EXPERIMENTAL STUDY ON SUBCOOLED FLOW BOILING OF WATER INSIDE A THREADED TUBE WITH DIFFERENT TUBE INCLINATIONS

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ABSTRACT

The objective of this paper is to investigate the influence on heat transfer coefficient of subcooled boiling of water inside a copper threaded tube at five different angles between 0^0& 90^0. The experimental data were obtained over a heat fluxes range from 17 kW/m^2 to 114 kW/m^{2}and mass fluxes range of 100 kg/m^{2}s to 300 kg/m^{2}s. The specifications of test section were as follows: length of test section, 1000 mm; outside diameter, 9.52 mm; inner diameter,7.52 mm; bottom wall thickness, 0.76 mm; tooth depth, 0.24 mm; tooth apex angle, 60^0; helix angle, 25^0. The influences of above parameters on the heat transfer coefficient with tube inclinations are presented. The heat transfer coefficients predicted by some available correlations are compared with the present data.

Keywords: Flow boiling, Subcooled, Inclined, Internal Threaded Copper Tube

I. INTRODUCTION

Flow boiling has long played a major role in many industrial applications due to its superior heat transfer performance such as water tube boilers, evaporators, nuclear power plants and high density electronic components. Boiling is a very complex process due to heterogeneous nature of heat transfer medium. In subcooled boiling vapour bubbles generates at the heater surface while the bulk temperature of the liquid is still below the saturation temperature. Bubbles detaching from the heated surface collapse and condense in the subcooled liquid bulk. Also the use of augmentative techniques, either active or passive, to increase heat transfer coefficient has been studied so many times. One of the passive techniques to enhance heat transfer coefficient is the applications of internally threaded tube. Internally threaded tube can increase heat transfer through creating turbulence and limiting the growth of thermal boundary layer by slight increase in pressure drop. Many researchers have conducted experimental studies with or without passive techniques. Also the heat transfer characteristics generally keeps changing as the flow pattern changes inside the test section in the other hand the flow regime influenced by interfacial shear stress, surface tension, buoyancy and gravitational force.

Nomenclature

Bo	Bond number
d_{in}	Inner diameter (mm)
d_{out}	Outer diameter (mm)
Fr	Froude number
G	Mass flux (kg/ m^2 s)
I	Current (A)
k	Thermal conductivity of copper (W/m^{0}C)
k_f	Thermal conductivity of water (W/ m^{0}C)
l	Length of test section (mm)
M	Mass flow rate (kg/s)
Nu	Nusselt number
n_g	Number of grooves
Pr_l	Prandtl number of water
Q	Heat flow (W)
q_w	Heat flux (W/ m^2)
Re_{eq}	Equivalent Reynolds number
Rx	Geometry enhancement factor
T_f	Fluid temperature (^{0}C)
T_{in}	Inlet temperature of water (^{0}C)
T_{out}	Outlet temperature of water (^{0}C)
T_{wi}	Inner surface temperature (^{0}C)
T_{ws}	Outer surface temperature (^{0}C)
T_{wo}	Average outer surface temperature (^{0}C)
u_{GO}	Velocity of gas phase with total flow rate (m/s)
V	Voltage (V)
x	Quality
μ	Dynamic viscosity (Pa s)
g	Gravity
α	Heat transfer coefficient (W/m$^{2\,0}$C)
ρ_l	Density of water (kg/m^3)
ρ_g	Density of vapour (kg/m^3)
σ	Surface tension (N/ m)
γ	Apex angle
β	Helix angle

II. LITERATURE SURVEY

As an example of conducted researchers, Sarafraz et al.[1] experimentally studied flow boiling heat transfer of dilute water- diethylene glycol mixtures inside a vertical annulus. He investigated the influence of heat flux, flow velocity, degree of subcooling and concentration of mixture on heat transfer coefficient in both the convection and nucleate boiling regimes. Barbosa et al. [2] conducted experiments in a vertical annulus in which heat was applied to the inner surface of the tube. A dominance of nucleate boiling was observed at low qualities. At high qualities, nucleate boiling was partly or totally suppressed and forced convection became the dominant mechanism. Thus, one may conclude that in internal flow boiling, the heat transfer coefficients a combination of two mechanisms: nucleate boiling and forced convection. Bin sun et al. [3] experimentally studied the flow boiling heat transfer characteristics of four nanorefrigerants in an internal threaded tube. They found that maximum heat transfer coefficient of four kinds of nanorefrigerants increased by 17-25%and the heat transfer coefficient increased by 3-20%.Akhvanbahabadi et al. [4] experimentally investigated evaporation heat transfer of R-134a inside a microfin tube for seven different tube inclinations ranging from -90^0 to $+90^0$. Results showed that at low vapour qualities the highest heat transfer coefficient was attained at $+90^0$& at higher vapour qualities the highest heat transfer coefficient when tube is horizontal or was inclined at -30^0. The vertical tube with inclination angle of 90^0 had the lowest heat transfer coefficient for entire range of vapour quality. Arijit kundu et al. [5] studied the heat transfer characteristics of R-407 inside smooth tube with different tube inclinations angles from 0^0-90^0. They found that heat transfer coefficient increases with mass flux and heat flux. The tube inclination angle affects heat transfer coefficient in a significant manner for the inclination of 90^0 the highest heat transfer was attained. Akhvan bahabadi et al. [6] studied experimentally evaporation heat transfer of R-134a inside a corrugated tube for seven different tube inclinations ranging from -90^0 to $+90^0$. They found that for low vapour quality region heat transfer coefficient for $+90^0$ inclined tube was about 62 % more than that of -90^0 inclined tube. Also for all mass velocities, the highest heat transfer coefficient were achieved for $+90^0$.

III. EXPERIMENTAL METHOD

3.1 Experimental Set-up

The schematic diagram of test apparatus has been shown in fig.1. It consists of a pre-heater, a pump, a rotameter, a test section, and water cooled condenser. Initially the fluid is heated in preheater at 80^0C, then that fluid flows to test section through rotameter. The liquid- vapour mixture that outflowed from test section flowed into the condenser, where it is condensed into liquid. The condensed liquid passed through circulation pump to pre- heater. The pre-heater consists of a tank with 1.5 kW capacity heater was installed in it and constant AC supply was given for preheating the fluid. The test section was heated by a flexible nichrome heater wire (of 4 kW capacity) wrapped around the outside of test tube. Heat flow to heater wire was monitored with a variety AC voltage controller. The voltage and current flow was measured by analog meter to determine applied heat flux.

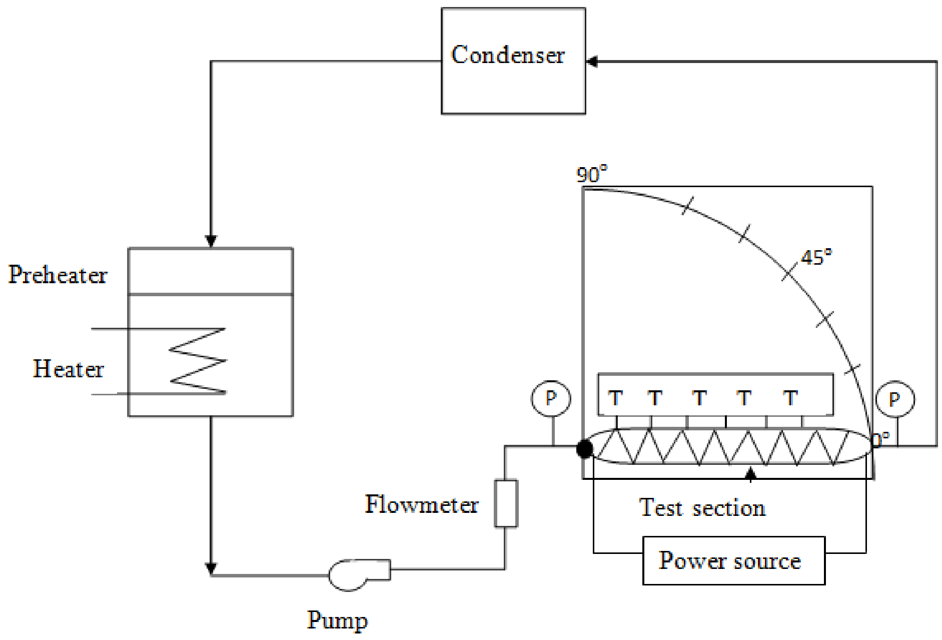

Fig.1Schematic Diagram of Experimental Setup

3.2 Test Section

The test section was made of copper tube. The specifications of test section were as follows: length of test section, 1000 mm; outside diameter, 9.52 mm; inner diameter, 7.52 mm; bottom wall thickness, 0.76 mm; tooth depth, 0.24 mm; tooth apex angle, 60^0; helix angle, 25^0. The test section was heated by a flexible nichrome heater wire (of 4 kW capacity) wrapped around the outside of test tube, and five cross sections without heater wire are reserved in order to adhere thermocouples, as shown in fig. 2. Ten K- type thermocouples are located at top and bottom sides of above five cross sections of the test tube to measure the outside tube wall temperatures. The test section was insulated with ceramic wool to reduce heat loss to the surroundings. Also two thermocouples and one pressure sensor to measure temperature and pressure of fluid are installed at inlet and outlet of test section respectively.

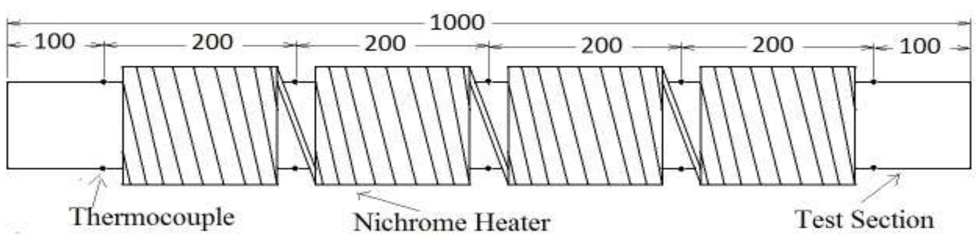

Fig.2 The Test Section and Layout (unit: mm)

3.3. Procedure

The heat transfer coefficient was calculated by using following equation,

$$\alpha = \frac{q}{T_{wi} - T_f}$$

Where q is the heat flux (W/m^2), T_{wi} is the inner surface temperature (^{o}C), and T_f is the fluid temperature (^{o}C) calculated as follows:

$$T_f = \frac{T_{in} + T_{out}}{2}$$

where T_{in} is the inlet temperature (°C) and T_{out} is the outlet temperature (°C) of fluid. As the outside wall temperature of the test section was measured at five axial locations. At each location, the temperature of the tube was measured at top and bottom positions.

$$T_{ws} = \frac{T_t + T_b}{2}$$

Thus, the average outside tube wall temperature of the test section, T_{wo}, was calculated as the arithmetic mean of outside tube wall temperature at five axial locations.

$$T_{wo} = \frac{\sum_{i=1}^{5} T_{ws}}{5}$$

Thermal resistance was used to measure the outside surface temperature of the tube, but the inner surface temperature was required to calculate the heat transfer coefficient. Therefore, according to Fourier's one-dimensional, radial, steady-state heat conduction equation for a hollow cylinder, based on the assumption that the heat flux is uniform inside the tube and a negligible heat loss to the surroundings, the inner surface temperature was calculated as follows:

$$T_{wi} = T_{wo} - \frac{Q_{test} \ln(d_{out}/d_{in})}{2\pi k l}$$

Where d_{out} is the outer diameter (mm), d_{in} is the inner diameter (mm), k is the thermal conductivity of copper (W/m°C), l is the length of the test section (mm), and Q_{test} is the heat flow to the test section (W) calculated from the voltage (*V*) and current (*I*) of the test section.

3.4 Experimental data validation

To verify the experimental data, obtained results for pure water have been compared with known correlations for the horizontal position ($\Theta = 0^0$) of test section. To examine the verification of obtained data related to single-phase convection region, Sider-Tate equation has been employed. Sider- Tate equation for predicting the forced convection heat transfer coefficient as follows [7]:

$$Nu = \frac{\alpha d_{in}}{k_l} = 0.0013\, Re^{1.2} Pr^{1/3} \left(\frac{\mu}{\mu_s}\right)^{0.14}$$

Results of this comparison demonstrate the well agreement of about 16% between the experimental data and the calculated results for single- phase convection zone. To examine the verification of obtained data related to nucleate boiling region, the experimental results of pure water were compared with the results obtained using the formula of Cavallini et al. [8]. The Cavallini pure working fluid experimental correlation is as follows:

$$Nu = \frac{\alpha d_{in}}{k_l} = 0.05\, Re_{eq}^{0.8} Pr_l^{1/3} Rx^2 (Bo.\ Fr)^{-0.26}$$

where,

$$Re_{eq} = 4M\left[(1-x) + x\left(\rho_l/\rho_g\right)^{1/2}\right]/(\pi d_{in}\mu_l)$$

$$Pr_l = \mu_l Cp_l / k_l$$

$$Rx = \{[2hn_g(1 - sin(\gamma/2))]/[\pi d_{in}\ cos(\gamma/2)] + 1\}/\ cos(\beta)$$

$$Fr = u_{GO}^2/(g d_{in})$$

$$Bo = g\rho_l h\pi d_{in}/8\sigma n_g$$

Results of this correlation express the agreement about +17% to - 4%with the experimental data. Fig. 3 and fig. 4 shows the result of comparison between experimental data related to pure water and well known correlations respectively.

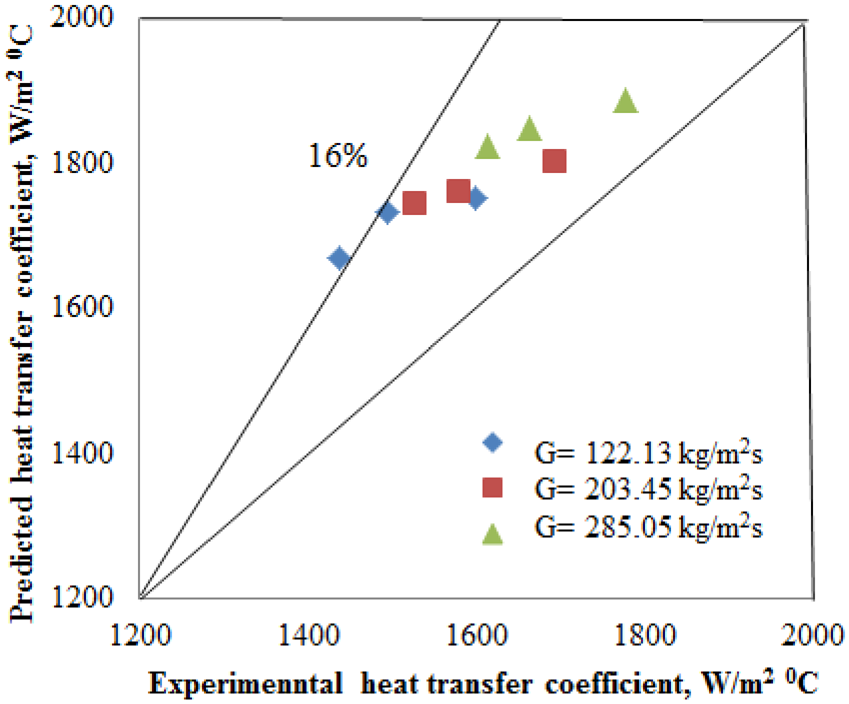

Fig. 3 Comparison of results obtained by Sider-Tate equation and experimental data in forced convective region

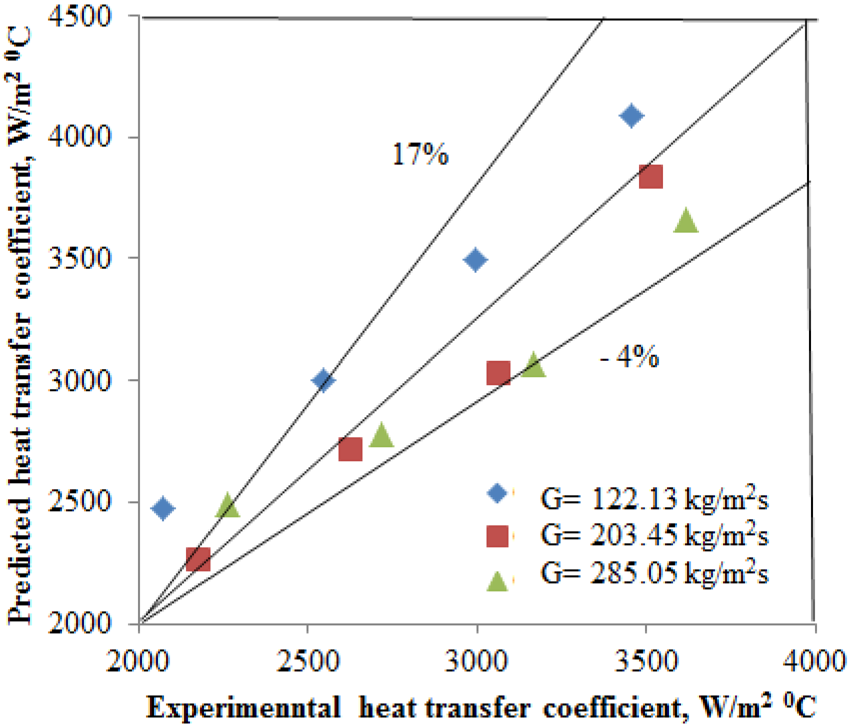

Fig. 4 Comparison of results obtained by Cavallini correlation and experimental data in nucleate boiling region

IV. RESULT AND DISCUSSION

The fluid enters the test section at 80^0C temperature. So the degree of subcooling ($\Delta T=20^0$C) is kept constant for whole experiment. In flow boiling two different regions of heat transfer has been considered: (1) convective region & (2) nucleate boiling region. There are many parameters affecting the flow boiling heat transfer coefficient. Accordingly, effect of heat flux, flow rate and inclination angle of test tube are separately discussed.

4.1. Effect of Heat Flux

Experimental data are shown in terms of heat transfer coefficient vs. heat flux. As shown in fig. 4, fig. 5 and fig. 6, with increasing heat flux, the flow boiling heat transfer coefficient increases. These increases are clearly observable in both convective and nucleate boiling zones. For convective heat transfer zone increase of heat transfer coefficient with heat flux is insignificant in comparison with nucleate boiling zone. In fig. 4, 5 and 6 for five different angle of test section, influence of heat flux on heat transfer coefficient is shown. As seen in convective zone, slope of changes of flow boiling heat transfer coefficient is less than that in nucleate boiling zone. The main reason for this is, at lower heat fluxes where no bubbles generated, hence lower heat transfer coefficient. As heat flux increases, the rate of bubble generation around the heated surface increases, so that there is rigorous interaction between bubbles at heated surface which induce the locally turbulence agitations and this results in heat transfer enhancement.

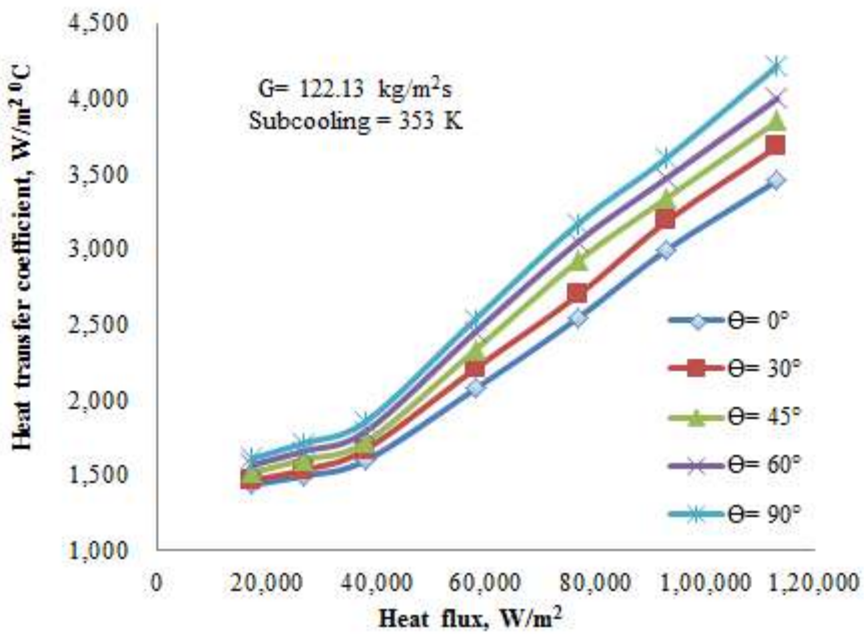

Fig. 5 Effect of heat flux and inclination angle of test section on heat transfer coefficient

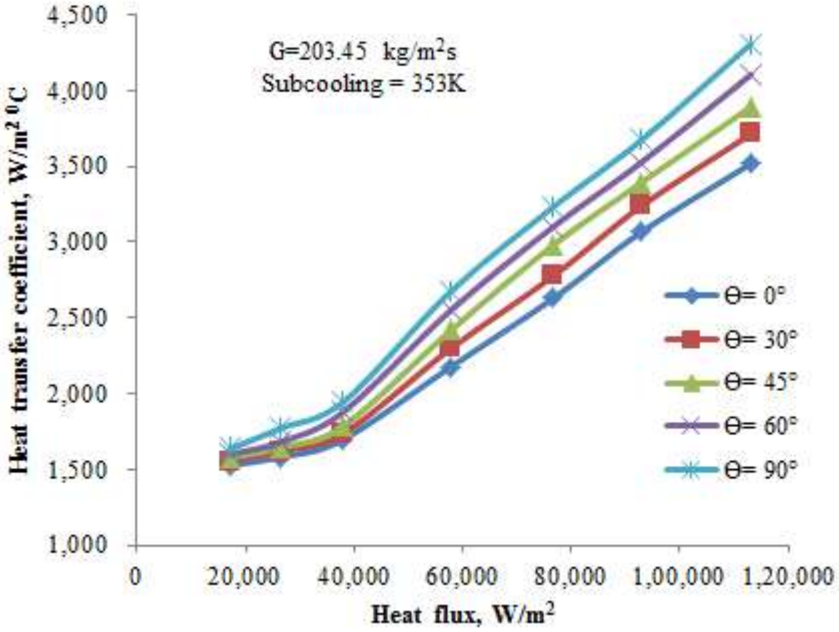

Fig. 6 Effect of heat flux and inclination angle of test section on heat transfer coefficient

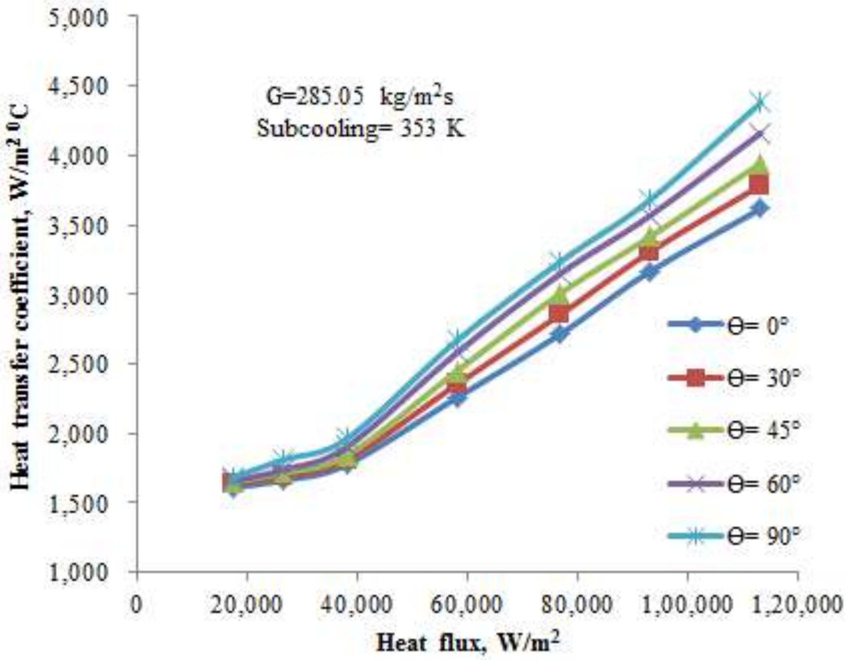

Fig. 7 Effect of heat flux and inclination angle of test section on heat transfer coefficient

4.2. Effect of Mass Flux

As in fig. 4, fig. 5 and fig. 6, flow boiling heat transfer coefficient increase with increasing mass flux in both convective and nucleate boiling regions. At all the mass fluxes the increase in the heat transfer coefficient in the nucleate boiling region is considerably higher than in the convective region. Also as the mass flux increases the difference in heat transfer coefficient for different inclination of tube goes on reducing in convective region. In nucleate boiling region, for vertical upward flow ($\Theta = 90^0$) the heat transfer coefficient increases with 4% for higher mass fluxin comparison with lower mass flux.

4.3. Effect of Inclination Angle of Test Section

As shown in fig. 4, fig. 5, and fig. 6, as the inclination angle increases the heat transfer coefficient also increases. For all flow rates, the increase in heat transfer coefficient with inclination angle 90^0in comparison with other inclination angles is probably due to the favourable flow of liquid inside grooves in convective region and due to high interfacial turbulence, also buoyancy force and fluid flow are unidirectional in nucleate boiling region, as a result the flow accelerates enhancing heat transfer from heated wall of the tube.

V. CONCLUSION

The effects of variation in heat fluxes, mass fluxes of water and inclination angle of test section on heat transfer coefficient were investigated experimentally inside threaded copper tube. The conclusions of this study can be summarized as follows:

1. The experimental results indicate that for all tube inclination angles, the heat transfer coefficient increases with heat flux and mass flux in both forced convective and nucleate boiling region.
2. The effect of tube inclination is also much more severe on the heat transfer coefficient. As the inclination angle increases the heat transfer coefficient also increases.
3. The highest heat transfer coefficient is attained at inclination angle of 90^0 (vertical upward flow).

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