Thermal Transport and Heat Exchanger Design for the Space Molten Salt Reactor Concept

THESIS

Presented in Partial Fulfillment of the Requirements for the Degree Master of Science in the Graduate School of The Ohio State University

By

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2012

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Abstract

Surface power and nuclear electric propulsion in space necessitate the development of high energy density, long term continuous power sources. Research at The Ohio State University under the NASA Ralph Steckler Space Grant Colonization Research and Technology Development Opportunity has identified molten salt reactors (MSRs) as a potentially appealing technology for high power, high temperature space fission systems. This thesis examines component specific design related to thermal transport in an attempt to further establish the feasibility of MSRs in space. Specifically, the optimization of the associated heat exchangers for a Brayton power cycle is discussed in detail. In addition, power cycle and secondary coolant selection, material selection, and general MSR design considerations are discussed.

The primary heat exchanger was optimized to maximize the margin to super prompt critical (MSPC), with a final MSPC equal to 513 pcm. The secondary and tertiary heat exchangers were optimized to minimize helium pressure drop, with a combined pressure loss of 2.62 kPa obtained.
Acknowledgements

I would like to thank Dr. Blue for allowing me to work on the Steckler project both as an undergraduate and as a graduate student, and for all of the guidance he has provided as the project expanded and evolved. I would like to thank Dr. Sun for the guidance and knowledge that he bestowed upon me throughout the years in his field of expertise, fluid mechanics and heat transfer. Without it, this research would not have turned out as it did. I appreciate the help from Michael Eades and all of the other students that were involved with the Steckler at some point. The teamwork of our group was exceptional. I would also like to thank Dr. Denning for his role as my undergraduate faculty adviser and convincing me to continue on to higher education at OSU. I must thank my parents, Ernest and Diane Flanders, for all of their love and support, and for guiding me in the right direction. And last, but certainly not least, I would like to thank my friends and my girlfriend for their understanding when I was pulling late nights and was unavailable.
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# Table of Contents

Abstract ....................................................................................................................... ii

Acknowledgements ................................................................................................. iii

List of Figures .......................................................................................................... vii

List of Tables ............................................................................................................ ix

1. Introduction ........................................................................................................... 1
   1.1. Problem Statement ......................................................................................... 1
   1.2. Project Objectives ......................................................................................... 1
   1.3. Background on Molten Salt Reactors ............................................................. 5
   1.4. System Overview ........................................................................................... 13

2. General Design Considerations ........................................................................... 15
   2.1. Shielding Considerations and Core Geometry ................................................. 15
   2.2. Fuel Selection ............................................................................................... 21
   2.3. Power Cycle Selection ................................................................................... 21
   2.4. Material Selection ......................................................................................... 23
   2.5. Secondary Coolant Selection ....................................................................... 24
   2.6. Fluid Thermophysical Properties .................................................................. 26

3. Heat Exchanger Optimization .............................................................................. 32
   3.1. Optimization Using fminsearch() ................................................................. 33
   3.2. Primary Heat Exchanger ............................................................................... 35
### 3.3. Secondary Heat Exchanger

---

3.4. Tertiary Heat Exchanger

---

### 4. Future Work

---

4.1. Future Use of Codes

---

4.2. Mission Specific Parameters

---

4.3. Limitations of Optimization

---

4.4. Secondary Coolant Considerations

---

4.5. Offset-Strip Fin Heat Exchanger Testing

---

4.6. Manufacturing of Components

---

### 5. Conclusions

---

5.1. Optimization Procedures

---

5.2. Closing Remarks

---

**References**

---

90
List of Figures

Figure 1: (Left) Optimal energy technology defined by duration of use and electrical output (Right)

Solar flux as a function of distance away the sun (Source: [1]) ......................................................... 4

Figure 2: Solid fueled reactor (U235) equivalent beta fraction for a MSR .................................................. 10

Figure 3: Power conversion loop overview .................................................................................................. 14

Figure 4: Effect of core and PHX on shadow shield shape ............................................................................. 16

Figure 5: CAD model of reactor core geometry for the SMSR ................................................................. 18

Figure 6: SMSR core CFD with inlet flanges ................................................................................................. 20

Figure 7: Effect of increasing inlet angles. From left to right – 0°, 15°, 30°, 45° ............................................ 20

Figure 8: Tube in shell heat exchanger (Source: [11]) ................................................................................. 36

Figure 9: CAD rendering of the PHX ........................................................................................................... 46

Figure 10: Section view of PHX .................................................................................................................... 47

Figure 11: Offset-strip fin heat exchanger configuration (Source: [17]) ......................................................... 48

Figure 12: Offset-strip fin heat exchanger dimensionless parameters (Source: [18]) ................................. 49

Figure 13: Effect of loss parameter beta on radiator area and total mass ..................................................... 52

Figure 14: SHX helium unit cell ................................................................................................................... 53

Figure 15: Two unit cells of helium side by side with 2 unit cells of lithium on top and bottom, viewing

downstream ..................................................................................................................................................... 54

Figure 16: Inlet and outlet concept for lithium and helium ............................................................................ 61

Figure 17: Side view of SHX depicting flow path of helium ........................................................................ 62

Figure 18: Section view of the helium inlet plenum ..................................................................................... 63

Figure 19: Section view of lithium inlet plenum .......................................................................................... 64

Figure 20: Top view of helium outlet header (left) and inlet header (right) ................................................... 65

vii
Figure 21: CAD model of helium inlet between plenum and helium channels ........................................65
Figure 22: Top view of lithium inlet header (left) and outlet header (right) ........................................66
Figure 23: CAD model of lithium outlet between plenum and lithium channels ................................66
Figure 24: Side view of centerline temperature profile for configuration 2 .......................................75
Figure 25: Helium exit temperature profile for modeled portion of configuration 2 .........................76
List of Tables

<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Table 1</td>
<td>Power conversion systems considered</td>
<td>23</td>
</tr>
<tr>
<td>Table 2</td>
<td>Summary of liquid metal melting and boiling temperatures at 1 ATM</td>
<td>26</td>
</tr>
<tr>
<td>Table 3</td>
<td>Thermophysical properties for LiF-UF$_4$ (65-35) at 1450 K</td>
<td>27</td>
</tr>
<tr>
<td>Table 4</td>
<td>Thermophysical properties of liquid lithium evaluated at 1344 K</td>
<td>28</td>
</tr>
<tr>
<td>Table 5</td>
<td>Thermophysical properties of helium at 15 bar</td>
<td>30</td>
</tr>
<tr>
<td>Table 6</td>
<td>Thermophysical properties for NaK evaluated at 556 K</td>
<td>31</td>
</tr>
<tr>
<td>Table 7</td>
<td>SMSR heat exchanger figures of merit</td>
<td>33</td>
</tr>
<tr>
<td>Table 8</td>
<td>PHX optimization parameters and optimization range</td>
<td>41</td>
</tr>
<tr>
<td>Table 9</td>
<td>PHX pressure drop terms, in the order of upstream to downstream</td>
<td>44</td>
</tr>
<tr>
<td>Table 10</td>
<td>PHX operating conditions</td>
<td>45</td>
</tr>
<tr>
<td>Table 11</td>
<td>PHX configuration specifications</td>
<td>46</td>
</tr>
<tr>
<td>Table 12</td>
<td>SHX optimization parameters and optimization range</td>
<td>55</td>
</tr>
<tr>
<td>Table 13</td>
<td>Helium pressure loss terms for the SHX, listed in the order they occur</td>
<td>70</td>
</tr>
<tr>
<td>Table 14</td>
<td>SHX operating conditions</td>
<td>71</td>
</tr>
<tr>
<td>Table 15</td>
<td>SHX specifications</td>
<td>71</td>
</tr>
<tr>
<td>Table 16</td>
<td>Three CFD configurations compared to MATLAB model</td>
<td>74</td>
</tr>
<tr>
<td>Table 17</td>
<td>Results of FLUENT and MATLAB comparisons</td>
<td>75</td>
</tr>
<tr>
<td>Table 18</td>
<td>THX operating conditions</td>
<td>79</td>
</tr>
<tr>
<td>Table 19</td>
<td>THX specifications</td>
<td>80</td>
</tr>
</tbody>
</table>
1. Introduction

1.1. Problem Statement

Research at The Ohio State University under the NASA Ralph Steckler Space Grant Colonization Research and Technology Development Opportunity has identified molten salt reactors (MSRs) as a potentially appealing technology for high power, high temperature space fission systems.[1] This project continues to develop the Space Molten Salt Reactor concept (SMSR) by examining application of the technology to the specific applications of 500kWe for surface power, 3MWe for surface power, and 15MWe for nuclear electric propulsions (NEP). Here, surface power refers to a power system deployed on an extra-terrestrial surface for electricity production or process heat. This thesis focuses on the thermal transport and power conversion systems of the 500kWe variant of the SMSR. The analytical methods and computer codes presented in this thesis will also aid in the study of SMSRs for other applications.

1.2. Project Objectives

Through the design of an effective thermal transport and power conversion system, my research explored the unique aspects of using molten salt reactor technology for space applications. Specifically, my work focused on the design and
optimization of the primary, secondary, and tertiary heat exchangers, a major step in validating the feasibility of the SMSR concept. In addition other topics covered include the advantages and disadvantages of the SMSR concept in comparison to a traditional solid fuel reactor, CFD validation of analytical methods, and a guide on using the CFD package FLUENT for applications relevant to SMSR development.

A major result of the research is an optimization tool for heat exchanger design, consisting of several codes developed using MATLAB. The intent of this tool is to allow further development of the space fission system, without requiring future researchers to redesign the heat exchanger. The codes are packaged in such a way that future researchers simply need to modify the input parameters prior to running the code. Using computational fluid dynamics, the results were validated and the procedure is documented in section 3.

This thesis covers the design and optimization of components required to convert the thermal output of the reactor into usable electricity, while also discussing some aspects of core design and fuel selection. Specifically, the following designs will be discussed in detail.

- Coolant loops selection
- Power cycle selection
- Heat exchangers selection/optimization
• Power cycle temperature/pressure

1.2.1. Fission Technology in Space

Fission technology in space is an active area of research at NASA. This is largely due to the inherent high energy densities and long operational lifetimes compared to other energy sources available for space. The technology is not immature, with over 30 nuclear reactors already having been launched into space by the Soviet and American space programs. If the world plans to undertake lengthy missions in space, fissions power systems are an attractive option.

Figure 1 contains two graphics. The leftmost one depicts approximate regions on a chart of power and lifetime where various energy technologies become advantageous over another for space applications. Where the SMSR is proposed to operate, at 500 kWe and above, for journeys on the order of years, it is clear that nuclear fission systems are the most suitable. The other graphic in Figure 1 shows the sharp decrease in solar flux as a function of distance away from the sun, which is not taken into account in the left graphic, further enhancing the benefit of fission systems alone for deep space missions. These graphics are from a NASA presentation to the Department of Energy’s Nuclear Research Advisory Committee [2]. Similar graphics have been presented in NASA’s draft Space Technology Road
Maps and the International Atomics Agency’s report “The Role of Nuclear Power and Nuclear Propulsion in the Peaceful Exploration of Space”.

![Diagram showing optimal energy technology and solar flux as a function of distance away the sun.](source: [1])

Space applications also benefit from lighter, more compact power systems. Every kilogram of system mass must be launched off the earth at considerable expense. Also, additional mass causes NEP systems to suffer from decreased overall system performance. The SMSR concept can yield a light and compact power system because of its compact core and efficient thermal transport system.
1.3. Background on Molten Salt Reactors

1.3.1. Fuel Structure and Heat Removal

A traditional light water reactor is comprised of fuel rods that are arranged with space in between. This space is occupied by the metal fuel bundle structure and moving water, which functions as both the moderator and coolant. The shape and spacing of a solid fueled reactor core is governed largely by the need to remove heat from the fuel rods to prevent overheating. MSRs have a fundamentally different structure because they do not have solid fuel rods. In an MSR, the fuel exists as a homogenous solution of fissile material and liquefied salt, which acts as both fuel and coolant.

The SMSR is not purposefully moderated like a traditional light water reactor. Some moderation occurs in the salt, but not enough to thermalize a significant portion of the neutrons. In a molten salt reactor where a thermal neutron spectrum is desired, solid in-core moderators such as graphite could be used. The SMSR is a fast reactor, so moderation is limited to what is inherent in the salt and Beryllium-oxide external reflectors.

One advantage the SMSR has in comparison to traditional solid fueled reactors is that the fuel salt is continuously pumped out of the core to the heat exchanger, while the cooled fuel salt is returned directly into the core. The removal of hot fuel salt from the core allows the peak fuel temperature to be extracted. This
contrasts with solid fueled reactors, where the fuel pin temperature peaks in the center, leaving a cooler surface for heat transfer. The result is a higher temperature hot reservoir for the Carnot cycle, allowing higher cycle efficiency. Also, by removing the necessity for in core heat transfer from a solid fuel to a circulating coolant, a major reduction in internal structure is allowed, producing a more compact core, which would also have lower mass.

1.3.2. Liquid Fuel and Reactivity

Another advantage of MSRs over traditional solid fueled reactors is the large negative temperature feedback coefficient associated with them. This property makes MSRs inherently stable. When liquid fuel heats up, thermal expansion causes fuel salt to be pushed out of the core into an accumulator outside of the core. Less fuel in the core results in a reduction in $k_{\text{eff}}$, and thermal power subsequently decreases. With heat being removed in the heat exchanger, the fuel temperature will approach an equilibrium value. In the event of extreme fuel heating, the fuel salt can boil, producing voids that rapidly decrease power levels. Boiling, however, is to be avoided because the system must be able to handle the pressure spike associated with it.

When a fission occurs in a MSR, the fission products are directly deposited in the fuel salt. The liquid nature of a MSR’s fuel allows for the removal of fission
products, which have an unfavorable effect on reactivity and thermal properties. Being a liquid, the fuel salt can be pumped through an online reprocessing or batch reprocessing system, where the fission products can be extracted with relative ease compared to a traditional solid fuel reactor.

One particularly desirable isotope for removal is Xenon-135, which exists as a gas in the fuel salt. Removing it provides two main benefits to the core reactivity. First, Xe-135 has an extremely high neutron cross-section when compared to the other fission products. By removing it, fewer neutrons will be absorbed, leaving a larger neutron inventory for fission. Second, noble gasses like Xenon have very low solubilities in molten salts and can form bubbles if the concentration of these gasses becomes too high. A bubble in the core is essentially a void in the fuel, reducing the quantity of fuel in the core, decreasing the reactivity.

1.3.3. Controllability and Delayed Neutrons

While fissions are occurring in a nuclear reactor, most of the neutrons are produced almost instantaneously. A small fraction of neutrons are created some time after the fission from the decay of fission products. The neutrons that are released after a fission event are called delayed neutrons, and the fission products that produce them are called precursor nuclei.
In a MSR, delayed neutrons are created both in the core and out of the core. A precursor nuclei within the fuel salt that decays while outside the core, has a very low probability of causing fission. For purposes of contributing to fission and controlling the reactor, the neutron is wasted when the precursor nuclei decays outside the core. Returning the fuel to the core in a timely fashion is therefore an important task in MSR design. This is obtained through efficient piping and more importantly, effective heat exchanger design.

Delayed neutrons are essential to the control of the reactor. The half-lives of the delayed neutron precursor nuclei range from less than a quarter of a second to 54 seconds. The delay in neutron generation after a fission event means that reactor power changes on the scale of seconds to minutes for small changes of reactivity. This is as opposed to a hypothetical reactor that operates on only prompt neutrons, where power level changes on the scale of microseconds for even small changes of reactivity. It is only because of the existence of delayed neutrons that nuclear reactors can be controlled with mechanical mechanisms and external feedback systems.

The larger the ratio of delayed neutrons to prompt neutrons in the core, the slower the reactor power will change with a reactivity insertion, and the larger the amount of reactivity that can be inserted into the core before becoming super
prompt critical, which refers to a state where the reactor is critical on prompt neutrons alone. In this state, the reactor period would be on the order of the average neutron life, which is between $10^{-6}$ and $10^{-5}$ seconds in the SMSR. Super prompt criticality creates a rapid power excursion, which can have devastating effects on the reactor. It is believed that the accident at Chernobyl in 1986 was a case of super prompt criticality.

The Margin to Super Prompt Critical (MSPC) is defined as the amount of reactivity, in pcm (mille-percent), that must be inserted for a critical reactor to become super prompt critical. The maximum possible MSPC occurs when 100% of the delayed neutrons are released within the core, while the minimum MSPC is 0, indicating that the reactor is super prompt critical. Using 6 delayed neutron groups, the MSPC can be calculated from knowing the time spent in the core and the time spent out of the core, as displayed in equation 1. Here, $\tau_c$ is the time spent in the core and $\tau_{hx}$ is the time spent out of the core. [3]

$$MSPC = \sum_{i=1}^{6} \left( \frac{\beta_i \times \lambda_i}{\lambda_i + \tau_c^{-1} \left(1 - \exp\left(-\tau_{hx} \lambda_i\right)\right)} \right) \times 10^5 \text{ [pcm]}$$ (1)

Figure 2 displays the solid fueled reactor equivalent beta fraction for the SMSR at varying fuel residency times. The MSPC is proportional to this value, with a maximum MSPC corresponding to an equivalent beta fraction of 1. There is a trend of increasing beta fraction moving towards increased in core time and reduced out
of core time. When the time out of the core approaches zero, this means that fuel isn’t being pumped out, just like in a solid fuel reactor. At this location, the equivalent beta fraction is equal to unity, indicating that the beta fraction is the same as for a solid fuel reactor. Similarly, when the time out of the core approaches infinity, the equivalent beta fraction is near zero, but never reaching zero, because there will always be some delayed neutrons that are created before the fuel exits the core. That is, unless the time in the core approaches zero. In that case, the equivalent beta fraction would approach zero.

![Figure 2: Solid fueled reactor (U$_{235}$) equivalent beta fraction for a MSR.](image)

1.3.4. Design Challenges of MSRs

When discussing molten salt reactors, it is important to discuss the design challenges posed by the unique technology. First, the corrosive nature of fluoride
salts limits the options for materials than can be used. Combine those restrictions with the high temperature, radioactive environment, and the options are severely limited.

Pumps and valves must be specially designed, both in material and functionality, to withstand the environment. The low electrical conductivity of molten salts prevents EM pumps from being used, as is common in many space fission systems. This represents a disadvantage because centrifugal pumps have moving parts that must be able to maintain integrity in the high temperature, corrosive environment.

On terrestrial molten salt reactors, valving was accomplished by melting and freezing portions of the fuel salt. This was the solution used in the Molten Salt Reactor Experiment (MSRE) at Oakridge National Lab. [4] Although it was found to be effective, these freeze valves impose design challenges for space. First, a method for cooling and heating the salt within the valve must be available. This means that a coolant loop must be implemented for each valve. For space applications, the additional mass of these coolant loops is undesirable, and may necessitate the development of mechanical valves for molten salts.

Fluoride salts also have poor thermal properties, such as a high viscosity and low thermal conductivity. The issues associated with poor thermal properties are
largely offset by the fact that fission energy is deposited directly into the fuel. The poor thermal properties greatly increase the need for an effective heat exchanger design when transporting the heat away from the fuel.

1.3.5. Terrestrial Benefits
The benefits associated with the development of nuclear power for space applications are not limited to space. Improvements to terrestrial power systems can be expected both directly through development and indirectly through the advancement of knowledge about molten salt fission systems.

Due to the nature of the thorium fuel cycle, MSR’s are ideal for thorium breeder reactors. Liquid fuel makes it easy to chemically separate $^{233}$U from the breeding sections, so it can be pumped into the fissioning sections. While the SMSR uses $^{235}$U as the fuel, its development promotes advancement in both molten salt and thorium technology by exploring the MSR design space and increasing the body of knowledge about MSR design.

Specifically, my work improves the knowledge of thermal transport systems for MSR’s. The design process of a molten salt to lithium heat exchanger, to maximize the MSPC, is something that has not been explored before. Also, liquid metal to helium heat exchangers are valuable for a variety of uses, including but not limited to high efficiency Brayton cycles for both terrestrial and space applications.
The development of high temperature, compact fission power systems would benefit industries for mobile electricity generation, such as naval transportation, as well as the development of modular reactors. In order to maximize the efficiency of a Carnot cycle on earth, where the cold sink temperature is essentially fixed, the maximum operating temperature must be increased. Currently, materials are a limiting factor for high temperature nuclear applications. Beyond the issue of melting temperature and creep, these materials must be compatible with high neutron fluxes while at high temperatures.

1.4. System Overview
A diagram of the SMSR’s thermal systems can be seen in Figure 3. The SMSR’s thermal systems are composed of four loops. The first loop contains the molten salt with fissile uranium dissolved in it. The loop traverses between the core and the primary heat exchanger (PHX). Within the PHX heat is transferred to the secondary coolant loop of liquid lithium. The liquid lithium travels from the PHX to the secondary heat exchanger (SHX), where the heat is transferred to the third coolant, which is the helium working fluid of the Brayton loop. The fourth coolant loop acts as the heat removal system for the Brayton cycle using the tertiary heat exchanger (THX). Heat is removed from the helium and given to a NaK loop, which delivers heat to the radiator system.
Figure 3: Power conversion loop overview
2. General Design Considerations

Due to the uniqueness of MSRs, special considerations must be made to ensure a successful design is produced. This section will deal with the design considerations not related to the heat exchangers, such as the core geometry, fuel selection, power cycle and secondary coolant selection. Section 2.6 will discuss the design of the heat exchangers in detail.

2.1. Shielding Considerations and Core Geometry

The general shape of a SMSR core is different from that of a solid fueled reactor. Instead of having metal cladding encapsulating fuel pellets, the homogenous mixture of fuel and salt exists within a chamber with minimal internal structure. Since the fuel is effectively the primary coolant, there is no need for coolant channels to pass through the core. This lack of internal components has a positive effect on $k_{eff}$. Any additional material within the core acts to absorb neutrons, which is undesirable because absorptions reduce the neutron availability.

If minimizing the core volume were the only requirement, the core could be designed as a reflected sphere. While this would benefit neutronics, two other design considerations drive the shape of the SMSR’s core geometry. First, thermal hydraulics benefits from having uniform flow distribution between the inlets and outlets to keep all portions of the fuel moving through the core. Hot spots are less dense and having them decreases $k_{eff}$ because less fuel is present. Second, the mass
of the radiation shield is strongly affected by the shape of the core. This is the case for all space fission systems.

![Diagram](image)

**Figure 4: Effect of Core and PHX on shadow shield shape**

All system components such as electrical equipment, the Brayton loop, and heat rejection loops, must be shielded from the radiation produced in the fuel salt. A shadow shield accomplishes this with minimum mass. Since the fuel salt travels into the PHX, both the core and the PHX must be located within the shielded region, formed by the cone of the shadow shield. By positioning the core and the PHX close together, the region requiring shielding can be kept small. As seen in Figure 4, the size of the radiation shield must be sufficient to block all mission critical systems.
from the radiation created and scattered by the core and PHX. The orange color represents radiologically “hot” components, while the blue represents radiologically “cool” components. It is not depicted in the drawing, but the pipes carrying coolant between the PHX and SHX must travel in such a path that a hole does not exist for radiation to escape.

The reactor core geometry was designed to account for the thermal hydraulics and reactor shielding. The main fuel chamber comprises a cylindrical pipe surrounded by a concentric annulus, as seen in Figure 5. Eight inlet pipes transport fuel into the top portion of the annulus. After passing down through the annulus, the fuel travels back up the central cylindrical pipe to the four outlet pipes. Although having no internal structure would be ideal for a reactor of this sort, the decision to enter and exit the core on the same side was made to reduce shielding requirements. By orienting all pipes on one end of the core, the widest portion that needs to be shielded is the outer diameter of the core and reflector. Since the shielded region is cone shaped, the piping is located at a larger diameter of shielded area.
2.1.1. Stagnant Fuel Issues

One design consideration for MSRs that is very different from solid fueled reactors is the concept of stagnant fuel. In a solid fueled reactor, the fuel is stationary, requiring only that coolant continue flow over the fuel rods. A MSR core must be designed to facilitate the continuous movement of fuel. If a region were to stagnate for a period of time, that portion of the liquid fuel would continue to generate heat and could potentially boil or otherwise damage the core structure, in addition to having a negative effect on $k_{\text{eff}}$. During the development of the SMSR core, several designs were considered in an attempt to prevent fuel stagnation.
Using FLUENT, simulations were developed to check the flow velocity at all regions. The areas between the inlets were of the biggest concern; the fuel had no reason to move through there. The first attempt at correction was to increase the number of inlet pipes, decreasing the spacing inlet flow. While there was a difference, this hardly corrected the situation, and the additional pressure drop from more inlets was undesirable. The second attempt has the opening of the inlet pipes spread out, much like a brass instrument. The motivation was that this would allow the flow to spread out to the low flow regions and even out the flow distribution. The result was opposite of what was expected. The widening of the inlet pipes at the opening decreased the flow velocity, resulting in significantly worse stagnation, as seen in Figure 6.

The final solution to this problem was obtained by angling the inlet pipes 45°, creating a vortex of molten salt that traveled through the core. CFD analysis was run for inlet pipes oriented at 0°, 15°, 30°, and 45°. As Figure 7 displays, the regions of stagnation were reduced as the inlet angle was increased. At 45°, the stagnation region is essentially nonexistent.
Figure 6: SMSR core CFD with inlet flanges

Figure 7: Effect of increasing inlet angles. From left to right – 0°, 15°, 30°, 45°
2.2. **Fuel Selection**

Uranium-235 was selected as the fissile material to fuel the SMSR. U-233 was not chosen on the basis that it does not occur naturally like U-235 and must be produced in a reactor by the transmutation of Thorium-232. Plutonium-239 was not chosen due to a lower solubility in fluoride salts and because of the large biological hazard it would pose in a launch accident scenario. Initially in the project, the fuel was selected to be a mixture of FLiNaBe-UF₄. Further research identified the mixture of LiF-UF₄ to be superior in a few important respects. The solubility of UF₄ in LiF is higher than in FLiNaBe. Another superior feature is that LiF moderates neutrons less, because of the exclusion of Be, allowing for the spectrum to remain predominantly in the fast region. LiF also has a lower viscosity than FLiNaBe due to the exclusion of Be and Na atoms. The thermal conductivity is also slightly higher in LiF, again due to the exclusion of heavier atoms like Be and Na. Lastly, vapor pressure relations exist for LiF-UF₄, which allows the determination of boiling points in various environments. One negative aspect of both of these fuels is the high cost of Li-7, which must be used to prevent the production of He-4 and H-3, as well as gamma production.

2.3. **Power Cycle Selection**

Several options for power cycles were considered: Potassium Rankine, Closed-Brayton, Water Rankine, and Stirling cycles. A trade study was performed
and selection was made by evaluating the cycles in terms of the following factors: efficiency, scalability, technological readiness, and system mass.

Water Rankine was rejected immediately because of the temperature and pressure limitations associated with the production of steam. Also, because the maximum temperature and efficiency is limited by cycle considerations, overall system mass increases significantly. Material corrosion from the water at the extreme temperatures and pressures need for a space fission system would also be an undesirable effect.

Potassium Rankine cycles are a promising technology for high temperature, high efficiency power systems on earth. The technological readiness level of these systems is fairly inadequate when compared to the large body of research conducted on Brayton systems. The technological readiness is further reduced for space applications because of the difficulty of modeling two-phase flow of potassium in micro-gravity. For this reason, the potassium Rankine cycle was not selected.

The Stirling cycle is another promising technology for high temperature, high efficiency energy production systems. For systems on the order of 10 kWe or less, the Stirling cycle has been shown to have a lower system mass than the Brayton system. Between 10 kWe and 20 kWe, Brayton and Stirling cycles have been evaluated to have comparable masses. As the considered power level increases
above 20 kWe, as the SMSR is proposed to operate, the mass of Brayton systems become significantly lower than for Stirling Systems.[5]

Ultimately, the closed-Brayton system was selected based on high efficiency, the lowest mass among other options, and its high technological readiness level.

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<thead>
<tr>
<th>Cycle</th>
<th>Rank</th>
<th>Reason</th>
</tr>
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<tr>
<td>Potassium Rankine</td>
<td>3</td>
<td>Low tech-readiness</td>
</tr>
<tr>
<td>Closed-Brayton</td>
<td>1</td>
<td>High efficiency, low mass, high tech-readiness</td>
</tr>
<tr>
<td>Water Rankine</td>
<td>4</td>
<td>Low efficiency, low temperature, high mass</td>
</tr>
<tr>
<td>Stirling</td>
<td>2</td>
<td>Higher mass than Brayton above 20 kWe[5]</td>
</tr>
</tbody>
</table>

2.4. **Material Selection**

The SMSR poses some very unique requirements on materials. The material used for the core and the heat exchangers must be able to withstand temperatures up to 1500 K, high flux fast neutron environments, and the corrosive nature of molten salts. An alloy composed of molybdenum (95%)–rhenium (5%) was selected on this basis and modeled as pure molybdenum for heat transfer purposes. Molybdenum alloys are employed in high temperature applications exceeding 1500K and have been tested in Molten Fuel Salts at 1370K.[6] Using creep data from Conway and Flagella, and the Larson-Miller parameter analysis, the creep rate was calculated to be less than 0.5% in 20 years at a stress of 300 kPa.[7] In the fast
spectrum, the effect of molybdenum on the reactivity of the core is minimal. It has also been shown to be very resistant to molten salt corrosion, with applications in coating steels with molybdenum to prevent corrosion. [8]

2.5. Secondary Coolant Selection

After selecting a Brayton cycle, the challenge remains to transfer the heat from the fuel salt to the helium in the loop. While it is advantageous from an efficiency standpoint to transfer the heat directly between the fuel salt and the helium, a secondary loop to transfer heat between the fuel salt and the helium was included. This secondary loop would allow a pressure transition between the low-pressure fuel salt and the high-pressure helium, reducing creep within the PHX. A secondary loop also allows the heat exchangers to each be optimized for a different parameter without conflicting. The PHX can be optimized for MSPC, while the SHX can be optimized for helium pressure drop.

In order to promote heat transfer, liquid metals with low Prandtl numbers were explored. An effective liquid metal must have a melting temperature low enough to be able to practically melt the coolant in the loop with electrical heat, while the boiling temperature needs to be sufficiently high as to not limit the maximum helium temperature. Currently, material considerations limit the turbine inlet temperature to around 1350 K. A summary of examined liquid metals can be
seen in Table 2. Clearly, beryllium and calcium are impractical because they melt at temperatures well above 1000 K. Even magnesium, with a melting temperature of 923 K, would result in a severe engineering challenge to fully melt the coolant before the reactor starts up. Of the three remaining coolants, lithium, sodium, and potassium, both sodium and potassium would require significant pressurization to remain a liquid up to 1350 K.

Another important factor to take into consideration is the effect of a neutron environment on these substances. Only lithium contains a naturally occurring isotope, Li-7, with an extremely low fast neutron cross-section. Enrichment is required to bring naturally occurring lithium from 92.5% Li-7 to approximately 99.7%. Although expensive, enriched Li-7 would provide excellent heat transfer without imposing limitations on the helium operating temperature, while eliminating the risk of H-3 and He-4 production.
Table 2: Summary of liquid metal melting and boiling temperatures at 1 Atm

<table>
<thead>
<tr>
<th>Liquid Metal</th>
<th>Melting Temperature (K)</th>
<th>Boiling Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lithium</td>
<td>453.7</td>
<td>1615</td>
</tr>
<tr>
<td>Sodium</td>
<td>370.9</td>
<td>1156</td>
</tr>
<tr>
<td>Potassium</td>
<td>336.5</td>
<td>1032</td>
</tr>
<tr>
<td>Magnesium</td>
<td>923</td>
<td>1363</td>
</tr>
<tr>
<td>Calcium</td>
<td>1115</td>
<td>1757</td>
</tr>
<tr>
<td>Beryllium</td>
<td>1560</td>
<td>2742</td>
</tr>
</tbody>
</table>

2.6. Fluid Thermophysical Properties

2.6.1. Fuel Salt

For design considerations related to neutronics, the fuel salt has a high percentage of uranium. The inclusion of heavy uranium nuclei results in a reduced thermal conductivity, reduced viscosity, increased density, and reduced specific heat capacity. With a 35% molar contribution of UF\(_4\), LiF-UF\(_4\) has the following thermophysical properties.

\[
\rho_f = 10^3 \left( \frac{2108 - 0.663T_f + 4.68 \times 10^{-5}T_f^2}{376.856 - 0.0662T_f} \right) \left[ \frac{\text{kg}}{\text{m}^3} \right] \tag{2}
\]

\[
k_f = 5 \times 10^{-4} T_f - 0.0856 \left[ \frac{\text{W}}{\text{m} \cdot \text{K}} \right] \tag{3}
\]

\[
\mu_f = \frac{10^{-1} e^{-0.987 \frac{T_f}{T_f}}}{108.96 + \frac{16.9}{T_f}} \left[ \frac{\text{Pa} \cdot \text{s}}{\text{m}^2} \right] \tag{4}
\]

\[
\mu_f = \frac{10^{-1} e^{-0.987 \frac{T_f}{T_f}}}{108.96 + \frac{16.9}{T_f}} \left[ \frac{\text{Pa} \cdot \text{s}}{\text{m}^2} \right]
\]

26
\( C_{p,f} = 816 \left[ \frac{J}{kg \cdot K} \right] \) \hspace{1cm} (5)

A summary of these properties at 1450 K can be seen in Table 3. Note the rather poor thermal conductivity, at 0.6394 W/m-K, which significantly impacts the PHX design.

Table 3: Thermophysical properties for LiF-UF\(_4\) (65-35) at 1450 K

<table>
<thead>
<tr>
<th>Property</th>
<th>Calculated Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel composition</td>
<td>LiF-UF(_4) (65-35) (99.7%w 7Li 93.15%w 235U)</td>
</tr>
<tr>
<td>Melting temp</td>
<td>860 K</td>
</tr>
<tr>
<td>Boiling temp @ 1 atm</td>
<td>2081 K</td>
</tr>
<tr>
<td>Density</td>
<td>4432 kg/m(^3)</td>
</tr>
<tr>
<td>Specific heat capacity</td>
<td>816 J/kg-K</td>
</tr>
<tr>
<td>Viscosity</td>
<td>0.0025 Pa-s</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>0.6394 W/m-K</td>
</tr>
<tr>
<td>Prandtl number</td>
<td>3.126</td>
</tr>
</tbody>
</table>

2.6.2. Lithium

Metals are unique in that the valence electrons are free to move within the material, allowing the conduction of electricity and heat to take place very readily. This phenomenon gives metals their high thermal conductivities. As a liquid, metals have very low Prandtl numbers, on the order of 10\(^{-3}\), which is much lower than water, with a Prandtl number of around 7. The implication of this is that the Nusselt
number is not heavily dependent on the Reynolds number, because heat is readily conducted through the fluid.

Specifically, lithium has the following thermophysical properties. [9]

\[
\rho_i = 10^3 \times (0.538 - 0.0160(T_i \times 10^{-3}) - 0.100(T_i \times 10^{-3})^2 + 0.0546(T_i \times 10^{-3})^3 \\
- 0.0151(T_i \times 10^{-3})^4 + 0.00270(T_i \times 10^{-3})^5 - 0.000315(T_i \times 10^{-3})^6 \quad \left[ \text{kg/m}^3 \right]
\]

(6)

\[
k_i = 24.8 + 45 \times 10^{-3}T_i - 11.6 \times 10^{-6}T_i^2 \quad \left[ \text{W/(m-K)} \right]
\]

(7)

\[
\log(\mu_i) = -4.164 - 0.6374 \times \ln(T_i) + \frac{292.1}{T_i} \quad \left[ \text{Pa-s} \right]
\]

(8)

\[
C_{pi} = \frac{10^3 \times (31.227 + 0.205 \times 10^6 T_i^{-2} - 5.265 \times 10^{-3} T_i + 2.628 \times 10^{-6} T_i^2)}{6.941} \quad \left[ \text{J/(kg-K)} \right]
\]

(9)

Table 4 displays these properties evaluated at \( T_i \) equal to the mean lithium temperature.

Table 4: Thermophysical properties of liquid lithium evaluated at 1344K

<table>
<thead>
<tr>
<th>Lithium Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>429 kg/m^3</td>
</tr>
<tr>
<td>Conductivity</td>
<td>64.3 W/m-K</td>
</tr>
<tr>
<td>Kinematic viscosity</td>
<td>1.96E-4 kg/m-s</td>
</tr>
<tr>
<td>Specific heat</td>
<td>4200 J/kg-K</td>
</tr>
<tr>
<td>Prandtl number</td>
<td>0.013</td>
</tr>
</tbody>
</table>
2.6.3. Helium

Helium is a widely used working fluid for a closed Brayton cycle. Its relatively high thermal conductivity also makes it an excellent gas for heat transfer, compared to gases such as nitrogen, and carbon dioxide.

Specifically, helium has the following thermophysical properties, with temperatures in Kelvin and pressures in bar.[10]

\[
\rho_h = \frac{0.176 P_h}{T_h} \left( 1 + 0.53 \times 10^{-3} \left( \frac{P_h}{T_h} \right)^{1.2} \right) \left[ \frac{\text{kg}}{\text{m}^3} \right]
\]  \hspace{1cm} (10)

\[
k_h = 0.144 \left( 1 + 2.7 \times 10^{-4} P_h \right) \left( \frac{T_h}{273} \right)^{0.7} \left( 1 - 2 \times 10^{-4} P_h \right) \left[ \frac{\text{W}}{\text{m} \cdot \text{K}} \right]
\]  \hspace{1cm} (11)

\[
\mu_h = 1.885 \times 10^{-5} \left( \frac{T_h}{273} \right)^{0.7} \left[ \text{Pa} \cdot \text{s} \right]
\]  \hspace{1cm} (12)

\[
C_{p,h} = 5195 \left[ \frac{\text{J}}{\text{kg} \cdot \text{K}} \right]
\]  \hspace{1cm} (13)

The thermophysical properties of helium at 15 bar are displayed in Table 5 for the minimum, maximum, and mean temperatures of helium within the SHX. The density nearly halves as the helium is heated, while the thermal conductivity nearly doubles and the viscosity increases significantly.
Table 5: Thermophysical properties of helium at 15 bar

<table>
<thead>
<tr>
<th>Property</th>
<th>729 K</th>
<th>1040 K</th>
<th>1350 K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m$^3$)</td>
<td>0.992</td>
<td>0.695</td>
<td>0.535</td>
</tr>
<tr>
<td>Thermal conductivity (W/m-K)</td>
<td>0.290</td>
<td>0.373</td>
<td>0.448</td>
</tr>
<tr>
<td>Viscosity (Pa-s)</td>
<td>3.75E-5</td>
<td>4.81E-5</td>
<td>5.77E-5</td>
</tr>
<tr>
<td>Specific heat (J/kg-K)</td>
<td></td>
<td>5195</td>
<td></td>
</tr>
<tr>
<td>Prandtl number</td>
<td>0.672</td>
<td>0.670</td>
<td>0.669</td>
</tr>
</tbody>
</table>

2.6.4. NaK

The heat rejection loop operates at temperatures below the freezing point of lithium and other metals. The NaK (22%-78%) eutectic, which melts at 260 K, is used for this loop. Specifically, the thermal properties of NaK are [9]

$$\rho_{\text{NaK}} = \left( \frac{0.22}{\rho_{\text{Na}}} + \frac{0.78}{\rho_{\text{K}}} \right)^{-1} \left[ \frac{\text{kg}}{\text{m}^3} \right]$$

(14)

where the density of sodium and potassium are

$$\rho_{\text{Na}} = 0.897 + 0.516(T \times 10^{-3}) - 1.83(T \times 10^{-3})^2 + 2.20(T \times 10^{-3})^4$$

$$-1.40(T \times 10^{-3})^4 + 0.449(T \times 10^{-3})^5 - 0.058(T \times 10^{-3})^6$$

(15)

$$\rho_{\text{K}} = 0.903 - 0.170(T \times 10^{-3}) - 0.269(T \times 10^{-3})^2 - 0.506(T \times 10^{-3})^4$$

$$-0.465(T \times 10^{-3})^4 + 0.204(T \times 10^{-3})^5 - 0.035(T \times 10^{-3})^6$$

(16)
\[ k_{NaK} = 15.0 + 30.29 \times 10^{-3} T + 1.12 \times 10^{-3} T^2 + \\
- 0.774 \times 10^{-6} T^3 + 0.200 \times 10^{-9} T^4 \left[ \frac{W}{m \cdot K} \right] \]  

(17)

\[ 10^8 \nu_{NaK} = 200.8 - 0.735 T + 1.12 \times 10^{-3} T^2 + \\
- 0.774 \times 10^{-6} T^3 + 0.20 \times 10^{-9} T^4 \left[ \frac{m^2}{s} \right] \]  

(18)

\[ C_{P,NaK} = 873 \left[ \frac{J}{kg \cdot K} \right] \]  

(19)

The thermophysical properties for NaK are evaluated at the mean temperature of 556 K within the THX in Table 6.

**Table 6: Thermophysical properties for NaK evaluated at 556 K**

<table>
<thead>
<tr>
<th>NaK Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>666 kg/m³</td>
</tr>
<tr>
<td>Conductivity</td>
<td>25.4 W/m-K</td>
</tr>
<tr>
<td>Kinematic viscosity</td>
<td>1.66e-4 kg/m-s</td>
</tr>
<tr>
<td>Specific heat</td>
<td>873 J/kg-K</td>
</tr>
<tr>
<td>Prandtl number</td>
<td>0.0057</td>
</tr>
</tbody>
</table>
Critical to the ultimate goal of a functional SMSR is the development of heat exchangers that allow the system to function effectively. For this reason, significant effort was spent investigating potential options for heat exchangers. Heat exchangers can have multiple direct and indirect effects on the overall system mass. As it was explained in section 2.1, a PHX that extends outside of the initial shielded region would require a larger region to be shielded. The heat exchanger mass can also be substantial if the design is ineffective. Furthermore, the heat exchanger shape must connect efficiently to the reactor core to prevent excess piping and the associated increase in pumping head.

As described in section 1.3.3, the MSPC is very important for the SMSR. The PHX will be designed to maximize the MSPC while maintaining reasonable pressure drops and dimensions, described in detail in section 3.2. The SHX will be designed to have a minimum pressure drop, while maintaining reasonable dimensions and mass, as described in detail in section 0.

The figures of merit for the heat exchangers have been defined in Table 7. Some of these figures of merit are more significant than others. MSPC and helium pressure drop were the optimization parameters for the PHX and SHX respectively. This does not, however, mean that the final designs selected were for the minimum MSPC and helium pressure drop configurations. The MATLAB optimization
produced a collection of local minimum configurations for the PHX and SHX. By applying the selection criteria to these configurations, the most desirable was selected.

<table>
<thead>
<tr>
<th>Figures of merit</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Margin to super prompt critical</td>
<td>PHX - optimization parameter</td>
</tr>
<tr>
<td>Low pressure drop</td>
<td>SHX - optimization parameter (helium)</td>
</tr>
<tr>
<td>Compact shape</td>
<td>PHX – limits set</td>
</tr>
<tr>
<td>Compact shape</td>
<td>Handled case by case</td>
</tr>
<tr>
<td>Low mass</td>
<td>As low as reasonably achievable</td>
</tr>
<tr>
<td>High average heat transfer coefficient</td>
<td>Governs size and performance</td>
</tr>
</tbody>
</table>

3.1. **Optimization Using fminsearch()**

Included in the MATLAB Optimization Toolbox is a function called fminsearch, which can be used to find the minimum value of a user-defined function, with multiple input parameters. A variant of fminsearch exists, written by John D’Errico, available on Mathworks File Exchange website, titled fminsearchbnd. Fminsearchbnd allows upper and lower bounds to be set on the optimization parameters. A user-defined function that calculates the heat exchanger minimization parameter as an output is created. By defining input parameters, fminsearch can be implemented to compute a local minimum in proximity to the input configuration. By nesting the fminsearch within a for-loop, it can be run for multiple initial guesses, generating a set of local minimums. This is a method for
estimating the global minimum. The final configuration can be selected by applying the figures of merit to these local minimum configurations. While the estimated global minimum might be selected, if the other figures of merit are undesirable, then a different local minimum can be selected.

To run a minimization, two MATLAB files must be created. The first file is the user-defined function file, which takes one set of input parameters, and evaluates the user defined function. In this case, the set of input parameters are the heat exchanger specifications, and the output is the parameter to be minimized. If maximization is desired, fminsearch can just be told to find the minimum of the reciprocal of the optimization parameter. The second MATLAB file has four purposes: to define all initial configurations to be tried, to define the upper and lower limits for each optimization parameter, to run the fminsearch function for each initial configuration by accessing the user-defined function, and to organize the output of each fminsearch run to allow for configuration selection.

In order to limit the potential outcomes, a method was developed that would guide the fminsearch algorithm towards desirable configurations. By setting hard limits on the non-optimized figures of merit, the output would only contain acceptable configurations. For example, while minimizing the reciprocal of the margin to super prompt critical (RMSPC) for the PHX, the tube-side pressure drop is
limited to 40 kPa. Within the user-defined function file, after calculating the RMSPC, before returning the RMSPC value to the fminsearch algorithm, the tube side pressure drop is checked. If the pressure drop is greater than 40 kPa, the RMSPC is modified by multiplying the RMSPC by the pressure drop and an arbitrarily large number, in this case $10^{10}$. By modifying the RMSPC in this way, we are creating an RMSPC with an artificial dependence on the tube-side pressure drop, but only when this pressure drop exceeds our hard limit. Since the RMSPC is now proportional to the tube-side pressure drop, the fminsearch algorithm systematically reduces the tube-side pressure drop in order to minimize the RMSPC. The minimization does not end until the pressure drop falls below 40 kPa, at which point the value for the RMSPC instantly drops by orders of magnitude. By using this procedure, if a local minimum is found with a pressure drop greater than the hard limit, the fminsearch algorithm will attempt to find a new local minimum, while maintaining a pressure drop below the hard limit.

### 3.2. Primary Heat Exchanger

The PHX transfers energy from the fuel salt to the lithium coolant. In order to maximize the MSPC, it must return the fuel salt to the core as quickly as reasonably possible. In this regard, it was found that a tube-in-shell design would be able to return the fuel salt to the reactor core very quickly, as opposed to a more compact
heat exchanger, where pressure drop limited the minimum heat exchanger residency time. A cross section of a tube in shell heat exchanger is displayed below in Figure 8. By putting the fuel salt on the tube side of the heat exchanger, a very direct path through the heat exchanger was created. Also, if the length of the heat exchanger exceeds the length required to stay within the shadow shield, it can be coiled to prevent the size of the shadow shield from increasing much beyond what is required for shielding the core. This coiling also allows the PHX inlets and outlets to be relatively close to the core inlets and outlets, with a radial distance approximately equal to the radius of the coil, further reducing time spent out of the core. For the 500 kWe design, the required length was not long enough to require coiling to take place. In section 3.2.3, the final design is described in detail.

![Figure 8: Tube in shell heat exchanger (Source:[11])](image)
3.2.1. PHX Design Process

The procedure to design a tube in shell heat exchanger has been well documented. Using procedures outlined in the *Handbook of Heat Exchanger Design* as well as the Wolverine Tube inc. *Engineering Data Book III*, a MATLAB user-defined function was generated. Given 7 input parameters, the user-defined function outputs the MSPC, required length, and pressure drops for the configuration. This design procedure was expanded to maximize the MSPC using fminsearch() in MATLAB. By varying the shell diameter, the tube diameter, the tube spacing, the baffle spacing, the baffle window size and the fuel salt and lithium temperature changes, a maximum MSPC was obtained.[12]

To determine the total thermal resistance within the tube and shell heat exchanger, the heat transfer coefficients on the tube and shell sides must be known. For the tube side, relationships for the Nusselt number within tubes are widely available. The Nusselt number for fully turbulent flow in this case is [13]

\[
\text{Nu}_t = 0.027 \text{Re}^{0.8} \text{Pr}^{1/3} \phi^{0.14}
\]

(20)

where \( \phi \) represents the ratio between the mean tube fluid temperature to the temperature of the wall on the tube side. With the Nusselt number known, the heat transfer coefficient is [13]

\[
h_t = \frac{\text{Nu}_t k_t}{D_h} \left[ \frac{\text{W}}{\text{m}^2 \cdot \text{K}} \right]
\]

(21)
The shell side heat transfer coefficient is calculated by modifying the ideal heat transfer coefficient of tubes in cross-flow.\[12\]

\[
h_{\text{ideal}} = j_{\text{ideal}} C_{p,s} \frac{m_s}{A_{\text{shell}}} Pr_s^{\frac{2}{3}} \phi_s^{0.14} \left[ \frac{W}{m^2 \cdot K} \right]
\]

\[
h_s = h_{\text{ideal},s} J_c J_l J_b J_r J_s \left[ \frac{W}{m^2 \cdot K} \right]
\]

(22)

(23)

where \( \phi_s \) represents the ratio between the mean shell fluid temperature to the wall temperature on the shell side. \( J_c, J_l, J_b, J_r, \) and \( J_s \) represent corrective factors taking into account segmented baffle windows, baffle leakage, bypass correction, laminar heat transfer correction (for Re<100 only), and unequal inlet/outlet baffle spacing respectively. These parameters are calculated from relationships given by Hewitt.\[12\]

The thermal resistance for a solid annular tube is given as \[12][13\]

\[
R_{\text{wall}} = \frac{\ln \left( \frac{D_t}{D_{t,i}} \right)}{N_{\text{tubes}} 2\pi L_{\text{tube}} k_{\text{moly}}} \left[ \frac{K}{W} \right]
\]

(24)

The total thermal resistance is the sum of the thermal resistances. \[12][13\]

\[
R_{\text{total}} = \frac{1}{h_s A_s} + \frac{1}{h_l A_l} + R_{\text{wall}} \left[ \frac{K}{W} \right]
\]

(25)
The dependence of the heat transfer coefficient on the shell side corrective factors, as well as the tube wall temperatures requires an iterative approach to determine the total length of the heat exchanger. \[12\]

To accurately obtain the tube side friction factor, a seventh order polynomial was created using MATLAB fitted to the Colebrook formula for a smooth pipe \[14\]

\[
\frac{1}{\sqrt{f}} = 2\log(Re \sqrt{f}) + 0.8
\]  

(26)

\[
f_t = 0.032 - 2.1 \times 10^{-6} \text{Re} + 1.3 \times 10^{-10} \text{Re}^2 - 4.6 \times 10^{-15} \text{Re}^3 + 9.4 \times 10^{-20} \text{Re}^4 - 1.1 \times 10^{-24} \text{Re}^5 + 6.689 \times 10^{-30} \text{Re}^6 - 1.7 \times 10^{-35} \text{Re}^7
\]  

(27)

Since a temperature change of 57 K produces negligible flow acceleration, the pressure drop within the tubes can be assumed to be the only component of frictional losses. \[14\]

\[
\Delta P_i = \frac{f_t L_{tube} \rho u_i^2}{2D_{i,i}} \quad \text{[Pa]}
\]  

(28)

Seven optimization parameters were considered for the PHX. The shell inner diameter \((D_s)\), which houses the tubes, was varied between 0.1 m and 0.6 m. The tube inner diameter \((D_{i,i})\) was optimized as the ratio of its diameter to the shell inner diameter. This was done to prevent wasted iterations from being calculated. The iterations that would have started with large tube inner diameter and small shell inner diameters would have had a very small number of tubes, resulting in a long
heat exchanger, while a small tube inner diameter and large shell inner diameter could result in thousands of tubes. The tube spacing ($t_s$), is the distance from centerline to centerline between tubes, in units of tube outer diameter. The distance between each baffle ($L_{bc}$) was varied between 2 cm and 25 cm. This affects the number of passes that the lithium makes across the tubes. The baffle window fraction is the fraction of the baffle that is left for lithium to flow through. A smaller baffle window causes more tubes to receive full cross flow, which is beneficial to heat transfer. However, since the heat transfer is limited by tube side convection, little is gained by reducing the baffle window. Since a smaller baffle window fraction mainly increased the pressure drop on the shell side, the baffle window fraction was held constant at 0.2, the recommended value. [12]

In addition to geometric optimization parameters, the fuel and lithium temperature changes within the heat exchanger were also optimized. The MSPC is affected by the fuel temperature change in opposing ways. A smaller temperature change enhances heat transfer by encouraging faster flow through the heat exchanger, which can shorten the length of the heat exchanger, reducing the time spent out of the core. This has a positive effect on the MSPC. The smaller temperature change also causes the fuel to spend less time in the core, which has a negative effect on the MSPC. By including the fuel salt temperature change as an
optimization parameter, a larger MSPC was obtained. A decreased lithium
temperature change enhanced heat transfer on the shell side, by increasing the flow
velocity, but subsequently increased the pressure drop. Since the lithium is not the
limiting fluid for heat transfer, the optimization was used to find a balance between
pressure drop and shell side convective heat transfer.

Table 8: PHX optimization parameters and optimization range

<table>
<thead>
<tr>
<th>Optimization Parameter</th>
<th>Range Minimum</th>
<th>Range Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shell inner diameter (D_s)</td>
<td>0.1 m</td>
<td>0.6 m</td>
</tr>
<tr>
<td>Tube inner diameter (D_{ti})</td>
<td>0.002 ×Ds</td>
<td>0.1 ×Ds</td>
</tr>
<tr>
<td>Tube spacing</td>
<td>1.5 ×(D_{ti}+0.001)</td>
<td>3.0 ×(D_{ti}+0.001)</td>
</tr>
<tr>
<td>Baffle spacing</td>
<td>0.02 m</td>
<td>0.25</td>
</tr>
<tr>
<td>Baffle window size (fraction)</td>
<td>0.2</td>
<td>0.25</td>
</tr>
<tr>
<td>Fuel temperature drop</td>
<td>10°C</td>
<td>150°C</td>
</tr>
<tr>
<td>Lithium temperature rise</td>
<td>10°C</td>
<td>150°C</td>
</tr>
</tbody>
</table>

The tube thickness was not included in the optimization parameters because
the optimization attempted to reduce the thickness to zero. Any thickness of tube
would increase the thermal resistance of the system, but below 0.5 mm, the thermal
resistance is negligible. The tube thickness was therefore set to 0.5 mm, to prevent
its minimization, since this thickness is large enough to withstand the pressures
being considered.
While mission specific hardware design will be driven by concerns that cannot be determined under the fidelity of this work, there will inevitably be limits on the fuel salt pressure drop. Limiting pressure drop is important for system reliability and pumping power. Acknowledging this limitation, a fuel salt side pressure drop of 40 kPa was chosen. This pressure drop is comparable to the pressure drop expected in the core.

3.2.2. PHX Optimization Adjustments
Throughout the optimization process, adjustments were made to improve the accuracy of the model, and improve the quality of the output configuration. At first, the maximization of MSPC was achieved by simply minimizing the time out of the core. This was done because the fuel salt temperature change was set by neutronic analysis. The temperature change was chosen to be large because that corresponded to a longer core residency time, which in theory improved the MSPC. Unfortunately, a large temperature change results in a low mass flow rate, and subsequently a lower fluid velocity. For LiF-UF₄, with a very high Prandtl number, momentum diffusivity dominates convection, and heat transfer suffers from low fluid velocities. This increased the required length of the heat exchanger, which increased the heat exchanger residency time. Instead of simply lowering the fuel temperature change, the optimization was expanded to maximize the MSPC directly,
in part by optimizing the fuel temperature change in addition to optimizing the geometric parameters. With this change the core residency time is directly controlled by the PHX optimization, allowing an even greater MSPC.

Another enhancement to the optimization was the inclusion of all pressure drop terms between exiting the core and reentering the core. Initially, the pressure drop limit of 40 kPa was set on just the pressure drop within the tubes. While the frictional pressure drop within the tubes should be the largest pressure drop term, the minor losses at the inlet and outlet can be significant, if nothing in the design limits them. The optimization produced many configurations with large shell diameters with widely spaced tubes, with a tube side pressure drop of 40 kPa. Since the fuel salt enters the PHX from an 8.4 cm pipe; pressure drops on the order of 100 kPa in the headers and nozzles due to flow expansion and contraction resulted. By expanding the optimization to require all pressure losses to be below 40 kPa, a more reasonable configuration was obtained. This is an important aspect to problem definition when using optimization algorithms. When it is assumed that variables will be insignificant, and are left unconstrained, the optimization might uncover configurations where they become significant.
Table 9: PHX pressure drop terms, in the order of upstream to downstream

<table>
<thead>
<tr>
<th>Pressure drop term</th>
<th>Value (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Expansion from piping to inlet nozzle</td>
<td>-0.94</td>
</tr>
<tr>
<td>Expansion from inlet nozzle to header</td>
<td>-4.03</td>
</tr>
<tr>
<td>Contraction from header to tubes</td>
<td>15.1</td>
</tr>
<tr>
<td>Friction within tubes</td>
<td>19.8</td>
</tr>
<tr>
<td>Expansion from tubes to header</td>
<td>-4.64</td>
</tr>
<tr>
<td>Contraction from header to outlet nozzle</td>
<td>11.9</td>
</tr>
<tr>
<td>Contraction from outlet nozzle to piping</td>
<td>11.3</td>
</tr>
<tr>
<td><strong>Total Pressure drop</strong></td>
<td><strong>40.0</strong></td>
</tr>
</tbody>
</table>

3.2.3. PHX Specifications

After the optimization was completed, 810 potential configurations were presented. Using Microsoft Excel, filters were applied to the data to narrow down the results. First, only options with a MSPC greater than 0.006 were considered. Then only tube lengths below 0.75 m were considered. This number was chosen because the total length of the heat exchanger, including the headers, must be kept below 1 m to fit behind the shadow shield properly. The proposed header design is 10 cm wide per side not including 1 cm on each side for the wall. This left 26 configurations to decide between. Seven more options were eliminated because they had shell side pressure drops greater than 50 kPa, while the remaining 19 options were all less than 10 kPa. 14 of the remaining options were duplicates, meaning they converged on precisely the same result. Of the remaining configurations, the one with the minimum mass also had the largest MSPC. The
operating conditions are displayed in Table 10, while the specifications for this configuration are displayed in Table 11. Upon reviewing the PHX design, it was noticed that a very high tube side heat transfer coefficient was obtained, equal to 17,000 W/m²-K, corresponding to a Nusselt number of 126. After verifying that the Reynolds number, Prandtl number, and L/d ratio used were within ranges for the equations used, a literature search was performed to find other examples of high Nusselt numbers for tube side heat transfer coefficients. Two papers showed experimental examples of Nusselt numbers in excess of 400 for water within small circular pipes with Reynolds numbers in up to 50,000.[15][16] The final design can be viewed as a CAD rendering in Figure 9 and as a section view in Figure 10.

**Table 10: PHX operating conditions**

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary fluid</td>
<td>LiF(65%)-UF4(35%)</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>1500 K</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>1385 K</td>
</tr>
<tr>
<td>Fuel mass flow rate</td>
<td>22.2 kg/s</td>
</tr>
<tr>
<td>Secondary fluid</td>
<td>Li (enriched to 99% Li-7)</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>1276 K</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>1426 K</td>
</tr>
<tr>
<td>Lithium mass flow rate</td>
<td>3.08 kg/s</td>
</tr>
<tr>
<td>LMTD</td>
<td>90.2 K</td>
</tr>
</tbody>
</table>
Table 11: PHX configuration specifications

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat exchanger length</td>
<td>1.14 m</td>
</tr>
<tr>
<td>Effectiveness</td>
<td>0.7</td>
</tr>
<tr>
<td>Shell Inner diameter</td>
<td>0.121 m</td>
</tr>
<tr>
<td>Shell thickness</td>
<td>0.01 m</td>
</tr>
<tr>
<td>Number of tubes</td>
<td>155</td>
</tr>
<tr>
<td>Tube inner diameter</td>
<td>6.1 mm</td>
</tr>
<tr>
<td>Tube wall thickness</td>
<td>0.5 mm</td>
</tr>
<tr>
<td>Baffle spacing</td>
<td>0.32561 m</td>
</tr>
<tr>
<td>Number of baffles</td>
<td>3</td>
</tr>
<tr>
<td>Heat exchanger mass</td>
<td>76.5 kg</td>
</tr>
<tr>
<td>Pipe inner diameter</td>
<td>0.084 m</td>
</tr>
<tr>
<td>MSPC</td>
<td>513 pcm</td>
</tr>
<tr>
<td>Tube pressure drop</td>
<td>10.4 kPa</td>
</tr>
<tr>
<td>Shell pressure drop</td>
<td>16.0 kPa</td>
</tr>
<tr>
<td>Tube velocity</td>
<td>1.09 m/s</td>
</tr>
<tr>
<td>Avg. heat transfer coeff.</td>
<td>6,251 W/m²-K</td>
</tr>
</tbody>
</table>

Figure 9: CAD rendering of the PHX
3.3. **Secondary Heat Exchanger**

The SHX’s function is to transfer heat from the liquid lithium coolant to the helium working fluid for the Brayton cycle. Since time spent within the heat exchanger is not important for these fluids, the design can focus on having a high heat transfer coefficient, low mass, and low pressure drop.

For liquid to gas heat transfer, offset-strip fin heat exchangers have been shown to be effective for producing a small, compact design. [17] Due to the repeating series of fins in the path of flow, the flow is constantly forced to redevelop.
on the surface of the fin. The result is greatly increased heat transfer at a cost of a slightly increased pressure drop. A smaller heat exchanger volume can be expected from the higher efficiency, which helps reduce the radiation shield mass, in addition to the heat exchanger mass. These benefits led to the decision to use an offset-strip fin heat exchanger as the SHX. Figure 11 and Figure 12 display the offset-strip fin heat exchanger configuration with the associated dimensionless parameters used in the calculation of the Fanning friction factor and Colburn factor.

Figure 11: Offset-strip fin heat exchanger configuration (Source: [17])
Originally, the SHX was to be optimized for minimum mass, yet initial optimization found that mass minimization produced undesirable results. The first was a massive pressure drop. The optimization uncovered that by creating extremely tiny channels with high Reynolds numbers, that the required size of a heat exchanger could be extremely small. With nothing restricting the pressure, an extremely tiny mass with an enormous pressure drop was the result, on the order of 10 MPa, which is nearly the magnitude as the working fluid pressure being considered. When the maximum pressure drop was limited to 40 kPa, the optimization produced multiple configurations with mass below 100 kg. While this result seemed to be a good compromise, having low mass, and low pressure drop,
further investigation showed that optimizing for minimum heat exchanger mass was not the ideal strategy. First, once the mass approaches 200 kg, optimizing it to reduce the mass is insignificant relative to the mass of the total system. Second, for space reactors, by far the largest component of system mass is the radiator setup for heat rejection.[19] It was realized that by reducing the pressure losses in a heat exchanger, the required radiator area can be reduced. Even a small increase in cycle efficiency caused by reducing pressure losses can result in a large mass savings. This is because for a given heat rejection temperature, the surface area of the radiator is proportional to the heat rejection rate. Higher cycle efficiency reduces the amount of heat that needs to be rejected. For this reason, the SHX was optimized for helium pressure drop. The goal for this exercise was to reduce the helium pressure drop to below 10 kPa.

The parameter $\beta$ is used to characterize the pressure losses of a Brayton cycle. In an ideal cycle, where pressure losses are not accounted for, the ratio of the turbine outlet pressure to the compressor inlet pressure would be one. The ratio of the compressor outlet pressure to the turbine inlet pressure would also be one. In reality, the pressure drop within the heat exchangers and piping increases both of these ratios to slightly above one. [20]
\[ \beta_{\text{loss}} = \left( \frac{P_{1,o} P_{c,o}}{P_{c,t} P_{t,d}} \right)^{\gamma-1} \gamma \]  

where \( \gamma \) is the isentropic expansion coefficient. When \( \beta \) is included in the cycle calculations, the turbine work becomes [20]

\[ \dot{W}_T = \eta_T \dot{m}_c P_{t,\text{in}} \left( 1 - \frac{\beta_{\text{loss}}}{r_p^{\gamma-1}} \right) \text{ [W]} \]  

where \( \eta_T \) is the isentropic efficiency of the turbine, and \( r_p \) is the compression ratio. Essentially, the turbine is unable to expand the gas at the same ratio as it was compressed within the compressor, because pressure was lost within the system.

Figure 13 demonstrates the effect of reducing \( \beta \). A reduction in \( \beta \) from 1.05 to 1.01 corresponds to a \( \sim 20\% \) reduction in radiator area and a 700 kg mass savings. A \( \beta \) of 1.05, which corresponds to a total pressure loss of about 12\%, is typically assumed in cases where pressure drop optimization is not performed. A \( \beta \) of 1.01 corresponds to a 2.5\% pressure loss in both heat exchangers.
3.3.1. SHX Design Process

The heat exchanger was first divided into identical unit cells. A unit cell of lithium would have a unit cell of helium both directly above and directly below it. In order to reduce the amount of heat that is lost to the environment, the top and bottom channels would be helium unit cells. An isometric view of a helium unit cell can be seen in Figure 14. Since lithium has such a high thermal conductivity, having offset-strip fins within the lithium channels did not significantly increase the overall heat transfer between cells. The lithium side instead has straight rectangular channels that extend the complete length of the heat exchanger. For structural considerations, the lithium channels were configured so that a vertical strip was
directly below each lithium strip and also directly below each lithium straight channel. Since the helium will be operating at a higher pressure, this would provide some structural rigidity, preventing the layer in between from bowing out into the lithium channels. Within each unit cell, 4 lithium channels exist. A graphic depicting the staggered helium channels can be seen in Figure 15, where two unit cells of helium are shown in the middle, with two unit cells of lithium above and below.

![Figure 14: SHX helium unit cell](image)
In order to obtain the minimum helium pressure drop, the geometric features to be optimized must be determined. Constraining the lithium channel width to the helium width leaves the helium channel height, lithium channel height, helium fin length, helium channel width, fin thickness, the number of cells per row, and the number of rows to be defined. Initially, these were all optimization parameters. Running the optimization revealed that the fin thickness would approach zero thickness in order to minimize pressure drop. This is because the form drag term for calculating pressure drop, seen in equation 38, is proportional to the fin thickness. The fin thickness was limited to a 1 mm minimum to prevent
difficulty with manufacturing and assembly. The minimum and maximum range considered for all optimization parameters is listed in Table 12.

Table 12: SHX optimization parameters and optimization range

<table>
<thead>
<tr>
<th>Optimization Parameter</th>
<th>Range Minimum</th>
<th>Range Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Helium channel width</td>
<td>0.01 m</td>
<td>0.1 m</td>
</tr>
<tr>
<td>Unit cell length</td>
<td>0.01 m</td>
<td>0.1 m</td>
</tr>
<tr>
<td>Helium channel height</td>
<td>0.01 m</td>
<td>0.12 m</td>
</tr>
<tr>
<td>Lithium channel height</td>
<td>0.005 m</td>
<td>0.1 m</td>
</tr>
<tr>
<td>Cells per row</td>
<td>2</td>
<td>24</td>
</tr>
<tr>
<td>Number of lithium rows</td>
<td>1</td>
<td>8</td>
</tr>
</tbody>
</table>

The mass flow rate within a helium cell is,

$$\dot{m}_{cell_h} = \frac{\dot{m}_{cells, frontal}}{n \times (r + 1)} \left[ \frac{\text{kg}}{\text{s}} \right]$$

and the hydraulic diameter, expressed as four times the minimum flow path area divided by the total heat transfer area per unit length is

$$D_{h} = \frac{4sh_hl}{2(sl + h_hl + th_h)} \left[ \text{m} \right]$$

The Reynolds number for both the lithium and helium can be written as

$$\text{Re}_D = \frac{4\dot{m}_{cell}}{\mu \pi D_h}$$
The Nusselt number for offset-strip fin geometries can be written as a function of the Colburn factor, the Reynolds number and the Prandtl number raised to the one-third power.[17]

\[ \text{Nu}_h = j \text{Re} \text{Pr}^{1/3} \]  \hspace{1cm} (34)

The helium heat transfer coefficient is[13]

\[ h_h = \frac{\text{Nu} k_h}{D_h} \left[ \frac{W}{m^2 - K} \right] \]  \hspace{1cm} (35)

Here, the Colburn factor is written in terms of the Reynolds number based on the length of the fin. [12]

\[ j = 0.665 \times \text{Re}^{-0.5} \]  \hspace{1cm} (36)

For laminar, transitional and turbulent flow with Reynolds numbers below 20,000, the friction factor is a function of the Reynolds number and the three non-dimensional geometric ratios, shown in Figure 12. [18]

\[ f = 9.62 \times \text{Re}^{0.74 \times 0.19 \times 0.31 \times 0.27 \times (1 + 7.7E^{-8} \times 4.4^{0.92 \times 3.8^{0.24}})^{0.1}} \]  \hspace{1cm} (37)

For Reynolds numbers above 20,000, the form drag becomes the dominating component in the pressure loss term.[12][21][22][23] This was confirmed by the CFD validation, discussed in section 0, where the pressure drop was shown to be more than double at a Reynolds number of 44,000. Equation 38 allows the friction factor to represent both the drag coefficient and the frictional losses on the walls.
The recommended value for \( C_d \), the drag coefficient, for offset-strip fin geometries is 0.88. \[ f = \frac{C_d t}{2l} + 1.328 \text{Re}^{-0.5} \] (38)

Since heating was occurring on only the top and bottom surfaces, surface efficiencies were considered for the fins. Fin heights were taken to be half of the channel height, to allow the top half of the fin to coincide with the top surface, while the bottom half of the fin coincided with the bottom surface. The fin and overall surface efficiencies are [13]

\[
\eta_{\text{fin}} = \frac{\arctan \left( \frac{m h_f}{2} \right)}{m \frac{h_f}{2}} = \frac{\arctan \left( \frac{2h_h(2l+2t)}{k_m \text{lt}} \right) h_h}{\frac{2h_h(2l+2t)}{k_m \text{lt}} h_h} \] (39)

\[
\eta_{\text{surface}} = 1 - \frac{A_{\text{fin}}}{A_h} \left( 1 - \eta_{\text{fin}} \right) \] (40)

With the heat transfer coefficient, the total surface area and the surface efficiency, the average thermal resistance for a helium unit cell is [13]

\[
R = \frac{1}{h_h A_h \eta_{\text{surface}}} \left[ \frac{\text{K}}{\text{W}} \right] \] (41)

Since within each unit cell there are four lithium channels, the mass flow rate of lithium per channel can be represented by
\[
\dot{m}_{\text{channel}} = \frac{\dot{m}}{\text{channels}} = \frac{P_{\text{thermal}}}{\Delta T_i C_{\text{p,i}}} \left[ \frac{\text{kg}}{\text{s}} \right]
\]

(42)

Four lithium channels plus three 1-mm fins equal the width of a helium channel. The lithium channel width becomes

\[
s_l = \frac{s - 3t}{4} \quad [\text{m}]
\]

(43)

With a hydraulic diameter of

\[
D_h = \frac{4s_i h_i}{2 (s_i + h_i)} = \frac{2s_i h_i}{s_i + h_i} \quad [\text{m}]
\]

(44)

a relationship was obtained for the Nusselt number of liquid metals in non-circular ducts from Hartnett and Irvine. [24]

\[
Nu_l = 5 + 0.025 \text{Re}^{0.8} \text{Pr}^{0.8}
\]

(45)

To obtain the thermal resistance of the lithium side, the same procedure used for the helium side was used. The last thermal resistance to be considered is the thermal resistance of the layer of molybdenum between the channels. The plate between the helium and lithium side is 3 mm thick, but half of the surface area has an additional layer of 1 mm, caused by the way the fins are constructed. By taking the thermal resistances of two plates in parallel, the total thermal resistance per unit cell is
\[
R = \frac{1}{h_h A_h \eta_h} + \frac{1}{h_l A_l \eta_l} + \left( \frac{2s l k_{\text{mol}}}{a + t} + \frac{2s l k_{\text{mol}}}{a} \right)^{-1} \left[ \frac{K}{W} \right]
\]  

(46)

Knowing the log-mean temperature difference and the thermal power, the number of cells can be calculated based on the heat transfer per unit cell.

\[
N_{\text{cells}} = \frac{P_{\text{thermal}}}{\frac{LMTD}{R}}
\]

(47)

This leads to the number of downstream cells required, which was rounded up to the nearest whole number.

\[
m = \frac{N_{\text{cells}}}{n \times r}
\]

(48)

### 3.3.1.1. Header Design

The purpose of the heat exchanger headers is to take the flow from the piping and distribute it within the heat exchanger properly. An effective header has minimal losses, and provides uniform flow into the heat exchanger. To minimize pressure drop, a header was designed with as few obstructions and minor losses as possible. Since the SHX has alternating layers of lithium and helium, the header must have a way of combining the layers of helium separately from the layers of lithium.

As the helium flows in from the piping, it is expanded into a rectangular channel, which is half of the width of the heat exchanger, and its height is equal to the sum of the SHX helium channels height. This channel forms the start of the
plenum, which is trapezoidal in shape when viewed from the side, as shown in Figure 17 and Figure 18. Within the plenum, the flow diverges into the channels that make the helium rows. By rounding the channels, just after the division, pressure losses are reduced. The helium is now flowing directly toward the opening in the heat exchanger. Just before entering, the helium channel is expanded so that its width now equals the heat exchanger width, as seen from the top view in Figure 20 and from the CAD model in Figure 21.

The lithium follows a path that is very similar to the helium. The lithium flow is expanded from the inlet pipe into a rectangular channel with the width equal to half the heat exchanger width. At this point, the lithium channel splits into two channels of equal area, and each curves slightly so that they are on opposite sides of the helium exit channel. This can be seen from the top view in Figure 16 and the CAD model in Figure 23, which gives an external view of the lithium inlet and helium outlet. Since the helium occupies the middle half of the header and expanded to the full width of the heat exchanger, the lithium channels each expand from the outer quarter until the two channels combine to have a channel width equal to the heat exchanger width, shown in Figure 22. Like the helium plenum, the lithium plenum is trapezoidal in shape and decreases in flow area as part of the flow diverts into the rows of the heat exchanger, shown in Figure 19.
The fluids enter and exit on opposite directions, forming an “S” shape, instead of each fluid entering and exiting in the same direction, which would form a “U” shape. While this adds some extra piping, this makes the path length for all channels of a common fluid to be the same. This keeps the theoretical pressure drop the same for each channel, which causes the flow distribution to be equal. This is further enhanced because the pressure drop associated with the plenum, the expansion just before entering the heat exchanger, and the contraction upon exiting the heat exchanger is considerably smaller than that for the channels themselves.

Figure 16: Inlet and outlet concept for lithium and helium
Figure 17: Side view of SHX depicting flow path of helium
Figure 18: Section view of the helium inlet plenum
Figure 19: Section view of lithium inlet plenum
Figure 20: Top view of helium outlet header (left) and inlet header (right)

Figure 21: CAD model of helium inlet between plenum and helium channels
Figure 22: Top view of lithium inlet header (left) and outlet header (right)

Figure 23: CAD model of lithium outlet between plenum and lithium channels
3.3.1.1. Pressure Drop Determination

The pressure drop is composed of the pressure drop within the heat exchanger, the expansion and contraction going between a pipe and the inlet header, the expansion and contraction going from the header to the channel, and the 90° bend in the rectangular header channel.

The pressure drop within the heat exchanger is calculated from the frictional losses as well as the flow acceleration as the helium is heated downstream. [23]

\[
\Delta P_{hx} = \rho_{h,in} \frac{u^2}{2} \left( 2 \left( \frac{\rho_{h,in}}{\rho_{h,out}} - 1 \right) + \frac{4 f_h (2 \text{l}_m) \rho_{h,in}}{D_h \rho_{h,mean}} \right) \text{ [Pa]} \tag{49}
\]

The pressure drops associated with transitioning between a pipe and the square inlet channel are calculated as a sudden expansion and sudden contraction with an entry radius equal to 0.1 times the pipe diameter. They are equal to [12]

\[
\Delta P_{\text{expansion}} = \rho_{h,in} \frac{u_{\text{pipe}}^2}{2} \left( A_{\text{ratio}}^2 - 1 + \left( 1 - A_{\text{ratio}}^2 \right)^2 \right) \text{ [Pa]} \tag{50}
\]

\[
\Delta P_{\text{contraction}} = \rho_{h,out} \frac{u_{\text{pipe}}^2}{2} \left( 1 - A_{\text{ratio}}^2 + \lambda K_c \right) \text{ [Pa]} \tag{51}
\]

where \(\lambda\) is related to the sharpness of the opening and \(K_c\) is related to the contraction ratio, obtained from Handbook of Heat Exchanger Design. [12]

Similarly, the pressure drop associated with the transition from the header to the heat exchanger (Figure 20) can be expressed as [12]
\[
\Delta P_{\text{head,expansion}} = \rho_{h,\text{in}} \frac{u_{\text{header}}^2}{2} \left( A_{\text{ratio}}^2 - 1 + \left( 1 - A_{\text{ratio}}^2 \right)^2 \right) \text{ [Pa]}
\] (52)

\[
\Delta P_{\text{head,contraction}} = \rho_{h,\text{out}} \frac{u_{\text{header}}^2}{2} \left( 1 - A_{\text{ratio}}^2 + \lambda K_e \right) \text{ [Pa]}
\] (53)

The pressure drop associated with a 45° bend within the inlet and outlet header (Figure 17) is equal to [25]

\[
\Delta P_{\text{bend,inlet}} = 0.0175 \lambda_{el} \frac{R_o}{D_h} \left( \frac{\pi}{4} \right) \times \rho_{h,\text{in}} \frac{u_{\text{channel}}^2}{2} \text{ [Pa]}
\] (54)

\[
\Delta P_{\text{bend,outlet}} = 0.0175 \lambda_{el} \frac{R_o}{D_h} \left( \frac{\pi}{4} \right) \times \rho_{h,\text{out}} \frac{u_{\text{channel}}^2}{2} \text{ [Pa]}
\] (55)

\[
\lambda_{el} = \frac{5}{\text{Re}_{\text{header}}^{0.45}} \left( \frac{h_{\text{c}} + 1}{2R_o} \right)^{0.275}
\] (56)

where \( R_o \) is the channel centerline radius, which is equal to the channel height. The loss coefficients associated with converging and diverging flow with a 45° side channel angle are given as [25]

\[
K_{l,\text{conv}} = 2 - 2 \left( \frac{r + 1}{r} \right) \left( 1 - \frac{1}{r + 1} \right)^2 - 1.41 \left( r + 1 \right) \left( \frac{1}{r + 1} \right)^2
\] (57)

\[
K_{l,\text{div}} = 0.24
\] (58)

which yields pressure losses of

\[
\Delta P_{\text{div}} = \rho_{h,\text{in}} \frac{u_{\text{channel}}^2}{2} K_{l,\text{div}} \text{ [Pa]}
\] (59)
\[ \Delta P_{\text{conv}} = \rho_{h,\text{out}} \frac{u_{\text{channel}}^2}{2} K_{l,\text{conv}} \quad \text{[Pa]} \]  

(60)

The total pressure drop, which is the term being minimized using fminsearch, is

\[ \Delta P_h[\text{Pa}] = \Delta P_{\text{HX}} + \Delta P_{\text{conv}} + \Delta P_{\text{div}} + \Delta P_{\text{bend,outlet}} + \Delta P_{\text{bend,inlet}} + \Delta P_{\text{contraction}} + \Delta P_{\text{expansion}} + \Delta P_{\text{head,contract}} + \Delta P_{\text{head,expansion}} \]  

(61)

Table 13 displays the helium pressure losses in the order of occurrence. The largest component of pressure loss is the contraction from the outlet channel of the plenum to the piping that carries the helium to the turbine, which exceeds the pressure drop within the heat exchanger itself by a factor of three. The diameter of this piping is 10 cm. The pressure drop can be reduced by a factor of two by increasing this diameter by 20% to 12 cm. This piping diameter was not included as an optimization parameter because the optimization would have just increased it until the pipe area was equal to the plenum channel area. The reality is that flow from a larger pipe is contracted less exiting the plenum, but will be contracted more when the helium enters the turbine. At this phase in the project, the inlet diameter of the turbine is unknown, which would drive the optimum piping diameter. Regardless, the pressure drops associated with the SHX are very low due to the pressure drop optimization.
Table 13: Helium pressure loss terms for the SHX, listed in the order they occur

<table>
<thead>
<tr>
<th>Pressure drop term</th>
<th>Value (Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Expansion from piping to inlet channel</td>
<td>-280.5</td>
</tr>
<tr>
<td>Dividing flow in plenum</td>
<td>45.4</td>
</tr>
<tr>
<td>45 degree elbow inlet</td>
<td>&lt;0.1</td>
</tr>
<tr>
<td>Expansion to full channel width</td>
<td>-16.7</td>
</tr>
<tr>
<td>Heat exchanger friction/drag losses</td>
<td>536</td>
</tr>
<tr>
<td>Contraction to half channel width</td>
<td>54.5</td>
</tr>
<tr>
<td>45 degree elbow exit</td>
<td>&lt;0.1</td>
</tr>
<tr>
<td>Combining flow in plenum</td>
<td>37.8</td>
</tr>
<tr>
<td>Contraction from outlet channel to piping</td>
<td>1861</td>
</tr>
<tr>
<td><strong>Total pressure losses</strong></td>
<td><strong>2237</strong></td>
</tr>
</tbody>
</table>

3.3.2. SHX Specifications

The optimization for the SHX began with a hard limit on the mass of 200 kg. Of the optimization results, thirty seven had a pressure drop below 10 kPa, and a mass below 150 kg. All of these options had very comparable pressure drops, masses and geometries. The only variable that had a wide distribution of values between them was the lithium pressure drop, with values between 59 kPa and 100 kPa. The configuration selected was the one with the lowest lithium side pressure drop at 59 kPa. The final specifications for the SHX are displayed in Table 15, with the corresponding operating conditions in Table 14.
### Table 14: SHX operating conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lithium inlet temperature</td>
<td>1426 K</td>
</tr>
<tr>
<td>Lithium outlet temperature</td>
<td>1276 K</td>
</tr>
<tr>
<td>Lithium mass flow rate</td>
<td>3.56 kg/s</td>
</tr>
<tr>
<td>Helium inlet temperature</td>
<td>862.0 K</td>
</tr>
<tr>
<td>Helium outlet temperature</td>
<td>352.5 K</td>
</tr>
<tr>
<td>Helium mass flow rate</td>
<td>0.586 kg/s</td>
</tr>
<tr>
<td>LMTD</td>
<td>256.66 K</td>
</tr>
</tbody>
</table>

### Table 15: SHX specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cells per row</td>
<td>29</td>
</tr>
<tr>
<td># Rows</td>
<td>4</td>
</tr>
<tr>
<td># Cells downstream</td>
<td>5</td>
</tr>
<tr>
<td>Helium channel width</td>
<td>5.3 mm</td>
</tr>
<tr>
<td>Fin thickness</td>
<td>1 mm</td>
</tr>
<tr>
<td>Fin length</td>
<td>89.9 mm</td>
</tr>
<tr>
<td>Helium channel height</td>
<td>80.1 mm</td>
</tr>
<tr>
<td>Lithium channel height</td>
<td>17.3 mm</td>
</tr>
<tr>
<td>Helium pressure drop</td>
<td>1.56 kPa</td>
</tr>
<tr>
<td>Lithium pressure drop</td>
<td>28.4 kPa</td>
</tr>
<tr>
<td>Thermal resistance per cell</td>
<td>0.0721 K/W</td>
</tr>
<tr>
<td>Mass</td>
<td>156.2 kg</td>
</tr>
<tr>
<td>Length</td>
<td>0.899 m</td>
</tr>
<tr>
<td>Width</td>
<td>0.188 m</td>
</tr>
<tr>
<td>Height</td>
<td>0.524 m</td>
</tr>
<tr>
<td>Overall heat transfer coefficient*</td>
<td>12,270 W/m²-K</td>
</tr>
</tbody>
</table>

* Overall heat transfer coefficient given with area in terms of area of interface between two cells. $A = (2 \times l) \times (s + t)$
3.3.3. CFD Validation

Little previous work exists on the optimization of a liquid to gas offset strip fin heat exchanger for space applications. As a result, the design process described above is newly developed for this application. In order to validate the code used for optimization, the results for three selected configurations were compared to simulated experiments using CFD. Since the numerical correlations for the friction factor and for the Colburn factor have been shown to be accurate within 20% of experimental results, it is to be expected that the MATLAB model results will be accurate to roughly 20% if the model is written correctly.

It was decided that in determining how similar the results from the simulation are to the optimization code, the figures of merit would be the heat transfer into a unit cell of helium, specifically the thermal resistance, and the helium pressure drop in a unit cell. This prevents variations in the LMTD, caused by only a portion of the heat exchanger being simulated, from interfering with the results of the study.

The pressure can be compared simply by computing the average pressure at the inlet and the outlet of the simulation and taking the difference. By dividing this difference by the number of cells, the average pressure drop per cell is obtained.

The thermal resistance is compared by first calculating the rate of heat transfer to the helium channel. Then by dividing by the number of unit cells in the
In a simulation, the heat transfer rate per unit cell can be obtained. The thermal resistance of a unit cell can be then calculated by dividing the log-mean temperature difference by the heat transfer rate per unit cell. The subscripts $i$ and $j$ represent the cell faces on the outlet and inlet boundaries respectively.

\[
\dot{Q} = \left( \sum_{i=1}^{N} A_i \rho_i v_i T_i C_p \right)_{out} - \left( \sum_{j=1}^{M} A_j \rho_j v_j T_j C_j \right)_{in} \quad [W] \tag{62}
\]

\[
\dot{Q}_{cell} = \frac{\dot{Q}}{N\text{Cells}} \quad [W] \tag{63}
\]

\[
R = \frac{\text{LMTD}}{\dot{Q}_{cell}} \quad \left[ \frac{\text{K}}{\text{W}} \right] \tag{64}
\]

Three different configurations, displayed in Table 16, were selected for the comparison. Each configuration was obtained by a \texttt{fminsearch} minimization. Two configurations were chosen and tested when the helium inlet and outlet temperatures were determined to be 575 K and 1200 K respectively. The third configuration was chosen after further literature review, where it was discovered that turbine inlet temperatures of 1350 K were possible. Regardless of the temperatures considered, the model should accurately determine the pressure drop and heat transfer for the configuration.
### Table 16: Three CFD configurations compared to MATLAB model

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Configuration 1</th>
<th>Configuration 2</th>
<th>Configuration 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cells per row</td>
<td>18</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td># rows</td>
<td>8</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>Helium channel width (mm)</td>
<td>7</td>
<td>16</td>
<td>9</td>
</tr>
<tr>
<td>Fin thickness (mm)</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Fin length (mm)</td>
<td>13</td>
<td>56</td>
<td>10</td>
</tr>
<tr>
<td>Helium channel height (mm)</td>
<td>120</td>
<td>105</td>
<td>98</td>
</tr>
<tr>
<td>Lithium channel height (mm)</td>
<td>30</td>
<td>5</td>
<td>10</td>
</tr>
<tr>
<td>Lithium inlet temp (K)</td>
<td>1251</td>
<td>1301</td>
<td>1419</td>
</tr>
<tr>
<td>Lithium outlet temp (K)</td>
<td>1151</td>
<td>1201</td>
<td>1319</td>
</tr>
<tr>
<td>Helium inlet temp (K)</td>
<td>575</td>
<td>575</td>
<td>729</td>
</tr>
<tr>
<td>Helium outlet temp (K)</td>
<td>1200</td>
<td>1200</td>
<td>1300</td>
</tr>
<tr>
<td>Helium inlet pressure (bar)</td>
<td>15</td>
<td>15</td>
<td>15</td>
</tr>
</tbody>
</table>

Three simulations were completed, and the results are displayed in Table 17. Out of the 6 total compared values, three were below 12% difference and three were between 20% and 25% difference. While 25% difference may seem high, it is important to consider the scope of this problem. Heat transfer problems, especially with this level of complexity, rarely have closed form solutions. The model takes into account 7 geometric parameters, helium inlet pressure, and 4 fluid temperatures, for a total of 12 input parameters. Using relationships, some of which are empirical in nature, the code calculates the theoretical heat transfer coefficient for both fluids and the thermal resistance for the molybdenum interface between them.
Table 17: Results of FLUENT and MATLAB comparisons

<table>
<thead>
<tr>
<th>Configuration</th>
<th>FLUENT</th>
<th>MATLAB</th>
<th>% Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$R_{cell}=1.49$ K/W</td>
<td>$R_{cell}=1.17$ K/W</td>
<td>21.1%</td>
</tr>
<tr>
<td></td>
<td>$\Delta P_{cell}=15.88$ Pa</td>
<td>$\Delta P_{cell}=17.72$ Pa</td>
<td>11.6%</td>
</tr>
<tr>
<td>2</td>
<td>$R_{cell}=0.238$ K/W</td>
<td>$R_{cell}=0.179$ K/W</td>
<td>25.0%</td>
</tr>
<tr>
<td></td>
<td>$\Delta P_{cell}=30.05$ Pa</td>
<td>$\Delta P_{cell}=32.91$ Pa</td>
<td>8.7%</td>
</tr>
<tr>
<td>3</td>
<td>$R_{cell}=0.896$ K/W</td>
<td>$R_{cell}=0.875$ K/W</td>
<td>2.4%</td>
</tr>
<tr>
<td></td>
<td>$\Delta P_{cell}=763.8$ Pa</td>
<td>$\Delta P_{cell}=590.4$ Pa</td>
<td>22.7%</td>
</tr>
</tbody>
</table>

Figure 24 displays the centerline temperature profile obtained from the CFD simulation for configuration 2. The red top and bottom portion corresponds to the lithium, while the blue and green portion corresponds to the helium. The segment modeled here represents six downstream cells of the required 32 designed. The exit temperature profile for the modeled portion of the heat exchanger is displayed in

![Figure 24: Side view of centerline temperature profile for configuration 2](image-url)
3.3.4. Potential Sources of Error

Some of the sources of error between the CFD validation and the MATLAB come from assumptions that were made in order to make computation in MATLAB possible. One assumption that was made for the SHX MATLAB code was that of constant thermal properties, while the FLUENT simulations took the temperature dependence of fluid properties into account for each cell. Using the thermal properties at the mean fluid temperatures allows the LMTD to be used. Had the temperature dependence of fluid properties in each cell been accounted for, the temperature in each cell would have to be known. Since the number of cells is
calculated based on the heat transfer between each cell, and the heat exchanger is flowing in counter flow, the number of required cells would affect the temperature in each cell, which would affect the heat transfer in each cell. Different heat transfer values would result in different number of cells. The calculation would have to be repeated until the number of cells remained unchanged. In order to perform a thorough minimization, there was simply not a practical way to accomplish this.

Another large source of error lies within the empirical relationships used. Whenever empirical relationships are employed in theoretical calculations, the potential for error is increased. Even though researchers developing empirical relationships attempt to make them applicable to the widest range of situations possible, there will usually be limitations. For this study, the empirical relationship employed for the friction factor, developed by Manglik and Bergles were used. While not explicitly mentioned in the paper, the data used to generate the friction factor relationship did not cover extremely high Reynolds numbers.[18] Even for Reynolds numbers approaching 20,000, this equation began to display deviation to experimental results, reported for an experiment performed by Michna, Jacobi, and Burton. [23] Another relationship was obtained that showed accuracy for Reynolds numbers above 20,000, shown in equation 38.[12] This equation, while simpler than equation 37, is more accurate at very high Reynolds numbers because the friction
factor at higher Reynolds numbers is largely governed by the form drag.

[12][21][22][23] Even with this improvement for higher Reynolds number flow, the empirical relationships still introduce error into the MATLAB model.

3.4. Tertiary Heat Exchanger

The THX acts to reject heat from the helium within the Brayton loop and transfer it to the heat rejection loop. Like the SHX, the THX was optimized for helium side pressure drop. The procedure for designing the THX is nearly identical to the SHX. The only differences being the reduced thermal power, because some energy was extracted by the turbine, and the different operating temperatures. To provide a high heat rejection temperature, an effectiveness of 90% was desired for the heat exchanger.

3.4.1. THX Specifications

While optimizing the THX, the strong mass dependence on the THX effectiveness was identified. For a heat exchanger effectiveness of 95%, the solution was unable to identify any configurations with a mass below 500 kg. By reducing the effectiveness to 90%, two solutions were found with masses of 350 kg. This demonstrates the huge impact that the effectiveness has on the heat transfer surface area. Essentially, a lower effectiveness reduces the temperature difference between
the fluids. The operating conditions and the final specifications for the selected THX configuration are displayed in Table 18 and Table 19 respectively.

When comparing the results of the SHX and THX, the THX is significantly larger than the SHX. This is because the LMTD of the THX is only 50 K, while the LMTD of the PHX is 250 K. Even a higher heat transfer coefficient associated with the THX is not enough to compensate for this. Regardless, 306 kg is not unreasonable for the heat exchanger mass, especially considering the PHX and SHX masses are only approximately 50 kg and 150 kg respectively.

Table 18: THX operating conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>NaK inlet temperature</td>
<td>295.9 K</td>
</tr>
<tr>
<td>NaK outlet temperature</td>
<td>816.0 K</td>
</tr>
<tr>
<td>NaK mass flow rate</td>
<td>3.41 kg/s</td>
</tr>
<tr>
<td>Helium inlet temperature</td>
<td>665.1 K</td>
</tr>
<tr>
<td>Helium outlet temperature</td>
<td>1350 K</td>
</tr>
<tr>
<td>Helium mass flow rate</td>
<td>0.586 kg/s</td>
</tr>
<tr>
<td>LMTD</td>
<td>51.12 K</td>
</tr>
</tbody>
</table>
Table 19: THX specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cells per row</td>
<td>17</td>
</tr>
<tr>
<td># Rows</td>
<td>8</td>
</tr>
<tr>
<td># Cells downstream</td>
<td>9</td>
</tr>
<tr>
<td>Helium channel width</td>
<td>5.8 mm</td>
</tr>
<tr>
<td>Fin thickness</td>
<td>1 mm</td>
</tr>
<tr>
<td>Fin length</td>
<td>148 mm</td>
</tr>
<tr>
<td>Helium channel height</td>
<td>50 mm</td>
</tr>
<tr>
<td>Lithium channel height</td>
<td>10.1 mm</td>
</tr>
<tr>
<td>Helium pressure drop</td>
<td>1.06 kPa</td>
</tr>
<tr>
<td>Lithium pressure drop</td>
<td>70.6 kPa</td>
</tr>
<tr>
<td>Thermal resistance per cell</td>
<td>0.0631 K/W</td>
</tr>
<tr>
<td>Mass</td>
<td>350.7 kg</td>
</tr>
<tr>
<td>Length</td>
<td>2.66 m</td>
</tr>
<tr>
<td>Width</td>
<td>0.122 m</td>
</tr>
<tr>
<td>Height</td>
<td>0.537 m</td>
</tr>
<tr>
<td>Overall heat transfer coeff*</td>
<td>12,313 W/m²·K</td>
</tr>
</tbody>
</table>

* Overall heat transfer coefficient given with area in terms of area of interface between two cells. $A = (2 \times l) \times (s + t)$
4. Future Work

4.1. Future Use of Codes

One of the aims of this research was to develop a series of codes that could be used to find optimum heat exchanger configurations for different power levels, or for different coolants. The code is structured in such a way that the inputs and material properties are easy to modify. It is also easy to modify optimization hard limits for non-optimized parameters. The Steckler project plans to include designing a 3 MWe SMSR for surface power and a 15 MWe SMSR for NEP. These codes will be used to obtain optimum heat exchanger configurations for these SMSRs.

Regarding the mass models for the SMSR, they assume that reducing the radiator area is of the greatest significance. As the results in section 3.4 have shown, the mass of the THX is driven directly by its effectiveness. The cycle efficiency, and subsequently the radiator area, is also driven directly by the THX effectiveness. This is because the heat rejection temperature is governed by both the Brayton cycle temperatures and the THX effectiveness. Using these optimization tools, the Steckler project team will investigate the optimum THX effectiveness in terms of overall system mass. This will further reduce the mass of the SMSR.

4.2. Mission Specific Parameters

Something this project didn’t consider is mission specific requirements that would limit mass and pressure losses. In the future, the Steckler project research
would like to investigate what sort of limitations would be imposed on a surface power station that would define these limitations. Ideally, the project would be able to use missions that fit into NASA’s research agenda. At this phase in the project, the goal was to show that optimization is possible and that solutions exist, while providing an example of a single desirable configuration. While the results of this research give a good idea of heat exchanger size and shapes, they represent a possible configuration that will inevitably be further constrained in the future.

4.3. Limitations of Optimization
As mentioned previously, mission specific parameters will ultimately determine limits such as mass, dimensions and pressure drop on system components. The optimization finds the local minimums for a single output variable and only considers other variables when hard limits are established. The algorithm will not stop minimizing when the marginal benefit becomes minimal. The component mass might increase ten percent for a pressure drop of less than one percent.

One possible way to overcome this problem would be to create a variable that is a function of multiple variables that are of importance. The relationship might be something like equation 65, where $Y$ is the variable to be minimized for.

$$Y = a \times X_1 + b \times X_2 + c \times X_3$$  \hspace{1cm} (65)
The coefficients must be determined by the significance of $X_1$, $X_2$, and $X_3$ on $Y$. These constants will most likely require iterations to obtain the proper ratios. After running the optimization and seeing the values of $X_1$, $X_2$, and $X_3$ that correspond to minimums of $Y$, the coefficients can be modified to obtain better results. If excess computational power is available, the variation of coefficients could be included in the for-loop and the set of initial guesses would be applied for each set of coefficients. This would produce a large amount of data, and a method would need to be developed to interpret the data and select the optimum coefficients.

4.4. **Secondary Coolant Considerations**

During startup, the lithium within the PHX and SHX, as well as in the piping must be heated to 453 K. Methods to do this have been developed for the SP-100 [26]. While melting the lithium can be done, it requires additional mass, engineering of components, and a startup energy source. In the future, a secondary coolant of NaK will be investigated, which is a liquid at room temperature. Since NaK has significantly worse thermal properties than lithium, it was not considered on the basis that heat transfer would suffer. However, based on the results of this study, the limiting case for heat transfer in the PHX and SHX lies not with the lithium. If NaK can be used without affecting performance significantly, then startup only requires that the molten salt be melted, which could simplify the startup procedure.
4.5. Offset-Strip Fin Heat Exchanger Testing

Although CFD is becoming more widely accepted, the SHX would still benefit from scaled testing for high Reynolds number flow, where the correlations are not as well developed. By performing a series high Reynolds number tests on offset-strip fin heat exchangers, more precise correlations could be obtained, further demonstrating their applicability, while reducing uncertainty in present calculations. This has benefits that exceed those for space applications. Interest in compact heat exchangers has increased over the years and high Reynolds number data would expand the knowledge of these

4.6. Manufacturing of Components

The PHX optimization resulted in tubes that were about seven mm. While tubes of this size can be built with current technology, the ability to obtain a finish on a surface on the inside of a tube this small has not been investigated. High performance tube in shell heat exchangers require that production of these tubes be done with high precision and consistency across the tubes.
5. Conclusions

5.1. Optimization Procedures

When designing an optimization problem, it is important to consider all constraints that are placed upon a system. For example, when minimizing mass, it might be convenient to leave pressure drop unconstrained, and simply select a configuration from the options with lower pressure drops. As this research discovered, optimization algorithms can abuse unconstrained parameters and allow them to approach undesirable values. If a hard limit is not set on the pressure drop, then as the algorithm steps towards a lower mass, the pressure drop can increase without bounds.

Occasionally, the algorithm will stumble upon configurations where the relationships applied are not applicable. This occurred during the PHX optimization, when the tube diameter approached the shell diameter, leaving room for only one tube. Some relationships on the shell side require that multiple tubes exist, or they yield nonsensical results. This specific problem was fixed by defining the tube diameter as a fraction of the shell diameter, preventing this from occurring.

Another example where the code found a nonsensical configuration was when the PHX outlet nozzle was defined to be 10 cm. Since the loss coefficients are linearly interpolated from a chart in Hewitt, the interpolation in MATLAB yields NaN, meaning not a number, when the area of the nozzle was larger than the area.
before the nozzle. This made the net tube side pressure drop to be NaN, preventing the hard limit of 40 kPa from being enforced. To eliminate this potential for error, the nozzle diameter is modified such that it will always be smaller than the header that precedes it.

5.2. Closing Remarks

This research investigated mechanisms of thermal transport and the optimization of heat exchangers for the SMSR concept. One of the major goals of this research was to demonstrate that heat exchangers can be designed for small molten salt reactors for space applications to be practical and lightweight. This has been demonstrated by the final specifications and performance of the heat exchangers described in section 3. By optimizing for MSPC and helium pressure drop, heat exchanger designs that are practical and lightweight were obtained. The total helium pressure drop within the SHX and THX was 2.62 kPa, which corresponds to less than 0.2% of the turbine inlet pressure, resulting in a loss coefficient, $\beta$, equal to 1.0014. This research also demonstrated that a high MSPC, equal to 513 pcm for this design, can be obtained for the 500 kWe SMSR when a tube and shell heat exchanger with the fuel salt on the tube side is used. Given a theoretical maximum MSPC for $^{235}\text{U}$ of 640 pcm, the solid fuel equivalent beta fraction is just over 0.8, which meets our design goals. This means that the SMSR is a very controllable, safe reactor. By
looking at component specific design, the feasibility of MSRs in space has been further established. These heat exchangers were the first of their kind for removing heat from a high power density MSR in space.

This research has also provided a heat exchanger optimization tool for the Steckler project team. Future SMSR designs will benefit from this because the heat exchangers will require little design work. By incorporating simulated experiments as validation of the optimization tool for the SHX, confidence is increased that the configurations obtained represent accurate estimations. The framework for CFD simulations was also created, which will allow the Steckler project team to perform meaningful CFD analyses, with the intention of performing multi-physics studies to better understand the way MSRs behave.

Another goal of this research was to improve the mass model estimations for the SMSR, which would allow for the minimization of the overall system mass. Previous research did not have accurate mass models for the heat exchangers. This prevented significant mass minimization from occurring within the heat exchanger, and also required a pressure loss parameter, $\beta$, to be estimated. This research demonstrates that very low pressure losses can be obtained, which can have significant mass savings in the radiator, since the thermal efficiency directly affects the quantity of heat rejected.
This research also showed that the prediction that minimum helium pressure loss corresponds to a total minimum system mass was not completely accurate. While reducing the pressure does reduce the radiator mass, there is a diminishing return on the savings, especially considering the increase in heat exchanger mass that results. In order to obtain a minimum overall system mass, these heat exchanger codes will be incorporated into a rigorous mass minimization algorithm.

Future scoping studies will also benefit from the knowledge that overall heat transfer coefficients on the order of 10,000 W/m²-K can be obtained in a salt to lithium tube in shell heat exchanger, while maintaining reasonable pressure losses. This performance was achieved by high Reynolds number flow within small tubes.

Ralph Steckler had a dream that one day in the near future, mankind would venture off our only planet to colonize other worlds and moons. Although he could not accomplish this in his own lifetime, he established a fund through NASA to finance research that would advance this mission after he passed on. It is because of his generosity that the Steckler project team has been able to undertake this research into demonstrating the validity of molten salt technology. It is in memory of his vision that we proudly explore the use of molten salt technology as a power source; not only for future colonies and spacecraft, nor for the much needed
terrestrial applications, but for the power of inspiration as we look ahead and dream of a wider horizon for human discovery and innovation.
References


