

Effect of Al_2O_3 Nanolubricant on a Passively Enhanced R134a Pool Boiling Surface

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This paper quantifies the influence of Al_2O_3 nanoparticles on the pool boiling performance of R134a/polyolester mixtures on a Turbo-BII-HP boiling surface. An Al_2O_3 nanolubricant (a lubricant containing dispersed nano-size particles) was made by suspending nominally 10 nm diameter Al_2O_3 particles in a synthetic polyolester to roughly a 1.6% volume fraction. The nanoparticles caused, on average, a 12% degradation in the boiling heat transfer relative to that for R134a/polyolester mixtures without nanoparticles for the three lubricant mass fractions that were tested. The degradation was nearly constant for heat fluxes between 20 kW/m^2 and 120 kW/m^2 . It was speculated that the boiling heat transfer degradation was primarily due to a combination of (1) film boiling in the reentrant cavity rendering the nucleate boiling enhancement mechanism of the nanoparticles ineffective and (2) a reduction in bubble frequency due to the increased surface wetting as caused by the nanoparticles.

Keywords: Additives, aluminum oxide, boiling, enhanced heat transfer, nanolubricant, nanotechnology, refrigerants, refrigerant/lubricant mixtures, structured surface

INTRODUCTION

In recent years, nanofluids, i.e., liquids with dispersed nano-size particles, have been shown to be a potential means for enhancing the performance of chillers (Liu et al., 2009; Kedzierski, 2009). A major motivation for improving chiller performance is that energy efficiency is a primary component for net zero energy, high performance green building-design (OSTP 2008, EPA 2008). Chillers, that provide air conditioning for U.S. buildings, account for nearly 13% of total building electric expendi-

tures (EIA, 2008). Consequently, a cost-effective means for improving the efficiency of chillers would facilitate meeting green building goals.

Lubricant-based nanofluids, i.e., nanolubricants, can facilitate the stability of the nanoparticles in a refrigerant cycle while delivering them to the components of the cycle where they can produce the most benefit. Bi et al. (2007a) have shown that nanoparticles in compressor lubricant can improve its performance. Likewise, nanoparticles in the lubricant excess layer that covers evaporator surfaces can interact with nucleating bubbles and cause a heat transfer

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enhancement (Kedzierski, 2009). The combined effects of nanoparticles on heat transfer and compressor performance were illustrated by Bi et al. (2007b) when they showed that nanolubricants produced energy savings of more than 25% in domestic refrigerators. These preceding studies suggest that it is worthwhile to investigate the potential benefits of nanolubricants for chillers.

Kedzierski (2010a) showed that aluminum oxide (Al_2O_3) nanolubricants can improve the refrigerant boiling heat transfer of a 2% by mass mixture on a smooth surface, on average, by roughly 150% for heat fluxes less than 40 kW/m^2 . The purpose of the present investigation was to determine if similar boiling heat transfer improvements could be obtained for a passively enhanced (structure) boiling surface. In order to investigate the influence of nanoparticle properties on refrigerant/lubricant pool boiling on a passively enhanced surface, the boiling heat transfer of three R134a/nanolubricant mixtures on a horizontal, flat, copper (Turbo-BII-HP)[†] surface were measured. A commercial polyolester lubricant (RL68H) with a nominal kinematic viscosity of $72.3\ \mu\text{m}^2/\text{s}$ at 313.15 K was the base lubricant that was mixed with nominally 10 nm diameter Al_2O_3 nanoparticles. Al_2O_3 nanoparticles have the advantages of a well-established, successful dispersion technology and being relatively inert with respect to lubricated compressor parts.

A manufacturer used a proprietary surfactant at a mass between 15% and 20% of the mass of the Al_2O_3 as a dispersant for the RL68H/ Al_2O_3 mixture (nanolubricant). The manufacturer made the mixture such that 40% of the mass was Al_2O_3 particles. The mixture was diluted in-house to a 5.6% mass fraction of Al_2O_3 by adding neat RL68H and ultrasonically mixing the solution for approximately 24 h. A Dynamic Light Scattering (DLS) technique was used to measure the average nanoparticle size on a number basis. The diameter of most of the particles was approximately 10 nm ($10.1\ \text{nm} \pm 1.3\ \text{nm}$) and the particles were well dispersed (Kedzierski, 2010a). Figure 1 shows a Transmission Electron Microscopy (TEM) image of

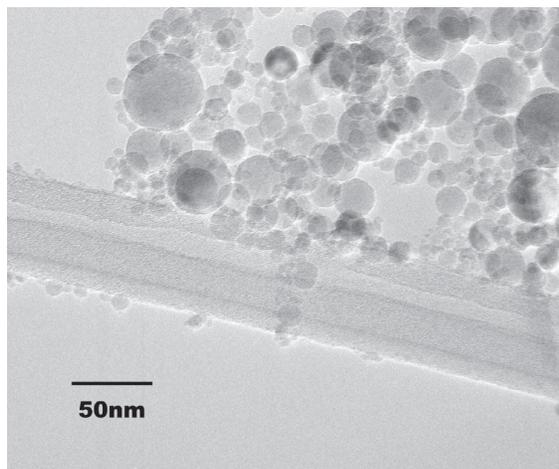


FIGURE 1
TEM of Al_2O_3 nanolubricant (Sarkas, 2009).

the nanoparticles as taken by Sarkas (2009). The image confirms the good dispersion and shows that the particles are spherical with most of them having diameters of approximately 10 nm or less and a few having diameters close to 50 nm.

The mass fraction was chosen so that it matched the nanoparticle mass fraction of the RL68H/ Al_2O_3 study on a roughened, flat surface (Kedzierski, 2010). The RL68H/ Al_2O_3 (98.4/1.6) volume fraction[‡] mixture, a.k.a. RL68H1A1O, was mixed with pure R134a to obtain three R134a/RL68H1A1O mixtures at nominally 0.5%, 1%, and 2% mass fractions for the boiling tests. In addition, the boiling heat transfer of three R134a/RL68H mixtures (0.5%, 1%, and 2% mass fractions), without nanoparticles, was measured to serve as a baseline for comparison to the RL68H1A1O mixtures.

APPARATUS

Figure 2 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

[†]Certain commercial equipment, instruments, or materials are identified in this paper in order to specify the experimental procedure adequately. Such identification is not intended to imply recommendation or endorsement by the National Institute of Standards and Technology, nor is it intended to imply that the materials or equipment identified are necessarily the best available for the purpose.

[‡]The equivalent mixture is RL68H/ Al_2O_3 (94.4/5.6) in terms of mass.

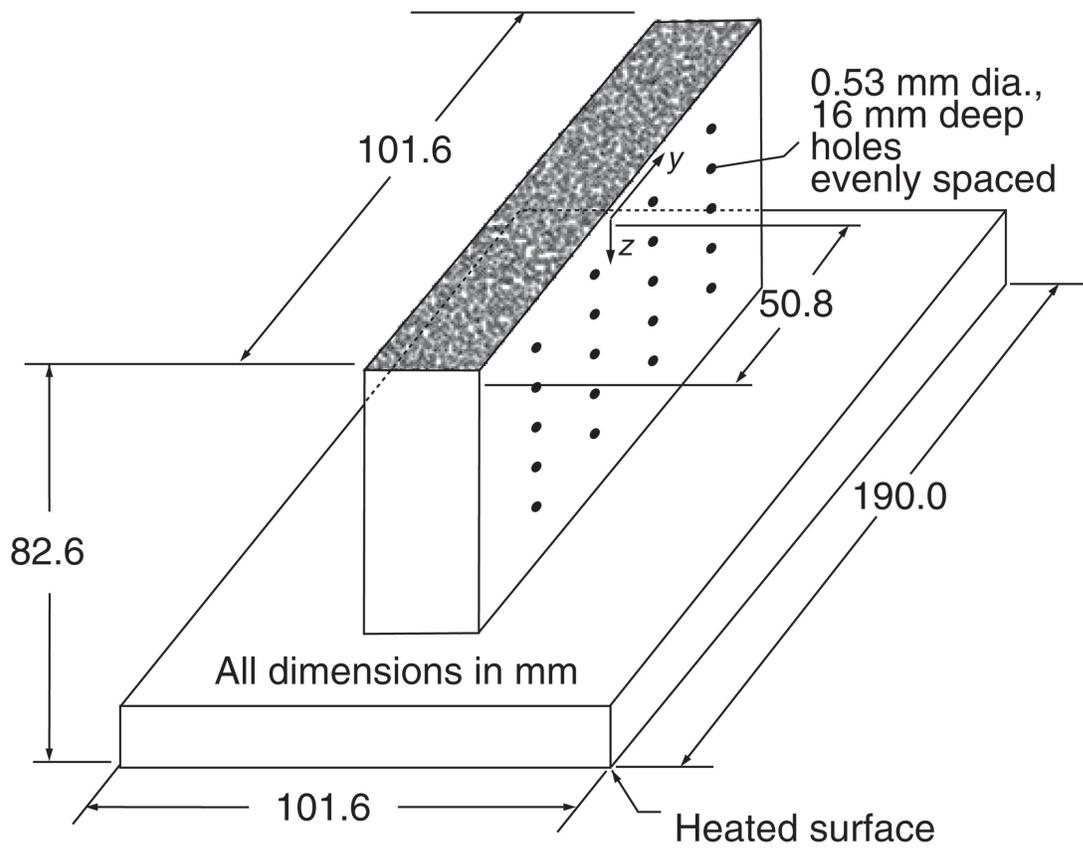
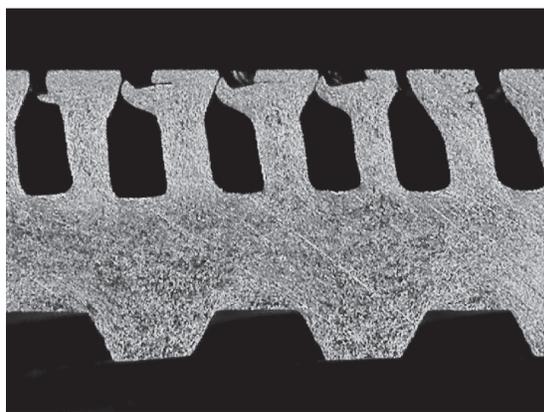
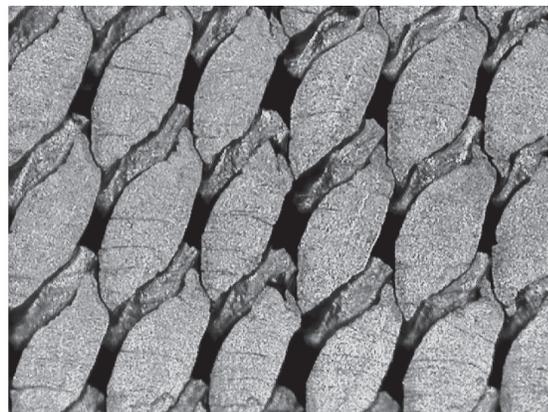


FIGURE 3
OFHC copper flat test plate with Turbo-BII-HP surface and thermocouple coordinate system.



— 0.1 mm

SIDE VIEW



— 0.1 mm

TOP VIEW

FIGURE 4
Photograph of Turbo-BII-HP surface.

other words, rather than using the boundary conditions to solve for the interior temperatures, the interior temperatures were used to solve for the boundary conditions following a backward stepwise procedure given in Kedzierski (1995). As shown in Fig. 3, the origin of the coordinate system was centered on the surface with respect to the y -direction at the heat transfer surface. Centering the origin in the y -direction reduced the uncertainty of the wall heat flux and temperature calculations by reducing the number of fitted constants involved in these calculations.

Fourier's law and the fitted constants from the Laplace equation were used to calculate the average heat flux (q'') normal to and evaluated at the heat transfer surface based on its projected area. The average wall temperature (T_w) was calculated by integrating the local wall temperature (T). The wall superheat was calculated from T_w and the measured temperature of the saturated liquid (T_s). Considering this, the relative expanded uncertainty in the heat flux ($U_{q''}$) was greatest at the lowest heat fluxes, approaching 8% of the measurement near 20 kW/m^2 . In general, the $U_{q''}$ remained approximately between 3% and 6% for heat fluxes greater than 50 kW/m^2 . The average random error in the wall superheat (U_{T_w}) remained between 0.04 K and 0.12 K. Plots of $U_{q''}$ and U_{T_w} versus heat flux can be found in Kedzierski (2010b).

EXPERIMENTAL RESULTS

The heat flux was varied between approximately 10 kW/m^2 and 140 kW/m^2 to simulate a range of possible operating conditions for R134a chillers. All pool-boiling measurements were made 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. Although not measured, it was believed that refrigerant/nanolubricant mixture boiling would suffer hysteresis similar to that for boiling without nanoparticles if not guarded against. Kedzierski (2010b) presents the measured heat flux and wall superheat for all the data of this study. A total of 2108 measurements were made over 46 days.

The mixtures were prepared by charging the test chamber (see Fig. 2) with pure R134a to a known mass. Next, a measured mass of nanolubricant or lubricant was injected with a syringe through a port in the test chamber. The refrigerant/lubricant solution was mixed by flushing pure refrigerant through the same port where the lubricant was injected. All compositions were determined from the masses of the charged components and are given on a mass fraction basis. The maximum uncertainty of the mass fraction measurement is approximately 0.02%, e.g., the range of a 2.0% mass fraction is between 1.98% and 2.02%. Nominal or target mass compositions are used in the discussion. For example, the "actual" mass composition of the RL68H in the R134a/RL68H (99.5/0.5) mixture was $0.50\% \pm 0.02\%$. Likewise, the RL68H mass fractions for R134a/RL68H (99/1) and the R134a/RL68H (98/2) mixtures were $1.00\% \pm 0.02\%$ and $1.99\% \pm 0.02\%$, respectively. Using the same uncertainties, the nanolubricant mass fractions as tested with R134a were 0.50%, 0.99%, and 2.00%.

Figure 5 is a plot of the measured heat flux (q'') versus the measured wall superheat ($T_w - T_s$) for pure R134a pool boiling on the Turbo-BII-HP surface at a saturation temperature of 277.6 K . These measurements serve as a baseline for comparison to the refrigerant/pure-lubricant measurements. The open triangles represent the measured data while the solid line is a cubic best-fit regression or estimated means of the data. Six days of boiling pure R134a produced 252 measurements over a period of nearly two weeks. Two of the 252 measurements were removed before fitting because they were identified as "outliers" based on having both high influence and high leverage (Belsley et al., 1980). The data sets for each test fluid presented in this manuscript exhibited a similar number of outliers and were regressed in the same manner. Table 1 gives the constants for the cubic regression of the superheat versus the heat flux for all of the fluids tested here. The residual standard deviation of the regressions – representing the proximity of the data to the mean – are given in Kedzierski (2010b) along with the magnitude of the 95% confidence intervals. The dashed lines to either side of the mean represent the lower and upper 95% simultaneous (multiple-use) confidence intervals for the mean. From the confidence intervals, the expanded uncertainty of the estimated mean wall superheat was, on average, 0.07 K.

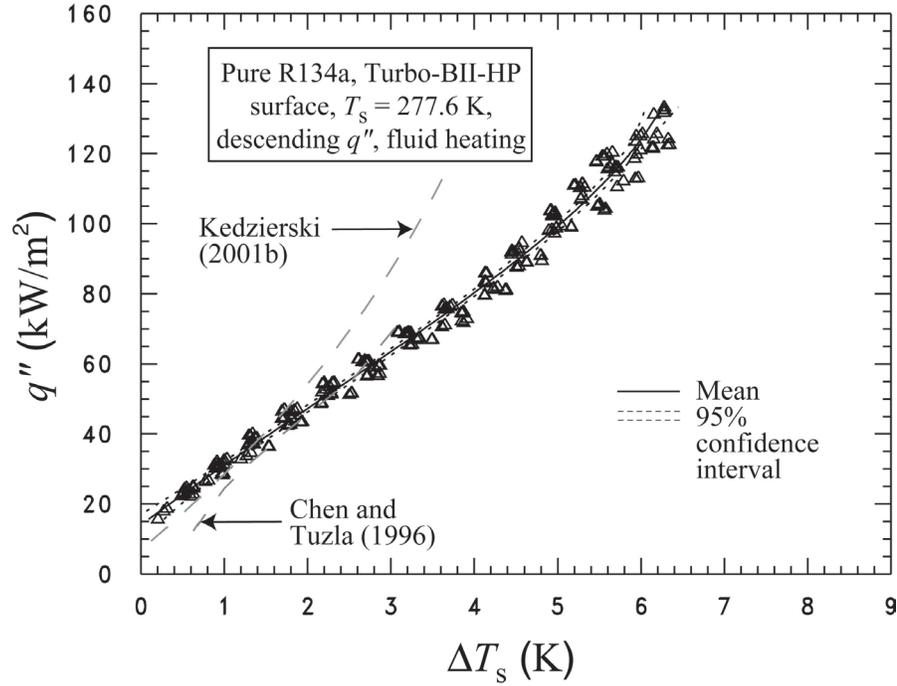


FIGURE 5
Pure R134a boiling curve for Turbo-BII-HP surface.

TABLE 1
Estimated parameters for cubic boiling curve fits for Turbo-BII-HP copper surface.

$$\Delta T_s = A_0 + A_1 q'' + A_2 q''^2 + A_3 q''^3$$

ΔT_s in kelvin and q'' in W/m^2

Fluid	A_0	A_1	A_2	A_3
Pure R134a				
0.1 K $\leq \Delta T_s \leq$ 6.3 K	-0.740527	4.95277×10^{-5}	2.63884×10^{-10}	-1.81569×10^{-15}
R134a/RL68H (99.5/0.5)				
0.12 K $\leq \Delta T_s \leq$ 3.8 K	0.252376	-4.07073×10^{-5}	2.76832×10^{-9}	-1.99733×10^{-14}
3.8 K $\leq \Delta T_s \leq$ 7.6 K	-3.29612	1.64852×10^{-4}	-1.00471×10^{-9}	2.90580×10^{-15}
R134a/RL68H (99/1)				
0.1 K $\leq \Delta T_s \leq$ 4.0 K	-0.051371	5.16322×10^{-6}	1.90984×10^{-9}	-1.39618×10^{-14}
4.0 K $\leq \Delta T_s \leq$ 7.6 K	-6.60753	2.89968×10^{-4}	-2.24421×10^{-9}	6.50065×10^{-15}
R134a/RL68H (98/2)				
0.1 K $\leq \Delta T_s \leq$ 5.0 K	-0.243947	1.95563×10^{-5}	1.91732×10^{-9}	-1.61984×10^{-14}
5.0 K $\leq \Delta T_s \leq$ 7.1 K	0.580891	1.10818×10^{-4}	-8.53909×10^{-10}	2.80652×10^{-15}
R134a/RL681A1O (99.5/0.5)				
0.17 K $\leq \Delta T_s \leq$ 3.3 K	-0.123590	-9.70313×10^{-6}	3.00869×10^{-9}	-2.98665×10^{-14}
3.3 K $\leq \Delta T_s \leq$ 7.8 K	-1.58427	1.24654×10^{-4}	-5.43675×10^{-10}	1.11426×10^{-15}
R134a/RL681A1O (99/1)				
0.18 K $\leq \Delta T_s \leq$ 3.7 K	-0.516802	4.44071×10^{-5}	1.55037×10^{-9}	-1.40552×10^{-14}
3.7 K $\leq \Delta T_s \leq$ 7.8 K	-3.88192	2.23548×10^{-4}	-1.61582×10^{-9}	4.49216×10^{-15}
R134a/RL681A1O (99/2)				
0.12 K $\leq \Delta T_s \leq$ 4.4 K	-0.482096	4.94608×10^{-5}	1.43488×10^{-9}	-1.34728×10^{-14}
4.4 K $\leq \Delta T_s \leq$ 7.1 K	-5.42332	3.05743×10^{-4}	-2.80148×10^{-9}	9.14222×10^{-15}

Two long-dashed lines in Fig. 5 compare measurements of other studies to the present study. The R134a pool boiling curve of Chen and Tuzla (1996) on a 19 mm OD Turbo-BII-HP tube at $T_s = 277.6$ K crosses the present measured mean at a superheat of approximately 2.6 K. The maximum deviation between the present measured mean and the Chen and Tuzla (1996) boiling curve is 0.6 K, while the average absolute deviation is less than 0.2 K. The R134a boiling curve of Kedzierski (2001b) for a Turbo-BII-HP surface on a flat plate crosses the present measured mean superheat at approximately 1.4 K. For the same heat flux range as the Chen and Tuzla (1996) boiling curve, the average absolute deviation of the Kedzierski (2001b) curve from the present mean curve is less than 0.4 K. It is believed that the above comparisons validate the present measurements.

Figure 6 is a plot of the measured heat flux (q'') versus the measured wall superheat ($T_w - T_s$) for the refrigerant/pure-lubricant mixtures at a saturation temperature of 277.6 K. The refrigerant/pure-lubricant measurements serve as the baseline for comparison to the refrigerant/nanolubricant pool boiling measure-

ments. Twenty boiling curves were measured over the span of approximately two months. The open circles, squares, and stars represent the measured heat flux (q'') versus the measured wall superheat ($T_w - T_s$) at a saturation temperature of 277.6 K for the R134a/RL68H (99.5/0.5), R134a/RL68H (99/1), and R134a/RL68H (98/2) mixtures, respectively. From the 95% multi-use confidence intervals, the expanded uncertainty of the estimated mean wall superheat was, on average, 0.09 K.

A general overview of the effect that the variation in the pure lubricant mass fraction has on R134a/lubricant pool boiling on the Turbo-BII-HP surface can be obtained from Fig. 6. Comparison of the three mean boiling curves shows that the superheats are within approximately 0.8 K of each other for the entire tested heat flux range. For the most part, the superheat for the refrigerant/lubricant mixtures is 0.3 K to 2.0 K greater than that for pure R134a indicating a heat transfer degradation with respect to pure R134a. Kedzierski (2001b) has shown that, in general, degradations associated with increased lubricant mass fractions occur when the concentration-induced bubble size reduc-

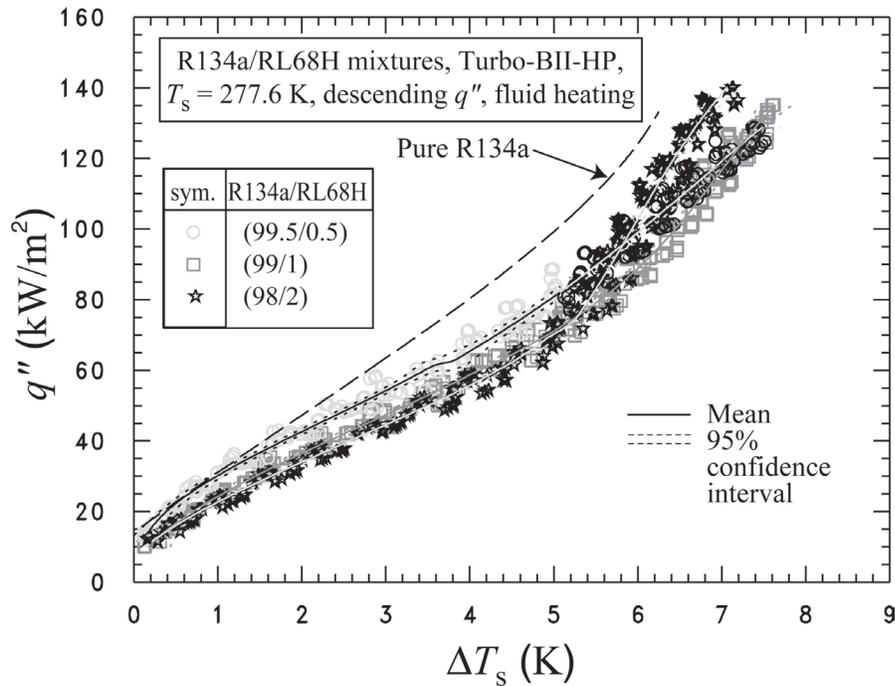


FIGURE 6
R134a/RL68H mixtures boiling curves for Turbo-BII-HP surface.

tion, and its accompanying loss of vapor generation per bubble, is not compensated by an increase in site density. Typically, heat transfer degradations have been observed to increase with respect to increasing lubricant mass fraction. The present measurements are consistent with this heat transfer performance trend for heat fluxes less than approximately 67 kW/m^2 . At a heat flux of approximately 67 kW/m^2 , the (98/2) mixture boiling curve crosses over the (99/1) mixture and the (99.5/0.5) mixture boiling curves, providing better boiling performance than either the (99/1) mixture or the (99.5/0.5) mixture for heat fluxes larger than 92 kW/m^2 .

A more precise comparison of the R134a/RL68H heat transfer performances relative to pure R134a is given in Fig. 7. Figure 7 plots the ratio of the R134a/RL68H mixture heat flux to the pure R134a heat flux (q''_{PL}/q''_p) versus the pure R134a heat flux (q''_p) at the same wall superheat. Figure 7 illustrates the influence of lubricant mass fraction on the R134a/RL68H boiling curve with solid and dashed lines representing the mean heat flux ratios for each mixture and shaded regions showing the 95% multi-use confidence level for each mean. A heat transfer degradation exists where

the heat flux ratio is less than one and the 95% simultaneous confidence intervals (depicted by the shaded regions) do not include the value one. For all compositions, the lubricant has caused a heat transfer degradation relative to the mean heat transfer of pure R134a for all measured q''_p . For the most part, the mean refrigerant/lubricant pool boiling heat flux resides between roughly 99% and 60% of that of the pure refrigerant. The heat flux ratio (q''_{PL}/q''_p) for all of the mixtures decreased slightly with respect to increasing heat flux for heat fluxes greater than 30 kW/m^2 . The average heat flux ratio for each mixture between 30 kW/m^2 and 100 kW/m^2 was: 0.86, 0.75, and 0.71 for the R134a/RL68H (99.5/0.5), the R134a/RL68H (99/1), and the R134a/RL68H (98/2) mixtures, respectively. The minimum heat flux ratio for each mixture (for confidence intervals that do not include the value one) is shown in Fig. 7 and occurred for heat fluxes greater than 100 kW/m^2 : 0.80 ± 0.02 , 0.72 ± 0.02 , and 0.69 ± 0.06 for the R134a/RL68H (99.5/0.5), the R134a/RL68H (99/1), and the R134a/RL68H (98/2) mixtures, respectively.

Figure 8 shows the measured heat flux (q'') versus the measured wall superheat ($T_w - T_s$) for three mixtures of R134a and the nanolubricant RL68H1A1O at

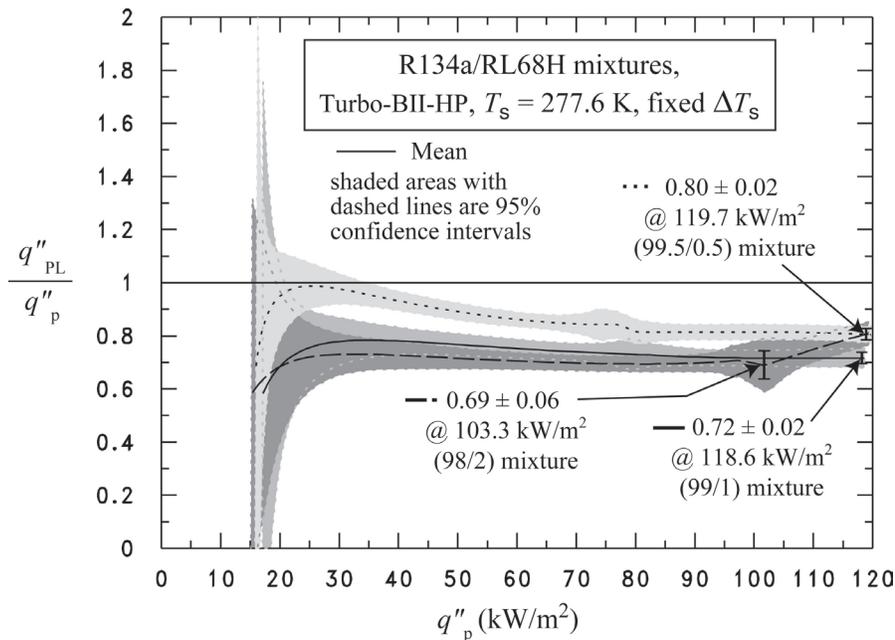


FIGURE 7
Boiling heat flux of R134a/RL68H mixture relative to that of pure R134a for Turbo-BII-HP surface.

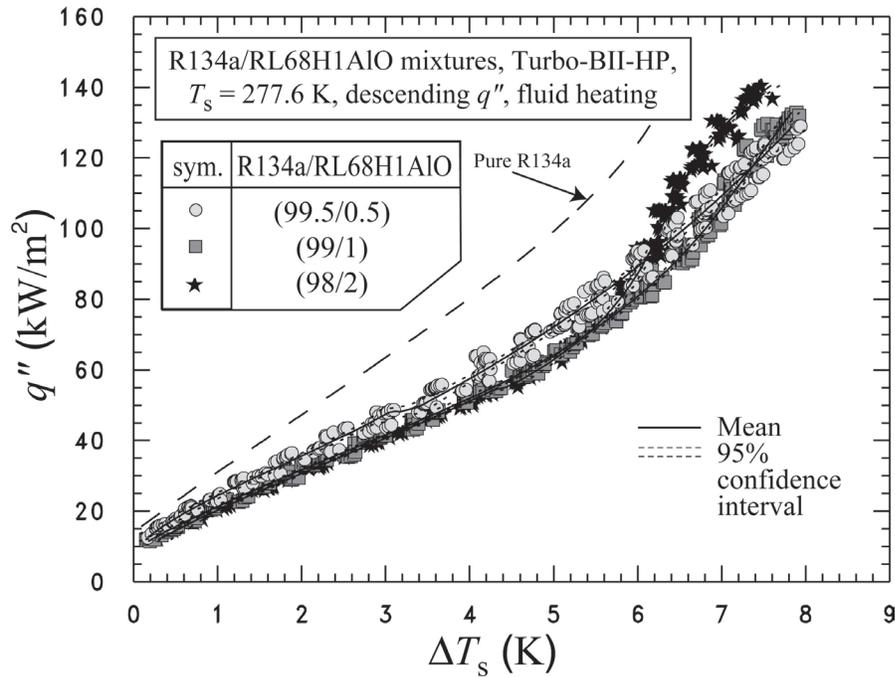


FIGURE 8
R134a/RL68H1AlO mixtures boiling curves for Turbo-BII-HP surface.

a saturation temperature of 277.6 K. Twenty-two boiling curves were measured over the span of approximately five weeks. The closed circles, squares, and stars represent the measurements for the R134a/RL68H1AlO (99.5/0.5), R134a/RL68H1AlO (99/1), and R134a/RL68H1AlO (98/2) mixtures, respectively. From the 95% multi-use confidence intervals, the expanded uncertainty of the estimated mean wall superheat was, on average, 0.17 K.

As Fig. 8 shows, a modest variation in the boiling performance is exhibited among the three nanofluids given the variation in composition. For example, the mean boiling curves of the refrigerant/nanolubricant mixtures differ by no more than 0.8 K. For heat fluxes less than 95 kW/m², the difference in superheat between the (99/1) mixture and the (98/2) mixture is less than 0.2 K. In this region, the boiling performance of the R134a/RL68H1AlO mixtures is ranked from best to worst as follows: (99.5/0.5), (99/1), and (98/2). The (98/2) nanolubricant mixture boiling curve exhibits the same crossover at approximately 67 kW/m² that was observed in Fig. 6 for the R134a/pure-lubricant mixtures.

Figure 9 summarizes the influence of Al_2O_3 nanoparticles on R134a/RL68H boiling heat transfer. The figure plots the ratio of the R134a/RL68H1AlO heat flux to the R134a/RL68H heat flux ($q''_{\text{Al}}/q''_{\text{PL}}$) versus the R134a/RL68H mixture heat flux (q''_{PL}) at the same wall superheat for the Turbo-BII-HP surface. The three different compositions are represented by three solid lines where each R134a/nanolubricant mixture is compared to the R134a/pure-lubricant mixture at the same mass fraction. Although the mean heat flux ratio for the (99/1) mixture is greater than one for heat fluxes less than 14 kW/m², neither an enhancement nor a degradation can be claimed in this region because the confidence intervals include the value one. Similarly, the confidence intervals show that the nanoparticles have caused a relatively constant heat transfer degradation compared to the boiling of the R134a/RL68H mixtures (without nanoparticles), for heat fluxes greater than 20 kW/m². The average heat flux ratio for heat fluxes greater than 20 kW/m² is approximately 0.87, 0.89, and 0.89 for the 0.5%, the 1%, and the 2% nanolubricant mass fractions, respectively. Consequently, the nanolubricant mass fraction had

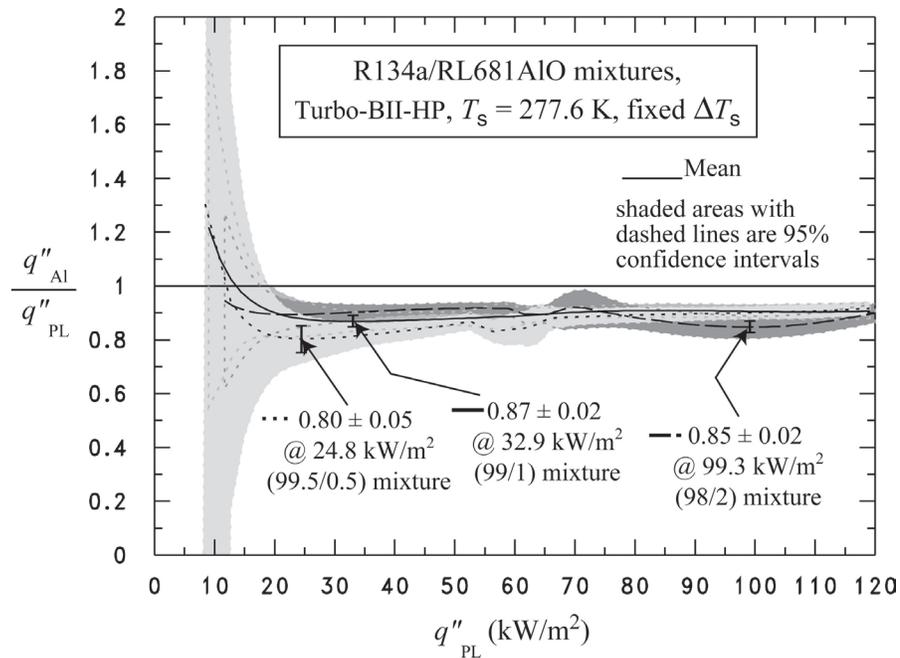


FIGURE 9
Boiling heat flux of R134a/RL68H1A1O mixtures relative to that of R134a/RL68H without nanoparticles for Turbo-BII-HP surface.

little influence on the magnitude of the heat transfer degradation.

DISCUSSION[§]

Figure 10 is used to illustrate why the same nanolubricant that was used to significantly enhance R134a boiling on a smooth surface in the Kedzierski (2010a) study caused a roughly 12% degradation for the Turbo-BII-HP reentrant cavity surface. The schematic shows the Turbo-BII-HP fin in cross section revealing its cavity with a mouth radius of r_c . The nanolubricant excess layer is shown, roughly to scale, being approximately $40\mu\text{m}$ thick (Kedzierski, 2002) on the approximately 0.7mm high fin. Kedzierski (2010a) proposed that the enhancement caused by the nanolubricant for a smooth boiling surface was due to surface work on bubbles as caused by momentum transfer between growing bubbles and nanoparticles suspended within the lubricant excess layer. It is proposed here that the momentum transfer between bubbles and particles

still occurs for reentrant cavities; however, thin film evaporation into large seed bubbles in the reentrant cavity suppresses the nucleate boiling in the nanolubricant excess layer, thus, reducing the interaction between nanoparticles and bubbles in the nanolubricant excess layer.

Arshad and Thome (1983) show that thin film evaporation primarily governs the heat transfer within reentrant cavities. Because of this, bubble nucleation at the wall is likely to be reduced, and thus, have less influence on the overall heat-transfer. Being that the nanoparticles enhance boiling heat transfer via momentum transfer between nanoparticles and bubbles during bubble growth from the wall, the enhancement effect of the nanoparticles is minimized for reentrant cavity boiling. For reentrant cavities, the boiling is controlled primarily by the cavity geometry, including the size of its opening and the contact angle that the bubble makes at the cavity mouth (Griffith and Wallis, 1960; Chien and Webb, 1998). In addition, Gerardi et al. (2010) have shown that nanoparticles can cause increased surface wetting, i.e., reduced contact angle, which leads to a

[§]The boiling mechanisms discussed here are not believed to be refrigerant specific.

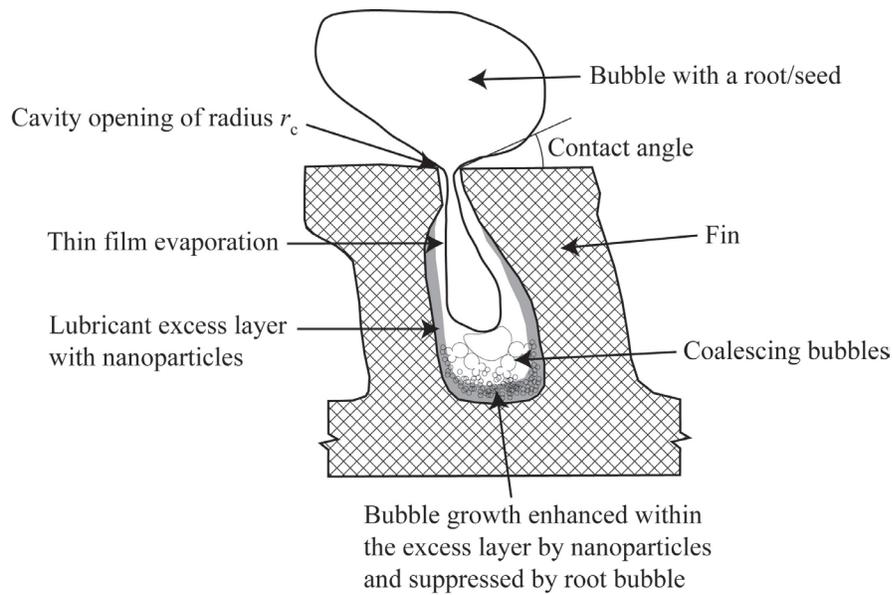


FIGURE 10
Mechanistic speculation of influence of nanoparticles on reentrant cavity bubble production.

small bubble departure frequency. A reduced bubble frequency could explain the approximately 12% heat transfer degradation exhibited by the refrigerant/nanolubricant boiling on the Turbo-BII-HP surface.

No white deposits were visible on the boiling surface after test, suggesting that the lubricant excess layer was successful at suspending the Al_2O_3 nanoparticles above the surface and, thus, limiting the number that become lodged in the surface cavities. For this reason, one would think that any heat transfer degradation that could have been attributed to nanoparticles filling surface cavities would have been compensated for by the heat transfer enhancement as caused by nanoparticles in the lubricant excess layer. However, the question remains if there were enough particles in the excess layer to cause an enhancement. Such a critical nanoparticle volume fraction for obtaining boiling improvement was observed for the smooth surface (Kedzierski, 2009). The reentrant cavity surface, because of its larger surface area, may require a larger nanoparticle volume fraction in the lubricant than the smooth surface, and larger than what was tested here, to obtain an enhancement in boiling. Kedzierski (2011) has shown that this is the case for the rectangular-finned surface. The rectangular-finned surface, because of its greater surface area, required greater

nanoparticle loading to achieve a boiling heat transfer enhancement of similar magnitude to that of a smooth surface. In light of these findings, future research is required to determine if a nanoparticle mass fraction larger than 5.6% is required to improve the boiling heat transfer of reentrant cavity surfaces.

CONCLUSIONS

The effect of Al_2O_3 nanoparticles on the boiling performance of R134a/polyolester mixtures on a flattened, horizontal Turbo-BII-HP surface was investigated. A nanolubricant containing roughly 10 nm diameter Al_2O_3 nanoparticles at 1.6% volume fraction with a polyolester lubricant was mixed with R134a at three different mass fractions. Even though a previous study had shown significant enhancements for a plain surface, the Al_2O_3 nanoparticles caused a heat transfer degradation relative to the heat transfer of pure R134a/polyolester for all three of lubricant mass fractions on the Turbo-BII-HP surface. For all nanolubricant mass fractions with refrigerant, the degradation in heat flux was approximately 12% and nearly independent of superheat. The average heat flux degradation for heat fluxes larger than 20 kW/m^2 was approximately 13%,

11%, and 11% for the 0.5%, the 1%, and the 2% mass fractions, respectively.

It was speculated that the boiling heat transfer degradation was potentially due to a combination of film boiling in the reentrant cavity rendering the nucleate boiling enhancement mechanism of the nanoparticles ineffective and a reduction in bubble frequency due to the increased surface wetting caused by the nanoparticles. It was also speculated that a nanoparticle loading larger than what was used in this study is required to improve the boiling heat transfer of reentrant cavity surfaces.

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NOMENCLATURE

English Symbols

A_n	regression constant in Table 1 $n = 0, 1, 2, 3$
q''	average wall heat flux, $W \cdot m^{-2}$
r_c	radius of cavity opening, m
T	temperature, K
T_w	temperature at roughened surface, K
U	expanded uncertainty

English Subscripts

Al	nanolubricant
p	pure R134a

PL	refrigerant/pure lubricant (R134a/RL68H) mixture
q''	heat flux
s	saturated state
T_w	wall temperature

REFERENCES

- Arshad, J., and Thome, J. R., 1983, "Enhanced Boiling Surfaces: Heat Transfer Mechanism and Mixture Boiling," ASME/JSME Joint Thermal Engineering Conference, Honolulu, Vol. 1, pp. 191–197.
- Belsley, D. A., Kuh, E., and Welsch, R. E., 1980, *Regression Diagnostics: Identifying Influential Data and Sources of Collinearity*, New York: Wiley.
- Bi, S., Shi, L., and Zhang, L., 2007a, "Performance Study of a Domestic Refrigerator Using R134a/Mineral Oil/Nano-TiO₂ as Working Fluid," *Proceedings of International Conference of Refrigeration*, Beijing, ICRO7-B2-346.
- Bi, S., Shi, L., and Zhang, L., 2007b, "Application of Nanoparticles in Domestic Refrigerators," *Applied Thermal Engineering*, Vol. 28, pp. 1834–1843.
- Chien, L. H., and Webb, R. L., 1998, "A Nucleate Boiling Model for Structured Enhanced Surfaces," *Int. J. Heat Mass Transfer*, Vol. 41, No. 14, pp. 2183–2195.
- Chen, J., and Tuzla, K., 1996, "Heat Transfer Characteristics of Alternative Refrigerants; Volume 3: Condenser and Evaporator Outside Tube," EPRI TR-106016-V3, project 3412–53, Palo Alto.
- Energy Information Administration (EIA), 2008, "2003 Commercial Buildings Energy Consumption Survey: Consumption and Expenditures Tables," http://www.eia.doe.gov/emeu/cbecs/cbecs2003/detailed_tables_2003/2003set15/2003pdf/c13a.pdf (January 2009).
- Environmental Protection Agency (EPA), 2008, "EPA Green Building Strategy," EPA-100-F-08-073.
- Gerardi C., Buongiorno, J., Hu, L. W., and McKrell, T., 2010, "Infrared Thermometry Study of Nanofluid Pool Boiling Phenomena," accepted for publication in *HKIE Transactions*.
- Griffith, P., and Wallis, J. D., 1960, "The Role of Surface Conditions in Nucleate Boiling," *Chem. Eng. Prog. Symp. Ser.*, Vol. 56, No. 49, pp. 49–63.
- Kedzierski, M. A., 2011, "R134a/Al₂O₃ Nanolubricant Mixture Boiling on a Rectangular Finned Surface with Extensive Measurement and Analysis Details," NIST Technical Note 1694, U.S. Department of Commerce, Washington, D.C.
- Kedzierski, M. A., 2010a, "Effect of Al₂O₃ Nanolubricant on R134a Pool Boiling Heat Transfer," submitted for publication in the *Int. J. Refrigeration*.
- Kedzierski, M. A., 2010b, "Effect of Al₂O₃ Nanolubricant on a Passively Enhanced R134a Pool Boiling Surface with Extensive Measurement and Analysis Details," NIST Technical Note 1677, U.S. Department of Commerce, Washington, D.C.

- Kedzierski, M. A., 2009, "Effect of CuO Nanoparticle Concentration on R134a/Lubricant Pool-Boiling Heat Transfer," ASME J. Heat Transfer, Vol. 131, No. 4, 043205.
- Kedzierski, M. A., 2002, "Use of Fluorescence to Measure the Lubricant Excess Surface Density During Pool Boiling," Int. J. Refrigeration, Vol. 25, pp. 1110–1122.
- Kedzierski, M. A., 2001a, "Use of Fluorescence to Measure the Lubricant Excess Surface Density During Pool Boiling," NISTIR 6727, U.S. Department of Commerce, Washington, D.C.
- Kedzierski, M. A., 2001b, "The Effect of Lubricant Concentration, Miscibility and Viscosity on R134a Pool Boiling" Int. J. Refrigeration, Vol. 24, No. 4., pp. 348–366.
- Kedzierski, M. A., 2000, "Enhancement of R123 Pool Boiling by the Addition of Hydrocarbons," Int. J. Refrigeration, Vol. 23, pp. 89–100.
- Kedzierski, M. A., 1995, "Calorimetric and Visual Measurements of R123 Pool Boiling on Four Enhanced Surfaces," NISTIR 5732, U.S. Department of Commerce, Washington.
- Liu, M. S., Lin, M.C.C., Liaw, J.S., Hu, R. Wang, C.C., 2009, "Performance Augmentation of a Water Chiller System Using Nanofluids," Proceedings of ASHRAE Winter Conference, Chicago, CH-09-058.
- Office of Science and Technology Policy (OSTP), 2008, "Federal Research and Development Agenda for Net-Zero Energy, High-Performance Green Buildings," National Science and Technology Council Committee on Technology, <http://www.ostp.gov/galleries/>
- NSTC%20Reports/FederalRDagendaforNetZeroEnergyHighPerformanceGreenBuildings.pdf (March 2009).
- Sarkas, H., 2009, Private Communications, Nanophase Technologies Corporation, Romeoville, IL.