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Horizontal flow boiling of pure and mixed refrigerants

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Abstract—The research involved determining experimental heat transfer coefficients for pure R152a and R13B1 and for four mixtures of these refrigerants. The mixtures yielded sharply lower heat transfer coefficients than either pure refrigerant. Correlative evidence suggests that full suppression of nucleate boiling is easier to achieve with mixtures than pure fluids. Existing correlations were compared to the data with success for both pure and mixed fluids.

1. INTRODUCTION

INTEREST has surged recently in the use of non-azeotropic mixtures of refrigerants as working fluids in heat pump applications, due to potential advantages in efficiency and capacity [1]. The prediction of their performance is complicated by uncertainty in thermophysical properties. Here, of particular concern is their behavior during evaporation and condensation.

The principal goal of this study therefore was to experimentally determine heat transfer coefficients, α, under a wide range of conditions, to assess and recommend models and correlations for their predictions, and to examine specific physical processes governing the heat transfer. The need for such work has been expressed widely in the recent literature [2–5].

The test fluids used in this investigation were pure R152a (CH_3CHF_2), pure R13B1 (CF_3Br) and mixtures of various compositions of these fluids. The two refrigerants are recommended as a mixture by their manufacturer for heat pump use due to their relatively wide difference in boiling point (35°C, 4.75 bar). In addition, the mixture can be used in existing machinery without major modification. Some experimental tests were also performed with pure R22 ($CHClF_2$), a much more widely utilized refrigerant. The R22 was used to ensure the experimental test rigs were operating correctly.

2. GENERAL DESCRIPTION OF TEST RIGS USED IN THIS INVESTIGATION

Through the course of this investigation, two experimental rigs were built and utilized. Both employed a horizontal stainless steel tube (length 2.7 m; inside diameter 0.9 cm; outside diameter 0.95 cm) which was electrically heated.

Rig 1 is described in Fig. 1. A semi-hermetic, oil free pump delivered subcooled liquid refrigerant to the test section. Heat was generated in the tube wall by applying a d.c. voltage difference along the tube. The test section itself was heavily insulated (approximately 15 cm radial thickness) to reduce heat gain from the surroundings: the remaining minimal heat gain was accounted for by calibration. The vapor generated in the test section was reliquified in an oversized condenser/receiver. The pump then drew on the liquid in the condenser to complete the cycle. Inlet subcooling and pressure level were controlled by valves in the liquid line. Subcooling and pressure level could also be modified by altering the condenser temperature by changing the flow rate or the temperature of the brine side of the condenser.

Pressures were also measured at the single phase liquid inlet and two phase outlet but not in the heated section. Tubular sight glasses located at the test section inlet and outlet allowed visual verification of the flow pattern.

Flow rates were determined by means of a calibrated turbine meter in the subcooled liquid line (estimated 2% error).

One of the features of Rig 1 was the use of a uniform heat flux (estimated 10% error) along a fixed tube length. To reach high exit qualities with low heat flux a tube length of more than 20 m is required in some cases. This straight length was unavailable. Alternately, if high exit vapor quality at a specified mass flux is desired, a large heat flux could be used. However, boiling phenomena is itself a strong function of heat flux. Therefore, the use of a large heat

NOMENCLATURE

ac	diffusivity
C_p	specific heat at constant pressure
D	diameter of tube
F	factor in Chen's method
G	mass flux
h	enthalpy
Δh_v	latent heat of vaporization
L	length of tube
$\dot{m}$	mass flow rate
P	pressure
Pr	Prandtl number, $\mu C_p/\lambda$
Q_s	suppression heat flux
q	heat flux
Re	Reynolds numbers, GD/μ
T	temperature
X	concentration of liquid
x	vapor quality
Y	concentration of vapor
y	distance from wall
y^+	viscous sublayer thickness, $\sqrt{(\tau/\rho)}y\rho/\mu$.

Greek symbols

α	heat transfer coefficient
δ	thickness (length) of layer
λ	conductivity
μ	viscosity
ρ	density
σ	surface tension
τ	wall shear.

Superscripts

$\bar{}$	molar value
$*$	vapor value in equilibrium with liquid.

Subscripts

B	bulk
D	mass
EPF	equivalent pure fluid
f	fluid
G	gas
L	liquid
LO	liquid only
MIX	mixture
out	outlet
pr	preheat
sat	saturation
T	thermal
v	vapor
w	wall.

flux to provide high vapor qualities is not strictly analogous to the more realistic heat flux levels found in the desired applications of this work (heat pumps).

In order to obtain data over the full quality range at the required heat and mass fluxes and without using a tube length greater than 5 m for practical reasons, a second experimental rig was constructed. It is displayed in Fig. 2. The principal change is to employ two distinct heating sections over the same length of tube, the first serving as a preheater to provide partially evaporated fluid to the new shorter test section.

By using a large preheat flux, qualities available in the test section could be made greater than with Rig 1. In the test section itself, lower heat fluxes were used to further vaporize the fluid. The test section itself was 0.6 m in length, so that quality changes across the test section were relatively small. The preheat section was also instrumented to determine heat transfer coefficients at the preheat flux.

Wall temperature measurement stations were located at axial positions shown in Figs. 1(b) and 2(b). At each station, measurements were made at 90° intervals around the outer tube circumference. The temperature measurements were made with 0.25 mm diameter thermocouples which had been silver soldered and flattened. The junction and wires were isolated electrically from the tube by a very thin layer of teflon tape (<0.01 mm). Good thermal contact was maintained by clamping the thermocouple to the tube and running the leads along a substantial tube length. Inside wall temperatures were calculated from the measured outside temperatures by use of the steady-state radial, conduction equation with uniform heat generation with a small calibrated correction for heat gain from the room.

For pure refrigerants the *fluid* temperature used in the defining relation for $\alpha(=\dot{q}/(T_W-T_F))$ was the saturation temperature at each thermocouple station given the local estimated pressure. Although the pressure drop was small, an attempt was made to account for its non-linear nature [6]. For *mixtures*, T_f was equated to T_{eqb}, the thermodynamic equilibrium temperature. The equilibrium temperature was calculated from an equation of state developed specifically for this mixture [7]. At each thermocouple station, pressure, enthalpy, and original feed compositions were known, and the equation of state allowed calculation of all relevant parameters.

Table 1 lists the range of tests which were performed with each testing.

The selected mixture compositions span those recommended by the refrigerant supplier for use in heat pumps. The mass fluxes and heat fluxes were typical of those which might be employed in heat pumps. Rather than controlling pressure, outlet temperatures were controlled with Rig 1 to be those typical in the desired application.

In all Rig 2 tests displayed in Table 2, outlet pressure

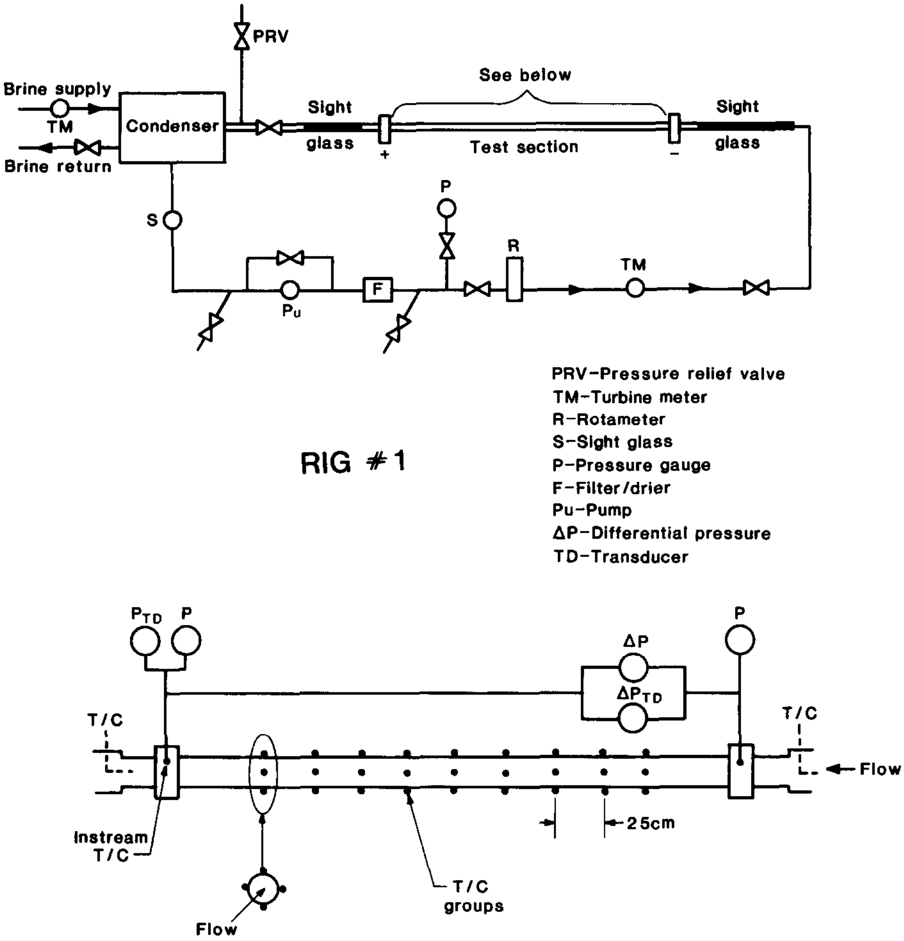

FIG. 1. Experimental test Rig 1. Test section length, 2.7 m; tube diameter, 0.9 cm.

was controlled at a fixed level typical of that which might be found with mixtures in a heat pump. By fixing the pressure, a conventional comparison of heat transfer coefficients was possible. The test section heat fluxes were set at the range of those which might be found in the latter portions of a crossflow heat exchanger in a heat pump. The preheat heat flux values ranged from 10 to 95 kW m^{-2} though film boiling conditions were generally avoided. Measurements of wall temperatures were made in the preheat section, so that heat transfer coefficients could be calculated at these heat fluxes as well.

3. DATA VALIDITY

In order to verify temperature measurements, several single phase liquid heating tests were made. The measured values compared within ±10% to the well-known equations†

$$\alpha_L = 0.023\frac{\lambda_L}{D}(Re_L)0.8(Pr_L)0.4 \tag{1}$$

as well as the more accurate equation [8]

$$\alpha_L = \frac{(f/8)Re_L\,Pr_L}{k_1+k_2\,Pr_L(f/8)^{1/2}(Pr_L)^{2/3}-1} \tag{2}$$

with

$$f = (1.85\log_{10}Re_L - 1.64)^{-2}$$
$$k_1 = 1+3.4f$$
$$k_2 = 11.7+1.8Pr_L^{1/3}.$$

In the process of these tests, the value for C_{pL} of R152a was shown to be in error by 15% [9]. Independent work in ref. [10] and a comparison to HTFS data [11] later confirmed this finding.

The effect of the preheater on the test section data was also examined. The last data validity check was the reproducibility of two-phase flow results, which is rarely discussed in the literature. In the case of refrigerants, only two literature values could be found: ±4% [12] and ±10% [13].

Some Rig 1 tests were repeated at various points in the day and sometimes from day to day. Agreement

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† This equation has been credited variously in the literature to Dittus and Boelter, McAdams, Colburn, or Kraubold (F.R.G.) The authors are not sure where credit belongs but will refer to it in this paper as the Dittus–Boelter equation.

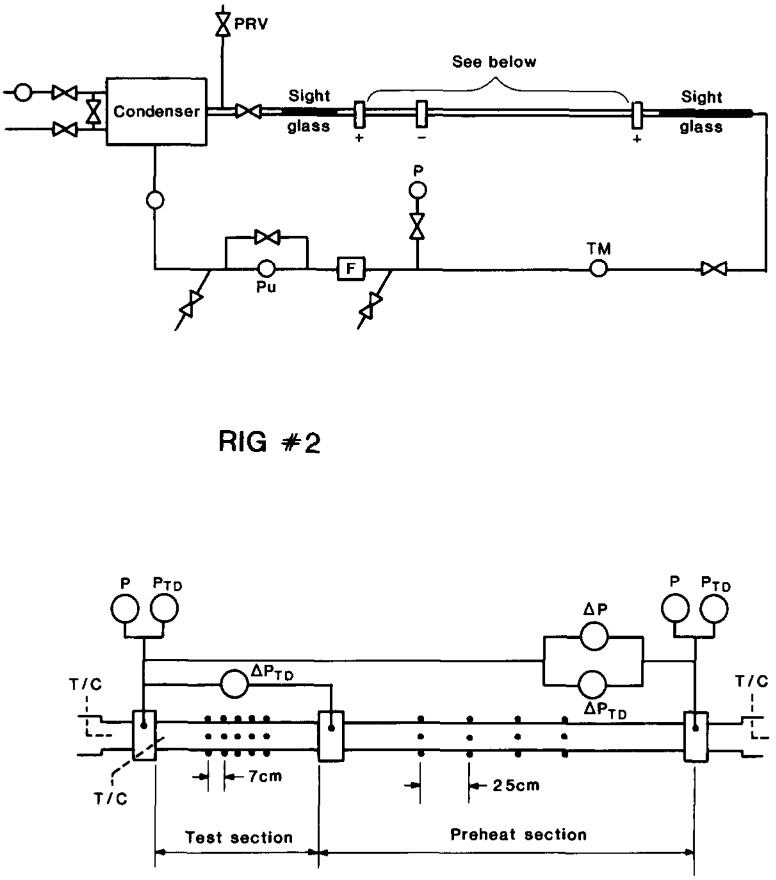

FIG. 2. Experimental test Rig 2. Test section length, 0.6 m; preheat section length, 2.1 m; tube diameter, 0.9 cm.

Table 1. Rig 1 test conditions

Composition	$G = \dot{m}/A_c$ (kg m^{-2} s^{-1})	$\dot{q}/A_s$ (kg m^{-2})	P (bar)
R152a	200–700	10–40	1.2–2.4†
R13B1	400–1200	20–40	5.7–7
0.833 wt % R13B1	200–600	30–40	5.7–8
0.750 wt % R13B1	200–600	30–40	5.7–6.6
0.706 wt % R13B1	200–550	30–40	4–7
0.662 wt % R13B1	200–550	30–40	4–7
0.454 wt % R13B1	150–300	20–40	5–7

† Isolated tests at higher pressure (~5 bar).

of Rig 1 heat transfer coefficients was within ±2% for tests done one day apart and ±5% for tests done one week apart. Eighteen tests with pure R152a were repeated approximately three weeks apart for Rig 2. Measured *preheat* values were compared. In the original tests, test section heat fluxes were 10 kW m^{-2}; in the later tests, the test section was set at 20 kW m^{-2}. Of the 54 data points, all agreed within 8%; most agreed much closer (about 4–5%). Most of this variation with Rig 2 was due to differences in mass flux. A few checks were made with mixtures, with similar findings.

In summary, a total of 1459 data points were collected with R152a/R13B1 in the following proportions: R152a: 409, R13B1: 170, Mixtures: 880. The data are tabulated with relevant variables in ref. [6]. Approximately 20% of the data is not in the annular flow regime and is not included in the further analysis.

The range of relevant variables for the reported test results includes: a heat flux of 10–95 kW m^{-2}, a mass flux of ~150–1200 kg m^{-2} s^{-1}, a composition of 0–1.0, and several intermediate values, a pressure of 1.7–8.0 bar, a quality of 0–1.0, a Martinelli parameter ($1/X_{tt}$) of 0.3–35, a Reynolds number of 3000–50,000, and a Prandtl number of ~3–4. All parameters, except Prandtl number, varied by nearly an order of magnitude. The data base then provided a strong basis for analysis of physical phenomena as well as the heat transfer coefficient.

To assist an understanding of mixture effects, the data was segregated into heat transfer regimes where first nucleate boiling dominated and then the evaporative process dominated. This was accomplished by examination of the dependence of the heat transfer coefficient on quality, heat and mass flux. A detailed examination of the pure refrigerant behavior was also made to provide a strong basis to analyze mixtures, behavior and existing mixture models.

Table 2. Rig 2 test conditions

Composition	G (kg m^{-2} s^{-1})	$\dot{q}_{ph}/A_c$ (kW m^{-2})	$\dot{q}_{ts}/A_c$ (kW m^{-2})	P_{ts} (bar)
R152a	100–300	10–95	10–20	4.70–4.80†
R13B1	200–500	10–50	10	4.70–4.80
0.80 wt R13B1	200–500	10–70	10	4.70–4.80
0.58 wt R13B1	200–500	10–70	20–30	4.70–4.80†
0.37 wt R13B1	100–400	10–80	10	4.70–4.80
0.18 wt R13B1	100–300	10–90	10	4.70–4.80

† Isolated tests at other pressures.

4. RESULTS: EVAPORATION DOMINATED HEAT TRANSFER REGIME

The situation where α_e is much greater than α_{nb} is characterized by a weak to negligible dependence on heat flux and a strong dependence on mass flux and local quality. There are several issues for both pure and mixed refrigerants which were addressed in the literature.

(a) *Suppression of nucleate boiling.* It has been suggested that fully suppressed nucleate boiling (FSNB) conditions are not achievable under most conditions with pure and mixed refrigerants due to their relatively low thermal conductivity [14]. The prediction of the FSNB point has never been adequately validated [15, 16]. It is not clear if and when for mixtures mass diffusion affects the FSNB point, and to what extent.

(b) *Heat transfer coefficient correlations.* The Chen correlation for pure fluids has been used with considerable success for water/steam data [17]. However, its predictive ability with refrigerants has been poor [18]. A Prandtl number correction has been recommended to improve its agreement with fluids other than water [19]; this correction has not been sufficiently verified [2].

(c) *Mass diffusion effect.* Analytic modelling of turbulent flow boiling of mixtures suggests mass diffusion effects are negligible under FSNB conditions [20]. However, some mixtures data were well predicted by including a diffusion correction to Chen's method [19].

Each of these issues will now be addressed.

5. SUPPRESSION OF NUCLEATE BOILING

The first issue to be addressed is whether FSNB conditions can reasonably be achieved with pure refrigerants. Indeed a review of the literature revealed that the pure refrigerant heat transfer data commonly showed a dependence on heat flux even at large qualities. Data taken at 4.75 bar and above as part of this investigation confirm this trend (Fig. 3). The effect of increasing mass flux and quality was not adequate to dissipate the dependence on heat flux. However, by lowering pressure to 1.5–2 bar, a different dependence on quality can be seen (Fig. 4). At low pressure a strong dependence on x is observed, indicating evaporation to be dominant. While this dependence is necessary it is not sufficient to show FSNB conditions. However, data taken at four levels of heat flux, as displayed on Fig. 5, show that the measured data merge into a single line, proving FSNB conditions to be achieved. Further data taken at larger mass fluxes yielded the same results, except that all data merged into a single line at lower quality. Thus FSNB can be achieved with pure refrigerants if pressure is lowered sufficiently.

It will now be shown that the quality at which FSNB conditions occur can be predicted from classical theory.

A comparison of the data with Hsu and Graham's model [21] for predicting the onset of nucleate boiling was made. The model requires a viscous sublayer thickness. Collier *et al.* [22] recommended using a dimensionless viscous sublayer thickness $y^+ = 7$. More recently, Bejan has analytically studied buckling and rolling of liquid layers in shear-driven flow (as occurs in annular flow boiling). He determined the value of $y^+ = 7.62$ as the viscous sublayer thickness which persists regardless of possible buckling or rolling [23].

The Hsu/Collier and Pulling suppression criterion was applied to the two pure refrigerants used in this investigation. The pressure gradient, needed to determine the wall shear stress and subsequently the thickness of the viscous sublayer, was estimated using the Martinelli–Nelson/Chisholm correlation [24, 25]. Contact angle was assumed to be 35°, typical for refrigerants on metal.

Measured mass fluxes and pressures were used. Assuming all cavities to be available, the criterion suggests a very small superheat requirement. Instead, a heat flux needed to activate a critical cavity size of 1.0 μm was calculated. Cavities of size greater than 1.0 μm were assumed to exist either in an insufficient number to affect the heat transfer, or to be fully wetted by refrigerant. This assumption then modifies the basic suppression criterion, so that a greater wall superheat is needed to initiate boiling than if all sizes were available in large numbers and were unwetted.

Three factors complicate the analysis, leading to substantial uncertainty. First, the estimate of the critical viscous sublayer thickness is not exact. It depends both on the determination of pressure drop and on a

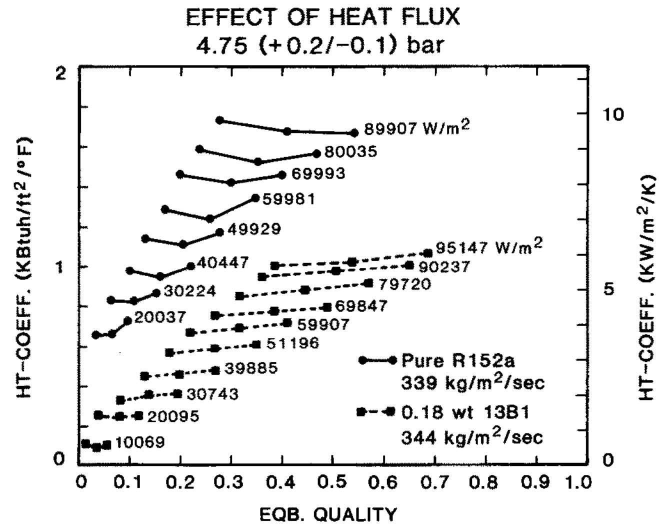

FIG. 3. Effect of heat flux on heat transfer coefficient for both pure fluids and mixtures under similar test conditions.

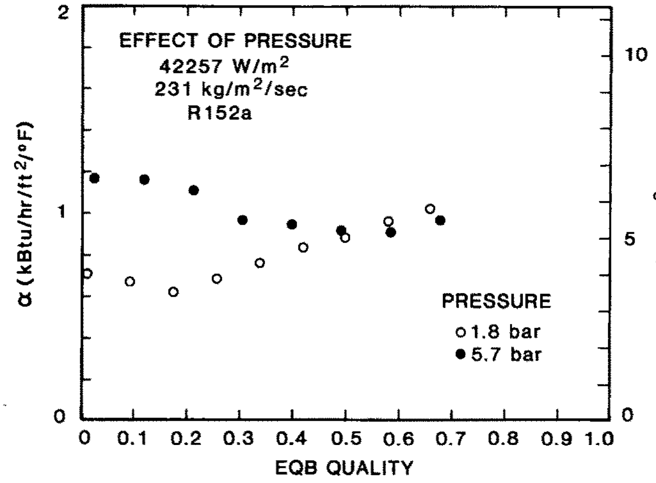

FIG. 4. Effect of pressure on measured heat transfer coefficient or pure R152a. At low pressure, evaporative contribution dominates.

selected critical value (Collier *et al.*'s $y^+ = 7$). Secondly, the selection of a critical cavity size may be in error. Collier has suggested 0.5 μm as a rough guideline for refrigerants [16]. This leads to a larger estimate of the suppression heat flux. Thirdly, vaporization may take place nearer the bubble base even in the absence of vaporization at the bubble cap. The required heat flux in this case would be less than as calculated. These complications lead to an estimated uncertainty of about ±40%. Despite the large uncertainty, the criterion was used with success.

Shown on Fig. 5 is the prediction of the suppression criterion for the heat flux level required to initiate boiling as a function of quality and the given flow rate. The criterion predicted fairly well the quality at which complete suppression occurs here for several mass fluxes.

The last FSNB issue is whether the criterion needs to be modified for use with mixtures. On one hand, Hsu and Graham's criterion is derived from a static force balance, so that no mass diffusion effect needs to be included. On the other hand, it is well established that bubble growth rates are reduced for mixtures as compared to an equivalent pure fluid. The criterion is attempting to predict bubble growth, so one might wish to include a mixture effect. Secondly, since active site sizes of refrigerants are very small, the inference of suppression via heat transfer data may in reality be reflective of boiling site density, rather than the growth of a bubble from a first single active site. It is known that mixture composition affects boiling site density [5]. Thus for the problem at hand, it is likely that some mixture effect needs to be accounted.

Ideally, data at sufficiently low pressures would have been taken for mixtures in this investigation. However, the experimental apparatus was incapable of lowering pressure for the mixture to the same levels as shown on Fig. 5. Instead, correlative evidence will

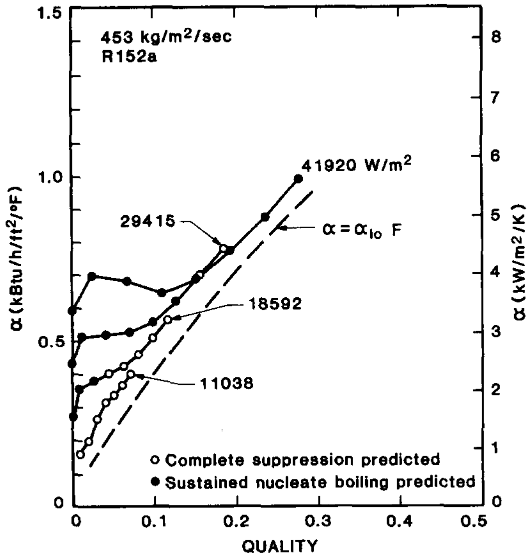

FIG. 5. Comparison of Chen's method (evaporative term only) with measured data. Shown also is the result of the method for predicting complete suppression of nucleate boiling.

be offered later in this paper, suggesting that mixtures affect the FSNB point.

6. EVAPORATION DOMINATED CORRELATIONS FOR PURE FLUIDS

As discussed earlier, there is a discrepancy in the literature regarding the predictive ability of Chen's correlation for non-aqueous pure fluids. While widely advocated for nuclear industry applications, its prediction of refrigerant heat transfer coefficients has been unacceptable. In general the refrigerant data is at qualities above $x = 0.2$. In Chen's original correlation development, the maximum quality for organic fluids was $x = 0.12$. This difference is, however, not the source of the discrepancy. Bennett and Chen [19] recently suggested that for fluids with Prandtl number greater than 1, a correction was needed whenever FSNB conditions were not achieved

$$\alpha = \alpha_{\mathrm{LO}} F Pr_{\mathrm{L}}^{0.296} + \alpha_{\mathrm{pool}} S. \tag{3}$$

Under conditions of FSNB, the Prandtl number correction is deleted. Polley [36] and Aounallah *et al.* [37] suggested that under such conditions, Chen's method predicted water/steam data best when $\alpha = \alpha_{\mathrm{LO}} F$, i.e. the suppression factor should be equal to zero. As discussed above, data were obtained which allowed comparison to refrigerants under FSNB conditions. As shown on Fig. 5, Chen's method without the nucleate boiling term and without the Prandtl correction predicted the data well.

Perhaps more significantly, the Prandtl number correction can also be validated. Figures 6 and 7 show R22 data ($Pr = 2.5$) from the literature [38] and R152a and R13B1 data ($Pr = 4$ and 3, respectively) taken in this investigation as compared with

$$\alpha = \alpha_{\mathrm{LO}} F Pr_{\mathrm{L}}^{0.296}. \tag{4}$$

While the Prandtl correction in equation (4) predicted the data very well, equation (3) overpredicted the data. Polley has suggested, for water/steam data that the suppression factor, S, be set equal to zero whenever the evaporative contribution is greater than the nucleate boiling contribution. The results here verify that this suggestion is correct for non-aqueous fluids as well, whether or not nucleation occurs. In fact, Bennett and Chen implicitly did the same in comparing their method to data of others [19].

7. MASS DIFFUSION EFFECT/CORRELATIONS FOR MIXTURES IN FSNB HEAT TRANSFER

An analytic treatment suggested that mass transfer resistance and sensible heating are negligible in turbulent flow evaporation (FSNB conditions) [26]. These conclusions are generally supported by multicomponent condensation data. The effect of these results are that non-mixture methods are adequate for predicting the heat transfer, specifically that Chen's original formulation for pure fluids is applicable for mixtures. If mass transfer resistance is included, then the analysis showed that it is dominant on the vapor side.

In sharp contrast, Bennett and Chen advanced a mixtures' model which includes both mass transfer resistance and sensible heat only in the liquid film; they correlated a large data set with their model. However, their correlation is semi-empirical in that the mass transfer coefficient was derived from the data set.

Neither treatment has been verified independently. The mixtures' data taken in this investigation at low heat flux but high quality is shown on Fig. 7. Shown also is the F function in the Bennett and Chen *pure fluid* method with and without the Prandtl correction. The following observations may be made:

(a) The pure fluid data is correlated well by the F function with Prandtl correction, suggesting some nucleation to be present.

(b) The mixtures' data is well below the pure fluid, indicating a degraded heat transfer.

(c) The mixtures' data is correlated fairly well by the F function without Prandtl correction, particularly at high quality (large $1/X_{\mathrm{tt}}$).

The tentative conclusion from these observations is that while the heat transfer is degraded with mixtures, the degradation is not due to mass transfer resistance at the annular interface. Instead, the degradation seems due to the effective suppression of nucleate boiling in the case of mixtures.

8. NUCLEATE BOILING DOMINATED SITUATIONS

We now move to the second heat transfer regime, where the data is characterized by a strong depen-

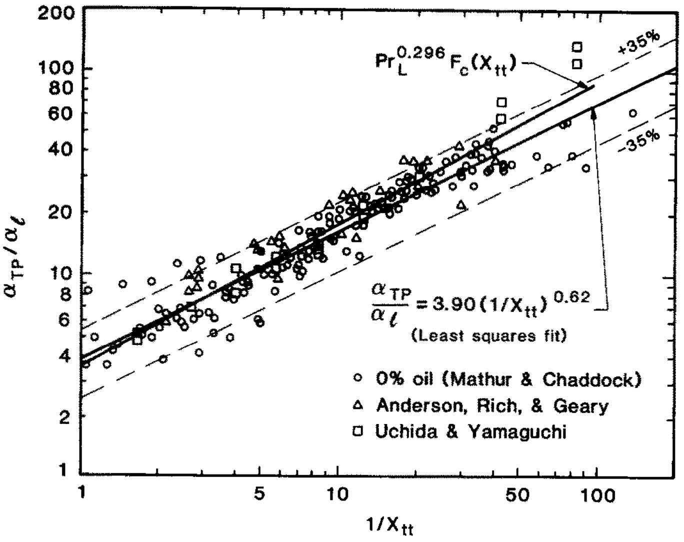

FIG. 6. Comparison of measured R22 data found in ref. [38] to that predicted by Chen's method evaporative term only with Prandtl number correction.

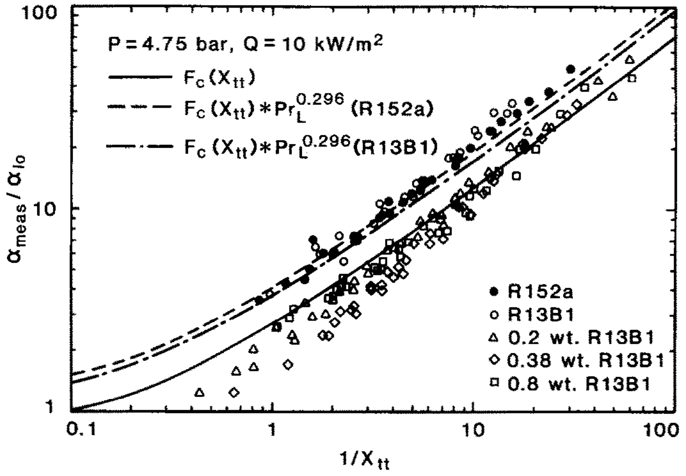

FIG. 7. Comparison of Chen's method to pure and mixed refrigerants. Pure fluids require Prandtl number correction, suggesting sustained nucleate boiling. Mixtures are fit fairly well by the same correlation without Prandtl correction, suggesting suppression of nucleate boiling.

dence on heat flux and weak effects of mass flux and quality. The issues here are less basic but of no less practical significance.

Experiments with pool boiling of mixtures have shown severe degradation in heat transfer coefficient when compared to ideal or equivalent pure fluid values. The degradation has been accounted by various correction factors. Since pool boiling methods have been extrapolated successfully in flow boiling of pure fluids, the same extrapolation has been hypothesized as valid for mixtures. Little data exists to confirm the hypothesis.

A second issue is around pressure drop prediction. Since it is well established that heat transfer is degraded in nucleate boiling dominated situations, some effect might be conjecture as to the effect of mixtures on ΔP. Little data exists to determine the effect, if any.

9. EXTENSION OF POOL BOILING METHODS TO FLOW BOILING

The first subissue to be addressed in this area is the ability of correlations to precict pure fluid behavior. This ability was analyzed by comparison to a data base consisting of about 200 points, with heat fluxes ranging from 10 to 95 kW m^{-2} and mass fluxes of 100–500 kg m^{-2} s^{-1}.

The correlation of Stephan and Abdelsalam [29] predicted both refrigerants exceptionally well with the data within $\pm 20\%$ and most predictions within 5–10%

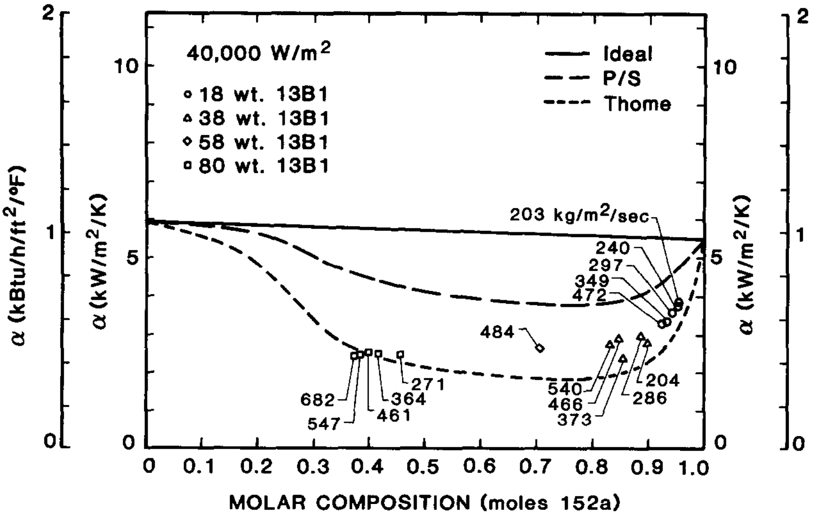

Fig. 8. Comparison of prediction methods to measured data. Data is from first measuring station in preheat section of Rig 2. P/S: Palen and Small [28]. Compositions of interest to heat pump application are between 0.35 and 0.5 mol composition of R152a.

$$\frac{\alpha d}{\lambda_L} = C_{SA} q^{0.745} \tag{5}$$

where C_{SA} is a constant determined from fluid properties at the saturation temperature.

Since equation (5) was developed from a large refrigerant data base, the results are consistent. Refrigerant R152a was not considered by Stephan and Abdelsalam so our experimental data extends the verification of the predictive ability of their method.

The effects of mass flux and quality are not accounted by equation (5). An attempt was made by Chawla to account for mass flux and quality [30, 31]. It however overestimated their effect in most cases.

Having in hand a correlation which predicts both pure refrigerants well, the effect of mixtures can be examined. Figure 3 shows some of the mixtures' data in a series of parametric tests with pressure, heat and mass flux levels similar to that of the pure fluids. A severe degradation (50%) as compared to pure fluids is seen. This data set showed a negligible dependence on mass flux and quality and thus represents well the domination of nucleate boiling. In all flow boiling tests, there is an uncertainty in the actual local liquid composition, given that a mass transfer resistance and a non-equilibrium condition are part of the physical process. These complications can be mitigated by examining only that experimental data closest to equilibrium but still in annular flow. At the first thermocouple group where $x > 0.05$, the vapor quality is relatively low, so that the liquid composition will still be close to the known subcooled inlet composition. At this point also, the evaporative mode should contribute only weakly, so that mass flux and vapor quality levels should not be significant. Figure 8 displays this data at one particular heat flux, along with the prediction methods.

Of the pool boiling of mixture relations, the simple method of Thome predicts the measured data fairly well [32]

$$\alpha = \alpha_{ID}\Delta T_{ID}/(\Delta T_{ID} + \Delta T_{DEW\text{-}BUB}). \tag{6}$$

Thome's method is both more conservative and more accurate in the mixture concentrations at which heat pumps are recommended for operation. It predicts the observed level of degradation at points away from the maximum degradation better than other mixture methods [6].

10. PRESSURE DROP FOR MIXTURES AND PURE FLUIDS

It has been suggested recently that conventional pressure drop prediction methods must be modified to account for mixture effects [33]. Others claim that there is no physical reason for requiring such a change [34]. Arguments might be made on either side. All prediction methods use the quantity x_{eqb} in determining pressure drop. For pure fluids at equilibrium qualities above zero, the vapor quality differences, $x_{act} + x_{eqb}$, is usually small in the annular flow regime. It is well established in nucleate boiling of non-azeotropic mixtures, mass transfer resistance leads to degraded heat transfer.

This phenomenon could change the vapor generation rate, such that the vapor quality difference, $x_{act} - x_{eqb}$, is significantly dissimilar between pure fluids and mixtures. The dissimilarity could lead to poor prediction using pure fluid methods.

The Martinelli–Nelson method [24] as modified by Chisholm [25] was selected for prediction due to its simple application and success with many different fluids. The Martinelli–Nelson method requires stepwise integration which has been well described elsewhere [33]. For the tests conducted here, the inlet

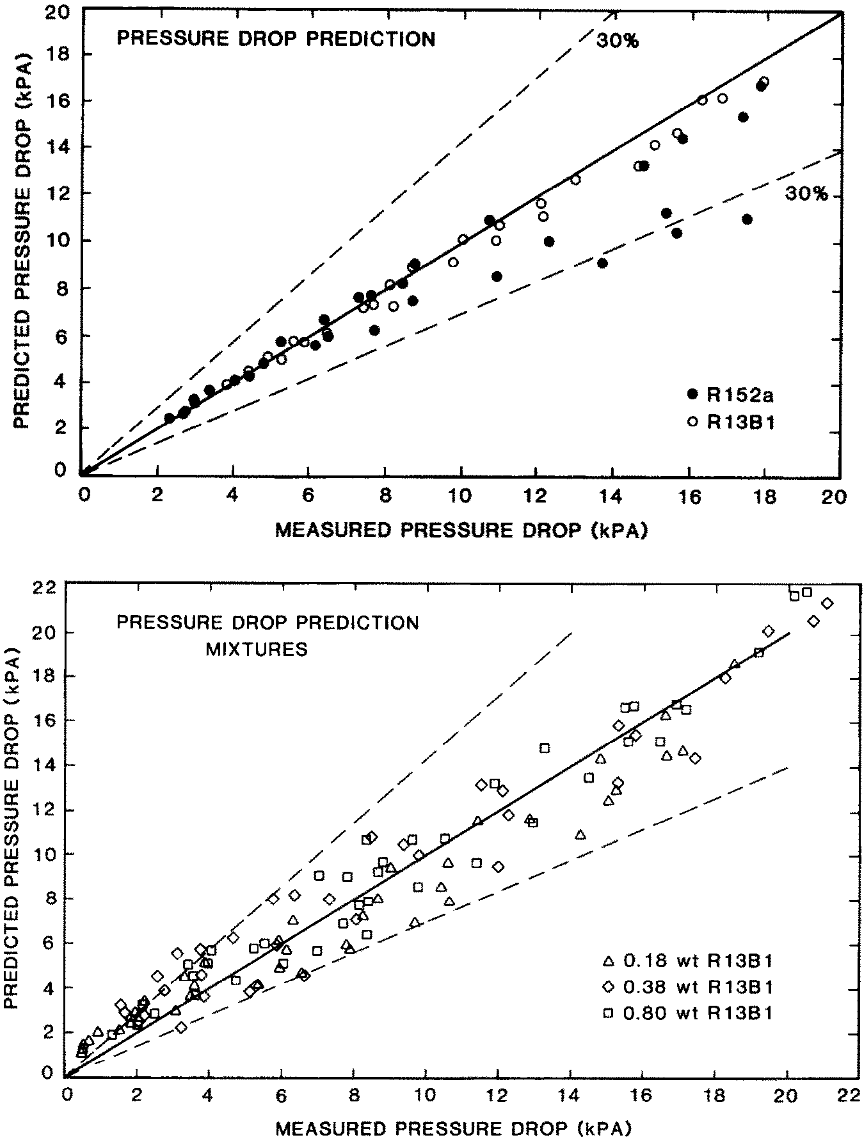

FIG. 9. Comparison of measured pressure drop to that predicted by the Martinelli–Nelson–Chisholm method.

conditions were only slightly subcooled, and the single phase pressure drop was neglected. The entire pressure drop was assumed to occur between the calculated position of $x_{\mathrm{eqb}} = 0$ and the outlet.

Figure 9(a) displays the comparison between measurement and prediction for the pure fluids in the test section and the entire tube length. Agreement is particularly good for pure R13B1, and within the stated accuracy of the method for pure R152a. The underprediction for R152a at large pressure drops is due partly to the neglect of the single phase region.

Figure 9(b) displays a similar comparison for the mixtures. Thermodynamic properties for the mixtures were calculated from the EOS. Transport properties were calculated using appropriate mixing rules from ref. [35]. While some increased scatter occurs, the method predicts within its intended accuracy. No correction for composition is required for this refrigerant pair.

11. A FURTHER EXAMINATION OF THE DATA: CIRCUMFERENTIAL VARIATION IN HEAT TRANSFER IN HORIZONTAL FLOW BOILING

In horizontal annular flow boiling the liquid film thickness is asymmetric around the inner circumference of the tube. Due to gravity, the film at the tube bottom is thicker than at the tube top. The effect of this thick film for pure fluids is to produce a lower heat transfer coefficient at the tube bottom due to increased resistance. This behavior has been observed widely in the literature and was seen with both rigs.

In the case of horizontal flow boiling of mixtures, a new opposite phenomenon was observed, i.e. the heat

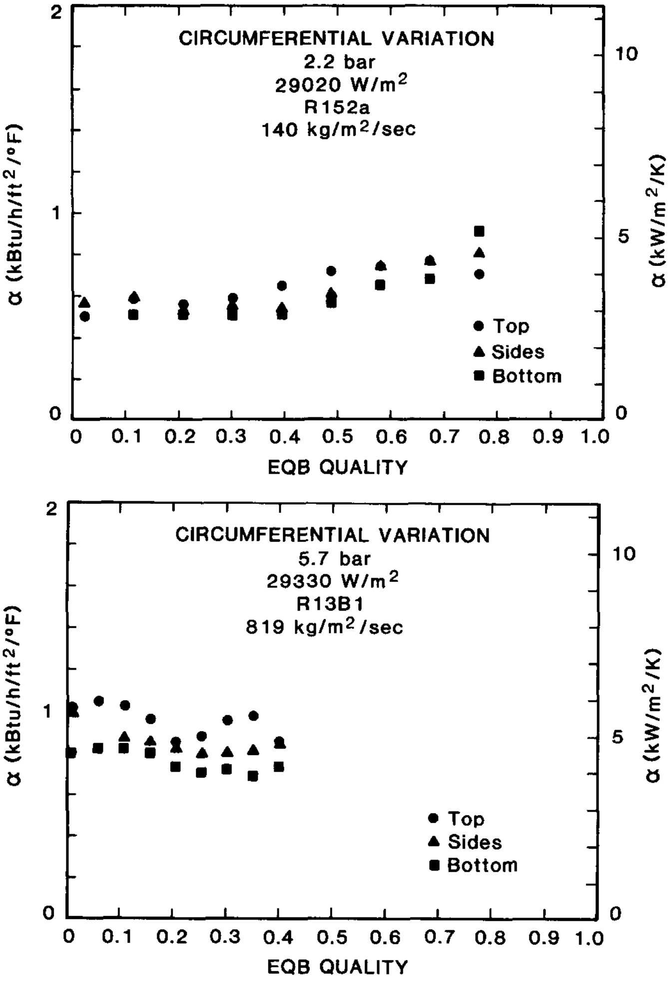

FIG. 10. Circumferential variation in heat transfer coefficients for pure refrigerants.

transfer coefficient at the tube bottom was higher than at the top by an amount far outside experimental uncertainty. The observation was made consistently with both rigs; the rigs utilized different tubes, thermocouples, and test section heaters, and thus should not be a function of this apparatus. The wall temperature measurements determine the circumferential variation and are not dependent on the data analysis technique. The effect was observed with and without a preheat section, with a variety of heat fluxes, flow rates, and mixture compositions, and over a range of pressures (left uncontrolled in the tests with the first rig).

Figure 10 displays a typical data set (Rig 1) for the pure fluids. The heat transfer coefficient at the bottom is seen to be lower than at the tube top, except near $x = 0.8$, where dryout is beginning. In contrast, Fig. 11 displays similar plots for mixtures. The reversal is noted at qualities greater than 0.2. Identical trends were observed with Rig 2 in both the preheat and the test sections.

One might question why this finding has not been discussed before in the literature. Three of the previous studies [14, 19, 26] involved vertical flow with only one thermocouple mounted around the tube circumference. Both the lack of instrumentation and the flow orientation preclude the possibility of observation.

Conjecture can be made regarding the reason for the reversal. Possibilities include:

(a) increased turbulence at the bottom of the tube;

(b) nucleate boiling at the bottom of the tube, and not at the top;

(c) flow pattern differences between mixtures and pure fluids;

(d) departure from nucleate boiling (DNB) events;

(e) different concentrations at the bottom of the tube than at the top.

The first two reasons may be dismissed in that they should appear for the pure fluids as well. Further, in the case of mixtures, nucleate boiling is more difficult to sustain, so it is unlikely that this is the cause. If any trend should appear in the nucleate boiling dominated region, it is that a greater disparity between top and bottom might occur with mixtures, not a reversal. The third reason may be dismissed in that flow patterns were observed at the outlet.

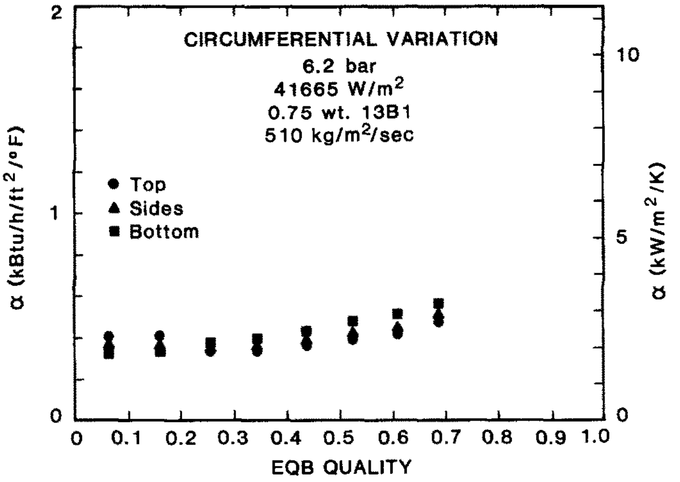

FIG. 11. Circumferential variation in heat transfer coefficients to mixed refrigerants.

The possibility of DNB events might be considered since it would occur first at the tube top. There are several reasons why this cause is unlikely: (a) the top wall temperatures were neither erratic nor wildly different from the bottom wall readings; (b) film boiling tends to be mitigated by the addition of a second less volatile component; (c) the behavior persisted at various qualities without showing a larger difference in heat transfer coefficient. Still, since a DNB event would cause a temperature difference in this same direction, such a possibility cannot be dismissed.

The last possibility is believed to be the root cause of the reversal: the higher heat transfer at the tube bottom is due to the relatively greater amount of the more volatile component (R13B1) at the bottom of the tube. The gravity-driven drainage of bulk liquid to the tube bottom provides a larger amount of fluid upon which to draw. Initially vaporization occurs at both tube top and bottom into the annular vapor core. The top portion of the tube, with its thinner film (and initially higher heat transfer rate) vaporizes most of the more volatile component. At some point it becomes starved of this component and the vaporization rate is diminished. In the bottom portion of the tube, vaporization is initially small then increases relative to the tube top; since it has a greater bulk amount of fluid, it becomes depleted of the more volatile component at a slower rate than the tube top.

There exists then a competition between the depletion of the more volatile component and film thickness. If this explanation is correct, one might expect to see in the early portions of the vaporization process α_{top} greater than α_{bot}, as both tube top and bottom have similar compositions. Then as the top becomes depleted of the more volatile component, the two values should merge and then reverse. Figure 11 verifies this conjecture.

An attempt was made to correlate the top-to-bottom wall temperature difference with local vapor–liquid composition difference. The latter quantity is correlated with local mass transfer resistance. No clear correlation was seen in the test section data.

12. SUMMARY OF FINDINGS

The research involved determining experimental heat transfer coefficients for both pure and mixed refrigerants. Over 1000 data points were collected, covering a wide range of pressure, composition, quality, heat and mass flux.

It was seen that with pure refrigerants complete suppression occurs only at rather low pressures. A suppression criterion was applied to pure refrigerants at a selected cavity size, and shown to predict quantitatively the quality at which suppression occurs for a given pressure, heat and mass flux. The method was extended to mixed refrigerants via several hypotheses; inclusion of a mass transfer resistance term suggests that suppression would be easier to achieve with mixtures than pure fluids. Correlative evidence suggested such a trend.

In the evaporation-dominated heat transfer regime, Chen's correlation was successfully applied to pure refrigerants with and without the occurrence of FSNB conditions. The Prandtl number correction suggested by ref. [19] is needed when some nucleation occurs. For mixtures, mass diffusion may not complicate the problem substantially under FSNB conditions, and the same correlation may be used with success.

In the nucleate boiling dominated regime, the method of ref. [29] was validated for pure fluids, and used successfully with ref. [32] for mixtures. Pressure drop correlations for pure fluids were also extended to mixtures without modification.

In the case of mixtures, previously unrecognized physical phenomena were noted: the circumferential variation in heat transfer coefficient may be opposite for mixtures than observed for pure fluids. This observation suggests the existence of a circumferential gradient in concentration and interfacial temperature. Modelling then of mixture heat transfer is further complicated in that gradients exist in axial, radial and circumferential directions. In the flow boiling of mixtures when the onset of nucleate boiling is more difficult to predict and when mass transfer resistance occurs, the actual quality may lag the 'equilibrium' quality in a different way than for pure fluids. This problem poses another difficulty for the correlation/models suggested to date.

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EBULLITION DE REFRIGERANTS PURS ET MELANGES EN ECOULEMENT HORIZONTAL

Résumé—La recherche faite est la détermination expérimentale des coefficients de transfert thermique pour R152 et R13B1 purs et pour quatre mélanges de ces corps. Les mélanges fournissent des coefficients de transfert nettement plus faibles que pour chacun des réfrigérants purs. L'observation suggère que la suppression complète de l'ébullition nucléée est plus facile à obtenir pour les mélanges que pour les fluides purs. Des formules existantes sont comparées aux résultats expérimentaux avec succès aussi bien pour les fluides purs que pour les mélanges.

STRÖMUNGSSIEDEN REINER KÄLTEMITTEL UND DEREN GEMISCHE BEI HORIZONTALER STRÖMUNG

Zusammenfassung—Die Untersuchung behandelt die experimentelle Bestimmung des Wärmeübergangskoeffizienten bei den reinen Kältemitteln R152a und R13B1 und bei vier Mischungen dieser Kältemittel. Die Mischungen zeigten deutlich geringere Wärmeübergangskoeffizienten als jedes der reinen Kältemittel. Die vollständige Unterdrückung des Blasensiedens ist mit Mischungen leichter zu erreichen als mit reinen Fluiden. Vorhandene Korrelationen wurden erfolgreich sowohl mit den Reinstoffen als auch mit den Mischungen verglichen.

ГОРИЗОНТАЛЬНОЕ ТЕЧЕНИЕ КИПЯЩИХ ЧИСТЫХ И СМЕШАННЫХ ХЛАДАГЕНТОВ

Аннотация—Экспериментально определены коэффициенты теплообмена для чистых охладителей R152 и R13B1 и четырех их смесей. Коэффициенты теплообмена для смесей существенно меньше коэффициентов теплообмена любого из чистых охладителей. Можно предположить, что достичь полного подавления пузырькового кипения значительно легче в смесях, чем в чистых жидкостях. Получено хорошее соответствие между известными зависимостями и настоящими данными как для чистых жидкостей, так и для смесей.