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Technical Note

Pool boiling of pure R134a from a single Turbo-BII-HP tube

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1. Background

In the last twenty years, great effort has been expended to improve refrigerantside heat transfer coefficients in commercial flooded evaporators. The improvements have come from enhanced tube surfaces; smooth and integral-finned tubing have been superceded by structured (re-entrant cavities) and porous surfaces. New refrigerants have also necessitated updated heat transfer studies.

Refrigerantside (outside) coefficients may be available from several sources: actual flooded evaporators, tube bundle segments, and pool boiling tests. Single tube pool (nucleate) boiling tests represent the conventional method. For refrigerant R114, coefficients have been measured with Turbo-B, Thermoexcel-E, and Thermoexcel-HE single tubes [1–3]. Similarly, for R124, data are available for the Turbo-B tube type [4]. Also, a Gewa-SE and a Turbo-B tube were subjected to R11, R12, R22, R123 and R134a under nucleate pool boiling conditions [5,6]. (Some of the cited studies also presented the effects of oil upon the heat transfer.) Although numerous studies exist for the determination of refrigerantside heat transfer coefficients for the standard Turbo-B tube, results are meager using R134a and non-existent, in the open literature, for the Turbo-BII-HP tube using R134a.

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2. Test tube and pool boiler construction

The experimental setup is displayed, in schematic form, in Fig. 1. It consists of the pool boiler adapted from a commercial filter-drier with its core removed. The nominal diameter is 121 mm and the overall length is 381 mm. The vapor refrigerant generated is condensed by a copper cooling coil made from nominal 9.53 mm copper tubing. The cooling medium is a 40% (by weight) ethylene glycol/water solution that is a side-stream from a nominal ten-ton, air-cooled external chiller.

The copper test tube is the Turbo-BII-HP, which is an advanced version of the conventional Turbo-B, and its re-entrant cavities are optimized for higher pressure refrigerants such as R134a. The tube nominal outer diameter is 19.05 mm and the length is 211.51 mm. The tube is fitted with a 1000 W cartridge heater having an active length of 203.20 mm; this type of heater is custom-designed for this application and identical heaters have been cut apart to verify that the resistance elements span the specified active length. The annular space between the heater and tube wall was filled with solder to insure good uniform contact. A variac and wattmeter are connected to this heating element.

For tube temperature readings, two thermocouple junctions are located at a distance of 38.10 mm from one end and centered in the tube wall. In this way, the thermocouple junctions rest well within the zone of heating. To accomplish this, the inside diameter of the tube was increased by milling out the wall up to 38.10 mm from one end. From another tube, a sleeve was formed to exactly replace the material milled away in

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the other one. Two square channels, 0.635 mm in width, were cut into the outside of the sleeve along its entire length. Following freeze-treating to shrink the sleeve, it was press-fitted into the test tube, completing the formation of a test tube with its two slots to accommodate thermocouples. The channels were located at the top dead center and bottom dead center around the circumference of the tube. Copper–constantan thermocouples, that measure 0.508 mm in diameter (including the stainless steel sheath), were inserted into each channel in the test tube. Two thermocouples in the bulk fluid are immediately adjacent to the test tube (Fig. 1). These thermocouple junctions are also fixed 38.10 mm from the end of the tube so that the bulk thermocouples have the same axial position as the tube wall ones.

3. Data collection and analysis

Heat transfer data are obtained from the thermocouple and wattmeter readings. A preheat is employed to eliminate any hysteresis effects. For this, the heater is set to 37,004 W/m^2 for 5 min. Note that heat flux is based on outside area derived from a plain, nominal 19.05 mm tube. Then the heat flux is slowly reduced to the desired level for data acquisition. A saturation of approximately 4.4°C was set for each run, although it could deviate slightly from this target between different runs since it is indirectly controlled by the external chiller's operating temperature.

Typically, about 20 min are required to establish steady-state. When all the settings are stable and steady-state is reached, data are taken by the data acquisition system and stored in the computer. All settings are held constant until the computer receives 20 continuous data points. Thus, each temperature is based upon an average of its 20 readings. The procedure is repeated for each heat flux, in descending order.

Once the thermocouple data are obtained, the refrigerantside heat transfer coefficient can be found from

$$h_o = q_o/(T_o - T_b) \quad (1)$$

where the heat flux is based on nominal 19.05 mm outside diameter and an active heating length of 203.20 mm. T_o and T_b represent the tube surface and bulk fluid temperatures, respectively. Since the tube thermocouples are located within the tube wall, T_o is not directly known. Instead, the measured tube wall temperature was extrapolated to find the outside surface temperature by utilizing an analytical expression for conductive heat transfer in the radial direction. The outside surface temperature is defined to exist at the actual fin-root diameter which was measured after machining away the finning; for the present tube batch, this diameter is 18.42 mm. For this type of tube, the average surface temperature can be taken as

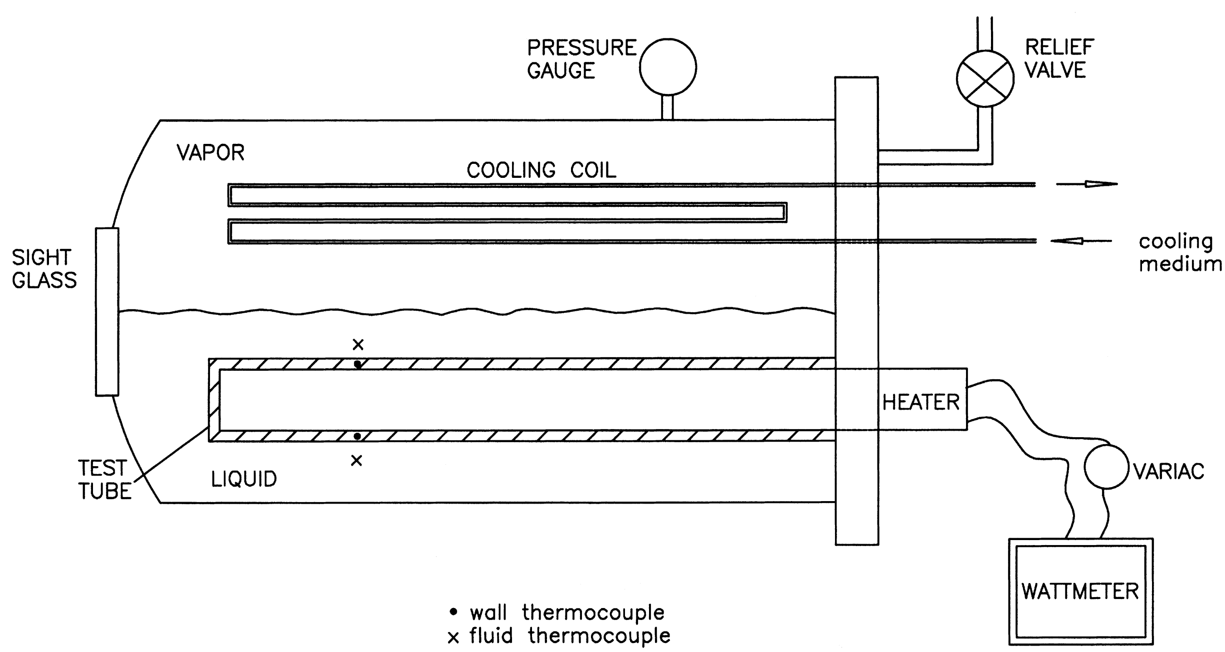

Fig. 1. Side-view of R134a pool boiler.

the extrapolated arithmetic mean of the upper and lower tube wall readings since this closely represents the average circumferential temperature [7]. The experimental heat transfer coefficients are presented based on the local tube bulk fluid temperature which is the average of the bulk temperature just above and below the tube.

Fig. 2 displays the heat transfer coefficient, h_o, plotted against the heat flux. The coefficients range from 19,294 to 28,117 W/m^2 °C and increase with heat flux. The corresponding saturation temperatures were 4.6, 5.4, 5.2, 4.2, 4.1 and 5.1°C at heat fluxes of 8,224, 12,335, 16,445, 24,669, 32,893 and 41,114 W/m^2, respectively. Displayed in Fig. 2 are error bars based on the uncertainty in measuring h_o and q_o. Uncertainty in heat flux is $\pm 1\%$ of setting. The thermocouples were calibrated with a precision platinum resistance thermometer and have an uncertainty estimated at ± 0.04°C. Not considering the effects of averaging readings for tube wall and bulk fluid temperatures, this implies a worst-case temperature difference error of ± 0.08°C. The temperature difference uncertainty is magnified at low heat fluxes where $T_o - T_b$ can be near 0.6°C for such an enhanced tube. The error bars indicate a deviation ranging from ± 20 to $\pm 6\%$ for increasing heat flux (increasing temperature difference).

The present coefficients are much greater than previously measured with the standard Turbo-B tubing.

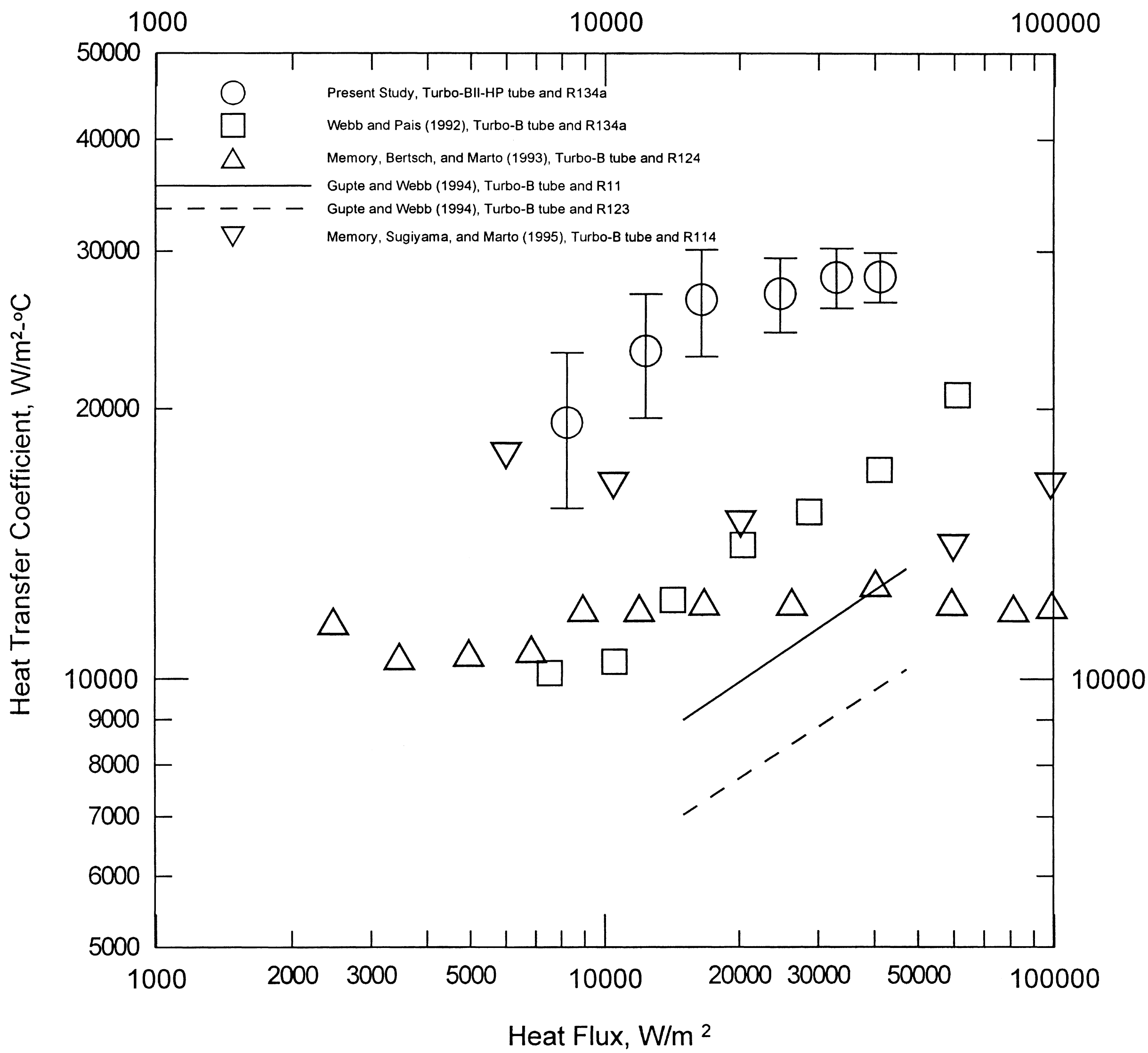

Fig. 2. Pool boiling of Turbo-B type single tubes.

For instance, the current values are 60–90% larger than those measured in R134a with the Turbo-B surface [5]; the difference decreases as the heat flux increases. At elevated heat duties, the flat region of the curve suggests that liquid cannot replenish the boiling cavities quickly enough. Also, the present coefficients are about 60% higher (at lower q_o) to more than double the performance (at higher q_o) of the Turbo-B with R124 [4]. With R114, the current values of h_o are slightly better at low heat fluxes and are twice greater at higher fluxes [3]. R11 and R123 heat transfer coefficients [6] are significantly lower compared to all the studies. All data were taken at 4.4°C except for the R114 and R124 values which used 2.2°C as the saturation temperature. It should be kept in mind that the tube type and/or refrigerants are varied from the present effort and any direct comparison is difficult. In all cases, the present data show the superior performance of the Turbo-BII-HP surface in the pool boiling mode.

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