

Gasoline Pump

ME559

Team 6: Fall 2025

Matt Stafford, Joseph Letteney, Dana Bulakh

Table of Contents

Table of Contents	2
Design Choices	3
Vane Geometry	3
Material Selection	4
Vanes	4
Shaft	4
Rotor	4
Housing	4
Inserts	4
Front & Back Plates	4
Bearing Caps	4
Clearances and Tolerances	5
Shaft Seal	5
Bearings	5
Rotor and Peek Insert	5
Fastener Selection	5
Fastener Torques	5
Surface Finishes	5
Bill of Materials	5

Design Choices

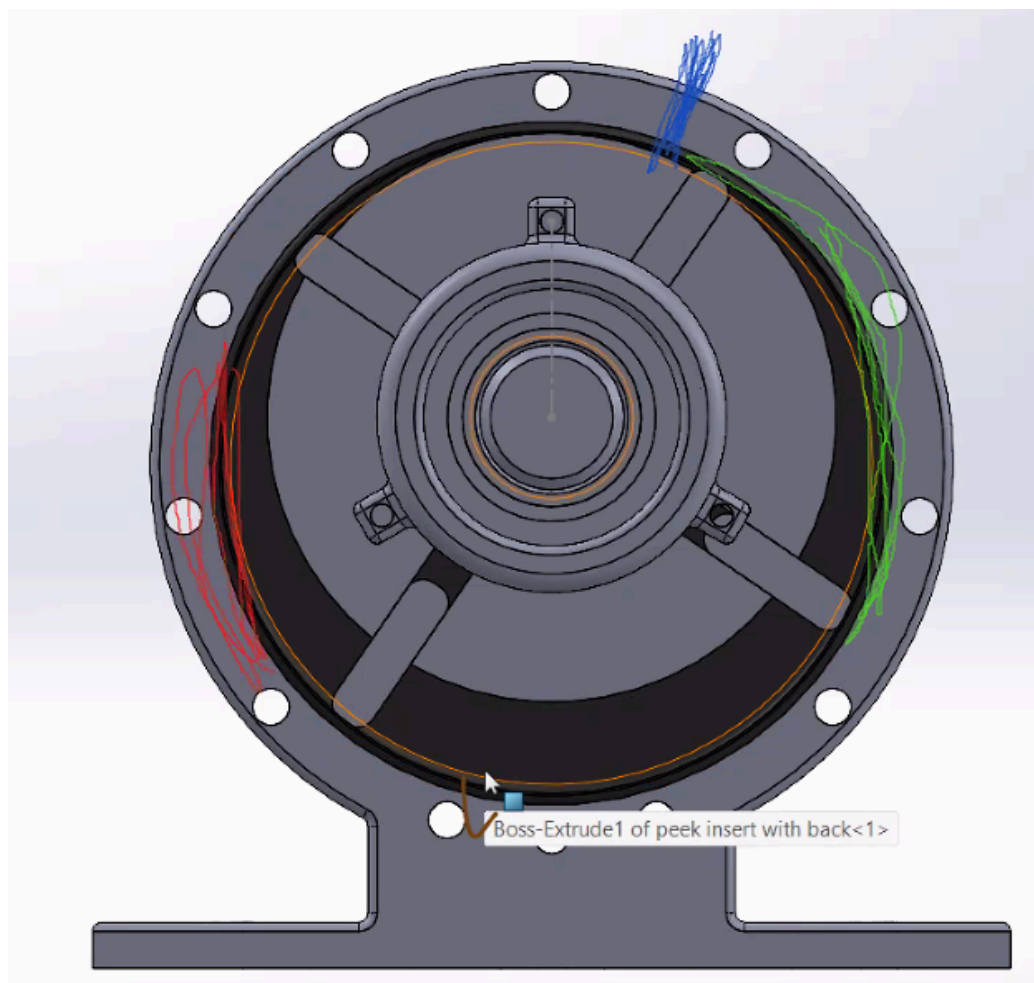
Housing Geometry

Through hole clearances and tolerances defined by manufacturer specifications (see: clearances and tolerances section)

Choose 4 vane slots for symmetry.

Bottom of side channels start at 45° from vertical ("6 o'clock" $\pm 45^\circ$)

Add pins for peek housing insert that would help with alignment



Housing Thread Calculations performed following the following equation (see spreadsheet for specific calculations)

Shear Area For Internal Thread

$$A_n = \pi n L_e D_s \min \left(\frac{1}{2n} + 0.57735 (D_s \min - E_n \max) \right)$$

Where:

$E_n \max$ = Maximum pitch diameter of internal thread.

$D_s \min$ = Minimum major diameter of external thread.

L_e = Fastener thread engagement

n = Number of threads per inch

Vane Geometry

Through-holes and side cutouts added to reduce radial fluid pressure.

Calculator that led to the selection of the vane geometry and material:

1. PV

In order to calculate the PV value required for our pump we first need to calculate the running velocity of the vanes given our inner diameter and rpm

$$\begin{aligned} v &= (rpm)(c) \\ v &= (2000 \frac{rev}{min})(\pi * 3.6in) \\ v &= 1884.96 \frac{ft}{min} \end{aligned}$$

In order to calculate the pressure acting at the tip we find the centrifugal force pushing the vanes out as they spin and then add that to the fluid pressure acting along the base of the vanes.

$$\begin{aligned} F_c &= mr\omega^2 \\ F_c &= (0.001832 \text{ slugs})(0.15ft)(209.44 \frac{ft}{s})^2 \\ F_c &= 12.0lbf \end{aligned}$$

To calculate the force due to pressure we multiply the highest pressure the vanes experience by the area at the base.

$$F_p = (4035.1 \frac{lbf}{in^2})(0.25in^2)$$

$$F_p = 1008.775lbf$$

To calculate the total force acting at the tip we add these two forces.

$$F_{total} = 1008.775 + 12.0$$

$$F_{total} = 1020.775lbf$$

The force due to pressure dominates this equation meaning that the density of the material does not play a significant role (in the equations above hardened tool steel was used for density). In order to calculate the required PV we multiple these numbers together.

$$PV = \left(\frac{1020.775lbf}{0.25in^2}\right)\left(1884.96\frac{ft}{min}\right)$$

$$PV = 1,924,120psi * \frac{ft}{min}$$

A PV value this high rules out plastics which have PV values in the tens of thousands. We can not improve this by reducing the area because whatever area is in contact with the housing is bearing the full force of the pressure and so that does not change. For this reason we decided to use hardened tool steel as our vane material for its higher durability and greater thermal conductivity.

2. Total force out.

In order to reduce the force out we decided to go with a vane design that minimizes the total force out. With this new design the vane is almost entirely pressure balanced. Calculating the force out we get.

$$\Sigma F_y = mr\omega^2 + P_{out}A$$

$$\Sigma F_y = (9.32 \times 10^{-4} slugs)(0.15ft)(209.44\frac{ft}{s})^2 + 4293.12psi(0.01in^2)$$

$$F_{total} = 49.06lb$$

3. Wear Rate

To calculate the total wear on the vanes we use the following equation.

$$w = KFVT$$

$$w = (5.73 \times 10^{-15} \frac{in^3}{lb \cdot ft \cdot min})(49.06LB)(22619.52\frac{in}{min})(402,000min)$$

$$w = 0.0026in^3$$

Therefore over the whole lifespan of the pump the vanes will wear around 2.4%. For the wear rate of the PEEK insert we can use only the time that any given section of it is in contact with a vane. This is the total area of the inner surface minus the total area in

contact with a vane. This works out to be about 96%. Therefore each section of the insert is only in contact with a vane for about 4% of each cycle. Therefore we can reduce our total time in our calculation by 96%.

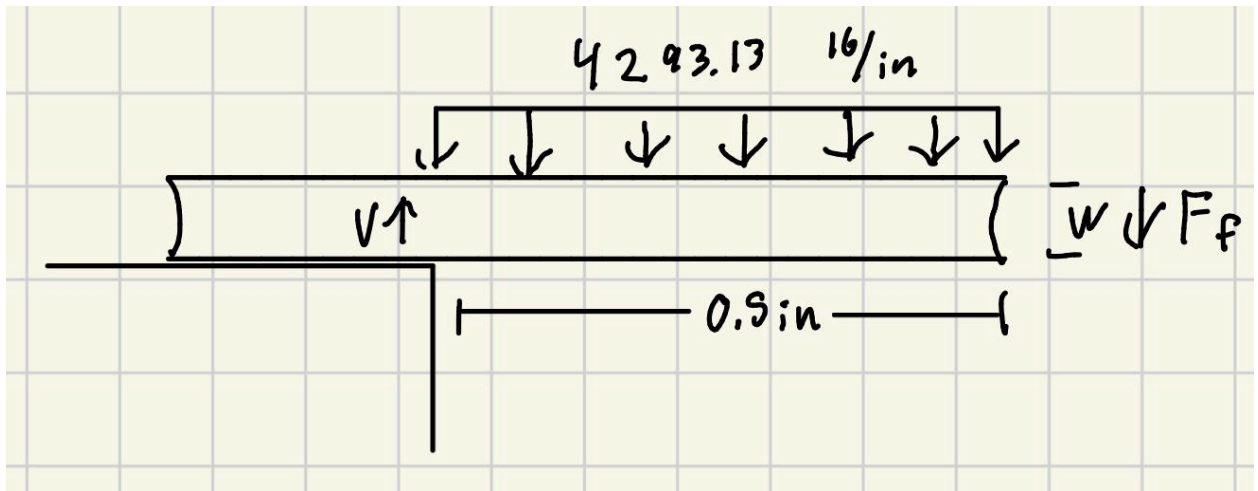
$$w = (5.73 \times 10^{-15} \frac{\text{in}^3}{\text{lb} \cdot \text{in}})(49.06 \text{LB})(22619.52 \frac{\text{in}}{\text{min}})(16,080 \text{min})$$

$$w = 0.000102 \text{in}^3$$

The real wear rate will be significantly less as the above calculations use the highest possible pressure which the vane will only experience for about a third of the time.

4. Vane thickness

In order to calculate the required vane thickness we can model the vane as a cantilevered beam experiencing the force laid out in the following image:



$$\Sigma F_y = 0 = - (4293.13 \frac{\text{lb}}{\text{in}} \times 0.5 \text{in}) + v - (92.702)(0.3)$$

$$v = 2174.38 \text{lb}$$

Now we have to find a thickness that allows us to bear this force.

$$\frac{\text{Yield stress}}{FOS} = \frac{v}{w \times 1 \text{in}}$$

$$\frac{185,000 \text{psi}}{10} = \frac{2174.38 \text{lb}}{w}$$

$$w = 0.117 \text{in}$$

Therefore our vane must be at least 0.117in wide in order to handle the force. We decided to go with 0.125in.

Insert Design

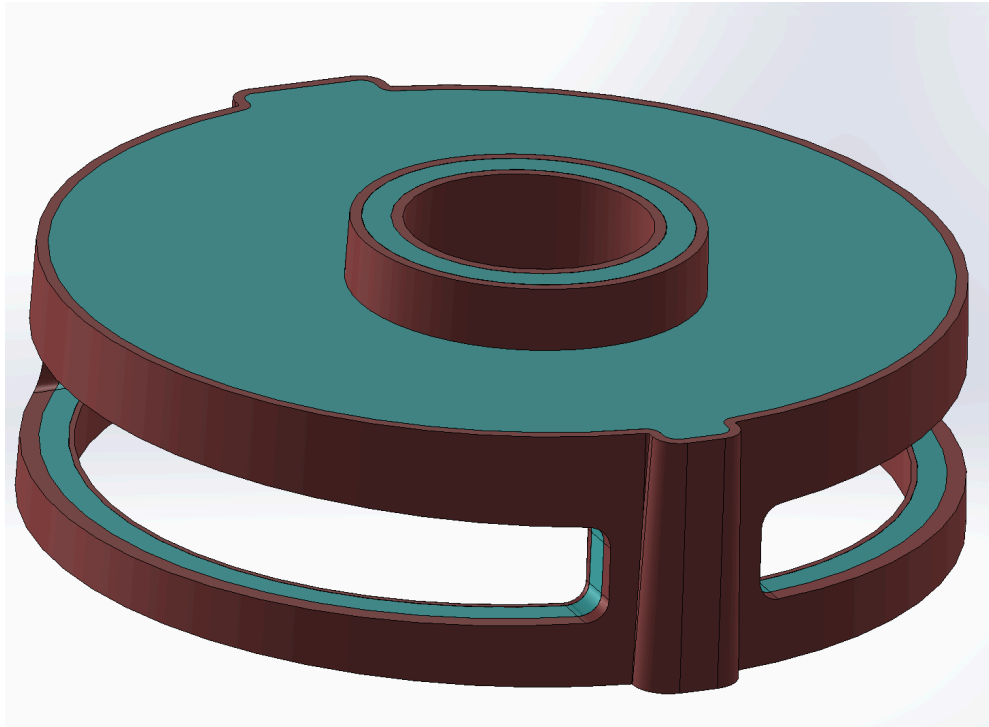
2° Draft added on precursor part

Extra material provided radially to allow removal of draft

Side porting slots given 2° for side-actions

Draft continued in housing port channels for smooth transition

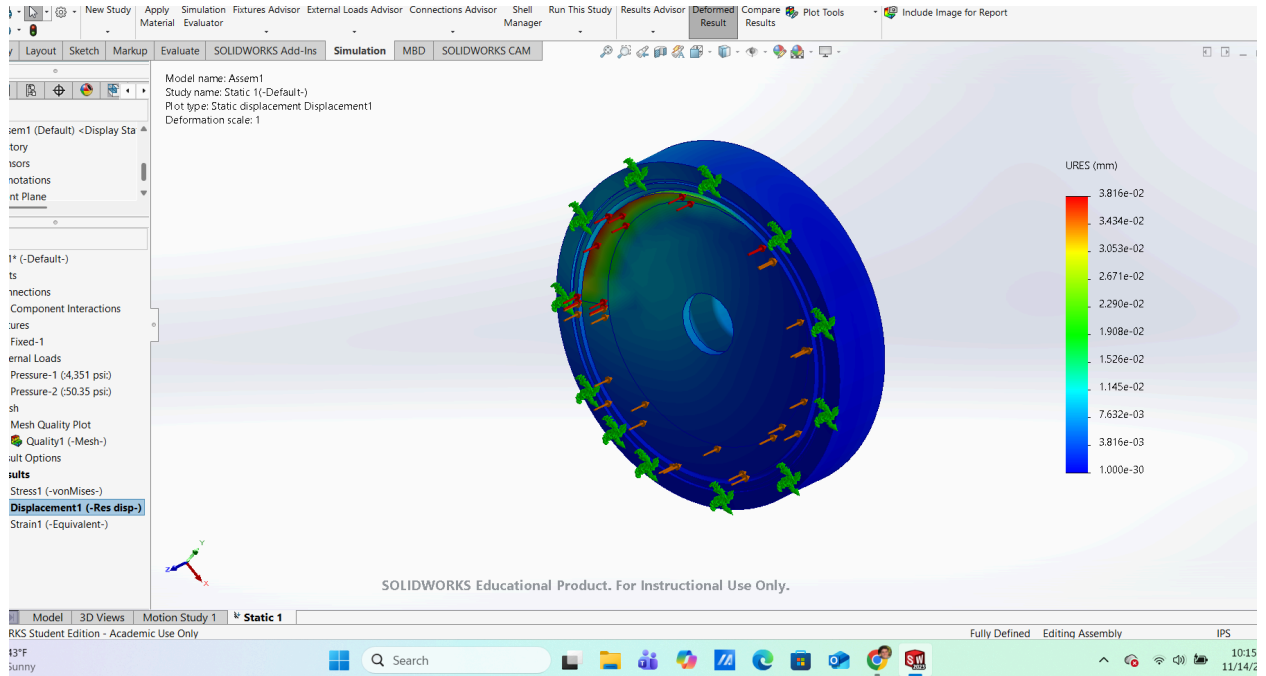
Side ports molded in to reduce machining setups



Red = Removed by Machining

Front Housing

Checked front plate pressure (**2079lb of clamping force required leads to about a thousandth of an inch deformation**)



O-Ring

O-ring used instead of gasket for reliability and to better tolerate potential front plate deformation

O-Ring material (Nitrile) and general dimensions determined from Parker O-Ring Handbook

O-Ring Gland dimensions and surface finish determined in same way

Parker Hannifin states that Backing rings (“ParBaks”) are not required for face seals

Bearing Mounting

Rear bearing provides alignment with G/H6 transition fit

Front bearing transition fit to accommodate thermal expansion

Mounting Bracket

Holes extended beyond body to improve tool access

Shaft

Key slot for 15mm shaft diameter:

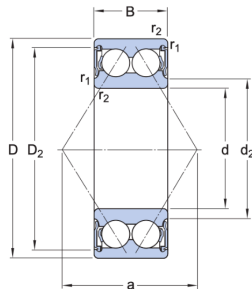
(<https://www.isccompanies.com/wp-content/uploads/2017/01/Keyway-and-Key-Size-Dimensions.pdf>)

Metric Standard Parallel Keyway and Key Sizes

Shaft Diameter (mm)		Keyway (mm)		Key (mm)*	
From	To	Width (W)	Depth (h)	Width (W)	Depth (T)
6	8	2	1.0	2	2
9	10	3	1.4	3	3
11	12	4	1.8	4	4
13	17	5	2.3	5	5
18	22	6	2.8	6	6
23	30	8	3.3	8	7

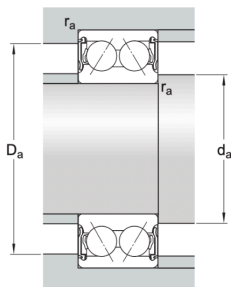
Shaft modifications for proper bearing installment:

(<https://www.skf.com/group/products/rolling-bearings/ball-bearings/angular-contact-ball-bearings/double-row-angular-contact-ball-bearings/productid-3202%20A-2RS1TN9%2FMT33>)



Dimensions

d	15 mm	Bore diameter
D	35 mm	Outside diameter
B	15.9 mm	Width
d ₂	≈ 20.2 mm	Recess diameter inner ring shoulder
D ₂	≈ 30.7 mm	Recess diameter outer ring shoulder
r _{1,2}	min. 0.6 mm	Chamfer dimension inner ring
a	21 mm	Distance pressure point(s)

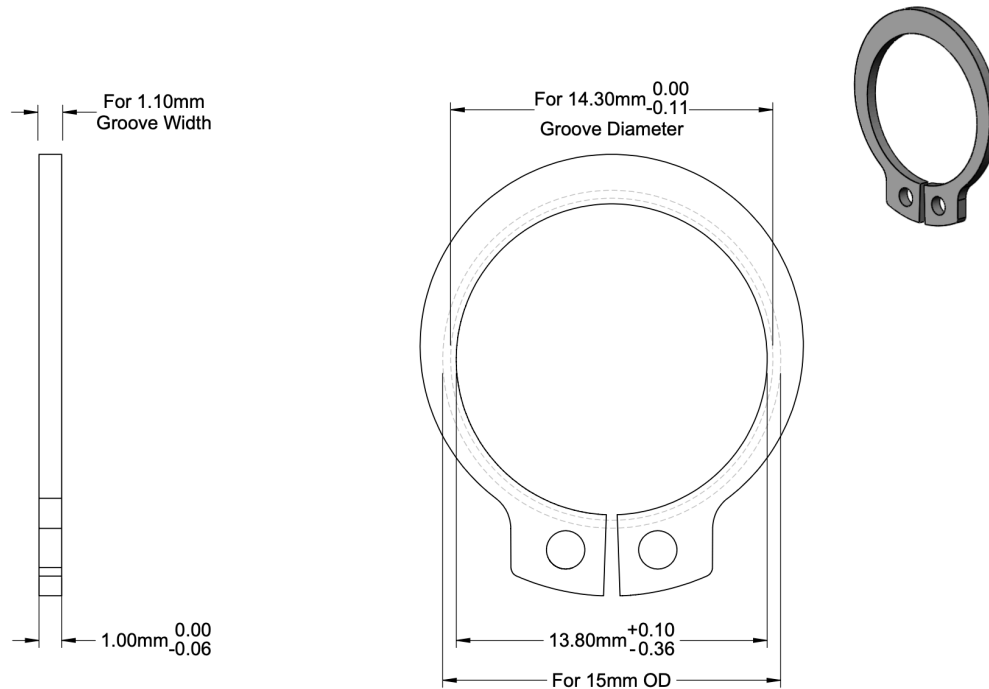


Abutment dimensions

d _a	min. 19.4 mm	Abutment diameter shaft
d _a	max. 20 mm	Abutment diameter shaft
D _a	max. 30.6 mm	Abutment diameter housing
r _a	max. 0.6 mm	Fillet radius

Snap ring groove geometry for proper installment:

(<https://www.mcmaster.com/catalog/131/3752/98541A410>)

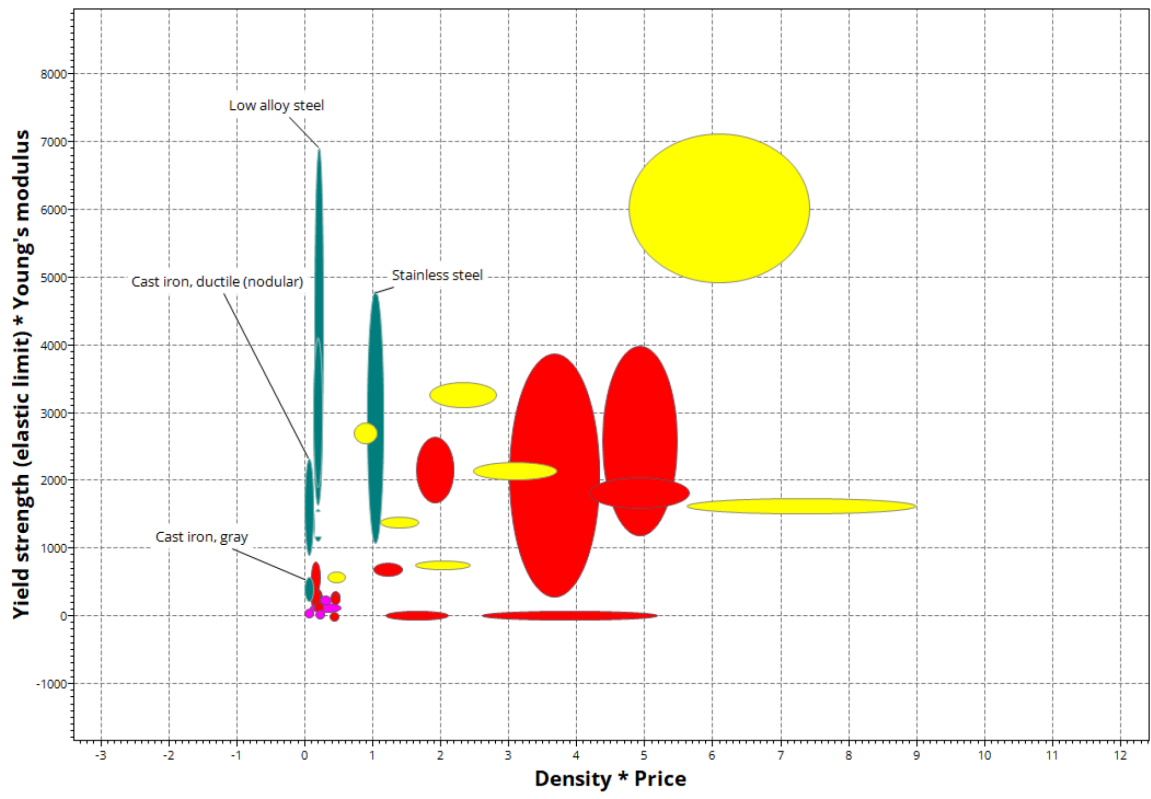


Material Selection & Manufacturing

The vanes were manufactured from hardened tool steel to withstand the extremely high PV values and demanding operating conditions. Tool steel was selected for its superior wear resistance and durability compared to plastic alternatives. Due to the complex vane geometry, powder metal manufacturing was chosen, as it provides excellent dimensional control while enabling the use of tool steel. Following fabrication, the vanes were heat treated to a hardness range of 46–52 HRC to further enhance wear resistance and extend service life.

Shaft & Rotor

For the shaft and rotor which experience a lot of pressure, yield strength and young's modulus are the main mechanical properties to look at, low alloy steel is best for this, considering the price and weight as well.

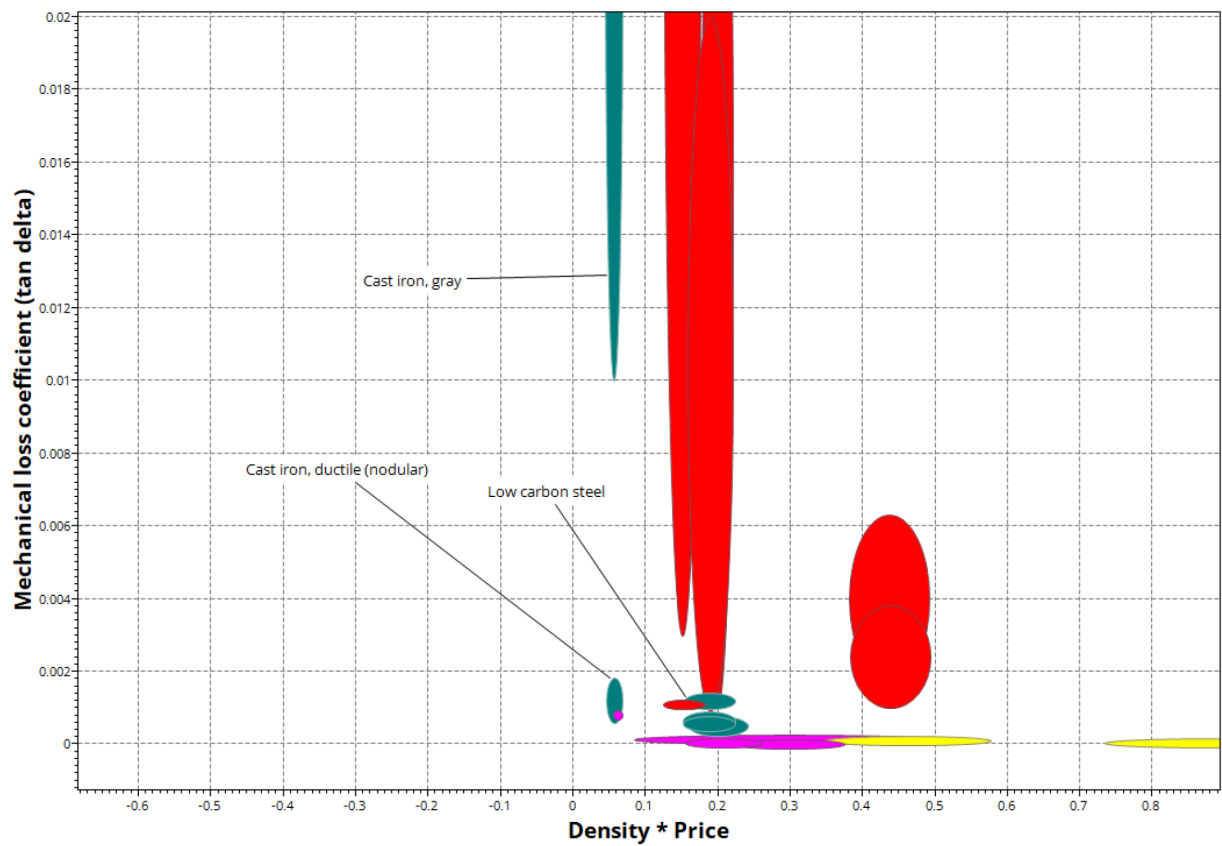
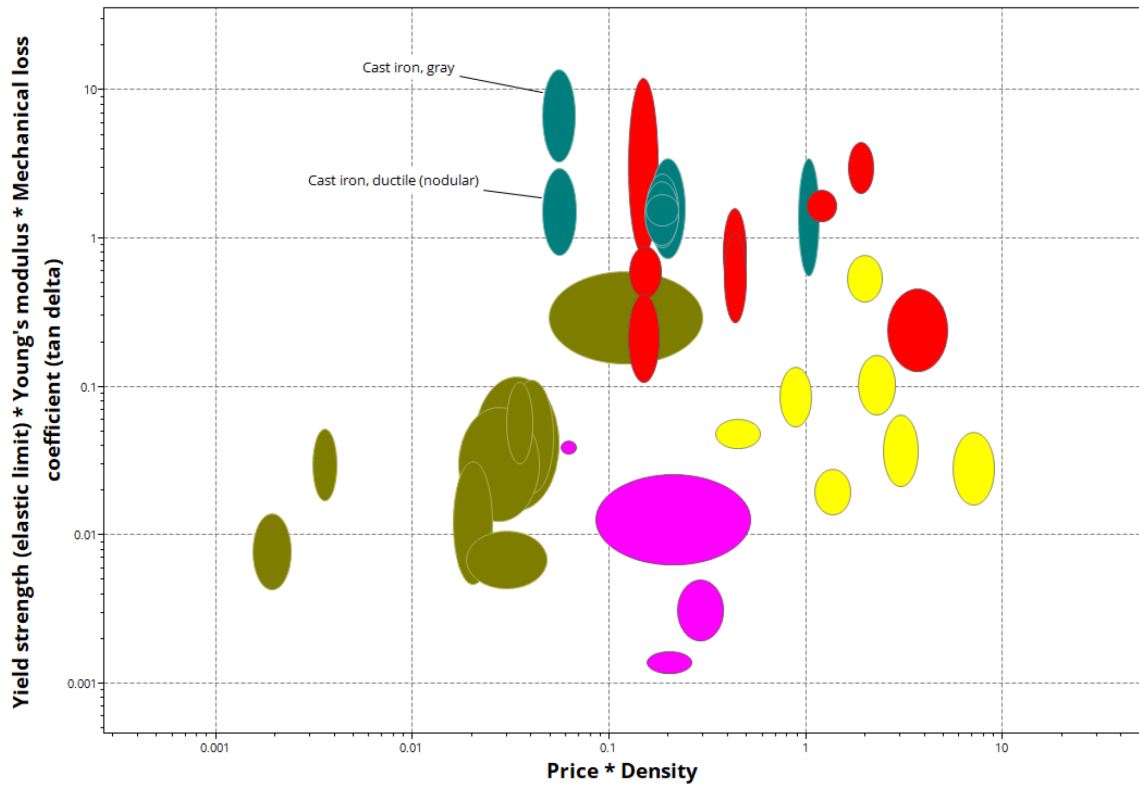


The shaft will be manufactured through purchasing stock parts 15mm (0.591 inches) in outer diameter for the shaft, then later cut to length and machines using the CNC lathe.

The rotor will be manufactured by purchasing stomach parts of 3.1 inches outer diameter, cut to length and machined using a CNC lathe.

Housing

For housing considering the mechanical loss coefficient, for good damping:



Ductile cast iron looks like the best material for these criteria, since it has good mechanical loss coefficient and is higher than gray cast iron for yield strength and has good price and weight.

For cost efficiency in terms of material stock, time and labor cost we chose to go with casting for the housing, which is then machined using a CNC mill, which will take care of any specific geometries that need tighter clearances and tolerances.

Inserts

The material for inserts was chosen based on the friction coefficient.

To machine the inserts we will first injection mold them and then machine to account for the draft angles that were required. The inner surface will be machined to final dimensions after press-fitting for accuracy.

Front Plate

Ductile Cast Iron (65-45-12)

Machined from stock using CNC Mill to appropriate specifications.

Bearing Caps

Made from Aluminum 6061-T6 for easy manufacturability and strength.

To machine we will buy stock and CNC Mill to appropriate specifications.

Clearances and Tolerances

Thermal Expansion Calculations

In order to calculate the thermal expansion we'll use the formula:

$$\Delta L = \alpha L_0 (T_{final} - T_{initial})$$

where,

ΔL - change in length/diameter

α - the coefficient of linear thermal expansion

$T_{initial}$ - 68F, or room temperature

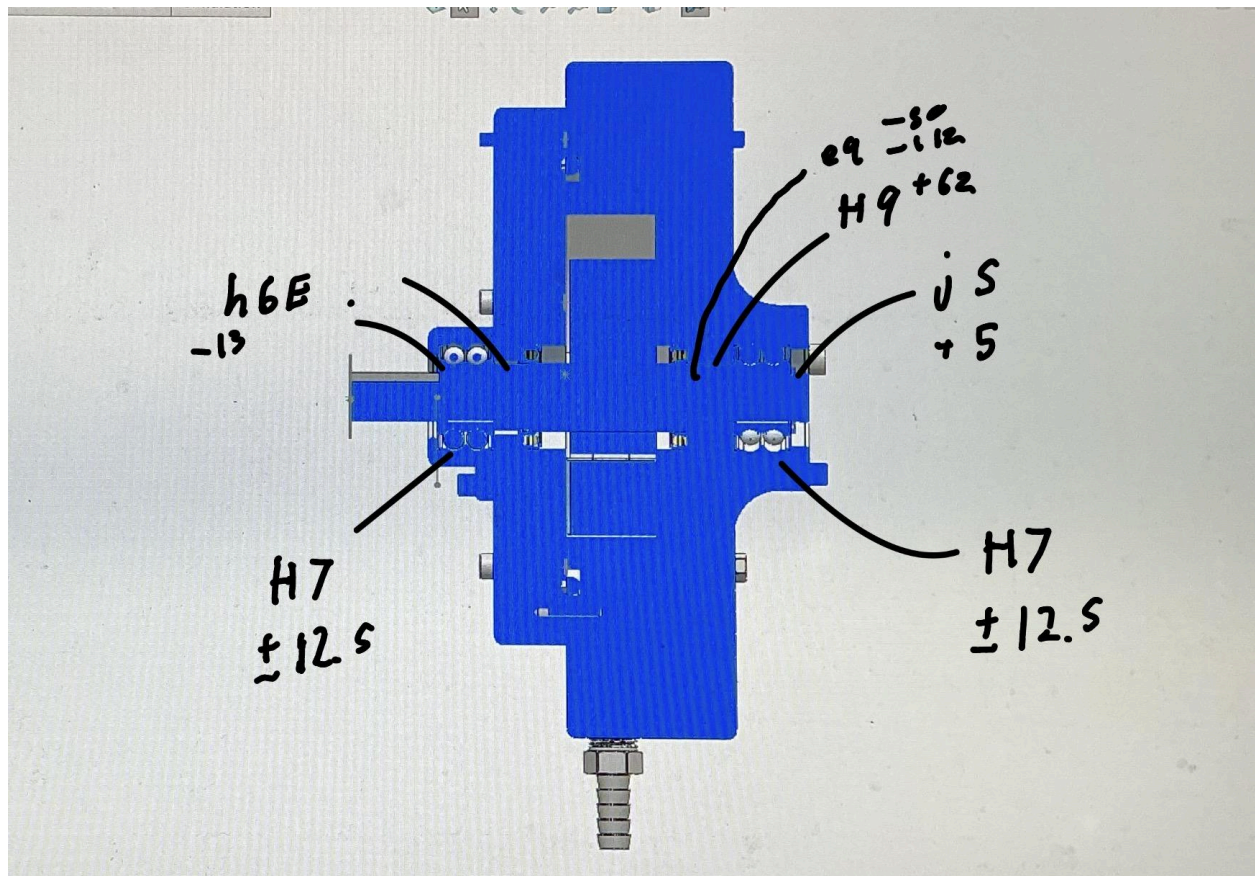
From problem statement we have two cases:

- Cold: $T_{final} = -40F$ (couldn't find thermal expansion coefficients for that range, most were ~40F+)
- Hot: $T_{final} = 140F$

Table 1: Thermal Expansion Coefficients for different parts of the pump and their materials, as well as their dimensions:

	Outer Diameter (in)	Inner Diameter (in)	Width (in)	Length (in)	Height (in)	Material	Thermal Expansion Coefficient (1/°F)	Link
Rotor	3.1	0.75	0.95	-	-	Alloy steel (AISI 4140)	0.00000678	Link
Housing	-	3.86	1.13	-	-	Cast Iron, ductile (65-45-12)	0.0000064	Link
Peek Insert	3.86	3.6	1.13	-	-	Peek (Vitrex 450G PEEK)	0.0000306	Link
Vanes	-	-	1	0.25	0.85	Hardened tool steel (AISI D3)	0.00000594	Link
Shaft	0.8	0.59	-	-	-	Alloy steel (AISI 4140)	0.00000678	Link

Total Thermal Expansion using Thot-Tcold					
Change in	Outer Diameter (in)	Inner Diameter (in)	Width (in)	Length (in)	Height (in)
Rotor	0.00378324	0.00097632	0.00115938	-	-
Housing	-	0.00444672	0.00130176	-	-
Peek Insert	0.02126088	0.0198288	0.00622404	-	0.0006885
Vanes	-	-	0.0010692	0.00013365	0.00090882
Shaft	0.00097632	0.000720036	0.004722948	-	-



Shaft

Seal Requirements

<https://www.balseal.com/wp-content/uploads/2024/03/Bal-Seal-Spring-Energized-Seal-Solutions-for-Rotary-Applications-DM-5-1.pdf>

Radial Clearance "E" (Inches) @70 °F (21 °C)					
Cross Section Code	Pressure psi (bar)				
	150 (10)	300 (21)	500 (34)	1000 (69)	1500 (103)
2	0.001 (0.03)	0.001 (0.03)	0.001 (0.03)	0.001 (0.03)	0.001 (0.03)
1	0.002 (0.05)	0.002 (0.05)	0.002 (0.05)	0.002 (0.05)	0.002 (0.05)
0	0.004 (0.10)	0.003 (0.08)	0.003 (0.08)	0.003 (0.08)	0.003 (0.08)
4	0.005 (0.13)	0.004 (0.10)	0.004 (0.10)	0.003 (0.08)	0.003 (0.08)
5	0.006 (0.15)	0.005 (0.13)	0.005 (0.13)	0.004 (0.10)	0.004 (0.10)
6	0.007 (0.18)	0.006 (0.15)	0.005 (0.13)	0.0045 (0.11)	0.0045 (0.11)
7	0.008 (0.20)	0.006 (0.15)	0.006 (0.15)	0.0055 (0.14)	0.0055 (0.14)
8	0.010 (0.25)	0.008 (0.20)	0.007 (0.18)	0.0065 (0.17)	0.0065 (0.17)
9	0.012 (0.30)	0.010 (0.25)	0.008 (0.20)	0.0075 (0.19)	0.0075 (0.19)

Note: For cross section dimensions of various seal designs, refer to pages 16, 17, 18, and 19.

Recommended Shaft and Housing Tolerances*

Shaft Diameter Range In. (mm)	Recommended Tolerance			
	Shaft Dimension In. (mm)		Bore Dimension In. (mm)	
	Min.	Max.	Min.	Max.
0.060 to 0.999 (1.52 to 24.39)	-0.0005 (0.013)	+0.0000 (0.00)	-0.0000 (0.00)	+0.0005 (0.013)
1.000 to 1.999 (25.40 to 50.79)	-0.001 (-0.025)	+0.000 (0.00)	-0.000 (0.00)	+0.001 (0.025)
2.000 to 3.499 (50.80 to 88.99)	-0.0015 (-0.038)	+0.000 (0.00)	-0.000 (0.00)	+0.0015 (0.038)
3.500 to 5.999 (88.90 to 152.37)	-0.002 (-0.051)	+0.000 (0.00)	-0.000 (0.00)	+0.002 (0.051)
6.000 to 14.999 (152.40 to 380.97)	-0.003 (-0.076)	+0.000 (0.00)	-0.000 (0.00)	+0.003 (0.076)
15.000 to 19.999 (381.00 to 507.97)	-0.004 (-0.102)	+0.000 (0.00)	-0.000 (0.00)	+0.004 (0.102)
20.000 to 34.000 (508.00 to 863.00)	-0.005 (-0.127)	+0.000 (0.00)	-0.000 (0.00)	+0.005 (0.127)

Note: For "A" and "B" dimensions refer to pages 16, 17, 18 and 19.

For shaft and housing diameters above 34 in. (863 mm), please contact us.

*We manufacture seals to accommodate larger tolerances. Please contact us for more information.

Bearings

Need G6 fit for shaft that is inside the bearing at front of the housing that is a float

- Shaft has to be 12.5 micro meters less than the nominal inner diameter of the bearing, and with a tolerance of 12.5 +/-

For the bearing at the back of the housing want a +12.5, tolerance at 12.5 micro meters for the tight fit between bearing and shaft

- Shaft has to be 12.5 micro meters bigger than the inner diameter of the bearing and then have +/- 12.5 micro meters tolerance

<https://www.skf.com/group/products/rolling-bearings/ball-bearings/angular-contact-ball-bearings/double-row-angular-contact-ball-bearings/productid-3202%20A-2RS1TN9%2FMT33>

Tolerances and clearances

GENERAL BEARING SPECIFICATIONS

- Tolerances: Normal, P6, P5
- Internal clearance: table, drawing no

BEARING INTERFACES

- Seat tolerances for standard conditions
- Tolerances and resultant fit

Shrink Fit between Rotor and Shaft

Area of Contact (in²)

$2\pi r \times width$

r - outer radius of shaft

w - width of rotor

Torque (lbf*in)	Area of contact (in ²)	Force (lbf)
1707.8568	2.387610417	4269.642

D2/D1	Torsional Tensile Stress (psi)	C - coefficient based on diameter ratio	Young's modulus (psi)	Interference between rotor and shaft(in)
3.875	1788.249025	0.45	29700000	0.000087305 08708

Total expansion in Diameter	Temperature for shrinkage fit (F)
0.001	147.4926254

Tolerance for diameters per the ANSI/ASME B4.1-1967 (R2020) standard for nominal sizes of diameter (0.71-1.19) using turning tolerance grade 7	0.0008
--	--------

Housing and Peek Insert

Clearances and tolerances calculated for a press fit per Machinist Handbook.

Fastener Selection

1. Bolt Requirements:

In order to calculate what bolts we need we use the equation below:

$$\text{Require bolt area} = \frac{F_{\text{applied}}}{\sigma_{\text{yield}}} \times FOS$$

$$A = \frac{9250N}{800MPa} \times 3$$

$$A = 34.69mm^2$$

Converting to diameter

$$A = \pi \left(\frac{d}{2}\right)^2 \times \text{number of screws}$$

$$34.69mm^2 = \pi \left(\frac{d}{2}\right)^2 \times 5$$

$$d = 2.972mm$$

Therefore we need 5xM3 grade 8.8 screws.

Fastener Torques

Torques for each of the fasteners were calculated using the following formula:

$$T = KdP$$

Where T is torque, K is a constant, d is the nominal diameter of the bolt, and P is the preload.

Fastener	Torque
M5	12.8 ft-lb
M4	6.5 ft-lb

Surface Finishes

For CNC Mills: 125 microinches

For CNC Lathe: 16 microinches Ra

For Casting: Surface tolerance for casting 250 to 500 microinches

For Injection Molding: 150 microinches

CNC Mill reasonable tolerances: ± 0.005 in, ± 0.002 in

CNC Lathe reasonable tolerances: ± 0.003 in, ± 0.001 – 0.002 in

Bill of Materials

Part	Part Number	Quantity	Material	Manufacturing Strategy	Manufacturer P/N	Price	Total	Link
Fitting for Steel Tubing	3-001	2	Zinc-Plated Steel	OTS	50695K61	\$2.67	\$5.34	Link
Contact ball bearing	3-002	2	-	OTS	3202 A-2RS1TN9 /MT33	\$141.67	\$283.34	Link
Seals	3-003	2	-	OTS	1306.708713		0	Link
Snap Ring	3-004	1	-	OTS		0.1282	0.1282	Link
O-ring	3-005	1	Nitrile (Parker-1500)	OTS	N1500-752-349		0	
Dowels	3-006	2	-	OTS		0.4648	0.9296	Link
Housing Fasteners	3-007	5	-	OTS	91290A193	0.94	4.7	Link
Rotor	1-002	1	Alloy steel	CNC Lathe			0	

			(AISI 4140)	and Mill				
Pump Housing	1-001	1	Cast Iron, ductile (65-45-12)	Post machining on CNC Mill			0	
Peek Insert	1-004	1	Peek (Viktrex 450G PEEK)	injection molding			0	
Vane part A	2-006	1	Hardened tool steel (AISI D3), hardened to 46-54 Rockwell C	powder metal manufacturing				
Vanes	1-006	4	Hardened tool steel (AISI D3), hardened to 46-54 Rockwell C	CNC Mill/drill			0	
Peek Front Insert	1-005	1	Peek (Viktrex 450G PEEK)	injection molding			0	
Shaft	1-003	1	Alloy steel (AISI 4140)	CNC lathe			0	

Front Plate	1-009	1	Cast Iron, ductile (65-45-12)				0	
Back Bearing Cap	1-007	1	Aluminum 6061-T6				0	
Front Bearing Housing	1-008	1	Aluminum 6061-T6	CNC Mill			0	
Bearing Housing Bolts	3-008	6	-	OTS		0.1423	0.8538	Link
Araldite AV 138/Hardener HV 998	3-009	1	-	OTS			0	
Thread Locker	3-010	1	-	OTS			0	
Housing Casting	2-001	1	Cast Iron, ductile (65-45-12)	Casting			0	
Housing Insert Blank	2-004	1	Peek (Viktrex 450G PEEK)	Injection Molding			0	
Front Insert Blank	2-005	1	Peek (Viktrex 450G PEEK)	Injection Molding			0	