Performance analysis of printed circuit heat exchanger for supercritical carbon dioxide and water

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Abstract. Printed circuit heat exchanger could be used as pre-cooler in the supercritical carbon dioxide (S-CO₂) Brayton power cycle. Due to the drastic variations of S-CO₂ properties in pre-cooler, segmental design method is used to investigate the thermal-hydraulic performance of PCHE. When the heat duty is given, the local heat transfer entropy generation number has extreme value, which appears near the intersection point of the heat capacity flow rates of both sides. In addition, the local and total entropy generation mainly depend on the local and total heat transfer entropy generation, and the flow imbalance increases entropy generation, especially under conditions of the higher local heat capacity flow rate of water.

1. Introduction

In view of its high efficiency, compactness, and superior economy, the supercritical CO₂ (S-CO₂) Brayton cycle gains lots of attention in the waste heat recovery system, nuclear and concentrated solar [1, 2]. Heat exchanger in the S-CO₂ cycle has an important effect on system efficiency. The printed circuit heat exchanger (PCHE) is one of the leading heat exchanger candidates considering its resistance to extreme conditions, compactness, high effectiveness, and reliability. However, the drastic variations of S-CO₂ properties, especially near the pseudo-critical (PC) point, greatly affect the fluid flow and heat transfer characteristics. So it is quite necessary to study the PCHE performance with S-CO₂.

Lots of the existing papers focused on the channel structure optimization and thermal-hydraulic performance analysis based on the first law of thermodynamics. However, heat transfer is inherently irreversible according to the second law of thermodynamics, which is quite significant in the heat transfer analysis, and Bejan [3] proposed an entropy generation number to optimize the heat exchanger. Sarkar concluded the heat exchanger irreversibilities are significantly more compared with the other parts in the S-CO₂ cycle [4]. For the heat exchanger design, the traditional constant property based heat exchanger design method is inapplicable due to the S-CO₂ properties. Utamura et al. proposed the Generalized Mean Temperature Difference (GMTD) based on the logarithm mean temperature difference (LMTD) [5]. Guo et al. analyzed the low-temperature recuperator using segmental design method [6, 7]. There is little research on the pre-cooler entropy generation analysis in terms of heat exchanger design. In the present study, the water is used for cooling down the S-CO₂ in the pre-cooler device before flowing into compressor, and the pre-cooler is decomposed into a number of elementary
units because of the rapid variations of S-CO\textsubscript{2} properties [8]. The local and whole performance of the pre-cooler is investigated from the viewpoint of heat exchanger design.

2. Theoretical analysis
The S-CO\textsubscript{2} recompressing cycle layout is shown in Figure 1. The fluid is split into two parts before entering the pre-cooler. One fraction of the fluid is cooled in the pre-cooler and the other is compressed by the recompressing compressor [1]. Notably, the splitting of flow may be changed [9], which could lead to changes in S-CO\textsubscript{2} heat capacity rates of the pre-cooler. The S-CO\textsubscript{2} properties change fiercely near the PC point as shown in Figure 2, so it is difficult for the whole heat exchanger design. The LMTD method is inapplicable in the whole heat exchanger design. To capture the influences of S-CO\textsubscript{2} thermo-physical properties, the pre-cooler is discretized into many elementary units whose properties are considered constant as shown in Figure 3 [7]. Each unit is considered as a black box which transforms two input temperatures to two output temperatures. These units are coupled to each other [8], the outlet temperature of hot side and the inlet temperature of cold side in one element is the inlet temperature of hot side and the outlet temperature of cold side in the next element.

For the pre-cooler, straight semi-circular channel, single banking and counter-flow configuration are adopted. The design parameters are outlined in Table 1 and the fixed heat duty is considered. As the total heat duty is constant, the total heat transfer rate is divided equally. The heat transfer rate in elementary units is obtained as:

\[ q_{h,j} = \frac{Q_{tot}}{N} = U_j A_j \Delta T_j \]  

(1)
where \( c_p \) is specific heat; \( m \) is the mass flow rate; \( T \) is thermodynamic temperature; \( Q \) is the total heat duty; \( N \) is the elementary units number; \( U \) is heat transfer coefficient; \( A \) is heat transfer area; \( \Delta T \) is logarithmic mean temperature difference; subscripts \( i \) and \( o \) represent inlet and outlet respectively; subscripts \( h \) and \( c \) indicate the hot and cold fluid respectively; \( j \) represents the local parameter.

Table 1. The initial design parameters.

| Parameters                      | Hot side | Cold side |
|---------------------------------|----------|-----------|
| Inlet pressure (MPa)            | 8.25     | 0.55      |
| Mass flow rate (kg/s)           | 0.40     | 0.86      |
| Inlet temperature (K)           | 373.85   | 292.15    |
| Diameter of channel (mm)        | 1.6      | 1.6       |
| Number of plates                | 24       | 24        |
| Number of channels in each layer| 19       | 23        |
| Number of elementary units      | 50       |           |
| Ideal gas constant (J/kg K)     | 189      |           |

The hot working fluid is in turbulent flow regime. For the straight channel, the Gnielinski correlation [10] was recommended by Hesselgreaves [11]:

\[
N_{u_{h,j}} = \frac{f_{h,j}}{8\left(Re_{h,j}-1000\right)Pr_{h,j}} \quad (2)
\]

\[
f_{h,j} = \left(0.79\log(Re_{h,j}) - 1.64\right)^2 \quad (3)
\]

where \( Nu \) is Nusselt number; \( Pr \) is Prandtl number; \( Re \) is Reynolds number; \( f \) is Darcy friction factor calculated from the Filonenko correlations.

The cold fluid is in the fully developed laminar flow state. Therefore, the semicircular channel Nusselt number and friction factor [11] is obtained as:

\[
N_{u_{c,j}} = 4.089 \quad (4)
\]

\[
f_{c,j} = \frac{15.78}{Re_{c,j}} \times 4 \quad (5)
\]

The S-CO\(_2\) fluid is regarded as an ideal gas and water fluid is regarded as an incompressible liquid. The local entropy generation rate caused by heat transfer [12] is calculated as follows:

\[
S_{g_{r,j}} = m_c c_p \ln \left( \frac{T_{h,i}}{T_{h,j}} \right) + m_c c_p \ln \left( \frac{T_{c,o}}{T_{c,i}} \right) \quad (6)
\]

The local entropy generation rate caused by pressure drop [13] is calculated as follows:

\[
S_{g_{p,j}} = -m_c Rg \ln \left( \frac{P_{h,i}}{P_{h,j}} \right) + \frac{m_c}{\rho_{c,j} T_{c,j}} \Delta P_{c,j} \quad (7)
\]

\( Rg \) is the ideal gas constant for CO\(_2\); \( P \) is the pressure; \( \Delta P \) is the pressure drop; \( \rho \) is the density.

The total entropy generation rate in elementary unit is obtained:

\[
S_{g,j} = S_{g_{r,j}} + S_{g_{p,j}} \quad (8)
\]

The local dimensionless entropy generation rate is calculated as [11]:

\[...\]
3. Results and discussion

The initial design data are outlined in Table 1. The inlet temperature of hot side, the mass flow rate and the inlet temperature of cold side are constant. The total heat duty is 73.6 kW. Figure 3 illustrates the distributions of the local heat capacity flow rate and temperature. When the local heat flux accumulation is less than 30 kW, the heat capacity flow rates of hot side show little change. This is because the heat exchanger inlet is far from the PC point. The heat capacity flow rates of hot side have peak value from $m_h = 0.30$ kg/s to $m_h = 0.40$ kg/s. When the S-CO$_2$ flow rate is larger than 0.40 kg/s, the heat capacity flow rate is increased generally along the passage. The heat capacity flow rate of cold fluid remains almost unchanged. As the S-CO$_2$ flow rate is increased, the intersection point of the heat capacity flow rates of both sides moves to the right along the passage, and $m_h = 0.30$ kg/s has two intersection points. The temperature has curve variations as shown in Fig. 3(b), and the temperature change is greater along the passage as the S-CO$_2$ flow rate is decreased. When the S-CO$_2$ flow rate is larger than 0.30 kg/s, the heat capacity flow rate of hot side is greater than that of cold side and closes to the PC point at the S-CO$_2$ outlet. Therefore, the temperature is decreased slowly.

![Figure 3](image-url)  
**Figure 3.** The relations of (a) local heat capacity flow rate and (b) temperature with the local heat flux accumulation.

![Figure 4](image-url)  
**Figure 4.** The relations of (a) $N_{sT,j}$ and (b) $N_{sP,j}$ with the local heat flux accumulation.
The distributions of the local entropy generation numbers caused by heat transfer and pressure drop are shown in Figure 4. $N_{ST, j}$ is generally decreased and then increased along the passage. The extremums of $N_{ST, j}$ appears near the intersection point of the heat capacity flow rates of both sides. As shown in Figure 4(b), $N_{SP, j}$ has the maximum value, which moves to the right along the passage with the increasing of the S-CO$_2$ flow rate.

To evaluate the whole heat exchanger performance, the total entropy generation numbers caused by heat transfer and pressure drop at different mass flow rates of hot fluid are illustrated in Figure 5. $N_{ST}$ and $N_{SP}$ are increased with the increasing of the S-CO$_2$ flow rate. However, $N_{ST}$ has a larger magnitude than $N_{SP}$. It can be seen from Figure 4 that $N_{ST, j}$ also has a larger magnitude than the $N_{SP, j}$ along the passage. $N_{ST, j}$ and $N_{SP, j}$ have different changing tendencies, the trade-off between the irreversibilities of heat transfer and flow is not easy to obtain [6], reducing the fluid flow irreversibility may be not good for the heat transfer irreversibility, and vice versa [12].

$$N_{ST} = \frac{m_{c, h} c_{ph, j}}{m_{c, pc, j}}$$

$$N_{SP} = \frac{m_{c, h} c_{ph, j}}{m_{c, pc, j}}$$

**Figure 5.** The relations of $N_{ST}$ and $N_{SP}$ with $m_{h}$. 

**Figure 6.** The relations of $N_{c, j}$ with the local heat capacity rate ratio.

It is concluded that the heat transfer entropy generation number is much greater than the frictional entropy generation number from Figure 4. Therefore, the contribution of pressure drop could be neglected. The entropy generation number of the elementary unit is calculated as follows [11]:

$$R_j = \frac{\min \left( m_{c, ph, j}, m_{c, pc, j} \right)}{\max \left( m_{c, ph, j}, m_{c, pc, j} \right)}$$ (11)
\[ \varepsilon_j = \frac{q_j}{\min\left(m_i c_{ph,i}, m_i c_{pc,i}\right) (T_{h,i,j} - T_{o,i,j})} \] (12)

\[
N_{s,j} = \frac{S_{v,j} T_{h,i,j}}{q_j} \left( c_{p,j} m_h > c_{p,i} m_e \right)
\]

\[
= \frac{1}{\varepsilon_j} \left\{ \frac{1}{1 - \left( \frac{T_{h,i,j}}{T_{o,i,j}} \right)} \left[ \log \left[ 1 + \varepsilon_j \left( \frac{T_{h,i,j}}{T_{o,i,j}} - 1 \right) \right] + \frac{1}{R_j} \log \left[ 1 - R_j \varepsilon_j \left( 1 - \frac{1}{\left( \frac{T_{h,i,j}}{T_{o,i,j}} \right)} \right) \right] \right] \right\}^{1/\varepsilon_j} \] (13)

\[
N_{s,j} = \frac{S_{v,j} T_{h,i,j}}{q_j} \left( c_{p,j} m_h \leq c_{p,i} m_e \right)
\]

\[
= \frac{1}{\varepsilon_j} \left\{ \frac{1}{1 - \left( \frac{T_{h,i,j}}{T_{o,i,j}} \right)} \left[ \frac{1}{R_j} \log \left[ 1 + R_j \varepsilon_j \left( \frac{T_{h,i,j}}{T_{o,i,j}} - 1 \right) \right] + \log \left[ 1 - \varepsilon_j \left( 1 - \frac{1}{\left( \frac{T_{h,i,j}}{T_{o,i,j}} \right)} \right) \right] \right] \right\}^{1/\varepsilon_j} \] (14)

Equation (13) and (14) indicate the local entropy generation number is determined by the local effectiveness, the inlet temperature ratio and the heat capacity rate ratio. The relations of local entropy generation number with the local heat capacity rate ratio are illustrated in Figure 6 when the inlet temperature ratio is 1.28. The local entropy generation number is decreased as the effectiveness and heat capacity rate ratio are increased, it is evident imbalance increases entropy generation [11]. As the heat capacity rate of both sides is the same, the local entropy generation number reaches a minimum value. On the basis of the second law analysis, the higher the local heat capacity flow rate of hot fluid is, the better the thermodynamic performance of elementary unit is. In the conventional heat exchanger design, the ratio of heat capacity rate keeps the same in the whole heat exchanger because the working fluid properties are constant. However, due to the extreme changes of S-CO2 properties, the whole PCHE is discretized into a number of elementary units. Many elementary units are in the unbalanced counter-flow case, which could increase the entropy generation. Notably, the local heat transfer entropy generation number is lower when the local heat capacity rate ratio is about 1 under the fixed heat duty conditions as shown in Figure 4(a). The changing trend of local entropy generation number and the heat transfer entropy generation number is similar because the heat transfer entropy generation is much greater than that caused by pressure drop. Therefore, more elementary units appear near the intersection point of the heat capacity flow rates of both sides, the total entropy generation would be less, such as the S-CO2 flow rate is 0.30 kg/s under the constant heat duty condition.

4. Conclusions
The performance analysis of PCHE is conducted in the present work. The local heat transfer entropy generation number has extremums, which is near the intersection point between the heat capacity flow rates of both sides. The local and total heat transfer entropy generation numbers are much greater than the local and total frictional entropy generation numbers. Therefore, the entropy generation contributed by pressure drop could be neglected. In addition, flow imbalance caused by the rapid
variations of S-CO₂ properties increases entropy generation, and the local thermodynamic performance is better when the S-CO₂ flow rate is higher based on second law analysis.

Acknowledgments
Our work was supported by the National Natural Science Foundation of China (No. 51606191), the National Key Research and Development Program-China (2017YFB0601803), and Key deployment project of Chinese Academy of Sciences (Y7220112H1).

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