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Investigation of Flow Boiling Characteristics of Surfactant/Water through Internal Threaded Copper Tube

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Abstract—*The subcooled flow boiling heat transfer characteristics of surfactant in an internal threaded copper tube were experimentally studied and compared to that of pure water. Triton X-100, non-ionic surfactant having negligible impact on environment, is taken as test surfactant and added separately in water with three concentrations i.e. 100,200 and 500 ppm. Heat transfer experiments were conducted with fluid inlet temperature of 80℃. The test runs were carried out by varying mass fluxes from 50 to 100 Kg/m^{2}s, heat fluxes between 10 and 90 kW/m^2 and tube inclination angles as 0° and 90°. The surface tension for water surfactant system was determined by Ostwald's Stalagmometer and viscosity by Brookfield viscometer. The addition of small amount of surfactant showed heat transfer enhancement upto 200 ppm concentration and above this range no heat transfer enhancement was observed. The result showed that the maximum heat transfer coefficient increased by 17.12% for horizontal position at mass flux 100 kg/m^{2}s.The entire study is carried out at low mass flux values. There is considerable enhancement in heat transfer coefficients in nucleate boiling region as compared to convective region at all mass fluxes and heat flux. The heat transfer coefficients predicted by some available correlations are compared with the present data.*

Keywords: *Flow boiling, surfactants, subcooled, inclined, internal threaded copper tube*

1. INTRODUCTION

Power generation and chemical process industries make extensive use of equipment like steam generators, heat exchangers and condensers which require rapid transfer of heat from a solid surface to a liquid. A lot of literature is available on methods of enhancing liquid- solid heat transfer rates, which allow the size, cost, and complexity of such equipment to be reduced. One promising approach has been the use of surfactants that increase the heat transfer rate by changing liquid properties. The boiling phenomena mainly depends on thermophysical properties of fluids that are density, surface tension and kinematic viscosity. The addition of small amount of surfactants doesn't affect the density of solution but it may slightly increase the viscosity in some cases. However it reduces the surface tension of liquid that results in nucleate boiling heat transfer enhancement remarkably. Also one of the passive techniques to enhance heat transfer rate is use of internal threaded tube. It can enhance the heat transfer by creating turbulence and limiting the growth of thermal boundary layer by slight increase in pressure drop. Thus by using surfactants in an internal threaded copper tube it is expected that it will result in heat transfer enhancement particularly in nucleate boiling region. The addition of surfactant may affect flow regimes with drag reduction thereby reducing frictional pressure drop. [1]

2. LITERATURE REVIEW

One of the earliest research works concerning the effect of surfactants on boiling phenomena is the study on flow boiling of water with a surfactant in a long tube vertical evaporator performed by Stroebe et.al.[2] in 1939.

Nomenclature

Bo	Bond number
d_{in}	inner diameter (m)
d_{out}	outer diameter (m)
Fr	Froude number
G	mass flux (Kg/m^{2}s)
h	heat transfer coefficient (W/m^{2}K)
I	current (A)
k_l	thermal conductivity of liquid (W/mK)
k	thermal conductivity of copper (W/mK)
l	length of test section (m)
Nu	Nusselt number
n_g	number of grooves
Pr_l	Prandtl number of water

Q_{test} heat flow (W)

q” heat flux (W/m^2)

Re_{eq} equivalent Reynolds number

R_X geometry enhancement factor

T_f fluid temperature (K)

T_{in} inlet temperature of water (K)

T_{out} outlet temperature of water (K)

T_{ws} outer surface temperature (K)

T_{wi} inner surface temperature (K)

T_{wo} average outer surface temperature (K)

u_{GO} velocity of gas phase with total flow rate (m/s)

V voltage (V)

x quality

μ dynamic viscosity (Pa-s)

μ_s dynamic viscosity at surface temperature (Pa-s)

g gravity

α heat transfer coefficient (W/m^2K)

ρ_l density of water (kg/m^3)

ρ_v density of vapour (kg/m^3)

σ surface tension (N/m)

γ apex angle

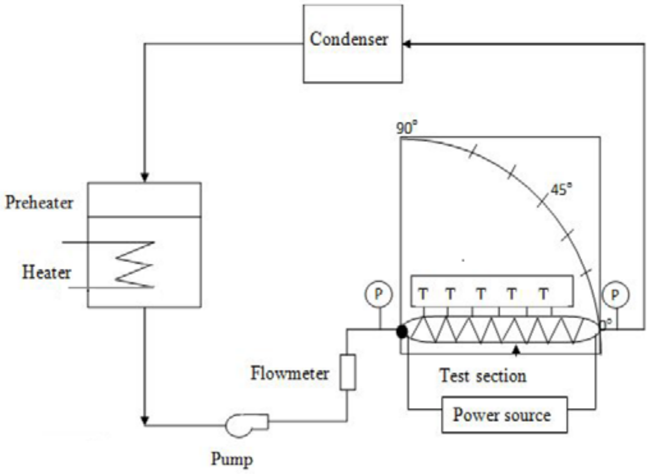
β helix angle

They found that flow boiling heat transfer coefficients were enhanced by addition of a surfactant Duponol and surface tension had a strong effect on the boiling heat transfer of water. Yang et.al. [3] took dynamic surface tension as one critical parameter in his study and investigated the effect of dynamic surface tension on boiling of aqueous surfactant solution. Frost and Kippenhan [4] also found that more boiling sites were nucleated and bubble growth was lower and thus flow boiling heat transfer was enhanced with Ultra wet 60L. Chang et.al. [5] investigated that flow boiling heat transfer was enhanced by addition of SLS. Hetsroni et.al. [6] conducted his study in vertical annular channel with Alkyl glycosides as surfactant and found that the average heat transfer coefficient may be enhanced upto four times. Klein et.al. [7] studied the effect of surfactants in microchannels. They used a kind of environmentally friendly surfactant and considered the environmental aspect. They concluded that an optimal value of mas flux was found for the flow boiling heat transfer. Quinn et.al. [8] studied the effect of surfactants on flow boiling heat transfer in a diverging open channel and found that surfactant solution created thick foam layer with high heat transfer rates and Nusselt numbers that are very weakly dependent on inlet flow rate or inlet Reynolds number. Akhavan Behabadi et al. [9] 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 heat transfer coefficient was highest when tube is horizontal or was inclined at -30^0. Bin sun et al. [10] 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 average heat transfer coefficient increased by 3-20%.

3. EXPERIMENTAL PROCEDURE

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℃, then it flows to test section through rotameter. The liquid- vapour mixture from outlet of test section flowed into the condenser, where it is condensed into liquid. The condensed liquid is passed through circulation pump to pre- heater. The pre-heater consists of a tank with 1.5 kW capacity heater installed in it and constant AC supply was given for preheating the fluid. The test section was heated by a nichrome wire heater (of 4 kW capacity) wrapped around the test tube. Heat input to heater was controlled with a variable AC voltage controller. The voltage and current flow was measured by analog meter to determine applied heat flux.



P- Pressure Transducer T- Thermocouple

Fig. 1: Experimental Set-up

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°; helix angle, 25°. The test section was heated by a cartridge heater (of 4 kW capacity) wrapped around the outside of test tube, and five cross sections 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 glass wool to reduce heat loss to the surroundings. Also two thermocouples and one pressure sensor to measure temperature and pressure drop at inlet and outlet of test section were used.

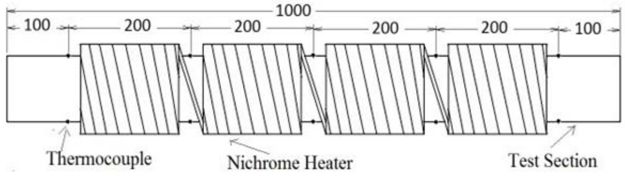


Fig. 2: Layout of Test Section

3.3 Procedure

The heat transfer coefficient was calculated by using following equation,

$$\alpha = \frac{q"}{T_{wi} - T_f} \quad (1)$$

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

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

where T_{in} is the inlet temperature (K) and T_{out} is the outlet temperature (K) of fluid. 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.

$$T_{ws} = \frac{T_t + T_b}{2} \quad (3)$$

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.

$$T_{wo} = \frac{\sum_{i=1}^{5} T_{ws}}{5} \quad (4)$$

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:

$$T_{wi} = T_{wo} - \frac{Q_{test} \ln(d_{out}/d_{in})}{2\pi k l} \quad (5)$$

where, d_{out} is the outer diameter (m), d_{in} is the inner diameter (m), k is the thermal conductivity of copper (W/mK), l is the length of the test section (m), 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 Uncertainty Analysis

The uncertainties of the experimental results are analyzed by the procedures proposed by the Schultz and Cole [4]. The method is based on careful specifications of the uncertainties in the various primary experimental measurements. The uncertainty in determination of the flow boiling heat transfer coefficients of the present study was found to be within ± 7.67 %.The detailed results from the present uncertainty analysis for the experiments conducted here are summarized in Table 1.

TABLE 1: SUMMARY OF THE UNCERTAINTY ANALYSIS

Sr. No.	Parameter	Uncertainty
1	Temperature (°C)	± 0.1
2	Pressure (bar)	± 0.001
3	Water flow rate (LPH)	± 1.2
4	Voltage (V)	± 10
5	Current (A)	± 1
6	Heat transfer coefficient (W/m2 K)	± 7.67 %

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.

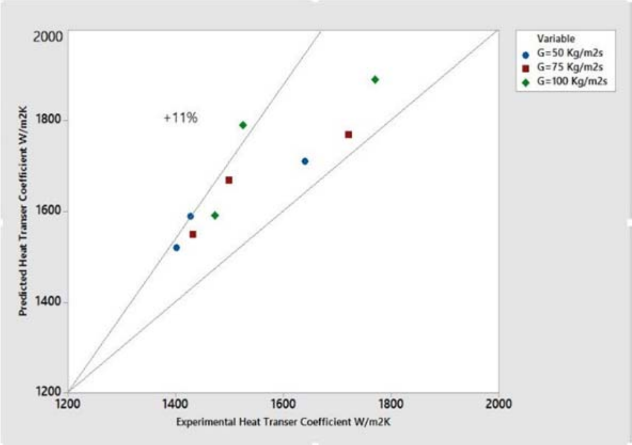


Fig. 3: Validation of forced convection region

Results of this comparison demonstrate the well agreement of about 11% 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.

$$Nu = \frac{\alpha d_{in}}{k_l} = 0.05\, Re_{eq}^{0.8}\, Pr_l^{1/3}\, Rx^2\, (Bo.\ Fr)^{-0.26} \quad (8)$$

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

$Pr_l = \mu_l Cp_l/k_l$

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

$Fr = u_{GO}^2\,/\,(gd_{in})$; $Bo = g\rho lh\pi d_{in}/8\sigma ng$

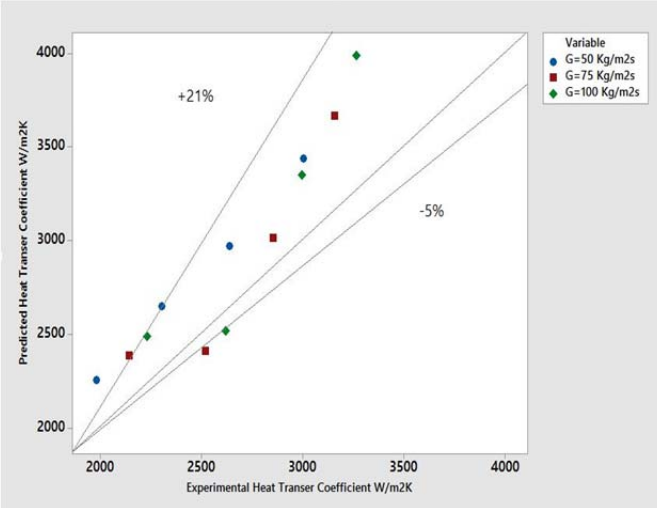


Fig. 4: Validation of nucleate boiling region

5. RESULTS AND DISCUSSION

The degree of subcooling ($\Delta T=20^0C$) 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 that are discussed below-

5.1 Effect of Concentration-

To investigate the influence of surfactant on heat transfer enhancement surfactants are added into water at three different concentrations i.e. 100,200 and 500 ppm. The obtained result shows that as concentration increases heat transfer coefficient increases upto critical micelle concentration. After cmc there is degradation in heat transfer enhancement. Due to addition of surfactants there is reduction in surface tension of water that results in easy bubble detachment from heated surface. For forced convective region, there is insignificant enhancement while for nucleate boiling region there is significant enhancement in heat transfer coefficient for 200 ppm concentration. For forced convective region heat transfer enhancement of 1.4 – 4.77 % and nucleate boiling region 6.88 -17.12% enhancement is obtained for mass flux 100 kg/m^{2}s.

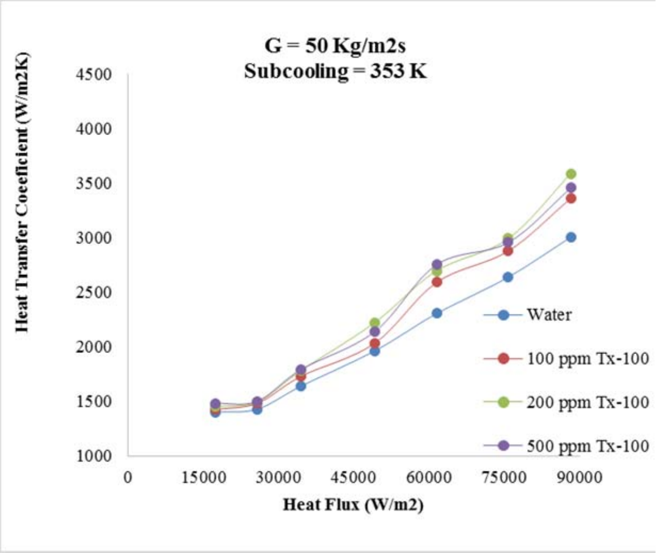


Fig. 5: Effect of surfactant concentration on heat transfer coefficient

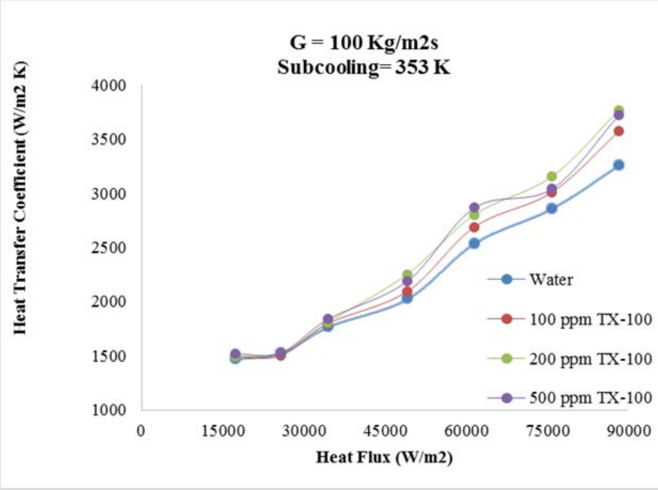


Fig. 6: Effect of surfactant concentration on heat transfer coefficient

5.2 Effect of heat flux -

As heat flux increases heat transfer coefficient increases for both convective and nucleate zone. The increase in heat transfer coefficient in convective zone is not significant because at lower heat flux no bubbles are generated. As heat flux increases rate of bubble generation and detachment also increases and it results in local turbulence agitations. There are drastic changes in slopes of graph for nucleate boiling zone.

5.3 Effect of mass flux-

The entire study is carried out at low values of mass flux. The enhancement in heat transfer coefficient for nucleate boiling zone is more than connective zone. There is slight decrease in enhancement for convective zone as mass flux increases. As concentration increases there is significant enhancement in nucleate boiling zone for higher mass flux.

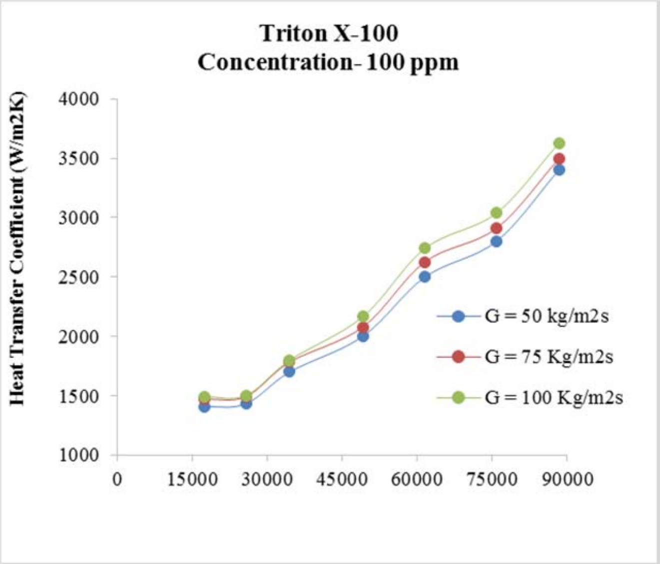


Fig. 7: Effect of heat flux and mass flux on heat transfer coefficient

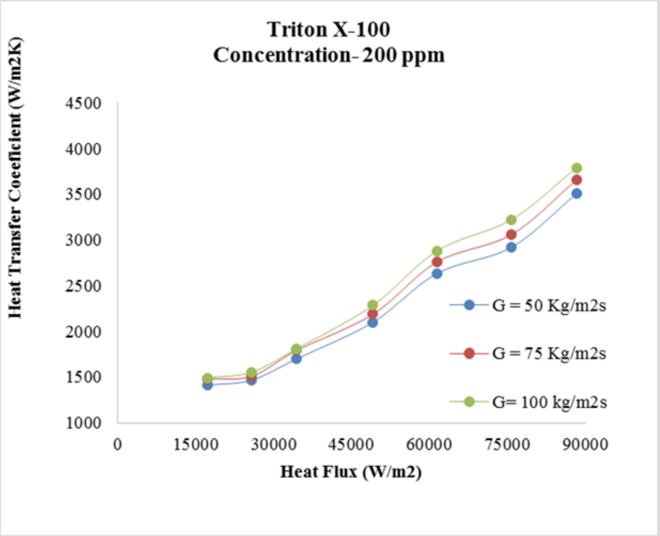


Fig. 8: Effect of heat flux and mass flux on heat transfer coefficient

5.4 Effect of inclination angle-

For all flow rate heat transfer coefficient increases for vertical position of test section. Since buoyancy force and fluid flow are unidirectional thus flow accelerates enhancing heat transfer in nucleate boiling zone. There is heat transfer enhancement of 12.5 % for water and 20.67% for water with surfactant in nucleate region for 200 ppm Triton X-100 solution with mass flux 100 kg/m^{2}s.

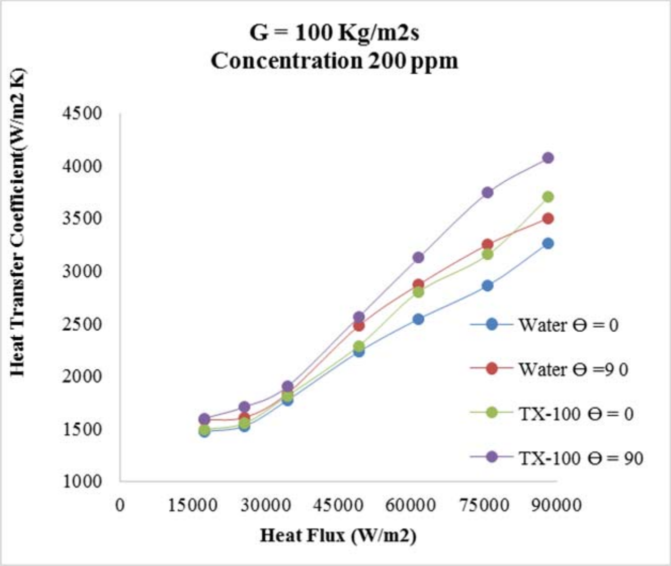


Fig. 9: Effect of inclination angle of test section on heat transfer coefficient

6. CONCLUSION-

The effect of variation in heat flux, mass flux, concentration of surfactant and inclination angle of test section were experimentally investigated. The result shows that

1. With increase in heat flux and mass flux there is considerable enhancement in heat transfer coefficient for nucleate zone as compared to convective zone. For mass flux 100 kg/m^{2}s and 200 ppm concentration heat transfer enhancement in convective zone is 3-4 % while in nucleate boiling zone it is 7-13 %.

2. Concentration of surfactant has significant effect on heat transfer enhancement. Upto cmc heat transfer enhancement has been observed but beyond this concentration there is degradation of enhancement. It may be due to viscous characteristics of surfactant. The surface tension of surfactant solution has remarkable effect on heat transfer enhancement. The maximum enhancement for horizontal position of tube is 17.12 % for cmc of surfactant at mass flux 100 kg/m^{2}s.

3. The inclination angle affects heat transfer coefficient in significant manner. The effect of inclination is prominent at high values of mass flux for nucleate boiling one. The heat transfer enhancement is highest i.e. 20.67 % at vertical position of test section with surfactant solution.

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