Duke University

ME421 – Fall 2025

Pressure Vessel Final Report

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Contents

1 Original Design

Our current pressure vessel design meets the requirements to safely contain 50–60 cc of liquid at 750 psig, incorporating at least one O-ring seal and a threaded connection. The vessel assembly, shown in the attached CAD drawing, consists of a cylindrical body with a removable end cap secured by bolts and sealed with an axial O-ring.

The vessel is composed of two main components:

- *Main Body:* A cylindrical chamber (internal diameter 63 mm, outer diameter 73 mm, length 38 mm) designed to hold the required internal volume. The wall thickness provides adequate safety margin under 750 psig internal pressure, verified analytically in subsequent sections.
- *End Cap:* A circular plate that mates with the main body through a flat flange joint. The cap sits against an axial O-ring groove machined into the body, ensuring a reliable seal under pressure.

A single $\frac{1}{4}$-in NPT port is threaded into the end cap to interface with the provided Swagelok plug connector (B-400-1-4). This port functions as both the fluid inlet and outlet. The tapped hole is through-drilled and threaded according to NPT standards to ensure a tight, leak-free connection.

The cap is held by six equally spaced M5 $\times$ 0.8 bolts arranged on a 90 mm bolt circle, providing uniform preload across the O-ring interface. The O-ring groove (ID 72 mm) accommodates the O-ring. The groove geometry and compression ratio are dimensioned as per in-class specifications.

The vessel is designed for aluminum 6061-T6 to balance strength, corrosion resistance, and ease of machining, consistent with course recommendations to avoid stainless steels for initial prototypes.

Please see the Solidworks Drawing appended to the end of this document.

2 Final Design - Changes Made

Following the initial prototype, several design modifications were made to simplify manufacturing and improve reliability under pressure. The revised pressure vessel maintains the same core function, to safely contain 50–60 cc of liquid at 750 psig with an O-ring seal and threaded connection, but has a more streamlined and robust construction.

The most significant change was the removal of the flange interface between the body and the lid. The flange had introduced unnecessary machining steps and potential alignment errors, so the updated design uses constant outer diameter of the main body instead. This adjustment makes the assembly easier to fabricate and assemble while maintaining a secure, leak-free seal. The inner diameter and volume calculations did not change.

The lid was also redesigned to be thicker, utilizing excess material ordered, to improve its strength under load. The increased thickness provides greater resistance to deflection and minimizes the likelihood of leakage at elevated pressures. The lid still contains a single $\frac{1}{4}$-in NPT threaded port for the Swagelok plug connector (B-400-1-4), which continues to function as both the inlet and outlet for the fluid.

To accommodate the thicker lid, the fasteners were upgraded to 1-inch-long #10-32 screws, ensuring sufficient thread engagement into the base. The longer screws distribute clamping forces more evenly across the lid and the main chamber, and provide additional mechanical security.

The outer diameter of the vessel is now constant and matches that of the lid (4.00 inches), simplifying the overall geometry and improving the vessel's aesthetics and symmetry. The internal cavity volume and wall thickness are still designed to safely handle the target operating pressure.

As with the original prototype, the vessel is machined from 6061-T6 aluminum, chosen for its combination of strength, corrosion resistance, and machinability. The O-ring seal is placed in an axial groove within the body to ensure consistent sealing performance under pressure.

Overall, the updated design is simpler, stronger, and easier to manufacture, while preserving the functionality and safety requirements established in the original design.

3 O-Ring Design

- *Type and fit:* Static axial flange/face seal rectangular gland with zero extrusion gap, appropriate for non-moving joints and defined by AS568 sizing practice. The seal relies on axial squeeze with slight installed stretch to seat the ring.

- *Material:* Buna-N (NBR) 70A. NBR is the standard O-ring elastomer for static service with oils/water, offering good mechanical strength and temperature capability up to 130 °C, well within our lab conditions.
- *Selected size:* ID 70 mm, CS 1.5 mm (McMaster 9262K534). For a face seal, this gives low ID stretch (target $\approx$ 0–5%) and allows axial compression when clamped.

4 O-Ring Groove

Dimensions

O-ring	
ID	70 mm
cross-section (CS) (diameter d)	1.5 mm
OD	73 mm
Groove	
Gland (groove) ID	72 mm
Gland width w_g	2.24 mm
Gland depth	1.125 mm

Analysis

1) Stretch

We compute the relative stretch as

$$\text{stretch} = \frac{\text{Gland ID} - \text{O-ring ID}}{\text{O-ring ID}} = \frac{72 - 70}{70} = \frac{3}{70} \approx 0.028571 \approx 2.86\%.$$

Here the calculation uses the gland ID value 72 mm minus O-ring ID 70 mm. This percentage is within the desired range of 1-5%.

2) Gland depth (compression check)

Using the O-ring cross-section (CS = 1.5 mm) and gland depth 1.125 mm:

$$\text{compression ratio} = \frac{\text{CS} - \text{gland depth}}{\text{CS}} = \frac{1.5 - 1.125}{1.5} = \frac{0.375}{1.5} = 0.25 = 25\%.$$

The ideal gland depth provides a compression ratio of about 25%.

3) Check: CS > gland depth

$$\text{CS} = 1.5\,\text{mm} > \text{gland depth} = 1.125\,\text{mm} \quad \text{(OK)}$$

4) Gland width and fill (cross-section area)

O-ring cross-sectional area Assuming a circular cross section of diameter $d = 1.5$ mm,

$$A_0 = \pi \left(\frac{d}{2}\right)^2 = \pi \left(\frac{1.5}{2}\right)^2 = \pi(0.75)^2 \approx 1.7671 \text{ mm}^2.$$

Gland cross-sectional area Gland area (rectangular approximation) is

$$A_g = (\text{gland depth}) \cdot w_g = 1.125 \times 2.24 \approx 2.52 \text{ mm}^2.$$

Gland fill fraction

$$\text{fill} = \frac{A_0}{A_g} = \frac{1.7671}{2.52} \approx 0.7004 \approx 70.0\%.$$

This lies within a common target range for gland fill ($\sim 65\% - 75\%$), so the selection looks reasonable.

5 Fastener Selection

Selection of appropriate fasteners, including the number of fasteners. Justify your choice with analysis.

Based on the initial CAD design:

$$D_b = 45.83 \text{ mm}, \quad d = 4.826 \text{ mm}$$

Let $N = 6$;

$$\frac{\pi D_b}{Nd} = \frac{\pi(45.83)}{6(4.826)} = 4.97 \implies [3 < 4.97 < 6]$$

Therefore, chosen number of bolts and bolt spacing seems sufficient.

6 bolts

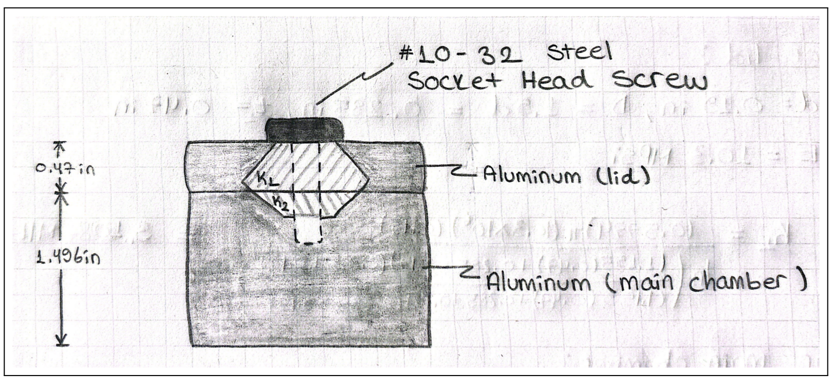


Figure 1: Steel socket head screws fastening an aluminum lid and aluminum main chamber.

$t_1 = 0.47$ in, $\quad t_2 = 1.496$ in, $\quad d = 0.19$ in

$h = t_1$, since no washer is present in the final design.

From Table 8.7 (for $t_2 > d$):

$$\ell = h + d/2 = 0.47 + (0.19/2) = 0.565 \text{ in}$$

$$L > h + 1.5d = 0.47 + (1.5)(0.19) = 0.755 \text{ in}$$

Based on the standard dimensions from McMaster, and leaving some room for error, we can decide on:

$$\boxed{L = 1 \text{ in}}$$

Bolt and Member Stiffness

Bolt stiffness k_b

$$k_b = \frac{A_d\, A_t\, E}{A_d\, \ell_t + A_t\, \ell_d}$$

$$A_d = \frac{\pi(0.19)^2}{4} = 0.0283\,\text{in}^2$$

From Table 8.2: $A_t = 0.0175\,\text{in}^2$.

To account for any potential changes in the dimensions of the lid or the main chamber during machining, a fully threaded bolt was utilized in the final pressure vessel. Thus:

$$\ell_d = 0 \text{ in}, \qquad \ell_t = \ell - \ell_d = 0.565 \text{ in}$$

From Table 8.8 (steel bolt): $E = 30$ Mpsi

$$K_b = \frac{(0.0175)(30 \times 10^6)}{0.565} = 0.93 \text{ Mlb/in}$$

Member (Lid) stiffness k_1

$$k = \frac{0.5774\,\pi\, E\, d}{\ln\left(\dfrac{(1.155\,t + D - d)(D + d)}{(1.155\,t + D + d)(D - d)}\right)}$$

Given: $d = 0.19\,\text{in}$, $t = 0.47\,\text{in}$, $D = 1.5d = 1.5(0.19) = 0.285\,\text{in}$.

From Table 8.8 (aluminum): $E = 10.3MPsi$

$$k = \frac{0.5774\,\pi\,(10.3 \times 10^6)(0.19)}{\ln\left(\dfrac{(1.155(0.47) + 0.285 - 0.19)(0.285 + 0.19)}{(1.155(0.47) + 0.285 + 0.19)(0.285 - 0.19)}\right)} = 3.108\,\text{Mlb/in.}$$

Member (Main Chamber) stiffness k_2

The depth of the effective load transfer can be approximated as:

$$t = \frac{d}{2} = \frac{0.19}{2} = 0.095\,\text{in}$$

Given: $d = 0.19\,\text{in}$, $t = 0.095\,\text{in}$, $D = 1.5d = 1.5(0.19) = 0.285\,\text{in}$.

From Table 8.8 (aluminum): $E = 10.3MPsi$

$$k = \frac{0.5774\,\pi\,(10.3 \times 10^6)(0.19)}{\ln\left(\dfrac{(1.155(0.095) + 0.285 - 0.19)(0.285 + 0.19)}{(1.155(0.095) + 0.285 + 0.19)(0.285 - 0.19)}\right)} = 6.34\,\text{Mlb/in.}$$

Member stiffness k_m

For two members in series:

$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} \quad \Rightarrow \quad k_m = \frac{k_1 k_2}{k_1 + k_2}.$$

$$k_m = \frac{(3.108)(6.34)}{(3.108) + (6.34)} = 2.085\,\text{Mlb/in}$$

6 Factors of Safety and Pre-Load Calculations

Calculations of factors of safety with respect to relevant joint failures. Additionally, calculate the necessary pre-load and torque to achieve that pre-load.

Joint constant

$$C = \frac{k_b}{k_b + k_m} = \frac{0.93}{0.93 + 2.085} = \boxed{0.308}$$

External load per bolt

Given the internal pressure

$$P = 750 \text{ psig},$$

Diameter of the O-Ring: 0.073 m = 2.874 in

projected area

$$A = \frac{\pi(2.874)^2}{4} = 6.49\,\text{in}^2,$$

and total separating force

$$F_{\text{ext}} = PA = (750)(6.49) = 4867.5\,\text{lbf}.$$

For $N = 6$ bolts,

$$F = \frac{F_{\text{ext}}}{6} = \boxed{811\,\text{lbf per bolt}}.$$

Maximum allowable bolt preload

$$F_{\max} = S_p A_t$$

From Table 8.2: $A_t = 0.0175\,\text{in}^2$.

Although not present in the textbook tables, Minimum Proof Strength of the used bolts was determined as 140 ksi based on data from McMASTER:

$$F_{\max} = (0.0175)(140 \times 10^3) = \boxed{2450\,\text{lbf}}.$$

(Maximum bolt preload without exceeding proof strength.)

Minimum preload to avoid separation

$$F_i = (1 - C)\,P_0, \qquad P_0 = F = 811\,\text{lbf},$$

$$F_i = (1 - 0.308)(811) = \boxed{561.212\,\text{lbf}}.$$

(Minimum bolt preload to avoid joint separation.)

Practical choice for assembly

For safety and practicality, use

$$F_i \approx 0.75\,F_{\max} = (0.75)(2450) = \boxed{1837.5\,\text{lbf}}.$$

Tightening torque

From Table 8.15: $K = 0.20$ (zinc–plated bolts).

$$T = K\,F_i\,d = (0.20)\big(1837.5\big)(0.19) = \boxed{69.825\ \text{lbf}\cdot\text{in}}.$$

Yielding factor of safety

$$n_y = \frac{F_p}{F_i + CP} = \frac{S_p A_t}{F_i + CP}.$$

With $S_p = 140$ ksi, $A_t = 0.0175$ in^2, $F_i = 1837.5$ lbf, $C = 0.308$, and $P = 811$ lbf:

$$n_y = \frac{(140 \times 10^3)(0.0175)}{(1837.5) + (0.308)(811)} = \boxed{1.173}\ (> 1).$$

The yielding factor of safety of 1.173 implies that the bolt is safely below its proof strength when both preload and the pressure loads are applied. Although this factor of safety is lower than typical design values, it is still greater than 1. Therefore, we can conclude that the bolt will not plastically deform under the determined operating pressure.

Overload factor of safety

$$n_{\text{load}} = \frac{S_p - F_i/A_t}{(CP)/A_t}.$$

$$n_{\text{load}} = \frac{(140 \times 10^3) - \dfrac{1837.5}{0.0175}}{\dfrac{(0.308)(811)}{0.0175}} = \boxed{2.45}\ (> 1).$$

The overload factor of safety of 2.45 implies that the bolt can withstand approximately two and a half times the external separating load before reaching its proof strength. Thus, we can conclude that even if the pressure vessel experiences an unexpected increase in pressure, the bolt would still remain within the elastic range.

Joint separation factor of safety

$$n_j = \frac{F_i}{P(1 - C)} = \frac{1837.5}{(811)(1 - 0.308)} = \boxed{3.27}\ (\gg 1).$$

The joint separation factor of safety of 3.27 implies that the initial bolt preload is more than three times larger than the external load necessary to overcome it. This means the clamping force is sufficient to keep the lid and base properly in contact as desired, preventing joint separation and any leakage.

7 Finite Element Analysis (FEA)

Stress results

Figure 2 shows the von Mises stress distribution. The absolute maximum stress occurs locally at the bolt–hole contact region, $\sigma_{\text{max}} \approx 4.67 \times 10^8\,\text{N/m}^2$, which is above the yield strength of the aluminum ($\sigma_y = 2.75 \times 10^8\,\text{N/m}^2$). However this peak is confined to a very small region at the hole edge, caused by a mesh singularity, and thus it does not control global failure.

Away from the hole edges a probe was put to get a more representative stress, coming to $\sigma_{\text{mem}} \approx 2.38 \times 10^8\,\text{N/m}^2$, giving a material factor of safety

$$n_y \approx \frac{\sigma_y}{\sigma_{\text{mem}}} \approx \frac{2.75 \times 10^8}{2.38 \times 10^8} \approx 1.2,$$

which is consistent with the bolt yield factor of safety $n_y = 1.173$ obtained from the hand calculations.

Displacement and sealing

Figure 3 shows the deformation of the lid under pressure. The maximum total displacement at the flange is $\delta_{\text{max}} \approx 2.8 \times 10^{-2}$ mm. This is an order of magnitude smaller than the designed O–ring compression $\Delta_c = 0.375$ mm. Thus, even under full internal pressure, the predicted flange opening in the seal region is much less than the initial O–ring squeeze, which supports the separation factor of safety $n_j = 3.27$ obtained from the joint analysis.

Overall these FEA results support the hand calculations for the O–ring groove design and bolted joint sizing.

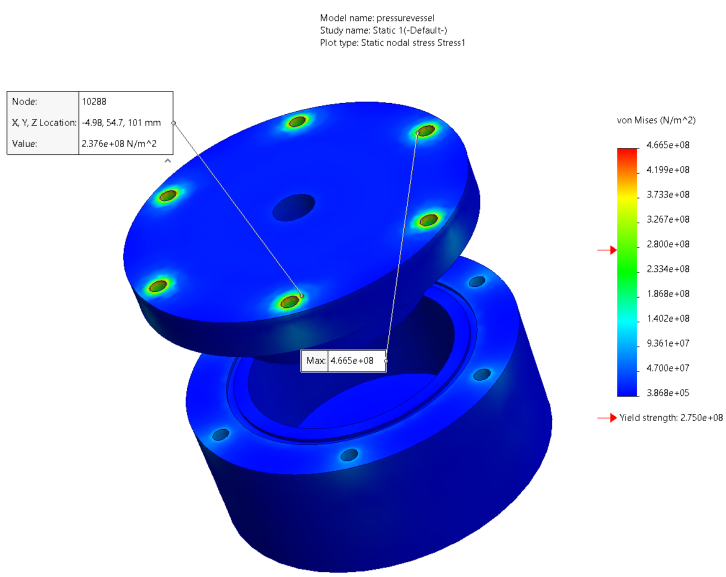


Figure 2: FEA showing Von Mies stress, max stress location, and a probe

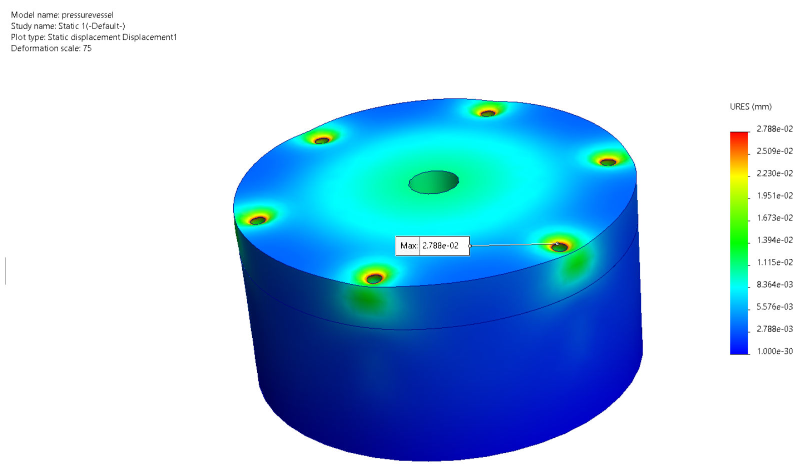


Figure 3: FEA showing displacement and max displacement location

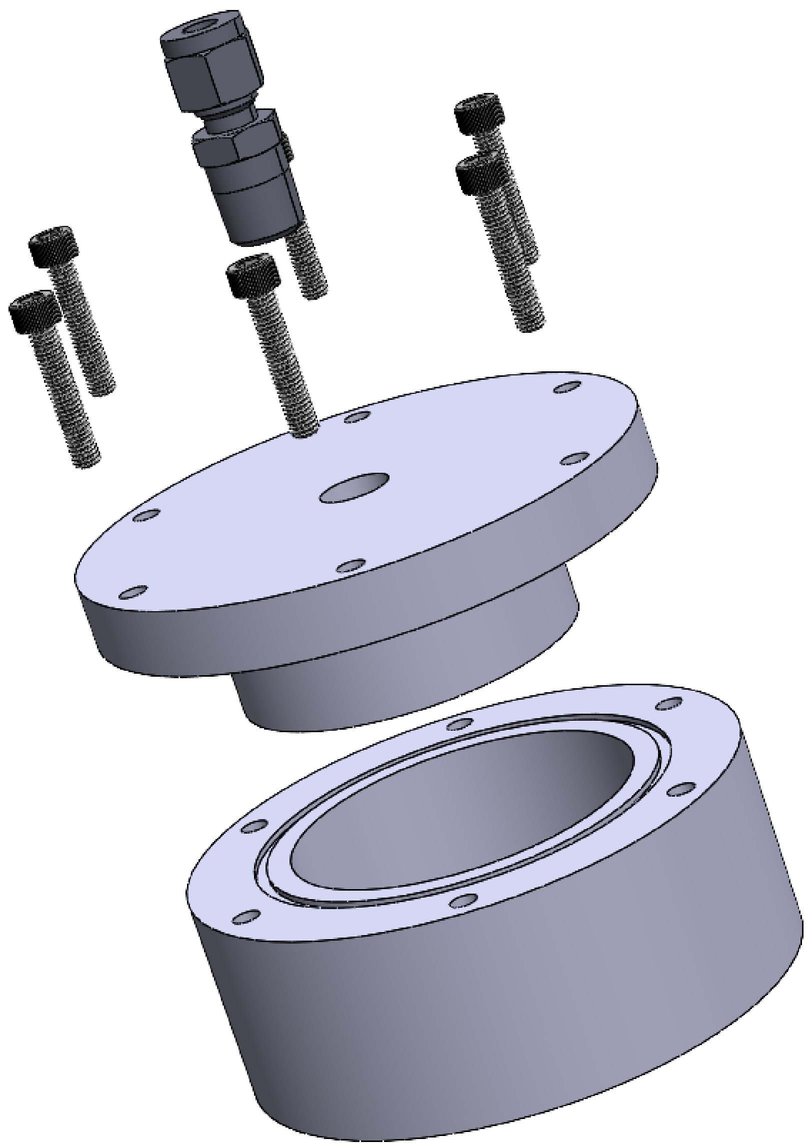
8 Bill of Materials

Material	Part No.	Description	Unit Price	Quantity
Multipurpose 6061 Aluminum Rod	McMaster 8974K97	Aluminum Rod with 4" Diameter for body and lid of vessel	$82.73	1
Oil-Resistant Buna-N O-Ring	McMaster 9262K534	O-Ring 1.5 mm Wide, 70 mm ID	$0.46	1
Zinc-Plated Alloy Steel Socket Head Screw	McMaster 90128A947	Socket Head Screw 10-32 Thread Size, 1" Long	Garage Lab $0.23	6
Brass Swagelok Tube Fitting	Swagelok B-4-TA-1-4AN	Male Tube Adapter, 1/4 in. Tube OD x 1/4 in. AN Tube Flare	Provided $12.50	1
Brass Swagelok Tube Fitting	Swagelok B-400-1-4	Male Connector, 1/4 in. Tube OD x 1/4 in. Male NPT	Provided $14.30	1
-	-	-	**$111.40***	-

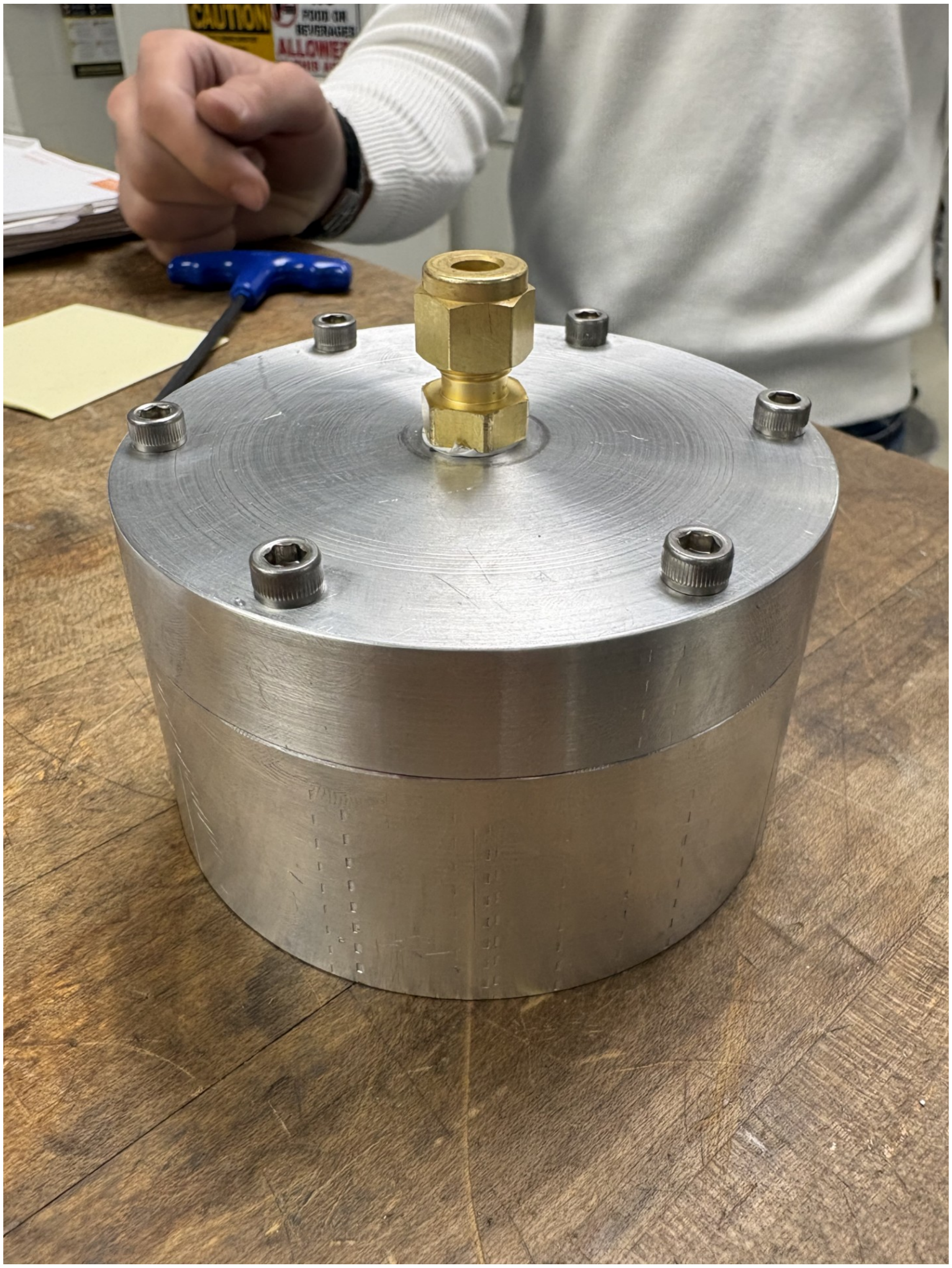
**the total expense is below our $100 budget when accounting for the provided parts*

Appendix

A Assembly

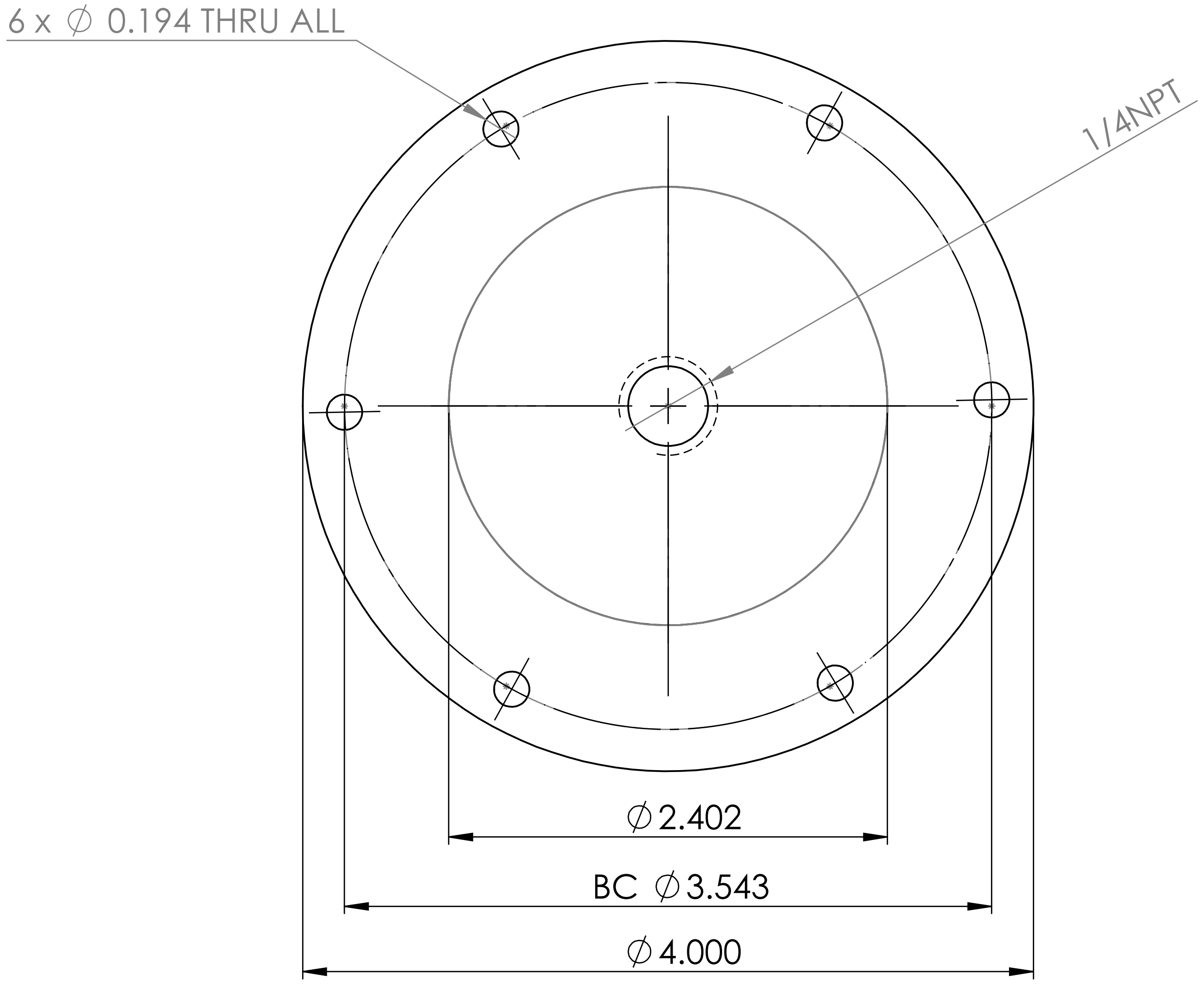


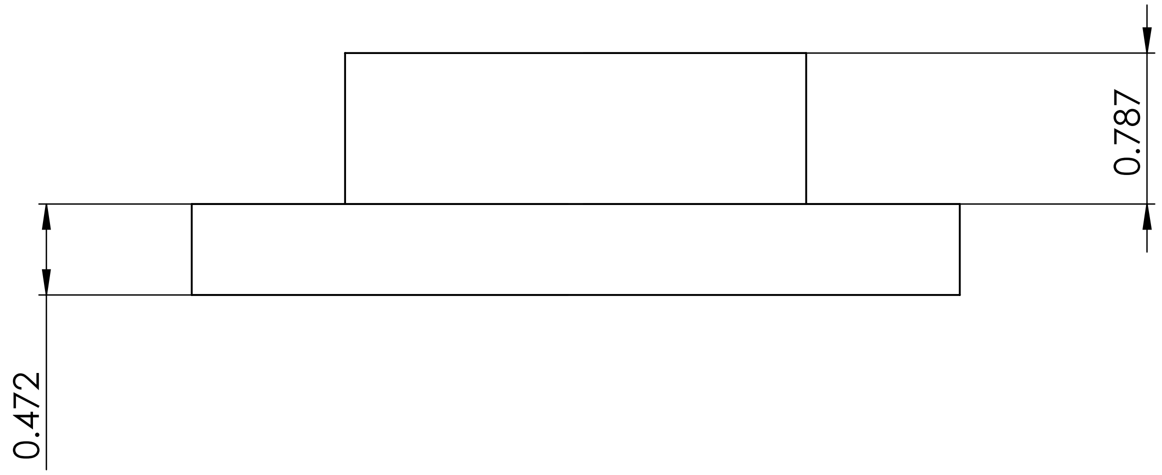
B Image of Completed Pressure Vessel

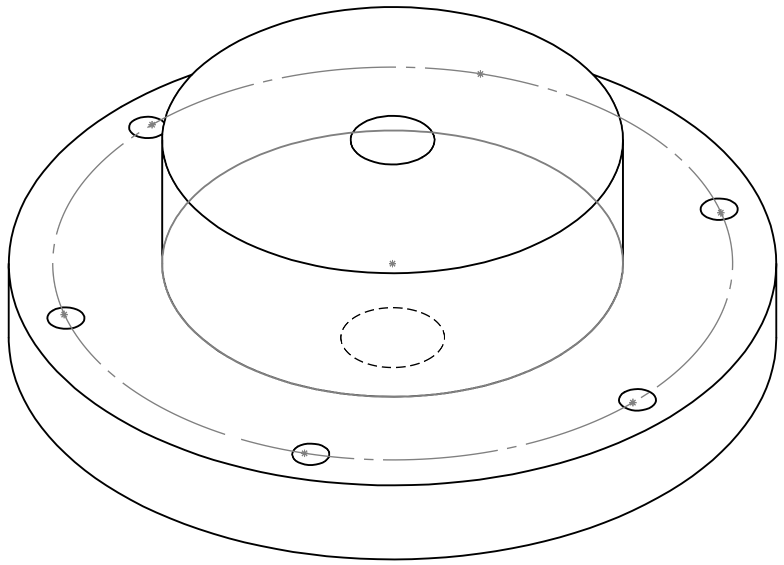


C Drawings

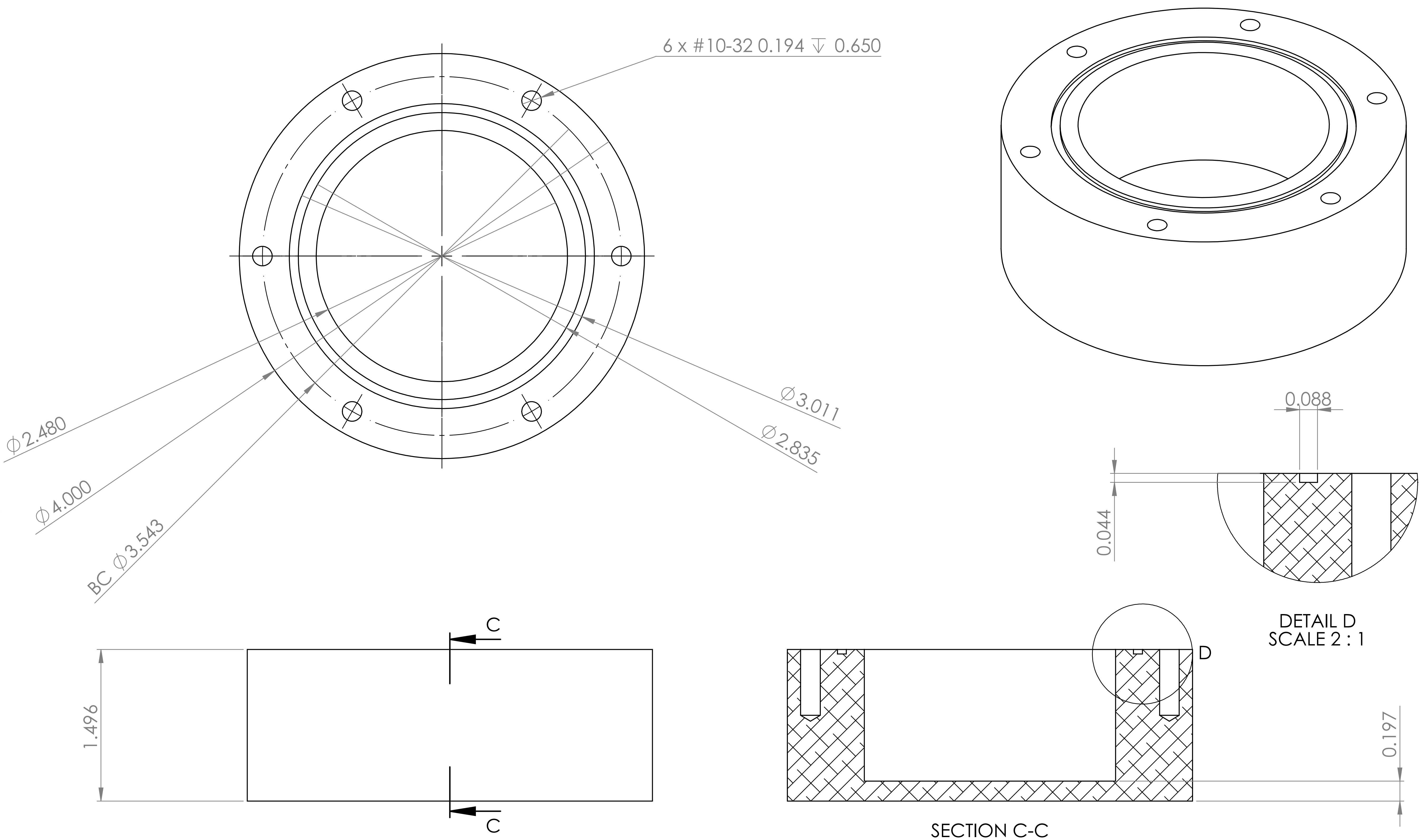
All machined parts were produced using standard shop practices. Specific dimensional tolerances are not required for this design, as the analysis and O-ring sealing performance are not sensitive to small manufacturing variation. Nominal dimensions shown are sufficient.







UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:			DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
	NAME	SIGNATURE	DATE		TITLE:	
DRAWN						
CHK'D						
APPV'D						
MFG						
Q.A				MATERIAL:	DWG NO. lid	A3
				WEIGHT:	SCALE:1:1	SHEET 1 OF 1



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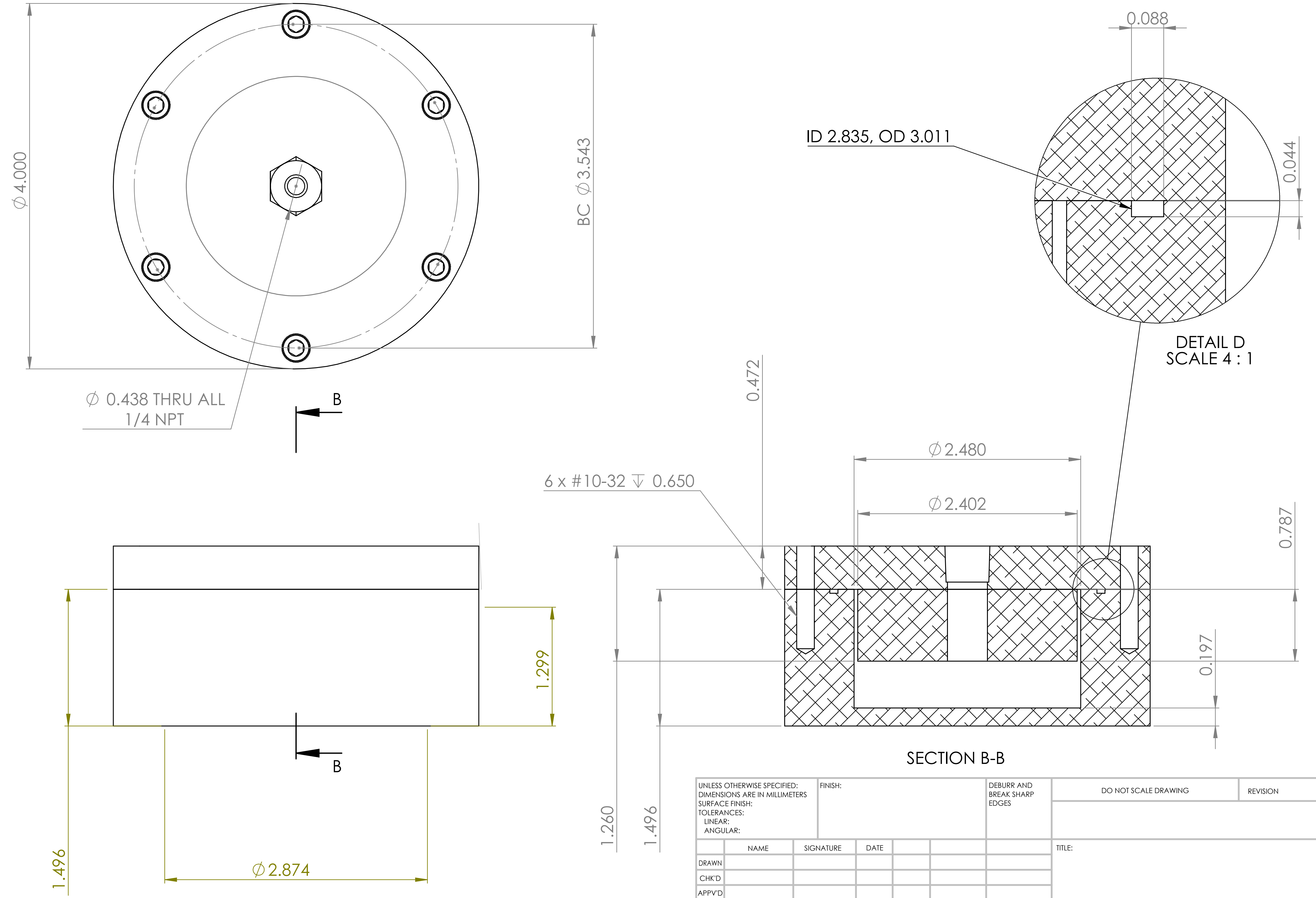
	NAME	SIGNATURE	DATE	TITLE:	
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A			MATERIAL:	DWG NO. main chamber	A3
			WEIGHT:	SCALE:1:1	SHEET 1 OF 1



SECTION B-B

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING / REVISION

	NAME	SIGNATURE	DATE	
DRAWN				
CHK'D				
APPV'D				
MFG				
Q.A				MATERIAL:
				WEIGHT:

TITLE:

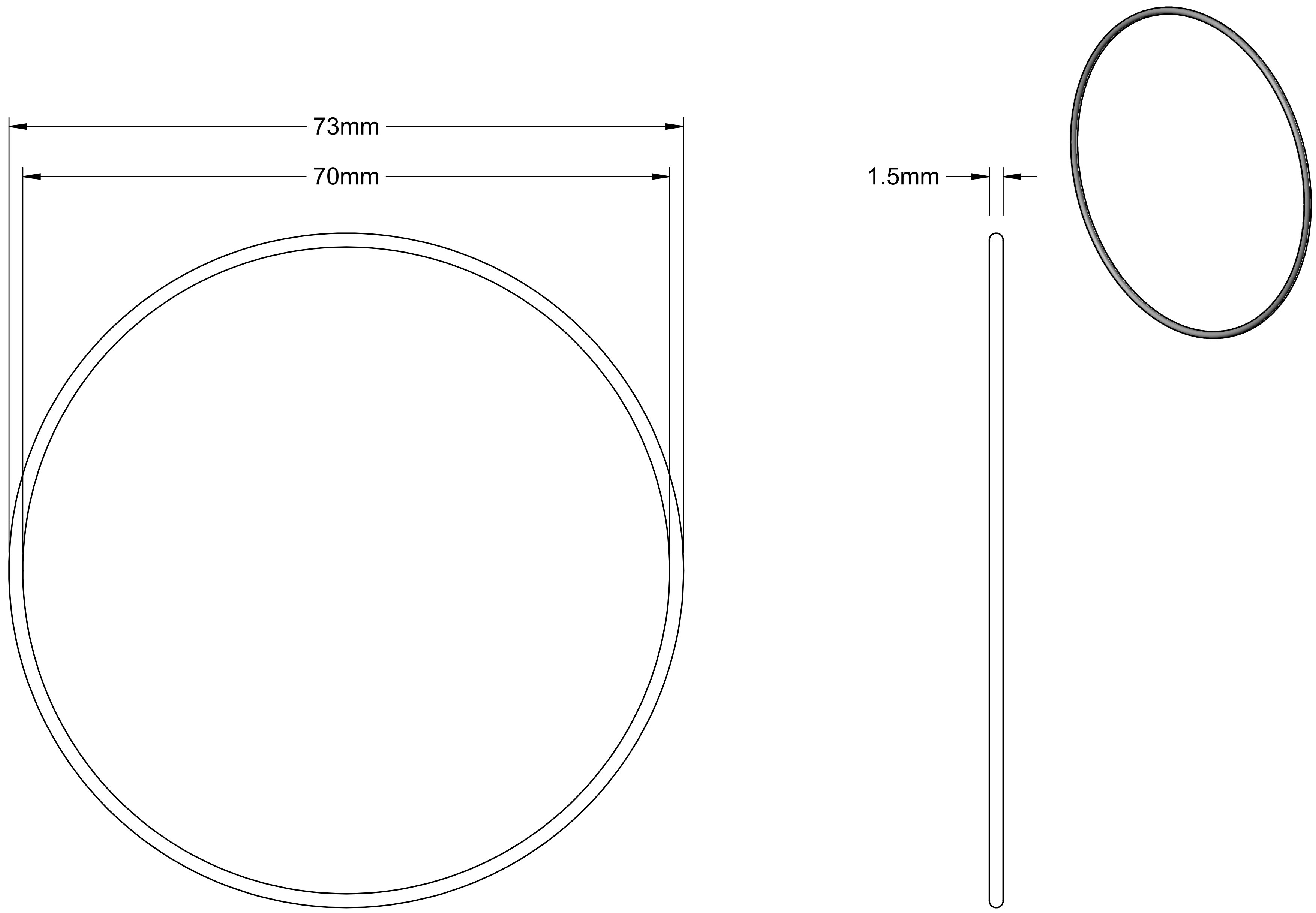
DWG NO.

Pressure Vessel

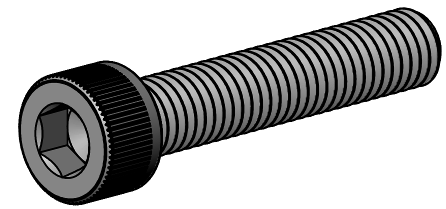
A3

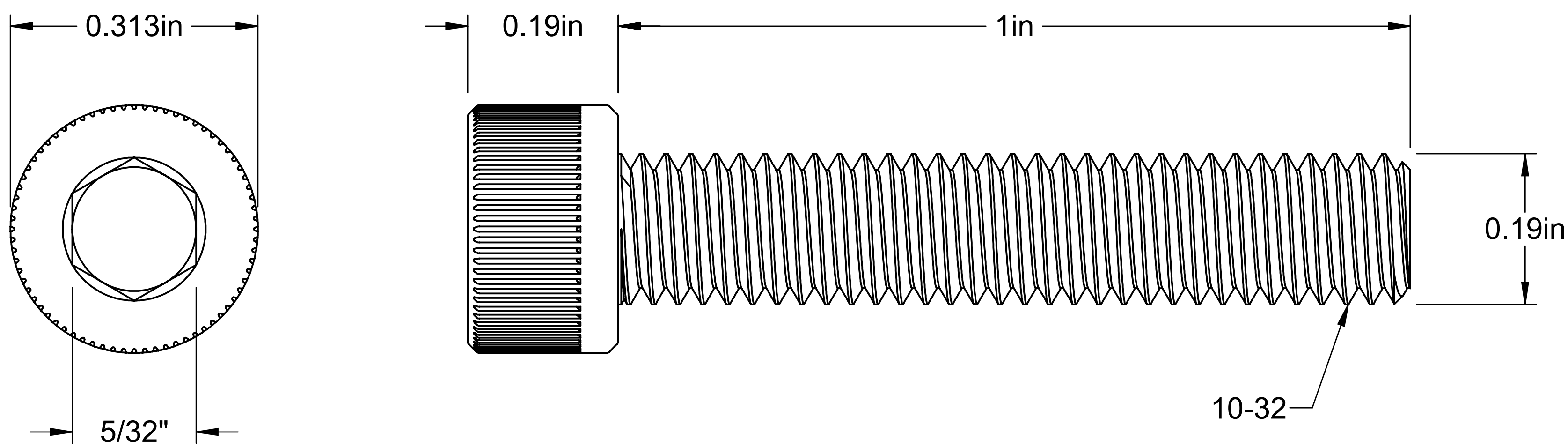
SCALE:1:1

SHEET 1 OF 1



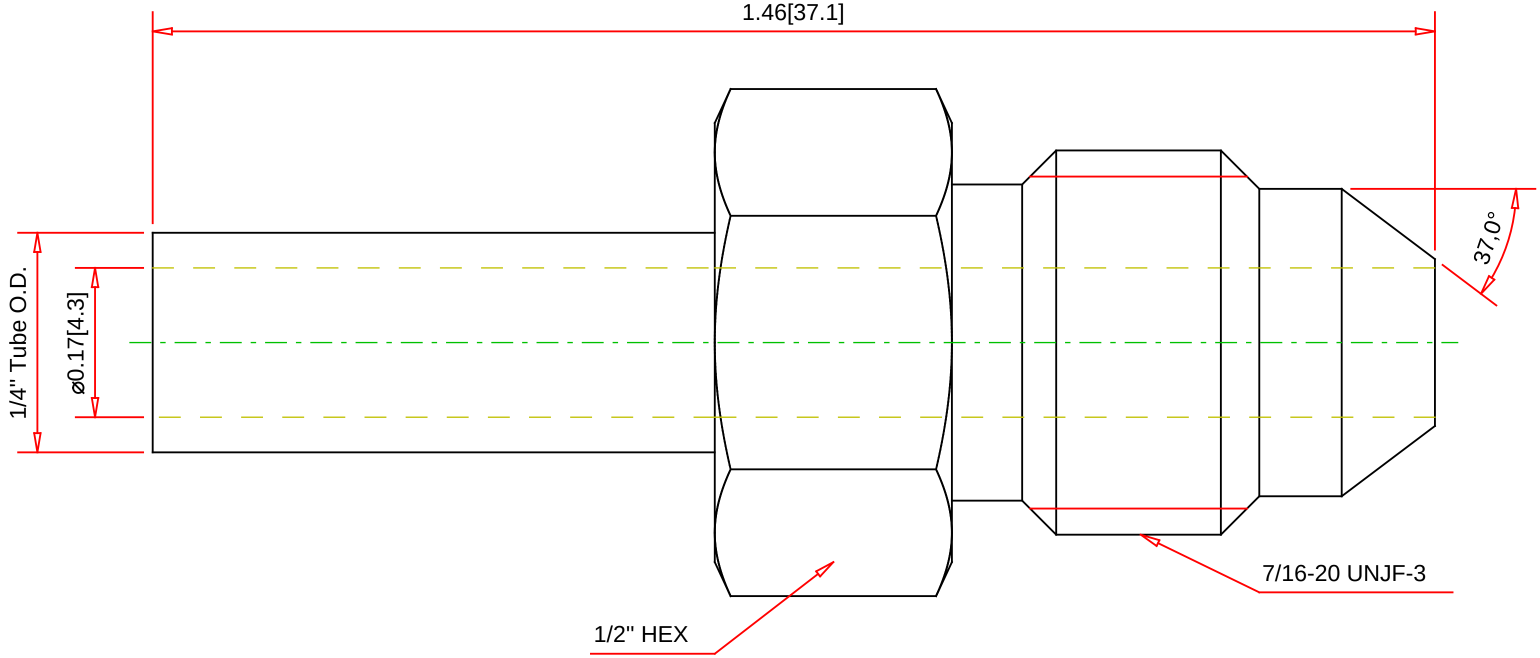
McMASTER-CARR® CAD		PART NUMBER	9262K534
http://www.mcmaster.com © 2025 McMaster-Carr Supply Company		Oil-Resistant Buna-N O-Ring	
Information in this drawing is provided for reference only.			





MCMASTER-CARR® CAD	PART NUMBER	90128A947
http://www.mcmaster.com Information in this drawing is provided for reference only.	Zinc-Plated Alloy Steel Socket Head Screw	

PART NO. B-4-TA-1-4AN



The internal diameter dimension is the minimum nominal opening. These fittings may have a larger opening at the pipe/straight thread end.

CUSTOMER DRAWING

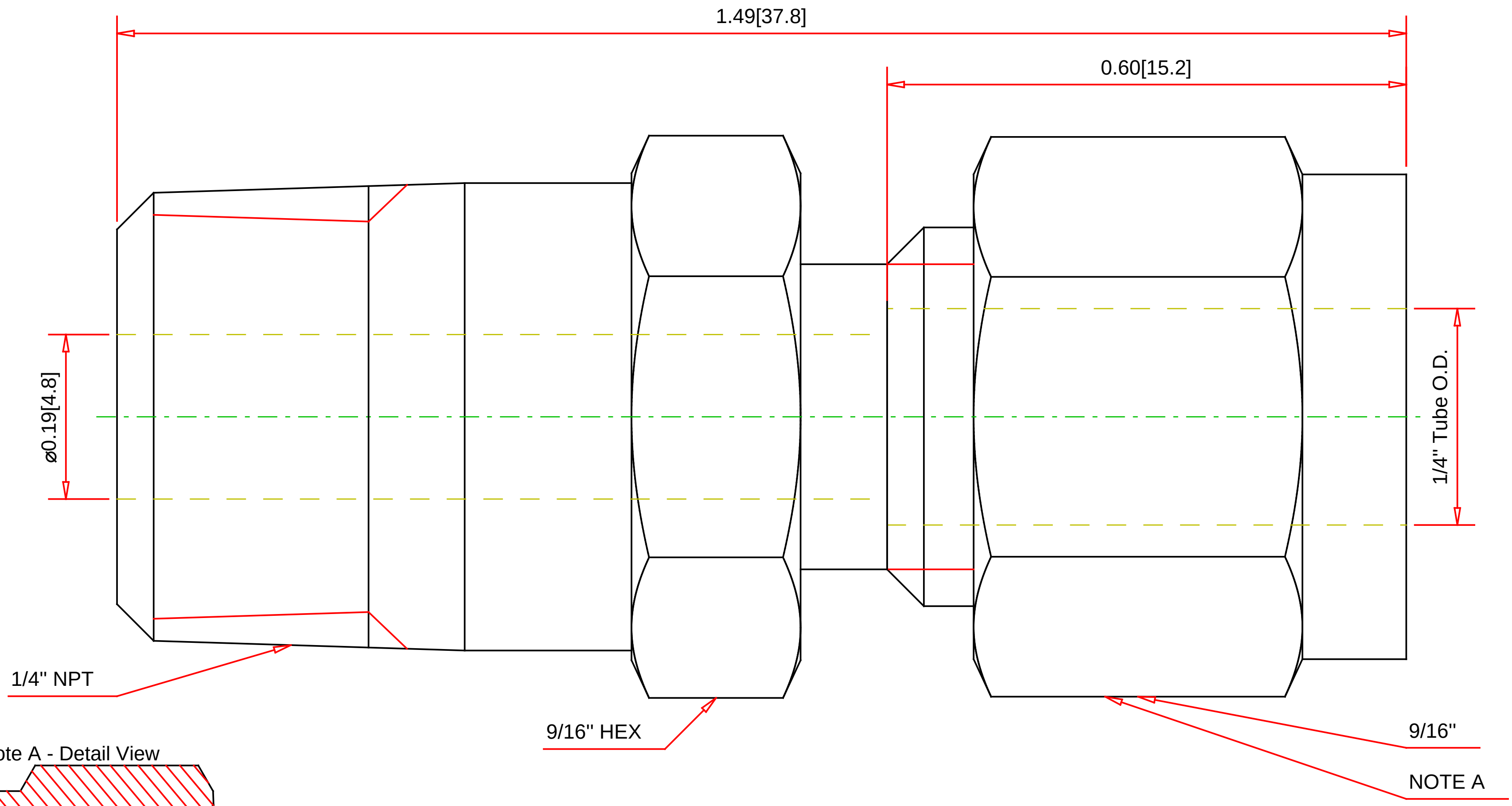
Swagelok

- DRAWING NOT TO SCALE
- DIMENSION ARE INCHES NEXT TO [MILIMETRES]
- DRAWING IS SUBJECT TO CHANGE WITHOUT NOTICE.
- ALL ASSEMBLED NUTS AND FERRULES ARE SHOWN AT FINGER TIGHT DIMENSIONS.
- ALL HEX CALL-OUT ARE ACROSS FLATS.

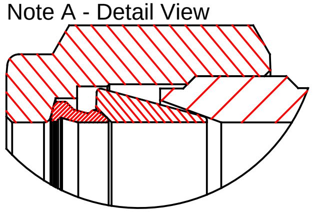
TITLE
AN Adapter - Fractional (to AN Flare Tube)

PART NO. B-4-TA-1-4AN

PART NO. B-400-1-4



Note A - Detail View



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CUSTOMER DRAWING		
Swagelok	- DRAWING NOT TO SCALE - DIMENSION ARE INCHES NEXT TO [MILIMETRES] - DRAWING IS SUBJECT TO CHANGE WITHOUT NOTICE. - ALL ASSEMBLED NUTS AND FERRULES ARE SHOWN AT FINGER TIGHT DIMENSIONS. - ALL HEX CALL-OUT ARE ACROSS FLATS.	TITLE Male Connector (Tapered Thread) - Fractional PART NO. B-400-1-4