127	7.879	6H	8.376	8.676	9.026	9.206	0.180	10.000
	1.25	M10 × 1.25	6g	0.028	9.972	9.760	9.160	9.042	0.118	8.439	8.231	6H	8.646	8.911	9.188	9.348	0.160	10.000
12	1.75	M12	6g	0.034	11.966	11.701	10.829	10.679	0.150	9.819	9.543	6H	10.106	10.441	10.863	11.063	0.200	12.000
	1.25	M12 × 1.25	6g	0.028	11.972	11.760	11.160	11.028	0.118	10.439	10.217	6H	10.646	10.911	11.188	11.368	0.180	12.000
14	2	M14	6g	0.038	13.962	13.682	12.663	12.503	0.160	11.508	11.204	6H	11.835	12.210	12.701	12.913	0.212	14.000
	1.5	M14 × 1.5	6g	0.032	13.968	13.732	12.994	12.854	0.140	12.127	11.879	6H	12.376	12.676	13.026	13.216	0.190	14.000
16	2	M16	6g	0.038	15.962	15.682	14.663	14.503	0.160	13.508	13.204	6H	13.385	14.210	14.701	14.913	0.212	16.000
	1.5	M16 × 1.5	6g	0.032	15.968	15.732	14.994	14.854	0.140	14.127	13.879	6H	14.376	14.676	15.026	15.216	0.190	16.000
18	2.5	M18	6g	0.038	17.958	17.623	16.334	16.164	0.170	14.891	14.541	6H	15.294	15.744	16.376	16.600	0.224	18.000
	1.5	M18 × 1.5	6g	0.032	17.968	17.732	16.994	15.854	0.140	16.127	15.879	6H	16.376	16.676	17.026	17.216	0.190	18.000
20	2.5	M20	6g	0.042	19.958	19.623	18.334	18.164	0.170	16.891	16.541	6H	17.294	17.744	18.376	18.600	0.224	20.000
	1.5	M20 × 1.5	6g	0.032	19.968	19.732	18.994	18.854	0.140	18.127	17.879	6H	18.376	18.676	19.026	19.216	0.190	20.000
22	2.5	M22	6g	0.042	21.958	21.623	20.334	20.164	0.170	18.891	18.541	6H	19.294	19.744	20.376	20.600	0.224	22.000
	1.5	M22 × 1.5	6g	0.032	21.968	21.732	20.994	20.854	0.140	20.127	19.879	6H	20.376	20.676	21.026	21.216	0.190	22.000
24	3	M24	6g	0.048	23.952	23.577	22.003	21.803	0.200	20.271	19.855	6H	20.752	21.252	22.051	22.316	0.265	24.000
	2	M24 × 2	6g	0.038	23.962	23.682	22.663	22.493	0.170	21.508	21.194	6H	21.835	22.210	22.701	22.925	0.224	24.000
27	3	M27	6g	0.048	26.952	26.577	25.003	24.803	0.200	23.271	22.855	6H	23.752	24.252	25.051	25.316	0.265	27.000
	2	M27 × 2	6g	0.038	26.962	26.682	25.663	25.493	0.170	24.508	24.194	6H	24.835	25.210	25.701	25.925	0.224	27.000
30	3.5	M30	6g	0.053	29.947	29.522	27.674	27.462	0.212	25.653	25.189	6H	26.211	26.771	27.727	28.007	0.280	30.000
	2	M30 × 2	6g	0.038	29.962	29.682	28.663	28.493	0.170	27.508	27.194	6H	27.835	28.210	28.701	28.925	0.224	30.000
33	3.5	M33	6g	0.053	32.947	32.522	30.674	30.462	0.212	28.653	28.189	6H	29.211	29.771	30.727	31.007	0.280	33.000
	2	M33 × 2	6g	0.038	32.962	32.682	31.663	31.493	0.170	30.508	30.194	6H	30.835	31.210	31.701	31.925	0.224	33.000
36	4	M36	6g	0.060	35.940	35.465	33.342	33.118	0.224	31.033	30.521	6H	31.670	32.270	33.402	33.702	0.300	36.000
	3	M36 × 3	6g	0.048	35.952	35.577	34.003	33.803	0.200	32.271	31.855	6H	32.752	33.252	34.051	34.316	0.265	36.000
39	4	M39	6g	0.060	38.940	38.465	36.342	36.118	0.224	34.033	33.521	6H	34.670	35.270	36.402	36.702	0.300	39.000
	3	M39 × 3	6g	0.048	38.952	38.577	37.003	36.803	0.200	35.271	34.855	6H	35.752	36.252	37.051	37.316	0.265	39.000

* Design form, see Figs. 2 and 5 of ANSI B1.13M-1979 (or Figs. 1 and 4 in 1983 revision).
† Required for high-strength applications where rounded root is specified.
SOURCE: [Appeared in ASME/SAE Interpretive document, Metric Screw Threads, B1.13 (Nov. 3, 1966), pp. 9, 10.] ISO 261-1973, reproduced by permission.

M4 Threads

$d_{m4} := 4mm$ Nominal diameter of the fastener

$A_{m4} := \frac{\pi}{4} \cdot d_{m4}^{2}$ Area of the unthreaded shank

$At_{m4} := 8.78mm^{2}$ Tensile stress area of M4 fastener (Shigley's)

$Am_{m4} := 7.75mm^{2}$ Minor Diameter area of the fastener (Shigley's)

$dp_{m4} := 3.545mm$ Pitch diameter of the thread form. (Mark's)

M5 Threads

$d_{m5} := 5mm$ Nominal diameter of the fastener

$A_{m5} := \frac{\pi}{4} \cdot d_{m5}^{2}$ Area of the unthreaded shank

$At_{m5} := 14.2mm^{2}$ Tensile stress area of the fastener (Shigley's)

$Am_{m5} := 12.7mm^{2}$ Minor Diameter area of the fastener (Shigley's)

$dp_{m5} := 4.361mm$ Pitch diameter of the thread form. (Mark's)

M6 Threads

$d_{m6} := 6mm$ Nominal diameter of the fastener

$A_{m6} := \frac{\pi}{4} \cdot d_{m6}^{2}$ Area of the unthreaded shank

$At_{m6} := 20.1mm^{2}$ Tensile stress area of the fastener (Shigley's)

$Am_{m6} := 17.9mm^{2}$ Minor Diameter area of the fastener (Shigley's)

$dp_{m6} := 5.212mm$ Pitch diameter of the thread form. (Mark's)

M8 Threads

$d_{m8} := 8mm$ Nominal diameter of the fastener

$A_{m8} := \frac{\pi}{4} \cdot d_{m8}^{2}$ Area of the unthreaded shank

$At_{m8} := 36.6mm^{2}$ Tensile stress area of the fastener (Shigley's)

$Am_{m8} := 32.8mm^{2}$ Minor Diameter area of the fastener (Shigley's)

$dp_{m8} := 7.042mm$ Pitch diameter of the thread form. (Mark's)

M10x1.5 Threads

$d_{m10} := 10mm$ Nominal diameter of the fastener

$A_{m10} := \frac{\pi}{4} \cdot d_{m10}^{2}$ Area of the unthreaded shank

$At_{m10} := 58mm^{2}$ Tensile stress area of the fastener (Shigley's)

$Am_{m10} := 52.3mm^{2}$ Minor Diameter area of the fastener (Shigley's)

$dp_{m10} := 8.9mm$ Pitch diameter of the thread form. (Mark's)

M12x1.75 Threads

$d_{m12} := 12mm$ Nominal diameter of the fastener

$A_{m12} := \frac{\pi}{4} \cdot d_{m12}^{2}$ Area of the unthreaded shank

$At_{m12} := 84.3mm^{2}$ Tensile stress area of the fastener (Shigley's)

$Am_{m12} := 76.3mm^{2}$ Minor Diameter area of the fastener (Shigley's)

$dp_{m12} := 9.543mm$ Pitch diameter of the thread form. (Mark's)

M16x2.0 Threads

$d_{m16} := 16mm$ Nominal diameter of the fastener (M16)

$A_{m16} := \frac{\pi}{4} \cdot d_{m16}^{2}$ Area of the unthreaded shank

$At_{m16} := 157mm^{2}$ Tensile stress area of the fastener (Shigley's)

$Am_{m16} := 14mm^{2}$ Minor Diameter area of the fastener (Shigley's)

$dp_{m16} := 14.503mm$ Pitch diameter of the thread form. (Mark's)

M18x2.5 Threads

$d_{m18} := 18mm$ Nominal diameter of the fastener (M18)

$A_{m18} := \frac{\pi}{4} \cdot d_{m18}^{2}$ Area of the unthreaded shank

$At_{m18} := 192mm^{2}$ Tensile stress area of the fastener (Shigley's)

$Am_{m18} := 14.75mm^{2}$ Minor Diameter area of the fastener (Shigley's)

$dp_{m18} := 16.25mm$ Pitch diameter of the thread form. (Mark's)

M20x2.5 Threads

$d_{m20} := 20mm$ Nominal diameter of the fastener

$A_{m20} := \frac{\pi}{4} \cdot d_{m20}^{\ 2}$ Area of the unthreaded shank

$At_{m20} := 245mm^2$ Tensile stress area of the fastener (Shigley's)

$Am_{m20} := 225mm^2$ Minor Diameter area of the fastener (Shigley's)

$dp_{m20} := 18.164mm$ Pitch diameter of the thread form. (Mark's)

M24x3 Threads

$d_{m24} := 24mm$ Nominal diameter of the fastener (M24)

$A_{m24} := \frac{\pi}{4} \cdot d_{m24}^{\ 2}$ Area of the unthreaded shank (M24)

$At_{m24} := 353mm^2$ Tensile stress area of the fastener (Shigley's)

$Am_{m24} := 324mm^2$ Minor Diameter area of the fastener (Shigley's)

$dm_{m24} := 20mm$ Minor diameter of the fastener (Mark's)

$dp_{m24} := 21.803mm$ Pitch diameter of the thread form. (Mark's)

M27x3 Threads

$d_{m27} := 27mm$ Nominal diameter of the fastener (M27)

$dm_{m27} := 23mm$ Minor diameter of the fastener

$dp_{m27} := 25mm$ Pitch diameter of the fastener

$A_{m27} := \frac{\pi}{4} \cdot d_{m27}^{\ 2}$ Tensile area of the unthreaded portion of the fastener.

$At_{m27} := \frac{\pi}{4} \cdot dm_{m27}^{\ 2} = 415.476 \cdot mm^2$ Tensile area of the threaded portion of the fastener

M30x3.5 Threads

$d_{m30} := 30mm$ Nominal diameter of the fastener (M30)

$A_{m30} := \frac{\pi}{4} \cdot d_{m30}^{\ 2}$ Area of the unthreaded shank (M24)

$At_{m30} := 561mm^2$ Tensile stress area of the fastener (Shigley's)

$Am_{m30} := 519mm^2$ Minor Diameter area of the fastener (Shigley's)

$dp_{m30} := 27.727mm$ Pitch diameter of the thread form. (Mark's)

Fastener Strengths

Property Class	Size Range, Inclusive	Minimum Proof Strength,* MPa	Minimum Tensile Strength,* MPa	Minimum Yield Strength,* MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	4.6
4.8	M1.6–M16	310	420	340	Low or medium carbon	4.8
5.8	M5–M24	380	520	420	Low or medium carbon	5.8
8.8	M16–M36	600	830	660	Medium carbon, Q&T	8.8
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	9.8
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	10.9
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	12.9

Class 8.8 fasteners

$Sp_{88} := 600MPa$ Proof Strength of 8.8 fastener

$Sy_{88} := 660MPa$ Yield Strength of 8.8 fastner

$Su_{88} := 830MPa$ Ultimate strength of fastener

Class 10.9 fastener

$Sp_{109} := 830MPa$ Proof strength of 10.9 fastener

$Sy_{109} := 940MPa$ Yield Strength of 8.8 fastner

$Su_{109} := 1040MPa$ Ultimate strength of fastener

Class 12.9 fastener

$Sp_{129} := 970MPa$ Proof strength of 12.9 fastener

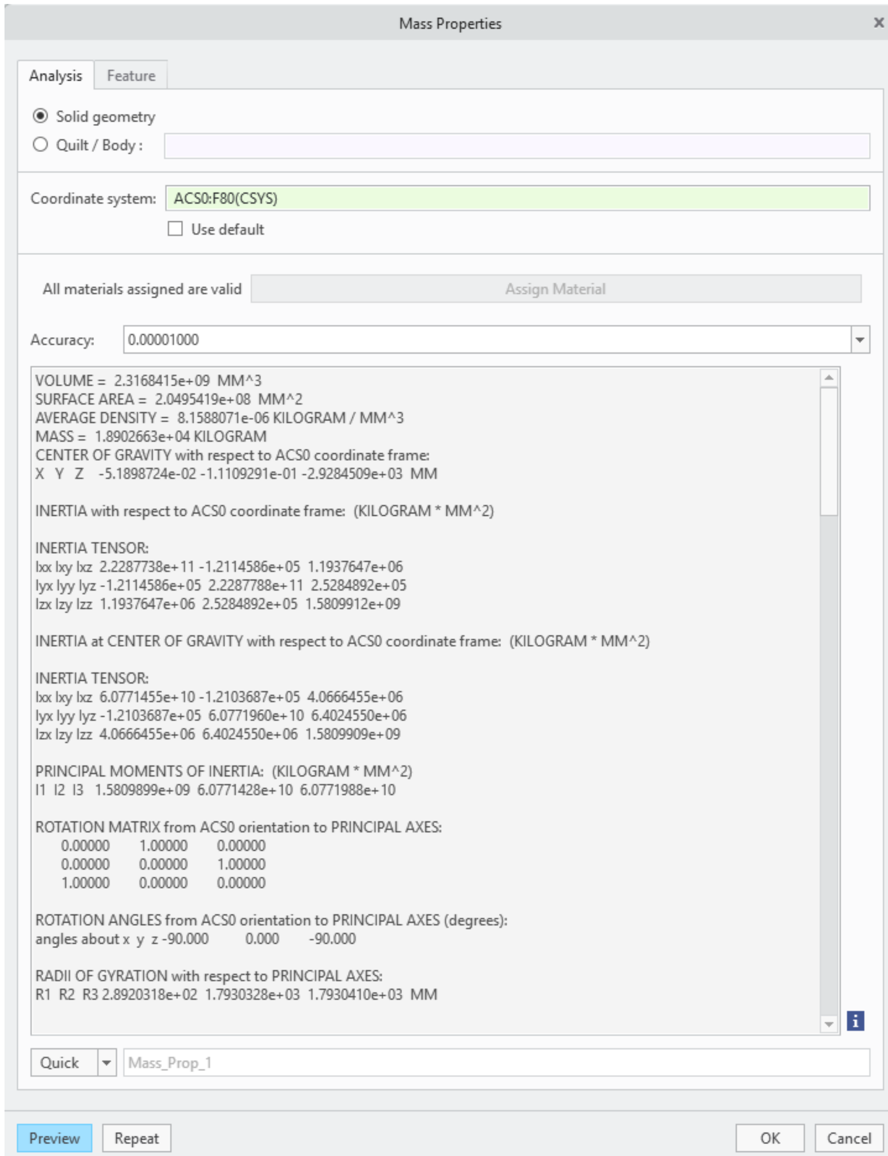
$Sy_{129} := 1100MPa$ Yield Strength of 8.8 fastner

$Su_{129} := 1220MPa$ Ultimate strength of fastener

Total System Properties

$m_{tot} := 18900kg$	Total mass of the target assembly
$m_{rot} := 17720kg$	Rotating mass, CAD model 0000004343
$I_{m\_tot} := 1.58 \cdot 10^{9} \cdot kg \cdot mm^{2}$	Mass moment of inertia, rotating system, CAD Model 000000434
$r_{iw} := 319mm$	Inner diameter of the Tungsten (from axis of rotation)
$r_{ow} := 571mm$	Outer diameter of the tungsten (from axis of rotation)
$h_{tot} := 7555mm$	Height from inlet to bottom of shaft
$l_{CG} := 3000mm$	Distance from bearing to CG of system.
$e_{imb} := 35kg \cdot m$	Imbalance of the target wheel (one segment missing)
$l_{bl} := 5633mm$	Distance from the beamline to the center of the bearing.

CAD Model Outputs, relative to bearing center



Loads

Seismic Loads

$A_v := 0.091g$ — Vertical acceleration, per S03020000-CAC10001

$A_h := 0.457g$ — Horizontal acceleration per S03020000-CAC10001

Live Loads

Task: Determine the loads (accelerations) due to constant rotation

$L_\omega(r) := r \cdot \omega_{rot}^{2}$ — Function to define centripetal acceleration

$L_{\omega pk} := L_\omega(r_{ow}) = 1.293 \cdot g$ — Maximum centripetal acceleration

Pressure loads

Task: determine the maximum pressure in the target system

$P_{cv} := 14.7psi$ — Vacuum "pressure" in the core vessel

$P := 75psi$ — Peak design pressure at the target assembly supply interface

$P_s := \rho_{water} \cdot h_{tot} \cdot g + \int_{r_{iw}}^{r_{ow}} \rho_{water} \cdot L_\omega(r)\, dr = 11.074 \cdot psi$ — Static head pressure

$P_{tot} := P + P_s = 86.074 \cdot psi$ — Maximum normal operating pressure in the target.

Note: There is additional fluid column height from the section of shaft below the target elevation, however, the centripatal forces add more pressure to the system than the additional height. The segment pressure dominates the system.

Dead Loads

$m_{dr\_stat} := 200kg$ — Mass of stationary portions of the drive housing

$m_{tot} := m_{rot} + m_{dr\_stat} = 1.792 \times 10^{4}\, kg$ — Total system mass

$D := m_{tot} \cdot g = 39.507 \cdot kip$ — Total dead load on the system

Estimate the overall loads on the bearing & core vessel interface.

$M_{imb} := e_{imb} \cdot \omega_{rot}^{2} \cdot l_{bl} = 4.378 \cdot kN \cdot m$ — Moment due to imbalance at speed.

Estimate the seismic moment and force reactions at the core vessel lid.

The sesimic motion of the target assembly is restrained by the snubber ring. The system is statically indeterminate and can be determined with a simple stiffness matrix. Assume the shaft contributes to the stiffness, but the segments have no stiffness contributions. Note - the stifness matrix in mathcad must of same units - we will manually have to keep track of units.

$K_{shaft} := 0 \cdot identity(6)$ — Initialize the global stiffness matrix for 2 nodes

$\delta_{max} := 12mm$ — Relative displacement of the snubber relative to the attachment base.

$i := 1..4$ $\quad$ $j := 1..4$ — Row and column counters

Element #1 - Upper Crown

$I_1 := 1.42 \cdot 10^{-4}$ — Area moment of inertia, upper crown (m^4)

$l_1 := 0.534$ — Length of the bearing race to lower crown (increased diameter) (m)

$k_1 := k_{loc}\left(\frac{E_{316}}{Pa}, I_1, l_1\right)$ — Flexural stiffness matrix for element #1 (v1, θ1,v2,θ2)

$K_{shaft_{0+i,0+j}} := k_{1_{i,j}} + K_{shaft_{0+i,0+j}}$ — Insert element 1 flexural stiffness matrix into the global stiffness matrix

Element #2 - Lower crown and shaft

$I_2 := 1.40 \cdot 10^{-3}$ — Area moment of inertia, shaft (m^4)

$l_2 := 5.180$ — Length of the crown to snubber

$k_2 := k_{loc}\left(\frac{E_{316}}{Pa}, I_2, l_2\right)$ — Flexural stiffness matrix for element #1 (v1, θ1,v2,θ2)

$K_{shaft_{2+i,2+j}} := k_{2_{i,j}} + K_{shaft_{2+i,2+j}}$ — Insert element 2 stiffness matrix into the global stiffness matrix.

Case #1 - Seismic event - target does not hit the snubber

Write the boundary conditions, displacement and seismic forces for horizontal acceleration, node 1 (bearing support) constrained.

$F_{ext} := \begin{pmatrix} 0 & 0 & m_{tot} & 0 & m_{tot} & 0 \end{pmatrix}^T \cdot \frac{A_h}{2 \cdot N}$ — External forces applied to degrees of freedom (v,θ)

$BC_{stat} := (1 \;\; 1 \;\; 0 \;\; 0 \;\; 0 \;\; 0)^T$ — Boundary condition status indicator

$BC_{app} := (0 \;\; 0 \;\; 0 \;\; 0 \;\; 0 \;\; 0)^T$ — Boundary condition values (must correspond to status flag)

Use the FEsolve program (appendix) to solve for displacements.

$\delta_{shaft} := FE_{solve}\left(K_{shaft}, BC_{stat}, BC_{app}, F_{ext}\right)$ — Displacements of DOF

$\delta_{shaft_5} = 0.03$ — The displacement (m) far exceeds the gap of the snubber; the moment is therfore limited by the motion of the shaft on the snubber.

Case #2 - Seismic displacement limited by snubber

The snubber, inits worst case position relative to the deflected core vessel lid is assumed to be permit 13mm of motion between snubber and mount. See S03000000-CAC10000 for derivations and rationale behind the maximum displacement.

$F_{ext2} := \begin{pmatrix} 0 & 0 & m_{tot} & 0 & m_{tot} & 0 \end{pmatrix}^T \cdot \frac{A_h}{2 \cdot N}$ — External forces applied to degrees of freedom (v,θ)

$BC_{stat2} := (1 \;\; 1 \;\; 0 \;\; 0 \;\; 1 \;\; 0)^T$ — Boundary condition status indicator

$BC_{app2} := (0 \;\; 0 \;\; 0 \;\; 0 \;\; 0.013 \;\; 0)^T$ — Boundary condition values (must correspond to status flag)

Use the FEsolve program to solve the displacements

$\delta_{shaft2} := FE_{solve}\left(K_{shaft}, BC_{stat2}, BC_{app2}, F_{ext2}\right)$ — Displacements

$\delta_{shaft2_5} = 0.013$ **OK** — Check to ensure that the displacements are valid

$R_2 := K_{shaft} \cdot \delta_{shaft2} - F_{ext2}$ — Check the reaction forces

$$R_2^T = \begin{pmatrix} -5.657 \times 10^4 & 1.152 \times 10^5 & 2.183 \times 10^{-11} & 2.328 \times 10^{-10} & -2.374 \times 10^4 & 1.164 \times 10^{-10} \end{pmatrix}$$

Select the force (DOF #1) and moment (DOF#2) as the bearing and assign as the peak moment the bearing will see.

$M_{seis} := R_{2_2} \cdot N \cdot m = 1.152 \times 10^5 \, J$ — Bearing moment computed from a freebody diagram without contact

$V_{seis} := \left| R_{2_1} \cdot N \right| = 5.657 \times 10^4 \, N$ — Seismic lateral force incurred by the bearing set

Select Primary Bearing for operation.

Select: SCHAEFFLER K81156-M Tapered Roller bearing for thrust and moment

Select: MORESCO-RP-42 oil for lubrication, under constant circulation

$r_{o\_brg} := 175mm$ Outer diameter of the bearing

$r_{i\_brg} := 140mm$ Inner diameter of the bearing

$C_{10} := 870kN$ Catalog - dynamic thrust rating

$C_{10\_revs} := 1 \cdot 10^{6} rev$ Revolutions at rated dynamic load

$C_{0a} := 3650kN$ Static Load Capacity

The bearing carries thrust and moment (by means of the duplex arrangement). Convert the loads to a per unit length measure for computing the bearing peak loads.

$$r_{avg\_brg} := \sqrt{\frac{r_{o\_brg}^{2} + r_{i\_brg}^{2}}{2}} = 158.469 \cdot mm$$

Midline of the circles (equal areas inside and outside)

$$A_{brg} := \pi \cdot r_{avg\_brg} = 497.846 \cdot mm$$

Circumference of the bearing

$$I_{brg} := \pi \cdot r_{avg\_brg}^{3}$$

Second moment of the bearing, treated as a line

$$P_{a\_brgn} := \frac{m_{rot} \cdot g}{A_{brg}} = 349.052 \cdot \frac{N}{mm}$$

Uniform force due to axial loads

$$P_{m\_brgn} := \frac{M_{imb} \cdot r_{avg\_brg}}{I_{brg}} = 55.495 \cdot \frac{N}{mm}$$

Peak force due to applied moment

$$P_{brgn} := P_{a\_brgn} + P_{m\_brgn} = 404.546 \cdot \frac{N}{mm}$$

Peak combined force in the bearing, due to applied moment (imbalance) and axial force

$$P_{10} := \frac{C_{10}}{A_{brg}} = 1.748 \cdot \frac{kN}{mm}$$

Lifetime capacity of the bearing, converted to force per unit length

Check Bearing Lifetime using simplified calculations

$F_{a\_brg} := m_{rot} \cdot g = 173.774 \cdot kN$

Axial load on the bearings during normal operation.

$\boxed{RI_{brg} := 7yr}$

Desired replacement interval of the bearing.

$L_{req} := RI_{brg} \cdot 5000 \frac{hr}{yr} = 3.5 \times 10^4 \cdot hr$

Desired operational hours before replacement

$$L_{10h} := \frac{C_{10\_revs}}{\omega_{rot}} \cdot \left(\frac{P_{10}}{P_{brgn}} \right)^{\frac{10}{3}} = 4.862 \times 10^4 \cdot hr$$

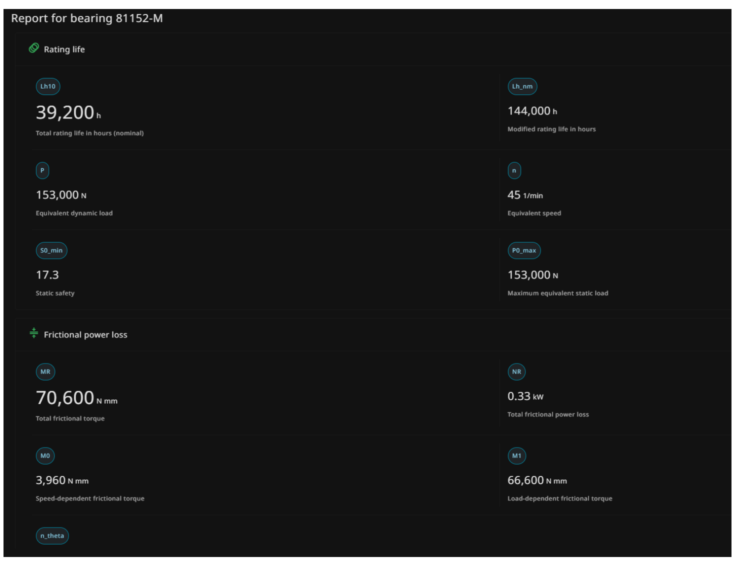
Basic Life of the bearing

$$\boxed{\frac{L_{10h}}{L_{req}} = 1.389}$$

Pass

The rated life of the bearing is greater than the that required for operation

Use the vendor rating for lifetime. LIfetime evaluations include oiling, cleanliness, etc.



Check the minimum Load

$C_0 := 3650\text{kN}$ Basic static load rating

$k_a := 1.4 \cdot \text{N}$ Minimum Axial load Factor for 811 series bearings

$$F_{a\_min} := 0.0005 \cdot C_{0a} + k_a \cdot \left[\frac{C_{0a} \cdot \omega_{rot}}{10^8 \cdot \left(\frac{\text{N}}{\text{s}}\right)}\right]^2 = 1.825 \cdot \text{kN}$$

$$\boxed{\frac{F_{a\_brg}}{F_{a\_min}} = 95.216}$$

OK

Factor to minimum loads

$T_{brg0} := 70600\text{N} \cdot \text{mm}$ Friction imposed on the system at operating speeds

Determine the torque on the nut to develop the preload (minimum load) on the lower bearing to prevent smearing damage (where rollers do not roll, but abrade or smear the surface of the race).

$d_{nut} := 260\text{mm}$ Nut thread major diameter (TR260)

$k_{nut} := 0.1$ Friction factor for a greased trapezoidal thread (estimated)

$$T_{pre\_trap\_nut} := k_{nut} \cdot d_{nut} \cdot F_{a\_min} = 47.451 \cdot \text{N} \cdot \text{m}$$

Check the capacity of the bearing to resist seismic loads (static failure - requirement is that the shaft is still supported).

The bearing carries thrust and moment (by means of the duplex arrangement). Convert the loads to a per unit length measure for computing the bearing peak loads.

$$P_{a\_brg} := \frac{m_{rot} \cdot g}{A_{brg}} = 349.052 \cdot \frac{\text{N}}{\text{mm}}$$

Uniform force due to axial loads

$$P_{m\_brg} := \frac{M_{seis} \cdot r_{avg\_brg}}{I_{brg}} = 1.461 \times 10^3 \cdot \frac{\text{N}}{\text{mm}}$$

Peak force due to applied moment

$$P_{brg} := P_{a\_brg} + P_{m\_brg} = 1.81 \times 10^3 \cdot \frac{\text{N}}{\text{mm}}$$

Peak combined force in the bearing, due to applied moment and axial force

$$P_{0a} := \frac{C_{0a}}{A_{brg}} = 7.332 \cdot \frac{\text{kN}}{\text{mm}}$$

Static capacity of the bearing, converted to force per unit length

$$\boxed{\frac{P_{0a}}{P_{brg}} = 4.051}$$

OK

Margin against exceeding the static capacity of the thrust bearing

Select needle bearing to resist lateral loads

Select SKF/280x265x50 needle roller bearing to resist lateral loads

Seismic Radial loads

$F_{r\_brg\_s} := \left|V_{seis}\right| = 5.657 \times 10^{4} \cdot N$ — Radial load on selected bearing.

$F_{r\_brg\_dlr} := 242 kN$ — Dynamic load rating of the bearing

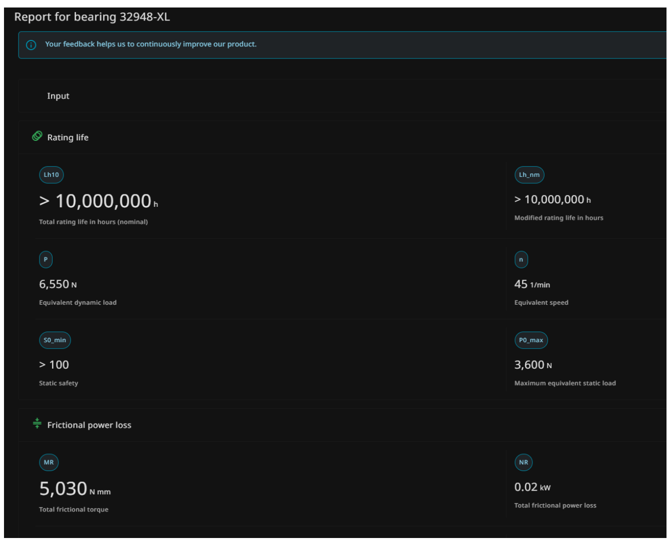
$F_{r\_brg\_cap} := 850 kN$ — Static radial load capacity. of selected bearing.

$$\boxed{\frac{F_{r\_brg\_cap}}{F_{r\_brg\_s}} = 15.026}$$

OK

The selected bearing has the capacity to handle the seismic radial loads

$T_{brg1\_min} := 5030 N \cdot mm$ — Frictional torque at selected prelaod



Check the minimum load on the bearing to avoid skipping or gliding

$F_{r\_brg\_min} := 0.02 \cdot F_{r\_brg\_dlr} = 4.84 \, kN$ — Required minimum radial load.

What level of eccentricity achieves this radial load?

$$e_{min} := \frac{F_{r\_brg\_min}}{m_{rot} \cdot \omega_{rot}^{2}} = 12.3 \, mm$$

Which would be nearly impossible to occur with machine tolerances.

Consider a bronze (babbit) bushing as an alternate f the pipe laods or imbalance are insufficient

$\mu_{babbit} := 0.08$

$$T_{brg1\_max} := F_{r\_brg\_min} \cdot \mu_{babbit} \cdot 265 \frac{mm}{2} = 51.304 \, N \cdot m$$

Which is more friction than the needle; use this value in motor sizing calcs

Determine the overall friction at steady state operation

$T_{brg} := 2 \cdot T_{brg0} + T_{brg1\_max} = 192.504 \cdot N \cdot m$ Bearing friction (from above), no preload

$T_{vac\_seal} := 75 N \cdot m$ Estimated friction in the vacuum seals (Chesterton)

$T_{grease\_seal} := 30 N \cdot m$ Estimated friction in the grease seals (lip)

$T_{rotary\_union} := 75 N \cdot m$ Estimated friction in the water seal at 45 rpm. DSTI Estimates 50-75 Nm

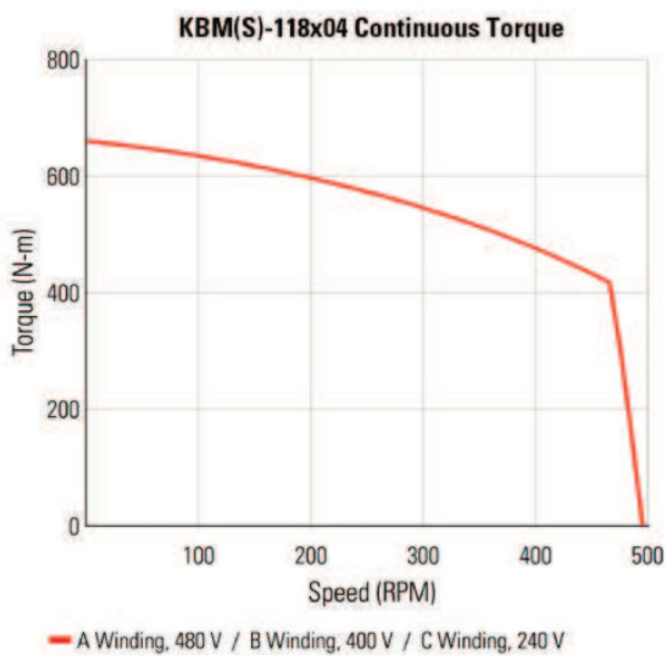
$T_{misc} := 10 N \cdot m$ Allowance for other friction sources

Overall torque due to firction

$$T_{fric} := T_{brg} + T_{vac\_seal} + 2 \cdot T_{grease\_seal} + T_{rotary\_union} + T_{misc} = 412.504 \cdot N \cdot m$$

$P_{mot} := \omega_{rot} \cdot T_{fric} = 1.944 \cdot kW$ Required steady state operating power for rotating the assembly

Select Kollmorge KBM-118H04 **series motor**



$T_{mot} := 650 N \cdot m$ Rated torque from the motor, at 45 rpm

$T_{pk} := 2400 N \cdot m$ Peak stall torque

$T_{cog} := 8.13 N \cdot m$ Cogging torque in the motor. 1% of Mo

$J_{motor} := 0.7 kg \cdot m^2$ Motor inertia

KBM(S)-118XXX PERFORMANCE DATA & MOTOR PARAMETERS									
Motor Parameter	Symbol	Units	TOL	KBM(S)-118X03-X			KBM(S)-118X04-X		
				A	B	C	A	B	C
Continuous Stall Torque at 25°C Amb. (1)	Tc	Nm	NOM	560	560	560	672	672	672
		lb-ft		413	413	413	495	495	495
Continuous Current	Ic	Arms	NOM	44.0	54.0	89.5	42.8	51.5	86.0
Peak Stall Torque (25°C winding temp)	Tp	Nm	NOM	1932	1932	1661	2400	2400	2068
		lb-ft		1425	1425	1224	1770	1770	1524
Peak Current	Ip	Arms	NOM	171	206	343	171	206	343.0
Rated Continuous Output Power at 25°C Amb. (1)	P Rated	Watts		17000	17000	17000	19850	19850	19850
	HP Rated	HP		22.8	22.8	22.8	26.6	26.6	26.6
Speed at Rated Power	N Rated	RPM		535	535	535	420	420	420
Torque Sensitivity (2)	Kt	Nm / Arms	+/-10%	12.8	10.7	6.40	16.0	13.4	8.00
		lb-ft / Arms		9.46	7.88	4.72	11.8	9.8	5.90
Back EMF Constant	Kb	Vrms/kRPM	+/- 10%	775	646	387	969	808	484
Motor Constant	Km	Nm/√watt	+/-10%	17.1	17.1	17.1	19.4	19.4	19.4
		lb-ft /√watt		12.6	12.6	12.6	14.3	14.3	14.3
Resistance (line to line)	Rm	Ohms	+/- 10%	0.373	0.259	0.093	0.455	0.298	0.112
Inductance	Lm	mH		4.3	3.0	1.1	4.5	3.0	1.2
Inertia (KBM)	Jm	Kg-m²		0.542			0.648		
		lb-ft-s²		0.400			0.478		
Weight (KBM)	Wt	Kg		71.7			88.5		
		lb		158			195		
Inertia (KBMS)	Jm	Kg-m²		0.591			0.698		
		lb-ft-s²		0.436			0.515		
Weight (KBMS)	Wt	Kg		73.9			90.7		
		lb		163			200		
Max Static Friction	Tf	Nm		12.8			16.0		
		lb-ft		9.42			11.8		
Cogging Friction (peak-to-peak)	Tcog	Nm		6.39			8.13		
		lb-ft		4.71			6.00		
Viscous Damping	Fi	Nm/ kRPM		81.3			100		
		lb-ft / kRPM		60.0			74.0		
Thermal Resistance (3)	TPR	°C / watt		0.078			0.069		
Number of Poles	P	-		38			38		
Recommended Kollmorgen AKD Drive				04807			04807		
Recommended Kollmorgen S700 Drive					S772	S772		S772	S772
Voltage Req'd at Rated Output	Vac Input	Vac		480	400	240	480	400	240
Peak Stall Torque (4) (Motor with Drive)	Tp Drive	Nm	+/-10%	1122	1358	850	1402	1698	1062
		lb-ft		828	1002	627	1034	1252	783
Cont. Stall Torque (4) (Motor with Drive)	Tc Drive	Nm	+/-10%	560	560	438	678	678	547
		lb-ft		413	413	323	500	500	403

Check operational capacity at steady state operation

$\boxed{M_{ss} := \frac{T_{mot}}{T_{fric}} = 1.576}$ OK Margin on steady state motor capacity

Check the ability to manually rotate the assembly with hand tools during maintenance

$\frac{T_{fric}}{1m} = 412.504\,N$ **OK** Force required to rotate at the end of a reasonable wrench.

$d_{pin\_spanner} := 5mm$ Diameter of the pin in the spanner

$d_{rot} := 234mm$ Diameter at which the pin spanner acts.

$F_{pin\_spanner} := \frac{2T_{fric}}{d_{rot}} = 792.603 \cdot lbf$ Force that is to be resisted in the pin

$\boxed{\frac{2300lbf}{F_{pin\_spanner}} = 2.902}$ **OK** margisn against shearing dowel pin (similar to 91595A178)

Check the ability to start and stop the system in the required timeframe

$\alpha_{start} := \frac{\omega_{rot}}{t_{start}} = 0.45 \cdot \frac{deg}{s^2}$ Angular acceleration during controlled start/shutdown.

$T_{start} := I_{m\_tot} \cdot \alpha_{start} + T_{fric} = 424.913 \cdot N \cdot m$ Torque required to start and stop rotation in the specified time.

$\boxed{M_{su} := \frac{T_{mot}}{T_{start}} = 1.53}$ **PASS** Margin for motor capacity tooperate in continuous duty range during start/stop

Check the inertial matching (Controllability)

$I_{reflected} := \frac{I_{m\_tot}}{J_{motor}} = 2.257 \times 10^3$

Typically the inertial matching is desired to be <200 for controllability (CNC motion, pick and place systems, etc.) We don't want dynamic performance, but instead are relying on the momentum of the flywheel to maintain pahse lock.
OK

Check Coast down time (no motor influence)

$I_{m\_tot} = 1.58 \times 10^{3}\, m^{2} \cdot kg$

Mass moment of inertia (repeated from within this calc package)

$\alpha_{decel} := \dfrac{T_{fric} + T_{cog}}{I_{m\_tot}} = 15.254 \cdot \dfrac{deg}{s^{2}}$

Rate of deceleration due to the torque in the system

$a_{block} := \alpha_{decel} \cdot r_{ow} = 0.016 \cdot g$

Equivalent acceleration at the the tip of the Tungsten

$\boxed{t_{stop} := \dfrac{\omega_{rot}}{\alpha_{decel}} = 17.701 \cdot s}$

Time required to bring the system to complete rest (approximate)

Determine the maximum torque imposed during a motor shunt or spurrious stop

$T_{stop} := T_{pk} + T_{fric} + T_{cog} = 2.821 \cdot kN \cdot m$

Maximum torque imparted to the structure

$\boxed{\alpha_{estop} := \dfrac{T_{stop}}{I_{m\_tot}} = 1.785 \cdot \dfrac{rad}{s^{2}}}$

Angular acceleration imposed during a shunted winding or similar event

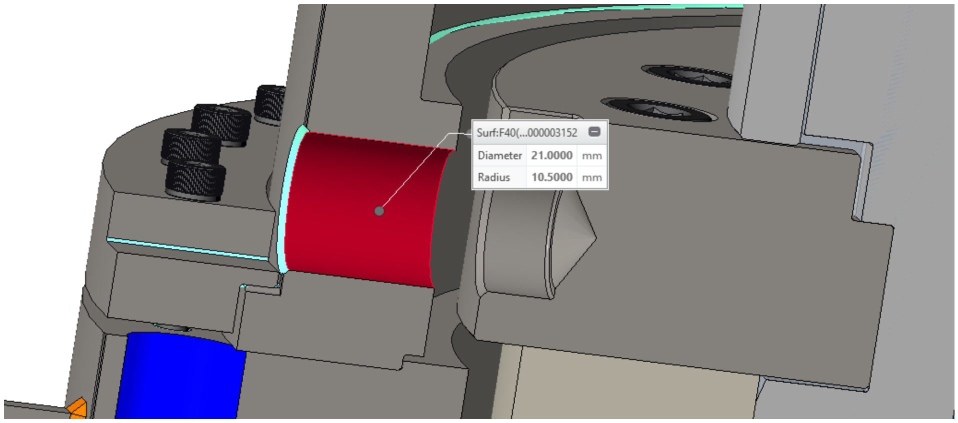
$t_{shunt} := \dfrac{\omega_{rot}}{\alpha_{estop}} = 2.64\, s$

Time to stop the system, shunted leads

Having a keyweay is not neccesary - the motor can't impart much force/ torque on the system. The motor is the fusible link, it simply isn't strong enough to do much harm. The angular decceleration is extremely small. Check that the angular deceleration does not harm the shaft or segments via FEA.

$\alpha_{estop} \cdot \dfrac{1.4m}{2} = 0.127 \cdot g$

Check the ability for the Lock-Out-Tag_out Pin to withstand the full motor torque. Mcmaster 93680A525 or similar



$dapp_{LOTO} := 330mm$

Diameter of the split ring where the LOTO pin engages

$$Fshear_{LOTO} := \frac{T_{pk} \cdot 2}{dapp_{LOTO}} = 14.545 \cdot kN$$

Shear force in the LOTO pin

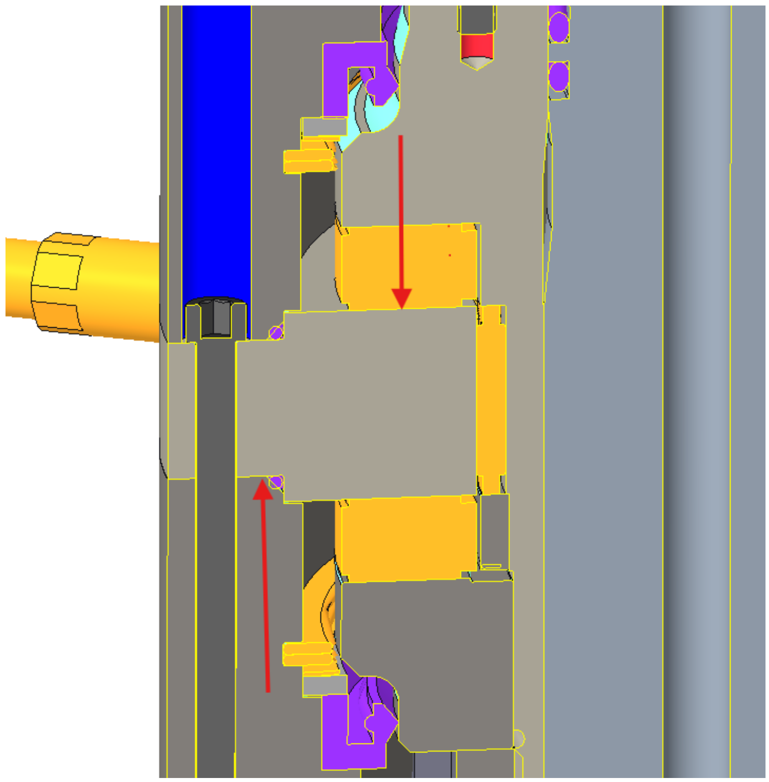
$$\boxed{\frac{90450lbf2}{Fshear_{LOTO}} = 55.322}$$

OK

The selected pin has 90,500# breaking strength. Significant margin

Component - Bearing Support Race

We must ensure that the bearing does not dish; causing premature wear on the bearing arrangement. Schaeffler states t"The bearings do not permit any skewing between the shaft and the housing. If angular misalignments occur between the locating surfaces on the shaft and in the housing, this will cause damage to the bearing and considerably reduce its operating life"



Dishing - Lower bearing race

Assume that the load acts at the inner edge of the race, and the race is simply supported at the inner diameter of the housing. Use Roark's Table 11.2, Case 1

$t_{race} := 50mm$ Thickness of the race

$do_{race} := 380mm$ Outer diameter of the bearing race

$di_{race} := 280mm$ Inner diameter of the race.

$d_{load} := 346mm$ Location of rollers (load application)

$w_{load} := \frac{m_{tot} \cdot g}{\pi \cdot d_{load}} = 0.162 \cdot \frac{kN}{mm}$ Distributed load on race.

Schaeffler does not publish allowable deflections their raceways. Instead, we have to interperet the allowables based upon the "tolerances" provided by a similar manufacturer, SKF.

$$\delta all_{race} := 25 \cdot 10^{-6} m$$ Allowable tolerance, for IT5 up to 315mm

$$\delta all_{race} = 0.025 \cdot mm$$

table 3 - Tolerances for solid steel shafts - seats for thrust bearings[1)]

Conditions	Shaft diameter	Dimensional tolerance[2)]	Total radial run-out tolerance	Total axial run-out tolerance	Ra
	mm	–	–	–	µm
Axial loads only on thrust ball bearings					
		h6	IT5/2	IT5	1,6[3)]
Combined radial and axial loads on spherical roller thrust bearings					
Stationary load on shaft washer	all	j6	IT5/2	IT5	1,6[3)]
Rotating load on shaft washer,	≤ 200	k6	IT5/2	IT5	1,6[3)]
or direction of load indeterminate	> 200 to 400	m6	IT5/2	IT5	1,6
	> 400	n6	IT5/2	IT5	1,6

Nominal mm		Standard Tolerance Grades µm						
		IT01	IT0	IT1	IT2	IT3	IT4	IT5
Above	Up to and including	µm						
—	3	0,3	0,5	0,8	1,2	2	3	4
3	6	0,4	0,6	1	1,5	2,5	4	5
6	10	0,4	0,6	1	1,5	2,5	4	6
10	18	0,5	0,8	1,2	2	3	5	8
18	30	0,6	1	1,5	2,5	4	6	9
30	50	0,6	1	1,5	2,5	4	7	11
50	80	0,8	1,2	2	3	5	8	13
80	120	1	1,5	2,5	4	6	10	15
120	180	1,2	2	3,5	5	8	12	18
180	250	2	3	4,5	7	10	14	20
250	315	2,5	4	6	8	12	16	23
315	400	3	5	7	9	13	18	25
400	500	4	6	8	10	15	20	27

Determine the constants for Roark's formulas Case 1 and 1e

$$D_{race} := \frac{E_{carbon\_steel} \cdot t_{race}^{3}}{12 \cdot \left(1 - \nu_{carbon\_steel}^{2}\right)} = 2.289 \times 10^{6}\,J$$

Cross section stiffness

$$C_1 := \frac{1 + \nu_{carbon\_steel}}{2} \cdot \frac{do_{race}}{di_{race}} + \frac{1 - \nu_{carbon\_steel}}{4} \cdot \left(\frac{do_{race}}{di_{race}} - \frac{di_{race}}{do_{race}}\right) = 0.991$$

$$C_4 := \frac{1}{2} \cdot \left[\left(1 + \nu_{carbon\_steel}\right) \cdot \frac{di_{race}}{do_{race}} + \left(1 - \nu_{carbon\_steel}\right) \cdot \frac{do_{race}}{di_{race}}\right] = 0.954$$

$$C_7 := \frac{1}{2} \cdot \left(1 - \nu_{carbon\_steel}^{2}\right) \left(\frac{do_{race}}{di_{race}} - \frac{di_{race}}{do_{race}}\right) = 0.282$$

$$L_3 := \frac{d_{load}}{4 \cdot do_{race}} \cdot \left[\left[\left(\frac{d_{load}}{do_{race}}\right)^{2} + 1\right] \cdot \ln\left(\frac{do_{race}}{d_{load}}\right) + \left(\frac{d_{load}}{do_{race}}\right)^{2} - 1\right] = 1.139 \times 10^{-4}$$

$$L_6 := \frac{d_{load}}{4 \cdot do_{race}} \cdot \left[\left(\frac{d_{load}}{do_{race}}\right)^{2} - 1 + 2 \cdot \ln\left(\frac{do_{race}}{d_{load}}\right)\right] = 3.761 \times 10^{-3}$$

$$L_9 := \frac{d_{load}}{do_{race}} \cdot \left[\frac{1 + \nu_{carbon\_steel}}{2} \cdot \ln\left(\frac{do_{race}}{d_{load}}\right) + \frac{1 - \nu_{carbon\_steel}}{4} \cdot \left[1 - \left(\frac{d_{load}}{do_{race}}\right)^{2}\right]\right] = 0.083$$

Case 1 - Simply supported outer edge

$$\delta i_{race} := \frac{w_{load} \cdot do_{race}^{3}}{4D_{race}} \cdot \left(\frac{C_1 \cdot L_9}{C_7} - L_3\right) = 0.281 \cdot mm$$

Deflection

$$\frac{\delta all_{race}}{\delta i_{race}} = 0.089$$

Not acceptable deflection

But it is not really simply supported, compare with case 1e of Roark's Table 11.2

$$\delta i_{race\_c} := \frac{w_{load} \cdot do_{race}^{3}}{4D_{race}} \cdot \left(\frac{C_1 \cdot L_6}{C_4} - L_3\right) = 3.674 \times 10^{-3} \cdot mm$$

$$\frac{\delta all_{race}}{\delta i_{race\_c}} = 6.805$$

OK

Acceptable deflection

Check Deflection with an FEA model

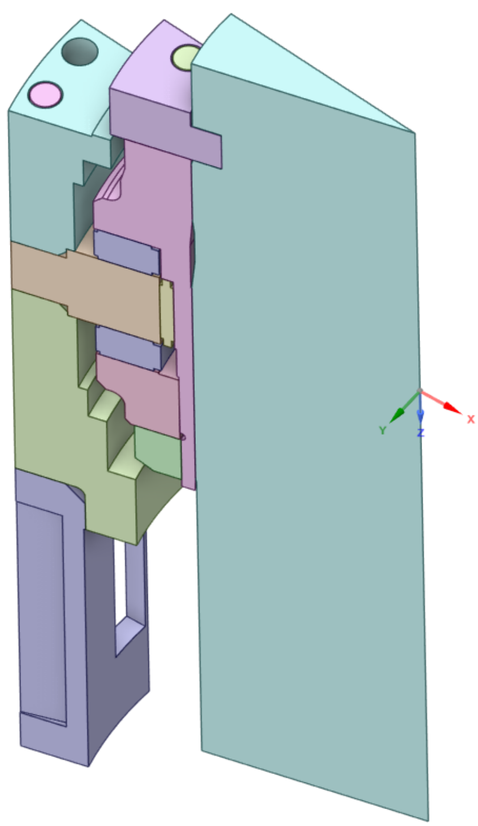
Since neither boundary condition is entirely appropriate,we turn to FEA to determine a more representative deflection. The ANSYS model is an axisymetric (cyclic) model that includes the effects of:

-Bolt pretension

-Contact surfaces (allows opening of gaps)

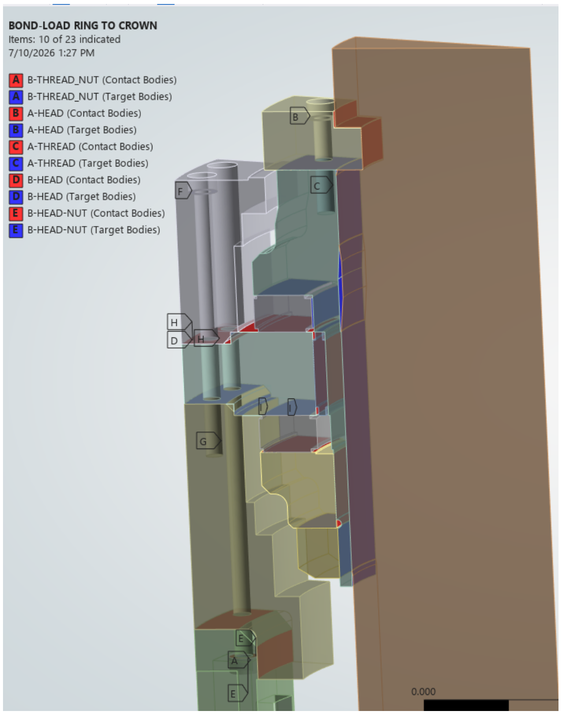
The model more accurately predicts the load path through the drive system than the simplifed structural model described in S03020000-DAC10002. The details of this Finite Element model are beyodn the scope of this document, but may be examined in the FILE: BEARING_DEFLECTION_3D.wbpj. This model is used to determine the magnitude of deflection.

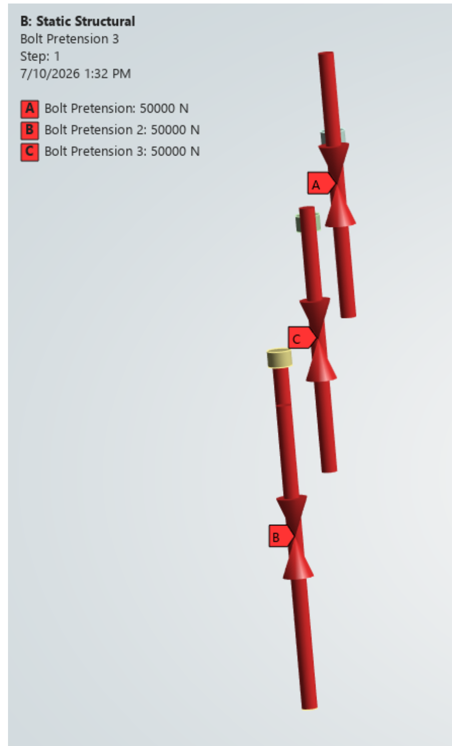
Geometry



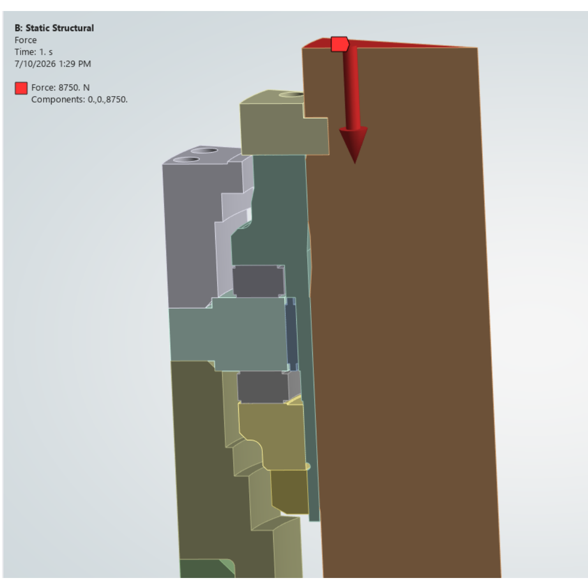
Mesh

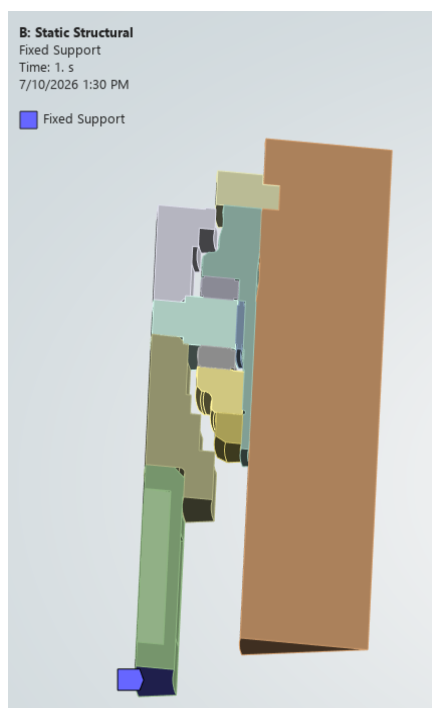
Contact pairs



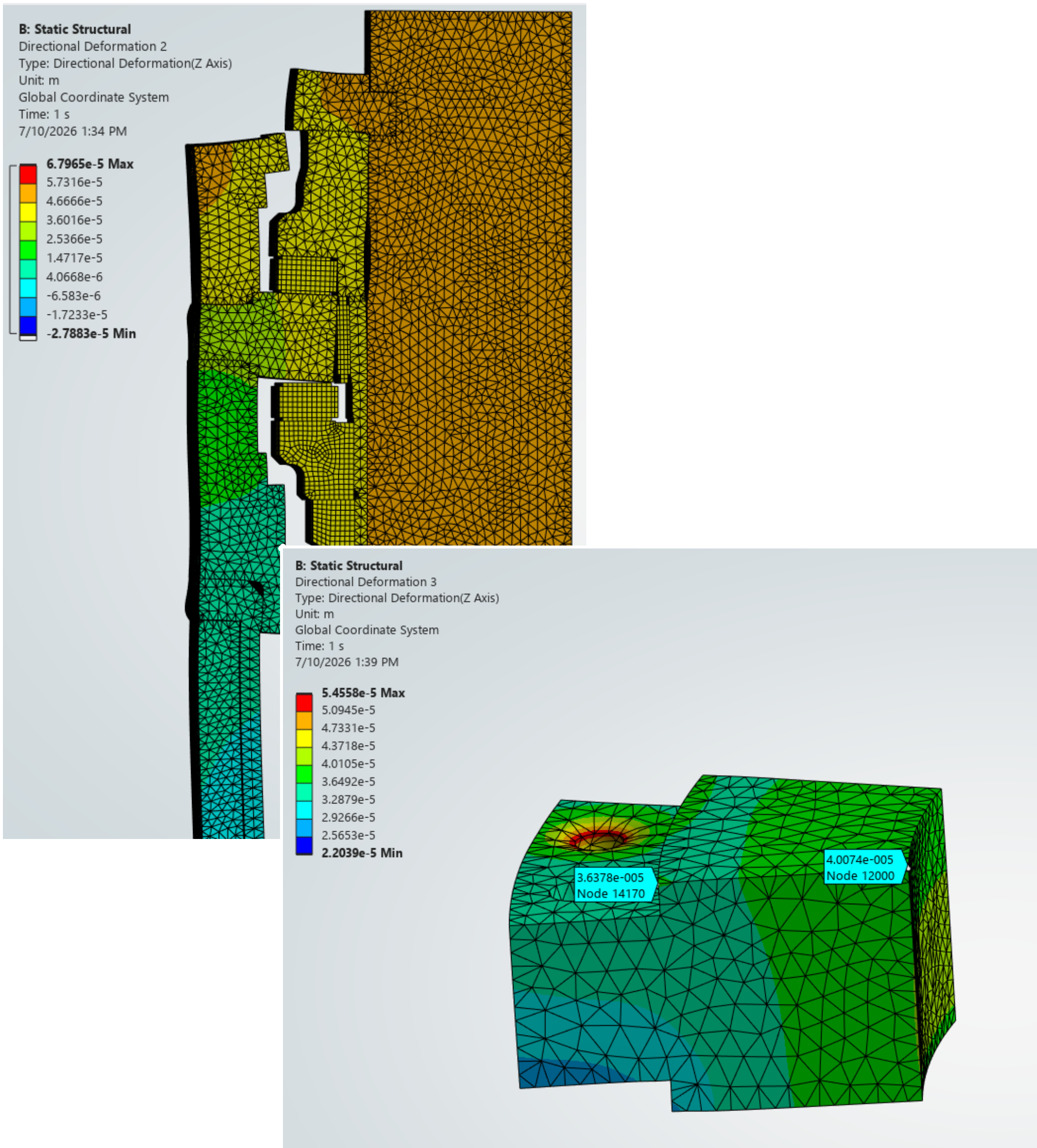


Boundary Conditions





Results

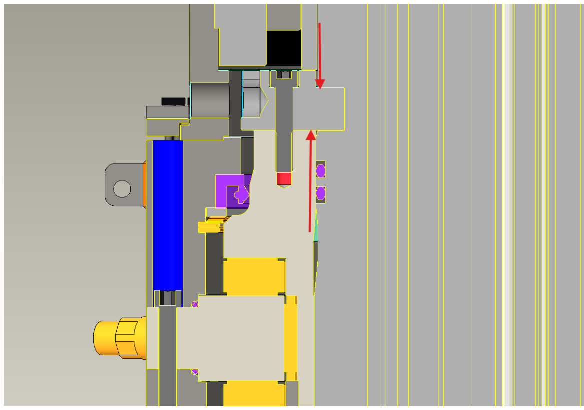


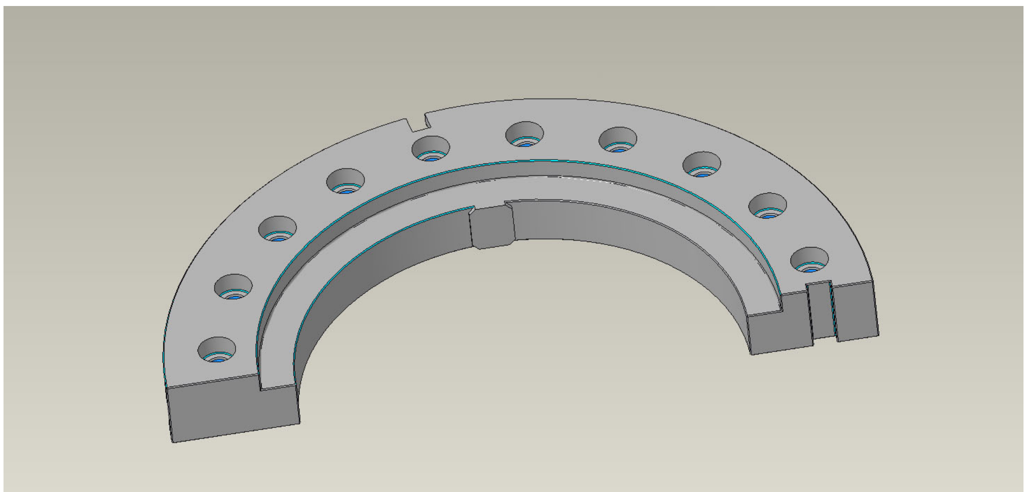
So extract (probe) the deflections between the inner and outer diameter to ensure that the race deforms within the manufacturing tolerances.

$$\delta_{raceFEA} := 4.0074 \cdot 10^{-5} m - 3.6378 \cdot 10^{-5} m = 3.696 \times 10^{-3} \cdot mm$$

$$\frac{\delta all_{race}}{\delta_{raceFEA}} = 6.764$$ **OK**

COMPONENT - Load Support RIng





Check the shear stress pof the ring supporting the weight of the assembly

$h_{lsr} := 25\text{mm}$ — Thickness of the ring though the shear plane

$d_{lsr} := 243\text{mm}$ — Diameter of the shear plane

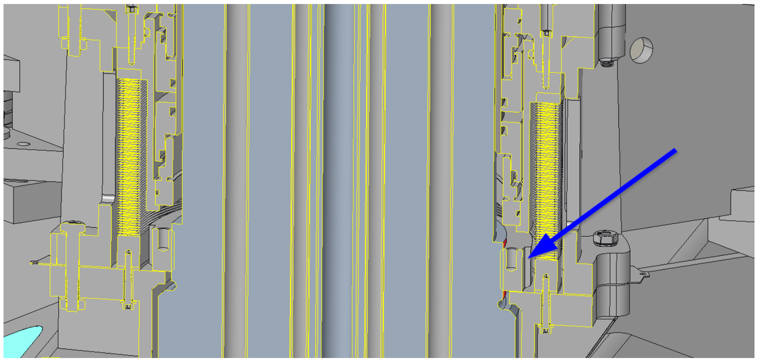
$Ashear_{lsr} := \pi \cdot d_{lsr} \cdot h_{lsr} = 1.909 \times 10^{4} \cdot \text{mm}^{2}$ — Shear area of the load support ring

$\tau_{lsr} := \dfrac{m_{tot} \cdot g}{Ashear_{lsr}} = 9.208 \cdot \text{MPa}$ — Shear stress in the load support ring

$\boxed{\dfrac{Sy_{4340}}{\sqrt{3} \cdot \tau_{lsr}} = 73.361}$ **OK** — Margin against exceeding shear capacity

COMPONENT - Maintenance Support Locknut

Select: Timken HML52T to support the load of the shaft during maintenance, with a trapezoidal thread of TR260x4.



Ensure that the crown trapezoidal thread has the capacity to support the load

$F_{thrd} := m_{tot} \cdot g = 175.735 \cdot kN$ — Vertical load that the nut has to support

$pitch := 4mm$ — Spacing between threads (metric)

$h_{thrd} := 29.5mm$ — Height of the threaded surface

$n_{thrd} := trunc\left(\frac{h_{thrd}}{pitch}\right) = 7$ — Number of threads engaged

$d_{thrd\_maj} := 259.7mm$ — Major diameter of thread form

$d_{thrd\_min} := 254.979mm$ — Minor diameter of the thread form

$a_{thrd} := 0.5mm$ — Major diameter clearance

$b_{thrd} := 1.5mm$ — Minor diameter clearance

$dm_{thrd} := \frac{d_{thrd\_maj} + d_{thrd\_min}}{2} = 257.339 \cdot mm$ — Mean diameter of the thread form

$fric := 0.15$ — Coeffecient of friction, oiled threads

$\alpha_{thrd} := 30\frac{deg}{2}$ — Thread angle, trapezoidal threads

Ensure that the bearing stress on the lid is acceptable

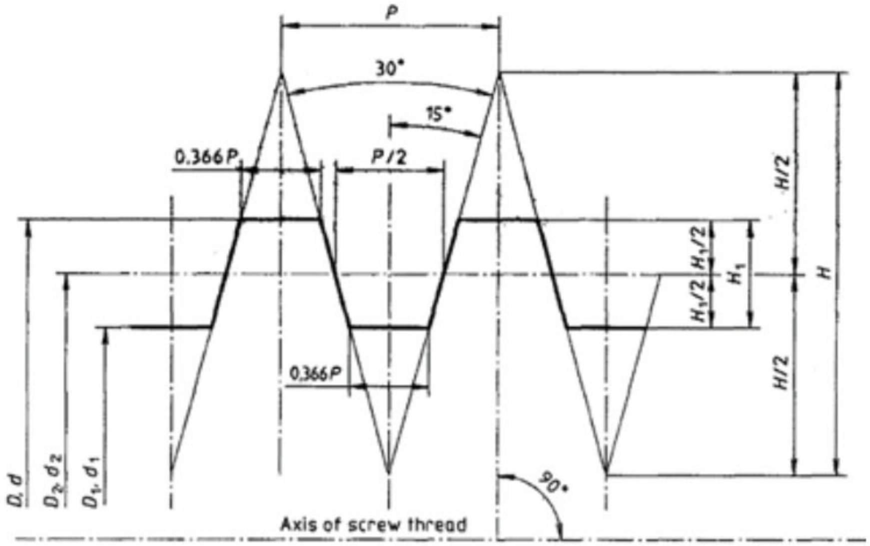
$do_{brg\_ln} := 290mm$ — Outer diameter of the bearing contact patch.

$do_{brg\_lid} := 270mm$ — Inned diameter of the core vessel lid.

$Abrg_{cv\_lid} := \frac{\pi}{4} \cdot \left(do_{brg\_ln}^{\ 2} - do_{brg\_lid}^{\ 2}\right) = 8.796 \times 10^3 \cdot mm^2$ — Bearing area of the locknut on the lid.

$\sigma_{bg\_lid} := \frac{m_{tot} \cdot g}{Abrg_{cv\_lid}} = 19.978 \cdot MPa$ **OK**

Estimate the thread stress in the shaft at the root of the thread form is acceptable



$H_{thrd} := 1.866 \cdot pitch = 7.464 \times 10^{-3}\, m$ — Per DIN 103

$h_{as} := 0.25 \cdot pitch = 1 \cdot mm$ — Per DIN 103

$h_s := 0.5 \cdot pitch + a_{thrd} = 2.5 \cdot mm$ — Per DIN 103

$w_{base} := 2 \cdot \left(\frac{H_{thrd}}{2} - h_{as} + h_s\right) \cdot \sin(\alpha_{thrd}) = 2.708 \cdot mm$ — Width of the base of the thread

$I_{base} := \frac{n_{thrd} \cdot \pi \cdot dm_{thrd} \cdot w_{base}^3}{12} = 9.368 \times 10^3 \cdot mm^4$ — Moment of inertia, thread form in bending

$A_{base} := \pi \cdot dm_{thrd} \cdot n_{thrd} \cdot w_{base} = 1.533 \times 10^4 \cdot mm^2$ — Cross sectional area of the thread from

$M_{thrd} := F_{thrd} \cdot \frac{h_s}{2} = 219.669\, J$ — Moment applied at the mean diameter of the thread

$\sigma_{b\_thrd} := \frac{M_{thrd} \cdot h_s}{2 \cdot I_{base}} = 29.311 \cdot MPa$ — Bending stress at the root of the thread form

$\tau_{thrd} := \frac{F_{thrd}}{A_{base}} = 11.466 \cdot MPa$ — Shearing stresses

$\sigma_{eff\_thrd} := \frac{1}{\sqrt{2}} \cdot \sqrt{\sigma_{b\_thrd}^2 + 6 \cdot \tau_{thrd}^2} = 28.705 \cdot MPa$ — Effective stress in the screw threads

$\boxed{\frac{Sy_{316}}{\sigma_{eff\_thrd}} = 8.407}$ **OK** — Margin against exceeding yield on the screw threads

Check the cross sectional area behind the screw threads of the bearing assembly

$d_{i\_ba} := 201\,mm$ Inner diameter of the bearing assembly

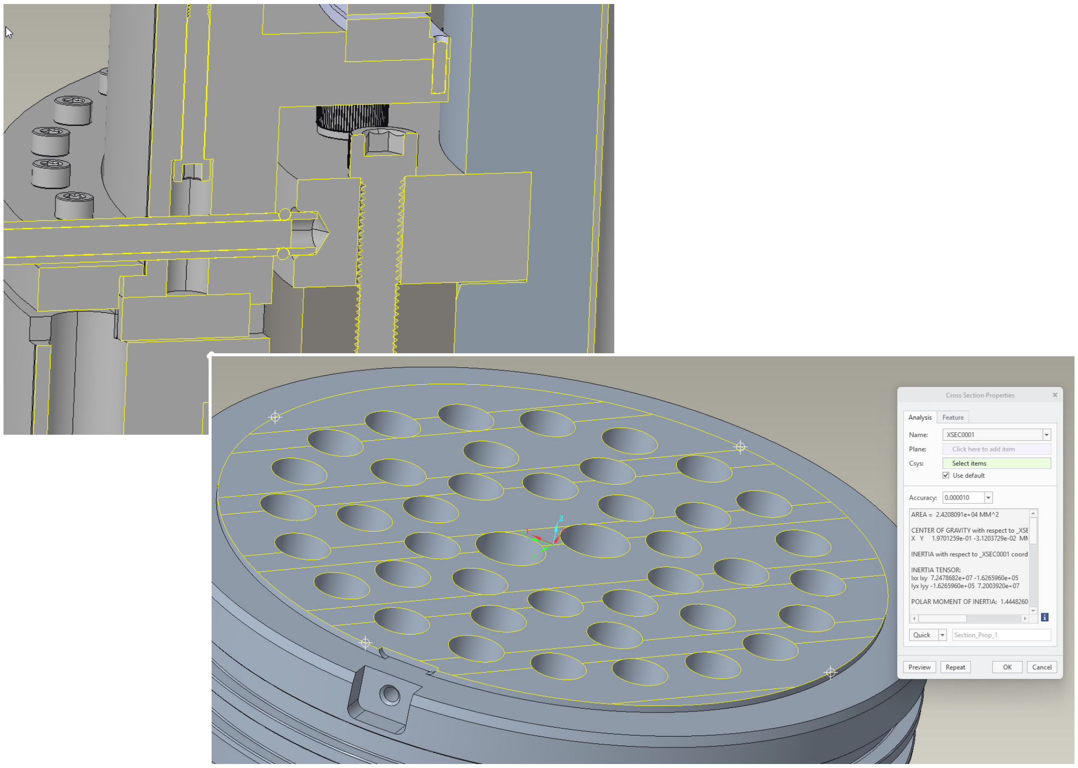
$A_{xc} := \frac{\pi}{4}\cdot\left(d_{thrd\_min}^{\ 2} - d_{i\_ba}^{\ 2}\right) = 0.019\,m^2$ Cross sectional area of the bearing assembly through the load support region

$\sigma_{ba\_ta} := \frac{m_{rot}\cdot g}{A_{xc}} = 8.989\cdot MPa$ Tensile stress through the cross section

$\boxed{\frac{S_{316}}{\sigma_{ba\_ta}} = 12.793}$ **OK** Check stresses against allowable stress (primary stress)

Component - Crown

The crown stresses are covered in detail in S03020000-DAC100002, but we will cover the direct load path briefly.



$$d_{crown} := 210mm$$

Diameter of the crown

$$XC_{crown} := 2.42 \cdot 10^{4} mm^{2}$$

Cross sectional area of the crown

$$I_{xc\_crown} := 7.2 \cdot 10^{7} mm^{4}$$

Second moment of inertia, crown

$$\sigma_{crwon\_op} := \frac{m_{tot} \cdot g}{XC_{crown}} + \frac{M_{imb} \cdot d_{crown}}{2 \cdot I_{xc\_crown}} = 1.979 \cdot ksi$$

Tensile stress in the cross section, operation

$$\sigma_{crown\_seis} := \frac{m_{tot} \cdot \left(g + A_{v}\right)}{XC_{crown}} + \frac{M_{seis} \cdot d_{crown}}{2 \cdot I_{xc\_crown}} = 25.523 \cdot ksi$$

Tensile stress in the crown, due to seismic forces

$$\boxed{\frac{S_{625}}{\sigma_{crown\_seis}} = 1.175}$$

OK

Margin to ASME allowable stress

COMPONENT - Load split collar

The load split collar supports the entire weight of the target assembly in shear and bears against the crown. Ensure that the collar is capable of handling the loads with sufficient margin, and ensure that the interaction does not damage the shaft.

$d_{o\_collar} := 320mm$ — Outer diameter of the split collar

$d_{i\_collar} := 210mm$ — Inner diameter of the split collar

$h_{collar} := 25mm$ — height of the split collar

$d_{load\_collar} := 241mm$ — Diameter at which the load is applied

Check shear capacity

$A_{s\_collar} := \pi \cdot d_{load\_collar} \cdot h_{collar} = 0.019 \cdot m^2$ — Shear area of the collar

$\tau_{collar} := \dfrac{m_{rot} \cdot g}{A_{s\_collar}} = 1.332 \cdot ksi$ — Shear stress in the load collar

$\boxed{\dfrac{Sy_{4130}}{\tau_{collar} \cdot \sqrt{3}} = 73.578}$ **OK** — Margin against shear (yield stress)

Check bearing against the crown (avoid deformation)

$A_{brg\_collar} := \dfrac{\pi}{4} \cdot \left(d_{load\_collar}{}^2 - d_{i\_collar}{}^2\right) = 0.011 \cdot m^2$ — Bearing surface area (to crown)

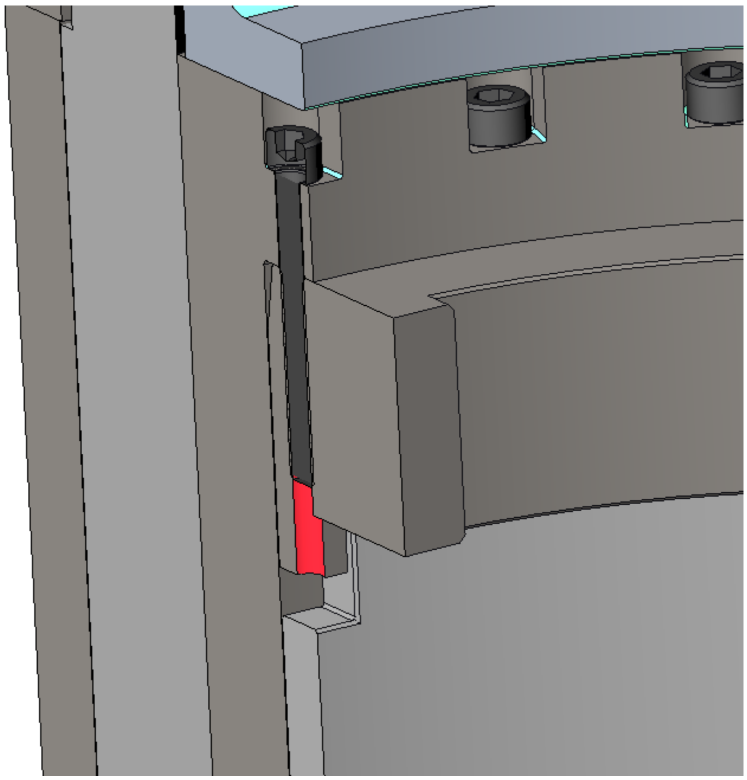
$\sigma_{brg\_collar} := \dfrac{m_{rot} \cdot g}{A_{brg\_collar}} = 2.295 \cdot ksi$ — Bearing stress against the crown

$\boxed{\dfrac{S_{316}}{\sigma_{brg\_collar}} = 7.267}$ **OK** — Factor against exceeding the ASME allowables in bearing.

Check the load path of the motor back to ground

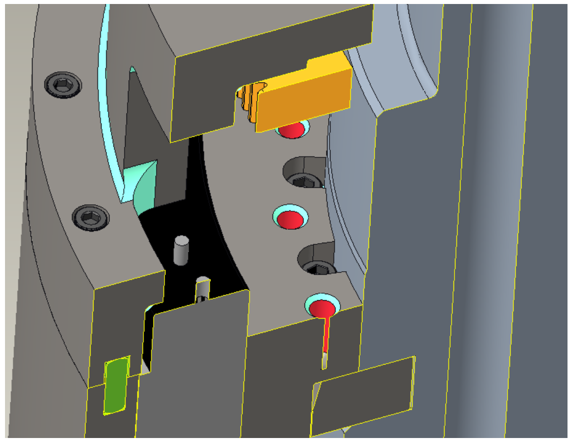
COMPONENT - Crown Keyway

Check the capacity for the keys to transmit torque. The crown keyway is to be durable and transmit the full capacity of the motor.



$d_{keyway} := 204mm$	Diameter of the crown at the keyway
$w_{key} := 20mm$	Width of key
$h_{key} := 35mm$	Height of the key.
$A_{shear} := w_{key} \cdot h_{key} = 700 \cdot mm^2$	Shear area of the key pattern
$S_{1040} := 90ksi$	Tensile capacity (ultimate) for 1040 steel
$\tau_{key} := \dfrac{2T_{pk}}{d_{keyway} \cdot A_{shear}} = 4.875 \cdot ksi$	Shear stress in the key at peak motor torque.
$\boxed{FS_{key} := \dfrac{S_{1040}}{\sqrt{3} \cdot \tau_{key}} = 10.658}$ **OK**	Factor of safety against exceeding the capacity of the keyway, during a maximum motor deceleration

Check the capacity of the bolts secruring the motor to transmit peak torque without slipping



$N_{bolts\_mot} := 11$ — Number of bolts securing the motor to the ring.

$d_{bolt\_mot} := 246mm$ — Diamter of the bolt circle

Select M6x1, 30mm long, class 12.9 SHCS

$d_{m6} = 6 \cdot mm$ — Nominal diameter of the selected bolt

$F_{pre} := 7.5kN$ — Desired preload in each fastener

$k_{bf} := 0.3$ — Bolt friction constant, black oxide (Shigley's)

$T_{pre} := F_{pre} \cdot k_{bf} \cdot d_{m6} = 13.5 \cdot N \cdot m$ — Fastener torque specification (drawing)

Check if there is enough induced friction from preload to transmit the torque

$\mu_{scj} := 0.3$ — Minimum friction factor for slip critical joint (AISC)

$$T_{scj\_mot\_cap} := N_{bolts\_mot} \cdot \frac{d_{bolt\_mot}}{2} \cdot F_{pre} \cdot \mu_{scj} = 3.044 \cdot kN \cdot m$$

Torque capacity of the joint, relying on friction

$$\frac{T_{scj\_mot\_cap}}{T_{pk}} = 1.268$$

Margin against exceedign the torque capacity of the joint between motor and ring.

Check capacity of fasteners to withstand the preload (no service loads)

$At_{m6} = 20.1 \cdot mm^2$ — Tensile area of M6 fastener (from above)

$Sp_{129} = 970 \cdot MPa$ — Proof strength of the fastener (from above)

$\sigma_{bolts\_mot} := \frac{F_{pre}}{At_{m6}} = 373.134 \cdot MPa$ — Tensile stress due to preload

$\frac{Sp_{129}}{\sigma_{bolts\_mot}} = 2.6$ — Margin against exceeding proof strength at pre-load

Check capacity of the tapped holes in the ring to withstand the preload without stripping

$l_{thrd} := 35mm$ — Length of the fastener in the tapped hole

$F_{thrd\_cap} := \frac{\pi \cdot dp_{m6} \cdot Sy_{316} \cdot l_{thrd}}{\sqrt{3} \cdot 3} = 26.615 \cdot kN$ — Pullout capacity of the threaded hole

$\frac{F_{thrd\_cap}}{F_{pre}} = 3.549$ **OK** — Margin against pulling threads out due to preload

Component - Motor housing shrink fit

The stator of the motor is secured to the houing by means of a thermal shrink/press fit. Esnure that the tolerances are appropriate to fully transmit the torque.

$l_{housing} := 254mm$	Height of the stator mounting surface.
$do_{housing} := 390mm$	Outer diameter of the stator housing
$do_{stator} := 361.11mm$	Outer diameter of the stator
$di_{stator} := 300mm$	Inner daimeter of the stator
$E_{housing} := 200GPa$	Elastic modulus of the housing
$\nu_{housing} := 0.2$	Poisson's ratio of 416 stainless steel (Matweb)
$E_{stator} := 70GPa$	Elastic modulus of the stator (assumed)
$\nu_{stator} := 0.3$	Assumed poisson's ratio for the stator.
$\delta_{MMC} := 0.22mm$	Diametral Interference at MMC (MAX) (2x gap)
$\delta_{LMC} := 0.04mm$	Interference at LMC
$\mu_{sf} := 0.3$	Conservative value for coeffecient fo static friction.

Using Equation 3-56 for interference fits from Shigley's Machine Design to build a reuasable formula; this fit willb e evaluated at MMC, LMC and after thermal expansion.

$$P_{pf}(\delta, r_o, R, r_i, E_o, E_i, \nu_o, \nu_i) := \frac{\delta}{R\cdot\left[\frac{1}{E_o}\cdot\left(\frac{r_o^2 + R^2}{r_o^2 - R^2} + \nu_o\right) + \frac{1}{E_i}\cdot\left(\frac{R^2 + r_i^2}{R^2 - r_i^2} - \nu_i\right)\right]}$$

Check at Max Material Condition (MMC)

$$P_{MMC} := P_{pf}(\delta_{MMC}, do_{housing}, do_{stator}, di_{stator}, E_{housing}, E_{stator}, \nu_{housing}, \nu_{stator}) = 4.36\cdot MPa$$

Check the torque capacity at this pressure

$$T_{MMC} := P_{MMC}\cdot(\pi\cdot do_{stator}\cdot l_{housing})\cdot\frac{do_{stator}}{2}\cdot\mu_{sf} = 6.805\times 10^4\,J$$

$$\boxed{\frac{T_{MMC}}{T_{pk}} = 28.353}$$

OK

The fit has the capacity to transmit torque at room temperature.

Check the housing stress.

$$\sigma t_{MMC} := \frac{do_{stator}^2\cdot P_{MMC}}{do_{housing}^2 + do_{stator}^2}\cdot\left(1 + \frac{do_{housing}^2}{do_{stator}^2}\right) = 4.36\cdot MPa$$

Tangential stress at the housing inner radii.

Check at Least Material Condition (LMC)

$$P_{LMC} := P_{pf}(\delta_{LMC}, do_{housing}, do_{stator}, di_{stator}, E_{housing}, E_{stator}, \nu_{housing}, \nu_{stator}) = 0.793\cdot MPa$$

Check the torque capacity at this pressure

$$T_{LMC} := P_{LMC}\cdot(\pi\cdot do_{stator}\cdot l_{housing})\cdot\frac{do_{stator}}{2}\cdot\mu_{sf} = 1.237\times 10^4\,J$$

$$\boxed{\frac{T_{LMC}}{T_{pk}} = 5.155}$$

OK

The fit has the capacity to transmit torque at room temperature.

Check the ability to assemble at reasonable temperatures/

To assemble the interference fit, we plan to shrink the housing into place. Kollmoregen reccomends against axially pressing the stator into the housing. CHeck the thermal fit if made at MMC.

$$\alpha_{416} := 11 \cdot 10^{-6} \cdot \frac{1}{K}$$

Coeffecient of Thermal Expansion, for 416 stainless, Matweb (0-300C)

$$\alpha_{crs} := 12.8 \cdot 10^{-6} \cdot \frac{1}{K}$$

Coeffecient of thermal expansion, for stator (email).

$$di_{housing\_rt} := do_{stator} - \delta_{MMC} = 360.89 \cdot mm$$

Minimum diameter of the housing at room temperature.

$$\Delta T_{assm} := 125\ °C$$

Temperature increase for thermal fit. Termperature does not overhat motor windings.

$$\delta_{ht} := \alpha_{416} \cdot di_{housing\_rt} \cdot \Delta T_{assm} = 1.581 \cdot mm$$

Diametral expansion of the ring.

$$\frac{\delta_{ht} - \delta_{MMC}}{2} = 0.68 \cdot mm$$

Radial gap between stator and housing at elevated temperatures.

Check the ability to maintain torque at increased operating temperatures.

Ensure that if the motor and housing are at the peak temperature (155C) that at LMC, they are still able to transmit the required torque. Assume that both bodies are at the same temperature. (The housing should be lower - it is convecting to the ambient) Assume that the

> The thermal element in the stator windings (PTC or KTY) must be wired to the control circuit of the application to make sure, that the motor temperature is supervised and the motor is protected from overheating. It must be ensured, that winding temperature never exceeds 155°C.

$$\Delta T_{op} := 125\ °C$$

Temperature rise at maximum motor temperature of 155C.

$$di_{housing\_rt\_LMC} := do_{stator} - \delta_{LMC} = 361.07 \cdot mm$$

Diameter of the housing at LMC, room temperature

$$\delta_{op\_housing} := di_{housing\_rt\_LMC} \cdot \alpha_{416} \cdot \Delta T_{op} = 1.581 \cdot mm$$

Diametral expansion of the housing.

$$\delta_{op\_stator} := do_{stator} \cdot \alpha_{416} \cdot \Delta T_{op} = 1.582 \cdot mm$$

Diametral expansion of the stator

$$\delta_{op} := \delta_{LMC} + \left(\delta_{op\_stator} - \delta_{op\_housing}\right) = 0.04 \cdot mm$$

Interference at peak operating temperature

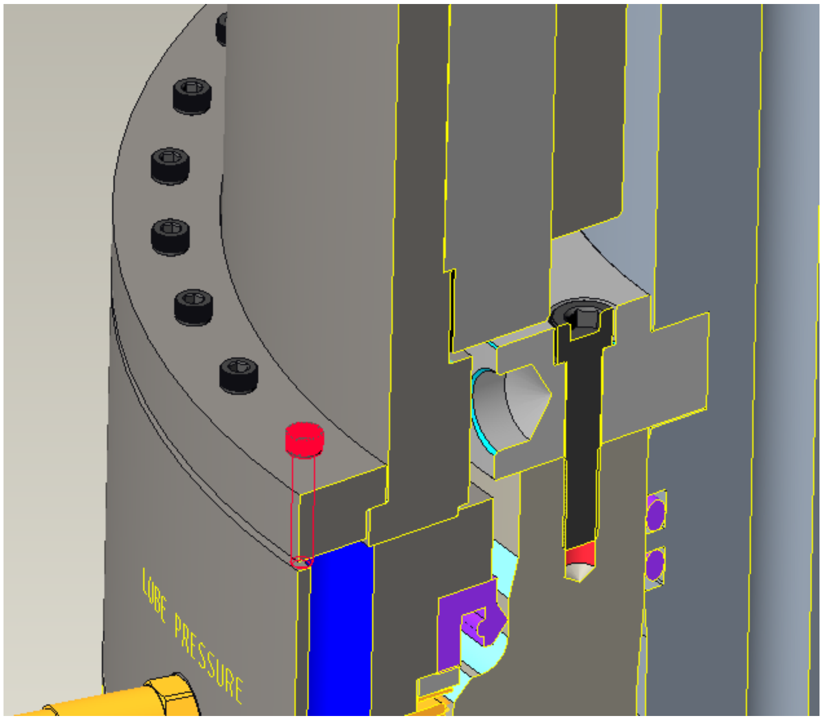
So the bore must be machined relative to the as-supplied motor diameter with the follwoing bias & tolerance.

D -

$$\delta_{MMC} = 0.22 \cdot mm$$

$$\delta_{LMC} = 0.04 \cdot mm$$

Component - Housing Clamping Ring



$N_{bolts\_clamp} := 40$ — Number of bolts securing the housing to the drive

$d_{clamp} := 390mm$ — Diamter of the clamping interface

Select M5x0.8, 30mm long, class 12.9 SHCS

$d_{m5} = 5 \cdot mm$ — Nominal diameter of the selected bolt

$F_{pre\_clamp} := 3kN$ — Desired preload in each fastener

$k_{bf} = 0.3$ — Bolt friction constant (from above), Shigley's

$T_{pre\_clamp} := F_{pre\_clamp} \cdot k_{bf} \cdot d_{m5} = 4.5 \cdot N \cdot m$ — Fastener torque specification (drawing)

Check if there is enough induced friction from preload to transmit the torque

$\mu_{scj} = 0.3$ — Minimum friction factor for slip critical joint (AISC)

$$T_{scj\_clamp} := N_{bolts\_clamp} \cdot \frac{d_{clamp}}{2} \cdot F_{pre\_clamp} \cdot \mu_{scj} = 7.02 \cdot kN \cdot m$$

Torque capacity of the joint, relying on friction

$$\frac{T_{scj\_clamp}}{T_{pk}} = 2.925$$

OK

Margin against exceedign the torque capacity of the joint between motor and ring.

Check capacity of fasteners to withstand the preload (no service loads)

$At_{m5} = 14.2 \cdot mm^2$ — Tensile area of M6 fastener (from above)

$Sp_{129} = 970 \cdot MPa$ — Proof strength of the fastener (from above)

$\sigma_{bolts\_clamp} := \frac{F_{pre\_clamp}}{At_{m5}} = 211.268 \cdot MPa$ — Tensile stress due to preload

$\frac{Sp_{129}}{\sigma_{bolts\_clamp}} = 4.591$ **OK** — Margin against exceeding proof strength at pre-load

Check capacity of the tapped holes in the drive to withstand the preload without stripping

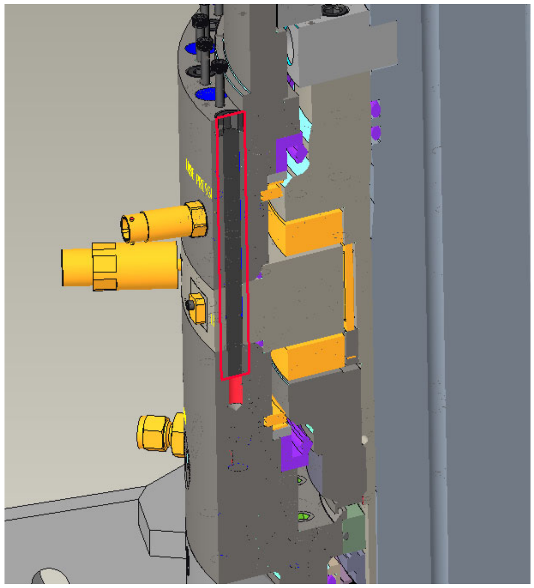
$l_{thrd\_clamp} := 10mm$ — Length of the fastener in the tapped hole

$F_{thrd\_clamp} := \frac{\pi \cdot dp_{m5} \cdot Sy_{1045} \cdot l_{thrd\_clamp}}{\sqrt{3} \cdot 3} = 8.174 \cdot kN$ — Pullout capacity of the threaded hole

$\frac{F_{thrd\_clamp}}{F_{pre\_clamp}} = 2.725$ **OK** — Margin against pulling threads out due to preload

Component - Upper Drive Housing

The upper drive housing reacts the otque of the motor and clamps the bearing race into place.



$N_{bolts\_ubh} := 16$

Number of bolts securing the upper bearing housing.

$d_{ubh} := 415mm$

Diamter of the clamping interface

Select M5x0.8, 30mm long, class 12.9 SHCS

$d_{m10} = 10 \cdot mm$

Nominal diameter of the selected bolt

$F_{pre\_ubh} := 5kN$

Desired preload in each fastener

$k_{bf} = 0.3$

Bolt friction constant (from above), Shigley's

$T_{pre\_ubh} := F_{pre\_ubh} \cdot k_{bf} \cdot d_{m10} = 15 \cdot N \cdot m$

Fastener torque specification (drawing)

Check if there is enough induced friction from preload to transmit the torque

$\mu_{scj} = 0.3$

Minimum friction factor for slip critical joint (AISC)

$$T_{scj\_ubh} := N_{bolts\_ubh} \cdot \frac{d_{ubh}}{2} \cdot F_{pre\_ubh} \cdot \mu_{scj} = 4.98 \cdot kN \cdot m$$

Torque capacity of the joint, relying on friction

$$\frac{T_{scj\_ubh}}{T_{pk}} = 2.075$$

OK

Margin against exceedign the torque capacity of the joint between motor and ring.

Check capacity of fasteners to withstand the preload (no service loads)

$At_{m10} = 58 \cdot mm^2$ Tensile area of M6 fastener (from above)

$Sp_{129} = 970 \cdot MPa$ Proof strength of the fastener (from above)

$\sigma_{bolts\_ubh} := \frac{F_{pre\_ubh}}{At_{m10}} = 86.207 \cdot MPa$ Tensile stress due to preload

$\frac{Sp_{129}}{\sigma_{bolts\_ubh}} = 11.252$ **OK** Margin against exceeding proof strength at pre-load

Check capacity of the tapped holes in the drive to withstand the preload without stripping

$l_{thrd\_ubh} := 16mm$ Length of the fastener in the tapped hole

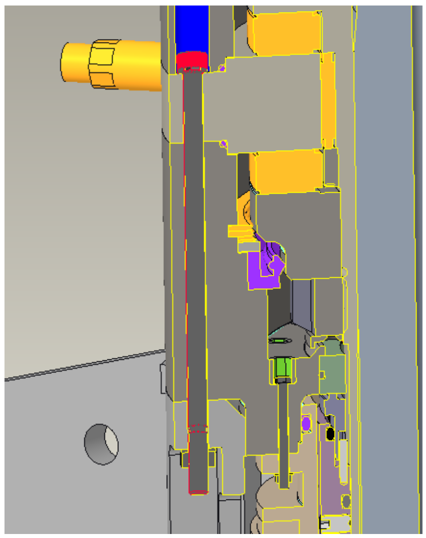
$F_{thrd\_ubh} := \frac{\pi \cdot dp_{m10} \cdot Sy_{1045} \cdot l_{thrd\_ubh}}{\sqrt{3} \cdot 3} = 26.689 \cdot kN$ Pullout capacity of the threaded hole

$\frac{F_{thrd\_ubh}}{F_{pre\_ubh}} = 5.338$ **OK** Margin against pulling threads out due to preload

From this point, the fasteners are large enough and at a large enough diameter that we are assured (without calcualtions) they can transmit the torque without slipping.

Bearing Mounting Bolts

The bearing mounting bolts secure the bearing housing to the base of the drive. The pitlo diameters provide location for radial loads (imbalance and seismic shear), while the bolts resist overturning moments during seismic events and the reaction forces from otor torque. We have shown previously tht the motor torques are well handled by the bolt sizes (previous section), so focus on resisting the overturning moment during a seismic event.



$N_{bolts} := 40$

Number of bolts in the pattern (note that two bolt patterns secure the race)

$d_{bolt\_cir} := 415mm$

Diameter of the bolt circle

$BDC_{brg} := BC\left[d_{bolt\_cir}, N_{bolts}, (0\ \ 1\ \ 0)\right]$

Write matrix for the location of each bolt in the pattern

Use the bolt force calcualtor to determine the bolt forces (3 space)

$BF_{matrix} := BoltForce\left[BDC_{brg}, (0\ \ 0\ \ 0), \left(0\ \ T_{pk}\ \ M_{seis}\right), \left(0\ \ -m_{rot} \cdot g\ \ 0\right)\right]$

$F_{pk} := \max\left(BF_{matrix}^{\langle 2 \rangle}\right) = 23.423 \cdot kN$

Peak tensile force in the bolt

Select: M10x1.5, Gr 12.9 fastener

$At_{m10} = 58 \cdot mm^2$

Tensile (min) area of the bolt minor diameter

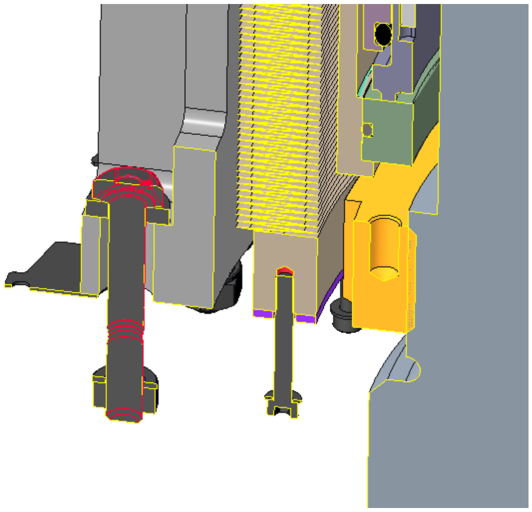
$\sigma b_{seis} := \dfrac{F_{pk}}{At_{m10}} = 403.847 \cdot MPa$

Tensile stress in the fastener

$\dfrac{Sp_{129}}{\sigma b_{seis}} = 2.402$

The nut develops greater capacity than the minor diameter of the bolt thread.

Interface Flange Mounting Bolts



$N_{b\_int} := 20$ — Number of bolts in the pattern

$d_{b\_int} := 415\text{mm}$ — Diameter of the bolt circle

$BDC_{int} := BC\left[d_{b\_int}, N_{b\_int}, (0\ \ 1\ \ 0)\right]$ — Write matrix for the location of each bolt in the pattern

$BDC_{int\_r} := \text{submatrix}\left(BDC_{int}, 2, 19, 1, 3\right)$ — Remove two bolts - first and last to account for the segment removal cutout.

$V_{seis} = 56.57 \cdot \text{kN}$ — Seismic shear demand (computed previously)

Use the bolt force calcualtor to determine the bolt forces (3 space)

$$BF_{int} := \text{BoltForce}\left[BDC_{int\_r}, (0\ \ 0\ \ 0), \left(0\ \ T_{pk}\ \ M_{seis}\right), \left(0\ \ -m_{tot} \cdot g\ \ m_{tot} \cdot A_h\right)\right]$$

$F_{pk\_int} := \max\left(BF_{int}^{\langle 2 \rangle}\right) = 46.729 \cdot \text{kN}$ — Peak external force in the bolt

Select: M10x1.5, Gr 12.9 fastener

$At_{m10} = 58 \cdot \text{mm}^2$ — Tensile (min) area of the bolt minor diameter

$Fi_{int} := 45\text{kN}$ — Selected preload of the fastener.

$\sigma b_{int} := \dfrac{F_{pk}}{At_{m10}} = 403.847 \cdot \text{MPa}$ — Tensile stress in the fastener

$$\frac{Sp_{129}}{\sigma b_{seis}} = 2.402$$

Check slip-critical capacity, AISC Specifications J3.8

The shear capacity is the capacity of all the bolts in the joint (theay all have different external loads)

$n_b := 1$ — Number of fasteners under consideration

$\mu := 0.30$ — Mean slip coefficient, Class A faying surface. AISC Specifications, J3.8

$D_u := 1.13$ — Bolt relaxation coefficient. AISC Specifications, J3.8

$h_f := 1.0$ — Factor for fillers (typically 1)

$n_s := 1$ — Number of slip planes in joint

Note that this requires a custom shim to be isntalled after everything has been aligned. Finger shims are not appropriate for faying surfaces.

$$R_n := \sum_{i=1}^{N_{b\_int}-2} \left[\mu \cdot D_u \cdot h_f \cdot Fi_{int} \cdot n_s \cdot \left[1 - \frac{\left(BF_{int}^{\langle 2 \rangle}\right)_i}{D_u \cdot Fi_{int} \cdot n_b} \right] \right]$$

Available slip capacity. Eq J3.5, AISC specifications.

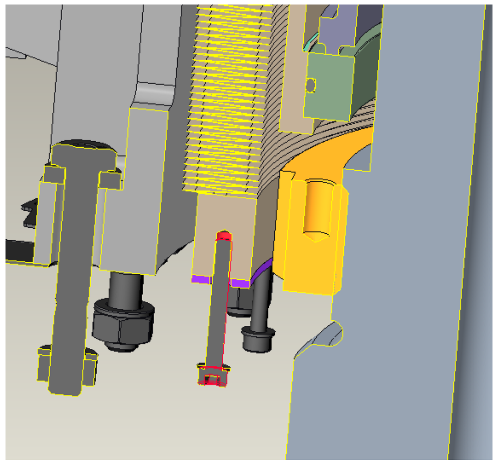
$R_n = 327.311 \cdot kN$

$\boxed{\frac{R_n}{V_{seis}} = 5.786}$ **OK** — Seismic capacity is greater than the demand.

Note that the outriggers will also contribute to the capacity, but are not computed here for conservative simplicity.

Vaccum seal flange bolting

Check to ensure that the bolts securing the bellows are adequately sized to compress the gasket.



Gasket:

$t_{gasket} := 2.5\text{mm}$ — Thickness of the gasket

$A_{gasket} := 17362\text{mm}^2$ — Area of the gasket

$E_{EPDM} := 10\text{MPa}$ — Elastic modulus of EPDM seal (worst case)

$\delta t_{seal} := 0.3 \cdot t_{gasket}$ — Motion required at 20-30% compression recomended for sealing.

$$F_{seal} := \frac{A_{gasket} \cdot E_{EPDM}}{t_{gasket}} \cdot \delta t_{seal} = 5.209 \times 10^4 \text{ N}$$

Total force required to compress the seal

Select: M5 screw, 316L

$N_{fast} := 20$ — Number of fasteners in the bolt pattern

$$F_{bolt} := \frac{F_{seal}}{N_{fast}} = 2.604 \times 10^3 \text{ N}$$

Force required of single bolt

$$\sigma_{bolt} := \frac{F_{bolt}}{At_{m5}} = 183.401 \cdot \text{MPa}$$

Axial stress in the bolt

$$\boxed{\frac{70\text{ksi}}{\sigma_{bolt}} = 2.632}$$

Facto against exceeding the strength of the bolt (DIN 912)

Check against thread pullout in the the bellows

$l_{thrd\_vac} := 11mm$

Thread engagement in the bellows

$$F_{thrd\_vac} := \frac{\pi \cdot dp_{m5} \cdot Sy_{316} \cdot l_{thrd\_vac}}{\sqrt{3} \cdot 3} = 6.999 \cdot kN$$

Pullout capacity of the threaded hole

$$\boxed{\frac{F_{thrd\_vac}}{F_{bolt}} = 2.687}$$

OK

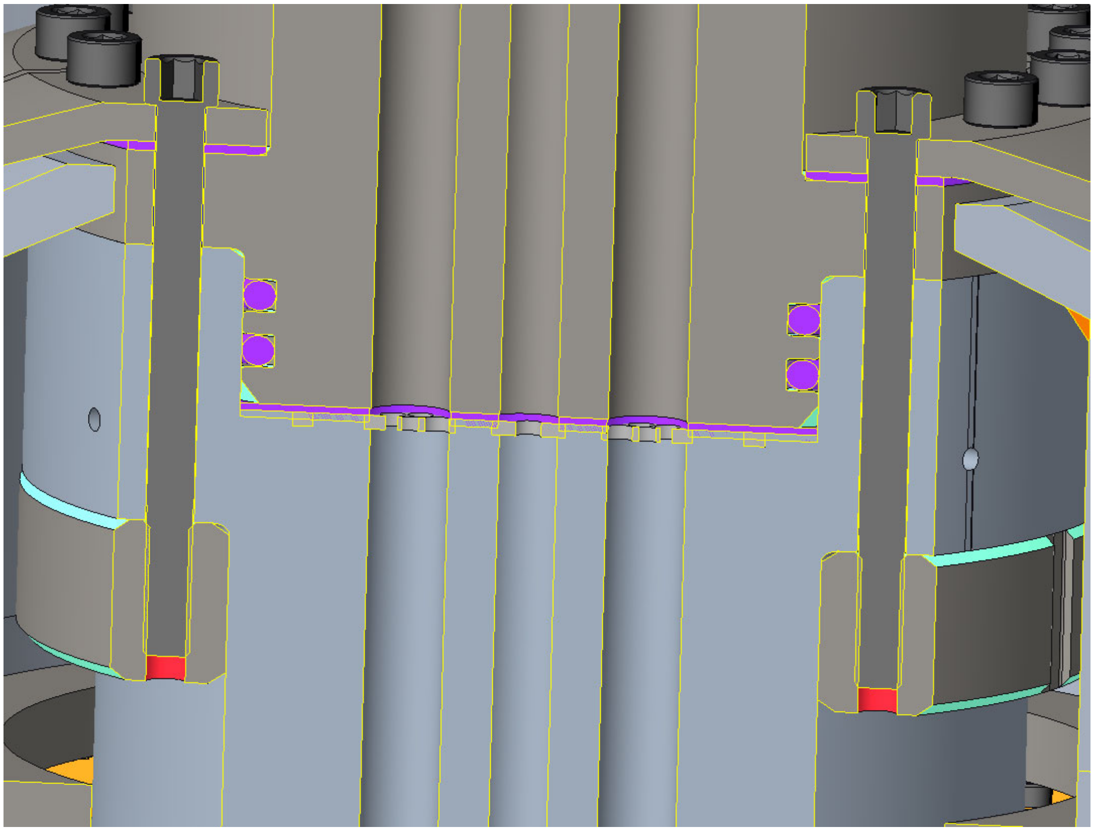
Margin against pulling threads out due to preload

Convert to a torque spec for drawing and field use.

$T_{thrd\_vac} := F_{bolt} \cdot k_{bf} \cdot d_{m5} = 3.906 \cdot N \cdot m$

Minimum torque.

COMPONENT - Crown Water gasket



The water gasket between the crown and the valve assembly has 21 circuits, consiting of 42 individual passages. We need to ensure that the gasket is kept under consistent pressure and external forces from the piping do not disrupt the seal. There are two load paths:
1) Gakset is compressed by the bolts until the flange contacts the crown.
2) The flange contacts the crown and the bolts are further tightened, clamping the faces
3) External loads seeks to unlaod the joint, without any gasket speeration.

Forces due to compressing the gasket and water pressure

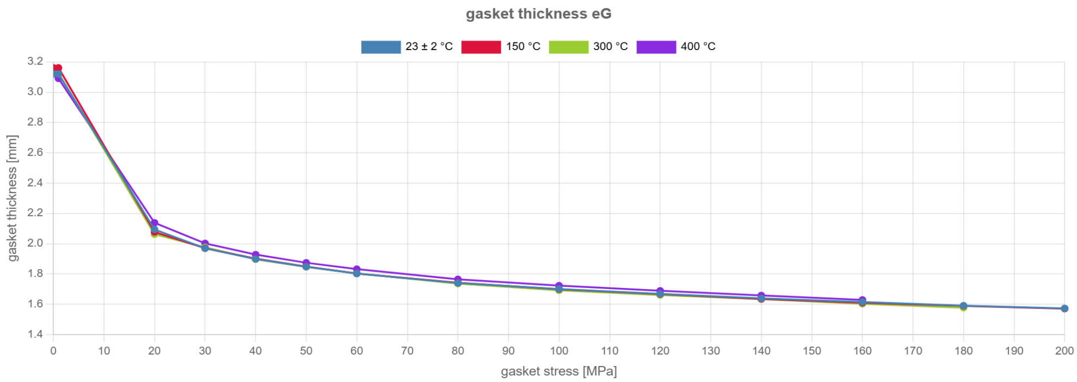
$do_{gasket} := 198mm$ Outer diameter of the gasket

$di_{gasket} := 18mm$ Inner diameter of the gasket (per hole)

$A_{va\_gasket} := \frac{\pi}{4} \cdot \left(do_{gasket}^{2} - 42di_{gasket}^{2}\right) = 2.01 \times 10^{4} \cdot mm^{2}$ Face area of gasket material

Determine the overall force required to compress the gasket to 80MPa (20 MPa min for sealing)

Select: Sigrafelx HOCHDRUCK V30011Z31 (3mm thick)



$\sigma i_{gasket} := 30MPa$ — Gasket stress, selected sealing force

$\boxed{tf_{gasket} := 2mm}$ — Total thickness of the gasket at the selected thikness. Unloaded =3mm

$Fi_{gasket} := \sigma i_{gasket} \cdot A_{va\_gasket} = 603.092 \cdot kN$ — Load required to compress the gasket

Determine the pressure force trying to unseat the valve body

$A_{press\_gasket} := 42 \cdot \frac{\pi}{4} \cdot \left(di_{gasket}^{\ 2}\right)$ — Pressure facing area of the gasket

$F_{press} := A_{press\_gasket} \cdot P_{op} = 1.242 \cdot kip$ — Force of water pressure

External interface loads from piping (from S01020500-IST10211)

Table 2. Primary Cooling Water Mechanical Connection

Description	Value
Supply connection	4" 150# rasied face flange
Supply connection height[d]	7.483m
Supply connection angle[c]	90°
Supply Screen	$< 8mm$
Return Connection	4" 150# rasied face flange
Return connection height[d]	7.775m
Return connection angle[c]	90°
Radial pipe loads[a b]	$\leq 6.7kN$
Axial pipe loads [a b]	$\leq 10.2kN$
Moment imposed perpendicular to pipe axis [a b]	$\leq 0.42kNm$
Torque imposed along piping axis [a b]	$\leq 1.7kNm$
Radial motion of the pipe flange due to target rotation[e f]	$\leq 1mm$
Axial motion of the pipe flange due to target rotation[e f]	$\leq 0.5mm$

$Fr_{ru} := 6.7\text{kN}$

Loads aligned to axis of rotation, imposed on the rotary union.

$Fa_{ru} := 10.2\text{kN}$

Loads perpendicular to axis of rotation, imposed on the rotary union.

$Ta_{ru} := 0.42\text{kN}\cdot\text{m}$

Torque along axis of rotation (or horisontal axis)

$Tr_{ru} := 1.7\text{kN}\cdot\text{m}$

Torque along axis pf water piping

$F_{ru} := \begin{pmatrix} Fa_{ru} & Fr_{ru} & Fa_{ru} \end{pmatrix}$

Collect the forces into a 3 psace vector

$M_{ru} := \begin{pmatrix} Tr_{ru} & Ta_{ru} & Ta_{ru} \end{pmatrix}$

Collect the moments into a 3 space vector

$r_{gu} := (0 \quad 1530 \quad 0)\text{mm}$

Vector between gasket flange and return pipe axis.

Compile all of the external forces and moments acting on the valve body

$$F_{app} := F_{ru} + (0 \quad 1 \quad 0)\cdot\left(F_{press} + Fi_{gasket}\right) = \begin{pmatrix} 1.02\times10^{4} & 6.153\times10^{5} & 1.02\times10^{4} \end{pmatrix}\text{N}$$

$$M_{app} := M_{ru} + \left({r_{gu}}^{T} \times {F_{ru}}^{T}\right)^{T} = (17.306 \quad 0.42 \quad -15.186)\cdot\text{kN}\cdot\text{m}$$

Moment at the bolted interface.

Select: M10 SHCS to secure the valve assembly to the crown.

$N_{fas\_gas} := 30$ — Number of fasteners in the group

$Fi_{gas} := 45kN$ — Reccomended preload in each fastener.

$T_{pre\_gas} := Fi_{gas} \cdot k_{bf} \cdot d_{m10} = 99.571 \cdot lbf \cdot ft$ — Torque to apply preload

$$\frac{Fi_{gas} \cdot N_{fas\_gas}}{Fi_{gasket}} = 2.238$$

Early check (without external load); determine if prelaod can compress the gasket.

Determine the ability of the bolted joint to react the external loads without lifting (lifting relieves the gasket).

$d_{bc\_gas} := 217.5mm$ — Diameter of the bolt circle.

$BC_{gas} := BC\left[d_{bc\_gas}, N_{fas\_gas}, \begin{pmatrix} 0 & 1 & 0 \end{pmatrix}\right]$ — Generate a matrix of bolt locations

$BF_{gas} := BoltForce\left[BC_{gas}, \begin{pmatrix} 0 & 0 & 0 \end{pmatrix}, M_{app}, F_{app}\right]$ — Compute the individual bolt forces

$F_{pk\_gas} := \max\left(BF_{gas}^{\langle 2 \rangle}\right) = 34.565 \cdot kN$ — Peak external applied load, single fastener.

Determine the joint constant to understand the motion and minimum gasket stress

Determine the stiffness of the bolt.

$l_{tot\_gas} := 90mm$ — Length of the joint

$l_{thrd\_gas} := 10mm$ — Length of the threaded portion of the joint

$l_{uthrd\_gas} := 80mm$ — Unthreaded length of the join

$$k_{b\_gas} := \frac{A_{m10} \cdot At_{m10} \cdot E_{carbon\_steel}}{A_{m10} \cdot l_{thrd\_gas} + At_{m10} \cdot l_{uthrd\_gas}} = 167.925 \cdot \frac{kN}{mm}$$

Spring stiffness of the bolt

Determine the stiffness of the clamped members

Shigley's reccomends developing a psring based upon the clamped members developing a frustum cone of approximately 30 degrees. The compact nature of this joint does not lend well to a fully developed frustum cone, and the accuracy of this approach is not needed; the gasket will dominate the stiffness. Treat the bolted joint areas as sectors of the circle.

Upper Ply spring rate (Valve body)

$do_{gas\_fas} := 234mm$ — Outer diameter of the flange

$di_{gas\_fas} := 198mm$ — Inned diameter of the flange

$A_{bj\_gas} := \frac{\pi}{N_{fas\_gas} \cdot 4} \cdot \left(do_{gas\_fas}^{\ 2} - di_{gas\_fas}^{\ 2}\right)$ — Area of the sector arc

$l_{1\_gas} := 28mm$ — Thickness of the upper ply

$km_1 := \frac{A_{bj\_gas} \cdot E_{316}}{l_{1\_gas}} = 2.907 \times 10^3 \cdot \frac{kN}{mm}$ — Spring rate of the upper ply

Lower member stiffness

$l_{2\_gas} := 60mm$ — Length of the lower clamped ply

$km_2 := \frac{A_{bj\_gas} \cdot E_{316}}{l_{2\_gas}} = 1.357 \times 10^3 \cdot \frac{kN}{mm}$ — Spring rate of the lower ply

Total clamped member stiffness

$k_{m\_gas} := \frac{1}{\frac{1}{km_1} + \frac{1}{km_2}} = 925.101 \cdot \frac{kN}{mm}$ — Spring rate of the clamped members

$C_{gas} := \frac{k_{b\_gas}}{k_{b\_gas} + k_{m\_gas}} = 0.154$ — Joint constant.

$F_{bolt\_gas} := C_{gas} \cdot F_{pk\_gas} + Fi_{gas} = 50.31 \cdot kN$ — Peak bolt force, preload and applied loads

$\sigma_{bolt\_gas} := \frac{F_{bolt\_gas}}{At_{m10}} = 867.421 \cdot MPa$ — Peak bolt stress

$\boxed{\frac{Sp_{129}}{\sigma_{bolt\_gas}} = 1.118}$ **OK** — Margin against exceeding proof load

$F_{sep} := \frac{Fi_{gas}}{1 - C_{gas}} = 53.168 \cdot kN$ — External load that would cause joint seperation.

$\boxed{\frac{F_{sep}}{F_{pk}} = 2.27}$ **OK** — Margin against joint seperation

Check against exceeding the fatigue capacity of the fasteners of the rotating joint

$$F_{cyc} := \max\left(\mathrm{BoltForce}\left(BC_{gas}, r_{gu}, M_{ru}, F_{ru}\right)^{\langle 2 \rangle}\right) = 14.278 \cdot kN$$

Amplitude (fully reversing) of the bolt force, piping loads only, due ot rotation

$$\sigma_a := \frac{C_{gas} \cdot F_{cyc}}{At_{m10}} = 37.821 \cdot MPa$$

Alternating stress in the bolt

$$\sigma_i := \frac{Fi_{gas}}{At_{m10}} + C_{gas} \cdot \frac{\left(F_{press} + Fi_{gasket}\right)}{N_{fas\_gas} \cdot At_{m10}} = 829.6 \cdot MPa$$

Mean stress due to pressure and gasket forces.

$$Se_{129} := 190MPa$$

Fully corrected endurance strength (Shigley's table 8-17)

$$Sa := \frac{Se_{129} \cdot \left(Su_{129} - \sigma_i\right)}{Su_{129} + Se_{129}} = 52.607 \cdot MPa$$

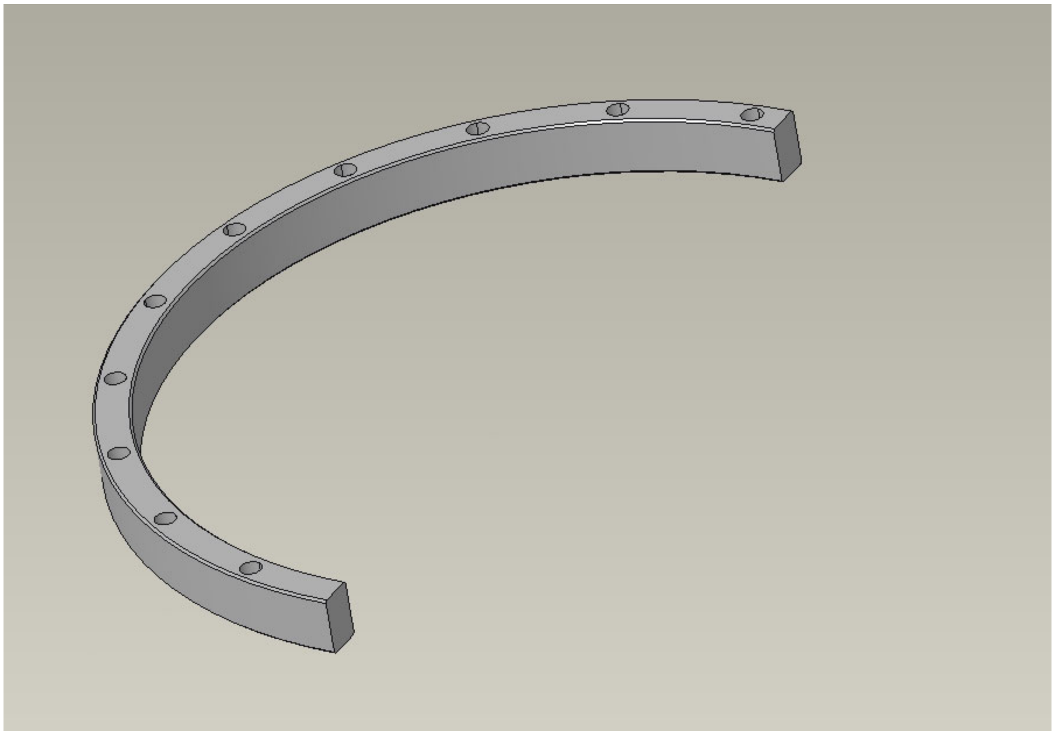
Fatigue limit at indicated intersection of load line and Goodman criterion (straight line) SHighley's Eq. 8-40

$$\boxed{n_f := \frac{Sa}{\sigma_a} = 1.391}$$

OK

Criteria against exceeding the fatigue strength of the selected fasteners.

COMPONENT - Gasket nut plate



$l_{np} := 30mm$ Thread engagement into the nut plate

$F_{u\_np} := Sy_{4340}$ Tensile strength of the connected material

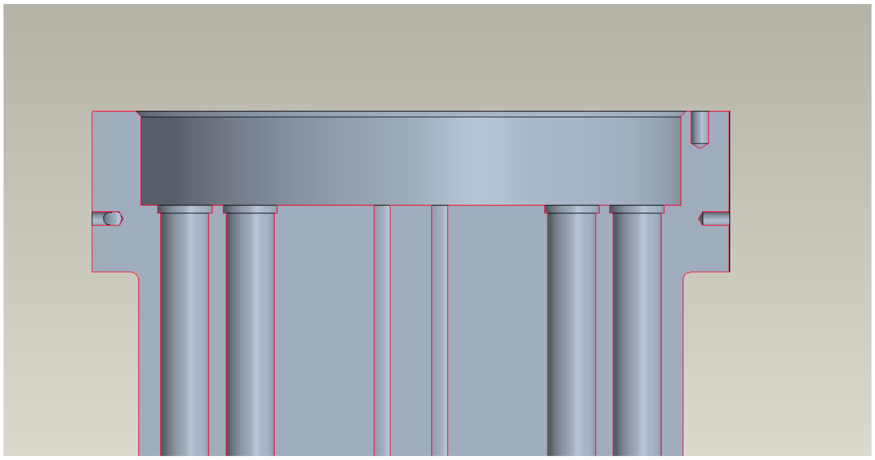
$$R_{po\_np} := \frac{\pi \cdot dp_{m10} \cdot F_{u\_np} \cdot l_{np}}{\sqrt{3} \cdot 3} = 42.46 \cdot kip$$

Pullout capacity of the threaded hole

$\boxed{\frac{R_{po\_np}}{F_{bolt\_gas}} = 3.754}$ **OK** Capacity margin, thread pullout

Check the capacity of the crown to be lifted directly during bearing removal

Assme that the lifting fixture clamps directly to the "neck" of the crown to lift and set it down on the maintenance supprt sleeve. Check the shear and bearing capacity.



Check shear

$d_{neck} := 200mm$ — Diameter of the "neck" of the crown.

$t_{neck} := 25mm$ — Thickness of the "neck" of the crown.

$$\tau_{lift} := \frac{m_{tot} \cdot g}{\pi \cdot d_{neck} \cdot t_{neck}} = 11.188 \cdot MPa$$

Shear stress in the "neck" of the crown.

$$\frac{S_{625}}{\sqrt{3} \cdot \tau_{lift}} = 10.674$$

OK

Margin against exceeding the capacity of the crown during lifting.

Check Bearing

$Abrg_{neck} := 7400mm^2$ — Bearing area of the underside of the crown.

$$\sigma brg_{neck} := \frac{m_{tot} \cdot g}{Abrg_{neck}} = 23.748 \cdot MPa$$

Bearing stress during lifting.

$$\boxed{\frac{S_{625}}{\sigma brg_{neck}} = 8.71}$$

OK

Margin against exceeding the bearing capacity of the crown face.

Check twisiting of flange.

Determine the stresses in accordance with Timoshenko, "Strength of Materials" Part II, Section 28. Treat the falnge as a pipe with flange similar to Example II. In reality, the pipe/flange will be much stiffer.

$d_{neck} = 200 \cdot mm$ Diameter of the neck of the crown (above)

$do_{neck} := 234mm$ Diameter of the flange

$c_{neck} := 100mm$ Inner diameter of the pipe (actually zero)

$h_1 := \frac{d_{neck} - c_{neck}}{2}$ Thickness of the "pipe" wall

$h := 25mm$ Height of the flange

$R_{dist} := \frac{m_{tot} \cdot g}{\pi \cdot d_{neck}} = 0.28 \cdot \frac{kN}{mm}$ Load distributed at the base of the neck.

$$\beta := \left[3 \cdot \frac{\left(1 - \nu_{625}{}^2\right)}{d_{neck}{}^2 \cdot h_1{}^2}\right]^{\frac{1}{4}} = 12.832 \frac{1}{m}$$

$$M_o := \frac{R_{dist} \cdot \left(do_{neck} - c_{neck}\right)}{1 + \frac{\beta \cdot h}{2} + \frac{1 - \nu_{625}{}^2}{2 \cdot \beta \cdot c_{neck}} \cdot \left(\frac{h}{h_1}\right)^3 \cdot \ln\left(\frac{do_{neck}}{c_{neck}}\right)} = 31.289 \cdot \frac{N \cdot m}{mm}$$

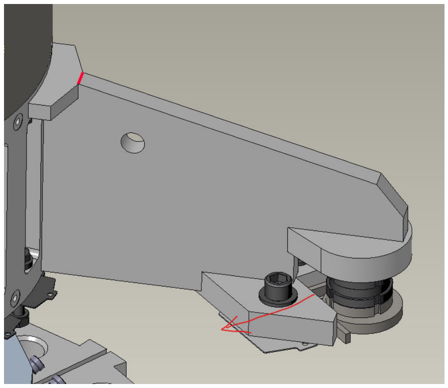
Twisting moment at the joint of pipe-flange

$\sigma b_{neck} := \frac{6 \cdot M_o}{h_1{}^2} = 75.093 \cdot MPa$ Bending stress die to overturning moment (circular twist)

$\frac{S_{625}}{\sigma b_{neck}} = 2.754$ **OK** Margin against exceeding the allowable on the flange.

Component - Support base legs

Check the capcity of the support base legs to be used for support during installation.



Note: the support leg will bear the full weight of the target assembly, but is not intended to push the full weight of the target assembly when it is resting on the vertical support ring. The crane will be used to relieve approximately half of the weight during horizonal pushing.

$w_{crane} := 0.5$ — Percent of the weight to be supported by the crane during installation & manipulation.

$t_{leg} := 19mm$ — Thickness of the support leg

$h_{leg} := 200mm$ — Height of the support leg.

$l_{leg} := 300mm$ — Moment arm for the support.

$w_{fl1} := 150mm$ — Width of the top flange

$w_{fl2} := 150mm$ — Width of the bottom flange

$t_{fl1} := 19.0mm$ — Thickness of the upper flange

$t_{fl2} := 19.0mm$ — Thickness of the lower flange

$F_{lift} := m_{tot} \cdot \frac{g}{2} = 87.868 \cdot kN$ — Maximum lifting force seen by leg

Check the capacity of the fillet weld leg (Consider only the flanges)

$t_{weld\_leg} := 10mm$ Thickness of the weld callout (dwg)

$A_{weld\_leg} := \frac{1}{\sqrt{2}} \cdot t_{weld\_leg} = 7.071 \cdot mm$ Throat area (per unit length) of the weld

$A_{weld\_patt} := 2 \cdot h_{leg} + 2 \cdot w_{fl1} + 2 \cdot w_{fl2}$ Area of the weld pattern

$I_{weld\_leg\_yy} := 2 \frac{w_{fl1} \cdot h_{leg}^{2}}{2} = 6 \times 10^{6} \cdot mm^{3}$ Second moment of welds (as lines)

$F_{b2} := \frac{F_{lift} \cdot l_{leg} \cdot h_{leg}}{2 \cdot I_{weld\_leg\_yy}} = 0.439 \cdot \frac{kN}{mm}$ Force per unit area (acts radially outward)

$F_{s2} := \frac{F_{lift}}{A_{weld\_patt}} = 0.088 \cdot \frac{kN}{mm}$ Force per unit area (direct shear) due to lifting

$$F_{eq} := \frac{1}{\sqrt{2}} \cdot \left[2 \cdot \left(F_{b2}\right)^{2} + 6 \cdot \left(F_{s2}^{2}\right)\right]^{\frac{1}{2}} = 0.465 \cdot \frac{kN}{mm}$$

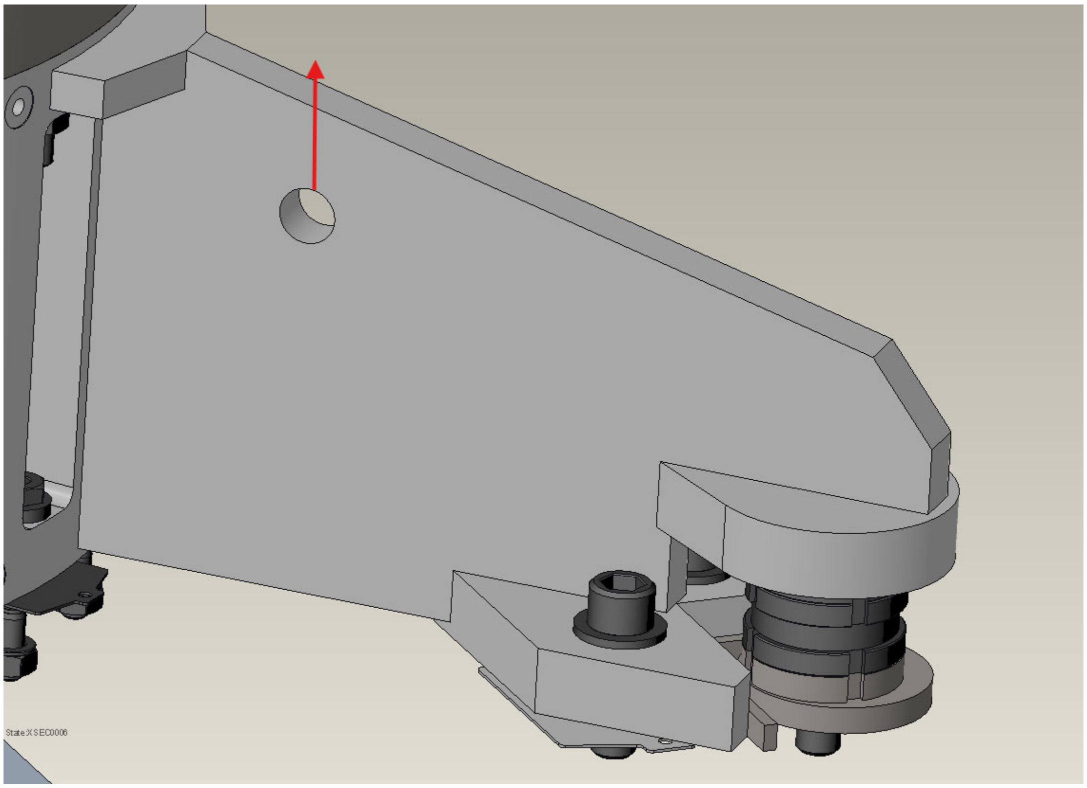
$\sigma_{weld} := \frac{F_{eq}}{A_{weld\_leg}} = 9.537 \cdot ksi$ Stress in weld leg

$\boxed{\frac{Sy_{A572\_50}}{\sigma_{weld}} = 5.243}$ **OK**

The welds are sufficiently sized (PJP), so there is no need to check the base material.

Check bearing stress at lifting holes

Ensure that the lifting holes are adequately sized to support the bearing forces. Use RCSC Section J3.10 for appropriate limit state.



$d_h := 19.75mm$ Diameter of the lfiting lug hole.

$l_{min} := 40mm$ Diatance from the hole to edge

$t_{leg} = 19 \cdot mm$ Thickness of the support leg

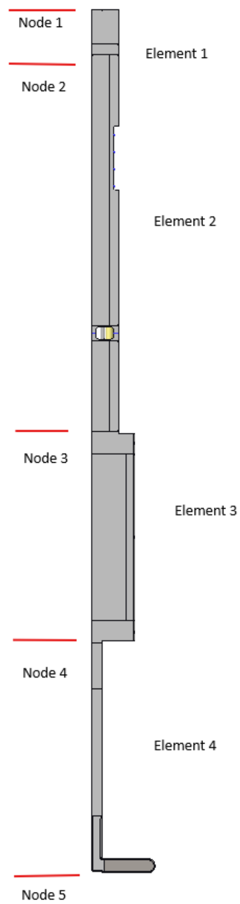
$\Omega := 2.0$ Bearing strength capacity/demand ratio

$R_n := 3.0 \cdot d_h \cdot t_{leg} \cdot Su_{A572\_50} = 5.045 \times 10^5 N$

$\frac{4R_n}{\Omega \cdot (m_{tot} \cdot g)} = 5.742$ **OK**

Segment Attachment bolt - loads

The segment is a statically indeterminate system with repsect to centripetal forces. To determine the reactions at the bolt, we will use the FE method (Stiffness matrix) in a similar fashion as we did with the shaft earlier in this package.



$DOF := 2$ Degrees of freedom per node

$N_{nodes} := 6$ Number of nodes

$i := 1 .. 2 \cdot DOF$ $j := 1 .. 2 \cdot DOF$ Row and column counters for local stiffness

$j := 1 .. DOF$ Row counters for local forces and displacements

$k := 1 .. N_{nodes} \cdot DOF$ Global counter for DOF

$n := 1 .. N_{nodes}$ Global node counter

$Kg_{seg} := 0 \cdot \text{identity}\left(N_{nodes} \cdot DOF\right)$ Initialize the global stiffness matrix for 5 nodes

$Mg_{seg_n} := 0$ Initialize the mass matrix

$Fg_{seg\_cent_k} := 0$ Initialize the load matrix

Element #1 - Upper segment

$el := 1$ Element number (for matrix and array indices)

$N_{conn} := (1 \quad 2)^T$ Nodal connectivity

$DOF_{conn_{el}} := (1 \quad 2 \quad 3 \quad 4)$ Local to global connectivity

$A_{seg_{el}} := 0.0127$ Area of the cross section

$I_{seg_{el}} := 2.8 \cdot 10^{-5}$ Area moment of inertia, segment (m^4)

$l_{seg_{el}} := 0.228$ Length of the upper segment to the bolt.

$X_{seg_{el}} := .0894 + 0.210$ CG Offset from the axis of rotation

$M_{seg_{el}} := A_{seg_{el}} \cdot l_{seg_{el}} \cdot \rho_{316} \cdot \frac{m^3}{kg}$ Mass of the element (total)

$k_{seg_{el}} := k_{loc}\left(\frac{E_{316}}{Pa}, I_{seg_{el}}, l_{seg_{el}}\right)$ Flexural stiffness matrix for element #1 (v1, θ1,v2,θ2)

$F_{cent} := M_{seg_{el}} \cdot X_{seg_{el}} \cdot \omega_{rot}^2 \cdot s^2 = 154.592$ Centripetal force in element (N)

$Kg_{seg} := \text{Assemble}\left(Kg_{seg}, k_{seg_{el}}, DOF_{conn_{el}}\right)$ Add local stiffnes to global stiffness matrix

$Fg_{seg\_cent} := \text{CVassm}\left[Fg_{seg\_cent}, \frac{F_{cent}}{2}\begin{pmatrix}1\\0\\1\\0\end{pmatrix}, DOF_{conn_{el}}^{\ T}\right]$ Add local forces (body) to global force vector

$Mg_{seg} := \text{CVassm}\left[Mg_{seg}, \frac{M_{seg_{el}}}{2} \cdot \begin{pmatrix}1\\1\end{pmatrix}, N_{conn}\right]$ Add local masses to global node mass vector

Element #2 - below bolt

$el := 2$ — Element number (for matrix and array indices)

$N_{conn} := (2 \;\; 3)^T$ — Nodal connectivity

$DOF_{conn_{el}} := (3 \;\; 4 \;\; 5 \;\; 6)$ — Local to global connectivity

$A_{seg_{el}} := 0.0127$ — Area of the cross section

$I_{seg_{el}} := 2.8 \cdot 10^{-5}$ — Area moment of inertia, segment (m^4)

$l_{seg_{el}} := 2.240$ — Length of the upper segment to the bolt.

$X_{seg_{el}} := .0894 + 0.210$ — CG Offset from the axis of rotation

$M_{seg_{el}} := A_{seg_{el}} \cdot \rho_{316} \cdot l_{seg_{el}} \cdot \frac{m^3}{kg}$ — Mass of the element (total)

$k_{seg_{el}} := k_{loc}\left(\frac{E_{316}}{Pa}, I_{seg_{el}}, l_{seg_{el}}\right)$ — Flexural stiffness matrix for element #1 (v1, θ1,v2,θ2)

$F_{cent} := M_{seg_{el}} \cdot X_{seg_{el}} \cdot \omega_{rot}{}^2 \cdot s^2 = 1.519 \times 10^3$ — Centripetal force in element (N)

$Kg_{seg} := \text{Assemble}\left(Kg_{seg}, k_{seg_{el}}, DOF_{conn_{el}}\right)$ — Add local stiffnes to global stiffness matrix

$$Fg_{seg\_cent} := \text{CVassm}\left[Fg_{seg\_cent}, \frac{F_{cent}}{2}\begin{pmatrix}1\\0\\1\\0\end{pmatrix}, DOF_{conn_{el}}{}^T\right]$$

Add local forces (body) to global force vector

$$Mg_{seg} := \text{CVassm}\left[Mg_{seg}, \frac{M_{seg_{el}}}{2} \cdot \begin{pmatrix}1\\1\end{pmatrix}, N_{conn}\right]$$

Add local masses to global mass vector

Element #3 - Donut

$el := 3$ Element number (for matrix and array indices)

$N_{conn} := (3 \quad 4)^T$ Nodal connectivity

$DOF_{conn_{el}} := (5 \quad 6 \quad 7 \quad 8)$ Local to global connectivity

$A_{seg_{el}} := 0.025$ Area of the cross section (m^2)

$I_{seg_{el}} := 1.25 \cdot 10^{-4}$ Area moment of inertia, segment (m^4)

$l_{seg_{el}} := 1.2$ Length of the donut. (m)

$X_{seg_{el}} := 0.144 + 0.210$ CG Offset from the axis of rotation (m)

$M_{seg_{el}} := A_{seg_{el}} \cdot \rho_{316} \cdot l_{seg_{el}} \cdot \frac{m^3}{kg}$ Mass of the element (total kg)

$k_{seg_{el}} := k_{loc}\left(\frac{E_{316}}{Pa}, I_{seg_{el}}, l_{seg_{el}}\right)$ Flexural stiffness matrix for element #1 (v1, θ1,v2,θ2)

$F_{cent} := M_{seg_{el}} \cdot X_{seg_{el}} \cdot \omega_{rot}^2 \cdot s^2 = 1.894 \times 10^3$ Centripetal force in element (N)

$Kg_{seg} := \text{Assemble}\left(Kg_{seg}, k_{seg_{el}}, DOF_{conn_{el}}\right)$ Add local stiffnes to global stiffness matrix

$$Fg_{seg\_cent} := \text{CVassm}\left[Fg_{seg\_cent}, \frac{F_{cent}}{2} \begin{pmatrix} 1 \\ 0 \\ 1 \\ 0 \end{pmatrix}, DOF_{conn_{el}}^T\right]$$

Add local forces (body) to global force vector

$$Mg_{seg} := \text{CVassm}\left[Mg_{seg}, \frac{M_{seg_{el}}}{2} \cdot \begin{pmatrix} 1 \\ 1 \end{pmatrix}, N_{conn}\right]$$

Add local masses to global mass vector

Element #4 -Below Donut

$el := 4$ Element number (for matrix and array indices)

$N_{conn} := (4 \quad 5)^T$ Nodal connectivity

$DOF_{conn_{el}} := (7 \quad 8 \quad 9 \quad 10)$ Local to global connectivity

$A_{seg_{el}} := 3.44 \cdot 10^{-3}$ Area of the cross section (m^2)

$I_{seg_{el}} := 1.94 \cdot 10^{-6}$ Area moment of inertia, segment (m^4)

$l_{seg_{el}} := 1.31$ Length of the donut. (m)

$X_{seg_{el}} := 0.034 + 0.210$ CG Offset from the axis of rotation (m)

$M_{seg_{el}} := A_{seg_{el}} \cdot \rho_{316} \cdot l_{seg_{el}} \cdot \frac{m^3}{kg}$ Mass of the element (total kg)

$k_{seg_{el}} := k_{loc}\left(\frac{E_{316}}{Pa}, I_{seg_{el}}, l_{seg_{el}}\right)$ Flexural stiffness matrix for element #1 (v1, θ1,v2,θ2)

$F_{cent} := M_{seg_{el}} \cdot X_{seg_{el}} \cdot \omega_{rot}^2 \cdot s^2 = 196.073$ Centripetal force in element (N)

$Kg_{seg} := \text{Assemble}\left(Kg_{seg}, k_{seg_{el}}, DOF_{conn_{el}}\right)$ Add local stiffnes to global stiffness matrix

$Fg_{seg\_cent} := \text{CVassm}\left[Fg_{seg\_cent}, \frac{F_{cent}}{2}\begin{pmatrix}1\\0\\1\\0\end{pmatrix}, DOF_{conn_{el}}^{\;T}\right]$ Add local forces (body) to global force vector

$Mg_{seg} := \text{CVassm}\left[Mg_{seg}, \frac{M_{seg_{el}}}{2}\cdot\begin{pmatrix}1\\1\end{pmatrix}, N_{conn}\right]$ Add local masses to global mass vector

Element #5 -segment pin

$el := 5$ Element number (for matrix and array indices)

$N_{conn} := (5 \quad 6)^T$ Nodal connectivity

$DOF_{conn_{el}} := (9 \quad 10 \quad 11 \quad 12)$ Local to global connectivity

$A_{seg_{el}} := 1.38 \cdot 10^{-3}$ Area of the cross section (m^2)

$I_{seg_{el}} := 7.9 \cdot 10^{-8}$ Area moment of inertia, segment (m^4)

$l_{seg_{el}} := 0.063$ Length of the element. (m)

$X_{seg_{el}} := 0.0325 + 0.210$ CG Offset from the axis of rotation (m)

$M_{seg_{el}} := A_{seg_{el}} \cdot \rho_{316} \cdot l_{seg_{el}} \cdot \frac{m^3}{kg}$ Mass of the element (total kg)

$k_{seg_{el}} := k_{loc}\left(\frac{E_{316}}{Pa}, I_{seg_{el}}, l_{seg_{el}}\right)$ Flexural stiffness matrix for element #1 (v1, θ1,v2,θ2)

$F_{cent} := M_{seg_{el}} \cdot X_{seg_{el}} \cdot \omega_{rot}^2 \cdot s^2 = 3.759$ Centripetal force in element (N)

$Kg_{seg} := Assemble\left(Kg_{seg}, k_{seg_{el}}, DOF_{conn_{el}}\right)$ Add local stiffnes to global stiffness matrix

$$Fg_{seg\_cent} := CVassm\left[Fg_{seg\_cent}, \frac{F_{cent}}{2}\begin{pmatrix}1\\0\\1\\0\end{pmatrix}, DOF_{conn_{el}}^{\;T}\right]$$

Add local forces (body) to global force vector

$$Mg_{seg} := CVassm\left[Mg_{seg}, \frac{M_{seg_{el}}}{2}\cdot\begin{pmatrix}1\\1\end{pmatrix}, N_{conn}\right]$$

Add local masses to global mass vector

Node 5 - Add Tungsten centripetal force and moment

$m_{tgt} := 45kg$ Approximate mass of the target foot.

$r_{tgt} := 483mm$ Approximate location of the target foot CG

$l_{tgt\_cg} := 175mm$ Approximate location of the target foot CG relative to pin axis.

$F_{cent} := m_{tgt} \cdot r_{tgt} \cdot \omega_{rot}^{2} \cdot \frac{1}{N} = 482.661$ Centripetal force of the target

$M_{dead} := m_{tgt} \cdot g \cdot l_{tgt\_cg} \cdot \frac{1}{N \cdot m}$ Moment of the target foot due to cantilevered mass.

$Fg_{seg\_cent_9} := Fg_{seg\_cent_9} + F_{cent}$ Add the centrifugal loads

$Fg_{seg\_cent_{10}} := Fg_{seg\_cent_{10}} + M_{dead}$ Add the moment due to the cantilevered target.

$Mg_{seg_5} := Mg_{seg_5} + \frac{m_{tgt}}{kg}$ Add the mass of the segment to the mass list

Review the global stiffness, force and mass matrices and vectors

$Kg_{seg} =$

$5.67 \cdot 10^{9}$	$-6.46 \cdot 10^{8}$	$-5.67 \cdot 10^{9}$	$-6.46 \cdot 10^{8}$	0	0
$-6.46 \cdot 10^{8}$	$9.82 \cdot 10^{7}$	$6.46 \cdot 10^{8}$	$4.91 \cdot 10^{7}$	0	0
$-5.67 \cdot 10^{9}$	$6.46 \cdot 10^{8}$	$5.67 \cdot 10^{9}$	$6.39 \cdot 10^{8}$	$-5.98 \cdot 10^{6}$	$-6.69 \cdot 10^{6}$
$-6.46 \cdot 10^{8}$	$4.91 \cdot 10^{7}$	$6.39 \cdot 10^{8}$	$1.08 \cdot 10^{8}$	$6.69 \cdot 10^{6}$	$5 \cdot 10^{6}$
0	0	$-5.98 \cdot 10^{6}$	$6.69 \cdot 10^{6}$	$1.8 \cdot 10^{8}$	$-9.74 \cdot 10^{7}$
0	0	$-6.69 \cdot 10^{6}$	$5 \cdot 10^{6}$	$-9.74 \cdot 10^{7}$	$9.33 \cdot 10^{7}$
0	0	0	0	$-1.74 \cdot 10^{8}$	$1.04 \cdot 10^{8}$
0	0	0	0	$-1.04 \cdot 10^{8}$	$4.17 \cdot 10^{7}$
0	0	0	0	0	0
0	0	0	0	0	0
0	0	0	0	0	0
0	0	0	0	0	...

$Fg_{seg\_cent} =$

	1
1	77.296
2	0
3	836.698
4	0
5	$1.706 \cdot 10^{3}$
6	0
7	$1.045 \cdot 10^{3}$
8	0
9	582.577
10	77.227
11	1.88
12	0

Centripetal force vector (with dead loads)

$$Mg_{seg} = \begin{pmatrix} 11.626 \\ 125.845 \\ 234.669 \\ 138.543 \\ 63.442 \\ 0.349 \end{pmatrix}$$

Mass vector

Case #1 Operational loads

Write the boundary conditions, where DOF 1,3, and 11 are fixed.

$BC_{stat} := (1\ \ 0\ \ 1\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 1\ \ 0)^T$ Boundary condition status indicator

$BC_{app} := (0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0\ \ 0)^T$ Boundary condition values (must correspond to status flag)

Use the FEsolve program (appendix) to solve for displacements.

$\delta_{seg} := FE_{solve}\left(Kg_{seg}, BC_{stat}, BC_{app}, Fg_{seg\_cent}\right)$ Displacements of DOF

$R_{seg} := Kg_{seg} \cdot \delta_{seg} - Fg_{seg\_cent}$ Determine the reaction forces in the segment

$$\delta_{seg} = \begin{bmatrix} 0 \\ 2.241 \cdot 10^{-5} \\ 0 \\ -4.482 \cdot 10^{-5} \\ 9.375 \cdot 10^{-4} \\ -5.055 \cdot 10^{-4} \\ 1.513 \cdot 10^{-3} \\ -4.513 \cdot 10^{-4} \\ 1.303 \cdot 10^{-4} \\ 1.949 \cdot 10^{-3} \\ 0 \\ 2.127 \cdot 10^{-3} \end{bmatrix} \qquad R_{seg} = \begin{bmatrix} 1.44 \cdot 10^{4} \\ 0 \\ -1.724 \cdot 10^{4} \\ 0 \\ -4.547 \cdot 10^{-12} \\ -7.276 \cdot 10^{-12} \\ -3.865 \cdot 10^{-11} \\ -2.137 \cdot 10^{-11} \\ 1.603 \cdot 10^{-11} \\ 3.268 \cdot 10^{-13} \\ -1.416 \cdot 10^{3} \\ 0 \end{bmatrix}$$

$$\boxed{F_{bolt\_op} := R_{seg_3} \cdot N = -1.724 \times 10^{4}\, N}$$

Tensile external load applied to the bolt, operation

Case #2 - Determine the forces applied during seismic event (seismic + centripetal)

$F_{seis_k} := 0$ Initialize a seismic force vector of correct size

$F_{seis_{2n-1}} := Mg_{seg_n} \cdot A_h \cdot \frac{s^2}{m}$ Determine the seismic force vector.

$Fg_{seg\_seis} := Fg_{seg\_cent} + F_{seis}$ Combine the seismic and operational loads

Use the FEsolve program (appendix) to solve for displacements.

$\delta_{seg\_seis} := FE_{solve}\left(Kg_{seg}, BC_{stat}, BC_{app}, Fg_{seg\_seis}\right)$ Displacements of DOF

$R_{seg\_seis} := Kg_{seg} \cdot \delta_{seg\_seis} - Fg_{seg\_seis}$ Determine the reaction forces in the segment.

$\delta_{seg\_seis} =$

	1
1	0
2	$3.556 \cdot 10^{-5}$
3	0
4	$-7.111 \cdot 10^{-5}$
5	$1.481 \cdot 10^{-3}$
6	$-7.934 \cdot 10^{-4}$
7	$2.383 \cdot 10^{-3}$
8	$-7.064 \cdot 10^{-4}$
9	$2.028 \cdot 10^{-4}$
10	$3.033 \cdot 10^{-3}$
11	0
12	$3.312 \cdot 10^{-3}$

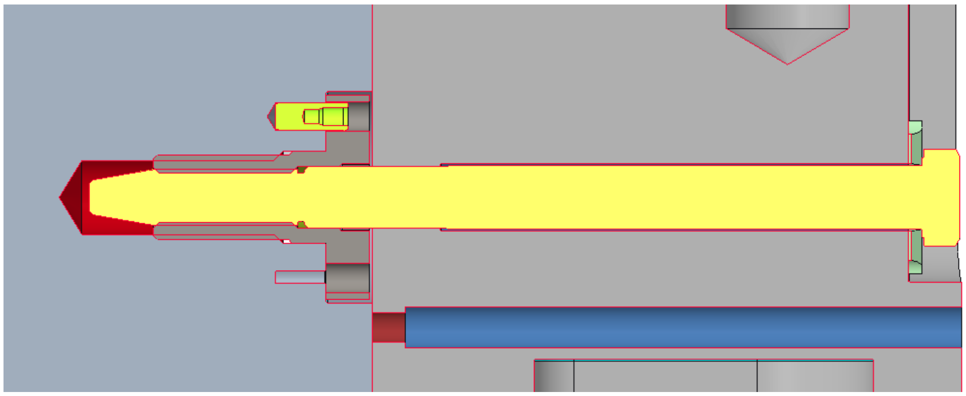
$R_{seg\_seis} =$

	1
1	$2.285 \cdot 10^{4}$
2	0
3	$-2.744 \cdot 10^{4}$
4	$4.547 \cdot 10^{-13}$
5	$7.731 \cdot 10^{-12}$
6	$-4.002 \cdot 10^{-11}$
7	$-2.069 \cdot 10^{-11}$
8	$-1.342 \cdot 10^{-11}$
9	$9.209 \cdot 10^{-12}$
10	$7.816 \cdot 10^{-13}$
11	$-2.229 \cdot 10^{3}$
12	$4.547 \cdot 10^{-13}$

$\boxed{F_{bolt\_seis} := R_{seg\_seis_3} \cdot N = 27.442 \cdot kN}$ Tensile external load applied to the bolt, seismic

Component - Segment Bolt

Select: M16 bolt, Inconel 718 N07718 per ASTM B637 & ASTM A1014



$Fi_{seg\_bolt} := 50kN$ — Tensile preload, bolt

$T_{seg\_bolt} := Fi_{seg\_bolt} \cdot k_{bf} \cdot d_{m16} = 240\ J$ — Torque required to achieve preload

Determine the clamping constant for the bolted connection

Estimate the member stiffness. The threaded end does not have the space to freely develop a 30 degree frustum cone. The segment angle is only 18 degrees; therefore the thread/nut side will only develop at 18 degrees. Assume that the stress field develop in both axes equally.

$\alpha_{frust} := 30deg$ — Angle of the unrestrained frustum cone

$\alpha_{seg} := 18deg$ — Angle of the segment

$l_{seg\_grip} := 200mm$ — Grip length of the segment

$ld_{seg} := 175mm$ — Unthreaded protion of the grip

$lt_{seg} := l_{seg\_grip} - ld_{seg} = 25 \cdot mm$ — Threaded portion of the grip

$d_{washer} := 24mm$ — Washer diameter (max head diameter)

$w_{seg} := 60mm$ — Segment width

$t_1 := \dfrac{w_{seg} - d_{washer}}{2 \cdot \tan(\alpha)} = 26.311 \cdot mm$ — Length to develop a full frustum cone, when cone hits side of part.

$t_2 := l_{seg\_grip} - t_1$ — Length of the nut side of the grip

Determine the spring stiffness of the clamped members

$$k1\_seg\_bolt := \frac{\pi \cdot E_{316} \cdot d_{m16} \cdot \tan(\alpha_{frust})}{\sqrt{3} \cdot \ln\left[\frac{(2 \cdot t_1 \cdot \tan(\alpha_{frust}) + d_{washer} - d_{m16}) \cdot (d_{washer} + d_{m16})}{(2 \cdot t_1 \cdot \tan(\alpha_{frust}) + d_{washer} - d_{m16}) \cdot (d_{washer} - d_{m16})}\right]} = 2.082 \times 10^3 \cdot \frac{kN}{mm}$$

Compute the stiffness of the nut side - simplify as if the insert were a part of the crown. Not perfect, but a reasonable approximation.

$$k2\_seg\_bolt := \frac{\pi \cdot E_{316} \cdot d_{m16} \cdot \tan(\alpha_{seg})}{\sqrt{3} \cdot \ln\left[\frac{(2 \cdot t_2 \cdot \tan(\alpha_{seg}) + d_{washer} - d_{m16}) \cdot (d_{washer} + d_{m16})}{(2 \cdot t_2 \cdot \tan(\alpha_{seg}) + d_{washer} - d_{m16}) \cdot (d_{washer} - d_{m16})}\right]} = 1.171 \times 10^3 \cdot \frac{kN}{mm}$$

$$k_m := \frac{1}{\frac{1}{k1\_seg\_bolt} + \frac{1}{k2\_seg\_bolt}} = 749.602 \cdot \frac{kN}{mm}$$

Overall spring stiffness fo the clamped members

Compute the stiffness of the bolt

$$k_b := \frac{A_{m16} \cdot At_{m16} \cdot E_{carbon\_steel}}{A_{m16} \cdot lt_{seg} + At_{m16} \cdot ld_{seg}} = 194.248 \cdot \frac{kN}{mm}$$

Bolt spring stiffness

Compute the overall joint constant and bolt load

$$C_{seg\_joint} := \frac{k_b}{k_b + k_m} = 0.206$$

Joint constant

$$F_{seg\_bolt} := Fi_{seg\_bolt} + C_{seg\_joint} \cdot F_{bolt\_seis} = 55.648 \cdot kN$$

Total bolt load; external and preload

$$F_{seg\_bolt\_cap} := At_{m16} \cdot Su_{718} = 200.175 \cdot kN$$

Bolt capacity

$$\boxed{\frac{F_{seg\_bolt\_cap}}{F_{seg\_bolt}} = 3.597}$$

OK

Tensile capacity/demand (2 required)

Component - Segment Bolt Thread Insert

Check the thread pullout within the insert

$l_{thrd\_ins} := 30mm$ — Effective length of the thread insert

$$F_{thrd\_ins} := \frac{\pi \cdot dp_{m16} \cdot Su_{N60} \cdot l_{thrd\_ins}}{\sqrt{3} \cdot 3} = 172.302 \cdot kN$$

Pullout capacity of the threaded hole in the thread insert.

$$\boxed{\frac{F_{thrd\_ins}}{F_{seg\_bolt}} = 3.096}$$

OK — Acceptable thread pullout margin

Check the tesnile capacity of the bolts holding the insert to the crown.

Select: M8 12.9 Stainless bolting per ASTM A193 to secure insert to crown.

$Nb_{ins} := 8$ — Number of fasteners holding the insert into the crown.

$$F_{seg\_ins\_bolt} := \frac{F_{seg\_bolt}}{Nb_{ins}} = 6.956 \cdot kN$$

Force in the securing bolt

$$\sigma b_{ins} := \frac{F_{seg\_ins\_bolt}}{At_{m8}} = 190.053 \cdot MPa$$

Stress of the fasteners securing the insert

$$\boxed{\frac{Sp_{129}}{\sigma b_{ins}} = 5.104}$$

Margin of the fasteners to yield

Check the pullout capacity of the bolt into the crown

$l_{thrd\_ext} := 13mm$ — Length of the thread

$$F_{thrd\_ext} := \frac{\pi \cdot dp_{m8} \cdot Su_{625} \cdot l_{thrd\_ext}}{\sqrt{3} \cdot 3} = 38.162 \cdot kN$$

Pullout capacity of the threaded hole in the thread insert.

$$\boxed{\frac{F_{thrd\_ext}}{F_{seg\_ins\_bolt}} = 5.486}$$

OK — Acceptable thread pullout amrgin

Check the capacity of the dowel to support the torque of preload or emergency extraction.

$Tmax_{bolt} := 550N \cdot m$ — Torque to induce yield in M16 bolt.

$d_{dowel} := 8mm$ — Outer diameter of the dowel pin

$r_{dowel} := 23mm$ — Distance to center of dowel

$$V_{dowel} := \frac{T_{seg\_bolt}}{r_{dowel}} = 10.435 \cdot kN$$

Shear demand on the dowel

$V_{d\_cap} := 50kip$ — Single shear capacity of the pull-out dowel

$$\boxed{\frac{V_{d\_cap}}{V_{dowel}} = 21.314}$$

OK — Margin capacity/demand for the dowel in single shear.

Check the ability of the bolts to support the torque imposed during prelaod or emergency extraction

$d_{tibc} := 26mm$ — Diameter of the thread insert bolt circle

$$V_{tibc} := \frac{Tmax_{bolt}}{d_{tibc} \cdot Nb_{ins}} = 2.644 \times 10^{3}\,N$$

Shear on single fastener, assuming all share torque load equally.

$$\tau_{tibc} := \frac{V_{tibc}}{At_{m8}} = 72.247 \cdot MPa$$

Shear stress in the bolt

$$\boxed{\frac{Sy_{129}}{\sqrt{3} \cdot \tau_{tibc}} = 8.791}$$

OK

Margin against exccedign the bolt shear capacity.

Check the ability of the insert to resist the torsion at max extraction torques

$do_{ins} := 26mm$ — Outer diameter of the insert

$$J_{ins} := \frac{\pi}{4} \cdot \left(do_{ins}^{\ 4} - d_{m16}^{\ 4} \right) = 3.074 \times 10^{5} \cdot mm^{4}$$

Radius of gyration

$$\tau_{ins} := \frac{Tmax_{bolt} \cdot do_{ins}}{2 \cdot J_{ins}} = 23.257 \cdot MPa$$

Shear stress in the insert due to max torque

$$\frac{Sy_{N60}}{\sqrt{3} \cdot \tau_{ins}} = 8.558$$

OK

Margin against exceeding plastic deformation, during peak torsion.

Check the bearing stress of the dowel in the crown

$l_{ins} := 12mm$ — Length of isnertion into the crown

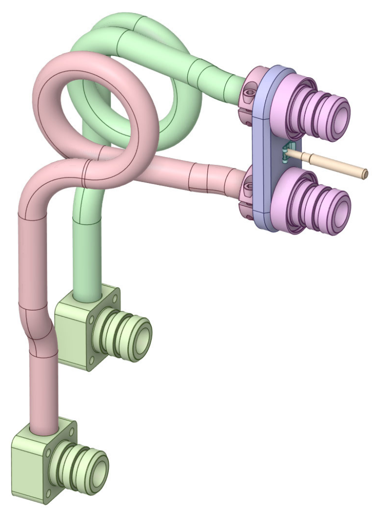
$$\sigma_{brg\_dowel} := \frac{V_{dowel}}{d_{dowel} \cdot l_{ins}} = 108.696 \cdot MPa$$

Bearing sress

$$\boxed{\frac{Sy_{625}}{\sigma_{brg\_dowel}} = 2.664}$$

OK

Component - water connection plate



The water connections enable the structural capacity of the segment to be seperated from the water connections. Specifically, motion and deflection of the segment does not compormise the sealing of the water passages.

We will check the following failure modes
-Kinking during withdrawal
-Ability of the bolt to draw into position
-Stress in the joiing plate

Check the ability of the piping to handle the pressure

$do_{tube} := 0.625in$ Outer diameter of the tube

$t_{tube} := 0.049in$ Wall thickness of the tube

$S_{3003} := 3.4ksi$ Allowable stress, AL 3003 (ASME SEC VIII, Part D, Table 1B)

$$\sigma_h := \frac{P_{MAWP}\cdot\left(do_{tube} - 2\cdot t_{tube}\right)}{2\cdot t_{tube}} = 4.635\cdot MPa$$

$$\frac{S_{3003}}{\sigma_h} = 5.058$$ **OK**

Check the ability of the tubes to withstand motion without kinking

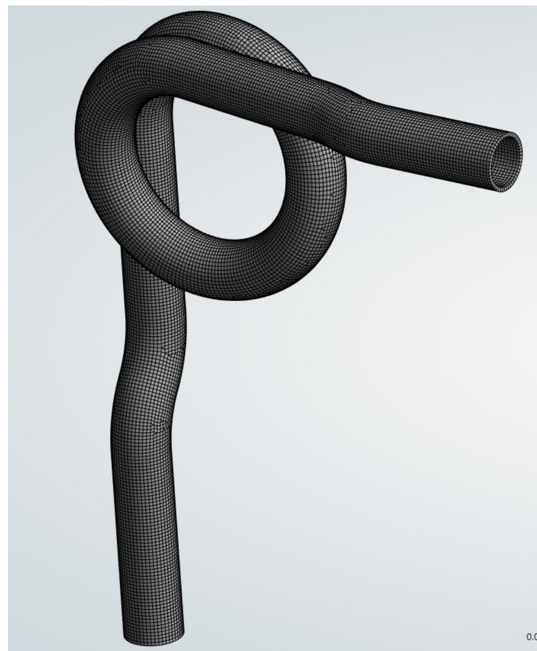
We will use a nonlinear Finite element model to impose displacement on the tubes, and monitor deflection, and reactions for "kinking" behavior. We are looking for smooth convergnece as displacement is imposed.

Finite Elment Model File: ANSYS model FLEX_TUBE.wbpj

Use a midsurface for 5/8"OD tube shell elements of 0.049" thick (1.245E-3m), per ASTM B483

Geometry

Mesh



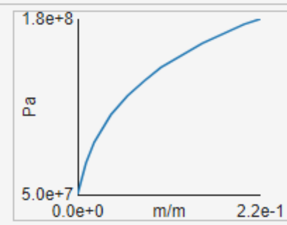
Material Properties

3003-0

General aluminum alloy. Fatigue properties come from MIL-HDBK-5H, page 3-277.

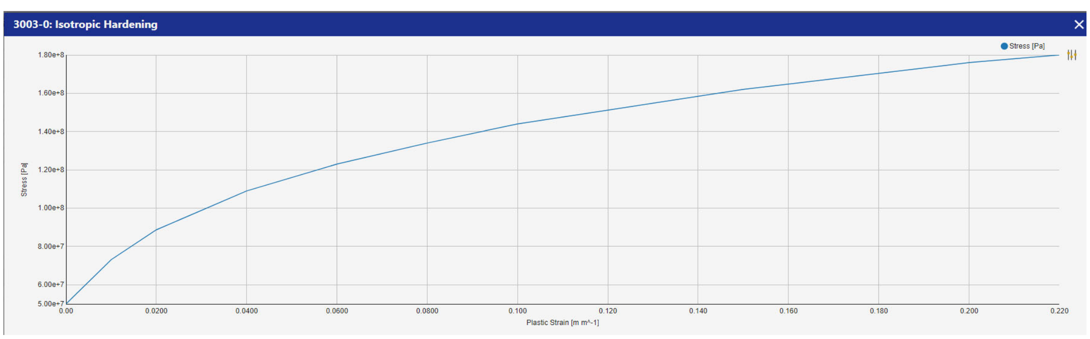
Density	2770 kg/m³

Structural

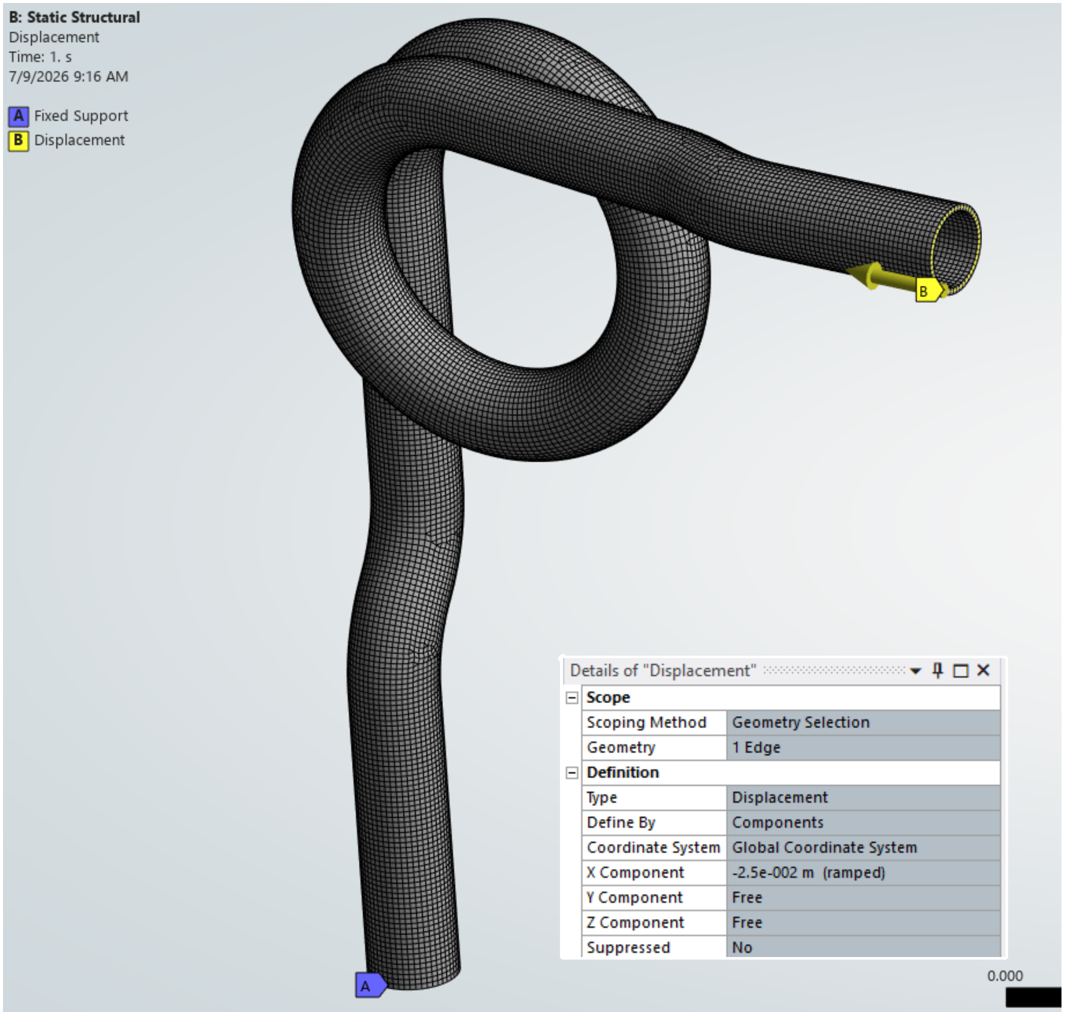
Isotropic Elasticity	
Derive from	Young's Modulus and Poisson's Ratio
Young's Modulus	7.1e+10 Pa
Poisson's Ratio	0.33
Bulk Modulus	6.9608e+10 Pa
Shear Modulus	2.6692e+10 Pa
Multilinear Isotropic Hardening	

Thermal

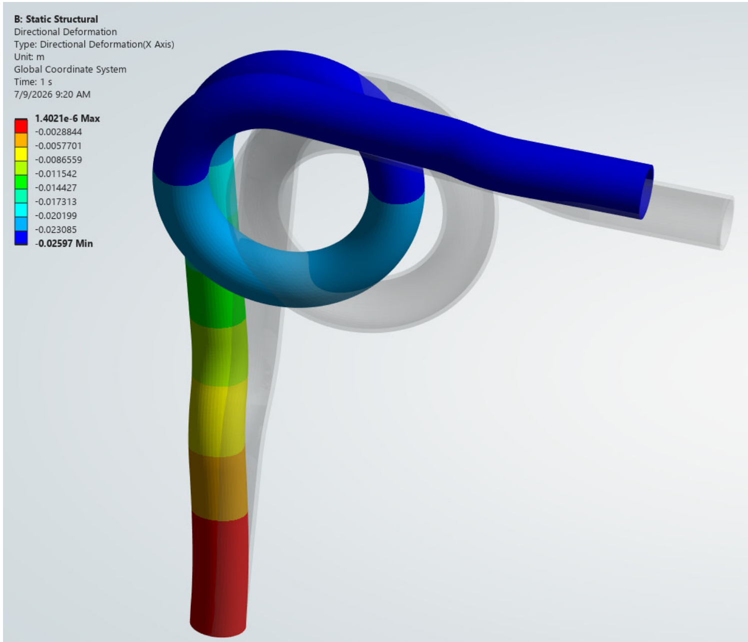
Specific Heat Constant Pressure	875 J/kg·°C

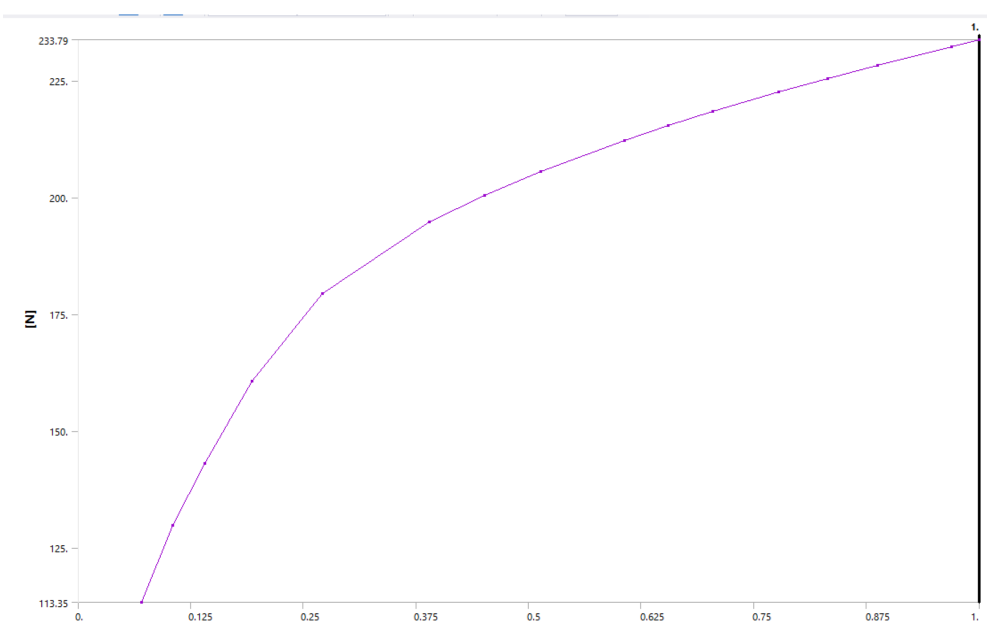


Boundary Conditions



Results

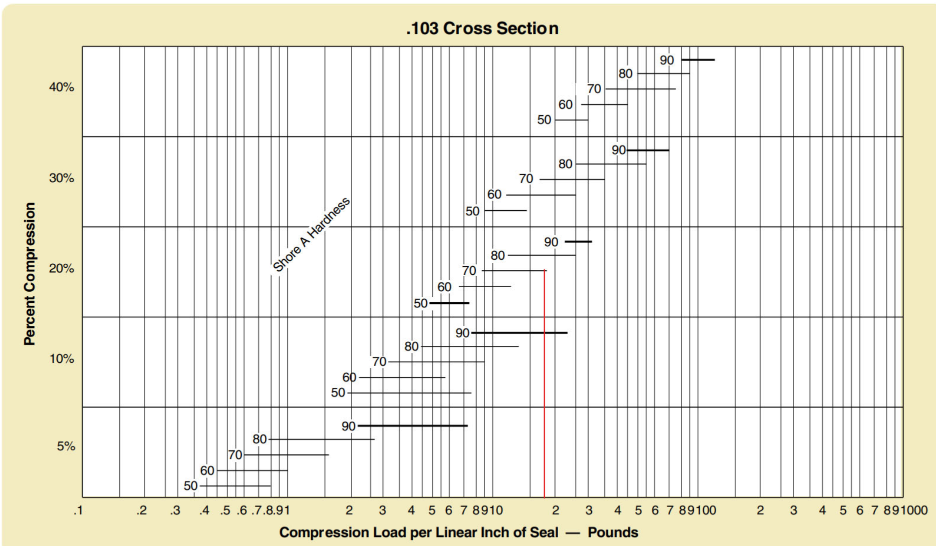




We see a very smooth and controlled reaction force showing that there are no sudden kinks, flattening or local buckling effects.

$F_{ins\_flex} := 250N$ Force required to withdraw 1 tube (of two)

Determine the axial force required to insert the seal with two O-rings



$Fc_{o\_ring} := 20\frac{lbf}{in}$ Compression load, per inch of circumference, Shore 70, 0.103" cross section. Parker O-ring handbook.

$\mu_{o\_ring} := 0.3$ Static coeffecient of friction for O-ring installation. (with lubrication)

$d_{o\_ring} := 25.4mm$ Diameter of the O-ring sealing face (1")

$F_{ins\_oring} := Fc_{o\_ring} \cdot \pi \cdot d_{o\_ring} \cdot \mu_{o\_ring} = 83.847\,N$ Force to "slide" and individual O-ring

$F_{ins} := 4 \cdot F_{ins\_oring} + 2 \cdot F_{ins\_flex} = 835.388\,N$ Total force required to insert/withdraw the two connections

Check the capacity of the draw-bolt.

$d_{db\_min} := 4mm$ Smallest diameter of the draw bolt

$A_{db} := \frac{\pi}{4} \cdot d_{db\_min}^{\;2} = 12.566 \cdot mm^2$ Cross sectional area of the draw-bolt

$\sigma_{db} := \frac{F_{ins}}{A_{db}} = 66.478 \cdot MPa$ Stress in the draw bolt

$\frac{Sy_{316}}{\sigma_{db}} = 3.63$ **OK** Margin against exceeding yield on the draw bolt

Check the stress in the joining plate

$w_{pl} := 20mm$	Width of the plate
$t_{pl} := 7mm$	Thickness of the plate
$d_{pl} := 7mm$	Diameter of the hole in the plate
$l_{cc} := 60mm$	Distance between centers of the tubes
$I_{pl} := \frac{(w_{pl} - d_{pl}) \cdot t_{pl}^{3}}{12} = 371.583 \cdot mm^{4}$	Moment of inertia for the plate in bending
$\frac{d_{pl}}{t_{pl}} = 1 \qquad \frac{d_{pl}}{w_{pl}} = 0.35$	Diameter to thickness ratio for determining stress concentration factor.
$K_{sc} := 1.6$	Stress concentration factor, Shigles Figure A-15-2
$M_{pl} := \frac{F_{ins} \cdot l_{cc}}{8} = 6.265\ J$	Peak force in the center of the plate
$\sigma b_{pl} := \frac{K_{sc} \cdot M_{pl} \cdot t_{pl}}{2 I_{pl}} = 94.424 \cdot MPa$	Bending stress in the plate
$\frac{Sy_{316}}{\sigma b_{pl}} = 2.556$ **OK**	Margin against exceeding yield in the plate.

Conclusions:

Future Work:
-Check the ability for the split ring to avoid overturning (turning inside out)
-Check capacity of LOTO Pin
-Add a machine to fit shim under the drive
-Frequency of accelerometr based upo overrolling frequencies of bearing
-Runot for motor
-Check pusher block threads
-Check bearing stress of the drive resting on the lid

Functions

FUNCTION - GENERATE A BOLT CIRCLE

$$
BC(d, N, Axis) \equiv \left|
\begin{array}{l}
\theta \leftarrow \dfrac{2\pi}{N} \\
r \leftarrow \dfrac{d}{2} \\
\text{for } i \in 1..N \qquad\qquad\qquad\qquad \text{if } Axis = (1 \ \ 0 \ \ 0) \\
\quad B^{\langle i \rangle} \leftarrow (0mm \quad r \cdot \sin(i \cdot \theta) \quad r \cdot \sin(i \cdot \theta))^{T} \\
\text{for } i \in 1..N \qquad\qquad\qquad\qquad \text{if } Axis = (0 \ \ 1 \ \ 0) \\
\quad B^{\langle i \rangle} \leftarrow (r \cdot \sin(i \cdot \theta) \quad 0mm \quad r \cdot \cos(i \cdot \theta))^{T} \\
\text{for } i \in 1..N \qquad\qquad\qquad\qquad \text{if } Axis = (0 \ \ 0 \ \ 1) \\
\quad B^{\langle i \rangle} \leftarrow (r \cdot \sin(i \cdot \theta) \quad r \cdot \cos(i \cdot \theta) \quad 0mm)^{T} \\
B^{T}
\end{array}
\right.
$$

FUNCTION - Determine the forces in an arbitrary bolt pattern

$$\mathrm{BoltForce}(B, s, M, F) \equiv$$

$$N_{bolts} \leftarrow \mathrm{rows}(B)$$

$$X_c \leftarrow \frac{\sum_{j=1}^{N_{bolts}} B_{j,1}}{N_{bolts}}$$

X Centroid

$$Y_c \leftarrow \frac{\sum_{j=1}^{N_{bolts}} B_{j,2}}{N_{bolts}}$$

Y Centroid

$$Z_c \leftarrow \frac{\sum_{j=1}^{N_{bolts}} B_{j,3}}{N_{bolts}}$$

Z Centroid

$$\mathrm{Centroid} \leftarrow \begin{pmatrix} X_c & Y_c & Z_c \end{pmatrix}$$

$$B_h \leftarrow B^T$$

Transfrom the bolt pattern to a column

$$\text{for } i \in 1 .. N_{bolts}$$

$$\quad B_h^{\langle i \rangle} \leftarrow B_h^{\langle i \rangle} - \mathrm{Centroid}^T$$

Shift the botl pattern so that bolt coordiantes are relative to the pattern CG

$$I_3 \leftarrow \mathrm{identity}(3)$$

$$I \leftarrow \sum_{j=1}^{N_{bolts}} \left[\left(B_h^{\langle j \rangle} \cdot B_h^{\langle j \rangle} \right) \cdot I_3 - B_h^{\langle j \rangle} \cdot B_h^{\langle j \rangle T} \right]$$

Calcuate the Area Moment of Inertia for the pattern

$$I_d \leftarrow \mathrm{diag}(I)^T$$

$$s_{loc} \leftarrow s - \mathrm{Centroid}$$

Determine the distance from the bolt pattern to the load application point

$$M \leftarrow M + \left(s_{loc}{}^T \times F^T \right)^T$$

Determine the moment associated with the force-distance

$$\text{for } i \in 1 .. N_{bolts}$$

$$\quad F_b^{\langle i \rangle} \leftarrow \left(\frac{M}{I_d} \right)^T \times B_h^{\langle i \rangle} + \frac{F^T}{N_{bolts}}$$

Determine the resultant moment

$$F_b{}^T$$

Appendix - FUNCTION - Determine stress under arbitrary load

A = Area, cross section under consideration
I= Moment of Inertia
Sxc= Unit vector normal to the cross section
Sloc= Vector from center of area to location where stress is desired
M=Moment imposed on the system
F=force imposed at the location
Q= Shear flow modifier for cross section (Assume that the worst case shear occurs on the surface)
Ktb=stress concetration, bending
kta=stress concetration, axial
Ktt=stress concetration, torsion

$$VM_{xc}\left(A, I, Q, s_{xc}, M, F, s_{loc}, Kt_a, Kt_b, Kt_t\right) \equiv \left| \begin{array}{l} Ref_{shear} \leftarrow \begin{pmatrix} 1 \\ 1 \\ 1 \end{pmatrix} - s_{xc}^{\ T} \\ \sigma \leftarrow \overrightarrow{\left[\left(\frac{M^T}{I} \times s_{loc}^{\ T} \cdot Kt_b + \frac{F^T}{A} \cdot Kt_a\right) \cdot s_{xc}^{\ T}\right]} \\ \tau \leftarrow \overrightarrow{\left[\left(\frac{M^T}{I} \times s_{loc}^{\ T} \cdot Kt_t + \frac{F^T}{A} \cdot Q\right) \cdot Ref_{shear}\right]} \\ \sigma_{eff} \leftarrow \frac{1}{\sqrt{2}} \cdot \sqrt{\left[\begin{array}{l} \left(\sigma_0 - \sigma_1\right)^2 + \left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_0\right)^2 \ldots \\ + 6 \cdot \left[\left(\tau_0\right)^2 + \left(\tau_1\right)^2 + \left(\tau_2\right)^2\right] \end{array}\right]} \end{array} \right.$$

Stress/force normal to XC

Stresses in plane of XC

Appendix - Generate flexural stiffness matrix of beam element

Function to generate flexural stiffnes matrix

$$k_{loc}(E,I,l) \equiv \frac{E \cdot I}{l^3} \cdot \begin{pmatrix} 12 & -6 \cdot l & -12 & -6 \cdot l \\ -6 \cdot l & 4 \cdot l^2 & 6 \cdot l & 2 \cdot l^2 \\ -12 & 6 \cdot l & 12 & 6 \cdot l \\ -6 \cdot l & 2 \cdot l^2 & 6 \cdot l & 4 \cdot l^2 \end{pmatrix}$$

Appendix - Assemble global stiffness matrix using local stiffness and connectivity

$$\text{Assemble}\left(K_{global}, k_{local}, M_{conn}\right) \equiv \left(\begin{array}{l} DOF \leftarrow \text{cols}\left(M_{conn}\right) \\ \text{for } i \in 1..DOF \\ \quad \text{for } j \in 1..DOF \\ \quad\quad \left| \begin{array}{l} row_{index} \leftarrow M_{conn_{1,i}} \\ col_{index} \leftarrow M_{conn_{1,j}} \\ K_{global_{row_{index}, col_{index}}} \leftarrow K_{global_{row_{index}, col_{index}}} + k_{local_{i,j}} \end{array} \right. \\ K_{global} \end{array} \right)$$

APPENDIX - Assemble global column vector (force, mass, etc)

$$\text{CVassm}\left(CV_{master}, CV_{local}, N_{conn}\right) \equiv \left| \begin{array}{l} Nodes \leftarrow \text{rows}\left(N_{conn}\right) \\ \text{for } i \in 1..Nodes \\ \quad CV_{master\left(N_{conn_i}\right)} \leftarrow CV_{master\left(N_{conn_i}\right)} + CV_{local_i} \\ CV_{master} \end{array} \right.$$

Functions

APPENDIX - MATRIX FE Solver

Write a program to solve the FE matrix problem usign the Direct Stiffness Matrix method
1) Arrange any forced displacements into the force side of the equation via F'=F-K*d
2) Make rows and columns of prescribed BC's (0s) all zero
3) Set the diagonal entry to 1 for prescribed BC's
4) Sole the FE problem K-1*F'

$$FE_{solve}(K, BC_{stat}, BC, F) \equiv \left| \begin{array}{l} \text{for } i \in 1..\text{length}(BC_{stat}) \\ \quad F \leftarrow F - K^{\langle i \rangle} \cdot BC_i \\ \text{for } i \in 1..\text{length}(BC_{stat}) \\ \quad \text{for } j \in 1..\text{length}(BC_{stat}) \\ \qquad \left| \begin{array}{l} K_{i,j} \leftarrow 0 \text{ if } BC_{stat_i} = 1 \\ K_{i,j} \leftarrow 0 \text{ if } BC_{stat_j} = 1 \\ K_{i,j} \leftarrow 1 \text{ if } i = j \wedge BC_{stat_j} = 1 \\ F_i \leftarrow BC_i \text{ if } BC_{stat_i} = 1 \end{array} \right. \\ K^{-1} \cdot F \end{array} \right.$$

Verification of function:

$$K_{test} := \begin{pmatrix} 12 & 6 & -12 & 6 & 0 & 0 \\ 6 & 4 & -6 & 2 & 0 & 0 \\ -12 & -6 & 24 & 0 & -12 & 6 \\ 6 & 2 & 0 & 8 & -6 & 2 \\ 0 & 0 & -12 & -6 & 12 & -6 \\ 0 & 0 & 6 & 2 & -6 & 4 \end{pmatrix} \quad BC_{stat\_test} := \begin{pmatrix} 1 \\ 1 \\ 0 \\ 0 \\ 1 \\ 0 \end{pmatrix} \quad BC_{test} := \begin{pmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0.5 \\ 0 \end{pmatrix} \quad F_{test} := \begin{pmatrix} 0 \\ 0 \\ 10 \\ 0 \\ 20 \\ -10 \end{pmatrix}$$

$$\delta_{test} := FE_{solve}(K_{test}, BC_{stat\_test}, BC_{test}, F_{test}) = \begin{pmatrix} 0 \\ 0 \\ 2.135 \\ 1.844 \\ 0.5 \\ -5.875 \end{pmatrix} \qquad R_{test} := K_{test} \cdot \delta_{test} = \begin{pmatrix} -14.563 \\ -9.125 \\ 10 \\ 0 \\ 4.563 \\ -10 \end{pmatrix}$$