

IMPROVED THERMAL BOUNDARY LAYER PARAMETER FOR SEMI-THEORETICAL REFRIGERANT/LUBRICANT POOL BOILING MODEL

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ABSTRACT

This paper outlines the framework for an improvement in the thermal boundary layer parameter for a semi-theoretical model for predicting the pool boiling heat transfer of refrigerant/lubricant mixtures on a roughened, horizontal, flat pool-boiling surface. Refrigerant/lubricant pool boiling is significantly influenced by the excess lubricant that resides in a thin layer on the boiling surface. The model is based on lubricant excess surface density measurements and a mass balance on lubricant deposition and removal from the wall. The original thermal boundary layer parameter that was derived for R123/lubricant mixtures predicts the R134a/lubricant data presented here for superheats below 6 K. The improvement in the thermal boundary layer enables the model to predict the full range of wall superheats for the R134a/lubricant mixture measurements. Further work on the new thermal boundary layer parameter is necessary to include the effects of the lubricant composition, viscosity, and critical solution temperature with the refrigerant. These effects were included in the original thermal boundary layer parameter. The new thermal boundary layer parameter increases the range of superheats for which the model is valid for the R134a/lubricant pool boiling heat transfer measurements presented here.

INTRODUCTION

The addition of lubricant to refrigerant can significantly alter the boiling performance due to lubricant accumulation at the heat transfer surface. Stephan (1963) 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 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 (2002a) 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 plain pool-boiling surface. The lubricant excess surface density measurements were used to derive a non-dimensional excess surface density parameter as a function of wall superheat and fluid properties (Kedzierski, 2003). The predictive model given in Kedzierski (2003) was based on lubricant excess surface density measurements and a conservation of mass of the lubricant at the boiling surface. In addition to the excess surface density measurements, the model relied on a thermal boundary layer parameter. The thermal boundary layer was developed for three R123/lubricant mixtures. The model was shown to predict the Kedzierski (2002b) data and other low pressure refrigerant/lubricant pool boiling data from the literature well. For example, the Bell *et al.* (1987) R113/lubricant pool boiling data was predicted on average to within $\pm 10\%$. In addition, Webb and McQuade (1993) measurements were predicted to within $\pm 25\%$ for heat fluxes greater than 20 kW/m^2 . Although the thermal boundary had provisions for accounting for the effects of lubricant properties, it had no variation with wall

superheat. Consequently, the present study attempts to improve the predictive capabilities of the refrigerant/lubricant pool boiling model presented in Kedzierski (2003) by deriving a wall superheat dependent thermal boundary layer parameter. The present study measures pool boiling heat transfer for a R134a/polyolester lubricant (POE) mixture at a (0.995/0.005) mass fraction. The viscosity of the POE investigated (DE589) was $21.76 \mu\text{m}^2/\text{s}$ at 313.15 K. The critical solution temperature (CST) of the R134a/DE589 mixture was approximately 270 K. The R134a/POE data are used to demonstrate the generality of the original pool boiling model that was developed using R123/naphthenic mineral oil heat transfer data, and to improve its prediction capabilities at higher heat fluxes.

1 APPARATUS

Figure 1 shows a schematic of the apparatus that was used to measure the pool boiling data of this study. More specifically, the apparatus was used to measure the liquid saturation temperature (T_s), the average pool-boiling heat flux (q''), and the wall temperature (T_w) of the test surface. The three principal components of the apparatus were test chamber, condenser, and purger. The internal dimensions of the test chamber were 25.4 mm \times 257 mm \times 1.54 m. The test chamber was charged with approximately 7 kg of refrigerant from the purger, giving a liquid height of approximately 80 mm above the test surface. As shown in Fig. 1, the test section was visible through two opposing, flat 150 mm \times 200 mm quartz windows. The bottom of the test surface was heated with high velocity (2.5 m/s) water flow. The vapor produced by liquid boiling on the test surface was condensed by the brine-cooled, shell-and-tube condenser and returned as liquid to the pool by gravity.

2 TEST SURFACE

Figure 2 shows the oxygen-free high-conductivity (OFHC) copper flat (plain) test plate used in this study. The test plate was machined out of a single piece of OFHC copper by electric discharge machining (EDM). A tub grinder was used to finish the heat transfer surface of the test plate with a crosshatch pattern. Average roughness measurements were used to estimate the range of average cavity radii for the surface to be between 12 μm and 35 μm . The relative standard uncertainty of the cavity measurements were approximately $\pm 12\%$. Further information on the surface characterization can be found in Kedzierski (2002a).

3 MEASUREMENTS AND UNCERTAINTIES

The standard uncertainty is the positive square root of the estimated variance. The individual standard uncertainties are combined to obtain the expanded uncertainty. The expanded uncertainty is calculated from the law of propagation of uncertainty with a coverage factor. All measurement uncertainties are reported for a 95 % confidence interval except where specified otherwise.

The copper-constantan thermocouples and the data acquisition system were calibrated against a glass-rod standard platinum resistance thermometer (SPRT) and a reference voltage to a residual standard deviation of 0.005 K. The NIST Thermometry Group calibrated the SPRT to two fixed points having expanded uncertainties of 0.06 mK and 0.38 mK. A quartz thermometer, which was calibrated with a distilled ice bath, agreed with the SPRT temperature to within approximately 0.003 K. Both the measured thermocouple electromotive force (EMF) and the measured 1 mV reference were regressed to the SPRT temperature. During a pool-boiling test, the 1 mV reference was measured prior to measuring each thermocouple EMF. The reference voltage was used to account for the drift in the acquisition measurement capabilities over time. Before each test run, the measurements of a thermocouple in the bath with the SPRT were compared. The thermocouple calibration was then adjusted so that bath thermocouple and the SPRT agreed. The mean absolute difference between the thermocouple and the SPRT before correcting for the drift was consistently around 0.07 K over the year of testing. Consequently, it is believed that the expanded uncertainty of the temperature measurements was less than 0.1 K.

Figure 2 shows the coordinate system for the 20 wells where individual thermocouples were force fitted into the side of the test plate. The wells were 16 mm deep to reduce conduction errors. Using a method given by Eckert and Goldstein (1976), errors due to heat conduction along the thermocouple leads were estimated to be well below 0.01 mK. The heat flux and the wall temperature were obtained by regressing the measured temperature distribution of the block to the governing two-dimensional conduction equation (Laplace equation). In other words, rather than using the boundary conditions to solve for the interior temperatures, the interior temperatures were used to solve for

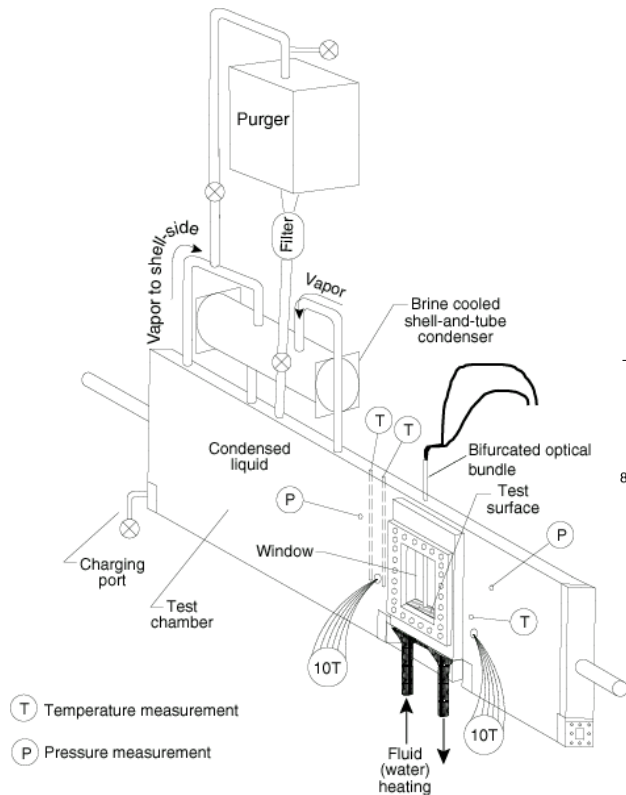


Figure 1: Schematic of test apparatus

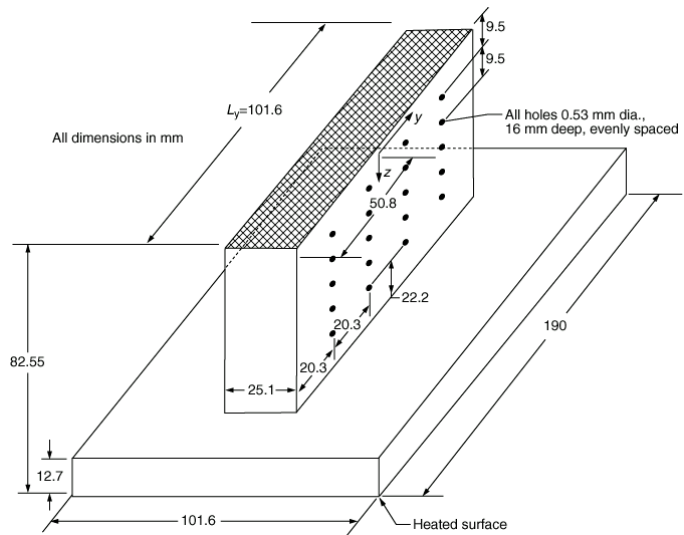


Figure 2: OFHC copper flat test plate with cross-hatched surface and thermocouple coordinate system

the boundary conditions following a backward stepwise procedure given in Kedzierski (1995). The average wall temperature (T_w) was calculated by integrating the local wall temperature that was obtained from the two-dimensional conduction solution. The relative expanded uncertainty in the heat flux was greatest at the lowest heat fluxes, approaching 8 % of the measurement at 10 kW/m^2 ; in general, it was relatively constant between 4 % and 5 % for heat fluxes above 25 kW/m^2 . The average random error in the wall superheat was between 0.02 K and 0.08 K.

4 EXPERIMENTAL RESULTS

The heat flux was varied from approximately 60 kW/m^2 to 1 kW/m^2 to simulate typical operating conditions of R134a chillers. All pool-boiling tests were taken at 277.6 K saturated conditions. The data were recorded consecutively starting at the largest heat flux and descending in intervals of approximately 4 kW/m^2 . The descending heat flux procedure minimized the possibility of any hysteresis effects on the data, which would have made the data sensitive to the initial operating conditions. The R134a/mixture was prepared by charging the test chamber with pure R134a and pure DE589 lubricant to known masses. The mass fraction of the R134a/DE589 lubricant mixture was $0.005 \pm 0.02 \%$ based on the lubricant.

Figure 3 is a plot of the measured heat flux (q'') versus the measured wall superheat ($T_w - T_s = \Delta T_s$) for R134a/DE589 (99.5/0.5) at a saturation temperature of 277.6 K. The open circles represent the measured data. Measurements were made over a period of approximately one month. Figure 3 shows that the boiling curve exhibits two characteristic regimes: a natural convection/boiling regime and a vigorous nucleate boiling regime. The regimes are separated by the cessation of vigorous nucleate boiling (CVNB). The CVNB occurs at a superheat of approximately 5.4 K.

5 MODEL DEVELOPMENT

5.1 Original Model

A brief overview of the refrigerant/lubricant pool boiling model present by Kedzierski (2003) is presented in this section and followed by a proposed modification to the calculation of the thermal boundary layer parameter given in

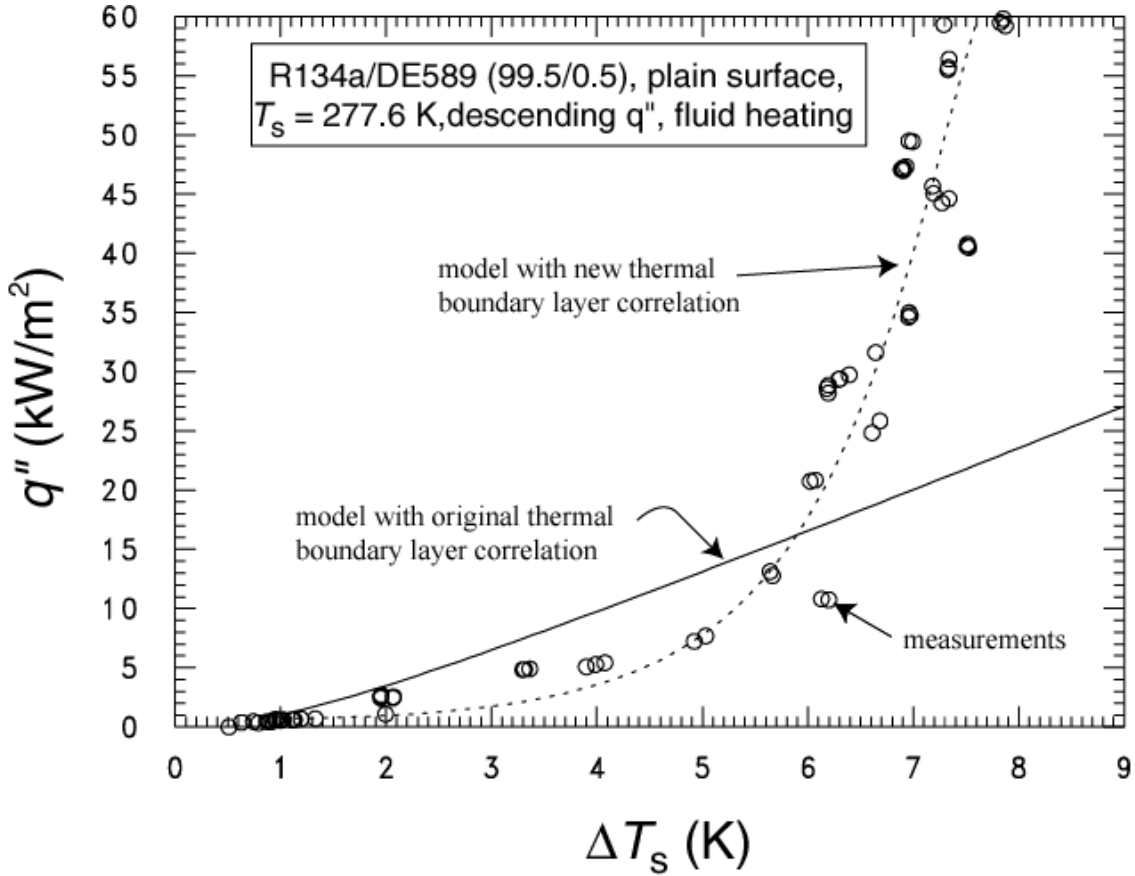


Figure 3: Heat flux versus wall superheat for R134a/DE589 (99.5/0.5) mixture

the model. The model is based on a non-dimensional lubricant excess surface density constant derived in Kedzierski (2003) and a mass balance on lubricant deposition and removal from the wall.

Figure 4 shows a hypothetical schematic of the average departure bubble for a refrigerant/lubricant mixture on a heated surface in a pool of liquid with corresponding thermal boundary layer temperature profiles and lubricant excess layer thickness. The hypothesis for the heat transfer model mechanism is that nearly pure lubricant resides on the heat transfer surface within the thickness of the lubricant excess layer, 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. The lubricant removal mechanism is assumed to be dominated by a mass transfer or a lifting of lubricant from the wall by the bubble. The thickness of the lubricant that is removed is limited by that which Van der Waals forces may hold to the bubble, which is approximate 25 \AA for lubricants (Laesecke, 2001).

Using the above assumptions, an expression for the departure bubble radius can be written based on a lubricant mass balance (Kedzierski, 2003):

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

where x_b is the mass fraction of the lubricant in the bulk mixture, ρ_L is the mass density of the lubricant, ρ_{rv} is the mass density of the refrigerant vapor, and 25 \AA was substituted for ζl_e per the preceding discussion. Equation (1) is valid for $x_b > 0$.

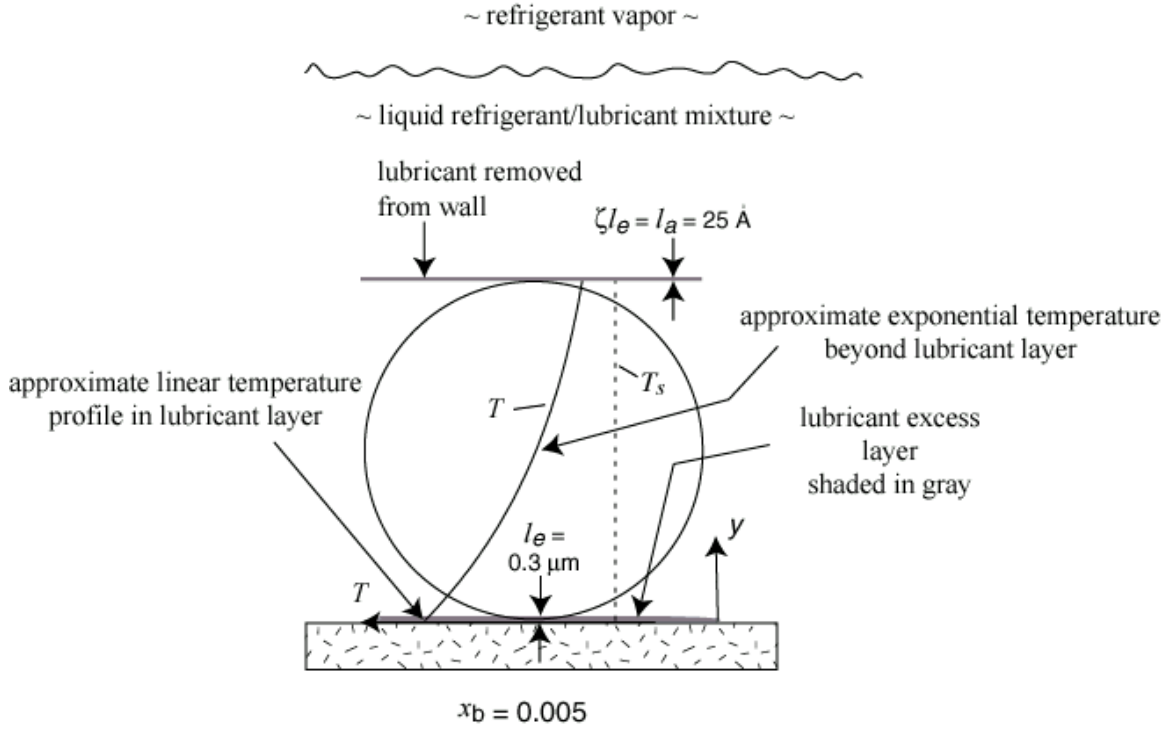


Figure 4: Schematic of the average departure bubble with excess layers (adapted from Kedzierski, 2003)

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. The non-dimensional constant that was derived from excess surface density (Γ) measurements in Kedzierski (2003) was given as:

$$\frac{x_b T_s \sigma (\rho_L - \rho_b x_b)}{\Gamma (1 - x_b) \rho_L h_{fg} \Delta T_s} = 5.9 \times 10^{-7} \quad (2)$$

where σ is the refrigerant liquid-vapor surface tension, h_{fg} is the latent heat of the refrigerant, T_s is the saturation temperature of the refrigerant, ρ_b is the liquid density of the bulk refrigerant/lubricant mixture, and ΔT_s is the wall superheat.

Equation (2) can be solved for l_e by using an approximate definition of the excess surface density, $\Gamma = l_e (\rho_L - \rho_b x_b)$:

$$l_e = \frac{\Gamma}{\rho_L - \rho_b x_b} = \frac{x_b T_s \sigma}{5.9 \times 10^{-7} (1 - x_b) \rho_L h_{fg} \Delta T_s} \quad (3)$$

The value of l_e is important because the temperature profile is assumed to be linear within the lubricant excess layer. Beyond the excess layer, the thermal boundary layer is assumed to decay exponentially to the saturation temperature of the bulk fluid (T_s). The thermal boundary layer parameter, λ , determines the rate of decay of the fluid temperature. Kedzierski (2003) gave λ as a function of the departure bubble radius and the ratio of the average mixture heat flux to the pure refrigerant heat flux:

$$\lambda = 0.27 + r_b \frac{q_m''}{q_p''} 10700 \text{ m}^{-1} \quad (4)$$

An expression for the average heat flux ratio was given as a function of the relative refrigerant/lubricant viscosity, the lubricant mass fraction, and the dimensionless critical solution temperature.

Finally, the refrigerant lubricant heat transfer coefficient (h_m) was given as:

$$h_m = \frac{q_m''}{T_w - T_s} = \frac{5.9 \times 10^{-7} (1 - x_b) \rho_L h_{fg} \Delta T_s k_L (1 - e^{-\lambda_e l_e})}{x_b T_s \sigma} \quad (5)$$

where eqs. (1), (3), and (4) are used to calculate r_b , l_e , and λ , respectively.

Figure 3 shows that eq. (5) (solid line) predicts the measured R134a/DE589 pool boiling data fairly well for superheats less than 6 K. However, the model fails to predict the pool boiling measurements for wall superheats greater than 6 K.

5.2 Modified Model

A new thermal boundary layer parameter λ expressed in terms of the wall superheat can be derived from equations that govern the formation of the excess layer. The excess layer does not exist prior to boiling of a refrigerant/lubricant mixture. From the moment that boiling begins on the surface ($t = 0$), the lubricant accumulates on the surface until a balance between lubricant deposition (\dot{m}_i) and removal (\dot{m}_o) has been established. The transient continuity equation for the lubricant mass (M) in the excess layer can be written as:

$$\frac{dM}{dt} = \frac{d\Gamma / N_b}{dt} = [\dot{m}_i - \dot{m}_o]_{oil} = f\pi \left\{ \frac{4r_b^3 \rho_{rv} x_b}{3(1-x_b)} - r_b^2 \rho_L \zeta l_e \right\} = B e^{-at} \quad (6)$$

where expressions for \dot{m}_i and \dot{m}_o were taken from Kedzierski (2003), N_b is the bubble site density, and f is the bubble frequency. The last term in the above equation is the assumed form of the rate of mass balance where B is a constant and a is a parameter.

Separating variables and integrating eq. (6) with respect to time (t) yields an expression for Γ :

$$\Gamma = N_b \int_0^{\infty} B e^{-at} dt = \frac{N_b B}{a} = \frac{N_b \dot{m}_i}{a} \quad (7)$$

The integration is for fixed superheat so the time constant a was taken as fixed. In addition, B represents the difference between the rate of lubricant deposition and removal at $t = 0$ which is taken as \dot{m}_i considering that no lubricant removal is possible at $t = 0$.

Solving eq. (7) for the time constant a gives:

$$a = \frac{N_b \dot{m}_i}{\Gamma} = \frac{N_b f 4\pi r_b^3 \rho_{rv} x_b}{3\Gamma(1-x_b)} \quad (8)$$

Neglecting conduction and convection from the wall, the heat flux can be approximated as:

$$q'' = \frac{4}{3} N_b f h_{fg} \pi r_b^3 \rho_{rv} \quad (9)$$

Solving eq. (9) for $N_b f$ and substituting the result into eq. (8) gives:

$$a = \frac{q'' x_b}{\Gamma h_{fg} (1-x_b)} \quad (10)$$

If it is assumed that the time constant varies exponentially with respect to the wall superheat ($a = A_0 e^{b\Delta T}$, where $b = 1 \text{ K}^{-1}$), the leading constant A_0 can be calculated using the measured q'' and ΔT_s and the Γ from eq. (3) as:

$$A_0 = \frac{q'' x_b}{\Gamma h_{fg} (1 - x_b) e^{b\Delta T}} \cong 5 \times 10^{-5} \text{ s}^{-1} \quad (11)$$

The thermal boundary layer parameter is related to the heat flux as (Kedzierski, 2003):

$$1 - e^{-\lambda_e / r_b} = \frac{q'' \Gamma}{k_L \Delta T_s (\rho_L - \rho_b x_b)} \quad (12)$$

Equation (12) can be linearized and solved for λ with use of the following approximation:

$$1 - e^{-\lambda_e / r_b} \cong \frac{\lambda_e / r_b}{1 + 0.62 \lambda_e / r_b} \quad (13)$$

which is valid for $0 < \lambda_e / r_b < 2.5$. Solving eq. (10) for q'' and substituting this and eq. (13) into eq. (12) and rearranging for λ yields:

$$\lambda = \frac{r_b \Gamma^2 h_{fg} / l_e}{\frac{k_L \Delta T_s (\rho_L - \rho_b x_b) x_b}{A_0 e^{b\Delta T_s} (1 - x_b)} - 0.62 \Gamma^2 h_{fg}} \quad (14)$$

The above expression for λ was used along with eqs. (1) and (3) to calculate the refrigerant/lubricant mixture heat transfer coefficient from eq. (5). The result is plotted as a dashed line in Fig. 3. Use of the new thermal boundary layer parameter enables eq. (5) to favorably predict the entire range of superheats while using a single fitting parameter A_0 . However, it is not expected that eq. (14) could be used to predict refrigerant/lubricant mixture other than the one presented here. Further work is required to generalize eq. (14) through the A_0 , where A_0 becomes a function of lubricant and refrigerant properties. In this way, eq. (14) would be able to predict the effect of lubricant and refrigerant properties. Nevertheless, the immediate purpose of eq. (14) has been served by demonstrating the relationship between the thermal boundary parameter and the wall superheat.

CONCLUSIONS

An improved thermal boundary layer parameter has been developed for use in a semi-theoretical model for pool boiling heat transfer of refrigerant/lubricant mixtures. 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 new thermal boundary layer parameter improves the range of superheats for which the model is valid, for the refrigerant/lubricant mixture presented here, and relies on a single fitting parameter. Further work is required to generalize the model to other refrigerant/lubricant mixtures.

NOMENCLATURE

a	time constant given by eq. (10), s^{-1}	\dot{m}_i	lubricant mass flow to the wall, kg/s
A_0	constant given by eq. (11)	\dot{m}_o	lubricant mass flow from the wall, kg/s
b	constant, 1 K^{-1}	M	lubricant mass, kg
B	constant in eq. (7), kg/s	N_b	bubble site density, m^{-2}
f	bubble frequency, s^{-1}	r_b	departure bubble diameter, m
h_{fg}	latent heat of refrigerant, J/kg	q''	heat flux, W/m^2
h_m	mixture heat transfer coefficient, $\text{W/m}^2\text{K}$	T_s	saturation temperature, K
k_L	thermal conductivity of lubricant, W/mK	T_w	wall temperature, K
l_e	thickness of lubricant excess layer, m		

x_b lubricant mass fraction of bulk mixture

ζ fraction of l_e that is lifted by bubble

Greek symbols

ΔT_s wall superheat ($T_w - T_s$), K
 Γ oil excess surface density eq. (3), kg/m²
 λ thermal boundary layer parameter
 ρ mass density, kg/m³
 σ refrigerant liquid-vapor surface tension, N/m

Subscripts

L pure lubricant
m refrigerant/lubricant mixture
p pure refrigerant
rv refrigerant vapor

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