Study on Condensation Heat Transfer Characteristics of Hydrocarbons Natural Refrigerants

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Key words: Condensaton heat transfer, Natural refrigerant, R-290 (Propane), R-600a (Iso-Butane)

Abstract

This study investigated the condensation heat transfer coefficients of R-22, R-290 and R-600a inside horizontal tube. Heat transfer measurements were performed for smooth tube with inside diameter of 10.07 mm and outside diameter of 12.07 mm and inner grooved tube having 75 fins whose height is 0.25 mm. Condensation temperatures and mass velocity were ranged from 308 K to 323 K and 51 kg/m^{2}s to 250 kg/m^{2}s, respectively. The test results showed that the local condensation heat transfer coefficients increased as the mass flux increased, and also the effects of mass velocity on heat transfer coefficients of R-290 and R-600a were less than those of R-22. Average condensation heat transfer coefficients of natural refrigerants were superior to that of R-22. The present results had a good agreement with Cavallini-Zecchin's correlation for smooth and inner grooved tubes.

Nomenclature

A : area [m^2]
c_p : specific heat [kJ/kg · K]
d : diameter [m]
G : mass velocity [kg/m^2 · s]
h : heat transfer coefficient [kW/m^2 · K]
i : enthalpy [kJ/kg]
i_{fg} : latent heat [kJ/kg]
m : mass flow rate [kg/s]
Q : heat capacity [kW]
T : temperature [K]
x : vapor quality [-]
z : length of test section [m]

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Greek Symbols

Δ : difference [-]

Subscripts

avg : average
cal : calculated
exp : experimental
i : inner
in : inlet
L : local
out : outlet
r : refrigerant
s : coolant
sub : subsection
wi : inside tube wall

1. Introduction

The use of HCFCs in new refrigeration equipments will be banned world wide in the near future. Because developments in technology have gone much faster than expected after the legislation on CFCs and green house gases including HCFCs was passed.[1] And it has already been suggested that HFCs may be decomposed by sunlight in the troposphere and form acid and poisonous substances.[2] Montreal (1987) was followed by London (1990) and Copenhagen (1992), and in Copenhagen not only the termination of the production of CFCs by January 1, 1995, in the industrial countries has been decided, but also a phase-out schedule for the HCFCs.[3]

HCs (Hydrocarbons) are well known flammable working fluids with favourable thermodynamics properties and material compatibility. They include propane, butane and their mixtures. The only important disadvantage of hydrocarbon refrigerants is that they are combustible, with a very low ignition concentration limit, and this drawback has been blown up to unreasonable proportions. In fact, they are popular fuels available everywhere and used with simple precautions even in private home. With reasonably careful design it must be even more simple to ensure safety in a hermetic closed refrigeration circuit. Direct cooling is possible in small systems, when the charge is low enough to avoid any explosion risk in rooms where leakage may occur.[4]

At present, hydrocarbons are not accepted as substitutes for refrigerants in the United States because of their flammability. However, thermodynamic properties of hydrocarbons, such as propane, are similar to those of R-12 and R-22. Another advantage of hydrocarbons is their solubility in mineral oil, which is traditionally used as a lubricant in the compressors.

Jung and Radermacher[5] simulated the performance of nonflammable pure and mixed refrigerants as substitutes for R-12 in single-evaporator domestic refrigerator/freezer units. They found no chlorine-free substitute that also yielded saving. James et al.[6] consider in detail the various hazards involved in the use of propane in small refrigeration systems, and explained how these can be overcome by appropriate design. They also point out that in the event of a fire of sufficient severity to raise the pressure of the refrigerant to a level that ruptures the circuit, propane (and similarly the other hydrocarbons) is actually safer than R-12, as the small quantity of hydrocarbon would make an insignificant contribution to the fire, while the R-12 could produce products such as carbonyl chloride (phosgene gas) of $COCl_2$.[7]

Despite their potential flammability, Richardson and Butterworth[8] have demonstrated that hydrocarbons can be safely used as refrigerants in hermetic vapour compression systems and achieve better COPs than R-12 under similar operating conditions. Mixtures of around 50 wt% propane and 50 wt% isobutane have very similar saturation characteristics to R-12 but COP would seem to improve as the proportion of propane is increased. In addition, propane was tested in a 2.5 ton air-conditioning unit by Treadwell.[9] After running the propane system with a larger compressor, 2% savings were achieved compared to the original HCFC22 system.

This study aims to concentrate in the hydrocarbon refrigerants R-600a, R-290, in order to develop the technology and expand the knowledge based on natural working fluids in compression heat pumping systems. Especially, focusing on the characteristics of condensation, which is the basis for the optimum condenser design for heat pump cycles using R-22 as a natural refrigerant. For the purpose of the study, we made a basic heat pump apparatus with a horizontal tube-in-tube type condensers used smoothed tube as well as grooved inner tube. It will confirm applicability of natural refrigerants to regulate refrigerant R-22, support presentation of alternative refrigerant data for optimum design in the refrigeration and air-conditioning systems.

2. Experimental apparatus and methods

2.1 Experimental apparatus

A schematic flow diagram of the experimental apparatus is presented in Figure 1. The compressor for refrigerant used a semi-open-drive, two-cylinder reciprocating type driven by a variable-speed, 1492 W (2 HP) electric motor. A compressor originally designed for HCFC22 was used in the experiment and was connected by an inverter, which could alter the revolutions per minute of the compressor. The suction line to the compressor contains an electrically heated refrigerant superheater to maintain a constant return gas temperature to the compressor if required. As shown in Figure 2, the refrigerant flows inside the inner tube and the secondary fluid (water) flows through the annular side in the opposite direction. The one, which was made of 6300 mm length of smoothed cooper tube, was constructed as double-tube types. There is the inner tube has thickness 1 mm and outer diameter 12.07 mm and the outer tube has thickness 2.5 mm and outer diameter 50 mm, respectively. Another one, which was made as inner grooved tube section has height 0.25 mm, tube thickness 0.4 mm, fin pich 75, fin angle 40°. Also, it has an outer diameter of 12.07 mm and 50 mm, respectively. Figure 2 shows detail of test section with inner grooved tube. Each heat exchanger is divided into two equal horizontal straight sections of 6300 mm long copper tubing joined with short U-tube interconnecting pieces. And each heat exchanger has eight subsections as a test section and 2 subsections as superheating or subcooling section. All test sections excluded external influence throughly with double adiabatic. Thermal expansion valve capillary tube and manual expansion valve are used as the throttling device for regulating refrigerant flow rate.

The pressure transducers were calibrated to ±0.25% (about 5 kPa) of full scale. Refrigerant mass flow rate was measured directly with the mass flow meter (OVAL mass flow meter, ±0.03 %) mounted in the liquid line leaving the condenser. Also the mass flow rate of source water was measured directly with a water mass flow meter (ORIFICE FLOW MATER, ±0.5%)

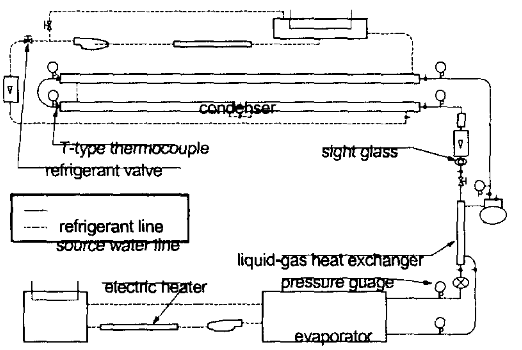

Fig. 1 Schematic diagram of experimental apparatus.

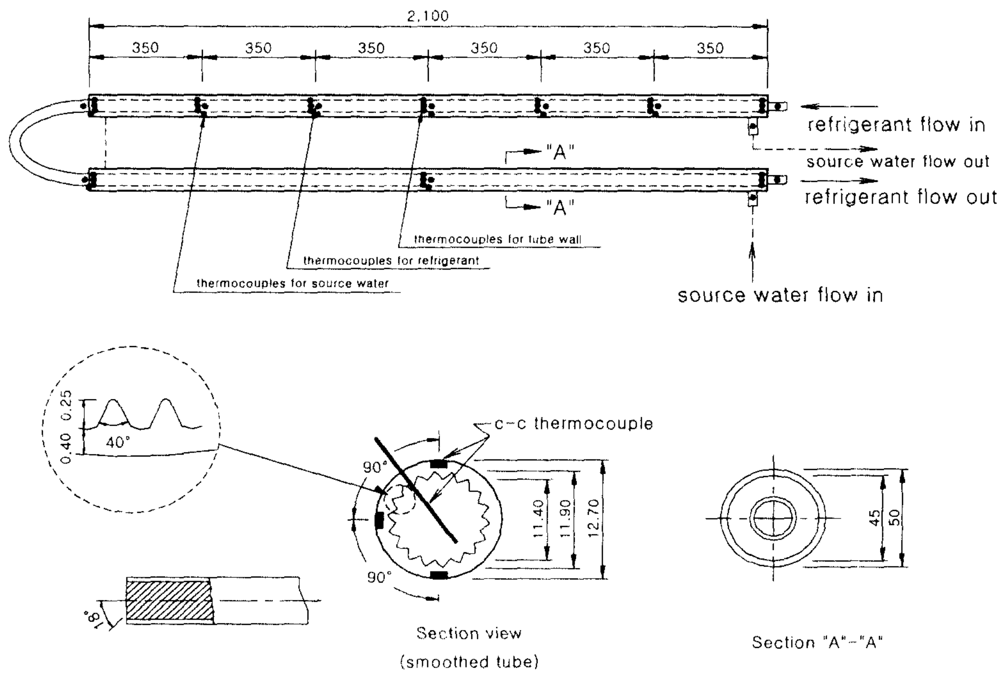

Fig. 2 Detail of test section (inner grooved tube).

installed in the source water inlet of the condenser. T-type thermocouples were used for temperature measurement : the accuracy of the thermocouples was estimated to be with ±0.2 ℃. All thermocouples were used after being revised with a standard thermometer. As shown in Figure 2, the tube wall temperatures of the inner tube were measured at three points (top, bottom, and side). Thus, test section has measured with 12 points of thermocouples that were reading for the bulk temperature of the refrigerants. Channel output signals from instrumentation points are fed to a data-acquisition control unit, and processed by a desktop computer, which communicates directly via an interface bus, RS232C.

The scope of condensing temperature is 303 K to 323 K which is the general temperature range for air-cooled and water-cooled condensers. The refrigerant mass velocity of R-22, R-290 and R-600a is 61 to 247 kg/m^2 · s, 30 to 200 kg/m^2 · s, and 30 to 139 kg/m^2 · s, respectively.

2.2 Data analysis

The thermodynamic data for the various refrigerants were obtained from the latest version of a refrigerant property package (NIST) that uses the Carnahan-Starling-DeSantis-Morrison (CSDM) equation of state.[10]

Raw data from the data-aquisition system were analyzed for each run to determine the heat transfer rate and the quality. The main equations used in processing the raw data were based on energy balances. The energy transferred in the test section was computed from an energy balance on the water side:

Heat capacity in condenser can be evaluated by the following equation (1).

$$Q_{exp} = m_s \cdot c_{ps} \cdot (T_{out} - T_{in}) \qquad (1)$$

Where m_s, T_{in}, T_{out}, and c_{ps} represent mass flow rate of water [kg/h], temperature of inlet and outlet [K], and specific heat of coolant [kJ/kg · K] respectively. And the local heat transfer coefficient in the process of condensation [kW/m^2 · K] can be evaluated by the following equation (2).

$$h_L = \frac{Q_{sub}}{A_{sub} \cdot (T_r - T_{wi})} \qquad (2)$$

Where, Q is heat capacity, and T_{wi}, A_{sub} represent the inside tube wall temperature of condenser and heat transfer area of a subsection ($= \pi \cdot d_i \cdot \Delta z$ [m^2]) respectively. Inside tube wall temperature can be expressed by using one-dimensional steady-state concentric conduction equation.

The vapor quality at the subsection exit of the condenser was calculated from an energy balance of the system. The quality can be calculated as

$$x_{sub,out} = x_{in} - \frac{q \cdot \pi \cdot d_i}{m_r \cdot i_{fg}} \qquad (3)$$

Where d_i, m_r, i_{fg} and q is the diameter of tube [m], mass flow rate of refrigerant [kg/h], the latent heat of condensation [kJ/kg] and the heat flux respectively. The averager heat transfer coefficients at condenser calculated to the means of local heat transfer coefficients h_L.

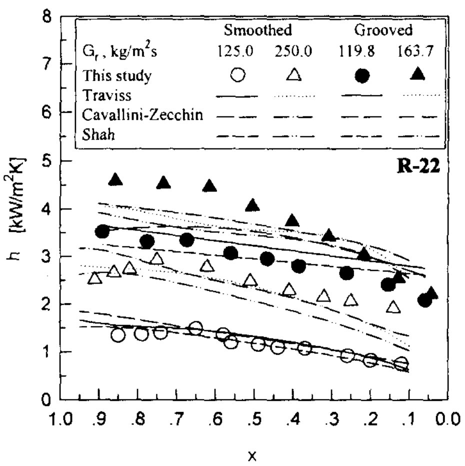

Fig. 3 Comparison of heat transfer coefficients with existing correlations for R-22.

3. Condensation heat transfer properties

3.1 Comparison of local condensation heat transfer correlations

Several correlations available in research literature have been verified for use with various refrigerants. Some of these correlations are described in this study. They are the correlation by Traviss et al.,[11] Cavallini-Zecchin,[12] and Shah.[13] As shown in Figures 3~5, the

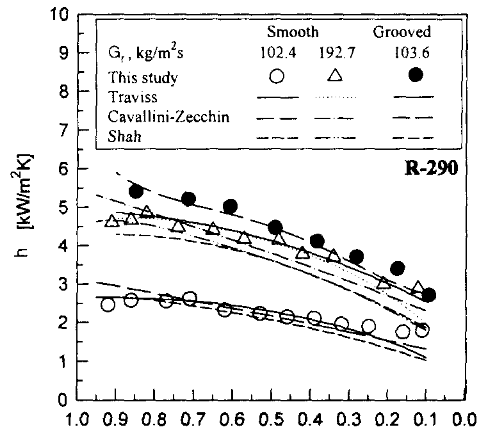

Fig. 4 Comparison of heat transfer coefficients with existing correlations for R-290.

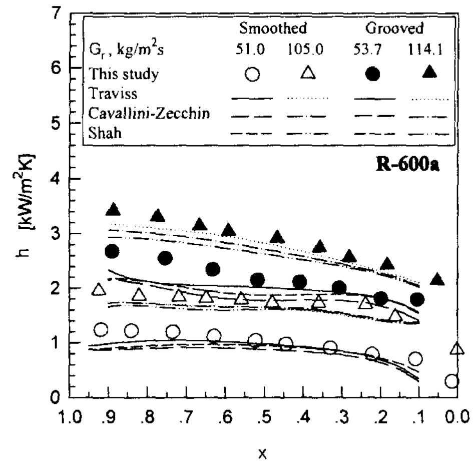

Fig. 5 Comparison of heat transfer coefficients with existing correlations for R-600a.

deviation between the experimental data and predicted data increased as the quality increases. In general, in the annular flow regime, the local condensation heat transfer coefficients gradually decreased as the quality decreased, which is the result of the annular film thickness increasing as condensation proceeded from the high-quality inlet of the condenser to the low-quality exit during the condensation process. This trend can be explained by the increase in the thermal resistance with increasing liquid film thickness and the reduced velocity due to the change from vapour to liquid.

Figures 3~5 damonstrate comparisons of the three correlations described above with the experimental data. These figures represent the variation of the condensation heat transfer coefficients of R-22, R-290 and R-600a according to the changes of quality when the mass velocity was changed.

Some correlations represent this trend well; the Cavallini-Zecchin and Shah correlations are the typical examples. According to these correlations, all condensation heat transfer coefficients showed almost the similar tendency; consistent decrease with decreasing quality, irrespective of refrigerants as well as tube geometry used. In this study, the initial increase in the condensation heat transfer coefficients was observed due to effect to superheating of compressor outlet. And then, it gradually decreased later as a stable annular flow field was achieved. And such a quantitative discrepancy between experimental and calculated data was due to the use of an annular flow model in analyzing the real condensation process. The correlation tendency seemed to be the same, such as shown in the Figures. According to the Figure 3, tendencies were observed that the experimental data for low mass velocity were smaller than those of the above three correlations and the experimental data for high mass velocity were larger than those of the correlations. Figure 4 present the local condensation heat transfer coefficients of R-290 according to the changes of quality when the mass velocity was 102 kg/m^{2}s and 192 kg/m^{2}s, respectively. As shown in Figures, tendencies were almost the same with R-22. The condensation heat transfer coefficients of R-600a according to the changes of quality are presented in Figure 5. Tendencies were observed that the distribution of condensation heat transfer coefficients seemed to be the same as that of R-22 and R-290.

As shown in the figures, the Cavallini-Zecchin correlation showed about ±30% mean deviations except in the high and low quality regions, where the unstable annular, slug or plug flows were expected, compared with the results of this experiment. So the present results had a good agreement with Cavallini-Zecchin's correlation for smooth and inner grooved tubes.

3.2 Average condensation heat transfer coefficients

Figure 6 described average condensation heat transfer coefficient according to mass velocity

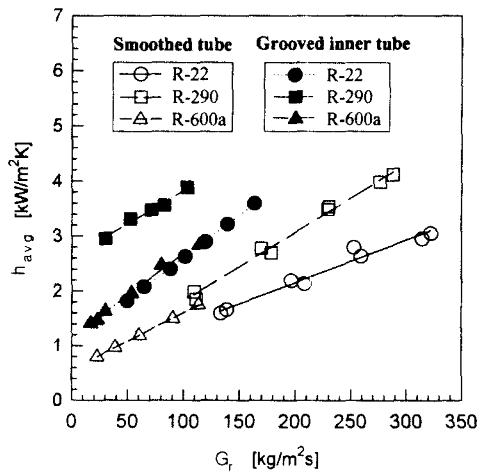

Fig. 6 Comparison of average heat transfer coefficients among the refrigerants mass velocity.

under all experimental conditions. Figure shows a general tendency that average condensation heat transfer coefficient lineally increases as mass velocity increases. As shown in the Figure the average condensation heat transfer coefficients of R-290 increased about 40% compared with R-22 in the smoothed tube. And the average condensation heat transfer coefficient of R-290 and R-600a increased approximately 50% and 5% in the inner grooved tube compared with smooth tube, respectively.

The condensation heat transfer coefficients of the R-290 and R-600a were higher than that of the R-22 in the smoothed tube as well as in the inner grooved tube. In addition, heat transfer coefficients of R-22 increased to a larger extent than R-290 and R-600a in the inner grooved tube compared with smooth tube. And average condensation heat transfer coefficient of natural refrigerants were superior to that of R-22. Especially, in the inner grooved tube, 2 times of the increment of the heat transfer coefficient was obtained compare with smoothed tube. Accordingly condensation heat transfer coefficients of natural refrigerants showed smaller effect on the mass velocity than that of R-22.

4. Conclusions

In this work, by taking R-22, R-290 and R-600a which are promising alternative refrigerants for R-22, the characteristics of condensation heat transfer are examined in the basic heat pump cycle.

From the data, we discussed the variation of heat transfer coefficients between the natural fluids and R-22. The stable annular flow regime was dominant in the inner grooved tube. And the all experimental data agreed with the available three correlations to within $\pm 30\%$. However, the Cavallini-Zecchin correlation matched with the experimental results quite well. Concerning the effect of mass velocity, in the range of low mass velocity under 120 kg/m^{2}s, Shah's correlation agreed well with the data, and in the rage of high mass velocity over 250 kg/m^{2}s, Cavallini-Zecchin correlation agreed well with the data.

The effect of condensation heat transfer coefficient of natural refrigerants was less than that of R-22 in the mass velocity. And the average condensation heat transfer coefficient of natural refrigerants is higher than that of R-22. The heat transfer coefficients in the smoothed tube using R-290 and R-600a increased compared with R-22 by 40% and 20%, respectively. In the inner grooved tube, the average heat transfer coefficient increased about 2 times compared with the smooth tube.

Therefore heat transfer properties of natural refrigerants, R-290 and R-600a, were presumed superior to those of R-22.

Acknowledgments

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