Experimental heat transfer coefficient during condensation of R-410A in horizontal micro-fin tubes

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Abstract. Condensation heat transfer coefficient has been calculated experimentally in micro-fin tubes of outer diameter 9.52mm with helix angles 15° and 18°. Condensation of refrigerant R-410A include mass fluxes from 200 to 600 kg/m²s, vapour qualities between 0.1 to 0.9 and saturation temperature at 35°C and 40°C. The experimental results indicate that the average heat transfer coefficients of R-410A in micro-fin (MF) tubes were increased 1.15-1.47 times larger than those of smooth tube (ST). The experimental results are compared with the existing correlations of heat transfer coefficient proposed by other authors. The comparison indicated good agreement with these existing correlations within ±30%. A correlation has also been developed for predicting condensation heat transfer coefficient in micro-fin tubes.

Keywords: Heat transfer coefficient, condensation, micro-fin, HFC.

1. Introduction
In two phase flow vapour refrigerant at outlet of compressor in refrigeration and air conditioning components normally cooled and condensed by the condenser. The condenser transfer or dissipates the heat to secondary substance like water or air. If condenser do not transfer the heat properly than pressure at outlet of compressor increases resulting in power consumption of compressor increases. The way of improving the efficiency of condenser is to increase its size. But it is not possible because of higher maintenance and material cost and increased quantity of charge. However to develop the efficient condensers, it is necessary to manufacture condensers as small as possible in size by using micro-fin (MF) (figure. 1) tube. MF increases the heat transfer rate inside tube during condensation of refrigerants by increasing heat transfer surface area. So it is more important to find out the heat transfer coefficients (HTCs) inside MF tubes of air cooled effective condensers. R-410A is more efficient and blends of gasses (50% R-32 and 50% R-125) and will not damage the ozone layer. Recently, air conditioning systems like air conditioners with R-410A has been manufactured and available in market.
The experimental work has been obtained the benefits of micro-fin tubes over smooth tubes in terms of higher HTC during condensation of R-410A (HFC). Also compare the experimental HTC results by using existing correlations.
Wijaya and Sparz [1] reported condensation HTC of R-410A inside horizontal ST and compared to R-22 in similar operating conditions. The authors concluded that the experimental HTCs of R-410A were 2-6% larger compare to R-22. Jung et al. [2] performed the condensation experiments using R-134a, and R-407C in horizontal ST and MF tubes at saturation temperature of 40°C, mass velocity from 10-30 kg/m²s of 9.52mm O.D. They propose that the R-134a and R-410A showed lower HTC in STs than in MF tubes. Nualboonrueng et al. [3] performed experiments using R-134a in a 9.52mm O.D. ST and MF tube at condensation temperature of 30°C and 40°C inside range of mass velocity 400-800 kg/m²s. The authors investigated the HTCs inside MF tubes were 10-85% compare to STs. Chamra et al. [4] carried out condensation HTC and pressure drop (PD) inside 15.88mm O.D. MF tube geometry. They obtained the condensation HTC of cross grooved tubes were larger compare to single grooved tubes. Infante Ferreira et al. [5] obtained the condensation HTC and PD of R-404A inside ST and MF tube of 9.52mm O.D. at saturation temperature 40°C by using mass velocity 200-600 kg/m²s. They showed that the influence of oil concentration on HTC and PD. Goto et al. [6] determined the HTC during condensation of R-410A in internally grooved horizontal tubes of 8mm O.D. They found HTC of R-410A in herring-bone grooved tubes were twice comparing to helical tubes. Cavallini et al. [7] reported the HTC of R-410A inside 11.6 mm O.D. MF tube at mass velocity between 100-800 kg/m²s and have compared with the equivalent inner diameter smooth tube. They concluded that heat transfer enhancement factor and PD for R-410A were 2.5-3 times larger between 400-800 kg/m²s mass flux than those with most common HFC and HCFC refrigerants. Nozu et al. [8] calculated HTC and PD of R-11 inside horizontal MF tubes. They suggested that PD inside MF tubes were 60% higher compare ST.

2. Experimental facility and procedure
The test facility is schematically shown in the figure 2. It consists of three loops. In first loop refrigerant flows inside the evaporator and converted into superheated vapour. The pre-condenser is counter flow heat exchanger in which the cold water flows in the annulus condenses the superheated vapour. Figure 3 shows test condenser, it is a counter flow condenser. In which the refrigerant flows in the copper tube. The test condenser is insulated by nitrile rubber to prevent the heat transfer between test condenser and environment. T-type thermocouples are brazed on outer wall of copper tube at six cross sectional locations of copper tube. Refrigerant temperatures and pressures at different locations are measured by using thermocouple wires and pressure sensors inserted into the refrigerant flow in tube wall. The coriolis flow meter installed between post condenser and evaporator for measuring refrigerant flow rate. The liquid and vapour phase from the test condenser flows to a post condenser. The post condenser is single shell and two tube heat exchanger in which refrigerant flows in the shell.

Figure 1. Enlarged image of Micro-fins.
and cooling water in copper tubes. This condensed liquid refrigerant then sent back to the evaporator by open type reciprocating compressor and it is compatible with refrigerant R-410A. The refrigerant flow can be controlled by needle type expansion valve placed at inlet of evaporator.

![Figure 2. Schematic diagram of experimental test facility.](image)

In the hot water loop, evaporator is tube coil and tank type heat exchanger. The refrigerant is converted into vapour to take the heat from hot water in the tank. The water is heated by resistance heater immersed in the tank. The experimental conditions are tabulated in Table 1. In cooling water loop, temperature of cold water in pre and test condenser is regulated from resistance heater and chiller unit.

The power supply to the resistance heater is regulated by dimmerstat. Cold water flow rates are measured by acrylic rotameter and temperature by resistance temperature detector (RTD). Thermal properties of refrigerant are taken from REFPROP version 7.0 [9]. Dimensions of ST and MF tubes are tabulated in Table 2.
Table 1. Experimental test conditions

| Operating parameters         | Range      |
|------------------------------|------------|
| Refrigerant                  | R-410A     |
| Mass flux, G, kg/m²s         | 200-600    |
| Saturation temperature, Tₛ, °C| 35-40      |
| Cooling water temperature, Tₘ, °C| 30-35     |
| Vapour quality               | 0.1-0.9    |
| Outer diameter (mm)          | 9.52       |

Figure 3. Schematic diagram of test section
3. Data reduction

3.1. Heat transfer coefficient (HTC)

The local condensation HTC for two phase flow in horizontal tubes as obtained as:

\[ h = \frac{Q}{A(T_{\text{sat}} - T_{\text{wi}})} \] (1)

Where \( h \), HTC \((W/m^2\cdot k)\), \( Q \) rate of heat transfer exchanged in the horizontal ST and MF tubes \((W)\), \( A \) heat transfer surface area of tube \((m^2)\), \( T_{\text{sat}} \) saturation temperature of the refrigerant \((k)\) and \( T_{\text{wi}} \) tube inner wall temperature \((k)\) respectively.

The heat absorbed by the cooling water \( Q \) was determined as

\[ Q = m_w C_{pw} \Delta T \] (2)

Where \( C_{pw} \) specific heat of water, \( m_w \) cooling water flow rate and \( \Delta T \) temperature difference of cooling water.

Vapour quality in test condenser were determined by energy balance in pre and test condenser. Then, mean vapour quality in test condenser is determined by average of vapour quality in test condenser.

4. Comparison of heat transfer coefficient correlations

The comparison of experimentally calculated Nusselt number \((Nu)\) and predicted Nusselt number calculated by existing correlations proposed by Cavallini et al. [11], Sapali and Patil [12], Kedzierski & Goncalves [13] and Yu & Koyama [14] for R-410A are shown in figures 4 and 5. In figure 4 the experimental Nusselt values show good association with Cavallini et al. [11] and Sapali and Patil [12] correlation of about ±30% but approximate 70% data are accumulated in ±20% range except some data points. Similarly in figure 5 experimentally calculated data are scattered ranging from -50% to +20% with Kedzierski & Goncalves [13] and ranging from -50% to +50% with Yu & Koyama [14].

The positive and negative value of average deviation suggest the under prediction and over prediction. From the comparisons between calculated and predicted values of Nusselt number, it has examined that the Sapali and Patil [12] shows good agreement with experimental Nusselt number data rather than other three correlations because correlation of Sapali and Patil [12] also formed by using multiple linear regression of data. Table 3 collects the results of experimental HTC data and existing correlation HTC data in terms of average, mean absolute, and standard deviation, determined as

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**Table 2. Horizontal smooth and micro-fin tube dimensions**

| Parameters                  | Smooth tube | Micro-fin tube-1 (MF-1) | Micro-fin tube-2 (MF-2) |
|-----------------------------|-------------|-------------------------|-------------------------|
| Outer diameter, OD (mm)     | 9.52        | 9.52                    | 9.52                    |
| Bottom wall thickness (BWT), (mm) | -          | 0.30                    | 0.28                    |
| Fins height (F.H.) (mm)     | -           | 0.17                    | 0.15                    |
| Fins angle (α)              | -           | 25                      | 30                      |
| Helix angle (β)             | -           | 18                      | 15                      |
| No. of fins, N              | -           | 70                      | 70                      |
| Length of tube, (m)         | 2.32        | 2.32                    | 2.32                    |
| Cross sectional area, mm²   | 48.97       | 62.45                   | 63.02                   |

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e (percent deviation) = \left[ \frac{h_{\text{Pre}} - h_{\text{Exp.}}}{h_{\text{Exp.}}} \right] \times 100 \tag{3}

e_r (average deviation) = \frac{1}{N_p} \sum(e) \tag{4}

e_{ab} (mean absolute deviation) = \frac{1}{N_p} \sum|e| \tag{5}

S_d (standard deviation) = \left[ \frac{\sum(e - e_r)^2}{N_p - 1} \right]^{1/2} \tag{6}

Table 3. Average, mean and standard deviations calculated in comparisons of predicted and experimental Nusselt number data

|                  | Cavallini et al.[11] | Sapali & Patil [12] | Kedzierski and Goncalves [13] | Yu and Koyama [14] |
|------------------|-----------------------|----------------------|-------------------------------|---------------------|
| \( e_r \) (%)    | 8.25                  | 5.71                 | 12.98                         | 36.47               |
| \( e_{ab} \) (%) | 16.04                 | 13.18                | 17.26                         | 41.55               |
| \( s_d \) (%)    | 17.69                 | 15.28                | 21.38                         | 32.37               |

Figure 4. (a) and (b) Predicted vs. Experimental Nusselt number for R410A in MF tubes (MF-1 and MF-2): correlation by Cavallini et al. [11] and Sapali & Patil [12].
5. Results and discussion

5.1 Heat transfer coefficient (HTC)

Figures 6 and 7 represent the comparison of experimental HTC data between the ST and MF tubes (MF-1, 18° helix angle and MF-2, 15° helix angle). Experiments have been conducted on refrigerant R-410A at condensing temperature 35°C and 40°C in the range of mass velocity 200-600 kg/m²s between vapour quality 0.1-0.9. The values of HTC, increases by increase vapour quality and mass flux and by decrease condensing temperature. The figures 6 and 7 show the experimental values of HTC of R-410A are higher in MF tubes (MF-1 and MF-2) than ST, it is 1.15-1.47 times. Due to the fins in micro-fin tubes thickness of liquid film decreases at the fin tip resulting in increased HTC. The condensation HTC inside MF tube (MF-1) is larger than MF (MF-2) because of high helix angle which increase the heat transfer surface area. A higher helix angle takes place the fins inside the tubes are more inclined which increases the interactions between the incoming fluid flow and the fins resulting in HTC increases. Further due to high fin height interactions between the fin tip and incoming axial fluid flow increases.

Figure 5. (a) and (b) Predicted vs. Experimental Nusselt number for R410A in MF tubes (MF-1 and MF-2) : correlation by Kedzierski & Goncalves [13] and Yu & Koyama [14].

Figure 6. Comparison of HTC of R-410A in a ST and MF tube (MF-1, helix angle 18°) at condensing temperature of 35°C and 40°C.
Figure 7. Comparison of HTC of R-410A in a ST and MF tube (MF-2, helix angle 15°) at condensing temperature of 35°C and 40°C

5.2. Developed correlation for micro-fin tubes
The correlation is obtained by using multiple linear regression technique [10] with 140 data points. The correlation is shown in Eq. 7. The form of correlation is developed by using geometrical characteristics of micro-fin tubes and thermo physical properties of R-410A. In Eq. 7 equivalent Reynolds number represent the relative predominance of inertia forces to the viscous forces appearing in the flow of fluid. The presence of pressure ratio forms the correlation is more suitable for large range of saturation temperatures. In MF tubes, heat transfer area enhancement factor is commanding parameter than the dimensionless numbers. The dimensionless number Bond and Froude involves the influence of inertia forces as well as surface tension forces. Figure 8 shows comparison between new obtained correlation data and experimentally calculated HTC data. The figure 8 also suggested that the new obtained correlation good interacts with experimentally calculated HTC data of R-410A inside two different MF tubes with in ±20% with coefficient of correlation 0.8668, mean absolute deviation 8.3% and standard deviation 9.52%.

Figure 8. Comparison of experimental HTC with new correlation in micro-fin tubes

\[
Nu = 0.1744 . Re_{Eq}^{0.5765} . \left( \frac{P}{Pr} \right)^{-0.2089} . R_x^{0.6232} . (Fr . Bo)^{-0.0014}
\]  

(7)
6. Conclusions
[1] The condensation HTC's of R-410A were experimentally calculated inside ST and MF tubes of mass flux 200-600kg/m2s, vapour quality 0.1-0.9 at condensation temperature of 35°C and 40°C, respectively.
[2] The experimental HTC of R-410A in MF tubes (MF-1 and MF-2) were 1.1-1.47 times larger than similar outer diameter ST.
[3] The condensation HTC's of R-410A increases with increase of mass flux and vapour quality and decreases with decreases of condensation temperature inside both ST and MF tubes.
[4] The following new correlation has been developed for the condensation HTC (140 data points) in MF tubes

\[ Nu = 0.1744 \cdot Re_{eq}^{0.5765} \cdot \left( \frac{P}{Pr} \right)^{-0.2889} \cdot R_x^{0.6232} \cdot (Fr \cdot Bo)^{-0.0014} \]

[5] The experimental HTC's of R-410A has been calculated and compared with the previously proposed correlations inside two different micro-fin tubes (MF-1 and MF-2). The correlation by Sapali and Patil [12] indicated good agreement with HTC data (average deviation 5.71%).

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