

An Investigation of Heat Transfer for Two-Phase Flow (Air-Water) in Shell and Tubes Heat Exchanger

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ABSTRACT: Overall Heat transfer coefficients were estimated for two-phase flow in shell and tubes heat exchanger for different flow patterns. A two-phase heat transfer experimental setup was built for this study and a total of 44 two-phase heat transfer data with different flow patterns were obtained. For these data, the superficial Reynolds number ranged approximately between (650-4,000) and (27,000-170,000) for the liquid (Re_{SL}) in first and second heat exchangers respectively and ranged approximately between (2,600-11,000) and (87,000-162,000) for the gas (Re_{SG}) in the first and second heat exchanger respectively. Results indicate to the overall heat transfer coefficient (U) and pressure drop (ΔP_{pipes}) increases with Re_{SG} increases for a fixed Re_{SL} . Overall heat transfer coefficient (U) increase for low Re_{SL} with increase Re_{SG} in which ranged between (2-8%) and (3-21%) for first and second heat exchanger respectively, but for high Re_{SL} the overall heat transfer coefficient is increased gradually when increase Re_{SG} for both heat exchangers in the range between 0.5-1% and from 10 to 14% for first and second heat exchangers respectively. Thus the performance of shell and tubes heat exchanger is more efficient and improved for two-phase flow in tubes than one phase, that leading to more efficient industrial applications.

KEYWORDS: Two-phase, Shell and tubes, Overall heat transfer coefficient, Pressure drop.

I. INTRODUCTION

Heat exchangers are one of the mostly used equipment in the process industries. Heat exchangers are used to transfer heat between two process streams. One can realize their usage that any process which involve cooling, heating, condensation, boiling or evaporation will require a heat exchanger for these purpose. Common applications include chemical plants, evaporators, oil wells and pipelines, fluidized bed combustors, evaporators and so on. Performance and efficiency of heat exchangers are measured through the amount of heat transfer using least area of heat transfer and pressure drop. A more better presentation of its efficiency is done by calculating overall heat transfer coefficient. Usually, there is lots of literature and theories to design a heat exchanger according to the requirements. A good design is referred to a heat exchanger with least possible area and pressure drop to fulfill the heat transfer requirements [1].

One of the most important design parameters involved in the different applications of two-phase flow pipes is the heat transfer coefficient. Most of the literature abounds with studies of two-phase flow hydrodynamics and flow boiling, while limited data are available on the circumferential heat transfer coefficient variation. Styrikovich and Miropolski [2]

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were the first to document a study of circumferential temperature variations around a horizontal tube, where found the temperature difference between the pipe top and bottom to vary with the superficial liquid and gas velocities. In the region of intermittent flow regime. Johnson and Abou-Sabe [3] measured the average heat transfer coefficient using thermocouples, through study found its dependence on air and water flow-rate.

Generally, there are relatively few published works on two-phase heat transfer, almost of them about refrigerants in small tubes based on flow pattern. Several studies have dealt with two-phase flow boiling in small tubes, such as reported by Zhang et al. [4], Tran et al. [5], Pettersen [6], Yun et al. [7]. Wang et al. [8], Wojtan et al. [9], and Pamitran et al., [10] have proposed flow pattern maps for two-phase flow boiling heat transfer. Most of the previous literature studies concern on the flow through a horizontal pipes only Baker [11], Kattan et al., [12] and Kim [13], or for vertical tubes such as Kim et al. [14] and Stevanovic et al. [15], but there is no studies about two-phase flow through a shell and tubes heat exchanger this may be due the complexity of the geometry. For this reason, several analytical, experimental and numerical studies have been carried out [16, 17], but concentrated on the certain aspects of the shell and tube heat exchanger design for one phase flow in tubes [18-20]. These studies are concerned for baffled shell and tube heat exchangers generally. So, this study aims at studying simple un-baffled heat exchanger, which is more similar to the double pipe heat exchangers. Almost no studies is found for an un-baffled shell and tube heat exchanger. Thus general correlations of heat transfer coefficients and pressure drop for straight pipes can be useful to get an idea of the design. So, the main purpose of this work was study heat transfer and pressure drop for one and two-phase flow through tubes of shell and tubes heat exchanger with different flow patterns.

II. THEORETICAL ANALYSIS

The main assumptions applied for the energy balance on the tested heat exchanger herein:

- Mass flow rate is constant.
- There is no heat losses to environment.
- No shaft work is done by the fluid.

Applying the 1st law of thermodynamics to the system shown in Figure (1), the following energy balance equation is derived for both sides;

$$Q_h = m_h Cp(T_{h,i} - T_{h,o}) \quad \dots\dots\dots (1)$$

$$Q_c = m_c Cp(T_{c,o} - T_{c,i}) \quad \dots\dots\dots (2)$$

Where; Q_c represents the amount of thermal gained by cold fluid (water or air/water mixture), while Q_h represent the amount of thermal heat from a heating fluid (hot water), m is the mass flow rate measured by orifice plate, (kg/sec); and Cp is a specific heat of a fluid is determined as a function of average temperature (J/kg.K).

In actual heat provided by a hot fluid to the cold fluid is not exactly equal due to losses and resistances in the form of wall fouling. Assumption is made that the amount of heat transferred from the hot fluid is equal to amount of heat transferred to the cold fluid. Usually, heat exchangers are made isolated to minimize the heat losses to the environment. So, can write as follow;

$$Q_h \cong Q_c = Q \quad \dots\dots\dots (3)$$

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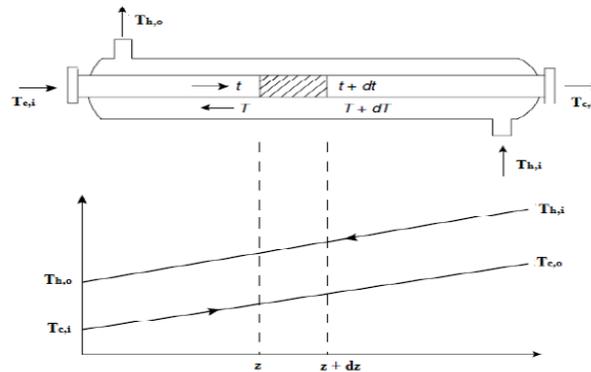


Fig 1. Heat Transfer in Counter-current Heat Exchanger

Graphically representation of these equations make the process easier and simple to understand (Figure.1). This figure are known as $T-Q$ diagrams. These graphs also helps in making sure that, 2nd law of thermodynamics is obeyed i.e. heat should always be transferred from higher temperature to lower temperature. The rate of heat transfer in a heat exchanger can also be expressed in an analogous manner to Newton's law of cooling as;

$$Q = UAF \Delta T_{LMTD} \dots\dots\dots (4)$$

Where;

- Q : Heat transfer rate, (W).
- A : Heat transfer area, (m^2)
- U : Overall heat transfer coefficient, ($W/m^2.K$)
- F : Fouling factor for system (For our cases ≈ 1)
- ΔT_{LMTD} : Logarithmic Mean Temperature Difference between both fluids, (K)

So, by using Eqn. (4) to evaluate overall heat transfer coefficient (U) then plot as a function of Reynolds number of cold water (Re_{SL}) and air (Re_{SG}) at superficial average velocities.

III. EXPERIMENTAL FACILITY AND MEASUREMENTS TECHNIQUE

3.1. Experimental Setup:

A schematic diagram of the experimental facility is presented in Figure.2. The water was drawn from a water tank (5) by a centrifugal pump (6). The water flow-rate was controlled by a flow regulator and measured by a calibrated Rotameter (7) with an accuracy of $\pm 2\%$. The air was delivered by a centrifugal blower (1) through a steel pipe connect with reduction pipe (4) then ended by air distributor. The air flow-rate was regulated by an air regulator (damper) (2) and measured by flow meter (orifice plate) (3) and related to differential manometer ($\Delta P_d, \Delta P_m$) with an accuracy of $\pm 2\%$. The water and the air were mixed in a mixing chamber and the two-phase mixture passed through pipeline of the flow development section (9). Then the flow entered the tubes of heat exchanger where hydrodynamic patterns and heat exchange were occurred. Afterwards, the flow entered the heat exchange section (shell and tubes heat exchangers) where the temperatures of inlet for air (T_1, T_2), inlet and outlet of two phase flow (T_3, T_5) respectively, and inlet and outlet of heating water (T_6, T_{10}) respectively were measured using a digital thermocouples connected to a control panel with accuracy ± 0.05 °C. Also, U-manometer is connected at the inlet and outlet points of heat exchanger to measure the pressure drop through heat exchanger. All the process controlled by a control panel including (electrical resistor power supply switch, blower power supply, thermocouple selector, four digit temperature display (resolution ± 0.01 °C), voltmeter and ammeter for the measurement of electrical resistors supply voltage and current, and electrical resistor power supply regulator).

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regulator display. Also, the cold water flow rate adjusting by means of regulator to desired value. Air flow rate adjusting by means of regulator to desired value. Finally, when steady state flow in tubes attained, the tests may be performed and record all the values of temperatures, flow rates and pressure drop for both sides shell and tubes. Then the collected data is analyzed in the computer and the heat transfer coefficient is estimated. Change both flow rates of air and cold water as required then repeat steps as proceed above without any particular time constraint.

IV. RESULTS & DISCUSSION

4.1. Heat Transfer & Pressure drops with Flow Patterns:

A number of verification runs were undertaken prior to the data logging. A series of tests for single-phase water flow were conducted to establish the validity of the system and test technique. All observations for the flow pattern by leaving the liquid flow rate fixed, flow patterns were observed for various air flow rates. The liquid flow rate was then adjusted and the process was repeated. All of the observed data points (11 points) were repeated four time to ensure of measurement. Then, evaluate U by using Eqn. (4) as a function of superficial Reynolds number of cold water (Re_{SL}) and air (Re_{SG}) [13]:

$$Re_{SL} = \frac{\rho_w u_{oL} d_i}{\mu_w} \dots\dots\dots (5)$$

$$Re_{SG} = \frac{\rho_g u_{oG} D}{\mu_g} \dots\dots\dots (6)$$

Where

- ρ_w : Density of water, (kg/m^3)
- d_i : Inside diameter of test tube section of heat exchanger, (m)
- u_{oL} : Superficial average velocity of water in the test tube section, (m/sec).
- μ_w : Viscosity of water at average temperature, ($kg/m.sec$).
- ρ_g : Density of air, (kg/m^3)
- u_{oG} : Superficial average velocity of air in the test tube section of steel pipe, (m/sec).
- D : Inside diameter of test tube section of steel pipe, (m)
- μ_g : Viscosity of water at average temperature, ($kg/m.sec$).

The measurements in two-phase flow were performed for different air–water flow regimes. The water superficial velocity changed in the range from 0.035 to 9.21 m/s while the air superficial velocity varied from 0.7 to 46.65 m/s . The average of uncertainty of the temperature and pressure drop measurements were ± 0.158 $^{\circ}C$ and ± 0.158 $mmHg$ respectively for replicate measurements of both heat exchangers in quadruplicate.

The overall heat transfer coefficient through both heat exchangers 1 and 2 between one phase (hot water) and two phase flow (mixture of cold water/air) are illustrate in Figures (3-8). The main behavior for each one are shown that, overall heat transfer coefficient increase due to appears air with water flow, this reveals to increase of turbulence through pipes leading to decrease thickness of boundary layer inside pipes.

Figures (3-5) show that an increase percent of the overall heat transfer coefficient for HEX.1 for $Re_{SL} = 651.05$, percent increase reaches to (2.1, 2.5, 6, and 6.2%) for Re_{SG} values (2692.7, 4071.37, 9499.86, and 10867.75) respectively. While for $Re_{SL} = 1953.14$, percent reach (5.2 and 7.6%) respectively, and for $Re_{SL} = 3906.28$ percent increase reach to (0.54, 0.62, 0.66, 0.97 and 1.05%) for Re_{SG} values (2035.68, 2692.7, 4071.37, 9499.86, and 10867.75%) respectively.

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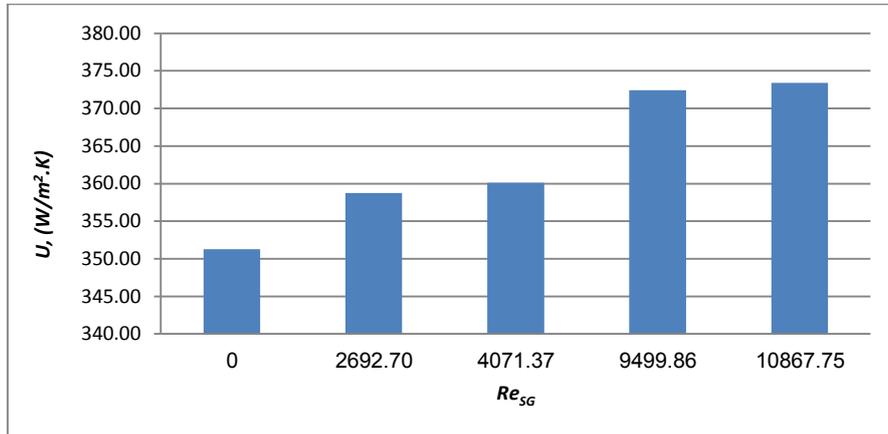


Fig 3. Comparison of Overall heat transfer coefficient vs. Air Reynolds number at constant liquid flow rate ($Re_{sl}=651.05$) for HEEx. No.1

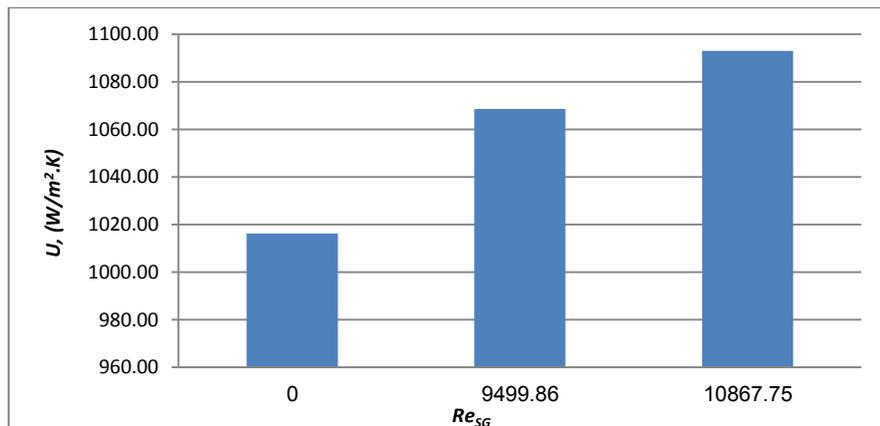


Fig 4. Comparison of Overall heat transfer coefficient vs. Air Reynolds number at constant liquid flow rate ($Re_{sl}=1953.14$) for HEEx. No.1

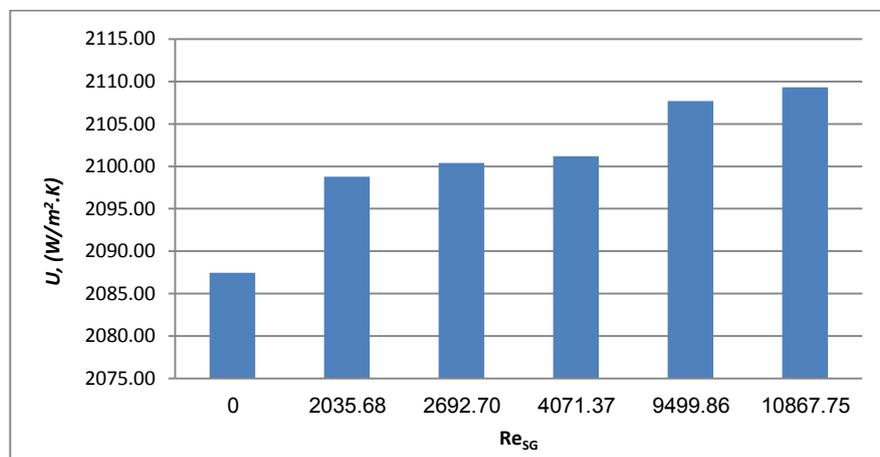


Fig 5. Comparison of Overall heat transfer coefficient vs. Air Reynolds number at constant liquid flow rate ($Re_{sl}=3906.28$) for HEEx. No.1

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This behavior described in (Figures 3-5) reveals to that, the overall heat transfer coefficient increase in two-phase of the gas Reynolds number increased for a fixed Re_{SL} in HEX.1. Generally increased as the air flow rate was increased for each fixed liquid flow rate. Also, the increase in U was more significant at low Re_{SL} than at high Re_{SL} due to at low liquid flow rates, the turbulence level in the liquid stream is small before being introduced into the gas stream. The introduction of the gas phase into the liquid stream increases the turbulence level which results in a high overall heat transfer coefficient. However, at high Re_{SL} the turbulence level is already high and the effect of the gas-phase on U is not that pronounced.

The identical behavior occur by the same way for HEX.2 as shown in Figures (6-8), whereas the overall heat transfer coefficient increase as a result of air introduces in stream with water. Also, continuously increase as a result of increase turbulence of flow as mentioned for HEX.1. Overall heat transfer coefficient increase with percent's (3.1, 10.4, 17.5, 18.9, and 20.6%) for Re_{SG} values (87566.69,112016.43,133558.06, 146483.03, and 161562.16) respectively when Re_{SL} =27931.66 as shown in Figure (6). While in Figure (7) reach to (18.9 and 20.6%) for Re_{SG} values (146483.03 and 161562.16) respectively when Re_{SL} =130209.39. But in Figure (18) increasing percent's reach to (10.99, 12, 13.1 and 13.8%) for Re_{SG} values (112016.43,133558.06, 146483.03 and 161562.16) respectively when Re_{SL} =169272.21.

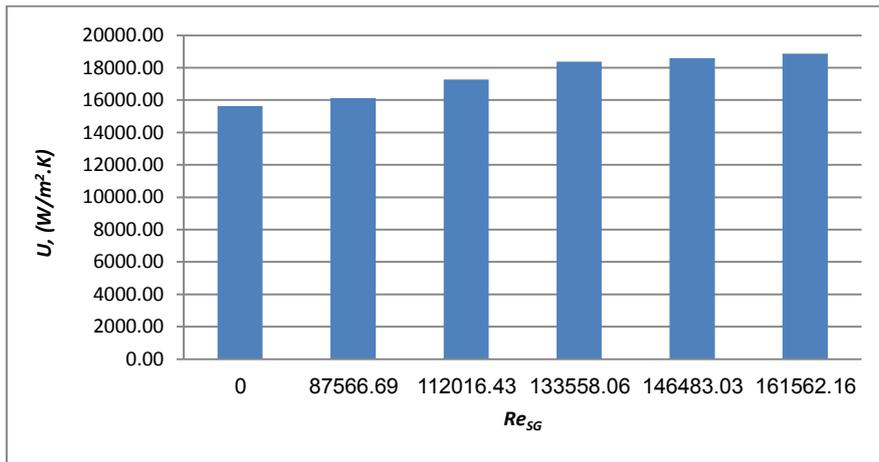


Fig 6. Comparison of Overall heat transfer coefficient vs. Air Reynolds number at constant liquid flow rate (Re_{SL} =27931.66) for HEX. No.2

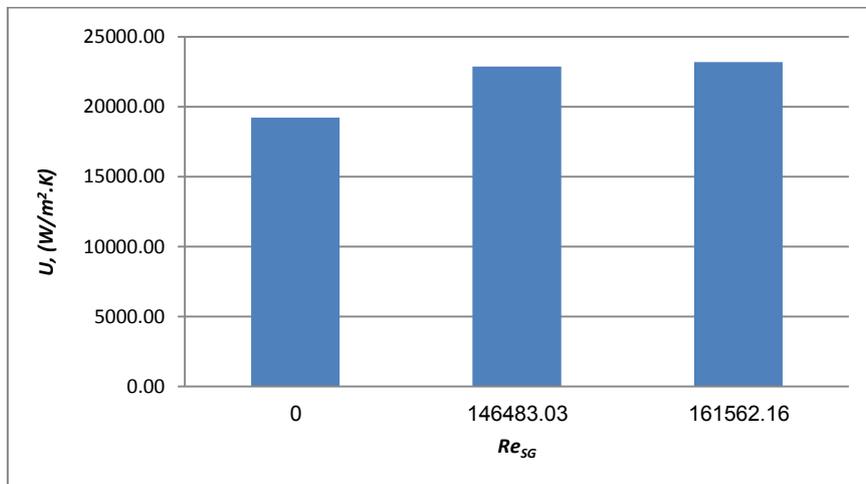


Fig 7. Comparison of Overall heat transfer coefficient vs. Air Reynolds number at constant liquid flow rate (Re_{SL} =130209.39) for HEX. No.2

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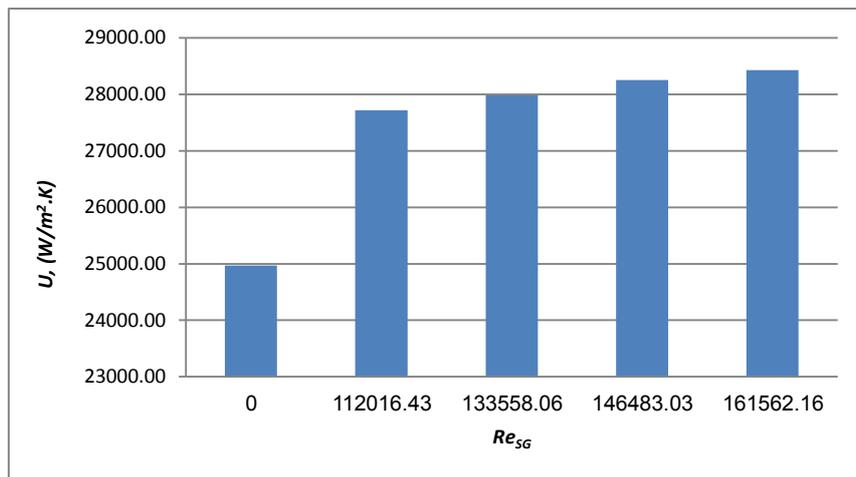


Fig 8. Comparison of Overall heat transfer coefficient vs. Air Reynolds number at constant liquid flow rate ($Re_{SL}=169272.21$) for HEX. No.2

The main difference between both heat exchangers for overall heat transfer coefficient whereas increasing progressively with increase Re_{SG} in HEX.2 more than as appear in HEX.1. This due to the scale of HEX.2 is greater than of HEX.1, leads to increase of the turbulence in HEX.2 more than of HEX.1. Comparison of the pressure drop of heat exchangers 1 & 2 between one phase (cold water) and two phase flow (cold water/air mixture) are illustrate in Figures (9-14). The main behavior for each one are shown that, pressure drop through pipe side increase due to appears air with water flow, this reveals to increase of turbulence through pipes leading to increase the pressure drop through pipes.

A comparison of increasing percent of pressure drop for HEX.1 Figures (9-11), note that for $Re_{SL} = 651.05$, percent increase reaches to (44.5,49.33,97.78, and 141.112%) for Re_{SG} values (2692.7, 4071.37, 9499.86, and 10867.75) respectively. While for $Re_{SL} = 1953.14$ percent reach (132.7, and 174.91%) for Re_{SG} values (9499.86 and 10867.75) respectively, and for $Re_{SL} = 3906.28$ percent increase reach to (86.94, 75.56, 92.93, 167.51, and 218.16%) for Re_{SG} values (2035.68, 2692.7, 4071.37, 9499.86, and 10867.76) respectively.

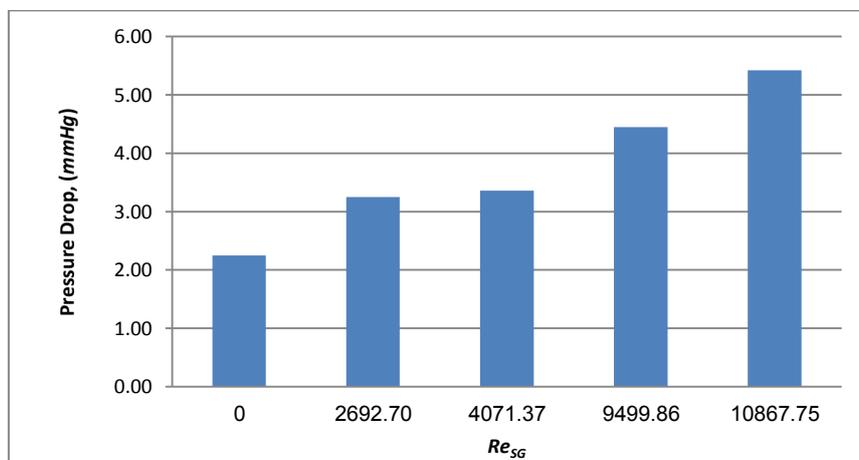


Fig 9. Comparison of Tubes Pressure Drop vs. Air Reynolds number at constant liquid flow rate ($Re_{SL}=651.05$) for HEX. No.1

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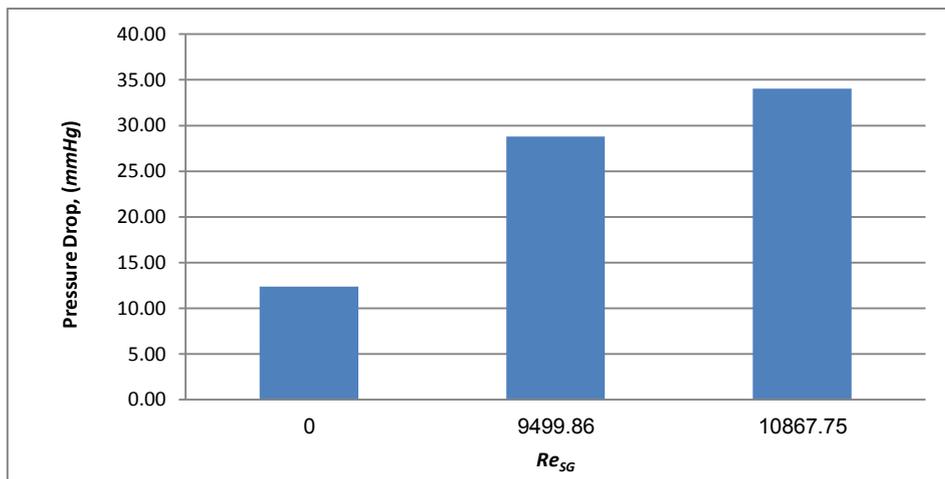


Fig 10. Comparison of Tubes Pressure Drop vs. Air Reynolds number at constant liquid flow rate ($Re_{SL}=1953.14$) for HEEx. No.1

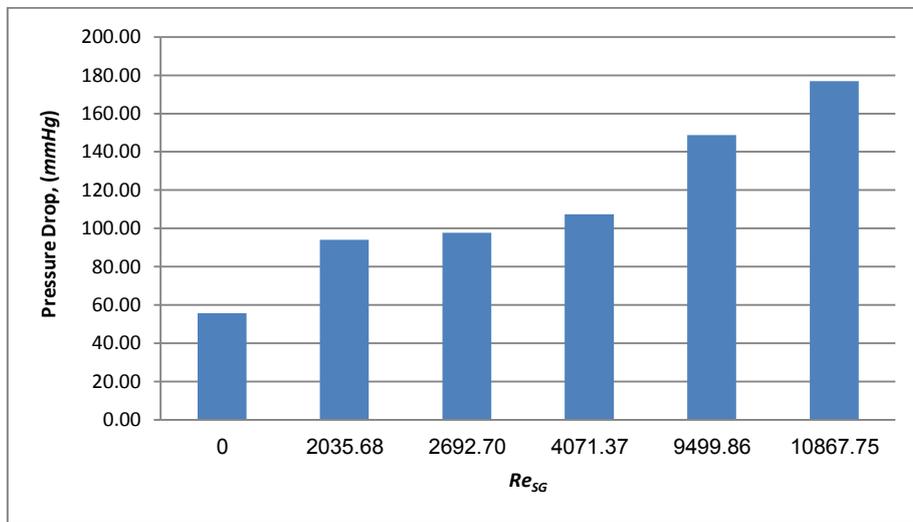


Fig 11. Comparison of Tubes Pressure Drops vs. Air Reynolds number at constant liquid flow rate ($Re_{SL}=3906.28$) for HEEx. No.1

While the comparison of increasing percent of pressure drop for HEEx.2 Figures (12-14), note that for $Re_{SL}=27931.66$, percent increase reaches to (8.87, 14.31, 26.97, 33.79, and 63.13)% for Re_{SG} values (87566.69, 112016.43, 133558.06, 146483.03, and 161562.16) respectively. While for $Re_{SL}=130209.39$, percent reach (88.49 and 101.18%) respectively, and for $Re_{SL}=169272.21$ percent increase reach to (189.14, 192.64, 197.13, and 200.75%) for Re_{SG} values (112016.43, 133558.06, 146483.03, and 161562.16) respectively.

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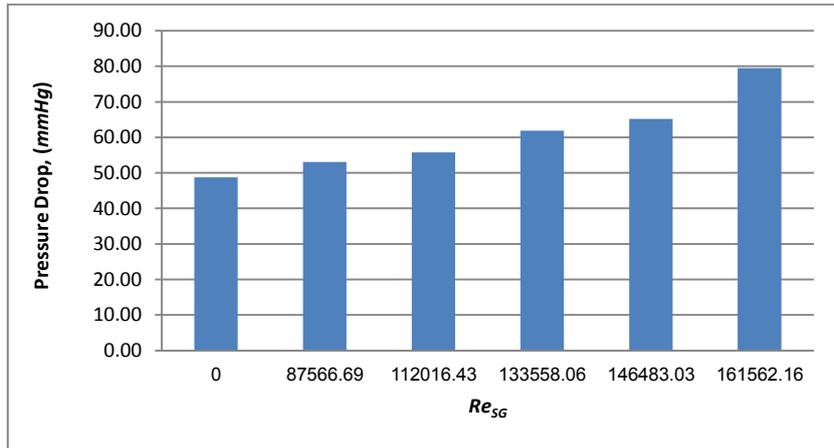


Fig 12. Comparison of Tubes Pressure Drop vs. Air Reynolds number at constant liquid flow rate ($Re_{SL}=27931.66$) for HEEx. No.2

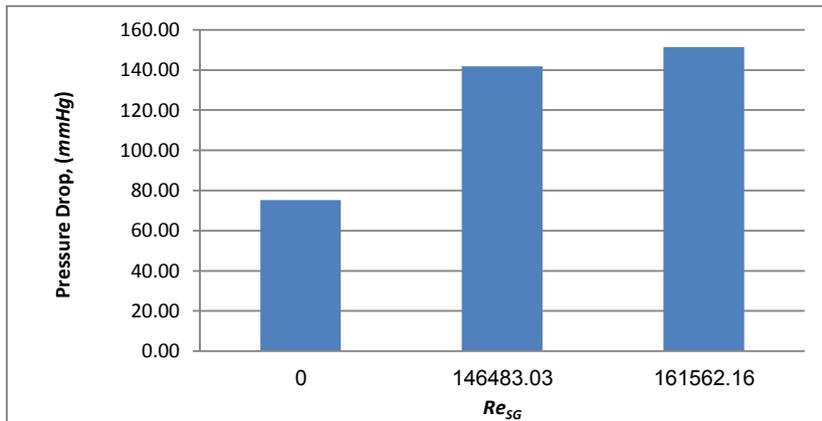


Fig 13. Comparison of Tubes Pressure Drop vs. Air Reynolds number at constant liquid flow rate ($Re_{SL}=130209.39$) for HEEx. No.2

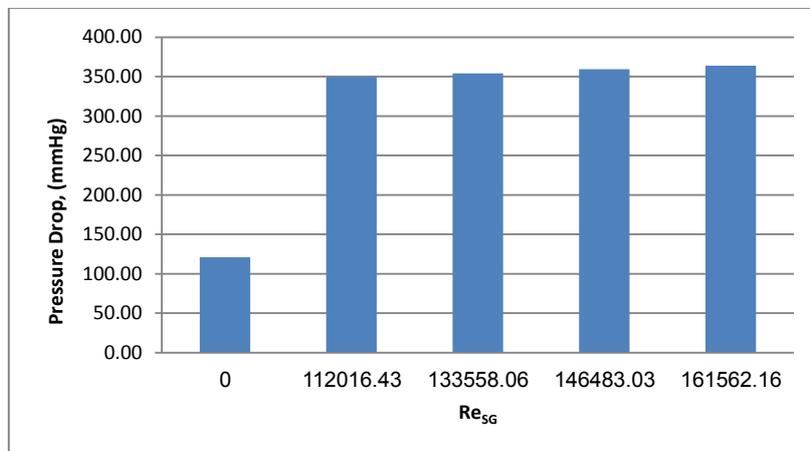


Fig 14. Comparison of Tubes pressure Drop vs. Air Reynolds number at constant liquid flow rate ($Re_{SL}=169272.21$) for HEEx. No.2

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4.2. Behavior of Overall Heat Transfer Coefficients:

Figures (15 and 16) show the variation of overall heat transfer coefficient parametrically versus superficial Reynolds numbers (Re_{SL} and Re_{SG}) for all of the water/air data obtained in this study. From these figures it can be seen that generally, as the gas superficial Reynolds number (Re_{SG}) increases for a fixed liquid superficial Reynolds number (Re_{SL}), the overall heat transfer coefficient increases. Some of the previous researchers also observed an increase in two-phase heat transfer as the gas Reynolds number increased for a fixed liquid Reynolds number. Zaidi and Sims [21] observed from the results of their two-phase heat transfer measurements in a vertical pipe that generally increased as the air flow rate was increased for each fixed liquid flow rate. Also, From Figures (15 and 16) note the increase in U was more significant at low Re_{SL} than at high Re_{SL} . The explained to the increase in U by the turbulence level already present in the liquid stream. At low liquid flow rates, the turbulence level in the liquid stream is small before being introduced into the gas stream. The introduction of the gas phase into the liquid stream increases the turbulence level which results in a high heat transfer coefficient. However, at high Re_{SL} , the turbulence level is already high and the effect of the gas-phase on U is not that pronounced.

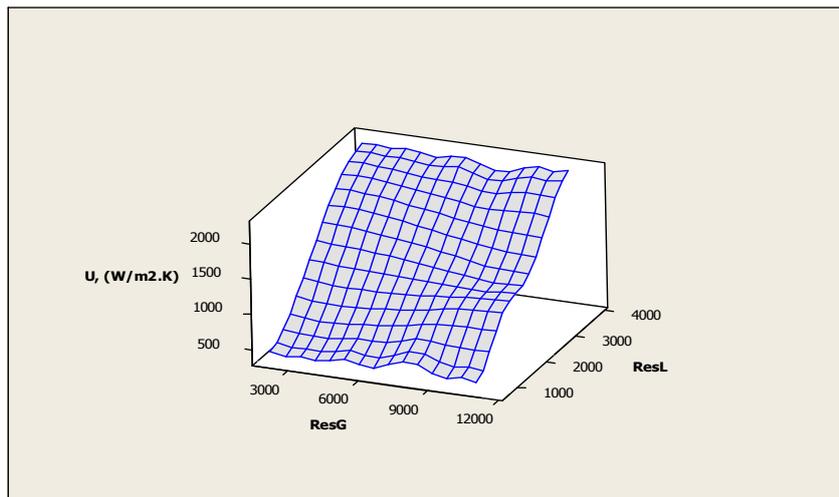


Fig 15. Variation of Overall Heat transfer Coefficients for HEX. No.1

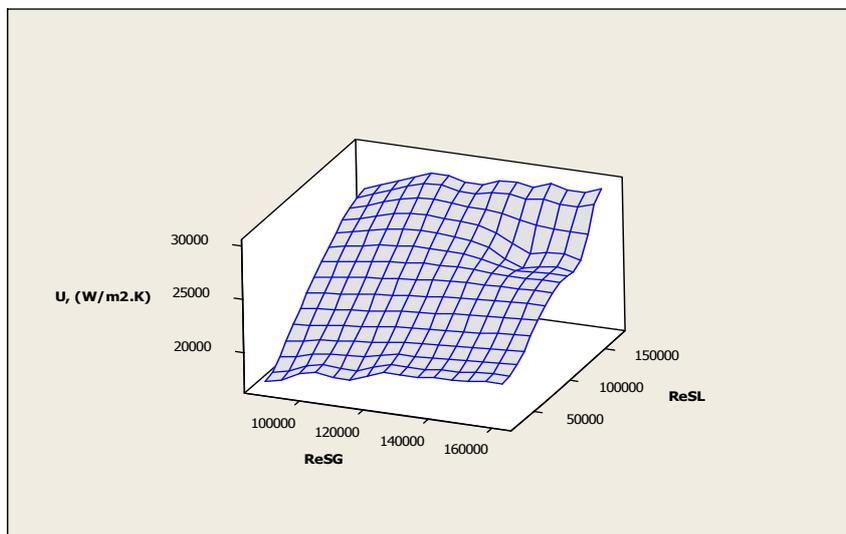


Fig16. Variation of Overall Heat Transfer Coefficients for HEX.No. 2

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These observed the increase in U caused by the addition of a gas phase into liquid flow from their air-water mixtures in pipes, due to the possible mechanisms: liquid and mixture velocity increase due to the addition of the gas phase increased turbulence and mixing action in the main stream due to continuous interaction of the two phases; and increased turbulence near the heated wall caused by gas bubbles, resulting in disturbances and a decrease in the effective thickness of the viscous boundary sub-layer by the fact that the eddies, present in the wake of the rising bubbles, penetrate in the viscous sub-layer. Groothuis and Hendl [22] mentioned in comparing their own air-oil results that the influence of air on heat transfer was most pronounced at the lowest Reynolds numbers because air would be more effective in promoting turbulence there. Also, Ravipudi and Godbold [23] found from their experimental data of air-toluene mixture in a vertical pipe that the introduction of air into the liquid increased the heat transfer coefficient substantially due to the reduction of the effective thickness of the viscous sub-layer. They also found that heat transfer coefficient of pipes increased, reached a maximum and then decreased. The maximum in heat transfer coefficient was observed to be in the transition zone between annular flow and mist flow. This situation is noticeable in Figure (15) of this work. The explained of the highly turbulent motion of the gas-liquid mixture with increasing amounts of air caused randomly distributed dry spots to appear on the wall, thereby decreasing the heat transfer rate. Also, they attributed the decrease in heat transfer coefficient of pipes at high Re_{SG} to firstly, the outlet liquid temperature decreased due to the mass transfer from liquid to air. Next, heat transfer coefficient of pipes did not increase in proportion to the increase in the temperature gradient. So this explanation satisfied with our work special for temperature difference has no proportional effect on U .

Finally, observe that there exists a maximum value of U as Re_{SG} increases for a fixed Re_{SL} . For Re_{SL} range (2,000-3,200) for HEx.1, and for range 10,000-15,000 for HEx.2, in which U reached a maximum, then U decreased as more air was added in these ranges. Previously, Pletcher and McManus [24] also observed this manner for air-water annular flow experiments in a horizontal pipe that heat transfer coefficient in pipes passes through a maximum for a given water flow rate and then decreases as the air flow rate increases. They explained this trend as the air flow rate increases, a countering mechanism comes into play which tends to reduce the heat transfer coefficients by depressing the final exit equilibrium temperature. This occurs since, as the ratio of air flow rate to water flow rate increases, more and more evaporation is possible. At some air flow rate, this latter mechanism begins to dominate. They also suggested that the decrease in heat transfer coefficient in pipes was due to liquid entrainment at the higher air rates.

4.3. Behavior of Pressure drop:

For both heat exchangers, the pressure drop through pipes versus Re_{SG} and Re_{SL} are shown in Figures (17 and 18).

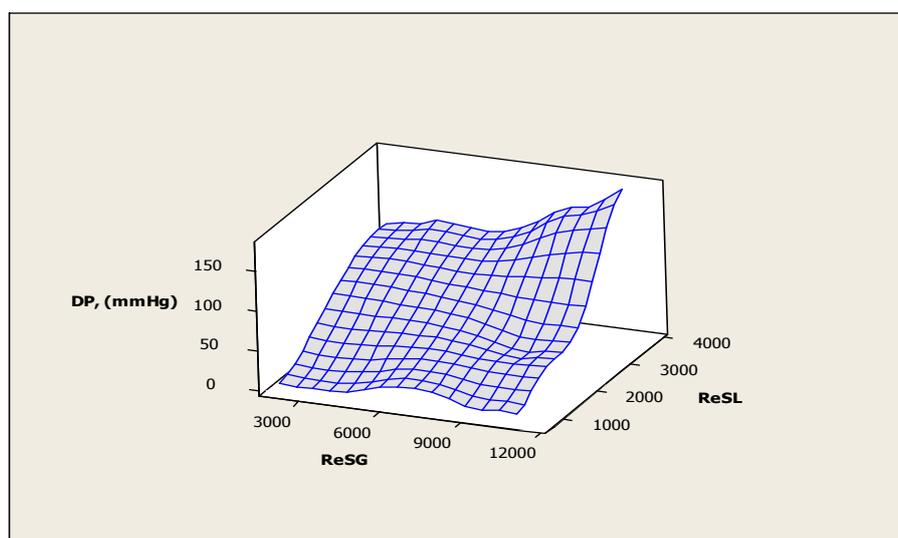


Fig 17. Variation of Pressure Drop Through Tubes of HEx. No.1

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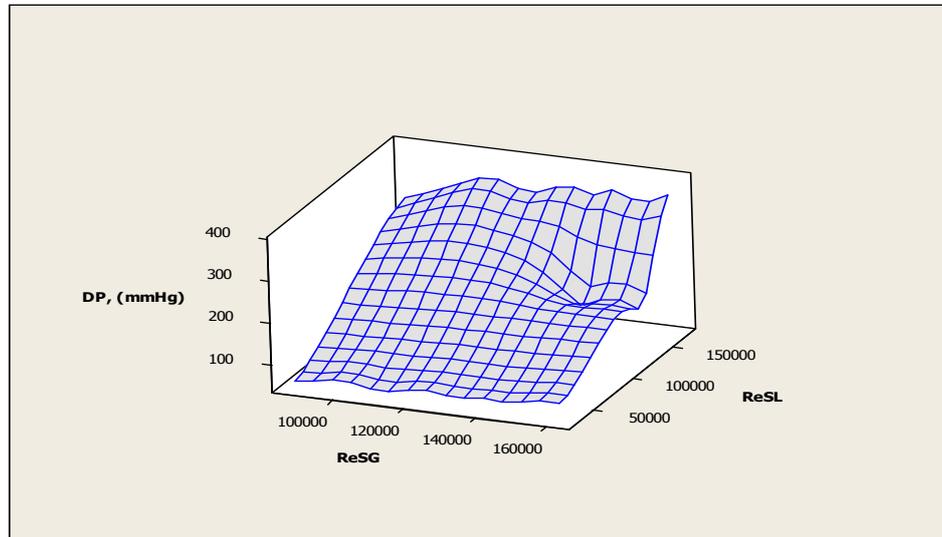


Fig18. Variation of Pressure Drop Through Tubes of HEEx. No.2

The same behavior in which increase pressure drop with increasing both Re_{SG} and Re_{SL} . Whereasthe peaks occur in the same range for decrease U shown in Figures. (15 and 16), due to probably appear annular flow for air and water mixture flow.

V. CONCLUSION

tubes exchangers were conducted. The water superficial velocity varied from 0.035 to 9.21 m/s and the air superficial velocity varied from 0.7 to 46.65 m/s .

The flow patterns were the main factors that influence the heat transfer, in which the overall heat transfer coefficient was estimated from the temperature measurements which were conducted by using digital thermocouples. The circumferential distribution of the overall heat transfer coefficients varies significantly in the flow regimes studied. The relationship between the overall heat transfer behavior and the hydrodynamics of the flow was clarified. The liquid superficial velocity was found to be a major factor which affects the overall heat transfer coefficient. Since the introduction of the gas phase into the liquid stream increases the turbulence level and mixing action in the main stream due to the continuous interaction of the two phases, generally the overall heat transfer coefficient increases as Re_{SG} increases for a fixed low Re_{SL} whereas generally ranged (2-21%) for low Re_{SL} , but for high Re_{SL} the overall heat transfer coefficient is low and generally ranged (0.5 to 14%). Also an increase of turbulence and mixing action in the main stream due to continuous interaction of the two-phases and presence of gas bubbles leading to increase the pressure drop through tubes of heat exchangers. Thus the performance of shell and tubes heat exchanger is more efficient and improved for two-phase flow in tubes more than one phase flow.

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