A comprehensive numerical study of energy efficiency analysis of a double pipe gas cooler based on second law analysis

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A numerical simulation of energy efficiency in commercially available double pipe heat exchangers in the market was investigated based on the second law of thermodynamics in this paper. The effects of CO\textsubscript{2} mass flow rate, water mass flow rate, pressure, CO\textsubscript{2} inlet temperature, and water inlet temperature of the double pipe heat exchanger were considered to evaluate the energy efficiency by analyzing entropy generation, exergy destruction, and entransy dissipation. The changes of the entropy generation, the changes of exergy destruction, and entransy dissipation are similar regardless of the operating conditions. Pressure has the most significant effect on the energy efficiency of the double pipe gas cooler compared to other operating conditions but negligible on the exergy destruction. The pressure, flow rate, and inlet temperature have completely different effects on energy efficiency depending on the region. The entropy generation and entransy dissipation at \(y=0\) m to \(y=0.05\) m (\(y\)-axis is the radial direction) decrease with increasing pressure and the opposite after that. The increase of CO\textsubscript{2} inlet temperature at \(y<0.5\) m is accompanied by an increase of entropy generation, exergy destruction, and entransy dissipation but this situation disappears after \(y=0.5\) m. Entropy generation, exergy destruction, and CO\textsubscript{2} and water mass flow rate are first negatively and then positively correlated with the cut-off point at \(y=0.1\) m.

Key words: entropy generation, exergy destruction, entransy dissipation, supercritical CO\textsubscript{2}

1. Introduction

The use of chlorofluorocarbons (CFCs) as refrigerants in refrigeration systems causes environmental degradation including global warming, ozone depletion, and air pollution due to greenhouse gas emissions, which has led international organizations to rush to find natural refrigerants.
to replace them. The advent of the Montreal Protocol[1] led countries to agree to find new non-polluting refrigerants to replace CFCs and hydrofluorocarbons (HFCs) at the end of the 20th century. A look back at the numerous studies on the properties of CO2[2-6] conducted in the last decade shows that CO2 is one of the most highly regarded natural refrigerants attributing to the fact that it is non-flammable, non-toxic, easily available, and cheap. In the above literature, both Liu et al. [7] and Zhang et al. [3] have used numerical simulation research to demonstrate that the buoyancy caused by the different structure of the heat exchange tube has a great influence on the heat transfer coefficient. Zhang et al., on the other hand, experimented with vertical heating tubes over a wide range of mass flow rates, and a large amount of experimental data led him to a new correlation equation. Joneydi et al. [5] and Wang et al. [6] are investigating trans-critical CO2 systems with the aim of obtaining high COP and reducing costs. More importantly, CO2 as a refrigerant has a high volume cooling capacity and good heat transfer performance[8]. Several heat transfer studies have been done in single tubes[7, 9-12] mainly exploring the secrets of complex flow heat transfer properties of CO2, where keeping wall temperature and heat flux constant are required as boundary conditions. Liu et al. [7] and Yang et al. [10] both studied how to suppress the deterioration of supercritical CO2 heat transfer in a vertical heating tube at constant heat flux and temperature, with the difference that Yang studies a spiral tube, while Zhang et al. [9] studied how to enhance CO2 heat transfer in a horizontal tube at a non-uniform heat flux. Luo et al. [12] reviewed and summarizes the heat transfer characteristics of CO2 in a vertical heating tube.

Most research on heat exchangers has focused partly on automotive air conditioning[13-15] limited by the higher air temperature of secondary flow[16] and partly on printed circuit heat exchangers[17-20] limited by the high cost. Ameur et al. [21] compared the effect of circular and elliptical perforated baffles on plate heat exchanger performance and concluded that elliptical was superior to circular. Menni et al. [22] tried to add vortex generators in the channels to improve heat exchanger performance. Karima et al. [23] designed a new butterfly baffle geometry configuration to improve tubular heat exchanger performance. Han et al. [24] presented a review of the micro-channel heat exchanger development applied in an air-conditioning system. He pointed out the importance of accurately predicting pressure losses and heat transfer characteristics before designing micro-channel heat exchangers. The steep increase and decrease in specific heat and the sudden drop in density cause the heat transfer characteristic of CO2 to be completely different from those of other constant masses. Therefore, a large number of investigations have mainly explored the secrets of complex flow heat transfer properties of CO2.

Liu et al.[2, 9] studied the heat transfer characteristics of supercritical CO2 in tubes with different structures. In the vertical tube, HTC (heat transfer coefficient) gradually increases first, then decreases rapidly, and peaks at the pseudo-critical point. However, in the horizontally oriented HC tube, The buoyancy effect causes a violent oscillation process in HTC, which indicates a weak heat transfer stability. As the inclination angle decreases in the vertical spiral tube, the unevenness of the circumferential heat transfer coefficient becomes more obvious. Jiang et al. [25-27] have done extensive research on the supercritical CO2 heat transfer characteristics in vertical tubes. They pointed out that the heat flux is always positively correlated with the heat transfer coefficient, but then the upward and downward flowing heat transfer coefficients move in the opposite direction when the heat flux continues to increase. Xu et al.[28] experimentally compared the heat transfer in a small serpentine vertical tube and a straight tube. They concluded that the reason why the serpentine tube
has better heat transfer performance than the straight tube due to the centrifugal force of the serpentine tube enhanced the heat transfer. Zhang et al.[29] conducted a numerical simulation investigation on the heat transfer characteristics of supercritical CO2 in horizontal semicircular micro-tubes. It was found that specific heat would be in occupying more percentage in influencing the heat transfer process with an absence of buoyancy. Lei et al.[30] did experiments in order to study the heat exchange situation of supercritical CO2 in a small vertical tube, especially in the case of low mass flux and high heat flux. They found that the buoyancy force in the small channel plays a decisive role in heat transfer under such conditions.

Despite the concept of exergy is of great important and the wide application of CO2 refrigeration cycles which is considered an efficient and environment-friendly system attributed to the complex properties of CO2 near the critical point widely used in automotive air conditioning, food refrigeration, and heat pumps[14]. Cao et al.[31] established a theoretical model to compare the heating performance with IHX and with an absence of IHX (Internal Heat Exchanger). The inlet water temperature of the IHX is the main factor affecting the coefficient of performance (COP). The presence of IHX at higher temperatures such as 313.15 K helps to improve the COP. Considering comprehensively, he suggested that the length of the heat exchanger should not exceed 2.5 m. Cao et al.[32] did an energy analysis with a trans-critical CO2 heat pump. It is the fact that the presence of IHX worsens the optimal discharge pressure. Reducing the ambient temperature and increasing the inlet and outlet water temperatures also can achieve that. The presence of IHX increases the coefficient of performance (COP) by 6.65% and reduces the total power consumption by 6.22%. Fang et al.[33] analyzes the optimal discharge pressure and cycling state parameters with various IHX efficiencies. The irreversibility and cycling of the main components are discussed from an entropy point of view, and the use of IHX increases the COP of CO2 systems by 14.5% to 18.5%.

There is little energy analysis of heat exchangers with specific structures. Mehran et al.[34] made a numerical study to design the optimal conical casing heat exchanger structure. The entropy generation, entropy generation number, heat exchanger reversibility norm (HERN), heat transfer improvement number, and effectiveness-NTU (number of transfers unit) were selected as the main parameters for the energy analysis, and the most suitable geometry was selected, where the efficiency and heat transfer improvement number increased by 55% and 40%, respectively. Hamed et al.[1] found in experimental studies that exergy destruction is positively correlated with mass flow rate, inlet temperature and coil diameter. The largest increase in fire losses occurs in parallel flow configurations. Coil spacing has almost no effect on exergy destruction. The curve direction of the dimensionless exergy destruction (e) differs from that of the exergy destruction (E). The lower the inlet temperature, the higher the hot water flow rate, and the higher the inlet temperature, the lower the cold water flow rate, which can increase the second law efficiency of the heat exchanger. Jafarzad et al.[35] proposed a combined approach applied on the annular side to experimentally study the energy efficiency including the exergy and energy of a vertical double tube heat exchanger. It was shown to indeed improve the performance with that method.

A numerical simulation of energy efficiency in commercially available double pipe heat exchangers in the market was investigated based on the second law of thermodynamics in this paper. The effects of CO2 mass flow rate, water mass flow rate, pressure, CO2 inlet temperature, and water inlet temperature of the double pipe heat exchanger were considered to evaluate the energy efficiency by analyzing entropy generation, exergy destruction, and entransy dissipation.
2. Numerical simulation

2.1. Numerical model

The structure of the double pipe gas cooler was shown in Figure 1 and mesh in Figure 1. The available product in the market consists of three sections in parallel, and one of them is taken as the simulated object to save simulation time. The main dimensions of the double pipe gas cooler are shown in Table 1. The CO₂ inlet is at \( x = 0.1775 \) m and the outlet is at \( x = -0.1775 \) m. Due to the special nature of the double pipe heat exchanger tube as a waist-shaped tube, 10 straight sections of the tube are taken as a reference to study the variation of its parameters along the \( y \)-axis, including the straight sections of the double pipe gas cooler inlet and outlet. CO₂ flows through the shell side of the double pipe gas cooler, and water flows through the tube side, counter-current heat transfer is adopted in the heat exchange process.

Following assumptions were required in the simulation:
1) uniform flow on the tube side.
2) uniform flow at the shell side inlet.
3) heat conduction exists only in the vertical direction of the tube wall.
4) ideal counter-flow between CO₂ and water.
5) the double pipe heat exchangers are placed horizontally.

![Figure 1. Numerical model](image1)

![Figure 2. Grid partition](image2)

<table>
<thead>
<tr>
<th>Table 1. The main dimensions of the double pipe gas cooler</th>
</tr>
</thead>
<tbody>
<tr>
<td>dimension</td>
</tr>
<tr>
<td>gas cooler length</td>
</tr>
<tr>
<td>gas cooler width</td>
</tr>
<tr>
<td>gas cooler height</td>
</tr>
<tr>
<td>inner tube diameter</td>
</tr>
<tr>
<td>outer tube diameter</td>
</tr>
<tr>
<td>tube wall thickness</td>
</tr>
</tbody>
</table>

The global heat transfer processes were simulated using ANSYS fluent 19.0. SST k-\( \omega \) turbulent model[36-39] was selected in this paper. NIST refprop9.11 was used to reference the properties. Mass-
flow inlet and pressure-outlet [40-45] were chosen as the boundary conditions. Table 2 shows the details of the simulated working conditions. Pressure-Velocity Coupling and SIMPLE scheme were selected. As for the residuals, there are all $10^{-3}$ except for the energy which is $10^{-6}$.

Table 2. The initial data for the simulations

<table>
<thead>
<tr>
<th>$m_w$(kg/s)</th>
<th>$m_e$(kg/s)</th>
<th>$T_{r,in}$(K)</th>
<th>$T_{w,in}$(K)</th>
<th>$P$ (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6</td>
<td>0.19</td>
<td>293.15</td>
<td>343.15</td>
<td>7.5</td>
</tr>
<tr>
<td>0.7</td>
<td>0.23</td>
<td>295.15</td>
<td>353.15</td>
<td>8.0</td>
</tr>
<tr>
<td>0.8</td>
<td>0.27</td>
<td>297.15</td>
<td>363.15</td>
<td>8.5</td>
</tr>
<tr>
<td>0.9</td>
<td>0.31</td>
<td>299.15</td>
<td>373.15</td>
<td>9.0</td>
</tr>
<tr>
<td>1.0</td>
<td>0.35</td>
<td>301.15</td>
<td>378.15</td>
<td>10.0</td>
</tr>
<tr>
<td>1.1</td>
<td>0.39</td>
<td>303.15</td>
<td>383.15</td>
<td>11.0</td>
</tr>
<tr>
<td>1.2</td>
<td>0.58</td>
<td>305.15</td>
<td>388.15</td>
<td>12.0</td>
</tr>
</tbody>
</table>

2.2. Governing equations

A steady-state method was used in the simulation. The equation for conservation of mass, or continuity equation, can be written as Eq. 1.

$$\frac{\partial (\rho u_j)}{\partial x_j} = 0 \tag{1}$$

Conservation of momentum is described by Eq. 2.

$$\frac{\partial}{\partial x_j} (\rho u_j u_j) = \frac{\partial}{\partial x_j} \left( \mu_{eff} \frac{\partial u_j}{\partial x_j} + \frac{2}{3} \mu_{eff} \frac{\partial u_k}{\partial x_k} \right) - \frac{\partial p}{\partial x_j} + \rho g_j \tag{2}$$

where $\mu_{eff}$ is the effective viscosity which is calculated as Eq. 3.

$$\mu_{eff} = \mu + \mu_t \tag{3}$$

where $\mu_t$ is the turbulent viscosity defined according to the turbulence model.

Conservation of momentum is described by Eq. 4.

$$\frac{\partial}{\partial x_j} (\rho u_j e_p T) = \frac{\partial}{\partial x_j} \left[ \mu_{eff} \left( \frac{\partial u_j}{\partial x_j} + \frac{\partial u_k}{\partial x_k} \right) - \frac{2}{3} \mu_{eff} \frac{\partial u_k}{\partial x_k} \right] + \frac{\partial}{\partial x_j} \left( k_{eff} \frac{\partial T}{\partial x_i} \right) \tag{4}$$

where $k_{eff}$ is the effective conductivity which is calculated as Eq. 5.

$$k_{eff} = k + k_t \tag{5}$$

where $k_t$ is the turbulent thermal conductivity defined according to the turbulence model.

$k$ - equation and $\omega$- equation are described by Eq. 6 and Eq. 7.

$$\frac{\partial (\rho u_j k)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + P_k - D_k + S_k \tag{6}$$

$$\frac{\partial (\rho u_j \omega)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_\omega} \frac{\partial \omega}{\partial x_j} \right) + C_\omega + P_\omega - D_\omega + S_\omega \tag{7}$$

2.3. Data reduction

In the thermal system, high-grade energy is partially converted into low-grade energy due to irreversible factors such as the dissipation effect, and entropy generation is produced. The more irreversible factors and the greater the degree of irreversibility, the more entropy generation is produced. Irreversible factors are, to a certain extent, a manifestation of the degree of disorder in a system, so entropy generation is a large amount of disorder. The entropy generation is the product of
the irreversibility of heat and friction. For any simulated pipe section j, the entropy generation ignoring thermal friction can be calculated by Eq. 8[46]

\[ S_{Rj} = \frac{Q_j}{T_{w,j}} \frac{Q_j}{T_{r,j}} \]  

where \( T_{w,j} \) and \( T_{r,j} \) were the bulk temperature of water in section j and the bulk temperature of CO2 in section j along the axial line, respectively, \( Q_j \) was heat exchange in section j which were calculated by Equation 9[47].

\[ Q_j = c_{p,w} \dot{m}_w (T_{w,\text{out},j} - T_{w,\text{in},j}) \]  

where \( c_{p,w} \) and \( \dot{m}_w \) were the specific heat of water and mass flow rate of water, respectively, \( T_{w,\text{out},j} \) and \( T_{w,\text{in},j} \) were the outlet and inlet temperature of water in section j.

When the system changes reversibly from any state to a state in equilibrium with the environment, the maximum amount that can theoretically be converted into useable work is called exergy. For a heat exchanger or a pipe section, the difference between input and recovery is the exergy destruction which was calculated by Equation 10

\[ E_k = (h_{r,\text{in},j} - h_{r,\text{out},j}) - T_0(S_{r,\text{in},j} - S_{r,\text{out},j}) \]  

where \( h_{r,\text{in},j} \) and \( h_{r,\text{out},j} \) were the import and export enthalpy of CO2 in section j, \( S_{r,\text{in},j} \) and \( S_{r,\text{out},j} \) were the import and export entropy of CO2 in section j, \( T_0 \) was the ambient temperature, take a constant 298.15 K.

Entransy dissipation was calculated by Equation 11

\[ E_h = \int_0^T Q \, dT \]  

2.4. Reliability verification

2.4.1 Model validation

Since experiments on supercritical CO2 double-pipe heat exchangers have been done previously, this paper directly used the previously obtained experimental data to verify the validity of the simulation. The specific equipment of the experiments and the working conditions can be found in Ref.[48]. Figure 1 showed a comparison of the numerical and experimental results for different pressures. The SST k - \( \omega \) turbulent model is proved to be accurate and reliable in the heat transfer prediction.

Figure 3. Comparison of numerical calculation results with experimental results
2.4.1.1 Grid independence

Table 3 shows the pressure drop for different grid numbers. It can be seen that the pressure drop in the double pipe heat exchanger is no longer significant for a grid partition of 2.69E+6. Therefore, a grid with the numbers 2.69E+6 was selected to save computational time.

<table>
<thead>
<tr>
<th>Number of meshes</th>
<th>Pressure drop</th>
<th>Error(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.28E+6</td>
<td>1721.6</td>
<td>1.446845</td>
</tr>
<tr>
<td>1.63E+6</td>
<td>1283.1</td>
<td>0.823621</td>
</tr>
<tr>
<td>1.86E+6</td>
<td>1043.9</td>
<td>0.483655</td>
</tr>
<tr>
<td>2.21E+6</td>
<td>804.5</td>
<td>0.143405</td>
</tr>
<tr>
<td>2.69E+6</td>
<td>730</td>
<td>0.037521</td>
</tr>
<tr>
<td>2.95E+6</td>
<td>718.3</td>
<td>0.020893</td>
</tr>
<tr>
<td>4.13E+6</td>
<td>703.6</td>
<td>0</td>
</tr>
</tbody>
</table>

3. Results and discussions

3.1. Effects of CO₂ mass flow rate

Figure shows the distribution of entropy generation, exergy destruction, and entransy dissipation for different CO₂ flow rates in the y-axis direction. CO₂ mass flow rate changed from 0.19 kg/s to 0.58 kg/s, while cooling water mass flow rate remained at 0.6 kg/s. In addition, the inlet temperature \((T_{\text{in}} = 343.15 \text{ K}, T_{\text{w,in}} = 299.15 \text{ K})\) is kept at a constant value. Both entropy generation and exergy destruction show a gradual decrease after CO₂ enters the heat exchanger tube in the radial direction (y-axis direction), and gradually decreases with increasing CO₂ mass flow rate in the inlet pipe section from \(y=0 \text{ m} \) to \(y=0.1 \text{ m} \) as shown in Figure (a) and (b). The reason is that the pressure caused by the specific frictional resistance gradually decreases along the length of the tube, resulting in a decrease in the potential difference for irreversible losses, which leads to a gradual decrease in entropy generation. CO₂ is gradually cooled along the length of the tube, resulting in a gradual decrease in the amount of energy that can be converted into usable work, so that the exergy destruction gradually decreases at the same time. When the CO₂ mass flow rate increases, the CO₂ bulk temperature decreases, and the heat transfer temperature difference between CO₂ and cooling water decreases, resulting in a lower imbalance potential and a lower entropy generation, while the energy that can be converted into usable work decreases with the lower CO₂ bulk temperature, so the exergy destruction also decreases. The high heat exchange in the inlet section leads to the high entransy dissipation at the inlet, and then the heat exchange decreases, and the fire entransy dissipation decreases gradually as can be seen in Figure (c). In fact, it can be seen from Figure that the increase in CO₂ mass flow rate has slight effect on the exergy destruction and entransy dissipation, after \(y=0.1 \text{ m} \) the entropy generation increases with increasing CO₂ mass flow rate.
3.2. Effects of water mass flow rate

Figure shows the distribution of entropy generation, exergy destruction, and entransy dissipation for different cooling water flow rates in the y-axis direction. Cooling water mass flow rate changed from 0.6 kg/s to 1.2 kg/s, while the CO2 mass flow rate remained at 0.58 kg/s. The same inlet temperature ($T_{\text{in}} = 343.15$ K, $T_{\text{in}} = 299.15$ K) is kept at a constant value. The entropy generation, exergy destruction, and entransy dissipation all decrease gradually in the y-axis direction as can be seen in Figure , which is due to the gradual decrease of pressure along the tube length and the temperature drop of CO2, and the gradual decrease of heat exchange in each section of the tube length, resulting in the decrease of unbalanced potential difference for irreversible losses, the decrease of energy that can be converted into usable work, and the decrease of "heat transfer capacity" losses. Entropy generation, exergy destruction, and entransy dissipation with the cooling water flow rate do not change significantly, which indicates that the increase in cooling water flow rate with the heat exchanger energy consumption is not significant, theoretically changing the mass flow rate of cooling water on the heat exchanger design guidance is not significant.

3.3. Effects of pressure

Figure shows the distribution of entropy generation, exergy destruction, and entransy dissipation for different pressure in the y-axis direction. The pressure changed from 7.5 MPa to 12 MPa, CO2 mass flow rate remained at 0.58 kg/s and the cooling water mass flow rate remained at 0.6
kg/s. The same inlet temperature \((T_{r,\text{in}}= 343.15 \text{ K}, T_{w,\text{in}}= 299.15 \text{ K})\) is kept at a constant value. The entropy generation, exergy destruction, and entransy dissipation decrease gradually along the y-axis at all pressures as shown in Figure. It is clear that the reduction of entropy generation and entransy dissipation at different pressures is significantly moderated when comparing different CO2 and water flow rates. The entropy generation and entransy dissipation at \(y=0 \text{ m to } y=0.05 \text{ m}\) decrease with increasing pressure and the opposite after that. The higher the pressure, the greater the potential difference causing irreversible losses. The greater the potential difference of irreversible loss, so the more entropy generation, when the pressure on the high pressure side is larger, the greater the temperature slip on the high pressure side, more heat can be released, so the more heat that can be converted into usable work, so the exergy destruction increases. This phenomenon is largely related to the complex physical properties of CO2 near the critical point, as the temperature of CO2 changes, the sudden change in specific heat causes a change in heat exchange resulting in a sudden change in entropy generation and entransy dissipation. By comparing the 7 pressure conditions, it is found that the higher the pressure, the higher the potential difference causing irreversible losses, so the higher entropy generation which is clearly shown in Figure (a). Changes in pressure have minimal effect on the exergy destruction.

![Figure 6](image-url)

**Figure 6. Effect of \(P\) on (a) entropy generation and (b) exergy destruction and (c) entransy dissipation**

### 3.4. Effects of CO2 inlet temperature

In order to clarify the effect of CO2 inlet temperature on the energy efficiency of the heat exchanger, the CO2 inlet temperature was varied from 343.15 K to 388.15 K and the cold water temperature was kept at 299.15 K. Meanwhile, the mass flow rates of cold water and CO2 were kept at 0.6 kg/s and 0.58 kg/s, respectively. Figure shows the distribution of entropy generation, exergy destruction, and entransy dissipation for different CO2 inlet temperatures in the y-axis direction. It is obvious that the trend of the entropy generation, exergy destruction are the same as entransy dissipation, all decreasing along the y-axis direction, and the increase of the inlet temperature between \(y=0\) and \(y=0.5\text{ m}\) leads to the increase of entropy generation, exergy destruction, and entransy dissipation but this situation disappears after \(y=0.5\text{ m}\). The high heat transfer temperature difference at the inlet is the reason for the existence of the above situation, and as the CO2 temperature decreases and the cooling water temperature increases, the temperature difference between them decreases, which naturally makes this trend disappear.
3.5. Effects of water inlet temperature

To investigate the effect of water inlet temperature on the energy efficiency of the heat exchanger, the water inlet temperature was varied from 299.15 K to 305.15 K and CO2 inlet temperature was kept at 343.15 K. Meanwhile, the mass flow rates of cold water and CO2 were kept at 0.6 kg/s and 0.58 kg/s, respectively. Figure shows the distribution of entropy generation, exergy destruction, and entransy dissipation for different water inlet temperatures in the y-axis direction. As shown in Figure, entropy generation, exergy destruction, and entransy dissipation show a strict decreasing trend along the y-axis direction. The entrance section is much higher than the other sections, in other words, the increase in temperature difference between the cold water inlet and the hot water inlet leads to an increase in entropy production.

4. Conclusion

In this study, numerical simulations based on the second law of thermodynamics were performed on heat exchanger products that already exist in the market. The effects of CO2 mass flow rate, water mass flow rate, pressure, CO2 inlet temperature, and water inlet temperature on the heat transfer effectiveness of the heat exchanger were considered to evaluate the energy efficiency of the heat exchanger. The indexes entropy generation, exergy destruction, and entransy dissipation for evaluating the energy efficiency of heat exchangers are analyzed in detail in this paper. The following conclusions were drawn from this study.
The changes of the entropy generation, the changes of exergy destruction, and entransy dissipation are similar regardless of the operating conditions (CO2 mass flow rate, water mass flow rate, pressure, CO2 inlet temperature, and water inlet temperature), and the similarity is that they are decreasing in the direction of temperature decrease (y-axis direction).

Compared to other operating conditions, it is clear that pressure has the most significant effect on the energy efficiency of the heat exchanger. But the effect of pressure on the exergy destruction is negligible.

The effect of pressure on the energy efficiency of the heat exchanger differs before and after y=0.05 m. The entropy generation and entransy dissipation at y=0 m to y=0.05 m decrease with increasing pressure and the opposite after that.

The effect of CO2 inlet temperature on the energy efficiency of the heat exchanger differs before and after y=0.5 m. The increase of the inlet temperature between y=0 and y=0.5 m leads to the increase of entropy generation, exergy destruction, and entransy dissipation but this situation disappears after y=0.5 m.

Both entropy generation and exergy destruction show a gradual decrease after CO2 enters the heat exchanger tube in the radial direction (y-axis direction), and gradually decreases with increasing CO2 and water mass flow rate in the inlet pipe section from y=0 m to y=0.1 m, but after that, the situation was completely opposite, entropy generation, exergy destruction, and CO2 and water mass flow rate are positively correlated.

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