Thermal and Hydraulic Performance of an SCO2 Printed Circuit Heat Exchanger (PCHE) with New Channel Geometry based on a Cosine Curve

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ABSTRACT
Printed circuit heat exchangers (PCHE) play a critical role in the performance and layout sizing of the new supercritical CO2 power cycle. The current study aims to predict the thermal and hydraulic performance of the new geometry of sCO2 PCHE. In this new geometry, conventional zig zag geometry has been replaced with the new channel geometry that is based on the curve of the cosine function. The new geometry has eliminated the recirculation zones that result in regions with low rates of heat transfer. Boundary reinitialization has been achieved by a discontinuous fin structure. Thermal and hydraulic performance of the printed heat exchanger were evaluated using the commercial ANSYS-CFX code. The computational model was validated by using conventional geometry and comparing numerical results with published correlations. Discretization of the computation domain was conducted using ANSYS ICEM-CFD and ANSYS-Mesh. Properties of supercritical carbon dioxide (sCO2) were computed using an in-house Matlab code that was linked with NIST Refprop. The same code was used to write the properties in the real gas property (rgp) table which was then supplied to ANSYS CFX in rgp file format.

Keywords: PCHE, sCO2 cycle, thermal and hydraulic performance.

NOMENCLATURE

\( C_p \) fluid specific heat capacity \([ J \cdot kg^{-1} \cdot K^{-1}]\)

\( D_{h,y,d} \) hydraulic diameter \([mm]\)

\( f \) friction factor

\( k \) thermal conductivity \([W \cdot m^{-1} \cdot K^{-1}]\)

\( m \) Flow rate (mass) \([kg \cdot s^{-1}]\)

\( \text{Nu} \) Nusselt number

\( \text{PCHE} \) printed circuit heat exchanger

\( p \) pressure \([Pa]\)

\( \text{Re} \) Reynolds number

\( n \) surface normal \([m]\)

\( T \) temperature \([K]\)

\( u \) velocity vector \([m \cdot s^{-1}]\)

GREEK SYMBOLS

\( \mu \) dynamic viscosity \([kg \cdot m^{-1} \cdot s^{-1}]\)

\( \rho \) density \([kg \cdot m^{-3}]\)

\( \Pi \) Reynolds stress tensor \([kg \cdot m^{-1} \cdot s^{-2}]\)

\( \tau \) stress tensor \([kg \cdot m^{-1} \cdot s^{-2}]\)

SUB AND SUPER SCRIPTS

\( \text{ave} \) Average

\( c \) cold side

\( ci \) cold side inlet

\( cf \) cell face

\( fl \) fluid

\( h \) hot side

\( hi \) hot side inlet

\( i \) interface plate

\( loc \) Local

\( nw \) near wall

\( sl \) Solid

\( t \) turbulent

\( w \) Wall

1. INTRODUCTION

Our increasing global energy demands could be fulfilled by a very efficient power cycle that would also minimize the risks of global warming, ozone depletion and pollution. In the current decade, supercritical carbon CO2 Brayton cycle (sCO\(_2\) – BC) has been very popular in the research concerning
power and modern cycles. $sCO_2 - BC$ operates in the critical region where compressor inlet conditions are placed very close to the critical point. Near the critical point, thermophysical properties change abruptly, and this is taken advantage of to increase the efficiency of the cycle. Based upon the operation of $sCO_2 - BC$ in the critical region, the said cycle combines the advantages of the Rankine cycle (i.e. low compression work, due to high density values near the critical point), and the high turbine inlet temperature that is a feature of the Brayton cycle. Moreover, high operating pressures make the turbomachinery and heat exchangers very compact, at almost one tenth the size required for the Rankine cycle [1]. The performance/layout size of the $sCO_2 - BC$ is highly dependent on the size and performance of the printed circuit heat exchangers (PCHE), which have very small diameters and are manufactured using photo etching technology. Due to this smaller hydraulic diameter, the active length of the PCHE is significantly reduced, but its heat transfer characteristics are largely unaffected (i.e. colburn j factor [2]). A comprehensive review of PCHE research is provided below.

Ishizuka et al. [3] performed an experimental study to evaluate the thermal and hydraulic characteristics of the zig zag PCHE that has been frequently cited in the literature for the validation of numerical models. A new S-shaped PCHE was analyzed for its thermal and hydraulic performance by Tsuzuki et al. [4]. They also compared the performance of the S-shaped printed heat exchanger with the conventional zig zag PCHE, and concluded that the pressure drop in an S-shaped heat exchanger is five times less than that in a zig zag PCHE, while heat transfer characteristics are identical. Abel et al. [5] conducted numerical simulations to compute the performance of a PCHE with rounded bend zig zag channels. They proposed that CFD simulations predict heat fluxes from 5-25%, while pressure loss predictions were broadly comparable with the experimental results. A comparative performance analysis of zig zag, air foiled, and S-shaped PCHE fins was conducted by Yoon et al. [6], who also developed the correlations for the air foiled PCHE. They concluded that, as intermediate heat exchangers, zig zag PCHE are the most suitable. Seo et al. measured experimentally the hydraulic performance (pressure losses) in a straight channel PCHE. Li et al. [7] computed the heat transfer characteristics for a PCHE based on the PDF modelling of turbulent convection. For the straight and zig zag printed circuit heat exchangers of a regenerator, thermal and hydraulic performance were computed using numerical techniques by Meshram et al. [8]. Kim et al. used a CFD approach to the design of a PCHE. They also computed the performance of a heat exchanger for an extended range of Reynolds numbers and developed correlation in the corresponding range of Reynolds numbers. Saeed and Kim [9] evaluated the thermal and hydraulic performance of a zig zag heat exchanger for a wide range of Reynolds numbers. They also computed the pressure and temperature distribution for the full length of the heat exchanger.

In the current study a new fin shape for the printed circuit heat exchanger was introduced. The conventional zig zag channel geometry was replaced by one which is based on the sine curve. The thermal and hydraulic performance of the new channel geometry was computed using ANSYS-CFX commercial software. The mesh was generated using ICEM-CFD and ANSYS-Mesh. As the properties of sCO2 change swiftly near the critical point in the supercritical region, the performance of the heat exchanger cannot be computed accurately without modelling the process with some provision for the accurate thermophysical properties of the working fluid. In this regard, properties of sCO2 were computed using an in-house Matlab code, coupled with NIST Refprop. These code properties were written in the same format as the real gas property table (RGP file) and supplied to CFX. Finally, the performance of the new channel geometry was compared with the zig zag channel geometry used by Ishizuka et al. [3]. The conventional zig zag geometry was used to validate the computer model.

2. COMPUTATIONAL DOMAIN AND MODEL

Conventional channel geometry

The conventional zig zag channel geometry of the printed heat exchanger is shown in Figure 1, while other details of the geometry are provided in Table 1. Printed circuit heat exchangers consist of stacked plates of hot and cold fluid channels such that each cold plate is surrounded by two hot plates. The geometries of the hot and cold flow channels are different, such that the pitches of the hot and cold channels are 9 mm and 7.24 mm respectively. The zig zag angle for the hot fluid channel is 115° and 110° for the cold flow channel.

<table>
<thead>
<tr>
<th>Geometrical Parameters</th>
<th>Hot channels</th>
<th>Cold channel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>SS316L</td>
<td>SS316L</td>
</tr>
<tr>
<td>Plate thickness [mm]</td>
<td>1.63</td>
<td>1.63</td>
</tr>
<tr>
<td>Pitch [mm]</td>
<td>9</td>
<td>7.24</td>
</tr>
<tr>
<td>Channel separation [mm]</td>
<td>0.6</td>
<td>0.7</td>
</tr>
<tr>
<td>Depth of channel [mm]</td>
<td>0.90</td>
<td>0.90</td>
</tr>
<tr>
<td>Hydraulic diameter [mm]</td>
<td>1.106</td>
<td>1.10</td>
</tr>
<tr>
<td>Heat transfer area [m²]</td>
<td>0.697</td>
<td>0.356</td>
</tr>
<tr>
<td>Channel active length [mm]</td>
<td>1000</td>
<td>1100</td>
</tr>
<tr>
<td>Length of channel [mm]</td>
<td>896</td>
<td>896</td>
</tr>
</tbody>
</table>

Conventional channel geometry

In the current study a new fin geometry for PCHE is proposed in order to reduce the pressure losses across the printed circuit heat exchanger. In the conventional zig zag channel, a recirculation regions form at all bends, and these are the main cause of high entropy and large pressure losses. In this study, the issue has been tackled by replacing the zig zag with a geometry based on the cosine curve, as shown in Figure 2. The geometry of the PCHE based on the new fin is shown in Figure 3. Similarly to the zig zag shaped printed circuit heat exchanger, the configuration of the proposed PCHE is the same (i.e. one cold fluid channel is surrounded by two hot fluid channels), but the fin geometry for the hot and cold channels is kept the same, in contrast to the zig zag PCHE.
Figure 1. Geometrical details of a conventionally used zig zag channel in the PCHE
Governing equations

In order to solve the conjugate heat transfer problem for the printed circuit heat exchanger using steady and compressible form of continuity, energy and momentum equations [9], [10]. To solve the set of governing Eqs (1-5), ANSYS-CFX commercial software was employed.

\[ \nabla \cdot \rho \mathbf{u} = 0 \]

\[ \rho (\mathbf{u} \cdot \nabla) \mathbf{u} = -\nabla p + \nabla \cdot \mathbf{T} \]

In the above equation \( \mathbf{T} \) is a stress tensor defined in the equation below.

\[ \mathbf{T} = (\mu + \mu_s) \left( \nabla \mathbf{V} + (\nabla \mathbf{V})^T - \frac{2}{3} \delta \nabla \cdot \mathbf{V} \right) \]

\[ = \mu \left( \nabla \mathbf{V} + (\nabla \mathbf{V})^T - \frac{2}{3} \delta \nabla \cdot \mathbf{V} \right) + \nabla \cdot \mathbf{\Pi} \]

Here \( \mathbf{\Pi} \) Reynolds stress tensor.

In the given stress tensor Eq. (3), \( \mathbf{\Pi} \) is the term that represents the Reynolds stress tensor that was resolved using the SST turbulence model. The energy Eq. for the fluid domain is given in Eq. (5).

\[ \nabla . (\rho \mathbf{u} h) = \nabla . \left( \frac{\lambda + \lambda_t}{c_p} \nabla h \right) \]

The energy equation for the solid domain is Eq. (5).

\[ k_s \nabla^2 T = 0 \]

Meshing of the computational domain

Meshes of the computational domains were generated using ICEM-CFD and ANSYS-CFX. The mesh for the zig zag channels was constructed using ICEM-CFD, while that of the proposed channel geometry was constructed using ANSYS MESH, as shown in Figure 4. In order to optimize the mesh, five meshes were generated, each with a different sizing, and the optimal mesh was chosen such that no variation in the results was observed when the mesh size was increased. In order to obtain the maximum advantage from the turbulence model, a value of less than 2 for y-plus was imposed on all the wall boundary conditions. Moreover, a minimum number of 15 elements were ensured within the boundary layer thickness. To fulfill these conditions, the O-grid was generated for the conventional zig zag geometry, while for the proposed geometry, 15 inflated prism layers were added to capture the boundary layer accurately.
Figure 3. Geometrical arrangement of the proposed PCHE
Figure 4. Mesh generated for the conventional and proposed geometries of the PCHE
Boundary conditions

In order to minimize the computational cost, instead of using a full length domain, all simulations were computed using a smaller domain length of 120 mm. In this situation, boundary conditions for the smaller length were obtained using the pressure and temperature distribution obtained by Saeed and Kim et al. [9]. The flow is in opposite directions in the hot and cold channels. Total pressure at the inlet while mass flow boundary conditions were imposed at the outlet. Mass flow rates were computed ‘in accordance with the Reynolds number, and the Reynolds number was varied from 2500-5000 for the hot channel and 5000-10000 for the cold channel. Boundary conditions obtained are listed in Table 2. Interface conditions were applied using Eq. (6) at the surface where fluid is in contact with a solid. Whereas top, bottom and sides surfaces were assigned with periodic boundary conditions.

\[
\begin{align*}
\mathbf{u} &= 0 \\
T_{sl} &= T_{fl} \\
\frac{\partial T_{sl}}{\partial n} &= k_{fl} \frac{\partial T_{fl}}{\partial n}
\end{align*}
\]

Table 2. Boundary conditions for the shorter length of the PCHE

<table>
<thead>
<tr>
<th>Hot channel</th>
<th>Cold channel</th>
</tr>
</thead>
<tbody>
<tr>
<td>(P_{in}) [kPa]</td>
<td>(T_{in}[^\circ C])</td>
</tr>
<tr>
<td>2545.5</td>
<td>279.9</td>
</tr>
</tbody>
</table>

3. RESULTS

In order to evaluate the hydraulic and thermal performance of the conventional and proposed geometries of the heat exchanger, Nusselt number and frictional factor parameters were used. Local values of the Nusselt number were computed using local heat transfer coefficient values, as given by Eq. (7). Eq. (8) defines the local heat transfer coefficient

\[
Nu_{loc} = \frac{h_{loc}D_{hyd}}{k}
\]

\[
h_{loc} = \frac{q_{s}}{T_{w} - T_{nw}}
\]

In the above two equations, subscript ‘loc’ is the local value, and \(D_{hyd}, k, q_{s}, T_{w}, T_{nw}\) are the hydraulic diameter, thermal conductivity, heat flux through the cell face, wall temperature and near wall temperature respectively. The average value of the Nusselt number \(\bar{Nu}\) was computed using Eq. (9), by integrating the local value of the Nusselt number over the whole length of the domain.

\[
\bar{Nu} = \frac{1}{n} \sum_{k=1}^{n} Nu_{k}
\]

Similarly, the value of the friction coefficient was computed using local values of the friction factor and then integrating over the whole length. Values of the local friction factor were computed using Eq. (10) while the average value of the friction factor was computed using Eq. (11).

\[
f_{loc} = \frac{dp}{dz} \cdot \frac{2}{\rho u^{2}} \cdot D_{hyd}
\]

\[
f_{ave} = \frac{1}{n} \sum_{i=1}^{n} f_{loc}
\]

Comparison of current results with the correlations

In order to validate the computational model, the results for the conventional geometry were compared with the existing correlations proposed by Ishizuka et al. and Kim et al. [3], [11], which are as shown in Table 3. Comparisons between the Nusselt numbers for the cold side with correlations listed in Table 3 are shown in Figure 5. It indicates that initially, at lower values of the Reynolds number, current numerical results are closer to those of Kim et al. (i.e. 4% to 6%). Then, at around the mid-range, current numerical results are closer to those of Ishizuka et al. (i.e. 6%). Finally, towards the higher Reynolds numbers, current numerical results bend back towards those of Kim et al. Maximum and minimum differences between the current numerical results and those of Ishizuka et al. are 6% to 10% respectively, while these minimum and maximum differences between the current study and that of Kim et al. are 4% to 8% respectively.

Table 3. Correlations available in the literature

<table>
<thead>
<tr>
<th>Correlation by Ishizuka et al. [3]</th>
<th>Local heat transfer coefficient (h = 0.2104 \times Re + 44.16)</th>
<th>Overall heat transfer coefficient (U = 0.1106 \times Re + 15.943)</th>
<th>Hot side (f = -2.0 \times 10^{-6}Re + 0.0467)</th>
<th>Cold side (f = -2 \times 10^{-6}Re + 0.0467)</th>
<th>Hot side (2400 \leq Re \geq 6000)</th>
<th>Cold side (5000 \leq Re \geq 13000)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Correlation by Kim et al. [11]</td>
<td>(\theta = 10^0) (\theta = 115^0) (\theta = 10^0) (\theta = 10^0)</td>
<td>(\theta = 100^0) (\theta = 115^0) (\theta = 100^0) (\theta = 100^0)</td>
<td>(\theta = 100^0) (\theta = 100^0) (\theta = 100^0) (\theta = 100^0)</td>
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Figure 6 shows a comparison of the Nusselt number with the existing correlations for the zig zag fin for the hot side. Comparison shows that, at lower Reynolds numbers, results are closer to those of Kim et al., and then at higher Reynolds numbers, they are closer to those of Ishizuka et al. The minimum and maximum difference between the current study and results from Ishizuka et al. is 5.6% to 9.2%, whilst the minimum and maximum difference between the current numerical work and that of Kim et al. is 0.54% to 13% respectively.

Comparisons of the friction factor results for the current numerical simulations and published correlations are shown in Figures 7 and 8. Comparison of the friction factor with the existing correlations for the zig zag fin for the cold side are shown in Figure 7. The results reveal that the whole range of Reynolds numbers are close to the results for Ishizuka et al., but deviate from those of Kim et al. Minimum and maximum deviations of the current results from the findings of Ishizuka et al. are 0.11% and 1.21% respectively. This shows that the numerical results are in close agreement with the Ishizuka correlation. On the other hand, the minimum and maximum differences between the current findings and those of Kim et al. are 1.09% and 2.29% respectively. Figure 8 shows a comparison of the friction factor with the existing correlations for the hot zig zag fin. The results indicate that, at lower Reynolds numbers, current numerical results are very close to those of Ishizuka et al., with maximum and minimum deviations of 1.47% and 3% respectively. On the other hand, the findings of Kim et al. are very different from the current numerical results and those of Ishizuka et al. this could be further elaborated from the minimum and maximum difference between the current numerical and Kim et al. are 14.96% to 16.74% respectively. It can be concluded from the above discussion that predictions of the current computational model are within the acceptable ranges and that the model could be used for further simulations.

The current section deals with the comparison of results between the conventional (zig zag channel) and proposed PCHE channel geometries. Figure 9 shows a comparison of Nusselt numbers between the zig zag and proposed fin for the cold channel. It shows that the Nusselt number increases as the value of the Reynolds number increases. The results reveal that the proposed fin geometry performs better thermal prediction. The difference between the Nusselt numbers of the conventional and proposed geometries increases as the value of the Reynolds number increases. An improvement in the value of the Nusselt number was recorded as 6.56%, corresponding to $Re = 5000$, and as 8% corresponding to $Re = 10000$. On the other hand, Figure 10 shows a comparison of the Nusselt number between the zig zag and proposed fin geometries for the hot channel, and it can be deduced from the results that the thermal performance of the proposed geometry is better than
that of the conventional zig zag channel. Improvement in the value of the Nusselt number ranges from 1% to 6.86% as it increases with increasing Reynolds number (i.e. 2500 to 5000).

![Figure 9. Variation of the Nusselt number with the Reynolds number for current numerical simulations compared with conventional geometry (cold side of PCHE)](image9)

![Figure 10. Variation of the Nusselt number with the Reynolds number for current numerical simulations compared with conventional geometry (hot side of PCHE)](image10)

Apart from the Nusselt number, which dictates the thermal performance, the hydraulic performance of the printed circuit heat exchanger geometry is another crucial parameter that must be considered when comparing the performance of two different geometries. In the current work, hydraulic performance is measured by the frictional factor. Figure 11 shows a comparison of the friction factor for the zig zag and proposed fin for the cold channel. The results show that the frictional factor for the proposed geometry decreases significantly. The hydraulic performance of the proposed geometry for the PCHE increases further with increasing Reynolds number. The proposed geometry showed an 18% reduction in friction factor for \( Re = 5000 \), while this reduction became almost 25% at \( Re = 10,000 \). A similar trend is observed for the hot side of the PCHE. The proposed geometry showed a significantly lower frictional factor that reflected a reduced drop in pressure for the proposed geometry. On the hot side, the friction factor for the proposed geometry reduced by 13% at \( Re = 2500 \), and up to 25% when the Reynolds number increased to 5000.

![Figure 11. Variation of the friction factor with the Reynolds number for current numerical simulations compared with conventional geometry (hot side of PCHE)](image11)

![Figure 12. Variation of the friction factor with the Reynolds number for current numerical simulations compared with conventional geometry (hot side of PCHE)](image12)

Local values of the heat transfer coefficient can provide a better insight into the physics of the heat transfer coefficient. To exploit this, values of the local heat transfer coefficient were plotted along the length of the fin pitch. Figures 13 and 14 show local values of the heat transfer coefficient along the fin pitch of the conventional zig zag and proposed channel geometries. It can be seen from Figure 13 that, at the first bend toward the lower wall, flow accelerates and the boundary layer is reinitialized at that point that corresponds to a peak in the local heat transfer values while corresponding to pitch length of 0.5 flow deaccelerate due to reticulations that decreased the local values of heat transfer coefficient. This recirculation region lasts till the normalized pitch length of 0.98, decreasing the local heat transfer coefficient throughout the second half of the fin pitch. This deficiency in the zig zag fin has been removed by designing the fin with a smooth cosine function, so that the total path remains the same and the boundary layer initialization has been done by discontinuous fin. It can be seen in Figure 14 that, instead of one peak in the zig zag fin, there are two peaks in the proposed geometry of the channel for PCHE. No recirculation zone exists on the fin surface, but rather in the wake of the fin, helping to accelerate the flow over the fin toward the mid length.
Figure 13. Fluctuation in the values of the local heat transfer coefficient for the conventional geometry along the fin pitch.

Figure 14. Fluctuation in the values of the local heat transfer coefficient for the proposed geometry along the fin pitch.
4. CONCLUSION

In the current study, the thermal and hydraulic performance of SCO2 PCHE have been evaluated numerically for a new channel geometry based on the cosine function curve. A computational model was validated by comparing the results of the conventional geometry with the existing correlations. The performance of the proposed geometry was compared with that of the conventional zig zag geometry. The following deductions have been made.

- A computational model has been validated using the conventional zig zag channel geometry of the PCHE by comparing the numerical results with published correlations.

- A comparison of Nusselt numbers from the proposed geometry with those of the conventional geometry shows that they increase by up to 7.8% on the cold side and 7.5% on the hot side.

- An increase in the thermal performance of the new geometry is caused by the discontinuous fin structure that allows the boundary layer re-initialization. The proposed cosine structure of the fin also increases the path of the fluid and the area available for heat transfer for the same length of heat exchanger.

- Comparison of friction factor results reveals that the friction factor decreases by up to 24.48% on the cold side and 25% on the hot side. This demonstrates a significant improvement in the hydraulic performance of the PCHE.

- A fin shape based on the cosine function changes the flow directed with smooth path gradient. The proposed fin structure halts the flow recirculation and irreversibility of the fluid that cause high pressure losses in the conventional fin geometry.

REFERENCES


