Experimental Investigation for the Condensation Heat Transfer of R-245Fa over Horizontal Plain Tube

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Abstract
An experimental investigation has been carried out for the condensation of R-245Fa vapour on horizontal plain copper tube. The experimental data have been acquired at the saturation temperature of 318.1 ±0.5 K. The condensation of R-245Fa refrigerant under examination consideration predicts the heat transfer co-efficient for the Nusselt's model by 19% and for the condensation of R-245Fa over a plain tube, the heat transfer co-efficient reduces with the rise in ΔT_f.

Key words: Condensation, Heat Transfer Coefficient, Refrigerants

1. Introduction

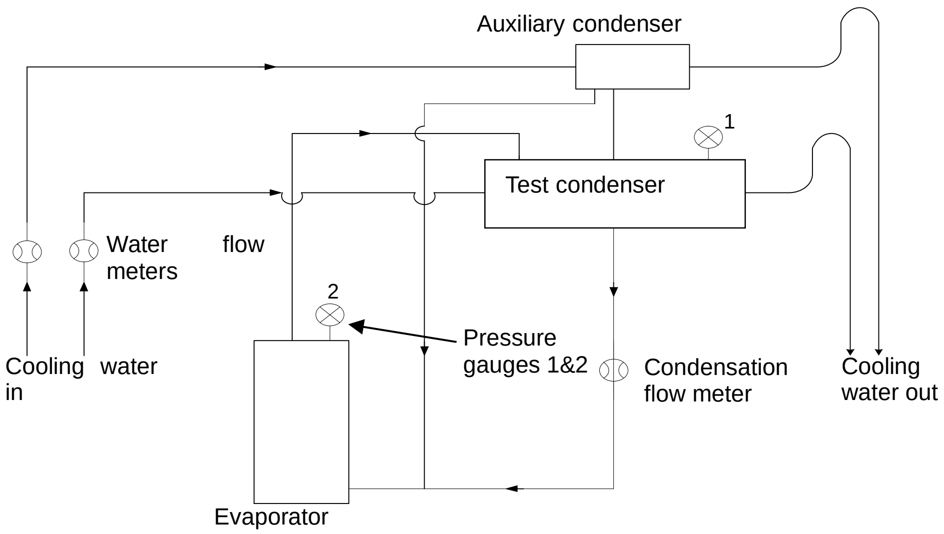
The enhancement of heat transfer has concerned the researchers and scientists for many years. It was discovered that chlorofluorocarbons (CFC's) such as R-11, R-114, R-113 etc. and hydrocholorofluorocarbons (HCFC's) viz. R-22 cause damage to the ozone layer of the earth's atmosphere and produce green house effect. In fact, life on earth depends upon a thin shell of gaseous ozone surrounding the earth, which stretches from 10-25 kilometers above the earth surface. Thus, these refrigerants have to be phased out completely in the coming years owing to their ozone depletion potential (ODP) and the global warming potential (GWP). Now, R-245Fa is a substitute for CFC-11 (R-11) in the field of air-conditioning, especially in low pressure water chillers typically used in large building air conditioning. The current lack of consistent data on thermophysical properties of R-245Fa is one impediment of wider use of this ozone friendly refrigerant.

In order to study heat transfer rate during condensation of pure R-245Fa vapor over single horizontal plain tubes, an experimental set-up has been designed and fabricated to carry out investigations. Being a relatively new refrigerant, the thermal behavior of R-245Fa is not known in some applications. Therefore, experiments were conducted for the condensation of R-245Fa over a horizontal plain tube and data was acquired for the condensation over same the horizontal plain tube. All the experimental data was acquired at nearly constant saturation temperature of condensing vapor. This is in agreement with the predicted heat transfer rate using Nusselt's model.

2. Experimental Setup

The experimental investigation was carried out for the condensation of R-245Fa vapor on an experimental setup illustrated in Figure 1. The vapor of R-245Fa was generated in an evaporator using electric immersion heaters. Three immersion heaters were mounted in the evaporator. A refrigerant level indicator was fitted to the evaporator to monitor the full submergence of immersion heaters in the pool of liquid refrigerant. The evaporator was connected to a test condenser. In the test condenser the vapor of R-245Fa was condensed on the outer surface of a single horizontal tube. The heat of the condensing vapor was carried away by the cooling water flowing inside the test section. The excess vapor was condensed in an auxiliary condenser connected to the test condenser. A purge valve was provided with the auxiliary condenser to remove the non condensable (i.e., air) from the system. The condensate formed in the test condenser and the auxiliary condenser returned to the evaporator. In the condensate return line, a glass tube flow meter was connected to visualize and measure the condensation flow. The refrigerant was charged in the setup with the help of a refrigerant cylinder. Inside the test condenser, the test section of 416 mm length was fixed with the help of chuck nuts. The rubber packing rings were used to compose a leak-proof fixing of the test section and to ensure further safety a lead Rope packing was fixed by a chuck nut .The condensation phenomenon inside the test condenser was observed through a viewing window.

Figure 1 Layout of experimental set up



Keeping in mind the compactness of the test condenser, a blind-ended parallel tube of 6.5 mm diameter, parallel and 35 mm above the test section, was connected to the vapor inlet port. This tube had 125 holes of 1.0 mm diameter along the tube length, facing the roof of the test condenser so that the vapor coming out from these holes struck the roof of the test condenser. The vapor lost its initial momentum and was distributed uniformly throughout the length of the test section. Municipal potable water was used as the cooling water and no scaling was observed inside the test section after its removal from the test condenser. As illustrated in Figure 1, the cooling water from a cooling tower was circulated inside the test section and the auxiliary condenser was metered separately using two rotameters The hydrodynamic stabilization of cooling water flow was achieved and an inverted U-bend after the downstream calming section was provided to ensure that the test section was fully filled with cooling water for all the flow rates. The hot water flowing out from the test condenser and auxiliary condenser was sent back to the cooling tower. The test section surface temperature was measured by J-type thermocouples. The thermocouples were fixed in the predetermined position (i.e., top, side, and bottom), in a vertical plane, on the tube wall surface at half of the tube length. The installation of thermocouples through the test-section wall is a difficult job on a tube of 417 mm length. In fact, a very high level of workmanship is required for this job. Therefore, the thermocouples were fixed on the test section surface from the vapor side only, as was also done by Kumar et al. (1998) [5]. Prior to installation of thermocouples, small holes of depth 1 mm were drilled on the tube surface and thermocouples were grounded on it. The bead was kept just below the surface of the test section. However, the fact remains that in spite of all precautions, the condensate drainage pattern at that particular location was disturbed due to fixation of the thermocouples on the tube surface. These locations constitute less than 0.1%of test-section surface area and, hence, hardly affect the performance of the test-section tube. The measurement of the test-section surface temperature was one of the most difficult jobs. This was due to fluctuations in the tube wall temperature and due to change in the condensate layer thickness on the tube surface and its subsequent drainage. Therefore, it became imperative to record the time-average value of the surface temperatures. In the present investigation the readings of thermocouples were noted with the help of an Adam Data Acquisition Modules-4019 (ADAM-4019), the cooling water temperature rise was measured with a J-TYPE thermocouples and the cooling water inlet temperature was also measured with the help of a J-TYPE thermocouple. The vapor saturation temperature was measured with the help of two J TYPE thermocouples fixed inside the test condenser.

Prior to charging the refrigerant, the experimental setup was tested for any leakage by pressurizing it up to 2.0 MPa gauge pressure for 24 h by the compressor followed by a vacuum of 600 mmHg for the same duration of time by interchanging the inlet and outlet connections of the compressor. The refrigerant was charged in the vacuum setup. When the desired quantity was charged inside the experimental setup, the cooling water was circulated inside the test condenser. The refrigerant inside the evaporator was heated and the energy input to the heaters was controlled by a variac. The vapor generated inside the evaporator rushed to the test condenser, where it was condensed after coming in contact with the cold surface of the test section. When the rate of vapor generation became equal to the condensation rate, the pressure inside the setup was stabilized and also the temperature of the condensing vapors as well. It was not possible to remove all the air inside the experimental setup prior to charging of the refrigerant. With one set of condensation in the test condenser and the auxiliary condenser, they became the low-pressure zones inside the system. Therefore, air and refrigerant vapor rushed to this part of the experimental setup. Since the air is lighter than the refrigerant, it accumulated in the upper portion of the test condenser and the auxiliary condenser as well. After 30 min of initial operation of the setup, electric supply to immersion heaters was switched off; the refrigerant and air were discharged to the atmosphere by the purging. Again, the electric supply was restored to the pool of refrigerant and the closed valves were opened. The above sequence of operations was repeated 4–5 times. Most of the air was removed from the system by this method. Still, traces of air may have remained inside the system. To cope with this problem, the experimental setup was kept operative and the gases inside the test condenser and the auxiliary condenser were purged four times at an interval of 4–5 h. The temperature measurements were taken when the temperature above and below the test section became equal and agreed with the vapor saturation temperature with an error of less than 0.2 K. This error could have been inherited due to the traces of the foreign vapors and gases in the system. Marto et al. (1990) and Sukhatme et al. (1990) have also observed this phenomenon. In their experiments the temperature of the vapor agreed with the corresponding saturation temperature of the vapor with errors of 0.3 K and 0.2 K, respectively. The cooling water flow rate was changed from 400 kg/h, with an increment of 50 kg/h up to a flow rate of 1,300 kg/h. The pressure of condensing vapor was kept constant throughout the investigation by regulating the energy supply to the heaters in the evaporator and the cooling water flow rate in the auxiliary condenser. Therefore, for all the test runs the temperature of condensing vapor remained 318.1 ± 0.5 K, approximately the saturation temperature inside the condenser of a refrigeration plant. At each cooling water flow rate, the temperature of condensing vapor, the surface temperature of the test section, the temperature at the cooling water inlet, and the cooling water temperature rise were recorded.

3. Experimental Uncertainty

All the thermocouples were calibrated prior to installation to an accuracy of 0.1 K and therotameters were calibrated for an accuracy of ±18 kg/h flow rate. Due to the error in the measurements, an uncertainty analysis has been carried out as suggested by Kline and McClintok (1953), the uncertainty in the coolant flow rate was 1.9% (at 1,300 kg/h) to 5.0% (at 400 kg/h). The uncertainty in the measurement of the temperature of condensing vapor was only 0.25%. Therefore, it is clear that the maximum uncertainty comes from that in the heat flux. However, the heat flux was determined by the heat carried away by the cooling water. Hence, the errors in the measurements of the coolant flow rate and the temperature rise of the cooling water cumulated in the error in the heat flux. In the present investigation, at low cooling water flow rates, the error in measuring the coolant flow rate controlled the error in the heat flux and at high flow rates of the error in the temperature rise of the cooling water controlled the error in the heat flux.

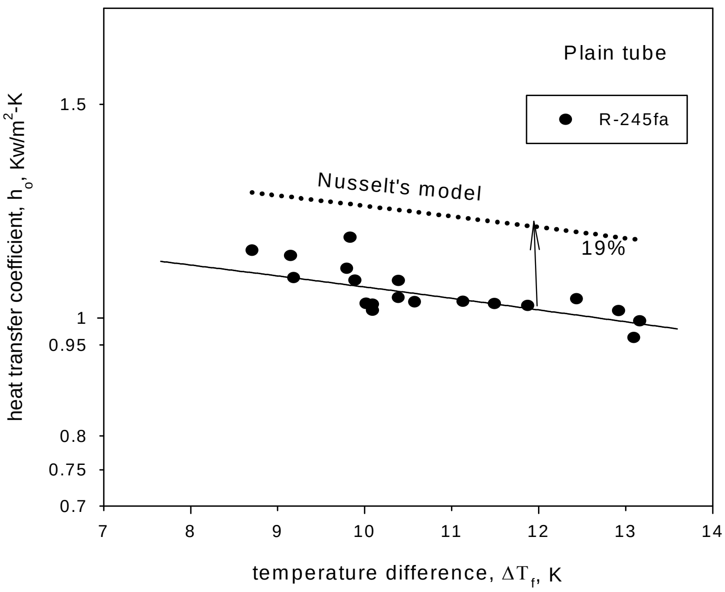
4. Result and Discussion

The result of the experimental investigation carried out for the condensation of pure vapor of R-245Fa over a plain tube is discussed below. For this investigation, the condensing-side heat transfer coefficient has been determined by the tube wall temperature measurement.

After the integrity of the experimental apparatus has been established and the reliability of experimental data has been ensured, the data for the condensation of the R245Fa over horizontal plain tubes have been further analyzed as below:

The variation of heat transfer coefficient during condensation of R-245Fa over a horizontal plain tube with the temperature difference across the condensate layer, ΔT_f, is shown in Figure 2.

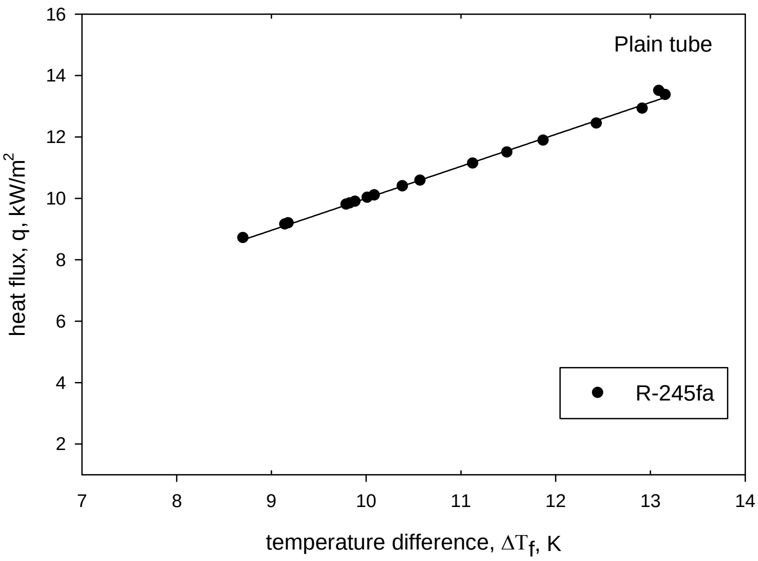
Figure 2 Variation of condensing side heat transfer coefficient
(Vapor to tube wall temperature difference for condensation of refrigerants over a plain tube)



The experimental values of heat transfer coefficient, h_o, of the plain tube for the condensation of R-245Fa vapor are in a range of 0.96 to1.13 kW/m2-K with an average value of 1.05-kW/m^2-K. The coefficients, ho of R-245Fa have also been computed by Nusselt's model, and are shown by separate dotted lines in Figure 2. Heat transfer coefficient reduces with the increase in temperature difference across the condensate layer, ΔT_f. This fact is also justified from the predictions of the Nusselt's model shown by the dotted line in the Figure 2.

The Nusselt's model also predicts the heat transfer coefficient for R-245Fa about 19 percent more at the same ΔT_f, To study the heat transfer behavior during condensation of refrigerant, Figure 2 is drawn, which shows that the heat transfer coefficient for refrigerant reduces with the increase in ΔT_f, and the data points tend to be on a straight line on a plot and the heat transfer coefficient has reduced with the rise in ΔT_f.

Figure 3 Variation of heat flux with vapor to tube wall temperature difference (Condensation of refrigerants over a plain tube)

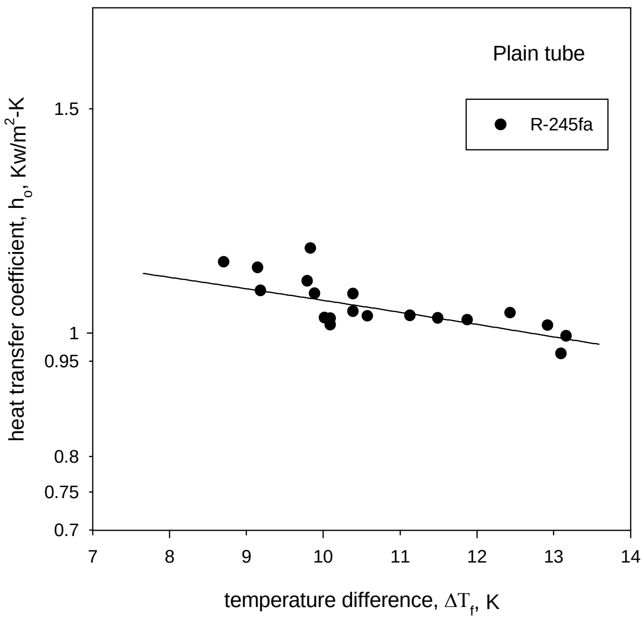


This is an expected trend since as the temperature difference across the condensate layer, ΔT_f, increases, more heat is transferred (heat flux increases) to the tube wall due to the generation of more condensate on the tube surface. This leads to the formation of a thick layer of condensate around the tube, and the heat transfer coefficient decreases.

Figure 3 shows the effect of ΔT_f, on heat flux for the condensation of refrigerants. As seen from the Figure, the heat flux increases with the rise in ΔT_f. Though the rise in the value of heat flux with ΔT_f clearly indicates that, with the rise in ΔT_f the heat transfer coefficient decreases the net effect is that more heat is passed from vapour to coolant.

The facts observed in Figure 3 become clear in plot between condensing side heat transfer coefficient and heat flux for the condensation of the refrigerant (Figure 4). The trends in this graph clearly testify that with the rise in heat flux, the heat transfer coefficient for a plain tube decreases during condensation of the refrigerant; this fact is in agreement with the findings of Marto (1990) for the condensation of refrigerants

Figure 4 Variation of condensing side heat transfer coefficient with heat flux
(Condensation of refrigerant over a plain tube)



5. Conclusion

The following conclusions have been drawn for the condensation of R-245Fa vapors over a horizontal plain tube.

1. For the condensation of R-245Fa over a plain tube, the heat transfer coefficient reduces with the rise in ΔT_f.
2. The Nusselt's model over predicts the heat transfer coefficient for the condensation of refrigerant R-245Fa. The heat transfer coefficients for R-245Fa predicted by Nusselt's model are nearly 19 percent more at a given value of ΔT_f .

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