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Condensation heat transfer of HFC134a on horizontal low thermal conductivity tubes ☆

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Abstract

Experimental studies of film condensation of HFC134a and CFC12 on single horizontal smooth tube and three enhanced tubes have been conducted. The objective is to obtain the basic data for film condensation of HFC134a on low thermal conductivity enhanced tubes. The results indicate that the predicted condensation heat transfer coefficients of HFC134a and CFC12 on smooth tube from Nusselt theory agree with the experimental data within ±10%. The condensation heat transfer coefficients of HFC134a are about 32.6% larger than that of CFC12 for the cupronickel Thermoexcel-C tube. The cupreous integral-fin tubes provide more than twice condensation heat transfer coefficients than that of the cupronickel Thermoexcel-C tube. The heat transfer coefficients for each integral-finned tube are also compared with existing single-tube models.

Keywords: Film condensation; HFC134a refrigerant; Enhanced tubes; Heat transfer coefficient

1. Introduction

The film-wise condensation outside horizontal tubes frequently occurs in the shell-and-tube condenser of air-conditioning systems. For the past decades, CFC12 had been extensively used in air-conditioning fields due to their excellent thermodynamic characteristics. Due to its hazard to the environment according to the Montreal protocol signed in 1987 and its subsequent amendments, CFC12 has been completely phased out up to 2006 in the world. Because of its thermophysical properties and environmental protection character HFC134a has been selected as one of the new alternative refrigerants of CFC12 [1], and has been widely used in compact air-conditioning systems.

The studies on the film condensation outside horizontal tubes have been conducted for more than half a century, and many valuable results have been obtained. It has been observed by many investigators that the condensation heat transfer coefficient of the traditional refrigerants such as CFC12 can be increased many folds by providing low integral fins and other enhanced surface structures on the outer surface of a tube [2–4]. Experimental study of film condensation of HFC134a was conducted in references [5–8]. It has been generally accepted that the condensation heat transfer of HFC134a is better than that of CFC12. For example, the condensation heat transfer performance of R134a on a horizontal tube was conducted in [5] and three various enhanced tubes, a plain tube, low fin tube and Turbo-C tube,

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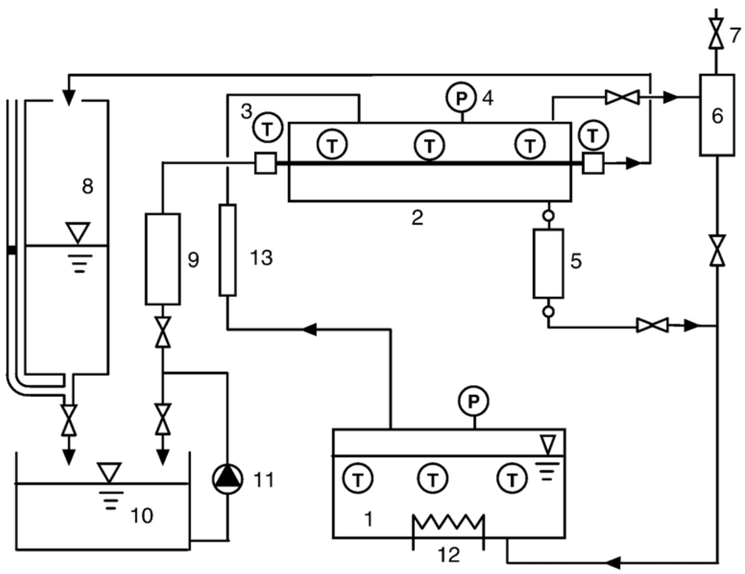
were studied. It is found that the condensation heat transfer coefficients of HFC134a on the three studied tubes are 0.0–31.8% higher than those of CFC12. For the condensation on a horizontal tube with low integral fin the optimum fin-per-meter for CFC12 and HFC134a is different because of their differences in the thermophysical properties. It is found in [6] that for HFC134a the best performance can be obtained from a tube with 1560 fpm (fin-per-meter), whose EF is about 5.6. Xie and Eckels [7] studied two enhanced tubes on HFC134a, one a two-dimensional (2D) finned tube and the other a three-dimensional (3D) enhanced tube. Specifically, the 2D tube had heat transfer coefficients that were 8.0 times better than the smooth tube, whereas enhancements of 11.8 times were obtained in the 3D tube. Gstoehl and Thome [8] studied the effect of condensate inundation on the thermal performance of a vertical array of horizontal tubes with plain and enhanced surfaces on HFC134a. At low liquid inundation rates, the tubes with 3D enhanced surface structures significantly outperformed the low finned tube. Increasing liquid inundation deteriorated the thermal performance of the 3D enhanced tubes, whereas the inundation had nearly no effect on the low finned tube.

It should be noted that most of the previous condensation heat transfer studies were conducted for tube materials with high thermal conductivity such as copper. In some special occasions, materials with low thermal conductivity such as cupronickel were used for the purpose of antisepsis. Mills et al. studied one special trapezoidal-fin tube (1420 fpm) which was made of copper, brass, copper/nickel (70–30) for condensation of steam [9]. It is found that compared with the smooth tubes of the same material the heat transfer enhancement factors for tubes of Cu, Br, Cu/Ni were 4.0, 3.6, 2.6, respectively, under the same temperature difference between the root wall temperature and the vapor saturation temperature. This implies that the thermal conductivity of the condensation tube is of a crucial importance. Marto also found that tube wall thermal conductivity was very important on condensation heat transfer [4]. Briggs and Rose found that the best fin spacing is more strongly dependent on fin thermal conductivity [10]. For some special cases such as for antisepsis the usage of tubes with low thermal conductivity is necessary. From the basic heat transfer theory, the importance of the tube thermal conductivity is expected, because the condensation usually has quite high heat transfer coefficient, hence its thermal resistance in the transmission of thermal energy may be comparable with the tube wall thermal resistance. However, the quantitative information of condensation on tubes with low thermal conductivity has to be determined by experiments. In the authors' knowledge, condensation on the lower thermal conductivity tube was not studied on HFC134a so far.

In this paper, experimental study results of film condensation heat transfer of HFC134a outside two horizontal cupreous integral-fin tubes, one cupronickel smooth tube, and one cupronickel Thermoexcel-C tube are presented. Comparisons with refrigerant CFC12 are conducted for the cupronickel tubes. In the following, the experimental apparatus will first be



(1)boiler; (2)condenser; (3)thermocouple;(4)pressure gauge; (5)condensate measuring container; (6)after-condenser; (7)flushing vent; (8)water rate measuring tank; (9)water flow meter; (10)water storage tank; (11)pump; (12)electric heater; (13)superheater

Fig. 1. Schematic diagram of experimental apparatus.

introduced, followed by the data reduction description and the uncertainty analysis. The test results for the smooth tube will first be compared with the well-known Nusselt solution to show the reliability of the test system, and then details of measurement results for the enhanced tubes will be provided. Finally some practical conclusions will be drawn.

2. Experimental apparatus and procedure

Fig. 1 is a schematic of the apparatus of the present study, which consists of a boiler, a condenser, an after-condenser, a water-cooling system and a measurement system. The condenser, the boiler and the vapor piping are made of stainless steel. The boiler has an inner diameter of 257 mm and a length of 1100 mm. It is fitted with four immersion heaters of 4.5 kW each. A stainless steel tube of 50 mm inner diameter connects the boiler to the condenser shell. Saturated vapor is generated in the boiler and passes through a superheater to the condenser. The inner diameter of the condenser shell is 147 mm and its length is 1500 mm. Four condensing tubes can be mounted in parallel within the shell. The saturated or slightly superheated vapor is admitted to the top part of the condenser shell. A specially fabricated metal sheet with a large number of small holes is mounted in the upper part of the shell to distribute the vapor uniformly along the test tube. After the condensation of the refrigerant vapor on the test tube, the condensate returns to the boiler by gravity. In the condensate return line, a box of fixed volume is mounted for measuring condensate flow rate. In order to remove noncondensable gas from the test section, an after-condenser is placed downstream from the test section.

The cooling water loop consists of a storage tank, a centrifugal pump, a flow rate measuring tank, a refrigeration system, and an electric heater. The condensing tubes mounted in the condenser shell are connected to the main circulating line in a parallel manner. The apparatus is well insulated.

Now the measurement system is introduced. Copper-constantan thermocouples are used to measure the temperatures of vapor in boiler and condenser, and the inlet water temperature. Since the coolant temperature rise is a very important measurement in this study, a six-junction, series-connected copper-constantan thermocouple is used for its measurement. A Keithley voltmeter having a resolution of 0.1 μV is used to measure the thermocouple emfs. The vapor pressures in the boiler and the condenser are measured by pressure gauges with an accuracy of 0.25%. The coolant flow rate is determined by a weight-time method. At a given water flow rate and vapor pressure, the following steady-state data are recorded: cooling water flow rate, inlet water temperature and temperature rise, vapor pressure and temperatures, and the condensate volume flow rate. All the data of thermocouples are collected by a data-acquisition system and reduced by a computer.

During the experiments, noncondensable gases in the test loop are reduced to a negligible value by repeatedly opening a flushing vent at the top of after-condenser. As a check of whether the noncondensable gases have been reduced to a negligible value, the vapor temperature in the condenser shell measured by the thermocouples is compared with the saturated temperature corresponding to the measured pressure. The allowed difference is limited to 0.3 K [3]. Under such condition, the estimated volume concentration of noncondensable gases is less than 1%.

The heat loss of the experimental system varies with the temperature difference between the refrigerant and the environment, and is measured via the following method. First the cooling water system for the condenser is closed and some power input to the evaporator is supplied. Then some evaporated vapor is condensed in both the condenser and the evaporator because of the heat loss to the environment. When the system is kept in steady state the power input ϕ indicates the total heat loss to the environment at the given temperature difference ($T_s - T_{sur}$). From these data the heat loss coefficient can be determined as follows:

$$C_{loss} = \phi/(T_s - T_{sur}) \tag{1}$$

The same procedure is repeated for different temperature differences. In our experimental process, the value of ($T_s - T_{sur}$) varies from 16 K to 20 K, and the heat loss coefficient is found to be 8 W/K.

The input power minus heat loss is considered as the heating power for evaporation, and the cooling water flow rate multiplied by its temperature rise is the cooling power in condensation. If the imbalance in heating power and the cooling power is larger than 3%, the reasons responsible for the large imbalance are analyzed and the test run should be repeated. Only those data with the relative imbalance less than 3% are recorded as the test data. And the average value of the heating power and cooling power is taken as the heat transfer rate.

Table 1
Geometries of studied tubes

Tube	Material	D_o (mm)	D_r (mm)	Fin height (mm)	Fin pitch (mm)	Fin density (fpm)	tt (mm)	tb (mm)	θ
No. 1	Cu/Ni	19.18	17.88	0.65	0.635	1575	–	0.25	–
No. 2	Cu	18.93	17.08	0.926	0.581	1720	0.21	0.21	–
No. 3	Cu	18.93	16.93	0.998	0.678	1475	0.266	0.352	2.5°
Smooth	Cu/Ni	19.03							

Attention is now turned to the test tubes. The tube specifications are listed in Table 1. Tube No. 1 is of Thermoexcel-C surface and tube No. 2 is an integral-fin tube with rectangular cross section and tube No. 3 is also integral-fin tube with trapezoidal-fin shape. The smooth tube and tube No. 1 are made of cupronickel, which contains copper and nickel with the percentages of 70 and 30, respectively.

3. Data reduction and uncertainty analysis

A subtraction of thermal resistances technique is used to obtain the vapor side heat transfer coefficients, h_o. A brief description of its implementation is given as follows. The total thermal resistance $1/k$ is related to the component thermal resistances by the following equation:

$$\frac{1}{k}=\frac{1}{h_o}+\frac{A_o}{A_i}\frac{1}{h_i}+R_f+R_w \tag{2}$$

It is a well-accepted practice that for the enhanced tubes of phase change heat transfer on outside surface with different kinds of complicated structures the outside area of the bare tube from which the enhanced tube was made is used as the heat transfer area A_o [11]. This practice is adopted in this study. Since all the tubes should be cleaned before experiment, the fouling resistance R_f is considered negligible. To obtain the average condensation heat transfer h_o from Eq. (2) with good accuracy, the water side heat transfer coefficient h_i must be determined as accurately as possible. In the present study, the Gnielinski equation is adopted [12]. It should be noted here that in Eq. (2) the term of R_w only accounts for the thermal resistance through the tube wall thickness. The thermal resistance of conduction within the saw-teeth is included in the condensation heat transfer coefficient h_o. When the material with high thermal conductivity such as copper is used, its effect on the condensation may be neglected, which means that each whole saw-tooth has the same temperature as the one of the tube root outside surface. However, when the material with much lower thermal conductivity is used, the effect of the saw-tooth conduction thermal resistance will be more significant, as can be seen from the following presentation.

According to the Nusselt theory for vapor condensation outside a single horizontal smooth tube [13], the condensation heat transfer coefficient can be determined by the following equation:

$$h_o=0.728\left(\frac{gr\rho^2\lambda^3}{\eta D_o\Delta T}\right)^{1/4} \tag{3}$$

In this study the predicted heat transfer coefficients from Nusselt theory are determined from Eq. (3), for which the mean temperature of the film, $(T_s-0.5\Delta T)$, is used as the reference temperature except the latent heat which is determined according to the condensation pressure.

An uncertainty analysis according to literature [11,14] has been carried out to reveal the possible uncertainty in the experimental data and the reduced results. All the thermocouples are calibrated prior to installation to an accuracy of 0.2 K, and that of thermopiles is 0.05 K. For tubes No. 1, No. 2, No. 3 and smooth tube, the estimated uncertainties of q are 2.1%, 2.0%, 2.1%, and 3.0%, and that of k are 3.0%, 2.9%, 3.0% and 3.7%, respectively. Since h_o is not directly measured, it is impossible to assign uncertainty estimation to it. However, the uncertainties involved in determination of h_o are estimated as follows. The value of the uncertainty in the calculation of h_i is considered 20% [12,13]. According to [12], the Gnielinski correlation deviates from 90% of its original test data within 20%. Thus this percentage is adopted as the uncertainty of the equation. In all experiments, the minimum percentage of water side thermal resistance (including the tube wall thermal resistance) is 31–41%, and the maximum percentage is about 40–49%, and that of the smooth tube are 24 and 30%. Based on such estimations, the calculated uncertainties of h_o are 13.3–18.3%, 18.8–24.9%, 18.3–24.2% and 11.2–13.9% for the tubes No. 1, No. 2, No. 3 and smooth tube, respectively.

4. Results and discussion

4.1. Condensation of HFC134a and CFC12 outside smooth tube

The reliability of the measurement system can be confirmed from the measured condensation heat transfer data of HFC134a and CFC12 outside a single horizontal smooth tube presented below. One set of measurements at a fixed vapor pressure was performed. The resulting h_o versus ΔT at the saturated temperature of 40°C is shown in Fig. 2(a), where the dash line and the dot line represent the Nusselt theory, Eq. (3) for HFC134a and CFC12 respectively. The solid lines are the curve-fittings of the experimental heat transfer coefficients. It can be seen that the Nusselt equation agrees with the experimental heat transfer coefficient very well, and the

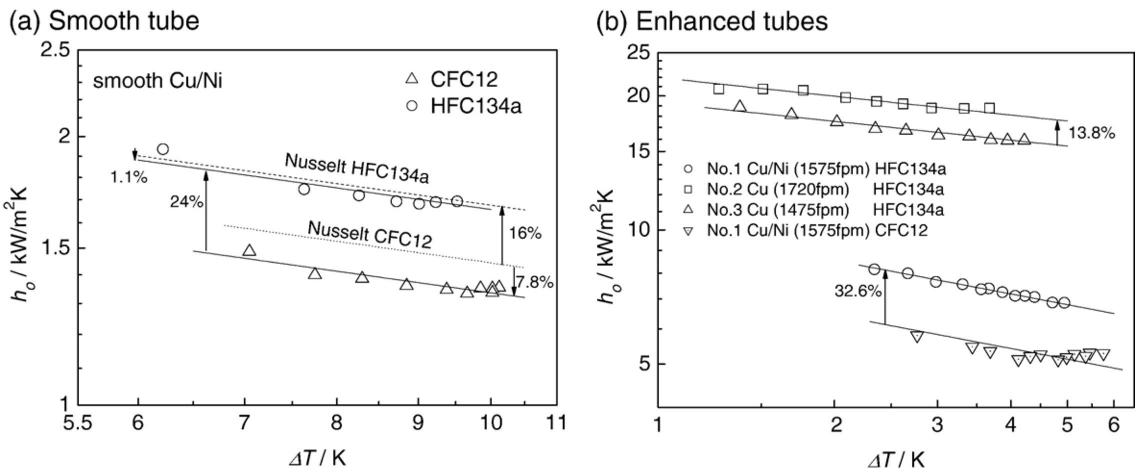


Fig. 2. Variation of h_o with ΔT for studied tubes.

difference between the predicted results and the experimental data is in a range of ± 10%. The experimental data of HFC134a is about 1.1% less than that of the theoretical results, and the value of CFC12 is about 7.8%. It is a general understanding that for the condensation of pure vapor outside single horizontal smooth tube, the Nusselt equation can be applied for most working media, including water and many refrigerants [11]. The above results indicate the reliability of the experimental data. The thermophysical properties are determined by using Refprop 6.0 by NIST.

It can be seen that the experimental heat transfer coefficient of HFC134a is approximately 24% more than that of CFC12. The Nusselt model predicts the condensation heat transfer coefficient for HFC134a to be about 16% more than that of CFC12 at the same temperature difference, due to the difference in thermophysical properties of the working fluids.

4.2. Condensation of HFC134a and CFC12 outside enhanced tubes

The presentation of the test results for the enhanced tubes is proceeded in three aspects. First the measurement results of the condensation heat transfer coefficients at the saturated temperature of 40°C are provided, followed by the enhancement factor of each tube. In addition, the experimental data of integral-fin tubes are compared with five theoretical models, which can give some further support to the reliability of the measurement system.

Fig. 2(b) shows the variation of the condensation heat transfer coefficients with the temperature differences. It can be seen that the condensation heat transfer coefficients of HFC134a are about 32.6% larger than that of CFC12 for enhanced tube No. 1, the difference is larger than that of the smooth tube (24%, Fig. 2(a)). The heat transfer coefficients of tube No. 2 are about 13.8% larger than that of tube No. 3 for condensation of HFC134a, the coefficients of the integral-fin tubes made of copper are larger than that of the tube made of cupronickel. As indicated above, this difference is mainly caused by the relatively large thermal resistance of saw-tooth, for the tube made of cupronickel.

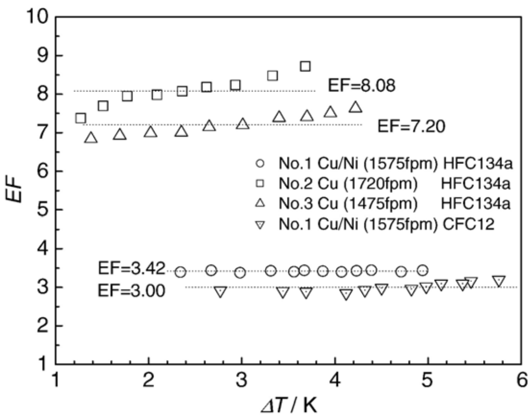


Fig. 3. Variation of EF with ΔT.

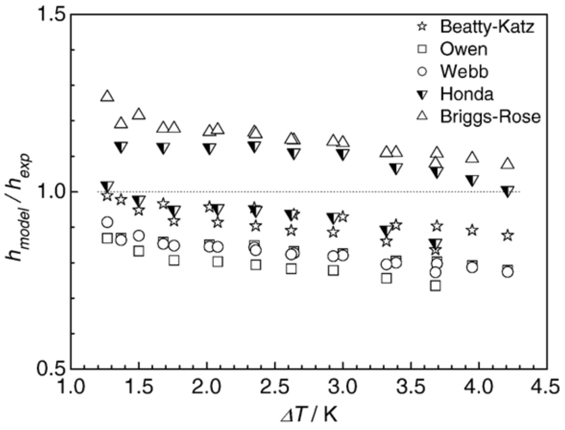


Fig. 4. Comparison with theoretical models.

Attention is now turned to the enhancement factor of the finned tubes. The enhancement factor, EF, is defined as the ratio of heat transfer coefficient of enhanced tube over that of the related smooth tube at the same temperature difference. Fig. 3 shows the EF of the three studied tubes. The average EF of the cupreous tubes No. 2 and No. 3 are 8.08 and 7.20, respectively. The cupronickel tube No. 1 has the EF value of 3.42 for HFC134a, and 3.00 for CFC12. The cupreous integral-fin tubes (No. 2 and 3) run more than twice condensation heat transfer coefficients than that of the cupronickel Thermoexcel-C tube (No. 1).

Generally speaking, the Thermoexcel-C tube possesses better performances than that of the integral-fin tubes for condensation heat transfer made of the identical material. The fin densities of the three tubes are not different very much. The difference is mainly due to the different tube thermal conductivities: The thermal conductivity of cupronickel is 28.9 W m^{-1} K^{-1}, and the value is 398 W m^{-1} K^{-1} for copper. From the experimental results we can confirm that the thermal conductivity is an important factor in enhanced condensation heat transfer. Because of the low thermal conductivity, the temperatures at the fin flank and fin tip are much higher than that of fin root, leading to lower fin efficiency, smaller local condensation temperature difference and lower local heat transfer rate.

Fig. 4 compares the experimental condensation heat transfer coefficient with that predicted by theoretical models, including Beatty–Katz model [2], Owen model [15], Webb model [3], Honda model [16] and Briggs–Rose model [10], for the integral-fin tubes No. 2–No. 3. It can be found that the five prediction models can predict the experimental results within a range of ±20% for the condensation of HFC134a, among whom the deviation of Honda model is in the range of about ±15%, showing a good agreement with the present experimental data. This generally good agreement between the predictions and our test data gives further support to the reliability of our measurement system.

5. Conclusion

Condensation heat transfer experiments outside a horizontal smooth tube and three finned tubes were conducted for HFC134a and CFC12 at saturated temperature of 40 °C with tube materials of different thermal conductivity. The major findings are as follows.

(1) The predicted condensation heat transfer coefficients for the smooth horizontal tube from Nusselt theory for HFC134a and CFC12 agree within ±10% with the experimental data.
(2) The condensation heat transfer coefficients of HFC134a are about 32.6% larger than that of CFC12 for the cupronickel Thermoexcel-C tube, indicating that for enhanced tubes HFC134a also has better heat transfer performance than that of CFC12.
(3) The cupreous integral-fin tubes provide more than twice condensation heat transfer coefficients than that of the cupronickel Thermoexcel-C tube, showing that the thermal conductivity has great influence on enhancing condensation heat transfer.

Nomenclature

A	surface area, m^2
C	heat loss parameter, W K^{-1}
D	diameter, m

EF	enhancement factor
g	gravitational acceleration, m s^{-2}
h	heat transfer coefficient, W m^{-2} K^{-1}
k	overall heat transfer coefficient, W m^{-2} K^{-1}
q	heat flux, W m^{-2}
r	latent heat, kJ kg^{-1}
R	thermal resistance, W^{-1} m^2 K
T	temperature, K
tt	fin thickness at fin tip, mm
tb	fin thickness at fin root, mm

Greek symbols

λ	thermal conductivity, W m^{-1} K^{-1}
Δ	vapor-to-wall temperature difference
η	liquid dynamic viscosity, Pa s
ϕ	heat transfer rate, W
θ	half fin tip angle
ρ	density, kg m^{-3}

Subscripts

exp	experimental result
f	fouling
i	inner surface
loss	heat loss
model	theoretical model
o	outer surface
r	fin root
s	saturated temperature
sur	surrounding temperature
w	tube wall

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