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THESIS

**A SYSTEMS ENGINEERING APPROACH
FOR THE SELECTION OF A HEAT EXCHANGER FOR A
SMALL-SCALE LIQUID AIR ENERGY STORAGE SYSTEM**

by

Alexander Fredrickson

June 2021

Thesis Advisor:

Co-Advisor:

Second Reader:

Anthony G. Pollman

Anthony J. Gannon

Paul T. Beery

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OF A HEAT EXCHANGER FOR A SMALL-SCALE LIQUID AIR ENERGY
STORAGE SYSTEM**

Alexander Fredrickson
Lieutenant, United States Navy
BS, North Carolina State University, 2011

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June 2021**

Approved by: Anthony G. Pollman
Advisor

Anthony J. Gannon
Co-Advisor

Paul T. Beery
Second Reader

Ronald E. Giachetti
Chair, Department of Systems Engineering

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ABSTRACT

The Department of Defense (DOD) is unequivocally the largest consumer of energy of all the branches in the federal government. The DOD's reliance on petroleum-based fuels increases the susceptibility of our forces stationed at forward-operating bases due to the continuous supply of these products that are required to remain operational. The implementation of renewable power sources that can be incorporated into a microgrid power system is a clear alternative to decrease our nearly sole-source reliance on fossil fuels. Small-scale liquid air energy storage systems (LAESS) are a promising novel technology that can aid in accomplishing this mission. This thesis presents a systems engineering approach with numerous systems engineering tools to aid in the selection of a heat exchanger for an existing small-scale Linde-Hampson cycle LAESS at the Naval Postgraduate School. Two alternative heat exchangers designs are analyzed and a trade-off study is conducted to determine the best alternative using systems engineering tools. The results of this analysis lead to the selection of a commercially manufactured printed circuit heat exchanger (PCHE). Once acquired, the PCHE will allow for a fully functional prototype small-scale LAESS and the potential for future analysis of LAESS viability in the force.

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LIST OF ACRONYMS AND ABBREVIATIONS

ε -NTU	Heat exchanger effectiveness-number of transfer units
ASME	American Society for Mechanical Engineers
CONOPS	Concept of operations
CV1	Control volume 1
CV2	Control volume 2
CV3	Control volume 3
DOD	Department of Defense
DoDAF	Department of Defense Architecture Framework
FTHE	Finned-tube heat exchanger
IDEF0	Integration definition for functional modeling
JT	Joule-Thomson
LAES	Liquid air energy storage
LAESS	Liquid air energy storage system
NAVFAC	Naval Facilities Engineering Command
NIST	National Institute of Standards and Technology
NPS	Naval Postgraduate School
NPT	National pipe taper
OV	Operational View
PCHE	Printed circuit heat exchanger
PE	Polyethylene
PSA	Pressure swing adsorption
PTFE	Polytetrafluoroethylene
SS304	Stainless steel 304

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EXECUTIVE SUMMARY

The Department of Defense (DOD) is actively investigating new technologies to decrease their reliance on refined petroleum fuels and implement alternative energy sources in their place as documented in the 2016 Operational Energy Strategy (Department of Defense 2016). The constant need to replenish refined petroleum reserves increases the susceptibility of our forces to attack. A 2009 report by the Army Environmental Policy Institute stated that on average, one personnel casualty occurred for every 24 fuel resupply missions conducted in Afghanistan during FY2007 (Eady et al. 2009). Incorporating alternative energy sources into microgrids on forward operating bases are one potential solution to decrease the DOD's reliance on fossil fuels and dependence on power supplied from a main power grid. Microgrids use a variety of power generation techniques, to include alternative energy sources, and can operate either in concert with, or independently from, the main power grid. To offset the innate intermittent characteristics associated with traditional alternative energy sources, such as solar and wind, this thesis explores liquid air energy storage systems (LAESS) as a means of electrical power generation and storage.

LAESSs can liquify and store atmospheric air by using power supplied from other alternative energy sources when demand for power is low. This allows the LAESS to operate independently from fossil fuels. Expanding the stored liquid air through a turbine, connected to a generator, provides power to a forward operating base when the alternative energy sources are not available (i.e., night-time or periods of low wind). This technology is still immature and requires additional analysis for future incorporation on DOD forward operating bases. To achieve this goal, the DOD can leverage the large existing infrastructure of commercially available cryogenic products to construct either a hybrid system or purchase a fully operational commercial system.

Dry atmospheric air consists of the following primary constituents: Nitrogen (78.08%), Oxygen (20.95%), Argon (0.934%), and Carbon Dioxide (0.0314%) (Wong and Embleton 1984). Due to safety and component icing concerns, this thesis performed an analysis of two nitrogen separation technologies. The two technologies that were considered were pressure swing adsorption (PSA) and membrane separation. Both

technologies provided a capability to remove impurities and unwanted elements (such as oxygen and carbon dioxide) from atmospheric air and had the ability to result in an effluent gas was inert 99%+ pure nitrogen. Using a decision evaluation display to assist in the selection of the best nitrogen separation technology for inclusion into the LAESS, it was determined that the nitrogen membrane separator was the preferred alternative. Weight and cost were ultimately the deciding factors between the two technologies.

The primary analysis performed in this thesis compared two types of heat exchangers for incorporation into an existing modified Linde-Hampson LAESS: a finned tube heat exchanger (FTHE) and a printed circuit heat exchanger (PCHE). The systems engineering process and numerous systems engineering tools were used to fully define each system and to aid in decision-making. Thermodynamic, fluid flow, and heat transfer analysis as well as a literature review on heat exchanger technologies were used to gather data about each system and support heat exchanger selection. Ultimately, the PCHE was selected as the best alternative heat exchanger for incorporation into the existing system due to its small footprint and compact design, theoretical ability to produce liquid air, its ease of integration, and its inherent safety features.

This thesis provides the current system description of the LAESS after it has undergone numerous changes and modifications since its arrival at the Naval Postgraduate School in 2017. Additionally, the thesis provides the supporting analysis and recommendations for future upgrades to the system to bring it to a fully operational state. As a result of this thesis, both a PCHE and a nitrogen membrane were ordered for incorporation into the system. Future work will consist of subsystem and system integration, test, and evaluation of the new systems components in the existing architecture. These steps will determine the LAESSs efficacy, utility, and feasibility in the goal to shift energy sources to more sustainable alternatives.

References

- Department of Defense. 2016. *Department of Defense: 2016 Operational Energy Strategy*. Washington, DC: Office of the Assistant Secretary of Defense for Energy, Installations and Environment. https://www.acq.osd.mil/eie/Downloads/OE/2016%20OE%20Strategy_WEBd.pdf.
- Eady, David S., Steven B. Siegel, R. S. Bell, and Scott H. Dicke. 2009. *Sustain the Mission Project: Casualty Factors for Fuel and Water Resupply Convoys*. Arlington, VA: Army Environmental Policy Institute.
- Wong, George S. K., and Tony F. W. Embleton. 1984. "Variation of Specific Heats and of Specific Heat Ratio in Air with Humidity." *The Journal of the Acoustical Society of America* 76 (2): 555–59. <https://doi.org/10.1121/1.391597>.

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I. INTRODUCTION

A. MOTIVATION AND BACKGROUND

The largest energy consumer in the United States federal government is indisputably the Department of Defense (DOD). According to a 2019 Congressional Research Service report to Congress, the DOD is responsible for the consumption of over three-quarters of the entire federal government's energy consumption (Greenley 2019). The Environmental Protection Agency has enacted numerous mandates, and the Secretaries of the Navy, Army, and Air Force have set their own energy goals to address the DOD's energy consumption; however, the DOD has realized very few. A shift to decrease the DOD's reliance on refined petroleum fuels and to replace them with alternative energy sources is becoming increasingly vital as the battlefield transitions to a progressively technologically dependent and more power-hungry environment. The DOD's dependence on refined petroleum fuels has also made our force more susceptible to enemy attack. A 2009 report by the Army Environmental Policy Institute stated that on average, one personnel casualty occurred for every 24 fuel resupply missions conducted in Afghanistan during FY2007 (Eady et al. 2009). The DOD must increase renewable and alternative energy sources to increase the survivability of our forces stationed abroad.

Traditional methods of power generation, which rely almost solely on fossil fuels, are becoming increasingly burdensome to the logistics community as our operations shift towards more land-locked and isolated theaters. The incorporation of microgrid technology at forward operating bases is a potential method to reduce the DOD's reliance on fossil fuels and dependence on power from a main power grid. Microgrids use a variety of power generation techniques and can operate either in concert with, or independently from, the main power grid. Microgrids often employ alternative energy generation sources, which allows for continued energy supply in the event the main power supply is interrupted. Due to the innate intermittent characteristics associated with these alternative energy sources, they require an energy storage system to prevent disruptions in electrical power supply. The traditional method of storage involves storing excess energy in batteries, which are not easily transportable and require extensive and continuous maintenance. Several storage

methods have been discovered in recent years, but most are not valid for military applications due to mobility and scalability concerns.

A promising new energy storage system that potentially suits the military is liquid air energy storage (LAES), but it still needs further refinement. There is a large existing infrastructure of commercially available cryogenic products that the DOD can leverage to construct either a hybrid system or purchase a fully operational commercial system. The Naval Postgraduate School (NPS) acquired a modified Linde-Hampson liquid air energy storage system (LAESS) in 2017 from the Naval Facilities Engineering Command (NAVFAC). Previous work with the system by a contracted company, Nitro-Turbodyne Inc., failed to get the system to produce liquid air (Amalla 2019). Considering the primary constituents of air, at 1atm, the liquefaction of oxygen occurs at approximately 90.2 K and the liquefaction of nitrogen occurs at approximately 77.4 K (National Institute of Standards and Technology 2018h; 2018g). While the LAESS was incapable of producing liquid air, it provided an opportunity for several graduate students to explore this technology.

Analysis of the LAESS at NPS began with specific component analysis. In 2018, a NPS thesis (Howe) performed an analytical analysis on the optimum pressure rating for a compressor in a Linde-Hampson cycle LAESS (Howe 2018). Additionally, Howe performed an energy and exergy analysis on the ideal system. His research concluded that the optimal compressor outlet pressure was between 20 to 50 MPa to achieve the highest energy and exergetic efficiencies as well as the highest liquid air yield. Pre-cooling the air also resulted in increased system performance. This research served as a baseline for the compressor selection in the current Linde-Hampson cycle LAESS.

The most recent analysis of the system suggests that the current heat exchanger is not properly designed and does not cool ambient air to the point of liquefaction. In 2019, a NPS thesis (Amalla) explored the modes of failure of the inherited LAESS from NAVFAC and determined the root cause to be a poorly designed heat exchanger (Amalla 2019). Amalla performed several experiments that manipulated the coiled heat exchanger's secondary side pressures and flow rates in an attempt to create liquid air. Despite his efforts, the system was never able to cool air below 213 K (Amalla 2019). Amalla continued his examination by conducting analytical research into two plausible heat exchanger designs

using high-level calculations. This resulted in two potential design options for an improved heat exchanger, a spiral tube and a finned tube heat exchanger. While the analysis was only surface level, the finned tube heat exchanger (FTHE) did show promising performance characteristics.

LAES is one method that has shown promise for potential incorporation into military microgrids. Since air is free and abundant around the globe, this alternative seems promising to store excess energy when demand on the main power grid is low. Integration of this technology into a microgrid would decrease the susceptibility of our forces to power interruptions on the main power grid and lessen a unit's dependence on fossil fuels during interrupted periods. In order to "expand and secure the supply of energy to military operations" and "build energy security into the future force" as defined in the 2011 Operational Energy Strategy, it is crucial that this technology is explored to possibly aid in achieving this goal (Department of Defense 2011, 1).

B. RESEARCH OBJECTIVE

The primary objective of this thesis is to determine what heat exchanger designs are possible to provide adequate heat transfer to produce liquid nitrogen in a modified, small-scale Linde-Hampson cycle LAESS. The requirement of the heat exchanger is to effectively cool atmospheric air to the point of liquefaction to make liquid nitrogen in a reasonable time frame. This thesis is a continuation of efforts to produce and store liquid air using a modified, small-scale Linde-Hampson cycle LAESS configuration.

C. SYSTEMS ENGINEERING APPROACH

Due to the cryogenic temperatures that are required for nitrogen to transform states from a gas into a liquid, material selection, adequate insulation and heat exchanger type are all vitally important to the success of the LAESS. The author will follow the systems engineering process to select the best heat exchanger for the current LAESS.

The systems engineering process throughout a system's life cycle is commonly illustrated using the V-model shown in Figure 1. The left side of the V-model focuses on defining and decomposing the system through the characterization of the high-level

operational concept, system requirements and specifications, and various system architectures used to perform high-level and detailed design. The right side of the V-model is focused on integration through the verification and validation of system components, subsystems, and eventually, the entire system. Testing is conducted throughout the verification and validation processes, in which the original requirements and design documentation are referenced continuously, to ensure the system is meeting all required specifications (Blanchard and Fabrycky 2010). This thesis utilizes the left side of the V-model, which is highlighted in red in Figure 1. Note that one of the initial steps of the process, a feasibility study, has already been conducted by previous students at NPS.

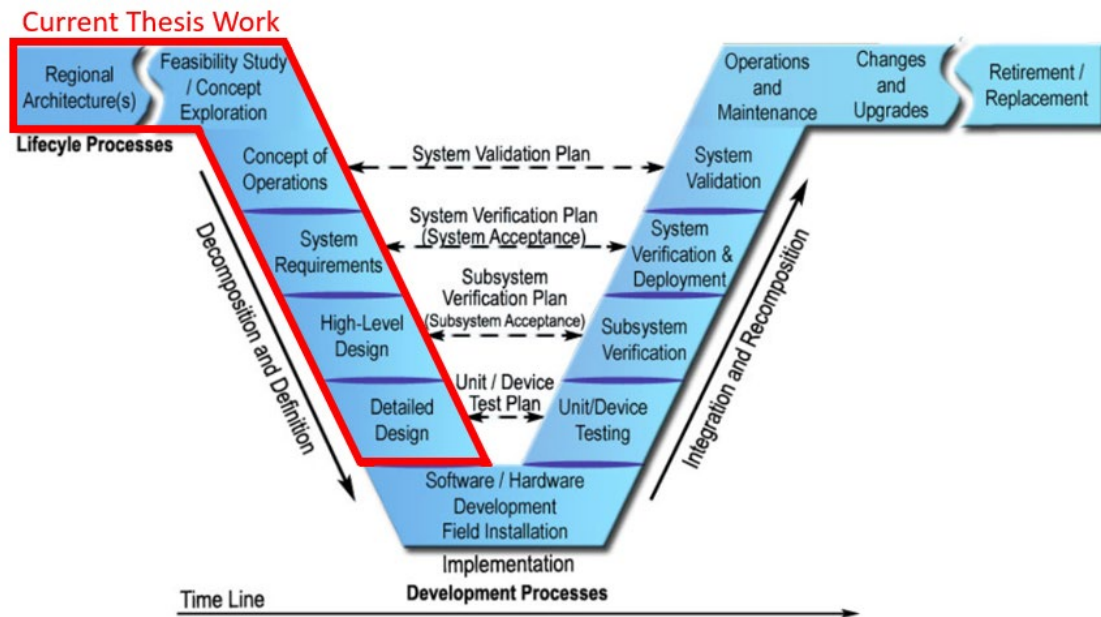


Figure 1. System Engineering Life cycle V-Model. Adapted from U.S. Department of Transportation (2020).

A finned tube heat exchanger (FTHE) and a printed circuit heat exchanger (PCHE) were analyzed. Thermodynamic, fluid flow, and heat transfer analysis as well as a literature review on heat exchanger technologies support the heat exchanger selection in this thesis. Numerous systems engineering tools and processes are employed for analysis of the potential heat exchanger designs. A Department of Defense Architectural Framework

(DoDAF) Operational View (OV), an integration definition for functional modeling (IDEF0) model, and two decision making models were utilized throughout this thesis to support the systems engineering process used for heat exchanger analysis and nitrogen separation technology selection. The use of systems engineering tools in this thesis provides enhanced traceability among system elements, greater visualization and communication of architectures and processes to stakeholders, and superior systems engineering products.

D. THESIS OVERVIEW

The previous sections have provided the background, motivation, prior work, research objective and the systems engineering approach associated with the modified Linde-Hampson cycle LAESS at NPS. Chapter II provides an overview of the modified, small-scale Linde-Hampson cycle LAESS and the concept of operations, defines the nitrogen separator and heat exchanger requirements, and includes an analysis of alternatives for two selected nitrogen separation technologies. Chapter III presents an analysis of alternatives for the FTHE and PCHE, which includes theoretical analysis and detailed designs to support heat exchanger selection. Lastly, Chapter IV contains the conclusions and future work associated with this thesis.

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II. METHOD

The modified Linde-Hampson Cycle LAESS that NPS inherited from NAVFAC was built by Nitro-Turbodyne Inc. and failed to produce liquid air. Following the system's delivery to NPS in 2017, it has undergone numerous transformations as former students and staff attempted to reconfigure the system. The original scope of full-scale integration into a microgrid has been modified to focus purely on liquid nitrogen generation.

Chapter II follows the systems engineering process by defining a hypothesized concept of operations (CONOPS) for the LAESS and describing each functional component in the system. To address the safety concerns of the stakeholders with regard to liquid oxygen generation, nitrogen separation technologies are discussed and requirements are generated for a nitrogen separator. Lastly, an analysis of alternatives is conducted for the nitrogen separation technologies and a technology is selected for future incorporation into the LAESS.

A. CONCEPT OF OPERATIONS

As previously stated, the theorized utility of the LAESS was to minimize the reliance on petroleum fuel based electrical generation methods on forward operating bases. This concept would be accomplished through the incorporation of one or more LAES systems into a microgrid comprised of other alternative sources of energy. LAES systems would be used to generate and store liquid air during periods of excess power generation and would be used to offset the intermittent power generation associated with alternative energy sources such as solar and wind technologies. This would ultimately decrease reliance from the main power grid and diesel generator power to meet the forward operating base's operational need for power. Figure 2 illustrates this CONOPS.

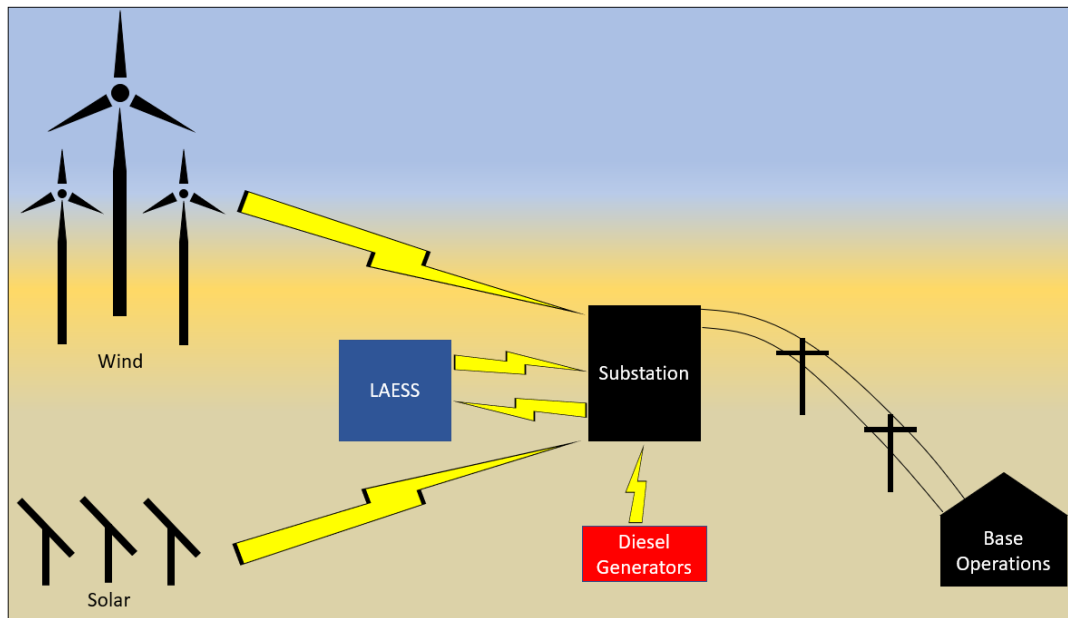


Figure 2. Concept of Operations for LAESS Inclusion on Forward Operating Bases.

Figure 2 provides a notional high level operational concept graphic, which is represented by a DoDAF OV-1. In this idealized situation, power is primarily generated by alternative sources of energy, such as wind turbines and active solar photovoltaic panels, and routed to a substation where power can be collected and distributed to support base operations. When excess energy is available due to low power demand by base operations, the excess power can be directed to the LAESS to convert atmospheric air into liquid air. The stored liquid air can be expanded through a turbine connected to a generator to provide power to the substation when the alternative energy sources are not available (i.e., night time or periods of low wind). Diesel generators remain directly connected to the substation should the power demand exceed what the alternate power sources can produce or for periods of routine or corrective maintenance. The inclusion of the LAESS and the alternate power sources greatly reduces the reliance on diesel generators for power and thus decreases the frequency of routine refined petroleum fuel replenishments. Additionally, decreased reliance on power from the main domestic power grid, if connected, is critical since power from this source is more vulnerable to attack, often unreliable, and frequently incompatible with standard military electronics.

B. MODIFIED LINDE-HAMPSON CYCLE LIQUID AIR ENERGY STORAGE (LAES) OVERVIEW

A simple Linde-Hampson cycle nominally consists of a compressor, a heat exchanger, an expansion valve, and a dewar (Kanoglu, Dincer, and Rosen 2008). Modifications to the simple cycle increase the liquid yield. Figure 3 illustrates the current modified Linde-Hampson cycle LAESS and its associated components.

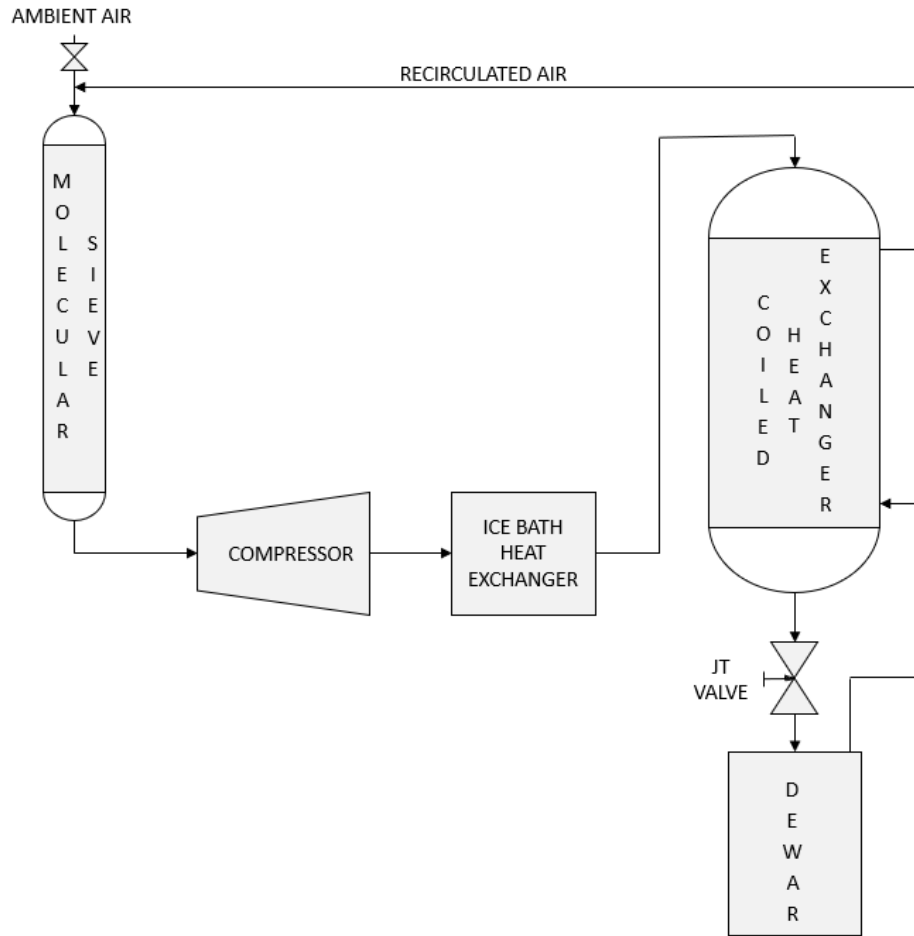


Figure 3. Current Modified Linde-Hampson Cycle LAESS Diagram.

As seen in Figure 3, the nitrogen liquefaction cycle of the modified Linde-Hampson cycle LAESS is accomplished using a regenerative process. Ambient air is drawn through a molecular sieve into an oil-less compressor, undergoes pre-cooling through an ice bath heat exchanger, enters the primary side of a coiled heat exchanger, passes through a

throttled Joule-Thomson (JT) valve, and enters a dewar. The dewar collects any liquified air that was produced and the remaining cooled air passes back through the secondary side of the coiled heat exchanger and returns to the beginning of the cycle to complete the cycle again. The cycle will run continuously until liquid air is produced, at which point makeup air is added to the system to ensure the cycle continues to produce liquid air.

The installed molecular sieve contains a mixed packed bed of adsorbing 13X and 4A sorbent beads that filters out moisture, carbon dioxide and particulate impurities from the atmospheric air. Nitro-Turbodyne, Inc. mandates that the sorbent is either regenerated or replaced after 8 hours of producing liquid air or 30 hours of system operation if liquid is not produced (Nitro-Turbodyne, Inc. 2016b). The removal of these aforementioned, undesirable atmospheric constituents is critical to the prevention of component icing within the LAESS and the JT valve.

The original system provided by NAVFAC included two compressors with approximately the same rating albeit different designs. Both compressors conformed to Howe's (2018) analysis on optimum pressure rating to achieve maximum liquid yield. The original compressor utilized was an Airetex 45 oil-flooded, three-stage compressor regulated to 3700 psi with a nominal flow rate of 3.5 SCFM (Nitro-Turbodyne, Inc. 2016b). Following initial system testing with the Airetex 45 compressor, undesirable oil residue buildup was noticed in other parts of the system downstream of the compressor. Due to the negative effects observed during initial testing, attributable to the inherent oil carryover associated with the oil-flooded design, the second compressor was used in lieu of the Airetex 45 compressor. The second and current high pressure compressor installed in the system is a RIX Industries oil-less, three-stage reciprocating air compressor, which has been regulated to output a pressure of 3300 psi at a nominal flow rate of 5.5 SCFM (RIX Industries 2016). By reducing the size of the compressor while still satisfying the optimum pressure range requirements identified by Howe, the current compressor requires less energy to power versus the original compressor, which improves the overall energy efficiency of the system. Additionally, the higher flow rate associated with the RIX compressor was more desirable because an increase in the mass flow rate entering the system would result in less potential fouling of the heat exchanger tubes.

The ice bath heat exchanger is designed to pre-cool the air exiting the compressor to expedite the initial production of liquid air due to increased air temperatures as a result of the compression process (Nitro-Turbodyne, Inc. 2016b). The ice bath heat exchanger utilizes 20 feet of 0.25 inch 304 stainless-steel (SS304) tubing with aluminum square plate fins (Nitro-Turbodyne, Inc. 2016a). The tubing is contained in a standard five-gallon bucket and is cooled by a mixture of ice, salt, and water. As validated in previous testing, it was assumed that the temperature of the air leaving the ice bath heat exchanger is approximately 273 K for analysis in this thesis (Amalla 2019).

The coiled tube heat exchanger contains 100 feet of 0.25 inch SS304 tubing that is wrapped in a 0.125 inch round copper wire and surrounded by 60 feet of polytetrafluoroethylene (PTFE) corrugated tubing and 20 feet of polyethylene (PE) corrugated tubing. The PTFE and PE tubing form the secondary side of the heat exchanger, and the copper wire provides a tortuous flow path to increase heat transfer. Additional insulation around the secondary side tubes and in the empty voids in the heat exchanger shell minimizes heat losses to the environment (Nitro-Turbodyne, Inc. 2016a).

The JT valve is where the majority of the cooling is produced in the LAESS. Throttling a gas, such as nitrogen, which possess a positive JT coefficient at ambient temperatures, produces a drop in temperature as the gas expands across the valve through the isenthalpic pressure drop (Flynn 2004). “The valve previously used as the JT valve was not readily apparent” upon system delivery to NPS (Amalla 2019, 10). This reality forced Amalla to substitute a SS304 needle valve with a remote operating handle in its place for system reconstruction (Amalla 2019).

The dewar collects liquid air is collected for later use. It receives the cooled air from the JT valve, collects any liquid generated, and passes the remaining gaseous cooled air back to the secondary side of the heat exchanger to provide cooling to the primary side of the heat exchanger. The original dewar provided by Nitro-Turbodyne, Inc. was constructed of SS304 with foam filled vacuum insulation and possessed a 30 L storage volume (Nitro-Turbodyne, Inc. 2016a). To minimize heat losses in the system due to a large storage volume, the dewar was replaced with a borosilicate glass dewar with a 1 L storage volume.

C. NITROGEN SEPARATION

Atmospheric air is primarily composed of nitrogen, oxygen, argon, carbon dioxide and water vapor. From these constituents, the primary concerns regarding LAESS operations were carbon dioxide, water vapor, and oxygen due to safety and component icing concerns. Due to these concerns, the stakeholders decided to incorporate a component into the LAESS to remove these constituents from atmospheric air and to use high purity nitrogen as the working fluid in the system. Although argon has a higher boiling point than nitrogen (i.e., will liquefy before nitrogen), its inherent inert properties and small proportion in atmospheric air was determined to be acceptable for the system (National Institute of Standards and Technology 2018a). Alternatively, liquid oxygen exacerbates combustion and can result in extremely violent explosions if proper infrastructure and precautions are not implemented to contain its volatility (CAMEO Chemicals 2020). The stakeholders mandated that oxygen gas must be removed from the incoming atmospheric air to prevent the formation and accumulation of liquid oxygen in the dewar. Experimentation with the existing LAESS in the past did not solicit the need to remove oxygen from the intake air since the system was never close to reaching the temperatures associated with the liquefaction of oxygen. However, since the incorporation of a new heat exchanger into the LAESS is likely to result in the production of liquid air, oxygen must be removed from the system. Carbon dioxide and water vapor were likely to cause issues with component icing, specifically in the JT valve, due to their high freezing points relative to nitrogen and, thus, were also required to be removed. Since the existing molecular sieve that originally filtered these atmospheric constituents from the intake air would likely need to be removed to allow for the installation of nitrogen separation equipment, it was imperative that the new nitrogen separation equipment incorporate a means to expel these constituents.

In industry, three primary means exist to separate nitrogen from air: cryogenic distillation, pressure swing adsorption (PSA), and membrane separation (GENERON 2020a). Since cryogenic distillation was to be avoided, this alternative was not considered. The other two methods considered mechanically extract nitrogen from air.

a. PSA

A PSA nitrogen generator relies on molecular sieves to preferentially adsorb a targeted chemical species from a heterogenous gas mixture. As atmospheric air passes through the molecular sieves, oxygen is trapped and embedded in the sieve and only nitrogen remains in the effluent gas. Once a molecular sieve is saturated, it can be desorbed by reducing the pressure in the system below the previous operating pressure, at which point the oxygen is released from the sieve and vented to the surrounding environment (GENERON 2020b). A nominal PSA system configuration is shown in Figure 4.

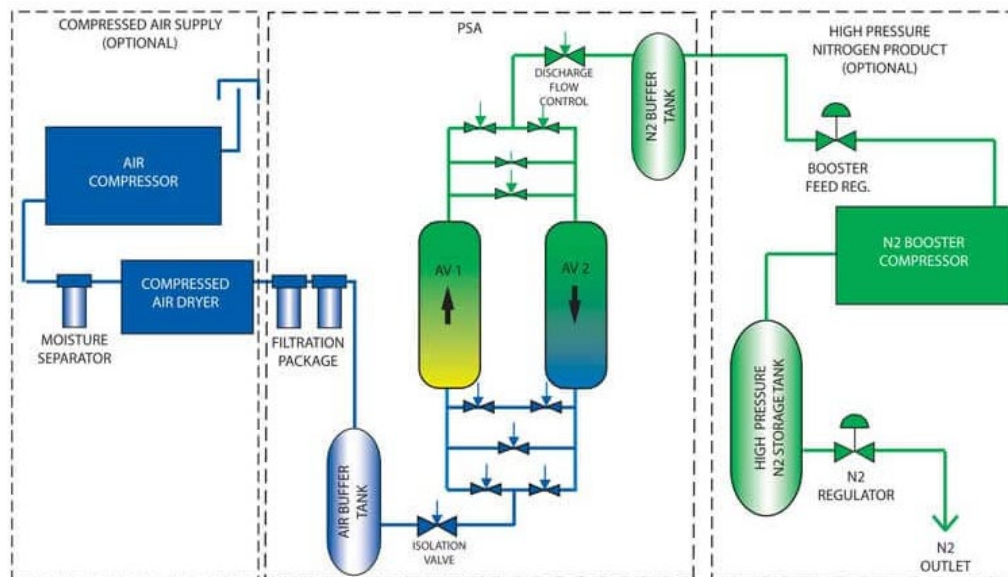


Figure 4. Nominal PSA Nitrogen Separation Cycle. Source: GENERON (2020b).

As illustrated in Figure 4, atmospheric air is drawn in and compressed. All moisture and particulates are removed from the compressed air by a series of moisture separators, an air dryer and filter banks. The dry compressed air then enters one of the adsorption towers while the other tower remains isolated. As the dry air passes through the adsorption tower, oxygen and carbon dioxide are removed, and primarily nitrogen remains. The purity of the nitrogen is adjustable depending on the quality desired. As the online adsorption tower nears oxygen saturation, the standby adsorption tower goes online and the previously

operational tower is isolated. Pressure reduces in the offline tower and the trapped oxygen and carbon dioxide vents back into the environment to regenerate the sieve material. Lastly, the nitrogen passes through additional buffer and storage tanks where it can be stored for later use (GENERON 2020b).

b. Membrane Separation

Conversely, nitrogen membrane modules employ hundreds of thousands of hollow fibers that selectively permeate oxygen, carbon dioxide, and water vapor from the air. As a heterogeneous gas mixture passes through the membrane, oxygen, carbon dioxide, and water vapor permeate out of the membrane and are vented to the environment (NANO Purification Solutions 2020a). Figure 5 provides a nominal nitrogen membrane system configuration.

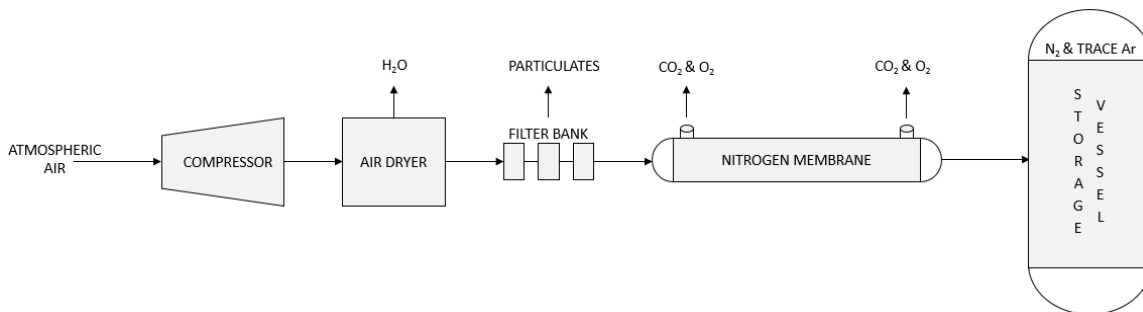


Figure 5. Nominal Nitrogen Membrane Separation System.

As shown in Figure 5, atmospheric air is drawn in and compressed by a compressor. The compressed air enters an air dryer assembly to remove moisture and contaminants. The dry air passes through an additional filter bank that is organic to the nitrogen membrane assembly to remove any additional contaminants present in the air. The clean, dry air passes through the nitrogen membrane's hollow fibers and oxygen and water vapor is selectively permeated and discharged into the surrounding environment. The purity of the effluent nitrogen is a function of compressor pressure, compressor flow rate, and the rating of the membrane. The nitrogen that is produced can be stored in a flask for later use (NANO Purification Solutions 2020b).

c. Analysis of Alternatives

Based on system parameters defined by existing equipment in the current LAESS and input from stakeholders, operational requirements for the nitrogen separator were generated. The requirements were determined to be as follows:

1. The nitrogen separator must be capable of providing a continuous supply of at least 99% pure nitrogen to the LAESS.
2. The nitrogen separator must conform to the 74.4 kg weight limit for two males carrying an object 10 m or less specified in MIL-STD-1472G-CHG 1 (Department of Defense 2019) to prevent the necessity for specialized lifting equipment.
3. The nitrogen separator shall provide a flow rate of at least 5.5 SCFM to ensure sufficient feed air is supplied to the high pressure compressor in the LAESS.

A comparison of the two alternative technologies was performed to determine which nitrogen separation solution would be more appropriate for integration with the existing LAESS. Since no status quo system existed in the current LAESS and cost was a concern to the stakeholders, a decision evaluation display was generated. Using brochure data provided by a popular manufacturer for nitrogen separation technologies, Nano Purification Solutions, LLC, and interacting with a local vendor for that company, performance data and the approximate manufacturer suggested retail prices (MSRP) were obtained for two comparable systems. The nitrogen separators that were considered were the NMG1126 nitrogen membrane and the GEN2 i4.0-2110 PSA nitrogen separator. The specific models were chosen because each could satisfy the minimum performance requirements demanded of the system at the lowest cost. The decision evaluation display that compares the alternatives is shown in Figure 6.

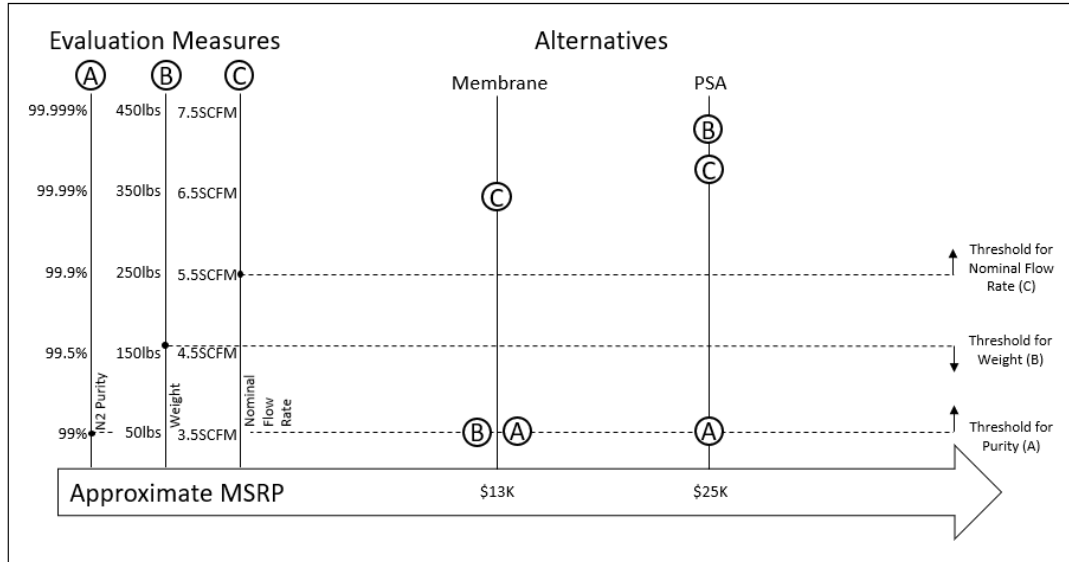


Figure 6. Decision Evaluation Display for Nitrogen Separation Alternatives.

Figure 6 illustrates the multi-attribute decision making process that was followed to select a viable nitrogen separation technology for integration into the existing LAESS. The decision evaluation display was generated by plotting the magnitudes of three evaluation measures selected to satisfy the requirements for the nitrogen separator for each respective technology. The three evaluation measures and their respective thresholds were nitrogen purity (greater than or equal to 99%), weight (less than or equal to 74.4 kg), and nominal flow rate (greater than or equal to 5.5 SCFM). The NMG1126 nitrogen membrane under consideration had an advertised weight of 51 lbs and was capable of producing 99% pure nitrogen by supplying the membrane with 200 psig inlet pressure, which resulted in a nominal flow rate of 6.47 SCFM (NANO Purification Solutions 2020a). The GEN2 i4.0-2110 PSA nitrogen membrane under consideration had an advertised weight of 437 lbs and was capable of producing 99% pure nitrogen by supplying the system with 100 psig inlet pressure, which resulted in a nominal flow rate of 6.83 SCFM (NANO Purification Solutions 2020a).

The results from the decision evaluation display yielded that the nitrogen membrane was the preferred technology for nitrogen separation due to its ability to satisfy all threshold requirements at a lower cost. The PSA nitrogen separator satisfied all operational

requirement thresholds except for the weight requirement; therefore, due to a combination of its larger weight and cost, it was determined to not be an ideal technology to implement into the LAESS.

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III. HEAT EXCHANGER ANALYSIS

To generate requirements for the heat exchanger in the LAESS, an integration definition for functional modeling (IDEF0) was generated for the system (Figure 7). The IDEF0 modeled all inputs, controls, mechanisms, and outputs of the associated major functions within the LAESS.

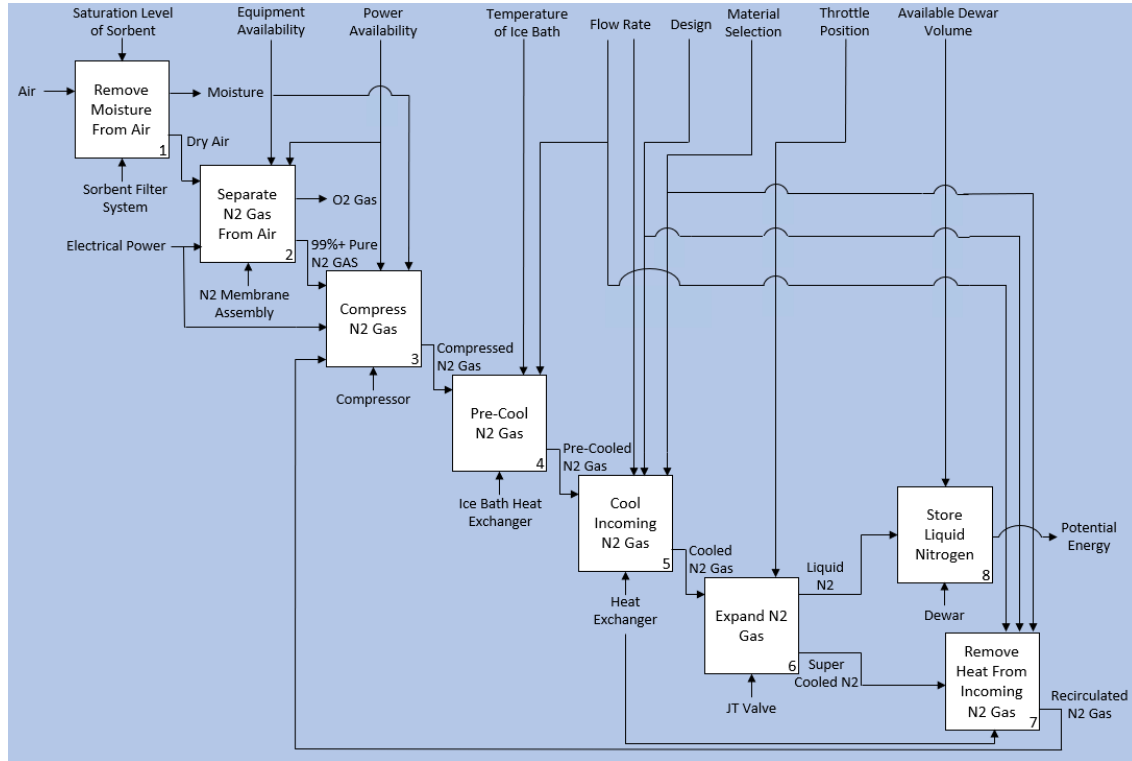


Figure 7. IDEF0 for the LAESS.

The observations in the IDEF0, system parameters defined by existing equipment in the current LAESS, and input from stakeholders' needs determined the operational requirements for the heat exchanger:

1. The heat exchanger must be capable of cooling gaseous nitrogen from a temperature of approximately 273 K (32°F) to 162.9 K (-166.5°F) at a nominal flow rate of 3.1×10^{-3} kg/s (5.5 SCFM).

2. The heat exchanger must be built to the American Society of Mechanical Engineers (ASME) codes to withstand the thermal and pressure stresses in the cycle.
3. The heat exchanger must possess interfaces that are compatible with existing 6.35×10^{-3} m (0.25 inch) National Pipe Taper (NPT) compression fittings to allow for integration into the existing system.
4. The heat exchanger must be compact to allow for undemanding transport, setup, and disassembly.

The subsequent sections provide a literature review for heat exchanger technologies under consideration. Cycle analysis was used to determine the dimensional requirements for the FTHE and the specific system parameters required for the PCHE. Lastly, an analysis of alternatives was conducted to select the preferred heat exchanger technology for implementation into the LAESS.

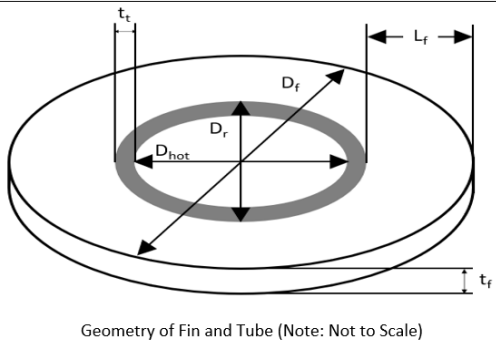
A. FINNED TUBE HEAT EXCHANGER (FTHE)

FTHEs are classified as compact heat exchangers since they are characteristically physically smaller than traditional shell and tube heat exchangers due to the increased heat transfer surfaces associated with the fins (Thulukkanam 2013). While FTHEs are traditionally used in refrigeration and air conditioning applications, this design was chosen as an alternative to investigate since previous work indicated that a FTHE was a potentially viable heat exchanger design for the LAESS (Amalla 2019).

Analysis was based on commonly available circular helical fin tubes for inclusion in the FTHE design (Amalla 2019). The tube and fins are constructed of austenitic stainless steel 304 (SS304) to prevent fracture in the high-pressure cryogenic environment. Despite a poor thermal conductivity relative to other metals, SS304 can withstand temperatures as low as 4 K and thus is one of the most commonly used materials for cryogenic applications (Thulukkanam 2013). A brazing process binds the fins and tube together, which creates a corrosion resistant bond at the interface (Fin Tube Products, Inc. 2020). This fin design is a circular tube and thus circular tube correlations were used for analysis (Thulukkanam

2013). Table 1 lists the dimensions and symbols used for analysis for the circular helical fin tube.

Table 1. Circular Helical Fin Dimensions. Adapted from Fin Tube Products, Inc. (2020).

Circular Helical Fin Tube Dimensions				
Description	Symbol	Dimension (inch)	Dimension (m)	
Hot Inside Diameter	D_{hot}	3/16	4.7625E-03	
Fin Root Effective Diameter	D_r	1/4	6.3500E-03	
Fin Tip Diameter	D_f	5/8	1.5875E-02	
Thickness of Tube	t_t	1/20	1.2700E-03	
Thickness of Fin	t_f	3/200	3.8100E-04	
Fin Length	L_f	3/16	4.7625E-03	
Fin Spacing	S	1/6	4.2333E-03	

The existing system originally had the recirculated air pass through the molecular sieve during each cycle, which is unnecessary since the air has already been filtered. The author has improved the existing system design by having the return nitrogen flow reenter the cycle after the nitrogen membrane assembly. In addition to creating a more efficient cycle, the nitrogen membrane assembly cannot tolerate temperatures below 41°F (278 K); therefore, the recirculated nitrogen connection was placed downstream of the nitrogen membrane assembly. Figure 8 provides a cycle diagram for the hypothesized modified Linde-Hampson LAESS, which includes the nitrogen membrane. The dashed boxes in Figure 8 correspond to the control volumes used for cycle analysis and process steps 1–8 are defined.

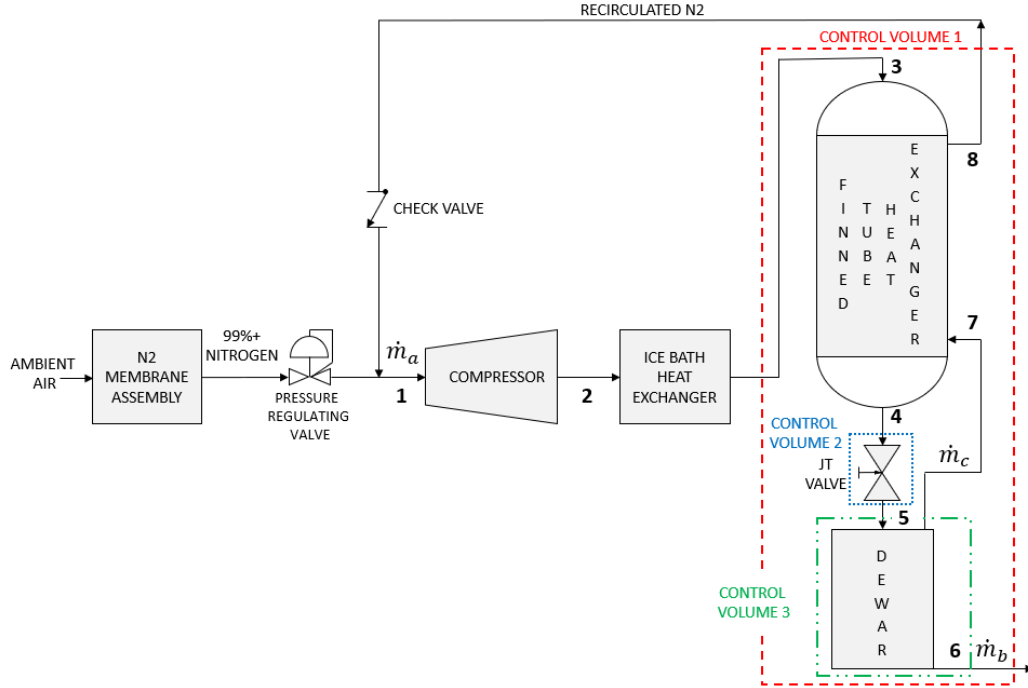


Figure 8. Hypothesized Modified Linde-Hampson LAESS with Incorporated Nitrogen Membrane.

The thermodynamic analysis began with the definition of the distinct process steps in the LAESS and of the assumed states associated with those stages (Figure 9). Figure 9 shows that the ideal modified Linde-Hampson cycle is isothermal during process steps 1–2 and 5–7, isobaric during process steps 2–4 and 7–1, and isenthalpic during process steps 4–5.

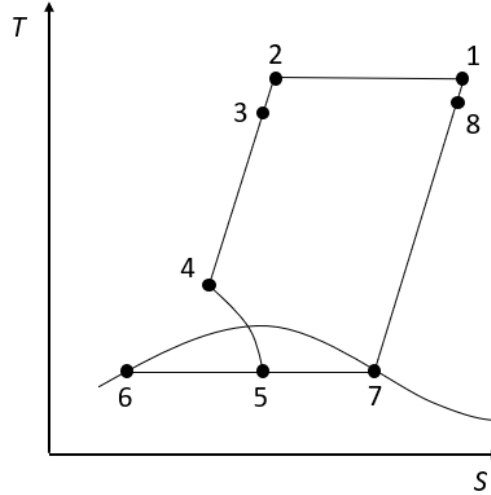


Figure 9. Temperature Entropy Diagram for Modified LAESS. Adapted from Kanoglu, Dincer, and Rosen (2008).

Following assignment of the process steps, thermodynamic properties were obtained from the National Institute of Standards and Technology (NIST). Based on the assumptions made in each of the stages, the estimated thermodynamic properties for process steps 1, 3, 6, and 7 were found for initial analysis (National Institute of Standards and Technology 2018b; 2018c; 2018d).

To assist in calculating the dimensional requirements for the FTHE, additional assumptions were made. In order to ensure laminar flow in the FTHE, a primary side Reynolds number of 1800 was originally assumed as a starting point for analysis to be below the laminar-turbulent transition zone that begins to occur around a Reynolds Number of 2000 (Granger 2012). The cycle requires an efficient heat exchanger; therefore, a cryogenic heat exchanger effectiveness value (ϵ^*), as defined in Equation (9), of 95% was assumed (Barron and Nellis 2016).

B. CYCLE ANALYSIS

Due to three distinct flows in this cryogenic cycle, the mass flows were annotated by subscripted variables versus distinct cycle process steps. These relationships are given by Equations (1)–(3).

$$\dot{m}_a = \dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}_5 \quad (1)$$

$$\dot{m}_b = \dot{m}_6 \quad (2)$$

$$\dot{m}_c = \dot{m}_7 = \dot{m}_8 \quad (3)$$

The problem was first approached by assigning a control volume (CV1) around the FTHE, the Joule-Thomson (JT) valve, and the dewar. Liquid Yield (Y) is defined as the ratio of $\dot{m}_b$, the mass flow out of the dewar (liquified nitrogen), and $\dot{m}_a$, the mass flow into the FTHE (Howe, Pollman, and Gannon 2018).

$$Y = \frac{\dot{m}_b}{\dot{m}_a} \quad (4)$$

The law of conservation of mass derives a relationship between the mass flow into and out of CV1.

$$\dot{m}_a = \dot{m}_b + \dot{m}_c \quad (5)$$

The mass flow entering CV1 is the same as the cryogenic fluid exiting the dewar and the unliquified portion of the flow exiting the secondary side of the heat exchanger. The steady-state energy equation leads to the following enthalpy flow relationship, where h is the specific enthalpy for the stated process step.

$$\dot{m}_a h_3 = \dot{m}_b h_6 + \dot{m}_c h_8 \quad (6)$$

Rearranging Equation (5) and substituting it into Equation (6) reveals the relationship between the mass flow through the compressor and the yielded fluid, which gives Equation (7).

$$\dot{m}_a (h_3 - h_8) = \dot{m}_b (h_6 - h_8) \quad (7)$$

Rearranging Equation (7) and noting that h_1 and h_8 are equivalent for an ideal cycle, an expression for the liquid yield of the cycle, Y , is obtained (Howe, Pollman, and Gannon 2018).

$$Y = \frac{\dot{m}_b}{\dot{m}_a} = \frac{h_3 - h_1}{h_6 - h_1} \quad (8)$$

Using the current operating temperatures and pressures in the cycle, the ideal yield was determined to be approximately 14.8%.

The actual system is not ideal. The cryogenic effectiveness used in this analysis, ε^* , is defined by Equation (9), and reflects that not all heat is transferred from the cold stream to the hot stream (Thulukkanam 2013). This impacts the real liquid yield significantly.

$$\varepsilon^* = \frac{\dot{Q}_{actual}}{\dot{Q}_{max}} = \frac{\dot{m}_c (h_8 - h_7)}{\dot{m}_c (h_1 - h_7)} = \frac{h_8 - h_7}{h_1 - h_7}, \quad (9)$$

where $\dot{Q}_{max}$ is the maximum possible heat transfer rate and $\dot{Q}_{actual}$ is the actual heat transfer rate limited by the return gas flow.

Rearranging Equation (9) allowed for the calculation of the specific enthalpy for the outlet of the secondary side of the heat exchanger (h_8), given by Equation (10).

$$h_8 = \varepsilon^* (h_1 - h_7) + h_7 \quad (10)$$

Equation (10) resulted in a specific enthalpy, h_8 , of approximately 290 kJ/kg. Using the calculated h_8 and assuming no pressure drop through the secondary side of the heat exchanger, the outlet temperature for the secondary side of the heat exchanger (T_8) was determined to be approximately 279.7 K (National Institute of Standards and Technology 2018e).

By combining Equations (7) and (10), real yield, Y_{real} , is defined in terms of the cryogenic heat exchanger effectiveness in Equation (11). The result was a yield of approximately 12.45%, which is a considerable drop when compared to the ideal yield of 14.8%.

$$Y_{real} = \frac{h_3 - h_7 - \varepsilon^*(h_1 - h_7)}{h_6 - h_7 - \varepsilon^*(h_1 - h_7)} \quad (11)$$

Applying a second Control Volume (CV2) around the JT valve, the law of conservation of mass derives a relationship between the specific enthalpies for the flows entering and exiting CV2.

$$h_4 = h_5 \quad (12)$$

Applying a third Control Volume (CV3) around the dewar, the law of conservation of energy derives a relationship between the mass flowrates and enthalpies for the flows entering and exiting CV3.

$$\dot{m}_a h_5 = \dot{m}_b h_6 + \dot{m}_c h_7 \quad (13)$$

Rearranging Equation (13), the specific enthalpy for the exit of the JT valve is defined in Equation (14).

$$h_5 = \frac{\dot{m}_b}{\dot{m}_a} h_6 + \frac{\dot{m}_c}{\dot{m}_a} h_7 \quad (14)$$

Combining Equations (4) and (5), the ratio of mass flow rate exiting the dewar and mass flow rate entering the FTHE is redefined by Equation (15).

$$Y_{real} = 1 - \frac{\dot{m}_c}{\dot{m}_a} \quad (15)$$

With the real yield calculated using the assumed cryogenic effectiveness, Equation (15) was rearranged to solve for the mass flow rate of the nitrogen gas exiting the dewar and entering the secondary side inlet of the FTHE.

$$\dot{m}_c = \dot{m}_a (1 - Y_{real}) \quad (16)$$

Substituting Equations (5) and (15) into Equation (14), the specific enthalpy for the JT valve exit was determined to be approximately 57.9 kJ/kg.

$$h_5 = (Y_{real} h_6) + (1 - Y_{real}) h_7 \quad (17)$$

Applying the relationship defined by Equation (12), the thermodynamic properties for the outlet of the primary side of the heat exchanger (process step 4) were found using the NIST nitrogen tables (National Institute of Standards and Technology 2018f).

With the mass flow rate of the nitrogen gas entering the secondary side inlet of the FTHE calculated, Equation (9) was rearranged to solve for the actual heat loss rate, assuming a cryogenic heat exchanger effectiveness of 95%. Equation (18) yielded a heat loss rate of approximately 557 W.

$$\dot{Q}_{actual} = \varepsilon^* \cdot \dot{Q}_{max} = \varepsilon^* \cdot \dot{m}_c (h_1 - h_7) \quad (18)$$

With the thermodynamic properties calculated for all applicable process steps in the LAESS cycle, the number of tubes to achieve laminar flow on the primary side of the FTHE was calculated. The Reynolds number is defined in Equation (19) in terms of the hydraulic diameter (D_h), the mass flow rate, the viscosity, and the minimum free flow area (A_o) (Thulukkanam 2013).

$$Re = \frac{D_h \cdot \dot{m}}{\mu \cdot A_o}, \quad (19)$$

where A_o is a function of the number of tubes in the FTHE (N_t) and the tube inner diameter (D_{hot}) (Thulukkanam 2013).

$$A_o = \frac{\pi \cdot D_{hot}^2}{4} N_t \quad (20)$$

For the primary side of the FTHE, the hydraulic diameter was equal to the inner diameter of the tube. Therefore, Equations (19) and (20) were rearranged in terms of the number of tubes required to maintain laminar flow.

$$N_t = \frac{4 \cdot \dot{m}_a}{\pi \cdot D_{hot} \cdot \mu_3 \cdot \text{Re}} \quad (21)$$

The number of tubes required on the primary side to maintain laminar flow, assuming a Reynolds number of 1800 and considering the flow into the primary side of the heat exchanger, was determined to be 19 tubes. Secondary side analysis was conducted in a similar manner. However, it was found that the primary side was more limiting than the secondary side.

Since 19 tubes did not provide for a desirable symmetric staggered tube-bundle arrangement, it was determined that 21 tubes would be used instead. By rearranging Equation (21), Equation (22) is used to find the actual Reynolds Number of 1625, based on the flow entering the primary side of the heat exchanger and the number of tubes used for the FTHE design.

$$\text{Re} = \frac{4 \cdot \dot{m}_a}{\pi \cdot D_{hot} \cdot \mu_3 \cdot N_t} \quad (22)$$

With the number of tubes determined, the length of the tubes was also required to be found. The heat exchanger effectiveness-number of transfer units (ϵ -NTU) method, an analysis of the Nusselt number, heat exchanger geometrical relationships, and the inclusion of the fin effects were used to determine the required length of the tubes.

First, the ϵ -NTU method was used to solve for the overall heat transfer coefficient of the heat exchanger (UA). The heat capacity rate ratio is defined in Equation (23) (Thulukkanam 2013).

$$C^* = \frac{C_{\min}}{C_{\max}}, \quad (23)$$

where $C_{\min}$ and $C_{\max}$ were associated with the smallest and largest temperature changes, respectively, and were calculated using Equations (24) and (25) (Thulukkanam 2013).

$$C_{\min} = \frac{\dot{Q}_{\max}}{(T_{h,i} - T_{c,i})} \quad (24)$$

$$C_{\max} = \frac{\varepsilon^* \cdot \dot{Q}_{\max}}{(T_{h,i} - T_{h,o})} \quad (25)$$

The heat capacity rate ratio was calculated to be 0.65 using Equation (23).

The heat exchanger was modeled as a counter-flow heat exchanger. The NTU for $C^* < 1$ was calculated to be approximately 5.8 using Equation (26) (Incropera et al. 2007).

$$NTU = \frac{1}{C^* - 1} \ln \left(\frac{\varepsilon^* - 1}{(\varepsilon^* \cdot C^*) - 1} \right) \quad (26)$$

The NTU was defined as the ratio of the overall heat transfer coefficient of the heat exchanger to the minimum heat capacity rate as shown in Equation (27).

$$NTU = \frac{UA}{C_{\min}} \quad (27)$$

By rearranging Equation (27), the overall heat transfer coefficient was determined to be approximately 0.9 W/K per tube.

$$UA = NTU \cdot C_{\min} \quad (28)$$

With the overall heat transfer coefficient calculated, the Nusselt numbers for the primary side and secondary side of the FTHE were calculated to find the primary and

secondary side convective heat transfer coefficients. Assuming a uniform heat flux wall boundary condition, fully developed laminar flow, and smooth, round tubes, Equation (29) provided the definition for the primary side Nusselt number (Thulukkanam 2013).

$$Nu_h = \frac{h_h \cdot D_h}{k_h} = 4.364 \quad (29)$$

Rearranging Equation (29) gives Equation (30), which yielded a primary convective heat transfer coefficient of approximately 36.5 W/m²·K using the hot inside diameter of the tube as the hydraulic diameter and the thermal conductivity of the fluid entering the heat exchanger.

$$h_h = \frac{Nu_h \cdot k_3}{D_{hot}} \quad (30)$$

Solving for the secondary side convective heat transfer coefficient was not as simple. The flow characteristics associated with circular helical fin designs are complicated; however, numerous empirical methods have been developed to predict the flow characteristics for circular fin designs using specific core geometries and flow parameters. The author chose to implement the Briggs and Young Correlation based on a regression model to support all tube bank and fin height configurations for circular tubes. The crimped fin interface was ignored and assumed to be completely circular. In order to use this correlation, a staggered, equilateral pitch core geometry with six or greater tube rows had to be incorporated into the design. Associated core geometric parameters were defined, as shown in Figure 10, where L_2 and L_3 are the respective heat exchanger width and height dimensions, P_t is the lateral pitch, and P_L is the longitudinal pitch.

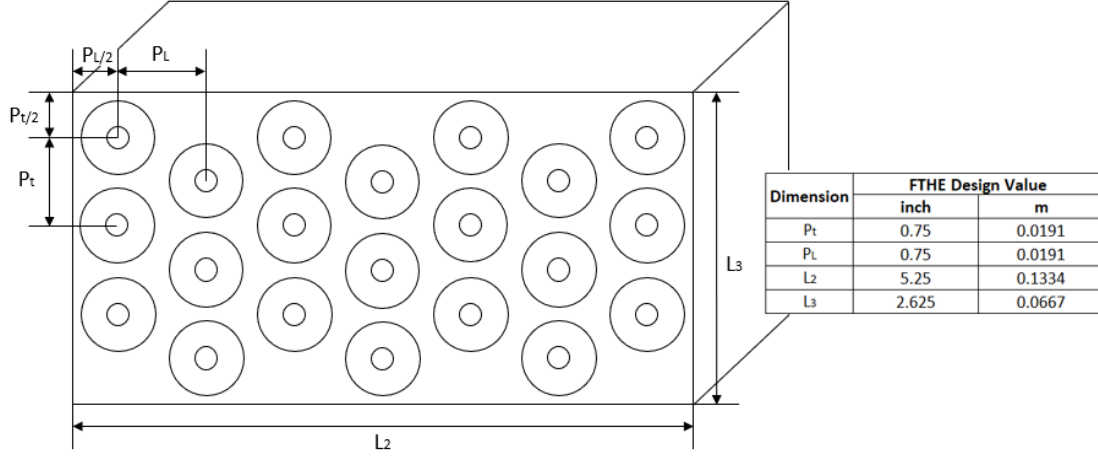


Figure 10. Staggered Equilateral Pitch FTHE Design Cross-Section.

Equation (31) is used to calculate the Reynolds Number based on the tube outer diameter (fin root effective diameter) for use in the Briggs and Young Correlation (Thulukkanam 2013).

$$Re = \frac{D_r \cdot \rho_7 \cdot V_{max}}{\mu_7}, \quad (31)$$

where V_{max} is the maximum velocity of the flow through the finned tube bank and is a function of the minimum free flow area (A_o) through the tube bank, the frontal area (A_{fr}), and the upstream fin side fluid velocity (Thulukkanam 2013).

$$V_{max} = \frac{\dot{m}_c}{\rho_7 \cdot L_2 \cdot L_3} \left(\frac{A_{fr}}{A_o} \right) \quad (32)$$

Substituting in the variables into Equation (32) yielded a maximum velocity of approximately 0.055 m/s. With V_{max} calculated, Equation (31) resulted in a secondary side Reynolds number of approximately 518.

The Briggs and Young Correlation is used for Reynolds Numbers ranging from 1,000–20,000 (Thulukkanam 2013). Acknowledging that the calculated secondary side Reynolds Number is outside of this range, the correlation is used to obtain an estimate of 11.5 for the secondary side Nusselt number in Equation (33) (Thulukkanam 2013).

$$Nu_c = 0.134 Re_d^{0.681} Pr_7^{1/3} \left(\frac{s}{L_f} \right)^{0.2} \left(\frac{s}{t_f} \right)^{0.1134} \quad (33)$$

Using the fin root effective diameter as the hydraulic diameter, the secondary side convective heat transfer coefficient was found using Equation (34) and yielded a primary convective heat transfer coefficient of approximately 14.9 W/m²·K.

$$h_c = \frac{Nu_c \cdot k_7}{D_r} \quad (34)$$

With the primary and secondary convective heat transfer coefficients calculated, the effects from the fins had to be considered. The primary purpose of fins is to increase the total surface area to increase the heat transfer rate. The effectiveness of a fin is a function of the fin's geometrical and metallurgical properties. A fin's thickness, length, shape, and composition determine the efficiency of the fin, which is a ratio of the actual heat transfer rate from the fin to the secondary side fluid to the maximum rate of heat transfer possible (Theodore 2011). If the fin efficiency achieves unity, it is perfectly isothermal in that the temperature of the fin surface is equivalent to the temperature of the base of the fin (Theodore 2011). The thermal conductivity of SS304 assumed for analysis was 11.45 W/m·K, which was based on an average of the secondary side entrance and exit temperatures (National Institute of Standards and Technology 2020).

Considering the FTHE fins as annular fins, the fin efficiency was approximated graphically in Figure 11. In order to use the graphical approach, additional geometrical variables and ratios had to be defined. These variables were the corrected outside radius (r_{2c}), the corrected height (L_c), the profile area (A_p), and a dimensionless relationship between the three variables (X) (Theodore 2011). The equations for these variables are embedded into Figure 11. Using the above variables, the fin efficiency was approximated to be 89%.

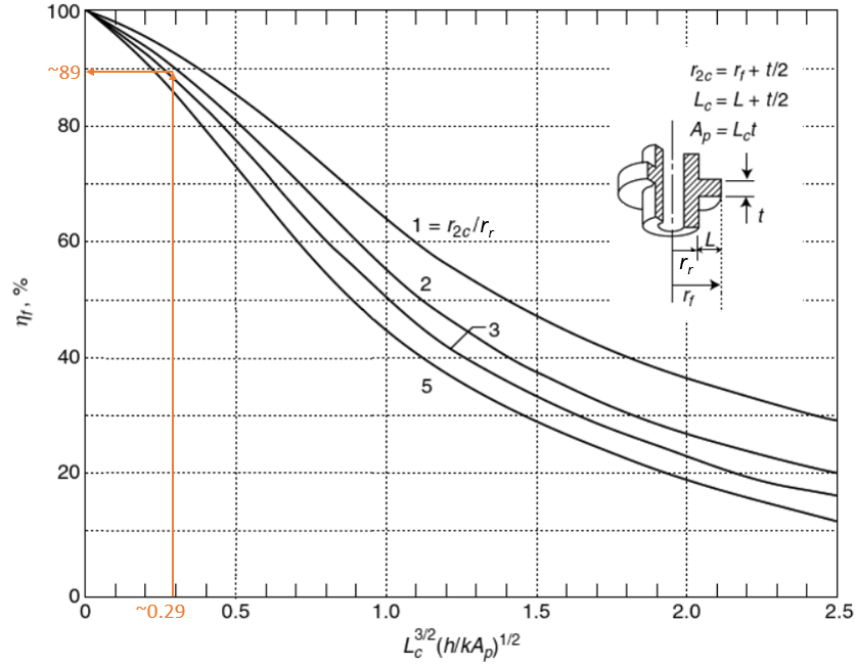


Figure 11. Annular Fin Efficiency. Adapted from Theodore (2011).

With the fin efficiency approximated, additional fin parameters were calculated to determine the required length of the tubes (L_t) in the FTHE. The surface area of the fin (A_f) is found using Equation (35) (Incropera et al. 2007).

$$A_f = 2\pi \left(r_{2c}^2 - \left(\frac{D_r}{2} \right)^2 \right) \quad (35)$$

In addition to the surface area of the fin, the area of the prime surface (A_b) and the total surface area of the tube (A_t) are found using Equations (36) and (37) (Incropera et al. 2007).

$$A_b = \pi \cdot D_r \left(L_t - \frac{L_t \cdot t_f}{s} \right) \quad (36)$$

$$A_t = \frac{L_t \cdot A_f}{s} + A_b \quad (37)$$

Substituting Equation (36) into (37), obtains an expression for the total surface area of the tube in terms of L_t .

$$A_t = L_t \left[\frac{A_f}{s} + \pi \cdot D_r \left(1 - \frac{t_f}{s} \right) \right] \quad (38)$$

With the pertinent areas defined, Equation (39) defines the overall fin efficiency (Incropera et al. 2007).

$$\eta_o = 1 - \frac{L_t \cdot A_f}{s \cdot A_t} (1 - \eta_f) \quad (39)$$

Substituting Equation (38) into Equation (39) yielded an expression for overall fin efficiency of approximately 92%.

$$\eta_o = 1 - \left(\frac{A_f}{A_f + \pi \cdot D_r (s - t_f)} \right) (1 - \eta_f) \quad (40)$$

Neglecting effects from surface fouling and accounting for fin effects on the secondary side, the overall heat transfer coefficient is redefined in terms of the thermal resistances of the primary and secondary fluids as well as the thermal resistance for the cylindrical tube wall in Equation (41) (Incropera et al. 2007).

$$\frac{1}{UA} = \frac{1}{\eta_o \cdot h_c \cdot A_t} + \frac{\ln(D_r/D_{hot})}{2\pi \cdot k_{304} \cdot L_t} + \frac{1}{h_h \cdot \pi \cdot D_{hot} \cdot L_t} \quad (41)$$

Substituting Equation (38) into Equation (41) and rearranging the Equation to solve for the tube length, Equation (42) yielded an estimated tube length of approximately 2.3 m.

$$L_t = UA \left[\frac{1}{\eta_o \cdot h_c \left(\frac{A_f}{s} + \pi \cdot D_r \left(1 - \frac{t_f}{s} \right) \right)} + \frac{\ln(D_r/D_{hot})}{2\pi \cdot k_{304}} + \frac{1}{h_h \cdot \pi \cdot D_{hot}} \right] \quad (42)$$

With the tube dimensions and configuration known, the heat exchanger header/nozzle configurations would have to be designed to ensure equal flow to all 21 tubes and to prevent maldistribution. Typically, for air-cooled heat exchangers, the industry standard is some version of a welded rectangular box with a common inlet and outlet port (Thulukkanam 2013). The FTHE shell would likely be welded together and possess the dimensions portrayed in Figure 10. To minimize heat loss to the external environment, sufficient insulation would have to be used to insulate the metal shell. Internal baffle design would also have to be considered to ensure a fluid counter flow. Baffles would not only allow for better heat exchange but would provide necessary tube structural support. Once fully designed, the total system pressure drop on both the primary and secondary sides would have to be evaluated to determine acceptability.

While it is possible to reduce the size of the FTHE by significantly decreasing the tube diameter and increasing the number of tubes, this action would force the use of specialized manufacturing processes. Since the FTHE did not possess a truly compact design, as dictated by requirement number four, another compact heat exchanger technology was explored.

C. PRINTED CIRCUIT HEAT EXCHANGER (PCHE)

PCHEs are also classified as compact heat exchangers and use a diffusion bonding process for construction. This design boasts heat exchanger effectiveness values that can peak around 98% for a single unit. PCHEs are a recommended choice for cryogenic applications since they are designed to withstand pressures up to 62 MPa (9000 psi) and temperatures ranging from cryogenic to 973 K (1292°F) (Thulukkanam 2013). In addition, these heat exchangers can be designed to accommodate counter-flow, parallel-flow, and cross-flow designs, and even a combination of flow types depending on the requirements for heat transfer. Generally, metal plates are chemically etched with the desired fluid flow

channel configuration(s), which are then stacked on each other to achieve the desired thermal capacity. Diffusion bonding these plates results in a single block architecture. This process inherently eliminates the need for gaskets or brazing processes for assembly, which increases the heat exchanger's reliability, availability, and maintainability. The absence of gaskets and brazed surfaces also decreases the magnitude and potential of fluid leakage throughout the system and temperature constraints associated with these standard methods of joining (Thulukkanam 2013). Due to their design, PCHEs have an average expected useful life of up to 30 years when used continuously, with little maintenance required during that period (Heatric 2020).

Fluid flow channels in PCHEs typically possess a semicircular geometry with a diameter of approximately 1–2 mm (Thulukkanam 2013). As the technology of PCHEs matures, new channel designs are being analyzed, which include the inclusion of finned surfaces within the PCHE that can increase the Colburn j -factor and the Nusselt number and thus overall heat exchanger performance (Li et al. 2011). The length, width, and height of the PCHE is based on the required heat load and the allowable pressure drop for the system. There are numerous companies that specialize in manufacturing PCHEs and each company safeguards its individual internal design structure, design parameters, and manufacturing process through patent protection. Given the proprietary nature of PCHE design, a generic illustration of a PCHEs fluid flow and design is shown in Figure 12.

The use of corrosion resistant alloys and materials that can be joined through the diffusion bonding process is imperative in PCHE design. During the diffusion bonding process, individual plates with etched fluid flow channels are joined together by applying high pressure to the plates at an elevated temperature that is below the metal's melting point (Thulukkanam 2013). Grain growth occurs between the plates, forming a high quality weld, where the optimal result is a solid block of stacked etched plates that possess a joint strength comparable to the parent metal strength (Thulukkanam 2013). Alloys of stainless steel, titanium, and copper are commonly employed plate materials; however, material selection can be adjusted based on the intended application (Thulukkanam 2013).

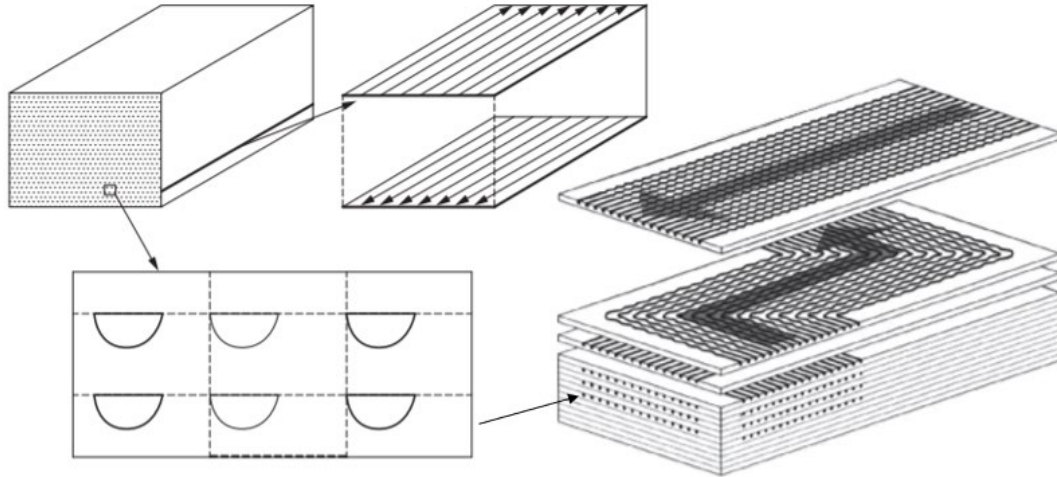


Figure 12. Generic Construction and Fluid Flow of a PCHE. Adapted from Thulukkanam (2013).

Due to the small channel diameter associated with PCHEs, channel fouling and blockage can become a major concern for operations. However, application in a LAESS involves a gas-to-gas application where this effect is minimal. With the addition of the nitrogen membrane assembly in the LAESS, the high purity nitrogen gas further mitigates this negative effect since the gas passes through a complex series of filters and all moisture is removed from the nitrogen prior to entering the heat exchanger. Another concern with the PCHE design is the potential for an unacceptably large pressure drop due to the small channel diameter and tortuous flow paths that are inherent to their design. While not generally unacceptable for high pressure applications, low to moderate pressure applications can cause unacceptable performance (Li et al. 2011).

PCHEs have already been incorporated into large-scale LAESSs in industry such as Highview Power's Pilsworth demonstration LAES plant in the United Kingdom (Highview Power Storage 2015). Potential primary drivers behind the selection of PCHEs for LAESS plants are their compact footprint, resilience to rapidly changing thermal cycles, inherent dynamic response, and the ability to precisely approach a specific temperature (Chen 2018; Thulukkanam 2013). Plate-fin heat exchangers are also a comparable technology in this industry.

Commercial vendors provided details on custom PCHEs for the LAESS to determine the feasibility of a PCHE as an alternative heat exchanger design. These details provided baseline specifications for the product to be used as a comparison to the FTHE. Their analysis resulted in a PCHE constructed of SS304 that would be approximately 4 inches wide, 12 inches long, and 3 inches high and would satisfy the previously defined heat exchanger requirements required to liquify nitrogen based on the existing system components and architecture. Additionally, the PCHE would be pressure tested to ASME code prior to delivery and standard 0.25 inch NPT compression fittings would be installed on both the inlet and outlet of the primary and secondary side. Due to the aforementioned proprietary nature of the PCHE industry, no further specifications were provided other than an estimated weight of 50 lbs.

D. ANALYSIS OF ALTERNATIVES

A comparison of the two alternative heat exchanger designs was performed using a Pugh Matrix (Figure 13) to compare the FTHE and PCHE to the existing coiled tube heat exchanger that was originally in the LAESS. The coiled tube heat exchanger did not provide acceptable performance and the system was never able to produce liquid air (Amalla 2019). The FTHE and the PCHE alternatives were compared to the datum and given an ordinal score based on being better than that datum (+), the same as the datum (0), or worse than the datum (-) using multiple evaluation criteria. The ordinal scores were summed and an overall total score was assigned for each alternative.

Evaluation Criteria	Alternatives		
	Coiled Tube Heat Exchanger	FTHE	PCHE
Foot Print (Volume)	DATUM	+	+
Theoretically Capable of Producing Liquid Nitrogen		+	+
Ease of Integration		0	+
Potential Operational Concerns		-	+
Build to ASME Codes		+	+
Sum of +		3	5
Sum of 0		1	0
Sum of -		1	0
Overall Score	N/A	2	5

Figure 13. Pugh Matrix for Heat Exchanger Decision Making.

The evaluation criteria that were used in the Pugh Matrix were directly traceable to the original requirements for the heat exchanger. Since both alternatives will employ and benefit from the nitrogen membrane assembly for the production of liquid nitrogen, the cost for this component was excluded from the decision making process. To compare the alternatives to the coiled tube heat exchanger (datum), the following evaluation criteria were chosen: the foot print or volume of the heat exchanger, the theoretical capability of the heat exchanger to produce liquid nitrogen, the ease of integration of the heat exchanger into the existing system, the potential operational concerns a design may have, and the ability to build and test the heat exchangers to ASME codes to ensure safety of the system.

The volume of the FTHE design was approximately 1/3rd the volume of the coiled tube heat exchanger and the PCHE was approximately 1/8th the volume of the FTHE. While it is possible to decrease the size of the FTHE to be comparable to the size of the PCHE by significantly decreasing the tube diameter and increasing the number of tubes, this action would force the use of specialized manufacturing processes as mentioned

previously. Despite the difference in size, both the FTHE and the PCHE were scored better than the coiled tube heat exchanger.

The thermodynamic analysis of the FTHE resulted in a plausible design that could produce liquid nitrogen. Although portions of the analysis used approximations due to empirical relationships and assumptions, it was determined that when compared to the previous heat exchanger, the system would be capable of producing liquid nitrogen when operational. The PCHE was designed based on the calculated heat load, entrance and exit temperatures and pressures for the heat exchanger, and the maximum allowable pressure drop. The manufacturer guaranteed performance based on the parameters provided.

The criteria of ease of integration is undoubtedly subjective. However, integration of the FTHE into the LAESS would require considerable effort to fabricate. Additionally, custom fittings would have to be constructed in order to incorporate the FTHE into the LAESS. The current coiled tube heat exchanger also possesses several custom fittings and thus it was determined to be of the same order of magnitude for ease of integration as the datum. Conversely, the PCHE would be manufactured with standard 0.25 inch NPT compression fittings, which would allow for simple integration into the existing system.

The FTHE would have numerous fittings and connections that would require individual pressure tests to ensure no leaks were present. Since the FTHE contained several more fittings and therefore points of failure than the coiled tube heat exchanger, it was scored worse than the datum due to a greater potential for operational concerns. Alternatively, the PCHE contained no gaskets and was diffusion bonded which resulted in minimal points of failure.

Lastly, component safety is an important evaluation criteria since the system operates under high pressure. The FTHE would be constructed of finned tubes manufactured to ASME code and the remaining components and manufacturing processes would also adhere to the same codes. Thus, the FTHE would have a better safety rating than the datum since the datum has been reassemble numerous times since it was delivered. PCHE manufacturers typically hydrostatically or pneumatically pressure test PCHEs prior to shipment to ensure safety and the heat exchanger is thus under guarantee to be built to

satisfy ASME codes. Due to the inherent design and fabrication process of the PCHE, the PCHE was also determined to have a better safety rating than the datum.

Summing the ordinal scores for each evaluation criteria in the Pugh Matrix resulted in the PCHE being designated as the stronger alternative. Cost was explicitly excluded from the Pugh Matrix analysis and considered as an independent variable. A quote for a PCHE was approximately \$26K, which was inclusive of all shipping and manufacturing costs. Based on the number of tubes required in the design, baffle material, case material, external and internal connectors, gasket material, fabrication materials, machining, shipping costs, and labor, the estimated cost to manufacture the FTHE would likely be comparable. Therefore, considering the benefits associated with the PCHE, as shown in the Pugh Matrix, at a comparable cost to the FTHE, the PCHE was selected as the preferred alternative.

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IV. CONCLUSION AND FUTURE WORK

This thesis presented a systems engineering approach to drive the selection of a heat exchanger for an existing modified, small-scale Linde-Hampson cycle LAESS for NPS. The analysis conducted assumes that the systems engineer has some background knowledge in thermodynamics, fluid flow, and heat transfer. However, the process was transparent and the Appendix provides the detailed calculations that were performed to validate the author's conclusions in the event the aforementioned is not the case.

Chapter I presented the motivation and background for this research as well as previous work that other graduate students have performed on the system at NPS. The current LAESS at NPS never created liquid air as intended and previous student analysis of the system determined the root cause to be an improperly designed heat exchanger. Since this technology may help the warfighter in the future by decreasing the DoDs reliance on petroleum-based fuels, the need to further investigate this technology was apparent.

Chapter II discussed the methodology that was used to decompose the system. The status quo LAESS components were described and a concept of operations was developed to illustrate how a small-scale LAESS could be implemented in theater. Requirements were generated for both the nitrogen separator and the heat exchanger by using an IDEF0 to define key system inputs, outputs, mechanisms, and controls for the stages associated with the LAESS cycle. An analysis of alternatives was conducted on two nitrogen separation technologies and compared using a decision evaluation display to select the best alternative. The analysis concluded with the selection of a nitrogen membrane system since it satisfied the nitrogen separation requirements at the lowest cost.

Chapter III contained a high-level design and analysis of two heat exchanger alternatives, a FTHE and a PCHE. Thermodynamic, fluid flow, and heat transfer analysis yielded a FTHE design that would contain 21 staggered tubes with equilateral pitch with each tube being approximately 2.3 m in length. A review of available literature on PCHEs was conducted and commercial vendors provided details on a custom PCHE with dimensional parameters for the second alternative. Using a Pugh Matrix, the two

alternatives were compared to the existing heat exchanger in the system using four evaluation criteria. The analysis of alternatives for the two heat exchanger candidates resulted in the PCHE being selected as the desired alternative due to its extremely compact footprint, capability of producing liquid nitrogen, ease of integration into the existing system, and its inherent safety features at a comparable cost.

Incorporating LAES systems into military microgrids remains a promising avenue to curb the dependence of the DOD on petroleum-based fuels and to achieve the goals defined in the DoDs Operational Energy Strategy. The work presented in this thesis continued to build on previously identified design flaws in the inherited LAESS from NAVFAC in an effort to achieve a fully functional prototype. As a result of this thesis, both the nitrogen membrane assembly and the PCHE were ordered for integration into the LAESS. Future work will include executing the right-hand side of the systems engineering V-model to include subsystem and system integration, test, and evaluation of the new systems components in the existing architecture. Once the LAESS can consistently produce liquid nitrogen, further analysis will have to be conducted on the energy recovery portion of the system. Following the demonstration of a fully operational prototype, operational test and evaluation must be conducted to test the system in an actual operational environment. These tests will determine the LAESSs efficacy, utility, and feasibility in the goal to shift energy sources to more sustainable alternatives.

APPENDIX. CYCLE ANALYSIS CALCULATIONS

Due to three distinct flows in this cryogenic cycle, the mass flows were annotated by subscripted variables versus distinct cycle process steps. These relationships are given by Equations (1)–(3).

$$\dot{m}_a = \dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = \dot{m}_5 \quad (43)$$

$$\dot{m}_b = \dot{m}_6 \quad (44)$$

$$\dot{m}_c = \dot{m}_7 = \dot{m}_8 \quad (45)$$

where the mass flow rate into the FTHE, $\dot{m}_a$, was found using Equations (4) and (5).

$$\dot{V}_a = \frac{5.5 \text{ ft}^3}{\text{min}} \cdot \frac{12^3 \text{ in}^3}{\text{ft}^3} \cdot \frac{0.0254^3 \text{ m}^3}{\text{in}^3} \cdot \frac{\text{min}}{60 \text{ s}} = 2.5957 \times 10^{-3} \frac{\text{m}^3}{\text{s}} \quad (46)$$

The density of nitrogen at an ambient temperature of 290 K is 1.1775 kg/m³ (National Institute of Standards and Technology 2018d). Therefore, the mass flow rate was calculated to be approximately 0.003 kg/s.

$$\dot{m}_a = 2.5957 \times 10^{-3} \frac{\text{m}^3}{\text{s}} \cdot 1.1775 \frac{\text{kg}}{\text{m}^3} = 3.0565 \times 10^{-3} \frac{\text{kg}}{\text{s}} \quad (47)$$

The known thermodynamic properties for nitrogen for process steps 1, 3, 6 and 7 are listed in Table 2 and were used for the cycle analysis.

Table 2. Known Thermodynamic Properties for Cycle Analysis. Adapted from National Institute of Standards and Technology (2018c; 2018b; 2018d).

Thermodynamic Properties							
Process Step 1 - Compressor Inlet				Process Step 3 - Inlet to Primary Side of FTHE			
Property	Symbol	Value	Units	Property	Symbol	Value	Units
Temperature	T1	290	K	Temperature	T3	273	K
Pressure	P1	101325	Pa	Pressure	P3	22752708	Pa
Mass Flow Rate	$\dot{m}_1$	3.0565×10^{-3}	kg/s	Mass Flow Rate	$\dot{m}_3$	3.0565×10^{-3}	kg/s
Enthalpy	h1	300780	J/kg	Enthalpy	h3	240199	J/kg
Viscosity	μ_1	1.7434×10^{-5}	Pa·s	Viscosity	μ_3	2.3940×10^{-5}	Pa·s
Thermal Conductivity	k1	2.5186×10^{-2}	W/m·K	Thermal Conductivity	k3	3.9823×10^{-2}	W/m·K
Specific Heat Capacity	cp1	1041.3	J/kg·K	Specific Heat Capacity	cp3	1393.9	J/kg·K
Process Step 6 - Dewar (Liquid Nitrogen)				Process Step 7 - Inlet to Secondary Side of FTHE			
Property	Symbol	Value	Units	Property	Symbol	Value	Units
Temperature	T6	83.76	K	Temperature	T7	83.76	K
Pressure	P6	202650	Pa	Pressure	P7	202650	Pa
Enthalpy	h6	-108765	J/kg	Enthalpy	h7	81599	J/kg
				Viscosity	μ_7	5.9398×10^{-6}	Pa·s
				Thermal Conductivity	k7	8.2023×10^{-3}	W/m·K
				Density	ρ_7	8.7685	kg/m ³
				Specific Heat Capacity	cp7	1181.6	J/kg·K

The problem was first approached by assigning a control volume (CV1) around the FTHE, the Joule-Thomson (JT) valve, and the dewar. Liquid Yield (Y) is defined as the ratio of $\dot{m}_b$, the mass flow out of the dewar (liquified nitrogen), and $\dot{m}_a$, the mass flow into the FTHE (Howe, Pollman, and Gannon 2018).

$$Y = \frac{\dot{m}_b}{\dot{m}_a} \quad (48)$$

The law of conservation of mass derives a relationship between the mass flow into and out of CV1.

$$\dot{m}_a = \dot{m}_b + \dot{m}_c \quad (49)$$

The mass flow entering CV1 is the same as the cryogenic fluid exiting the dewar and the unliquified portion of the flow exiting the secondary side of the heat exchanger.

The steady-state energy equation leads to the following enthalpy flow relationship, where h is the specific enthalpy for the stated process step.

$$\dot{m}_a h_3 = \dot{m}_b h_6 + \dot{m}_c h_8 \quad (50)$$

Rearranging Equation (7) and substituting it into Equation (8) reveals the relationship between the mass flow through the compressor and the yielded fluid, which gives Equation (9).

$$\dot{m}_a (h_3 - h_8) = \dot{m}_b (h_6 - h_8) \quad (51)$$

Rearranging Equation (9) and noting that h_1 and h_8 are equivalent for an ideal cycle, an expression for the liquid yield of the cycle, Y , is obtained (Howe, Pollman, and Gannon 2018).

$$Y = \frac{\dot{m}_b}{\dot{m}_a} = \frac{h_3 - h_1}{h_6 - h_1} \quad (52)$$

$$Y_{ideal} = \frac{h_3 - h_1}{h_6 - h_1} = \frac{(240199 - 300780) J/kg}{(-108765 - 300780) J/kg} = 0.1479 \quad (53)$$

Using the current operating temperatures and pressures in the cycle, the ideal yield was determined to be approximately 14.8%.

The actual system is not ideal. The cryogenic effectiveness used in this analysis, ε^* , is defined by Equation (12), and reflects that not all heat is transferred from the cold stream to the hot stream (Thulukkanam 2013). This impacts the real liquid yield significantly.

$$\varepsilon^* = \frac{\dot{Q}_{actual}}{\dot{Q}_{max}} = \frac{\dot{m}_c (h_8 - h_7)}{\dot{m}_c (h_1 - h_7)} = \frac{h_8 - h_7}{h_1 - h_7}, \quad (54)$$

where $\dot{Q}_{max}$ is the maximum possible heat transfer rate and $\dot{Q}_{actual}$ is the actual heat transfer rate limited by the return gas flow.

Rearranging Equation (12) allowed for the calculation of the specific enthalpy for the outlet of the secondary side of the heat exchanger (h_8).

$$h_8 = \varepsilon^*(h_1 - h_7) + h_7 \quad (55)$$

$$h_8 = 0.95(300780 - 81599) J/kg + 81599 J/kg = 289.8 kJ/kg \quad (56)$$

Equation (14) resulted in a specific enthalpy, h_8 , of approximately 290 kJ/kg. Using the calculated h_8 and assuming no pressure drop through the secondary side of the heat exchanger, the outlet temperature for the secondary side of the heat exchanger (T_8) was determined to be approximately 279.7 K (National Institute of Standards and Technology 2018e).

By combining Equations (9) and (13), real yield, Y_{real} , is defined in terms of the cryogenic heat exchanger effectiveness in Equation (15). The result was a yield of approximately 12.45%, which is a considerable drop when compared to the ideal yield of 14.8%.

$$Y_{real} = \frac{h_3 - h_7 - \varepsilon^*(h_1 - h_7)}{h_6 - h_7 - \varepsilon^*(h_1 - h_7)} \quad (57)$$

$$Y_{real} = \frac{240199 J/kg - 81599 J/kg - 0.95(300780 - 81599) J/kg}{-108765 J/kg - 81599 J/kg - 0.95(300780 - 81599) J/kg} = 0.1245 \quad (58)$$

Applying a second Control Volume (CV2) around the JT valve, the law of conservation of mass derives a relationship between the specific enthalpies for the flows entering and exiting CV2.

$$h_4 = h_5 \quad (59)$$

Applying a third Control Volume (CV3) around the dewar, the law of conservation of energy derives a relationship between the mass flowrates and enthalpies for the flows entering and exiting CV3.

$$\dot{m}_a h_5 = \dot{m}_b h_6 + \dot{m}_c h_7 \quad (60)$$

Rearranging Equation (18), the specific enthalpy for the exit of the JT valve is defined in Equation (19).

$$h_5 = \frac{\dot{m}_b}{\dot{m}_a} h_6 + \frac{\dot{m}_c}{\dot{m}_a} h_7 \quad (61)$$

Combining Equations (6) and (7), the ratio of mass flow rate exiting the dewar and mass flow rate entering the FTHE is redefined by Equation (20).

$$Y_{real} = 1 - \frac{\dot{m}_c}{\dot{m}_a} \quad (62)$$

With the real yield calculated using the assumed cryogenic effectiveness, Equation (20) was rearranged to solve for the mass flow rate of the nitrogen gas exiting the dewar and entering the secondary side inlet of the FTHE.

$$\dot{m}_c = \dot{m}_a (1 - Y_{real}) \quad (63)$$

$$\dot{m}_c = 3.0565 \times 10^{-3} \text{ kg/s} (1 - 0.1245) = 2.676 \times 10^{-3} \text{ kg/s} \quad (64)$$

Substituting Equations (7) and (20) into Equation (19), the specific enthalpy for the JT valve exit was determined to be approximately 57.9 kJ/kg.

$$h_5 = (Y_{real} h_6) + (1 - Y_{real}) h_7 \quad (65)$$

$$h_5 = (0.1245 \cdot -108765 \text{ J/kg}) + (1 - 0.1245) 81599 \text{ J/kg} = 57.9 \text{ kJ/kg} \quad (66)$$

Applying the relationship defined by Equation (17), the thermodynamic properties for the outlet of the primary side of the heat exchanger (process step 4) were found using the NIST nitrogen tables as shown in Table 3.

Table 3. Thermodynamic Properties of Nitrogen at Process Step 4. Adapted from National Institute of Standards and Technology (2018f).

Thermodynamic Properties			
Process Step 4 - Outlet of Primary Side of FTHE			
Property	Symbol	Value	Units
Temperature	T4	162.91	K
Pressure	P4	22752708	Pa
Enthalpy	h4	57900	J/kg
Viscosity	μ_4	4.0994×10^{-5}	Pa·s
Thermal Conductivity	k4	6.4300×10^{-2}	W/m·K
Specific Heat Capacity	cp4	1919.8	J/kg·K

With the mass flow rate of the nitrogen gas entering the secondary side inlet of the FTHE calculated, Equation (12) was rearranged to solve for the actual heat loss rate, assuming a cryogenic heat exchanger effectiveness of 95%. Equation (25) yielded a heat loss rate of approximately 557 W.

$$\dot{Q}_{actual} = \varepsilon^* \cdot \dot{Q}_{max} = \varepsilon^* \cdot \dot{m}_c (h_1 - h_7) \quad (67)$$

$$\dot{Q}_{actual} = 0.95 (2.676 \times 10^{-3} \text{ kg/s}) (300780 - 81599) \text{ J/kg} = 557.2 \text{ W} \quad (68)$$

With the thermodynamic properties calculated for all applicable process steps in the LAESS cycle, the number of tubes to achieve laminar flow on the primary side of the FTHE was calculated. The Reynolds number is defined in Equation (27) in terms of the hydraulic diameter (D_h), the mass flow rate, the viscosity, and the minimum free flow area (A_o) (Thulukkanam 2013).

$$\text{Re} = \frac{D_h \cdot \dot{m}}{\mu \cdot A_o}, \quad (69)$$

where A_o is a function of the number of tubes in the FTHE (N_t) and the tube inner diameter (D_{hot}) (Thulukkanam 2013).

$$A_o = \frac{\pi \cdot D_{hot}^2}{4} N_t \quad (70)$$

For the primary side of the FTHE, the hydraulic diameter was equal to the inner diameter of the tube. Therefore, Equations (27) and (28) were rearranged in terms of the number of tubes required to maintain laminar flow.

$$N_t = \frac{4 \cdot \dot{m}_a}{\pi \cdot D_{hot} \cdot \mu_3 \cdot Re} \quad (71)$$

$$N_t = \frac{4 \cdot 3.0565 \times 10^{-3} \text{ kg/s}}{\pi (4.7625 \times 10^{-3} \text{ m}) (2.394 \times 10^{-5} \text{ Pa} \cdot \text{s}) (1800)} = 18.96 \quad (72)$$

The number of tubes required on the primary side to maintain laminar flow, assuming a Reynolds number of 1800 and considering the flow into the primary side of the heat exchanger, was determined to be 19 tubes. Secondary side analysis was conducted in a similar manner. However, it was found that the primary side was more limiting than the secondary side.

Since 19 tubes did not provide for a desirable symmetric staggered tube-bundle arrangement, it was determined that 21 tubes would be used instead. Rearranging Equation (30), Equation (31) is used to find the actual Reynolds Number of 1625, based on the flow entering the primary side of the heat exchanger and the number of tubes used for the FTHE design.

$$Re = \frac{4 \cdot \dot{m}_a}{\pi \cdot D_{hot} \cdot \mu_3 \cdot N_t} \quad (73)$$

$$Re = \frac{4 \cdot 3.0565 \times 10^{-3} \text{ kg/s}}{\pi (4.7625 \times 10^{-3} \text{ m}) (2.394 \times 10^{-5} \text{ Pa} \cdot \text{s}) (21)} = 1625 \quad (74)$$

With the number of tubes determined, the length of the tubes was also desired to be found. The heat exchanger effectiveness-number of transfer units (ϵ -NTU) method, an

analysis of the Nusselt number, heat exchanger geometrical relationships, and the inclusion of the fin effects were used to determine the required length of the tubes.

First, the ε -NTU method was used to solve for the overall heat transfer coefficient of the heat exchanger (UA). The heat capacity rate ratio is defined in Equation (33) (Thulukkanam 2013).

$$C^* = \frac{C_{\min}}{C_{\max}}, \quad (75)$$

where $C_{\min}$ and $C_{\max}$ were associated with the smallest and largest temperature changes, respectively, and were calculated using Equations (34) and (35) (Thulukkanam 2013).

$$C_{\min} = \frac{\dot{Q}_{\max}}{(T_{h,i} - T_{c,i})} \quad (76)$$

$$C_{\max} = \frac{\varepsilon^* \cdot \dot{Q}_{\max}}{(T_{h,i} - T_{h,o})} \quad (77)$$

The heat capacity rate ratio was calculated to be 0.65 using Equation (33).

$$C^* = \frac{(300780 - 81599) J/kg / (290 - 83.76) K}{0.95 \cdot (300780 - 81599) J/kg / (290 - 162.91) K} = 0.649 \quad (78)$$

The heat exchanger was modeled as a counter-flow heat exchanger. The NTU for $C^* < 1$ was calculated to be approximately 5.8 using Equation (37) (Incropera et al. 2007).

$$NTU = \frac{1}{C^* - 1} \ln \left(\frac{\varepsilon^* - 1}{(\varepsilon^* \cdot C^*) - 1} \right) \quad (79)$$

$$NTU = \frac{1}{0.649 - 1} \ln \left(\frac{0.95 - 1}{(0.95 \cdot 0.649) - 1} \right) = 5.8 \quad (80)$$

The NTU was defined as the ratio of the overall heat transfer coefficient of the heat exchanger to the minimum heat capacity rate as shown in Equation (39).

$$NTU = \frac{UA}{C_{\min}} \quad (81)$$

By rearranging Equation (39), the overall heat transfer coefficient was determined to be approximately 0.9 W/K per tube.

$$UA = NTU \cdot C_{\min} \quad (82)$$

$$UA = 5.8 \cdot \frac{(300780 - 81599) J/kg}{(290 - 83.76) K} \cdot \left(\frac{1}{21 \text{ tubes}} \right) = 0.897 W/K \cdot \text{tube} \quad (83)$$

With the overall heat transfer coefficient calculated, the Nusselt numbers for the primary side and secondary side of the FTHE were calculated to find the primary and secondary side convective heat transfer coefficients. Assuming a uniform heat flux wall boundary condition, fully developed laminar flow, and smooth, round tubes, Equation (42) provided the definition for the primary side Nusselt number (Thulukkanam 2013).

$$Nu_h = \frac{h_h \cdot D_h}{k_h} = 4.364 \quad (84)$$

Rearranging Equation (42) gives Equation (43), which yielded a primary convective heat transfer coefficient of approximately 36.5 W/m²·K using the hot inside diameter of the tube as the hydraulic diameter and the thermal conductivity of the fluid entering the heat exchanger.

$$h_h = \frac{Nu_h \cdot k_3}{D_{hot}} \quad (85)$$

$$h_h = \frac{4.364 \cdot (3.9823 \times 10^{-2} W/m \cdot K)}{4.7625 \times 10^{-3} m} = 36.49 W/m^2 \cdot K \quad (86)$$

Solving for the secondary side convective heat transfer coefficient was not as simple. The flow characteristics associated with circular helical fin designs are complicated; however, numerous empirical methods have been developed to predict the flow characteristics for circular fin designs using specific core geometries and flow parameters. The author chose to implement the Briggs and Young Correlation based on a regression model to support all tube bank and fin height configurations for circular tubes. The crimped fin interface was ignored and assumed to be completely circular. In order to use this correlation, a staggered, equilateral pitch core geometry with six or greater tube rows had to be incorporated into the design. Associated core geometric parameters were defined as shown in Figure 10 where L_2 and L_3 are the respective heat exchanger width and height dimensions, P_t is the lateral pitch, and P_L is the longitudinal pitch.

Equation (45) is used to calculate the Reynolds Number based on the tube outer diameter (fin root effective diameter) for use in the Briggs and Young Correlation (Thulukkanam 2013).

$$\text{Re} = \frac{D_r \cdot \rho_7 \cdot V_{\max}}{\mu_7}, \quad (87)$$

where $V_{\max}$ is the maximum velocity of the flow through the finned tube bank and is a function of the minimum free flow area (A_o) through the tube bank, the frontal area (A_{fr}), and the upstream fin side fluid velocity (Thulukkanam 2013).

$$V_{\max} = \frac{\dot{m}_c}{\rho_7 \cdot L_2 \cdot L_3} \left(\frac{A_{fr}}{A_o} \right) \quad (88)$$

$$A_{fr} = P_t \cdot \left(\frac{12in}{ft} \right) = 0.75in \cdot \left(\frac{12in}{ft} \right) = 9in^2/ft \quad (89)$$

$$A_o = \left[(P_t - D_r) \cdot \left(\frac{12in}{ft} \right) \right] - \left[2 \cdot \left(\frac{\#fins}{in} \right) \cdot \left(\frac{12in}{ft} \right) \cdot t_f \cdot L_f \right] \quad (90)$$

$$A_o = \left[(0.75in - 0.25in) \cdot \left(\frac{12in}{ft} \right) \right] - \left[2 \cdot \left(\frac{6fins}{in} \right) \cdot \left(\frac{12in}{ft} \right) \cdot \left(\frac{3}{200}in \right) \cdot \left(\frac{3}{16}in \right) \right] = 5.595in^2/ft \quad (91)$$

$$V_{max} = \frac{2.676 \times 10^{-3} kg/s}{(8.7685 kg/m^3) \cdot (0.13335m) \cdot (0.06668m)} \left(\frac{9}{5.595} \right) = 0.055 m/s \quad (92)$$

$$Re = \frac{(6.35 \times 10^{-3} m) \cdot (8.7685 kg/m^3) \cdot (0.055 m/s)}{5.9397 \times 10^{-6} Pa \cdot s} = 517.57 \quad (93)$$

Substituting in the variables into Equation (46) yielded a maximum velocity of approximately 0.055 m/s. With V_{max} calculated, Equation (45) resulted in a secondary side Reynolds number of approximately 518.

The Briggs and Young Correlation is used for Reynolds Numbers ranging from 1,000–20,000 (Thulukkanam 2013). Acknowledging that the calculated secondary side Reynolds Number is outside of this range, the correlation is used to obtain an estimate of 11.5 for the secondary side Nusselt number in Equation (52) (Thulukkanam 2013).

$$Nu_c = 0.134 Re_d^{0.681} Pr_7^{1/3} \left(\frac{s}{L_f} \right)^{0.2} \left(\frac{s}{t_f} \right)^{0.1134} \quad (94)$$

$$Nu_c = 0.134(517.57)^{0.681} \left(\frac{(1181.6 \frac{J}{kg \cdot K}) \cdot (5.9397 \times 10^{-6} Pa \cdot s)}{8.2023 \times 10^{-3} \frac{W}{m \cdot K}} \right)^{1/3} \left(\frac{4.2333 \times 10^{-3} m}{4.7625 \times 10^{-3} m} \right)^{0.2} \left(\frac{4.2333 \times 10^{-3} m}{3.81 \times 10^{-4} m} \right)^{0.1134} = 11.51 \quad (95)$$

Using the fin root effective diameter as the hydraulic diameter, the secondary side convective heat transfer coefficient was found using Equation (54) and yielded a primary convective heat transfer coefficient of approximately 14.9 W/m²·K.

$$h_c = \frac{Nu_c \cdot k_7}{D_r} \quad (96)$$

$$h_c = \frac{(11.51) \cdot (8.2023 \times 10^{-3} W/m \cdot K)}{6.35 \times 10^{-3} m} = 14.869 \frac{W}{m^2 \cdot K} \quad (97)$$

With the primary and secondary convective heat transfer coefficients calculated, the effects from the fins had to be considered. The primary purpose of fins is to increase the total surface area to increase the heat transfer rate. The effectiveness of a fin is a function of the fin's geometrical and metallurgical properties. A fin's thickness, length, shape, and composition determine the efficiency of the fin, which is a ratio of the actual heat transfer rate from the fin to the secondary side fluid to the maximum rate of heat transfer possible (Theodore 2011). If the fin efficiency achieves unity, it is perfectly isothermal in that the temperature of the fin surface is equivalent to the temperature of the base of the fin (Theodore 2011). The thermal conductivity of SS304 assumed for analysis was 11.45 W/m·K, which was based on an average of the secondary side entrance and exit temperatures (National Institute of Standards and Technology 2020).

Considering the FTHE fins as annular fins, the fin efficiency was approximated graphically in Figure 11. In order to use the graphical approach, additional geometrical variables and ratios had to be defined. These variables were the corrected outside radius (r_{2c}), the corrected height (L_c), the profile area (A_p), and a dimensionless relationship between the three variables (X) (Theodore 2011). The equations for these variables are embedded into Figure 11. Using the above variables, the fin efficiency was approximated to be 89%.

$$r_{2c} = r_f + \frac{t_f}{2} = \left(\frac{1.5875 \times 10^{-2} m}{2} \right) + \left(\frac{3.81 \times 10^{-4} m}{2} \right) = 8.128 \times 10^{-3} m \quad (98)$$

$$L_c = L_f + \frac{t_f}{2} = 4.7625 \times 10^{-3} m + \left(\frac{3.81 \times 10^{-4} m}{2} \right) = 4.953 \times 10^{-3} m \quad (99)$$

$$A_p = L_c \cdot t_f = (4.953 \times 10^{-3} m) \cdot (3.81 \times 10^{-4} m) = 1.887 \times 10^{-6} m \quad (100)$$

$$X = L_c^{3/2} \left(\frac{h_c}{k_{304} \cdot A_p} \right)^{1/2} = (4.953 \times 10^{-3} m)^{3/2} \cdot \left(\frac{14.869 \frac{W}{m^2 \cdot K}}{11.45 \frac{W}{m \cdot K} \cdot 1.881 \times 10^{-6} m} \right)^{1/2} = 0.289 \quad (101)$$

$$\frac{r_{2c}}{r_r} = \frac{8.128 \times 10^{-3} m}{\left(\frac{6.35 \times 10^{-3} m}{2} \right)} = 2.56 \quad (102)$$

With the fin efficiency approximated, additional fin parameters were calculated in order to determine the required length of the tubes (L_t) in the FTHE. The surface area of the fin (A_f) is found using Equation (61) (Incropera et al. 2007).

$$A_f = 2\pi \left(r_{2c}^2 - \left(\frac{D_r}{2} \right)^2 \right) \quad (103)$$

$$A_f = 2\pi \left((8.128 \times 10^{-3} m)^2 - \left(\frac{6.35 \times 10^{-3} m}{2} \right)^2 \right) = 3.5176 \times 10^{-4} m^2 \quad (104)$$

In addition to the surface area of the fin, the area of the prime surface (A_b) and the total surface area of the tube (A_t) are found using Equations (63) and (64) (Incropera et al. 2007).

$$A_b = \pi \cdot D_r \left(L_t - \frac{L_t \cdot t_f}{s} \right) \quad (105)$$

$$A_t = \frac{L_t \cdot A_f}{s} + A_b \quad (106)$$

Substituting Equation (63) into (64), obtains an expression for the total surface area of the tube in terms of L_t .

$$A_t = L_t \left[\frac{A_f}{s} + \pi \cdot D_r \left(1 - \frac{t_f}{s} \right) \right] \quad (107)$$

With the pertinent areas defined, Equation (66) defines the overall fin efficiency (Incropera et al. 2007).

$$\eta_o = 1 - \frac{L_t \cdot A_f}{s \cdot A_t} (1 - \eta_f) \quad (108)$$

Substituting Equation (65) into Equation (66) yielded an expression for overall fin efficiency of approximately 92%.

$$\eta_o = 1 - \left(\frac{A_f}{A_f + \pi \cdot D_r (s - t_f)} \right) (1 - \eta_f) \quad (109)$$

$$\eta_o = 1 - \left(\frac{3.5176 \times 10^{-4} m^2}{3.5176 \times 10^{-4} m^2 + \left[\pi \cdot (6.35 \times 10^{-3} m) \cdot (4.2333 \times 10^{-3} m - 3.81 \times 10^{-4} m) \right]} \right) (1 - 0.89) = 0.923 \quad (110)$$

Neglecting effects from surface fouling and accounting for fin effects on the secondary side, the overall heat transfer coefficient is redefined in terms of the thermal resistances of the primary and secondary fluids as well as the thermal resistance for the cylindrical tube wall in Equation (69) (Incropera et al. 2007).

$$\frac{1}{UA} = \frac{1}{\eta_o \cdot h_c \cdot A_t} + \frac{\ln(D_r/D_{hot})}{2\pi \cdot k_{304} \cdot L_t} + \frac{1}{h_h \cdot \pi \cdot D_{hot} \cdot L_t} \quad (111)$$

Substituting Equation (65) into Equation (69), and rearranging the Equation to solve for the tube length, Equation (70) yielded an estimated tube length of approximately 2.3 m.

$$L_t = UA \left[\frac{1}{\eta_o \cdot h_c \left(\frac{A_f}{s} + \pi \cdot D_r \left(1 - \frac{t_f}{s} \right) \right)} + \frac{\ln(D_r/D_{hot})}{2\pi \cdot k_{304}} + \frac{1}{h_h \cdot \pi \cdot D_{hot}} \right] \quad (112)$$

$$L_t = 0.897 \frac{W}{K} \left[\frac{1}{(0.923) \left(14.869 \frac{W}{m^2 \cdot K} \right) \left(\frac{3.5176 \times 10^{-4} m^2}{4.2333 \times 10^{-3} m} + \left[\pi \cdot (6.35 \times 10^{-3} m) \cdot \left(1 - \frac{3.81 \times 10^{-4} m}{4.2333 \times 10^{-3} m} \right) \right] \right)} + \frac{\ln(6.35 \times 10^{-3} m / 4.7625 \times 10^{-3} m)}{2\pi \cdot (11.45 \frac{W}{m \cdot K})} + \frac{1}{\pi \cdot (36.49 \frac{W}{m^2 \cdot K}) \cdot (4.6725 \times 10^{-3} m)} \right] = 2.292 m \quad (113)$$

LIST OF REFERENCES

- Barron, Randall F., and Gregory F. Nellis. 2016. *Cryogenic Heat Transfer*. 2nd ed. Boca Raton, FL: CRC Press. <https://doi.org/10.1201/b20225>.
- Blanchard, Benjamin, and Wolter Fabrycky. 2010. *Systems Engineering and Analysis*. 5th ed. New York: Pearson.
- CAMEO Chemicals. 2020. "Oxygen Gas, Refrigerated Liquid, Oxidizing, N.O.S." Accessed October 25, 2020. <https://cameochemicals.noaa.gov/chemical/4145>.
- Chen, Minghui. 2018. "Performance Testing and Modeling of Printed Circuit Heat Exchangers for Advanced Nuclear Reactor Applications." PhD diss., University of Michigan. <http://hdl.handle.net/2027.42/143945>.
- Department of Defense. 2011. *Energy for the Warfighter: Operational Energy Strategy*. Washington, DC: Assistant Secretary of Defense for Operational Energy, Plans & Programs. <https://www.secnav.navy.mil/eie/ASN%20EIE%20Policy/DODOperationalEnergyStrategy.pdf>.
- . 2016. *Department of Defense: 2016 Operational Energy Strategy*. Washington, DC: Office of the Assistant Secretary of Defense for Energy, Installations and Environment. https://www.acq.osd.mil/eie/Downloads/OE/2016%20OE%20Strategy_WEBd.pdf.
- . 2019. MIL-STD-1472G CHANGE 1, Department of Defense Design Criteria Standard Human Engineering. Washington, DC: Department of Defense.
- Eady, David S., Steven B. Siegel, R. S. Bell, and Scott H. Dicke. 2009. *Sustain the Mission Project: Casualty Factors for Fuel and Water Resupply Convoys*. Arlington, VA: Army Environmental Policy Institute.
- Fin Tube Products, Inc. 2020. "CLADFIN® Finned Tube." Accessed November 8, 2020. <http://www.fintube.com/cladfin-nickel-chrome-coating.html>.
- Flynn, Thomas. 2004. *Cryogenic Engineering*. 2nd ed. Boca Raton, FL: CRC Press. <https://doi.org/10.1201/9780203026991-9>.
- GENERON. 2020a. "How to Separate Nitrogen from Air – Nitrogen Extraction from Air." <https://www.generon.com/how-separate-nitrogen-air-extraction/>.
- . 2020b. "PSA Nitrogen Generator." Accessed October 25, 2020. https://www.generon.com/product_type/psa-nitrogen-generator/.
- Granger, Robert A. 2012. *Fluid Mechanics*. Newburyport: Dover Publications.

- Greenley, Heather L. 2019. *Department of Defense Energy Management: Background and Issues for Congress*. Washington, DC: Congressional Research Service. <https://fas.org/sgp/crs/natsec/R45832.pdf>.
- Heatric. 2020. “Heatric Printed Circuit Heat Exchangers.” Accessed December 16, 2020. <https://www.heatric.com/heat-exchangers/>.
- Highview Power Storage. 2015. “Highview’s 5MW Liquid Air Energy Storage Demonstrator Starting Operations This Winter.” <https://www.highviewpower.com/wp-content/uploads/2018/04/15.08.19-HPS-Press-Release-Pilsworth-Update-19.08.15.pdf>.
- Howe, Todd A. 2018. “Thermodynamic System Analysis of a Liquid Air Energy Storage System.” Master’s Thesis, Monterey, CA, Naval Postgraduate School. <https://calhoun.nps.edu/handle/10945/59687>.
- Howe, Todd A., Anthony G. Pollman, and Anthony J. Gannon. 2018. “Operating Range for a Combined, Building-Scale Liquid Air Energy Storage and Expansion System: Energy and Exergy Analysis.” *Entropy* 20 (10): 770. <https://doi.org/10.3390/e20100770>.
- Incropera, Frank P., David P. Dewitt, Theodore L. Bergman, and Adrienne S. Lavine. 2007. *Introduction to Heat Transfer*. 5th ed. Hoboken, NJ: John Wiley & Sons, Incorporated.
- Kanoglu, Mehmet, Ibrahim Dincer, and Marc A. Rosen. 2008. “Performance Analysis of Gas Liquefaction Cycles.” *International Journal of Energy Research* 32 (1): 35–43. <https://doi.org/10.1002/er.1333>.
- Li, Qi, Gilles Flamant, Xigang Yuan, Pierre Neveu, and Lingai Luo. 2011. “Compact Heat Exchangers: A Review and Future Applications for a New Generation of High Temperature Solar Receivers.” *Renewable and Sustainable Energy Reviews* 15 (9): 4855–75. <https://doi.org/10.1016/j.rser.2011.07.066>.
- NANO Purification Solutions. 2020a. “High-Purity Membrane Nitrogen Generators.” Accessed October 25, 2020. <https://www.n-psi.com/nmg-membrane-nitrogen-generators>.
- . 2020b. “Ultra-High Purity Nitrogen Generators.” Accessed October 29, 2020. <https://www.n-psi.com/gen2-nitrogen-generators>.
- National Institute of Standards and Technology. 2018a. “Argon.” 2018. <https://webbook.nist.gov/cgi/cbook.cgi?ID=C7440371&Units=SI&Mask=E8F>.

- . 2018b. “Isobaric Properties for Nitrogen.” 2018. https://webbook.nist.gov/cgi/fluid.cgi?Action=Load&ID=C7727379&Type=IsoBar&Digits=6&P=.202650&THigh=100&TLow=75&TInc=.5&RefState=DEF&TUnit=K&PUnit=MPa&DUnit=kg%2Fm3&HUnit=kJ%2Fkg&WUnit=m%2Fs&VisUnit=Pa*s&STUnit=N%2Fm.
- . 2018c. “Isobaric Properties for Nitrogen.” 2018. https://webbook.nist.gov/cgi/fluid.cgi?Action=Load&ID=C7727379&Type=IsoBar&Digits=6&P=22.7527&THigh=274&TLow=271&TInc=1&RefState=DEF&TUnit=K&PUnit=MPa&DUnit=kg%2Fm3&HUnit=kJ%2Fkg&WUnit=m%2Fs&VisUnit=Pa*s&STUnit=N%2Fm.
- . 2018d. “Isobaric Properties for Nitrogen.” 2018. https://webbook.nist.gov/cgi/fluid.cgi?Action=Load&ID=C7727379&Type=IsoBar&Digits=6&P=0.101325&THigh=291&TLow=289&TInc=1&RefState=DEF&TUnit=K&PUnit=MPa&DUnit=kg%2Fm3&HUnit=kJ%2Fkg&WUnit=m%2Fs&VisUnit=Pa*s&STUnit=N%2Fm.
- . 2018e. “Isobaric Properties for Nitrogen.” 2018. https://webbook.nist.gov/cgi/fluid.cgi?Action=Load&ID=C7727379&Type=IsoBar&Digits=6&P=.202650&THigh=280&TLow=278&TInc=.1&RefState=DEF&TUnit=K&PUnit=MPa&DUnit=kg%2Fm3&HUnit=kJ%2Fkg&WUnit=m%2Fs&VisUnit=Pa*s&STUnit=N%2Fm.
- . 2018f. “Isobaric Properties for Nitrogen.” 2018. https://webbook.nist.gov/cgi/fluid.cgi?Action=Load&ID=C7727379&Type=IsoBar&Digits=6&P=22.752708&THigh=163&TLow=162&TInc=.01&RefState=DEF&TUnit=K&PUnit=MPa&DUnit=kg%2Fm3&HUnit=kJ%2Fkg&WUnit=m%2Fs&VisUnit=Pa*s&STUnit=N%2Fm.
- . 2018g. “Nitrogen.” 2018. <https://webbook.nist.gov/cgi/cbook.cgi?ID=C7727379&Mask=4>.
- . 2018h. “Oxygen.” 2018. <https://webbook.nist.gov/cgi/cbook.cgi?ID=C7782447&Mask=4>.
- . 2020. “Material Properties: 304 Stainless (UNS S30400).” Accessed December 10, 2020. https://trc.nist.gov/cryogenics/materials/304Stainless/304Stainless_rev.htm.
- Nitro-Turbodyne, Inc. 2016a. “Process Flow Diagram (P&ID) LAESS.” Revision 2. Nitro-Turbodyne, Inc., Cheyenne, WY.
- . 2016b. “Operations Manual Liquid Air Energy Storage System (LAESS).” Under Noble Supply Purchase Order P659307 For Naval Facilities Command Engineering and Expeditionary Warfare Center. August 12, 2016.

- RIX Industries. 2016. "Operating Instructions and Parts List for RIX Air Compressor Model SA-6B." RIX Industries, Benicia, CA.
- Theodore, Louis. 2011. *Heat Transfer Applications for the Practicing Engineer*. Hoboken, UNITED STATES: John Wiley & Sons, Incorporated.
<http://ebookcentral.proquest.com/lib/ebook-nps/detail.action?docID=693262>.
- Thulukkanam, Kuppan. 2013. *Heat Exchanger Design Handbook*. 2nd ed. Boca Raton, Fla: CRC Press.
- U.S. Department of Transportation. 2020. "Systems Engineering for Intelligent Transportation Systems." Accessed October 1, 2020. <https://ops.fhwa.dot.gov/publications/seitsguide/section3.htm>.
- Wong, George S. K., and Tony F. W. Embleton. 1984. "Variation of Specific Heats and of Specific Heat Ratio in Air with Humidity." *The Journal of the Acoustical Society of America* 76 (2): 555–59. <https://doi.org/10.1121/1.391597>.

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