Convective Evaporation on Plain Tube and Low-Fin Tube Banks Using R-123 and R-134a

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ABSTRACT

This experimental study investigates the convective evaporation heat transfer in a tube bundle. Heat transfer experiments were performed using R-134a and R-123 on a plain tube and a fin tube having 15.9 mm (0.626 in.) O.D. over a range of vapor qualities (0.03~0.34) with low mass velocities (8 to 40kg/m²s, or 1.6 to 8.2 lbm/ft²s) for a wide range of heat flux. Pool boiling data were also tested in the same apparatus. The fin tube having fins 0.6 mm (0.024 in.) high, with 0.6 mm (0.024 in.) fin pitch and 0.3 mm (0.012 in.) fin thickness, results in up to 170% boiling performance enhancement over the plain tube.

The present experimental data were compared with correlations using superposition and asymptotic models. The superposition-type correlation provided better prediction than the asymptotic model.

INTRODUCTION

Convective boiling on tube banks is an important heat transfer mode in a flooded evaporator. The study of convective evaporation on tube banks is an important foundation in the design of flooded evaporators in chillers. Casciaro and Thome (2001) and Browne and Bansal (1999) have surveyed the previous work on flooded evaporators, including experimental work and predictive models. Many researchers have tested the convective boiling heat transfer performance of tube banks in various conditions. For example, Cornwell and Scoones (1988), Jensen and Hsu (1987), and Webb and Chien (1994) have tested tube bank performance for R-113 and R-123. Some researchers have also provided correlations to predict the heat transfer performance in two-phase conditions. Gupte and Webb (1992) surveyed correlations for prediction of convective vaporization in tubes and tube banks. The phenomenological model results from combining the nucleate boiling and convective terms. In general, this may be written as

$$h = [(h_{nb})^n + (h_{cv})^n]^{1/n}, \tag{1}$$

which consists of nucleate boiling (h_{nb}) and convective (h_{cv}) contributions. Two main types of correlations have been used. If $n = 1$, Equation1 is called the "superposition model"; if $n > 1$, it is called the "asymptotic model." Chen (1966) proposed the superposition model and argued that the flow velocity suppresses nucleate boiling. Hence, he proposed that the nucleate boiling heat transfer in two-phase flow be calculated by $h_{nb} = Sh_{nbp}$, where h_{nbp} is nucleate pool boiling, and S is the suppression factor ($0 < S < 1$). It is assumed that h_{nbp} is the heat transfer coefficient for pool boiling on a single tube. The term h_{cv} in Equation 1 is the convective contribution. The h_{cv} is assumed to be the product of the "two-phase multiplier" (F) and the heat transfer coefficient for the liquid phase flowing alone h_l. To evaluate the F-factor experimentally, one calculates

$$F = \frac{(h^n - h_{nb}^n)^{1/n}}{h_l}, \tag{2}$$

where the total heat transfer coefficient in two-phase flow (h) and nucleate pool boiling coefficient (h_{nb}) can be tested separately. The exponent n in Equation 2 is the same in Equation 1. For example, $n = 1$ for the superposition model.

The prior work of Chen (1966) and Bennett and Chen (1980) used the superposition model ($n = 1$ in Equation 1) to correlate their data for flow in vertical, plain tubes. They proposed that the convective two-phase flow suppresses

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nucleate boiling, and they developed a correlation to account for the suppression factor (S). Steiner and Taborek (1992), Cornwell and Scoones (1988), Kenning and Cooper (1989), and Gupte and Webb (1992) have argued against use of a suppression factor. Steiner and Taborek (1992) correlated their vertical, smooth tube data using Equation 1 with $n = 3$ and $S = 1$. Webb and Chien (1994) evaluated the magnitude of possible nucleate boiling suppression by evaluating several data sets for refrigerants boiling in plain tube bundles. Webb and Chien (1994) found that the heat transfer coefficient of the convective boiling is lower than that of the pool boiling value (corrected for suppression) in some cases. Such behavior was not expected. They also found that the pool boiling performance for a single heated tube in a tube bundle is different from the test where the test cell contains only one tube.

For the asymptotic model, the suppression factor S is generally assumed to be one because the exponent n ($n > 1$) in the asymptotic model inherently accounts for suppression by inhibiting the smaller of the contributing components in the region between the asymptotic limits of h_{nb} and h_{cv}.

The present work provides more experimental data of tube bundles of plain and low-fin tubes using R-134a and R-123. R-123 is tested at 15°C and 30°C saturation temperature, and R-134a is tested at 10°C saturation temperature. The low-fin tube has 0.6 mm (0.024 in.) height, 0.3 mm (0.012 in.) thick fins, and 0.6 mm (0.024 in.) fin pitch. The asymptotic model and superposition model were compared with the present data.

EXPERIMENTAL DESIGN

Experimental Apparatus

A diagram of the test facility is shown in Figure 1. Refrigerant of known vapor quality enters from the bottom of the test section, where a bundle of tubes has a 15.87 mm (0.625 in.) outer diameter and 9.2 mm (0.362 in.) inner diameter with 23.85 mm (0.94 in.) tube spacing. The evaporated two-phase mixture enters a condenser, and then condensed liquid flows into a receiver. The condenser is cooled by glycol water, which was circulated between the condenser and a constant temperature bath. The glycol water tank is maintained at a constant temperature by an R-22 chiller. Following the receiver, a gear pump (Cole-Parmer variable flow drive P-75225-00 equipped with 07003-04 pump head) is used to circulate the refrigerant. A turbine flow meter (Cole-Parmer U32249-00: flow rate = 0.3 to 3.0 GPM, accuracy = 1% of reading) is connected after the pump to measure the flow rate. After passing through the flow meter, the liquid enters a pre-heater, where a given amount of heat is supplied. The maximum heating power of the preheater is 4.2 kW, and its heating power is controlled by a variac. The fluid temperatures before and after entering the preheater were measured for the calculation of the vapor quality at the inlet of the test section.

Figure 2 shows the details of the test cell made from 20 mm (0.78 in.) thick stainless steel plates. The rectangular internal space of the test cell is 150 mm (5.9 in.) high, 71 mm (2.8 in.) wide, and 120 mm (4.7 in.) long. A sight glass is made on one side of the test cell to observe the liquid level and boiling phenomena during the test. Tubes are soldered to the brass end flange to form a five-row staggered array in an equilateral triangular pitch of 23.85 mm (0.94 in.). An O-ring is inserted between the end flange and the test cell for sealing. All tubes are 120 mm (4.7 in.) long, and have a 15.87 mm (0.625 in.) outside diameter and 9.2 mm (0.362 in.) inner diameter. Except for the tubes in the top and bottom rows, 9.1 mm (0.358 in.) diameter 250 W cartridge heaters were inserted in these tubes. The total length of the cartridge heater is 100 mm (3.94 in.), but the actual heated region is 60 mm (2.36 in.) long, located in the middle of the heater. The heat flux of the test tube is calculated based on the actual heated length of the heater and outer diameter of the tube. The heat input of the cartridge

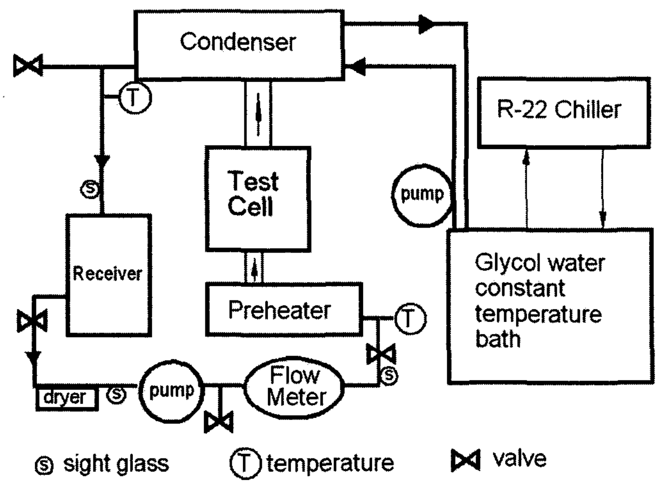

Figure 1 Diagram of tube bundle test facility.

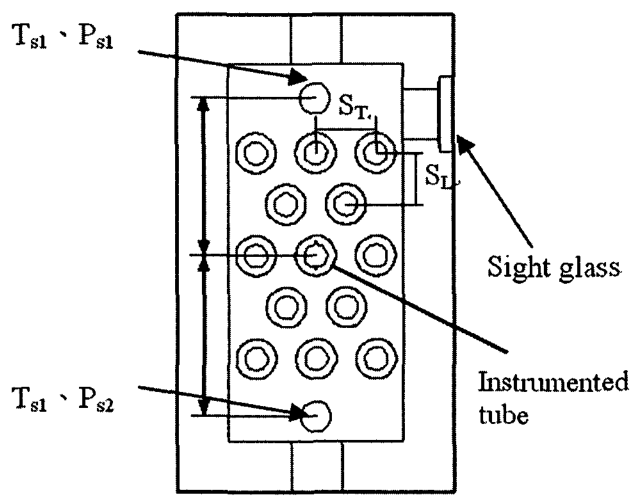

Figure 2 Cross section of tube bundle test cell.

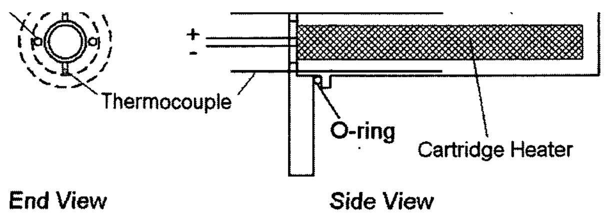

Figure 3 *Test tube instrumentation and heater assembly.*

heater in the test tube was controlled by a variable power transformer, and the current and voltage were measured to determine the heating power. Saturation pressure and temperature are measured at the top (P_{s1}, T_{s1}) and bottom (P_{s2}, T_{s2}) of the tube bundle, and the saturation temperature is calculated at the test tube location by interpolation between the saturation temperature at the top and bottom (T_{s1} and T_{s2}). During the tests, the difference in the saturation temperatures at the top and bottom is less than 0.5°C (0.9°F). The Cole-Parmer 68001-24 pressure transducers (pressure range 0 to 100 psig; accuracy 0.028% of 100 psig) were used for the R-134a tests, and the Cole Parmer 07356-50 pressure transducers (pressure range 0 to 30 in.-Hg; accuracy 0.04% of 30 in.-Hg) were used for R-123 tests. The mass velocity (G) is calculated based on the minimum flow area in the tube bundle. Heat is supplied through 250 W cartridge heaters inserted in all tubes in the test cell except the top and bottom rows. The function of unheated tubes is to mix the inlet flow from the preheater. The instrumented tube is located at the center of the third row from the bottom.

Figure 3 shows the end view and the side view of the instrumented tube. The instrumented tube is fixed on the end flange with two 3.0 mm (0.118 in.) thick screws. An O-ring is placed between the tube and the end flange for sealing. The tube is designed with the precise length to fit in the distance between two flanges, and the location of the O-ring is carefully designed to ensure proper compression of the O-ring. Two diametrically opposite axial grooves of 0.6 mm are made along two opposite sides of the test tube. The center of the groove is 0.6 mm (0.024 in.) away from the tube wall of the plain tube or 0.6 mm (0.024 in.) from the root of the fins on the fin-tube. T-type thermocouples of 0.5 mm (0.02 in.) sheath diameter are inserted in the grooves for tube wall temperature measurement. The one-dimensional steady-state heat conduction equation in cylindrical coordinates is used to correct for the conduction temperature drop between the thermocouple and the boiling surface. The grooves are located at the top and bottom positions in the tube wall. A heat sink compound is used to ensure good thermal contact of heaters and thermocouples with the tube wall.

Data Reduction and Experimental Uncertainty

The temperature and pressure of the refrigerant entering the preheater were measured to determine the thermodynamic state of the subcooled refrigerant. Knowing the heat input to the preheater, the mass flow rate, and the thermodynamic properties, the thermodynamic state of refrigerant leaving the preheater is calculated assuming an isobaric process. By knowing the pressure at the bottom of the tube bundle, the vapor quality can be calculated. The vapor quality in the test is calculated at the center position of the test tube from the heat input up to that point.

All instruments are connected to an Agilent 34970A data acquisition system and transmit data through RS-232 interface to a personal computer. The thermocouples were frequently calibrated in a constant temperature bath and are repeatable within ±0.1°C (0.18°F). Based on the instrumentation accuracy of the heat flux, saturation pressures, and wall temperature, an error analysis shows that heat transfer coefficients are calculated within 2.5% for the maximum heat flux condition and within 7.1% for the minimum heat flux condition for R-134a. Similarly, the uncertainty is within 1.1% for the maximum heat flux condition and within 2.5% for the minimum heat flux condition for R-123.

EXPERIMENTAL PROCEDURE

Convective vaporization of a tube bundle and pool boiling data of a heated tube in a tube bundle were tested in R-134a and R-123 boiling on a plain tube and a fin tube having 0.6 mm fin height, 0.3 mm fin thickness, and 0.6 mm fin pitch. Data were taken for three saturation temperatures: 10°C (50°F), 15°C (59°F), and 30°C (86°F) for various mass velocities, heat fluxes, and vapor qualities at the test section. The heat flux was varied decreasingly to avoid boiling hysterysis. The instrumented tubes were cleaned with acetone and pure water before each test. Then they were immersed in de-ionized water for ten minutes and then dried by blowing air. The pool boiling test was performed according to the following procedures:

- A leak-tight test is performed before charging. The system was considered leak-tight if the system was able to hold vacuum (less than 5 kPa) for at least 24 hours.
- Evacuate the system for at least twenty minutes by a vacuum pump.
- Charge the working fluid into the system. About 22 kg of refrigerant is needed.
- Turn the heater to the maximum output (about 250 W). Keep the pool temperature at the desired saturation temperature by adjusting the cooling water chiller system and preheater and maintain for at least an hour.
- Then, evacuate the system for some three seconds to remove the noncondensable gases.
- Keep the system boiling at the maximum heat flux for at least thirty minutes, and adjust the glycol water system and preheater heat input to keep the saturation temperature at the desired value.