Design Study and Cost Assessment of Straight, Zigzag, S-shape, and OSF PCHEs for a FLiNaK-SCO₂ Secondary Heat Exchanger in FHRs

In Hun Kim*, Xiaoqin Zhang, Richard Christensen, and Xiaodong Sun

Nuclear Engineering Program, Ohio State University, Columbus, OH 43210, USA

Phone: +1-614-247-4684, Fax: +1-614-247-3163

Email: ihkim0730@gmail.com, zhang.3836@osu.edu, christensen.3@osu.edu,
sun.200@osu.edu

Abstract

This study focused on designing a cost-effective heat exchanger for a 20-MW FLiNaK-SCO₂ Secondary Heat Exchanger in Fluoride salt-cooled High-temperature Reactors. Specific Printed Circuit Heat Exchanger geometries were investigated for those cases where Fanning factor and Nusselt number correlations were available. Straight, S-Shape, Offset Strip Fin, and zigzag 52° channels were considered for the SCO₂ side while straight, zigzag 15°, rectangle OSF 7.565 mm, and rectangle OSF 2.40 mm channels were considered for the FLiNaK side. Thermal-hydraulics, mechanical aspects, and corrosion rate were taken into account. A cost analysis was performed to combine the effects of heat transfer performance and pressure drop in heat exchangers. Single banking and double banking were also considered. Finally, the best PCHE channel configurations on SCO₂ and FLiNaK sides were proposed.

Keywords

PCHE; Optimization; FHR

Abbreviations

FHR: Fluoride salt-cooled High-temperature Reactor

HTGR: High Temperature Gas-cooled Reactor

MSR: Molten Salt Reactor

SFR: Sodium Fast Reactor

DRACS: Direct Reactor Auxiliary Cooling System

PCHE: Printed Circuit Heat Exchanger

SCO₂: Supercritical Carbon Dioxide

IHX: Intermediate Heat Exchanger

OSF: Offset-Strip Fin

SHX: Secondary Heat Exchanger

1. Introduction

The Fluoride salt-cooled High-temperature Reactor (FHR) adopts advantages of High Temperature Gas-cooled Reactors (HTGRs), Sodium Fast Reactors (SFRs), and Molten Salt Reactors (MSRs). Accordingly, graphite-matrix coated-fuels, a Direct Reactor Auxiliary Cooling System (DRACS), and liquid salt working fluids are applied to the FHR, where a three-loop system is currently considered. The primary coolant is FLiBe, which is a mixture of LiF and BeF₂, with a melting point of 459°C and a boiling point of 1433°C. LiF-NaF-KF (FLiNaK), KF-ZrF₄, and KCl-MgCl₂ are considered as potential secondary coolants. A helium Brayton cycle, a supercritical CO₂ Brayton cycle, a supercritical Rankine steam cycle, and a subcritical Rankine steam cycle are considered as potential candidates for the power conversion cycle.

In the FHR, the heat exchangers are some of the most important components. It is strongly pertinent to overall plant thermal efficiency. In addition, heat exchanger failure due to corrosive salts results in a significant economic loss due to purchasing a new heat exchanger, replacing the failed one with a new one, and generating no profit during the shut-down period. Therefore, more reliable and compact heat exchangers are required.

A Printed Circuit Heat Exchanger (PCHE) is currently considered as one of the most promising heat exchangers for the FHR. The PCHE is manufactured by chemical etching, stacking, and diffusion bonding processes. Compared to welding, diffusion bonding achieves higher structural integrity and corrosion resistance. Micro-wavy channels made by the chemical etching increase heat transfer performance as well as pressure drop.

Previous research evaluated thermal-hydraulic performance for specific PCHE

channel geometries and suggested an improved PCHE design than a zigzag channel. Ngo et al. (2007) designed and constructed zigzag 52° and S-shape PCHEs for CO_2 and Supercritical CO_2 (SCO_2) conditions. Their subsequent analysis resulted thermal-hydraulic correlations for both zigzag 52° and S-shape fin channels. They showed the S-shape fin channel has equivalent heat transfer performance and $1/6 \sim 1/7$ pressure drop compared to the zigzag 52° . Kim et al. (2008) numerically investigated thermal-hydraulic performance under SCO_2 condition for a zigzag PCHE (Ishizuka et al., 2005) and an air-foil finned PCHE. Their results indicated that the air-foil PCHE has the same heat transfer performance and $1/20$ smaller pressure drop compared to the reference zigzag PCHE. Kim et al. (2009) and Kim and No (2011 and 2013) investigated thermal-hydraulic performance of a zigzag 15° channel through experiments and numerical analysis for He-He, He-Water, and Mixture-Water conditions. Kim and No (2012) suggested thermal-hydraulic correlations for various zigzag channels ($5^\circ \sim 45^\circ$) based on a numerical analysis. Through a cost analysis, they showed a $10 \sim 15^\circ$ zigzag channel has the lowest total cost for the Intermediate Heat Exchanger (IHX) in the PBMR among $5^\circ \sim 45^\circ$ zigzag channels. Yoon et al. (2014) accomplished a cost assessment of straight channel, zigzag channel, S-shape fin, and airfoil fin PCHEs for the IHX in HTGRs and SFRs. For the IHX (He-He) in HTGRS, their results showed laminar flow operation of a zigzag PCHE is the most promising. For the IHX (Sodium- SCO_2) in SFRs, their results indicated turbulence flow operation of a straight channel PCHE is the most promising. Manglik and Bergles (1995) proposed generalized correlations of friction factor and Colburn factor for Offset-Strip Fin (OSF) heat exchangers.

However, Ngo et al. (2007) and Kim et al. (2008) did not combine both heat transfer performance effect and pressure drop effect as a same unit. Kim et al. (2012) did not

consider different geometries other than zigzag channels when the optimized design was proposed. Yoon et al. (2014) did not calculate required plate thickness and displacement based on mechanical strength and corrosion, which were located between the two neighboring channels on the same plate.

Now, when an optimized PCHE channel design for a specific inlet/outlet operating condition is suggested, it is still not clear on how to optimize PCHE channel geometries among different channels. In this study, we focus on developing an optimized PCHE design for a 20-MW FLiNaK-SCO₂ Secondary Heat Exchanger (SHX) in FHRs. Heat transfer and pressure drop correlations for straight, S-shape, OSF, and zigzag channels were investigated. Considering those geometries for which thermal-hydraulic correlations are available, we determined PCHE cross-sectional dimensions. Thermal-hydraulics, mechanical aspect, and corrosion characteristics were considered for the PCHE design. Single banking and double banking are considered for the current design study. A cost analysis, which mainly consider capital cost and operating cost due to material cost and pumping power, is performed to downselect the most cost-effective PCHE channel geometry for both the SCO₂ and FLiNaK sides.

2. Considerations for PCHE designs

A reference operating condition of the 20-MW FLiNaK-SCO₂ SHX in the FHR, Sabharwall et al. (2011), was selected as design parameters. Details are presented in Table 1. FLiNaK and SCO₂ properties were obtained from Williams et al. (2006) and a NIST chemistry web-book (NIST website). Single banking and double banking were considered.

Table 1. 20-MW SHX operating conditions (Sabharwall et al. 2011)

SHX	T_{in} (°C)	T_{out} (°C)	P (MPa)	Mass flow rate (kg/s)
FLiNaK	676.2	575.0	0.153	102.2
SCO ₂	494.0	651.2	20.8	102.2

2.1 PCHE correlations for FLiNaK and SCO₂

Heat transfer and pressure drop correlations could be obtained for a limited number of PCHE channel geometries from literature: straight (Incropera and Dewitt, 2002), zigzag 52 and S-shape (Ngo et al., 2007), Zigzag 15° (Kim and No, 2013), and OSF (Manglik and Bergles, 1995). They are summarized in Table 2. Correlations for straight, zigzag 15°, and OSF channels are available in laminar region, while those for straight, S shape, zigzag 52°, and OSF channels are available in turbulence region.

2.2 PCHE channel geometry for FLiNaK and SCO₂ sides

Reynolds number in the flow channel is defined as:

$$Re = \frac{\rho v D_h}{\mu} = \frac{\dot{m} D_h}{\mu A_f} \quad (1)$$

If very similar values of hydraulic diameter, flow area, and mass flow rate are used for the different PCHE shapes, Reynolds number is inversely proportional to viscosity.

At the SHX operating condition shown in Table 1, Reynolds number on the SCO₂ side is 100 times larger than that on the FLiNaK side because the FLiNaK viscosity is much higher than the SCO₂ viscosity. Here, laminar flow in the range of $100 < Re < 600$ is currently considered on the FLiNaK side. Accordingly, The SCO₂ side is in the turbulent flow regime based on Reynolds number justification.

Table 2. Heat transfer and pressure drop correlations available for specific PCHE flow channel geometries

Shape	Top view (Black- wall, White- working fluid)	Channel Cross- section	Laminar flow	Turbulent flow
Straight (Incropera and Dewitt, 2002)		Circle Generally available	$f Re = 15.78$ $Nu = 4.089$	$f = (0.790 \ln Re - 1.64)^{-2}$ $Nu = \frac{(f/8)(Re - 1000) Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$ $3,000 < Re < 5 \times 10^6$ $0.5 < Pr < 2000$
S shape (Ngo et al., 2007)	$\theta = 52^\circ$ $p_f^x = 7.565mm$ $p_f^y = 2.11mm$ $t_f = 0.8mm$	Rectangle Width (w_f) = 1.31 mm Depth = 0.94 mm $D_h = 1.09$ mm	N/A	$f = 0.4545 Re^{-0.405}$ $Nu = 0.1740 Re^{0.593} Pr^{0.430}$ $3.5 \times 10^3 < Re < 2.3 \times 10^4$ $0.75 < Pr < 2.2$
Zigzag 52° (Ngo et al., 2007)	$\theta = 52^\circ$	Rectangle Width (w_f) = 1.31 mm Depth = 0.94 mm $D_h = 1.09$ mm	N/A	$f = 0.1924 Re^{-0.091}$ $Nu = 0.1696 Re^{0.629} Pr^{0.317}$ $3.5 \times 10^3 < Re < 2.2 \times 10^4$ $0.75 < Pr < 2.2$

	$p_f^x = 7.565mm$ $p_f^y = 2.11mm$ $t_f = 0.8mm$	mm		
Zigzag 15° (Kim and No, 2013)	$\theta = 15^\circ$ $p_f^x = 6.15mm$ $p_f^y = 2.62mm$ $t_f = 1.1mm$	Semi-Circle Width (w_f) = 1.51 mm $D_h = 0.922$ mm	$f Re = 15.78 + 0.557 Re^{0.82}$ $Nu = 4.089 + 0.00497 Re^{0.95} Pr^{0.55}$ $Re < 3000$ $0.66 < Pr < 13.41$	N/A
OSF (Manglik and Bergles, 1995)	$0.134 < \alpha < 1.034$ $0.012 < \delta < 0.060$ $0.038 < \gamma < 0.195$	Rectangle $\alpha = s/h$ $\delta = t/l$ $\gamma = t/s$	$f = 9.6243 Re^{-0.7422} \alpha^{-0.1856} \delta^{0.3053} \gamma^{-0.2659}$ $\times [1 + 7.669 \times 10^{-8} Re^{4.429} \alpha^{0.920} \delta^{3.767} \gamma^{0.236}]^{0.1}$ $Nu = 0.6522 Re^{-0.5403} \alpha^{-0.1541} \delta^{0.1499} \gamma^{-0.0678}$ $\times [1 + 5.269 \times 10^{-5} Re^{1.340} \alpha^{0.504} \delta^{0.456} \gamma^{-1.055}]^{0.1} Re Pr^{1/3}$ $120 < Re < 3000, \quad 0.5 < Pr < 15$ $0.646 < D_h < 3.414$	

Based on the correlations available for laminar and turbulent flow regimes in Table 2, we decided to take into account the straight, zigzag 15°, and OSF channels for the FLiNaK side and straight, S-shape, zigzag 52°, and OSF channels for the SCO₂ side, respectively. The Reynolds number, Prandtl number, and geometric parameters were kept within the applicable range of the available correlations. Similar hydraulic diameters were selected to ignore a hydraulic diameter effect in all designs. Single banking and double banking configurations as shown in Figure 1 were considered. Double banking is the most effective when one fluid has a much lower heat transfer coefficient than the other fluid.

Fig. 1. Banking configuration: Red (FLiNaK) and Blue (SCO₂)

2.3 Plate thickness

Figure 2 is a cross-sectional view of a PCHE. To determine the plate thickness, mechanical and corrosion aspects need to be carefully taken into account. Alloy 800 HT is assumed as the plate material for this study. For a thick-walled cylinder, a tangential stress is expressed by Oh and Kim (2008) as:

$$\sigma_D = \frac{p_i \left[\left(\frac{r_o}{r_i} \right)^2 + 1 \right] - p_o \left[\left(\frac{r_o}{r_i} \right)^2 + \left(\frac{r_o}{r_i} \right)^2 \right]}{\left(\frac{r_o}{r_i} \right)^2 - 1} \quad (2)$$

where p_i is the inside pressure, p_o is the outside pressure

The required ratio of r_o/r_i is determined by Eq. (2). The required thickness of $r_o - r_i$ is estimated in the range of 0.3~0.5 mm according to the selected geometry. However, the plates are pressed by diffusion bonding which requires high pressure and temperature so that the height of the plate would be reduced. Considering the diffusion bonding process

and the tolerance of channel size, r_i , which is 0.46~0.755 mm according to the selected geometry, is considered as a margin in addition to r_o-r_i . Eventually, t_{m_m} is considered to be r_o . Corrosion also needs to be considered. The corrosion rate of Alloy 800 HT is 0.0033 mm per year for FLiNaK from Sabharwall et al. (2010). The corrosion allowance, t_{m-c} , is determined by assuming a 30-year operation life. Then, a 20% margin is added to the sum of the material and corrosion thickness to determine t_m :

$$t_m = 1.2(t_{m-m} + t_{m-c}) \quad (3)$$

Fig. 2. Cross-sectional view of a PCHE

2.4 Distance between two neighboring flow channels

The fin thickness, the distance between two neighboring channels, for a plate-fin heat exchanger is calculated using (Hesselgreaves, 2001):

$$t_f = \frac{1}{\left[\left(\frac{\sigma_D}{\Delta p} \right) + 1 \right] N_f} \quad (4)$$

where t_f is the fin thickness, Δp is the pressure difference between the hot and the cold sides, σ_D is the maximum allowable stress, and N_f is the number of fins per meter.

The number of fins per meter is defined as:

$$N_f = \frac{1}{P} \quad (5)$$

According to Figure 2, pitch for the hot side P_{w_hot} can be expressed:

$$P_{w_hot} = d + t_f \quad (6)$$

Inserting Eqs. (5) and (6) into Eq. (4) yields:

$$t_{f_hot} = \left(\frac{\Delta p}{\sigma_D} \right) d \quad (7)$$

For the cold side, Eq. (8) can be obtained. Accordingly, a different spacing was used

for the hot and cold side during the design process.

$$t_{f_cold} = \left(\frac{\Delta p}{\sigma_D} \right) w_f \quad (8)$$

3. Cost-based PCHE design optimization

3.1 Assumptions

The following assumptions were used in the design process:

- (1) The fabrication limit for the size of a PCHE module (Gezelius, 2004): width $\leq$ 600 mm and length $\leq$ 1500 mm
- (2) Reynolds number must be greater than 100, because of significant axial conduction at $Re \leq 100$ (Rosa et al., 2009)
- (3) Etching process is used in making the micro-channels

3.2 Cost-optimization procedure

Following an iterative process shown in Figure 3, a cost-optimized PCHE is proposed. This process begins with assuming a FLiNaK Reynolds number. Thermo-physical properties of the FLiNaK and SCO_2 were determined at the average of the inlet and outlet temperatures. Once Reynolds number is known, total flow area can be calculated

by Eq. (1). The overall heat transfer coefficient and the required surface area are estimated by Eqs. (9) and (10).

$$\frac{1}{U} = \frac{1}{h_{FLiNaK}} + \frac{A_{s_FLiNaK} \Delta t_w}{A_{s_w} k_w} + \frac{A_{s_FLiNaK}}{A_{SCO_2} h_{SCO_2}} \quad (9)$$

$$A_s = \frac{Q}{U \Delta T_{lm}} \quad (10)$$

Then, the required transparent length was calculated as:

$$L = \frac{A_s}{S} \quad (11)$$

where S is the surface area per unit length.

Kim and No (2012) assumed the annual total cost of a heat exchanger consists of the capital cost and the operating cost as:

$$C_t = C_{cp} + C_o \quad (12)$$

For the capital cost analysis, only the material cost was taken into account. Fabrication cost, labor cost, and others were not considered as shown in Eq. (13). The units of capital cost (\$) has to be identical with the units of operating cost (\$/y). Therefore, the capital cost was converted to the payback cost as shown in Eq. (14) assuming 5% interest and a 30-year payback period.

$$C_c = C_M M = C_M \rho_M V \quad (13)$$

where $C_M = 150$ \$/kg for Alloy 800 HT

$$C_{cp} = \frac{C_c IR(1 + IR)^n}{[(1 + IR)^n - 1]} \quad (14)$$

For the operating cost estimation, the pumping power (W) calculated from Eq. (15) could be converted to the operating cost (\$/y) by multiplying the required pumping power by the electricity cost rate (\$/W-hr).

$$W_p = \Delta p \frac{\dot{m}}{\rho} \quad (15)$$

where Δp is pressure drop across the heat exchanger.

$$C_o = C_E \sum_{i=1}^2 W_{pi} \quad (16)$$

where C_E is 0.0000987 \$/W-hr (EIA, 2012), $i = 1$ is for the FLiNaK side, and $i = 2$ is for the SCO_2 side

This process for the total cost estimation was repeated changing FLiNaK Reynolds number from 100 to 600. Figure 3 is the iterative process.

Fig. 3. Iterative process to determine an optimized PCHE

3.3. Single banking

3.3.1 SCO₂ side optimization

For the flow channel geometry optimization on the SCO₂ side, SCO₂ channel configurations were varied, keeping the same channel configuration on the FLiNaK side. From Table 2, the zigzag channel 15° with a semi-circle cross-section ($D = 1.51$ mm) was used for the FLiNaK side, while the zigzag 52°, straight, S shape, and OSF channels with a rectangular cross-section ($W 1.31$ mm $\times$ $H 0.94$ mm) were considered for the SCO₂ side. Two different fin lengths of 7.565 mm and 2.40 mm were taken into account for the OSF channel. The Fanning factor and Nusselt number for each channel configuration on the SCO₂ side were plotted versus Reynolds number as shown Figures 4 and 5.

Fig. 4. Fanning factor vs. Re for SCO₂ channel configurations

Fig. 5. Nusselt number vs. Re for SCO₂ channel configurations

The order of the channel configuration from smallest to largest Fanning factor is:

Straight < S shape < OSF 7.565 mm < OSF 2.40 mm < Zigzag 52°

The order of channel configuration from largest to smallest Nusselt number is:

Zigzag 52° > OSF 2.40 mm > OSF 7.565 mm $\approx$ S shape > Straight

The heat transfer coefficients for the SCO_2 and FLiNaK sides are plotted in Figures 6 and 7. The heat transfer coefficient on the FLiNaK side is several times higher than that on the SCO_2 side.

Fig. 6. Heat transfer coefficient (SCO_2) vs. Re for channel configurations

Fig. 7. Heat transfer coefficient (FLiNaK) vs. Re for the zigzag 15° channel

Surface area per unit length for each configuration on the SCO_2 side was also investigated. The ratio of surface area per unit length (S) for each configuration to the surface area of the OSF 7.565 mm is:

Straight: OSF 7.565 mm: OSF 2.40 mm: Zigzag 52°: S shape = 1: 1.077: 1.241: 1.624: 1.780

This indicates that the S-shape flow channel is the best selection to maximize the surface area in a unit length.

Following the iterative process mentioned above, the total cost was investigated for

the single banking configuration as shown in Figure 8. The lowest total cost can be obtained by using the S-shape finned channel. The second and the third promising options would be zigzag 52° and straight channels.

Fig. 8. Total cost vs. Re for SCO₂ channel configurations

3.3.2 FLiNaK side optimization

For the channel geometry optimization on the FLiNaK side, various FLiNaK channel configurations were used, keeping the same channel configuration on the SCO₂ side. Based on the result that the S-shape was the best SCO₂ channel configuration, the S-shape was used for the SCO₂ side, while the semi-circle straight ($D=1.51$ mm), semi-circle zigzag 15° ($D=1.51$ mm), rectangle OSF 7.565 mm (1.51 mm $\times$ 0.755 mm), and rectangle OSF 2.40 mm (1.51 mm $\times$ 0.755 mm) channels were considered for the FLiNaK sides. Fanning factor and Nusselt number for each channel configuration on the

FLiNaK side were plotted versus Reynolds number as shown in Figures 9 and 10.

Fig. 9. Fanning factor vs. Re for FLiNaK channel configurations

Fig. 10. Nusselt number vs. Re for FLiNaK channel configurations

The order of the channel configuration from smallest to largest Fanning factor is:

Semi-circle straight < Semi-circle zigzag 15° $\approx$ Rectangle OSF 7.565 mm < Rectangle OSF 2.40 mm

The order of channel configuration from largest to smallest Nusselt number is:

Rectangle OSF 2.40 mm > Rectangle OSF 7.565 mm > Semi-circle zigzag 15° > Semi-circle straight

The heat transfer coefficient for the FLiNaK and SCO_2 sides are plotted in Figures 11 and 12, respectively.

Fig. 11. Heat transfer coefficient (FLiNaK) vs. Re for channel configurations

Fig. 12. Heat transfer coefficient (SCO₂) vs. Re for the S-shape channel

Surface area per unit length for each configuration on the FLiNaK side was also investigated. The ratio of the surface area per unit length (S) for each configuration to the surface area of the semi-circle straight is:

Semi-circle straight: Semi-circle zigzag 15°: Rectangle OSF 7.565 mm: Rectangle OSF 2.40 mm = 1: 1.035: 1.077: 1.241

This indicates that the OSF 2.40 mm is the best to maximize the surface area in a unit length among those surfaces considered.

Following the iterative process mentioned above, the total cost was estimated for the single banking as shown in Figure 13. The lowest total cost was obtained by using the

rectangle OSF 2.40 mm channel. The second and the third promising options would be the rectangle OSF 7.565 mm and semi-circle zigzag 15°.

Fig. 13. Total cost vs. Re for FLiNaK channel configurations

3.4. Double banking

3.4.1 SCO_2 side optimization

For the channel geometry optimization for the SCO_2 side, we varied SCO_2 channel configurations, keeping the same channel configuration on the FLiNaK side. The zigzag channel 15° with a semi-circle cross-section ($D = 1.51$ mm) was used for the FLiNaK side, while the zigzag 52°, straight, S-shape, and OSF channels with a rectangle cross-

section (W 1.31 mm $\times$ H 0.94 mm) were considered for the SCO_2 side. Following the same procedure used for the single banking, the total cost was investigated for the double banking configuration as shown in Figure 14. The lowest total cost can be obtained by using the S-shape channel, similar to the single banking case. The second and the third promising options would be the zigzag 52° and straight channels. The total cost difference between the S-shape and zigzag 52° or straight channel in the double banking is smaller than that in the single banking configuration.

Fig. 14. Total cost vs. Re for SCO_2 channel configurations (Double banking)

Figure 15 is a cost comparison between the single and double banking configurations for the zigzag 15° (FLiNaK)- S-shape (SCO_2) combination. It shows that the single banking is more cost-effective than the double banking. However, the difference is not significant. The double banking could be a good selection to balance heat transfer

multiplied by surface area (hA_s) because the heat transfer coefficient on the FLiNaK side is over three times higher than that on the SCO_2 side.

Fig. 15. Total, capital, and operating costs vs. Re for Zigzag 15° (FLiNaK)- S-shape (SCO_2) for the single and double banking

3.4.2 FLiNaK side optimization

For the channel geometry optimization on the FLiNaK side, we varied FLiNaK channel configurations, keeping the same channel configuration on the SCO_2 side. The S-shape was used for the SCO_2 side, while semi-circle straight ($D= 1.51$ mm), semi-

circle zigzag 15° ($D= 1.51$ mm), rectangle OSF 7.565 mm (1.51 mm $\times$ 0.755 mm), and rectangle OSF 2.40 mm (1.51 mm $\times$ 0.755 mm) channels were considered for the FLiNaK sides. Following the same procedure used for the single banking configuration, total cost of the heat exchangers was investigated for the double banking as shown in Figure 16. The lowest total cost can be obtained by using the rectangle OSF 2.40 mm channel. The second and third promising options would be the rectangle OSF 7.565 mm and semi-circle zigzag 15° channels.

Fig. 16. Total cost vs. Re for FLiNaK channel configurations (Double banking)

For the OSF channel, Reynolds number, Prandtl number, and the hydraulic diameter studied here are in the applicable range of the OSF correlation. However, the geometry parameters such as α , δ , and γ are out of the OSF correlation range because quite large

thickness t_f is required due the pressure difference between the FLiNaK and SCO_2 sides. Experimental data for the geometry used for this study need to be obtained in order to confirm the present OSF correlations or suggest new ones.

3.5 Selection

For the SCO_2 side, the S-shape shows the lowest total cost for both the single and double banking configurations. However, the total cost difference between the single and double banking is small. The straight and the zigzag 52° channels could be the second and third choice in terms of the total cost. For the FLiNaK side, the OSF 2.40 mm channel gives the lowest cost. However, the total cost difference between the single and double banking is quite small, similar to the SCO_2 channel optimization.

The OSF 2.40 mm (FLiNaK)- S-shape (SCO_2) with the single banking is the most cost-effective selection for 20 MW FLiNaK- SCO_2 SHX for the FHR. However, the OSF 2.40 mm (FLiNaK)- S shape (SCO_2) with the double banking would be a good option to balance heat transfer multiplied by surface area (hA_s) because the heat transfer coefficient in the FLiNaK side is over three times higher than that in the SCO_2 side. Here, we would like to suggest the OSF 2.40 mm (FLiNaK)- S shape (SCO_2) with the double banking for the 20 MW SHX in the FHR. If the S-shape fin is not appropriate due to the manufacturing considerations, OSF 2.40 mm (FLiNaK)- Zigzag 52° or the OSF 2.40 mm (FLiNaK)- Straight shape (SCO_2) with the double banking would also be promising options. A summary of the design details in these types of heat exchangers is

presented in Table 3.

Table 3. Suggested PCHE Designs

		OSF 2.40 mm – S-shape	OSF 2.40 mm – S-shape	OSF 2.40 mm – Zigzag 52°	OSF 2.40 mm – Straight
Banking configuration		Single	Double	Double	Double
Total volume (m ³)		0.381	0.390	0.398	0.500
Module number		4	4	4	4
Module transverse section		600 mm (W) × 667.4 mm (H)	600 mm (W) × 676.6 mm (H)	600 mm (W) × 947.1 mm (H)	600 mm (W) × 540.9 mm (H)
Length (mm)		237.8	239.9	212.8	385.4
FLiNaK Re _{inlet}		210	280	200	350
SCO ₂ Re _{inlet}		20,577	13,718	9,798	16,541
Heat transfer coefficient (W/m ² K)	FLiNaK	12,848	14,713	12,557	16,358
	SCO ₂	3,246	2,552	2,934	2,684
Pressure drop (kPa)	FLiNaK	48.437	79.510	32.891	186.751
	SCO ₂	27.733	14.276	24.896	12.852
Power density (MW/m ³)		52.47	51.29	50.16	39.72
Total cost (\$/y)		35,895	36,013	36,108	49,362
Capital cost (\$/y)		29,504	30,184	30,863	38,764
Operating cost (\$/y)		6,391	5,829	5,245	10,598

Here, a chemical etching process was assumed to fabricate the flow channels. Both the remaining solid metal and empty channel space etched away should be considered to estimate the total metal mass used for the PCHEs. If it becomes feasible to manufacture the OSF geometry considered here as a fin type and to perform diffusion bonding, a significant reduction of the total cost would be expected in the future.

5. Summary and Conclusions

To propose the cost effective heat exchanger for 20-MW FLiNaK-SCO₂ SHX in the FHR, PCHE channel geometries for which Fanning factor and Nusselt number correlations were available were summarized. Straight, S-Shape, OSF, and zigzag 52° channels were considered for the SCO₂ side while straight, zigzag 15°, rectangle OSF 7.565 mm, and rectangle OSF 2.40 mm channels were considered for the FLiNaK side. Thermal-hydraulic performance, mechanical aspect, and corrosion characteristics were taken into account. The results of the cost analysis indicated the most cost-effective design for the SCO₂ and FLiNaK sides based our assumptions were the S-shape fin and the OSF 2.40 mm flow channels. The zigzag 52° and straight could be next promising selections for the SCO₂ side if the fabrication of the S-shape finned channels is difficult. A significant reduction of the total cost could be expected in the future, if the OSF geometry considered here can be fabricated as a fin type and the diffusion bonding process for those finned plates are available. Additional experimental data for the geometries considered for this study are necessary in the future in order to confirm or suggest the OSF heat transfer and pressure drop correlations because a large thickness t_f is required for the OSFs due to 20 MPa pressure difference between the FLiNaK and SCO₂ sides.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the financial support by U.S. Department of Energy Nuclear Energy University Program (NEUP).

NOMENCLATURE

Re	Reynolds number	
Nu	Reynolds number	
ρ	Density	[kg/m ³]
v	Velocity	[m/s]
D_h	Hydraulic diameter	[m]
μ	Viscosity	[Pa·s]
$\dot{m}$	Mass flow rate	[kg/s]
A_f	Flow area	[m ²]
T	Temperature	[K]
P	Pressure	[Pa]
σ_D	Allowable stress	[Pa]
r	Radius	[m]
t_m	Required metal thickness	[m]
t_{m-m}	Required thickness determined by mechanical aspect	[m]

t_{m-c}	Required thickness determined by corrosion aspect	[m]
t_f	Fin thickness= Distance between two neighboring channels	[m]
Δp	Pressure difference	[Pa]
P	Pitch	[m]
P_{w_hot}	Pitch in the hot side	[m]
P_{w_cold}	Pitch in the cold side	[m]
d	Diameter	[m]
t_{f_hot}	Fin thickness in the hot side	[m]
t_{f_cold}	Fin thickness in the cold side	[m]
w_f	Channel width	[m]
d_f	Channel height	[m]
U	Overall heat transfer coefficient	[W/m ² K]
h	Heat transfer coefficient	[W/m ² K]
A_s	Surface area	[m ²]
Q	Heat exchanger thermal load	[W]
ΔT_{lm}	Log Mean Temperature Difference	[K]
L	Transparent length	[m]
N	The total number of channels	
N_s	The number of plate stacks	
f	Fanning factor	

C_t	Total cost per year	[\$/y]
C_{cp}	Payback cost per year for the capital cost	[\$/y]
C_o	Operating cost per year	[\$/y]
C_M	Material cost per kg	[\$/kg]
M	Total material mass	[kg]
ρ_M	Material density	[kg/m ³]
V	Total heat exchanger material volume	[m ³]
IR	Interest rate	
W_p	Pumping power	[W]
C_E	Average retail price of electricity to customers	[\$/W-hr]

Subscript

i: Inside

o: outside

in: inlet

out: outlet

FLiNaK or _FLiNaK	FLiNaK side
SCO ₂ or _SCO ₂	SCO ₂ side
w	Metal Wall

REFERENCES

Ngo, T.L., Kato, Y., Nikitin, K., Ishizuka T., 2007. Heat transfer and pressure drop correlations of microchannel heat exchangers with S-shaped and zigzag fins for carbon dioxide cycles. *Experimental Thermal and Fluid Science* 32, pp. 560-570.

Kim, D.E., Kim, M.H., Cha, J.E., Kim, S.O., 2008. Numerical investigation on thermal-hydraulic performance of new printed circuit heat exchanger model. *Nuclear Engineering and Design* 238, pp. 3269-3276.

Ishizuka, T., Kato, Y., Muto, Y., Nikitin, K. Ngo, T.L., Hashimoto, H., 2005. The 11th International Topical Meeting on Nuclear Reactor Thermal-Hydraulics (NURETH-11),

Ishizuka, T., Kato, Y., Muto, Y., Nikitin, K., Tri, N.L., Hashimoto, H., 2005. The 11th International Topical Meeting on Nuclear Reactor Thermal-Hydraulics (NURETH-11), Thermal-hydraulic characteristics of a printed circuit heat exchanger in a supercritical CO₂ loop, pp. 218–232.

Kim, I.H., NO, H.C., Lee, J.I., Jeon, B.G., 2009. Thermal hydraulic performance analysis of the printed circuit heat exchanger using a helium facility and CFD simulations. *Nuclear Engineering and Design* 239, pp. 2399–2408.

Kim, I.H. and No, H.C., 2011. Thermal-Hydraulic Performance Analysis of a Printed Circuit Heat Exchanger using a Helium-Water Test Loop and Numerical Simulations. *Applied Thermal Engineering* 31, pp.4064-4073.

Kim, I.H. and No, H.C., 2012. Physical model development and optimal design of PCHE for intermediate heat exchangers in HTGRs. Nuclear Engineering and Design 243, pp. 243-250.

Kim, I.H., No, H.C., 2013. Thermal-hydraulic physical models for a Printed Circuit Heat Exchanger covering He, He-CO₂ mixture, and water fluids using experimental data and CFD. Experimental Thermal and Fluid Science 48, pp. 213-221.

Yoon, S.H., No, H.C., Kang, G. B., 2014. Assessment of straight, zigzag, S-shape, and airfoil PCHEs for intermediate heat exchangers of HTGRs and SFRs. Nuclear Engineering and Design 270, pp. 334-343.

Manglik, R.M., and Bergles, A.E., 1995. Heat transfer and pressure drop correlations for the rectangular offset strip fin compact heat exchanger. Experimental Thermal and Fluid Science 10, 171-180.

Sabharwall, P., Kim, E.S., Mckellar, M., Anderson, M., 2011. Process Heat Exchanger Options for the Advanced High Temperature Reactor. INL/EXT-11-21584, Idaho National Laboratory. June 2011.

Williams, D.F., Toth, L.M., Clarno, K.T., 2006. Assessment of Candidate Molten Salt Coolants for the Advanced High-Temperature Reactor (AHTR). ORNL/TM-2006/12, Oak Ridge National Laboratory. March 2006.

Incropera, F.P., DeWitt, D.P., 2002. Fundamentals of Heat and Mass Transfer, fifth edition, pp. 508.

Sabharwall, P., Ebner, M., Sohal, M., Sharpe, P., Anderson, M., Sridharan., K., Ambrosek, J., Olson, L., Brooks., P., 2010. Molten Salts for High Temperature Reactors: University of Wisconsin Molten Salt Corrosion and Flow Loop Experiments-Issues Identified and Path Forward. INL/EXT-10-18090, Idaho National Laboratory. March 2010.

Hesselgreaves, J.E., 2001. Compact heat exchangers. Pergamon, pp.267.

Gezelius, K., 2004. Design of Compact Intermediate Heat Exchanger for Gas Cooled Fast Reactors, B.S. and M.S. Thesis MIT, Cambridge, MA, U.S.A..

Rosa, P., Karayiannis, T.G., Collins, M.W, 2009. Single-phase heat transfer in microchannels: The importance of scaling effects. Applied Thermal Engineering 29, pp. 3447-3468.

Energy Information Administration (EIA), 2012. Average retail price of electricity to ultimate consumers by end-use sector. www.eia.doe.gov.

Chang H. Oh and Eung S. Kim. Heat Exchanger Design Options and Tritium Transport Study for the VHTR System, INL/EXT-08-14799, 2008.