Vapor Condensation of Iso-Butane over Plain Tube and 24 fpi finned tube placed horizontally

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Abstract

In this study, vapor condensation heat transfer rate of iso-butane over plain tube and 24 fpi finned tube placed horizontally was determined experimentally and results were compared to get optimum heat transfer coefficient. Condensation heat transfer coefficients of R-600a showed the trend that external condensation HTCs decrease as wall sub cooling temperatures increase. All data were taken under saturation temperatures of 39 ^{0}C with wall sub-cooling of 5-12 ^{0}C on a tube of 19 mm outside diameter under a heat flux of 11-55 kW/m^2. Based upon the Data taken in this study, different graphs are plotted varying different parameters to show their dependency on other parameters. The experimental data are validated by comparing them against the standard model for condensation.

***Keywords*:** vapor condensation, iso-butane, horizontal tube, external heat transfer coefficient;

1. Introduction

The concept of integral fin-tubes for the enhancement of heat transfer rate was developed in the late 1940's. The first idea to introduce integral fin-tubes was to increase the heat transfer area but later on, many models like gravity drainage mechanism, surface tension drainage mechanism, and condensate retention model due to surface tension came into existence. The first effort to predict the heat transfer rate for outside condensation of a horizontal integral fin-tube was carried out by Beatty and Katz [1]. They proposed film condensation and gravity drainage mechanism and derived a correlation to predict heat transfer coefficients (HTCs) for the condensation of horizontal integral finned tubes by reconstituting the Nusselt's correlations for film condensation over horizontal tube and vertical plate together resembling the region of inter-fin spacing of integral fin tubes and fin-flank respectively. Their correlation was based upon the assumption that the condensate is not retained in between the fins and the surface force can be neglected with only the gravitational force considered for draining out the condensate. Canvas [2] conducted experiments using CFC11 as the working fluid for different fin densities of low integral fin tubes and determined the optimum fin density for the condensation of CFC11 over fin tubes. Rudy and Webb [3] distributed the whole fin surface and interfin spacing into two regions: unflooded and flooded. They assumed that heat transfer occurred only by the unflooded region and neglected the heat transfer by the flooded region. Their model predicted most of the test data for water, n-pentane, and R-11 on 19 mm dia integral-fin tubes within ± 10 percent. Owen et al. [4] included the flooded region into consideration for its contribution in heat transfer and improved Beatty and Katz model which explicitly accounted for flooded region as well. The extended model predicted the mean condensation heat transfer coefficient of methyl chloride, sulphur dioxide, propane, n-butane, n-pentane, acetone, water, R-11, R-12 and R-22 and found all data within ± 30 percent of the values predicted by the model. Previous all models were based upon the gravity drainage mechanism of condensate and were the extension models of Beatty and Katz model but Webb, Rudy and Kedzierski [5] proposed surface tension drainage mechanism on the fin surface and gravity drainage mechanism in the interfin spacing. This model predicted the R-11 condensation coefficient on horizontal, integral-fin tubes having 748, 1024, and 1378 fpm and the predicted values were within ± 20 percent of the experimental values. Yau et al. [6] performed experiments with low fin tubes of various fin heights and fin spacings using R-12 as a working fluid. Wanniarachchi et al. [7] measured HTCs of outside condensation of steam over 24 finned tubes of different geometry and found best fin spacing between 1.5 to 2.0 mm. Masuda and Rose [8] found 0.5 mm as optimum fin spacing for R-113 and for this tube, enhancement ratio was 7.3. Honda and Nozu [9] presented a method for predicting the average heat transfer coefficient for film condensation on horizontal low integral-fin tubes. Their model predicted most of the available experimental data including 11 fluids and 22 tubes within ± 20 percent. The objective of this paper is to compare the heat transfer characteristics of vapor condensation of Iso-butane (R-600a) over plain tube and commercially important 24 fpi tube and to check the feasibility of 24 fpi tube.

2. Experimental Facility

The schematic diagram of the set-up is illustrated in Fig. 1, which comprised of major components like condenser, evaporator and data collection unit. The test condenser (5) is a stainless steel-304 cylinder of thickness 3 mm, inside diameter 100 mm and length 414 mm. The vapor of refrigerant is supplied through a dead end pipe (6) of 350 mm length and 6.5 mm diameter. This pipe has equally spaced 125 holes, each of 1 mm diameter, in a straight line; holes facing upwards so that vapor coming out of it does not strike directly to the test section. The test-section (3) (horizontal smooth copper tube) is fixed inside the test- condenser with the help of chuck nut assembly. The refrigerant-vapor generated in the evaporator (10) condenses over the test-section releasing latent heat of vaporization to the coolant (water), flowing inside the test-section. Thus the temperature of the water rises, which is recorded with the help of data collection unit. The viewing window is provided in the middle of the test-condenser fitted with the Teflon glass to observe condensation. The vapor of liquid refrigerant is generated in the evaporator. The evaporator is a stainless steel -304 cylinder of thickness 3 mm, inside diameter 140 mm and length 670 mm. Three immersion heaters (12) each of 3 KW heating capacity are fixed in the bottom of the evaporator to transfer the heat directly to the liquid refrigerant. The cylinder is closed by a flange (13), sealed with a 12 mm diameter hardened rubber O-ring and 8 holes of 10 mm hole diameter. At the top of the evaporator, a port is provided to supply the generated vapor to the test-condenser. Also at the top of the evaporator, another port is provided to monitor the evaporator pressure with a pressure gauge and pressure transducer. In the bottom part of the evaporator, a port is provided to feedback the condensate to the evaporator. The auxiliary condenser (2) is connected to the test-condenser. The auxiliary condenser is a stainless steel -304 cylinder of thickness 3 mm, inside diameter 120 mm and length 140 mm.

3. Instrumentation and measurement

The data collection unit comprises three types of measuring units: a.) temperature measuring unit which consists of thermocouples, data acquisition system (Adam-4000 series) and laptop b.) Pressure measuring unit consists of pressure gauge and pressure transducer c.) Coolant flow measuring unit which consists of turbine type flow meters and display units. Temperature measurement includes the measurement of the condensing vapor temperature, surface temperatures of the test section and the temperature rise of the cooling water. J-type thermocouples of 20 gauge thickness and an 8 channel data acquisition module (ADAM-4019) are used to fulfil this requirement. The pressure of condensing vapor is measured with the help of a dial type pressure gauge and pressure transducers. The cooling water flow rate is measured with the help of turbine type flow meters.

4. Experimental procedure and data reduction

4.1 Experimental procedure

To ensure leak proof experimental set-up, a pneumatic gauge pressure of 2.0 MPa was maintained inside the set-up for 24 hours, followed by a vacuum of 76 cm of Hg for same duration. No pressure drop inside the set-up ensured that the set-up was leak-proof. Assuring the leak proof of the set-up, the refrigerant was charged in the set-up under vacuum. While charging the refrigerant, the required purging was done via purging pipe (1) to escape air and other non-condensable gases from the set-up. After charging, the cooling water was circulated inside the test-condenser and the auxiliary condenser by centrifugal pumps (8) and (11) respectively. The refrigerant in the evaporator was heated with the electric immersion heaters (12). The vapor of refrigerant thus generated rushed to the test-condenser (5) and passing through the perforated pipe (6), it was subsequently condensed by coming in contact with the cold surface of test-section (3). The condensate was then sent back to the evaporator and thus the refrigerant flow cycle was completed. The rate of vapor generation was controlled by a Variac. The pressure of condensing vapor was maintained constant throughout the investigation by regulating the energy supply to immersion heaters in the evaporator and cooling water flow rate in the auxiliary condenser.

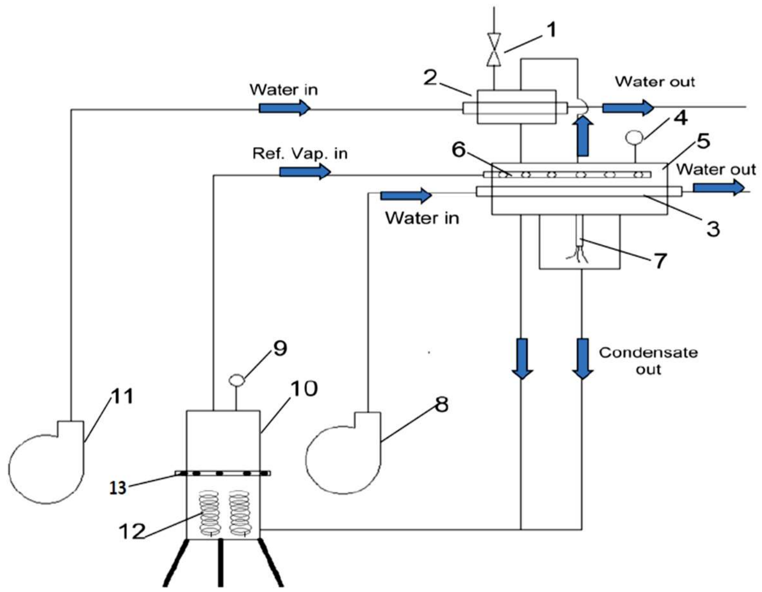

Fig. 1. Schematic diagram of the experimental set-up

1-purging pipe 2-auxiliary condenser 3-test-section 4,9-pressure gauge & pressure transducer 5-test- condenser 6-perforated pipe 7-thermocouple exit port 8,11-centrifugal pump 10-evaporator 12-heaters 13-flange

For each test run, data were collected only when the set-up attained a steady state condition. The steady state condition occurred when the rate of vapor generation became equal to the condensation rate, consequently the pressure inside the set-up became stable and so, the temperature of condensing vapor. For each test run, the temperature of condensing vapor, the surface temperature of the test-section, the temperature of cooling water at the inlet and outlet of the test-section and the cooling water flow rates were recorded.

4.2 Data Reduction

The condensing-side heat transfer coefficient, h, was determined by taking the ratio of heat flux, q”, and the wall sub-cooling temperature, ΔT_f:

$$ho = q'' / \Delta T_f \quad (1)$$

$$\Delta T_f = T_s - T_w \quad (2)$$

Where T_s and T_w are refrigerant vapor temperature and tube wall temperature respectively.

The heat flux, q”, was calculated by taking the ratio of heat transfer rate, q, and the outer surface area of the plain tube, Ao:

$$q'' = q / Ao \quad (3)$$

The heat transfer rate, q, was determined by calculating the heat carried away by the cooling water from the test-condenser, which is measured by a simple energy balance equation as follows:

$$q = \dot{m}_{cw}\, Cp_{cw} \,(\, T_{cwo} - T_{cwi}\,) \quad (4)$$

Where $\dot{m}_{cw}$, Cp_{cw}, T_{cwo}, T_{cwi} are the mass flow rate, specific heat, and temperatures of the cooling water at the outlet and inlet of the tube, respectively.

5. Results and Discussion

Fig. 2 shows that heat flux (q”) increases as wall sub-cooling temperature increases for both plain and 24 fpi tubes. The possible reason can be easily explained with the help of Equation 1. From Fig. 3, it is clear that heat transfer coefficients (HTCs) decrease as wall sub-cooling temperatures increase for both plain and 24 fpi tubes. With increase in ΔT_f, the heat flux increases, resulting in an increased rate of vapor condensation on the tube surface which brings a thicker condensate layer on the tube surface. The condensate layer offers higher thermal resistance decreasing the heat transfer coefficient. Fig. 4 shows enhancement factor (EF) of 24 fpi tube for different wall sub-cooling temperatures. The enhancement factor (EF) is defined as the ratio of the heat transfer coefficient of finned tube to that for a plain tube at the same value of wall sub cooling temperature.

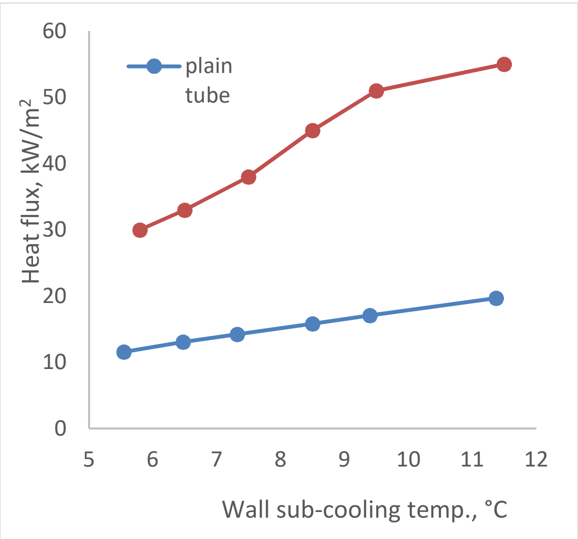

Fig. 2. Variation in heat flux with wall sub-cooling temperature

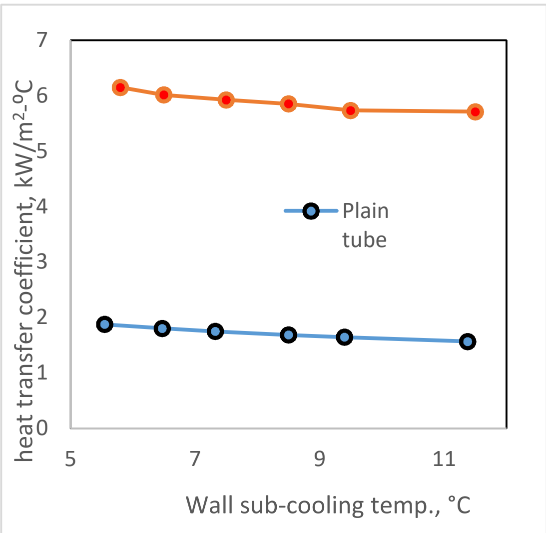

Fig. 3. Variation in heat transfer coefficient with wall sub-cooling temperature

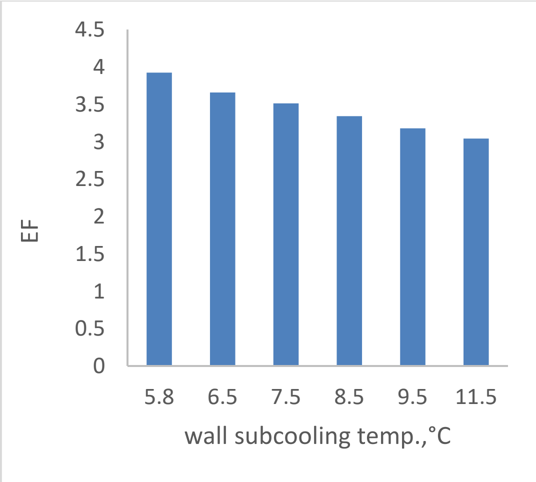

Fig. 4. Variation in enhancement factor with wall sub-cooling temperature

From Fig. 4, it is clear that heat transfer rate of 24 fpi tube is increased 3 to 4 times in comparison to plain tube. The reason behind this is that fins not only increases the surface area but also promote heat transfer characteristics by surface drainage mechanism.

6. Conclusion

1 Heat flux increases as wall sub-cooling temperatures increase for both plain and 24 fpi tubes
2 Heat transfer coefficients decrease as wall sub-cooling temperatures increase for both plain and 24 fpi tubes.
3 Heat transfer rate of 24 fpi tube is increased 3 to 4 times in comparison to plain tube.

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