

Performance Enhancement of the Vertical Double Pipe Heat Exchanger by Applying of Bubbling Generation on the Shell Side

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ABSTRACT

The performance of a vertical double pipe heat exchanger was analyzed experimentally under effect of bubble generation by air injection through an air diffuser has many hundreds of very small holes placed at the bottom of the vertical heat exchanger around the (inner) hot tube. The air is supplied by using a small compressor with an air flow meter, and a small boiler with a temperature controller which used for heating the hot fluid, the inlet and outlet temperatures are measured by using digital thermocouples, the tests are performed for parallel and counter flow configurations with different flow rates of cold fluid and air injection. The results show that the exergy loss was affected significantly by bubble generation with positive effects on (Heat Transfer Rate, Effectiveness, Number of Transfer Units and Overall Heat Transfer Coefficient).

Keywords: heat exchangers, bubble generation, performance improving, vertical heat exchangers, exergy loss.

أداء المبادل الحراري المزدوج تحت تأثير توليد الفقاعات داخل الأنبوب الخارجي

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الملخص

يتضمن البحث دراسة وتحليل أداء المبادل الحراري المزدوج العمودي عمليا تحت تأثير توليد الفقاعات من خلال حقن الهواء باستخدام ناشر ذو مئات الثقوب الصغيرة جدا حيث تم وضعه في قعر المبادل الحراري العمودي حول أنبوب المائع الساخن (الأنبوب الداخلي)، وباستخدام ضاغط هواء صغير تم تجهيز الهواء بقياس كميته باستخدام مقياس الجريان، ومن خلال منظومة كاملة مزودة بمرجل صغير ومنظم لدرجة حرارة المائع الساخن ومقاييس رقمية لقياس درجات الحرارة ومعدل الجريان لكلا المائعين الداخل والخارج تم الحصول على البيانات اللازمة. وأجريت القياسات للجريان التوازي والمتعاكس لمعدلات جريان مختلفة للمائع البارد وحقن الهواء. وأظهرت النتائج بان توليد الفقاعات ذو تأثير واضح على خسائر الطاقة المتاحة وتأثيرات ايجابية على (معدل انتقال الحرارة، الفاعلية، عدد انتقال الوحدات ومعدل انتقال الحرارة الكلي).

الكلمات الدالة: المبادلات الحرارية، توليد الفقاعات، تحسين الأداء، المبادلات العمودية، خسائر الطاقة المتاحة.

1. Introduction

A standard and typical double pipe heat exchanger consists of one pipe placed concentrically inside another pipe of a larger diameter with appropriate fittings to direct the flow from one section to the other. One fluid flows through the inner pipe (tube side) and other flows through the annular. The major use of the double pipe heat exchanger is for sensible heating or cooling of process fluids when small heat transfer areas are required [1]. These heat exchangers can be used when one stream is a gas viscous liquid or small in volume and can be used also under severe flouting conditions because of the ease of cleaning and maintenance. The heat exchangers are used in a wide range and size for cooling or heating of fluids, the applications of this devices are observed clearly for many industries process, and great efforts were devoted by engineers and scientists to increase heat transfer rate and improving its performance using several methods which can be classified into two principal types [1], the first type (passive techniques) which requires fluid additives (Nano particles) without direct application of external power ,this method was used by [2,3] also coarsening heat exchanger surfaces and inserting fluid turbuletors such as variation of tube design [4,5], propellers were used by [6], fins by [7], the internal spring by [8], a swirl generation by [9] and helical wires by [10], in the second method (active techniques) the external power was needed for instance surface vibration and electrostatic fields such as pulsation which was used by [11], pipe rotation used by [12] and air injection (bubble generation). The effect of gas bubbles on heat transfer have been studied and can be classified into boiling two-phase heat transfer [13,14] and non-boiling two-phase heat transfer [15,16], the results showed that the air injection (bubble generation) has a big effect on the heat transfer coefficient due to liquid motion and velocity fluctuations [17]. Experimental investigation was done by [18]. Some parameters were studied for observing their effects on exergy loss. The present research is aimed to study the effect of air injection (bubble generation) on the double pipe vertical heat exchanger performance and exergy loss using both configurations (parallel and counter) flow, the application of this method is suitable among the others in terms of controlling and adjusting the air flow rate for best results, this method (bubble generation) was selected and applied in this work.

2. Theoretical Analysis:

The heat Transfer in the double pipe heat exchangers can be calculated by [1]:

$$Q_h = \dot{m}_h C_{p_h} (T_{hi} - T_{ho}) \text{ Or } Q_c = \dot{m}_c C_{p_c} (T_{co} - T_{ci}) \quad (1)$$

and

$$Q_{ave} \frac{Q_h + Q_c}{2} = \quad (2)$$

where $\dot{m}_h$ = mass flow rate of hot fluid , $\dot{m}_c$ = mass flow rate of cold fluid, C_{p_h} and C_{p_c} represent the specific heat of hot and cold fluids respectively and T_h = hot fluid temperature and T_c = the cold fluid temperature .

The subscripts (i) and (o) represent the inlet and outlet hot and cold temperature respectively.

The overall heat transfer coefficient can be calculated experimentally from: [18]

$$U_{exp} = \frac{Q_{ave}}{A \Delta T_{LMTD}} \quad (3)$$

Where A is the surface area of the tube and

$$\Delta T_{LMTD} = \frac{(T_{h,out} - T_{c,out}) - (T_{h,in} - T_{c,in})}{\ln\left(\frac{T_{h,out} - T_{c,out}}{T_{h,in} - T_{c,in}}\right)} \quad \text{For parallel flow} \quad (4)$$

And

$$\Delta T_{LMTD} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln\left(\frac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}}\right)} \quad \text{For counter flow} \quad (5)$$

Effectiveness – NTU method is the versatile and powerful method for designing and analyzing the heat exchangers and can be calculated by: [1, 18 and 19]

$$\text{Effectiveness} = \varepsilon = \frac{\text{actual heat transfer rate}}{\text{maximum possible heat transfer rate}} \quad (6)$$

$$\text{And maximum possible heat transfer rate} = Q_{max} = (\dot{m}_c)_{\min} (T_{h,in} - T_{c,out}) \quad (7)$$

Where $(\dot{m}_c)_{\min}$ = the lesser value of $(\dot{m}_h C p_h)$ or $(\dot{m}_c C p_c)$

NTU (number of transfer units) is indicative of the size of the heat exchanger and it is evaluated by:

$$NTU = \frac{A U}{(\dot{m}_c)_{\min}} \quad (8)$$

Where A is the heat transfer area (m^2) and U is the overall heat transfer coefficient ($W/m^2 \text{ } ^\circ C$)

3. Exergy Analysis:

The exergy can be defined as the maximum theoretical work which extracted from a given entity when it is brought to equilibrium with its environment. The pressure drop and temperature gradient are the main sources of exergy losses in heat exchangers. The exergy [E] for a given control volume can be calculated as [1, 18]:

$$\sum E_i = \sum E_o - \sum E_{product} \quad (9)$$

Where E_i = inlet exergy, E_o = outlet exergy .and

$E_{product}$ = amount of exergy produced in the control volume.

Despite the perfect thermally insulating of the outer surfaces of the outer tube and their joints, there was heat transfer as a (*heat flux*) from the shell surface (outer wall surface) of the vertical heat exchanger used in this research due to less adequateness of the insulation therefore the heat transfer from the outer surfaces is taken to be [12%] which means that the heat losses from the hot fluid (Q_h) and that gained by the cold fluid (Q_c) are not equal [18].

$$\text{Or } Q_c \neq Q_h \text{ where } Q_c = \dot{m}_c C p_c \Delta T_c \quad \text{and} \quad Q_h = \dot{m}_h C p_h \Delta T_h \quad (10)$$

exergy loss of a double pipe heat exchanger under steady state conditions, can be calculated by using: [18]

$$E = E_h + E_c \quad (11)$$

Where E_h = hot fluid exergy change and E_c =cold fluid exergy change, and can be expressed mathematically as [1, 18]:

$$E_h = T_e [\dot{m}_h (S_{ho} - S_{hi})] \quad \text{and} \quad E_c = T_e [\dot{m}_c (S_{eo} - S_{ei})] \quad (12)$$

Where T_e is the ambient temperature.

By taking into account the entropy change that caused by temperature difference and neglecting the loss due to frictional pressure drop (due to short length of the heat exchanger [1, 18]) can be evaluated as:

$$S_o - S_i = C_p \ln \left(\frac{T_o}{T_i} \right) \quad (13)$$

Where S_o and S_i represent the outlet and inlet specific entropy of mass fluxes. By substituting equations (13) and (12) in equation (11) the following equation was obtained which represents the exergy loss as [1, 18 and 19]:

$$E = T_e \left[\dot{m}_h C_{p_h} \ln \left(\frac{T_{ho}}{T_{hi}} \right) + \dot{m}_c C_{p_c} \ln \left(\frac{T_{co}}{T_{ci}} \right) \right] \quad (14)$$

Where T_{hi} , T_{ho} and T_{ci} , T_{co} represent the inlet and outlet temperatures of the hot and cold fluids respectively and T_e is the environment temperature.

4. Experimental Procedures and Instrumentation:

The first step of this research was to designing and making a double pipe heat exchanger **Table (1)** (technical specification) with an air distributor (ring shape) on a shell side (cold fluid), the air distributor has about two hundred small (0.5mm in diameter) holes and during the experiments the heat exchanger was positioned to be vertical and air distributor was placed at the bottom of the heat exchanger for utilization of the buoyancy force of the air bubbles which were formed inside the fluid to move along the tube of the heat exchanger. The second step was preparing a small boiler with a hot fluid and cold fluid mass flow rates controllers and temperature indicators (the system that was used from TQuepment). The third step was preparing an air compressor with flow rate controller to supplying the air distributor (diffuser, **Fig. (1)**) which was placed inside the heat exchanger on the shell side around the hot tube. A thick layer of glass wool was used for insulating the shell (outer tube) to prevent and reduce the heat losses to surrounding. The system was completed as in **Fig. (2)**.

Table (1): Technical Specification of the Heat Exchanger

Heat Exchanger Length	60Cm
Shell Diameter	14Cm
Tube Diameter	2.85Cm
Tube Thickness	0.0125Cm



Fig. (1): The Air Distributor (Diffuser)



Fig. (2): The Test Rig Apparatus

5. Experimental Procedure

The two configurations were tested (parallel flow and counter flow) in all cases the constant flow rate for the hot fluid (tube side) was selected to be (1.5 L/min) or (0.025 kg/sec) while the cold fluid flow rate (shell side) was adjusted and controlled at (1.5, 1.125, 0.75 and 0.25) L/min or (0.025, 0.01875, 0.0125 and 0.00417) kg/sec and for each flow rate on shell side the air flow rate controlled at (0, 2, 4, 6, 8 and 10) L/min and the inlet temperatures of (57°C) and (15°C) for hot and cold fluids respectively were measured, where the low temperature of the cold fluid was due to winter weather [February 2017] in [Kirkuk City], the measured inlet temperature of the injected air was about [18°C =Laboratory temperature] and the system has been operated with neglecting the effect of air temperatures (inlet and outlet) due to little mass fraction of the injected air to the liquid. Finally the results were recorded when the steady state condition achieved (outlet temperatures constant in time) for both (parallel and counter) flow configurations at all cases.

6. Results and discussion:

The following figures represent the effects of air injection (bubble generation) at different injection rates for both types of double pipe heat exchanger (parallel flow and counter flow). **Fig. (3)** and **Fig. (4)** represent the relationship between the air injection rates (0, 2, 4, 6, 8 and 10) liter per minutes on the shell side (cold fluid) and temperature difference ($T_{hi}-T_{ho}$ and $T_{co}-T_{ci}$) of the both fluids (hot and cold) for heat capacity ratio of ($\frac{C_{min}}{C_{max}}=1$) and two flow types (parallel and counter). It is observed that the bubbling generation has significant positive effect on the temperature differences due to the agitation or bubble motion through the fluid which leads to variation in the flow type from laminar to turbulent and increases the temperature difference which is a great factor (driving force) that heat transfer rate depends on it. All tests show that the maximum effect is occurred at air injection rates between (2 – 4) liter/min. And the increase in temperature difference will be little after that.

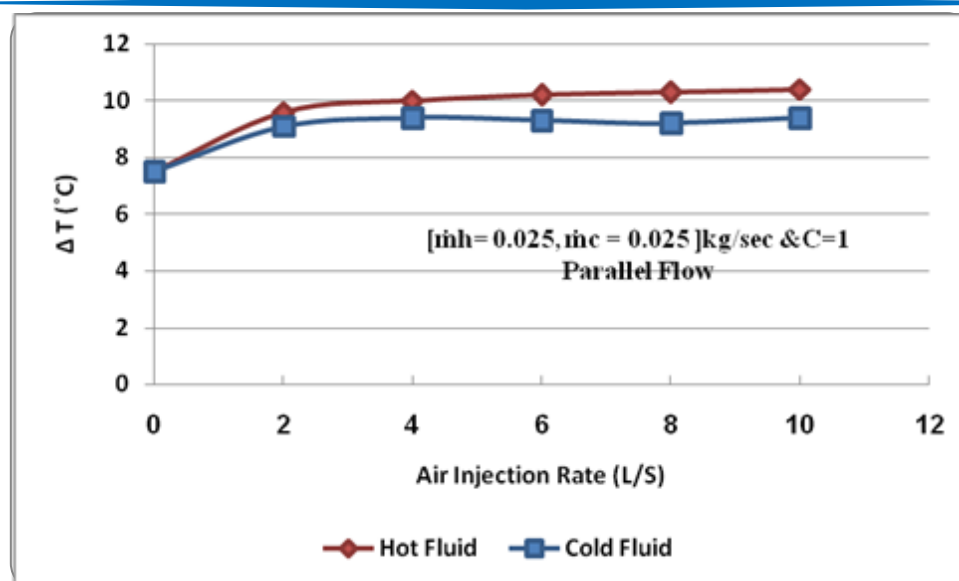


Fig. (3): The Effect of Bubble Generation on the Temperature Difference of the Both Fluids

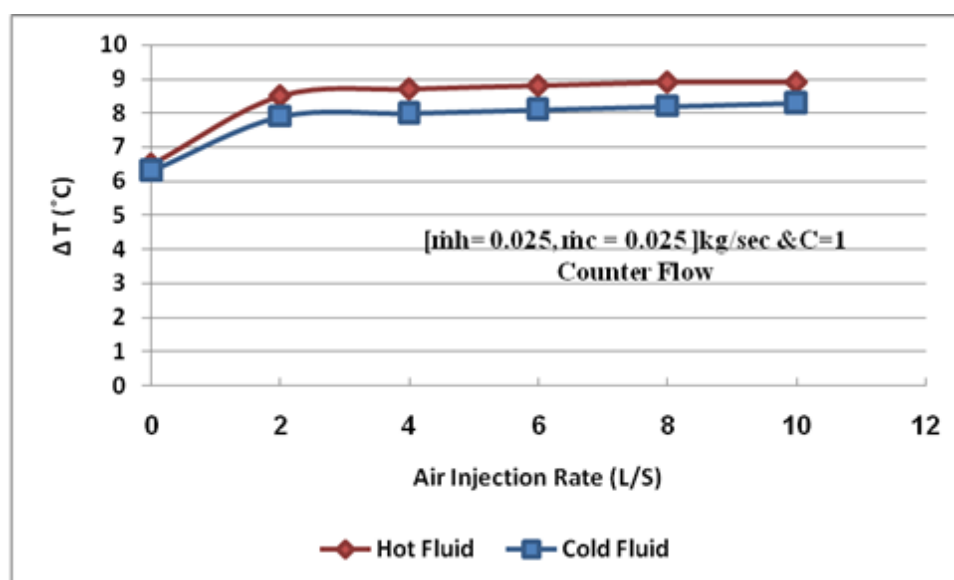


Fig. (4): The Effect of Bubble Generation on the Temperature Difference of Both Fluids.

Fig. (5) and Fig. (6) represent the effect of air injection rates on the exergy loss for both (parallel and counter) flow configuration and flow rate of (1.5 L/min.) for hot fluid (tube side). It is observed that the increasing of air flow rate (bubble generation) increases the exergy loss strongly for the case of ($C=1$), or (cold fluid flow rate equal to 1.5 L/min. too) which means that the bubble generation leads to more entropy generation and exergy consumption (this exergy is due to temperature variation with neglecting the exergy loss generated by pressure drop and friction).

In the author cases in which the flow rate of the cold fluid (shell side) reduced to satisfy the heat capacity ratios of (0.75, 0.5 and 0.25) the exergy loss reduces to minimum value at the air flow rate of (2 L/min.) and by increasing the air flow rate more than (2 L/min.) the exergy loss increases proportionally, which means that the controlling of the exergy loss can be done by controlling the air flow rate (bubble generation) in these three cases .

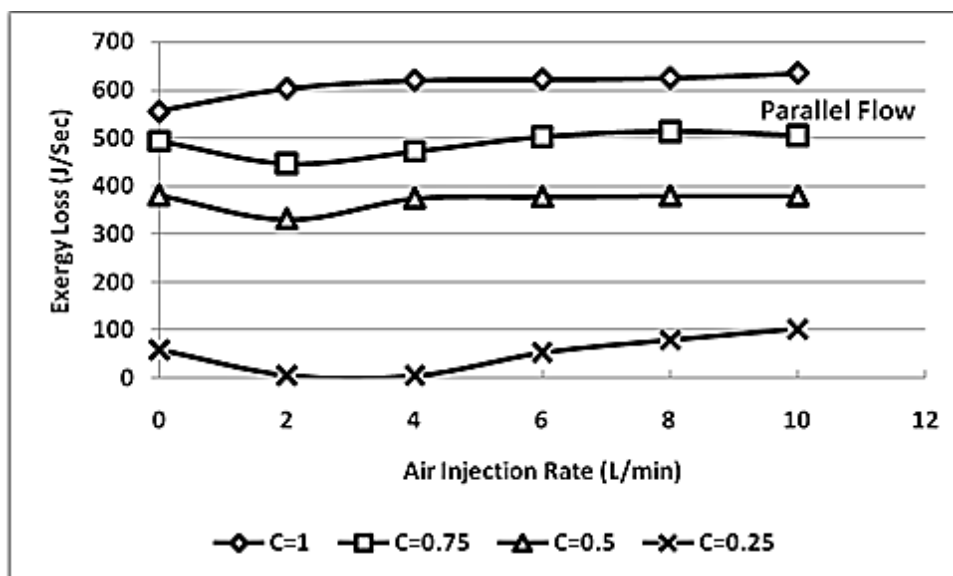


Fig. (5): The Effect of Air Injection Rate (Bubble Generation) on the Exergy Loss.

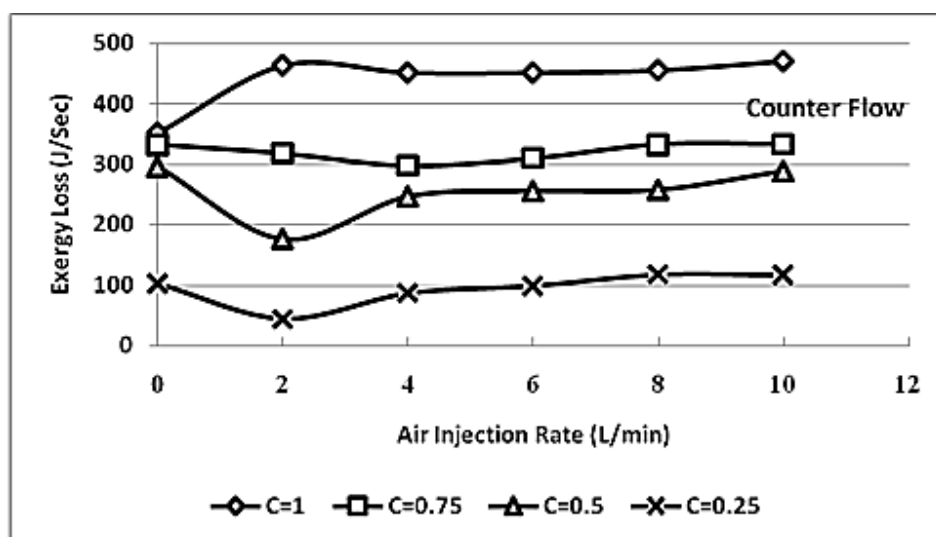


Fig.(6): The Effect of Air Flow Rate (Bubble Generation) on the Exergy Loss.

Fig. (7) and Fig. (8) represent the relationship between air injection rates and actual heat transfer rates for both types (parallel and counter) at different values of heat capacity ratios of

(1, 0.75, 0.5 and 0.25). It is observed that the air injection rates (bubble generation) increases the actual heat transfer rate about (15-27) percent and the positive effect is clearly noted, these increases are due to the bubble traveling through the cold fluid which changes the flow from laminar to turbulent flow. A great amount of heat was transferred at the first steps of air injection rates of (2-4) liter/min because at the low injection rates the size of bubbles was too small which makes the heat transfer rates better than the other air injection rates.

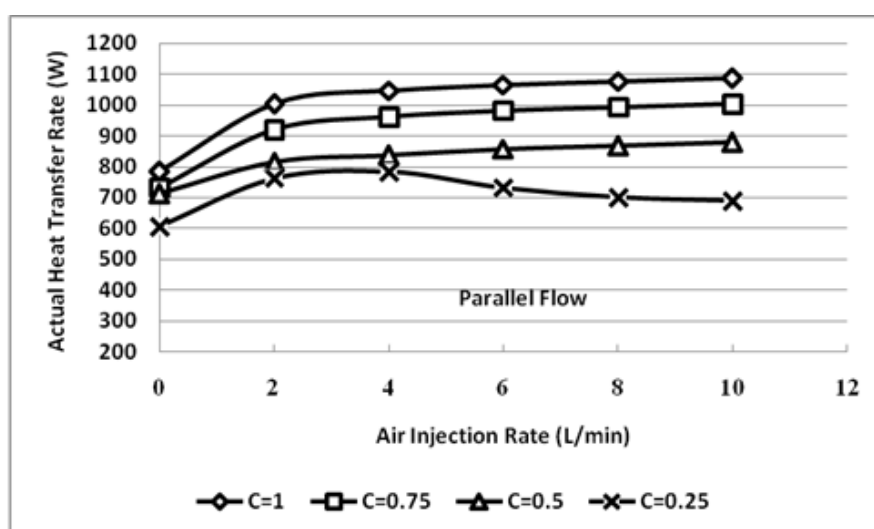


Fig. (7): The Effect of Air Injection Rate on The Actual Heat Transfer Rate

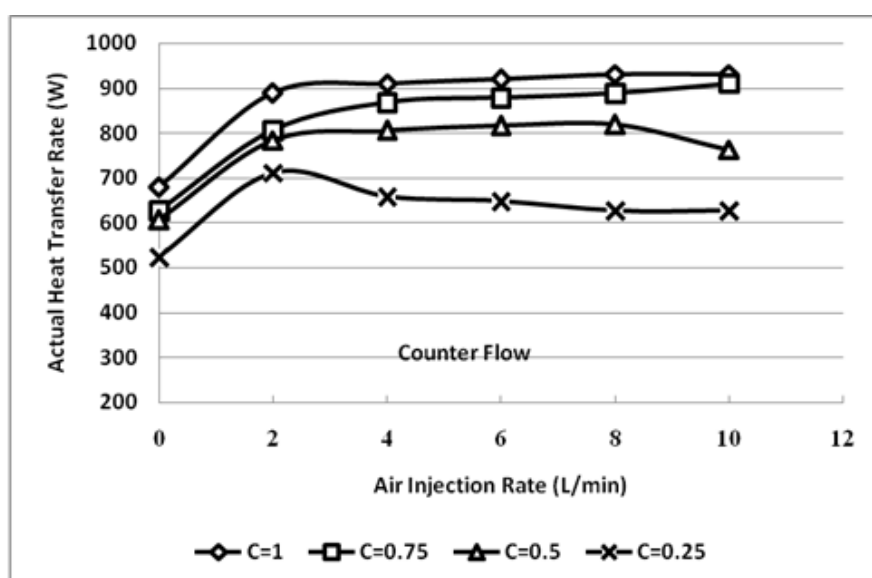


Fig. (8): The Effect of Air Injection Rate on The Actual Heat Transfer Rate

Fig. (9) and Fig. (10) represent the relationship between the air injection rates (bubble generation) and effectiveness for both flow types (parallel and counter) at different heat capacity ratios of (1, 0.75, 0.5 and 0.25). It is observed that the air injection rates (bubble generation) increases the effectiveness about (5%-15%) for all cases and at the case of heat capacity of (0.25) the effectiveness increases more than the others for the air injection rates of (2 and 4) liter/min and then decreases, this fact indicates that the air injection rates should be limited for obtaining the best performance according to the operating case.

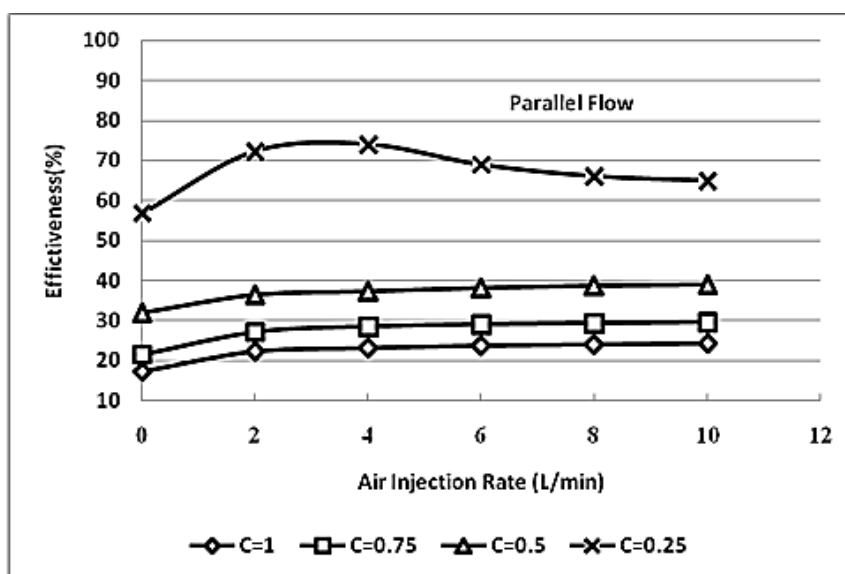


Fig. (9): The Effect of Air Injection Rate on The Actual Heat Transfer Rate.

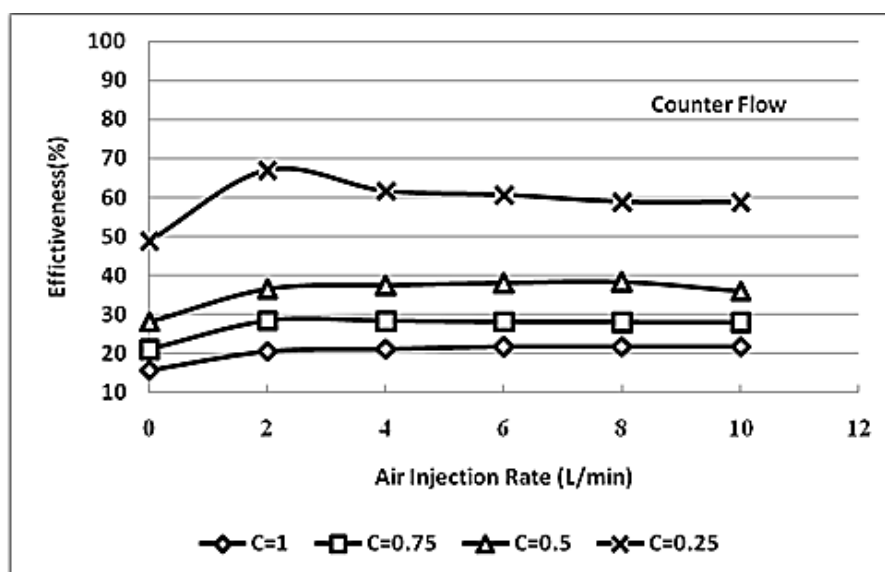


Fig. (10): The Effect of Air Injection Rate on the Actual Heat Transfer Rate

Fig. (11) and Fig. (12) represent the relationship between the air injection rates (bubble generation) and NTU(number of transfer units) for both flow types (parallel and counter) at different heat capacity ratios of (1, 0.75, 0.5 and 0.25). It is observed that the air injection rates (bubble generation) increases the NTU about (25%-40%) for the cases of ($C=1$ and $C=0.75$) and at the other cases of heat capacity of (0.5 and 0.25) the NTU increases about (25%-30%). In case of ($C=0.25$) the NTU increases for air injection rates of (2 and 4) liter/min and then decreases, this fact indicates that the air injection rates should be limited for obtaining the best performance according to the conditions of operating.

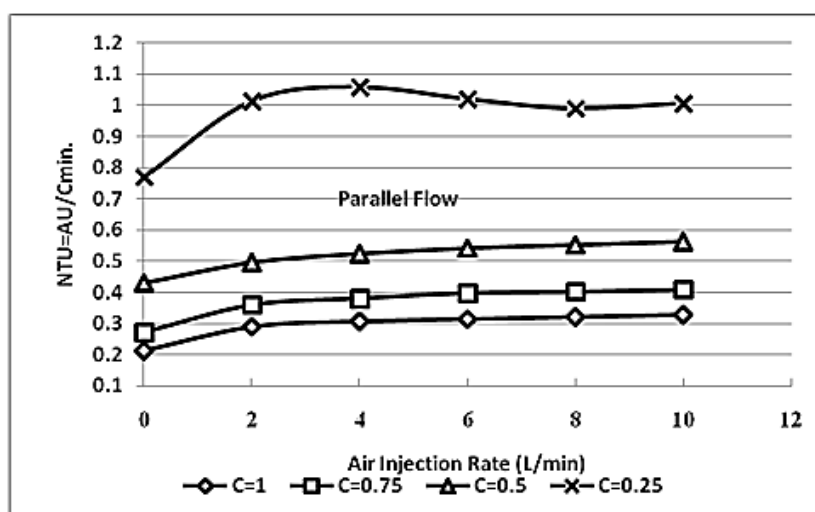


Fig. (11): The Effect of Air Injection Rate on The Number of Transfer Units.

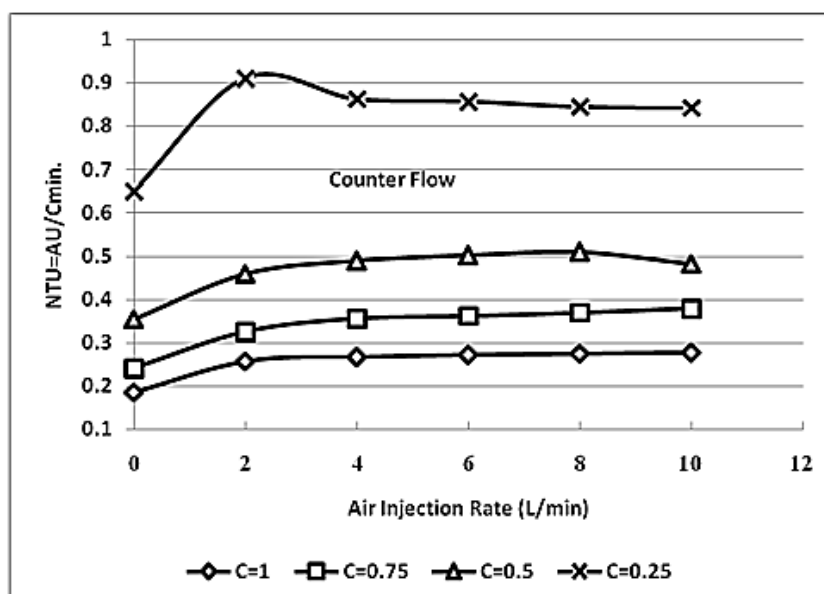


Fig. (12): The Effect of Air Injection Rate on the Number of Transfer Units

Fig. (13) and Fig. (14) represent the relationship between air injection rates (bubble generation) and overall heat transfer coefficient for both flow types (*parallel and counter*) at different heat capacity ratios of (1, 0.75, 0.5 and 0.25). It is observed that the air injection rates (bubble generation) increases the overall heat transfer coefficient more than 20%-30% and greatest (rapidly)effect was noted at case of air injection rate of (2L/min) comparing with other cases for both configurations and the positive effects were clearly noted except the case of heat capacity ratio of (0.25) which indicates that the overall heat transfer coefficient begins to downward after (4L/min for *parallel flow*) and (2L/min for *counter flow*), in these cases the operating conditions must be taken into accounts according to flow configuration.

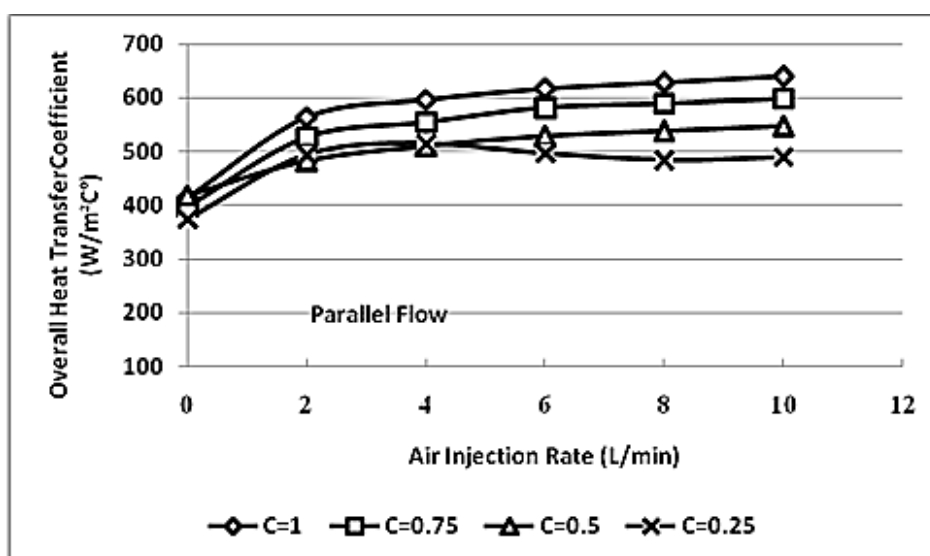


Fig. (13): The Effect of Air Injection Rate on the Overall Heat Transfer Coefficient.

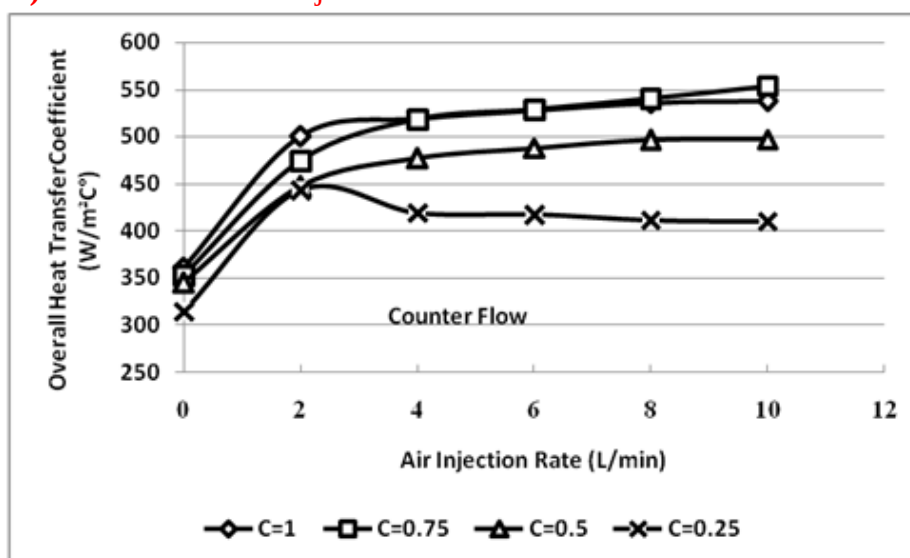


Fig. (14): The Effect of Air Injection Rate on the Overall Heat Transfer Coefficient.

7. Conclusions:

- 1- The bubble generation can improve the heat transfer rate about 30 percent for some cases and the other cases require an adjustment process by controlling the air injection rates (bubble generation) for obtaining the better results according to the operating conditions.
- 2- The features of this application are simple, less requirements, and proved a reasonable activity for improving the heat exchangers performance.
- 3-controlling and adjusting the bubble generation can be applied very easy according to the case requirements and operating conditions (parallel or counter) configurations.

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