

## HORIZONTAL FLOW BOILING OF ALTERNATIVE REFRIGERANTS WITHIN A FLUID HEATED MICRO-FIN TUBE

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### ABSTRACT

This paper presents local, fluid-heated, flow boiling measurements for three refrigerant mixtures: (1) R410B (R32/125, 45/55 % mass), (2) R32/R134a (28/72 % mass) and, (3) R407C (R32/125/134a, 25/23/52 % mass). Flow boiling heat transfer coefficients for the mixtures' pure components and R22 were also measured to establish a baseline for the heat transfer degradation calculations. The heat transfer degradation was shown to be a relatively strong function of heat flux and thermodynamic mass quality. The heat transfer degradation associated with the R32/125 mixture was believed to be mostly due to nonlinear property effects. The R410B appears to be a good alternative refrigerant for R22 having approximately a 20% larger heat transfer coefficient. The degradation of the R407C was shown to increase with heat flux for all qualities except for 0.6. The dependency of the R32/134a degradation was not consistent with heat flux. The influence of nonlinear property effects was apparently responsible for the inconsistency. Correlations of the two-phase Nusselt number were derived for each fluid.

**Keywords:** Enhanced heat transfer, micro-fin, refrigerant mixtures, fluid heating

### NOMENCLATURE

#### English symbols

$A_c$  cross sectional flow area inside tube ( $m^2$ )  
 $Bo$  local boiling number,  $\frac{q''}{Gh_{fg}}$   
 $c_p$  specific heat ( $kJ/kg \cdot K$ )  
 $D_h$  hydraulic diameter ( $m$ )  
 $D_e$  equivalent inner diameter of micro-fin tube,  $\sqrt{\frac{4A_c}{\pi}}$  ( $m$ )  
 $G$  mass velocity ( $kg/m^2 \cdot s$ )  
 $h_{2\phi}$  local two-phase heat-transfer coefficient ( $W/m^2 \cdot K$ )  
 $h_{fg}$  latent heat of vaporization ( $kJ/kg$ )

$k$  liquid thermal conductivity ( $W/m \cdot K$ )  
 $Nu$  local Nusselt number based on  $D_i$   
 $\dot{m}$  mass flow rate ( $kg/s$ )  
 $m$  wetted perimeter of inner micro-fin tube ( $m$ )  
 $P$  local fluid pressure ( $Pa$ )  
 $q''$  local heat flux ( $W/m^2$ )  
 $Re$  all liquid, Reynolds number based on  $D_e$   
 $T$  temperature ( $K$ )  
 $U$  expanded uncertainty  
 $x$  mole fraction  
 $x_q$  thermodynamic mass quality  
 $z$  axial distance ( $m$ )

#### Greek symbols

$\Delta h_{2\phi}$  heat transfer degradation  
 $\rho$  liquid density ( $kg/m^3$ )  
 $\mu$  liquid dynamic viscosity ( $kg/m \cdot s$ )

#### subscripts

$f$  water  
 $i$  inlet  
 $l$  liquid  
 $L$  linear interpolation  
 $m$  mixture  
 $r$  refrigerant  
 $w$  heat transfer surface  
 $v$  vapor

## INTRODUCTION

The local flow boiling heat transfer coefficient of three refrigerant mixtures and their pure components were experimentally determined as part of the R22 Alternate Refrigerant Evaluation Program (AREP). There were two main objectives of the study. The first was to characterize the flow boiling performance of the R22 alternatives in a micro-fin tube. The knowledge gained can assist in the design and development of new air-conditioning and refrigerant products. The second objective was to fundamentally characterize the heat transfer degradation/enhancement of the mixed refrigerants relative to the performance of the pure components. A fundamental understanding of mixture heat transfer mechanisms can be used to refine the selection of mixture composition and/or mixtures for alternative refrigerants.

Most of the refrigerant mixture flow boiling data in the literature is for smooth tubes. Works such as those by Ross et al. (1987), Jung et al. (1989), and Niederkrueger et al. (1992) present local measurements obtained from an electrically heated test section. Local heat transfer measurements provide a greater understanding of the phenomena. However, electric heating is not a physically realistic boundary condition for refrigerant applications. The most pertinent and useful data are local measurements obtained from a fluid heated rig. Even so, the majority of studies where the test section is fluid heated are not local heat transfer measurements. For example, Eckels and Pate (1991), Torikoshi and Ebisu (1993), and Tuzla et al. (1994) present global heat transfer coefficient measurements from fluid heated test sections. Consequently, there is a need for local flow boiling heat transfer measurements obtained by a fluid heating boundary condition.

Local-averaged fluid heated data can be obtained from a test section that is divided into several segments. The heat transfer coefficient is averaged over a particular range of qualities for each segment providing quasi-local (local-averaged) data. Kattan, et al. (1995), Goto, et al. (1995), and Conklin and Vineyard (1992)

provide examples of local-averaged flow boiling experiments. If the variation of the heat transfer coefficient over the quality range for which it is averaged is small, only a small uncertainty is introduced by the averaging process. This criterion cannot always be satisfied for flow boiling experiments. In some cases, true local measurements are necessary to accurately capture flow boiling physics.

Unfortunately, few local, fluid-heated, flow-boiling heat transfer data are available for refrigerant mixtures. Consequently, the purpose of this study is to provide local flow boiling measurements for some pertinent refrigerant mixtures: (1) R410B (R32/125, 45/55 % mass), (2) R32/R134a (28/72 % mass) and, (3) R407C (R32/125/134a, 25/23/52 % mass). Flow boiling heat transfer coefficients for the mixtures' pure components and R22 were also measured to establish a baseline for the heat transfer degradation calculations. Local measurements are required to measure the heat transfer degradation with a relatively small uncertainty. In this way, the data can contribute to a fundamental understanding of refrigerant mixture heat transfer.

## TEST APPARATUS

Figure 1 shows a sketch of the experimental apparatus used to measure the flow boiling. The test section consisted of a pair of 3.34 m long, horizontal micro-fin tubes connected by a U-bend. The test refrigerant flowed inside the test section. Two condensers balanced the heat load of the test section and maintained a fixed test pressure. A magnetically coupled gear pump delivered the test refrigerant to the entrance of the test section with a few degrees of subcooling. The range of mass flow rates were 0.019 kg/s to 0.022 kg/s. Distilled water flowed counterflow in the annulus that surrounded the micro-fin tube. Another magnetically coupled gear pump was used to maintain the water flow rate. The temperature of the water loop was maintained with a water chilled heat exchanger. The refrigerant and water flow rates were controlled by varying the pump speeds with frequency inverters.

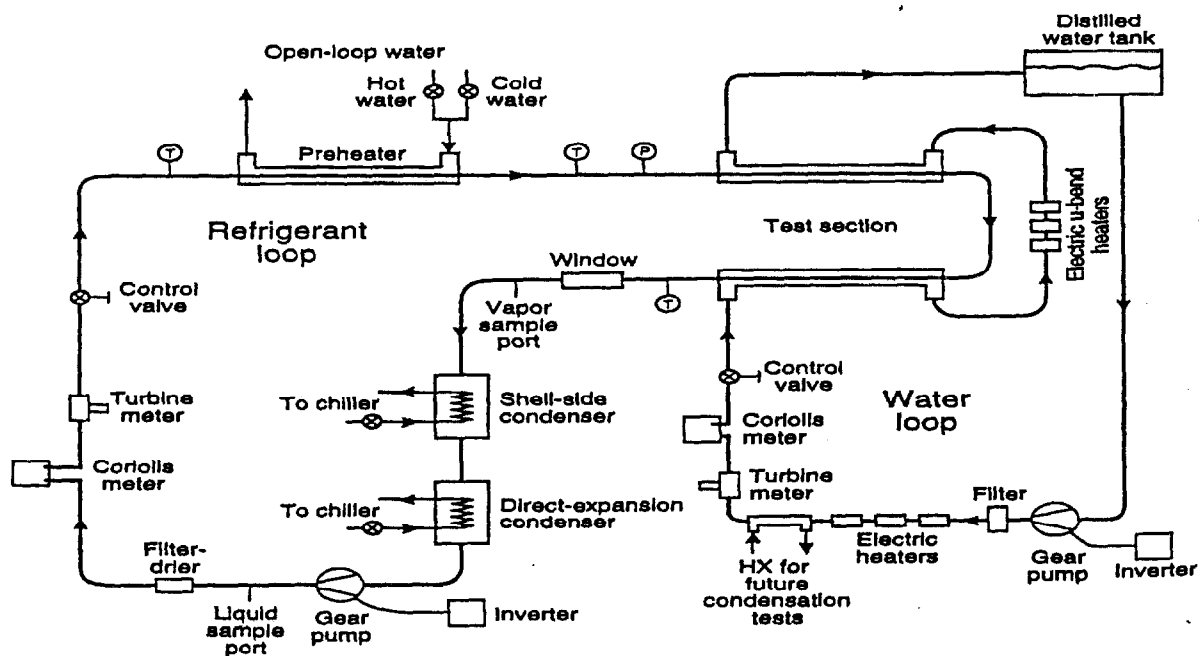


Figure 1. Schematic of test rig

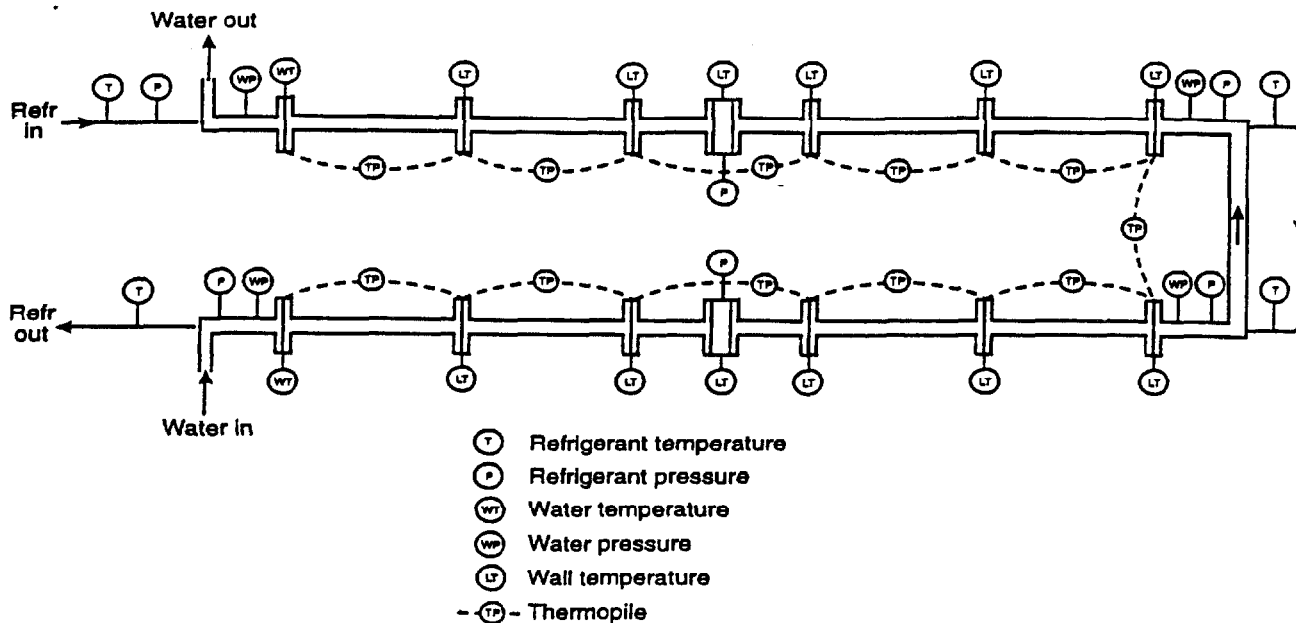


Figure 2. Detailed schematic of test section annulus

Figure 2 shows how the annulus was constructed by connecting a series of tubes with stainless steel flanges. Figure 3 shows a cross section of the test section. Thermopile and thermocouple wires pass between gasketed flanges to measure local water and refrigerant-tube wall temperatures. The annulus gap was 2.2 mm, and the micro-fin tube wall thickness was 0.3 mm. A series of teflon half-rings attached to the refrigerant tube centered the tube in the annulus. The half-rings were circumferentially baffled to mix the water flow. Mixing was further ensured by a high water Reynolds number (Kattan et al. 1995).

Six refrigerant pressure taps along the test section allowed the measurement of the upstream absolute pressure and five pressure drops along the test section. Two sets of two water pressure taps were used to measure the water pressure drop along each tube. Three sets of type-T thermocouples were mounted on the test section to measure the wall, water, and refrigerant temperatures. The first set consisted of 16 wall thermocouples on the outside of each refrigerant tube. At five locations for each tube, thermocouples were mounted on the top, side, and bottom of the tube wall. These locations were separated by 0.6m on average, and

they were located near the intersection of the shell flanges. The sixteenth wall thermocouple for each tube was mounted at the same point as the middle pressure taps. The thermocouple junction was soldered to the outside surface and was sanded to a thickness of 0.5 mm. The leads were strapped to a thin non-electrically-conducting epoxy layer on the wall for a distance of 14.3 mm before they passed between a pair of the shell flanges. The wall temperature was corrected for a heat flux dependent fin effect. The correction was typically 0.05 K.

A chain of thermopiles measured the change in the water temperature between six points separated by approximately 0.6 m on each tube and across the U-bend. Each thermopile consisted of ten thermocouples in series, with the ten junctions at each end evenly spaced around the circumference of the annulus. Because the upstream junctions of one thermopile and the downstream junctions of another enter the annulus at the same axial location (except at the water inlet and outlet), the junctions of the adjacent piles were alternated around the circumference.

A sheathed thermocouple measured the refrigerant temperature at each end of the two refrigerant tubes, with the junction of each positioned at the center radially. Only the thermocouple at the inlet of the first tube was used in the calculations. The entire test section was wrapped with 50 mm of foam insulation to minimize heat transfer between the water and the ambient.

### HEAT TRANSFER COEFFICIENTS

The flow boiling heat transfer coefficient ( $h_{2\phi}$ ) was calculated as:

$$h_{2\phi} = \frac{q''}{T_w - T_r} \quad (1)$$

where the measured wall temperatures ( $T_w$ ) were fitted to their axial position to reduce the uncertainty in the measurement. The enthalpy rise of the refrigerant was calculated with a energy

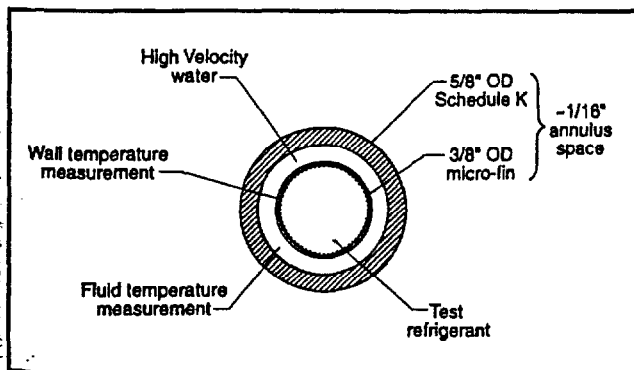


Figure 3. Test section cross section

Table 1. Typical Estimated Relative Expanded Uncertainties (%) for High-Quality Flow Condition

$h_{sp}$	$q''$	$T_w$	$T_f$	D	$c_p$	$dT/dz$	$m_f$	$\rho$	$dP/dz$	P	$T_f$	Re
13	0.7	5	3.3	0.3	0.3	0.6	0.1	0.1	7.7	0.4	1	0.1

Table 2. Coefficients For Nu Correlation

Fluid	K	a RE	b REX	c REXX	d BO	e BOX	f BOXX	gd
R22	6.355	0.321	-0.339	0	0.367	-0.466	0	0
R32	6.008	0.317	0	-0.366	0.313	0	-0.495	0
R125	5.371	0	2.161	-2.587	-0.137	2.333	-2.841	0
R134a	3.814	0.224	1.205	-1.293	0	0.992	-1.134	0
R32/R125	4.093	0.363	1.166	-1.378	0.150	1.215	-1.460	0
R32/R134a	2.865	0.886	-0.637	0	0.621	-0.800	0	0
R407C	-7.450	-0.397	1.500	-1.351	0	1.384	-1.239	-9.031

balance along the test section. The refrigerant pressures were measured at six pressure taps along the test section. The pressure was linearly interpolating between the taps. The equilibrium refrigerant temperature ( $T_f$ ) was calculated with REFPROP (Huber et al. 1995) with enthalpy and pressure as inputs. The average  $T_f$  was held constant at 4.4°C for each test.

The heat flux ( $q''$ ) was determined from a fit of the water temperature profile and the water mass flow rate. The water temperature fit was determined from the location of the thermopiles and the temperature drops obtained from each thermopile. The

$$q'' = \frac{m_f}{\pi p} \left( C_{pr} \frac{dT_f}{dz} \right)_{x_1} + \frac{1}{\rho} \frac{dP_f}{dz} \quad (2)$$

specific heat ( $C_p$ ) and the density ( $\rho$ ) of the water were axially calculated as a function of temperature. The axial water temperature gradient ( $dT/dz$ ) was also calculated. The water pressure gradient ( $dP/dz$ ) was assumed to be linear between the pressure taps. The heat flux was calculated as:

Table 1 shows expanded uncertainty  $U$  of the various measurements. The  $U$  is commonly referred to as the law of propagation of uncertainty. All measurement uncertainties are reported for a 95% confidence interval and are evaluated by statistical methods. The estimates shown in Table 1 were typical for the high quality region where the greatest  $U$  were observed.

## RESULTS

The flow boiling Nusselt numbers ( $Nu$ ) were correlated to the boiling number ( $Bo$ ), the Reynolds number ( $Re$ ), the quality ( $x_q$ ), and the R32 mole composition difference ( $x_r - x_l$ ) with:

$$Nu = \frac{hk_l}{D_h} = K Re^\alpha Bo^\beta (x_r - x_l)^\epsilon \quad (3)$$

where:  $\alpha = a + bx_q + cx_q^2$   
 $\beta = d + ex_q + fx_q^2$

Table 2 provides the constants  $K$  and  $a$  through  $g$  that were determined for each of the test fluids. Typically, the correlation is within  $\pm 1.5\%$  of the measured value for 95% confidence. The

quadratic exponent form was used with good results by Kedzierni and Kim (1996) to correlate several other pure refrigerants and mixtures for a wide range of qualities for both evaporative and condensing flows.

Figures 4 through 9 show plots of the flow boiling heat transfer coefficient versus either the heat flux or thermodynamic quality. All figures are presented for a mass flux of 314 kg/(m<sup>2</sup> s). Figures 4 through 6 show how the effect of the heat flux on the heat transfer coefficient changes for increasing quality. As expected, the heat flux has the greatest influence on the heat transfer coefficient at qualities less than 0.4 where nucleate boiling is expected to be prevalent. However, the influence of the heat flux remains significant for  $x_q = 0.6$ . Specifically, the average slope of the  $h_{sp}$  versus  $q''$  plot for all the fluids in the low-quality region was reduced by only 35% for a transition to  $x_q = 0.6$ . A similar heat flux dependence in the high-quality region was evident in the data of Kattan et al. (1995). Although, the heat flux dependence in the convective region ( $x_q = 0.6$ ) may suggest that nucleate boiling is present, this conjecture must be verified by visual evidence. Possibly, the convective region is influenced by heat flux in the absence of nucleate boiling.

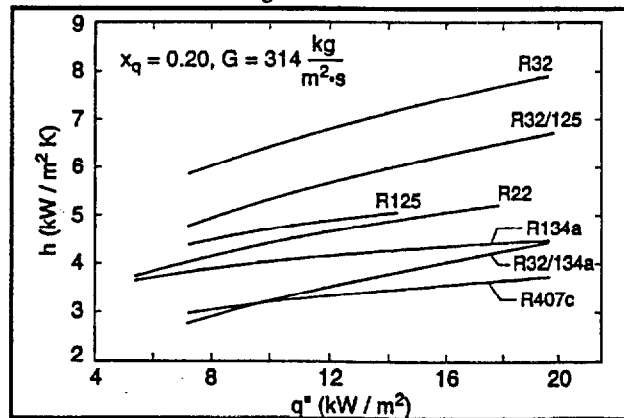


Figure 4. Influence of heat flux on flow boiling heat transfer coefficient for  $x_q = 0.2$

Figures 7 through 9 show the dependence of the heat transfer coefficient on the thermodynamic quality. The heat transfer coefficient increases modestly with respect to quality. Comparison of figures 7 through 9 reveal that variation of the heat

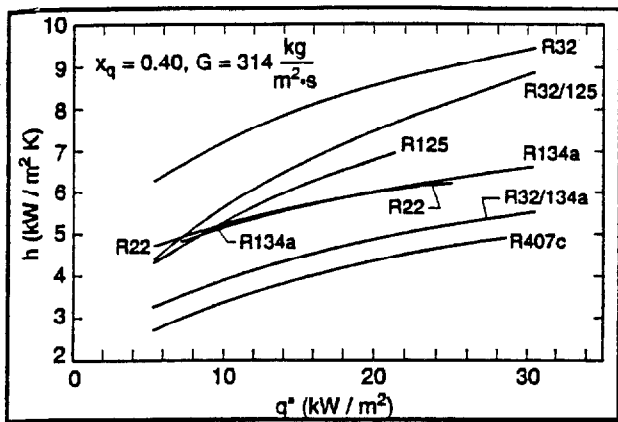


Figure 5. Influence of heat flux on flow boiling heat transfer coefficient for  $x_q = 0.4$

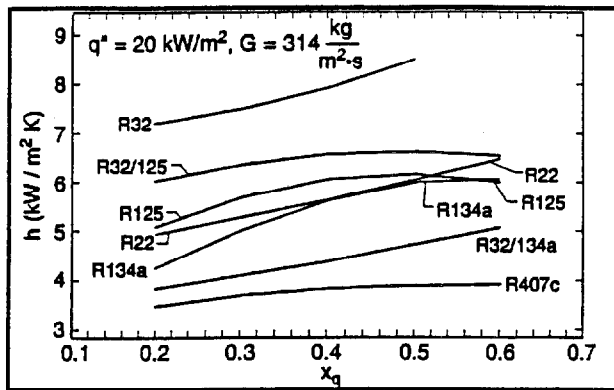


Figure 8. Influence of quality on flow boiling heat transfer coefficient for  $q'' = 20 \text{ kW/m}^2$

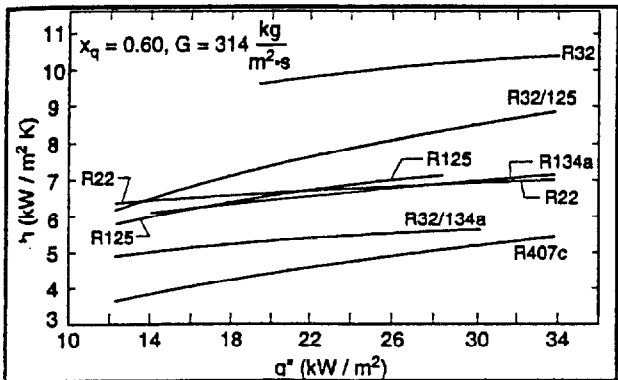


Figure 6. Influence of heat flux on flow boiling heat transfer coefficient for  $x_q = 0.6$

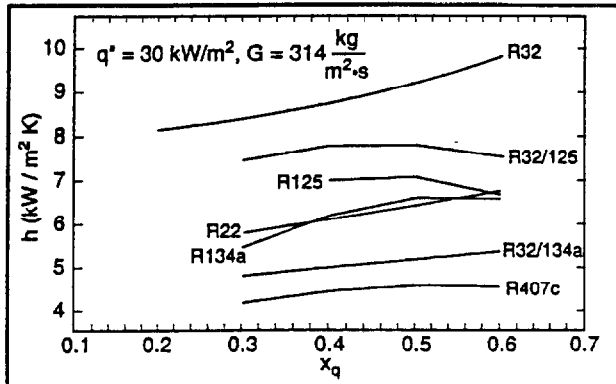


Figure 9. Influence of quality on flow boiling heat transfer coefficient for  $q'' = 30 \text{ kW/m}^2$

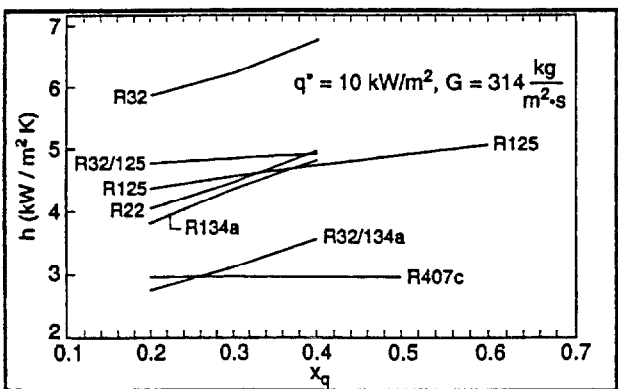


Figure 7. Influence of quality on flow boiling heat transfer coefficient for  $q'' = 10 \text{ kW/m}^2$

performance compared to R22 because it is a near-zeotropic mixture of two refrigerants that each have a greater heat transfer performance than R22. The performance of R32/125 lies between the performances of R32 and R125 due to the lack of concentration gradients in the liquid. When R32 and R125 are mixed with R134a, which has a heat transfer performance similar to R22, the heat transfer performance of the resulting zeotropic ternary mixture (R407C) is considerably lower than R22. The addition of R134a to the near-zeotropic (R410B) has created a zeotropic mixture (R407C) with a significant heat transfer degradation.

flux effect with quality varies from fluid to fluid. For some fluids, the influence of heat flux remains relatively constant with quality. Other fluids show a decreasing, increasing, or parabolic relationship with respect to quality.

The R410B (R32/125) appears to be a good alternative refrigerant for R22 which has approximately a 20% larger heat transfer coefficient. The R410B has a greater heat transfer

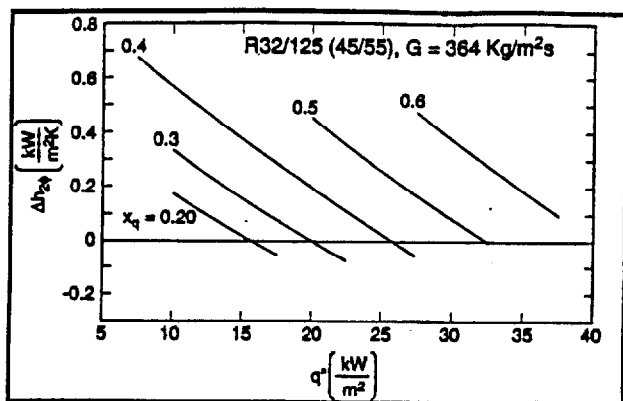


Figure 10. Heat transfer degradation for R32/125 at  $G=364 \text{ kg/m}^2\text{s}$

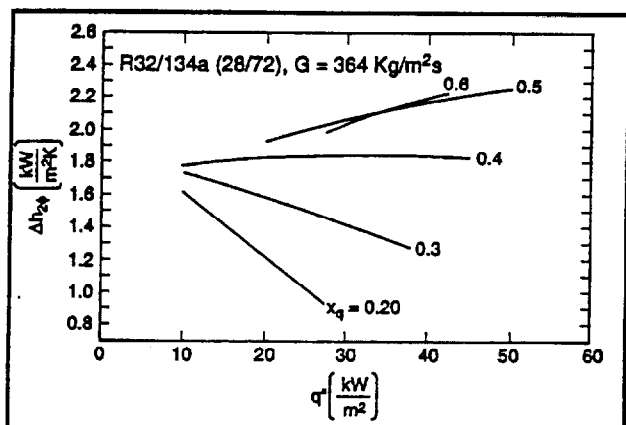


Figure 11. Heat transfer degradation for R407C at  $G=364 \text{ kg/m}^2\text{s}$

In general, pure R32 and the ternary R407C have the highest and the lowest heat transfer coefficients, respectively. The R407C has the lowest heat transfer of the fluids tested because it is the zeotropic mixture containing the least amount of R32. R32 contributes to favorable convective properties and possibly the dissipation of concentration gradients. As described by Kedzierski et al. (1992), small concentration gradients tend to exist for mixtures where the more volatile component has a small molecular mass. Light molecules diffuse more quickly to dissipate concentration gradients. The molecular mass of R32 is the smallest of the pure refrigerants tested. Likewise, because the R32/R134a mixture contains more R32 than the R407C mixture, the R32/R134a mixture has a slightly greater performance.

Figures 10 through 12 plot the heat transfer degradation ( $\Delta h_{24}$ ) as a function of heat flux for the three mixtures at a mass flux of  $364 \text{ kg/m}^2\text{s}$ . The  $\Delta h_{24}$  was calculated from the correlations using the same definition as given in Kedzierski et al. (1992):

$$\Delta h_{24} = h_L - h_{24} \quad (4)$$

where  $h_L$  is the heat transfer coefficient obtained from a linear interpolation of the pure components at a given composition. Each plot contains five lines of constant quality spanning 0.2 to 0.6.

Figure 10 shows that the degradation for the R32/125 mixture was, on average with respect to quality and heat flux, approximately  $215 \text{ W/m}^2\text{K}$ . The average difference between the liquid and vapor R32 mole fraction for this mixture was approximately 0.03. Mass diffusion is expected to be insignificant for this near-azeotropic refrigerant. Consequently, most of the degradation should be due to nonlinear property effects as described by Jung et al. (1989). Figure 10 shows a negative degradation, or an enhancement, of the heat transfer with respect to the pure components. Kedzierski et al. (1992) describe the enhancement as being possible for near-azeotropes where the heat transfer characteristics of the less volatile component are more favorable for heat transfer than those of the more volatile component. They suggested the use of the essence of the Dittus-Boelter (1930) equation  $(k_f^{0.6} (c_p/\mu)^{0.4}) \rho_1^{0.8}$  to predict the impact of each component's properties on the overall heat transfer. The R32 Dittus-Boelter parameter is 15% greater than that for R125 at the test temperature suggesting that the R32/125 should exhibit a degradation. Considering the above argument, the relatively small enhancement, and the uncertainty of the  $h_{24}$  measurement, the enhancement probably does not exist.

Figure 10 shows that the heat transfer degradation decreases with increasing heat flux. The reduction in  $\Delta h_{24}$  is a consequence of the relative effects of heat flux on the  $h_{24}$  of R125, R32 and the R32/125 mixture. The R32/125 mixture heat transfer was enhanced at a greater rate with respect to the heat flux than the pure components R32 and R125 were. Consequently, the heat transfer degradation decreased as the mixture heat transfer increased at a greater rate than the pure components.

Figure 11 shows that the degradation for the R407C mixture was, on average with respect to quality and heat flux, approximately  $2500 \text{ W/m}^2\text{K}$ . The average difference between the liquid and vapor R32 mole fraction for this mixture was approximately 0.12. Consequently, most of the heat transfer degradation is expected to be due to concentration gradients. The concentration gradients are accentuated by increasing heat flux. Accordingly, the heat transfer degradation should increase with heat flux. The R407C mixture exhibits an increasing degradation with increasing heat flux for all qualities except  $x_q = 0.6$ .

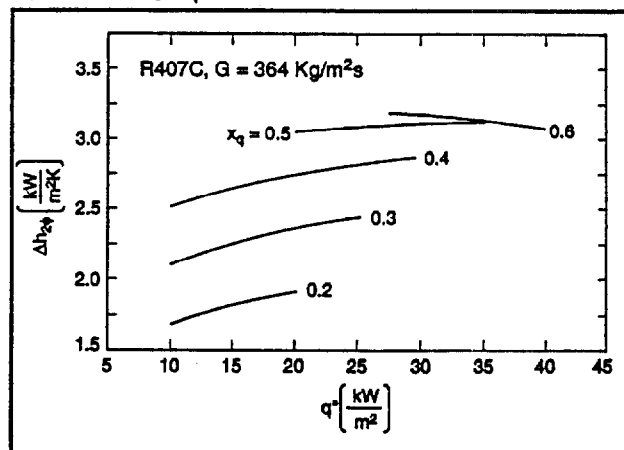


Figure 12. Heat transfer degradation for R32/134a at  $G=364 \text{ kg/m}^2\text{s}$

Figure 12 shows that the degradation for the R32/134a mixture was, on average with respect to quality and heat flux,

approximately 1800 W/m<sup>2</sup>·K. The average difference between the liquid and vapor R32 mole fraction for this mixture was approximately 0.19. Again, some heat transfer degradation due to concentration gradients is expected. The heat transfer degradation increases with heat flux for qualities above 0.4. However, the degradation decreases with heat flux for qualities less than 0.4. Because this behavior is similar to that observed for the near-zeotropic mixture, it is possibly that nonlinear property effects have a strong influence on the R32/R134a mixture for qualities less than 0.4.

## CONCLUSIONS

Local, fluid-heated, flow boiling measurements for three refrigerant mixtures: (1) R410B (R32/125, 45/55 % mass), (2) R32/R134a (28/72 % mass) and, (3) R407C (R32/125/134a, 25/23/52 % mass) were presented. Flow boiling heat transfer coefficients for the mixtures' pure components and R22 were also measured to establish a baseline for the heat transfer degradation calculations. The heat transfer degradation was shown to be a relatively strong function of heat flux and thermodynamic quality. Correlations of the two-phase Nusselt number were given for each fluid.

The heat transfer degradation associated with the R32/125 mixture was believed to be due primarily to nonlinear property effects. Although the mixture exhibits a heat transfer degradation relative to its pure components, it appears to be a good alternative refrigerant for R22, having approximately a 20% larger heat transfer coefficient. The favorable performance of the R32/125 mixture is largely due to the favorable transport properties of the mixture's pure components.

The R407C has the lowest heat transfer of the fluids tested perhaps because it is the zeotropic mixture containing the least amount of R32. Also, concentration gradients contributed to the heat transfer degradation of R407C mixture. The degradation was shown to increase with heat flux for all qualities except the highest tested. This is generally consistent with increasing concentration gradients with heat flux.

Most of the degradation of the R32/134a mixture was due to concentration gradients in the liquid. However, the dependency of the degradation was not consistent with heat flux. The heat transfer degradation increased and decreased with heat flux for qualities greater than and less than 0.4, respectively. Presumably, nonlinear property effects may become important for qualities less than 0.4.

## POSTSCRIPT

The heat transfer and thermodynamic properties of R407C and R410B influence their use in existing and new refrigeration systems, respectively. The R407C was tailored for existing "R22" refrigeration systems because it has vapor pressure similar to R22. The R407C lowers the capacity and/or efficiency of the existing equipment as it compares to R22. By contrast, R410B systems require smaller heat exchangers while maintaining a similar or greater efficiency than R22 systems. Smaller heat exchangers are a result of the greater heat transfer of R410B as compared to R22. The higher vapor pressure of R410B as compared to R22 results in smaller and less expensive systems to manufacture. Further improvements in system capacity and efficiency could be achieved by increasing the concentration of R32 in either the R410B or the R407C mixture. However, due to its flammable nature, safety considerations limit the concentration of R32 in the mixture to the present amounts.

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