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Comparative Performance Analysis of Exhaust Air Heat Recovery Devices Based on Microchannel heat pipe

Qianfan Dong[a], Zhiyong Li* and Yuqing Zhao

North China University of Technology, Civil Engineering Department, Shijingshan, Beijing, China

Abstract. In response to the problem of low recovery efficiency of sensible heat recovery devices, this paper designs a microchannel heat pipe heat recovery device (MHPHRD) and compares its heat transfer performance during summer use through experiments and simulation calculations. The experimental results show that the microchannel heat pipe (MHP) has strong temperature uniformity, and the temperatures of the evaporation and condensation sections are basically the same. When the inlet temperature of fresh air changes between 32℃ and 40℃, the recovery efficiency of the device varies between 67%-80%, and the linear relationship between the two is obvious. The established calculation model has high accuracy, with a maximum error of 0.7% between measured and simulated values. Simulation calculations show that the cooling capacity recovery efficiency of the MHPHRD is 4% -16% higher than that of the cylindrical heat pipe heat recovery device (CHPHRD), and 24%-30% higher than that of the plate heat recovery device (PHRD). The research in this article can provide reference for the practical application of the device.

1 Introduction

The fresh air energy consumption of air conditioning system accounts for about 25%-50% of the total energy consumption of air treatment [1], and studies have shown that the use of reasonable exhaust air heat recovery technology can reduce air conditioning energy consumption by 40%-50% [2-3]. Exhaust air heat recovery refers to the use of some technical means to transfer the energy in the exhaust air to the new air to pre-treat the new air. However, the traditional method of sensible heat recovery has problems such as additional consumption of electrical energy by auxiliary equipment, not close fit with the heat exchange surface, inflexible arrangement, easy to block, and occupying a large space [4-8], which leads to low heat exchange efficiency.

In order to solve the problem of low efficiency of the above mentioned sensible heat recovery method, this study proposes an exhaust air heat recovery device based on microchannel heat pipe heat recovery device (MHPHRD). The microchannel heat pipe (MHP) is shaped as a plate, with multiple parallel and independent heat transfer microchannels inside. MHP not only has advantages such as large heat transfer coefficient of traditional heat pipe, but also solves the problems of small heat transfer contact area, complex installation and slow response speed of cylindrical heat pipe(CHP), which has attracted more and more attention from scholars.Diao Y H [9] used nanofluid as a work material to test the heat transfer performance of MHP. Later, MHPHRD was designed and experimentally investigated [10], and the experimental results showed that the recovery efficiency of the device was 51%-78% in summer,and 56%-77% in winter, which is a significant recovery effect. Shen Chao[11] also designed a comprehensive performance test device for MHPHRD, and the experimental results showed that the MHP had a fast response speed and the thermal resistance was only 0.06 K/W. Yang Jingang[12] proposed a new type of low-temperature soot heat recovery collector based on the MHPHRD, and carried out experimental research on it, and the results showed that the heat recovery efficiency of the MHPHRD was stabilized at 77-97%, which also corroborated the recovery effect of the device in the summer of 51-78%, and in the winter of 56-77%. The excellent performance of the MHPHRD applied in heat recovery was also corroborated.

Through the above analysis, it can be found that the current research on the MHPHRD is mostly to explore its heat transfer performance, while the comparative research on the MHPHRD and other sensible heat recovery devices is less. Therefore, in order to accurately analyze the performance of this device, this paper builds a performance test device, and establishes mathematical models of different sensible heat recovery methods, and analyzes the advantages of this device relative to other sensible heat recovery devices when used in summer through experiments and simulation comparisons, and the research in this paper can provide references to the practical application of this device.

[a] qianfan19990306@163.com, * lizhiyongnbe@163.com

2 Experimental setup of MHPHRD

2.1 Construction of experimental device

The device consists of inclined flow duct fan, air duct, finned tube heater, regulator, microchannle heat pipe (MHP) and data acquisition system. The heating section of the duct on the fresh air side is made of 3cm high-temperature-resistant aluminum silicate heat-insulating board, the duct on the exhaust side and the remaining part on the fresh air side are made of 5cm polystyrene foam board, polyurethane foam caulking is used in the joints and fan connections to ensure the duct's sealing, and the gaps in the adiabatic section of the heat pipe are filled with aluminum-foil insulating cotton.

When the equipment is turned on, the fresh air is heated by the fin heater and flows to the evaporation section of the MHP, where the work material is heated and boils, and the steam produced by the work material carries heat into the condensation section of the MHP, where the steam passes through the fins and heats the cold air, and then returns to the evaporation section to cool down the hot air under the effect of gravity after the vapor is condensed, and so on and so forth. The data acquisition system is connected to the LU-924U04Y four-channel measurement and control module, and a T-type thermocouple with an accuracy of ±0.2°C and an AR866 hot-wire anemometer are used to collect the temperature and wind speed. The schematic diagram of the experimental setup and the distribution of temperature and air velocity measurement points are shown in Figure 1, where t_1 is the fresh air inlet temperature, t_2 is the fresh air outlet temperature, t_3 is the exhaust air inlet temperature, and t_4 is the exhaust air outlet temperature.

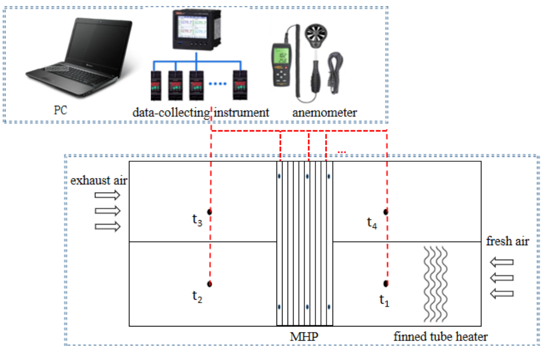

Fig. 1. Experimental procedure and distribution of measurement points

2.2 Experimental conditions

The actual test time of this article is from May 25th to June 20th, and the test location is a typical natural ventilation room, which size of 5.2m×4.5m×3m. The experimental device is placed vertically, through the hot wire anemometer shows the value of real-time control of the air duct wind speed at about 1m / s, the same amount of fresh air and the exhaust volume, the temperature of the room is 26℃. Control the fin heater temperature of the fresh air heating, fresh air inlet temperature were set to 30°C, 32°C, 34°C, 36°C, 38°C, 40°C, and test 6 sets of data, to be stable air temperature every 2 min to record the temperature of the measurement point once. In order to ensure the accuracy of the data, each set of data testing 3 times to take the average value as the point of the working conditions of the test data.

2.3 Performance evaluation indicators

MHPHRD takes MHP as the core heat transfer element, so the evaluation of device performance first evaluates the performance of MHP, and after confirming that the MHP is in good operating condition, then the recovery efficiency is used as the evaluation index to measure the performance of the device, and the definitions of the evaluation parameters and formulas are as follows.

The MHP can be evaluated by the average temperature difference between the evaporation section and the condensation section to evaluate the performance of the heat pipe. The performance of MHPHRD can be evaluated by the cold recovery efficiency of the device, defined as the ratio of the actual heat transfer to the theoretical maximum heat transfer, and the cold recovery efficiency E_C is calculated as follows:

$$E_c = \frac{L_x(t_1 - t_2)}{L_{\min}(t_1 - t_3)} \times 100\% \tag{1}$$

In the formula, L_x is the mass flow rate of fresh air, kg/h; L_{min} is the minimum value of mass flow rate of fresh and exhaust air, kg/h.

3 Mathematical modeling

3.1 Analysis of heat transfer process

Sensible heat recovery units are surface heat exchangers where hot and cold fluids do not contact each other. If the inlet temperature t_1 and flow rate m_1 on the fresh air side, the inlet temperature t_2 and flow rate m_2 on the exhaust air side, and the thermophysical parameters of the air are known, we can apply the method of heat exchanger efficiency to establish a mathematical model of the heat transfer process and combine it with the actual heat transfer of the device. Neglecting the phase change heat transfer process inside the heat pipe, we can get the general model of the heat pipe and plate heat recovery device as follows:

The formula for calculating the heat exchange capacity of the unit is as follows:

$$Q = E_c C_{min}(t_1 - t_3) \tag{2}$$

In the formula, E_C' is the heat transfer effectiveness of the device, defined as the ratio of the actual heat transfer to the theoretical maximum heat transfer; C_{min} is the minimum value of the heat capacity on the fresh air side and the exhaust air side, kJ/h·K.

In the above formula minimum heat capacity and heat transfer effectiveness are calculated as follows:

$$C_{\min} = \min(m_1 c_{p1}, m_3 c_{p3}) \tag{3}$$

In the formula, C_{p1} is specific heat of air in the fresh air stream, kJ/(kg·K); m_1 is the mass flow rate of fresh air inlet air, kg/h; C_{p3} is specific heat of air in the exhaust stream, kJ/(kg·K); m_3 is the mass flow rate of exhaust air inlet air, kg/h.

$$E_C' = \frac{1-\exp[-NTU(1-C_{min}/C_{max})]}{1-(C_{min}/C_{max})\exp[-NTU(1-C_{min}/C_{max})]} \quad (4)$$

In the formula, NTU is the number of heat transfer units; C_{max} is the maximum value of the heat capacity of the fresh air side and the exhaust air side, kJ/h·K.

The formula for the number of heat transfer units NTU of the heat exchanger in the above equation is as follows:

$$NTU = \frac{KA}{C_{min}} \quad (5)$$

In the formula, K is the heat transfer coefficient between air and heat pipe wall, W/(m^2·K); A is the effective heat transfer area, m^2.

Subsequently, the outlet temperatures of the fresh and exhaust air are calculated from the calculated heat exchanger Q, using the following formula:

$$t_2 = t_1 - \frac{Q}{m_1 c_{p1}} \quad (6)$$

$$t_4 = t_3 + \frac{Q}{m_3 c_{p3}} \quad (7)$$

In the formula, the calculation process first assumes that the exit temperature value of the fresh air side is t_2' (t_2'=t1+(4~6) ℃), and compares the E_C calculated according to Equation (1) with the E_C' calculated according to Equation (4), and if the error between the two is less than 0.1, it is considered that the exit temperature value of the fresh air side is the actual exit temperature value, and if it is not, then reset the t_2' for calculation. It is worth noting that in order for the model to achieve higher accuracy, the thermophysical parameters of the air in the modeling process are not fixed values, but change with the change of the average temperature in the duct, so with the change of the assumed value of t_2', the subsequent values of the parameters change until the error meets the requirements.The heat transfer efficiency algorithm of the three heat exchangers is basically the same,the difference lies in the formula (5) in the heat transfer coefficient K and the calculation of the effective heat transfer area A, the calculation of K can be referred to the "Principles and Design of Heat Exchangers" [13] in the relevant formulas, the calculation of A is relatively simple, and will not be repeated here. The model is simulated by MATLAB software, and the specific calculation steps are shown in Fig. 2.

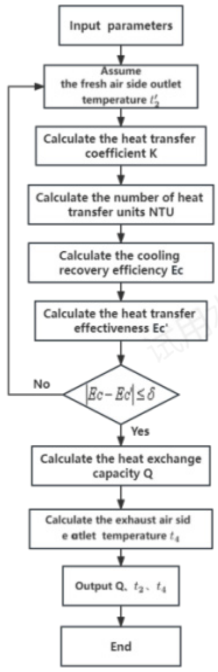

Fig. 2. Flowchart of the model of heat pipe heat recovery device

3.2 Calculation of working conditions

In order to compare the performance of the return air heat recovery device scientifically, according to the principle of heat exchanger volume proximity, cylindrical heat pipe (CHP) and plate heat exchanger (PHE) are selected for comparative analysis.CHP and MHP have the same piping and material, and the length of the fins along the direction of the airflow is 50mm, and the dimensions are different as shown in Fig. 3 and Table 1 below.CHPHRD is arranged in forked rows, with dimensions of 170mm×170mm×660mm and MHPHRD is arranged in parallel rows, with dimensions of 500mm×100mm×190mm after the arrangement. PHE model is B3-60-66, which manufactured by a certain manufacturer, with dimensions of 476mm×140mm×174mm, and an effective heat exchange area of 3.52 m^2.The plate material of PHE is stainless steel, with a thermal conductivity of 14.4 W/(m·K). , there are 66 plates, the distance between the plates is 1.5mm, the width of the channel is 140mm, and the thickness of the plate is 1.2mm.

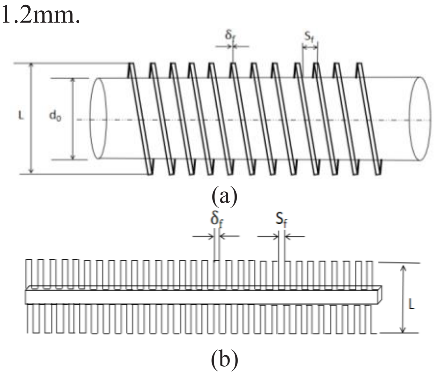

Fig. 3. (a)Calculated dimensions for CHP (b)Calculated dimensions for MHP

Table 1. CHP and MHP size

	CHP	MHP
Dimension	The length of the tube is 660mm and the diameter d_0 is 25mm	500mm×50mm ×3mm
Effective heat transfer area A/m²	2.89	3.12
thickness of fin δ_f /mm	0.2	0.5
Pitch of fin S_f/mm	5	1.6
distance between tubes S_1/mm	60	16
quantities	8	20

3.3 Model accuracy analysis

The accuracy of the model calculation was analyzed through the experimental study of the MHPHRD. As can be seen from Fig. 4,the average relative error between measured and simulated values is 0.38%, and the maximum relative error is 0.7%, which proves that the established model can be used for the simulation and analysis of heat exchange of the device.

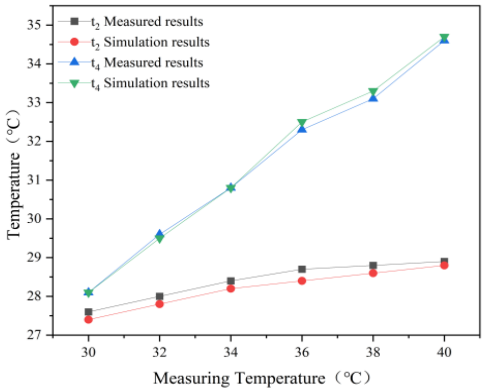

Fig. 4. Comparison results between simulated and measured results of t_2, t_4

4 Analysis of findings

4.1 MHP Thermal homogeneity

The temperature distribution of the evaporation section and condensation section measurement points when the MHP is working is shown in Fig. 5, from which it can be seen that the highest temperature of the evaporation section is 29.6°C and the lowest temperature of the condensation section is 28.6°C under condition 1, with a temperature difference of 1°C.The temperature difference between the evaporation section and condensation section under other conditions is about 0.7°C, which indicates that the MHP has good temperature uniformity and high heat transfer efficiency.

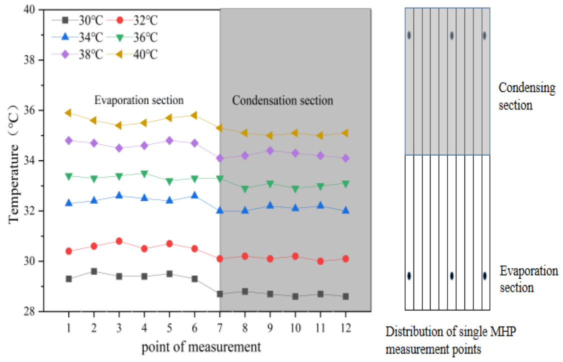

Fig. 5. MHP evaporation section, condensation section measurement point distribution and temperature

4.2 Cooling Recovery Efficiency

Fig. 6 shows the cold recovery efficiencies at different temperatures.

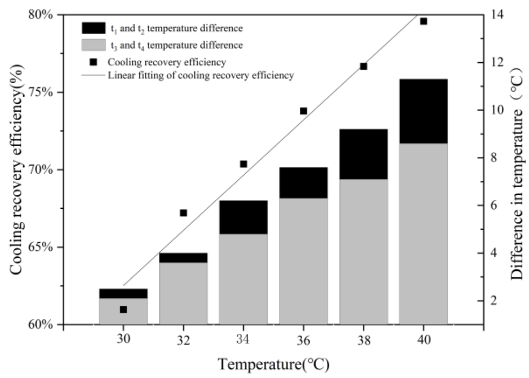

Fig. 6. Results of MHPHRD temperature efficiency variation with fresh air temperature

As shown in Fig. 6, the cold recovery efficiency of the device as a function of outdoor temperature can be derived as:

$$E_c = 0.0176t_1 + 0.091 \quad (8)$$

From the formula (8) can be calculated under any outdoor temperature corresponding to the device cooling recovery efficiency. Combined with Table 2 and Figure 6, it can be seen that in the fresh air inlet temperature of 30°C, the fresh air import and export temperature difference is only 2.5°C, with the increase in fresh air temperature of the import and export temperature difference increases, in the fresh air inlet temperature of 40°C can be up to 11.3°C, the exhaust air inlet and outlet temperature difference has a similar phenomenon. Exhaust air inlet and outlet temperature difference are smaller than the fresh air inlet and outlet temperature difference, which is due to the device in the operation process has a certain heat loss caused by. As the temperature of the fresh air increases, the cooling recovery efficiency also increases, and the efficiency changes linearly with the fresh air inlet temperature.

4.3 Comparative analysis of sensible heat recovery for CHPHRD, MHPHRD, and PHRD

According to the established model, the efficiency of three heat recovery devices was calculated, and the results are shown in Figure 7. The efficiency fitting curves of CHPHRD, MHPHRD, and PHRD efficiencies are from the experimental studies in the literature [6], [10], and [8], respectively. From the figure, it can be seen that the trend of the calculated values of the cooling recovery efficiency of the three devices are all with all increase with increasing temperature. The recovery efficiencies from high to low are: MHPHRD, CHPHRD, PHRD. Among them, the cold recovery efficiencies of MHPHRD and CHPHRD are more obvious with the change of fresh air inlet temperature. the cold recovery efficiencies of MHPHRD are 4%-16% higher than those of CHPHRD, and 24%-30% higher than those of PHRD.

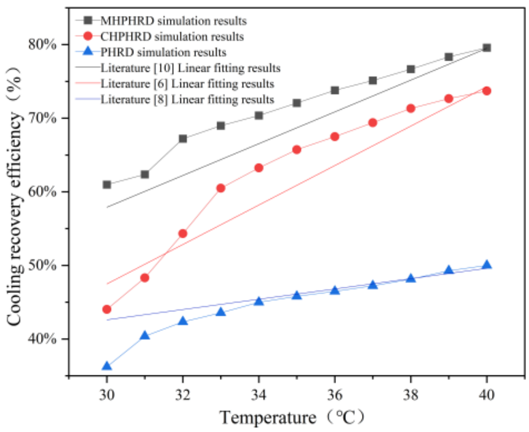

Fig. 7. Results of cold recovery efficiency E_C as a function of outdoor temperature for CHPHRD, MHPHRD,PHRD and literature fitting results

5 Conclution

Through experiments and comparative simulation calculations, it can be found that the temperature of the evaporation section and condensation section of MHP is basically the same, and the average temperature difference is basically 0.7 ℃, which proves that MHP has small thermal resistance and high heat transfer efficiency in the process of heat transfer.The heat recovery efficiency of MHPHRD is mainly concentrated in the range of 67%-80%, which is greater than that stipulated in GB/T 21087-2020 Heat Recovery Fresh Air Units in most of the operation time. 65% heat recovery efficiency in summer. The simulation results show that the cooling recovery efficiency of MHPHRD is 4%-16% higher than that of CHPHRD and 24%-30% higher than that of PHRD, indicating that MHPHRD has a more obvious advantage over other device in terms of recovery efficiency.

Reference

1. Fangyuan C, Yang C. Analysis of carbon emission reduction of fresh air heat recovery device used in summer for office buildings in three types of climate areas in China [J]. Building Science, 2013,**29**(12):103-107.DOI:10.13614/j.cnki.11-1962/tu. 2013.12. 020.
2. Yi J. Building energy consumption status and effective energy-saving ways in China. HVAC [J], 2005,35 (5): 30-40.
3. Fehrm M,Reiners W,Ungemach M. Exhaust air heat recovery in buildings[J]. International Journal of Refrigeration,2002,**25**(4).
4. Ewa Zender.A Review of Heat Recovery in Ventilation[J].Energies, 2021, 14(6):1759.DOI: 10.3390/en14061759.
5. Jie Z,Lihong Z, Jianning G, etc. Application of heat pipe in heat recovery of air conditioning [J]. Environmental Engineering, 2009,**27**(03): 72-74 + 107.
6. Xia Y, Wen W, Ruzhu W. Application of heat pipe in air conditioning [J]. HVAC, 2004(05): 26-29 + 46.
7. M.Ahmadzadehtalatapeh，Y.H.Yau.The application of heat pipe heat ex changers to improve the air quality and reduce the energy consumption of the air conditioning system in a hospital ward-A full year model simulation [J].Energy and Buildings,2011,**43**(9)：2344-2355.
8. Zhaoming Q, Zhimei H. Performance test of plate heat recyclator [C] / / Warm and Air Conditioning Academic Committee of Architectural Society of China, fifth Professional Committee of Chinese Refrigeration Society. National HVAC refrigeration 1994 academic annual conference data set. [Publisher unknown], 1994:4.
9. Zhang J , Diao Y.H , Zhao Y.H ,et al.Experimental study on the heat recovery characteristics of a new-type flat micro-heat pipe array heat exchanger using nanofluid[J].Energy Conversion and Management, 2013, 75:609-616.DOI:10.1016/j.enconman.2013.08.003.
10. Diao Y.H , Liang L , Kang Y.M ,et al.Experimental study on the heat recovery characteristic of a heat exchanger based on a flat micro-heat pipe array for the ventilation of residential buildings[J].Energy and Buildings, 2017, **152**(oct.):448-457.DOI:10.1016/j.enbuild.2017.07.045.
11. Chao S, Shaolun Y, Dongwei Z, etc. Experimental study on heat transfer performance of parallel heat flow heat exchanger [J]. Journal of Engineering Thermophysics, 2018,**39**(06): 1339-1343.
12. Jingang Y,Weiyu S, Zhenhua Q, etc. Study on the performance of lampblack heat exchanger based on micro-heat pipe array [J]. Building Science, 2019,**35**(12):49-54.DOI:10.13614/j.cnki.11-1962/tu. 2019.12.0
13. Jianzu Y. (2006) Design and principle of heat exchanger. Beihang University Press, Beijing.