HYDROGEN TANK DESIGN FOR THE MATERIAL HANDLING INDUSTRY

Design Group #8

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EXECUTIVE SUMMARY

This report presents the work of the design group on its project in hydrogen tank design for the material handling industry. Presently, the fuel cell system in a hydrogen-powered fork lift truck is contained within a package that replaces the lead-acid battery of the lift truck. Eventually the fuel cell system, including its compressed hydrogen tank, will be distributed optimally around the lift truck. The design group was tasked with creating three proposals for compressed hydrogen tanks that could be integrated into a Raymond Corporation fork lift truck.

During the fall semester research phase the project included; the determination of basic design requirements, a review of applicable design standards, and the generation, evaluation, and selection of design concepts. Among other requirements, it was determined that the tanks must store approximately 1kg (43L) of compressed hydrogen at 5,000 psi and that they must be composed of steel to compensate for the lost weight of the lead-acid battery. It was decided that design standard compliance would focus on the DOT Part 178 and HGV5 design standards. The three design concepts chosen for detailed design were; a traditional cylindrical vessel, an assembly of nested high pressure tubing coils, and a welded assembly of rectangular box tubing.

During the spring semester design phase the project included an iterative design process for each design concept and the creation of functional specifications for the finished design proposals. Consultation with industry and the acquisition of quotes were also achieved.

Two versions of the cylindrical vessel were designed; one DOT-3A, 316L stainless steel cylinder with a ½ inch wall and one DOT-3AA 4130 Q&T steel cylinder with a ¼ wall. Quotes for the manufacture of both cylinders were acquired from Taylor-Wharton. The tubing coil tank was designed using 316L stainless steel high pressure tubing and terminated with fittings and caps by Swagelok. Tubing quotes were acquired from both Swagelok and Handy & Harman Tube Company. The rectangular tank involved the design of a custom box tubing cross-section. The material used was 4130 Q&T steel. A workable cross-section was developed and the full assembly designed. The performance of the various junctions in the assembly were analyzed and found to be sufficient. Consultation with Louisiana Steel was conducted during the design. The functional specifications developed for each design proposal presented the physical, manufacturing, and operational requirements dictated by the designs. This revealed in what ways the design proposals met or failed to meet the basic requirements as well as those of the DOT and HGV5 design standards.

In general, the project achieved all of its critical objectives; three design proposals with functional specifications were delivered; history of the design iterations and the engineering evidence of their success was documented; and a quote for the manufacture of at least one design proposal was acquired.

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1. INTRODUCTION

This report will present the spring semester work of the design group on its project in hydrogen tank design for the material handling industry. This section will present the problem addressed by the design group, the description of the project and its requirements, and the scope and format of this report.

1.1. PROBLEM STATEMENT

For years now, the material handling industry has been experimenting with hydrogen fuel cell-powered fork lift trucks. Such lift trucks improve warehouse productivity in high throughput applications because compressed hydrogen tanks can be refueled in only a few minutes, while the lead-acid batteries of the lift trucks used throughout the industry can require an entire day to charge and cool [1]. Material handling is an ideal environment for hydrogen technology as compared to the mass transportation industry because it does not require refueling stations to be distributed over large geographic areas. Fork lift trucks do not travel hundreds of miles from a warehouse, therefore it is only necessary to provide refueling facilities at the warehouse. So far the design of hydrogen fuel cell-powered lift trucks has focused strictly on the replacement of the lead-acid batteries with fuel cell systems of the same size, weight, and energy capacity [1]. In the future, the components of the fuel cell system will be optimally distributed and integrated into the lift truck, which will be designed to take full advantage of the modular nature of the fuel cell system [1]. The problem addressed by the design group is how to design a compressed hydrogen tank, part of the fuel cell system, to be integrated into a lift truck. The client of the design group, Raymond Corporation, requires three design proposals for compressed hydrogen tanks that could be integrated into the design of its lift trucks.

1.2. PROJECT DESCRIPTION

The primary goal of the project was summarized in the preceding problem statement; the project shall result in three design proposals for compressed hydrogen tanks that could be integrated into the design of Raymond Corporation lift trucks. The additional project requirements are below.

The project shall

- Involve a thorough review of all applicable design standards and a determination of what can and cannot be achieved in each design proposal to comply with those standards
- Involve the generation, evaluation, and selection of design concepts to be iterated into the final design proposals
- Include, as necessary throughout its duration, consultation with industry experts, suppliers, and manufacturers
- Include a history of all design iterations and evidence of the capacity of each design proposal to meet the design requirements
- Include the generation of a functional specification detailing the technical requirements for each of the final design proposals
- Include acquisition of a quote for tooling and manufacture in volume for at least one of the final design proposals

The final project deliverables consist entirely of the three design proposals with standards review and functional specification included. For at least one of the design proposals a quote is included. It is important to note that there are no physical deliverables; no prototypes were constructed.

1.3. PROJECT SCOPE

In general, the scope of the project has been conveyed in the preceding problem statement and project description; the project is focused entirely on the compressed hydrogen tank of a hydrogen fuel cell-powered fork lift truck. The project is not concerned with any other component of the fuel cell system. It should be stated that the project will not focus any one particular model of lift truck. It follows, generally, that any design proposal should be applicable to any model of lift truck and should not involve modification of existing lift truck components. If a design proposal requires mounting of the tank to an existing lift truck component, the mounting design is assumed to not interfere with the operation of the tank and is therefore considered outside the scope of the project.

The various tasks of the project, as broken down by semester, provide additional insight into its scope and are depicted in the flowcharts of Figure 1 and Figure 2. The fall and spring semesters were broken down into research and design phases, respectively. Figure 1 illustrates the research phase that was carried out last semester. The research phase consisted primarily of standards review, generation of basic design requirements, and selection of design concepts. The broad purpose of the research phase was to flush out all aspects of the problem to be addressed by the design group in preparation for the detailed design phase. Figure 2 illustrates the design phase that was carried out this semester. This phase consisted of an iterative design process for each of the design concepts and resulted in the final, client-approved, design proposals. It should be noted that industry input was sought throughout the design phase for all design proposals.

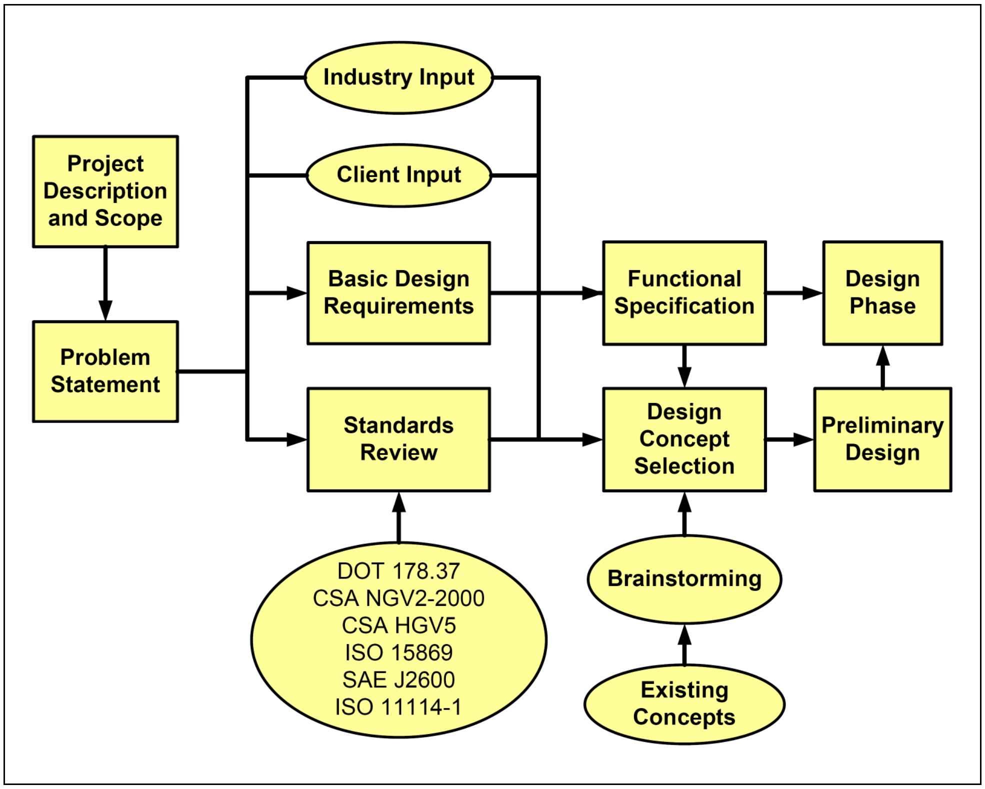


Figure 1 – Fall Semester Research Phase Flowchart

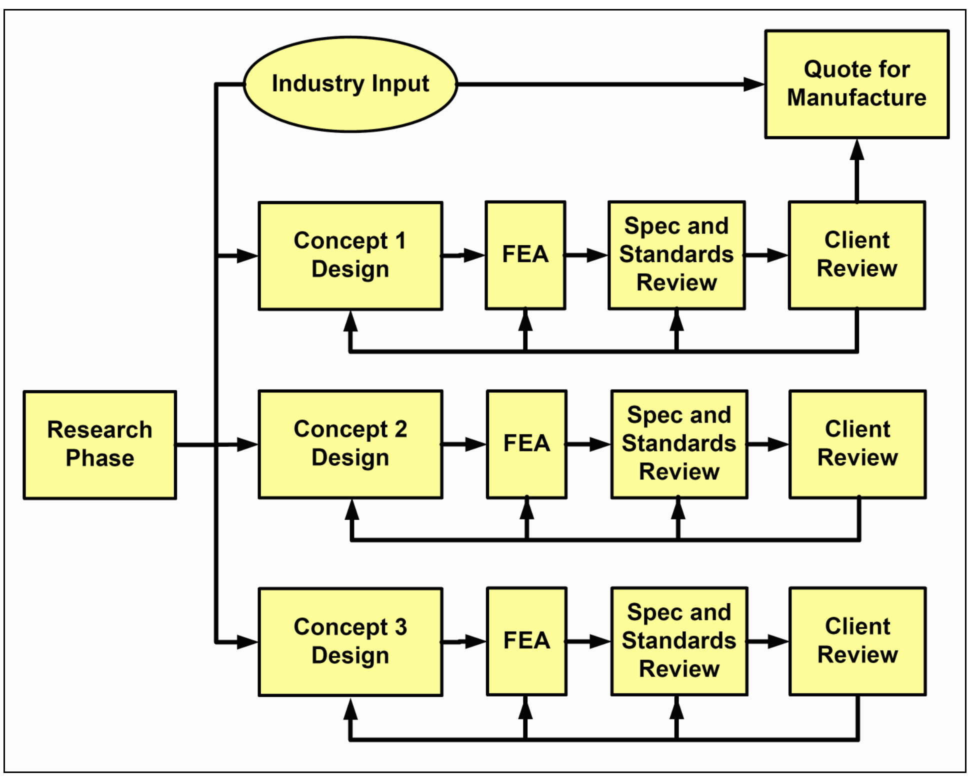


Figure 2 – Spring Semester Design Phase Flowchart

1.4. REPORT FORMAT

This report will begin by reviewing the project background information and design requirements. It will then present the three design processes and subsequent proposals outlined in the Spring Semester Design Phase Flowchart of Figure 2. The project description, project scope, and problem statement have already been discussed. The remainder of the report is broken down into the following sections;

- **Background** – This section will provide important information about the client and the properties of hydrogen, as well as review the selection of design concepts that was carried out during the fall semester
- **Design Requirements** – This section will present the design requirements for the compressed hydrogen tanks
- **Traditional Cylindrical Vessel** – This section will review the design concept, discuss the design evolution, and present the functional specifications for the traditional cylindrical vessel (concept #1)
- **Nested High Pressure Tubing Coils** – This section will review the design concept, discuss the design evolution, and present the functional specifications for the nested high pressure tubing coils (concept #2)
- **Rectangular Tank (Overhead Guard)** – This section will review the design concept, discuss the design evolution, and present the functional specifications for the rectangular tank (concept #3)

Following the above mentioned sections of the report is the conclusion, references, and appendices. The conclusion summarizes the project accomplishments and makes recommendations for any future work that might be carried out on the design proposals.

2. BACKGROUND

This section of the report will provide important background information regarding the client, Raymond Corporation, and the properties of hydrogen.

2.1. RAYMOND CORPORATION

Raymond Corporation is the client of the design group and also the industry sponsor of the project. Raymond Corporation has a long history in the material handling industry that began in 1922 with the acquisition of Lyon Iron Works in Greene, New York by George Raymond, Sr., an industrial engineer from Brooklyn. He refocused the efforts of Lyon, eventually Raymond Corporation, on the development of new kinds of material handling equipment [1]. In the decades since its founding, Raymond has remained in the forefront of fork lift truck development through continual pioneering. From walk-behind stackers to the sophisticated 9000 Series *Swing-Reach* seen at left, Raymond produces a full line of electric fork lift trucks for a wide variety of material handling applications. Raymond leads the market in the United States and sells, rents, or leases its lift trucks “throughout North and South America, as well as in Australia, China, the Pacific Rim, and the Middle East, through a network of independent dealers” [1]. Raymond remains headquartered in Greene, New York and has additional manufacturing facilities in Muscatine, Iowa and Brantford, Ontario. Raymond Corporation was acquired by BT Industries of

Sweden in 1997, which was purchased by Toyota Industries Corporation in 2000. Toyota combined all of its lift truck operations into the Toyota Material Handling Group in 2007, which has 13,000 employees worldwide and $4 billion in annual sales [1].

2.1.1. FUEL CELL EXPERIMENTATION

Raymond began investigating the use of fuel cells in the material handling industry in 2004 due to queries by its customers. A financial model was developed that explored the economics of converting an entire warehouse from lead-acid batteries to fuel cells. Raymond concluded that fuel cell technology can improve warehouse productivity while lowering operating costs [1]. The following year, Raymond began to actively work with suppliers to gain experience with fuel cell technology [1]. Raymond was awarded a $750,000 contract by the New York State Energy and Development Authority (NYSERDA) in 2007 to study the performance of hydrogen fuel cell-powered fork lift trucks and to demonstrate the feasibility of an indoor hydrogen-fueled environment [2]. During the two-year study, Raymond installed a hydrogen refueling station inside its Greene, New York manufacturing facility and employed hydrogen fuel cell-powered lift trucks in real applications [2]. Raymond is currently evaluating the results of its study and determining how best to move forward with its pursuit of fuel cell technology. The sponsorship by Raymond of this project is evidence of its continuing interest in how hydrogen fuel cell systems will be integrated into its fork lift trucks in the future.

2.2. HYDROGEN

From the problem statement it was clear that it would be important to develop an understanding of the fuel to be contained by the tanks being designed. For this reason, general research on hydrogen was conducted to understand how it is produced and stored, its energy density, and how it is affected by temperature, pressure, mass, and volume.

2.2.1. PRODUCTION

Although hydrogen gas produces clean emissions of water when reacted with oxygen, it is the production and storage of hydrogen gas that impedes its widespread use. Hydrogen gas can be derived from natural gas or fossil fuels but only at a fraction of the original chemical energy [3]. Hydrogen can also be obtained from the electrolysis of water, a very energy demanding process [3]. During electrolysis electricity is used to chemically decompose water into its constituent elements. The sustainability of hydrogen production via electrolysis really depends on the method used to produce the electricity. It is not generally sustainable to use fossil fuels to generate the electricity because it is more straightforward to use the fossil fuels to generate the hydrogen directly. If the electricity is generated from sources such as solar, wind, or nuclear power the sustainability of hydrogen generation can be greatly improved. Even though electrolysis is the preferred production method for extremely pure applications, it is generally difficult for it to compete on a large scale with other production methods. The most cost-efficient method currently employed in the industrial manufacture of hydrogen is steam hydrocarbon reforming. In this process natural gas is treated with high temperature steam that causes a chemical breakdown of the natural gas and releases hydrogen [4].

2.2.2. STORAGE

The development of efficient storage options for hydrogen creates further challenges for its use. Hydrogen can be stored as a gas, liquid, or a solid. Each storage method has advantages and disadvantages. Liquid storage is complex because hydrogen vaporizes at -253°C. This requires the tank to be cryogenically cooled in order to maintain the liquid state of the hydrogen [5]. Cryogenic cooling requires considerable amounts of energy and an extremely well insulated tank. This is generally not practical for many applications, including those in the material handling industry. Hydrogen can also be stored as a solid hydride; most commonly as a reversible metal hydride [5]. A reversible metal hydride can be recharged with hydrogen after release. One of the major challenges of metal hydrides is the percent weight of metal as compared to hydrogen. Hydrogen makes up only a small percentage of the total weight in a metal hydride, which requires that a large quantity of metal be used in order to supply the desired quantity of hydrogen [5].

Storing hydrogen as a compressed gas is perhaps the most practical method because it can be maintained at ambient temperatures and it is technically simple to contain using high-pressure gas cylinders. Hydrogen must be compressed due to its low density; to achieve a usable quantity of hydrogen at a practical volume compression is the only solution. High-pressure gas cylinders are generally metal based or composite based. Metal based high-pressure gas cylinders are relatively simple but the compatibility of the metal with the hydrogen gas can be a concern. Composite high-pressure gas cylinders are complex structures that contain multiple layers for hydrogen confinement, rupture strength, and impact resistance. In general, high-pressure tanks must be cylindrical or near cylindrical in shape to optimally withstand the pressure [5]. Many of the significant dangers associated with hydrogen are derived from the simple fact that it is commonly stored as a highly compressed gas. In the event that a tank were to fail catastrophically, the energy released would be comparable to that of an explosive. This risk can be successfully managed via the proper design and use of equipment for storing high pressure hydrogen gas. It is also important to note that industry experts generally agree that hydrogen is no more dangerous than other fuels including propane, natural gas, and gasoline [5].

2.2.3. STATE RELATIONSHIPS

To determine how hydrogen is affected by pressure, temperature, mass and volume, the Ideal Gas Law was employed. It is well known, however, that hydrogen deviates significantly from the Ideal Gas Law at high pressures. In order to use results from the Ideal Gas Law it must be corrected to give reasonably accurate results. The Van der Waals modification of the Ideal Gas Law takes into account the volume of the gas particles and the intermolecular forces that the Ideal Gas Law assumes to be negligible. Although there are other equations of state in addition to the Van der Waals modification, it is relatively simple and accurate enough for the purposes of this project. Writing the Van der Waals modification into a Matlab function allowed solving for pressure, temperature, volume, or mass as a function of the other three variables. The details of this Matlab function and its associated plotting script are shown in Appendix A: Hydrogen Property Calculation. Figure 3 illustrates the dependency of pressure on the temperature of hydrogen gas. It should be noted that for every 10°C change in temperature, the pressure changes by 200 psi. In creating the plot of Figure 3 the mass of hydrogen was fixed at 1 kilogram and the volume at 43 liters.

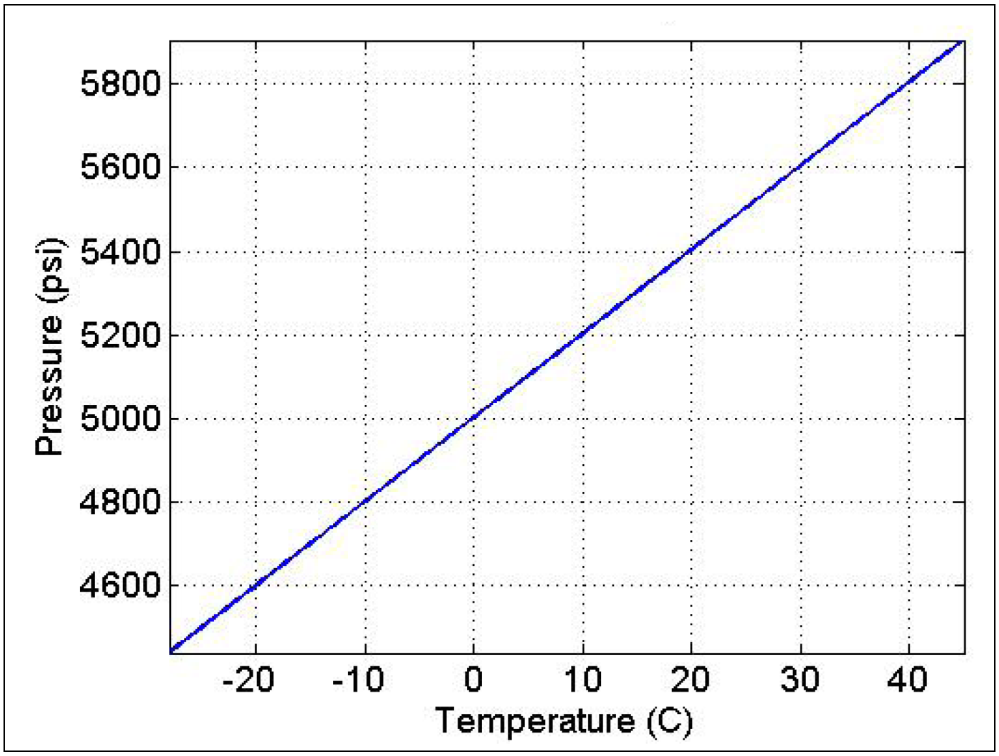


Figure 3 – Temperature vs H_2 Tank Pressure

Another relationship of interest involves the volume required to hold a fixed mass of hydrogen as pressure varies. The dependency of volume on pressure is illustrated in Figure 4. It should be noted that the volume reduction gained by increasing the pressure is gradually diminished. For example, a doubling of the pressure from 2,000 to 4,000 psi will decrease the volume of a tank by approximately 50% of its original volume. A doubling of the pressure from 5,000 to 10,000 psi will only decrease the volume of a tank by approximately 33% of its original volume.

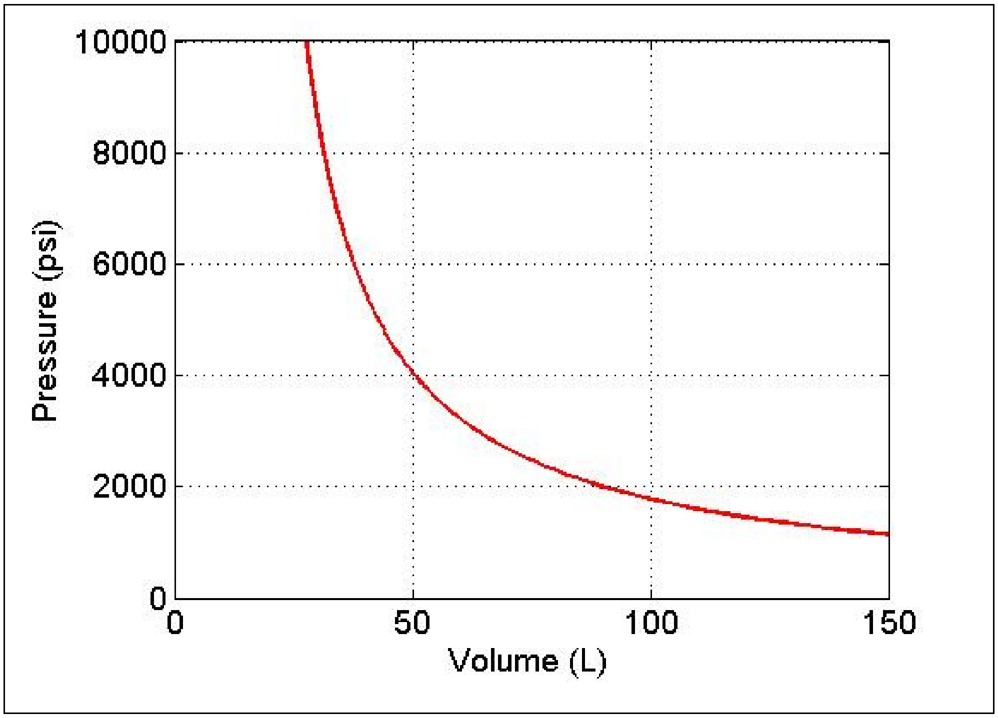


Figure 4 - Pressure vs Volume for Tank Containing 1kg of H_2 at 0°C

2.2.4. ENERGY DENSITY

Energy density refers to the amount of energy contained within a certain volume or mass of material. Looking at the energy density of hydrogen provides some additional insight into why it is used and the challenges associated with its use. Table 1 provides data on the energy available per unit volume of hydrogen (at 5,000 psi, 10,000 psi, and as a liquid), as well as gasoline. Table 2 provides data on the energy available per unit mass of hydrogen. Both tables provide data for total energy of the fuel and the energy as adjusted for the efficiency of the system consuming it. It can be observed from Table 1 that hydrogen stored at 5,000 psi has approximately 90% less energy than the same volume of gasoline. Hydrogen stored at 10,000 psi has approximately 85% less energy than the same volume of gasoline, not a significant improvement. Liquid hydrogen has approximately 70% less energy than the same volume of gasoline.

Table 1 – Available Energy per Unit Volume of Hydrogen and Gasoline

	Total Energy (MJ/L)	Energy × System Efficiency (MJ/L)
5000 psi H_2	3.2	1.6
10000 psi H_2	5.0	2.5
Liquid H_2	10	5.0
Gasoline	32.4	8.1

If the efficiencies of a PEM fuel cell (50%) and an internal combustion engine (25%) are taken into account [5], then hydrogen stored at 5,000 psi has approximately 80% less energy than gasoline, an improvement of 10% as compared to the total energy. Hydrogen at 10,000 psi is also improved with approximately 70% less energy, as well as liquid hydrogen with approximately 60% less energy.

The values for the total energy of compressed hydrogen presented in Table 1 were calculated by multiplying the enthalpy of combustion of 142 MJ/kg [7] by 1 kilogram and dividing by the volume in liters. Because the volume varies with the pressure, the previously described Van der Waals modification was used to determine the volume of 1 kilogram of hydrogen at 5,000 and 10,000 psi and fixed temperature of 20℃. A similar procedure was carried out for gasoline using an enthalpy of combustion of 45 MJ/kg (at a density of 720 kg/m^3) [8]. The value for the total energy of liquid hydrogen was found directly through research [5]

It can be observed from Table 2 that hydrogen has approximately 300% more energy than the same mass of gasoline. When the efficiencies are factored in as before hydrogen has 600% more energy than the same mass of gasoline.

Table 2 – Available Energy per Unit Mass of Hydrogen and Gasoline

	Total Energy (MJ/kg)	Energy × System Efficiency (MJ/kg)
H_2	142	71
Gasoline	45	11

From the data on hydrogen energy density it is clear that hydrogen is inferior to gasoline in terms of volumetric energy density but is superior in terms of mass energy density. In general, when considering the type of fuel to use in a consumer land vehicle, volumetric energy density is heavily favored over mass energy density [5]. This helps explain why it is desirable to store hydrogen as a compressed gas; to increase the volumetric energy density it must be compressed.

2.2.5. EMBRITTLEMENT OF STEELS

Lowering of the load-bearing and energy absorbing ability of steel by the influence of hydrogen is termed hydrogen embrittlement [5]. The mechanism behind hydrogen embrittlement involves the high pressurization of hydrogen within internal micro-cracks and voids, which generates plastic deformation and leads to the coalescence of micro-cracks or voids [5]. Hydrogen embrittlement leads to a reduction in ductility and tensile strength and can ultimately lead to failure of the part. High-strength and low-alloy steels (HSLA) are most susceptible to embrittlement. Steels with an ultimate tensile strength of less than 128 ksi are generally not considered susceptible to hydrogen embrittlement [9]. Embrittlement must generally be considered for any steels exposed to hydrogen during service but is especially important for steel gas cylinders containing highly pressurized hydrogen due to the safety concerns involved.

2.3. DESIGN CONCEPT SELECTION

The generation, evaluation, and selection of design concepts to be developed during the spring semester design phase was carried out during the fall semester research phase. The research phase report discussed the generation of design concepts, the methods and criteria used to evaluate them, and the results of the evaluation [12]. The design concepts selected for detailed design were as follows;

- **Seamless cylindrical vessel**
  Refers to a traditional high pressure vessel that is seamless and cylindrical in shape. A seamless tank is manufactured by piercing a billet and shaping it through spinning, or by spinning closed the ends on a piece of seamless tubing. Seamless cylindrical vessels are commonly used in a wide variety of industrial applications and fields.
- **Nested coiling of high pressure tubing (HPT)**
  High pressure tubing equipped with fittings at each end shall be bent into large spring-like coils. A similar coil of smaller diameter shall be placed within the center of the larger coil and so on creating a set of nested high pressure tubing coils. This is desirable because the shape of the coils can be customized as necessary to fit within abnormally shaped spaces.
- **Rectangular tank from structural members**
  Involves sealing of existing, rectangular structural members of a lift truck and pressurizing the members for use as a tank. The structural members could be of the mast, chassis, or overhead guard of the lift truck.

The three selected design concepts are discussed in further detail at the beginning of their corresponding report sections; Traditional Cylindrical Vessel, Nested High Pressure Tubing Coils, and Rectangular Tank (Overhead Guard).

3. DESIGN REQUIREMENTS

This section will present the design requirements for the compressed hydrogen tanks. The design requirements include basic requirements established in the project proposal as well as requirements set forth by applicable design standards identified during the fall semester research phase.

3.1. BASIC REQUIREMENTS

The basic design requirements are those requirements that were presented to the design group in the project proposal submitted to the Watson School of Engineering and Applied Science, or ascertained in meetings with industry advisor Bryce Gregory from Raymond Corporation.

3.1.1. CAPACITY

The total capacity of the compressed hydrogen tanks to be mounted on an individual fork lift truck shall be at least 1kg. According to the research conducted by Raymond Corporation, a capacity of 1kg of hydrogen provides a reasonable period of continuous operation for a lift truck.

3.1.2. PRESSURE

The tanks shall be able to hold compressed hydrogen at a minimum of 5,000 psi, at a standard temperature of 0℃, without leaking. As described in the section on hydrogen properties, there is relatively little volume reduction to be gained by increasing the pressure of hydrogen beyond 5,000 psi. For this reason, and due to the technical challenges of compressing hydrogen, it is not commonly compressed beyond 5,000 psi in the industry [10]. Implications for the effect of temperature on the service pressure are discussed in the operating temperature requirements.

3.1.3. TEMPERATURE

The tanks shall be able to operate in a temperature range from -28℃ to 45℃. This temperature range is based on the operating temperature range for a lift truck. The lower limit of the temperature range allows a lift truck to be operated inside a warehouse freezer, though it is not generally recommended for a lift truck to be left in such an environment for extended periods of time [10]. In general, design standards specify maximum pressures and test pressures based on the service pressure as defined at a particular temperature value (such as 0℃, 15℃, or 20℃). For this reason, the upper limit of the operating temperature range will not be used to calculate a maximum possible service pressure. The service pressure is defined at a temperature of 0℃ and all calculations will be made from that service pressure and not from a temperature adjusted maximum pressure.

3.1.4. MATERIAL

The tanks shall be composed of steel in order to keep the weight of the tanks high to compensate for the lost weight of the lead-acid batteries, which can weigh several thousand pounds. The heavier the tanks, the less additional counterweight needs to be added to the fork lift truck. There is one extenuating circumstance regarding weight; if the tanks are to be mounted to the fork lift truck in a way that may negatively alter the dynamics of the vehicle the weight of the tanks shall not exceed 500 lbs.

3.1.5. CYCLING

The tanks shall survive a minimum of 15,000 refueling cycles. The use of a lift truck varies significantly over its lifetime; it may begin its life in a high use application where it is refueled several times a day but end its life in an application where it is refueled once a week [10]. According to the US Fuel Cell Council's Fork Lift Task Force, 12,000 cycles is the 6σ value for the maximum number of refueling cycles expected for a fuel cell system that replaces a lead-acid battery [11]. Because this project is concerned with a fuel cell system that is integrated into a lift truck, a more conservative 15,000 cycles was suggested [10].

3.1.6. MISCELLANEOUS

The tanks shall have an appropriate protective coating, such as paint or powder coat, as necessary to protect the exterior surface. The tanks shall have threaded openings to accept standard CGA valve fittings such as those found on standard seamless pressure cylinders.

3.2. DESIGN STANDARDS

During the fall semester research phase applicable design standards were identified, obtained, and reviewed by the design group [12]. At the beginning of the spring semester design phase the design group decided that it would focus on conforming to the DOT Part 178 and HGV5 design standards. Basic information regarding each of these standards is presented within this section. It is important to note that there are many requirements contained within the DOT Part 178 and HGV5 design standards that are not presented within this section but are still referenced in the functional specification for the design proposals. Additional design standards, such as ISO 11114-1 and ASME Article KD-10, may also be referenced within the functional specifications but will not be discussed within this report. For further information regarding such additional design standards refer to the fall semester research phase report [12].

3.2.1. DOT PART 178

Part 178 of the federal Department of Transportation regulations details the manufacturing and testing specifications for packaging and containers used for the transportation of hazardous materials in commerce [14],[15]. Section 36 of Part 178 covers specification 3A and 3AX seamless steel pressure cylinders. Section 37 of Part 178 covers specification 3AA and 3AAX seamless steel pressure cylinders. The cylinder specification (3A, 3AX, 3AA, 3AAX) is determined based on conditions in the standard.

The applicability of the DOT Part 178 standard was identified early in the project by the project industry advisor. The standard only applies to seamless cylindrical vessel design. A summary of the requirements presented in the DOT Part 178 standard is presented in the web diagrams of Figure 5 and Figure 6. It was found that the DOT Part 178 standard is particularly useful to the design group because it includes equations for calculating bending stress, longitudinal stress, and wall stress. The standard also defines acceptable limits for stress levels in the cylinder. The DOT standard is unique as compared to other design standards because it requires every individual pressure cylinder to be certified and then recertified every five years through inspection and testing.

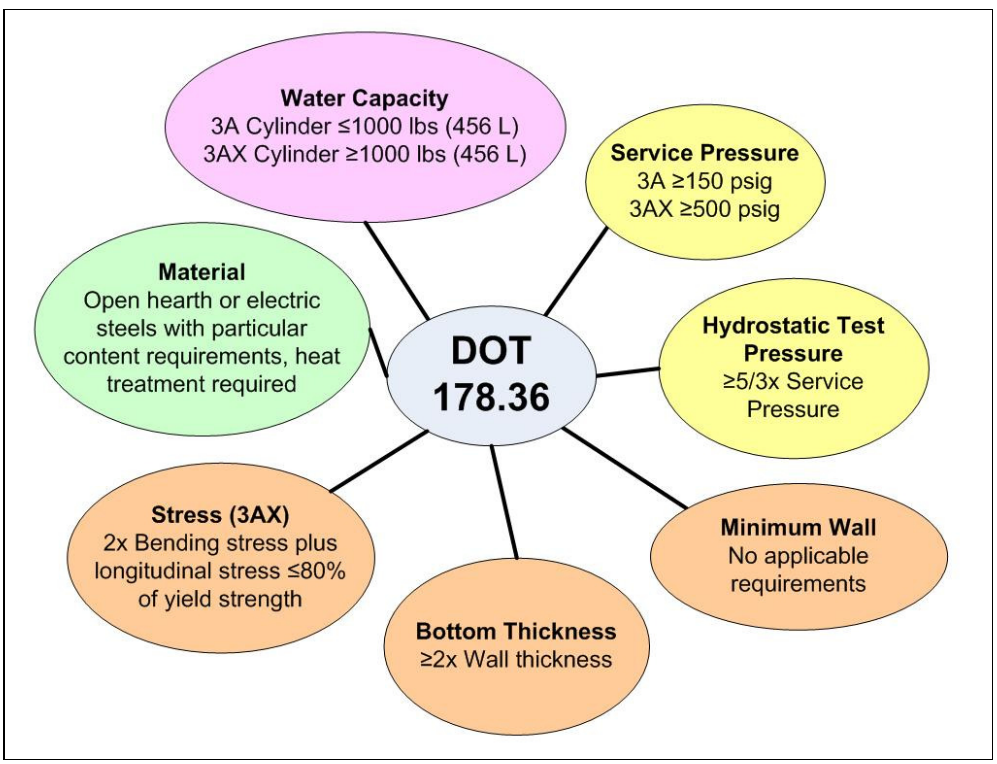


Figure 5 - DOT 178.36 Web Diagram

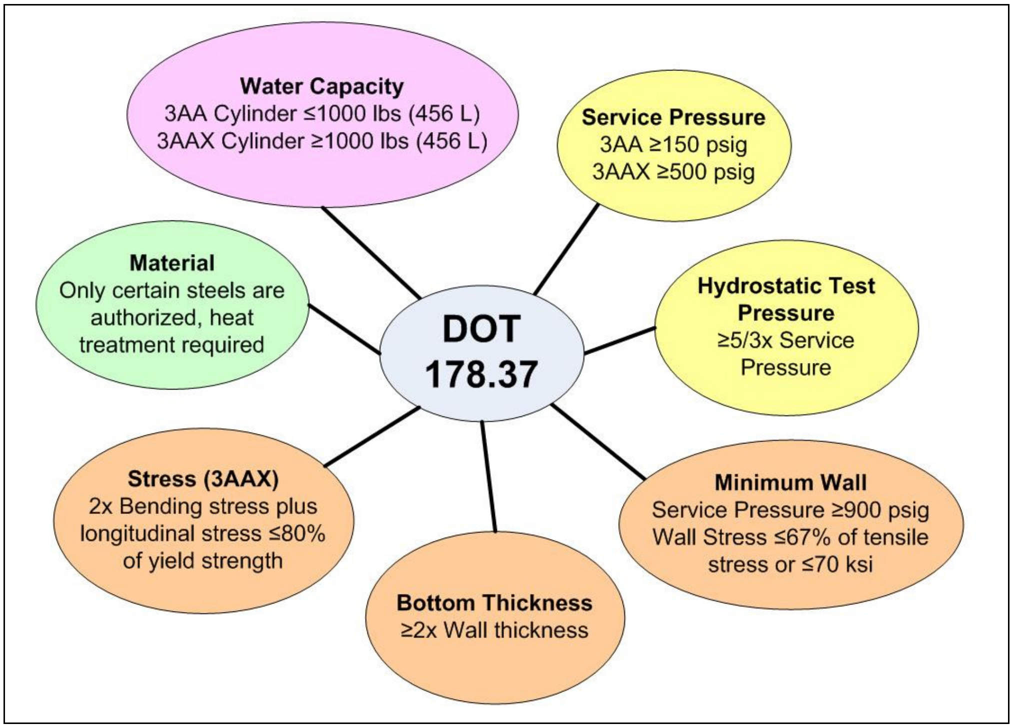


Figure 6 - DOT 178.37 Web Diagram

3.2.2. HGV5

HGV5 is a draft standard under development by the American National Standard Institute and the Canadian Standards Association. The design group was able to obtain a copy of the draft standard, dated October 2009, because the project industry advisor, Bryce Gregory, is a member of the technical advisory committee contributing to the creation of the standard. The language of HGV5 originates from HGV2, a standard similar to the NGV2 standard for compressed natural gas fuel tanks. HGV5 contains requirements for the material, design, manufacture and testing of containers that are intended only for the storage of compressed hydrogen gas and that are installed in powered industrial truck applications [13]. Like the NGV2 standard, HGV5 covers both metal and composite containers; metal containers are referred to as type HGV5-1 containers [13]. A summary of the requirements presented in the HGV5 draft standard is presented in the web diagram of Figure 7.

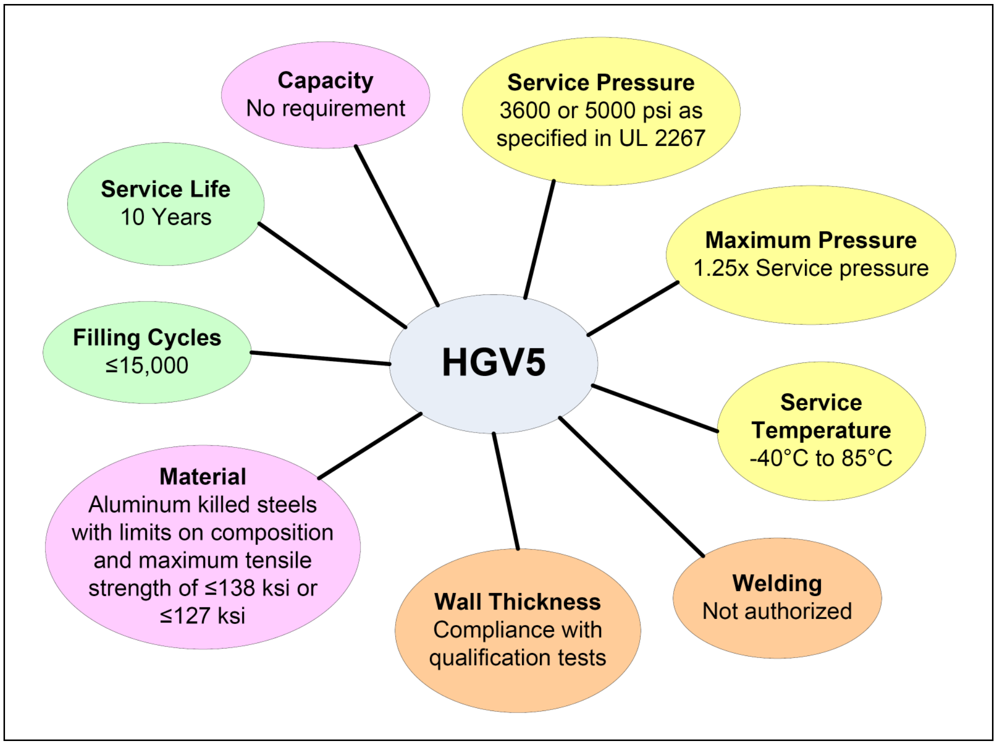


Figure 7 – HGV5 Web Diagram

4. TRADITIONAL CYLINDRICAL VESSEL

This section of the report will review the design concept for the traditional cylindrical vessel, discuss the design evolution, and present the functional specifications for the final design proposal.

4.1. CONCEPT DESCRIPTION

The traditional cylindrical vessel is a relatively straightforward concept; specially designed seamless pressure cylinders are to be mounted to either side of the mast of a typical fork lift truck, as shown in Figure 8. The cylinders are to be mounted to the fixed portion of the mast so as to minimize any adverse effects on the dynamics of the vehicle. The cylinders may have two hemispherical ends with threaded openings, as shown in Figure 8, or they may have one hemispherical end (with a threaded opening) and one closed, flat-bottomed end.

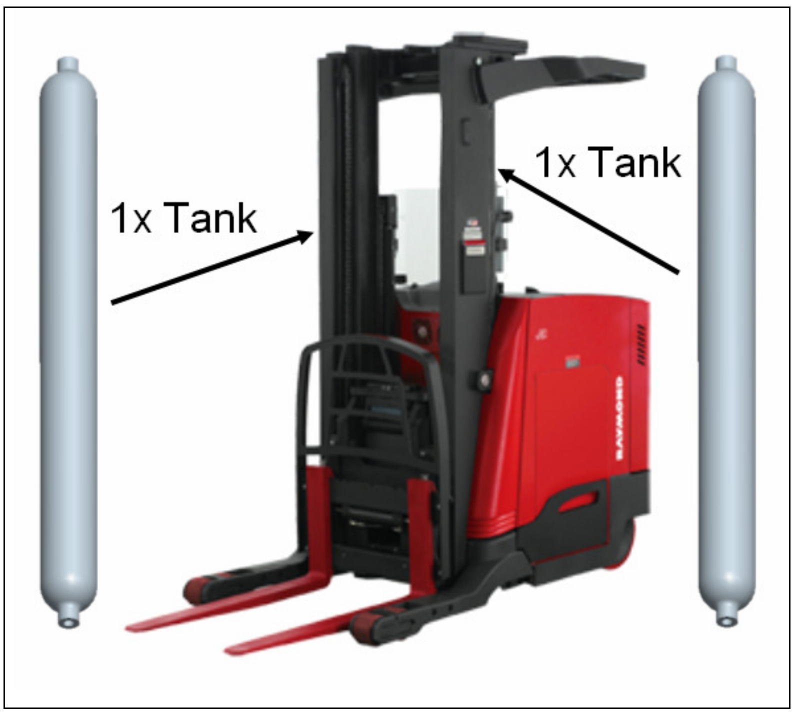


Figure 8 – Mast-Mounted Traditional Cylindrical Vessel

4.2. DESIGN EVOLUTION

4.2.1. INITIAL SIZING

The first step in designing the traditional cylindrical vessel was to conduct initial sizing based on the mounting space available. Using information provided by the project industry advisor, the space available on the mast of a typical fork lift truck was determined to be 100 inches in height, 9 inches in width, and 4 inches in depth. Based on this, the outside diameter of the cylinder was limited to the smallest dimension of 4 inches. The length of the cylinder was selected to be 90 inches to provide adequate space for valve connections at either end. The small outside diameter of the cylinder made it possible to specify the mounting of two cylinders to either side of the mast (for a total of four cylinders). This was necessary in order to achieve the desired total hydrogen volume of 43 liters.

Seeking a professional opinion on the initial sizing, Taylor-Wharton, an international metal working company specializing in the production of high pressure compressed gas cylinders, was contacted. Jim Wedding, a senior design engineer, provided valuable information concerning the manufacturability of the cylinder design. It was learned that the capabilities of Taylor-Wharton would not allow a cylinder with such a small outside diameter to be more than about half the desired length [16]. The planned length of the cylinder was easily reduced to 45 inches, which subsequently required the mounting of four cylinders to either side of the mast (for a total of eight cylinders) as shown in Figure 9.

It was also learned that there is some difficulty in completely closing one end of a stainless steel cylinder; the end must be drilled out and plugged in order to prevent future cracking [16]. This essentially required that the cylinder have two hemispherical ends with threaded openings. From discussion with the project industry advisor this was determined to actually be desirable because it eases the process of connecting the matrix of cylinders together; threaded openings on both ends allow for easy daisy-chaining of one cylinder to the next. Figure 9 illustrates the matrices of four cylinders that will be placed on either side of the fork lift truck mast.

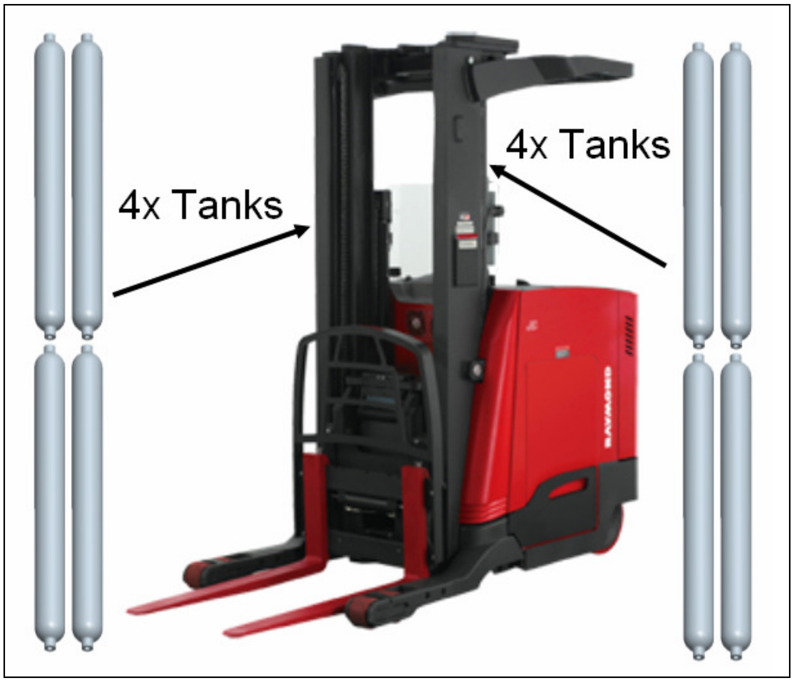


Figure 9 – Revised Mast-Mounted Configuration

Two additional aspects of the cylinder design discussed with Taylor-Wharton were the type of threading to be used and the incorporation of valve protection. It was learned during the fall semester research phase that ¾-14 NGT threading is typically used on high pressure cylinders [12]. Based on this ¾-14 NGT threading was selected for the cylinder. It was assumed that the integration of the cylinder into the fork lift truck would provide for the mounting and protection of the cylinder and any valves and connecting piping.

4.2.2. MATERIAL SELECTION

The research conducted during the fall semester indicated that 316L stainless steel (SS) is generally good at withstanding hydrogen embrittlement [12]. For this reason 316L stainless steel was chosen for the cylinder material. Based on average results from tensile-testing finished 316L cylinders, Taylor-Wharton uses a conservative value of 34,000 psi for the yield strength [16]. The use of 316L was also desirable because the fatigue limit (based on 10^6-10^9 cycles to failure) was found to be 39,000 psi [17]. This fatigue limit reduces the risk of failure due to refueling for two reasons. First, the 10^6 cycles on which the fatigue limit is based is much higher than the 15,000 refueling cycles allowed by the basic design requirements. Second, it is *above* the yield strength and would, theoretically, never be reached because the cylinder was be designed to operate below its yield strength at all times.

Due to the cost of 316L stainless steel, the project industry advisor suggested designing an alternate version of the cylinder using a less expensive steel. Based on knowledge acquired during the fall semester research phase of the project, 4130 Q&T steel was selected for the alternate version of the cylinder. It was found that an appropriate yield strength for 4130 Q&T is 102,000 psi [18]. The fatigue limit (based on 10^6-10^9 cycles to failure) was found to be 71,000 psi [18]. It was learned that, based on common practice, Taylor-Wharton would design a 4130 Q&T cylinder assuming a maximum allowable wall stress of 67,000 psi (even though the yield strength is much higher) [16]. This assumption makes sense because it keeps a significant margin of safety between the maximum stress and the yield strength and is *below* the fatigue limit. The fatigue limit will, theoretically, never be reached because the cylinder was designed to operate below its maximum stress at all times.

It should be mentioned that because the yield strength of the 316L stainless steel is so low (as compared to the 4130 Q&T) it is not possible for there to be much of a safety margin between the maximum stress and the yield strength. This will be revealed further in the section on stress analysis. It will also be revealed that in order to achieve a large safety margin it would be necessary for the cylinder to have a very thick wall.

During the fall semester research phase it was learned that, according to ISO 11114-1, Q&T steels must have an ultimate tensile strength below 137,000 psi to prevent issues with hydrogen embrittlement [19]. Because it was found that 4130 Q&T has an ultimate tensile strength of 118,000 psi [18], it should not suffer significantly from hydrogen embrittlement. It is important to note that the 4130 Q&T used for the cylinder material could be qualified for high pressure hydrogen storage by carrying out the qualification tests of ASME Article KD-10 [12]. This relatively new addition to Section VIII, Division 3 of the ASME Boiler and Pressure Vessel Code details fracture control rules for vessels containing high pressure hydrogen gas. Even though these tests could not be carried out by the design group it is important to note that they *could be* carried out to further qualify 4130 Q&T for use with high pressure hydrogen.

With the materials selected for two different versions of the cylinder it was possible to complete the sizing of each version by determining the appropriate wall thicknesses. The determination of the wall thicknesses, and the subsequent stress analysis, is described in the following section.

4.2.3. STRESS ANALYSIS

Before stress analysis could begin it was necessary to determine the appropriate wall thicknesses for each version of the cylinder using thick-walled cylinder equations from Shigley's *Mechanical Engineering Design, 8th Edition* [20]. The simplified thick-walled equations were inserted into the expression for Von Mises stress [29] and then solved for the inside radius (effectively, the wall thickness). The resulting equation required a stress limit, outside radius, and internal pressure in order to calculate the wall thickness. Details of the equation derivation are shown in Appendix K.

The stress limits were, as previously discussed, 34,000 psi for the 316L SS version and 67,000 psi for the 4130 Q&T version. DOT Part 178 and HGV5, the design standards previously described, would require the cylinder to be hydrostatically tested at a pressure of 5/3 and 1.5 times the service pressure, respectively [14],[15],[13]. The basic design requirements call for a service pressure of 5,000 psi which results in a hydrostatic test pressure of 8,333 psi and 7,500 psi for the DOT Part 178 and HGV5 design standards, respectively. For the wall thickness calculation and the subsequent stress analyses, the DOT Part 178 test pressure of 8,333 psi was adopted as it was the higher of the two test pressures. This test pressure was confirmed to be appropriate for meeting the DOT Part 178 requirements by Taylor-Wharton [16].

Using all this information, a minimum wall thickness of 0.483 inches was calculated for the 316L version of the cylinder while a minimum wall thickness of 0.228 inches was calculated for the 4130 Q&T version. These wall thicknesses were rounded up to 1/2 inch and 1/4 inch, respectively, in order to select a simple nominal wall thickness for each version of the cylinder. Inserting the rounded wall thicknesses back into the equation and using the same test pressure resulted in stresses of 32,990 psi and

61,580 psi for the 316L SS and 4130 Q&T versions, respectively. This means that there are factors of 1.03 and 1.09 between the theoretical stress and the stress limit for the 316L SS and 4130 Q&T versions, respectively. The selected nominal wall thicknesses were confirmed to be appropriate by Taylor-Wharton for the desired service pressure [16]. While the design factors above may seem low, it is important to note that nearly all the calculations and analyses carried out during the project, for all designs, were carried out at the hydrostatic test pressures not the service pressure. This means there is already a significant design factor between the service pressure and the hydrostatic test pressures.

After a thorough review of DOT Part 178, and with some input from Taylor-Wharton [16], it was determined that the 316L version of the cylinder is classified as a DOT-3A cylinder and the 4130 Q&T version a DOT-3AA cylinder. These classifications are based on the water capacity and service pressure of the cylinders [14],[15]; refer back to the DOT Part 178 web diagrams of Figure 5 and Figure 6. According to DOT Part 178.37, the wall stress in the 4130 Q&T version of the cylinder must be less than or equal to 67% of the tensile stress of the material, *or* less than or equal to 70,000 psi. Using the equations provided in 178.37, and the selected 1/4 inch wall thickness, the calculated wall stress was 57,109 psi; this was below 67% of the 118,000 psi tensile stress of the material *and* below 70,000 psi. This calculation was shown on the final cylinder drawing in Appendix E. According to DOT Part 178.36 there are no applicable wall stress requirements for the 316L SS version of the cylinder.

The first computer model of the 316L SS cylinder was 45 inches long with 1/2 inch thick walls along the 38 inch straight section, see Figure 10. At the transition between the straight section and the hemispherical caps, however, the walls gradually increased in thickness to 1 inch. This was done to meet the DOT Part 178 requirement that the bottom thickness be twice the wall thickness. The thickness of the necks around the threaded openings was made to be 3/4 inch. These end thicknesses were also chosen to compensate for any stress concentrations that might occur at the ends of the cylinder. The radii shown in Figure 10 were visually selected in Pro/Engineer to create a smooth transition between the straight section and the necks of the cylinder.

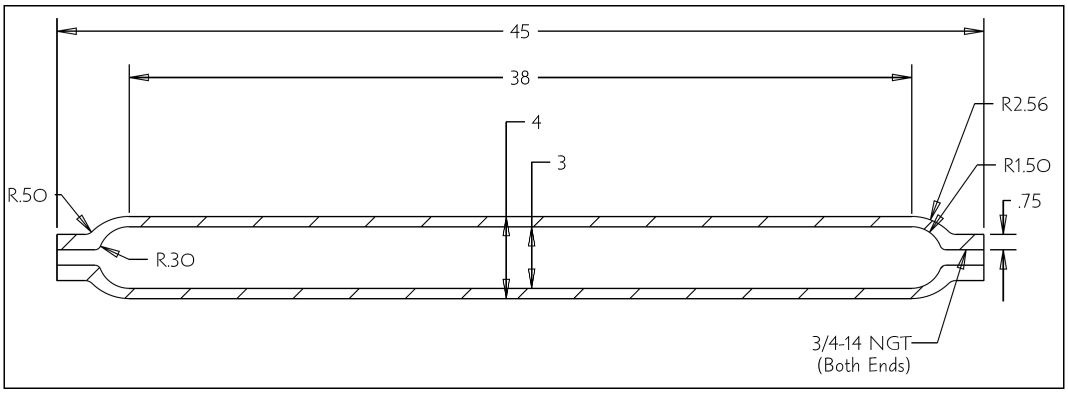


Figure 10 – First Computer Model of 316L SS Cylinder

The drawing shown in Figure 10 was sent to Taylor-Wharton for feedback concerning the dimensions and manufacturability of the cylinder. It was learned that the term *bottom* in DOT Part 178 refers strictly to flat-bottomed cylinders and does not apply to cylinders with hemispherical ends. This meant it was not necessary for the wall thickness to increase to 1 inch at the hemispherical ends. It was also learned that the thickness of the neck on such a cylinder typically arrives at 3/8 inch during manufacturing. Finally, it was suggested that the radii shown on the drawing be removed because it is not possible to guarantee specific radii due to the nature of the spinning process used to close the cylinder. Based on what was learned from Taylor-Wharton, the hemispherical caps were changed to match the 1/2 inch wall thickness, the thickness of the neck was made to be 3/8 inch, and the radii were removed from the drawing. See Figure 11 on the following page.

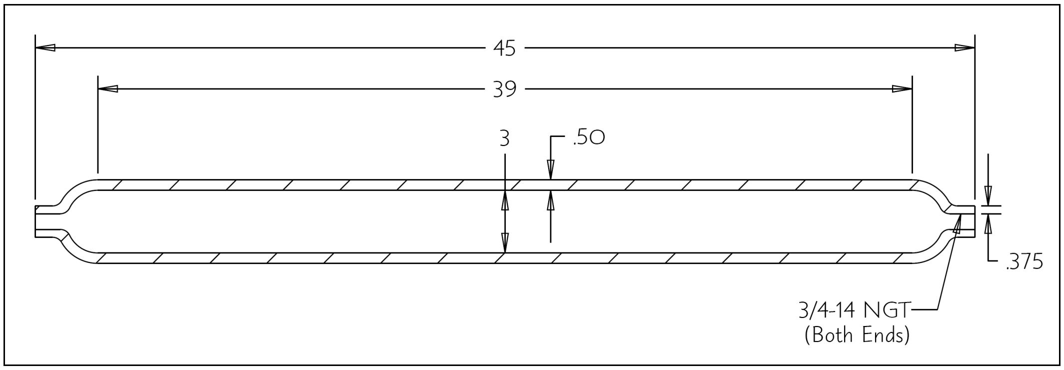


Figure 11 – Revised Computer Model of 316 SS Cylinder

With the computer model revised, it was time to conduct a finite element analysis (FEA) in Pro/Mechanica. A displacement constraint was placed on the inner walls of the neck opening where the threads would be located. The DOT Part 178 test pressure of 8,333 psi was applied to the remaining inside surfaces of the cylinder. The resulting Von Mises stress was visualized using the fringe plot shown in Figure 12. The maximum stress observed was 33,660 psi, only 2% higher than the 32,990 psi calculated theoretically. This is a positive result because it is beneath the stress limit of 34,000 psi. It is also important to note that no significant stress concentrations were observed. Several different views of the FEA results shown in Figure 12, as well as a convergence plot, are available in Appendix H: FEA Results (Cylindrical Vessel).

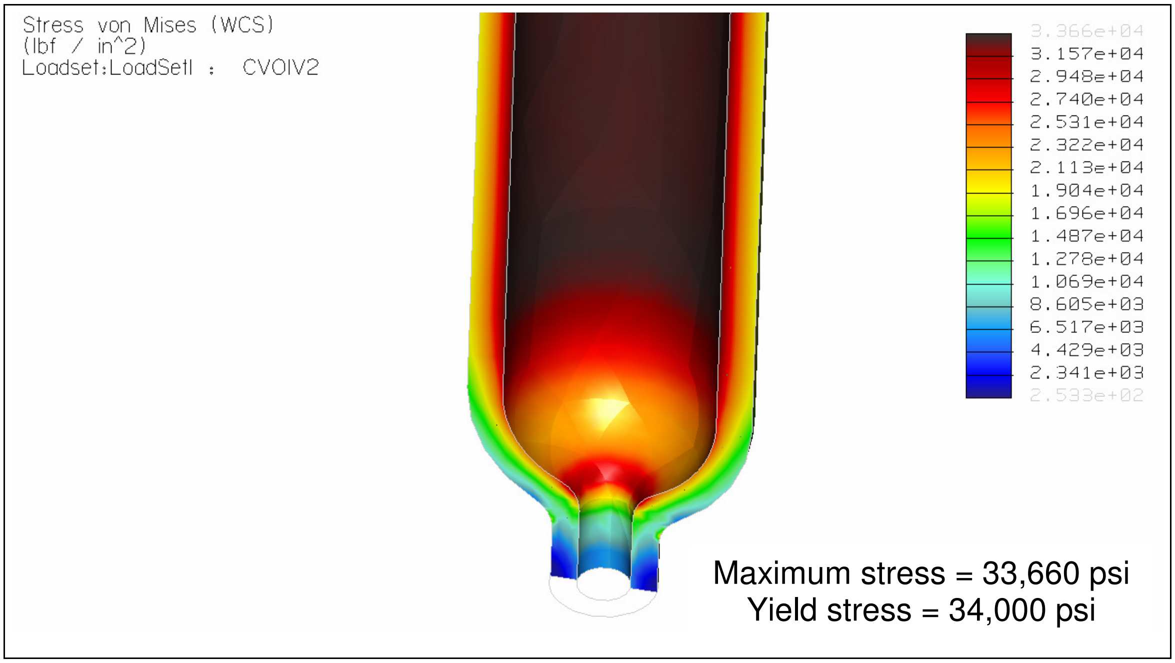


Figure 12 – 316L SS Cylinder Fringe Plot – Longitudinal Cut Close-Up

With the success of the 316L SS cylinder, the next step was the 4130 Q&T cylinder model and analysis. A model was created with the same basic dimensions as the 316L SS cylinder except the wall thickness was reduced to 1/4 inch, see Figure 13. A finite element analysis was carried out in the exact same manner as with the 316L SS cylinder. The resulting Von Mises stress was visualized using the fringe plot shown in Figure 14. The maximum stress observed was 63,390 psi, only 3% higher than the 61,580 psi calculated theoretically. This is a positive result because it is beneath the stress limit of 67,000 psi. Even though DOT Part 178.37 requires the wall stress to be calculated from the equations it provides, it is important to note that the maximum wall stress observed in the FEA is still less than 67% of the 118,000 psi tensile stress of the material. As before, no significant stress concentrations were observed.

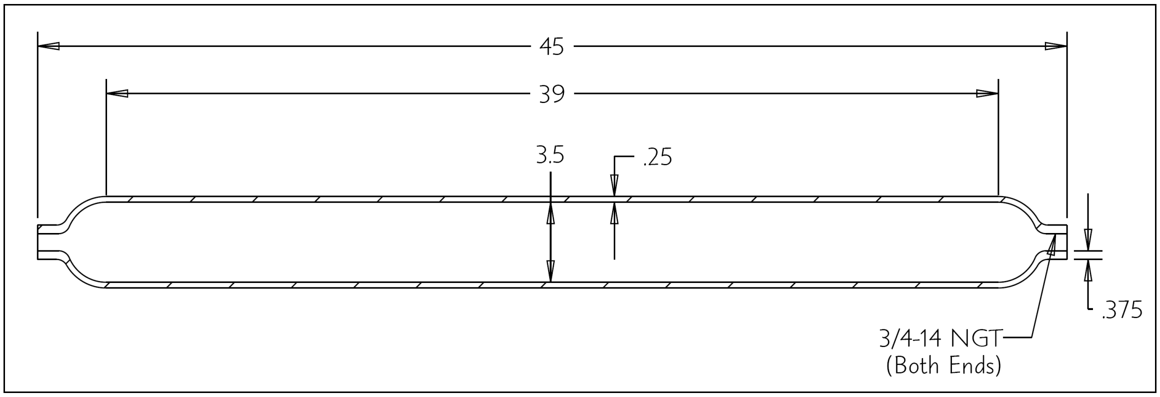


Figure 13 – Computer Model of 4130 Q&T Cylinder

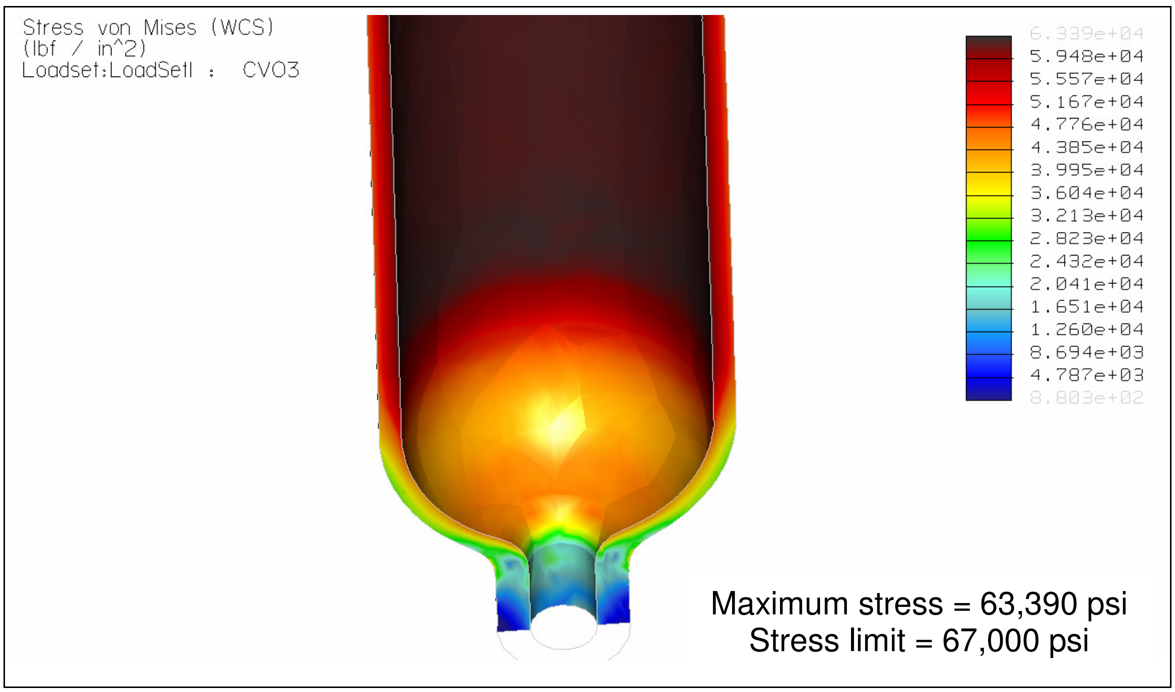


Figure 14 – 4130 Q&T Cylinder Fringe Plot – Longitudinal Cut Close-Up

Several different views of the FEA results shown in Figure 14, as well as a convergence plot, are available in Appendix H. With the initial stress analysis complete the drawings of each version of the cylinder were sent to Taylor-Wharton for quoting. The finished drawings for the cylinder designs are available in Appendix E: Functional Specification (Cylindrical Vessel). Finally, the capacity and weight of each cylinder design (as well the matrix of cylinders to be mounted to the fork lift truck mast) are shown in Table 3.

Table 3 – Capacity and Weight of Cylinders

	Single Cylinder	Eight Cylinders
316L Stainless	4.75 L at 66.1 lb	38.0 L at 529 lb
4130 Q&T	6.52 L at 36.5 lb	52.1 L at 292 lb

4.2.4. HYDROGEN-RESISTANT COATINGS

It was suggested by the project industry advisor that the design group investigate the possibility of applying a protective coating to the inside of the cylinders, especially the 4130 Q&T cylinder, to reduce the permeation of hydrogen into the cylinder material. To begin this investigation the design group contacted Endura Coatings, a company that has been specialized in coating technology for more than 38 years. The opinion of Bill Naschak, an engineering manager at Endura, was that it would be incredibly difficult to create a barrier against diatomic hydrogen using any kind of thin-film coating [21]. During the fall semester research phase it was learned that many composite tanks are equipped with a plastic liner to help protect the metal boss from the hydrogen. When this was mentioned to Bill he suggested contacting Zeus Plastics, a company specializing in plastic extrusions.

Zeus Plastics was contacted and the problem was discussed with Eric Trimble in the Engineering Extrusions Division. According to him, Zeus does not have the capabilities to extrude plastic tubes large enough to be used as liners for the cylinders. When it was clarified that the plastic lining would need to be applied *after* the cylinder ends have been spun down to necks with 3/4 inch openings, Eric suggested two possible solutions; either a heat expandable tube small enough to fit through the neck opening or a spray coating [22]. Unfortunately, Zeus Plastics does not have any experience working with either of these solutions and could not provide the design group further assistance.

Having already discussed some possibilities with two contacts in industry, the design group received a suggestion to consider using rotational molding to coat the interior of the cylinders with plastic linings. Rotational molding is a process whereby a mold is heated, causing a plastic raw material inside to melt, and then rotated on several axes until the plastic evenly coats the inside of the mold. The idea would be to use a cylinder as the mold in order to evenly coat the inside with a plastic lining. To investigate this idea, the design group contacted Formed Plastics, Inc., a company specializing in vacuum forming and rotational molding. Tom Crowe, the Vice President of Sales, indicated that Formed Plastics has not had any experience with such an application but suggested contacting either Paul Nugent or the Association of Rotational Molders (ARM) [23].

Paul Nugent, an independent, international consultant in rotational molding, was contacted. According to Paul, using rotational molding to add a lining to a pipe is carried out on a regular basis [24]. To his knowledge, there are a couple of polyethylene materials offered by Equistar and ICO Polymers that have been modified for such a purpose. They required modification because polyethylene needs an adhesive component added to it in order for it to bond with the steel [24]. On the subject of the heating required, Paul said the temperatures required during rotational molding are typically 450°F to 550°F and would not likely have a serious effect on the base properties of the steel [24]. He cautioned, however, that the temperature should be checked against the heat treatment data for the steel to investigate the possible effects [24]. Finally, Paul indicated that the cycle required to rotational mold the cylinders would likely be longer than normal due to the heavy wall thickness as compared to a typical mold; a half hour cycle would likely be required for a 1/8 inch plastic coating [24]. Due to the other demands of the project the design group did not have an opportunity to pursue the rotational molding idea further than the communication with Paul.

Though incomplete, the design group did make significant progress in assessing the possibilities for applying hydrogen-resistant coatings to the cylinders. While Bill at Endura Coatings indicated that a thin-film coating would be very difficult, he did not suggest that it would be altogether impossible to engineer a solution that could at least reduce the rate of hydrogen permeation into the cylinder material. Eric at Zeus Plastics offered a plausible idea involving heat expandable tubing that could have been explored further if the other demands of the project had allowed. Finally, Paul Nugent provided promising insights regarding the use of rotational molding as a means of coating the cylinders with plastic linings.

4.2.5. MANUFACTURING QUOTE

In parallel with the design work conducted on the cylinders, Taylor-Wharton developed manufacturing quotes for orders of 1,000-1,999 and 10,000 units of each cylinder version. A summary of the manufacturing quote received is shown in Figure 15 – Taylor-Wharton Cylinder Pricing. The complete manufacturing quote received is included in Appendix K.

DOT3A5000 4.0" OD 316 Stainless Steel Double Ended Cylinder

Part# (To Be Assigned)

Quantity:	**1,000 – 1,999**	**10,000**
Price:	**$777.48 ea**	**$726.970 ea**

DOT3AA5000 4.0" 4130 Steel Double Ended Cylinder

Part# (To Be Assigned)

Quantity:	**1,000 – 1,999**	**10,000**
Price:	**$225.98 ea**	**$198.42 ea**

Figure 15 – Taylor-Wharton Cylinder Pricing

Along with the quote, the design group was informed that the prices would be good for only 30 days due to the constantly fluctuating costs of the raw material, especially the raw stainless steel tubing [16]. From discussion with the project industry advisor it was desirable to try and ascertain what portion of the cylinder costs are for raw material versus labor. To estimate this it was necessary to obtain an approximate market value of 316L stainless steel and 4130 Q&T steel. The design group was conveniently in communication with Louisiana Steel concerning the rectangular tank design and was able to obtain a cost per length and weight per length for an extruded box tubing in both materials. Since the raw tubing used for the cylinders is also an extruded product, this seemed like a reasonable way to approximate the cost per weight. The cost of the 316L SS and 4130 Q&T box tubing was $140 per foot and $75 per foot, respectively [25]. Both materials were 16.25 pounds per foot [25]. The material costs were calculated to be $8.62 per pound and $4.62 per pound for the 316L SS and 4130 Q&T, respectively. The total costs of the cylinders, based on the weights in Table 3, were calculated to be $569.48 and $168.46 for the 316L SS and 4130 Q&T, respectively. A comparison of these costs to those in Figure 15 revealed that (for 1,000-1,999 units) approximately 73% of the 316L SS cylinder cost is for material and 75% of the 4130 Q&T cylinder cost is for material.

The design group was also informed that Taylor-Wharton would conduct its own design of the cylinders based on a minimum allowable wall thickness of 0.213 inches for the 316L SS version and 0.426 inches for the 4130 Q&T version [16]. These minimum wall thicknesses are, respectively, almost 7% and 12% lower than those calculated by the design group using the theoretical equations from *Shigley's* Mechanical Engineering Design [20]. This discrepancy raised some concerns about the technique used to calculate the minimum wall thickness. Discussion with Jim at Taylor-Wharton revealed that he went to the CSA B339 design standard to conduct the wall thickness calculation [16]. During the discussion it was realized that it was not really important to the design group what minimum wall thickness Taylor-Wharton calculated. What was important to the design group was the maximum dimensional variations in the raw tubing that would be considered acceptable by Taylor-Wharton. According to Jim, the tolerances on the raw 316L SS tubing are taken from the ASTM A269 or A511 standards and the tolerances on the raw 4130 Q&T tubing are taken from the ASTM A519 standard [16]. From these standards, the allowable variation in the diameter is +/-0.015 inches for the 316L SS and +/-0.025 for the 4130 Q&T [16]. The allowable variation in the wall thickness for both materials is +/-10%.

The allowable variation in the wall thickness was satisfactory because it prevents the minimum values calculated by Taylor-Wharton from ever being reached; raw tubing with a wall thickness that varies by more than 10% from the nominal value will be rejected. On the other hand, the allowable variation in the wall thickness was not satisfactory because it does allow values that are below those minimum values theoretically calculated by the design group during the initial sizing and stress analysis. The nominal wall thickness of the 316L SS cylinder minus 10% is 0.45 inches, which is 6.8% lower than the theoretically calculated minimum. The nominal wall thickness of the 4130 Q&T cylinder minus 10% is 0.225 inches, which is 1.3% lower than the theoretically calculated minimum. This made it clear that the effect of the dimensional variation in both the diameters and the wall thicknesses on the wall stress needed to be fully investigated. To begin, the most extreme cases of the possible dimensional variations were illustrated for the 4130 Q&T cylinder as shown in Figure 16. Each case illustrates a different combination of the maximum possible variations in the inside and outside diameters. The minimum or maximum wall thickness was reached in each case by offsetting the centers of the inside and outside diameters.

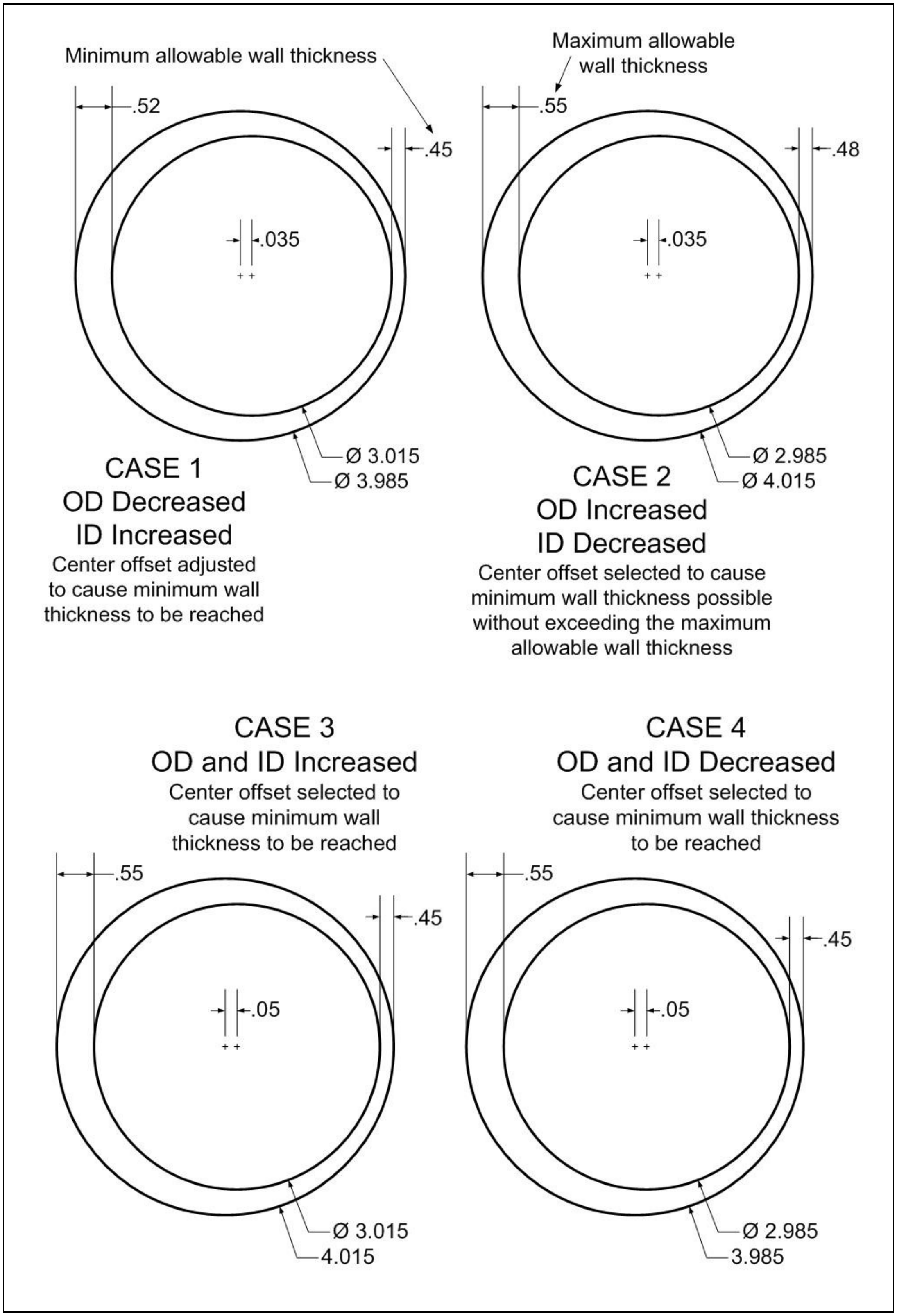


Figure 16 – Maximum Dimensional Variations in 316L Cylinder Tubing

It was decided from Figure 16 that Case 2 and Case 4 would not cause the most extreme wall stresses of the four cases. It was known from *Shigley's* Mechanical Engineering Design that the thicker the wall and the smaller the diameter the smaller the inner wall stress [20]. Case 2 clearly has the thickest walls while Case 4 has the smallest diameters. It was not immediately clear whether Case 1 or Case 3 would cause the most extreme wall stresses. Case 1 clearly as the smallest wall thicknesses but Case 3 also has a larger outside diameter. To investigate which case caused the most extreme wall stress finite element analyses were carried out on models with the given dimensional variations. It was found that both cases were almost identical in terms of maximum wall stress; Case 1 was just slightly higher than Case 3. The results of the analyses for Case 1 dimensional variations in the 316L SS cylinder and 4130 Q&T cylinder are shown in Figure 17 and Figure 18, respectively. For the full set of FEA results see Appendix H.

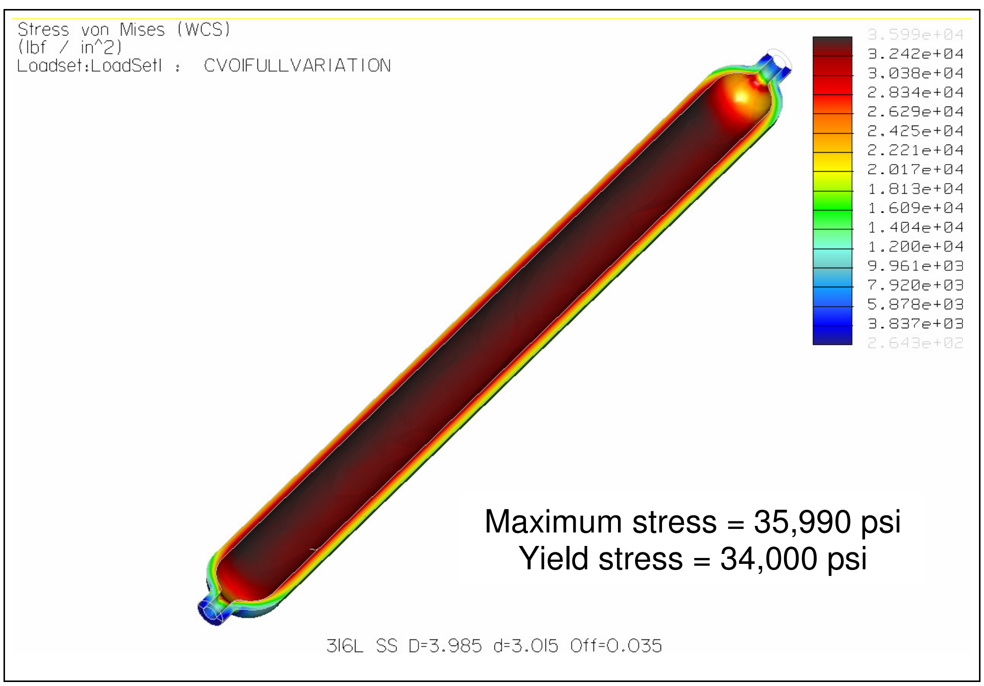


Figure 17 – 316L SS Cylinder Fringe Plot – Case 1 Variation

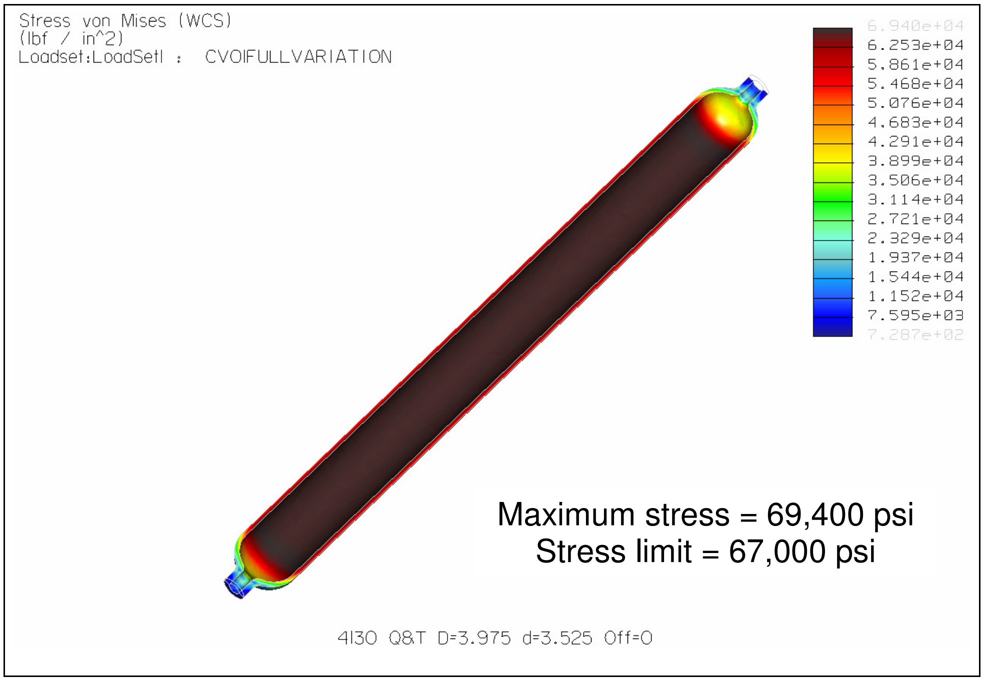


Figure 18 – 4130 Q&T Cylinder Fringe Plot – Case 1 Variation

The maximum wall stress observed in the 4130 Q&T cylinder with the Case 1 dimensional variation was 69,400 psi. Though this is beyond the stress limit previously defined, this was not a problem because it is still below 67% of the tensile stress of the material (67% of 118,000 psi is 79,060 psi) per DOT Part 178.37 as well as below the fatigue limit (71,000 psi) of the material. The maximum wall stress observed in the 316L SS cylinder with the Case 1 dimensional variation was 35,990 psi. This value is beyond the 34,000 psi yield strength suggested by Taylor-Wharton [16]. It should be recalled that Taylor-Wharton suggested this value for the yield strength because it was the average value observed from tensile testing finished 316L SS cylinders. To solve this problem, it was decided that a minimum yield strength of 36,000 psi would be made a requirement in the functional specification for the cylinder. From discussion with Taylor-Wharton it was determined that such a requirement would be passed on to the manufacturer of the raw tubing and might ultimately increase the cost of the cylinder [16].

4.3. FUNCTIONAL SPECIFICATION

This section of the report presents the functional specifications for the cylindrical vessel designs. A condensed version of the functional specification is presented in Appendix E. The functional specifications are the detailed requirements dictated by the finished design in order to meet the basic requirements presented previously in the report as well as the DOT Part 178 and HGV5 design standard requirements. Unless specifically indicated, the requirements presented in this functional specification apply to both the 316L SS and 4130 Q&T versions of the cylinder. Note, however, that requirements of DOT Part 178.36 (for specification 3A cylinders) apply *only* to the 316L SS version of the cylinder. Requirements of DOT Part 178.37 (for specification 3AA cylinders) apply *only* to the 4130 Q&T version of the cylinder.

4.3.1. PHYSICAL REQUIREMENTS

This section includes the geometry, material, and physical features that characterize the design and are required to meet the basic requirements.

Geometry – The geometries of the cylinders shall conform to the drawings contained within Appendix E of this report. These geometries have been designed to meet the *Wall thickness* requirements of DOT Part 178.36 and DOT Part 178.37.

316L SS Cylinder Capacity – One 316L SS cylinder will allow for the storage of approximately 4.75 liters of 5,000 psi compressed hydrogen. Eight 316L SS cylinders shall be used on a single fork lift to allow for the combined storage of approximately 38 liters of compressed hydrogen. ***Comments:*** *Because the 38 liter capacity achieved is 88% of the 43 liter basic requirement it was considered acceptable. It was not possible to increase the volume past 38 liters given the available space.*

4130 Q&T Cylinder Capacity – One 4130 Q&T cylinder will allow for the storage of approximately 6.52 liters of 5,000 psi compressed hydrogen. Eight 4130 Q&T cylinders shall be used on a single fork lift to allow for the combined storage of approximately 52.1 liters of compressed hydrogen.

316L SS Cylinder Material – The material of the raw tubing shall be 316L stainless steel to reduce the risk of hydrogen-induced embrittlement over time. The 316L stainless steel shall have a yield strength no lower than 36,000 psi. The 316L SS shall have a fatigue limit (based on 10^6-10^9 cycles to failure) above the specified yield strength to reduce the risk of fatigue failure caused by refueling cycling. The 316L stainless steel must meet the *Material Qualification Tests and Requirements* set forth in HGV5 and the *Steel* requirements set forth in DOT Part 178.36.

4130 Q&T Cylinder Material – The material of the raw tubing shall be 4130 Q&T steel. The steel shall have a yield strength no lower than 102,000 psi. The steel shall have an ultimate tensile strength of no more than 137,000 psi in accordance with ISO 11114-1. The steel should have a fatigue limit (based on 10^6-10^9 cycles to failure) of not less than 71,000 psi to reduce the risk of fatigue failure caused by refueling cycling. The steel must meet the *Material and Qualification Tests and Requirements* set forth in HGV5. The steel must meet the *Authorized steel* requirements in DOT Part 178.37. The steel should be qualified for use with high pressure hydrogen according to the requirements of Article KD-10 of Section VIII, Division 1 of the ASME Boiler and Pressure Vessel Code. ***Comments:*** *It was not possible for the design group to assess in detail the requirements set forth in Article KD-10, but it was known that Article KD-10 uses a robust approach, based on fracture mechanics, to qualify materials for use with high pressure hydrogen.*

Tolerances – The raw tubing that will be spun into the cylinders shall meet the following requirements for maximum allowable dimensional variations;

- Maximum of 10% variation in the wall thickness (from the nominal) as measured at various points around the circumference of the tube
- For the 4130 Q&T cylinder; Maximum of +/-.025" deviation in the diameters per ASTM A519
- For the 316L SS cylinder; Maximum of +/-0.015" deviation in the diameters per ASTM A269 or ASTM A511

External Surfaces – The external surfaces of the 4130 Q&T cylinder shall be protected by a coating of paint or powder coat according to appropriate industry practice. The protective coating on the 4130 Q&T cylinder shall meet the requirements for *External Surfaces* set forth in HGV5. The external surfaces of the 316L SS cylinder shall meet the requirements for *External Surfaces* set forth in HGV5.

Termination – The cylinders shall have two standard ¾-14 NGT threaded openings per the drawings in Appendix E for connection to appropriate valves and the hydrogen supply system on the fork lift truck. The threads shall comply with *Threaded Openings* in HGV5 and *Openings in cylinders* in DOT Part 178.36 and DOT Part 178.37.

4.3.2. MANUFACTURING REQUIREMENTS

This section includes tests that must be carried out in order to qualify the design for subsequent manufacturing, as well as the manufacturing procedures and tests required to confirm a quality final product that meets the other requirements set forth in this functional specification.

HGV5 Qualification Testing – Prior to shipment of a completed cylinder the design qualification tests listed below must be carried out, according to HGV5, with satisfactory results. Any change in the cylinder design may require some or all qualification tests to be repeated according to *Change of Design* in HGV5.

- *Ambient Cycling Test*
- *Extreme Temperature Cycling Test*
- *Hydrostatic Burst Test*
- *Bonfire Test*
- *Penetration Test*
- *Leak Before Break Test*
- *NDE Defect Size Determination*
- *Expected Service Performance Test*

Comments: *Because the design group did not construct or test a prototype the qualification tests above are extremely important. Only these tests can determine the real-world robustness of the final design.*

Production Unit and Batch Testing – Unit and batch testing must be carried out during the manufacturing process according to the applicable requirements set forth in *Production Tests and Examinations* and *Batch Tests* of HGV5.

DOT Unit Testing – Prior to shipment of any completed cylinder the tests listed below must be carried out, according to DOT Part 178.36 and DOT Part 178.37, with satisfactory results. These tests must be carried out in accordance with *Inspections and analyses* requirements of DOT Part 178.35 *General requirements for specification cylinders*.

- *Hydrostatic Test*
- *Flattening Test*
- *Physical Test*
- *Leakage Test*

Comments: *Because the design group did not construct or test a prototype the unit tests above are extremely important. Only these tests can determine the real-world robustness of the final design.*

Quality Assurance – In general, manufacturing must be carried out according to the sections *Manufacture* in HGV5, DOT Part 178.36, and DOT Part 178.37. Quality assurance practices must be established and operated to ensure all cylinders will be manufactured according to the qualified design. Quality assurance practices must meet the requirements of the *Quality Assurance* section in HGV5. The rules for *Identification of Material, Heat treatment,* and *Rejected cylinders* in DOT Part 178.36 and DOT Part 178.37 must also be followed.

Marking, Dispatch, and Records – Each cylinder must be marked per the requirements for *Markings* in DOT Part 178.35. Each cylinder must be marked and dispatched from the manufacturing facility per the requirements of *Marking and Dispatch* set forth in HGV5. The manufacturer shall follow the requirements for *Records of Manufacture* set forth in HGV5.

4.3.3. OPERATIONAL REQUIREMENTS

This section includes basic end-user operational requirements such as operating temperature, operating pressure, hydrogen purity, periodic inspection, and mounting effect on forklift dynamics.

Mounting – Four cylinders shall be mounted in a 2x2 matrix on either side of the fixed portion of the fork lift truck mast. The mounting system must be designed according to appropriate industry practice so as to 1) not interfere with the normal operation of the cylinders, 2) protect the cylinders from accidental damage, and 3) prevent the build-up of hydrogen gas should a leak occur in or around the cylinders.

Comments: *The mounting design will depend very much on the model of fork lift truck on which the cylinders are to be used. Because the combined weight of eight cylinders is approximately 219 pounds for the 4130 Q&T and 529 pounds for the 316L SS it may be possible to mount the cylinders in a way that might negatively alter the dynamics of the vehicle (such as to the telescoping portion of the mast). This is generally not recommended unless the appropriate analysis has been carried out to prove there is no dangerous effect on the vehicle dynamics.*

Service and Maximum Pressure – In accordance with HGV5, the service pressure of the cylinders is 5,000 psi and the service life shall be 10 years or 15,000 refueling cycles, whichever is reached first. The maximum pressure is not to exceed 6,250 psi immediately after filling, in accordance with HGV5.

Temperature – The hydrogen gas temperature and container temperature shall meet the requirements for *Settled Gas Temperatures* and *Container Temperatures* set forth in HGV5. In general, the cylinders shall not be placed in an environment with an ambient temperature below -25°C or above 45°C for an extended period of time.

Comments: *It is important to note that temperature was not a major consideration during the design process. It should also be noted that the temperatures at which HGV5 (15°C) and DOT (21°C) define the service pressure are different than the 0°C temperature at which the basic design requirement for service pressure is defined. It is recommended that the service pressure be redefined in the basic requirements to more closely match the definitions in HGV5 and DOT. The environmental requirement shown here is based on the recommendation that a fork lift truck should not be operated in such an environment for an extended period of time.*

Hydrogen Composition – The purity and composition of the hydrogen gas used in the cylinders shall meet the requirements set forth in *Gas Composition* of HGV5.

Inspection – Each cylinder shall be visually and ultrasonically inspected periodically while in service according to the *Periodic In-Service Inspection* requirements set forth in HGV5. Cylinders shall undergo periodic tests according to the *Periodic qualification and marking of cylinders* requirements set forth in DOT Part 173.34. Any cylinder involved in a collision, fire, or other event that may have caused damage to the cylinder shall be handled according to the *Conditions Requiring Immediate Inspections* set forth in HGV5. Any cylinder which has been pressurized beyond the maximum allowable pressure shall be handled according to the *Over-Pressurization* requirements set forth in HGV5.
Comments: *Based on a lack of experience in the industry it is not possible for the design group to recommend an appropriate inspection frequency for HGV5. The HGV5 minimum is every 36 months.*

5. NESTED HIGH PRESSURE TUBING COILS

This section of the report will review the design concept for the nested high pressure tubing coils, discuss the design evolution, and present the functional specifications for the final design proposal.

5.1. CONCEPT DESCRIPTION

The nested high pressure tubing coil design concept involves the creation of an assembly of several high pressure tubes that have been bent into individual coils. The individual coils are then nested inside one another. The diameter and bend radius of each tube generally decreases from the outside of the assembly to the inside. Like the traditional cylindrical vessel concept, the tubing coil assemblies are to be mounted to either side of a typical fork lift truck mast. This is illustrated in Figure 19 below.

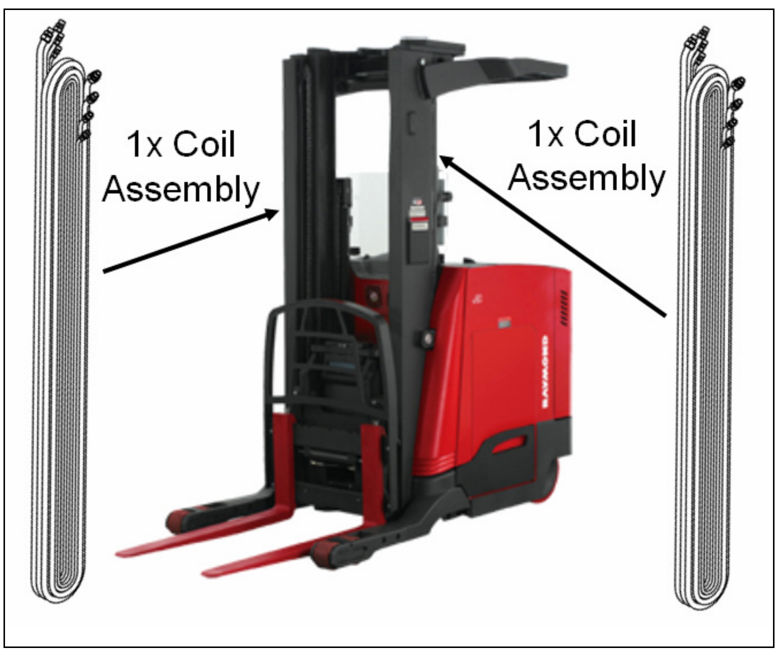


Figure 19 – Mast-Mounted Nested High Pressure Tubing Coils

Also as with the traditional cylindrical vessel concept, the assemblies are to be mounted to the fixed portion of the mast so as to minimize any adverse effects on the dynamics of the vehicle. The nested high pressure tubing concept is unique and relatively practical because the tubing is readily available and the shape and volume of the tank can be easily altered by changing the layout of the bends or increasing the number of coils. This means the design could be readily modified to fit within other unusually shaped spaces onboard a fork lift truck.

5.2. DESIGN EVOLUTION

5.2.1. TUBING SELECTION

The first step in the design of the tubing coils was to select the material, size, and configuration of the tubing to fit within the space available. As before, using information provided by the project industry advisor, the space available on the mast of a typical fork lift truck was determined to be 100 inches in height, 9 inches in width, and 4 inches in depth. The material selected for the tubing was 316L stainless steel for two reasons. First, as discussed previously, 316L SS is generally good at withstanding hydrogen embrittlement [12]. Second, since the tubing would be relatively small in diameter, it was thought that keeping the wall stress adequately below the yield strength of the 316L would be less difficult than it was in the cylindrical vessel design. The yield strength used was based on Certified Material Test Reports received with 316L SS tubing from Sandvik, a tubing supplier [28]. The lowest yield strength seen among the reports was approximately 36,000 psi (see Appendix L). The fatigue limit of 39,000 psi used in the design of the 316L SS cylinder was also used for the tubing coils [17]. Knowing both the space available and the material it was time to select tubing sizes and design the configuration of the tubing coils to fit within the space available.

Because the project industry advisor had previous experience working with Swagelok, a company specializing in fluid systems technology, this was the first source of tubing considered. During the fall semester research phase tubing with a 20,000 psi rated service pressure was used to create a preliminary design [12]. It was learned that a rated service pressure much closer to the 5,000 psi requirement should be used in order to maximize the volume and reduce the weight [12]. The Swagelok Tubing Data catalog [30] was used to select sizes of fractional stainless steel seamless tubing with service pressures of approximately 5,000 psi, see Table 4. Excerpts from the catalog are shown in Appendix L.

Table 4 – Swagelok Fractional Stainless Steel Seamless Tubing [30]

Tube OD in.	**Tube Wall Thickness**, in.																Swagelok Fitting Series
	0.010	**0.012**	**0.014**	**0.016**	**0.020**	**0.028**	**0.035**	**0.049**	**0.065**	**0.083**	**0.095**	**0.109**	**0.120**	**0.134**	**0.156**	**0.188**	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)																
1/16	5600	6800	8100	9400	12 000												100
1/8						8500	10 900										200
3/16						5400	7 000	10 200									300
1/4						4000	5 100	7 500	10 200①								400
5/16							4 000	5 800	8 000								500
3/8							3 300	4 800	6 500	7500①②							600
1/2							2 600	3 700	5 100	6700							810
5/8								2 900	4 000	5200	6000						1010
3/4								2 400	3 300	4200	4900	5800					1210
7/8								2 000	2 800	3600	4200	4800					1410
1									2 400	3100	3600	4200	4700				1610
1 1/4										2400	2800	3300	3600	4100	4900		2000
1 1/2											2300	2700	3000	3400	4000	4900	2400
2												2000	2200	2500	2900	3600	3200

① For higher pressures, see the Swagelok *Medium-Pressure Fittings* catalog, MS-02-335, or the Swagelok *High-Pressure Fittings* catalog, MS-01-34.
② Rating based on repeated pressure testing of the Swagelok tube fitting with a 4:1 design factor based upon hydraulic fluid leakage.

The design group was referred to the tubing of Table 4 by Dino Dutcher, a local sales representative for Swagelok [26]. It should be noted that Dino's initial recommendation was to use 2507 Super Duplex stainless steel tubing due to the need to maximize the strength-to-weight ratio. From research conducted by Sandia National Labs, it was learned that Super Duplex has a two-phase structure consisting of austenite and ferrite [27]. It is this two phase structure that makes duplex stainless steel especially strong but also contributes to its questionable compatibility with hydrogen [27]. Austenite is generally not very susceptible to hydrogen-assisted fracture but ferrite is extremely susceptible [27]. Due to the questionable compatibility with hydrogen it was decided that the material originally selected for the tubing, 316L stainless steel, would be used. It was at this point that the design group was referred to Table 4. Information on the bend radius allowed for each size of tubing was found in the Swagelok Tubing Tools and Accessories catalog [31]. An excerpt from the catalog is shown in Appendix L.

With the available tubing sizes and allowable bend radii identified, the design group required a method to calculate the tubing coil specifications. More specifically, the capacity and weight of each coil was required given the width, length, and bend radius of each coil, as well as the number of loops in each coil. Equations were developed for this purpose and are detailed in Appendix L. Using Table 4, the allowable bend radii, and the Appendix L equations, the specifications shown in Table 5 were developed.

Table 5 – Coil Specifications from Selected Standard Tubing Sizes

Outside Diameter [in]	1	0.875	0.625	0.5
Wall Thickness [in]	0.120	0.109	0.095	0.083
Swagelok Working Pressure [psi]	4700	4800	6000	6700
Minimum Bend Radius [in]	3.2	2.6	1.8	1.4
Bend Radius Utilized [in]	3.75	2.75	2	1.4
Number of Loops in Coil	3	4	5	6
Depth of Coil [in]	3.25	3.875	3.625	3.625
Length of Tubing in Coil [ft]	50.81	67.68	84.41	100.95
Capacity of Coil [L]	4.53	4.51	2.47	1.74
Weight of Coil [lb]	58.46	61.56	46.30	38.07

Total Length of Tubing in Assembly [ft]	303.85
Total Capacity of Coils in Assembly [L]	13.25
Total Weight of Assembly [lb]	204.39

In Table 5 the width and height of the outside coil was designed to fit within the perimeter of the space available. The width and height of the subsequent coils were designed to nest within one another. A gap of 1/8 inch was left between neighboring coils. It was possible to fit only four coils in the space available. The total capacity of the coils was found to be 13.25 liters. For two coil assemblies the total capacity was found to be only 26.5 liters, approximately 62% of the 43 liter requirement. Given the space available and the required wall thickness of the tubing, it would not be possible to improve the capacity significantly by changing the coil specifications. The weight of two coil assembles was found to be approximately 409 pounds. An illustration of the coil configuration specified in Table 5 is shown in Figure 20.

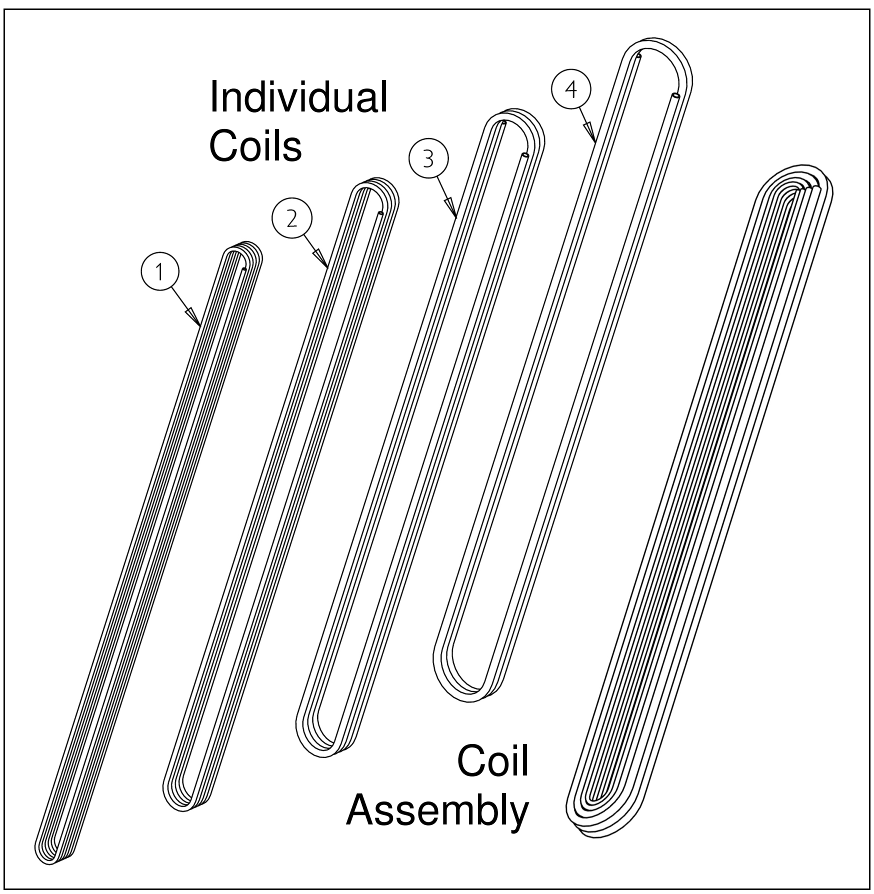


Figure 20 – Coil Configuration

It is important to note that two of the tubing sizes shown in Table 5, selected from Table 4, have suggested working pressures just below the service pressure requirement of 5,000 psi. This decision was made based on discussion with Wendy Caparco, a field engineer at Swagelok. According to Wendy, the allowable working pressures shown in Table 4 are conservative values that have been calculated using the maximum allowable dimensional variation of the tubing as well as a 0.94 usage factor. Based on this it was believed that there should be no problem using the two tubing sizes with working pressures just below 5,000 psi.

Wendy also indicated that the allowable working pressures shown in Table 4 are based on the availability of Swagelok fittings for terminating the tubing [28]. It is not recommended to use Swagelok fittings on tubing sizes that do not have working pressures shown [28]. Additionally, it was learned that the allowable working pressure is defined so as to provide for a 4:1 factor between it and the failure point of the associated fitting [28]. Because the design group wanted the ability to use Swagelok fittings, it decided to select only tubing sizes from Table 4 for which working pressures were shown.

5.2.2. STRESS ANALYSIS

Having the tubing sizes selected and the coil specifications worked out, it was possible to conduct stress analysis. The thick-walled cylinder equations from *Shigley's* Mechanical Engineering Design were used to calculate the maximum component stresses, which occur along the inside wall of each tube [R7]. Because 316L SS is a ductile material the Von Mises yield criterion could be used to predict yielding. The Von Mises stress was calculated using the equation from *Shigley's* [29]. A test pressure was needed to carry out the calculations. As previously described, the DOT Part 178 and HGV5 design standards would require the tank to be hydrostatically tested at a pressure of 8,333 psi and 7,500 psi, respectively [14],[15],[13]. The DOT Part 178 test pressure of 8,333 psi was adopted as it was the higher of the two test pressures. Even though the DOT Part 178 technically only applies to cylinders, it was decided to investigate whether the tubing would withstand the more demanding test pressure.

The results of the theoretical stress calculations are presented in Table 6. It should be noted that for all four tubes the Von Mises stress is less than the 36,000 psi yield strength indicating the wall thicknesses are adequate to withstand the DOT Part 178 test pressure. It was interesting to see that tubes with working pressures around 5,000 psi can withstand a test pressure 67% higher than that. Clearly the working pressures of Table 4 are indeed conservative values, as Wendy indicated [28]. It should also be noted that only the 1 inch OD tube has a stress just slightly above the conservative yield strength of 34,000 psi used by Taylor-Wharton, as discussed during the cylindrical vessel design [16]. The design factor between the theoretically calculated Von Mises stress and the 36,000 psi yield strength are also shown in Table 6. It is important to note, however, that the theoretically calculated Von Mises stresses of Table 6 applied to straight tubing only. The effect of bending on the maximum inside wall stress had yet to be investigated.

While the design factors in Table 6 may seem low, it is important to note that nearly all the calculations and analyses carried out during the project, for all designs, were carried out at the hydrostatic test pressures not the service pressure. This means there is already a significant design factor between the service pressure and the hydrostatic test pressures.

Table 6 – Stress Analysis using Thick-Walled Cylinder Equations

D[in]	1	0.875	0.625	0.5
t[in]	0.120	0.109	0.095	0.083
Swagelok Working Pressure [psi]	4700	4800	6000	6700
Test Pressure [psi]	8333	8333	8333	8333
Tangential Stress [psi]	31122	29815	23917	21762
Longitudinal Stress [psi]	11395	10770	7829	6715
Radial Stress [psi]	-8333	-8275	-8259	-8333
Von Mises Stress [psi]	**34169**	**32987**	**27866**	**26063**
316L Yield Strength [psi]	36000	36000	36000	36000
Design Factor	**1.05**	**1.09**	**1.29**	**1.38**

The next step in the stress analysis was to carry out finite element analysis (FEA) to verify the stresses obtained theoretically and investigate the effect of bending. The setup of the model in Pro/Mechanica was as follows; a J-shaped section of tube had a displacement constraint placed on the end of the straight section and an internal test pressure of 8,333 psi applied throughout the inside. Only 90 degrees of the bend was included in the model and a symmetry constraint was placed at its end. This meant the model being analyzed was effectively a long straight section of tube, followed by a complete 180 degree bend, followed by a long straight section of tube. The Von Mises stress results were visualized using a fringe plot like the one shown in Figure 21 for the 1 inch OD tube. It is important to note that the maximum stress observed was located directly at the displacement constraint and was therefore ignored because such a constraint would not actually exist. The maximum inside wall stress was queried manually for the straight sections of the tubing. The maximum inside wall stress for the bent sections was retrieved from graphs of the stress as a function of the arc length around the inside rim at the end of the bend (See graphs in Appendix L). These values are shown on the fringe plots for each tubing size. The fringe plots for the other tubing sizes can be found in Appendix I: FEA Results (Tubing Coils).

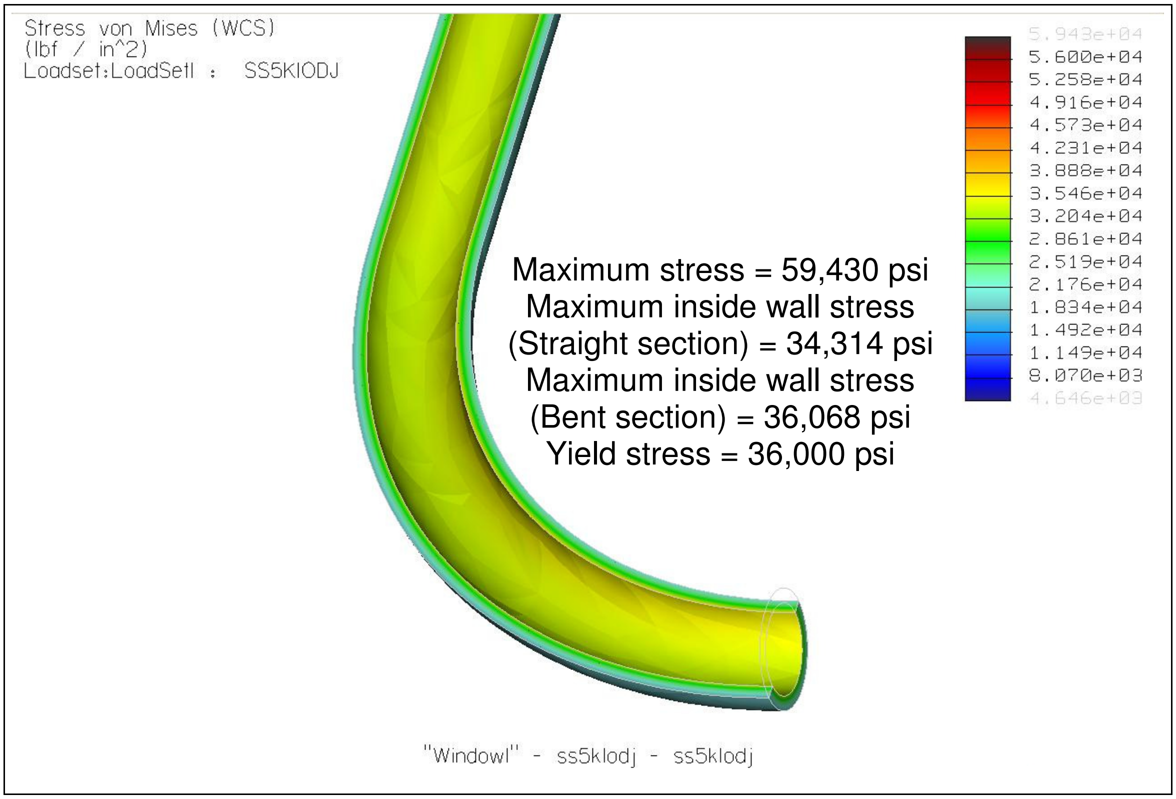


Figure 21 – 1" OD Tube FEA Results

A comparison of the FEA results to the theoretically calculated stresses can be seen in Table 7. From the table it is clear that the theoretical results matched up very well with the FEA results for the straight section of the tube. A close look at the fringe plot of Figure 21, however, reveals that there is variation in the wall stress on each side of the bend as the tube is bent. The FEA results for the bent section of the tube are higher than the theoretical results. The 1 inch and 7/8 inch OD tubes were most highly effected. The stress ratio between the straight and bent sections of the tube was calculated for each tubing size and included in Table 7. It should be noted that for all of the tubing sizes, the stress ratios between the straight and bent sections of the tubes are the same. The design factors between the Von Mises stress from the straight and bent sections and the material yield strength are also shown in Table 7.

Table 7 – Stress Analysis using Finite Element Analysis

Outside Diameter [in]	1	0.875	0.625	0.5
Von Mises Stress by Hand [psi]	34169	32987	27866	26063
Von Mises Stress from FEA (Straight Section) [psi]	34314	33213	28084	26317
Von Mises Stress from FEA (Bent Section) [psi]	**36068**	**34950**	**29514**	**27583**
Stress Ratio Between Straight and Bent Sections	1.05	1.05	1.05	1.05
316L Yield Strength [psi]	36000	36000	36000	36000
Design Factor (Straight Section)	1.05	1.08	1.3	1.4
Design Factor (Bent Section)	**0.998**	**1.03**	**1.2**	**1.3**

From Table 7 it can be observed that only the design factor of the 1 inch OD tube dropped below a value of one when the stress from the bent section was considered. This indicated that the 1 inch OD tube could not be approved for testing at the DOT Part 178 pressure of 8,333 psi. This was not a significant issue because the DOT Part 178 design standard is written specifically for cylindrical vessels and not alternative designs like nested coils of high pressure tubing. This meant that the HGV5 hydrostatic test pressure of 7,500 psi must be used. Due to the close proximity of the 1 inch OD design factor to a value of 1, it was assumed that dropping the test pressure by 1,500 psi and carrying out another FEA would certainly result in an acceptable design factor. Based on this assumption FEA was not carried out for any of the tubing sizes at the 7,500 psi test pressure.

5.2.3. DEFLECTION ANALYSIS

It was decided by the design group that the deflection of each tube, due to the internally applied pressure, should be investigated to make sure the spacing between the coils was adequate and that they would not require any special containment structures. It was relatively easy to create a fringe plot of total deflection from the finite element analysis already carried out. The displacement magnitude fringe plot for the 1 inch OD tube is shown in

Figure 22. Displacement magnitude is the total deflection calculated from the deflections in each of the coordinate directions (longitudinal, radial, and tangential). The applicable deflection equations, taken from *Shigley's* Mechanical Engineering Design, are shown in Appendix L [32]. The maximum displacement magnitude of 0.02035 inches observed in Figure 22 is much smaller than the 1/8 inch gap between the coils. The largest maximum displacement magnitude of 0.03475 inches was observed in the 1/2 inch OD tube. This value was again much smaller than the 1/8 inch gap between the coils. Based on this it was decided that the gap was sufficient in order to allow the coils to expand and that no special containment structures would be necessary.

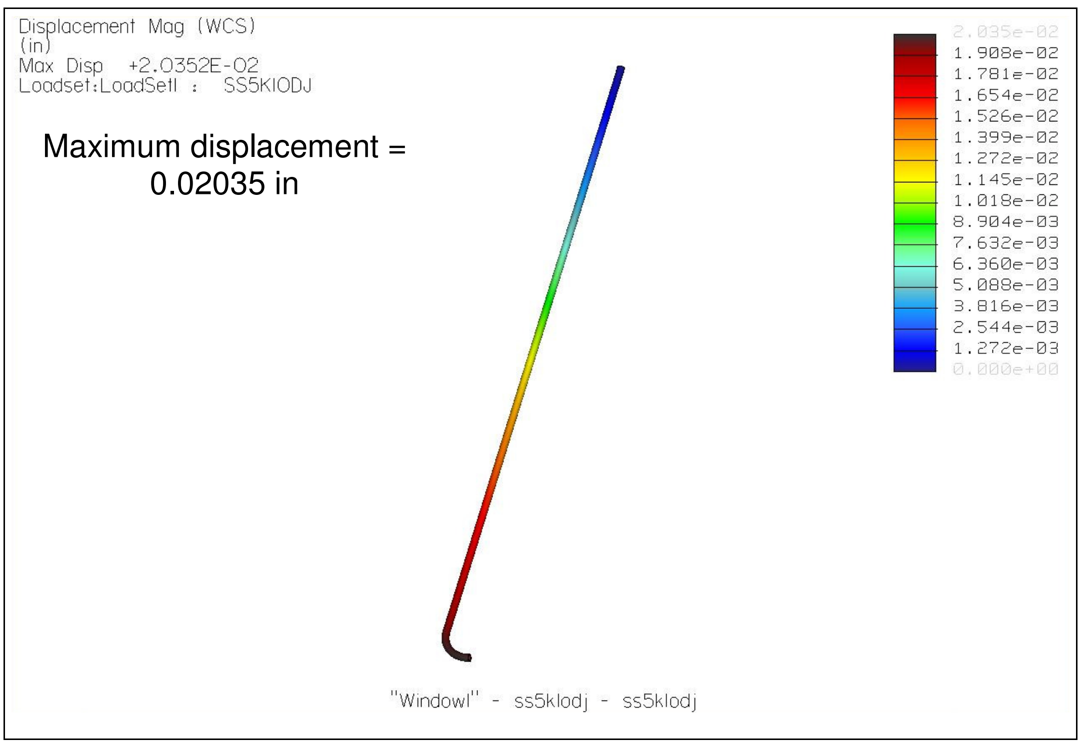


Figure 22 – 1" OD Tube Displacement

While this would have been sufficient, the design group decided to try verifying its deflection findings using the theoretical equations in Appendix L. This proved to be impossible due to the differences in the finite element technique and the analytical technique. The FEA required a displacement constraint that the theoretical equations did not require. Additionally, it was not possible to account for the bend in the theoretical equations. Every attempt to match the results of the theoretical equations to the FEA results was unsuccessful. This does not discredit the FEA results because they actually overstate the magnitude of the displacement as compared to the theoretical equations. Because the displacement magnitude was acceptable when determined from the FEA it would surely be acceptable from the theoretical equations as well.

5.2.4. TERMINATION DESIGN

With the stress and deflection analysis of the coil configuration complete, it was possible to design the way in which the ends of the coils would be terminated. It was decided that each coil would be terminated with a Swagelok fitting at one end and a Swagelok cap at the other. This was possible, as previously mentioned, because all of the tubing sizes selected from Table 4 had suggested allowable working pressures shown. This meant one end of each coil would have a threaded opening allowing it to be connected to the hydrogen supply system of a fork lift truck, while the other end would be sealed with a cap. It was desirable to use fittings with ¾-14 NGT threads, the same thread type commonly used on high pressure cylinders [12]. Wendy at Swagelok aided in the selection of the proper fittings and caps for the selected tube sizes. Drawings of the fittings and caps are included with the drawings of the finished tubing coils in Appendix F. Knowing the size of the fittings and caps it was possible to design the terminating bends of the coil. Terminating bends were necessary to allow enough space between the tubes for the fittings and caps to be properly installed. The finished termination design is illustrated in the screenshot of Figure 23.

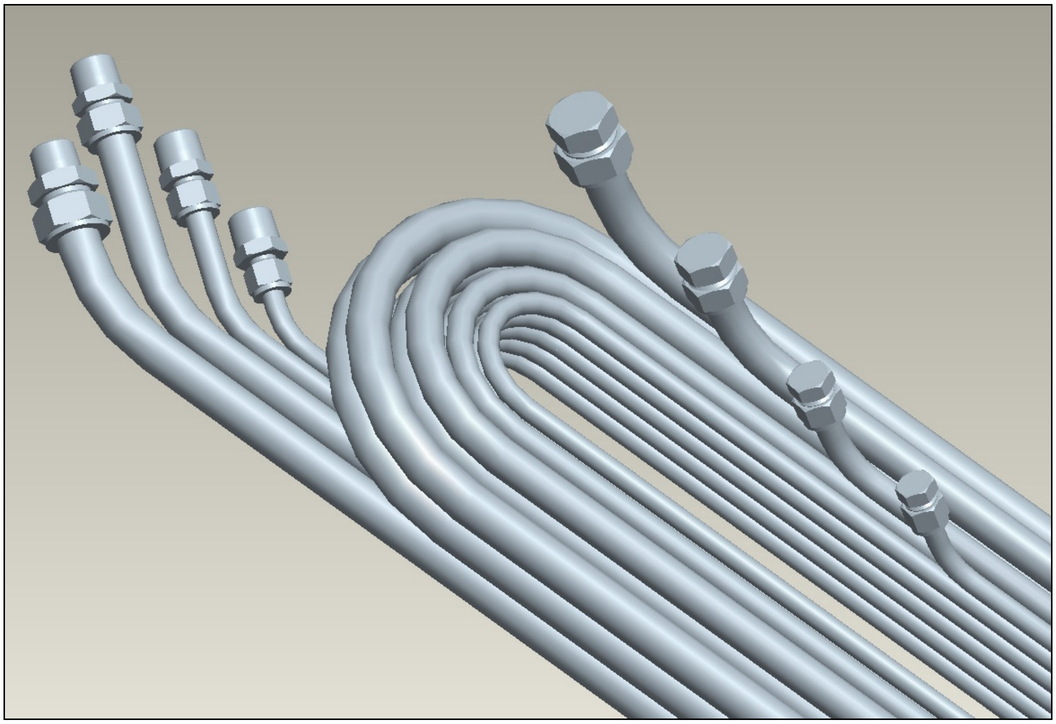


Figure 23 – Termination Design

As shown in Figure 23, the bottom loop in each coil was extended past the assembly and then bent so as to space it out from the other tubes. The length of tubing supplied after the terminating bend was selected so as to provide access for a wrench during installation of the fitting. Due to the need for the coils to nest together it was not possible for the sealed ends of the tubes to deviate from the path of their respective coils. To allow for the installation of the caps the top loop in each coil was bent upward at different positions so as to space it out from the other tubes. This termination design requires that the caps be installed only after the coils have been nested together. In carrying out the termination design it was found that it would be impossible to terminate the ends of the coils without deviating from the available space. This was discussed with the project industry advisor and determined not to be a critical problem because the design was not being developed for a specific fork lift truck model. The available space was a soft requirement that was necessary to establish the approximate dimensions allowed. If the coil specifications had been adjusted to allow the terminating bends, fittings, and caps to fit within the available space the volume would have dropped significantly. The addition of the terminating bends had little effect on the total length, capacity, and weight of the tubing coils, as shown in Table 8.

Table 8 – Corrected Coil Specifications Accounting for Terminations

Outside Diameter [in]	1	0.875	0.625	0.5	Total
Wall Thickness [in]	0.120	0.109	0.095	0.083	
Length of Tubing in Coil [ft]	50.81	67.68	84.41	100.95	303.85
Corrected Length [ft]	52.09	68.62	84.83	100.82	306.36
Capacity of Coil [L]	4.53	4.51	2.47	1.74	13.25
Corrected Capacity [L]	4.65	4.58	2.48	1.74	13.44
Weight of Coil [lb]	58.46	61.56	46.30	38.07	204.39
Corrected Weight [lb]	59.93	62.43	46.53	38.02	206.91

From the data shown in Table 8, it was determined that the addition of the terminating bends led to a 0.8% increase in the length of tubing used, a 1.4% increase in the hydrogen capacity, and a 1.2% increase in the total weight. It should be mentioned that if the tubing coil assemblies were to be produced in a large-scale manufacturing operation the termination design would most likely change. Based on discussion with the project industry advisor, it would not be especially practical in a large-scale manufacturing operation to use the kind of fittings and caps offered by Swagelok. It would be more practical for the ends of the four tubes to be welded into a solid manifold. Each set of tube ends would likely be welded into separate manifolds; one manifold would supply hydrogen to the supply system of the fork lift truck while the other would be equipped with an emergency pressure regulator or release valve. Though more practical for a large-scale manufacturing operation, a manifold-based termination approach was beyond the scope of the work that could be carried out.

5.2.5. MATERIAL QUOTE

Though it was not practical for the design group to seek out a quote for manufacturing of the tubing coils, it was possible to obtain a quote for the materials that would be required. Because the tubing, fittings, and caps were all selected from Swagelok catalogs it was relatively straightforward to acquire a quote for those materials from Wendy and Dino. Based on discussion with the project industry advisor, it was important to investigate how the cost of the materials would vary depending on the number of tubing coils being produced. In other words, what discount might be applied for a large order? The design group and industry advisor defined a large order as approximately 5,000 fork lift trucks, or 10,000 tubing coil assemblies. The lengths required for the tubing, shown in Table 8, were rounded up to 75 feet, 100 feet, 100 feet, and 125 feet for the 1 inch, 7/8 inch, 5/8 inch, and 1/2 inch OD tubes, respectively. This was done in order to provide allowances for scrap during manufacturing. The quote obtained from Swagelok for 10,000 assemblies is shown in Table 9. The total cost of the material was found to be $75.28 million. Approximately $1.55 million (or 2%) of the total cost was attributed to the fittings and caps with the remaining $73.73 million (or 98%) being attributed to the tubing.

Table 9 – Swagelok Quote for 10,000 Tubing Coils [26] [28]

Part #	Description	List Price	Qty	Discount	Extended Cost
SS-1610-C	S.S. 1" Tube Cap	$31.70	10000	26%	$234,580.00
SS-1410-C	S.S. 7/8" Tube Cap	26.60	10000	26%	$196,840.00
SS-1010-C	S.S. 5/8" Tube Cap	$12.45	10000	26%	$92,130.00
SS-810-C	S.S. 1/2" Tube Cap	$10.36	10000	26%	$76,664.00
SS-1610-1-12	S.S. 1" OD Tube x 3/4" MNPT Connector	$44.50	10000	26%	$329,300.00
SS-1410-1-12	S.S. 7/8" OD Tube x 3/4' MNPT Connector	$41.90	10000	26%	$310,060.00
SS-1010-1-12	S.S. 5/8" OD Tube x 3/4" MNPT Connector	$21.90	10000	26%	$162,060.00
SS-810-1-12	S.S. 1/2" OD Tube x 3/4" MNPT Connector	$20.40	10000	26%	$150,960.00
SS-T16-S-120-20	S.S. 1" OD x .120 wall tubing, per foot	$33.80	1250000	40%	$25,350,000.00
SS-T14-S-109-20	S.S. 7/8' OD x .109 wall tubing, per foot	$40.60	1250000	40%	$30,450,000.00
SS-T10-S-095-20	S.S. 5/8" OD x .095 wall tubing, per foot	$17.73	1000000	40%	$10,638,000.00
SS-T8-S-083-20	S.S. 1/2" OD x .083 wall tubing, per foot	$16.20	750000	40%	$7,290,000.00

Given the incredible costs involved, the lengths of tubing required were reduced to a more conservative 57 feet, 75 feet, 93 feet, and 111 feet for the 1 inch, 7/8 inch, 5/8 inch, and 1/2 inch OD tubes, respectively. These lengths were determined by increasing those shown in Table 8 by a fixed 10%, rather than by an arbitrary rounding-up. This simply reduced the allowances for scrap during manufacturing. Using the prices shown in Table 9, the cost of the tubing only was found to be $50.51 million; this represented a 31% reduction in the cost of the tubing. Obviously, the initial estimation of the lengths of tubing required was unnecessarily wasteful.

While the total costs were impressive it was more useful to observe the difference in the unit cost of one tubing coil assembly with and without the discounts shown in Table 9. The unit cost of one tubing coil assembly with the discounts was found to be $5,206. The unit cost of one tubing coil assembly without the discounts was found to be $8,629. The per-unit discount received when ordering 10,000 units was therefore found to be approximately 40%.

It was learned during the design that Swagelok was probably not the best option for the supplier of the tubing should the tubing coils be manufactured on a large-scale. First, Swagelok can only offer tubing in discrete lengths of 20 feet, which would require orbital welding multiple tubes together in order to manufacture a single coil. Second, Swagelok receives its tubing from a manufacturer and distributes it with a mark-up in price. It made more sense for the design group to locate a supplier that manufactured the tubing itself and could supply it in long, continuous lengths. The supplier that was eventually located was Handy & Harman Tube Company of Camdel Metals Corporation. Through discussion with manufacturing engineer Michael Bauman it was learned that H&H could provide the desired sizes of 316L SS tubing in long, continuous lengths that could be wound on spools or coiled in large crates for shipping [34].

The quote eventually received from the sales department at H&H is shown in Table 10. Using the revised lengths of tubing from the Swagelok quote calculations, the total cost of the tubing was determined to be $22.12 million. The total cost of the tubing if supplied from H&H was found to be 56% lower than the total cost if supplied by Swagelok. When the $1.55 million cost of the fittings and caps provided by Swagelok was added to the H&H tubing cost, the total was found to be $23.67 million. The unit cost of one tubing coil assembly, using the Swagelok discounted fittings and caps and the H&H tubing, was found to be $2,375. This unit cost represented a 54% reduction in price compared to the discounted unit cost offered by Swagelok.

Table 10 – Handy and Harman Quote for 10,000 Tubing Coils [34]

QUANTITY	*UNIT*	*PRODUCT NAME*	*ALLOY*	*DESCRIPTION*	*UNIT PRICE*
125,000- 1,250,000	Feet	.500" OD X .083" Wall	316/316L	Smls, ASTM A269 Annealed 450' Min Coil Lengths	$4.01
125,000- 1,250,000	Feet	.625" OD X .095" Wall	316/316L	Smls, ASTM A269 Annealed 325' Min Coil Lengths	$5.98
100,000- 1,000,000	Feet	.875" OD X .120" Wall	316/316L	Smls, ASTM A269 Annealed 175' Min Coil Lengths	$8.90
75,000- 750,000	Feet	1.00" OD X .134" Wall	316/316L	Smls, ASTM A269 Annealed 125' Min Coil Lengths	$9.67

The official quotes received from both Swagelok and Handy and Harman Tube Company can be found in Appendix L: Additional Materials (Tubing Coils). Having already dealt with dimensional variations in the design of the traditional cylinders, the design group moved on to the issue of dimensional variations in the tubing supplied by H&H and how it might impact the work already carried out.

5.2.6. DIMENSIONAL VARIATIONS

Michael at Handy and Harman Tube Company provided the design group with important information regarding the allowable dimensional variations of the 316L SS tubing to be used for the tubing coils. According to Michael, and the ASME Code for Pressure Piping B31, wall thickness can vary circumferentially within 10% of the nominal wall thickness while concentricity can vary axially within 5% of the nominal wall thickness [34]. Additionally, the outside diameter can vary by +/-0.005 inches for the tubing sizes being used [34]. To investigate the effect of the dimensional variation on the inner wall stress the theoretical equations from *Shigley's* were again used [20]. This time, the outside diameter of the tubing was increased by 0.005 inches and the wall thickness was decreased by 10% in order to investigate the worst-case dimensional variation. The calculations for the maximum dimensional variation with the DOT Part 178 8,333 psi test pressure are shown in Table 11.

Table 11 – Maximum Dimensional Variation with 8,333 psi Test Pressure

Outside Diameter [in]	1.005	0.880	0.630	0.505
Wall Thickness [in]	0.108	0.098	0.086	0.075
Test pressure [psi]	8333	8333	8333	8333
Von Mises Stress (Straight Section) [psi]	40546	39342	34094	32451
Stress Ratio Between Straight and Bent Sections (From Table 7)	1.05	1.05	1.05	1.05
Estimated Von Mises Stress (Bent Section) [psi]	**42573**	**41309**	**35798**	**34073**
Yield Strength [psi]	36000	36000	36000	36000
Design Factor (Straight Section)	0.89	0.92	1.06	1.11
Design Factor (Bent Section)	**0.85**	**0.87**	**1.01**	**1.06**

Recall from the finite element analysis that the maximum inner wall stress was observed along the rim at the end of the bent section of the tube. In Table 7 the ratio between the inner wall stress of the straight section and the inner wall stress of the bent section was calculated for all tubes to be approximately 1.05. This was calculated in order to estimate what the maximum inner wall stress of the bent section is without conducting any FEA. In Table 11 the 1.05 stress ratio was used to estimate the maximum inner wall stress of the bent section *from* the theoretically calculated maximum inner wall stress of the straight section. It was clear in Table 11 that, with the DOT Part 178 test pressure applied and the maximum dimensional variations possible, the 1 inch and 7/8 inch OD tubes exceeded the yield strength of the material. This was not a concern because it was already determined during the stress analysis that the 1 inch OD tube was not acceptable for testing with the DOT Part 178 test pressure.

The next step in the investigation was to reduce the pressure from the DOT Part 178 test pressure to the HGV5 test pressure of 7,500 psi; these results are shown in Table 12. Even though the performance of the 1 inch and 7/8 inch OD tubes improved, it was again clear that they exceeded the yield strength of the material. It was also clear from the design factors that both tubes were much closer to being acceptable than previously.

Table 12 – Maximum Dimensional Variation with 7,500 psi Test Pressure

Outside Diameter [in]	1.005	0.880	0.630	0.505
Wall Thickness [in]	0.108	0.098	0.086	0.075
Test pressure [psi]	7500	7500	7500	7500
Von Mises Stress (Straight Section) [psi]	36493	35409	30685	29207
Stress Ratio Between Straight and Bent Sections (From Table 7)	1.05	1.05	1.05	1.05
Estimated Von Mises Stress (Bent Section) [psi]	**38317**	**37179**	**32219**	**30667**
Yield Strength [psi]	36000	36000	36000	36000
Design Factor (Straight Section)	0.98	1.02	1.17	1.23
Design Factor (Bent Section)	**0.94**	**0.97**	**1.12**	**1.17**

There were generally two options available to solve the problem with the 1 inch and 7/8 inch OD tubes. First, the yield strength could be required in the functional specification to be higher than the stresses observed. Second, the allowable dimensional variation on the 1 inch and 7/8 inch tubes could be specified in the functional specification such that the stresses would not exceed the yield strength of the material. Due to the proximity of the observed stresses in Table 12 (38,317 psi and 37,179 psi) to the fatigue limit of the material (39,000 psi), it was decided it made more sense to limit the allowable dimensional variation. That led to the question of how much the allowable dimensional variation needed to be reduced. It was found in Table 13 that a 5% variation in wall thickness was acceptable for both the 1 inch and 7/8 inch OD tubes. It is unknown what effect this requirement would have on the availability or cost of the tubing.

Table 13 – Max OD Variation, 5% Wall Reduction (7,500 psi Test Pressure)

Outside Diameter [in]	1.005	0.880
Wall Thickness [in]	0.114	0.098
Test pressure [psi]	7500	7500
Von Mises Stress (Straight Section) [psi]	33754	32712
Stress Ratio Between Straight and Bent Sections (From Table 7)	1.05	1.05
Estimated Von Mises Stress (Bent Section) [psi]	**35441**	**34347**
Yield Strength [psi]	36000	36000
Design Factor (Straight Section)	1.07	1.10
Design Factor (Bent Section)	**1.02**	**1.05**

The completion of the dimensional variation investigation nearly concludes the discussion of the design evolution of the nested high pressure tubing coils. Some additional considerations concerning manufacturing of the tubing coils will now be discussed.

5.2.7. MANUFACTURING CONSIDERATIONS

One significant aspect of the tubing coils is the manner in which they would be manufactured. Several professional opinions were sought concerning this but no definitive answers were received. Michael at Handy and Harman Tube Company was not sure whether tube bending specialists would have the ability to straighten such long lengths of tube and perform such tight bends, both of which would almost have to take place at the same time [34]. He imagined that such a design would typically be manufactured by welding or brazing together a series of J-shaped bent tubes [34]. Michael referred the design group to another company, Precision Tube Bending. A contact at PTB, Philip Stephen, indicated that coil type bends require specialized tooling that most companies do not have readily available [35]. He also indicated that the approach to manufacturing such a coil design depends on several factors; 1) the quantity, 2) the investment the customer wants to make, and 3) the type of machinery being used [35]. No other useful information concerning manufacturing was received as a result of communication carried out with industry.

Based on the communication carried out with industry, and with the project industry advisor, it was deemed very plausible that manufacturing the assembly might require welding or brazing to connect smaller bent pieces of tubing together. Such an approach might prove more effective in a large scale manufacturing operation. It may be possible to manufacture a continuous coil but the investment by Raymond Corporation would have to be significant in order for the process necessary to be developed.

5.3. FUNCTIONAL SPECIFICATION

This section of the report presents the functional specifications for the tubing coil tank design. A condensed version of the functional specification is presented in Appendix F. The functional specifications are the detailed requirements dictated by the finished design in order to meet the basic requirements presented previously in the report as well as the HGV5 design standard requirements for Type 1 compressed hydrogen tanks.

5.3.1. PHYSICAL REQUIREMENTS

This section includes the geometry, material, and physical features that characterize the design and are required to meet the basic requirements.

Geometry – The geometries of the raw tubing, the individual tubing coils, and the finished assembly shall conform to the drawings contained within Appendix F of this report.

Capacity – One assembly of tubing coils will allow for the storage of approximately 13.4 liters of 5,000 psi compressed hydrogen. Two assemblies shall be used on a single fork lift to allow for the combined storage of approximately 26.8 liters of 5,000 psi compressed hydrogen. ***Comments:*** *Given the available space of 4x9x100 inches it was not possible to incorporate enough tubing into the coils to reach the capacity requirement of 43 liters for two tubing coil assemblies. Relatively minor modifications to the design, such as increasing the number of loops in each coil, would allow the capacity requirement to be reached.*

Tubing Material – The material of the raw tubing shall be 316L stainless steel to reduce the risk of hydrogen-induced embrittlement over time. The 316L stainless steel shall have a yield strength no lower than 36,000 psi. The 316L stainless steel shall have a fatigue limit (based on 10^6-10^9

cycles to failure) above the specified yield strength to reduce the risk of fatigue failure caused by refueling cycling. The 316L stainless steel must meet the *Material Qualification Tests and Requirements* set forth in HGV5.

Tubing Tolerances – The raw tubing shall meet the following requirements for maximum allowable dimensional variations;

- For the 1" and 7/8" OD tubes; Maximum of 5% variation in the wall thickness (from the nominal) as measured at various points around the circumference of the tube

  Comments: *This restriction was necessary in order to prevent the theoretical stress during the HGV5 hydrostatic test from exceeding the yield strength of the material. It may be possible in the real-world for these tubes to pass the hydrostatic test with a maximum of 10% variation but that could not be determined by the design group due to the inability to build a prototype and carry out the hydrostatic test.*
- For the 5/8" and 1/2" OD tubes; Maximum of 10% variation in the wall thickness (from the nominal) as measured at various points around the circumference of the tube
- Maximum of 5% variation (from the nominal wall thickness) in the concentricity of the inner and outer diameter
- Maximum of +/-.005" deviation in the outside diameter

Termination – The fittings and end caps specified in the drawings of Appendix F shall be used to terminate the tubing coils of the tank. The fittings shall provide a standard ¾-14 NGT thread for connection to an appropriate valve and the hydrogen supply system on the fork lift truck.

External Surfaces – The external surfaces of the tubing coils shall meet the requirements for *External Surfaces* set forth in HGV5.

5.3.2. MANUFACTURING REQUIREMENTS

This section includes tests that must be carried out in order to qualify the design for subsequent manufacturing, as well as the manufacturing procedures and tests required to confirm a quality final product that meets the other requirements set forth in this functional specification.

Qualification Testing – Prior to shipment of a completed tubing coil tank the design qualification tests listed below must be carried out, according to HGV5, with satisfactory results. Any change in the tubing coil tank design may require some or all qualification tests to be repeated according to *Change of Design* in HGV5.

- *Ambient Cycling Test*
- *Extreme Temperature Cycling Test*
- *Hydrostatic Burst Test*
- *Bonfire Test*
- *Penetration Test*
- *Leak Before Break Test*
- *NDE Defect Size Determination*
- *Expected Service Performance Test*

Comments: *Because the design group did not construct or test a prototype the qualification tests above are extremely important. Only these tests can determine the real-world robustness of the final design.*

Tube Bending – The custom manufacturing process required to produce the bends detailed in the drawings of Appendix F shall be designed and implemented according to appropriate industry practice.

Comments: *It may not be practical to manufacture a continuous coil because the investment would have to be significant in order for the* *process to be developed. The language necessary to allow the orbital welding of smaller pieces of tubing together was not included above due to the complexities that would result in order to meet the HGV5 standard. If welding were to take place during manufacturing the section in HGV5 concerning* Alternative Construction and Materials *would have to be met.*

Production Unit and Batch Testing – Unit and batch testing must be carried out during the manufacturing process according to the applicable requirements set forth in *Production Tests and Examinations* and *Batch Tests* of HGV5.

Quality Assurance – In general, manufacturing must be carried out according to the section *Manufacture* in HGV5. Quality assurance practices must be established and operated to ensure all tubing coil tanks will be manufactured according to the qualified design. Quality assurance practices must meet the requirements of the *Quality Assurance* section in HGV5.

Marking, Dispatch, and Records – Each tubing coil tank must be marked and dispatched from the manufacturing facility per the requirements of *Marking and Dispatch* set forth in HGV5. The manufacturer shall follow the requirements for *Records of Manufacture* set forth in HGV5.

5.3.3. OPERATIONAL REQUIREMENTS

This section includes basic end-user operational requirements such as operating temperature, operating pressure, hydrogen purity, periodic inspection, and mounting effect on forklift dynamics.

Mounting – The tubing coil tanks shall be mounted on either side of the fixed portion of the fork lift truck mast. The mounting system must be designed according to appropriate industry practice so as to 1) not interfere with the normal operation of the tank, 2) protect the tank from accidental damage, and 3) prevent the build-up of hydrogen gas should a leak occur in or around the tank.
Comments: *The mounting design will depend very much on the model of fork lift truck on which the tubing coil tank is to be used. Because the combined weight of two tanks is approximately 414 pounds it may be possible to mount the tanks in a way that might negatively alter the dynamics of the vehicle (such as to the telescoping portion of the mast). This is generally not recommended unless the appropriate analysis has been carried out to prove there is no dangerous effect on the vehicle dynamics.*

Service and Maximum Pressure – In accordance with HGV5, the service pressure of the tubing coil tank is 5,000 psi and the service life shall be 10 years or 15,000 refueling cycles, whichever is reached first. The maximum pressure is not to exceed 6,250 psi immediately after filling, in accordance with HGV5.

Temperature – The hydrogen gas temperature and container temperature shall meet the requirements for *Settled Gas Temperatures* and *Container Temperatures* set forth in HGV5. In general, the tubing coil tank shall not be placed in an environment with an ambient temperature below -25°C or above 45°C for an extended period of time.
Comments: *It is important to note that temperature was not a major consideration during the design process. It should also be noted that the temperatures at which HGV5 (15°C) and DOT (21°C) define the service pressure are different than the 0°C temperature at which the basic design requirement for service pressure is defined. It is recommended that the*

service pressure be redefined in the basic requirements to more closely match the definitions in HGV5 and DOT. The environmental requirement shown here is based on the recommendation that a fork lift truck should not be operated in such an environment for an extended period of time.

Hydrogen Composition – The purity and composition of the hydrogen gas used in the tubing coil tank shall meet the requirements set forth in *Gas Composition* of HGV5.

Inspection – Each tubing coil tank shall be visually and ultrasonically inspected periodically while in service according to the *Periodic In-Service Inspection* requirements set forth in HGV5. Any tubing coil tank involved in a collision, fire, or other event that may have caused damage to the tank shall be handled according to the *Conditions Requiring Immediate Inspections* set forth in HGV5. Any tubing coil tank which has been pressurized beyond the maximum allowable pressure shall be handled according to the *Over-Pressurization* requirements set forth in HGV5.
Comments: *Based on a lack of experience in the industry it is not possible for the design group to recommend an appropriate inspection frequency. The HGV5 minimum is every 36 months.*

6. RECTANGULAR TANK (OVERHEAD GUARD)

This section of the report will review the design concept for the rectangular tank, discuss the design evolution, and present the functional specifications for the final design proposal.

6.1. CONCEPT DESCRIPTION

The rectangular tank design concept was based on the overhead guard of a fork lift truck. The overhead guard consists of a welded grid of box tubing that protects the operator of the lift truck from falling objects. Figure 24 identifies an overhead guard on a fork lift truck and illustrates the rectangular tank created by the design group to take its place. It should be noted that the design group did not deal with specific geometry from the overhead guard of a particular lift truck, it simply created its own representation of the overhead guard and explored the implications of using it as a compressed hydrogen tank. In practice, a rectangular-based design would not have to be limited to the overheard guard, it might also be applied to other box tube structures on a fork lift truck, such as the chassis or mast.

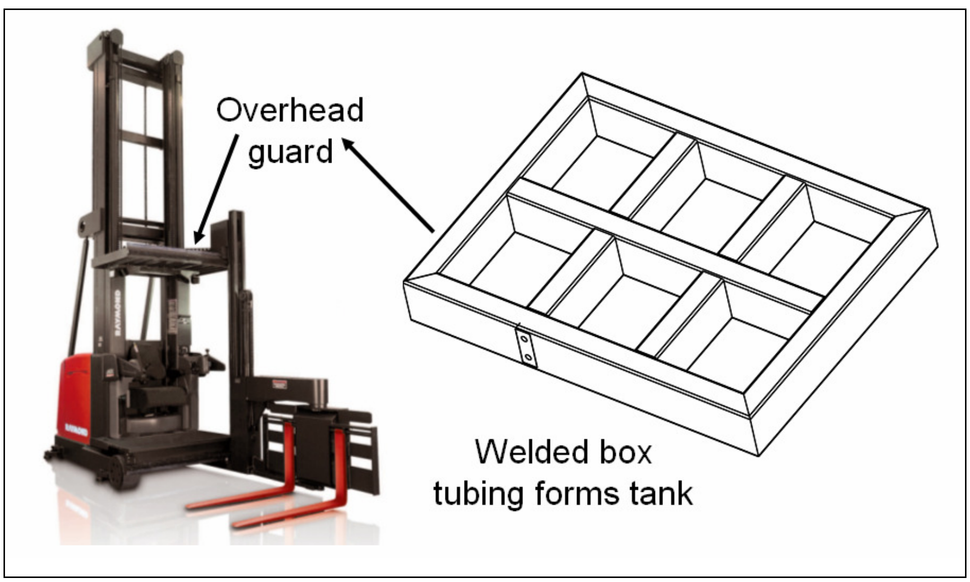


Figure 24 – Overhead Guard Becomes Tank

6.2. DESIGN EVOLUTION

6.2.1. MATERIAL SELECTION

As with both the cylindrical vessels and the tubing coils, there was a desire to use 316L stainless steel (SS) because it is generally good at withstanding hydrogen embrittlement [12]. It was decided to use the conservative yield strength value of 34,000 psi provided by Taylor-Wharton [16]. As before, the fatigue limit (based on 10^6-10^9 cycles to failure) was found to be 39,000 psi [17]. This fatigue limit reduces the risk of failure due to refueling for two reasons. First, the 10^6 cycles on which the fatigue limit is based is much higher than the 15,000 refueling cycles allowed by the basic design requirements. Second, it is *above* the yield strength and will, theoretically, never be reached because the rectangular tank was designed to operate below its yield strength at all times.

6.2.2. CROSS-SECTION DESIGN

With the material selected it was possible to move on to initial sizing of the tank and the cross-section design of the box tubing members that would make up the tank. Depending on the model of lift truck, an overhead guard might have outer dimensions in the neighborhood of 5 feet by 3 feet. At the beginning of the design phase the project industry advisor suggested 4 feet by 3 feet for the outer dimensions. There were no dimensional requirements regarding the height and width of the box tubing members. It was decided that the width of the members would be allowed to vary between 2 and 5 inches while the height would be allowed to vary between 2 and 8 inches. These dimensions seemed appropriate based on the outer dimensions specified. Because the overheard guard of a fork lift truck is a grid, the number of rows and columns in the grid had to be decided. Given the outer dimensions of 4 feet by 3 feet and the allowable variation in the box tubing members, the 2-row, 3-column configuration shown in Figure 24 of the concept description was selected. It was decided that this configuration would not be altered unless it proved to be impossible to achieve the desired hydrogen capacity. If determined later that the size of the spaces in the grid does not provide adequate protection to the fork lift truck operator, additional protective structures could be inserted to reduce the size of the spaces. These additional protective structures would not contain hydrogen; they would simply break the tank up into more rows and columns so as to prevent objects from falling through the tank.

In order to determine which box tubing member dimensions would result in an acceptable volume, an Excel spreadsheet was created. In order to maximize the volume this spreadsheet assumed that an open, unreinforced cross-section would be used. What was just as important as an acceptable volume was the ability to withstand stress. The Excel spreadsheet was expanded to include equations for unreinforced rectangular vessels from Appendix 13 of the 2004 ASME Section VIII, Division 1, Boiler and Pressure Vessel code [33]. Appendix 13 of the code presents equations for "Vessels of non-circular cross-section" [33]. An excerpt from the equations and the Excel spreadsheet is shown in Appendix M. An example of unreinforced construction is shown in Figure 25. The pressure used in the theoretical calculation of the stress from the equations was the maximum allowable pressure stated in HGV5 (1.25 times the working pressure of 5,000 psi, which yielded a maximum allowable pressure of 6,250 psi) [13]. Due to the known difficulties of designing a rectangular vessel to withstand high internal pressure, the 6,250 psi maximum allowable pressure, rather than the 7,500 psi hydrostatic test pressure, was used throughout the initial design.

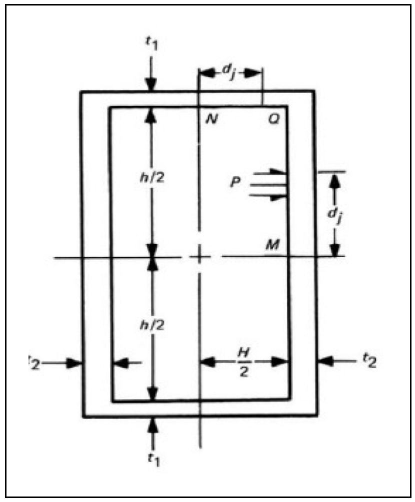


Figure 25 – Example of Unreinforced Construction

In the Excel spreadsheet for unreinforced construction the various theoretical stresses were calculated (membrane stress, bending stress, total stress, etc) and compared against the 34,000 psi yield strength of the material. The stresses were calculated for each possible combination of the box tubing member dimensions. Both 1/2 inch and 3/4 inch thick walls were considered. It was generally found that where the stress was acceptable the volume was completely inadequate, and vice versa. From these results it was obvious that unreinforced construction would not work.

A new version of the Excel spreadsheet was created using the equations for "stayed vessels of rectangular cross-section" [33]. See Appendix M for excerpts. A *stay* is a reinforcing structure placed across the middle of the open rectangular cross-section in order to hold the long sides together. This effectively divides the box tubing member into two separate cavities. An example of reinforced construction is shown in Figure 26. As before, in the Excel spreadsheet for reinforced construction, the various theoretical stresses were calculated and compared against the 34,000 psi yield strength of the material. The stresses were calculated for each possible combination of the box tubing member dimensions. Both 1/2 inch and 3/4 inch thick walls and stays were considered.

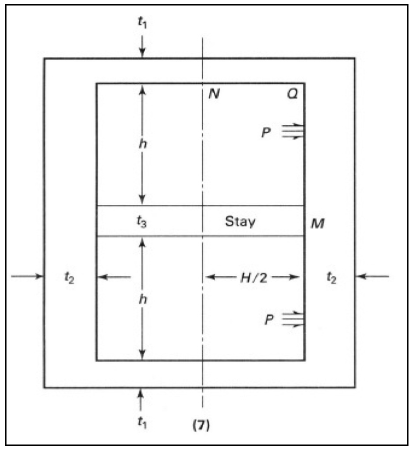


Figure 26 – Example of Reinforced Construction

As before, there was a general trend that where the stress was acceptable the volume was inadequate, and vice versa. Though no combination of dimensions with an acceptable volume produced perfectly acceptable component stresses, there were a few that showed promising results. The 4 inch by 7 inch cross-section with a 3/4 inch thick wall and stay had theoretical stresses below 44,000 psi that could probably be reduced by introducing a radius on the inside corners. To explore this, a model of a single box tubing member was created in Pro/Mechanica. Initial radii were selected and applied to the outside and inside corners of the model. The 6,250 psi pressure was applied to the inside surfaces, a displacement constraint was applied to one end, and an FEA was carried out. After some iteration in which the corner radii were adjusted, the FEA shown in Figure 27 was completed. The maximum stress was observed to be 29,840 psi, below the 34,000 psi yield strength of the material. The cross-section arrived at from the FEA iteration is shown in Figure 28. The outside corner radii were made to be 1/4 inch while the inside corner radii were made to be 7/8 inch. The outside dimensions remained at 4 inches by 7 inches and the thickness of the walls and stays remained at 3/4 inch.

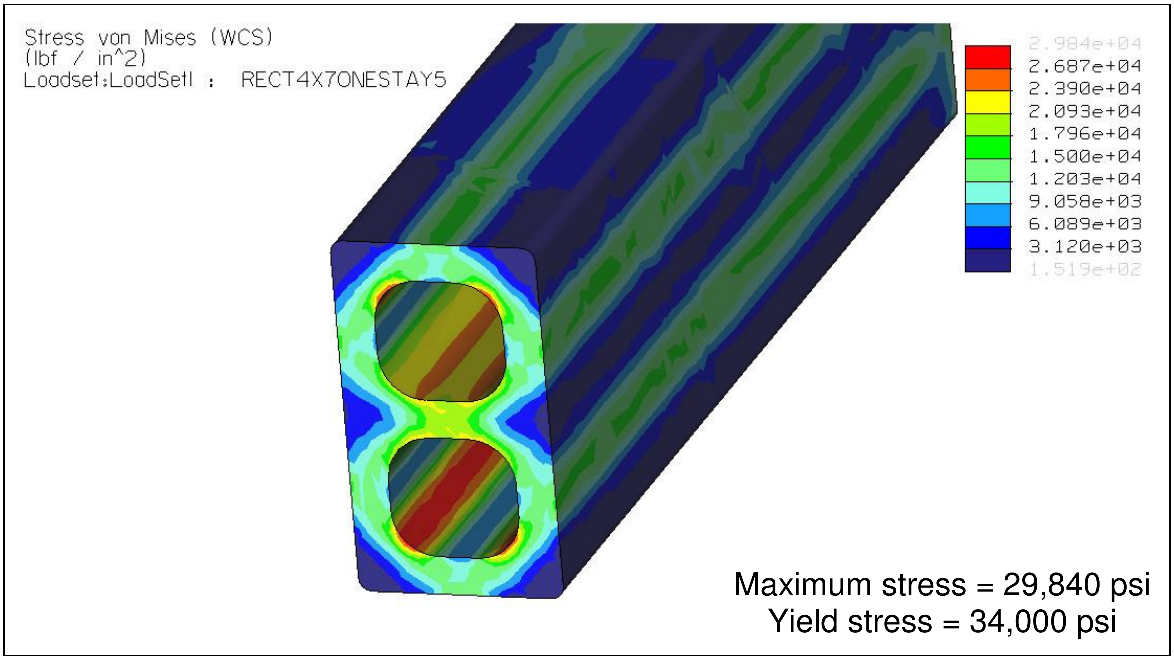


Figure 27 – 316L SS Straight Section Fringe Plot

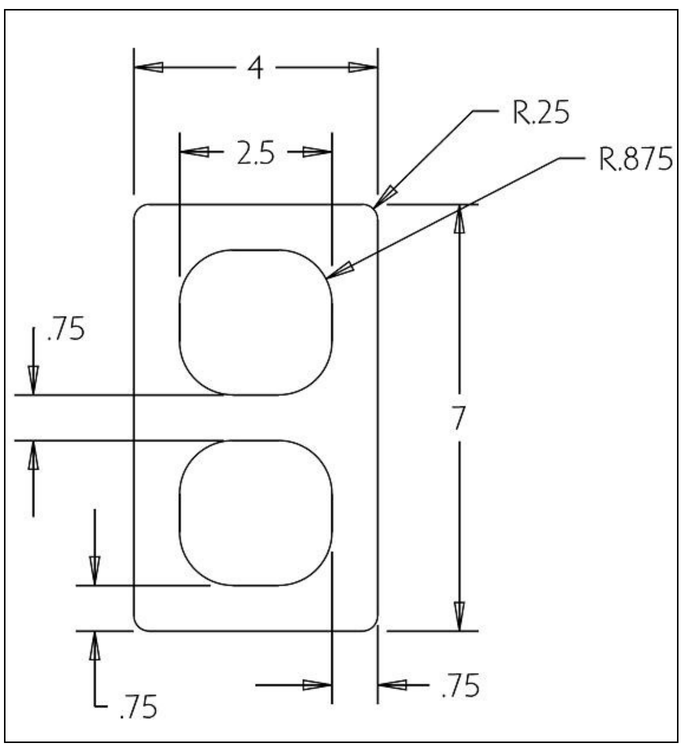


Figure 28 – 316L SS Cross-Section Design

The next step was to examine the stress that would result at a junction in the rectangular tank assembly. L-junctions, T-junctions, and X-junctions all had the potential to result in higher stresses than were observed in the single box tubing member. The details concerning how the junctions would be designed were not initially considered. It was assumed that perfect junctions could be produced. It was decided to first analyze a T-junction of box tubing members with the cross section shown in Figure 28. The model and FEA result is shown in Figure 29. The maximum stress observed was 52,830 psi, a 77% increase from the stress observed in the single box tubing member. More importantly, the maximum stress exceeded the yield strength of the material by 55%. Based on this, it was concluded that it would not be possible to design the rectangular tank assembly from 316L stainless steel material. To make the cross-section work the inside radius would have had to be increased to the point that there was simply a circular cavity inside a solid block of stainless steel. The project industry advisor suggested the design group consider a stronger material.

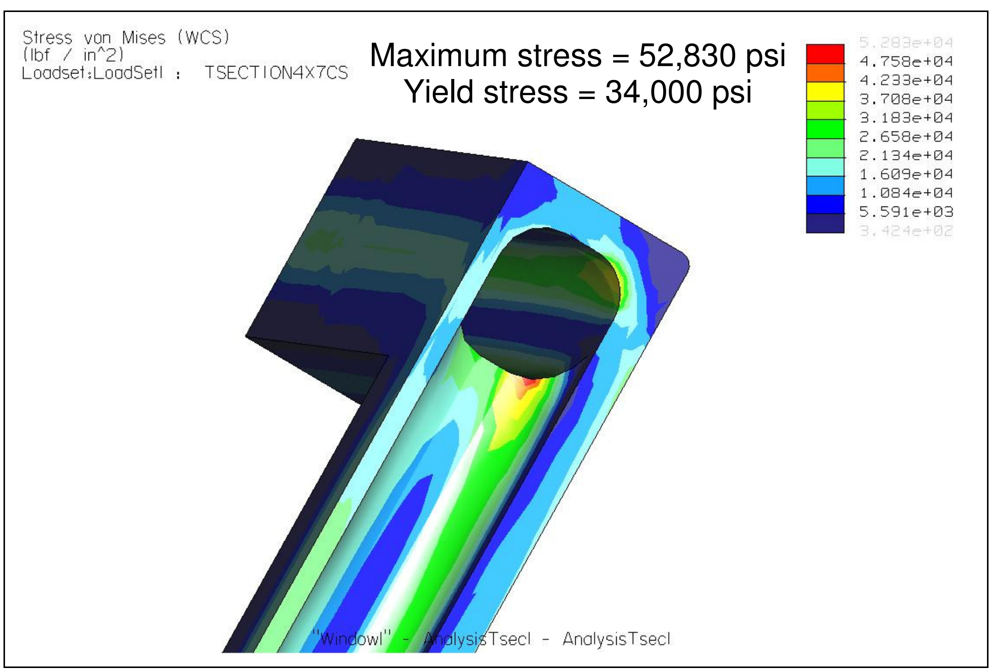


Figure 29 – 316L SS T-Junction Fringe Plot

6.2.3. CROSS-SECTION REVISION

Based on knowledge acquired during the fall semester research phase, and experience gathered during the cylindrical vessel design, 4130 Q&T steel was the most logical choice for a stronger box tubing material. As before, an appropriate yield strength for 4130 Q&T is 102,000 psi [18]. The fatigue limit (based on 10^6-10^9 cycles to failure) was found to be 71,000 psi [18]. Due to the known difficulties of designing a rectangular vessel to withstand high internal pressure, the 102,000 psi yield strength, rather than the 71,000 psi fatigue limit, was used as the stress limit throughout the design. It was accepted that the ability of the tank to withstand refueling cycles would have to be explored during qualification testing because it could not be properly addressed by the design group.

The procedure used to arrive at the 316L SS cross-section was repeated in order to arrive at a workable cross-section for the 4130 Q&T. The final FEA iteration and cross-section dimensions are shown in Figure 30 and Figure 31, respectively. The maximum stress observed in a straight section of box tubing was 67,370 psi, well below the yield strength of the material and still below the fatigue limit. From Figure 31 it was clear that changing materials from 316L SS to 4130 Q&T allowed the wall and stay thickness to be reduced to only 1/2 inch from 3/4 inch. It was also possible to reduce the inside corner radii to 1/2 inch from 7/8 inch. The outside dimensions required to achieve an acceptable volume were reduced somewhat from 4 inches by 7 inches to 3 inches by 6-1/2 inches.

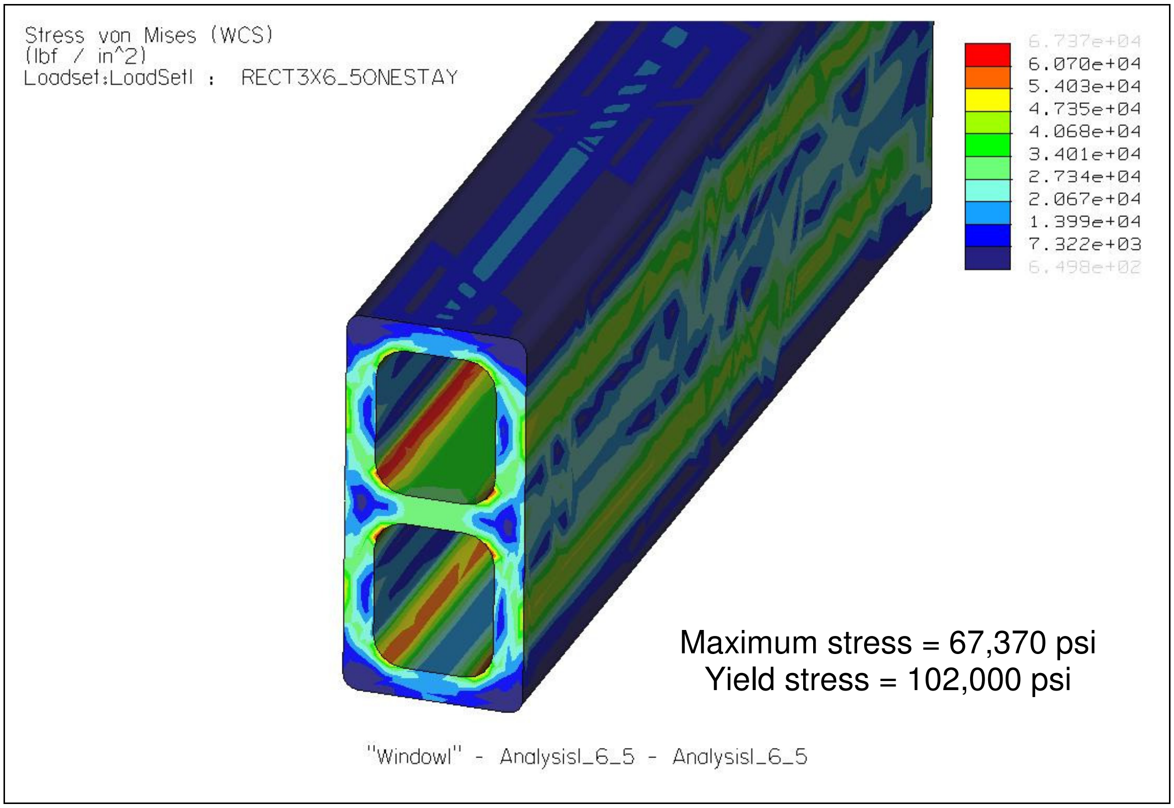


Figure 30 – 4130 Q&T Straight Section Fringe Plot

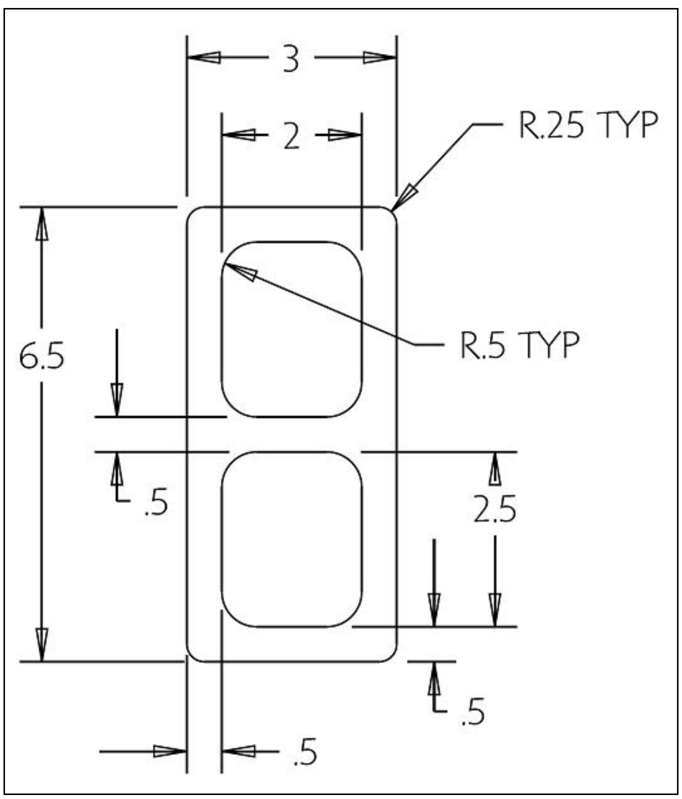
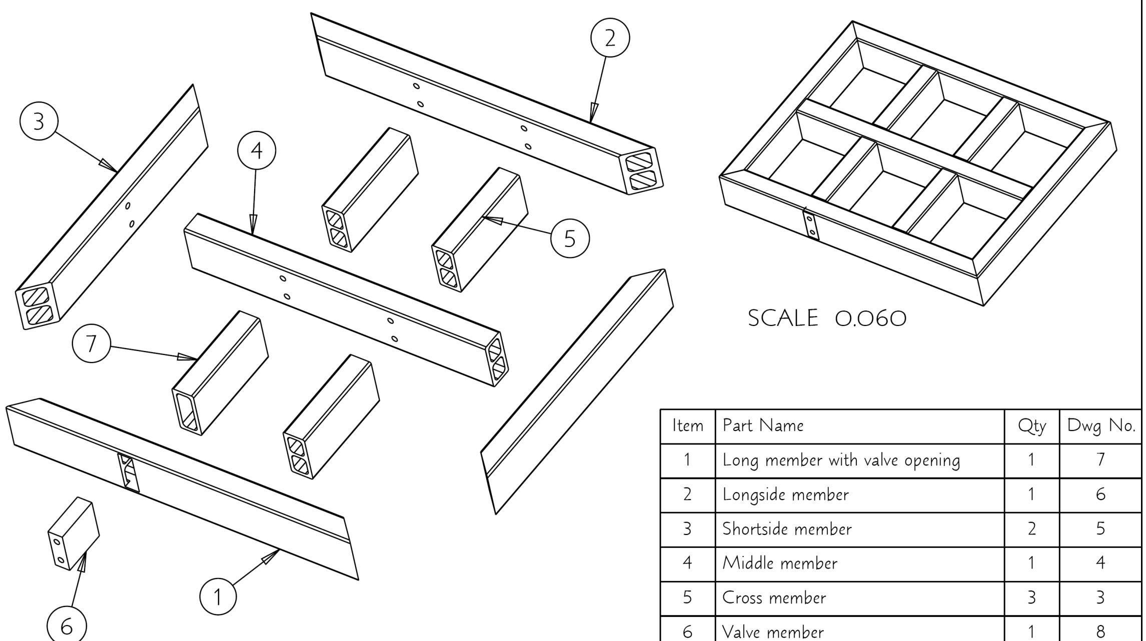


Figure 31 – 4130 Q&T Cross-Section Design

As before, the next step was to examine the stress that would result at a junction in the rectangular tank assembly. L-junctions, T-junctions, and X-junctions would all result in higher stresses than were observed in the single box tubing member. The details concerning how the junctions are designed were considered this time; the assembly design was carried out along with the FEA. It was still assumed that perfect junctions could be produced. Before the FEA results are discussed the assembly design will be presented. The finished assembly created by the design group is shown in Figure 32 below.



Item	Part Name	Qty	Dwg No.
1	Long member with valve opening	1	7
2	Longside member	1	6
3	Shortside member	2	5
4	Middle member	1	4
5	Cross member	3	3
6	Valve member	1	8
7	Cross member with valve opening	1	9

Figure 32 – 4130 Q&T Assembly Design

It is important to note in Figure 32 that L-junctions are to be created using members mitered at 45 degrees while T-junctions and X-junctions are to be created by butting one member against the side of the other. It was the small outside corner radius on the box tube that allowed the T-junctions and X-junctions to be assembled this way. It was thought that the small gap that would result along the edges of the box tube would not interfere with the welding of the assembly. To allow hydrogen to pass between a member and one butted against its side, 3/4 inch holes are to be drilled through the side of the member. One T-junction (between parts 1 and 7 in Figure 32) was specially designed to allow the insertion of a solid valve block (part 6). The valve block was designed with two ¾-14 NGT threaded holes that allow for connection of the tank to the hydrogen supply system on the fork lift truck. Detailed drawings for the parts shown in Figure 32 are included in Appendix G. The rectangular tank assembly was calculated to have a hydrogen capacity of about 41.3 liters and a total weight of approximately 719 pounds.

The FEA results for the various junctions shown in Figure 32 are summarized in Table 14. The maximum stresses for the single box tube member and each possible junction type (L-junction, T-junction, X-junction, and T-junction with valve block) are included. The highest stress was observed in the L-junction while the lowest was observed in the single member and T-junctions. The high stress observed in the L-junction was likely caused by the placement of the displacement constraint in the FEA model. Only one end of the L-junction was grounded resulting in an asymmetric loading that caused a relatively high stress concentration at the inside corner of the box tube. In a complete assembly, the stress observed in the L-junction would likely be lower than that observed in the model of the junction alone. As expected, the addition of the solid material of the valve block to the T-junction caused the stress to be reduced slightly (by about 2%). The design factors shown in Table 14 were calculated by dividing the yield strength of 102,000 psi by the maximum stress. From the design factors, it was clear that the performance of all of the junctions is acceptable. Table 14 also includes the corresponding figure number in Appendix J that illustrates the FEA result with a fringe plot of Von Mises stress. Included with each fringe plot is a strain energy convergence plot.

Table 14 – 4130 Q&T FEA Results

	Maximum Stress* [psi]	Design Factor	Appendix J Figure No.
Single Member	67370	1.51	5
L-Junction	82497	1.24	7
T-Junction	67260	1.52	9
X-Junction	71840	1.42	11
Valve T-Junction	66240	1.54	13

*The maximum stress as observed away from the displacement constraint

Despite the positive results shown in Table 14 the design group decided to reassess the single box tube member and all of the junctions using the HGV5 hydrostatic test pressure of 7,500 psi (1.5 times the 5,000 psi service pressure) [13]. This decision was made because the rectangular tank assembly would have to be able to pass the hydrostatic test required in HGV5. The results of the FEA reassessment are shown in Table 15. The T-junction with the addition of the valve block was not reassessed because it was known that it would result in a maximum stress less than the normal T-junction. All of the same stress relationships seen in Table 14 were observed in Table 15 (the L-junction had the highest stress, the single member and T-junction had the lowest stress, etc). What is important, however, is that none of the maximum stresses observed exceeded the 102,000 psi yield strength of the material. Though all of the design factors were reduced none of them dropped to a value less than or equal to one. From the results of Table 15, it was clear that the performance of all of the junctions is acceptable even at the HGV5 hydrostatic test pressure.

Table 15 – 4130 Q&T Hydrostatic FEA Results

	Maximum Stress* [psi]	Design Factor	Appendix J Figure No.
Single Member	80850	1.26	27
L-Junction	98990	1.03	29
T-Junction	79370	1.29	31
X-Junction	96050	1.06	33

*The maximum stress as observed away from the displacement constraint

6.2.4. CROSS-SECTION MANUFACTURABILITY

During the course of the rectangular tank design Louisiana Steel, a manufacturer of custom steel tubing, was contacted for consultation regarding the manufacturability of the designed cross-section. Joe Renick, a sales representative for the company, provided the consultation. Upon receiving the cross-section design, Joe informed the design group that he was not aware of any extruding technique that could extrude a box tube with a stay across the middle [25]. According to Joe, if the desired material was more like Aluminum it would be relatively easy to extrude, but a high strength steel like 4130 would be very difficult [25]. Joe's suggestion was to mimic the desired cross-section by welding together two square box tubes that Louisiana Steel would be able to manufacture. This suggested cross-section configuration is shown in Figure 33.

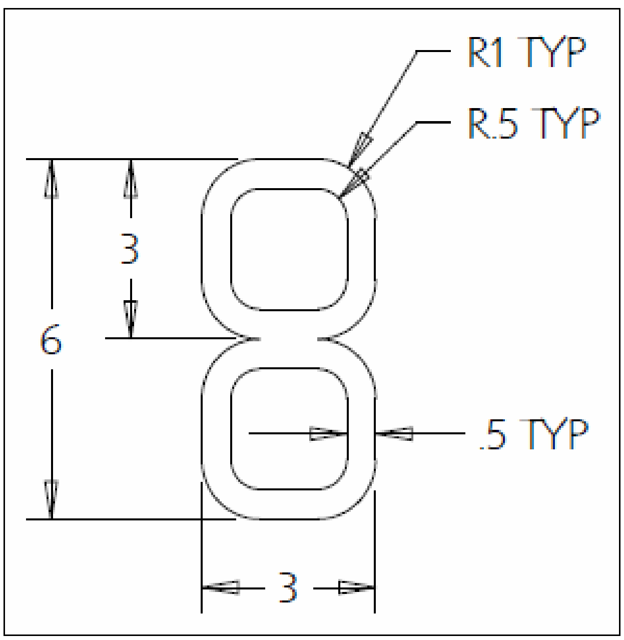


Figure 33 – 4130 Q&T Boxes v1 Cross-Section

It was the general opinion of the design group that if the configuration shown in Figure 33 could withstand the stress it would be an acceptable alternative to the original cross-section design. The other problem, however, was the junction design. Due to the large outside radius shown in Figure 33 it would not be possible to simply butt one box tube member against another in order to produce a T-junction or X-junction. An alternative solution had to be developed if the rectangular tank assembly were to be made from the configuration shown in Figure 33. Two potential solutions were developed; First, the idea of using a solid junction block was proposed, see Figure 34. Second, the idea of carefully mitering the box tubing members and fitting them together like a puzzle was proposed, see Figure 35.

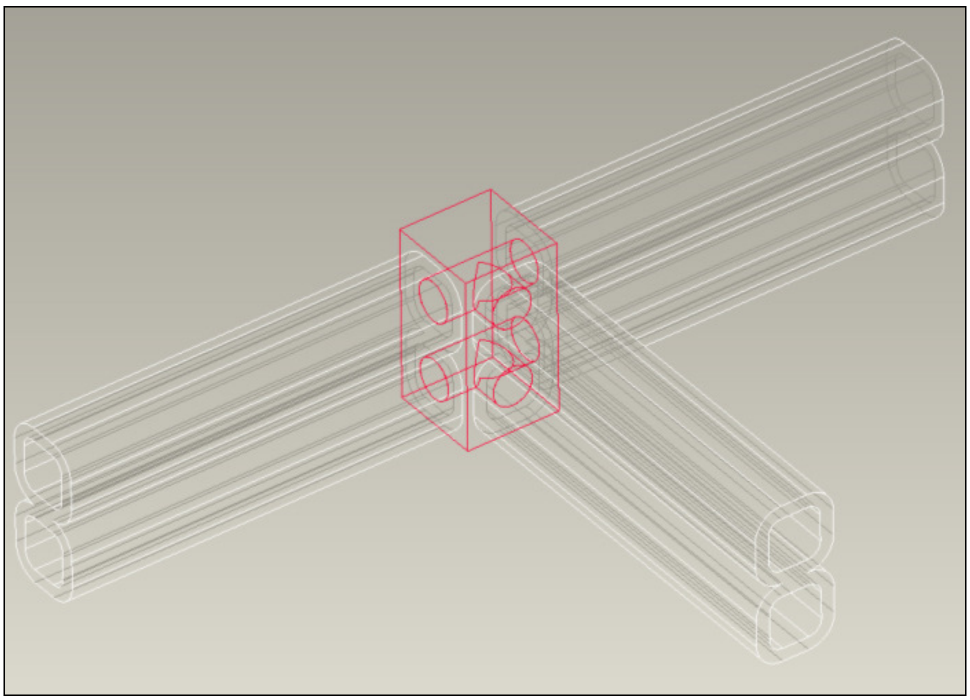


Figure 34 – Solid Junction Block Proposal

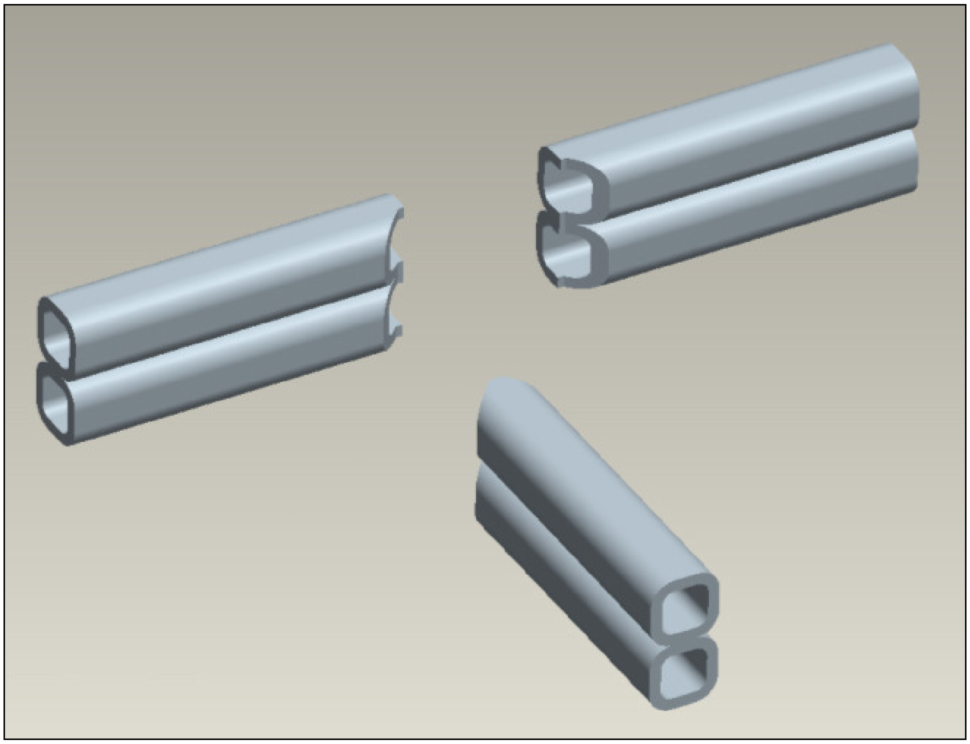


Figure 35 – Mitered Member Proposal

Both potential solutions were discussed with the project industry advisor and it was decided that the mitered proposal shown in Figure 35 would be more practical than the introduction of solid junction blocks. The solid junction blocks would consume a large amount of material, increasing the weight and decreasing the hydrogen capacity of the tank assembly. Additionally, the junction blocks would require machining during manufacturing whereas the mitering might be achieved in a laser cutting operation. Having identified a workable solution, the design group turned to carrying out FEA (at the HGV5 maximum allowable pressure of 6,250 psi) on a single box tube member and on the various junctions using the configuration shown in Figure 33. The FEA results for the 4130 Q&T welded box tubes are shown in Table 16. While the results for the single member were promising, both the T-junction and X-junction had maximum stresses higher than the yield strength of the material (resulting in design factors below one).

Table 16 – 4130 Q&T Boxes v1 FEA Results

	Maximum Stress* [psi]	Design Factor	Appendix J Figure No.
Single Member	51910	1.96	15
L-Junction	84500	1.21	17
T-Junction	**105000**	**0.97**	19
X-Junction	**123000**	**0.83**	21

*The maximum stress as observed away from the displacement constraint

During communication with Joe at Louisiana Steel it was found that there was an error in the dimensions shown in Figure 33. The inside radius of the box tube that Louisiana Steel is capable of manufacturing is only 1/4 inch as opposed to 1/2 inch [25]. Given the failure of the T-junction and the X-junction in Table 16 the design group was sure that both would fail with the change in radius as well. The single member and L-junction were reassessed via FEA with the 1/2 inch inside radius, the results are shown in Table 17. While the single member was still acceptable the L-junction now also had a maximum stress higher than the yield strength of the material. From the FEA results of Table 16 and Table 17 it was clear that the welded box tubes shown in Figure 33 would not be acceptable for the rectangular tank assembly. More specifically, it was learned that the key to the success of the cross-section in Figure 31 was the combination of inside and outside corner radii. Having a larger inside corner radius than outside corner radius maximizes the amount of material in the corner and allows it to withstand more stress.

Table 17 – 4130 Q&T Boxes v2 FEA Results

	Maximum Stress* [psi]	Design Factor	Appendix J Figure No.
Single Member	91200	1.12	23
L-Junction	**147000**	**0.69**	25

*The maximum stress as observed away from the displacement constraint

Given the inability of Louisiana Steel to manufacture the cross-section shown in Figure 31, and the inability of the cross-section shown in Figure 33 to withstand the stress, the design group needed to find another source of the box tubing for the rectangular tank. More specifically, the design group needed to find a manufacturer that could extrude box tubing that has a larger inside corner radius than outside corner radius. At this point the design group was concerned only with locating square box tubing that could be welded together to mimic the cross-section of Figure 31. Joe at Louisiana Steel referred the design group to Timken, a leading manufacturer of alloyed steels [25].

Jeff Hoerr of Timken indicated that such a box tube would be difficult to produce but suggested that they have done something similar in the past [36]. Jeff suggested that the design group consider using welded tubing versus seamless tubing and recommended the company EMJ Metals [36]. Given the quantity of welding that would be required to assemble the rectangular tank the design group decided it would probably not be a good idea to use box tubing that had already been welded. Jeff also suggested two other companies, Keystone Profiles and American Extruded Products (Amerex) [36]. Kathy Meteney at Keystone Profiles indicated that they could not produce the desired cross-section [37]. Efforts to establish communication with Amerex were unsuccessful. Based on the information received from Timken the design group concluded that it may indeed be possible to produce square tube with the desired corners, but it would be very difficult and require a sizeable investment. Additional research should be done to further assess the manufacturability of the original cross-section design of Figure 31 as well as the possibility of using welded tubing in place of seamless tubing.

6.2.5. OTHER CONSIDERATIONS

There are several other considerations regarding the rectangular tank design that have not yet been discussed including 1) welding of the box tubes together, 2) the effect of refueling cycling, 3) alternative valve and junction designs, and 4) the effect of dimensional variations.

To begin with welding; the design group generally had to assume that it would be possible to weld the box tube members together without significantly harming the mechanical properties of the steel. It also had to assume that the mechanical properties of the weld would match the properties of the steel. The extent to which these assumptions are valid is generally unknown. Joe at Louisiana steel made several attempts to contact someone regarding welding considerations; he was unsuccessful in his attempts [25]. It is important to mention that HGV5 only allows welded construction if it is carried out "in accordance with reasonable concepts of safety, substantiality and durability" [13]. It is possible to satisfy the specifications of HGV if the welded construction of the rectangular tank provides "at least equivalent performance" to the prescribed construction methods in the standard [13]. It also states that "additional tests may be required to evaluate potential failure modes… that are not specifically addressed" in the standard [13]. From what little it was able to determine about welding, the design group decided to simply include in the functional specification that the welding must be designed and carried out according to appropriate industry practice.

On the effect of refueling cycles; it was not possible for the design group to use the 71,000 psi fatigue limit as a stress limit in designing the rectangular tank due to the large stresses that were observed. Additionally, time did not allow the design group to carry out a fatigue life analysis in order to confirm that the 15,000 cycle requirement was met. It was generally decided by the design group that the tests specified in HGV would properly evaluate its ability to withstand cycling, as well as its overall performance, if they were carried out.

On alternative valve and junction designs; it was suggested by the project industry advisor that placing a ¾-14 NGT threaded hole directly in the side of one of the box tube members would be adequate for the connection of a valve. Based on the fact that the holes in the various members of Figure 32 are 3/4 inch, there would be probably be no stress problems associated with doing this. Such a design modification would be made in order to eliminate the complexity of the valve block design. Specifically, no substantial modification of the box tube members would be required if the valve block was not used. It should also be mentioned that the design of the junctions could be altered to employ the mitered idea presented in Figure 35. Mitered construction at all of the junctions would provide smoother seems for welding. Because no member would be butted against the side of another member all of the outside corner radii would flow smoothly into one another.

Finally, on dimensional variations in the box tubing; it was not possible to determine what dimensional variations would be expected in the box tubing due to the difficulties of designing the cross-section and of locating a supplier. It was seen with both the cylindrical vessels and the tubing coils that dimensional variations had a significant effect on the ability to withstand stress. It is unclear to what extent dimensional variations would have an influence on the ability of the rectangular tank to withstand stress.

An equally important concern is the extent to which dimensional variations would complicate the ability to fit and weld the box tubes together. This would be an especially significant concern if a mitered junction approach were used because dimensional variations could prevent the box tube members from locking tightly together. Not knowing so much about the effect of dimensional variations, the design group could only include in the functional specification that allowable dimensional variations must be determined based on their effect on the ability to withstand stress and on the ability to manufacture the assembly.

This discussion of other considerations concludes the evolution of the rectangular tank design. The next section will present the functional specifications for the design.

6.3. FUNCTIONAL SPECIFICATION

This section of the report presents the functional specifications for the rectangular tank design. A condensed version of the functional specification is presented in Appendix G. The functional specifications are the detailed requirements dictated by the finished design in order to meet the basic requirements presented previously in the report as well as the HGV5 design standard requirements for Type 1 compressed hydrogen tanks.

6.3.1. PHYSICAL REQUIREMENTS

This section includes the geometry, material, and physical features that characterize the design and are required to meet the basic requirements.

Geometry – The geometries of the raw material, the individual parts, and the finished assembly shall conform to the drawings contained within Appendix G of this report.

Capacity – One rectangular tank assembly will allow for the storage of approximately 41.3 liters of 5,000 psi compressed hydrogen.
Comments: *Because the 41.3 liter capacity achieved is 96% of the 43 liter basic requirement it was considered acceptable.*

Material – The raw material shall be 4130 Q&T steel. The steel shall have a yield strength no lower than 102,000 psi. The steel shall have an ultimate tensile strength of no more than 137,000 psi in accordance with ISO 11114-1. The steel should have a fatigue limit (based on 10^6-10^9 cycles to failure) as high as possible to reduce the risk of fatigue failure caused by refueling cycling. The steel must meet the *Material and Qualification Tests and Requirements* set forth in HGV5. The steel should be qualified for use with high pressure hydrogen according to the requirements of Article KD-10 of Section VIII, Division 1 of the ASME Boiler and Pressure Vessel Code.

Comments: *It was not possible for the design group to assess in detail the requirements set forth in Article KD-10, but it was known that Article KD-10 uses a robust approach, based on fracture mechanics, to qualify materials for use with high pressure hydrogen.*

Dimensional Variations – The allowable dimensional variations of the raw material, the individual parts, and the finished assembly shall be determined according to appropriate industry practice in order to;

- Minimize any negative effects on the ability to withstand stress
- Minimize any negative effects on manufacturability (includes abilities to fit pieces together as designed and to perform welding)
- Minimize any negative effects on the ability to mount the tank and integrate it with the hydrogen supply system on a fork lift truck

Comments: *Due to the difficulties of designing a workable cross-section it was not possible for the design group to consider the effect of dimensional variations on the stress or manufacturability.*

Termination – The rectangular tank shall be terminated as shown in the drawings of Appendix G using a solid valve block. The valve block shall provide two standard ¾-14 NGT threaded openings for connection to appropriate valves and the hydrogen supply system on the fork lift truck. The threads shall comply with *Threaded Openings* in HGV5.

External Surfaces – The external surfaces of the rectangular tank shall be protected by a coating of paint or powder coat according to appropriate industry practice. The protective coating shall meet the requirements for *External Surfaces* set forth in HGV5.

6.3.2. MANUFACTURING REQUIREMENTS

This section includes tests that must be carried out in order to qualify the design for subsequent manufacturing, as well as the manufacturing procedures and tests required to confirm a quality final product that meets the other requirements set forth in this functional specification.

Qualification Testing – Prior to shipment of a completed rectangular tank the design qualification tests listed below must be carried out, according to HGV5, with satisfactory results. Any change in the rectangular tank design may require some or all qualification tests to be repeated according to *Change of Design* in HGV5.

- *Ambient Cycling Test*
- *Extreme Temperature Cycling Test*
- *Hydrostatic Burst Test*
- *Bonfire Test*
- *Penetration Test*
- *Leak Before Break Test*
- *NDE Defect Size Determination*
- *Expected Service Performance Test*

The ability of the tank to act as an overhead guard must also be tested according to the typical industry practice for testing overhead guards. The tank must not depressurize during the overhead guard testing.

Comments: *Because the design group did not construct or test a prototype the qualification tests above are extremely important. Only these tests can determine the real-world robustness of the final design.*

Welding – The welding of the joints in the rectangular tank shall be designed, carried out, and inspected according to appropriate industry practice and according to the requirements set forth in *Alternative Construction or Materials* in HGV5.

Comments: *Due to the difficulties of designing a workable cross-section it was not possible for the design group to consider the welding design or implementation. The section of HGV5 referred to allows for welded construction if it* provides *"at least equivalent performance" to the prescribed construction methods.*

Production Unit and Batch Testing – Unit and batch testing must be carried out during the manufacturing process according to the applicable requirements set forth in *Production Tests and Examinations* and *Batch Tests* of HGV5.

Quality Assurance – In general, manufacturing must be carried out according to the section *Manufacture* in HGV5. Quality assurance practices must be established and operated to ensure all rectangular tanks will be manufactured according to the qualified design. Quality assurance practices must meet the requirements of the *Quality Assurance* section in HGV5.

Marking, Dispatch, and Records – Each rectangular tank must be marked and dispatched from the manufacturing facility per the requirements of *Marking and Dispatch* set forth in HGV5. The manufacturer shall follow the requirements for *Records of Manufacture* set forth in HGV5.

6.3.3. OPERATIONAL REQUIREMENTS

This section includes basic end-user operational requirements such as operating temperature, operating pressure, hydrogen purity, periodic inspection, and mounting effect on forklift dynamics.

Mounting – The rectangular tank shall be mounted on the fork lift truck so as to act as an overhead guard (to protect the operator from falling objects). Due to the 719 pound weight of the rectangular tank, its effect on the vehicle dynamics must be analyzed and found not to be dangerous. The mounting system must be designed according to appropriate industry practice so as to 1) not interfere with the normal operation of the tank, 2) allow it to function as an overhead guard, and 3) prevent the build-up of hydrogen gas should a leak occur in or around the tank.
Comments: *The mounting design will depend very much on the model of fork lift truck on which the rectangular tank is to be used.*

Service and Maximum Pressure – In accordance with HGV5, the service pressure of the rectangular tank is 5,000 psi and the service life shall be 10 years or 15,000 refueling cycles, whichever is reached first. The maximum pressure is not to exceed 6,250 psi immediately after filling, in accordance with HGV5.

Temperature – The hydrogen gas temperature and container temperature shall meet the requirements for *Settled Gas Temperatures* and *Container Temperatures* set forth in HGV5. In general, the rectangular tank shall not be placed in an environment with an ambient temperature below -25°C or above 45°C for an extended period of time.
Comments: *It is important to note that temperature was not a major consideration during the design process. It should also be noted that the temperatures at which HGV5 (15°C) and DOT (21°C) define the service pressure are different than the 0°C temperature at which the basic design requirement for service pressure is defined. It is recommended that the service pressure be redefined in the basic requirements to more closely match the definitions in HGV5 and DOT. The environmental requirement shown here is based on the recommendation that a fork lift truck should not be operated in such an environment for an extended period of time.*

Hydrogen Composition – The purity and composition of the hydrogen gas used in the rectangular tank shall meet the requirements set forth in *Gas Composition* of HGV5.

Inspection – Each rectangular tank shall be visually and ultrasonically inspected periodically while in service according to the *Periodic In-Service Inspection* requirements set forth in HGV5. Any rectangular tank involved in a collision, accident, fire, or other event that may have caused damage to the tank shall be handled according to the *Conditions Requiring Immediate Inspections* set forth in HGV5.
Comments: *Based on a lack of experience in the industry it is not possible for the design group to recommend an appropriate inspection frequency. The HGV5 minimum is every 36 months.*

7. CONCLUSION

The project can best be concluded by reviewing the accomplishments the design group has made in terms of the project objectives laid out in the project description at the beginning of this report. It is also important to present the recommendations of the design group for future work that might take place on the design proposals presented in this report.

7.1. ACCOMPLISHMENTS

Listed below are the project objectives presented in the project description. Following each objective is a review of the accomplishments of the design group with respect to that project objective.

- ***The project shall involve a thorough review of all applicable design standards and a determination of what can and cannot be achieved in each design proposal to comply with those standards***
  The design group made significant gains during the fall semester research phase in obtaining and reviewing design standards applicable to the project. Six design standards (DOT Part 178, NGV2-2000, HGV5, ISO 15869, SAE J2600, and ISO 11114-1) were thoroughly reviewed [12]. At the beginning of the spring semester design phase the design group chose to focus on meeting the requirements of the DOT Part 178 (for the traditional cylindrical vessel only) and HGV5 design standards. The functional specifications created for each design proposal detail how the designs did or did not fulfill the requirements of the design standards.

- ***The project shall involve the generation, evaluation, and selection of design concepts to be iterated into the final design proposals***
  The generation, evaluation, and selection of design concepts were all accomplished by the design group during the fall semester research

phase of the project [12]. A variety of design concepts were generated through brainstorming and the prior art offered by the patent application submitted by the project industry advisor. The generated concepts were evaluated using Pugh's and Pahl and Beitz methods. The results of the evaluation of alternatives were interpreted and the three design concepts to be pursued in detail during the design phase were selected based on those interpretations and input from the industry advisor.

- ***The project shall include, as necessary throughout its duration, consultation with industry experts, suppliers, and manufacturers***
  The design group communicated with more than a dozen industry experts, suppliers, or manufacturers throughout the course of the project. Valuable information was gathered regarding design standards used in the industry, manufacturing techniques, material availability, and the progress being made in the world of hydrogen applied to the material handling industry. The success of the project was made possible by the input of these industry experts and their support is acknowledged at the end of this report.

- ***The project shall include a history of all design iterations and evidence of the capacity of each design proposal to meet the design requirements***
  The evolution of each design proposal has been presented in detail in this report. The history of the design iterations was documented and evidence of the capacity to meet the requirements was presented. The evidence presented included theoretical calculations, numerical finite element analyses, and expert opinion from industry. Additionally, digital data of all of the work carried out by the design group is either included with this report or available through the various parties listed on the cover page.

- ***The project shall include the generation of a functional specification detailing the technical requirements for the final design proposals***
  Functional specifications were created for each design proposal. The functional specifications detailed the requirements dictated by the finished designs in order to meet the basic requirements as well as the requirements of the design standards. The functional specifications included engineering drawings for each of the design proposals.

- ***The project shall include acquisition of a quote for tooling and manufacture in volume for at least one of the design proposals***
  A quote for tooling and manufacture in volume of both versions of the traditional cylindrical vessel was acquired from Taylor-Wharton [16]. This alone was enough to satisfy the requirement above. Additionally, material quotes for the nested high pressure tubing coils were acquired from both Swagelok and Handy and Harman Tube Company [26].[28],[34].

7.2. RECOMMENDATIONS

Listed below are the recommendations of the design group for future work that might take place on the design proposals presented in this report;

- ***Revisit design steps that might require revision***
  Certain steps in the design could certainly have been approached differently. In the case of the cylindrical vessels, for example, the dimensional variations could have been considered up-front during the calculation of the appropriate wall thickness. This also goes for the tubing coils, the dimensional variations could have been considered up-front during tubing selection. In addition, for the tubing coils, the tubing selection was based on the availability of Swagelok fittings. If this had not been the catalyst for the tubing selection it is quite possible that different tubing sizes would have been selected that might have improved the

performance of the design. In the case of the rectangular tank, the effect of constraints on some of the FEA results could have been explored further. Analyses of larger sections of the tank might also have been carried out in order to more completely evaluate its real-world ability to withstand stress. These are a few immediate examples of some of the steps in the design that could have been approached differently.

- ***Investigate design aspects that could not be addressed***
  There were many aspects of the designs that could not be properly addressed by the design group, especially for the tubing coils and the rectangular tank. For the tubing coils the impact of the manufacturability could have been addressed in much more depth than it was. For example, if orbital welding J-shaped tubes together would be the most practical large-scale manufacturing technique, the design group could have called for the use of this technique and analyzed the potential impact on the ability to withstand stress. Additionally, it was decided that the ends of the tubes would likely be welded to manifolds in a large-scale manufacturing operation. The design group could have designed such manifolds rather than simply terminating the tubes with fittings and caps. The effect of welding the tubes to the manifolds could also have been considered. In the case of the rectangular tank, the impact of the welding on the design of the assembly in general could have been addressed. It is possible that the junction design might be optimized in order to facilitate the welding. Again, the effect of the welding on the ability to withstand stress could have been addressed. The impact of dimensional variations in the box tubing might have been cause for design changes as well. Finally, for all of the design proposals, theoretical approaches to considering the effect of refueling cycling could have been carried out.

- ***Conduct a more expansive, up-to-date review of the design standards to further sharpen the functional specifications***
  The federal Department of Transportation regulations applicable to the cylindrical vessel design are more broad than the few parts considered by the design group. It would be valuable to conduct a more expansive review of the DOT regulations as they relate to one another in order to ensure that all of the critical aspects of the regulations were addressed. Because the HGV5 standard reviewed by the design group was a draft version, it should be reviewed in the future whenever significant changes are made. In addition, HGV5 does refer to other standards documents that could be investigated. Finally, the ASME Article KD-10 and its referenced documents could be thoroughly reviewed.

- ***Construct prototypes and conduct testing***
  A key recommendation for future work is the construction of prototypes and the testing of those prototypes. As stated in the functional specifications, qualification tests are necessary in order to evaluate the real-world robustness of the final design proposals. This is especially true for the less conventional tubing coil and rectangular tank designs. Certainly some additional revision of the design proposals may be required before an investment could be made in prototypes, but prototypes should be the eventual goal for future work. Depending on the success of the prototypes, the next goal would be full-scale implementation as a product on a Raymond Corporation lift truck.

To conclude, a review of the accomplishments of the design group has shown that all of the project objectives have been achieved. Much was accomplished throughout the course of the project; the final result was the completion of three design proposals, complete with functional specifications, for hydrogen tanks that could be integrated into the design of Raymond Corporation fork lift trucks.

APPENDIX A: HYDROGEN PROPERTY CALCUALTION

VAN DER WAALS FUNCTION

```
function output = vdwaals(Pr,Vo,m,Te)
% Ideal Gas Law with Van Der Waals correction for hydrogen gas
% output = vdwaals(P,V,m,T)
% Input -1 for the variable to solve for.
% Inputting vdwaals(1,-1,1,273) will solve for V
% Pr = Absolute pressure in (psi)
% Vo = Volume in (L)
% ma = mass in (kg)
% Te = Temperature in (C)

P = Pr*6.894757293e3;
V = Vo/1000;
T = Te+273;

% Constants for Hydrogen Gas Only
a = 6093; %(m^6)*Pa/(kg^2)
b = 0.0132; %(m^3)/kg
R = 4124; %(m^3)*Pa/(kg*K)

if nargin < 4, error('Not enough inputs!'); end
if Pr(1) ~= -1 && Vo(1) ~= -1 && m(1) ~=-1 && Te(1) ~= -1
   error('You must input -1 in place of the unknown variable!')
end

if Pr(1) == -1
   output = m*R.*T./(V-b*m) - (a*m.^2)./V.^2;
   output = output/6.894757293e3;
elseif Vo(1) == -1
   c1 = P;
   c2 = -(P*b.*m + m*R.*T);
   c3 = a*m.^2;
   c4 = -a*b*m.^3;
   volume = zeros(1,length(m));
   for n = 1:length(m)
      Vx = @(x) c1(n)*x.^3 + c2(n)*x.^2 + c3(n)*x + c4(n);
      volume(n) = fzero(Vx,0.05);
   end
   output = volume*1000;
elseif m(1) == -1
   c1 = a*b./V.^2;
   c2 = -a./V;
```

```
    c3 = P*b + R*T;
    c4 = -P.*V;
    mass = zeros(1,length(V));
    for n = 1:length(V)
        mx = @(x) c1(n)*x.^3 + c2(n)*x.^2 + c3(n)*x + c4(n);
        mass(n) = fzero(mx,1);
    end
    output = mass;
elseif Te(1) == -1
    output = (P + a*m.^2./V.^2).*(V - b*m)./(m*R);
    output = output-273;
end
```

PLOTTING SCRIPT

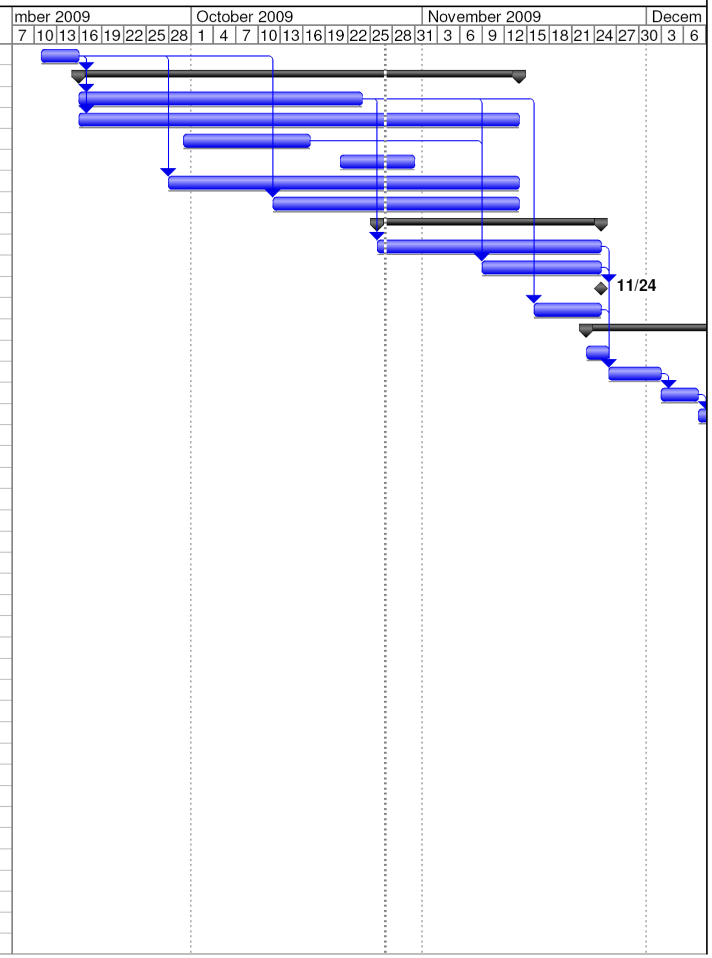
```
% Plot of volume required to hold 1 kg of H2 at 0 C from 15 - 10000 psi

P = linspace(15,10000,200); V = -1; m = 1; T = 0;
one = ones(1,length(P));
T = T*one;
m = m*one;
vary = vdwaals(P,V,m,T);
set(0,'defaultaxesfontsize',14)
plot(P,vary,'LineWidth',2);
axis([0 10000 0 150])
title('Volume versus Pressure of Tank containing 1 kg of H_2 at 0^oC')
xlabel('Pressure (psi)')
ylabel('Volume (L)')
grid on

% Plot of pressure required to hold 1 kg of H2 in a 43 liter tank from
% -28 C to 45 C

P = -1; V = 43; m = 1; T = linspace(-28,45,200);
one = ones(1,length(T));
V = V*one;
m = m*one;
vary = vdwaals(P,V,m,T);
figure(2)
plot(T,vary,'LineWidth',2);
axis tight
title('Effects of Temperature on H_2 Tank Pressure')
xlabel('Temperature (C)')
ylabel('Pressure (psi)')
grid on
```

APPENDIX B: PROJECT SCHEDULE

Project Schedule (1st Semester Projection)

MDP #8 Hydrogen Tank Design, Prelimary Project Schedule (Rev 2)

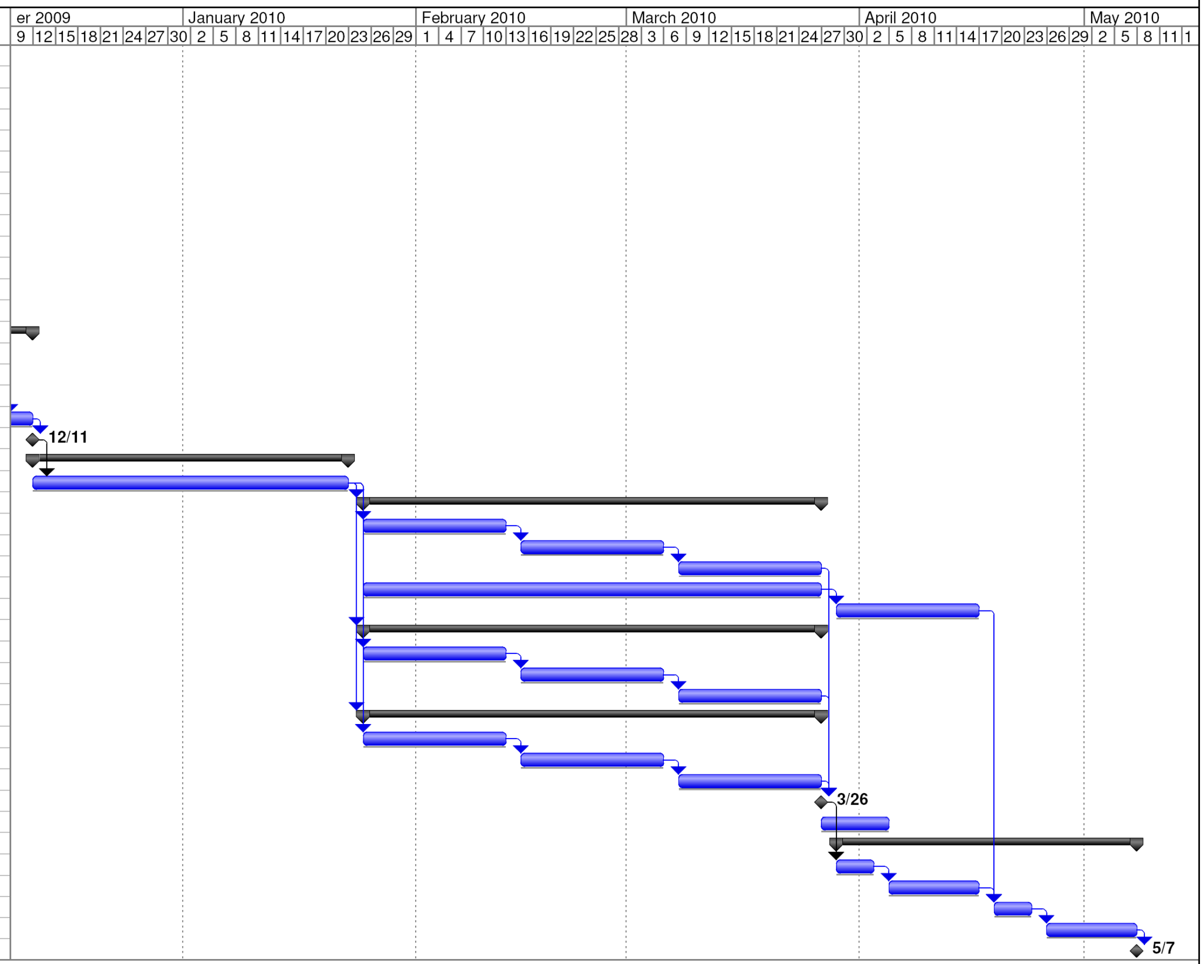
ID		Task Name	Duration	Start	Finish	Predecessors
1		Project Kick-Off	3 days?	Fri 9/11/09	Tue 9/15/09	
2		**Project Research**	**43 days?**	**Wed 9/16/09**	**Fri 11/13/09**	**1**
3		Industry Research	28 days?	Wed 9/16/09	Fri 10/23/09	1
4		Standards Review	43 days?	Wed 9/16/09	Fri 11/13/09	1
5		Requirements Matrix	13 days?	Wed 9/30/09	Fri 10/16/09	
6		Project Schedule/Budget	8 days?	Wed 10/21/09	Fri 10/30/09	
7		Industry Communication	35 days?	Mon 9/28/09	Fri 11/13/09	1
8		Supplier Communication	25 days?	Mon 10/12/09	Fri 11/13/09	1
9		**Design Preparation**	**22 days?**	**Mon 10/26/09**	**Tue 11/24/09**	
10		Napkin Design	22 days?	Mon 10/26/09	Tue 11/24/09	3
11		Functional Specifications	12 days?	Mon 11/9/09	Tue 11/24/09	3,5
12		Approval of Funcational Specs	0 days	Tue 11/24/09	Tue 11/24/09	11
13		Design Concept Selection	7 days?	Mon 11/16/09	Tue 11/24/09	3
14		**Report/Presentation**	**15 days?**	**Mon 11/23/09**	**Fri 12/11/09**	
15		Report/Presentation Outline	3 days?	Mon 11/23/09	Wed 11/25/09	
16		Report/Presentation Draft	5 days?	Thu 11/26/09	Wed 12/2/09	13,11,10
17		Report/Presentation Complete	3 days?	Thu 12/3/09	Mon 12/7/09	16
18		Presentation Rehearsal	4 days?	Tue 12/8/09	Fri 12/11/09	17
19		First Semester Presentation	0 days	Fri 12/11/09	Fri 12/11/09	18
20		**Between Semesters**	**31 days?**	**Sat 12/12/09**	**Fri 1/22/10**	
21		Between Semesters	31 days?	Sat 12/12/09	Fri 1/22/10	19
22		**Design Concept 1 Iteration**	**45 days?**	**Mon 1/25/10**	**Fri 3/26/10**	**21**
23		Iteration 1	15 days?	Mon 1/25/10	Fri 2/12/10	21
24		Iteration 2	15 days?	Mon 2/15/10	Fri 3/5/10	23
25		Iteration 3	15 days?	Mon 3/8/10	Fri 3/26/10	24
26		Manufacturer Communication	45 days?	Mon 1/25/10	Fri 3/26/10	
27		Concept 1 Manufacturing Quote	15 days?	Mon 3/29/10	Fri 4/16/10	26
28		**Design Concept 2 Iteration**	**45 days?**	**Mon 1/25/10**	**Fri 3/26/10**	**21**
29		Iteration 1	15 days?	Mon 1/25/10	Fri 2/12/10	21
30		Iteration 2	15 days?	Mon 2/15/10	Fri 3/5/10	29
31		Iteration 3	15 days?	Mon 3/8/10	Fri 3/26/10	30
32		**Design Concept 3 Iteration**	**45 days?**	**Mon 1/25/10**	**Fri 3/26/10**	**21**
33		Iteration 1	15 days?	Mon 1/25/10	Fri 2/12/10	21
34		Iteration 2	15 days?	Mon 2/15/10	Fri 3/5/10	33
35		Iteration 3	15 days?	Mon 3/8/10	Fri 3/26/10	34
36		End of Design Iterations	0 days	Fri 3/26/10	Fri 3/26/10	35,31,25
37		Spring Break	6 days?	Sat 3/27/10	Sun 4/4/10	
38		**Report/Presentation**	**30 days?**	**Mon 3/29/10**	**Fri 5/7/10**	
39		Report/Presentation Outline	5 days?	Mon 3/29/10	Fri 4/2/10	36
40		Report/Presentation Draft	10 days?	Mon 4/5/10	Fri 4/16/10	39
41		Report/Presentation Revision	5 days?	Mon 4/19/10	Fri 4/23/10	40,27
42		Presentation Rehearsal	10 days?	Mon 4/26/10	Fri 5/7/10	41
43		2nd Semester Presentation	0 days	Fri 5/7/10	Fri 5/7/10	42



Project Schedule (1st Semester Projection)

MDP #8 Hydrogen Tank Design, Prelimary Project Schedule (Rev 2)

ID	Task Name
1	Project Kick-Off
2	**Project Research**
3	Industry Research
4	Standards Review
5	Requirements Matrix
6	Project Schedule/Budget
7	Industry Communication
8	Supplier Communication
9	**Design Preparation**
10	Napkin Design
11	Functional Specifications
12	Approval of Funcational Specs
13	Design Concept Selection
14	**Report/Presentation**
15	Report/Presentation Outline
16	Report/Presentation Draft
17	Report/Presentation Complete
18	Presentation Rehearsal
19	First Semester Presentation
20	**Between Semesters**
21	Between Semesters
22	**Design Concept 1 Iteration**
23	Iteration 1
24	Iteration 2
25	Iteration 3
26	Manufacturer Communication
27	Concept 1 Manufacturing Quote
28	**Design Concept 2 Iteration**
29	Iteration 1
30	Iteration 2
31	Iteration 3
32	**Design Concept 3 Iteration**
33	Iteration 1
34	Iteration 2
35	Iteration 3
36	End of Design Iterations
37	Spring Break
38	**Report/Presentation**
39	Report/Presentation Outline
40	Report/Presentation Draft
41	Report/Presentation Revision
42	Presentation Rehearsal
43	2nd Semester Presentation



Project Schedule (2nd Semester Projection vs Actual)

										Spring Break					Final Week
Task Name	**Responsible**	**Week of**	**1-Feb**	**8-Feb**	**15-Feb**	**22-Feb**	**1-Mar**	**8-Mar**	**15-Mar**	**22-Mar**	**29-Mar**	**5-Apr**	**12-Apr**	**19-Apr**	**26-Apr**	**3-May**
Initial Sizing	**All**	Exp	75%	100%												
		Act	75%	100%												
Cycling Considerations	**Matt**	Exp	25%	50%	75%	100%										
		Act	0%	25%	25%	25%	50%	100%								
Model Iteration	**Jeff**	Exp		30%	60%	70%	80%	90%	100%							
		Act		50%	60%	70%	80%	100%								
Pressure FEA Test	**Jeff**	Exp		30%	60%	70%	80%	100%								
		Act		30%	60%	70%	100%									
DOT FEA Test	**Jeff**	Exp		30%	60%	70%	80%	100%								
		Act		30%	60%	100%										
Drawing Iteration	**Jeff**	Exp		30%	60%	70%	80%	90%	100%							
		Act		30%	60%	70%	80%	100%								
Manufacturer Identification	**Jeff/Matt**	Exp	30%	60%	100%											
		Act	50%	75%	100%											
Quote Acquisition	**Jeff/Matt**	Exp			10%	20%	40%	60%	70%	80%	90%	100%				
		Act		10%	20%	20%	40%	75%	100%							
Report Preparation	**Jeff**	Exp							25%	50%	75%	100%				
		Act							10%	25%	25%	50%	75%	100%		
Tube Selection	**Wui/Matt**	Exp	30%	60%	100%											
		Act	50%	70%	100%											
Bending and Joining	**Wui/Matt**	Exp		30%	60%	100%										
		Act		30%												
Coiled Tube Model Creation	**Wui**	Exp		30%	60%	75%	100%									
		Act		30%	60%	70%	85%	100%								
Pressure FEA Test	**Wui**	Exp			30%	60%	75%	100%								
		Act		15%	100%											
Drawings	**Wui**	Exp			33%	66%	100%									
		Act			33%	66%	75%	85%	100%							
Report Preparation	**Wui**	Exp							25%	50%	75%	100%				
		Act							0%	50%	100%					

Project Schedule (2nd Semester Projection vs Actual)

Task Name	Responsible	Week of	1-Feb	8-Feb	15-Feb	22-Feb	1-Mar	8-Mar	15-Mar	22-Mar	29-Mar (Spring Break)	5-Apr	12-Apr	19-Apr	26-Apr	3-May (Final Week)
Rectangular Model Creation	Anil	Exp	25%	50%	75%	100%										
		Act	0%	25%	50%	100%										
Pressure FEA Test	Anil	Exp	15%	30%	45%	60%	75%	100%								
		Act	0%	30%	60%	100%										
Model Drawings	Anil	Exp			33%	66%	100%									
		Act			0%	0%	50%	75%	100%							
Assembly Drawings	Anil	Exp					25%	50%	75%	100%						
		Act					50%	75%	100%							
Welding Specifications	Anil/Matt	Exp		20%	40%	60%	80%	100%								
		Act		0%	0%	0%	0%	25%	25%							
Report Preparation	Anil	Exp							25%	50%	75%	100%				
		Act							0%	0%	0%	50%	100%			
Report/Presentation Outline	Matt	Exp					30%	60%	100%							
		Act					0%	0%	50%	75%	75%	75%	75%	100%		
Report/Presentation Draft	Matt	Exp									25%	50%	100%			
		Act								10%	10%	10%	10%	75%	100%	
Report/Presentation Revision	Matt	Exp											25%	50%	100%	
		Act											0%	0%	100%	
Presentation Rehearsal	All	Exp													25%	100%
		Act													25%	100%

APPENDIX C: REQUIREMENTS MATRIX

Requirement	Derived Requirement	Derived-Derived Requirements	Test Plan
The tanks shall be able to hold at least 5,000 psi of compressed hydrogen without leaking	The tanks shall be able to provide for the storage of approximately 1 kg of compressed hydrogen on an individual fork lift truck		A: FEA Verification of ability to hold pressure B: Calculation of tank volume and comparison to volume of hydrogen
	The tanks shall be able to operate in a temperature range from -28 to 45 degrees Celsius		C: FEA Verification of stresses due to varying temperature of hydrogen
The tanks shall survive cycling of 3x per day for their life cycle	The tanks shall survive a minimum of 15,000 refueling cycles		D: Fracture mechanics or low cycle fatigue calculation using available fatigue data F: Compare the tank design to the standard
	The tanks shall be protected from embrittlement due to hydrogen exposure	The tanks shall be composed of austenitic stainless steel, such as 316L, or aluminum	
The tanks shall not have an adverse effect on the dynamics of the fork lift trucks	The tanks shall be composed of steel in order to compensate for the lost weight of the fork lift truck battery		
	If the tanks are to be mounted on the fixed portion of the fork lift truck mast the weight shall not exceed 500 pounds		E: Calculation of tank weight using tank volume and material density
The project shall develop 2-3 hydrogen tank designs for use in fork lift trucks	The project shall develop 3 hydrogen tank designs for use in fork lift trucks, at least one of which involves an awkward shape and at least one of which involves welding	One tank design shall involve coiled high pressure tubing as presented in the patent application and as specified by Raymond	

Requirement	Derived Requirement	Derived-Derived Requirements	Test Plan
		Corporation	
		One tank design shall involve a mast-mounted cylinder as presented in the patent application	
		One tank design shall involve a non-circular cross-section and employ welding	
The project shall identify potential suppliers of required materials			
The project shall seek quotes for tooling and manufacture in volume for 1 of the tank designs			
The project shall explore the following standards and what must be done in order to comply with them; DOT Section 178 Subsection 36-38, ANSI/CSA NGV2-2002, ISO 15869, ISO 11114-1, HGV5 Draft	Requirements of DOT Section 178 the tanks may comply with include water capacity, service pressure, longitudinal stress, wall thickness, material, welding, marking, openings	The tanks shall meet the requirement for water capacity and service pressure	F: Compare the tank design to the standard
		The tanks shall meet the requirement for longitudinal stress	G: Perform hand calculation and compare to FEA modeling, compare stress to material yield
		The tanks shall meet the requirement for wall thickness	G: Perform hand calculation and compare to FEA modeling, compare

Requirement	Derived Requirement	Derived-Derived Requirements	Test Plan
			stress to material yield
		The tanks shall meet the requirement for material	
		The tanks shall meet the requirement for welding	F: Compare the tank design to the standard
		The tanks shall meet the requirement for marking	
		The tanks shall meet the requirement for openings	F: Compare the tank design to the standard
	Requirements of ANSI/CSA NGV2-2002 the tanks may comply with include water capacity, service life, nominal service pressure, maximum pressure, maximum number of filling cycles, gas temperature range, container temperature range, external surfaces, chemical composition of steel, burst/service pressure ratio, openings, container end contour, brazing, welding, end closing, marking	The tanks shall meet the requirements for water capacity, service life, nominal service pressure, maximum number of filling cycles, gas temperature range, container temperature range, chemical composition of steel, openings, container end contour, brazing, welding, end closing	F: Compare the tank design to the standard
		The tanks shall meet the requirement for maximum pressure	A: FEA Verification of ability to hold pressure
		The tanks shall meet the requirements for external surfaces, burst/service pressure ratio, marking	
	Requirements of ISO 15869 the tanks may comply with include working pressure, maximum filling pressure, filling cycles, design temperature, external surfaces, material, wall thickness, construction,	The tanks shall meet the requirements for working pressure, filling cycles, design temperature, material, neck threads, construction	F: Compare the tank design to the standard

Requirement	Derived Requirement	Derived-Derived Requirements	Test Plan
	neck threads, marking		
		The tanks shall meet the requirement for maximum filling pressure and wall thickness	A: FEA Verification of ability to hold pressure
		The tanks shall meet the requirements for external surfaces and marking	
	Requirements of ISO 11114-1 the tanks may comply with include material with exposure to hydrogen		F: Compare the tank design to the standard
	Requirements of HGV5 Draft the tanks may comply with include service life, service pressure, maximum pressure, maximum number of filling cycles, temperature range, container temperature range, external surfaces, material, wall thickness, welding, threaded openings, container end contour, brazing, marking	The tanks shall meet the requirements for service life, service pressure, maximum number of filling cycles, temperature range, container temperature range, material, welding, threaded openings, container end contour, and brazing	F: Compare the tank design to the standard
		The tanks shall meet the requirement for maximum pressure and wall thickness	A: FEA Verification of ability to hold pressure
		The tanks shall meet the requirements for external surfaces and marking	

APPENDIX D: PROJECT BUDGET

MD 8 Total Material Budget (Fall)

Item #	Item Name	Cost / unit	Quantity	Total Cost
	Standards			
1	SAE (J2579) - $48.8 for members	$61.00	1	$61.00
2	ANSI/CSA NGV2-2000 (purchased)	$53.00	1	$53.00
3	ANSI NGV2-2007	$455.00	1	$455.00
4	ISO/TS 15689:2009 (purchased)	$140.00	1	$140.00
5	ISO 11114-1 (found online)	$0.00	1	$0.00
6	DOT sec178 (acquired from IA)	$0.00	1	$0.00
7	ASME KD-10	$555.00	1	$555.00
8	SAE J2600 (acquired from IA)	$0.00	1	$0.00
	Design Software			
8	Pro-E license (school owns)	$5,000.00	1	$5,000.00
9	ANSYS license (school owns)	$9,000.00	1	$9,000.00
			Total Materials	**$15,264.00**

Total Project Costs (fall)	$67,464.00

= Fall Labor plus total materials

MDP 8 Actual Material Budget (Fall)

Item #	Item Name	Cost / unit	Quantity	Total Cost
	Standards			
1	SAE (J2579) - $48.8 for members	$61.00	1	$61.00
2	ANSI/CSA NGV2-2000 (purchased)	$53.00	1	$53.00
3	ANSI NGV2-2007	$455.00	1	$455.00
4	ISO/TS 15689:2009 (purchased)	$140.00	1	$140.00
5	ISO 11114-1 (found online)	$0.00	1	$0.00
6	DOT Section 178 (acquired from IA)	$0.00	1	$0.00
7	ASME KD-10	$555.00	1	$555.00
8	SAE J2600 (acquired from IA)	$0.00	1	$0.00
	Design Software			
8	Pro-E license (school owns)	$5,000.00	0	$0.00
9	ANSYS license (school owns)	$9,000.00	0	$0.00
			Total	**$1,264.00**

MDP 8 Estimated labor (Fall)

	Name	Task Name	Estimated Hours
1	Jeffrey Chen	Industry research about Raymond corporation	10
2		Research on hydrogen, pressure vessels and hydrogen embrittlement	35
3		Review NGV2-2000, ISO 15689 and EIGA doc 100/03	35
4		Experiment with Norris high pressure cylinder	8
5		Experiment with bent, rectangular and straight high pressure tubing	25
6		Initial design concept selection	10
		Total Hours	123
		Total Labor Cost	**$12,300.00**
1	Wui On Wong	Industry research about Raymond corporation	10
2		Research on hydrogen, pressure vessels and hydrogen embrittlement	40
3		Research on high pressure tubing	10
4		Review NGV2-2000, ISO 15689, ISO 11114-1 and DOT 178.36-178.37	45
5		Research on functional specification	20
6		Initial design concept selection	10
		Total Hours	135
		Total Labor Cost	**$13,500.00**
1	Anil Kumar Chebrolu	Industry research about Raymond corporation	10
2		Research on hydrogen, pressure vessels and hydrogen embrittlement	35
3		Review NGV2-2000, ISO 15689, ISO 11114-1 and DOT 178.36-178.37	45
4		Identifying and compiling relevant information from standards	8
5		Project budget	4
6		Research on functional specification	15
7		Initial design concept selection	10
		Total Hours	127
		Total Labor Cost	**$12,700.00**
1	Matthew Grenier	Industry research about Raymond corporation	10
2		Research on hydrogen, pressure vessels and hydrogen embrittlement	30
3		Review EIGA doc 100/03, NGV2-2000 and ISO 15689	25
4		Identifying and compiling relevant information from standards	8
5		Composing requirement matrix	5
6		Project shedule	4
7		Industry and supplier communication	45
8		Initial design concept selection	10
		Total Hours	137
		Total Labor Cost	**$13,700.00**
		Total Project Labor Costs	**$52,200.00**

MD 8 Total Material Budget (Spring)

Item #	Item Name	Cost / unit	Quantity	Total Cost
1	Report printing	$100.00	1	$100.00
			Total Materials	**$100.00**

Total Project Costs (Spring)	$41,600.00

= Spring Labor plus total materials

MD 8 Actual Material Budget (Spring)

Item #	Item Name	Cost / unit	Quantity	Total Cost
1	Report Printing	$100.00	1	$100.00
			Total	**$100.00**

MDP 8 Estimated labor (Spring)

	Name	Task Name	Estimated Hours
1	**Jeffrey Chen**	Model Iteration	20.00
2		Pressure FEA Test	15.00
3		DOT FEA Test	25.00
4		Drawing Iteration	15.00
5		Report Preparation	25.00
		Total Hours	**100.00**
		Total Labor Cost	**$10,000.00**
1	**Anil Kumar Chebrolu**	Retangular Model Creation	30.00
2		Pressure FEA Test	20.00
3		Model Drawings	15.00
4		Assembly Drawings	10.00
5		Report Preparation	25.00
		Total Hours	**100.00**
		Total Labor Cost	**$10,000.00**
1	**Matthew Grenier**	Fatigue Life Calculator	10.00
2		Manufacturer Identification	20.00
3		Quote Acquisition	5.00
4		Welding Specifications	25.00
5		Report & Presentation	50.00
		Total Hours	**110.00**
		Total Labor Cost	**$11,000.00**
1	**Wui On Wong**	Tube Selection	10.00
2		Bending & Joining	5.00
3		Coiled Tube Model Creation	15.00
4		Pressure FEA Test	30.00
5		Drawings	20.00
6		Report Preparation	25.00
		Total Hours	**105.00**
		Total Labor Cost	**$10,500.00**
		Total Project Labor Costs	**$41,500.00**

APPENDIX E: FUNCTIONAL SPECIFICATION (CYLINDRICAL VESSEL)

FUNCTIONAL SPECIFICATION

Cylindrical Pressure Vessels

This functional specification includes the detailed requirements dictated by the finished cylindrical vessel designs in order to meet the basic design requirements as well as the DOT Part 178 and HGV5 design standard requirements. Unless specifically indicated, the requirements presented in this functional specification apply to both the 316L SS and 4130 Q&T versions of the cylinder. Note, however, that requirements of DOT Part 178.36 (for specification 3A cylinders) apply *only* to the 316L SS version of the cylinder. Requirements of DOT Part 178.37 (for specification 3AA cylinders) apply *only* to the 4130 Q&T version of the cylinder.

PHYSICAL REQUIREMENTS

Geometry, material, and physical features.

Geometry – The geometries of the cylinders shall conform to the drawings included with this specification. These geometries have been designed to meet the *Wall thickness* requirements of DOT Part 178.36 and DOT Part 178.37.

316L SS Cylinder Capacity – One 316L SS cylinder will allow for the storage of approximately 4.75 liters of 5,000 psi compressed hydrogen. Eight 316L SS cylinders shall be used on a single fork lift to allow for the combined storage of approximately 38 liters of compressed hydrogen.

4130 Q&T Cylinder Capacity – One 4130 Q&T cylinder will allow for the storage of approximately 6.52 liters of 5,000 psi compressed hydrogen. Eight 4130 Q&T cylinders shall be used on a single fork lift to allow for the combined storage of approximately 52.1 liters of compressed hydrogen.

316L SS Cylinder Material – The material of the raw tubing shall be 316L stainless steel to reduce the risk of hydrogen-induced embrittlement over time. The 316L stainless steel shall have a yield strength no lower than 36,000 psi. The 316L SS shall have a fatigue limit (based on 10^6-10^9 cycles to failure) above the specified yield strength to reduce the risk of fatigue failure caused by refueling cycling. The 316L stainless steel must meet the *Material Qualification Tests and Requirements* set forth in HGV5 and the *Steel* requirements set forth in DOT Part 178.36.

4130 Q&T Cylinder Material – The material of the raw tubing shall be 4130 Q&T steel. The steel shall have a yield strength no lower than 102,000 psi. The steel shall have an ultimate tensile strength of no more than 137,000 psi in accordance with ISO 11114-1. The steel should have a fatigue limit (based on 10^6-10^9 cycles to failure) of not less than 71,000 psi to reduce the risk of fatigue failure caused by refueling cycling. The steel must meet the *Material and Qualification Tests and Requirements* set forth in HGV5. The steel must meet the *Authorized steel* requirements in DOT Part 178.37. The steel should be qualified for use with high pressure hydrogen according to the requirements of Article KD-10 of Section VIII, Division 1 of the ASME Boiler and Pressure Vessel Code.

Tolerances – The raw tubing that will be spun into the cylinders shall meet the following requirements for maximum allowable dimensional variations;

- Maximum of 10% variation in the wall thickness (from the nominal) as measured at various points around the circumference of the tube
- For the 4130 Q&T cylinder;
  Maximum of +/-.025” deviation in the diameters per ASTM A519
- For the 316L SS cylinder;
  Maximum of +/-0.015” deviation in the diameters per ASTM A269 or ASTM A511

FUNCTIONAL SPECIFICATION

Cylindrical Pressure Vessels

External Surfaces – The external surfaces of the 4130 Q&T cylinder shall be protected by a coating of paint or powder coat according to appropriate industry practice. The protective coating on the 4130 Q&T cylinder shall meet the requirements for *External Surfaces* set forth in HGV5. The external surfaces of the 316L SS cylinder shall meet the requirements for *External Surfaces* set forth in HGV5.

Termination – The cylinders shall have two standard ¾-14 NGT threaded openings per the drawings in Appendix E for connection to appropriate valves and the hydrogen supply system on the fork lift truck. The threads shall comply with *Threaded Openings* in HGV5 and *Openings in cylinders* in DOT Part 178.36 and DOT Part 178.37.

MANUFACTURING REQUIREMENTS

Qualification tests, manufacturing procedures and tests.

HGV5 Qualification Testing – Prior to shipment of a completed cylinder the following design qualification tests must be carried out, according to HGV5, with satisfactory results; *Ambient Cycling Test, Extreme Temperature Cycling Test, Hydrostatic Burst Test, Bonfire Test, Penetration Test, Leak Before Break Test, NDE Defect Size Determination, Expected Service Performance Test.* Any change in the cylinder design may require some or all tests to be repeated according to *Change of Design* in HGV5.

Production Unit and Batch Testing – Unit and batch testing must be carried out during the manufacturing process according to the applicable requirements set forth in *Production Tests and Examinations* and *Batch Tests* of HGV5.

DOT Unit Testing – Prior to shipment of any completed cylinder the following tests must be carried out, according to DOT Part 178.36 and DOT Part 178.37, with satisfactory results; *Hydrostatic Test, Flattening Test, Physical Test, Leakage Test.* These tests must be carried out in accordance with *Inspections and analyses* requirements of DOT Part 178.35 *General requirements for specification cylinders*.

Quality Assurance – In general, manufacturing must be carried out according to the sections *Manufacture* in HGV5, DOT Part 178.36, and DOT Part 178.37. Quality assurance practices must be established and operated to ensure all cylinders will be manufactured according to the qualified design. Quality assurance practices must meet the requirements of the *Quality Assurance* section in HGV5. The rules for *Identification of Material, Heat treatment,* and *Rejected cylinders* in DOT Part 178.36 and DOT Part 178.37 must also be followed.

Marking, Dispatch, and Records – Each cylinder must be marked per the requirements for *Markings* in DOT Part 178.35. Each cylinder must be marked and dispatched from the manufacturing facility per the requirements of *Marking and Dispatch* set forth in HGV5. The manufacturer shall follow the requirements for *Records of Manufacture* set forth in HGV5.

FUNCTIONAL SPECIFICATION

Cylindrical Pressure Vessels

OPERATIONAL REQUIREMENTS

Operating temperature, pressure, hydrogen purity, inspection, mounting.

Mounting – Four cylinders shall be mounted in a 2x2 matrix on either side of the fixed portion of the fork lift truck mast. The mounting system must be designed according to appropriate industry practice so as to 1) not interfere with the normal operation of the cylinders, 2) protect the cylinders from accidental damage, and 3) prevent the build-up of hydrogen gas should a leak occur in or around the cylinders.

Service and Maximum Pressure – In accordance with HGV5, the service pressure of the cylinders is 5,000 psi and the service life shall be 10 years or 15,000 refueling cycles, whichever is reached first. The maximum pressure is not to exceed 6,250 psi immediately after filling, in accordance with HGV5.

Temperature – The hydrogen gas temperature and container temperature shall meet the requirements for *Settled Gas Temperatures* and *Container Temperatures* set forth in HGV5. In general, the cylinders shall not be placed in an environment with an ambient temperature below -25°C or above 45°C for an extended period of time.

Hydrogen Composition – The purity and composition of the hydrogen gas used in the cylinders shall meet the requirements set forth in *Gas Composition* of HGV5.

Inspection – Each cylinder shall be visually and ultrasonically inspected periodically while in service according to the *Periodic In-Service Inspection* requirements set forth in HGV5. Cylinders shall undergo periodic tests according to the *Periodic qualification and marking of cylinders* requirements set forth in DOT Part 173.34. Any cylinder involved in a collision, fire, or other event that may have caused damage to the cylinder shall be handled according to the *Conditions Requiring Immediate Inspections* set forth in HGV5. Any cylinder which has been pressurized beyond the maximum allowable pressure shall be handled according to the *Over-Pressurization* requirements set forth in HGV5.

CONTENT APPROVAL

Content approval by the designers and client.

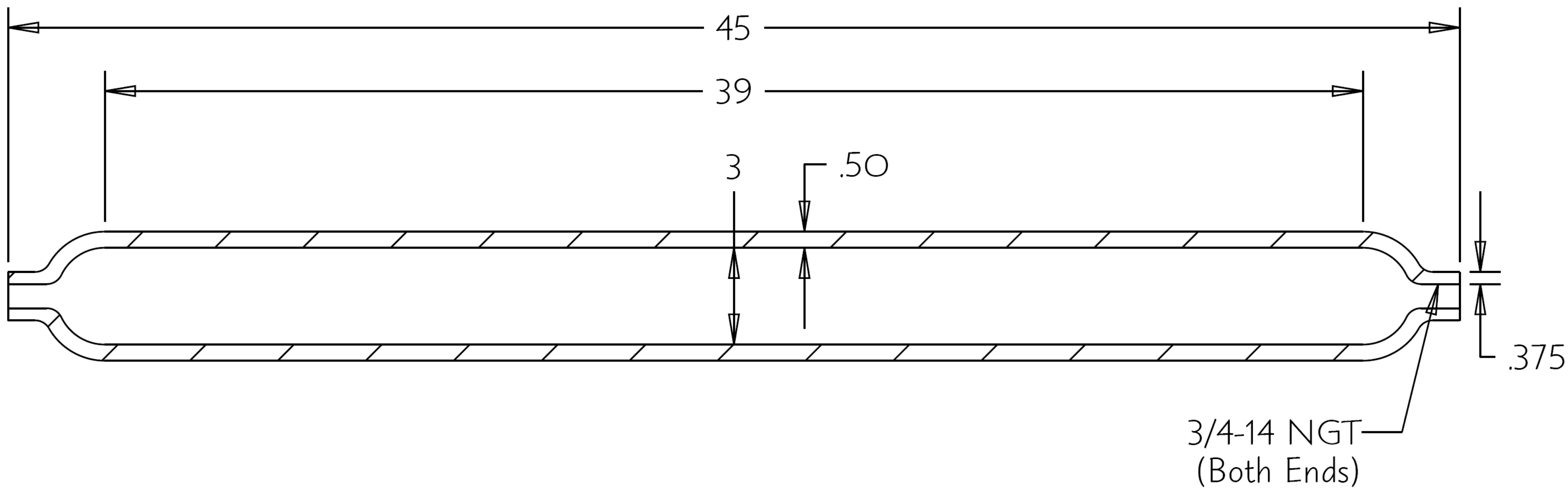
The content of this functional specification and the work carried out
on this design proposal is herby approved by:

______________________________ ______________________________

Matthew Grenier, MDP #8 Jeffrey Chen, MDP #8

____

Bryce Gregory, Raymond Corporation



SPECIFICATION: DOT 3A

1. Principal Elements
 - DOT Service Pressure: 5000 psi
 - Minimum Test Pressure: 8333 psi

2. Material:
 - AISI 316L Stainless Steel

PART AND DRAWING NOTES

All dimensions shown in inches

DRAWN Jeffrey Chen

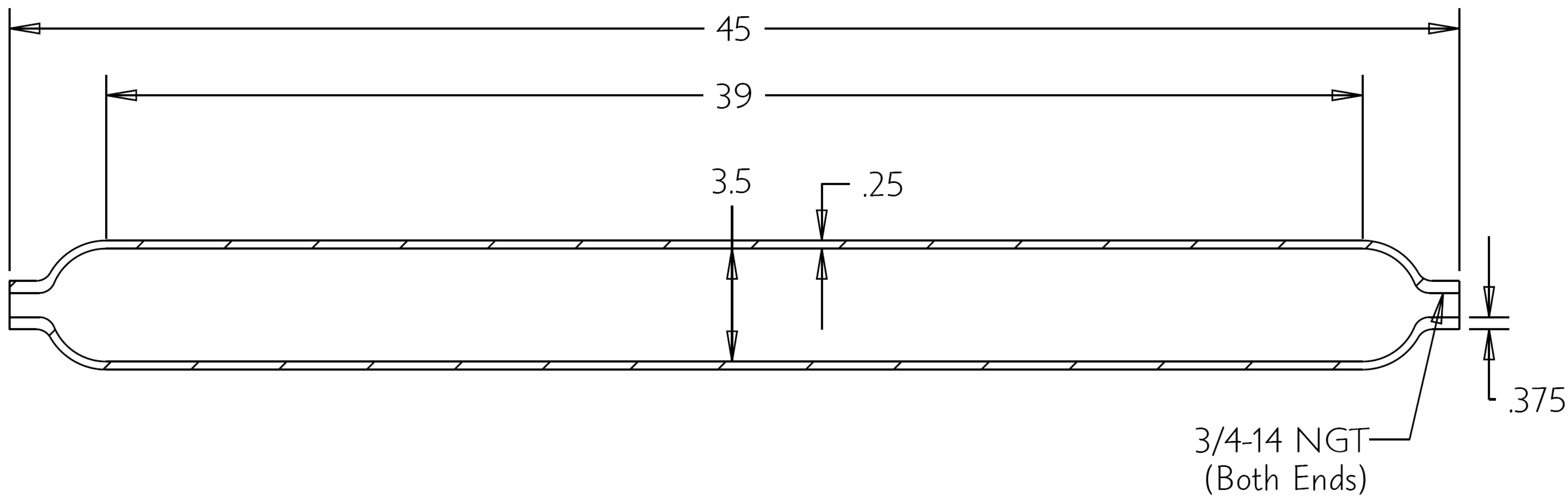
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

TITLE

Cylindrical Vessel

SIZE	FSCM NO.	DWG NO.	REV
A	N/A	CV01	6

DATE	SCALE	SHEET	OF
4/29/10	0.2 in = 1 in	1	1



SPECIFICATION: DOT 3AA

1. Principal Elements
   - DOT Service Pressure: 5000 psi
   - Minimum Test Pressure: 8333 psi

2. Material:
   - AISI 4130 Q&T Steel

3. DOT Wall Stress Calculations

   S = Wall Stress, psi
   P = Min. Test Pressure, psi
   D = Outer Diameter, in
   d = Inner Diameter, in

   $S = [P(1.3D^2+0.4d^2)]/(D^2-d^2)$

   $S = [8333(1.3(4)^2+0.4(3.5)^2)]/(4^2-3.5^2)$

   $S = 57109$ psi

   Required Minimum Tensile:
   = 57109 psi / 0.67 = 85237 psi

PART AND DRAWING NOTES	Binghamton University Watson School of Engineering Major Design Project #8, Hydrogen Tank Design
	TITLE Cylindrical Vessel
All dimensions shown in inches	SIZE A · FSCM NO. N/A · DWG NO. CV03 · REV 3
DRAWN Jeffrey Chen	DATE 3/08/10 · SCALE 0.2 in = 1 in · SHEET 1 OF 1

APPENDIX F: FUNCTIONAL SPECIFICATION (TUBING COILS)

FUNCTIONAL SPECIFICATION

Nested High Pressure Tubing Coils

This functional specification includes the detailed requirements dictated by the finished tubing coil tank design in order to meet the basic design requirements as well as the HGV5 design standard requirements for Type 1 compressed hydrogen tanks.

PHYSICAL REQUIREMENTS

Geometry, material, and physical features.

Geometry – The geometries of the raw tubing, the individual tubing coils, and the finished assembly shall conform to the drawings contained within Appendix F of this report.

Capacity – One assembly of tubing coils will allow for the storage of approximately 13.4 liters of 5,000 psi compressed hydrogen. Two assemblies shall be used on a single fork lift to allow for the combined storage of approximately 26.8 liters of 5,000 psi compressed hydrogen.

Tubing Material – The material of the raw tubing shall be 316L stainless steel to reduce the risk of hydrogen-induced embrittlement over time. The 316L stainless steel shall have a yield strength no lower than 36,000 psi. The 316L stainless steel shall have a fatigue limit (based on 10^6-10^9 cycles to failure) above the specified yield strength to reduce the risk of fatigue failure caused by refueling cycling. The 316L stainless steel must meet the *Material Qualification Tests and Requirements* set forth in HGV5.

Tubing Tolerances – The raw tubing shall meet the following requirements for maximum allowable dimensional variations;

- For the 1” and 7/8” OD tubes; Maximum of 5% variation in the wall thickness (from the nominal) as measured at various points around the circumference of the tube
- For the 5/8” and 1/2” OD tubes; Maximum of 10% variation in the wall thickness (from the nominal) as measured at various points around the circumference of the tube
- Maximum of 5% variation (from the nominal wall thickness) in the concentricity of the inner and outer diameter
- Maximum of +/-.005” deviation in the outside diameter

Termination – The fittings and end caps specified in the drawings of Appendix F shall be used to terminate the tubing coils of the tank. The fittings shall provide a standard ¾-14 NGT thread for connection to an appropriate valve and the hydrogen supply system on the fork lift truck.

External Surfaces – The external surfaces of the tubing coils shall meet the requirements for *External Surfaces* set forth in HGV5.

MANUFACTURING REQUIREMENTS

Qualification tests, manufacturing procedures and tests.

Qualification Testing – Prior to shipment of a completed tubing coil tank the following design qualification tests must be carried out, according to HGV5, with satisfactory results; *Ambient Cycling Test, Extreme Temperature Cycling Test, Hydrostatic Burst Test, Bonfire Test, Penetration Test, Leak Before Break Test, NDE Defect Size Determination, Expected Service Performance Test.* Any change in the tubing coil tank design may require some or all qualification tests to be repeated according to *Change of Design* in HGV5.

FUNCTIONAL SPECIFICATION

Nested High Pressure Tubing Coils

Tube Bending – The custom manufacturing process required to produce the bends detailed in the drawings of Appendix F shall be designed and implemented according to appropriate industry practice.

Production Unit and Batch Testing – Unit and batch testing must be carried out during the manufacturing process according to the applicable requirements set forth in *Production Tests and Examinations* and *Batch Tests* of HGV5.

Quality Assurance – In general, manufacturing must be carried out according to the section *Manufacture* in HGV5. Quality assurance practices must be established and operated to ensure all tubing coil tanks will be manufactured according to the qualified design. Quality assurance practices must meet the requirements of the *Quality Assurance* section in HGV5.

Marking, Dispatch, and Records – Each tubing coil tank must be marked and dispatched from the manufacturing facility per the requirements of *Marking and Dispatch* set forth in HGV5. The manufacturer shall follow the requirements for *Records of Manufacture* set forth in HGV5.

OPERATIONAL REQUIREMENTS

Operating temperature, pressure, hydrogen purity, inspection, mounting.

Mounting – The tubing coil tanks shall be mounted on either side of the fixed portion of the fork lift truck mast. The mounting system must be designed according to appropriate industry practice so as to 1) not interfere with the normal operation of the tank, 2) protect the tank from accidental damage, and 3) prevent the build-up of hydrogen gas should a leak occur in or around the tank.

Service and Maximum Pressure – In accordance with HGV5, the service pressure of the tubing coil tank is 5,000 psi and the service life shall be 10 years or 15,000 refueling cycles, whichever is reached first. The maximum pressure is not to exceed 6,250 psi immediately after filling, in accordance with HGV5.

Temperature – The hydrogen gas temperature and container temperature shall meet the requirements for *Settled Gas Temperatures* and *Container Temperatures* set forth in HGV5. In general, the tubing coil tank shall not be placed in an environment with an ambient temperature below -25°C or above 45°C for an extended period of time.

Hydrogen Composition – The purity and composition of the hydrogen gas used in the tubing coil tank shall meet the requirements set forth in *Gas Composition* of HGV5.
Inspection – Each tubing coil tank shall be visually and ultrasonically inspected periodically while in service according to the *Periodic In-Service Inspection* requirements set forth in HGV5. Any tubing coil tank involved in a collision, fire, or other event that may have caused damage to the tank shall be handled according to the *Conditions Requiring Immediate Inspections* set forth in HGV5. Any tubing coil tank which has been pressurized beyond the maximum allowable pressure shall be handled according to the *Over-Pressurization* requirements set forth in HGV5.

FUNCTIONAL SPECIFICATION

Nested High Pressure Tubing Coils

CONTENT APPROVAL

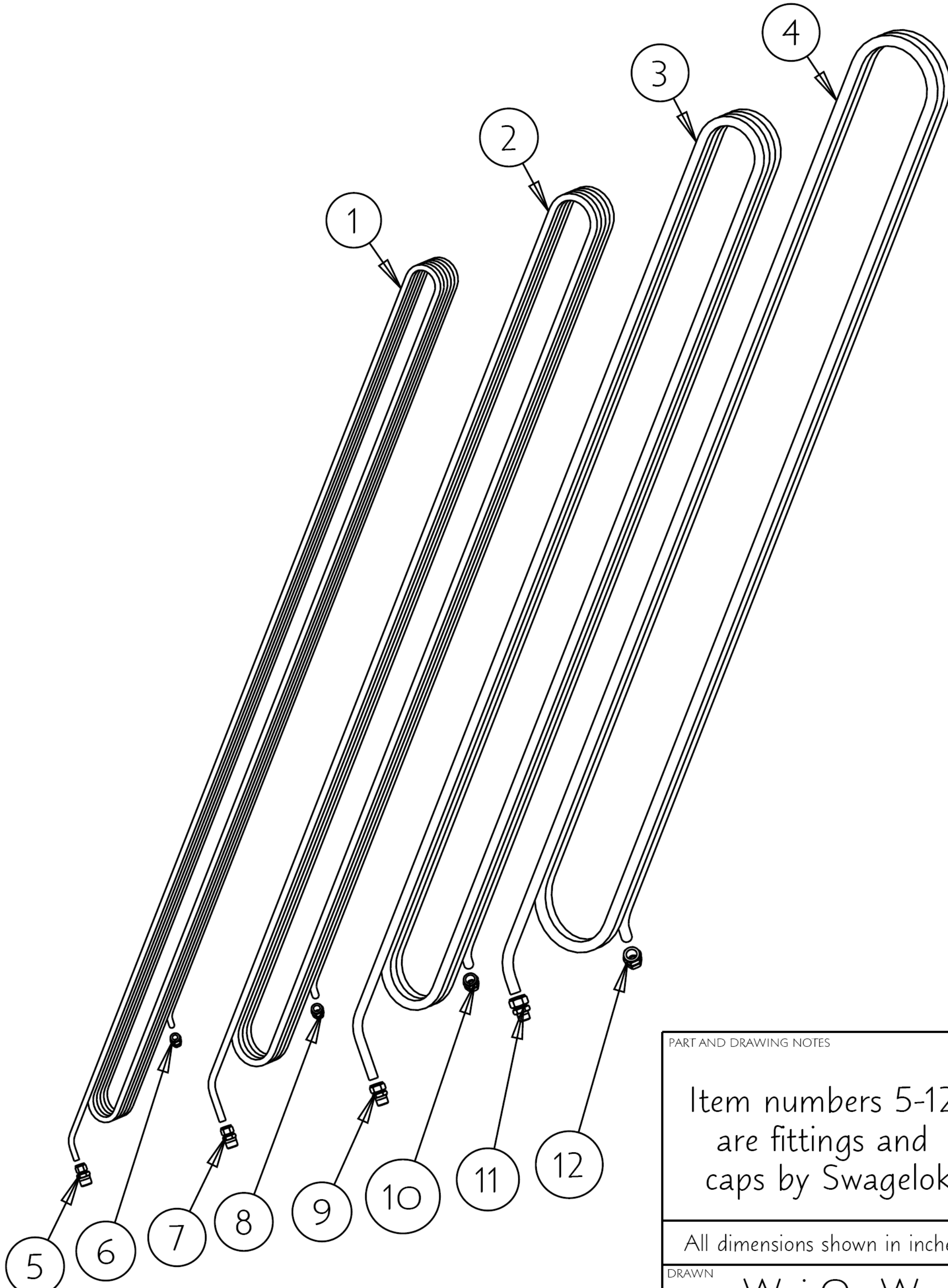
Content approval by the designers and client.

The content of this functional specification and the work carried out on this design proposal is herby approved by:

Matthew Grenier, MDP #8

Wui On Wong, MDP #8

Bryce Gregory, Raymond Corporation



Item	Part Name	Material(s)	Qty	Dwg No.
0	Coil Assembly	316L SS,SS	1	B
1	Tubing Coil 1	316L SS	1	1
2	Tubing Coil 2	316L SS	1	2
3	Tubing Coil 3	316L SS	1	3
4	Tubing Coil 4	316L SS	1	4
5	SS-810-1-12	SS	1	N/A
6	SS-810-C	SS	1	N/A
7	SS-1010-1-12	SS	1	N/A
8	SS-1010-C	SS	1	N/A
9	SS-1410-1-12	SS	1	N/A
10	SS-1410-C	SS	1	N/A
11	SS-1610-1-12	SS	1	N/A
12	SS-1610-C	SS	1	N/A

PART AND DRAWING NOTES

Item numbers 5-12 are fittings and caps by Swagelok

All dimensions shown in inches

DRAWN Wui On Wong

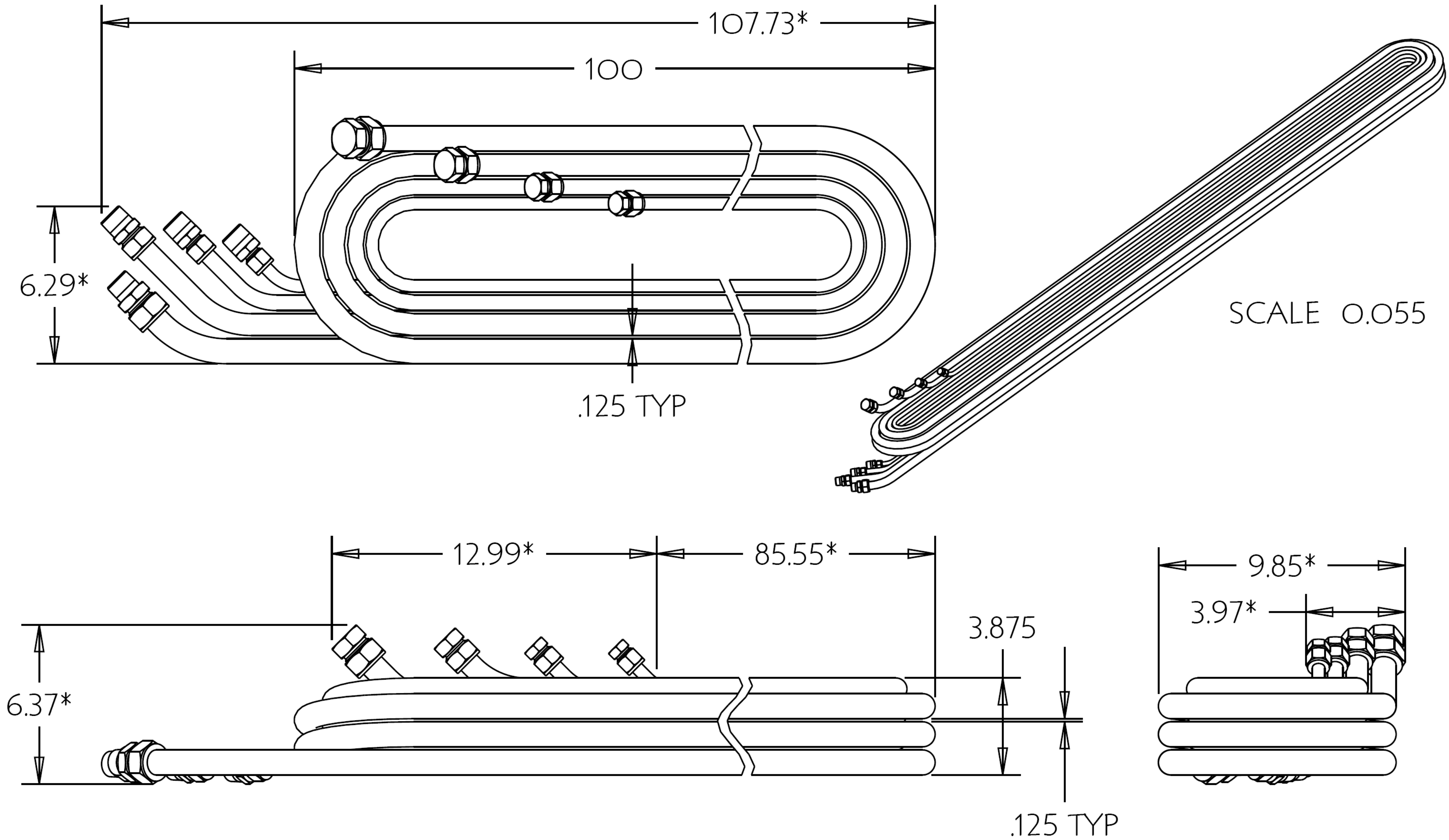
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

TITLE

Coil Assembly

SIZE	FSCM NO.	DWG NO.	REV
A	N/A	A	2

DATE 3/17/10 | SCALE 0.065in = 1in | SHEET 1 OF 6



PART AND DRAWING NOTES

*For reference only

All dimensions shown in inches

DRAWN Wui On Wong

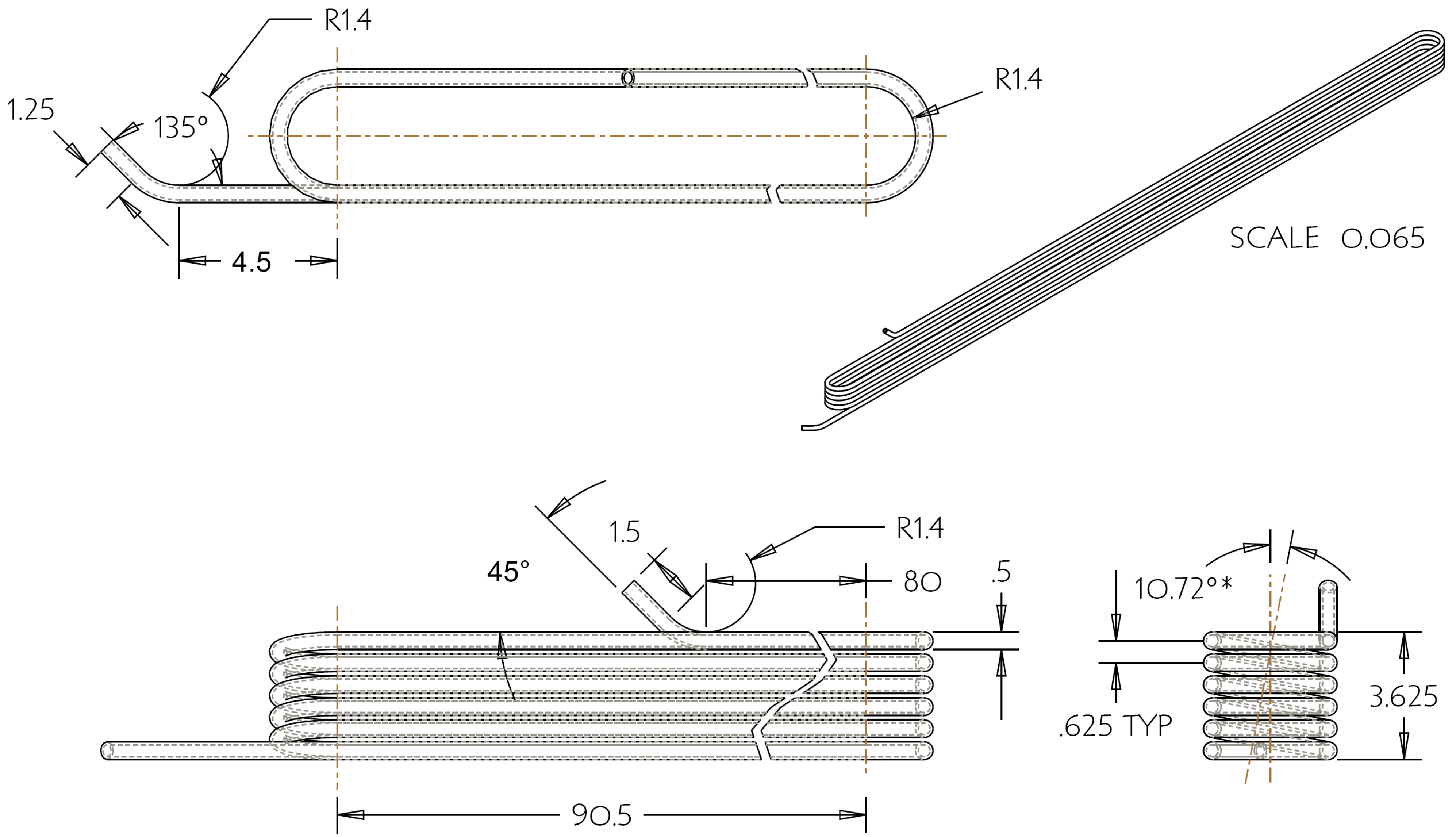
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

TITLE

Coil Assembly

SIZE	FSCM NO.	DWG NO.	REV
A	N/A	B	2

DATE 3/17/10 | SCALE 0.175 in = 1 in | SHEET 2 OF 6



PART AND DRAWING NOTES

OD=.5, t=.083

*Angle for reference only

All dimensions shown in inches

DRAWN Wui On Wong

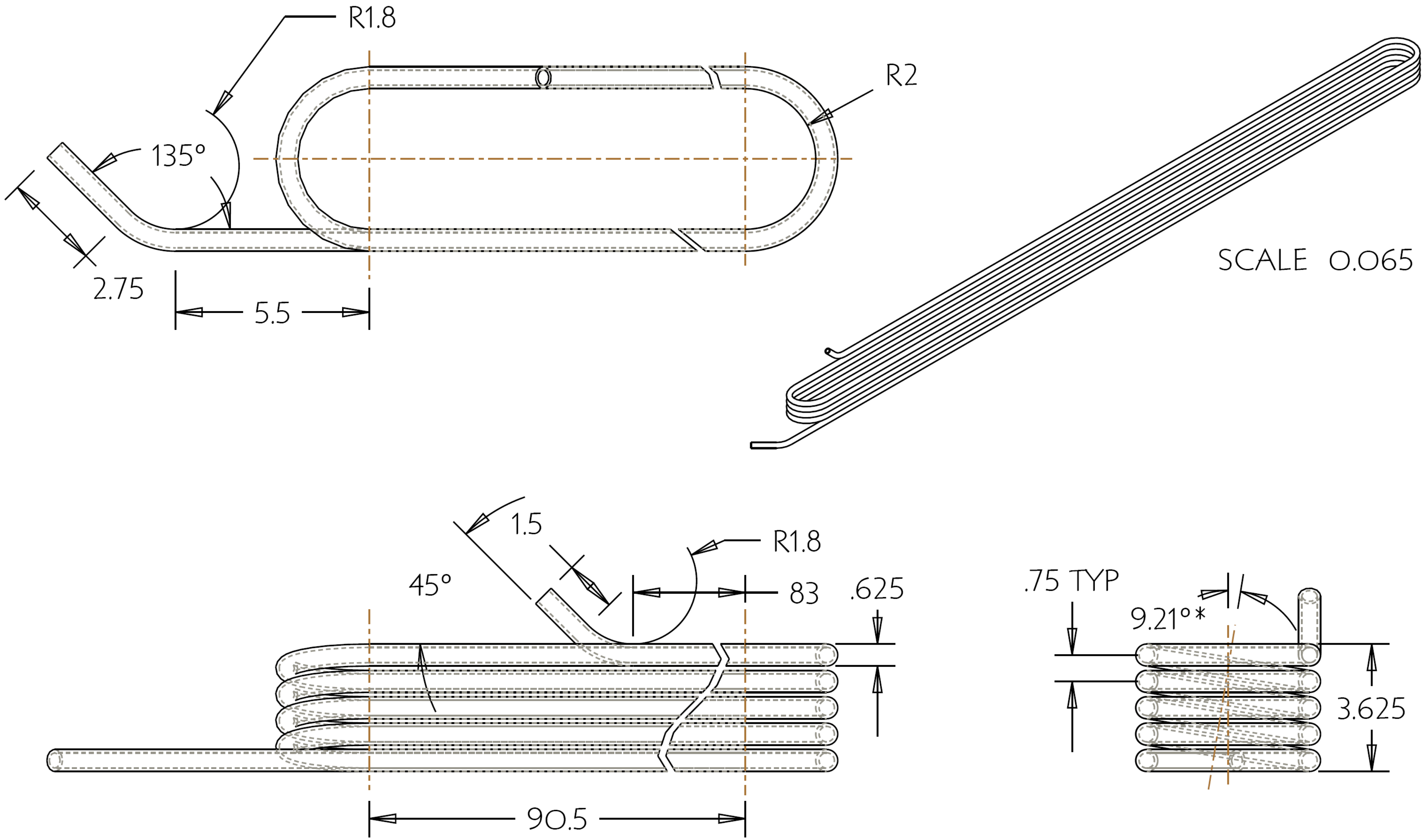
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

TITLE

Tubing Coil 1

SIZE	FSCM NO.	DWG NO.	REV
A	N/A	1	2

DATE 3/17/10 | SCALE 0.25in = 1in | SHEET 3 OF 6



PART AND DRAWING NOTES

OD=.625, t=.095

*Angle for reference only

All dimensions shown in inches

DRAWN Wui On Wong

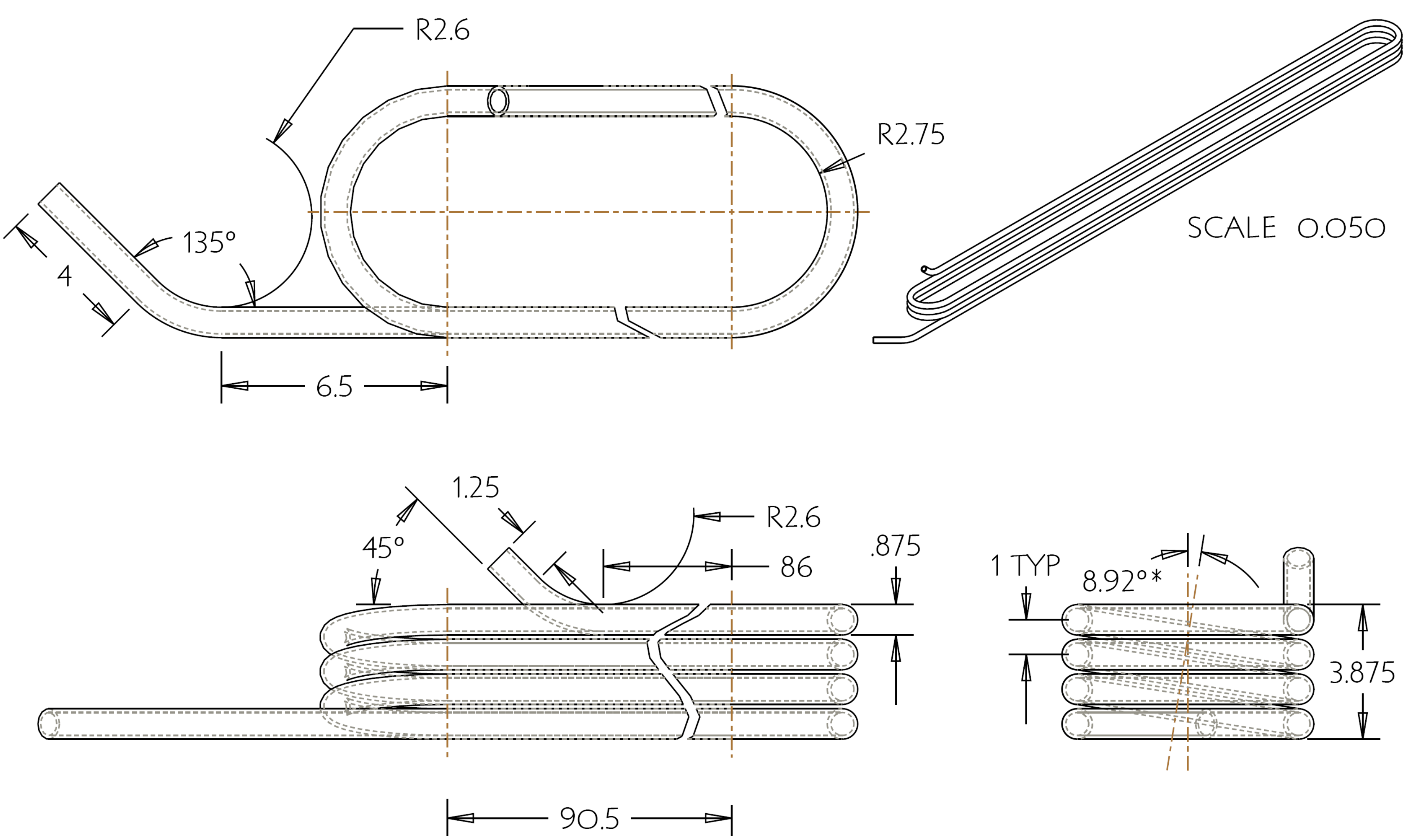
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

TITLE

Tubing Coil 2

SIZE	FSCM NO.	DWG NO.	REV
A	N/A	2	2

DATE 3/17/10 | SCALE 0.25in = 1in | SHEET 4 OF 6



PART AND DRAWING NOTES

OD=.875, t=.109

*Angle for reference only

All dimensions shown in inches

DRAWN Wui On Wong

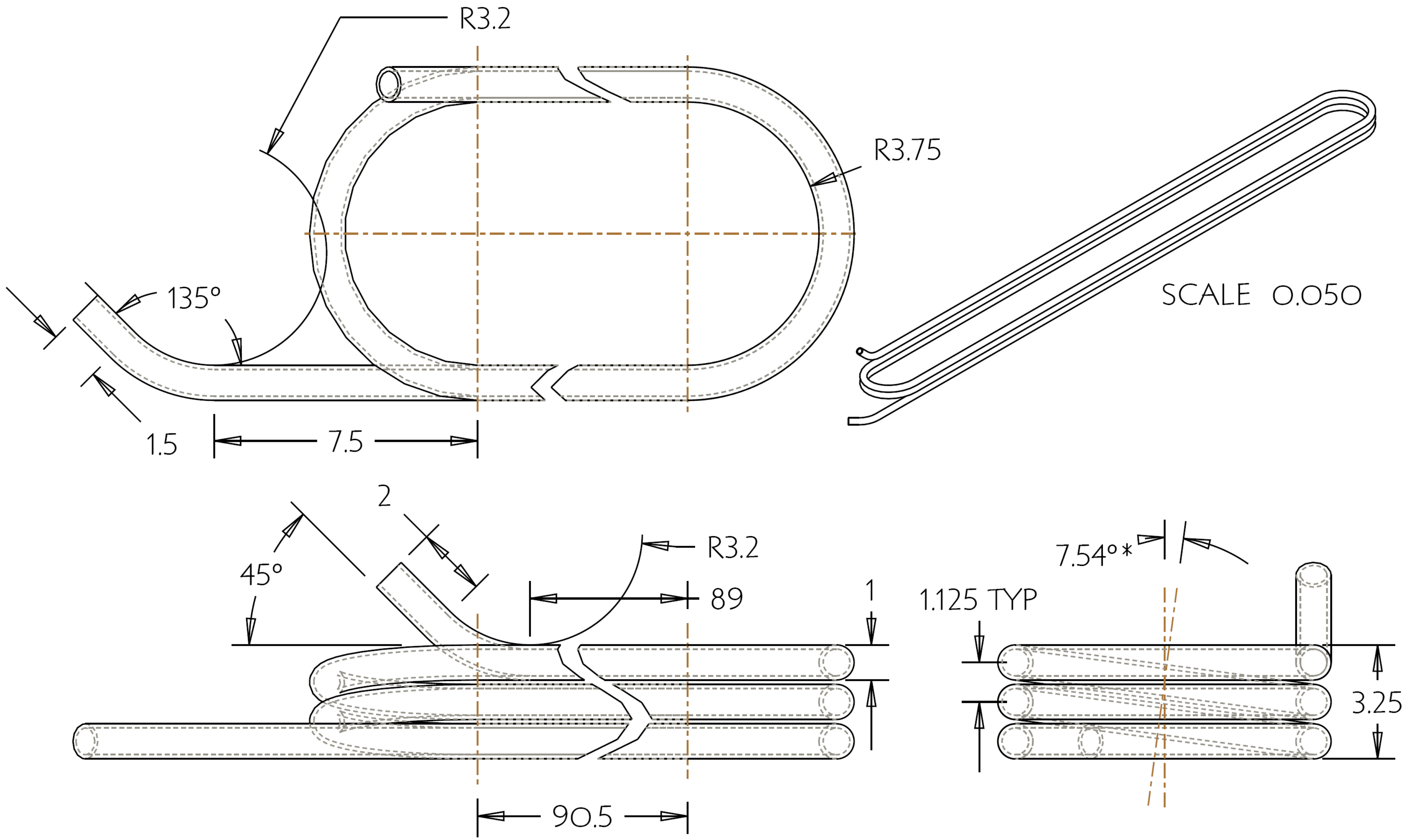
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

TITLE

Tubing Coil 3

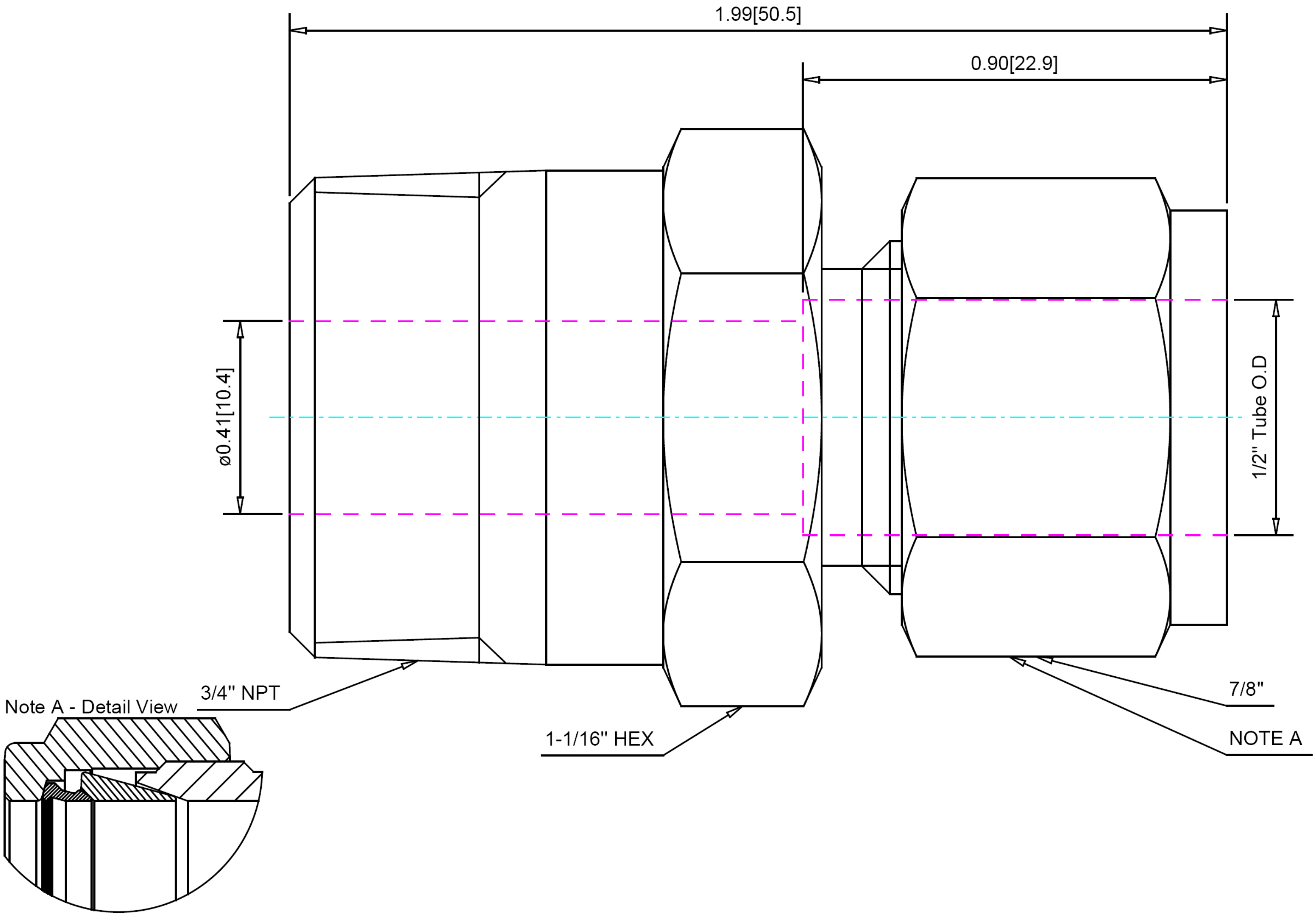
SIZE	FSCM NO.	DWG NO.	REV
A	N/A	3	2

DATE 3/17/10 | SCALE 0.25in = 1in | SHEET 5 OF 6



PART AND DRAWING NOTES OD=1, t=.120 *Angle for reference only	Binghamton University Watson School of Engineering Major Design Project #8, Hydrogen Tank Design
	TITLE Tubing Coil 4
All dimensions shown in inches	SIZE A · FSCM NO. N/A · DWG NO. 4 · REV 2
DRAWN Wui On Wong	DATE 3/17/10 · SCALE 0.25in = 1in · SHEET 6 OF 6

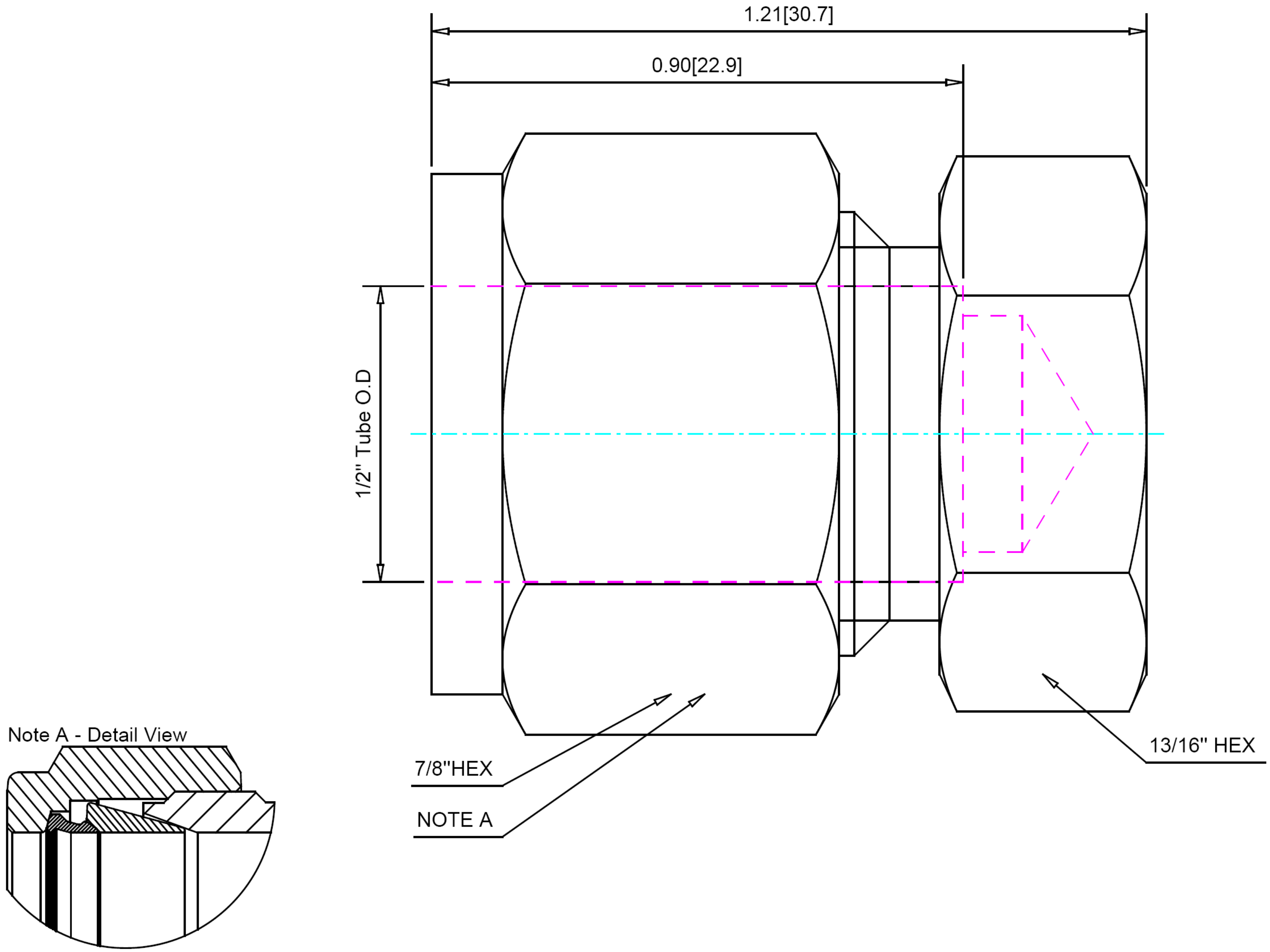
PART NO. SS-810-1-12



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 Male Connector (Tappered Thread) PART NO. SS-810-1-12

PART NO. SS-810-C



CUSTOMER DRAWING

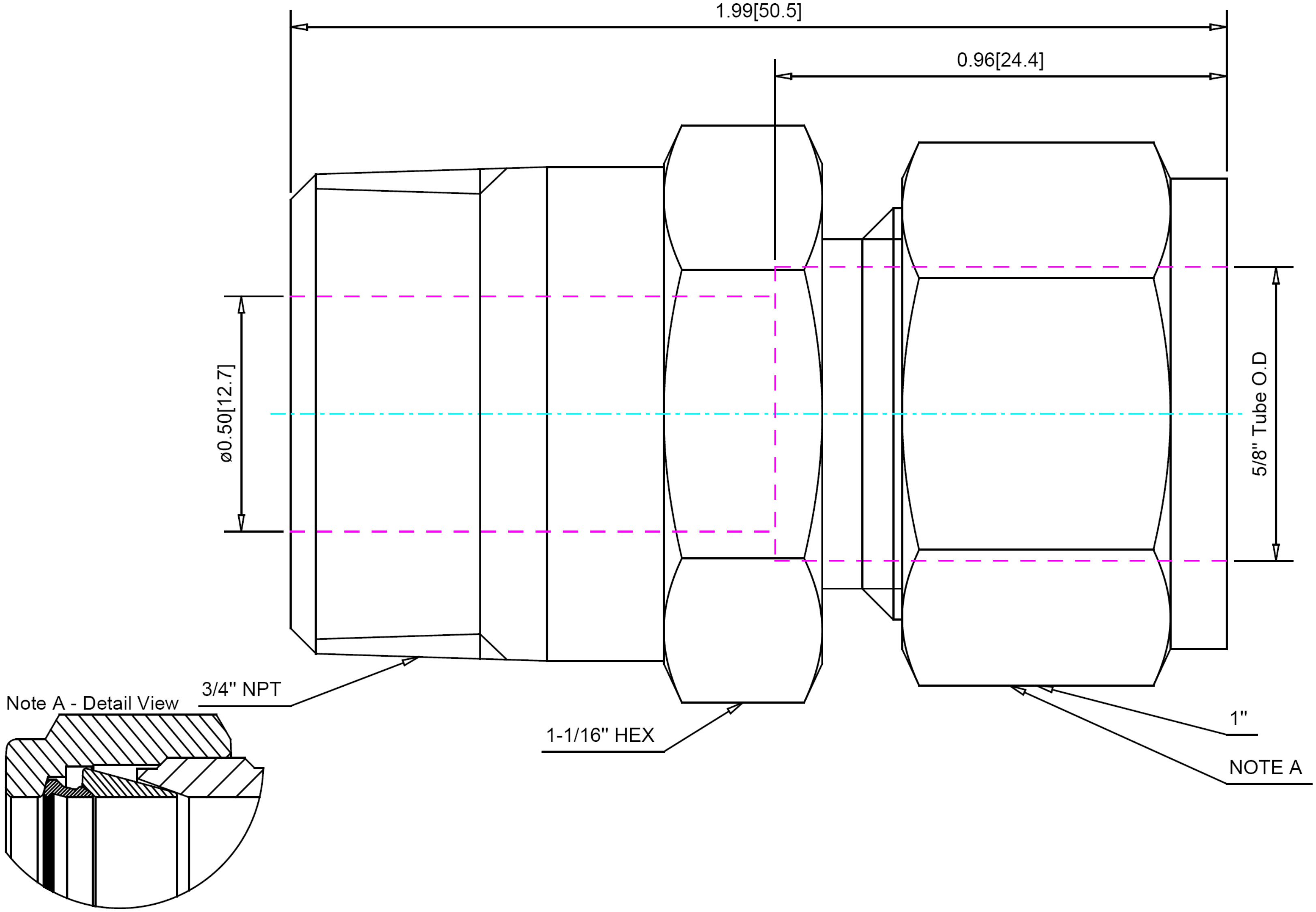
Swagelok

- 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
Cap

PART NO. SS-810-C

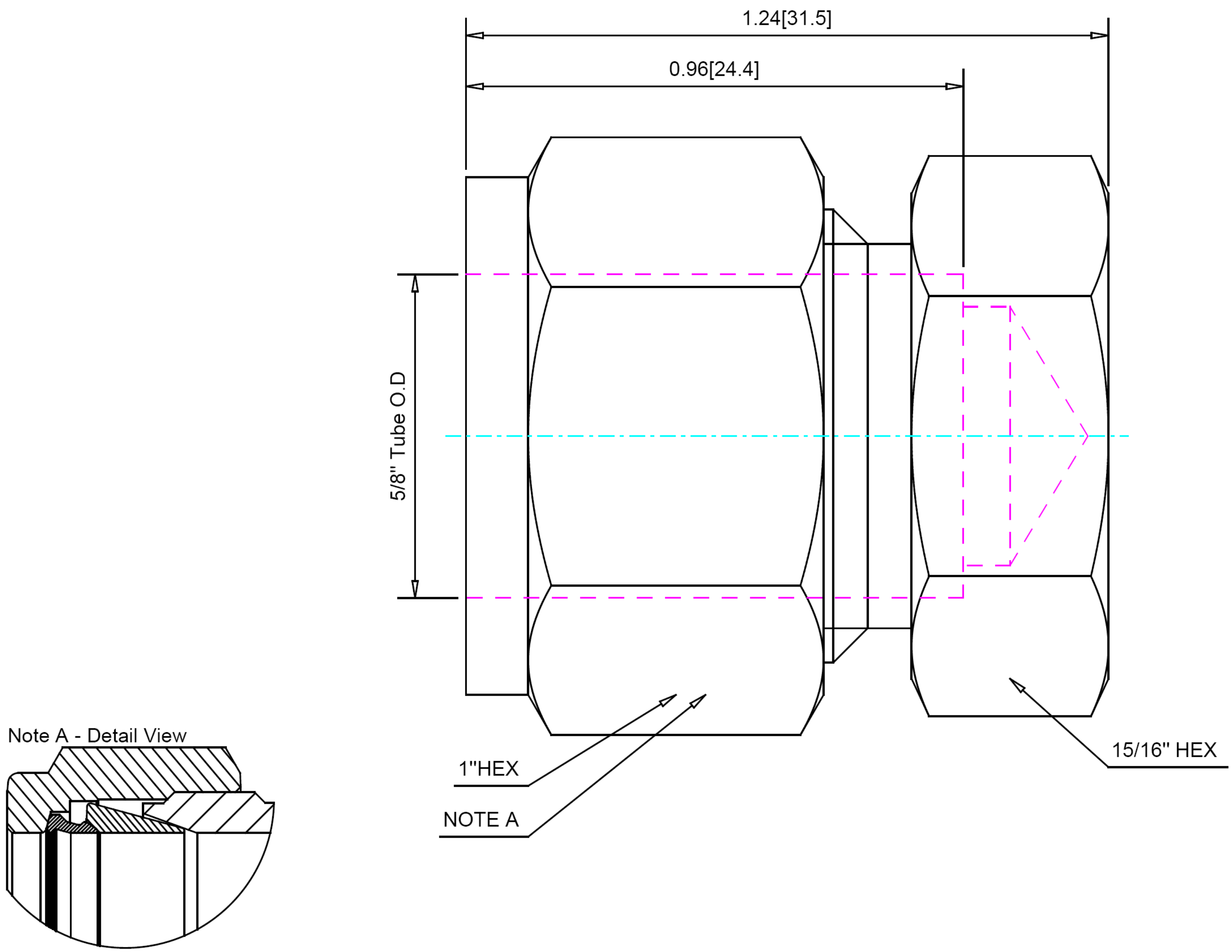
PART NO. SS-1010-1-12



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 Male Connector (Tappered Thread) PART NO. SS-1010-1-12

PART
NO. SS-1010-C



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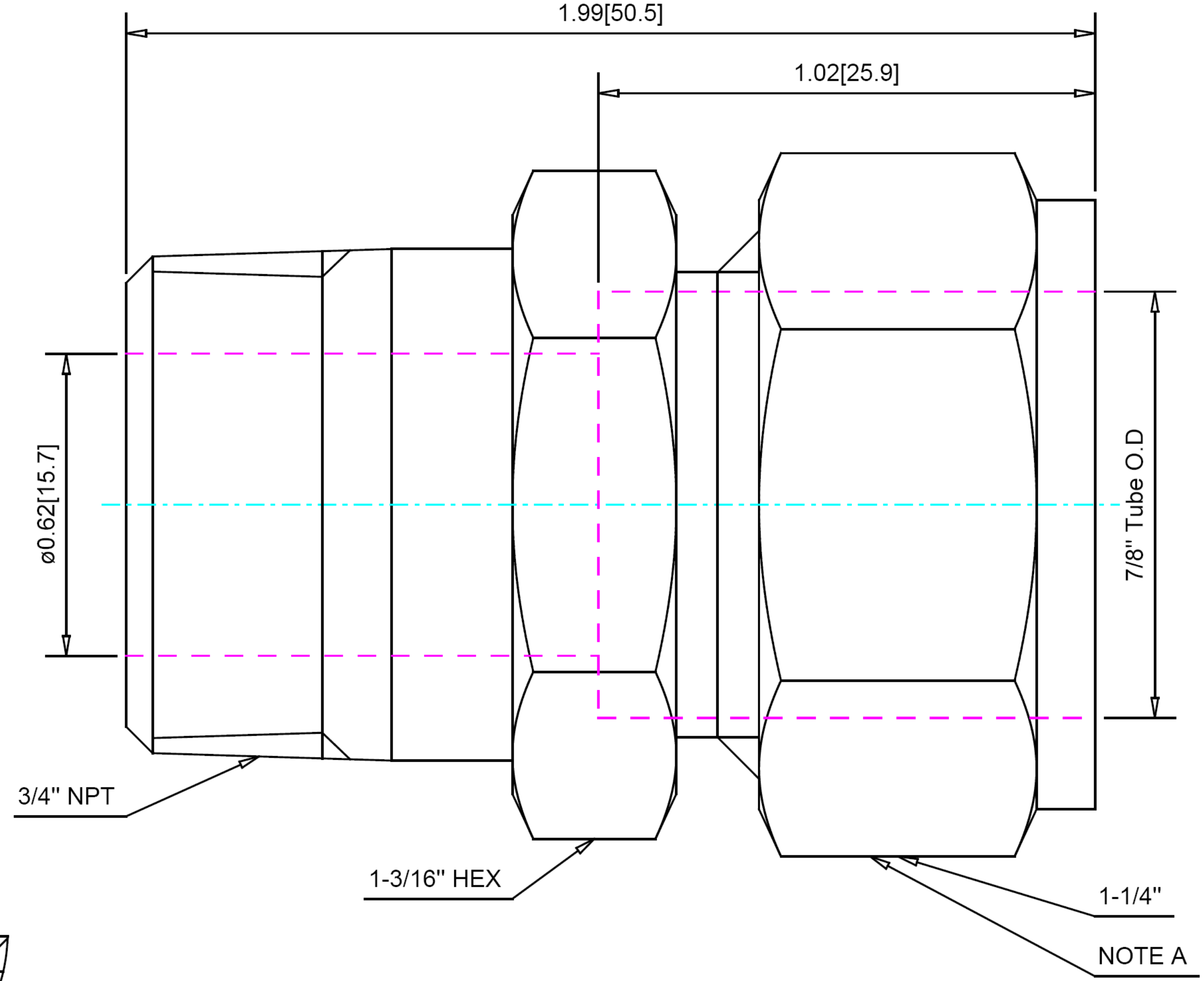
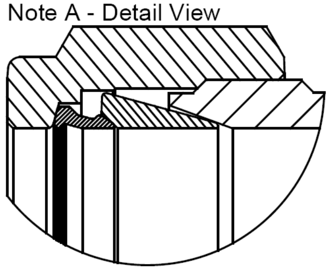
PART NO. SS-1410-1-12



Note A - Detail View



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

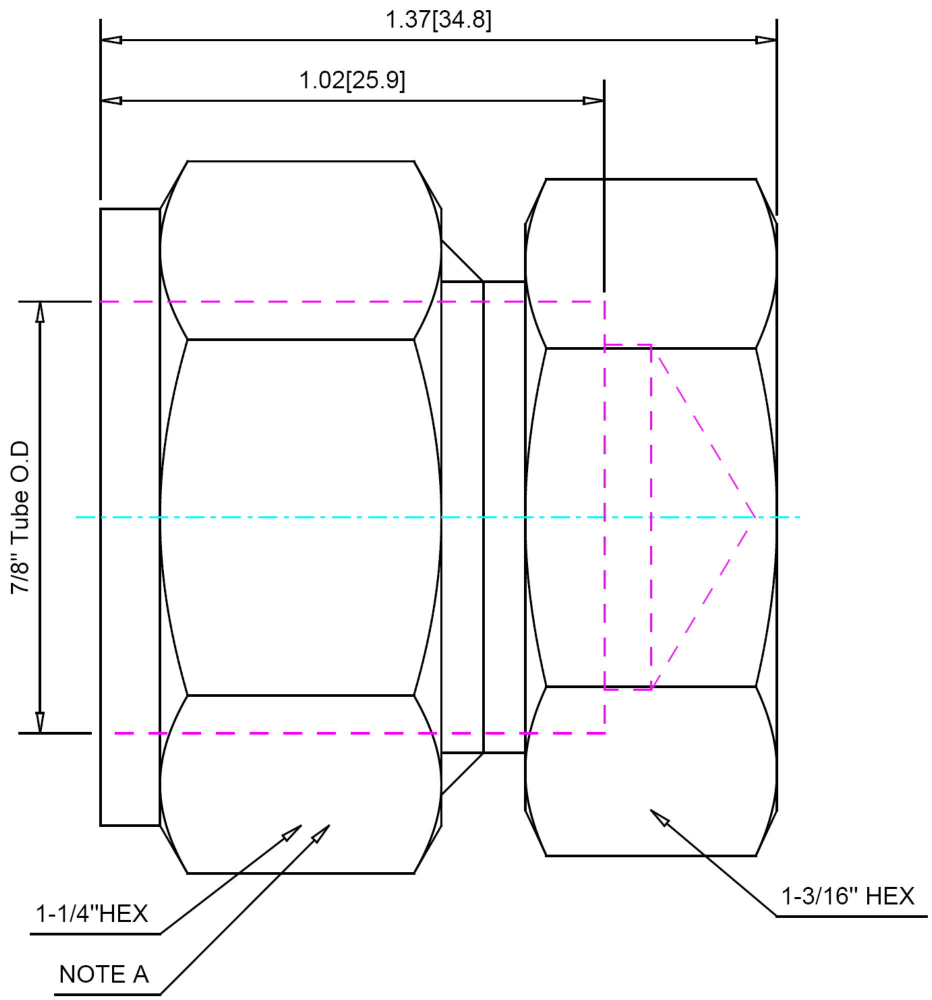
CUSTOMER DRAWING

Swagelok

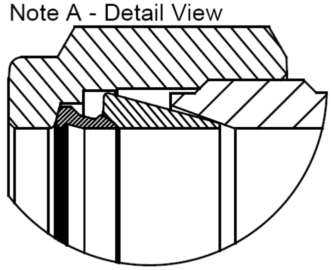
- DRAWING NOT TO SCALE
- DIMENSION ARE INCHES NEXT TO [MILIMETRES]
- DRAWING IS SUBJECT TO CHANGE WITHOUT NOTICE.
- ALL ASSEMBLED NUTS AND FERRULES ARE SHOWN AT FINGER TIGHT DIMENSIONS.
- ALL HEX CALL-OUT ARE ACROSS FLATS.

TITLE Male Connector (Tappered Thread)

PART NO. SS-1410-1-12



Note A - Detail View



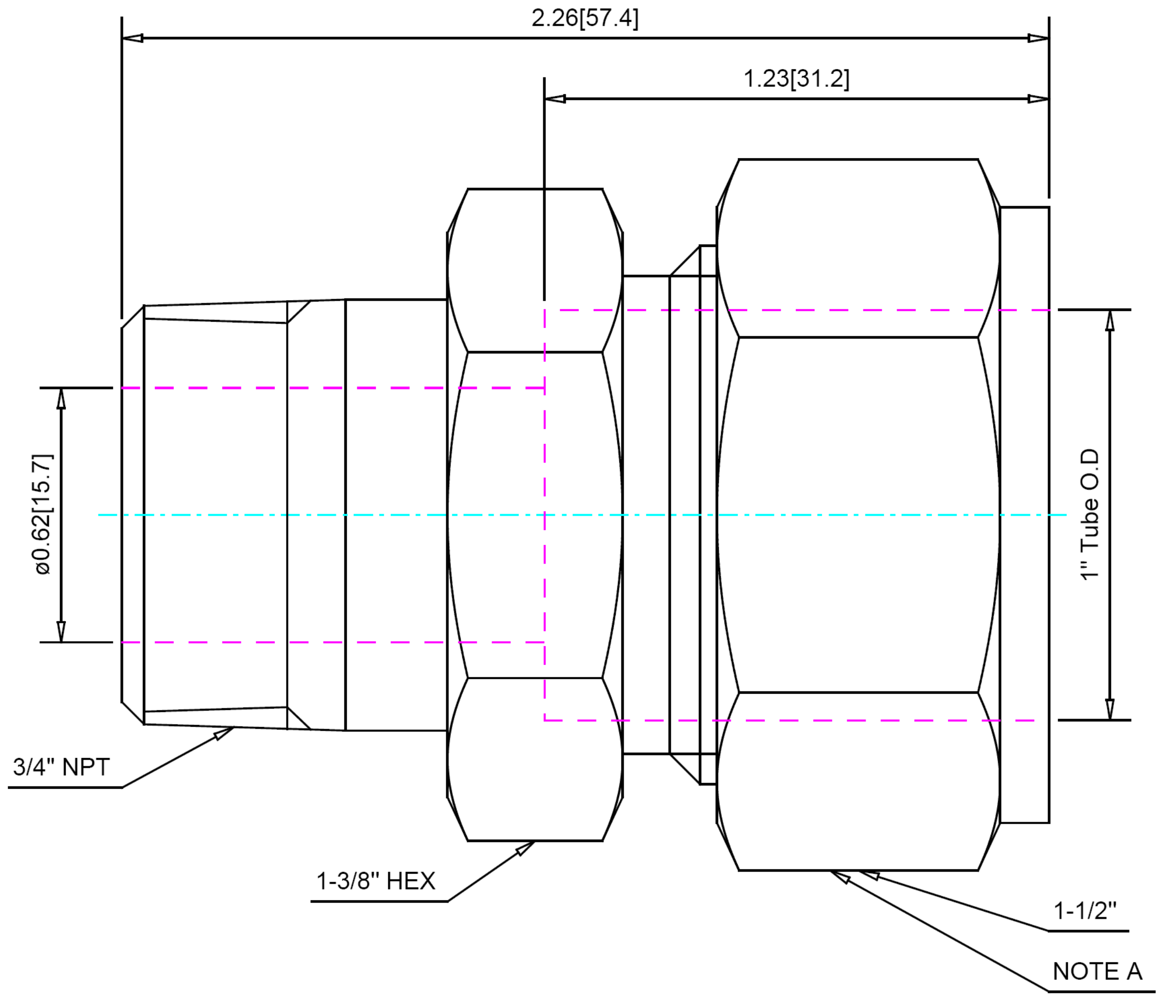
CUSTOMER DRAWING

Swagelok

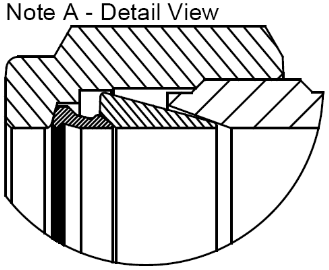
- DRAWING NOT TO SCALE
- DIMENSION ARE INCHES NEXT TO [MILIMETRES]
- DRAWING IS SUBJECT TO CHANGE WITHOUT NOTICE.
- ALL ASSEMBLED NUTS AND FERRULES ARE SHOWN AT FINGER TIGHT DIMENSIONS.
- ALL HEX CALL-OUT ARE ACROSS FLATS.

TITLE Cap

PART NO. SS-1410-C



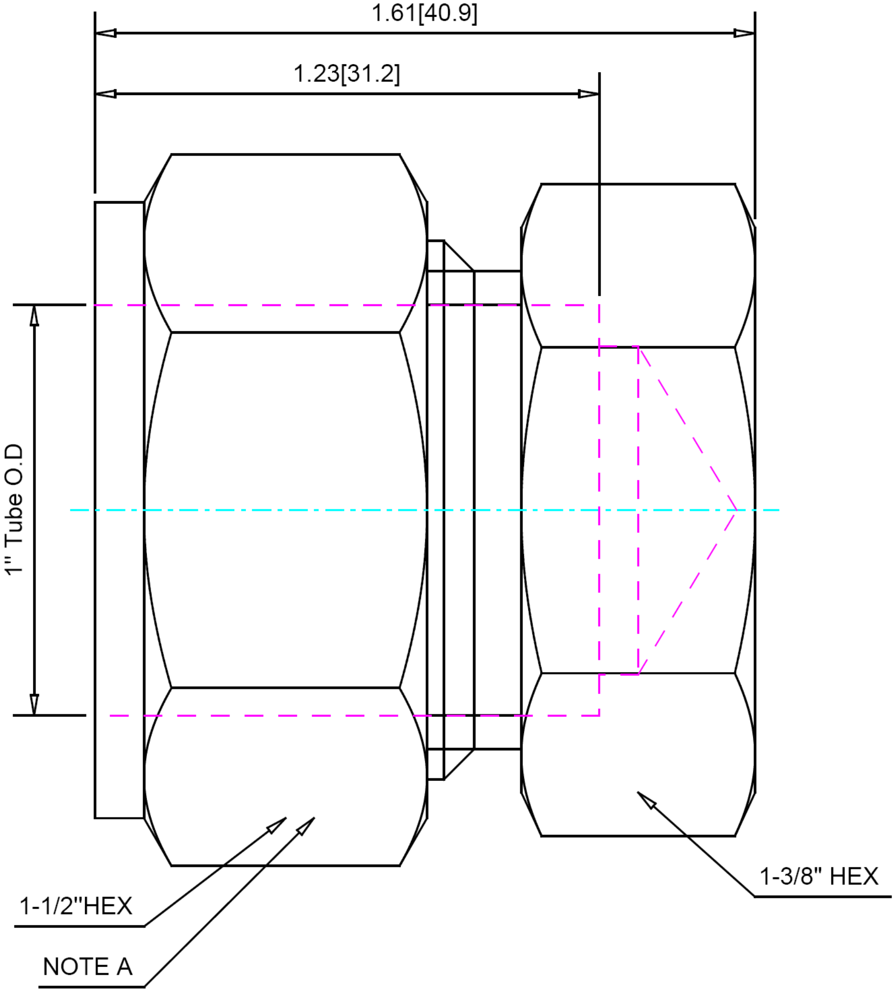
Note A - Detail View



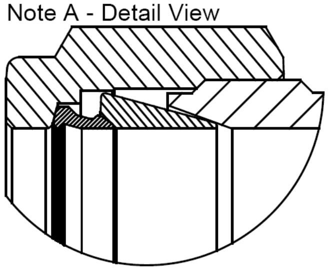
The internal diameter dimension is the minimum nominal opening. These fittings may have a larger opening at the pipe/straight thread end.

CUSTOMER DRAWING		TITLE
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.	Male Connector (Tappered Thread) PART NO. SS-1610-1-12

PART NO. SS-1610-C



Note A - Detail View



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 Cap PART NO. SS-1610-C

APPENDIX G: FUNCTIONAL SPECIFICATION (RECTANGULAR TANK)

FUNCTIONAL SPECIFICATION

Rectangular Tank (Overhead Guard)

This functional specification includes requirements dictated by the finished rectangular tank design in order to meet the basic requirements presented previously in the report as well as the HGV5 design standard requirements for Type 1 compressed hydrogen tanks.

PHYSICAL REQUIREMENTS

Geometry, material, and physical features.

Geometry – The geometries of the raw material, the individual parts, and the finished assembly shall conform to the drawings contained within Appendix G of this report.

Capacity – One rectangular tank assembly will allow for the storage of approximately 41.3 liters of 5,000 psi compressed hydrogen.

Material – The raw material shall be 4130 Q&T steel. The steel shall have a yield strength no lower than 102,000 psi. The steel shall have an ultimate tensile strength of no more than 137,000 psi in accordance with ISO 11114-1. The steel should have a fatigue limit (based on 10^6-10^9 cycles to failure) as high as possible to reduce the risk of fatigue failure caused by refueling cycling. The steel must meet the *Material and Qualification Tests and Requirements* set forth in HGV5. The steel should be qualified for use with high pressure hydrogen according to the requirements of Article KD-10 of Section VIII, Division 1 of the ASME Boiler and Pressure Vessel Code.

Dimensional Variations – The allowable dimensional variations of the raw material, the individual parts, and the finished assembly shall be determined according to appropriate industry practice in order to;

- Minimize any negative effects on the ability to withstand stress
- Minimize any negative effects on manufacturability (includes abilities to fit pieces together as designed and to perform welding)
- Minimize any negative effects on the ability to mount the tank and integrate it with the hydrogen supply system on a fork lift truck

Termination – The rectangular tank shall be terminated as shown in the drawings of Appendix G using a solid valve block. The valve block shall provide two standard ¾-14 NGT threaded openings for connection to appropriate valves and the hydrogen supply system on the fork lift truck. The threads shall comply with *Threaded Openings* in HGV5.

External Surfaces – The external surfaces of the rectangular tank shall be protected by a coating of paint or powder coat according to appropriate industry practice. The protective coating shall meet the requirements for *External Surfaces* set forth in HGV5.

MANUFACTURING REQUIREMENTS

Qualification tests, manufacturing procedures and tests.

Qualification Testing – Prior to shipment of a completed rectangular tank the following design qualification tests listed below must be carried out, according to HGV5, with satisfactory results; *Ambient Cycling Test, Extreme Temperature Cycling Test, Hydrostatic Burst Test, Bonfire Test, Penetration Test, Leak Before Break Test, NDE Defect Size Determination, Expected Service Performance Test.* Any change in the rectangular tank design may require some or all qualification tests to be repeated according to *Change of Design* in HGV5.

FUNCTIONAL SPECIFICATION

Rectangular Tank (Overhead Guard)

Welding – The welding of the joints in the rectangular tank shall be designed, carried out, and inspected according to appropriate industry practice and according to the requirements set forth in *Alternative Construction or Materials* in HGV5.

Production Unit and Batch Testing – Unit and batch testing must be carried out during the manufacturing process according to the applicable requirements set forth in *Production Tests and Examinations* and *Batch Tests* of HGV5.

Quality Assurance – In general, manufacturing must be carried out according to the section *Manufacture* in HGV5. Quality assurance practices must be established and operated to ensure all rectangular tanks will be manufactured according to the qualified design. Quality assurance practices must meet the requirements of the *Quality Assurance* section in HGV5.

Marking, Dispatch, and Records – Each rectangular tank must be marked and dispatched from the manufacturing facility per the requirements of *Marking and Dispatch* set forth in HGV5. The manufacturer shall follow the requirements for *Records of Manufacture* set forth in HGV5.

OPERATIONAL REQUIREMENTS

Operating temperature, pressure, hydrogen purity, inspection, mounting.

Mounting – The rectangular tank shall be mounted on the fork lift truck so as to act as an overhead guard (to protect the operator from falling objects). Due to the 719 pound weight of the rectangular tank, its effect on the vehicle dynamics must be analyzed and found not to be dangerous. The mounting system must be designed according to appropriate industry practice so as to 1) not interfere with the normal operation of the tank, 2) allow it to function as an overhead guard, and 3) prevent the build-up of hydrogen gas should a leak occur in or around the tank.

Service and Maximum Pressure – In accordance with HGV5, the service pressure of the rectangular tank is 5,000 psi and the service life shall be 10 years or 15,000 refueling cycles, whichever is reached first. The maximum pressure is not to exceed 6,250 psi immediately after filling, in accordance with HGV5.

Temperature – The hydrogen gas temperature and container temperature shall meet the requirements for *Settled Gas Temperatures* and *Container Temperatures* set forth in HGV5. In general, the rectangular tank shall not be placed in an environment with an ambient temperature below -25°C or above 45°C for an extended period of time.

Hydrogen Composition – The purity and composition of the hydrogen gas used in the rectangular tank shall meet the requirements set forth in *Gas Composition* of HGV5.

Inspection – Each rectangular tank shall be visually and ultrasonically inspected periodically while in service according to the *Periodic In-Service Inspection* requirements set forth in HGV5. Any rectangular tank involved in a collision, accident, fire, or other event that may have caused damage to the tank shall be handled according to the *Conditions Requiring Immediate Inspections* set forth in HGV5.

FUNCTIONAL SPECIFICATION

Rectangular Tank (Overhead Guard)

CONTENT APPROVAL

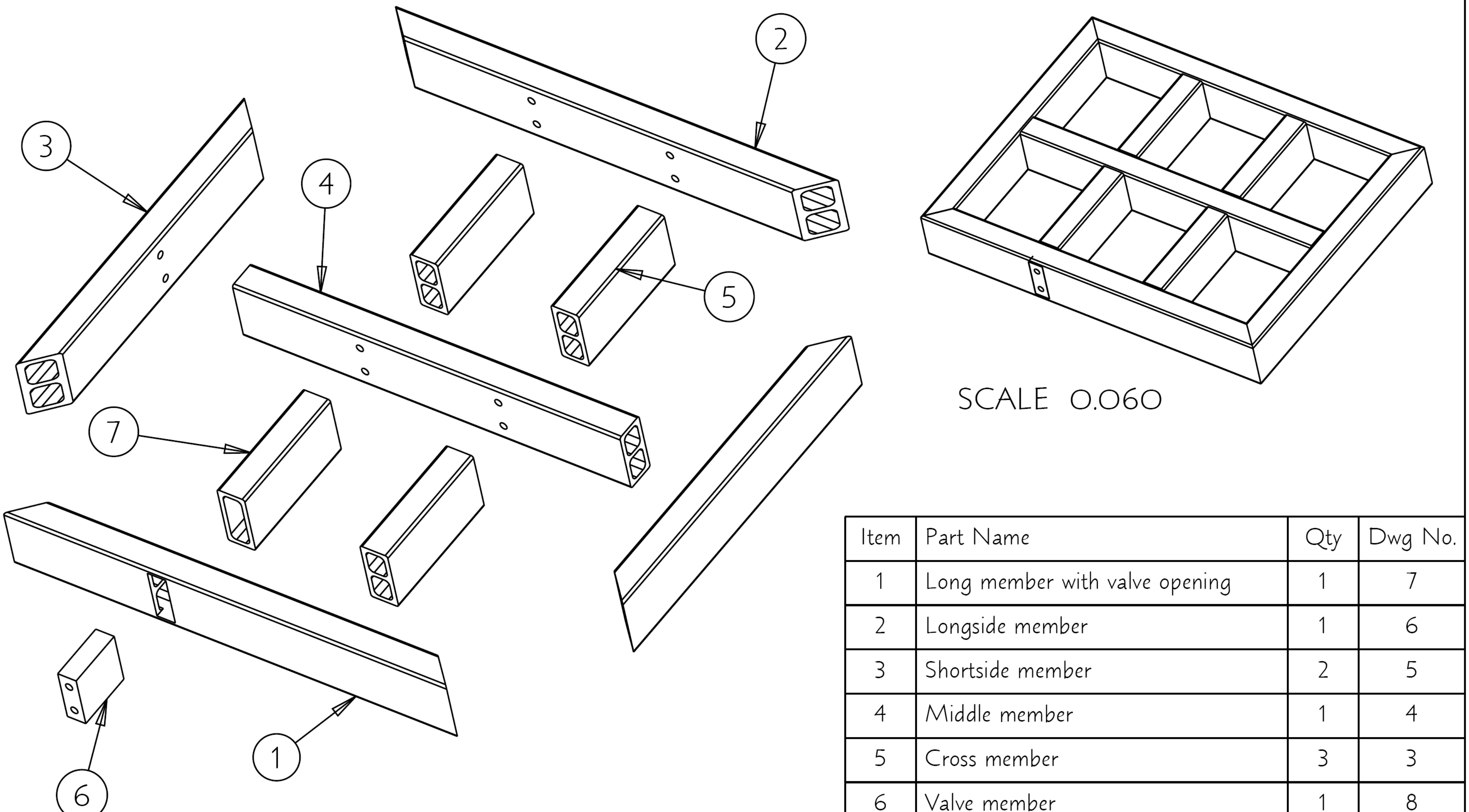
Content approval by the designers and client.

The content of this functional specification and the work carried out on this design proposal is herby approved by:

Matthew Grenier, MDP #8

Anil Kumar Chebrolu, MDP #8

Bryce Gregory, Raymond Corporation



Item	Part Name	Qty	Dwg No.
1	Long member with valve opening	1	7
2	Longside member	1	6
3	Shortside member	2	5
4	Middle member	1	4
5	Cross member	3	3
6	Valve member	1	8
7	Cross member with valve opening	1	9

PART AND DRAWING NOTES

All materials are 4130 Q&T with an UTS not greater than 137 Ksi

All dimensions shown in inches

DRAWN Anil Kumar Chebrolu

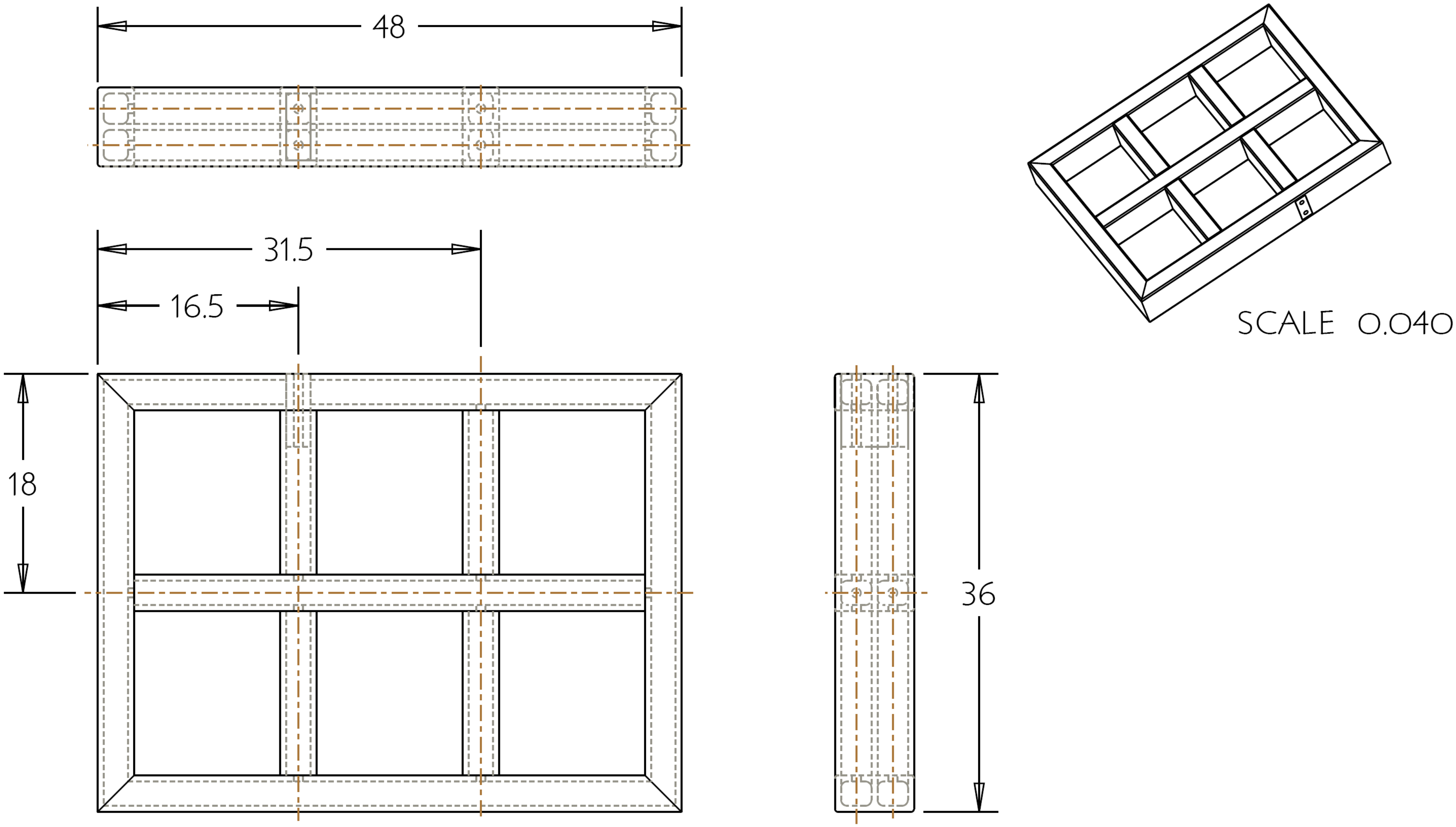
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

TITLE

Over head guard
rectangular tank assembly

SIZE A | FSCM NO. N/A | DWG NO. 1 | REV 5

DATE 04/20/10 | SCALE 0.08 In = 1 In | SHEET 1 OF 9



PART AND DRAWING NOTES

All materials are 4130 Q&T with an UTS not greater than 137 Ksi

All dimensions shown in inches

DRAWN Anil Kumar Chebrolu

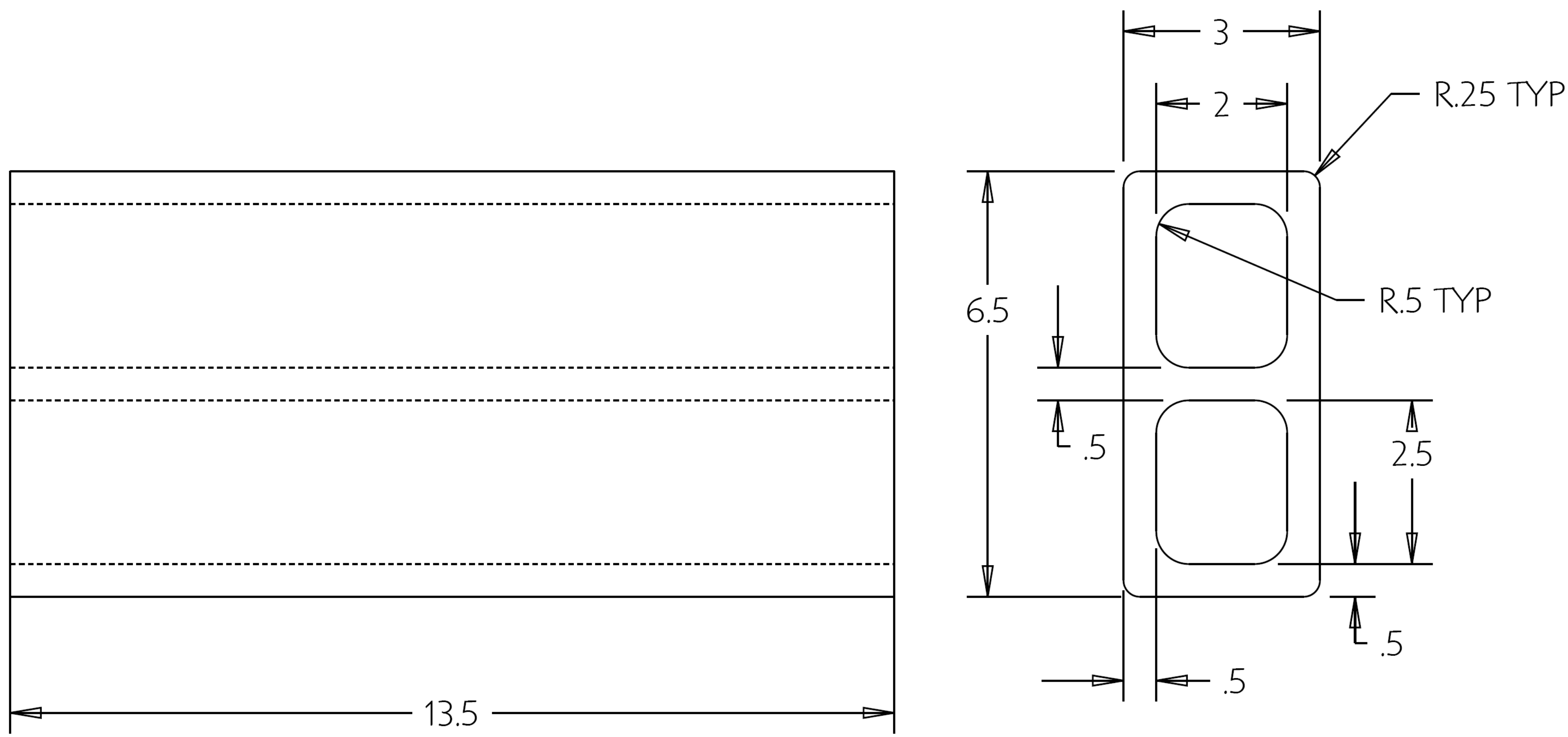
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

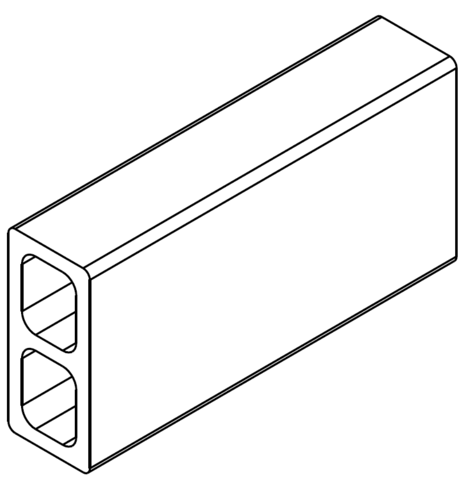
TITLE

Assembly detail drawing

SIZE	FSCM NO.	DWG NO.	REV
A	N/A	2	4

DATE	SCALE	SHEET
04/20/10	0.08 in = 1 in	SHEET 2 OF 9





SCALE 0.200

PART AND DRAWING NOTES

Material is 4130 Q&T with an UTS not greater than 137 Ksi

All dimensions shown in inches

DRAWN Anil Kumar Chebrolu

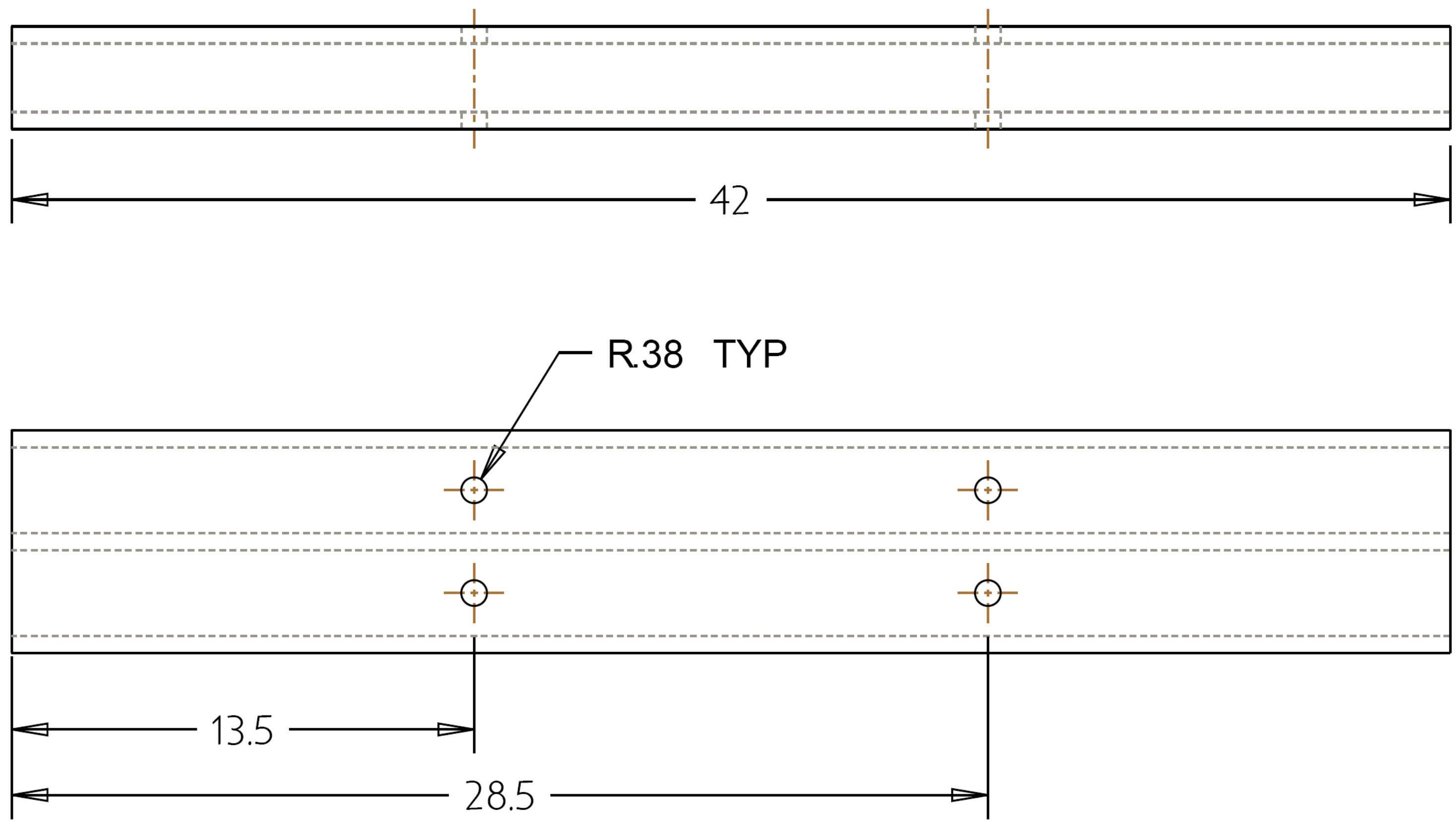
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

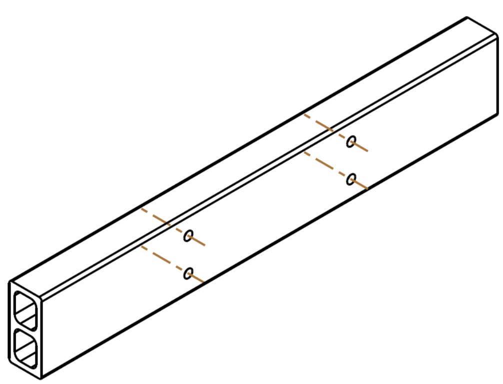
TITLE

Cross member

SIZE	FSCM NO.	DWG NO.	REV
A	N/A	3	4

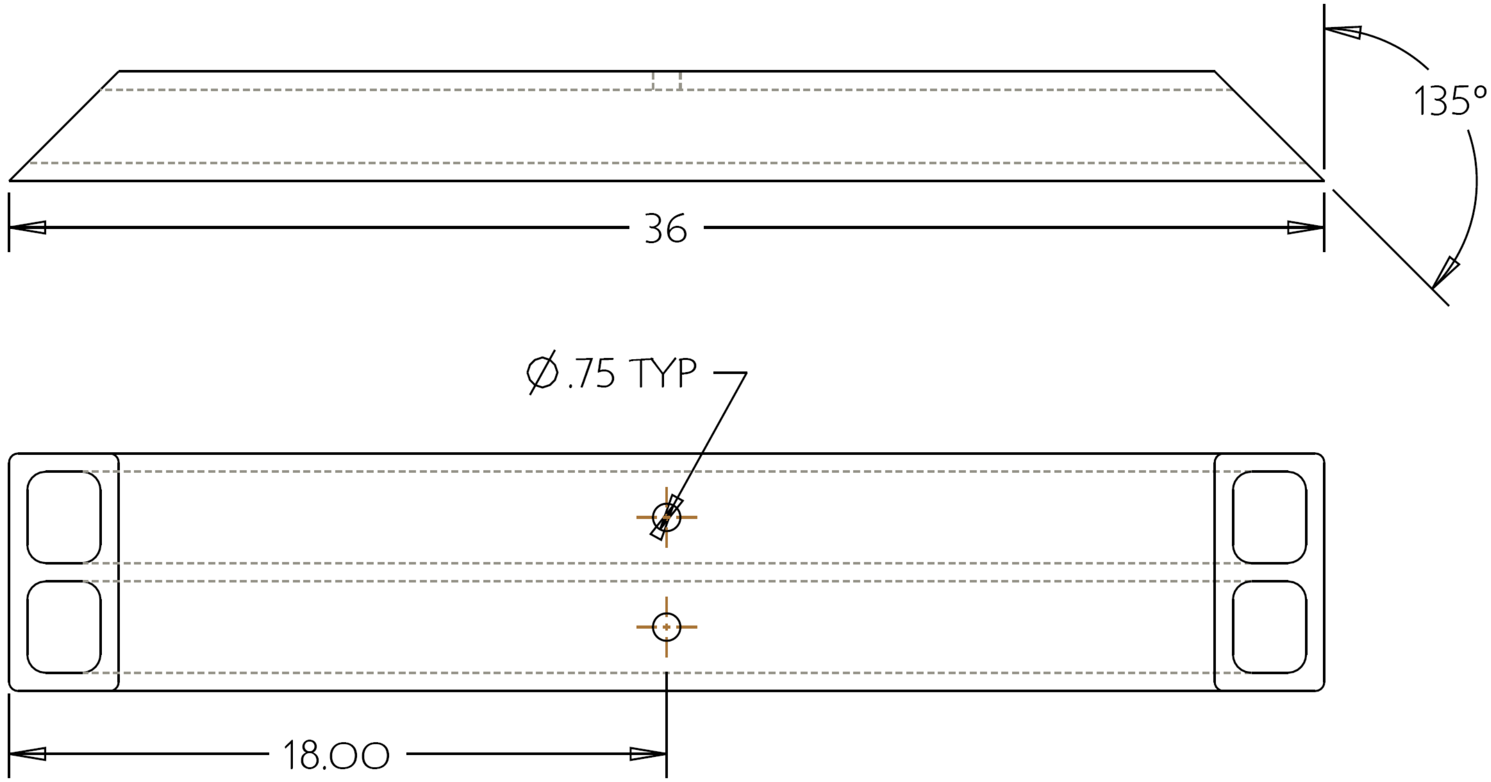
DATE 03/09/10 SCALE 0.4 in = 1in SHEET 3 OF 9

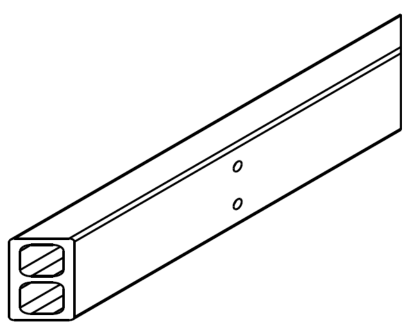




SCALE 0.080

PART AND DRAWING NOTES	Binghamton University Watson School of Engineering Major Design Project #8, Hydrogen Tank Design			
Material is 4130 Q&T with an UTS not greater than 137 Ksi	TITLE: Middle member			
All dimensions shown in inches	SIZE: A	FSCM NO.: N/A	DWG NO.: 4	REV: 4
DRAWN: Anil Kumar Chebrolu	DATE: 04/20/10	SCALE: 0.18 in = 1 in	SHEET 4 OF 9	





SCALE 0.080

PART AND DRAWING NOTES

Material is 4130 Q&T with an UTS not greater than 137 Ksi

All dimensions shown in inches

DRAWN Anil kumar Chebrolu

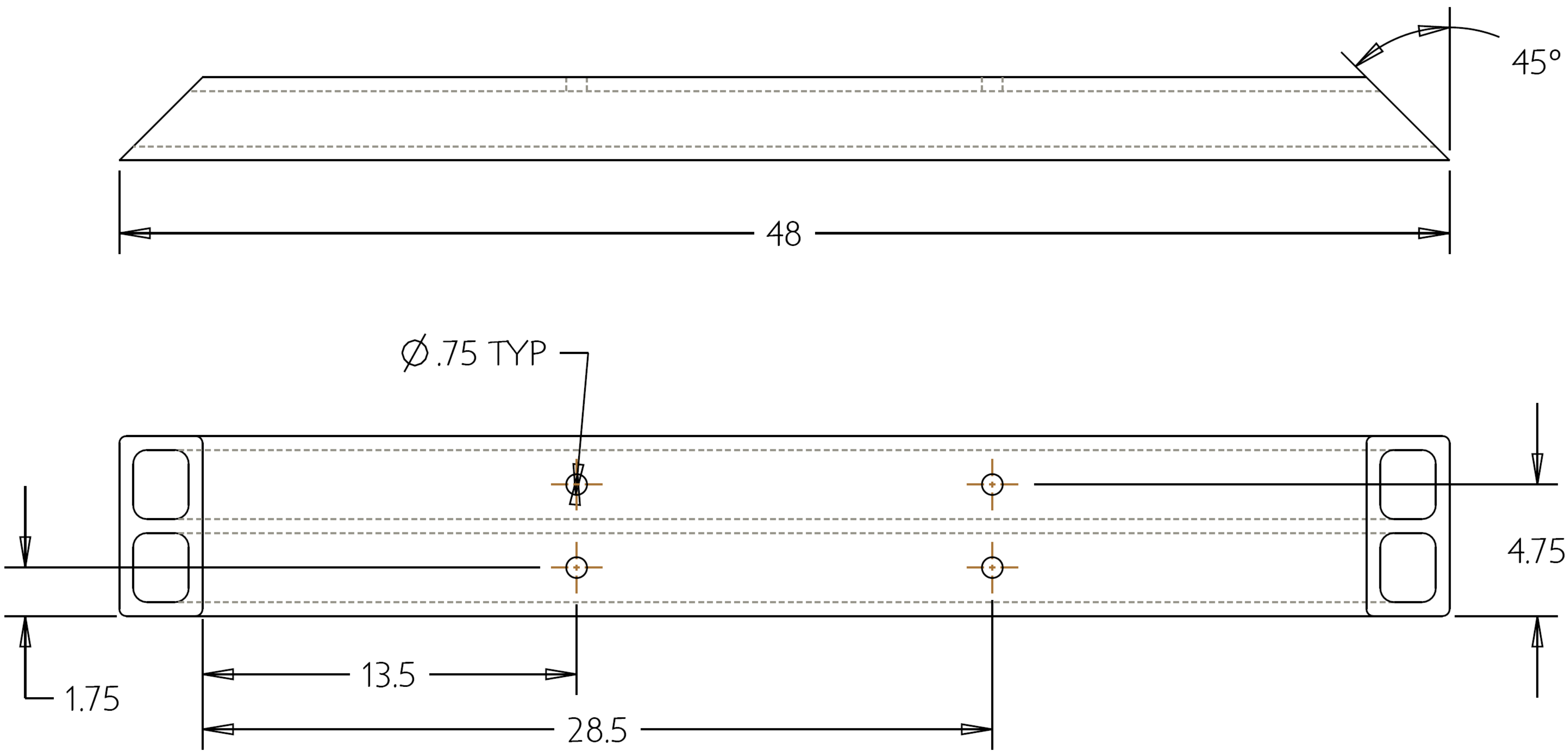
Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

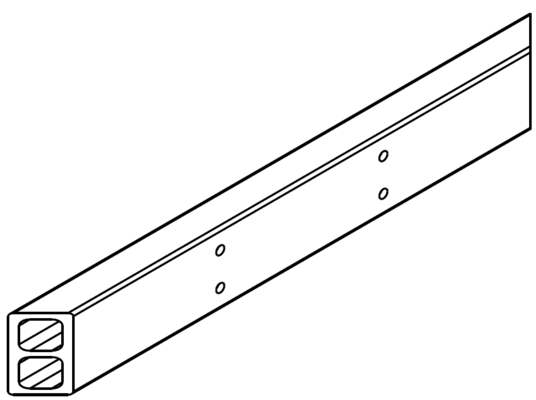
TITLE

Short side member

SIZE	FSCM NO.	DWG NO.	REV
A	N/A	5	4

DATE	SCALE	SHEET	
04/20/10	0.19 in = 1in	SHEET 5 OF 9	





SCALE 0.080

PART AND DRAWING NOTES

Material is 4130 Q&T with an UTS not greater than 137 Ksi

All dimensions shown in inches

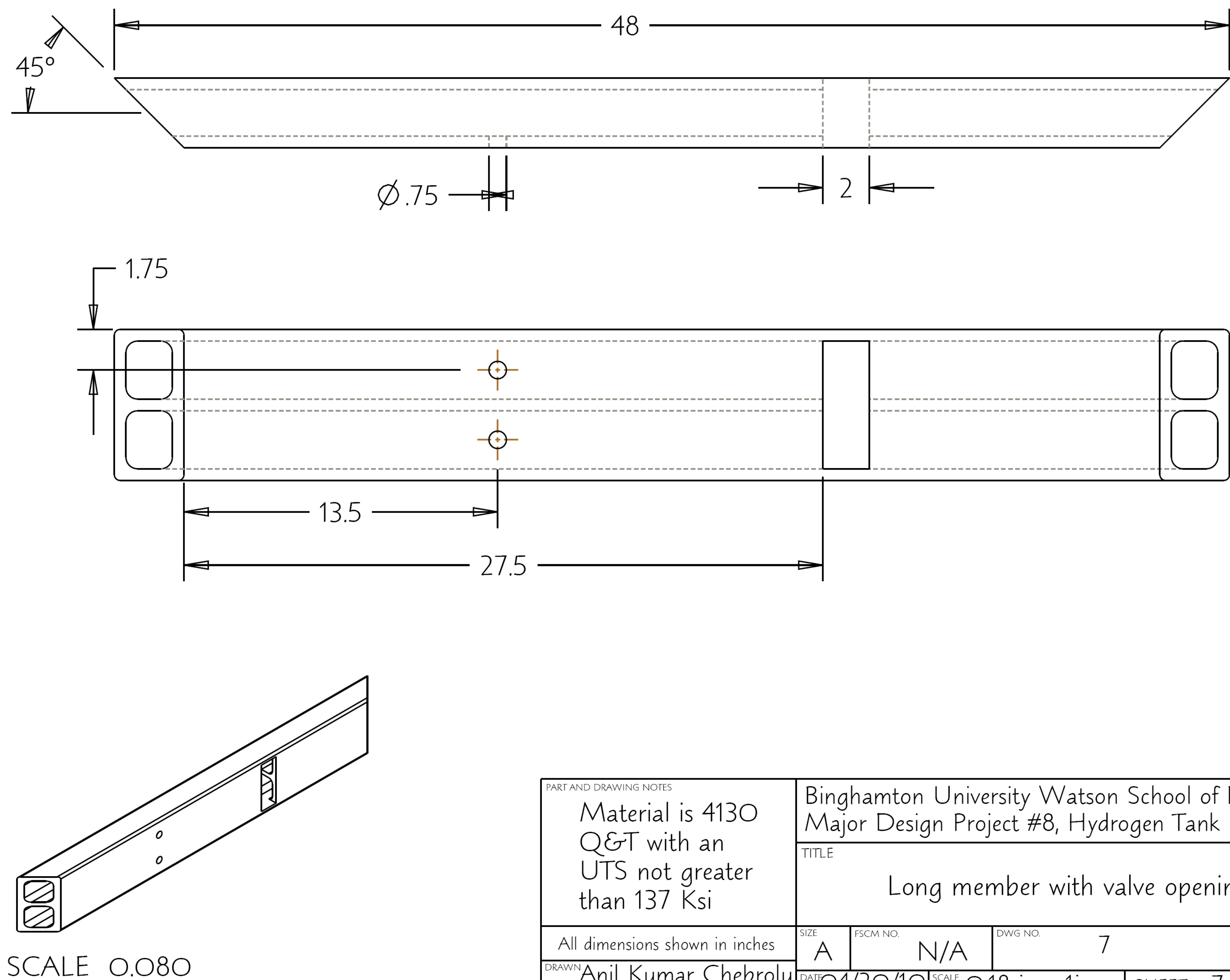
DRAWN Anil Kumar Chebrolu

Binghamton University Watson School of Engineering
Major Design Project #8, Hydrogen Tank Design

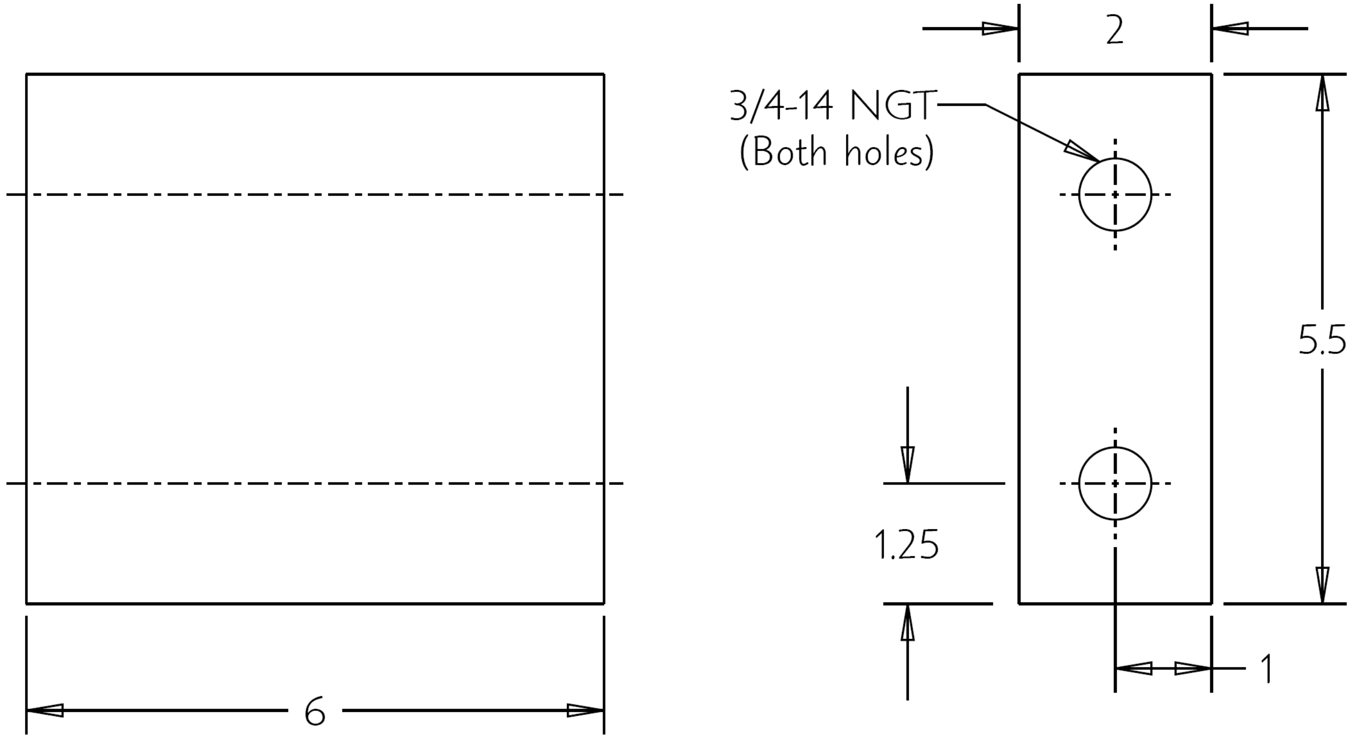
TITLE Long side member

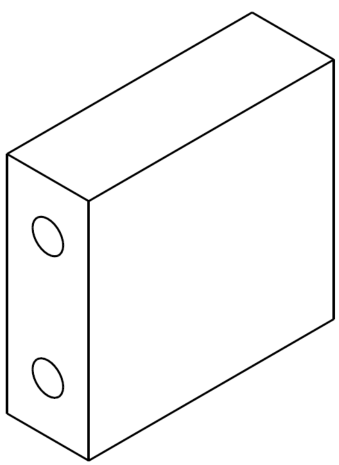
SIZE	FSCM NO.	DWG NO.	REV
A	N/A	6	4

DATE	SCALE	SHEET
04/20/10	0.17 in = 1in	6 OF 9



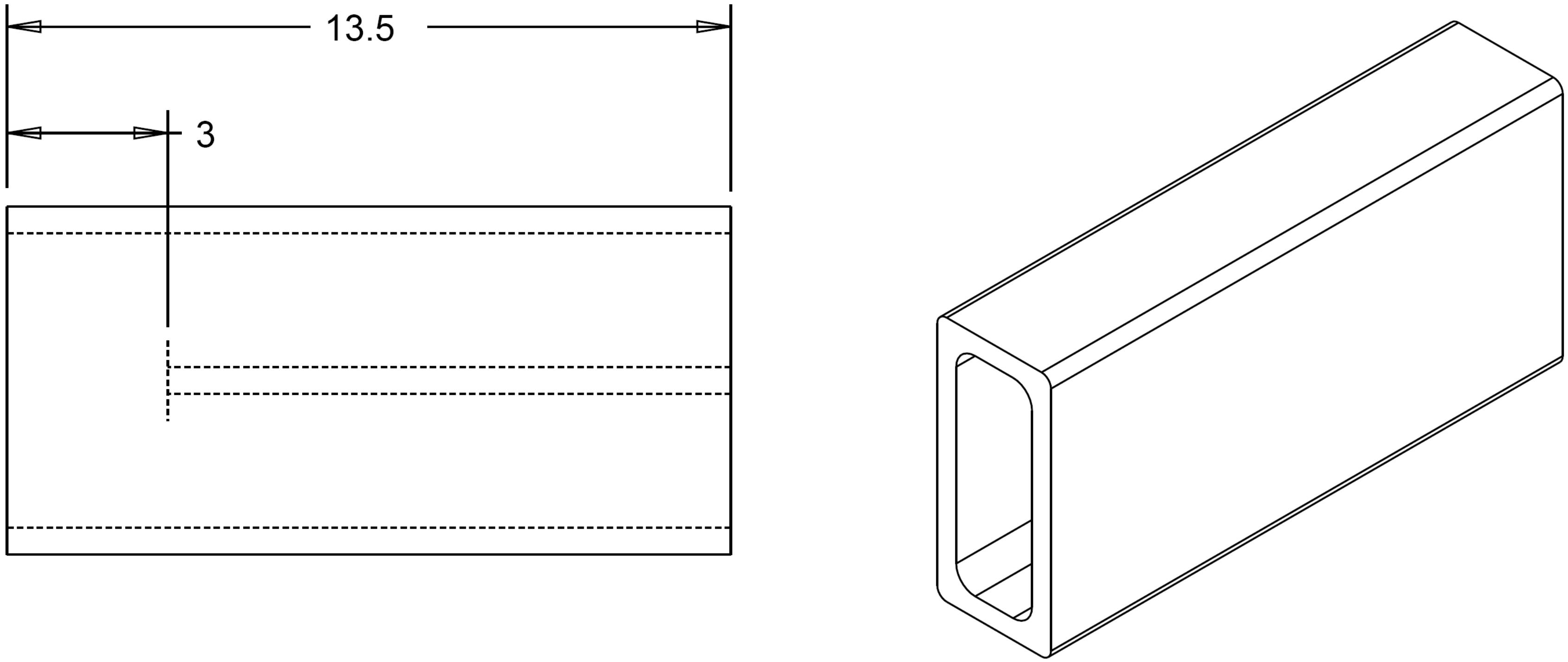
PART AND DRAWING NOTES	Binghamton University Watson School of Engineering Major Design Project #8, Hydrogen Tank Design			
Material is 4130 Q&T with an UTS not greater than 137 Ksi	TITLE Long member with valve opening			
All dimensions shown in inches	SIZE A	FSCM NO. N/A	DWG NO. 7	REV 4
DRAWN Anil Kumar Chebrolu	DATE 04/20/10	SCALE 0.18 in = 1in	SHEET 7 OF 9	





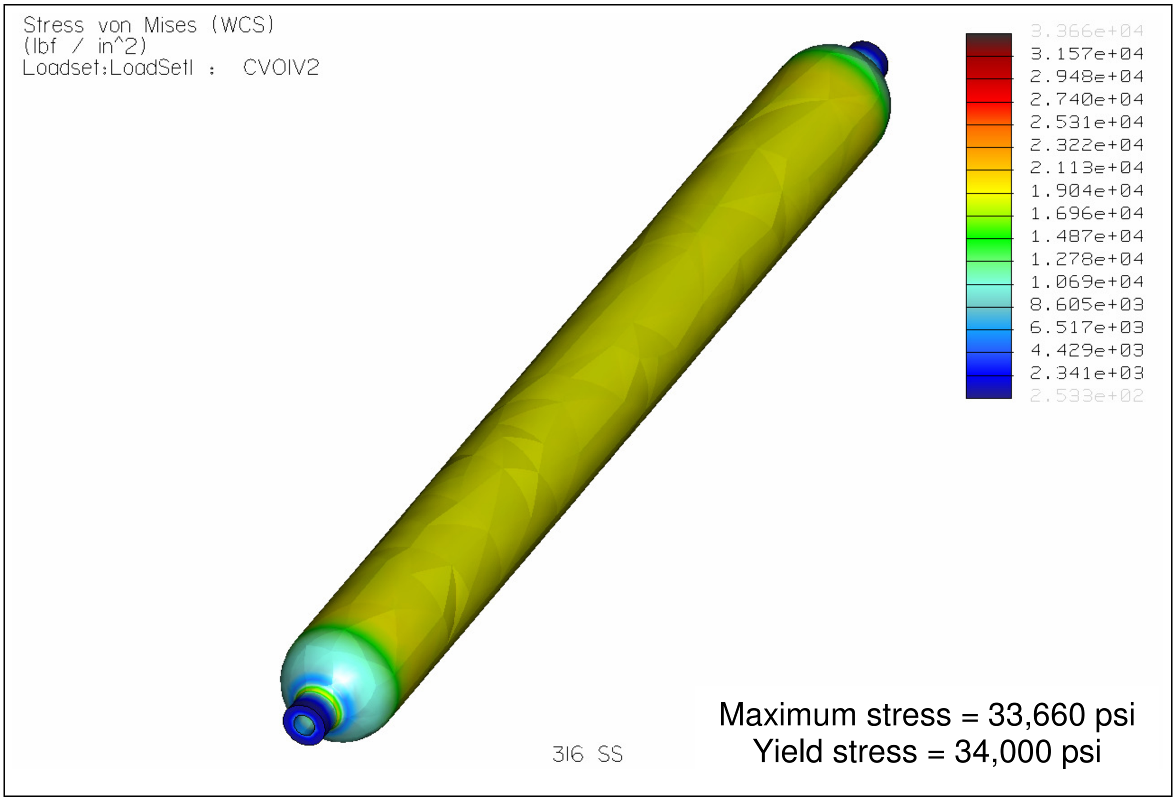
SCALE 0.300

PART AND DRAWING NOTES: Material is 4130 Q&T with an UTS not greater than 137 Ksi	Binghamton University Watson School of Engineering Major Design Project #8, Hydrogen Tank Design			
	TITLE: Valve member			
All dimensions shown in inches	SIZE: A	FSCM NO.: N/A	DWG NO.: 8	REV: 3
DRAWN: Anil kumar Chebrolu	DATE: 03/09/10	SCALE: 0.5 in = 1in	SHEET 8 OF 9	

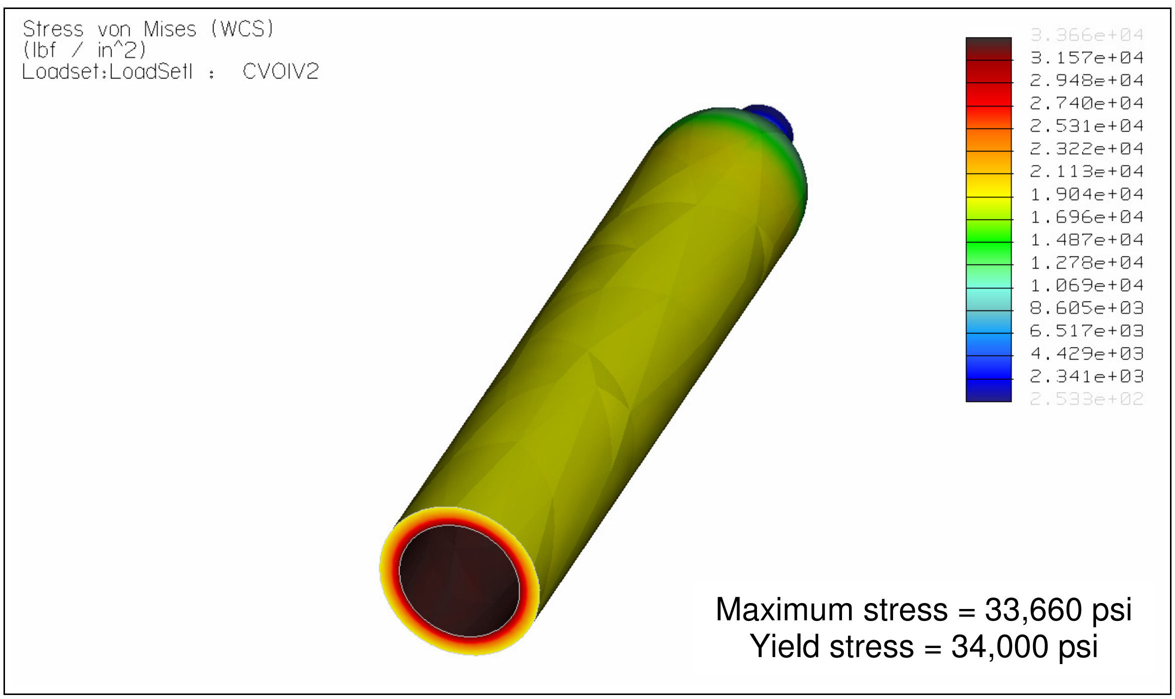


PART AND DRAWING NOTES	Binghamton University Watson School of Engineering Major Design Project #8, Hydrogen Tank Design
Material is 4130 Q&T with an UTS not greater than 137 Ksi	TITLE Cross member with valve opening
All dimensions shown in inches	SIZE A · FSCM NO. N/A · DWG NO. 9 · REV 3
DRAWN Anil Kumar Chebrolu	DATE 03/09/10 · SCALE 0.3 in = 1in · SHEET 9 OF 9

APPENDIX H: FEA RESULTS (CYLINDRICAL VESSEL)

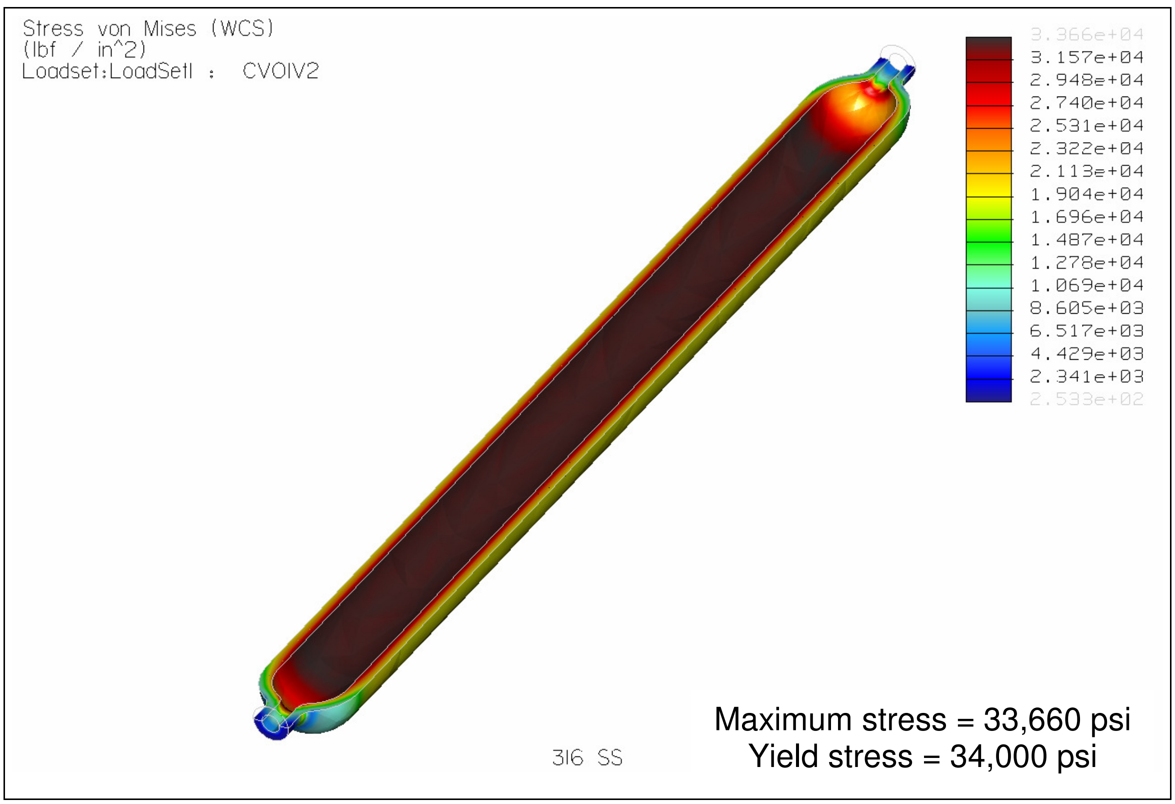


A-H Figure 1 – 316L SS Cylinder Fringe Plot

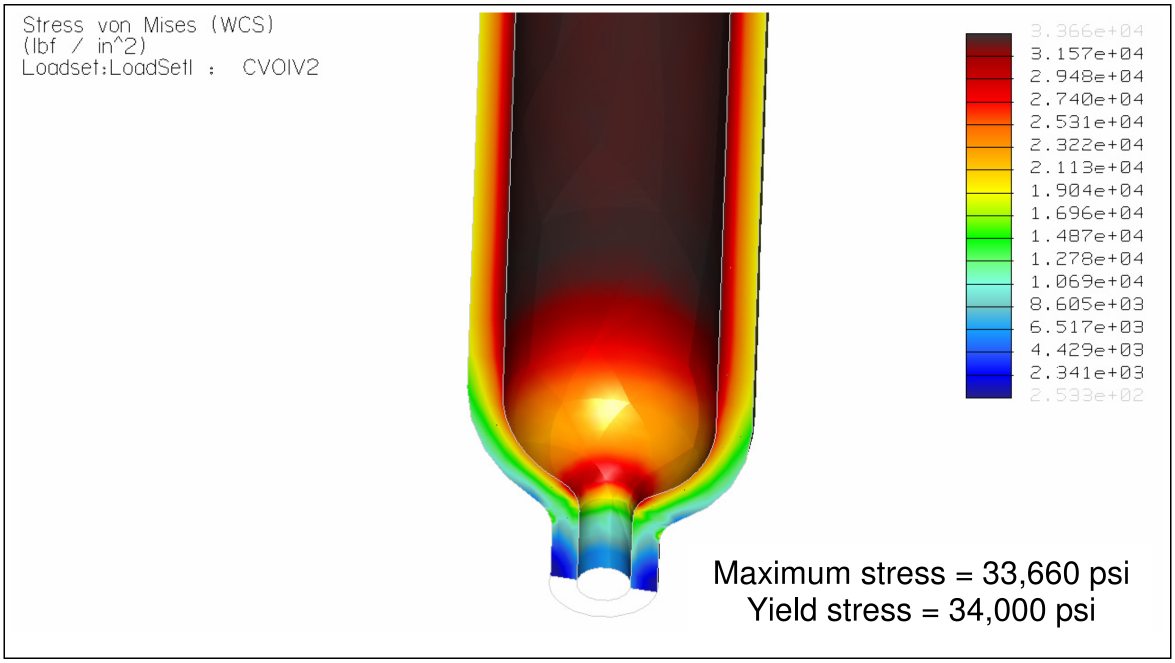


A-H Figure 2 – 316L SS Cylinder Fringe Plot – Transverse Cut

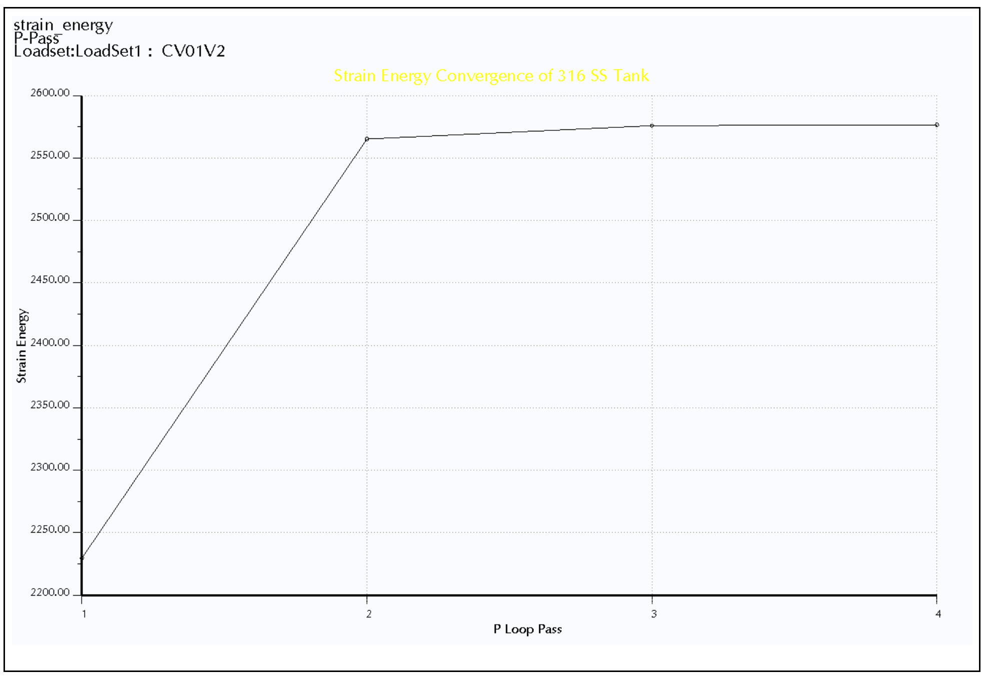
A-H Figure 3 – 316L SS Cylinder Fringe Plot – Transverse Cut Close-Up



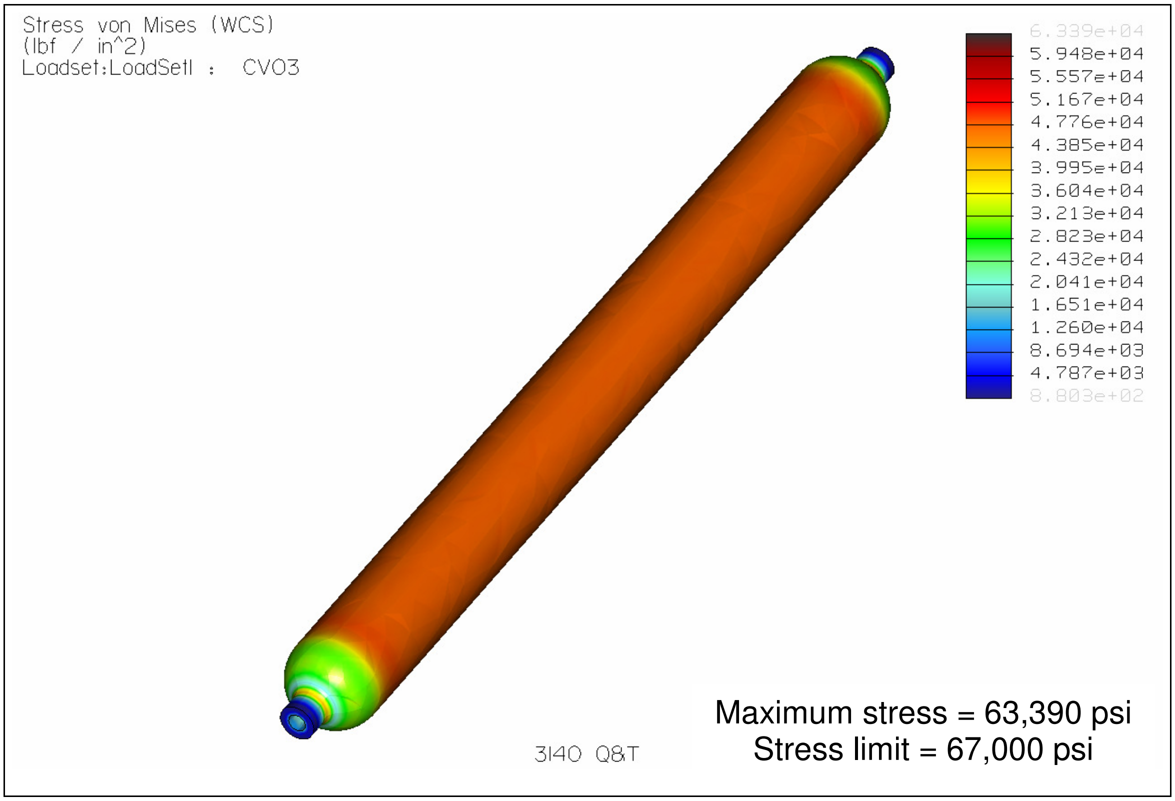
A-H Figure 4 – 316L SS Cylinder Fringe Plot – Longitudinal Cut



A-H Figure 5 – 316L SS Cylinder Fringe Plot – Longitudinal Cut Close-Up

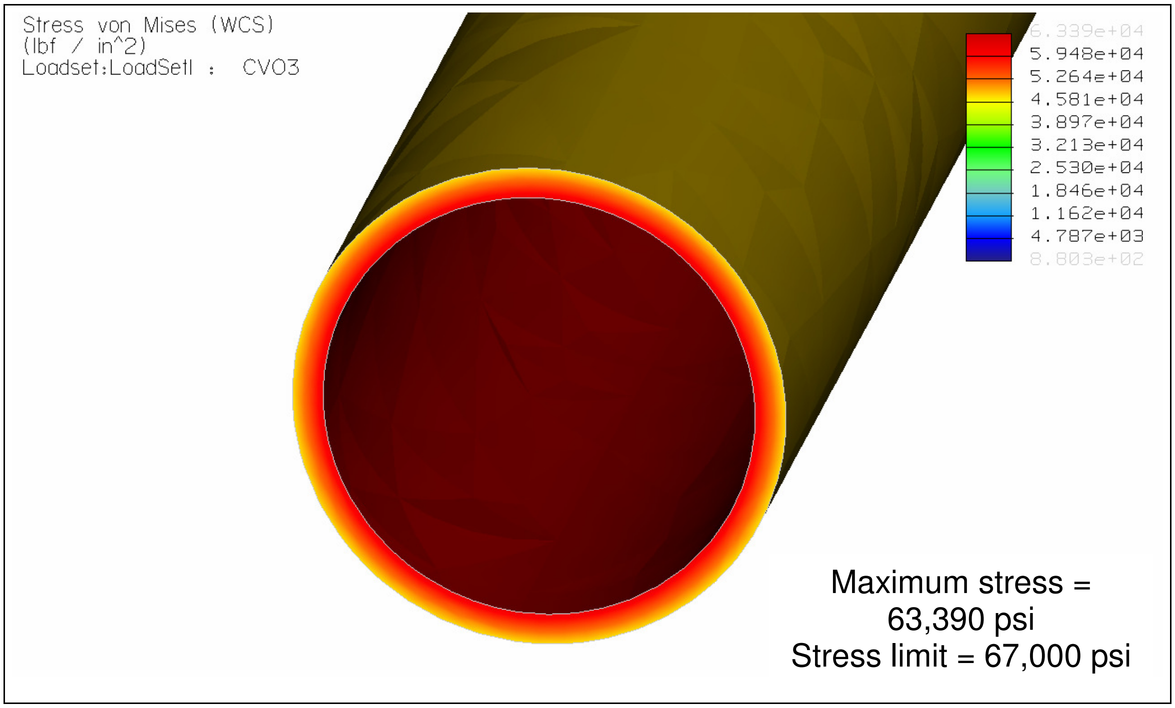


A-H Figure 6 – 316L SS Cylinder 5% Convergence Plot

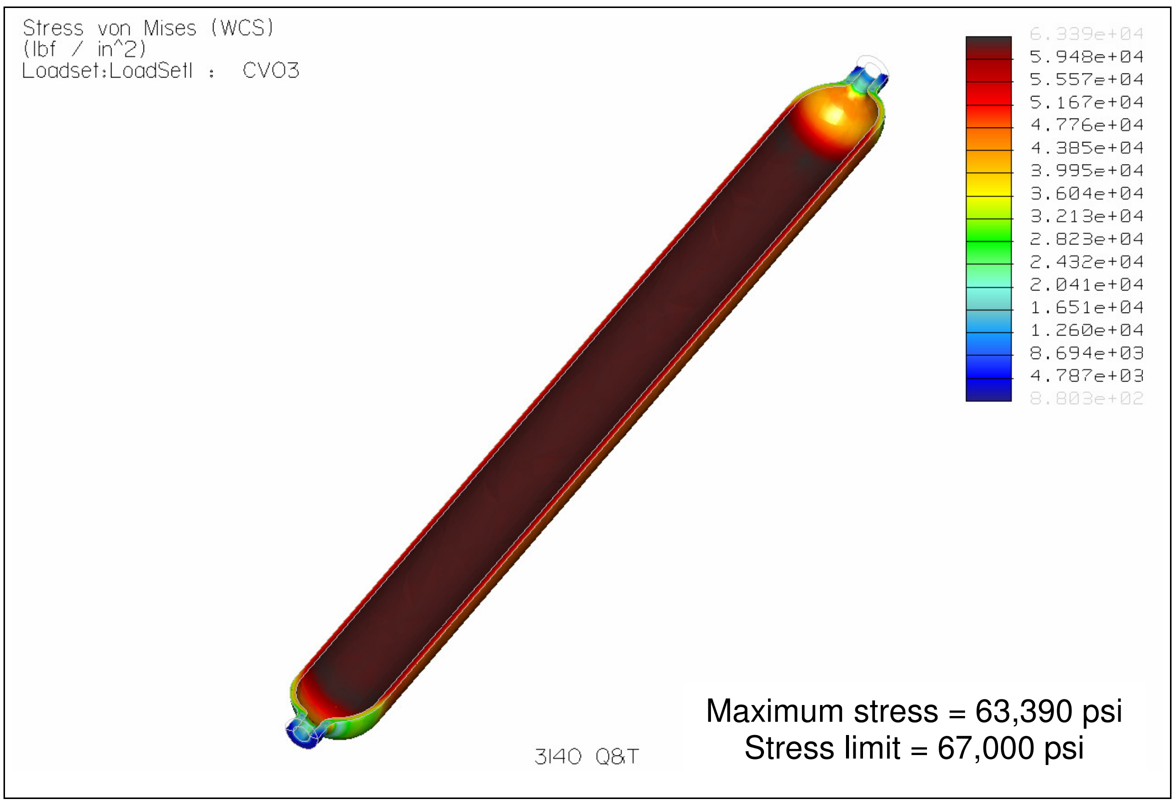


A-H Figure 7 – 4130 Q&T Cylinder Fringe Plot

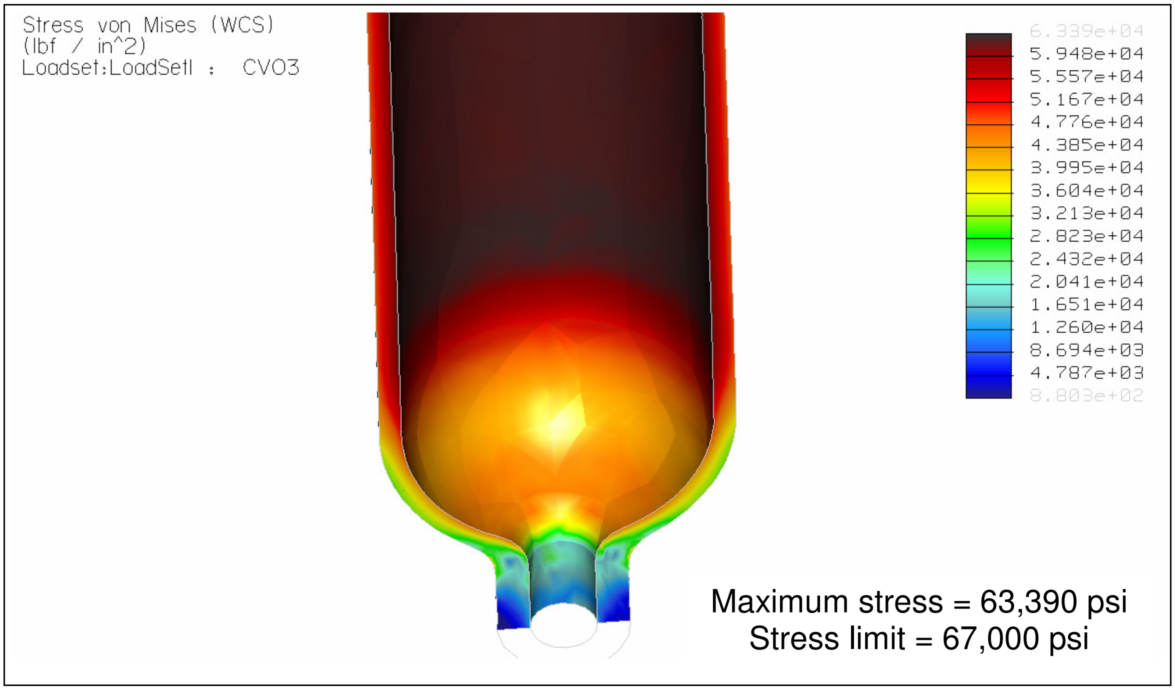
A-H Figure 8 – 4130 Q&T Cylinder Fringe Plot – Transverse Cut



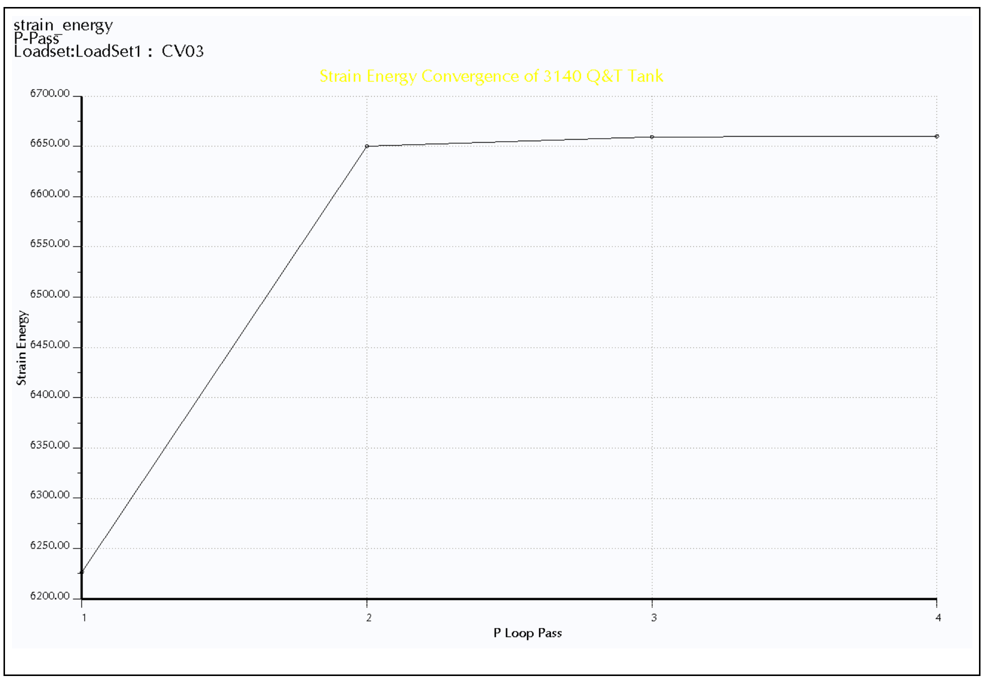
A-H Figure 9 – 4130 Q&T Cylinder Fringe Plot – Transverse Cut Close-Up



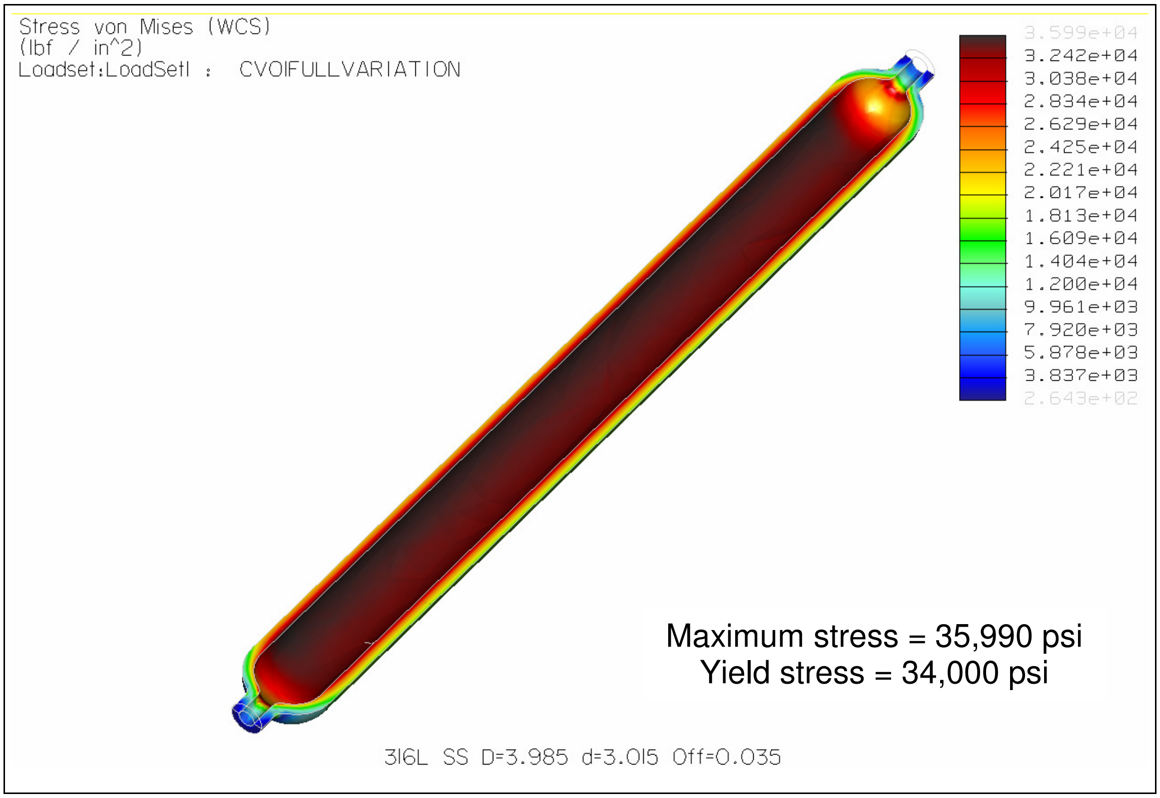
A-H Figure 10 – 4130 Q&T Cylinder Fringe Plot – Longitudinal Cut



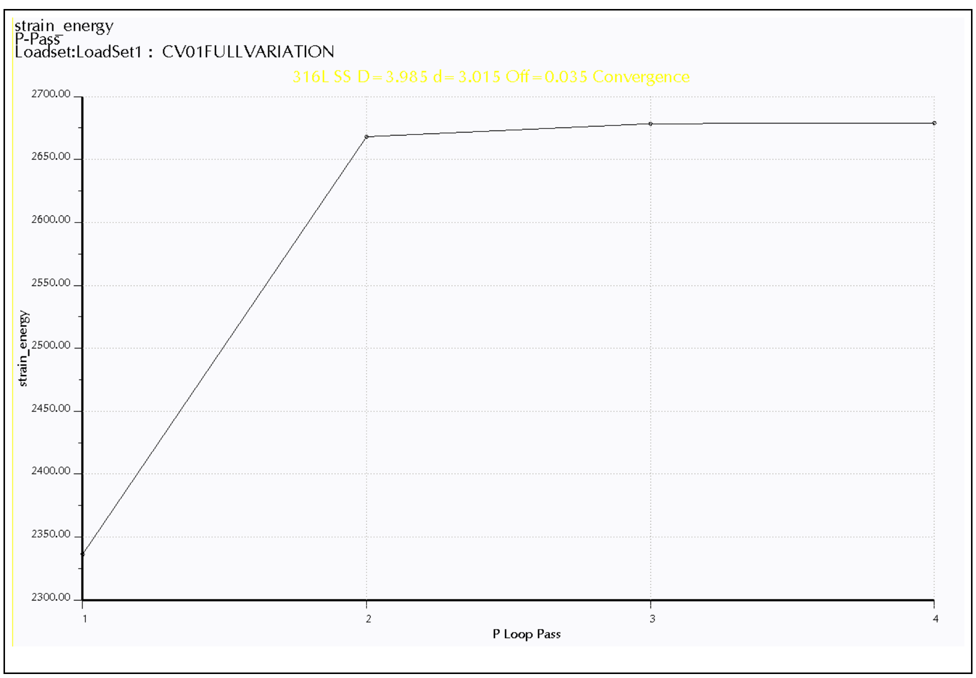
A-H Figure 11 – 4130 Q&T Cylinder Fringe Plot – Longitudinal Cut Close-Up



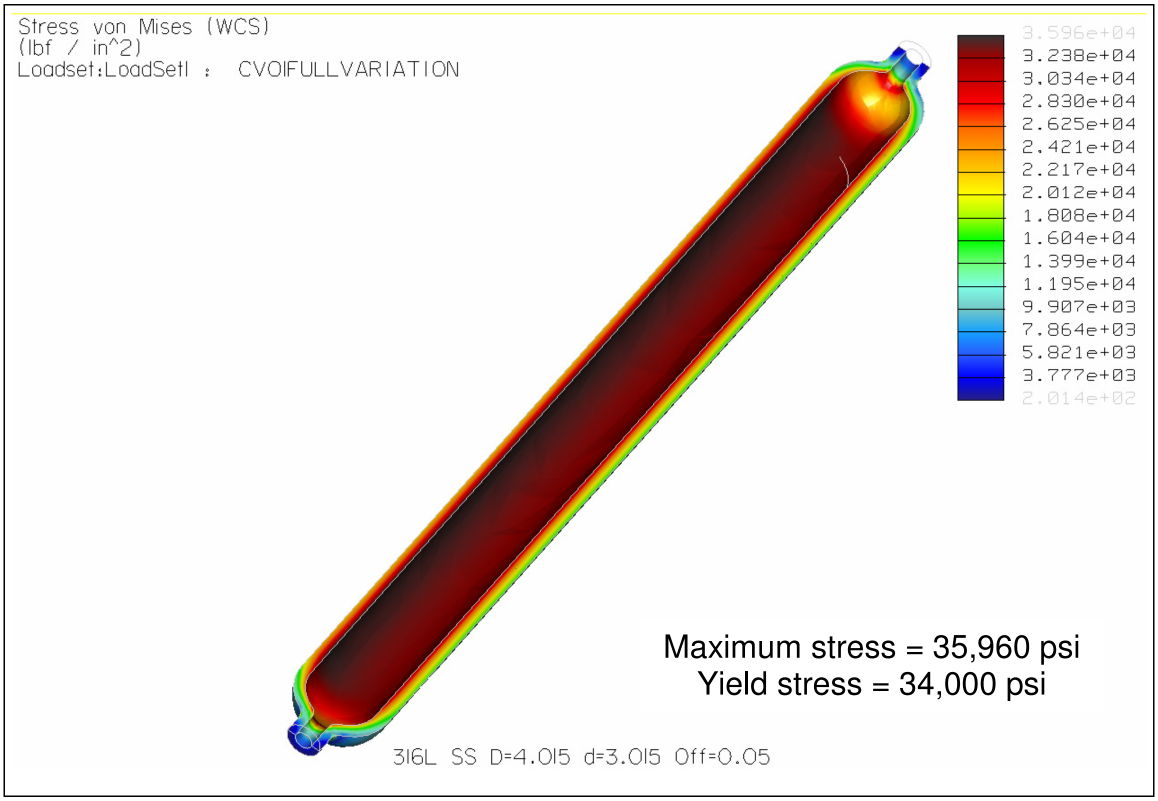
A-H Figure 12 – 4130 Q&T Cylinder 5% Convergence Plot



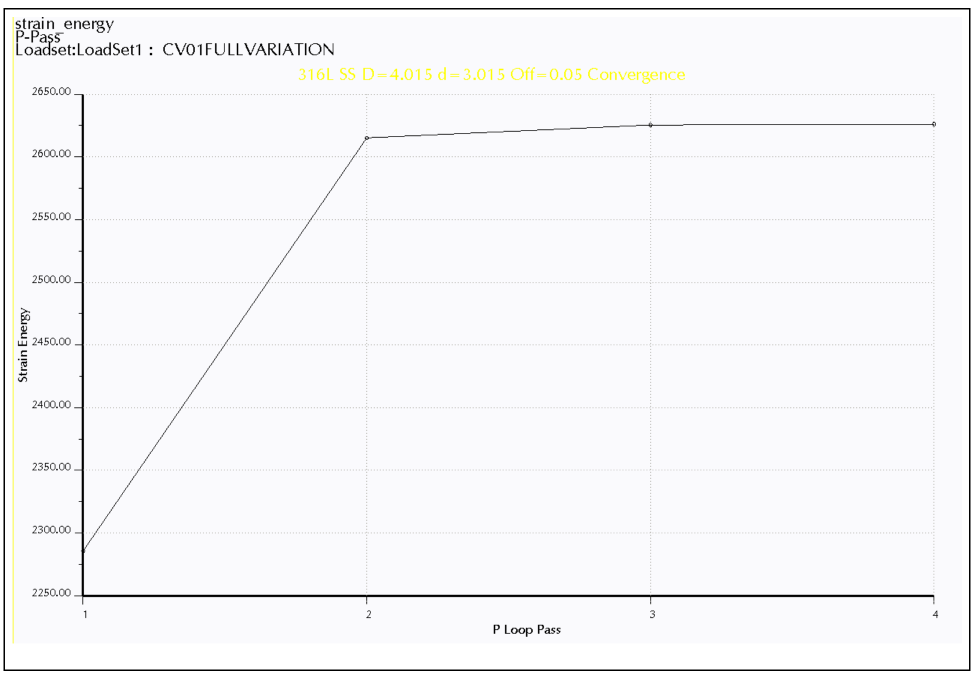
A-H Figure 13 – 316L SS Cylinder Fringe Plot – Case 1 Variation



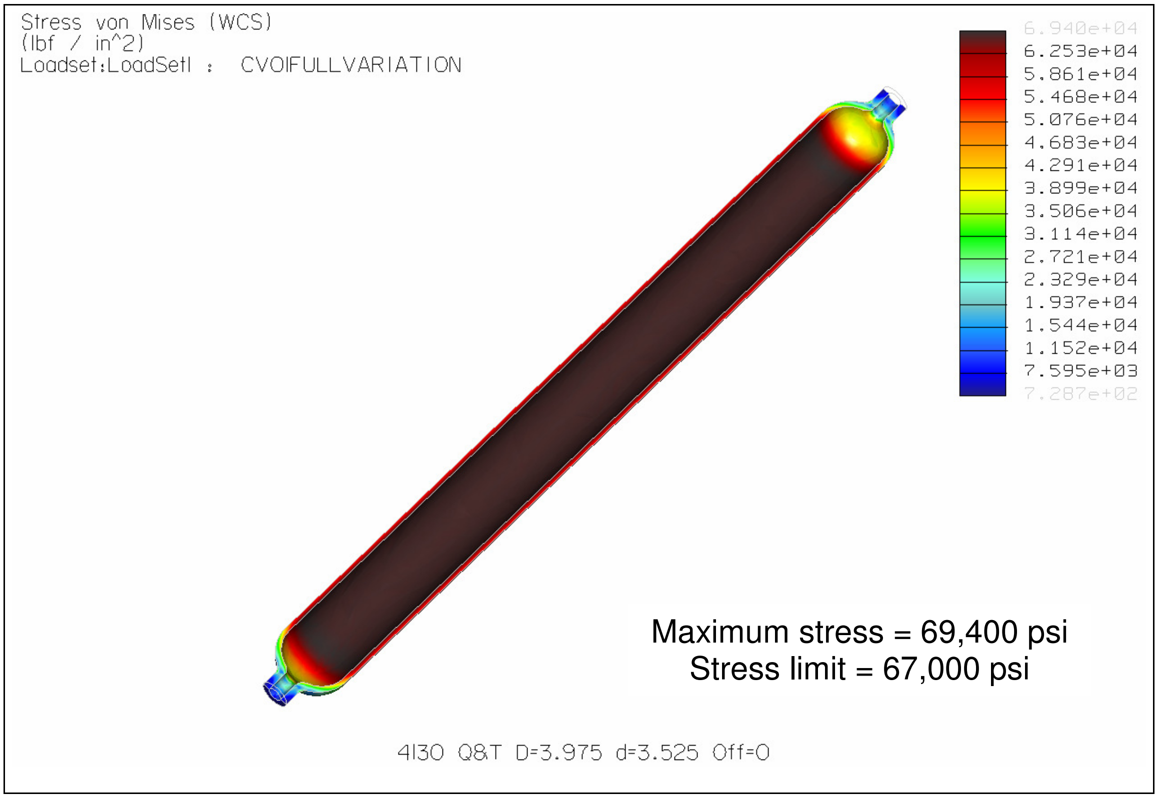
A-H Figure 14 – 316L SS Cylinder 5% Convergence – Case 1



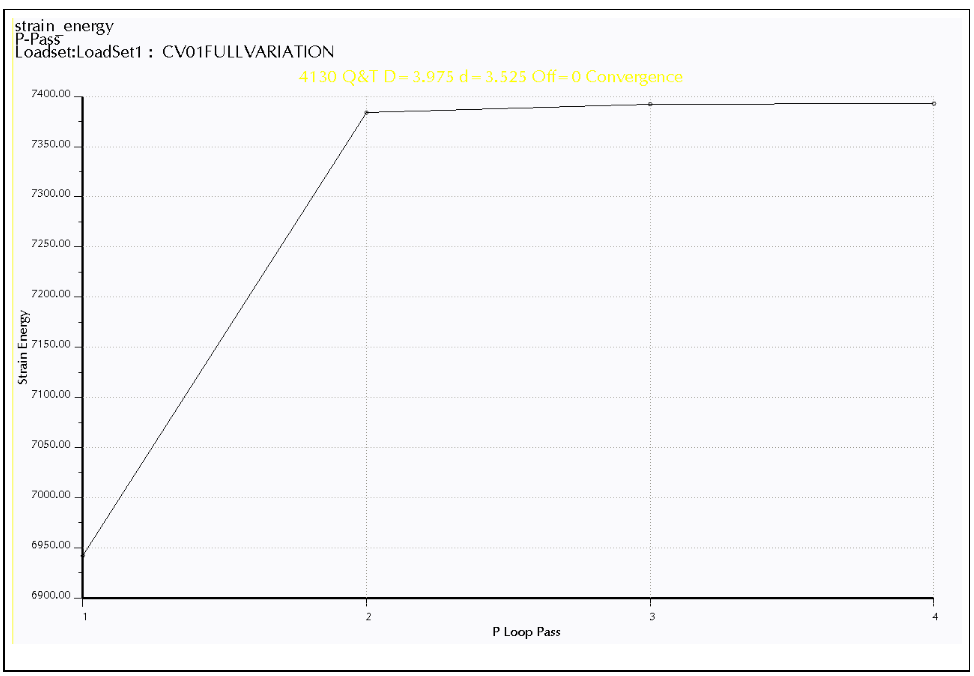
A-H Figure 15 – 316L SS Cylinder Fringe Plot – Case 3 Variation



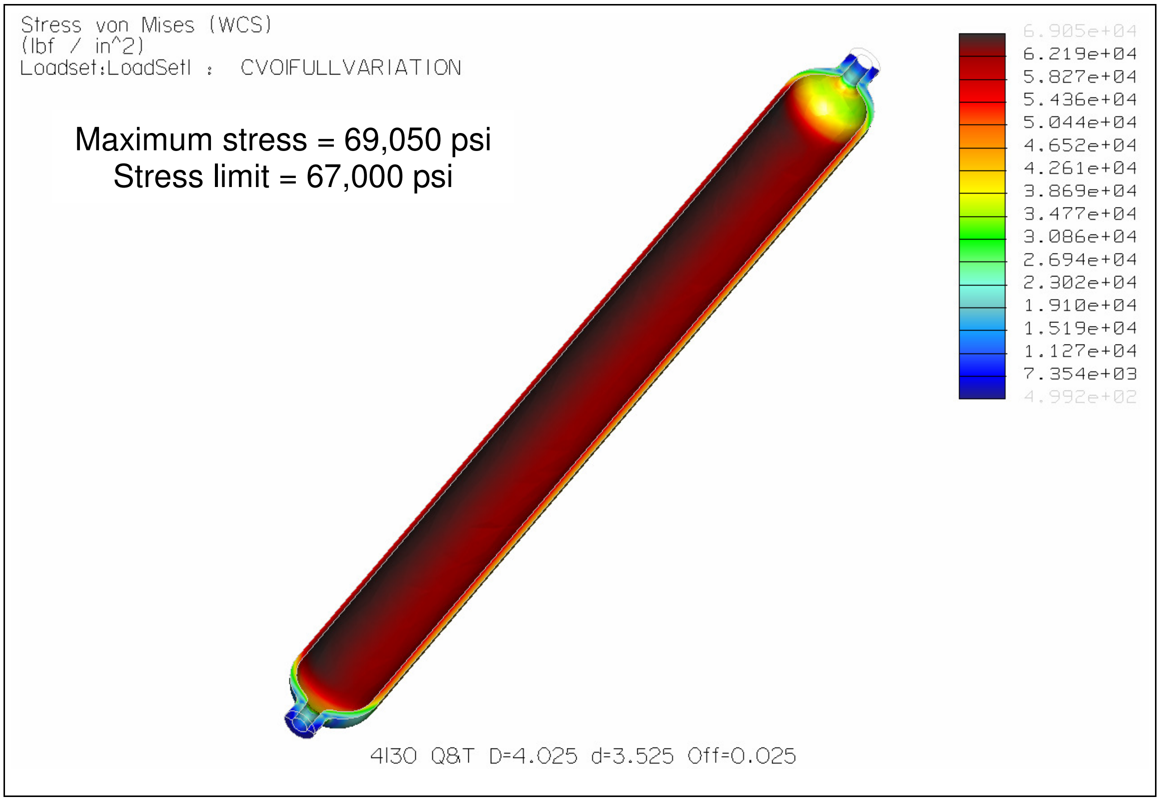
A-H Figure 16 – 316L SS Cylinder 5% Convergence – Case 3



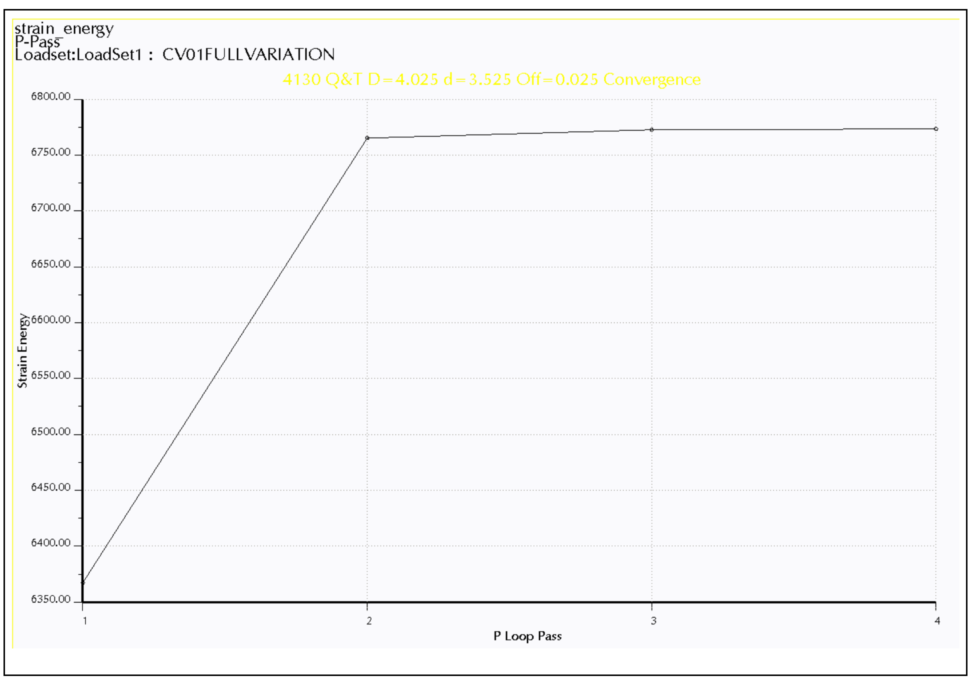
A-H Figure 17 – 4130 Q&T Cylinder Fringe Plot – Case 1 Variation



A-H Figure 18 – 4130 Q&T Cylinder 5% Convergence – Case 1

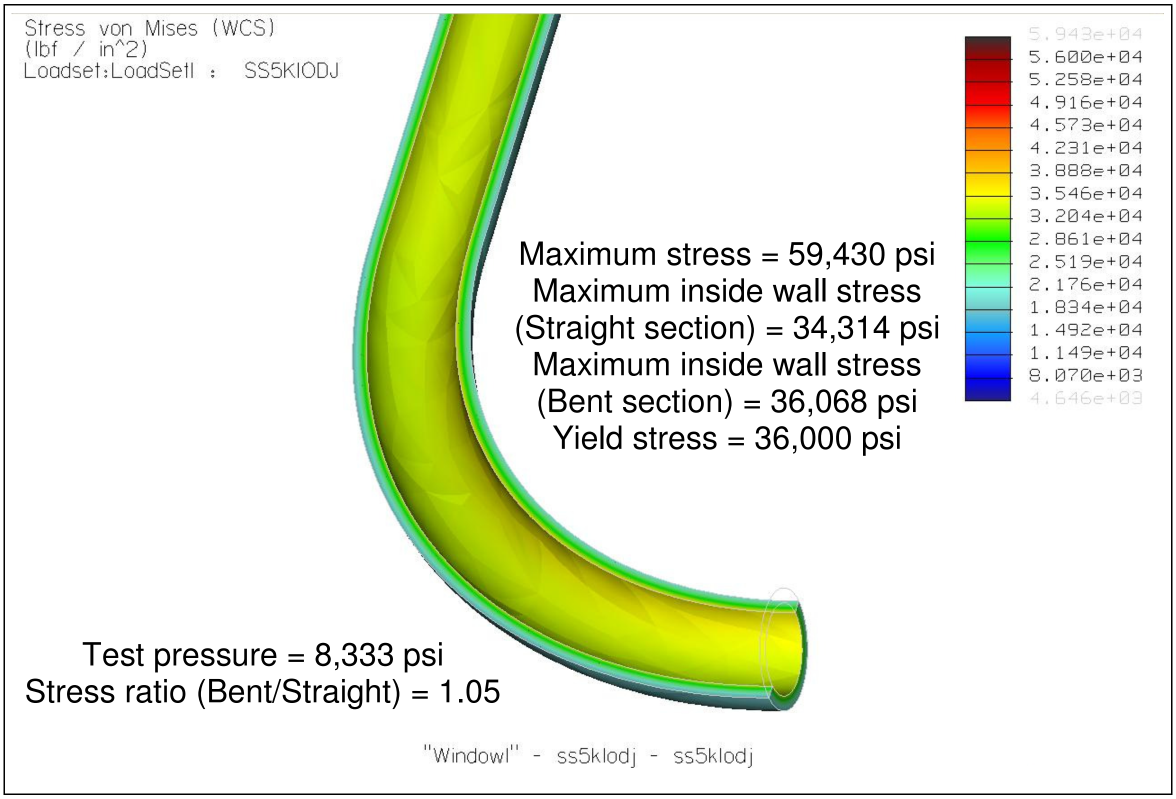


A-H Figure 19 – 4130 Q&T Cylinder Fringe Plot – Case 3 Variation

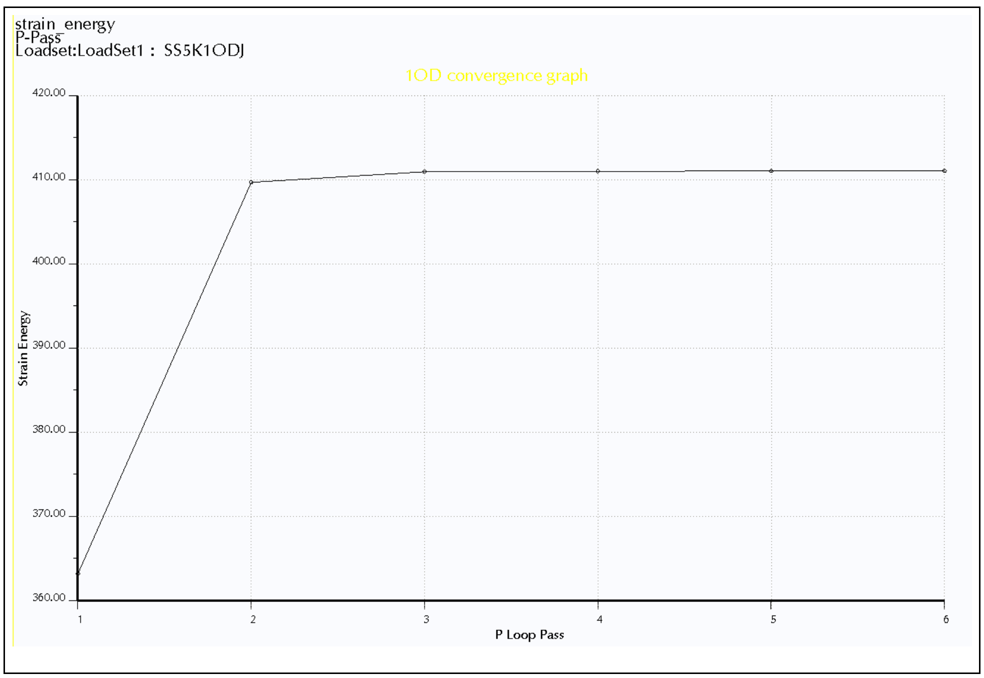


A-H Figure 20 – 316L SS Cylinder 5% Convergence – Case 3

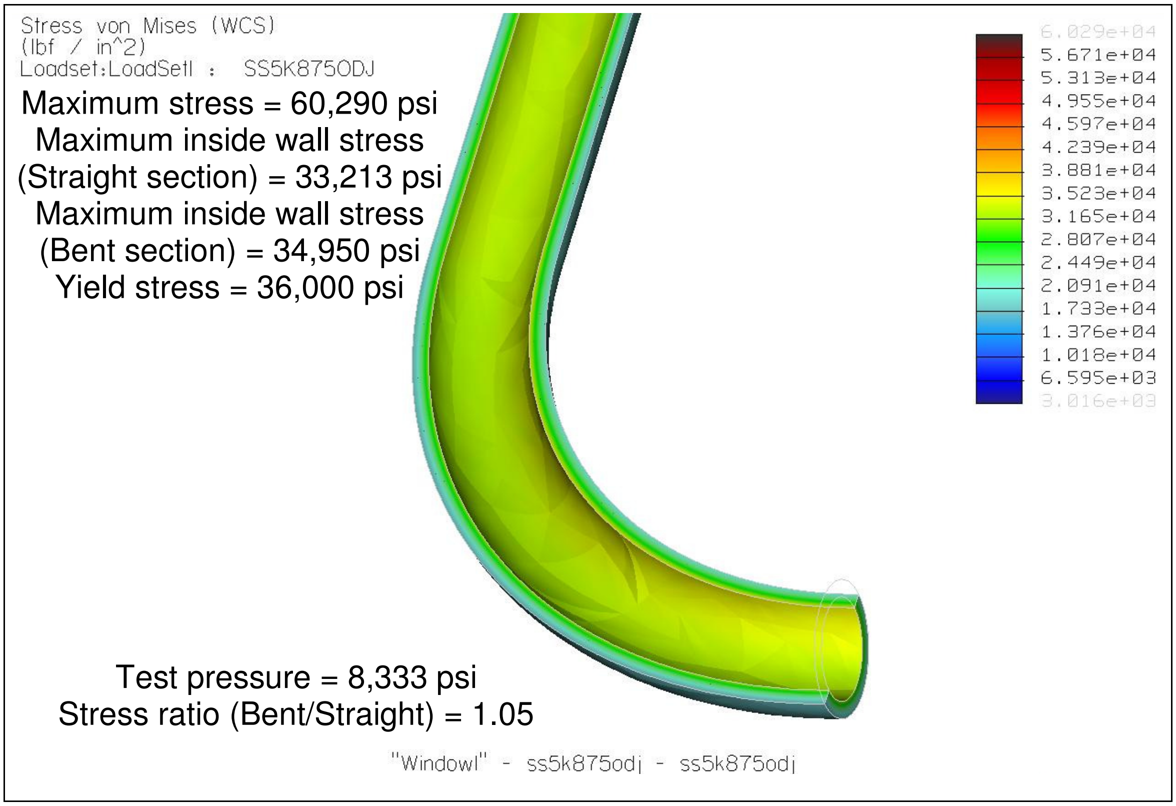
APPENDIX I: FEA RESULTS
(TUBING COILS)



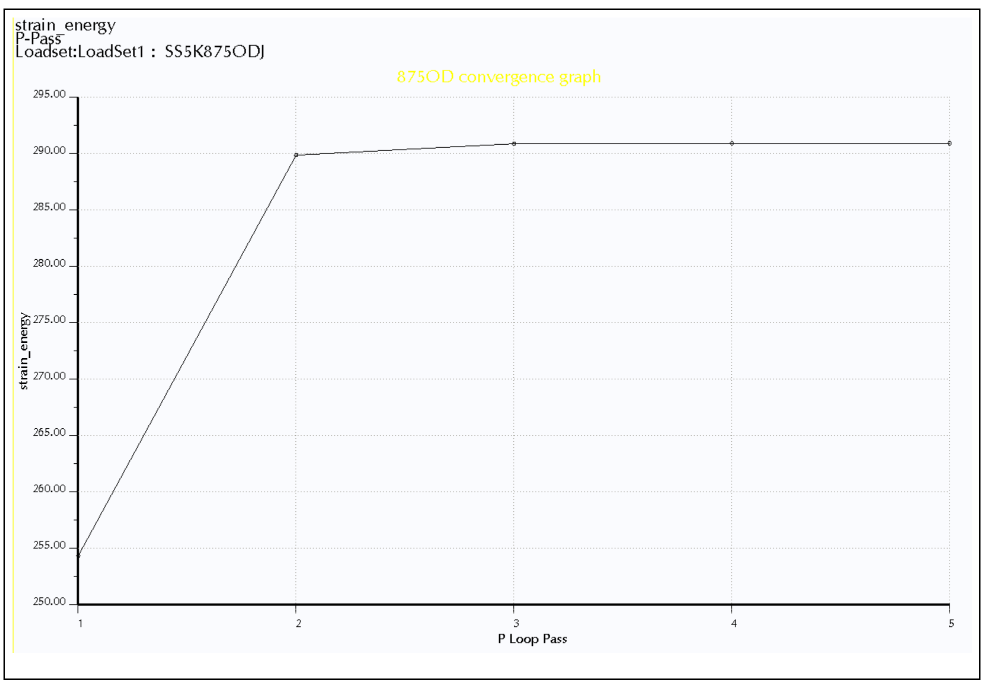
A-I Figure 1 – 1" OD Tube, 8,333 psi Fringe Plot



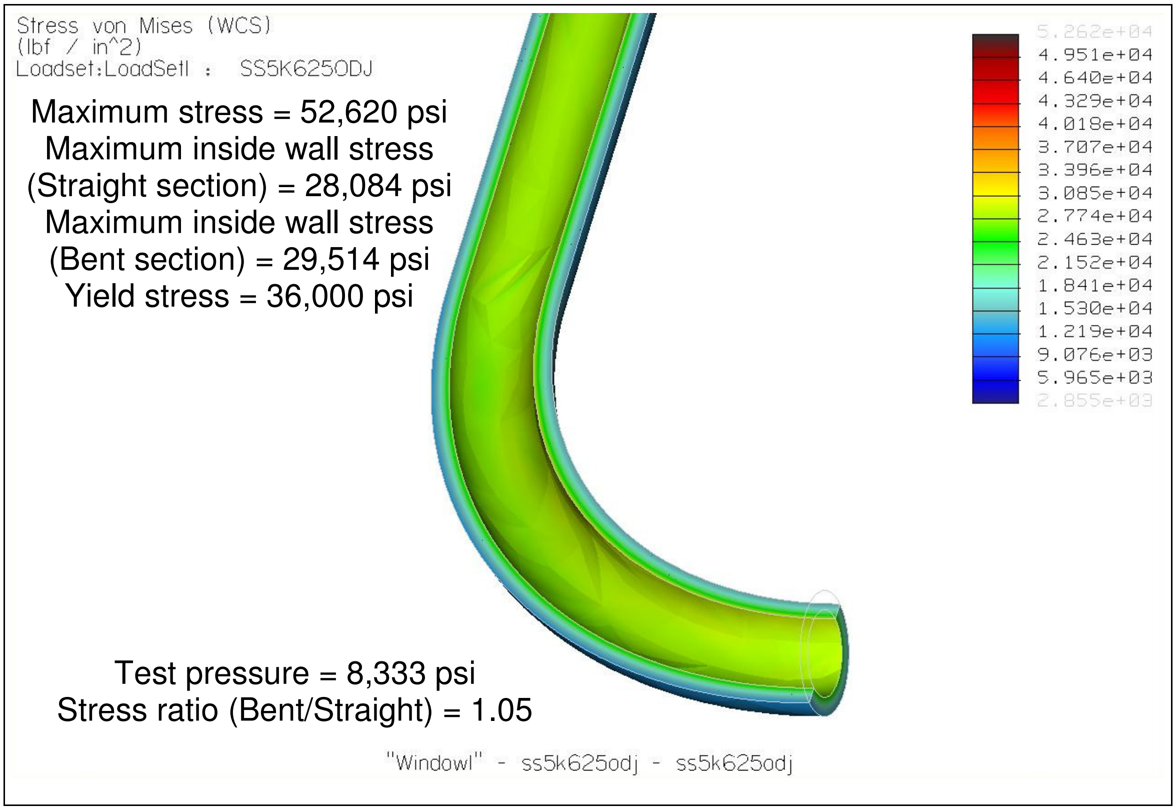
A-I Figure 2 – 1" OD Tube, 8,333 psi 5% Convergence Plot



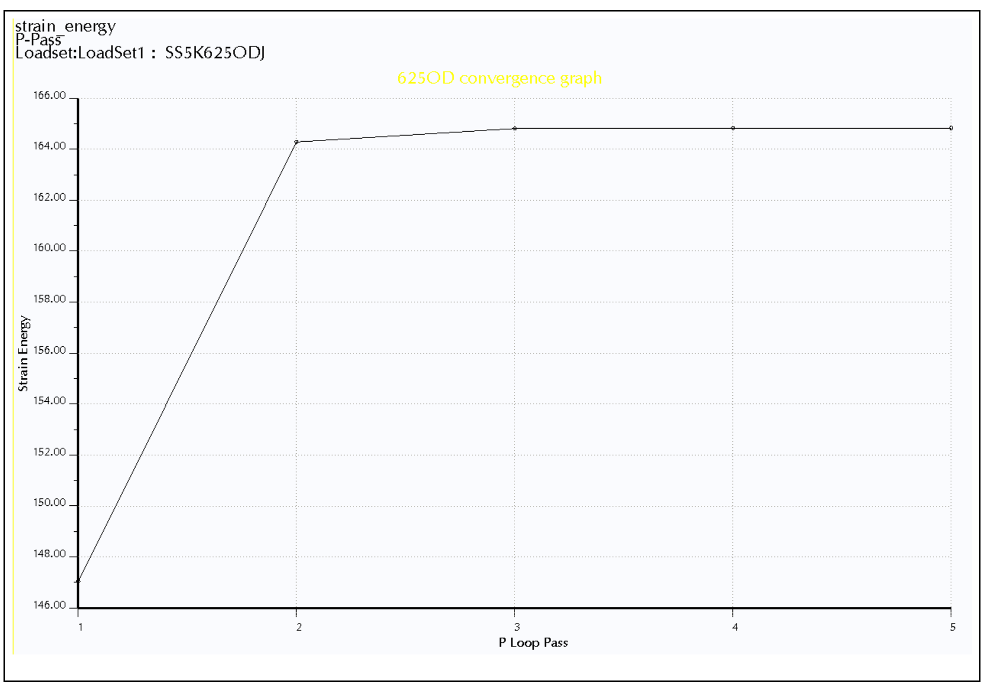
A-I Figure 3 – 0.875" OD Tube, 8,333 psi Fringe Plot



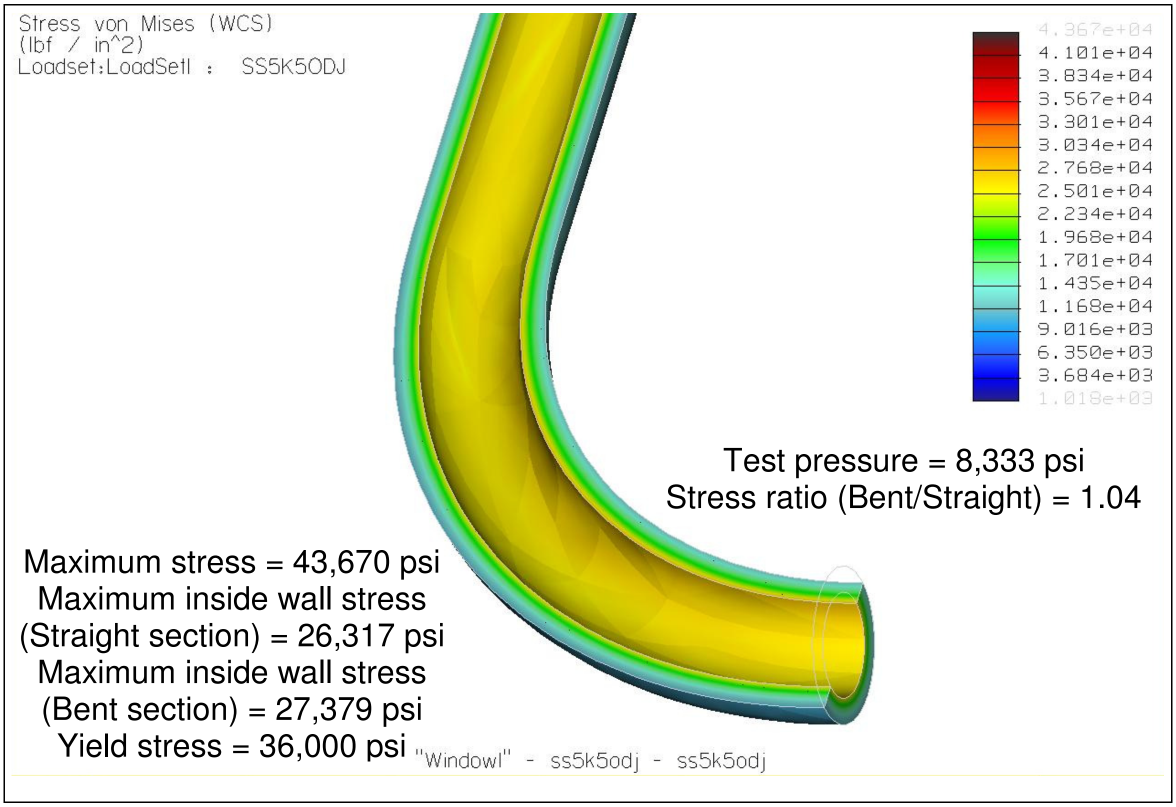
A-I Figure 4 – 0.875" OD Tube, 8,333 psi 5% Convergence Plot



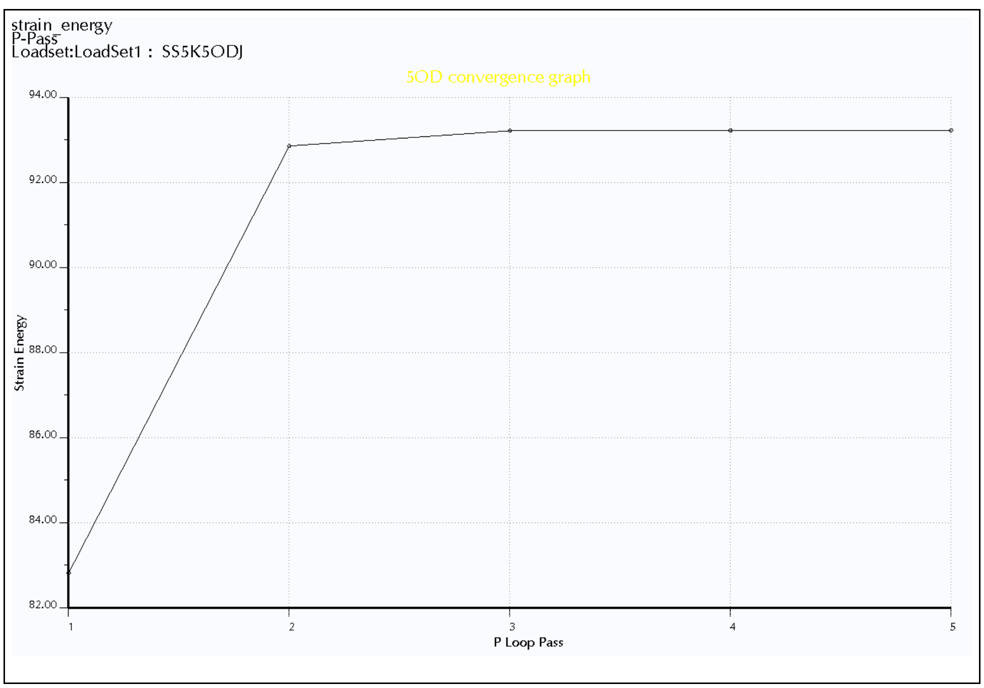
A-I Figure 5 – 0.625" OD Tube, 8,333 psi Fringe Plot



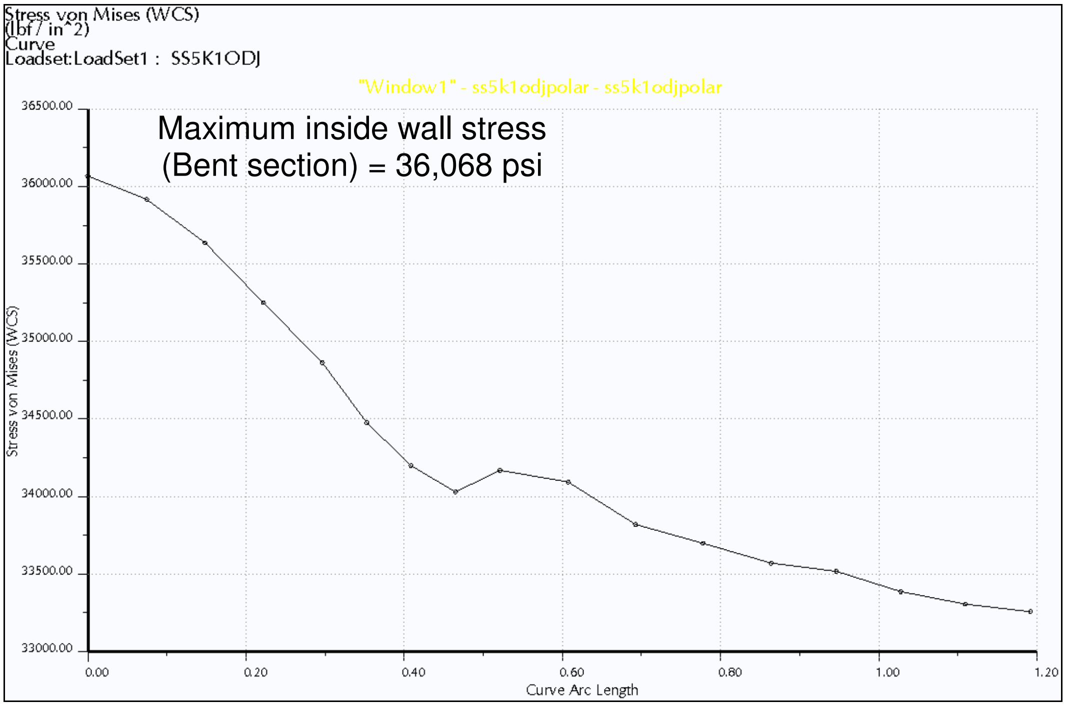
A-I Figure 6 – 0.625" OD Tube, 8,333 psi 5% Convergence Plot



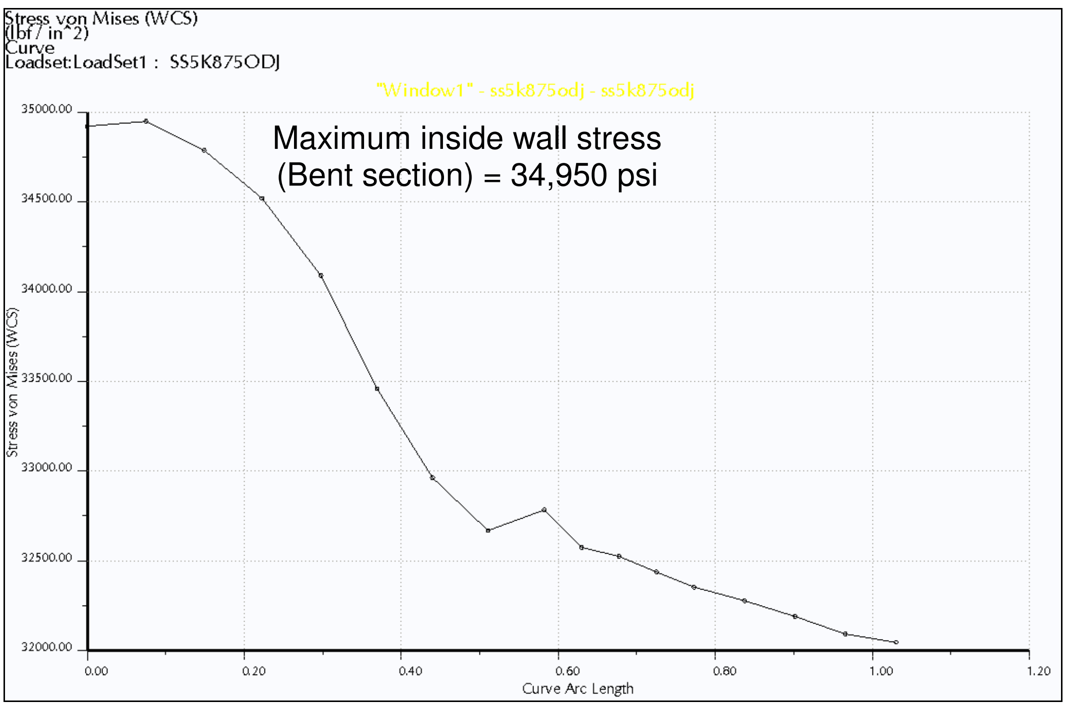
A-I Figure 7 – 0.5" OD Tube, 8,333 psi Fringe Plot



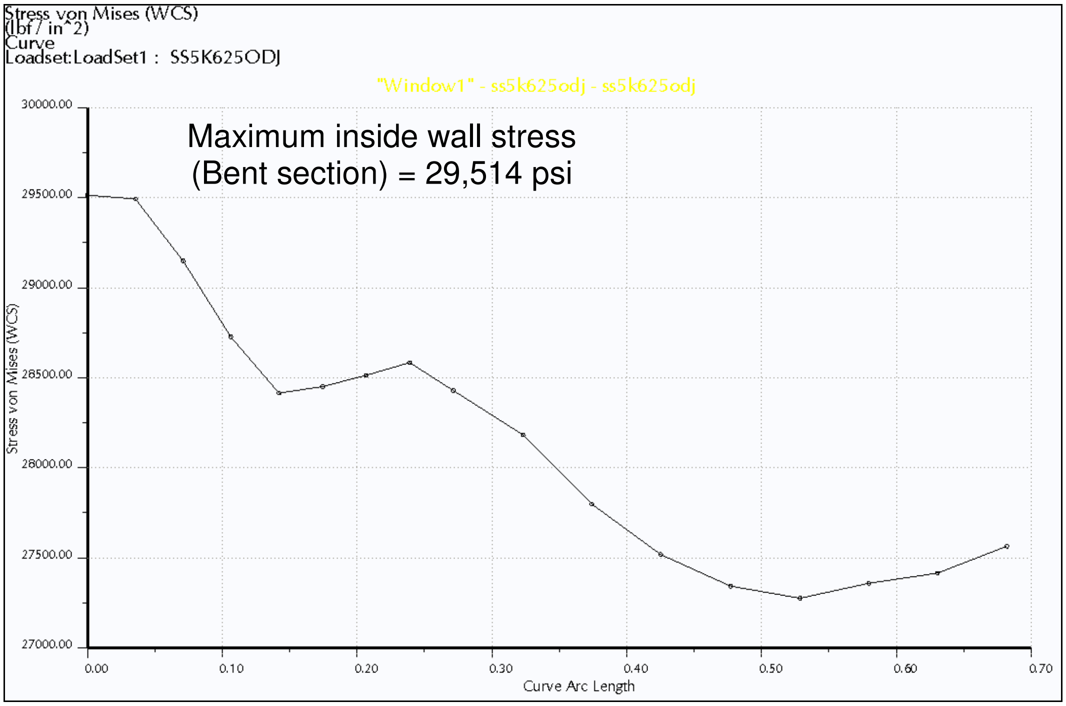
A-I Figure 8 – 0.5" OD Tube, 8,333 psi 5% Convergence Plot



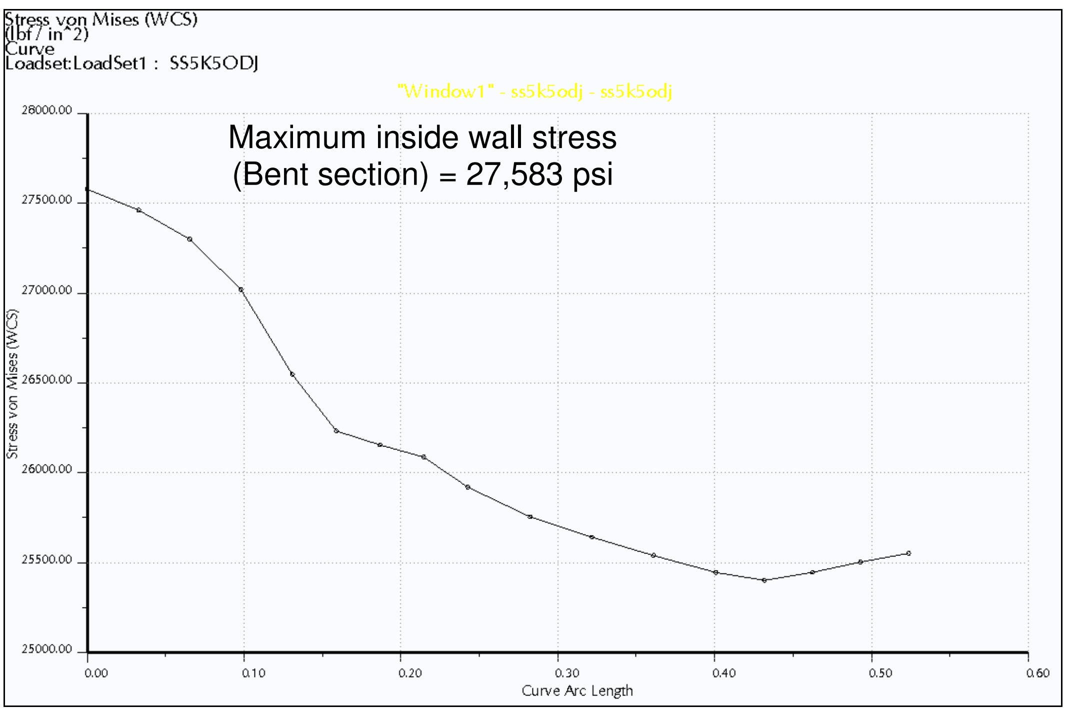
A-I Figure 9 – Stress vs Arc Length (Inside Rim of Bent End, 1" OD)



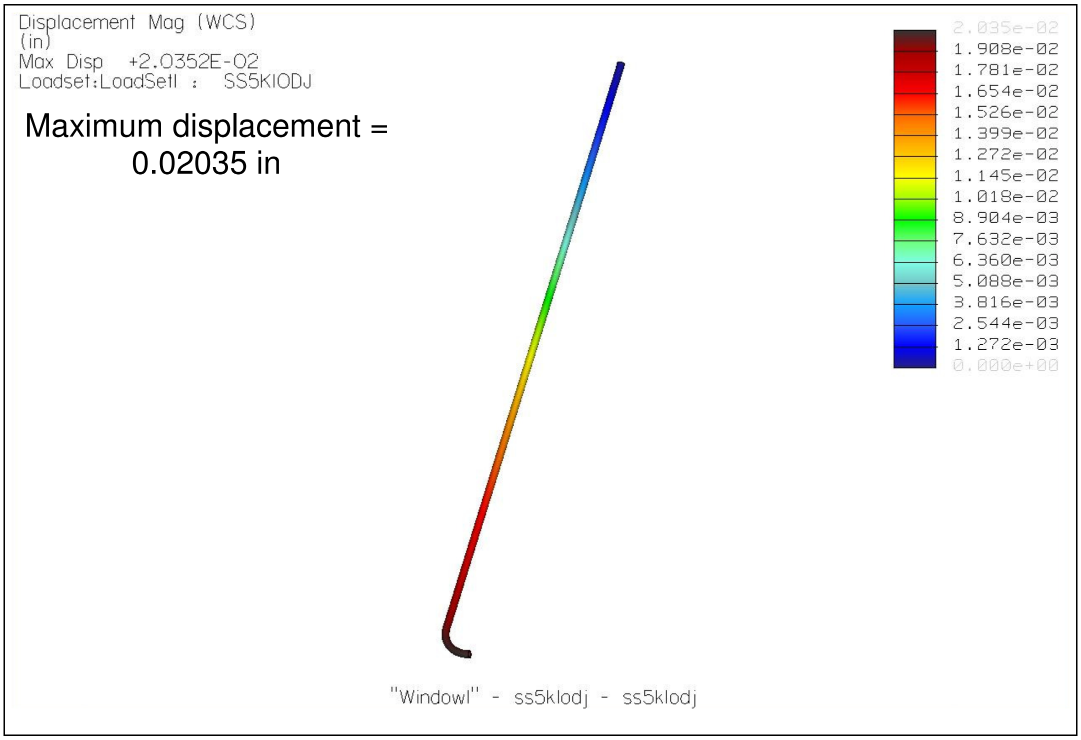
A-I Figure 10 – Stress vs Arc Length (Inside Rim of Bent End, 0.875" OD)



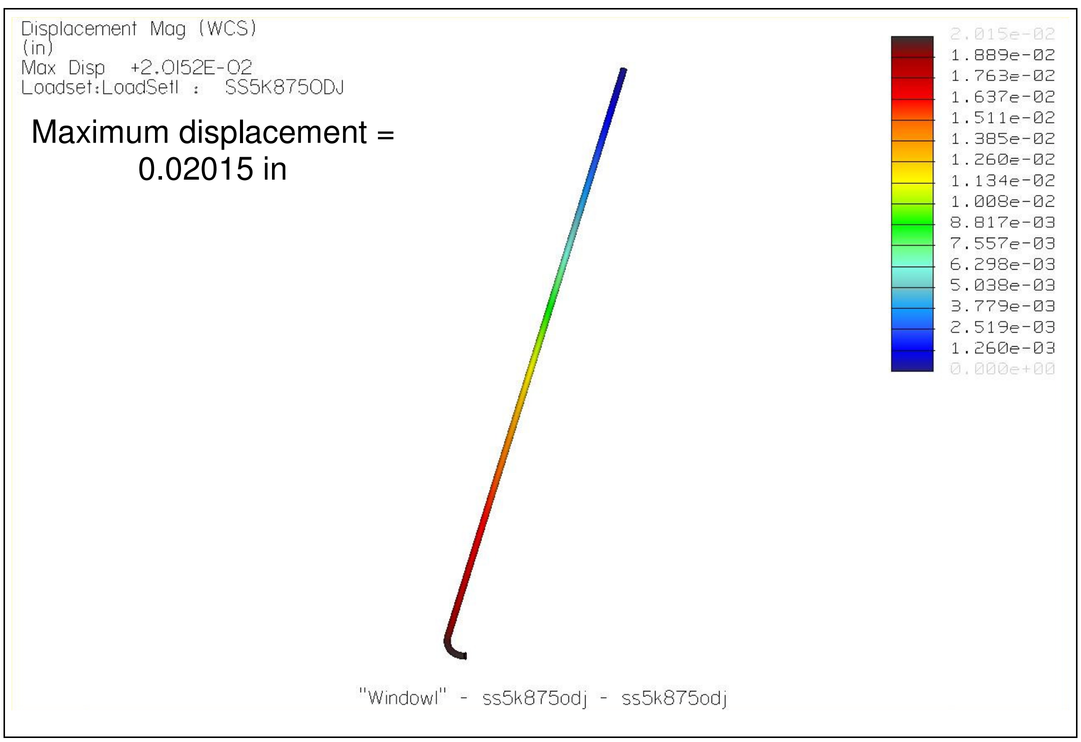
A-I Figure 11 – Stress vs Arc Length (Inside Rim of Bent End, 0.625" OD)



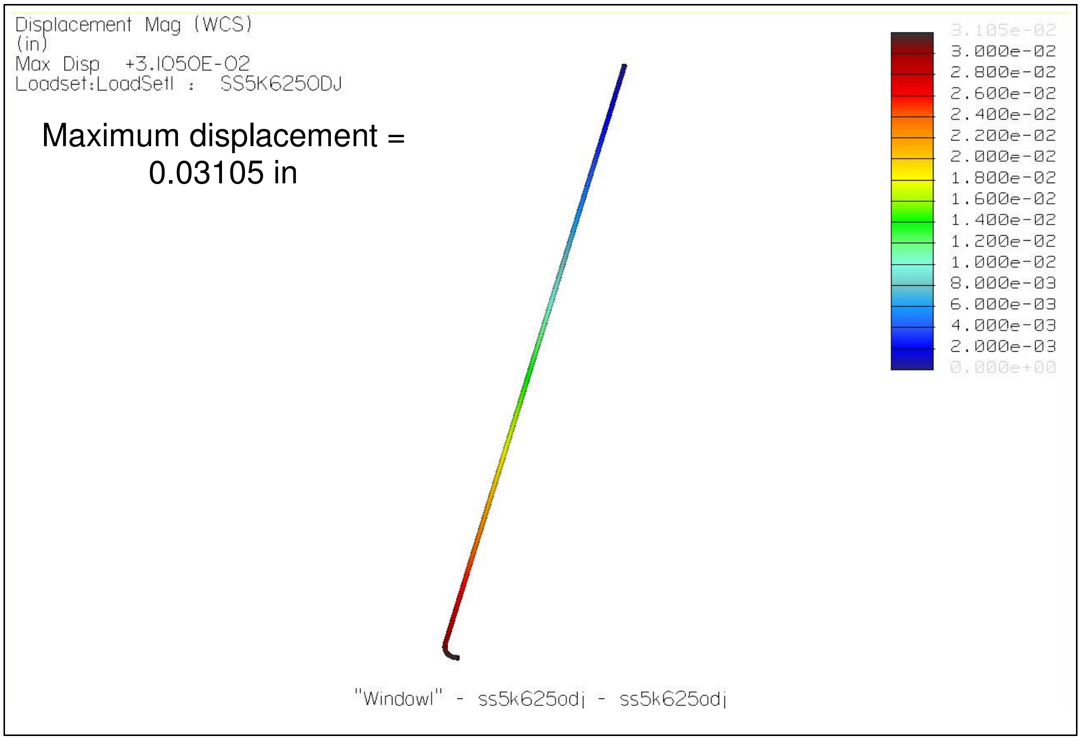
A-I Figure 12 – Stress vs Arc Length (Inside Rim of Bent End, 0.5" OD)



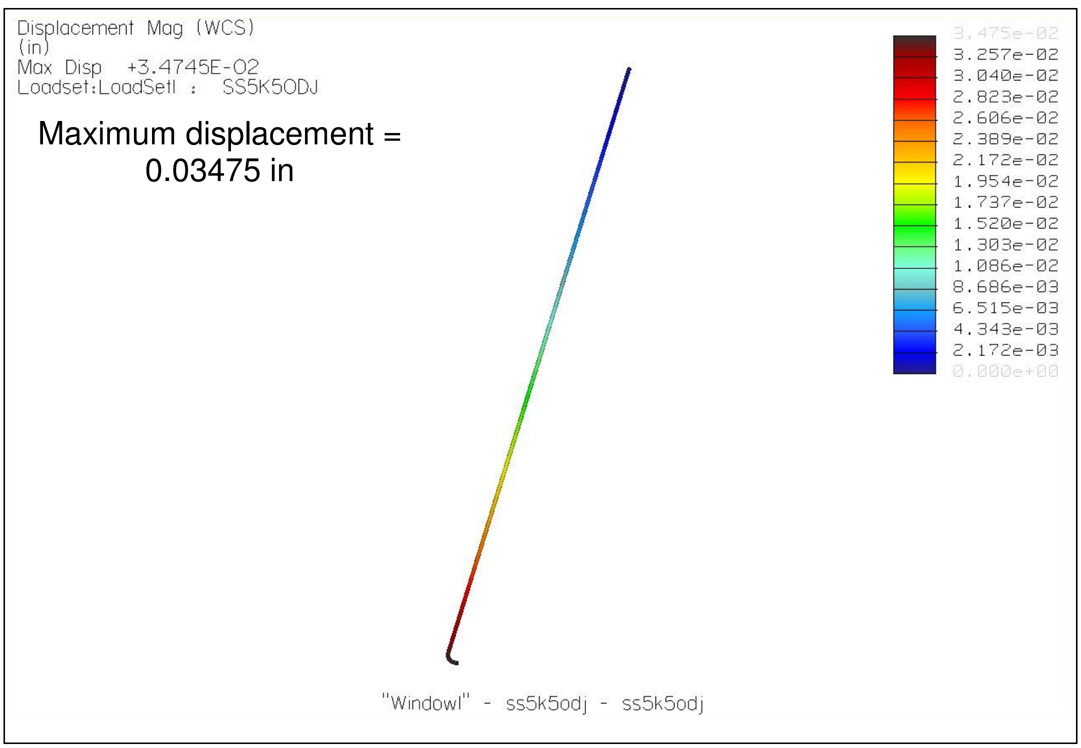
A-I Figure 13 – 1" OD Tube Displacement



A-I Figure 14 – 0.875" OD Tube Displacement

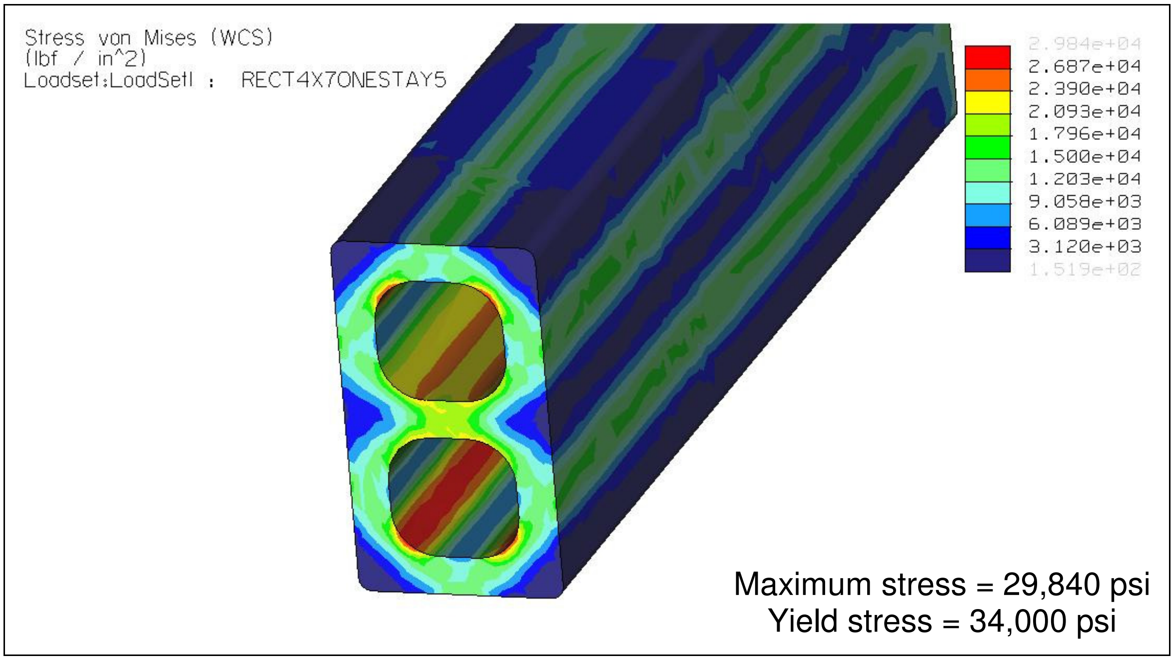


A-I Figure 15 – 0.625" OD Tube Displacement

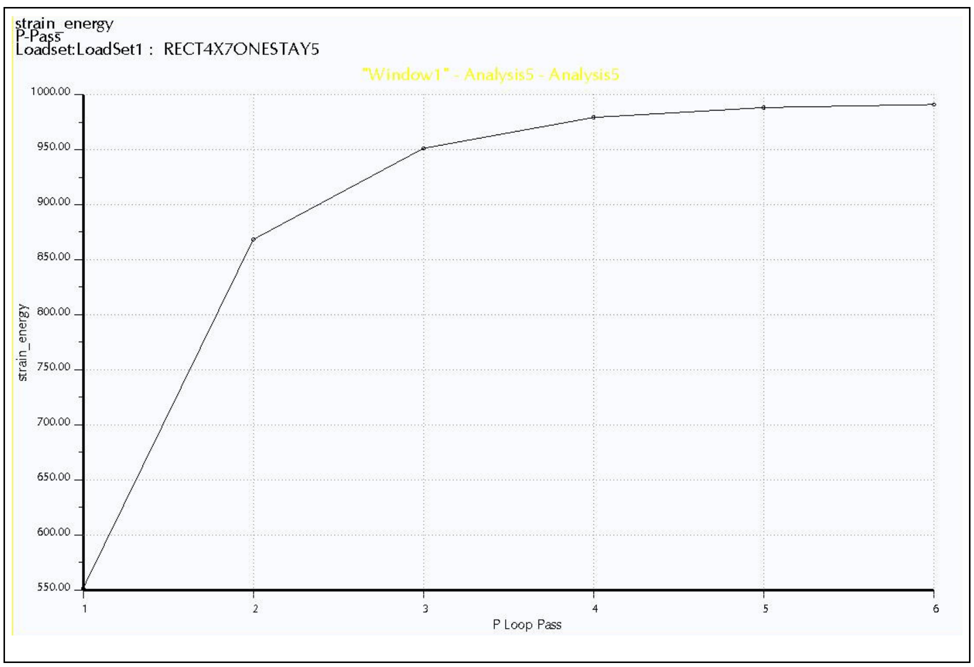


A-I Figure 16 – 0.5" OD Tube Displacement

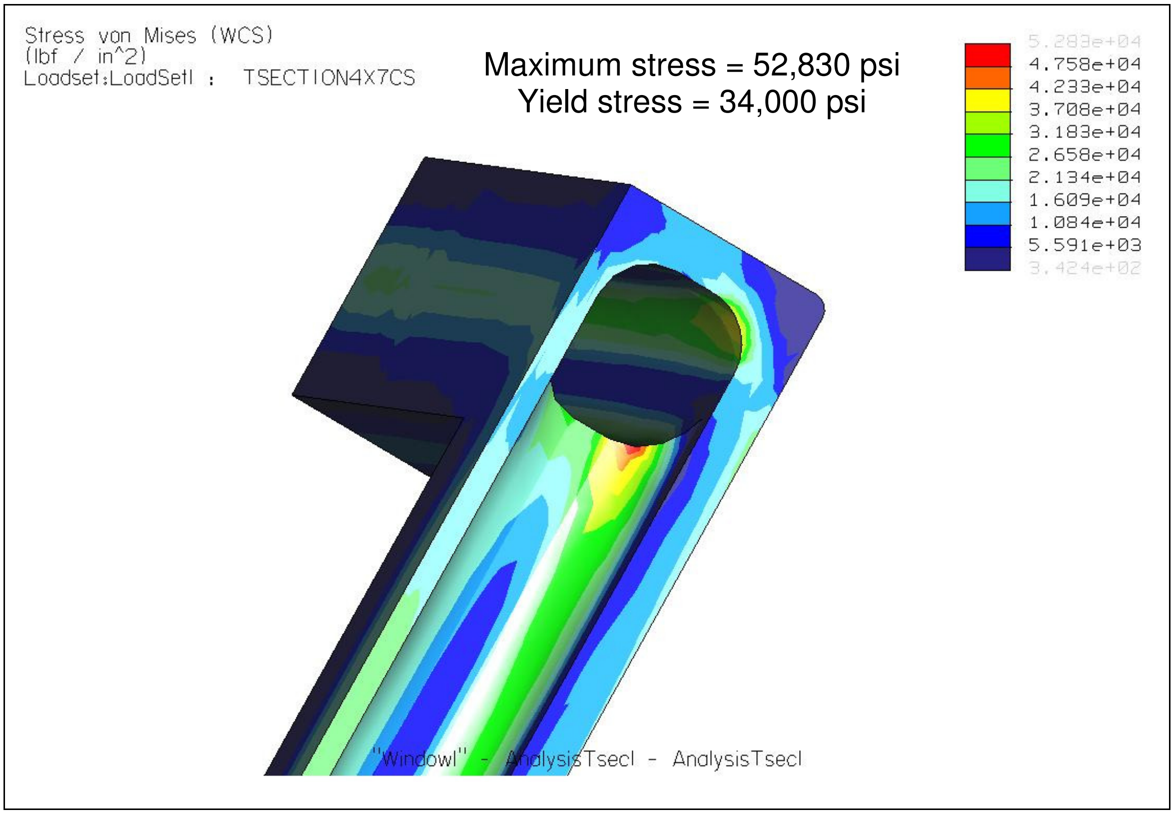
APPENDIX J: FEA RESULTS
(RECTANGULAR TANK)



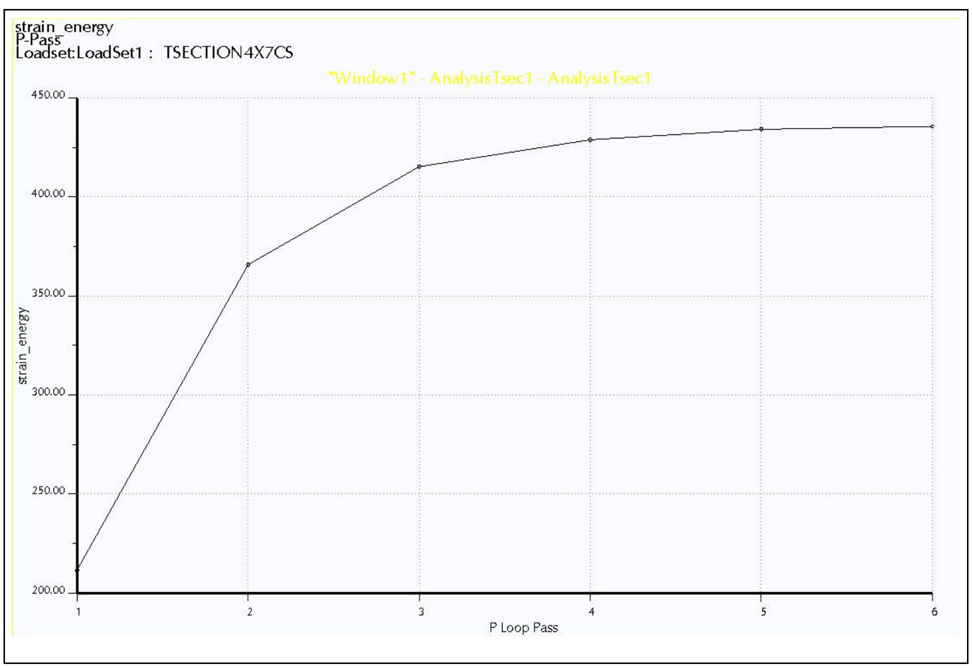
A-J Figure 1 – 316L SS Single Member Fringe Plot



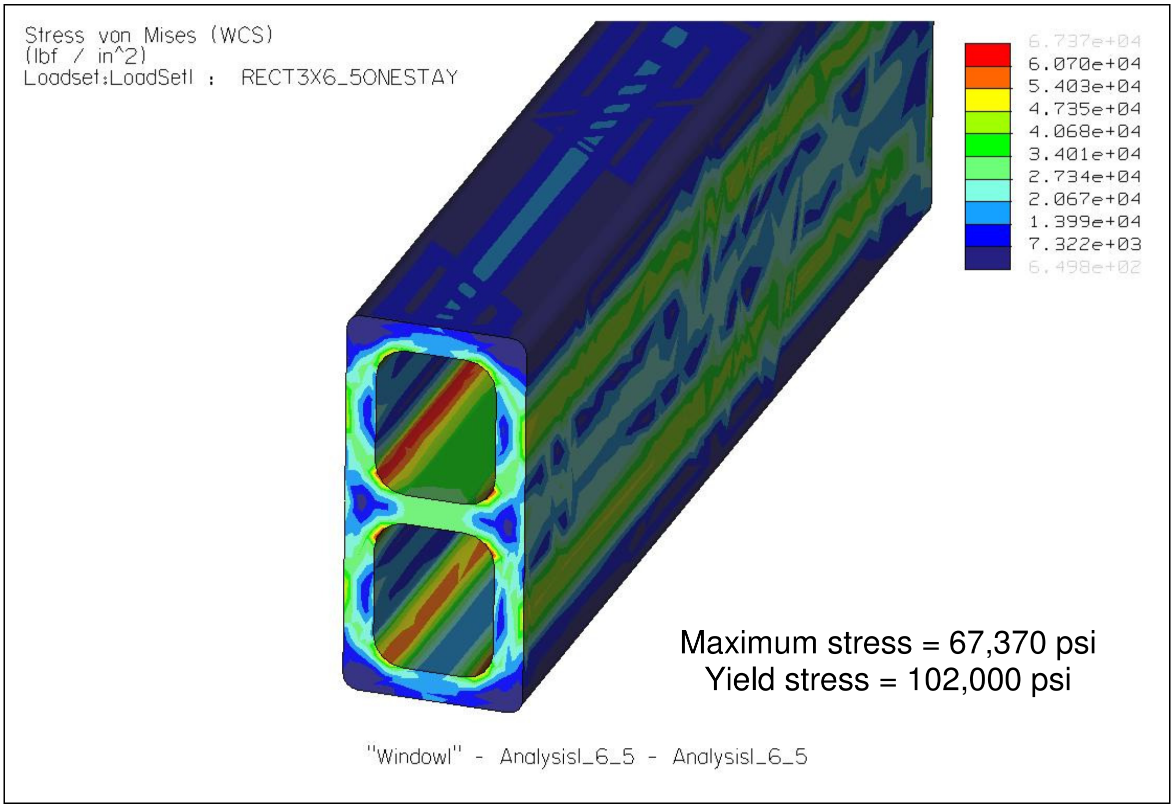
A-J Figure 2 – 316L SS Single Member 10% Convergence Plot



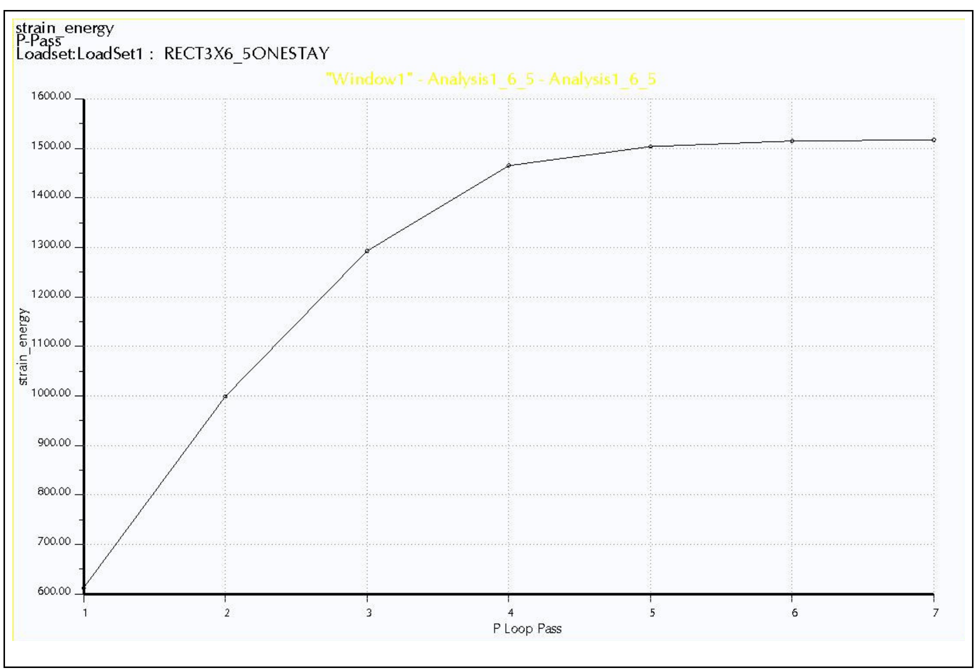
A-J Figure 3 – 316L SS T-Junction Fringe Plot



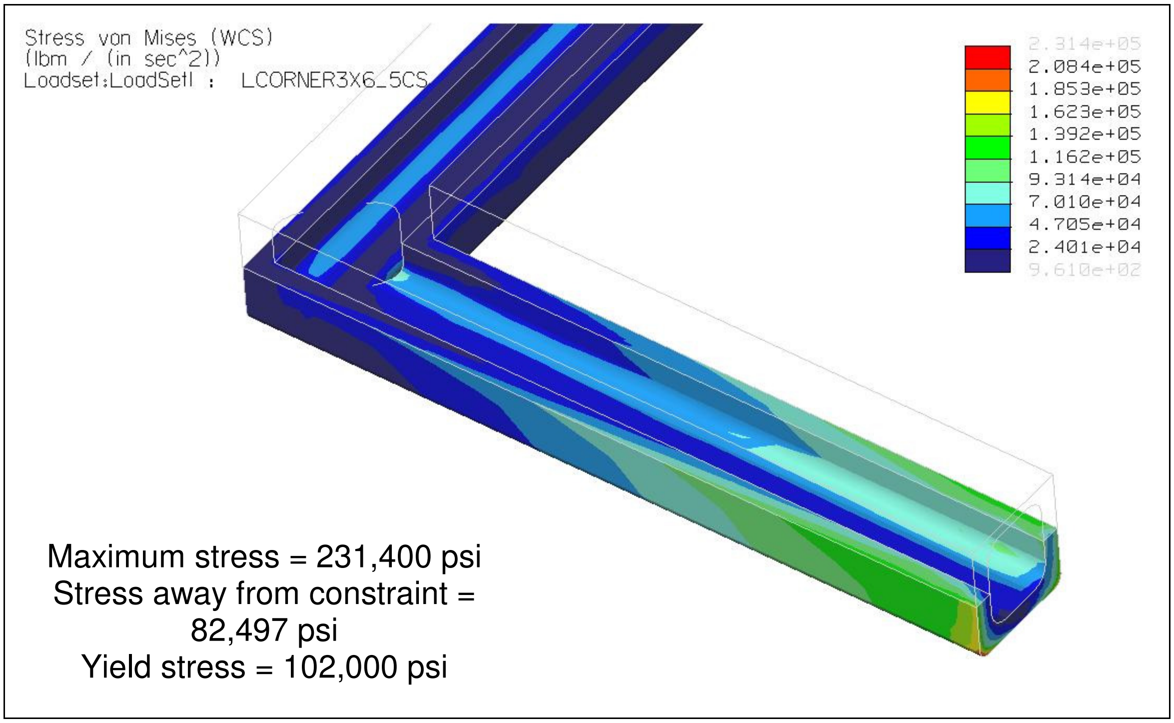
A-J Figure 4 – 316L SS T-Junction 10% Convergence Plot



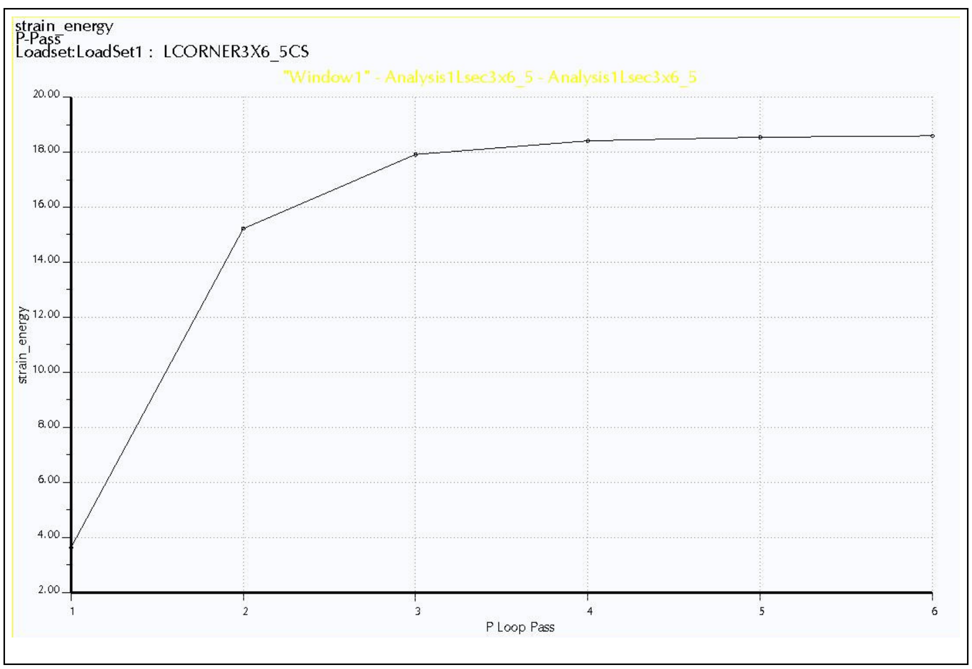
A-J Figure 5 – 4130 Q&T Single Member Fringe Plot



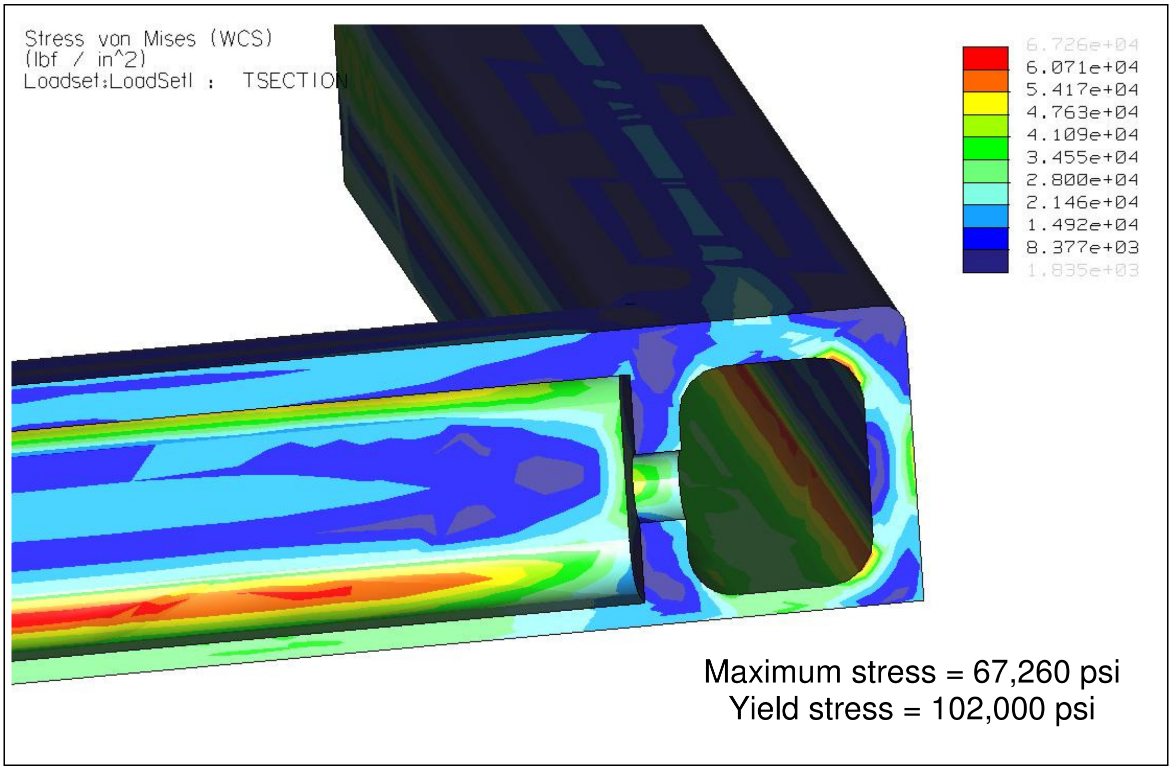
A-J Figure 6 – 4130 Q&T Single Member 10% Convergence Plot



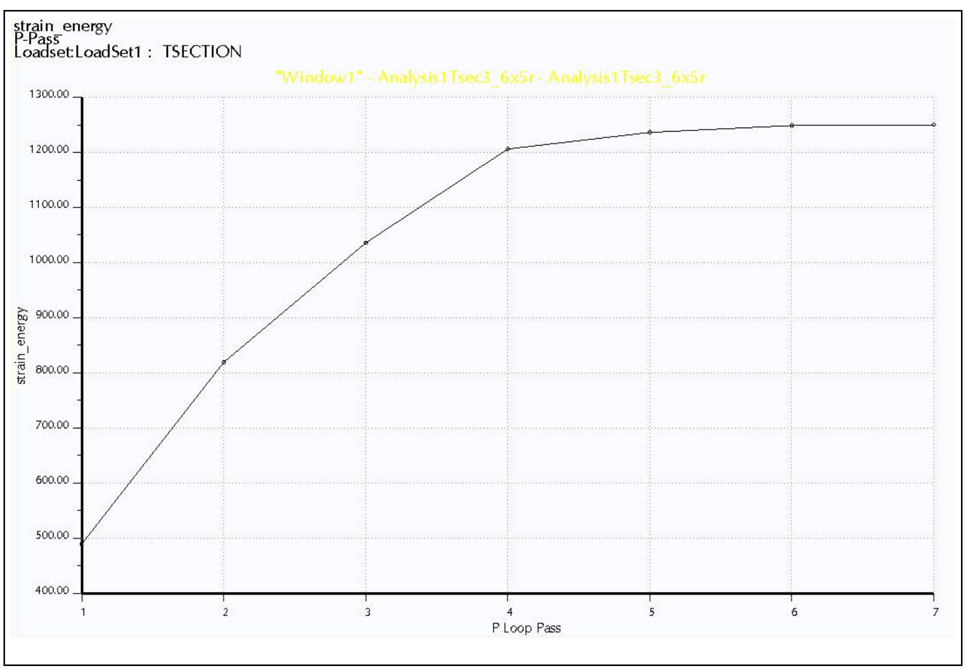
A-J Figure 7 – 4130 Q&T L-Junction Fringe Plot



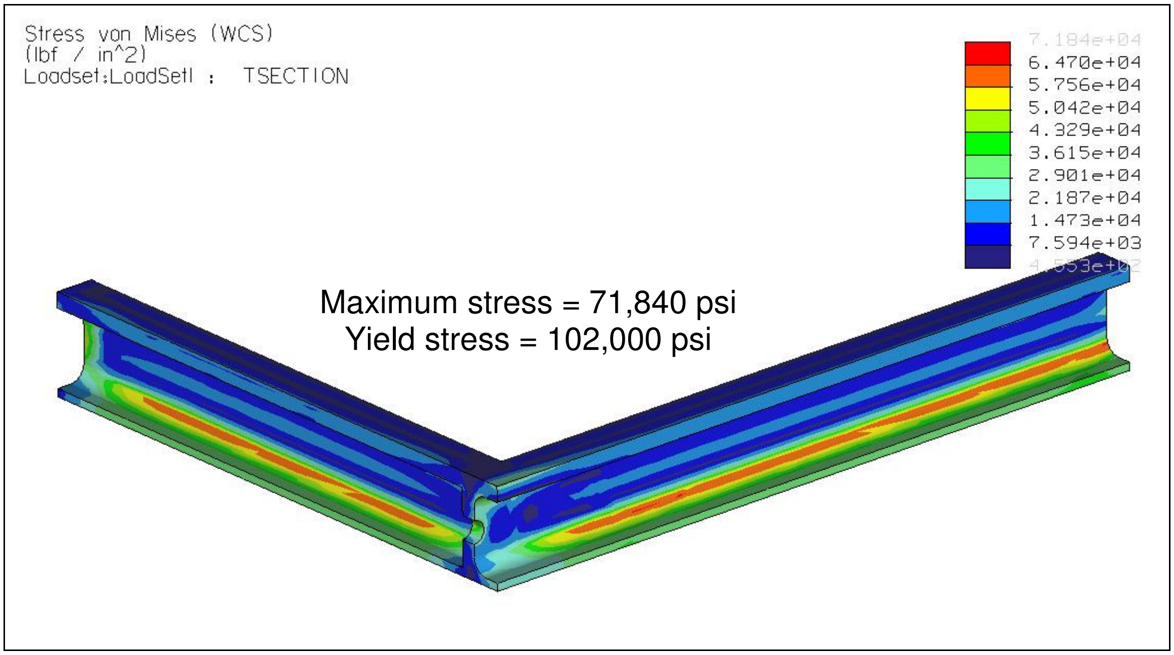
A-J Figure 8 – 4130 Q&T L-Junction 10% Convergence Plot



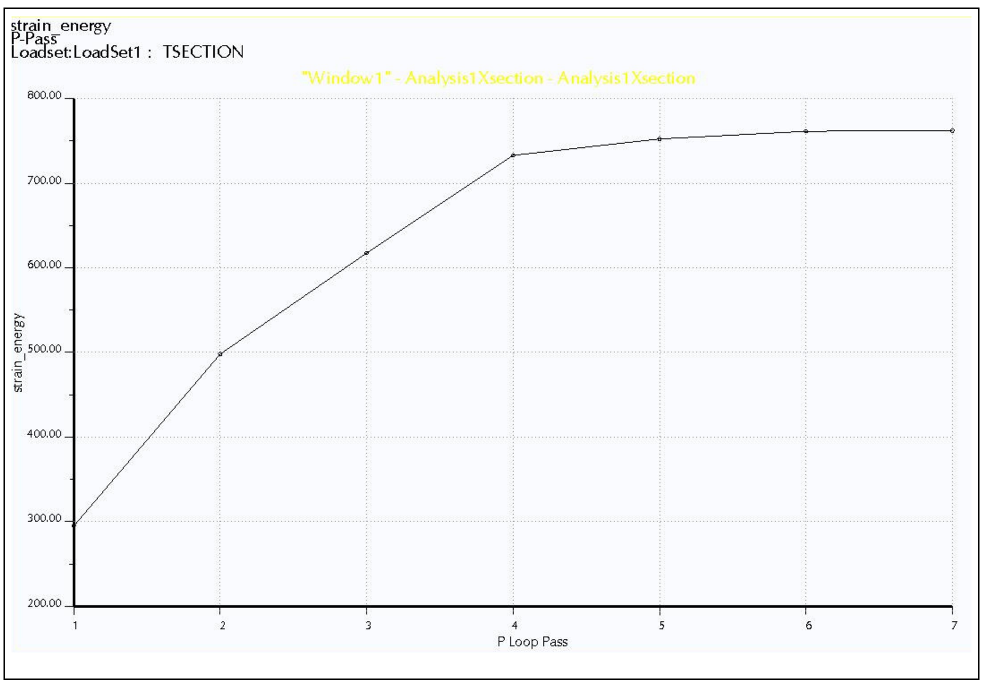
A-J Figure 9 – 4130 Q&T T-Junction Fringe Plot



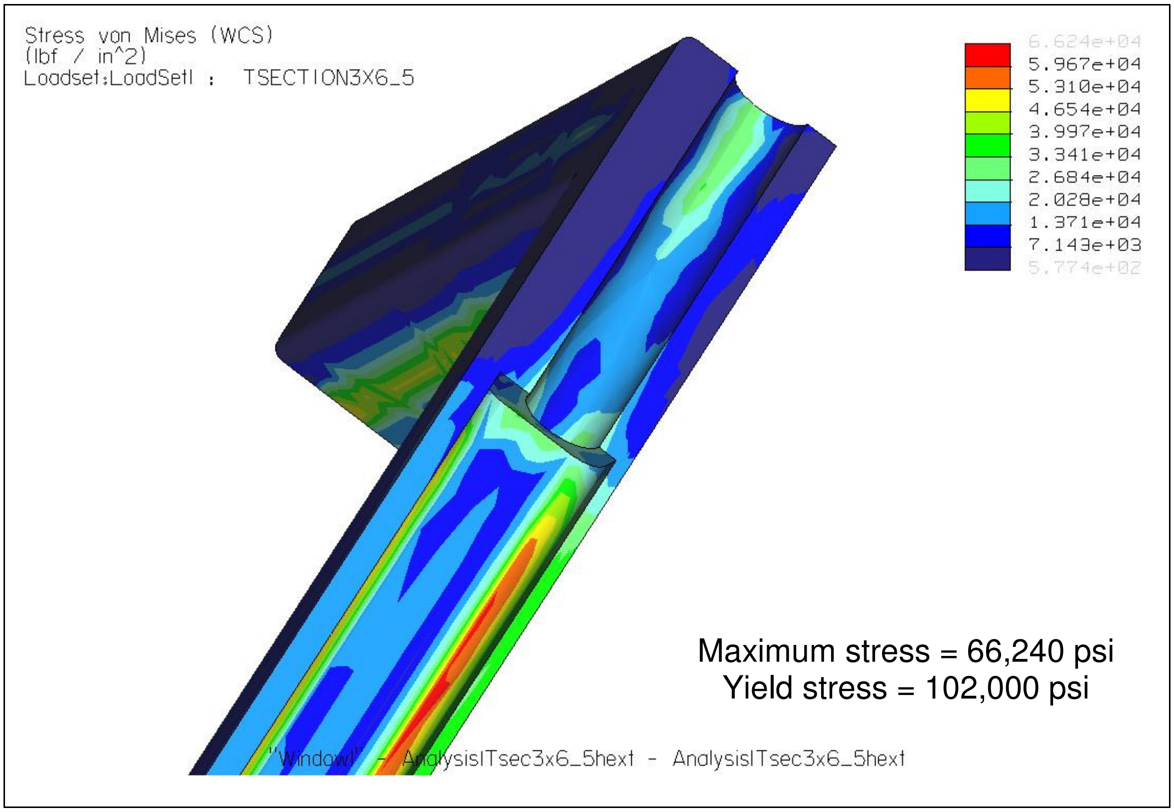
A-J Figure 10 – 4130 Q&T T-Junction 10% Convergence Plot



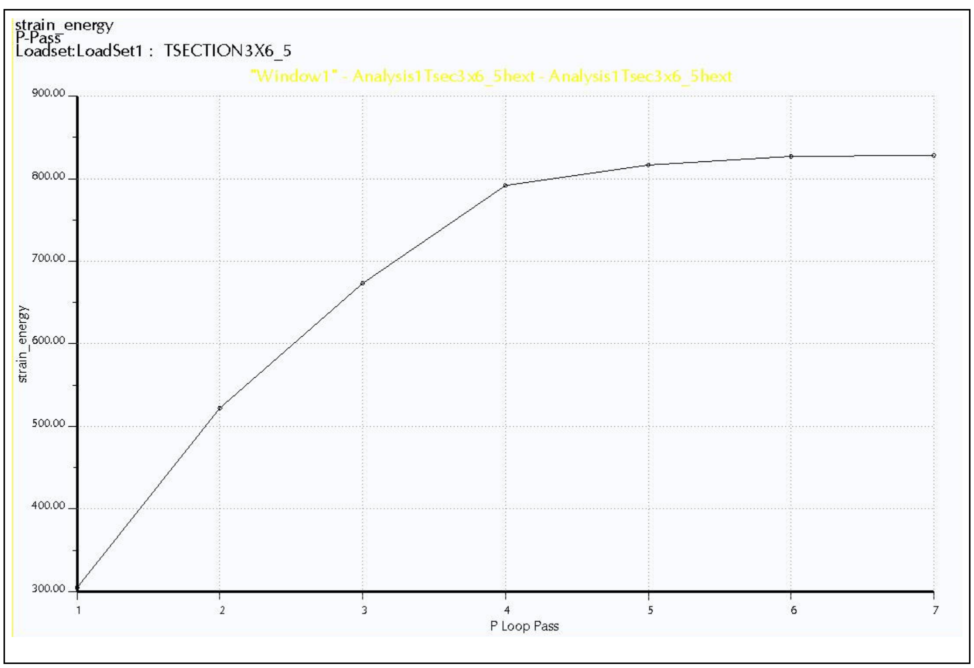
A-J Figure 11 – 4130 Q&T X-Junction Fringe Plot



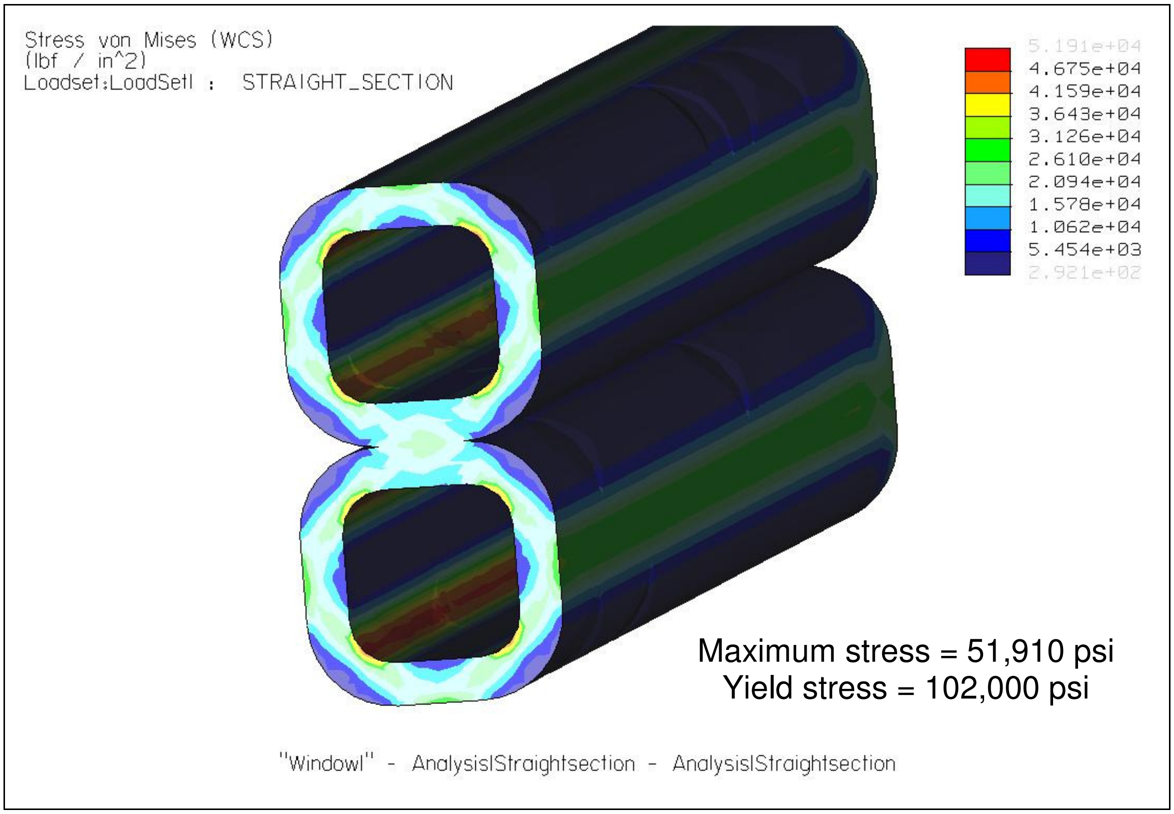
A-J Figure 12 – 4130 Q&T X-Junction 10% Convergence Plot



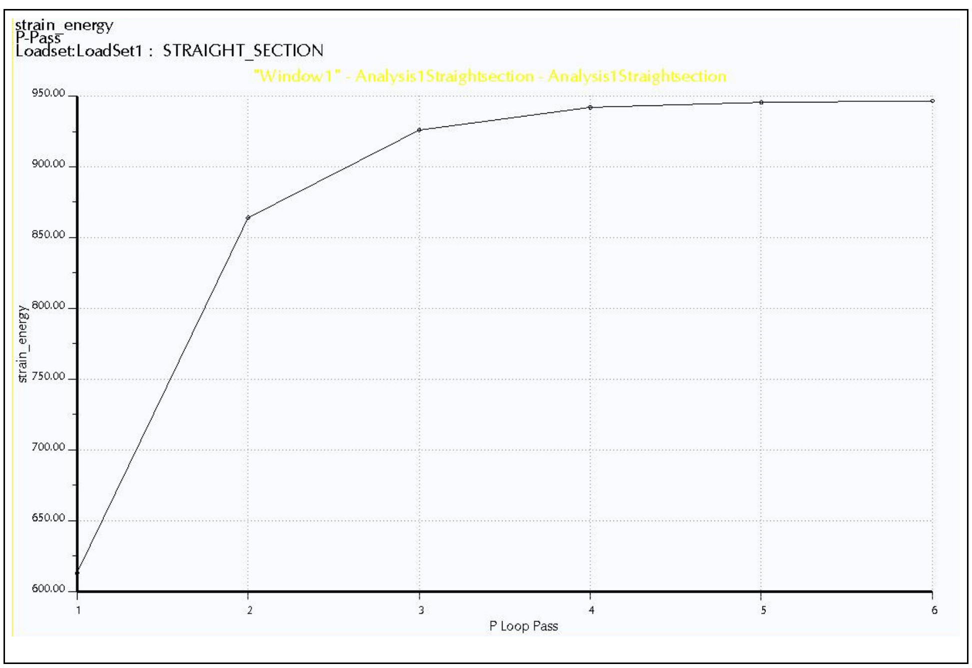
A-J Figure 13 – 4130 Q&T T-Junction Valve Block Fringe Plot



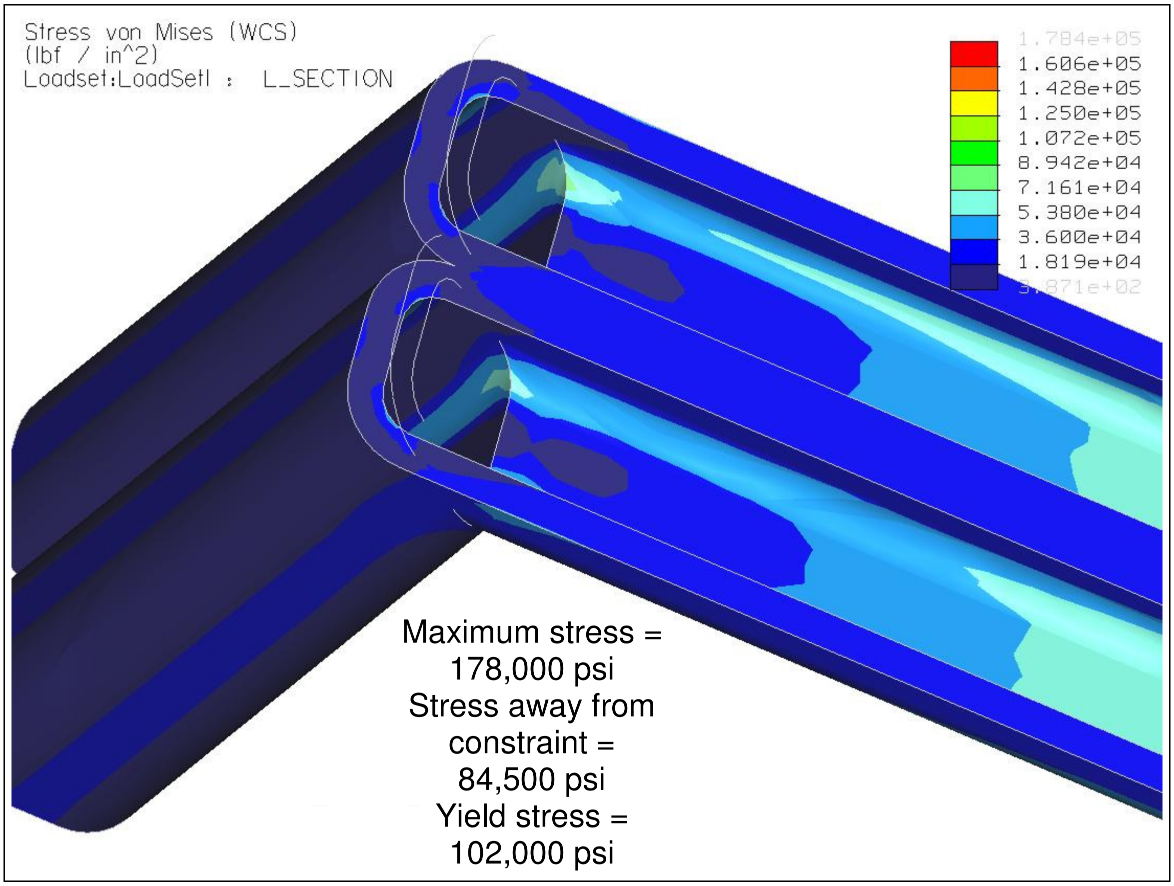
A-J Figure 14 – 4130 Q&T T-Junction Valve Block 10% Convergence Plot



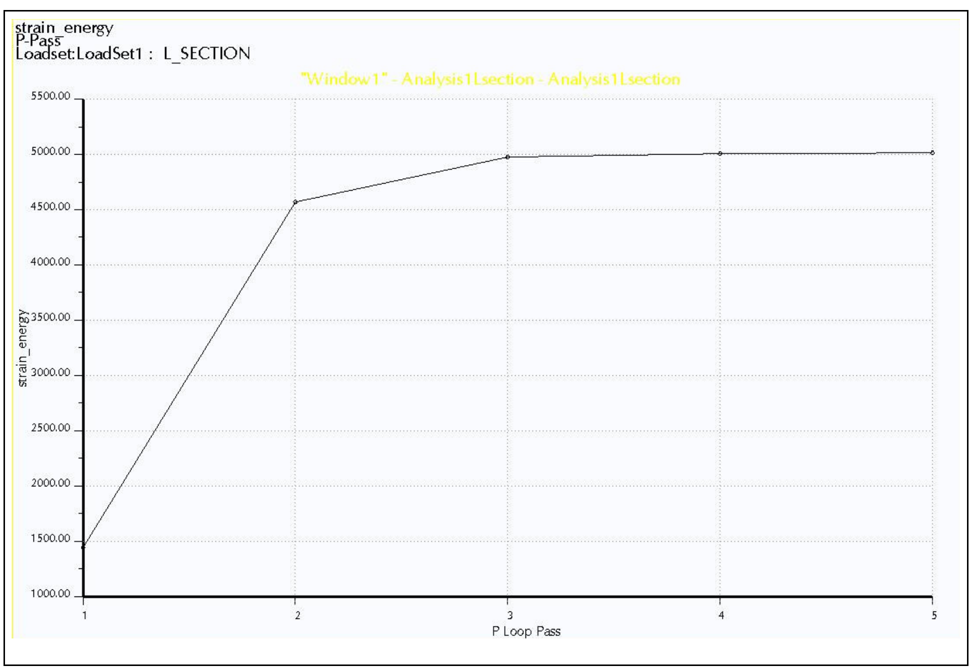
A-J Figure 15 – 4130 Q&T Boxes v1 Single Member Fringe Plot



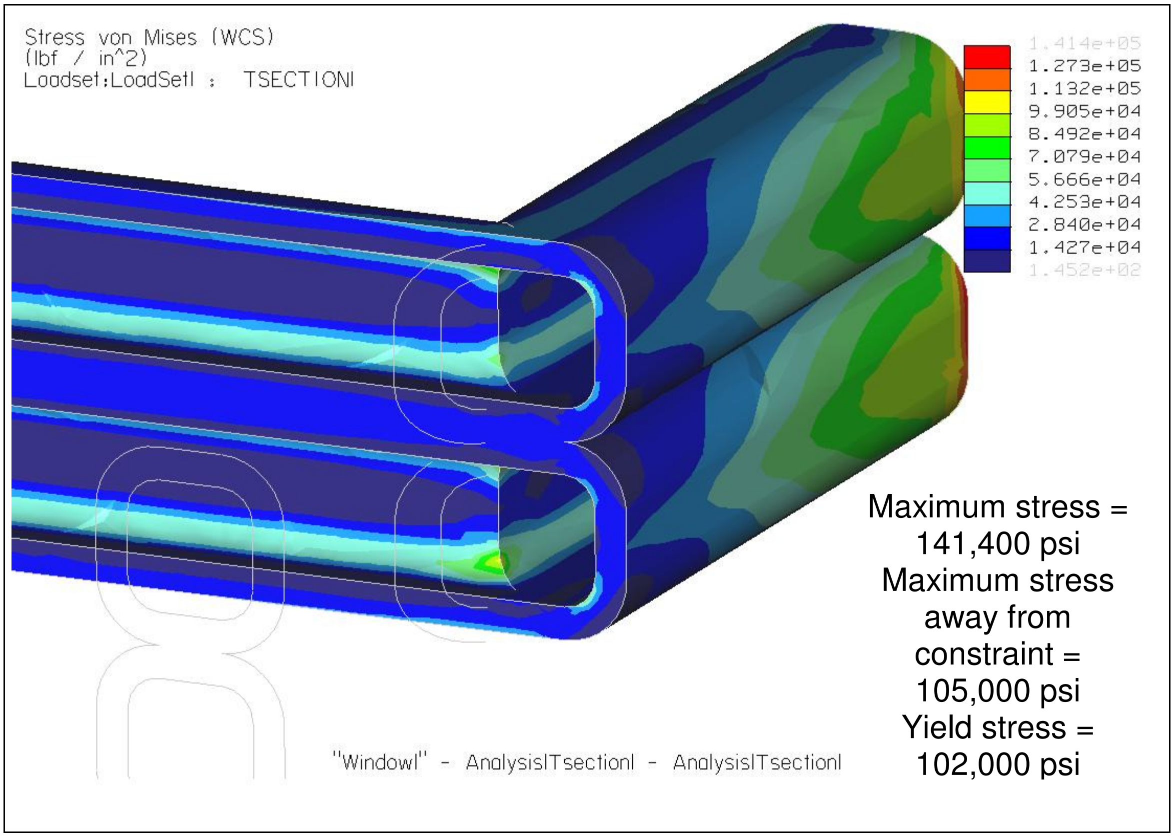
A-J Figure 16 – 4130 Q&T Boxes v1 Single Member 10% Convergence Plot



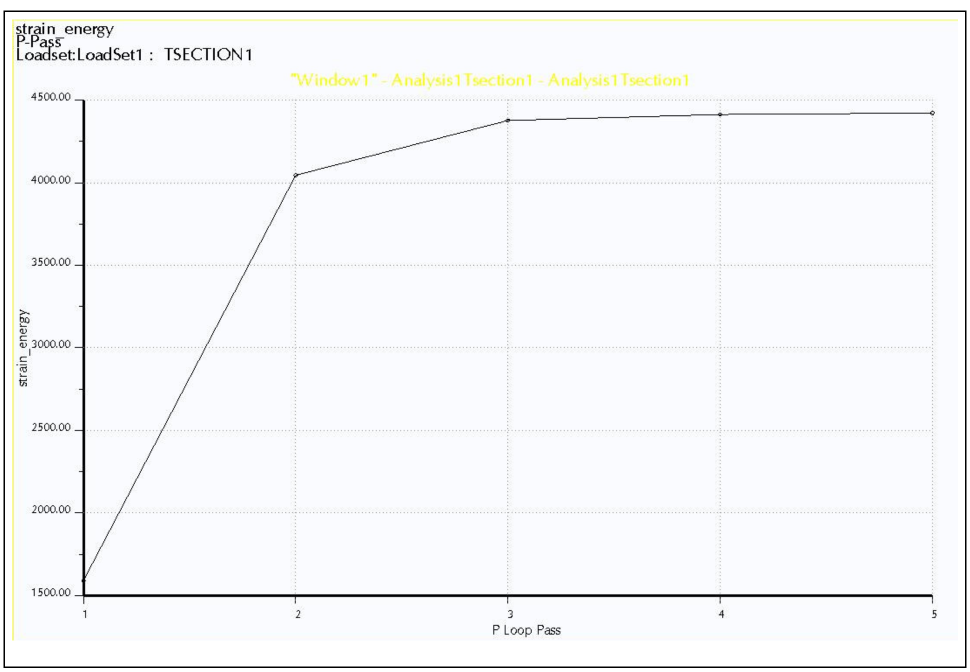
A-J Figure 17 – 4130 Q&T Boxes v1 L-Junction Fringe Plot



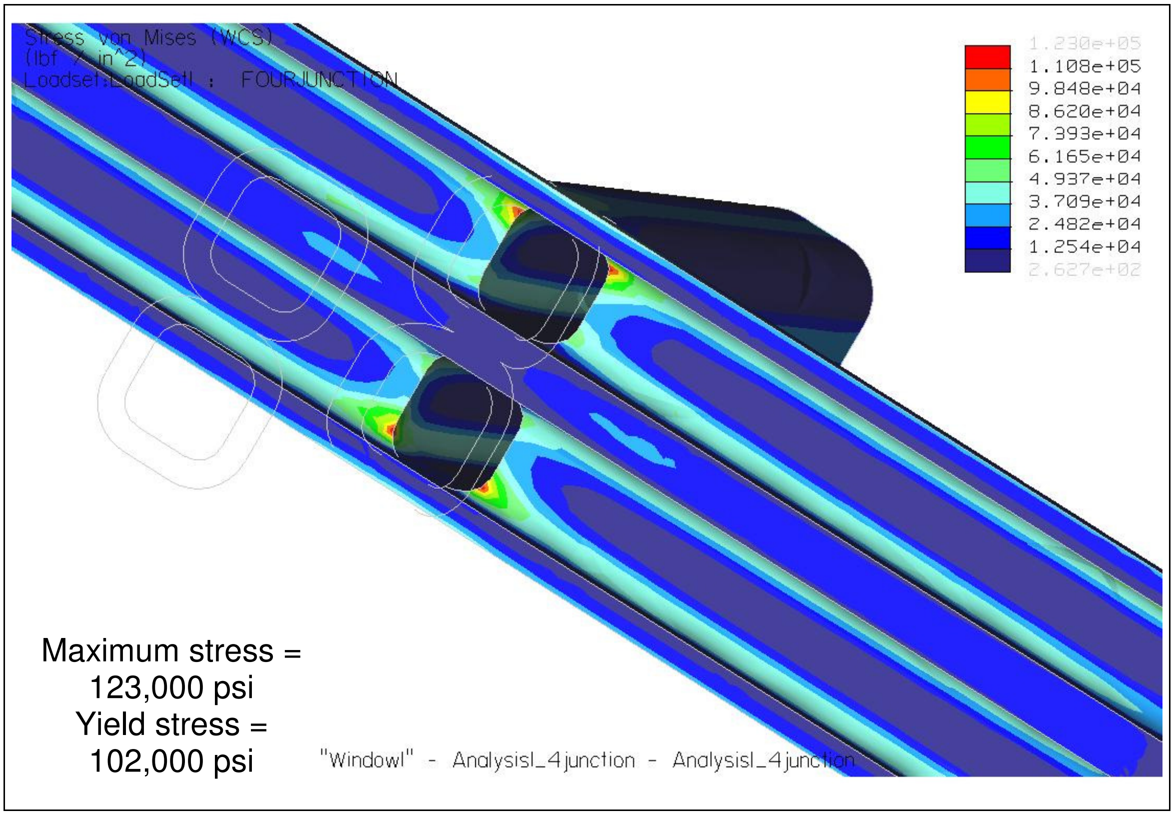
A-J Figure 18 – 4130 Q&T Boxes v1 L-Junction 10% Convergence Plot



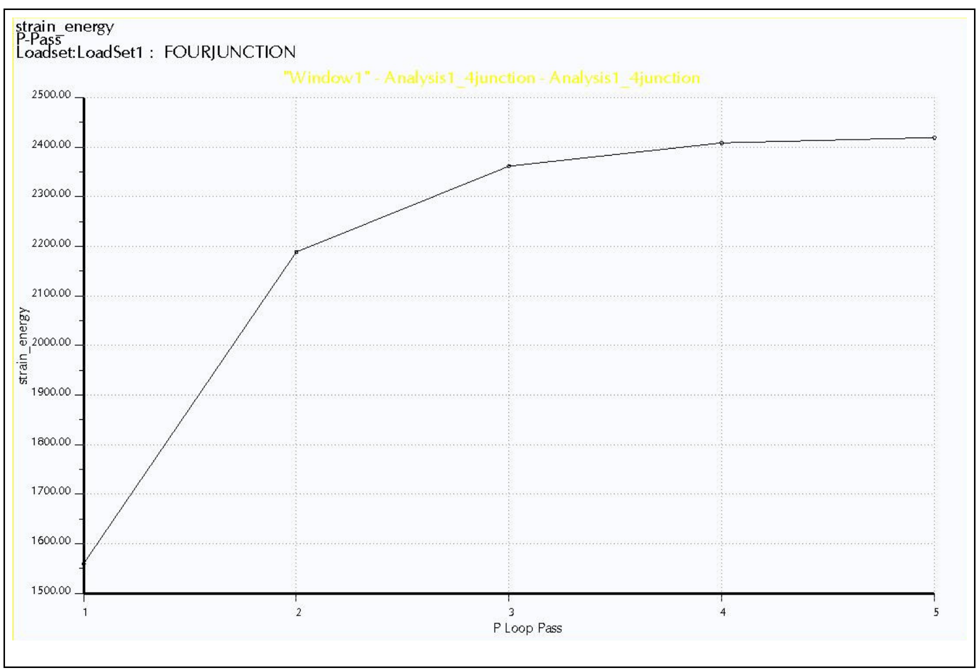
A-J Figure 19 – 4130 Q&T Boxes v1 T-Junction Fringe Plot



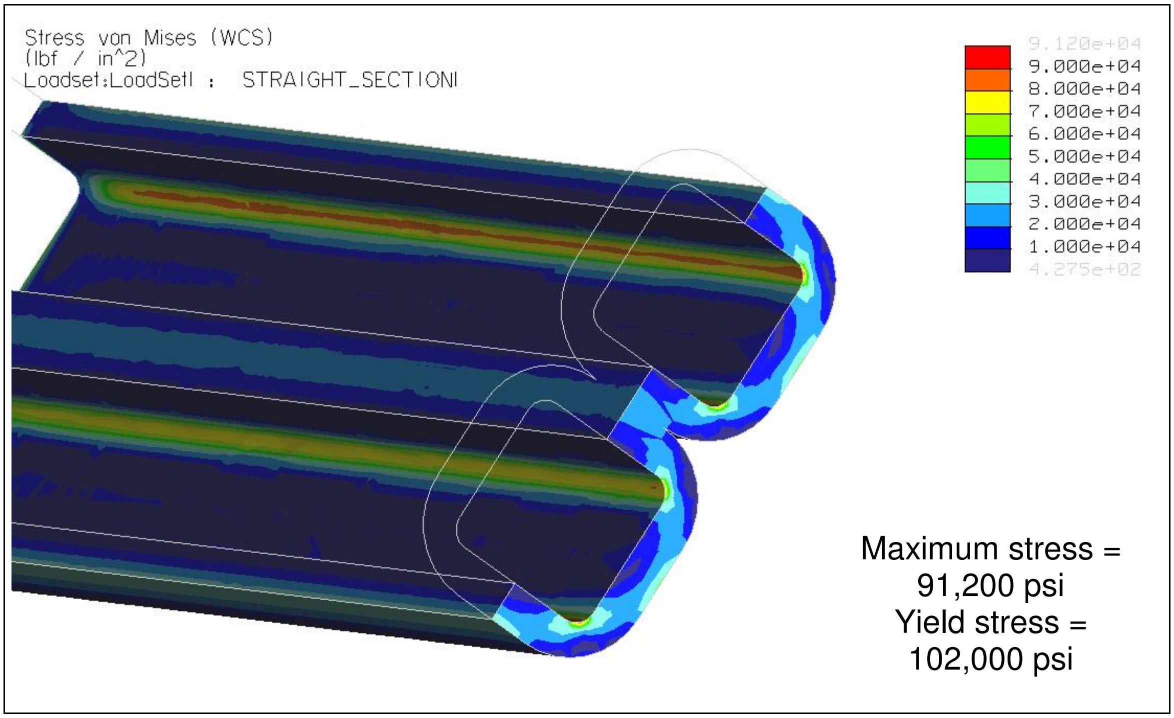
A-J Figure 20 – 4130 Q&T Boxes v1 T-Junction 10% Convergence Plot



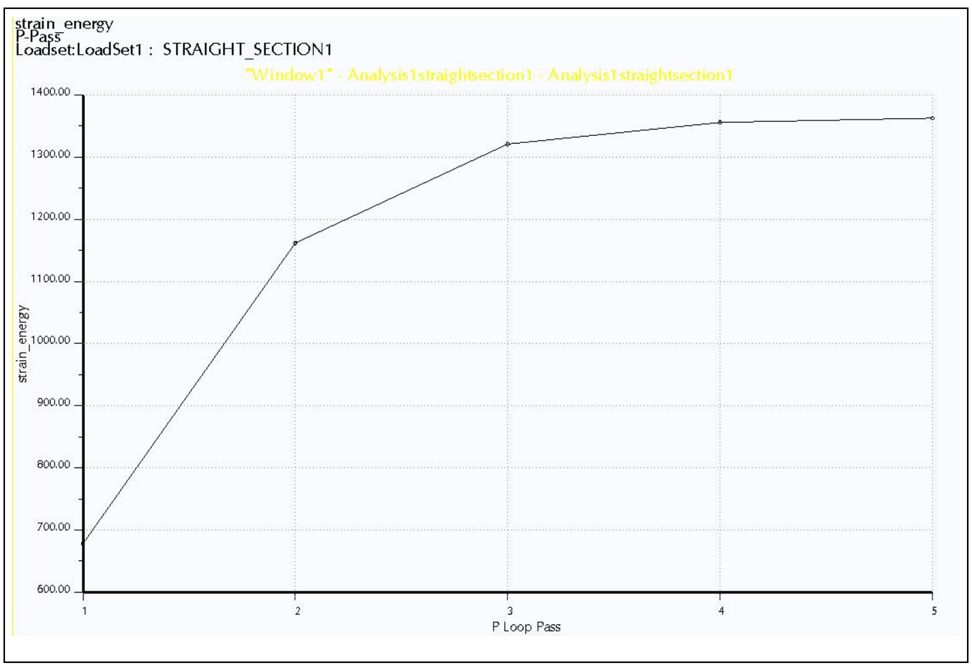
A-J Figure 21 – 4130 Q&T Boxes v1 X-Junction Fringe Plot



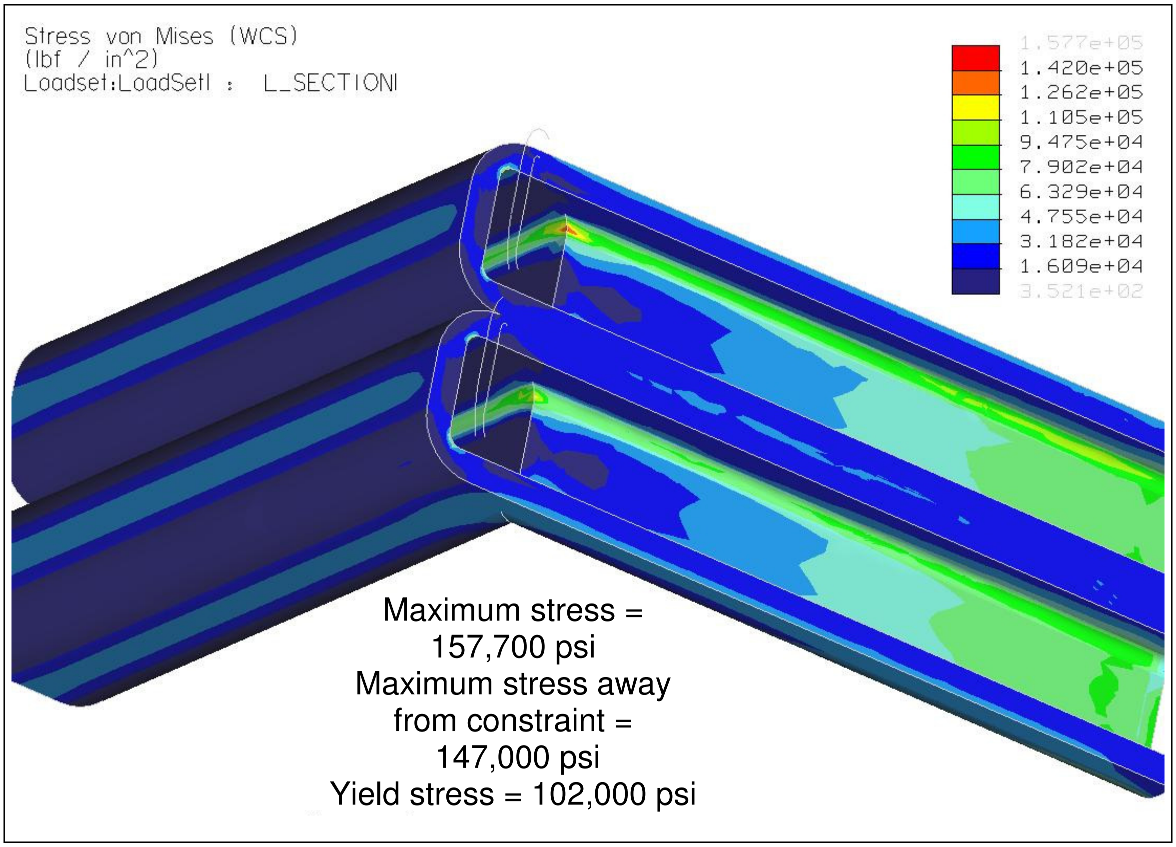
A-J Figure 22 – 4130 Q&T Boxes v1 X-Junction 10% Convergence Plot



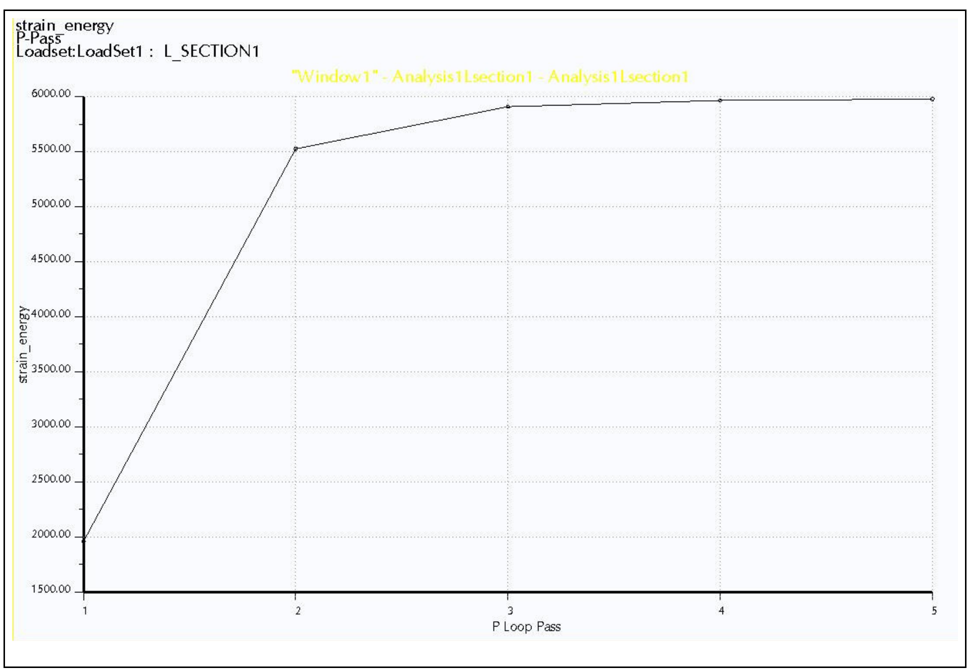
A-J Figure 23 – 4130 Q&T Boxes v2 Single Member Fringe Plot



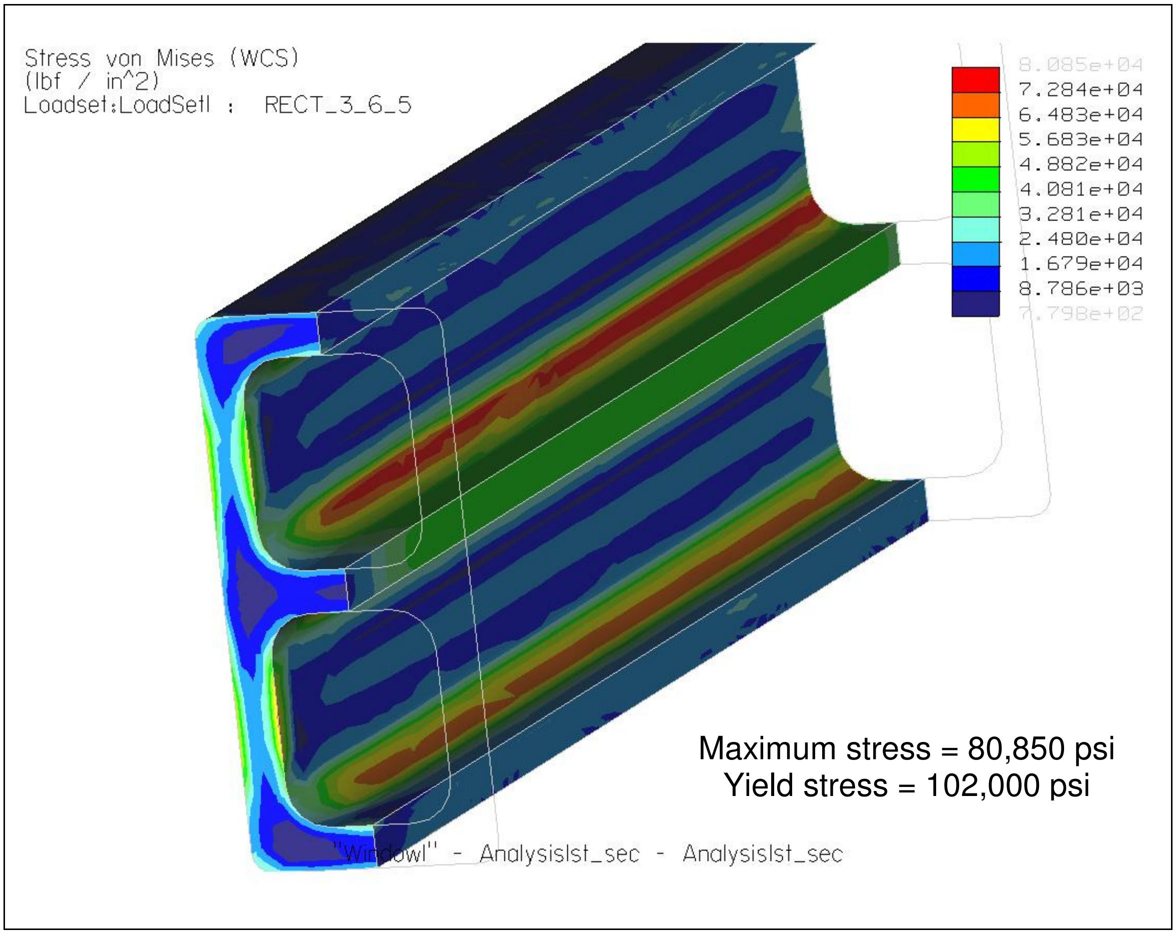
A-J Figure 24 – 4130 Q&T Boxes v2 Single Member 10% Convergence Plot



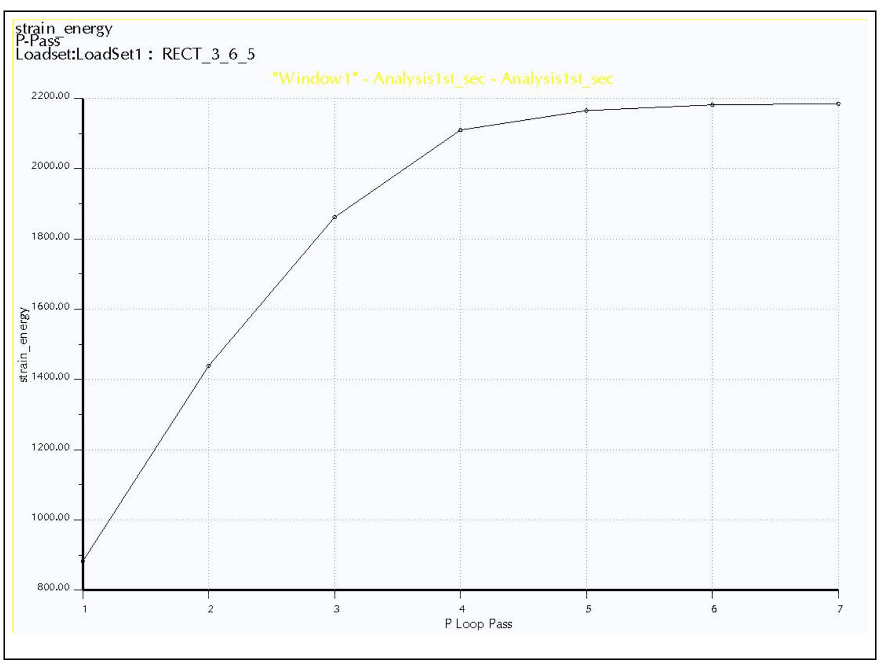
A-J Figure 25 – 4130 Q&T Boxes v2 L-Section Fringe Plot



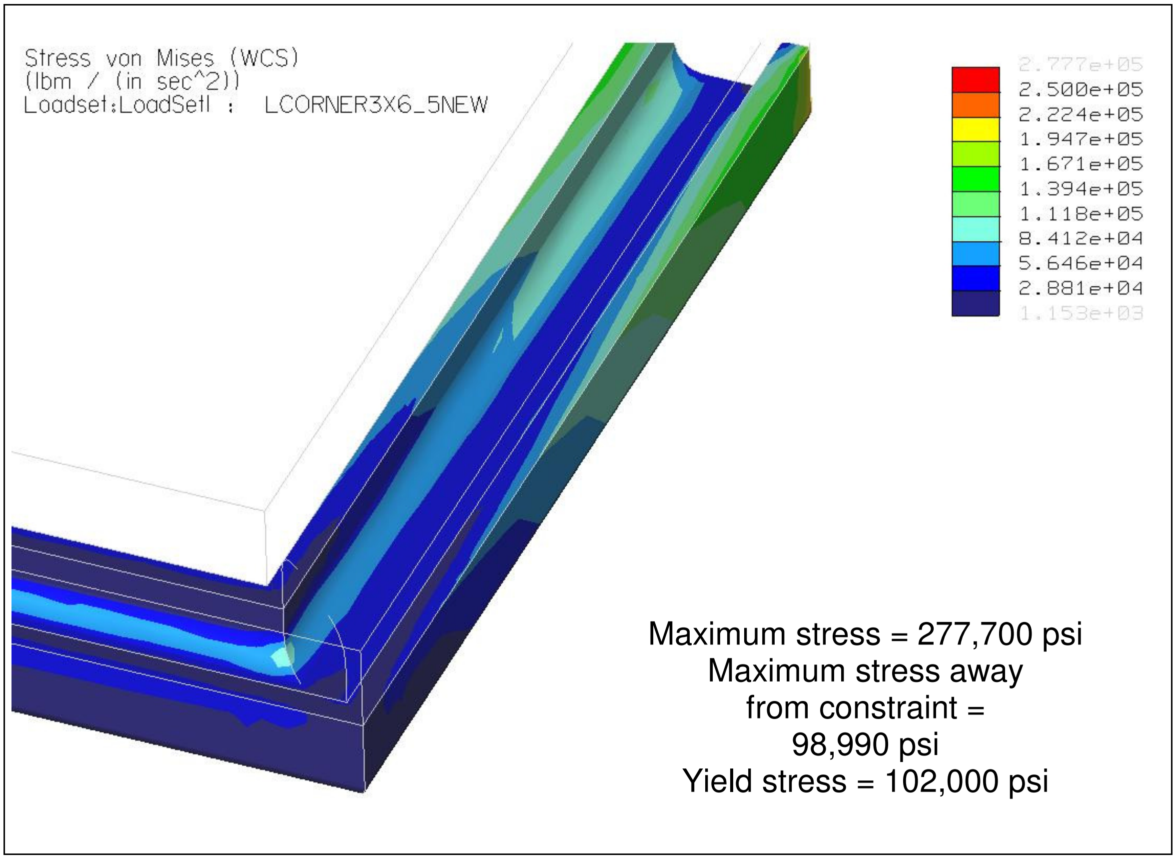
A-J Figure 26 – 4130 Q&T Boxes v2 L-Section 10% Convergence Plot



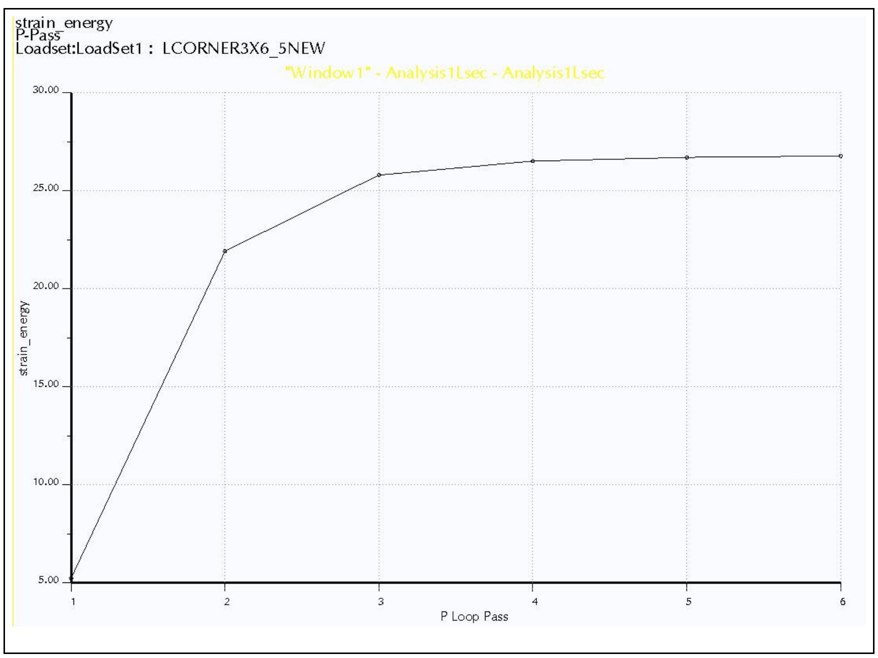
A-J Figure 27 – 4130 Q&T Hydrostatic Straight Sect Fringe Plot



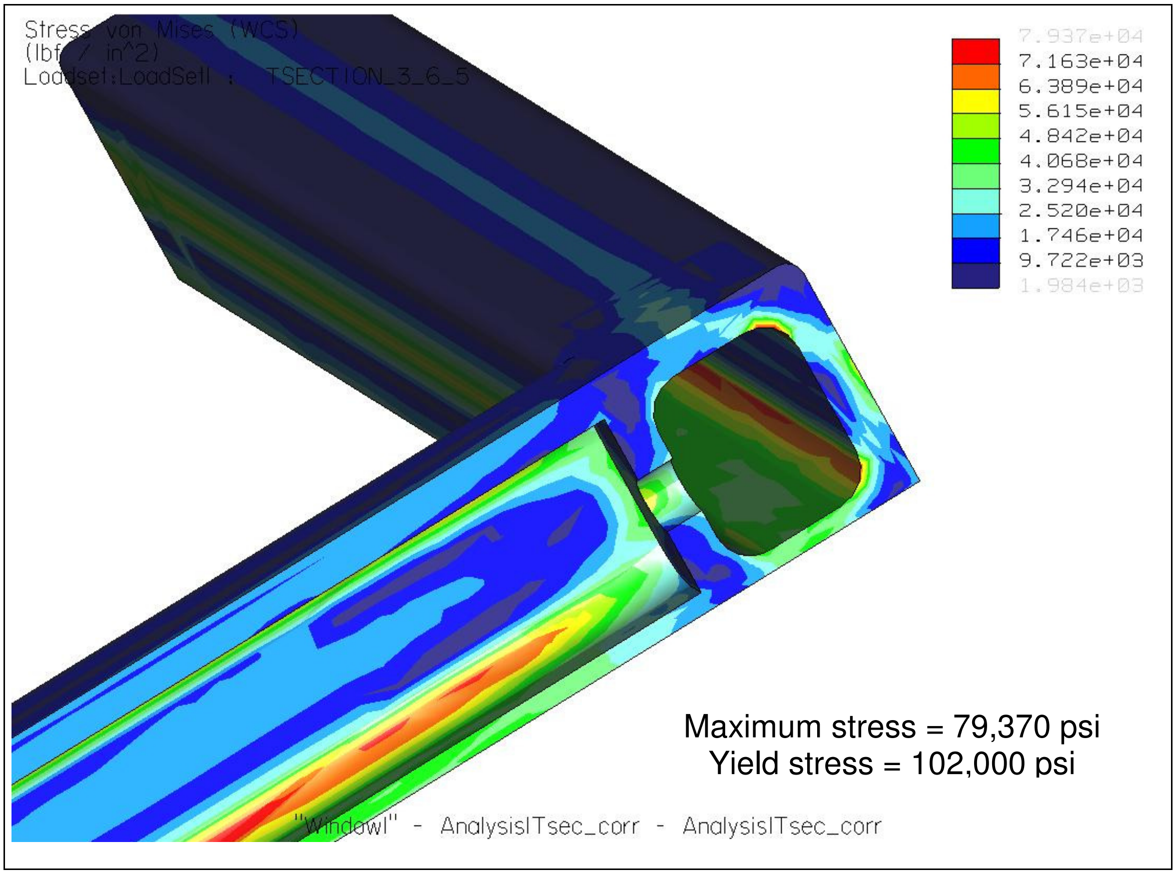
A-J Figure 28 – 4130 Q&T Hydrostatic Straight Sect 10% Convergence Plot



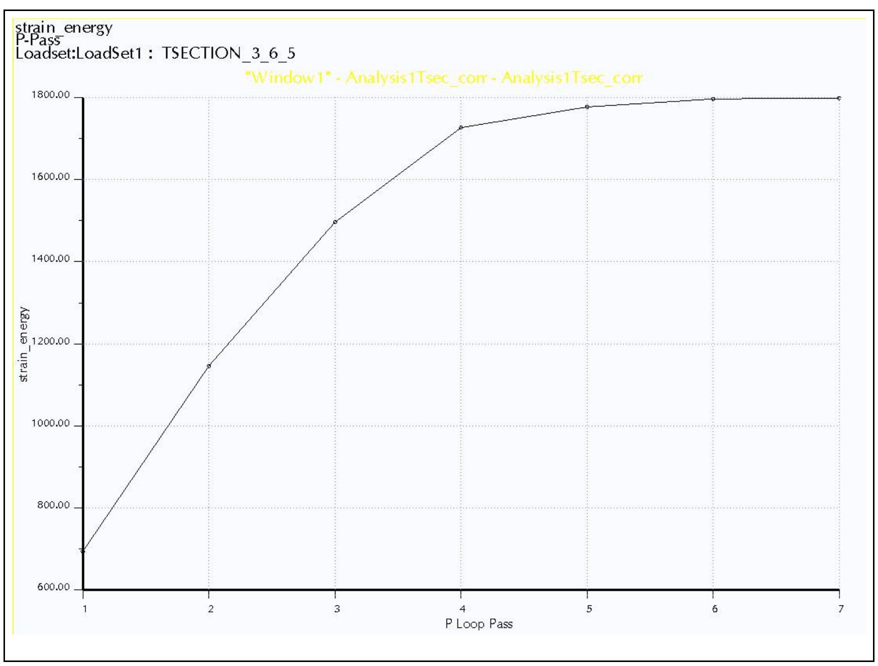
A-J Figure 29 – 4130 Q&T Hydrostatic L-Section Fringe Plot



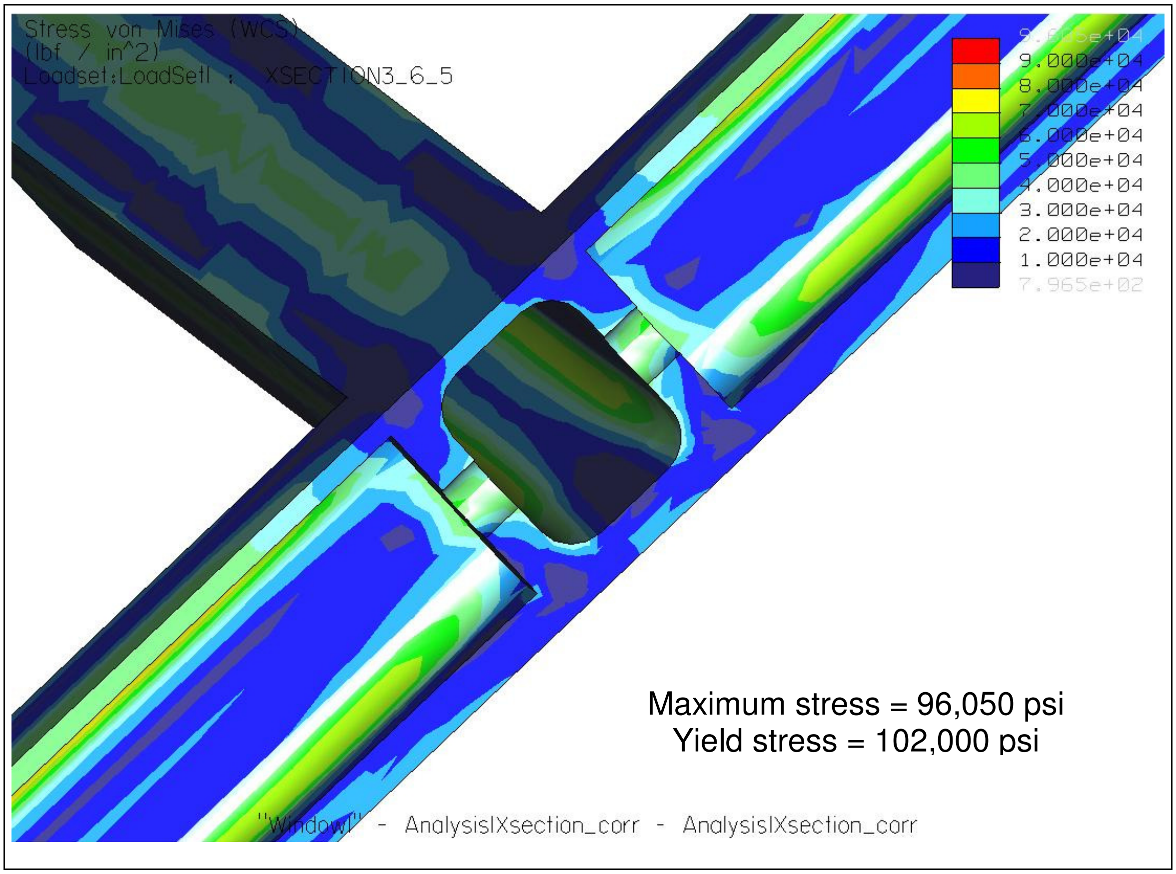
A-J Figure 30 – 4130 Q&T Hydrostatic L-Section 10% Convergence Plot



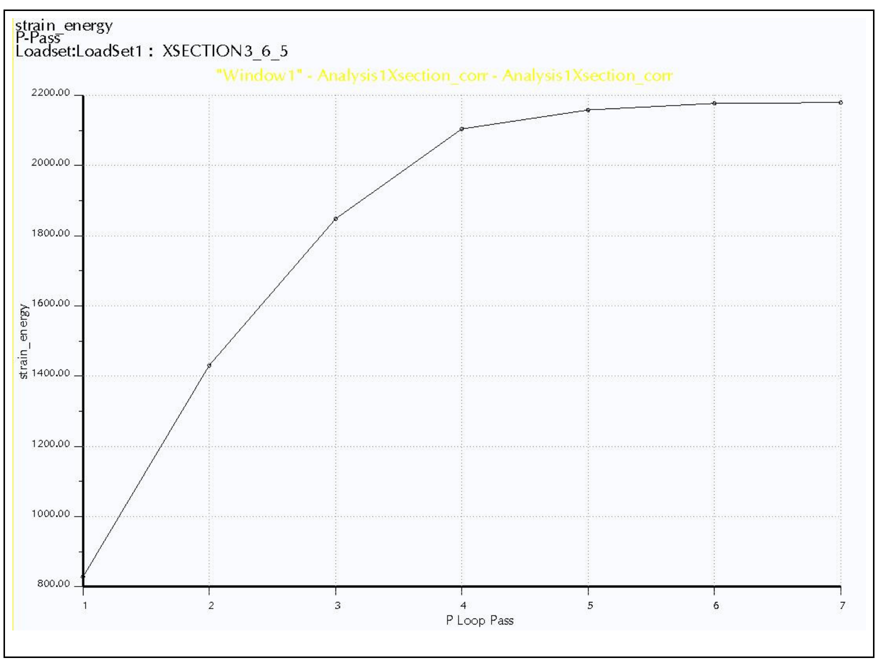
A-J Figure 31 – 4130 Q&T Hydrostatic T-Section Fringe Plot



A-J Figure 32 – 4130 Q&T Hydrostatic T-Section 10% Convergence Plot



A-J Figure 33 – 4130 Q&T Hydrostatic X-Section Fringe Plot



A-J Figure 34 – 4130 Q&T Hydrostatic X-Section 10% Convergence Plot

APPENDIX K: ADDITIONAL MATERIALS (CYLINDRICAL VESSEL)

A-K Figure 1 – Wall Thickness Equations [20],[29]

$$Tangential\ Stress\ \sigma_t = \frac{r_i{}^2 p_i}{r_o{}^2 - r_i{}^2}\left(1 + \frac{r_o{}^2}{r^2}\right)$$

$$Radial\ Stress\ \sigma_r = \frac{r_i{}^2 p_i}{r_o{}^2 - r_i{}^2}\left(1 - \frac{r_o{}^2}{r^2}\right)$$

$$Longitudinal\ Stress\ \sigma_l = \frac{r_i{}^2 p_i}{r_o{}^2 - r_i{}^2}$$

Where:

r_i *is the inner radius*

r_o *is the outer radius*

p_i *is the inside pressure*

Let the query radius $r = r_i$ *to find the peak stress along the inner wall.*

$$Von\ Mises\ Stress\ \sigma' = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_2)^2}{2}}$$

Where $\sigma_1, \sigma_2, \sigma_3$ *are the principal stresses*

Plug in $\sigma_t, \sigma_r, \sigma_l$ *for the principal stresses in Von Mises Stress equation and reducing terms:*

$$\sigma' = \frac{\sqrt{3}p_i}{1 - \frac{r_i{}^2}{r_o{}^2}}$$

Rewriting the equation in terms of r_i *as a design guide:*

$$r_i = r_o\sqrt{1 - \sqrt{3}\frac{p_i}{\sigma'}}$$

A-K Figure 2 – Manufacturing Quote from Taylor-Wharton [16]

DOT3A5000 4.0" OD 316 Stainless Steel Double Ended Cylinder

Part# (To Be Assigned)

Quantity:	**1,000 – 1,999**	**10,000**
Price:	**$777.48 ea**	**$726.970 ea**

DOT3AA5000 4.0" 4130 Steel Double Ended Cylinder

Part# (To Be Assigned)

Quantity:	**1,000 – 1,999**	**10,000**
Price:	**$225.98 ea**	**$198.42 ea**

SPECIFICATIONS: Cylinder Is Manufactured to Applicable DOT Specifications and In Accordance With TW Manufacturing Tolerances. Cylinders are furnished with Standard DOT Markings and Serial number. Cylinder design required spun bottoms to be plugged. Inlet threads: 3/4 x14NGT.

DELIVERY: Delivery to be determined based on quantity ordered, material lead-time and production schedules at time or order.

FREIGHT: FOB Huntsville, Alabama 35803

PACKAGING: Packing is standard commercial, boxed on pallet, banded and stretch-wrapped.

PAYMENT TERMS: **.5% - 10-days, net 30 days. All Orders Subject to Credit Approval**.

Taylor-Wharton appreciates and thanks you for allowing us an opportunity to quote you cylinder requirements. If you should have any questions, please feel free to give me a call.

Mike Camp

General Manager, Cylinders

TW Cylinders LLC

Division of TAYLOR - WHARTON INTERNATIONAL

P: 256-650-9111

F: 256-650-9195

C: 256-755-0845

email: mcamp@taylorwharton.com

A-K Figure 3 – Manufacturing Quote from Taylor-Wharton [16]

OPERATIONS FOR DOUBLED ENDED 5000 PSI CYLINDERS

316 L MATERIAL	4130 STEEL MATERIAL
1. CUT TUBING TO LENGTH	1. CUT TUBING TO LENGTH
2. SPIN TOP	2. SPIN TOP
3. SPIN BOTTOM	3. SPIN BOTTOM
4. ASSIGN LOT NUMBER FOR TRACEABILITY	4. HEAT TREAT
5. MOVE TO INVENTORY	5. ASSIGN LOT NUMBER FOR TRACEABILITY
6. THREAD TOP	6. PHYSICAL TESTS
7. THREAD BOTTOM	7. MOVE TO INVENTORY
8. ROUGH POLISH CYLINDER	8. WHEELABRATE OUTSIDE SURFACE
9. COLD WORK CYLINDER (PRESSURIZE TO WP)	9. THREAD TOP
10. HEAT TREAT	10. THREAD BOTTOM
11. PHYSICAL TESTS	11.HYDROSTATIC TEST EACH CYLINDER
12. HYDROSTATIC TEST EACH CYLINDER	12. STAMP EACH CYLINDER (DOT MARKS)
13. STAMP EACH CYLINDER (DOT MARKS)	13. PAINT, IF REQUIRED
14. FINE POLISH CYLINDER	14. FINAL INSPECT
15. FINAL INSPECT	15. INSERT PLASTIC PLUGS AND PACKAGE
16. INSERT PLASTIC PLUGS AND PACKAGE	16. SHIP
17. SHIP	

APPENDIX L: ADDITIONAL MATERIALS (TUBING COILS)

A-L Figure 1 – Swagelok Tubing Data Catalog [30]

Suggested Allowable Pressure Tables

Figure and tables are for reference only. No implication is made that these values can be used for design work. Applicable codes and practices in industry should be considered. ASME Codes are the successor to and replacement of ASA Piping Codes.

- All pressures are calculated from equations in ASME B31.3, Process Piping. See factors for calculating working pressures in accordance with ASME B31.1, Power Piping.
- Calculations are based on maximum OD and minimum wall thickness, except as noted in individual tables.

  Example: 1/2 in. OD × 0.035 in. wall stainless steel tubing purchased to ASTM A269:

  OD Tolerance ± 0.005 in. / Wall Thickness ± 10 %

  Calculations are based on 0.505 in. OD × 0.0315 in. wall tubing.
- No allowance is made for corrosion or erosion.

Suggested Allowable Working Pressure for *Stainless Steel Tubing*

Table 3—Fractional Stainless Steel Seamless Tubing

Allowable working pressures are calculated from an S value of 20 000 psi (137.8 MPa) for ASTM A269 tubing at –20 to 100°F (–28 to 37°C), as listed in ASME B31.3 and ASME B31.1, except as noted.

For Welded Tubing

For welded and drawn tubing, a derating factor must be applied for weld integrity:

- for double-welded tubing, multiply working pressure by 0.85
- for single-welded tubing, multiply working pressure by 0.80.

Tube OD in.	**Tube Wall Thickness**, in.																Swagelok Fitting Series
	0.010	**0.012**	**0.014**	**0.016**	**0.020**	**0.028**	**0.035**	**0.049**	**0.065**	**0.083**	**0.095**	**0.109**	**0.120**	**0.134**	**0.156**	**0.188**	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)																
1/16	5600	6800	8100	9400	12 000												100
1/8						8500	10 900										200
3/16						5400	7 000	10 200									300
1/4						4000	5 100	7 500	10 200①								400
5/16							4 000	5 800	8 000								500
3/8							3 300	4 800	6 500	7500①②							600
1/2							2 600	3 700	5 100	6700							810
5/8								2 900	4 000	5200	6000						1010
3/4								2 400	3 300	4200	4900	5800					1210
7/8								2 000	2 800	3600	4200	4800					1410
1									2 400	3100	3600	4200	4700				1610
1 1/4										2400	2800	3300	3600	4100	4900		2000
1 1/2											2300	2700	3000	3400	4000	4900	2400
2												2000	2200	2500	2900	3600	3200

① For higher pressures, see the Swagelok *Medium-Pressure Fittings* catalog, MS-02-335, or the Swagelok *High-Pressure Fittings* catalog, MS-01-34.
② Rating based on repeated pressure testing of the Swagelok tube fitting with a 4:1 design factor based upon hydraulic fluid leakage.

Suggested Ordering Information

High-quality, fully annealed (Type 304, 304/304L, 316, 316/316L, 317, 317/317L) (seamless or welded and drawn) stainless steel hydraulic tubing, ASTM A269 or A213, or equivalent. Hardness not to exceed 90 HRB or 200 HV. Tubing to be free of scratches, suitable for bending and flaring. OD tolerances not to exceed ± 0.003 in. for 1/16 in. OD tubing.

Note: Certain austenitic stainless tubing has an allowable ovality tolerance double the OD tolerance and may not fit into Swagelok precision tube fittings. Dual-certified grades such as 304/304L, 316/316L, and 317/317L meet the minimum chemistry and the mechanical properties of both alloy grades.

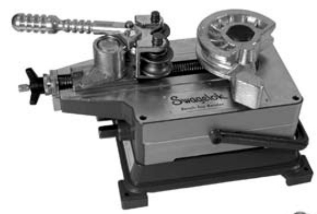
A-L Figure 2 – Swagelok Tube Bending Information [31]

Tube Benders

Bench Top Tube Benders

Features

- Rugged, lightweight aluminum construction
- 1 to 180° bending range
- 1/4 to 1 1/4 in. outside diameter (0.028 to 0.120 in. wall thickness) and 12 to 30 mm outside diameter (1.0 to 3.0 mm wall thickness) tubing range
- Steel bend shoes required for:
  - 1 in. outside diameter tubing with greater than 0.095 in. wall thickness
  - 25 mm tubing with greater than 2.4 mm wall thickness
  - all sizes of SAF 2507™ tubing
  - all sizes of heavy-wall annealed stainless steel tubing
  - all sizes of cold-drawn 1/8-hard stainless steel seamless tubing.
- Includes grease gun and metal carrying case for storage
- Manual model can be operated with a 1/2 in. drill motor using optional torque clutch and support arm.
- CE compliant



Manual Model

Electric Model

Technical Data

- Dimensions—tube bender in case:

  14 1/2 in. (37 cm) high, 21 in. (53 cm) wide, 11 in. (28 cm) deep
- Weight—tube bender in case, excluding tools:

  Manual model—75 lb (34 kg)

  Electric model—79 lb (36 kg)
- Power requirements (electric model)

  MS-BTB-1—110 V (ac), 50/60 Hz; maximum current—10 A

  MS-BTB-2—230 V (ac), 50/60 Hz; maximum current—5 A

See Ordering Information, page 4, and Options and Accessories, page 5.

Tubing Data

Minimum tube length, bend radius, and wall thickness limits required to make a 90° bend in annealed tubing are listed below. See Swagelok *Tubing Data,* MS-01-107, for suggested tubing wall thickness for use with Swagelok tube fittings.

Fractional Tubing

<table>
<tr><th rowspan="2">Tube OD</th><th rowspan="2">Min Tube Length</th><th rowspan="2">Approx Bend Radius</th><th colspan="4">Wall Thickness, Min/Max</th></tr>
<tr><th>Carbon Steel</th><th>Stainless Steel</th><th>Heavy-Wall Annealed SS</th><th>Cold-Drawn 1/8-Hard SS</th></tr>
<tr><td colspan="7">Dimensions, in.</td></tr>
<tr><td>1/4</td><td rowspan="3">7.00</td><td rowspan="3">1.4</td><td colspan="2">0.028/0.065</td><td>0.065/0.095</td><td>0.028/0.065</td></tr>
<tr><td>3/8</td><td>0.035/0.065</td><td>0.035/0.083</td><td>0.083/0.134</td><td>0.035/0.083</td></tr>
<tr><td>1/2</td><td colspan="2">0.035/0.083</td><td>0.083/0.188</td><td>0.049/0.109</td></tr>
<tr><td>5/8</td><td>8.50</td><td>1.8</td><td>0.035/0.095</td><td>0.049/0.095</td><td colspan="2" rowspan="5">—</td></tr>
<tr><td>3/4</td><td>9.75</td><td>2.2</td><td colspan="2" rowspan="2">0.049/0.109</td></tr>
<tr><td>7/8</td><td>10.5</td><td>2.6</td></tr>
<tr><td>1</td><td>12.2</td><td>3.2</td><td>0.049/0.120</td><td>0.065/0.120</td></tr>
<tr><td>1 1/4</td><td>15.0</td><td>4.4</td><td>0.065/0.120</td><td>0.083/0.120</td></tr>
</table>

A-L Figure 3 – Equations to Determine Coil Specifications

The height of the coil is given by:

$$h = D + s(n-1) < 4in$$

where

D = outer diameter of the tubing

n = number of loops

s = spacing between loops

The outer width of the coil is given by:

$$W = 2r_b + 2D < w$$

where

r_b = bend radius

= radius of inside curve

$w = 2r_b$

= inner width of the bigger tubing

The length of the straight portion is given by:

$$L_s = 100 - 2r_b - 2D$$

For one loop, the length the coil is given by:

$$L_1 = 2(100 - 2D - 2r_b) + 2\pi r_c$$

$$L_1 = 200 - 4D - 4r_b + 2\pi r_c$$

where

r_b = bend radius

r_c = center radius = $(r_b + D/2)$

For multiple loops, the length of the coil is given by:

$$L_n = nL_1 - \pi r_c$$

The fuel capacity of the coil is given by:

$$C = \frac{\pi d^2}{4} L_n$$

where

d = inner diameter = D − 2t

t = thickness of the tubing

The weight of the coil is given by:

$$W = \frac{\pi(D-d)^2}{4} L_n \rho_{ss}$$

where

$\rho_{ss} = 0.289 \frac{lb}{in^3}$ for 316L SS

A-L Figure 4 – Sandvik Certified Material Test Report 1/2" OD Tube [28]

SANDVIK

www.smt.sandvik.com
www.smt.sandvik.com/nafta

Sandvik Materials Technology
Product Area Tube
P.O. Box 1220
Scranton
PA USA 18501
570-585-7500

200724407

Page 1
DDT

Plant Location: 982 Griffin Pond Road, Clarks Summit, PA 18411

Sold To: 8442	Ship To:
SWAGELOK FINANCE ACCTS PAYABLE	WESTERN NEW YORK FLUID SYSTEM
MEDINA OH	TONAWANDA NY
Customer Order No: N44513	Certification Date: 20070604
Sandvik Order No: 23963/1	
Work Order/Lot: 332194	F96860

ASTM A511-04, ASTM A269-04, ASTM A213-06a, ASME SA-213
ASME Section II, 2004 Edition, 2006 Addenda, NACE MR0175 / ISO 15156
First edition 2003-12-15, (Austenitic)

Cold Finished BRIGHT ANNEALED Seamless Tube
Type MT 316/MT 316L/TP316/TP316L   Size: .500" X .065"
Heat: 035592

ANALYSIS %

	C	Si	Mn	P	S	Cr	Ni
Heat	.016	.47	1.71	.026	.011	16.78	12.62
Prod	.017	.46	1.68	.026	.010	16.77	12.51

	Fe	Mo	Co	Al	Pb
Heat		2.00	.130	.005	.002
Prod		2.00	.12	.003	.0001

Mechanical Tests:

Yield Strength 0.2% psi	Yield Strength 0.2% MPa	Yield Strength 1.0% psi	Yield Strength 1.0% MPa	Tensile Strength psi	Tensile Strength MPa	Elongation in % E2"	E10"	E4d	E5d	Reduction Of Area %
40000	275.9		N/A	81000	558.6	64	N/A	N/A	N/A	N/A

Hardness Test Results: 74HRB, 74HRB

Flare Test per ASTM A1016,
No. samples: 2 Result: Acceptable
Flattening Test per ASTM A1016: Acceptable
Tensile Test sample width (1=Full-Size 2=1/2" Strip): 1
Country Of Origin: Canada
All material subjected to a final solution annealing heat treatment with material at a temperature of 1900 deg. F minimum followed by rapid quenching to below 800 deg. F in less than three (3) minutes.
The material has not come in contact with Mercury or Mercury containing compounds.
No welding has been performed on this material.
Material has been eddy current tested in accordance with ASTM A450, ASTM A1016 and is acceptable.
Material has been manufactured/supplied in accordance with Sandvik Materials Technology Quality Manual-Standard Products Revision 10 dated May 29, 2007. Quality system has been approved to ISO 9001:2000.
Certificate produced in accordance with EN 10204 3.1 (formerly EN 10204 3.1.B)
Melt Source: Cogne Acciai Speciali, Italy
Melt Method: Electric Arc Furnace-AOD Refining
This is to certify that the contents of this certificate are correct and accurate as contained in Sandvik's records,

A-L Figure 5 – Sandvik Certified Material Test Report 5/8" OD Tube [28]

SANDVIK
www.smt.sandvik.com
www.smt.sandvik.com/nafta

Sandvik Materials Technology
Product Area Tube
P.O. Box 1220
Scranton
PA USA 18501
570-585-7500

200550443

Page 1
NPD

Plant Location: 982 Griffin Pond Road, Clarks Summit, PA 18411

Sold To: 8442
SWAGELOK FINANCE ACCTS PAYABLE
MEDINA OH

Ship To:
WESTERN NEW YORK FLUID SYSTEM
TONAWANDA NY

Customer Order No: M90372
Certification Date: 20051028
Sandvik Order No: 62715/1
Work Order/Lot: 771852

ASTM A213-04b, ASME SA-213, ASTM A269-04, ASME Section II
2004 Edition

Cold Finished BRIGHT ANNEALED Seamless Tube
Type TP316/TP316L (UNS S31600/S31603)

Size: .625" X .065" AW
Heat: 506909

ANALYSIS %

	C	Si	Mn	P	S	Cr	Ni
Heat	.022	.37	1.53	.025	.008	16.90	12.39
Prod	.02	.36	1.53	.026	.007	16.90	12.38
	Fe	Mo	Al	Pb			
Heat		2.06	.004	.0001			
Prod		2.07	.004	.0001			

Mechanical Tests:

Yield Strength 0.2% psi	Yield Strength 0.2% MPa	Yield Strength 1.0% psi	Yield Strength 1.0% MPa	Tensile Strength psi	Tensile Strength MPa	Elongation in % E2"	Elongation in % E10"	Elongation in % E4d	Elongation in % E5d	Reduction Of Area %
38100	262.8	N/A		86900	599.3	49	N/A	N/A	N/A	N/A
40200	277.2			88400	609.7	47				

Hardness Test Results: 66HRB, 68HRB

Flare Test per ASTM A1016: Acceptable
Flattening Test per ASTM A1016: Acceptable
Tensile Test sample width (1=Full-Size 2=1/2" Strip): 1
Country Of Origin: United States
All material subjected to a final solution annealing heat treatment with material at a temperature of 1900 deg. F minimum followed by rapid quenching to below 800 deg. F in less than three (3) minutes.
The material has not come in contact with Mercury or Mercury containing compounds.
No welding has been performed on this material.
Material has been eddy current tested in accordance with ASTM A450, ASTM A1016 and is acceptable.
Material has been manufactured/supplied in accordance with Sandvik Materials Technology Quality Manual-Standard Products Revision 8 dated August 17, 2005. Quality system has been approved to ISO 9001:2000.
Certificate produced in accordance with EN 10204 (DIN 50049) 3.1.B.
This is to certify that the contents of this certificate are correct and accurate as contained in Sandvik's records, and that all above test results and operations performed are in compliance with the requirements of the purchase order and the specification(s) listed above.

A-L Figure 6 – Sandvik Certified Material Test Report 3/4” OD Tube [28]

SANDVIK

www.smt.sandvik.com
www.smt.sandvik.com/nafta

Sandvik Materials Technology
Product Area Tube
P.O. Box 1220
Scranton
PA USA 18501
570-585-7500

200726684

Page 1
DDT

Plant Location: 982 Griffin Pond Road, Clarks Summit, PA 18411

Sold To: 8442
SWAGELOK FINANCE ACCTS PAYABLE
MEDINA OH

Ship To:
WESTERN NY FLUID SYSTEM TECH
HENRIETTA NY

Customer Order No: N36039
Certification Date: 20070620
Sandvik Order No: 16468/1
Work Order/Lot: 799592
F88068

ASTM A511-04, ASTM A269-04, ASTM A213-05c, ASME SA-213
ASME Section II, 2004 Edition, 2006 Addenda, NACE MR0175 / ISO 15156
First edition 2003-12-15, (Austenitic)

Cold Finished BRIGHT ANNEALED Seamless Tube
Type MT 316/MT 316L/TP316/TP316L
Size: .750" X .065" AW
Heat: 514221

ANALYSIS %

	C	Si	Mn	P	S	Cr	Ni
Heat	.016	.46	1.56	.032	.010	16.76	12.39
Prod	.016	.48	1.54	.032	.008	16.86	12.43

	Fe	Mo	Co	Al	Pb
Heat		2.04	.16	.003	.0001
Prod		2.02	.16	.003	.0001

Mechanical Tests:

Yield Strength 0.2% psi	Yield Strength 0.2% MPa	Yield Strength 1.0% psi	Yield Strength 1.0% MPa	Tensile Strength psi	Tensile Strength MPa	Elongation in % E2"	E10"	E4d	E5d	Reduction Of Area %
35900	247.6	N/A		83500	575.9	54	N/A	N/A	N/A	N/A
36000	248.3			83600	576.6	58				

Hardness Test Results: 67HRB, 71HRB

Flare Test per ASTM A1016,
No. samples: 2 Result: Acceptable
Flattening Test per ASTM A1016: Acceptable
Tensile Test sample width (1=Full-Size 2=1/2" Strip): 1
Country Of Origin: United States
All material subjected to a final solution annealing heat treatment with material at a temperature of 1900 deg. F minimum followed by rapid quenching to below 800 deg. F in less than three (3) minutes.
The material has not come in contact with Mercury or Mercury containing compounds.
No welding has been performed on this material.
Material has been eddy current tested in accordance with ASTM A450, ASTM A1016 and is acceptable.
Material has been manufactured/supplied in accordance with Sandvik Materials Technology Quality Manual-Standard Products Revision 10 dated May 29, 2007. Quality system has been approved to ISO 9001:2000.
Certificate produced in accordance with EN 10204 3.1 (formerly EN 10204 3.1.B)
Melt Source: AB Sandvik MT, Sweden
Melt Method: Electric Arc Furnace-AOD Refining
This is to certify that the contents of this certificate

A-L Figure 7 – Sandvik Certified Material Test Report 1” OD Tube [28]

M A T E R I A L   C E R T I F I C A T E

Date: 02/16/2006 No. 200500721W

SNPR55 80 EC244119

Sandvik Materials Technology, Tubular Products Area
P.O. Box 1220, Scranton, PA USA 18501, Ph: 570-585-7500
Plant Location: 982 Griffin Pond Road, Clarks Summit, PA 18411

Sold To: 8442
SWAGELOK COMPANY
MEDINA OH

Ship To:
WESTERN NY FLUID SYSTEM TECH
HENRIETTA NY

Customer Order No: M77407
Certification Date: 20050105
Sandvik Order No: 45766/4
Work Order/Lot: 765375

ASTM A213-03b, ASME SA-213, ASTM A269-02a, ASME Section II
2004 Edition

Cold Finished BRIGHT ANNEALED Seamless Tube
Type MT 316/MT 316L/TP316/TP316L

Size: 1.000" X .083" AW
Heat: 503964

ANALYSIS %

	C	Si	Mn	P	S	Cr	Ni
Heat	.021	.42	1.60	.027	.009	16.81	12.52
Prod	.018	.41	1.59	.025	.006	16.80	12.47

	Fe	Mo	Al	Pb
Heat		2.18	.007	.0001
Prod		2.17	.003	.0001

Mechanical Tests:

Yield Strength				Tensile Strength		Elongation in %				Reduction Of Area
0.2%		1.0%								
psi	MPa	psi	MPa	psi	MPa	E2"	E10"	E4d	E5d	%
47600	328.3	N/A		86000	593.1	54	N/A	N/A	N/A	N/A
47300	326.2			86700	597.9	53				

Hardness Test Results: 74HRB, 78HRB

Flare Test per ASTM A450: Acceptable
Flattening Test per ASTM A450: Acceptable
Tensile Test sample width (1=Full-Size 2=1/2" Strip): 1
Country Of Origin: United States
100% Positive Material Identification performed
All material subjected to a final solution annealing heat treatment with material at a temperature of 1900 deg.F. minimum followed by rapid quenching to below 800 deg.F in less than three (3) minutes.
The material has not come in contact with Mercury or Mercury containing compounds.
No welding has been performed on this material.
Material has been eddy current tested in accordance with ASTM A450, ASTM A1016 and is acceptable.
Material has been manufactured/supplied in accordance with Sandvik Materials Technology Quality Manual-Standard Products Revision 7 dated September 9, 2004. Quality system has been approved to ISO 9001:2000.
Certificate produced in accordance with EN 10204 (DIN 50049) 3.1.B.
This is to certify that the contents of this certificate are correct and accurate as contained in Sandvik's records, and that all above test results and operations performed are in compliance with the requirements of the purchase order and the specification(s) listed above.

A-L Figure 8 – Equations to Determine Tubing Deflection [32]

The normal strain in each direction is given by:

$$\epsilon_t = \frac{1}{E}[\sigma_t - v(\sigma_l + \sigma_r)]$$
$$\epsilon_l = \frac{1}{E}[\sigma_l - v(\sigma_t + \sigma_r)]$$
$$\epsilon_r = \frac{1}{E}[\sigma_r - v(\sigma_t + \sigma_l)]$$

where

v = Poisson's ratio = 0.3 for metals

E = Elastic Modulus

σ_t = Tangential stress

σ_l = longitudinal stress

σ_r = Radial stress

The strain in each direction is given by:

$$\epsilon_t = \frac{\Delta t}{t_0}$$
$$\epsilon_l = \frac{\Delta l}{l_0}$$
$$\epsilon_r = \frac{\Delta r}{r_0}$$

where

$\Delta t, \Delta l, \Delta r$ = deflection of each parameter

t_0, l_0, r_0 = the original values of the parameters

t_0 = circumference

l_0 = length

r_0 = radius

The total deflection can be calculated using the Pythagorean Theorem given by:

$$\Delta_{tot} = \sqrt{\Delta t^2 + \Delta l^2 + \Delta r^2}$$

A-L Figure 9 – Swagelok Quote for 10,000 Tubing Coils [26],[28]

Swagelok

WESTERN NEW YORK FLUID SYSTEM TECHNOLOGIES
Buffalo - Rochester - Syracuse

Per Your Request for Quote…

Co:	Binghamton University	**From:**	Dino Dutcher
Attn:	Matt Grenier	**Pages:**	1
Fax:		**Date:**	3/24/10
Phone:	607-244-0390	**Re:**	Hydrogen Tank Project

Thank you for the opportunity to provide the following quotation. Per our discussion, Swagelok Tubing conforms to ASTM A213/A269 (refer to Swagelok Catalog MS-01-153-SCS).

Part #	Description	List Price	Qty	Discount	Extended Cost
SS-1610-C	S.S. 1" Tube Cap	$31.70	10000	26%	$234,580.00
SS-1410-C	S.S. 7/8" Tube Cap	26.60	10000	26%	$196,840.00
SS-1010-C	S.S. 5/8" Tube Cap	$12.45	10000	26%	$92,130.00
SS-810-C	S.S. 1/2" Tube Cap	$10.36	10000	26%	$76,664.00
SS-1610-1-12	S.S. 1" OD Tube x 3/4" MNPT Connector	$44.50	10000	26%	$329,300.00
SS-1410-1-12	S.S. 7/8" OD Tube x 3/4' MNPT Connector	$41.90	10000	26%	$310,060.00
SS-1010-1-12	S.S. 5/8" OD Tube x 3/4" MNPT Connector	$21.90	10000	26%	$162,060.00
SS-810-1-12	S.S. 1/2" OD Tube x 3/4" MNPT Connector	$20.40	10000	26%	$150,960.00
SS-T16-S-120-20	S.S. 1" OD x .120 wall tubing, per foot	$33.80	1250000	40%	$25,350,000.00
SS-T14-S-109-20	S.S. 7/8' OD x .109 wall tubing, per foot	$40.60	1250000	40%	$30,450,000.00
SS-T10-S-095-20	S.S. 5/8" OD x .095 wall tubing, per foot	$17.73	1000000	40%	$10,638,000.00
SS-T8-S-083-20	S.S. 1/2" OD x .083 wall tubing, per foot	$16.20	750000	40%	$7,290,000.00

- FOB: Syracuse, New York
- TERMS: NET 30
- Prices are based on single shipment and valid for 30 days from date of quote
- Availability subject to prior sales
- Deliveries quoted will take effect after acceptance of order (ARO)
- Requests for return of SWAGELOK items must come within 60 days of receipt of order. Standard price list items with a good prior volume sales history and in perfect condition may be returned and applied as a Credit Memo, less a 20% restocking charge.
- Factory specials are non-cancelable and non-returnable
- Please note the SWAGELOK Limited Lifetime warranty applies to Swagelok components only. The other manufacturer's warranty applies to their components.

* Safe Product Selection:
When selecting a product, the total system design must be considered to ensure safe, trouble-free performance.
Function, material compatibility, adequate ratings, proper installation, operation and maintenance are the responsibilities of the system designer and user.

1. We reserve the right to alter our quote if it is not accepted in the quantity quoted.
2. We reserve the right to alter our quote if we experience a National price change.
3. All agreements are contingent upon strikes, accidents and other delays unavoidable or beyond our control.

Buffalo
Phone 716.875.9365
Fax 716.877.6903

Rochester
Phone 585.359.8470
Fax 585.359.8475
Email : info@wnyfst.swagelok.com

Syracuse
Phone 315.437.1287
Fax 315.437.3825

A-L Figure 10 – Handy & Harman Quote for 10,000 Tubing Coils [34]

Handy & Harman

Quote

ISSUED: 2010-03-22
END DATE: 2010-04-22
QUOTE #: Project Pricing

Handy & Harman Tube Co. - Camdel Metal
124 Vepco Blvd ▪ Camden, DE 19934
Tel: 302.697.9521 ▪ Fax: 302.697.7405

Binghamton University

Attention: Matt Grenier
Email: mgrenie1@binghamton.edu

QUANTITY	UNIT	PRODUCT NAME	ALLOY	DESCRIPTION	UNIT PRICE
125,000-1,250,000	Feet	.500" OD X .083" Wall	316/316L	Smls, ASTM A269 Annealed 450' Min Coil Lengths	$4.01
125,000-1,250,000	Feet	.625" OD X .095" Wall	316/316L	Smls, ASTM A269 Annealed 325' Min Coil Lengths	$5.98
100,000-1,000,000	Feet	.875" OD X .120" Wall	316/316L	Smls, ASTM A269 Annealed 175' Min Coil Lengths	$8.90
75,000-750,000	Feet	1.00" OD X .134" Wall	316/316L	Smls, ASTM A269 Annealed 125' Min Coil Lengths	$9.67

Notes: Surcharges included with price.

SHIPPING QUANTITIES +/- 10% OF ORDER UNLESS PREVIOUSLY AGREED UPON.

TERMS: NET 30 (Pending Credit Approval)
DELIVERY: 8-10 weeks ARO

SHIPPING: EXW
Thank you, *JoAnna Troubetaris*

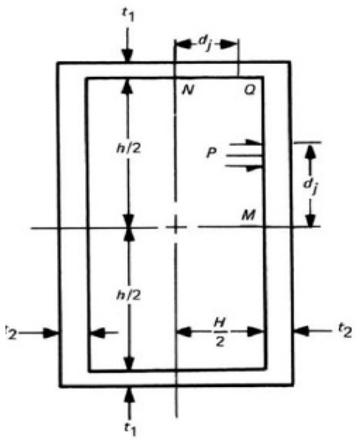
A-L Figure 11 – Handy & Harman Quote Terms and Conditions [34]

TERMS and CONDITIONS

1. PRICES - Prices on material covered may be adjusted at Seller's option. Seller may adjust prices (a) to those Seller has in effect of the time of shipment; (b) in the event that Seller cancels, to cover labor expanded, material procured, processed or partly processes, and reasonable overhead expenses applicable thereto; and in the event of (c) delays, acceptable by Seller or (d) Seller's specific consent to deliver material beyond a 12 month period from the initial contract date, to cover the additional charges involved through (c) or (d).
2. TAXES - The Buyer shall reimburse the Seller for all taxes, excises or other charges which the Seller may be required to pay to any Government (National, State or Municipal) upon the sale, production or transportation of the products sold hereunder.
3. DELIVERY - Unless otherwise specified by written agreement, shipments shall be FOB common carrier most convenient to the Seller's plant. Seller shall not be liable for any delay in the delivery or shipment of products or for any damage suffered to the Buyer by reason of such delay. When such delay I directly or indirectly caused by or in any manner arises from fires, floods, accidents, riots, war, government interferences, strikes, shortages of labor or materials, inadequate transportation facilities or any other cause or causes beyond its reasonable control. Nor shall Seller be under obligation to insure any material affected by their contract. Acceptance by carriers of the materials shall constitute delivery, and all risk and loss thereafter is assumed by the Buyer.
4. STANDARDS - The product furnished hereunder shall be produced and their amounts determined in accordance with Seller's standard practices. Buyer agrees that among Seller's standards practices are included the measurement of tubing by weight and converting the weight to footage or pieces, in addition to direct counting. All products however, including those produced to meet exact specification, shall be subject to trade practices, tolerances and variances.
5. CANCELLATIONS - Buyer may not cancel this contract or any part thereof unless by written agreement. Seller reserves its right to cancel this contract owing to (a) its inability to perform due to war, fire, flood, storms, strikes, lock-outs, riots, civil commotions, embargoes and acts of God; (b) a change in the identity of the Buyer; (c) failure on the part of the buyer to pay within 30 days any amount owing to Seller on account hereof; or (d) for any other reason that Seller finds it impossible, despite its best efforts, to perform hereunder.
6. WARRANTIES - Seller neither makes nor assumes any warranties, expressed or implied, unless by a written agreement executed by it subsequent to the making hereof. Without limiting the foregoing in any way, Seller makes no warranty of fitness of material for any purpose. Anything herein to the contrary notwithstanding, should Buyer be entitled to collect damages hereunder or in any connection whatsoever with the transaction herein represented. Buyer agrees that that damages shall not consist of any sum or sums or anything other than a replacement of the tubing covered hereby, or its dollar value of the time(s) of occurrence of any alleged damage.
7. WAIVER - The failure of Seller to insist, in any one or more instances, upon the performance of any of the terms, covenants or conditions of this contract or to exercise any right hereunder shall not be construed as a waiver or relinquishment of the future performance of any such term, covenant or condition or the future exercise of such rights, nor shall it be deemed to be a waiver or relinquishment of any other term, covenant or condition, or the exercise of any other rights under this contract.
8. MODIFICATION OF CONTRACT - This document contains the entire contract between Buyer and Seller in relation to the items listed on it an supersedes any documents or any understanding, verbal or written, between the parties prior hereto. From the date hereof onward no terms and conditions other than those states herein, and no agreement or understanding, oral or written, in any way purporting to modify these terms or conditions, whether contained in Buyers purchase or shipping release forms, or elsewhere, shall be binding on Seller hereinafter unless made in writing and signed by Seller's authorized representative.
9. ERRORS - Anything to the contrary herein notwithstanding, Seller reserves the right to correct all stenographic or arithmetical errors which may appear on this contract, or any invoice relating thereto.
10. ASSIGNMENTS - Buyer, in making this contract, does not do so in dependence upon the performance thereof by Seller but merely looks to Seller to provide to Buyer at the times and places agreed, and for the amounts specified, all subject to the terms hereof, the materials as described herein, and therefore accords to Seller a free right to assign to such assignees as it may see fit or all of in obligation hereunder and to retain Seller's right to collect that amounts due from the Buyer hereunder.
11. APPORTIONMENT - In the event that for any reason whatsoever Seller has performed some portion of its obligations hereunder and is prevented from performing the balance thereof, including by reason of Seller's cancellation hereunder pursuant hereto. Seller reserves the right to collect from Buyer and appropriate proportionment of the total amount Buyer agrees to pay Seller.
12. ACCELERATION - Should this agreement comprehend deliveries in installments by Seller, and after demand therefore Buyer fails to pay Seller for any such installment within 30 days. Seller at his election may forthwith declare the amount owing for a portion or all of the future installments as immediately due, and may collect therefore upon tendering delivery of such portion or whole of the material.

APPENDIX M: ADDITIONAL MATERIALS
(RECTANGULAR TANK)

A-M Figure 1 – ASME Equations for Unreinforced Rectangular Vessels [33]



13-7 UNREINFORCED VESSELS OF RECTANGULAR CROSS SECTION

For the equations in these paragraphs, the moments and moments of inertia are calculated on a per-unit-width basis. That is, $I = bt^3/12$, where $b = 1.0$. The moments M_A and M_r have dimensions [Force × Length/Length] = Force. See para. 13-4(k).

(a) Vessel per Fig. 13-2(a) Sketch (1)

(1) Membrane Stress

Short-Side Plates

$$S_m = Ph/2t_1 \tag{1}$$

Long-Side Plates

$$S_m = PH/2t_2 \tag{2}$$

(2) Bending Stress

Short-Side Plates

$$(S_b)_N = \frac{Pc}{12I_1}\left[-1.5H^2 + h^2\left(\frac{(1+\alpha^2K)}{1+K}\right)\right] \tag{3}$$

$$(S_b)_Q = \frac{Ph^2c}{12I_1}\left(\frac{1+\alpha^2K}{1+K}\right) \tag{4}$$

Long-Side Plates

$$(S_b)_M = \frac{Ph^2c}{12I_2}\left[-1.5 + \left(\frac{(1+\alpha^2K)}{1+K}\right)\right] \tag{5}$$

$$(S_b)_Q = \frac{Ph^2\,c}{12I_2}\left(\frac{1+\alpha^2\,K}{1+K}\right) \tag{6}$$

(3) Total Stress

Short-Side Plates

$$(S_T)_N = \text{Eq. (1)} + \text{Eq. (3)} \tag{7}$$

$$(S_T)_Q = \text{Eq. (1)} + \text{Eq. (4)} \tag{8}$$

Long-Side Plates

$$(S_T)_M = \text{Eq. (2)} + \text{Eq. (5)} \tag{9}$$

$$(S_T)_Q = \text{Eq. (2)} + \text{Eq. (6)} \tag{10}$$

(4) An example illustrating use of these rules is given in 13-17(a).

α = rectangular vessel parameter = H/h

t_1 = thickness of short-side plates of vessel

t_2 = thickness of long-side plates of vessel

K = vessel parameter $(I_2/I_1)\alpha$

I_1 = moment of inertia of strip of thickness[2] t_1

I_2 = moment of inertia of strip of thickness[2] t_2

P = internal design pressure (see UG-21)

c = distance from neutral axis of cross section to extreme fibers (see c_i and c_o). The appropriate c_i or c_o value shall be substituted for the c term in the stress equations.

h = inside length of long side of unstayed rectangular vessel; or dimension perpendicular to the H dimension in stayed vessels as shown in Fig. 13-2(a) sketches (7), (8), (9), and (10) in which case h may be greater than, equal to, or less than H,
= $2(L_2 + L_{21})$ for equations in 13-8(d) for Fig. 13-2(a) sketches (5) and (6)
= $2L_2$ for equations in 13-8(d) for Fig. 13-2(b) sketch (2)

H = inside length of short side of rectangular vessel
= $2(L_1 + L_{11})$ for equations in 13-8(d) for Figs. 13-2(a) sketches (5) and (6)

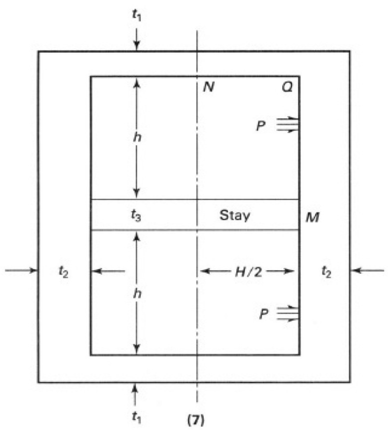
A-M Figure 2 – Stress Analysis Excerpt for Rectangular Tank (316L SS)

Unreinforced rectangular tank with six sections

Length (in)	Width (in)	Height (in)	Wall (in)	Volume (in^3)	Volume (L)	t1	t2	H	h	P	c	I1	I2	α	K
252	3	2	0.5	504	8.26	0.5	0.5	2	1	6,250	0.25	0.0104	0.0104	2.00	2
252	3	2.5	0.5	756	12.39	0.5	0.5	2	2	6,250	0.25	0.0104	0.0104	1.33	1
252	3	3	0.5	1008	16.52	0.5	0.5	2	2	6,250	0.25	0.0104	0.0104	1.00	1
252	3	3.5	0.5	1260	20.66	0.5	0.5	2	3	6,250	0.25	0.0104	0.0104	0.80	1
252	3	4	0.5	1512	24.79	0.5	0.5	2	3	6,250	0.25	0.0104	0.0104	0.67	1
252	3	4.5	0.5	1764	28.92	0.5	0.5	2	4	6,250	0.25	0.0104	0.0104	0.57	1
252	3	5	0.5	2016	33.05	0.5	0.5	2	4	6,250	0.25	0.0104	0.0104	0.50	1
252	3	5.5	0.5	2268	37.18	0.5	0.5	2	5	6,250	0.25	0.0104	0.0104	0.44	0
252	3	6	0.5	2520	41.31	0.5	0.5	2	5	6,250	0.25	0.0104	0.0104	0.40	0
252	3	6.5	0.5	2772	45.44	0.5	0.5	2	6	6,250	0.25	0.0104	0.0104	0.36	0
252	3	7	0.5	3024	49.57	0.5	0.5	2	6	6,250	0.25	0.0104	0.0104	0.33	0
252	3	7.5	0.5	3276	53.70	0.5	0.5	2	7	6,250	0.25	0.0104	0.0104	0.31	0
252	3	8	0.5	3528	57.84	0.5	0.5	2	7	6,250	0.25	0.0104	0.0104	0.29	0

Membrane (SS)	Membrane (LS)	Bending (SSM)	Bending (SSC)	Bending (LSM)	Bending (LSC)	Total Stress (SSM)	Total Stress (SSC)	Total Stress (LSM)	Total Stress (LSC)
6250	12500	-37500	37500	18750	37500	-31250	43750	31250	50000
9375	12500	-34375	40625	-1562	40625	-25000	50000	10938	53125
12500	12500	-25000	50000	-25000	50000	-12500	62500	-12500	62500
15625	12500	-9375	65625	-51563	65625	6250	81250	-39063	78125
18750	12500	12500	87500	-81250	87500	31250	106250	-68750	100000
21875	12500	40625	115625	-114063	115625	62500	137500	-101563	128125
25000	12500	75000	150000	-150000	150000	100000	175000	-137500	162500
28125	12500	115625	190625	-189063	190625	143750	218750	-176563	203125
31250	12500	162500	237500	-231250	237500	193750	268750	-218750	250000
34375	12500	215625	290625	-276563	290625	250000	325000	-264063	303125
37500	12500	275000	350000	-325000	350000	312500	387500	-312500	362500
40625	12500	340625	415625	-376563	415625	381250	456250	-364063	428125
43750	12500	412500	487500	-431250	487500	456250	531250	-418750	500000

A-M Figure 3 – ASME Equations for Stayed Rectangular [33]



13-9 STAYED VESSELS OF RECTANGULAR CROSS SECTION [FIG. 13-2(a) SKETCHES (7) AND (8)]

For the equations in these paragraphs, the moments of inertia are calculated on a per-unit-width basis. That is, $I = bt^3/12$, where $b = 1.0$. See para. 13-4(k).

(a) Three types of stayed construction are considered as shown on Fig. 13-2(a) sketches (7) through (10). In these types of construction the staying members may be plates welded to the side plates for the entire length of the vessel; or, the stays may be bars of circular cross section fastened to the side plates on a uniform pitch. For the former case, the stay plates shall not be constructed so as to create pressure containing partitions (see UG-19 for vessels containing more than one independent pressure chamber). For the latter case the rules of UG-47(a), UG-48, UG-49, and UG-50 must be met. End plates are subject to the rules of 13-4(f).

(b) Vessel Stayed by a Single Plate. Figure 13-2(a) sketch (7) shows a vessel with a central stay plate.

(1) Membrane Stress

Short-Side Plates

$$S_m = \frac{Ph}{4t_1}\left\{4 - \left[\frac{2 + K(5 - \alpha^2)}{1 + 2K}\right]\right\} \qquad (1)$$

Long-Side Plates

$$S_m = PH/2t_2 \qquad (2)$$

Stay Plate

$$S_m = \frac{Ph}{2t_3}\left[\frac{2 + K(5 - \alpha^2)}{1 + 2K}\right] \qquad (3)$$

(2) Bending Stress

Short-Side Plates

$$(S_b)_N = \frac{Pc}{24I_1}\left[-3H^2 + 2h^2\left(\frac{1 + 2\alpha^2 K}{1 + 2K}\right)\right] \qquad (4)$$

$$(S_b)_Q = \frac{Ph^2 c}{12I_1}\left(\frac{1 + 2\alpha^2 K}{1 + 2K}\right) \qquad (5)$$

Long-Side Plates

$$(S_b)_M = \frac{Ph^2 c}{12I_2}\left[\frac{1 + K(3 - \alpha^2)}{1 + 2K}\right] \qquad (6)$$

$$(S_b)_Q = \frac{Ph^2 c}{12I_2}\left(\frac{1 + 2\alpha^2 K}{1 + 2K}\right) \qquad (7)$$

(3) Total Stress

Short-Side Plates

$$(S_T)_N = \text{Eq. (1)} + \text{Eq. (4)} \qquad (8)$$

$$(S_T)_Q = \text{Eq. (1)} + \text{Eq. (5)} \qquad (9)$$

Long-Side Plates

$$(S_T)_M = \text{Eq. (2)} + \text{Eq. (6)} \qquad (10)$$

$$(S_T)_Q = \text{Eq. (2)} + \text{Eq. (7)} \qquad (11)$$

Stay Plate

$$S_T = \text{Eq. (3)} \qquad (12)$$

t_3 = thickness or diameter of staying member

α = rectangular vessel parameter = H/h

t_1 = thickness of short-side plates of vessel

t_2 = thickness of long-side plates of vessel

K = vessel parameter $(I_2/I_1)\alpha$

I_1 = moment of inertia of strip of thickness^2 t_1

I_2 = moment of inertia of strip of thickness^2 t_2

P = internal design pressure (see UG-21)

c = distance from neutral axis of cross section to extreme fibers (see c_i and c_o). The appropriate c_i or c_o value shall be substituted for the c term in the stress equations.

h = inside length of long side of unstayed rectangular vessel; or dimension perpendicular to the H dimension in stayed vessels as shown in Fig. 13-2(a) sketches (7), (8), (9), and (10) in which case h may be greater than, equal to, or less than H,
= $2(L_2 + L_{21})$ for equations in 13-8(d) for Fig. 13-2(a) sketches (5) and (6)
= $2L_2$ for equations in 13-8(d) for Fig. 13-2(b) sketch (2)

H = inside length of short side of rectangular vessel
= $2(L_1 + L_{11})$ for equations in 13-8(d) for Figs. 13-2(a) sketches (5) and (6)

A-M Figure 4 – Stress Analysis Excerpt for Stayed Rectangular Tank (316L SS)

Reinforced (1-stay) tank with six sections

Length (in)	Width (in)	Height (in)	Wall (in)	Volume (in^3)	Volume (L)	t1	t2	t3	H	h	P	c	I1	I2	α	K
252	3	2	0.5	252	4.13	0.50	0.50	0.50	2.0	0.25	6,250	0.25	0.0104	0.0104	8.00	8.00
252	3	2.5	0.5	504	8.26	0.50	0.50	0.50	2.0	0.50	6,250	0.25	0.0104	0.0104	4.00	4.00
252	3	3	0.5	756	12.39	0.50	0.50	0.50	2.0	0.75	6,250	0.25	0.0104	0.0104	2.67	2.67
252	3	3.5	0.5	1008	16.52	0.50	0.50	0.50	2.0	1.00	6,250	0.25	0.0104	0.0104	2.00	2.00
252	3	4	0.5	1260	20.66	0.50	0.50	0.50	2.0	1.25	6,250	0.25	0.0104	0.0104	1.60	1.60
252	3	4.5	0.5	1512	24.79	0.50	0.50	0.50	2.0	1.50	6,250	0.25	0.0104	0.0104	1.33	1.33
252	3	5	0.5	1764	28.92	0.50	0.50	0.50	2.0	1.75	6,250	0.25	0.0104	0.0104	1.14	1.14
252	3	5.5	0.5	2016	33.05	0.50	0.50	0.50	2.0	2.00	6,250	0.25	0.0104	0.0104	1.00	1.00
252	3	6	0.5	2268	37.18	0.50	0.50	0.50	2.0	2.25	6,250	0.25	0.0104	0.0104	0.89	0.89
252	3	6.5	0.5	2520	41.31	0.50	0.50	0.50	2.0	2.50	6,250	0.25	0.0104	0.0104	0.80	0.80
252	3	7	0.5	2772	45.44	0.50	0.50	0.50	2.0	2.75	6,250	0.25	0.0104	0.0104	0.73	0.73
252	3	7.5	0.5	3024	49.57	0.50	0.50	0.50	2.0	3.00	6,250	0.25	0.0104	0.0104	0.67	0.67
252	3	8	0.5	3276	53.70	0.50	0.50	0.50	2.0	3.25	6,250	0.25	0.0104	0.0104	0.62	0.62
240	4	2	0.5	360	5.90	0.50	0.50	0.50	3.0	0.25	6,250	0.25	0.0104	0.0104	12.00	12.00

Total Stress (SSM)	Total Stress (SSC)	Total Stress (LSM)	Total Stress (LSC)
-3171	71829	-9881	59605
-16667	58333	-5208	57292
-21066	53934	1439	55715
-22500	52500	10000	55000
-22121	52879	20424	55246
-20265	54735	32670	56534
-17047	57953	46705	58933
-12500	62500	62500	62500
-6628	68372	80031	67281
577	75577	99279	73317
9130	84130	120226	80642
19048	94048	142857	89286
30344	105344	167161	99273
-5531	163219	-34094	126781

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