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EXPERIMENTAL DETERMINATION OF FORCED CONVECTION EVAPORATIVE HEAT TRANSFER COEFFICIENTS FOR NON-AZEOTROPIC REFRIGERANT MIXTURES

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ABSTRACT

A number of experiments were conducted in the 1960's to determine the two phase heat transfer coefficient, h_{TP}, of pure refrigerants. Recently energy conservation requirements spurred interest in non-azeotropic refrigerant mixtures, because such mixtures can improve theoretically the COP of certain refrigerant cycles. The two phase heat transfer coefficient of such mixtures under forced convection conditions is virtually unknown. An experimental rig has been constructed to investigate whether it is possible to predict the heat transfer coefficient of the mixture based on the coefficients of the components. Initially data was taken on R-22 and compared to literature data and to existing predictive correlations. Good agreement was found with the literature's data on forced convection, single phase heat transfer correlations, and some two phase evaporative correlations. Single phase heating and two phase boiling experiments were conducted then on R152a and R13B1 individually and in various mixture concentrations. The results of these tests are reported in the paper.

NOMENCLATURE

h	= heat transfer coefficient
Q	= heat flux per unit area
T	= temperature
p	= pressure
k	= thermal conductivity
F	= dimensionless coefficient in Chen's correlation
S	= dimensionless coefficient in Chen's correlation
Re	= Reynolds Number
Pr	= Prandtl Number
D	= diameter of tube
f	= friction factor
G	= mass flux
C_p	= specific heat
x	= quality
$\bar{y}$	= vapor composition
$\bar{x}$	= liquid composition
Bo	= boiling number
X_{tt}	= Martinelli parameter
H	= enthalpy
A	= tube surface area
ϕ	= Martinelli parameter related to X_{tt}

Subscripts

TP	= two phase
W	= tube wall
sat	= saturation
f	= fluid
g	= gas
top	= top of tube
ls	= left side of tube (viewed from upstream)
rs	= right side of tube
bot	= bottom of tube
inlet	= inlet of test section
L	= liquid
fg	= diference between vapor and liquid
L_o	= liquid only
FCV	= forced convection vaporization
NB	= nucleate boiling regime
2ϕ	= two phase (used in Reynolds number)

Greek Symbols

μ	= absolute viscosity
ρ	= density
ν	= kinematic viscosity
σ	= surface tension

INTRODUCTION

In order to overcome certain limitations in the vapor compression cycle which are due to the employment of single working fluid mixtures, the use of non-azeotropic refrigerant mixtures has been widely suggested [1]. Design of machinery systems employing these non-azeotropic mixtures is among others a strong function of the knowledge of mixture transport properties. Determination of these mixtures' properties has been generally limited to mole fraction weighting of the components' properties. However, for the heat transfer coefficient this type of estimation is questionable since a new and possibly more limiting phenomena has been introduced by the existence of the mixture, that of mass diffusion. At least one author has concluded

that prediction of the local heat transfer coefficient of a non-azeotropic mixture is a function of the rate at which the more volatile component can diffuse to the vapor bubble region [11]. In order to provide reliable data as to the significance of this phenomena for the temperature, pressure and composition ranges found in heat pumps, NBS has begun a study using R13B1 and R152a from 27/73 to 69/31 percent molar range, the results of which are reported in this paper. As will be seen in the description of the work it is necessary to know the local thermodynamic properties in order to determine the heat transfer coefficient from the data. Therefore, a separate simultaneous study (both theoretical and experimental) was conducted to determine an appropriate equation of state [15].

THE EXPERIMENT

In order to measure the evaporative heat transfer coefficient, a test facility was designed as shown in Fig. 1. The test section is a horizontally-marked

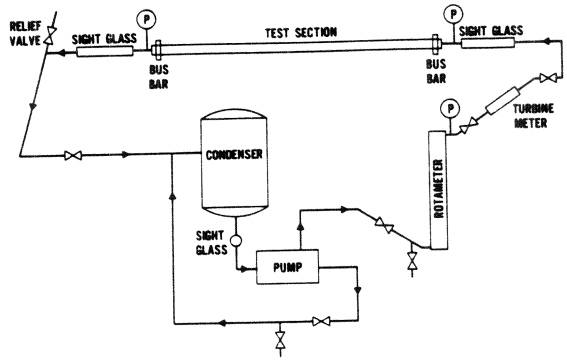


Figure 1: Experimental Test Facility

stainless steel tube with .25 mm wall thickness, a length of 2.70 m and an outer diameter of .95 cm. It is heated by applying a DC voltage difference along the tube using the tube itself as the heater. The liquid refrigerant is pumped out of the condenser through the test section. By adjusting the valves at the test section inlet and outlet the degree of subcooling could be set. Then the heated and partially evaporated refrigerant enters the condenser to be liquified and recirculated by the pump. The flow pattern is observed in the glass tubes which have the same inside diameter as the test section and which are installed at each end of the test section.

All tests were run with subcooled liquid entering the test section. Refrigerant temperatures were measured at the inlet and outlet by metal sheathed thermocouples which had an exposed junction located in the centerline of the flow. The refrigerant flow rate was determined by a calibrated turbine meter. The outlet pressure and the inlet-outlet pressure difference were also measured. The outside wall temperature was measured by 36 Copper-Constantan thermocouples which were clamped to the tube in 9 groups of thermocouples, each insulated electrically from the tube by a thin layer of teflon tape (< .01 mm). The thermocouples within each group were located at the top, sides, and the bottom of the tube. The junctions of the thermocouples were silver soldered, and run along the tube for more than 0.1 m. The thickness of the wires was .25 mm. The entire test section was heavily insulated with about 12 cm (radial) fiberglass.

For each run an energy balance was conducted over the entire loop and data were considered valid when the energy balance agreed within 7%.

In order to simulate conditions which occur in an evaporator for a residential heat pump, the measurements were performed in a temperature range between 20°C and 0°C. Because of the temperature gradient across the insulation of the test section, about 30W of heat from the room were picked up by the refrigerant while it passed through the test section, causing the wall thermocouples to read slightly higher than they would have under truly adiabatic conditions. This deviation was quantified by a calibration test run with a very high liquid refrigerant flow rate, without electric heat at the lowest temperature in the range of this study. The difference between each wall thermocouple and the centerline temperature was recorded (typically less than 0.5°C). The wall temperatures measured during an experimental test run, with the electric heat turned on, were then corrected accordingly, taking the previously determined background heat flux (i.e., between the tube wall and room) into account. The correction did not cause more than a 5% change in the measured wall temperature. Also, during the experiments, the radial temperature gradient within the wall was taken into account by using the appropriate equation for steady state heat conduction with simultaneous heat generation in a cylindrical wall; axial conduction was considered negligible. The temperature-dependence of the tube's electrical resistance was also determined to have a negligible effect on the heat transfer coefficient determination.

DATA ANALYSIS

The local heat transfer coefficient, h_{TP}, was determined by $h_{TP} = \dot{Q}/(T_w - T_f)$. $\dot{Q}$ is the heat flow per unit area and T_w and T_f are the inside wall temperature and the fluid temperature at the tube center line. At each thermocouple station, the coefficient value was averaged by $(h_{top} + h_{ls} + h_{rs} + h_{bot})/4$. The variables $\dot{Q}$ and T_w were measured directly; T_f was determined by the equation of state. For pure refrigerants T_f was assumed equal to the saturation temperature. To determine the saturation temperature, local pressure values were used. Because pressure measurements were made only at the inlet and outlet of the test section, local pressure values had to be calculated. Given T_{inlet} and P_{inlet} and knowing the rate of heat addition (and thus the enthalpy), the location of the onset of boiling could be determined. The small pressure drop measured over the entire test section was assumed to occur linearly between the start of boiling and the outlet of the test section.

For the mixtures, a more complicated procedure was required. The location of the onset of boiling was determined from the knowledge of inlet composition, pressure, temperature and the rate of heat addition by using the NBS equation of state. From this information, the bubble point temperature and the location of the start of boiling was also determined. Again a linear pressure drop was assumed between the onset of boiling location and the outlet. In this case, a linear temperature difference was also assumed which provided the needed local values of T_f. Given temperature and pressure, local values of composition and enthalpy, the quality was then calculated. It was determined that the slight nonlinearity that existed between calculated enthalpy increase and temperature increase during boiling for these mixtures could be neglected.

RESULTS

A series of tests were run with pure refrigerants and mixtures of R13B1 and R152a in various concentrations. In all of the evaporative tests, only the data at thermocouple locations with a quality greater than 10% were included in the analysis. This criterion was selected so that an annular flow pattern was assured and so that entrance effects would be eliminated (the criterion of 10% quality usually meant that the first 100 diameters of the test section were not considerd in the data reduction). Because of the rather thin liquid annular layer that exists under this flow condition it is not clear as to whether the phase change was actually boiling, evaporative or a combination. For the analysis of the data and in comparison to predictive correlations, thermodynamic properties were calculated from the NBS equation of state. Thermodynamic properties for pure refrigerants were taken from the ASHRAE Handbook of Fundamentals 1981 [13]; for calculation of transport properties of the mixtures, the method described by Kandlikar, Bijlani and Sukhatme was used [9]. The composition of the mixtures was determined by gas chromatograph sample analysis.

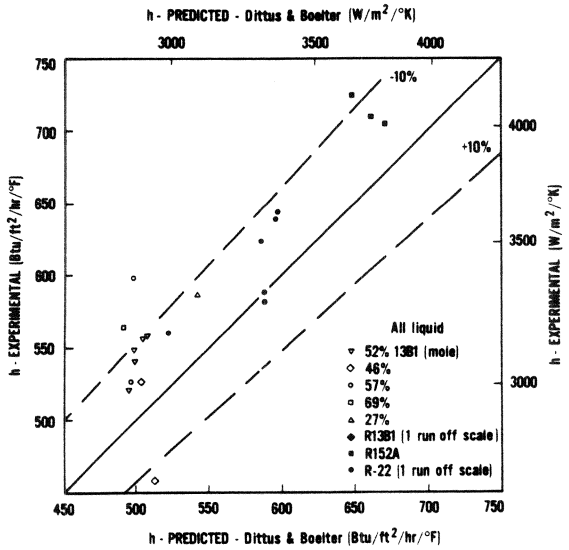
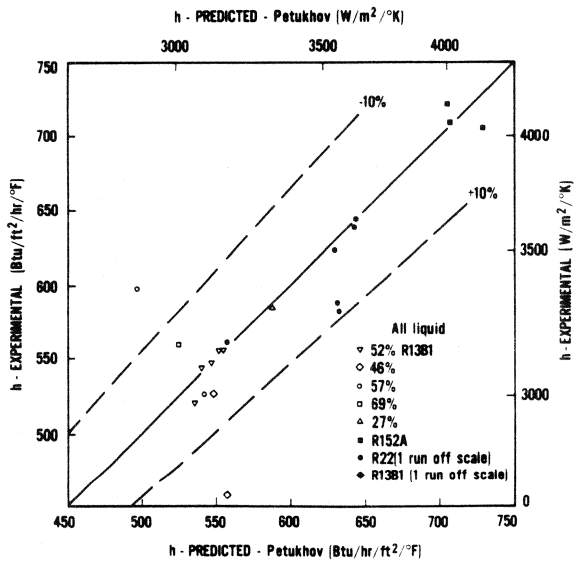


Figure 2a, 2b: Comparison of Correlations to Single Phase Runs



The first set of tests (24 runs) were single phase liquid runs, the results of which are plotted against two predictive correlations on Fig. 2a and b. For all but two runs, the experimental heat transfer coefficients, h_L, were found to be within the published accuracy of $\pm$ 13% [4] with the well-known Dittus-Boelter correlation:

$$h_L = 0.023 \frac{K_L}{D} Re_L^{0.8} Pr_L^{0.4} \quad (1)$$

$$\text{with } Re_L = \frac{GD}{\mu}, \; Pr_L = \frac{\mu_L C_{p_L}}{K_L} \quad (1a)$$

Seventy percent (70%) of the experimental results were found to be within the published accuracy of $\pm$ 6% of the Petukhov correlation [6]:

$$h_L = \frac{K_L}{D} \frac{(f/8) Re_L Pr_L}{K_1 + K_2 Pr_L (f/8)^{1/2} (Pr_L^{2/3} - 1)} \quad (2)$$

$$\text{with } f = (1.85 \log_{10} Re_L - 1.64)^{-2} \quad (2a)$$

$$K_1 = 1 + 3.4 f \quad (2b)$$

$$K_2 = 11.7 + 1.8 Pr_L^{1/3} \quad (2c)$$

All but two runs were within $\pm$ 10% of the Petukhov correlation. The prediction of heat transfer for the mixtures was as accurate as for pure refrigerants. In the course of these tests it was felt that the values of specific heat capacity (liquid phase) for R152a differ from those published by ASHRAE [13]. Using the published values resulted in a sharply increasing single phase heat transfer coefficient along the test section a multiplication of the published specific heat capacity values by 1.15 yielded the expected roughly constant heat transfer coefficient and a good energy balance throughout the refrigerant loop; these calculated values were employed throughout the experiments with R152a and the mixtures.

The second set of tests was with phase change (35 runs) involving pure refrigerants and a comparison of local heat transfer coefficients to five predictive correlations (Schrock and Grossman, Dengler and Addoms, Chen, Bennett, and Lavin and Young). The details of the comparison are described in Appendix A. As shown in Tables A1 to A3, the Chen and Lavin and Young correlations proved to be the best predictors of local heat transfer coefficients based on this set of experiments. Previous applications of the Chen correlation for horizontal refrigerant flow have not yielded satisfactory agreement [16, 2] though the correlation is widely recommended for vertical water flows [5, 7]. In these tests, a simple increase of 15-17% of the expression in the Chen correlation would have brought 80% of the data within -22/+18%.

The third set of tests was with phase change (33 runs) involving refrigerant mixtures of R13B1 and R152a in various concentrations. The local and average heat transfer coefficients were lower significantly than either pure refrigerant, as shown in Fig. 3. This result is contrary to the findings of Singel et al who find the average heat transfer coefficient for R-12/13 to be greater than pure R-12 (they did not report

measurements of pure R-13) [12]. Most likely the reduced heat transfer in our case is due to a mass diffusion limitation (and reduction of available superheat effects) as reported in other literature [4].

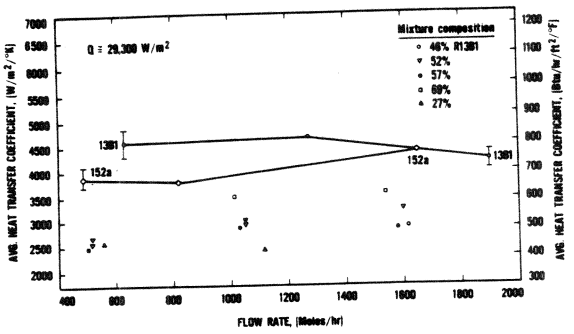


Figure 3: Heat Transfer Coefficient for Various Mixtures

The mixture with the highest average heat transfer coefficient was 69% R13B1 (by moles) whose average heat transfer coefficient was roughly 90% of the value obtained with pure R152a; the composition of 27% R13B1 (by moles) had the lowest h_{TP}, about 35% below pure R152a. Since R13B1 had a higher heat transfer coefficient than R152a, this trend appears logical. Clearly there is sensitivity to composition; however, further investigation needs to be done to determine the optimal mixture proportions so that a basis for prediction may be established. Dependence of the heat transfer coefficient on heat flux was also apparent, as is commonly found in the literature for single component flow boiling. The dependence on quality is not clear. It was expected that larger heat transfer coefficients would be observed consistently as the quality increased. In single component fluids in this range of mass/heat flux, annular flow patterns form at qualities much less than 10% and as the liquid film is evaporated downstream heat transfer coefficients are expected to become larger. At higher mole fractions of R13B1, the heat transfer coefficient rose initially, then decreased to a minimum at roughly 200 diameters and then rose again. With greater heat flux, this behavior was more pronounced. Chaddock and Mathur observed the same phenomena in their oil/refrigerant tests [16]. Throughout our tests annular flow was observed at the outlet sight glass except for those cases with an outlet quality higher than about 95% where mist flow occurred. Fig. 4 displays a typical mixture run with a high mole fraction of R13B1.

A potential explanation for the observed variation in heat transfer was a dependence on the quantity $|\bar{y} - \bar{x}|$, the difference between vapor and liquid compositions, since such a dependence has been shown in pool boiling [14]. All other conditions being equal, the heat transfer coefficient is inversely proportional to $|\bar{y} - \bar{x}|$. In the set of flow boiling tests reported in this paper, this phenomena could not be observed on individual tests, but is noted on a comparison of different tests. Fig. 5 shows the local h_{TP} value plotted against the local $|\bar{y} - \bar{x}|$ for three different test runs with the same molar flow rate and heat flux, but different compositions. If the pool boiling relation held strictly, one would expect a negative slope. But this relationship is observed on comparing tests: the test labelled SVV 226 (Fig. 5) has a lower average $|\bar{y} - \bar{x}|$ (denoted hereafter as $\overline{|y - x|}$) and higher average $\bar{h}_{TP}$ value than the other tests shown. This fact

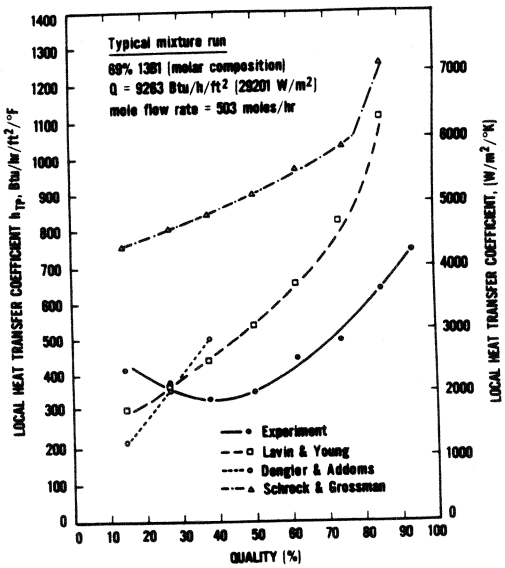


Figure 4: Typical Mixture Run

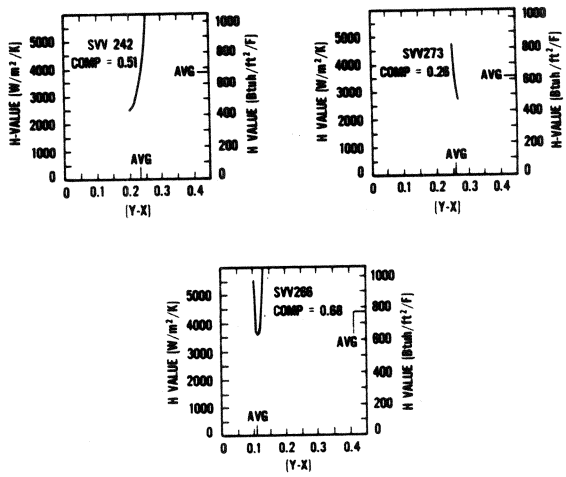


Figure 5: Local H_{TP} Value as a Function of Liquid-Vapor Composition Difference

can be used to predict roughly the average heat transfer coefficient of these mixtures, $\bar{h}_{mix}$ with the following form:

$$\bar{h}_{mix} = \bar{h}_{lv}(1 - \overline{|\bar{y} - \bar{x}|}) \qquad (3)$$

where $\bar{h}_{lv}$ is the average heat transfer coefficient of the less volatile component (in our case, R152a)at the same mole flow rate and heat flux. This form has the advantage that there are no empirical constants (or else by fortune they are one for this mixture). This form predicted the experimental results to within experimental error for 54% of the tests, ±15% for ~90% of the tests and -12%/+22% for all the tests. From

pool boiling experiments, Happel and Stephan suggested the following, more general form:

$$\bar{h}_{mix} = \bar{h}_{id} (1 - K|\bar{y} - \bar{x}|^n) \quad (4)$$

where $\bar{h}_{id}$ is the average heat transfer coefficient, based on a mole fraction weighting of the pure refrigerants, and K and n are empirically determined constants. This form, using a least squares fit to determine the constants, tended to underpredict our experimental data.

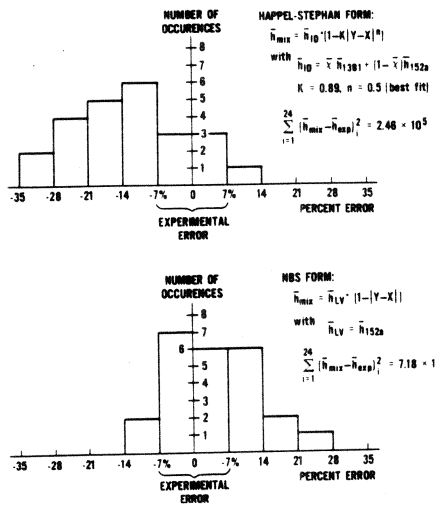


Figure 6: Comparison of Predicted Average h_{mix} to Experiment

Figure 6 is a histogram of the errors of prediction using equations (3) and (4) the sum of the square of the errors for equation (3) is roughly three times smaller than for equation (4). In an individual test, where temperature, pressure and quality changes cause a varying $|\bar{y} - \bar{x}|$, the $|\bar{y} - \bar{x}|$ dependence (when $\bar{y}$ and $\bar{x}$ are local values) does not appear to hold (see Figure 5.) In an individual test, other effects such as local quality probably dominated the heat transfer (from the correlations, one expects h_{TP} to be a function of at least mass and heat flux, quality, and composition). It is hoped that in further planned work, local non-intrusive measurements of composition, quality and deduced flow patterns will clarify the relative importance of these individual dependencies. Once these interdependencies are established improved correlations which include a composition variable may be possible.

CONCLUSIONS

Existing correlations for single phase liquid heat transfer were shown to be in a satisfactory range for mixtures. Two phase heat transfer correlations for pure refrigerants were shown to be within the acepted practice for accuracy scatter existing currently in the literature today. The heat transfer coefficient measured for mixed refrigerants was significantly lower than that measured for either pure refrigerant, and a varying dependence on quality was observed for unknown reasons. The 69% R13B1 mixture had the highest heat transfer coefficient of those mixtures tested since R13B1 had a higher measured heat transfer coefficient than R152a the result is not surprising. The local heat transfer correlations for single component flow do not appear adequate for these non-azeotropic mixtures (Fig. 4); however, the reduction in average heat transfer coefficient may be correlated with some success. Additional empirical work as well as simultaneous heat and mass transfer modelling will be necessary to establish a better composition sensitive correlation.

The selection of a particular best refrigerant mixture for heat pump applications is not apparent. Local heat transfer coefficients cannot be predicted simply, and values lower than that found for pure refrigerants may occur. Further research needs to be done to refine the prediction of the heat transfer coefficient. It is planned that future work will provide measurements of local composition and quality for these mixtures to improve the overall understanding of the heat transfer phenomena.

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APPENDIX A: Correlations for Two Phase Heat Transfer with Pure Refrigerants

The second set of tests (35 runs) involved pure refrigerants and a comparison of local heat transfer coefficients to five predictive correlations listed below:

Schrock and Grossman (as modified by Chaddock and Noeranger)

$$\frac{h_{TP}}{h_L} = 1.85 \left[10^4 Bo + 1.5\left(\frac{1}{X_{tt}}\right)\right]^{2/3} \quad (3)$$

with h_L = Dittus-Boelter relation (1)

and $$X_{tt} = \left(\frac{x}{1-x}\right)^{.9}\left(\frac{\nu_L}{\nu_g}\right)^{0.5}\left(\frac{\mu_L}{\mu_g}\right)^{0.1} \quad (3a)$$

$$Bo = \frac{Q/A}{GH_{fg}} \quad (3b)$$

Dengler and Addoms (as modified by Chaddock and Noeranger)

$$\frac{h_{TP}}{h_L} = 3.0\left(\frac{1}{X_{tt}}\right)^{2/3} \quad (4)$$

with h_L = equation (1)

Lavin and Young

$$\frac{h}{h_{Lo}} = \left(\frac{1+x}{1-x}\right)^{1.16}\left(Bo\right)^{-0.1}\left(6.59\right) \quad (5)$$

with $$h_{Lo} = 0.23\frac{K_L}{D}\left[\frac{GD(1-x)}{\mu_L}\right]^{0.8}\left[Pr_L\right]^{0.33}\left[\frac{\mu}{\mu_w}\right]^{0.14}$$

Chen

$$h_{TP} = h_{NB} + h_{FCV} \quad (6)$$

where $$h_{FCV} = 0.023\left[\frac{G(1-x)D}{\mu_L}\right]^{0.8}\left[\frac{C_{pL}\mu_L}{k_L}\right]^{0.4}\left[\frac{k_L}{D}\right]F$$

and $$h_{NB} = .00122\left[\frac{k_L^{0.79} c_{pL}^{0.45} \rho^{0.49}}{\sigma^{0.5}\mu_L^{0.29} H_{fg}^{0.24} \rho_g^{0.24}}\right]$$

$$\left(\frac{H_{fg}}{T_{sat}\nu_{fg}}\right)^{0.75}(T_w - T_{sat})^{0.99} S \quad (6b)$$

with $F = 1.0$ for $1 \le X_{tt} \le 0.10$

$$= 2.35(1/X_{tt} + 0.213)^{0.736} \text{ for } 1/X_{tt} > 1.0 \quad (6c)$$

and $$S = \left[1 + 0.12(Re_{2\phi})^{1.14}\right]^{-1} \quad Re_{2\phi} < 32.5 \times 10^4$$

$$= \left[1 + 0.42(Re_{2\phi})^{0.78}\right]^{-1} \quad 32.5 \times 10^4 \le Re_{2\phi} < 70.0 \times 10^4$$

$$= 0.1 \quad Re_{2\phi} \ge 70 \times 10^4$$

and the two-phase Reynolds number $Re_{2\phi} = Re_L(1.25)$ [7].

Bennett

Same as Chen, except

$$F = \phi_{L_{tt}}.89\left[\frac{P_{r_L} + 1}{2}\right]^{.44} \quad [10]$$

with $$\phi_{L_{tt}} = 1 + 20/X_{tt} + 1/(X_{tt})^2 \quad [11]$$

Because the correlations are intended to be predictive, no "fitting" by modification of published coefficients was attempted to improve the agreement with the data. In the case of the correlations by Schrock and Grossman (SG) and Dengler and Addoms (DA), coefficients published by Chaddock and Noeranger [2] were used as they were more appropriate to the flow geometry. The Chen correlation requires surface tension values; unfortunately these values were only available for the R-22 tests. Vapor viscosity values needed for the SG and DA correlations were unavailable for R152a, however it was assumed 10% greater than the R13B1 values (roughly the same difference as the liquid viscosities). The results are not sensitive particularly to this assumption. Doubling or halving the R152a vapor viscosity changed the correlation values by a total less than 3%. As shown in Tables A1 to A3, the Chen and Lavin and Young (LY) correlations proved to be the best predictors of local heat transfer coefficient based on this set of experiments. The Chen correlation, derived originally for vertical flow, tends to underpredict consistently the experimental values. One possible reason for underprediction was the Prandtl number assumed in the Chen correlation is 1.

Bennett's modification of the Chen correlation was intended for use with fluids whose Prandtl Number was greater than 2 [10]. Since R-22 has a Pr ~ 2.7, the Bennett modification was applicable. Unfortunately, it tends to overcompensate and overpredict badly our experimental values. A second explanation for Chen's underprediction is flow geometry. Since in this range of mass flow rate and heat flux, horizontal flow yields a higher h_{TP} than vertical flow, the bias is not surprising. Previous applications of the Chen correlation for horizontal refrigerant flow have not yielded satisfactory agreement [2, 17] though the correlation is widely recommended for vertical water flows [5, 7]. In

these tests, simple multiplication by 1.15 of the expression in the Chen correlation brought 80% of the data within 22/+18%.

Table A1

LOCAL H VALUE FOR R-22
EXPERIMENT VS. PREDICTIVE CORRELATION

Correlation	No. of data points	Avg. neg. error (all data)	Avg. pos. error (all data)
Schrock & Grossman	50	-9.1% (9)	33.5% (41)
Dengler & Addoms	38	-21.9% (10)	18.7% (28)
Chen	50	-28.9% (46)	1.4% (4)
Bennett	50	-14.2% (9)	42.8% (41)
Lavin & Young	50	-16.7% (24)	18.8% (26)

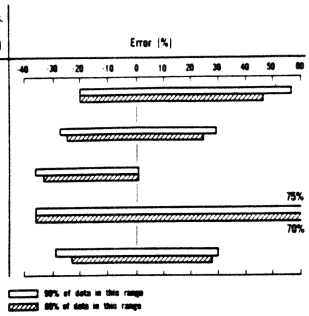

Table A2

LOCAL H VALUES FOR R-152A
EXPERIMENT VS. PREDICTIVE CORRELATION

Correlation	No. of data points	Avg. pos. error	Avg. neg. error
Schrock & Grossman	55	38.9 (55)	0 (0)
Dengler & Addoms	28	33.8 (25)	-40.6 (3)
Chen	–	–	–
Lavin & Young	55	12.3 (22)	-9.3 (33)

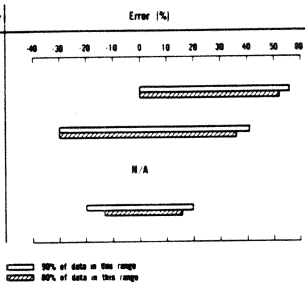

Table A3

LOCAL H VALUES FOR R-1381
EXPERIMENT VS. PREDICTIVE CORRELATION

Correlation	No. of data points	Avg. pos. error	Avg. neg. error
Schrock & Grossman	52	27.0 (34)	-19.7 (18)
Dengler & Addoms	44	17.0 (7)	-35.9 (37)
Chen	–	–	–
Lavin & Young	51	21.1 (14)	-22.3 (38)

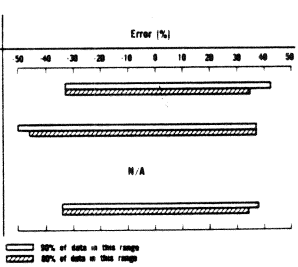

Table A4

LOCAL H VALUES FOR MIXTURES
EXPERIMENT VS. PREDICTIVE CORRELATION

Correlation	No. of data points	Avg. Pos. error	Avg. neg. error
Schrock & Grossman	214	127.8% (214)	0 (0)
Dengler & Addoms	138	42.2% (115)	-22.2% (24)
Chen	–	–	–
Lavin & Young	214	47.1% (184)	-13.7% (29)