Investigation of Variation of Heat Transfer Coefficient of Water in Flow Boiling in Horizontal Plain Tube

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Abstract— An experimental facility is developed to investigate heat transfer coefficient in two-phase flow boiling of water in horizontal plain tube at different mass flux and heat flux. Experimentation include three type of arrangements of test section Internal diameter of tube 8 mm, Internal diameter of tube 6 mm ID 6 mm by varying angle of it at an angle of 100 from horizontal other specifications are same for all three like, length 1.5 m, heat flux ranges are from 2 KW/m2 to 10 KW/m2, the water mass velocities set to the discrete values in the range of 40-250 kg/(m2s) and the pressure at the inlet is close to atmospheric pressure so it is considered to be at atmospheric and respective temperature is consider to be the saturation temperature. The study analyses heat transfer, through the local heat transfer coefficient along the flow and its variation under different parameters. It is possible to observe effect of heat flux, mass flux on the heat transfer coefficient. As many familiar engineering application involve the boiling heat transfer like, household refrigerator condensers, heat exchangers, boilers and host of the process equipment, this study is very much important to improve performance.

***Keywords*:** Heat transfer, Flow Boiling, Varying Angle, Water, Calculations, Correlations.

I. INTRODUCTION

Study of heat transfer coefficient is having the number of application in the field of refrigeration. For designing evaporator system we must have the knowledge about the behavior of the refrigerant properties and best operating parameters to get more heat transfer rate in two-phase condition. Here water is used as a refrigerant, many researchers have done lots of work on water before. Purpose of this experimentation is to verify the results which were concluded by some researchers and mach that data with this data to see at what percentage it is different from the standard.

The interest in this experimentation is to decrease material and also operating cost. Variety of refrigerant which are present in the markets according to their application but cost of these refrigerants are comparatively much more than the water hence can't be used in the huge amount. Now a day's conventional ways are preferred more because they are more efficient and safe for both to humans and nature too. For using water as a refrigerant properties of water are need to be reconsidered at the two phase condition.

Designing of the condenser or the evaporator is require actual heat transfer rate which will help to reduce the size of the equipment otherwise problem of over-design or under-design of evaporator, condenser or such type of two phase process equipment may occurs. The correct prediction of the heat transfer coefficient in two phase flow is being considered as very important, as we seen from the past few year experimental investigations conducted in this field [3].

Why use water? this is first question comes in our mind that, we have so many other supplements for water available in the market then why prefer the water. Because water is not hazardous to human health so it can be used in the food industry and some applications like solar plants if we reduce its operating by reducing pumping cost.

Now days everything is gating compact in size so as the mechanical equipments but the efficiency is need to be the same, for that new techniques are used to enhance the heat transfer rate. There are number of refrigerants present according to the applications but still there are some areas like power plants where huge amount of water is used to generate an electricity at such places heat transfer rate of the water is need to be reconsidered.

Need for this topic for experimentation is first started in 1996 by chan for flow boiling heat transfer in vertical tube and still there are series of works have been recently pursued at the Purdue University boiling and two- phase flow laboratory (PU – BTPFL) based on the methodology that were used to predict critical heat flux (CHF) for water in tube [7].

Boiling process is categorized in two type pool boiling and flow boiling, in the pool boiling heat transfer rate can be calculated very easily but in case of the flow boiling it is hard to find out the heat transfer rate as the bulk of the fluid in contact with the hot surface is not the same it changes with the time hence it hard to calculate the heat transfer rate from the wall to the flowing fluid. Previously some researchers have done so much work on this and developed some relations using these relations one can find out the heat transfer coefficient at that section.

Flow boiling involves bulk motion of liquid and buoyancy effect that is why generalized theories are not available for the flow boiling due to various flow complexities i.e. bubble growth, separation, coalescence, effect of flow hydrodynamics, velocities and the variation in the flow patterns of the two phase flow. Although two-phase flow maps are available for distinguishing the flow regions within two phase flows such as slug and bubbly flow in the pipe but it should be noted that the subject of flow transition between these regimes is an area of active research. so recent activity carried out in order to investigate the behavior of the flow

boiling heat transfer in small diameter channels there is a still lack of information and reliable data of engineering design and application. Analysis of these earlier works shows that the major parameters affecting the HTC under flowboiling is heat flux saturation pressure and thermo-physical properties of the working fluid [8].

Designing heat treatment equipments like heat exchangers, two phase condensers and evaporators is very complicated task for calculating the exact heat transfer rate and for this exact analysis of the water properties are need to be studied apart from the issues like running performance and economic aspects. Reducing the production and running cost is the major challenge here also keeping size of the equipment as compact as possible without affecting to its performance. There are some other practical applications where heat exchangers are used like ocean thermal energy conversion plant (OTEC) and working fluid in this plant is water this plant requiresheat transfer area of nearly 10000 m2/MW. From the start of the 21st century one more parameter is added in the designing procedure which is environmental friendly. for testing the properties of the water three different arrangements are made-1] Plain copper tube is used of internal diameter 8 mm, length 1.5 m, thickness of the pipe is 1mm and pipe is placed horizontal. 2] In the second test only internal diameter of the pipe is changed to 6 mm keeping all the other specifications to be same.

3] In the third test 6 mm ID pipe is fixed at an angle of 100 from the horizontal and other specification are same as above.

II.EXPERIMENTAL APPARATUS AND PROCEDURE

The purpose of this experimentation is to determine the effects of different parameters like mass flow, heat flux on the hat transfer coefficient in the two-phase flow boiling. The detail of the facility is shown in the fig. The experimentation setup consists of different parts which are designed to vary the parameters in the different ranges so that number of readings can be taken. The main parts of the setup are pre-heater, centrifugal pump, flow meter, display board, condenser and test section. To assure that the pump should work continuously the placement of the pump is lower than the water level in the pre-heater so that cavitations problem can be avoided.

The pre-hater is used to establish the experimental condition entering the test section. The pre- heater is rectangular tank in to which an electric resistance heater is used to increase the temperature of the water near to the saturation temperature. This is done because increasing temperature of water from room temperature to the saturation temperature inside the pipe is difficult.

The test section is a copper pipe with internal diameter 8 mm and length of the test section is 1.5 meters. This test section is heated externally by using a flexible heating coil which is raped around the tube to get the uniform heating all over the surface. For avoiding the heat loss to the atmosphere thermal insulation is provided using fiberglass cover on the heating tape.

The test section is divided into five, and T-Type of thermocouples are used to measure the surface temperature of the tube, at each section there are 4 thermocouples attached along its periphery separated by an angle of 900 from each other, like there are four section so total 16 number of thermocouples are used. 4 k- Type of thermocouples are used at inlet, outlet, condenser and pre-heater one at each place.For measuring the pressure drop across the test tube differential manometer is used with the mercury as a manometric fluid.

The centrifugal pump is used to circulate fluid through the loop which gives the constant flow rate. To vary this a by-pass arrangement is provided so that the excess water can flows back to the pre-heater. For measuring the flow rate, flow meter is used with 0-100 LPH capacity.

2.1 Device overview:

In this device the main portion is test section which is a 1.5 m long copper pipe with outer diameter 10 mm as shown in the fig: .This contains 4 thermocouples at one section along the periphery of the circular pipe expressed as T1, T2, T3, T4 like that there are four different sections so total 16 thermocouples at the test section and two separate thermocouples at the inlet and outlet of the test section also one at pre heater isused. A flexible heating coil is wound around the pipe to provide uniform heating along the pipe and insulation is provided to reduce the heat loss to the atmosphere.

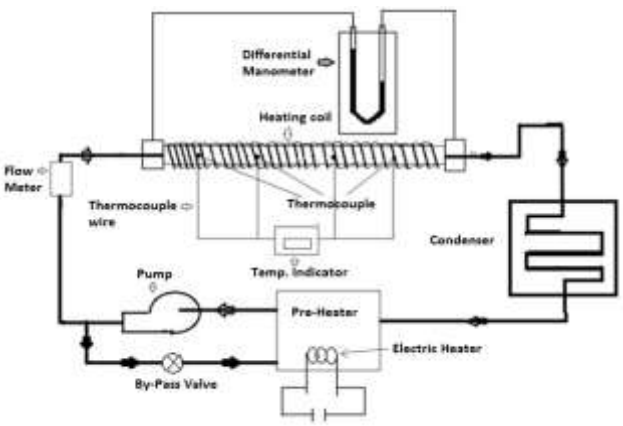

Fig.1 Experimental setup

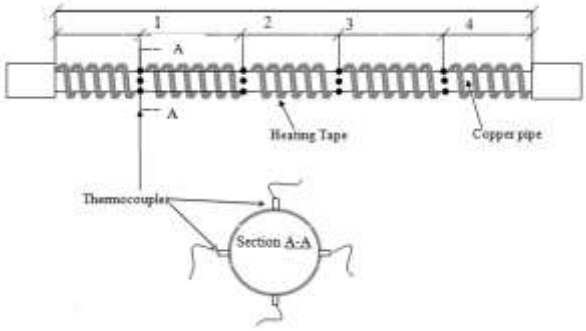

Fig. 2 Test section close-up view

III.MEASUREMENT PROCEDURE AND DATA REDUCTION

Efficiency: before taking main readings test section is tested on single phase condition to calculate heat transfer efficiency of the test section, as the heat is provided from heating coil total heat produced will not pass on to the copper some of it will dissipate in to the atmosphere. For calculating heat transfer efficiency following formula is used.

$$\eta = \frac{\dot{m}(h_{out} - h_{in})}{P} \quad (1)$$

Where η is efficiency, $\dot{m}$ is mass flow rate, p is electric power input and h_{out} and h_{in} are enthalpies at outlet and inlet of the test section. Enthalpy can be determined from the steam table using temperature and pressure values.

Heat efficient is directly promotional with mass flow and inversely with the power supplied hence average heat efficiency of test section is considered and it expected to be in between 80 to 90%. This will be used in all the further calculations.

Experimental conditions: water is used as working fluid in this experimentation, two copper pipes with diameter of 8mm and 6mm are used. Saturation temperature of the water is achieved at the pre-heater.

Data Reduction : In this experimentation data will be gathered which include parameters like mass flow rate, wall temperature at the different test sections, pressure at the inlet, outlet and some other parameters which are used to calculate the heat flux, enthalpies and heat transfer coefficient using thermodynamic property table of the water.

These calculations are based on the following considerations for calculating heat transfer coefficient.

- Pressure drop along the inlet and outlet of test section is linear function of tube length.
- Heat transfer only in radial direction is considered and in te axial direction is neglected.
- Heat flux is uniform along the tube in test section due to volumetric heat generation.

Enthalpy at the entrance of the test section is directly calculated from the property table directly, for calculating the vapor quality in the first part and the consecutive parts of the test section following formula are used.

$$h_1 = (q_1\eta / \dot{m}) + h_i \quad (2)$$

$$X_1 = h_1 - h_f / h_{fg} \quad (3)$$

Where h_1 is the enthalpy at the first section, q_1 is heat supplied by the heater, η efficiency of heat transfer, h_i is enthalpy at inlet of test section, h_f and h_{fg} are the enthalpies of liquid and mixture of both liquid and vapor respectively and X_1 is the vapor quality in the first section similarly,

Section 2

$$H_2 = (q_2\,\eta / \dot{m}) + h_1 \quad (4)$$

$$X_2 = h_2 - h_f / h_{fg} \quad (5)$$

Section 3

$$H_3 = (q_3\eta / \dot{m}) + h_2 \quad (6)$$

$$X_3 = h_3 - h_f / h_{fg} \quad (7)$$

Section 4

$$H_4 = (q_4\eta / \dot{m}) + h_3 \quad (8)$$

$$X_4 = h_4 - h_f / h_{fg} \quad (9)$$

Local heat transfer coefficient h is calculated by using the following formula.

$$h = (q''\,\eta / T_{wi} - T_{sat}) \quad (10)$$

where T_{wi} and T_{sat} are the internal wall temperature and saturation temperature at the local pressure calculated respectively and q" is the imposed heat flux.

At each section along the test section the wall temperature is considered to be the average of the four temperatures measured across the section it can be calculated as,

$$T_{wo} = (T_{w,top} + T_{w,R\ side} + T_{w,L\ side} + T_{w,bottom}) / 4 \quad (11)$$

Where T_w represents wall temperature and subscripts top, R side, L side, bottom represents top, right side, left side and bottom position temperature at each cross section along the test section. T_{wi} is the inner wall temperature of the test section which can be calculated by assuming radial condition to the wall, subjected to the internal heat generation.

$$T_{wi} - T_{wo} \quad \left(\frac{qD_i}{4k}\right)\left(\frac{(D_i/D_o)^2 \quad 2ln(D_i/D_o) \quad 1}{1-(D_i/D_o)^2}\right) \qquad (12)$$

3.1 Uncertainty analysis:

The uncertainty associated with sensors used in the experimentation is estimated by considering accuracy of the different sensors as mentioned in the table 1. Some test runs are performed on setup by changing parameters to calculate the uncertainty for the heat transfer coefficient. The maximum uncertainty for heat transfer coefficient calculated is (h_{max})35.3% the uncertainty is decreases with increase in heat flux when mass velocity is constant.

Table 1
Measuring equipments

Variable	Equipments	Accuracy	Range
Mass flow rate	Rotameter	±1 lph	0 to 100 lph
Temperature	Thermocouples	$\pm 0.03^0$ C	0 to 300^0 C
Pressure difference	Pressure sensor	±0.05 bar	0 to 1
Voltage	Voltmeter	±0.2%	0 to 30 V
Current	Amperometer	±0.2%	0 to 220 A

IV. RESULTS AND DISCUSSION

In this study the flow boiling of water inside the horizontal plain tube with inside diameter 6mm at the atmospheric pressure. Here different mass flux.

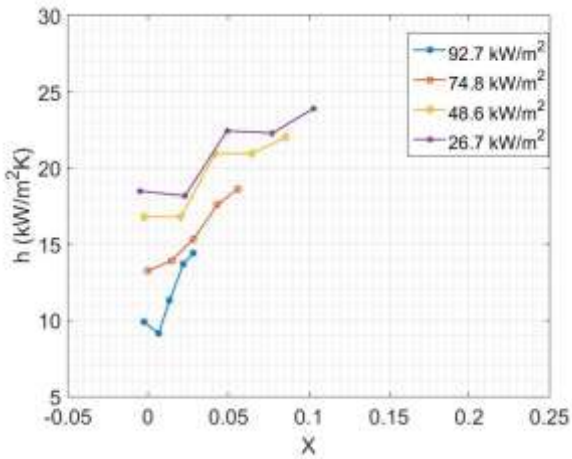

Fig. 3 Heat transfer coefficient Vs quality (211 kg/m^{2}s)

These graphs are results of test data obtained from the wall temperatures at different section, heat flux and mass flow inside the 6mm diameter tube and 8mm dia. Graph shows that at the low dryness fraction value of the heat transfer coefficient is likely to be more dependent on it.

Results in the Fig. 4 shows there are five different mass fluxes which shows that heat transfer coefficient have negligible effect of mass flux for the wall superheat up to the 5^0 C. As per the heat balancing the heating length changes with the inlet temperature readings are also following the same here.

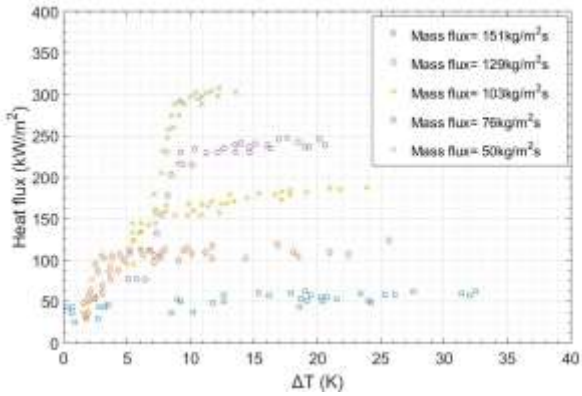

Fig. 4 Heat flux Vs wall superheat

Heat transfer coefficient above the 8^0 C is dependent on the mass flux but it is having minimal effect of degree of wall superheat.

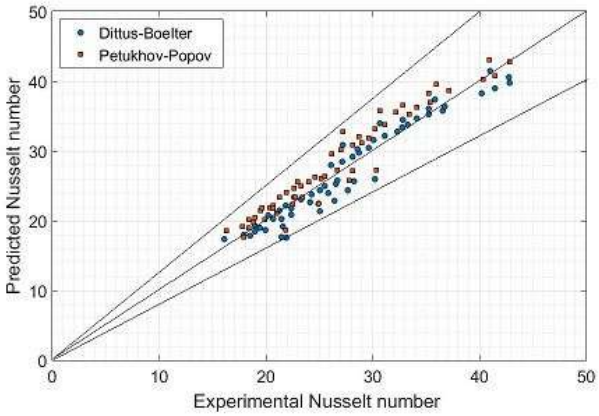

Fig. 5 Predicted Nusselt number Vs Experimental Nusselt number

In the Fig. 5 shows results for the single phase heat transfer, here predicted Nusselt number and experimental Nusselt number are compared and all data is under the ±20%.

V.CONCLUSIONS

In this experimentation flow boiling heat transfer coefficient for the water is calculated at atmospheric pressure for different mass flow rate and heat flux. Dryness fraction is varying from 0 to 0.25 considering these parameters some result are concluded which are as follows.

1) Heat transfer coefficient is more dependent on the heat flux as compared to the mass flux but it is under-predicted for the low heat flux. The under-prediction is decreases with increase in heat flux.
2) The pressure drop in the smaller diameter tube is lower as compared to the larger diameter for the two phase flow at the same mass flux. This comparison is done by using Chisholm correlation for larger channels.

REFERENCES

1] Kandlikar S.G.(2004) “Heat Transfer Mechanisms during flow boiling in microchannels”, journal of Heat Transfer, vol.126, pp.8-16.

2] Thome J.R.(2006) “stat of the art overview of boiling and two phase flow in microchannel”, Heat Transfer Engineering,27(9),pp.4-19.

3] Lee J., Mudawar I.(2005) “Two phase flow in high-heat-flux microchannel heat sink for refrigeration cooling application: part I-pressure drop characteristics, part II-heat transfer characteristics", Int. Journal of heat and mass transfer, 48,pp.928-955.

4] Samuel M. Sami, Joseph D. Comeau(2002) "Influence of the thermo-physical on two-phase flow convective boiling of refrigerant mixture”, applied thermal engineering 22, pp.1535-1548.

5] J. C. Chen(1966) “Correlation for boiling heat transfer to saturated fluids in convective flow,” I&EC Process Design and Development, **5** (3), pp. 322-329.

6] M. M. Shah(1982)“Chart correlation for saturated boiling heat transfer: equations and further study,” ASHRAE Transactions, **88**, pp.185-196.

7] D.L. Bennett, M.W. Davies, and B.L. Hertzler(1980)"The suppression of saturated nucleate boiling by forced convective flow," American Institute of Chemical Engineers Symposium Series, **76** (199), pp. 91-103.
8] K. E. Gungor and R. H. S. Winterton(1986)"A general correlation for flow boiling in tubes and annuli," Int. J. Heat Mass Transfer, **29** (3), pp. 315-358.
9] S. G. Kandlikar(1990) "A general correlation for saturated two-phase flow boiling heat transfer inside horizontal and vertical tubes," J. Heat Transfer, **112**, pp. 219-228.
10] D. Steiner and J. Taborek(1992)"Flow boiling heat transfer in vertical tubes correlated by an asymptotic model," Heat Transfer Engineering, **13** (2), pp. 43-68.
11] Arjit Kundu, Ravi Kumar and Akhilesh Gupta(2014) "Flow boiling heat transfer characteristics of R407C inside smooth tube with different tube inclinations." International journal for refrigeration 45,pp I-I2.
12] Wojtan, Ursenbacher, Thome(2005) "Investigation of flow boiling in horizontal tube: part II-development of new heat transfer model for stratified wavy, dry-out and mist flow regimes", Int. journal of heat and mass transfer48,pp.2970-2985.

APPENDIX

Notation

A Heat transfer area (m^2)
C_p specific heat at constant pressure (J/kg K)
$\dot{m}$ mass flow rate (kg/m^2s)
P pressure (Pa)
T temperature
K thermal conductivity (W/m K)
q" heat flux (W/ m^2)
Q heat transfer rate (W)
x vapor quality
H heat transfer coefficient (W/ m^2 K)
ΔP pressure drop
Re Reynolds number
d diameter of pipe (m)
l length of test section (m)

Greek symbols

f friction factor
Ø two phase multiplier
ṇ efficiency
ρdensity (kg/m^3)
μ dynamic viscosity (pa s)

Subscripts

tp test pipe
ph pre-heater
i internal
wi inner wall
sat saturation
G vapor
L liquid
h enthalpy

