

**Research Report on Heat Transfer and Pressure
Drop Performance of Small-diameter Heat Exchanger of
Refrigerated cabinet Unit for Refrigerated Cabinet**



University of Shanghai for Science and Technology
Institute of Refrigeration Technology

April 2019

Abstract

The vigorous development of fresh electronic commerce in China has stimulated the demand for refrigerated vehicles and refrigerated cabinet. The popularity of refrigerated cabinet will increase dramatically, which will drive the rapid development of this industry. Under the favorable situation of national policy and market demand, facing the development environment of coexistence of opportunities and challenges, Chinese mobile environmental refrigeration devices will make great progress in various fields through different industries, and the refrigeration system-related manufacturing industry and corresponding manufacturing technology will flourish. At present, most heat exchangers for refrigerated cabinet use copper tube-fin heat exchangers with tube diameters of 9.52 mm and 7.94 mm. If the diameter of copper tubes used in heat exchangers in these fields can be reduced to 5 mm, the replacement of small-diameter copper tubes for heat exchangers can not only save a lot of production costs and significantly reduce refrigerant charge, but also reduce the weight of heat exchangers and improve heat exchangers. Seismic resistance, so as to achieve energy saving and emission reduction and extend the service life of equipment. Obviously, the development of small-diameter heat exchangers suitable for refrigerated cabinet is of great significance to the whole industry.

However, as the diameter of heat exchanger decreases, the performance of refrigeration system will also be affected, especially the heat transfer and pressure drop characteristics of refrigerants in heat exchanger tubes will change, which will lead to the performance of the system, will also change. In order to make the small-diameter heat exchanger be well used in refrigerated cabinet, the necessity and feasibility of replacing the small-diameter heat exchanger are affirmed. It is necessary to analyze the influence of the small-diameter copper tube on the heat transfer and pressure drop performance of the heat exchanger and the performance of the whole system. In this context, it is necessary to design the replacement prototype first, and apply it in the refrigerated cabinet. Then, according to the test data after the replacement of the small-diameter heat exchanger and the working data of the original traditional heat exchanger, the thermal and economic performance advantages of the replacement of the small-diameter heat exchanger are determined. This paper mainly introduces the performance of heat transfer and pressure drop of small-diameter heat exchanger of refrigeration unit for refrigerated cabinet.

This project is commissioned by the Shanghai Representative Office of the International Copper Professional Association of the United States (China). Under the guidance of

Professor Liu Jianhua, it is completed by the collaboration of Yu Xiaoxiao, Wang Haoyu, He Kuan and Feng Guangdong, Postgraduates of refrigeration specialty of Shanghai Polytechnic University. In the course of the project, I was guided and helped by Song Ji, manager of Shanghai Representative Office of American International Copper Professional Association (China).

Catalog

Abstract	2
-----------------------	----------

Introduction	5
Chapter I Theoretical Analysis of Small Diameter Substitution.....	7
Chapter II Study on heat transfer correlation of refrigerant condensation in small tube diameter	10
2.1 Research Status of Refrigerant Condensation in Copper Tube.....	10
2.2 Experimental Device and Data Processing	11
2.2.1 Experimental device	11
2.2.2 Experimental condition.....	14
2.2.3 Data processing.....	14
2.2.4 Uncertainty analysis	15
2.3 Research on the Adaptability of Experimental Data Analysis and Model.....	16
2.3.1 Study on Condensation Heat Transfer Model of Small Tube Diameter.....	17
2.3.2 Study on condensation pressure drop model of small diameter tube	21
Reference.....	24

Introduction

With the development of social economy and the quickening pace of social life, people's pursuit for better quality of life is also increasing. In recent years, the growth demand of mobile environmental refrigeration devices is very strong, which is mainly reflected in the field of cold chain logistics.

The vigorous development of fresh electronic commerce in China has stimulated the demand for refrigerated vehicles. According to the monitoring data of the E-commerce Business Research Center, fresh e-commerce has developed rapidly. The scale of fresh e-commerce has reached 56 billion Yuan in 2015, more than doubling from 26 billion Yuan in 2014, and breaking through 100 billion Yuan in 2017. China is a big agricultural and animal husbandry country, and a big consumer of frozen and cold food and fruits and vegetables. It produces 60 million tons of meat food, more than 10 million tons of quick-frozen food, more than 10 million tons of cold drinks, and 50 million tons of aquatic products, 35,000 tons of vegetables and 61 million tons of fruits annually. Thus, more than 100 million tons of food need to be refrigerated to meet consumer demand. The "13th Five-Year plan" and "one belt and one road" strategy are driving the development of cold chain logistics. The implementation of the food safety law, the cold chain logistics development plan and other important laws and regulations indicates that the government has intensified the management of food safety. At present, the phenomenon of "broken chain" in the transportation of cold chain logistics is more common in China, which affects the overall service quality and security capability of cold chain logistics. The Ministry of Communications hopes that the problem of "broken chain" will be basically solved by 2020. Accordingly, the popularity of refrigerated trucks as food carriers will increase dramatically, which will drive the rapid development of the industry.

All the above phenomena show that, under the favorable situation of national policy and market demand, facing the development environment of opportunities and challenges, Chinese mobile environmental refrigeration devices will be greatly developed through different industries in all aspects, and the refrigeration system-related manufacturing industry and corresponding manufacturing technology will flourish. At present, most heat exchangers for refrigerated cabinet use copper tube-fin heat exchangers with tube diameters of 9.52 mm and 7.94 mm. If the diameter of copper tubes used in heat exchangers in these fields can be reduced to 5 mm, the replacement of small-diameter copper tubes for heat exchangers can not only save a lot of production costs and significantly reduce refrigerant charge, but also

reduce the weight of heat exchangers and improve heat exchangers. Seismic resistance, so as to achieve energy saving and emission reduction and extend the service life of equipment. Obviously, the development of small-diameter heat exchangers suitable for refrigeration cabinet systems is of great significance to the whole industry.

However, the heat transfer and pressure drop characteristics of refrigerant in the copper tube of heat exchanger are directly related to the diameter of the tube. As the diameter of the tube decreases, the main heat transfer and pressure drop properties of refrigerant in the copper tube also change, which leads to the change of the system performance. Therefore, for the refrigeration system heat exchangers of refrigerated cabinet to be replaced by small-diameter copper tubes, not only need to improve efficiency, but also need to solve practical application problems. Before the technology promotion, we need to carry out repeated theoretical research and experimental verification. In order to analyze the necessity and feasibility of the application of small-diameter copper tube in refrigeration system of refrigerated cabinet, it is necessary to systematically analyze the influence of small-diameter copper tube on heat transfer performance and pressure drop performance of heat exchanger and on the performance of the whole system, and to analyze the results according to theory. In this paper, the unique heat transfer and pressure drop characteristics of small-diameter heat exchangers are analyzed, and the structural design process of heat exchangers is described in detail.

Chapter I Theoretical Analysis of Small Diameter Substitution

In the heat exchangers of refrigerated cabinet, if the diameter of tubes is reduced from 9.52 mm or 7.94 mm to 5 mm, can the copper consumption of heat exchangers be effectively reduced and the production cost be saved under the premise of reasonable heat transfer and pressure drop? It is necessary to conduct theoretical analysis of heat transfer characteristics first. The direct influence of small diameter substitution on heat transfer performance of heat exchanger is the increase of heat transfer area per unit copper consumption and the increase of heat transfer coefficient of refrigerant side in tube. The increase of heat transfer area per unit copper consumption of heat exchanger is beneficial to heat transfer of heat exchanger. Under the same heat transfer requirement, the production cost of heat exchanger can be effectively reduced and the miniaturization and lightweight of refrigeration unit can be promoted due to the reduction of copper consumption replaced by small tube diameter. Small diameter substitution increases the heat transfer coefficient of refrigerant side in heat exchanger tube, and also improves the heat transfer performance of heat exchanger. However, when the small diameter is replaced, the channel diameter of heat exchanger decreases. If the overall channel area of refrigerant is not changed, the flow velocity of refrigerant will be greatly increased, the resistance loss along the refrigerant tube will be increased, and the heat transfer performance of heat exchanger will be reduced. In addition, when a small-diameter heat exchanger is used as an evaporator, as the diameter of the heat exchanger decreases, the flow resistance of the refrigerant in the tube increases with the increase of the flow velocity and friction coefficient in the tube, which will lead to the increase of the pressure drop on the side of the refrigerant, which will inevitably lead to the decrease of the evaporation temperature in the evaporator, thus affecting the refrigeration effect of the whole system.

However, how much influence the heat transfer performance of heat exchanger is caused by reducing the pipe diameter, which factors play a key role in the heat transfer performance of heat exchanger after reducing the pipe diameter, and which parameters have little influence, so far, there is no systematic research and analysis. Therefore, after reducing the diameter of the tube, to ensure the original heat transfer performance of the heat exchanger, the shape of the heat exchanger is basically unchanged under the premise of whether it can really save copper consumption has become the primary task of this study.

Taking refrigerated cabinet as examples, this paper studies the replacement of small diameter of heat exchanger. Condenser is one of the main heat exchange equipment of

refrigeration vehicle and refrigerated cabinet. Its main task is to cool the high pressure superheated refrigerant vapor discharged from compressor through environmental medium and condense it into saturated liquid or even super cooled liquid. According to the general structure of the refrigeration cabinet, the condenser selected in this project is the forced ventilation air-cooled fin condenser. Under the premise of keeping the heat transfer capacity and shape size of the condenser basically unchanged, by changing the length and arrangement of the tube-fin heat exchanger, the size and spacing of the fins, the small diameter of the tube can be replaced. At present, parts of refrigerated cabinet that do not require high temperatures above - 10 degrees Celsius use R134a, while most refrigerated cabinet use R404A. In this paper, the heat transfer characteristics of R134a and R404A as refrigerants were studied. From the point of view of theoretical analysis, the differences of heat transfer coefficient, heat transfer area, heat exchange tube length, channel number and copper consumption of condensers with 9.52 mm, 7.94 mm conventional diameter and 5 mm small diameter are preliminarily obtained, which provides a basis for the study of small diameter replacement of heat exchangers. The structural parameters of 9.52 mm, 7.94 mm conventional diameter and 5 mm small diameter internal threaded pipe are listed in Table 1-1 below.

Table1-1 9.52 mm, 7.94 mm and 5 mm internal threaded pipe structural parameters

Outer diameter /mm	9.52	7.94	5
Wall thickness /mm	0.32	0.28	0.2
Helix angle / degree	18	18	18
Tooth vertex angle	50	40	40
Tooth number	70	50	56
Tooth height /mm	0.12	0.12	0.12

The calculation results show that when the diameter of copper tube is reduced from 9.52 mm or 7.94 mm to 5 mm in refrigeration vehicle and refrigerated cabinet condenser, the mass flow rate increases and the equivalent Reynolds number increases with the decrease of the diameter of heat exchanger tube, and the heat transfer coefficient increases by 25-40%. When the diameter of the tube decreases, the equivalent diameter of the narrowest interface on the outer side of the tube decreases, which increases the fin efficiency. With the decrease of the fin spacing, the heat transfer area of the fin increases and the heat transfer coefficient on the outer side of the tube increases, so the total heat transfer coefficient of the heat exchanger increases by 10-20%. Through preliminary pipeline arrangement, the number of branches is increased appropriately, and the length of a single flow path is reduced. The actual copper saving is calculated to be 8-34%.

Chapter II Study on heat transfer correlation of refrigerant condensation in small tube diameter

Small-diameter heat exchangers have the advantages of compact structure, larger contact area per unit volume and smaller refrigerant charge. For HFC and HCFC refrigerants widely used at present, although the heat transfer coefficient can be significantly increased with the decrease of the diameter of heat exchangers, the following problem is the increase of pressure drop in the process of heat transfer. However, in the theoretical calculation process of using 5mm small diameter copper tube for condenser, the reference heat transfer models of refrigerant R134a and R404A in 5mm horizontal threaded copper tube are rare, and the relevant literature is very rare. At present, there is still no mature correlation which can be applied in practice to obtain the heat transfer coefficient in the tube under the corresponding working conditions, so it is still questionable to verify that the substitution of small diameter can save copper consumables by theoretical calculation alone, and a pertinent heat transfer model is also needed after it is applied to the market.

However, because there are too many factors affecting the condensation heat transfer of R134a and R404A refrigerants, it is very difficult to obtain the correlations of condensation heat transfer suitable for R134a and R404A refrigerants in 5mm internal screw copper tubes by theoretical analysis. Therefore, the research group chose to conduct the condensation experiments of R134a and R404A in 5mm internal screw copper tubes, and obtain the pertinent heat transfer based on the combination of experiment and theoretical research. The correlation provides an accurate theoretical basis for the design of the new heat exchanger and the whole machine.

2.1 Research Status of Refrigerant Condensation in Copper Tube

At present, the research of refrigerant condensation heat transfer in copper tubes with conventional diameter has a mature foundation at home and abroad.

Qin Yan and others used smooth copper tube with outer diameter of 7mm and inner thread tube as condensers to carry out experiments. The effects of two kinds of copper tube condensers on the performance of the refrigeration system were analyzed. It was concluded that the energy efficiency ratio of the inner thread tube system was 3.3% higher than that of the smooth copper tube system. Cavallini et al. proposed a heat transfer correlation for inner threaded tubes based on the condensation heat transfer correlation in optical tubes, and expressed that the tube diameter, thermal conductivity, surface tension, mass flow rate,

dynamic viscosity, tooth height, tooth apex angle, helix angle, tooth number and other geometric parameters of copper tubes would have an impact on condensation heat transfer in tubes. Greco A et al. explored the heat transfer performance of R404A and R410A in 6mm horizontal stainless steel tube through experiments, and analyzed the influence of evaporation pressure, heat flux density and mass flux density on the heat transfer performance. Ren Fan et al. carried out an experimental study on the flow condensation pressure drop of R410A-oil mixture in a 5mm small diameter inner threaded copper tube. The experimental results show that the friction pressure drop in 5mm tube increases with the increase of mass flow density and vapor quality. Boissieux X et al. studied the condensation heat transfer coefficients of R404A in optical tubes, and screened out the correlations with good prediction accuracy.

In addition, many scholars at home and abroad have studied the condensation heat transfer of refrigerants in copper tubes. By referring to a large number of literatures and drawing on the rich experience of predecessors, the research group has built a refrigerant condensation heat transfer test-bed in a 5mm single tube to verify the adaptability of the existing but not directly adapted heat transfer and pressure drop correlations in the small diameter of tubes and revise them. The small-diameter heat exchangers for refrigerated cabinet provide theoretical guidance.

2.2 Experimental Device and Data Processing

2.2.1 Experimental device

The research group designed and built an experimental platform for condensation heat transfer in refrigerant tube, which was used to study the condensation heat transfer of different refrigerants in different specifications of heat exchanger tubes. As shown in Figure 2-1, the experimental system consists of three parts: refrigerant cycle system, water-borne refrigerant cycle system and data acquisition system.

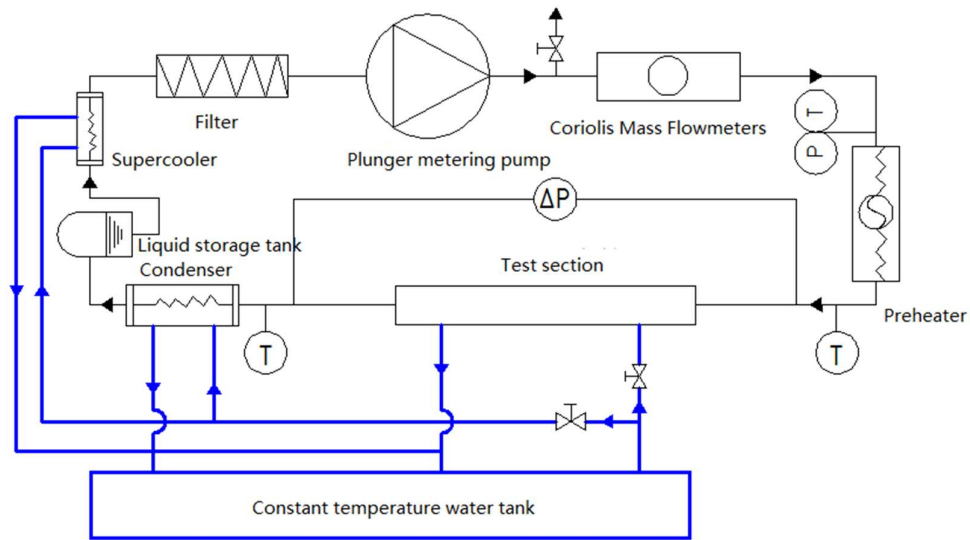


Figure 2-1 schematic diagram of the experimental system

(1) Refrigerant cycle system

The refrigerant circulation system is composed of electric heater, test section, condenser, liquid storage tank, super-cooler, plunger metering pump, Coriolis mass flowmeters, valves and connecting pipe. The piston metering pump replaces the compressor to provide power for refrigerant flow, avoiding the influence of lubricant on the experimental results, and the refrigerant flow can also be adjusted by the stroke knob of the piston pump. The circulating flow rate of the liquid working substance after the plunger pump is measured by Coriolis mass flowmeter, and then it enters the electric heater, heating and evaporating into gas. The vapor quality of the refrigerating working substance at the entrance of the test section is adjusted by adjusting the power of electric heating. Then the refrigerant vapor enters the experimental section to condense, then enters the secondary condensation of the condenser and controls the pressure, then enters the liquid storage tank, which can ensure that the refrigerant in the circulating system is sufficient, and also has the function of cushioning the pressure change of the system. The refrigerant from the liquid storage tank enters the re-heater after super cooling and then flows into the plunger metering pump again. The system completes a cycle.

(2) Refrigerant Cycle System

The refrigerant-carrying circulating system includes refrigeration unit, electric heating, constant temperature water tank, condenser, re-cooler, plate heat exchanger, circulating pump, temperature controller and other components. The refrigeration unit, electric heating and temperature controller ensure that the water in the water tank is constant temperature.

The constant temperature water tank is divided into three ways to provide cooling capacity for three parts. The refrigerant can be condensed in the experimental section by providing cooling capacity for the experimental section; the refrigerant can be supplied with cooling capacity for the condenser and controlling the pressure of the system; and the refrigerant can be supplied with cooling capacity for the re-cooler to ensure that the refrigerant is completely condensed into a single phase. The refrigerant circulating system can adjust the flow rate of water and realize water supply at any temperature of 10-40 C.

(3) Data Acquisition System

The data acquisition system can realize the functions of acquisition, storage, monitoring and data processing of physical parameters in the experimental process, including hardware and software. The sensor, acquisition instrument and computer constitute the hardware part. The system uses T-type thermocouple and capacitive pressure sensor to measure the required temperature and pressure respectively. The mass flow rate of refrigerant is measured by the Coriolis mass flowmeter. The development platform of the experimental acquisition software is VB6.0 programming language. The input and output function of I/O port is realized by using DLL file loaded by VB.

In order to effectively inspect the surface heat transfer coefficient and avoid the entrance effect ($L/D > 60$), the length of the test section is set at 1500mm. The experimental section is counter-current sleeve, refrigerant flows in the inner tube, refrigerant water flows between the outer tube and the inner tube to cool the refrigerant, and the outer tube is made of PVC tube. The inner layer of the insulation material outside the PVC pipe is PEF insulation material (polyethylene foam), the thermal conductivity is less than $0.04 \text{ W}/(\text{m} \cdot \text{K})$, the outer layer is glass wool, and the thermal conductivity is $0.052 \text{ W}/(\text{m} \cdot \text{K})$.

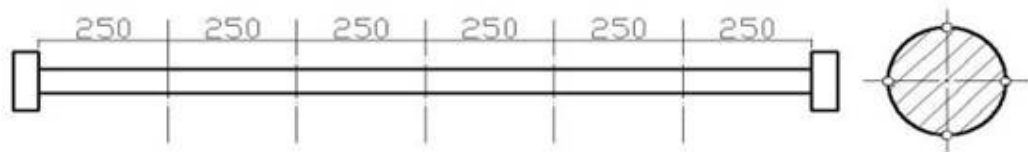


Figure 2-2 Layout of Measuring Points in Test Section

In order to ensure the accuracy of temperature measurement on the outer wall of the inner copper tube, the copper tube is evenly divided into six sections. The temperature of the upper, lower, left and right thermocouples at each node is measured simultaneously and then the average value is calculated. Twenty T-type thermocouples are installed on the copper

tube under test, as shown in Fig. 2-2. The accuracy of the thermocouple has been calibrated before use, and it is confirmed that the temperature is less than 0.3°C.

2.2.2 Experimental condition

Because the mass flow rate, condensation temperature and inlet vapor quality have great influence on the condensation of refrigerant in copper tube, considering the above factors, the test conditions are determined as shown in Table 2-1. When the system is relatively stable (thermal balance error is less than 5%), the experimental data are recorded.

Table 2-1 Test Conditions

Test parameters	Test range	Company
Test tube diameter	5	mm
Length of test section	1500	mm
Density of mass flow	200~ 700	Kg/(m ² ·s)
Condensation temperature	40~ 60	Temperature
Inlet vapor quality	0.1~ 0.9	/

2.2.3 Data processing

(1) Condensation pressure drop

Condensation pressure drop refers to the pressure difference between the inlet and outlet of refrigerant in the test section. In this experiment, the condensation pressure drop can be measured by the pressure difference sensor installed in the test section.

(2) Entrance vapor quality

Vapor quality refers to the percentage of dry vapor contained in wet vapor per kilogram. The entrance vapor quality of refrigerant in the experimental test section, that is, the outlet vapor quality of electric heating, can be obtained by the energy balance of electric heating.

Heat from electric heating of refrigerant in preheating section come from two parts, Q_{pre} latent heat Q_{sens} and sensible heat Q_{lat}

$$Q_{pre} = Q_{sens} + Q_{lat} \quad (1)$$

$$Q_{sens} = m_r c_{p,r} (t_{sat} - t_{pre,i}) \quad (2)$$

$$Q_{lat} = m_r (h_s - h_f) x_{pre,o} \quad (3)$$

Vapor quality of refrigerant at entrance of test section

$$x_{in} = x_{pre,o} = \frac{Q_{pre} - Q_{sens}}{m_r (h_s - h_f)} \quad (4)$$

(3) Heat transfer coefficient

Firstly, the heat transfer coefficient of water side is calculated according to Nusselt number, and then the heat transfer coefficient of refrigerant side is calculated by the method of separation thermal resistance.

Total Heat Transfer Coefficient of Test Section

$$K = \frac{(Q_r + Q_w)/2}{A_0 \cdot \Delta T_m} \quad (5)$$

Logarithmic Mean Temperature Difference of Water in Experimental Section

$$\Delta T_m = \frac{\Delta T_{max} - \Delta T_{min}}{\ln(\Delta T_{max}/\Delta T_{min})} \quad (6)$$

Using Gnielinski empirical correlation to obtain Nusselt number on water side

$$N_u = \frac{(f/2)(R_e - 1000)P_r}{1 + 12.7\sqrt{f/2}(P_r^{2/3} - 1)} \left(\frac{\mu_f}{\mu_w}\right)^{0.14} \quad (7)$$

From Petukhow Formula

$$f = (1.58 \ln R_e - 3.28)^{-2} \quad (8)$$

After calculating the heat transfer coefficient on the water side according to the Nusselt number on the water side, the heat transfer coefficient on the refrigerant side is obtained by the thermal resistance separation method.

$$h_r = \frac{1}{A_i \left(\frac{1}{kA_0} - \frac{1}{h_w A_0} \right)} \quad (9)$$

2.2.4 Uncertainty analysis

The data measured in the experiment are not completely accurate. There are random errors and systematic errors in the measurement process. Because of the deviation of the accuracy of the measuring instrument, the read data is deviated from the true value. And some data need to be calculated twice, and the errors of the basic data will be accumulated and transmitted to the exported data.

Table 2-2 Error Analysis of Experimental System

Experimental parameters	Measuring device	Maximum error
Refrigerant mass flow rate	Mass flowmeter	0.48%
Volume flow rate of cooling water	Electromagnetic flowmeter	1.86%
Temperature	T-type thermocouple	0.1℃
Pressure	Pressure transducer	0.38%
Heat flux density	Calculated value	1.94%
Condensation heat transfer coefficient	Calculated value	3.71%

In this paper, Moffat's error transfer analysis method is used to analyze the reliability of experimental data and calculate the uncertainty of each parameter. The uncertainty of each experimental parameter is summarized as shown in Table 2-2.

2.3 Research on the Adaptability of Experimental Data Analysis and Model

According to the mechanism that the heat transfer performance of inner threaded tube is better than that of smooth tube, different scientists have drawn different conclusions. Yu. J. et al. pointed out that for copper tubes with the same inner diameter, the local heat transfer performance of inner threaded tubes is more than twice that of smooth tubes, and the enhancement of heat transfer is attributed to the increase of heat transfer area. Kedzierski M A et al. proposed a scheme for measuring the local convective condensation heat transfer of four refrigerants (R134a, R410A, R125 and R32) in an internal screw tube. Based on the above literature, it is considered that there are two reasons for the enhancement of heat transfer in inner threaded tubes:

1. The inner threads have a certain surface roughness, which enhances the flow disturbance, resulting in the uneven distribution of fluid around the tubes and the occurrence of turbulence;
2. The increase of the effective area of the inner surface enhances the convective heat transfer.

For the characteristics of flow condensation pressure drop in smooth tube and intensified tube, Cavallini et al., Miyara et al., Wijaya and Spatz, Eckels and Tesene, Goto et al., Hart and Lee and Kedzierski and Goncalves have been studied. The diameter of the condenser studied ranges from 4 to 14.61 mm. Refrigerants include R22, R134a, R12 and R410A. These studies show that: 1. Friction pressure drop in intensified tube is 1.2-2.1 times of heat transfer coefficient in smooth tube with the same diameter under the same working condition, and the bigger the diameter of tube, the more obvious the effect of pressure drop in intensified tube is; 2. The diameter of tube has an effect on the flow pattern transformation, which results in different friction pressure drop characteristics of heat exchanger tubes with different diameter.

Undoubtedly, the heat transfer performance of inner-threaded tubes is much stronger than that of light tubes. Therefore, the heat transfer performance of inner-threaded tubes can be predicted on the basis of heat transfer performance of light tubes. The existing correlations of light tubes should also be investigated when selecting correlations. The heat transfer correlations of inner-threaded copper tubes can be obtained by multiplying a correction factor

on the basis of light tubes. Based on the principle of preferential selection of refrigerants with similar properties and tube diameters, a more suitable correlation is selected for verification and correction.

Louay M. Chamra et al. proposed the condensation heat transfer model of pure refrigerant in internal threaded tubes. The average absolute deviation (MAD) can be used as a criterion to consider the effectiveness of the heat transfer model. MAD is defined as the average normalized difference between the heat transfer coefficient predicted by correlation and the heat transfer coefficient obtained by experiment.

$$MAD = \frac{1}{N} \sum \frac{|h_{exp} - h_{calc}|}{h_{exp}} \quad (10)$$

The model is used to demonstrate the availability of the proposed correlation between heat transfer and pressure drop. The deviation is expressed as follows.

$$\varepsilon = \frac{h_{calc} - h_{exp}}{h_{exp}} \quad (11)$$

It can also be used as one of the principles for selecting correlation. In this study, the basic definition of MAD is used as the judgment basis for the applicability of the correlation. If the deviation between the predicted value and the experimental value of the correlation is less than 20%, the correlation is considered to be selected.

2.3.1 Study on Condensation Heat Transfer Model of Small Tube Diameter

Based on the principles and methods mentioned above, a large number of correlations are carefully compared and repeatedly screened from the applicable refrigerants, structural parameters of copper tubes and applicable operating conditions. Finally, Dobson and Chato condensation heat transfer calculation model is selected as the object of verification and correction based on the comparative analysis of experimental parameters.

The Dobson and Chato's condensation heat transfer calculation model mainly considers the condensation heat transfer process including annular flow and wavy flow. It is considered that the two flow patterns cover a wide range in the whole condensation process. Gravity plays a dominant role in annular flow and shear stress plays a dominant role in wavy flow. The two-phase heat transfer coefficient of refrigerant in a horizontal smooth tube is discussed in this model. The heat transfer coefficient of annular flow is calculated by formula (12-15).

$$Nu = 0.023 Re_L^{0.8} Pr_L^{0.4} \left(1 + \frac{2.22}{X_{tt}^{0.89}}\right) \quad (12)$$

$$Re_L = \frac{G(1-x)D}{\mu_L} \quad (13)$$

$$Pr_L = \frac{C_{p,L}\mu_L}{\lambda_L} \quad (14)$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_L}\right)^{0.5} \left(\frac{\mu_L}{\mu_v}\right)^{0.1} \quad (15)$$

The film condensation at the top of the tube and the forced convection condensation at the bottom of the tube are considered simultaneously. The heat transfer coefficient of wave flow can be calculated by equation (16-20):

$$Nu = \frac{0.2Re_{vo}^{0.12}}{1 + 1.11X_{tt}^{0.58}} \left(\frac{G_a Pr_L}{Ja_L}\right)^{0.25} + \left(1 - \frac{\theta_L}{\pi}\right) Nu_{forced} \quad (16)$$

$$Nu_{forced} = 0.0195Re_L^{0.8} Pr_L^{0.4} \sqrt{1.376 + \frac{c_1}{X_{tt}^{c_2}}} \quad (17)$$

C_1C_2 is defined as follows:

$$\begin{aligned} 0 < Fr_L < 0.7 \quad & \begin{aligned} c_1 &= 4.172 + 5.48Fr_L - 1.564Fr_L^2 \\ c_2 &= 1.773 - 0.169Fr_L \end{aligned} \\ Fr_L > 0.7 \quad & \begin{aligned} c_1 &= 7.242 \\ c_2 &= 1.655 \end{aligned} \end{aligned}$$

In addition, $\left(1 - \frac{\theta_L}{\pi}\right)$ is approximately equal to $(\arccos(2\alpha - 1)/\pi)$ in Formula (16)

The radian angle, called porosity, is defined as the ratio of the area occupied by steam to the total cross-sectional area of the channel in a two-phase flow. Its physical meaning is shown in Figure 2-3.

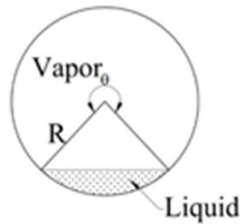


Figure 2-3 Flow Occupancy Area of Channel Section

Reynolds number for a single gas phase

$$Re_{vo} = \frac{GD}{\mu_v} \quad (18)$$

Galileo number

$$Ga = \frac{g\rho_L(\rho_L - \rho_v)D^3}{\mu_L^2} \quad (19)$$

Jakob number in liquid phase

$$Ja_L = \frac{C_{p,L}(T_{sat} - T_w)}{h_{fg}} \quad (20)$$

The boundary between annular flow and wavy flow can be determined by the modified Froude number. Fr_{so} is defined:

$$Rr_{so} = c_3 Re_L^{c_4} \left(\frac{1 + 0.09X_{tt}^{0.039}}{X_{tt}} \right)^{1.5} \frac{1}{Ga^{0.5}} \quad (21)$$

	$C3=0.025$
$Re < 1250$	$C4=1.59$
	$C3=1.26$
$Re > 1250$	$C4=1.04$

The annular flow equation (12) can be applied to all vapor quality values of $G > 500$ kg/m²·s. For $G < 500$ kg/m²·s, equation (16) is applicable to wavy flow for $Fr_{so} < 20$.

And h is available.

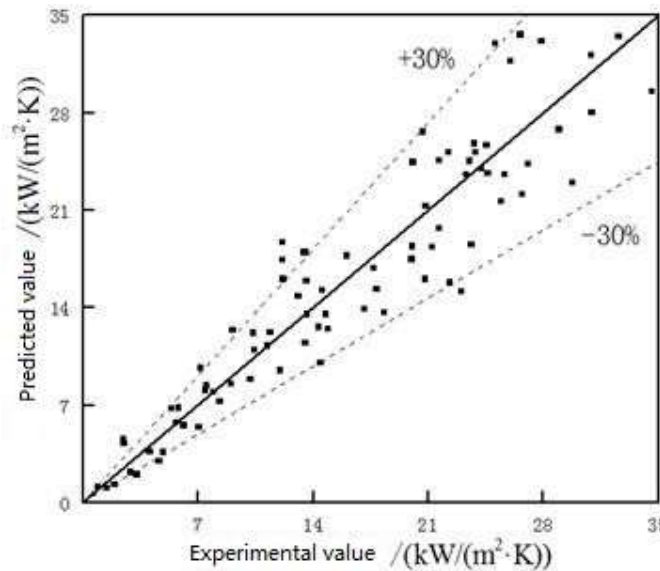
The range of test conditions applicable to Dobson and Chato's condensation heat transfer calculation model is shown in Table 2-3.

Table 2-3 Dobson and Chato's Model Applicable Test Conditions

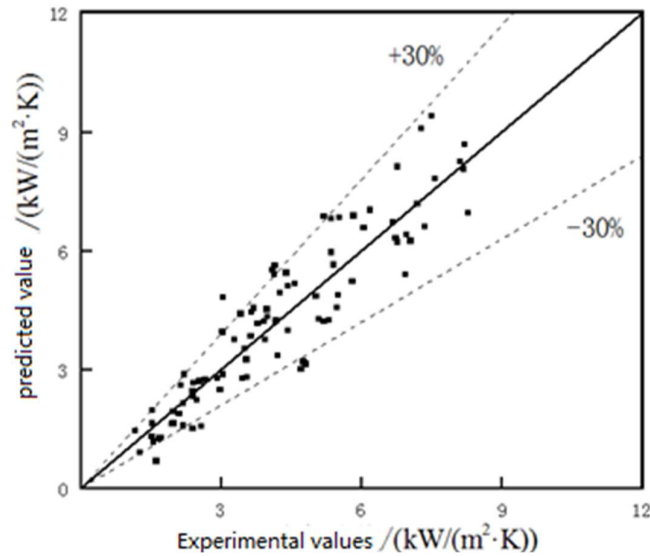
Applicable test conditions	Range
Density of mass flow	$25 < G < 800$ kg/m ² ·s
Heat flux density	$5 < q < 15$ kW/m ²
Vapor quality	$0.1 < x < 0.9$
Condensation temperature	$35 < T_{sat} < 45$ °C
Pipe diameter	$3.14 < D < 7$ mm

Because this correlation is based on optical tube, and this research mainly focuses on internal threaded tube, so it needs to be revised. The heat transfer coefficients of R134a and R404A in the 5 mm internal threaded tube are calculated from the experimental data. The experimental parameters are directly substituted into the Dobson and Chato's correlation. The linear relationship between the experimental results and the model results is found to fit a new correlation.

Through the analysis of experimental data, it is found that the experimental heat transfer coefficient of R134a in small diameter is about 1.3 times of that calculated by Dobson and Chato's correlation under the conditions of mass flow density ranging from $500 \text{ kg}/(\text{m}^2 \cdot \text{s})$ to $700 \text{ kg}/(\text{m}^2 \cdot \text{s})$ and inlet vapor quality of test section ranging from 0.1 to 0.9. On the basis of experiments, the correction factor of the correlation is determined to be 1.32 by multiple linear regression of least square method. Similarly, the experimental heat transfer coefficient of R404A in a small diameter tube is about two times as much as that calculated by Dobson and Chato's correlation in the mass flow density range of $200\text{--}400 \text{ kg}/(\text{m}^2 \cdot \text{s})$ and the inlet vapor quality of the test section of 0.1–0.9. On the basis of experiments, the correction factor of the correlation was determined to be 2.05 by using the least square method and multiple linear regression.



(a) Model deviation of R134a



(b) Model deviation of R404A

Figure2-4 Deviation between predicted and experimental values of heat transfer coefficient by graph correlation

Whether the new correlation is applicable or not requires further verification of the deviation between the experimental and theoretical values, as shown in Figure 2-4. The deviation statistics show that the prediction error of R134a model is within (+30%) as high as 82.8%, and the average absolute deviation MAD is only 15.2%. For R404A model, the prediction error is 87.6% and the average absolute deviation MAD is only 18.2%. From the data point of view, the prediction accuracy of these two models is relatively high.

2.3.2 Study on condensation pressure drop model of small diameter tube

Based on the principle and method of selecting correlation formulas before, a large number of correlation formulas are carefully compared and repeatedly screened from the applicable refrigerants, structural parameters of copper tubes and applicable operating conditions. Based on the comparative analysis of experimental parameters, Huang's condensation pressure drop model is selected as the object of verification and correction.

Huang's correlation of frictional pressure drop in small-diameter inner-threaded tubes is based on the condensation pressure drop characteristics of R410A refrigerant in small-diameter intensified tubes as a condensation pressure drop model. The test working fluid is R410A refrigerant with RB68EP lubricant. The existence of lubricating oil will cause some changes in the physical properties of refrigerant, such as dynamic viscosity, surface tension, specific heat and density, and will also affect the pressure drop characteristics. The calculation formulas are as follows:

$$\Delta P_{r,o,frict} = \frac{2lf_v G^2 x_{r,o}^2 (1 + 1.457 X_{tt}^{0.604})^2}{\rho_{r,v} d_h} \quad (22)$$

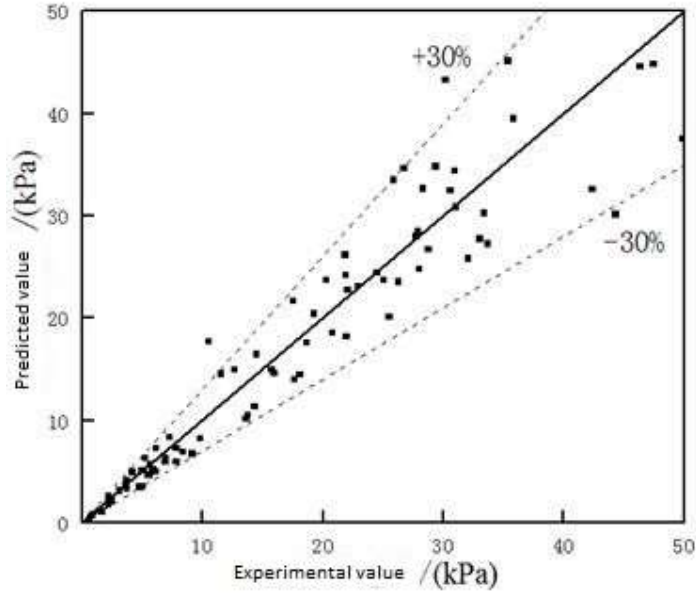
$$X_{tt} = \left(\frac{1 - x_{r,o}}{x_{r,o}} \right)^{0.9} \left(\frac{\rho_{r,v}}{\rho_{r,o,L}} \right)^{0.5} \left(\frac{\mu_{r,o,L}}{\mu_{r,v}} \right)^{0.1} \quad (23)$$

$$f_v = \frac{0.128}{Re_{h,v}^{0.267}} \quad (24)$$

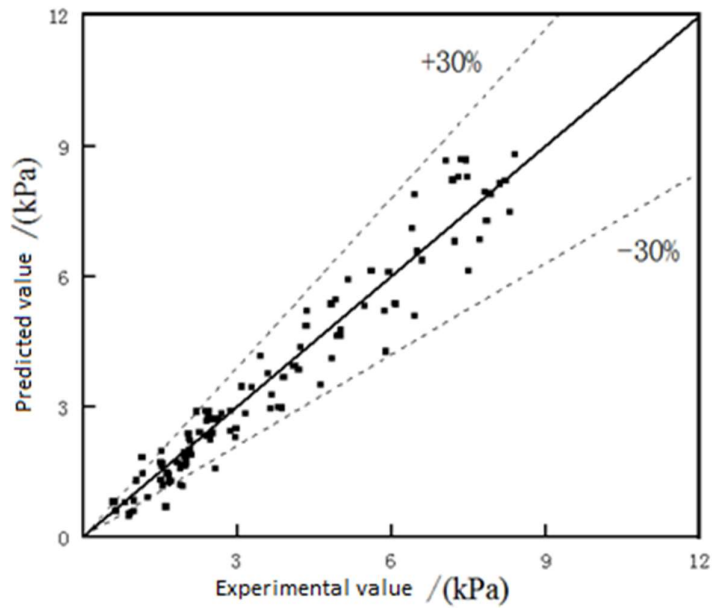
$$Re_{h,v} = \frac{G x_{r,o} d_h}{\mu_{r,v}} \quad (25)$$

The applicability of the above correlations is as follows: tube diameter 5mm, mass flow density 200-500 kg/ (m²•s), heat flow density 4.21-7.91 kW/m², steam vapor quality 0.3-0.9, oil concentration 0-5%.

The refrigerants in this study are R134a and R404A. It is not enough to replace the physical parameters of R410A with those of R134a and R404A in the modification of the correlation formula, and the correlation formula needs to be further modified by experimental fitting method. The results show that a small amount of lubricant in the refrigerant can reduce the friction pressure drop slightly. The pressure drop of R134a and R404A in the 5 mm internal threaded tube under different mass flow density and inlet vapor quality of refrigerant test section is obtained from the experimental data. The same experimental conditions are directly substituted into the model of pressure drop correlation proposed by Huang to calculate a theoretical predicted condensation pressure drop value. The numerical comparison shows that in the mass flow density range of 500-700 kg/ (m²•s), the test section enters. Under the condition of 0.1-0.9 mouth vapor quality, the experimental R134a condensation pressure drop is about 98% of the value calculated by Huang's overpressure drop correlation. On the basis of the experiment, the correction factor of the correlation is determined to be 0.982 by the least squares multiple linear regression method. Similarly, under the conditions of mass flow density ranging from 200 to 400 kg/(m²•s) and entrance vapor quality of test section ranging from 0.1 to 0.9, the experimental R404A condensation pressure drop is about 95% of that calculated by Huang's overpressure drop correlation. On the basis of the experiment, the correction factor of the correlation is determined to be 0.948 by the least squares multiple linear regression method.



(a) Model deviation of R134a



(b) Model deviation of R404A

Figure 2-5 The deviation between predicted and experimental values of heat transfer coefficient

Whether the new correlation is applicable or not requires further verification of the deviation between the experimental and theoretical values, as shown in Figure 2-5. The deviation statistics show that the prediction error of R134a model is less than (+30%) and the average absolute deviation MAD is only 17.4%. For R404A model, the prediction error is less than (+30%) and the average absolute deviation MAD is only 14.7%. From the data point of view, the prediction accuracy of these two models is relatively high.

Reference

- [1] 秦妍, 张剑飞. 关于内螺纹管及光管冷凝器换热效果的实验分析[J]. 制冷与空调, 2011, 11(1): 59-61.
- [2] CAVALLINI A, COL D D, DORETTI L, et al. Heat transfer and pressure drop during condensation of refrigerants inside horizontal enhanced tubes [J]. International journal of refrigeration, 2000, 23(1): 4-25.
- [3] GRECO A, VANOLI G P. Flow boiling heat transfer with HFC mixtures in a smooth horizontal tube. Part I: Experimental investigations [J]. Experimental Thermal&Fluid Science, 2005, 29(2): 189-198.
- [4] 任凡, 丁国良, 黄翔超, 胡海涛, 邓斌, 等. 高屹峰.R410A-油混合物在 5mm 内螺纹强化管内流动冷凝的摩擦压降实验研究[J]. 化工学报, 2008, (第 A2 期).
- [5] Boissieux x, Heikal M R, Johns R A. Two—phase heat transfer coefficients of three HFC refrigerants inside a horizontal smooth tube, part II: Condensation [J]. International Journal of Refrigeration, 2000, 23(4): 269-283.
- [6] APREA C, GRECO A, VANOLI G P. Condensation heat transfer coefficients for R22 and R407C in gravity driven flow regime within a smooth horizontal tube [J]. International journal of refrigeration, 2003, 26(4): 393-401.
- [7] Moffat R J. Describing the uncertainties in experimental results [J]. Experimental Thermal & Fluid Science, 1988, 1(1): 3-17.
- [8] Yu J, Koyama S. Condensation heat transfer of pure refrigerants in micro-fin tube [J]. British Journal of heatmatology, 2007, 139(1): 281-288.
- [9] Kedzierski M A, Goncalves J M. Horizontal convective condensation of alternative refrigerants within a micro-fin tube [J]. Journal of Enhanced Heat Transfer, 1999, 6(2): 161-178.
- [10] F.W. Dittus, M.L.K. Boelter. Heat transfer in automobile radiators of the tubular type, University of California. Publications on Engineering, Berkeley, 1930, CA 2(13): 443.
- [11] Churchill, S.W., "Comprehensive correlating equations for heat, mass, and momentum transfer in fully developed flow in smooth tubes, " Ind. Eng. Chem. Fundam, 1977(16):109-116.
- [12] V. Gnielinski. New equations for heat and mass transfer in turbulent pipe and channel flow. International Chemical Engineering,, 1976 (16): 359-368.
- [13] Elmahdy A H, Biggs R C. Finned tube heat exchanger: Correlation of dry surface heat transfer data [J]. ASHRAE Transactions, 1979, 85(2): 262-273.
- [14] Chen Z Q, Ren J X. Effect of fin spacing on the heat transfer and pressure drop of a two-row plate fin and tube heat exchanger [J]. International journal of refrigeration, 1988, 11(6): 356-360.
- [15] Huang Jeng-Min, Hsieh Wen-Chien, Ke Xin-Ji, Wang Chi-Chuan. The effects of frost thickness on the heat transfer of finned tube heat exchanger subject to the combined influence of fan types [J]. Applied Thermal Engineering, 2008, 28(7): 728-737.