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TUBESIDE CONDENSATION OF NONAZEOTROPIC REFRIGERANT MIXTURES FOR TWO ENHANCED SURFACES

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Abstract

As part of the Building Equipment Research program at Oak Ridge National Laboratory (ORNL), nonazeotropic refrigerant mixtures (NARMs) are being investigated to replace chlorofluorocarbon compounds. The condensation of NARMs is not isothermal, and this can improve the heat exchanger effectiveness of a condenser as well as improve thermodynamic cycle efficiencies. The total condensing heat transfer coefficients for refrigerant R22 and for four nonazeotropic mixtures of refrigerants R143a and R124 were measured and are presented as a function of mass flux for two inside tube surfaces, one having spiral ridged fins and the other having a spirally corrugated or fluted surface. The total condensing coefficient for the finned tube is higher than that for the fluted tube at any given refrigerant mass flux for all the refrigerant mixtures. The measured irrecoverable pressure drop for the finned tube was approximately half that for the fluted tube; thus, the finned tube has the better thermal performance of the two enhanced tubes.

The condensing heat transfer coefficient is also presented as a function of the mass fraction of R143a for three values of mass flux. Degradation of the condensing coefficient for intermediate values of R143a mass fraction is apparent, but has different trends with respect to mass flux for the two enhanced surfaces; thus, the geometry of the enhanced surface appears to affect the physical mechanism for condensation of NARMs.

Nomenclature

- A area (m^2)
- a dimensionless coefficient used in equation (4)
- c specific heat at constant pressure ($J \cdot kg^{-1} \cdot K^{-1}$)
- d diameter (m)
- e fluted tube indentation (m)
- f fin height (m)
- h heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
- k thermal conductivity ($W \cdot m^{-1} \cdot K^{-1}$)
- m mass flow rate ($kg \cdot s^{-1}$)
- p fluted tube pitch (m)
- q'' heat flux ($W \cdot m^{-2}$)
- Re Reynolds number ($md_h/A\mu$)
- T temperature (K)
- U overall heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
- μ dynamic viscosity (Pa·s)

Subscripts

- h hydraulic
- i inlet, inner
- $lmtd$ log mean temperature difference
- o outlet, outer
- r refrigerant
- w water

Introduction

Replacements for chlorofluorocarbon (CFC) refrigerants are being investigated as part of the Building Equipment Research program at Oak Ridge National Laboratory (ORNL). One class of replacements is called nonazeotropic refrigerant mixtures (NARMs). For NARMs in the wet (two-phase) region, the saturation temperature at constant pressure is a function of the vapor quality, that is, the condensation process is not isothermal. This nonisothermal phase-change behavior can improve the temperature effectiveness of a heat exchanger (Granryd and Conklin, 1990) as well as thermodynamic cycle efficiencies (McLinden and Radermacher, 1987). Other investigators (DeGrush and Stoecker, 1987; Stoecker and Kornota, 1985; Bokhanovskiy, 1980), however, have measured heat transfer coefficients for binary NARMs condensing in smooth tubes that were less than the heat transfer coefficients for either of the pure constituent refrigerants. Also, the same degradation of condensing coefficient for mixtures of R22 and R114 was reported by Koyama et al. (1990) inside a horizontal tube with internal spiral grooves. Additional heat transfer area for NARM condensers would thus be required; hence, the effect that additional enhanced heat transfer surfaces have on the NARM condensing heat transfer coefficient was investigated.

The non-CFC refrigerants R143a and R124 were chosen for testing based on a previous research effort (Vineyard et al., 1989) that screened refrigerant pairs using such factors as boiling point, stability, ozone depletion potential, and coefficient of performance to determine suitable candidates for residential heat pump operation. No azeotropic mixtures of R143a and R124 are believed to occur. Four different mixtures of R143a/R124 were condensed at various mass flow rates in two heat exchangers: one with extruded ridged fins and the other with a fluted inside tube surface. Refrigerant R22 was also condensed in the two heat exchangers. Because water-to-refrigerant heat exchangers were used in the experimental apparatus, the inlet water temperatures were selected to yield refrigerant saturation pressures equivalent to those determined for air-to-refrigerant heat exchangers at the U.S. Department of Energy's standard rating conditions for heat pumps.

Apparatus

The heat transfer from enhanced tubes during tubeside condensation of binary NARMs is being investigated at ORNL in a highly instrumented apparatus, shown in Fig. 1, that consists of the following components: a variable-speed compressor having a range of 500 to 3000 rpm that allows for variable heat exchanger loadings, a variable-orifice refrigerant metering device (needle valve), and two sets of counterflow concentric-tube heat exchangers having two different enhanced tubeside surfaces. The refrigerant circulates inside the central tube and water circulates in the annulus. The variable-speed compressor provides different heat loadings of the heat exchangers. The needle valve allows flexibility in controlling refrigerant conditions; this flexibility is lacking for capillary tubes and thermal expansion valves.

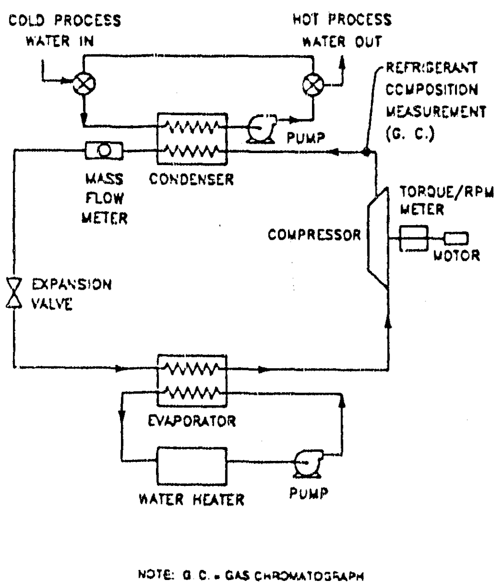

Fig. 1 NARM test loop

The heat exchangers, with instrumentation as shown in Fig. 2, consist of a central tube and an outer tube forming an annulus. There are eight horizontal passes where the outer surfaces of the water annulus and the refrigerant bends are insulated. Each horizontal section is 1.94 m long. Two separate heat exchangers were used for both condensing and evaporation, each having an enhanced surface on both the refrigerant and water sides for investigation of NARM heat transfer. Complete condensation results are presented here; results for evaporation were reported by Conklin and Vineyard (1990).

A sketch of both condenser tube geometries is presented in Fig. 3. The fluted, or corrugated, enhanced copper tube has an outer diameter of 18.9 mm and a wall thickness of 1.065 mm. The spiral indentations are approximately 0.8 mm deep, there are two indentations at a given cross-section, and the pitch (i.e., axial distance between indentations) is 7.1 mm. This pitch results in a helix angle of approximately 80° with respect to the longitudinal axis of the tube. The surface roughness is 2 μm (80 μin.).

The finned tube, also copper, has ten extruded spiral ridges on the

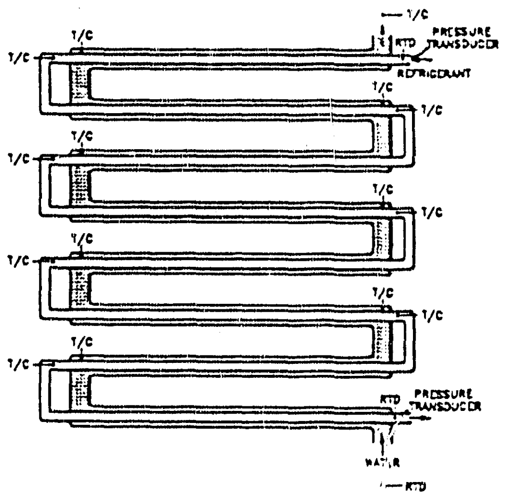

Fig. 2 Heat exchanger schematic

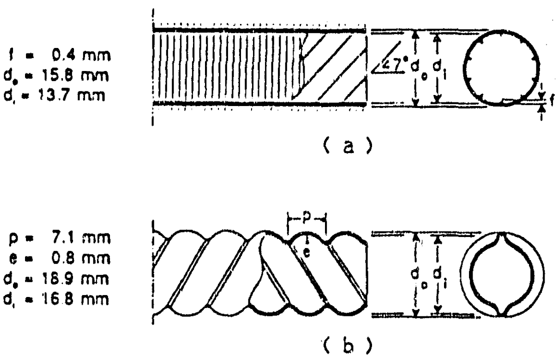

Fig. 3 Tube geometry: (a) finned, (b) fluted

Test Procedure

The output of instrumentation such as mass flowmeters, pressure transducers, thermocouples (T/Cs), and resistance temperature detectors (RTDs), as shown in Figs. 1 and 2, was recorded on a PDP-11 minicomputer. In addition, a dedicated gas chromatograph, located at the compressor exit, was used to monitor the composition of the circulating charge in real time. Data were taken every fifteen seconds and averaged over a fifteen-minute period during steady-state operation. The gas chromatograph and all pressure transducers, flowmeters, thermocouples, and RTDs were calibrated before testing.

Water flow rates to the heat exchangers were pneumatically controlled to allow for rapid changes in test conditions. The inlet temperature to the condenser was maintained with a mixing valve that introduces cool process water while bleeding off warmer water that has passed through the condenser. The inlet water temperature to the evaporator was controlled by a bank of resistance heaters in the closed-loop system.

Test conditions were based on the U. S. Department of Energy's condensation was approximately 1500 kPa, and the saturation temperature was approximately 45°C. At this pressure, the temperature glide (i.e., amount of temperature change in the two-phase region) was approximately 6°C and 10°C for 73% R143a / 27% R124 and 30% R143a / 70% R124, respectively. The heat rates for the refrigerant side and the water side were determined from measured inlet and outlet temperatures and pressures and measured flow rates and were then checked for a steady-state heat balance. The electrical power measurements for the closed-loop system were also used to check for a heat balance in steady-state operation. The needed refrigerant thermophysical properties were determined from algorithms presented by Morrison and McLinden (1986) that implement the Carnahan-Starling-DeSantis equation of state. During the testing, minimal subcooling at the condenser exit and minimal superheating at the evaporator exit were maintained to insure two-phase conditions in the heat exchangers.

Baseline tests were performed with R22 at different compressor speeds and subsequently varying heat exchanger loads to compare the two enhanced tube surfaces, as well as to compare similar published investigations. Four compositions by mass of R143a and R124 were tested: 30% R143a / 70% R124, 51% R143a / 49% R124, 73% R143a / 27% R124, and 77% R143a / 23% R124. A fifth composition of 63% R143a / 37% R124 was also tested, but the resulting condensing coefficients were anomalously high for both tubes and are not presented. Both the R22 and the R143a/R124 mixtures had an alkylbenzene oil present to lubricate the compressor. The concentration of oil was about 1%. The oil's effect on the heat transfer coefficient or pressure drop was not investigated.

Heat Transfer Analysis

One of the objectives of the investigation at ORNL is to measure the thermal performance of the NARMs in condensers. The refrigerant enters the condenser with some superheat (approximately 110% quality) and leaves with slight subcooling (approximately −5% quality). The superheat is removed in the first pass of condenser tubing, and the remaining seven passes constitute the condensing section. A total, or average, condensing heat transfer coefficient is computed from the measured temperatures and flows.

A "standard" log mean temperature difference is computed from the measured inlet and outlet temperatures of the condensing section with the following equation:

$$T_{lmtd} = \frac{(T_{r,o}-T_{w,i})-(T_{r,i}-T_{w,o})}{\ln\left(\dfrac{T_{r,o}-T_{w,i}}{T_{r,i}-T_{w,o}}\right)} \quad . \tag{1}$$

Here, i represents the corresponding flow inlet, and o represents the corresponding flow outlet. Because the phase-change process of NARMs at constant pressure is nonisothermal, a finite specific heat can be defined in the wet region (Granryd and Conklin, 1990). For the refrigerant mixtures of interest, this specific heat varies from the inlet to the outlet, hence, the log-mean temperature difference in the condenser is not strictly applicable. A preliminary correction of the log mean temperature difference to account for the varying specific heat of the refrigerant mixture is being developed at ORNL; therefore, the resultant condensing heat transfer coefficients are preliminary and are presented to qualitatively compare the heat transfer performance of the two enhanced surfaces.

The average heat flux to the refrigerant during the condensation process is then determined from the measured water temperatures and flow rate with the following equation:

$$q'' = \frac{\dot{m}_w c_w (T_{w,o}-T_{w,i})}{A_w} \quad , \tag{2}$$

An overall heat transfer coefficient is then determined as follows:

$$U = \frac{q''}{T_{lmtd}} \quad . \tag{3}$$

The heat transfer coefficient for the water was obtained from the following empirical relationship:

$$h_w = \frac{k_w}{d_h} a Re_w^{0.8} \quad , \tag{4}$$

where Re_w represents the Reynolds number of the flow based on the hydraulic diameter of the annulus. The coefficient a was obtained from a series of tests with R22 using a modified Wilson plot method (McAdams, 1942). For the fluted tube, a had a value of 0.0403. For the finned tube, a had a value of 0.0828.

The heat transfer coefficient of the refrigerant is then computed with the following equation:

$$h_r = \frac{1}{\dfrac{A_r}{A_w}\left(\dfrac{1}{U}-\dfrac{1}{h_w}\right)} \quad , \tag{5}$$

where A_r represents the heat transfer area of the inner tube surface to the refrigerant. The flow area and the heat transfer area are those of the equivalent unenhanced surface—which for the finned tube is the root diameter and for the fluted tube is the undimpled diameter, given as d_r on Fig. 3. These dimensions were chosen so that the thermal performance of enhanced heat transfer geometries can be compared with the equivalent performance of a smooth tube. Testing of these mixtures of R143a and R124 in a smooth tube condenser is underway at ORNL, and the results will be used to quantify the condensing coefficients of these particular NARMs in a smooth tube. The thermal resistance of the tube material was neglected because the tubes are copper, and no fouling factor was assigned because the tubes were new.

In each condensing pass of the tubes, the flow pattern was determined by first computing the Froude and Weber numbers and then using the criteria of Nitheanandan et al. (1990).

Heat Transfer Results

The heat transfer coefficient computed with equation (5) for the series of tests with R22 is presented in Fig. 4 as a function of mass flux. Linear regression analysis was used to fit a straight line to the data. Approximate uncertainties of 9% at high mass fluxes and 15% at low mass fluxes for the heat transfer coefficient were computed with a propagation of error analysis that included the uncertainties of (1) the water-side heat transfer coefficient (estimated), (2) 0.28°C for the RTDs, and (3) 3% of measured flow for the flowmeters.

From inspection of Fig. 4 it can be seen that the heat transfer coefficient is higher for the finned than for the fluted tube surface. The flow pattern for the finned tube was annular for all tube passes at all qualities, and the flow pattern for the fluted tube was annular for most of the tube passes, with wavy flow being computed for the last pass of the fluted tube where the quality was less than 25%. The condensing coefficient for the finned tube as shown in Fig. 4 generally agrees with the R22 condensing coefficients presented by Schlager et al. (1989) for different finned tube geometries.

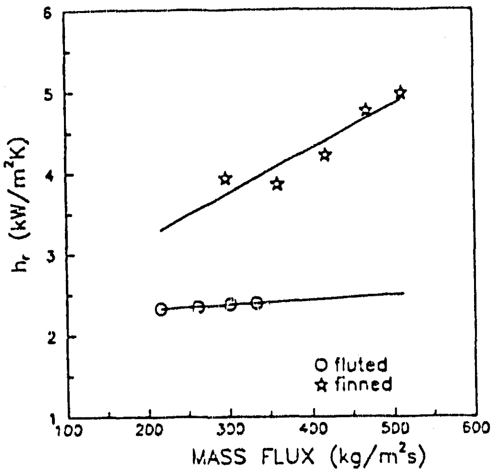

Fig. 4 Average condensing coefficient for R22

The total or average condensing heat transfer coefficients for the four mixtures of R143a and R124 are presented in Figs. 5 to 8. Linear regression analysis was also used to fit a straight line to these data. By inspection it can be seen that the measured condensing heat transfer coefficients for the finned tube are greater than the condensing coefficients for the fluted tube; however, the fitted straight line for the 30% R143a / 70% R124 mixture as shown in Fig. 5 appears to cross the line for the fluted tube at approximately 200 kg/m^{2}s. Because no experimental data were obtained for the finned tube at these low values of mass flux, extrapolation of this straight line may not predict the condensing coefficient at low mass flux for this particular mixture.

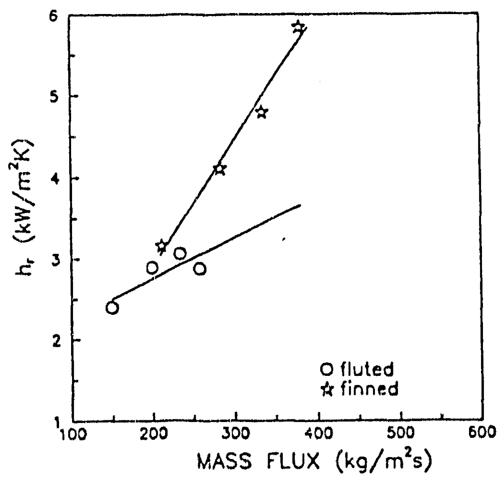

Fig. 5 Average condensing coefficient for 30% R143a / 70% R124

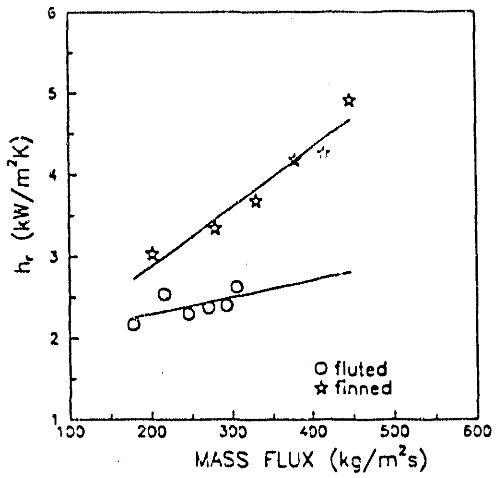

Fig. 6 Average condensing coefficient for 51% R143a / 49% R124

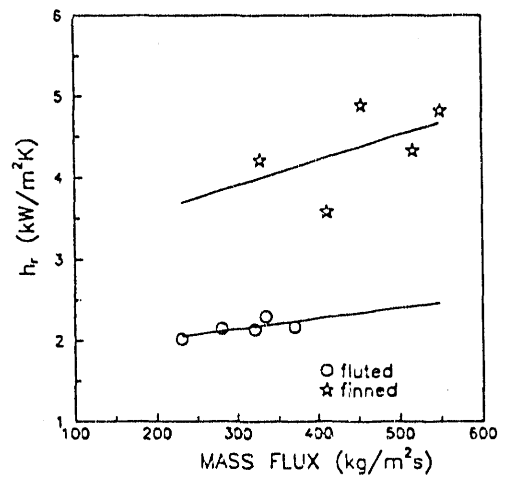

Fig. 7 Average condensing coefficient for 73% R143a / 27% R124

In general, the flow pattern computed for each refrigerant mixture was annular for one tube pass more in the finned condenser (annular for four to five passes, dependent on mass flux) than in the fluted (annular for three to four passes). Also, the Reynolds number for the flow in the finned tube will be higher than the Reynolds number for the flow in the fluted tube for any given mass flow rate because the finned tube has a smaller inside diameter. Reynolds numbers for all flows were greater than 2.5 x 10^4, hence turbulent flow existed for all test conditions.

In the fluted tube, the pressure drop ranged from 2% to 4% of the system pressure (~1500 kPa) for the range of presented mass fluxes. The irreversible pressure drop in the finned tube for all mixtures was approximately half that of the fluted tube. Since the finned tube showed larger heat transfer, the finned tube has the better thermal performance. No relative thermal performance parameter is presented here for these preliminary results. A relative thermal performance

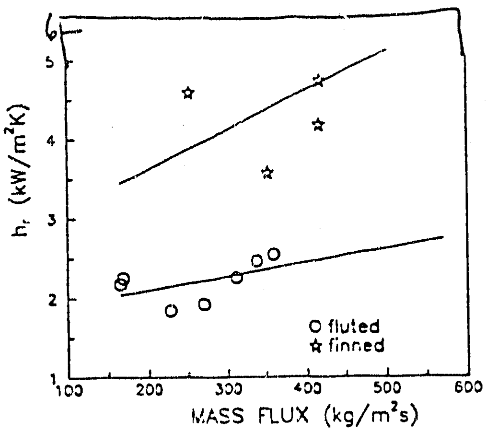

Fig. 8 Average condensing coefficient for 77% R143a / 23% R124

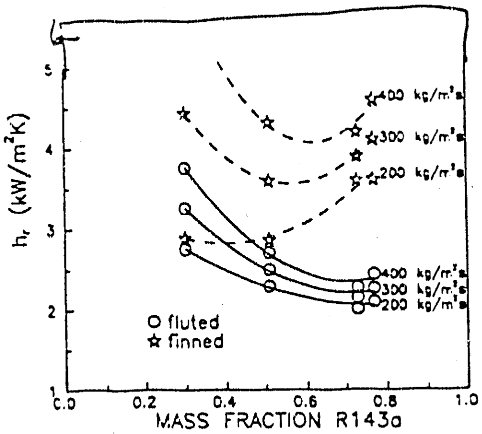

Fig. 9 Average condensing coefficient for R143a/R124 mixtures computed from linear regression of data in Figs. 5-8

The condensing coefficients of R22 and the NARMs in both tubes can be compared at any given mass flux to give the relative heat transfer area needed if R22 is replaced by these particular NARMs. From inspection of Figs. 4 and 7 it can be seen that the fluted-tube condensing coefficient for the 73% R143a / 27% R124 mixture is approximately 10% less than that for R22. This figure of 10%, however, is approximately the uncertainty in the condensing coefficient as given earlier; therefore, only slightly more heat transfer area for a fluted tube heat exchanger may be needed if R22 is replaced with a 73% R143a / 27% R124 refrigerant mixture. Conversely, the fluted tube condensing coefficient for the 30% R143a / 70% R124 mixture, as shown in Fig. 5, is higher than that for R22, as shown in Fig. 4, indicating that less heat transfer area would be needed for replacement of R22. The finned-tube condensing coefficient for the 30% R143a / 70% R124 mixture, as shown in Fig. 5, is approximately 15% greater than that for R22, indicating that slightly less heat transfer area of the finned tube geometry would be needed if R22 is replaced with a 30% R143a / 70% R124 refrigerant mixture.

The total condensing heat transfer coefficient obtained from the linear regression relationships for each refrigerant mixture as shown in Figs. 5 to 8 from both enhanced tubes was computed at the three mass fluxes of 200, 300, and 400 kg/m²s. It is presented in Fig. 9 as a function of the mass fraction of R143a. The computed points for both tubes appear to be best represented by quadratic polynomials determined with a least-squares analysis.

The condensing heat transfer coefficients from the finned and fluted tubes show similar trends as the mass fraction of R143a increases from the lowest (30%) to the highest (77%) mass fraction tested: the condensing coefficient initially decreases, reaches a minimum, then increases as the mass fraction of R143a is increased. Where the minimum condensing coefficient occurs, however, the mass fraction appears to be a function of mass flux for the finned tube, while for the fluted tube it occurs at a constant value of approximately 70% mass fraction of R143a (80% by mole).

No other published work was found for the condensation of the R143a/R124 NARMs in the subject finned and fluted enhanced tube geometries. Koyama et al. (1990), however, report condensing coefficients for mixtures of R22 and R114 inside horizontal tubes that have an enhanced surface consisting of spiral grooves. Condensing coefficients were also reported for flow in smooth tubes for R12/R22 mixtures (Bokhanovskiy, 1980) and R12/R114 mixtures (Stoecker and Kornota, 1985; DeGrush and Stoecker, 1987). Those publications report a degradation in the condensing heat transfer coefficient for the mixtures as compared with the pure constituents, although at differing mole fractions for the different NARMs; thus, a degradation, or minimum, of condensing coefficient as a function of composition was expected. The effect on the condensing coefficient with respect to composition that is apparently caused by the enhanced surface geometry characteristics is shown in Fig. 9.

Conclusions and Recommendations

Although a fair amount of scatter exists for the total, or average, condensing heat transfer coefficient, particularly that for 73% R143a / 27% R124 (Fig. 7) and that for 77% R143a / 23% R124 (Fig. 8), some conclusions can be drawn. The finned enhanced tube has a higher condensing heat transfer coefficient for the NARMs as compared with the fluted tube at any mass flux, with the possible exception of the 30% R143a / 70% R124 mixture at low mass fluxes. Because the irreversible pressure drop for the finned tube was approximately half that for the fluted tube in all the tests, the finned tube has the better thermal performance.

The difference in the total condensing coefficient may be partially explained by the annular flow pattern's being computed for one more tube pass in the finned tube condenser than in the fluted, as well as by a lower Reynolds number flow in the fluted tube as compared with the finned tube at any given mass flux. Other factors may affect the condensing heat transfer, however, such as surface finish, extended surface helix angle, fin height, and surface wetting characteristics. None of these were investigated; hence, the condensing heat transfer coefficients presented here are preliminary and should be considered only within the specific conditions of this work, which simply measured the thermal performance of two specified enhanced surfaces.

Within these limitations, then, the major conclusion of this work is that the geometry used to enhance heat transfer appears to affect the physical mechanism for condensing a nonazeotropic mixture of refrigerants. Additional experiments using different enhanced tubeside geometries and other NARMs are recommended to further investigate the physical mechanism of condensation of NARMs and to identify geometries that optimize the heat transfer and pressure drop characteristics.

Acknowledgment

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