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A COMPARISON OF EXPERIMENTAL MEASUREMENTS OF LOCAL FLOW BOILING HEAT TRANSFER COEFFICIENTS FOR R11 AND R123

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ABSTRACT

This paper presents a comparison of the measured horizontal, smooth-tube, flow boiling heat transfer coefficient of R11 to that of its proposed ozone safe replacement, R123. The fluid properties of R11 and R123 are similar. The flow boiling data for the two fluids are similar for the convective region. However, the heat transfer coefficient for R11 in the nucleate flow boiling region was consistently observed to be, on average, 8.5% to 33% larger than that for R123. The influence of Reynolds number and heat flux on the heat transfer-thermodynamic quality relationship is also presented. Predictions of the heat transfer coefficient with two open literature flow boiling correlations were compared to the measured data. The heat transfer coefficients predicted with the correlations were, on average, from 13% to 57% greater than the measured heat transfer coefficients. Not until recently has there been sufficient thermodynamic or transport data for R123 to perform an accurate heat transfer analysis for that fluid. Great care was taken to ensure that the most carefully measured property data were used for this analysis. For the convenience of the reader, both the R123 and R11 property data used in this study are presented.

NOMENCLATURE

English symbols

a	coefficients of CSD equation of state
b	coefficients of CSD equation of state
Bo	Boiling number, $q''\pi D_i^2/(4\dot{m}\lambda)$
c	coefficients of ideal gas heat capacity
Co	Convection number, $((1-x)/x)^{0.8}\,(\rho_v/\rho_l)^{0.5}$
C_p	heat capacity (kJ/kmol-K)
D_i	internal tube diameter (m)
E_c	percent overprediction of data for $x < 0.13$
E_n	percent overprediction of data for $x > 0.13$
F	enhancement factor in eq. 7
F_l	fluid factor in eq. 8
k	thermal conductivity
h	horizontal flow boiling heat transfer coefficient (W/m^2-K)
h_l	single phase, all liquid, heat transfer coefficient (W/m^2-K)
h_n	pool boiling heat transfer coefficient in eq. 7 (W/m^2-K)
i	refrigerant enthalpy (kJ/kg)
L	length (m)
$\dot{m}$	mass flow rate (kg/s)
N	nucleate factor in eq. 7
P	pressure (Pa)
P_r	reduced pressure, P/P_c
q''	uniform heat flux (W/m^2)
Re	Reynolds number, $\rho_l u D_i/\mu_l$
T	temperature (K)
T_c	critical temperature (K)
T_r	reduced temperature, T/T_c
T_{wi}	internal wall temperature (K)
u	average liquid velocity (m/s)
x	thermodynamic quality
x_t	transition quality
z	coordinate along tube axis (m)

Greek symbols

ρ_l	density of liquid (kg/m^3)
ρ_v	density of vapor (kg/m^3)
λ	heat of vaporization (kJ/kg-K)
μ	dynamic viscosity (kg/m-s)

Subscripts

c	critical
l	liquid
s	saturation state
v	vapor

INTRODUCTION

This paper presents a comparison of measured horizontal flow boiling heat transfer coefficients for the refrigerant trichlorofluoromethane (R11) to that of its proposed replacement, the alternative refrigerant dichlorotrifluoroethane (R123). The measurements for both fluids were obtained from the same smooth-tube test rig to ensure that observed differences were due to the characteristics of the fluids.

By far, the largest use of R11 as a heat transfer fluid is in centrifugal chillers for large water chilling/air-conditioning systems. The evaporators of most centrifugal chillers are shell-and-tube heat exchangers with the tubes nearly filling the entire shell, leaving little space for flow through the bundle. For this configuration, water flows inside the tubes, and R11 enters the bottom of the bundle flowing at a high velocity across the outside of the tubes. The resulting heat transfer mechanism is convective boiling, i.e., a two-phase mixture travels past the tubes while nucleation occurs on the outside of the tube wall.

Some of the mechanisms of flow boiling inside tubes at low quality can be considered valid for exterior tube flow as well. In the bubbly flow regime, nucleation occurs at the inner tube wall as the bulk of the flow, which is liquid, flows past the bubbles and aids in their detachment. Shell-side boiling is strongly dependent upon several variables: the tube bundle layouts, pass arrangements, flooded condition, and many other factors. It is difficult to conduct a single experiment that will be versatile enough to satisfy all flow conditions. However, an experiment having relatively few flow parameters can be used to obtain a direct comparison of the boiling characteristics of R11 and R123 which are due solely to the differences in the fluids. This comparison can, in turn, be used to project differences in heat transfer for processes of similar mechanisms such as boiling in a flooded evaporator for R11 as compared to R123. This is not to say that the results of this paper, internal flow boiling with constant heat flux, can be used to design chillers. However, the results can be used as a starting point from which a relative size of R11 versus R123 chiller can be anticipated.

TEST APPARATUS

Following is a discussion of the test apparatus and the accuracies associated with the individual measurements. A schematic of the test apparatus with the location of measurement devices is shown in Fig. 1. The test section consisted of two insulated, 4 m lengths (connected with an unheated u-bend) of 9.1 mm internal diameter type 304 stainless steel tube with a .25 mm wall thickness. The tubes were heated electrically with a D.C. power source giving a range of uniform heat fluxes (q'') from 10 to 40 kW/m^2. The accuracy of the heat flux measurement was within approximately ±1.5%. The refrigerant was pumped with an oil-free pump through the inside of the tube giving a range of Reynolds numbers from 4000 to 30000. The refrigerant mass flow rate was measured with a turbine flow meter. The accuracy of the mass flow rate ($\dot{m}$) measurement was approximately ±1-2%, resulting in a slightly higher uncertainty cf 1.4-2.2% for the Reynolds number calculation. The outside tube wall temperature was measured approximately every 0.3 m along the tube length at four circumferencial positions (north, south, east, and west). A line drawn through the north and south positions was vertical. The accuracy of the wall temperature measurement was approximately ±0.2K. The fluid pressure and instream

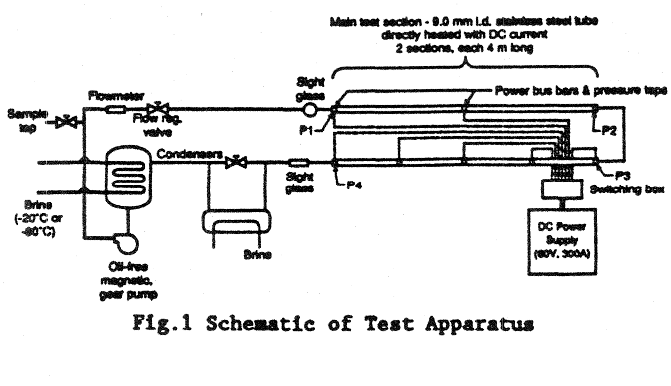

Fig.1 Schematic of Test Apparatus

fluid temperature were measured at four locations (see Fig. 1). The accuracy of the pressure measurement was approximately ±2 kPa. The fluid pressure at intermediate locations was linearly interpolated between the measured pressures P1 and P2 for leg 1 of the rig and P2 and P3 for leg 2 of the rig (see Fig. 1). The linear interpolation should introduce negligible errors since the heat flux is uniform. For example, the two phase pressure drop correlation given by Pierre (1964) and the homogeneous pressure drop model (for constant heat flux) is linear with respect to quality.

Several quantities, i.e., the saturation temperature, the refrigerant enthalpy, the thermodynamic quality (x), and finally, the two-phase heat transfer coefficient were calculated using the previously discussed measured quantities. The first three quantities were obtained directly from a thermodynamic equation of state which will be discussed in a later section. The local fluid saturation temperature (T_s) was obtained from that temperature which corresponded to the measured (or interpolated) local saturation pressure. The change in the refrigerant enthalpy (Δi) for a given heat input ($q''\pi D_i \Delta L$) was calculated at every thermocouple station as follows:

$$\Delta i = \frac{q''\pi D_i \Delta L}{\dot{m}} \tag{1}$$

The inlet to the test section was always subcooled. The point of incipient saturation was calculated from the amount of energy required to raise the subcooled liquid to the saturated state. At that axial position within the tube, all thermodynamic properties were at the saturated liquid state; the thermodynamic quality was zero. The thermodynamic quality (x) was calculated along the tube length (z) as follows:

$$x = \frac{\Delta i}{\lambda} \tag{2}$$

The temperature of the inside tube wall (T_{wi}) was obtained from the outside wall temperature measurement and the measured heat flux, accounting for conduction through the thin tube wall. The flow boiling heat transfer coefficient (h) was calculated from the corrected wall temperature, the measured heat flux (q''), and the fluid saturation temperature (T_s):

$$h = \frac{q''}{(T_{wi} - T_s)} \tag{3}$$

The calculation of the heat transfer coefficient is accurate, for 99.7% confidence, to within ±3% in the nucleate regime where relatively large driving temperatures (20 K) exist, and ±8% in the annular regime where the temperature difference is relatively small (1 K).

PROPERTIES OF R123

Until recently, very few data have existed for the thermodynamic and transport properties of R123. Therefore, it is important to present the property data that were used in the calculation of the flow and heat transfer parameters and to note the claimed accuracies to quantify the uncertainty associated with the results of this study. Following is a presentation of the fluid properties used for R123.

There were three sources of measured R123 data which were available for use in this study. Saturation pressure, temperature, and liquid densities were obtained from the measured data of Morrison and Ward (1990). The vapor density was obtained from Weber (1990). The transport properties of R123 were obtained from Shankland (1990).

Morrison and Ward (1990) have measured the thermodynamic vapor pressure and temperature, and also both the subcooled and saturated liquid densities for R123. The accuracy of their temperature, pressure, and density measurements was ± 0.001 K, 0.2 kPa, and 0.3%, respectively. They checked their measurements against two independent measurement methods and also against the small amount of existing data. The comparison proved that their saturation densities agreed to within ±0.2%, and the saturation pressures were within 2 kPa of other existing data for the 273 to 373 K temperature range.

Weber (1990) measured the vapor temperature, pressure, and density accurately with a Burnett/isochoric PVT apparatus. The uncertainty of the measurements were as follows: ±1 mK, $\pm 2\times10^{-4}$ MPa, and ±0.03% for the temperature, pressure and vapor density measurements, respectively.

The transport properties for R123 used in this study were taken from Shankland (1990). The accuracy of the vapor thermal conductivity (k_v) measurements is approximately ±3%. Shankland (1990) presents a fit for the k_v data with respect to the vapor temperature which is within ±0.3% of the measurements:

$$k_v(\mathrm{mW/m\text{-}K}) = 8.2342\left[1 + 10.5887\times10^{-3}(T_v(\mathrm{K})-273.15)\right] \quad (4)$$

The accuracy of the liquid thermal conductivity (k_l) measurements is within ±1% of the measurements. The fit of the k_l data with respect to the liquid temperature given by Shankland (1990) is within ±0.1% of the measurements:

$$k_l(\mathrm{mW/m\text{-}K}) = 81.1234\left[1 - 2.6878\times10^{-3}(T_l(\mathrm{K}) - 273.15)\right] \quad (5)$$

The liquid dynamic viscosity measurements (μ_l) are accurate to within ±1%. The fit of the μ_l to T_l is within ±0.1% of the measured data and is given by Shankland (1990) as follows:

$$\mu_l(\mathrm{mPa\text{-}s}) = \exp\left[1.07460/T_r - 0.7084 - 2.57586T_r\right] \quad (6)$$

where the critical temperature of R123 used in equation 6 was 457.15K.

The pressure, temperature, and liquid density data of Morrison and Ward (1990) and the vapor density of Weber (1990) were fitted to the Carnahan-Starling-DeSantis (CSD) equation of state which owes its final development to DeSantis, et al. (1976). The coefficients obtained from the fit are valid from 273 to 423 K, and are given in Table 1. The equation of state coefficients were then used as input to the algorithms of Morrison and McLinden (1986) which applies the CSD equation of state to refrigerants. Table 2 shows the values of the thermodynamic properties of R123 as predicted by using the Morrison and McLinden (1986) algorithm. For a given temperature, the prediction of the pressure differed from the measured values by less than ±0.3%. The predictions of the vapor and liquid densities differed an average of ±0.7 and ±0.3%, respectively, from the measured values. The resulting uncertainty of the latent heat (λ) calculation is less than 1%.

Table 3 shows the values of the thermodynamic properties of R11 as predicted with the Morrison and McLinden (1986) algorithm. The R11 properties obtained

<u>Table 1 CSD Eq. of State Coefficients for R123 and R11</u>

	Subscripts of Coefficients (n) 0	1	2
R123			
a_n	5.8856290E+3	-2.2446861E-3	-1.0342735E-6
b_n	1.9576620E-1	-1.6784282E-4	-9.9567402E-8
c_n	2.9260400E+1	3.0299400E-1	-1.9290700E-4
R11			
a_n	4.9715400E+3	-2.2466900E-3	-5.1194300E-7
b_n	1.7665900E-1	-1.7453100E-4	-3.4971700E-8
c_n	2.2041800E+1	2.6089500E-1	-2.4531900E-4

Note: $a(\mathrm{kJ\ m^3/kmol^2}) = a_0 \exp(a_1 T + a_2 T^2)$
$b(\mathrm{m^3/kmol}) = b_0 + b_1 T + b_2 T^2$
$C_p^0(\mathrm{kJ/(kmol\ K)}) = c_0 + c_1 T + c_2 T^2$

<u>Table 2 Thermodynamic Properties of R123</u>

Ts (K)	Ps (kPa)	ρ_l (kg/m³)	ρ_v (kg/m³)	λ (kJ/kg)
300.00	97.85	1458.4	6.242	170.9
305.00	117.03	1445.7	7.386	168.9
310.00	139.04	1432.8	8.689	166.8
315.00	164.16	1419.7	10.166	164.7
320.00	192.67	1406.4	11.833	162.5
325.00	224.87	1392.9	13.709	160.3
330.00	261.07	1379.1	15.813	158.0
334.10	293.92	1367.6	17.721	156.1
335.00	301.57	1365.1	18.164	155.6
340.00	346.70	1350.8	20.785	153.2
345.00	396.79	1336.1	23.700	150.7
350.00	452.19	1321.1	26.935	148.1
355.00	513.23	1305.8	30.517	145.4
360.00	580.27	1290.1	34.479	142.6

<u>Table 3 Thermodynamic Properties of R11</u>

Ts (K)	Ps (kPa)	ρ_l (kg/m³)	ρ_v (kg/m³)	λ (kJ/kg)
300.00	112.87	1471.6	6.474	179.8
305.00	133.71	1459.7	7.585	177.8
310.00	157.41	1447.6	8.838	175.9
315.00	184.24	1435.4	10.247	173.9
320.00	214.46	1423.1	11.824	171.8
325.00	248.34	1410.5	13.583	169.7
330.00	286.15	1397.7	15.540	167.5
335.00	328.18	1384.7	17.709	165.2
337.70	352.73	1377.6	18.976	164.0
340.00	374.72	1371.5	20.109	162.9
345.00	426.07	1358.0	22.758	160.5
350.00	482.52	1344.2	25.673	158.1
355.00	544.39	1330.1	28.878	155.6
360.00	611.97	1315.8	32.395	152.9

from the equation of state were used in the analysis of the data. Agreement of ±0.5% was achieved between the equation of state predictions for the thermodynamic properties of R11 and the R11 properties given in ASHRAE (1989).

TEST RESULTS

Detailed Discussion

Results are presented as plots of the local heat transfer coefficient (h) versus thermodynamic quality (x) for fixed reduced pressure, $P_r = 0.08$, (353 kPa for R11 and 294 kPa for R123). The tests were run for two constant, wall heat flux conditions (20 and 30 kW/m²) and two liquid Reynolds numbers (18000 and 24000). The all-liquid Reynolds numbers and the reduced pressures were evaluated just prior to the onset of boiling condition. Typically, the pressure dropped 50 to 70 kPa from the inlet to the outlet of the test section.

The values for the heat transfer and flow parameters presented in the graphs and discussion are nominal values. Table 4 shows the actual values of the flow parameters for each test. These values were steady with a standard deviation of less than 1% for any test.

Heat transfer coefficients obtained from the last three consecutive sensors at the exit of the test section were ignored due to suspected bend-flow effects. The questioned data exhibited an upward curvature which appeared regardless of the fluid and test condition. The most revealing signature of the systematic error associated with the last three sensors was that the data taken with them did not agree with data for similar heat transfer conditions taken with other sensors. Also, data from the sensor immediately after the u-bend were ignored for similar reasons.

Figures 2-7 are plots of the measured heat transfer data of this paper. There are two distinct regions which are visible on most plots. In the low quality region, the h is either decreasing with respect to quality, or it is relatively independent of it. This is the nucleate flow boiling region, where the phase change is governed by the growth and release of bubbles from the inner tube wall. Both of the tested fluids had sufficiently low thermal conductivities. This is typical for refrigerants where the nucleate flow boiling region had a heat flux dependency. This dependency had the characteristic of exhibiting larger h's for higher q"'s at the lowest qualities. The convective flow boiling region is nominally independent of heat flux and strongly dependent on mass flow rate. The dominant heat transfer mechanism of the second region is convective evaporation from the annular vapor-liquid interface. Flows with larger Reynolds numbers result in a greater change in heat transfer coefficients with quality. Thus, the heat transfer coefficient is relatively independent of quality for x approximately less than 13% (nucleate region) and increases rapidly with quality for x approximately greater than 13% (convective region).

Figure 2 is a comparison of the measured h versus x for R11 to that of R123 for q" = 20 kW/m^2 and Re = 18000. The figure represents a nearly complete synopsis of the magnitude of the flow boiling h for R11 as compared to that for R123 for all the data presented here. That is the magnitude of the flow boiling heat transfer coefficient for R11 and that for R123 are similar over the quality range tested. This is not really surprising since the thermodynamic and transport properties of the two fluids are nearly identical. A closer look at the data reveals that the R11 data are marginally an average of 8.5% higher than that of R123 for thermodynamic qualities less than 30%. There is a 95.5% probability that this difference is larger than that which could be attributed to the measurement uncertainty, which is ±5% for this quality range. For qualities above 30%, only one measurement is available

Table 4 Actual Heat Transfer Parameters for Data

Data Symbol	P_s (kPa)	q" (kW/m^2)	Re	μ_1 (kg/m-s)
○	355.0	20.065	18133	3.017E-4
●	294.1	19.974	17999	3.254E-4
□	352.9	19.984	24097	3.022E-4
■	294.0	19.920	24054	3.254E-4
◇	352.2	30.075	17944	3.023E-4
◆	293.9	26.522	18069	3.254E-4
△	358.5	30.035	24055	3.009E-4
▲	291.4	26.983	23823	3.267E-4

(x = 35%) for R123; it is within 2% of the R11 data.

The data of Fig. 3 are for the same heat flux as that of Fig. 2, but for a larger Reynolds number (Re = 24000). Figure 3 is consistent with Fig. 2, in that the heat transfer coefficients for the two fluids are very close (within 2%) for a quality of 20%. However, the Fig. 3 data exhibit a difference in h between R11 and R123 in the low quality region which is larger than that for the data of Fig. 2. For qualities below 20%, the R123 data are an average of 12.5% lower than the R11 data for q" = 20 kW/m^2 and Re = 24000. It is not known why a larger difference between the h's for the two fluids in the nucleative region would exist for the larger Reynolds number.

As a consequence of the increase in Re, the h is

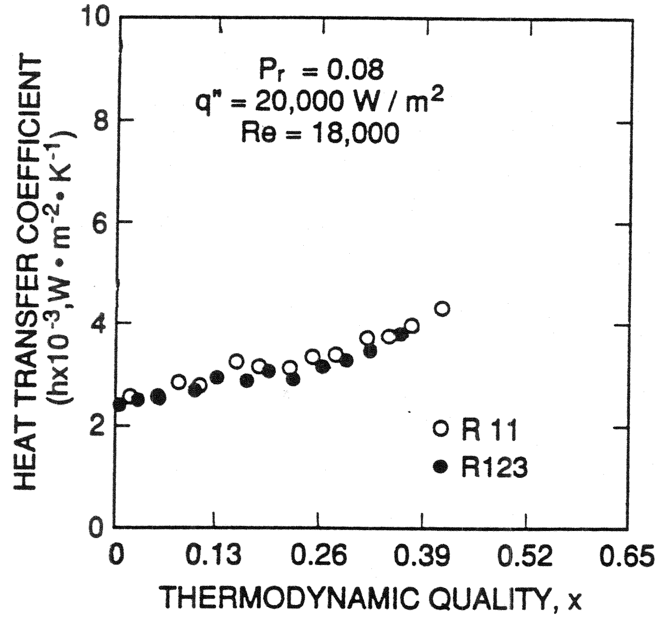

Fig.2 Comparison of h Versus x for R11 and R123 at Re = 18000 and q" = 20 kW/m^2

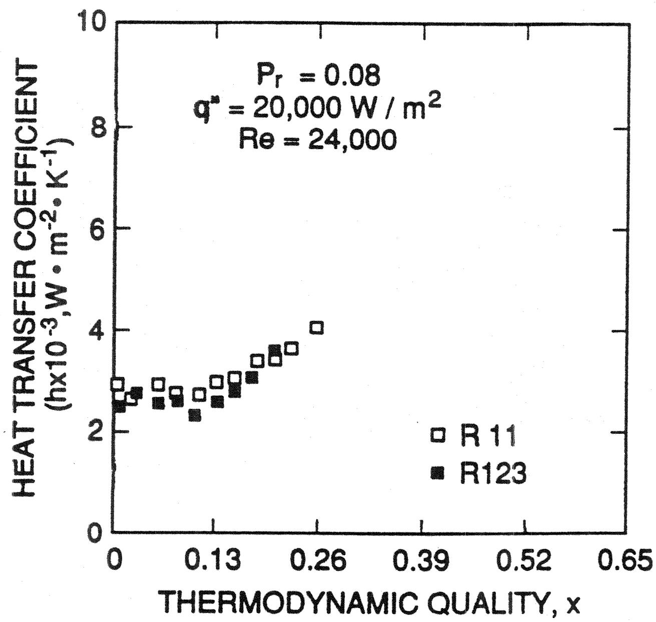

Fig.3 Comparison of h Versus x for R11 and R123 at Re = 24000 and q" = 20 kW/m^2

shown to increase more rapidly than that of Fig. 2 for qualities above 15%. In other words, the slope of the data for the larger Reynolds number is greater than the slope of the data for the smaller Reynolds number. The difference between the heat transfer in the convective region can be seen more clearly in Fig. 4, which is a composite of Fig. 2 and Fig. 3. Figure 4 shows that, at a quality of 26%, R11 exhibits a flow boiling heat transfer coefficient for Re = 24000 that is approximately 19% greater than that for Re = 18000. That is, a 33% increase in Reynolds number leads to a 19% increase in heat transfer at the same thermodynamic quality. Although this demonstrates that the mass flow has a strong effect on h in the convective region, the same plot also demonstrates that a 33% increase in Reynolds number has a slight effect on the heat transfer coefficient in the nucleative region which has increased on average by 6%.

Figure 5 presents a comparison of the heat transfer for the two fluids for a Reynolds number of 18000 and a higher heat flux (30 kW/m^2). The heat transfer coefficient for the R11 data is 10% and 6% higher than that for R123 for qualities of 1% and 5%, respectively. However, the difference between the heat transfer coefficients for the two fluids is less than 1% for both fluids at x = 10% and x = 14%. For qualities greater than 20%, the supremacy of the R11 heat transfer coefficient varies sporadically from 5% to 22% greater than the R123 coefficient. The irregular R11 values complicate the comparison of the data sets. But, for the most part, the flow boiling heat transfer coefficient for R11 is larger than that for R123.

The flow boiling heat transfer coefficient for R11 and R123 for q" = 30 kW/m^2 and Re = 24000 are presented in Fig. 6. The convective region appears to begin between 13% and 17% quality for both fluids. In the convective region, the data agree within 1% to 3% of each other. In the nucleative region, the h for R11 varies from 34% to 18% greater than that for R123 with the percent difference decreasing for increasing quality. The great difference between the heat transfer coefficients for the two fluids may be partly attributed to the lower heat flux (27 W/m^2) of the R123 data. This contribution should be small considering that only a 10%

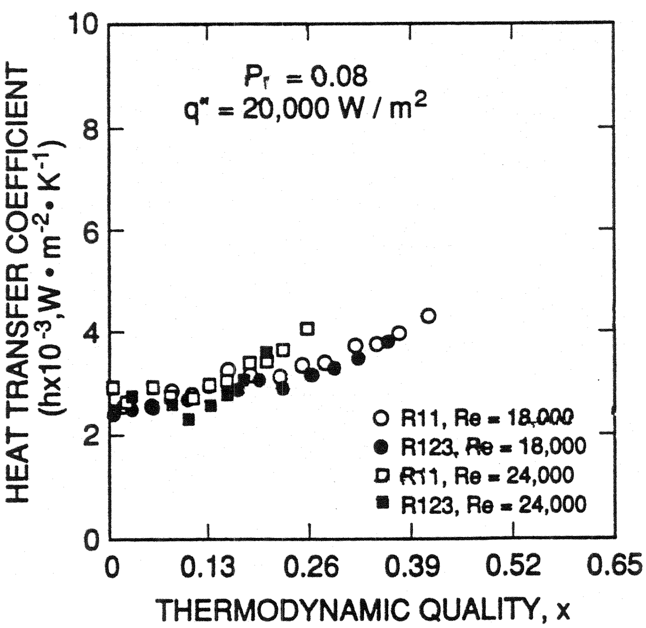

Fig.4 Effect of Reynolds Number on the Heat Transfer-Quality Dependence for both R11 and R123

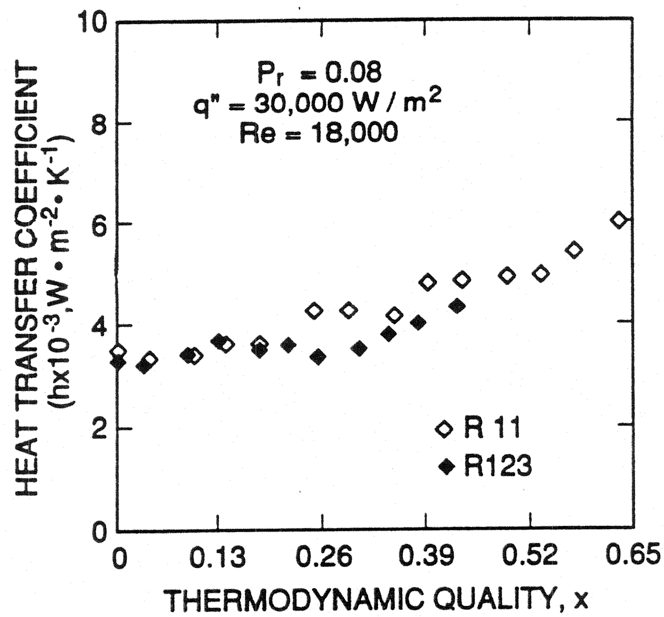

Fig.5 Comparison of the Flow Boiling Characteristics of R11 to R123 for q" = 30 kW/m^2 and Re = 18000

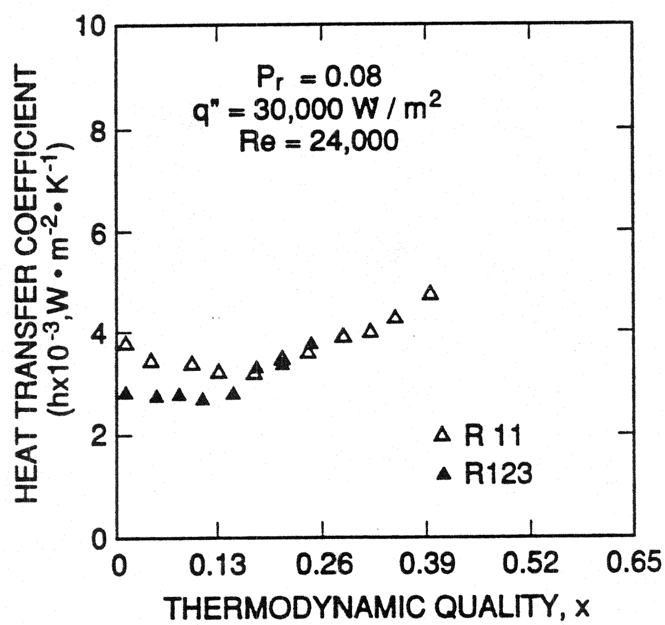

Fig.6 Comparison of h Versus x for R11 and R123 at q" = 30 kW/m^2 and Re = 24000

increase in the R123 heat transfer coefficient (Re = 24000) was achieved for a 35% increase in the heat flux. Correspondingly, the R123 data in Fig. 6 should be approximately 3% higher if they were taken at 30 kW/m^2 instead of 27 kW/m^2. The difference between the heat transfer coefficients for R11 and R123 in the nucleative region could be reduced at most by only 1%. Hence, the R11 and R123 data at q" = 30 kW/m^2 and Re = 24000 show a distinct transition from the nucleative to the convective region. In the former, the flow boiling heat transfer coefficient for R11 is from 33% to 17% greater than that for R123, and where in the latter region there is only 1% to 3% difference.

Figure 7 is Fig. 6 with the inclusion of data at q" =

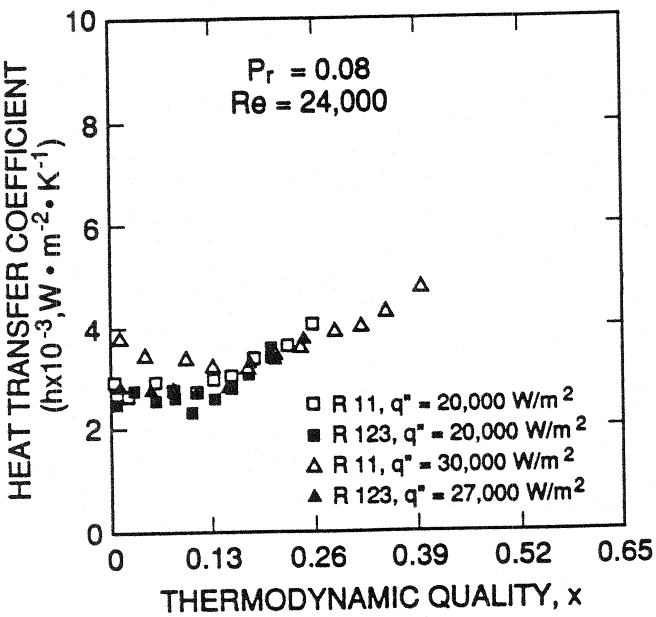

Fig.7 Effect of Heat Flux on the Flow Boiling Characteristics of R11 and R123

20 kW/m² and the same Reynolds number (24000). Figure 7 demonstrates the effect of increased heat flux on the heat transfer-quality relationship. For qualities above 15%, all of the data are within ±3% of each other, which is within the ±5% accuracy of the data. Implying the following: 1) the h in the convective region is relatively unaffected by the 50% increase in heat flux from 20 to 30 kW/m²; and 2) for Re = 24000, there is no significant difference between the convective flow boiling for R11 and that for R123 for either heat flux. However, the h for the quality range of 0 to 13% is strongly influenced by the change in heat flux. The effect of heat flux is greatest for the lowest qualities and becomes negligible as the quality increases into the convective region. The R11 flow boiling heat transfer coefficient for q" = 30 kW/m² is 32% and 12.5% larger than that for q" = 20 kW/m² at 1% and 13% quality, respectively. The R123 flow boiling heat transfer coefficient for q" = 27 kW/m² is 15% and 9% larger than that for q" = 20 kW/m² at 1% and 13% quality, respectively.

Overview Discussion

Tables 5 through 7 contain summaries of the effects of heat flux, Reynolds number, and fluid on the measured heat transfer coefficient. The definition of the transition quality, x_t, varies for these tables. In general, it represents the quality at which the investigated characteristic under goes a step change.

Table 5 shows the effect of heat flux on the heat transfer coefficient for all test conditions. For thermodynamic qualities less than 15%, a 50% increase in heat flux results in an average increase of 23.4% in the h for R11 at a Re = 24000. The enhancement of h, for the same fluid and Reynolds number, due to the same increase in q", drops sharply to 2.3% for qualities above 15%. Although the transition quality seems to vary randomly, it is evident for all of the measured data that the effect of heat flux is more prevalent for qualities below x_t than for above it.

Table 6 contains a summary of the effect of increased Reynolds number on all of the measured data. All but the data for R123 at q" = 30 kW/m² follow the trend of a larger percentage increase in h due to an increase in Reynolds number for the highest quality regions than for the lowest qualities. The transition quality for R11 occurs at 20% and 30%, while that for R123 occurs at 18% and 15% for the 20 kW/m² and 30 kW/m² condition, respectively.

The division between the nucleative sensitive regime and that dominated by convection cannot be precisely obtained from the data. For convenience, a non-rigorous definition for the thermodynamic quality where the heat transfer characteristics transition from nucleative to convective is defined as the average of the heat flux and Reynolds number transition qualities given in Tables 5 and 6, respectively. According to this definition, the nucleative regime is below x = 24% for R11 and below x = 19% for R123.

Table 7 contains a comparison of the flow boiling heat transfer coefficient for R11 to that of R123 for all of the measurements. The R11 heat transfer coefficient is from 7.6% to 27% greater than that for R123 for qualities below the transition quality. The difference between the heat transfer coefficients for the two fluids diminishes to an average of ±2.1% for thermodynamic qualities above the transition quality. The transition quality varies from 10% to 30% depending upon the fluid and heat transfer conditions and has an average value of 18.75% for all tests. Notice that for Re = 24000, where there appears to be a visible change in the slope of the data from the nucleative to the convective region, the transition quality has a small range of 15% to 20%. Contrast this to the data for Re = 18000, where the data lacks a well-defined slope change, the transition quality is not as clearly defined, varying from 10% to 30%.

Table 5 Effect of Heat Flux on Heat Transfer Coefficient

		%h ↑ for $x < x_t$		%h ↑ for $x > x_t$		%x_t	
		R11	R123	R11	R123	R11	R123
50% ↑ in q" (35% for R11)	Re = 18000	22.3	25.0	9.0	4.0	30	24
	Re = 24000	23.4	10.3	2.3	2.0	15	20

Table 6 Effect of Reynolds Number on Heat Transfer

		%h ↑ for $x < x_t$		%h ↑ for $x > x_t$		%x_t	
		R11	R123	R11	R123	R11	R123
33% ↑ in Re	q" = 20 kW/m²	6.1	5.9	16.5	23.3	20	18
	q" = 30 kW/m² (27 kW/m² for R11)	6.9	18.3	3.0	1.9	30	15

COMPARISON OF DATA TO EXISTING CORRELATIONS

Two horizontal, smooth-tube, flow boiling heat transfer correlations from the open literature were compared to the heat transfer data. Both correlations

Table 7Comparison of Heat Transfer of R11 to that of R123

		% h that R11 higher than R123 for $x < x_t$	% difference h between R11 and R123 for $x > x_t$	%x_t
q" = 20 kW/m²	Re = 18000	8.5	2.0	30
	Re = 24000	12.5	2.0	20
q" = 30 kW/m²	Re = 18000	7.6	1.9	10[1]
	Re = 24000	27.0	2.4	15
Average over all q" and Re		13.9	2.1	18.75

[1] For x_t > 20% heat transfer coefficient for R11 was an average of 12% higher than that for R123.

are a superposition type, where the nucleative and convective terms are multiplied by factors and then summed to make up the total two-phase heat transfer coefficient. The Jung (1988) correlation consists of a pool boiling heat transfer coefficient (h_n) by Stephan and Abdelsalam (1980), modified by a nucleate factor, N, added to the single phase heat transfer coefficient (h_l) from the Dittus-Boelter (1930) equation, modified by and enhancement factor, F:

$$h = N\ h_n + F\ h_l \tag{7}$$

The Kandlikar (1987) correlation is a sum of a nucleative term consisting of the Boiling number (Bo) and a convective term consisting of the Convective number (Co), which is multiplied by h_l:

$$h = C_1\ Co^{C2}\ h_l + C_3\ Bo^{C4}\ F_l\ h_l \tag{8}$$

where C_1 through C_4 are constants given in Kandlikar's paper and depend upon the flow regime. The fluid factor, F_l, is a function of the fluid properties and is not given for R123. Since the properties of R11 and R123 are similar, the same fluid factor for R11 (1.3) was used in the prediction of R123.

The main difference between the convective portion of the two correlations is that the Jung (1988) correlation puts more emphasis on this term than the Kandlikar (1987) correlation does. This is an unavoidable consequence of fitting the correlations to two different data sets. The basic difference between the nucleative term of the correlations is that Jung (1988) alters a two-phase correlation and Kandlikar alters a single-phase correlation.

Figure 8 shows a comparison of data from Fig. 2 with the two flow boiling correlations. The correlations were compared for the same Reynolds number (Re = 18000), the same heat flux ($q'' = 20\ kW/m^2$), and the same reduced pressure ($P_r = 0.08$) as the data. The most apparent observation is that both correlations go through the data at x = 13%. The extreme deviations from the data occur at x = 0.7%. This is where the Jung correlation is 50% higher than the data and where the Kandlikar correlation is 37% higher than the data. At x = 26% both correlations intersect each other and are 18% greater than the data. As the quality increases to 7%, the over-predictions decrease to 7% and 18% for the Jung and the Kandlikar methods, respectively. The Kandlikar (1987) prediction, because of its lesser emphasis on the convective term, is lower than the Jung (1988) prediction for qualities greater than 30%.

Most of the data for R11 and R123 was over-predicted by both correlations. The exceptions were the two data points shown in Fig. 8 and five data points from the R11 data of Fig. 5. Tables 8 and 9 contain the percentage that the measured heat transfer coefficients were over-predicted by the correlations. The second and third columns contain the percent difference between the data below 13% quality and the predictions (E_n) with the Jung (1988) and the Kandlikar (1987), respectively. The forth and fifth columns contain the percent difference between the data above 13% quality and the predictions (E_c) with the Jung (1988) and the Kandlikar (1987) correlations, respectively. The best guess of nucleate flow boiling regime for both the data and the correlations is from x = 0% to x = 13%. Table 8 shows that the nucleative portion of the R11 data for all heat fluxes and Reynolds numbers is more closely predicted with the Jung (1988) correlation than with the Kandlikar (1987) correlation. This is a 13% versus 22.5% over-prediction, respectively. It might be argued that the use of a pool boiling correlation for the nucleative portion of a flow boiling correlation results in closer predictions than

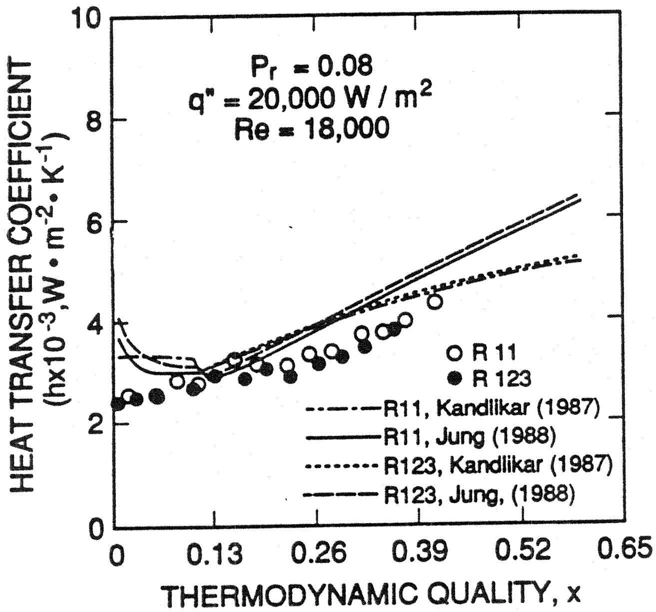

Fig.8 Comparison of Data to Existing Correlations

the use of just the Boiling number. However, there appears to be no advantage associated with the use of the pool boiling component over the use of the Boiling number for the prediction of the nucleative portion of the R123 flow boiling data. Here the data is over-predicted by 38% with the Jung correlation and over-predicted by 36% with the Kandlikar correlation. The convective region is predicted equally well by both correlations. The R11 convective data and the R123 convective data are over-predicted by approximately 16% and 25%, respectively, by both correlations. This is probably due to similar convective terms and also due to the quality range tested. Since the two correlations begin to diverge at about x = 30%, see Fig. 8., a greater difference in the predictions between the two correlations would have been seen for data with a higher

Table 8 Percent Over-prediction of R11 Data by Correlations

		E_n		E_c		max. x for data
		Jung (1988)	Kandlikar (1987)	Jung (1988)	Kandlikar (1987)	
$q'' = 20\ kW/m^2$	Re = 18000	12	21	11	12	41%
	Re = 24000	11	25	18	20	26%
$q'' = 30\ kW/m^2$	Re = 18000	12	24	6	3	64%
	Re = 24000	17	20	29	31	39%
Average over all q" and Re		13	22.5	16	16.5	42.5%

Table 9 Percent Over-prediction of R123 Data by Correlations

		E_n		E_c		max. x for data
		Jung (1988)	Kandlikar (1987)	Jung (1988)	Kandlikar (1987)	
$q'' = 20\ kW/m^2$	Re = 18000	33	25	23	20	36%
	Re = 24000	30	33	28	28	26%
$q'' = 30\ kW/m^2$	Re = 18000	31	28	18	20	43%
	Re = 24000	57	57	31	34	25%
Average over all q" and Re		37.75	35.75	25	25.5	32.5%

quality range.

A surprising outcome is that both correlations indicate that the heat transfer for R123 is higher than that of R11 for all thermodynamic qualities. Contrary to this, the measured data for R11 at low quality were shown to be almost always greater than that for R123. The greatest deviation between R123 and R11 is predicted with the Jung (1988) correlation at x = 3% where the h prediction for R123 is 8% greater than that for R11. The same correlation predicts a shrinking of the difference between the h of R123 and R11 to 2% at x = 50%. The Kandlikar (1987) correlations predicts a lesser difference between the h for R11 and the h for R123 than does the Jung (1988) correlation for x = 3% (R123 less than 1% greater than R11). Like the Jung correlation, the Kandlikar correlation predicts that the h for R123 at x = 50% is only 2% higher than that for R11.

The data of Fig. 8 shows that the heat transfer coefficient of R11 is roughly 8.5% greater than that of R123 in the nucleative regime. Recall that this study showed anywhere from 8.5% to 33% difference between the heat transfer of R123 and R11 in the nucleative regime, with the R11 heat transfer higher than that for R123. Contrary to this, the Jung (1988) correlation predicts an 8% heat transfer difference in the nucleate regime; however, it is in the opposite direction (R123 higher than R11). The data agrees with the correlation predictions because the difference in heat transfer between the two fluids is negligible in the convective evaporation regime.

If one is to use these correlations to predict the flow boiling characteristics of R11, the Jung (1988) correlation is recommended. However, neither correlation has successfully captured the fundamental mechanisms of horizontal flow boiling well enough to predict the heat transfer coefficient for R123.

CONCLUSIONS

The results show that the flow boiling characteristics of R11 and R123 are similar for the same reduced pressure, Reynolds number, and heat flux. This is especially true in the convective evaporation region, where the horizontal flow boiling heat transfer coefficient for the two fluids only differs by an average of ±2.1%. However, the heat transfer coefficient for R11 was consistently observed to be on average anywhere from 8.5% to 27% higher than that for R123 in the nucleate flow boiling region.

The influence of the Reynolds number and the heat flux on the heat transfer coefficient was investigated for fixed reduced pressure. An increase in the Reynolds number was shown, for most of the data for both fluids, to increase the heat transfer coefficient by an average of 14%, for thermodynamic qualities greater than from 15% to 30%, depending upon the heat transfer conditions. For qualities less than this, the Reynolds number had comparatively little effect, on average, a 6% increase in the measured heat transfer coefficient. In contrast, a 50% (35%) increase in the heat flux caused, on average, a 23% (18%) increase in the heat transfer coefficient for R11 (R123) in the nucleative region. The 50% (35%) increase in heat flux resulted in a modest average increase of 5.6% (3%) in the heat transfer coefficient for R11 (R123) in the convective region.

The measured heat transfer coefficients were compared to predictions from two open literature horizontal flow boiling correlations. Both correlations resulted in an over-prediction of the R11 heat transfer coefficients in the convective region by an average of 16%. The predictions with the Jung (1988) correlation were closer to the measured flow boiling coefficients for R11 (13% over-prediction) in the nucleative region than were the predictions with the Kandlikar (1987) correlation (22.5% over-prediction). Both correlations resulted in an over-prediction of the R123 heat transfer coefficient which was greater than that for R11. The heat transfer coefficient for R123 predicted with either correlations was an average of 36% and 25% greater than the measurements in the nucleative and the convective regions, respectively.

Only the predictions with the Jung (1988) correlation show a substantial difference in the heat transfer between the fluids in the nucleative regime, with the predicted heat transfer coefficient for R123 8% higher than that for R11. Predictions with the Kandlikar correlation show the R123 nucleative heat transfer coefficient to be 1% greater than that for R11. Contrary to this, the measurements showed an average of 8.5% to 27% difference in the heat transfer coefficient in the opposite direction, with the heat transfer coefficient for R11 higher than that for R123.

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