Evaporative Characteristics of R-134a, MP-39, and R-12 at Low Mass Fluxes

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ABSTRACT

Experimental heat transfer coefficients for R-134a, MP-39, and R-12 are reported herein. Tests were conducted using a single-tube evaporator test facility. The test section used was a 2.43 m (8-foot) long, 7.04 mm (0.277-in) inside diameter, horizontal, smooth copper tube. Heat was applied to the test section using electric resistance heaters. Test parameters were varied as follows: mass flux, 25-100 kg/m^2-s (19-76 klb$_m$/ft^2-hr); heat flux, 2-10 kW/m^2 (600-3200 Btu/hr-ft^2); quality, 10-90 percent; saturation temperature, -15 to 5 °C (5 to 41°F). The heat transfer coefficients for R-134a were higher than those of R-12 using equivalent mass flux and cooling capacity bases. The heat transfer coefficients for MP-39 were lower than those of R-12 using equivalent mass flux and cooling capacity bases. The correlations of Shah and Kandlikar were compared with the experimental heat transfer coefficients. An empirical correlation was developed for the wavy-stratified flow pattern occurring with the low mass fluxes. For correlating purposes, some high mass flux data with annular flow patterns, reported previously, are also included.

NOMENCLATURE

a	exponent in Eq. (2)
A	constant in Eq. (2)
A_s	surface area
b	exponent in Eq. (2)
B	constant in Eq. (2)
Bo	$=\dfrac{q}{Gi_{fg}}$, boiling number
D	diameter
F	$=\dfrac{\alpha_{cb}}{\alpha_l}$, two-phase convection multiplier
Fr	$=\dfrac{G^2}{\rho^2 gD}$, Froude number
g	gravitational acceleration
G	mass flux
i_{fg}	latent heat of vaporization
K	$=\dfrac{\Delta x i_{fg}}{Lg}$, Pierre boiling number
L	tube length
M	molecular weight
n	exponent in Eq. (5), (7)
P_r	$=\dfrac{P}{P_{crit}}$, reduced pressure
Pr	$=\dfrac{\mu c_p}{k}$, Prandtl number
q	heat flux
R	reduction parameter in Eq. (7)
Re_l	$=\dfrac{GD(1-x)}{\mu_l}$, liquid Reynolds number
Re_{lo}	$=\dfrac{GD}{\mu_l}$, total liquid Reynolds number

S suppression factor

T_s surface temperature

T_b bulk fluid temperature at saturation

x vapor quality

X_{tt} $=\left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}$, Lockhart-Martinelli parameter

X_{tt}' $=\left(\frac{1-x}{x}\right)^{0.9}\Omega$, modified Lockhart-Martinelli parameter

Greek symbols

α heat transfer coefficient

Ω $=\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} = 0.551P_r^{0.492}$, transport property ratio parameter

ρ density

μ viscosity

Δ change in

Subscripts

b bulk fluid

cb convective boiling

crit critical

l liquid phase

lo liquid only

nb nucleate boiling

s surface

TP two-phase

v vapor

INTRODUCTION

The prediction of heat transfer coefficients and pressure drops during forced-convective evaporation is important for a wide range of processes, which include applications relating to the power, chemical process, and refrigeration industries. The first phase of extensive study of these heat transfer coefficients took place during the 1950s and 1960s. Appropriate mechanisms for forced-convective evaporation were identified and dimensionless parameters representing these mechanisms were used in developing correlations. However, magnitudes of resulting terms and transitions between flow regimes were not extensively examined. Later, generalized correlations were developed in the 1970s and 1980s from compiled databases of these earlier studies. Improvements in predicting heat transfer coefficients were made, but the magnitudes of each mechanism and transitions between these mechanisms were still not clearly defined.

Recently, because of the phaseout of chlorofluorocarbons outlined by the Montreal Protocol and tightening energy efficiency standards imposed by the federal government, the refrigeration industry, with the aid of chemical companies, has identified alternative, ozone-safe refrigerants whose thermal and heat transfer properties have not been determined. Experimental research to measure heat transfer coefficients of these new chemicals has begun, including development of a test apparatus at the Air Conditioning and Refrigeration Center. In addition to collecting baseline data for the refrigerant industry, another opportunity has arisen to extend the existing databases regarding evaporative heat transfer coefficients.

Most of the correlations developed in the literature are for annular flow situations inside horizontal or vertical tubes. Smaller data bases have been generated regarding wavy-stratified flows, which is the predominant flow pattern found in household refrigerators and supermarket display case evaporators. These systems require much smaller heat loads per tube than stationary air conditioning systems, and therefore need much smaller mass flow rates.

A study was conducted to examine heat transfer coefficients of alternative refrigerants for R-12 for low mass flux conditions. Experimental data was taken with refrigerants R-134a and MP-39, alternative refrigerant replacements for R-12, and were compared with baseline tests using R-12. Heat transfer coefficients were compared with correlations of Kandlikar [1990] and Shah [1982]. An empirical correlation developed for annular flow data using an asymptotic form was modified to account for the decrease in heat transfer due to the wavy-stratified flow pattern in the low mass flux cases.

LITERATURE REVIEW

Initial methods were developed for convective boiling heat transfer along a liquid-vapor interface. These methods were based on the premise that the mechanism of heat transfer in forced convection was similar to single-phase forced convection [Chaddock and Noerager, 1967]. By applying the Reynolds analogy that relates the energy transport mechanism to momentum transport in convection, it was shown that the ratio between the two-phase flow and the single-phase liquid heat transfer coefficients could be exclusively correlated by the Lockhart-Martinelli parameter, X_{tt}. The form of this correlation was

$$\frac{\alpha_{TP}}{\alpha_l} = F(X_{tt}) \tag{1}$$

It was also observed that nucleate boiling could occur simultaneously with evaporation along an extensive liquid-vapor interface. Pierre[1956] developed a model for the average heat transfer coefficient over a large quality change using Reynolds number and a modified boiling number as follows:

$$\alpha_{TP} = AK^a + B\,Re_{lo}^b \tag{2}$$

To add nucleate boiling effects into the form of Eq. (1), Shrock and Grossman [1962] introduced the boiling number, Bo, as follows:

$$\frac{\alpha_{TP}}{\alpha_l} = F(Bo, X_{tt}) \qquad (3)$$

Rohsenow [1952] first proposed an additive model of the nucleate and convective boiling heat transfer coefficients, and Chen [1966] utilized this based on the superposition of heat transfer coefficients as follows:

$$\alpha_{TP} = S\alpha_{nb} + F\alpha_l \qquad (4)$$

where S is a suppression factor for nucleate boiling and F is the two-phase convective multiplier for heat transfer as defined in the right hand side of Eq. (1). Recently, Jung and Radermacher [1989] developed a correlation using this basic form.

Shah [1976] later utilized a different form by developing a generalized correlation using several data bases from the literature which broke flow boiling into distinct regions: a nucleate boiling dominated regime, a bubble suppression regime where nucleate boiling and convective boiling are important, and a convective boiling dominated regime. The correlation was in graphical form and was evaluated by taking the larger of the three heat transfer coefficients calculated for the nucleate boiling, bubble suppression, and convective boiling regions. The nucleate boiling term was characterized by the boiling number while the convective boiling term was characterized by the convection number, Co, which is a modified form of the Lockhart-Martinelli parameter. Shah [1982] later put this correlation in equation form.

Recognizing that the boiling number could not accurately model the nucleate boiling term alone, Kandlikar [1990] developed a similar correlation that multiplied the boiling number by a fluid specific term which accounted for the different nucleate boiling effects that occurred from fluid to fluid. This can be described as a "greater of the two" (nucleate and convective boiling dominated) method.

The generalized correlations proposed by Shah and Kandlikar also account for the loss of tube wetting in horizontal flow for low mass fluxes. For wavy-stratified flows, part of the tube wall remains dry and there is a loss of convective heat transfer area, thereby decreasing the heat transfer coefficient. In these correlations, this loss of tube wetting is accounted for by introducing a Froude number based correction factor to the convective term.

Connecting the Chen and Shah methods is the form Kutateladze [1961] proposed which is an asymptotic, power-type addition model for the nucleate and convective boiling components:

$$\alpha_{TP} = \left[\alpha_{nb}^{n} + \alpha_{cb}^{n}\right]^{1/n} \tag{5}$$

For n equal to 1, the form becomes that of Chen. As n approaches infinity, the form becomes the "greater of the two" method similar to that of Shah. This method was used by Churchill [1972] to correlate the transition between forced convection and natural convection heat transfer. Bergles and Rohsenow [1964], Steiner and Taborek [1992], and Liu and Winterton [1988] have developed correlations utilizing this form. Choices of n have ranged from 2 to 3.

EXPERIMENTAL APPARATUS AND METHODS

Refrigerant Loop

Figure 1 is a schematic of the experimental test facility. A variable-speed gear pump was used to circulate the refrigerant around the loop, eliminating the need for a compressor and an expansion device found in vapor-compression refrigeration systems. This allowed testing capabilities in a pure refrigerant environment without the influence of the oil from the compressor. A 6-kW preheater was used to control inlet qualities to the test section. The preheater was a 3-pass, horizontal, serpentine, copper coil wrapped with strip heaters to provide heat input. Heat input rates to the preheater were set using variacs which were manually controlled and were measured with a watt transducer. Heat was removed from the refrigerant using two parallel condensers cooled by an ethylene glycol-water mixture from a chiller. Flow rates were measured using a Coriolis-type mass flow meter. Absolute pressure was measured at several locations using strain-gage pressure transducers. Fluid temperatures were measured using calibrated, type T thermocouples.

Test section

The horizontal test section was a 2.43 m (8 foot) long copper tube with a 7.04 mm (0.277 in) inside diameter. Heat input was provided to the test section using variac-controlled, surface-wrapped heaters. The heat rate was measured by a watt transducer. To reduce heat gain from the environment, the test section was covered with 50 mm (2 in) of foam insulation. Sixteen type T thermocouples were soldered in grooves cut longitudinally along the test section to measure the surface temperatures. Figure 2 shows the location of these thermocouples. Bulk fluid temperatures were measured at the inlet ant outlet of the test section using type T thermocouple probes. Differential pressure across the test section was measured using a pressure transducer across the pressure taps shown in Fig. 2. Sight glasses with the same inside diameter as the test section were installed at the inlet and outlet of the test section. Flow visualization was enhanced using a strobe light.

Data Collection

Prior to running, the system loop was evacuated and then charged with refrigerant. Testing showed that the saturation pressure based on thermocouple readings matched the measured pressure within 2 kPa (0.3 psi). Thermocouples were calibrated in a thermostatic bath over the testing temperature range and pressure transducers were calibrated using a dead weight tester. The estimated uncertainty of the thermocouples was ±0.15°C (±0.27 °F) and the estimated uncertainty of the pressure transducers was ±2 kPa (0.3 psi). Uncertainty of the differential pressure transducers was estimated at ±0.15 kPa (0.02 psid). Watt transducers were factory calibrated within ±10 W and this number has been verified during single-phase energy balance testing. Heat gains to the test section were also determined through single-phase energy balance testing.

Data collection was performed using a micro-computer and data acquisition system. The data acquisition system consisted of 5 terminal panels for 40 transducer connections, four data acquisition cards connected to the computer via Nubus slots, and data acquisition software. Testing was conducted at steady-state conditions, which were monitored and controlled by the above-mentioned system.

Parameters controlled during tests were mass flux, heat flux, inlet quality, and saturation temperature. Steady-state conditions, reached in approximately 15 minutes to two hours, were assumed when the time variation of saturation temperature was less than 0.1°C (0.18°F) for five minutes. The controlled parameters also had to be within the following range of target values: mass flux, ±5%; heat flux, ±5%; saturation temperature, ±0.5°C. Once steady-state conditions were achieved, the transducer signals were logged into a data acquisition output file. The channels were scanned once every second for a period of 60 seconds. The inlet pressure to the test section was then recorded at 100 Hz for 20 seconds. The data was then reduced using a spreadsheet macro. Refrigerant transport and thermodynamic properties used during data collection and reduction were obtained from McLinden et. al [1991].

<u>Experimental Results and Uncertainties</u>

Experimental heat transfer coefficients were determined by the convective law of cooling using the circumferentially averaged values of surface temperatures, the linearly interpolated values of bulk fluid temperature, the surface area of the test section, and the heat input rate to the test section as follows:

$$h = \frac{q / A_s}{\left(T_s - T_b\right)} \tag{6}$$

Since surface temperatures were measured inside grooves located along the external surface of the copper tube, the temperature drop across the tube wall needed to be considered. However, its value was determined to be negligibly small, and, therefore, it was disregarded. Axial heat conduction along the length of the tube was also neglected.

Heat gain from the environment was determined during single-phase energy balance testing.

Uncertainties for the experimental heat transfer coefficients were determined using the method of sequential perturbation as outlined by Moffat [1988]. Uncertainties in each of the independent variables used to calculate the heat transfer coefficient from Eq. (6) were estimated based on calibration and examination of system-sensor interaction errors, system disturbance errors, and other errors. Uncertainties ranged from ±3% to ±12.5% for low mass flux testing.

ANNULAR FLOW DATA

Tests were previously conducted for high mass fluxes and heat fluxes similar to those found in automotive air conditioning and stationary air conditioning evaporators. Initial results were given in Wattelet et. al [1992] and the complete set of data is summarized in Panek [1991]. The test section consisted 10.92 mm (0.43 in) inside diameter, 2.43 m (8 foot) long copper tube. Test parameters for the high mass flux testing are given in Table 1.

Table 1. High mass flux test parameters

Parameter	Value
Saturation temperature	-5 to 15°C (23 to 59 °F)
Mass flux	200-500 kg/m^2-s (152-380 klb$_m$/ft^2-hr)
Heat flux	5-30 kW/m^2 (1600 to 9600 Btu/hr-ft^2)
Quality	10-90 %

Refrigerants R-134a, R-12, MP-39 were used as test fluids. Flow patterns were determined by strobe-light enhanced visual observation through sight glasses at the inlet and outlet of the test section. For higher mass flux testing, the predominant flow pattern was annular flow.

Figure 3 shows the variation of heat transfer coefficient versus quality for annular flow tests with a fixed mass flux and varying heat flux for R-134a, R-12, and MP-39. As can be noted, for low heat fluxes the heat transfer coefficient increases with quality. Intense evaporation at the liquid-vapor interface diminishes the liquid film thickness, reducing the thermal resistance, which is associated with heat conduction across the film. Nucleate boiling appears to be largely suppressed for these low heat flux cases. As heat flux increases, the heat transfer coefficients increase in the lower quality region and eventually merge at higher qualities with the heat transfer coefficients for low heat flux cases. Nucleate boiling at these lower qualities enhances the heat transfer coefficient. At higher qualities, nucleate boiling is again largely suppressed due to significant surface cooling promoted by the thinning of the annular film.

R-134a and R-12, both single-component fluids, exhibit similar behavior qualitatively. For an equivalent, mass flux, heat flux, saturation temperature, and quality, R-134a had approximately a 25% higher heat transfer coefficient on average than R-12 for annular flow tests. On an equivalent cooling capacity basis, R-134a had approximately a 2% higher heat transfer coefficient on average than R-12 for annular flow tests.

MP-39, a 52% / 33% /15% near azeotropic mixture of R-22 / R-124 / R-152a, has similar convective heat transfer properties to R-134a. For low heat flux tests shown in Fig. 3, the heat transfer coefficients for MP-39 are roughly the same as R-134a. However, its nucleate boiling characteristics are significantly diminished compared with R-134a and R-12. For higher heat flux testing where nucleate boiling effects are more important, R-134a has a higher heat transfer coefficient than MP-39. Overall, MP-39 has a 10% higher heat transfer coefficient on average compared with R-12 for annular flow tests. As will be noted in the next section, this is quite different than the comparisons between MP-39 and R-12 for low mass flux tests with wavy-stratified flow patterns.

STRATIFIED FLOW DATA

Test parameters for the low mass flux testing are given in Table 2. Refrigerants R-134a, R-12, MP-39 were used as test fluids. For low mass flux testing, the predominant flow pattern was wavy-stratified flow.

Table 2. Low mass flux test parameters

Parameter	Value
Saturation temperature	-15 to 5°C (5 to 41 °F)
Mass flux	25-100 kg/m^2-s (19-76 klb$_m$/ft^2-hr)
Heat flux	2-10 kW/m^2 (600 to 3200 Btu/hr-ft^2)
Quality	10-90 %

For wavy-stratified tests, there is no major effect of quality on the circumferentially averaged heat transfer coefficients, as can be seen in Fig. 4 for R-134a, R-12, and MP-39. However, as the heat flux increases, the heat transfer coefficient also increases. Compared with the results in the annular flow regime, convective boiling is diminished while nucleate boiling does not appear to be suppressed at higher qualities or lower heat fluxes. The decrease in convective boiling can be attributed to the reduction in available surface area for convective boiling, a decrease in turbulence due to the decreased Reynolds numbers for the low mass fluxes, and a decrease in slip ratio between the vapor and liquid streams.

R-134a again has a higher heat transfer coefficient than R-12 based on equivalent mass flux and cooling capacity comparisons for wavy-stratified tests. However, MP-39 shows a significant decrease in heat transfer coefficient for low mass flux cases. Figure 5 is a comparison of average heat transfer coefficients over the quality range of 20-90 percent for R-134a, R-12, and MP-39. For high mass fluxes, MP-39 has comparable heat transfer coefficients to R-134a. As mass flux is decreased, the heat transfer coefficients for MP-39 fall below R-12 by a significant amount. For the wavy-stratified cases, MP-39 has a 20% lower heat transfer coefficient on average compared with R-12. In addition to the decrease in nucleate boiling due to concentration gradients in the liquid, the lower slip ratios and turbulence in the wavy-stratified flows also seem to decrease the mixing in the vapor stream and concentration gradients may also be set up in the vapor.

CORRELATION COMPARISON

Two heat transfer correlations were selected from the literature to compare with the experimental values of heat transfer coefficient. The correlations of Shah [1982] and Kandlikar [1990] were selected because of their use of R-12 as a test fluid and the presence of a Froude number dependent term to account for wavy-stratified flow. The Shah correlation was the first generalized correlation and evaluates equations for the nucleate boiling, bubble suppression, and convective boiling dominated regimes and takes the largest of the three values. The Kandlikar correlation uses the greater of the nucleate boiling and convective boiling dominated forms. It differs from the Shah correlation in its use of boiling number for both the nucleate boiling and convective boiling regimes. It also uses a fluid specific term to account for variations of the nucleate boiling component of each form.

Figures 6, and 7 are plots of the predicted heat transfer coefficients for R-12 and R-134a from the correlations of Shah and Kandlikar, respectively, versus experimental heat transfer coefficients for R-12 and R-134a for annular and wavy-stratified flow. The mean deviation of the three correlations are given in Table 3 for both wavy-stratified and annular flow tests. The Kandlikar correlation does a slightly better job predicting the data than the Shah correlation.

Table 3. Comparison of mean deviation between the various correlations and the present experimental data for R-134a and R-12

Correlation	R-134a Annular	R-134a Wavy	R-12 Annular	R-12 Wavy
Kandlikar	9.9%	12.2%	9.7%	14.0%
Shah	11.8%	20.1%	16.4%	16.9%
Wattelet et. al	7.6%	6.6%	6.8%	7.1%

Figure 8 is a plot of the predicted heat transfer coefficients of MP-39 from the correlation of Shah with the experimental heat transfer coefficients of MP-39. Because of the reduction of both nucleate boiling and convective boiling heat transfer for the zeotrope, the unmodified Shah correlation severely overestimates by more than 50% the experimental heat transfer coefficients in the wavy-stratified flow regime. However, for annular flows, the convective correlating form does an adequate job of predicting the MP-39 heat transfer coefficients. As was discussed earlier, nucleate boiling effects are dramatically reduced for zeotropic mixtures. To properly correlate zeotropic mixtures, modifications will need to be made to the existing correlations in the literature, modifying the nucleate boiling term for annular flow and both the nucleate boiling and convective boiling terms for wavy-stratified flow. Although not discussed in this paper, Kandlikar proposed an initial modification to his generalized correlation for exactly this purpose for binary mixtures [1991]. These modifications to the Kandlikar correlation among others will become even more important as many zeotropic mixtures are currently the leading candidates for replacement of R-22.

MODEL SELECTION AND COMPARISON

The superposition model, the "greater of the two" model, and the asymptotic model were all discussed earlier in the literature review. After extensive evaluation of these forms, the asymptotic model was chosen to be the best form to correlate the experimental data. Correlations from nucleate pool boiling can be used for the nucleate boiling term while a convective form similar to the proposed form of Chen [1966] can be evaluated experimentally and used for the convective boiling term.

The main feature of this form is the "built in" suppression of the weaker component. Table 4 shows an example of this with n equal to two for the asymptotic form. For a large convective component and a small nucleate boiling component, the total two-phase heat transfer coefficient is made up almost entirely of the convective boiling component. For a mixed situation where both nucleate boiling and convective boiling both occur, the total two-phase heat transfer coefficient is made up of a combination of the two components. For a nucleate boiling dominated situation, the total two-phase heat transfer coefficient is made up almost entirely by the nucleate boiling component.

Table 4. Asymptotic form examples

h (nucleate boiling)	h (conv. boiling)	h (two-phase)
5000	1000	5036
3000	3000	3959
1000	5000	5036

Using annular flow data for R-134a and R-12 discussed above, an asymptotic correlation was developed using the following equations:

$$\alpha_{TP} = \left[\alpha_{nb}^{n} + \alpha_{cb}^{n}\right]^{1/n} \quad n = 2.5 \qquad (7)$$

$$\alpha_{nb} = 55M^{-0.5}q^{0.67}P_r^{0.12}[-\log P_r]^{-0.55}$$
$$\alpha_{cb} = F\alpha_l R$$
$$F = 1 + 1.925X_{tt}^{-0.83}$$
$$\alpha_l = 0.023\frac{k_l}{D}Re_l^{0.8}Pr_l^{0.4}$$
$$Re_l = \frac{GD(1-x)}{\mu_l} \qquad Pr_l = \frac{\mu_l c_{p_l}}{k_l}$$
$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\Omega$$
$$\Omega = 0.551P_r^{0.492}$$
$$R = 1.32Fr_l^{0.2} \quad \text{if } Fr_l < 0.25$$
$$R = 1 \quad \text{if } Fr_l \geq 0.25$$
$$Fr_l = \frac{G^2}{\rho_l^2 gD}$$

To account for the decease in convective heat transfer due to loss in convective boiling surface area and a loss of turbulence for lower Reynolds number flows, a Froude dependence has been added to the convective term. Compared with the Shah and Kandlikar correlations, this term begins to take effect at a much higher Reynolds number. This offsets the overestimation of the single-phase liquid heat transfer coefficient through use of the Dittus-Boelter correlation [McAdams, 1942] for tests with Reynolds numbers below 10,000. Kandlikar [1991] and others have recently suggested using the Gnielinski correlation [1976] for test conditions with Reynolds numbers below 10,000. However, the Gnielinski correlation only works for Reynolds numbers above 4,000 and is not suggested for extrapolation below this value because of the presence of a (Re-1000) term in the correlation. Many practical uses of refrigerants inside horizontal tubes, such as in household refrigerator evaporators, have Reynolds numbers below 4,000. Because the form of the Dittus-Boelter correlation is more tractable to modification, the Dittus-Boelter correlation was selected for use in the convective boiling term of Eq. (7).

Several nucleate pool boiling correlations have been developed over the past several decades. Initial correlations suggested by Chen [1966], such as the Forster-

Zuber correlation [1955] for nucleate pool boiling, are difficult to calculate and require knowledge of data such as surface temperatures and surface tension, which are not always available. This lack of surface tension data may become even more important for correlating heat transfer coefficients for zeotropic mixtures. Two recent pool boiling correlations have been developed that are more accurate than some of the original correlations and are easier to evaluate. The Cooper correlation [1984] is based on reduced pressure, heat flux, and molecular weight and is of the same order of accuracy as the Forster-Zuber correlation, but is much easier to evaluate. The other recent correlation developed in the literature is that of Stephan and Abdelsalam[1980] , and is used in the Jung-Radermacher correlation [1989]. Insufficient data has been collected in the present study for nucleate boiling dominated regimes of flow boiling to adequately distinguish between the two. Because of model simplicity, need for surface tension in the Stephan-Adelsalam correlation, and similar accuracy in correlating the present flow boiling data, the Cooper correlation was selected for the nucleate boiling term in Eq. (7).

A modified form of the convective term in the Chen correlation was selected. Kenning and Cooper [1989] have shown this to be the appropriate form for this term. However, the Chen correlation has been found to underestimate their data and others. The convective boiling dominated experimental data in this paper also is underestimated by the Chen correlation. The form for the two-phase multiplier, F, in Eq. (7) is approximately 10 to 30% higher than the Chen two-phase multiplier between qualities of 10 and 90 percent for refrigerants R-134a and R-12.

The value of n selected in the asymptotic model was 2.5. This values was determined by a regression analysis for values of n between 1 to 3.

Figure 9 is a comparison of the experimental values of heat transfer coefficient for annular and wavy-stratified flow data for R-134a and R-12 and the predicted values of Eq. (7). The mean deviation of the correlation for the experimental data is shown in Table 3. These results compare very favorably to those of the other correlations shown in Table 3 and should give much credibility to the asymptotic form used in Eq. (7). MP-39 heat transfer coefficient values, however, are also severely overestimated by Eq. (7), mainly due to the stronger suppression of nucleate boiling for the zeotropic mixture. Use of the convective boiling term only results in a mean deviation similar to that of Shah's correlation for the annular flow data. Further testing and analysis needs to be performed before corrections to the nucleate boiling and convective boiling terms of Eq. (7) can be made.

CONCLUSIONS

Experimental heat transfer coefficients for R-12, R-134a, and MP-39 were determined for both annular flow and wavy-stratified flow. For equivalent mass flux, heat flux, saturation temperature, and quality, the heat transfer coefficients for R-134a were on average 25% higher than those of R-12. MP-39 heat transfer coefficients for similar conditions were 10% higher on average than R-12 for annular flow tests and 20% lower on average than R-12 for wavy-stratified tests. For equivalent cooling capacities, R-134a had slightly higher heat transfer coefficients than R-12 and MP-39 had lower heat transfer coefficients than R-12 for both wavy-stratified and annular flows.

Heat transfer coefficient correlations from the literature predicted heat transfer coefficients for low mass flux conditions for R-134a and R-12 with mean deviations of under 20%. MP-39 experimental heat transfer coefficients were overestimated by all correlations for wavy-stratified flows. Significant reductions in both nucleate and convective boiling due to concentration gradients in the liquid and vapor streams and the lack of turbulent mixing in the wavy-stratified flow contribute to this degradation from the predicted values using the weighted averages of liquid and vapor properties.

A new heat transfer coefficient correlation was developed using an asymptotic form. The Cooper correlation was selected for the nucleate boiling term. A two-phase multiplier, F, was developed using a modified Lockhart-Martinelli parameter for the convective term. For wavy-stratified flow, a Froude number correction was added to the convective boiling term to account for the decrease in available convective heat transfer area and the decrease in turbulence as predicted by the Dittus-Boelter single-phase heat transfer correlation. This correlation predicted both the annular and wavy-stratified flow data well.

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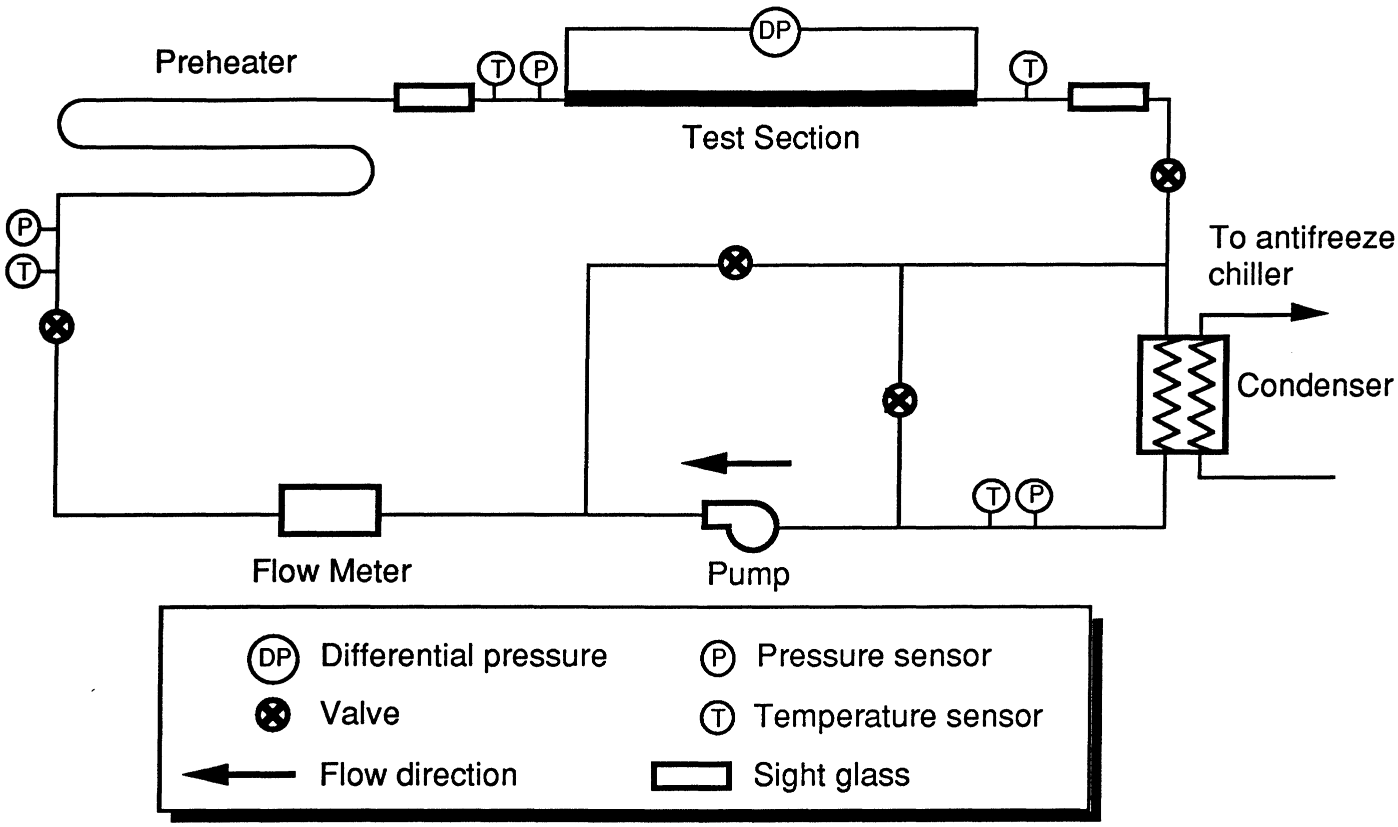


Figure 1. Refrigerant Loop

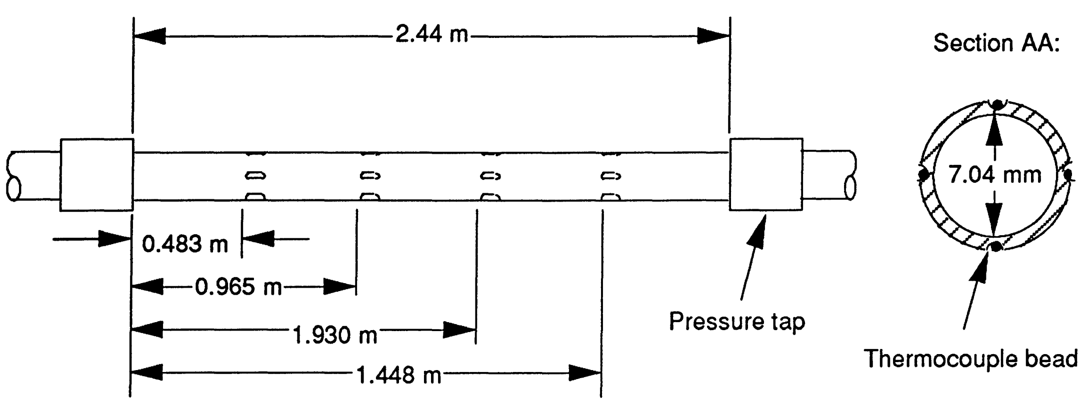
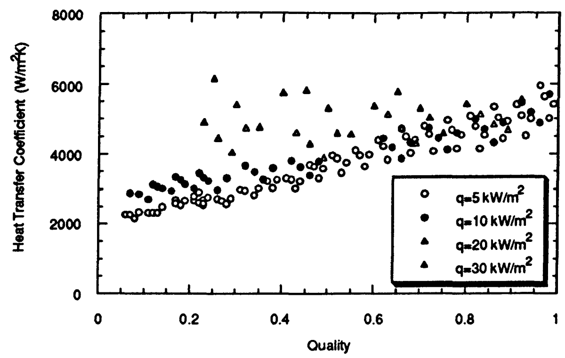
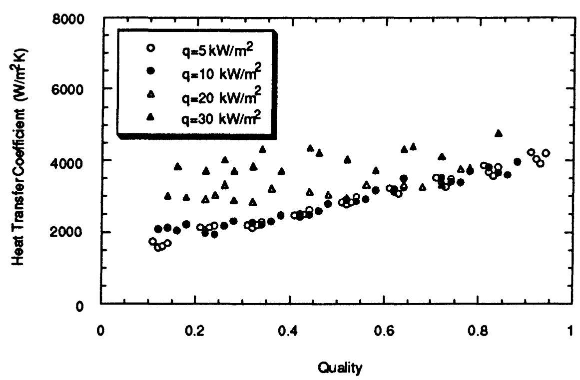


Figure 2. Evaporation Test Section



(a)



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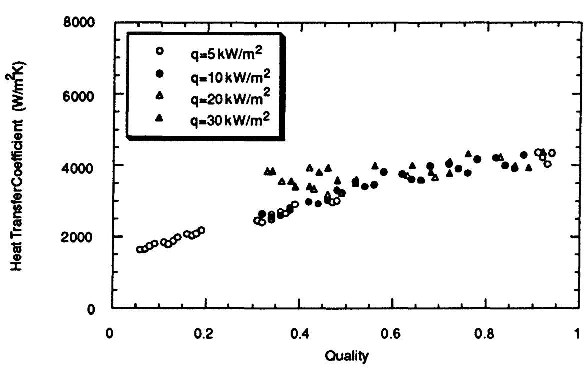
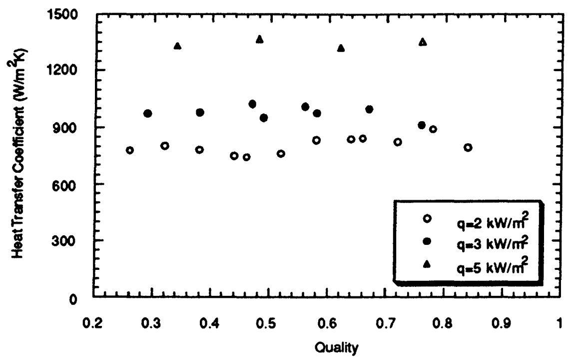
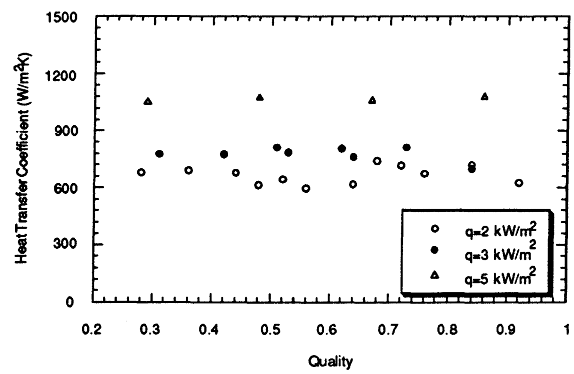


(c)

Figure 3. Heat transfer coefficient versus quality during annular flow for (a) R-134a; (b) R-12; (c) MP-39. Mass flux: 300 kg/m^2-s; Saturation temperature: 5°C; Tube diameter: 10.92 mm.



(a)



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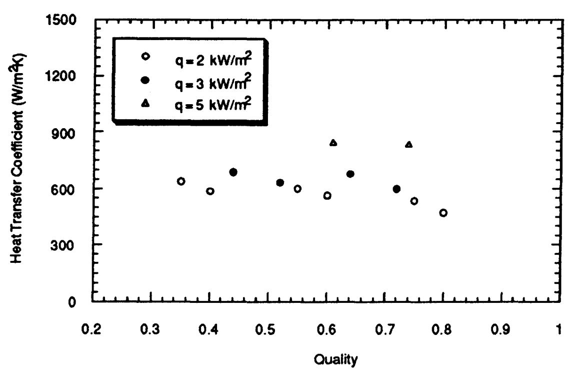


(c)

Figure 4. Heat transfer coefficient versus quality during wavy-stratified flow for (a) R-134a; (b) R-12; (c) MP-39. Mass flux: 50 kg/m^2-s; Saturation temperature: 5°C; Tube diameter: 10.92 mm.

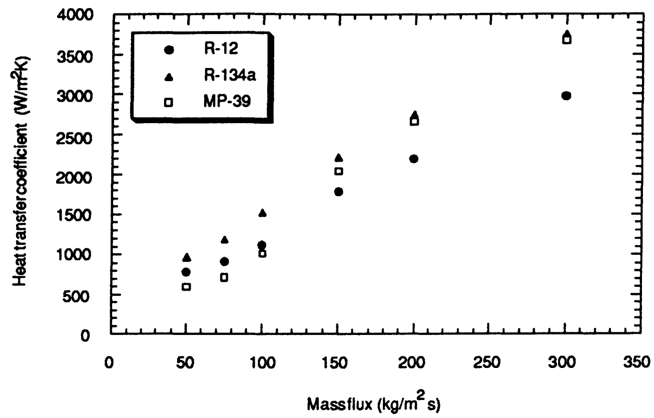


Figure 5. Heat transfer coefficient versus mass flux for R-134a, R-12, and MP-39. Heat flux: 5 kW/m^2; Saturation temperature: 5°C; Tube diameter, 7.04 mm.

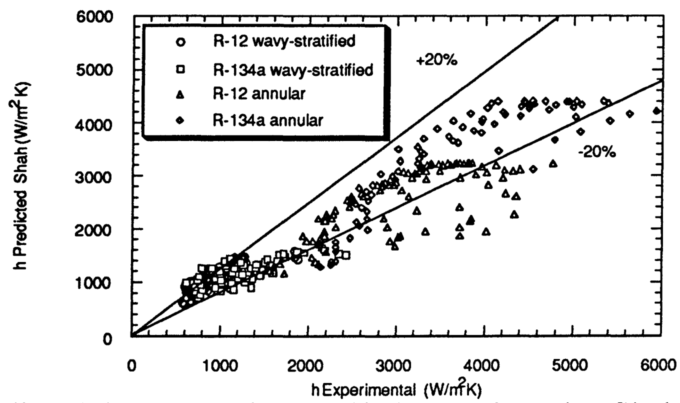


Figure 6. Predicted heat transfer coefficient using the Shah correlation versus experimental heat transfer coefficient for R-134a and R-12 during wavy-stratified and annular flows.

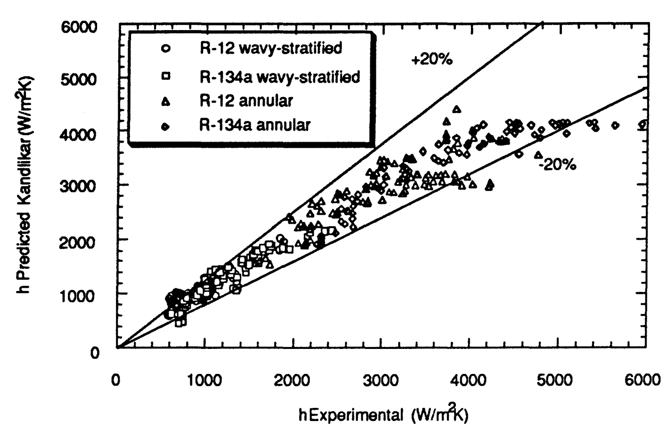
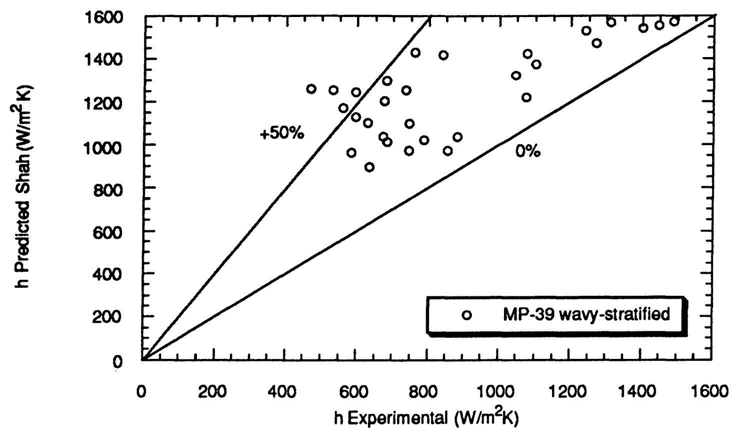
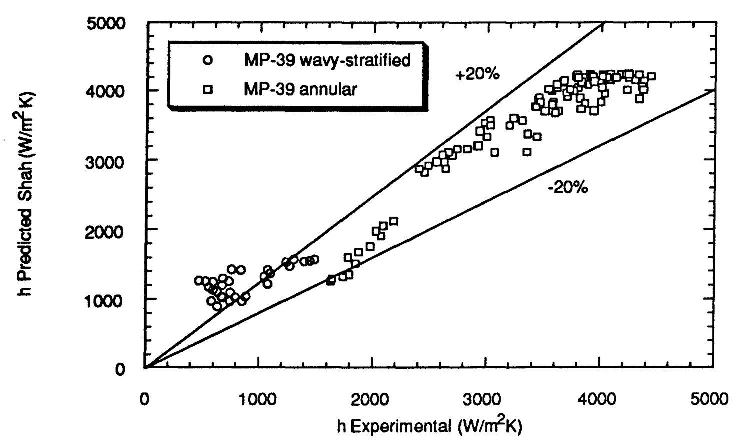


Figure 7. Predicted heat transfer coefficient using the Kandlikar correlation versus experimental heat transfer coefficient for R-134a and R-12 during wavy-stratified and annular flows.



(a)



(b)

Figure 8. Predicted heat transfer coefficient using the Shah correlation versus the experimental heat transfer coefficient for MP-39 during wavy-stratified and annular flows.

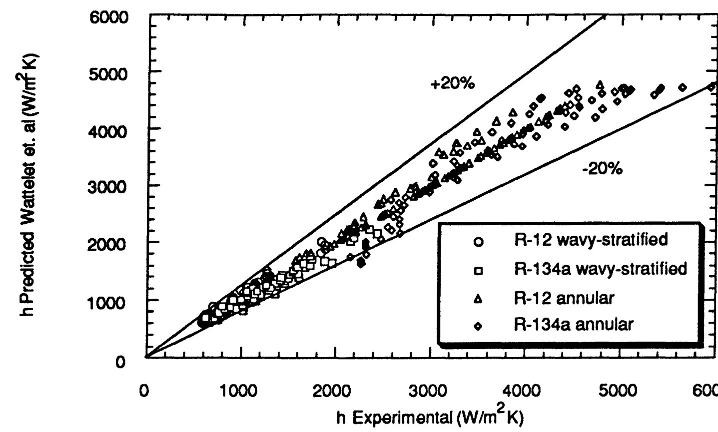


Figure 9. Predicted heat transfer coefficient using the Wattelet et. al correlation versus the experimental heat transfer coefficient for R-134a and R-12 during wavy-stratified and annular flows.