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TWO PHASE FLOW HEAT TRANSFER OF BINARY MIXTURES INSIDE ENHANCED SURFACE TUBING

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ABSTRACT

The primary heat transfer parameters such as coefficient of heat transfer and pressure drop observed during condensation of binary azeotropic refrigerant mixtures R-410a (R125/R32: 50/50), and R-507 (R125/R143a: 50/50) are presented in this paper.
Experiments showed that for Reynolds numbers higher than 4.2 E06, R-410a appears to have greater heat transfer rates more than the other blends under investigation. Furthermore, it is quite evident from this data that R-507 has the highest pressure drop among the refrigerants under investigation.

Introduction

The boiling and condensation heat transfer coefficients of pure refrigerants inside enhanced surface tubing have been reported in the literature by Web et al. [2], Webb and Gupte [10] and Darabi et al. [2]. As of January 1996, new alternatives to CFCs have been developed and some of them are currently available on the market. Some of proposed new alternatives for HCFC 22 and CFC 502 are either azeotropic or zeotropic binary mixtures (ARI [1]).

A review of the literature reveals that a very limited number of studies have been reported on heat transfer characteristics of new zeotropic or azeotropic alternatives to HCFC 22 (Sami et al. [5,7,8]) in smooth surface tubing. However, to the authors' knowledge none is available on enhanced surface tubing other than Sami and Poirier [6]. Therefore, this research work has been undertaken to enhance our understanding of the subject.

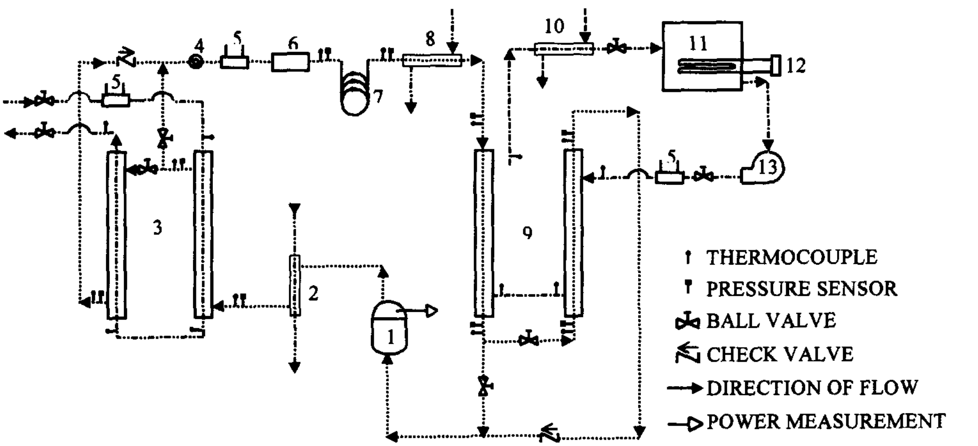
Experimental Apparatus and Measurements

Figure 1 shows a schematic diagram of the experimental setup, which is a water/water vapor compression heat pump composed mainly of an 3 kW compressor, oil separator, condenser, pre-condenser, pre-evaporator, adjustable expansion device, and a test condenser. The oil content in the refrigerant loop was estimated to be about ±1%. Schematic views of the condenser's test section are shown in Figure 2. The horizontal 0.90-m condenser section was constructed to eliminate entry length effects. The experimental setup test sections were composed of a double fluted tube condenser, and evaporator. During condensation tests, the refrigerant flows in the outer annulus of the double fluted tube with 0.0324-m envelope diameter and water flows counter-currently inside the inner tube. The inner tube was made of copper with dimension as follows; bore inside diameter 0.0212 m, bore outside diameter 0.0226 m, number of flutes 4 and total effective heat transfer area 0.431 m^2. The outer tube has a smooth surface and is made of copper with an inner diameter 0.0375 m. The outer annulus was subdivided into 20 sections to measure the temperatures. This was necessary to measure the local heat transfer characteristics. Pressure, temperature and flow rate measuring stations are shown in Fig .1. All pressures were measured using calibrated pressure transducers (0-3400 kPa). The accuracy of pressure transducers was ±2.5%. Differential pressure transducers were employed to measure the refrigerant pressure drop. Temperatures were measured by thermocouples type J and K. Temperature measurements accuracy was within ±1K.

All recorded measurements were obtained at a sink water entering temperature of 21°C. On the other hand, the water-cooled pre-condenser was employed to control the quality of the saturated vapor entering the condenser. The quality of the refrigerant entering the test condenser was kept at 0.90. The temperature difference across the condenser was controlled to achieve saturated liquid at the outlet of the condenser.

A calibrated orifice installed in the liquid line after a liquid receiver was used to measure the refrigerant mass flow rate. Both orifice pressure taps were connected to a differential pressure transducer (0-70 kPa). Water mass flow rate was also measured by a calibrated orifice. The accuracy of the mass flow measurements was ±3% of the nominal flow. Power supplied to the compressor and fans was measured in order to complete heat balance. An AC/DC clamp-on device was calibrated for power measurements with accuracy of ±2.5%.

Data collection was carried out using a Pentium-133 equipped with a data acquisition system having a capacity of 112 channels. This enabled us to record with a single scan, local properties such as pressure drops, pressures, temperatures, refrigerant as well as coolant flow rates, heat flux and power. All tests were performed under steady state conditions. The channels were scanned every second and stored every 10 seconds.



LINE

···········	REFRIGERENT
-·-·-·-·-	SOLUTION
-··-··-··-	WATER
---------	STEAM

LEGEND:

1. COMPRESSOR
2. PRE-CONDENSER
3. TEST CONDENSER
4. SIGHT GLASS
5. ORIFICE (FLOW METER)
6. FILTER
7. CAPILLARY TUBES (WITH VALVES)
8. PRE-EVAPORATOR
9. TEST EVAPORATOR
10. HEAT EXCHANGER
11. SOLUTION STORAGE TANK
12. ELECTRIC HEATER
13. SOLUTION PUMP

FIG. 1
Schematic Diagram of Water/Water Heat Pump Test Facility

FIG. 2
Schematic View of Test Condenser

The measured values were averaged over a period of 10 seconds and were mixture mass flux, heat flux, and quality for refrigerants; R-502 (R22/R115: 48.8/51.95%), R-22 as well as azeotropic refrigerant mixtures R-410a (R125/R32: 50/50%), R-507 (R125/R143a: 50/50%). Mass flow rates ranged from 5 to 30 g/s. Input quality was kept constant at 0.9 and condensation took place so as to reach saturation condition at the exit of the condenser. Other key parameters test conditions are given in Table 1. Readers interested in further details concerning experimentations are advised to consult Sami and Poirier [6].

The primary heat transfer condensation parameters observed during the course of this study were mixture mass flux, heat flux, and quality for refrigerants, R-502 (R22/R115: 48.8/51.95%), R22 as well as azeotropic refrigerant mixtures R-410a (R125/R32: 50/50%), R-507 (R125/R143a: 50/50%). Mass flow rates ranged from 5 to 30 g/s. Input quality was kept constant at 0.9 and condensation took place so as to reach saturation condition at the exit of the condenser. Other key parameters test conditions are given in Table 1. Readers interested in further details concerning experimentations are advised to consult Sami and Poirier [6].

TABLE 1
Test Conditions

Temperature of Refrigerant at Condenser Inlet	30) 80° C
Temperature of Refrigerant at the Evaporator Inlet	-25EC) +4EC
Reynolds Number at Condenser	4E06) 2E07
Refrigerant Mass Flow Rate	0.007) 0.028 kg/s
Condenser Pressure	1250) 1723 kPa
Evaporator Pressure	180) 350 kPa
Inlet water Temperature at Condenser	21EC
Reynolds Number for Evaporator	5E03)1E04

TABLE 1-a
Test Section Specifications

Number of flutes	4
Bore inside diameter (m)	0.0212
Bore outside diameter(m)	0.0226
Envelope diameter(m)	0.0324
Length of condenser(m)	0.94
Length of evaporator(m)	0.812

In order to develop the proposed correlations describing the flow condensation heat transfer characteristics, the thermodynamic properties as well as transport properties of pure and non-azeotropic refrigerant mixtures should be known. The REFPROP version 5.12(Gallagher et al. [9]) was employed with caution, in selecting the interaction parameters, to determine the transport and thermodynamic properties of the mixed refrigerants.

Results and Discussion

In the following sections, the results of the heat transfer characteristics such as pressure drops, heat transfer coefficients at different conditions will be presented and discussed. Summary of the test conditions is presented in Table 1.

Heat Transfer

The following equations have been employed to calculate the heat transfer coefficients from the data stored during each particular test at equilibrium conditions. The heat transfer rate in the condenser test section can be determined from the heat balance of the water flow in the annulus;

$$Q_w = \dot{m}_w C_{p_w} \left(T_{wout} - T_{win}\right) \tag{1}$$

The vapor quality at the exit to the test section was calculated from an energy balance of the system;

$$x_{out} = x_{in} + \frac{Q_w}{\dot{m}_r \; h_{fg}} \tag{2}$$

The refrigerant properties were determined at saturation conditions in the test section.
The total heat transferred to the refrigerant is;

$$Q_n = \dot{m}_r \, h_{fg} \, \Delta x \tag{3}$$

Where x is the quality change in the test section.
The overall heat transfer coefficient based on the outside surface area of the test section is;

$$U = \frac{Q_n}{A_o \, LMTD} \tag{4}$$

Where LMTD is the mean logarithmic temperature difference based on the inlet/outlet temperatures of water/refrigerant flows. Assuming no fouling and R_w is the thermal resistance in the copper wall of tube, the refrigerant heat transfer coefficient h_r can be calculated as follows;

$$\frac{1}{h_r \; A_i} = \frac{1}{U \, A_o} - \left(\frac{1}{h_w \, A_o} + R_w\right) \tag{5}$$

Where h_w is the water heat transfer coefficient and is calculated using the Wilson plot technique as described in Khartabil et al. [10]. R_w is the wall resistance evaluated using the actual thickness and the outside diameter of the tube. During the course of this study for data resolution purposes, the enhanced surface tube (doubly fluted tube) annulus has been treated as a plain tube annulus with an equivalent diameter. Using the equivalent diameter is consistent with the approach suggested by others Sami et al. [8] for the annuli of a horizontal double-tube condenser with an enhanced inner tube. The reliability of the testing facility has been checked out by comparing the correlations proposed by Richards et al. [11], and Sami and Schnotale [7] for single phase turbulent water flow and pure, as well as refrigerant mixtures.

Excellent agreement was obtained. Interested readers are advised to consult Sami and Schnotale [7] and Sami et al. [8].

Condensation Characteristic

Samples of series of tests on the condensation for R-410a (R32/R125; 50/50%), R507 (R125/R143a; 50/50%), R-22, as well as R-502 (R22/R115: 48.8/51.2%) have been presented in Figures 3 and 4 where the condensation heat transfer coefficient and Nusselt number are plotted versus Reynolds number. The data presented in these figures clearly indicated that the condensation heat transfer coefficient of the blends increases as the mass flux and Reynolds number increase. However, through examination of the data displayed in these figures also demonstrates that R-507 blend has the highest heat transfer coefficient among the blends under investigation as well as R-22 at Reynolds number lower than 4.2 E06. However, for Reynolds numbers higher than 4.2E06, R-410a appears to have greater heat transfer rates more than the other blends under investigation. It is believed that the significant increase in the heat transfer coefficient of R-507 is due to higher latent heats of two components of this blend; R-32 and R-125.

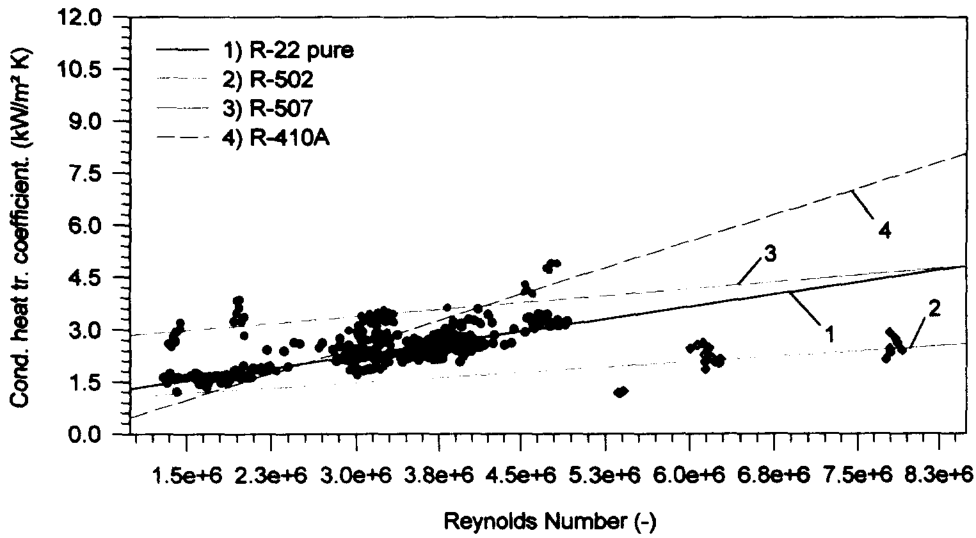


FIG. 3
Condensing heat transfer coefficient versus Reynolds Number

Furthermore, to study the impact of heat flux on the heat transfer coefficient, Figure .5 has been established where the binary refrigerant blends under investigation R-410a and R-507 have been compared to R-502 as well as R-22. The data showed that R-502 has the highest heat transfer rate at heat flux lower than

25.3 kw/m^2 However, at higher heat flux, R-507 appears to have similar heat transfer rates to that of R-502. It is quite clear from the data that R-410a has the lowest heat transfer rate. This is attributed to the lower latent heat of R-32 compared to that of R-143a.

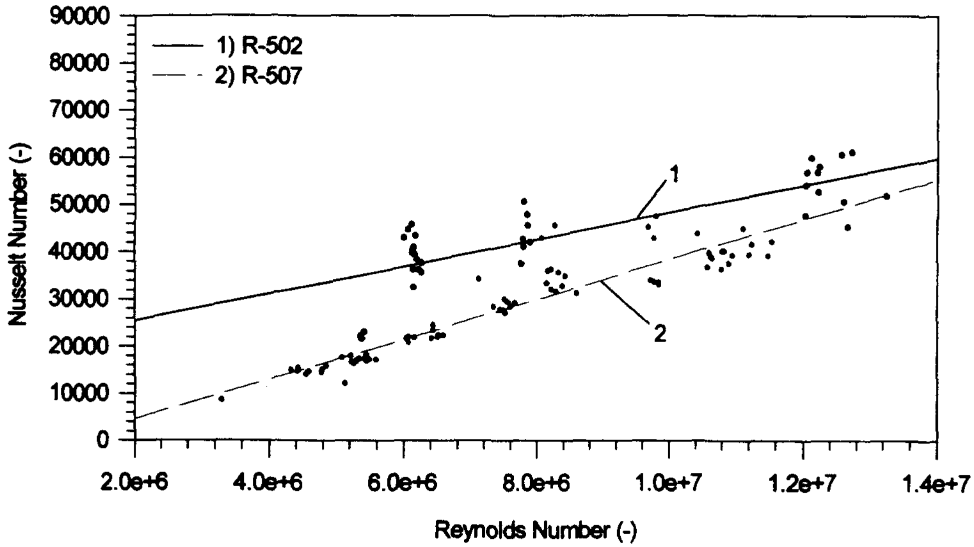


FIG. 4
Nusselt Number for condensation versus Reynolds Number.

Pressure Drop Characteristics

In the flow conditions under investigation, the total two-phase pressure drop during boiling and condensation of pure or azeotropic refrigerant mixtures is given by;

$$\Delta P_{total} = \Delta P_f + \Delta P_{ac} + \Delta P_g \qquad (6)$$

where ;Δp_{total} ;total two-phase pressure drop, Δp_f ; frictional pressure, Δp_{ac} ; acceleration pressure drop, and Δp_g ; gravitational pressure drop.

In horizontal geometry, the gravitational pressure drop can be neglected [8] and Equation (6) can be rewritten as follows:

$$\frac{dP}{dZ} = \frac{dP}{dZ}\Big|_f + \frac{dP}{dZ}\Big|_M . \qquad (7)$$

The two-phase flow pressure drop for a working fluid undergoing evaporation and/or condensation process over a length Z [7] can be written as follows:

$$\frac{dP}{dZ}\Big|_{tp} = \frac{dP}{dZ}\Big|_{f} + \frac{dP}{dZ}\Big|_{M}. \tag{8}$$

The momentum pressure drop for a working fluid undergoing evaporation and/or condensation process over a length Z [7] can also be written in the following form:

$$\frac{dP}{dZ}\Big|_{M} = G^{2}\,\frac{d}{dZ}\left[\frac{x^{2}}{\rho_{v}\,\alpha} + \frac{(1-x)^{2}}{\rho_{l}(1-\alpha)}\right] \tag{9}$$

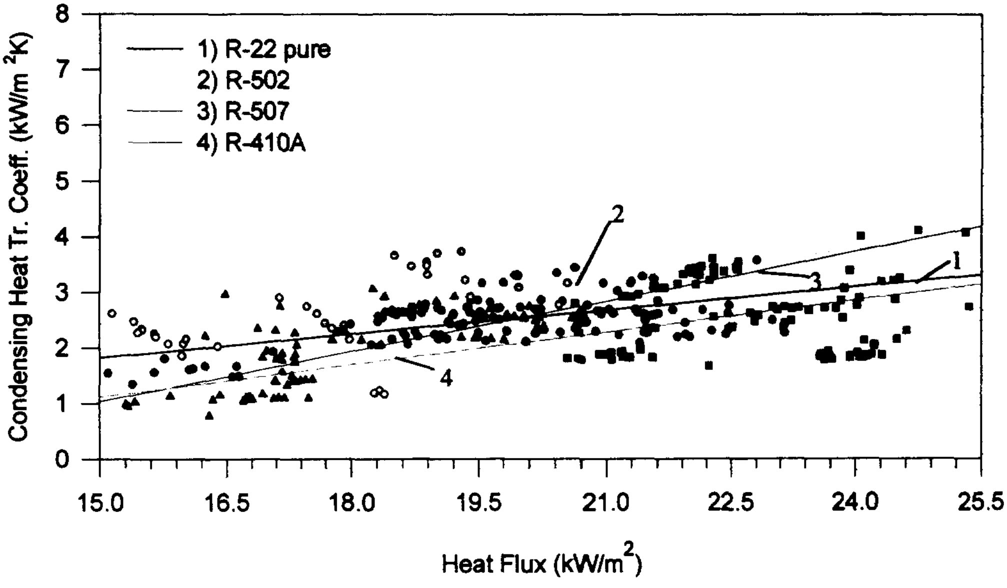


FIG. 5
Condensing Heat Transfer Coefficient versus Heat Flux.

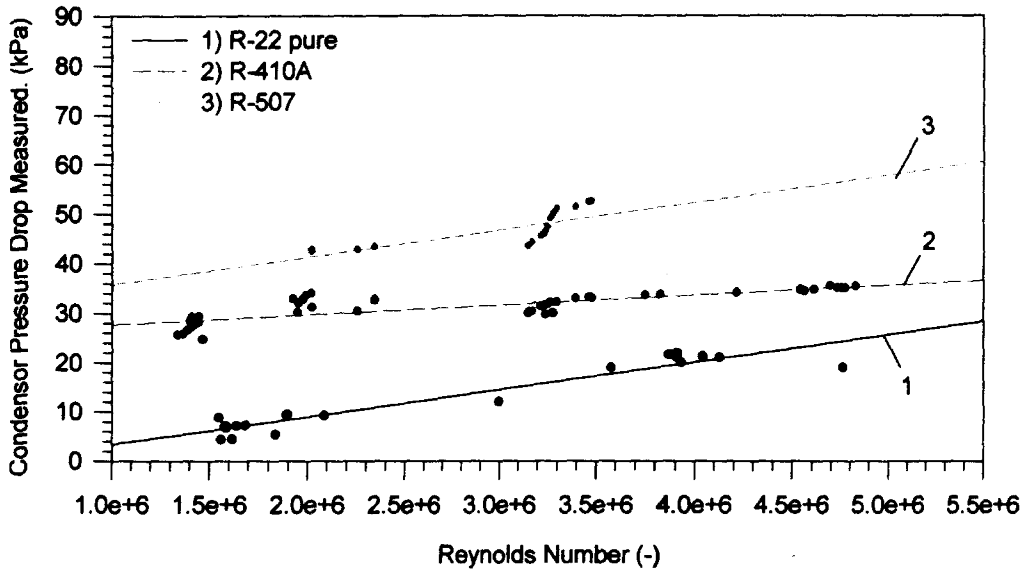


FIG. 6
Pressure Drop across Condenser versus Reynolds Number.

where α is the void fraction and x is the flow quality. Sami and Schnotale [7] have reported that the acceleration term in Equation (9) is negligible. Therefore, only the friction pressure drop is considered in this study.

The frictional pressure gradient for R-410a, R-507, and R-22 as well as R-502 obtained during forced convective condensation have been plotted against the refrigerant Reynolds number in Figures 6 and 7. The data presented in these figures show that as the refrigerant Reynolds number increases the pressure drop increases. Furthermore, it is quite evident from this data that R-507 has the highest pressure drop among the refrigerants under investigation.

It appears from the condensation and boiling pressure drop data that the use of new alternative refrigerants such as R-507 will require compressors with larger volumetric capacities than that of R-22 and R-502. In addition, in order to obtain thermal capacities similar to that of R-22 and R-502 better designed heat exchangers will be needed.

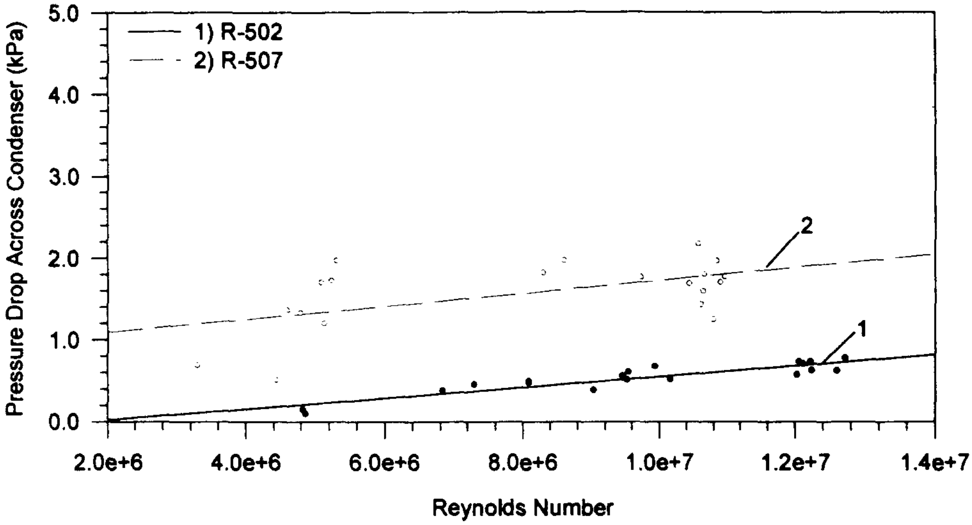


FIG. 7
Pressure Drop across Condenser versus Reynolds Number.

Conclusions

During the course of this comparative study, the condensation heat transfer characteristics of the newly proposed binary refrigerant blends as substitutes for R-22 and R-502 applications have been investigated inside enhanced-surface tubing under different conditions. The results showed that caution should be exercised when retrofitting existing systems using R-22 and R-502 with alternatives such as R-507.

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