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EXPERIMENTAL AND NUMERICAL STUDY ON THERMAL HYDRAULIC PERFORMANCE OF TRAPEZOIDAL PRINTED CIRCUIT HEAT EXCHANGER FOR SUPERCRITICAL CO_2 BRAYTON CYCLE

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ABSTRACT

The supercritical carbon dioxide (sCO_2) Brayton cycle is the preferred power cycle for future nuclear energy, fossil energy, solar energy, and other energy systems. As the preferred regenerator in the cycle, the printed circuit heat exchanger (PCHE) exhibits a high heat transfer efficiency, compactness, and robustness. The structure design of its internal flow channel is one of the most important factors to enhance the heat transfer and reduce pressure loss. In the present work, trapezoidal channel structure is developed and its thermal–hydraulic performances are compared with the straight, the S-shape, and the zigzag structures. Further, a trapezoidal PCHE prototype is manufactured and experimentally studied as a regenerator in the sCO_2 test loop. The overall heat transfer coefficient exceeds 1.10 kW/(m^2·K) and reaches a maximum of 2.53 kW/(m^2·K) with the changes in the inlet temperature, the working pressure, and the mass flow rate. Correlations of the Nusselt numbers are proposed on both sides, with the Reynolds numbers ranging from 10,000 to 30,000 and 4800 to 14,000, and the Prandtl numbers ranging from 0.91 to 1.61 and 0.77 to 0.98 on the cold side and hot side, respectively. The pressure drop of the channels calculated by the peeling method using a single-plate straight prototype is less than 7 kPa and 15 kPa on the hot and the cold side, respectively. The heat recovery efficiency is analyzed to evaluate the performance of the PCHE used as a regenerator. Finally, simulation works are carried out to verify the experimental results and expand the Reynolds numbers ranging from 3796 to 30,000 and 1821 to 14,000, on the cold side and hot side, respectively. This work provides the test methods and experimental correlations for the development of an efficient PCHE in the sCO_2 Brayton cycle.

INTRODUCTION

Printed circuit heat exchanger (PCHE) is a preferred heat-exchange device in supercritical carbon dioxide (sCO_2) Brayton cycle because of its high heat transfer efficiency, compactness, and robustness. The flow microchannels, formed by chemical etching on the metal plates, have a decisive influence on the thermal-hydraulic performance of PCHE. Many kinds of channel structures have been studied such as straight, zigzag, S-shaped fin, and airfoil fin. The straight channel is the basic structure with a simple etching process. Mylavarapu et al. [1] fabricated two straight-channel PCHEs and connected them in series to a high-temperature helium test facility (HTHF). The heat transfer and friction characteristics were analyzed based on the experimental data under the conditions corresponding to the laminar to turbulent transition region. Chen et al. [2] developed a numerical dynamic model and successfully predicted the steady-state and transient behaviors of a straight PCHE by comparing with the experimental results. Chu et al. [3] studied the PCHE thermo-hydraulic performance on the sCO_2-water experiment platform at the transcritical and supercritical states, indicating that the comprehensive performance reduced by about 17.6% at the transcritical state. The zigzag structure can significantly improve the heat transfer area and coefficient, accompanied by the

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disadvantage of increased pressure loss. Nikitin et al. [4] investigated the heat transfer performance and pressure drop of a zigzag PCHE through experiments and numerical simulations. The overall heat transfer coefficient ranged from 300 to 650 W/(m^2·K) with a compactness of approximately 1050 m2/m3 and a maximum power density of 4.4 MW/m^3. Kim et al. [5–7] carried out a detailed study on the zigzag-structure PCHE using He, CO_2, and water as working fluids. The correlations of Nusselt numbers and Fanning friction factors were fitted, and the effects of the channel geometric parameters were analyzed. Zhou et al. [8] designed and manufactured a 100 kW class zigzag PCHE prototype as a recuperator and tested using sCO_2 on both sides. The effectiveness was over 95% and the pressure drop was less than 50 kPa on both sides. S-shaped structure was proposed by Tsuzuki et al. [9], which can reach the same thermal performance as the zigzag flow channel, but its pressure loss reduction can be reduced to one - fifth. Ngo et al. [10,11] developed a new S-shaped-fin PCHE and compared its thermal-hydraulic performance with that of zigzag fins. The empirical correlations of Nusselt numbers and pressure-drop factors were proposed, which proved that the pressure drop factor of the S-shaped microchannels was 4–5 times less than the zigzag one through a 24–34% reduction in the Nusselt numbers. Kim et al. [12] proposed an airfoil fin structure which may have a smaller pressure drop than S-shaped. Chen et al. [13] compared the performance of four types of NACA 00XX airfoil structures with zigzag and found that the airfoil structure can significantly reduce the flow pressure drop loss while maintaining heat transfer performance. Pidaparti et al. [14] investigated two kinds of discontinuous PCHEs with an offset rectangular and NACA0020 airfoil fin. Empirical correlations for the friction factor and the Nusselt number were proposed, which could match the experimental results.

From the above studies, it can be found that balancing heat transfer and pressure drop and developing flow channel structures with higher comprehensive performance have always been the focus of PCHE research. Especially in regenerators with the largest heat load, the comprehensive performance of PCHE will significantly affect the efficiency of sCO_2 cycle system. However, most existing experiments focused on precoolers, which studied the heat transfer characteristics between sCO_2 and water, and the conditions for testing were usually near the CO_2 critical point. It also needs to study the performance of PCHE regenerators by using sCO_2 as the heat transfer medium on both sides. In this work, a trapezoidal channel structure model is developed and compared with the straight, the S-shape, and the zigzag structures. Then a trapezoidal PCHE prototype is manufactured and experimentally studied as a regenerator in the sCO_2 test loop. The heat transfer coefficient and heat recovery efficiency are calculated and analyzed under different thermal parameters. Correlations for Nusselt numbers on both trapezoidal channels are proposed with respect to Reynolds numbers and Prandtl numbers, and the pressure drop in the flow channels is evaluated and peeled off by designing a single-plate straight channel prototype. In addition, the numerical simulation results verify and expand the experimental conclusions. This work provides new trapezoidal channel experimental results and heat transfer correlations for an advanced PCHE regenerator design in the sCO_2 Brayton cycle.

TRAPEZOIDAL CHANNEL STRUCTURE

A counterflow trapezoidal channel PCHE heat transfer unit model is established for numerical simulation study as shown in Figure 1, the channels had semicircular cross - sections with 2 mm diameters and a 0.50 mm space in between. The period and amplitude of the trapezoid are 10 mm and 1 mm, respectively. Further, to compare the performance with other structures, straight, zigzag and S-shaped channels are established. The four models only have differences in geometric shapes. The unstructured tetrahedral mesh is divided and the SST k-omega turbulence model is chosen due to the Reynolds number range and flow bending [15-17]. Periodic boundary conditions are applied on the top, the bottom, the left, and the right sides, and adiabatic boundary conditions are applied on the front and the back surfaces. The inlet temperature and pressure boundary conditions on the hot and cold channels are 726.85 K, 7.6 MPa and 388.75 K, 20.2 MPa, respectively. The inlet mass flow rate range on both sides is from 4.82×10^{-4} to 14.45×10^{-3} kg/s. All conditions are selected based on the regenerator parameters of the 200-kW class sCO_2 simple recuperation cycle demonstration system.

Table 1 summarizes the heat transfer coefficient and pressure drop results by simulations. The trapezoidal structure has the highest heat transfer coefficient over 5000 W/(m^2·K). The hot side and cold side are 47.84% and 52.31% higher than that of the straight channel, 13.49% and 17.10% higher than that of the S-shaped, and 7.69% and 10.84% higher than that of the zigzag channel, respectively. However, the pressure drop of the trapezoid is also the highest in the four structures, especially on the hot side, which pulls down the comprehensive thermal-hydraulic performance.

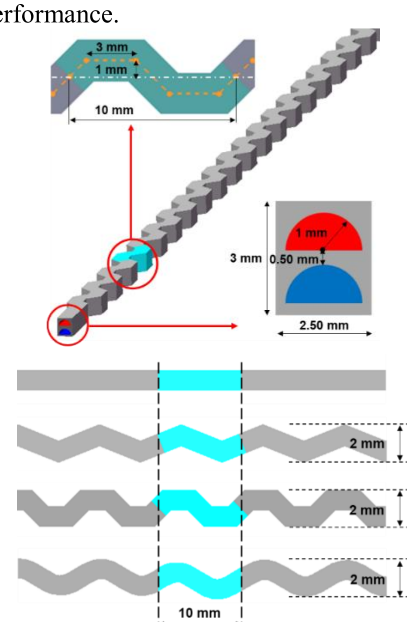

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Figure 1: Channel structure models for numerical simulation

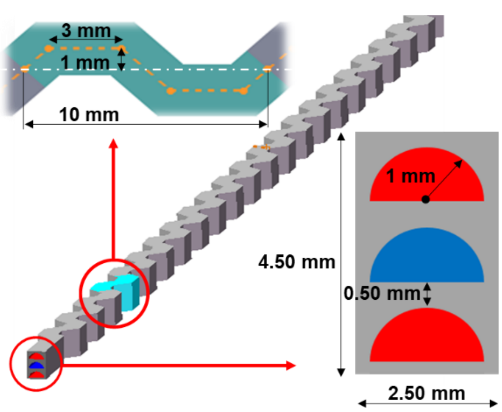
To reduce the pressure loss problem, a sandwiched trapezoidal flow channel structure is designed as shown in Figure 2 (a), where the cross-section of the flow unit model is increased to 2.50 mm × 4.50 mm with one cold flow channel sandwiched between two hot flow channels. Table 2 shows that the significant reductions in pressure drop loss in the hot channel are obtained, where the values are 75.4% (from 154.81 kPa to 38.37 kPa) and 74.7% (from 39.65 kPa to 10.05 kPa) at 42 kPa and 11 kPa pressure loss cases in the cold side, respectively. Figure 2(b) also indicates the heat recovery efficiency is increased by about 5% in the sandwich structure.

Table 1: Simulation results of four channel structures

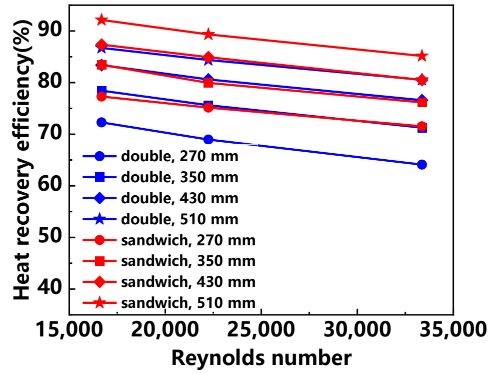
Channel structure	Heat transfer coefficient $[W/(m^2 \cdot K)]$		Pressure drop (kPa)		Heat recovery efficiency (%)
	Hot side	Cold side	Hot side	Cold side	
Straight	3402.20	3302.66	26.32	7.85	52.12
Zigzag	4670.64	4538.54	104.57	30.11	60.93
S-shaped	4431.90	4295.62	56.78	17.10	59.67
Trapezoid	5029.80	5030.37	159.30	45.80	64.09

Table 2: Pressure loss comparisons between sandwich and double-channel structure

Reynolds Number of the Cold Side	Pressure Loss (kPa).			
	Double-Channel Structure		Sandwich Structure	
	Cold Side	Hot Side	Cold Side	Hot Side
33,366	42.62	154.81	41.94	38.37
22,244	19.73	71.11	18.60	17.77
16,683	11.07	39.65	11.04	10.05



(a) sandwich trapezoidal channel model



(b) heat recovery efficiency comparison

Figure 2: Sandwich trapezoidal channel structure and comparison of heat recovery efficiency

EXPERIMENTAL SYSTEM AND PROTOTYPE

A supercritical carbon dioxide heat transfer and circulation test loop was constructed to investigate the heat transfer and pressure drop characteristics, as shown in Figure 3. The loop can be roughly divided into four parts, namely the CO_2 gas source and pump, the cooling system, the PCHE test part, and the heat transfer test section. It also includes a pulsation damper, a mass flowmeter, a filter, various valves, thermocouples, pressure/differential pressure sensors, etc. The maximum temperature of the loop can reach 500 ℃, the working pressure can be adjusted within 7 ~ 15 MPa, and the maximum mass flow rate is 60 kg/h.

The test trapezoidal PCHE prototype is shown in Figure 5, which is the main research object in this work. There are 20 channels on one plate, each of which has a length of 120 mm, including ten periods and two 10 mm long straight channel zones at both ends, and a 4 mm interval is left between two adjacent channels. The prototype has two hot plates and one cold plate, resulting in the mass flow of the hot channel being half of the cold side in each heat transfer unit to balance the flow velocity and heat capacity. Using diffusion bonding, the plates are combined into a 168 mm × 90 mm × 10.50 mm device with a 120 mm × 80 mm × 4.50 mm heat transfer core. The total heat transfer area is about 0.36 m^2 with a 0.30 m^2 area in the heat transfer core. Four 90 mm long pipes are welded on the top of the prototype and connected to the test loop by tube fittings.

The experimental test conditions are shown in Table 3, and the direct measuring instruments and accuracy are shown in Table 4. In order to evaluate the pressure drop of trapezoidal channel, as shown in Figure 4, a straight channel PCHE prototype with only one layer of plates was designed and manufactured for pressure drop peeling, which has the same structure and size as trapezoidal channel. In the pressure drop peeling experiment, the temperature and pressure of the straight channel PCHE prototype were controlled in accordance with the conditions of the trapezoidal one. Since the pressure drop of the straight channel PCHE was found to be very small, the pressure

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loss of the trapezoidal channel core could be considered equivalent to the difference between the measurement results of trapezoidal channel PCHE and the straight channel PCHE.

Table 3: The experimental test conditions

Parameters	Hot side	Cold side
Inlet temperature, °C	200 – 400	40 – 100
Inlet pressure, MPa	7.50 - 12	8.25 – 12.75
Mass flow rate, kg/h	20 - 60	20 - 60
Reynolds number range	4,800 – 14,000	10,000 – 30,000

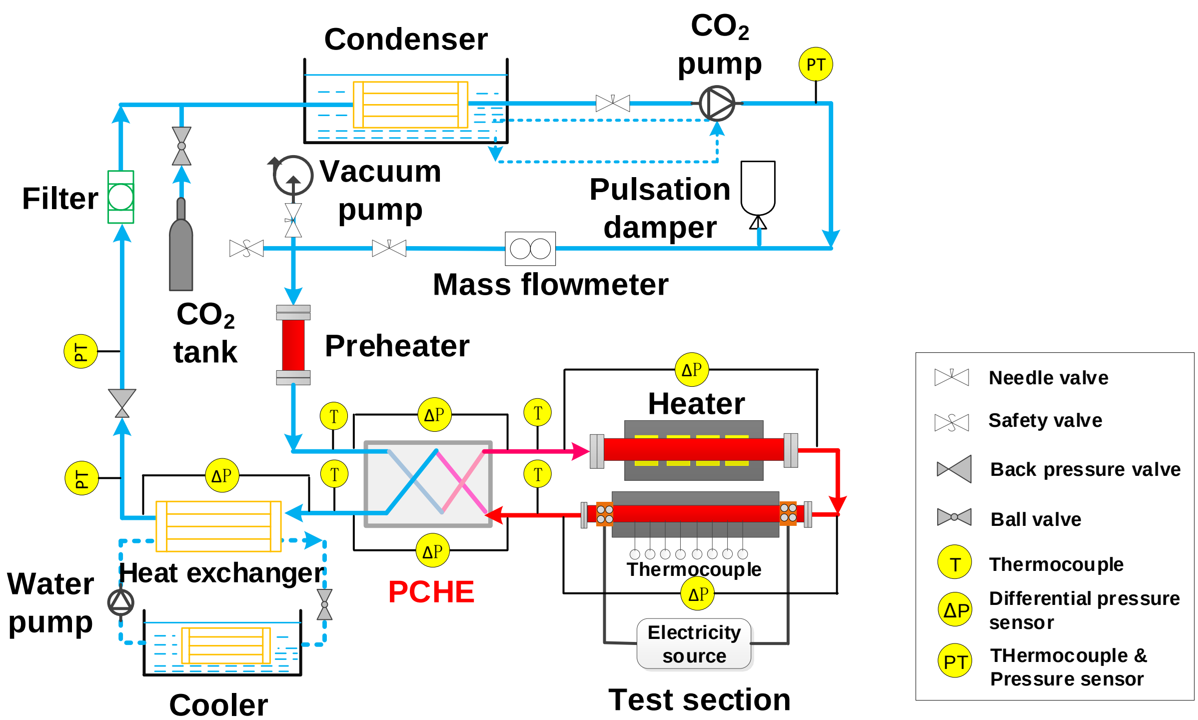

Figure 3: Schematic of the sCO_2 heat transfer and circulation test loop

Table 4: The direct measuring instruments and accuracy

Parameters	Position	Instruments	Range	Accuracy
Mass flow rate	Pump outlet	Coriolis flowmeter	0 ~ 300 kg/h	0.20%
Temperature	Preheat outlet	PT100	-50 ~ 200 °C	A level
Temperature	PCHE cold inlet	K-type thermocouple	0 ~ 1100 °C	0.75%
Temperature	PCHE cold outlet	K-type thermocouple	0 ~1100 °C	0.75%
Temperature	PCHE hot inlet	K-type thermocouple	0 ~1100 °C	0.75%
Temperature	PCHE hot outlet	K-type thermocouple	0 ~1100 °C	0.75%
Pressure loss	PCHE cold side	Differential pressure sensor	0 ~ 500 kPa	0.10%
Pressure loss	PCHE hot side	Differential pressure sensor	0 ~ 500 kPa	0.10%

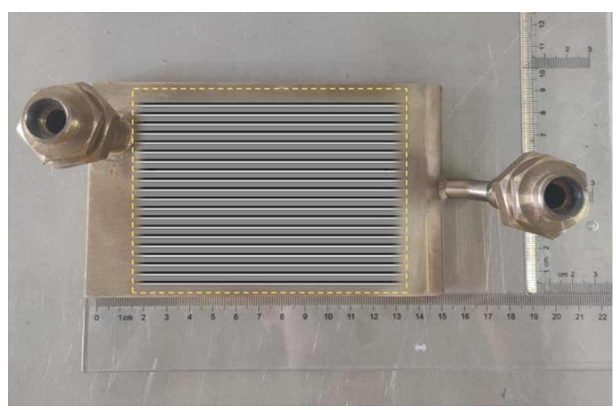

Figure 4: The straight channel PCHE prototype for pressure drop peeling

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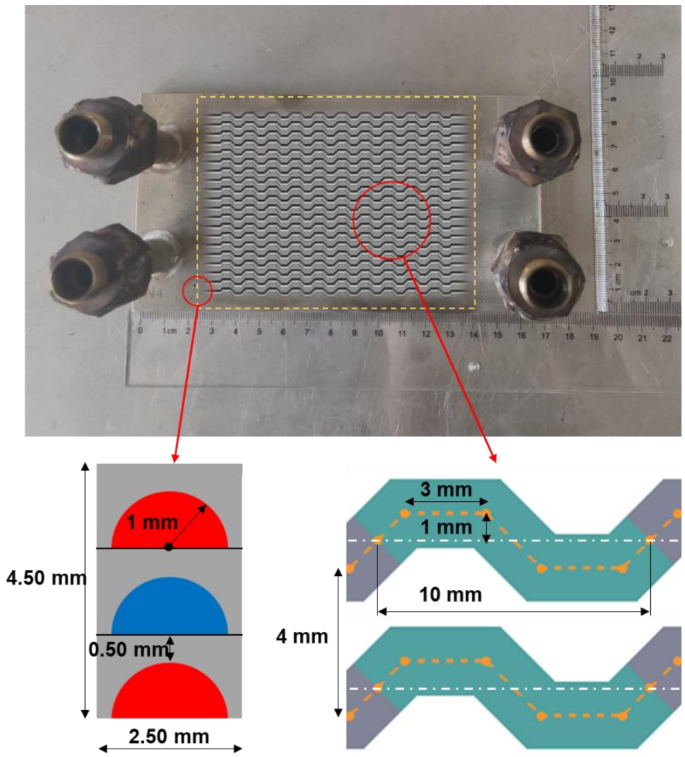

Figure 5: The test trapezoidal PCHE prototype and its channel geometric parameters

RESULTS AND DISCUSSION

Firstly, the overall heat transfer coefficient is introduced to evaluate the performance of the PCHE prototype. The effects of the inlet temperature on both sides, the working pressure, and the mass flow rate on the cold overall heat transfer coefficient are shown in Figure 6. It can be seen that the U_c exceeds 1.10 kW/(m2·K) and reaches a maximum of 2.53 kW/(m2·K) with the changes of various conditions.

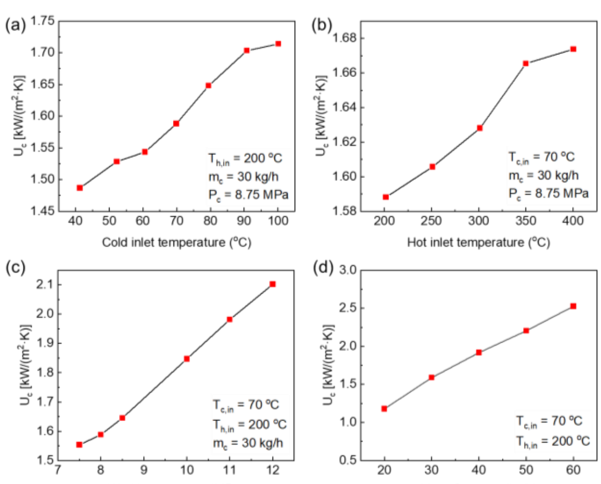

Figure 6: The effects of (a) cold inlet temperature, (b) hot inlet temperature, (c) working pressure, and (d) mass flow rate on the cold overall heat transfer coefficient

Then the average convective heat transfer coefficient is used to analyze the heat transfer characteristics of sCO_2 on each side. And dimensionless Nusselt numbers are calculated and fitted the correlations as Equations (1) and (2) with the average Reynolds numbers and Prandtl numbers on the cold and the hot sides, respectively.

On the cold side:

$$Nu_c = 0.8937 Re_c^{0.5176} Pr_c^{0.1106} \quad (1)$$

$$\begin{bmatrix} 10{,}000 \le Re_c \le 30{,}000 \\ 0.91 \le Pr_c \le 1.61 \end{bmatrix}$$

On the hot side:

$$Nu_h = 0.1817 Re_h^{0.6741} Pr_h^{0.6980} \quad (2)$$

$$\begin{bmatrix} 4{,}800 \le Re_h \le 14{,}000 \\ 0.77 \le Pr_h \le 0.98 \end{bmatrix}$$

Figure 7 reflects the difference between the correlations and the experimental results. All the correlations values are within 15% deviation with the experimental results, and 92% and 86% values are within 10% deviation on the cold and the hot sides, respectively.

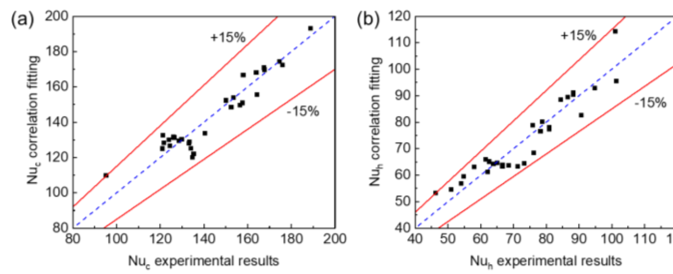

Figure 7: The difference between the correlations and the experimental results on (a) the cold side and (b) the hot side

Finally, the heat recovery efficiency and pressure drop are discussed in Figure 8 under different conditions in Table 3. The heat recovery efficiency is generally low at about 60% level, the maximum value is only 63.10%, mainly because the size of the prototype is small and the heat transfer area is limited. The pressure drop on the cold side is generally higher than that on the hot side. It is greatly affected by the working pressure and mass flow rate, but little changes with the inlet temperature of hot and cold sides. The cold pressure drop increases significantly when the cold inlet temperature rises from 40 ℃ to 50 ℃ due to drastic changes in physical properties of sCO_2. Besides, it can be concluded increasing the working pressure is helpful to reduce the pressure loss and improve the cycle performance.

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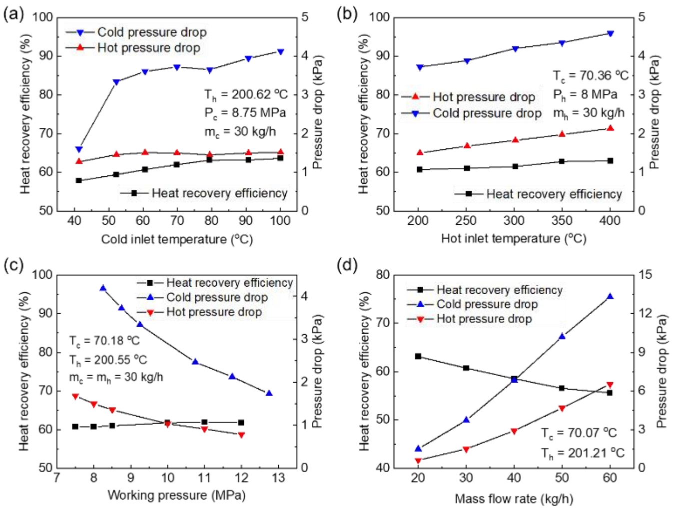

Figure 8: The effects of (a) the cold inlet temperature, (b) the hot inlet temperature, (c) the working pressure and (d) the mass flow rate on the efficiency and pressure drop

SIMULATION VERIFICATION

A three-dimensional CFD model is established as shown in Figure 9 for numerical simulation verification, which is the same as the internal heat transfer unit of the test trapezoidal prototype. The model unit size is 2.50 mm × 4.50 mm× 120 mm with 2,240,000 unstructured tetrahedral meshes after grid independence verification.

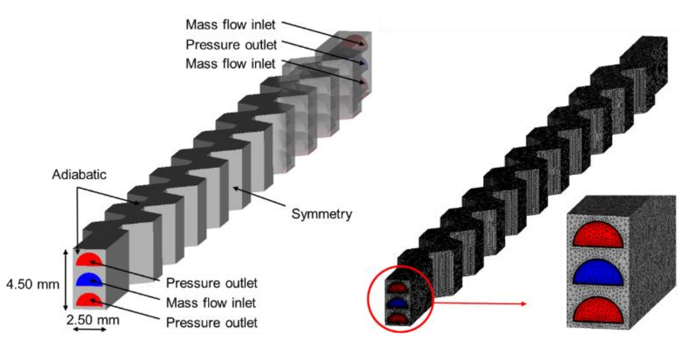

Figure 9: The mesh model unit and boundary conditions

The same setup is applied as the trapezoidal PCHE unit model in Figure 1, and the simulation results are compared with the test values under the experimental conditions, as shown in Figure 10. The outlet temperature between simulation and experimental results are in good agreement with each other except at the point of 40 °C due to the drastic changes in physical properties in Figure 10(a) and (c). The maximum deviation between them is 2.80% and 7.92% on the cold and the hot side, respectively. As for the pressure drop, the variation trend of simulation and experiment results is consistent in Fig. 10(b) and (d), with a maximum deviation of 9.22% and 24.54% on the cold and the hot side. The larger deviation on the hot side may be due to the roughness of the hot plate channels exceeding the design requirements. In general, it can be considered that the simulation model can accurately reflect the experimental situations.

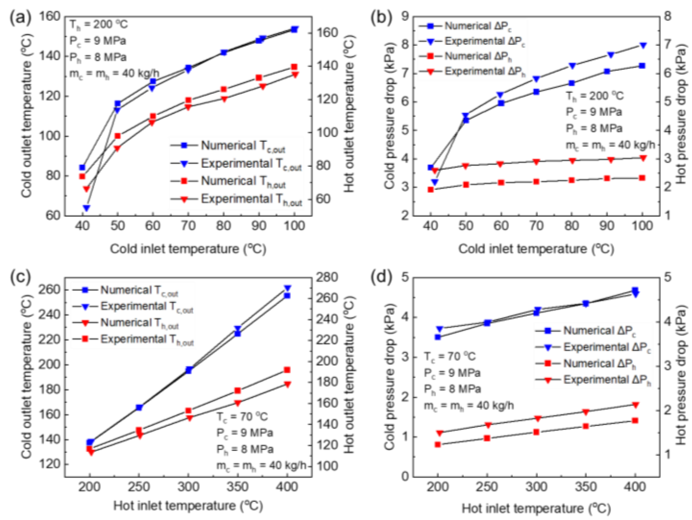

Figure 10: Comparison of simulation and experimental results of (a) & (c) outlet temperature and (b) & (d) pressure drop on both sides

Due to the fluctuations caused by pump operation, it is difficult to accurately measure the heat transfer parameters when the inlet mass flow rate is less than 15 kg/h in the circulation loop. The above verified numerical simulation model can effectively solve this problem, thereby extending the Nusselt number correlations to a lower Reynolds number range as Equations (3) and (4). The deviation of new correlations with all experimental and extended numerical simulation results are within 16%, and 85% of the data on both cold and hot side deviate within 10% as shown in Figure 11.

On the cold side:

$$Nu_c = 0.1232 Re_c^{0.7193} Pr_c^{0.1007} \tag{3}$$

$$\begin{bmatrix} 3{,}796 \le Re_c \le 30{,}000 \\ 0.91 \le Pr_c \le 1.61 \end{bmatrix}$$

On the hot side:

$$Nu_h = 0.0501 Re_h^{0.8131} Pr_h^{0.5540} \tag{4}$$

$$\begin{bmatrix} 1{,}821 \le Re_h \le 14{,}000 \\ 0.77 \le Pr_h \le 0.98 \end{bmatrix}$$

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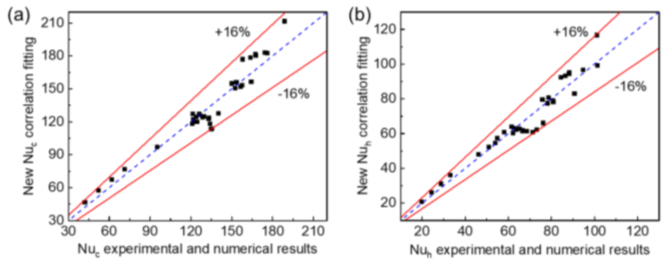

Figure 11: The difference between the new correlations and the experimental and numerical results on (a) the cold side and (b) the hot side

CONCLUSIONS

In this work, a trapezoidal channel structure model is developed and compared with the straight, the S-shape, and the zigzag structures. It indicates that trapezoidal structure owns the highest heat transfer coefficient but also the largest pressure loss. A sandwich model is to optimize the problem, improving the heat recovery efficiency by 5% while reducing the pressure loss by about 75% on the hot side.

The experimental tests of trapezoidal PCHE prototype are carried out in the sCO_2 test loop. The correlations of the Nusselt numbers related to the average Reynolds numbers and Prandtl numbers are proposed on both sides, within 15% deviation with the experimental results, and 92% and 86% values are within 10% deviation on the cold and the hot sides, respectively. The pressure drop on the hot and the cold side is less than 7 kPa and 15 kPa under test conditions, respectively. The heat recovery efficiency is defined to evaluate the performance of PCHE as a regenerator. It decreases with the increase of mass flow rate, and basically remains unchanged with the increase of inlet temperature and working pressure.

A simulation model is established for verification and expansion. It has been proved to reflect the experimental results well with a maximum temperature deviation of 2.80% and 7.92% on the cold and hot side, respectively. Extended simulations are studied based on the model, which expand the Reynolds numbers ranging from 3796 to 30,000 and 1821 to 14,000 on the cold side and hot side, respectively. And new Nusselt numbers correlations are obtained within 16% deviation.

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STATEMENTS

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[1] Ji Y, X. K. Numerical study on flow and heat transfer characteristics of trapezoidal printed circuit heat exchanger. China, Hangzhou.

[2] Ji Y, W. Z. Experimental and numerical study on thermal hydraulic performance of trapezoidal printed circuit heat exchanger for supercritical CO_2 Brayton cycle. China, Hangzhou.

REFERENCES

[1] Mylavarapu SK, S. X. (2014). Thermal hydraulic performance testing of printed circuit heat exchangers in a high-temperature helium test facility. USA, Columbus.

[2] Chen M, S. X. (2016). Experimental and numerical study of a printed circuit heat exchanger. USA, Columbus.

[3] Chu W, Li. X. (2017). Experimental investigation on SCO_2 -water heat transfer characteristics in a printed circuit heat exchanger with straight channels. China, Shaanxi.

[4] Nikitin K, K. Y. (2006). Printed circuit heat exchanger thermal–hydraulic performance in supercritical CO_2 experimental loop. Japan, Tokyo.

[5] Kim IH, N. H. (2009). Thermal hydraulic performance analysis of the printed circuit heat exchanger using a helium test facility and CFD simulations. Republic of Korea, Daejeon.

[6] Kim IH, N. H. (2011). Thermal hydraulic performance analysis of a printed circuit heat exchanger using a helium–water test loop and numerical simulations. Republic of Korea, Daejeon.

[7] Kim IH, N. H. (2013). Thermal-hydraulic physical models for a Printed Circuit Heat Exchanger covering He, He-CO_2 mixture, and water fluids using experimental data and CFD. Republic of Korea, Daejeon.

[8] Zhou J, C. K. (2020). Test platform and experimental test on 100 kW class printed circuit heat exchanger for supercritical CO_2 Brayton cycle. China, Beijing.

[9] Tsuzuki N, K. Y. (2007). High performance printed circuit heat exchanger. Japan, Tokyo.

[10] Ngo TL, K. Y. (2006). New printed circuit heat exchanger with S-shaped fins for hot water supplier. Japan, Tokyo.

[11] Ngo TL, K. Y. (2007). Heat transfer and pressure drop correlations of microchannel heat exchangers with S-shaped and zigzag fins for carbon dioxide cycles. Japan, Tokyo.

[12] Kim DE, K. M. (2008). Numerical investigation on thermal-hydraulic performance of new printed circuit heat exchanger model. Republic of Korea, Daejeon.

[13] Chen F, Z. L. (2017). Comprehensive performance comparison of airfoil fin PCHEs with NACA 00XX series airfoil. China, Beijing.

[14] Pidaparti SR, A. M. (2019). Experimental investigation of thermal-hydraulic performance of discontinuous fin printed circuit heat exchangers for supercritical CO_2 power cycles. USA, Atlanta.

DOI: 10.17185/duepublico/77317

[15] Xu X, M. T. (2014). Optimization of fin arrangement and channel configuration in an airfoil fin PCHE for supercritical CO_2 cycle. China, Shaanxi.
[16] Lee S, K. K. (2014). A parametric study of the thermal-hydraulic performance of a zigzag printed circuit heat exchanger. Republic of Korea, Incheon.
[17] Wang K, X. X. (2015). Numerical investigation on heat transfer of supercritical CO_2 in heated helically coiled tubes. China, Chongqing.

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