Experimental study on heat transfer resistance of condensation heat transfer in high efficiency heat transfer

L X Ma¹² and K Lu¹²
¹Huadian Electric Power Research Institute Co, LTD, Zhejiang Hangzhou, 310030, China
E-mail: malongxin2004@163.com/kun-lu@chder.com

Abstract. The condensation heat transfer experiment of R134a inside the horizontal enhanced tube was operated under the condensation temperatures of 35°C, 40°C and 45°C. Firstly, correlations were used to predict the heat transfer coefficient inside the tube to check the experimental data reliability. Then mass flux and condensation temperature were selected as the variables, and the total heat transfer coefficient $K$, the water heat transfer coefficient $h_w$ and the refrigerant heat transfer coefficient $h_r$ were seen as performance evaluation index to study the heat resistance of the enhanced tube. Experimental results show that: both the total heat transfer coefficient $K$ and the refrigerant heat transfer coefficient $h_r$ increase with increasing mass flux, and get bigger with temperature decreasing. But the water heat transfer coefficient $h_w$ slightly decreases with mass flux increasing. And condensation temperature has a small influence on it. Analyze the thermal resistance and find that the water thermal resistance increases and the refrigerant thermal resistance decreases with mass flux increasing but the refrigerant thermal resistance is always less than the water thermal resistance. The difference between them gets bigger with the temperature decreasing.

1. Foreword
With the worsening of the energy crisis and the aggravation of environmental pollution, the efficient use of energy has gradually become the focus of research. With the wide application of heat transfer equipment in petroleum, chemical, electric power, refrigeration and other fields, people have higher and higher requirements for the compactness, high efficiency and low cost of heat transfer equipment. Since the introduction and manufacture of the reinforced tube by the Japanese Hitachi Company in the 1970s, the reinforced tube has been greatly developed from the original two-dimensional rectangle and trapezoidal rib to the current three-dimensional reinforced tube [1]. And its internal complicated geometry makes the already complex flow evaporation/condensation heat transfer even more complex [2].

R134a has been widely used in the field of automobile air conditioning due to its superior thermophysical properties and environmentally-friendly performance [3]. Many scholars have done a lot of research on the strengthening mechanism of R134a used in various intensified tubes. The effects of different tube sizes and hydraulic conditions on heat transfer coefficient and pressure drop are analyzed under flow boiling/condensation heat transfer conditions. Great achievements have been achieved [4-10].

Dang C et al [4] carried out an experimental study on the flow boiling heat transfer characteristics of R134a in a rectangular microchannel. The change of heat transfer coefficient under the 7-flow type of the flow pattern from bubble flow to dry flow was analyzed. Zhang X Y et al [5] mainly analyzed...
the condensation heat transfer characteristics of R134a in the annular channel under experimental conditions such as heat flux density, condensing pressure, dryness and other experimental conditions. Ou Y X P et al [6] analyzed the influence of micro-toothed tube structure parameters on the condensation heat transfer mechanism inside the tube.

In order to strengthen the research on the mechanism of heat transfer in the enhanced tube, this paper uses R134a as the working medium to carry out experiments on the newly built single-tube heat transfer test bench, study the heat transfer performance of both sides of tubes under the conditions of changing the condensation temperature and mass flow, analyze the heat transfer resistance between the two fluids in detail, and provides a theoretical basis for the study of heat transfer enhancement.

2. Experimental device
Detailed system principle of the test platform for condensation heat transfer in tube is shown in figure 1. The test platform consists of four cycles: heat transfer tube test cycle, refrigerated water cycle, cryogenic cold source and data acquisition system.

![Figure 1. Experimental device diagram.](image-url)

The heat transfer tube test cycle, the supercooled refrigerant flows out of the reservoir under the driving of the diaphragm pump, enters into the preheater through the pulse damper and the mass flow meter and is heated to the set state in the preheater. The state of the refrigerant is observed by optic liquid microscope, and the superheated refrigerant vapor completes the condensation test in the experimental section. The supercooled refrigerant that completes the condensation experiment is throttled by the electronic expansion valve and then enters into the reservoir through the drying filter, and the next cycle is repeated. During the experiment, the saturation pressure of the experimental section is controlled by adjusting the opening of the electromagnetic expansion valve. The refrigerant circulation flow rate in the system is regulated by adjusting the diaphragm pump operating frequency.

The refrigerant/freezing water temperature in the system is measured by WZPK series PT100 platinum resistance with 0.1°C accuracy. The refrigerant pressure in the system is measured by Drucker TXY800 pressure transmitter. The measuring range is 0-10 MPa, and the measurement accuracy is 0.25%; KLB-CMFI type Coriolis mass flowmeter is used to measure the refrigerant circulation flow in the system. The measurement range is 0.12-900 m³/h and the measurement accuracy is ±0.25%.
accuracy is 0.2%; the LDG-MIK electromagnetic flowmeter is used to measure the system. The water circulation flow rate has a measurement range of 0-500 m$^3$/h and a measurement accuracy of 0.5%.

![Cooling water](image)

**Figure 2. Assembly section of test section.**

The experimental section is designed as a horizontal tube heat transfer. The refrigerant flows in the tube, and the frozen water flows in the annular channel, and the two flow in opposite directions, as shown in figure 2.

In this paper, an internally threaded pipe with a rib base diameter of 5.5 mm, a rib spiral angle of 28°, a rib tip angle of 30° and a rib height of 0.15 mm is selected as the test tube, and the effective heat transfer length is 2000 mm.

The experimental conditions are set as following: condensation temperature is 35°C, 40°C, 45°C, refrigerant mass flow is 45-80 kg/h, and the refrigerant at the inlet and outlet of the experimental section maintain the temperature of 2-3°C overheat / undercooling. The state of the refrigerant is observed through the liquid microscope inside the device. The physical properties of R134a under experimental conditions are shown in table 1.

| Temperature (°C) | 35   | 40   | 45   |
|-----------------|------|------|------|
| Pressure (MPa)  | 0.887| 1.0166| 1.1599|
| Liquid density (kg/m$^3$) | 1167.5| 1146.7| 1125.1|
| Gas density (kg/m$^3$) | 43.416| 50.085| 50.085|
| Liquid devaluation (kJ/kg) | 249.01| 256.41| 263.94|
| Gas devaluation (kJ/kg) | 417.19| 419.43| 421.52|
| Liquid kinematic viscosity (uPa/s) | 172 | 161 | 151|
| Gas kinematic viscosity (uPa/s) | 12.1| 12.4| 12.6|

3. Experimental data processing
The data can be collected by the instrument includes refrigerant mass flow $G_m$, refrigerant inlet and outlet temperature $T_3/T_4$, frozen water flow $G_w$, frozen water inlet and outlet temperature $T_7/T_8$ and experimental section pressure $P$.

The refrigerant condensation heat dissipation in the experimental section mainly includes sensible heat and latent heat. The proportion of sensible heat transfer in the total heat transfer is less than 3%. Therefore, this paper takes the refrigerant heat dissipation as the calculation standard of heat transfer in the experimental section. According to the temperature/pressure of refrigerant at the inlet and outlet of the experimental section, the corresponding enthalpy of refrigerant is calculated, and then the heat transfer capacity of refrigerant is obtained as follows:

$$\Phi_r = G_m \cdot (h_{r,\text{out}} - h_{r,\text{in}})$$  \hspace{1cm} (1)

Where: $G_m$ is the refrigerant circulation flow rate, kg/s; $h_{r,\text{in}}$, $h_{r,\text{out}}$ is the inlet and outlet refrigerant
enthalpy value of the experimental section, kJ/kg.

The heat transfer coefficient of the tube is calculated by the thermal resistance separation method. The total heat resistance between the refrigerant and the frozen water in the experimental section is equal to the sum of the thermal resistance inside the tube (refrigerant side) the thermal resistance of the tube wall, and the thermal resistance outside the tube (freezing water) [11]. Since the test tube is an unused copper tube, the wall scale thermal resistance is ignored, namely:

$$\frac{1}{KA_o} = \frac{1}{h_A A_i} + \frac{1}{h_w A_w} + \frac{\delta}{\lambda \Delta T}$$

(2)

Where: $A_o/A_i$ is the inner surface area of the heat transfer tube, $m^2$; $\delta$ is the wall thickness of the heat transfer tube, m; $\lambda$ is the heat transfer coefficient of the heat transfer tube, $W/(m \cdot K)$; $\Delta T$ is the inner and outer walls of the heat transfer tube Temperature difference, °C.

Calculate the total heat transfer coefficient $K$ of the experimental section by the outer surface area of the heat transfer tube, namely:

$$K = \frac{\Phi_r}{h_w \cdot \Delta T_m}$$

(3)

Where: $\Delta T_m$ is the logarithmic mean temperature difference of the heat transfer tube, °C.

Calculate the heat transfer coefficient $h_w$ of the frozen water in the annular channel using the Dittus-Doelter [12] formula, that is:

$$h_w = 0.023 \frac{\lambda}{D_h} Re_w^{0.8} Pr_w^{0.3}$$

(4)

Where: $\lambda_w$ is the thermal conductivity of frozen water, $W/(m \cdot K)$; $D_h$ is the hydraulic diameter of the heat transfer tube, m;

The integrated heat transfer coefficient $h_r$ can be obtained from (2), (3), (4), that is:

$$h_r = A \left( \frac{1}{KA_o} + \frac{1}{h_w A_w} + \frac{\delta}{\lambda \Delta T} \right)$$

(5)

In order to ensure the accuracy of the calculation of experimental parameters, the uncertainty of each value is calculated using equations (6) [13].

$$\partial R = \left[ \sum_{i=1}^{b} \left( \frac{\partial R}{\partial y_i} \right)^2 \right]^{0.5}$$

(6)

Where: $\partial R$ is the total uncertainty of the independent variable $R$, $y$ is the influencing factor, and $\partial y$ is the uncertainty of the variable.

For example, the uncertainty of the total heat transfer coefficient $K$ can be calculated by equation (7):

$$\frac{\partial K}{K} = \left( \frac{\partial \Phi_r}{\Phi_r} \right)^2 + \left( \frac{\partial A_o}{A_o} \right)^2 + \left( \frac{\partial \Delta T_m}{\Delta T_m} \right)^2$$

(7)

It is calculated that the uncertainty of the total heat transfer coefficient $K$ is less than 5%, the uncertainty of the water side heat transfer coefficient $h_w$ is less than 0.5%, and the uncertainty of the refrigerant side heat transfer coefficient $h_r$ is less than 5%.

4. Experimental results

Although the spiral angle of fins, the number of fins, the top angle of teeth and the width of alveolar
groove all have great influence on the flow mechanism of the fluid in the tube, due to the limitation of the experimental equipment, this paper mainly analyses the influence of the experimental conditions (mass flow rate, condensation temperature) on the flow condensation heat transfer characteristics of R134a in the tube.

4.1. System reliability testing
In order to ensure the accuracy of each measurement parameter of the experimental platform, the temperature value, pressure value and mass flow value of the refrigerant inlet and outlet of the experimental section are selected, and the temperature value and mass flow value of the frozen water in the experimental section are subjected to repeated experiments. It has been verified that when the measurement error of each parameter is kept within ±0.5%, it can be said that the parameter measurement accuracy meets the requirements.

In addition, in order to verify the accuracy of the experimental data, correlations [14, 15] were used to predict the two-phase heat transfer coefficient in the enhanced tube. Among them, Cavallini used the equivalent Reynolds Re_{eq} to characterize the two-phase flow turbulence in the tube, and used the Bond number Bo and the Froude number Fr to characterize the combined effects of two-phase flow shear force, gravity and surface tension on the fluid flow mechanism. Rx was used to characterize the enhancement effect of the increase of heat transfer area and the disturbance of fins to two-phase flow on heat transfer in tubes.

\[
Nu = 0.05Re^{0.8}Pr_{L}^{1/3}Rx(Bo \times Fr)^{t}
\] (8)

Koyama assumed that the enhancement effect of the enhanced tube structure on heat transfer was mainly expressed by the enhanced convection Nusselt number Nu_{F}, and the expression formula was determined based on the experimental data of R22 and R410A in the internally threaded tube:

\[
Nu = (Nu_{2}^{2} + Nu_{b}^{2})^{0.5}
\] (9)

\[
Nu_{F} = 0.152(0.3 + 0.1Pr_{l}^{1.1})(\Phi_{V}/X_{tt})Re^{0.68}
\] (10)

\[
\Phi_{V} = 1.1 + 1.3(GX_{tt}/(gd_{i}\rho_{v}(\rho_{l} - \rho_{v}))^{0.5})^{0.35}
\] (11)

Finally, the average error of the correlation between the predicted value and the experimental value of the heat transfer coefficient is calculated using equation (12), and the correlation prediction effect is evaluated.

\[
MRD(\%) = \left( \frac{1}{n} \sum_{i=1}^{n} \frac{y_{pred} - y_{exp}}{y_{exp}} \right) \times 100
\] (12)

The prediction effect of the Cavallini correlation and the Koyama correlation on the two-phase heat transfer coefficient in the enhanced tube is shown in figure 3. It can be seen from the figure that although the correlation method overestimates the heat transfer coefficient hr in the tube and the experimental conditions (quality) Flow rate and saturation temperature have little effect on the prediction accuracy of the correlation, but both correlations show higher prediction effects. The calculated values of Cavallini and Koyama correlation and the experimental values of heat transfer coefficient inter-error is 13.25% and 9.78%, respectively, and the predicted average error of less than 15% is sufficient to verify the reliability of the experimental data measurement.
4.2. Heat transfer coefficient

The condensing temperature is (35±0.3)°C, (40±0.3)°C, (45±0.3)°C, and the mass flux is 45 kg/h-80 kg/h. The influence law of K is shown in figure 4. It can be seen from the figure that the total heat transfer coefficient K increases as the mass flow rate increases and the condensation temperature decreases. It can be known from equation (2) that the heat transfer coefficient of the refrigerant, the heat transfer coefficient of the frozen water, and the thermal conductivity of the tube wall jointly determine the total heat transfer coefficient, while the mass flow rate and the condensation temperature mainly affect the heat transfer coefficient of the refrigerant. The thermal conductivity has no influence at all and has little effect on the heat transfer coefficient of the frozen water. Therefore, the influence of the experimental conditions on the total heat transfer coefficient can be explained by its influence on the heat transfer coefficient of the refrigerant. In addition, as the condensing temperature decreases, the latent heat value of R134a increases. As the mass flux increases (the refrigerant state at the inlet and outlet of the experimental section remains unchanged), the heat flux in the experimental section increases, and the heat transfer area changes. In the case where the thermal temperature difference does not change, both of them cause an increase in the total heat transfer coefficient K.

The condensing temperature is (35±0.3)°C, (40±0.3)°C, (45±0.3)°C, and the mass flux is 45 kg/h-80 kg/h. The experimental conditions are for refrigerant/frozen water. The influence law of the thermal coefficient is shown in figure 5. It can be seen from the figure that the heat transfer coefficient of the refrigerant increases with the increase of the mass flow rate and the decrease of the condensation temperature, and the change trend is similar to the total heat transfer coefficient K and the heat transfer coefficient of the frozen water With the increase of mass flow rate and the decrease of condensing temperature, the change trend is opposite to the total heat transfer coefficient K, and the change trend is not obvious.

The heat transfer coefficient of the refrigerant increases as the mass flow rate increases, and the lower the condensation temperature, the larger the heat transfer coefficient. This is because: (1) For R134a, the gas-phase refrigerant increases as the condensation temperature increases, and the liquid-phase refrigerant decreases as the condensation temperature increases, causing the gas-liquid density ratio to decrease with the condensation temperature, which leads to greater gas-liquid velocity difference (characterizes the larger gas-liquid interface shear force), and the increase of the shear force between the gas-liquid interface reduces the thickness of the heat transfer boundary layer, thereby enhancing the heat transfer effect; (2) The flow rate of the gas-liquid phase refrigerant in the tube increases with the increase of the mass flow rate, and the difference of the gas-liquid phase density makes the gas-liquid phase velocity increase value different, so that the gas-liquid velocity difference
increases with the increase of the mass flow rate. It can also enhance the fluid turbulence in the tube and enhance the heat transfer effect. The experimental structure is similar to that of Hajeri et al [16].

![Figure 5](image_url) **Figure 5.** Influence of working conditions on the heat transfer coefficient of water/refrigerant.

![Figure 6](image_url) **Figure 6.** Influence of working conditions on the thermal resistance of water/refrigerant.

During the experimental operation, as the mass flow rate of the refrigerant increases, the heat flux in the experimental section increases. In order to maintain the state of the refrigerant at the inlet and outlet of the experimental section, the water flow in the experimental section is kept constant, mainly by controlling the inlet water temperature. Achieve the regulation of heat flux. Therefore, as the mass flow rate increases, inlet water temperature of the frozen water decreases, and the frozen water viscosity increases, resulting in an increase in the thickness of the water side heat transfer boundary layer, so that the frozen water heat transfer coefficient decreases slightly with the increase of the mass flow rate. The latent heat value of R134a increases with the decrease of condensing temperature, which increases the heat transfer capacity of the experimental section. The frozen water also needs to increase the heat transfer by lowering the inlet temperature. Therefore, the heat transfer coefficient of frozen water decreases with the decrease of condensing temperature.

4.3. Heat transfer resistance

The basic principle of heat transfer enhancement is to strengthen the main heat resistance part of the heat transfer surface, but the main thermal resistance is not constant with the change of experimental conditions. Therefore, in order to obtain heat transfer enhancement of heat transfer under different experimental conditions, it is necessary to make a detailed analysis of the total heat resistance of the heat transfer under different working conditions. Based on the outer surface of the reinforced tube, the total heat transfer coefficient \( K \), the frozen water heat transfer coefficient \( h_w \) and the refrigerant heat transfer coefficient \( h_r \) can be used to obtain the total heat transfer resistance, the frozen water heat transfer resistance, and the refrigerant heat transfer resistance. For the same type of reinforced tube, structural parameters remain unchanged, that is, the thermal conductivity of the tube wall does not change with the experimental conditions, so it can be neglected in the thermal resistance analysis calculation.

The change of the ratio of the thermal resistance of the reinforced refrigerant to the total thermal resistance of the frozen water is shown in figure 6. It can be obtained: (1) In the experimental range, the thermal resistance of the refrigerant is smaller than the thermal resistance of the frozen water, therefore, The next step of heat transfer enhancement measures of the reinforced tube should be mainly concentrated on the frozen water side; (2) When the mass flow rate increases, the frozen water flow rate remains unchanged, and the heat transfer capacity of the frozen water is mainly achieved by
reducing the inlet water temperature. The influence of mass flow rate and condensing temperature is not great, that is the thermal resistance of frozen water changes little with the experimental conditions; the mass flow rate and condensing temperature mainly affect the heat transfer coefficient of the refrigerant, and the heat transfer coefficient of the refrigerant with the mass flow rate The increase and the decrease of the condensation temperature increase, and the refrigerant thermal resistance gradually decreases, as shown in figure 5. Therefore, as the mass flow rate of the refrigerant increases and the condensation temperature decreases, the ratio of the thermal resistance of the frozen water to the total thermal resistance increases gradually, and the ratio of the refrigerant gradually decreases.

In summary: the experimental conditions in the tube have little effect on the heat transfer coefficient of the frozen water. Under the same mass flow rate, the lower the condensation temperature, the greater the gas-liquid shear force in the tube, and the smaller the thickness of the heat transfer boundary layer, the two factors are beneficial to the increase of the heat transfer coefficient on the refrigerant side and the decrease of the heat transfer resistance in the tube.

5. Conclusion
In this paper, the relationship between total heat transfer coefficient $K$, frozen water heat transfer coefficient $h_w$, refrigerant heat transfer coefficient $h_r$ with mass flow rate and condensing temperature is analyzed, and the thermal resistance of the strengthened tube is analyzed according to the experimental data. The research direction provides a certain experimental basis.

- The prediction errors of the heat transfer coefficient $h$ of the enhanced tube by Cavalini correlation and Koyama correlation are 13.25% and 9.78%, respectively, which can verify the reliability of the data measurement of the experimental bench.
- The total heat transfer coefficient $K$ and the heat transfer coefficient $h_r$ of the refrigerant increase with the increase of the mass flow rate, and the lower the corresponding condensation temperature, the larger the value; the heat transfer coefficient $h_w$ of the frozen water increases with the increase of the mass flow rate. There is a decrease, and the condensation temperature has little effect on its value.
- In the thermal resistance analysis, as the mass flow rate increases, the ratio of the frozen water thermal resistance to the total thermal resistance increases gradually, the refrigerant thermal resistance ratio is always smaller than the frozen water thermal resistance ratio, and the lower the condensation temperature, the difference between the two value is larger.

References
[1] Wu X M and Wang X L 2003 Flow evaporation heat transfer and pressure drop in horizontal micro-fin tubes Chem. Ind. Eng. 54 1215-9
[2] Carey V P 1992 *Liquid-Vapor Phase-Change Phenomena* (Bristol, USA: Taylor & Francis)
[3] Zhang D C, Wang K and Tao W Q 2007 Boiling heat transfer of R134a outside horizontal doubly-enhanced tubes Chem. Ind. Eng. 58 2710-4
[4] Dang C, Jia L and Huang Q 2017 Experimental study on flow boiling heat transfer characteristics of R134a in a rectangular micro-channel J. Eng. Thermophys 38 1327-32
[5] Zhang X Y, Hao P and Zhang Y P 2017 Condensation heat transfer performance for R134a in horizontal annular channels Environ. Eng. 35 161-5
[6] Ouyang X P and Chen Q W 2012 Experimental study of the condensation heat transfer of coolant R134a in a horizontal micro-fin tube J. Eng. Therm. Energ. Power 27 361-5
[7] Li P W, Chen M and Tao WQ 1997 An experimental study on forced-convective condensation heat transfer of R134a inside a horizontal tube J. Eng. Thermophys 18 73-6
[8] Jatuporn K, Kittipong S and Somchai W 2011 Flow boiling heat transfer of R134a in the multiport minichannel heat transfers Exp. Therm. Fluid Sci. 35 364-74
[9] Kittipong S, Jatuporn K and Ahmet S D 2013 Condensation heat transfer characteristics of R-134a flowing inside the multiport minichannels Int. J. Heat Mass Tran. 64 976-85
[10] Al-Hajeri M H and Koluiib A M 2007 Heat transfer performance during condensation of R-134a
inside helicoidal tubes Energ Convers. Manage. 48 2309-15

[11] Tao W Q, Kang H J, Xin R C et al 1997 Determination of thermal resistance of forced convection heat transfer in air cooled tube formation Hvaca 10 64-7

[12] Yang S M and Tao W Q 2006 Heat Transfer 2016 246-7

[13] Min X L and Chao B D 2012 Flow boiling heat transfer of HFO1234yf and R32 refrigerant mixtures in a smooth horizontal tube: Part 1 experimental investigation Int. J. Heat Mass Tran. 55 3437-46

[14] Cavallini A, Col D D, Mancin S et al 2009 Condensation of pure and near-azeotropic refrigerants in microfin tubes: A new computational procedure Int. J. Refrig. 32 162-74

[15] Yu J and Koyama S 1998 Condensation heat transfer of pure refrigerants in microfin tubes Rerig Conf at Purdue 35 325-30

[16] Al-Hajeri M H, Koluib A M, Mosaad M et al 2007 Heat transfer performance during condensation of R-134a inside helicoidal tubes Energ Convers. Manage. 48 2309-15