

DEVELOPMENT OF A THERMAL VACUUM SYSTEM FOR APPLICATION
IN SPACE HARDWARE ENVIRONMENTAL TESTING

by

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DEDICATION

I dedicate this work to my parents, whose constant love and support have made me the person I am today.

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ABSTRACT

Since 2001 the Space Science and Engineering Lab (SSEL) at Montana State University has designed and built a variety of space hardware, as well as developed the facilities necessary for environmental testing of flight hardware. In late 2005, the SSEL began developing a system to simulate the vacuum environment of near-space which would allow for rudimentary outgassing testing as well as thermal testing of electronics and spacecraft components. To truly test hardware and validate hardware analysis and design, the ability to cycle between the expected temperature extremes in a vacuum environment was essential.

The usage and operation of thermal vacuum systems was investigated, requirements for a thermal vacuum system were defined, and possible design options were considered. A Finite Element Analysis (FEA) was performed to predict the heat load in Watts on the system when a 40 kg nanosatellite (50 cm x 50 cm x 60 cm) was cycled between -40°C and $+80^{\circ}\text{C}$ at rates of temperature change from $1^{\circ}\text{C}/\text{min}$ to $5^{\circ}\text{C}/\text{min}$. Additional research, analysis, and design was performed on a thermal shroud surrounding the same nanosatellite and operating under identical conditions. Radiation heat transfer between the satellite on the shroud's inside and the vacuum chamber on the shroud's outside was calculated using the radiation network approach. The LU decomposition method was used to solve the resulting set of simultaneous equations.

From the results, a design was selected for the system base plate on which the nanosatellite rested, with the capability of sinking or sourcing 833 W of heat power. Similarly, the thermal shroud was designed to sink or source up to 704 W through the shroud body, and up to 229 W through the shroud top. Testing was performed to validate both the FEA model and the physical hardware. Temperature measurements were taken during system testing to validate the design, and as a means to compare the FEA model of the base plate and the radiation heat transfer calculations with the performance of the system hardware. The results indicated that the system functioned as designed, that it met the design requirements, and that it was capable of completely and safely testing satellites and other space hardware.

INTRODUCTION

Motivation

The Space Science and Engineering Lab (SSEL) at Montana State University (MSU) was formed in 2001 with the goal of educating students in the design and development of space hardware. Since then, hundreds of students from a variety of engineering disciplines and the physical sciences have participated in the concept development, engineering, fabrication, testing, and operation of numerous satellite, rocket, and payload projects. While some programs were purely educational, others proceeded through the entire design process and were delivered for launch into Low Earth Orbit (LEO).

During the design process students became intimately familiar with the design and operation of their particular project, including the testing required to verify its readiness for flight. Some of the tests required expensive and unique equipment which neither the SSEL nor MSU possessed. Thus, many relationships were built with industry partners to use their facilities and equipment. In many cases the required testing was far less complicated than the typical testing that these facilities would perform, so the testing was offered to the SSEL for a significantly reduced price, or was free.

As the frequency of these environmental tests increased, so too did the scope of SSEL projects. Thermal vacuum testing was one of several tests performed on flight hardware. The travel to testing facilities and the testing itself cost \$10k to \$20k and at some point the recurring cost of using outside facilities was expected to surpass the cost of purchasing and operating in-house equipment. An inexpensive commercially built system would have cost at least \$150k, would have included capabilities far

beyond the SSEL's needs and required resources that the SSEL did not have. Thus, a less expensive custom design was sought which would more closely match the SSEL's needs and available resources.

Owning and operating an in-house thermal vacuum system had several distinct advantages. First, operational costs at MSU would be significantly less than those incurred at industry facilities while providing the same testing capability. Second, once a system was established at MSU, other groups would seek to use SSEL's facilities and reduce their testing costs, thereby helping the SSEL to offset any costs above that of a commercial system. Finally, MSU students would be directly involved in the design, fabrication, and operation of the Thermal Vacuum System for simulating the near space thermal environment.

The Space Environment

The space environment is classified into regions by the proximity to Earth. Because the SSEL develops space hardware designed for LEO satellites, this document focuses on this region between altitudes of 300 and 1000 km. In LEO satellites have an orbital velocity of approximately 7 km/s, with an orbital period of approximately 90 minutes. Depending on the orbital inclination and eccentricity of the orbit, a satellite can experience a wide variety of thermal variation.

With the exception of some higher inclination orbits, a satellite will pass into the Umbra (the shadow of the Earth) for about 30 minutes and back into the sunlight once per orbit. At inclinations above 70° a satellite will pass through the Umbra during most orbits but will occasionally be in sunlight for one or more complete orbits because of orbital precession. Seasonal precession will also cause the satellite to have several orbits in complete sunlight. When a satellite passes into or out of the Umbra it

experiences rapid temperature changes. Nearing the Umbra a satellite is at its highest temperature, sometimes as high as 70°C. Leaving the umbra temperatures can reach as low as -30°C. Satellites in dawn-dusk sun-synchronous orbits are constantly at the hot end of the spectrum. The solar flux, Earth Albedo, extreme cold of deep space, and orbital trajectory play a role in a satellite's temperature as a function of time in all orbits. In addition, the vacuum of space also plays a large role.

At atmospheric pressure, electronics are able to dissipate heat via forced or natural convection with air, as well as through conduction and radiation. Personal computers use fans to blow air across finned heat sinks to remove heat from electronics, and household radiators rely on natural convection to circulate heat throughout a room. While convenient and reliable on Earth, convection heat transfer in space is negligible because the mean free path between particles is on the order of 10 km at a pressure of 100 pTorr. On Earth the mean free path of atmospheric air is roughly 10 μm at a pressure of 760 Torr at sea level [1] [2]. Thus, engineers can rely only on conduction and radiation when designing space hardware, and it follows that thermal simulation of LEO satellites must also rely only on these two modes of heat transfer.

Given the nature of the thermal environment in space, it is easy to see why the thermal design of a satellite is crucial to the survival and operation of the instrumentation and electronics on board. If a component or spacecraft subsystem were to fail prematurely, there is no way to retrieve and repair small spacecraft in orbit. For these reasons it is essential to perform thorough ground testing prior launch.

Space Hardware Environmental Testing

Thorough ground testing will validate all aspects of a spacecraft design for its desired lifetime. Vibration and shock testing, along with thermal testing, ensures the

structural integrity of a spacecraft during launch and orbit, while thermal testing also validates electrical functionality and reliability at the expected temperature extremes and transitions during orbit. Gilmore [2] describes three standard types of thermal testing: thermal cycle, thermal vacuum, and thermal balance testing – each having a unique purpose.

Thermal Cycle Testing

Thermal cycle testing is performed during the the design and development process at the component or subsystem level. As the first environmental test, and because it is significantly less expensive to conduct than thermal vacuum and balance tests, its primary purpose is to quickly expose faulty designs and manufacturing defects. This test is done at atmospheric pressure which makes convection the primary mode of heat transfer and allows for rapid cycling between the temperature extremes. It also creates large temperature gradients. Early in the design process the number of cycles will often be high (10 to 20) to thoroughly stress the hardware, while the number drops off to between 5 and 10 as the hardware matures. If there is a failure before the last cycle, the design engineer will make changes and repeat the test until the article passes the test.

Thermal Vacuum Testing

Thermal vacuum testing follows thermal cycle testing, sometimes at the subsystem level, but more often at the fully integrated system level because of the cost and equipment needed to perform multiple cycles. Its primary purpose is to verify the functionality of the test article in vacuum and at the temperature extremes. At this point any major design errors and manufacturing defects have been found and corrected through thermal cycle testing. By operating a test article in vacuum (below

100 μ Torr), convection no longer plays a significant role in heat transfer and the test article is therefore tested in an environment that more closely matches what it will experience in space.

In the design phase, the thermal engineer chooses materials, surface finishes, and fastener designs to minimize the thermal gradients that the satellite will experience during its life in orbit, keeping in mind that each component in the satellite has temperature limits. Operational temperature limits are the maximum and minimum temperatures under which a component, subsystem, or system can operate; survival temperatures are the maximum and minimum temperatures under which a component, subsystem, or system can survive when not in use. A thermal vacuum test verifies that a component, subsystem, or system will function within and at the expected operational temperature extremes.

Thermal Balance Testing

The final test of the complete system is the thermal balance test. In the design phase the thermal engineer builds a computer model of the entire satellite, including factors such as geometry, material properties, surface finish, contact resistances, heat generation, absorbed and radiated heat, satellite orbit, and satellite operation modes. The goal of the thermal modeling is to predict as closely as possible the temperature distribution throughout the satellite over the course of its mission, especially the worst case scenarios. The thermal balance test exposes a satellite to an environment as close as possible to that which it will experience in orbit while it is operated in its various modes. Temperature is measured at critical locations and compared to the temperatures predicted by the model. The thermal control subsystem of the satellite is also tested, to ensure that it can maintain the temperature of critical components within their operational range, and also that its design specifications, as predicted

by the thermal model, are correct. The thermal model and the satellite design are validated in this way.

LITERATURE REVIEW

Established Thermal Vacuum Practices

While the focus of this thesis was the design and testing of the thermal system, the vacuum system and its interaction with the thermal system could not be ignored. Their interaction and the degree to which they were able to function together determined the overall performance of the system. Convection was removed as a mode of heat transfer because of the vacuum environment, and therefore limited the heat transfer modes in the system to conduction and radiation.

Most if not all of today's standards for thermal vacuum testing are derived from the military requirements developed over the past several decades, specifically the several versions of MIL-STD-1540 [3] that have been released since the mid-1970's. Gilmore [2] dedicates an entire chapter to thermal testing of spacecraft and their various components and relies heavily on the military standards. Military standards require a pressure of 100 μ Torr or less to remove the effects of convection from testing. This same pressure is required by NASA standards [2]. Both NASA and the Military Standards require very high levels of testing for confidence in flight hardware. Military testing levels as well as their dependence on thermal modeling are discussed at length by Gilmore.

The temperature extremes used in spacecraft and component level testing are derived from thermal modeling performed during the design phase of a project. Early in the design phase there is a high degree of uncertainty in the predicted temperature extremes, but this tends to decrease over the design life of a project. Once the nominal temperature extremes have been established for a component or spacecraft, two margins are added. The first is an Acceptance margin of 11°C and the second is

a Qualification margin of 10°C. The Acceptance test temperature for the hot side is the greater of 61°C or the sum of the predicted hottest temperature, the hot side uncertainty margin, and the Acceptance margin. The Qualification test temperature is 71°C or the Acceptance test temperature plus 10°C, whichever is greater. On the cold side, the Acceptance test temperature is -24°C or the predicted cold side temperature, less the cold side uncertainty margin, less the Acceptance margin, whichever is colder. The Qualification test temperature is -34°C or the Acceptance test temperature less the Qualification margin, whichever is colder. These temperatures and margins are illustrated in Figure 2.1.

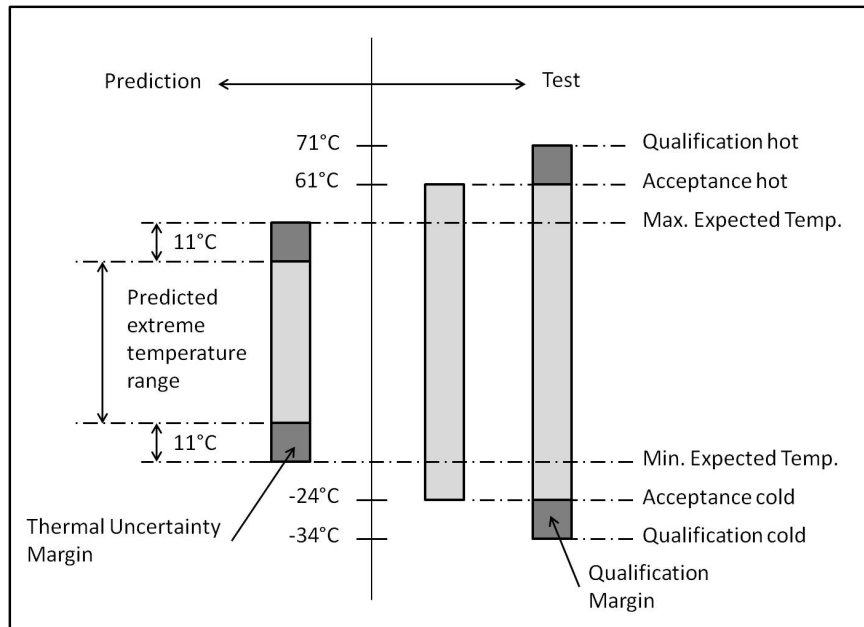


Figure 2.1: Acceptance and Qualification Temperatures

Qualification testing demonstrates the ruggedness of flight hardware and its design, and subjects the hardware to conditions more extreme than expected during mission life, including temperature, duration, and number of cycles. A unit or subsystem is typically temperature cycled 24 times in thermal vacuum if no thermal

cycles are performed at ambient pressure. However, this number is reduced to 3 if 24 thermal cycles are performed prior to thermal vacuum testing. At each temperature extreme the unit is held there (soaked) for 6 hours on the first and last cycles and 1 hour on every cycle in between. In comparison, a full satellite system is temperature cycled at least 4 times and soaked for 8 hours on the first and last cycle and 4 hours on intermediate cycles. Although a satellite will experience far more thermal cycles in orbit than it does during testing, testing at the extreme temperatures mitigates the risk associated with fewer temperature cycles. Because of the severity of the test, hardware that undergoes Qualification testing is ineligible for flight.

Identical units undergo Acceptance testing at the less severe levels later in the testing process and are therefore available for flight. For Acceptance testing, a unit or subsystem is typically temperature cycled 8 times in thermal vacuum if no thermal cycles are performed at ambient pressure. However, this number is reduced to 1 if 8 thermal cycles are performed prior to thermal vacuum testing. At each temperature extreme the unit is soaked for 6 hours on the first and last cycles and 1 hour on every cycle in between. In comparison, a full satellite system is temperature cycled at least 4 times and soaked for 8 hours on the first and last cycle, and 4 hours on intermediate cycles. The cost associated with a Qualification and Acceptance testing program has prompted the creation of alternative testing requirements and procedures, namely Protoqualification testing.

A protoqualification unit undergoes a modified version of qualification testing at levels sufficient to validate the design and hardware, but not so severe as to be ineligible for flight, thereby saving cost. Organizations like NASA utilize the protoqualification process and have established their own requirements for this type of testing. NASA defines the testing temperatures in a similar way as the Military requirements, but uses slightly different terminology. An Allowable Flight Tempera-

ture (AFT) range is determined through thermal modeling, and a Flight Acceptance (FA) margin of 5°C is added to both ends of the range. The AFT range includes the thermal uncertainty from the modeling. A thermal reliability margin of -15°C and +20°C is added to the AFT range to establish qualification temperature extremes. Unlike the military standard, NASA has no minimum bounds on Acceptance and Qualification temperatures, though -35°C to +75°C are typical of many current NASA programs. The number of temperature cycles and their duration are identical to that of Acceptance testing [2].

An additional test parameter is the rate of temperature change during transition periods. The Military standards establish this rate at a minimum of 1°C/min for thermal vacuum testing and no higher than 5°C/min. At atmospheric pressure a 5°C/min rate of temperature change is easily achievable because of convection, but in vacuum this is much more difficult. To achieve this higher rate in vacuum, equipment with much higher heat pumping capability is needed.

Additional clarification is given regarding thermal soak, as well as thermal dwell. The thermal soak period is defined as the time during which a unit or system is operating and its base plate is continuously maintained within the allowable tolerance at the test temperature (typically $\pm 2^\circ\text{C}$). Thermal dwell is defined as the time it takes for all internal parts of a unit or system to come to equilibrium.

Commercial Thermal Vacuum Systems

Thermal vacuum systems have been in used since the 1950's for space hardware testing, so their concept is not new. Companies like High Vacuum Equipment Corporation (HVEC) and Bemco have been designing thermal vacuum components and systems for over 60 years [4] [5]. Their systems boast a very wide temperature range

and low ultimate pressure. For example, the HVEC system can reach pressures as low as 1 nTorr and temperatures of -255°C (20K) and $+150^{\circ}\text{C}$ with a uniformity of 2°C . It can also ramp up to $7^{\circ}\text{C}/\text{min}$ [4]. Bemco systems have similar capabilities. The cost of a basic version of one of these systems is at minimum \$150k [6].

Previous Work

Although commercial thermal vacuum systems are available, custom chambers are frequently used as well. Dgany [7] provided an excellent framework for the design process of a custom thermal vacuum system. His particular system was intended for qualification and acceptance testing of electronics units, requiring an estimated 150 W of heat power. Of primary interest was a thorough understanding of the physical constraints and functional requirements of the system. The temperature range and other functional specifications were defined by a thorough understanding of the testing that was to be performed. Similar to Gilmore [2], Dgany [7] defined the temperature extremes according to established standards.

One of Dgany's goals was to find a cost effective solution to the design. As previously mentioned, in thermal cycle testing a temperature change rate of $5^{\circ}\text{C}/\text{min}$ is easily achievable because of convection, but achieving this rate in vacuum is more difficult without convection and requires very large and/or expensive equipment. For this reason Dgany choose a $1^{\circ}\text{C}/\text{min}$ rate of temperature change for his system to save on equipment costs and still meet the established standard. The temperature range of the system was further refined by limiting it to a range that could be handled by a single re-circulating fluid.

For simplicity and to keep costs low Synth-60 was used as the re-circulating fluid, having an operating temperature range of -40°C to $+60^{\circ}\text{C}$. Although this range was

not as broad as that specified by the Military standards, it was sufficient for testing electronic units in the range of -10°C and $+10^{\circ}\text{C}$, as measured on orbit from the UOSAT II satellite. Temperatures from -30°C to $+50^{\circ}\text{C}$ were measured on outer panels of the UOSAT II mission, which could easily be achieved using Dgany's system. Having this understanding of expected temperature limits provided a baseline from which to define this system's requirements and select appropriate control hardware.

Dgany sized his thermal plate to the intended test items, and thermal analysis was performed on the plate to determine if it remained within specified performance characteristics. The rate of temperature change selected was meant to apply to the entire thermal plate, and so a difference of 2°C was specified as the design criteria. The plate was designed with several parallel flow paths, and appropriate convection and conduction calculations were performed to determine the resulting temperature distribution. His configuration resulted in temperature variation of less than 2°C across the plate.

Also of interest were three options for a radiation heat shroud. First was a removable radiation heat shield which effectively isolated the chamber walls from the test item. By blocking the radiation heat exchange between the two surfaces, conduction based models could be more easily validated. Second was a conduction controlled mini thermal shroud attached to the thermal plate. Ideally this mini shroud operated at the same temperature as the thermal plate and served to speed the rate at which a test item would come to equilibrium. And third, the chamber wall itself was utilized as a means of temperature control. The chamber was fabricated with a double-wall and a recirculating fluid passed through the annular volume between the two walls. It was Dgany's conclusion that these three radiative temperature control methods increased reliability and decreased long-term cost compared to their counterparts found in much larger systems.

Built in system safety features that prevent inadvertent damage to the system or the test article were of interest for the SSEL's thermal vacuum system. SSEL's thermal vacuum system was to be designed so that students without any prior experience could safely operate the system. Both Dgany [7] and the Microwave Journal [8] discussed the advantages of fully autonomous control of the vacuum and thermal systems. A pre-programmed process would eliminate the risk of human error, guarantee correct sequencing, and reduce test time. Today's computers could not only perform the automation sequences, but could also record all system and test data.

Closing Remarks

The SSEL wanted the ability to simulate the vacuum and thermal environments of Low Earth Orbit and to perform thermal vacuum testing at Montana State University. Although similar systems were in use by other institutions, they were not commercially manufactured, nor were the full designs published. Commercial systems were too costly to purchase and had capabilities far exceeding those needed by the SSEL. Therefore, a low-cost system that meets the needs of the SSEL while not exceeding the available resources was designed and built so that SSEL hardware, as well as hardware from other organizations and collaborators, could be tested at low operational costs.

VACUUM SYSTEM DESIGN

Vacuum System Requirements

In 2005, the SSEL began procuring a vacuum chamber and all of the auxiliary equipment needed to operate the chamber at pressures below 100 μ Torr. It was sized to allow for testing of CubeSats (10 cm cubes), nanosatellites (approximately 50 cm cubes), as well as subsystems of each satellite type. Discussions within the SSEL [9] as well as with industry contacts established a set of requirements for the vacuum system which would maximize the functionality of the system as well as allow for future additions. The following requirements were defined for the vacuum system based on the needs described above. The Vacuum System shall:

1. Fit within a 1.219 m (4 ft) by 2.743 m (9 ft) area, and be no taller than the light fixtures in the SSEL's Small Satellite Lab.
2. Have internal dimensions of 91.4 cm (3 ft) in diameter and 91.4 cm (3 ft) tall.
3. Be capable of reaching 5 μ Torr without a load in two hours so that with a load it can reach 100 μ Torr.
4. Be as quiet as possible during operation.
5. Have viewports on 3 axes.
6. Have extra ports of various sizes to allow for connection to pumps, gauges, and valves; and to allow for future expansion of capabilities.

In May of 2006, purchasing of components for the SSEL Vacuum System began. The chamber and two pumps were the first to be purchased, and the remainder of

the components soon followed. Before discussing the vacuum system design and the reasons for choosing the various components and their configuration, it is worthwhile to explain some terminology that is used with vacuum technology.

Vacuum Terminology

There are several common units used in the scientific and engineering community to measure pressure, the most common of which are Atmospheres, psi, Pascals, Bar, and Torr. In this document, Torr will be the units for all pressure measurement. For reference, atmospheric pressure is equal to 760 Torr at sea level, and 1 Torr is equal to 1 mmHg. Pressures in vacuum are categorized into regimes for quick reference. Rough vacuum ranges from 760 Torr to 1 mTorr, and high vacuum ranges from 1 mTorr to 10 nTorr.

One other pressure term will also be referenced. The ultimate pressure of a chamber is the lowest pressure it can reach and is dependent on several factors, most notably the cleanliness of the chamber, the size of the chamber, the pumps used, and whether there is a load on the pumps. A load is anything which slows the process of reaching a chamber's ultimate pressure. Examples include water vapor, oils (which outgass and increase the particle count in the chamber), and leaks.

Vacuum chambers also come in a variety of shapes and configurations. The chamber chosen for the SSEL's system is called a vertical chamber. This style of chamber is most commonly seen as a glass bell jar type. It is made up of two halves, either a flat plate for a bottom or a hollowed out cavity. Also very common are cylindrical horizontal chambers which lay on their side and are opened from one or both circular end caps.

Selection of Components

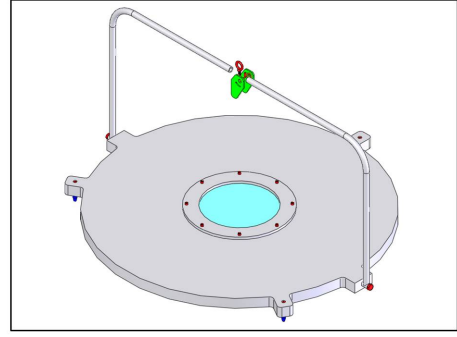
Chamber

The chamber that was selected is a custom vertical chamber designed by LACO Technologies and built to the SSELs specifications. It is a domed vertical chamber measuring approximately 90.2 cm (35.5 in) inside diameter and 91.4 cm (36 in) tall at the base of the dome. It separates into two halves at roughly 45.7 cm (18 in) from the base. Located axially on the base of the chamber is a 25.4 cm (10 in) Con-Flat (CF) port for the high vacuum pump. Located radially are 12 ports (four each of three sizes: 7 cm (2.75 in) CF, 15.4 cm (6 in) CF, and 20.3 cm (8 in) CF) for pressure gauges and instruments, as well as for future additions to the chamber. This design was chosen over a horizontal chamber with similar ports because a horizontal chamber with the same capabilities would have been too large to fit into the 1.219 m (4 ft) by 2.743 m (9 ft) floor area, whereas a vertical chamber and all of the pumps, valves, etc. would fit in this area. With the exception of the size requirement (91.4 cm inside diameter), this chamber, shown in Figure 3.1a, met all of the requirements for the SSEL's vacuum chamber.

The domed vacuum chamber lid has viewports on the X- and Y-axes but none on the Z-axis, so a second lid was designed to provide a Z-axis viewport. This flat lid design, shown in Figure 3.1b, would allow for faster pumping because the chamber volume was roughly halved. With the dome lid the chamber can be pumped to below 5 μ Torr in under 2 hours without a load, and so the flat lid was unnecessary to meet this requirement. Thus, the requirement for viewports on three axes was relaxed and the final design has only the two viewports on the domed lid. As of this writing the flat lid design has not been manufactured.



(a) SSEL's vacuum chamber after mounting it to the table



(b) Chamber flat lid assembly concept

Figure 3.1: SSEL Vacuum Chamber and Flat Lid Concept

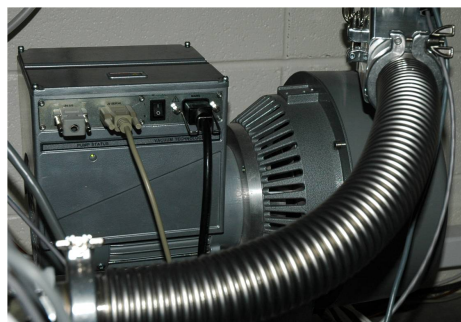
Pumps

To achieve the desired base pressure under load (i.e. with a satellite in the chamber) a high vacuum pump was needed, and several types were available. A cryogenic pump uses baffles at very low temperatures (4.2 Kelvin) to adsorb and entrap molecules that collide with the cold surface, thus removing them from the chamber volume. These pumps require a compressor unit for liquid Helium which can take up almost a square meter of lab space and are very noisy and expensive. Another alternative was a diffusion pump which uses a high velocity fluid (with a low vapor pressure) to impart momentum to gas molecules and move the molecules from a low pressure to a high pressure region. While quiet, diffusion pumps are known for backstreaming where some of the pumping fluid vaporizes and enters the vacuum chamber against the pressure gradient [1]. Because satellites have solar cells and other sensitive optics, the backstreamed particles can form a monolayer on the optics and change their optical properties. Turbomolecular pumps have an ultimate pressure of

approximately 0.1 nTorr, are quiet, very clean, and require infrequent maintenance. Thus, a turbo pump (Figure 3.2a) was the best option for the SSEL's needs.



(a) Varian TV-1000 HT Navigator Turbomolecular Pump



(b) Varian TS300 TriScroll Pump

Figure 3.2: The SSEL's High and Rough Vacuum Pumps

Turbo pumps are electrically driven and require only a small controller to operate. They function similarly to the compression region of jet engine. A turbo pump's rotors spin at 30,000 RPM or more and impart momentum to any molecules that come in contact with the blades. Multiple rows of blades and compression regions force molecules from the low pressure inlet to the high pressure outlet (called the foreline) where they are removed by a backing pump [1]. The backing pump (also called a roughing pump or foreline pump) serves two functions. First, it maintains the foreline of the turbo pump at or below 50 mTorr. Turbo pumps cannot operate above approximately 50 mTorr without causing damage to the rotors and the pump housing. Second, the backing pump allows for pumping the chamber to rough vacuum before switching on the turbo pump, or for tests that do not require high vacuum.

Several types of roughing pumps exist, but many of them suffer the same problem as diffusion pumps - they backstream because they use mechanisms that require lubrication. A scroll pump (Figure 3.2b) does not suffer from this problem. It uses two orbiting spiral regions to compress and force gasses from the low pressure inlet to the higher pressure outlet. It is also a quiet pump and takes up no more space than any of its alternatives.

For these reasons a Varian TV-1000 HT Turbomolecular Pump (Figure 3.2a) and a Varian TriScroll TS300 Scroll Pump (Figure 3.2b) were chosen to meet the pumping needs of the SSEL's vacuum system. Both pumps have simple user controls, either on their respective front panels or through serial interfaces with a computer.

Valves

As shown in Figure 3.3 a series of valves were used to achieve proper system functionality. Each valve is listed below with a brief description of its purpose.

- Scroll Valve: Isolates the scroll pump from the foreline. It prevents large volumes of atmospheric air from backstreaming into the foreline when the scroll pump is shut off, thereby maintaining a clean foreline and preventing damage to the scroll pump.
- Foreline Valve: Along with the gate valve, it isolates the turbo pump from the chamber and foreline, thereby helping keep the turbo pump clean. It also allows for direct pumping on the chamber through the roughing valve.
- Gate Valve: Along with the foreline valve, it isolates the turbo pump from the chamber and foreline, thereby helping to keep the turbo pump clean, and also allowing for rough vacuum testing in the chamber.

- Roughing Valve: Allows for roughing the chamber without going through the turbo pump. This valve also isolates the foreline from the chamber when the turbo pump is used.
- Ion Gauge Valve: Isolates the Bayert-Alpert Gauge from the chamber in rough vacuum which reduces the amount of contaminants that deposit on the gauge filament. By keeping the filament clean, degassing the gauge is required less frequently which increases its lifetime and helps maintain measurement accuracy.
- RGA Valve: Similar to the ion gauge valve, this valve isolates the RGA from the chamber in rough vacuum to help keep it clean.
- Chamber Nitrogen Vent Valve: Used to vent the chamber with dry Nitrogen gas and bring the pressure back to ambient after a test.
- Turbo Nitrogen Vent Valve: Used to vent the turbo pump with dry Nitrogen gas after it has been turned off. Venting the turbo pump is necessary to stop the turbine blades from spinning.

Pressure Gauges

The driving requirements for the pressure gauges were (a) the desired base pressure of 5 μ Torr and (b) system safety. In the rough vacuum range (760 Torr to 1 mTorr) a ConvecTorr gauge was used because the low end of its pressure range overlaps with many of the high vacuum gauges. In high vacuum range a Bayert-Alpert Gauge was chosen (Figure 3.4). Similar to the ConvecTorr gauge it measures the pressure by measuring the current generated as a result of ionizing the media in the chamber. A heated element boils off electrons which then ionize the media in the chamber. These ions are then attracted to a collector element by a grid at a lower potential and the

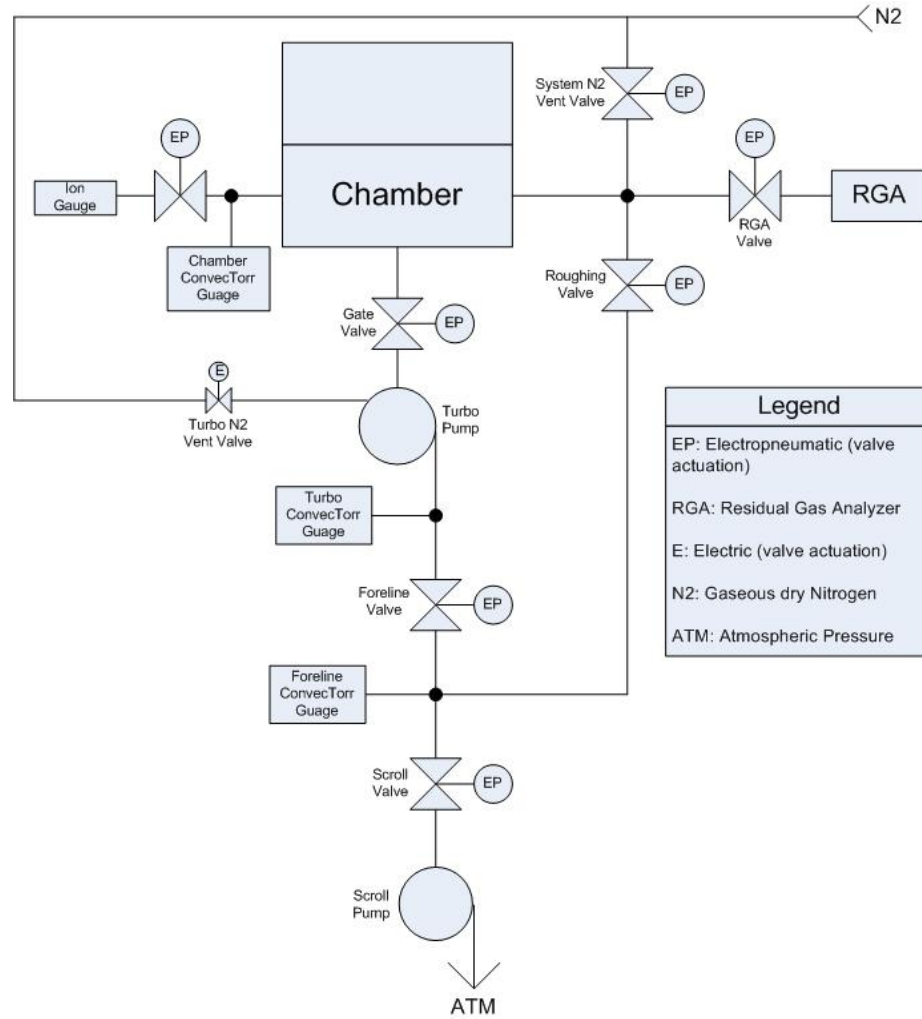


Figure 3.3: Vacuum System Schematic

resulting bombardment on the collector generates a current which is proportional to the number of molecules in the chamber, and hence, the pressure.

Because several valves were used in the system, multiple gauges were needed to measure the pressure on both sides of a closed valve. If one of the valves were to be opened with a large pressure differential across its sealing surface, damage to the valve or to one of the two pumps could occur. For example, if the turbo pump was on, the gate valve was closed, and the chamber pressure was above 50 mTorr, then opening the gate valve and exposing the turbo pump to high pressure could damage

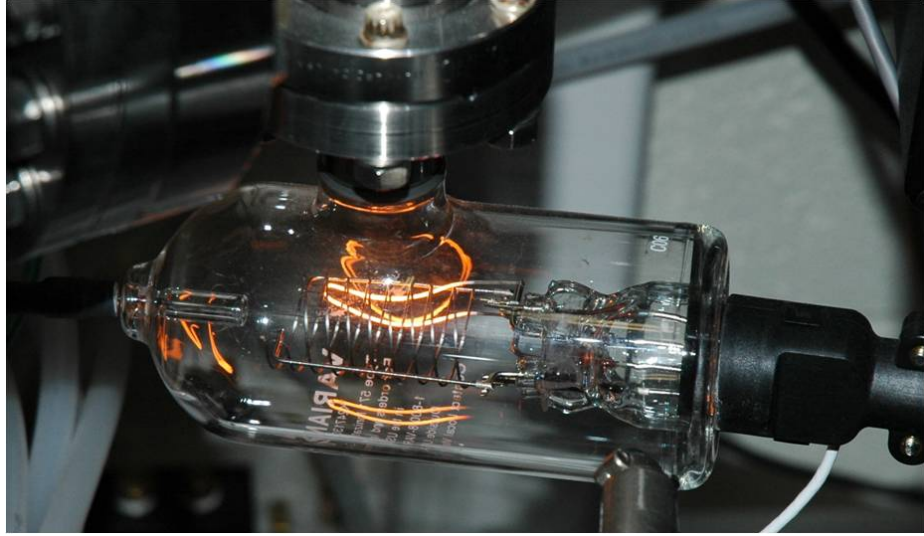


Figure 3.4: Bayert-Alpert Gauge

or destroy the pump, and would pose a potential safety concern for anyone in the room. Thus, the ability to determine when it is safe to open a valve or turn on a pump was essential to protect the equipment and the operator.

Three ConvecTorr gauges were used in the system. The first was situated in the foreline adjacent to the scroll pump and between both the chamber and the turbo pump. Here, the user or the control code could verify the foreline pressure before opening the foreline valve. Also, the user or the control code could verify the foreline pressure before opening the roughing valve. A second ConvecTorr gauge was situated at the foreline of the turbo pump but before the foreline valve. In this location the user or control code could determine when it was safe to open the foreline valve and also when it was safe to turn on the turbo pump. The third ConvecTorr gauge was situated on the chamber itself to indicate its rough pressure. A single Bayert-Alpert gauge was also connected to the chamber to measure the pressure below the range of the chamber ConvecTorr gauge.

With this combination of gauges, shown in Figure 3.3, the user or control code could determine when it was safe to operate valves or pumps, could measure the pressure throughout the entire desired pressure range, and could control all the gauges from a single control unit. Having one instrument (a Varian MultiGauge Controller) and only one electrical interface made controlling and reading the pressure gauges simple.

Residual Gas Analyzer

Along with the pumps and valves, another important piece of hardware was a Residual Gas Analyzer (RGA) (Figure 3.5) which measured the relative amount of gases as a function of their mass to charge ratio. Any matter that goes into the vacuum chamber outgasses, or gives off particles of itself, when the pressure drops below its partial pressure. Gasses like Oxygen, Nitrogen, water vapor, Carbon Dioxide, and Argon are common as they are large components of atmospheric air. Gasses from hydrocarbons are undesirable, so the ability to detect them is important in maintaining cleanliness in a vacuum chamber.

Another function of the RGA was leak detection. The ultimate pressure of a chamber and the time it takes to reach the ultimate pressure are functions of how leak-tight the chamber is, and the RGA is the tool for determining the type and location of leaks. Two types of leaks are common: real leaks and virtual leaks. As the name implies, a real leak is a leak from the outside to the inside of the chamber, and is characterized by the species and relative amounts detected by the RGA. Prominent peaks at mass to charge ratios of 28 (N_2), 32 (O_2), 44 (CO_2), and 14 (N) indicate an air leak. Pumping on the chamber will not reduce the pressure beyond a certain point, and the pressure will increase linearly when the chamber is isolated from the

pumps. Virtual leaks are not true leaks but instead are a result of outgassing from the chamber itself or from materials inside the chamber [1].



Figure 3.5: Extorr XT200 RGA

To detect a leak, the chamber is first pumped to its ultimate pressure or as low as it can go in a given amount of time. The chamber is then isolated from the pumps and the pressure recorded at regular intervals. If the pressure increases linearly over the course of the measurements, a real leak exists. If a virtual leak exists the pressure will increase steadily and then plateau, indicating a stop to the outgassing that has caused the pressure to increase. In either case, it can take several hours to determine which type of leak is causing the pressure to increase.

If a real leak exists, an RGA will help the user to locate the source. The RGA can be set to a ‘trend mode’ where it looks at only at a mass to charge ratio of 4 (Helium). Helium molecules can easily get into the chamber and can be pumped out quickly because they are small and energetic. The user can open the chamber to the pumps again and use a helium source (often a bottle of helium connected to a hose, set at a small flow rate) while in trend mode with the RGA. The leak is found when the RGA indicates a spike in the Helium level.

With an RGA attached to the SSEL’s vacuum system, leaks could be detected and test objects could be monitored for outgassing. While an RGA can not measure the Total Mass Loss (TML) and Collectible Volatile Condensable Matter (CVCM), both of which are used to quantify the outgassing of an object, it provides an indication that outgassing is occurring and helps to determine what is outgassing. To determine the TML and CVCM of an object in the vacuum chamber a Thermoelectric Quartz Crystal Microbalance (TQCM) would have been necessary. These were rather expensive and were not essential for the SSEL’s needs, thus an RGA provided the needed outgassing instrumentation for the SSEL’s vacuum system.

Table and Crane

Much of the work that went into selecting the vacuum components was driven by the need to fit all of the functionality into a 1.219 m (4 ft) by 2.743 m (9 ft) area, to keep the cost down as much as possible, and to leave room for future expansion. The chamber and components were not self-supporting, thus a table or suitable support structure was necessary. Three undergraduate students designed and fabricated a custom table and crane for the vacuum system [10].

MSU and SSEL students machined the parts or coordinated with local machinists to have the parts made, and assembled the table and crane in the MSU Physics



(a) A student welds parts of the table together



(b) Students with the assembled table before powder coating

Figure 3.6: Fabrication of the SSEL's Vacuum Chamber Table



(a) A student installs the crane to the table



(b) Students installing the table in the lab after powder coating

Figure 3.7: Fabrication and Assembly of the SSEL's Vacuum Chamber Table



(a) Machining the crane mast



(b) Assembling the trolley

Figure 3.8: Building the Crane

Department Machine Shop. Figures 3.6 and 3.7 show the students building and assembling the table. The crane was fabricated after the table and the chamber had been installed. Figure 3.8 shows several steps of this process.

Installation

Installing the vacuum system components as well as the table and crane required careful planning. The table itself was assembled and situated first as shown in Figure 3.7b. Following that, each half of the chamber was lifted and set on a cart, rolled through the lab door, and placed appropriately on the table (see Figure 3.9) [10]. All additional components (valves, pumps, pressure gauges) were installed to the chamber following its installation. Following this the pumps were wired to the PC, the valves were wired to the Data Acquisition device, and the compressed air and dry Nitrogen lines were installed.

Both pumps, the RGA, and the MultiGauge Controller interfaced to a host computer over serial RS-232. A National Instruments NI-9172 Data Acquisition (DAQ) device utilized a set of user configurable modules to control and monitor the eight



(a) Unpacking the chamber



(b) Moving the chamber top



(c) Moving the chamber base



(d) The chamber after installation

Figure 3.9: Installation of the SSEL's Vacuum Chamber

system valves. Each valve was opened by applying 120 Volts AC to its solenoid using two 4 channel NI-9481 relay switching modules. A 5 Volt DC signal was applied to each of the limit switches on the valves. Each valve had ‘open’ and ‘closed’ limit switches which were read by two NI-9401 TTL modules. Thus, if a valve was working properly only one limit switch would be active at a time.

A custom power distribution box split the 120 Volt AC power to each of the 8 valves, and contained a 5 Volt DC power supply for the limit switches. This power distribution box had one main switch on the front panel (Figure A.35) that turned on power to the valves and limit switches, as well as to the DAQ, the RGA, and several power supplies used for the thermal system (discussed later). Power to the entire system (except the pumps) could thus be removed with this one switch, shutting down the system quickly and safely in an emergency. The rear panel, illustrated in Figures A.36, A.37, and A.38 had a connection for power to each of the valves, an input for the valve limit switches, and two power outlets for switched 120 Volt AC instruments.

Vacuum System Operation: LabVIEW Software

National Instruments LabVIEW Software was used to develop a hierarchy of Virtual Instruments (VIs) to safely control the SSEL vacuum system. The code for operating the vacuum and thermal systems went through several iterations, beginning with a completely automated version as suggested by Dgany [7] and the Microwave Journal [8]. The code asked the user a series of questions and configured the program to follow a sequence based on the user’s input. This version was not developed beyond the initial concept because after speaking with several professionals in industry [11]

it was determined that a completely automated system was not in the best interest of this system or of any vacuum system.

When big budget satellites are tested in thermal vacuum chambers, the customer will want to initiate every action the system takes. That customer will have defined and thoroughly reviewed a test procedure, requiring the system operator to command each step (with a second person present) as a way to prevent unwanted action by the system. This rules out a completely automated system. A completely automated process was initially desired for the SSEL's vacuum system because it would prevent unexperienced system operators from performing actions which could cause damage to the system or to the test article. However, with well defined and thoroughly reviewed test procedures already in use at the SSEL, a fully automated system was not needed. Since a completely automated system was not acceptable and the system and test object had to be protected from damage, several safety features were included in a manually operated system.

Designing this control system posed a significant challenge. The user had to be able to command each action, but he or she had to be prevented from commanding actions which would damage the system. This was implemented by enabling and disabling user interface controls based on the current state of the system. The valve and pump states, along with pressure criteria determined whether a user control was enabled or disabled.

Each of the pumps and other instruments was configured and set up the same way each time the program initialized, as was each of the user controls, to ensure that the system hardware and software started in a known and safe configuration. Once the instruments and code were initialized, the code would wait for the user to choose one of the allowed commands, and the system then responded accordingly and updated the user interface. At any point during operation the user could choose actions which

would continue taking the system toward reaching high vacuum, or choose actions which would bring the system back to atmospheric pressure. Once the user finished using the system the code would then put the hardware into a safe state (if were not there already) and proceeded to close communication sessions with the instruments.

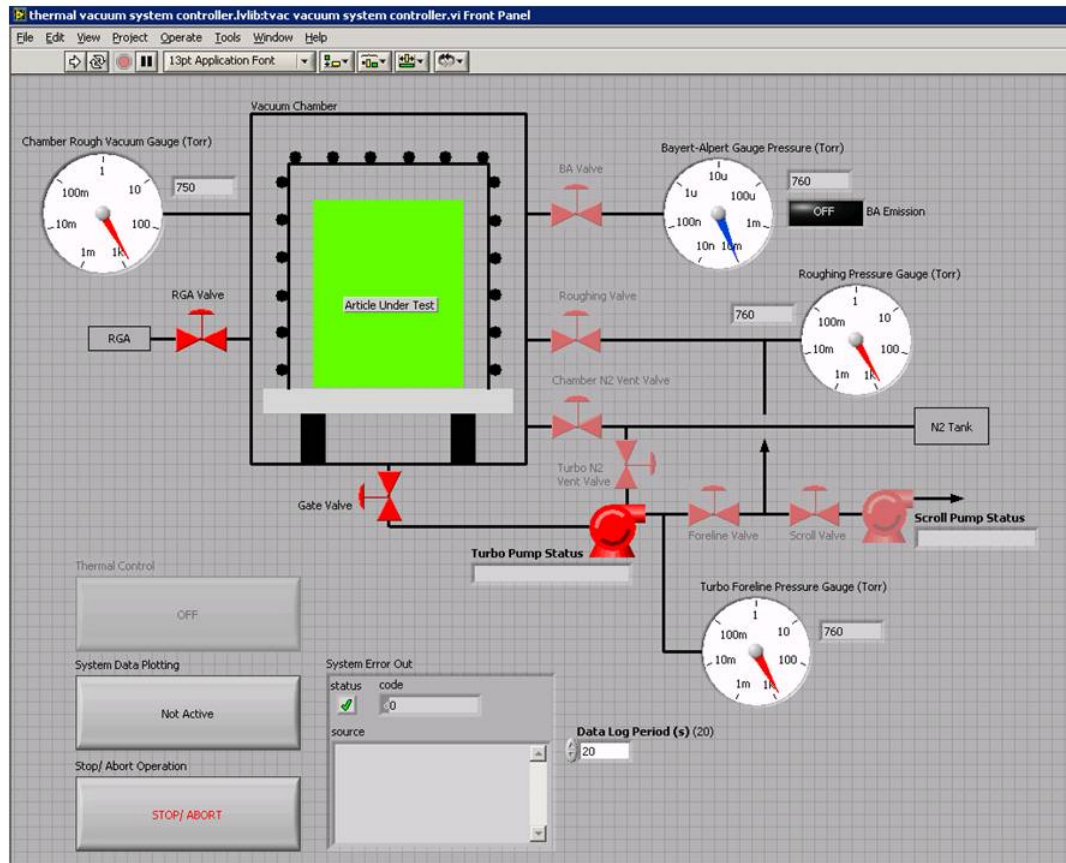


Figure 3.10: Vacuum System Controller User Interface

While the TVac System Controller VI (Figure 3.10) ran, several other features also ran. From startup to shutdown the code regularly logged all of the system pressures and user actions to a text file for later processing.

As part of the development process for this code, LabVIEW instrument drivers were written for both pumps and for the MultiGuage controller because they were not available for download from National Instruments or from their manufacturer.

These drivers initialized communication between the host computer and the instruments over RS-232, sent commands, received and parsed responses into usable data in appropriate units, and closed communication between the host computer and the instruments. Each instrument command was built into an Application Program Interface (API) VI and tested with the instrument. The application VI was then built containing all of the appropriate API VIs, and was also tested with the instrument. After each of the application VIs was built and tested, they were integrated into the TVac System Controller VI. Following this bottom-to-top testing approach ensured that all low level commands functioned before they were ever used in a higher level application. When an instrument application VI was tested, so too was the instrument itself.

Closing Remarks

The SSEL Vacuum System requirements, as laid out at the beginning of this chapter, were met and the system has performed up to specification. It reached pressures as low as $0.5 \mu\text{Torr}$ in approximately two hours when clean and without a load. All of the vacuum system components did fit within the allowable physical envelope, the chamber itself met the size requirement and included two viewports, the two pumps added minimal noise to the lab while running, and the chamber had adequate ports for adding thermal capability. User interface software was written and performed admirably.

BASE PLATE DESIGN

Thermal System Requirements

In the fall of 2008, the requirements for the thermal system were outlined in a similar way as for the vacuum system. At this time the vacuum system components had been purchased and installed, and a first iteration of the software needed to safely operate the vacuum system equipment had been developed. The thermal system, therefore, was a retrofit to the vacuum system, operating within the physical constraints of the already built vacuum system. The process used to define the thermal system requirements was more formal than the process used for the vacuum system. It began with an overall statement of intent for the system: The SSEL thermal vacuum system will provide the capability for thermal vacuum testing and thermal balance testing for nanosatellites, picosatellites, and other space hardware. The following are the thermal system requirements, derived from the statement of intent.

1. The thermal system shall be capable of cooling a test object to -40°C and heating it to 80°C to allow for both thermal vacuum testing and thermal balance testing.
2. The thermal system shall be capable of maintaining the temperature within 2°C during soak periods.
3. The in-vacuum components shall be capable of operating in pressures as low as 1 nTorr.
4. The in-vacuum components shall be designed or chosen such that a test object can be mounted to the base plate of the thermal system.
5. The in-vacuum components shall surround the test object in all directions.

6. Users shall be able to operate the system easily from a control panel on the external equipment or through a computer interface.
7. The thermal system shall be capable of heating or cooling a test object at a rate of at least $1^{\circ}\text{C}/\text{min}$, with a goal of $5^{\circ}\text{C}/\text{min}$, at the interface between the object and the base plate.
8. The thermal system shall be capable of thermal soak at the extreme temperatures for durations lasting up to 8 hours at each soak.
9. The thermal system shall be capable of thermal cycling a test object, including maintaining soak temperatures, for up to 24 cycles.
10. The in-vacuum components shall be designed such that test objects can be placed in the chamber from above using the crane or by hand.
11. All in-vacuum thermal components shall fit inside the chamber which is 90.2 cm (35.5 in) in diameter and 91.4 cm (36 in) tall.
12. The in-vacuum thermal components shall not obstruct the chamber flanges located radially or axially.
13. The in-vacuum thermal components shall allow for testing of an object up to 61 cm (24 in) in diameter 50.8 cm (20 in) tall.
14. All thermal system feedthroughs shall extend out of the chamber radially through one or more of the following flanges: 20.3 cm (8 in) CF (4 available), 15.2 cm (6 in) CF (1 available), or 7 cm (2.75 in) CF (2 available).
15. The in-vacuum thermal components shall be designed such that electrical connections can easily be made to the test objects through the two 15.2 cm (6 in) CF flanges with D-Sub electrical connections.

16. The thermal system shall be designed such that the total cost does not exceed \$25k, with a goal of \$20k.

Base Plate Requirements

Requirements for the base plate itself were derived from the Thermal System Requirements, and are shown below. The base plate shall:

1. Be capable of reaching and maintaining the temperature at the interface between the base plate and the object between -40°C and $+80^{\circ}\text{C}$.
2. Be capable of heating or cooling a test object at a rate of at least $1^{\circ}\text{C}/\text{min}$, with a goal of $5^{\circ}\text{C}/\text{min}$, at the interface between the object and the base plate.
3. Be able to safely support a test object with a mass up to 50 kg.
4. Be designed to minimize conduction heat loss to the bottom of the chamber, with a goal of less than 25 W.
5. Be designed to minimize conduction heat loss to the thermal shroud, with a goal of less than 25 W.
6. Be designed to minimize radiative heat loss to the chamber, with a goal of less than 25 W.
7. Have regularly spaced holes drilled on the top surface to a depth of at least 6.35 mm (0.25 in) for mounting a test object and to allow for adequate thermal contact between the test object and the base plate.

8. Be at least 7.6 cm (3 in) above the bottom inside of the chamber, measured from the lowest part of the base plate (not including the feet and tubing) to the chamber bottom.
9. Be no higher than 20.3 cm (8 in) from the chamber bottom to the top of the base plate (the height of the center of the radial flanges on the chamber).
10. Have at least 7.6 cm (3 in) between the base plate and the chamber walls for access around and under the base plate.

Heat Load Analysis

The base plate was the central component of the thermal system. It was used whether performing a thermal vacuum or thermal balance test, with or without the thermal shroud, and it supported any test object in the chamber. For these reasons, great care was given to the analysis and design of the base plate. Design of the base plate depended significantly on the physical constraints of the vacuum chamber, and was just as dependent on the thermal needs which have been defined only in terms of temperatures and the rate of temperature change to this point. The heat load on the base plate was determined from these values and provided the means to determine an appropriate thermal control method and mechanical design. Analysis was performed to determine the largest heat load, in Watts (W), that the base plate would experience during testing. The resulting heat load was then used as a baseline for the thermal design of the base plate.

A better definition of the worst case heat load was needed before proceeding, and so a simple thought experiment was performed. There are several points of interest in the temperature profile (Figure 4.1), one of which is where the maximum heat load

occurs. The first point of interest is the knee at the start of the first hot soak, at the end of the transition from room temperature (22°C) to hot soak (80°C). The second is the knee at the start of the first cold soak, at the end of the transition from hot soak to cold soak (-40°C). Coincidentally, the same heat load value is found at the end of the transition from cold soak to hot soak because the temperature difference between the two extremes is 120°C in both cases. This is only true if the satellite is at uniform temperature at hot or cold soak before the transition begins. The third point of interest is the knee at the very end of the profile, at the end of the transition from cold soak to room temperature.

The difference between the start and stop temperatures for each transition is indicative of the gradient size in the test object at the end of the transition. The temperature gradient is largest at the end of the transition because of the thermal lag in the test object. The base of the test object changes temperature at the prescribed rate while everything above the base changes as a result of heat flow. The temperature of everything above the base is not be able to keep pace with the base of the test object, and thus the gradient grows with time. Since the largest temperature difference occurs between hot soak and cold soak (or between cold and hot soaks) with a difference of 120°C it follows that the gradient is largest at the end of this transition.

Given the results of this though experiment, a Finite Element Analysis (FEA) was performed to estimate the worst case heat load on the base plate during the cold to hot (or hot to cold) transition. All temperatures and rates of temperature change were based on the requirements as outlined above.

Finite Element Model

In order to estimate the maximum heat load of the base plate, several assumptions were made regarding the geometry of the satellite. First, adopting a standard satellite

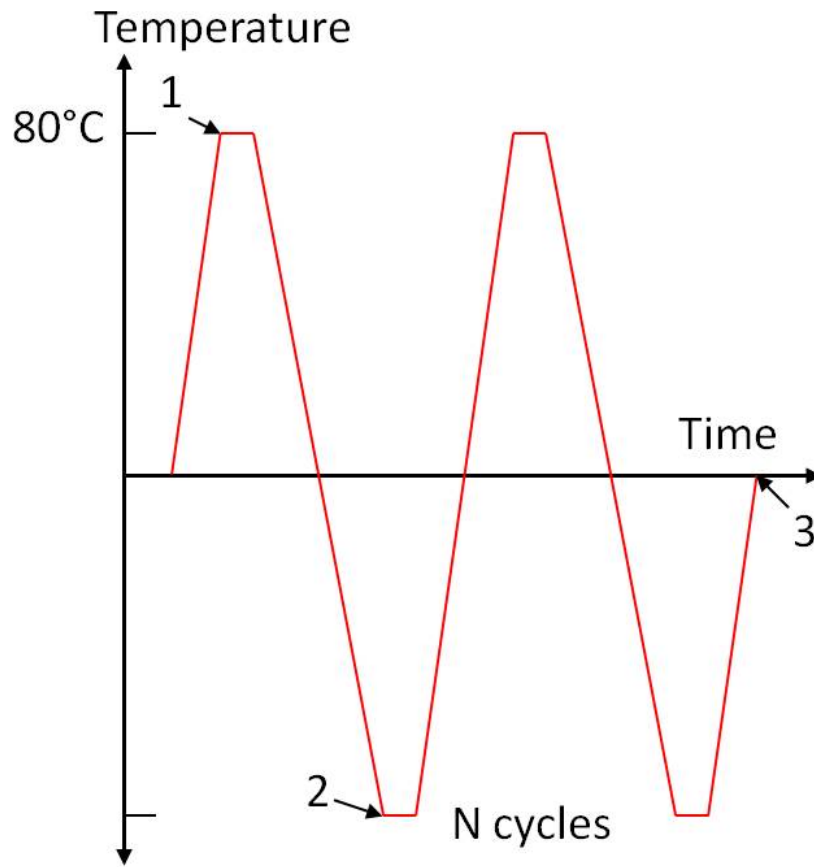


Figure 4.1: Basic Temperature Profile with Points of Interest Indicated

bus gave a baseline for the model size. The University Nanosatellite Program 6 (UNP-6) physical envelope of 50 cm x 50 cm x 60 cm (l x w x h) was used for this satellite because it represented the largest class of satellite that would fit in the SSEL vacuum chamber. Second, the volume of the satellite was assumed to be made of several members, including a base, top, four sides, and twelve interior walls which served as structural supports. The sides and walls were 4 mm thick each and the top and base pieces were 10 mm thick each. The mass of this satellite model would not exceed the 50 kg maximum mass specified by the UNP, and was constructed with 6061-T651 Aluminum (one of the more common metals used in satellite structures).

The mass of the satellite model was approximately 35 kg with the above assumptions. The mass was increased to 40.5 kg by adding ribs to the walls where they attach to other components, as well as adding plates to create electronics boxes. Given that this satellite model was meant to simulate an actual satellite, more mass was needed. However, it was difficult to add more mass without adding mission specific components such as batteries, circuit boards, sensors, actuators, payloads, etc. The base, top, sides, walls, ribs, and subsystem plates were all sized appropriately for an actual satellite. Thus, this model at 40.5 kg was as detailed as could be made without adding more mass. Figure 4.2 shows the satellite model created in CAD Software [12] with one of the walls, one of the sides, and the top hidden from view so that the inside is visible.

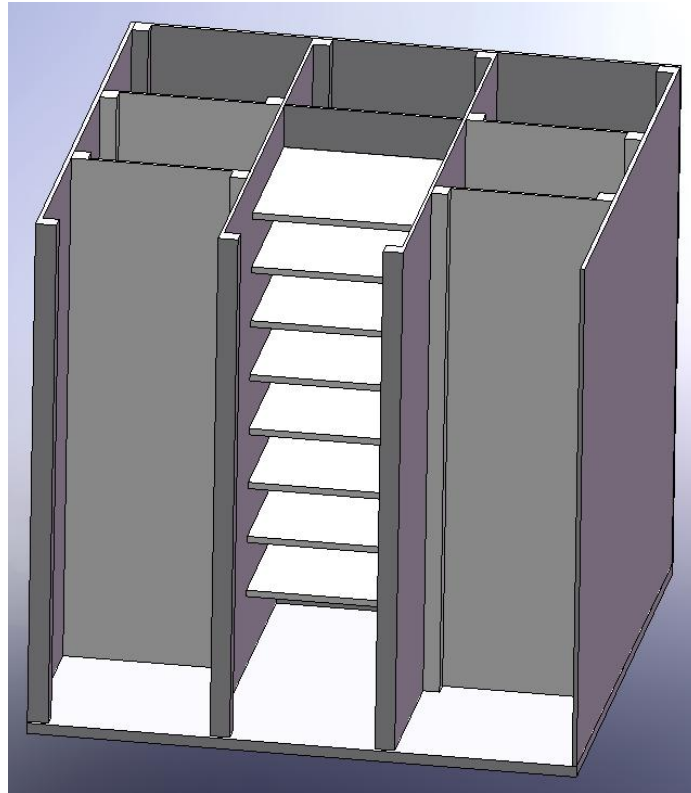


Figure 4.2: Satellite Model Used in the FEA Calculation of the Base Plate Heat Load, with the Top and the Near Side Hidden from View

The boundary conditions and temperatures for this FEA model were selected to closely match the conditions under which an actual satellite would be tested. Convection heat transfer from the satellite walls and top was negligible because the chamber would be operated at a pressure below 100 μ Torr. The surfaces of the satellite model and the chamber wall were polished Aluminum and Stainless Steel, respectively, and so both had low surface emissivity ($\epsilon \approx 0.1$) so radiation heat transfer was assumed to be negligible for simplicity. Thus, all surfaces except the bottom were set as insulated. Initially the satellite would be at the cold soak temperature (-40°C) and would then be heated at the bottom surface. Upon reaching the desired temperature the bottom surface was held at that temperature (soaking) while the remainder of the satellite reached equilibrium. For the purposes of this analysis the hot-side soak temperature was 80°C, the maximum temperature at which this system would operate under normal testing conditions.

The thermal analysis software [13] required two steps to simulate what the satellite model would experience during a heating cycle. First, the entire satellite was set to an initial temperature of -40°C by running a steady state case. The result from the first case was then used as the starting temperature distribution for a transient analysis where the bottom boundary temperature was raised from -40°C to 80°C over 24 minutes (for a 5°C/min temperature increase).

Finding a proper mesh for the study required running the analysis several times. The analysis software [13] recommended a mesh density but it could not determine if that density would be sufficient for the results to converge. To be sure that the results from a given mesh converged, the results from two identical studies with increasing mesh density were compared. When the results agreed to within a reasonable percent difference the mesh was sufficient and the results had converged. As a baseline study, the recommended mesh (4.9 cm) from the thermal analysis software was used with

the boundary conditions previously mentioned. The mesh, as viewed from the side and bottom of the satellite model, is shown in Figures 4.3a and 4.3b, respectively.

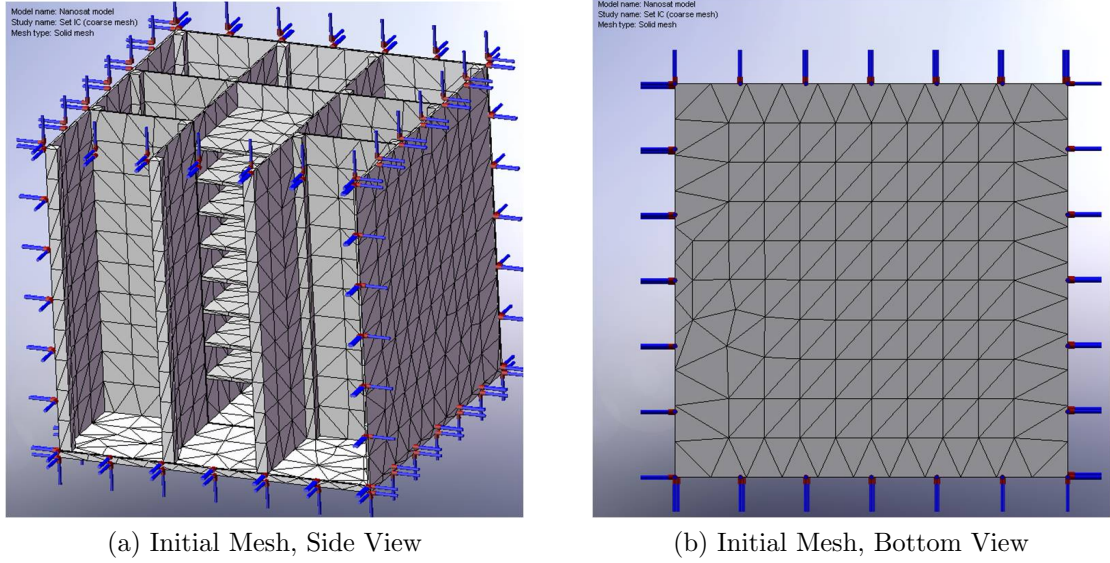


Figure 4.3: Initial Mesh of the Satellite Model

The temperature distribution made physical sense, with the base at 80°C (353.15 K) and the temperature decreasing from bottom to top (Figure 4.4). However, the heat flux through the bottom surface was negative in places which was not physically possible. Figures 4.5 and 4.6 show areas with a negative heat flux (darker blue), which was believed to be the result of the nodes being spaced too far apart; thus, a more dense mesh was required. The analysis was re-run several times, each iteration with a smaller mesh spacing, until the heat flux was a positive value at all nodes. The thermal analysis software [13] output several calculated results including average flux, the maximum and minimum fluxes, and the RMS heat flux at the bottom boundary. The minimum flux value (see Table 4.1) indicated that a negative flux did not exist with a mesh size of 1.37 cm as it did in previous studies.

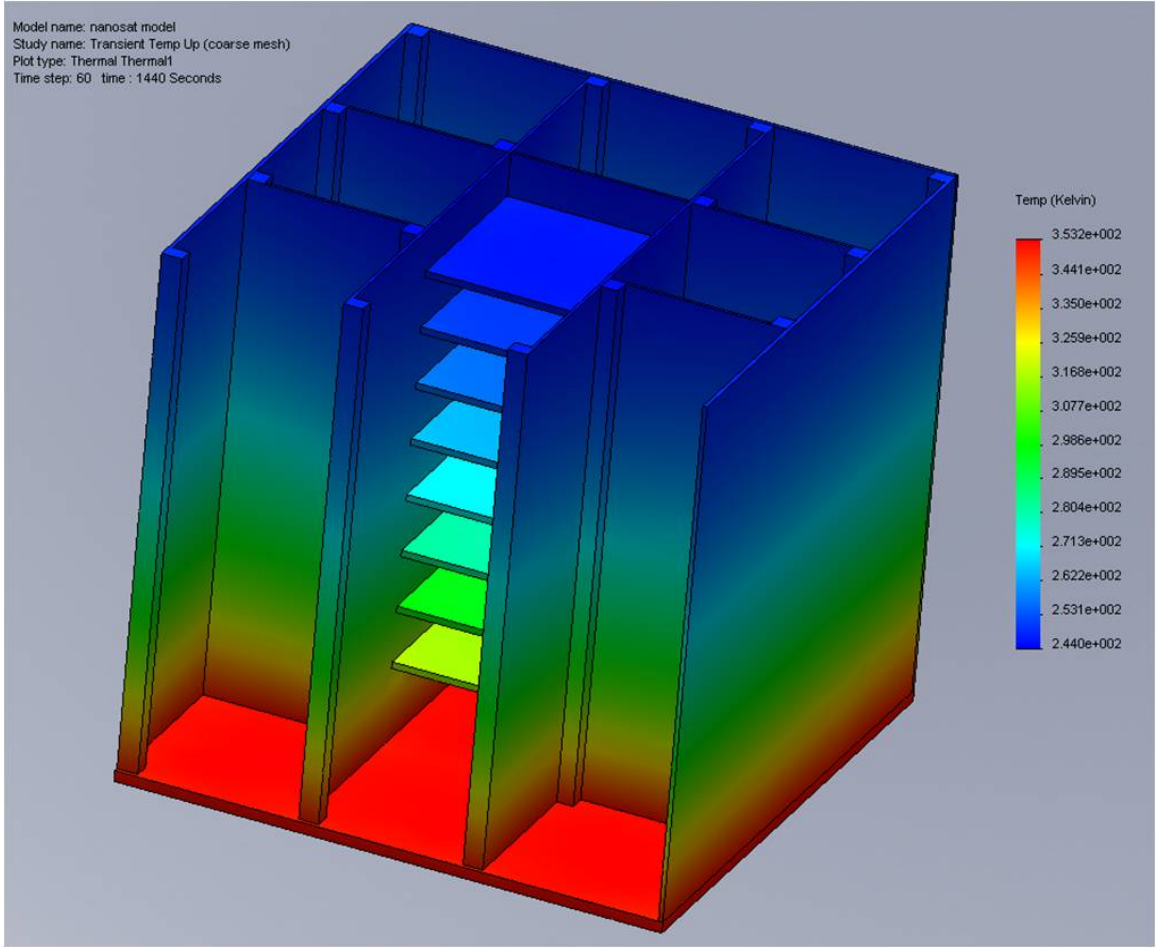


Figure 4.4: Temperature Distribution for 4.9 cm Mesh

While the thermal analysis software [13] provided the average power through the bottom surface of the satellite model, it was unclear whether this value was the arithmetic mean or the integral mean. To find out, raw data from the final time step (the time at which the transition ends and the soak begins) was exported to data analysis software [14], and the integral mean flux of the model's bottom surface was found using Equation 4.1 with trapezoidal integration.

$$Q_{surface} = \frac{1}{ab} \int_0^a \int_0^b f(Q_{local}) dx dy \quad (4.1)$$

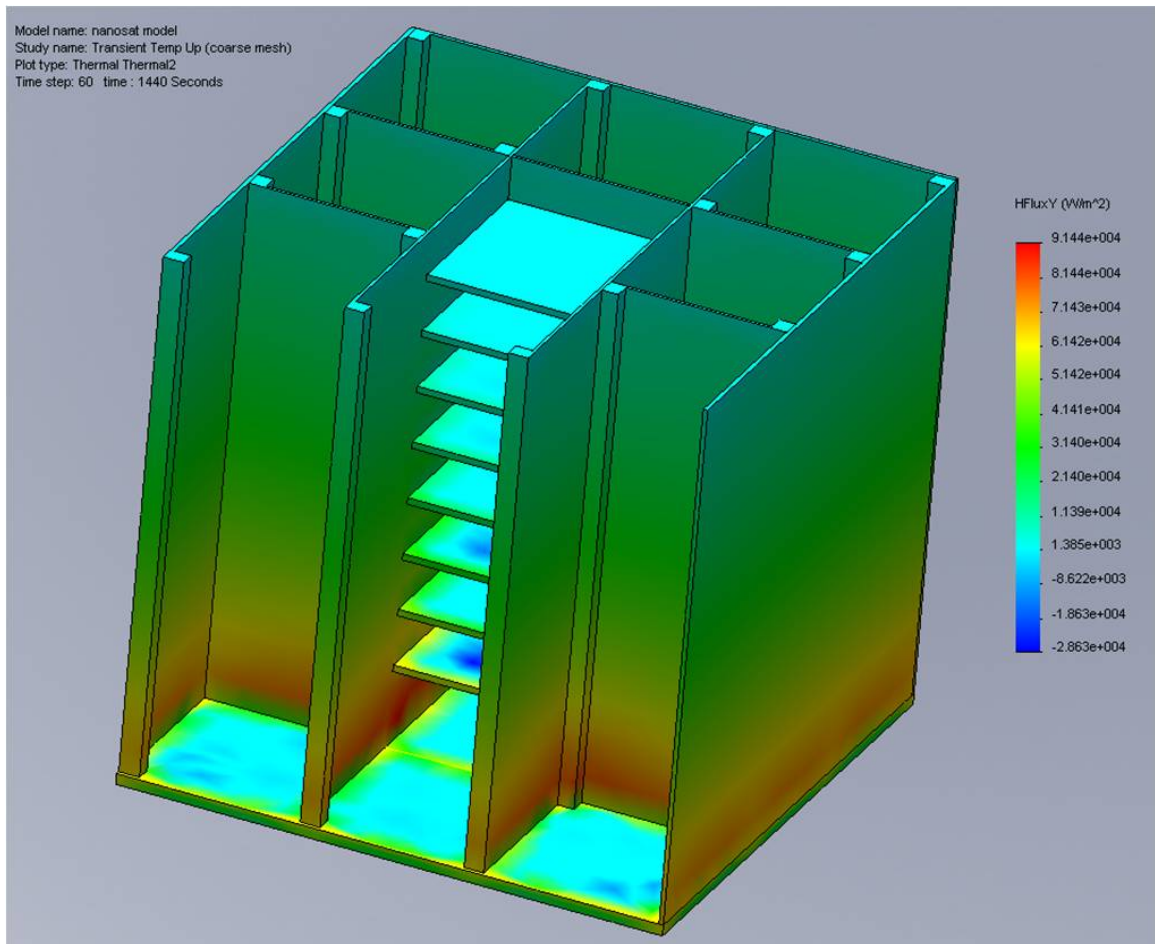


Figure 4.5: Heat Flux Distribution for 4.9 cm Mesh (Vertical Direction)

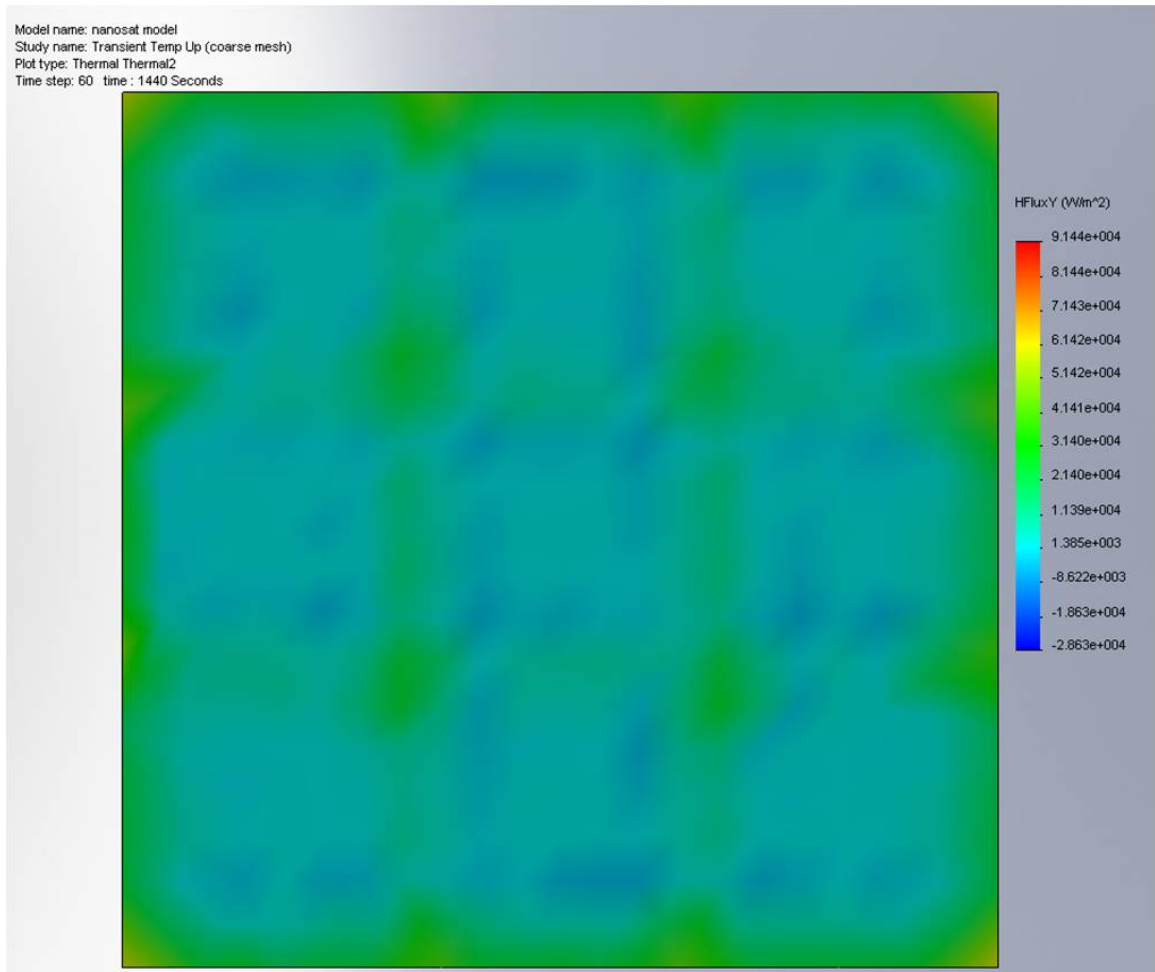


Figure 4.6: Heat Flux on the Bottom Surface for 4.9 cm Mesh (Vertical Direction)

The comparison between the unspecified mean calculated by the thermal analysis software [13] and the integral mean values from the data analysis software [14] (Table 4.1) showed that the values from the thermal analysis software were not the integral mean and that the integral mean would have to be calculated in the data analysis software for each unique mesh. As previously mentioned, several identical studies were performed, each having a more dense mesh. The average, minimum, maximum, RMS, and integral mean values were calculated and compared to the values from the study with the next highest mesh spacing, and the percent relative error, E_{pr} , between each of the corresponding values was determined. Equation 4.2 was used to calculate the percent relative error.

$$E_{pr} = 100 \times \frac{CurrentValue - LastValue}{CurrentValue} \quad (4.2)$$

Table 4.1 shows the results of running these studies. Performing the studies down to a mesh size of 0.8 cm was necessary to achieve a percent error of less than 5 percent in both the average and integral mean values. The result for a 5 °C/min rate of temperature change was a heat power value of 1881 W as a maximum value for the system. With a mesh spacing value established, identical studies were done with a boundary condition of 3 °C/min, resulting in a heat power of 1352 W, and 1.5 °C/min, resulting in a heat power of 833 W.

The values above were worst case heat load values for the base plate for several reasons. First, the satellite was representative of a real heat load because of its geometry, material and size. Second, the assumptions made for the boundary conditions were representative of a typical testing scenario. Finally, because contact resistance was neglected between the various members of the satellite model, heat would flow

Table 4.1: Iterative Mesh Study for a 5°C/min Rate of Temperature Change

Mesh Spacing (cm)	4.9	1.58	1.37	1.22	0.8*	Power (W)
Number of Nodes	18439	133785	179697	224376	155064	-
Values Provided by FEA Software [13]						
Average Flux ($\frac{W}{m^2}$)	9513.3	7958	7807.5	7629.2	7699	1924.75
Relative Error(%)	-	-19.54	-1.93	-2.34	0.91	-
Max Flux ($\frac{W}{m^2}$)	60288	61493	60411	60729	61149	15287.25
Relative Error(%)	-	1.95	-1.79	0.52	0.69	-
Min Flux ($\frac{W}{m^2}$)	-4250.1	-243.06	1137.3	2214.3	2219.1	554.78
Relative Error(%)	-	-1648.58	121.37	48.64	0.22	-
RMS Flux ($\frac{W}{m^2}$)	15450	12741	12448	12021	11955	2988.75
Relative Error(%)	-	-21.26	-2.35	-3.55	-0.55	-
Values Provided by the Data Analysis Software [14]						
Integral Mean Flux ($\frac{W}{m^2}$)	-	-	5722.34	7211.43	7526.53	1881.63
Relative Error(%)	-	-	-	20.65	4.19	-

*The number of nodes decreased between the 0.8 cm and 1.22 cm cases because of a limitation in the thermal analysis software. There were more nodes with a 0.8 cm mesh than the thermal analysis software could handle, so the bottom surface was set to a 0.8 cm mesh and the remainder of the model had a spacing of 2.01 cm.

more quickly from the heated bottom to the top of the satellite model than in an actual satellite. A real satellite would have contact resistances which would reduce the rate of heat flow within the satellite and therefore reduce the required power, given a constant rate at the satellite's base. With the heat power values known, the thermal and mechanical designs began on the system base plate.

Design Process

The requirements listed earlier outline the required functionality and physical constraints of the base plate. With the above requirements in mind, an Aluminum disc, 74.9 cm (29.5 in) in diameter and 1.9 cm (0.75 in) thick, having 2.54 cm (1 in) center-to-center $\frac{1}{4}$ -20 holes drilled on the top surface was designed (Figures A.1, A.2, A.3, and A.4). It must be noted that all drawings in Appendix A were designed in English units for ease of communication with machine shops in the United States. Aluminum was chosen because of its excellent thermal conductivity, structural strength, and low cost. Thermal gradients were also minimized by using a material with a high thermal conductivity. This Aluminum plate was intended to function with a thermal control system, regardless of what system was chosen.

Thermal Control Hardware Options

Common methods for controlling temperature in vacuum systems were researched in parallel with the design of the base plate. Any system that heats or cools inside a vacuum chamber must pass heat through the chamber or generate the heat inside the chamber. Conduction through the chamber itself was first considered, but because it was constructed of 316 stainless steel the response time would be very slow and the process would be terribly inefficient given stainless steel's low thermal conductivity

and the chamber's large thermal mass. Other means were needed to move heat into or out of the vacuum chamber.

Commercial systems commonly utilize recirculating fluid systems, some of which use synthetic fluids, while others use gaseous and Liquid Nitrogen (LN₂). Several companies [5] [4] [15] design and build custom thermal control hardware and software for systems like what was needed for the SSEL. Less expensive were commercial recirculating chillers that could control the temperature within the range that the SSEL needed. Companies were contacted for quotes for complete thermal systems as well as for stand-alone recirculating chillers. Complete systems ranged from \$100k to \$200k, which greatly exceeded the project budget and design goal. Recirculating chillers that met our requirements (up to 2 kW) were much less expensive at \$30k but also exceed the project budget and design goal. While these recirculating chillers could provide the needed temperature control and heat pumping capability over the entire temperature range, they were far too large for the lab space available and would have produced a great deal of undesired heat and noise in the lab.

Other common systems used resistive heaters and liquid Nitrogen to regulate temperature. These were much less expensive but required more setup with each use. Electrical power could be regulated to control the temperature above ambient, and could be used in parallel with the liquid Nitrogen to regulate temperature below ambient. Liquid Nitrogen at atmospheric pressure has a boiling temperature of 77K, costs approximately \$0.06 per Liter, and is readily available, so it is an excellent choice for use in cooling TVac systems.

In addition to the above two commercial thermal control options, a novel control method using Thermoelectric (TE) devices was considered. TE devices operate by applying power to a Bismuth Telluride semiconductor. These semiconductors are doped, some positive and some negative, and are arranged in pairs between two

substrates. They are connected in series electrically and in parallel thermally. When the temperature difference across the two substrates is minimum (0°C) the TE device is at its maximum pumping capacity. When the temperature difference is at its maximum (typically 70°C) the pumping capacity is at its minimum. Applied power causes the flow of heat from one side of the device to the other and by reversing the polarity heat can be moved in the opposite direction. Because the amount of heat flowing through the devices can be easily regulated by controlling the applied voltage, they are ideally suited for fine temperature control of better than 0.1°C [16]. The SSEL TVac system could benefit from this capability, but it was not necessary as 2°C control is all that was required for the SSEL's system.

A possible operational scenario using TE devices is as follows. On one side of a TE device is the base plate and on the other is a heat sink of some kind, all of which are initially isothermal meaning that the pumping capacity of the TE device is at its maximum. The rate of temperature change of the base plate is set at $5^{\circ}\text{C} / \text{min}$ starting at 25°C . After 11 minutes the base plate temperature is now 80°C and a difference of 75°C exists across the TE device, assuming that the heat sink is maintained at 25°C .

From the FEA model, the maximum heat load is expected to be approximately 1881 W. One TE device cannot handle this heat load even at its maximum pumping capacity (20-150 W depending on the model); therefore, a number of TE devices must be set up in parallel. In the scenario described above, the maximum heat load on the TE devices occurs when the temperature difference is greatest (75°C) and where the pumping power of the TE device is at its minimum. If a high-end (150 W) model is assumed, it could be possible to pump up to 10 W with a 75°C difference across the TE devices. With this assumption, 188 TE devices are necessary. A 150 W TE device has a physical footprint of 4 by 4 cm which means that 0.3 m^2 are required for

all 188 TE devices. Only 0.441 m^2 are available on one of the base plate's circular faces. Thus, TE devices are not a viable option at this heat load.

If the $5^\circ\text{C}/\text{min}$ requirement were relaxed to $1.5^\circ\text{C}/\text{min}$, only 833 W would be required. Repeating the same analysis as above for this heat load only 84 TE devices are needed, requiring only 0.1344 m^2 . While this area is less than the available area, it does not factor in mounting the TE devices. To mechanically mount each of these 84 TE devices, roughly 2 cm is needed between each row and column in an array of TE devices to allow for spacers and room for wiring them together. For a base plate diameter of 74.93 cm, the largest square that can fit in the area of the circular base plate has a side length of 52.98 cm as calculated by using Equation 4.3.

$$L = \sqrt{\frac{\text{Diameter}^2}{2}} \quad (4.3)$$

To find the maximum number of rows and columns of TE devices that can fit within this square center area with adequate spacing, Equation 4.4 is used.

$$L = (W + S)N + W \quad (4.4)$$

Here, W is 4 cm (the width of a TE device), S is 2 cm (the spacing between TE devices), and N is the number of rows and columns in the square area. Thus, N is 8.16 which is rounded down to 8. Therefore, only 64 TE devices will fit in the center square leaving 20 that must be mounted in the sectors outside the square center area. If S is reduced to 1 cm, N increases to 9 after rounding down. This leaves only 3 TE devices to be mounted in the sectors outside the square center area, which could easily be done.

In addition to the TE devices, a heat source or sink would be necessary. The afore-mentioned recirculating fluid or resistive heater/liquid Nitrogen system would be necessary in addition to the TE devices for such an arrangement to operate within the bounds of the First Law of Thermodynamics. Either of the heat sink/source methods would reduce the temperature difference across the TE devices making them more efficient, but also adding additional complexity, cost, and an additional failure mode. Therefore, a TE controlled system was not a cost effective or practical solution for the SSEL TVac base plate. A resistive heater and liquid Nitrogen system was the best choice for thermal control of the base plate.

Liquid Nitrogen Cooling System Design

Two coinciding designs, one for the heaters and the other for the liquid Nitrogen, were used to achieve the desired temperature control. Research into commercial TVac systems showed that many systems utilize a serpentine tube mounted to the bottom of the base plate for the flow of liquid Nitrogen [15]. Others use more of a jacket type plate which provides a much more uniform temperature distribution [5]. These jacket style plates cost an order of magnitude more than the serpentine tube style plates, around \$50k which is beyond the design goal. The serpentine tube style is often made with a Copper tube brazed or soldered to a flat Copper plate. The Copper plate is then attached to the bottom side of an Aluminum plate. The serpentine tube design was used for the SSEL TVac system to reduce cost and complexity.

The first design concept for the serpentine tube design sought to mimic the multiple flow paths typical to jacket style plates. These plates split the incoming fluid flow into several parallel paths in order to achieve a uniform temperature distribution. For the tube design, this proved to be a significant mechanical challenge. In order to accommodate a Nanosatellite sized test object and also fit in the vacuum chamber,

the base plate had to be a round disc which meant that evenly spaced, parallel flow paths had to be of differing lengths. To do this meant using multiple tubes, connectors, elbows, etc., and mounting these components within the base plate envelope and maintaining good thermal contact proved to be very difficult. The butt-type connections required for tube connectors prevented the tube surface from mounting flush with the flat Copper plate, leaving a gap of between 1 and 2 mm. Therefore, an alternate design was required.

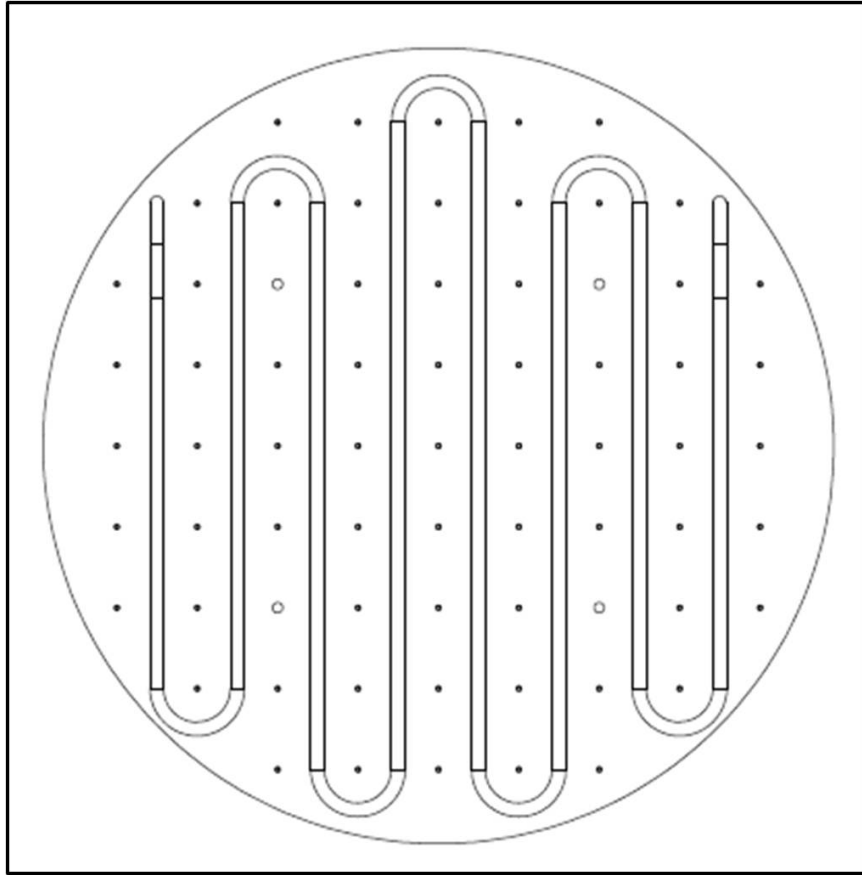


Figure 4.7: First Base Plate Tube Design (Not Final)

The next best design was a single tube bent into a shape that would fit the circular disc. This design, shown in Figure 4.7, had eight parallel lengths of tubing in the serpentine path. To check the temperature distribution of this design, the thermal

analysis software [12] was again used to run a transient analysis. Much was unknown about how the liquid Nitrogen would behave in this system, and for that reason the flow rate and phase of the fluid were both unknown. Therefore, estimating the fluid velocity, Reynold's number, and the convection coefficient in the tube became a non-trivial problem, one which, if attempted, would have provided questionable results at best. For this reason, the inner surface of the tube was assumed to be at a uniform temperature throughout. It must be noted that this assumption did not accurately represent how the system would actually behave – the temperature was expected to change as a function of distance along the tube length.

Having narrowed the choices to this Copper plate and tube design, a compliment to the Aluminum base plate was designed. The Copper plate was designed with the same outer diameter as the Aluminum plate and with a hole pattern to match the underside of the Aluminum base plate (Figures A.5, A.6, and A.7). The plates were designed to be machinable, as was the tube which has adequate separation between each row and standard bend radii. This mechanical design was initially considered for use with a re-circulating chiller, and it remained an option once liquid Nitrogen became the chosen cooling method. However, while a re-circulating chiller was being considered, a thermal analysis was performed on the serpentine tube design to assess the temperature distribution on the base plate assembly while transitioning from -40°C to 80°C . Rather than discarding that analysis, it was used to assess the temperature distribution when liquid Nitrogen was used.

For that analysis, the Aluminum plate, Copper plate, and Copper tube were all set to an initial temperature of -40°C . The tube was then heated at a rate of $5^{\circ}\text{C}/\text{min}$ to 80°C . At the final time step where the temperature difference across the plates was expected to have the widest spread, the temperature distribution in the center of the plate (top surface) was $79^{\circ}\text{C} \pm 1^{\circ}\text{C}$, well within acceptable limits. Around the edges,

however, the temperature was as low as 55°C near the ends of the tube. One could expect a similar large temperature difference between the center and the edge of the plate when cooling to -40°C with liquid Nitrogen. This large difference indicated that modifications to the design were necessary.

An additional row was added to each end of the tube to better cover the entire area. An identical analysis as just described was performed, and the results indicated a much better distribution on the top of the plate around the edge, as shown in Figure 4.8 (where the temperature is shown in Kelvin). In several areas the temperature was as low as 73°C, 5°C colder than the warmest places on the same surface. This difference was not of great concern because a test object would be centered on the plate, and these cooler areas around the edge of the plate would not affect the test object's temperature distribution to any great extent. The final Copper tube design (Figures A.8, A.9, and A.10) fit between the rows of bolt holes on the Aluminum and Copper plates and attached to these plates as shown in Figure A.11. The results of the above analysis also indicated that if liquid Nitrogen were used and the base plate assembly were cooled, the warmest places along the edge of the plate would be -33°C, only 5°C warmer than the coldest places on the same surface. The center would have been -39 +/- 1°C, well within the acceptable limits. For these reasons, the analysis performed for the re-circulating chiller indicated that the temperature distribution during cooling, with the serpentine tube design, would be adequate when liquid Nitrogen was used.

Heater System Design

As previously mentioned, heaters also had to be mounted to the base plate. Two types of heaters, namely power resistors and tape heaters, were available to supply the needed power. These heaters would both work in vacuum, and would not conflict

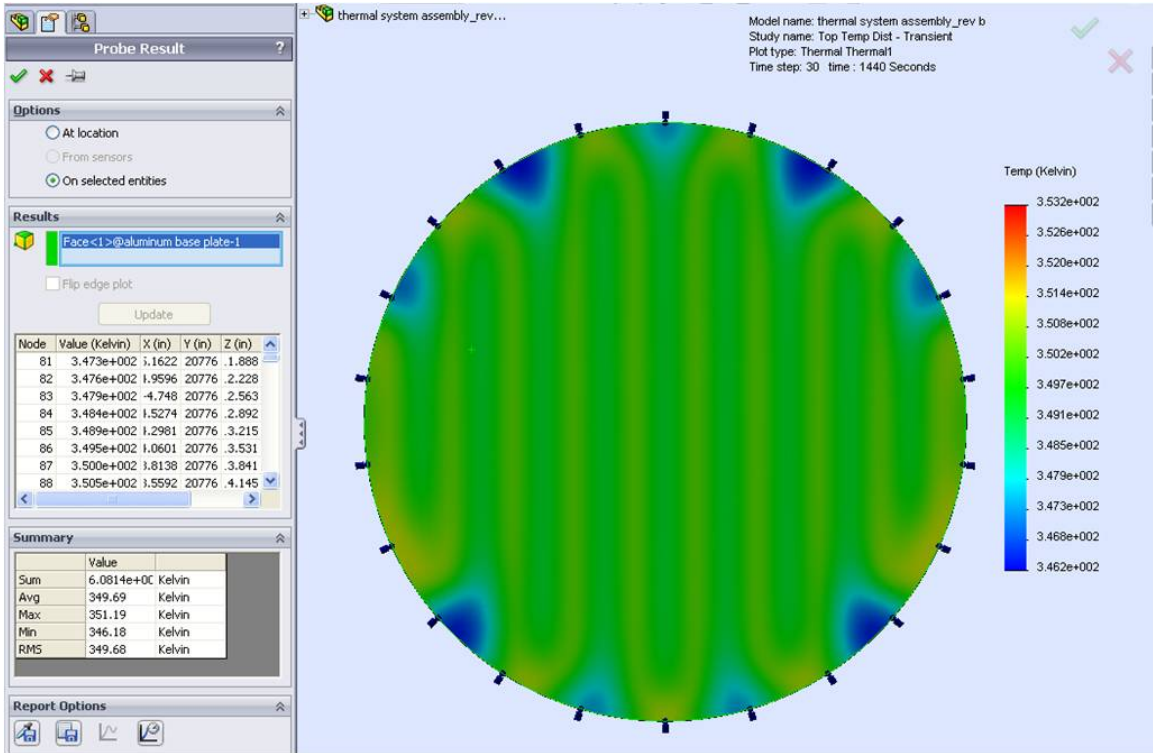


Figure 4.8: Thermal Analysis Results of the Second Base Plate Tube Design

with the tubing design. Tape heaters were ideal for this application because they could apply heat to the base plate far more uniformly than power resistors. One company in particular [17] was well known for its robust and reliable tape heaters, and had several options for using their heaters in vacuum – the deciding factor for choosing them over any other supplier. From Table 4.1, the minimum power that these heaters had to be able to supply was 833 W, but because the goal was to have a rate of temperature change of 5°C/min, the initial design criteria was 1881 W.

For the SSEL TVac system, RF noise had to be kept to a minimum, thereby prohibiting AC power inside the chamber. AC Power would have been the simplest and least expensive option because it would have required the least amount of support equipment. Joule’s Law for direct current (DC), Equation 4.5, dictates the relationship between electrical power, current, and voltage. Thus, a suitable combination of

voltage and current had to be found to meet the power requirement. Before selecting the IV pair, the DC power supply, electrical feedthrough for the chamber, and wiring had to be considered. Electrical feedthroughs were available with capabilities up to 2kV and several hundred Amperes – more than adequate for 1881 W – so little consideration was given to feedthroughs when selecting the IV pair. DC power supplies up to 2 kW were available from companies like Agilent Technologies. However, these supplies cost between \$5k and \$10k – almost half of the project budget, and so something else was needed.

$$P = I \times V \tag{4.5}$$

The SSEL had a 1 kW DC power supply that was rarely used, and so the decision was made to use this power supply solely for the TVac base plate. The consequence of this was that the rate of temperature change of the Base Plate would not exceed approximately 2°C/min, but since this met the minimum requirement it was an acceptable trade-off. This supply could regulate 0 to 60 Volts DC, and source up to 16.67 Amps. All wiring, therefore, had to meet or exceed these ratings. Wiring on the outside of the chamber had to be at least 10 AWG. Inside the chamber the ratings did not follow AWG standards because of the vacuum environment, so all wire inside the chamber was selected according to vacuum rated wire specifications.

Having established the power criteria and the IV limits, tape heaters were disqualified as an option because commercially available heaters could not handle the power per unit length that was required; custom heaters would have been necessary which would have added to the overall system cost. A design using power resistors was therefore considered. The mechanical design of the base plate had 65 holes that

mount the Copper plate to the Aluminum plate. Shunt resistors (a type of power resistor) were available which could be bolted to the Copper plate via the same bolts that held the two plates together. Because they were mounted to the bottom of the base plate assembly, there was no force applied to these heaters other than the force from tightening the bolts. Therefore, this was an acceptable way to attach the heaters to the base plate.

The number of heaters was set at 65; however, the resistance of the heaters still had to be decided. The entire 60 Volt range of the power supply would be used to regulate temperature. At 1 kW and 60 Volts, the equivalent resistance, R_{eq} , was found by substituting Ohm's Law, Equation 4.6 into Equation 4.5, giving an equivalent resistance of 3.629Ω . How the resistors were wired together determined the resistance per resistor. Table 4.2 lists the three simplest configurations and the required resistance for each. Also listed are the string resistance R_s , string current I_s , and the power through each resistor P_R .

$$V = I \times R \tag{4.6}$$

The style of resistor that would mount to the base plate limited the choices of resistance values. Resistors within a few Ohms of 234Ω were not available. Choosing a resistance of 0.05Ω (the closest to 0.055Ω) resulted in a string current of 18.5 Amps which was greater than the DC power supply could source. A resistance of 10Ω (nearest value to 9.36Ω) resulted in a total power of 936 W and a power per resistor of 14.4 W when the resistors were arranged in 13 strings of 5 resistors each. At this power, the base plate could be heated at approximately $2^\circ\text{C}/\text{min}$ which met the minimum requirement. The resistor model FPR 2-T218 from Reidon was chosen

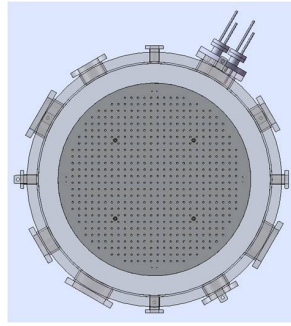
Table 4.2: Heater Resistor Selection

Configuration	Resistance Function	R (Ω)	R_s (Ω)	I_s (A)	P_R (W)
All Series	$R_{eq} \div 65$	0.055	3.629	16.67	15.38
All Parallel	$R_{eq} \times 65$	234	234	0.256	15.38
13 Strings of 5 Resistors in Series	$R_{eq} \times \frac{13}{5}$	9.36	46.8	1.28	15.38
As built: 13 Strings of 5 Resistors in Series	$R_{eq} \times \frac{13}{5}$	10	50	1.2	14.4

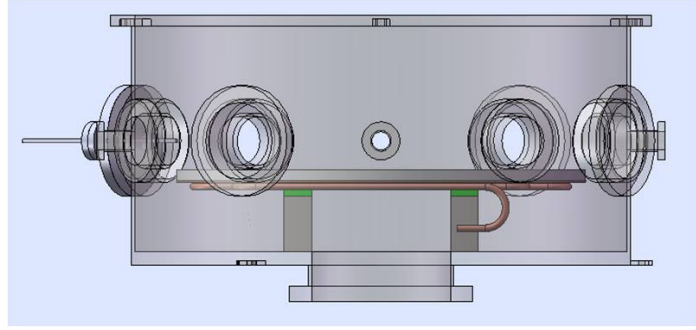
as the heater resistor for the base plate. Its power rating of 30 W (when mounted to a heat sink – the base plate) was not exceeded even at the power supply’s maximum output, it could operate from -40°C to +130°C (within the temperature limits of the base plate), and it was made from vacuum safe materials. Thus, an appropriate heater had been selected.

Insulator Design

Three base plate requirements specify that, as a goal, no more than 25 W are to be lost to the chamber, ambient environment, or the thermal shroud. The base plate itself must rest on the bottom of the chamber, and the thermal shroud rests on the top of the base plate. Insulators of some kind had to be used to meet these 25 W goals. Between the base plate and the chamber bottom, four legs support the base plate and test object as well as thermally insulate the base plate from the bottom of the chamber. On the top of the base plate, eight insulators situate the thermal shroud and insulate the two. Two materials were considered for these insulators: Delrin 507 (Black Acetal Copolymer), and Ultem (Polyetherimide). Both exceed the expected temperature range of the system and would remain structurally rigid over the entire temperature range, as well as met low outgassing requirements for use in high vacuum.



(a) Solid Model of the Base Plate and Chamber, Top View



(b) Solid Model of the Base Plate and Chamber, Side View

Figure 4.9: Solid Model of the Base Plate and Chamber

The results of the mechanical design process are shown in Figure 4.9. The chamber is included to show the base plate in perspective (the chamber is transparent in these views so that the base plate is visible). Between the inside of the chamber and the bottom of the base plate was a 12.7 cm (5 in) gap, the height of the insulators (Figure A.13) for the tubing and access space (see Figure 4.9b). Around the edge of the base plate was a 7.62 cm (3 in) gap for access to the underside of the base plate (see Figure 4.9a). Also, the top of the base plate did not exceed the height of the radial chamber flanges. Requirements 8, 9, and 10 for the base plate were met with this design. The thermal requirements were also be considered.

The heat transfer through bottom insulators were considered first. Figure A.13 shows the physical dimensions of the part. For a worst case heat flow analysis, one end was set at 77 Kelvin (saturation temperature of liquid Nitrogen at STP) and the other at room temperature (295.15 Kelvin). The heat power through both Delrin (having conduction coefficient, k , of $0.4 \frac{W}{mK}$) and Ultem ($k = 0.122 \frac{W}{mK}$), was calculated using Fourier's Law, Equation 4.7, where A was the axial cross sectional area and $\frac{dT}{dx}$ was the temperature gradient along the length of the part. The total heat transferred through

the four Delrin insulators was 3.18 W, and through the four Ultem insulators was 0.97 W. Both materials easily met the 25 W goal and were easily machinable. However, Delrin was chosen because Delrin stock was one fifth the cost of Ultem per cubic volume.

$$q = -kA \frac{dT}{dx} \quad (4.7)$$

The compressive strength of the bottom insulators also had to be considered in the design. Base plate Requirement 3 specifies that they must be able to support a test object with a mass of 50 kg. These insulators not only support the mass of the test object, but also the mass of the base plate and the thermal shroud when it is in use. Having not yet considered the thermal shroud, its mass is not known and therefore this calculation will have to be addressed after the shroud has been designed.

The heat transferred through the shroud insulators was calculated in a similar manner, except the worst case temperatures were more extreme. During thermal balance tests the thermal shroud would be operated as cold as possible (77 Kelvin) and the base plate could be operated as high as 353.15 Kelvin (80°C). Figure A.12 shows the dimensions of these parts. The bottom surface was set to 353.15 Kelvin and the 1.27 cm by 3.81 cm (0.5 in by 1.5 in) step (on which the shroud rests) was set to 77 Kelvin. Repeating the analysis using Equation 4.7 yielded 22.45 W through the eight Delrin insulators and 6.85 W through the eight Ultem insulators. Given that both of these met the 25 W goal, Delrin was again chosen as the material for these insulators.

Although not a physical insulator, the design of the base plate affected its heat transfer with the chamber walls. Aluminum and Copper were the chosen materials

of the base plate components. In their finished state they were polished and so they had very low emissivity ($\epsilon = 0.03$). However, as they age their surfaces can oxidize and the emissivity increases, up to $\epsilon = 0.3$. The chamber was assumed to remain at room temperature (295.15 Kelvin) and the base plate heated to 80°C, which simulated the conditions under which the worst case heat loss would occur. All surfaces were considered diffuse, gray, opaque, and isothermal. For the purposes of this analysis, a two surface enclosure was formed: the chamber bottom and walls (up to the height of the top of the base plate), and the other a disc of equal radius as the chamber bottom (simulating the base plate surfaces).

The assumptions made for a relatively straightforward calculation. Incropera [18] presented a solution for a two surface enclosure with diffuse, gray surfaces (Equation 4.8). Substituting the values appropriately yielded a heat loss between the base plate and the chamber of 74.5 W at the beginning of the base plate's life ($\epsilon = 0.03$). If the surfaces oxidized over time the heat loss could increase up to 329.6 W. This design did not meet the 25 W goal, but it did meet the requirement by choosing materials with low emissivity and thereby minimizing the heat loss as much as possible during the worst case conditions. Oxidation could be prevented or slowed in part if the base plate were maintained in a Nitrogen rich environment when not in use, and by minimizing the time that the chamber was open to atmosphere.

$$q_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\epsilon_1}{\epsilon_1 A_1} + \frac{1}{A_1 F_{12}} + \frac{1-\epsilon_2}{\epsilon_2 A_2}} \quad (4.8)$$

Support Equipment Selection and Design

Equipment outside of the chamber was necessary to heat and cool the base plate. As previously mentioned, the heaters were driven by a 1 kW DC power supply. Cooling the base plate was done by forcing liquid Nitrogen through the base plate tubing, which could be accomplished either by a gravity feed system or a pressurized system. A gravity feed system was the simpler of the two options, and required little setup. Liquid Nitrogen was readily available through the MSU Physics Department, and filling a dewar and transporting it to the SSEL Lab would have required no special equipment. However, a gravity fed system would have required the means to raise the dewar above the height of the base plate as well as prevent the dewar from falling on nearby personnel. It would have been possible to raise the dewar 1.5 m above the base plate, resulting in a feed pressure of approximately 6.89 kPa (1 psi).

As mentioned earlier, much was unknown of how the liquid Nitrogen would behave in the base plate tubing and what flow rate would be required. Therefore, the ability to control the pressure and hence the flow rate was desired. A pressurized tank would allow much better control of the feed pressure. Pressurized tanks could be rented at a daily rate from local gas supply companies and could be delivered directly to the lab. These tanks were rated to high pressures, up to 1551.3 kPa (225 psi), and could supply cold gas or liquid Nitrogen. It was useful to cool the liquid Nitrogen line between the tank and the base plate before feeding liquid Nitrogen to prevent very fast boil-off of liquid Nitrogen and a rapid pressure rise in the lines. The cold gas output provided the means to do this. Another benefit of using a liquid Nitrogen tank over a gravity feed system was that the flow rate of cold gas or liquid Nitrogen could be regulated by the hand valves on the tank. The pressure in the tank (feed pressure) could also be regulated to control the flow rate. A tank, therefore, would provide a

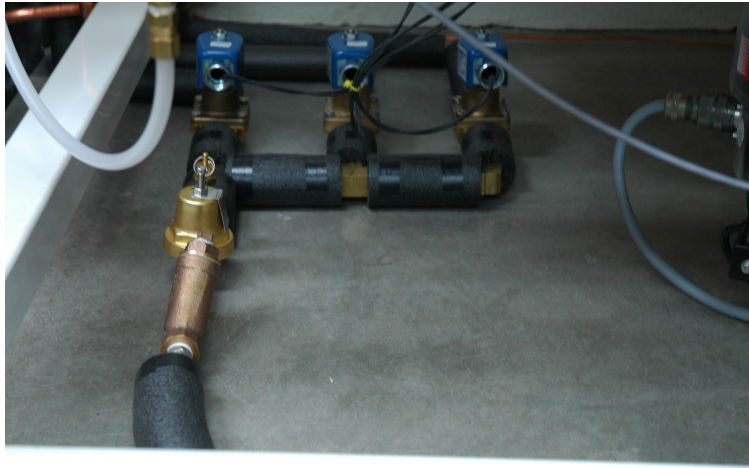
much larger range of flow rates than a gravity feed system, and would provide the best level of control of the flow rate.

Built-in safeties had to be included in the liquid Nitrogen lines. Liquid Nitrogen has an expansion ratio of nearly 1:700, so if it were trapped in an enclosed space the pressure would increase as the liquid turned to gas which, if not released, could cause the vessel to explode. Pop-safety valves set above the expected operating pressure were included in the line to prevent this from happening. Cryogenic rated lines and on/off valves (Figure 4.10a) were used to reduce the risk of component failure. Custom bent copper tubes (Figure 4.10b) connected the cryogenic valves to the fluid feedthrough on the chamber, a commercial part designed for liquid Nitrogen use.

Temperature Control Software

Just as the Vacuum System was controlled by a user friendly computer interface, so too was the base plate temperature. A user interface was created with National Instruments LabVIEW to interface with the thermal system hardware. A common technique for controlling temperature are PID (Proportional Integral Derivative) algorithms, which have been used for many years and are extremely robust and versatile. The PID algorithm is ideally suited for systems that respond quickly to changes and are analog controlled. The base plate heater was controlled by an analog DC power supply, but the liquid Nitrogen cryogenic valve was digital (ON/OFF). For this reason an alternative control algorithm was implemented.

The base plate temperature was completely user controlled via the computer interface. The user supplied the desired temperature setpoint, the desired rate of temperature change, and a hysteresis value and the code responded accordingly. If the temperature setpoint was above the current temperature, the heaters were turned on. The heater power was determined by the rate of temperature change value that



(a) Cryogenic Valves in the Liquid Nitrogen Line



(b) Copper Tubing Connecting to the Chamber

Figure 4.10: Liquid Nitrogen Line Hardware

the user supplied. Once the base plate reached the desired temperature, the code turned off the heater and waited for the temperature to drop below the setpoint less the hysteresis value. When it did, it turned the heaters back on until the temperature reached the setpoint. This method of control is often called a bang-bang controller.

When the setpoint temperature was below the current temperature the heaters were turned off and the liquid Nitrogen valve was opened. Once the base plate reached the desired temperature, the valve was closed again, and cycled open and closed based on the hysteresis, just like during heating. The rate of temperature change was dependent on the flow rate of cold gas or liquid Nitrogen, both of which were adjusted by hand valves on the liquid Nitrogen tank.

The hysteresis value provided a way to protect the hardware (valves and power supply) from frequent cycling and premature failure. With a reasonable hysteresis value the power supply and valve were cycled on and off every few minutes instead of every few seconds. Built into the automatic mode control were safeties that prevent the temperature from going beyond the operational limits of the base plate and the heaters. Another feature of the code was the ability to control both the heater and the valve manually.

Base Plate Manufacturing and Assembly

Manufacturing of the base plate components was largely contracted to outside shops, and as much as possible local shops were used to save on shipping costs [19]. The Aluminum and Copper plates, as well as the Delrin insulators, were machined locally [19]. The serpentine Copper tube was contracted to an out-of-state machine shop that specialized in custom tube bending [20]. Joining the Copper plate and serpentine tube together was contracted to vacuum furnace brazing facility [21] according

to the assembly shown in Figure A.25. This process was chosen because it maintained the parts in a clean environment, and more importantly, prevented the Copper plate from warping during the process. Brazing and soldering were alternatives that are typically done using a torch to heat the material at the location of the joint, but these methods are known to cause warping of thin, flat plates. Good thermal contact between the Copper and Aluminum plates was assured by minimizing this warping. Welding metals together is also very common, but because Copper has a low melting temperature the Copper parts would melt and deform. The vacuum furnace brazing heated both parts uniformly and thereby prevented warping. Thus, vacuum furnace brazing was chosen to join the parts together.

Before assembling the Aluminum plate and Copper plate/tube together, both were thoroughly cleaned. The Aluminum plate had not been cleaned at the machine shop and so the machine oils, dirt and metal flakes had to be removed. This first cleaning was performed using a car wash sprayer with mild soapy water to remove roughly 90 percent of the contaminants on the plate. Second, a water and Liquinox bath with an ultrasonic cleaner loosened and removed much of the contaminants in the many drilled holes and on the surfaces. Finally, the plate was soaked in an alcohol bath and a syringe used to force alcohol into the holes and remove any remaining oils or grit (Figure 4.11a). Even this cleaning did not remove all of the metal flakes and oils from the holes, so tweezers, q-tips and pipe cleaners were used to clean each of the holes until the q-tips or pipe cleaners came out of the holes clean. The $\frac{1}{4}$ -20 helicoils were then inserted into each of the holes on the top surface of the plate. The Copper plate and tube assembly was scrubbed in a large sink to remove oils and residue and then wiped with alcohol. It returned from the furnace brazing facility very clean, but with oils from handling it with bare hands. An alcohol bath and pipe cleaners

were used in a similar way to clean it before mounting it to the bottom surface of the Aluminum plate along with the heaters.

The thirteen 5-resistor heater strings were soldered together and scrubbed with alcohol to remove the solder flux from each of the connections. Kynar shrink tubing covered the terminals of each of the heaters to prevent them from shorting to the copper surface of the base plate. The 5-resistor strings were connected in parallel, and the resulting resistor network was then connected to the DC power supply.

Assembly of the entire base plate required having each of the components cleaned prior to starting the process so that it could be installed in the chamber as soon as it was finished. All of the bolts and washers were cleaned in an ultrasonic bath with Liquinox detergent, rinsed with tap water, and then soaked in an alcohol bath for at least two hours. The Aluminum plate sat with its top surface on a clean table surface. The Copper plate/tube assembly rested on the bottom surface of the Aluminum plate and the heater strings were arranged with their mounting holes lined up with the mounting holes of the Aluminum and Copper plates, as shown in Figure 4.11b. The #4-40 socket head cap screws fed through the heaters and the Copper plate, and threaded into the bottom of the Aluminum plate, thus holding everything together. The Delrin standoffs were then mounted in their respective locations. Type E thermocouples were then mounted to the bottom of the base plate in two places, and the flexible hoses for liquid Nitrogen were attached to the serpentine Copper tube. The final assembled base plate is shown in Figure 4.11c.

With the help of an SSEL staff member [10], the base plate assembly was placed into the chamber as seen in Figure 4.11d. The ends of the thermocouples were then connected to pins on a low power electrical feedthrough, and the liquid Nitrogen hoses connected to their respective feedthrough, shown in Figure 4.11e (before it was

installed on the chamber). The shroud insulators were then bolted to the top of the Aluminum plate.

Closing Remarks

Design of the base plate centered around its functional requirements as well as the physical limitations of the vacuum chamber. Analysis showed that it was capable of heating and cooling a 40.5 kg test object at a rate of 1.5°C/min minimum over a range of -40°C to +80°C. The requirements for insulating the base plate from the chamber and thermal shroud were achieved by plastic Delrin insulators, and all materials were vacuum safe.



(a) Cleaning the base plate in an alcohol bath



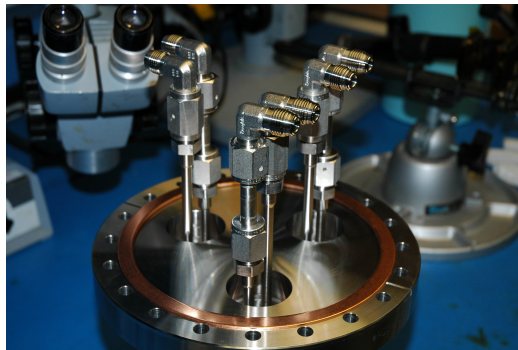
(b) Assembling the heaters to the Aluminum and Copper plates



(c) After installing the thermocouples and liquid Nitrogen lines



(d) Installing the base plate in the chamber



(e) Assembly of the Liquid Nitrogen feedthrough

Figure 4.11: Base Plate Assembly Process

THERMAL SHROUD DESIGN

Thermal Shroud Requirements

A set of requirements was defined for the thermal shroud as it was for the base plate. These requirements were derived from the Thermal System Requirements (found in Chapter 4) and specify the minimum and required functionality. The thermal shroud shall:

1. Be capable of cooling to temperatures as low as -100°C , with a goal of -196°C .
2. Be capable of heating to 80°C .
3. Be capable of increasing or decreasing temperature at a rate of at least $1^{\circ}\text{C}/\text{min}$ with a goal of $5^{\circ}\text{C}/\text{min}$.
4. Have a high emissivity inner surface (> 0.8) to achieve good radiative coupling with the test object.
5. Be closed on top.
6. Have two independently controllable zones: one for the top and the other for the side(s).
7. Be insulated to minimize the heat loss to the baseplate, with a goal of less than 25 W.
8. Be insulated to minimize the heat loss to the chamber, with a goal of less than 25 W.
9. Rest on top of the base plate during operation.

10. Have an outer diameter of less than 74.9 cm (29.5 in) so that there is at least 7.62 cm (3 in) of clearance between the shroud and the chamber on all sides.
11. Have an inner diameter of greater than 66 cm (26 in) to accommodate a 60.7 cm (24 in) diameter spacecraft.
12. Be at least 55.9 cm (22 in) tall on the inside to allow 5.08 cm (2 in) clearance for a 50.8 cm (20 in) tall nanosatellite.
13. Be no more than 61.3 cm (24.125 in) tall on the outside, including the top, to allow adequate clearance between the shroud and the chamber lid posts when moving the chamber lid.

Design Process

The options presented in Chapter 4 were also considered for use with the thermal shroud, and for the same reasons resistive heaters and liquid Nitrogen were selected as the heating and cooling mechanisms for both zones (side and top). The thermal and mechanical designs for the thermal shroud were highly dependent on one another. Material selection and possible mechanical design options were considered first, followed by the thermal analysis once the material was selected. The final configuration of the system hardware was then chosen based on the results.

Mechanical Design of the Shroud Body

Requirement 9 above specifies how the shroud and the base plate interface during operation, and Requirement 7 specifies the allowable heat transfer between the two. The Aluminum base plate (Figures A.1, A.2, A.3, and A.4) has a pattern of #6-32 holes on its top surface for insulators (Figure A.12) which serve to situate the thermal

shroud and to insulate it from the base plate. Their circular pattern suggested a 71.1 cm (28 in) diameter cylinder for the shroud body which met the minimum size requirement and left adequate room on the outside for heating and cooling hardware. Requirements 12 and 13 specify the allowable height range of the shroud. Thus, a height of 58.4 cm (23 in) for the shroud body was chosen to maximize the internal height and also to allow space above the shroud top for its heating and cooling hardware.

A commercial solution as well as several custom design options were considered, all of which closely matched the sizes listed above. Schedule concerns initially drove the need to find a commercial product that could be purchased and easily installed; however, the cost for a commercially available shroud was roughly three quarters of the allocated budget for the entire project making it a non-viable option. While commercial shrouds boasted excellent temperature uniformity and engineering techniques unavailable within our limited budget, similar techniques were applied to a simpler custom design. For the shroud body, three designs were considered: a single tube in the shape of a helical spiral, and wrapped around the shroud body; multiple parallel tubes, connected in parallel; and multiple parallel tubes, connected in series.

The helical design consisted of a single tube bent into a helix, extending from the bottom to the top of the shroud body and fitting snugly with the outer surface of the shroud itself. While this design could be manufactured, several limitations made it difficult. First, the tube had to be bent into the helical shape. Tube bending companies with CNC tube benders [20] had the manufacturing capability to do this at reasonable cost, but a single, long tube was needed to accommodate multiple revolutions. Several tubes could have been connected together, but the joints of those tubes would have formed gaps between the tube and the shroud and localized hot spots would have been present. A single helical tube had to be approximately

15.24 m (50 ft) long and continuous to achieve adequate thermal uniformity, and the only available material found in continuous lengths greater than 6.1 m (20 ft) was Copper.

Mechanically, this design was simpler than either of the other two parallel tube designs in that it required fewer pieces. However, like the other two options, it had to be designed to be lifted out of the chamber and moved out of the way while a test object was being installed. Copper is typically sold in a nearly annealed (soft) state and the only way to harden it is to cold work the material, thereby deforming it. Regardless of whether the tubing and the shroud were to be attached using soldering, brazing, or vacuum furnace brazing, the resulting assembly would have been annealed after assembly. If it were possible to harden Copper by dip-quenching then it would have been an acceptable choice, but because Copper can only be hardened by cold working, it did not suffice as an appropriate material for the shroud. The only way to use a Copper shroud would be in its fully annealed state and in the process of moving the shroud, a bump against anything would have easily dented or damaged the shroud. Thus, another material had to be chosen.

Ideally, the chosen material would also have a high coefficient of thermal conduction to minimize thermal gradients as well as to reduce the time it would take to come to equilibrium at a given temperature. The material used for the shroud body and the tubing needed to be the same to avoid mismatched coefficients of thermal expansion, minimizing thermally induced stresses. The shroud had the widest temperature range of all of the thermal system components, ranging from -196°C to 80°C , and would likely be cycled over this range several times in the course of one usage. Using two different materials would have caused stress during each cycle and would have reduced the usable lifetime of the shroud. Since Copper was not a feasible material choice, the helical design could not be used for the thermal shroud.

Meeting the requirements previously discussed meant using a material that could hold up to frequent installation and removal from the chamber, as well as meeting the needed thermal properties. Having reduced the design options to parallel tubes mounted to the shroud, the tubes needed to be 2.44 m (8 ft) long (the circumference of the shroud). Also of concern was cleanliness, or more importantly the ability to clean the shroud easily each time it was removed from the chamber. Typical vacuum chambers and components are made from stainless steel because of its structural strength, because it can be electropolished (i.e. smoothed, which makes cleaning easier, reduces surface adhesion of unwanted contaminants, and reduces outgassing rates), and because it does not tarnish. These properties made stainless steel an excellent choice mechanically.

Thermally, stainless steel was not a good choice because of its low thermal conduction coefficient. While stainless steel did not have very good thermal properties, it easily met the structural strength, availability, machinability, and cleanliness needs for the system. Given the few possible material choices for the shroud, the choice between the series and parallel tubes designs had to factor in the material choice. Both tube design concepts involved a set of evenly spaced tubes wrapped circumferentially around the shroud body. In the case of the series tubes design, the tubes were connected end-to-end with the connections on alternating ends of the tubes (Figure 5.1a). For the parallel tubes design (Figure 5.1b), two bulkhead tubes connected the ends of all 8 tubes together.

As discussed in Chapter 4, there were many unknowns regarding the liquid Nitrogen and how it would behave in the system and for this reason a qualitative, rather than quantitative, analysis was used to select the tube design for the shroud body. Ideally, liquid Nitrogen would flow through the parallel connected tube design evenly and provide a uniform temperature distribution. However, because the tubes

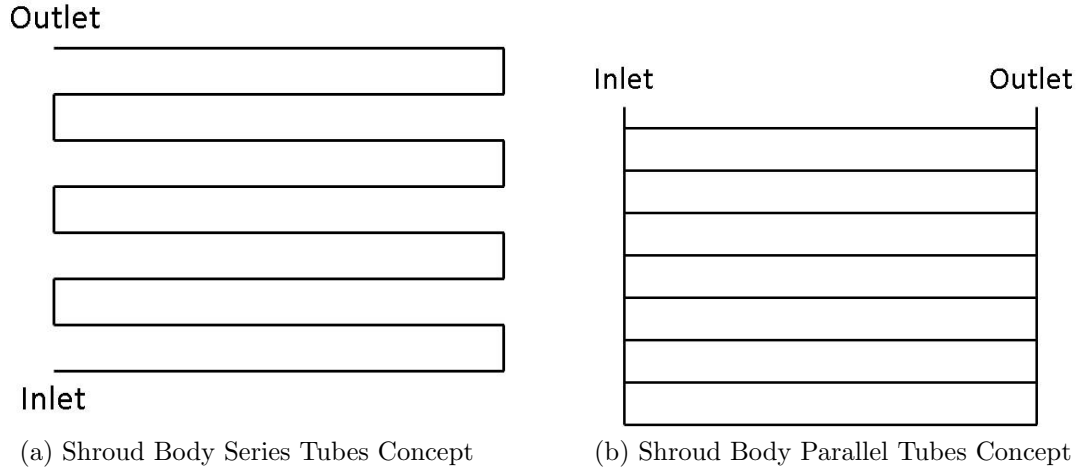


Figure 5.1: Satellite Model Mesh and Results

are stacked vertically, the lower tube would fill first and so on to the top because of gravity, creating a potentially large temperature gradient from bottom to top.

The series connected tube design would also fill from bottom to top because of gravity, also creating a potentially large temperature gradient. However, because its assembly was simpler and because there were fewer parts involved, it was less expensive to fabricate. For these reasons the series connected tube design was chosen for the shroud body. As with the base plate design, this analysis did not assume that the shroud body would have uniform temperature during operation – the temperature was expected to change as a function of distance along the tube length.

In addition to the tubing on the outer surface, the inner surface of the shroud body had to be considered. Requirement 4 specifies that the inner surface must have a high emissivity value to ensure adequate radiative coupling with the test object. Discussion with an industry contact [11] led to the selection of Aeroglaze Z306 flat black absorptive polyurethane, a low outgassing, high emissivity paint that was very commonly used in vacuum applications. Its emissivity was specified as 0.9, which exceeded the requirement.

The design considerations, as discussed above, lead to a custom design for the thermal shroud. The shroud body was a 58.4 cm (23 in) tall, 71.1 cm (28 in) diameter cylinder made of stainless steel (Figures A.14 and A.15). Wrapped around the shroud body were eight parallel tubes connected in series. Seven of the eight tubes were identical (see Figures A.16 and A.17) while the eighth had a custom bend on one end to facilitate connecting the tube to a flexible hose and remain within the allowable size envelope (see Figures A.30, A.31 and A.32). Connecting the tubes together were several 90° elbows and a short tube connector (Figure A.20). The fully assembled thermal shroud body is illustrated in Figures A.18 and A.19. Attached to each end of the shroud body tubing were 1.27 cm (0.5 in) VCR connections that attached to flexible hoses for the liquid Nitrogen. VCR connections were used because they provided a leak-proof seal (since they were to be used in vacuum) and could be easily disconnected and re-connected each time the thermal shroud was removed or installed into the chamber. By design, Requirements 7 and 9 through 13 were met.

Mechanical Design of the Shroud Top

The discussion above for the shroud body material also applied to the shroud top, and so stainless steel was also the material choice for the shroud top. For simplicity, and to meet the shroud height requirement, the shroud top was a flat plate with a diameter slightly larger than the diameter of the shroud body (Figure A.21). Given that the shroud top was a flat disc like the base plate, the tubing for the shroud top took a similar shape to that of the base plate Copper tube, forming a serpentine tube on top of the shroud top (Figures A.22, A.23, and A.24). The top plate and the serpentine tube were assembled together as illustrated in Figures A.26, A.27, and A.28. The inner surface of the shroud top was also painted with the Aeroglaze Z306 paint to meet Requirement 4.

Mounting the shroud top to the shroud body was accomplished by using eight insulating spacers made of Delrin plastic (Figure A.29). These were bolted to the painted underside of the shroud top plate using the #6-32 holes on the insulators, and an eye bolt was attached to four of the eight insulators through the #10-24 helicoil holes. A lifting harness was then attached to the eye bolts, providing a single lift point for the entire shroud assembly (body and top). The body was then attached to the the top via the 6.76 mm (0.266 in) through holes in the insulators. A ring pin fit snugly through these holes and through a corresponding set of holes on the top of the shroud body so that the two could be lifted together, and could also be separated easily should the need arise. This design was intended to insulate the shroud body from the shroud top and meet Requirements 5 and 6, and is shown in Figure 5.2.



Figure 5.2: Thermal Shroud Insulator Assembly

Attached to each end of the shroud top tubing were 1.27 cm (0.5 in) VCR connections that attached to flexible hoses for the liquid Nitrogen. VCR connections were used because they provided a leak-proof seal (since they were used in vacuum) and they could be easily disconnected and re-connected each time the thermal shroud was removed or installed into the chamber.

At this point, the design of the base plate and thermal shroud was complete and the total mass was known. In Chapter 4 the insulators under the base plate were designed, but their analysis could not be completed without the total mass of the system. Added up, the base plate and thermal shroud components had a mass of approximately 70 kg. The base plate requirements specify that the base plate (including the insulators) must be able to support the mass of a 50 kg test object. Therefore, the 4 base plate insulators had to support 120 kg (260 lbf). Each insulator had a cross sectional area of 11.4 cm^2 , which meant that the compressive force on each insulator was 253.72 kPa. The ultimate strength of the Delrin insulators is 151,685 kPa. Thus, the base plate insulators (the weakest component of the base plate) could easily support a 50 kg test object.

Heat Load Analysis

The heat exchange between the thermal shroud, the nanosatellite test object and the rest of the thermal vacuum system posed a non-trivial problem in terms of the geometry and the temperature conditions. Given that the thermal shroud would be used to heat and cool, both cases were examined to determine the worst case heat load on the shroud.

Thermal balance testing put the largest loads on the system because of its extreme nature. In a worst-case cold balance test, the satellite would be held at -40°C and the

shroud heated to 80°C. The reverse was true for a worst-case hot balance test. This 120°C drove the worst case heat load on the thermal shroud. It was assumed that the shroud body and top were isothermal, and that the satellite and the base plate were isothermal.

The entire thermal mass of the shroud had to be heated or cooled and so an energy balance was performed. Each of the shroud zones was considered a control mass, thus a First Law balance resulted in Equation 5.1 where $\dot{Q}_{in}$ is the energy (rate) required to heat or cool the zone, $\dot{Q}_{out}$ is the energy (rate) lost to the surroundings, and $\dot{Q}_{store}$ is the energy (rate) required to heat up the material of the shroud itself. Losses to the chamber, to the base plate, and to the test object itself were the most difficult to calculate and so were undertaken first.

$$\dot{Q}_{in} = \dot{Q}_{out} + \dot{Q}_{store} \quad (5.1)$$

Inside the shroud, the geometry of the components as well as the enclosure formed by the surfaces necessitated that a circuit network approach be used to determine the heat transfer between the surfaces. For simplicity, the nanosatellite was assumed to be a 45.7 cm (18 in) diameter, 45.7 cm (18 in) tall cylinder. All surfaces in the enclosure were assumed to be diffuse, gray, opaque, and isothermal. Figure 5.3 shows the five surfaces formed by this enclosure and are referenced as follows:

1. Nanosatellite Body
2. Shroud Body
3. Shroud Top
4. Base Plate Annular Area

5. Nanosatellite Top

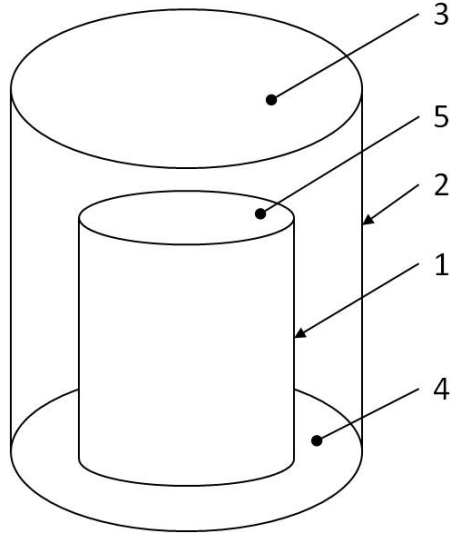


Figure 5.3: Thermal Shroud Surface Designations

The net radiation heat transfer, $\dot{q}_i$, from each surface was found using Equation 5.2, but in order to make use of this equation the radiosity, J_i , of each surface had to be found first.

$$\dot{q}_i = \frac{E_{bi} - J_i}{\frac{1 - \epsilon_i}{\epsilon_i A_i}} \quad (5.2)$$

$$E_{bi} = \sigma T_i^4 \quad (5.3)$$

The total emissive blackbody power, E_{bi} , is defined in Equation 5.3, ϵ_i is the emissivity of surface i , and A_i is the surface area of surface i . A set of five equations was created, one for each surface, using Equation 5.4 and solved simultaneously for the five radiosity values.

$$\frac{E_{bi} - J_i}{\frac{1 - \epsilon_i}{\epsilon_i A_i}} = \sum_{j=1}^N \frac{J_i - J_j}{(A_i F_{ij})^{-1}} \quad (5.4)$$

All variables in the set of equations generated from Equation 5.4 were known with the exception of the many view factors, F_{ij} . The view factor between surfaces 3 and 5, shown in Figure 5.4, was the most straightforward to find and used a formulation, Equation 5.5, presented by Incropera [18].

$$F_{ij} = \frac{1}{2} \left[S - \sqrt{S^2 - 4 \left(\frac{r_j}{r_i} \right)^2} \right] \quad \text{where}$$

$$S = 1 + \frac{1 + R_j^2}{R_i^2} \quad \text{and}$$

$$R_i = \frac{r_i}{L} \quad \text{and}$$

$$R_j = \frac{r_j}{L}$$
(5.5)

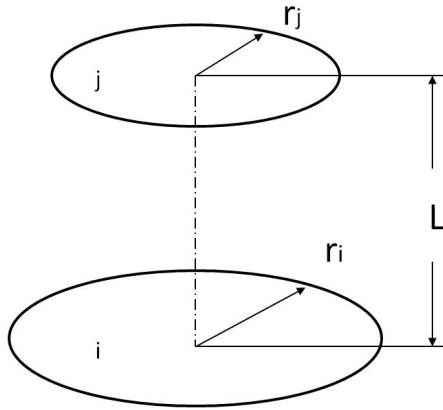


Figure 5.4: Coaxial Parallel Disks View Factor Geometry

The view factor between the satellite top and the shroud body, F_{52} , was found using a formulation presented by Leuenberger [22], shown in Equation 5.6. Figure

5.5 shows the geometry used for this calculation. It must be noted that the satellite top (Surface 5) does not see the entirety of the shroud body surface (Surface 2). Therefore, the height of the cylinder, L , used in this calculation was the difference between the shroud and satellite heights.

$$F_{ij} = \frac{1}{2} \left\{ 1 - \frac{R^2 + L^2}{r^2} + \sqrt{1 + \left(\frac{R^2 + L^2}{r^2} \right)^2 - 4 \left(\frac{R^2}{r^2} \right)} \right\} \quad (5.6)$$

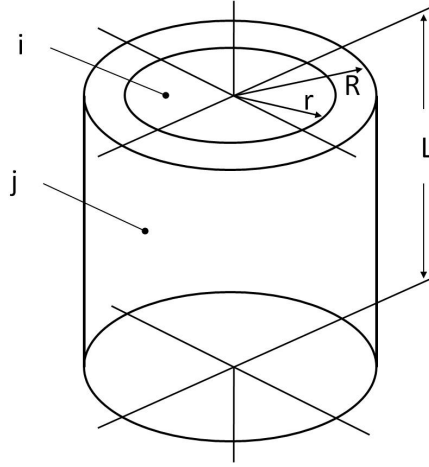


Figure 5.5: Geometry Between the Satellite Top and Shroud Body

The remaining view factors (F_{12} , F_{13} , and F_{14}) were found using a closed form technique developed by Rea [23] which utilized several cylinder-disk view factors to calculate the desired result. This method referenced two co-axial cylinders of differing lengths and radii, one inside the other (Figure 5.6). The cylinder-disk view factors were substituted into Equation 5.7 to obtain the view factor between the inner and outer cylinders, F_{ij} .

$$F_{ij} = 1 + \frac{bF_b}{L} + \frac{cF_c}{L} - \frac{(L+b)F_{L+b}}{L} - \frac{(L+c)F_{L+c}}{L} \quad (5.7)$$

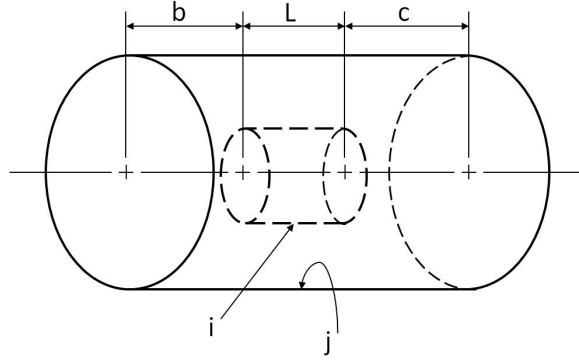


Figure 5.6: Geometry of the Concentric Cylinders

The cylinder-disk view factor, F_{mn} , was found by solving Equation 5.8, where h and r were the height and radius of the cylinder, respectively, and R was the radius of the disk. The notation given in Equations 5.7 and 5.8 is a view factor shorthand where, for example, F_b is the cylinder-disk view factor F_{mn} with $h/r = b/r$.

$$F_{mn} = \frac{1}{2\pi} \left[\cos^{-1} \left(\frac{a}{b} \right) - \frac{r}{2l} \left\{ \left(\frac{c^2}{r^4} - 4 \left(\frac{R}{r} \right)^2 \right)^{\frac{1}{2}} \cos^{-1} \left(\frac{ra}{Rb} \right) + \left(\frac{a}{r^2} \right) \sin^{-1} \left(\frac{r}{R} \right) - \left(\frac{\pi}{2} \right) \left(\frac{b}{r^2} \right) \right\} \right] \quad \text{where} \quad (5.8)$$

$$a = h^2 - R^2 + r^2 \quad \text{and}$$

$$b = h^2 + R^2 - r^2 \quad \text{and}$$

$$c = h^2 + R^2 + r^2$$

The constants listed below are a summary of the values already discussed, or were derived from the mechanical design of the thermal system parts, as detailed in Appendix A. Substituting these constants appropriately into Equations 5.5, 5.6, and 5.7 and solving yielded the view factors F_{12} and F_{14} (F_{14} was one of the cylinder-disk view factors found in the process of solving for F_{12}). By inspection of the enclosure

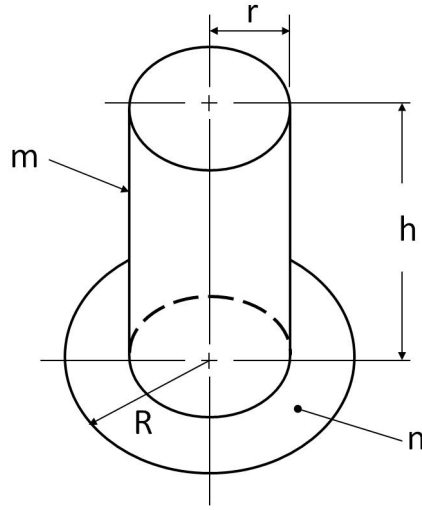


Figure 5.7: Geometry of the Cylinder-Disk View Factor

geometry, F_{13} was calculated using the summation rule, Equation 5.9, where the enclosure was formed by surfaces 1 through 4, shown in Figure 5.3.

$$F_{13} = 1 - F_{12} - F_{14} \quad (5.9)$$

A fifth surface was also part of the enclosure of Equation 5.9, that being an additional cylindrical surface having a radius equal to that of the satellite and a height equal to the difference between the shroud and satellite heights. This surface cannot be seen by Surface 1 and its view factor is zero, so it was not included in Equation 5.9.

The final view factors were found using a method also developed by Leuenberger [22] in which two equal annular areas were considered. In Figure 5.3, surface 4 and an equal annular area on surface 3 were the two surfaces of interest. The satellite was nearly 80 percent the height of the shroud, and so for simplicity the two were both assumed to have the height of the shroud for this calculation. Equation 5.10 states the method developed by Leuenberger, where F_{22full} is the self view factor

of surface 2, and F_{21full} is the view factor between surfaces 2 and 1. The ‘full’ designation is indicating that these view factors were calculated with the full height assumption mentioned above. Fortunately, Leuenberger provided a graphical method for determining these view factors, which were found to be $F_{22full} = 0.23$ and $F_{21full} = 0.55$.

$$F_{34} = 1 - \frac{1}{2} \left(\frac{L}{R^2 - r^2} \right) \left[r - R(F_{22full} + 2F_{21full} - 1) \right] \quad (5.10)$$

View factors F_{23} and F_{24} were equal by examination when the geometry of the annular enclosure used for calculating F_{34} was considered. This assumed again that the satellite and the shroud were the same height. The summation rule was applied, F_{23} was substituted into the resulting equation for F_{24} , and the equation solved for F_{23} . The constants used for all of the view factor calculations, along with the resulting view factors are listed below.

Constants:

$$r_1 = 0.2286m, h_1 = 0.4572m, \epsilon_1 = 0.85$$

$$r_2 = 0.3556m, h_2 = 0.5842m, \epsilon_2 = 0.9$$

$$r_3 = 0.3556m, \epsilon_3 = 0.9$$

$$\epsilon_4 = 0.1$$

$$r_5 = 0.2286m, \epsilon_5 = 0.85$$

View Factors:

$$F_{53} = 0.837$$

$$F_{52} = 0.163$$

$$F_{12} = 0.847$$

$$F_{13} = 0.026$$

$$F_{14} = 0.100$$

$$F_{34} = 0.562$$

$$F_{23} = 0.110$$

$$F_{24} = 0.110$$

With the view factors now known and the constants defined, the set of simultaneous equations generated from Equation 5.4 was solved for the radiosity values, J_i . For example, Surface 1 took the form shown in Equation 5.11 when expanded, and when further re-arranged took the form shown in Equation 5.12.

$$\frac{\sigma T_1^4 - J_1}{\frac{1-\epsilon_1}{\epsilon_1 A_1}} = \frac{J_1 - J_2}{\frac{1}{A_1 F_{12}}} + \frac{J_1 - J_3}{\frac{1}{A_1 F_{13}}} + \frac{J_1 - J_4}{\frac{1}{A_1 F_{14}}} \quad (5.11)$$

$$\frac{\epsilon_1 A_1 \sigma T_1^4}{1 - \epsilon_1} = J_1 \left(A_1 F_{12} + A_1 F_{13} + A_1 F_{14} + \frac{\epsilon_1 A_1}{1 - \epsilon_1} \right) - J_2 A_1 F_{12} - J_3 A_1 F_{13} - J_4 A_1 F_{14} \quad (5.12)$$

The other four equations were rearranged similarly. There were five equations and five unknowns, so the equations had to be solved simultaneously and the simplest technique for doing this was the LU (Lower-Upper) Decomposition matrix method. The five equations were separated into three matrices: Constants, Resistances, and Radiosities.

$$Constants = \begin{pmatrix} \frac{\epsilon_1 A_1 \sigma T_1^4}{1-\epsilon_1} \\ \frac{\epsilon_2 A_2 \sigma T_2^4}{1-\epsilon_2} \\ \frac{\epsilon_3 A_3 \sigma T_3^4}{1-\epsilon_3} \\ \frac{\epsilon_4 A_4 \sigma T_4^4}{1-\epsilon_4} \\ \frac{\epsilon_5 A_5 \sigma T_5^4}{1-\epsilon_5} \end{pmatrix}$$

$$Resistances = \begin{pmatrix} \alpha & -A_1 F_{12} & -A_1 F_{13} & -A_1 F_{14} & 0 \\ -A_1 F_{12} & \beta & -A_2 F_{23} & -A_2 F_{24} & -A_5 F_{52} \\ -A_1 F_{13} & -A_2 F_{23} & \gamma & -A_3 F_{34} & -A_5 F_{53} \\ -A_1 F_{14} & -A_2 F_{24} & -A_3 F_{34} & \delta & 0 \\ 0 & -A_5 F_{52} & -A_5 F_{53} & 0 & \zeta \end{pmatrix}$$

where

$$\begin{aligned} \alpha &= A_1 F_{12} + A_1 F_{13} + A_1 F_{14} + \frac{\epsilon_1 A_1}{1-\epsilon_1} \\ \beta &= A_1 F_{12} + A_2 F_{23} + A_2 F_{24} + A_5 F_{52} + \frac{\epsilon_2 A_2}{1-\epsilon_2} \\ \gamma &= A_1 F_{13} + A_2 F_{23} + A_3 F_{34} + A_5 F_{53} + \frac{\epsilon_3 A_3}{1-\epsilon_3} \\ \delta &= A_1 F_{14} + A_2 F_{24} + A_3 F_{34} + \frac{\epsilon_4 A_4}{1-\epsilon_4} \\ \zeta &= A_5 F_{52} + A_5 F_{53} + \frac{\epsilon_5 A_5}{1-\epsilon_5} \end{aligned}$$

$$Radiosities = \begin{pmatrix} J_1 \\ J_2 \\ J_3 \\ J_4 \\ J_5 \end{pmatrix}$$

Here the five equations were written succinctly as one equation (5.13) and solved for the radiosity values.

$$Radiosities = Constants \times Resistances^{-1} \quad (5.13)$$

$$Radiosities = \begin{pmatrix} 276.87 \\ 879.95 \\ 862.01 \\ 715.04 \\ 277.18 \end{pmatrix} \frac{W}{m^2}$$

Having found the radiosities, the net radiation heat transfer to the shroud body and shroud top was found using Equation 5.2, resulting in $\dot{q}_2 = 388.4$ W (shroud body) and $\dot{q}_3 = 107.0$ W (shroud top). Losses to the base plate and the chamber also had to be found. Using Fourier's Law of Conduction (Equation 5.14), the heat loss through each of the insulators ($N = 8$) was calculated. Delrin, the material used for the insulators, has a thermal conduction coefficient (k) of $0.4 \frac{W}{mK}$ and geometry as shown in Figure A.12. The surface in contact with the shroud (notched step, 3.8 cm (1.5 in) by 1.27 cm (0.5 in)) was assumed to be the cross sectional area, A , through which the heat flows. Under the same temperature conditions as described above, the heat loss through all eight standoffs was $\dot{q}_{insulator} = 9.75$ W. This was a more liberal estimation of the heat lost through these insulators than was found in Chapter 4 because it assumed a smaller temperature difference across the insulators. In either case, the respective requirements of the base plate and thermal shroud were met, as were the design goals.

$$\dot{q}_{insulator} = -NkA \frac{\Delta T}{\Delta x} \quad (5.14)$$

The heat loss to the cylindrical chamber wall was found using Equation 5.15 from Incropera [18]. The inner surface of the chamber and the outer surface of the shroud body were both polished stainless steel which had a surface emissivity of $\epsilon_{stainless} = 0.1$. This provided a heat loss to the chamber of $\dot{q}_{\text{chamber,shroud body}} = 40.45$ W where the chamber temperature was assumed to be 20°C.

$$\dot{q}_{\text{chamber,shroud body}} = \frac{\sigma A_{\text{chamber}}(T_1^4 - T_2^4)}{\frac{1}{\epsilon_{stainless}} + \frac{(1-\epsilon_{stainless})r_2}{\epsilon_{stainless}r_1}} \quad (5.15)$$

Another formulation from Incropera [18] was used to calculate the heat loss from the shroud top to the chamber, $\dot{q}_{\text{chamber,shroud top}}$. To use Equation 5.16, it was assumed that the shroud top and the chamber top form a two surface enclosure. There was, in fact, a 7.62 cm (3 in) radial gap between the circular shroud top and the chamber walls, but this area was coplanar with the shroud top so its influence on the result was negligible. For simplicity, the surface area of the chamber top was assumed to be hemispherical with radius equal to that of the chamber walls (compare to Figure 3.1a). Solving this equation yielded a heat loss of $\dot{q}_{\text{chamber,shroud top}} = 17.45$ W. The total heat loss from the shroud body and top was equal to 57.9 W, which, while exceeding the design goal of 25 W (Requirement 8), did meet the requirement of minimizing the heat loss by virtue of its design.

$$\dot{q}_{\text{chamber,shroud top}} = \frac{\sigma(T_3^4 - T_{\text{chamber}}^4)}{\frac{1-\epsilon_{\text{chamber}}}{\epsilon_{\text{chamber}}A_{\text{chamber}}} + \frac{1}{F_{3,\text{chamber}}A_{\text{chamber}}} + \frac{1-\epsilon_{\text{chamber}}}{\epsilon_{\text{chamber}}A_3}} \quad (5.16)$$

The final quantities needed were the $\dot{Q}_{\text{store}}$ values for each of the two shroud zones, found using Equation 5.17. Stainless steel has a specific heat, C_p , of $470 \frac{J}{kgK}$

at 20°C (the mean temperature of the shroud's range) and the shroud was to be heated at a rate similar to that of the base plate, 1.5 °C/min. Under these conditions, $\dot{Q}_{\text{shroud body store}} = 265.4 \text{ W}$ for the shroud body ($m = 22.6 \text{ kg}$), and $\dot{Q}_{\text{shroud top store}} = 104.5 \text{ W}$ for the shroud top ($m = 8.9 \text{ kg}$).

$$\dot{Q}_{\text{store}} = mC_p \frac{\Delta T}{\Delta t} \quad (5.17)$$

The results thus far were summed according to Equation 5.1, and the required heating power input for the shroud body, Equation 5.18, was found to be 704.0 W. For the shroud top, Equation 5.19, the required heating power input was 228.9 W.

$$\dot{Q}_{\text{shroud body in}} = \dot{q}_2 + \dot{q}_{\text{insulator}} + \dot{q}_{\text{chamber body}} + \dot{Q}_{\text{shroud body store}} = 704.0 \text{ W} \quad (5.18)$$

$$\dot{Q}_{\text{shroud top in}} = \dot{q}_3 + \dot{q}_{\text{chamber top}} + \dot{Q}_{\text{shroud top store}} = 228.9 \text{ W} \quad (5.19)$$

These values dictate the heat power required to raise the temperature of each thermal shroud zone to 80°C and keep it there while a nanosatellite size test object is under test at -40°C. These values were the worst case values for several reasons. First, if a smaller test object were under test, it would have less surface area thereby reducing the heat transferred to it from either shroud zone. Second, the surface emissivity value for the test object was fairly high ($\epsilon_1 = 0.85$). In very rare instances this value could go up as high as 0.9, but lower values would be more common, thereby reducing the required heat transfer from either of the shroud zones. Finally, actual operating temperatures would likely not be as extreme as those assumed in the above analysis, which was an extreme operating scenario of the system.

The above analysis was performed for the cold balance case of thermal balance testing. The hot balance test operated with the temperature extremes reversed and the rate negative (for cooling the shroud), but with the dimensions and all other constants the same. Performing an identical analysis for the hot balance case yielded $\dot{Q}_{\text{shroud body in}} = -685.5 \text{ W}$ for the shroud body, and $\dot{Q}_{\text{shroud top in}} = -220.9 \text{ W}$ for the shroud top (the negative sign refers to the direction of heat flow, in this case out of the shroud). The difference between the results of the hot and cold balance cases was due to the T^4 difference in the numerators of Equations 5.15 and 5.16.

Liquid Nitrogen Tubing Thermal Design

As previously mentioned, the behavior of liquid Nitrogen in the tubing was very unpredictable and so a thorough analysis of the shroud using liquid Nitrogen would have provided, at best, mediocre results. It was still desired to have some level of confidence in the mechanical design as it affected the thermal performance of the shroud. Like the base plate, the shroud top consisted of a flat plate attached to a serpentine tube, and so a similar qualitative analysis based on the base plate results was appropriate. Given that the shroud top tubing and the base plate tubing had similar spacing, that the shroud top had a thermal mass of almost half that of the base plate, that the shroud top plate was 1.9 mm (0.075 in) thick – an order of magnitude thinner than the base plate – and assuming that the tube was a uniform temperature throughout, it was reasonable to conclude that the temperature distribution in the shroud top would be no worse than the results of the base plate analysis.

As for the shroud body, the parallel tubes were spaced 7.62 cm (3 in) apart – the same separation as for the base plate tubing – and the tubes also surrounded the shroud body from bottom to top. The shroud body was made from 1.52 mm (0.060 in) thick stainless steel, also much thinner than the base plate. Therefore, it followed

that the temperature distribution of the shroud body and top would be acceptable for the needs of the SSEL thermal vacuum system.

Heater Design

Based on the heat load analysis, heaters were sized for both zones of the thermal shroud. Unlike the base plate, the shroud body was not a flat surface and so a flexible heater was needed. A heat tape supplier [17] was contacted to design custom tape heaters for both the shroud body and top. For the shroud body, a single tape heater operating at 60 Volts DC and having a length of 31.7 m (104 ft) was designed. This heater wrapped around the shroud body between the parallel tubes. For the shroud top a set of heaters was designed, each a unique length to fit between the rows of tubing. The heaters were connected together in series electrically and were designed to provide uniform heating across the top plate. The heaters for both the shroud top and body were designed for use in vacuum and to meet at least the minimum temperature range of the shroud. The acrylic adhesive used to attach the tape heaters to the shroud had a minimum operational limit of -100°C which did not meet the goal of Requirement 1, but did meet the requirement itself.

Support Equipment

Equipment outside of the chamber was necessary to heat and cool the thermal shroud. The heaters were driven by two DC power supplies located in the PDB and controlled by the LabVIEW Software. Unlike the base plate power supply, these power supplies had only ON and OFF modes, but no analog control of the voltage output. The output power was therefore controlled by duty cycling the output so that the desired heat power was applied to the attached heater. In this way, the same

control of heat power was achieved with the thermal shroud body and top as with the base plate power supply.

Cooling the shroud was done by forcing liquid Nitrogen through the shroud tubing. For the same reasons as discussed in Chapter 4 a pressurized tank was used to move the liquid Nitrogen through the tubing. The same tank was used for both the base plate and the shroud body and top, so adjustments to the tank's regulating valves affected all three zones. Both the shroud top and body had liquid Nitrogen ON/OFF valves that were controlled by the temperature control software.

Temperature Control Software

The same control software used to control the base plate was used to control both thermal shroud zones, with the primary difference being the method for controlling the heater power supplies as mentioned above. The user supplied the desired temperature setpoint, the desired rate of temperature change, and a hysteresis value, and the heater duty cycle and the code responded accordingly. If the setpoint was above the current temperature, the heaters were turned on. The heater power was determined by the rate of temperature change value that the user supplied. Once the base plate reached the desired temperature, the code turned off the heater and waited for the temperature to drop below the setpoint less the hysteresis value. When it did, it turned the heaters back on until the temperature reached the setpoint. In this way the software could maintain the desired setpoint temperature.

When the setpoint temperature was below the current temperature the heaters were turned off and the liquid Nitrogen valve was opened. Once the base plate reached the desired temperature, the valve was closed again, and cycled open and closed based on the hysteresis, just like during heating. The rate of temperature

change was dependent on the flow rate of cold gas or liquid Nitrogen, both of which were adjusted by hand valves on the liquid Nitrogen tank.

Thermal Shroud Manufacturing and Assembly

As with the base plate, parts were manufactured locally if possible. The shroud body and top as well as the top insulating spacers were machined locally [19]. The shroud body and top tubing were fabricated out-of-state by a custom tube bending shop [20]. These parts were then shipped to a vacuum furnace brazing facility [21] where they were assembled according to the drawings shown in Figures A.16, A.17, A.26, A.27, and A.26. Although the vacuum furnace brazing process provided better results than welding or standard brazing, it was not perfect. Those imperfections, coupled with small manufacturing imperfections in the tube bending process, and in the process for rolling the shroud body to shape and welding the seam, led to an alignment issued between the shroud top insulators and the shroud body as shown Figures 5.8a and 5.8b.

The non-circular shape of the shroud body, seen in Figure 5.8c and the warping of the shroud top, seen in Figure 5.8d, led to the design of a new set of shroud top insulators that would compensate for these manufacturing defects and still meet the original design goal. The new set of shroud top insulators, shown in Figures A.33 and A.34, allowed the shroud body and top to be attached to one another and the four lift points still used for moving the shroud in its entirety. This modified design did not incorporate all eight insulators as the original design did, but the original design intent was preserved.

Prior to installing the tape heaters, the shroud body and top assemblies were painted with the Aeroglaze Z306 paint [24]. The paint was given a 40 days cure



(a) Shroud Alignment Horizontal Mismatch



(b) Shroud Alignment Vertical Mismatch



(c) Shroud Body Seam



(d) Warped Shroud Top

Figure 5.8: Shroud Manufacturing Process

time to ensure a full cure. The assemblies and heaters were thoroughly cleaned with alcohol to remove oils, and the heaters were then attached to the shroud body and top. The heater resistance was checked against the design, the paint was inspected for cracking and durability, and the liquid Nitrogen hoses attached appropriately prior to installation in the chamber.

Closing Remarks

Like the base plate, design of the thermal shroud centered around its functional requirements as well as the physical limitations of the vacuum chamber. Analysis

predicted the necessary heat power needs given the shroud requirements and an appropriate hardware solution was designed.

SYSTEM TESTING AND VALIDATION

System testing ensured that the thermal vacuum system would perform as designed and would also provide a baseline from which the design could be compared to the actual hardware performance. Both the vacuum and thermal systems had to meet their respective functional requirements and each test performed served to verify one or more of those requirements. Because the vacuum system was always used whether the thermal system was used or not, its functionality could be verified at the same time as the thermal system. While a failure of the vacuum system during a thermal system test would not damage the thermal system, the results from the thermal system test would be meaningless. Therefore, careful and complete testing of the vacuum system was performed before testing the thermal system.

Vacuum System Testing

Basic functional testing of the vacuum system hardware was performed prior to or during installation, ensuring problem free usage in the final integrated system. Components like the Turbo Pump were tested in conjunction with the Scroll Pump and the necessary valves by necessity and while using their respective hardware controls. Testing the hardware first eliminated many of the variables during software testing.

The LabVIEW control code for the vacuum system was composed of instrument driver Application Program Interface (API) Virtual Instruments (VIs), wrapper VIs that called the instrument driver VIs in a specific sequence, and higher level application VIs which called the wrapper VIs, handled the graphical user interface (GUI), and logged all system data. This architecture necessitated that the low level driver VIs were first tested individually, and then together as part of the wrapper VIs. Once

the wrapper VIs had been tested they were then integrated into the top level application VIs and then tested together. By testing at the lowest level first and gradually adding complexity, software bugs were found early and could be fixed before they caused large problems.

Built-in hardware safety was also tested. System valves were held open using compressed air which meant that the valves automatically closed with a loss of air pressure, resulting in a safe configuration even if the pumps did not turn off. The valves and pumps also required electrical power to operate and so a loss of power closed the valves and shut off the pumps. The Scroll Pump required 208 Volts AC which meant that a failure of the 120 Volt AC would not necessarily shut down the Scroll Pump. The system power distribution box (PDB), which operated on 120 Volts AC, compensated for this by switching the Scroll Pump's interlock connection, thus insuring that a loss of the 120 Volts AC would also shut down the Scroll Pump.

Integration of an Uninterrupted Power Supply (UPS) was considered but not implemented. Without the UPS in the system, a loss of power closed all of the valves and turned off both vacuum pumps, putting the hardware into a safe configuration. It also shut down the computer without going through the shutdown sequence in the LabVIEW software. Therefore, a UPS was only needed if a proper shutdown of the computer was desired. No damage would occur to the vacuum system with a power loss.

Adverse user commands were also tested to be sure that no damage would occur to the vacuum system hardware. The software interface had built-in safeties to prevent the valves, pumps and pressure gauges from being operated when damage could occur. For example, the Gate Valve could only be opened if the pressure in the chamber and the pressure in the Turbo Pump were between 1 mTorr and 50 mTorr, and if the Turbo Pump was turned off. This prevented damage to the Turbo Pump from a

sudden pressure spike at its inlet. Whenever a user performed an allowed action, the software checked the system state and determined what the user could and could not to do next, and no further action was allowed until this determination had been made. All combinations of possible allowed states were tested to verify that the built-in software safeties would prevent damage to the system.

Without a load on the vacuum pumps, the system was able to pump from atmospheric pressure to 50 mTorr within one hour, and from 50 mTorr to below $5\mu\text{Torr}$ in under 45 minutes. In addition, several successful tests lasting over 24 hours took place, indicating that the vacuum system software and the configuration of the hardware (fail safes) were acceptable for safe and long term usage, and that the vacuum system hardware and software were ready to be used in conjunction with the thermal system.

Thermal System Testing

In many ways the thermal system testing was more critical to the safety of the system and to the test object than the vacuum system testing. More severe damage could occur to the system and to any test objects if the thermal system did not function properly and within its design limitations. The thermal system requirements define how the system had to function, and the limitations were derived from the design itself. A series of tests was constructed to verify that the design met the requirements and to test the system limitations. Data was compared to the requirements, models, and calculations used in the system design. These tests sought to verify full functionality of the base plate and thermal shroud, their respective control systems, as well as validate the overall system design.

Validation of the base plate functionality was accomplished by cycling the base plate and the test satellite without the thermal shroud. The base plate design was

validated by cycling the base plate and thermal shroud at the same temperatures. In the satellite model the sides and top were assumed to be insulated, and this condition was approximated in testing by removing convection as a mode of heat transfer (vacuum) and by minimizing the temperature difference between the test satellite and the thermal shroud (minimal temperature difference means minimal radiation heat transfer). In this way the shroud functionality was also validated. As a point of comparison, the data needed to validate the base plate design was gathered and compared with the same data when only the base plate was used. Although not necessary to validate the system design, doing so provided a quantitative demonstration of how the system behaved with and without the thermal shroud.

To validate the shroud design, the base plate and satellite were cooled to -40°C and held there, and then the shroud was heated to 80°C . This maximized the temperature difference between the satellite and shroud and simulated the conditions that were used in the shroud design. Details for each of these tests are explained in greater detail along with each test. All tests required use of the base plate, so that testing was performed first.

Test Setup

The thermal models used in the design could only be validated if the system were operated under the same conditions. Unfortunately, this was not possible for a number of reasons. Most importantly was that an exact match to the 40.5 kg satellite model was not available. However, a similar 26.65 kg satellite structure was available as a thermal load. This structure consisted of all the same structural elements of the satellite model: a bottom plate, top plate, wings, center boxes, and outer skins. The test satellite differed from the satellite model in that it had eight sides instead of four, and its mass was approximately 14 kg less than that of the satellite model. Given that

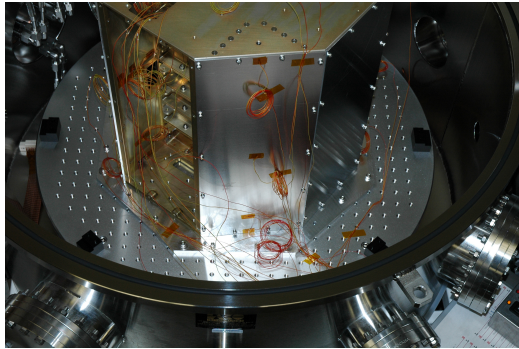
building a test satellite identical to the satellite model would have cost approximately \$10k to \$15k and that it would have had no purpose beyond this thermal testing, the decision was made to use the available test satellite structure and to analyze the test results in light of the mass and geometric differences.

System temperatures were measured using a set of calibrated thermocouples. Two distinct groups of thermocouples were configured for the system. The first was a set that was specifically designated for measuring system temperatures. These thermocouples were mounted to the base plate and thermal shroud at critical locations. On the base plate, two thermocouples were mounted to the underside, one at the inlet end of the serpentine tube and the other at the outlet. Two thermocouples were also mounted to the top of the base plate. On the thermal shroud body, one thermocouple was mounted near the tube inlet, a second was mounted half way between the top and bottom, and a third was mounted near the tube outlet. The thermal shroud top had two thermocouples, one near the tube inlet and the other near the outlet.

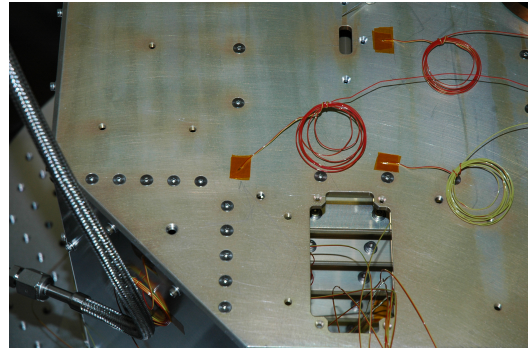
The second set of thermocouples was designated for use with any test object. All were terminated at a common connector so that the thermocouples could be installed to the test object prior to installing it in the chamber. There were twenty five thermocouples available, and they were attached to the test satellite such that the measurements taken would provide a comprehensive temperature distribution, and would also provide the ability to measure the heat flow through the bottom of the test satellite. On four of the eight sides, three thermocouples were attached: one at the top, one half way between the top and bottom and the third at the base (Figure 6.1a). On the front side (closest to the front of the chamber), a fourth thermocouple was mounted 3.81 cm (1.5 in) above the bottom thermocouple and was used to measure the heat flow through the bottom of the satellite. Figure 6.1b shows the locations of the three thermocouples on the top of the test satellite. Inside the satellite there

were a number of thermocouples on one of the wings, and on bottom, middle, and top component boxes (Figures 6.1c, 6.1d, and 6.1e, respectively). Each thermocouple was bare at the junction, so a piece of Kapton tape electrically insulated the thermocouple wires from the surface of the test satellite. A piece of reflective Aluminum tape then covered each thermocouple to reduce the influence of radiation on the measured surface temperature. Figure 6.1f shows a closeup of one of the front thermocouples, minus the reflective tape.

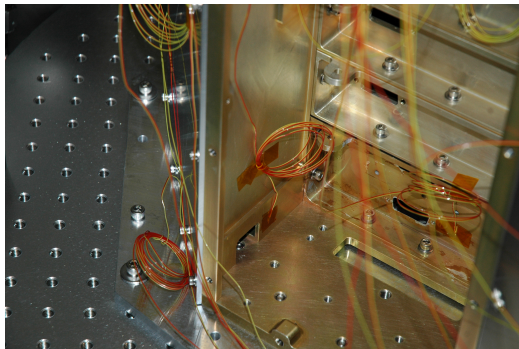
Prior to installing these thermocouples to the test satellite, the test satellite was thoroughly cleaned. It had been previously used as a demonstration structure and had been handled many times by bare hands. It had not been thoroughly cleaned after it had been machined so the surfaces and many threaded holes had remnants of machine oil. The test satellite was disassembled, cleaned and re-assembled. The first step in cleaning the structural pieces involved soaking them in a water and Liquinox bath and using an ultrasonic cleaner to dislodge dirt from inside the threaded holes (see Figure 6.2a). The parts were then thoroughly cleaned in an alcohol bath to remove any remaining oils and contaminants. Starting from the bottom plate (Figure 6.2b) the test satellite was re-assembled, beginning with the component boxes and an interface plate (bare Aluminum) seen in Figure 6.2c. The wings and top plate were then installed (Figure 6.2d), followed by the sides, shown in Figure 6.2e. All of the fasteners were tightened using a torque wrench to a torque appropriate for the size fastener so that the contact resistances between various components could be approximated if desired.



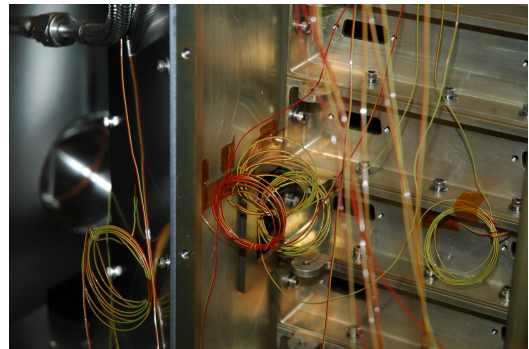
(a) Outside Body Thermocouples



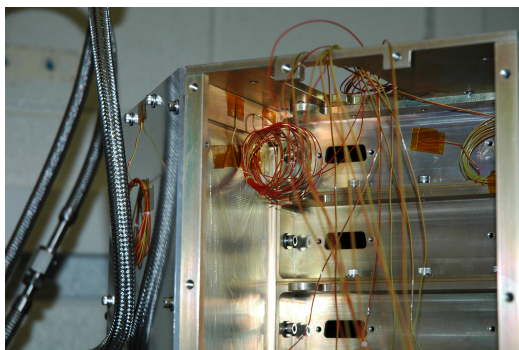
(b) Outside Top Thermocouples



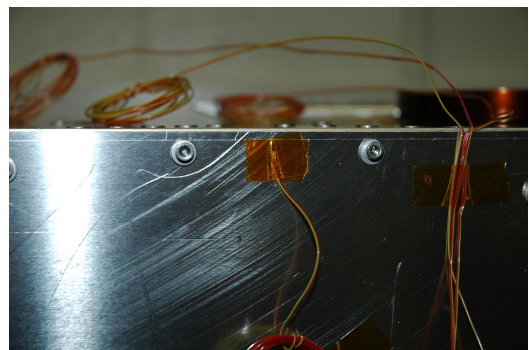
(c) Inside Bottom Thermocouples



(d) Inside Middle Thermocouples

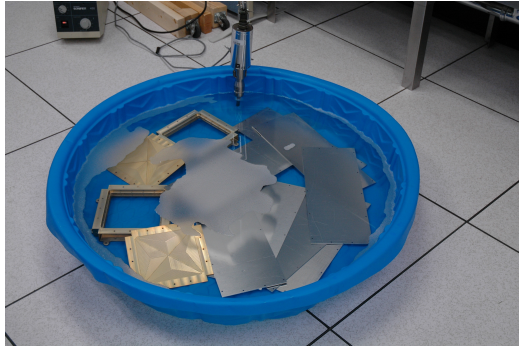


(e) Inside Top Thermocouples

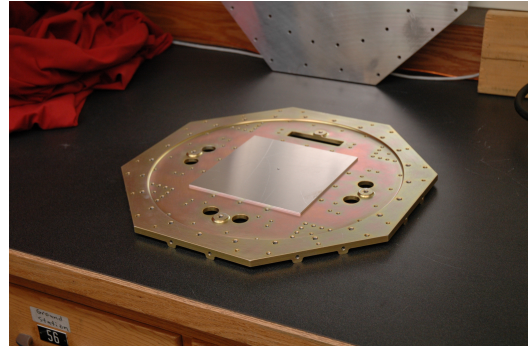


(f) Thermocouple Closeup

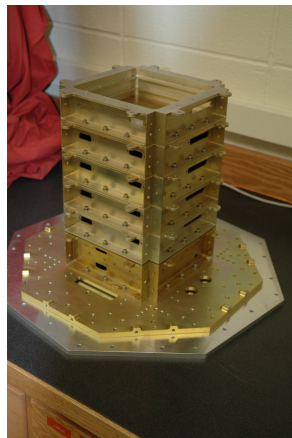
Figure 6.1: Thermocouples on the Test Satellite



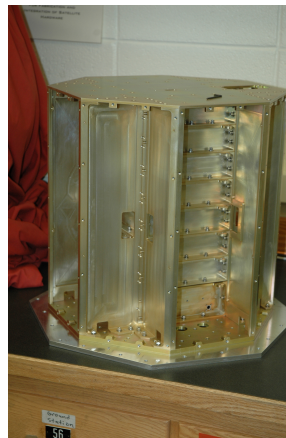
(a) Cleaning the Test Satellite in an Ultrasonic Bath



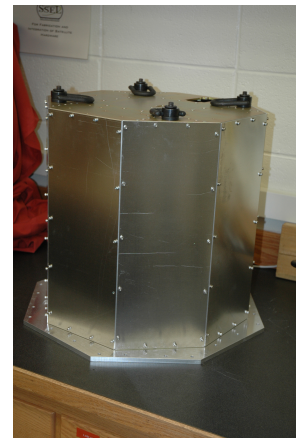
(b) Beginning the Assembly with the Bottom Plate



(c) After Installing the Component Boxes and Interface Plate



(d) After Installing the Wings and Top Plate



(e) After Installing the Sides

Figure 6.2: Cleaning and Assembly Process of the Test Satellite

Base Plate Testing

Preliminary Base Plate Testing

In Chapter 4, the conditions of the base plate thermal analysis were described in detail, and these conditions were duplicated as closely as possible. However, the inverse temperature profile was used in place of the profile used in the analysis. Until this time, no hardware other than the thermal system itself had been installed in the vacuum chamber. Prior to this test the base plate itself had been ‘vacuum baked’ for several hours at 80°C. Vacuum baking is a common and recommended practice in vacuum science because it speeds the outgassing process of newly introduced materials to vacuum, and in so doing reduces the time it takes to achieve the desired system pressures during subsequent pumping cycles. The test satellite, too, had to be vacuum baked prior to its first use (along with thorough cleaning before installation) to maintain system cleanliness as well as to reduce pumping time of all subsequent tests.

Heating first, instead of cooling as in the model analysis, was acceptable given the discussion in Chapter 4. The heat load at the point in time when temperature transition ended and soak began was the same regardless of whether heating or cooling was performed first - only the direction of heat flow changed. Therefore, starting at 22°C (room temperature) the base plate and test satellite were first heated to 80°C at a rate of 1.5°C/min. Once the temperature of the test satellite reached 80°C $\pm$ 2°C, it was held there for several hours for system bakeout and then cooled at 1.5°C/min until the bottom surface of the test satellite reached -40°C. The bottom surface of the test satellite remained at -40°C until the entire satellite reached -40°C $\pm$ 2°C, after which it was brought back to room temperature. This was the ideal temperature profile at the base plate/ satellite interface.

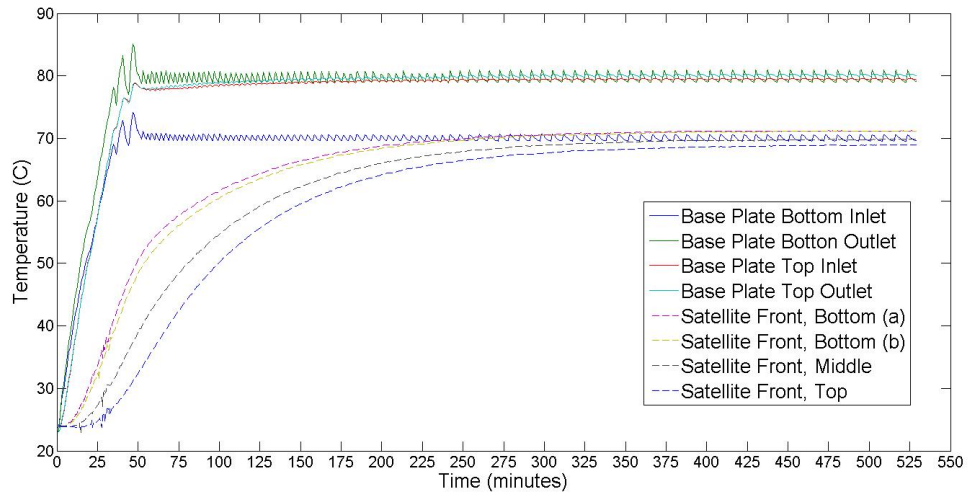
This temperature profile was attempted so that a measurement could be taken of the heat transfer through the base of the test satellite. In the analysis, the maximum heat transfer was found at the end of the transition from -40°C to 80°C. In the test, the heat transfer was found at this same point in time by measuring the temperatures at two locations and using Fourier's Law (Equation 6.1) to calculate the heat transfer between the two points at the moment when the interface between the base plate and test satellite reached 80°C. The two thermocouples were located one above the other, labeled 'Satellite Front, Bottom (a)' and 'Satellite Front, Bottom (b)'. Figure 6.3a shows the measured temperature profile during the heating cycle of the base plate. Measured temperatures from the base plate, in four locations, and on the test satellite are also shown for reference. During the course of the heating cycle the 10°C temperature difference between the base plate thermocouples was attributed to contact resistance between the mechanical parts. Upon completing the heating cycle the base plate and test satellite were allowed to cool to room temperature and the following day the first cooling cycle was attempted, as shown in Figure 6.3b.

$$\dot{q} = -kA \frac{\Delta T}{\Delta x} \quad (6.1)$$

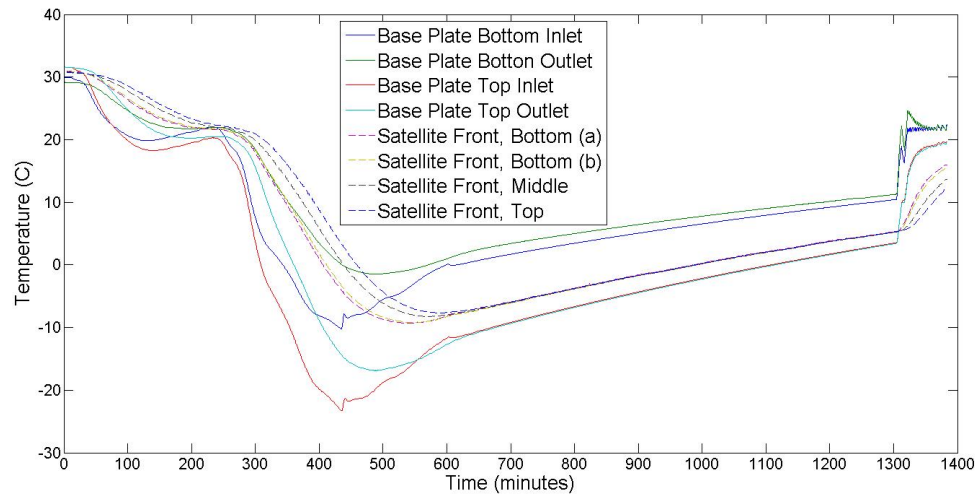
The ideal temperature profile as described above was not followed because the system behaved differently than expected, and so deviations from the ideal temperature profile were made in order to better understand the system. The oscillations in the base plate temperatures, shown in Figure 6.3a, were due to the thermal control code turning the base plate heaters on and off. The difference between the temperatures of the base plate thermocouples was found to be the result of configuring the data acquisition device for the wrong thermocouple type. Except for the 'Base Plate Bot-

tom Inlet' thermocouple, all other base plate thermocouples had been configured in software for type K when in fact they were type E, and so the temperature of the base plate and the satellite reached only 70°C during heating. The thermocouple configuration error was unknown until after the cooling cycle was finished. Fortunately, the correlation between the measured temperature and the thermal Electromotive Force (EMF) generated by a thermocouple is well understood. The temperatures recorded in the data (type E) were converted back to the measured EMF and that EMF was then converted to the temperature for the correct thermocouple type (type K). The corrected results for both heating and cooling are shown in Figures 6.4a and 6.4b, respectively.

The conversion was not perfect as evidenced by the consistent temperature difference of the base plate thermocouples for the entire duration of the testing, and by the temperatures of the satellite being higher than the temperatures of the base plate. The data does demonstrate the stability of the temperature control software that was operating the base plate. The temperature increased steadily at 1.5°C/min and remained within approximately 2°C of the setpoint temperature during soak. Contact resistance between components slowed the flow of heat through the base plate and into the satellite which caused the temperature gradient in the satellite, and was not unexpected. The cooling cycle attempted to cool the base plate and satellite to -40°C, but, as Figures 6.3b and 6.4b show, -40°C was never reached. In both the original and corrected temperature plots the temperature uniformity in the base plate was very good during the heating cycle. Temperatures across the base plate were within 5°C after correction, even with the conversion error. During cooling, the temperature difference across the base plate from inlet to outlet was colder at the inlet than at the outlet as expected.

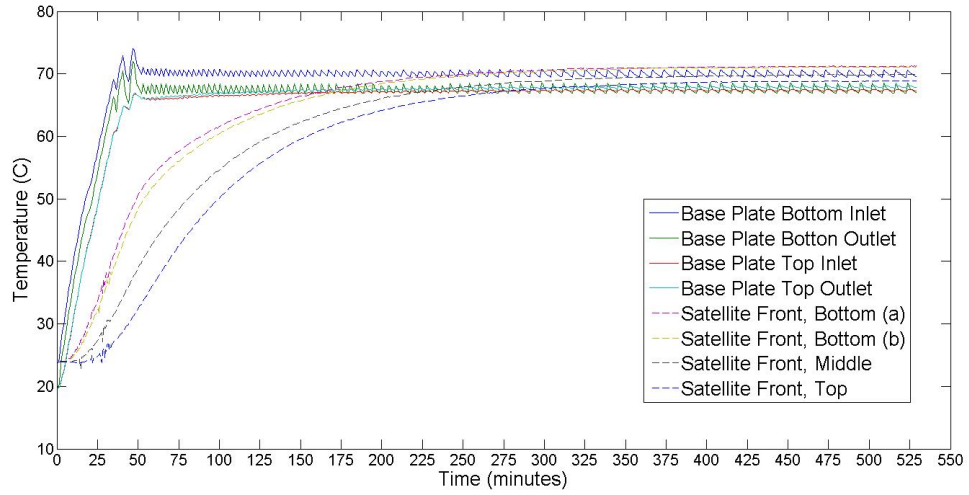


(a) Base Plate and Satellite Temperatures During Heating, As Measured

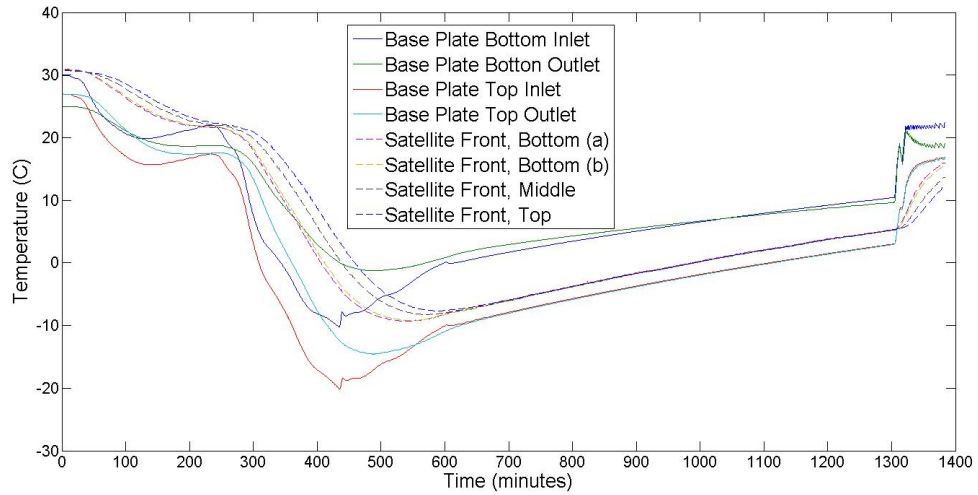


(b) Base Plate and Satellite Temperatures During Cooling, As Measured

Figure 6.3: Base Plate and Satellite Temperatures, As Measured



(a) Base Plate and Satellite Temperatures During Heating, Corrected



(b) Base Plate and Satellite Temperatures During Cooling, Corrected

Figure 6.4: Base Plate and Satellite Temperatures, Corrected

Having discovered the thermocouple setting issue and repairing the problem, the full thermal cycle was repeated. During the course of the heating cycle, a system malfunction occurred, which caused damage to the plastic parts of the base plate. The system was left unattended overnight and at 3:30 AM the computer that was operating the system re-booted because of a Microsoft Windows automatic update. At the time the computer shut down the base plate heaters were on, and once the computer re-booted, the control software did not re-acquire control of the heaters; it was not designed for this type of fault. As a result, the heaters continued heating the base plate beyond the 100°C design limit. When the computer shut down, it also halted all data logging activity, so no record of the actual temperature was created.

At the time the damage was discovered, the temperature was found to be in excess of 200°C and possibly in excess of 270°C. The plastic Delrin insulators that maintained the base plate off of the chamber bottom melted and the weight of the satellite and base plate compressed the insulators. This in turn put tension on the electrical wires connecting the heaters to the chamber feedthrough which caused the feedthrough leads to come in contact with the chamber and short the heater power supply. Fortunately, the heater power supply was current limited and it shut itself off at that time. Had it not done so, the power supply could have been damaged, temperature could have gone much higher, and the damage could have been much worse. Another fortunate occurrence was that when the computer shut down, it also shut down the vacuum system safely so no damage was done to the vacuum system components.

Although this test did not verify correct functionality of the base plate, it did bring to light several defects in the design. First, the thermal system needed hardware temperature limits that would operate independently from the computer. Second, the thermal vacuum system should not have been left unattended. The damage could

have been prevented had the issue been caught within 30 minutes of the re-boot. Third, automatic updates on the computer were turned completely off. Fourth, the system design prevented cooling the base plate faster than approximately $0.25^{\circ}\text{C}/\text{min}$ which did not meet the minimum rate requirement, and was why the cooling test was halted at around -20°C . Fifth, cooling the system at a constant and controllable rate was difficult to achieve and so a better control method during cooling was needed.

Base Plate Repairs

The base plate was removed from the chamber and inspected to gauge the extent of the damage. In short, the Delrin insulators on the bottom and the top of the base plate had melted and conformed to the holes and tubing on the top and bottom of the plate, shown in Figure 6.5a. Six of the sixty-five resistive heaters on the bottom of the base plate had melted and fallen apart and the rest showed signs of heat damage (Figure 6.5b). Pieces of the Delrin and the heaters were found on the bottom of the chamber along with an oil-like substance from the melted plastics; both of these contaminants made it imperative the the chamber be cleaned thoroughly before operating the vacuum system again.

After removing the base plate from the chamber, the Delrin insulators and the heaters were removed from the base plate. The chamber was thoroughly cleaned in preparation for re-installing the base plate once it had been repaired. The base plate was also thoroughly cleaned and a set of resistive tape heaters were installed, having an equivalent resistance as the original heaters. Custom tape heaters were chosen over the previous power resistor design because they were less expensive than an identical replacement of the power resistors, required less fabrication time, and were easier to install. The need to replace the Delrin insulators provided the opportunity to revisit which material to use and to improve the system. Delrin and Ultem were

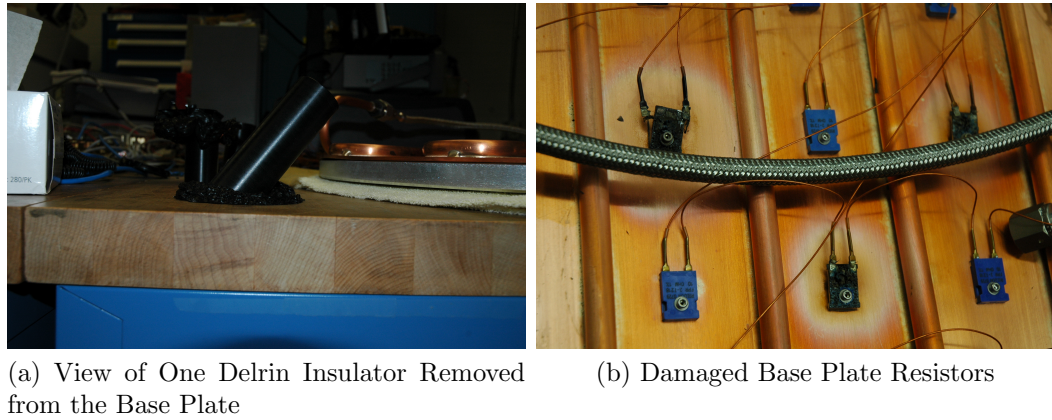


Figure 6.5: Damage to the Base Plate from Preliminary Testing

the two choices, where Delrin was less expensive but had a higher thermal conduction coefficient. Ultem had a lower thermal conduction coefficient and a wider operating temperature range, and so it was chosen as the new material despite the added expense. Figures 6.6a and 6.6b show the new heaters and insulators installed on the bottom and top of the base plate, respectively.

Besides implementing a policy where the system was never left unattended while in operation, hardware temperature limits were also added to the system to protect against unintended user error. Each thermal zone had its own temperature limit controller which measured the temperature at a critical location and would physically disconnect power to the corresponding heater or the liquid Nitrogen valve if the temperature exceeded pre-set limits. As a result, an additional thermocouple was added to each of the three zones. On the base plate and thermal shroud body, a fourth thermocouple was added to each near the liquid Nitrogen inlet, between the inlet and center thermocouples. On the shroud top, a third thermocouple was added near the liquid Nitrogen inlet, closer to the edge than the center. In these locations, the temperature controllers could prevent each zone from exceeding its cold and hot

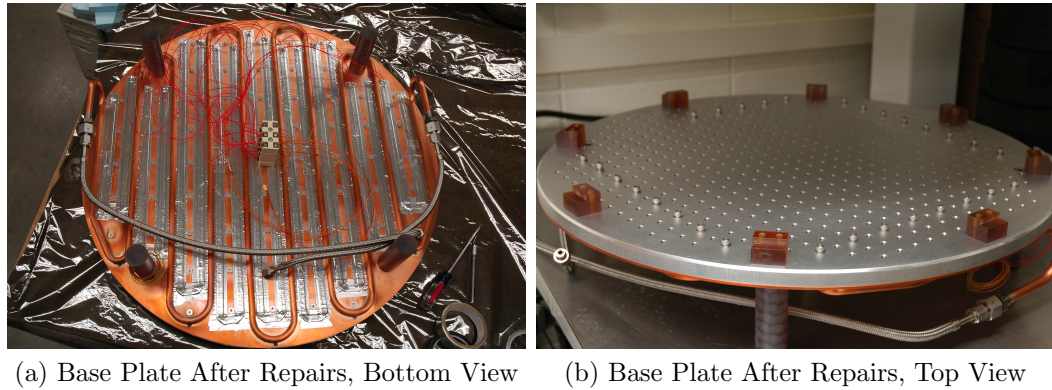


Figure 6.6: Repaired Base Plate

temperature limits. The temperature limit controllers operated independently of the computer which ran the heaters and liquid Nitrogen valves and controlled the system temperatures. The controllers were hard wired into the existing thermal system such that the heaters and liquid Nitrogen valves could be controlled by the computer if the controllers were not turned on. This configuration ensured that a power outage, computer re-boot, or operator error would not cause future damage to the thermal system.

Another necessary change involved the regulator in the liquid Nitrogen line between the liquid Nitrogen tank and the three valves. The regulator was installed to prevent the pressure in the line from exceeding 137.9 kPa (20 psi). However, during the first cooling test it was not possible to drive the cooling rate any faster than approximately 0.25°C/min even with the tank pressure at 482.6 kPa (70 psi) (the design pressure). The pressure regulator was restricting the flow in the line and not ever actuating to prevent the pressure at it's outlet from exceeding 137.9 kPa (20 psi). For these reasons the pressure regulator was removed from the line without any detriment to the system safety.

First Base Plate Functional Test

Having made the necessary repairs to the system, a full temperature cycle was again attempted. The test satellite was again installed to the base plate and the system was operated without the thermal shroud. Beginning at room temperature, the system was heated to 80°C and held there long enough for the satellite to come to near uniform temperature ($\pm 2^\circ\text{C}$). The heaters were then turned off and the system cooled to -40°C and held there long enough for the satellite to come to near uniform temperature. The system was then returned to room temperature. During the transitions a rate of temperature change of 1.5°C/min was attempted. Figure 6.7 shows the ideal temperature profile as described above, the resulting temperature distribution measured by the base plate (BP) thermocouples, and the satellite 'Front' thermocouples.

This test did follow the ideal temperature profile (described above) fairly closely. The base plate was heated to 80°C and was maintained there until the satellite temperature was within 3°C of the setpoint temperature. The oscillations seen in the 'BP Bottom Inlet Side (#4)' thermocouple during soak were the result of the heater being turned on and off by the temperature control software. As cooling proceeded, the rate of temperature change remained difficult to control as evidenced by the large temperature fluctuations. Measures were taken to more closely control the temperature change during cold transition, such as adjusting the code that measured the temperature and controlled the liquid Nitrogen valves, as well as making smaller adjustments to the regulating valve on the liquid Nitrogen tank (experience gained from the first test). Even in doing these things, the large temperature swings during cold transition and soak were still present. Removing the regulating valve did allow the system to meet the 1.5°C/min goal during the cold transition.

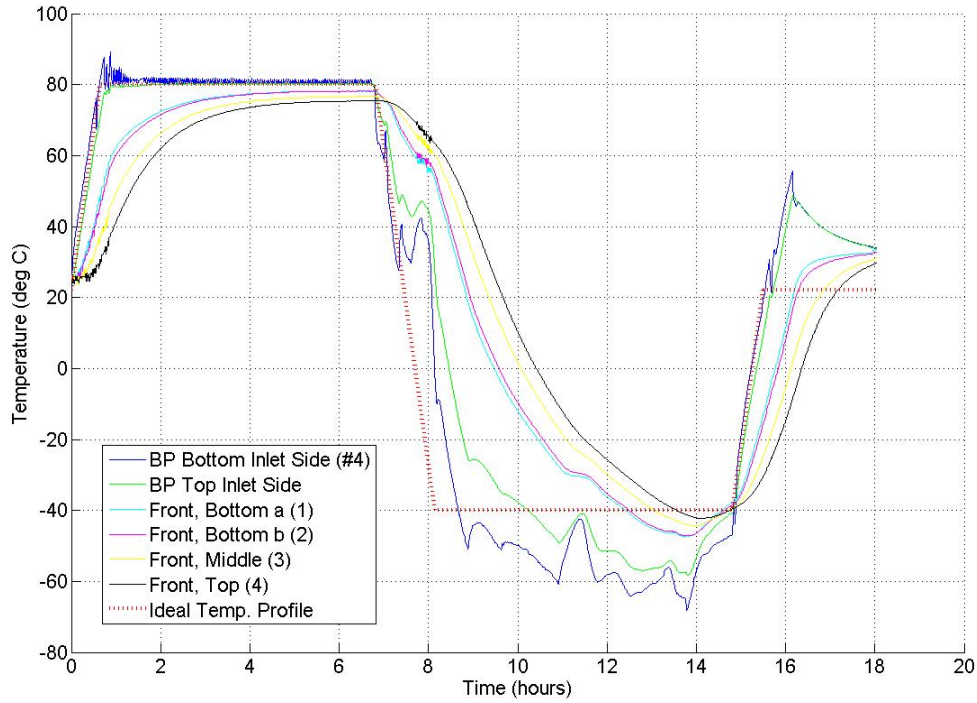


Figure 6.7: Temperature Profiles of Selected Thermocouples from the First Base Plate Functional Test

The four thermocouples on the test satellite were only a sample of the 25 total thermocouples. The front four thermocouples were chosen to show the temperature profile from the bottom to top of the test satellite. Data from other thermocouples showed nearly isothermal planes from bottom to top, to within 3°C. Here, as in the plots from the first set of tests, it was apparent that the temperature was much more easily controlled during heating than during cooling. Because the analysis assumed a much more linear rate of temperature change between temperature extremes, a more linear rate was desired in the test so that a more precise value could be measured. As a result of this test, it was clear that to continue trying to acquire the heat power at the point between cold transition and cold soak was not a valuable use of time or resources. Even small adjustments to the liquid Nitrogen flow had large effects on

the temperature profile of the base plate. Therefore, the test was repeated, but the heat transfer was found at the point between hot transition and hot soak.

Second Base Plate Functional Test

The second base plate functional test sought to acquire the heat power through the base of the test satellite at the end of a transition from -40°C to 80°C . The chamber was first baked to 80°C and held there for several hours for the purpose of vacuum bakeout. An abbreviated bakeout was performed for this test, lasting only two hours and taking the satellite temperature to a minimum of 70°C . The second base plate test had performed a full bakeout lasting six hours and the chamber had been opened only briefly between the previous and current tests, so an additional full bakeout was not necessary. At the conclusion of the bakeout, the heaters were turned off and the system was allowed to cool naturally overnight. The next morning, at approximately hour fourteen, a liquid Nitrogen tank was delivered and the system was forcefully cooled at approximately $1.5^{\circ}\text{C}/\text{min}$ until -40°C was reached. For the next seven hours and 30 minutes the base plate was maintained at or just below -40°C (soaking) until the entire satellite reached the soak temperature. After the soak a hot transition began, taking the base plate from -40°C to 80°C at a rate of $1.5^{\circ}\text{C}/\text{min}$. Upon reaching 80°C the system was shut down and allowed to cool because the data point occurred at the instant in time that the system changed from hot transition to hot soak. The resulting temperature profile is shown in Figure 6.8.

Several important observations were made regarding the system dynamics. Again, maintaining a steady cooling rate using liquid Nitrogen was difficult. The large fluctuations in the base plate temperature between hours 15 and 17 were the result of trying to fine tune the liquid Nitrogen flow rate. The small fluctuations between hours 17 to 19 were due to the base plate liquid Nitrogen valve opening and

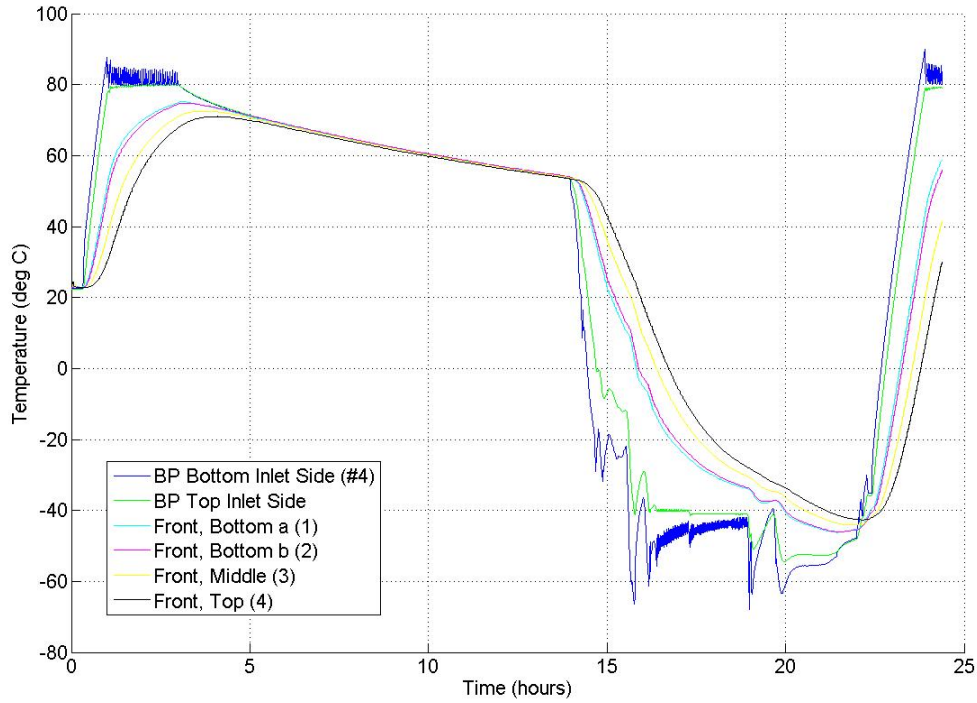


Figure 6.8: Temperature Profiles of Selected Thermocouples from the Second Base Plate Functional Test

closing. The temperature control software was operating correctly and attempting to maintain the setpoint temperature. During the hot soak period the system maintained the temperature within 2 degrees of the setpoint (80°C). During the cold soak, this was more difficult to achieve, but it maintained the temperature steadily on the top surface of the base plate, except for the large temperature swing just before the temperature was overdriven to -55°C. Toward the end of the soak period the rate of temperature change in the test satellite slowed dramatically because the temperature difference between it and the base plate was getting smaller. To compensate for this and to speed up the process, the base plate was overdriven to approximately -55°C, beginning just before hour 20. Although the base plate was designed to cool a test

object no colder than -40°C , the base plate, the heaters, and the tubing are all capable of getting as cold as -100°C for this very purpose.

On the hot side, the base plate was also designed to be overdriven, up to 200°C . For safety and to prevent damage to a test object, the base plate temperature limit controller was set to 90°C on the hot side and -90°C on the cold side. The low setpoint temperature was set at -90°C so that any liquid Nitrogen left in the tubing could evaporate and warm up and so that the thermal momentum of the base plate mass does not take the temperature below -100°C . The 90°C hot setpoint was chosen so that temperature oscillations around 80°C would not set off the temperature limit controller. These values can be changed to suit the individual needs of each test object for all future tests.

As previously mentioned, cycling the base plate temperature with the test satellite was not sufficient to validate the base plate design. However, after successfully cycling the temperature several times between its designed extremes, its functionality was proven. Two pieces of data were needed so that a comparison could be made between this test and an identical test including the thermal shroud (presented later). First, the heat power through the base of the satellite; and second, a comparison between the predicted temperature distribution in the satellite and the measured temperature distribution. Figure 6.9 shows the heat rate through the base of the satellite over the entire duration of the test, calculated using Fourier's Law and using the measured temperatures from the two thermocouples named 'Front, Bottom a (1)' and 'Front, Bottom b (2)'. For reference, the 'BP Top Inlet Side' thermocouple temperature profile is also included in the plot. The point of interest was at the peak at 24 hours on the x-axis, when the transition from cold to hot ended. The heat power per unit area at this point in time was $19,286 \text{ W/m}^2$. Referring back to the results in Chapter

4, the predicted heat power per unit area through the base of the satellite model was 833 W (or $3,332 \text{ W/m}^2$, through the bottom surface of the satellite).

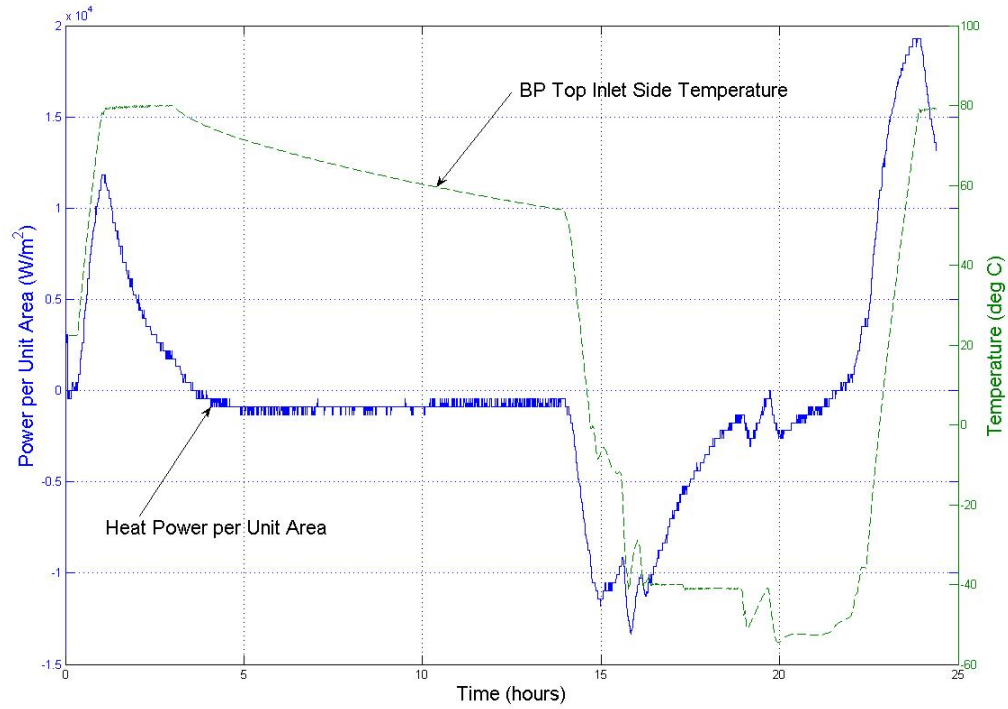


Figure 6.9: Measured Power Through the Bottom of the Satellite

This difference was significant. There were several reasons for this, not the least of which was the difference between the assumptions used in the satellite model analysis and the physical construction of the test satellite. In the satellite model all components were assumed to have zero contact resistance at their interfaces. However, the test satellite was assembled using a number of fasteners and had many contact surfaces, and so it did have significant contact resistance between components, as evidenced in Figures 6.7 and 6.8. It should also be noted that the heat power applied to the bottom side of the base plate during the transition from -40°C to 80°C was 926 W, nearly 100 W more than the satellite model analysis predicted (see Chapter 4).

This was done so that the base plate capabilities were tested in addition to its basic functionality. The operational limit of the system was 930 W, and by testing near this limit the rate of temperature change was expected to be greater than $1.5^{\circ}\text{C}/\text{min}$, which would suggest that the minimum design requirement was met.

Using the ‘BP Top Inlet Side’ thermocouple, the rate of temperature change was calculated for the duration of the test, as shown in Figure 6.10. For reference, the ‘BP Top Inlet Side’ thermocouple temperature profile is also included in the plot. The rate of temperature change increased to at least $1.5^{\circ}\text{C}/\text{min}$ when transition from -40°C to 80°C began, and dropped off as the temperature approached 80°C . This was because of radiation heat transfer between the base plate and test satellite and the chamber. While the base plate temperature was less than room temperature, heat moved from the chamber to the base plate, helping to increase the achievable rate of temperature change. Once the temperature passed room temperature, heat moved from the base plate to the chamber, reducing the achievable rate. The result was the sudden increase and slow decrease of the rate of temperature change, seen in Figure 6.10.

During the entire transition time, the rate exceeded $1^{\circ}\text{C}/\text{min}$ which met the minimum requirement. However, the heat power input was 926 W and so the rate of temperature change was expected to be greater than $1.5^{\circ}\text{C}/\text{min}$ given the results of the satellite model analysis from Chapter 4. While the rate reached approximately $1.7^{\circ}\text{C}/\text{min}$ and the requirement was met, several factors suggest that the system was under-designed. First, the satellite model analysis did not factor in the thermal mass of the base plate, so while 833 W was predicted to raise the temperature at the base of the satellite at a rate of $1.5^{\circ}\text{C}/\text{min}$, more power was needed when the base plate thermal mass was added to the control mass. Second, per the discussion above, heat loss to the chamber was also not factored in to the base plate analysis. Finally, the test

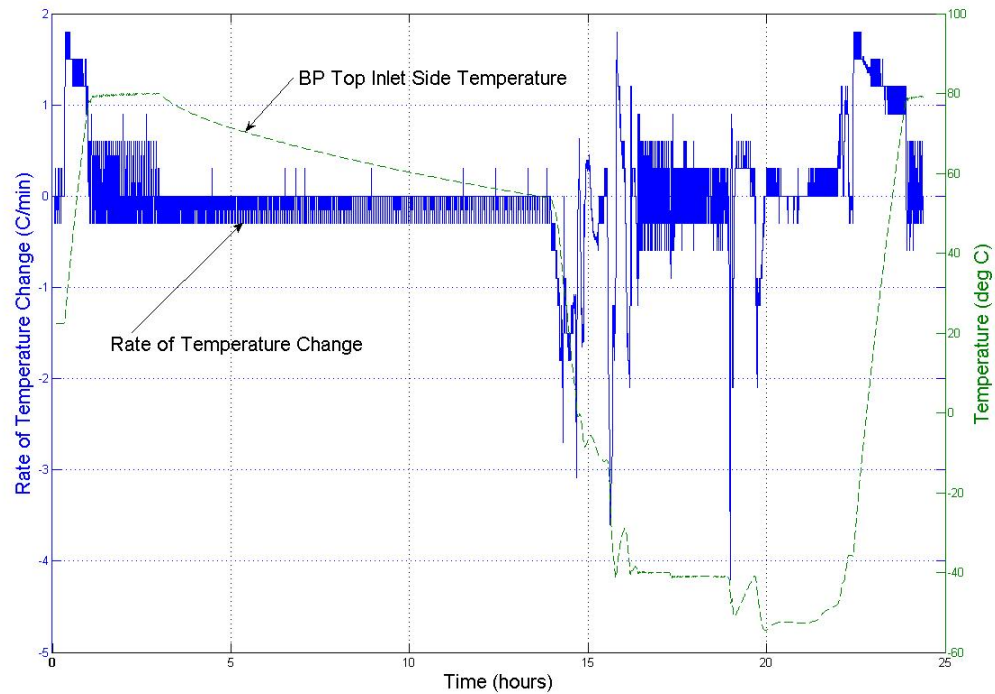


Figure 6.10: Measured Rate of Temperature Change at the Base Plate/ Satellite Interface

satellite had a mass of approximately 14 kg less than the mass of the satellite model, so if a larger test satellite had been used, the rate of temperature change would have been less than the results seen thus far, and the $1^{\circ}\text{C}/\text{min}$ requirement would likely have not been met. These indicators were confirmed by the results shown in Figure 6.10. The logical conclusion was that even at the base plate's maximum heat power input, the rate of temperature change of a 40.5 kg test object may not have met the $1^{\circ}\text{C}/\text{min}$ requirement.

This conclusion did not mean, however, that the base plate had to be re-designed or re-tested. The requirements stated that the thermal system, including the thermal shroud, must meet the $1^{\circ}\text{C}/\text{min}$ requirement. Testing with the thermal shroud would indicate if this requirement had been fully met.

Another comparison between the test data and the satellite model prediction was the temperature distribution throughout the entire satellite. Figure 6.11 shows the predicted temperatures across the entire satellite, taken at the time when hot transition becomes hot soak, on the outside as well as at the internal components. Figure 6.12 indicates the locations of the temperature probes in the model which correspond directly to the locations of the thermocouples on the test satellite, on the outside (Figure 6.12a) and on the inside (Figures 6.12b and 6.12c). Table 6.1 lists a comparison between the predicted and measured temperatures at the point when the transition period ends.

Within the satellite model (Figure 6.11) there was consistent uniformity of temperature at any given height. The trend from bottom to top was a decrease in temperature, as expected. Also, the temperature tended to decrease from the outside to the inside of the satellite. Many of these same trends were present in the measured temperatures, with some exceptions. Most noticeably was that the measured temperatures were significantly lower than the predicted temperatures. The center stack of the test satellite was warmer at a given height than the outside of the satellite. The ‘Inside, Top Box’ temperature was warmer than the ‘Inside, Center Box’ when it was 21.4 cm (8.5 in) farther from the base plate. The Front and Left side temperatures of the test satellite were 7°C to 8°C warmer than the Back and Right side temperatures.

A number of variables could have been the cause of these apparent measurement inconsistencies. Thermocouples are known to have measurement errors on the order of 1°C to 2°C which could explain why the center stack was warmer than the outside of the satellite. However, there were more fasteners and there was more surface area between components in the center stack, which would have helped speed the flow of heat. Contrarily, there was more thermal mass in the center stack, which would slow the flow of heat. So, errors in the thermocouple measurements may or

Table 6.1: Predicted and Measured Temperatures of the Test Satellite During the Base Plate Test

No.	Probe Name (Location)	Height Above Base, cm (in)	Predicted Temp. (°C)	Measured Temp. (°C)	% Diff.
1	Front, Bottom a	2.54 (1)	76.18	43.40	43.03
2	Front, Bottom b	6.35 (2.5)	70.35	39.00	44.56
3	Front, Middle	21.6 (8.5)	35.58	19.70	44.63
4	Front, Top	43.2 (17)	14.40	6.00	58.33
5	Left, Bottom	2.54 (1)	76.18	44.50	41.59
6	Left, Middle	21.6 (8.5)	35.58	20.30	42.95
7	Left, Top	43.2 (17)	14.40	7.44	48.33
8	Back, Bottom	2.54 (1)	76.18	36.90	48.02
9	Back, Middle	21.6 (8.5)	35.58	17.50	50.82
10	Back, Top	43.2 (17)	14.40	4.65	67.71
11	Right, Bottom	2.54 (1)	76.18	37.40	50.91
12	Right, Middle	21.6 (8.5)	35.58	15.60	56.16
13	Right, Top	43.2 (17)	14.40	4.44	69.17
14	Top, Wing Corner	45.7 (18)	12.25	2.47	79.84
15	Top, Box Wall Center	45.7 (18)	11.55	1.88	83.73
16	Top, box Center	45.7 (18)	11.15	1.13	89.87
17	Inside, Bottom Box	2.54 (1)	74.05	42.70	42.34
18	Inside, Center Box	21.6 (8.5)	35.65	3.03	91.50
19	Inside, Top Box	43.2 (17)	12.35	4.18	66.14
20	Inside, Wing Bottom, Center	2.54 (1)	75.95	10.80	85.78
21	Inside, Wing Middle, Skin	21.6 (8.5)	36.15	12.90	64.32
22	Inside, Wing Middle, Center	21.6 (8.5)	35.35	15.00	57.57
23	Inside, Wing Middle, Box	21.6 (8.5)	34.35	46.40	-35.08
24	Inside, Wing Top, Center	43.2 (17)	12.65	1.04	91.78
25	Inside, Wing Top, Box	43.2 (17)	12.35	11.20	9.31

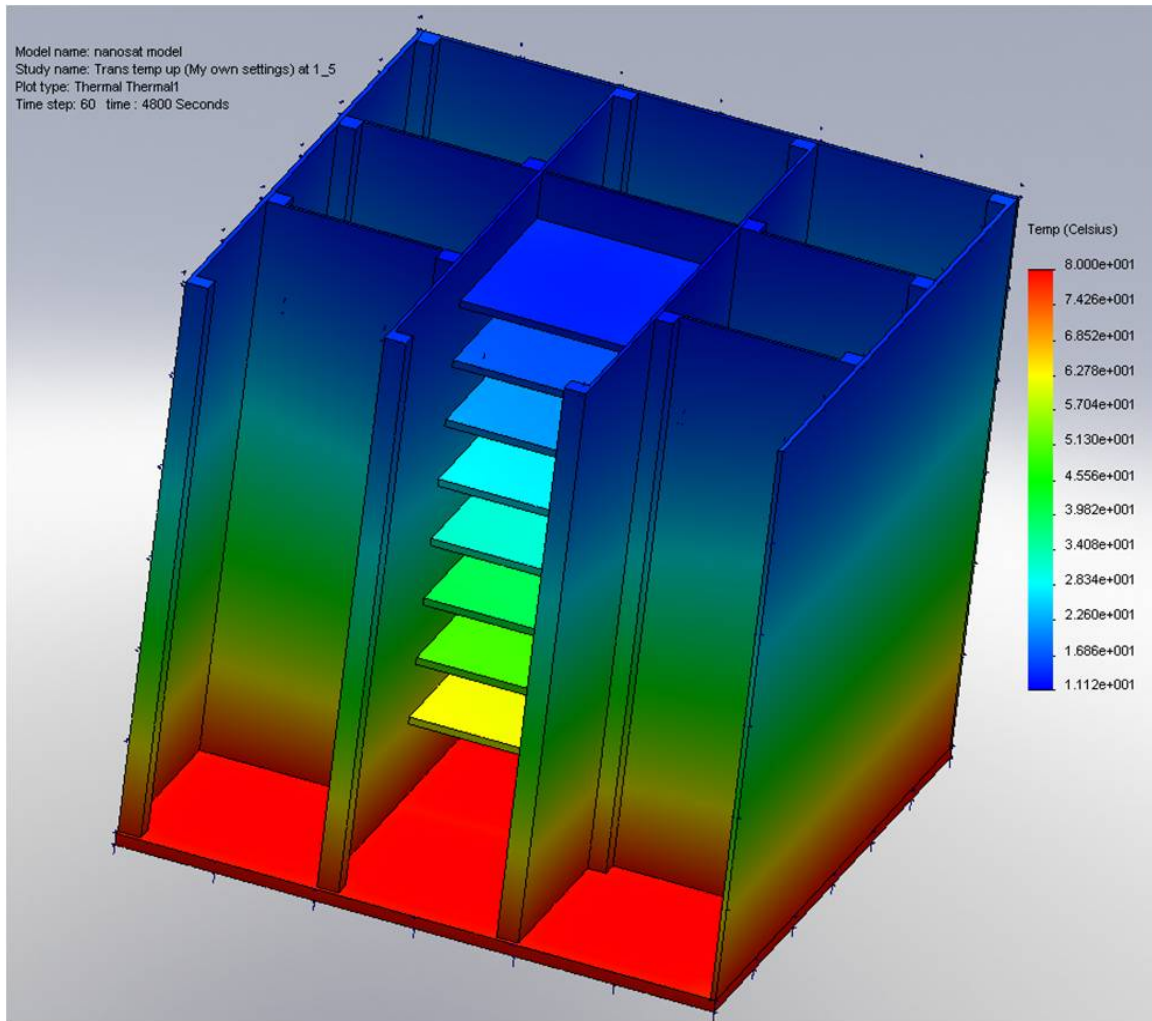


Figure 6.11: Temperature Distribution for 4.9 cm Mesh at 1.5°C/min

may not have been the cause. A more likely culprit was contact resistance between components, or more accurately, uneven or asymmetric contact resistance across the test satellite. Notes from the test satellite assembly process indicated that there was a design or manufacturing defect in the test satellite where the threaded inserts on the Front/Left and Back/Right sides were different sizes. This meant different fasteners and different tightening torques, which would explain the 7°C to 8°C difference between the Front/Left and Back/Right sides of the test satellite. The sides did not touch the base plate, so heat had to flow through the satellite bottom plate and then into the

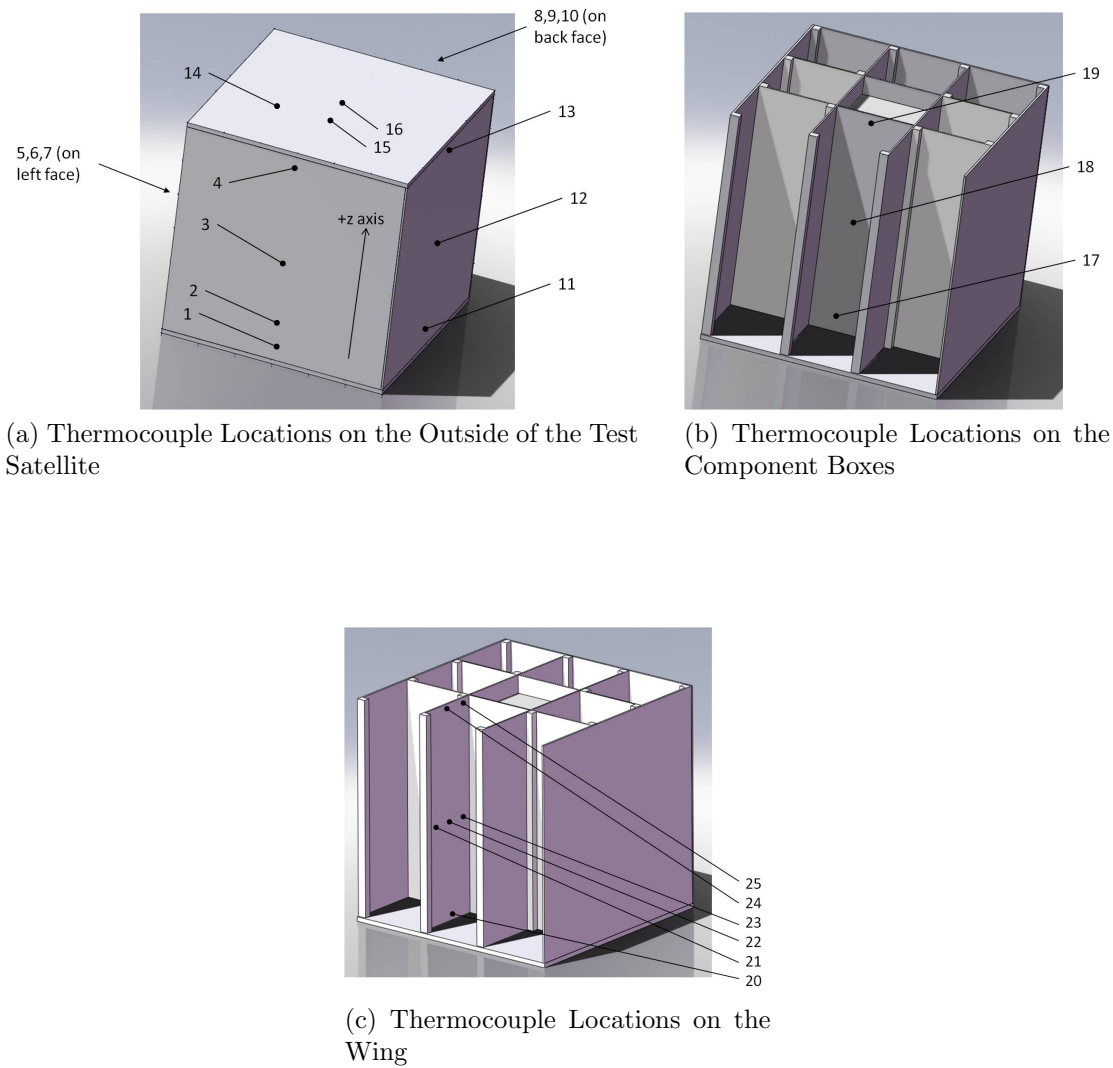


Figure 6.12: Thermocouple Locations on the Test Satellite

sides through the fasteners. The interface plate between the satellite and the base plate may also have been tightened unevenly, but there were no notes to this effect in the test satellite assembly process.

Variations in how the thermocouples were attached at each location also could have been a cause. The end of each thermocouple had a piece of Kapton tape applied to electrically insulate it from the test satellite, and then a piece of Aluminum tape placed over it to secure it to the surface. Care was taken to attach them uniformly with similar sized pieces of tape, but not so much care as to ensure that they were attached identically.

The base plate itself also could have been the cause of this uneven temperature distribution, but this was unlikely. The heaters maintained a constant resistance over the course of the test duration indicating that the generated heat was evenly distributed across the bottom of the base plate. If the resistance had changed it would have indicated that one or more of the tape heaters had been damaged, and generating significantly more or less heat than designed. Therefore, the temperature distribution of the tape heaters was not a likely cause of the uneven temperature distribution in the test satellite.

Another noteworthy observation from the base plate testing was the temperature difference formed between the inlet and outlet of the base plate tubing. This trend was first observed during the first base plate test, shown in Figures 6.3b and 6.4b. When liquid Nitrogen was introduced into the base plate tubing a difference developed, sometimes as large as 40°C, as shown in Figure 6.13 at approximately hour 16. While the temperature difference was not unexpected, the severity was surprising. During all tests, steps were taken to minimize this difference as much as possible. Maintaining a steady but slow flow rate of liquid Nitrogen was found to be the most effective way to minimize this difference.

The assumptions made during the base plate design were, as previously mentioned, made for simplicity and because the behavior and properties of the liquid Nitrogen in the tubing were unknown. Figure 6.13 indicates that a more thorough analysis of the temperature distribution in the base plate should have been attempted. Knowledge of how the liquid Nitrogen behaved and under what conditions the system was operated also made it apparent that a more thorough analysis should have been performed, and that a tubing design with several parallel paths would have been the better choice, despite the mechanical difficulties.

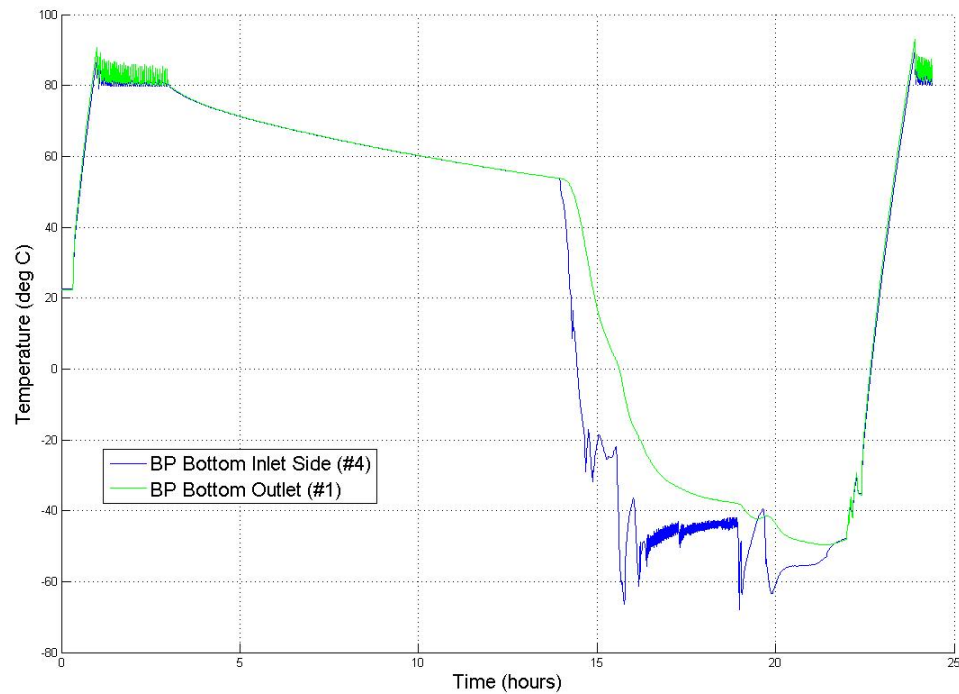


Figure 6.13: Measured Temperatures on the Base Plate Bottom Side

The base plate was functioning properly and was capable of heating and cooling a test object within the required temperature limits. It's performance was acceptable given the results of the testing. The performance of the full system was therefore

expected to improve with the addition of the thermal shroud operating in conjunction with the base plate.

Thermal Shroud Testing

Like the base plate, the thermal shroud was installed in the chamber and operated under conditions which would allow for validation of the design and verification of functionality. The test satellite was installed to the base plate with the 25 test object thermocouples, which were not removed from the satellite following the base plate test. The thermal shroud was situated on top of the insulators, on the top of the base plate, and surrounding the test satellite. To the outside of the thermal shroud were attached eight thermocouples as previously discussed.

Two distinct tests were performed with the thermal shroud installed in the system. The first was an identical temperature profile used in the second base plate functional test. This served to verify functionality of the thermal shroud, as well as to simulate the conditions needed to validate the base plate design as previously discussed. The second type of test was meant to validate the thermal shroud design. The base plate and satellite were cooled to -40°C and held there, and then the thermal shroud was heated to 80°C . This maximized the temperature difference between the satellite and shroud and simulated the conditions that were used in the shroud design.

First Thermal Shroud Test

Once installed and set up, the thermal shroud was operated in vacuum for the first time. Both zones (body and top) were set to heat at their maximum rate (100 percent duty cycle) to test the system capabilities and to verify that the heater design would meet the requirements. Unfortunately, an odor from the output of the scroll pump

was present after about 30 minutes which smelled like something burning, so the test was shut down immediately until the source could be found. The source was found to be the tape heaters on the thermal shroud. An electrical short had developed between the heater element and the metallic body of the shroud itself in several locations, which was grounded to earth for safety. As a result, for some portion of the 30 minutes of operation the heater power supplies had been sourcing their maximum current, and all of the heat power generated by the heaters was constrained to the section of the heater between the positive terminal and the short location.

The short was found using temperature information from the thermocouples. Near the liquid Nitrogen inlet the temperature had increased rapidly while the other side of the plate barely increased, indicating that the short was located near the inlet. The short was verified by measuring the electrical resistance between the heater terminals and the shroud body, where the heater end with the smallest measured resistance was closest to the short. The heat tape nearest the short was removed and inspected, and a burn mark from the dielectric material and adhesive was present. The heat tape supplier [17] was contacted to ascertain the reason for the failure, and it was determined that the heat tape used for the heater was not designed for application to a conductive, metallic surface - an oversight during the design process. The correct product would have been manufactured with a double layer of dielectric material to prevent shorts from occurring when applied to metallic surfaces.

The compromised tape heaters, from the top and body of the thermal shroud, were removed and discarded, and an identical set of double dielectric heaters were purchased as a replacement. Prior to testing the rebuilt heaters, the heater power supplies were tested to be certain that no damage had occurred to them as a result of running them at or exceeding their limits. A high power heating element was connected to each power supply, with a resistance approaching the maximum recom-

mended load, and each power supply was run for 30 minutes to verify that no damage had been done. Both power supplies passed the test and neither require replacement.

The replacement heaters were installed and checked to be sure that the resistance was correct and that there were no shorts to the shroud. While the heaters were being constructed by the manufacturer, a much needed addition and improvement was added to the thermal vacuum system. Until this point, all of the system components had to be operated through the LabVIEW software which, while assuring system safety, made performing simple system diagnostics very difficult and also did not allow for emergency manual overrides. A set of switches was added to the Power Distribution Box (PDB) so that in an emergency or for basic operation (advanced users only) a user could operate the two pumps, all vacuum system valves, the liquid Nitrogen valves and the shroud heaters. The scroll pump could be turned on or off by its switch, while the turbo pump could be turned off or set to an enabled state (the turbo pump controller actually operated the pump). Each of the valve and heater switches had three positions: Auto (for using LabVIEW), manual ON, and manual OFF. In wiring the switches this way, the main PDB power switch continued to function as a system emergency shutoff switch. By turning off the power to the PDB, the entire vacuum system was put into a safe configuration, the liquid Nitrogen valves were closed, and the shroud heaters were turned off. Only the base plate heater was not affected by this switch.

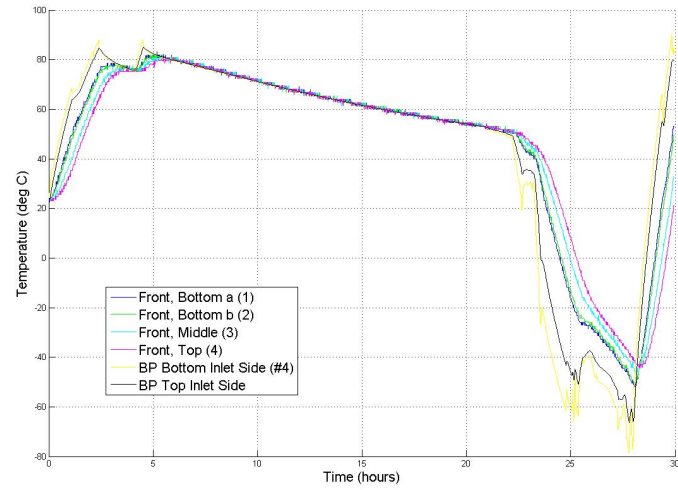
Thermal Shroud Functional Test

The repaired thermal shroud was again installed in the chamber and a second attempt to test the system with the shroud was undertaken. The temperature profile from the final base plate test was followed so that a valid comparison could be made between use with and without the thermal shroud. Both were heated to 80°C and

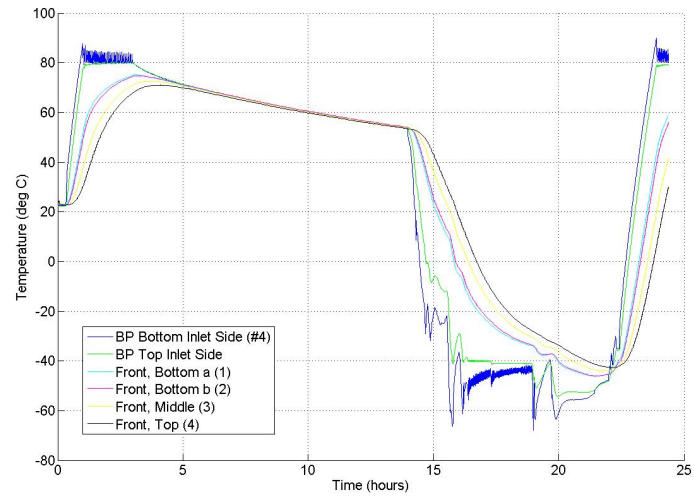
cooled to -40°C which validated the thermal shroud functionality. The base plate design was validated by gathering the same data as before and comparing the results to the base plate design. Figure 6.14a shows base plate and satellite temperatures measured with the thermal shroud installed in the system, and for comparison the same temperature measurements without the thermal shroud (from the final base plate test) are shown in Figure 6.14b. With the thermal shroud the temperature gradients in the test satellite were smaller. Also of interest were the shroud body temperatures (Figures 6.15a) and shroud top (cover) temperatures (Figure 6.15b). All of the shroud and satellite temperatures are combined and shown in Figure 6.16. The Front temperatures of the satellite are included in each figure for easy comparison between the four plots.

At the transition from heating to hot soak at approximately hour 29, the heat power through the base of the test satellite was $17,094\text{ W/m}^2$. Compared to the test with only the base plate, which had $19,286\text{ W/m}^2$, this was closer to the predicted value of $3,332\text{ W/m}^2$, but only an 11% reduction. The reasons for it being higher than the predicted value were discussed in the Chapter 6. The reduction in measured heat power was the direct result of using the thermal shroud in the system. Heat moved to the satellite from the shroud as well as from the base plate, so less heat was required from the base plate to achieve the same temperature change rate in the test satellite.

Figure 6.17 shows the rate of temperature change of the test satellite along with the thermal shroud in use. For reference, the ‘BP Top Inlet Side’ thermocouple temperature profile is also included in the plot (this was the temperature measured at the interface between the base plate and test satellite). During the cold transition the rate exceed the $1^{\circ}\text{C}/\text{min}$ rate, but not for the entire cooling transition as had been the case during hot transitions. Based on experience, this rate could be

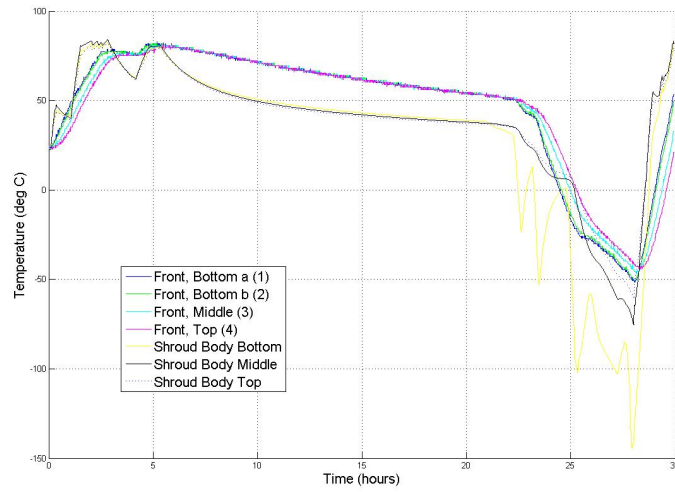


(a) Base Plate and Satellite Temperatures with the Thermal Shroud

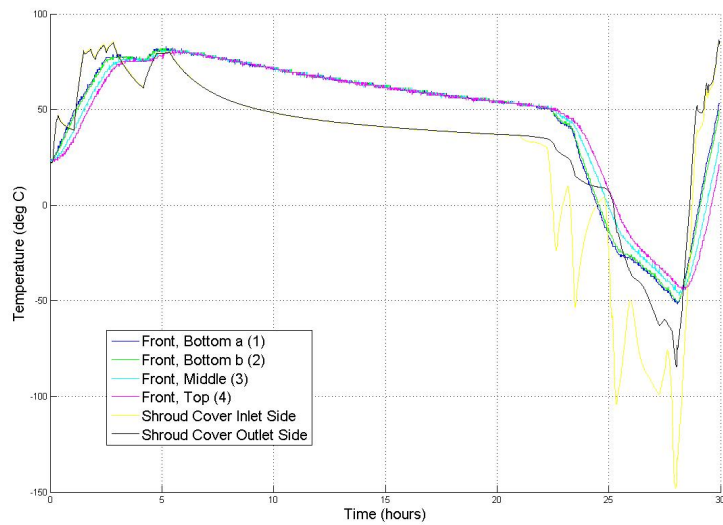


(b) Figure 6.8 Repeated for Comparison

Figure 6.14: Satellite and System Temperatures with and without the Shroud



(a) Shroud Body and Satellite Temperatures



(b) Shroud Top and Satellite Temperatures

Figure 6.15: Shroud and Satellite Temperatures

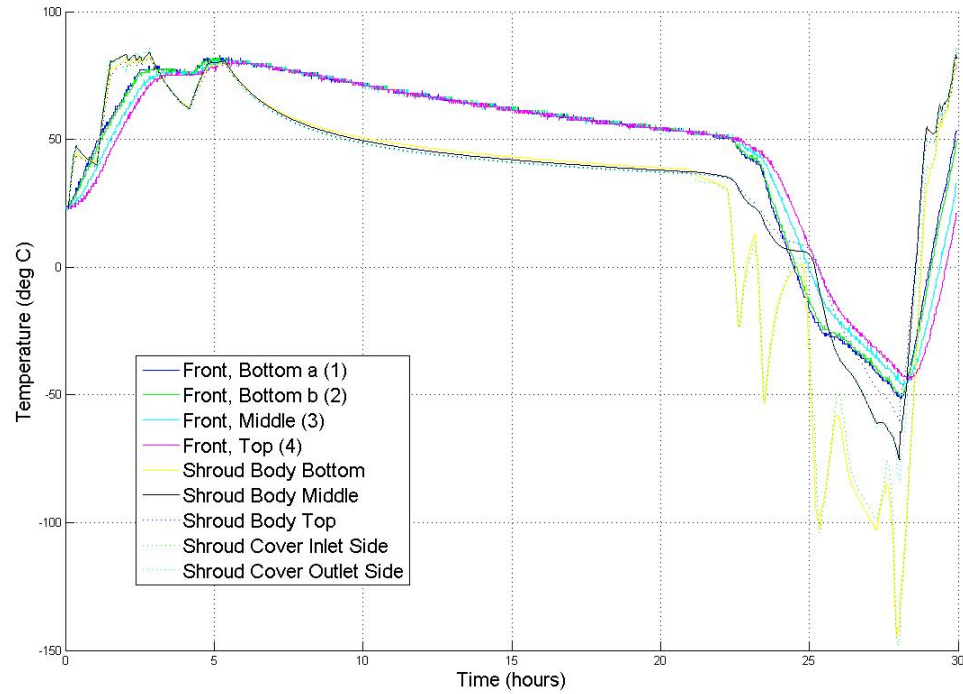


Figure 6.16: All Shroud and Satellite Temperatures

achieved with adjustments to the liquid Nitrogen tank valve, and so an additional test to verify this specific requirement was not performed. During heating the rate reached approximately $2.7^{\circ}\text{C}/\text{min}$ and decreased over the course of the hot transition to $1^{\circ}\text{C}/\text{min}$. Thus, testing showed that the rate requirement was met during heating but not during cooling. A test remained to validate the thermal shroud design, and the cold transition attempted to validate this requirement.

The second point of comparison was the measured temperature distribution in the test satellite. Temperatures with and without the thermal shroud are shown in Table 6.2. With the thermal shroud, temperatures on the front of the test satellite were closer to the predicted temperatures from the satellite model. At each location, the temperature with the shroud was greater than the corresponding temperature

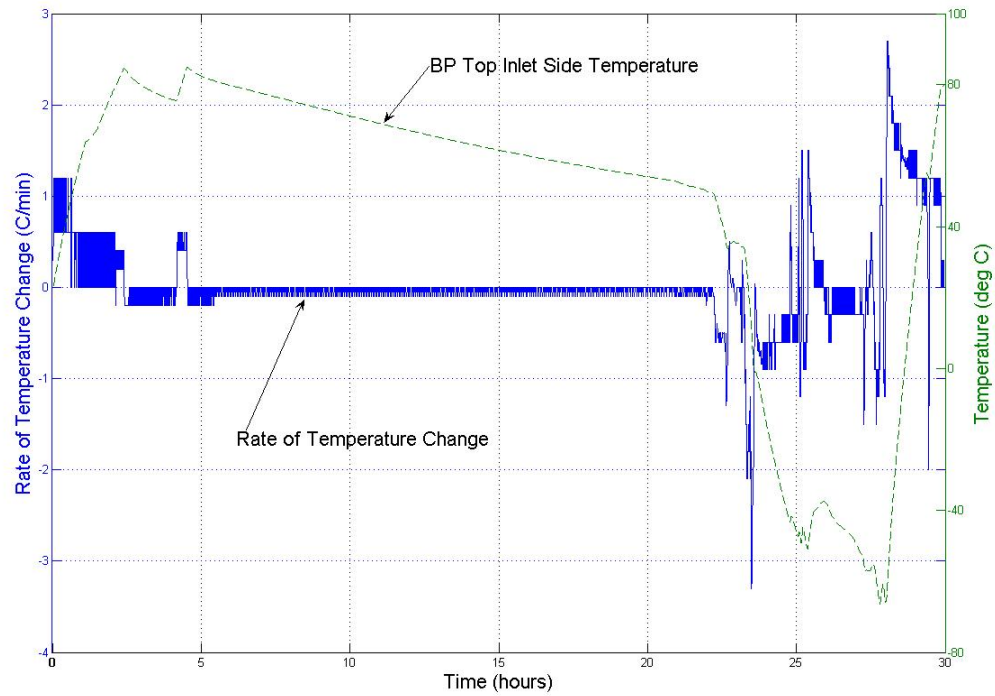


Figure 6.17: Measured Rate of Temperature Change at the Base Plate/ Satellite Interface with the Thermal Shroud

from the base plate test indicating again that the thermal shroud did improve the performance of the system. The temperature difference from the bottom to the top of the test satellite dropped from approximately 36°C to approximately 32°C. The shroud did not remove the gradient from the Front/Left to the Back/Right sides, although it was reduced.

This temperature cycling verified the full functionality of the thermal shroud. The final test is intended to validate the design of the thermal shroud after which the entire thermal system design and functionality will have been validated.

Table 6.2: Comparison of Measured Temperatures on the Test Satellite with and without the Thermal Shroud

No.	Probe Name (Location)	Height Above Base, cm (in)	Shroud Test Temp. (°C)	Base Plate Test Temp. (°C)	% Diff.
1	Front, Bottom a	2.54 (1)	47.00	43.40	8.51
2	Front, Bottom b	6.35 (2.5)	43.10	39.00	9.51
3	Front, Middle	21.6 (8.5)	26.10	19.70	24.52
4	Front, Top	43.2 (17)	15.50	6.00	61.29
5	Left, Bottom	2.54 (1)	48.40	44.50	8.06
6	Left, Middle	21.6 (8.5)	26.40	20.30	23.1
7	Left, Top	43.2 (17)	17.10	7.44	56.49
8	Back, Bottom	2.54 (1)	42.00	36.90	12.14
9	Back, Middle	21.6 (8.5)	23.80	17.50	26.47
10	Back, Top	43.2 (17)	13.20	4.65	64.77
11	Right, Bottom	2.54 (1)	40.90	37.40	8.56
12	Right, Middle	21.6 (8.5)	22.50	15.60	30.67
13	Right, Top	43.2 (17)	13.20	4.44	66.36
14	Top, Wing Corner	45.7 (18)	10.60	2.47	75.98
15	Top, Box Wall Center	45.7 (18)	10.7	1.88	76.70
16	Top, box Center	45.7 (18)	9.41	1.13	87.99
17	Inside, Bottom Box	2.54 (1)	46.00	42.70	7.17
18	Inside, Center Box	21.6 (8.5)	10.60	3.03	71.42
19	Inside, Top Box	43.2 (17)	11.90	4.18	64.87
20	Inside, Wing Bottom, Center	2.54 (1)	16.00	10.80	32.50
21	Inside, Wing Middle, Skin	21.6 (8.5)	19.00	12.90	32.11
22	Inside, Wing Middle, Center	21.6 (8.5)	21.80	15.00	31.19
23	Inside, Wing Middle, Box	21.6 (8.5)	49.40	46.40	6.07
24	Inside, Wing Top, Center	43.2 (17)	8.80	1.04	88.18
25	Inside, Wing Top, Box	43.2 (17)	16.90	11.20	33.73

Thermal Shroud Design Validation

The same test setup was used for this test as in the thermal shroud functionality test. Operationally, the base plate and thermal shroud were cooled until the test satellite reached a uniform temperature of approximately -40°C . The base plate and test satellite were then held at -40°C while the thermal shroud was driven at its maximum heater power until the temperature reached 80°C or as hot as it could go, whichever came first. In the thermal shroud analysis the shroud heater power was calculated under these conditions, and if the design was correct, the thermal shroud was expected to reach 80°C .

Figure 6.18 shows the resulting temperature profile, including the ‘Shroud Body Bottom’, ‘Shroud Cover Inlet Side’, ‘BP Top Inlet Side’, and two test satellite thermocouple measurements. Temperatures in the thermal shroud, both top and bottom, reached 80°C while the satellite and base plate remained at -40°C and so the design of the thermal shroud was validated by successfully operating the thermal shroud under the same conditions used in its design. The oscillations in the base plate and thermal shroud body and top were due to adjustments in the regulating valve on the liquid Nitrogen tank. Because all three thermal zones of the thermal system were controlled by the same valve these oscillations correlated between the zones (base plate, shroud body, shroud top (cover)). The base plate was affected less than the shroud because it had a much higher thermal mass, and because the test satellite was in direct contact with it.

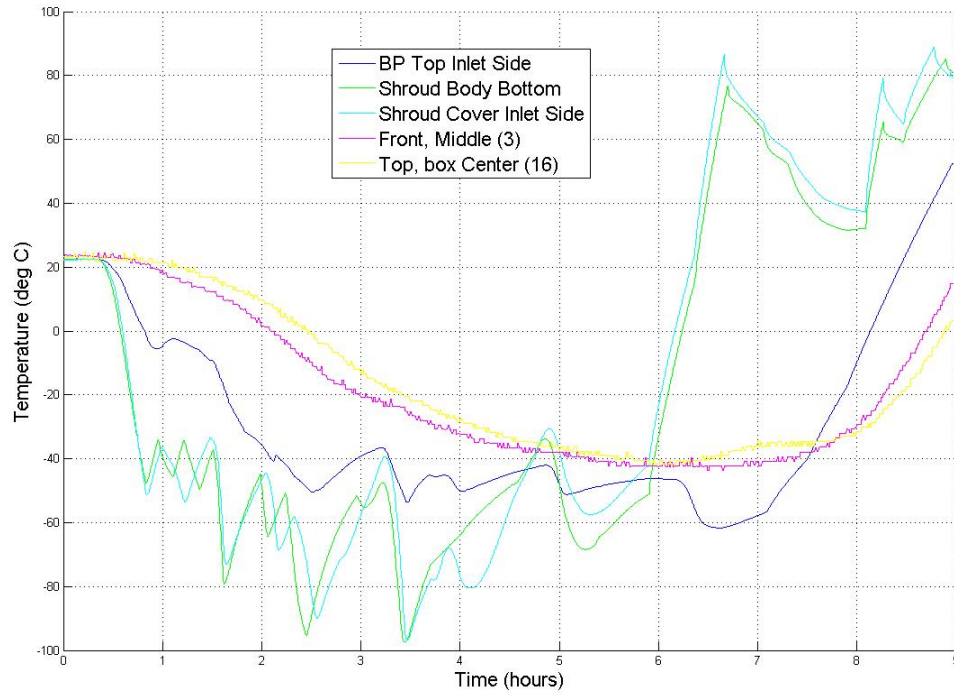


Figure 6.18: Temperature Profile During the Shroud Design Validation Test

Requirements Verification

Having completed all system testing, the requirements for the vacuum system, thermal system, base plate and thermal shroud have been met. The tables in Appendix B summarize how and where each requirement was verified. Table B.1 refers to the vacuum system, Table B.2 refers to the thermal system, Table B.3 refers to the base plate, and Table B.4 refers to the thermal shroud.

The budget requirement of the thermal system was the only requirement not addressed except when comparing prices qualitatively between several design options. Despite cost saving measures taken throughout the design of the thermal system, the total cost of the system components added up to \$25,265 which exceeded the cost

requirement by \$265. Parts required for repairing the damage to the base plate and thermal shroud totaled approximately \$3,500. In addition to the parts used for the system itself, a pair of extreme cold gloves and two oxygen displacement sensors were purchased for user safety. This safety equipment totaled \$788. The grand total spent on the thermal vacuum system was \$29,553.

Although the total spent on the thermal vacuum system exceeded the \$25,000 requirement, it did so by 18%. Had the damage to the system not occurred, the total would not have exceeded \$26,053, which was only 4% more than the requirement. In all, the cost of the system was far less than a commercially available system (\$100,000 or more).

Future Work

The effort put into the design and fabrication of the base plate and thermal shroud yielded a fully functional piece of test equipment that will serve the SSEL for many years. The system provides the means for full satellite system thermal vacuum and thermal balance testing, as well as testing of individual satellite systems, payloads, or other space hardware. The design requirements were developed such that the system would be capable of testing beyond the required limits of a typical test, and also to accommodate a variety of testing scenarios.

Despite the effort to design and build a robust system, testing indicated that the system has room for improvement. The base plate design did not include a thorough analysis of the temperature distribution during cooling with liquid Nitrogen. A 40°C temperature difference developed between the inlet and outlet of the tubing on the underside of the base plate. This was the result of using a single tube rather than a series of parallel tubes to better distribute the liquid Nitrogen across the base plate

bottom surface. If the base plate Copper plate and tube assembly were replaced, a parallel tube design should be implemented so that this temperature difference would be minimized. This same phenomenon was present in the thermal shroud body and top. Similar considerations should be made if the shroud body and/or top were to be replaced. It has come to the author's attention that the mechanical difficulties that were mentioned in the design of each of these components could be overcome with the selection of appropriate hardware.

The results of the testing indicated that using the thermal shroud improved system performance, but using the shroud may not always be desirable, in which case storage of the shroud outside of the chamber would be necessary. A method for easily storing the thermal shroud when it is not being used has not been implemented. A movable cart or permanent shelf in the SSEL lab may be possible solutions. Under normal operation, the shroud is used and stored in the chamber, and so this concern is not significant.

Improved temperature control during cooling could be possible by substituting electrically controlled regulating valves for the ON/OFF liquid Nitrogen valves. While running all three zones, an adjustment to the hand valve on the liquid Nitrogen tank or the opening and closing of an electric valve changed the flow rate of liquid Nitrogen into each of the other two zones. The flow of liquid Nitrogen through each zone could be independently controlled, and the effects of one zone on each of the others could be mitigated. Additional software would be needed to control each electric regulating valve.

In addition to hardware improvements, the thermal control software written for the system could be improved. Data sampling was performed at a fairly slow rate so that the data files were not too large to analyze. The amount of data recorded could be reduced further, given that the data were not useful for system analysis. The data

could be useful for system health monitoring, but that can be decided by future users. Data display can also be improved by implementing smoothing algorithms as well as real time plotting functions. The plotting functions may also include the chamber pressure so that easy viewing and correlation between the pressure and temperatures is possible.

Closing Remarks

The SSEL Thermal Vacuum System has been designed and built to meet the requirements specified for each functional subsystem. Through design, analysis, and/or testing each of the requirements has been verified. System testing has also shown that the system is capable of performing thermal vacuum and thermal balance testing of small satellites or other space hardware.

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- [10] MSU Mechanical Engineering student Jordan Evans was the engineer behind the design of the vacuum system table and crane. All components were designed and drafted by Jordan under the author's supervision using SolidWorks CAD software. MSU (and SSEL) students Rubin Meuchel and Justin Rettenmeyer, both of Mechanical Engineering Technology, were the hands behind building and assembling the vacuum system table and crane. Both were supervised by the author during fabrication. Students Brian Larson, Drew Eldeen, Cory Wiltshire, Rubin Meuchel, and Justin Rettenmeyer assisted in installing the chamber to the table. SSEL Staff member Keith Mashburn was instrumental in providing knowledge gained from experience using thermal vacuum systems, as well as for helping to install various pieces of hardware on several occasions. Many thanks are due for the help given by those mentioned.

- [11] Conversation with industry contact at the Southwest Research Institute. Discussions with Roy Graham included the topics of software automation needs of the SSEL vacuum system, industry standard thermal coatings used in thermal vacuum systems, and recommended hardware temperature limit controllers.
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- [20] Out-of-state parts machining. The serpentine copper tubes for the base plate and thermal shroud top, as well as the parallel tubes for the shroud body, were fabricated by Tube Bending, Incorporated, located in Hesperia, California.
- [21] Vacuum furnace brazing. Vacuum furnace brazing of the base plate Copper plate and tube, as well as for the shroud body and top, was contracted to Wall Colmonoy Aerobrazo in Cincinnati, Ohio.
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APPENDICES

APPENDIX A

THERMAL SYSTEM MECHANICAL DRAWINGS

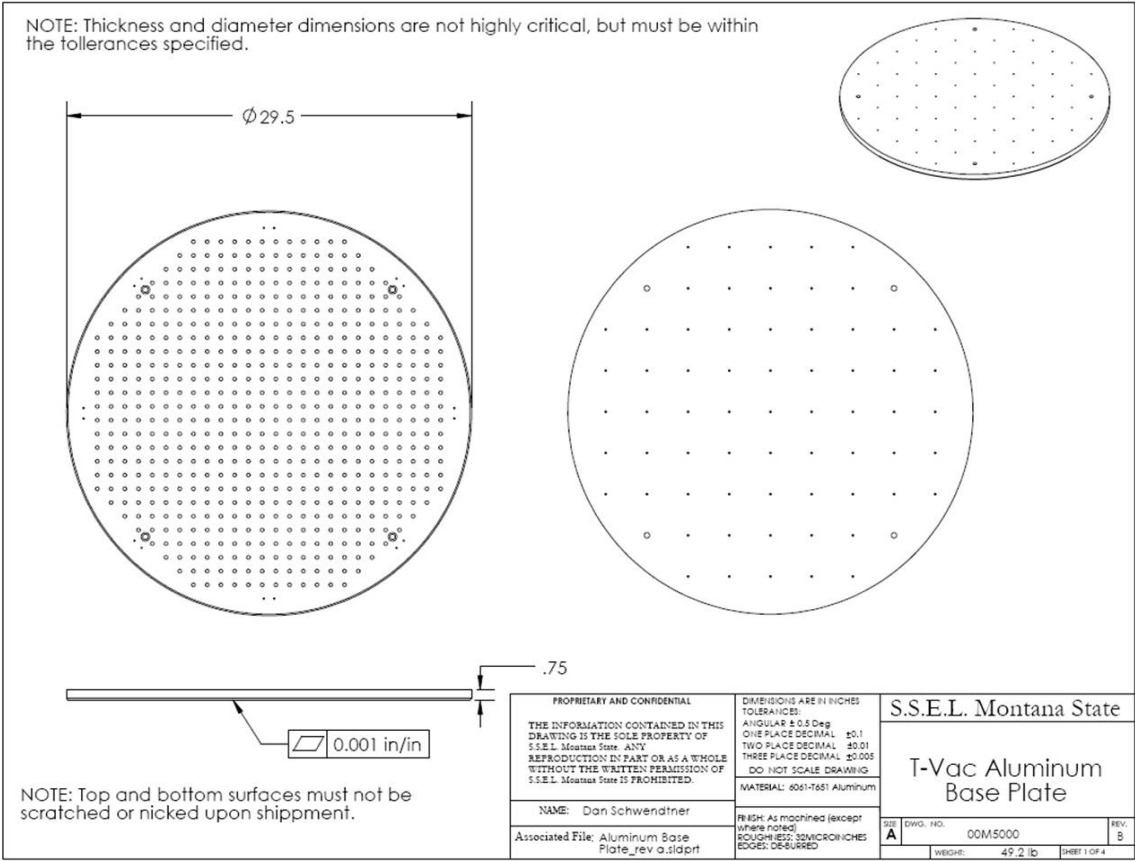


Figure A.1: Thermal Vacuum System Base Plate Mechanical Drawing, 00M5000, Page 1 of 4

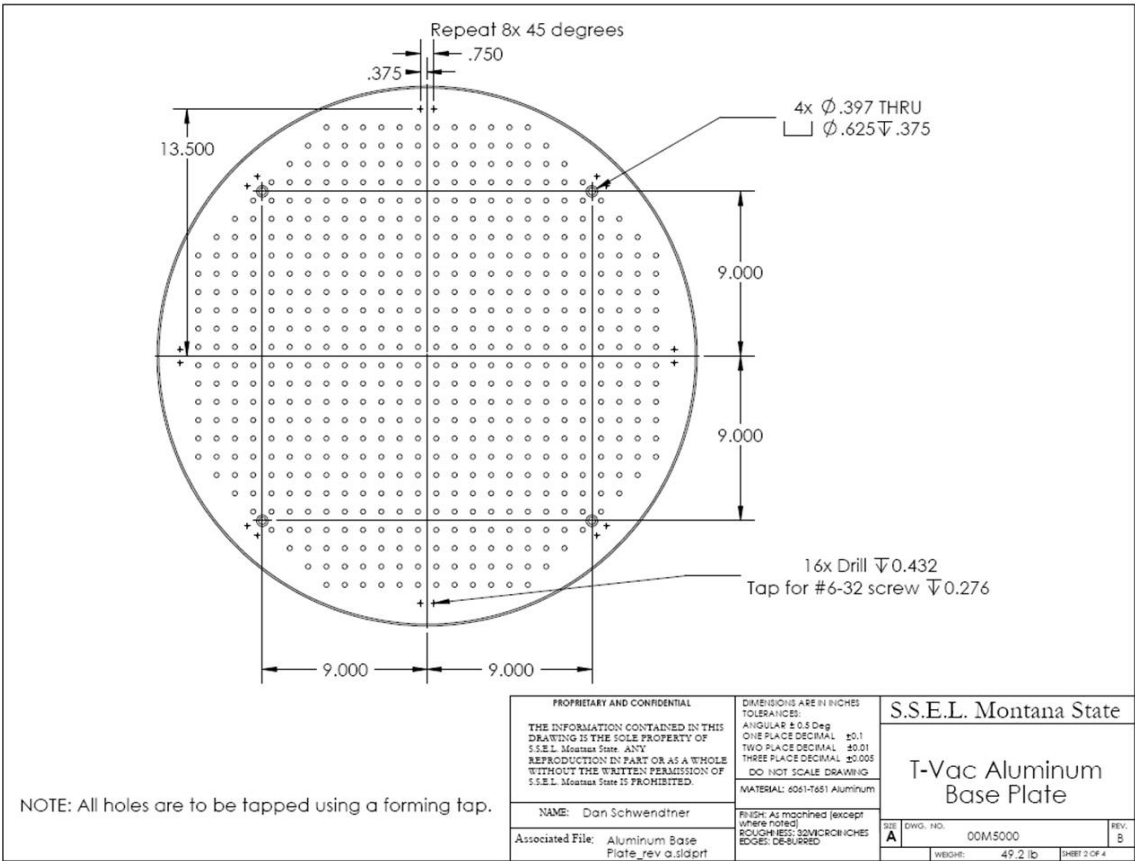


Figure A.2: Thermal Vacuum System Base Plate Mechanical Drawing, 00M5000, Page 2 of 4

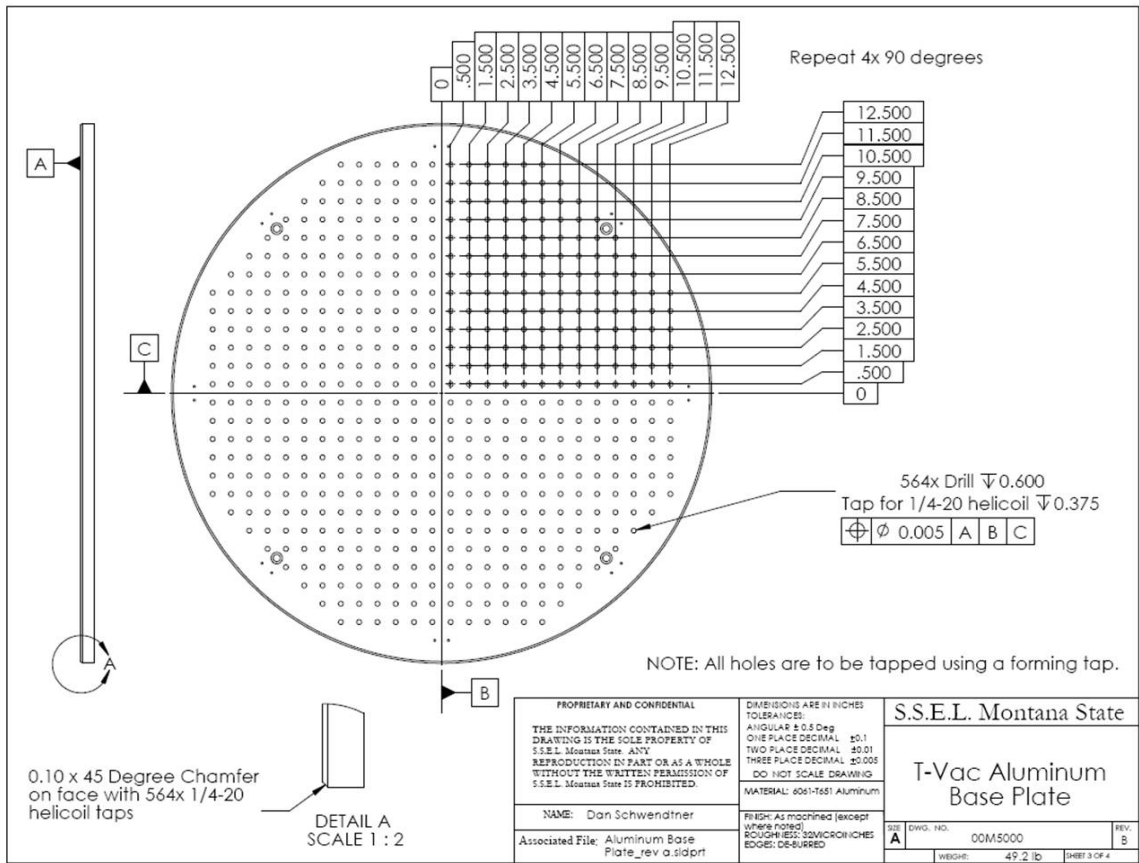


Figure A.3: Thermal Vacuum System Base Plate Mechanical Drawing, 00M5000, Page 3 of 4

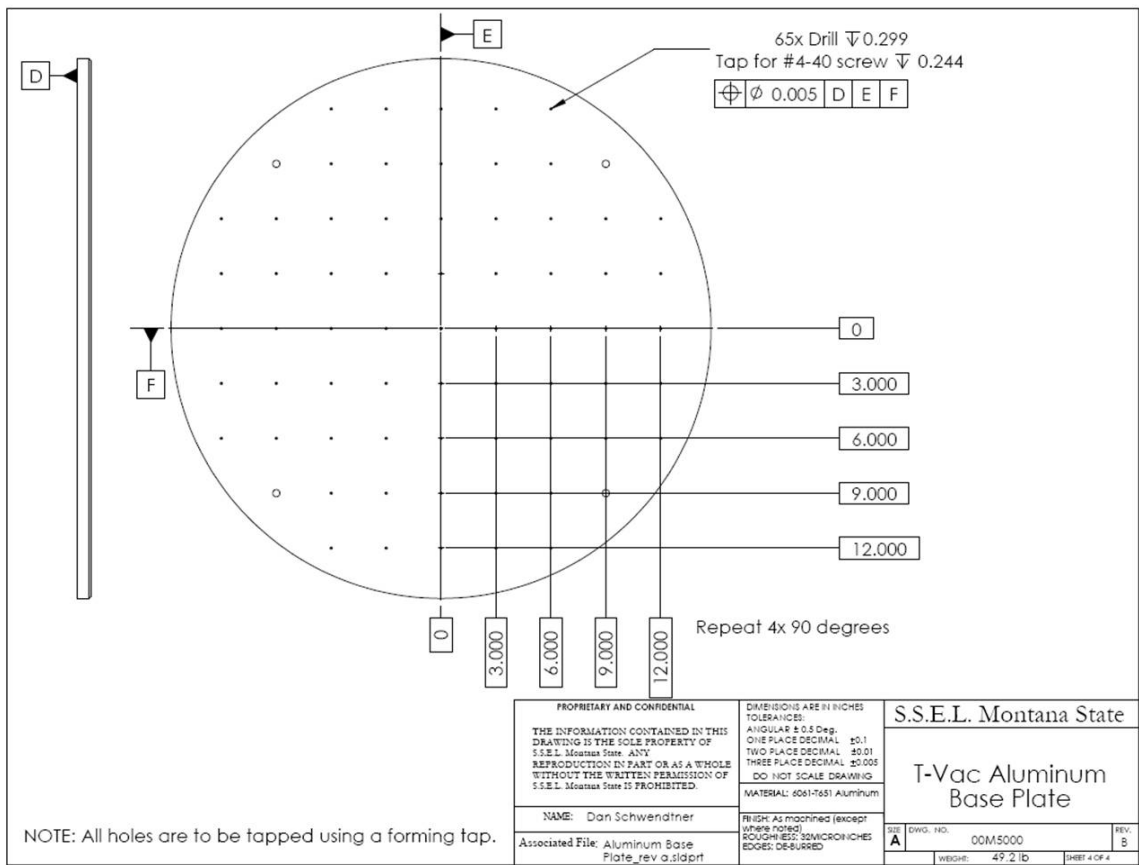


Figure A.4: Thermal Vacuum System Base Plate Mechanical Drawing, 00M5000, Page 4 of 4

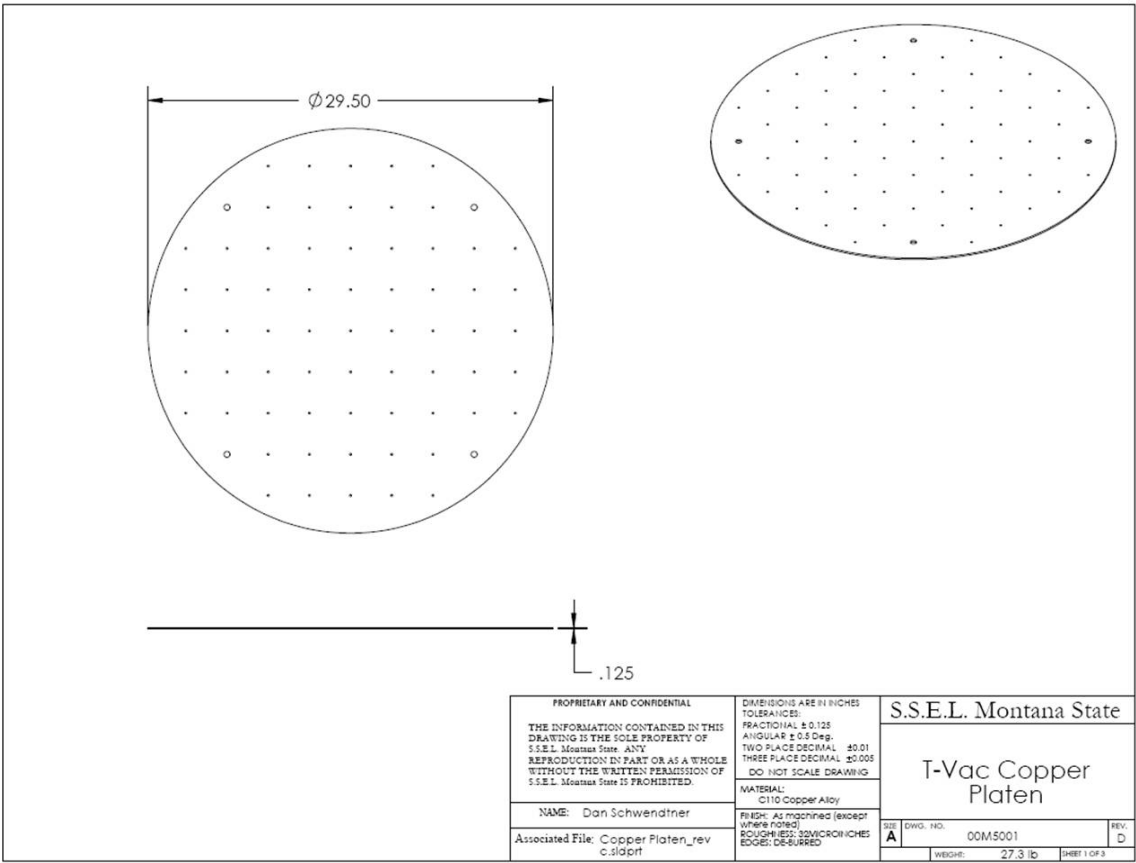


Figure A.5: Thermal Vacuum System Copper Platen Mechanical Drawing, 00M5001, Page 1 of 3

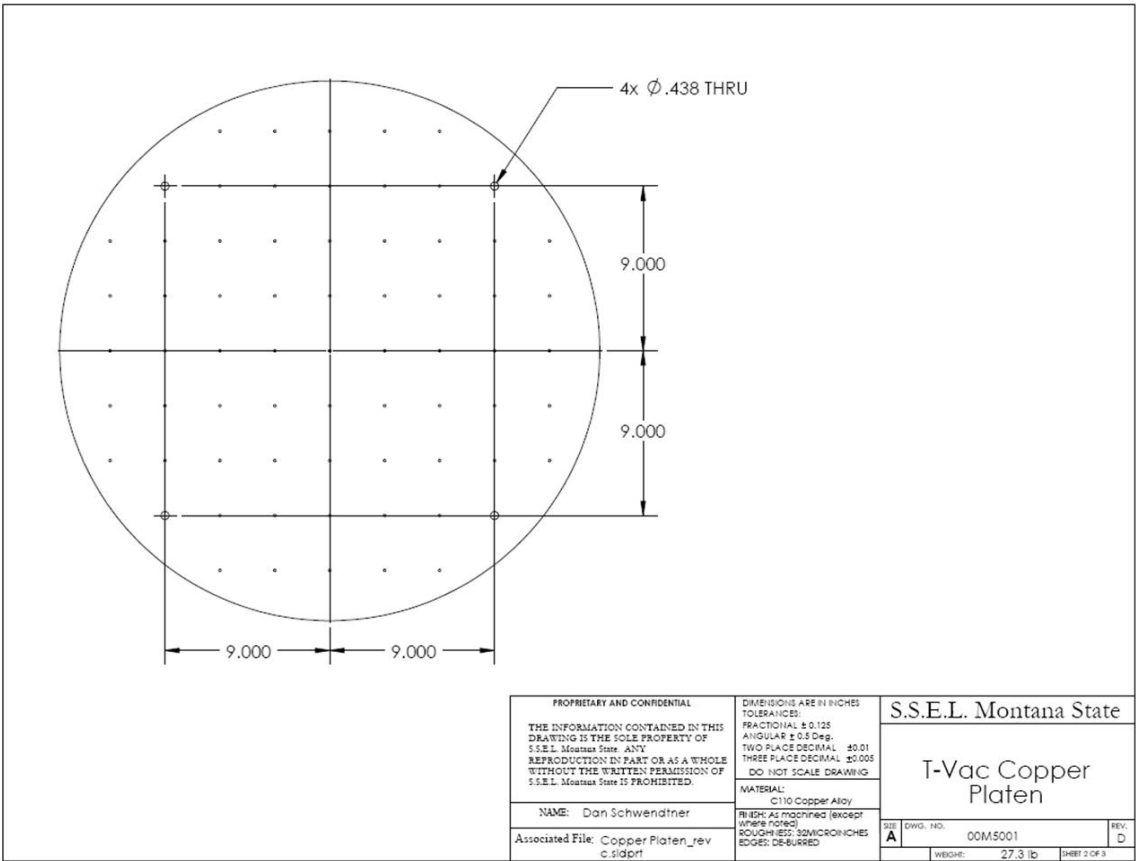


Figure A.6: Thermal Vacuum System Copper Platen Mechanical Drawing, 00M5001, Page 2 of 3

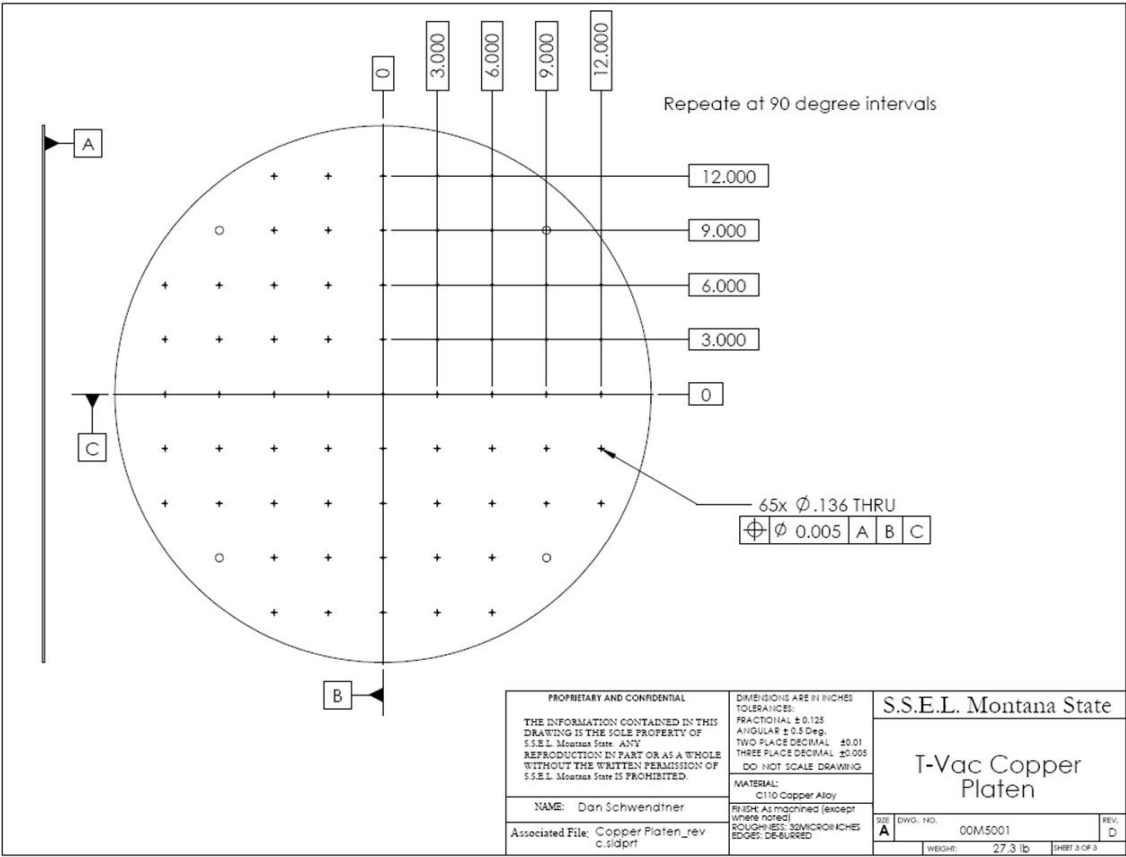


Figure A.7: Thermal Vacuum System Copper Platen Mechanical Drawing, 00M5001, Page 3 of 3

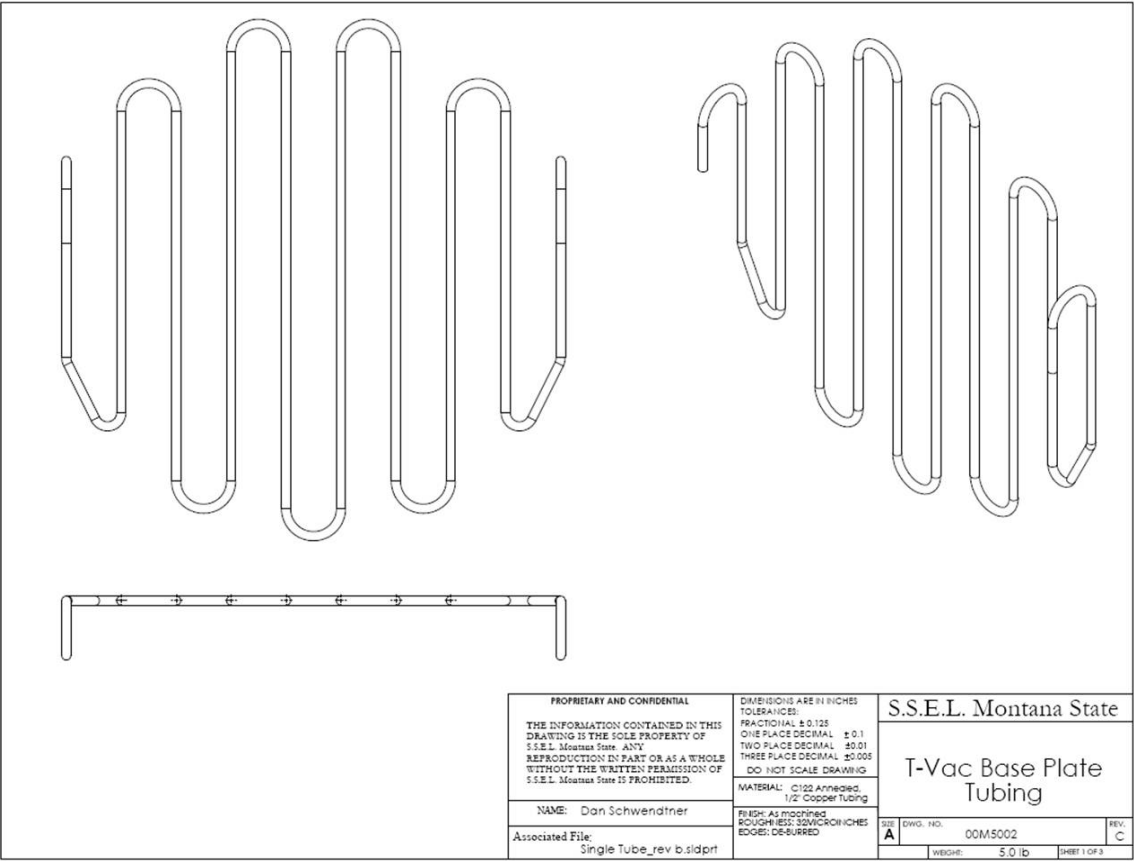


Figure A.8: Thermal Vacuum System Base Plate Tubing Mechanical Drawing, 00M5002, Page 1 of 3

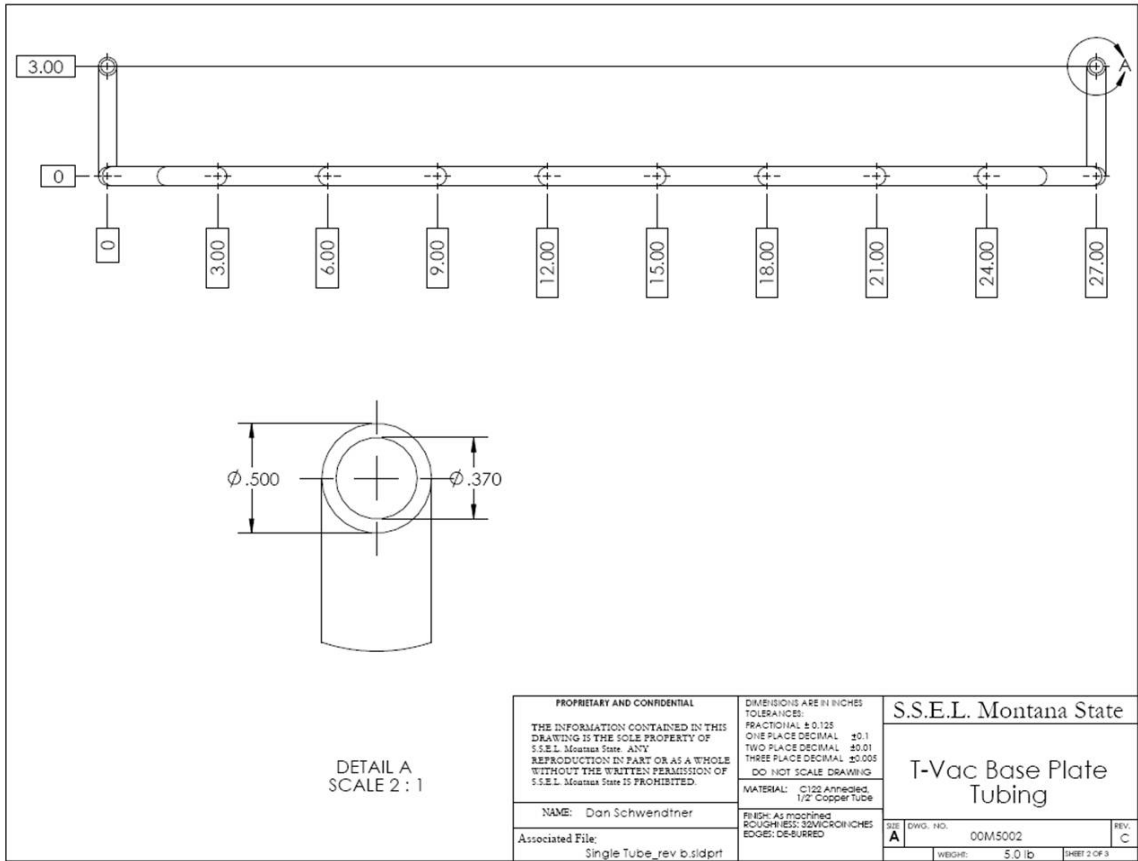


Figure A.9: Thermal Vacuum System Base Plate Tubing Mechanical Drawing, 00M5002, Page 2 of 3

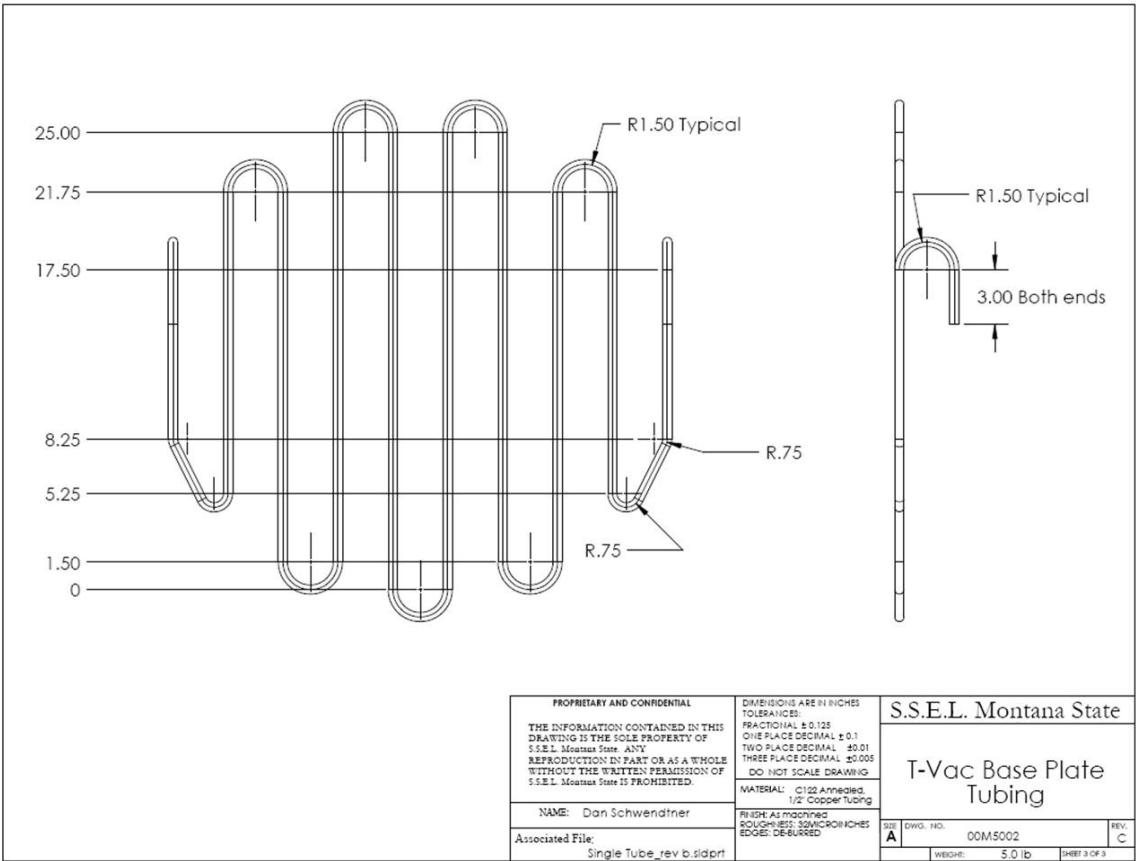


Figure A.10: Thermal Vacuum System Base Plate Tubing Mechanical Drawing, 00M5002, Page 3 of 3

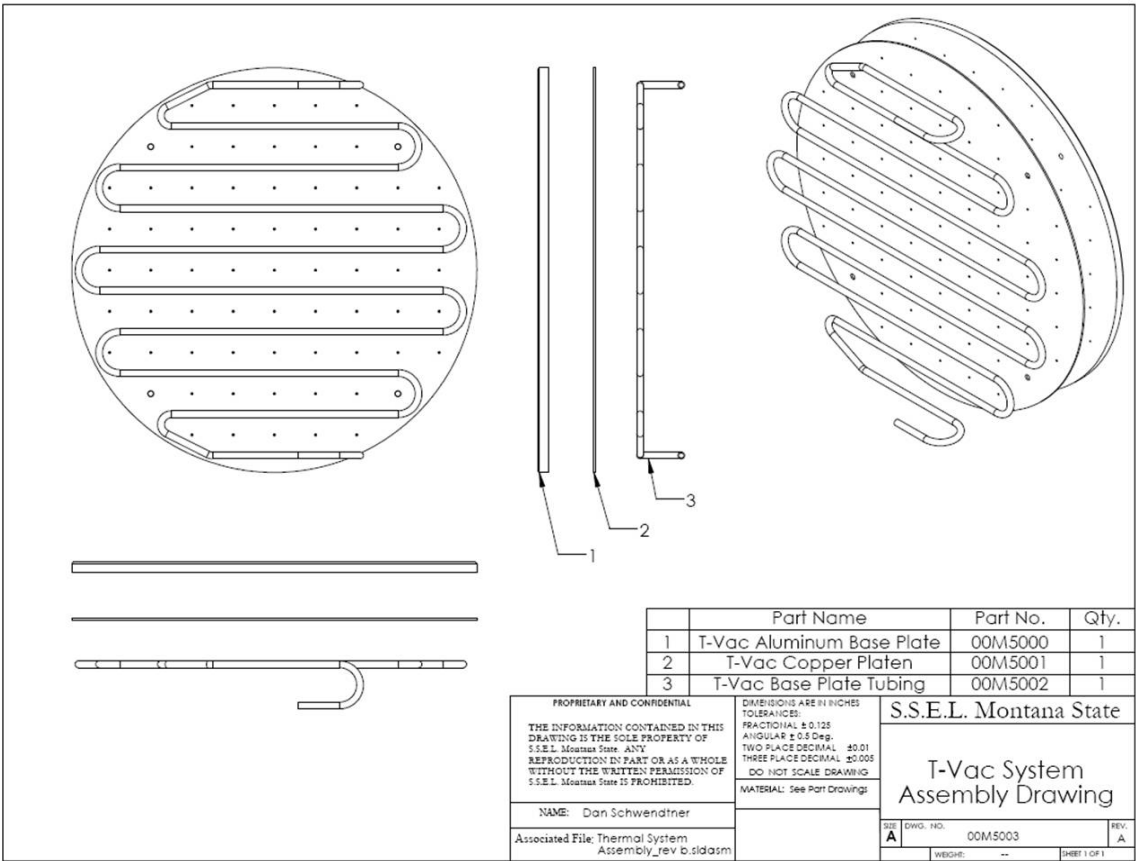


Figure A.11: Thermal Vacuum System Base Plate Assembly Drawing, 00M5003, Page 1 of 1

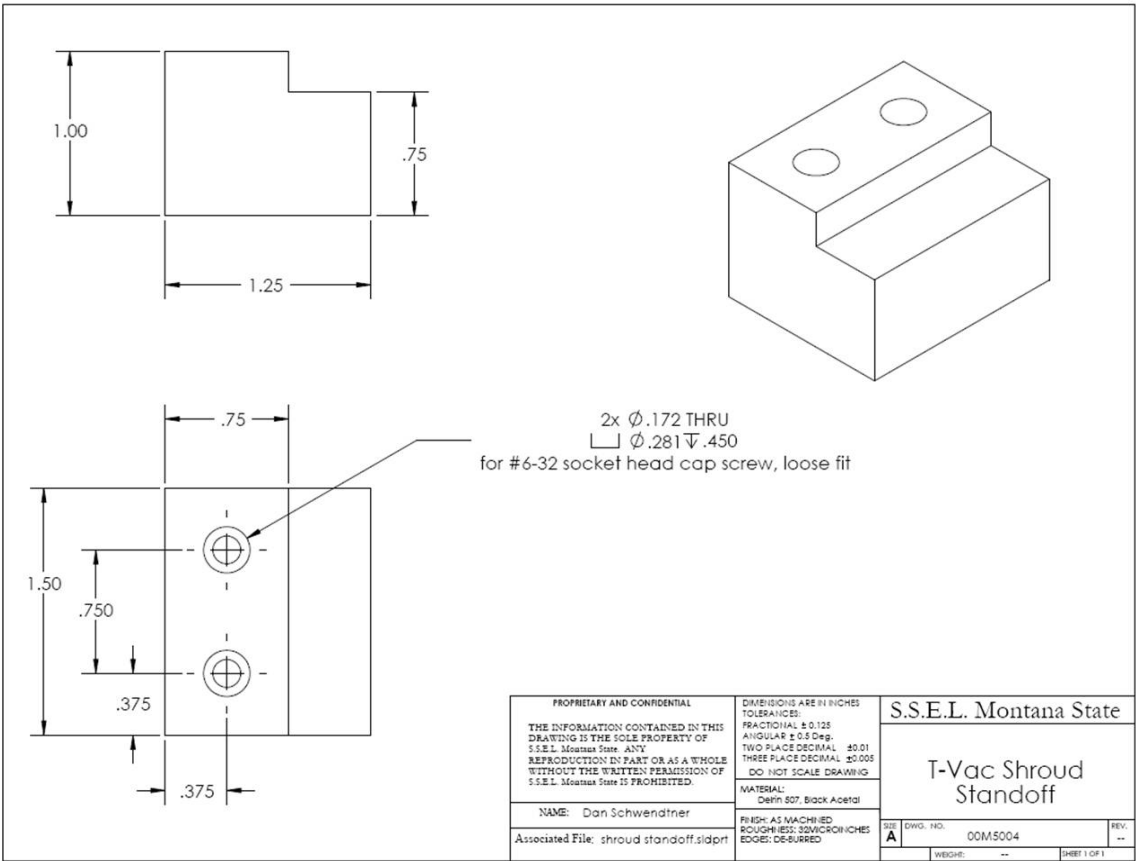


Figure A.12: Thermal Vacuum System Shroud Insulator Mechanical Drawing, 00M5004, Page 1 of 1

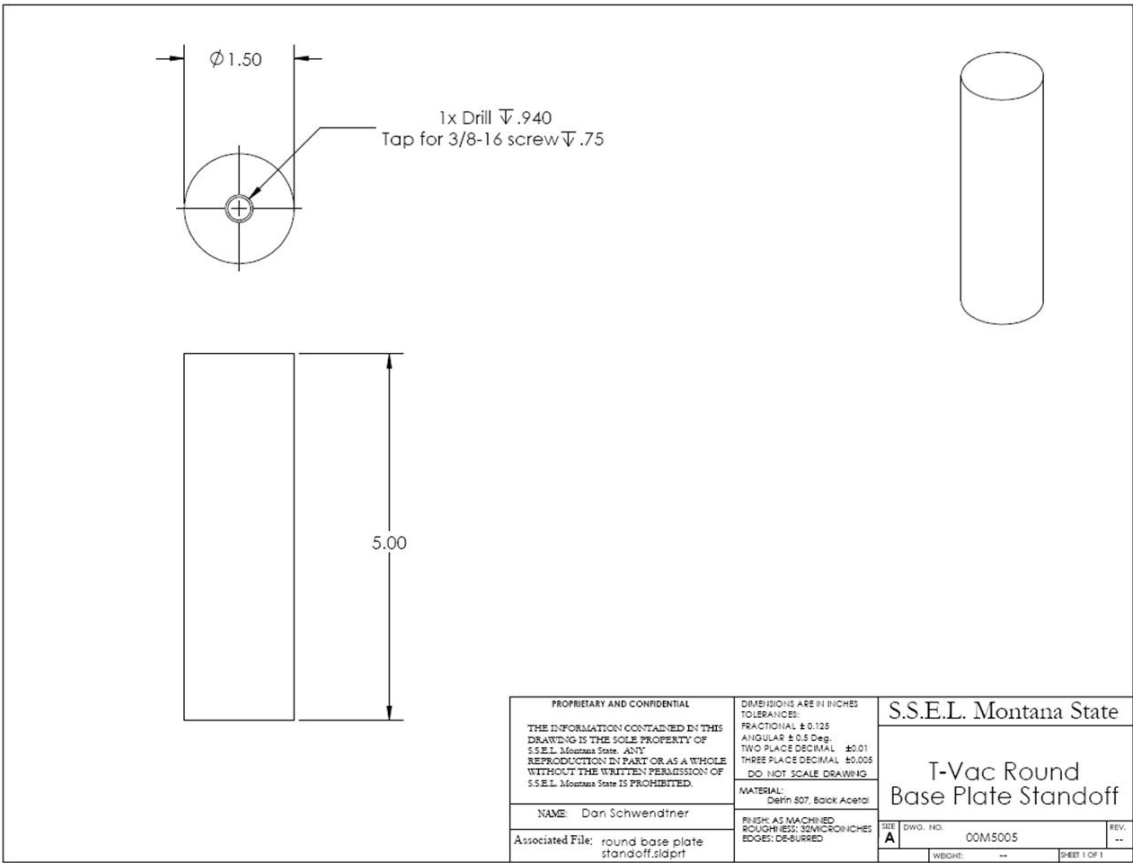


Figure A.13: Thermal Vacuum System Base Plate Insulator Mechanical Drawing, 00M5005, Page 1 of 1

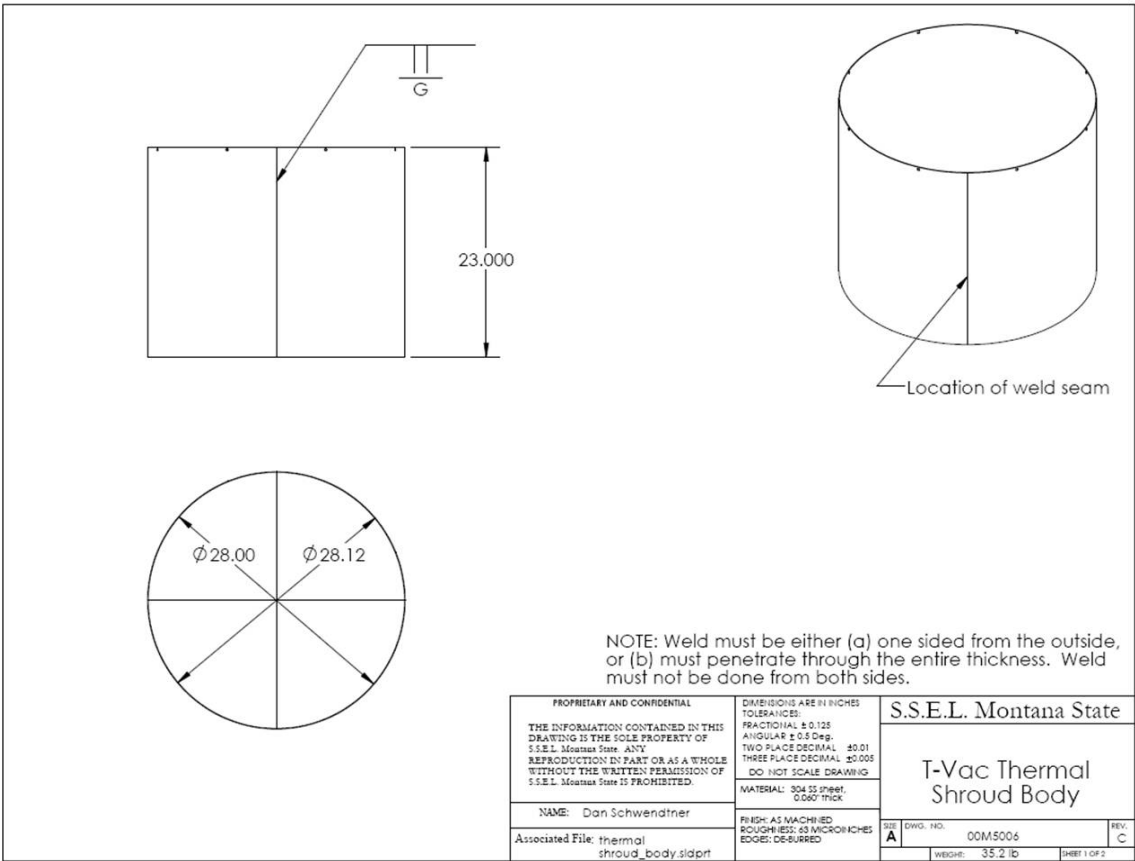


Figure A.14: Thermal Vacuum System Thermal Shroud Body Mechanical Drawing, 00M5006, Page 1 of 2

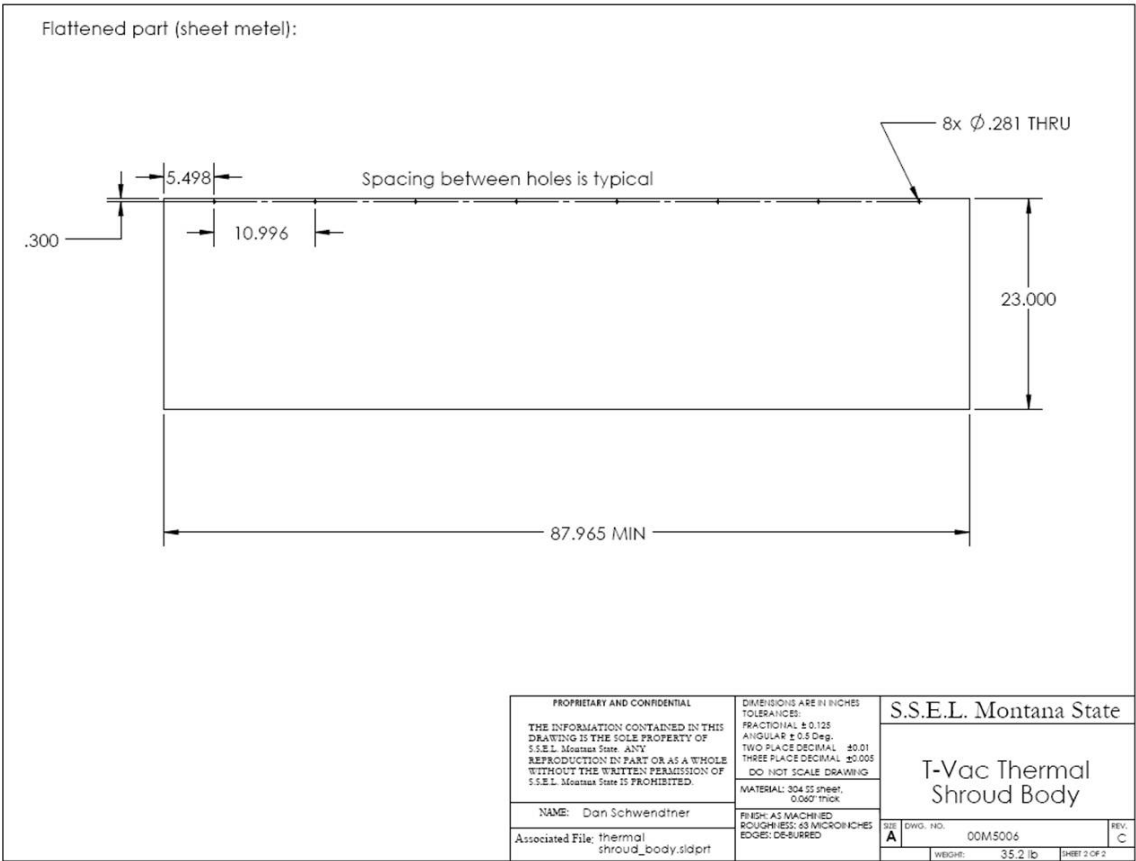


Figure A.15: Thermal Vacuum System Thermal Shroud Body Mechanical Drawing, 00M5006, Page 2 of 2

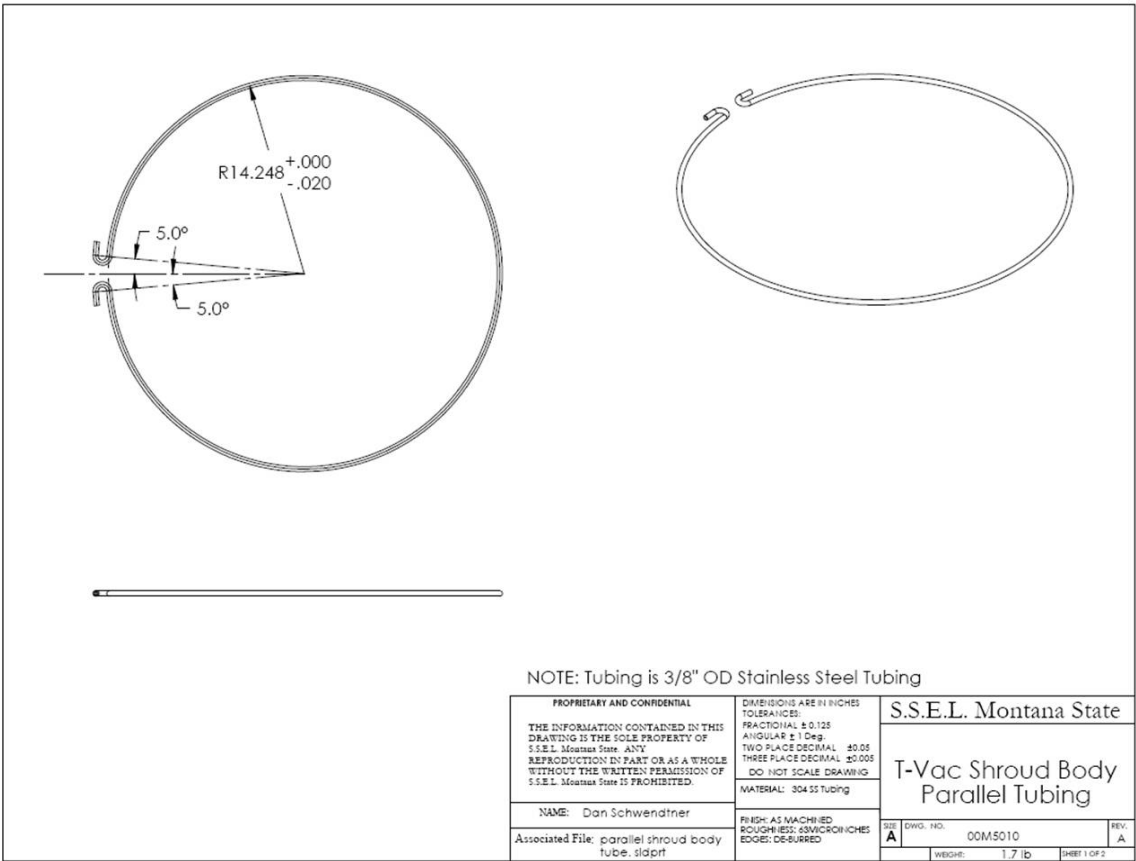


Figure A.16: Thermal Vacuum System Thermal Shroud Body Parallel Tubing Mechanical Drawing, 00M5010, Page 1 of 2

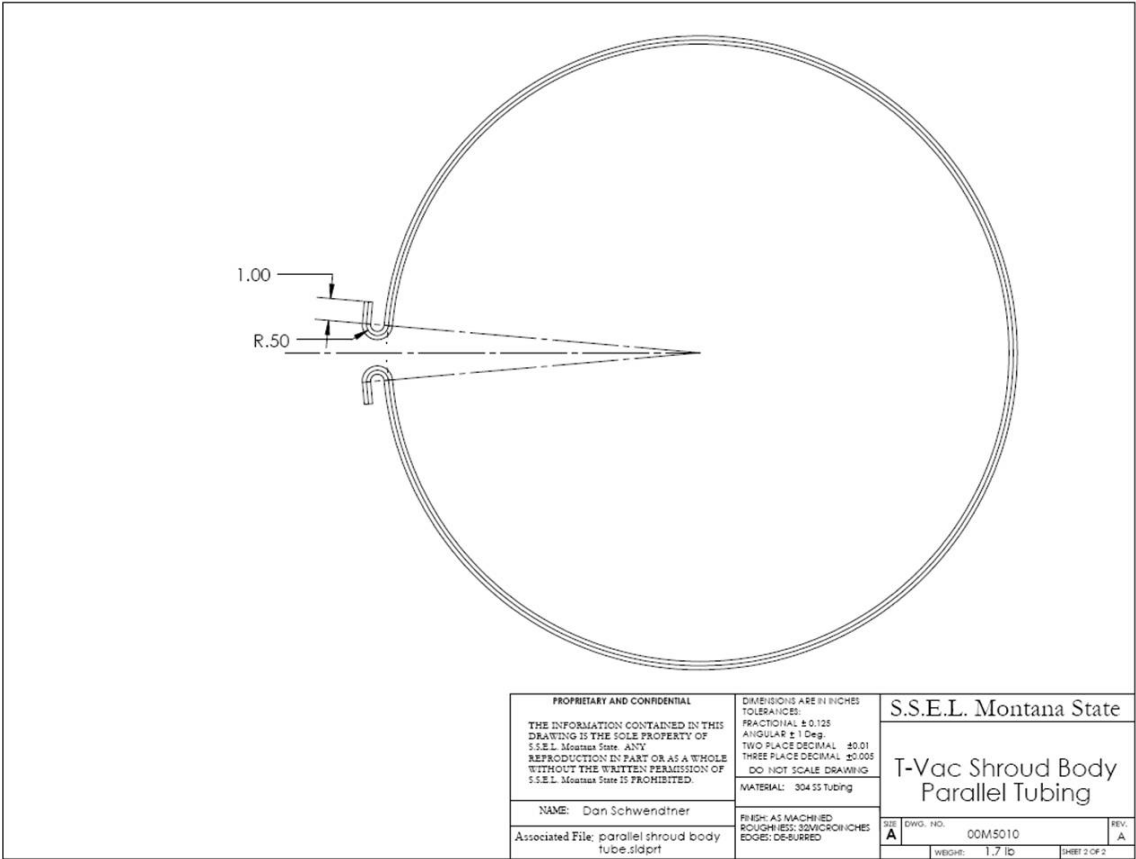


Figure A.17: Thermal Vacuum System Thermal Shroud Body Parallel Tubing Mechanical Drawing, 00M5010, Page 2 of 2

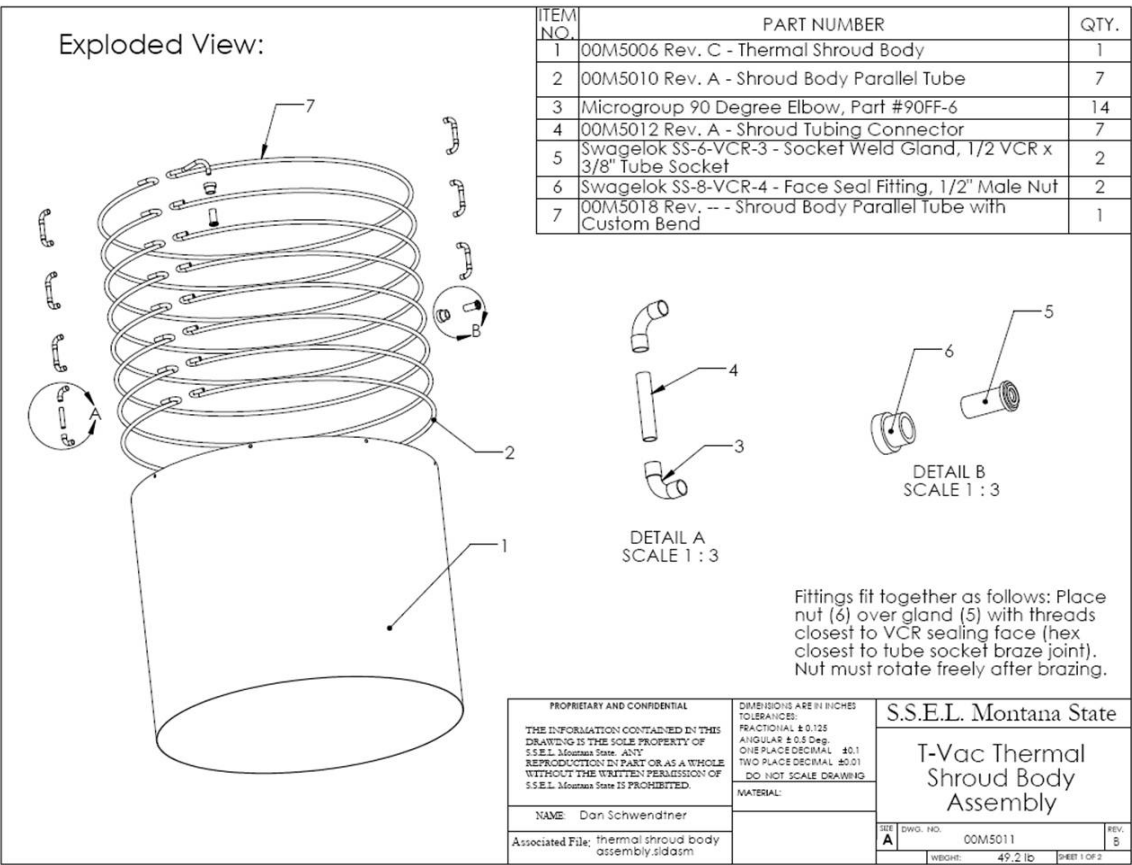


Figure A.18: Thermal Vacuum System Thermal Shroud Body Assembly Drawing, 00M5011, Page 1 of 2

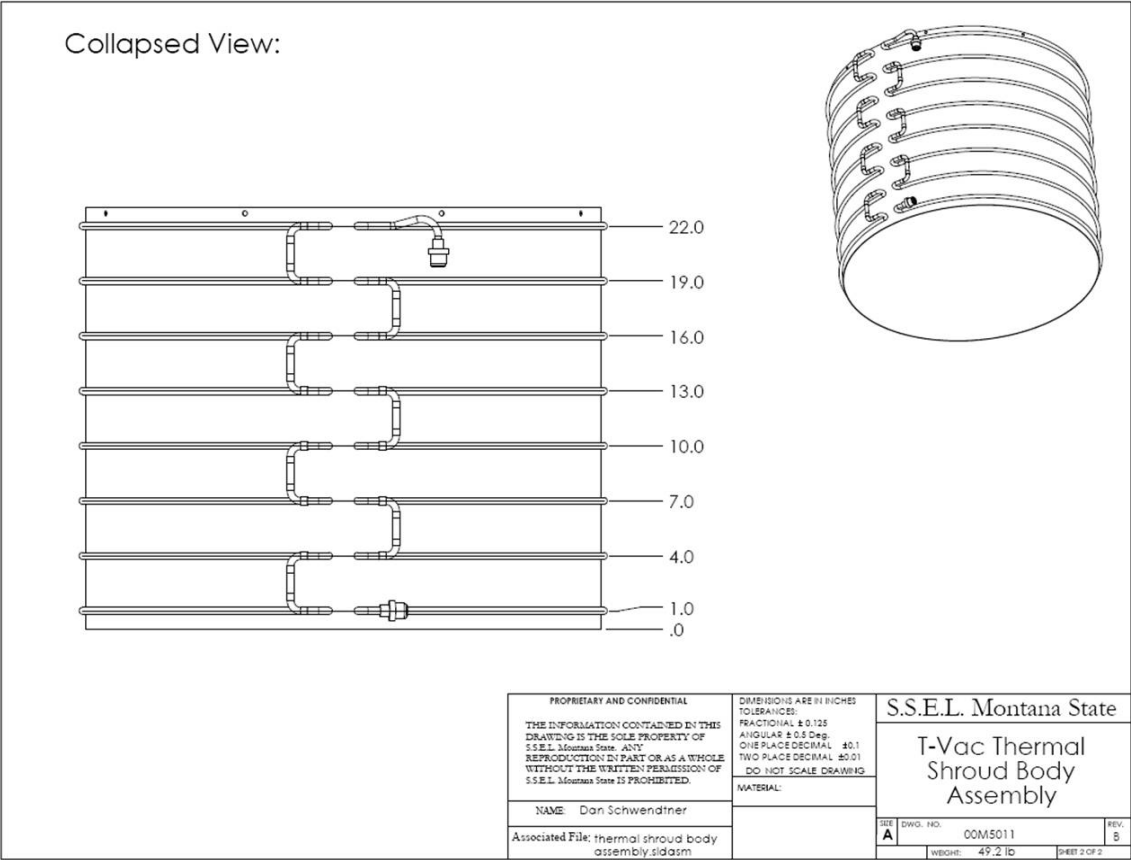


Figure A.19: Thermal Vacuum System Thermal Shroud Body Assembly Drawing, 00M5011, Page 2 of 2

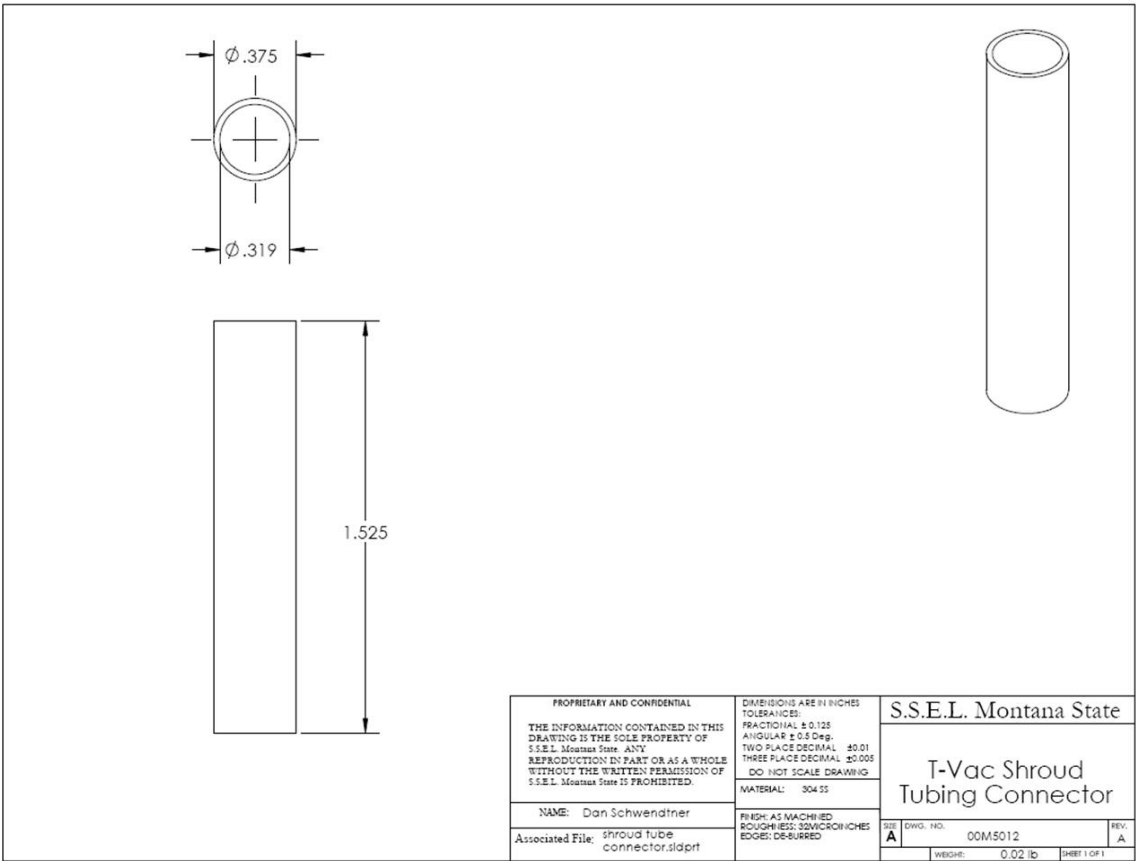


Figure A.20: Thermal Vacuum System Thermal Shroud Tubing Connector Mechanical Drawing, 00M5012, Page 1 of 1

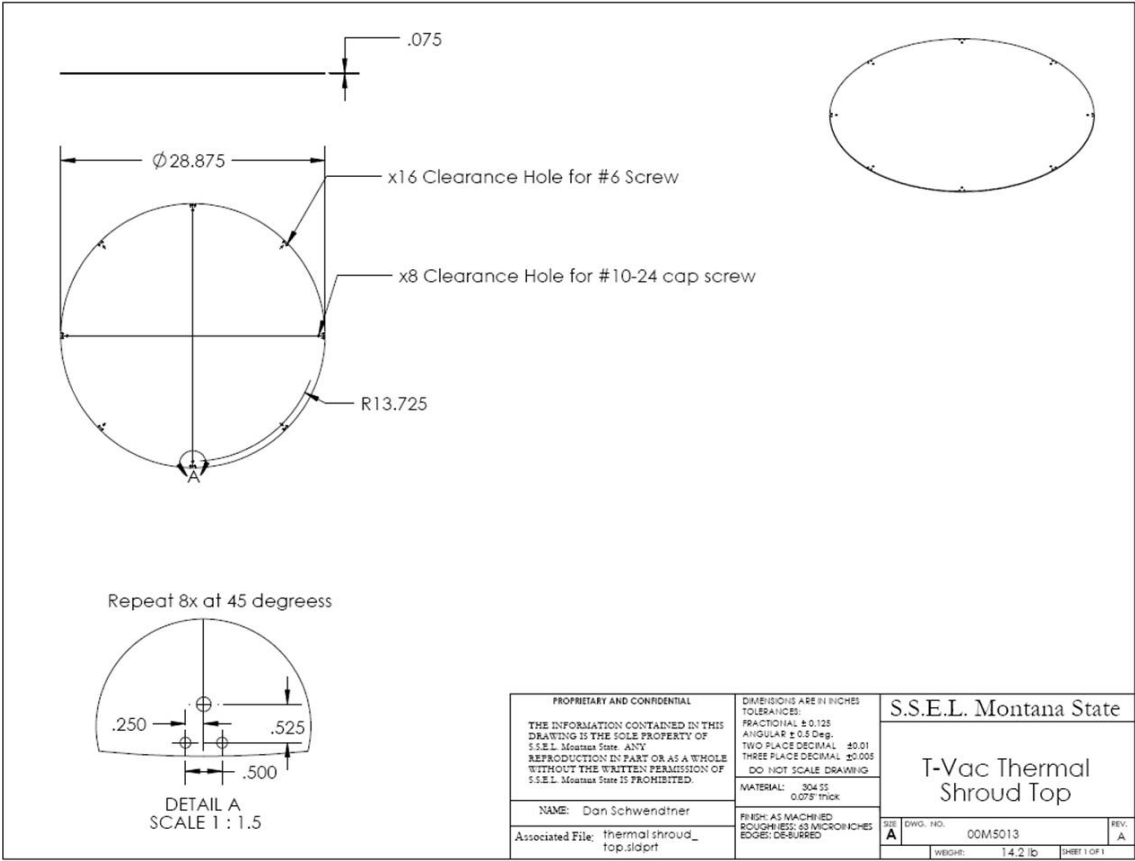


Figure A.21: Thermal Vacuum System Thermal Shroud Top Plate Mechanical Drawing, 00M5013, Page 1 of 1

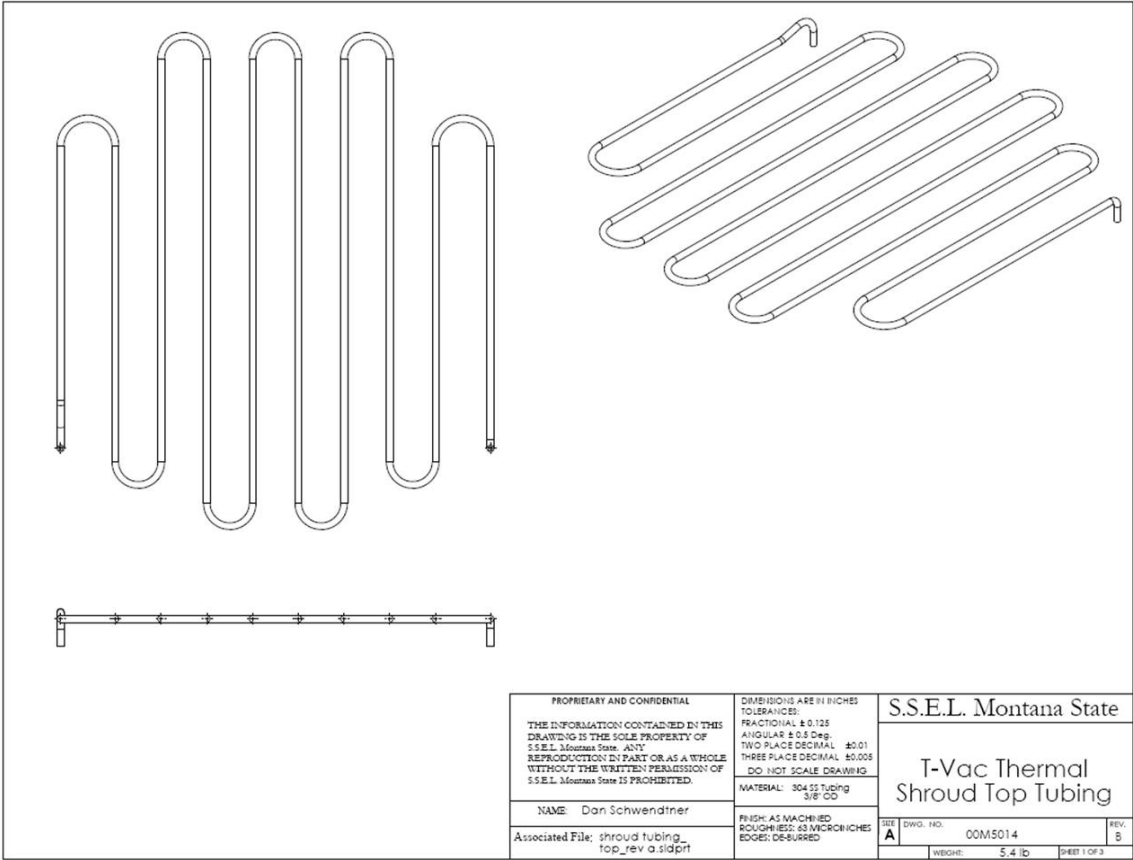


Figure A.22: Thermal Vacuum System Thermal Shroud Top Tubing Mechanical Drawing, 00M5014, Page 1 of 3

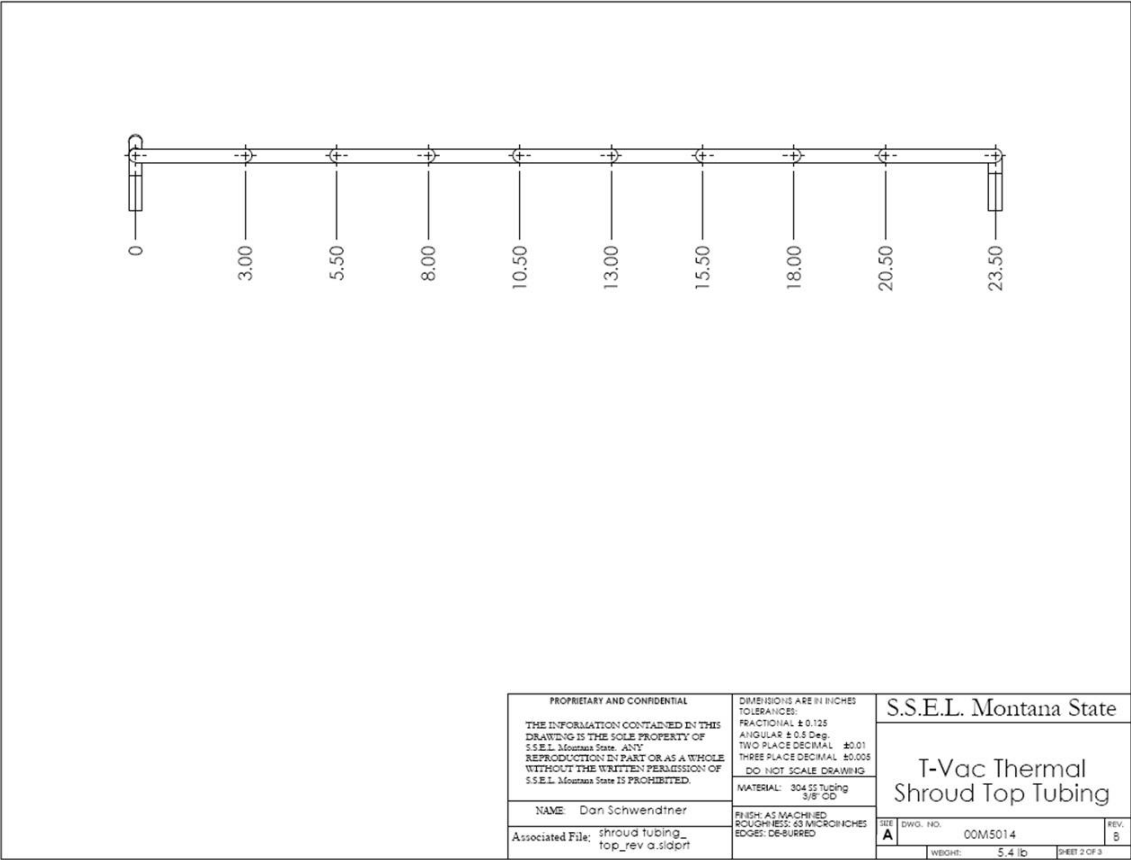


Figure A.23: Thermal Vacuum System Thermal Shroud Top Tubing Mechanical Drawing, 00M5014, Page 2 of 3

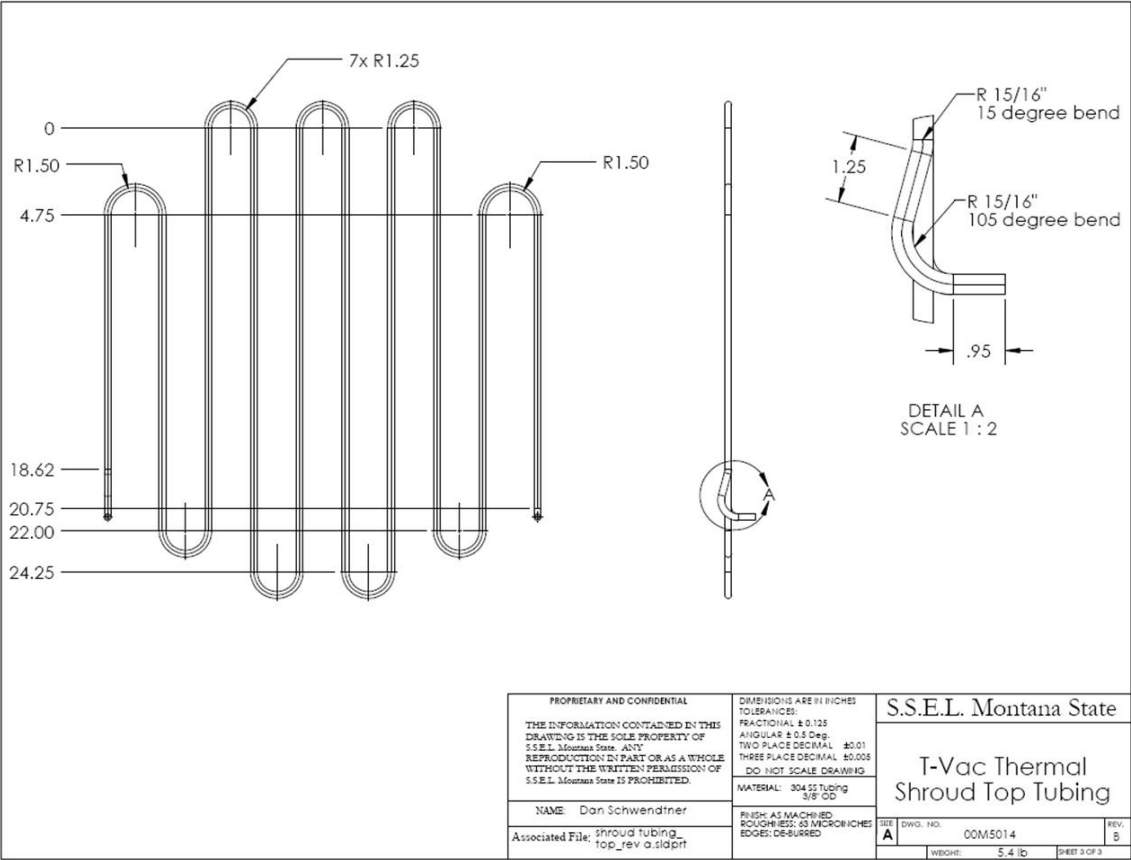


Figure A.24: Thermal Vacuum System Thermal Shroud Top Tubing Mechanical Drawing, 00M5014, Page 3 of 3

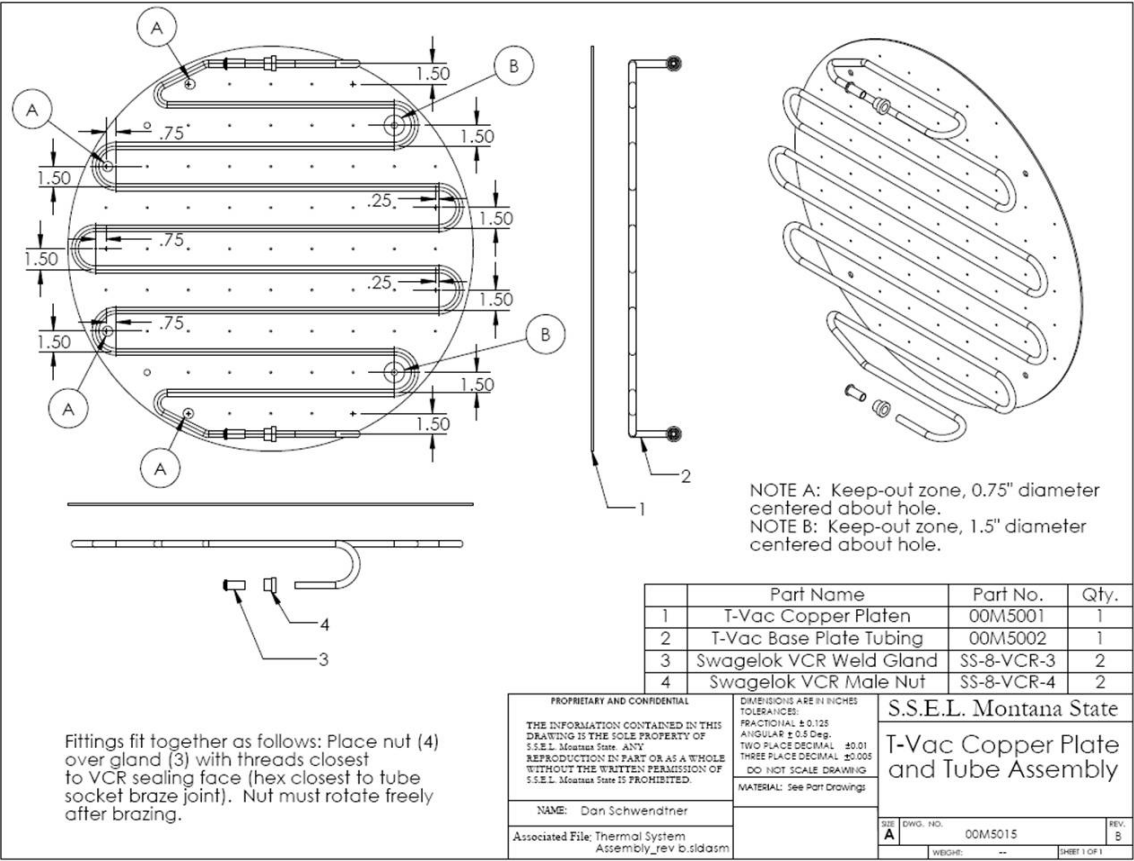


Figure A.25: Thermal Vacuum System Copper Plate and Tube Assembly Drawing, 00M5015, Page 1 of 1

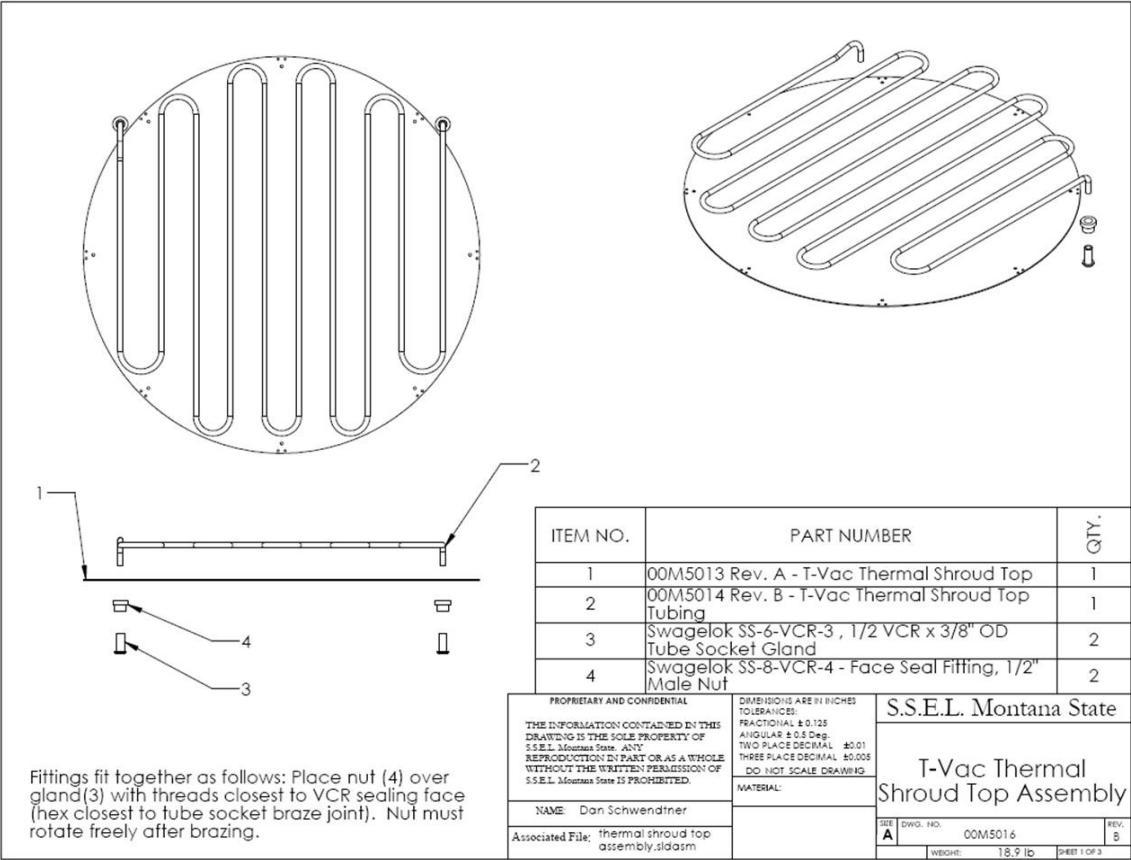


Figure A.26: Thermal Vacuum System Thermal Shroud Top Assembly Drawing, 00M5016, Page 1 of 3

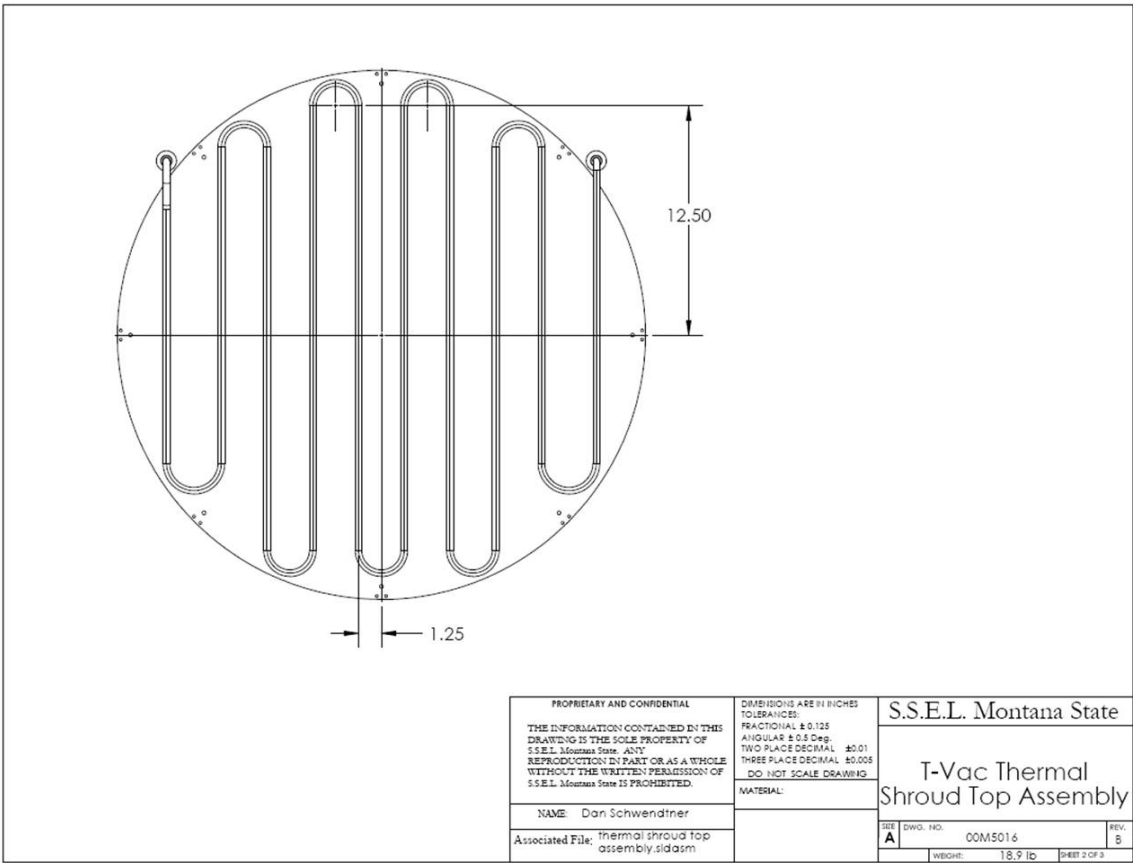


Figure A.27: Thermal Vacuum System Thermal Shroud Top Assembly Drawing, 00M5016, Page 2 of 3

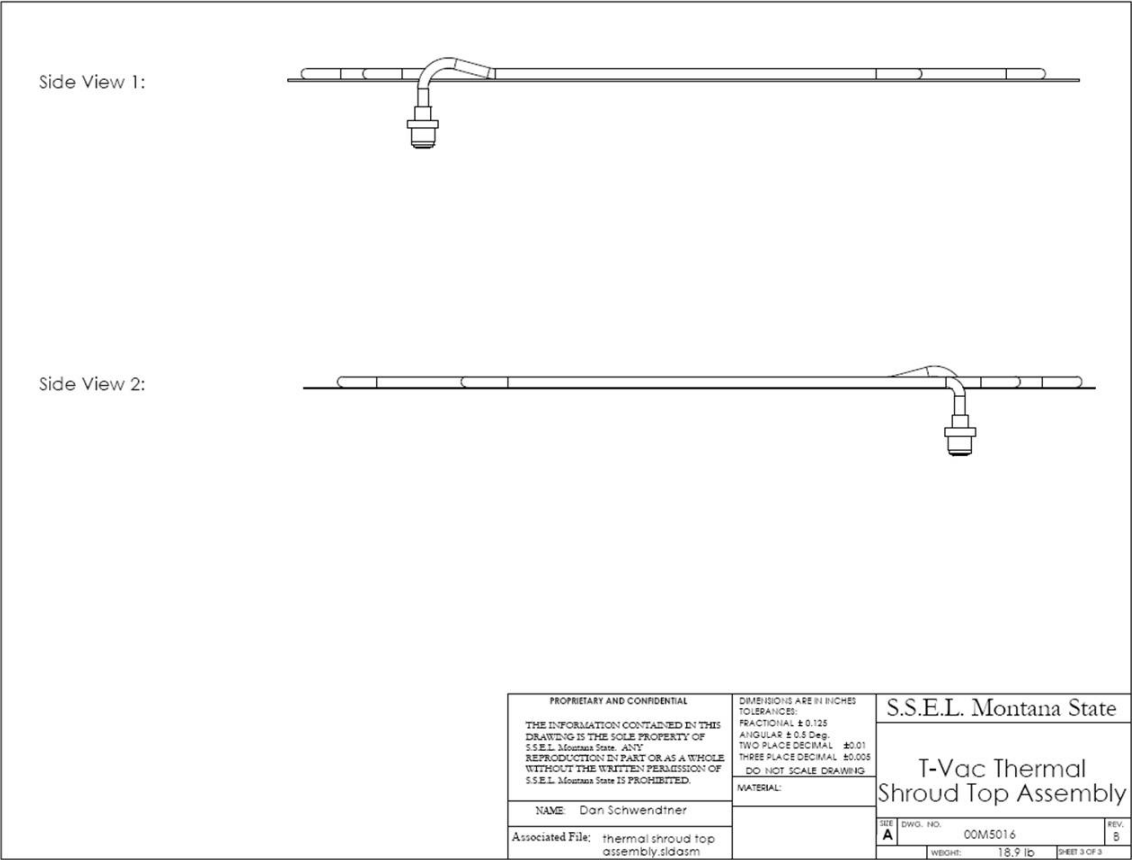


Figure A.28: Thermal Vacuum System Thermal Shroud Top Assembly Drawing, 00M5016, Page 3 of 3

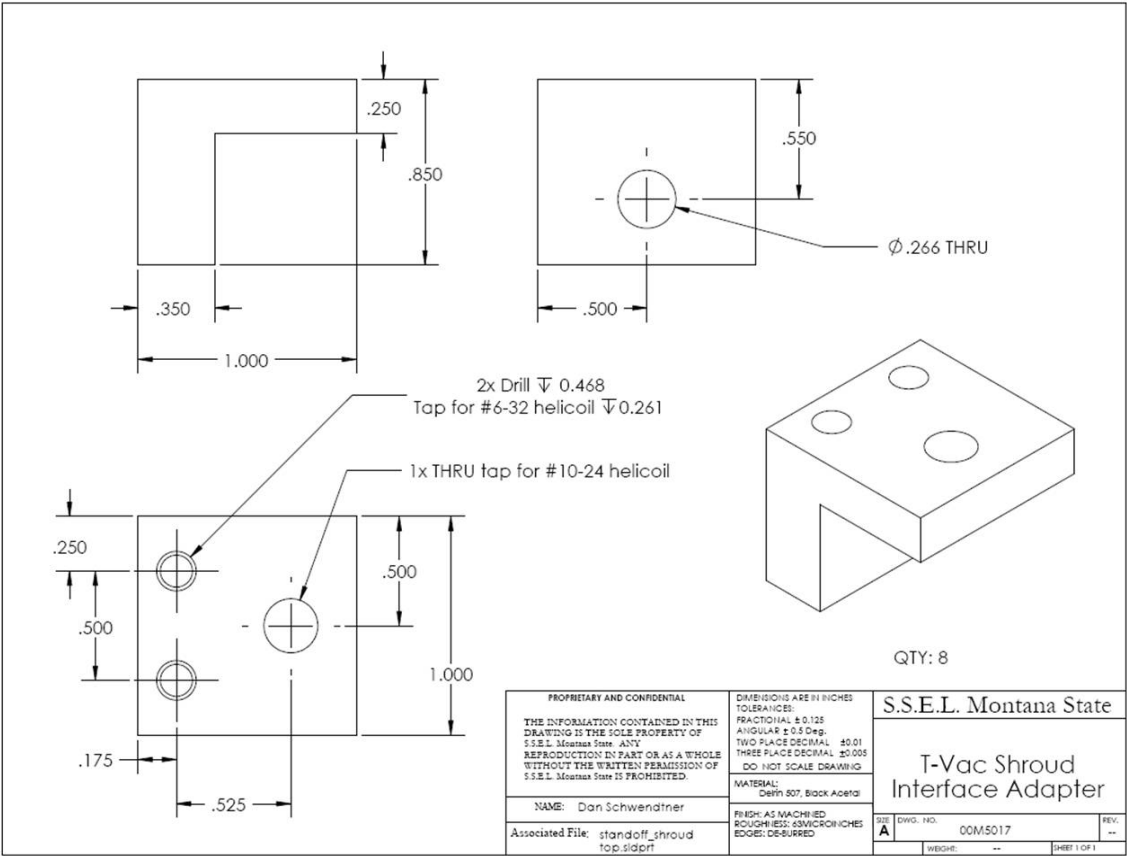


Figure A.29: Thermal Vacuum System Shroud Top Interface Adapter Mechanical Drawing, 00M5017, Page 1 of 1

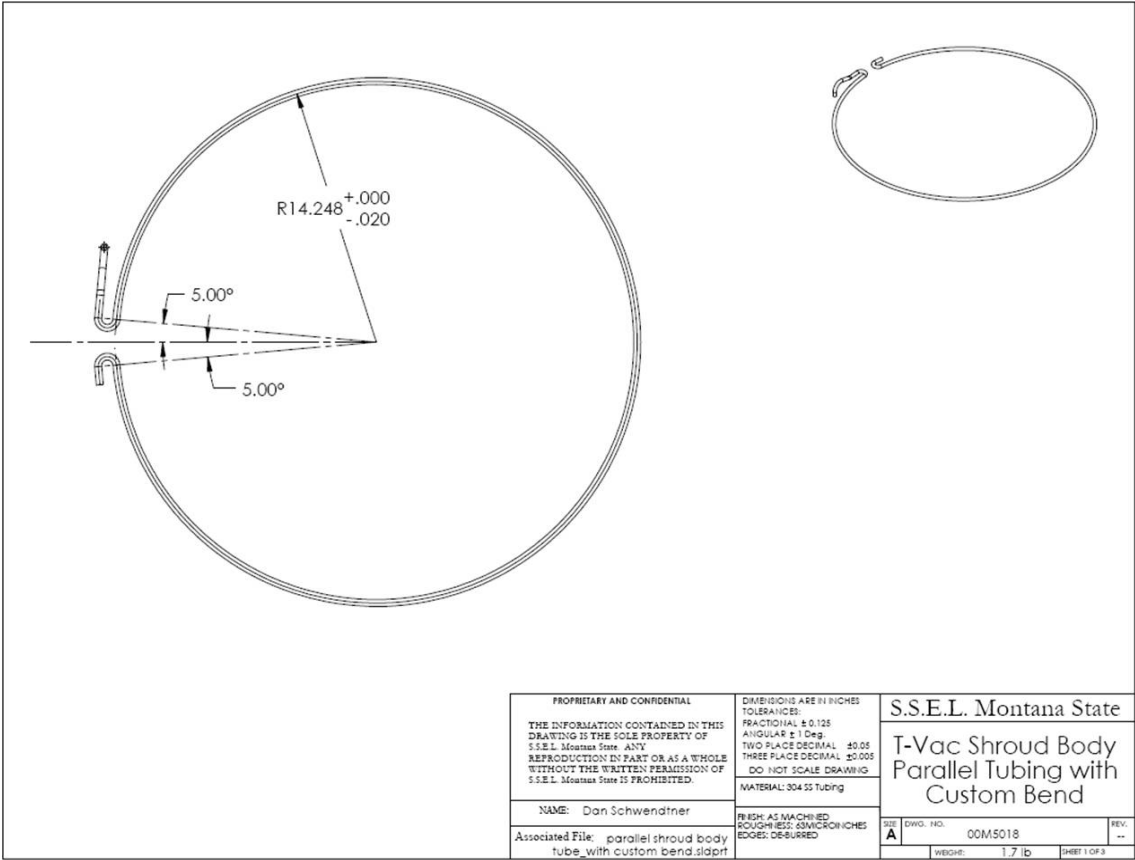


Figure A.30: Thermal Vacuum System Shroud Body Parallel Tubing with Custom Bend Mechanical Drawing, 00M5018, Page 1 of 3

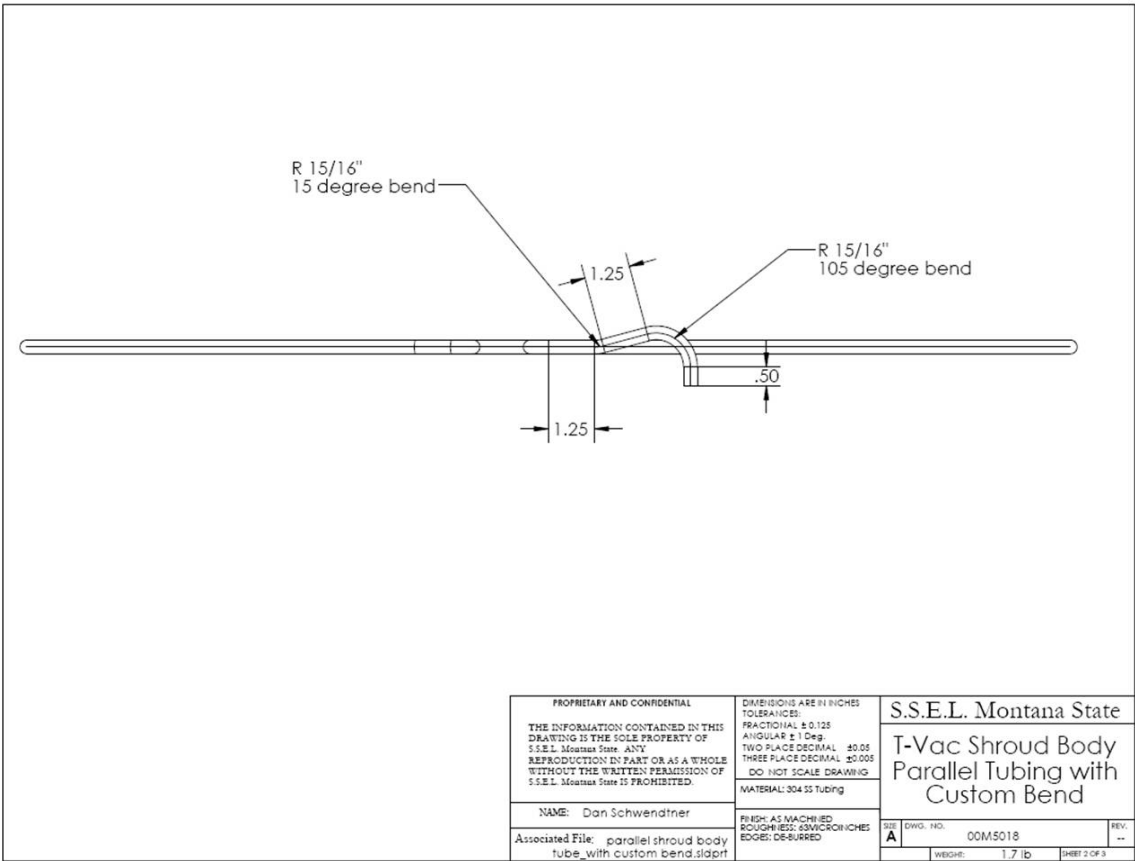


Figure A.31: Thermal Vacuum System Shroud Body Parallel Tubing with Custom Bend Mechanical Drawing, 00M5018, Page 2 of 3

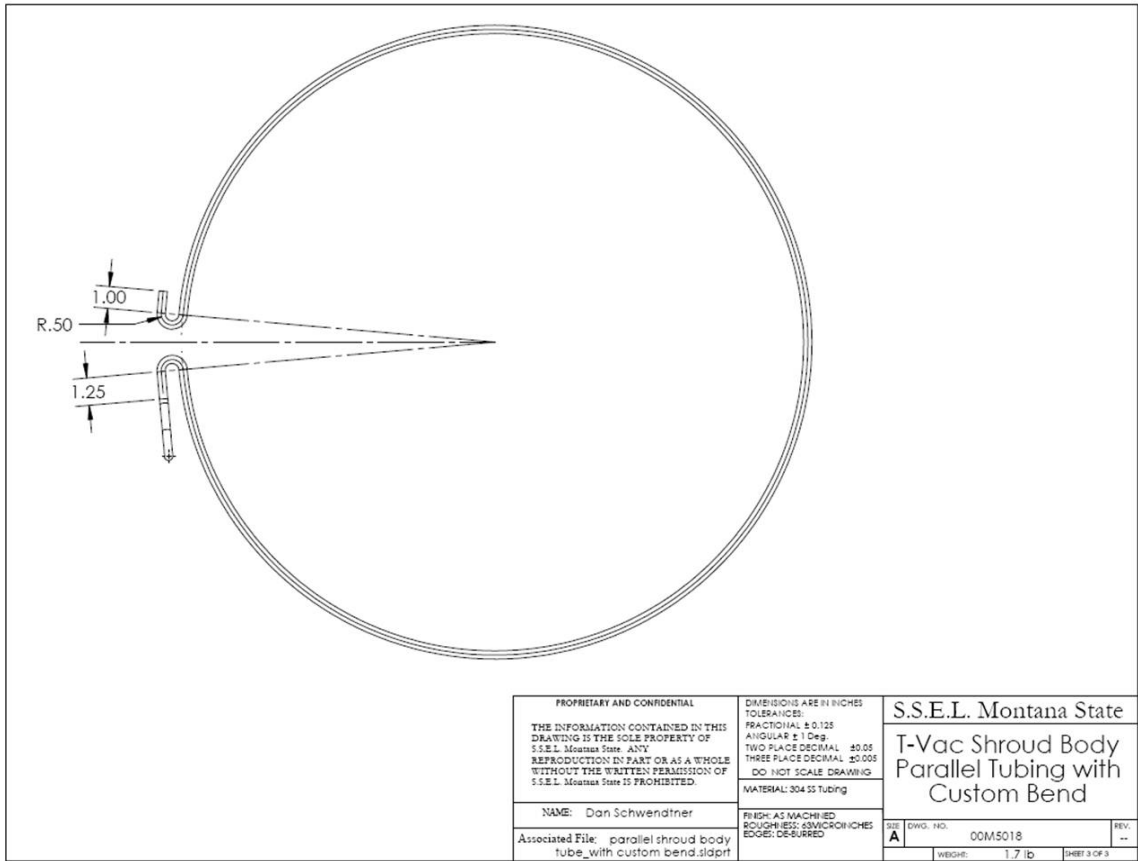


Figure A.32: Thermal Vacuum System Shroud Body Parallel Tubing with Custom Bend Mechanical Drawing, 00M5018, Page 3 of 3

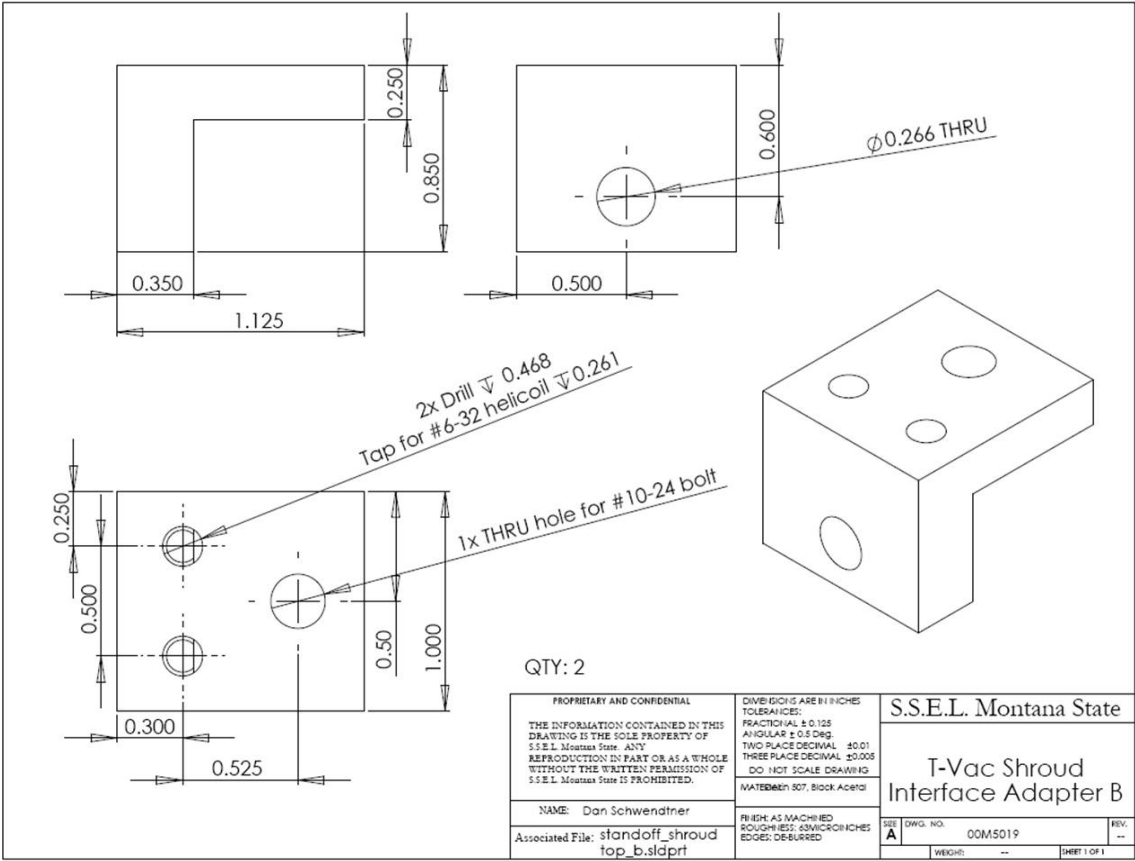


Figure A.33: Thermal Vacuum System Shroud Top Interface Adapter B Mechanical Drawing, 00M5019, Page 1 of 1

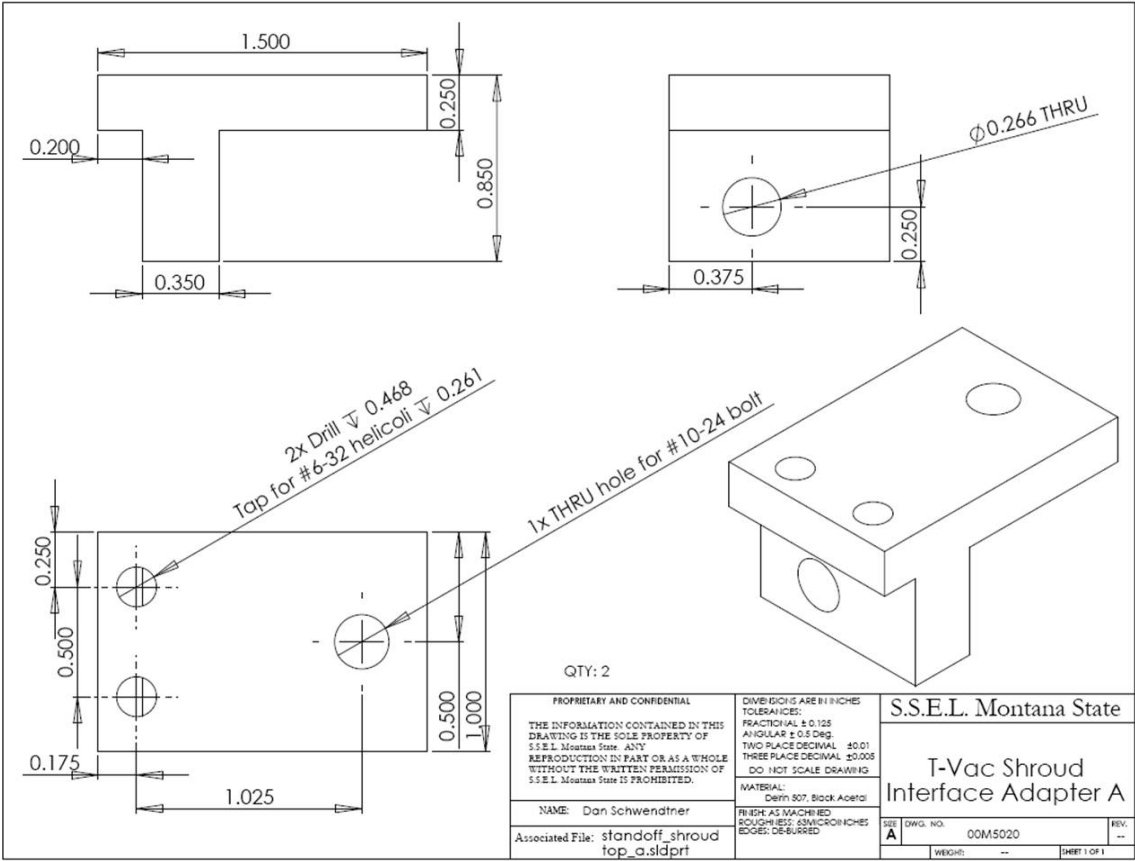


Figure A.34: Thermal Vacuum System Shroud Top Interface Adapter A Mechanical Drawing, 00M5020, Page 1 of 1

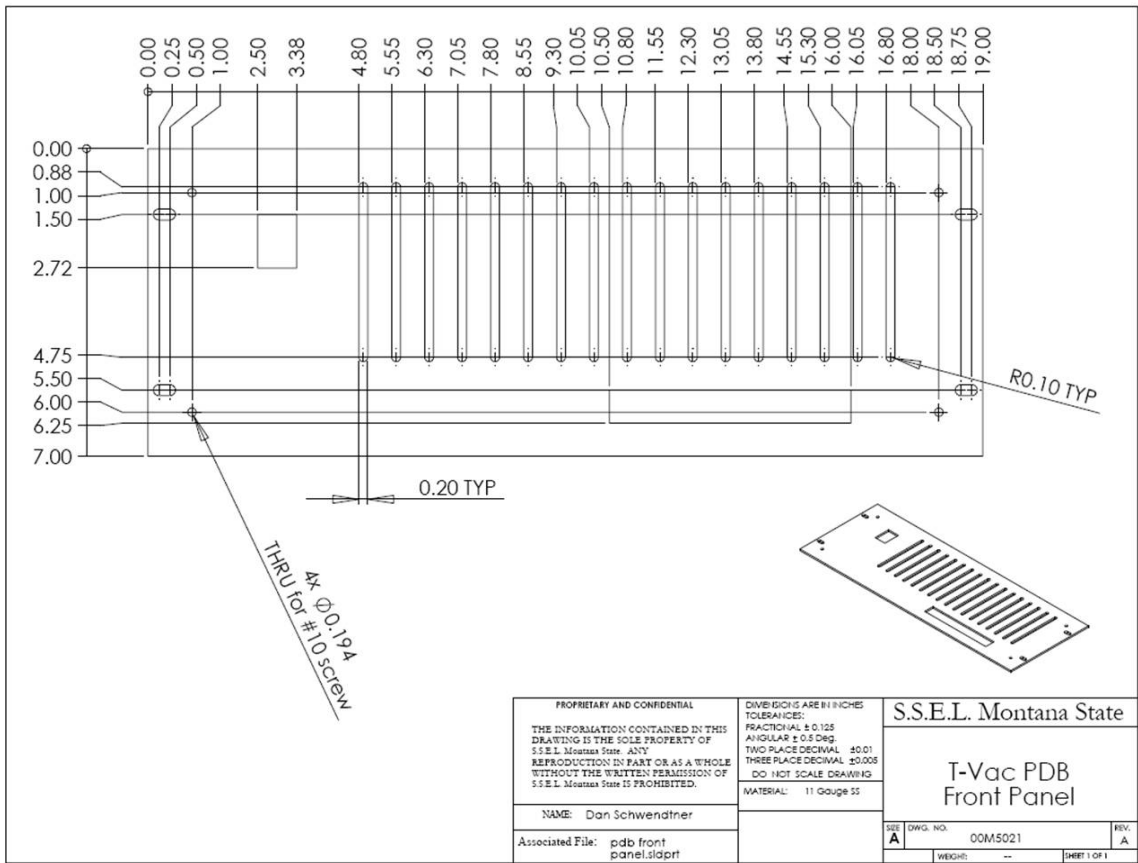


Figure A.35: Thermal Vacuum System PDB Front Panel Mechanical Drawing, 00M5021, Page 1 of 1

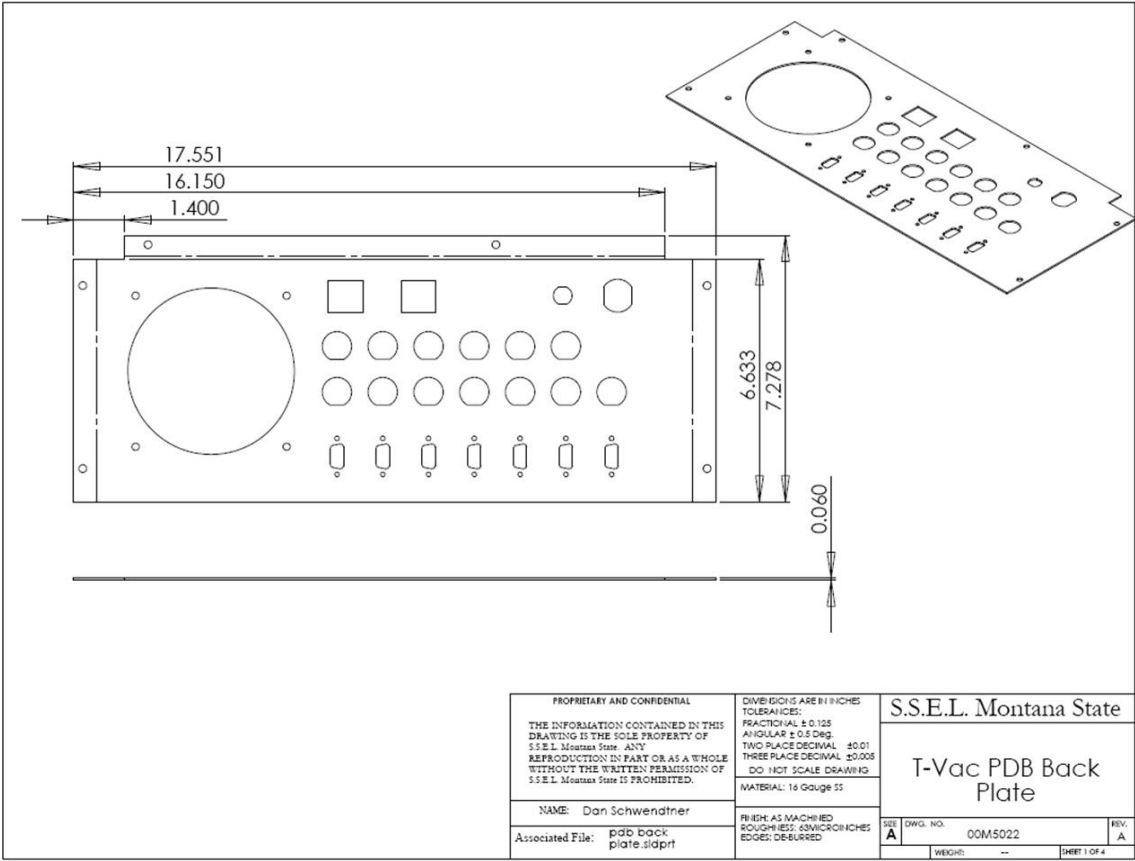


Figure A.36: Thermal Vacuum System PDB Back Panel Mechanical Drawing, 00M5022, Page 1 of 4

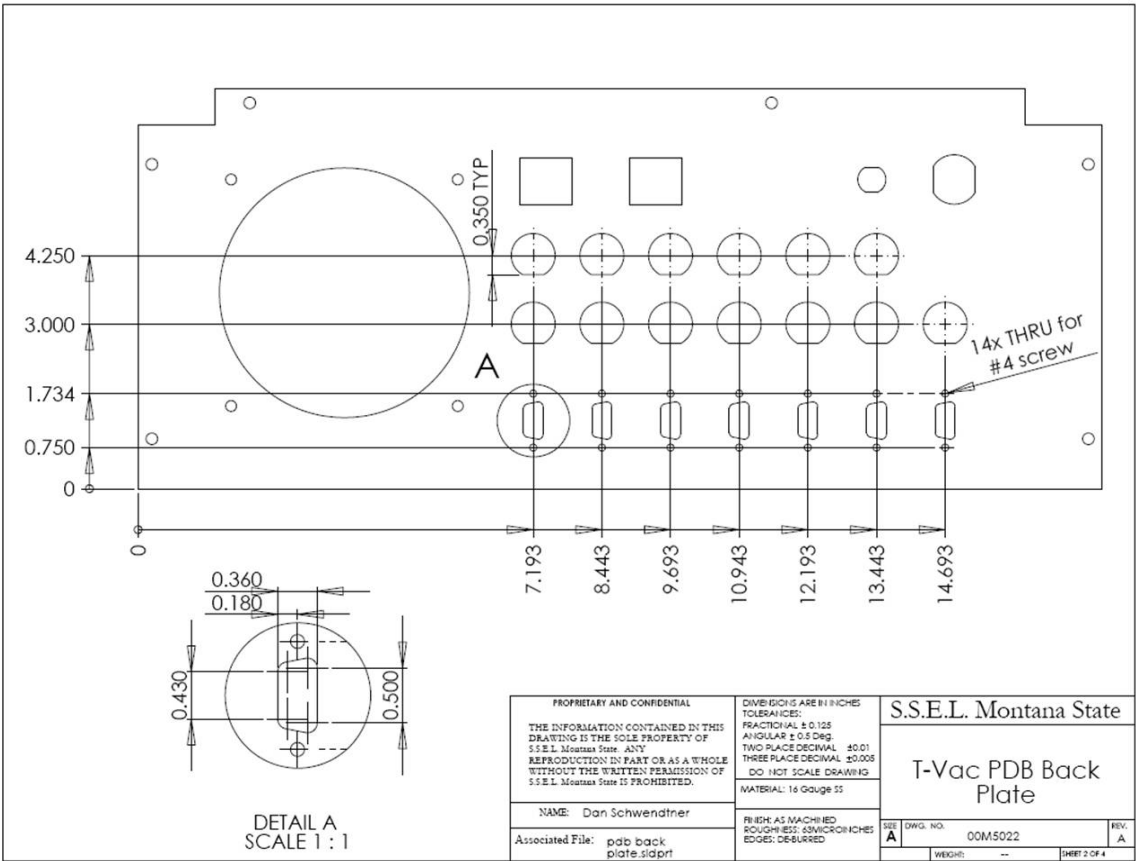


Figure A.37: Thermal Vacuum System PDB Back Panel Mechanical Drawing, 00M5022, Page 2 of 4

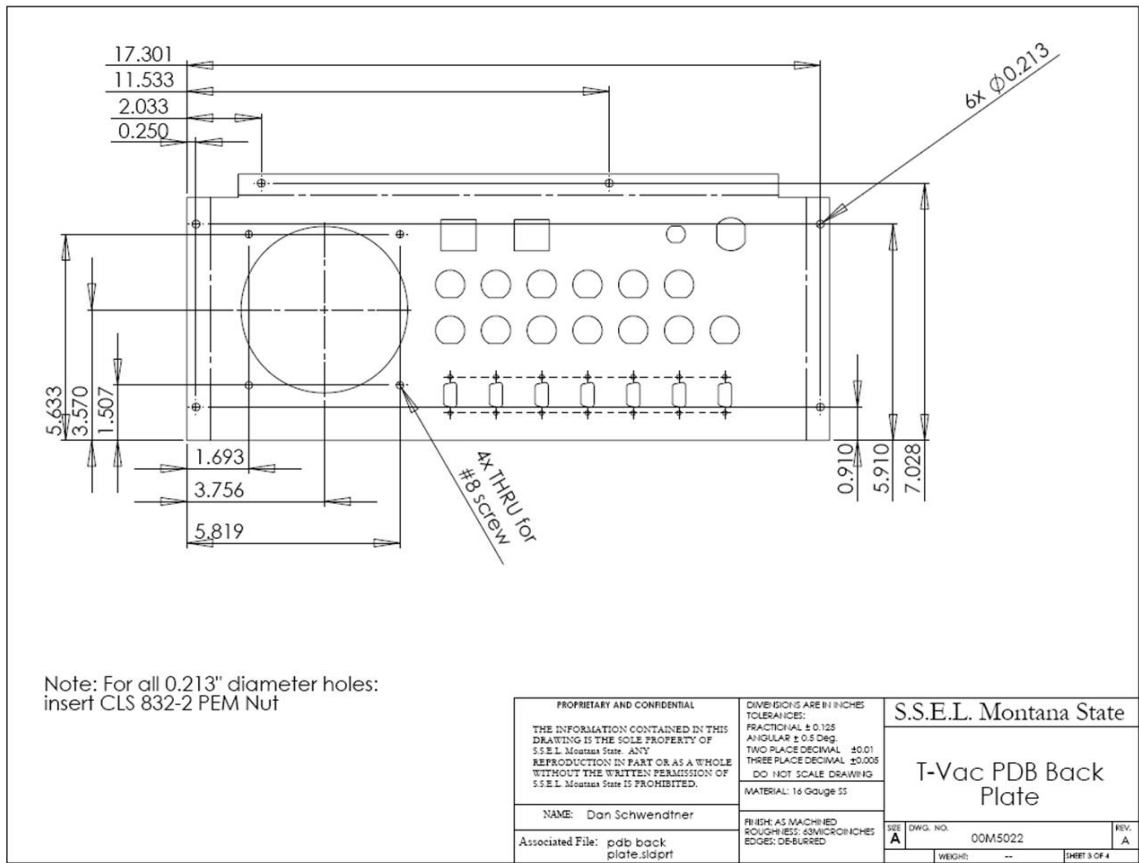


Figure A.38: Thermal Vacuum System PDB Back Panel Mechanical Drawing, 00M5022, Page 3 of 4

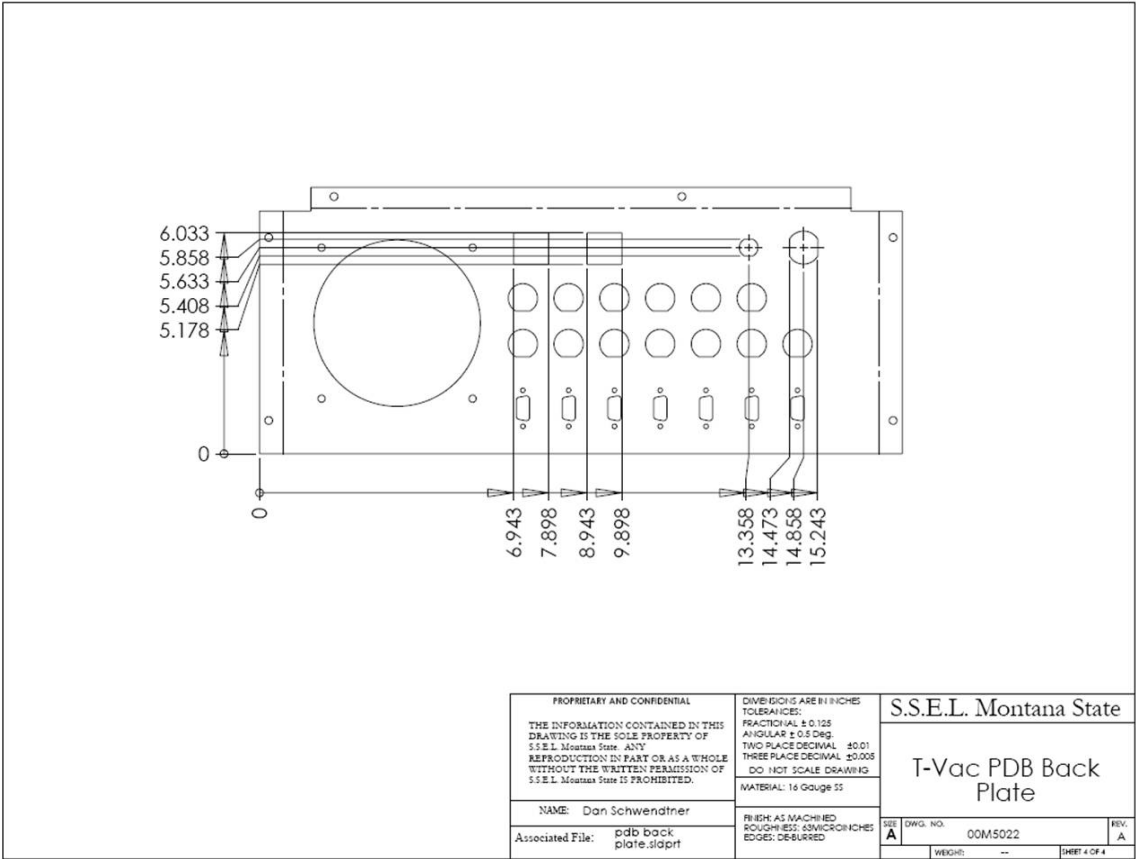


Figure A.39: Thermal Vacuum System PDB Back Panel Mechanical Drawing, 00M5022, Page 4 of 4

APPENDIX B

REQUIREMENTS VERIFICATION

Table B.1: Requirements Verification of the Vacuum System

Requirement No.	Verification Method	Section Reference
1	Design	3
2	Design	3
3	Test	3, 6
4	Design	3
5	Design	3
6	Design	3

Table B.2: Requirements Verification of the Thermal System

Requirement No.	Verification Method	Section Reference
1	Test	6
2	Test	6
3	Design, Test	4, 5, 6
4	Design	4
5	Design	4, 5
6	Design, Test	4, 5
7	Design, Test	6
8	Analysis, Test	6
9	Analysis, Test	6
10	Design	4, 6
11	Design	4, 5
12	Design	4
13	Design	4, 5
14	Design	4
15	Design	6
16	Design	6

Table B.3: Requirements Verification of the Base Plate

Requirement No.	Verification Method	Section Reference
1	Test	6
2	Design, Test	4, 6
3	Design	5
4	Analysis	4
5	Analysis	4
6	Analysis	4
7	Design	4
8	Design	4
9	Design	4
10	Design	4

Table B.4: Requirements Verification of the Thermal Shroud

Requirement No.	Verification Method	Section Reference
1	Design, Test	6
2	Design, Test	6
3	Design, Test	6
4	Design	5
5	Design	5
6	Design	5
7	Design, Analysis	4, 5
8	Design, Analysis	5
9	Design	5
10	Design	5
11	Design	5
12	Design	5
13	Design	5