AIM

Association for Information and Image Management

1100 Wayne Avenue, Suite 1100
Silver Spring, Maryland 20910
301/587-8202

Centimeter

Inches

MANUFACTURED TO AIM STANDARDS
BY APPLIED IMAGE, INC.

**ENHANCED SHELL-AND-TUBE HEAT EXCHANGERS
FOR THE POWER AND PROCESS INDUSTRIES**

Final Report

By A. E. Bergles, M. K. Jensen, E. F. Somerscales,
L. A. Curcio, Jr., R. R. Trewin

August 1994

Work Performed Under Contract No. DE-FC07-88ID12772

Prepared for the
U.S. Department of Energy
Under DOE Idaho Operations Office
Sponsored by the Office of the Assistant Secretary
for Energy Efficiency and Renewable Energy
Office of Industrial Technologies
Washington, D.C.

Prepared by
Rensselaer Polytechnic Institute
Troy, New York

REPORT DOCUMENTATION PAGE	1. REPORT NO.	2.	3. Recipient's Accession No.
4. Title and Subtitle ENHANCED SHELL-AND-TUBE HEAT EXCHANGERS FOR THE POWER AND PROCESS INDUSTRIES			5. Report Date December 1992
7. Author(s) A.E. Bergles, M.K. Jensen, E.F.C. Somerscales, L.A. Curcio, Jr., and R.R. Trewin			6.
9. Performing Organization Name and Address Rensselaer Polytechnic Institute Department of Mechanical Engineering, Aeronautical Engineering, and Mechanics Troy, New York 12180-3590			8. Performing Organization Rept. No. HTL-14
12. Sponsoring Organization Name and Address U.S. Department of Energy Idaho Operations Office, Idaho Falls, Idaho Office of Industrial Programs Washington, D.C.			10. Project/Task/Work Unit No.
			11. Contract(C) or Grant(G) No. (C) (G) DE FC07 88ID12772
15. Supplementary Notes			13. Type of Report & Period Covered Final
			14.
16. Abstract (Limit: 200 words) Single-tube pool boiling tests were performed with saturated pure refrigerants and binary mixtures of refrigerants. Generally, with the pure refrigerants, the High Flux surface performed better at the higher heat fluxes compared to the Turbo-B tube, and both enhanced surfaces performed significantly better than the smooth surface. The overall results of tests of R-11/R-113 mixtures was that the enhanced surfaces had a much smaller degradation in the heat transfer coefficient due to mixture effects compared to the smooth tube results; the largest degradation occurred at a mixture of 25% R-11/75% R-113. Under boiling conditions in a saturated aqueous solution of calcium sulfate, with a single tube, the effects of fouling were more pronounced at the higher heat fluxes, for all surfaces. Two staggered tube bundles were tested with tube pitch-diameter ratios of 1.17 and 1.50. For the pure refrigerant, tests on the smooth-tube bundle indicated that the effects on the heat transfer coefficient of varying mass flux, quality, and tube-bundle geometry were small, except at low heat fluxes. Neither enhanced surface showed any effect with changing mass flux or quality. The binary mixture bundle-boiling tests had results that were very similar to those obtained with the pure refrigerants. When boiling a refrigerant-oil mixture, all three surfaces (smooth, High Flux, and Turbo-B) experienced a degradation in its heat transfer coefficient; no surface studied was found to be immune or more vulnerable to the presence of oil than another surface.			
17. Document Analysis a. Descriptors heat exchangers enhancement fouling b. Identifiers/Open-Ended Terms shell-and-tube heat exchangers enhanced tubes power and process industries fouling due to water and oil in refrigerant c. COSATI Field/Group			
18. Availability Statement: Available from OSTI, P.O. Box 62 Oak Ridge, TN 37830 or NTIS 5285 Port Royal Road, Springfield, VA 22161		19. Security Class (This Report) unclassified	21. No. of Pages 13+99=112
		20. Security Class (This Page) unclassified	22. Price

ACKNOWLEDGEMENTS

This study received financial support from the U.S. Department of Energy under project No. FC07 88ID12772; Mr. Gary Peterson and Dr. Manohar Sohal were project monitors. Enhanced tubing and advice was provided by P. Thors of Wolverine Tube and E.G. Raji of U.O.P. Engineered Products. Undergraduate students involved with this work were Scott D. Coe, Timothy J. Riley, Sandra L. King, Mark E. Shorey, John K. McFarland, and Richard M. Hatley. Additional student support was given by Rensselaer Polytechnic Institute. All of this support is gratefully acknowledged.

Thanks also are due to the Rensselaer technical support staff without whose advice and help this project would not have been possible. Ms. Elizabeth Schoonmaker provided valuable assistance throughout the project with monthly, quarterly, and other reports.

CONTENTS

	Page
ABSTRACT	ii
ACKNOWLEDGEMENTS	iv
LIST OF TABLES	vii
LIST OF FIGURES	viii
NOMENCLATURE	xi
1. INTRODUCTION	1
2. POOL BOILING ON ENHANCED TUBES	3
2.1 Introduction.	3
2.2 Experimental Apparatus and Procedure	4
2.3 Smooth and Enhanced Tube Results with Pure Refrigerants	9
2.3.1 Pure R-113 Results	9
2.3.2 Pure R-11 Results.	15
2.4 Binary Mixture Results	16
2.5 Conclusions	19
3. SHELLSIDE BOILING ON ENHANCED TUBES IN CROSSFLOW	20
3.1 Introduction.	20
3.2 Experimental Apparatus and Procedure	22
3.3 Smooth Tube Reference Results with Pure Refrigerant	26
3.4 Enhanced Tube Results with Pure Refrigerant.	29
3.5 Binary Mixture Results	32
3.6 Conclusions	45
4. FOULING OF ENHANCED TUBES IN POOL BOILING	47
4.1 Introduction.	47
4.2 Experimental Apparatus and Procedure	52
4.3 Experimental Results	57
4.4 Physical Modelling of the Fouling Processes on Enhanced Heat Transfer Surfaces Under Pool Boiling Conditions.	63
4.4.1 Introduction.	63
4.4.2 The Model Surfaces and Experimental Procedure	64
4.4.3 Experimental Results	67
4.5 Conclusions.	70

	Page
POOL BOILING HEAT TRANSFER IN REFRIGERANT-OIL MIXTURES ON ENHANCED HEAT TRANSFER SURFACES	72
5.1 Introduction	72
5.2 Literature Survey	73
5.3 Experimental Approaches and Procedure	77
5.4 Results and Discussion	80
5.5 Conclusions	89
6. SUMMARY AND RECOMMENDATIONS	90
7. REFERENCES	95

LIST OF TABLES

	Page
Table 4.1 Precision of Measured Quantities in Fouling Experiments	57
Table 5.1 Enhancement Ratios ($\Delta T_p/\Delta T_e$) in R-114/oil Mixtures at a Heat . . . Flux of 30 kW/m ² (Wanniarachchi et al., 1986, 1987, Murphy 1987)	77
Table 5.2 Precision of Measured Quantities in Oil-Refrigerant Tests	81
Table 5.3 Ratio of Observed Boiling Coefficients (h'/h'') at a Heat Flux of 30 kW/m ²	82
Table 5.4 Observed Enhancement Ratios $\Delta T_p/\Delta T_e$ at a Heat Flux of 30 kW/m ²	86

LIST OF FIGURES

	Page
Figure 2.1 Pool boiling apparatus (Jung and Bergles 1989)	2
Figure 2.2 Test Section Schematic for the High Flux tube	5
Figure 2.3 Boiling curves for tubes with smooth, Turbo-B (medium opening), and High Flux surfaces at $P = 103$ kPa.	9
Figure 2.4 Axial cross-section of the Turbo-B tube (enlarged 9X)	12
Figure 2.5 Cross-section of the High Flux surface enlarged 100X (Gottzmann et al. 1973)	13
Figure 2.6 Comparison of Turbo-B tube pool boiling data in R-113 at $P = 103$ kPa	13
Figure 2.7 Saturated pool boiling curve for the smooth tube in pure R-11 and pure R-113 at $P = 103$ kPa	15
Figure 2.8 Saturated pool boiling curve for the smooth tube in mixtures of R-11 and R-113 at $P = 103$ kPa	17
Figure 2.9 Saturated pool boiling curve for the Turbo-B tube (medium opening) in mixtures of R-11 and R-113 at $P = 103$ kPa.	18
Figure 2.10 Saturated pool boiling curve for the High Flux tube (small porosity) in mixtures of R-11 at $P = 103$ kPa	18
Figure 3.1 Flow loop schematic	22
Figure 3.2 Cross-section of the test chamber	23
Figure 3.3 Effect of heat flux, pressure, tube row/quality, and mass flux on the heat transfer coefficient, and comparison with pool boiling results	27
Figure 3.4 Effect of the tube pitch-to-diameter ratio on the heat transfer coefficient	28
Figure 3.5 Heat transfer coefficients for a tube bundle using the Turbo-B surface.	30
Figure 3.6 Heat transfer coefficients for a tube bundle using the High Flux surface.	30
Figure 3.7 Effect of mass flux on the heat transfer coefficients for High Flux bundle.	31
Figure 3.8 Effect of heat flux on the heat transfer coefficient for all three surfaces in a mixture of 75.8% R-11 and $G = 49$ kg/m ² s	33

	Page
Figure 3.9 Effect of heat flux on the heat transfer coefficients for all three surfaces in a mixture of 75.8% R-11 and $G = 224 \text{ kg/m}^2\text{s}$	35
Figure 3.10 Effect of heat flux on the heat transfer coefficients for all three surfaces in a mixture of 23.8% R-11 and $P = 200 \text{ kPa}$	37
Figure 3.11 Effect of mixture concentration and pressure on the heat transfer coefficients for $q'' = 60 \text{ kW/m}^2$	39
Figure 3.12 Effect of mixture concentration and heat flux on the High Flux transfer coefficients.	40
Figure 3.13 Trends in heat transfer coefficients and the difference between the liquid and vapor mixture concentrations (Stephan, 1982).	41
Figure 3.14 A comparison between (a) the difference between the liquid and vapor concentrations of R-11 and (b) experimental trends in heat transfer coefficients at $q'' = 60 \text{ kW/m}^2$ and $P = 200 \text{ kPa}$	43
Figure 3.15 Comparison of experimental values for smooth tube heat transfer coefficients and predicted values using a Stephan-Körner type correlation	44
Figure 4.1 The variation of wall superheat with time of a plain smooth surface in a saturated aqueous solution of calcium sulfate at two heat fluxes.	58
Figure 4.2 The variation of wall superheat with time of High Flux large surface in a saturated aqueous solution of calcium sulfate at two heat fluxes.	58
Figure 4.3 The variation of wall superheat with time of a High Flux small surface in a saturated aqueous solution of calcium sulfate two heat fluxes.	59
Figure 4.4 The variation of a wall superheat with time of a Turbo-B very open surface in a saturated aqueous solution of calcium sulfate at two heat fluxes.	59
Figure 4.5 The variation of wall superheat with time of a Turbo-B very closed surface in a saturated aqueous solution of calcium sulfate at two heat fluxes.	60
Figure 4.6 Apparatus used in simulation of UOP High Flux surface. Note this shows the earliest version of this apparatus in which the sintered metal particles were simulated by ball bearings (not shown) of various sizes. In the latest version randomly distributed cylinders, providing a number of conductive paths from the lower heated surfaces, are mounted in the test chamber	65

	Page
Figure 4.7(a) Comparison of metallographic section of the Wolverine Turbo-B surface and the model surface.	65
Figure 4.7(b) Comparison of metallographic section of the High Flux surface and the model surface using ball bearings of of various sizes	66
Figure 4.8 Plots of the fouling deposit on model of Wolverine Turbo-B surface.	69
Figure 4.9 Temperature records made with the High Flux surface model The plot is the average reading of the eight thermocouples in the aluminum heater block, with the room temperature subtracted from the readings in order to smooth out temperature fluctuations introduced by the variation in the laboratory temperature	69
Figure 5.1 Pool boiling curves of a smooth surface in R-113 at 0%, 1%, 5% and 10% mass concentrations of 3GS oil	82
Figure 5.2 Pool boiling curves of UOP High Flux "Large" surface in R-113 at 0%, 1%, 5%, and 10% mass concentrations of 3GS oil	83
Figure 5.3 Pool boiling curves of UOP High Flux "Small" surface in R-113 at 0%, 1%, 5%, and 10% mass concentrations of 3GS oil	84
Figure 5.4 Pool boiling curves of Wolverine Turbo-B "Very Open" surface in R-113 at 0%, 1%, 5%, and 10% mass concentrations of 3GS oil	85
Figure 5.5 Pool boiling curves of Wolverine Turbo-B "Very Closed" surface in R-113 at 0%, 1%, 5%, and 10% mass concentrations of 3GS oil	85

NOMENCLATURE

B	temperature coefficient of solubility, $\text{kg}/(\text{m}^3\text{C})$
C,D	correlation constants used in Eqs. 3.3 and 3.4, respectively
C_1	constant in Eq. (2.4)
C_p	specific heat at constant pressure
d	tube diameter, m
e	enhancement ratio = $\Delta T_p/\Delta T_e$
G	mass flux based on minimum flow area, $\text{kg}/\text{m}^2\text{s}$
h	heat transfer coefficient, $\text{kW}/\text{m}^2\text{K}$
h'	heat transfer coefficient without oil, $\text{kW}/\text{m}^2\text{K}$
h''	heat transfer coefficient with oil, $\text{kW}/\text{m}^2\text{K}$
I	current, amps
i_{fg}	enthalpy of vaporization, J/kg
K_1	constant in Eq. (4.6), $\text{kg}/(\text{W-s})$
K_2	constant in Eq. (4.5) [$= (n+1) K_1(T_s-T_b)^n R_c^n$, $\text{m}^2\text{C}^2/(\text{W}^2\text{s})$]
k_f	thermal conductivity of fouling deposit, $\text{W}/(\text{mK})$
k_L	thermal conductivity of liquid, $\text{W}/(\text{mK})$
m_f	mass of fouling material per unit area, kg/m^2
$\dot{m}_g$	rate of growth of the fouling deposit per unit area, $\text{kg}/(\text{m}^2\text{s})$
$\dot{m}_r$	rate of removal of the fouling deposit per unit area, $\text{kg}/(\text{m}^2\text{s})$
n	empirical exponent in equations (4.3), (4.4) and (4.5)
p	tube pitch, m
P	pressure, kPa
q''	heat flux, kW/m^2
R_{total}	total thermal resistance at the fouled heat transfer surface ($= R_c + R_f$), $\text{m}^2\text{C}/\text{W}$

R_c	convective thermal resistance at the heat transfer surface, (= $R_c + R_f$), $m^2{}^{\circ}C/W$
R_f	fouling deposit thermal resistance, $m^2{}^{\circ}C/W$
S	solubility of scaling resistance, kg/m^3
T	temperature, $^{\circ}C$
T_b	bulk fluid temperature = boiling saturation temperature, $^{\circ}C$
$\bar{T}_b$	bulk fluid temperature averaged over an entire experimental run, $^{\circ}C$
T_s	surface temperature, $^{\circ}C$
T_{sat}	boiling saturation temperature at prevailing atmospheric pressure, $^{\circ}C$
$\bar{T}_w$	average value of $T_{w,i}$ ($i = 1,2,3,4$), $^{\circ}C$
ΔT	$\bar{T}_w - \bar{T}_b$, $^{\circ}C$ (for oil-refrigerant tests); $\bar{T}_w - T_{sat}$, $^{\circ}C$ (for $CaSO_4$ -water test)
$\Delta T'$	ΔT value without oil, $^{\circ}C$
$\Delta T''$	ΔT value with oil, $^{\circ}C$
ΔT_a	defined by Eq. (3.1), $^{\circ}C$
ΔT_e	ΔT value on enhanced surface, $^{\circ}C$
ΔT_{id}	defined in Eq. 3.2, $^{\circ}C$
ΔT_p	ΔT value on plain surface, $^{\circ}C$
ΔT_{sat}	wall superheat, $^{\circ}C$
x_{in}	inlet quality
x_{out}	outlet quality
$\tilde{x}'_{Ethanol}$	mole fraction of ethanol in liquid, mole/mole

$\tilde{x}_{\text{Ethanol}}''$	mole fraction of ethanol in vapor, mole/mole
$\tilde{Y}_{\text{R113}}, \tilde{Y}_{\text{R11}}'$	overall mole fraction of R-113 and R-11, respectively, in the fluid, kmol/kmol
$\tilde{Y}_{\text{R11}}''$	mole fraction in the vapor phase of R-11, kmol/kmol
y''	mass fraction in vapor phase (kg/kg)
y'	mass fraction in liquid phase (kg/kg)
κ	constant in Eq. (4.3), $\text{kg}^{1-m}\text{K}^n/(\text{m}^{2-3m-2n}\text{W}^n)$
κ_c	combined mass transfer coefficient and reaction rate constant, m/s
κ_r	parameter in Eq. (4.4)
μ_L	viscosity of liquid, N-s/m ²
ρ_f	density of fouling deposit, kg/m ³
ρ_L	density of liquid, kg/m ³
ρ_v	density of vapor, kg/m ³
σ_s	surface tension, kg/m
ω	precision of quantity indicated by subscript, units same as those of the quantity indicated by the subscript
θ	time, s; parameter defined in Eq. 3.3

Subscripts

average	average for a run
clean	measured with clean surface
fouled	measured under fouling conditions

1. INTRODUCTION

Shell-and-tube heat exchangers represent the most widespread and commonly used basic heat exchanger configuration in the process industries. This type of heat exchanger provides a comparatively large ratio of heat transfer area to volume and weight. The area is readily provided via the tubes which can have plain (smooth) surfaces or can have surface modifications to enhance their heat transfer behavior. The tubes can be rather easily cleaned and replaced if necessary.

Shell-and-tube heat exchangers are used for both single- and two-phase flow applications. Multi-tube bundles with boiling on the shellside are commonly used in the petroleum and chemical process industries to vaporize process fluids and to generate steam. If enhanced surfaces are used in such heat exchangers, the thermal result can be a greater heat duty for a given size heat exchanger, a smaller heat exchanger for a specified heat duty or, for a fixed size and heat duty, the mean temperature difference can be reduced. Substantial energy savings in many different forms can be realized from the higher performance or smaller heat exchanger.

While many enhanced surfaces have already been developed, can be economically manufactured and are in limited use in shell-and-tube heat exchangers, this investigation addressed several of the impediments to more widespread adoption of a specific type of heat transfer enhancement — structured surfaces — for use in vaporization/boiling applications. "Structured" surfaces result from modifications of a surface due to machining, forming, multilayering, or coating processes. Remarkable improvements in heat transfer coefficient compared to a smooth tube have been obtained in a variety of tests. However, these tests primarily have involved single enhanced tubes in pool boiling of a non-fouling, single-component liquid. These conditions are not typical of industrially relevant processes. The lack of data from more realistic operating conditions is a major problem. Hence, the objectives of this investigation were to:

- obtain data from a multi-tube bundle so that procedures for scale-up of single tube data to tube bundles could be developed;
- obtain data on structured surfaces when they are boiling in a liquid that can foul the surface;
- obtain data on structured surfaces for binary mixtures;
- and interpret the data relative to reference smooth tube data and to the physical processes governing the heat transfer behavior.

This investigation was primarily experimental in nature. Smooth surfaces and two enhanced surfaces were studied under a variety of operating conditions. The two enhanced surfaces were the Linde High Flux surface, which is obtained by sintering a porous metallic coating onto a smooth tube, and the Wolverine Turbo-B surface, which is obtained by cold-working a surface to form an array of interconnected reentrant cavities on the surface; figures of these surfaces are given in a subsequent chapter.

The project was divided into several distinct but related parts. Three topical reports resulted from this division and were submitted to the U.S. Department of Energy, Idaho Operations Department:

"Evaluation of Commercial Enhanced Tubes in Pool Boiling," C. Jung and A.E. Bergles, HTL-2, Rensselaer Polytechnic Institute, Troy, NY, March 1989.

"Pool Boiling of Enhanced Heat Transfer Surfaces in Refrigerant-Oil Mixtures and Aqueous Calcium Sulfate Solutions," L.A. Curcio, Jr. and E.F.C. Somerscales, HTL-5, Rensselaer Polytechnic Institute, Troy, NY, August 1991.

"Shellside Boiling on Horizontal Tube Bundles with Enhanced Surfaces," R.R. Trewin, A.E. Bergles, and M.K. Jensen, HTL-13, Rensselaer Polytechnic Institute, Troy, NY, August 1992.

The remainder of this Final Project Report summarizes the material contained in these three topical reports, as well as additional work not included in them.

2. POOL BOILING ON ENHANCED TUBES

2.1 Introduction

A number of enhancement technologies for pool boiling of a saturated liquid are reported in the literature (Bergles 1985). In particular, some of the most common enhanced surfaces are Wieland's Gewa-T, Hitachi's Thermoexcel-E, Wolverine's Turbo-B, and Linde's High Flux. Webb and Pais (1991) have compiled a literature survey of pool boiling on single tubes with enhanced surfaces that are produced by rolling the surface and by sintering small particles to the surface. They report that the enhanced surfaces perform significantly better than smooth tubes, often by an order of magnitude. Pool boiling heat transfer from horizontal tubes with enhanced surfaces was also reported by Yilmaz and Westwater (1981) who found that the heat transfer from enhanced surfaces can be an order of magnitude larger than that from smooth tubes. The increased heat transfer possible with enhanced tubes over smooth tubes makes them a strong candidate for use in tube bundles also.

A possible disadvantage of using enhanced surfaces is the delayed onset of nucleation resulting in a boiling curve hysteresis. Pool boiling from horizontal tubes with a porous metallic coating was investigated by Bergles and Chyu (1982) who first reported a large boiling curve hysteresis for this surface in wetting fluids. Possible explanations include the reduced tendency for vapor from the first nucleation sites to spread over the surface and activate other nucleation sites.

In this project pool boiling experiments were performed in order to determine the thermal performance of single, horizontal tubes with different surfaces boiling in a pool of saturated refrigerant. The results of the experiments were used as a reference for the tube bundle data and to validate the experimental tube bundle apparatus and procedure. Single tubes with plain and enhanced surfaces were tested in the pool boiling apparatus containing R-113, R-11, or mixtures of both. Heat flux and wall temperature superheat data were

collected in order to determine the effects of changing heat flux and various fluids on the heat transfer coefficients.

2.2 Experimental Apparatus and Procedure

The general pool boiling apparatus shown in Fig. 2.1 was designed similar to the one used by Bergles and Chyu (1982) and is described in detail by Jung and Bergles (1989). It consists of a Pyrex tank containing a water bath and a smaller Pyrex tank that contains the refrigerant and the test section. The electrically-heated cylindrical test section was mounted to a rigid, vertical supporting wall on one end and was insulated with a Plexiglas plug on the other end. Electrical connections supplying the power to the test section were inserted through the lid of the smaller Pyrex tank along with the thermocouple wires used to instrument the test section and measure the fluid temperature. A cooling coil was

Figure 2.1 Pool boiling apparatus (Jung and Bergles, 1989)

mounted just below the lid of the tank and had cold water circulating through it so that the refrigerant vapor condensed and refluxed back into the pool. In addition, an open reflux condenser, inserted through the lid, ensured that atmospheric pressure was maintained at the surface of the refrigerant without allowing refrigerant vapor to escape. The refrigerant in the smaller tank was insulated by surrounding it with constant temperature water in the larger tank. Water from the bath tank was recirculated through an external, temperature-controlled source that kept the water at 1 to 2 °C above the saturation temperature of the refrigerant.

The test section was manufactured out of a copper cylinder 120.7 mm long with an outside diameter of 19.1 mm and an inside diameter of 6.4 mm. The test section with the High Flux surface was mounted in the same way as in the tube bundle apparatus (see Fig. 2.2). Therefore, the sealing method which was required for the tube bundle experiments

Figure 2.2 Test section schematic for the High Flux tube

also was used in the pool boiling experiments even though sealing was not required. One end of the cylinder was machined down to create a neck 19.1 mm long, with an outside diameter of 15.1 mm, which protruded through the mounting wall. The neck was machined with a snap-ring groove and was inserted through the mounting plate. A snap ring secured the shoulder of the test section to the mounting plate so that they were kept perpendicular to each other. The mounting plate had an O-ring groove machined in it, and an O-ring was used to provide a seal between the test section and the mounting plate.

The test sections with the smooth and Turbo-B surfaces were mounted differently than what was used for the High Flux test sections. These tubes were attached to the mounting wall using two threaded screws. (See Jung and Bergles 1989.) The test sections were insulated from the mounting wall with a Buna-N gasket.

The test sections were instrumented with thermocouples inserted into four thermocouple wells as shown in Fig. 2.2(a). The thermocouple wells were manufactured by drilling four holes located 90 degrees apart in the tube parallel to the axis from the end of the tube neck to a location halfway to the end of the heated surface. A relatively large hole 2.39 mm in diameter was drilled so that a sufficiently rigid drill bit could be used to provide a straight thermocouple well. Sleeves made from small copper tubing, with an outside diameter approximately equal to the diameter of the thermocouple well (to provide a snug fit), were forced into the wells as shown in Fig. 2.2(b). The inside diameter of the sleeves was equal to 0.91 mm which provided for a snug fit between the sleeve and the 36-gage T-type (copper-constantan) thermocouple wire. The thermocouple beads were pressed against the bottom of the thermocouple wells, and the thermocouples were secured to the test sections with epoxy.

Test section heating was provided by inserting a 101.6 mm long, 6.35 mm diameter, 500W electric resistance cartridge heater into the test section, and power was supplied from a 0-120V variable transformer. The cartridge heater was manufactured 0.05 mm smaller in diameter than the inside diameter of the test section to provide a snug fit and a uniform

heat flux from the heater to the copper tube. Both ends of the test section were sealed to the cartridge heater with epoxy.

The current supplied to the cartridge heater was measured using a 15 A/100 mV shunt. The voltage across and current supplied to the cartridge heater were measured with a Hewlett–Packard family 9000, model 310 computer and an HP–3241A data acquisition system. In addition, the data acquisition system measured the temperature of the refrigerant, the water bath, and the four test–section wall temperatures. The atmospheric pressure was determined using a precision mercury barometer, and a precision manometer was used to verify that the pressure difference between the room and the vapor space inside the apparatus was negligible.

When experiments involving refrigerant mixtures were performed, three test sections, one with each type of surface, were attached to the mounting plate 0.1254 m apart so that the tests for each surface could be completed without removing the lid and allowing the concentration to change. Tests with three smooth tubes in pure refrigerant were conducted to ensure that the multiple mounting did not affect the heat transfer coefficients.

Experiments were conducted in pure R–113, pure R–11, and three mixture concentrations: nominally 25%, 50%, and 75% of R–113 by mass. Because R–11 was the pure fluid used just prior to the mixture tests, mixture concentrations were determined by measuring the volume of R–113 added to the pool. Prior to each set of experiments for a particular concentration, a volume of refrigerant was removed and a measured volume of pure R–113 was added so that the mixture concentration could be calculated.

Prior to taking data for each test section, the pool of refrigerant was heated to the saturation temperature and degassed by heating the pool with the test section and the water jacket. The test section boiled at a heat flux of approximately 30 kW/m^2 for at least 30 minutes in order to degas the refrigerant. The removal of dissolved gas from the refrigerant was necessary to prevent early nucleation from the tube surface.

The power to the cartridge heater was then turned off and the temperature of the test section was allowed to decrease to the saturation temperature. Power was then supplied again to the test section, and the temperature of the test section and pool were allowed to come to steady state. The criterion used for steady state was a change of less than 0.1% of the indicated level over 30 seconds for all the voltages and temperatures measured.

After the data were taken and stored, the power to the cartridge heater was increased slightly, and the temperature of the test section again was allowed to come to steady state. This procedure was repeated for each data point until the maximum cartridge heater power was reached. The power to the cartridge heater was then decreased slightly and the temperature of the test section and the pool were again allowed to come to steady state. In this way, data were taken at different heat flux values as the power was decreased. The experiment for a particular test section was complete when the minimum power to the cartridge heater was reached.

The values for the heat transfer coefficient was obtained by dividing the heat flux by the difference between the average tube surface temperature and the saturation temperature of the fluid. The surface temperature of the test section wall was calculated using the one-dimensional conduction equation for a cylinder assuming that the heat flux is uniform, which was ensured by providing a snug fit between the cartridge heater and the test section.

The saturation temperatures for the two pure fluids, R-113 and R-11, were evaluated as a function of pressure through the use of the property tables listed in ASHRAE (1985) and verified experimentally. When the working fluid was a mixture of R-113 and R-11, the bubble point was used in place of the saturation temperature. The bubble point was measured using a thermocouple placed in the bubbly flow just above the test section during the degassing process. The pressure at the center line of the test section was calculated by adding the hydrostatic pressure contribution (calculated using the

nominal height of liquid above the test section) to the measured atmospheric pressure.

The temperature measurements as input from the data acquisition system were calibrated experimentally to 0.1°C . A propagation-of-uncertainty analysis was performed. The analysis shows that for uncertainties in the surface temperature were less than 0.12°C and that the uncertainty in the saturation temperature, resulting from uncertainties in the pressure measurement, were less than 0.04°C . Therefore, the uncertainty in the wall superheat was calculated as 0.13°C . The uncertainties in the heat flux, as a result of uncertainties in surface area and voltage measurements, were less than 5%. Details can be found in Trewin (1992).

2.3 Smooth and Enhanced Tube Results with Pure Refrigerants

2.3.1 Pure R-113 Results

The boiling curves for tubes with the smooth, knurled (Turbo-B, medium opening), and sintered (medium and small porosity High Flux) surfaces in pure R-113 are shown in

Figure 2.3 Boiling curves for tubes with smooth, Turbo-B (medium opening), and High Flux Surfaces at $P = 103\text{ kPa}$

Fig. 2.3. The natural convection correlation curve developed by McAdams (1954) is plotted for comparison. The smooth tube results for increasing heat fluxes (from approximately 0.2 kW/m^2 up to approximately 2 kW/m^2) were predicted fairly well using the McAdams correlation, which validated the experimental apparatus and procedure.

Data from the smooth tube for heat fluxes from 2 kW/m^2 to 7 kW/m^2 deviates slightly from the natural convection line due to the presence of nucleate boiling at a few sites. The entirety of the heated surface did not nucleate until the heat flux was increased to approximately 8 kW/m^2 , at which time the nucleate boiling resulted in a significantly smaller wall superheat value that shifted the boiling curve to the left. Occasionally this shift in the boiling curve occurred over a range of heat fluxes, which was not repeatable. Visual observations revealed that nucleation was initiated in patches on the heated surface and that these patches grew with increasing heat flux. Similar observations were noted by Bergles and Chyu (1982).

Data taken at heat fluxes greater than 8 kW/m^2 were generally the same regardless of whether the heat flux was increased or decreased. Data obtained for decreasing heat fluxes less than 8 kW/m^2 were to the left of the data obtained for increasing heat fluxes due to the continued presence of nucleation sites, thereby creating the so-called boiling curve hysteresis.

Figure 2.3 also shows that data from the small porosity High Flux surface have the smallest wall superheat for decreasing heat fluxes greater than 1.5 kW/m^2 and so performed the best in this heat flux range. The Turbo-B surface performed better than the smooth and medium porosity High Flux surfaces for heat fluxes less than 10 kW/m^2 , but the medium porosity High Flux surface performed better than the Turbo-B and smooth surfaces for heat fluxes greater than 10 kW/m^2 . At low heat fluxes, the heat transfer coefficients for a given wall superheat are generally at least an order of magnitude better for the enhanced tubes than the smooth tubes.

At higher heat fluxes, the difference between the results for the smooth and enhanced surfaces diminishes and the boiling curves for the smooth and Turbo-B surfaces merge together. Some tests showed at higher heat fluxes a sudden shift in the boiling curve to the right which indicates that the critical heat flux (CHF) has been reached. This was verified by visual observations of film boiling. It appears that the CHF value for the Turbo-B and medium porosity High Flux tubes will be approximately the same as that of the smooth tube (approximately 200 kW/m^2), which indicates that the surface enhancement is ineffective when operating too close to the CHF value.

The results shown in Fig. 2.3 agree with results of other researchers. For example, the results of the smooth and small porosity High Flux surfaces are similar to those obtained by Chyu (1979). Webb and Pais (1991) report very good agreement between heat transfer coefficients from their Turbo-B tube and values from the present study.

The small porosity High Flux surface produced the largest hysteresis in the boiling curve of approximately 10°C while the Turbo-B produced almost no overshoot. A possible explanation for the lack of a large temperature overshoot for the Turbo-B surface is that the subsurface geometry of the Turbo-B tube is connected by a spiraling channel. Figure 2.4 shows a cross section of the Turbo-B surface enlarged 20 times. Note that the subsurface channels remain open even though the pore openings are sometimes closed and the outside surface is sometimes compressed an additional amount. This channel would make it possible for vapor from just a few active nucleation sites inside the channel on the bottom of the tube to travel circumferentially around the tube and activate other extinct nucleation sites. Such a process would be less likely for tubes with the High Flux surface where the connecting subsurface channels are interrupted as shown in Fig. 2.5. Indeed, visual observations verified that the onset of nucleation for the High Flux tubes occurred in patches on the tube surface and at varying heat fluxes.

Figure 2.4 Axial cross section of the Turbo-B tube (enlarged 9X)

In the fully established nucleate boiling range, the size of the pore opening has very little influence on the boiling curve for the Turbo-B surface. As reported by Jung and Bergles (1989), three boiling curves were obtained for Turbo-B tubes with surface opening sizes designated small, medium, and large, and the results were nearly identical. Subsequently, Turbo-B tubes with surface opening sizes designated "very open" and "very closed" were tested and the results are shown in Fig. 2.6. Even for these extreme pore sizes, the boiling curves show very little effect, especially for heat fluxes greater than 2000 W/m². These results support the conclusion that the basic heat transfer mechanism takes place in the subsurface channel alone and that the bubble departure process at the pore has little influence.

Figure 2.5 Cross-section of the High Flux surface enlarged 100X (Gottzmann et al. 1973)

Figure 2.6 Comparison of Turbo-B tube pool boiling data in R-113 at $P = 103$ kPa

The porosity of the High Flux surface, on the other hand, has a strong effect on the nucleate boiling heat transfer, as seen by comparing the "High Flux, small" and "High Flux, medium" results shown in Fig. 2.3. Evidently, the bubble departure process is very important in the heat removal process for the High Flux surface.

It is interesting to note that while the slope of the $q'' \sim \Delta T$ boiling curves for the smooth and High Flux surfaces are steep, the slope of the boiling curve for the Turbo-B surface is approximately unity. This implies that the heat transfer coefficients for the Turbo-B tubes varied only slightly with heat flux. The heat transfer coefficients for the High Flux and smooth tubes, on the other hand, depended strongly on the heat flux so that the two enhanced surfaces performed very differently in this respect. The weak heat flux dependency on the heat transfer coefficients for the Turbo-B tubes is possibly the result of thin film vaporization.

A "least-squares" type of linear regression was performed on the pool boiling data for the three sets of surfaces used in the tube bundle experiment. The correlation equation for the smooth tube data is:

$$q'' = 1.067 (\Delta T_{\text{sat}})^{4.012} \quad (2.1)$$

The correlation approximates the data to within 2.4% in the range of heat fluxes from 2.3 kW/m² to 123 kW/m². The exponent on ΔT_{sat} , which is the slope of the correlation curve, is very nearly the same as given by other researchers including Stephan and Abdelsalam (1980) who use a value of 3.92. For the Turbo-B data, the equation is:

$$q'' = 6149 (\Delta T_{\text{sat}})^{1.283} \quad (2.2)$$

which approximates the data to within 1.1% in the range of heat fluxes from 1.6 kW/m² to 152 kW/m². The exponent on ΔT_{sat} is close to unity and confirms that the heat transfer coefficient is a weak function of heat flux. For the High Flux data, the equation is:

$$q'' = 4227 (\Delta T_{\text{sat}})^{2.033} \quad (2.3)$$

which approximates the data to within 3.1% in the range of heat fluxes from 1.6 kW/m² to 43 kW/m². The value of the exponent on ΔT_{sat} is between that for the smooth and Turbo-B surfaces for the applicable range of heat fluxes. Note, however, that while a linear regression appears to correlate the data for the smooth and Turbo-B surfaces fairly well, a non-linear regression would have been necessary to predict the gradually decreasing slope in the data for the High Flux surface.

2.3.2 Pure R-11 Results

As seen in Fig. 2.7 for the smooth tube, the pool boiling data in pure R-11 are nearly the same as in pure R-113 for fully established nucleate boiling. The performance is only slightly better in R-11 than in R-113 at low heat fluxes (less than 10 kW/m²).

Figure 2.7 Saturated pool boiling curve for the smooth tube in pure R-11 and pure R-113 at $P = 103$ kPa

The relative performance of the smooth in R-113 and R-11 can be determined by examining a model for nucleate pool boiling, such as the one described by Forster and Zuber (1955):

$$q'' = C_1 \left[\frac{i_{fg}^{1/2} \rho_v^{3/8} \rho_L^{1/2} c_p^{11/24}}{\sigma_L^{1/2} T_{sat}^{3/2} \mu_L^{7/24} k_L^{25/24}} \right] (T_w - T_{sat})^2 \quad (2.4)$$

In general, the heat transfer coefficient in R-11 should be similar to those in R-113 due to their similar properties. For example, the vapor density and surface tension of R-113 is slightly greater than for R-11 but the thermal conductivity and heat of vaporization are slightly greater for R-11 than for R-113. At 100 kPa and a given wall superheat, equation 2.4 predicts that the heat flux for R-113 is approximately 15% larger than for R-11. Alternatively, the ratio $(T_{sat}/p_v i_{fg})^{1/2}$ (Trewin 1992) used to predict the pressure effect also predicts a slightly larger heat transfer coefficient in R-113 than in R-11. The lack of any significant differences between the experimental data obtained in R-113 and R-11 can be attributed to experimental uncertainty.

The performance of the Turbo-B tube (medium opening) in R-11 was better than in R-113 for heat fluxes greater than 5 kW/m². The performance was poorer in R-11 than in R-113 for heat fluxes less than 1 kW/m² and was approximately equal for intermediate heat flux values. The performance of the small porosity High Flux surface was better in R-11 than in R-113 for heat flux values less than 30 kW/m² with the greatest increase in the performance occurring at the smallest heat flux value.

2.4 Binary Mixture Results

Figure 2.8 shows the boiling curves for the smooth tube in mixtures of R-113 and R-11. Boiling curves for the smooth tube in pure R-113 and pure R-11 are also shown for comparison. In general, the performance of the smooth surface does not change

Figure 2.8 Saturated pool boiling curve for the smooth tube in mixtures of R-11 and R-113 at $P = 103$ kPa.

significantly with mixture concentration. The performance is only slightly better in pure R-113 than in 25% R-11 for heat flux values greater than 10 kW/m^2 . For a mixture of 50% R-11 and 50% R-113, the performance is only slightly better in the range from 1 kW/m^2 to 10 kW/m^2 than the performance in the same range of heat fluxes for a mixture with 25% R-11 and for pure R-113. The smooth tube performed better in a mixture of 75% R-11 and 25% R-113 than in pure R-11 for a range of heat flux values from 1 to 10 kW/m^2 , worse for heat flux values less than 1 kW/m^2 , and the same for heat flux values greater than 10 kW/m^2 .

Boiling curves for the Turbo-B surface (medium opening) are shown in Fig. 2.9. Generally, the performance is best in pure R-11, slightly poorer for pure R-113, and as the amount of R-113 in the mixture is increased, the boiling curve is shifted to the right. The same trends were found for the small porosity High Flux tube as shown in Fig. 2.10. It is

Figure 2.9 Saturated pool boiling curve for the Turbo-B tube (medium opening) in mixtures of R-11 and R-113 at $P = 103$ kPa

Figure 2.10 Saturated pool boiling curve for the High Flux tube (small porosity) in mixtures of R-11 and R-113 at $P = 103$ kPa

interesting to note that for the High Flux surface, the onset of fully established nucleate boiling occurred between 2 kW/m^2 and 3 kW/m^2 .

2.5 Conclusions

Pool boiling tests were conducted for each of the three surfaces so that reference values of the heat transfer coefficient could be obtained. Values of the heat transfer coefficient for the enhanced tubes were generally an order of magnitude larger than for the smooth tubes. Heat transfer coefficients for the porous surface were generally the largest; however, a large hysteresis in the boiling curve was found. Visual observations revealed that only patches of the High Flux surface nucleated initially and that nucleation covered more of the surface as the heat flux increased. Heat transfer coefficients for the Turbo-B tube, on the other hand, showed very little hysteresis in the boiling curve.

The pool boiling experiments conducted in pure R-11 showed that, as for the pure R-113 experiments, the enhanced tubes performed better than the smooth tubes. Indeed, the performance of the smooth and High Flux tubes in pure R-11 was approximately equal to that in pure R-113, except for heat fluxes less than 10 kW/m^2 where the performance in R-11 was slightly greater than in R-113. The performance of the Turbo-B tube was also slightly better in R-11 than in R-113.

Pool boiling heat transfer coefficients in fluid mixtures were less than those in pure fluids for all three surfaces. However, for a given heat flux and mixture concentration, the enhanced surfaces consistently performed better than the smooth tube. The High Flux tube performed slightly better than the Turbo-B tube, as was the case for the pure fluid experiments. The performance of all three surfaces decreased as the concentration of R-11 decreased, with the largest reduction occurring in a mixture of 25% R-11/75% R-113.

3. SHELLSIDE BOILING ON ENHANCED TUBES IN CROSSFLOW

3.1 Introduction

Shellside boiling with crossflow in horizontal multi-tube bundles is used extensively in a variety of applications, such as kettle and thermosyphon reboilers in the chemical process industry, submerged evaporators in the refrigeration industry, waste-heat boilers, etc. As noted in recent reviews by Palen (1983) and Jensen (1988), design methods for heat exchangers of this type tend to be conservative, primarily because of the lack of data to fully explain what does occur in these heat exchangers. Recent research (e.g., Cornwell and Schuller 1982, Hwang and Yao 1986, Jensen and Hsu 1988, Jensen et al. 1989, Fujita et al. 1990, and Cornwell 1990) has begun to shed light on the heat transfer and fluid mechanics of the two-phase flow across multi-tube bundles. Most of the investigations have used smooth tubes in their bundles. However, there have been several studies that have tested enhanced tubes in the bundles to determine if, relative to a smooth tube, the significant enhancement in the heat transfer obtained in saturated-liquid pool boiling on a single enhanced tube also is realized in large multi-tube bundles in a two-phase crossflow, or whether there is a further enhancement due to convection (the so-called bundle effect).

Finned tubes have been studied, for example, by Myers and Katz (1953), Wall and Park (1978), Danilova and Dyundin (1972), Hahne and Mueller (1983), Yilmaz and Palen (1981), and Cornwell and Scoones (1988). Generally, these investigators found that the finned tubes exhibit the same behavior as previously obtained with smooth tubes, except for a higher starting basis due to area enhancement. That is, at low heat fluxes, the heat transfer coefficient increased significantly at rows higher in the bundle; as the heat flux increased, the separate curves for each tube row converged to a single curve that was representative of a single finned tube. Induced convection and thin-film evaporation are credited with enhancing the heat transfer performance.

Finned tubes that have been further cold worked (e.g., Gewa-T and Thermoexcel-E) have been used by Arai et al. (1977), Yilmaz et al. (1981), and Stephan and Mitrovic (1981). These tubes have had their fins bent over to form reentrant type cavities that are interconnected below the surface. While there are some inconsistencies among the results from the different investigations, several conclusions can be drawn. A single enhanced tube of this type outperforms both a single plain tube or a plain-tube bundle. However, the enhanced tube's performance relative to a single enhanced tube was much smaller than that for a plain-tube bundle relative to a single plain tube. Hence, it appears that convective effects and/or thin-film evaporation, which enhance the plain bundle performance, are not as significant for these enhanced tubes.

The third enhancement technique used in bundle boiling is the bonding of a porous, sintered metallic matrix to the base tube (e.g., Union Carbide's High Flux tubing). This enhancement technique has been investigated, for example, by Fujita et al. (1984), Czikk et al. (1970), O'Neill et al. (1980), Tatara and Payvar (1986), and Koyama and Hashizume (1983). In a comparison between a bundle with this enhanced surface and a plain-tube bundle, about a factor of ten increase in the heat flux at a given overall temperature difference was obtained. The most striking aspect of this surface's performance was that there was essentially no difference between its heat transfer characteristics in a bundle or on a single tube in a pool (Czikk et al. 1970).

In investigations where natural circulation test rigs were used, neither the mass flux nor quality were known for any of the tests. Likewise, the range of conditions studied have been limited. Hence, an experimental investigation has been performed in which mass flux, pressure, local quality, and test-section geometry (bundle layout and surface geometry) were controlled and varied over a wide range of conditions so that the effect of two-phase crossflow over enhanced tubes could be more definitively evaluated and information developed to help clarify the governing heat transfer mechanisms. Three surfaces (smooth,

Turbo-B, and High Flux) were tested in two staggered tube bundles using R-113 as the working fluid.

3.2 Experimental Apparatus and Procedure

The basic flow loop used in this experiment (described by Dykas and Jensen (1992)) consisted of a positive displacement pump, electric preheaters, condenser/aftercooler, storage/surge tank, and the test-section assembly (see Figure 3.1). Degassed R-113 was the working fluid.

The tube-bundle test chamber, designed to produce shellside boiling over horizontal tubes where the predominant flow direction is vertically upward, is described in detail by Trewin (1992). The inlet to the test-chamber shell consisted of brass screens, a diverging

Figure 3.1 Flow loop schematic

section containing flow vanes, and a section containing a honeycomb flow straightener, as shown in Figure 3.2. The screens were placed at the end of the inlet pipe so that fluid entering the test chamber was evenly distributed across the flow area. The stainless-steel flow vanes allowed for a smooth transition from the inlet pipe to the rectangular body of the test chamber. Any swirl in the flow was removed by the stainless-steel flow straightener.

The 68 tubes that made up the bundle were securely sealed in the test chamber with a precise orientation and spacing. This number of tubes represents one of the largest bundles ever tested and insures that both columns and rows are representative of actual bundles. The holes used for mounting the heated tubes were laid out in 5 columns and 15 rows according to two equilateral triangular spacings, a pitch-to-diameter ratio of

Figure 3.2 Cross-section of the test chamber

$p/d=1.50$ on one and 1.17 on the other. Six pressure taps were drilled in the mounting plate between tube holes in the middle column so that the local saturation temperature and two-phase pressure drop could be measured. The pressure taps were located at the inlet, exit, and just upstream of the third, seventh, eleventh, and fifteenth tube rows.

Thermocouple ports were constructed in the mounting plates just upstream of the bundle and at the exit so that fluid temperatures could be measured. The front plate of the test chamber was made of Lexan so that visual assessment of the hydrodynamics of the two-phase flow could be made.

The 101.6 mm long, 19.1 mm O.D./5.08 mm I.D. heated tubes were constructed of copper and were attached horizontally to the mounting plate. The middle column of tubes was instrumented with thermocouples that penetrated axially through to the midpoint of the heated surface. The construction of the instrumented tubes is shown on Fig. 2.2. The uninstrumented tubes were constructed identically except for the absence of thermocouple wells. In each bundle, half of each tube in the outside columns was imbedded in the chamber wall.

The cartridge heaters used to heat the tubes were connected to one of three variable transformers. Each transformer supplied up to 14 kW to a pair of bus bars.

Approximately one-third of the cartridge heaters were connected electrically in parallel to each set of bus bars. Each variable transformer had its own current shunt for measurement of the total current and power to each of the three banks of tubes. In addition, a battery of eight shunts was installed so that the current through each of the instrumented tubes could be measured individually.

The test-section inlet pressure was measured with a Bourdon-tube gauge. A differential pressure transducer was used to measure the pressure drop between six pressure taps distributed along the tube bundle. Type T copper-constantan thermocouples placed at the inlet and the exit of the test section, and elsewhere in the loop, were located in the center of the flow channels. The mass flow rate was measured upstream of the preheaters

by means of a turbine flowmeter. Thermocouple voltages, flowmeter frequencies, instrumented tube voltage drops, shunt voltage drops, and pressure drop transducer voltages were measured using a computer-controlled data-acquisition system; pressure-level readings were manually entered into the computer.

After the test-section flow and pressure levels were established, the electric preheaters were set to raise the R-113 temperature to that necessary to attain the desired saturated liquid condition at the inlet to the bundle. A high pressure was maintained in the preheaters through the use of the flashing valves (see Fig. 3.1). The start-up heat flux in the three sections of the tube array was then set to the same value, generally at the highest heat flux to be tested so that hysteresis effects could be avoided. Data were taken once conditions in the loop had reached steady state, as defined by inlet temperatures and flows not varying by more than 0.1 K or 2%, respectively.

Mass fluxes from 50 to 500 kg/m²s, absolute pressures of 0.2 and 0.6 MPa, heat fluxes from 5 to 80 kW/m², and qualities from 0 to 80% were tested. Maximum preheater capacity restricted the range of quality at higher mass fluxes and pressures, especially in the $p/d = 1.50$ test section. Mass flux was calculated through the minimum flow area within the tube array. Local pressures at tubes along the test section were calculated by linear interpolation between pressure taps. The saturation temperature at any of the instrumented tubes was then derived from the local pressure. The heat transfer coefficient was calculated from the heat flux, the average wall temperature, and the local saturation temperature. Qualities at the test section inlet and along the test section were determined by assuming an isenthalpic process across the flashing valves and by an equilibrium energy balance along the test section. Uncertainties in the experimentally measured quantities were estimated (Trewin, 1992) through a propagation of uncertainty analysis as: G , $\pm 3.0\%$; P , $\pm 0.5\%$; x , $\pm 0.5\%$; q'' , $\pm 5.0\%$.

3.3 Smooth Tube Reference Results with Pure Refrigerant

Single-phase heat transfer tests on both tube bundle layouts and the three surface geometries were run over a wide range of Reynolds numbers. The smooth-tube heat transfer coefficients were compared to two correlations; Zukauskas (1972) and ESDU (1973). The experimental values were within 5% of the predicted values. There were essentially no differences between the smooth-tube results and the single-phase data of the Turbo-B and the High Flux tubes. Evidently, the surface roughness on those two tubes is sufficiently small so that it does not have an effect on the heat transfer.

Single-phase and two-phase pressure drop data also were obtained. The single-phase data agreed well with predictions from the ESDU (1979) correlation. The two-phase pressure drop was consistent with previous results (e.g., Jensen et al. 1989). For the same operating conditions (whether single or two-phase), the same pressure drop was obtained with all three surfaces. As with the single-phase heat transfer, the surface roughness on the enhanced tubes was sufficiently small to have no discernible effect on the pressure drop characteristics of the flow.

Typical boiling data for the smooth tube in the tube bundle are shown on Fig. 3.3. The convective bundle effect (increasing heat transfer coefficient between inlet tube row and exit tube row) is not significant in these data. The effect of a varying mass flux is minimal; as shown on Fig. 3.3a, a factor of ten variation in mass flux has a negligible effect on the heat transfer coefficient. At lower heat fluxes (see Fig. 3.3b), some convective effects begin to be evident, but they are not large. This pattern is the same at low and high pressures (Fig. 3.3a), and is caused by a trade off between convection and nucleation effects that govern the heat transfer process. Nucleation becomes the dominant mechanism at relatively low heat fluxes, even though the two-phase velocity probably is quite large toward the outlet of the bundle. The abscissa on Fig. 3.3 is "Row Number" but since there was a uniform heat flux applied to the bundle, the abscissa is directly proportional to

Figure 3.3 Effect of heat flux pressure, tube row/quality, and mass flux on the heat transfer coefficient, and comparison with pool boiling results

quality. Note that on Fig. 3.3a, the outlet quality approaches 80% for two of the runs shown, but there is no appreciable increase in the heat transfer coefficients from the second instrumental tube to the exit of the bundle.

The minimal role played by convection is further illustrated by the data shown on Fig. 3.4. Data from the two smooth-tube bundles show that the results from the bundle with $p/d=1.17$ were slightly larger than those of the $p/d=1.50$ bundle, but with increasing heat flux the data overlap and the effect of bundle layout becomes small. In single-phase convection, these two bundles do have different heat transfer coefficients at the same mass flux. The small differences in boiling indicate that nucleation probably does have the dominant role in governing the heat transfer process.

Figure 3.4 Effect of the tube pitch-to-diameter ratio on the heat transfer coefficient

While these results are consistent with previous bundle research (e.g., Jensen et al. 1989), it is not clear what all the governing mechanisms are or how they interact. Mechanistic models, such as suggested by Cornwell (1990), have been developed; however, they generally are applicable to low mass flux, bubbly flow situations and, thus, are not necessarily appropriate for the high-quality, high-mass-flux flows used in this experiment. In addition, all the previous bundle research have used R-113 as the working fluid, as does this experiment. This choice of fluid may confound the interpretation and analysis of the data because of possible peculiar behavior first noted in in-tube boiling.

Kandlikar (1990) has gathered a large data bank of heat transfer coefficients for intube evaporation of a variety of fluids. Some of the R-113 data show a decreasing heat

transfer coefficient with increasing quality, which is opposite of typical behavior. Using guidance from the Kandlikar correlation, it appears that this and other investigations were performed at conditions where the Kandlikar correlation would estimate a relatively flat curve of heat transfer coefficient versus quality for an intube flow. If the non-typical behavior exhibited in in-tube flows also occurs in shellside boiling, then clarification of the mechanisms governing the process will be much more difficult.

3.4 Enhanced Tube Results with Pure Refrigerant

Typical data from the tube bundles using the Turbo-B and High Flux surfaces are shown on Figs. 3.5 and 3.6, respectively. Generally, the heat transfer trends with both enhanced tube bundles were comparable to those of the smooth-tube bundle. There were very little convective effects noted in the data, even at the lowest heat the lowest heat fluxes tested (about 10 kW/m^2). Figure 3.6 shows that the heat transfer coefficients for the High Flux tube bundle varied significantly with heat flux, as was the case for pool boiling. However, the heat transfer coefficients for the Turbo-B tube bundle did not vary as much, especially for heat fluxes greater than 20 kW/m^2 , as shown on Fig. 3.5. Again, pool-boiling values showed similar trends. Pool boiling values are not shown in Figs. 3.5 and 3.6 due to the range of heat flux shown but comparisons can be made with Fig. 3.3. The heat transfer coefficient versus tube row curves were essentially flat across the bundle at all heat fluxes, although there were some "entrance effects" noted with the High Flux surface (Fig. 3.6). We suspect that difficulty in initiating boiling over the complete surface of the tubes in the first few rows of the bundle contributed to this behavior. Patchiness in the boiling was noted through visual observations of the boiling process during both the preliminary pool-boiling tests and bundle-boiling tests. As shown on Fig. 3.5, exit effects occasionally caused an increased heat transfer coefficient at the last row in the bundle. This is possibly the result of a dramatic change in the flow pattern due to the sudden expansion at the exit of the tube bundle.

Figure 3.5 Heat transfer coefficients for a tube bundle using the Turbo-B surface

Figure 3.6 Heat transfer coefficients for a tube bundle using the High Flux surface

The effects of pressure are shown on Fig. 3.3 for the smooth surfaces, and the results for the two enhanced surfaces are consistent. Among the three surfaces, the smooth tube showed the largest increase in the heat transfer coefficient with pressure. Comparison of the two enhanced tubes showed that the effect of pressure was smaller for the Turbo-B tube and larger for the High Flux tube.

Figure 3.7 shows that for a factor of ten increase in mass flux, at a moderate heat flux the High Flux surface's heat transfer coefficient is essentially constant. The lack of any influence on the heat transfer coefficient due to convection is not surprising given the results of the smooth tube and the fact that both of these enhanced tubes were designed to improve the nucleate boiling heat transfer behavior. These surfaces are designed with very small surface openings, and the dominant heat transfer processes in nucleate pool boiling occur inside the cavities and internal channels. Convective effects from the gross external

Figure 3.7 Effect of mass flux on the heat transfer coefficients for High Flux bundle

flow past the tubes apparently cannot intrude into or influence the processes in these small spaces. Hence, as in pool boiling, for these types of enhanced surfaces, subsurface heat transfer mechanisms also govern the processes in forced-convective two-phase flow.

3.5 Binary Mixture Results

Two-phase heat transfer coefficients were obtained in the same tube bundle used to collect data in pure R-11. The bundle had a tube pitch-to-diameter ratio of 1.50 and the bottom five rows of the tube bundle contained tubes with the High Flux surface, rows six through ten contained tubes with the smooth surface, and the top five rows contained tubes with the Turbo-B surface. Experiments were conducted in mixtures of R-113 and R-11 with inlet concentrations of approximately 25%, 50%, and 75% R-11 by mass. The experiments were conducted for pressures of 200 and 600 kPa, mass flux values of 50 and 225 kg/m²s, and heat flux values from 5 to 80 kW/m² so that the effect of these parameters on the tube bundle performance could be determined.

Results from the 75% R-11/25% R-113 mixture

Heat transfer coefficients for tubes with each of the three surfaces are shown in Fig. 3.8 for a mass flux of 50 kg/m²s, a pressure of 200 kPa, and heat flux values from 5 to 60 kW/m². The inlet mass concentration of R-11 in the liquid is 75.8% and decreases with increasing tube row due to evaporation of the mixture. The rate at which the concentration of R-11 in the liquid decreases depends on the mass flux and heat flux. For example, at a heat flux of 60.4 kW/m², the mass concentration of R-11 in the liquid decreased to 72.5% at the seventh tube row, which is the first row of instrumented smooth tubes. At the eleventh tube row, which is the first Turbo-B tube row, the mass concentration of R-11 in the liquid decreased to 70.4%. The mass concentration of R-11 in the liquid at the outlet of the tube bundle was 67.1%. The concentration changes

Figure 3.8 Effect of heat flux on the heat transfer coefficients for all three surfaces in a mixture of 75.8% R-11 and $G = 49 \text{ kg/m}^2\text{s}$

throughout the bundle at different rates depending on the heat flux. At a heat flux of 10 kW/m^2 , the mass concentration of R-11 in the liquid changed by only 1.6% from the inlet to the exit of the bundle.

The heat transfer coefficients follow the same general trends as those obtained in the pure fluids. The heat transfer coefficients for the High Flux tubes in the first, third, and fifth tube rows exhibit a strong dependency on the heat flux and generally the heat transfer coefficient at the first row is the smallest. The smooth tube heat transfer coefficients were consistently smaller than those for the enhanced tubes. A significant heat flux effect was also observed for the smooth tubes. The heat transfer coefficients for the Turbo-B tubes (shown in rows 11, 13, and 15) exhibit much less variation with heat flux as was the case in the pure fluids, and some exit effects are still present.

The effect of the mass flux on the heat transfer coefficients can be seen by comparing Figs. 3.9 and 3.8; Figure 3.9 shows data at a mass flux of $225 \text{ kg/m}^2\text{s}$ while Fig. 3.8 shows data at $49 \text{ kg/m}^2\text{s}$. Heat transfer coefficients for each of the three surfaces at a given heat flux are not significantly different for the different mass flux conditions. Note that because the mass concentration of R-11 in the liquid is not an independent parameter, comparing data taken at a similar heat flux, pressure, and inlet concentration but different mass flux results in a comparison between data at different local mixture concentrations as well. For the same nominal inlet concentration and at a given tube row, the mass concentration of R-11 in the liquid is lower for the data taken at a mass flux of $49 \text{ kg/m}^2\text{s}$ than at $224 \text{ kg/m}^2\text{s}$, especially for the tubes higher in the bundle and at larger heat fluxes. For the same inlet concentrations and a heat flux of 60 kW/m^2 , the mass concentration of R-11 at the bundle exit is 73.3% when the mass flux is $49 \text{ kg/m}^2\text{s}$, which is 6.3 percentage points higher than at the bundle exit when the mass flux is $224 \text{ kg/m}^2\text{s}$. However, the mass concentration change and any changes in the free-steam convection due to a larger mass flux apparently do not appreciably affect the concentration gradient at the heated surface and so do not change the mechanisms that reduce the heat transfer.

The magnitudes of the heat transfer coefficients are, however, significantly lower in the binary mixture than in the pure fluids. For example, the heat transfer coefficient for the smooth tubes obtained in a mixture with 72.3% R-11 in the liquid at 60 kW/m^2 and 200 kPa is approximately $3.4 \text{ kW/m}^2\text{K}$. This value is 19% smaller than the value obtained for similar operating conditions in pure R-113 ($4.2 \text{ kW/m}^2\text{K}$) and 12.8% smaller than the value obtained in pure R-11 ($3.9 \text{ kW/m}^2\text{K}$). This trend has been well established and is described in detail by several researchers. Thome and Shock (1984) give some of the reasons for the lower heat transfer coefficient in mixtures: a higher concentration of the less volatile component at the heated surface causing an increased local boiling temperature and a reduced bubble growth rate; a higher enthalpy of vaporization; varying bubble departure frequencies; and reduced nucleation site density due to reduced contact angle.

Figure 3.9 Effect of heat flux on the heat transfer coefficients for all three surfaces in a mixture of 75.8% R-11 and $G = 224 \text{ kg/m}^2\text{s}$

The magnitudes of the heat transfer coefficients for the enhanced surfaces obtained in the binary mixture also are significantly lower than those obtained in pure fluids. The average heat transfer coefficient obtained in the binary mixture for the High Flux tubes in rows three and five at 60 kW/m^2 is approximately $10.8 \text{ kW/m}^2\text{K}$. The average value at similar operating parameters is $12.5 \text{ kW/m}^2\text{K}$ in pure R-113 and $12.2 \text{ kW/m}^2\text{K}$ in pure R-11. Therefore, the reduction in the heat transfer coefficients obtained in the mixture is approximately 13.6% and 11.5% compared to the pure R-113 and R-11 data, respectively. The heat transfer coefficient for the Turbo-B tubes with a mass concentration of 70.0% R-11 in the liquid upstream of the eleventh tube row is $8.4 \text{ kW/m}^2\text{K}$. This value is 2.3% smaller than the value of $8.6 \text{ kW/m}^2\text{K}$ obtained in pure R-113 and 14.3% smaller than the value of $9.8 \text{ kW/m}^2\text{K}$ obtained in pure R-11. For a given heat flux larger than $30 \text{ kW/m}^2\text{K}$, the High Flux heat transfer coefficients generally were larger than those for the

Turbo-B tubes. The heat transfer coefficients for the enhanced surfaces are always larger than for the smooth tubes at similar heat fluxes, pressures, and inlet mixture concentrations.

Results from the 50% R-11/50% R-113 mixture

The heat transfer coefficients obtained in a mixture with a nominal inlet mass concentration of 50% exhibited the same general trends as the data obtained in 75% R-11. The data show that the High Flux tubes performed slightly better than the Turbo-B tubes and that both enhanced tubes performed better than the smooth tubes. However, the variation in the mixture concentration throughout the bundle makes a direct comparison of data obtained at different tube rows difficult. Data for the Turbo-B tubes were obtained in mixtures with local liquid concentrations up to 7.6 percentage points lower in R-11 than the local mixture concentration at the High Flux tubes. The lower concentration of R-11 could account for some of the difference between the High Flux and Turbo-B tubes but lower heat transfer coefficients for the Turbo-B tubes were also found in the pure fluids, and so the variation in the liquid concentration of R-11 probably plays a minor role.

The heat transfer coefficients for the Turbo-B tubes do not vary as much with heat flux as do the heat transfer coefficients for the High Flux tubes. A possible explanation is that the concentration of R-11 in the liquid at a given tube row was lower for the larger heat fluxes, and the heat transfer coefficients decrease with decreasing liquid concentrations of R-11. Therefore, the variation in the local concentration of liquid R-11 could contribute to the reduced variation with heat flux in the Turbo-B tubes. However, a similar trend was found in the data obtained in pure fluids and so, again, the variation in the liquid concentration of R-11 probably plays a minor role.

Comparisons show that the heat transfer coefficients for the 50% inlet concentration are generally smaller than those for the 75% inlet concentration by as much as 48% for the High Flux tubes, 37% for the Turbo-B tubes, but only 6% for the smooth tubes. Large

reductions in the heat transfer coefficients for the enhanced tubes and smaller reductions for the smooth tube were also observed in the binary mixture pool boiling results. Similar mechanisms between pool boiling and bundle boiling appear to play significant roles in the heat transfer process.

Results from the 25% R-11/75% R-113 mixture

Data obtained in a mixture with an inlet mass concentration of 23.8% R-11 in the liquid are shown in Fig. 3.10 for a mass flux of 217 kg/m²s and a pressure of 200 kPa. At a given heat flux, the heat transfer coefficients for the High Flux tubes are larger than those for the smooth tubes but are slightly smaller than those for the Turbo-B tubes in general. The slightly larger heat transfer coefficients for the Turbo-B tubes compared to the High

Figure 3.10 Effect of heat flux on the heat transfer coefficients for all three surfaces in a mixture of 23.8% R-11 and $P = 200 \text{ kPa}$

Flux data is contrary to the trends obtained in pure fluids, and might be the result of the lower local mass concentration of R-11 in the liquid at the Turbo-B tubes.

Insight into the trends of the heat transfer coefficients for the Turbo-B tubes can be gained by comparing heat transfer coefficients obtained in a 25% R-11 mixture to those obtained in the 50% R-11 mixture. Comparisons show that, at a given heat flux, the heat transfer coefficients decreased for each of the surfaces but that those for the High Flux tubes decreased the most. For example, at a heat flux of 60 kW/m^2 and a liquid mass concentration of 22.7% R-11, the heat transfer coefficient of the High Flux tube is 13.3% smaller than the High Flux heat transfer coefficient obtained at a liquid mass concentration of 45.4% R-11. At a heat flux of 60 kW/m^2 and a liquid mass concentration of 20.6% R-11, the smooth tube heat transfer coefficient was 5.9% smaller than that obtained in a mixture of 43.2% R-11. At a heat flux of 60 kW/m^2 and a liquid mass concentration of 18.5% R-11, the heat transfer coefficient for the Turbo-B tube was 5.9% smaller than that obtained in a mixture of 41.1% R-11 in the liquid. The smaller decrease in the heat transfer coefficients for the Turbo-B tubes compared to the High Flux tubes could be the result of an end in the decreasing trend in the heat transfer coefficient with decreasing liquid mass concentration of R-11 or perhaps even the result of a beginning in an increasing trend in the heat transfer coefficient with decreasing liquid mass concentration of R-11.

In general, increasing the inlet pressure results in an increase in the heat transfer coefficients for all three tube surface types and at all inlet mixture concentrations. The trends at 600 kPa are similar to those at 200 kPa, with the heat transfer coefficients larger for the enhanced tubes than for the smooth tubes. The heat transfer coefficients for the High Flux tubes increase with pressure but generally by only 11%. This increase obtained in the mixtures is significantly less than for the data obtained in pure R-113, which resulted in an average increase of approximately 30%. The smooth tube heat transfer coefficients obtained in the mixtures were approximately 33% larger at 600 kPa than at

200 kPa, which is consistent with data obtained in pure R-113. The Turbo-B heat transfer coefficients obtained at 600 kPa increased by approximately 9% over the data obtained at 200 kPa for inlet mass concentrations of 75.8% and 45.8% but increased by 23% for an inlet mass concentration of 23.1% R-11.

Examples of these data are shown in Fig. 3.11 which is a plot of the heat transfer coefficients versus the mass concentration of R-11 in the liquid for the High Flux tube at the third row, the smooth tube at the seventh row, and the Turbo-B tube at the eleventh row for an average mass flux of 204 kg/m²s, and a heat flux of 60 kW/m². The Turbo-B heat transfer coefficients obtained at a pressure of 600 kPa and a local liquid mass concentration of 20.0% R-11 are now slightly larger than for a liquid concentration of

Figure 3.11 Effect of mixture concentration and pressure on the heat transfer coefficients for $q'' = 60 \text{ kW/m}^2$

41.5%. The heat transfer coefficients are beginning to increase with decreasing liquid mass concentration of R-11 and, apparently, the minimum heat transfer coefficients occur in mass concentrations greater than 20.0% R-11 at 600 kPa.

Comparison of results

The effect of the mass concentration of R-11 in the liquid on the heat transfer coefficient can be more easily seen in a plot such as Fig. 3.12 where heat transfer coefficients for High Flux tubes in the third tube row are plotted versus the local mass concentration of R-11 in the liquid at a pressure of 200 kPa and an average mass flux of $204 \text{ kg/m}^2\text{s}$. Note that the mass flux is large enough so that the change in the mass concentration with heat flux is small at the third tube row, even for heat fluxes as large as 80 kW/m^2 . Therefore, the data shown in Fig. 3.12 were taken in mixture concentrations approximately equal to the inlet concentration.

Figure 3.12 Effect of mixture concentration and heat flux on the High Flux heat transfer coefficients

At a given heat flux, the heat transfer coefficients decrease from the values obtained in pure R-11 to minimum values obtained in the mixture with an inlet mass concentration of 23.1% R-11. Figure 3.12 shows that the heat transfer coefficients obtained in mixtures with small liquid mass concentrations of R-11 are generally smaller than those obtained in mixtures with larger liquid mass concentrations. This non-symmetric decrease in the heat transfer coefficient with liquid mass concentration of R-11 has been reported by other researchers using binary mixtures of different fluids. In a mixture of ethanol and water, for example, Stephan (1982) reports much lower heat transfer coefficients than the ideal values obtained by a linear interpolation between the pure component values. Figure 3.13 shows that the reduction below the ideal value is greatest at a liquid mole fraction of ethanol of approximately 15%.

Figure 3.13 Trends in heat transfer coefficients and the difference between the liquid and vapor mixture concentrations (Stephan 1982)

Stephan (1982) reviewed several heat transfer coefficient correlations and noted that the decrease in the heat transfer coefficient is most pronounced when the difference between the liquid and vapor mole fractions is largest. Heat transfer coefficients are then correlated using a wall superheat which is a function of the difference between the vapor and liquid mole fractions instead of correlating heat transfer coefficients for the pure components using only the liquid mole fraction of the least volatile component. The difference between the vapor and liquid mole fractions versus the liquid mole fraction is plotted in Fig. 3.14a for the R-11/R-113 mixture used in the present study. Also shown is a plot of the heat transfer coefficients versus the liquid mole fraction for all three surfaces at a heat flux of 60 kW/m² and a pressure of 200 kPa in Fig. 3.14b. It appears that the increase in the concentration difference between vapor and liquid R-11 does follow the same trend as the decrease in the heat transfer coefficient. Therefore, the same mechanisms which reduce the heat transfer coefficients for smooth tubes also play a major role in the enhanced surfaces, and a method of correlating the enhanced surfaces similar to that used to correlate smooth tubes might be appropriate. Specifically, Stephan and Körner (1970) suggests that the heat transfer coefficient be defined as

$$h = q'' / \Delta T_a \quad (5.35)$$

where

$$\Delta T_a = \Delta T_{id} + \Delta T_e = \Delta T_{id} (1 + \theta) \quad (3.1)$$

and

$$\Delta T_{id} = \tilde{y}'_{R113} \Delta T_{sat, R113} + \tilde{y}'_{R11} \Delta T_{sat, R11} \quad (3.2)$$

In equation (3.2), the terms $\Delta T_{sat, R113}$ and $\Delta T_{sat, R11}$ refer to the wall superheat for the pure fluids. The only unknown in equation (3.1) is θ which Stephan and Körner correlate as

$$\theta = C_{R113, R11} (\tilde{y}''_{R11} - \tilde{y}'_{R113}) \quad (3.3)$$

Figure 3.14

A comparison between (a) the difference between the liquid and vapor concentrations of R-11 and (b) experimental trends in heat transfer coefficients at $q'' = 60$ kW/m² and $P = 200$ kPa

where $C_{R_{113}, R_{11}}$ is a positive number independent of the mole fraction. Stephan and Körner (1970) obtained values of $C_{1,2}$ for various binary mixtures but the R-11/R-113 combination was not included. Alternatively, Bier et al. (1982), recommend the following correlation:

$$h/h_{id} = 1 - D_{R_{113}, R_{11}} (\tilde{y}_{R_{11}}'' - \tilde{y}_{R_{11}}')^{0.5} \quad (3.4)$$

where

$$h_{id} = h_{R_{11}} \tilde{y}_{R_{11}}' + h_{R_{113}} (1 - \tilde{y}_{R_{11}}') \quad (3.5)$$

and $D_{R_{113}, R_{11}}$ is a function of pressure only. Again, Bier et al. (1982) did not include the R-11/R-113 combination in their study.

The correlation by Stephan and Körner was chosen as representative of the types of correlations reported in the literature. In order to determine the applicability of these types of correlations, the experimental data for the smooth tube were used to determine values of $C_{R_{113}, R_{11}}$. Generally, the values ranged from 0.5 to 2.2 with an average value of 1.5. This value was used to generate predicted heat transfer coefficients which are represented by the solid lines in Fig. 3.15. The experimental data for liquid mass

Figure 3.15

Comparison of experimental values for smooth tube heat transfer coefficients and predicted values using a Stephan-Körner type correlation

concentrations of approximately 25% is generally overpredicted while the data for mass concentrations of approximately 75% is generally underpredicted, though the trends are followed well.

Experimental data for the enhanced surfaces was also used to determine values of $C_{R_{113}, R_{11}}$ but with little success. Values of $C_{R_{113}, R_{11}}$ for the High Flux tubes ranged from 1.1 to 18.6 which indicates that this type of correlation is not applicable for enhanced surfaces. A possible explanation is that the enhanced surfaces is conducive to more of a buildup of the less volatile component next to the heated surface than smooth tubes due to the inability of the free stream to affect the heat transfer process in the subsurface cavities.

3.6 Conclusions

An experimental investigation has been run to examine forced—convective, two—phase crossflow through two staggered tube bundles with three different surface geometries. Data were obtained over a range of mass fluxes, heat fluxes, pressures, and qualities. The following conclusions can be drawn from the data:

1. Heat transfer coefficients of smooth tubes are affected by changes in mass flux only at very low heat fluxes.
2. The effect of quality changes on the heat transfer coefficients on smooth tubes is negligible, except at low heat fluxes.
3. The pitch—to—diameter ratio on the heat transfer coefficient of smooth tubes is important only at low heat fluxes and large mass fluxes.
4. The enhanced—tube bundles have significantly larger heat transfer coefficients than those with smooth tubes. The values of the enhanced—bundle heat transfer coefficients are comparable to those obtained in saturated—liquid pool boiling.

5. Changes in mass flux and quality have a negligible effect on the heat transfer coefficient of the enhanced-tube bundles. It is concluded that the heat transfer process on such tubes is governed by subsurface heat transfer mechanisms because the external flow cannot intrude into the small surface cavities.

Thus, the heat transfer behavior of these types of enhanced tubes in shellside boiling has been confirmed. However, the mechanisms and their interactions governing the heat transfer of smooth tubes in bundles remain an issue for discussion and research. Likewise, the possible influence of the particular working fluid (R-113) on the heat transfer processes also must be clarified. Research is continuing on these unresolved issues.

4. FOULING OF ENHANCED TUBES IN POOL BOILING

4.1 Introduction

Although the topic of scaling from salt solutions has received a good deal of attention since at least 1924, only a handful of articles deal with the effect of precipitation fouling on nucleate boiling heat transfer and still less investigate the effect on enhanced boiling surfaces. Conflicting reports of the boiling performance of enhanced heat transfer surfaces in a fouling environment suggest a need to more systematically approach the problem.

The present work studies the effects of precipitation fouling on enhanced nucleate boiling surfaces such as those used in kettle reboilers, thermosiphon reboilers, and evaporators. Specifically two varieties of both the UOP High Flux and the Wolverine Turbo-B nucleate boiling tubes are examined. The goals of the present research are:

1. To compare the boiling performance of a fouled surface to a clean surface.
2. To observe the variation in the heat transfer coefficient with time under fouling conditions.
3. To determine the effects of heat flux on fouling.
4. To compare the results obtained with the types of surfaces examined.

Reported investigations on the fouling of enhanced transfer surfaces under nucleate pool boiling conditions appear to be limited to results presented by Gottzmann, O'Neill and Minton (1973) and by Chyu (1984). However, the work on this type of fouling with plain surfaces is comparatively much more extensive. Although surface differences may be important, it seems appropriate, in the circumstances, to review the investigations on plain surfaces as well as those on enhanced surfaces.

The basic equation of fouling is due to Kern and Seaton (1959a,b) who hypothesized that the observed rate of fouling on a heat transfer surface is the result of two competing processes: the growth and removal of a fouling deposit. That is, if $dm_f/d\theta$ is the observed rate of fouling, where m_f is the mass of deposit on unit area of the surface at any instant (θ), then

$$\frac{dm_f}{d\theta} = \dot{m}_g - \dot{m}_r \quad (4.1)$$

where $\dot{m}_g$ = rate of growth of the deposit and $\dot{m}_r$ = rate of removal of the deposit (the units of both quantities are mass per unit area per unit time).

Equation (4.1) gives the fouling effect in terms of mass (m_f) of the deposit formed on unit area of the heat transfer surface. To relate this to the fouling thermal resistance (R_f) of the deposit per unit area we have

$$R_f = \frac{m_f}{\rho_f k_f} \quad (4.2)$$

where ρ_f is the density and k_f is the thermal conductivity of the deposit. These quantities can change during the course of the fouling process, but this is usually ignored in the formulation of mathematical models of fouling.

Under boiling conditions it is reasonable to suppose that the fouling of the heat transfer surface is a consequence of the precipitation on the heat transfer surface of salts having inverse solubility properties (the solubility decreases as temperature increases in contrast to normal solubility salts, such as, sodium chloride). Presumably, examination of the deposits has suggested to previous investigators that under boiling they are sufficiently hard and tenacious, that deposit removal by hydrodynamic effects is unlikely. Consequently models of the fouling of heat transfer surfaces with boiling usually ignore the removal rate term ($\dot{m}_r$ in the Kern and Seaton model).

A comprehensive review, with particular reference to boilers, of fouling under boiling conditions up to 1929 has been published by Partridge and White (1929a,b,c), and by Partridge (1930). The earliest investigation on the mechanism of fouling deposit formation reported by Partridge is due to Couste (1854), who observed that calcium sulfate (a frequent contributor to fouling in boilers with untreated water) is an inverse solubility salt. He further proposed that local super-saturation of the solution in the vicinity of the heat transfer surface could be the cause of fouling. No further fundamental work on fouling under boiling conditions was reported until Hall (1925) presented the results of an investigation that appeared to support the hypothesis of Couste.

Partridge extended the ideas of Couste and of Hall by considering the role of steam bubbles on the surface. According to Partridge, the excess solute in the supersaturated solution is deposited as crystals in the shape of rings at the triple interface formed by the surface, the vapor and the liquid. Such rings were observed by Partridge on test surfaces exposed to boiling calcium sulfate solutions, and subsequently by Freeborn and Lewis (1962). The latter authors discuss the proposed mechanism in detail. The important role of supersaturation appears to be confirmed by experiments reported by Banchero and Gordon (1960) and by Chandler (1964).

Partridge (1930) proposed that the rate of growth of the fouling deposit could be expressed by

$$\dot{m}_g = -\kappa \left[\frac{dS}{dT} \right]^m q''^n \quad (4.3)$$

Partridge made no attempt to evaluate the empirical constants, m and n .

Partridge also measured the thermal conductivity (k_f) of the fouling deposit formed when a test surface was exposed under boiling conditions to saturated calcium sulfate solution.

The work, previously referred to, of Freeborn and Lewis (1962) extended the work of Partridge to include other hard water salts besides calcium sulfate (magnesium hydroxide, calcium hydroxide, sodium chloride) and tap water.

An important paper published by Reitzer (1964) represents the first attempt to formulate a theory for the rate of formation of fouling deposits under boiling conditions. He proposed that the rate of growth of the deposit was given by

$$\dot{m}_g = \kappa_T^n B^n (T_i - T_b)^n \quad (4.4)$$

where n was an undefined constant, presumably to be determined empirically. Under the assumptions that the temperature difference $T_s - T_b$ is constant and that there is no removal of deposit ($\dot{m}_r = 0$) he showed that the total heat transfer resistance $R_{\text{total}} (= R_c + R_f)$ at the heat transfer surface is given by

$$R_{\text{total}}^{n+1} = R_c^{n+1} + K_2 \theta \quad (4.5)$$

where R_c is the thermal resistance on the clean heat transfer surface which is assumed to be the same as that at the deposit–fluid interface (a dubious assumption).

Equation (4.4), with $n = 1$, is identical to that proposed by McCabe and Robinson (1924) for fouling in evaporators. Epstein (1979, 1981) subsequently suggested that Eq. (4.5) can be used to represent data from the fouling of heat transfer surfaces on which boiling is occurring. In particular, he proposed that $n = 1$ would apply in relatively low velocity non–stirred systems, in which the fouling process is mass transfer controlled, and $n = 2$ in high velocity and/or well agitated systems where the fouling process is controlled by the crystallization process on the heat transfer surface.

Palen and Westwater (1966) conducted pool boiling experiments with strips of electrically heated aluminum immersed in a saturated calcium sulfate solution. They measured the "wall" superheat ($= T_s - T_b$, where, in this case, T_b = boiling saturation temperature at the prevailing atmospheric pressure) as a function of time. They also measured, at the conclusion of a run, the mass of deposit formed on the surface and observed the initial form of the surface deposit.

Rings of calcium sulfate crystals, similar to those observed by Partridge (1930), were observed by Palen and Westwater on the surface at points where intermittent nucleation was detected. Where there were no boiling sites, the deposit formed initially as specks.

On the assumption that the fouling deposit was formed at a constant rate ($\dot{m}_g = \text{constant}$), Palen and Westwater used their measurements of the mass of the deposit accumulated at the end of a run to formulate the following empirical equation

$$\dot{m}_g = K_1 q^{n/2} \quad (4.6)$$

Assuming the convective resistance (R_c) at the deposit–fluid interface is constant, Eq. (4.6) can be written

$$\dot{m}_g = \frac{K_1}{R_c^2} (T_i - T_b)^2 \quad (4.7)$$

If Reitzer's Eq. (4.4) and Eq. (4.6) are identical, then $n = 2$ in Eq. (4.4) and $K_1 = R_c \kappa_f^2 B^2$.

The variation of fouling with time was investigated by Palen and Westwater by studying the variation of the "wall" superheat $T_s - T_b$ at constant heat flux when an initially clean surface was exposed to saturated calcium sulfate solution, over a total observation period of about 110 minutes. The wall superheat initially increased very

rapidly, then decreased and finally appeared to be increasing at a steady rate. This behavior was explained on the basis of the intersection between the nature of the deposit on the surface and the number of boiling sites.

Gottzmann et al. (1973) reported that a commercial enhanced heat transfer (UOP High Flux) surface, consisting of sintered metal particles, was unaffected by light fouling (a condition not defined by the authors) under boiling conditions in aqueous solutions. They attributed the resistance of these surfaces to fouling to: (a) the high liquid circulation through the porous matrix formed by the sintered particles; (b) the high capillarity of the porous matrix, resulting in wetting of the surface, which prevents local hot spots and dryout; (c) the high heat transfer coefficients resulting in surface temperatures close to the bulk liquid temperatures. To substantiate their claims, a long-term pool boiling test in concentrated seawater was carried out. They claimed that their measurements of the variation of the overall heat transfer coefficients over a test duration of 20 days indicated that no solids deposited on the surface.

The only other observation of fouling on an enhanced surface was reported by Chyu (1984) on a UOP High Flux porous type enhanced surface which was apparently accidentally contaminated with a cleaning compound. Chyu noticed a substantial degradation in the pool boiling performance of the surface as a consequence of the assumed surface contamination.

The conflicting reports of Gottzman et al. and of Chyu, and a consideration of the surface structure of enhanced heat transfer surfaces, which appear to be potentially sensitive to fouling, suggest a need to carry out a systematic investigation of fouling on this type of surface under nucleate boiling conditions.

4.2 Experimental Apparatus and Procedure

Single-tube nucleate pool boiling experiments were conducted in a saturated aqueous solution of calcium sulfate at atmospheric pressure. Samples of the following

commercially available enhanced heat transfer surfaces were used in the tests: UOP High Flux R-113 Large Pore, High Flux R-113 Small Pore, Wolverine Turbo-B Very Open, Turbo-B Very Closed. In addition a smooth copper surface was employed. Tests were carried out at two heat fluxes. The effects of precipitation fouling were investigated by measuring the change in the tube wall superheat with time and the weight of the fouling deposit.

All test sections were copper tubes with a heated length of 101.6 mm, an outer diameter of 19.05 mm, and an inner diameter of 6.35 mm. These test sections were given the following designations:

<u>Complete Tube Name</u>	<u>Abbreviated Designation</u>
Plain, Smooth	Plain
UOP High Flux R-113 Large Pore	Large
UOP High Flux R-113 Small Pore	Small
Wolverine Turbo-B Very Open	Very Open
Wolverine Turbo-B Very Closed	Very Closed

The plain test section had a smooth surface polished with 600 grit paper. The High Flux surfaces consisted of a porous layer of copper particles with an approximate layer thickness of 0.381 mm. The particle sizes differed for the Large Pore and Small Pore surfaces [see Curcio (1989) for details]. the Wolverine surfaces were manufactured by finning followed by successive cross-grooving and rolling operations, giving the general appearance of a grid of rectangular flattened blocks. The blocks of the "Very Closed" surface appeared to be more flattened than those of the "Very Open" surface.

Power was supplied to the test section by means of an electric resistance cartridge heater press-fit into the test section. Voltage to the heater was adjusted using a variable transformer. The power input to the heater was determined by measuring the voltage across the heater and the current through a precision shunt in series with the heater. The heat flux to the test section was computed by dividing the power input to the heater by the nominal surface area of the test section. It was assumed that both an enhanced surface and

a plain smooth surface with equivalent outer diameters and lengths had equivalent surface areas.

Temperatures were measured using copper–constantan thermocouples referenced to an electronic ice point. Four thermocouples were located inside the tube wall at 12/3/6/9 o'clock at the midpoint of the heated length. Two additional thermocouples were located in the pool, one located approximately 51 mm under the test section and the other approximately 3.2 mm above the surface of the test section in the vapor stream. Conversion of thermocouple voltages to temperatures was accomplished using the National Bureau of Standards curve fit (National Bureau of Standards, 1979).

The apparatus consisted of two rectangular glass tanks, one placed inside the other (Fig. 2.1). The inner tank (238 × 181 × 324 mm) contained the pool of saturated aqueous calcium sulfate solution and the test section; the outer tank (308 × 308 × 305 mm) was filled with ethylene glycol. An immersion heater was placed in the outer jacket and was kept at a constant temperature of about 130° C by means of a temperature controller. A 6.35 mm thick polycarbonate lid was clamped in place on the top of the inner tank. A strip of closed-cell sponge rubber placed under the lid sealed the tank. Several holes were drilled in the lid to allow access to the pool for instrumentation. Glass wool enclosed the entire apparatus.

The test section was mounted on a polycarbonate plate using a spring clip (see Fig. 2.2). The polycarbonate plate was attached to the lid. To prevent leakage of solution into the test section, a Buna–N cap sealed one end of the test section while a protection tube (attached to an elbow) carried the heater and thermocouple wires to the lid of the inner tank. A copper coil condenser was located beneath the lid while an open glass condenser was placed in the lid to provide atmospheric pressure in the vapor space above the pool. Tap water was the cooling fluid for both condensers.

The following procedure recommended by Palen (1965) was used to obtain a saturated, aqueous solution of calcium sulfate:

1. Dissolve 25.7 g of reagent grade calcium sulfate dihydrate ($\text{CaSO}_4 \cdot 2\text{H}_2\text{O}$) in 10 L of distilled water.
2. Boil of approximately 1/4 of the total volume.
3. Allow solution to stand and cool overnight.

Since precipitate was always present at room temperature, the solution was also saturated when again heated to the boiling point.

Pool boiling experiments were conducted in the following manner:

1. Turn on tap water to the condensers (approximately 0.1 L/min).
2. Heat the pool to the saturation temperature of pure water (as predicted from the pressure at the tube axis) by means of the temperature controlled outer jacket of ethylene glycol.
3. Adjust the tap water flowrate so that a small amount of vapor condenses inside the glass condenser.
4. Degas the pool and test section for about one hour by boiling from immersed cartridge heaters in the pool. Remove the heaters at the end of the period.
5. Seal all openings to the pool, except for the glass condenser.
6. Measure and record the atmospheric pressure, the height of the liquid above the tube centerline, the room temperature and the temperature of the outer jacket.
7. Measure and record the thermocouple voltages at fixed intervals of time.
8. Apply power to the test section with the AC variable transformer.
9. Measure and record the heater and shunt voltages at fixed intervals.
10. Steps 6 and 8 continued until a general trend is evident, typically between 15 and 24 hours.

Steps 1 through 6 took about six hours. On rare occasions, saturated solution was added to the pool when the liquid level decreased by more than about 50 mm.

At the end of a run, the specimen (containing the fouling deposit) was boiled vigorously in a pool of pure distilled water for about 30 minutes to remove the fouling deposit. It was then placed in a second pool of pure distilled water and again boiled vigorously for another 30 minutes. EDTA titration (Water Pollution Control Federation, 1962) of the water samples provided a measure of the amount of fouling deposit (assumed to be only calcium sulfate anhydrite) removed from the specimen.

The saturation temperature of pure water was obtained via a curvefit (Curcio 1989) of the ASME (1967) steam tables based on the absolute pressure at the axis of the test section. This pressure was determined by measuring the atmospheric pressure of the laboratory with a mercury barometer and the height of liquid above the test section to account for the hydrostatic pressure of the pool. The tube wall superheat, ΔT , was defined as the difference between the average of the four wall thermocouples and the saturation temperature of pure water:

$$\Delta T = \bar{T}_w - T_{\text{sat}} \quad (4.8)$$

It was decided to employ the saturation temperature of pure water as the reference value for heat transfer when it was discovered that the pool boiling experiments in distilled water were conducted under slightly sub-cooled conditions. The average measured bulk temperature for these runs was 99.7°C whereas the average saturation temperature predicted from the pressure was 100.3°C . To be consistent, the results of the tests under fouling conditions were also referred to the pure water saturation temperature.

The change in wall superheat due to fouling ($\Delta T_{\text{fouled}} - \Delta T_{\text{clean}}$) is taken to be the difference between the wall superheat at twenty hours in the fouling experiment and the clean value obtained in a separate pure water test at the same heat flux. When calculating the mass of the deposit on the surface (by titration), the deposit was assumed to be calcium sulfate anhydrite (Ca SO_4) only.

A propagation of uncertainty analysis was performed and Table 4.1 shows the estimated precision of a typical data point. Details of this analysis can be found in Curcio (1989).

Table 4.1 Precision of Measured Quantities in Fouling Experiments

Measured Quantity	Nominal Value	Precision	Percent of Nominal Value
q''	30,000 W/m ²	± 1581 W/m ²	5.3
ΔT	2.0°C	± 0.4 °C	17.4
$\Delta T_{\text{fouled}} - \Delta T_{\text{clean}}$	2.0°C	± 0.5 °C	24.6
Deposit mass	177 mg	± 2.84 mg	1.6

4.3 Experimental Results

The results of the experiments are given in Figs. 4.1–4.5. These show the wall superheat, defined as the difference between the saturation temperature for distilled water obtained from the pressure at the tube axis and the average wall temperature (see above), as a function of the time of exposure of an initially clean surface to a saturated solution of calcium sulfate with heat transfer by boiling at the test surface. Measurements were made at two heat fluxes (10 kW/m² and 80 kW/m²).

The dashed horizontal line in Figs. 4.1–4.5 shows the wall superheat under clean conditions. This was obtained from interpolation on the results of a series of boiling heat transfer tests carried out in pure distilled water prior to the fouling tests.

All the surfaces tested, except for the High Flux Small surface exhibited an increase in the wall superheat with time at the higher heat flux. In general, the superheat was seen

Figure 4.1 The variation of wall superheat with time of a plain smooth surface in a saturated aqueous solution of calcium sulfate at two heat fluxes

Figure 4.2 The variation of wall superheat with time of High Flux large surface in a saturated aqueous solution of calcium sulfate at two heat fluxes

Figure 4.3 The variation of wall superheat with time of a High Flux small surface in a saturated aqueous solution of calcium sulfate at two heat fluxes

Figure 4.4 The variation of wall superheat with time of a Turbo-B very open surface in a saturated aqueous solution of calcium sulfate at two heat fluxes

Figure 4.5 The variation of wall superheat with time of a Turbo-B very closed surface in a saturated aqueous solution of calcium sulfate at two heat fluxes

to increase within the first five hours of exposure or less. This very rapid initial fouling may account for the observation by Gottzmann, O'Neill and Minton (1973) that no fouling occurred on the enhanced surface used in their tests. If observations were made at infrequent intervals, as is likely in the industrial situation where their tests were made. Then an initial rapid increase in fouling could be overlooked.

At the lower heat flux, only the smooth surface gave a statistically significant increase in the wall superheat over the time of exposure. Examination of Figs. 4.1–4.5 shows that the results from the enhanced surfaces could be interpreted as showing "negative fouling" because the superheat under fouling conditions is lower than the clean value. These differences are not considered to be statistically significant and merely represent an artifact associated with unavoidable random variations in the observed wall superheat. To assist in this assessment, error bars corresponding to an estimate of one standard deviation (Curcio 1989) of the statistical distribution of the measured wall superheats are shown in Figs. 4.1–4.5. Overlap of these error bars for corresponding sets of

fouled and clean data is taken to indicate a statistically insignificant difference between the observed wall superheats under clean and fouled conditions.

The very small observed effects of fouling at the low heat flux may also provide an alternative explanation for the observations of Gottzmann, O'Neill and Minton (1973) noted above. In other words, their tests may have been carried out below the threshold heat flux at which significant fouling effects are observed.

Both the Wolverine Turbo-B surfaces appeared to give negative superheat values at the lower heat flux (Figs. 4.4 and 4.5). These surfaces operated with exceptionally small superheat values under the clean condition. In fact, at a heat flux of 10 kW/m^2 the superheat value on the Very Closed surface is of the same order of magnitude as the uncertainty of the measurement, and in the case of the Very Open surface the superheat value is probably buried in the noise of the measurement (measurements on the clean surface were not taken at such low heat fluxes with this surface). It appears that at a heat flux of 10 kW/m^2 the superheat on the Wolverine Turbo-B surfaces that were tested is immeasurably small under both clean and fouled conditions.

It will be noted that in the case of the plain smooth surface (Fig. 4.1) that by the criterion given above, there is a statistically significant difference between the clean wall superheat when measured in distilled water (star and circle symbols) and when it is initially measured in the saturated calcium sulfate solution (triangle and square symbols). This is believed to be a consequence of the procedure used to degas the solution prior to the commencement of a test. This required the test surface to be immersed in the solution for an hour before the actual run began (see above). Fine white crystals, probably mobilized from those lying in the tank bottom (required to ensure the solution is saturated) by the vigorous motion in the tank induced by the immersion heaters used in the degassing procedure, were observed to deposit on the upper surface of the test specimens.

The significant difference in the initial superheats with the plain surface was not observed with the enhanced surfaces (Figs. 4.2–4.5). The only reason for this difference is

behavior that can be advanced at this time is associated with the observed effect of fouling being greater on the plain surface than on the enhanced surfaces. this difference in behavior, as indicated by the greater fouling of the plain surface, could just as well be present on the first exposure of the surface to fouling conditions as at some later time.

The difference between the wall superheat at the conclusion of a fouling test and the wall superheat measured on the same surface at the same heat flux when it is clean, was compared with the measured amount of material on the surface removed by boiling at the end of each test, as described above. All the surfaces showed significant amounts of material had been deposited on the surface. In addition it appears that the amount of deposit increases with the wall superheat. This is in agreement with the model of Reitzer (1964) described above.

A further important observation is that surfaces with the lower wall superheat under clean conditions appear to exhibit less fouling than surfaces with higher clean wall superheats. Since the clean wall superheat depends on the structure of the enhanced heat transfer surface, then clearly the fouling tendency of an enhanced surface in pool boiling conditions depends on the form of the enhancement.

The importance of actually measuring the amount of deposited material is demonstrated by the results of visual observations of the surface during the test. The plain smooth surfaces showed a thin white scale, which was found to be strongly adherent after the surface was removed from the tank. Finely divided white crystals were deposited on the surface of the enhanced tubes, but these were only visible after the specimen was removed from the tank.

The fouling curves shown in Figs. 4.1–4.5 show, with the exception of Fig. 4.4, an asymptotic behavior. That is, the fouling increases when the clean surface is first exposed to the calcium sulfate solution, but the rate of change of the wall superheat gradually decreases and eventually goes to zero. This is different from the behavior usually observed with precipitation fouling under nonboiling conditions. For example, data for precipitation

fouling of calcium sulfate presented by Ritter (1983) exhibits linear, constant rate, fouling, this would correspond to no removal of the deposit once it is formed.

Assuming no deposit removal, the results shown in Figs. 4.1–4.5 suggest one of three possibilities. Either the precipitation at the deposit–fluid interface has ceased, or the deposit thermal resistance and the convective thermal resistance at the deposit–fluid interface are simultaneously increasing and decreasing at about the same rate, or, if deposit removal processes are present, then the rate of growth of the deposit ($\dot{m}_r$) is equal to the rate of removal ($\dot{m}_r$) of the deposit. It is possible that the high temperature gradients in the deposit, which could promote spalling, and the vigor of the fluid motion, due to the boiling, in the vicinity of the test surface, could have promoted deposit removal. Tests extending over a much longer duration than the 20 hours used here would be required to evaluate this hypothesis. Neither time nor the limited number of available test surfaces permitted such extensive tests.

4.4 Physical Modelling of the Fouling Processes on Enhanced Heat Transfer Surfaces Under Pool Boiling Conditions

4.4.1 Introduction

In the investigation described in sections 4.1–4.3, it was noticed that when pool boiling saturated aqueous calcium sulfate solution from UOP High Flux and Wolverine Turbo–B surfaces that the fouling thermal resistance (R_f) was significantly smaller than that measured on a plain surface under comparable conditions. Similar observations had been made on High Flux surfaces by Gottzmann et al. (1973) both with aqueous solutions and other fluids. They ascribed this to the vigorous fluid motion in the cavities formed by the sintered metal which constitutes the heat transfer surface. Possibly similar conditions prevail in the cavities of the Turbo–B surface.

Direct observation of the processes occurring in the surface cavities is not possible because of their very small size. In these circumstances, the use of enlarged models of the two surfaces seemed appropriate. It was recognized that the results would probably be qualitative because the appropriate scale-up rules relating the dimensions of and physical conditions in the model and in the prototype (real) surfaces were unknown.

4.4.2 The Model Surfaces and Experimental Procedure

Two-dimensional models, approximately 100 times larger than the prototype, were constructed for both the High Flux and Turbo-B surfaces. Essentially they each consisted of an aluminum block (4 in. $\times$ 5 in. $\times$ 1.5 in.) that was heated electrically by two cartridge heaters, inserted in horizontal, cylindrical cavities, and two pieces of glass or Hyzod (transparent plastic) (5 in $\times$ 4.5 in.), about 0.125 in. thick and separated by a distance of 0.5 in. The glass or plastic sheets were mounted parallel to one another on the upper surface of the aluminum block so that they formed a narrow cavity with transparent walls (see Fig. 4.6). The cavity was filled with saturated aqueous calcium sulfate solution. When sufficient energy was supplied to the cartridge heaters, the solution boiled on the surface of the block at the bottom of the cavity. Two devices of this general form were made. One had a plane surface at the base of the cavity (The other had a fin that stood clear of the upper surface of the block. "Keyholes", roughly representing the form of the Turbo B surface as observed in a metallographic section of the surface (Fig. 4.7(a)) were machined in the fin.

Ball bearings, of various sizes, were placed in the cavity with the plane base to simulate the form of the High Flux surface when viewed in a metallographic section (Fig. 4.7(b)). Experience showed that the ball bearings were not entirely satisfactory because they moved when the liquid was boiling and because their three-dimensional form made it difficult to detect any fouling deposits that might form. A second version employed aluminum machining 'shavings.' These were packed in the cavity at a sufficiently low

Figure 4.6

Apparatus used in simulation of UOP High Flux surface. Note this shows the earliest version of this apparatus in which the sintered metal particles were simulated by ball bearings (not shown) of various sizes. In the latest version randomly distributed cylinders, providing a number of conductive heat paths from the lower heated surface, are mounted in the test chamber.

Figure 4.7(a)

Comparison of metallographic section of the Wolverine Turbo-B surface and the model surface

Figure 4.7(b) Comparison of metallographic section of the High Flux surface and the model surface using ball bearings of various sizes

density that numerous cavities allowed direct observation through the shavings from one side of the cavity to the other. These remained stationary under boiling conditions, but their three dimensional form and surface texture did not permit fouling deposits to be detected.

The current version of the High Flux surface model consists of a number of aluminum cylinders of various sizes arranged in a random pattern that simulates the prototype surface. These are located in depressions in the clear plastic sidewalls of the cavity, so they do not move. Their two-dimensional form allows the formation of fouling deposits to be observed with appropriate illumination.

The test chamber is mounted on an optical bench and illuminated by a combination of front and back lighting. This particular lighting arrangement was developed in the course of the research program, and was found to ensure that the fouling

deposit formed a clear, 'silhouette' image on photographic film, which allowed its dimensions to be measured without difficulty.

The temperatures at various points in the aluminum block are measured by thermocouples. The records of the variation of temperature with time are correlated with the data obtained from the photographs.

Since the tests could run for durations up to eight hours, the calcium sulfate solution was continuously replenished by a syphon.

A typical test involved boiling the calcium sulfate solution for experimental durations up to eight hours. System temperatures, and electrical power were recorded, and photographs were taken at intervals of approximately one hour during a test. The power was temporarily decreased while the photographs were being made, in order to suppress the bubbles that would otherwise interfere with the observation of the fouling deposits.

In order to obtain quantitative data on the formation of the fouling deposit, negatives of the photographs were projected on an enlarged scale on white paper and the form of the surface and the deposit traced onto the paper. To ensure that successive projected images are correctly located and that the magnification was identical in all cases, a fiduciary mark was arranged in the field of view. The sequence of recorded projected images showed the form and growth of the fouling deposit with time.

4.4.3 Experimental Results

Wolverine Turbo-B surface. Typical results for the Wolverine Turbo B surface are shown in Fig. 4.8. From Fig. 4.8 it will be noticed that most of the fouling deposit is formed on the cylindrical surfaces of the cavity, with a little on the upper, plane surface. This suggests that the observed small effect of fouling on the performance of the prototype surface is due to either or both of the following causes:

- (a) As the thermal resistance increases on the portions of the surface that are fouled, the heat flow decreases on these sections of the surface, but increases on other portions of the surfaces where the deposit is lighter or non-existent. This would result in a change in the temperature distribution in the metal, which would provide a means for confirming this hypothesis.
- (b) The roughness of the interface between the fouling deposit and the liquid provides more boiling sites than are provided by the clean surface. This increase in the bubble formation rate compensates to some extent for the additional thermal resistance introduced by the fouling deposit. It was noticed that the presence of a fouling deposit on the surface of the model resulted in bubbles of a smaller size, and that the rate of release of these bubbles was higher than from the clean surface.

Plots of the temperatures from the eight thermocouples for a test duration of about eight hours showed an initial rapid increase in temperature due to the combined effects of fouling and the heating-up of the heavy aluminum heater block.

After this initial period, the temperature increased at a rate of about 0.25°C per hour.

UOP High Flux surface. The data obtained with the model of the High Flux surface is not as extensive as that provided by the Turbo-B model. This is a consequence of the difficulties experienced in obtaining a satisfactory simulation of this type of surface.

A heavy deposit was noted on the bottom surface of the test chamber. Much lighter deposits, which could not be detected on photographs, but could be seen by eye, were formed on the cylinders. This distribution of deposit suggests that, as in the Turbo-B surface, there could be a redistribution of the temperature field in the heated surface. In addition, or alternatively, the fouling could increase the effectiveness of the boiling heat transfer process. No specific observations were made on this last point, so this can only be treated as a hypothesis at this stage.

The variation with time of the temperatures in the heated block is shown in Fig. 4.9. This was obtained in a run of 24 hours total duration, made up of three segments of 10, 7 and 7 hours, respectively, because the tests were not conducted overnight. The temperatures increased very rapidly in the first 20–25 minutes of operation, this was a

Figure 4.8 Plots of the fouling deposit on model of Wolverine Turbo-B surface

Figure 4.9 Temperature records made with the High Flux surface model. The plot is the average reading of the eight thermocouples in the aluminum heater block, with the room temperature subtracted from the readings in order to smooth out temperature fluctuations introduced by the variation in the laboratory temperature.

consequence of both fouling effects and the thermal response of the heavy aluminum heater block. The rate of change of temperature then decreased to about 1.2°C per hour for the next 10 hours, for the following 6 hours the rate of change was about 6°C per hour, and for the final seven hours of exposure the temperature increased at about 2°C per hour.

4.5 Conclusions

The results obtained in these experiments indicate that under nucleate pool boiling conditions, the amount of precipitation fouling on both plain smooth surfaces and the tested enhanced surfaces increases with the heat flux. The results also indicate that there is a threshold value of heat flux that must be exceeded before significant fouling effects are seen on enhanced surfaces. Most of the fouling occurs within the first five hours, or less, of the exposure of the surface to fouling conditions. The measured amount of material deposited and the increase in the wall superheat as a consequence of fouling, were both smaller on the enhanced heat transfer surfaces than on the plain smooth surface. In other words the enhanced heat transfer surfaces tested are not so susceptible to precipitation fouling under pool boiling conditions in a saturated calcium sulfate solution as is the plain smooth surface. Furthermore, the lower the wall superheat observed on an enhanced heat transfer surface under clean conditions, the lower was the amount of fouling material deposited on the surface. Since wall superheat under clean conditions is closely related to the geometry of the surface enhancement, this observation suggests that the effect of precipitation fouling under nucleate pool boiling conditions is related to the type of surface.

Observations with simulated Wolverine Turbo-B and UOP High Flux surfaces suggest that enhanced surfaces of this type experience only a slight degradation in performance due to fouling because deposits are restricted to certain portions of the surface, or because the fouling induces more vigorous boiling or both. The non-uniform formation of the fouling could result in a redistribution of temperature and heat flux in the heat transfer

surface that compensates to some extent for the effects of the fouling. Measurements of the temperature distribution in the heat transfer surface should provide confirmatory evidence of this observation. Tests to provide this information are currently underway.

5. POOL BOILING HEAT TRANSFER IN REFRIGERANT–OIL MIXTURES ON ENHANCED HEAT TRANSFER SURFACES

5.1 Introduction

The results of a substantial body of research work are available on the pool boiling of plain and enhanced heat transfer surfaces in pure liquids. This has been reviewed by Yilmaz and Westwater (1981). However refrigeration systems with oil–lubricated compressors typically contain a mixture of oil and refrigerant as the working fluid. The oil is generally less than 10 per cent by mass, but it alters the thermodynamic and transport properties of the refrigerant (Baustian et al., 1986). As a result, the heat transfer performance of the components of the system (condensers and evaporators) may be adversely affected. In spite of its importance to refrigerator engineering, the topic of boiling on plain and enhanced heat transfer surfaces with oil–refrigerant mixtures has received much less attention (Nakayama et al., 1979) than the same process with pure liquids. This has prevented the development of generalized and reliable correlations.

The work reported here was intended to evaluate the performance of various enhanced heat transfer surfaces in single–tube pool boiling of R–113/Suniso 3 GS oil mixtures. It is recognized that refrigerants R–12 and R–22 are more widely used than R–113, furthermore it is appreciated that the fluorocarbon based refrigerants, such as R–12, R–22 and R–113, are about to be replaced by refrigerants that are environmentally more benign. Yet the fundamental information developed in this research program should be useful, in at least a qualitative sense, to the designers of heat transfer equipment in which boiling mixtures of lubricating oil and refrigerants are present. Furthermore, the experimental techniques developed in the research described here should be useful for studying boiling heat transfer from enhanced heat transfer surfaces in mixtures of the new refrigerants and oil.

5.2 Literature Survey

Two opposing effects on the heat transfer coefficient have been observed in refrigerant-oil mixtures (Sauer et al. 1980, Stephan 1963). First, as the refrigerant evaporates at the boiling surface, it is speculated that an oil-rich layer develops near the surface. Since the liquid-vapor surface tension and surface adhesive tension are greater in this oil-rich layer, more energy is required for the formation and growth of bubbles (Stephan 1963). Also, increased liquid viscosity reduces turbulence, decreasing liquid flow to the surface resulting in a lower heat transfer coefficient (Sauer et al., 1980). A second effect sometimes observed is the enhancement of the heat transfer coefficient at small oil concentrations. This effect is attributed to foaming which results in more vigorous refrigerant motion around the tube, augmenting the heat transfer (Stephan (1963); Chaddock (1976)) presents a detailed discussion on the performance of evaporators with refrigerant-oil mixtures.

This paper is concerned with the performance of externally enhanced heat transfer surfaces in refrigerant-oil mixtures. One could anticipate significantly larger concentrations of oil in the plentiful re-entrant cavities of an enhanced surface than in the bulk fluid. As a result, the effect of oil on an enhanced surface may be more pronounced than on a smooth surface.

Gottzmann et al. (1973) presented data on the effect of compressor oil on the boiling performance of the High Flux surface in propylene. They reported about a 3% addition to the overall heat transfer resistance at a concentration of 4% oil by volume. They claimed that upon removal of the oil, the clean values were restored.

Sauer et al. (1980) investigated the performance of low finned tubing in an R-11/naphthene based oil mixture. Two oils at various concentrations were studied. They observed decreased heat transfer coefficients at higher oil concentrations and an increase in the saturation temperature at greater oil concentrations. The effect of oil, though, was no greater for the finned tube than for the plain tube. They concluded that an

increase in oil in R-11 up to 3% by mass does not significantly affect the heat transfer coefficient and may increase it slightly. For oil concentrations greater than 5%, the heat transfer coefficient was significantly reduced.

Stephan and Mitrovic (1981) studied boiling of R-12/Shell Clavus G68 oil mixtures from Gewa-T tube bundles. They observed a decrease of at most 20% in the boiling heat transfer coefficient at a concentration of 6% oil by mass. At higher concentrations, substantial degradations in performance were observed.

Wanniarachchi et al. (1986) made the following observations with a UOP High Flux surface boiling in R-114/oil mixtures. With pure R-114, the High Flux tube resulted in an enhancement of the boiling heat transfer coefficient of about ten times that of a smooth tube. With up to 3% oil by mass, a 35% reduction in the heat transfer coefficient for the enhanced surface was observed at all heat fluxes. At concentrations of 6% and higher with heat fluxes greater than 30 kW/m^2 , the performance of the enhanced tube substantially decreased. Also, the effect of oil was observed to be more severe with the High Flux surface than with the smooth surface. The boiling heat transfer coefficient of the High Flux tube was still about seven times that of a smooth tube with up to 10% oil over the practical operating range of heat fluxes (10 kW/m^2 to 40 kW/m^2).

McManus et al. (1987) evaluated both a spirally-roped High Flux tube and a Wolverine Turbo-B tube in R-114/oil mixtures. They found at a heat flux of 40 kW/m^2 with pure R-114, the boiling coefficients of the High Flux and Turbo-B tubes were 14.6 and 6.4 times, respectively, those of a smooth tube. These ratios (commonly called enhancement ratios) decreased to 7.0 and 4.9 respectively at 6% oil by mass. The rate of degradation with increasing oil concentration was seen to be less pronounced for the Turbo-B tube than for the High Flux tube.

Wanniarachchi et al. (1987) observed the Gewa-T, Thermoexcel-E and -HE tubes in R-114/York Oil C mixtures. Comparing their results with those obtained previously by Wanniarachchi et al. (1986), they concluded that the Thermoexcel-E and -HE tubes

performed better than the other tubes studied (High Flux, Gewa-T, and plain) for heat fluxes below 18 kW/m^2 . The High Flux tube showed the best performance at higher heat fluxes up to 6% oil by mass. At heat fluxes greater than 45 kW/m^2 with 6% or more oil, the Gewa-T tube gave the best performance.

Murphy (1987) compared the Wolverine Turbo-B surface to 26 fpi (1024 fpm) and 40 fpi (1575 fpm) finned tubes in R-114/oil mixtures. He claimed the Turbo-B tube performed the best for both the pure and mixture situations up to 10% oil by mass. The boiling coefficient of the Turbo-B tube was also observed to be relatively insensitive to heat flux at various oil concentrations. Comparing his results with Wanniarachchi et al. (1986), he noted that the Turbo-B tube provided about the same enhancement at a heat flux of 30 kW/m^2 as the High Flux tube. For lower heat fluxes, the Turbo-B tube performed better than the High Flux tube while the reverse was true at higher heat fluxes and oil concentrations less than or equal to 6%.

Grant et al. (1989) examined the UOP High Flux tube in R-113/Shell Clavus 68 oil mixtures. They reported an enhancement ratio between two (at 80 kW/m^2) and three (at 10 kW/m^2) for pure R-113. The High Flux tube was more affected than the plain tube at low oil concentrations. The rate of degradation decreased with increasing oil concentrations (up to 10% by volume). A 40% reduction in the heat transfer coefficient was observed at 10% oil and a flux of 80 kW/m^2 . Also, the researchers could not restore the performance of the High Flux tube in pure R-113 to the original values.

Marvillet (1989) studied a copper finned tube (26 fpi) and an aluminum plasma sprayed tube in R-12 and R-22 mixtures with Mobil oil SHC 234 and 326. The R-12/SHC 234 mixture is completely miscible while the R-22/SHC 326 mixture is partially miscible. He concluded that in the R-12/SHC 234 mixture at low heat fluxes (about 20 kW/m^2), the plasma sprayed tube out-performed the finned tube over the range of oil concentrations (0 to 16% by volume). At a high heat flux (40 kW/m^2), the plasma sprayed tube still offered better performance but for a smaller range of concentrations (0 to

4%). The original performance was not restored with the plasma sprayed tube in pure R22. This was not true for the case with the completely miscible mixture of R12/SHC 234.

Table 5.1 lists values of the enhancement ratio at a heat flux of 30 kW/m^2 for various enhanced surfaces and oil concentrations reported in Wanniarachchi et al. (1986, 1987) and Murphy (1987). The surfaces with the greatest enhancement for the pure refrigerant case generally suffer the largest degradation with the presence of oil. Wanniarachchi et al. (1987) sketch the probable mechanism present. As the mixture boils inside the re-entrant cavities, tunnels, or pores, the more volatile refrigerant evaporates, leaving behind the less volatile oil. Since a larger concentration of oil exists locally inside the cavities than in the bulk mixture, oil then diffuses back to the bulk liquid. At a given bulk concentration and heat flux, steady-state conditions are reached when the flow of oil into the cavities is exactly balanced by the diffusion of oil to the bulk. An oil-enriched layer provides both a thermal and mass transfer resistance. As a result, the boiling performance is degraded.

The boiling heat transfer coefficient is also seen to decrease with increasing heat flux at a given oil concentration. At higher fluxes, the refrigerant evaporates at a greater rate, increasing the rate at which oil is left behind. Since the diffusion of oil back to the bulk must increase, the local oil concentration is necessarily higher.

The local oil concentration inside the cavities is also expected to increase as the cavity size decreases. A greater resistance to mass transfer exists in smaller passages. In addition, as the cavities become smaller, the effects of liquid convection within the pores are reduced. Thus, one would predict the UOP High Flux surface to suffer the greatest degradation in performance. Table 5.1 supports this prediction.

Table 5.1 Enhancement Ratios ($\Delta T_p/\Delta T_e$) in R-114/Oil Mixtures at a Heat Flux of 30 kW/m² (Wanniarachchi et al., 1986, 1987, Murphy 1987)

Surface	Oil Concentration [Ⓢ]		
	0%	3%	10%
UOP High Flux (Wanniarachchi et al., 1986)	9.1	7.5	5.5
Hitachi Thermoexcel-E (Wanniarachchi et al., 1987)	8.2	5.5	3.8
Hitachi Thermoexcel-HE (Wanniarachchi et al., 1987)	6.8	5.5	3.8
Wolverine Turbo-B (Murphy 1987)	5.2	5.0	4.7
Wieland Gewa-T (Wanniarachchi et al., 1987)	4.4	3.8	2.9
40 fpi* (1575 fpm) (Murphy 1987)	3.8	3.5	3.2
26 fpi* (1024 fpm) (Murphy 1987)	2.8	2.6	2.6

[Ⓢ] Only Wanniarachchi et al., (1987) specifies York Oil C; no other references specify oil used

* fpi = fins per inch, fpm = fins per meter

5.3 Experimental Approaches and Procedure

Single-tube nucleate pool boiling experiments were conducted in R113/oil mixtures at atmospheric pressure. Samples of the following commercially available enhanced heat transfer surfaces were used in the tests: UOP High Flux R-113 Large Pore, High Flux

R-113 Small Pore, Wolverine Turbo-B Very Open, Turbo-Very Closed. In addition, a smooth copper surface was employed. Pool boiling curves (heat flux vs. wall superheat) were generated for 0%, 1%, 5%, and 10% mass concentrations of 3GS (155 SSU) oil in R-113. Maximum heat fluxes of $60,000 \text{ W/m}^2$ were achieved.

All test sections were copper tubes with a heated length of 101.6 mm, an outer diameter of 19.05 mm, and an inner diameter of 6.35 mm. These test sections were given the following designations:

Complete Tube Name	Abbreviated Designation
Plain, Smooth	Plain
UOP High Flux R-113 Large Pore	Large
UOP High Flux R-113 Small Pore	Small
Wolverine Turbo-B Very Open	Very Open
Wolverine Turbo-B Very Closed	Very Closed

The plain test section had a smooth surface polished with 600 grit paper. The High Flux surfaces consisted of a porous layer of copper particles with an approximate layer thickness of 0.381 mm. The particle sizes differed for the Large Pore and Small Pore surfaces [see Curcio (1989) for details]. The Wolverine surfaces were manufactured by finning followed by successive cross-grooving and rolling operations, giving the general appearance of a grid of rectangular flattened blocks. The blocks of the "Very Closed" surface appeared to be more flattened than those of the "Very Open" surface. The apparatus used was as described in Section 4.2.

Pool boiling experiments were conducted in the following manner, with steps 1-7 identical to those described in section 4.2.

8. Measure and record the thermocouple voltages upon stabilizing.
9. Apply power to the test section with the Variac.

10. Measure and record the heater and shunt voltages.
11. Repeat steps 7–9, increasing the voltage in about 10 volt increments to the maximum desired heat flux (typically $5.8 \times 10^4 \text{ W/m}^2$).
12. Repeat steps 7–9, decreasing the voltage in about 20 volt decrements to zero power.
13. Repeat step 6.

Steps 1–6 took approximately three hours while steps 7–12 required another two hours.

After each mixture run, the surface was soaked in de-natured alcohol for at least 48 hours to clean it.

The heat flux to the test section was computed by dividing the power input to the heater by the nominal surface area of the test section. It was assumed that both an enhanced surface and a plain smooth surface with equivalent outer diameters and lengths had equivalent surface areas.

The temperature of the bulk pool was obtained from the thermocouple located approximately 3.2 mm above the surface of the test section in the vapor stream. An average bulk temperature for the entire run was calculated by averaging the bulk temperatures for each data point ($\bar{T}_b$). The average temperature of the outer tube surface was determined from the four thermocouples located inside the tube wall, accounting for one-dimensional conduction through the wall ($\bar{T}_w$). The temperature difference between the surface and the bulk pool, known as the wall superheat, is then:

$$\Delta T = \bar{T}_w - \bar{T}_b.$$

The enhancement ratio (e) is used to compare the performance of an enhanced surface to a plain surface at a given heat flux and oil concentration, it is defined as:

$$e = \Delta T_p / \Delta T_e.$$

To compare the effect of oil concentration for a given surface at a given heat flux, a ratio of boiling coefficients is computed by:

$$h''/h' = \Delta T' / \Delta T''$$

where the equality applies because the heat flux is the same in both cases.

Table 5.2 lists the precision of the measured quantities discussed above for a typical data point.

5.4 Results and Discussion

Figures 5.1 to 5.5 present pool boiling curves for the five surfaces examined at four mass concentrations of 3GS (155 SSU) oil. It is important to note that the temperature difference is referenced to an average over time of the measured bulk temperature (from the thermocouple located 1/8" above the test section) and not on a saturation temperature predicted from the pressure at the axis of the test section. The primary motivation for choosing this *in situ* temperature was the lack of any data on the saturation temperature of refrigerant-oil mixtures in general or for R-113 at various mass concentrations of 3GS oil in particular. The pure R-113 data were also based on a measured bulk temperature to be consistent. It was observed that the measured bulk temperature increased with increasing oil concentration.

The effect of oil concentration on a particular surface may be observed by comparing the various pool boiling curves of that surface (Figs. 5.1 to 5.5) Table 5.3 also shows the effect of oil at a heat flux of 30 kW/m² by presenting the ratio $h'/h'' (= \Delta T' / \Delta T'')$ for a given surface at various oil concentrations.

The boiling curves for the plain tube (Fig. 5.1) show a general decrease in heat transfer performance with increasing oil concentration, particularly at heat fluxes above 20 kW/m². At the highest heat flux the degradation is seen to be quite dramatic at 10% oil, considering the logarithmic nature of the plot. Table 5.3 supports this observation at a heat flux of 30 kW/m². While no statistically significant change in the ratio of boiling coefficients is observed between 1% and 5% oil, there is a significant decrease in the ratio between 0% and 1% oil as well as between 5% and 10% oil.

Table 5.2 Precision of Measured Quantities in Oil-Refrigerant Tests

	$w_{\Delta T}$ (°C)	Nominal ΔT (°C)	$\frac{w_{\Delta T}}{\Delta T}$	w_e	Nominal e	$\frac{w_e}{e}$	$w_{\Delta T'}/\Delta T''$	Nominal $\frac{h''}{h'}$	$\frac{w_{\Delta T'}/\Delta T''}{\Delta T'/\Delta T''}$
Plain	± 0.367	10	± 0.037	—	—	—	± 0.10	0.53	± 0.19
Enhanced	± 0.367	2.0	± 0.18	± 0.936	5.0	± 0.19	± 0.10	0.50	± 0.20

Figure 5.1 Pool boiling curves of a smooth surface in R-113 at 0%, 1%, 5% and 10% mass concentrations of 3GS oil

Table 5.3 Ratio of Observed Boiling Coefficients (h'/h'') at a Heat Flux of 30 kW/m²

Oil Concentration [⊗]	Surface				
	Plain	High Flux Large	High Flux ¹ Small	Turbo-B ² Very Open	Turbo-B Very Closed
1%	0.7	0.6	0.9	1.0	0.8
5%	0.6	0.5	0.7	0.7	0.6
10%	0.3	0.4	0.4	0.6	0.4

[⊗] Suniso 3GS (155 SSU) Oil

¹ Universal Oil Products Incorporated (formerly Linde Division, Union Carbide Corporation).

² Wolverine Tube Incorporated

The High Flux Large surface (Fig. 5.2) experienced a degradation with the addition of oil (compared to the pure R-113 curve) at all heat fluxes. For fluxes above 30 kW/m^2 , no significant difference in performance was observed between the 1% and 5% oil runs while a degradation was observed between 5% and 10% oil. At 30 kW/m^2 , Table 5.3 shows a significant change in the ratio of boiling coefficients between the pure R-113 run and the runs with oil, although no significant difference was observed between the various concentrations of oil.

The effect of 1% oil on the High Flux Small tube (Fig. 5.3) was seen to be significant at only the highest heat flux while a reduction in performance between 0% and 5% oil was seen at fluxes above about 30 kW/m^2 . The effect of 10% oil was observed at fluxes above about 6 kW/m^2 with a dramatic degradation at the highest flux. Table 5.3 also shows a change in the ratio of boiling coefficients at 30 kW/m^2 between 0% and 5% oil as well as between 5% and 10% oil.

Figure 5.2 Pool boiling curves of UOP High Flux "Large" surface in R-113 at 0%, 1%, 5%, and 10% mass concentrations of 3GS oil

AIM

Association for Information and Image Management

1100 Wayne Avenue, Suite 1100
Silver Spring, Maryland 20910
301/587-8202

Centimeter

Inches

MANUFACTURED TO AIM STANDARDS
BY APPLIED IMAGE, INC.

2 of 2

Figure 5.3 Pool boiling curves of UOP High Flux "Small" surface in R-113 at 0%, 1%, 5%, and 10% mass concentrations of 3GS oil

No significant effect was observed between 0% and 1% oil with the Turbo-B Very Open tube (Fig. 5.4) at all fluxes. A degradation was noted with the addition of 5% or 10% oil at all fluxes. Table 5.3 supports these observations at a heat flux of $30 \text{ kW}/m^2$, also suggesting no significant effect between 5% and 10% oil.

The Turbo-B Very Closed tube (Fig. 5.5) was affected by the presence of 5% or 10% oil at fluxes above about $2 \text{ kW}/m^2$ with no effect with 1% oil. Table 5.3 confirms this at a flux of $30 \text{ kW}/m^2$, and suggests no change in the ratio of boiling coefficients between the 1% and 5% oil cases or the 5% and 10% oil runs.

Table 5.3 shows that the High Flux Large surface is the one most affected by the presence of 1% and 5% oil (compared to pure R-113). The addition of 10% oil is seen to affect the plain tube more than the other four surfaces. The Very Open surface was least affected by oil at all concentrations.

Figure 5.4 Pool boiling curves of Wolverine Turbo-B "Very Open" surface in R-113 at 0%, 1%, 5%, and 10% mass concentrations of 3GS oil

Figure 5.5 Pool boiling curves of Wolverine Turbo-B "Very Closed" surface in R-113 at 0%, 1%, 5%, and 10% mass concentrations of 3GS oil

To compare the performance of various surfaces at a given oil concentration and heat flux, it is convenient to introduce the enhancement ratio $\Delta T_p/\Delta T_e$. Table 5.4 presents enhancement ratios for a flux of 30 kW/m². In pure R-113, the High Flux Small surface performed better than the Turbo-B surfaces. At 1% and 5% oil, the High Flux Small surface provided a lower boiling coefficient than both the High Flux Large and Turbo-B Very Closed surfaces. With 10% oil, the High Flux Small and Turbo-B Very Open surfaces out-performed the Turbo-B Very Closed surface. Some researchers (Wanniarachchi et al. 1986 and 1987, McManus et al. 1987) have observed a decrease in the enhancement ratio with increasing oil concentration. No general trend with oil concentration was observed in this study.

Table 5.4 Observed Enhancement Ratios $\Delta T_p/\Delta T_e$ at a Heat Flux of 30 kW/m²

Surface	Oil Concentration [⊗]			
	0%	1%	5%	10%
High Flux ¹ Small	7	8	7	8
High Flux Large	5	4	4	6
Turbo-B ² Very Open	4	6	5	8
Turbo-B Very Closed	3	4	4	4

[⊗]Suniso 3GS (155 SSU) Oil

¹Universal Oil Products Incorporated
(formerly Linde Division, Union Carbide Corporation).

²Wolverine Tube Incorporated.

Boiling data were collected for the four enhanced surfaces in pure R-113 before and after the oil mixture runs. Between each run, every surface was soaked in de-natured alcohol for at least 48 hours to clean the surface. The High Flux Small surface showed a decrease in the boiling coefficient from its pretest value at only the highest heat flux examined in the post-oil experiment. All other surfaces were observed to experience no significant change from their performance as measured before the commencement of tests with oil/refrigerant mixtures.

The presence of oil in the R-113 resulted in mixture with a syrup-like consistency, increasing with increasing oil concentration. The pool also became cloudy. Foam was observed at the top of the pool when boiling with oil. More foam existed at higher concentrations. An oily residue was observed on the plain tube following experiments in R113/oil mixtures. No noticeable difference was observed in the time it took to achieve steady-state conditions between the pure refrigerant experiments and the mixture experiments. In both cases, the wall temperature became steady within five minutes of adjusting the power.

The results obtained in the tests described here appear to support the proposal of Wanniarachchi et al. (1987) that the observed decrease in heat transfer performance with increasing oil concentration is the result of an oil-rich layer adjacent to the tube surface. This layer develops as the refrigerant (the more volatile component) evaporates at the boiling surface, leaving behind the less volatile oil. The liquid-vapor surface tension and the surface adhesive tension associated with the oil-rich layer are significantly higher than those of the pure refrigerant (Baustian et al., 1986) a result, more energy is needed for the evolution of bubbles. Also, since the oil/refrigerant mixture has a greater liquid viscosity than the pure refrigerant, the turbulent motion in the pool is reduced, lessening the flow of liquid to the boiling surface. Both effects increase as the oil concentration increases. The result is a decrease in the boiling heat transfer coefficient.

No surface examined in this study was found to be especially immune or more vulnerable to the phenomenon than another surface. In general, the surfaces experience only a slight decrease in performance when 1% oil is present while all the surfaces suffer a substantial degradation at 10% oil (see Table 5.3).

To compare the results obtained in Table 5.4 with those in Table 5.1, it is important to note that the specific surface structures of the High Flux and Turbo-B surfaces examined in Grant et al. (1989) and Marvillet (1989) are not known. Thus, it is not likely that the surfaces examined in this study have the same structure as those in Table 5.1. In the pure refrigerant tests all observed enhancement ratios ($\Delta T_p / \Delta T_e$) were between 30% and 82% lower than those reported in Table 5.1. At 10% oil, the observed enhancement ratios for the High Flux Large and Turbo-B Very Closed surfaces were about the same (less than 20% difference) as those reported in Grant et al. (1989) and Marvillet (1989). Both values of the enhancement ratio for the High Flux Small and Turbo-B Very Open surfaces were between 30% and 40% larger than those obtained by Wanniarachchi et al. (1986) and Murphy (1987). Table 5.1 shows a monotonic decrease in the enhancement ratio with increasing oil concentration for all surfaces. No general trend in the enhancement ratio was observed in this study.

The fact that the original performance of all the enhanced tubes was practically restored when soaked in de-natured alcohol and re-tested in pure R-113 suggests that the miscible mixture of R-113 and 3GS oil does not lead to an irreversible degradation of the heat transfer surface. Other miscible mixtures such as R-12 and Mobil oil SHC 234 have yielded similar results (Marvillet 1989).

5.5 Conclusions

Pool boiling data of structured surfaces in R-113/3GS oil mixtures support the following conclusions:

1. A general decrease in heat transfer performance occurs with increasing mass concentration of oil.
2. A substantial degradation in the performance of all surfaces exists at 10% oil.
3. No surface is especially immune or more vulnerable to the presence of oil than another surface.
4. The structured surfaces maintain their original enhancements over the plain surface when both surfaces are exposed to oil. Thus, there is no general trend of the enhancement ratio with increasing oil concentration.
5. The original performance of the enhanced surfaces can be restored after cleaning in de-natured alcohol.

Based on the difficulty in determining the wall superheat, the following recommendations are made:

1. More precise temperature sensors, such as thermistors, should replace the thermocouples.
2. A method of instrumentation should be devised to measure the temperature difference directly using differential temperature sensors.
3. A design should be proposed to minimize the effect of cooling the pool due to the reflux of condensate from the condensers.

6. SUMMARY AND RECOMMENDATIONS

This investigation sought to establish the boiling heat transfer performance of individual horizontal tubes with enhanced surfaces in a tube bundle. In doing so, insight has been gained that will aid in the design of shell-and-tube heat exchangers with horizontal tube bundles. The investigation included experimental studies of single tubes in pool boiling and of tube bundles with the fluid forced through the bundle in the vertically upward direction.

Pool boiling tests were conducted for each of the three surfaces so that reference values of the heat transfer coefficient could be obtained. Values of the heat transfer coefficient for the enhanced tubes were generally an order of magnitude larger than for the smooth tubes. Heat transfer coefficients for the porous surface were generally the largest; however, a large hysteresis in the boiling curve was found. Visual observations revealed that only patches of the High Flux surface nucleated initially and that nucleation covered more of the surface as the heat flux increased. Heat transfer coefficients for the Turbo-B tube, on the other hand, showed very little hysteresis in the boiling curve. The difference was attributed to the sub-surface structure of the enhanced tubes and the ability of vapor generated near the bottom of the tube to travel around the perimeter and initiate boiling in extinct nucleation sites. The pool boiling experiments conducted in pure R-11 showed that, as with the pure R-113 experiments, the enhanced tubes performed better than the smooth tubes and that the performance of the smooth and High Flux tubes in pure R-11 was similar to that in R-113.

Pool boiling heat transfer coefficients in fluid mixtures were less than those in pure fluids for all three surfaces. However, for a given heat flux and mixture concentration, the enhanced surfaces consistently performed better than the smooth tube. The High Flux tube performed slightly better than the Turbo-B tube, as was the case for the pure fluid experiments. The performance of all three surfaces decreased as the concentration of R-11

decreased, with the largest performance reduction relative to the pure refrigerant performance occurring in a mixture of 25% R-11/75% R-113.

Boiling heat transfer experiments in the tube bundles conducted in pure R-113 revealed that the heat transfer coefficients for the smooth tubes increased significantly with heat flux and pressure. However, the heat transfer coefficients increased only slightly with increasing mass flux and only at very low heat fluxes. The pitch-to-diameter ratio did not affect the heat transfer coefficients appreciably either, except at low heat fluxes and large mass fluxes. In addition, the heat transfer coefficients increased with tube row (quality) only at very low fluxes.

The heat transfer coefficients for the enhanced surfaces in the tube bundles were significantly larger than those for the smooth tubes and were comparable to the values obtained in pool boiling. Indeed, the increasing trend in the heat transfer coefficient obtained in the tube bundle due to with increasing heat flux was comparable to the trends observed in pool boiling for all three surfaces. The heat transfer coefficients for the enhanced surfaces did not vary significantly with mass flux, quality, or pitch-to-diameter ratio. These results indicated that the enhanced tubes were unaffected by the convective flow in the free stream. The enhanced surface caused the boiling process to occur within the sub-surface structure, and the external flow could not intrude into the sub-surface cavities and change the heat transfer mechanisms.

No significant variation of the heat transfer coefficient with tube row was found except at the first row in the High Flux bundle (and the last row in the Turbo-B bundle). The reduced heat transfer coefficient in the first row of the High Flux tube bundle was attributed to non-nucleating areas on the surface of the High Flux tube, caused by the free-stream convection increasing the heat transfer from the non-nucleating regions and preventing the wall superheat from increasing enough to initiate nucleation everywhere on the tube surface. Heat transfer coefficients obtained in pure R-11 generally followed the same trends as in pure R-113.

Heat transfer coefficients decreased significantly for all surfaces when refrigerant mixtures were used but the same trends in the mass flux, heat flux, and pressure effects generally were observed, and the enhanced surfaces consistently performed better than the smooth tubes. The reduction in the heat transfer coefficient was largest for an inlet mixture concentration of 23.8% R-11. The degradation in the smooth tube heat transfer coefficients due to the fluid mixtures was predicted fairly well using a correlation of the form given by Stephan and Körner (1970), which predicts the wall superheat as a function of the difference between the molar concentration in the liquid and vapor. However, experimental data for the enhanced surfaces were not well predicted using a correlation of the same form and indicates that the heat transfer process has been significantly altered by the enhanced surface structure, possibly because of the inability of the free stream to affect the heat transfer process near the heated surface.

In order to elucidate the fouling behavior responsible for the unexpected behaviour of the enhanced surfaces with pool boiling in a saturated calcium sulfate solution, enlarged models of the test surfaces were constructed. Photographic observation of the test surface showed that in the case of the simulated Wolverine Turbo-B surface that most of the fouling deposit formed in the cavities on the surface. It was hypothesized that this caused either a redistribution of the temperature distribution in the enhancement elements and/or the effective roughening of the surface increases the local heat transfer coefficient. Further tests would be required to support this hypothesis.

The tests with the simulated UOP "High Flux" surface were inconclusive. This was due to difficulties in photographing the test surface and to uncertainty concerning the accuracy of the simulation of the actual surface using ball bearings.

The heat transfer measurements made with boiling oil-refrigerant mixtures indicated that there was a decrease in the heat transfer performance of a surface with increasing concentrations of oil. This supports the observations of earlier investigators. Cleaning the surface at the conclusion of a test restored the performance of the surface

when it was tested in pure refrigerant which demonstrated that the degradation in heat transfer performance was not permanent.

The results of this investigation have made apparent several areas which require further investigation. Perhaps the most obvious is the need to determine if a combination of nucleate boiling and single-phase convection could also account for the lack of a mass flux effect on the smooth tube heat transfer coefficients. Although fully established nucleate boiling was found to dominate the heat transfer process in this study, the alternative possibility might apply to data obtained at low heat flux values and might explain the lack of a mass flux effect of internal flow, such as reported by Kandlikar (1990).

The invariance in the heat transfer coefficient with tube row and quality also might be explored further in light of a possible balance between nucleate boiling and single-phase convection. The explanation presented depends on a decrease in the boiling heat transfer with increasing liquid velocity, which depends on the liquid-to-vapor density ratio. This analysis could explain the density ratio effect reported by Kandlikar (1990).

The presence of thin film vaporization in the sub-surface channels of the Turbo-B tubes would be supported by verifying that liquid enters the pores near the bottom and exits as vapor through pores near the top. This could be facilitated with visual studies, using enlarged models as described in the fouling tests, of the path of dye injected into the liquid.

Finally, the onset of nucleation for High Flux tubes should be investigated further. Numerical calculations could be performed to determine the maximum surface temperature variation in high conductivity, thick-walled tubes with a uniform heat flux as the inside boundary condition and a variable heat transfer coefficient as the outside boundary condition. In combination with experimental temperature overshoot information, such a study would determine whether the temperature of small patches of non-nucleating surface area would be low enough to prevent the onset of nucleation.

The behavior of enhanced surfaces when operating under boiling conditions, and exposed to categories of fouling other than the precipitation fouling studied in the work reported here, should be investigated. Particulate fouling and chemical reaction fouling resulting from the degradation of organic fluids would be particularly important. The details of the fouling process using enlarged models should be continued and extended to other categories of fouling. Information derived from tests of this type should suggest ways in which minimum fouling enhanced heat transfer surfaces for operation under boiling conditions can be designed.

The tests on oil–refrigerant mixtures should be extended to the new refrigerants, such as R–134(a), which are being developed.

7. REFERENCES

- American Society of Mechanical Engineers, 1967, Thermodynamic and Transport Properties of Steam, American Society of Mechanical Engineers, New York, NY.
- Arai, N., Fukushima, T., Arai, A., Nakajima, T., Fujie, K., and Nakayama, Y., 1977, "Heat Transfer Tubes Enhancing Boiling and Condensation in Heat Exchangers of a Refrigerating Machine," ASHRAE Transactions, Vol. 83, Part 2, pp. 58–70.
- ASHRAE, 1985, Handbook of Fundamentals, ASHRAE, Atlanta, pp. 17.2–17.19.
- Banchero, J.T. and Gordon, K.F., 1960, Scale Deposition on a Heated Surface, Advances in Chemistry, No. 27, pp. 105–114.
- Baustian, J.J., Pate, M.B., and Bergles, A.E., 1986, "Properties of Oil–Refrigerant Liquid Mixtures with Applications to Oil Concentration Measurement: Part I – Thermophysical and Transport Properties," ASHRAE Transactions, Vol. 92, Part 1, pp. 55–73.
- Bergles, A.E., 1985, "Techniques to Augment Heat Transfer," Chapter 3 of Handbook of Heat Transfer Applications, McGraw–Hill, New York, pp. 3–1–3–80.
- Bergles, A.E. and Chyu, M.C., 1982, "Characteristics of Nucleate Pool Boiling From Porous Metallic Coatings," ASME Journal of Heat Transfer, Vol. 104, pp. 279–285.
- Bier, K., Schmadl, J., and Gorenflo, D., 1982, "Pool Boiling Heat Transfer to Mixtures of SF₆ and CF₃Br at Elevated Saturated Pressures," 7th Int. Heat Transfer Conf., Munich, Vol. 4, pp. 35–40.
- Chaddock, J.B., 1976, "Influence of Oil on Refrigerant Evaporator Performance," ASHRAE Trans., Vol. 82, pp. 474–486.
- Chandler, J.L., 1964, The Effect of Supersaturation and Flow Conditions on the Initiation of Scale Formation, Trans. Instn. Chemical Engineers, Vol. 42, pp. T24–T34.
- Chyu, M.C., 1984, Falling Film Evaporation on Horizontal Tubes with Smooth and Structured Surfaces, Ph.D. Thesis, Iowa State University, Ames, Iowa.
- Cornwell, K., 1990, "The Influence of Bubbly Flow on Boiling From a Tube in a Bundle," International Journal of Heat and Mass Transfer, Vol. 33, pp. 2579–2584.
- Cornwell, K. and Schuller, 1982, "A Study of Boiling Outside a Tube Bundle Using High Speed Photography," International Journal of Heat and Mass Transfer, Vol. 25, pp. 683–690.
- Cornwell, K. and Scoones, D.J., 1988, "Analysis of Low Quality Boiling on Plain and Low-finned Tube Bundles," Second United Kingdom Conference on Heat Transfer, Glasgow, Vol. 1, pp. 21–32.
- Couste, E., 1854, Researches on Boiler Scale, Ann. Mines, Vol. 5, No. 5, pp. 69–164.
- Curcio, L.A., Jr., 1989, Pool Boiling of Enhanced Heat Transfer Surfaces in Refrigerant–Oil Mixtures and Aqueous Calcium Sulfate Solutions, M.E. thesis, Rensselaer Polytechnic Institute, Troy, New York.

Czikk, A.M., Gottzmann, C.F., Ragi, E.G., Withers, J.G., and Habdas, E.P., 1970, "Performance of Advanced Heat Transfer Tubes in Refrigerant-Flooded Liquid Coolers," ASHRAE Transactions, Vol. 76, pp. 96-109.

Danilova, G.N. and Dyundin, V.A., 1972, "Heat Transfer with Freons 12 and 22 Boiling at Bundles of Finned Tubes," Heat Transfer - Soviet Research, Vol. 4, No. 4, pp. 48-54.

Dykas, S. and Jensen, M.K., 1992, "Critical Heat Flux on a Tube in a Horizontal Tube Bundle," Experimental Thermal and Fluid Science, Vol. 5, No. 1, pp. 34-39.

Epstein, N., 1979, Fouling in Heat Exchanges, Heat Transfer 1978 - Proceedings 6th International Heat Transfer Conference. Hemisphere Publishing Corporation, Washington, DC, Vol. 6, pp. 235-253.

Epstein, N., 1981, Fouling: Technical Aspects (Afterword to Fouling in Heat Exchangers), Fouling of Heat Transfer Equipment, E.F.C. Somerscales and J.G. Knudsen, eds., Hemisphere Publishing Corp., Washington, DC, pp. 31-53.

ESDU, 1973, "Convective Heat Transfer During Crossflow of Fluids Over Plain Tube Banks," Engineering Sciences Data Unit, London, Item No. 73031.

ESDU, 1979, "Crossflow Pressure Loss Over Banks of Plain Tubes in Square and Triangular Arrays Including Effects of Flow Direction," Engineering Science Data Unit, London, Item No. 79034.

Freeborn, J. and Lewis, D., 1962, Initiation of Boiler Scale Formation, J. Mech. Eng. Sci., Vol. 4, pp. 46-52.

Fujita, Y., Ohta, H., Yoshida, K., Hidaka, S., and Nishikawa, K., 1984, "Nucleate Boiling Heat Transfer in Horizontal Tube Bundles (1st Report: Experimental Investigation on Tube Bundle Effect)," Memoirs of the Faculty of Engineering, Kyushu University, Vol. 44, No. 4, pp. 427-446.

Fujita, Y., Ohta, H., Hoshida, K., and Hidaka, S., 1990, "Heat Transfer in Nucleate Boiling Outside Horizontal Tube Bundles: Part 2 - Prediction of the Tube Bundle Effect," Heat Transfer - Japanese Research, Vol. 19, No. 2, pp. 25-40.

Gottzmann, C.F., O'Neill, P.S. and Minton, P.E., 1973, High Efficiency Heat Exchangers, Chem. Eng. Prog., Vol. 69, No. 7, pp. 69-75.

Grant, I., Hewitt, G.F., and Qureshi, J., 1989, "Pool Boiling of R113/Oil Mixtures on a High Flux Tube," Eurotherm No. 8, Advances in Pool Boiling Heat Transfer, Paderborn, FRG, May, pp. 169-172.

Hahne, E. and Mueller, J., 1983, "Boiling on a Finned Tube and a Finned Tube Bundle," International Journal of Heat and Mass Transfer, Vol. 26, pp. 849-859.

Hall, R.E., 1925, A System of Boiler Water Treatment Based on Chemical Equilibrium, Ind. Eng. Chem., Vol. 17, pp. 283-290.

Hwang, T.H. and Yao, S.C., 1986, "Forced Convective Boiling in Horizontal Tube Bundles," International Journal of Heat and Mass Transfer, Vol. 29, pp. 785-794.

Jensen, M.K., 1988, "Boiling on the Shellside of Horizontal Tube Bundles," Two-Phase Flow Heat Exchangers Thermal-Hydraulic Fundamentals and Design, Eds. S. Kakac, A.E. Bergles, and O.E. Fernandes, Kluwer Academic Publishers: Dordrecht, pp. 707-746.

Jensen, M.K. and Hsu, J.-T., 1988, "A Parametric Study of Boiling Heat Transfer in a Tube Bundle," Journal of Heat Transfer, Vol. 110, pp. 976-981.

Jensen, M.K., Reinke, M.J., and Hsu, J.-T., 1989, "The Influence of Tube Bundle Geometry on Cross-Flow Boiling Heat Transfer and Pressure Drop," Experimental Thermal and Fluid Science, Vol. 2, pp. 465-476.

Jung, C. and Bergles, A.E., 1989, "Evaluation of Commercial Enhanced Tubes in Pool Boiling," U.S. Department of Energy Topical Report No. DE-FC07-88ID12772. Also Heat Transfer Lab Report HTL-2, Rensselaer Polytechnic Institute, Troy, New York, March, 1989.

Kandlikar, S.G., 1990, "A General Correlation for Saturated Two-Phase Flow Boiling Heat Transfer Inside Horizontal and Vertical Tubes," Journal of Heat Transfer, Vol. 112, pp. 219-228.

Kern, D.C. and Seaton, R.E., 1959a, A Theoretical Analysis of Thermal Surface Fouling, Brit. Chem. Eng., Vol. 4, pp. 258-262.

Kern, D.C. and Seaton, R.E., 1959b, Surface Fouling: How to Calculate Limits, Chem. Eng. Prog., Vol. 55, No. 6, pp. 71-73.

Koyama, Y. and Hashizume, K., 1983, "Boiling Heat Transfer on Outside of Horizontal Tube Bundles," 16th International Congress of Refrigeration, Proceedings, Vol. 2, pp. 171-177.

Marvillet, C., 1989, "Influence of Oil on Nucleate Pool Boiling on Refrigerants R12 and R22 from Porous Layer Tube," Eurotherm No. 8, Advances in Pool Boiling Heat Transfer, Paderborn, FRG, May, pp. 173-177.

McAdams, W.H., 1954, Heat Transmission, 3rd Ed., McGraw Hill Book Company, New York, p. 176.

McCabe, W.L. and Robinson, C.S., 1924, Evaporator Scale Formation, Ind. Eng. Chem., Vol. 16, pp. 478-479.

McManus, S.M., Marto, P.J., and Wanniarachchi, A.S., 1987, "An Evaluation of Enhanced Heat Transfer Tubing for Use in R-114 Water Chillers," Heat Transfer in Air Conditioning and Refrigerating Equipment, ASME, HTD-Vol. 65, pp. 11-19.

Murphy, T.J., 1987, Pool Boiling of R-114/Oil Mixtures from Single Tubes and Tube Bundles, M.S. Thesis, Naval Postgraduate School, Monterey, California, September.

Myers, J.E. and Katz, D.L., 1953, "Boiling Coefficients Outside Horizontal Tubes," Chemical Engineering Progress Symposium Series No. 5, Vol. 49, pp. 107-114.

Nakayama, W., Daikoku, T. Kuwahara, H. and Nakajima, T., 1979, "Dynamic Model of Enhanced Boiling Heat Transfer Surfaces," Advances in Enhanced Heat Transfer, ASME, pp. 31-43.

National Bureau of Standards, 1979, Thermocouple Reference Tables, United States Government Printing Office, Washington, DC.

O'Neill, P.S., King, R.C., and Ragi, E.G., 1980, "Application of High Performance Evaporator Tubing in Refrigerant Systems of Large Olefins Plants," AIChE Symposium Series No. 199, Vol. 76, pp. 289–300.

Palen, J.W., 1983, "Shell and Tube Reboilers," Heat Exchanger Design Handbook, Hemisphere Publishing Corp.: Washington, D.C.

Palen, J.W., 1965, Fouling of Heat Transfer Surfaces, M.S. Thesis, University of Illinois, Urbana, Illinois.

Palen, J.W. and Westwater, J.W., 1966, Heat Transfer and Fouling Rates During Pool Boiling of Calcium Sulfate Solution, Heat Transfer – Los Angeles, Chem. Eng. Prog. Symposium Series, J.G. Knudsen, ed., No. 64, Vol. 62, pp. 77–86.

Partridge, E.P., 1930, Formation and Properties of Boiler Scale, Engineering Research Bulletin, No. 15, Department of Engineering Research, University of Michigan, Ann Arbor, Michigan.

Partridge, E.P. and White, A.H., 1929a, Mechanism of Formation of Calcium Sulfate Boiler Scale, Ind. Eng. Chem., Vol. 21, pp. 834–838.

Partridge, E.P. and White, A.H., 1929b, Thermal Effects of Boiler Scale, Ind. Eng. Chem., Vol. 21, pp. 839–844.

Partridge, E.P. and White, A.H., 1929c, The Formation and thermal Effects of Calcium Sulfate Boiler Scale, Trans. American Soc. Mech. Engrs., Vol. 51 (FSP–51–49), pp. 383–394.

Reitzer, B.J., 1964, Rate of Scale Formation in Tubular Heat Exchangers, Ind. Eng. Chem. Process Design and Development, Vol. 3, pp. 345–348.

Ritter, R.B., 1983, Crystalline Fouling Studies, ASME Journal of Heat Transfer, Vol. 105, pp. 374–378.

Sauer, H.J., Davidson, G.W. and Chongrungreong, S., 1980, "Nucleate Boiling of Refrigerant–Oil Mixtures From Finned Tubing," Joint ASME/AIChE National Heat Transfer Conference, Orlando, ASME Paper No. 80–HT–111.

Stephan, K., 1963, "Influence of Oil on Heat Transfer of Boiling Freon 12 (Refrigerant 12) and Freon 22 (Refrigerant 22)," Proc. of the Int'l. Congress (XI) of Refrigeration, Vol. 1, pp. 369–380.

Stephan, K. and Körner, M., 1969, "Berechnung des Wärmeübergangs verdampfender binäre flüssigkeitsgemische," Chemie–Ing.–Techn., Vol. 41, No. 7, pp. 409–417.

Stephan, K. and Abdelsalan, M., 1980, "Heat–Transfer Correlations for Natural Convection Boiling," International Journal of Heat and Mass Transfer, Vol. 23, pp. 73–87.

- Stephan, K. and Mitrovic, J., 1981, "Heat Transfer in Natural Convective Boiling of Refrigerants and Refrigerant-Oil Mixtures in Bundles of T-Shaped Finned Tubes," Advances in Enhanced Heat Transfer - 1981, Eds. R.L. Webb et al., ASME, New York, HTD-Vol. 18, pp. 131-146.
- Stephan, K., 1982, "Heat Transfer in Boiling of Mixtures," 7th Int. Heat Transfer Conf., Munich, Vol. 1, pp. 59-81.
- Tatara, R.A. and Payvar, P., 1986, "Pressure Drop and Heat Transfer Measurements of Boiling Heat Transfer in Normal Flow Through a Porous Coated Tube Bundle," Heat Transfer in Air-Conditioning and Refrigeration Equipment, Ed. J.A. Kohler and J.W.B. Lu, ASME, New York, HTD-Vol. 65, pp. 1-9.
- Thome, J.R. and Shock, R.A.W., 1984, "Boiling of Multicomponent Liquid Mixtures," Advances in Heat Transfer, Eds. J.P. Hartnett and T.F. Irvine, Vol. 16, pp. 59-156.
- Trewin, R.R., 1992, "Enhanced Boiling Heat Transfer in Horizontal Tube Bundles," Ph.D Thesis in Mechanical Engineering, Rensselaer Polytechnic Institute, Troy, NY.
- Wall, K.W. and Park, E.L., Jr., 1978, "Nucleate Boiling of N-Pentane, N-Hexane and Several Mixtures of the Two from Various Tube Arrays," International Journal of Heat and Mass Transfer, Vol. 21, pp. 73-75.
- Wanniarachchi, A.S., Marto, P.J., and Reilly, J.T., 1986, "The Effect of Oil Contamination on the Nucleate Pool-Boiling Performance of R-114 from a Porous-Coated Surface," ASHRAE Trans., Vol. 92, Part 2, June, pp. 525-538.
- Wanniarachchi, A.S., L.M. Sawyer, and P.J. Marto, 1987, "Effect of Oil on Pool-Boiling Performance of R-114 from Enhanced Surfaces," 2nd ASME-JSME Thermal Engineering Joint Conference, Honolulu, Hawaii, March.
- Water Pollution Control Federation, 1962, Standard Methods of Chemical Analysis, N.H. Furman, ed., 6th edition, Water Pollution Control Federation, Washington, DC, p. 271.
- Webb, R.L. and Pais, C., 1991, "Pool Boiling Data for Five Refrigerants on Three Tube Geometries," ASHRAE Transactions, Vol. 97, pt.XX pp. 72-78.
- Yilmaz, S. and Palen, J.W., 1981, "Performance of Finned Tube Reboilers in Hydrocarbon Service," ASME Paper No. 84-HT-91.
- Yilmaz, S. and Westwater, J.W., 1981, "Effect of Enhanced Surfaces on the Boiling Heat Transfer Curve," Advances in Enhanced Heat Transfer, ASME, HTD-Vol. 18, pp. 73-91.
- Yilmaz, S., Palen, J.W., and Taborek, J., 1981, "Enhanced Boiling Surfaces as Single Tubes and Tube Bundles," Advances in Enhanced Heat Transfer - 1981, Eds. R.L. Webb et al., ASME, New York, HTD-Vol. 18, pp. 123-129.
- Zukauskas, A., 1972, "Heat Transfer from Tubes in Crossflow," Advances in Heat Transfer, Eds. J.P. Hartnett and T.F. Irvine, Jr., Academic Press, New York, Vol. 8, pp. 137-142.

DATE

FILMED

11/14/94

END