Design Considerations of a 15kW Heat Exchanger for the CSPonD Project

by

Adrian A. Adames

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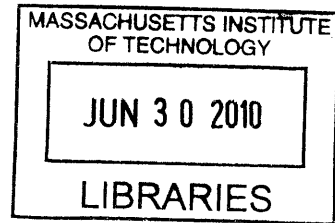in partial fulfillment of the
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Abstract

The objective of this work was to develop a 15 kW heat exchanger model for the CSPonD molten salt receiver that will shuttle the molten salt's thermal energy for conversion to electric power. A heat extraction system comprising of an intermediate heat exchanger placed inside the molten salt receiver is proposed. Using an intermediate working fluid to shuttle heat energy between the receiver and a power cycle working fluid, thermal loads in the heat exchanger are reduced when compared to a similar alternative heat extraction system without a buffer-like intermediate working fluid. Select salt compositions were examined as intermediate working fluid candidates. The fluoride salt composition LiF-NaF-KF showed the best heat transfer performance, providing the greatest heat transfer rate per unit area relative to all the other candidate working fluids. For the LiF-NaF-KF salt composition to attain a thermal power output of 15 kW in two 0.75 in schedule XXS pipes at a flow velocity of 2 m/s, 0.053 m^2 of heat exchanger surface area is required. The fluoride salts examined were the easiest to prepare and use but also the most expensive salts considered. The LiF-NaF-KF salt composition was priced at $7.82/kg. A selection of high temperature alloys capable of withstanding the high operating temperatures of molten salts were chosen as candidate materials for the heat exchanger and ranked based on the allowable stress levels near the operating temperature range of the heat exchanger. Alloys Inconel 671, Inconel 625, and Haynes 230 had the highest allowable stresses at 750 °C, while the stainless steel alloys 310S and 321 have the lowest allowable stresses at 750 °C. Corrosion data for the examined molten salts and high temperature materials near the heat exchanger operating temperatures are very limited and fragmented requiring that any design decisions regarding intermediate fluid selection or materials selection be made after thorough testing is conducted.

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1. Introduction

In today's world, alternative clean methods of energy are needed to meet growing energy demands. The challenge with green energy sources, however, is the inability to store the energy. One such novel approach stores solar energy in a molten salt pond which couples into a concentrated solar power (CSP) plant. Concentrated solar power is a method that reflects the sun's rays to one central location thus heating the area to high temperatures and storing the sun's energy as heat. The Concentrated Solar Power on Demand (CSPonD) project at MIT involves using concentrated solar power to heat and melt a salt mixture that can store the sun's energy for periods greater than 24 hours. The energy from the molten salt can be harvested and converted to electricity. MIT's CSPonD group and Cyprus have partnered to develop a prototype of the CSPonD concept. The focus of this paper is to provide an alternative method of heat extraction from the molten salt pond that can reduce the thermal loads.

The task at hand is to provide a heat exchanger design that can remove heat from the molten salt, convert it into useful energy, while at the same time minimizing thermal loads. One of the main challenges in working with the molten salt is the high temperatures the salt can reach. Conventional heat exchangers are not designed for temperatures greater than 600° C. The high temperature requirements demand the use of unconventional materials and fluids in the design of the CSPonD heat exchanger. The heat exchanger design is for a 15 kW thermal system (~4 kW electric). Ideally, the lessons learned from the small prototype will aid in the design of the larger megawatt power plant. If the technology proves to be a viable means to produce and store energy then it can serve as a means to begin addressing the growing energy demands.

1.1. Concentrated Solar Power on Demand

Concentrated solar power (CSP) systems convert the sun's radiant energy into heat energy by focusing the sun's rays onto a central area called a receiver, heating up the contents inside the receiver which can then be used to produce electricity. CSP systems are early in their inception but have already begun implementation. Parabolic troughs and solar towers are two CSP energy production systems already in existence. As shown in Figure 1, a parabolic trough uses a parabolic reflector to concentrate the sun's rays into a central focus line directed at the receiver, a tube filled with a working fluid that collects the sun's energy as heat. The receiver is often a pipe filled with oil that is circulated through a steam generator to produce electricity. Solar towers use a field of heliostats, essentially large mirrors that track the sun's rays, to focus sunlight onto a receiver at the top of a tower where steam is generated and used to produce electricity. The PS10 project shown in Figure 2 is the first solar power tower in the world to commercially generate electricity and deliver it reliably to the grid. Drawbacks associated with current CSP systems include large capital costs and the inability to store the sun's energy for electricity production after sundown. The CSPonD project addresses these flaws to make CSP a more viable part of the solution to the world's growing energy demands.

Figure 1: Array of parabolic troughs in the Negev desert of Israel. A parabolic trough uses a parabolic reflector to concentrate the sun's rays into a central focus line directed at the receiver, a tube filled with a working fluid that collects the sun's energy as heat. The receiver is often a pipe filled with oil that is circulated through a steam generator to produce electricity [7].

The CSPonD project aims to harvest and store solar energy as heat energy in a molten salt medium that can be used for continuous on demand 24/7 electricity production. Figure 3 illustrates the CSPonD method of harvesting and storing the sun's energy. Hillside mounted heliostats focus the sun's rays into a molten salt receiver where molten salt volumetrically absorbs the sun's energy. Instead of beaming up the sun's rays to a higher elevation like in a solar tower, the hillside heliostats beam down the sun's rays into the molten salt receiver. Placing the heliostats on steep, undevelopable terrain eliminates the costs associated with a large tower on which to place the receiver. The molten salt receiver is an integral part of the CSPonD project.
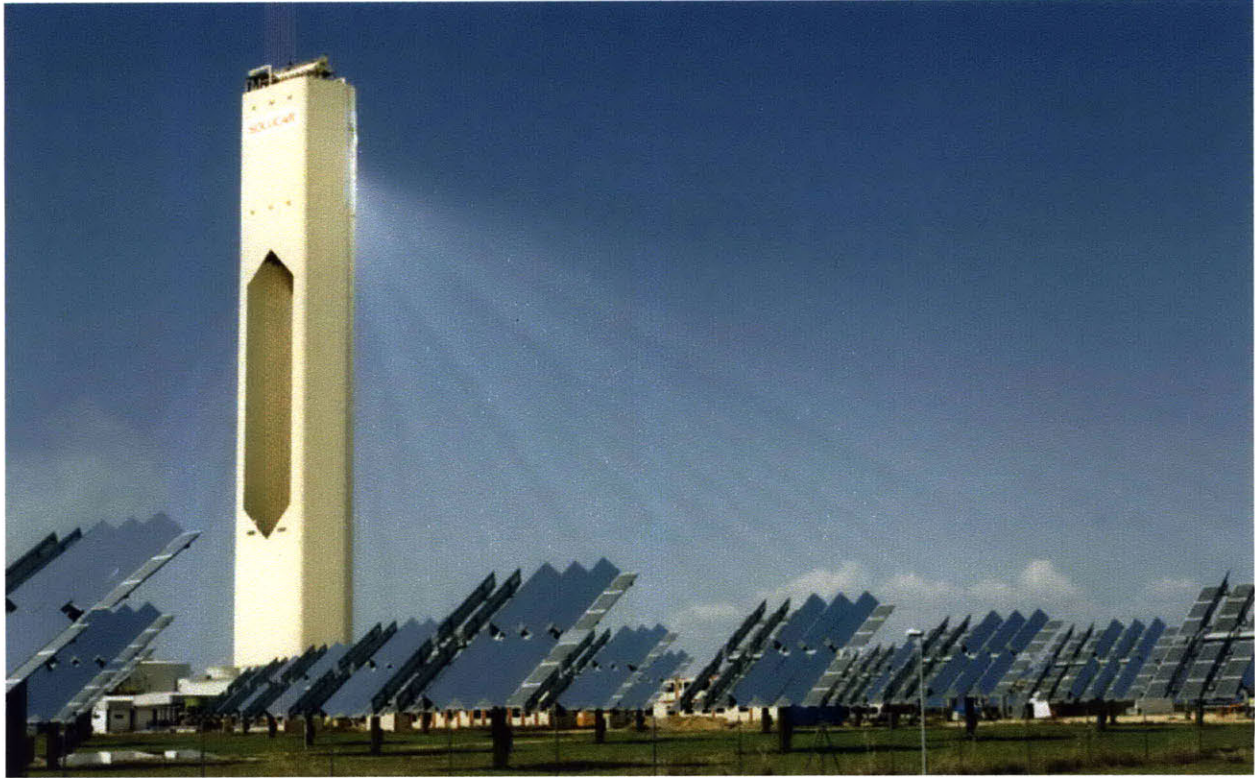
Figure 2: The PS10 project, an 11 MW solar thermal tower plant in Southern Spain, is the first solar power tower in the world to commercially generate electricity and deliver it reliably to the grid. A field of large mirrors called heliostats focuses the sun's rays onto a central receiver at the top of the tower where water is heated into saturated steam which in turn is used to generate electricity [8].

The molten salt receiver, as shown in Figure 4, also acts as the CSPonD energy storage vessel. Molten salts have high melting temperatures and high thermal capacitances meaning that they can retain their heat energy for long periods of time. This makes it possible to heat up the salt with solar energy in the daytime and use the thermal energy stored within the salt in the nighttime. Harvesting the sun's energy in a salt bath makes the CSPonD system less sensitive to weather fluctuations relative to other CSP systems which couple the collection of the sun's energy directly with the production of steam in the receiver.
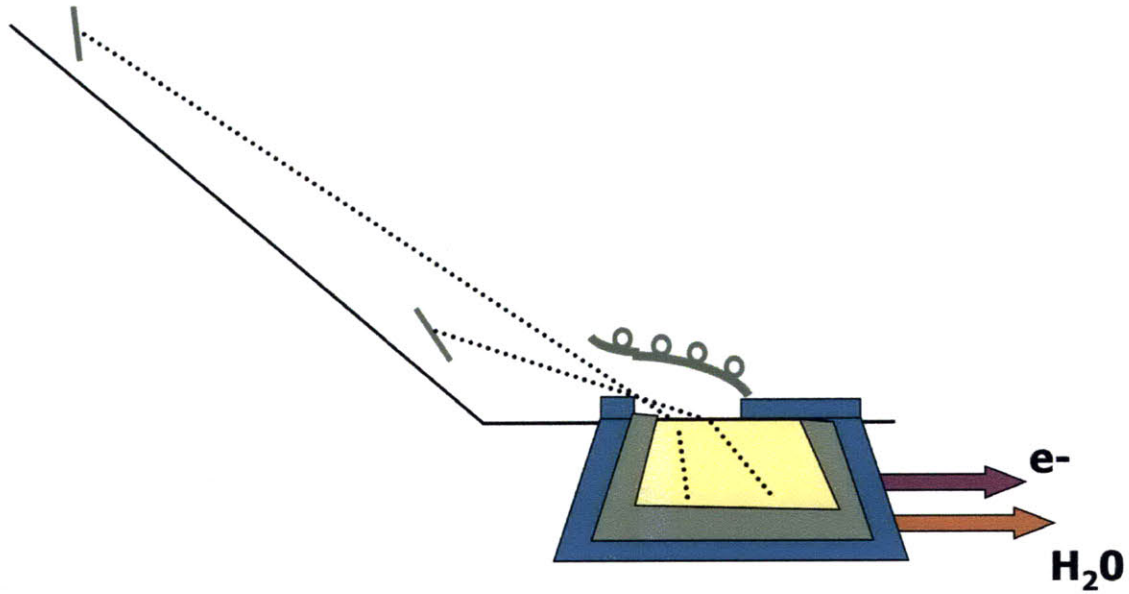
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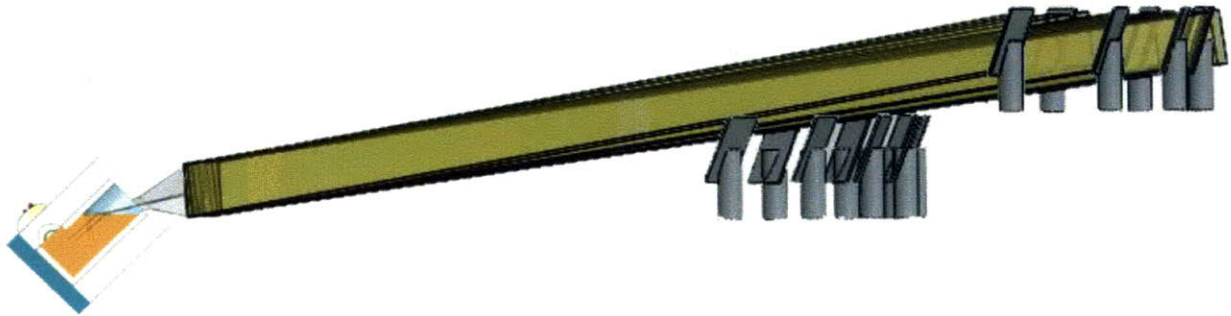
(b)

Figure 3: CSPonD heliostat beam down configuration. (a) Sketch of heliostat beam down. The heliostats beam down sun's rays to output electricity and clean water [9]. (b) A solid model showing the layout of the heliostats on a hill [10]. The heliostats focus the sun's rays into the tub, melting the salt, creating a reservoir of heat energy from solar energy [11].
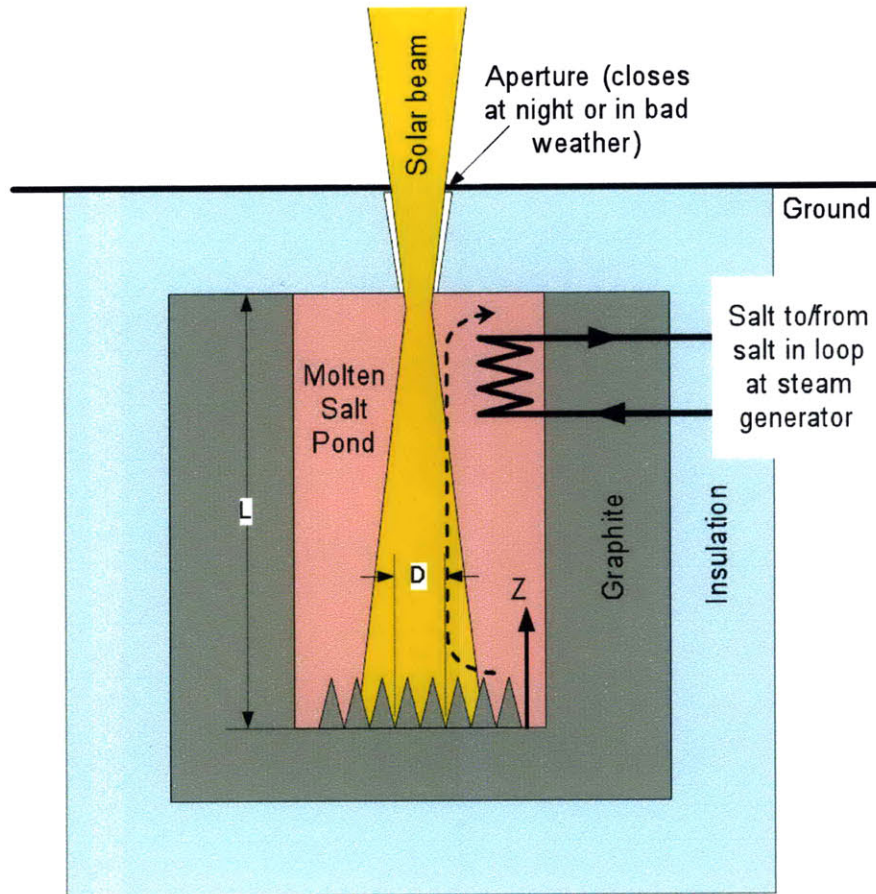
Figure 4: The CSPonD molten salt receiver uses molten salt to couple the collection and storage of the sun's energy. Concentrated sunlight is directed into the receiver where the molten salt volumetrically absorbs the sun's energy storing it as heat energy for later use. The heat energy stored in the molten salt can then be used as the heat source in a thermodynamic power cycle to generate electricity [10].

1.2.   Heat Exchangers

Heat exchangers are devices used to transfer heat from one medium to another. They are used to facilitate heat transfer in applications such as space-heating, air-conditioning, power production, waste heat recovery, and chemical processing [2].

A common type household heat exchanger is the water boiler used to provide hot water for use in everyday life. Figure 5 shows the inside of a water boiler. The two u-shape heating elements, similar to those found on electric stoves, and the tank make up the heat exchanger. Cold water is pumped into the tank, heated by the two heating elements, and then pumped out as hot water. The hot water, being less dense than cold water, rises to the top of the tank is pumped out first.

hot water out

cold water in

heating elements
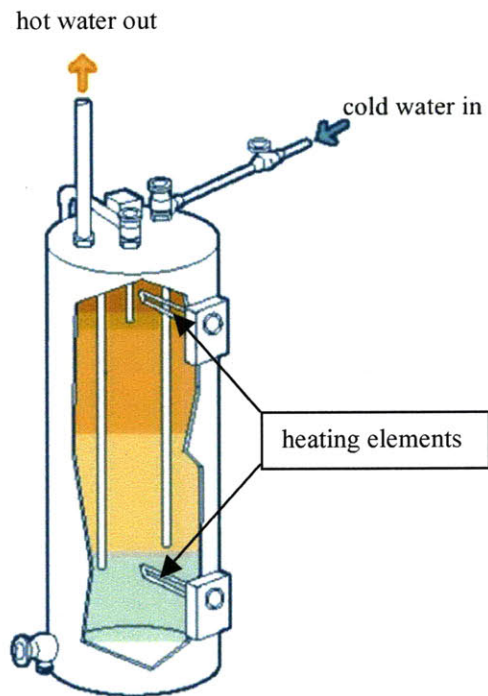
Figure 5: An electric water boiler. Cold water enters the tank and is heated by two u-shaped electrical resistance heaters. The hottest water rises to the top and is pumped out first [12].

1.3. Using heat to generate electricity

Heat exchangers are integral in the process of generating electricity through a thermodynamic power cycle known as the Rankine cycle, used to generate the majority of the world's electricity. In the Rankine cycle heat energy is converted to mechanical energy which in turn can be converted into electric energy through a generator.

The Rankine cycle consists of four main processes as shown in Figure 6. The working fluid starts off at a cold temperature as a liquid and is pumped from low to high pressure. Heat is then transferred into the fluid from an external heat source such as burning fuel through the use of a heat exchanger, which boils the fluid into a vapor. The vapor then expands inside a turbine creating a work output as it drops in pressure and temperature. The working fluid is then sent to a condenser, a heat exchanger that extracts heat energy from the wet vapor exiting the turbine, condensing the fluid into a pure liquid [5].
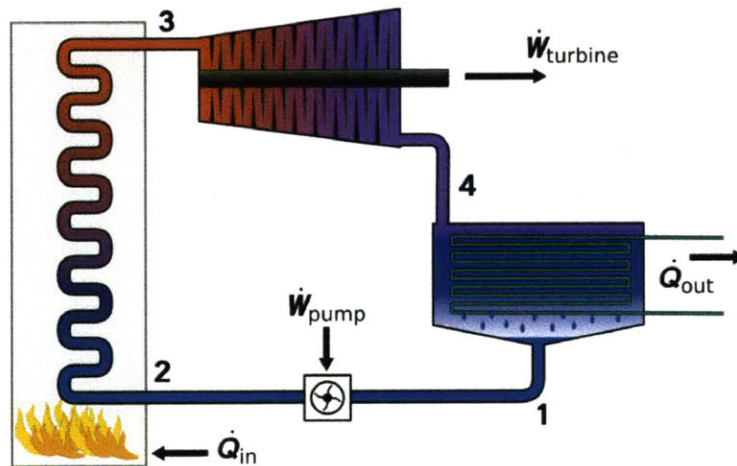
Figure 6: The Rankine cycle. In the first part of the Rankine cycle the working fluid is in its liquid phase and is pumped from low pressure to high pressure. The working fluid is then sent into a heat exchanger where the liquid boils into vapor as heat is transferred into the fluid by an external heat source such as burning fuel. The high pressure, high temperature vapor is then sent into a turbine where it expands creating a work output while it drops in temperature and pressure. In the last step of the cycle, the fluid flows through a condenser, a heat exchanger that extracts energy from the working fluid, cooling it and condensing it into a pure liquid [13].

2. Extracting energy from the CSPonD molten salt bath

This section details the reasoning behind the proposed method of heat extraction from the CSPonD molten salt bath. Section 2.1 serves as an introduction as to how energy can be transferred by looping working fluids of different temperatures next to each other. Section 2.2 goes over the functional requirements of the proposed heat exchanger design. Section 2.3 goes over some of the challenges of using the molten salt bath as a heat source. Section 2.4 proposes a method of heat extraction from the molten salt bath to be used as the basis of analysis for the rest of the paper.

2.1. *Looping working fluids in a heat exchanger*

An important consideration in laying out a potential power cycle is how to provide the source of heat energy to the working fluid. The Rankine power cycle requires that the working fluid be heated by an external source and then be sent into a turbine. One way to provide heat energy to the working fluid in the Rankine power cycle is to flow a hot fluid next to the wall enclosing the cold working fluid.

A schematic of a power cycle heat exchanger wherein a "hot" fluid is used to heat a "cold" fluid is shown in Figure 7. The hot fluid, fluid A, acts as the source of heat energy for the working fluid of the power cycle, labeled as fluid B. Fluid A enters the heat exchanger at a hotter temperature than fluid B, giving fluid B some or all of its heat energy.
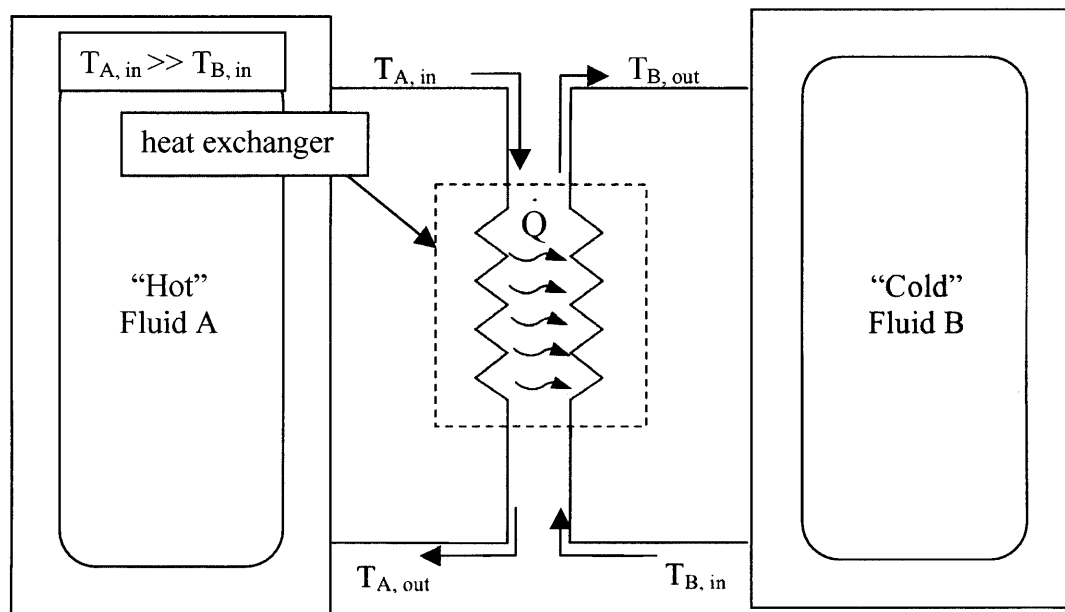
Figure 7: Two working fluids being looped as they exchange energy in a heat exchanger. Fluid A acts as the source of heat energy for fluid B resulting in a positive heat transfer for fluid B and an equal but opposite heat transfer for fluid A. This results in fluid A exiting the heat exchanger at a lower temperature than it entered and fluid B exiting the heat exchanger at a higher temperature than it entered.

2.2. Heat exchanger functional requirements

The proposed heat exchanger design must meet certain functional specifications in order for it to be considered a viable part of the CSPonD project. The detailed design is covered in detail in section 4. The system must generate an average of 15 kW of continuous thermal power, minimize thermal loads, withstand high temperatures, etc. For a detailed list of the functional requirements evaluation refer to Table 5 (functional requirement evaluation table is referred as FRDPARRC). The FRDPARRC table organizes the design functional requirements, design parameters, essential analysis proceedings, references, risks, and countermeasures in table form.

2.3. Using a molten salt working fluid in a heat exchanger

The molten salt bath can act as the heat source in a power cycle such as the Rankine cycle by using a heat exchanger to shuttle heat energy from the molten salt bath to the power cycle working fluid, however, the temperature behavior of the molten salt precludes the use of the simple heat exchanger design illustrated in Figure 7. The high temperatures reached by the molten salt can fluctuate due to varying magnitudes of solar energy absorption resulting from variability in weather. Varying the rate of energy extraction from the bath can also cause temperature fluctuations in the molten salt. The varying temperature of the molten salt has important implications in the design of the heat exchanger used to extract energy from the bath.

2.3.1 Minimizing thermal loads using an intermediate working fluid

In exchanging heat energy between working fluids, the effects of the temperature gradient of the heat exchanger must be taken into consideration to assure the safe operation of the heat exchanger. Consider a Rankine cycle, as shown in Figure 8, using the CSPonD molten salt bath as the heat source, water as the Rankine cycle working fluid, and the simple heat exchanger from Figure 7 as the means by which the two working fluids exchange energy. Figure 8 does not capture the complexity of a real power cycle system but serves to highlight the challenges involved in running the molten salt from the bath directly next to the power cycle working fluid, which in this case is steam.

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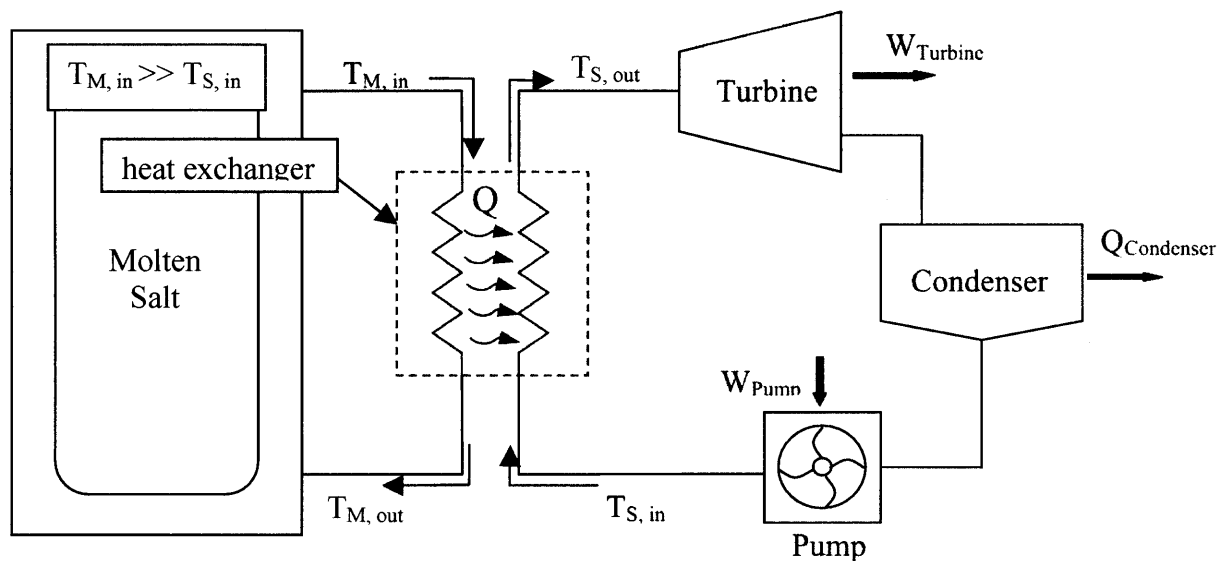
Figure 8: A Rankine cycle with the simple heat exchanger set up from Figure 7. T_m and T_s represent the temperature of the molten salt and steam respectively. Due to varying magnitudes of solar energy absorption resulting from variability in weather, the molten salt varies in weather creating undesired thermal loads on the heat exchanger.

Due to the varying solar fluxes seen by the molten salt and other factors that dictate the molten salt's temperature the incoming temperature of molten salt is not stable and fluctuates. Thermal stresses and large temperature gradients in the heat exchanger material between the molten salt and steam can cause it to lose its mechanical integrity and possibly fail. In addition, the varying molten salt temperature can result in undesirable variations in the steam exit temperature. Precise and accurate control of the exit temperature of the steam is advantageous as it can minimize variations in temperatures as seen by the downstream power components, thus extending the lifecycle of the components that the steam interacts with and improving the efficiency of the turbine. Another consequence of direct exchange of thermal energy between molten salt and steam are the risks associated with the steam getting too hot or the molten salt getting too cold. Having the steam approach the temperature of the molten salt is undesirable due to the highly corrosive nature of steam near those temperatures on metals and refractory alloys. Drawing the molten salt out of the bath risks the molten salt temperature falling below its melting point and solidifying.

Placing an intermediate heat exchanger between the molten salt and the steam is a means of attaining a constant temperature gradient on the primary heat exchanger loop providing energy to the steam. Figure 9 shows a schematic of an intermediate heat exchanger and an intermediate working fluid, fluid C, being used to collect heat from the molten salt and shuttle it to the steam. Thermal loads can be reduced using an intermediate heat exchanger between the molten salt and steam. The working fluid of the intermediate heat exchanger, fluid C, must be chosen so that it can safely interact with the high temperature of the molten salt without risking failure in the material composing the heat exchanger. Similarly, fluid C must also transfer heat with the steam

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without making the steam too hot or risking failure in the material composing the primary heat exchanger. Fluid C allows better control over the temperatures the steam interacts with by serving as a buffer between molten salt and the steam.

Compared to the heat exchanger setup in Figure 7, the series of loops in Figure 9 offers better control over the temperatures seen by the power cycle working fluid, but the additional intermediate heat exchanger loop adds complexity and costs to the heat extraction system. The addition of the intermediate heat exchanger loop requires an additional pump, heat exchanger, working fluid, and storage tank. The added complexity of running three loops instead of two also increases maintenance and supervision costs.
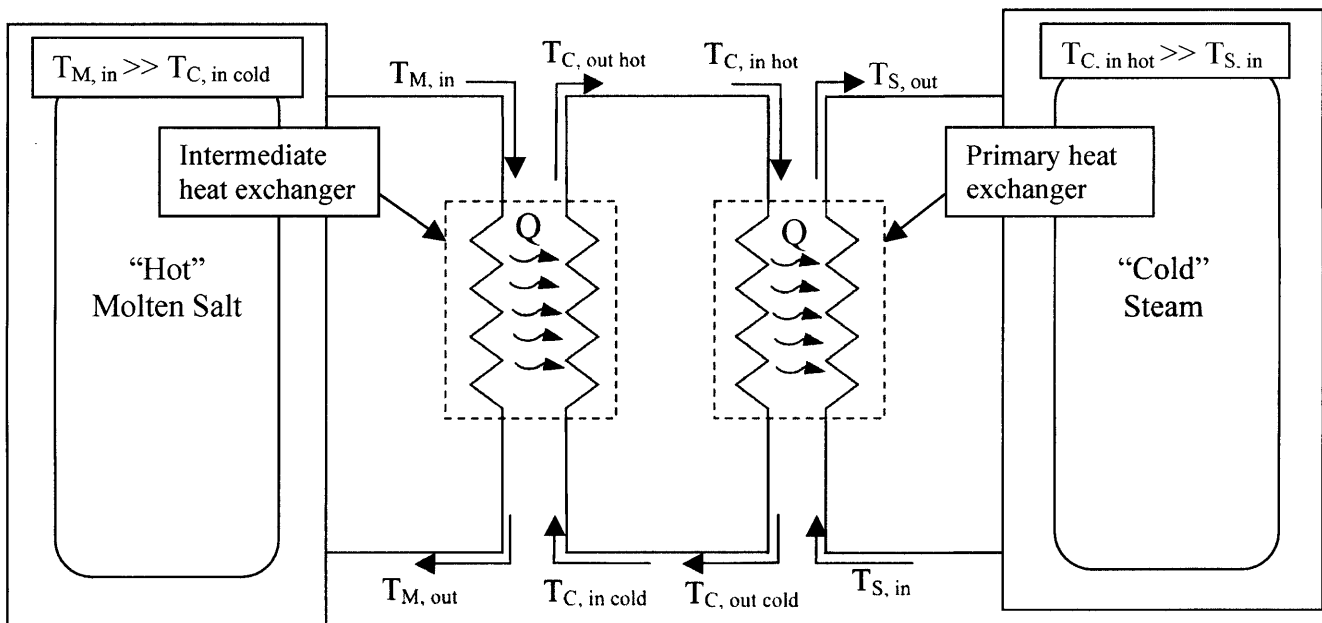
Figure 9: Minimizing thermal loads through the use of an intermediate working fluid. An intermediate working fluid, fluid C, is looped within two heat exchangers as it shuttles heat energy from the molten salt bath to the steam which is commonly used as a power cycle working fluid. Fluid C allows better control over the temperatures the steam interacts with by serving as a buffer between the molten salt and the steam.

2.4. The proposed heat extraction system

To lessen the drawbacks associated with looping three working fluids in the heat extraction system, a heat extraction system, shown in Figure 10, combines the simplicity of Figure 7 and the complexity of Figure 8. The proposed heat extraction system loops two working fluids instead of three. Instead of looping the molten salt, the intermediate working fluid is looped inside the molten salt bath where it experiences a positive heat transfer from the molten salt and shuttles it to the power cycle working fluid. Not looping the molten salt as before results in a lower rate of heat transfer between the intermediate working fluid and the molten salt, but it makes the heat extraction system easier to implement and run. Looping the molten salt out of the bath risks damaging the pipes due to the high temperatures of the molten salt, a problem further examined in section 4.3. The flow rate of the intermediate working fluid can be varied to vary the heat transfer rate in the two heat exchangers.
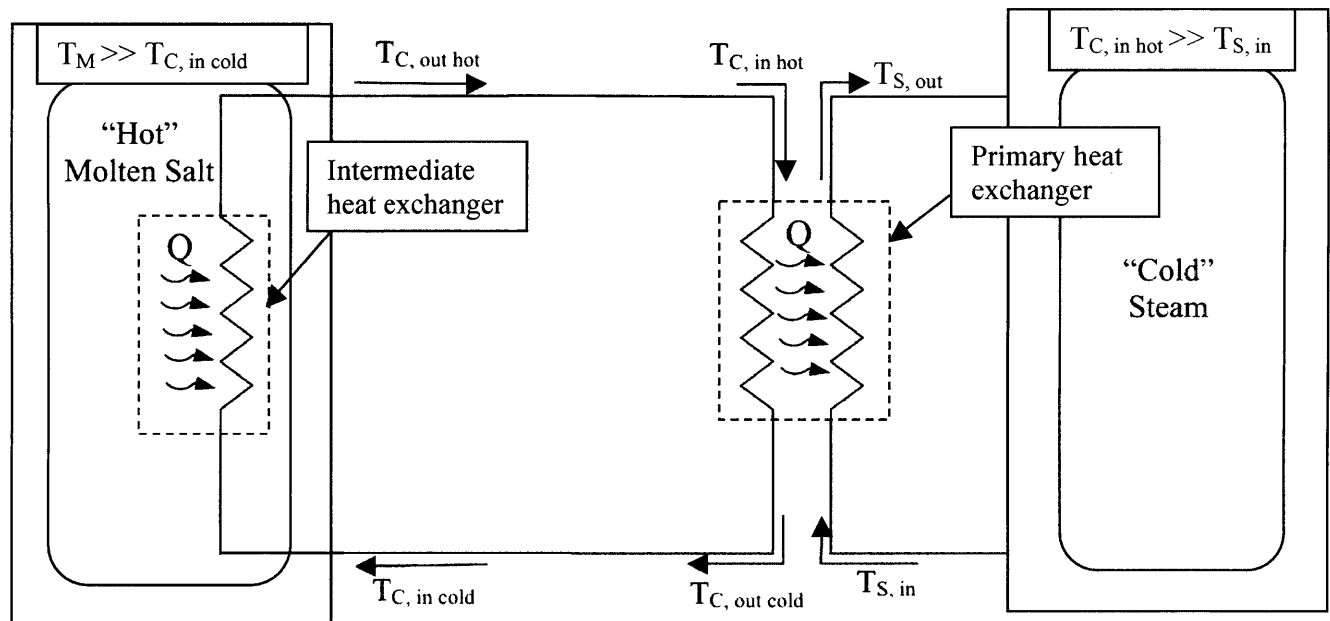
Figure 10: The proposed heat extraction system loops two working fluids instead of three. Instead of looping the molten salt, the intermediate working fluid is looped inside the molten salt bath where it experiences a positive heat transfer from the molten salt and shuttles it to the power cycle working fluid.

3. Background

This section gives a general introduction to heat exchanger design parameters, such as: flow arrangements, heat transfer, molten salts, and high temperature metal alloys. The heat transfer phenomenon in heat exchangers is discussed in section 3.1. Molten salts and their use as working fluids in high temperature applications are discussed in section 3.2. A general overview of the different high temperature alloys is discussed in section 3.2.

3.1. Heat Exchangers and Heat Transfer

Heat exchangers are classified by their flow arrangement. Concentric flow heat exchangers have one of the simplest flow arrangements. The arrangement can consist of a pair of concentric tubes, one inside the other, with two fluids flowing in each tube exchanging heat through the pipe wall. Two concentric flow heat exchanger arrangements are: parallel flow and counter-flow, as shown in Figure 11.
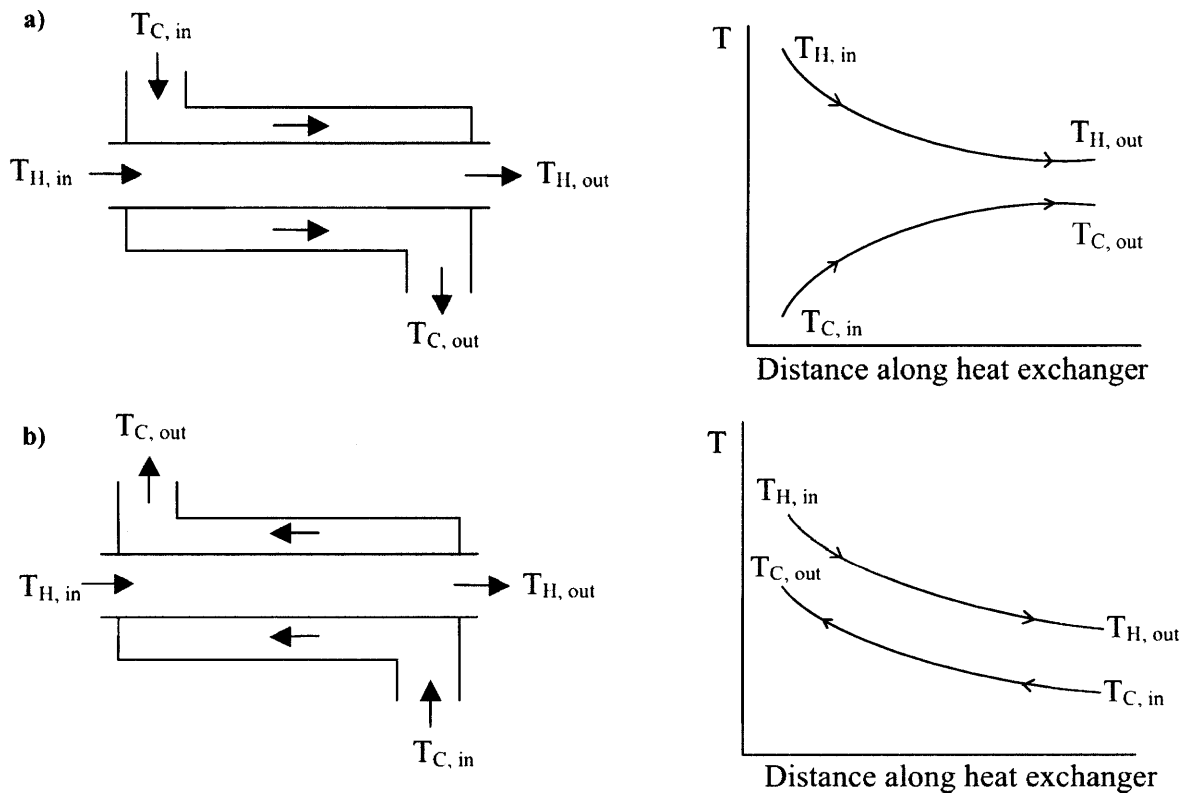
Figure 11: Flow arrangements and temperature distributions along the heat exchanger. T_H and T_C represent the hot and cold fluids going through the heat exchanger respectively. (a) Parallel flow arrangement. (b) Counter-flow arrangement [5].

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A heat transfer model is necessary for predicting and optimizing the performance of a heat exchanger design. A heat transfer model can aid to evaluate the effect of adjusting different parameters such as working fluids, flow rates, heat exchanger material, etc.

Given a simple heat exchanger defined by a hot and cold fluid separated by a solid wall as the two fluids flow next to each other, the differential form of the first law of thermodynamics shows that the only source of power generation within the heat exchanger is through heat transfer. The heat transfer for the hot and cold leg, respectively are shown in Eq 1 and Eq 2.

$$\dot{Q}_h = \dot{m}_h c_{p,h} (T_{i,h} - T_{o,h}) \tag{1}$$

$$\dot{Q}_c = \dot{m}_c c_{p,c} (T_{i,c} - T_{o,c}) \tag{2}$$

where $\dot{Q}$ is heat transfer rate, $\dot{m}$ is mass flow rate, c_p is specific heat at constant pressure, T_i is inlet temperature, and T_o is outlet temperature, with subscripts h and c representing the hot and cold fluid respectively.

The rate of this heat transfer is driven by the difference in temperature between the two fluids but limited by thermal resistances between the two fluids. The total thermal resistance, R_{tot} see Eq 3, is defined as the ratio of the difference in temperature between the two fluids to the corresponding heat transfer rate.

$$R_{tot} = \frac{\Delta T_{overall}}{\dot{Q}} \tag{3}$$

where R_{tot} is the sum of the thermal resistances and $\Delta T_{overall}$ is overall temperature difference.

In a heat exchanger design an overall heat transfer coefficient is used to characterize the overall resistance to heat transfer between the two fluids. The overall heat transfer coefficient, U see Eq 4, lumps the effect of all the individual thermal resistances into one resistance parameter. It is defined as the inverse of the product of the sum of thermal resistances and area.

$$U = \frac{1}{R_{tot} A} \tag{4}$$

where U is the overall heat transfer coefficient and A is the surface area of the vessel with which the fluid is in contact. With the overall heat transfer coefficient, a form of Newton's law of cooling, shown in Eq 5, can be used to determine the heat exchanger area required to meet the thermal output of the heat exchanger.

$$\dot{Q} = UA\Delta T_{lm} , \tag{5}$$

where ΔT_{lm} is the log mean temperature difference. The log mean temperature difference, a weighted average used in thermal applications, is defined in Eq 6,

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln \frac{T_2}{T_1}} \tag{6}$$

where ΔT_1 represents the temperature difference of the two fluids at one end of the heat exchanger, and ΔT_2 represents the temperature difference of the two fluids at the other end of the heat exchanger . ΔT_1 and ΔT_2 are defined in Eq 7 and Eq 8, respectively.

$$\Delta T_1 = T_{i,h} - T_{o,c} \tag{7}$$

$$\Delta T_2 = T_{o,h} - T_{i,c} \tag{8}$$

In the example of two fluids exchanging heat energy through a solid wall, convection is the dominant mode of heat transfer, and the convection heat transfer coefficient is the term of proportionality of the heat transfer rate between the two fluids and the temperature difference between the two fluids. The rate of heat transfer due to convection is defined in Eq 9,

$$\dot{Q}_{conv} = hA(T_s - T_\infty) \tag{9}$$

where $\dot{Q}_{conv}$ is the heat transfer rate due to convection, h is the convection heat transfer coefficient, T_s is the temperature of the wall with which the fluid is in contact, and T_∞ is the temperature of the fluid.

The convective heat transfer coefficient can be found through the dimensionless Nusselt number, equal to the ratio of convective to conductive heat transfer of the fluid.

$$Nu = \frac{hL_c}{k} \tag{10}$$

where k is the thermal conductivity of the fluid and L_c is the characteristic length scale of the vessel through which the fluid is flowing. To determine the convective heat transfer coefficient the Nusselt number must be known.

Through experimentation correlations have been developed to calculate the Nusselt number under different flow conditions. These correlations express the Nusselt number in terms of the dimensionless Reynolds and Prandlt numbers which serve to characterize the behavior and properties of the fluid flow. The Reynolds number is used to characterize flow rate as turbulent or laminar. As shown in Eq 11, the Reynolds number is the ratio of inertial forces to viscous forces acting on the fluid.

$$Re = \frac{\rho v L_c}{\mu} \tag{11}$$

where Re is the Reynolds number, ρ is density, v is velocity, and μ is kinematic viscosity. The Prandtl number is the ratio of thermal to momentum diffusivity of the fluid.

$$Pr = \frac{\mu c_p}{k} \tag{12}$$

In modeling the heat transfer of a heat exchanger, the overall heat transfer coefficient and by association, the convective heat transfer coefficient, are important in determining the heat transfer rate of the heat exchanger. By finding an expression for the overall heat transfer coefficient, appropriate design decisions can be made to achieve a desired heat transfer rate; namely, the selection of working fluids, wall material, and fluid flow rates.

3.2. Molten Salts

Molten salts are salts in their liquid phase that are normally solid at standard temperature and pressure. They can make excellent heat transfer fluids in high temperature applications (260°C - 1400°C) because of their high melting points, stability at high temperatures, and low viscosities. The size of pipes and other equipment used for molten salts together with the pumping power required is much smaller than if gases were employed in the same high temperature applications. The wall thicknesses required in using molten salts are also much smaller than those required in high-pressure steam systems operating near the same temperatures [14].

Certain molten salts can serve as energy storage mediums. Their high melting temperatures and high thermal capacitances make it so they retain heat for periods longer than 24 hours. They also have high energy densities reducing the amount of molten salt needed in applications normally carried out by traditional working fluids. Their ability to store this heat energy means that the sun's radiant energy can be stored in the salt as heat energy to produce electric energy at any time of the day [14].

Precautions must be taken when using molten salts in heat exchangers due to the corrosive nature of certain molten salts on select heat exchanger construction materials. The heat exchanger must be designed for a high degree of leak tightness to prevent contamination of the salt by water vapor or by oxygen to keep corrosion rates low. The low freezing points of molten salts also require that the system be designed to preheat materials in contact with the salts and with good drainage to prevent the salt from freezing in the system.

3.3. High Temperature Metal Alloys

The high operating temperatures of molten salts limit the range of possible candidate materials for the construction of the heat exchanger. Corrosion, creep, and thermal loads are especially pronounced at high temperatures precluding the use of common heat exchanger materials such as copper and aluminum. Steels and a class of metals called superalloys, alloys developed specifically for high temperature applications, are two groups of alloys commonly used in high temperature applications because of their resistance to high temperature corrosion, creep, and thermal loads. The oxidation resistance is based on the alloying element chromium and the protective chromia layer formed on the metal surface through oxidation. Resistance to oxidation, one of the most dominant forms of corrosion at high temperatures, is especially important in high temperature applications [15].

The oxidation of a metal occurs when a metal is exposed to oxygen and a non-metallic oxide layer forms on the surface of the metal, degrading away the surface of the metal that the oxygen is in contact with. This oxide layer can serve to protect the metal from further oxidation and removal of material by forming a protective layer that acts as a diffusion barrier between oxygen and the metal beneath the protective layer [15]. If the oxide layer does not adhere to the metal however, the material slowly deteriorates at the oxide layer and the oxidation reaction continues unimpeded. The adherence of the oxide layer to the metal surface is determined by the ratio of molar volume of the metal to the molar volume of the oxide layer formed on it, known as the Pilling-Bedford ratio (PBR). When the ratio is close to one the oxide layer adheres to the metal surface and when the ratio is large the oxide layer does not adhere to the metal surface and the corrosion of the material proceeds at a faster rate. High temperature metals are often alloyed with chromium to generate a protective chromium oxide layer, i.e. chromia (Cr_2O_3), which is stable at high temperatures. In addition to resistance to corrosion, heat exchanger construction

materials exposed to the high temperatures of molten salt must also have exceptional creep resistance and be strong enough to withstand thermal loads [15].

Creep is the time-dependent deformation of a material under a static mechanical stress. It can cause undesired deformations in a material and cause a vessel to rupture. Materials are especially vulnerable to creep deformation at elevated temperatures ($T > 0.4\ T_{melting}$) [16]. The creep behavior of a material is an important factor in determining the integrity of the material at elevated temperatures. In addition to creep, thermal loads can contribute to the failure of a heat exchanger component. If a metal is constrained thermal expansion can cause the material to deform and yield, encouraging crack formation and growth that can lead to the failure of the material.

4. Heat Exchanger Design

This section describes the methodology behind major heat exchanger design decisions. Section 4.1 lays out the heat transfer equations for the heat exchanger flow arrangement described in section 2.4. Section 4.2 compares the different working fluids being considered to serve as the intermediate working fluid that shuttles energy from the molten salt bath to the power cycle working fluid. Section 4.3 goes over the selection of the heat exchanger construction material.

4.1. Modeling Heat Transfer

The heat transfer model of the proposed heat exchanger layout was developed through the heat transfer modeling process described in section 3.1 of the background. In order to model heat transfer the Nusselt number correlation for the specific geometry of the heat exchanger and the fluid flow regime (turbulent or laminar) is necessary. The Nusselt number correlation for turbulent, fully developed flow in a cylinder describes the flow conditions of the proposed heat exchanger layout [5].

$$Nu = \frac{(\frac{f}{8})(\text{Re}-1000)\,\text{Pr}}{1+12.7(\frac{f}{8})^{\frac{1}{2}}(\text{Pr}^{\frac{2}{3}}-1)} \tag{13}$$

where f is the Darcy friction factor. The Darcy friction factor correlation for smooth pipes in Eq 14 is valid for $3000 \leq \text{Re} \leq 5\text{x}10^{6}$.

$$f = (0.790\ln\text{Re}-1.64)^{-2} \tag{14}$$

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4.2. Molten Salt Considerations

To meet the thermal power demands of the bath-side heat exchanger, an appropriate intermediate working fluid must be chosen to shuttle energy from the molten salt bath to the power cycle working fluid. Different compositions of molten salts are being considered to serve as the intermediate working fluid due to the physical and chemical properties of molten salts that allow them to be stable and manageable (able to work with) at very high temperatures(>700 °C). The molten salt chosen as the working fluid for the heat exchanger must remain physically and chemically stable in its liquid form while it exchanges heat energy from the "hot" molten salt in the bath (up to ~1000 °C; as low as ~750 °C) and from the "cold" superheated steam in a steam generator (exit temp of ~550 °C). In addition, the working fluid molten salt has to be chemically inert enough as to not corrode the materials it flows through. For insight into the intricacies in the selection of an appropriate intermediate working fluid, a FRDPARRC table, refer to Table 6 in the appendix.

There are three families of salts that fall within the parameters set by the functional requirements; fluoride salts, alkali fluoroborates, and chloride salts. In evaluating each of the three families of molten salt as a possible working fluid candidate select individual salt compositions representative of each salt family were chosen to compare against one another as shown in Table 1. Certain metrics of performance were used to help rank each of the salts. The three metrics used to compare each of the candidate molten salt working fluids are heat transfer performance, ease of preparation and corrosion behavior, and cost.

Much of the research focused on using molten salt as a working fluid has been done in the context of using nuclear energy to generate useful electricity. The working fluid molten salts in these nuclear applications are selected with special consideration given to compatibility of the molten salt with fissile materials. In researching different reports done on using molten salts as working fluids in power generating applications, D.F. Williams's report, "Assessment of Candidate Molten Salt Coolants for the NGNP/NIH Heat-Transfer Loop", done in the Nuclear Science and Technology Division of the Oak Ridge National Laboratory, differs from most work done in investigating molten salts as working fluids in that the molten salt working fluid stays "clean" throughout its application, free from fissile contamination [4]. The selection of salt compositions examined as intermediate working fluid candidates for the CSPonD heat extraction system are drawn from Williams's report. Although the specific application of the molten salt working fluid in Williams's report differs from the CSPonD application in that the molten salt working fluid shuttles heat from the Next Generation Nuclear Plant (NGNP) to the Nuclear Hydrogen Initiative (NHI) hydrogen-production plant, the screening parameters and metrics of molten salt working fluid performance are quite similar. Both the CSPonD and NGNP applications require that the working fluids be stable at high temperatures, melt at useful temperatures, and are compatible with high temperature materials [4].

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Table 1: Summary of important properties of the working fluid candidates at 700 °C [4]

Salt Composition (mole percent)	Melting Point (°C)	Density (kg/m^3)	Heat capacity (J/kg-K)	Viscosity (N-s/m^2)	Thermal Conductivity (W/m-K)
LiF-NaF-KF (46.5-11.5-42)	454	2020	1886	0.0029	0.92
KF-ZrF$_4$ (58-42)	390	2800	1047	<0.0051	0.45
NaF-ZrF$_4$ (59.5-40.5)	500	3140	1173	0.0051	0.49
KF-KBF$_4$ (25-75)	460	1700	1305	0.0009	0.38
RbF-RbF$_4$ (31-69)	442	2210	909	0.0009	0.28
LiCl-KCl (59.5-40.5)	355	1520	1198	0.00115	0.42
NaCl-MgCl$_2$ (58-42)	445	1680	1097	0.00136	0.5
KCl-MgCl$_2$ (68-32)	426	1660	1160	0.0014	0.4

4.2.1 Heat transfer comparisons

To differentiate between the different salts and their heat transfer performance, the heat exchanger heat transfer rate as a function of heat exchanger surface area was plotted under realistic heat exchanger operating conditions. As shown in Figure 12, the fluoride salt LiF-NaF-KF outperforms all the other candidate salts, providing the greatest heat transfer rate per unit area translating into lower heat exchanger construction materials costs relative to the other working fluid candidates.
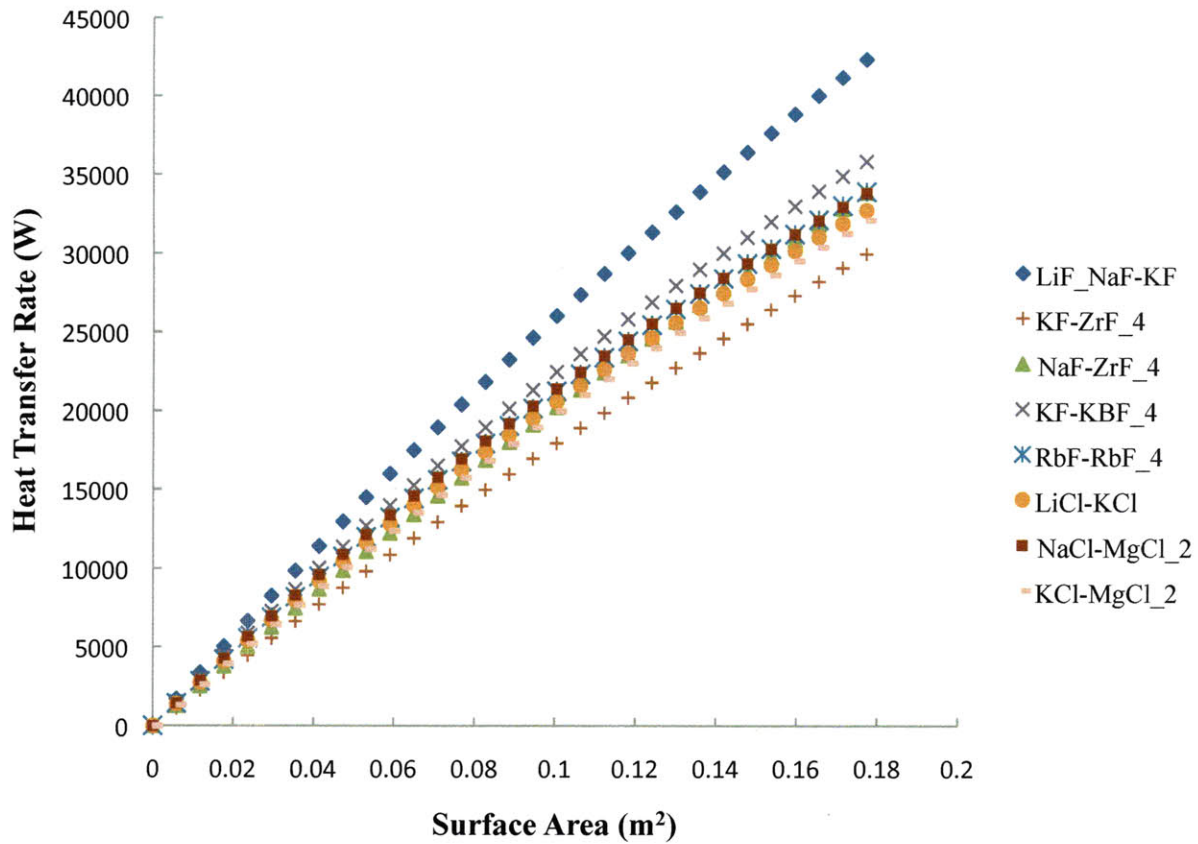
Figure 12: Heat transfer rate vs. surface area for the candidate working fluids flowing through two schedule XXS (double extra strong) 0.75 in. steel pipes (~0.0188 m inner diameter, ~ 0.0267 m outer diameter) with a fluid velocity of 2 m/s, giving an effective surface are of 0.0591 m^2/m. Under the aforementioned parameters, the LiF-NaF-KF salt composition delivers the highest heat transfer rate per unit area, resulting in lower heat exchanger material costs when compared to the other working fluids.

4.2.2 Ease of use and corrosion behavior

The high operating temperatures of molten salts coupled with their chemical compositions make them particularly corrosive to containment materials. Although molten salt corrosion data is extensive, data useful in choosing corrosion resistant is limited and fragmented and corrosion data for molten chlorides and molten fluoroborates as working fluids at temperatures approaching the levels anticipated in the CSPonD project is nonexistent [6] [4]. Corrosion data from tests performed with chloride salts near the operating temperatures in consideration do not conform to any expected or predictable trends. This is likely due to the variability of impurities in the chloride salts tested [4]. Although corrosion data is limited several general rules should be observed. The material should form a passive oxide film with molten salt to reduce the oxidation rate of the material and the entry of oxidizing species such as oxygen and water should be minimized [6]. In addition, before the salts are used as working fluids they should be prepared as to minimize the amount of oxidizing impurities in the salt.

The molten salt candidates have varying degrees of ease of preparation and use. Chloride salts are troublesome to prepare because of the relatively high levels of oxygen-containing impurities found in the salts and the lack of treatments available to remove these impurities [4].

Fluoroborate salts present the greatest challenges because of the presence of a highly toxic high pressure ($P > 0.5P_{atm}$) BF_3 cover gas at the operating temperatures in consideration. In order to maintain the fluoroborate salt composition the BF_3 cover gas pressure must be maintained. The monitoring and control of the BF_3 cover gas is necessary because the gas can escape through shaft seals and bearings on pumps or other machinery. Another concern in using fluoroborate salts is the highly corrosive byproduct of the BF_3 cover gas whenever in contact with moist air [4].

Fluoride salts are the easiest to prepare and use. They have been used as high temperature working fluids in nuclear power applications and should therefore be ranked as the best candidate in terms of corrosion behavior and ease of use [4].

4.2.3 Cost Comparisons

Table 2 shows the costs of the raw materials associated with the various candidate salt mixtures. Based on these raw material costs, it is clear that many of the chloride salts are the least expensive, followed by fluoroborate salts, and then fluoride salts.

Table 2: Estimated raw material costs for various salt mixtures[4]

Salt Mixture	Raw Material Cost ($/kg)	Cost/Volume ($/L at 700°C)
LiF-NaF-KF	7.82	15.79
KF-ZrF$_4$	4.85	13.58
NaF-ZrF$_4$	4.02	12.63
KF-KBF$_4$	3.68	6.26
RbF-RbF$_4$	*	*
LiCl-KCl	5.07	7.71
NaCl-MgCl$_2$	0.25	0.42
KCl-MgCl$_2$	0.21	0.35

*Commodity price unavailable.

4.3.   Heat Exchanger Material Considerations

The high operating temperatures of the heat exchanger limits the selection of heat exchanger construction to high temperature metal alloys. The proposed heat exchanger design has a pipe immersed in the molten salt bath. Temperatures inside the molten salt bath can reach as high as ~1000°C at the top of the bath and as cool as 750°Cat the bottom of the bath. The material selected for the heat exchanger pipes has to maintain its structural integrity under these high temperatures. In addition, the material must be resistant to the corrosive nature of the molten salt it will constrain. A FRDPARRC table, as shown in Table 7in the appendix, summarizes the constraints that limit what materials can be used to make the heat exchanger along with the objectives to be used in ranking the materials that meet the design constraints.

In researching metals capable of performing under the high service temperatures required of the heat exchanger design, seven candidate metal alloys were identified as capable of meeting the functional requirements summarized in Table 7. To determine the best candidates for the heat exchanger, the alloys were evaluated based on their high temperature mechanical properties (creep and creep fatigue properties), physical properties, resistance to corrosion, heat transfer properties, and price.

Table 3: Candidate materials for the heat exchanger [18] [19] [20] [21] [22] [23] [24]

Metal	Melting Point (°C)	Yield Stress (MPa)	Cost ($/kg)
310S SS	1454	147 @ 760 °C 56 @ 982 °C	27.14
321 SS	1400 - 1425	130 @ 732 °C 115 @ 816 °C	(n/a)
Inconel 617	1398 - 1446	381 @ 760 °C 75 @ 982°C	25.83
Inconel 625	1290 - 1350	381 @ 760 °C 75 @ 982 °C	64.53
Incoloy 800HT	1357 - 1385	130 @ 550 °C 90 @ 760 °C	34.09
Hastelloy X	1260 - 1355	237 @760 °C 91 @982 °C	69.8
Haynes 230	1301 - 1371	253 @760 °C 118 @982 °C	131.97

4.3.1 Evaluating the heat exchanger candidate materials

Due to the continuous high temperature operating conditions of the heat exchanger, the candidate material must have exceptional creep resistance. Creep is the slow accumulation of strain under stress at an elevated temp over a period of time [16]. There is little or no correlation between the creep properties of a material and its room temperature mechanical properties, so the material must be tested to determine its creep behavior [16].

The *ASME Boiler and Pressure Vessel Code* recognizes creep and creep deformation as a high temperature design limitation and provides allowable stresses for all alloys used in the creep range [17]. The allowable stress values for the candidate materials as a function of service temperature are plotted in Figure 13. Inconel 671, Inconel 625, and Haynes 230 have the highest allowable stresses at 750 °C, while the stainless steel alloys 310S and 321 have the lowest allowable stresses at 750 °C. As the service temperature approaches 900 °C the allowable stresses of all the alloys start to cluster together, except for the stainless steels which have lower operating temperatures than the other candidate alloys.
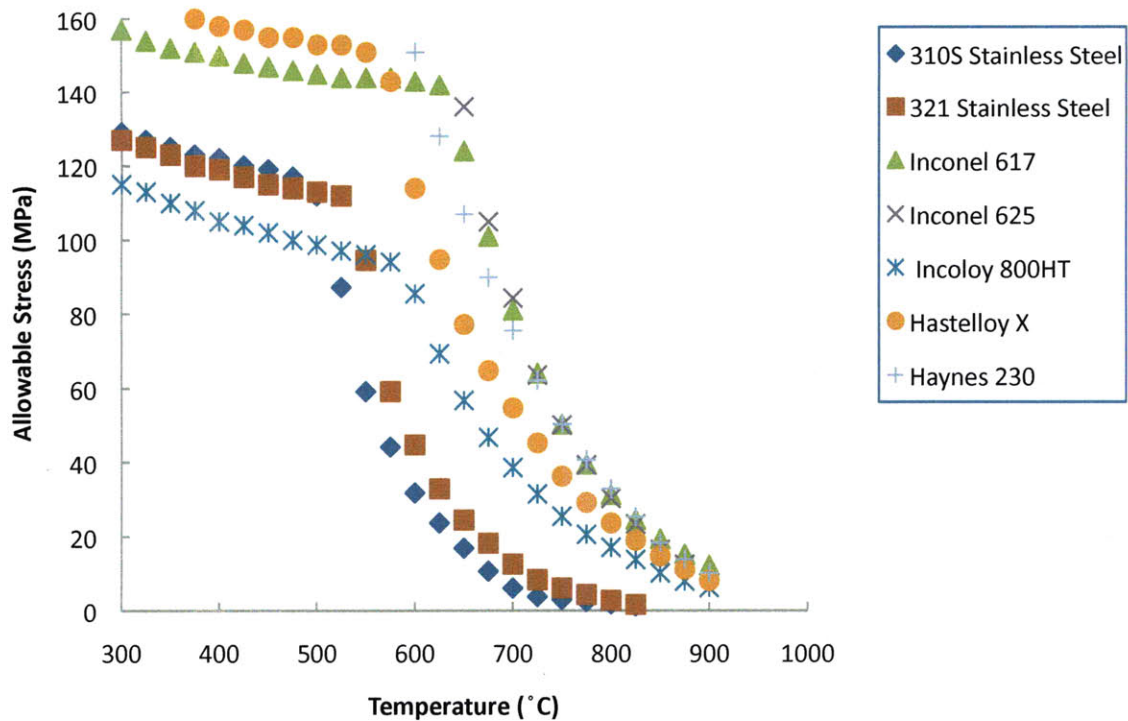
Figure 13: A plot of the allowable stresses as a function of temperature for the seven candidate materials as specified by the ASME Boiler and Pressure Vessel Code, Section II-Materials, Part D [17].

Table 4 presents oxidation resistance data collected by Haynes International, a manufacturer of alloys used in high temperature and corrosive applications [15]. Relative to the oxidation data collected at the higher temperatures, the oxidation resistance data collected at 980 °C is the most relevant due to it being collected at the temperature closest to the operating temperature of the heat exchanger. Metal alloys Haynes 230 and Inconel 625 are the most resistant to oxidation, followed by Hastelloy X and 310 stainless steel. Incoloy 800HT is the most vulnerable to oxidation.

Table 4: Oxidation resistance of selected superalloys and steels [15]

Material	Average Metal Affected*			
	980 °C Mm	1095 °C μm	1150 °C μm	1205 °C μm
310S SS	28	58	112	262
321 SS	n/a	n/a	n/a	n/a
Inconel 617	33	46	86	318
Inconel 625	18	122	462	>1209
Incoloy 800HT	46	188	226	345
Hastelloy X	23	69	147	>899
Haynes 230	18	33	86	201

*All figures shown with greater than stated value represent extrapolation of tests for which samples were consumed in less than 1000 hours.

5. Discussion

The proposed CSPonD heat extraction system is composed of an intermediate heat exchanger shuttling energy between the CSPonD molten salt bath and a power unit. The intermediate working fluid serves to reduce thermal loads on the heat exchanger induced by cyclic temperature variations between the molten salt inside the bath and the power cycle working fluid.

The proposed heat exchanger layout requires the selection of an intermediate working fluid that could safely interact with the high temperature of the molten salt without risking failure. Similarly, the proposed layout also required that the intermediate working fluid transfer heat with the power cycle working fluid without making the power cycle working fluid too hot. Three families of molten salts were examined, each having their own advantages in consideration as the intermediate working fluid.

One particular fluoride salt composition, LiF-NaF-KF was by far the best working fluid in terms of heat transfer performance, providing the highest rates of heat transfer per unit surface area relative to the other salt candidates. The fluoride salt candidates however are the most expensive salt type, with the LiF-NaF-KF salt composition ranking as the most expensive at an estimated cost of $7.82/kg.

The least expensive salt compositions under consideration were the chloride salts with the KCl-MgCl$_2$ salt composition having an estimate cost of $0.21/kg. The disadvantages associated with the chloride salts are the relatively large amounts of corrosion inducing impurities in the salts and the lack of treatments available to remove these impurities. The chloride salts have the worst heat transfer performance, i.e. the lowest heat transfer rates per unit surface area. However, the low cost of the chloride salts make them a viable candidate for further investigation, provided that a method for removing impurities is developed. The fluoroborate salt were discarded due to the highly reducing nature, toxicity, and corrosive byproducts of the BF$_3$ _

cover gas. The fluoride salts are the best candidates for the CSPonD working fluid due to their proven use as high temperature working fluids and the limits of the other salt families under consideration. The chloride salts however deserve further investigation. If a method to remove the impurities in the chloride salts can be developed then the chloride salts might be a better intermediate working fluid due low costs.

The heat exchanger materials under consideration are limited to high temperature alloys that could withstand the operating temperatures and corrosive properties of molten salt. Alloys Haynes 230, Inconel 617, Inconel 625, Hastelloy X were identified as the best candidate materials for the heat exchanger due to their creep behavior at high temperatures. The most favorable material candidates also have maximum service temperatures within the range of temperatures inside the CSPonD molten salt bath.

Most of the information available for the candidate intermediate working fluids and heat exchanger construction material is not available under the conditions these materials would face as part of the CSPonD heat extraction system. Due to the lack of data, thorough experiments must be performed to evaluate how these materials perform under the proposed operating conditions. The candidate intermediate working fluids should be tested along with the candidate heat exchanger materials to better understand the corrosion behavior of these materials under the proposed operating conditions.

It might be the case after testing the various intermediate working fluid candidates with the various heat exchanger candidate materials that the risks of running a pipe inside the molten salt bath is too high. Future work in the design of the CSPonD heat extraction system should be conducted with the mechanical design and heat transfer behavior being conducted in parallel. The high operating temperatures of the heat exchanger require thorough understanding of the heat exchanger's stress behavior as well as the stress behavior of any element in contact with the molten salt such as the pumps moving the salt whatever fasteners that come in to contact with the salt. Finite element analysis should be performed in high risk areas in addition to developing appropriate maintenance procedures to ensure the safe operation of the heat extraction system.

In addition to testing and mechanical design analysis, a thorough cost analysis of the CSPonD project as a whole should be conducted to serve as a basis of comparison with existing energy production systems. The cost analysis should entail the costs associated with the material and maintenance costs of the system as well as cost-saving and profitable opportunities in the heat extraction system. Without a clear cost advantage, it will be difficult for CSPonD to displace existing energy production systems.

6. Conclusion

The objective of this work was to develop a 15 kW heat exchanger model for the CSPonD molten salt receiver that will shuttle the molten salt's thermal energy for conversion to electric power. A heat extraction system comprising of an intermediate heat exchanger placed inside the molten salt receiver is proposed. Using an intermediate working fluid to shuttle heat energy between the receiver and a power cycle working fluid, thermal loads in the heat exchanger are reduced relative to an alternative heat extraction system without a buffer-like intermediate working fluid. Fluoride, fluoroborate, and chloride salts were three salt families chosen as intermediate working fluid candidates because of their abilities to safely interact with both the "hot" molten salt inside the receiver and the "cold" power cycle working fluid. The fluoride salt composition LiF-NaF-KF showed the best heat transfer performance, providing the greatest heat transfer rate per unit area relative to all the other working fluids. For the LiF-NaF-KF salt composition to attain a thermal power output of 15 kW in two 0.75 in schedule XXS pipes at a flow velocity of 2 m/s, 0.053 m^2 of heat exchanger surface area is required, while the best fluoroborate and chloride salts required surface areas of 0.59 m^2 and 0.60 m^2 respectively. The fluoride salts were also determined the easiest to prepare and use due to nuisance factors in the use of fluoroborate salts and high corrosion causing impurities in chloride salts. Although the easiest to prepare fluoride salts were also the most expensive. The LiF-NaF-KF salt composition was priced at $7.82/kg while the typical KF-KBF$_4$ fluoroborate salt and KCl-NaCl were priced at $4.08/kg and $0.21/kg respectively. A selection of high temperature alloys capable of withstanding the high operating temperatures of molten salts were chosen as candidate materials for the heat exchanger and ranked based on the allowable stress levels near the operating temperature range of the heat exchanger. Alloys Inconel 671, Inconel 625, and Haynes 230 have the highest allowable stresses at 750 °C, while the stainless steel alloys 310S and 321 have the lowest allowable stresses at 750 °C. Corrosion data for the examined molten salts and high temperature materials near the heat exchanger operating temperatures are very limited and fragmented requiring that any design decisions regarding intermediate fluid selection or materials selection be made after thorough testing is conducted.

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7. Appendices

Table 5: FRDPAARC table summarizing critical functional specifications for the proposed CSPonD heat exchanger

Functional Requirements	Design Parameters	Analysis	References	Risks	Countermeasures
-generate average of 15kW thermal power (~4kW electric)	-syphon out salt in heat exchanger directly next to the power cycle working fluid	-first law of thermodynamics -log mean temperature method -E-NTU method	-Fund. of Heat and Mass Transfer (Incropera and DeWitt) -2.006 Course Notes (Brisson, Cravalho, McKinley, Smith)	-temp. fluctuations and resulting thermal stresses in pipes -freezing of salt in pipes due to the cold power cycle working fluid	-control system that only lets salt of certain temperature near the power cycle working fluid -place heaters along pipe or have powerful mobile ones on site
	-syphon salt out w/ an intermediate HEX, whose working fluid interacts with the power cycle working fluid	-first law of thermodynamics -log mean temperature method -E-NTU method	-Fund. of Heat and Mass Transfer (Incropera and DeWitt) -2.006 Course Notes (Brisson, Cravalho, McKinley, Smith)	-extra costs due to extra HEX, working fluid, pump, and storage tank -added complexity in running multiple loops -higher maintenance costs	-low-cost intermediate working fluid -downsize intermediate heat exchanger components
	-run a heat exchanger inside the tub w/ an intermediate working fluid interacting b/t salt and power cycle working fluid	-first law of thermodynamics -log mean temperature method -E-NTU method	-Fund. of Heat and Mass Transfer (Incropera and DeWitt) -2.006 Course Notes (Brisson, Cravalho, McKinley, Smith)	-high temperatures of tub can damage HEX -seal between tube and HEX can leak -diffucult to access the heat exchanger in the bath	-use high temperature materials -extra mechanical sealing elements (i.e. welding)

Table 5 Continued

-high temperature use	-use high temperature materials	-mechanical properties (yield stress, melting point, creep properties) -manufacturability -failure analysis/ safety factors -FEA in critical areas	-ASME Code -Ashby Charts -matweb -manufacturers -FEA software	-high cost -limited availability: manufactured in limited forms (only certain sizes of pipes, rod, bar, etc.)	-minimize HEX material required -use regular, low cost materials wherever possible -design around existing forms of the high temp metals -custom manufacture
-operable for continuous 24 hours	-reduce heat losses of the pond and HEX -	-thermal run-down to equilibrium -thermal losses analysis -thermal resistance and cost-benefit analysis (i.e. cost of insulation vs. savings from smaller heat loss)	--Fund. of Heat and Mass Transfer (Incropera and DeWitt) -2.005 Course Notes (Brisson, Cravalho, McKinley, Smith)	-salt pond might lose too much heat overnight and begin to solidify -consecutive days with limited sun	-constant supervision of temperature -external heating of pipes if freezing occurs in pipes -choose location with small risk of undesired weather (i.e. sunny climate)
-reasonable cost	-minimize HEX area	-model heat transfer vs. HEX area	-Fund. of Heat and Mass Transfer (Incropera and DeWitt) -2.006 Course Notes (Brisson, Cravalho, McKinley, Smith)	-weak HEX structure	-failure analysis/ safety factors
	-reduce working fluid cost	-compare different working fluids' heat transfer performance and costs	-Fund. of Heat and Mass Transfer (Incropera and DeWitt) -2.006 Course Notes (Brisson, Cravalho, McKinley, Smith) -salt cost tables	-large volume of intermediate working fluid necessary for desired work transfer	-proper cost benefit analysis in choosing intermediate working fluid
	-run near maximum power production whenever possible	-upper limit failure analysis of critical HEX components	-FEA (thermal and structural)	-failure -wear	-calc. appropriate safety factor -frequent inspections of critical HEX components

Table 6: FRDPPARC table for use in selecting a molten salt working fluid

Functional Requirements	Design Parameters	Analysis	References	Risks	Countermeasures
Operable at high temperatures	-minimum operating temperature of 750°C	-molten salt spec sheets; lookup salts with operating temperatures >750°C	-D. F. Williams's paper	-impurities or change in chemical structure can change temperature behavior	-regularly inspect salt composition
Low melting point	-minimum melting point of 550°C	-lookup salts with melting temperatures <550°C	-D. F. Williams's paper		
Transfer heat energy	-loop and convect through the material it's enclosed in	-convective heat transfer coefficient	-Fund. of Heat and Mass Transfer (Incropera and DeWitt) -2.006 Course Notes (Brisson, Cravalho, McKinley, Smith)	-thermal stresses in material enclosing salt -corrosion in materials enclosing salt	-choose a strong, high temperature metal alloy with appropriate safety factor
Low cost	-use small amount -use cheap salts -use salts that are cheap to prepare and maintain	-lookup prices and calculate amount of salt needed	-D. F. Williams's paper	-high maintenance costs	-weigh in maintenance costs in cost analysis
Compatible with high temp alloys	-choose salt with favorable corrosion properties	-look up literature for corrosion tables and information -test candidate materials	-D. F. Williams's paper -High temp corrosion and materials applications (G.Y. Lai) -High temp corrosion in molten salts (C. Sequeira)	-corrosion of metal alloy	-regularly inspect material corrosion -thorough testing of material for corrosion behavior

Table 7: FRDPAARC table summarizing critical functional specifications of the heat exchanger construction material

Functional Requirements	Design Parameters	Analysis	References	Risks	Countermeasures
Transmit heat	-overall heat transfer coefficient, UA	-first law of thermodynamics -log mean temperature method -E-NTU method	-Fund. of Heat and Mass Transfer (Incropera and DeWitt) -2.006 Course Notes (Brisson, Cravalho, McKinley, Smith)	-molten salt corrosion	-choose appropriate safety factor (make pipe thick) -select corrosion resistant material
Contain a pressure	-pipe shape to enclose and transport salt	-pressure vessel eqns. -ASME B31.8 pipe thickness eqn.	- Mechanics of Materials (Hibbeler) -ASME Section B	-yielding -fracture	-choose appropriate safety factor (large pipe thickness) -frequent inspection
Reasonable cost	-minimize construction material	-$/kg	-quotes from high temperature metal manufacturers	-lots of material required -expensive material required	-metal coatings -use different metal alloys in the HEX depending on its operating environment
High operating Temperature	- service temp >/= 750C	-melting point of metal -stress and creep behavior	-AMSE Section II-Materials, part D	-creep -thermal-mechanical fatigue -corrosion	-choose appropriate safety factor (large pipe thickness) -frequent inspection

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Table 8: Allowable stress @ 900 °C

Metal	Alloy designation/UNS No	Maximum Service Temperature (Celsius)	Allowable Stress at Maximum Service Temperature (MPa)
310S SS	S31008	816	1.15
321 SS	S32100	816	1.62
Inconel 617	N06617	982	12.3
Inconel 625	N06625	871	12.5
Incoloy 800HT	N08811	899	6.2
Hastelloy X	N06002	899	8.14
Haynes 230	N06230	982	10.2

Note: Data past 900 °C is not in the ASME Boiler & Pressure Vessel Code

Table 9: Thermal conductivity and Coefficient of thermal expansion of the candidate materials

Material	Thermal Conductivity (W/C-m)	Coeff. Of Therm. Exp. (10^{-6}/C°)
310S SS	18.7 @500 °C	17.1@20-500 °C
321 SS	21.4 @500 °C	18.9 @20-600 °C
Inconel 617	25.5 @800 °C	15.4 @25-800 °C
Inconel 625	20.8 @760 °C	15.7 @20-760 °C
Incoloy 800HT	24.7 @800 °C	18.0 @21-800 °C
Hastelloy X	25.1 @816 °C	15.8 @26-732 °C
Haynes 230	24.4 @800 °C	15.2 @26-800 °C