FLOW BOILING ENHANCEMENT OF R-22 AND A NONAZEOTROPIC MIXTURE OF R-143A AND R-124 USING PERFORATED FOILS

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ABSTRACT

This paper reports an investigation of the flow-boiling heat transfer from the inside of a smooth tube and a novel surface designed to improve nucleate boiling heat transfer. This surface consists of a perforated brass foil inserted into the smooth tube. Two fluids were investigated: pure R-22 and a nonazeotropic mixture of 75% R-143a and 25% R-124 (by mass). The experiment was conducted in an apparatus consisting of a variable-speed compressor having a range of 500 to 3,000 rpm and a counterflow concentric-tube heat exchanger. The refrigerant circulates inside the central tube and water circulates in the annulus.

For R-22, the perforated foil insert increased the heat transfer coefficient of the smooth tube, particularly at lower qualities and heat fluxes, indicating augmentation of nucleate boiling. For this particular nonazeotropic refrigerant mixture, however, the perforated foil insert did not increase the heat transfer coefficient, which is most likely due to nucleate boiling suppression in nonazeotropic refrigerant mixtures at the qualities observed in the tests.

INTRODUCTION

Nonazeotropic refrigerant mixtures (NARMs) are being investigated as replacements for the chlorofluorocarbon (CFC) refrigerants presently used in air conditioners and heat pumps. For NARMs in the wet (two-phase) region, the saturation temperature at constant pressure varies with respect to quality, and thus the evaporation process is not isothermal. This nonisothermal phase-change behavior can improve the heat exchanger effectiveness of an evaporator (Granryd and Conklin 1990) as well as the thermodynamic cycle efficiencies (McLinden and Radermacher 1987). However, other investigators (Jung et al. 1989; Ross et al. 1987) have measured heat transfer coefficients for binary NARMs undergoing evaporation in smooth tubes that were less than the heat transfer coefficients for either of the pure constituent refrigerants. Additional heat transfer area for NARM evaporators would thus be required. Hence, the effect of an enhanced surface on the NARM evaporating heat transfer coefficient was investigated here.

For this particular series of tests, a novel heat transfer surface—a perforated brass foil insert—was selected for investigation. The perforated brass foil heat transfer augmentation surface, protected by United States and European patents (Granryd and Palm 1988, 1990), has been used to augment heat transfer in pure refrigerants from the outside surface of a tube in pool boiling (Palm 1991b) and from the inside surface of a tube in flow boiling (Palm 1990). This surface has the capability of being used as a retrofit heat transfer enhancement device for existing evaporators. The actual increase in the heat transfer varies considerably with application, hole geometry, and pure refrigerant (Palm 1991a). Palm reports a doubling in heat transfer coefficient for flow boiling with pure R-22 in these investigations. Hence, the foils were inserted into the smooth tube evaporator to test the possible enhancement performance with a NARM.

The non-CFC refrigerants R-143a and R-124 were chosen for testing based on a previous research effort (Vineyard et al. 1989), which screened refrigerant pairs using such factors as boiling point, stability, ozone depletion potential, and coefficient of performance (COP) to determine suitable candidates for residential heat pump operation. Based on a previous investigation (Conklin and Vineyard 1990), a mixture of 75% R-143a and 25% R-124 (by mass) was selected for evaporation at various mass flow rates. R-22 was also evaporated for comparison of the surface performance. Since water-to-refrigerant heat exchangers were used in the experimental apparatus, the inlet water temperatures were selected to yield refrigerant saturation pressures equivalent to those determined for air-to-refrigerant heat exchangers in the U.S. Department of Energy standard rating conditions for heat pumps (Miller 1989).

EXPERIMENTAL APPARATUS

The heat transfer from enhanced surfaces during tube-side evaporation of binary NARMs is being investigated using a highly instrumented apparatus, shown in Figure 1, that consists of the following components: a variable-speed compressor having a range of 500 to 3,000 rpm, which allows for variable heat exchanger loadings; a variable-orifice refrigerant metering device (needle valve); and two sets of counterflow concentric-tube heat exchangers having two different enhanced tube-side surfaces. The refrigerant was circulated inside the central tube and water was

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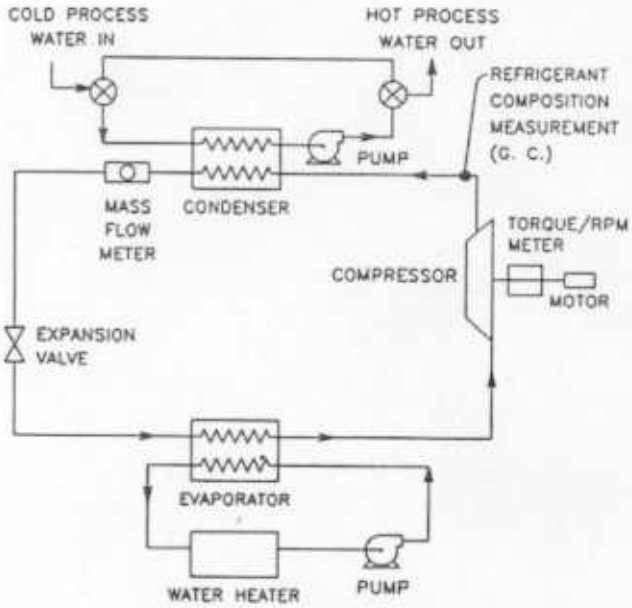

Figure 1 *NARM test loop.*

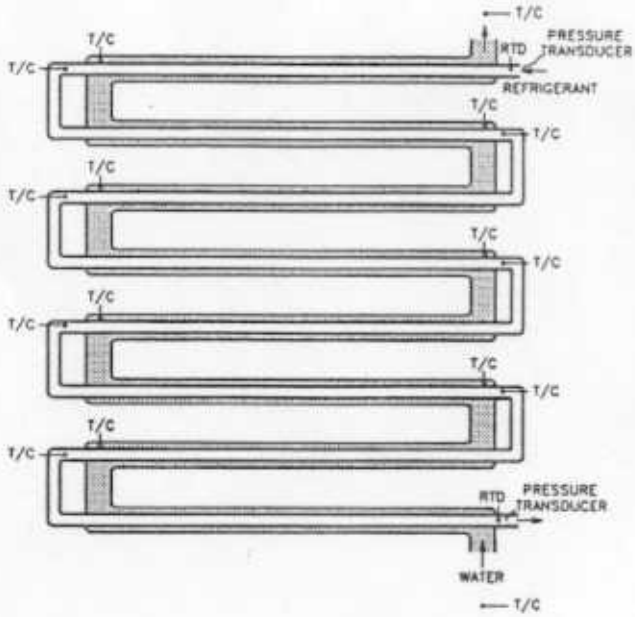

Figure 2 *Heat exchanger schematic.*

circulated in the annulus. A variable-speed compressor was chosen to provide for different heat loadings of the heat exchangers. A needle valve was chosen for flexibility in controlling refrigerant conditions; this flexibility is lacking in capillary tubes and thermal expansion valves.

The heat exchangers, with instrumentation as shown in Figure 2, consist of a central tube and an outer tube forming an annulus. There are eight horizontal passes where the outer surfaces of the water annulus and the refrigerant bends are insulated. The smooth surface tube is copper and has an outer diameter of 15.875 mm (0.625 in.) and an inner diameter of 13.85 mm (0.545 in.). Each heated horizontal section is 2.75 m (9.02 ft) in length. The brass foils—approximately foil number 2 of Palm (1990)—were 0.05 mm (0.002 in.) thick. The foils were perforated by a simple method with the resultant approximate hole shape shown in Figure 3. The holes were evenly spaced, with 50 holes in an area of one square centimeter (i.e., a hole density of 50 cm^{-2} [323 $in.^{-2}$]). The extruded material from the perforation operation has a height of approximately 0.23 mm (0.009 in.). The side of the foil with the extruded material was placed against the smooth tube wall because that placement resulted in the greatest enhancement for the tests of Palm (1990). The foils are held against the tube wall by the stiffness of the brass. The holes were oriented with the longest dimension in the flow direction.

The first tube at the refrigerant inlet shown in Figure 2 is defined as tube-pass number 1, with tube-pass numbers incrementing in the direction of the refrigerant flow. Because the perforated foils were designed to improve nucleate boiling, they were inserted only in passes 2 through 5, where the quality would be low with the highest likelihood of liquid. The foils were not inserted into tube-pass number 1 because of accessibility constraints due to the location of the evaporator in the experimental apparatus.

TESTING PROCEDURE

The output of instrumentation such as mass flow meters, pressure transducers, and the thermocouples and resistance temperature devices (RTDs) shown in Figures 1 and 2 was recorded on a minicomputer. In addition, a dedicated gas chromatograph, located at the compressor exit, was used to monitor the composition of the circulating charge in real time. Data were taken every 15 seconds and averaged over a 15-minute period during steady-state operation. All pressure transducers, flowmeters, thermocouples, and RTDs were calibrated prior to testing.

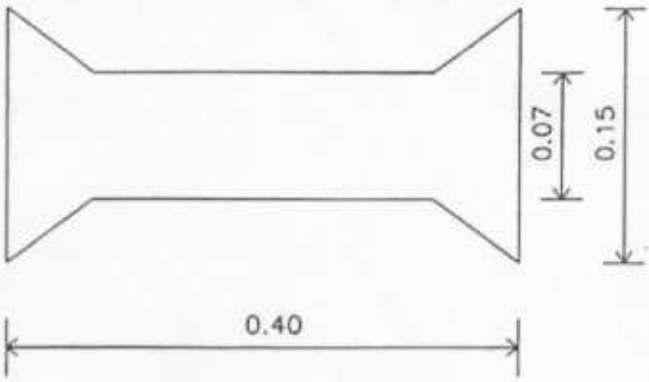

Figure 3 *Approximate hole geometry (units in millimeters).*

Water flow rates to the heat exchangers are pneumatically controlled to allow for rapid changes in test conditions. Inlet temperatures to the condenser are maintained with a mixing valve that introduces cool process water while bleeding off warmer water that has passed through the condenser. Inlet water temperatures to the evaporator are controlled by a bank of resistance heaters in the closed-loop system.

Test conditions were based on the U.S. Department of Energy standard rating conditions for an air-source heat pump. Inlet water temperatures were adjusted to yield refrigerant saturation pressures equivalent to those determined from previous testing with an air-source system (Miller 1989). The heat rates for the refrigerant side and the water side were determined from measured inlet and outlet temperatures and measured flow rates and then checked for a steady-state heat balance. The electrical power measurements for the closed-loop system were also used to check for a heat balance in steady-state operation. The needed refrigerant thermodynamic properties were determined from algorithms presented by Morrison and McLinden (1986) that implement the Carnahan-Starling-DeSantis equation of state. During the testing, minimal subcooling at the condenser exit and minimal superheating at the evaporator exit were maintained to ensure two-phase conditions in the heat exchangers.

Baseline tests were performed with R-22 at different compressor speeds with subsequent heat exchanger loads as a basis for comparison of the performance of the smooth tube and perforated foil inserts. Both R-22 and the R-143a/R-124 mixture had an alkylbenzene oil present for lubrication of the compressor. The concentration of oil, which was not monitored during the tests, was on the order of 1%. The effect of oil on the heat transfer coefficient or pressure drop was not investigated.

HEAT TRANSFER ANALYSIS

Since one objective is to measure the thermal performance of the NARMs in the evaporator, an evaporating heat transfer coefficient is computed from the measured temperatures and flows. First, a "standard" log mean temperature difference is computed from the measured evaporator inlet and outlet temperatures with the following equation:

$$T_{lmtd} = \frac{(T_{w,i} - T_{r,o}) - (T_{w,o} - T_{r,i})}{\ln\left(\dfrac{T_{w,i} - T_{r,o}}{T_{w,o} - T_{r,i}}\right)}, \tag{1}$$

where T represents a temperature. Subscript r represents the refrigerant, w represents the water, i represents the corresponding flow inlet, and o represents the corresponding flow outlet.

The total heat flux to the refrigerant during the evaporation process is then determined from the measured water temperatures and flow rate with the following equation:

$$q'' = \frac{\dot{m}_w c_w (T_{w,i} - T_{w,o})}{A_w}, \tag{2}$$

where A_w represents the heat transfer area of the outer surface of the refrigerant tube to the water. Heat gains from the environment were neglected because the outer annulus was insulated and the total heat rate transferred from the water was within 5% of the total heat rate transferred to the refrigerants for an acceptable test.

An overall heat transfer coefficient is then determined as follows:

$$U = \frac{q''}{T_{lmtd}}. \tag{3}$$

Axial heat losses along the copper tube were neglected in Equation 2. To justify this assumption, the ratio of the convection conductance ($U \cdot A_w$) and the axial thermal conductance ($k_{cu} \cdot A_{cu}/l$) was computed. This ratio was on the order of 3×10^4, indicating a much greater resistance to the flow of heat along the tube wall compared to the flow of heat convected to the fluids through the tube wall. Thus, axial heat transfer was neglected for computing the overall heat transfer coefficient.

The heat transfer coefficient to the water was obtained from the following empirical relationship:

$$h_w = \frac{k_w}{d_{hyd}}\, a\, \mathrm{Re}_w^{0.8}\, \mathrm{Pr}_w^{0.4}, \tag{4}$$

where Pr_w represents the Prandtl number of the water and Re_w represents the Reynolds number of the flow based on the hydraulic diameter of the annulus. The coefficient a, equal to 0.0164 for the smooth tube annulus used in this study, and the exponent of the Reynolds number were obtained from a series of tests with R-22 using a modified Wilson plot method that Shah (1990) attributes to Briggs and Young (1969). Viscosity variations were neglected because the difference in temperature between the tube wall and the water was less than 10 C° (18 F°).

The heat transfer coefficient of the refrigerant is then computed by the following equation:

$$h_r = \frac{1}{\dfrac{A_r}{A_w}\left(\dfrac{1}{U} - \dfrac{1}{h_w}\right)}, \tag{5}$$

where A_r represents the heat transfer area of the inner tube surface to the refrigerant. The dimension used for the flow area and heat transfer area of the inserts was the smooth tube's inner diameter, i.e., the unenhanced diameter. Because the tubes are copper and the foils are brass, the thermal resistance of the materials was neglected due to the high thermal conductivity of these materials. Also, because the tubes were new, no fouling factor was assigned. No contact thermal resistance of the brass foils was also

assigned, since any contact resistance would be included in the overall performance of the foils.

Where needed, the refrigerant thermodynamic properties were determined from algorithms presented by Morrison and McLinden (1986) that implement the Carnahan-Starling-DeSantis equation of state. The transport properties of pure R-22 were computed with the equations of Kandlikar et al. (1975), and the transport properties of the R-143a/R-124 mixture were computed with the equations and methods presented by Jung and Radermacher (1991).

HEAT TRANSFER RESULTS AND DISCUSSION

The heat transfer coefficient computed with Equation 5 has an approximate uncertainty of 20%. This uncertainty was computed from a propagation of error analysis (Box et al. 1978) that included the uncertainties of 0.28 C° (0.5 F°) for the thermocouples and 3% of measured flow for the flowmeter. Because of instrumentation system limitations, the heat transfer coefficient of the individual tube passes was not computed; instead, the heat transfer coefficient for tube passes 2-3 and passes 4-5 is reported. Thus, the heat transfer coefficient reported here is a local or a mean value over two tube lengths and not a mean value for the entire evaporation length. The resultant heat transfer coefficients for the series of tests with pure R-22 and the 75% R-143a/25% R-124 NARM in the smooth tubes are compared with the heat transfer coefficients of the perforated brass foil inserts.

R-22 Results

The local heat transfer coefficient in the smooth tube for passes 2-3 and 4-5 with R-22 is compared in Figure 4 with the predictions of two published correlations for flow boiling in a smooth tube. The two correlations are those of Shah (1982) and Kandlikar (1990). As shown in Figure 4, both correlations generally overpredict to within 25% of the measured heat transfer coefficient, with the one exception of the lowest reported value of heat transfer coefficient, which occurs at the lowest mass flow rate. More data points, however, are needed to draw a definite conclusion about the agreement.

The Shah (1982) correlation appears to yield the best agreement with the measured heat transfer coefficient, albeit consistently higher in value. The Kandlikar (1990) correlation predicts a higher value of heat transfer coefficient than that predicted by the Shah (1982) correlation. The reason that the measured heat transfer coefficient was always less than that predicted by either correlation is not obvious, although the alkylbenzene lubricant is suspect. The measurements are deemed to be of sufficient accuracy because the intent of this investigation was to compare the performance of the perforated foil inserts with that of a smooth tube.

The measured heat transfer coefficients for the smooth tube and the perforated foil inserts are plotted in Figure 5

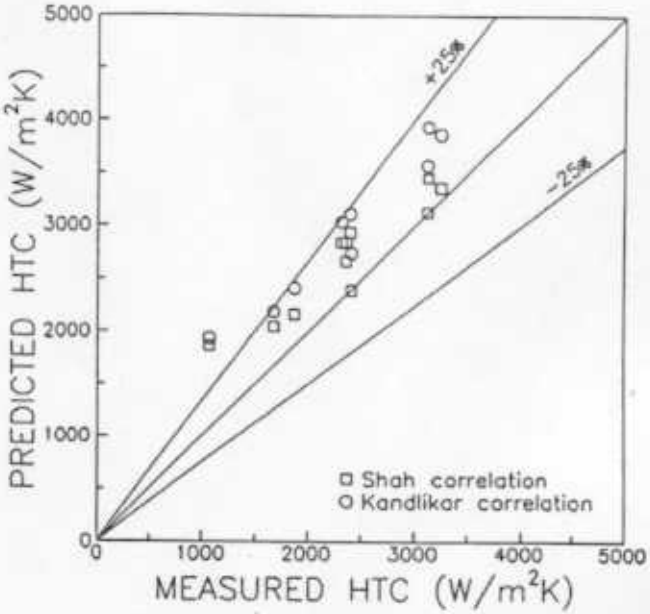

Figure 4 *Smooth-tube heat transfer coefficient for R-22.*

as a function of mass flux. There are two data points for each mass flux value, the lower of each pair representing the local heat transfer coefficient for tube passes 2-3 and the higher representing tube passes 4-5. In general, the heat flux and quality were lower for tube passes 2-3. The quality in passes 2-3 will always be less because the refrigerant inlet is at pass 1. The heat flux for passes 2-3 is generally less because the temperature difference between the water and refrigerant is less at this position in the heat exchanger; the heat flux is not controlled in this experimental apparatus, but the total heat load is. The points for the perforated foils are much closer together than

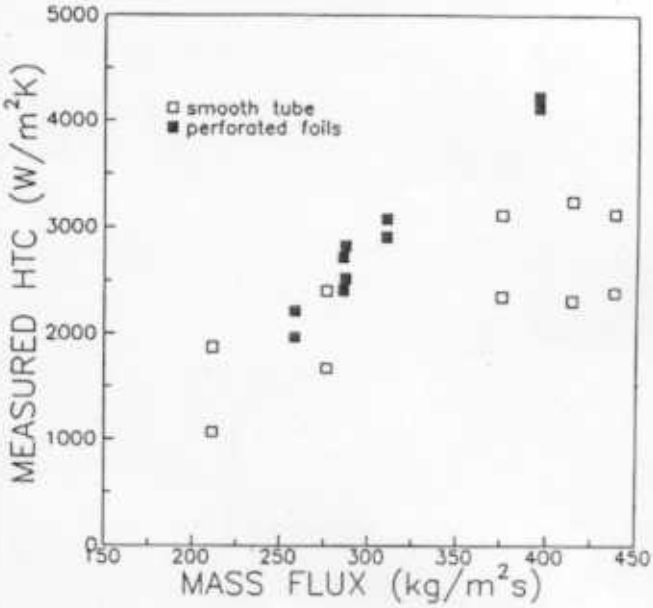

Figure 5 *Heat transfer coefficient vs. mass flux for R-22.*

those representing the smooth tube, indicating a more noticeable enhancement of heat transfer at lower qualities and heat fluxes. Palm (1991a) states that nucleate boiling is suppressed in flowing fluids due to an increase in the temperature gradient at the wall, which results in less available superheat to form bubbles. Figure 5 shows that the foils delay the suppression of nucleate boiling at the higher mass fluxes, believed to be due to the foils' isolating the liquid immediately adjacent to the wall from the bulk fluid velocity.

The measured heat transfer coefficients for the smooth tube and the perforated foil inserts are plotted in Figure 6 as a function of heat flux. The heat flux reported here is referenced to the inner radius of the tube. Here, also, there are two points for each value of mass flux, where the leftmost of each pair of points represents the local heat transfer coefficient for tube passes 2-3. From inspection of Figure 6, the enhancement in heat transfer due to the perforated foils is more at the lower heat fluxes experienced on the inserts in tube passes 2-3, where the quality is lower. The enhancement effect due to the perforated foils in tube passes 4-5, which occurs at higher qualities, is small for the heat flux levels tested here, with one exception of the highest heat flux. A decrease in nucleate boiling should be expected as the quality increases and, hence, any nucleate boiling enhancement technique can be expected to decrease in effectiveness also as the quality increases. There is a fair amount of scatter in the data, and additional data are needed for consideration.

The measured heat transfer coefficients for the smooth tube and the perforated foil inserts are plotted in Figure 7 as a function of the average local quality. Here also are two points for each value of mass flux, where the leftmost of each pair of points represents the local heat transfer coefficient of tube passes 2-3. From inspection of Figure 7, the enhancement effect due to the foils is more noticeable at the lower qualities experienced in tube passes 2-3 than the higher qualities experienced in tube passes 4-5, where the enhancement effect is scarcely noticeable.

75% R-143a/25% R-124 Results

The local heat transfer coefficient in the smooth tube for passes 2-3 and 4-5 with the NARM is compared in Figure 8 with the predictions of two published correlations for flow boiling in a smooth tube (Shah 1982; Kandlikar 1990). As shown in Figure 8, both correlations generally overpredict the measured heat transfer coefficient by approximately 25% for the Kandlikar correlation and approximately 20% for the Shah correlation. This indicates that the measured heat transfer coefficient is less than might be expected for a pure fluid having the same thermophysical properties of the NARM. This result is consistent with other investigations of flow boiling evaporation of binary NARMs (Jung et al. 1989; Ross et al. 1987) in that a degradation in heat transfer coefficient was observed, albeit for different nonazeotropic refrigerant mixtures. This degradation may have three components (Jung et al. 1989): variation of physical properties due to mixing, mass transfer resistance between the two components, and the loss of wall superheat due to preferential evaporation of the higher volatility component near the wall. The mechanism for the heat transfer degradation in NARMs is actively undergoing research.

The computation of the heat transfer coefficient from the measured temperatures and flows as described earlier assumes a constant specific heat of the refrigerant in the derivation of the log mean temperature difference, Equation 1. This equation, however, may introduce an error when applied to some NARMs. Because the phase-change

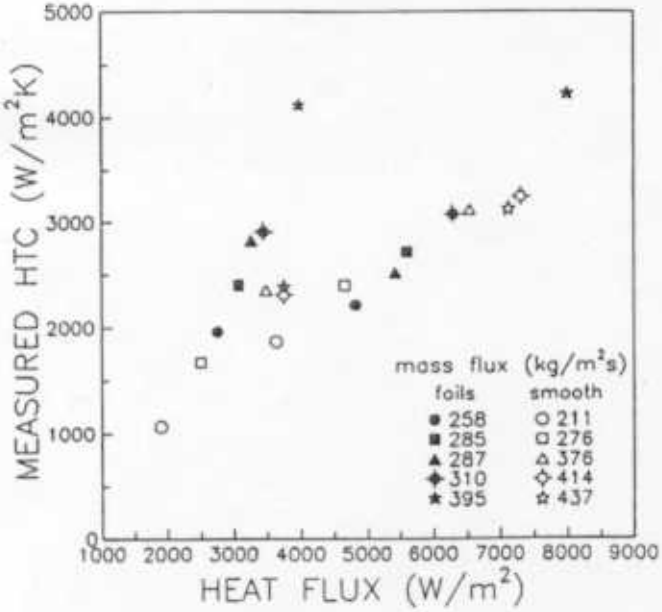

Figure 6 *Heat transfer coefficient vs. heat flux for R-22.*

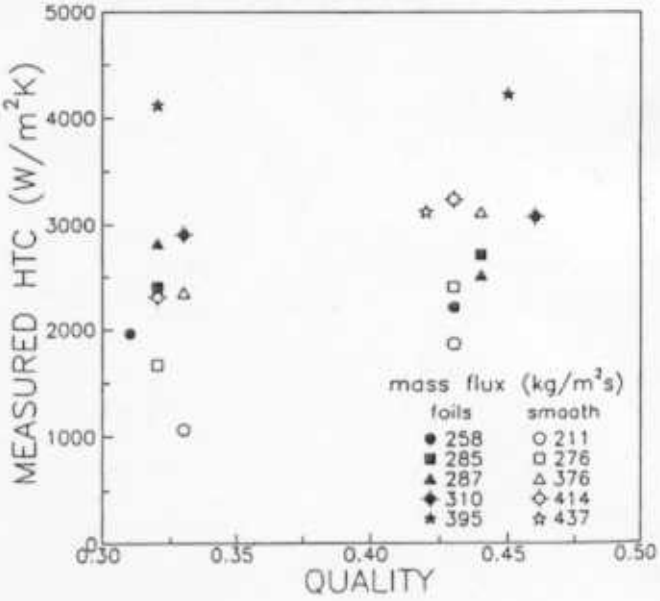

Figure 7 *Heat transfer coefficient vs. quality for R-22.*

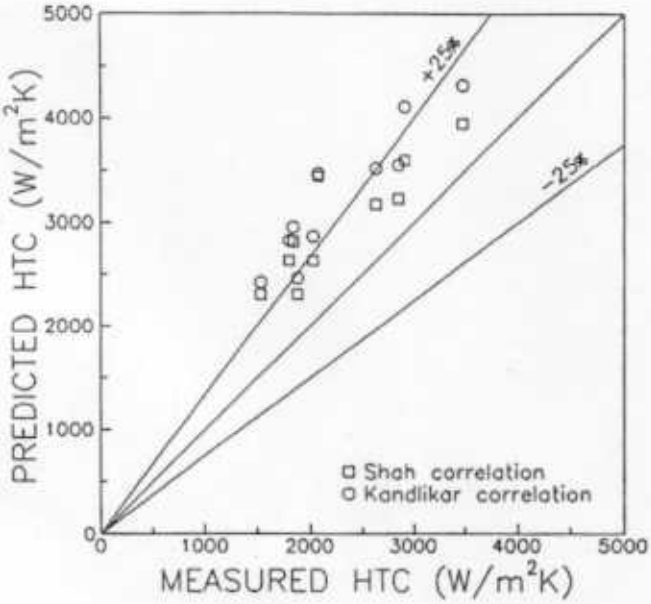

Figure 8 *Smooth-tube heat transfer coefficient for R-143a/R-124.*

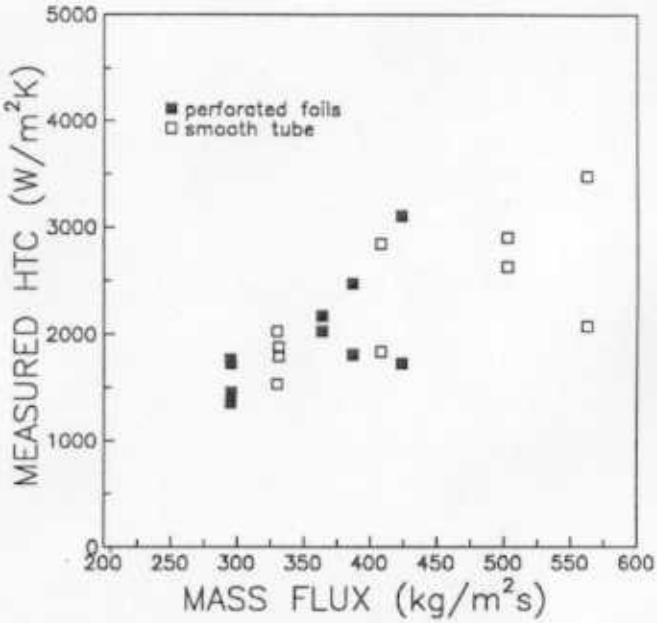

Figure 9 *Heat transfer coefficient vs. mass flux for R-143a/R-124.*

process of NARMs occurs over a finite temperature difference, a specific heat can be defined for the two-phase region. As described by Granryd and Conklin (1990), this two-phase specific heat can vary as a function of quality—and does so for this 75% R-143a and 25% R-124 (by mass) mixture—affecting the thermal performance of a heat exchanger. The refrigerant quality, however, varies only on the order of 10% over the length of tube passes 2-3 and 4-5. Hence, the error committed by assuming a constant specific heat over the tube pass length should be small and is neglected by the use of Equation 1. Work to quantify this error is in progress. This error should not affect the basic intent of this investigation, which, again, was to compare the performance of the perforated foil inserts with that of a smooth tube for a specific refrigerant (R-22) and a specific nonazeotropic refrigerant mixture (75% R-143a/25% R-124, by mass).

The measured heat transfer coefficient for the subject NARM of 75% R-143a/25%R-124 is plotted in Figure 9 as a function of mass flux, in Figure 10 as a function of heat flux, and in Figure 11 as a function of quality. From inspection of Figure 9, there seems to be no enhancement of heat transfer as a function of mass flux—the points representing the perforated foils are neither closer together nor higher in value than those representing the smooth tube. This is a first indication that the perforated foils may have little or no enhancement effect in this NARM. Since the foils were designed to enhance nucleate boiling, either nucleate boiling may be suppressed at these experimental conditions or the perforated foils have no effect with respect to mass flux for nucleate boiling heat transfer in this particular binary NARM.

The perforated foils apparently degrade the heat transfer coefficient for this particular NARM, as shown in Figure 10. The heat flux is referenced to the inner radius of the tube. For the lower range of heat fluxes, the perforated foils show a slight decrease in heat transfer coefficient. Palm (1991a) noticed a decrease in enhancement at the higher qualities of his tests; the foils acted as an additional heat transfer resistance where there apparently

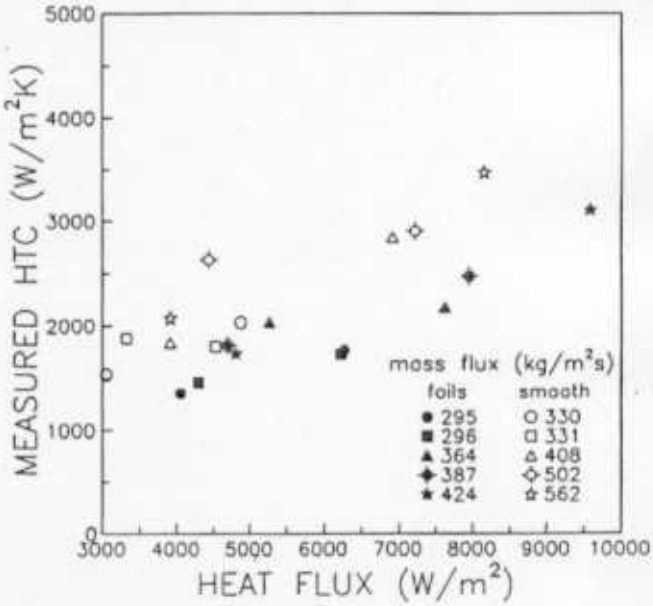

Figure 10 *Heat transfer coefficient vs. heat flux for R-143a/R-124.*

was no nucleate boiling to enhance. For the higher range of heat fluxes shown in Figure 10, the perforated foils significantly degrade the heat transfer coefficient. Palm (1991a) also measured a significant decrease in heat transfer coefficient for heat fluxes above 10 kW/m^2 (3,200 Btu/h·ft^2) for flow boiling and speculated that dryout of the surface under the foil was occurring. Similar degradation in heat transfer coefficient is indicated here for these tests but at a lower value of approximately 6 kW/m^2 (1,900 Btu/h·ft^2). Dryout of the heated surface under the foils is the most likely reason.

Figure 11 also shows a slight degradation in heat transfer coefficient at the lower qualities and a significant degradation at the higher quality values. The lowest quality investigated here was 35% for the NARM. Jung et al. (1989) claim that nucleate boiling was suppressed in nonazeotropic mixtures of R-22 and R-114 at qualities of 20% or less. Hence, there may have been no nucleate boiling to enhance with the perforated foil in this particular series of tests. The scatter, however, is noticeable in Figure 11, and more data are needed to make a definitive statement.

Thus, the nucleate boiling enhancement effect of the perforated foils observed in pure R-22 is not apparent in this particular NARM for the experimental conditions of this investigation.

PRESSURE DROP RESULTS

The irreversible pressure drops over the total length of the evaporator for these tests are reported in Figure 12. The experimental data were fitted by a quadratic polynomial obtained from a linear least-squares analysis. The experimental uncertainty of the pressure transducers is 0.69 kPa (0.1 psi). This irreversible pressure drop was calculated by taking the pressure difference measured by the pressure transducers at the inlet and outlet of the evaporator and subtracting the acceleration pressure drop due to the phase change. Thus, this irreversible pressure drop represents the loss due to friction and the loss in the tube bends at the end of each heated tube pass.

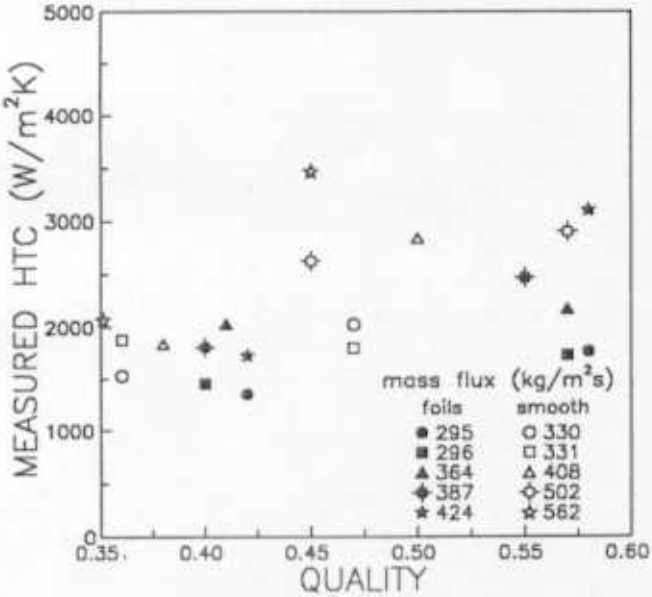

Figure 11 Heat transfer coefficient vs. quality for R-143a/R-124.

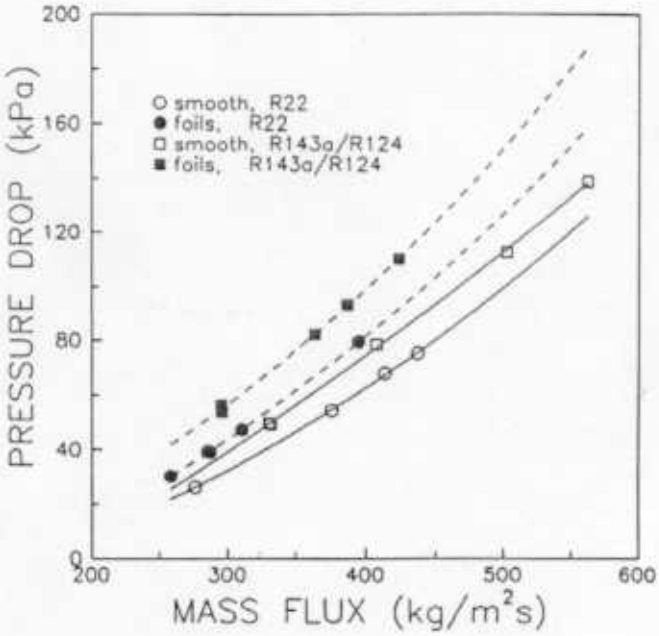

Figure 12 Irreversible pressure drop.

R-22 Results

The inlet pressure of the R-22 for the tests was approximately 750 kPa (109 psi), which is 15% of critical pressure. The irreversible pressure drops over the total length of the evaporator for both the completely smooth tube evaporator and the evaporator with half of the passes having the foil inserts are presented in Figure 12. From inspection of Figure 12, the presence of the perforated foil inserts in half of the tube passes resulted in an approximately 25% to 30% increase in irreversible pressure drop for the total evaporator length at a given mass flux.

75% R-143a/25% R-124 Results

The inlet pressure of the NARM for these tests was also approximately 750 kPa (109 psi), which is 17% of critical pressure. As for the pure R-22, the presence of the perforated foil inserts in half of the tube passes resulted in an approximately 25% to 30% increase in irreversible pressure drop over the total evaporator length at a given mass flux.

CONCLUSIONS AND RECOMMENDATIONS

For these tests with R-22, the perforated foil insert augments the flow-boiling heat transfer coefficient where nucleate boiling may be expected to occur, particularly at the lower qualities and heat fluxes, but also delays the suppression of nucleate boiling to higher mass fluxes than observed in the smooth tube. These results are consistent with the observations of Palm (1991a), and, thus, the nucleate boiling enhancement effect of the perforated brass foil insert as reported by Palm (1990) is corroborated for flow boiling of R-22. The enhancement effect observed here was not as high as Palm observed, but his tests were performed at lower mass fluxes and under constant, controlled wall heat flux conditions.

For these tests with the 75% R-143a/25% R-124 (by mass) nonazeotropic refrigerant mixture, there is no apparent enhancement effect of the perforated foil inserts for the range of flow parameters investigated, and, indeed, there is a significant degradation in heat transfer coefficient for most flow conditions. Because the experimental apparatus shown in Figure 1 has an accessibility constraint so that the foils cannot be easily inserted in tube-pass 1, the foils cannot be tested at the lowest qualities of the existing apparatus without extensive modifications. Also, because the refrigerant enters the evaporator at an approximate quality of 15% to 20% and the perforated foils showed no enhancement effect at a quality of 35%, the perforated foil insert may not affect the overall, total rate of heat transfer in the evaporator sufficiently to improve the efficiency of the complete cycle. Hence, no further testing of the perforated foils as nucleate heat transfer augmentation devices is planned for this particular NARM. Use of the perforated foil insert concept, however, to increase heat transfer in NARMs for other situations should not be dismissed without further testing.

These perforated foils may be used as a retrofit to existing smooth-tube evaporators to increase heat transfer but should only be used in regions where nucleate boiling might be expected to occur. Because Palm (1991a) reports different enhancement effects of the foils with different refrigerants, an optimum hole geometry and density may exist that is dependent on refrigerant thermophysical properties and flow conditions. Further study of the heat transfer enhancement due to the perforated foil inserts is recommended for pure refrigerants as well as other NARMs at low qualities. Particularly, it appears from this investigation that the penalty paid in increased pressure drop is relatively small.

ACKNOWLEDGMENTS

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NOMENCLATURE

A	=	area, heat transfer (m^2)
c	=	specific heat at constant pressure ($J \cdot kg^{-1} \cdot K^{-1}$)
d	=	diameter (m)
h	=	heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
k	=	thermal conductivity ($W \cdot m^{-1} \cdot K^{-1}$)
l	=	evaporator length (m)
$\dot{m}$	=	mass flow rate ($kg \cdot s^{-1}$)
q''	=	heat flux ($W \cdot m^{-2}$)
T	=	temperature (K)
U	=	overall heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)

Subscripts

cu	=	copper
hyd	=	hydraulic
i	=	inlet
$lmtd$	=	log mean temperature difference
o	=	outlet
r	=	refrigerant
w	=	water

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DISCUSSION

Akira Yabe, Senior Researcher, National Mechanical Engineering Laboratory, Tsukuba Science City, Japan: Evaporation of nonazeotropic mixtures is deteriorated by the accumulation of less evaporative components near the heat transfer surface. Would your perforated foil be effective for preventing the accumulation? From the experiments, the above important effect was not expected.

James C. Conklin: The perforated foil tends to isolate the fluid immediately adjacent to the wall from the bulk flow and thus probably will not prevent the accumulation of the less volatile component of the nonazeotropic binary mixture next to the wall.

Vijay Dhir, Professor, School of Engineering, University of California, Los Angeles: This may enhance heat transfer only under partial nucleate boiling. At high quality, it is probably annular flow and nucleate boiling is suppressed. The perceived enhancement is probably data scatter.

Conklin: I admit that there is some data scatter and I agree that the flow pattern is probably annular. But nucleate boiling is not precluded in annular flow, just more easily suppressed by bulk flow conditions. The perforated foil tends to isolate the fluid immediately adjacent to the wall from the flowing bulk fluid and thus mitigates nucleate boiling suppression due to flow conditions. While what little enhancement shown in Figure 7 at the higher range of tested qualities (42% to 46%) for R-22 may be due to data scatter, I believe the heat transfer enhancement is indeed real at the lower range of qualities (30% to 35%) shown in Figure 7 for the R-22 series of tests. Also, these observations and conclusions corroborate an independent investigation (Palm 1990).