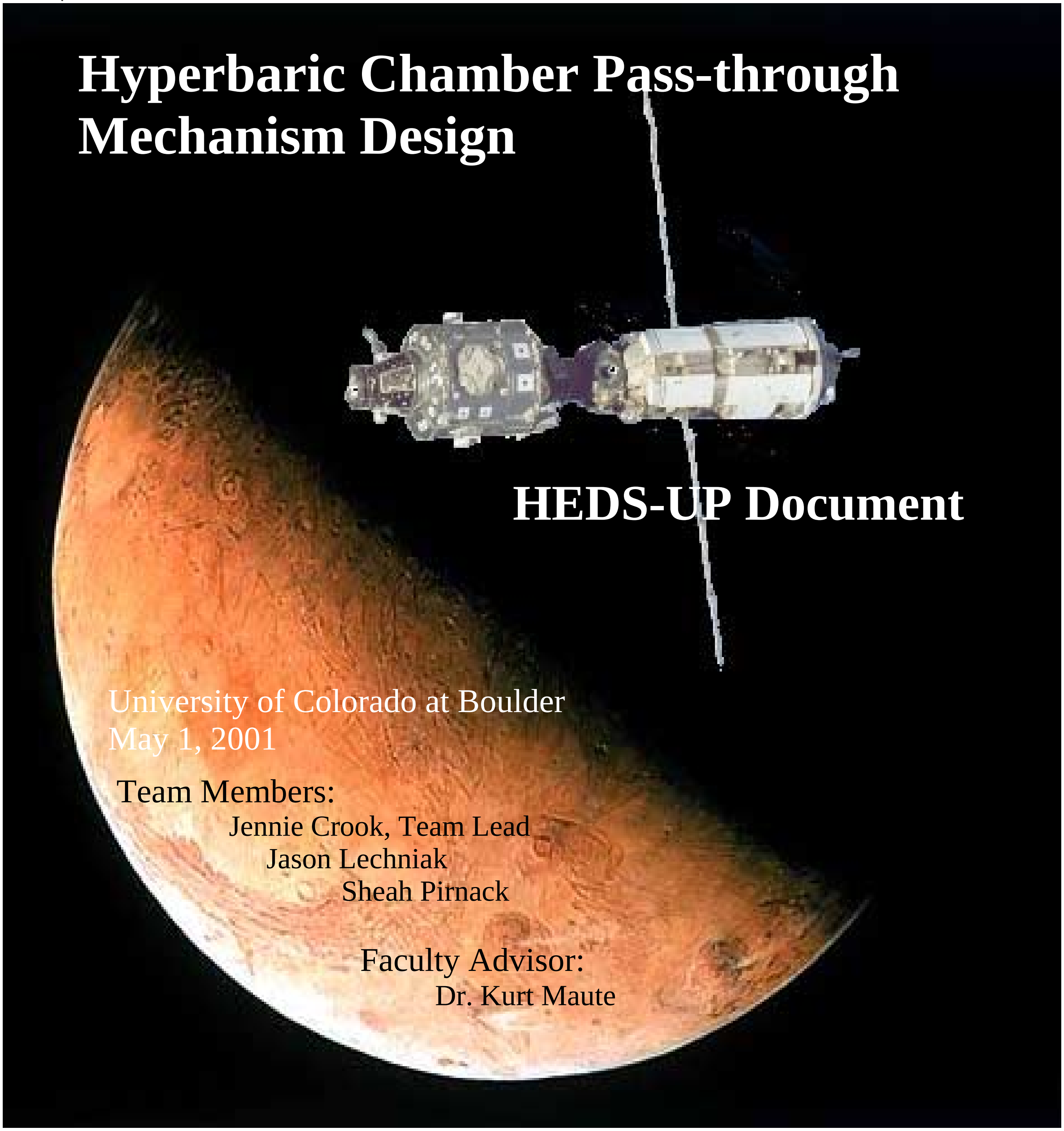
Hyperbaric Chamber Pass-through Mechanism Design



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Abstract

This project encompassed the design, analysis, fabrication, and testing of a prototype pass-through mechanism to be used on a portable hyperbaric chamber in space. The contents of this report include the scope of the project requirements, how they were met, and improvements for future studies. The main objective of the design was to build a prototype of a mechanism to allow supplies to be passed through the hatch of a hyperbaric chamber and retrieved by the patient, without depressurization. A feasible design, one which could be built while upholding the project budget and schedule, was created using IDEAS CAD software. The design was analyzed and manufactured by the team, then assembled and tested. Following a brief setback in testing, the pass-through mechanism was confirmed to seal effectively and maintain pressure. Strain gages placed in critical stress areas indicated an increase in strain with pressure and a decrease in strain with depressurization; however, no unbearable strain was reached. A total pressure of 56psi was achieved during testing. The pass-through mechanism performed optimally while withstanding the safety margin pressure.

Table of Contents

1.0 Introduction 4

2.0 Approach to the Problem 4
2.1 Background 4
2.2 Application 4
2.3 Design Problem 5
2.4 Approach 5
2.4.1 Design Phase 5
2.4.2 Building Phase 5
2.4.3 Testing Phase 5

3.0 The Design 5

4.0 Results: Theoretical Design Analysis 7
4.1 Materials / Parts 7
4.2 Static Loading Analysis 9
4.2.1 Pressure Assumptions and Theory 9
4.2.2 Chamber Cap 9
4.2.3 The Pressure Vessel (Inner Cylinder) 10
4.2.4 Bolt Calculation for Latch Attachment 11
4.2.5 Bolt Calculations for Flange Attachment 11
4.3 Sealing Capabilities 12
4.4 3-D CAD Model (In IDEAS) 12
4.5 Finite Element Analysis 12

5.0 Results: Building Phase 13

6.0 Results: Testing Phase 13
6.1 Results 14

7.0 Lessons Learned / Future Studies 15
7.1 Lessons Learned 15
7.2 Improvements to Design 15
7.3 Unlimited Resource Solutions / Future Studies 16
7.3.1 Elliptical body and top 16
7.3.2 Honeycomb Construction of heavy parts 16
7.3.3 Woven Kevlar Inflatable 16

8.0 Conclusions 16

9.0 Acknowledgments 17

10.0 References 17

11.0 Appendix: Critical FEM 18

1.0 Introduction

Maintaining a healthy crew on the long trip to Mars will be one of the many challenges faced by those who plan the mission. A solution to many of the medical needs encountered in space is a hyperbaric chamber. A hyperbaric chamber is a pressurized vessel containing excess O_2 that can be used to decrease the recovery time for almost any injury. The project that was completed for the HEDS-UP Forum involved the design, building and testing of a prototype pass-through mechanism that can be attached to the door of a hyperbaric chamber to be used on the International Space Station. The mechanism will permit medical supplies to be passed into the chamber and retrieved by the patient, without depressurization of the chamber. Although the initial design is for use on the international space station, any space application, including a trip to Mars, is feasible for the chamber.

2.0 Approach to the Problem

2.1 Background

A hyperbaric chamber is a pressurized vessel used for medical purposes containing excess O_2. Chambers are normally kept at pressures greater than atmospheric. Hyperbaric Oxygen Therapy (HBO) is a type of medical treatment that is effective for many clinical conditions [3].

2.2 Application

NASA Johnson Space Center is in the process of testing a prototype chamber for use on the International Space Station (see Figure 2.2-1). The chamber may also be used at NASA's Weightless Environment Training Facility (WETF, the water tank containing a mockup of the shuttle's payload bay, where astronauts train in a simulated micro-gravity environment) [4].

Figure 2.2-1. Hyperbaric Chamber Prototype [1]

2.3 Design Problem

Design a pass-through mechanism to be added to the door of the prototype of the hyperbaric chamber. The mechanism will permit medical supplies to be passed into the chamber and retrieved by the patient, without depressurization of the chamber. Ideally, the mechanism will be removable, attaching to the hatch only when functionally needed. The pass-through module should be approximately coffee-can sized and be able to withstand pressures up to 3.6 atm (safety factor)[1].

2.4 Approach

2.4.1 Design Phase

A number of designs were considered in order to optimize the pass through mechanism for its intended purpose. Each design was evaluated through several criteria to determine the designs' relative functionality when integrated into the system. The optimum design was based on:

1. Ease of integration into the system
2. Most feasible for use in zero-g environment
3. Minimum maintenance requirements
4. Cost and feasibility of construction

Once a design was decided upon, it was analyzed through several avenues:

Material Property Considerations
Static Loading Analysis
Variable Pressure and Contact Force Analysis
Sealing Capability / Analysis
3-D CAD Model (In IDEAS)
Integrated System Analysis
Finite Element Analysis

2.4.2 Building Phase

It is essential to build a prototype in order to determine if the design is functional and operative. The building phase required the necessary parts to be on-hand or built by the engineering students working on the project (in order to stay within the small budget of the project).

2.4.3 Testing Phase

The prototype was tested through several avenues:

Structural Strain Measurements - to confirm finite element analysis
Simulation of usage over a range of pressures
Testing for sealing confirmation

3.0 The Design

Figure 3.0-1 is a schematic of the design chosen by the team, assembled in its entirety.

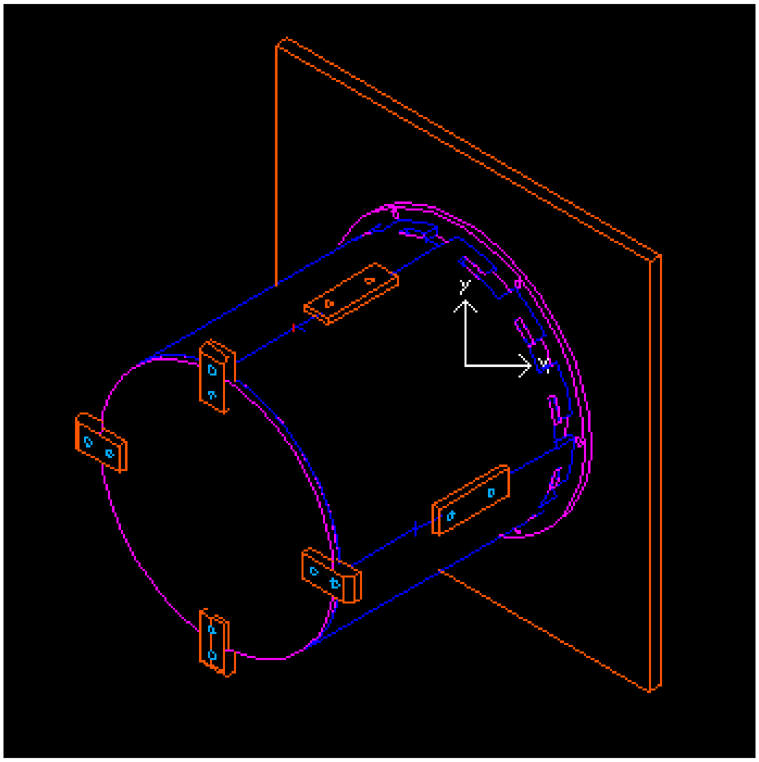


Figure 3.0-1 Assembly Schematic of the Design

The design incorporates a dual-cylinder configuration. The interior cylinder maintains the pressure seal, and the outer cylinder houses the latch attachments, the interior cylinder, and connects to a flange on the chamber door. Because of the complications associated with screw-on devices that require strength in a 0-g environment, the design has a nearly effortless toothed-locking-mechanism (to attach to the hatch itself), instead of a more standard flange design. The exploded view of all parts and the corresponding bill of materials can be found in figure 3.0-2.

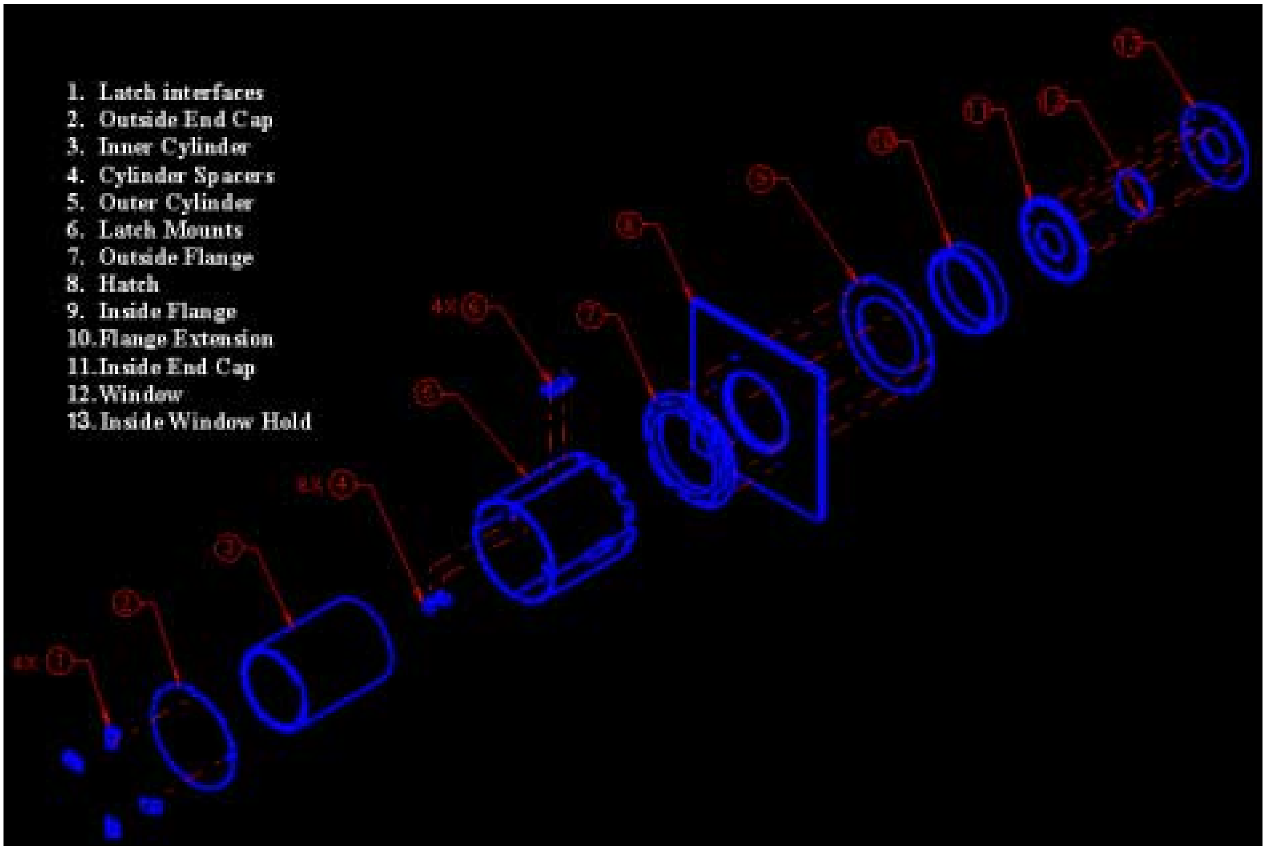


Figure 3.0-2 Exploded view of all fabricated parts

4.0 Results: Theoretical Design Analysis

The analysis performed in this section includes all off-the-shelf part specifications and material properties. Simplifying assumptions were made concerning the homogeneity of the materials and the equal distribution of forces resulting from pressurization (uniform pressure).

4.1 Materials / Parts

There were many factors that were taken into consideration in the process of designing this small pressure chamber. Two of the most important considerations were the availability of materials that were within the budget of the project, and the machinability of parts. The inner and outer cylinders are off-the-shelf Aluminum 6061-T6 stock piping. The inner cylinder was purchased from The Marmon/Keystone Corporation for $80.00. All the sheet metal components, such as the inner and outer flanges and the doors, are made of Aluminum 7075-T6. The outer cylinder and the sheet metal were purchased from Alreco for a total cost of $297. Due to a machining error, an additional slab of Aluminum had do be purchased from Alreco for $87. Material characteristics for both types of Aluminum are shown in table 4.1-1. The material properties were obtained from the MatWeb Materials Property Database website [5].

Table 4.1-1 Material Properties for Machined Aluminum Parts

Material Characteristics	Aluminum 7075-T6	Aluminum 6061-T6
Density (g/cc)	2.81	2.7
Tensile Strength, Ultimate, MPa	570	310
Tensile Strenth, Yield, MPa	505	275
Elongation %; break	11	12
Modulus of Elasticity, Gpa	72	69
Notched Tensile Strength, MPa		324
Ultimate Bearing Strength, MPa		607
Bearing Yield Strength, Mpa		386
Poissons Ratio	0.33	0.33
Fatigue Strength, MPa	160	95
Fracture Toughness, MPa-m(1/2)	29	29
Machinability, %	70	50
Shear modulus, GPa	26.9	26
Shear Strength, MPa	330	205

The attachment parts, such as the nuts, bolts and washers are all made from steel and were purchased from a local hardware store for under $10.00. The specifications for the bolts are given in table 4.1-2 and were obtained from the book *Machine Design: An Integrated Approach [7]*.

Table 4.1-2 SAE Specifications and Strengths for Steel Bolts [7]

SAE Grade Number	Size Range Outside Diameter (kpsi)	Minimum Proof Strength (kpsi)	Minimum Yield Strength (kpsi)	Minimum Tensile Strength (kpsi)	Material
1	0.25-1.5	33	36	60	low or medium Carbon
2	0.25-0.75	55	57	74	low or medium Carbon
2	0.875-1.5	33	36	60	low or medium Carbon
4	0.25-1.5	65	100	115	medium carbon, cold drawn
5	0.25-1.0	85	92	120	medium carbon, Q&T
5	1.125-1.5	74	81	105	medium carbon, Q&T
5.2	0.25-1.0	85	92	120	low-carbon martensite, Q&T
7	0.25-1.5	105	115	133	medium-carbon alloy, Q&T
8	0.25-1.5	120	130	150	medium-carbon alloy, Q&T
8.2	0.25-1.0	120	130	150	low-carbon martensite, Q&T

The Latches are also constructed of stainless steel. The latches were catalog parts purchased from Southco, costing almost $100 for all 4 latches. The latch strength specifications are shown below in table 4.1-3 [9]. The entire latch mechanism is made up of 300 series stainless steel and all parts within the latch are passivated. A picture of the latch is shown in figure 4.1-1 [9].

Table 4.1-3 Latch Specifications

Southco Draw Latches (Latched at center of radius)	
Maximum working load	700 lbs
Average ultimate load	1100 lbs

Figure 4-1-1 Picture of the latch mechanism [9]

4.2 Static Loading Analysis

4.2.1 Pressure Assumptions and Theory

The function of the pass-through mechanism, a small pressure chamber, dictates that it will withstand an internal pressure of 3.6 atmospheres (this value includes a safety margin). The main structural concern is the distributed load resulting from a differential pressure from the 'inside' of the chamber to the 'outside'. The gage pressure, p_g, is a measure of this differential pressure as shown in equation 4.2.1-1.

(4.2.1-1) $p_g = p - p_{atm}$ = 3.6atm – 1 atm = 2.6 atm = 38.22psi = .26337MPa

The pressure chamber is built to withstand a total pressure of 3.6atm (52.92psi or .36467MPa). For the analysis of the chamber, the gage pressure will be used instead of total pressure. The chamber, however, will theoretically be able to handle the total pressure inside a vacuum. This is important because the mechanism will be in space, and although it will be kept in a module that is pressurized to atmospheric conditions, an emergency situation may cause that the pressure outside the chamber to approach 0, in the vacuum of space.

4.2.2 Chamber Cap

The exterior cap of the pass-through mechanism is attached to the containment unit by 4 latches (specifications given previously in table 4.1-3). The equivalent force, F, produced by the differential pressure is given in equation 4.2.2-1. This force represents the distributed force acting on the pressurized portion of the cap (assuming uniform pressure along the area). The force calculation utilizes the actual gage pressure exerted on the chamber and the interior diameter of the containment unit, since the pressure will only act on the sealed interior portion of the mechanism, not the entire surface area of the cap.

(4.2.2-1) $F = pA = (38.22\text{psi})\pi(7.25\text{in}/2)^2 = 1577.82$ pounds

The point at which the line of action of the equivalent force intersects the surface, or the center of pressure, is in the center of the circular cap. The free body diagrams in figure 4.2.2-1 reflects the equivalent force, F of the pressure as the large arrow, and the resultant force of the latches as the 4 arrows on the opposing circumference of the cap.

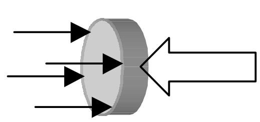


Figure 4.2.2-1 FBD of the Cap or Top

According to the calculation made in Equation 4.2.2-1, each of the four latches has to withstand a quarter of the pressure equivalent force (1577.82 pounds), providing that they are placed symmetrically about the center. The latches have to withstand a minimum of 394.46 pounds in order to satisfy the safety pressure, plus an additional error factor since it is unlikely that will be perfectly symmetrical about the perimeter. The latches chosen met and exceeded this requirement, as can be seen from the specifications given in table 4.1-3.

4.2.3 The Pressure Vessel (Inner Cylinder)

When in use, the pass-through mechanism is essentially a thin-walled pressure vessel that is subject to loading in all directions. In general, "thin wall" refers to a vessel having an inner-radius-to-wall–thickness ratio of 10 or more (r/t • 10) [7]. In the design of the inner pressure cylinder, the cylinder inner diameter is 7.25 inches and the thickness of the wall is 0.375. Therefore the radius/wall-thickness ratio is approximately 10, so a thin-walled pressure vessel analysis is valid. The inner cylinder is subject to normal stresses in the circumferential, or hoop, direction and in the longitudinal, or axial, direction. Both of these stress components exert tension on the material. Loads are developed by the uniform hoop stress, •$_1$, acting through the vessel wall. Figure 4.2.3-1 illustrates the basic free body diagram of the interior cylinder.

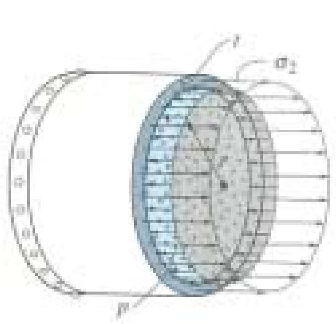


Figure 4.2.3-1 FBD of Interior Cylinder (thin-wall pressure vessel)

Assuming equilibrium in the x direction (the sum of the forces in x are 0), the hoop stress can be calculated by equation 4.2.3-1, where p is the pressure, r is the inner radius and t is the thickness of the cylinder.

(4.2.3-1) $$\sigma_1 = \frac{pr}{t}$$

The maximum hoop stress on the inner cylinder was calculated from the safety pressure of 51.84psi (3.6atm) to be 511.56 psi (3.52516Mpa). If the gage pressure of 2.6 atm is used, the maximum hoop stress is 369.46psi (2.5459Mpa), using the thin-walled pressure vessel assumption. The longitudinal stress, •$_2$, acts uniformly throughout the wall and is calculated from equation 4.2.3-2. The mean radius is assumed to be approximately equal to the inner radius with the requirement that there is equilibrium in the y direction.

(4.2.3-2) $$\sigma_2 = \frac{pr}{2t}$$

The longitudinal stress is half of the hoop stress. Therefore it is approximately 250.56 psi for the inner cylinder, again with a thin walled pressure vessel assumption.

The maximum hoop stress that the pressure exerts on any given section of the interior cylinder was calculated to be 511.56 psi (3.52516Mpa). The material characteristics of the cylinder, or Aluminum 6061-T6, are shown in table 4.1-1. Given a value for fatigue strength of 160Mpa for the cylinder, there is a wide margin between the allowable stress and the stress exerted on the cylinder.

4.2.4 Bolt Calculation for Latch Attachment

The steel latches attach with mounting interfaces to the exterior cylinder in a single-shear connection (lap joint) with two quarter-inch steel bolts. In this analysis, the friction between the members and the mounting interfaces is neglected because it can be assumed that the nuts are not tightened significantly enough to create more than a negligible friction force.

The bolts connecting the latches are subject to shear forces resulting from the pressurization of the mechanism, causing a load to be applied to the cap of the pressure chamber, which propagates through each latch. The free body diagram of the internal uniform shear stress acting on each bolt is shown in figure 4.2.4-1 [7]. Since there are two bolts per latch, each bolt is required to hold a shear stress of 197.23 psi, or half the force on each latch. Equation 4.2.4-1 gives the relationship that the bolts must satisfy in order to effectively stabilize the latches to the outer cylinder given the pressurization of the mechanism.

(4.2.4-1) $$\tau_{allow} = \frac{P}{A}$$

Each bolt is a quarter of an inch in diameter, resulting in a area, A, of 0.0490874 inches squared. This requires that each bolt have an allowable shear stress, τ_{allow}, of at least 4.018 kpsi. Therefore each bolt would easily be able to hold the requirement of .19723 kpsi.

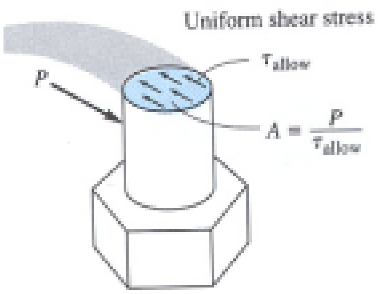


Figure 4.2.4-1 FBD of internal shear on each bolt

4.2.5 Bolt Calculations for Flange Attachment

There are a couple of issues concerning the calculation of the number of bolts and the type of bolts that are needed to attach the interior and exterior flanges together (in compression). First, the threading must be able to withstand the pre-load on each bolt, as well as the load associated with pressurization. In order to avoid breaking contact between the nut, washer and the contact surface (flanges), the pre-load has to be equal to or greater than the load imposed upon each bolt when pressurization occurs. This is important for maintaining an effective seal on the mechanism.

The pre-load on each bolt and the number of bolts depends on the total force applied, the length of each bolt, and the fastener's material, size (cross-sectional area), thread area, and percentage of the minimum proof strength (as given in table 4.1-2). Generally, for statically loaded assemblies, such as the pass-through mechanism, a pre-load that generates bolt stress as high as 90% of the proof strength, Sp, can be used. Assuming that the bolts are suitably sized for the applied loads, these high pre-loads make it very unlikely that the bolts will break during use if they do not break while being tightened. The total length of the bolts, l_{bolt}, is 1.5 inches. The thread length on the bolts is the full 1.5 inches, although only the last .25 inches of threading will be used for attaching to the nut. Each bolt is a ¼ inch in diameter, d, giving a cross-sectional area, Ab, of 0.0490874 inches squared. Course threaded bolts are used (20 threads/inch), with a thread tensile stress area, At, of approximately 0.0318 inches squared [7}. Therefore the preload can be as high as 944.46 pounds, using equation 4.2.4-1.

(4.2.5-1) Preload=Fi=(90%)SpAt

The material, construction, and stiffness of each bolt was taken into account when determining the resultant loads in the bolt and the maximum interior tensile force that each bolt could withstand. It was found that each bolt has to be able to withstand a force of 576.825 pounds. Given the force and the thread area that it acts upon, the actual stress on the bolt may be calculated. This stress was found to be 18,139.15psi for each bolt. The maximum tensile stress allowed in each bolt is approximately 30,717.76 psi. There is a large margin between allowable and actual stress and the safety factor against yielding was calculated to be 1.17196 with a separation point of 7.295 given the loads applied.

4.3 Sealing Capabilities

Elastimetric Viton o-rings were used for sealing the pass-through mechanism. Each sealing interface consisted of an o-ring installed in a gland (cut to specifications). O-ring seals are very dependable and are generally very rugged. Static sealing is needed in the design of the pass through mechanism and o-rings have been proven to seal at high pressures despite slight irregularities in the sealing surfaces, when implemented in a static seal. O-rings are also easily maintained, are compact and lightweight, and no adhesives are needed. Another advantage of o-rings is that if failure does occur, it is usually gradual and detectable, which is an important safety consideration. The metal parts that are integrated with the o-rings were all finished on a lath with a smooth surface in order to potentially increase the sealing capability.

4.4 3-D CAD Model (In IDEAS)

A model of each part to be machined for the hyperbaric pass through mechanism was drawn in IDEAS Master Series 8. An assembly drawing was then completed in order to confirm that all the parts would interface correctly if built to specification. Technical drawings were produced for geometric specifications and tolerances and then each part was fabricated. The final assembly drawing is shown previously in Figure 3.0-1.

4.5 Finite Element Analysis

Finite Element Modeling and Analysis were performed on all loaded parts to confirm their structural integrity and verify that the pass through mechanism would hold under the forces associated with pressurization. The Finite Element Analysis was done in IDEAS under the simulation module. First each part was drawn up as simply as possible. Second, boundary conditions were applied and forces associated with pressurization were included in the simulation. The model was free-meshed and then evaluated for any possible errors. The model was broken down into solid elements with an average size of a ¼ inch. A solution was run to obtain strain, stress and deflection for each part. The two most critical sections on the structure, the flange teeth/cylinder teeth and the latch mounts, were further analyzed to insure that they would not fail. Illustrated results of the Finite Element Analysis for the critical points can be found within the Appendix.

5.0 Results: Building Phase

During the fabrication process the team members, along with the help of a few experienced machinists, were able to machine all the necessary parts. All machining was completed in campus facilities.

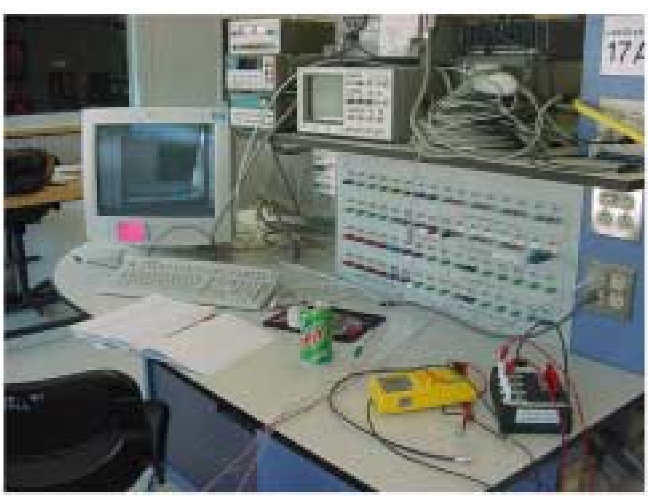
6.0 Results: Testing Phase

All testing was completed at the Integrated Teaching and Learning Laboratory (ITLL), a facility at the University of Colorado Engineering Center.

The Pass Through Mechanism was assembled with pre-loads on the latches and the flange bolts. The preload is essential for sealing purposes (in order to keep the interfaces tight with no leaks through the o-rings).

The interior door was clamped because the design requires that there would be a pressure match between the interior of the cylinder and interior of the chamber. The chamber itself could not be simulated, so it was not possible to obtain a pressure match.

In order to safely conduct the experiment, the chamber was partially filled with water, leaving some volume for the addition of compressed air. Testing with water is safer than air because it allows high pressures to be obtained without extensive fluid compression. In addition to diminishing safety concerns, it allows for easy leak detection. Air was then pumped into vessel until the system reached and exceeded its design pressure.



Before testing, strain gages were applied at two critical stress areas on the pass through mechanism. Reference (dummy) gages were also applied to non-stressed points of equivalent material to serve as the fourth resister in each Wheatstone bridge. A Lab-station (shown above) at the ITLL was configured to obtain readouts of the test data.

6.1 Results

The initial phase of testing was to check for leaks at all interfaces of the module. The first time the chamber was filled with water, significant leaks were detected at the weld on the interior flange. Due to time constraints, it was not feasible to reconstruct the weld. Epoxy, rated to 2500 psi, was applied to the inner surface of the weld interface, in an attempt to maintain a proper seal.

Once the epoxy set, the chamber was refilled with water and checked for leaks. No leaks were detected, so the pressurization of the chamber commenced. The chamber attained its maximum pressure, 56 psi, about 30 seconds into the test. This pressure was maintained for approximately 45 seconds, at which point the air compressor was turned off and the chamber was allowed to decompress.

The results obtained from the strain gages indicated that there was micro-strain on the latches and the outer cylinder teeth. The strain on the latch, seen in figure 6.1, increased abruptly during pressurization, reaching a plateau once the desired pressure was attained. The rapid increase in strain is a result of the manner in which the latch hooked onto the exterior door. The single point force on the latch from the top of the chamber was a quarter of the total force on the exterior door, since there were four latches. The slight dip in the trend of the data is likely due to slippage of the bolts that attach the latch to the outer cylinder.

The strain on the aluminum tooth gradually increased with pressure, as shown in figure 6.2. The steady increase is due to a combination of two factors. First, the geometry of the tooth is such that the strain gage had to be mounted in a place where it would only detect an indirect load. Second, the total load is distributed among 12 teeth, so the single tooth where the gage was mounted was only affected by a twelfth of the load.

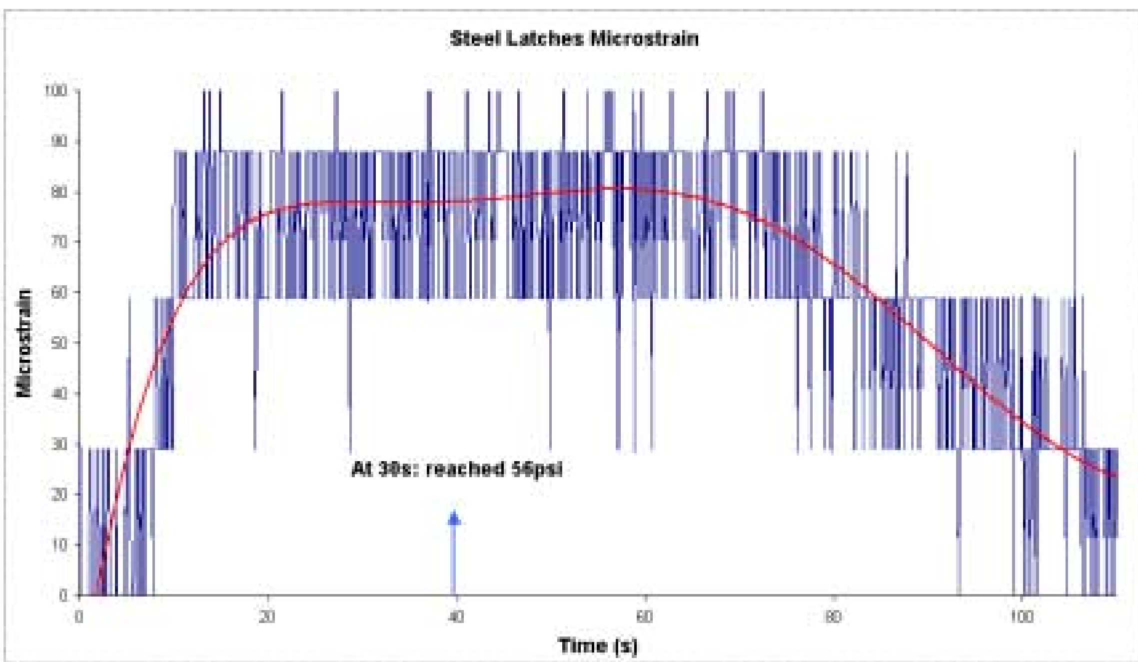


Figure 6.1 Microstrain on the Steel Latch

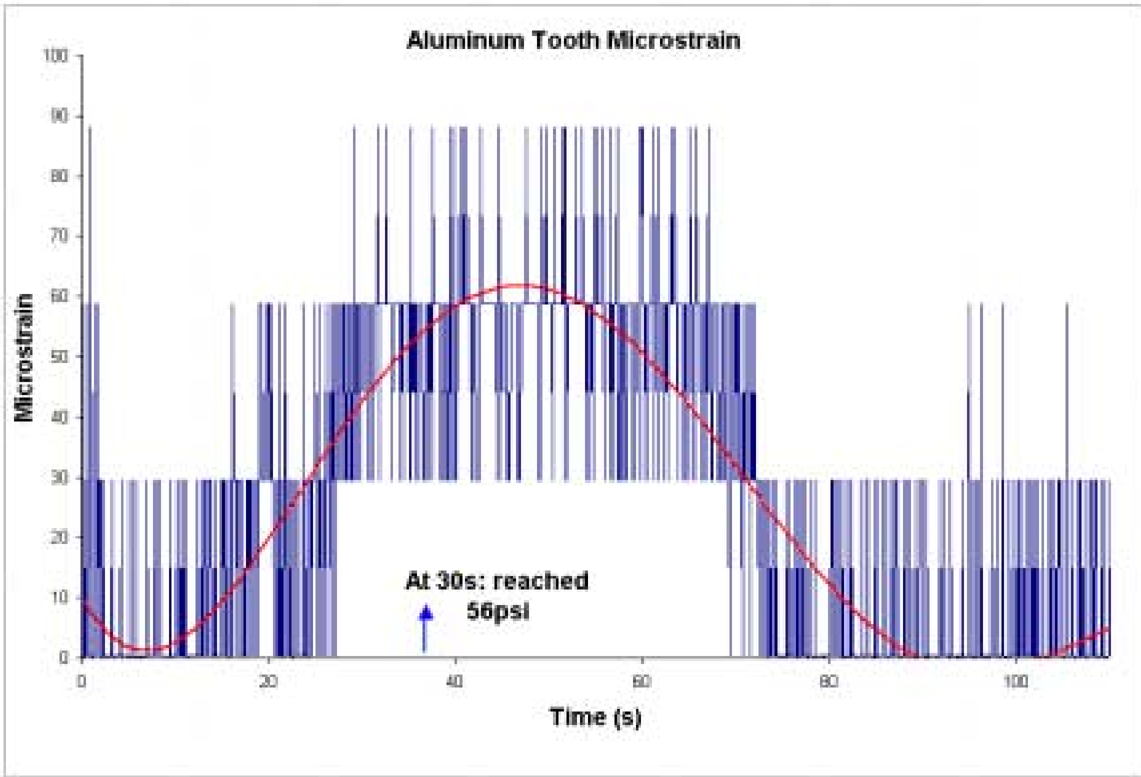


Figure 6.2 Microstrain on the Aluminum Tooth of Exterior Cylinder

7.0 Lessons Learned / Future Studies

7.1 Lessons Learned

- Everything that affects a design cannot possibly be thought of before building a prototype (now we know why sometimes several production models are built before the flight model)
- Ordering parts in small quantities from manufacturers or distributors is often hard, if not impossible
- Allow more time for improvements and changes to the design
- Strain gages are very difficult to correctly and effectively apply
- Money does not go very far on a prototype relying on a limited student project budget

7.2 Improvements to Design

- Do not have an opening on both sides, have only one opening on the side to be attached to the hatch (more coffee-can like instead of latches and a cap), this would allow for less concern with the sealing, leading to a more reliable design

- Obtaining a preload on the latches was physically strenuous (difficult to close)- we didn't think of the implications of the magnitude of preload that was needed
- It was difficult to determine whether the preload was distributed evenly among all four latches

- Use a Metal Inert Gas (MIG) weld instead of a tungsten inert gas (TIG) weld to get better penetration into the crack and maintain better contact with the surfaces ("If you want something done right, do it yourself" – this was the only thing that we didn't do ourselves)
- While positive pressure sealing would have been more difficult, it would be a more reliable design
- If this design were actually implemented for the situation it was built for, the patient would have to deal with o-rings falling out of the mechanism when the interior door was opened: this is a problem that would have to be solved using a integrated sealing surface attached to the inner flange or door
- Sections could be cut out on the exterior cylinder to reduce the weight
- Make a single body (one cylinder) that could both be the sealed containment unit and a locking mechanism

7.3 Unlimited Resource Solutions / Future Studies

7.3.1 Elliptical body and top

Building the body and top in an elliptical formation, as opposed to circular, would allow all seals to be positively sealed with the pressurization of the mechanism. This was not feasible within time constraints and budget constraints, so the design that was built only using off-the-shelf circular, cylindrical piping and aluminum sheets developed and refined on the lathe for the top.

7.3.2 Honeycomb Construction of heavy parts

A relatively simple way to make the mechanism less massive would be to construct the 'tubing' and caps from hollowed-thin cylinder and sheets with honeycomb core for strength and stiffness. This would significantly reduce the weight of the mechanism and is within technological means today. This was not feasible for the initial prototype because of time and budget constraints.

7.3.3 Woven Kevlar Inflatable

With an inflatable structure that can withstand loads associated with pressurization, the mechanism could be significantly less massive and could be easily stowed.

8.0 Conclusions

The senior design project consisted of the design, analysis, manufacturing, and testing of a prototype pass-through mechanism for a space-compliant portable hyperbaric chamber. The development of such a hyperbaric chamber would aid in preserving the health of astronauts in space, especially in long-term applications such as a manned-mission to Mars. The overall design was successful and through the setbacks encountered, many lessons were learned. In the course of assembling and testing the mechanism, an understanding of the importance of building a prototype was realized. Problems such as the o-rings falling out, the latch pre-load issues, and weld failures became evident. A prototype makes apparent previously un-addressed obstacles, allowing for improvements in the design. Concerning the o-ring and the latch difficulties, a second design could incorporate a single door plan where sealing would only be necessary on a single face of the cylinder. Instead of the TIG weld a MIG weld would be recommended. Many problems would be avoided if the parts could be made with mold casting technology. Addition future studies would entail using more advanced lightweight materials and refining the design to be more convenient for astronaut use.

9.0 Acknowledgments

We would like to thank the following people for time they gave in the tasks outlined below:

Jim Ray, Student:	Helped machine several parts on the lathe
Lee Thornhill, Student:	Gave advise on machining and welding
Al Williams, Student:	Helped machine several parts: threading and dremmel detail
Chris Muhler, Student:	Helped on the Assembly Drawings
Matt Rhode, Aerospace Machine Shop Manager:	Assisted in machining several parts as well as giving access the Aerospace Machine Shop
Bill Ingino, ITLL Machine Shop Manager:	Assisted in machining several parts as well as giving access the ITLL Machine Shop
Kurt Filsinger, LASP Facility Manager	Gave advise on o-ring seals
Walt Lund, Aerospace Electronic Shop	Helped provide information and supplies concerning electronics
Catherine Flanders & Trudy Shwartz: ITLL Modulation Integration Engineers	Helped set-up the lab station and electronic for testing

10.0 References

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11.0 Appendix: Critical FEM