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Abstract – *Next generation nuclear reactors such as the advanced high temperature reactor (AHTR) are designed to increase energy efficiency in the production of electricity and provide high temperature heat for industrial processes. The efficient transfer of energy for industrial applications depends on the ability to incorporate effective heat exchangers between the nuclear heat transport system and the industrial process heat transport system. This study considers two different types of heat exchangers—helical coiled heat exchanger and printed circuit heat exchanger—as possible options for the AHTR secondary heat exchangers with distributed load analysis and comparison. Comparison is provided for all different cases along with challenges and recommendations.*

I. INTRODUCTION

Next generation nuclear reactors (NGNR) such as the advanced high temperature reactor (AHTR) are intended to increase energy efficiency in the production of electricity and/or provide high temperature heat for industrial processes. The primary loop reference salt for an AHTR is Li_2BeF_4, referred to as "flibe." Heat in an AHTR is transferred from the reactor core by the primary liquid-salt coolant to an intermediate heat-transfer loop through intermediate heat exchangers (IHXs). The intermediate heat-transfer loop uses an intermediate liquid-salt coolant through a secondary heat exchanger (SHX) to move the heat to a power conversion system (Rankine cycle). These heat exchangers (HXs) are considered key components because they are operated under a severe environment and their performance is directly related to the overall system efficiency and safety.

The AHTR appears to have excellent safety attributes. The combined thermal capacity of the graphite core and the molten salt coolant pool offer a large time buffer to reactor transients. Compared to light water reactors, AHTRs should be more economical[1] because of higher power conversion efficiency, low pressure containment, and the absence of active safety systems. The efficient transfer of energy for power production depends on the ability to incorporate effective heat exchangers between the nuclear heat transport system and the power production process. The selection of a HX is most strongly influenced by the application and operational conditions requirement; with its individual requirement for cost, size, reliability, robustness, maintenance, expected life, etc.

The performance of any HX is described by the overall heat transfer coefficient, and pressure drop. Operating the HX in the turbulent regime ensures higher flow rates, which does enhance the overall heat transfer coefficient but leads to a higher pressure drop. Ideally, the heat transfer coefficient should be as high as possible with lowest possible pressure drop. Having a lower pressure drop reduces the pumping cost while maintaining the required pressures downstream of the HX.

This study carries out the SHX thermal design for both supercritical and subcritical Rankine cycles. The thermal design of HXs determines channel size, required length, and number of layers or tubes to meet a given thermal duty. The HX of an AHTR is subjected to a unique set of conditions that impose several design challenges not encountered in standard HXs. Corrosive molten salts, for example, especially at temperatures in excess of 700°C, require specialized materials throughout the system to avoid corrosion and ensure a long life expectancy.

This study considers helical coil heat exchangers (HCHE) and printed circuit heat exchangers PCHEs as possible options for the AHTR secondary HXs with the three different options discussed later. A preliminary cost comparison will be provided for all different cases along with challenges and recommendations.

This study took the following three options into consideration as HX arrangement:

- *Option 1*: A single HX transfers all of the heat (3,400 MW(t)) from the intermediate heat transfer loop to the power conversion system or process plants.
- *Option 2*: Two HXs share heat to transfer total heat of 3,400 MW(t) from the intermediate heat transfer loop to the power conversion system or process plants. Each exchanger transfers 1,700 MW(t) with a parallel configuration.
- *Option 3:* Three HXs share heat to transfer total heat of 3,400 MW(t) from the intermediate heat transfer loop to the power conversion system or process plants.

Each exchanger transfers 1,130 MW(t) with a parallel configuration.

The AHTR SHX design requirements and operating conditions are given in Table I.

Table I**.** SHX design requirements and basic conditions for the AHTR.

Parameter	Requirements
Reference System Configuration	AHTR + Supercritical Steam Rankine PCS:
HX Type	- Helical Coiled - Printed Circuit HX
HX Duty (MW)	3400/1700/1130
Primary Coolant	KF-ZrF4
Secondary Coolant	Water/Steam
Primary Temperature (Tin/Tout)	679/587°C (supercritical Rankine cycle) 679/586.1°C (subcritical Rankine cycle)
Secondary Temperature (Tin/Tout)	251/593°C (supercritical Rankine cycle) 241.7/550°C (supercritical Rankine cycle)
Primary Pressure (MPa)	0.103
Secondary Pressure (MPa)	24 (supercritical Rankine cycle) 17 (subcritical Rankine cycle)
Tube Material	Hastelloy N
Shell Material	Hastelloy N

II. PRELIMINARY DESIGN ANALYSIS

This study generally describes HCHE and PCHE followed by thermal design analyses of each. The function of the process SHX is to transfer heat from the secondary salt to the supercritical/subcritical Rankine cycle. Very high efficiency heat conversion is desired for the HX because any heat that is not transferred to the process side reduces overall plant efficiency. PCHEs can provide significant reductions in volume and material usage. With tightly packed channels making adjacent streams able to effectively translate temperatures across their boundaries, approach temperatures (the difference between the outlet temperature of one fluid stream and the inlet temperature of the opposing fluid stream at their common header location) closer to 1°C are possible with compact HXs. Shell-and-tube HXs are often closer to 12°C in approach temperature, another demonstration of their inferior performance.[2]

For the same inlet and exit temperature values of two fluids, (Log Mean Temperature Difference, LMTD), attains a larger magnitude for countercurrent flow than the parallel flow[3]. Thus, the counter-flow configuration is more effective in transferring heat than the parallel-flow arrangement and has been assumed for further thermal design in this study.

The analysis carried out here is appropriate for the preliminary design analysis (for scoping and comparison purposes). In order to make the design of a HX more manageable, the following assumptions were made for preliminary designs:

- The fluid properties are constant throughout the exchanger and are evaluated at the respective mean temperature.
- Heat transfer coefficients of both sides are constant along the exchanger.
- Flows in HXs experience abrupt contractions at the entry and expansions at the exit, from the headers or ducts. In this analysis entry and exit losses are neglected.
- Flow is assumed to be steady-state to avoid the complications of making transient calculations.

Two cases are individually analyzed:

- Supercritical Rankine Cycle (24 MPa)
- Subcritical Rankine Cycle (17 MPa).

A specified heat load , is given by the heat transfer and rate equation.

. (1)

The equation written in terms of hot and cold fluids yields:

and (2)

(3)

The product of is called the heat capacity rate and is denoted by C; the highest and lowest values of C between hot and cold streams being C_{max} and C_{min}. If the ratio of C_{min} to C_{max} is unity, the exchanger is said to be balanced as explained[4].

The theoretically maximum possible heat exchanged between the hot and cold streams can be expressed as:

, (4)

which represents the idealized performance with infinite surface area, as explained[4]. The point of equal temperatures, called the pinch point, ideally corresponds to zero temperature difference, but in general it refers to the minimum temperature difference in the HX.

After defining the maximum possible heat transfer, HX effectiveness can be defined as:

(5)

where C_c and C_h are the cold and hot stream heat capacity rates.

The overall heat transfer coefficient is now determined followed by the calculation of LMTD. These values and the Q calculated above will finally be used to determine the surface area through which heat transfer takes place.

HX performance is normally evaluated by the overall heat transfer coefficient U that is defined by the equation:

(6)

where, is the LMTD between the streams and F is a correction factor that depends on the flow configuration. UA_s is the product of the overall heat transfer coefficient and reference area, also known as heat transfer conductance. is calculated with the equation:

(7)

where ; .

Obtaining the respective values for , overall heat transfer coefficient (U), and LMTD (, the size and number of plates/tubes were determined by solving for A_s. Dividing it further with inner cylinder's circumference could yield the required tube length needed for the HX for the given heat duty. Using this approach as a tool, the next section further discusses the preliminary design for both HX types.

III. HEAT EXCHANGERS

This section describes the general features of HCHEs and PCHEs.

III.A. HCHEs

HCHEs are shell-and-tube type HXs that consist of tubes spirally wound into bundles and fitted into a shell.

Because of the tube bundle geometry, a considerable amount of surface can be accommodated inside the shell. These HXs are used for gas-liquid heat transfer applications, primarily when the operating temperature and/or pressure are very high. Cleaning an HCHE is very challenging.[5]

Primary and the secondary channels in PCHEs are geometrically identical so the coolant type in each channel does not affect HX design. However, the design of HCHEs is significantly affected by selection of the coolant inside and outside of the tubes. Two options were evaluated for the HCHE design:

- Tube (Molten Salt) and Shell (Water/Steam)
- Tube (Water/Steam) and Shell (Molten Salt).

III.A.1 Mechanical Design

HCHE: Determination of Tube and Shell Thickness. This study assumes that loading is caused by pressure (internal and external) and is loaded axisymmetrically.

General Case: In the case of a thick wall cylinder subjected to uniform internal and external pressure the radial (σ_r) and tangential (σ_θ) stresses are defined as[6]:

(8)

(9)

where:

R_i = Inside radius
P_i = Internal pressure
R_o = Outside radius
P_o = External pressure
r = radius

The maximum numerical value of radial stress from Equation (8) is at r = R_i (if P_i is greater than P_o). If P_o is greater than P_i, the maximum radial pressure occurs at $r=R_o$ and it is equal to the external pressure (P_o).

Internal Pressure Dominant: In a special case where the cylinder is exposed to dominant internal pressure, from Equations (8) and (9), the radial (σ_r) and tangential (σ_θ) stresses are defined as:

(10)

(11)

It is obvious from Equations (10) and (11) that the magnitude of the tangential stress is greater than the radial stress. The tangential stress ($\sigma_{\theta\text{-}max}$) is always positive (tensile) for all the r values and it is maximum at r equal to inner radius by expressed as:

(12)

External Pressure Dominant. In the case of presence of dominant external pressure from Equations (10) and (11), radial (σ_r) and tangential (σ_θ) stresses are defined as:

(13)

(14)

The maximum radial stress occurs at $r=R_o$ (always compressive/negative). As it can be seen from Equations (13) and (14), the numerical value of the tangential stress is greater than the radial stress. The maximum tangential stress (compressive/negative, $\sigma_{\theta\text{-}max}$) occurs at $r=R_i$ and is expressed as:

(15)

Tables II and III show preliminary HCHE and PCHE Mechanical design results for both supercritical and subcritical Rankine cycles. In this stress analysis, tube and shell inner radius were assumed to be 0.0124 m and 1.65 m, based on the IHX design for the High Temperature Test Reactor operated by the Japan Atomic Energy Agency. While this HX transfers heat from helium to water, it was judged representative enough to compare Options 1 and 2. In this analysis, safety factors were assumed to be 1.0.

Table II(A) shows the stress analysis and mechanical design calculations for Option 1 in the supercritical Rankine cycle. In this option, molten salt is inside the tubes and water/steam is on the shell side. According to the calculation, required minimum thickness for the tube is only 0.0016 m. However, the minimum thickness for the shell is 0.341 m, which is almost 15% of the vessel diameter. For larger safety factors and shell diameters, the required shell thickness will further increase, resulting in an unrealistic design. For this reason, Option 1 is not recommended Table II(B) shows the stress analysis and mechanical design calculations for Option 2 where the molten salt is on the shell side and water/steam is inside the tubes. Therefore, the dominant load for both tube and shell is an internal load. However, the shell's inside pressure is from molten salt (200,000 Pa) and only marginally higher than the pressure outside (101,000 Pa). In Option 2 then, the required minimum thickness for the tube and shell are estimated to be only 0.0014 m and 0.0015 m, respectively (for a safety margin of 1.0). Therefore, Option 2 is strongly recommended for the final design.

Tables III(A) and III(B) show similar results for the Subcritical Rankine Cycle. The recommended option is also Option 2, with molten salt on the shell side and water/steam inside the tubes. The required minimum thickness for the tube and shell are estimated to be only 0.0012 m and 0.0015 m respectively.

The mechanical design method describes how this study determined tube, shell, and plate geometrical dimension in order to satisfy operating and temperature requirements. Thermal design method describes how this study determined overall HX geometries and sizes.

Table II. Mechanical design of the tube and shell thicknesses (Supercritical Rankine).

(A) Option 1

Tube (Molten Salt)			Shell (Water/Steam)		
Parameter	unit	value	Parameter	unit	value
Tube inner radius	m	0.015	Shell inner radius	m	1.65
Tube inside pressure	kPa	100	Shell inside pressure	kPa	24000
Tube outside pressure	kPa	24000	Shell outside pressure	kPa	100
Yield stress (Hastelloy N)	kPa	217874	Yield stress (Incoloy 800H)	kPa	129627
Maximum allowable stress	kPa	217874	Maximum allowable stress	kPa	129627
Tube minimum wall thickness	m	0.0016	Shell minimum wall thickness	m	0.341

(B) Option 2

Tube (Water/Steam)			Shell (Molten Salt)		
Parameter	unit	value	Parameter	unit	value
Tube inner radius	m	0.0124	Shell inner radius	m	1.65
Tube inside pressure	kPa	24000	Shell inside pressure	kPa	200
Tube outside pressure	kPa	100	Shell outside pressure	kPa	100
Yield stress (Hastelloy N)	kPa	217874	Yield stress (Hastelloy N)	kPa	217874
Maximum allowable stress	kPa	217874	Maximum allowable stress	kPa	217874
Tube wall minimum thickness	m	0.0014	Shell minimum wall thickness	m	0.0015

Table III. Mechanical design of the tube and shell thicknesses (Subcritical Rankine).

(A) Option 1

Tube (Molten Salt)			Shell (Water/Steam)		
Parameter	unit	value	Parameter	unit	value
Tube inner radius	m	0.015	Shell inner radius	m	1.65
Tube inside pressure	kPa	100	Shell inside pressure	kPa	17300
Tube outside pressure	kPa	17300	Shell outside pressure	kPa	100
Yield stress (Hastelloy N)	kPa	217874	Yield stress (Incoloy 800H)	kPa	129627
Maximum allowable stress	kPa	217874	Maximum allowable stress	kPa	129627
Tube minimum wall thickness	m	0.0013	Shell minimum wall thickness	m	0.247

(B) Option 2

Tube (Water/Steam)			Shell (Molten Salt)		
Parameter	unit	value	Parameter	unit	value
Tube inner radius	m	0.0124	Shell inner radius	m	1.65
Tube inside pressure	kPa	17300	Shell inside pressure	kPa	200
Tube outside pressure	kPa	100	Shell outside pressure	kPa	100
Yield stress (Hastelloy N)	kPa	217874	Yield stress (Hastelloy N)	kPa	217874
Maximum allowable stress	kPa	217874	Maximum allowable stress	kPa	217874
Tube wall minimum thickness	m	0.0012	Shell minimum wall thickness	m	0.0015

Preliminary Thermal Design Analysis. A general methodology for determining the heat transfer surface area was presented previously. The following equations were used to expand on the calculation for A_s to determine tube dimensions and other details for the HCHE:

- Radius of tube:

  a = (16)

  where:

- Radius of curvature:

  (17)

- Number of tubes in the bundle:

  (18)

  where:

- Nusselt number for straight pipe[7]

) (19)

- Nusselt number for Helical Coil[8]

  (20)

Therefore, the heat transfer coefficient (h_t) inside the tube can be estimated as follows:

(21)

where, K_{tube} = thermal conductivity of the tube material.

In the tube side, heat transfer correlations are based on the heat transfer in inline tube bundles in smooth pipes.

To estimate heat transfer in the inline tube bundles, some parameters, e.g., the correction factor for the tube rows and Prandtl number for the tube wall, are defined as:

- Correction factor (C): 1 (for large number of tubes)
- Prandtl number for the tube wall: $Pr_s = Pr_w$ (Prandtl number for the tube wall was assumed to be the same as the bulk phase)
- Nusselt number for the shell side can be estimated as follows[9]:

  (22)

Therefore, the heat transfer coefficient on the shell side can be estimated as follows:

(23)

where K_{shell} – thermal conductivity of the shell material.

Since the tube thickness is small, the effect of heat transfer resistance at the wall has been neglected. The overall heat transfer coefficient is calculated as

. (24)

The heat transfer surface area is calculated as

. (25)

The tube length is calculated as

. (26)

Number of rotations of the tube bundle is calculated as

. (27)

Therefore, shell length can be calculated as:

. (28)

After establishing tube dimensions and parameters in this manner, pressure drop inside the tube can be estimated by the following equation[10]:

- Tube side:

  Friction factor inside the helical coiled tubes:

  (29)

  Pressure drop inside the helical tube:

  (30)

  where:

- Shell side:

  (31)

  Hagen number for inline tube bundles[11]:

  (32)

  (33)

  (34)

  Number of effective tube bundles:

  (35)

Pressure drop on the shell side is calculated as

—— —* Hg (36)

Table IV presents preliminary thermal designs for the helical coiled HX for the supercritical Rankine cycle. In this design, three different options were considered based on the number of HXs. Table V shows the similar thermal design for the subcritical Rankine cycle.

Table IV. Preliminary design specifications for AHTR secondary HX (helical coiled) – Supercritical Rankine cycle.

Spec.	Unit	1 SHX	2 SHXs	3 SHXs
Heat Duty	MW(t)	3400.00	1700.00	1130
Number of Units	-	1	2	3
Primary coolant (Shell)	-	Molten Salt	Molten Salt	Molten Salt
Secondary coolant (Tube)	-	Water/Steam	Water/Steam	Water/Steam
Primary inlet T	◦C	679.00	679.00	679.00
Primary outlet T	◦C	587.00	587.00	587.00
Secondary inlet T	◦C	251.00	251.00	251.00
Secondary outlet T	◦C	593.00	593.00	593.00
Primary pressure (Shell)	MPa	0.10	0.10	0.10
Secondary pressure (Tube)	MPa	24.00	24.00	24.00
Tube diameter	m	0.03	0.03	0.03
Tube pitch	m	0.05	0.05	0.05
Tube thickness	mm	1.4	1.4	1.4
Number of tubes	-	740.00	600.00	480.00
Number of tube rotations	-	4.12	4.16	4.39
Tube material	-	Hastelloy N	Hastelloy N	Hastelloy N
Shell inside diameter	m	3.30	2.00	1.60
Shell outside diameter	m	7.00	5.00	4.00
Shell material	-	Hastelloy N	Hastelloy N	Hastelloy N
Overall heat transfer coefficient	W/m K	3804.16	3422.89	3366.13
Heat transfer surface area	m2	4871.922	2707.29	1829.90
Pressure drop in shell	kPa	15.61	13.10	14.87
Pressure drop in tube	kPa	36.30	10.83	6.7

Table V. Preliminary design specifications for AHTR SHX (helical coiled) – Subcritical Rankine cycle.

Spec.	Unit	1 SHX	2 SHXs	3 SHXs
Heat Duty	MW(t)	3400.00	1700.00	1130
Number of Units	-	1	2	3
Primary coolant (Shell)	-	Molten Salt	Molten Salt	Molten Salt
Secondary coolant (Tube)	-	Water/Steam	Water/Steam	Water/Steam
Primary inlet T	◦C	679.00	679.00	679.00
Primary outlet T	◦C	586.10	586.10	586.10
Secondary inlet T	◦C	241.70	241.70	241.70
Secondary outlet T	◦C	550.00	550.00	550.00
Primary pressure (Shell)	MPa	0.10	0.10	0.10
Secondary pressure (Tube)	MPa	17.30	17.30	17.30
Tube diameter	m	0.03	0.03	0.03
Tube pitch	m	0.05	0.05	0.05
Tube thickness	mm	1.24	1.24	1.24
Number of tubes	-	740.00	600.00	480.00
Number of tube rotations	-	3.33	3.35	3.53
Tube material	-	Hastelloy N	Hastelloy N	Hastelloy N
Shell inside diameter	m	3.30	2.00	1.60
Shell outside diameter	m	7.00	5.00	4.00
Shell material	-	Hastelloy N	Hastelloy N	Hastelloy N
Overall heat transfer coefficient	W/m K	3959.51	3571.62	3522.38
Heat transfer surface area	m^2	3914.71	2169.92	1462.52
Pressure drop in shell	kPa	11.66	9.76	11.05
Pressure drop in tube	kPa	52.57	15.64	9.66

III.A.2 PCHEs

The PCHE is a relatively new concept that has been commercially manufactured by Heatric™ since 1985. PCHEs are robust, combining compactness, low pressure drop, high effectiveness, and the ability to operate with a very large pressure differential between hot and cold sides.[12] These HXs are especially well suited to applications where compactness is important. As the name implies, PCHEs are manufactured by the same technique used to produce standard printed circuit boards for electronic equipment. Individual plates are etched to produce channels. These etched plates are thereafter joined by diffusion welding, resulting in extremely strong all-metal HX cores. The diffusion welding process includes a thermal soaking period to allow grain growth across the joint and between the plates, which creates a joint with nearly the same strength as the base material. Hence, the joint has very high pressure containment capability and avoids the creation of corrosion cells observed in traditional welding, as explained[4]. The fluid passages are semicircular in cross-section, typically being 1.0 to 2.0 mm wide and 0.5 to 1.0 mm deep with a hydraulic diameter of 1.5 to 3 mm. After bonding, any number of core blocks can be welded together to provide the required flow capacity.

The thermal design subjects PCHEs to very few constraints. Fluids may be liquid or gas, or two-phase, multistream and multipass configurations can be assembled and flow arrangements can be truly counter-current, co-current, or cross-flow, or a combination of these, at any required pressure drop.

Where required, high heat exchange effectiveness (over 98%) can be achieved through very close temperature approaches in counter-flow. To simplify control or further maximize energy efficiency, more than two fluids can exchange heat in a single core. Heat loads can vary from a few watts to many megawatts, and these exchanger's can weigh a few to thousands of kilograms.

Flow induced vibration, an important source of failure in shell-and-tube exchangers, is largely absent from PCHE.

III.A.2.a. Thermal Design Guideline and Constraints

This sectiont summarizes the guidelines and criteria for designing the PCHE for IHX[13,14]:

- No gasket or brazing (risk of leak is considerably reduced); two orders of magnitude lower
- Very low vibration damage
- No fouling under clean gas conditions
- Surface area density: about 2,500 m^2/m^3
- No heat transfer and friction factor correlations are available
- Semicircular cross-section
- Width: 1.0 to 2.0 mm (2.0 mm shows maximum thermal performance and economic efficiency but for nuclear application, 1.2 mm is suggested.)
- Depth: 0.5 to 1.0 mm
- Weight based costing: \$30/kg for stainless steel, \$120/kg for titanium, expected to be less than \$40/kg for nuclear application (not known for Hastelloy N)
- Carbon steel is typically not used because of the small channel diameter vulnerable to corrosion and unsuitability for diffusion bonding.
- Average mass-to-duty ratio: 0.2 tones/MW (13.5 tones/MW in shell-and-tube design)
- No constraint to the pressure drop
- Multiport PCHE module size: width: 0.5 m (1.5 m is max), height: 0.6 m, depth: 0.4 to ~0.6 m.
- Fatigue can be caused by thermal transient.
- Only pressure drop restrict the velocity.
- Estimated minimum life ~20 years.

PCHE: Determination of Channel Pitch and Plate Thickness

Diffusion welding strength of compact PCHE is believed to be that of the parent metal, because of the crystal grain growth across the interface during bonding. A simple stress analysis was preformed for the IHX, assuming that it is a compact HX of the type designed by Heatric[15]. The design of the HX channel is defined by the channel diameter (d) and plate thickness (tp) as illustrated in Figure 1. Each plate contains either hot or cold fluid, but not both. Adjacent plates contain the other fluid. Following the method used previously[16], the minimum wall thickness between channels (t_f) can be approximated as

$$t_f \geq \frac{p}{\frac{\sigma_D}{\Delta P}+1} \tag{37}$$

where σ_D is the allowable stress and ΔP is the differential pressure between the hot and cold streams. Expressing in terms of pitch-to-diameter ratio (relation between pitch and thickness is shown by equation 39), yields

$$\frac{p}{d} \geq 1+\frac{\Delta P}{\sigma_D}\,. \tag{38}$$

The required plate thickness can also be calculated based on this method[16]. The plate is assumed to be a thick-walled cylinder, with an inner radius (Ri) of d/2 and an outer radius (Ro) of tp. Equations (37) and (38) can be used to calculate the thickness-to-diameter and pitch-to-diameter ratio for the IHX as a function of allowable stress and various pressure of the hot and cold streams.

Preliminary Thermal Design Analysis. In this study, two adjacent hot channels are assumed for each cold channel in the PCHE. Figure 1 shows the channel arrangement of a PCHE. Using the general methodology for the thermal deign presented in Section II, the equations that follow were used for the PCHE design.

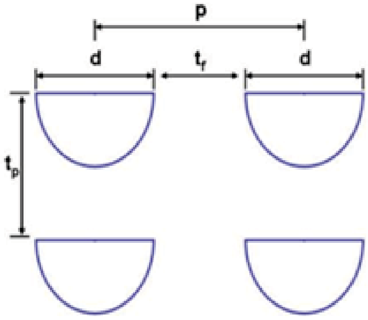

Figure 1. Channel arrangement of PCHE.[16,17]

HX geometrical parameters[4,17]

- Channel horizontal distance:

(39)

- Ratio of free flow area to frontal area:

(40)

- Effective diameter:

(41)

Heat Transfer Coefficient. This section estimates heat transfer for the compact HX. In this thermal design study the channel is assumed to be straight throughout the flow paths[18].

(42)

where are the hot side and cold side Nusselt numbers. The Reynolds numbers, , are based on the effective diameters of the and the respective hot side and cold side Reynolds numbers.

From the above equation, the convective heat transfer coefficient can be calculated as:

; , and (43)

the overall heat transfer coefficient can be calculated as:

(44)

where R_w is the resistance in heat transfer wall .

Heat transfer surface area:

(45)

Number of channels on the hot and cold side, respectively:

(46)

(47)

HX Channel length for hot and cold side:

(48)

(49)

After establishing channel dimensions and parameters in this manner, the pressure drop inside the channel can be estimated by the following equation[19]:

(50)

Therefore, pressure drop in the hot channel can be estimated as:

(51)

Similarly for the cold side

(52)

Therefore, pressure drop in the hot channel can be estimated as:

(53)

where:

f_h friction factor for the hot side
f_c friction factor for the cold side
V_h and V_c is the coolant velocity in the hot and cold channel respectively.

Tables VI and VII summarize the preliminary design specifications for AHTR SHXs (PCHE). Three different options for the number of HXs and arrangement were also considered in this preliminary design.

Existing Challenges for Fluoride Salt Process HX. Using molten salt as coolant requires lots of challenges for designing and operating HXs[20,21]. The following summarizes some of existing challenges identified:

- High melting point temperature:
  - Prevent or accommodate freezing in HXs and piping

- Materials and corrosion control strategies needed:
  - Fluoride salts remove oxide layers from metals, which in turn enhances corrosion
  - Reactions with metal ions occur (HF + metal → metal fluoride + H_2)
  - Minimum Cr but sufficient to provide oxidation resistance
  - Limited Al, Ti, and carbon contents minimize fabrication and corrosion problems
  - Limited Boron content prevents weld cracking
- Development needs in fabrication and joining technology is a must (thin sections, grain size effects, etc.)
- High uncertainty in fluoride salts and Hastelloy N data at high temperatures
- Impurities in salts must be controlled
- Oxidation potential (redox control) of salts can influence material performance (embrittlement of material)
- Mass transfer:
  - Oxidation at high temperature and deposition at lower temperature
  - Differences in chemical activities of different metals (dissimilar materials)
- Salt to gas transfer issues:
  - Pressure differentials between the primary and secondary systems are large (the pressure difference is about 17 and 24 MPa for subcritical and supercritical Rankine cycle, respectively)
  - Increase in gas velocities or large surface areas to compensate for low thermal conductivity of gas
  - Enhance material performance, thin walls are problematic for high pressure differences and corrosion
  - High temperature alloys and ceramics with required mechanical and fluoride salt corrosion resistance properties

Table VI. Preliminary design specifications for AHTR Secondary HX (PCHE) – Supercritical Rankine Cycle.

Spec.	Unit	1 SHX	2 SHXs	3 SHXs
Heat Duty	MW(t)	3400	1700	1130
Number of Units	—	1	2	3
Primary coolant (primary)	—	Molten Salt (KFZr4)	Molten Salt (KFZr4)	Molten Salt (KFZr4)
Secondary coolant (secondary)	—	Water/Steam	Water/Steam	Water/Steam
Primary inlet T	◦C	679	679	679
Primary outlet T	◦C	587	587	587
Secondary inlet T	◦C	251	251	251
Secondary outlet T	◦C	593	593	593
Primary pressure	Pa	1.00E+05	1.00E+05	1.00E+05
Secondary pressure	Pa	2.40E+07	2.40E+07	2.40E+07
Channel diameter	m	0.003	0.003	0.003
Channel pitch	m	3.33E-03	3.33E-03	3.33E-03
Plate thickness	m	3.17E-03	3.17E-03	3.17E-03
Number of total primary channels	—	3678247	3678247	3678247
Number of total secondary channels	—	1839123	1839123	1839123
Module width	m	0.6	0.6	0.6
Module height	m	0.6	0.6	0.6
Module length	m	0.85	0.85	0.86
Overall Heat Transfer Coefficient	W/m K	759	686	631
Primary pressure drop	kPa	38	10.5	52.86
Secondary pressure drop	kPa	1.7	0.3	0.1

Table VII. Preliminary design specifications for AHTR Secondary HX (PCHE) – Subcritical Rankine Cycle.

Spec.	Unit	1 SHX	2 SHXs	3 SHXs
Heat Duty	MW(t)	3400	1700	1130
Number of Units	—	1	2	3
Primary coolant (primary)	—	Molten Salt (KFZr4)	Molten Salt (KFZr4)	Molten Salt (KFZr4)
Secondary coolant (secondary)	—	Water/Steam	Water/Steam	Water/Steam
Primary inlet T	◦C	679	679	679
Primary outlet T	◦C	586.1	586.1	586.1
Secondary inlet T	◦C	241.7	241.7	241.7
Secondary outlet T	◦C	550	550	550
Primary pressure	Pa	1.00E+05	1.00E+05	1.00E+05
Secondary pressure	Pa	1.73E+07	1.73E+07	1.73E+07
Channel diameter	m	0.003	0.003	0.003
Channel pitch	m	3.24E-03	3.24E-03	3.24E-03
Plate thickness	m	3.17E-03	3.17E-03	3.17E-03
Number of total primary channels	—	3678247	3678247	3678247
Number of total secondary channels	—	1839123	1839123	1839123
Module width	m	0.6	0.6	0.6
Module height	m	0.6	0.6	0.6
Module length	m	0.7	0.7	0.7
Overall U	W/m K	764	692	638
Primary pressure drop	kPa	37.8	10.4	5
Secondary pressure drop	kPa	3.2	0.5	0.2

VI. SUMMARY AND CONCLUSIONS

Preliminary HX design has been performed in this study. The AHTR secondary HX transfers heat from the molten salt intermediate loop to the power conversion system based on steam Rankine cycles. Three different options were investigated for HCHEs and PCHEs such that the thermal duty varied from 3400 MW(t), to 1700 MW(t) and further to 1130 MW(t) as the duty load was split into three HXs.

A simple stress analysis was used for determining tube and shell thicknesses of the HCHEs, and for determining channel pitch and plate thickness of the PCHEs. As a result, it was concluded that for HCHEs, the coolant inside the tube is water/steam and the shell side coolant is molten salt because of much higher pressure drop.

A simple thermal design method was used for determining overall design specifications including geometry, sizing, and configurations. Tables IV-VII summarizes the design specifications for helical coiled heat exchangers and PCHEs, respectively. Mainly, for the same heat duty helical coil heat exchanger showed 5 times higher overall heat transfer coefficient, though also had a much higher pressure drop when compared with the printed circuit heat exchanger. Further experiments should be carried out for validation and for overall uncertainty quantification.

In this paper, the cost of steam generators considered for a super-critical Rankine power cycle was estimated. The steam generators considered are shell and tube HXs and printed circuit HXs, PCHE. For this analysis, the HXs must have a total duty of 3400 MW(t). This duty is supplied in one of three ways: a single HX; the duty is split between two HXs; or the duty is divided between three HXs. The focus of the study was on the Section VIII requirements of the SHX designed to the requirements of ASME Section VIII, which will transfer this heat to downstream applications such as hydrogen production, process heat, and electricity generation.

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