Chapter 16

Effects of Oil on Thermal
Performance of Heat Exchangers

SUMMARY: Miscible lubricating oils in refrigeration system working fluids can have a dramatic effect on boiling heat transfer performance and two-phase pressure drops in direct-expansion evaporators and flooded evaporators, representing one of the old unresolved problems of refrigeration heat transfer. This survey presents a summary of recent experimental work of the influence of oil on nucleate pool boiling, bundle boiling and flow boiling heat transfer coefficients and two-phase pressure drops for plain and enhanced tubes. Some methods for predicting the effect of oil on intube evaporation and intube pressure drops are also described. The influence of oil on condensation tends to be less than for evaporation and less information is available. This chapter addresses the influence of oil on boiling heat transfer only, but not on condensation heat transfer.

The general trends for evaporation of refrigerant-oil mixtures inside plain tubes are that (i) oil increases the local boiling heat transfer coefficient at low to intermediate vapor qualities on the order of 10-30% and (ii) this is followed by a sharp reduction by up to –90% at high vapor qualities with respect to pure refrigerant performance. The general trend for microfin tubes is (i) little (5-10%) or no enhancing effect of the oil on heat transfer at low and medium vapor qualities but sometimes a detrimental influence of as much as -30% and (ii) then a substantial reduction up to –90% at high vapor qualities.

Presently, there is no model that can explain why oil has an enhancing effect in plain tubes at the low and medium vapor qualities. Instead, at high vapor qualities, the large increase in local liquid viscosity explains the sharp reduction in heat transfer coefficients at high vapor qualities. Using the flow boiling model and flow pattern map of Kattan, Thome and Favrat (1998a, 1998b, 1998c), where the refrigerant-oil viscosity, $\mu_{ref-oil}$, is used in place of the pure refrigerant value, μ_{ref}, heat transfer to refrigerant-oil mixtures has been successfully predicted. No method is currently available for predicting the oil effect for microfin tubes and hence only the plain tube approach can be extrapolated to microfin tubes at present. Oil also increases two-phase pressure drops sharply at high local vapor qualities for evaporation in plain and microfin tubes. Using an oil-to-refrigerant viscosity correction factor, local two-phase pressure drops are adequately predicted for foaming and non-foaming refrigerant-oil mixtures evaporating inside plain tubes. No such method is available for microfin tubes so for now the plain tube method is extrapolated to applications to microfin tubes.

For evaporation on a single plain tube or a single low finned tube, oil normally tends to increase the boiling heat transfer coefficient up to some limiting % mass composition and then reduce it, although some tests show no augmentation effect of oil but only a detrimental effect. For evaporation on single enhanced boiling tubes, oil always tends to reduce the boiling coefficient with increasing % mass of oil composition. Effects of oil on tube bundle boiling are not yet well understood.

16.1 Introduction

Evaporation inside tubes and on the outside of tube bundles is an important heat transfer process in direct-expansion and flooded evaporators widely used for refrigeration, air-conditioning and heat pump systems. Plain tubes may be used but tubes with internal microfins (such as *Turbo-A* and *Turbo-D*) or external boiling enhancements (such as the various versions of *Turbo-B*) are now widely and routinely used in these services. Plain tubes are used in some applications, primarily for air-heated coils where the limiting

thermal resistance is on the airside of the tube; however, for these units microfin tubes are often used to achieve a more compact, economical design.

Lubricating oil from the compressor's bearings typically enters the refrigerant as a mist in a vapor compression cycle, dissolves into the liquid refrigerant in the condenser, and then circulates around the flow circuit to the evaporator. The oil may leave the evaporator as a liquid mist in the refrigerant vapor but may also build up in the evaporator tubes in the high vapor quality zone, held in place by the oil's high surface tension and viscosity. The amount of oil may seem to be negligible in these systems, often ranging from only 0.5-1.0 (% mass) of the total refrigerant charge but up to 3-5 (% mass) in severe cases. Yet, even a small oil mass fraction may have a very detrimental effect on evaporator thermal capacity in one or more of the following manners:

- decreasing the evaporating heat transfer coefficient;
- increasing the two-phase pressure drop;
- elevating the boiling point temperature;
- preventing all the refrigerant from evaporating;
- reducing the log mean temperature difference.

Considering lubricating oils in general, their normal boiling points are about 300°C (572°F) or above; hence it is very high relative to that of a refrigeration system evaporator. In addition, the vapor pressure of oil at 20°C (68°F) is only about 1 millionth that of the refrigerant, so its partial pressure in the vapor-phase is extremely negligible. Thus, the oil is essentially non-volatile at normal operating conditions, which means that the oil remains in its liquid-phase, either dissolved in the liquid refrigerant (if miscible) or is in the form of entrained droplets at the exit of the evaporator. Consequently, for a certain mass of oil in the refrigerant charge, the oil mass fraction in the liquid-phase will tend to increase along the evaporation flow path through a direct-expansion evaporator as more and more liquid refrigerant is evaporated. Thus, the highest oil mass fraction in the liquid-phase will be at the outlet of the evaporator.

As an example, let us assume that refrigerant with 1.0 (% mass) oil in the total flow of subcooled liquid leaves the condenser for the expansion device, i.e. 1 g of oil for every 100 g of liquid refrigerant and oil. For a local vapor quality of 0.25 at the exit of the expansion device (valve or capillary tube), the local oil mass fraction in the remaining liquid will be 1.33 (% mass), i.e. 1 g of oil per 75 g of the remaining liquid (74 g of liquid refrigerant + 1 g of oil) at that location. Near the exit of the evaporator at a local vapor quality of 0.95, the local oil mass fraction in the remaining liquid will be 20 (% mass), i.e. 1 g of oil in every 5 g of liquid. At the outlet of the evaporator, the local vapor quality when oil is present cannot in fact reach 1.0 since the oil cannot be evaporated at this temperature. For a nominal 1.0 (% mass) oil circulating in the unit, the highest local vapor quality that can be attained is less than 0.99 since 0.01 represents the oil in the liquid and some refrigerant will remain as liquid too. Since the local oil mass fraction increases with local vapor quality, it can thus be expected to have its dominant effect on thermal performance (heat transfer, two-phase pressure drop and local boiling point temperature) at the high quality end of a direct-expansion evaporator.

The evaporation of refrigerant/lubricating oil mixtures is thus a thermodynamic problem as well as a heat transfer problem, analogous to evaporation of zeotropic refrigerant mixtures (e.g. R-407C). However, a refrigerant-oil mixture has a temperature glide on the order of 300°C (572°F) since the oil increases the local bubble point temperature with increasing oil mass fraction, i.e. the local saturation temperature at which evaporation of a mixture takes place. Furthermore, to be thermodynamically correct and to respect the definition of the boiling heat transfer coefficient, the local bubble point temperature should be used to calculate the boiling wall superheat between the tube wall and the refrigerant-oil mixture, not the saturation temperature of the pure refrigerant, and also to determine the log-mean temperature difference.

In addition, the physical properties of the liquid-phase tend to be significantly affected by the oil, especially the liquid dynamic viscosity. Lubricating oils have dynamic viscosities on the order of 1000 times that of a liquid refrigerant at normal evaporation temperatures. Therefore, a nominal 1.0 (% mass) oil mixture will substantially increase the local liquid viscosity at the point in the evaporator where the local vapor quality reaches 0.95 and thus the oil's local mass fraction in the liquid is 20 (% mass). Hence, locally the liquid viscosity may go as high as 0.035 Ns/m^2 (35 cp) compared to 0.0002-0.00035 Ns/m^2 (0.2-0.35 cp) for pure refrigerants such as R-134a or R-22, i.e. an increase of 100-175 times! Other key physical properties are also affected, but to a much lesser extent.

The thermodynamics of refrigerant-oil mixtures are discussed in detail in Chapter 15. Refer to Chapter 15 for more information on this topic.

The first part of this review is on the influence of oil on evaporation of refrigerant-oil mixtures inside tubes. The second part will cover oil effects on evaporation outside of tubes. Other previous reviews on this subject have been presented by Schlager, Pate and Bergles (1987) and by Thome (1990, 1994b, 1995, 1996). The present review will cover the most important recent work on this topic.

16.2 Summary of Oil Effects on Evaporation inside Tubes

Figure 16.1 from Zürcher, Thome and Favrat (1997a) depicts some local flow boiling data for R-134a/oil mixtures evaporating inside a plain horizontal tube of 10.92 mm (0.430 in.) internal diameter at 3.43 bar (49.7 psia), where the local measured heat transfer coefficients with and without oil have been normalized using calculated values for pure R-134a obtained with the new intube flow boiling model of Kattan, Thome and Favrat (1998a, 1998b, 1998c). The test conditions are two fixed mass velocities of 100 and 300 kg/m^{2}s and nominal inlet oil mass fractions of 0, 0.5, 1.0, 3.0 and 5.0. These results demonstrate most of the trends typically observed in local flow boiling data for refrigerant-oil mixtures inside plain tubes:

- At low mass velocities and intermediate vapor qualities, the oil enhances heat transfer by 10% to 60% depending on the inlet oil mass fraction and flow conditions in the study;
- At higher mass velocity and intermediate vapor qualities, the oil may still enhance heat transfer but by a smaller amount (0 to 20%);
- At vapor qualities above about 0.70 for all mass velocities, oil has a very adverse influence on heat transfer, reducing heat transfer coefficients by up to 90%.

Therefore, the effect of oil on evaporation heat transfer may be very positive, insignificant or very negative, depending on the local operating conditions at that point in the evaporator. A direct-expansion evaporator, in any case, must operate in the high vapor quality region where oil effects on heat transfer are particularly adverse.

Referring again to Figure 16.1, the two mass velocities shown are 200 kg/m^{2}s (147,158 lb/ h ft^2) and 300 kg/m^{2}s (220,738 lb/ h ft^2) over the wide vapor quality range from 0.18 to 0.93. At these diverse conditions, the Kattan-Thome-Favrat flow-boiling model is seen to predict all the pure R-134a data (0% curves) to within -8% and +21%. For accurate thermal design of direct-expansion evaporators, the influence of oil on the local heat transfer coefficient must be taken into account, especially at high vapor qualities where the local coefficient becomes very low and is hence the controlling thermal resistance in the overall heat transfer coefficient. In fact, the local flow-boiling coefficient with oil at high vapor qualities may even be as low as the airside heat transfer coefficient and hence appropriate design measures should be taken if possible.
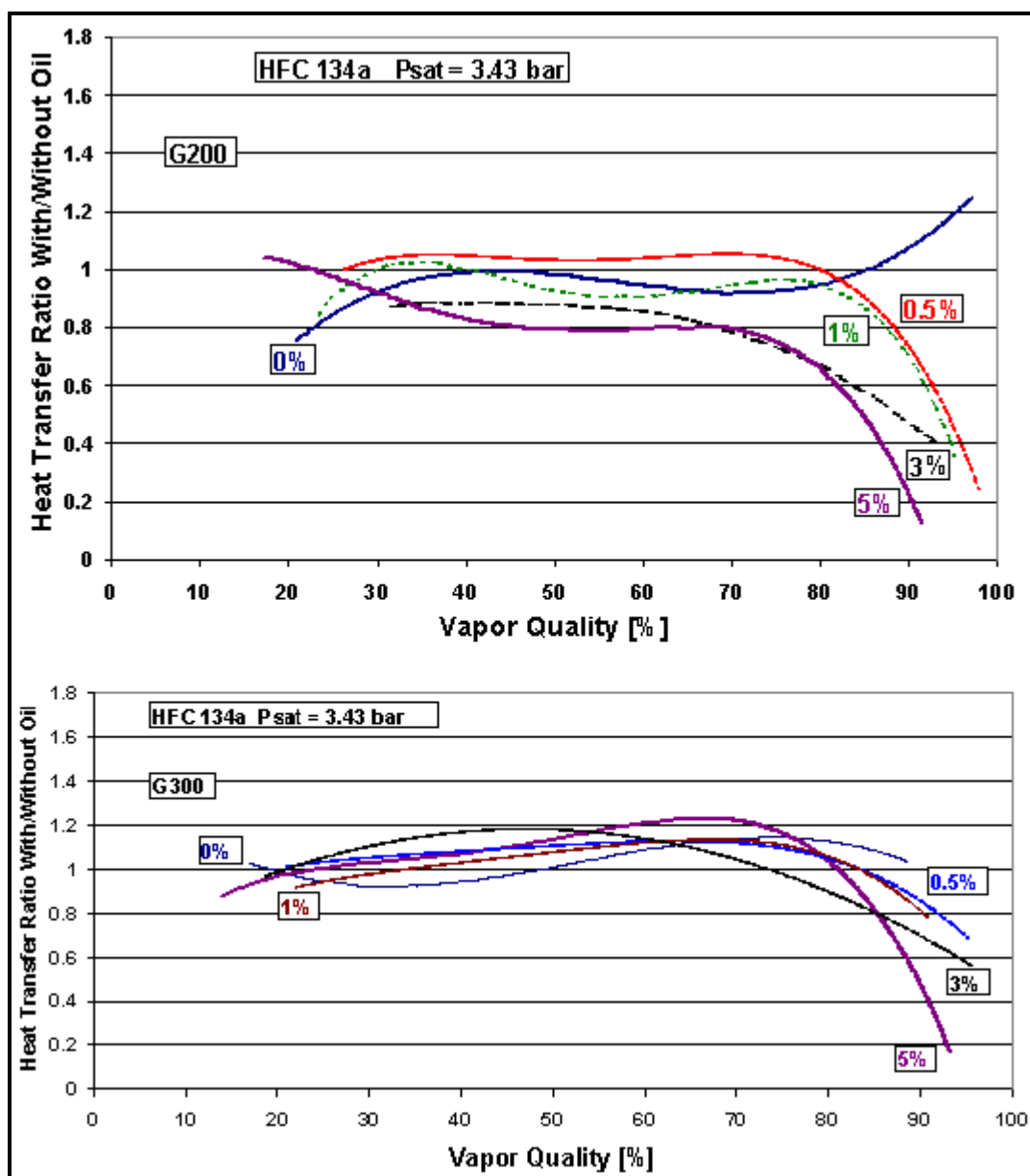
Figure 16.1. Heat transfer ratios for evaporation of R-134a/oil mixtures in a plain tube from Zürcher, Thome and Favrat (1997a) at two mass velocities at 3.43 bar (49.7 psia) [reprinted from *HVAC&R Research,* ASHRAE, Atlanta, Georgia].

For evaporation in microfin tubes, Figure 16.2 from Zürcher, Thome and Favrat (1998a) illustrates the typical effects of oil in the same format as in Figure 16.1 but for R-407C/oil mixtures. The microfin tube had an internal diameter of 11.90 mm (0.469 in.) at the root on the internal fins, 70 helical fins of 0.254 mm (0.010 in.) height and 18° helix angle, and an internal area ratio of 1.74. The local heat transfer coefficients with oil were normalized using the experimental data for pure R-407C. The test conditions were at two fixed mass velocities of 200 and 300 kg/m²s (147,158 and 220,738 lb/ h ft²) and nominal inlet oil mass fractions of 0, 0.5, 1.0, 3.0 and 5.0. All these data were reduced experimentally utilizing the Thermodynamic Approach described in Chapter 15. The typical trends observed in local microfin flow boiling data for refrigerant-oil mixtures are:

- At low mass velocities not shown and intermediate vapor qualities, the oil can either have a very detrimental effect or nearly no effect on heat transfer (from about -30% to -1%);
- At higher mass velocity and intermediate vapor qualities, the oil may either degrade heat transfer or slightly increase it (from about -20% to +10%);
- At high vapor qualities for all mass velocities, the oil has a very detrimental influence on local heat transfer, resulting in lowering heat transfer coefficients by up to 90%.
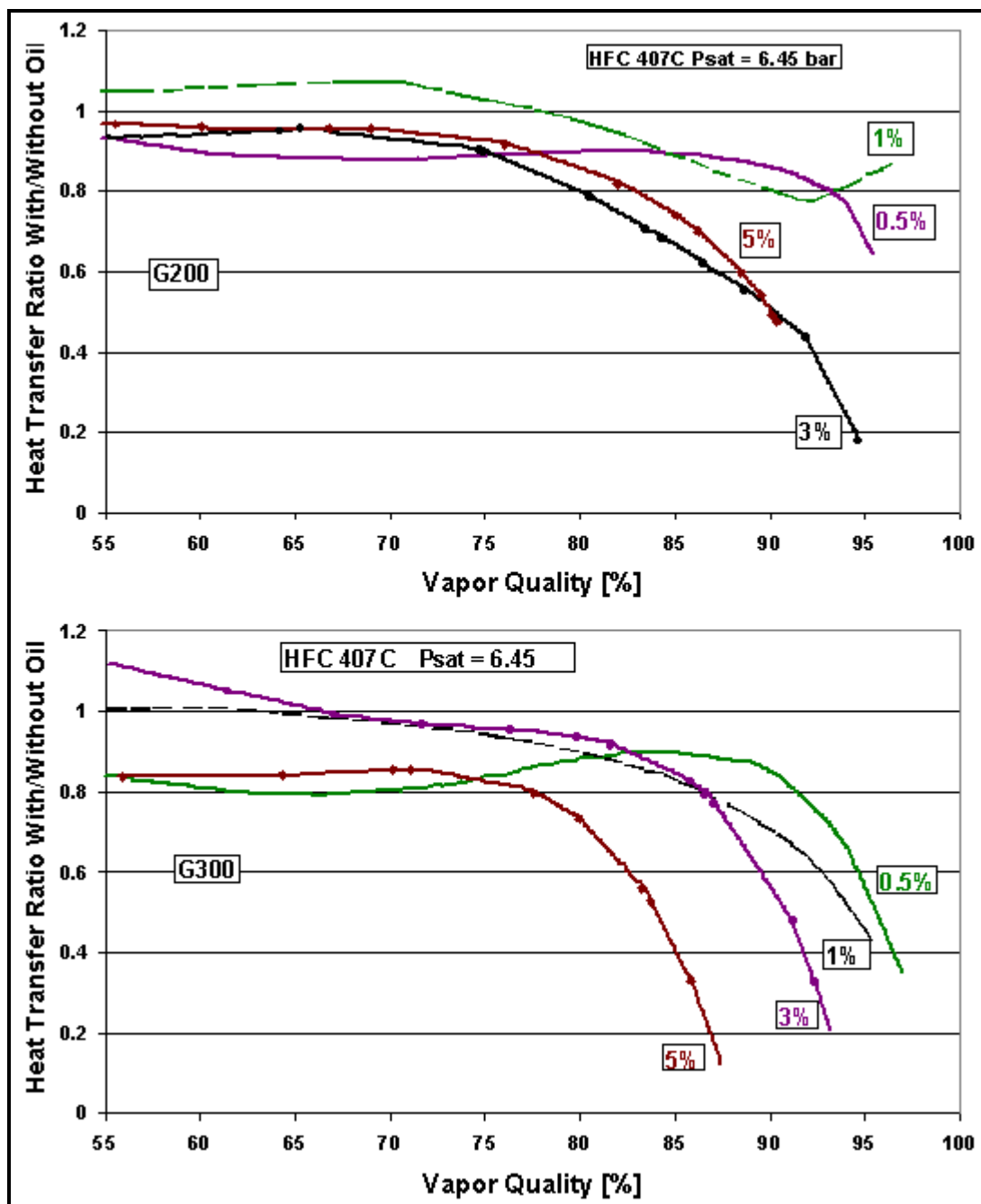

Figure 16.2. Heat transfer ratios for evaporation of R-407C/oil mixtures in a microfin tube from Zürcher, Thome and Favrat (1998a) at two mass velocities at 6.45 bar (93.5 psia) [reprinted from *HVAC&R Research,* ASHRAE, Atlanta, Georgia].

Hence, the effect of oil on evaporation heat transfer can either be slightly positive, insignificant or very negative, depending on the local operating conditions at that point in an evaporator. The overall effect of oil on the thermal capacity of an evaporator requires a thermal rating of the unit to be performed.

Twelve thermal effects of miscible lubricating oils on evaporation of refrigerants inside plain and microfin tubes have been identified, based on the results shown above and those in the literature. They are as follows:

- *Bubble point temperatures of refrigerant-oil mixtures*. Oil has a much higher boiling point than common refrigerants. Hence adding oil to a refrigerant increases its bubble point temperature relative to the saturation temperature of the pure refrigerant. This becomes increasingly important at local oil mass fractions greater than 10-20% oil mass that occur at high vapor qualities in an evaporator.

- *Oil effects on thermodynamic and transport properties*. Lubricating oils have much larger molecular weights than common refrigerants. For refrigerant-oil mixtures, this significantly affects the liquid density, where often the density difference between the oil and refrigerant is as much as 300-400 kg/m^3 (19-25 lb/ft^3), the liquid viscosity, the liquid specific heat, the critical pressure and the surface tension. Since the oil is essentially non-volatile at refrigeration operating conditions, the oil only affects liquid-phase properties but not vapor-phase properties.

- *Oil effects on enthalpy*. Adding oil to a refrigerant has primarily two effects. First, it increases the bubble point temperature (as noted above) as the mixture evaporates along the heat transfer tube. Secondly, it changes the liquid specific heat, which becomes a function of oil mass fraction. The combined influence of these two effects is that the change in enthalpy during evaporation of a refrigerant-oil mixture includes both sensible and latent heat while isobaric evaporation of a pure refrigerant only involves latent heat.

- *Oil viscosity effects on convection*. The rapid rise in local liquid viscosity with local oil mass fraction as the local vapor quality approaches 1.0 has a detrimental effect on the liquid only heat transfer coefficient, α_L, typically used as the starting basis in modeling flow boiling heat transfer (α_L in turbulent flow is inversely proportional to μ_L).

- *Oil holdup effects*. The actual local oil mass fraction may be much larger than expected at some flow conditions in the high vapor quality zone of the evaporator because the vapor shear may not be strong enough to effectively drag the viscous refrigerant-oil liquid film along the tube, and may prevent adequate oil return to the compressor. This occurs in both plain and microfin tubes.

- *Foaming*. Some oils promote the formation of foam when evaporated with particular refrigerants. The consequences of foam on the heat transfer process are poorly understood at present. The foam can be postulated to influence the onset of dry out in evaporating annular flows and shift its location to lower than expected vapor qualities (by thinning the annular liquid film) and it may have the positive affect of wetting the top of the tube under otherwise stratified flow conditions (by virtue of the height of the foam above the normal liquid level). Significant increases in local evaporating heat transfer coefficients by the oil at intermediate vapor qualities are often ascribed by researchers to foaming. However, for R-134a/oil results at 100 kg/m^2s (73,579 $lb/ h ft^2$) in the stratified-wavy flow regime in the previously mentioned study, there was significant augmentation by the oil even though *no* foaming was observed). Consequently, the influence of foaming is not well understood.

- *Wetting effects*. The larger surface tension of oil and its surfactant effect tends to increase the wetted fraction of the tube perimeter in stratified flows in horizontal tubes, which occur at low mass

velocities, and hence could be the mechanism responsible for an oil's heat transfer augmentation effect at low to medium vapor qualities in plain tubes mentioned above.

- *Influence on flow pattern transitions*. The oil's effects on physical properties have the potential to modify the threshold of transition between different flow patterns, such as annular flow and stratified-wavy flow. Manwell and Bergles (1990), for instance, have reported two-phase flow patterns for R-12/oil mixtures and discussed their observations. Videos were also obtained for the tests described earlier in Figure 16.1 and Figure 16.2.

- *Mass transfer effect of oil on heat transfer*. Oil builds up at evaporating interfaces just like the heavier component(s) in zeotropic refrigerant blends, creating a mass fraction gradient and thus increasing the bubble point temperature at the vapor-liquid interface. This in turn reduces the superheat across the evaporating film, which drives the evaporation process, and consequently adversely affects the rate of heat transfer.

- *Viscous transition to laminar film flow*. As oil becomes increasingly concentrated in the liquid phase during evaporation in the annular flow regime, i.e. in the liquid film, the film is prevented from drying out at the top of the tube at high vapor quality because the bubble point temperature of the film rises towards that of the heated wall. For refrigerant-oil mixtures, the less profound peaks in plots of heat transfer coefficient versus local vapor quality (α vs. x at constant mass flux) observed experimentally may represent a transition in the liquid annular film from a *turbulent* convection regime to the less thermally effective *laminar* convection regime. In fact, such slow flowing laminar films have been observed and reported in the literature.

- *Nucleate boiling augmentation*. Oil may increase the nucleate boiling contribution to flow boiling over the low and intermediate vapor quality range {as discussed for nucleate pool boiling of refrigerant-oil mixtures in Thome (1995)}. Even so, this does not seem likely to be particularly significant in flow boiling conditions where the nucleate boiling contribution is typically of secondary importance to the convective contribution.

- *Solubility effects*. Some refrigerant-oil combinations are soluble (and miscible) over the entire mass fraction range while others are only soluble for oil mass fractions less than 5-10 (% mass) depending on the temperature. Consequently, at high vapor qualities where the liquid can reach 30-60% oil mass, the local mixture may no longer be in the soluble range {for instance, refer to Hewitt and McMullan (1995)}.

The complex trends observed experimentally for evaporation inside plain and microfin tubes with oil in the refrigerant charge are a composite of these effects, and their impact on heat transfer depends on local flow conditions (oil mass fraction, mass velocity, vapor quality, heat flux, flow pattern, plain or enhanced surface, foaming if any, etc.).

16.3 Experimental Studies on Intube Flow Boiling

Some flow boiling studies still investigate the effect of oil on *mean* evaporating coefficients obtained over a wide change of vapor quality from inlet to outlet of the test section rather than on *local* coefficients. This is typical of industrial experimental facilities and some university test rigs too. Also, most experimental studies reported in the literature have used the pure refrigerant saturation temperature T_{sat} rather than T_{bub} to calculate the heat transfer coefficient (i.e. the oil contamination approach described in Chapter 15), and since $T_{bub} > T_{sat}$, such heat transfer coefficients are smaller than the thermodynamically

correct values, especially for vapor qualities greater than 0.8. For x < 0.8, the influence of oil on T_{bub} is relatively small, especially for mean data. This difference becomes significant at high oil mass fractions and also for microfin tubes, which have small boiling wall superheats. It should be pointed out that the only studies in the technical literature that have apparently correctly reduced their raw test measurements to refrigerant/oil boiling heat transfer coefficients utilizing bubble point temperatures and enthalpy curves (i.e. the Thermodynamic Approach described in Chapter 15) are those by Thome and coworkers. Hence, experimental heat transfer coefficients from other studies may include some combination of the real oil effects and those attributable to the improper use of T_{sat} of the pure refrigerant rather than T_{bub} of the mixture. Therefore, only the recent experimental work of Thome and coworkers will be discussed here (all though the qualitative trends in other studies are similar to these).

Nidegger, Thome and Favrat (1997) ran local R-134a/oil evaporation tests for a *microfin* tube with an internal diameter of 11.90 mm (0.470 in.), seventy 18° helical fins of 0.254 mm (0.010 in.) and internal area ratio of 1.74 at nominal oil mass fractions of 0, 0.5, 1.0, 3.0 and 5.0 (% mass). They measured the actual oil mass fractions circulating in their loop with an accurate online density flowmeter {cf. Bayini, Thome and Favrat (1995) in Chapter 15}. They used counter-current hot water in a double-pipe test section to obtain their data using the modified-Wilson plot approach. Hence, mean heat transfer coefficients were measured for subsections of 1.0 m (3.37 ft) length in their 3.0 m (10.1 ft) long test section, which gave *quasi-local* heat transfer test data for a change in vapor quality of about 0.045 to 0.12 in a subsection. The Thermodynamic Approach for refrigerant/oil mixtures was used to reduce their raw data to heat transfer coefficients, determining the bubble point temperatures of the local refrigerant-oil mixture at the inlet and outlet of their test zones and the corresponding enthalpy curves. Mobil Arctic Grade EAL 68 ester oil was used, which has a kinematical viscosity of 62.5 mm^2/s at 40°C (104°F) and was completely miscible over all test conditions.

Figure 16.3 shows their data at a mass velocity of 200 kg/m^{2}s (147,158 lb/ h ft^2) plotted in two different ways, first in the traditional way as a function of the nominal inlet oil mass fraction in the subcooled liquid and then in a more appropriate way as a function of the local oil mass fraction corresponding to the local vapor quality. Adding oil to R-134a always reduced heat transfer at vapor qualities before the peak in α vs. x, but after the peak a few data were higher than the pure R-134a values, apparently because a small amount of oil retarded the onset of dry out (or eliminated it since the refrigerant-oil mixture cannot dry out). At high local oil mass fractions, the heat transfer coefficients were well below those for pure R-134a. They reported that no foaming occurred in any of their R-134a/oil tests.

Zürcher, Thome and Favrat (1997a) ran *plain* tube tests with R-134a/oil at the same test conditions, i.e. constant inlet saturation pressure of 3.40 bar (49.3psia) corresponding to T_{sat} = 4.4°C (40°F) for pure R-134a. Figure 16.4 depicts their test data for a plain tube of 10.92 mm (0.430 in.) internal diameter. The oil mass fractions of 0.5% and 1.0% tended to increase boiling performance at low to intermediate vapor qualities while 3.0% and 5.0% oil tended to decrease it. At high vapor qualities, the 0.5% oil mixture had a performance similar to pure R-134a until after the peak in α vs. x. The mixtures with higher levels of oil suffered very severe performance degradations at high vapor quality, falling by as much as –90% relative to the pure R-134a data and noticeably shifted the peak in α vs. x towards lower vapor qualities and to lower amplitudes.
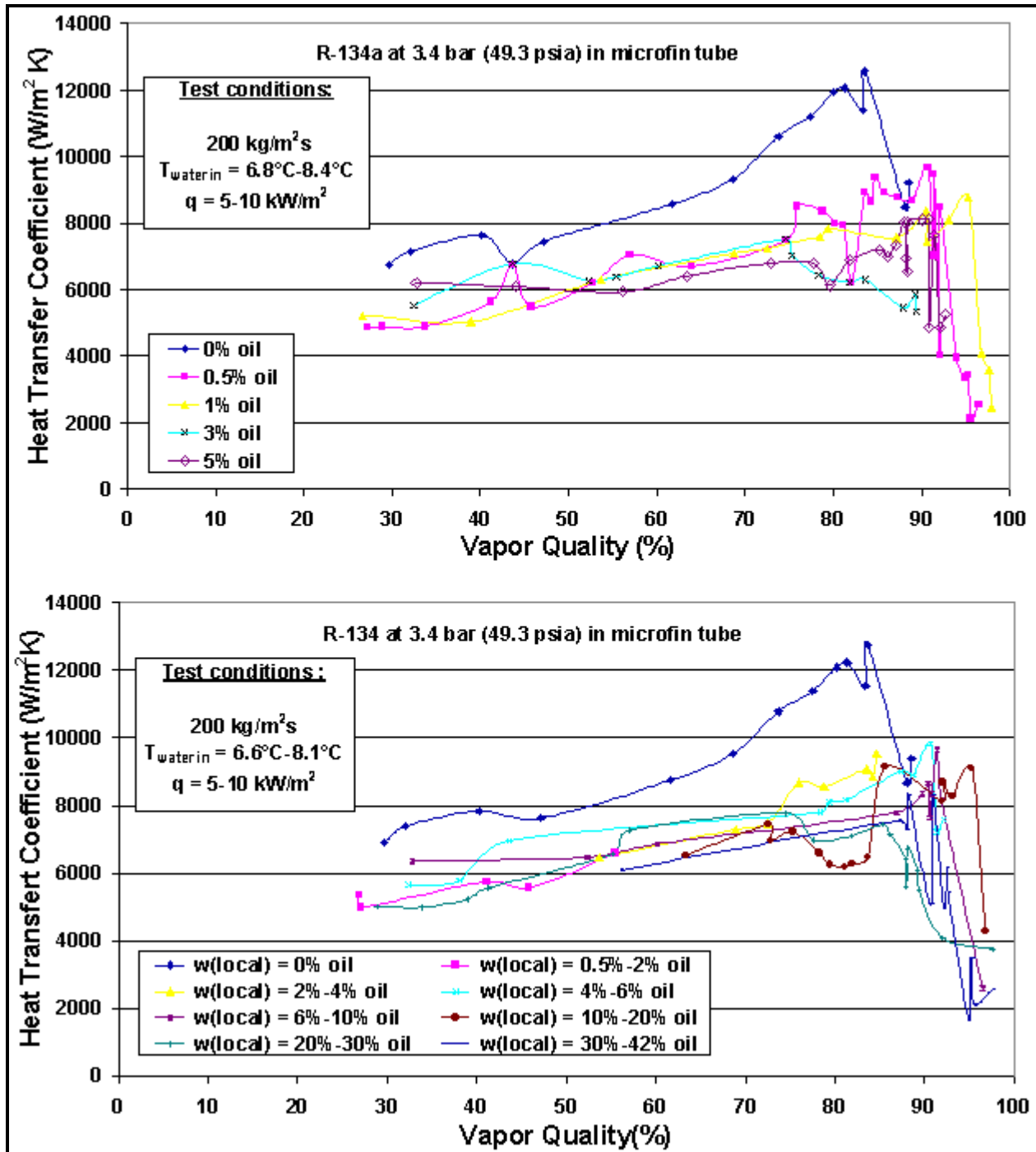
Figure 16.3. Nidegger, Thome and Favrat (1997) flow boiling data for R-134a/oil mixtures in a microfin tube. Top: Plotted as inlet oil mass fraction. Bottom: Plotted as local oil mass fraction [reprinted from *HVAC&R Research,* ASHRAE, Atlanta, Georgia].

Zürcher, Thome and Favrat (1998a) ran local R-407C/oil evaporation tests for a *microfin* tube test section for nominal inlet oil mass fractions of 0, 0.5, 1.0, 3.0 and 5.0 (% mass) at 6.45 bar. The microfin tube was the same tested previously with R-134a/oil mixtures and described above. They reduced their data using the Thermodynamic Approach for refrigerant/oil mixtures and oil mass fractions were measured online. They observed that their inlet and outlet temperature measurements for the test fluid compared well with

their calculated values of bubble point temperatures of the temperature glides for both R-407C and R-407C/oil mixtures as a function of vapor quality. [They also presented a complete set of thermodynamic equations for describing R-407C mixtures valid from -5°C to +50°C (23°F to 122°F) for reducing test data and for thermal design of R-407C evaporators and condensers]. The same Mobil Arctic Grade EAL 68 ester oil as in the prior R-134a tests was used (always completely miscible according to the oil manufacturer). Their data were quasi-local data obtained with modified-Wilson plot tests. Figure 16.2 presented earlier shows their data as normalized values relative to pure R-407C. Adding oil to R-407C always reduced heat transfer at all vapor qualities except for a minor increase of 6-8% over part of the range for 3.0 % oil mass that probably is not significant. This combination of refrigerant and oil caused a lot of foaming to occur, which seemed to intensify with increasing oil mass fraction, mass velocity and local vapor quality.

Zürcher, Thome and Favrat (1998b) ran the analogous *plain* tube tests with R-407C/oil at similar test conditions, i.e. constant inlet saturation pressure of 6.45 bar (93.5 psia) corresponding to a saturation temperature of 4.4°C (40°F) for pure R-407C at a vapor quality of 0.0. Figure 16.5 depicts their 10.92 mm (0.430 in.) bore, plain tube test data at 300 kg/m^{2}s (220,738 lb/h ft^2) that have been normalized by the Kattan, Thome and Favrat (1998c) flow boiling model. At high vapor qualities, the heat transfer coefficients of the refrigerant-oil mixtures suffered very severe degradations with increasing oil mass fraction, falling by as much as -90%. Note that the Kattan-Thome-Favrat flow boiling model, including the mass transfer effect of the mixture, predicts their pure R-407C data (0% curve) to within a maximum error of ±20% and a mean error of less than 10%.

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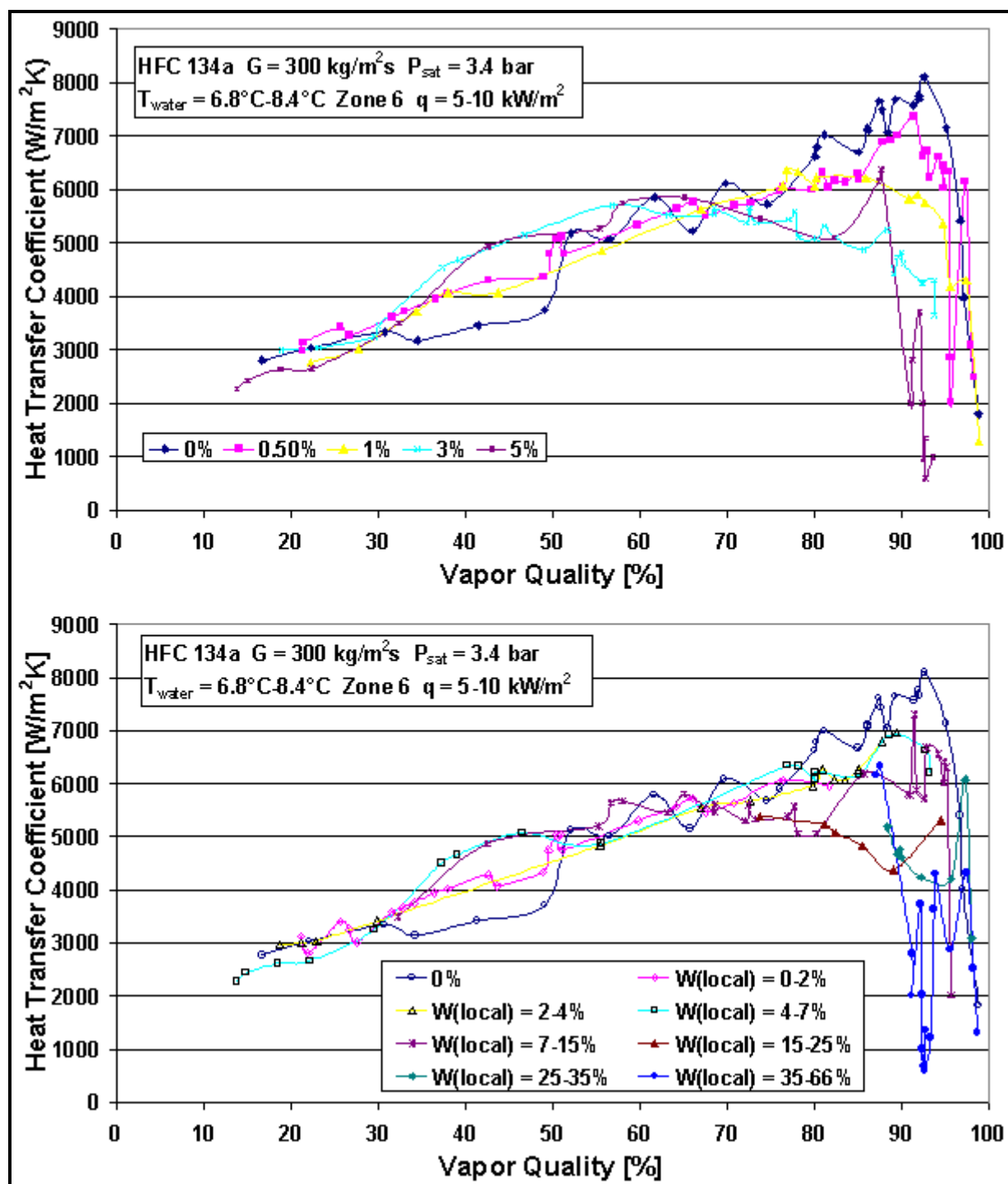*Engineering Thermal Innovation™*

Figure 16.4. Zürcher, Thome and Favrat (1997a) flow boiling data for R-134a/oil mixtures in a plain tube. Top: Plotted as inlet oil mass fraction. Bottom: Plotted as local oil mass fraction [reprinted from *HVCA&R Research*, ASHRAE, Atlanta, Georgia].
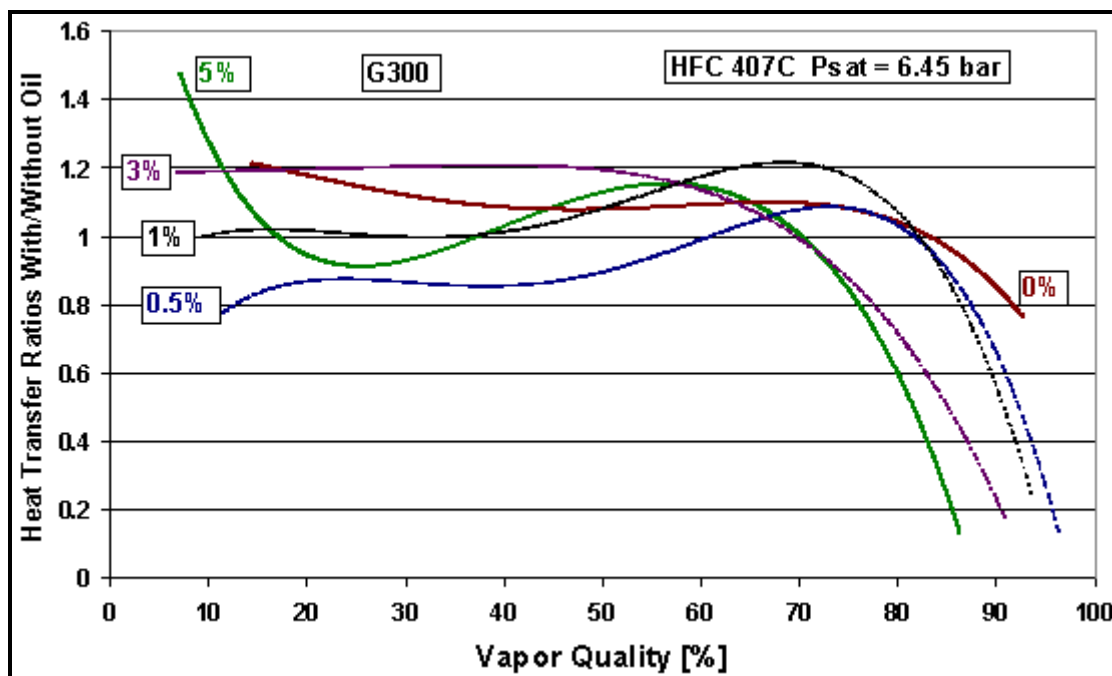
Figure 16.5. Zürcher, Thome and Favrat (1998a) flow boiling data for R-407C/oil mixtures in a plain tube plotted as normalized coefficients at 6.45 bar (93.5 psia) [reprinted from *HVAC&R Research,* ASHRAE, Atlanta, Georgia].

The phenomenon of oil build up in plain and microfin tubes during evaporation was quantified in Zürcher, Thome and Favrat (1998a) for the first time. Figure 16.6 depicts their oil mass fractions measured online for subcooled refrigerant-oil liquid flowing into their preheater compared to the 5 (% mass oil) mixture charged into the flow boiling loop. At local vapor qualities below about 0.72 (which corresponds to a local oil mass fraction of 18%), no oil held up in the two 3-meter long microfin tube test sections connected in series, as indicated by the online value which matches that charged into the loop. In sharp contrast, at high outlet vapor qualities, oil did build up, becoming more severe with decreasing mass velocity. The oil does not actually remain stationary within the microfin tube but the flow of the very viscous refrigerant-oil mixture at high vapor qualities is extremely slow, as observed in a tubular sight glass located directly at the exit of the microfin tube. The flow of the annular film has in fact become laminar and the local oil mass fraction builds up beyond that estimated from equilibrium flow conditions. As a consequence, the local oil mass fraction at high vapor qualities in microfin tubes is higher than that calculated, which means that the actual local bubble point temperature is higher than that estimated assuming no local build up of oil. *In similar tests with the plain tube test section, the same oil build up phenomenon manifested itself, beginning at only slightly higher vapor qualities.*

The amount of oil build up and its vapor quality threshold is directly dependent on the nominal oil composition circulating in the loop. The above example is perhaps a worst-case scenario, i.e. 5% oil, while oil compositions of 0.5 to 1% (more typical of industrial practice) will only reach that point at close to complete evaporation. For example, extrapolating the above threshold for oil build up at a vapor quality of 0.72 for 5% oil to a 1% oil mixture at that same mass velocity, the threshold to oil build up in the evaporator would occur at a vapor quality of 0.944 while for a 0.5% oil mixture it would occur at 0.972. Hence, while oil build up is a possible problem for proper operation, the actual conditions will have a large influence on its onset and the magnitude of its effect.
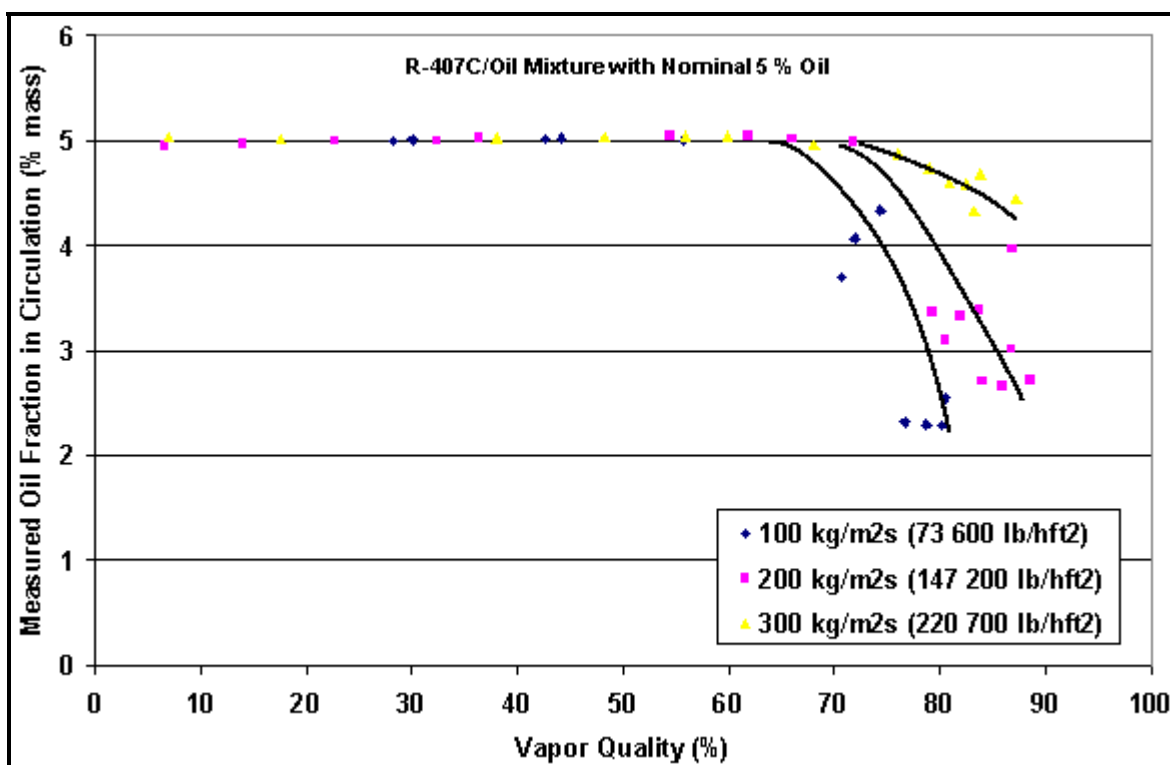
Figure 16.6. Online oil mass fraction measurements illustrating the effect of mass flow rate on oil holdup in their microfin tube test section with R-407C for a nominal mass fraction of 5 (% mass) oil [reprinted from *HVAC&R Research,* ASHRAE, Atlanta, Georgia].

16.4 Modeling Oil Effects on Flow Boiling Heat Transfer in Plain Tubes

Zürcher, Thome and Favrat (1998b) showed that the sharp reduction in heat transfer coefficients at high vapor quality caused by oil can be accurately predicted with the Kattan, Thome and Favrat (1998a, 1998c) flow boiling model and flow pattern map, if the pure refrigerant's liquid viscosity μ_{ref} is replaced with that of the local refrigerant-oil viscosity $\mu_{ref\text{-}oil}$. For the mass velocities of 200 and 300 kg/m²s (147,158 and 220,738 lb/h ft²) for R-134a/oil and R-407C/oil at x > 0.70, i.e. where there is the most severe effect, they predicted these data to a standard deviation of 22.3%, which is only slightly higher than for the same refrigerants without oil in this range of vapor quality, i.e. about 17%.

To incorporate this viscosity effect, the local liquid dynamic viscosity of the refrigerant-oil mixture $\mu_{ref\text{-}oil}$ is calculated using the Arrhenius mixing law, which after rearranging into a convenient form is:

$$\mu_{ref-oil} = \left(\mu_{ref}\right)^{(1-w)}\left(\mu_{oil}\right)^{w} \qquad\qquad [16.4.1]$$

where w is the local oil mass fraction in the liquid, which in terms of the local vapor quality x and the inlet oil mass fraction w_{inlet} is:

$$w = w_{inlet}/(1-x) \qquad\qquad [16.4.2]$$

where μ_{ref} is the liquid dynamic viscosity of the pure refrigerant and μ_{oil} is the liquid dynamic viscosity of the pure oil, all evaluated at the local bubble point temperature of the mixture. Their data cover local liquid viscosities up to about 0.0035 Ns/m^2 (35 cp), which is up to 150 times the values of pure R-134a and R-407C.

Figure 16.7 shows their simulation applying this method to R-134a with Mobil EAL 68 oil for several inlet oil compositions. The resulting curves capture the shifts of the peak in α vs. x to lower vapor qualities together with their smaller amplitude peaks.

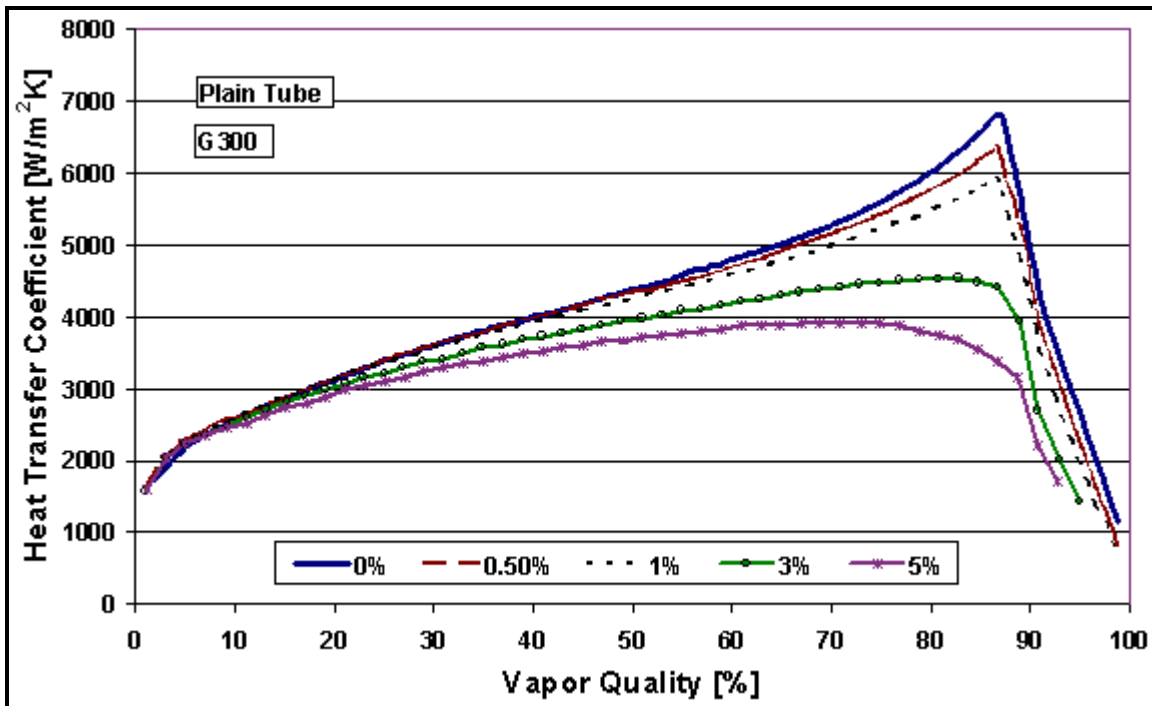**Figure 16.7. Predicted variation in R-134a/oil flow boiling coefficients in a plain tube at q = 10 kW/m^2 (3170 Btu/h ft^2) at 3.43 bar (49.7 psia) [reprinted from *HVAC&R Research*, ASHRAE, Atlanta, Georgia].**

As another approach, Thome (1997b) proposed a multiplier be applied to pure refrigerant heat transfer coefficients, either predicted or measured, to obtain the flow-boiling coefficient with oil, $\alpha_{ref-oil}$. The multiplier $[\mu_{ref}/\mu_{oil}]^{0.26w}$ for plain tubes is applied to the pure refrigerant heat transfer coefficient α_{ref} as:

$$\alpha_{ref-oil}/\alpha_{ref} = [\mu_{ref}/\mu_{oil}]^{0.26w}$$ [16.4.3]

where w is the local oil mass fraction. This approach and that above are only reliable if the mass flow rate is at or above about 200 kg/m^{2}s (147,158 lb/h ft^2) and $0.05 \geq w_{inlet}$.

16.5 Modeling Oil Effects on Flow Boiling Heat Transfer in Microfin Tubes

No prediction method for refrigerant-oil effects on microfin tubes has yet been proposed. In the absence of other alternatives, the multiplier $[\mu_{ref}/\mu_{oil}]^{0.26w}$ for plain tubes can be applied as a correction to local pure refrigerant microfin experimental data as an interim approach.

16.6 Modeling Oil Effects on Two-Phase Pressure Drops for Plain Tubes

Numerous test measurements of two-phase pressure drops for evaporation of refrigerant-oil mixtures are available for plain and microfin tubes, both for quasi-local pressure drops over a narrow change in vapor quality and mean pressure drops over a wide change in vapor quality. Space limitations do not allow all this experimental work to be reviewed here. Similar to evaporation, quasi-local two-phase pressure drop data are more valuable than mean data since the higher mass fraction of oil in the liquid-phase at high vapor qualities results in a significant increase in two-phase frictional pressure drop while only a minor effect of oil occurs at lower vapor qualities.

As an example, Figure 16.8 depicts the plain tube pressure drop data for R-134a and R-407C mixed with Mobil Arctic EAL 68 oil obtained by Nidegger, Thome and Favrat (1997) and Zürcher, Thome and Favrat (1998b), respectively, during their boiling tests described earlier. Their quasi-local refrigerant-oil measurements were normalized by the corresponding pure refrigerant values, and polynomial curve fits are shown in the graphs at a mass velocity of 300 kg/m^{2}s (220,738 lb/h ft^2) for vapor qualities greater than 55%. The influence of oil increases with its local oil mass fraction in the liquid-phase, where the local refrigerant-oil liquid viscosity can be calculated by Equation (7).

Friedel (1979) is just one of many two-phase frictional pressure drop correlations available in the literature for plain tubes and has been recommended for general use in Collier and Thome (1994, 1996). The frictional two-phase pressure drop gradient (dp/dz)$_{frict}$ is defined as

$$(dp/dz)_{frict} = 2 f_L \, \dot{m}_{total}^2 (\rho_L d_i) \phi_{Lo}^2 \qquad [16.4.4]$$

where ρ_L is the density of the liquid-phase, f_L is the Fanning friction factor for turbulent flow for all flow as liquid, $\dot{m}_{total}$ is the total mass velocity, d_i is the tube internal diameter, and ϕ_{fo}^2 is the two-phase friction multiplier. The influence of the oil on liquid viscosity is the dominant oil related factor on the two-phase pressure drop. Adding an empirical correction parameter n to the Arrhenius mixing law for liquid viscosities, i.e. Eq. [16.4.1], and rearranging, the following factor was obtained by Zürcher, Thome and Favrat (1998b):

$$f_{oil} = \left[\mu_{ref-oil}/\mu_{ref}\right]^n = \left[\mu_{oil}/\mu_{ref}\right]^{nw} \qquad [16.4.5]$$
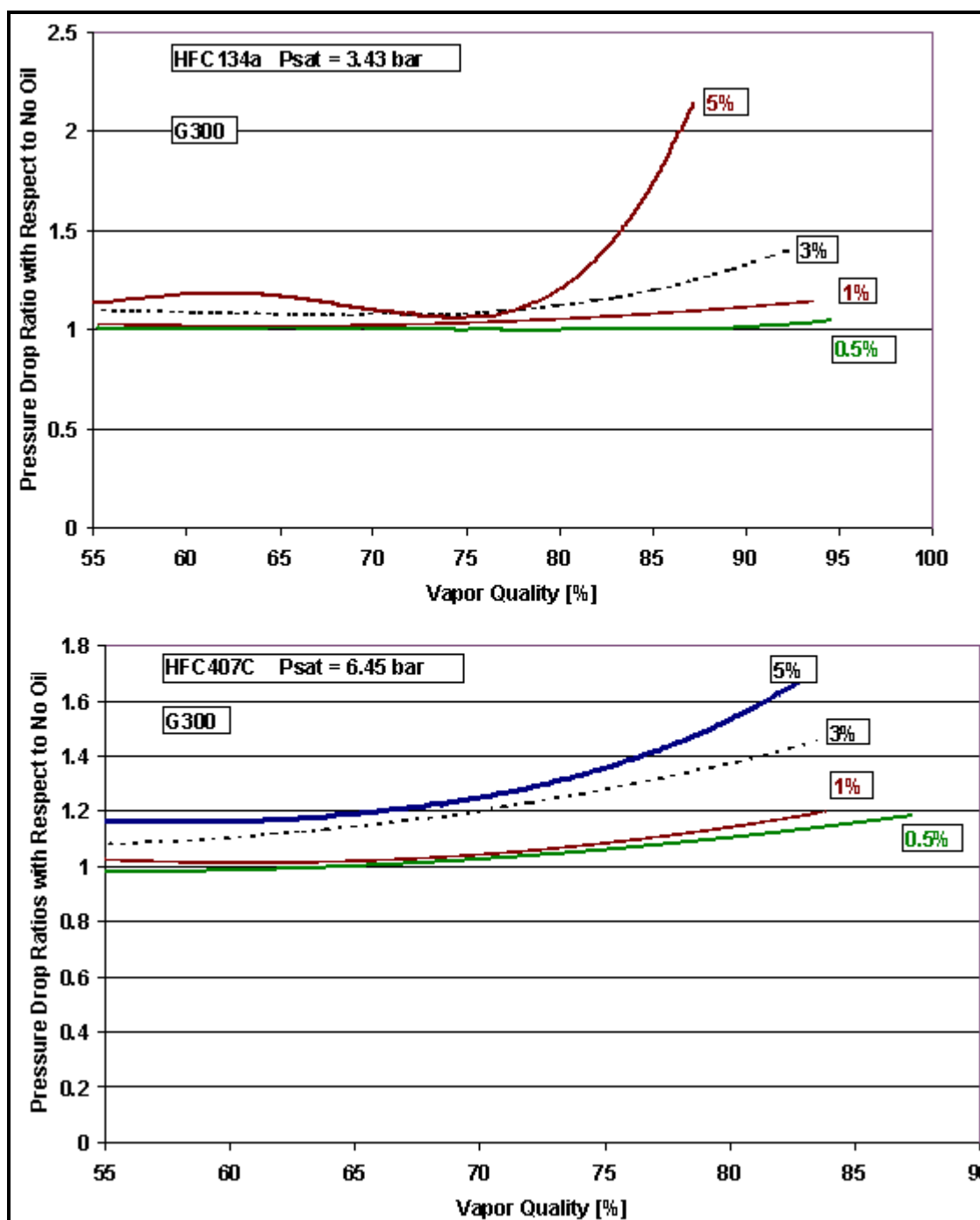
Figure 16.8. Two-phase pressure drop ratios for evaporation of refrigerant-oil mixtures in a 10.92 mm (0.430 in.) diameter tube at 300 kg/m2s (220,738 lb/h ft2). Top: R-134a; Bottom: R-407C [reprinted from HVAC&R Research, ASHRAE, Atlanta, Georgia].

where w is the local oil fraction in the liquid-phase determined with Eq. [16.4.2]. Thus the two-phase friction multiplier for refrigerant-oil mixtures becomes:

$$\left(\phi_{Lo}^2\right)_{ref-oil} = \left(\phi_{Lo}^2\right)_{ref} f_{oil} = \left(\phi_{Lo}^2\right)_{ref} \left[\mu_{oil}/\mu_{ref}\right]^{nw} \tag{16.4.7}$$

where n = 0.18355 without foaming (R-134a/oil mixtures) and

$$n = w\left(3.583w_{inlet} + 0.0616\right)$$
[16.4.7]

with foaming (R-407C/oil mixtures). Overall standard deviations at mass velocities of 200 and 300 kg/m²s (147,158 and 220,738 lb/h ft²) with inlet oil fractions from 0.005 to 0.05 and w from 0.01-0.4 were 9.2% and 7.3% for R-134a/oil and R-407C/oil mixtures, respectively. Thus, the effect of oil can easily be included in local two-phase pressure drop calculations.

16.7 Modeling Oil Effects on Two-Phase Pressure Drops for Microfin Tubes

For microfin tubes, in the absence of a specific method, the above approach for plain tubes can be utilized to estimate the influence of oil on microfin tubes.

16.8 Nucleate Pool Boiling of Refrigerant-Oil Mixtures

In refrigeration systems with flooded evaporators, operational oil mass fractions are normally in the range from 0.5% to 3.0% oil, depending on the type of compressor and whether or not an oil separator is used and its effectiveness. This range in values refers to the oil mass in the subcooled liquid prior to the expansion device; the local oil mass fraction in the liquid rises rapidly in the evaporator as the refrigerant evaporates into the vapor-phase. At the exit of an evaporator, the local mass fraction in the liquid can reach 50% oil or more.

In recent years it has become common for academic tests to include oil effects as part of nucleate pool boiling studies for refrigerants on enhanced boiling surfaces, which has been reviewed in Thome (1996). The present review will present the highlights and conclusions drawn from them.

Before proceeding, the bubble point temperature T_{bub} and the definition of mixture heat transfer coefficients need to be discussed. The effect of oil on bubble point temperatures in mixture pool boiling tests is not severe since 1-5% oil has only a minimal effect on T_{bub} compared to the pure refrigerant saturation temperature, e.g. about 0.1-0.2°C (0.2-0.4°F). Yet, for an enhanced boiling surface, this represents about 10-20% of the measured boiling superheat. Therefore, it is very important as pointed out in Thome (1994b) to use the bubble point temperature T_{bub} of the mixture to determine the boiling heat transfer coefficient α from its definition, i.e.

$$\alpha = q/\left(T_w - T_{bub}\right)$$
[16.9.1]

and not the saturation temperature T_{sat} of the pure refrigerant with the expression

$$\alpha = q/\left(T_w - T_{sat}\right)$$
[16.9.2]

where T_w is the heated wall temperature. The bubble point temperature can be calculated either with the generalized method presented in Chapter 15 or measured as the bulk liquid temperature during the tests. Only experimental studies that have correctly reduced their boiling data using the bubble point temperature of the refrigerant-oil mixture are described below to avoid showing incorrect trends in the data.

Memory, Bertsch and Marto (1993) ran nucleate pool boiling tests for pure refrigerant R-124 and its mixtures with an alkyl benzene oil. These single tube tests studied boiling on plain and enhanced tubes at a saturation temperature of 2.2°C (36°F). Figure 16.9 depicts the effect of oil mass fraction on these tubes boiling heat transfer coefficients at q = 25,000 W/m^2 (7926 Btu/h ft^2), normalized by their corresponding values for pure R-124. For the plain tube, boiling performance increased up to 22% at 6% oil while for a small amount of oil (0.5%) no change occurred relative to the pure R-124 value. For the plain tube this is quite a common trend where the maximum occurs for an oil mass fraction in the range from 3-6% and the maximum increase in performance is typically from 20-40%.
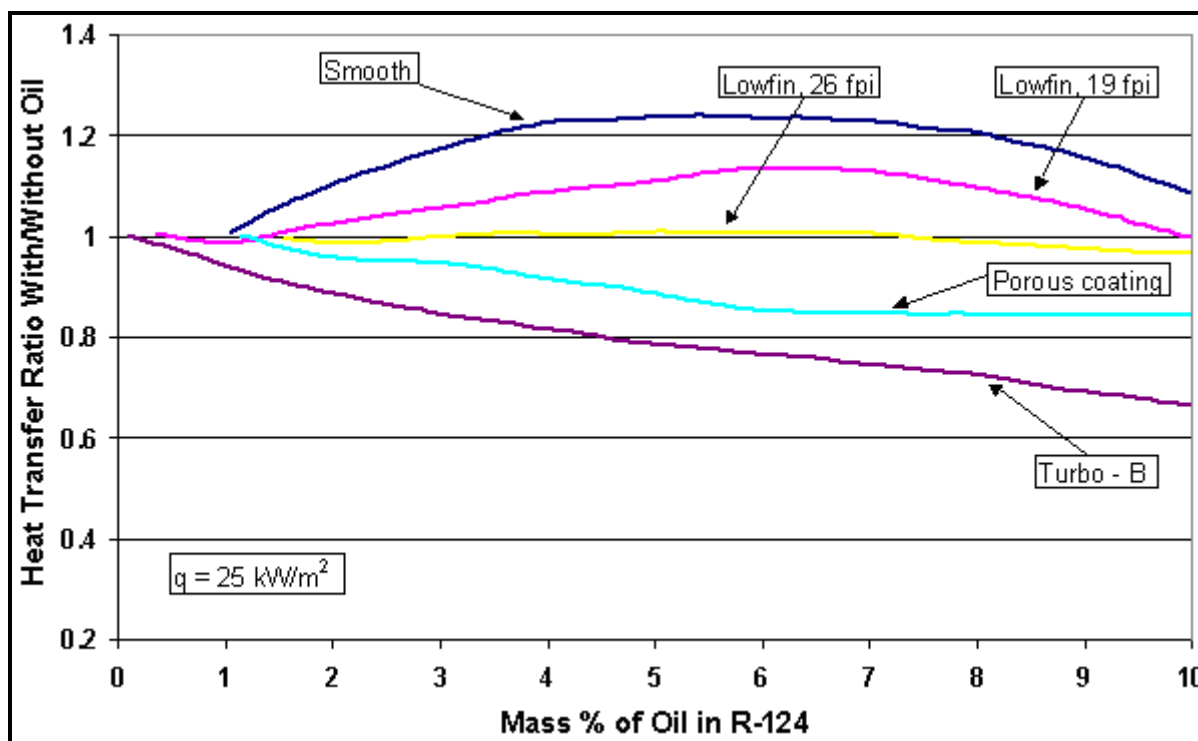

Figure 16.9. Boiling of R-124 with oil on plain and enhanced tubes by Memory, Bertsch and Marto (1993).

For the 19-fpi, low-finned tube they tested, a similar trend to the plain tube was observed. On the other hand, their 26 fpi low-finned tube had essentially no oil effect over the entire oil mass fraction range tested. For the High Flux tube instead, the boiling coefficients dropped off monotonically to an 18% reduction at 10% oil. Similarly, the Turbo-B tube was adversely affected, its performance decreasing by 35% for 10% oil. While it would be extremely unusual operating conditions for a refrigeration system to be operating with 10% oil in its charge, at local evaporating conditions within the tube bundle this local oil fraction is easily reached but only near the top of the bundle. The level of improvement or degradation caused by the oil is a function of heat flux, where the plain and low finned tube coefficients tend to improve with increasing heat flux while the two enhanced boiling tubes declined further. Even so, the two enhanced boiling tubes still significantly outperformed the two low finned tubes.

Memory, Sugiyama and Marto (1995) continued the above tests with equivalent tests for R-114/oil mixtures at 2.2°C (36°F) using the same tubes for oil mass fractions from 0-10%. They tested a completely miscible mineral oil (York-C), which caused foaming during their tests. They observed that a small amount of oil (3%) increased the boiling coefficient compared to pure R-114 for the two low finned

tubes they tested. In contrast, oil decreased performance for their plain tube and for all reentrant types of enhanced boiling tubes (High Flux, Turbo-B, Gewa-YX, Gewa-T and Thermoexcel-HE).

Webb and McQuade (1993) investigated the effect of oil on boiling of R-123 and R-11 using a mineral oil. Opposed to the above study, their plain tube displayed a monotonic degradation in performance with increasing oil mass fraction relative to pure R-123 at a heat flux of 30 kW/m^2, i.e. no maximum as in the above study. For 5% oil in fact, the plain tube's performance decreased by 15%. In comparison, the Gewa SE enhanced boiling tube experienced more degradation than the plain tube, falling by 24% at 5% oil. Similar to the R-124 tests, the Turbo-B tube had a large degradation with its boiling performance dropping by 28% relative to its pure refrigerant value.
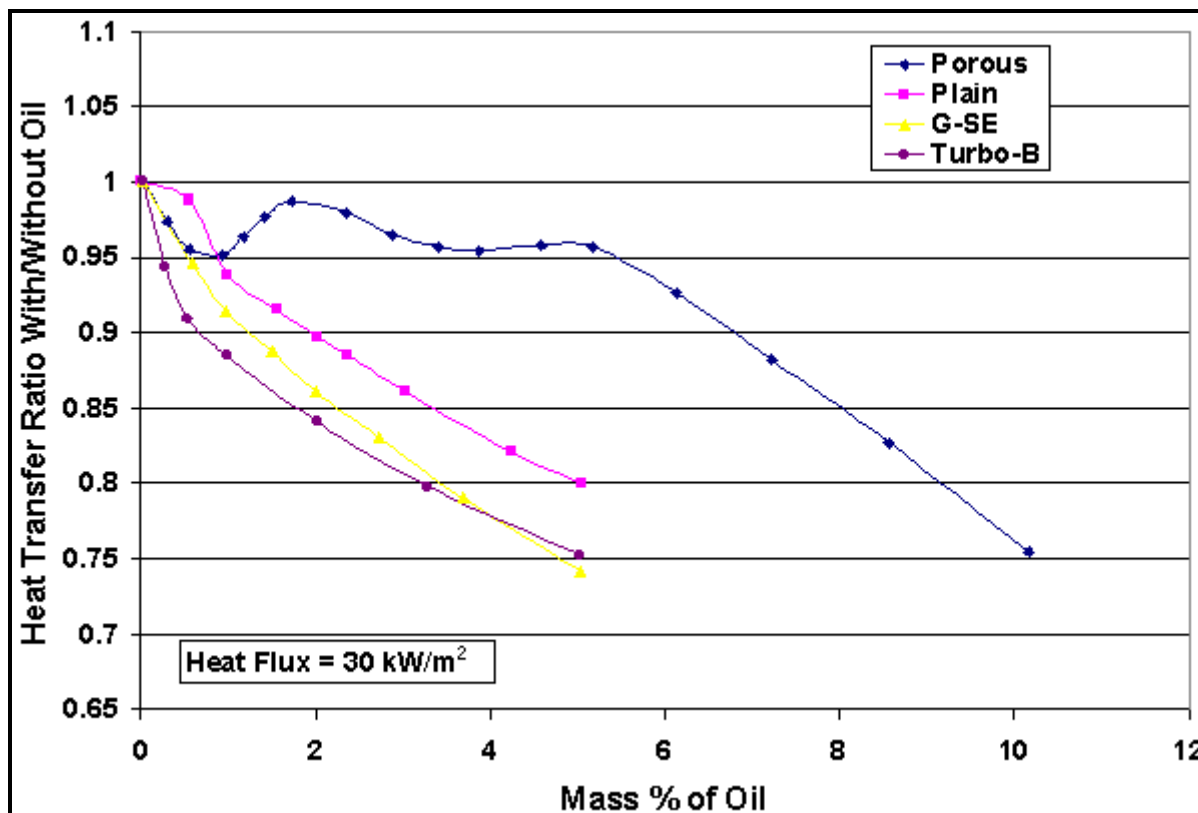

Figure 16.10. Webb, Chien, McQuade and Imadojemu (1995) R-123/oil results normalized by their pure fluid coefficients.

Webb, Chien, McQuade and Imadojemu (1995) ran further tests for R-123/oil mixtures on plain, High Flux, Gewa-SE, and Turbo-B tubes over the range of 0-10% oil. The tests were run at a temperature of 4.4°C (40°F) and they correctly used the measured value of T$_{bub}$ in the liquid pool to reduce their data. They tested a mineral oil identified as GA-155 supplied by the Mobil Research and Development Company. They observed foaming for oil mass fractions of 2% or larger. Figure 16.10 depicts their test results normalized by the pure refrigerant heat transfer coefficients at a heat flux of 30 kW/m^2 (9512 Btu/h ft^2). Surprisingly, the High Flux tube had little degradation in performance for oil mass fractions less than 5%, which is contrary to what Marto and coworkers found for R-114/oil and R-124/oil mixtures on this tube. The other results for R-123/oil were similar to those of Marto and coworkers with performance dropping off monotonically with increasing oil.

Palm (1995) did tests with R-134a and oil for plain and enhanced tubes. Figure 16.11 depicts his results, which give similar trends to those illustrated above.
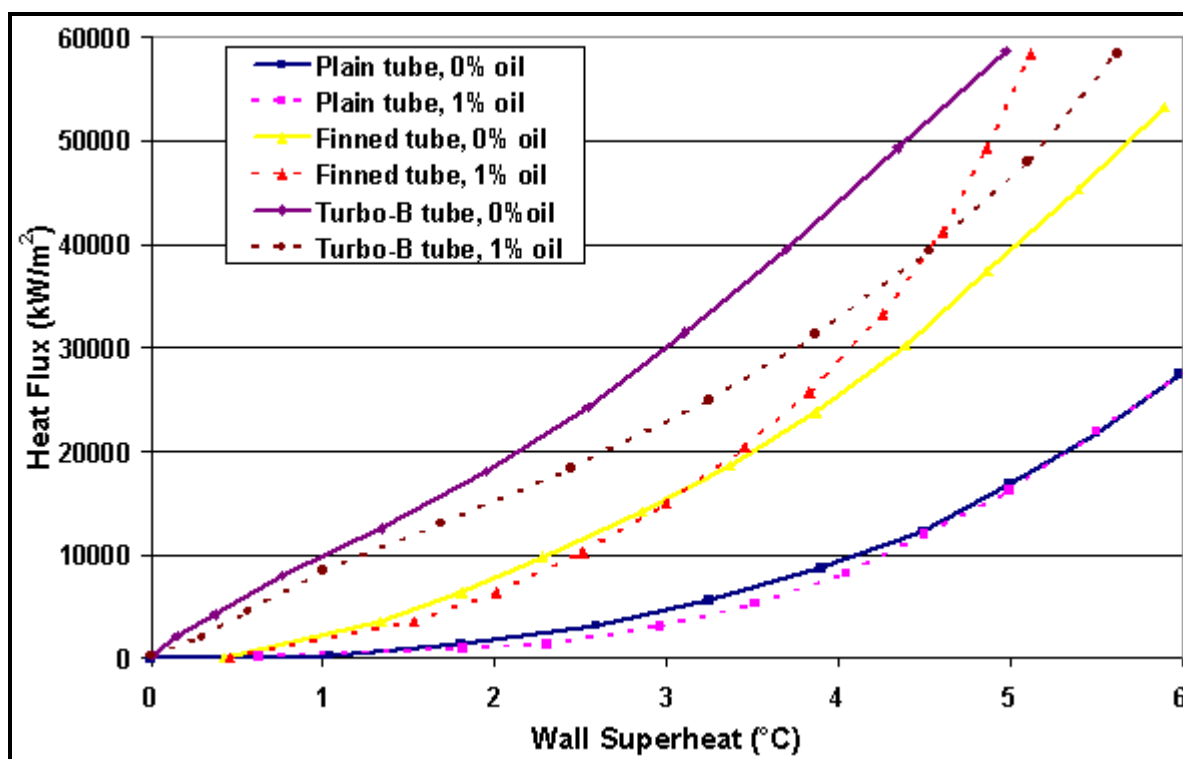
Figure 16.11. Palm (1995) R-134a/oil pool boiling results.

Chen and Tuzla (1996) obtained Turbo-BII pool boiling coefficients at 4.4°C (40°F) for R-22, R-134a and three refrigerant blends with 1% and 5% oil mass fractions. They used hot water heating in accurately run tests. Figure 16.12 shows their results with 5% oil. R-22 was tested with a petroleum based York-C oil while the others were tested with an ester type Mobil EAL Arctic 46 lubricant. With 1% oil, the Turbo-BII boiling performance for R-22, R-134a and the azeotropic blend fell by 10-30% while about 25-40% with 5% oil. The Turbo-BII tube did not suffer a fall in performance from the oil for the two-zeotropic blends. Hence, while their non-oil boiling performance penalty in an evaporator with respect to R-22 is substantial, but it is less so once oil effects are included.
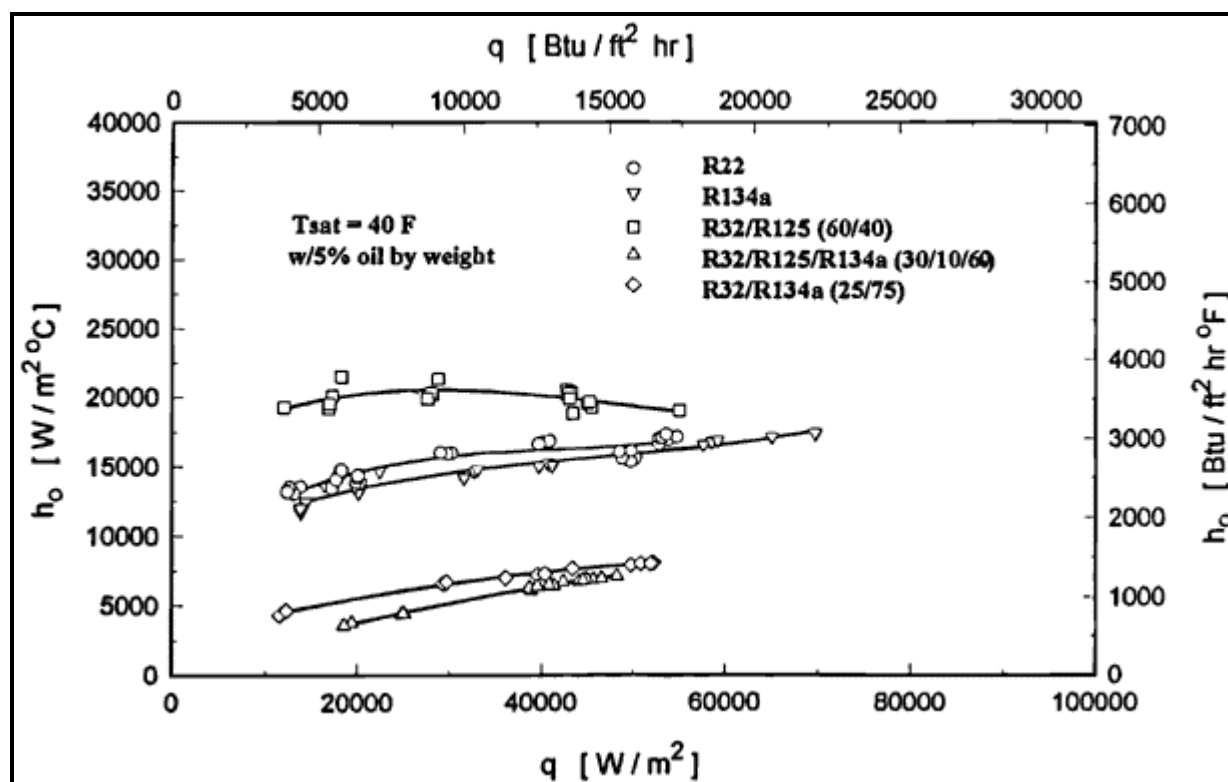
Figure 16.12. Chen and Tuzla (1996) boiling data for Turbo-BII tube with 5 % oil [their fig. 16 reprinted by permission of EPRI].

Gorenflo and Koster (1997) compared the oil effects measured in four independent refrigerant-oil studies, depicted in Figure 16.13. In the top graph, Moller, Spindler and Hahne (1993) found that oil increased boiling performance in R-134a by up to 20% depending on heat flux and oil mass fraction. In the lower graphs from tests by Marto and coworkers, the oil either increased or decreased performance depending on the type of tube. For a tube bundle, the effect of oil was to enhance performance for plain and low finned tubes more than in the comparable single-tube tests while for the enhanced tubes (High Flux and Turbo-B) single-tube and bundle results were similar.
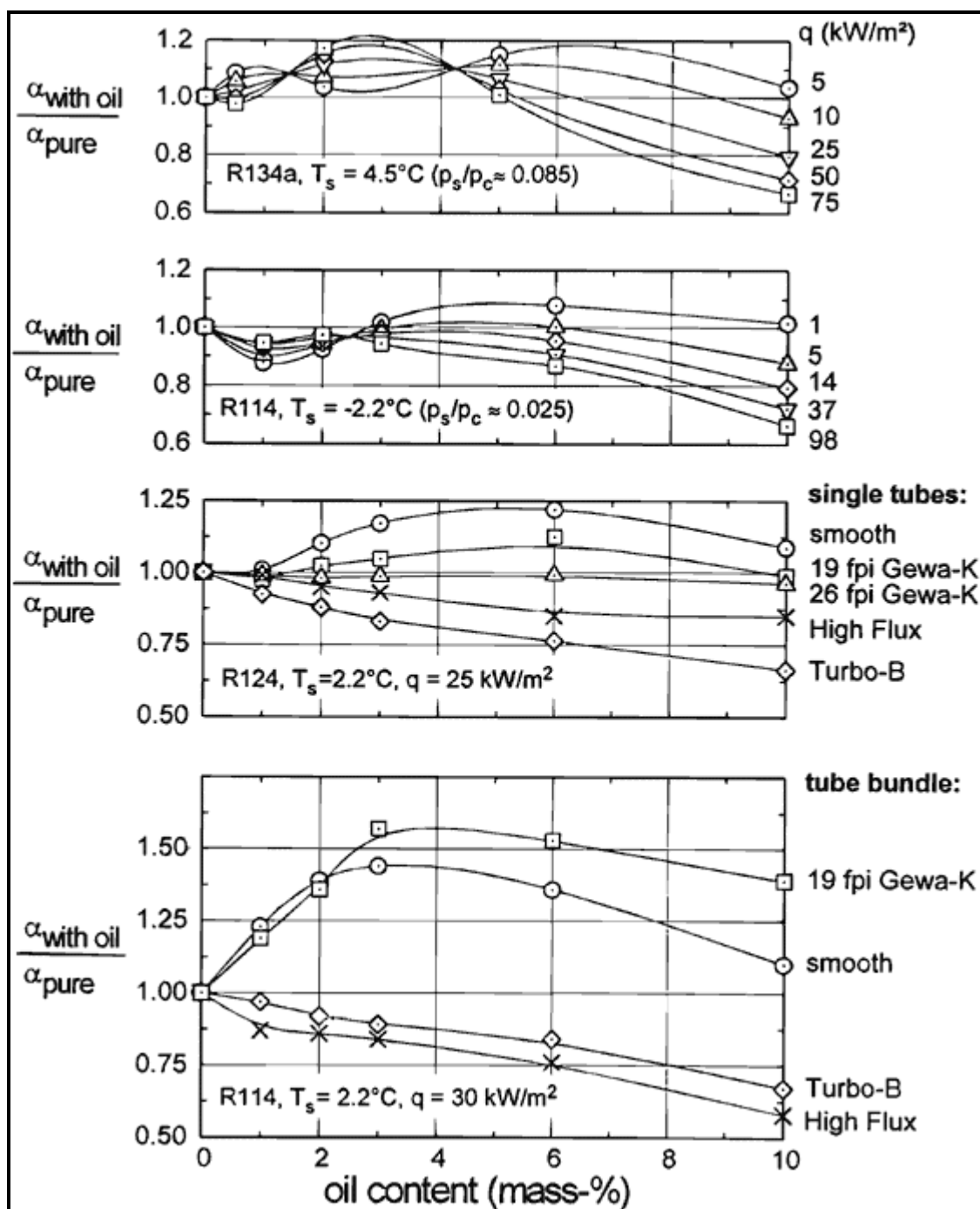
Figure 16.13. Relative influence of oil on boiling from Gorenflo and Koster (1997). *Top*: R-134a/polyester oil on a stainless steel tube from Moller, Spindler and Hahne (1993); *Second*: R-114/oil on a copper tube from Warriarachchi, Marto and Reilly (1986); *Third*: R-124/alkylbenzene oil on single tubes from Memory, Bertsch and Marto (1993); *Bottom*: R-114/mineral oil on small bundles from Memory, Akcasayar, Eraydin and Marto (1995) [their fig. 4 reprinted by permission of D. Gorenflo].

For boiling in mixtures, it is common to describe the degradation in heat transfer caused by mass transfer effects with an expression proposed by Stephan and Körner (1969) that can be adapted for refrigerant-oil mixtures as:

$$\frac{\alpha}{\alpha_I} = \frac{(T_w - T_{bub})}{(T_w - T_{bub}) + dT_{bub}}$$

[16.9.3]

where the ideal boiling heat transfer coefficient α_I is for an equivalent pure fluid with physical properties identical to the mixture and the mixture heat transfer coefficient α is defined as in expression [16.9.1]. The rise in the local bubble point temperature at the interface of a growing bubble, dT_{bub}, due to the congregation of oil (which gets transported to the interface but cannot evaporate), decreases the ratio of α/α_I, such that α of the mixture is less than or equal to α_I. The larger degradation in α/α_I typically observed for enhanced boiling tubes compared to plain and low finned tubes is expected, as pointed out in Thome (1987), since $(T_w\text{-}T_{bub})$ for an enhanced tube is much less that of a plain or low finned tube. This increases the relative importance of dT_{bub} in [16.9.3]. For instance, if $dT_{bub} = 0.5°C$ (0.9°F) and $(T_w\text{-}T_{bub})$ = 10°C (18°F) for a plain tube, the relative degradation in the mixture coefficient is only 4.8%; for an enhanced boiling tube with $(T_w\text{-}T_{bub})$ = 1.0°C (1.8°F), the degradation is 33.3%!

In addition to the mass diffusion effect noted above, two other aspects play a role in boiling of refrigerant-oil mixtures: (i) a change in physical properties and (ii) a modification of the underlying heat transfer mechanisms. Both of them may be important. First of all, lubricating oils have critical pressures much lower than refrigerants so adding oil to a refrigerant substantially decreases the critical pressure of the mixture. Other properties affected significantly by the oil are liquid viscosity, surface tension and contact angle. Secondly, oil may promote foaming at the boiling surface that in turn influences bubble growth and departure. This foaming action may modify the underlying heat transfer mechanisms for plain and low finned tubes and the thin film evaporation process occurring inside reentrant channels of an enhanced tube. More fundamental study of these mechanisms is required in order to understand their consequences before generalized predictive methods can be developed.

In summary, oil effects on the nucleate pool boiling heat transfer coefficient are numerous and quite complex. For plain tubes, oil can either cause a slow monotonic decrease in α with increasing oil mass fraction relative to the pure refrigerant coefficient or first increase and then decrease with increasing oil to create a local maximum in α. No concrete explanation with conclusive experimental proof is available explain why oil sometimes increases plain tube boiling performance by 30-40% but in other cases has no effect or monotonically degrades performance. The same comments are applicable to low finned tubes. For enhanced boiling tubes, even a small amount of oil usually decreases the boiling performance in nucleate pool boiling tests. A better understanding of the influence of oil is required for enhanced tubes. Also, test results tend to depict a complicated influence of heat flux on the oil effect for all types of tubes. Not surprisingly, there are not any prediction methods for the influence of oil on nucleate pool boiling performances of plain, low fin and enhanced tubes to be recommended at this time.

16.9 Bundle Boiling of Refrigerant-Oil Mixtures

For evaporation of a refrigerant-oil mixture on a horizontal tube bundle, the local oil mass fraction increases from bottom to top in the bundle as the local vapor quality rises since no oil evaporates into the vapor-phase and consequently it builds up in the liquid-phase. Hence, the local oil mass fraction may reach levels, under exceptional conditions, as high as 20% to 50% near the top of the tube bundle. Therefore, using single-tube nucleate pool boiling test data for say 1% oil typical of industrial practice to size an enhanced tube flooded-evaporator does not realistically account for the oil's influence on local performance near the top of the bundle. Unfortunately, single-tube boiling tests rarely go beyond oil mass fractions of 5%, although some tests up to 10% have been published. Hence, caution is required in using single-tube tests with 1% oil for thermal design. The best approach would be to use test data for a range

of oil mass fractions and apply these to the mass fraction profile that occurs up through the bundle as the refrigerant evaporates out of the mixture.

Not many boiling tests on tube bundles that have correctly reduced their data using the Thermodynamic Approach described in Chapter 15 are available. Results reduced using the saturation temperatures of the pure refrigerant are not thermodynamically correct and do not abide by the definition of the heat transfer coefficient for a mixture given by expression [16.9.1]. There is, however, the excellent study by Memory, Akacasayer, Eraydin and Marto (1995) on R-114/oil mixtures on plain and enhanced tube bundles.

The Thermodynamic Approach presented in Chapter 15 for prediction of bubble point temperatures of refrigerant-oil mixtures and preparation of temperature-enthalpy-vapor quality (T-h-x) curves for thermal design should also be used to determine the LMTD, either global or incremental, for a flooded-evaporator with oil in the evaporating refrigerant. Current design practice in the refrigeration industry ignores these aspects of oil on flooded-evaporator design and hence do not account for oil effects in a proper manner.

16.10 Comments of Practical Importance

From an overall perspective, it is particularly important to think of oil in a refrigerant charge not as an *impurity* but as a *component in a binary mixture*. Hence, the theory and methodologies available for dealing with heat transfer and flow of mixtures can be brought to focus on the problem.

Running fundamental tests, coil and bundle tests and equipment validation tests with disregard to the quantity of oil still remains common practice; however, it is only by accounting for the oil as a binary component in a mixture that we can arrive at a better understanding of this complex problem. Disregarding the mass fraction of oil circulating in the flow loop and not applying an enthalpy curve of the mixture to reduce test data on an evaporator is not correct. Thus, it is very easy to arrive at results that do not respect the correct energy balance and consequently to arrive at wrong conclusions.

As noted in Chapter 15, it is clear that the oil can have a significant effect on the temperature profile of the evaporating fluid. A local oil mass fraction of 0.50 at high vapor qualities, which means that one-half of the local liquid is oil, is within the normal operating range of many units. For example, a circulating mass fraction of 0.02 of oil at a local vapor quality of 0.96 achieves this local mass fraction and hence results in a rise in the local boiling point temperature by 2.3°C (4.1°F) for R-22. Similarly, an oil mass fraction of 0.01 at a vapor quality of 0.98 produces the same result. Hence, it is easy to mistake a temperature measurement at the exit of an evaporator as having achieved 2.3°C (4.1°F) of superheat when in fact not all the refrigerant has been evaporated and naturally none of the oil. What is actually being measured is the local bubble point temperature. Thus, the wrong conclusions about performance will be reached unless the enthalpy approach is applied. Since obtaining test results is expensive, it makes sense to implement the enthalpy approach to get better, more accurate information from such tests.

Example Calculation: Assume that 3% oil (mass) is circulating in a refrigeration system with a direct-expansion evaporator. The viscosity of the pure refrigerant is 0.4 cp while that of the oil is 400 cp at the operating temperature of the evaporator. At a local vapor quality of 0.95, what will be the local reduction in the boiling heat transfer coefficient?

Solution: The local oil mass fraction w for $w_{inlet} = 0.03$ and x = 0.95 is obtained using [16.4.2]:

$$w = w_{inlet}/(1-x) = 0.03/(1-0.95) = 0.60$$

The ratio of $\alpha_{\text{ref-oil}}/\alpha_{\text{ref}}$ for this situation is obtained with [16.4.3]:

$$\alpha_{\text{ref-oil}}/\alpha_{\text{ref}} = \left[\mu_{\text{ref}}/\mu_{\text{oil}}\right]^{0.26w} = \left[0.4/400\right]^{0.26(0.60)} = 0.340$$

Hence, the boiling heat transfer coefficient with oil will only be 0.34 that of the pure refrigerant without oil or a reduction of 66%.