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Experimental study on the parallel-flow heat pipe heat exchanger for energy saving in air conditioning

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Abstract: Energy saving significantly and positively affects energy conservation and releases less carbon dioxide. This study aims to investigate the energy-saving characteristics of a novel parallel-flow heat pipe heat exchanger (PFHP-HE) in air conditioning. It merged the characteristics of high axial heat transfer performance of heat pipe with high external heat transfer performance of parallel-flow heat exchanger (PF-HE). The test rig for the PFHP-HE was built on the basis of the heat transfer wind tunnel in this work. The heat transfer performance of parallel-flow heat pipe heat exchanger charged with R600A as the working fluid was experimentally studied. The filling rate of the heat pipe was 26%, and its thermal behavior was analyzed. The evaporation section of the PFHP-HE was heated by heating belt. The experiment was conducted with heating power of 500W. The experiments were performed with cooling air flow rates ranging from 63.36 m^3/h to 118.08 m^3/h. The results indicate that when the airflow rate was 118.08m^3/h, the minimum thermal resistance of the PFHP-HE was 0.06 K/W, and the heat transfer efficiency was up to 98% in the experiments. As the pulsation phenomenon was caused by its small diameter, the temperature distribution in condenser had stratified phenomenon in PFHP-HE. The PFHP-HE has excellent heat transfer performance and working stability. The PFHP-HE is a promising heat exchanger for energy-saving in air conditioning. Its superior heat transfer performance and working stability make it an attractive alternative to conventional heat pipe heat exchangers.

Keywords: Heat pipes; Parallel-flow heat exchangers; Waste heat recovery; Energy

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saving; Air conditioning

Nomenclature

R	Thermal resistance, K/W
Q	Input power, W
$\overline{T_e}$	Average temperatures of the evaporation section, ℃
$\overline{T_c}$	Average temperatures of the condensation section, ℃
η	Efficiency of the PFHP-HE
$c_{p,\mathrm{air}}$	Specific heat capacity of air at constant pressure，kJ/(kg·℃)
ρ_{air}	Density of the air，kg/m^3
A	Cross-sectional area of the rectangle wind tunnel，m^2
v	Wind speed, m/s
δT_1	Measurement accuracy of T-type thermocouple, ℃
δT_2	Measurement accuracy of Agilent data collector, ℃
T_{min}	Minimum temperature, ℃
δT	Maximum temperature error, ℃
T	Temperature, ℃
δQ	Measurement accuracy of input power, W
δU	Measurement accuracy of output voltage, V
U	Output voltage, V
δI	Measurement accuracy of current of the power source, A
I	current of the power source, A
δR	Measurement accuracy of Thermal resistance, K/W
COP	Heat efficiency ratio of air conditioning
Abbreviations	
PFHP-HE	Parallel-flow heat pipe heat exchanger
PF-HE	Parallel-flow heat exchanger
HP-HE	heat pipe heat exchange
PFHP	parallel-flow heat pipe

1. Introduction

The energy consumption of buildings has accounted for about one-third of the whole consumption of energy in China, in which consumed by air conditionings is more than 50% of the energy consumption and still grows year by year [1-4]. Globally, heating, ventilation, and air conditioning systems consume approximately 20% of electricity consumed in buildings, contributing to 10% of global electricity consumption [5]. Besides, the cooling load of sensible and latent heat often appears in air conditioning systems. Therefore the energy recovery of heating and cooling system attracts attention on decreasing air conditioning energy consumption [6,7]. Waste heat recovery through heat pipes is an effective method. It consumes minor energy and releases less carbon dioxide [8,9]. The numerous advantages of heat pipe cover include improved heat recovery effectiveness, lighter weight and lower pressure drop, and separation of hot and cold fluids[10]. Hasan et al. [11] devoted themselves to the control of fouling and antifouling techniques to make the heat exchanger high-efficiency. Among the various types of heat pipes used in air conditioning systems, gravity heat pipes are commonly used, along with capillary and separated heat pipes [12]. Gravity heat pipes are regarded as a promising heat transfer component and widely used in many fields due to their high thermal conductivity, uncomplicated construction, and cost-effectiveness. For example, in contrast to traditional pure copper heat conductor, conceptual equipment equipped with gravity heat pipe has higher heat recovery efficiency in the course of recovering energy, specially provides a large heat load[13]. The development of heat pipe technology for energy recovery has gained significant attention from the academic and industrial communities.

In recent years, a number of researchers have researched the flow characteristics and thermal performance of heat pipe and parallel-flow heat exchanger (PF-HE） by numerical simulation and experimental testing. For example, Ghani et al. [14] conducted an experimental investigation of dual-tube heat exchangers applied in domestic air-conditioning systems. As is exhibited in the experiment, the COP improved and the energy consumption reduced. Meanwhile, Zhang et al. [15] designed a gravity-assisted heat exchanger to replace traditional air conditioning systems during transitional seasons. The experiment showed that the novel heat exchanger reduced both operating expenses and running time. Mostafa et al. [16]developed a novel wind tunnel with two streams of fresh and return air, which was connected by

HP-HE. The effectiveness of heat recovery was tested and analyzed by adjusting the fresh air ratio to verify the temperature change and the heat transfer of fresh air. To recover the waste heat in the printing industry, Tao et al. [17] designed a novel type of HP-HE. The result showed that the novel type of HP-HE could reduce energy consumption by up to 15%. Eidan et al. [18] used an innovative heat pipe in air conditioning systems. The result indicated that the COP increased by 17%. Jouhara et al. [19] constructed a heat recovery system for air conditioning using multiple gravity heat pipes. The study revealed the impact of the evaporator inlet temperature and the inclination of the HP-HE on energy recovery efficiency. The HP-HE demonstrated superior heat transfer performance when inclined at 90°. Gedik et al. [20] conducted an experimental investigation of the heat transfer characteristics of thermosiphon heat exchangers charged with R134a and R401a, respectively, which were applied in recovering the waste heat from the flue gas of industry. The experiment demonstrated that the effectiveness of HP-HE varied from 35.6% to 57.7%. Mathur [21] investigated the impact of HP-HE retrofits on reducing air conditioning energy consumption under the climate conditions of Missouri. They studied the performance of air conditioning systems equipped with HP-HE that incorporated six rows of independent heat pipes arranged horizontally. The experiment demonstrated that the HP-HE was effective in enhancing the efficiency of existing air conditioning systems. Longo et al. [22] designed a double wind channel and carried out a series of theoretical and experimental studies of the HP-HE. The length of evaporator, condenser, and adiabatic section of the HP-HE were 270mm, 270mm, 160mm, respectively. The results indicated that the heat transfer efficiency of the HP-HE is higher even when the pipe is applied horizontally, and the inner surface of the pipe is spiral micro-fin. Ahmadzadehtalatapeh and Yau [23] simulated the time-of-day air supply and indoor air environment used by HP-HE in a hospital ward. The results revealed that the HP-HE significantly improved indoor air quality and reduced air conditioning energy consumption. The heat transfer efficiency was enhanced with the quantity of heat pipe rows increasing, but the increase was not obvious. The average heat transfer efficiency of HP-HE was 45% as heat pipe rows of eight. Yang et al. [10] developed a novel type of HP-HE to recover energy in exhaust air from air conditioning systems. The results helped to clarify whether the HP-HE with R134a could achieve great start-up and operational capabilities. Deng et al. [24] presented a novel model of flat-plate solar collector that combines separate heat pipes into an array. The experiment indicated that the micro-channel heat pipe array had the superior isothermal capability and possessed a rapid thermal respond speed. Liang et al. [25] proposed a flat micro-heat pipe array to improve heat transfer performance in the application of energy recovery. Shen et al. [26] presented a novel model of a heat pipe with parallel flow pipes. It has the characteristics of high axial heat transfer efficiency and compact structure.

The HP-HE is a kind of high-efficiency passive device that can transfer a lot of thermal over relatively long distances, while the temperature difference between the heat source and sink is small [27]. The application of HP-HE in recovering waste energy from exhaust air in air conditioning systems has been widely studied and proved to be an effective method for energy conservation [28]. The effects of flow configuration on a double-pipe evaporative heat exchanger have been numerically studied by Abishek [29]. This study highlighted that the net heat transfer efficiency of parallel-flow is about 6% higher than counter flow. Lu et al. [30] developed a novel type of HP-HE with a parallel flow structure to reduce its thermal resistance and enhance its heat transfer characteristics through numerical simulation and experimental research. As introduced by present work, the combined heat pipe heat exchanger has attracted attention due to its many advantages, including its lightweight construction, economic feasibility, low pressure drop on the air side, large heat transfer area per unit volume, and reliable performance[14]. Some researchers have proposed the use of independent heat pipes to cool air before entering the condenser, but their limited external heat transfer characteristics have resulted in low waste heat recovery efficiency in air conditioning systems. In contrast, the parallel-flow heat exchanger offers several benefits, such as high heat transfer coefficient, high surface area-volume ratio, and low heat transfer temperature difference, making it suitable for various applications, such as automobile air conditioning, and refrigeration evaporators and condensers [31].

In this study, it was aimed to search how to combines the advantages of PF-HE and heat pipe. Combining parallel flow heat exchangers with previous research on heat pipes, this parallel flow heat pipe heat exchanger was proposed and applied in the cold and thermal recovery of air conditioning systems. Therefore, the test rig for the PFHP-HE was built on the basis of the heat transfer wind tunnel in this work. Experimental studies on the heat transfer and flow characteristics of the PFHP-HE have been performed to reveal the heat transfer and flow characteristics, which can be a good reference for the design and operation in energy recovery of air conditioning systems.

2. Experimental apparatus and methods

2.1 Experimental setup

Figure 1 and Table 1 show the experimental schematic and key parameters of the PFHP-HE. As illustrated in Fig. 1 (a), the PFHP-HE is a novel model of the thermosyphon heat exchanger, in which a series of pipes are arranged in parallel at the center and connected by two closed tubes at the both end. Within each flat plate, parallel pipes are connected, allowing for interconnection and uniform distribution of working fluids through two connecting tubes. The flat plates are made of aluminum belts and enable easy installation of fins, which increase the effective heat exchange area per unit volume. Fig.1 (b) presents a single flat plate unit with parallel pipes in PFHP-HE. Fig.1 (c) is A-A section view of the unit. The unit has 20 parallel micro-channels with micro-fins and separated by partitions. Micro-channel and micro-fin were processed in small channels which can strengthen the internal tube heat transfer performance. Table 1 provides the geometric parameters of the PFHP-HE.

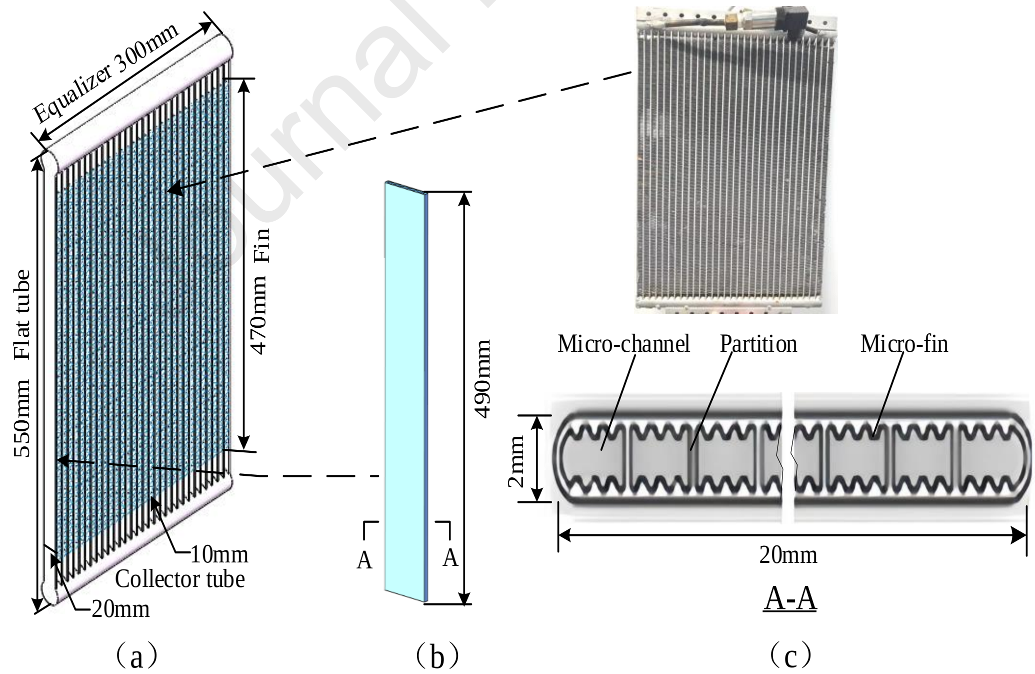
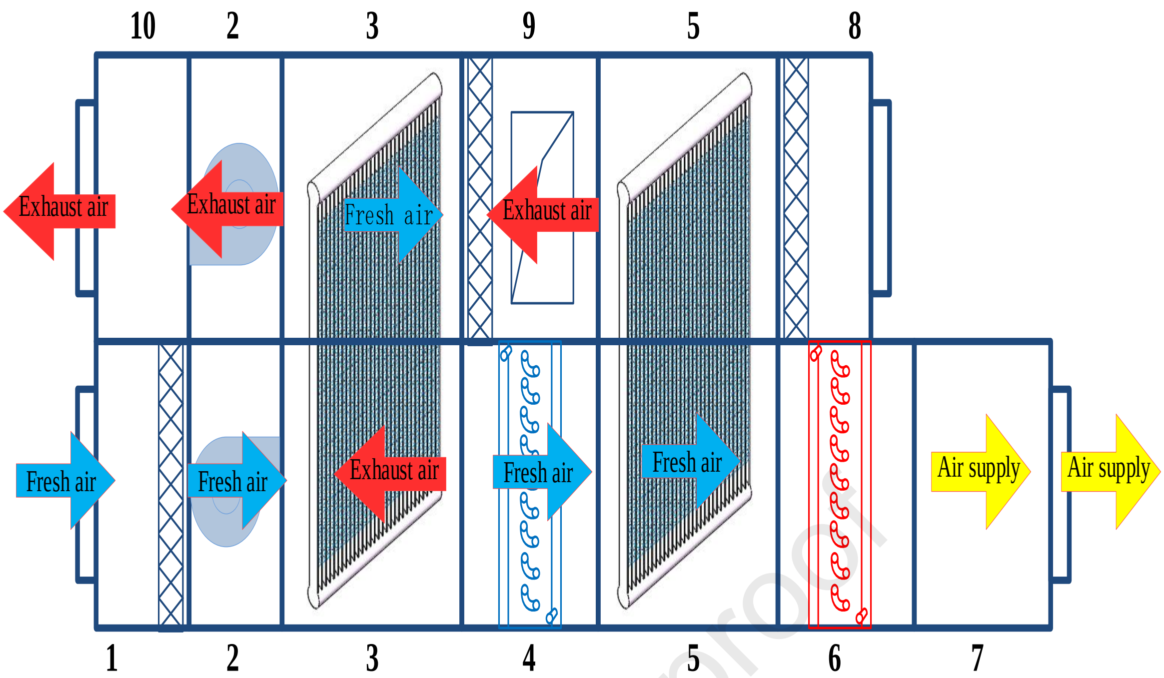


Fig. 1 Schematic diagram of PFHP-HE

Table 1 Geometric dimensions of the PFHP-HE

Parameters	Values	Parameters	Values
Length of the heat exchanger (mm)	300	Length of collector tube and equalizer (mm)	300
Width of the heat exchanger (mm)	20	Outer diameter of collector tube and equalizer（mm）	24
Height of the heat exchanger (mm)	550	Height of condenser section（mm）	300
Width of louvered fin (mm)	20	Section length of microchannel flat tube（mm）	2
Overall height of louvered fin (mm)	470	Section width of microchannel flat tube（mm）	20
Number of louvered fins（row）	30	Height of microchannel flat tube（mm）	490
Column spacing of louvered fins (mm)	10	Number of microchannel flat tube（mm）	29
Thickness of the fin (mm)	0.08	Volume of heat exchanger（mL）	315

Figure 2 illustrates the application of the PFHP-HE in a central air-conditioning cold/heat energy recovery system. For winter running mode, fresh air passed through the condenser section of the PFHP-HE, while exhaust air was directed through the evaporator section, as indicated by the arrows in the figure. The PFHP-HE transferred heat from the exhaust air to the condenser, preheating the fresh air and effectively recovering heat from the exhaust air. This process helped to reduce the heat load on the air conditioning unit in winter. In summer, the direction of airflow was reversed, with fresh air flowing through the evaporator and exhaust air passing through the condenser, as shown in the figure 2. This design ensures efficient energy recovery throughout the year and demonstrates the versatility of the PFHP-HE in various air conditioning applications.



1- Fresh air filtration section; 2-Fan section; 3-Winter heat recovery section; 4-Surface cooling section; 5-Summer heat recovery section; 6-Heating section; 7-Exhaust section; 8-Summer exhaust section; 9-Winter exhaust section; 10- Exhaust section

Fig. 2 Application of PFHP-HE in central air-conditioning cold/heat energy recovery

Figure 3 shows the sketch map of the experimental system. The heating power of the PFHP-HE was modulated by adjusting the voltage of the AC power source through a regulator. The wind speed in the wind tunnel was regulated by manipulating the fan air valve and frequency converter to cool the condenser section of the PFHP-HE. The pressure drop between the front and back of the PFHP-HE was measured using a Fluke 922 differential manometer with a precision of 0.001 in water. The duct velocity was measured by the Fluke 933 hot wire anemometer in this study. The error for the Fluke 933 hot wire anemometer is ±0.02 m/s. In addition, the surface temperature and the pressure changes of the PFHP-HE were recorded by the Agilent data collector.

R600A is an environmentally friendly refrigerant, which possesses several desirable properties, such as a high latent heat of evaporation, robust cooling capability, excellent fluidity and no harm to the ozone layer. Therefore, R600A was selected as the working fluid for the heat pipe heat exchanger (PFHP-HE) in this study. The filling rate of working fluids is defined as the ratio of the cross-sectional area filled with working fluid inside the heat pipe to the cross-sectional area of the heat pipe. The filling rate of working fluids is 26% in the test. The experiment was conducted with heating power of 500W. Two thermocouples were arranged in front of and behind the PFHP-HE for testing the temperature variation of the air before and after the PFHP-HE, respectively. The internal pressure of the PFHP-HE was measured by pressure sensor, and the pressure signal was collected by Agilent Data Acquisition System. The surface temperature of the PFHP-HE was well uniform during normal operation. The surface temperature of the PFHP-HE was monitored by eighteen T-type thermocouples. Nine of them were placed on the condenser part, three on the adiabatic part and six on the evaporator part. Figure 4 shows the arrangement of the T-type thermocouples. The signals of all thermocouples are connected to the computer through the Agilent data acquisition system, and they are recorded every 5 seconds. The tolerance of the T-type thermocouples is ±0.1°C.

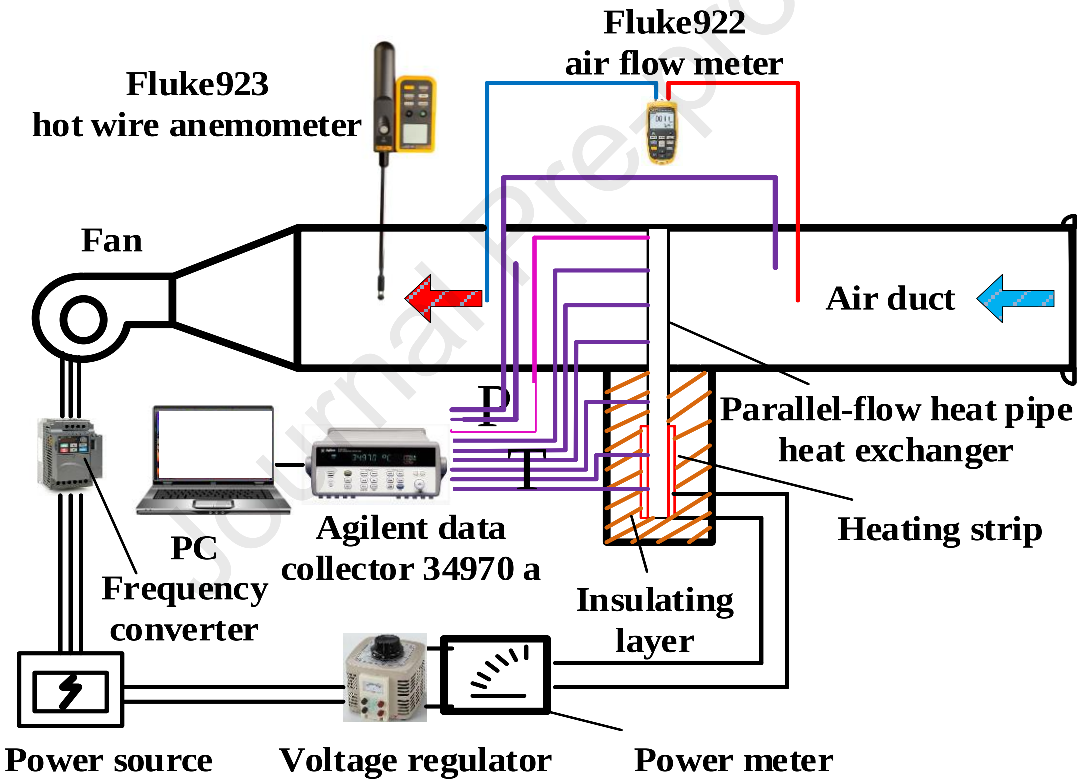


Fig.3 Schematic diagram of the experimental system

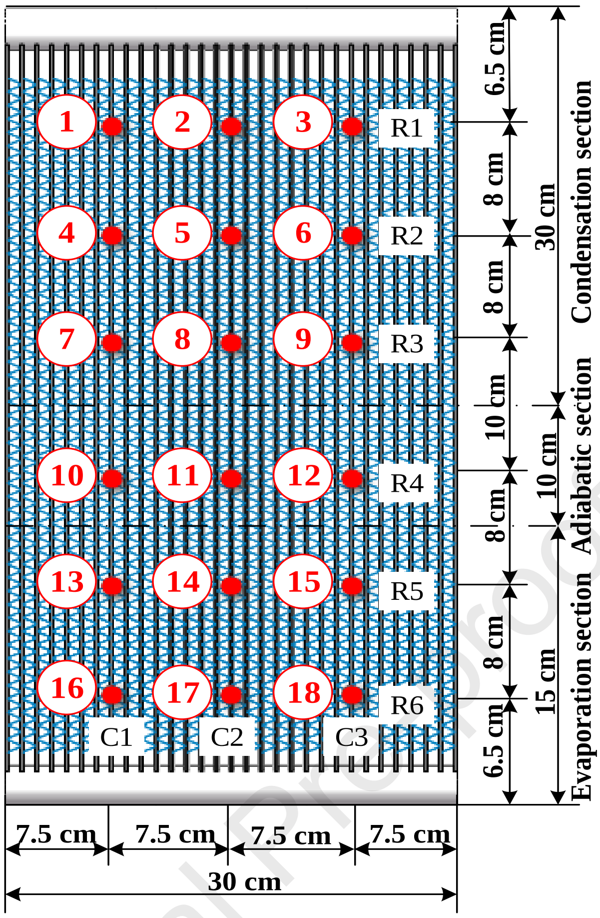


Fig. 4 The distribution of the thermocouples

2.2. Date processing

The thermal resistance of the PFHP-HE was adopted to indicate its heat transfer performance, the lower the thermal resistance, the better thermal performance of the PFHP-HE. The thermal resistance of the PFHP-HE is given by Eq. (1)[32]:

$$R = \frac{\overline{T}_e - \overline{T}_c}{Q} \tag{1}$$

Where Q is input power, $\overline{T}_e$ and $\overline{T}_c$ are the average temperatures of the evaporation section and condensation section, respectively.

The efficiency of the PFHP-HE is determined as the ratio of the heat dissipated in the condensation section to that absorbed in the evaporation section, assuming a well-insulated evaporation section and negligible heat exchange between the heat transfer tunnel and external environment. The efficiency is calculated by Eq. (2)[33]:

$$\eta = \frac{c_{p,\mathrm{air}} \rho_{\mathrm{air}} A v \Delta T}{Q} \tag{2}$$

Where $c_{p,\mathrm{air}}$ is the specific heat capacity of air at constant pressure and with the value of 1.004 kJ/(kg·°C); ρ_{air} is the density of the air with the value of 1.205 kg/m^3; A is the cross-sectional area of the rectangle wind tunnel with the value of 0.2 m×0.2 m; v is the measured wind speed, and its unit is m/s; ΔT is the temperature difference of air between front and rear of the PFHP-HE when the system reaches thermal equilibrium.

2.3 Measurement accuracy and uncertainties

In this experiment, the measurement accuracy of T-thermocouple and Agilent data collector are both 0.1℃. The minimum temperature measured by a thermocouple in the experiment is 20℃, so the maximum temperature error in this experiment is calculated by Eq. (3)[34]:

$$\frac{\delta T}{T}=\sqrt{\left(\frac{\delta T_1}{T_{\min}}\right)^2+\left(\frac{\delta T_2}{T_{\min}}\right)^2}=\sqrt{\left(\frac{0.1}{20}\right)^2+\left(\frac{0.1}{20}\right)^2}\times 100\%=0.71\% \tag{3}$$

Where δT_1 is the measurement accuracy of T-type thermocouple; δT_2 is the measurement accuracy of Agilent data collector; $T_{\min}$ is the minimum temperature measured by a thermocouple in the experiment.

The maximum output voltage and current of the power source are 60 V and 60 A, respectively. The measurement accuracy is 0.15 V and 0.015 A, respectively. When the heating power is 100 W, the output voltage is 24 V, and the current is 4.17 A, the relative uncertainty of the input power Q of the evaporation section is calculated by Eq. (4)[35]:

$$\frac{\delta Q}{Q}=\sqrt{\left(\frac{\delta U}{U}\right)^2+\left(\frac{\delta I}{I}\right)^2}=\sqrt{\left(\frac{0.15}{24}\right)^2+\left(\frac{0.015}{4.17}\right)^2}\times 100\%=0.72\% \tag{4}$$

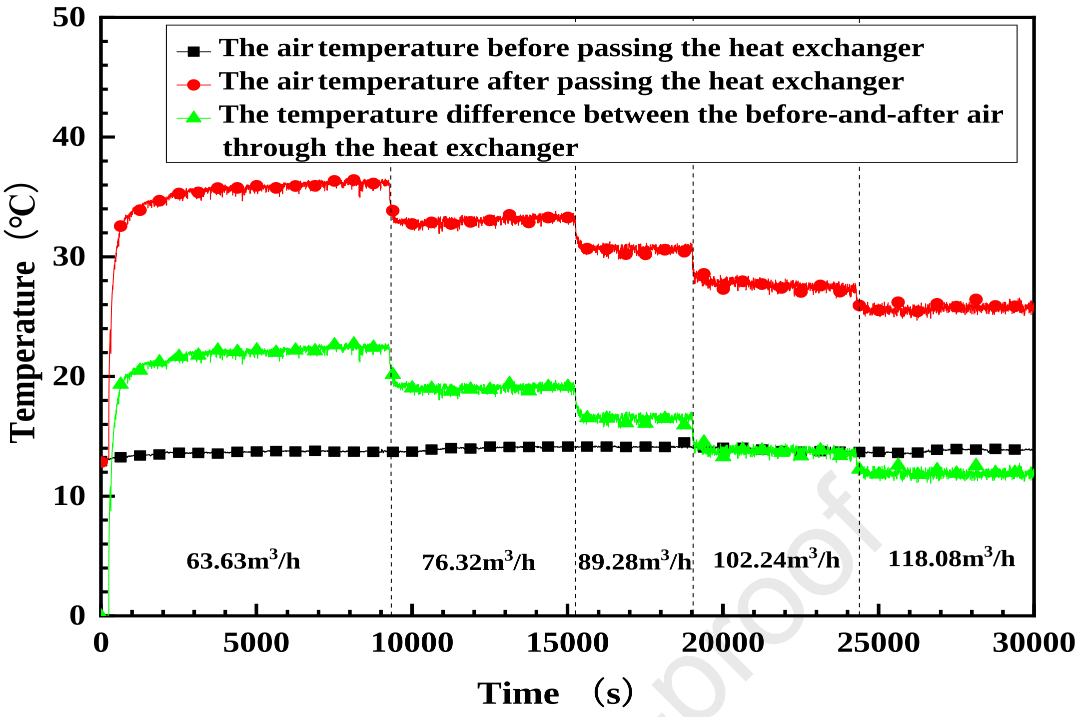
Hence, the maximum uncertainty about thermal resistance is given by Eq. (5)[35]:

$$\begin{aligned}\frac{\delta \mathrm{R}}{\mathrm{R}}&=\sqrt{(\frac{\delta \mathrm{Q}}{\mathrm{Q}})^2+(\frac{\delta \mathrm{T}}{\mathrm{T}})^2}\\&=\sqrt{(0.72\%)^2+(0.71\%)^2}\times 100\%=1.01\%\end{aligned} \tag{5}$$

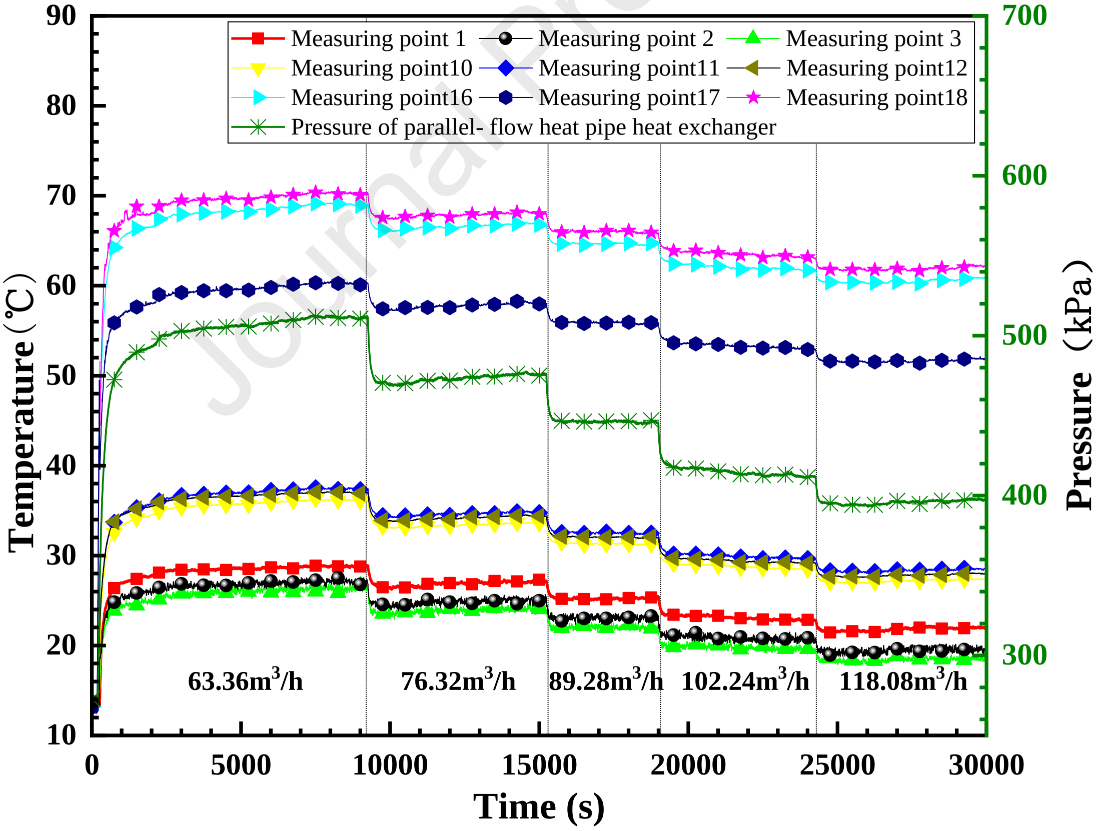
3. Results and discussion

The experiments were carried out with different wind speeds of 0.44, 0.53, 0.62, 0.71 and 0.82 m/s, which have the corresponding cooling air flow rates of 63.36m^3/h, 76.32m^3/h, 89.28m^3/h, 102.24 m^3/h and 118.08 m^3/h, respectively. When the wind volume reached the maximum, the pressure drop of the PFHP-HE had the maximum value of 1 Pa/m.

Figure 5 shows the instantaneous temperature of air at the front and rear position of PFHP-HE and the internal pressure of the PFHP-HE changes with time. Initially, the temperature difference between the working medium and the air in the wind tunnel was negligible. As the electric heater applied heat to the evaporator section, the temperature of the evaporator section rose sharply, followed by temperature increases in the adiabatic and condensation sections. With the continuous heating, the temperature increases in the evaporation section, condensation section and adiabatic section of the PFHP-HE gradually slowed down, and finally reaches thermal balance. From the figures, it also can be seen that the change in tendency of pressure was the same as the change in tendency of temperature. When the temperature reached the boiling point, the working fluid started evaporating, and the pressure of the evaporator section also increased dramatically. Through the transmission of the working fluid, high-temperature vapor carried heat along the adiabatic section until it was released in the condenser section. The heat was then absorbed by the surrounding air in the wind tunnel through convection. Meanwhile, the thermal response of the PFHP-HE was very fast and the temperature rapidly increased. However, the increases in temperature and pressure at different parts gradually slowed down and eventually tended to dynamic equilibrium. This indicated that the PFHP-HE had reached stable working condition. Furthermore, different heat flux has been applied to various measuring points in the evaporation section. It can be seen that although the temperature at measuring point 17 of the evaporation section was relatively low, the temperature at the adiabatic section and the condensing section, as well as the temperature at the measuring points in the same row as the other two columns, were not significantly different. Although the heating power of the evaporation section is different, the working medium can be evenly distributed in the condensing section because of the mutual oscillation between different pipelines, so as to have better temperature uniformity. When the cooling air flow rate was increased in the wind tunnel, the pressure and temperature drastically reduced due to the larger heat absorption of the air. But the new thermal balance was quickly established for the rapid evaporation and condensation of working fluids in the PFHP-HE. So the pressure and temperature would reach a new stead state for the different cooling air flow rates, as shown in Fig.5.



(a)



(b)

Fig. 5 The instantaneous temperature and pressure of the PFHP-HE changes with time

The thermal resistance can be calculated by Eq. (1) in which the average temperatures of the evaporator section and the condenser section were measured when the system reached thermal balance. Heat transfer efficiency of PFHP-HE could be determined by Eq. (2). The results, presented in Fig. 6, demonstrate that the thermal resistance of the PFHP-HE decreased rapidly as the cooling air flow rate increased, while the heat transfer efficiency gradually increased. When the airflow rate was 118.08m^3/h, the thermal resistance of the PFHP-HE reduced to 0.0600 K/W, and the heat transfer efficiency was up to 98%. Notably, the thermal resistance did not reach a constant value, indicating that the PFHP-HE still had the potential for enhanced heat transfer. In fact, the minimum efficiency could reach 95%, and it is significantly higher than that of fin HP-HE[16]. The thermal resistance was primarily influenced by the resistances of the evaporation and condensation sections, with a reduction in the local thermal resistance leading to an overall reduction in thermal resistance. Additionally, the cooling air flow rate has a significant effect on heat dissipation in the condenser section. Increasing air flow rate enhanced the heat dissipation, resulting in a decrease in the thermal resistance of the condenser section and an increase in the heat transfer efficiency of the PFHP-HE. These findings demonstrate that a larger air flow rate can lead to a smaller thermal resistance and higher heat transfer efficiency in the PFHP-HE.

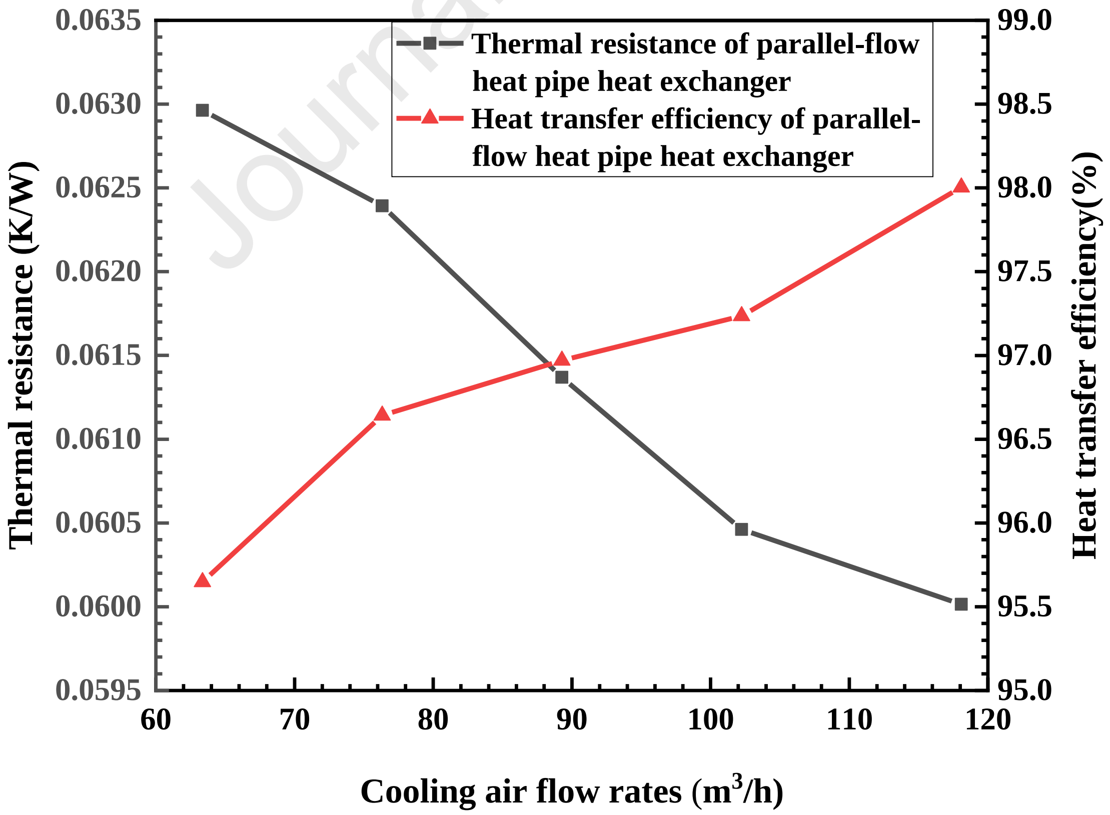
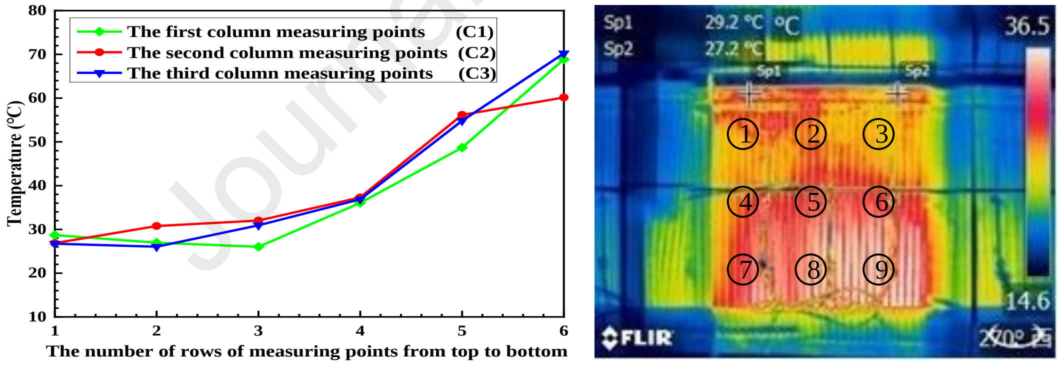
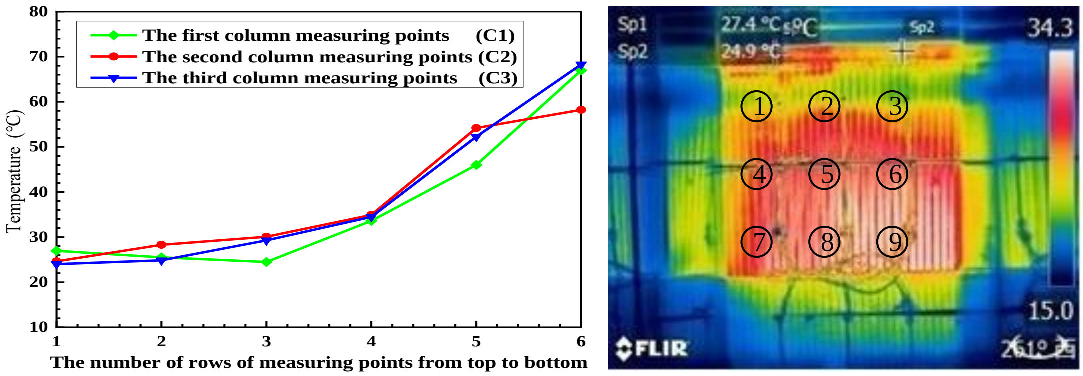


Fig. 6 Thermal resistance and heat transfer efficiency of the PFHP-HE changes with airflow rates

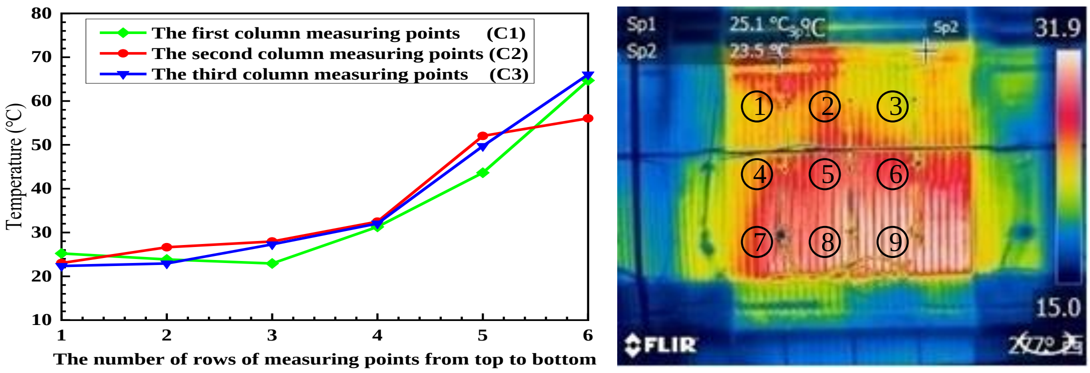
Figure 7 illustrates the impact of different cooling airflow rates on the surface temperature of the PFHP-HE and the temperature field of the condenser section in the wind tunnel. Notably, as the cooling airflow rates increased, the overall temperature of the PFHP-HE decreased. It is worth mentioning that the temperature distribution of the PFHP-HE differed from that of the HP-HE, which was composed of multiple independent heat pipes. In addition, the effects of heat flux on temperature distribution in the evaporation section were investigated by applying different heat flux at different measurement points. Notably, the temperature at the bottom of the evaporator section (column 2, row 6) was observed to be lower than that at the measurement points in columns 1 and 3 of the same row. This observation highlights the non-uniformity of temperature distribution within the evaporation section with different heat flux. In contrast, the condensing section demonstrated more uniform temperature distribution due to the mutual oscillation between different pipelines, which promotes uniform distribution of the working medium. The infrared image displayed a delamination in the vertical surface temperature of the PFHP-HE, with the top section of the condenser having lower temperatures than the lower middle section. This phenomena may be caused by the presence of some non-condensable gas accumulated on the top of the HP-HE.



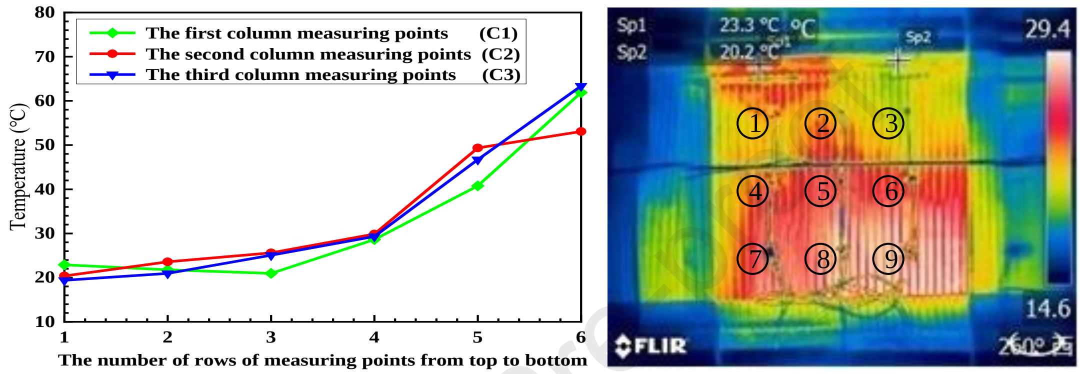
(a) The cooling air flow rate was 63.36m^3/h



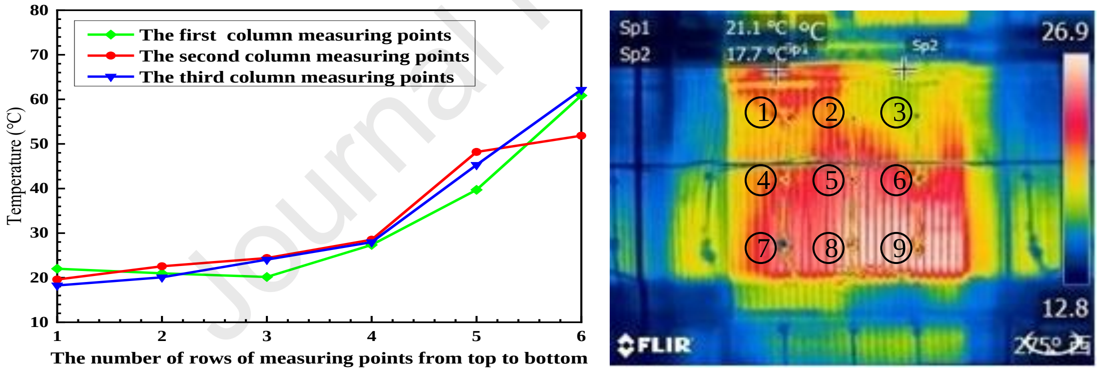
(b) The cooling air flow rate was 76.32m^3/h



(c) The cooling air flow rate was 89.28m^3/h



(d) The cooling air flow rate was 102.24m^3/h



(e) The cooling air flow rate was 118.08m^3/h

Fig. 7 The surface temperature of the PFHP-HE and the temperature field of the condenser section with different cooling air flow rates

4. Conclusions

In this work, the heat transfer and flowing performance test system for the PFHP-HE was built based on the heat transfer wind tunnel and the experimental study on heat transfer and flowing performance of the PFHP-HE was conducted, the conclusions are as follows:

(1) The PFHP-HE merged the characteristics of high axial heat transfer performance of heat pipe with high external heat transfer performance of PF-HE, applied in the cold and thermal recovery of air conditioning systems. It has superior

heat transfer performance with the minimum thermal resistance of 0.06 K/W, and the heat transfer efficiency could reach 98%, which was far more than a fin heat pipe exchanger composed of many independent heat pipes

(2) The parallel-flow heat pipe heat exchanger has good temperature uniformity because all pipes are interconnected and the working fluid oscillates with mutual excitation in different pipes. Even with uneven heating of pipes, the adiabatic and condenser sections achieved uniform temperature. In the worst case for heat transfer efficiency, the maximum temperature difference in the evaporator section of the same row was 10.071℃, whereas the maximum temperature difference between the condenser section and adiabatic section of the same row was only 1.969℃ and 1.246℃, respectively.

In future work, the cold energy and heat energy recovery of PFHP-HE in actual air conditioning systems will be analyzed, and experimental research on PFHP-HE will be conducted. The influence of filling rate and heating power on parallel flow heat pipe heat exchanger should be studied so as to determine the maximum heat transfer capacity of the PFHP-HE and the amount of energy saving from the waste heat of the air conditioning system.

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Highlights

- Develop a novel heat pipe heat exchanger for air conditioning energy saving.
- Experimentally test the heat transfer performance of the heat pipe heat exchanger.
- The minimum thermal resistance of the novel heat pipe heat exchanger is 0.06 K/W.
- The heat transfer efficiency of the heat pipe heat exchanger up to 98%.

CRediT author statement

Chao Shen: Investigation, Methodology, Writing - Review & Editing, Writing - Review & Editing. **Bowen Zhang**: Investigation, Methodology, Project administration, Validation. **Dongwei Zhang**: Conceptualization, Methodology, Supervision. **Yizhe Zhang**: Investigation, Methodology. Shen Wei: Investigation, Project administration. **Shaolun Yang**: Investigation, Formal analysis.

Declaration of interests

☒ The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

☐ The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: