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EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER TO AIR–WATER FLOW IN DOWNWARD INCLINED SQUARE PIPE

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ABSTRACT

Two-phase local heat transfer coefficient and flow pattern in 4.5 cm inner length of glass square test section downwardly pipe inclination angles of 5°, 10°, and 15° were investigated experimentally. Two superficial velocities for each air and water were used 0.164 m/s, 0.411 m/s and 1.1506 m/s, 1.4815m/s respectively. Local heat transfer coefficient was calculated from the temperature measured by using temperature recorder with SD memory card data logger with sensor type K thermocouple positioned along the test section. It was found that the heat transfer coefficient of air-water flow increase with increasing flow rate of water, air, and inclination angle. Maximum local heat transfer coefficient calculated where superficial velocity of water and air 1.4815 m/s, 0.411 m/s respectively and inclination angle 15° . In addition a high speed camera was used in order to obtain images sequence of the flow under different selected conditions. The flow regimes, which are stratified, intermittent, and annular flow are observed and recorded by high speed camera. At 5° inclination angle and low air-water superficial velocity stratified flow observed, transition from stratified to intermittent flow and annular flow occurs at increasing the inclination angle and air-water superficial velocity. Correlations on average Nusselt number were obtained for three inclination angle 5°, 10°, and 15°.

KEYWORDS: Local Heat Transfer Coefficient; Two-Phase Flow; Inclined Pipe; Downward Flow; Square Pipe; Flow Regime.

INTRODUCTION

Two–phase flow is a characteristic term for a gas-liquid, gas-solid, or a liquid-solid flow, flowing simultaneously in a pipe, channel, or other conduit. Gas-liquid two-phase flow is the best described as the interactive motion of two different kinds of media. They are extremely important in many industrial applications: solar collectors, chemical plants, nuclear reactors, oil wells and pipelines, etc. The knowledge of non-boiling two-phase, two-component (liquid and permanent gas) heat transfer is required. When a gas-liquid mixture flows in pipe, a variety of flow pattern may occur, depending primarily on flow rates, the physical properties of the fluids, and pipe inclination angle that reported by (Ghajar, 2005).

Downward simultaneous flow of gas and liquid, although rare, is important in the chemical process industry and also in petroleum production. Gas-liquid two-phase flow heat transfer and flow patterns in inclined pipes have been studied by many researchers.

Experimental work with 10.16 cm diameter pipe, 36 m long ±2°, ±15°, and ±30° inclinations angle, containing oil and carbon dioxide as working fluids was observed by (Kang et al., 2002) Superficial oil velocity between 0.2 and 2.0 m/s and superficial gas velocity between 1 and 4 m/s were used. They were observed that the dominant regime for upward inclinations was slug flow. No stratified flow was observed for these conditions. As the angle of upward inclination increases from 2° to 15°, the transition from plug to slug flow occurs at higher superficial gas velocities. As the angle of downward inclination increases, the transition from stratified to slug flow occurs at higher liquid and gas flow rates. The transition from slug to annular flow occurred at slightly higher gas flow rates when inclination was changed from -2° to -15°. However, little difference was found during the transition from slug to annular flow when the inclination was changed from -15° to -30°.

Experimental work to study the flow regimes and heat transfer in air-water flow in 8° inclined tube of inner diameter of 49.2 and 25 mm were performed by (Hetsroni et al., 2003) The thermal pattern on the heated wall and local heat transfer coefficients were obtained by infrared thermography. They observed the local heat transfer coefficient on the upper part of the pipe is described by empirical correlation with standard deviation of 18%. Under conditions of dry out in open annular air-water flow the heat transfer coefficient is about 2.5 to 8 times higher than that for single-phase airflow. Two-phase air-water flow and heat transfer in a 25 mm diameter horizontal pipe, the water superficial velocity varied from 24.2 m/s to 41.5 m/s and the air superficial velocity varied from 0.02 m/s to 0.09 m/s were investigated experimentally (Zimmerman et al., 2006) The flow patterns were visualized using a high speed video camera, and the film thickness was measured by conductive tomography technique. The heat transfer coefficient was calculated from the temperature measurements using the infrared thermography method. It was found that the heat transfer coefficient at the bottom of the pipe is up to three times higher than that at the top, and becomes more uniform around the pipe for higher air flow-rates. Correlations on local and average Nusselt number were obtained and compared to other authors. The behavior of local heat transfer coefficient was analyzed and the role of film thickness and flow patterns was clarified. A 27.9 mm stainless steel test section with a length to diameter ratio of 100 in horizontal and slightly upward inclined (2^o, 5^o, and 7^o) positions, many experiments have been carried out by (Ghajar and Tang, 2007). The heat transfer data points were collected under a uniform wall heat flux boundary condition ranging from about 1800 to 10900 W/m^2, the superficial Reynolds numbers ranging from 740 to 26000 for water and 560 to 48000 for air. They observed that the heat transfer coefficient was significantly depending on the superficial liquid, gas Reynolds numbers, inclination angle, and flow pattern.

The experimental data indicated that even in a slightly upward inclined pipe, there is a significant effect on the two-phase heat transfer of air-water flow. Two-phase air-water flow in test pipe (6.5 m) long positioned at angles between -20^o downwards and vertical upwards was described by (Perez, 2007) Two pipe diameters were used 38 mm and 67 mm, superficial velocities for air ranged from 0.15 to 8.9 m/s and from 0.04 to 0.7 m/s for water. High speed video system was used to obtain image sequence of the flow under different selected conditions. It was found that for upward inclined flow most of the experiments fall within the slug flow regime whereas for inclined downward flow the dominant flow pattern is stratified flow. An effect of the pipe diameter was also found under certain flow conditions mainly on the liquid holdup, pressure drop and structure velocity. He was investigated that increase pipe diameter displaces the bubbly-slug transition to the right hand side on the flow pattern map for inclined flow, and for horizontal pipe the stratified-slug transition is moved up. A CFD code was used to successfully model the hydrodynamics of the slug flow pattern, using the volume of fluid model based on the Euler-Euler approach.

The principal interest of this study was to identify the effect of inclination angle in downward air-water flow on the heat transfer coefficient of two phases as well as study the effect of inclination angle in the pattern of the flow. Connection between flow parameters and heat transfer was also studied.

Experimental Apparatus and Procedure

Experiments were carried out in a smooth transparent test section glass pipe with square cross section. The square test section with an inner length of 4.5 cm, outer length of 6.5 cm, hydraulic diameter is 4.5 cm, and over length of 400 cm was used in the experiments. The inclination angles of the test section are 5^o, 10^o, and 15^o. The schematic description of the experimental facility is presented in Fig. 1. Water and air is used as working fluids. The water and air supply systems work independently. Water in the system was controlled by water flow meter with different measure range. The water was heated to 60 ^{o}C by two heaters, and was pumped to mixer, and test section.

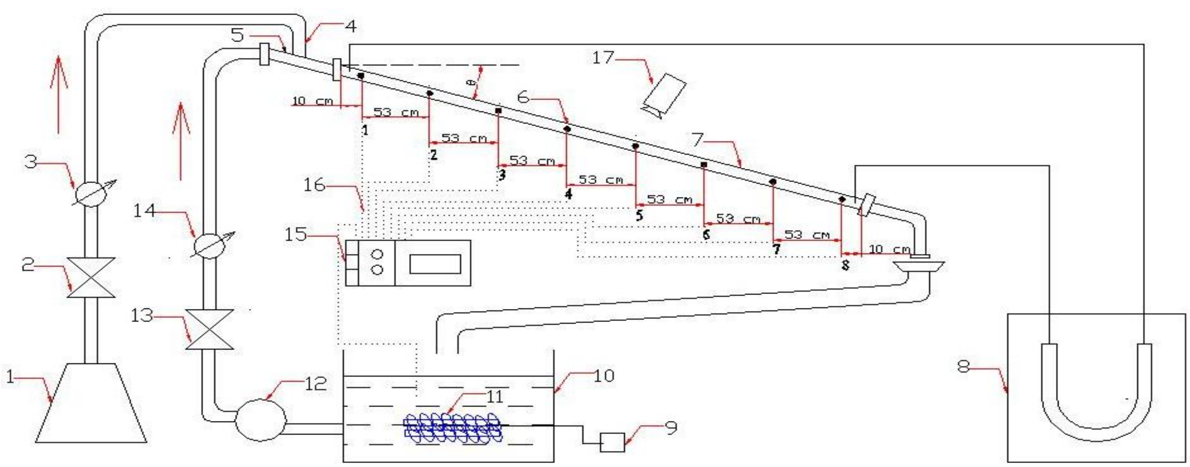


Fig. 1. Experimental facility: (1) compressor, (2) air valve, (3) air flow meter, (4) air injector, (5) air-water mixer, (6) thermocouple, (7) test section, (8) U-tube manometer, (9) power supply, (10) water tank, (11) heater, (12) pump, (13) water valve, (14) water flow meter, (15) data logger, (16) 12 channels temperature recorder device, (17) camera.

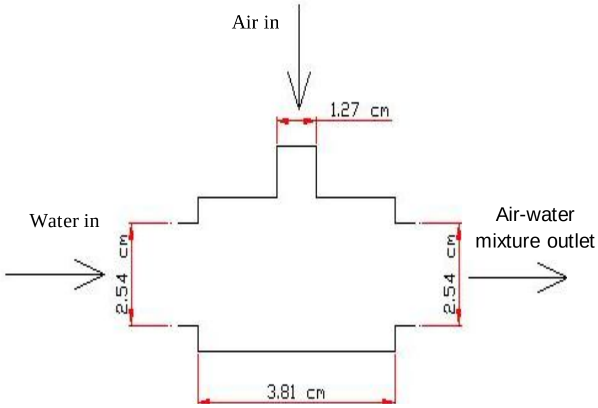


Fig. 2. Schematic diagram of air-water

The air at 30 ^{o}C and atmospheric pressure from the air compressor was pumped through air valve, the air flow meter, the air-water mixer, and the test section. Air-water mixer was made of mild steel. Both the air and water were brought together in a mixer. Water flows through the mixer at diameter of 2.54 cm, and air from 1.27 cm of diameter, and overall length of 3.81 cm. The test section of air-water mixer was shown in Fig.2.

Eight thermocouples type K (chromium^{-} - aluminum^{+}) were fixed inside the flow of test section on the equal distance between them 53cm to measure the two-phase air-water temperature. Temperature recorder with SD memory card data logger with sensor type K thermocouple was used to read the temperature with time. This device has calibration certificate with accuracy (± 0.4% ^{o}C + 1 ^{o}C). Flow pattern was detected by using a digital camera, which was used to take video capture for the flow. It uses an SD card memory with 8 GB capacity for recording and it has 14.1 Mega pixels.

All measurement equipments were calibrated by measuring actual reading with different instruments. Then, the water mass flow rate was measured with flow meter (accuracy ± 3%). Also, the air mass flow rate was measured with flow meter (accuracy ± 0.15%). Experiments were carried out at two values of water superficial velocities: 1.1506 m/s and 1.4815 m/s .Then, for each water velocity two values of air superficial velocities were used (0.164 m/s and 0.411 m/s) through every experimental test. For each combination of water and air velocities experiments were carried out for the following values of pipe inclination angles: 5^{o}, 10^{o}, 15^{o}. For each inclination angle of the pipe and for each air and water velocity considered temperatures were recording from all eight thermocouples to calculate heat transfer coefficient of two phase flow experimentally. Photograph type-of flow happened run in each case was detected by using a digital camera.

Analytical Considerations

A heat transfer coefficient for the two-phase flow was calculated experimentally as reported by (Ghajar and Tang, 2007) :

$$h_{TP_{EXP}} = \frac{1}{L}\int \bar{h}\ dz = \frac{1}{L}\sum_{k=1}^{N_{ST}} \bar{h}_k\ \Delta z_k \qquad (1)$$

Where $Q = h\ A\ \Delta T$, then

$$h_{TP_{EXP}} = \frac{1}{L}\sum_{k=1}^{N_{ST}} \left(\frac{Q}{A\,(T_w - T_b)}\right)_k \Delta z_k \qquad (2)$$

Where $\bar{h}$ (W/m^{2}.^{o}C) is the local mean heat transfer coefficient, L (cm) is the length of the test section, k is the index of the thermocouple stations, N_{ST} is the number of the thermocouple station, and Δz (cm) is the distance between each thermocouple and equal to 53 cm. Fig. 3 shows the axial distance along the test section.

Q (W) is the electrical power supply to heat the water in the tank until 60 ^{o}C, a single digital voltmeter device was used to measure the voltage, and the current was measured by single digital ammeter device. Then, the electrical power (P=VI) supply to heaters as electrical energy rate Q, which equal (4034 W). A is the wetted area in each thermocouple station (cm^{2}), which was calculated (A= 4×a×b) as shown in Fig. 4. T_b is the bulk temperature in each thermocouple station, which equal the temperature recorded by temperature recorder with SD card data logger with time (air-water temperature). $\bar{T}_w$ is the wall temperature of the test section.

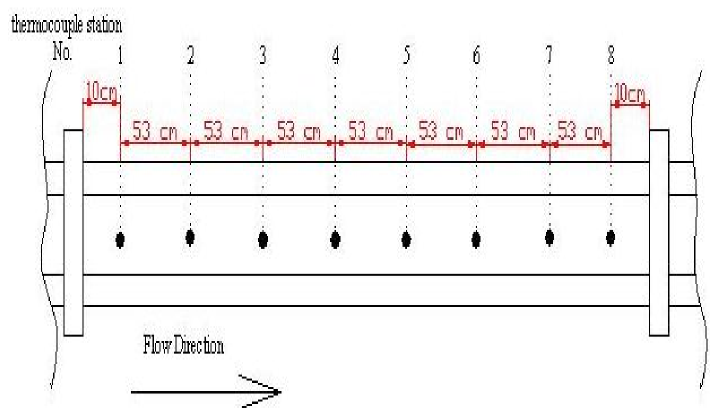


Fig.3. Axial distance along the test section

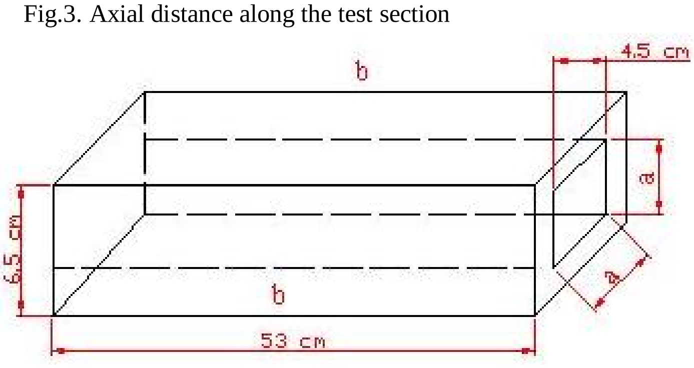


Fig. 4. Element of wetted area

All the temperature readings with time in the experimental work of these channels were calibrated with accuracy factor ±1.004 in order to get the true value of temperature readings. $\overline{T}_w$ is the wall temperature of test section in each thermocouple station its calculated by using finite difference analysis by using heat conduction balance at steady state condition eq. (3). In this calculation, axial conduction was assumed negligible, but heat conduction through glass test section wall was included as shown in Fig.5.

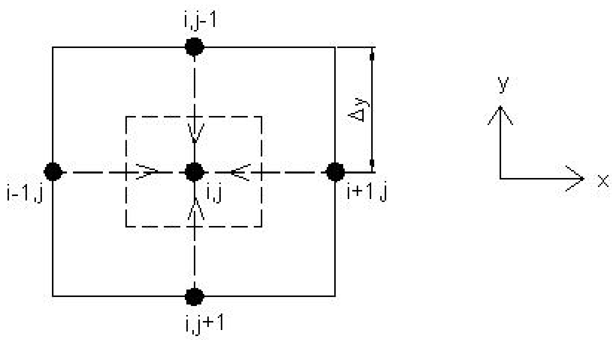


Fig. 5 Finite-difference grid arrangement

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = 0 \qquad (3)$$

Represent eq. (3) in finite difference form

$$\frac{T_{i+1,j} - 2T_{i,j} + T_{i-1,j}}{\Delta x^2} + \frac{T_{i,j+1} - 2T_{i,j} + T_{i,j-1}}{\Delta y^2} = 0 \qquad (4)$$

As assumption the axial conduction negligible, then eq. (4) becomes

$$\frac{T_{i,j+1} - 2T_{i,j} + T_{i,j-1}}{\Delta y^2} = 0 \qquad (5)$$

Where $T_{i,j-1}$ was represented the ambient temperature of laboratory, $T_{i,j}$ was the wall temperature of test section, which was calculated by eq. (5), and $T_{i,j+1}$ was represented the air-water temperature measured by sensor .

The non-dimensional parameters chosen for the analysis are liquid Nusselt number (Nu_L), liquid Froude number (Fr_L), and the gas Reynolds number (Re_G). (Nu_L), (Re_G) are calculated by the laws which are described by (Zimmerman et al., 2006) as:

$$Nu_l = \frac{hD_h}{k_l} \qquad (6)$$

Where h (W/m^2 ^{o}C) is the heat transfer coefficient, D_h is the hydraulic diameter of test section 4.5 cm, and k_L (W/m ^{o}C) is the thermal conductivity of the water at 60 ^{o}C.

The Reynolds number of air (Re_G) was calculated as:

$$Re_G = \frac{u_G D_h}{\nu_G} \qquad (7)$$

Where u_G (m/s) is the velocity of air, D_h (cm) is the hydraulic diameter of test section, ν_G (m^2/s) is the kinematics viscosity of air at 30 ^{o}C

The Froude number of liquid was presented by (Malhotra, 2004) as:

$$Fr_l = \frac{u_l}{\sqrt{g\, D_h \sin\theta}} \qquad (8)$$

Where u_L (m/s) is the velocity of water, g is the acceleration gravity (9.81 m/s^2), θ is the inclination angle of test section, and D_h (cm) is the hydraulic diameter of test section.

RESULT AND DISCUSION

Heat Transfer Results

Figures (6) and (7) represent the thermocouple readings along the test section at different stations for test run in two-phase flow from values of water Reynolds numbers 108094.106 to 139180.791 and with two values of air Reynolds numbers 461.222 and 1153.055 with different angle values of inclination at interval time 5 seconds. Each figure shows the relationship between average temperature of air-water and the distance of thermocouple station along the square test section. The response was a downward relationship. The average temperature of air-water flow decreases at different station in test section. This is due to the heating water was cooled down by air. The heat was dissipation from the wall of the rig tube by heat transfer. This dissipation of heat was calculated in the analysis in order to calculate local heat transfer coefficient in each thermocouple station along the test section.

It has been shown the level of average temperature of air-water decreasing with increasing the inclination angles and also with increasing the water superficial velocity, and air superficial velocity (i.e. increasing the values of Reynolds numbers of water and air). Each run dependent upon three factors water velocity, air velocity and temperature of water in tank. These conditions will control on temperature readings of air-water flow. These experimental results were confirmed with the experimental results of (Malhotra, 2004).

Figures (8) and (9) illustrate the relationship between the local heat transfer coefficients of air-water flow at each thermocouple station along the test section. Each figure demonstrated upward relation with one value of water velocity, and two values of air velocities 0.164 and 0.411 m/s with inclination angle values are 5^o, 10^o, and 15^o. The local value of heat transfer coefficient was calculated through every thermocouple station dependent upon the analysis considerations, which were detailed in section 3. These results show that the calculation of local value of heat transfer coefficient increasing upward along the distance of test tube. This is due to the heat dissipation between air-water, and wall of the test section. That leads to decrease the temperature difference between the bulk temperature of two phases, and wall temperature of test tube. Then, the value of heat transfer coefficient was increased. This is due to the temperature difference was inversely proportional with heat transfer coefficient. These results show the value of (h) was increased with increasing the values of water velocity, and air velocity. Also, these results were pointed the effective of inclination angle values of a square test section on the values of (h) through the run test. It has been shown higher values of (h) with increasing the values of inclination angle.

It has been pointed from these two figures the value of (h) in station number 1 higher than another stations. This is due to the increment length from the starting of test tube to station number 1 was smaller than another increment length of others as shown in Fig.3. This produces smaller wetted area of this station than another station. Where the value of (h) was inversely proportional with wetted area as mention in eq.(2). These results consistent with result of (Zimmerman et al., 2006).

Figure (10) describes the relationship between the calculations the overall mean values of two-phase heat transfer coefficient in all thermocouples station. These were calculated through every run for water superficial velocity 1.150 to 1.481 m/s, and air superficial velocity 0.164 to 0.411 m/s. This figure show the influence of inclination angle on the overall mean ($h_{TP\ EXP.}$). Due to gravitational effect, inclining the tube affects the flow characteristics of two-phase flow, which in turn affects the air-water heat transfer coefficient ($h_{TP\ EXP.}$). Overall, the mean ($h_{TP\ EXP.}$) increased when the inclination angle value was increased. Also, it has been shown higher values of ($h_{TP\ EXP.}$) with increasing the air and water velocity through the experimental test. It has noted maximum value of($h_{TP\ EXP.}$) was calculated at 15^o, which was about 6900 W/m^2.^{o}C at water superficial velocity 1.48 m/s, and air velocity 0.411 m/s. These results were consistent with the results of many authors such as (Hetsroni et al., 2003 and Ghajar and Tang, 2007)

Figure (11) shows the dependence of the local Nusselt number on the parameter $Fr_L Re_G$ at $\theta=5^o$, $\theta=10^o$, and $\theta=15^o$. These results were calculated the average values of these parameters the local Nusselt number and $Fr_L Re_G$ with different conditions. These conditions are different values of water and air velocities through many run test of experimental work by inclining the square test section at three values of angle, from these curves of this figure, it has been produced the fitting equations for every value of inclining angle. The following equations were found to describe the experimental data:

$$Nu_L(5^o) = 4.953(Re_G Fr_L)^{0.089} \quad (9)$$

$$Nu_L(10^o) = 5.230(Re_G Fr_L)^{0.081} \quad (10)$$

$$Nu_L(15^o) = 5.755(Re_G Fr_L)^{0.027} \quad (11)$$

These results were conformed to the result of (Zimmerman et al., 2006).

Flow Pattern

It is not possible to understand the two-phase flow phenomenon without a clear understanding of the flow patterns encountered. Flow patterns play very important roles in two-phase flow. Each regime has certain hydrodynamic characteristic, occurrence in nature and many applications in industries. The observation in the present work are illustrated in Fig. 12 by a set of photograph that show mainly the effect of inclination angle ,change the air and water flow rate on the flow pattern. The observed flow patterns can be described as follows:

1. Stratified smooth flow: For low superficial air and water velocities the flow is gravity dominated, and the phases are segregated. Fig 12 (a), and (g) illustrate stratified smooth flow v_w =1.1506 m/s with two air velocity vg = 0.164 m/s, v_g = 0.411 m/s, and θ= 5^o. At the same angle, an increase of superficial velocity v_w =1.4815 m/s causes the appearance of stratified wavy flow as shown in Fig.12 (d), and (j). That was consistent with experimental work of (Tzotzi, 2011). It has been noted stratified flow was only observed when the pipe inclined downwards 5^o. Due to the buoyancy force the gas phase tends to the flow along the upper wall of the tube than the lower wall, as the gas velocity is increased some droplets appeared at the upper wall of the test section.
2. Intermittent flow: plug or slug of liquid which fill the pipe are separated by gas zones which contain liquid layer flowing along the bottom of the pipe. At low superficial velocity of air and with increase the inclination angle of test section this flow pattern was notched, Fig. 12 (b) depicts plug flow regime with bubble appeared at the top of the tube, where v_w= 1.1506 m/s, v_a=0.164 m/s , and θ= 10^o. The transition from plug to slug flow at higher superficial air velocity v_a= 0.411 m/s at the same angle θ=10^o as shown in Fig.12 (h). Slug flow was also presented at angle 15^0 with v_w=1.1506 m/s, and v_g=0.164 m/s as shown in Fig. 12 (c). This due to increase the superficial velocity of air, and the angle of downward inclination, these results confirmed with experimental results of (Kang et al., 2002).
3. Annular flow: the annular flow consists of a water film around the pipe and an air core. This flow dominated at angle 15^o with superficial velocity of water and air v_w=1.4815 m/s, v_g=0.164 and 0.411 m/s as shown in Fig. 12 (f) and (L). Also, the annular flow regime appears at v_w=1.4815 m/s with air velocity v_g=0.411 m/s, and θ=10^o as pointed in Fig. 12 (k). In case of smaller superficial velocities, droplets appeared at the top of the test section as can be seen in Fig. 12 (e), and (i). This is called annular bubbly flow regime. These patterns results consistent with results of (Zimmerman et al., 2006).

CONCLUSIONS

Local heat transfer coefficients and flow pattern were measured for air-water in 4.5 cm, inner length of square section pipe and downward inclined 5^o, 10^o, and 15^o positions. The water superficial velocity varied from 1.150 to 1.481 m/s and the air superficial velocity varied from 0.164 to 0.411 m/s.

1. These results showed the heat transfer coefficient increases with increasing pipe inclination angle, air and water superficial velocity.
2. Stratified flow pattern was observed in the case of inclination angle 5^o as V_w= 1.150 m/s with two values of air velocity 0.164 and 0.411 m/s. Also, stratified wavy regime was dominated when the water superficial velocity increasing to 1.48 m/s.
3. Plug flow pattern was observed at V_w=1.150 m/s and V_a=0.164 m/s with inclination angle 10^o.
4. The transition from plug to slug flow occurred when increasing the inclination angle to 15^o at the same values of air and water velocities.
5. The transition from slug to annular flow occurred with higher superficial water velocity with increase in downward inclination angle to15^o.
6. Empirical correlations on average Nusselt number were developed. The results of the present study can be used to predict the wall temperature for the heat transfer to air-water flow in inclination pipe, in the range of studied superficial velocities**.**
7. The experimental results showed that the inclination angle, air and water velocities have significant effect on the heat transfer coefficient and flow pattern.

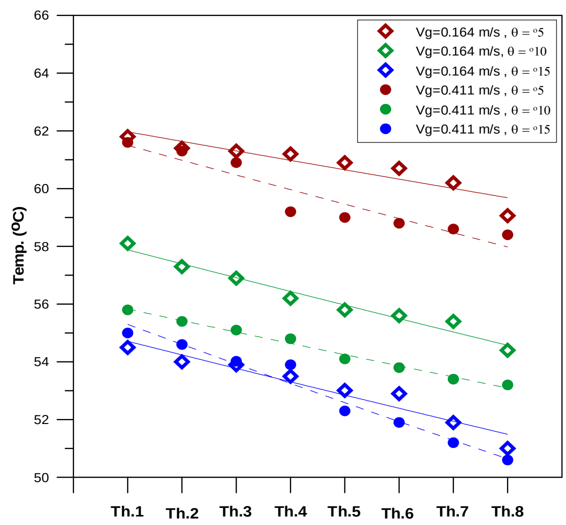


Fig. 6. Temperature reading in two phase air-water flow at V_w= 1.1506 m/s

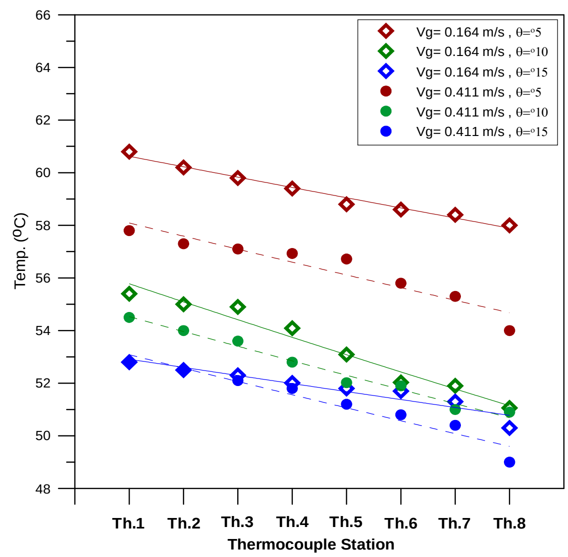


Fig. 7. Temperature reading in two phase air-water flow at V_w= 1.**4815** m/s

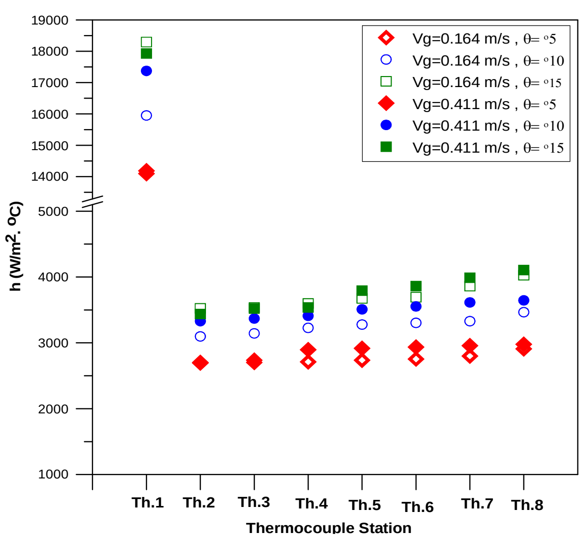


Fig. 8. Effects of inclination angle on two-phase heat transfer coefficient at V_W=1. **1506**m/s.

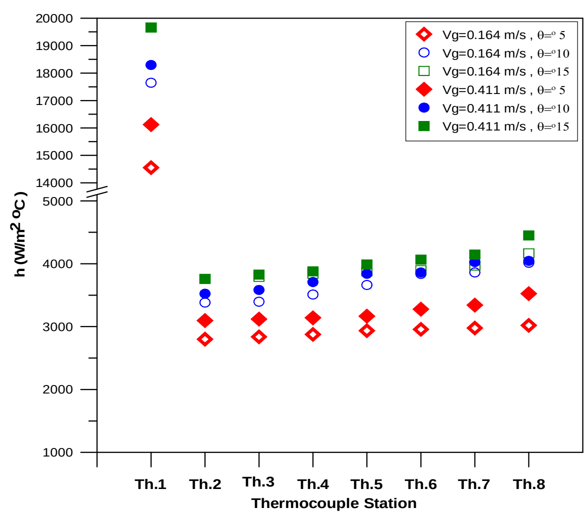


Fig. 9. Effects of inclination angle on two-phase heat transfer coefficient at V_W =1.**4815** m/s.

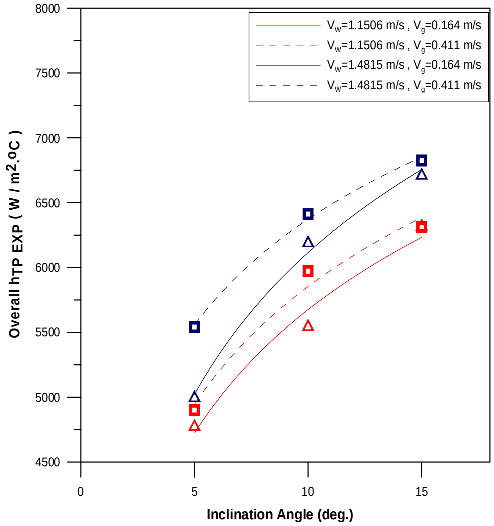


Fig. 10. Effect of inclination angle on the overall mean two-phase heat transfer coefficient

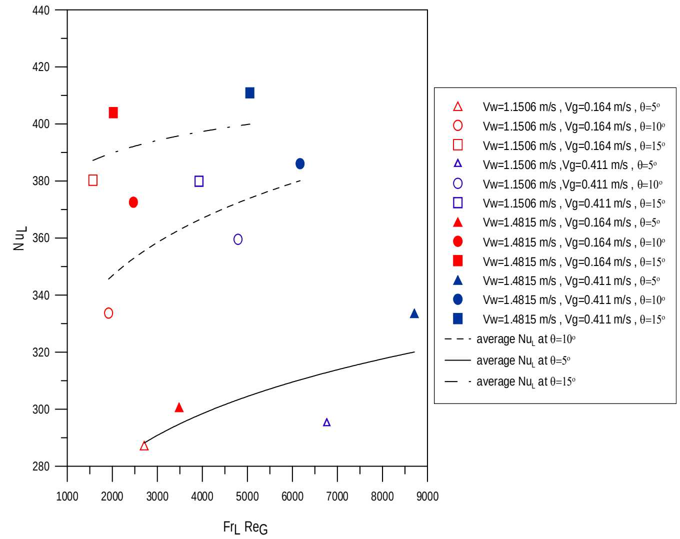
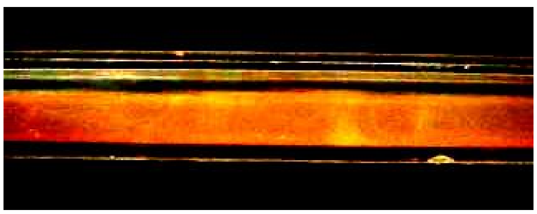
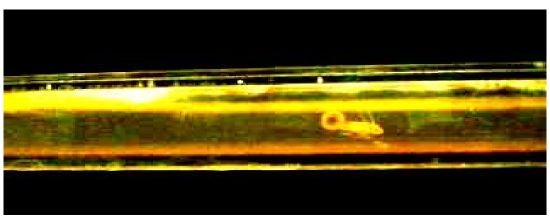


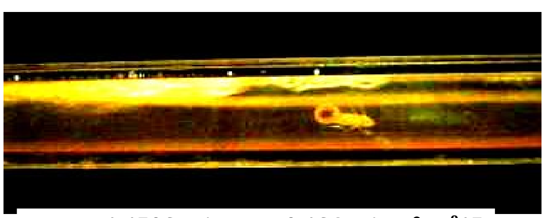
Fig. 11. Nusselt number correlation



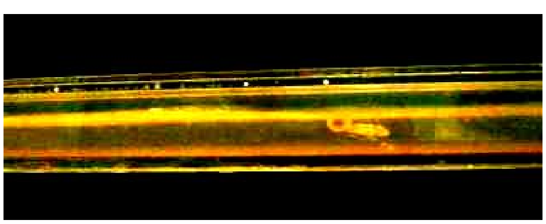
a: v_w = 1.1506 m/s , v_a = 0.164 m/s , θ = °5



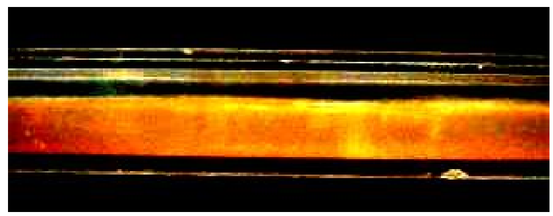
b: v_w = 1.1506 m/s , v_a = 0.164 m/s , θ = °10



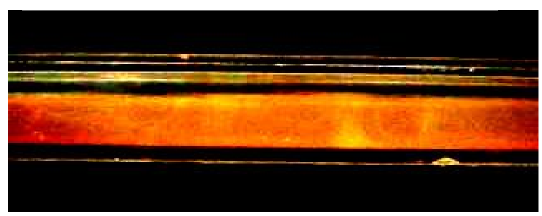
c: v_w = 1.1506 m/s , v_a = 0.164 m/s , θ = °15



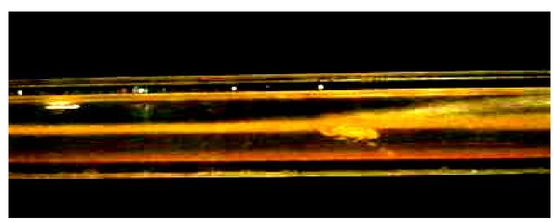
d: v_w = 1.48156 m/s , v_a = 0.164 m/s , θ = °5



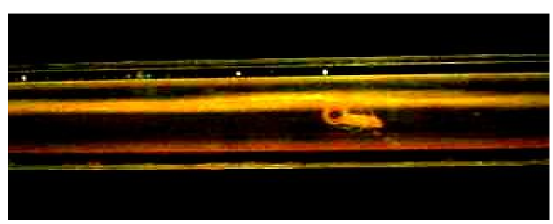
g: v_w = 1.1506 m/s , v_a = 0.411m/s , θ = °5



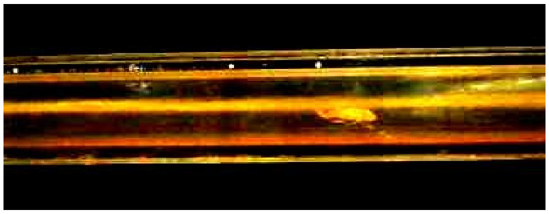
h: v_w = 1.1506 m/s , v_a = 0.411m/s , θ = °10



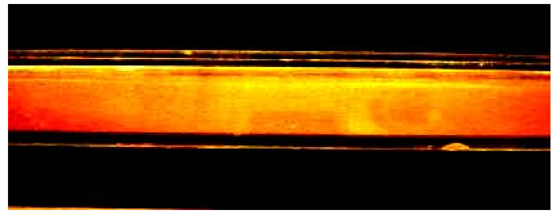
i: v_w = 1.1506 m/s , v_a = 0.411m/s , θ = °15



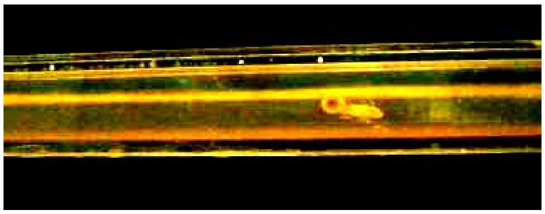
j: v_w = 1.48156 m/s , v_a = 0.411 m/s , θ = °5



e: $v_w = 1.48156$ m/s , $v_a = 0.164$ m/s , $\theta = {}^o10$



k: $v_w = 1.48156$ m/s , $v_a = 0.411$m/s , $\theta = {}^o10$



f: $v_w = 1.48156$ m/s , $v_a = 0.164$ m/s , $\theta = {}^o15$

L: $v_w = 1.48156$ m/s , $v_a = 0.411$m/s , $\theta = {}^o15$

Fig. 12. Flow patterns of downward air-water flow in inclined square pipe

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