COMPREHENSIVE PERFORMANCE ANALYSIS OF TWO INTERNAL THREADED TUBE AND DIFFERENT HELICAL ANGLE FINS HEAT-EXCHANGE FACILITY

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Two types of internal threaded tubes with the same fin base diameter and different fin helical angle were analyzed through heat transfer experimentation of flow condensing. In the paper, heat transfer ratio and heat transfer temperature difference, together with pressure drop and friction factor were utilized to evaluate the heat transfer performance and flow power consumption of working medium. Consequently, the per unit pressure drop of heat transfer ratio and entropy production were utilized to study the energy quality loss as well as the comprehensive performance of interchanger throughout heat transfer. Entropy increase mainly represented the irreversible loss throughout heat transfer, which was positively correlated to the thermal transmission temperature difference. Therefore, the thermal transmission entropy in the system also decreased as the saturation temperature decreased, while the Reynolds number and spiral angle of fins increased.

Key words: Internal threaded tube; Thermal transmission temperature difference; Friction factor; Unit pressure drop heat transfer ratio; Entropy production.

1. Introduction

Internal threaded tube heat-exchange facility has been widely put into service in air conditioning and refrigeration field, on account of the corresponding high heat transfer efficiency. As an example, for various finned tube heat-exchange facility uses, micro-finned heat-exchange facility were applied to enhance the heat transfer efficiency of working substances in tubes. Also, double-side enhanced tubes are utilized in shell-tube heat-exchange facility, in

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addition to updating the structure of aluminum fins to enhance air-side heat transfer. The research on the impact mechanism of surface strengthening structure on heat transfer performance is quite important for heat transfer performance improvement and new heat-exchange facility development.

In addition to the heat transfer enhancement mechanism study of enhanced tubes, the comprehensive performance evaluation of heat-exchange facility tubes could also indirectly make the overall performance of heat exchanger better through the best suitable tube type selection, depending on actual needs. When heat-exchange facility are evaluated, many aspects are often involved: thermal performance (heat transfer and resistance), economy and operation safety of heat exchanger, while no index exists to achieve the evaluation significance of all heat-exchanging unit. Therefore, various evaluation indicators are required to be selected for different purposes[1].

In the researches on the thermal transmission problem and heat transfer enhancement of the thin tube, in microchannel and smooth microchannel tube, the pressure drop characteristics of R134a was looked into by Ammar et al.[2]. The results show that: in microchannel and smooth microchannel tube, pressure drop gradient is 80 kPa/m and 52 kPa/m respectively when the cross-section mass flux is 2650 kg/(m·s). For smooth microchannel tube with a hydraulic semidiameter of 0.48~0.508 mm, the heat transfer characteristics was analyzed by Chien et al.[3]. It was manifested that the heat transfer ratio in the tube diminished with the decrement of the pipe diameter at a cross-section velocity of 2100 kg/(m·s). With regard to microchannel with "V" wing and hydraulic diameter of 1.5mm, Alneama et al.[4] studied the heat transfer performance. The results show that heat transfer coefficient of "V" wing microchannel was about 1.13 times of that of smooth microchannel, and the resistance pressure drop was about 2.5 times of that of smooth microchannel. Lotfi et al.[5] discussed the thermal performance of the vortex generator fins with delta wing, triangular folded plate and conical crest. The *N*u of the folded plate vortex generator fins is about 1.4 times that of the ordinary flat fins, while the friction factor of the folded plate vortex generator fins is only slightly increased.

Many single indicators exist to characterize heat transfer performance and resistance power consumption, such as heat transfer coefficient, heat transfer temperature difference, Nusselt number, pressure drop, friction factor and pump power. Still, for most heat-exchange facilitys, the enhancement of heat transmission effect is usually accompanied by the working medium power consumption increase[6]. For this reason, certain scholars[7-16] proposed to utilize per unit pressure drop of heat transfer ratio (K/P) and Nusselt number per unit resistance factor (*N*u/f), to assess the comprehensive performance of heat-exchange facility. As the research continued, the effective heat transfer enhancement technology could not produce a significant improvement of heat transfer at any unit pressure drop, but it could be highly improved under the same power consumption. Therefore, $(Nu/f)^{1/3}$ was proposed as an evaluation index of heat transfer at the same power consumption.

On account of the second law of thermodynamics, Bejan[17] proposed that the generation of entropy could be made use of to evaluate the heat transfer quality and irreversibility of thermal transmission in a heat-exchange facility. The entropy generation of heat-exchange facility mainly included the generation of entropy brought about by fluid flow and heat transfer temperature difference. Lai X, et al.[18] utilized the total entropy increment rate to evaluate the comprehensive performance of the flower plate and baffle plate heat-exchange facilitys, fully considering the temperature difference heat transfer loss and flow resistance loss. It was deduced that the total entropy increment rate changed with the shell-side fluid velocity, which provided a direction for the heat-exchange facility optimization. On the basis of the deficiency of exergy efficiency and dimensionless entropy yield in practical applications, Wu S, et al.[19] proposed a new thermodynamic performance evaluation index of heat-exchange facility, dimensionless entropy yield, which not only had a clear physical meaning, but it also had a broader scope of application. Wang S, et al.[20] deduced that the existing pyrolysis theory only considered the influence of heat transfer quantity or temperature difference of the heat-exchange facility, without pressure drop or pump power consideration. Therefore, for the sake of characterize the thermal transmission quantity under unit pressure drop and unit pump power, a new pyrolysis evaluation index EPEC was proposed.

In this paper, with regard to R134a flow condensation heat transfer experiments were carried out on an existing experimental bench, while two types of internal threaded tubes with the same fin base diameter and different fin helical angles were analyzed. Firstly, heat transfer ratio, pressure drop, thermal transmission temperature difference and friction factor were utilized to evaluate the flow power consumption and thermal transmission performance of heat-exchange facility. Following, on the basis of the first law and second law of thermodynamics, entropy production and per unit pressure drop of thermal transmission ratio were utilized to evaluate the comprehensive performance of heat-exchange facility, while the effects of experimental variables on each index were analyzed. Simultaneously, data support and theoretical basis were improved for the heat-exchange facility selection.

2. Experimental equipment

For the sake of flow condensation thermal transmission in tube, the schematic diagram of the measurement platform is presented in Fig. 1. In general, all equipment in the system served to provide an experimental simulation environment for heat-exchange facility tubes. Preheater, superheater, condenser, supercooler and cryogenic source together, maintained the energy conservation in the system.

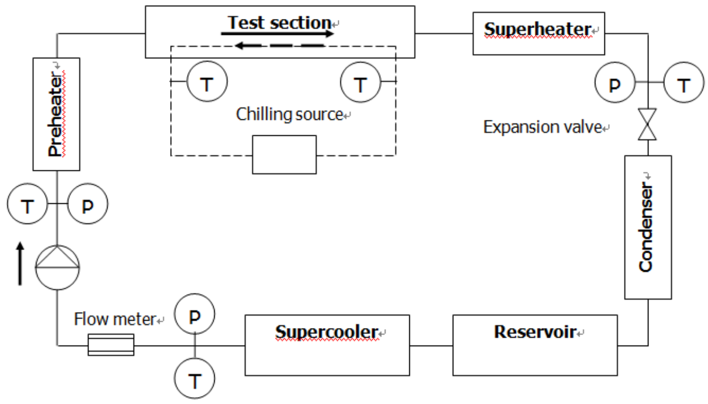


Fig.1 Schematic diagram of experimental system

A tube heat-exchange facility is acted as the experimental test section. Refrigerant flowed in the tube (red arrow pointing), while chilled water flowed in the annular channel (blue arrow pointing), as presented in Fig. 2.

Fig.2 Experimental test section schematic diagram

Two inner threaded tubes of 7.00 mm in outer diameter were selected as the test tubes. Except for the different spiral angles of fins, the other structural parameters were the same: rib height of 0.2 mm, pitch of 0.4 mm, groove width of 0.2 mm, tooth apex angle of 27° and rib number of 50.

3. Experimental treatment

Heat transfer in experimental section $\dot{Q}_{test}$:

$$\dot{Q}_{test} = \dot{m}_r \left(h_{r,post} - h_{r,pre} \right) - \dot{Q}_{pre} - \dot{Q}_{post} \tag{1}$$

where $\dot{m}_r$ is mass flow rate of working medium, for the purpose to measure the circulating flow velocity of working medium, DMF-1 mass flowmeter was selected, its measurement range was 0-100 kg/h and the measurement accuracy was ±0.5%. $h_{r,pre}/h_{r,post}$ are the enthalpy values at the outlet and inlet of the preheating section, kJ/kg, respectively. Because the working fluid was in liquid/gas phase at this time, the enthalpy could be calculated through temperature and pressure measurements. CYYZ11 pressure transmitter was put to use to survey working fluid pressure in the experiment. The corresponding measuring range was 0-42 bar and the measurement accuracy level was 0.1. Pt100 platinum resistance was selected to measure the working medium temperature. The corresponding range was 0~100 °C and the measurement accuracy was ±0.5 °C. $\dot{Q}_{pre}$ is the electric heating quantity in the preheating section, which is calculated from the applied voltage (V_{pre}) and current (I_{pre}), as:

$$\dot{Q}_{pre} = V_{pre} \cdot I_{pre} \tag{2}$$

$\dot{Q}_{past}$ is the electric heating quantity in the superheated section, which is also calculated by the applied voltage (V_{past}) and current (I_{past}), as:

$$\dot{Q}_{post} = V_{post} \cdot I_{post} \tag{3}$$

For the sake of ensure the precision of wall temperature measurement, the wall temperature was measured in all four directions at the same location, while the average value of four thermocouples was selected as the reference value of wall temperature, as presented in Fig. 3.

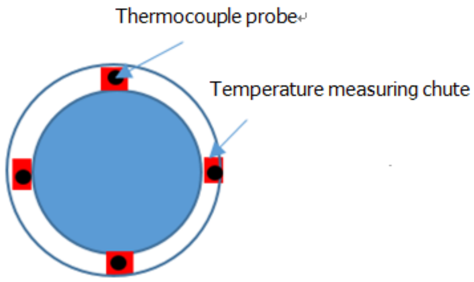


Fig.3 Schematic diagram of wall temperature measurement

Four locations existed along the tube axis and four thermocouples were arranged in the circumference of each location. The average value of 16 thermocouples was selected as the wall temperature calculation value, for the heat transfer temperature difference to be obtained as:

$$T_{wall} = \frac{1}{16}\sum_{i=1}^{16} T_i \tag{4}$$

$$\Delta T = T_{wall} - T_{sat} \tag{5}$$

Heat transfer coefficient in tube hr:

$$h_r = \dot{Q}_{test} / A_i \Delta T \tag{6}$$

where, A_i is the surface area of the heat-exchange facility tube, which is calculated directly from the diameter of the fin base of the heat-exchange facility tube, m^2.

CYY3051 differential pressure transmitter was used to measure the pressure drop in the test section. The corresponding range was 0-250 kPa and the measurement accuracy was ±0.25 kPa

4. Verification of Test Platform

For the sake of confirm the measurement accuracy of the experimental platform, single-phase condensation heat transfer experimentation was carried out in the heat-exchange facility tube at the first step. The experimental conditions were as follows: refrigerant mass flow rate of 65 kg/h, working medium temperature at the inlet and outlet of heat-exchange facility tube of 30±0.5 ^{o}C /15±0.5 ^{o}C and the supercooling temperature of heat-exchange facility tube was 3±0.5 ^{o}C /18±0.5 ^{o}C.

According to the experimental results, the accuracy of wall temperature measurement was verified. It was assumed that the liquid temperature in the tube decreased linearly along the axis. The results demonstrated that the fluctuation ranges of wall temperature measurements at four locations were 1.2 ^{o}C, 1.4 ^{o}C, 1.5 ^{o}C, 0.85 ^{o}C, respectively. In single-phase experimentation, the difference of heat transfer temperature was below 3.5±0.3 ^{o}C. A small measurement deviation could confirm the high accuracy of each thermocouple, as presented in Fig. 4.

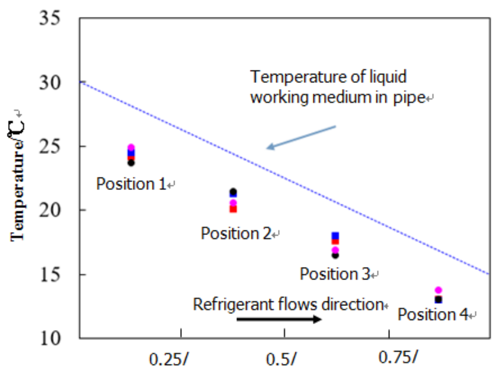


Fig.4 Measurement accuracy of tube wall temperature

In addition, for further check the wall temperature measurements, the liquid phase working fluid temperature and wall temperature at corresponding locations were calculated. Equation (6) was utilized to work out the heat transfer ratio in the tube, while Gnielinski correlation [21] was utilized to predict the heat transfer characteristics. The calculated values of the correlation were compared to the thermal transmission ratio experimental values. The results demonstrated that the average errors among the experimental values of thermal transmission ratio and the calculated values of correlation equation were 3.5%, 4.2%, 8.5% and 6.8%, respectively. With reference to the research of other scholars[22], the prediction errors within ±10% were sufficient to verify the accuracy of wall temperature measurements.

5. Analysis of experimental data

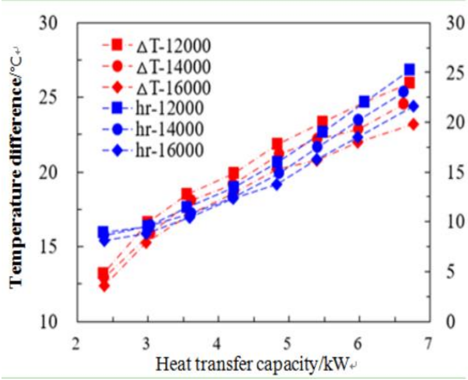
Firstly, heat transfer efficiency of heat-exchange facility tube was based on thermal transmission ratio and temperature difference of heat-exchange facility. Pressure drop and friction factor were the indices of working fluid flow power consumption. The effects of experimental variables (mainly including Reynolds number *R*e, saturation temperature, rib structure parameters) on heat transfer efficiency and power consumption of fluid flow in heat-exchange facility tubes were analyzed. Subsequently, the overall performance of heat-exchange facility tube was evaluated through unit pressure drop heat transfer coefficient HHP, while the energy loss in heat exchange process was analyzed through the system entropy increase.

5.1 Heat transfer characteristics

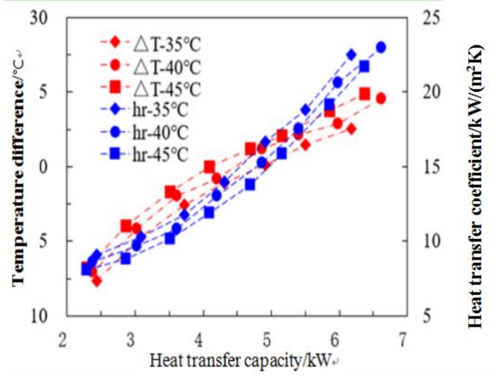
Under the same experimental conditions and thermal transmission requirements, the heat transfer coefficient was negatively correlated to the heat transfer temperature difference, signifying that the better the heat transfer performances of heat-exchange facility tubes were, the smaller the heat transfer temperature differences were. Therefore, when the influence of experimental variables on heat transfer performance was analyzed, the heat transfer temperature difference could also indirectly characterize the thermal transmission effects of heat-exchange facility tubes.

Reynolds number *R*e acted in the same way as saturation temperature did, which changed the heat transfer performance by affecting the physical properties of refrigerants. However, the larger the Reynolds number of cooling water was, the smaller the temperature gradient between the pipe wall and the working fluid was. This meant that the working fluid film temperature was closer to the saturation temperature. Consequently, the *R*e increase of cooling water and saturation temperature increase produced the same influence on heat transmission performance of heat-exchange facility tubes.

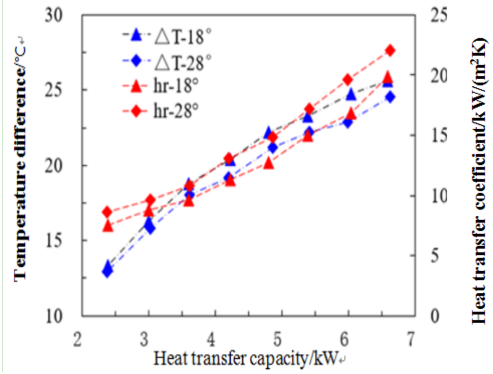
The liquid thermal conductivity of R134a decreased as the saturation temperature will be ascended. The fluid density was negatively correlated to temperature, while the gas density was positively correlated to temperature. Therefore, the heat transfer coefficient HR increased as the saturation temperature and Reynolds number Re decreased, but the temperature difference T was positively correlated to the saturation temperature and negatively correlated to the Reynolds number Re. This occurred because the reduction of Reynolds number resulted in a weakening of the total heat transfer coefficient compared to the enhancement of the total thermal transmission ratio caused by the increase of the heat transfer coefficient in the tube. Therefore, the total thermal transmission ratio decreased as the Reynolds number downsized, resulting in the lower Reynolds number and the higher difference of heat transfer temperature, as presented in Fig. 5 (a) and (b).



(a) Reynolds number *Re*



(b) Saturation temperature



(c) Structural parameters of ribs

Fig.5 Influence of experimental variables on heat transfer performance

The fins in the tube mainly enhanced the heat transfer effect through the turbulence increase of working fluid in the tube. The higher the spiral angles of the fins were, the higher the heights of the fins were. Also, the higher number of fins and the lower tip angle of the teeth would directly enhance the disturbance of the fins to the fluid, consequently enhancing the heat transfer effect.

Fig. 5 (c) presents that under the same heat transfer requirements, the temperature difference of 28^o helical angle heat-exchange facility tube was approximately 0.36~1.92 lower than the 18^o helical angle heat-exchange facility tube, which directly proved that the 28^o helical angle heat-exchange facility tube had better heat transfer coefficient than the 18^o helical angle heat-exchange facility tube.

The flow pressure drop of working agent in pipe is mainly composed of friction drop of pressure and accelerated pressure drop between pipe wall and working agent, the latter is mainly caused by the change of working fluid gas content. In the experiments, the outlet and inlet conditions of heat-exchange facility tube remained constant, while the accelerated drop of pressure was mainly caused by the increase of thermal transmission (equal to the increase of mass flow rate of working substance).

With reference to Hirose M. et al.[23], for horizontal heat-exchange facility tubes, the accelerated pressure drop accounted for approximately 15% of the sum pressure drop of the working agent flow. Therefore, the friction factor f could be directly used to characterize the pressure drop in tubes, through influence analysis of experimental variables on the aggregate pressure drop in the tubes. Subsequently, the effects of experimental variables on the friction factor f and pressure drop were similar.

For intensified tubes, since the disturbance of fins led the flow of two-phase flow to become more complex, while no formula existed for the friction factor F calculation with relatively high accuracy, the Chio correlation[24] and the total pressure drop of heat-exchange facility tubes were utilized to calculate the friction factor, as:

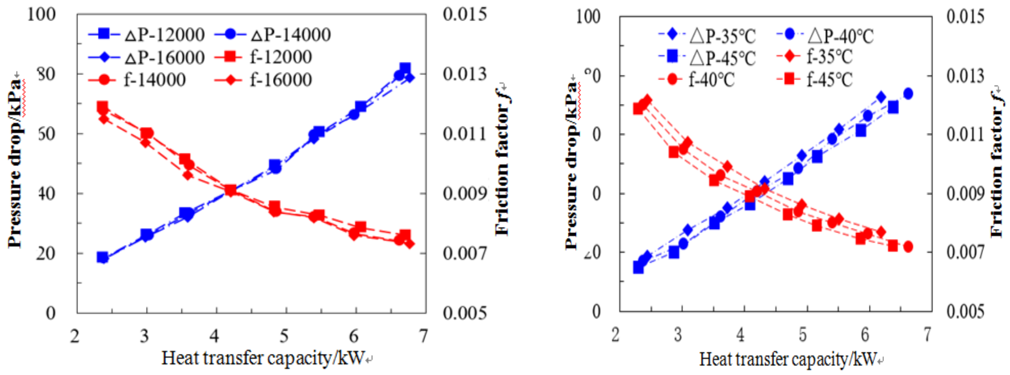
$$f_{tw} = 0.00506\ \mathrm{Re}_{fo}^{-0.0951} K_f^{0.1554} \tag{7}$$

where, f_{tw} is the friction factor of two-phase flow ; Re_{fo} is Re of total flow assumed liquid; K_f is two-phase number.

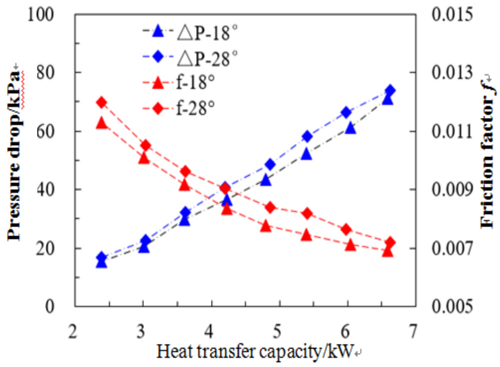
$$\Delta P_{fric} = 0.85 \Delta P_{total} \tag{8}$$

where, Δp_{fric} is pressure drop of the friction; Δp_{total} is total pressure drop.

Under the identical thermal transmission requirement, the pressure drop of refrigerant flow increased as the Reynolds number (Re) and saturation temperature decreased, whereas it increased as the heat transfer increased. Reynolds number Re and saturation temperature had the same effect on friction factor f as the pressure drop did, but friction factor f decreased as the heat transfer increased, as shown in Fig. 6 (a) and (b).



(a) Reynolds number Re (b) Condensation temperature



(c) Structural parameters of ribs

Fig.6 Influence of experimental variables on pressure drop

Due to the thermal transmission requirement, the heat transfer performance of refrigerated water dropped (Reynolds number decrease), while the difference of heat transfer temperature must be increased to meet the heat transfer requirement. Therefore, the effects of Reynolds number reduction and condensation temperature reduction on the working fluid in the tube were the same, also affecting the working fluid flow pressure drop through the physical characteristics change of the working agent (gas-liquid viscosity & density) in the tube.

Pressure reduction of two-phase streaming in pipe mainly included friction pressure reduction among inner wall of pipe and liquid flow , friction pressure reduction between inner wall of pipe and gas flow, as well as friction pressure reduction between gas-liquid interface and accelerated pressure drop caused by change of void fraction of two-phase streaming. The friction pressure reduction between the inner wall of the tube and the liquid flow occupied the main position in the sum pressure reduction in the experimental operation range (two-phase current in the tube was circular current). Therefore, the influence of liquid working fluid viscosity on the pressure reduction in the tube was very important. While the saturation temperature raised, the liquid viscosity of R134a decreased, in the wake of the saturation temperature and Reynolds number decreased, the pressure drop will be magnified. The enlargement of thermal transmission demonstrated the increase of mass flux of two-phase flow in tube, while the pressure reduction increased parabolically with mass flow. Consequently, the pressure drop in tube was also positively correlated to the heat transfer.

Differently from in-pipe pressure drop, the friction factor was negatively correlated to heat transfer, because friction factor could only be used to characterize the friction pressure drop. Moreover, the proportion of accelerated pressure drop in total pressure drop increased as the heat transfer increased (increased mass flow). The calculation results demonstrated that as the heat transfer increased, the proportion increase of accelerated pressure drop in total pressure drop was higher compared to heat transfer. Therefore, the friction factor diminished as the heat transfer performance improves.

The fins mainly increased the surface roughness of heat-exchange facility tubes to produce additional resistance of working fluid flow. The larger the helical angle of fins was, the higher the number of ribs per unit length was, and the higher the additional resistance to fluid flow was. Therefore, the pressure drop of 28^o helical angle heat-exchange facility tube was approximately 1.34~5.89 kPa, higher compared to 18^o helical angle heat-exchange facility tube. The friction factor of 28^o helical angle heat-exchange facility tube was approximately 2.62×10^{-4}~7.14×10^{-4} higher compared to 18^o helical angle heat-exchange facility tube, as presented in Fig. 6(c).

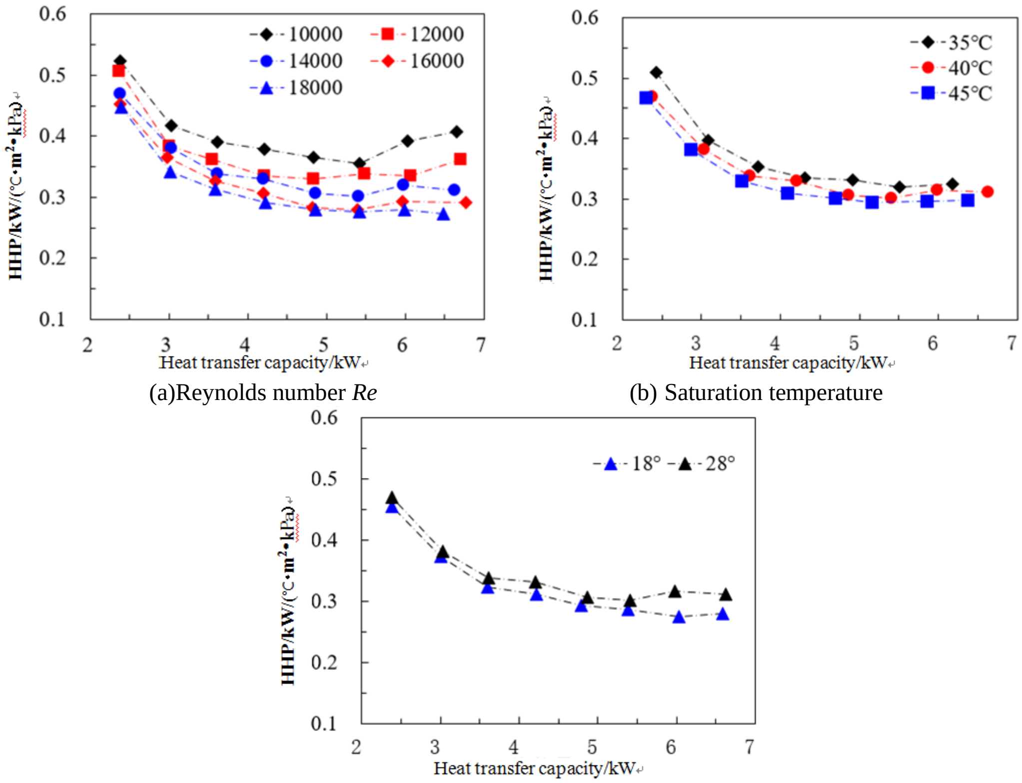
5.2 Comprehensive performance evaluation

Temperature difference/thermal transmission ratio and friction factor/pressure drop could only contribute to evaluate the thermal transmission characteristics and fluid flow loss of heat-exchange facilitys, but not the performance of heat-exchange facilitys in terms of power loss and heat transfer characteristics. Therefore, the performance of heat-exchange facility was evaluated comprehensively through unit pressure drop thermal transmission coefficient HHP and entropy.

The effect of experimental variables on HHP was essentially the relative magnitude of its impact on pressure drop and thermal transmission coefficient. The experimental variables could promote both pressure drop and heat transfer coefficient. When the effect of experimental variables on heat transfer coefficient was higher than on pressure drop, the experimental variables were positively correlated to HHP. When the effect of experimental variables on thermal transmission coefficient was lower than of pressure drop, the experimental variables were negatively correlated to HHP.

$$HHP = h_r / \Delta P \tag{9}$$

The unit pressure drop heat transfer coefficient HHP increased as the Reynolds number, condensation temperature and spiral angle of fins decreased. Therefore, the Reynolds number Re decrease, the decrease of condensation temperature and the increase of spiral angle of fins had higher effect on thermal transmission coefficient than on pressure drop. Also, the influence of experimental variables on temperature difference/thermal transmission coefficient, pressure drop/friction factor and other indicators was indirectly reflected, as presented in Fig. 7.



(a)Reynolds number *Re*
(b) Saturation temperature
(c) Structural parameters of ribs
Fig.7 Influence of experimental variables on heat transfer coefficient per unit pressure drop

Entropy constitutes a state parameter closely related to the second thermodynamics law. It is mainly utilized to determine the direction of the actual process, whether the process could be conducted or not, or whether the process is reversible. On the basis of the second law of thermodynamics, two agents with different temperatures conduct thermal transmission through the heat exchange wall. Heat could be transferred automatically from the high-temperature agent to the low-temperature agent. However, the thermal transmission from the low-temperature object to the high-temperature object and the system returning to its original state could not be carried out spontaneously, requiring external assistance. Therefore, the increase of system entropy could also indicate the decrease of energy quality throughout thermal transmission.

To facilitate the analysis of energy quality degradation in heat-exchange facility, a unified mathematical model was established to characterize the heat transfer loss of temperature difference in heat exchanges, as presented in Fig. 8.

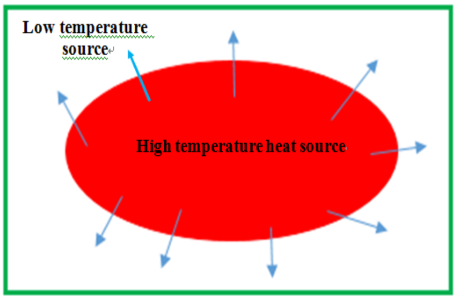


Fig.8 Entropy production of heat transfer in the adiabatic system

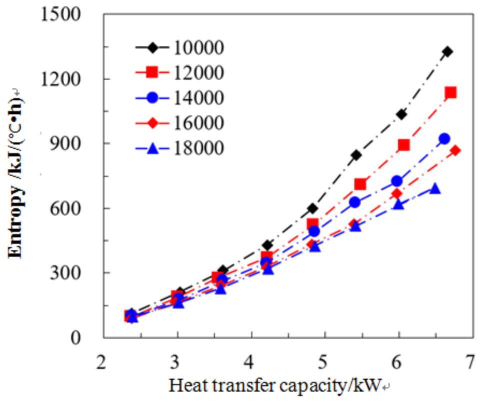
In the heat transfers, it was assumed that only thermal transmission between high-temperature heat source and low-temperature heat source existed in the heat transfers, while the system was insulated from the external environment. Regardless of the working fluid flow entropy increase, only the increase of thermal transmission entropy brought about by the difference of heat transfer temperature was calculated, as:

$$\Delta S = \dot{Q}/(T_l - T_h) \tag{10}$$

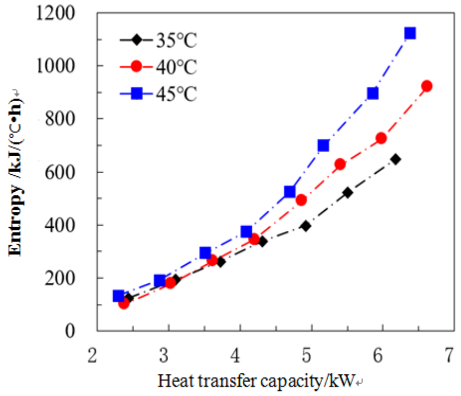
where, $\dot{Q}$the thermal transmission between high and low temperature zones in the system, kW; T_l/T_h are low temperature heat source temperature and high temperature heat source temperature in the system, respectively, K.

Because the working fluid state at the outlet and inlet of the heat-exchange facility tube remained unchanged, it could be regarded as the working fluid heat transfer performance of the heat-exchange facility tube remaining unchanged under the same heat transfer rate. Under different Reynolds number conditions, the difference of entropy change in the system was mainly caused by the differences of heat transfer performances on the cooling water side. The larger the Reynolds number of the cooling water was, the better side exchange performance of cooling water was, while the smaller the difference of thermal transmission temperature between the two refrigerants was. Therefore, the lower the energy quality of the system was, the higher the entropy of the system was, increasing as the Reynolds number decreased, as presented in Fig. 9 (a).

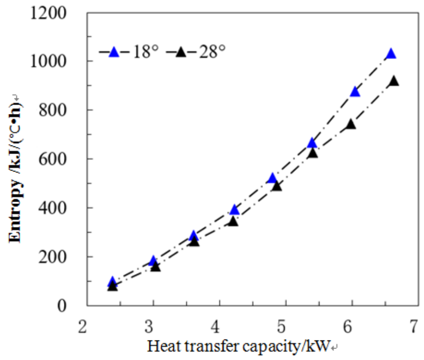
Under the same Reynolds number, the thermal transmission characteristics of cooling water were the same. Therefore, the effect of condensation temperature on the system entropy was essentially realized through the heat transfer characteristics effects of working medium in the tube. The thermal transmission coefficient increased as the condensation temperature decreased, while the difference of thermal transmission temperature decreased as the condensation temperature decreased. The smaller the difference of heat transfer temperature was, the smaller the increase of system entropy was. Therefore, the increase of system entropy decreased as the condensation temperature decreased, as presented in Fig. 9 (b).



(a) Reynolds number *Re*



(b) Saturation temperature



(c) Structural parameters of ribs

Fig.9 Influence of experimental variables on system entropy production

Similarly, to the influence mechanism of condensation temperature, the fin structure in the tube also affected the system entropy increase by affecting the thermal transmission characteristics of the working medium in the tube. The thermal transmission coefficient in the tube increased as the helical angle of the fin increased, while the temperature difference decreased as the helical angle of the fin increased. Therefore, the increase of system entropy decreased as the spiral angle of fins increased, as presented in Fig. 9 (c).

In essence, the increase of entropy caused by heat transfer of objects in adiabatic system was closely related to the difference of heat transfer temperature between high-temperature heat source and low-temperature heat source. The larger the difference of thermal transmission temperature between the two heat sources was, the higher the irreversible losses of heat transfer to objects were and the higher the increase of system entropy was. Therefore, the experimental variables could lead to the thermal transmission performance decrease of heat-exchange facility and to the increase of heat transfer temperature difference, which would inadvertently lead to higher irreversible heat transfer losses within the system.

6. Conclusions

According to the experiment of flow condensation thermal transmission in tube, the main conclusions were as follows:

1. The irreversible heat transfer loss in the system can be reduced by changing the factors that cause the decrease of the heat exchange temperature of the heat exchanger, therefore, the performance of heat exchanger is improved.

2. The pressure drop and thermal transmission coefficient increased as the Reynolds number Re decreased, whereas the condensation temperature decreased and the spiral angle of fins increased.

3. The heat transfer temperature difference and friction factor increased as the Reynolds number *R*e decreased, whereas the condensation temperature increase and the spiral angle of fins decreased. The friction factor was negatively correlated to heat transfer, because friction factor could only be used to characterize the friction pressure drop. Moreover, the proportion of accelerated pressure drop in total pressure drop increased as the heat transfer increased (increased mass flow).

4. It was observed that: with the decrease of Reynolds number *Re*, the thermal transmission coefficient HHP of unit pressure reduction will be increased, whereas the condensation temperature decreased and the of spiral angle of fins increased. Moreover, the system entropy decreased as the Reynolds number Re increased, whereas the condensation temperature decreased and the spiral angle of fins increased. The decrease of entropy indicates the improvement of heat exchanger performance.

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R E F E R E N C E S

[1] *Xie G, Wang Q, Luo L.* Entropy generation analysis and performance evaluation of heat transfer enhancement through internal flow. Journal of chemical industry, **vol. 57**, no.2, 2006, pp. 241-245.

[2] *Ammar S M, Abbas N, Abbas S, et al.* Experimental investigation of condensation pressure drop of R134a in smooth and grooved multiport flat tubes of automotive heat-exchange facility. International Journal of Heat and Mass Transfer, **vol. 130**, 2019, pp. 1087-1095.

[3] *Chien N B, Choi K I, Oh J T, et al.,* An experimental investigation of flow boiling heat transfer coefficient and pressure drop of R410A in various minichannel multiport tubes. International Journal of Heat and Mass Transfer, **vol. 127**, 2018, pp. 675-686.

[4] *Al-Neama A F, Khatir Z, Kapur N, et al.,* An experimental and numerical investigation of chevron fin structures in serpentine minichannel heat sinks. International Journal of Heat and Mass Transfer, **vol. 120**, 2018, pp. 1213-1228.

[5] *Lotfi B, Sundén B, Wang Q.,* An investigation of the thermo-hydraulic performance of the smooth wavy finand-elliptical tube heat-exchange facilitys utilizing new type vortex generators. Applied Energy, **vol. 162**, 2016, pp. 1282-1302.

[6] *Yu W, Huang W.* The enhanced heat transfer of spirall corrugated tube.Journal of Inner Mongolia University of technology, **vol. 25**, no.4, 2016, pp. 301-304.
[7] *Song R, Cui X, Wang X.* Performance evaluation criteria of spirally fluted tubes. Refrigeration and Air-conditioning, **vol. 26**, no.2, 2012, pp. 205-208.
[8] *Lu X, Zhang G, Chen Y, et al.* Effect of geometrical parameters on flow and heat transfer performances in multi-stream spiral-wound heat-exchange facilities. Applied Thermal Engineering, **vol. 89**, 2015, pp. 1104-1116.
[9] *Xu G, Deng X, Xu X, Wang Z.* Analysis on heat transfer enhancement performance evaluation. Journal of Huaihai Institute of Technology (Natural Sciences Edition), no.2, 2005, pp. 42-44.
[10] *Zhang Z, Yan H, Guan C, et al.* Heat transfer enhancement performance evaluation of a rotorinserted tube. Journal of Beijing University of Chemical Technology (Natural Science), **vol. 40**, no.2, 2013, pp. 90-94.
[11] *Shen D, Gui C, Xia J,* et al. Comprehensive performance analysis of the heat-exchange facility with rifled tube. Cryogenics, **vol. 231**, no.5, 2019, pp. 15-22.
[12] *Shen D., He W., Xia J., et al.* Experimental verification and double particle diameter calculation model of thermal conductivity of frozen soil considering latent heat. U.P.B. Sci. Bull., Series D, **vol. 81**, no.4, 2019, pp. 71-86.
[13] *Shen D., Gui C., Xia J. et al.* A novel method for calculating correlation of flow condensation pressure drop in micro-ribbed tube. U.P.B. Sci. Bull., Series D, **vol. 82**, no.2, 2020, pp. 143-160.
[14] *Ouyang X, Gao M, Liu B.* Experimental study on heat transfer and flow resistance of long-lead double-head spirally grooved tubes.Journal of Power Engineering, **vol. 82**, no.2, 2020, pp. 143-160.
[15] *Haghighi E B, Saleemi M, Nikkam N, et al*. Cooling performance of nanofluids in a small diameter tube. Experimental Thermal and Fluid Science, **vol. 49**, 2013, pp. 114-122.
[16] *Kubin M, Hirs J, Plasek J.* Experimental analysis of steam condensation in vertical tube with small diameter. International Journal of Heat and Mass Transfer, **vol. 94**, 2016, pp. 1403-410.
[17] *Bejan A.,* The Concept of Irreversibility in heat-exchange facility Design: Counterflow heat-exchange facilitys for Gas-to-gas Applications. Transaction of the ASME, Series C, Journal of Heat T ransfer, **vol. 99**, 1977, pp. 374-380.
[18] *Lai X, Li W, Huang S, Guo Z.,* Performance evaluation of total entropy increase flower baffle heat-exchange facility and single segmental baffle heat-exchange facility. Petro-Chemical Equipment, **vol. 38**, no.3, 2009, pp. 1-4.
[19] Wu S, Zeng D, Li Y. Thermodynamic performance evaluation of a heat-exchange facility system.Journal of Engineering for Thermal Energy &Power, no.3, 2002, pp. 237-240.
[20] *Wang S, Wang M, Gu X, er al.* Research on the heat transfer enhancement evaluation criterion of heat-exchange facility based on entransy theory.Journal of Xi'an Jiaotong University, **vol. 50**, no.1, 2016, pp. 1-7+15.
[21] *V. Gnielinski*. New equations for heat and mass transfer in turbulent pipe and channel flows. Int. Chem, Eng, **vol. 16**, 1976, pp. 359-368.
[22] *Liu N, Li J.* Experimental study on condensation heat transfer of R32, R152a and R22 in horizontal minichannels. Applied Thermal Engineering, **vol. 90**, 2015, pp. 763-773.
[23] *Hirose M , Ichinose J, Inoue N.* Development of the general correlation for condensation heat transfer and pressure drop inside horizontal 4mm small-diameter smooth and microfin tubes. International Journal of Refrigeration, **vol. 90**, 2018, pp. 238-248.
[24] *Choi J Y, Kedzierski M A, Domanski P A.* Generalized pressure drop correlation for evaporation and condensation of alternative refrigerants in smooth and micro-fin tubes [C]//Proceedings of IIF-IIR Commission.B1: Paderborn, Germany: B4, 2001, pp. 9-16.