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Advancing Experimental Study of Heat Transfer Characteristics of Refrigerant R-600a Condensing Over Horizontal Copper Coated Tubes

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Abstract: In this study, heat transfer characteristics of vapor condensation of R-600a over horizontal copper coated tubes were evaluated experimentally at saturated temperature of 40 ℃, wall sub-cooling temperatures of 4-18 ℃ and heat flux of 32-176 kW/m^2. The present study emphasized on surface modification through coating so that dropwise condensation could be promoted and condensate could be easily drained out from the tube surface. Also, this study investigated the omniphobic behavior of different coating parameters by comparing their heat transfer characteristics. The effect of different operating parameters on required outcomes were analyzed and represented through different graphs. Results were validated through modified Wilson plot method which underpredicted heat transfer coefficient by 11.9 to 26.8 %.

Keywords: Iso-Butane, Copper Coating, Dropwise Condensation, Heat Transfer Characteristics

1. Introduction

A Refrigerant is the working fluid for refrigeration and air-conditioning system. In spite of its high potential of depleting ozone layer and increasing global warming, it has been widely used for many industrial purposes. Good thermodynamic properties as well as low environmental impact are big concerns in refrigeration, heating and air-conditioning industry. Scientists are continuously starving in the quest of environment-friendly refrigerant without compromising its thermodynamic properties responsible for enhanced heat transfer. Though, some eco-friendly refrigerants have been developed but at the cost of desirable thermodynamic properties.

Also, the performance of a refrigerant depends on system variables. Thermodynamic properties and environmental impact of any refrigerant as well as system variables like tube geometry offer a big permutation and combination for getting optimum result. Condensation heat transfer over horizontal tubes has been continuously improved by researchers. Nusselt's film condensation theory [1] was first derived for vertical wall which was later modified for horizontal outer smooth surface. The biggest challenge in condensation heat transfer is the formation of condensate film which becomes thicker as condensation heat transfer increases; consequently heat transfer decreases due to shielding effect of condensate film. Many modifications in the outer surface of horizontal tube have been suggested by researchers to overcome this challenge. Beatty and Katz [2] suggested two dimensional integral-fin tubes to drain out the condensate film by gravitational effect. In the analysis, both condensate drainage and condensate retention by surface tension effect were neglected. Karkhu and Borovkov [3] were the first who included surface tension effect on heat transfer of condensing refrigerant on a tube. Further, many researchers performed experimental and analytical analysis varying different fin geometries and working fluids. Honda and Nozu [4] analyzed the effect of condensate retention angle on condensation heat transfer. Some scientists found that three dimensional finned tubes showed better heat transfer rate than two dimensional finned tube due to two directional condensate drainage mechanism. Jung *et al.* [5] investigated experimentally and found that three dimensional Turbo-c finned tube showed better heat transfer characteristics than two-dimensional low finned tube and plain tube with enhancement factor of 8 for all refrigerants R-11, R-123, R-12 and R-134a. Sajjan *et al*. [6] studied the effect of position of 3-D fins over tubes and found that position of 3-D fins over the lower part of tubes showed better condensation heat transfer rate with enhancement factor of 5.85 for R-600a refrigerant. Some scientists studied condensation heat transfer characteristics over tubes coated with nano-particles. Kim *et al*. [7] performed experimental investigations to analyze condensation heat transfer characteristics over carbon nanotube (CNT) coatings and concluded that enhancement in heat transfer coefficient was approximately 16 % in comparison to the bare surface. Adera *et al*. [8] used porous silica inverse opal coatings over copper tubes which showed almost sevenfold higher heat transfer rate (80 kW/m^2K) compared to smooth copper tube (12 kW/m^2K). The possible reason explained in this study was axial transport of condensate through cracks developed due to shrinkage of coating material. Liu *et al.* [9] investigated dropwise condensation of steam at low pressures over PFA-coated horizontal tube and examined the dynamics of condensate-droplet by optical observation. They found that average sweeping frequency increased by 122% as vapor pressure increased from 5 kPa to 100 kPa. Kuzma-Kichta *et al*. [10] developed a structured hydrophobic coating amalgamating nanoparticles with microstructure which achieved the contact angle of 158°. Cheng *et al*. [11] studied vapor condensation heat transfer characteristics of steam over stainless steel and brass coated tubes; and found that stainless coated tube exhibited maximum 22.2% increased heat transfer rate in comparison to plain tube whereas brass coated tube exhibited maximum 70.7% increased heat transfer rate.

Condensers are major components in RAC systems, heat pumps and different process industries. In shell and tube condensers where condensing vapor (in many cases, refrigerant vapor), flowing in the shell, condenses over tubes through which coolant (water) is flowing. The bigger challenge is to drain out the condensate collected over the tube which is decreasing the heat transfer rate. Our objective is to modify the tube surface through coating so that dropwise condensation can be promoted and condensate can be easily drained out from the tube surface. The present study also investigates the omniphobic behavior of different coating parameters by comparing their heat transfer characteristics. Since, a meagre variation in any parameter causes the significant change in results, a consistent effort is still required to get the best performing set of parameters. The present investigation caters ample span of data for outside heat transfer of R-600a condensing over copper coated tubes.

2. Experimental set-up and Procedure

The schematic flow diagram of an experimental facility is shown in Figure 1. An evaporator (5) is a cylindrical vessel built from Stainless Steel in which two immersion heaters (3) each of 3 kilo watt heat rating are fitted at the base. A test (main) condenser (12) is also a cylindrical vessel built from Stainless Steel. Test section (9) is installed inside the test condenser by flange-nut-bolt arrangement so that attachment and detachment of test section can easily be done. Liquid refrigerant is heated by immersion heaters to generate vapour which is transported to condenser through copper tube. Vapour generation rate is controlled with the help of variac. Vapour is uniformly distributed along test tubes through dead end perforated tube (8) having 1 mm hole diameter. Dead end perforated tube is a tube of 6.0 mm diameter and 400 mm length made of SS-304 and closed from one end in which 130 equidistant holes are made in a straight line at the top of the tube. Perforated tube is placed above test tubes with holes facing upwards so that vapour shear effect can be minimized as vapour strikes the inner surface of condenser and loses its momentum. Refrigerant vapour condenses over test tubes and sent back to evaporator in liquid form. Cooling water (coolant) is flowing inside test-tubes. An inverted U-tube (15) is attached to the exit side of cooling water to ensure availability of water when set-up is inoperative. An auxiliary condenser (13) attached to test condenser is placed over the test condenser to collect non-condensable gases and little amount of refrigerant vapour coming with non-condensable gases. Purging valve (14) is attached to auxiliary condenser to expel non-condensable gases from the experimental facility. Viewing window (11) fitted with sight glass made of Teflon is attached at the center of main condenser to monitor the flooding pattern of condensate. Turbine flow meters (4), J-type thermocouples, pressure gauges and transducers (7, 16) synchronized with data acquisition system are installed appropriately according to the schematic flow diagram. Figure 2 shows the pictorial image of the investigating facility. The details of different components are listed in table 1.

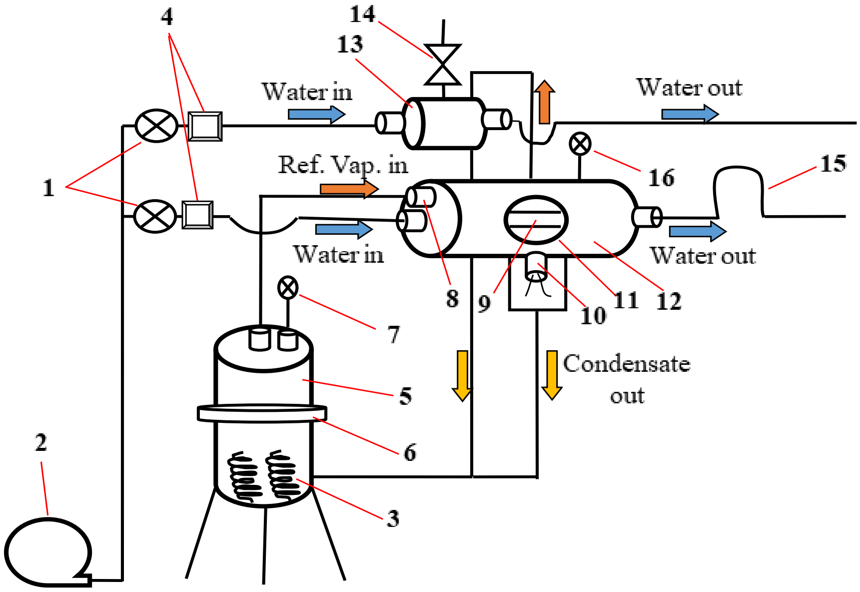
Air pressure of 290 psi was acquired in the experimental facility with the help of compressor and kept the set-up in the pressurized condition for 24 hours. No drop in pressure after the stipulated time ensured that there was no leakage in the heat exchanger. Pressure was released and set-up was kept under vacuum of 80 kPa for the same stipulated time. After confirming that there was no variation in vacuum pressure, the experimental facility was charged with refrigerant under vacuum condition. In the evaporator, liquid refrigerant is heated with the help of emersion heaters to produce vapor. During this phase change process, temperature of vapor remains constant and close to saturated temperature at that particular vapor pressure. The vapor pressure was sustained during experimentation by controlling immersion heaters with the help of variac. Data were recorded after attaining steady state condition in which vapor generation-rate and condensation-rate both attained equilibrium to each other resulting into the stability of pressure and sub-cooling temperature inside the experimental set-up. The associated pressure during the experiments was 0.53 MPa. Thermocouples are connected at four different equidistant peripheral positions of the tube. Average reading of these four thermocouples represents tube wall temperature (T_w). Three thermocouples are inserted inside condenser, entry and exit tubes of the cooling water which represents saturated vapour temperature (T_S), inlet coolant temperature (T_{wi}) and outlet coolant temperature (T_{wo}) respectively. A gas flame spraying method is used to coat copper powder over a smooth copper tube of outside diameter of 19 mm and thickness 3 mm. Acetylene and Oxygen are mixed and ignited with the help of torch to generate high temperature flame. High pressure air works as carrier to carry copper powder in the high temperature flame where copper powder melts and further accelerates towards substrate. Molten particles impact the substrate and solidify rapidly to form a coating over the substrate. Four copper coated tubes with different coating parameters have been prepared to study condensation heat transfer characteristics and to estimate heat transfer enhancement factor in comparison to plain tube. Surface profile meter of accuracy 0.01 microns and Scanning electron microscope (SEM) are used to measure coating parameters of the test tubes. The photographic view of smooth and coated surfaces is shown in Figure 3. SEM images of coated and plain surfaces are illustrated in Figure 4. Details of coating variables are listed in table 2.

Table 1. Details of components of experimental set-up

Component	Thickness (mm)	Inside diameter (mm)	Length (mm)
Condenser	3	100	410
Evaporator	3	140	600
Auxiliary condenser	3	110	140
Perforated tube	1	6.5	370

Table 2. Details of coating parameters

Test tube (*Nomenclature*)	Thickness of coating, t_c (μm)	Porosity of coating, ε (%)	Mean pore diameter of coating, d_{mp} (μm)
Plain tube (CT-0)	–	–	–
CT-1	57	13.03	2.68
CT-2	132	11.67	2.45
CT-3	289	11.79	2.52
CT-4	431	10.36	2.41



1. Valve, 2. Centrifugal pump, 3. Immersion heater, 4. Turbine flow meter, 5. Evaporator, 6. Flange, 7 & 16. Pressure gauge & pressure transducer, 8. Perforated pipe, 9. Test section, 10. Thermocouple exit port, 11. Viewing window, 12. Test condenser, 13. Auxiliary condenser, 14. Purging valve, 15. U-bend tube

Figure 1. Schematic flow diagram of the experimental set-up

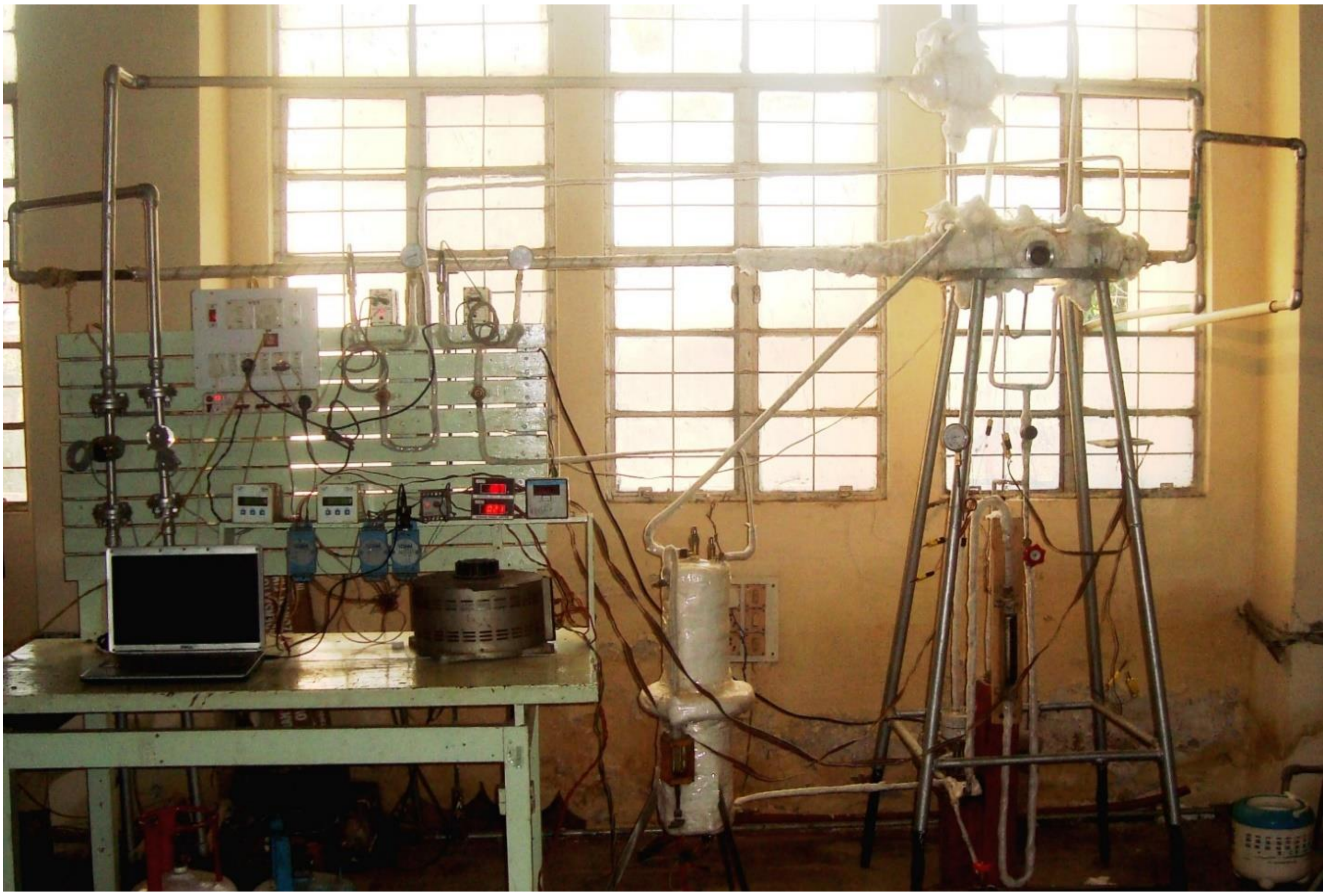


Figure 2. Pictorial image of investigating facility

(a) plain tube (CT-0)

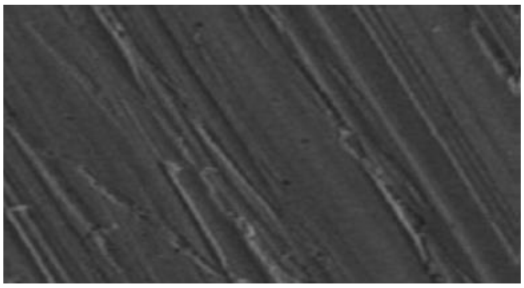
(b) 57 μm thick coating (CT-1)

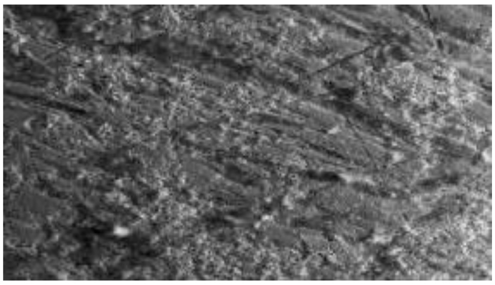
(c) 132 μm thick coating (CT-2)

(d) 289 μm thick coating (CT-3)

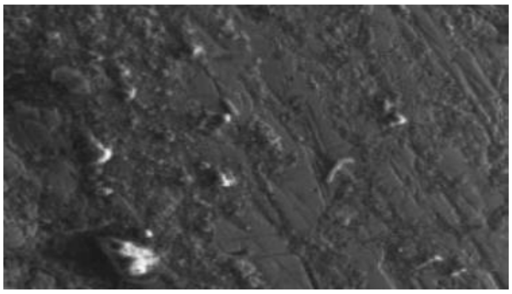
(e) 431 μm thick coating (CT-4)

Figure 3. Photographic view of coated and plain surfaces





(a) Plain tube (CT-0) (b) 57 µm thick coating (CT-1)



(c) 132 µm thick coating (CT-2) (d) 289 µm thick coating (CT-3)

(e) 431 µm thick coating (CT-4)

Figure 4. Coated and Plain Surfaces Illustrated By SEM

3. Data reduction and Error analysis

Heat transfer rate is estimated by simple heat balance equation between refrigerant vapour condensing over test tube and coolant flowing inside the test tube. The heat released by condensing vapour is equal to the heat absorbed by cooling water (coolant). Heat transfer rate, q, can be estimated either by calculating heat lost by condensing vapour or by calculating heat gain by cooling water. Since cooling water flow rate is much higher than refrigerant flow rate, hence low flow rate measuring instrument requires more accuracy and is expensive also. Therefore, in this study, cooling water flow meter has been used and heat transfer rate, q, has been estimated by evaluating rate of heat gain by coolant.

$$q = \dot{m}_{cw} C_{p_{cw}} (T_{cw_o} - T_{cw_i}) \quad (1)$$

Heat flux, q'' has been evaluated by dividing q by A_o,

$$q'' = \frac{q}{A_o} \quad (2)$$

Table 3. Maximum uncertainties induced in measurements and experimental data

Measurements	InducedUncertainty
Temperature measurement (T)	1.1 ℃
Pressure measurement (p)	5.5 kPa
Mass flow rate of coolant ($\dot{m}_{cw}$)	1 kg/h
Length of the test section (L)	1.0 mm
Diameter of the test section (d)	10 µm
Coating thickness (t_c)	0.01 µm
Outer surface area (A_o)	0.28 %
Heat flux (q'')	3.42 %
Heat transfer coefficient (h_o)	9.89 %

Outside condensation heat transfer coefficient, h_o has been assessed by dividing q'' by ΔT_f:

$$h_o = \frac{q''}{\Delta T_f} \quad (3)$$

$$\Delta T_f = T_S - T_W \qquad (4)$$

Errors in measurements transit in the results through different correlations and equations. An uncertainty analysis is carried out to quantify the errors and interpret the results accurately. It also helps to select the appropriate measuring instrument.

Correlations developed by Kline and McClintock [12] are adopted for error analysis in this study. Maximum uncertainties associated with measurements and experimental data are presented in the Table 3.

4. Results and Discussion

This investigation analyzes HTC of R-600a condensing over copper coated tubes placed horizontally under steady state condition. Experiments were performed under controlled atmospheric condition of vapour temperature of 40 ℃, wall sub-cooling temperatures of 4-18 ℃ and heat flux of 32-176 kW/m^2. The effect of different operating parameters on required outcomes were analyzed and represented through different graphs. Figure 5 depicts the proportionality relationship between q'' and ΔT_f which is in the agreement with basic convection heat transfer equation. Also, all coated tubes exhibit better heat transfer performance compared to smooth tube. Coating improves surface condition to promote dropwise condensation. 57 µm (CT-1), 132 µm (CT-2), 289 µm (CT-3) and 431 µm (CT-4) coating raised heat flux to 107.3 kW/m^2, 175.8 kW/m^2, 104.9 kW/m^2 and 78.5 kW/m^2 respectively whereas HTC to 7.7 kW/m^2-°C, 11.5 kW/m^2-°C, 9.8 kW/m^2-°C and 9.6 kW/m^2-°C respectively. It is clear from table 2 that porosity and mean pore diameter of the coatings decrease as coating thickness increases. As porosity of the coating decreases, grain density of the coating increases. Since, grains of the coating act as nucleation sites for dropwise condensation; higher the grain density, higher the condensation sites. However, on increasing grain density beyond a certain limit, condensate drops coalesce to form a film which further reduces the heat transfer rate.

Tube having 132 µm thick coating (CT-2) showed highest heat flux value among all coated tubes. Since favorable surface condition for dropwise condensation depends upon thickness of the coatings (coating parameters like grain density and porosity depends upon thickness of the coating), it improves upto a certain coating thickness thereafter deteriorates. As explained earlier, increasing the thickness of the coatings increases condensation sites to increase heat transfer rate, but also increases the chance of condensate drops to coalesce to form film which decreases heat transfer rate. CT-2 attains balance between these two effects to get optimum heat transfer rate. Zhang *et al.* [13] supported the findings by their results in which pure coating layers showed 3.1 to 7.4% higher heat transfer rate in comparison to composite coating. Although condensate film thickness reduced due to enhanced surface condition, it could not eliminated completely. Gain in wall sub cooling temperature enhances heat transfer rate. A thin blanket of condensate is wrapped around the tube whose thickness increases gradually due to enhanced rate of heat transfer. Consequently, an additional thermal resistance causes reduction in HTC as indicated in Figure 6 and Figure 7. Performance of the copper coated tubes is measured by the parameter called Enhancement factor (EF). It is expressed as the ratio of HTC of the test tube calculated experimentally to that of smooth tube calculated theoretically at same ΔT_f. HTC of smooth tube is determined by Nusselt's equation [1] since same ΔT_f is difficult to generate for both test tube and smooth tube. Performance of copper coated tubes is depicted in Figure 8 and 9. CT-2 is showing highest average value of EF which is equal to 7. It is in agreement with Sajjan *et al*. [6] with some improvement in EF where EF was equal to 5.85. Results interpreted in Figure. 6-9 show superior heat transfer characteristics of coated tube in comparison to plain tube which are supported by the findings of Rabbi *et al.* [14].

They concluded that P-HFDS coating improved HTC by 274%, 374%, 636% and 688% for ethyl alcohol, hexane, pentane, and R1233zd (E) respectively, compared to bare metal surfaces. Coolant flow rate also affects condensation heat transfer performance of coated tubes. Figure 10 and 11 depict the effects of cooling water flow rate on condensation heat transfer characteristics over test tubes. Coolant flow rate is characterized by Reynolds Number.Figure10 represents the influence of coolant Reynolds number onΔT_f. From the figure, it is manifested that ΔT_f increases with increase in coolant Reynolds number. From Sieder-Tate [15] equation (5), it is clear that increase in coolant Reynolds number increases coolant side heat transfer coefficient (hi). Consequently, overall heat transfer coefficient (U_o) and wall sub-cooling temperature (ΔT_f) also increase. Fig. 11 represents the trend in which heat flux ascends with increase in coolant Reynolds number. As discussed earlier, ΔT_f shows ascending trend with Reynolds number of the coolant. Gain in ΔT_f increases heat flux which is in agreement with Newton's law of cooling. Due to increased heat flux, a thin film of condensate blankets over tubes working as a shield. This shield acts as an additional thermal resistance causing a meagre decrease in HTC which is evident in Figure 12. & 13 illustrates the validity of experimental results with the help of modified Wilson plot method which was initially developed by Wilson [16] and later on, modified by two scientists Briggs and Young [17].This method underpredicts heat transfer coefficient by 11.9 to 26.8 % because this method assumes that condensate flow is laminar but in reality, flow is turbulent due to fast drainage of condensate.

Sieder-Tate equation:

$$h_i = c_i\left(\frac{k_b}{D_i}\right)Re_b^{0.8}Pr_b^{0.33}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \quad (5)$$

Where C_i (Sieder-Tate constant for coolant flow inside the tube) = 0.027

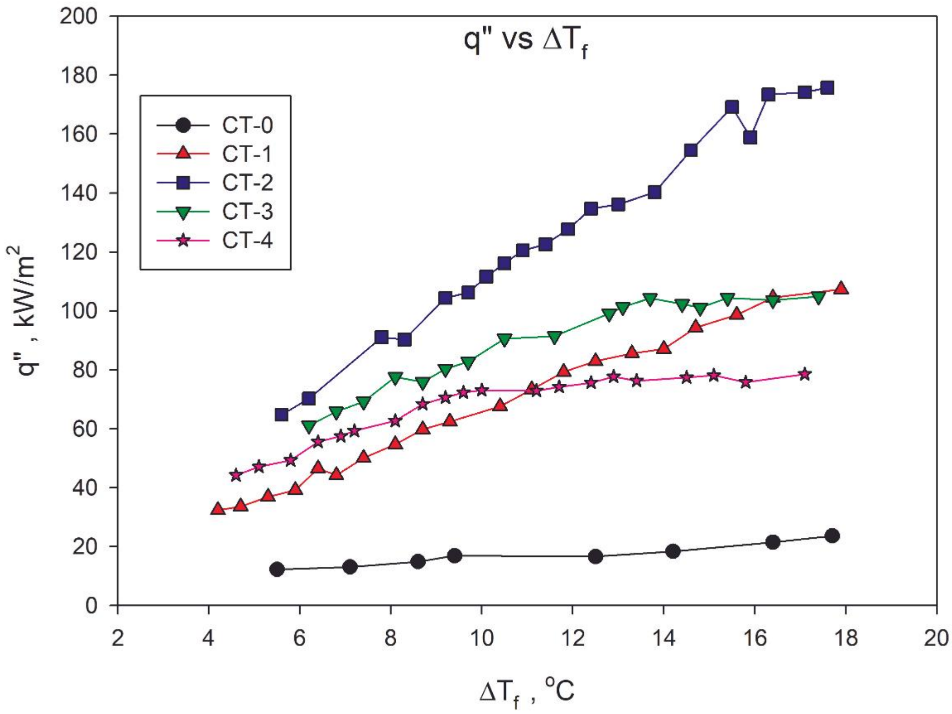


Figure 5. Effect of ΔT_f on q''

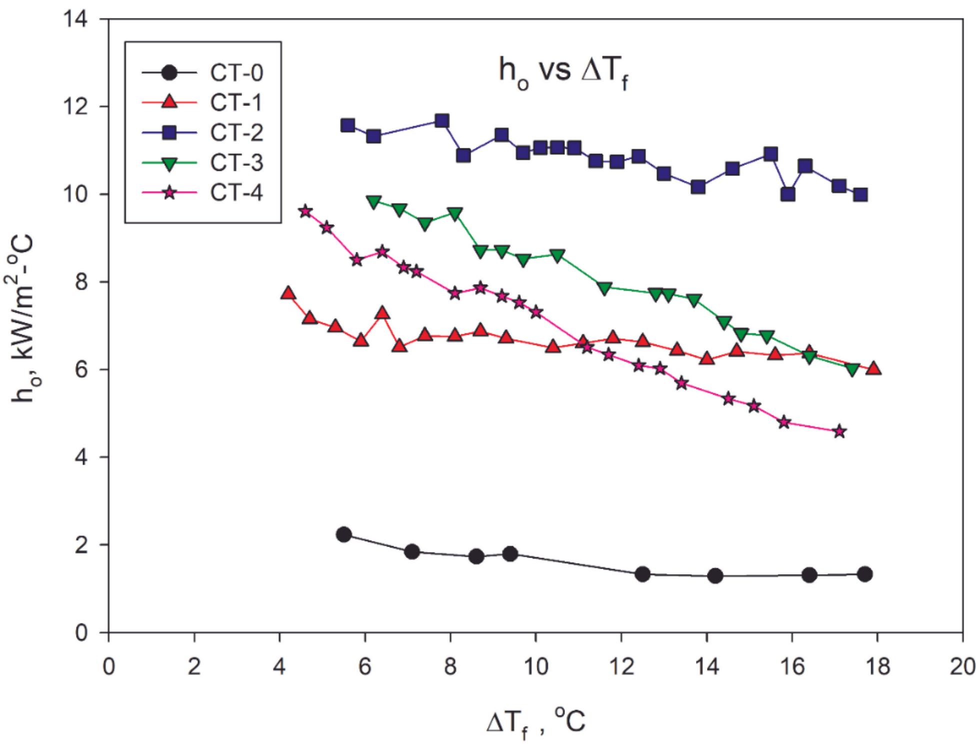


Figure 6. Effect of wall sub cooling temperature on HTC

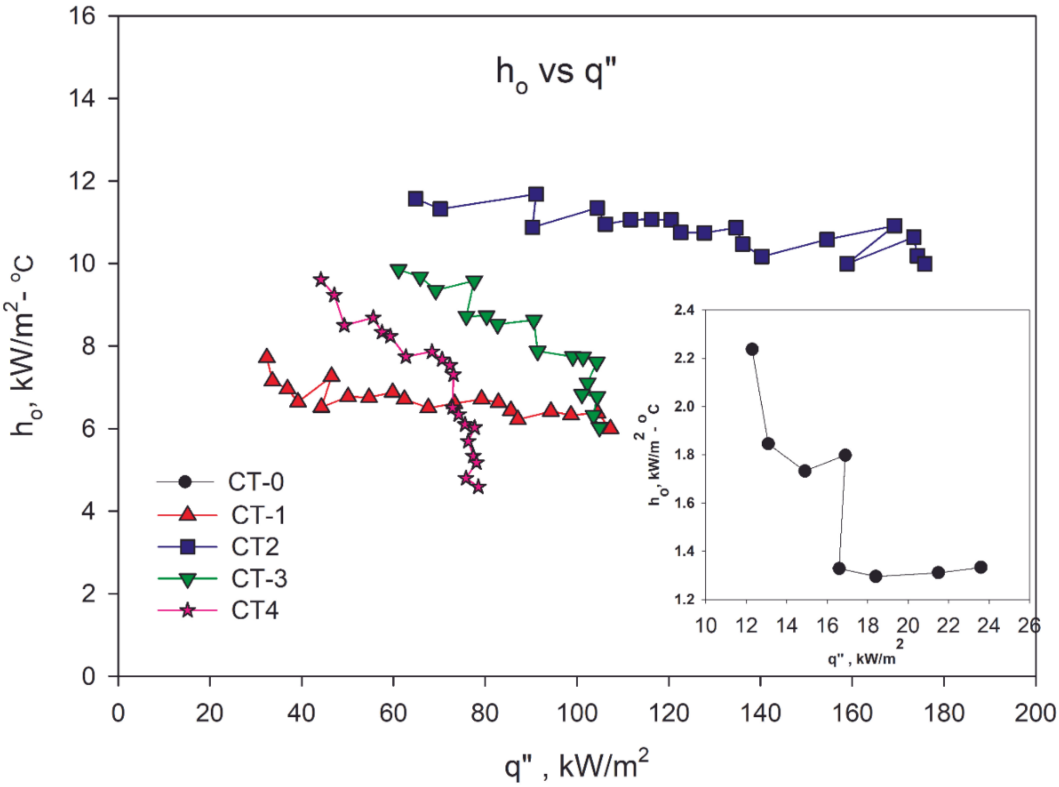


Figure 7. Dependency of heat flux on HTC

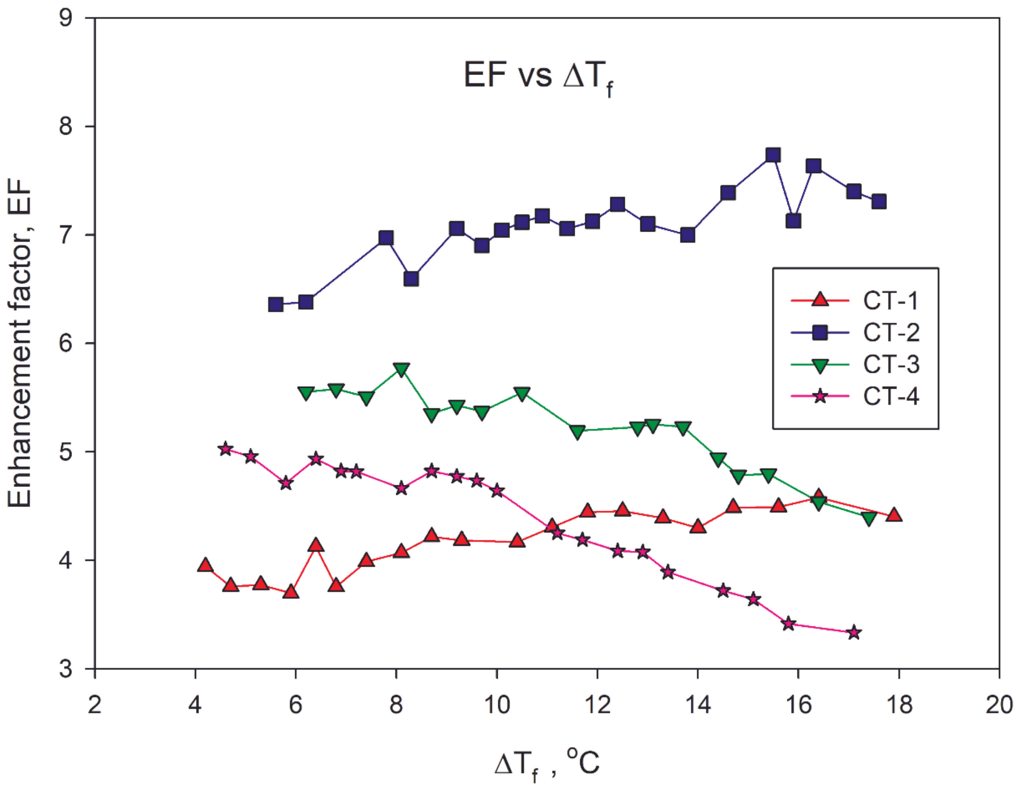


Figure 8. Dependency of ΔT_f on enhancement factor

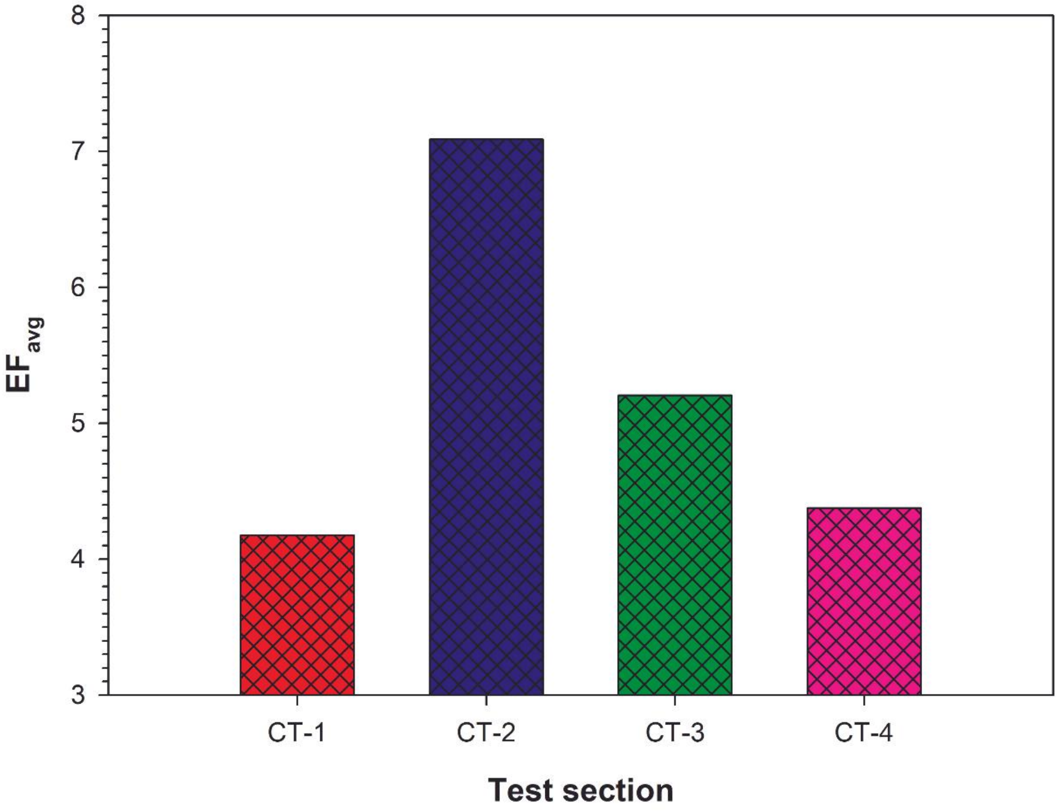


Figure 9. Performance of different coated tubes in terms of EF

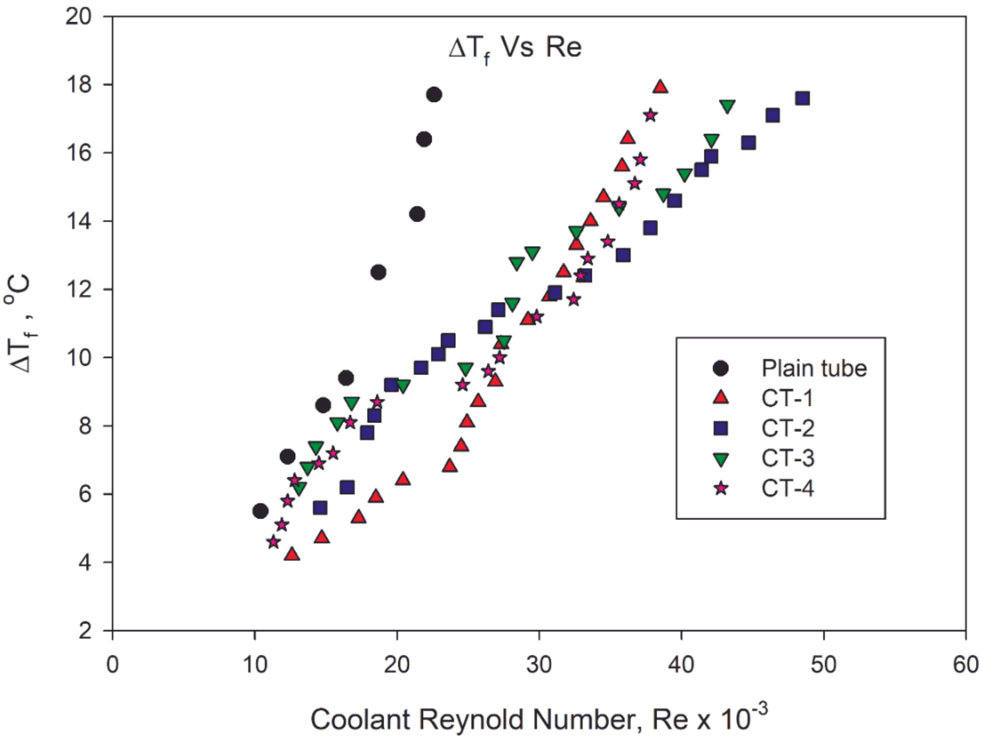


Figure 10. Dependency of coolant Reynolds number on ΔT_f

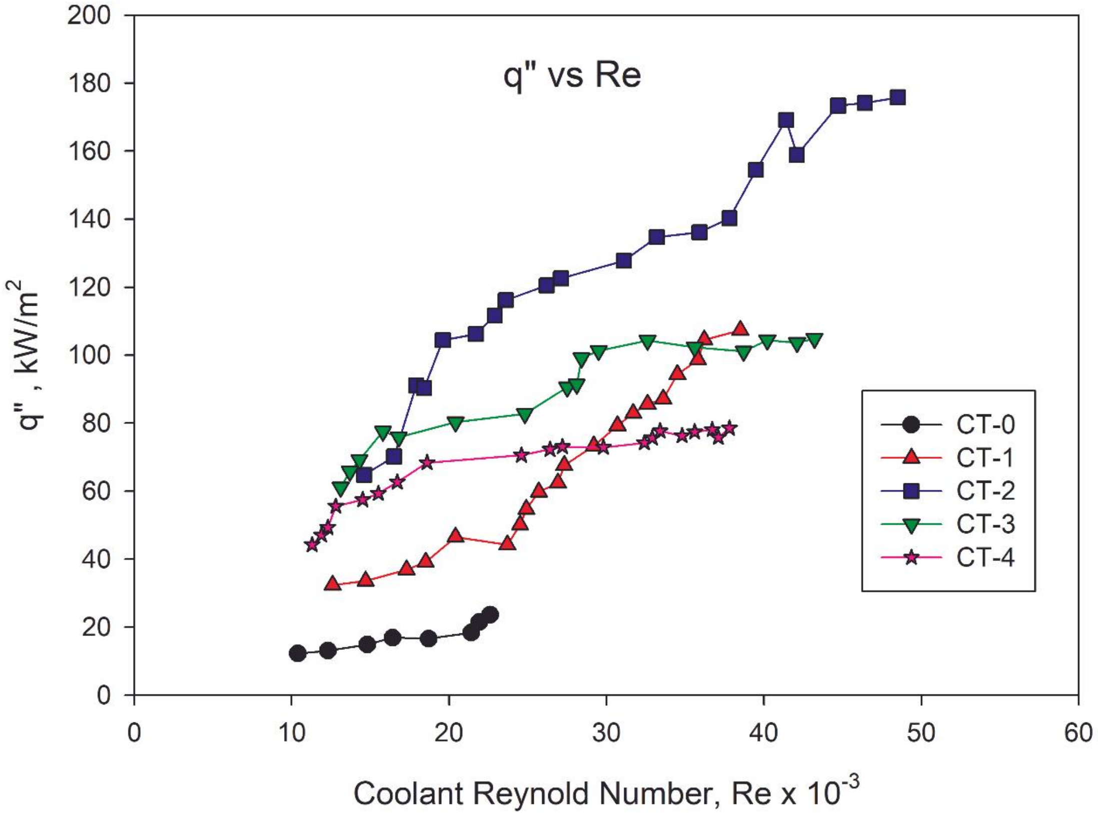


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Figure 11. Relationship between coolant Reynolds number and heat flux

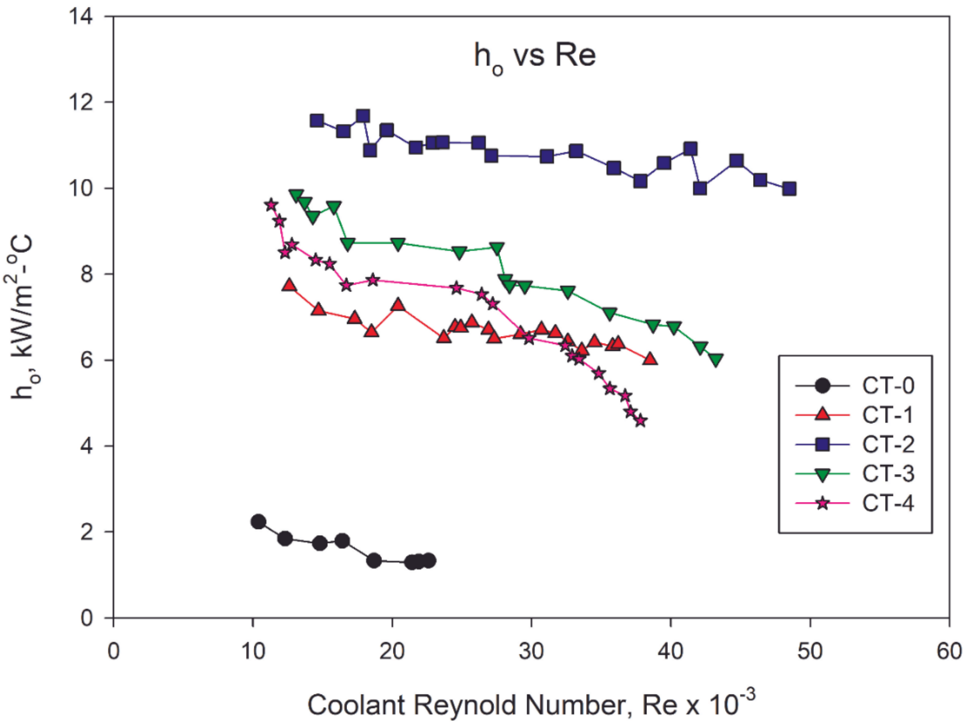

Figure 12. Relationship between coolant Reynolds number and HTC

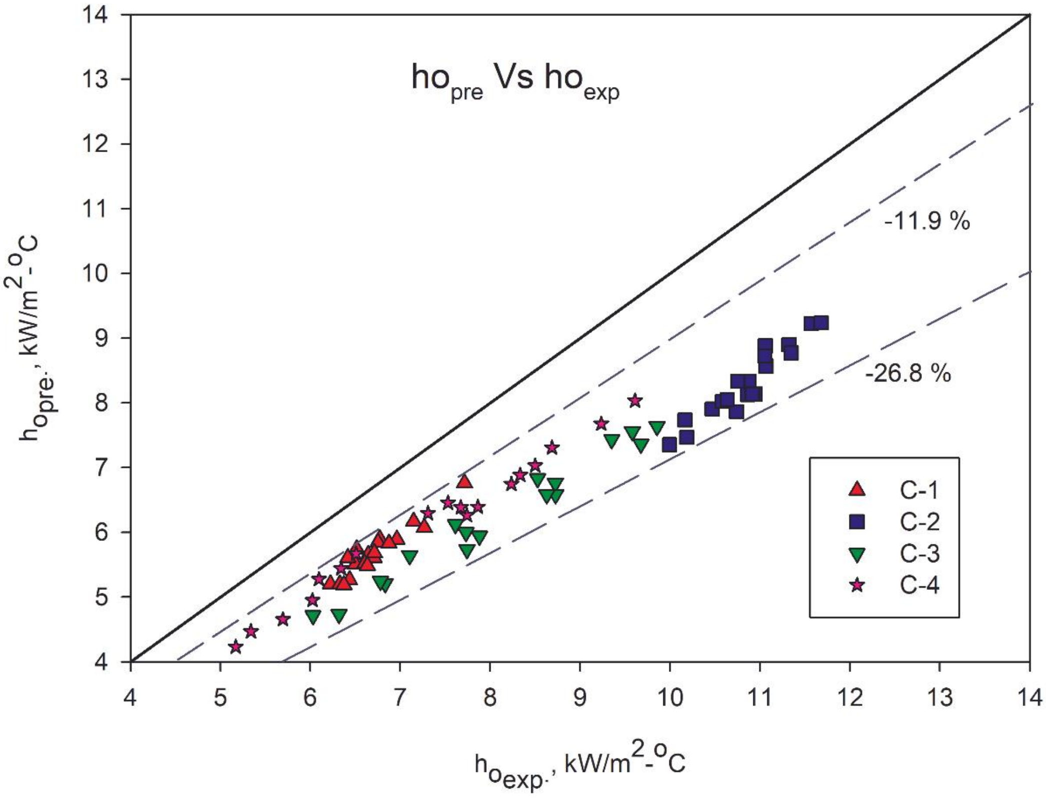

Figure 13. Validation of experimental HTC by modified Wilson plot method

6. Conclusion and Future Works

All coated tubes manifest enhanced heat transfer rate compared to plain tube. Coating improves surface condition to promote dropwise condensation. 57 µm (CT-1), 132 µm (CT-2), 289 µm (CT-3) and 431 µm (CT-4) coating raised heat flux to 107.3 kW/m^2, 175.8 kW/m^2, 104.9 kW/m^2 and 78.5 kW/m^2 respectively whereas HTC to 7.7 kW/m^2-°C, 11.5 kW/m^2-°C, 9.8 kW/m^2-°C and 9.6 kW/m^2-°C respectively. Tube having 132 µm thick coating (CT-2) showed highest heat transfer characteristics among all coated tubes. CT-2 is showing highest average value of EF which is equal to 7. Elevated difference in temperature of refrigerant vapor and tube wall enhances heat transfer rate which causes increased retention of condensate over tubes reducing the heat transfer coefficient. Coolant flow pattern puts significant impact on heat transfer rate. Evidently, elevation in Reynolds number results into elevation of wall sub-cooling temperature, and thus leads to enhanced heat flux. Due to increased heat transfer rate, a thin film of condensate blankets over tubes causing a slight decrease in heat transfer coefficient. Modified Wilson plot method underestimates HTC by 11.9 to 26.8 % because this method assumes that condensate flow is laminar but in reality, flow is turbulent due to fast drainage of condensate.

Nomenclature

A	surface area (m^2)
h_o	Condensing-side heat transfer coefficient (kW/m^2-^{0}C)
HTC	Heat transfer coefficient (kW/m^2-^{0}C)
C_p	specific heat (kJ kg^{-1} K)
T	temperature (K or ℃)
$\dot{m}$	mass flow rate (kg s^{-1})
ΔT_f	wall sub-cooling temperature (K or ℃)
q	heat transfer rate (kW)
Re	Reynolds number
Pr	Prandtle number
q''	heat flux (kW/m^2)
μ	viscosity (μPas)
ρ	density (kg/m^3)
Subscripts	
b	at coolant bulk temperature
cw	coolant or cooling water
i	inlet, inside or coolant-side
o	outet, outside
exp	experimental values
s	saturated vapor
w	surface of tube or wall of tube

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Authors Contribution Statement

Sanjeev Kumar Sajjan: Conceptualization, Methodology, Investigation. Ashok Kumar Dewangan: Software, Resources. Manish Kumar: Formal Analysis, Data Curation. Akash Kashyap: Visualization, Validation. Gopal Nandan: Supervision, Writing-Reviewing and Editing. All authors read andapproved the final manuscript.

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Data Availability

The data supporting the findings of this study can be obtained from the corresponding author upon reasonable request.

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