

A semi-theoretical model for predicting refrigerant/lubricant mixture pool boiling heat transfer

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Received 20 February 2001; received in revised form 20 June 2002; accepted 15 September 2002

Abstract

This paper outlines the framework of a semi-theoretical model for predicting the pool boiling heat transfer of refrigerant/lubricant mixtures on a roughened, horizontal, flat pool-boiling surface. The predictive model is based on the mechanisms involved in the formation of the lubricant excess layer that exists on the heat transfer surface. The lubricant accumulates on the surface in excess of the bulk concentration via preferential evaporation of the refrigerant from the bulk refrigerant/lubricant mixture. As a result, excess lubricant resides in a thin layer on the surface and influences the boiling performance, giving either an enhancement or degradation in heat transfer. A dimensionless excess layer parameter and a thermal boundary layer constant were derived and fitted to data in an attempt to generalize the model to other refrigerant/lubricant mixtures. The model inputs include transport and thermodynamic refrigerant properties and the lubricant composition, viscosity, and critical solution temperature with the refrigerant. The model predicts the boiling heat transfer coefficient of three different mixtures of R123 and lubricant to within $\pm 10\%$. Comparisons of heat transfer predictions to measurements for 13 different refrigerant/lubricant mixtures were made, including two different refrigerants and three different lubricants.

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Keywords: Heat transfer; Boiling; Refrigerant; Lubricant; Mixture; Heat transfer coefficient; Modelling

Un modèle semi-théorique de calcul des coefficients de transfert de chaleur en cas d'ébullition de mélange frigorigène/lubrifiant

Mots clés : Transfert de chaleur ; Ébullition ; Frigorigène ; Lubrifiant ; Mélange ; Coefficient de transfert de chaleur ; Modélisation

1. Introduction

The addition of lubricant to refrigerant can significantly alter the boiling performance due to lubricant accumulation at the heat transfer surface. Stephan [1]

was one of the first researchers to note that a lubricant-rich layer exists near the tube wall. The excess concentration (excess surface density) arises from the low vapor pressure of the lubricant relative to the refrigerant. The lubricant can be locally drawn out of solution as a consequence of refrigerant evaporation at the heat transfer surface. The refrigerant/lubricant liquid mixture travels to the heated wall, and the refrigerant preferentially evaporates from the surface leaving behind a

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Nomenclature*Symbols*

h_{fg}	latent heat of vaporization, kJ/kg
h_m	heat transfer coefficient of refrigerant/lubricant mixture, $W/m^2 K$
k	thermal conductivity, $W/m K$
l_e	thickness of excess layer, m
l_a	thickness of adiabatic/van der Waals excess layer, m
M	mass, kg
q''	average wall heat flux, W/m^2
r_b	bubble departure radius, m
T	temperature, K
T_c	refrigerant/lubricant critical solution temperature (lower limit), K
T_c	temperature at excess layer/bulk fluid interface, K
T_w	temperature at roughened surface, K
x	mass fraction of lubricant

Greek symbols

Γ	excess surface density, kg/m^2
ΔT_s	wall superheat: $T_w - T_s$, K
ΔT_{ie}	temperature drop across excess layer, K
ζ	fraction of excess layer removed per bubble
θ	dimensionless thermal boundary layer temperature profile
λ	thermal boundary constant
ρ	mass density of liquid, kg/m^3
σ	liquid-vapor surface tension, kg/s^2
ν	viscosity, m^2/s

Subscripts

b	bulk
L	lubricant
m	measured, mixture
p	pure refrigerant
r	refrigerant liquid
rv	refrigerant vapor
s	saturated state of mixture

Superscripts

–	average
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liquid phase enriched in lubricant. A balance between deposition and removal of the lubricant establishes the thickness of the excess lubricant at the surface. It is hypothesized that the lubricant excess layer controls the bubble size, the site density and, in turn, the magnitude of the heat transfer.

Kedzierski [2] developed a fluorescence measurement technique to verify the existence of the lubricant excess layer at the wall during pool boiling. The fluorescent properties of the lubricant were used to measure the mass of lubricant on a pool-boiling surface. The study indicated that the excess layer was nearly pure lubricant with a thickness ranging from 0.04 mm to 0.06 mm depending on the heat flux. The fluorescence measurement technique was later applied to two more mixtures of R123 and a $63.8 \mu m^2/s$ (63.8 cSt at 313 K) naphthenic mineral oil, York-CTM [3]. The excess surface density (Γ) measurements were used as a basis for a pool-boiling model present in Kedzierski [3]. The present study expands the physics and applicability of the previous model by including a dimensionless excess layer parameter and a fitted thermal boundary layer constant which is determined by lubricant properties.

Prediction methods for refrigerant/lubricant pool boiling typically rely on bulk mixture properties in parameters common to pure component models to capture the lubricant property effects. For example, Chongrungreong and Sauer [4] based their correlation on selected parameters from existing single component pool boiling correlations and used mixture properties where appropriate and also included the use of the lubricant volume fraction. Jensen and Jackman [5] included the effects of lubricant concentration more directly with the calculation of an effective oil concentration; however, the effect of lubricant viscosity was indirectly modeled via bulk properties. Mohrlök [6] modified the leading coefficient of the Chongrungreong and Sauer [4] correlation to better reproduce the effect of concentration. Although Mohrlök et al. [6] discuss the importance of miscibility, none of the above prediction methods attempt to model its effect. Consequently, the purpose of this paper is to build a model for refrigerant/lubricant pool boiling that captures the effects of lubricant properties (viscosity, miscibility, and concentration) more directly than through the use of mixture properties in traditionally pure component parameters.

2. Refrigerant/lubricant model

2.1. Background

The pool-boiling model presented here is valid only for mixtures of refrigerants and lubricants because both

¹ Certain trade names and company products are mentioned in the text or identified in an illustration in order to adequately specify the experimental procedure and equipment used. In no case does such an identification imply recommendation or endorsement by the National Institute of Standards and Technology, nor does it imply that the products are necessarily the best available for the purpose.

the underlying physics and the supporting data for the model rely on the existence of a lubricant excess layer. Neither a lubricant excess layer nor lubricant mass continuity exists nor make sense for a refrigerant without lubricant. Considering that the key elements of the present model are the lubricant excess layer and a mass balance on the lubricant, the model cannot be expected to be valid for pure refrigerants.

The excess surface density (Γ) represents the mass of lubricant per unit surface area within the layer (l_e) at the wall that is in excess of the lubricant mass that would have been within the layer l_e had the excess layer not existed. There is strong evidence to suggest that the excess layer is nearly pure lubricant [2]. For this situation, Γ is defined as:

$$\Gamma = l_e(\rho_l - \rho_b x_b) \tag{1}$$

where l_e is the thickness of the excess layer, x_b is the bulk lubricant mass fraction, and ρ_b and ρ_l are the densities of the refrigerant liquid mixture at the bulk composition and the liquid lubricant, respectively. Note that for a dilute solution of lubricant, Γ may be thought of as simply the mass of lubricant per unit surface area.

2.2. Existing model

Fig. 1 shows a schematic of three bubbles on a heated wall in a pool of liquid with corresponding thermal boundary layer temperature profiles and lubricant excess layer thickness. Each bubble is shown to reside in a different refrigerant/lubricant liquid mixture with a wall heat flux of approximately 39 kW/m². As shown in the figure, an increase in the bulk lubricant mass fraction causes the average departure bubble diameter to become smaller.

The average departure bubble diameters for the three refrigerant/lubricant mixtures shown in Fig. 1 were calculated with excess surface density measurements as input to a model presented in Kedzierski [3] and described here. The excess surface density, and thus, the lubricant excess layer were measured in the same study with a spectrofluorometer that was specially adapted for use with a bifurcated optical bundle so that fluorescence measurements could be made perpendicular to the horizontal-flat-roughened-copper pool boiling heat transfer surface. The mixture boiling heat transfer measurements were simultaneously taken with the fluorescence measurements. The fluorescence measurements provided the means for the direct measure of lubricant mass on the boiling surface, i.e., Γ .

The hypothesis for the heat transfer model mechanism is that nearly pure lubricant resides on the heat transfer surface within the thickness l_e . The lubricant excess layer is formed on the wall by preferential evaporation of the refrigerant. The model assumes that all of the lubricant carried to the wall by the bulk liquid/lubricant mixture is deposited on the wall while the entire refrigerant leaves the wall as refrigerant vapor. For a spherical bubble of radius r_b , the mass of refrigerant vapor (M_{rv}) leaving the wall in a single bubble is:

$$M_{rv} = \frac{4}{3} \pi r_b^3 \rho_{rv} \tag{2}$$

where ρ_{rv} is the density of the refrigerant vapor.

The mass lubricant (M_l) deposited on the surface due to the evaporation of the refrigerant for a single bubble can be calculated from Eq. (2) and the definition of the lubricant mass fraction ($x_b = \frac{M_l}{M_r + M_l}$) as:

$$M_l = \frac{4}{3} \pi r_b^3 \rho_{rv} \frac{x_b}{(1 - x_b)} \tag{3}$$

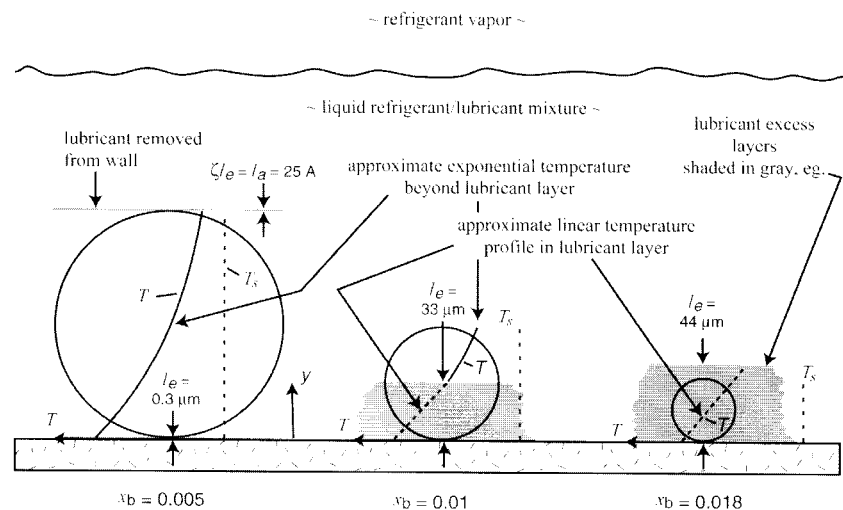


Fig. 1. Schematic of the average departure bubble for three R123/lubricant mixtures with corresponding excess layers [3].

The lubricant removal mechanism is assumed to be dominated by a mass transfer or a lifting of lubricant from the wall by the bubble. This mechanism differs from that proposed by Jensen and Jackman [5] and Mitrovic [7] where the lubricant excess layer surrounds the entire bubble and was formed by preferential evaporation of the refrigerant at the bubble liquid–vapor interface. The present model presumes that diffusion is of secondary importance because of the question of whether or not there is sufficient time for diffusion to establish an excess layer at the bubble liquid–vapor interface. Rather, the bubble lifts the existing lubricant excess layer from the wall by boiling through the lubricant. The thickness of the lubricant that is removed is limited by that which Van der Waals forces may hold to the bubble. Thus, the thickness is independent of the bulk lubricant mass fraction. Typically the thickness of an adiabatic or a Van der Waals excess layer (l_a) is approximately 2 monolayers thick [8], which is approximately 25 Å for lubricants [9]. This value is essentially invariant with temperature and pressure [10].

As illustrated in Fig. 1 for the $x_b = 0.005$ bubble, the removed lubricant resides in the vicinity of the top of the bubble in the thickness of an adiabatic excess layer (l_a) and represents a fraction (ζ) of the excess layer thickness (l_c). The mass of the lubricant “cap” on the bubble (M_L) may be approximated as a disk with a radius equal to the departure bubble radius (r_b) and thickness equal to l_a :

$$M_L = \pi r_b^2 \rho_L l_a = \pi r_b^2 \rho_L \zeta l_c \quad (4)$$

where ρ_L is the density of the liquid lubricant.

Writing a mass balance between lubricant deposition and removal [setting Eqs. (3) and (4) equal] and rearranging to solve for the refrigerant bubble radius yields:

$$r_b = \frac{0.75 \zeta l_c \rho_L (1 - x_b)}{x_b \rho_{rv}} = \frac{18.75 \text{ Å} \rho_L (1 - x_b)}{x_b \rho_{rv}} \quad (5)$$

where 25 Å was substituted for ζl_c per the preceding discussion and note that Eq. (5) is valid for $x_b > 0$.

In addition to lubricant continuity, the temperature profile of the thermal boundary layer (see Fig. 1) is a key element in the boiling model. The thermal boundary layer was approximated with two different functions similar to the approach for modeling the turbulent single-phase flow thermal boundary layer [11], which represents the viscous sublayer, the buffer layer, and the inertial sublayer with three different equations. By analogy to the turbulent flow boundary layer a linear temperature profile is assumed for the excess layer, while an exponential temperature profile approximates the temperature field beyond the excess layer. A linear temperature profile at the wall is appropriate given the relatively thin lubricant excess layer. The dimensionless

form of the temperature profile valid beyond the excess layer (θ) is given as:

$$\theta = \frac{T - T_s}{T_w - T_s} = e^{-\lambda y/r_b} \quad (6)$$

where T is the local temperature of the fluid, y is a coordinate direction measured perpendicularly from the wall, and λ is a constant that is obtained for each mixture from a fit of the measured pool boiling curve as described in the following.

Fourier’s law gives the heat flux normal to the heated surface (q'') for a linear temperature gradient valid in the excess layer as:

$$q'' = \frac{k_L \Delta T_{lc}}{l_c} \quad (7)$$

where k_L is the thermal conductivity of the lubricant, and ΔT_{lc} is the temperature drop across the lubricant excess layer ($T_w - T_c$).

Eq. (6) is used to calculate the temperature at the excess layer/bulk fluid interface (T_c). The T_c is, in turn, used to solve for ΔT_{lc} . The expression for ΔT_{lc} is substituted into Eq. (7) and rearranged to obtain the wall superheat (ΔT_s) valid for $x_b > 0$:

$$\begin{aligned} \Delta T_s = T_w - T_s &= \frac{q'' l_c}{k_L (1 - e^{-\lambda l_c/r_b})} \\ &= \frac{q'' \Gamma}{k_L (1 - e^{-\lambda l_c/r_b}) (\rho_L - \rho_b x_b)} \end{aligned} \quad (8)$$

where Eq. (1) was used to express ΔT_s in terms of Γ .

Fig. 2 shows measured excess surface densities for each of the R123/York-CTM mixtures as a function of heat flux taken from Kedzierski [3].² The Γ measurements along with the ΔT_s and q'' data were used as input to Eq. (8) and was rearranged to solve for the constant λ which was essentially invariant for a fixed bulk lubricant mass fraction. The value of the fitted constant λ was 1.74, 1.09, and 0.57 for the 0.005 mass fraction lubricant mixture [(99.5/0.5) mixture], the 0.010 mass fraction lubricant mixture [(99/1.0) mixture], and the 0.018 mass fraction lubricant mixture [(98.2/1.8) mixture], respectively. The mean absolute difference between the post-predicted and measured mean wall superheat was 0.46 K, 0.07 K, and 0.22 K for the (99.5/0.5), (99/1.0), and (98.2/1.8) mixtures, respectively. This comparison was made for the heat flux range for which excess surface density measurements were available.

2.3. Modified model

Two modifications were made to the above-outlined model in an effort to predict or generalize λ based on

² All confidence intervals are on the mean of the regression and not the individual measurements.

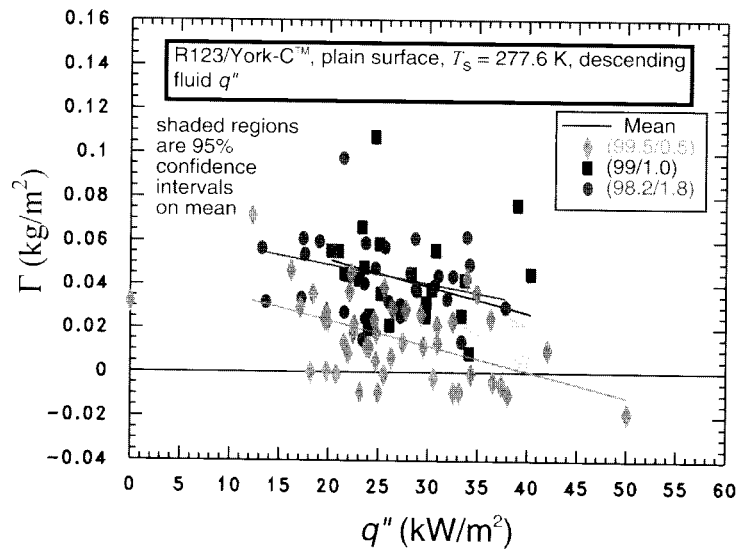


Fig. 2. Lubricant excess surface density for three R123/York-C™ mixtures as a function of heat flux [3].

refrigerant and lubricant properties such as viscosity, miscibility, and composition. The first modification was to collapse the excess surface density measurements shown in Fig. 2 into a single line. To facilitate this, the following expression for Γ was derived:

$$\Gamma = \frac{8(\rho_L - \rho_b x_b) x_b T_s \sigma}{3\zeta(1 - x_b) \rho_L h_{fg} \Delta T_s} \quad (9)$$

where σ and h_{fg} are the liquid–vapor surface tension and the latent heat of the refrigerant, respectively.

Eq. (9) was obtained by eliminating the bubble radius between Eq. (5) and the Clausius–Clapeyron–Laplace expression [12] for the required superheat for a heterogeneous bubble in a uniform temperature field and solving for l_e . Eq. (1) was used to express l_e in terms of Γ . Strictly speaking, Eq. (9) is valid for only a uniform temperature field. However, the sole use of Eq. (9) is to determine a regression parameter for Γ . Following this reasoning, it is hypothesized that the parameter group of Eq. (9), excluding the $8/3\zeta$, sufficiently represents the functional form of Γ for the purpose of regressing Γ against data.

Fig. 3 shows a statistical regression of the same Γ measurements given in Fig. 2 against only the parameters given in the right side of Eq. (9). The dashed lines are the 95% confidence intervals for the mean of a linear fit of the data, which are approximately ± 0.01 kg/m² about the mean. The linear fit gave a non-zero intercept. A second solid line, which was forced to have a zero-intercept, is within ± 0.01 kg/m² of the linear fit and exhibits a similar uncertainty. By using the second fit, a non-dimensional excess surface density group for refrigerant/lubricant pool boiling can be expressed as the following constant:

$$\frac{x_b T_s \sigma}{l_e (1 - x_b) \rho_L h_{fg} \Delta T_s} = \frac{(\rho_L - \rho_b x_b) x_b T_s \sigma}{(1 - x_b) \rho_L h_{fg} \Delta T_s \Gamma} = 5.9 \times 10^{-7} \quad (10)$$

where the constant 5.9×10^{-7} is the reciprocal of the slope of the linear fit of Γ versus the abscissa of Fig. 3.

Solving for l_e yields:

$$l_e = \frac{\Gamma}{\rho_L - \rho_b x_b} = \frac{x_b T_s \sigma}{5.9 \times 10^{-7} (1 - x_b) \rho_L h_{fg} \Delta T_s} \quad (11)$$

Note that the large scatter of the excess surface density data does not transfer to the calculation of l_e —and in turn the heat transfer predictions—due to the use of the fitted non-dimensional constant in the predictions.

The second modification to the original lubricant pool-boiling model was to introduce the influence of lubricant properties on the heat transfer. Mohriok et al. [13] and Kedzierski [14] have illustrated the importance of refrigerant/lubricant miscibility and lubricant viscosity on pool boiling heat transfer. Consequently, a correlation of the effects of lubricant mass fraction, viscosity (ν_L) and the lower Critical Solution Temperature (CST)³ on R134a pool boiling on a Turbo-BII™-HP surface was integrated in the modified model. The

³ Throughout this manuscript reference to the CST implies the lower CST limit. In addition, the chemistry and freezing points of lubricants and refrigerants are different enough for a CST to always exist for a refrigerant/lubricant mixture [15]. Also, the value of the CST does not depend on whether the mixture is miscible or non-miscible at the operating temperature. Consequently, the proposed model is valid for mixtures that are “miscible” and it is valid for mixtures that are “non-miscible.”

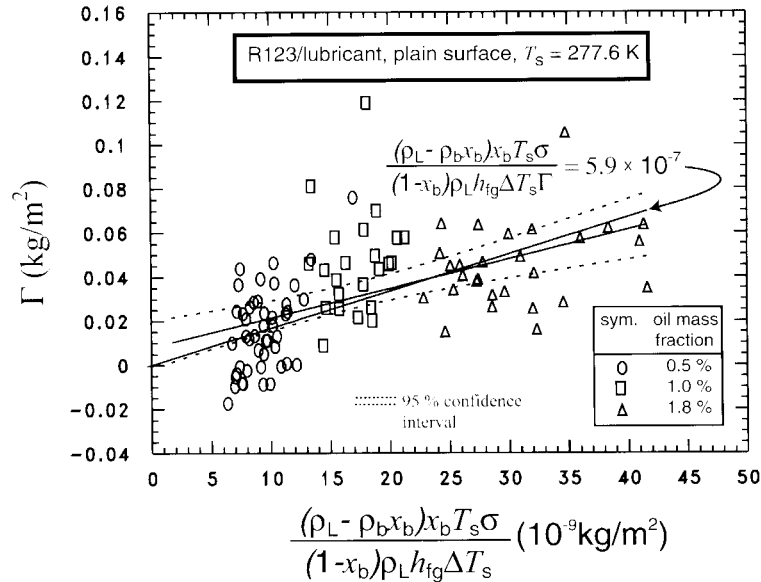


Fig. 3. Lubricant excess surface density for three R123/lubricant mixtures as a function of non-dimensional excess surface density.

following correlation gives the ratio of the refrigerant/lubricant heat flux (q''_m) to that of the pure refrigerant (q''_p) for the same superheat [14]:

$$\frac{q''_m}{q''_p} = 1.25 - x_b \left\{ 91.9 - \left(\frac{\nu_L - \nu_r}{\nu_r} \right) [0.529 - 1.92 \left(\frac{T_s - T_c}{T_s} \right)] - 211 \left(\frac{T_s - T_c}{T_s} \right) \right\} \quad (12)$$

where $(\nu_L - \nu_r)/\nu_r$ is the difference between the liquid viscosity of the lubricant and that of the refrigerant normalized by the refrigerant viscosity all evaluated at 313 K, and $(T_s - T_c)/T_s$ is the difference between the saturated fluid mixture temperature and the lower critical solution temperature (T_c) of the refrigerant/lubricant mixture normalized by the mixture saturation temperature (T_s). The heat flux ratio was fitted to within ± 0.1 for 95% confidence for q''_p between approximately 10 kW/m² and 80 kW/m².

The data used to generate Eq. (12) were re-regressed so that the heat flux range coincided with that for which the excess surface density measurements were made for the R123/lubricant data. The modified correlation is valid for q''_p between approximately 25 kW/m² and 45 kW/m²:

$$\frac{q''_m}{q''_p} = 1.27 - x_b \left\{ 99.1 - \left(\frac{\nu_L - \nu_r}{\nu_r} \right) [0.578 - 2.09 \left(\frac{T_s - T_c}{T_s} \right)] - 226 \left(\frac{T_s - T_c}{T_s} \right) \right\} \quad (13)$$

valid for $\{x_b > 0\}$ and $\left\{ 99.1 - \left(\frac{\nu_L - \nu_r}{\nu_r} \right) [0.578 - 2.09 \left(\frac{T_s - T_c}{T_s} \right)] - 226 \left(\frac{T_s - T_c}{T_s} \right) \right\} \geq 0$

Fig. 4 plots the measured heat flux ratio versus that predicted by Eq. (13) along with 95% confidence intervals for the fit. The original R134a heat transfer data agree to within ± 0.1 , which is the same agreement that was obtained with Eq. (12) and this data. Confirming the similarity between Eqs. (12) and (13) is the agreement between the prediction of the R123 data using Eqs. (12) and (13). Open circles represent R123/York-C heat transfer measurements predicted by Eq. (12), while closed circles represent the data and predictions for Eq. (13).

Four different sets of predictions are shown for the three R123/ York-C mixtures. Predictions made with Eqs. (12) and (13) with a CST of 235 K [$(T_s - T_c)/T_s = 0.15$] for the R123/lubricant system are circled in Fig. 4 and lie outside of the confidence intervals for the correlation. Eqs. (12) and (13) predictions made with a CST of 214 K [$(T_s - T_c)/T_s = 0.23$] fall within in the confidence intervals because the CST was determined from an optimization of Eq. (13). Two different CSTs were used because the CST for R123/lubricant was not available. However, Kaimai [16] presents the CST for 10 different paraffinic mineral oil/R123 systems with viscosities (at 313 K) between 56 $\mu\text{m}^2/\text{s}$ and 68 $\mu\text{m}^2/\text{s}$. The average CST was 235 K with maximum of 3 K variation between mixtures. In addition, ASHRAE [17] provides a lower CST of 235 K for a R123/68 $\mu\text{m}^2/\text{s}$ naphthenic mineral oil system. Consequently, it is highly probable that R123/York-CTM has a CST of 235 K, but this was not confirmed by measurement. The 214 K CST was used in an attempt to compensate for the fact that the use of Eq. (13) to predict the performance of R123/naphthenic-mineral-oil mixture pool boiling on a flat

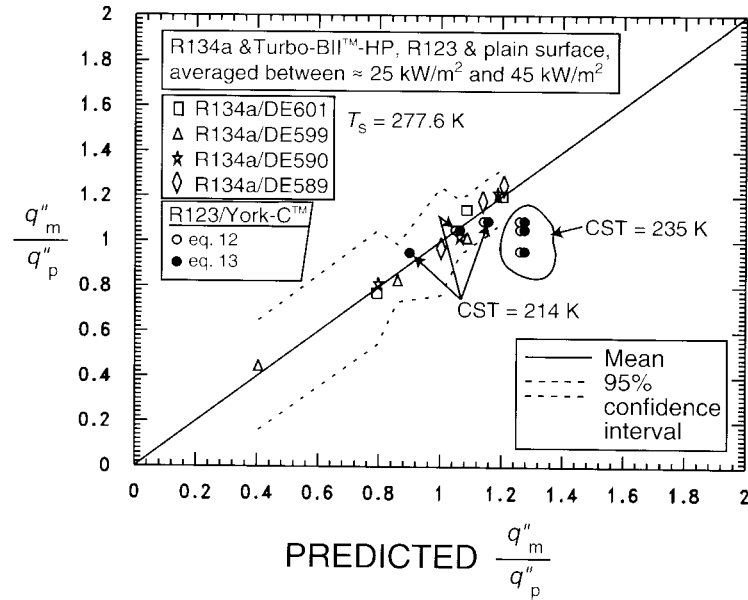


Fig. 4. Comparison of measured heat flux ratio to that predicted by Eq. (13).

surface is questionable given that Eq. (13) was derived from R134a/polyolester lubricant pool boiling on an enhanced surface. Nevertheless, it is hypothesized that Eq. (13) captures the interaction between lubricant properties and composition in a way that produces trends that are universally valid but offset from systems other than R134a/polyolester lubricants and the Turbo-BII™-HP surface. In addition, there is less than a 2% difference in the predicted heat transfer coefficient using the different CSTs for the R123 data set. In this light, a CST of 214 K was used to ensure that Eq. (13) predicts the R123 data and, in turn, demonstrates the concept of the modified model. Future work for improving the model would be to improve the validity of Eq. (13) for the fluid system in question.

The heat transfer data for the three R123/York-C™ mixtures and corresponding excess surface densities which were calculated from Eq. (9) were used to obtain new values for the thermal boundary layer parameter, λ . The value of the fitted constant λ was 1.76, 0.95, and 0.65 for the (99.5/0.5) mixture, the (99/1.0) mixture, and the (98.2/1.8) mixture, respectively. Fig. 5 shows that the product of the bubble radius and the heat flux ratio is nearly linear with respect to λ :

$$\lambda = 0.27 + r_b \frac{q''_m}{q''_p} 10700/m \quad (14)$$

Given that the thermal boundary layer controls the heat transfer, it is logical that Eq. (14) is a function of key heat transfer properties: r_b and the heat flux ratio. Nevertheless, more work is required to confirm Eq. (14) given the scant data that was used to produce it.

At this point the modifications to the model are complete, and Eqs. (8) and (11) can be simultaneously solved for the refrigerant/lubricant mixture heat transfer coefficient (h_m) valid for $x_b > 0$:

$$h_m = \frac{q''_m}{T_w - T_s} = \frac{5.9 \times 10^{-7} (1 - x_b) \rho_l h_{fg} \Delta T_s k_l (1 - e^{-\lambda k / r_b})}{x_b T_s \sigma} \quad (15)$$

where Eqs. (5), (11), and (14) are used to calculate r_b , l_e , and λ , respectively.

Fig. 6 compares the measured heat transfer coefficients for the three mixtures at three superheats to the model given by Eq. (15). The comparison was made for a fixed saturation temperature of 277.6 K for three different lubricant mass fractions of the R123/York-C™ mixtures. The average absolute difference between the predicted and measured heat transfer coefficient was 2.2%, 1.8%, and 5.3% for the $\Delta T_s = 19$ K, $\Delta T_s = 17$ K, and $\Delta T_s = 14$ K superheats, respectively. Given that the model physics are based on the existence of an excess layer, the model is not valid for the pure refrigerant case.⁴ Thus, rather than plotting the model to $x_b = 0$, a dashed-line was used to extrapolate heat transfer coefficients from the 0.001 mass fraction mixture to the pure R123 data. The dashed line is only the author's

⁴ The pure component heat flux can be calculated by substituting the numerical result of q''_m from Eq. (15) into Eq. (13) (using the same x_b , etc. that was used in Eq. (15) and solving for q''_p .

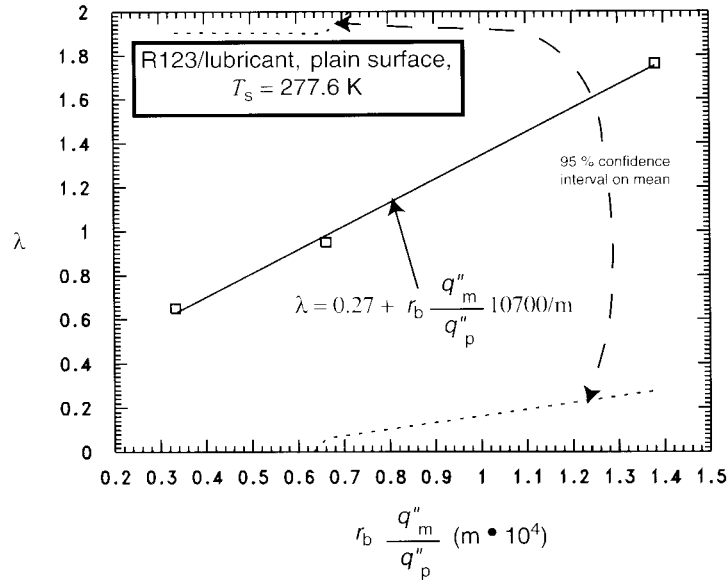


Fig. 5. Regression of thermal boundary layer constant.

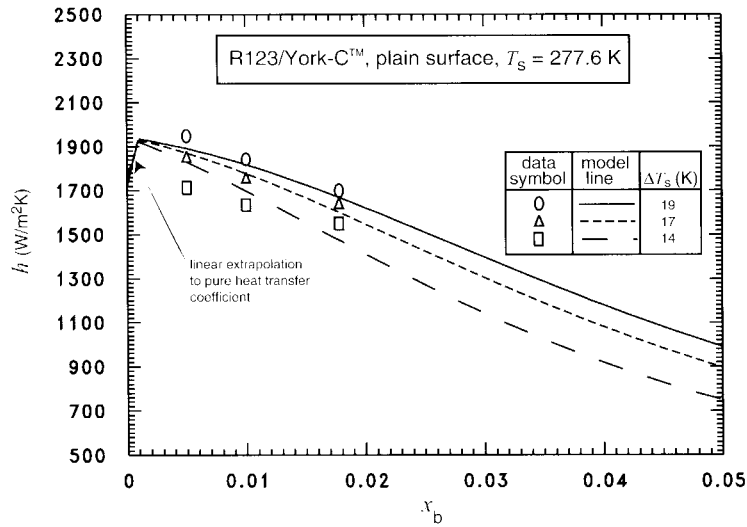


Fig. 6. Comparison of R123/York-CTM pool boiling heat transfer coefficients of model and experiment.

interpretation of what “should” happen near the zero mass fraction. For lubricant mass fractions greater than 0.001, the model predicts a decrease in the heat transfer coefficient. The heat transfer coefficients predictions become much less dependent on the superheat at the 0.001 mass fraction and are approximately 10% greater than the pure R123 heat transfer coefficient measurements. The lubricant excess layer thickness approaches only a few monolayers of thickness as the lubricant mass fraction approaches 0.001. A thin excess layer enhances heat transfer because the mass of lubricant on the wall is sufficient to reduce the liquid solid surface

energy (similar to a surfactant) while not enough to burden the bubbles with lubricant removal from the wall.

Fig. 7 compares the Eq. (15) predictions to the measured R123/lubricant plain tube pool boiling data of Webb and McQuade [18] at $T_s = 277.6$ K. The commercial lubricant (Mobil GA-155TM) was a $30.1 \mu\text{m}^2/\text{s}$ (30.1 cSt at 313 K) naphthenic mineral oil. The CST for the R123/GA-155TM combination was estimated to be the same as that used for the R123/lubricant of the present study because a measured value was not available from the manufacturer. Use of the same CST for the Webb and McQuade [18] mixture is justified because the

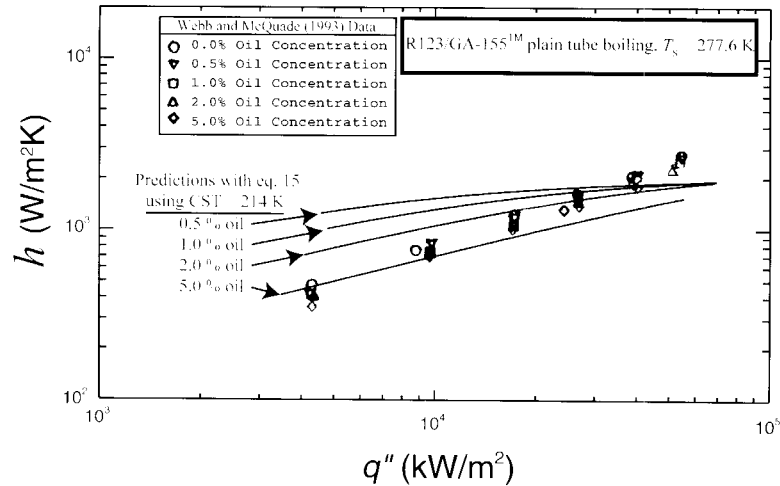


Fig. 7. Comparison of R123/GA-155TM pool boiling heat transfer predictions to measurements.

refrigerants are the same and the lubricants are both naphthenic mineral oil. Fig. 7 illustrates that the smallest overall difference between the predictions of the pool boiling heat transfer coefficient and the Webb and McQuade [18] measurements for the entire heat flux range was achieved for the 5.0% lubricant mass fraction mixture. For example, the heat transfer coefficient for the 5.0% mixture was overpredicted, and underpredicted by approximately 25% for a heat flux approximately 4.2 kW/m², and 52 kW/m², respectively. In addition, the Webb and McQuade (1993) measurements and the predictions for the 5.0% mixture nearly coincide for the 9.8 kW/m² heat flux condition. In contrast, the largest overprediction (150%) was exhibited for the 0.5%

lubricant mass fraction mixture at approximately 9.8 kW/m². Considering all the mixtures, improved agreement between measurements and predictions was apparent for heat fluxes larger than 20 kW/m². Generally, the agreement for all the mixtures is within ±25% for heat fluxes larger than 20 kW/m². Also, it should be noted that greater variation in the heat transfer coefficient with respect to lubricant concentration was exhibited in the Bell et al. [19] data and the Sauer et al [20] data as compared to the Webb and McQuade [18] measurements.

Fig. 8 compares the Eq. (15) predictions to the measured R113/lubricant plain tube pool boiling data of Bell et al. [19] at $T_s = 320.7$ K. The viscosity of the

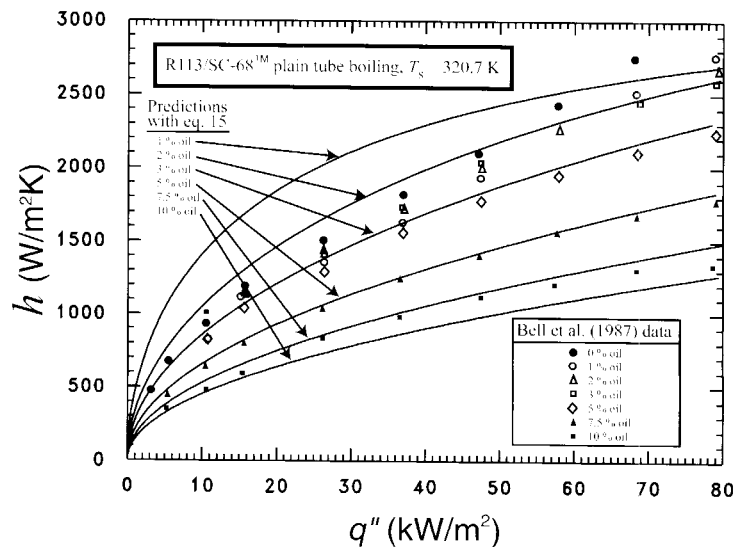


Fig. 8. Comparison of R113/SC-68TM pool boiling heat transfer predictions to measurements.

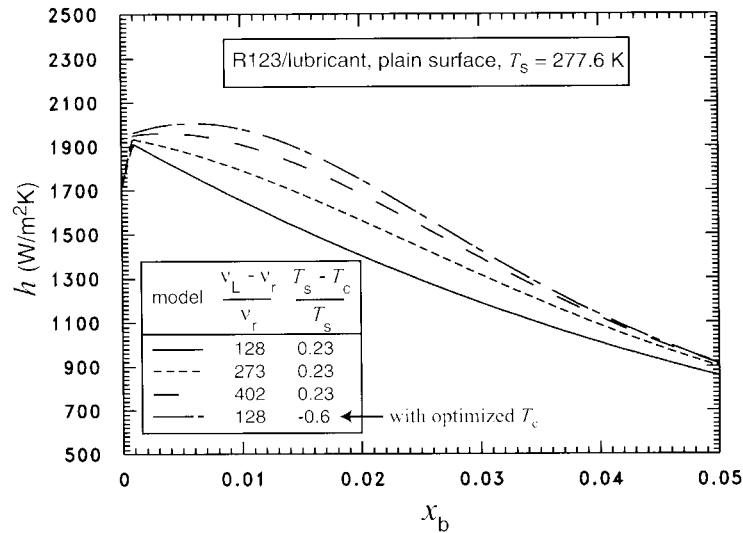


Fig. 9. R123/lubricant pool boiling heat transfer coefficient predictions for a range of lubricant viscosities.

commercial naphthenic mineral oil (Shell Clavus-68TM) was $68 \mu\text{m}^2/\text{s}$ (68 cSt at 313 K). The CST for the R113/SC-68TM combination was estimate to be the average of three CST's given in ASHRAE [17] for naphthenic oils and refrigerants (263 K). The resulting heat transfer coefficient predictions are compared to the Bell et al. [19] data for six different lubricant mass fractions in Fig. 8. The agreement between the measured and the predicted heat transfer coefficient varies between approximately 1% to 2% and some maximum difference for each composition. For example, the approximate maximum deviation of the predicted from the measured heat transfer coefficient is 12%, 20%, 20%, 10%, 10%, and 40% for the 10%, 7.5%, 5%, 3%, 2%, and 1% compositions, respectively. In general, the slope of the predictions and the measurements agree and the relative difference between compositions for the predictions is roughly consistent with the respective relative difference for the measurements. Similar agreement was achieved if a CST of 214 K was used in the prediction.

3. Discussion

Fig. 9 shows R123/lubricant pool boiling heat transfer coefficient predictions using Eq. (15) for three different viscosity ratios, $(v_L - v_r)/v_r = 128, 273,$ and 402 , which correspond to mixtures of R123 and a $32 \mu\text{m}^2/\text{s}$, a $68 \mu\text{m}^2/\text{s}$, and a $100 \mu\text{m}^2/\text{s}$ lubricant, respectively. In general, a larger viscosity gives a larger pool boiling heat transfer coefficient for the fixed $(T_s - T_c)/T_s$ of 0.23 . The largest heat transfer coefficients shown in Fig. 9 for lubricant mass fractions below 0.04 were obtained for $(T_s - T_c)/T_s = -0.6$. For this case, the CST was optimized by setting the quantity within the outermost brackets

that multiplies x_b in Eq. (13) to zero and solving for the CST that satisfies the viscosity ratio. An approximate 25% increase in the heat transfer coefficient was obtained at $x_b = 0.01$ for the R123/ $32 \mu\text{m}^2/\text{s}$ lubricant by decreasing the $(T_s - T_c)/T_s$ (i.e., the lubricant miscibility) from 0.23 to -0.6 . This illustrates that the model predicts that the lubricant viscosity and the lubricant miscibility are influential in determining the magnitude of the heat transfer coefficient.

Fig. 10 shows the lubricant viscosity/CST combination that will give the maximum heat transfer for a given lubricant mass fraction as predicted by Eq. (13). Six different refrigerant/lubricant combinations are illustrated on the plot for R123 and R134a and a $32 \mu\text{m}^2/\text{s}$, a $68 \mu\text{m}^2/\text{s}$, and a $100 \mu\text{m}^2/\text{s}$ lubricant. The optimum heat transfer performance for five of the mixtures presented is obtained for a $(T_s - T_c)/T_s$ of approximately 0.2 , which is in the miscible range. Only the R123 with the $32 \mu\text{m}^2/\text{s}$ lubricant mixture requires an immiscible lubricant to obtain the maximum heat transfer performance. The model also predicts that refrigerant/lubricant combinations with a $(v_L - v_r)/v_r$ that is slightly less than that of the R123/ $32 \mu\text{m}^2/\text{s}$ lubricant require a completely miscible lubricant with a $(T_s - T_c)/T_s$ greater than 0.45 . These results should only be used to guide the readers thought process in exploring new refrigerant/lubricant mixtures considering that Eq. (13) has not been thoroughly validated for R123 and heat transfer surfaces other than the Turbo-BIITM-HP.

In its present state, the modified model should be cautiously used as a tool for guiding research in the direction of choosing lubricant properties for optimizing refrigerant heat transfer for a particular application. Further work is required for the model to fully capture the important influence of other controlling parameters

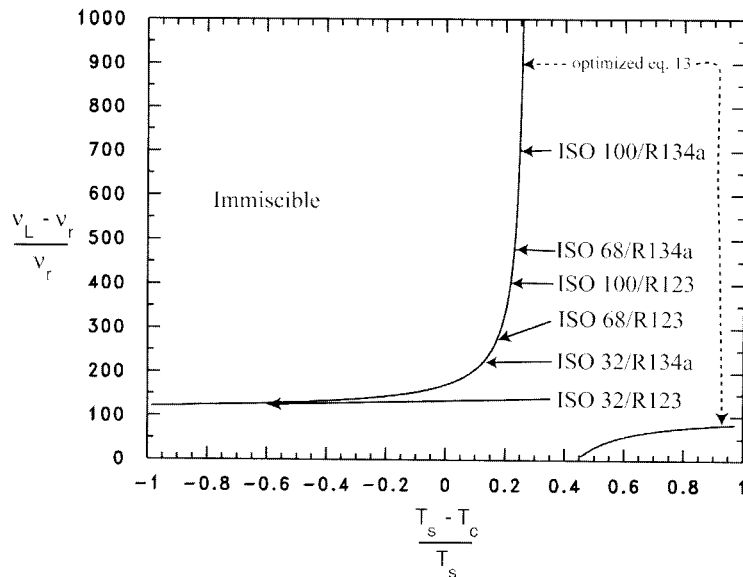


Fig. 10. Optimized CST versus lubricant viscosity ratio for refrigerant/lubricant pool boiling heat transfer.

including refrigerant saturation pressure as illustrated by Morhlok [6]. Future improvements in the model should also be focused on generalizing Eqs. (10) and (13) to other surfaces and fluids. This requires heat transfer and lubricant excess surface density measurements for more refrigerant/lubricant mixtures boiling on various surfaces for various saturation pressures. The form of Eq. (13) should also be improved. Presently, the quantity within the outermost brackets that multiplies x_b must be greater than zero to provide for a decreasing enhancement with respect to increasing lubricant mass fraction, which is consistent with current experience. Eq. (13) also predicts an enhancement ratio that is limited to 1.27 and, in addition, does not correctly predict the pure refrigerant ratio. Given that few, if any, models attempt to correlate lubricant effects on pool boiling based on lubricant properties, the modified model establishes a valuable framework on which to build these capabilities.

4. Conclusions

The importance of the lubricant excess layer on refrigerant/lubricant pool boiling has been illustrated with the development of a semi-theoretical model for predicting the pool boiling heat transfer of refrigerant/lubricant mixtures on a roughened, horizontal flat pool-boiling surface.

The predictive model is based on the mechanisms involved in the formation of the lubricant excess layer that exists on the heat transfer surface. A dimensionless excess layer parameter and a thermal boundary layer constant were fitted to excess surface density and heat

transfer data in an attempt to generalize the model to other refrigerant/lubricant mixtures. The model is fitted to excess surface density and heat transfer measurements. The model relies on three key assumptions: (1) lubricant is lifted from the excess layer as lubricant caps on bubbles, (2) the temperature profile in the thermal boundary layer can be approximated piecewise with a linear function within the excess layer and an exponential function beyond, and (3) the excess layer consists of entirely lubricant.

The model requires transport and thermodynamic refrigerant properties and the lubricant composition, viscosity, and critical solution temperature with the refrigerant as input. The model predicts the boiling heat transfer coefficient of three different mixtures of R123 and lubricant to within $\pm 10\%$. Comparisons of heat transfer predictions to measurements for other refrigerant/lubricant mixtures were also made. All in all, the model was compared against 13 different refrigerant/lubricant mixtures, which included two different refrigerants and three different lubricants. The pool-boiling model should only be used to guide the reader's thought process in exploring new refrigerant/lubricant mixtures. Additional heat transfer and lubricant excess surface density measurements for more refrigerant/lubricant mixtures boiling on various surfaces are required to extend the validity of the model.

Acknowledgements

This work was funded by NIST. Thanks go to the following people for their constructive criticism of the

first draft of the manuscript: Dr. W.V. Payne, and Dr. P.A. Domanski of NIST, and Professor M.S. Kim of Seoul National University. Mr. J. Judge and Mr. K. Starner of York International facilitated the acquisition of the naphthenic mineral oil (York-C™) for use with R123.

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