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Nucleate pool boiling of R-114 and R-114–oil mixtures from smooth and enhanced surfaces—II. Tube bundles

S. B. MEMORY

Department of Mechanical Engineering, University of Miami, Coral Gables, FL 33124, U.S.A.

and

N. AKCASAYAR, H. ERAYDIN and P. J. MARTO

Department of Mechanical Engineering, Naval Postgraduate School, Monterey, CA 93943, U.S.A.

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Abstract—Measurements of pool boiling heat-transfer coefficients in pure R-114 and R-114–oil mixtures are reported for a bundle of smooth tubes and three enhanced tube bundles (finned, structured and porous). Each bundle contained 15 electrically heated tubes (five of which were instrumented) in a staggered triangular-pitch layout. Tests were carried out at atmospheric pressure while decreasing the heat flux. For pure refrigerant, the structured and porous bundles provide the highest average enhancements of between 4 and 6. For all but the porous bundle, the bundle effect is only significant at low heat fluxes: the porous tube bundle exhibits no bundle effect. With addition of oil, the performance of the smooth and finned tube bundles at first increases before dropping off slightly. For the structured and porous bundles, oil addition leads to a steady decrease in performance, especially for the porous bundle at high heat fluxes.

INTRODUCTION

Significant progress has been made in recent years in understanding pool boiling heat transfer in refrigerant flooded-type evaporators [1, 2]. In particular, the influence of tube position within a bundle of smooth tubes has been studied extensively [3–17] for a variety of fluids, operating conditions, bundle layouts (in-line and staggered) and tube spacing (pitch-to-diameter ratios of 1.2–2.0). It has been well documented that lower tubes within a bundle can significantly increase nucleate boiling heat transfer from the upper tubes at low heat fluxes: this is referred to as bundle effect (see below). The works of Cornwell [3–7] indicate that convection effects as well as bubbles from lower tubes impinging on, and sliding around, upper tubes can account for these increases and that nucleation only occurs on the lower tubes. At high heat fluxes in the fully developed nucleate boiling region (typically >30 kW m^{-2}), these influences disappear and the data for all tubes merge onto a single curve, representative of a single isolated tube. Similar results have also been obtained for finned tube bundles [18–25].

There is much confusion in the literature regarding the use of the terms bundle *effect* and bundle *factor* and it is often not made clear which is being reported.† Bundle effect is defined as the ratio of the heat-transfer coefficient for an upper tube in a bundle with lower tubes activated to that for the same tube activated alone in the bundle. Bundle factor is defined as the ratio of *average* heat-transfer coefficient for the whole bundle to that of a single isolated‡ tube of similar surface. In many cases, however, the data for a single isolated tube (which should ideally be taken in the same apparatus as the bundle) is not available and data taken from a single tube activated alone within the bundle (usually an upper tube) is used. Bundle factor is typically slightly lower than bundle effect and is of more use to the designer who can simply use it together with single tube data to estimate average bundle coefficients. To understand the mechanisms which affect heat transfer in a bundle, however, bundle effect is often more informative.

Based on results from smooth and finned tube bundles, Palen [26] recommended that bundle heat transfer behavior could be predicted by simple interpolation formulae, combining the contributions of pool boiling and liquid forced convection. Cornwell [7] suggests adding a third contribution due to sliding bubbles, which accounts not only for the increased turbulence induced by the impinging bubbles, but also microlayer evaporation under the bubbles. However, the second and third contributions cannot be readily evaluated as vapor qualities and induced liquid circulation rates through the bundle are often not

† In some cases, the present authors have had to use their judgement as to whether it is bundle factor or bundle effect which is being cited.

‡ The term isolated means a single tube with no other tubes around it.

NOMENCLATURE

$\overline{T_w}$	average outside wall temperature
ΔT	wall superheat, $(\overline{T_w}-T_{sat})$
$\overline{h_{av}}$	average bundle heat-transfer coefficient for pure refrigerant
$\overline{h_{av,mix}}$	average bundle heat-transfer coefficient for refrigerant–oil mixture
$\overline{h_s}$	average heat-transfer coefficient for a single tube in pure refrigerant.

Greek symbols

ε_{bf}	bundle factor in pure refrigerant
ε_{oil}	oil addition factor.

known. Consequently bundle modeling relies heavily on empirical correlations for the three components, which in turn rely on accurate experimental data.

Although there are less data available, similar conclusions to those above for smooth and finned tube bundles (bundle effect > 1 at low heat fluxes, decreasing to unity at high heat fluxes) have also been obtained for other types of enhanced tube bundles in a variety of refrigerants, including finned tubes with modified fin profiles (GEWA-T, Stephan and Mitrovic [27]) and structured surface tubes (THERMOEXCEL-E, Arai *et al.* [20] and TURBO-B, Memory *et al.* [28]). However, for porous coated tubes, Fujita *et al.* [13] and Memory *et al.* [29] have both reported a bundle effect of unity over a wide range of heat fluxes for a 'bundle' of two tubes in a pool of R-113 and R-114, respectively. Czikk *et al.* [30] conducted extensive tests using a 20-tube porous coated bundle in R-11. Although no single tube tests were done, comparison with similarly coated disks also indicated a bundle effect of unity.

Effects of oil on refrigeration equipment can also be significant [31, 32]. For refrigerant–oil mixtures, the authors know of no smooth tube bundle data. For low integral-fin tube bundles, Danilova and Dyundin [18] and Heimbach [19] have studied the effects of oil on overall bundle performance.† Using a 19-tube 16 fpi bundle in R-12, Danilova and Dyundin [18] tested only one oil concentration (8%) and found a decrease in average bundle heat-transfer coefficient of up to 50% at low heat fluxes, decreasing to about 10% at high heat fluxes. For all tests carried out with oil, significant foaming was also reported. Heimbach [19] tested two oils of different viscosity in R-12 using a 10-tube bundle of 19 fpi tubes. He found that the addition of between 3 and 7% of either oil actually *increased* the average bundle heat-transfer coefficient (compared to pure refrigerant values) by up to 40% at the highest heat fluxes. He attributed this increase in heat transfer to the foaming, which increased in intensity with increases in both oil concentration and heat flux. Stephan and Mitrovic [27] added up to 9% oil with their GEWA-T bundle. Above a heat flux of 20 kW m^{-2}, they also obtained a small increase in the heat-transfer coefficient of about 8% for oil concentrations up to 6%. Above this oil concentration, performance started to drop-off. No reason was given for the increase and no mention of foaming was made. Arai *et al.* [20] added up to 4% oil with their structured surface bundle and showed that, for tests conducted with the top rows *below* the foaming region, a drop-off in performance for the bundle was obtained. However, with the top rows immersed *in* the foaming region, they found that this drop-off was offset by the increased activity of the refrigerant around these upper tubes, thereby maintaining the high performance of the bundle. Although no actual increase in bundle performance was reported, the maximum heat fluxes used were not very high. Their work also suggests that bundle performance is extremely sensitive to refrigerant level.

From the above, it is clear that two-phase interactions that occur in tube bundles during boiling are very complex and can vary with heat flux, operating pressure, fluid properties, tube surface and pool height; bundle layout seems to be less important. Furthermore, as with single tubes, some investigators have found increases in average bundle heat transfer with addition of oil and some have found decreases: some of this discrepancy may be attributed to varying degrees of miscibility of the oil used. It is therefore not very wise to use information from one fluid mixture–bundle combination and apply it to another. Instead, experimental data are needed that cover these parameters so that models can be formulated and appropriately evaluated. This is particularly important in the refrigeration industry where new, alternative refrigerants and refrigerant–oil mixtures are being proposed.

The objective of this paper is to provide a comprehensive pool boiling database for R-114 and R-114–oil mixtures from smooth and enhanced tube bundles as well as to shed further light on the mechanisms which affect bundle heat transfer. Of particular importance is information pertaining to the influence of lower tubes on upper tubes (bundle effect) and the influence of oil on bundle performance, especially for the enhanced tubes. A companion paper [33] devotes

† The definitions of bundle effect and bundle factor can still be applied to refrigerant–oil mixtures, with comparison made at the same oil concentration. However, the performance of a bundle in a refrigerant–oil mixture is often gaged by how it compares to the same bundle in pure refrigerant under identical conditions.

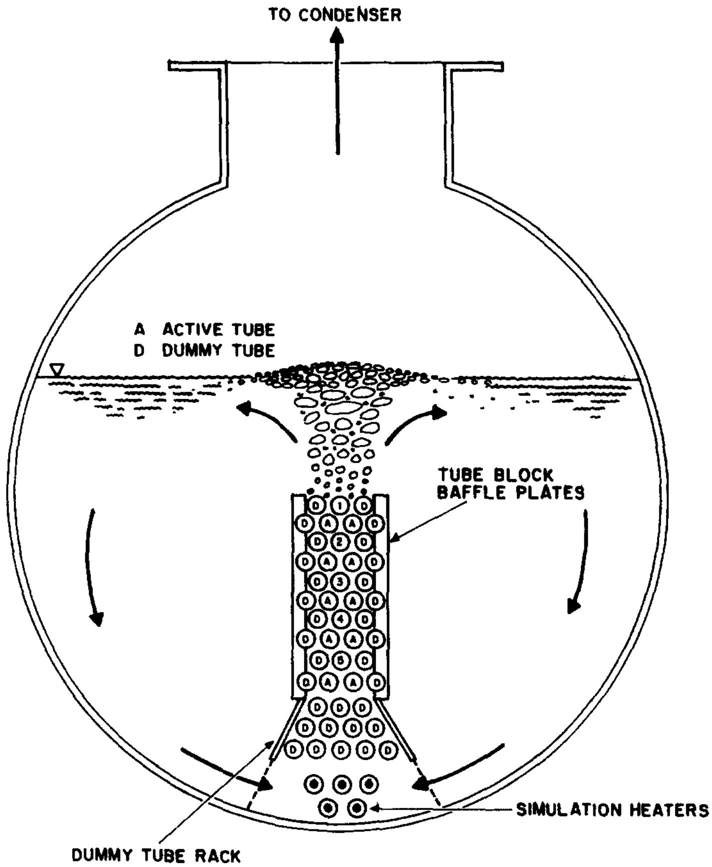

Fig. 1. Sectional view of evaporator.

attention to single tube studies. R-114 is a CFC and is to be completely phased out over the next few years due to environmental concerns. However, by having such a database available, heat-transfer characteristics of future replacement refrigerants may be more readily evaluated.

EXPERIMENTAL APPARATUS

Full details of the experimental facility can be found in Marto and Anderson [17]. Essentially, it consisted of an evaporator and condenser arranged to provide reflux operation. The evaporator was designed to simulate a small slice of a refrigerant flooded-type evaporator, shown schematically in Fig. 1. Vapor from the evaporator flowed upward to a condenser through a riser section and was distributed axially and circumferentially to the top of the condenser by a vapor shroud. Non-condensibles were removed from the condenser by means of a vacuum pump. The condensate collected in the bottom of this shroud returned to the evaporator by gravity.

The bundle consisted of 15 electrically-heated tubes which were cantilever-mounted from the back wall of the evaporator to permit easy viewing along the axis of the tubes through the lower of two glass windows mounted on the front. Ten of these tubes were active (marked 'A') and contained 1 kW heaters; each of the remaining five were instrumented tubes (numbered 1–5) which, in addition to 1 kW heaters, contained six wall thermocouples. The instrumented tubes were located along the centerline of a symmetrical, staggered tube bundle, as shown. Figure 2 is a sketch of an instrumented tube, showing the tube construction and the location of the wall thermocouples. The heater was the same as that used in the single tube study [33] with a heated length† of 190 mm. Both the smooth and enhanced tubes were fabricated in the same way. Full details of the construction and soldering process of the tubes are provided by Marto and Anderson [17]. In measuring boiling heat transfer coefficients, great care must be exercised with the cartridge heater and temperature measuring instrumentation to ensure good accuracy: this is discussed in detail in ref. [33].

Five simulation heaters, each capable of 4 kW, were mounted below the test bundle to simulate additional tube rows and to provide inlet vapor quality into the bottom of the test bundle. Each set of heaters (bundle and simulation) could be independently activated using two separate rheostat controllers. The bundle also contained a number of unheated dummy smooth tubes (marked 'D') that were used to guide the two-

† The heaters used were Watlow Firerod heaters which were continuously wound with a 203 mm nominal length and a 190 mm heated length.

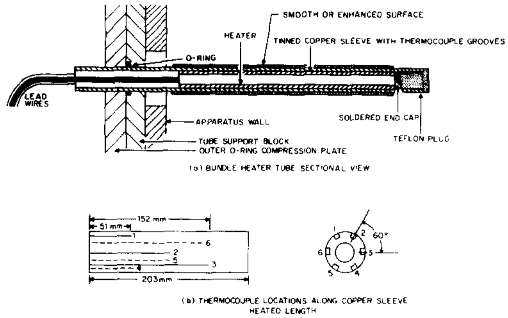

Fig. 2. Instrumented tube construction.

phase mixture through the bundle. Two vertical baffle plates were installed to restrict side circulation into and out of the bundle. To permit side entry of liquid into the bottom of the dummy tube rack, the area adjacent to the simulation heaters was left open. Thus, liquid circulation was vertically upward over the heated tubes with no net horizontal component. Each bundle layout was identical and arranged in an equilateral triangular pitch with a centerline-to-centerline spacing of 19.1 mm and a nominal pitch-to-diameter† ratio of 1.2. A Plexiglass template was fitted to the front end of the bundle (within the pool) to maintain the bundle layout and ensure that each tube was horizontal. Details of the bundle installation in the evaporator are given by Marto and Anderson [17].

Vapor temperatures were measured by two thermocouples at the top of the condenser and one thermocouple near the top of the evaporator. Liquid temperatures were measured by three thermocouples, two located close to the free surface of the liquid and a third located at the bottom of the pool close to the dummy tube rack. During operation, the top two liquid thermocouples were in the frothy, two-phase mixture and were considered to be well representative of the saturation temperature at the free surface. The refrigeration oil used was mineral based (York-C) and was completely miscible in R-114 (characteristics of which are given in ref. [33]). It was added to the evaporator under vacuum via a graduated container. The desired amount of oil was calculated from a knowledge of the initial volume of R-114 within the evaporator.

TUBES TESTED

The finned tube was a 19 fpi trapezoidal low integral-fin tube (GEWA-K). Some dimensions of this and the other tubes used are given in Table 1. These tubes were the same as those used in ref. [33] and further details are given there. Sketches of the enhanced tube surfaces and fabrication details are also given in ref. [33].

EXPERIMENTAL PROCEDURE

With the bundle installed and the system integrity checked, the evaporator was charged with R-114 to a level of 10 cm above the top row of tubes in the bundle. With 10% oil added to the system, the mixture level increased to a maximum of 16 cm above the top row of tubes. The liquid level has been found to be an important parameter [20] and was therefore maintained at a constant value for a given oil concentration.

The average outer wall temperatures of the five instrumented tubes were obtained by averaging the six wall thermocouples in the copper sleeve and correcting for the small radial temperature drop due to conduction across the copper wall. The temperature drop across the solder joint between the copper sleeve and the test tube (estimated to be 0.05 mm) was neglected. For a given tube, the average heat flux was calculated by dividing the electrical power (after it was corrected for small axial losses from the end of the test tube) by the tube surface area based on the diameter to the base of the enhancement‡ (see Table 1) and an active heated length of 190 mm. The local saturation temperature for each tube in the bundle was calculated using a hydrostatic pressure correction between the tube location and the free surface of the liquid (any pressure drop effects due to the two-phase flow in the bundle were neglected).

During this investigation, data were obtained following the same kind of surface aging procedure§ as used in ref. [33]. The heat flux of each heated tube in the bundle was first set at a maximum (≈ 100 kW m^{-2}) for approximately 30 min, such that any non-condensables could collect in the condenser and be vented off. The apparatus was then secured and the test tube was left to stand overnight in the pool of R-114, reaching room temperature. The following morning, the tube heat flux was again set at maximum for

† Using diameter to the outside of the enhancement (envelope diameter).

‡ A full explanation of the significance of surface area is given in ref. [33].

§ Marto and Anderson [17] report four different surface aging procedures for R-113.

Table 1. Specifications of tubes tested

Tube description	Diameter to base of enhancement [mm]	Thickness of enhancement or fin height [mm]
Smooth (used as received)	15.9	—
Finned (19 fpi GEWA-K)	12.9†	1.50
Structured (TURBO-B)	14.2‡	0.85
Porous (HIGH FLUX)	15.7‡	0.1

† Diameter to root of fin.
‡ Diameter to base of enhancement.

another 10 min. Data collection then commenced with decreasing heat flux in pre-determined steps down to around 1 kW m^{-2}. Once the required heat flux in the evaporator had been fixed, the coolant flow through the condenser was adjusted to maintain the required saturation temperature at the pool surface of 2.2°C, corresponding to a pressure of approximately 1 atm.† All the data were obtained and reduced with a computer-controlled data acquisition system: the thermophysical properties for R-114 were taken from REFPROP [34].

In addition to pure refrigerant, oil concentrations of 1, 2, 3, 6 and 10% were used. For each tube bundle, seven independent tests were conducted for all six oil concentrations (making 42 independent tests for each bundle). Test numbers 1–5 progressively activated only the five instrumented tubes within the bundle, starting with the top tube alone, to see the effect of lower heated tubes on the performance of the tubes above. Test number 6, in addition, activated the remaining 10 heated tubes around these instrumented tubes. All these tests were essentially carried out with zero quality entering the bottom of the bundle. By activating the simulation heaters below the bundle, test number 7 included an inlet quality which increased with heat flux up to a maximum of about 20%. For every data point, the heat flux on each of the heated tubes was the same. For test 7, the heat flux of the simulation heaters was set such that, at any point, they simulated a bundle of 15 additional heated tubes, each at the same heat flux as the tubes above.

RESULTS AND DISCUSSION

Before discussing the results, a general observation that should be mentioned (and which applies to both pure refrigerant and refrigerant–oil mixtures) was the fluid oscillation that consistently occurred within each bundle at heat fluxes higher than about 20 kW m^{-2}. This could clearly be seen as a periodic side-to-side movement of the bubbly mixture as it flowed up through the bundle. As with the single tube apparatus [33], the frequency‡ was found to be very repeatable at around 1 Hz, and surprisingly independent of heat flux and oil concentration.

Uncertainty in the measured data

The uncertainty in the experimental data was estimated using a propagation of error analysis. The largest source of uncertainty in the data is that associated with the wall superheat, which in turn is dominated by the uncertainty in the average wall temperature measurement. The wall superheat is taken as the average of the six measured wall temperatures minus the saturation temperature of the bulk fluid. For a high (60 kW m^{-2}) and low (2 kW m^{-2}) heat flux for each bundle, Table 2 lists the maximum variation in the wall superheat for the top tube alone (test 1) as well as for the bundle averaged over the five instrumented tubes (test 7). These were calculated from local variations in the wall superheat taken from the six wall thermocouples. It was noted that for tests 6 and 7, the maximum variation of the wall thermocouples decreased slightly compared with test 5, suggesting that circulation through the bundle becomes more longitudinally uniform as surrounding tubes and simulated tubes below are activated. Although some of the variation is therefore due to the flow characteristics within the bundle, the majority appear to be random and independent of thermocouple orientation, caused either by non-uniformities in the cartridge heater coils or (more likely) by the tube soldering and assembly procedure. The estimated uncertainties in both the wall superheat and the average heat-transfer coefficient for each bundle during tests 1 and 7 are also listed in Table 2. The corresponding uncertainty in the measured heat flux was estimated to be $\pm 1.5\%$ at the highest heat flux, increasing to 5% at the lowest heat flux. Like ref. [33], high values of uncertainty occur at low heat fluxes when the measured wall superheat is typically < 1°C. This is so for both re-entrant cavity bundles as well as for the finned bundle during test 7, where convection effects significantly reduce the measured wall superheat. Uncertainty bands are included in some of the figures below.

Pure R-114

Throughout the investigation, the five instrumented tubes located along the centerline of each bundle were

† This is a typical operating pressure in a centrifugal flooded evaporator using R-114.
‡ One cycle is taken to be a side-to-side motion of the pool surface back to its starting position.

Table 2. Uncertainty analysis for tests 1 (tube 1 alone) and 7 (bundle average) at a low (≈ 2 kW m^{-2}) and high (≈ 60 kW m^{-2}) heat flux

Tube type	Maximum variation† in ΔT ($\pm$°C)		% Uncertainty in ΔT		% Uncertainty in $\overline{h_{av}}$	
Heat flux	Low	High	Low	High	Low	High
Smooth (test 1)	0.20	0.90	4.5	5.6	4.8	7.5
Smooth (test 7)	0.17	1.10	7.9	6.7	8.0	8.4
Finned (test 1)	0.14	0.50	6.5	7.6	6.7	9.1
Finned (test 7)	0.16	0.74	16.7	11.6	16.7	12.6
Structured (test 1)	0.10	0.45	14.9	10.2	15.0	11.4
Structured (test 7)	0.10	0.55	19.6	14.1	19.7	15.0
Porous (test 1)	0.09	0.35	14.1	15.2	14.1	16.0
Porous (test 7)	0.08	0.38	13.6	11.7	13.6	12.7

† For the average bundle (test 7), maximum variation is taken to be the maximum found from a study of each individual tube.

numbered consecutively from the top downward as 1, 2, 3, 4 and 5. Figure 3 shows a comparison between tube 1 activated alone as a single tube within the bundle (test 1) and the single tube data† reported in ref. [33] for the four types of surface. Typical uncertainty bands for ΔT have been indicated on the figure at low and high heat fluxes for all but the smooth tube (which is too small to be included on the figure). For the smooth tube at low heat fluxes, where the nucleation sites have died out, the single tube data reported in ref. [33] have higher wall superheats (lower heat-transfer coefficients). This is thought to be due to the effect of stronger local convection currents felt in the bundle due to 'squeezing' of the flow around the other tubes. This has also been observed by Memory *et al.* [29], who placed an unheated smooth tube below a heated one in the single tube apparatus and found a similar increase in heat transfer to that seen in Fig. 3. As heat flux increases, agreement improves as convection effects become less important.

For the re-entrant cavity tubes, this trend is reversed at low heat fluxes since the surfaces remain nucleating: any fluid dragged along by the bubbles, as well as the circulating bubbles themselves, will have a greater effect in a smaller pool. The 19 fpi tube bundle falls between these two extremes and gives the best agreement at low heat fluxes. Figure 3 also shows that the re-entrant cavity tubes display significantly higher heat-transfer coefficients than the smooth and finned tubes, similar to those reported in ref. [33]. These differences become more apparent at lower heat fluxes where the nucleation sites on the smooth and finned tubes die away.

Due to the way in which the five instrumented tubes were individually manufactured, direct comparisons between data taken on different tubes within the same bundle might include uncertainties due to small differences in thermocouple installation. Of more use, perhaps, is what happens to a given tube as succesive tubes below it are activated, i.e. the bundle effect. Any trends are then associated with heat transfer and fluid flow mechanisms within the pool and not with tube manufacture. Figures 4–7 show the data for tube 1 only during all seven tests for each bundle. Also shown in Fig. 4 for the smooth bundle are the data of Marto and Anderson [17] for tube 1 during tests 1 and 7 using the same apparatus with R-113 at the same saturation pressure. The close agreement between the data for R-113 and R-114 is indicative of the similarity in thermophysical properties of the two fluids at the same pressure. Also shown for comparison on each figure are the data for the bundle average taken from test 7 and calculated by averaging the five instrumented tubes.‡ Table 3 lists the average heat-transfer coefficient for the top tube during tests 1 and 7, as well as the bundle average for all four bundles at five heat fluxes.

Also listed in Table 3 are the bundle effect and bundle factor at each heat flux. Strict definition of bundle factor requires the use of isolated single tube data, which exist in ref. [33]. However, due to differences in convection patterns between the data in the bundle apparatus and those in the much smaller single tube apparatus [33] (highlighted in Fig. 3), it was felt that using data from test 1 would be preferable as effects of pool volume are reduced. For the smooth and finned tube bundles (Figs. 4 and 5), there is a significant bundle effect at low heat fluxes. By adding heated tubes below, the mechanisms of convection and sliding bubbles postulated by Cornwell and co-workers [3–7] (and documented by other investigators) create these significant bundle effects at low heat fluxes. A third mechanism which could further explain the observed increase in heat transfer in the upper part of the bundle when sliding bubbles are

† It should be noted that the bundle pool volume was approx. 23 times the single tube pool volume.

‡ Due to the bundle effect, averaging only the top five tubes in a simulated larger bundle may cause the average bundle heat-transfer coefficients to be slightly high, especially at low heat fluxes.

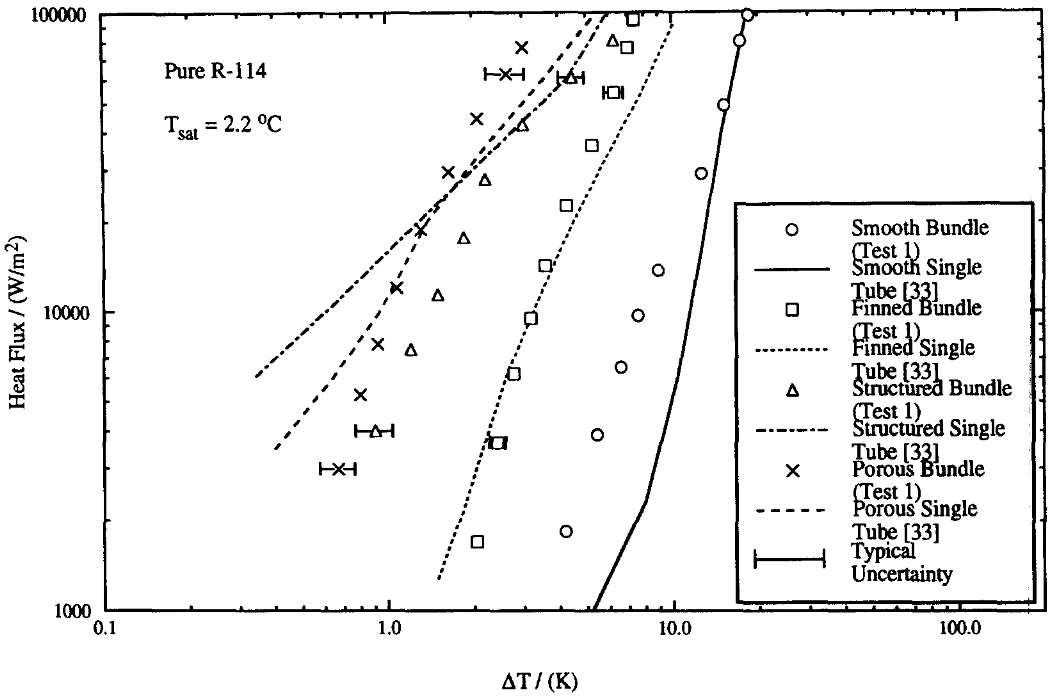

Fig. 3. Test 1 results in the bundle vs results from single tube apparatus [33].

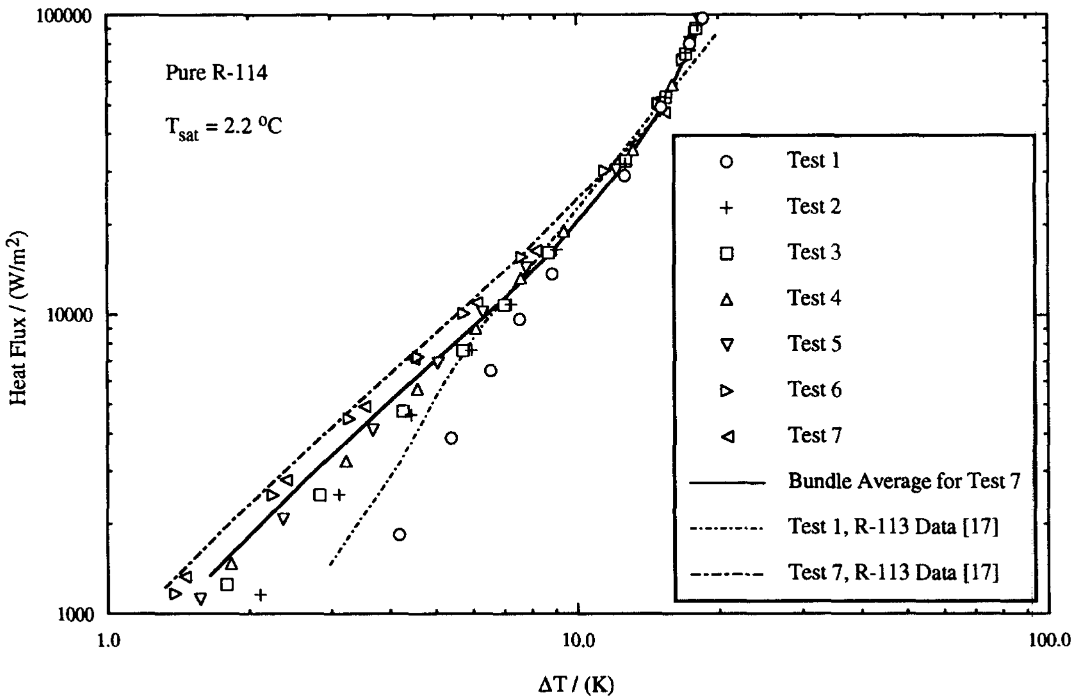

Fig. 4. Top smooth tube data for all 7 tests.

present is secondary nucleation.† Mesler and Mailen [36] found that, when a bubble bursts through a thin liquid film (such as an evaporating microlayer), many new microscopic-sized bubbles grow from that location due to entrained vapor nucleii created from the bursting process. For the smooth and finned bundles, the heat-transfer from each of the instrumented tubes decreases as one moves down the bundle, indicating that convection is the dominant mechanism in improving heat transfer performance. Bundle factor is always slightly lower than bundle effect due to the reduced heat transfer from the lower tubes. As heat flux increases, bundle effect and bundle factor

† A fuller explanation of secondary nucleation is given in ref. [33].

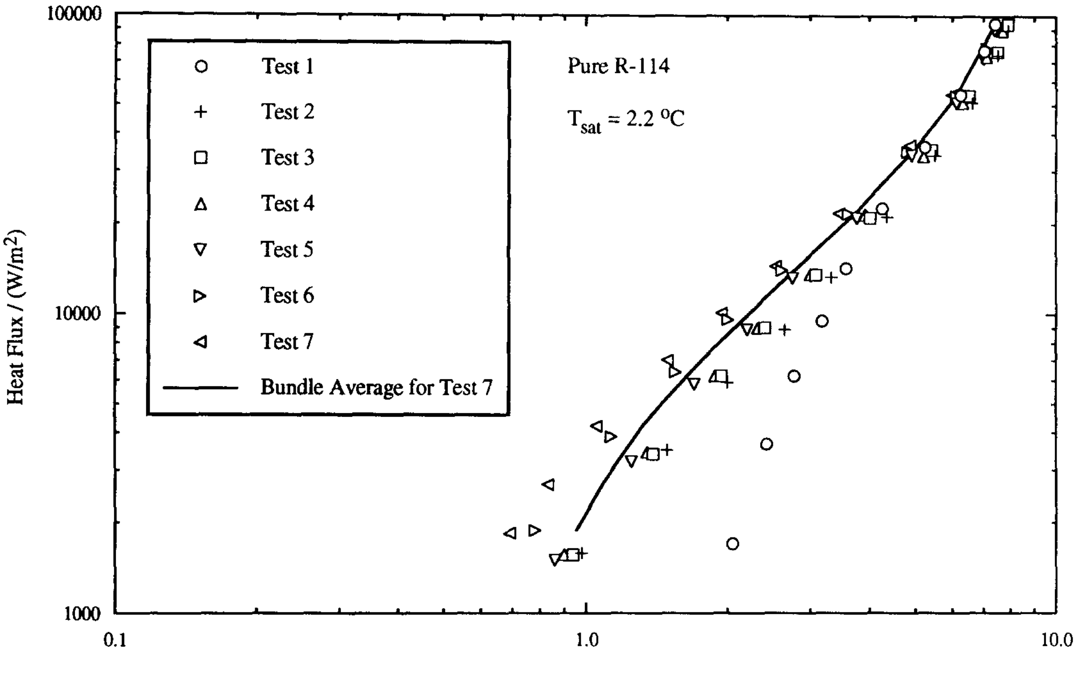

Fig. 5. Top finned tube data for all 7 tests.

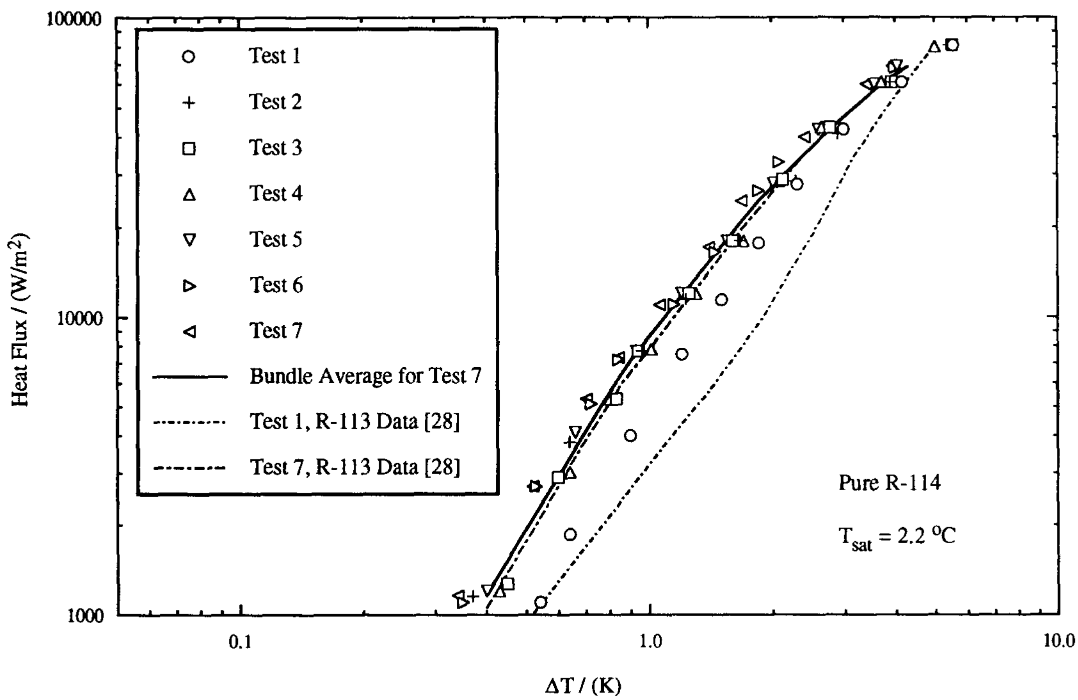

Fig. 6. Top structured surface tube data for all 7 tests.

decrease, reaching values of unity at the highest heat fluxes. This is where boiling from the surface itself begins to dominate, diminishing the influence of these other mechanisms.

The structured surface bundle (TURBO-B, Fig. 6) exhibits certain similar trends as the smooth and finned bundles with both bundle effect and bundle factor decreasing to unity as heat flux increases. This trend is similar to that reported by Memory *et al.* [28] for the same bundle in R-113 (also given in Fig. 6 for tube 1 during tests 1 and 7) and Arai *et al.* [20] for a different type of structured surface bundle (THERMOEXCEL-E). As mentioned in ref. [33], this is not too surprising when one considers that struc-

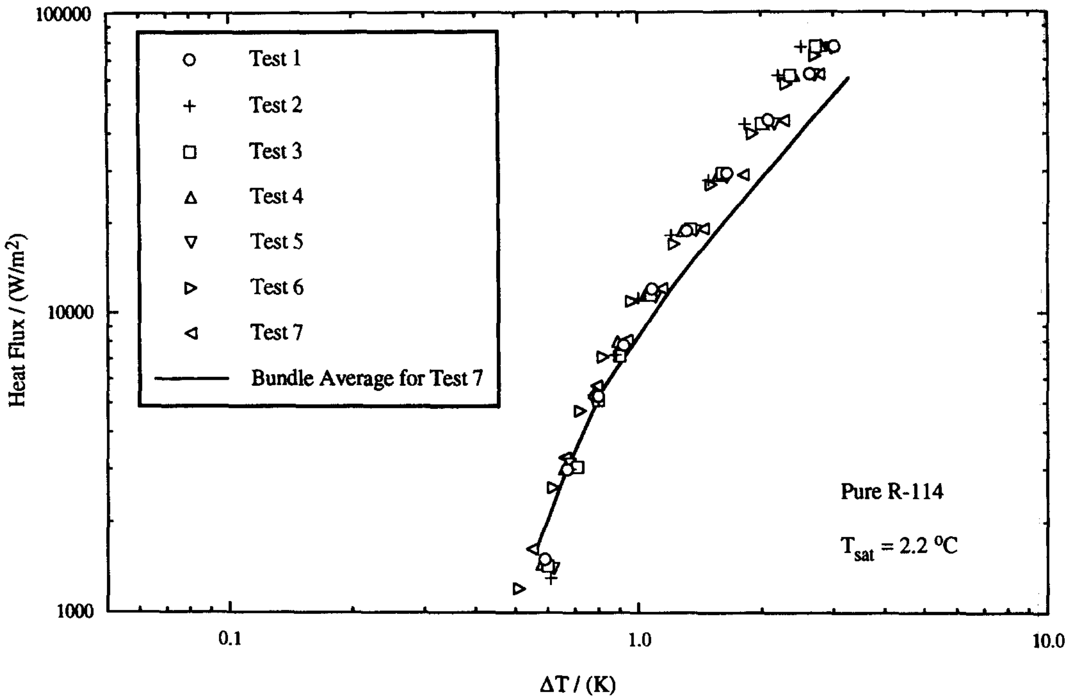

Fig. 7. Top porous tube data for all 7 tests.

Table 3. Comparison of bundle heat transfer behavior for pure R-114

	Heat-transfer coefficient [kW m^{-2} K^{-1}]				
Heat flux [kW m^{-2}]	5	15	30	50	80
Smooth bundle					
Tube 1 (test 1)	0.84	1.61	2.36	3.25	4.65
Tube 1 (test 7)	1.41	2.00	2.54	3.29	4.67
Bundle average (test 7)	1.28	1.83	2.50	3.29	4.67
Bundle effect†	1.68	1.24	1.08	1.01	1.00
Bundle factor‡	1.52	1.14	1.06	1.01	1.00
Finned bundle					
Tube 1 (test 1)	1.89	4.05	6.19	8.33	11.27
Tube 1 (test 7)	4.03	5.66	6.98	8.77	11.01
Bundle average (test 7)	3.38	5.17	6.67	8.48	11.11
Bundle effect†	2.13	1.40	1.13	1.05	0.98
Bundle factor‡	1.79	1.28	1.08	1.02	0.99
Structured bundle					
Tube 1 (test 1)	5.05	8.72	12.77	14.50	14.55
Tube 1 (test 7)	7.41	11.54	15.63	16.67	16.00
Bundle average (test 7)	6.49	10.64	14.29	15.39	15.10
Bundle effect†	1.47	1.32	1.22	1.15	1.10
Bundle factor‡	1.29	1.22	1.12	1.06	1.04
Porous bundle					
Tube 1 (test 1)	5.95	12.93	19.36	24.39	28.07
Tube 1 (test 7)	5.97	11.54	16.67	20.41	23.19
Bundle average (test 7)	6.33	10.95	14.93	17.86	19.75
Bundle effect†	1.00	0.89	0.86	0.84	0.83
Bundle factor‡	1.06	0.85	0.77	0.73	0.70

† Bundle effect is the ratio of the heat-transfer coefficient for the top tube during bundle operation (test 7) to that of the top tube in the bundle activated alone (test 1).

‡ Bundle factor is the ratio of the average heat-transfer coefficient for the bundle (test 7) to that of the top tube in the bundle activated alone (test 1).

tured surface tubes are fabricated from finned tubes and indicates that the mechanisms listed above for finned and smooth bundles are still important for structured surface tube bundles. However, since the number of bubbles at low heat fluxes is higher, sliding bubbles may be relatively more important, with convection having a reduced influence.

The porous bundle (Fig. 7) contradicts data from the other bundles, showing a bundle effect ≈ 1 for the whole range of heat flux covered, similar to that found by Fujita *et al.* [13] and Memory *et al.* [29]. It seems, therefore, that nucleation from the porous coated surface is the dominant mechanism at low as well as high heat fluxes. Added influences from tubes below make little or no difference to the heat transfer performance from upper tubes. At high heat fluxes, bundle effect actually drops below unity: this could be due to the significant voidage (high quality) that exists around the top tube during test 7.

Figure 8(a) (with uncertainty bands) compares the average bundle heat-transfer coefficients (from test 7) for all four bundles. At a given heat flux, the ratio of $\overline{h}_{av}$ for a given bundle to that for the smooth bundle gives the bundle enhancement, which is listed in Table 4. In general, the enhancements agree well with those reported in ref. [33] for single tubes of similar surface, especially at higher heat fluxes (>50 kW m^{-2}). At low heat fluxes, bundle enhancements are more conservative (especially for the re-entrant cavity bundles), due to convection increasing the smooth bundle $\overline{h}_{av}$. Maximum uncertainty in enhancement values for the enhanced bundles is about 25% at low heat fluxes decreasing to around 20% at high heat fluxes.

Any comparisons between the present enhanced tube bundle data and those found in the literature should be made with great caution due to varying definitions of tube outer surface area; this is discussed fully in ref. [33] for single tubes. If envelope diameter had been used rather than root diameter in the calculation of heat flux, then enhancements for the finned, structured and porous tube bundles in Table 4 would be reduced by factors of 1.23, 1.12 and 1, respectively. If actual wetted surface area had been used† for the 19 fpi tube bundle, then enhancements would be reduced by a factor of 3. This suggests that the bundle enhancement given in Table 4 for the 19 fpi tube bundle is mainly due to an increase in surface area.

The present results for pure R-114 confirm that, in general, a bundle factor should be incorporated into the design of flooded evaporators and that the use of single tube data will be conservative. The one exception to this is the porous coated bundle, where single tube data may even overestimate the heat transfer, especially at higher heat fluxes. Furthermore, bundle enhancements agree fairly well with single tube enhancements.

R-114–oil mixtures

Figure 8(b) and (c) compares the average bundle heat-transfer coefficients (from test 7) for all four bundles at 3 and 10% oil concentrations, respectively. Enhancements taken from these figures as well as the other oil concentrations tested (1, 2 and 6%) are given in Table 4 for the same three heat fluxes as above. Comparing these values to those given in ref. [33] for single tubes, it can be seen that trends with increases in oil concentration are similar. As with pure R-114, bundle enhancements at low heat fluxes are conservative (especially for the re-entrant cavity bundles), due to convection significantly improving the smooth tube bundle performance. At higher heat fluxes, these influences are diminished and enhancements are more in line with those found from single tubes. At high heat fluxes and oil concentrations, the performance of the porous tube bundle drops off dramatically and the finned tube bundle gives the best performance: this behavior is similar to that reported for single tubes [33] and attributed to clogging of the pores with oil.

In general, data for refrigerant–oil mixtures are presented as an 'oil addition factor', ε_{oil}, which is a ratio of the average bundle heat-transfer coefficient for a given oil concentration to that for the same bundle under identical conditions in pure refrigerant.‡ Figure 9 presents ε_{oil} as a function of oil concentration in a similar way to that of Danilova and Dyundin [18] and Heimbach [19] at a heat flux of 30 kW m^{-2}. The most surprising aspect is the increase in ε_{oil} for the smooth and finned tube bundles at low oil concentrations. Although no data were taken for oil concentrations between 3 and 6%, it appears that ε_{oil} reaches a maximum at an oil concentration of around 3–4%. This agrees well with that reported for single tubes [33], where it was postulated that this increase in heat transfer could be attributed to the foaming that occurs with addition of oil: significant foaming was also observed for the present data when oil was added. Further addition of oil (>6%) leads to a drop-off in performance for both the smooth and finned bundles, although not as large a drop-off as found for single tubes [33].

For the porous and structured surface tube bundles, performance is seen to drop off for any oil addition ($\varepsilon_{oil} < 1$): this is also similar to that found in ref. [33]. For a different structured bundle (THERMOEXCEL-E), Arai *et al.* [20] found similar behavior to that shown in Fig. 9. However, they also showed that the position of the tubes with respect to the foaming was very important and that, if the top rows were immersed in the foaming region, drop-off in performance could be offset by the increased activity of the mixture around these upper tubes. For the present tests, the top rows were always well below the foaming region.

Similar trends to those given in Fig. 9 were also

† Due to the complexity of the re-entrant cavity tubes, the actual wetted surface area is unknown.

‡ It should be noted that ε_{oil} is not necessarily the same as that found for single tubes.

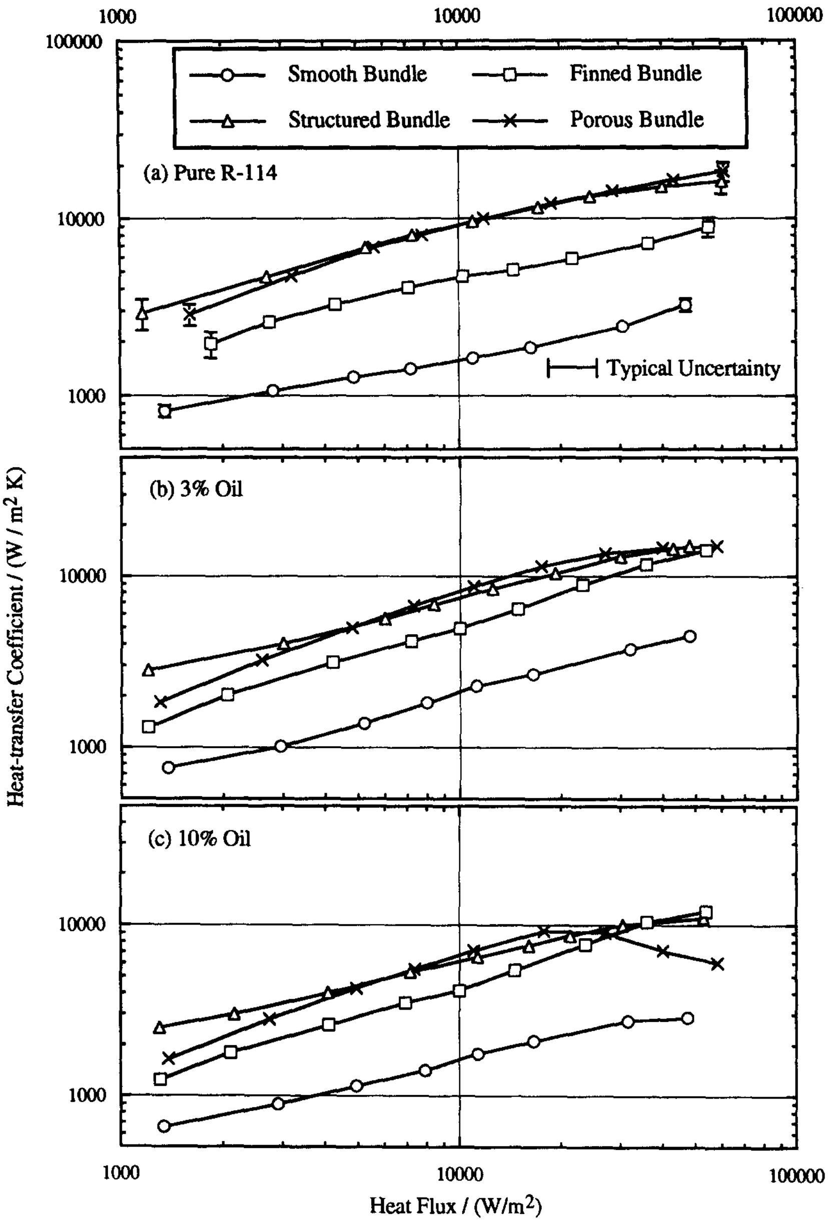

Fig. 8. Average bundle coefficients with pure refrigerant, 3% and 10% oil.

Table 4. Comparison of bundle enhancements with varying oil concentration

	Heat flux [kW m^{-2}]	Enhancement Oil concentration [%] 0	1	2	3	6	10
19 fpi bundle	15	2.8	2.5	2.3	2.5	2.4	2.6
	30	2.7	2.6	2.6	2.9	3.0	3.4
	50	2.6	2.7	3.0	3.2	3.5	4.2
Structured surface bundle	15	5.8	4.1	3.4	3.4	3.4	3.7
	30	5.7	4.5	3.8	3.6	3.5	3.5
	50	4.7	4.0	3.7	3.3	3.2	3.8
Porous bundle	15	6.0	4.2	3.6	3.5	3.6	3.8
	30	6.0	4.2	3.7	3.5	3.3	3.2
	50	5.4	4.2	3.7	3.4	2.6	2.3

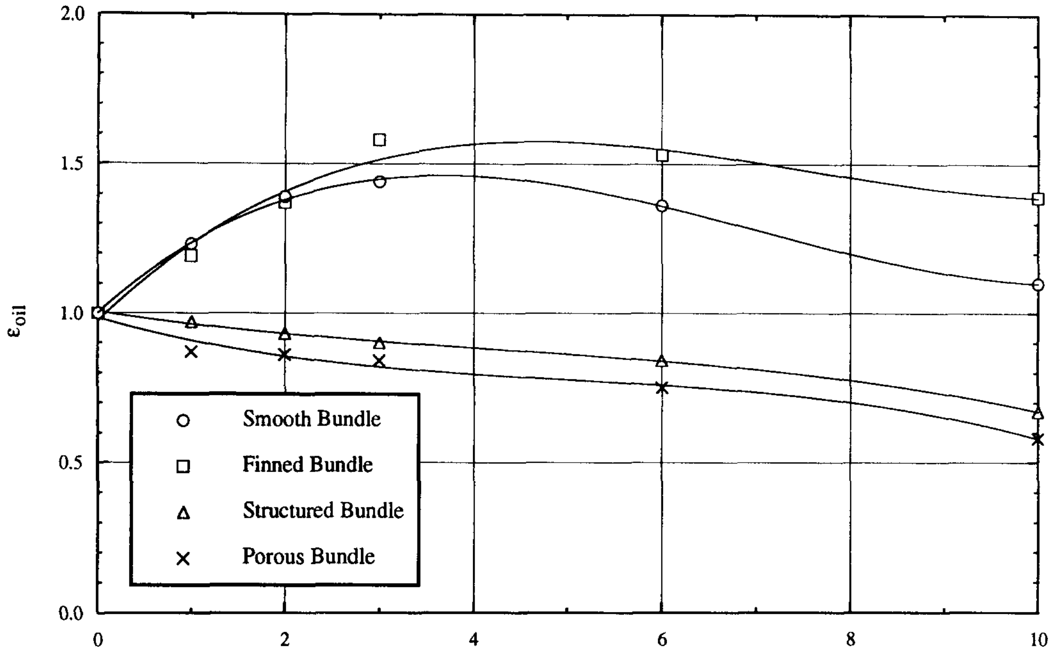

Fig. 9. Effect of oil at 30 kW m^{-2}.

found at other heat fluxes, although magnitudes were slightly different: with increasing heat flux, the maximum value of ε_{oil} decreased slightly for the smooth, but increased slightly for the finned tube bundle. As mentioned above, the drop-off in performance for the porous coated bundle was particularly significant at high heat fluxes and large oil concentrations where, at 50 kW m^{-2} and 10% oil, the value of ε_{oil} dropped to 0.38. However, at design heat fluxes (15–30 kW m^{-2} for most commercial flooded evaporators) and oil concentrations (<3% with separators), the re-entrant cavity bundles still provide the best overall performance.

Certain trends in Figs. 8 and 9 are very similar to those found in ref. [33] for single tubes:

- the addition of small amounts of oil increases the performance of the smooth and finned tube bundles;
- the addition of oil decreases the performance of the re-entrant cavity bundles at all heat fluxes;
- the best overall thermal performance at high oil concentrations and high heat fluxes (using a surface area based on root diameter) is for the finned bundle;
- the best overall thermal performance at typical operating oil concentrations and heat fluxes is for the porous bundle;
- the very poor performance of the porous bundle is at high heat fluxes and high oil concentrations.

The designer can estimate an average bundle performance at a given oil concentration, $\overline{h_{av,mix}}$, from appropriate† single tube data (for the same surface) in pure refrigerant ($\overline{h_s}$) from:

$$\overline{h_{av,mix}} = \overline{h_s}\varepsilon_{oil}\varepsilon_{bf}, \quad (1)$$

where ε_{oil} is the oil addition factor and ε_{bf} is the bundle factor for pure refrigerant, taken respectively from Fig. 9 and Table 3 at a heat flux of 30 kW m^{-2}. The value of $\overline{h_s}$ can also be taken from Table 3. If only single smooth tube data are available, then equation (1) could be multiplied by the enhancement ratio listed in Table 4 to obtain estimates of the average enhanced tube bundle heat-transfer coefficient.

† Appropriate here means either isolated tube data or data taken from a single tube within a bundle.

CONCLUSIONS

An experimental database has been established for pool boiling of pure R-114 and R-114–oil mixtures from smooth and three enhanced tube bundles. Based upon these bundle measurements, the following conclusions may be made.

Pure R-114

- Significant bundle effect and bundle factors have been found for a smooth, finned and structured surface bundle at low heat fluxes. As heat flux increases, these decrease to unity. For a porous bundle, these were around unity (or below) for all heat fluxes covered.
- Both the porous and structured surface bundles provide larger enhancements than the 19 fpi tube bundle at all heat fluxes. At high heat fluxes, these enhancements are comparable to those found from

single tube tests, whereas, at low heat fluxes, they are somewhat conservative.

R-114–oil mixtures

- For small quantities of oil (up to 3%), the smooth and finned tube bundles exhibit an increase in average heat transfer. Further increases in oil concentration lead to a slight drop-off in bundle performance.
- For the re-entrant cavity bundles, any addition of oil leads to a reduction in bundle performance for all heat fluxes. This is especially significant for the porous coated bundle at high heat fluxes and large oil concentrations.
- At practical design heat fluxes (15–30 kW m^{-2}) and oil concentrations (3%) for flooded type evaporators, the re-entrant cavity bundles provide the best thermal performance.

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