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NUMERICAL ANALYSIS OF A FIN ARRANGEMENT FOR AN OPTIMAL DESIGN OF AIRFOIL FIN PCHE

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ABSTRACT

Printed Circuit Heat Exchanger (PCHE) has increasingly used due to its wide range of applicability and operability up to high temperature and pressure conditions while keeping the excellent heat transfer performance per unit volume thanks to the combined techniques of the Photo-Chemical Etching (PCE) the Diffusion Bonding (DB). One of drawbacks is the high pressure drop due to the micro-channel configuration. Recently various channel shapes and configurations have been studied and proposed. One of innovative concepts was an airfoil-shape fin configuration in PCHE proposed by POSTECH in 2008. The model showed an excellent thermal-hydraulic performance compared to other conventional channel shapes of the PCHE including straight, zigzag, wavy etc., showing above 90% pressure drop decrease for the similar thermal performance. However, the optimal performance of the PCHE design has not been well studied due to its complicated configuration.

In this research, for the optimal design of the PCHE, the airfoil fin arrangement was studied with the CFD simulation using the ANSYS CFX software. Approximately two million numerical grids represents the computational domain and the k- ε turbulence model is applied to analyze the complex flow over the airfoil shaped channels. The initial and boundary conditions for the analysis were selected from the SCIEL high temperature recuperator design conditions. For the optimization of the fin arrangement, two geometric parameters such as the horizontal pitch and the vertical pitch were selected. The objective function is obtained from reasonable weight of heat transfer performance and friction factor with cost analysis. As the result of analysis, the objective function and optimal configuration of airfoil type PCHE are obtained for the limited case of the SCIEL recuperator design.

INTRODUCTION

Energy consumption with environmental consideration becomes important issue in many areas of society. Price of fossil fuels has increased because the total amount of the fuel is limited. As a result, the interest to high efficient power cycle becomes high due to its economic and ecological effects. The Supercritical Carbon Dioxide (SCO2) Brayton cycle has high power efficiency and relative small size to existed power cycle, so it is attractive cycle for the future power system.

The Brayton cycle (BC) is a single-phase gas cycle that requires less components than two-phase cycles like the Rankine cycle (RC). However, the cycle has larger energy consumption of compressor than RC due to the variation of volume of gaseous phase required much compression work for high pressure. BC also has larger size components because the density of a working fluid is very low (Feher, 1967).

However, the selection of SCO2 for a working fluid in BC could improve these disadvantages since the critical pressure (7.38MPa) and the critical temperature (31.25°C) of SCO2 are low operating conditions for the power cycle. The supercritical conditions have the advantage of liquid and gas phase: it has less compress compression work since its density is high like liquid. In addition, it has low viscosity than liquid and thermodynamic properties rapidly change with small temperature and pressure variation. It makes high heat transfer and high specific heat capacity. These features could make SCO2 BC efficient and compact (Russick et al., 1996).

In general, heat exchanger is one of the key components to determine the power cycle size. The Printed Circuit Heat Exchanger (PCHE), a micro-channel heat exchanger, produced by photochemical etching and diffusion bonding techniques is selected to improve performance as well as to provide compactness of the power cycle size. It allows the complex shape of channel with the small size of flow region. As a result, it has compact size, but it has an incredible heat capacity per unit volume because the size of flow region and wall are in mm-scale (Ahn et al., 2013).

Many researchers have tried to modify the shape of the PCHE channel because photochemical etching allows various shapes in mm-scale size. Ishizuka et al. (2005) performed the experimental study of the zigzag type PCHE with various conditions such as different pressure, temperature, and mass flow rate. Kim et al. (2008) studied the numerical simulation results on the bending angle effect and the validation of numerical models. They also developed a new type of PCHE (POSTECH-PCHE) that has a channel shape adopted “airfoil-shaped fin” and it could improve the performance of the PCHE. Tsuzuki et al. (2009) performed numerical analysis to compare heat transfer and operation consumption of the S-shaped fin to the zigzag channel PCHE. Choi (2010) performed numerical study about the effect of

POSTECH-PCHE (the airfoil shaped fin channel) on the heat transfer and pressure drop. These previous studies show that the performance of the airfoil type PCHE was notably superior to other designs of PCHEs.

For the optimization of PCHE, Lee and Kim (2012) performed analysis to find optimal configurations of the zigzag type PCHE with using the proposed objective functions and found the Pareto optimal front of the zigzag type PCHE. Kim and No (2012) performed the cost factor analysis of the zigzag type PCHE for HTGRs. Yoon et al. (2014) performed analysis to find optimal configuration of POSTECH-PCHE. These studies attempt to search the optimal PCHE with indetermination of a suitable objective function for the airfoil type PCHE.

Lately, a collaborative effort of Korean Atomic Energy Research Institute (KAERI), KAIST and POSTECH under the sponsor of the Korean National Foundation designed and constructed an experimental loop of S-CO2 Brayton cycle, called SCIEL at the KAERI, Korea. SCIEL designed for generating the output power of 100kWe. POSTECH is responsible to design heat exchangers (HX) including the SCIEL recuperator with the innovative concept such as POSTECH-PCHE, however we have not known the optimized configurations of the airfoil type PCHE.

Since the various complex shapes of channels impede vigorous numerical analysis of PCHE channel, a unit channel with dimensionless scale parameters was examined in this study. The correlations of the dimensionless numbers could be obtained from the simulation result and applied the previous methods to define an adequate objective function to investigate the optimal configurations of the airfoil type PCHE.

NUMERICAL SIMULATION

Geometry

Since full geometry analysis is extremely expansive in computation, one channel, the symmetric unit part of the entire PCHE channel, is simulated in this analysis that provides simulation results with sufficient accuracy in reasonable computation time and efforts. The channel length of the unit model is also limited because the predicted length of the recuperator is unnecessarily long to represent the channel performance of the PCHE. The airfoil, NACA0020-63, with the length of 4mm is determined by our previous analysis (Choi, 2010). The thickness between channel and plate which divides the hot and cold channel is 0.8mm. The length of the simulation domain is 50 mm. The working fluids of the channels for the hot and cold sections are SCO2. The geometry and boundary conditions of the simulation domain are shown in Figure 1.

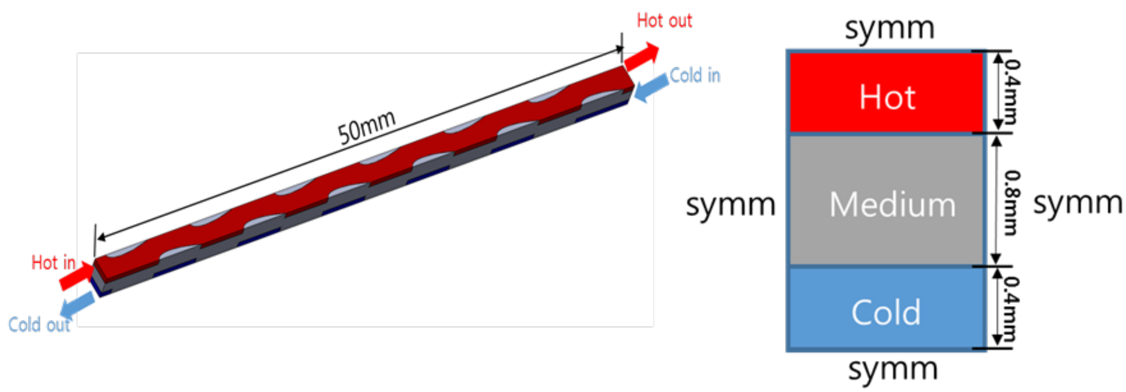

Figure 1. Schematic of simulation domain and boundary conditions

Mesh Generation

Each numerical condition has different number of meshes (the maximum number of meshes up to 2 million) to provide reasonable accuracy of the numerical analysis. The number of meshes is determined after the vigorous mesh dependence test. The shape of the meshes is tetrahedral employed in the ANSYS ICEM CFD package.

Validation of Numerical Methods

Although we set the limited number of meshes for the numerical analysis, the model should be validated for its suitability with the experimental data. However, there is no directly applicable experimental data for the airfoil type PCHE from the previous studies. In particular, the selection of a proper turbulence model in this analysis is important and thus two different experimental data; straight PCHE channel (Kruizenga et al., 2012) and pressure drop data of a single airfoil (Boutilier and Yarusevych, 2012), were chosen for the validation.

The studies (Kruizenga et al., 2012) from the University of Wisconsin-Madison's group performed heat exchange experiments for a straight channel PCHE with a SCO2-CO2 system. They used the experimental data to validate numerical simulations to examine various turbulence models. The SST k-ω model provided good agreement (less than 3%) with experimental data. However, this model required very fine meshes near the wall region. As a result, it is inadequate for our case due to the limitation of computational power. Therefore, the alternative k- ε model with the standard wall function (kε-SWF) is selected since the results showed that the results were within the maximum error of 7% from experimental results.

Nonetheless, since kε-SWF is only validated the straight channel PCHE with SCO2, the validation for the airfoil-finned PCHE is still required. For the purpose, the validation was studied in terms of pressure drop only due to the rack of experimental data to compare. A wind tunnel experimental data of an airfoil shows that kε-SWF showed sufficient accuracy to simulate (Boutilier and Yarusevych, 2012). Therefore kε-SWF is decided to use in the present study for the performance of POSTECH-PCHE. Simulations results were obtained from the commercial CFD code: ANSYS CFX. The temperature and pressure dependent thermo-physical properties of SCO2 were employed by the NIST chemistry web-book.

Table 1. Boundary values of working fluid

Conditions	Hot channel	Cold channel
Inlet temperature	451.3 °C	216.1 °C
Inlet pressure	7.8 MPa	19.8 MPa
Mass flux	937.5 kg/m^{2}s	937.5 kg/m^{2}s

Initial and Boundary Conditions

The initial and boundary conditions of the simulation in the present study are based on the experimental design conditions of the SCIEL loop that had been constructed at KAERI by the collaborative efforts among KAERI, KAIST and POSTECH. The inlet temperature, the working pressure and the mass flow rate are defined as shown in Table 1. The width and height of the recuperator are defined after the selection of the adequate mass flux of the unit channel for simulation.

DATA REDUCTION

Geometry Analysis

For the optimization of the airfoil type PCHE, numerical simulations carried out by varying the characteristic size parameters of the airfoil arrangement. Three dimensionless parameters such as the staggered length ζ_s , the horizontal pitch length ζ_h and the vertical pitch length ζ_v represent how densely airfoil arrangement is populated. The present analysis sets ζ_s as 1 since Yoon et al (2014) showed that ζ_s could be optimized at 1. For the limitation of computing power, we selected 11 cases for each type of pitches from 1.1 to 4. Definition of dimensionless parameters and required features for defining are shown in Figure 2 and Table 2 respectively.

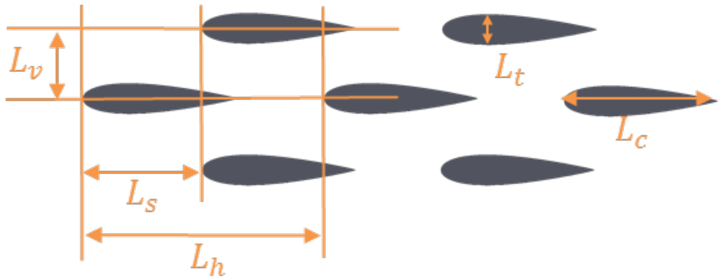

Figure 2. Geometrical parameters of airfoil array

Table 2. Definitions of dimensionless parameters about geometry

Dimensionless Parameter	Definition	range
Staggered arrangement	$\zeta_s = 2L_s / L_h$	1
Horizontal pitch	$\zeta_h = L_h / L_c$	1.1 to 4
Vertical pitch	$\zeta_v = L_v / L_t$	1.25 to 4

Heat Transfer

There are many dimensionless numbers that represent the performance of heat transfer. Traditionally, the Corburn-j factor has been used to show the performance of heat exchanger for evaluating heat transfer. It is the modified Stanton number with considering moderate variation of the Prandtl number. However, the Corburn-j factor limitedly shows the effect of channel geometry. The Nusselt number, the ratio of the convection heat transfer to the conduction heat transfer, includes the hydraulic diameter which represents the geometric characteristic of the channel selected in this research. It could represent the complex shape of the airfoil type PCHE channel on heat transfer performance.

- Colburn-j factor

$$j = St \cdot \mathrm{Pr}^{2/3} = \frac{Nu \cdot \mathrm{Pr}^{-1/3}}{\mathrm{Re}} \tag{1}$$

- Nusselt number

$$Nu = \frac{h}{k / D_h} = \frac{q'' D_h}{k(T_w - T_m)} \tag{2}$$

Pressure Drop

Traditionally, the Fanning friction factor is used for the typical pressure drop coefficient, the ratio of the wall shear stress to the flow kinetic energy per unit volume. Although it is slightly dependent on the channel geometry in the turbulence flow and strongly dependent on the shape of the channel in the laminar flow, it is used to predict the pressure drop of the flow channel. Some of researchers tried to use the Euler number as the performance of pressure drop. This number shows the normalized pressure drop of the channel, but it is not the adequate selection to predict the pressure drop of the designed channel. Therefore, the Fanning friction factor is selected to the dimensionless number for the pressure drop (Lee and Kim, 2012; Kim and No, 2012).

- Fanning-friction factor

$$f = \frac{\tau_w}{\rho u_m^2} \tag{3}$$

SIMULATION RESULTS

Heat Transfer Analysis

For the unit channel of the 50mm in the channel length, the averaged bulk properties are used for obtaining the Nusselt number, the Reynolds number, and the Prandtl number. The plots from Figures 3 to 8 are the predicted Nusselt number in terms of the mass flux, the vertical pitch and the horizontal pitch for the hot and cold sides of PCHE. The dotted points show the results of the numerical analysis, and lines indicates the predicted correlations for the prediction. The Nusselt number increases with the horizontal pitch, but decreases with the vertical pitch.

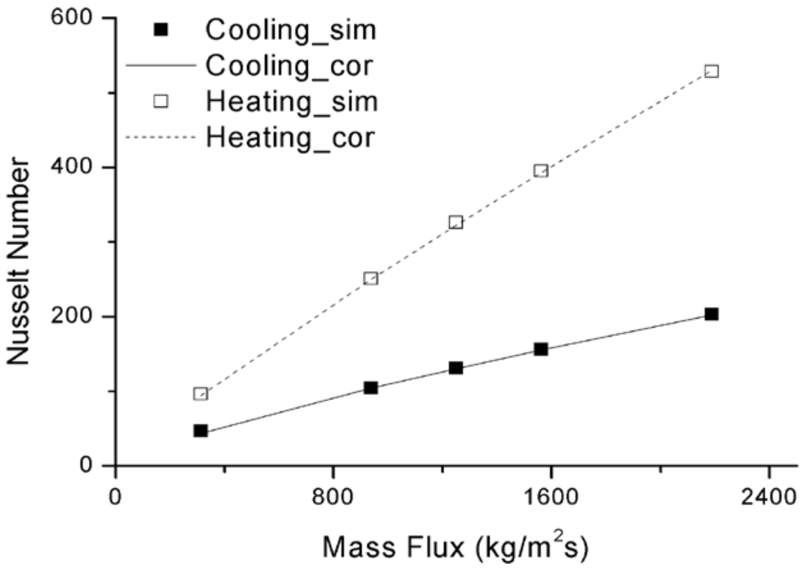

Figure 3. Effect and correlation of mass flux variation

Also the Nusselt number almost linearly increases with the increase of the mass flux. Since the mass flux at the inlet is the same for the horizontal and the vertical cases, the larger effect of the horizontal pitch on the Nusselt number means that the volume of the unit channel is smaller than the small horizontal pitch case. As a result, since the velocity of fluid at the channel becomes larger, heat transfer (the Nusselt number) increases. The vertical case shows the inverse effect in the same manner. There exists an evitable tendency to mass flux, horizontal pitch, and vertical pitch. It shows the potential of correlation existence.

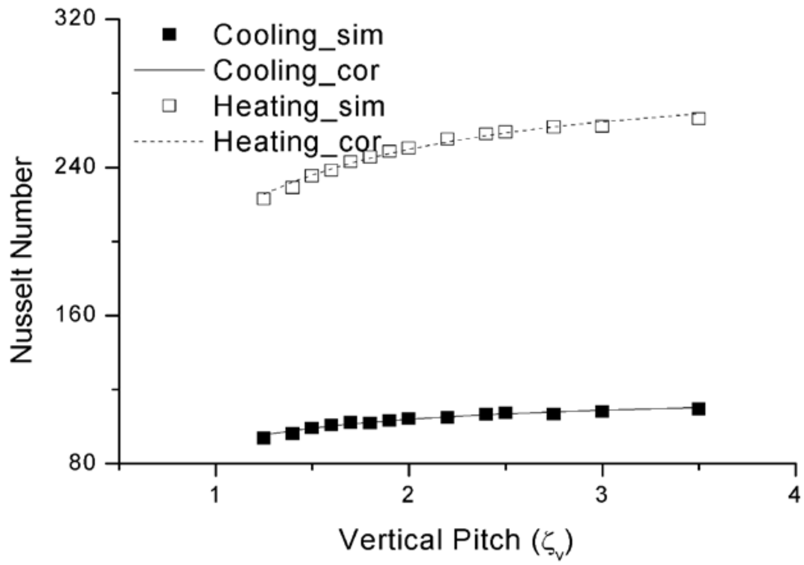

Figure 4. Effect and correlation of vertical pitch variation

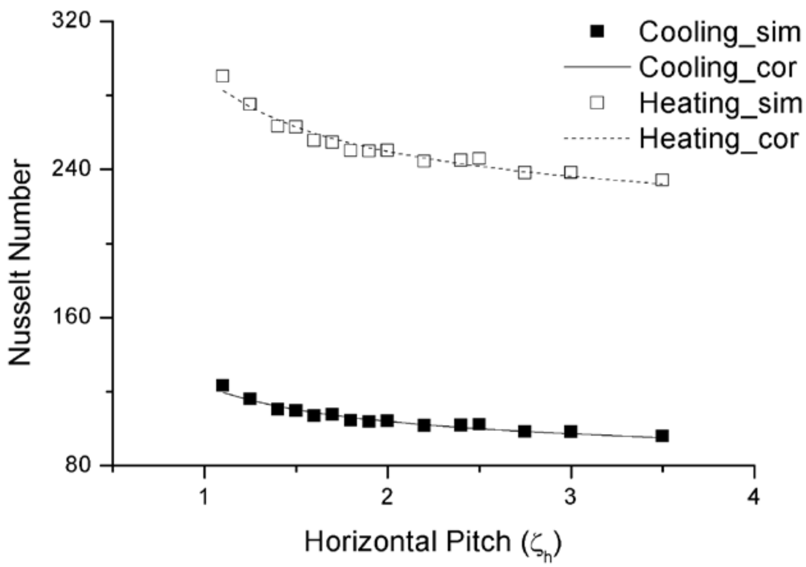

Figure 5. Effect and correlation of horizontal pitch variation

The Nusselt number of the in-pipe flow is usually predicted by the Dittus-Boelter equation presented by the Prandtl number and the Reynolds number. Although this type of the channel shape is not in similar to pipe, we assume that both straight and airfoil channels have similar physical properties without airfoil effects. Therefore we proposed the modified Dittus-Boelter equation with a correction factor that includes the horizontal and vertical pitch effect.

We also assume that both heating and cooling modes of SCO2 have different correlations because the previous researcher Dang and Hihara (2004) showed that traditional correlations of the Nusselt number could not predict accurately experimental results at the same time.

- The Dittus-Boelter equation

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{n} \quad \text{(n=0.3 for the cooling mode and n=0.4 for heating mode)} \tag{4}$$

- The Modified Dittus-Boelter equation

$$Nu = a\,\mathrm{Re}^{b}\,\mathrm{Pr}^{c}\,\zeta_v^{d}\zeta_h^{e}(a, b, \mathrm{c}, \mathrm{d}, \mathrm{e} = \mathrm{constant}) \tag{5}$$

Each coefficient for the correlation is calculated by the multiple regression analysis with the simulated results and shown in Table 3. The errors of the analysis are lower than 5.7 and 2.6% for the cooling and the heating modes, respectively.

Table 3. Coefficients and errors of modified Dittus-Boelter equation

Mode	a	b	c	d	e	R2	Average Error	Maximum Error
Cooling mode	0.0314	0.794	0.3	-0.0509	-0.0846	0.996	0.09%	5.72%
Heating mode	0.0113	0.889	0.4	-0.0488	-0.0492	0.998	0.02%	2.63%

Pressure Drop Analysis

Pressure drop is also an important factor for the design of PCHEs because it determines the operating pumping power. It can be separated into three components in terms of its physical origins such as the surface friction, the airfoil pressure drop, and the acceleration effect. The pressure drop due to the surface friction shows the same result of a normal in-pipe flow. The pressure drop due to airfoil is an additional contribution in the PCHE channel. Finally the acceleration effect appears due to the differene of density at the inlet and the outlet of the channel. Those are expressed in Eq (6).

$$\Delta P_{total} = \Delta P_{friction} + \Delta P_{airfoil} + \Delta P_{acceleration} \tag{6}$$

The density of fluid at the inlet and the outlet conditions are determined by the temperature and pressure that are already known by the CFD calculation or the design requirement. As a result, we can consider the corrected pressure drop without the acceleration effect. The pressure drop due to the surface friction is assumed to be similar physical phenomena to an in-pipe flow. With this assumption, the friction factor that is the dimensionless number of pressure drop due to the surface friction can be obtained from the Colebrook-White equation that describes the moody chart as an equation at the straight circular channel. The pressure drop of a single airfoil can also be expressed by the similar equations with the drag coefficient. Therefore we could obtain the integrated friction factor that includes both the airfoil effect and the surface friction.

The Colebrook-White equation represents the Moody chart well in a form of an implicit equation. For the Darcy-friction factor from the Colebrook equation, the iteration process is required. However, considering the uncertainty of the experimental data and relative roughness, the approximated equation of the Colebrook equation as known as modified the Blasius equation is used since this equation does not require the iteration process.

- The Colebrook-White equation

$$\frac{1}{\sqrt{f_{Darcy}}} = -2.0\log\left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f_{Darcy}}}\right) \tag{7}$$

- The Blasius equation

$$f = 0.079\,\mathrm{Re}^{-0.25} \tag{8}$$

The friction factor is divided into two parts for the source of pressure drop and it showed in Eq (9). To describe the friction factor, we use the modified Blasius equation.

$$f_{total} = f_{surface} + f_{airfoil} = a\,\mathrm{Re}^{b} + c\,\mathrm{Re}^{d}\,\zeta_v^{e}\zeta_h^{f} \tag{9}$$

From the result of our numerical analysis, the pitch-friction factor as shown in Figures 10~14 shows that the friction factor decreases with the increase of the horizontal pitch and the vertical pitch. Also mass flux rate and friction factor have an inversely proportional relationship. The growth of the horizontal pitch and the vertical pitch mean that there exist small numbers of airfoil fin at the unit channel. As a result, the drag force over the airfoil fins is reduced to produce the lower friction factor. This characteristic prevents to come up with an adequate objective function that the high Nusselt number at the low pressure drop is the best situation when its horizontal and vertical pitches becomes large. However we should consider high pitches meaning the small heat transfer area per unit channel. Finally the entire size of heat exchanger becomes larger with the same requirement of total heat transfer and the large size is undesirable feature of heat exchangers. Therefore in order to establish the objective function for PCHE, we should do quantitative analysis with correlations.

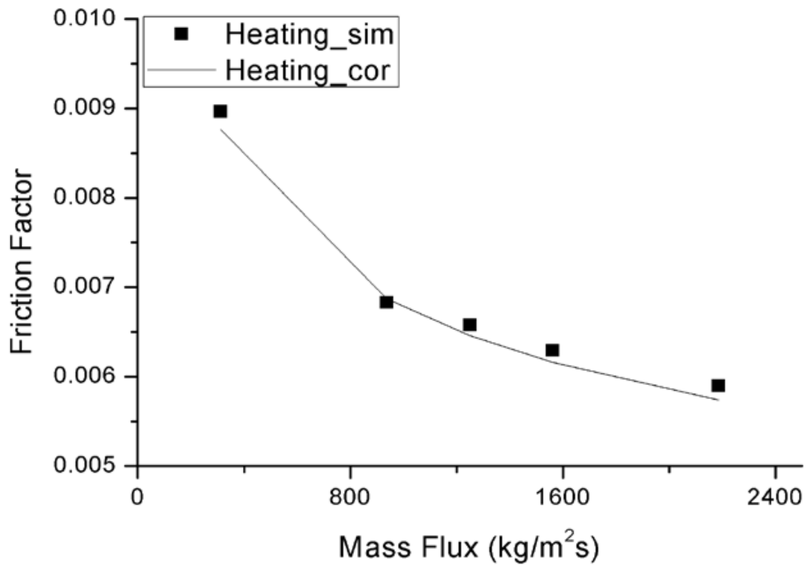

Figure 6. Friction factor variation with variation of mass flux

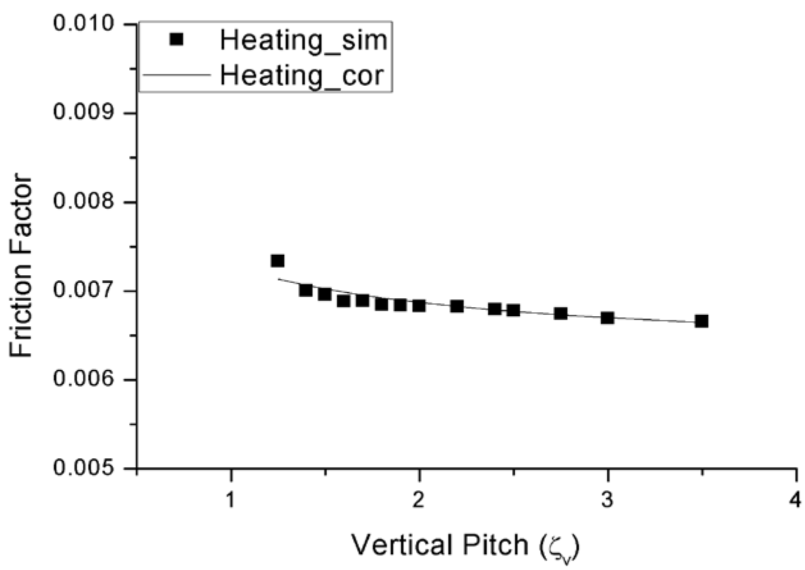

Figure 7. Friction factor variation with variation of vertical pitch

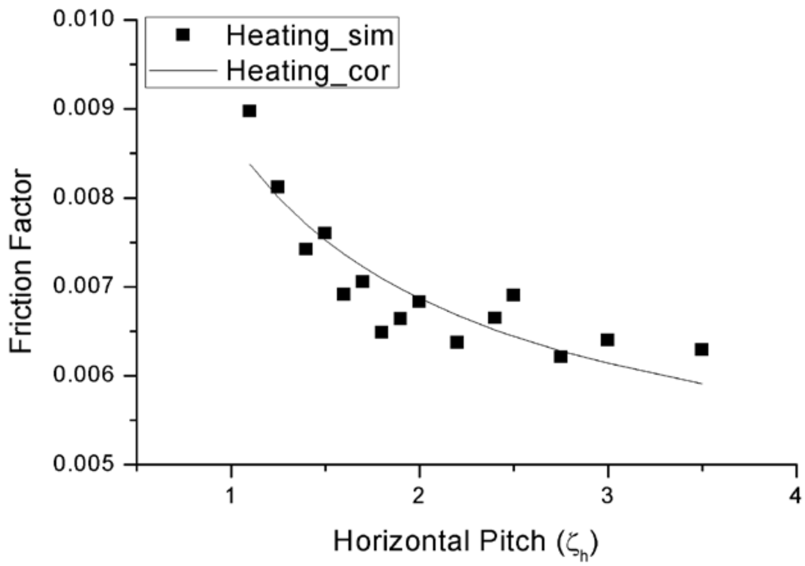

Figure 8. Friction factor variation with variation of horizontal pitch

Based on the form of the correlation defined previously, each coefficient of the correlations as shown in Table 4 is calculated by the multiple regression analysis at its adjusted coefficient of determination of about 0.85. However the errors of the friction factor are acceptably lower than 9.9%. Therefore we could consider that the correlations have sufficient accuracy in this airfoil configuration and thus can be used to design PCHE for considering an objective function.

Table 4. Coefficients and errors of the modified Blasius equation

Mode	a	B	c	d	e	f	R^2	Average Error	Maximum Error
Cooling mode	0.0237	-0.211	0.0306	-0.182	-0.768	-0.153	0.83	0.05%	9.9%
Heating mode	0.0087	-0.301	0.0171	-0.113	-0.726	-0.0346	0.85	0.03%	9.4%

OPTIMIZATION PROCESS

Design process

A design factor of the recuperator of the SCIEL loop is given in Table 5. For the given effectiveness value in the loop, the value of heat transfer at the recuperator is fixed. Therefore we should find the volume enough to satisfy the requirement. The effectiveness-NTU method is generally used for the design of heat exchanger because it removes the iterations when the LMTD method is used.

Table 5. Design factors of the SCIEL recuperator

Conditions	Hot channel	Cold channel
Inlet temperature	451.3 °C	216.1 °C
Inlet pressure	7.8 MPa	19.8 MPa
Mass flow rate	4.2 kg/s	4.2 kg/s
Effectiveness	0.95	

The effectiveness of a heat exchanger is the ratio of heat transfer rate to the maximum heat transfer rate of a heat exchanger as given in Eq. (10). It is calculated from the assumed infinite length counter-flow heat exchanger that the inlet temperature of the cold side is the same as on the outlet temperature of the hot side; the inlet temperature of the hot side is also the same as the outlet temperature of the cold side.

$$\varepsilon = \frac{q}{q_{\max}} = \frac{q}{C_{\min}(\mathrm{T}_{h,i} - T_{c,i})} \quad C_{\min} = \min\{\mathrm{C}_{cold}, C_{hot}\} \tag{10}$$

Since the required effectiveness is 0.95, with the inlet temperature of the hot and cold sides are 451.3 °C and 216.1 °C. NTU (Number of Transfer Unit) is calculated from the effectiveness and the specific heat capacity with the assumption of counter-flow at concentric pipe as below.

$$NTU = \frac{UA}{C_{\min}} = \frac{1}{C_r - 1}\ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right), \; C_r = \frac{C_{\min}}{C_{\max}} \tag{11}$$

where UA is the overall heat transfer coefficient defined as the total heat resistance between hot and cold fluid. If we assume that heat resistance of wall is negligible, it could be calculated from the convection heat transfer coefficient obtained by the Nusselt number. The areas for heat transfer are the same at the hot and cold channels.

$$\frac{1}{UA} = \frac{1}{(hA)_c} + \frac{1}{(hA)_h} = \frac{1}{A}\left(\frac{1}{h_c} + \frac{1}{h_h}\right) \tag{12}$$

As a result, the required heat transfer area satisfied the design factor is determined, and the total length could be calculated from the unit surface area and length.

Also the friction factor could be determined by the boundary conditions of the recuperator; similarly the pressure drop from the designed length satisfied the requirement of heat transfer characteristics. The correlations about the Fanning-friction factor could then determine the value of pressure drop when its length is known.

$$\Delta p = 4 f_{total} \frac{L}{D_h} \frac{1}{2} \rho v^2 + p_{accelaration} \tag{13}$$

From these results about the size of the recuperator that satisfies the requirements and pressure drop, the total cost could be calculated and it works as an objective function.

Previous Optimizations

Although there existed few attempts to obtain an adequate objective function for PCHE, there had some challenges on the real size applications. Lee and Kim (2012) defined two objective functions for a zig-zag type PCHE and found the Pareto optimal front. Their objective functions to describe heat transfer and pressure drop are the effectiveness of channel and the Euler number. In our case, however, the Pareto optimal front was not considered. However Figure 9 shows the inflection point similar to one observed in the Lee and Kim’s case. However, this type of multi-objective optimization excludes the weight of each objective function. In addition, their attempts to make the one objective function with an adequate weight (Eq.15), but there was no reason of setting weight of each factor (Lee and Kim, 2012, ICAPP12). As a result, for the application of heat exchanger, defining the weight of each factor related to the manufacturing cost should be considered.

$$f_{objective} = 1/effectiveness + 0.09Eu \tag{14}$$

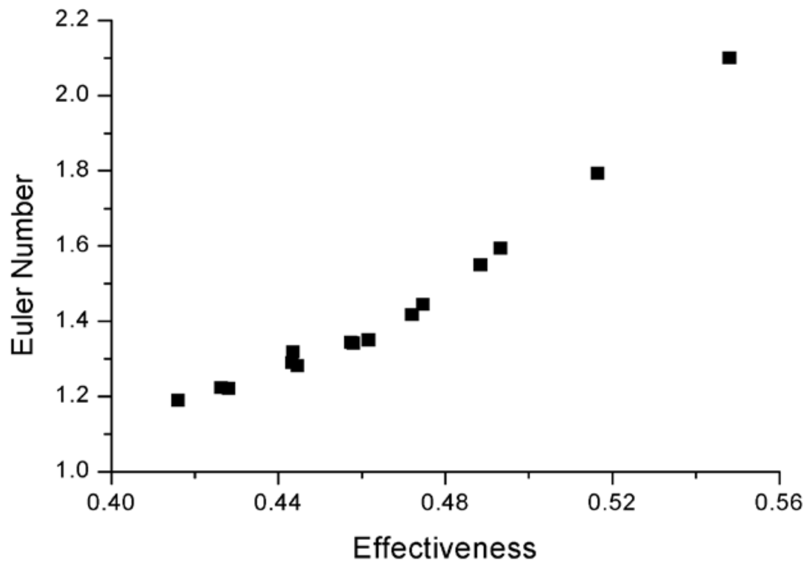

Figure 9. Effectiveness - Euler Number optimization (horizontal cases, heating mode)

Optimization Based on the Cost Analysis

Kim and No (2012) performed the cost analysis of a zigzag type PCHE. Their method seems reasonably suitable to the airfoil type PCHE because it considered the main cost factors of general heat exchanger. So this method is applied to our case. To determine the objective function of our airfoil type PCHE, analysis considering the cost factor is performed.

In the cost analysis, the total cost of a power cycle could be divided into two parts: the initial cost and the operation cost. The initial cost occurs due to the price of heat exchanger and the construction cost due to volume of heat exchanger. However we only consider the price of heat exchanger because of difficulty in considering the construction cost. The operation cost occurs due to the power consumption determined by heat exchanger pressure drop of pump that drives the fluid at the design conditions.

Also, both costs could not represent the absolute value of capital cost. Therefore we employed the compound interest of Korea to calculation to obtain the value of the total capital cost after the end of power cycle lifetime. The interest rate R is assumed to be 3.82% in average value of last 5 years. After adapting below equations (Eqs. 15-17), the initial and the operation costs could be compared with the same unit cost.

Both parts of the total cost are obtained from design progress. The SCIEL loop is the small size power cycle experimental facility that can generate up to 100kW. And thus we assume the small generating system designed to match the same conditions and organization. Using the results of the numerical analysis are based on the boundary conditions of the recuperator, the cost analysis is only performed for the recuperator.

$$C_{total} = C_{i\_\mathrm{ci}} + C_{o\_\mathrm{ci}} \quad (15)$$

$$C_{i\_ci} = C_i \times (1+R)^{year} \quad (16)$$

$$C_{o\_ci} = C_o \frac{(1+R)^{year}}{R} \quad (17)$$

Heatric Company, PCHE producer at England hosted workshop at MIT on October 2, 2003. Question and answer of this workshop were summarized in the MIT thesis (Gezelius, 2004). They provided useful information of the cost of PCHE. The cost of PCHE is generally defined in cost/kg since it is independent to the thermal conditions not like price/MW that varies with the operating condition. The cost for non-nuclear applications is approximately $50/kg when customers ordered large amount. It includes the price for manufacturing a complete heat exchanger including the header welding and customer designed unit. For nuclear applications, however, cost is approximately $30/kg lower than one in non-nuclear applications because nuclear applications required the large amount with standardization. However, this data is too old to apply the present conditions, also the SCIEL loop had the PCHE for precooling the whole system that could be used to make the standard cost for the PCHE unit. Although the channel shape of the precooler is zigzag type at present, the width and the depth of each channel is the same as the recuperator. From price of the precooler and its dimension, the proper estimation is obtained for each unit volume and its value becomes 1.072$/cm^3. Since the number of plates is a dominant factor of the total price, the unit of cost should be changed to the price per volume because the mass could be changed by the shape of channel while the total volume could not be changed.

$$C_i = 1.072\$ / cm^3 \times L \times W \times H \tag{18}$$

The operating cost could be represent by the production of pressure drop of the recuperator and the mass transfer rate. Also we have to determine the cost of power consumption. The retail cost of the electricity is higher than the production cost of unit power. However since we consider the power system, we use the generation cost of nuclear power plants in Korea, its value of 0.04$/kWh.

$$C_o = \Delta p \frac{\dot{m}}{\rho} \cdot 1 year \cdot 0.04\$ / kWh \tag{19}$$

For obtaining the optimal point, we vary the mass flux with various horizontal and vertical pitches. When we control the vertical pitch, the horizontal pitch is fixed to two. The same as before, the vertical pitch is fixed to two when the horizontal pitch is varied. Both conditions show that the mass flux is the dominant factor for 20-year total operation cost and it is nearly proportional to the total cost when the mass flux is large. In both cases, the total cost is nearly inversely proportional to pitch while the low mass flux makes it to horizontal or linear relation. In the vertical pitch analysis, there is a local minimum point of analyzed conditions. The vertical and horizontal pitches at this point are 2.75 and 2.0, respectively. In the horizontal pitch analysis, the low horizontal pitch point at the low mass flux is the local minimum. The vertical and horizontal pitches at this point are 2.0 and 1.1, respectively.

Furthermore, the cost analysis on the mass flux shows that the local minimum occurs at 312.5kg/m^2. The growth of volume is the dominant factor at the low mass flux region because the low fluid speed makes the operation cost to be relatively low portion of total cost. In contrast, the high mass flux with the high speed of fluid induces high pressure drop. Therefore the reduction of pressure drop makes low total cost.

From these results, we assume that vertical and horizontal pitches of the optimized design are approximately 2.75 and 1.1 with mass flux of 312.5kg/m^2, respectively. However the combined variation of the vertical and horizontal pitches might make unpredicted changes on fluid flow and heat transfer. Therefore further investigation is needed to predict the detailed analysis on pressure drop and heat transfer at such a complex geometry of PCHE.

CONCLUSIONS

This paper studied the airfoil-shape fin channel of POSTECH-PCHE for finding the optimum configuration of the fin channel by defining the objective function. Numerical simulation performed by ANSYS CFX and defined by the NIST data for SCO2 properties was performed. From the simulation results, we developed the relationships on heat exchange between bot hot and cold channels of SCO2 with the dimensionless airfoil configuration parameters. The recuperator was designed by the resulting correlations, and designs are evaluated by the objective functions with the cost analysis. It suggests the guideline for the manufacture of our airfoil-fin PCHE. The study should be validated and needed for further development in future since these correlations are obtained from the limited configurations.

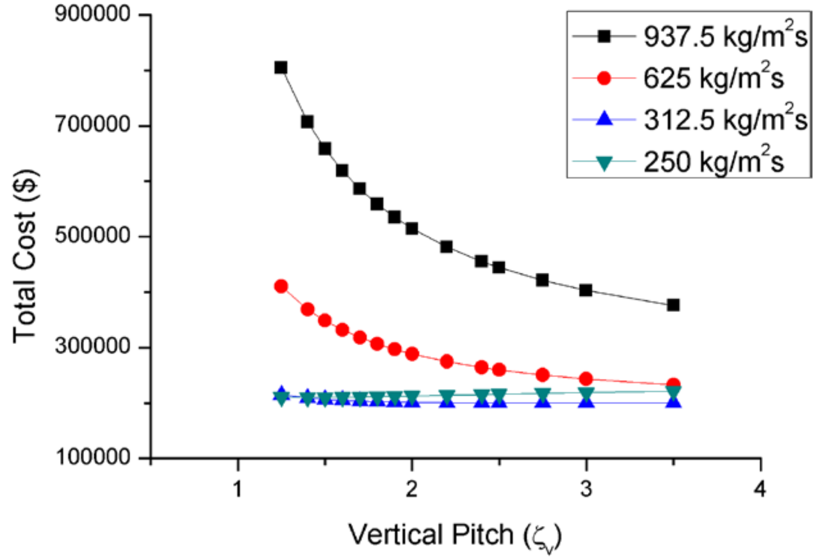

Figure 10. Total cost of the recuperator with vertical pitch variation

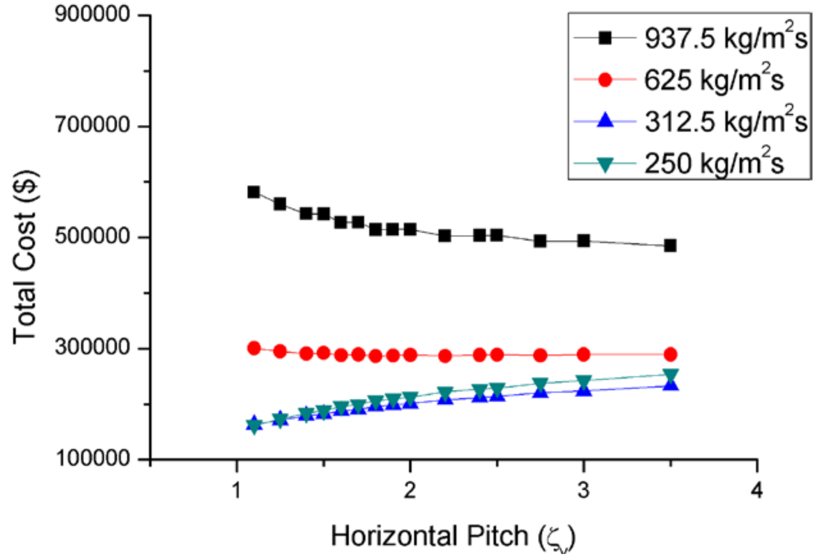

Figure 11. Total cost of the recuperator with vertical pitch variation

NOMENCLATURE

SCIEL	=	the S-CO2 Brayton cycle test facility	
KAERI	=	Korean Atomic Energy Research Institute	
C	=	specific heat at constant pressure(kJ/kg·K)	
C	=	cost	($)
D_h	=	hydraulic diameter	(m)
Eu	=	Euler number	(-)
f	=	Fanning friction factor	(-)
h	=	heat transfer coefficient	(W/m^2·K)
j	=	Colburn-j factor	(-)
k_s	=	solid thermal conductivity	(W/m·K)
k_f	=	fluid thermal conductivity	(W/m·K)
L	=	length	(m)

Nu	=	Nusselt number	(-)
P	=	Pressure	(Pa)
Pr	=	Prandtl number	(-)
q``	=	heat flux	(W/m^2)
R	=	interest rate	(%)
St	=	Stanton number	(-)
T	=	temperature	(K)

Greek symbols

ζ	dimensionless symbol	(-)
μ	dynamic viscosity	(Pa·s)
ρ	density	(kg/m^3)
τ	shear stress	(Pa)

Subscripts

m	material
o	operating cost per year
o_ci	operating cost after applying interest rate
h	horizontal
i	initial cost
i_ci	initial cost after applying interest rate
s	staggered
t	width
v	vertical
w	wall

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