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HEAT TRANSFER UNDER CONVECTIVE BOILING OF REFRIGERANTS R-404A AND R-407C IN A HORIZONTAL COPPER TUBE ELECTRICALLY HEATED

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ABSTRACT

Convective boiling of refrigerants R-404A and R-407C in a horizontal copper tube of 12.7 mm internal diameter has been experimentally investigated. A 2 m long test section electrically heated has been used for that purpose. Tests have been performed preferentially at the average test section temperature of 8°C. Effects over the heat transfer coefficient of such parameters as quality, mass velocity and heat flux have been considered. Kandlikar's correlation has been evaluated against experimental results for values of the fluid parameter obtained in this investigation.

INTRODUCTION

Convective boiling designates the liquid to vapor change of phase that takes place under forced flow of a heated fluid. In this study major attention has been focused on conditions prevailing in direct expansion evaporators, especially covering a region from the evaporator inlet to the section where dryout occurs. This is by large the most important region in the evaporator, and that where thermal equilibrium conditions prevail. Post dryout conditions encompass the misty two-phase flow and the superheated vapor regions. Regarding heat transfer, misty flow has been generally considered as a single phase saturated vapor region.

Nucleate boiling and evaporation at the liquid-vapor interface are the mechanisms associated to convective boiling heat transfer, occurring either isolated or simultaneously. The latter is designated as convective boiling by some authors, more restrictive than the one used in this study. Throughout the paper this condition will be designated as "strictly convective boiling", Saiz Jabardo et al (1999). Some of the current heat transfer correlations for convective boiling actually assume a superposition of those effects, after Chen (1966), see, for example, Jung et al (1989), Jung & Radermacher (1991), Gungor & Winterton (1986), Wattelet (1994), and others.

Heat transfer is affected by the flow regime, as one should expect, due to changes in the topology of the liquid/vapor interface, as suggested by Collier & Thome (1996). Three major physical parameters affect the flow regime transition in convective boiling: mass velocity, heat flux, and quality. For a given heat flux and reduced mass velocity, typically lower than 100 kg/(s m^2), stratified (wavy) regime occurs over the range of qualities encompassing those prevailing in a typical evaporator (say $x>5\%$), Kattan et al (1998a), Wattelet et al (1992). Nucleate boiling might occur on the wall in contact with the liquid filling the bottom of the horizontal tube, specially at lower qualities. As the liquid layer gradually turns thinner, bubble nucleation might be suppressed. The heat transfer coefficient is not significantly affected by quality and remains essentially constant up to the complete dryout of the wall, though, at very low mass velocities, it has been observed that it gradually diminishes as the liquid layer at the bottom of the tube evaporates.

Higher mass velocities prompt the transition from a bubbly and intermittent, mostly slug, nucleate boiling dominated regime, to the annular one. The heat transfer coefficient presents a typical behavior, being strongly dependent upon the heat flux in the nucleate boiling region. As the annular regime sets in, the heat transfer coefficient no longer depends upon the heat flux but increases gradually as the film thickness at the wall becomes thinner as a result of intense evaporation at the liquid/vapor interface, Jung et al (1989).

The heat transfer coefficient behavior along the evaporator described in previous paragraphs will be carefully cross examined in subsequent sections on the basis of experimental results obtained under convective boiling conditions of refrigerants R-404A and R-407C flowing in a horizontal copper tube electrically heated.

EXPERIMENTAL BENCH

An schematic circuit diagram of the experimental bench used in present study is shown in Fig. 1. The refrigerant is pumped from the condenser through a filter drier and a sight glass (SG) to the mass flow meter and preheater before reaching the entrance of the test section (TS). In order to allow for the flow development a 1.5 m length, 12.7 mm internal diameter copper tube is interposed between the preheater exit and the TS entrance. The results reported herein were obtained in a 2 m long, 12.7 mm internal diameter copper tube test section. The preheater and test section are heated by tape electrical resistors, uniformly wrapped on the external surface of the tube in such a way to guarantee a uniform heat flux. Refrigerant bulk temperature is measured at the TS inlet and outlet through type T sheathed thermocouples, whereas surface temperature is measured at four equally spaced cross sections along the TS by type T AWG#30 thermocouples. These thermocouples are nested in longitudinal grooves of a couple of centimeters long in such a way to reduce possible fin effects in temperature readings. At each measuring cross section, the surface temperature is read at three locations, 90° spaced, from the bottom to the top of the tube. The heaters are wrapped around the tube and covered by successive layers of fiber glass and foam thermal insulation. Details of installation of the test section electrical heaters and surface thermocouples are shown in Fig. 2.

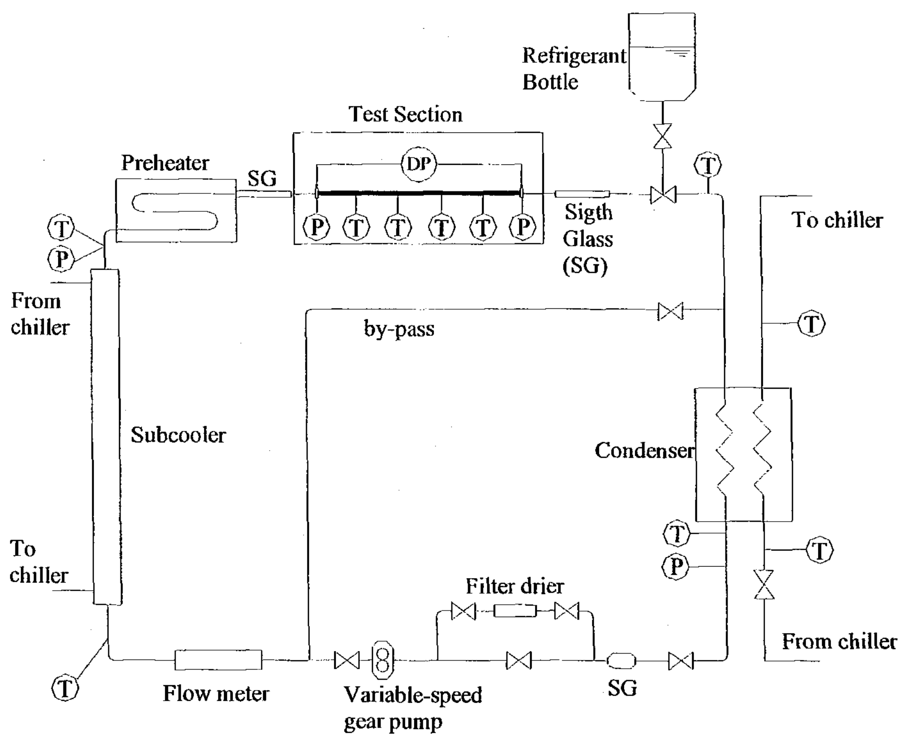


Figure 1. An schematic diagram of the experimental set up.

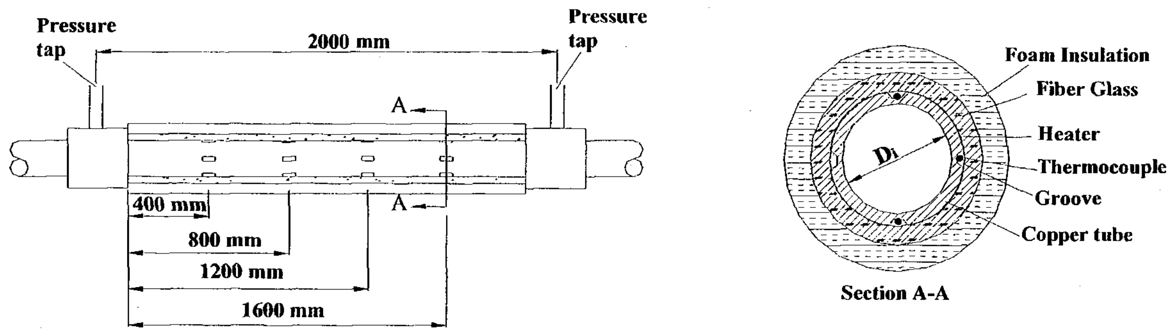


Figure 2. Detail of the test section.

The electrical output in the preheater and TS is manually adjusted by voltage converters. The quality of the refrigerant at the TS entrance is set by adequately adjusting the electrical output in the preheater. The refrigerant vapor produced in the preheater and TS is condensed in a shell and tube heat exchanger cooled by a 60% solution of ethylene glycol/water, as shown in Fig. 1. The solution is cooled by a chiller specially built for operation in the experimental set up.

Readings from the transducers are processed through a 12 bit resolution data acquisition system including terminal panels, for both temperature and electrical cable connections, and A/D boards. The system was provided with a software for monitoring and logging experimental data.

The average heat transfer coefficient in each measuring cross section along the TS was evaluated as

$$h = \frac{\phi}{T_w - T_f} \tag{1}$$

where T_w is the average surface temperature corresponding to the readings of the three surface thermocouples at the particular cross section, and T_f is the value that results from the interpolation of the bulk temperature of the refrigerant at the entrance and exit of the TS. The TS average heat transfer coefficient corresponds to the average of the ones associated to each of the four measuring cross sections. Refrigerant quality at the TS entrance and exit is evaluated through energy balances at the preheater and TS. The refrigerant at the preheater entrance must always be sub cooled so that its state is unambiguously known to allow for the enthalpy evaluation. For similar reasons, saturated equilibrium conditions must be warranted at the preheater exit for the quality evaluation.

Measured physical parameters in the reported experiments were: temperature, pressure, mass flow rate and electrical power. As already noted, temperature was measured by type T thermocouples, whereas pressure transducers were used to measure the inlet and exit pressures at the TS. Mass flow rate was measured with a Coriolis type meter, and electrical power with power transducers. The measuring accuracy of these parameters is shown in Table 1. The propagated uncertainty in the evaluation of the heat transfer coefficient was obtained according to Moffat (1988) and Albernethy & Thompson (1980), varying from 2.4% to 9.2%, for refrigerant R-404A, and from 1.88% to 12.1%, for R-407C, depending upon the particular experimental condition. It must be noted that the accuracy of the temperature measurement is the most influential parameter in the propagated uncertainty of the heat transfer coefficient.

Table 1. Measuring accuracy of physical parameters.

Parameter	Accuracy
Temperature	±0,2°C
Pressure	±0,3%
Mass flow rate	±0,15%
Electrical power	±0,5%

ANALYSIS OF RESULTS

During the experimental campaign reported in this paper, up to 600 data points were gathered involving both refrigerants. Data have been organized in such a way to parametrically investigate the effects of physical parameters that might significantly influence the heat transfer coefficient under convective boiling conditions, namely quality, mass velocity and heat flux. Effects of the mass velocity are summarized in Fig. 3, where the heat transfer coefficient is plotted against the quality, for three values of the mass velocity and a constant heat flux of 10 kW/m^2. The heat transfer coefficient increases with the mass velocity, though its effect is related to quite dissimilar flow regimes. In fact, whereas for the lower mass velocity (100 kg/s.m^2) the heat transfer coefficient tends to remain constant, for the higher values (200 and 300 kg/s.m^2), it presents a clear rising trend, specially for qualities larger than 30%. At low mass velocities, the stratified-wavy regime sets in at very low qualities, and the liquid at the bottom of the tube progressively dries with limited variations in the heat transfer coefficient. Increments in the flow rate prompt the transition to annular flow, characterized by an asymmetric liquid film covering the whole surface of the tube. As the film thickness diminishes with increasing qualities, the heat transfer coefficient experiences a progressive increment, shown in Fig. 3, associated to the reduction in the thermal resistance of the liquid film.

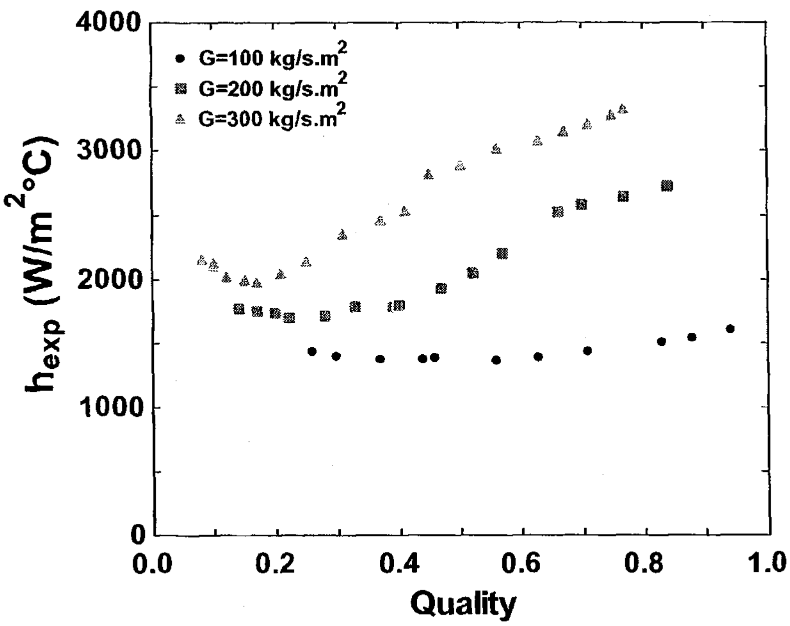
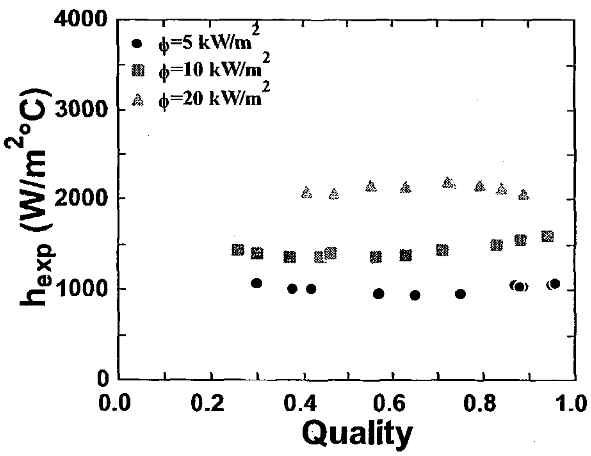
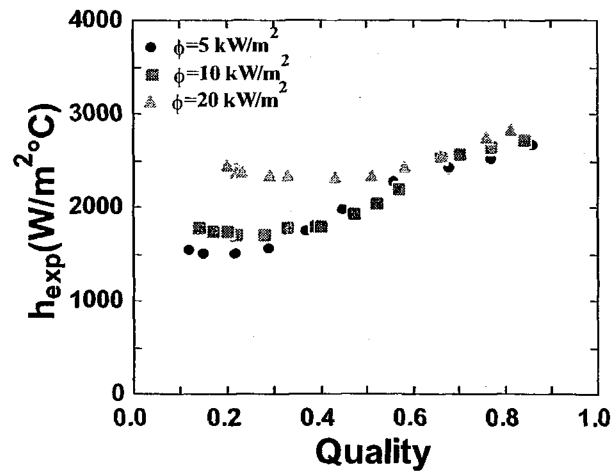


Figure 3. Variation of the heat transfer coefficient with quality for different mass velocities. Refrigerant R-407C; T_{avg}=8°C; ϕ=10 kW/m^2.

The heat flux affects the heat transfer coefficient in different manners depending upon the range of qualities and mass velocities. Figures 4a and b display the variation of the heat transfer coefficient with quality for different heat fluxes. Figure 4a depicts the typical behavior for the low range of mass velocities (100 kg/s m^2), whereas Fig. 4b displays results for a mass velocity of 200 kg/s.m^2. In the first case, the heat transfer coefficient increases with the heat flux, with possible occurrence of nucleate boiling over the whole range of qualities. In fact, through a sight glass at the exit of the TS bubbles were observed in the liquid at the bottom of the tube in experiments under stratified flow regime. The heat transfer coefficient variation with quality in the case of Fig. 4b is characterized by two regions. In the low quality region (x<50% roughly), it is affected by the heat flux, with nucleate boiling effects being a significant mechanism of heat transfer. Higher qualities are associated to the annular flow regime, with a reduced thickness liquid film attached to the tube surface The cooling effect of intensive evaporation at the liquid-vapor interface of the film inhibits the occurrence of nucleate boiling. Nucleate boiling suppression must be responsible for the observed trend at higher qualities in Fig. 4b, according to which the heat transfer coefficient is no longer dependent upon the heat flux (curves collapsing into a single one).



(a)



(b)

Figure 4. Variation of the heat transfer coefficient with quality for different heat fluxes. Refrigerant R-407C; $T_{avg}=8^{\circ}C$. (a) G=100 kg/s.m^2; (b) G=200 kg/s.m^2.

Figure 5 displays the comparison between results obtained for refrigerants R-404A and R-407C under similar operating conditions. Results are rather interesting. In fact, the R-404A heat transfer coefficient is higher over the whole range of qualities for the low mass velocity (100 kg/s.m^2), whereas for the the higher mass velocity (300 kg/s.m^2) the opposite happens, except for low qualities. This interesting behavior has been confirmed over all the operating conditions considered in this study. It is apparently related to the relative value of transport properties of these refrigerants. As noted before, for low mass velocities, stratified flow occurs over the range of qualities considered in this study. Under free surface conditions, the heat transfer coefficient depends upon the Froude number, Fr, increasing with it. Since Fr is inversely proportional to the density, the lower the density the higher the heat transfer coefficient will be, all other physical parameters being equal. Since the density of refrigerant R-404A is lower than that of refrigerant R-407C by a factor of 1.20, the relative value of the heat transfer coefficient must behave as in Fig. 5 for the low mass velocity. For higher values of the mass velocity, the annular flow regime sets in at higher qualities. Under this regime, the heat transfer coefficient is roughly proportional to the thermal resistance of the liquid film attached to the surface of the tube. In other words, it is proportional to the thermal conductivity of the liquid. Thus, the higher the conductivity the higher the heat transfer coefficient, all other physical parameters kept constant. This explains the inversion in the relative values of the heat transfer coefficient noted in Fig. 5 when passing from low to high values of the mass velocity, since thermal conductivity of refrigerant R-404A is lower than that of R-407C by a factor of 1.29. Finally, it must be noted that, for high values of the mass velocity and low quality range (lower than 30%), nucleate boiling effects are dominant. This effects are correlated by the Boiling number, which is inversely proportional to the latent heat of vaporization, for a given mass velocity and heat flux. Since the latent heat of refrigerant

R-404A is lower that that of R-407C (by a factor of 1.27), its Bo must be higher. Thus refrigerant R-404A heat transfer coefficient must be higher than the one associated to R-407C, as confirmed in Fig. 5 for low qualities.

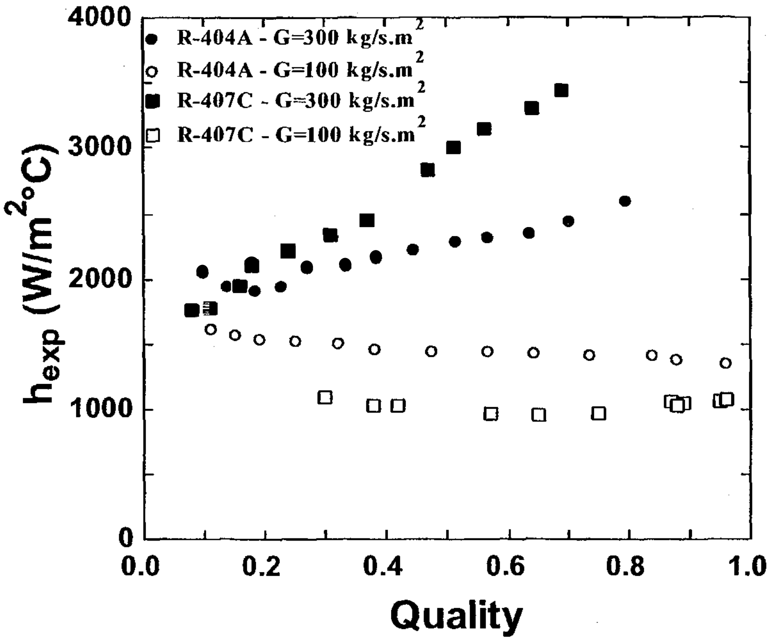


Figure 5. Comparison of results for refrigerants R-404A and R-407C for the indicated mass velocities. ϕ=5 kW/m^2; T_{avg}=8°C.

COMPARISON WITH CORRELATIONS

Experimental results obtained in this study have been used to asses the performance of heat transfer correlations developed elsewhere, but based upon data from other refrigerants. In this paper only the correlation proposed by Kandlikar (1990) will be considered, since it has been widely recognized as accurate and reliable for halocarbon refrigerants during the last decade. The current form of the correlation will not be presented to save space. Kandlikar's correlation requires the introduction of a fluid dependent numerical coefficient. Since this coefficient is not available for refrigerants R-404A and R-407C, it has been determined using the data from present research. By searching the fluid parameter that minimizes the average absolute deviation between correlation and experimental results, the following values were obtained: 1.55 for R-404A and 1.50 for R-407C. Figure 6 has been included as an example of the performance of Kandlikar's correlation fitted with present data. It can be noted that both regions are very well correlated: the low quality, where nucleate boiling effects are significant, and the high quality. The largest deviations occur at the highest qualities. However, these deviations are within the uncertainty range of the heat transfer coefficient.

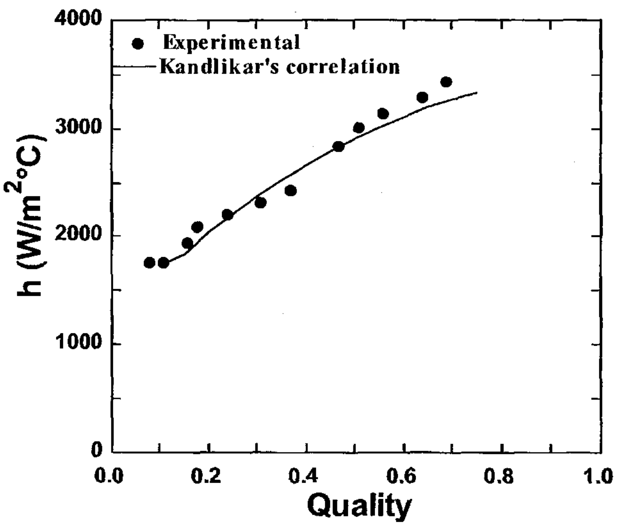
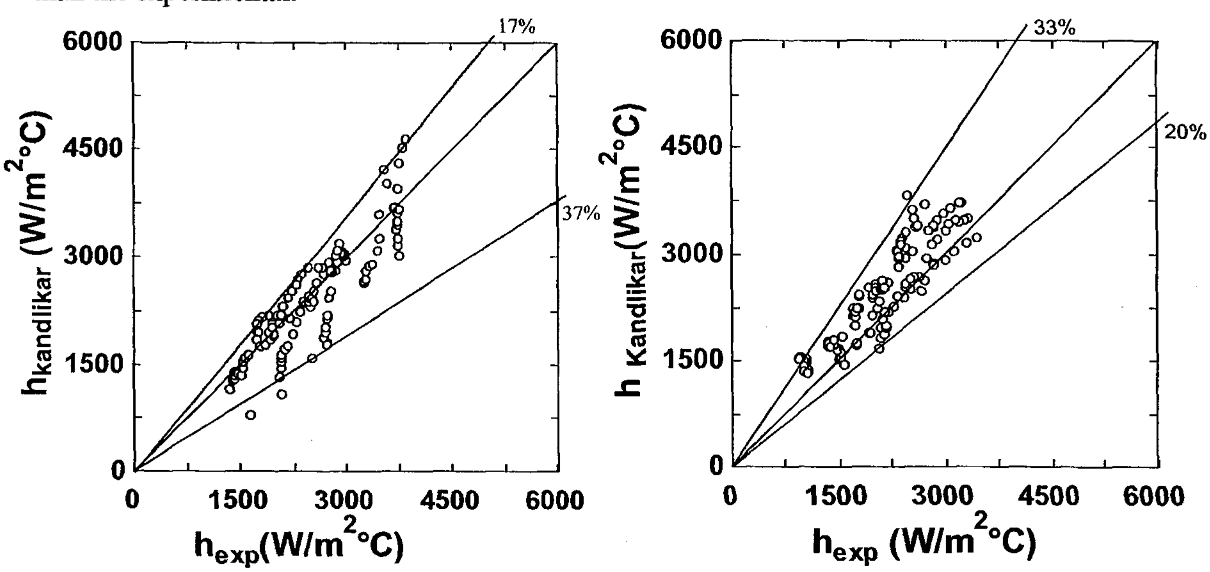


Figure 6. Heat transfer coefficient from Kandlikar's correlation against experimental results for refrigerants R-404A and R-407C. T_{avg}=8°C, G=300 kg/s.m^2; ϕ=5 kW/m^2.

Figures 7a and b display Kandlikar's heat transfer coefficient against the experimental for all the data points involving separately refrigerants R-404A and R-407C. Data dispersion is roughly within 20% and 33%, which can be deemed rather acceptable. Clearly, Kandlikar results tend to be higher than the experimental.



(a) (b)

Figure 7. Kandlikar heat transfer coefficient against the experimental for all the data points. (a) R-404A; (b) R-407C.

CONCLUSIONS

Experimental evaluation of the heat transfer coefficient of refrigerants R-404A and R-407C under convective boiling conditions in a horizontal copper tube has been performed. Results indicate that the relative value of the heat transfer coefficient for both refrigerants depends upon the range of qualities and mass velocities. In addition, it has been found that results are reasonably correlated by Kandlikar's correlation, for values of the fluid parameter obtained in this investigation.

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