Design of Clamping Mechanism for Securing Sections of Unmanned Submarine

by

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Abstract

A clamping mechanism was designed for securing together two sections of a 12.75" diameter autonomous underwater vehicle. Two semicylindrical sections are secured together by sixteen ¼"-20 bolts around the machined ends of the vehicle sections. An angled interface between the clamps and the sections, with an angle of 25°, transmits the clamp tension into axial tension, which pulls the sections together and apply force on the gaskets and bulkhead that divides them. Within this paper, the design requirements and details are discussed, including strength, tolerance, sealing, pressure endurance, and material requirements.

This project was developed as part of the MIT class 2.014: Engineering Systems Development, which designed and developed a prototype of a power supply section for a REMUS 600 autonomous underwater vehicle system.

Acknowledgements

Table of Contents

List of Figures

9

List of Tables

1. Introduction

1.1.   Project Overview

This thesis discusses the design of a clamping mechanism to hold together parts of an unmanned submarine for use in a prototype. It was developed as part of the MIT class 2.014: Engineering Systems Development, which focused on designing a power supply section for a REMUS 600 autonomous underwater vehicle. The clamp holds the main power supply section to the two auxiliary sections, which are hollow, mimic the appearance of a REMUS 600, and contain a relatively simple propulsion system to enable in-water testing. The clamp consists of two semi-cylindrical clamping sections that bolt together around the machined ends of the power and auxiliary sections, which are separated by a bulkhead and gaskets for sealing. An angled face between the clamps and the section ends transmits the tension in the bolts into gasket compression.

Details of the class project and purpose of the clamp are first presented in the Background section. Next, in the Requirements section, the design requirements imposed by the use conditions and purpose of the clamp are discussed. An Approach section follows, explaining in greater detail the workings and advantages of the system and including instructions on the operation of the clamp. The Analysis section discusses in greater depth the motivation behind particular design decisions and provides analysis to show that the design meets the requirements specified earlier. In the Conclusions section, the results are summarized and comments are made concerning the overall process.

1.2.   2.013/4 Engineering Systems Design and Development

This project was developed as part of the MIT class series 2.013 and 2.014, Engineering Systems Design and Development, which were held in the fall 2011 and spring 2012 semesters. For this class, students worked together to design a power supply section for a REMUS 600, a widely-used autonomous underwater vehicle (AUV). AUVs are in common use for oceanographic and surveillance

missions, but their usefulness is reduced by their limitation to a maximum of 72 hours on a battery

charge. In the class, a power supply system was developed which consisted of an internal combustion

engine, water and air snorkels, a fuel buoyancy compensation system, and electronic controls, as well as

all auxiliary features such as cooling, mounting, packaging, and routing systems. The improved system

was designed to increase the REMUS's operating life and enable it to complete up to forty 12-hour

missions without needing to return to ship. This project was developed in conjunction with MIT's Lincoln

Laboratories and the Woods Hole Oceanographic Institute; funding was provided by Lincoln Labs.

2. Background

2.1.   Submarine Specifications

This clamp was designed in order to interface with the submarine power supply section made in the MIT class 2.014: Engineering Systems Development. The section developed in that class was a 60" long, 12.75" diameter submarine section including an engine, a snorkel system for air and water intake, a fuel buoyancy compensation mechanism, and an electronics management system. The section was designed to be neutrally buoyant so it can stay underwater without additional power requirements. Its hull is made of 60" of Schedule 80XS Aluminum 6061-T6 pipe, which has an outer diameter of 12.75" and an inner diameter of 11.75". The internal components of the main section were mounted by attaching them to rails, which were in turn supported by holding in tension two end caps on either side of the main section.

For testing and display purposes, a mockup vehicle was made in which to situate the power supply system; the clamp described here was designed as part of this mockup. The mockup consists of two 37" empty sections of pipe on either side of the power supply system. These auxiliary sections are, in turn, held to the main section by this clamp. Fiberglass fins and nose cones were added to the empty sections, and they were outfitted with a simple propulsion system. The empty sections were comprised of the same pipe material as the main section hull, though they were left empty of key components; they were merely filled partially with foam to maintain neutral buoyancy throughout the system. The ends of both the main section and the auxiliary sections are available for machining, but the available material is limited. The clamping system must be designed to interface between the primary section and auxiliary sections, and must also be made from the same hull material as they are.

3. Requirements

3.1. Introduction

The first step in designing this clamping system is to formally lay out the requirements for the design. One fundamental requirement is that the clamp system must be made from the same stock that is being used for the hull of the main system, both to maintain a smooth profile and appearance and also to limit material costs. The primary purposes of the clamp are to mechanically hold the system together when it is out of the water and to seal the electronics from the water when the vehicle is submerged. The system must also account for water pressure when at depth. Since the clamp is designed for use in a prototype, it is also important that the clamp be relatively easy to assemble and remove.

TABLE 3.1 summarizes the design requirements set forth in this section.

TABLE 3.1: Compilation of design requirements.

Property	Range
Stock material	12" Sch 80XS 6061-T6 Al pipe (12.75"OD, 11.75"ID)
Bending load	425 ft-lb
Gasket material	Resist weather and seawater corrosion
Gasket width	Absorb tolerance variation of clamp
Gasket sealing	Prevents water from passing to main section from either outside hull or from side section
Gasket compression set	Good
Depth	20 m
Number of bolts	Minimized
Tension before inserting bolts	None

3.2. Aesthetics and material

Since this clamp will be used for display, it should adequately represent the appearance of a REMUS, for which the power section has been designed. One aspect of this requirement is that the clamp must retain the smooth profile of the rest of the system; a flanged design with a large protrusion both interferes with the flow profile around the submarine and appears slapdash rather than professional

when on display. To maintain a smooth profile, and to limit material costs, the clamp must be made from a section of the same stock as the hull of the REMUS. The stock used is 12" Schedule 80XS 6061-TG aluminum tube stock, with an outer diameter of 12.75" and an inner diameter to 11.75". This restriction requires that any features of the clamp be contained within the 0.5" walls of this pipe.

3.3.   Bending and Joint Loads

The strength requirement for the clamping system requires it to hold the pipe sections together safely while the vehicle is being handled and displayed out of the water. When the section is submerged, the buoyancy force will support the weight of the submarine. However, during transport, storage, and display, the submarine section must stay together and the clamping mechanism must not fail.

Figure 3.1 shows a diagram of the systems that the clamping system must hold together. The weights are determined using the known dimensions of the submarine hull: 12.75" outer diameter, 11.75" inner diameter. The central section is assumed to be slightly heavier than neutrally buoyant, so their weight depends only on the pipe diameter and length (275 pounds). The outer sections are assumed to be empty and flooded at depth, and thus their weight out of water is merely the weight of the hull. This weight is calculated from the pipe dimensions as well as the density of the aluminum stock from which the pipe is made. Fixing the clamps, the bending moment on each is 425 ft-lbs; thus, each clamp must be able to support a bending moment of 425 ft-lbs.
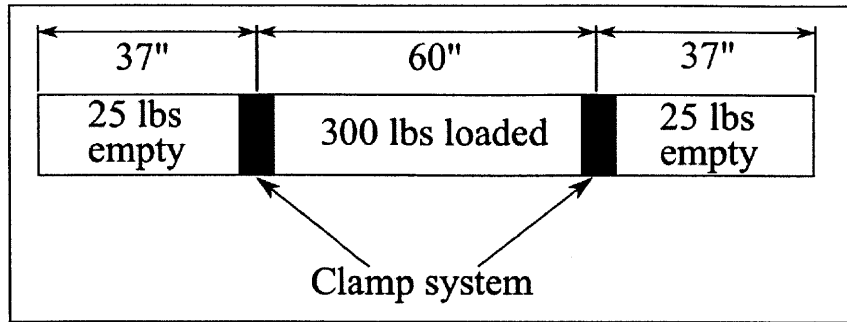
Figure 3.1: Diagram of sections that must be clamped together, with associated weights and distances.

3.4.   Gasket and Sealing

As the last section concerned operation outside of water, this section considers operation in water. The primary goal of the section when submerged is to keep water out of the sensitive electronic equipment. In addition, since the sections adjacent to the engine section will be flooded, the water must be kept from passing through the clamp section. Thus, a bulkhead is added to the system to separate the sections. This bulkhead is also necessary to support rails that run through the main section and hold the components.

In order to accomplish this sealing, gaskets must be added in order to absorb tolerances and block the passage of water. The gasket material must be selected to resist weathering and degrading in the seawater environment. Furthermore, as it will undergo relatively frequent assembly and disassembly, as well as cyclic pressure loadings (though see next subsection), it must have good compression set and recovery characteristics. The additional purpose of a gasket is to absorb tolerance mismatch in the parts. Thus, the gasket must be thick enough to deform up to the tolerance mismatch while staying in the elastic region.

3.5.   Pressure Loads

The effect of water pressure acting on the submarine at depth must be studied. The radial pressure on the hull will not be considered in this design, since pipe selection for hull strength was determined as

19

part of the project and the same material is used for the section joints. However, the axial pressure will act to compress the gaskets further. Though this added pressure will increase the effectiveness of the seal, the additional force gained at depth must be considered when determining the preload on the gasket. If the force on the gasket applied by the water pressure exceeds the force applied by the preload, the clamp force will drop to zero and the clamp could come loose. Furthermore, the gasket must be strong enough to withstand the pressure supplied at depth (or by the preload, if that is greater).

The entire submarine section was designed to withstand pressures up to 600 m depth. For this clamp, however, ease of assembly and material selection was prioritized over nominal depth specifications. Thus, for this clamp, the maximum depth rating is 20 m, which is chosen to be larger than the maximum depth of the Charles River [1], in which the submarine will be tested.

3.6.   Assembly

Since the system is a prototype, the internals of the system will need to be accessed often. Therefore, it must not be exceedingly difficult to open and close the clamping mechanism to access the electronics. While this requirement is somewhat less important than the above requirements (that is, sealing should not be sacrificed for ease of opening), it is nevertheless another important aspect for analyzing the system. One method of formulating this requirement is by specifying that the number of operations to open or close the clamp be limited. Another important aspect is the alignment accuracy required during assembly. Ideally, the clamp should not need to be pre-compressed before inserting and tightening any bolts. Instead, the dimensions and tolerances should be set up so that aligning the clamp requires no force, and the only application of force comes in when tightening bolts. While requiring the assembler to lift a 300-lb center section is unavoidable, thus needing at least two people, no additional force should be required.

4. Approach

4.1. Overview

In order to meet all the requirements set out in the previous section, a design was selected that consists of two semi-cylindrical clamps that bolt together on the outside of the sections and, through an angled feature, induce axial compression force on two gaskets and a bulkhead separating the two sections. An exploded view of the system is shown below, in
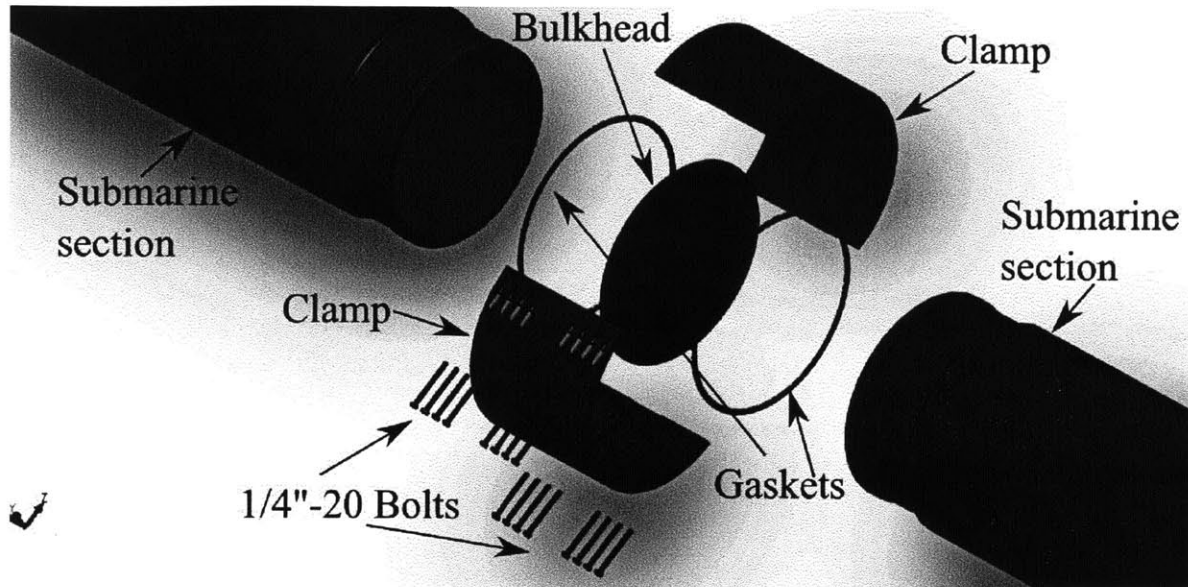
Figure 4.1. The two sections of submarine have machined ends to allow them to interface with the clamping system; the machined length is $4\frac{7}{8}$ in. The sections come together in the center around the bulkhead, a machined aluminum plate with maximum thickness of 0.5" and inset rims for the insertion of gaskets. Once the sections and bulkhead are aligned, the clamps come radially inward from either side to compress around the machined section ends and are secured by up to sixteen ¼"-20 stainless steel bolts. The angled interface between the two pieces transmits the bolt tension to compression force along the submarine main axis, thus compressing the gaskets and forming a seal. The details of this interface are shown in cross-section in Figure 4.2.
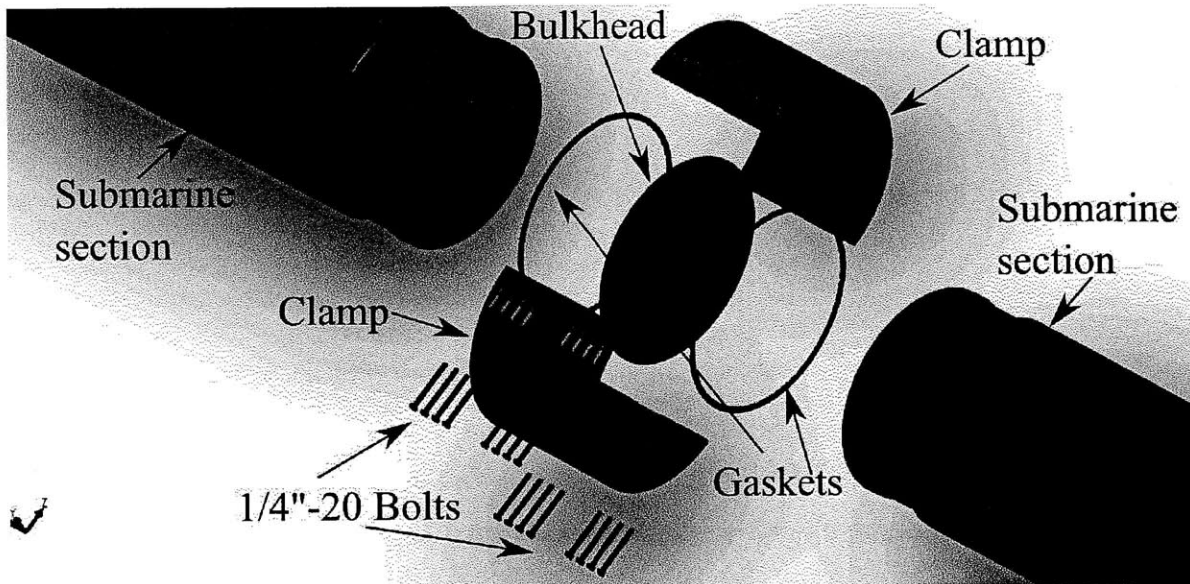
Figure 4.1: Exploded view of clamping system, with names of components shown.
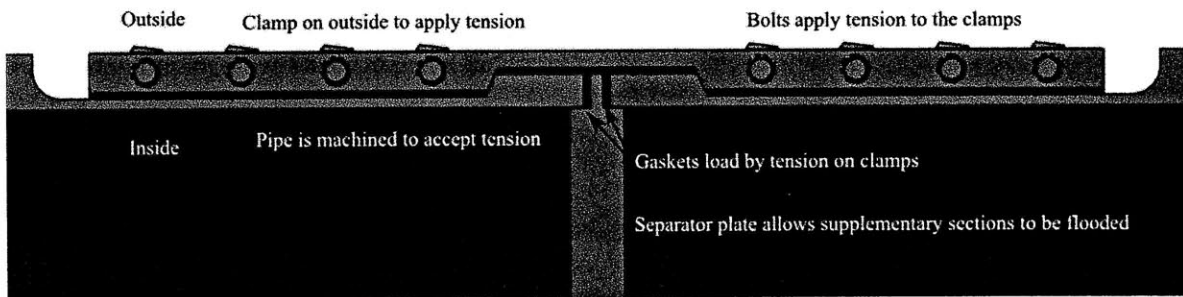
Figure 4.2: Cross-sectional view of the clamping system, illustrating the angled feature that transmits tension on the clamps to compression of the gaskets.

The advantages of this design are that it maintains the smooth outer profile and appearance of a submarine and does indeed fulfill the material constraints by being producible from Schedule 80XS pipe. In addition, the recessed bolts inserted orthogonal to the submarine axis both intrude little into the flow profile and also remain accessible from the outside. The angled connection between the clamps and the section interface enables this bolt position and also unites the clamp attachment and gasket compression forces, which would not be possible from, say, a concealed flange design. The bulkhead

and gasket separator enables the components of the main section to be isolated from both the adjacent sections and the water outside the assembly, while additionally supplying the mounting location for the support rails within the main section. Adequate design of the critical dimensions will allow the clamps to sit directly against the angled face before any tightening so that the only force required in assembly is supplied during the bolt tensioning process.

The design, as explained above, requires certain critical parameters to be determined. The wall thicknesses of various parts of the clamp and section interface can be selected using the strength and tolerance requirements of the clamp. The angle is chosen based on this tolerance and the required gasket preload. The preload is also used to determine the bolt strength and quantity, as well as the gasket dimensions and material. The details of the design are explained in Section 0.

4.2.   Assembly Instructions

The method of assembling the section joint is summarized in Figure 4.3. First, the on-axis components are placed together. Put the gaskets in the grooves in the bulkhead plate and align the sections to be attached with the bulkhead. If applicable, fasten the bulkhead to the rails in the primary section, making sure to place the gasket between the main section and the bulkhead. Then, align the remaining section (or both, if the rails have not been affixed) to the bulkhead and bring them together as close as reasonably possible. Close alignment is required to ensure that the angled surface of the clamp touches the angled surfaces of the sections.

Next, join the two halves of the section clamp around the interface section, ensuring that the angled faces of the main section and the clamps touch. Make sure also to use one "head" clamp (with bolt clearance holes and access holes for screwing accessible) and one "thread" clamp (with smaller, threaded holes).

23

Finally, screw the bolts into the appropriate slots according to a bolt rotation plan, detailed below in Figure 4.4. The bolts should be tightened in the order shown, to at least 1220 pounds each. For less critical applications, not all 16 bolts need be used; bolts can be removed in groups of four according to their order of tightening. For example, during transport one could use only bolts one through twelve or one through eight. For underwater use, however, all sixteen bolts should be installed.
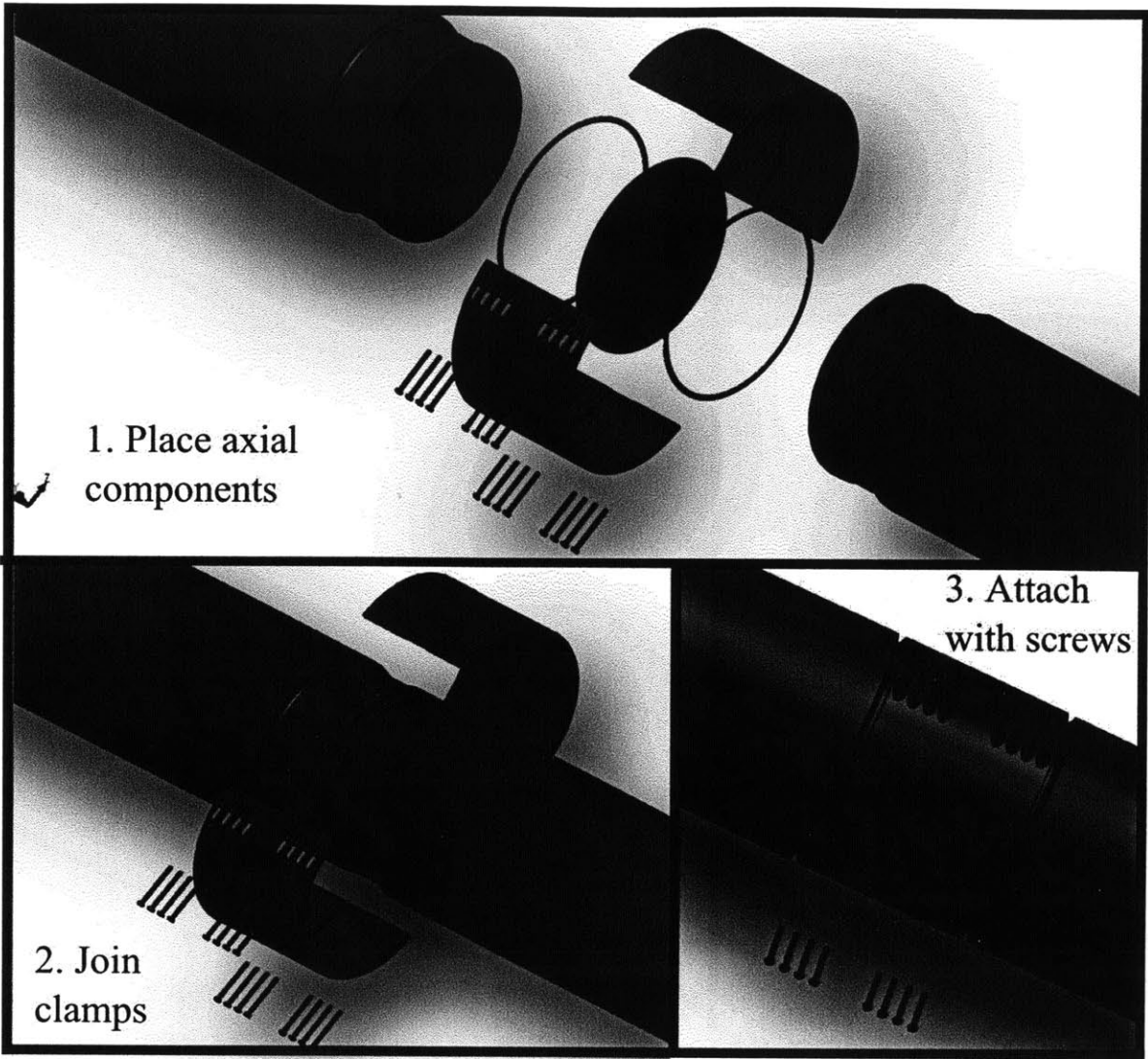
Figure 4.3: Steps required to install the clamping mechanism. First, all the parts that mate axially are aligned. Next, the clamp sections are placed around this axial section so that the angled surfaces are in contact. Finally, the system is bolted together.
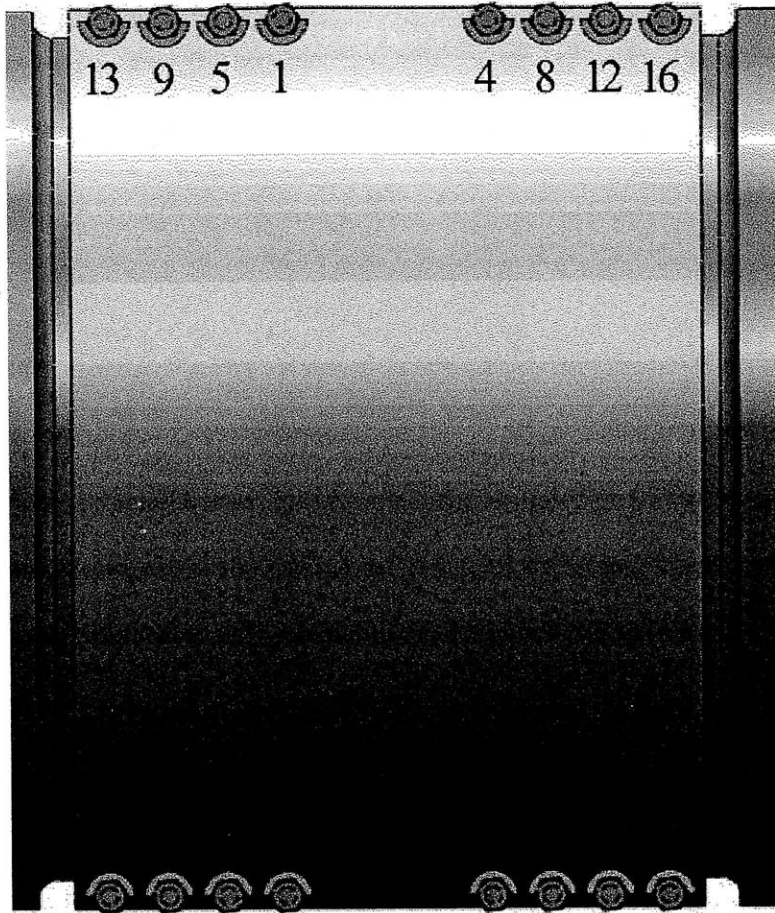
Figure 4.4: Diagram showing order for tightening bolts. To evenly distribute stress on the gasket, bolts should be tightened in an alternating pattern as shown. For temporary clamping, the last four or eight bolts may be omitted.

5. Analysis

5.1.   Introduction

In order to meet the design requirements, various specifics of the design must be determined. First,

the minimum allowable thickness of the bored-out features must be determined based on the strength

requirements on the clamp; stress concentrations due to the bolt holes must also be accounted for.

Next, gasket preload must be considered, depending upon the expected compression force at depth;

from this property, gasket material and dimensions can be selected. Based on the required compression

force and tolerance stackup in the central axial region, the angle of interface can be determined.

Following this angle, and using the required bolt preload, the number and diameter of the bolts are

selected.

5.2.   Strength

In this section, the restrictions placed on the design by the strength requirement (425 ft-lbs of

torque) will be examined. First, the minimum allowable thickness of a cylindrical feature will be

determined analytically. Next, stress concentrations and the strength of the overall assembly are

examined using finite element analysis. Thus, the design is shown to be strong enough to support the

weight of the components.

5.2.1. Thickness Analysis

The first analysis that can be done is an estimate of the minimum thickness allowable of a cylindrical

shell at the radii under consideration. The equation for bending stress in a region with known external

moment is

$$\sigma = \frac{My}{I_x} \ [2]$$
(5.1)

with I_x for a cylindrical shell found to be

27

$$I_x = \frac{\pi}{4}\left(R_0^4 - R_i^4\right) \approx \pi R_0^3 t, \tag{5.2}$$

assuming the shell is thin with thickness t. Equations (5.1) and (5.2) can be solved together for the minimum thickness as a function of applied moment and yield strength, using R_0 as the maximal distance from the neutral axis:

$$t = \frac{M}{\pi R_0^2 \sigma_y}. \tag{5.3}$$

Using $\sigma_y = 40$ ksi [3], $M = 425$ft-lbs, and $R_0 = 5.875"$ for maximal t, it is found that for a cylindrical shell to support the given moment,

$$t > 0.002 \text{ in.} \tag{5.4}$$

This limit is easily accomplished in the design; in fact, the minimum thickness is smaller than the tolerance specifications for machining. Thus, ease of machining is a more critical consideration than strength. Hence the thickness restriction is chosen to be

$$t \geq 0.1 \text{ in.} \tag{5.5}$$

5.2.2. Numerical Analysis

For more detailed analysis, numerical simulations must be produced, specifically with finite element analysis. Using the SolidWorks Simulation package, the assembly was set up in two orientations, horizontal and vertical. Each orientation was supported by two rigid, fixed support pieces, emulating the state of the system during transportation and storage. The assembly was allowed to fall under its own weight, and a 105 kg distributed mass was added to the central section, simulating the weight of the internal components, assuming it is neutrally buoyant. A picture of the starting configuration is shown in Figure 3.1.
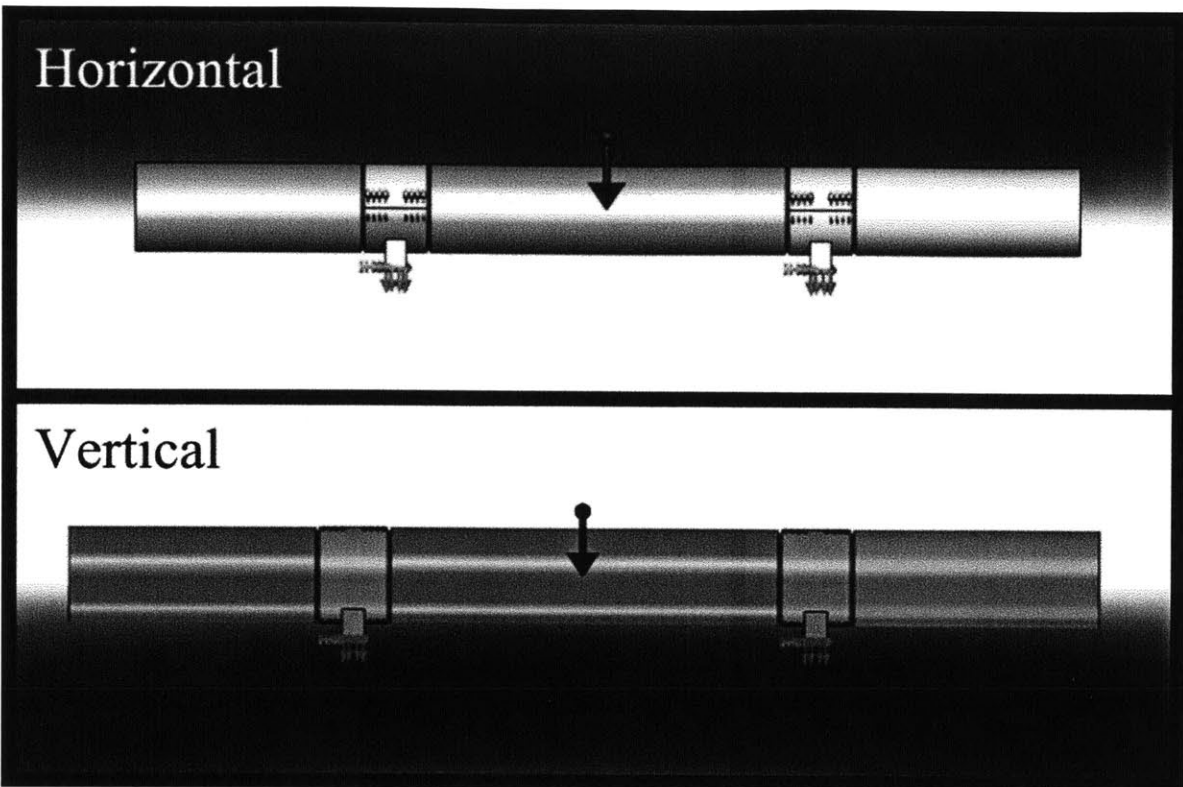
Figure 5.1: Setup for finite element analysis. The assembly in each orientation is supported by a two fixed, rigid supports simulating the support during transport or storage. It is subjected to loading due to gravity, with an additional 105 kg distributed mass in the central section to emulate the contained components.

The simulation was then performed on this setup; results appear in Figure 5.2. As shown there, the minimum safety factor is over 200 for either case. Thus, the clamping system is sufficiently strong to hold up the weight of the submarine sections. One confirmation of this analysis is that the safety factor is the same order of magnitude as the ratio between the smallest thickness used (0.1 in.) and the minimum allowable thickness (0.002 in.), as determined above. The agreement between the two types of analysis suggests their accuracy. A further point of interest is that the clamp is stronger when held horizontally than when held vertically. Though the clamp is strong in the vertical direction, it should ideally be held horizontally to increase its strength. In general, though, this design presents no concerns about its ability to support the weight of the submarine sections.
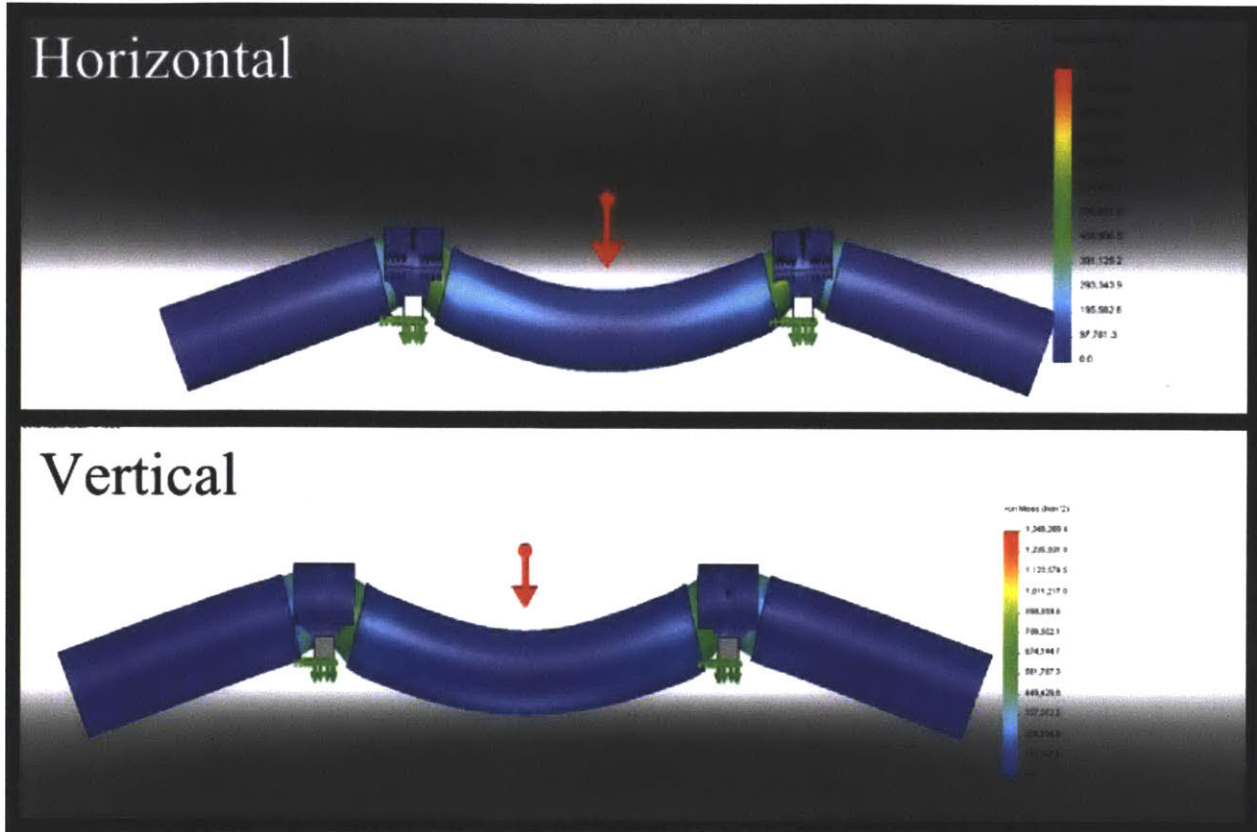
Figure 5.2:   FEA results on the systems in Figure 5.1. The maximum von Mises stresses for the two setups were 1.17 MPa and 1.35 MPa respectively, corresponding to safety factors of 234 and 204. The maximum strains found were 72.6 µstrain and 80.6 µstrain.

5.3.   Angle Selection

Another parameter that must be determined for the design is the interface angle between the clamp and the machined section piece. This angle influences two factors: the transmission of clamping tension to gasket compression and the axial travel distance allowed during compression. The force transmission factor controls how much bolt tension is transmitted to gasket compression, and thus affects the number of bolts required to obtain a given gasket preload, which is in turn determined by the gasket material and specified water pressure. The angle must also be steep enough to overcome friction between the two faces. The axial length of interference between the two faces is also determined by the angle, and it governs the allowable variation in length of the components under compression. As long as

31

the components start near enough that the clamp can engage the angled face of the submarine

section, the bolt tension will compress the gaskets and seal the vessel.

5.3.1. Force Transmission

First, the force transmission angle dependence will be considered. To do so, consider the free-body

diagram in Figure 5.3. A force balance on each component leads to the equations

$$
\begin{aligned}
T &= F\cos\theta + N\sin\theta \\
P &= N\cos\theta - F\sin\theta \\
F &= \mu N,
\end{aligned}
\tag{5.6}
$$

where the last equation comes from the static/dynamic friction model. Solving these equations leads to

an equation for the force transmission factor:

$$
\frac{P}{T} = \frac{1 - \mu\tan\theta}{\mu + \tan\theta}.
\tag{5.7}
$$

In order to select an appropriate angle, this relationship must be examined. Figure 5.4 shows a plot

of this relationship, using as the coefficient of dynamic friction $\mu = 1.4$ [4]. The first notable

characteristic of this plot is that above a certain angle, the friction between the plates cancels the

transmission of force across the face. This critical angle can also be calculated from Equation (5.7) as

$$
\theta_{max} = \tan^{-1}\left(\frac{1}{\mu}\right) = 35.5°.
\tag{5.8}
$$

From this equation, we find an upper limit on the face angle. As the angle gets smaller, the factor

increases approximately linearly, to a nominal value of 0.71 near 0°. Note, though, that at 0° there is no

normal force and the system is no longer static; thus, the equation is not valid for very small angles. The

equation is also inversely dependent on µ; thus, if the system produced does not have sufficient force,

the surfaces can be polished to reduce friction as a substitute for remachining the face with a steeper

angle. Nevertheless, the overall force constraint is that the angle must be less than about 30° for

adequate transmission, and that a larger angle is better.

33
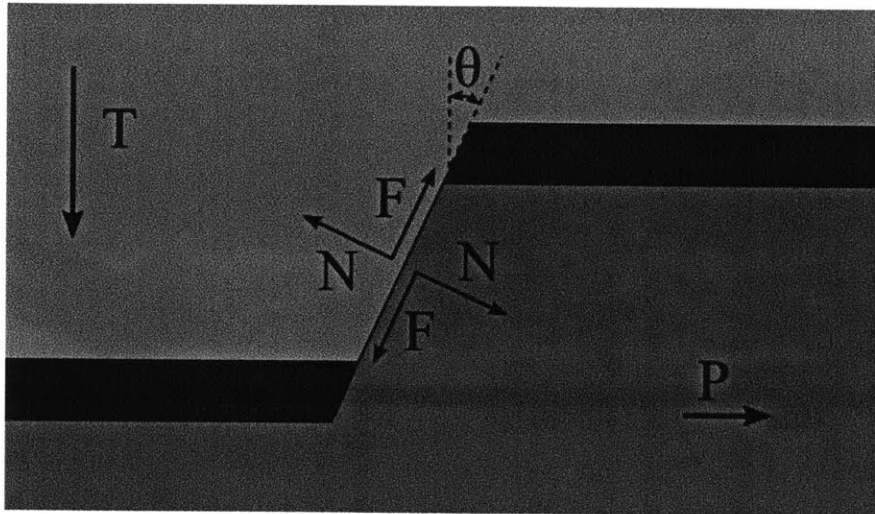
Figure 5.3: Free-body diagram of forces at the angled interface between the clamp and section. An external tension T is applied to the clamp by the tightened bolts, which goes through the angled face to cause an axial compression force P.
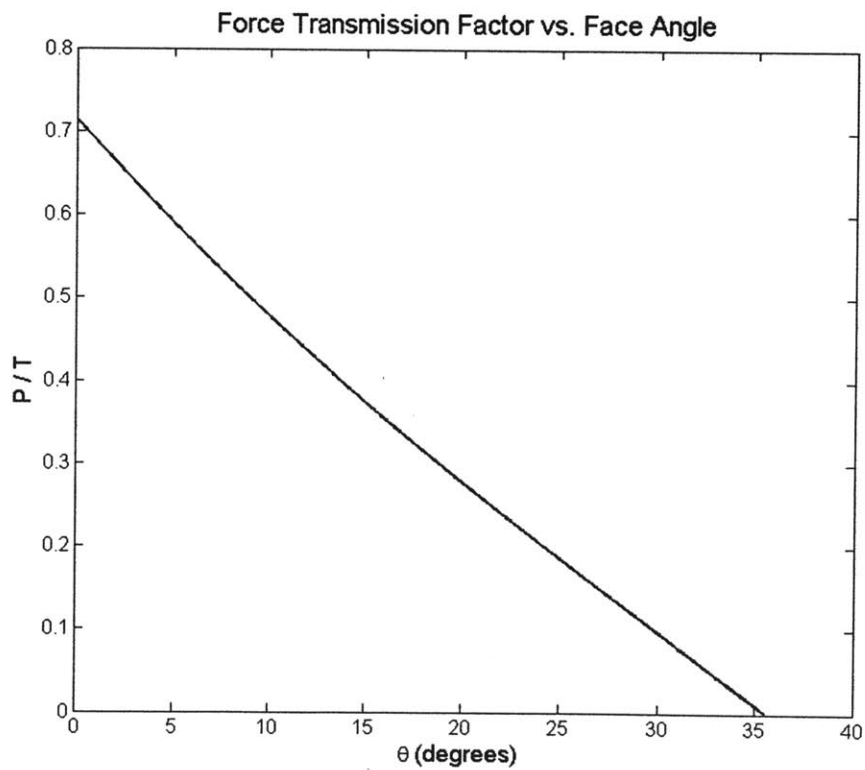
Figure 5.4: Plot of the force transmission factor dependence on face angle. The plot assumes a coefficient of dynamic friction of 1.4, for aluminum sliding on aluminum. The factor decreases approximately linearly from an initial value of 0.71 to zero at 35.5°, at which angle the friction exactly counteracts the applied tension.

5.3.2. Tolerance

The other primary constraint on angle selection is providing enough length axially to absorb the tolerance variation of the contained components. Figure 5.5 illustrates the expected tolerance variation of the clamped region as well as the usable height variation. The distance that the angle covers is limited by the strength of the material and, critically, by the limited allowable total wall thickness for the two parts. Thus, only 0.200" are available for the interface radial tolerance. This variation must match the required tolerance in the axial direction; they are related by

$$\delta_z = \delta_r \tan \theta. \tag{5.9}$$

The stated tolerance requirement is 0.047". However, since the clamp is used for a prototype, the pieces can be machined to match tolerances. In particular, the bulkhead is machined last, and its thickness can be varied to account for the large tolerance in the gaskets. Thus, the new tolerance requirement is determined by setting 0.005" measurement variation for each piece. Accounting for a minimum allowable gasket compression strain of 0.5, the axial variation must be

$$\delta_z = 5(0.005") + 0.5(0.063" + 0.016") = 0.065". \tag{5.10}$$

Thus, under this restriction,

$$\theta_{min} = \tan^{-1}\left(\frac{\delta_r}{\delta_z}\right) = 17.8°. \tag{5.11}$$

As determined earlier, in Equation (5.8), $\theta_{max} = 35.5°$. Thus, an angle is selected evenly between the extremes, and at a natural value for machining: $\theta = 25°$.

35
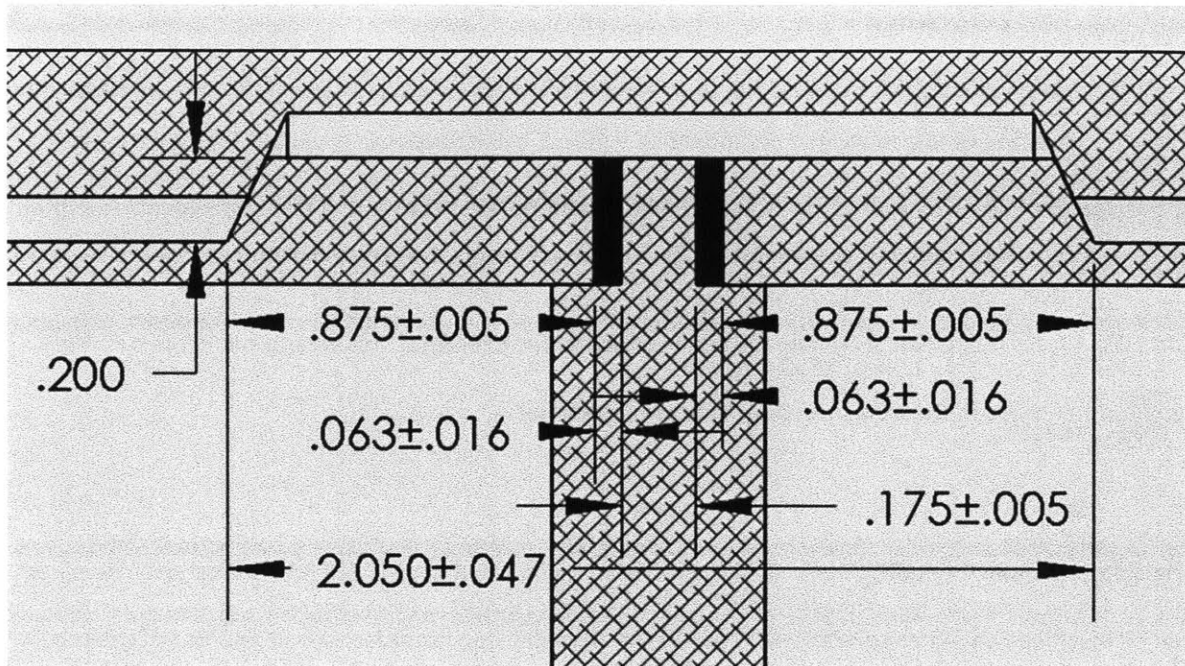
Figure 5.5: Diagram illustrating tolerance stackup of clamped portion. The angle must be shallow enough to enable the angled face to engage at any tolerance.

5.4. *Gasket Selection*

Many of the gasket requirements were defined in Section 3.4 above. However, now that design specifics have been determined, a more detailed examination of the requirements for the gasket is now possible. Previously, weathering and sealing requirements were specified, and tension/compression requirements were left unspecified. Now that a specific design has been selected and the angle has been determined, these requirements can be made specific. A summary of the requirements is shown in TABLE 5.1.

TABLE 5.1: Summary of gasket requirements.

Gasket property	Range
Material	Resist weather and seawater corrosion
Temperature	Down to 32°F
Thickness	0.063"±0.016"
Pressure	800 psi maximum
Strain	< 0.5 at maximum pressure

5.4.1. Material Properties

As discussed earlier, the gasket must be able to withstand environmental conditions, in particular

seawater, low temperatures, and weathering. Reference [5] recommends the use of "synthetic rubber,

rubber bonded aramid or synthetic fibre" for use with stainless steel pipes in seawater environments. It

also cautions against the use of PTFE, since the material helps corrode the steel. Though the hull is

aluminum in this case, the hull for the actual system would be made from stainless steel, and it is

appropriate that gasket selection be done to accommodate this limitation, assuming a gasket can be

found that is appropriate. Additionally, the gasket must be able to withstand low temperatures, as low

as the freezing temperature of water.

5.4.2. Pressure

The gasket must be strong enough to withstand the pressure at its maximum depth. The pressure

can be calculated as hydrostatic, at the 20 m depth specified earlier:

$$P_w = \rho g h = 72 \text{ psi.} \tag{5.12}$$

For axial force, multiply this pressure by the submarine hull cross-sectional area:

$$F_w = P_w \frac{\pi}{4} D^2 = 9080 \text{ lbs.} \tag{5.13}$$

When distributed over the area of the gasket, which is an annulus of outer diameter 12.35" and inner

diameter 11.75", the gasket pressure becomes

$$P_g = \frac{F_w}{\frac{\pi}{4}(OD^2 - ID^2)} = 800 \text{ psi.} \tag{5.14}$$

The gasket must be strong enough to withstand at least this pressure with some suitable margin of

safety included.

5.4.3. Hardness

As mentioned earlier, the tolerance was determined assuming a maximum gasket strains of $\varepsilon = 0.5$.

The gasket's Young's modulus must then be at least

$$E_{min} = \frac{\sigma_{max}}{\varepsilon_{max}} = 1.6 \, \text{ksi.} \tag{5.15}$$

Gaskets, however, are usually specified using durometer hardness readings. The relationship between

Young's modulus and durometer reading is complicated; a roughly corresponding restriction is a

durometer reading 80A or harder.

5.4.4. Compression Set

The gasket preload (see next section) is calculated in order to minimize the cyclic loading

encountered by the gasket when surfacing and lowering. However, since the device is for a prototype, it

is expected to be opened and resealed often. Thus, the gasket selected must have good compression set

characteristics. Unfortunately, this property is difficult to numerically identify, since its value depends

strongly on usage conditions. Nevertheless, a rating of "good" or better is necessary.

5.4.5. Gasket Selection

Ultimately, the gasket selected was High-Strength Weather-Resistant EPDM rubber, available from

McMaster. This grade of EPDM has good weatherability and water resistance, as well as a temperature

range that extends down to -55°F [6]. Its tensile strength is rated to 1700 psi, its durometer hardness is

80A, as specified, and it is rated as having "good" compression set characteristics. An additional benefit

is that this material is available from McMaster in appropriate sizes for cutting into annular gaskets at a

relatively cheap price.

5.5.   Bolt Selection

5.5.1. Gasket Pretension

The only remaining question left to be determined is the number and size of the bolts to be used to tighten the clamps together. The required pretension on the gasket is determined from the maximum force supplied by the water pressure at depth. If the pressure from the water ever puts more force on the gasket than is supplied by the bolts, there will be two problematic effects: the clamp force will drop to zero, possibly leading to the clamp coming loose, and the gasket force will increase, leading to cyclic loading on the gasket and potential gasket failure. Thus, the bolts must be set to apply the maximum force that the system will experience during operation.

The maximum gasket force was calculated earlier, in Equation (5.13) above, to be 9080 lbs. However, as indicated in Section 5.3.1, the force applied by the bolts must be transmitted through the angled face before it comes to the gasket. The transmission factor is defined in Equation (5.7), and for the given $\theta = 25°$ and $\mu = 1.4$, it works out to be

$$\frac{P}{T} = 0.186. \tag{5.16}$$

Thus, the total tension supplied by the bolts must be

$$T = \frac{F_w}{0.186} = 1.95 \times 10^4 \text{ lbs.} \tag{5.17}$$

5.5.2. Bolt Size and Count

Based on the thickness calculations in Sections 5.2.1 and 5.3.2, there is 0.35" available thickness for bolting the two clamp sections together. The bolt size to be selected must be small enough that a clearance hole can fit within this thickness. Moreover, the stress concentrations caused by the regions between the holes cannot exceed the yield stress. The bolt size selected was ¼"-20, to conform to a

standard bolt size; the numerical analysis in Section 5.2.2 confirms that the stress concentrations do not cause a problem.

Next, the required number of bolts must be determined. The yield stress of Stainless Steel Grade 18-8 bolts is $\sigma_y = 70$ ksi, which leads to a proof load of

$$F_b = \sigma_y A = 3440 \text{ lbs.} \tag{5.18}$$

Combining this number with the required bolt tension, the minimum number of bolts is

$$N = \left\lceil \frac{T}{F_b} \right\rceil = 6, \tag{5.19}$$

each at maximum tension. To account for the variability in bolt tensioning and to add a safety factor, holes are added for up to 16 bolts. Each should be tensioned to a tightness of

$$T_b = \frac{T}{16} = 1220 \text{ lbs} \tag{5.20}$$

to supply the required gasket pretension. For more casual uses, several of the bolts may be ignored, so long as at least eight remain in use (for symmetry). Thus, an arrangement allowing for up to 16 ¼"-20 bolts is sufficient to supply a gasket preload greater than the expected force from water pressure at depth.
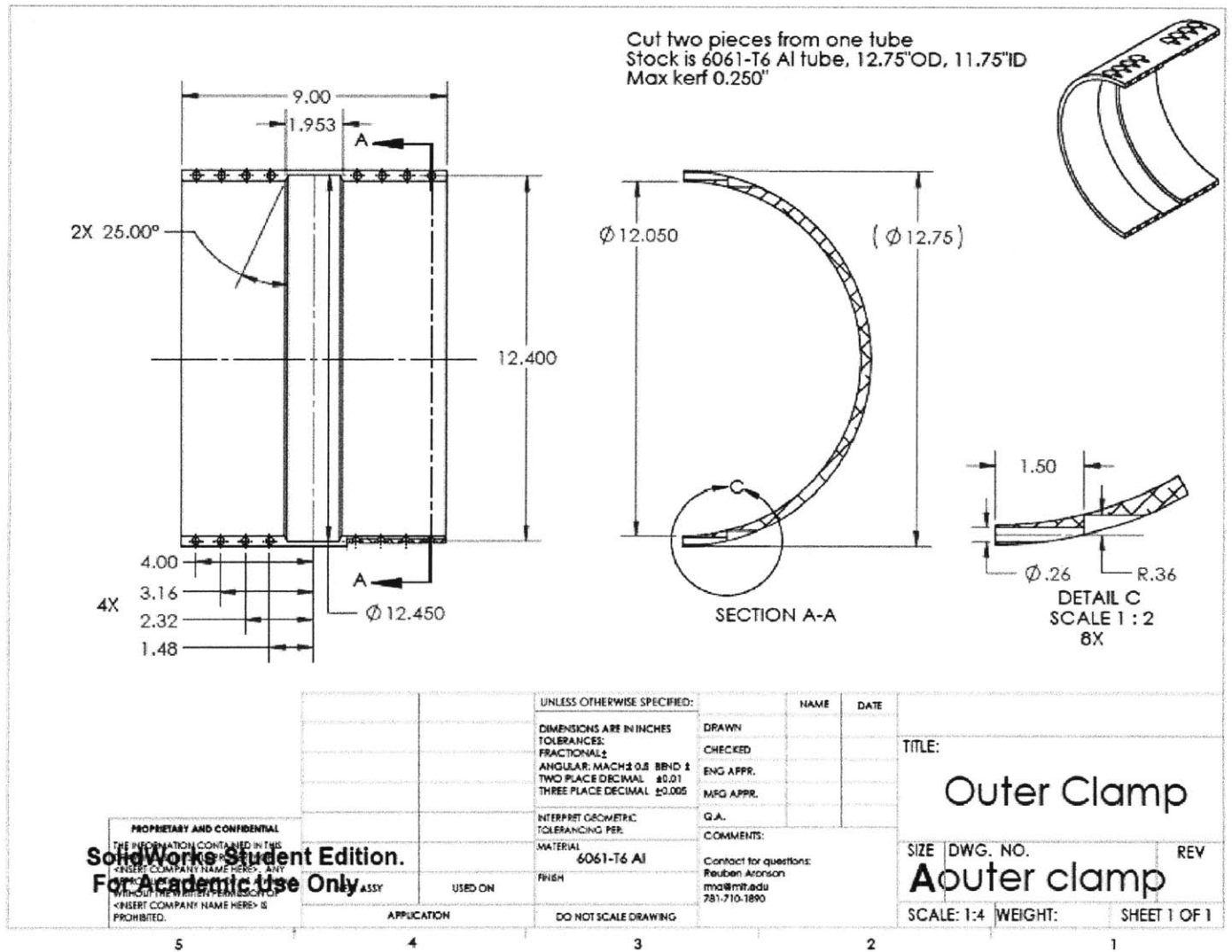
6. Conclusions

6.1.   Design Summary

This design for a clamping mechanism between two sections of a small unmanned submarine successfully fulfills the design requirements as set out in Section 0. The material limitation on the clamp, that it must be made from the same pipe as the original system, did not prevent the system from retaining all necessary features to be successful. Through both analytical and numerical analysis, the clamp has been shown to be strong enough by at least a factor of 200 to support the weight of the submarine. The angle of the interference face has been selected so that sufficient force is transmitted to the gasket and the axial tolerance variation can be absorbed by the gasket. A gasket material was selected that has the desired corrosion resistance and compression properties. Bolts have been chosen, and their tension determined in order to match the gasket preload to the maximum force at depth while not being too difficult to assemble. The design contained here will successfully provide the structure for testing and displaying the power section developed in 2.014.

6.2.   Personal Statement

This project was an excellent learning experience for the author. It was exciting to have the opportunity to work on a project jointly with many of the most experienced undergraduate engineers at MIT. While a group of 30 MIT seniors has the potential to produce an excellent final project, which 2.014 has done, it is also an organizational and managerial challenge of the highest order. To experience first-hand the amount of work is required to direct a large group of engineers so that all the tasks are accomplished and all the pieces fit together has been an excellent learning experience.

In addition, the design of the clamp itself introduced the author to a variety of design challenges. This project included a variety of different analysis techniques, from geometry to stress calculations to numerical simulations to determining machining tolerances. While the topics have appeared separately
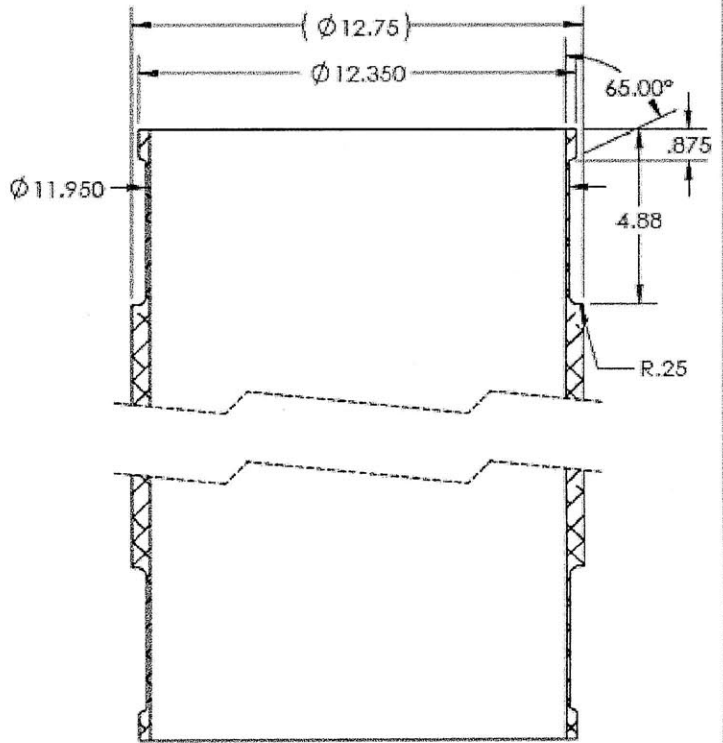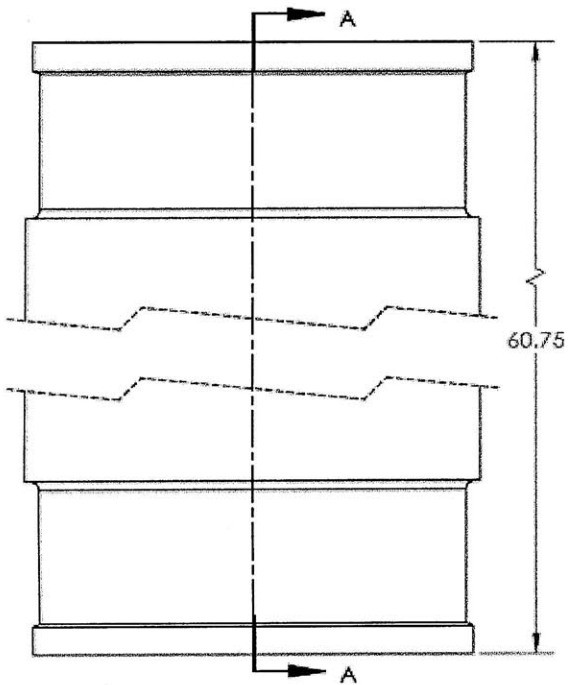
and in a very theoretical manner throughout MIT's coursework, the experience of combining all these features to make a successful design has been an excellent way to unify all the knowledge acquired over an undergraduate career and prepare to enter the working world as a well-trained engineer.

Appendix A: Engineering Drawings

Cut two pieces from one tube
Stock is 6061-T6 Al tube, 12.75"OD, 11.75"ID
Max kerf 0.250"

9.00

1.953

A

2X 25.00°

12.400

$\emptyset$12.450

4.00
3.16
2.32
1.48

4X

A

$\emptyset$12.050   ($\emptyset$12.75)

C

SECTION A-A

1.50

$\emptyset$.26   R.36

DETAIL C
SCALE 1 : 2
8X

		UNLESS OTHERWISE SPECIFIED:	NAME	DATE		
		DIMENSIONS ARE IN INCHES	DRAWN		TITLE:	
		TOLERANCES:	CHECKED			
		FRACTIONAL±				
		ANGULAR: MACH± 0.8 BEND ±	ENG APPR.		**Outer Clamp**	
		TWO PLACE DECIMAL ±0.01				
		THREE PLACE DECIMAL ±0.005	MFG APPR.			
PROPRIETARY AND CONFIDENTIAL		INTERPRET GEOMETRIC	Q.A.			
THE INFORMATION CONTAINED IN THIS		TOLERANCING PER:	COMMENTS:			
DRAWING IS THE SOLE PROPERTY OF <INSERT COMPANY NAME HERE>. ANY		MATERIAL 6061-T6 Al	Contact for questions: Reuben Aronson		SIZE	DWG. NO.	REV
REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF	NEXT ASSY	USED ON	FINISH	rna@mit.edu 781-710-1890		**A**outer clamp	
<INSERT COMPANY NAME HERE> IS PROHIBITED.	APPLICATION		DO NOT SCALE DRAWING			SCALE: 1:4 WEIGHT:	SHEET 1 OF 1

5   4   3   2   1

Cut from 6061-T6 Al tube stock, 12.75OD, 11.75ID

A

60.75

A

$\emptyset$ 12.75

$\emptyset$ 12.350

65.00°

.875

$\emptyset$ 11.950

4.88

R.25

SECTION A-A

			UNLESS OTHERWISE SPECIFIED:		NAME	DATE		
			DIMENSIONS ARE IN INCHES	DRAWN			TITLE:	
			TOLERANCES: FRACTIONAL± ANGULAR: MACH± 0.5 BEND ± TWO PLACE DECIMAL ±0.01 THREE PLACE DECIMAL ±0.005	CHECKED				
				ENG APPR.				
				MFG APPR.			Section	
			INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
PROPRIETARY AND CONFIDENTIAL				COMMENTS:				
THE INFORMATION CONTAINED IN THIS <INSERT COMPANY NAME HERE>. ANY		MATERIAL 6061-T6 Al	Contact for questions: Reuben Aronson	SIZE	DWG. NO.		REV	
WITHOUT THE WRITTEN PERMISSION OF <INSERT COMPANY NAME HERE> IS PROHIBITED.	ASSY	USED ON	FINISH	rma@mit.edu 781-710-1890	A	section		
	APPLICATION		DO NOT SCALE DRAWING		SCALE: 1:4	WEIGHT: 108 lbs	SHEET 1 OF 1	
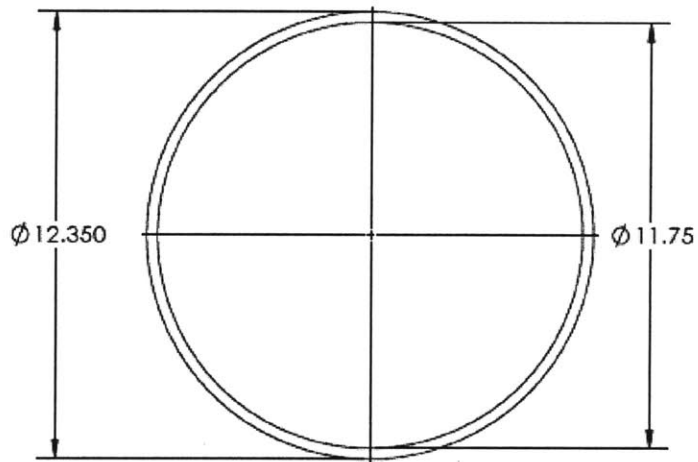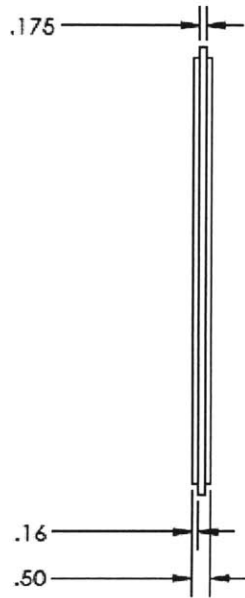
.175

.16

.50

Ø12.350

Ø11.75

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE			
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ±0.01 THREE PLACE DECIMAL ±0.005	DRAWN			TITLE:		
			CHECKED					
			ENG APPR.				Inner Disc	
			MFG APPR.					
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.					
			COMMENTS:					
		MATERIAL 6061-T6 Al				SIZE A	DWG. NO. inner disc	REV
NEXT ASSY	USED ON	FINISH						
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:4	WEIGHT:	SHEET 1 OF 1

5   4   3   2   1

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