

MIT Open Access Articles

*Experimental Investigation of the Flow
Maldistribution Inside an Air-Cooled Heat Exchanger*

The MIT Faculty has made this article openly available. **Please share**
how this access benefits you. Your story matters.

Citation: Habib, M. A., S. A. Said, A. Khalifa, M. A. Nemitallah, and T. Ayinde. "Experimental Investigation of the Flow Maldistribution Inside an Air-Cooled Heat Exchanger." *Arabian Journal for Science and Engineering* 39, no. 11 (October 4, 2014): 8187–8198.

As Published: <http://dx.doi.org/10.1007/s13369-014-1384-0>

Publisher: Springer Berlin Heidelberg

Persistent URL: <http://hdl.handle.net/1721.1/103147>

Version: Author's final manuscript: final author's manuscript post peer review, without publisher's formatting or copy editing

Terms of Use: Article is made available in accordance with the publisher's policy and may be subject to US copyright law. Please refer to the publisher's site for terms of use.



Experimental investigation of the flow maldistribution inside an air cooled heat exchanger

By:

M.A. Habib¹, S.A. Said¹, A. Khalifa¹, M.A. Nemitallah^{1,2} and T. Ayinde¹

¹ Mechanical Engineering Department, Faculty Of Engineering, KFUPM, Dhahran 31261, Saudi Arabia.

² Mechanical Engineering Department, Massachusetts Institute of Technology, 77 Mas Avenue, Cambridge, USA

Abstract

Flow maldistribution in heat exchanger tubes can significantly affect its performance. In this work, 16 tubes are connected between the inlet and the exit headers forming the heat exchanger. The feed nozzle is connected to the inlet header and its connection point can be altered. The influences of inlet flow Reynolds number (Re), nozzle diameter, number of nozzles and nozzle location on the flow maldistribution are experimentally investigated. Water is chosen to be the working fluid inside the heat exchanger set of tubes. At lower flow rates, the results showed that the flow Reynolds number has a significant effect on the flow maldistribution inside the heat exchanger set of tubes; however at higher flow rates, this effect was insignificant. Locating the nozzle at the center of the inlet header resulted in about 25% to 30% reduction in the standard deviation (STD) of the flow rate inside the tubes. Increasing the number of inlet nozzles resulted in an insignificant effect on the flow maldistribution. Increasing the nozzle diameter resulted in increased standard deviation of the flow rate distribution among the tubes and pressure drop across the tubes at the considered heat exchanger geometry and water flow rate.

Key words: Flow maldistribution; Air cooled; Heat exchangers; Experimental study.

Corresponding author: M.A. Nemitallah, E-Mail: mahmed@mit.edu and medhatahmed@kfupm.edu.sa, Tel: +966-3-860 4959.

Nomenclature

$\phi_{i)_{avg}}$	Average value of any variable
CFD	computational fluid dynamics
GPM	Gallons per minute
hp	Horse power
PVC	Polyvinyl chloride pipes
Re	Reynolds number
STD	Standard deviation

1. Introduction

It is of highly importance to understand through experiments and CFD modeling the influence of operating parameters on the flow distribution to the individual passes. There are many studies in the literature considering this phenomenon numerically; however, small number of studies were carried out on certain types of heat exchanger

were performed experimentally. The flow maldistribution mechanism between the tubes of the heat exchangers needs to be well studied in order to address the main parameters causing this flow maldistribution. Maldistribution of flow in the header can be affected by the header orientation, velocity of the inlet flow and geometry. The main goal while designing a heat exchanger is to obtain a uniform flow distribution inside the heat exchanger tubes in order to obtain a heat exchanger with uniform cooling.

Anjun et al. [1] conducted an experimental study on the influences of both the diameter of the inlet and the first and the second header diameters on the flow maldistribution inside a plate-fin heat exchanger (PFHE). Correlation of the dimensionless flow maldistribution parameter and Reynolds number was obtained under different header configurations. An experimental investigation was carried out by Prabhakara et al. [2] to measure the difference in pressure through the port to the channel inside plate heat exchangers using low corrugation angle plates under large range of operating Reynolds number starting from 1000 to 17000 and for varied number of channels, 20 and 80. The working fluid was water for both cases of hot and cold fluids. The results indicated that as the flow maldistribution increases the overall pressure losses in the plate heat exchangers increases. Hoffmann et al. [3] described experimentally flow maldistribution for a plate fin and-tube heat exchanger. However, the authors have not considered its impact on the heat exchanger thermal efficiency. The topic was presented in more detail by Hoffmann et al. [4]. Pipatpaiboon et al. [5] performed an experimental study on a 17-tube thermo-syphon heat exchanger (TPHE) using different working fluids including methanol, distilled water, and refrigerant 134-a. The heat exchanger in their work was used in a factory to cool the biodiesel. They reported that, under uniform cooling of the heat exchanger, the biodiesel temperature can be reduced from 120 to 80 °C.

There are many numerical models which were applied in order to understand the flow maldistribution phenomenon. Different models that take into account flow maldistribution effects in plate and cross flow heat exchangers were described by Luo and Roetzel [6]. In their work, a system of governing equations was solved using both numerical inverse transform and Laplace transform algorithms. They showed that, for plate-fin heat exchangers made of aluminum, the influence of the lateral heat conduction resistance of fins on the flow temperature and flow maldistribution is insignificant and they attributed this to the high fin efficiency. Ranganayakulu and Seetharamu [7] carried out a study, using a finite element method, on a plate fin, compact and cross flow heat exchanger, considering the influences of both the exchanger wall and non-uniform inlet fluid flow distribution on both hot and cold fluid sides and the two-dimensional longitudinal heat conduction. Using a finite element code, the mathematical equations were solved for different types of inlet flow and temperature maldistributions. Based on that, the heat exchanger effectiveness and its deteriorations due to flow and temperature maldistributions were calculated. They reported a significant effect of the flow and temperature maldistribution on the heat exchanger performance deteriorations. The effect of flow maldistribution on the thermal performance of cross-flow heat exchanger and the deterioration or promotion due to the flow maldistribution has been investigated numerically by Yuan [8]. They indicated that the best flow maldistribution mode promotes the thermal performance of cross-flow heat exchanger occurs when the number of transfer units (NTU) and heat capacity rate ratios are large.

A review of the flow distribution performance in a plate-fin heat exchanger has been studied by Jiao et al. [9]. The study of Rao et al. [10] proved that the optimum design

of the header configuration can greatly improve the performance of flow distribution in plate fin heat exchanger. A better approach of the analysis of the heat transfer data for plate heat exchangers was suggested by Rao et al. [11]. Habib et al. [12], based on numerical studies, provided correlations of flow maldistribution parameters in air-cooled heat exchangers and indicated that the inlet flow Reynolds number and nozzle geometry do not greatly influence flow maldistribution. In addition, the results indicated that reducing the nozzle diameter results in an increase in the flow maldistribution. It was found that increasing the number of nozzles results in a significant influence on the maldistribution. The results indicated that incorporating a second header tends to reduce the flow maldistribution. Bhramara et al. [13] carried out a CFD analysis of two phase flow of refrigerants inside a horizontal tube using homogeneous model under adiabatic conditions. The analysis was performed to evaluate the local frictional pressure drop at different flow rates and saturation temperatures. More recently, Habib et al. [14] evaluated flow maldistribution in air-cooled heat exchangers. They evaluated the effects of number of nozzles, nozzles location, geometry and diameter on maldistribution inside the heat exchangers. Josedite et al. [15] studied the thermal fluid dynamics of water/ultra-viscous heavy oil separation process in a hydro-cyclone. They presented a steady state mathematical model which simulates the performance of a non-isothermal separation process. The Eulerian-Eulerian approach for the interface of the phases involved (water/ultra-viscous heavy-oil) is used and the two-phase flow is considered as incompressible, viscous and turbulent. It was determined from their study that the separation efficiency was higher for higher fluid inlet velocity of the mixture, when the average temperature of the fluid in the hydro-cyclone was increased and for bigger oil droplets size (10^{-3} m). Luo et al. [16] and Meyer and Kroger [17] concluded similar results about minor up to 5% effects of this phenomenon. The effects of maldistribution in fin-tube heat exchangers have been investigated by Aganda et al. [18]. It was found that flow maldistribution greatly influences the mean and standard deviation. Hetsroni et al. [19] performed experiments to study the flow regimes and heat transfer in water-air flow in inclined tubes. Conductive Tomography and infrared tomography was used in their investigation. Their analysis showed that dry out took place in the open annular flow regimes with motionless or slowly moving droplets. Under the investigated conditions, the heat transfer coefficient was determined about 10 times higher than that for single phase airflow. The problem of flow maldistribution can have serious effects on heat exchangers performance especially, in case of highly turbulent flow [20]. Flow maldistribution has been studied also in other applications like industrial air heaters by Jonas et al. [21]. In their work, they studied the influences of inlet flow conditions on the air side hydraulic resistance and flow maldistribution inside an industrial air heater. There is a lack of data of flow maldistribution in air cooled heat exchangers which have lot of industrial applications. The objective of the present study is to experimentally investigate the effects of flow parameters, and different geometrical dimensions on the flow maldistribution inside an isothermal single-phase air-cooled heat exchanger. The specific objectives include 1) Experimental investigation of the influence of the flow Reynolds number on the flow maldistribution and 2) Experimental investigation of the influence of the number of inlet nozzles, nozzle diameter and location on the flow maldistribution. It should be noted that the present study is of the experimental type. The previous studies by Habib et al [12, 14] are based on numerical investigations.

2. Experimental setup and Instrumentations

The main parameters influencing maldistribution of flow in the header are the header inlet flow rate and number and location of inlet nozzles. The present work aims at experimentally, studying the effect of these parameters on the distribution of flow into the individual passes. The present work focuses on the flow inside the header and the tubes and excludes the heat transfer process. To evaluate the flow maldistribution in the heat exchanger tubes, sixteen tubes are connected between the two headers, as shown in Fig.1. The headers are made of plexiglass sheets formed in square duct shape, with removable cover. The diameter of each tube is 3/8 inches, separated a part by a distance of 3 inches. In most of the conducted experiments, the flow was passed through only eight tubes in order to increase the flow rates and the other tubes were left closed. Different sizes of nozzles were tested at different locations with respect to header center. Data were collected and compared for different number of nozzles at different locations (Fig.1 shows the setup for two nozzles separated apart by a distance of 500 mm, from center to center, with a diameter of 70 mm for each nozzle).

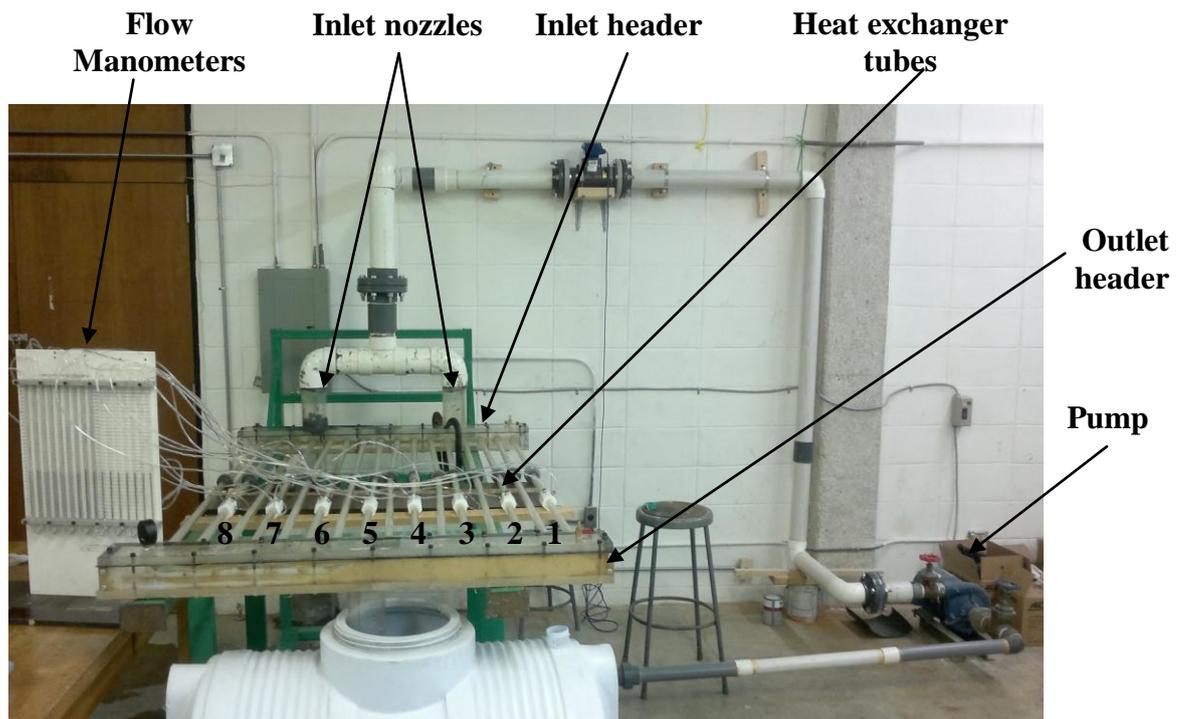


Fig.1 photograph showing the main components of the test ring and tube numbering.

2.1 Experimental facility

As shown earlier by Habib et al. [14], the maldistribution problem is serious in the first set of tubes. In the following passes, the flow does not exhibit any significant maldistribution and the flow becomes very uniform. Therefore, the present study does consider a single pass air-cooled heat exchanger. The experimental setup shown in Fig.1 is composed of two main parts, namely, the flow loop and the test section. The

flow loop (closed-type) consists of a centrifugal-type water pump of 5 hp, a PVC piping system fitted with valves for flow control and two (upper and lower) reservoirs. The two reservoirs were made of fiberglass with 2 m³ total volume. The working fluid, water, was pumped from the lower reservoir to the inlet header and through the test section back to the lower reservoir. The inlet header works as the main distributing chamber to the test section tubes with an inside height of 10 cm. The volume flow rate through the test section was controlled through the delivery valve of the pump in addition to using a ball valve. Standard venturi meters were used for flow measurement inside the eight numbered tubes, as shown in Fig.1. A set of U-tube manometers were used to measure the pressure drop in the venturi meters. The system was supported by a steel structure designed to match the purpose of the experiment. The total volume flow rate to the heat exchanger was measured using a turbine flow meter. The total water supply was measured by an inline turbine flow meter and the values were compared with the sum of tubes flow rates, in order to check for the accuracy of the flow measurements.

2.2 Operating conditions

The header has a length of 1231 mm and a cross section of 100 mm width and 58.9 mm height. These dimensions represent the geometry of a real industrial air-cooled heat exchanger. The nozzle location effect was investigated in the present work. Two locations were considered for a single nozzle operation, one is at the center and the other location is at the right side of the inlet header. The considered nozzle has a diameter of 92 mm for all experimental sets except for the case of using two inlet nozzles. In this case, the flow was divided between the two nozzles and the diameter for each nozzle was changed to be 70 mm in order to keep the same velocity at nozzle exit like the case of using a single nozzle. Eight venturi meters were used to measure the fluid flow rates inside the tubes. The tube sheet has sixteen tubes; eight of them are installed with standard calibrated venturi meters for flow rate measurements. All the components of the test section, Fig.1, were made out of Plexiglas. The test section was manufactured in a way that allows testing the location and number of nozzles as well as different Reynolds numbers.

In this work, five sets of measurements were conducted. The first set considers the influence of the inlet flow rate (Re at the exit of the inlet nozzle) on the standard deviations. In this case, the considered flow rates were ranging from 45 GPM ($2.8395 \times 10^{-3} \text{ m}^3/\text{s}$) to 94 GPM ($5.9314 \times 10^{-3} \text{ m}^3/\text{s}$), and the corresponding Reynolds numbers ranges from 36,000 to 76,000. The Reynolds numbers were calculated at the exit of the inlet nozzle and were based on a nozzle diameter of 92 mm. In this set of measurements, all of the 16 tubes were left opened. For the second set of experiments, the flow went through only 8 tubes and the other 8 tubes were blocked. In order to investigate the effect of the inlet nozzle location on the maldistribution through the heat exchange, a third set of experiments was conducted considering only 8 opened tubes. In this regard, two nozzle locations were tested. These locations were at the header center and the header end side. The number of inlet nozzles was also investigated through a fourth set of experiments considering only 8 opened tubes. The last set of experiments was performed in order to investigate the nozzle diameter effects. The considered nozzle diameters were 2, 3 and 4 inches with only 8 tubes are opened. The nozzles in the last set of experiments were attached to the center of the inlet header.

The uncertainty analysis was performed using the method described by Holman et al. [22]. In the present experiments, water temperature and flow rates were measured

using appropriate instruments as described above. The water temperature was measured in order to calculate the appropriate values of the fluid dynamic viscosity and density in order to be used in the Reynolds number calculations. A k-type thermocouple was used for the temperature measurements. The uncertainties in the exhaust gas temperature and the water volume flow rate are ± 0.5 °C and $\pm 3\%$, respectively. Reproducibility of results was checked by repeating a set of tests for two times under the same conditions and taking the average values.

3. Results and discussions

The main parameters influencing the flow maldistribution that are considered in the present work are the inlet flow rate and the number and location of the nozzles. The results were obtained for a single and double nozzle. The nozzle diameter was kept unchanged. Different inlet volume flow rates in the range from 40 GPM (2.524×10^{-3} m³/s) to 100 GPM (6.31×10^{-3} m³/s) were used. The main criterion used for the evaluation of flow maldistribution is the standard deviation (STD) in the mass flow rate distribution in the tubes. The standard deviation, STD, of a variable ϕ_i is given as:

$$STD = \sqrt{\left[\frac{1}{n-1} \sum_{i=1}^n \left(\frac{\phi_i}{\phi_i}_{avg} - 1 \right)^2 \right]} \quad (1)$$

Where ϕ_i is the average value and n is the number of tubes in the case of ϕ_i = mass flow rate.

3.1 All tubes are opened

The distribution of the mass flow rate in the exchanger tubes (numbered as shown in Fig.1) is shown in Fig.2a. The considered values of inlet flow rates were 45, 60, 69, 80 and 94 GPM (corresponding to 2.8395×10^{-3} , 3.786×10^{-3} , 4.3539×10^{-3} , 5.048×10^{-3} , and 5.9314×10^{-3} m³/s, respectively). In this case, a nozzle of 92 mm diameter was attached at the center of the feed water header. The figure indicates that the flow maldistribution is similar for all the cases of the different flow rates; however, the values are different. This behavior reflects the fact that the maldistribution is higher at low flow rates in comparison to the high flow rate cases. For all the cases, the figure shows drop in flow rate in the two tubes located at the header end and at the center of the header. The figure indicates that the inlet flow (or Reynolds number) has a slight influence on the maldistribution. Upon dividing the average mass flow rate for each case and normalizing each value as shown in Fig.2b. It was found that the distribution is same for all mass flow rates, except at $Re = 36148$, which corresponds to the lowest flow rate. The flow in the tubes is correlated with the pressure head across each of the tubes. Therefore, the maldistribution among the tubes may be attributed to the pressure variation along the header length. These results were confirmed in our previous numerical work, Habib et al. [14]. At low values of inlet flow rates, the amount of flow recirculation in both sides of the inlet section to the feed header (nozzle exit) is increased. As a result, low pressure regions are created in the front of the center tubes that are close to the inlet nozzle, and this should result in reducing the flow rates to those tubes. A back flow from the outlet header may happen if the inlet flow rates dropped below a certain value due to the back pressure effects. There is

also another low pressure region at the end side of the feed header and this should result also in a reduction of the flow rates through the end tubes as shown in Fig.2. However, the low pressure region at the end side of the header is due to the pressure losses through the header itself. This reduction in pressure depends on the length and cross sectional area of the inlet header and in most of the cases the flow maldistribution is small in the end tubes as compared to the center tubes close to the inlet nozzle. The chances for back flow in the end tubes are small as compared to those for the center tubes due to the intense flow recirculation close to the inlet nozzle [14]. At high values of inlet flow rates, the flow inertia force is increased and the amount of flow recirculation is reduced and this justifies the lower values of flow maldistribution in the cases of high inlet volume flow rates.

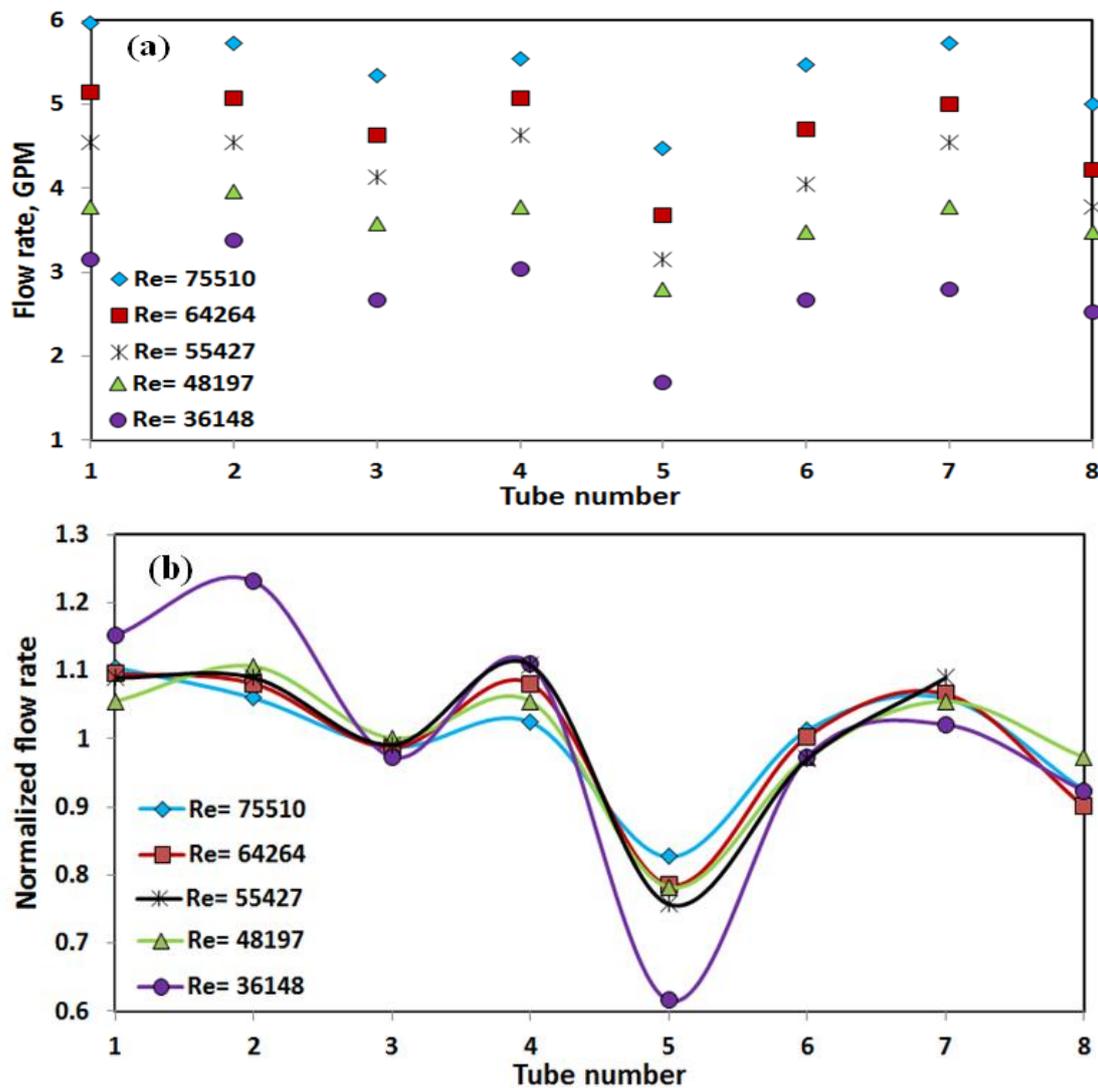


Fig.2 Maldistribution of flow rate, (a), and maldistribution of normalized flow rate, (b), in the air cooled heat exchanger tubes at different values of inlet Reynolds number for the case of all the 16 tubes are opened.

The effect of the inlet flow Reynolds number on the standard deviations in the values of the mass flow rates is shown in Fig.3 for the case of all 16 tubes are opened. The figure indicates that the standard deviation (STD) in the mass flow rate which causes the flow maldistribution to decrease as the Reynolds number (inlet volume flow rate)

increases. For the case of low value of inlet volume flow rate and according to Fig.3, it can be seen that the standard deviation (STD) in the mass flow rate is significantly high. This can be attributed to the increased chances for pressure variations around the inlet nozzle which will result in reduced pressure values at the inlet section of the tubes which are close to the inlet nozzle. In contrast, at high values of volume flow rates, it is clear in Fig.3 that the variations in the standard deviation between the cases of different inlet flow rates are very small. The results of the inlet volume flow rates effect on the standard deviations (STD) in the volume flow rates inside the tubes inside the header tubes are summarized in Table 1 (first row of data). The table assures that the variations in the STD of the mass flow rate are less than 0.02 GPM ($1.262 \times 10^{-6} \text{ m}^3/\text{s}$) at higher values of inlet flow rates. This clearly indicates that the Re has significant influence at lower values of volume flow rates and its effect is reduced by increasing the inlet flow rate through the inlet nozzle.

Table 1 Calculated values of the standard deviation (STD) in the inlet mass flow rate based on the measurements at different operating conditions.

Water flow rate, GPM ($10^{-3} \text{ m}^3/\text{s}$)	45 (2.84)	50 (3.15)	60 (3.79)	69 (4.35)	70 (4.42)	80 (5.05)	88 (5.55)	92 (5.81)	94 (5.93)
Using single nozzle (center-located) and 16 opened tubes	0.31	-	0.16	0.18	-	0.14	-	-	0.088
Using two nozzles and 16 opened tubes	0.27	-	0.21	-	0.18	0.15	-	0.11	-
Using single nozzle (center-located) and 8 opened tubes	-	0.09	0.079	-	0.081	0.082	0.086	-	-
Using single nozzle (end-located) and 8 opened tubes	-	0.106	0.109	-	0.11	0.107	0.084	-	-

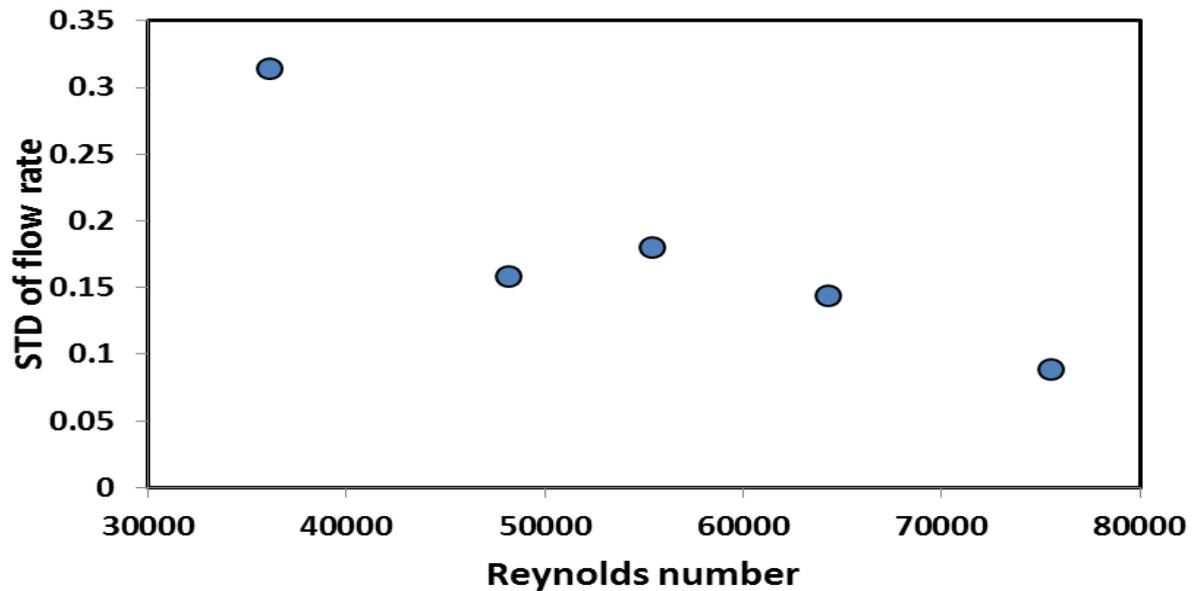


Fig.3 Standard deviation of the flow rate distribution at different inlet Reynolds numbers for the case of all the 16 tubes are opened.

3.2 Half tubes are opened

In order to validate the above results, another experiment was conducted using only 8 opened tubes. In this case, the flow through each tube was increased as a result of closing half of the exchanger tubes. The maldistribution of the flow inside the considered tubes is shown in Fig.4a at 50, 60, 70, 80, and 88 GPM (corresponds to Reynolds numbers of 43657, 52388, 61120, 69851, and 76836, respectively). The flow maldistribution is small at higher values of the inlet flow rates. The most affected tubes by the maldistribution at lower values of the inlet flow rates are the tubes at the center and end side of the inlet header. However, there are some differences which are clear from the comparison of Figs. 2a and 4a. In the case of all tubes are opened as shown in Fig.2, the maldistribution resulted in a reduction in the flow in the mostly affected tubes at the center and side locations of the inlet header. This can be attributed to the pressure build up in the region beside the inlet nozzle when the adjacent tube to the nozzle is closed (tube beside tube number 5). At the end tube and due to the pressure losses inside the inlet header, the flow inside the tube is reduced. Thus, it can be concluded that the flow maldistribution is much more profound at low flow rates. The normalized profiles are presented in Fig.4b for the same operating conditions. In this figure, the flow rates in the different tubes were normalized with respect to the average flow value. The figure shows that, apart from the region close to the header and in the center region, all profiles coincide. In case of closing half of the tubes, the low pressure zone was moved from the zone just beside the inlet nozzle (in case of all tubes are opened) to the tubes just following the closed one beside the inlet nozzle. The pressure at the exit header is almost atmospheric which is lower than the pressure in the inlet header. That's why when a tube is closed, the pressure will build up in the front of this tube and this should result in movement of the low pressure region to the next opened tubes.

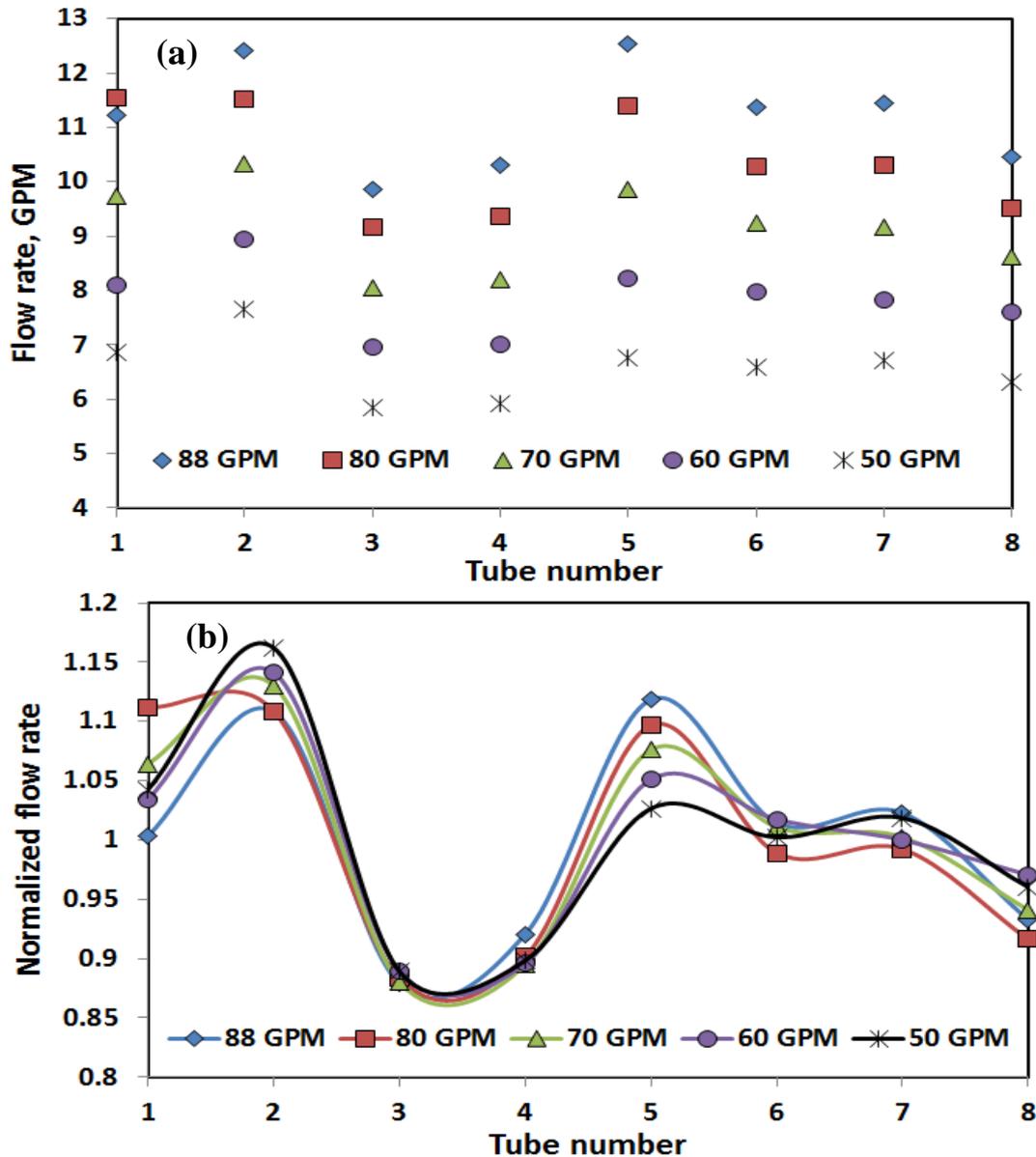


Fig.4 Maldistribution of flow rate, (a), and maldistribution of normalized flow rate, (b), in the air cooled heat exchanger tubes at different values of inlet total flow rate for the case of only 8 tubes are opened.

The root mean square is shown in Fig.5 and confirms the above conclusions and indicates that the STD of the flow in the tubes decreases monotonically with the increase in the inlet flow rate. However, the rate of decrease in maldistribution diminishes at high flow rates and further increase shows insignificant influence on flow maldistribution. It is shown that the figure exhibits a smooth profile for the case of 16 nozzles with eight of them blocked in comparison to the case of sixteen open tubes. The results of the influence of the inlet volume flow rates on the standard deviations in the mass flow rates in the tubes inside the inlet header are summarized in Table 1 (third row of data).

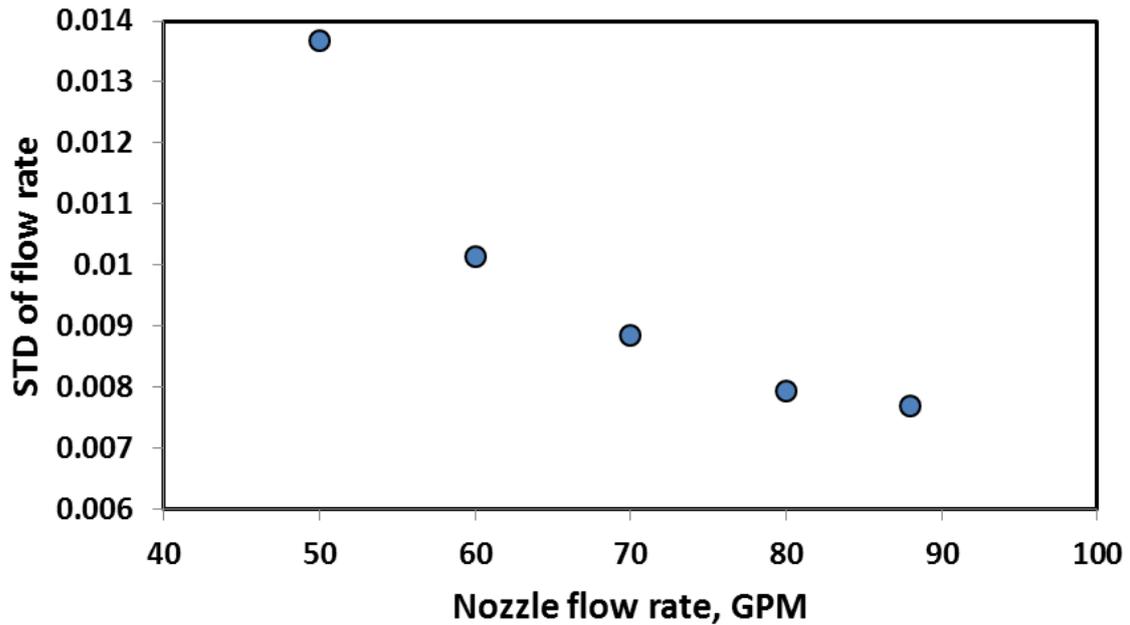


Fig.5 Standard deviation of the flow rate distribution at different inlet total flow rates for the case of only 8 tubes are opened.

3.3 Influence of the Nozzle Location

The effect of nozzle location on the flow maldistribution was studied considering only 8 open tubes and using the same nozzle with a diameter of 92 mm. The variations in volume flow rates inside the tubes were measured for two cases with nozzle location being located at the center and at the end side of the header. The influence of the nozzle location is shown in Fig.6a. The figure indicates that moving the inlet nozzle from the center of the inlet header towards its end side results in an increase in the flow maldistribution through the exchanger tubes. This is attributed to the loss of the benefit of flow symmetry around the inlet nozzle. In order to further investigate the influence of nozzle location, the flow rates were divided by their average value. The results are plotted in Fig.6b. Apart from the case of the lowest flow rate, the plots almost coincide with each other indicating lower maldistribution at higher velocity for both cases. However, for each mass flow rate, the figure indicates that the flow distribution among the tubes is very much different for the two cases of center and end side of the header.

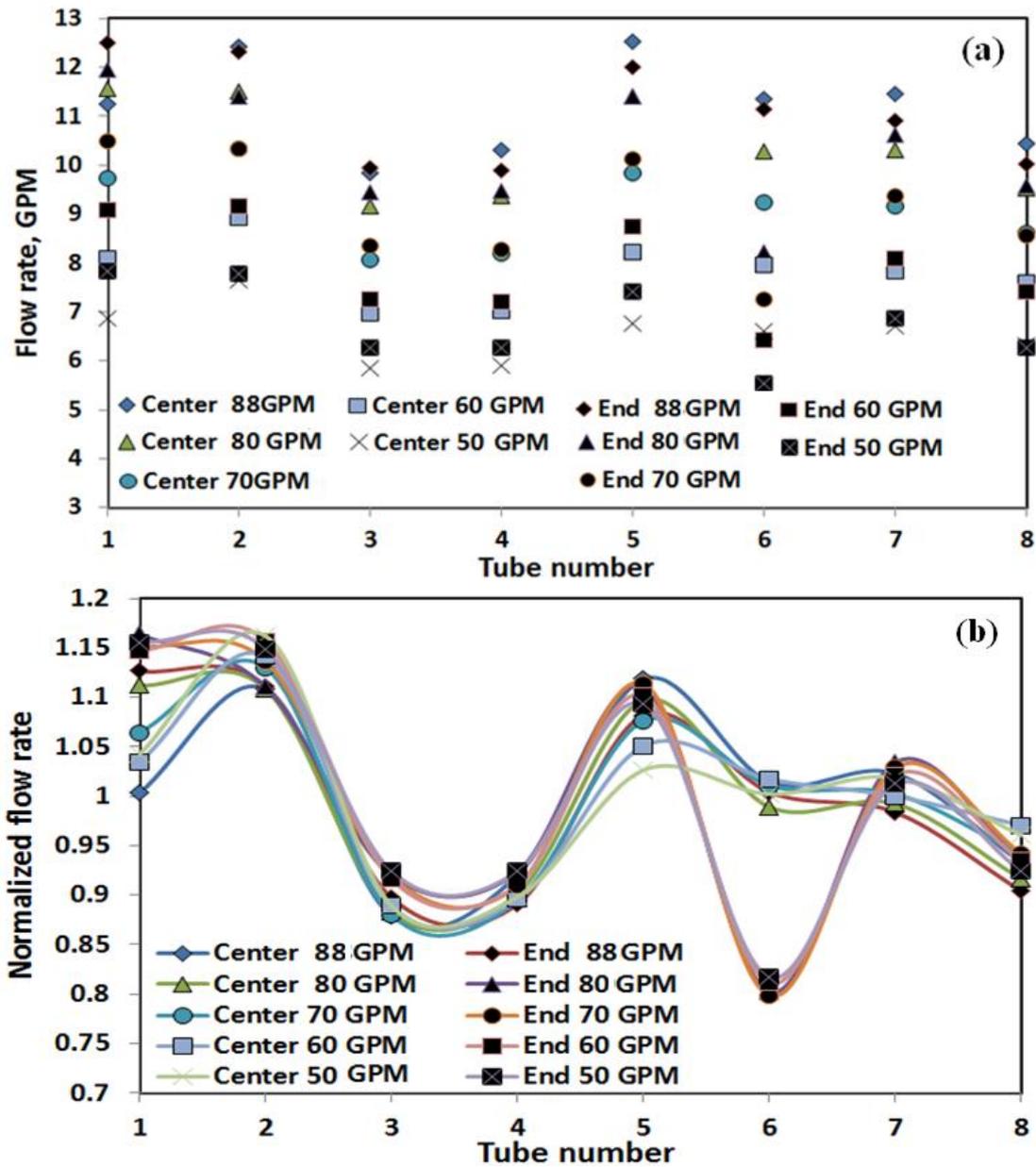


Fig.6 Maldistribution of flow rate, (a), and maldistribution of the normalized flow rate, (b), in the air cooled heat exchanger tubes at different values of inlet total flow rate and nozzle location for the case of only 8 tubes are opened.

The STD of the variations in the volume flow rates for the two cases of nozzle location is shown in Fig.7. As shown in the figure, 25% to 30% decrease in the standard deviation is obtained as a result of moving the nozzle location towards the center of the header. The results of the nozzle location effect on the standard deviations (STD) in the mass flow rates inside the tubes of the inlet header are summarized in Table 1 (fourth row of data). The numbers in the table also confirms the bad effects of losing the flow symmetry around the inlet nozzle when the nozzle is moved to the end side of the header.

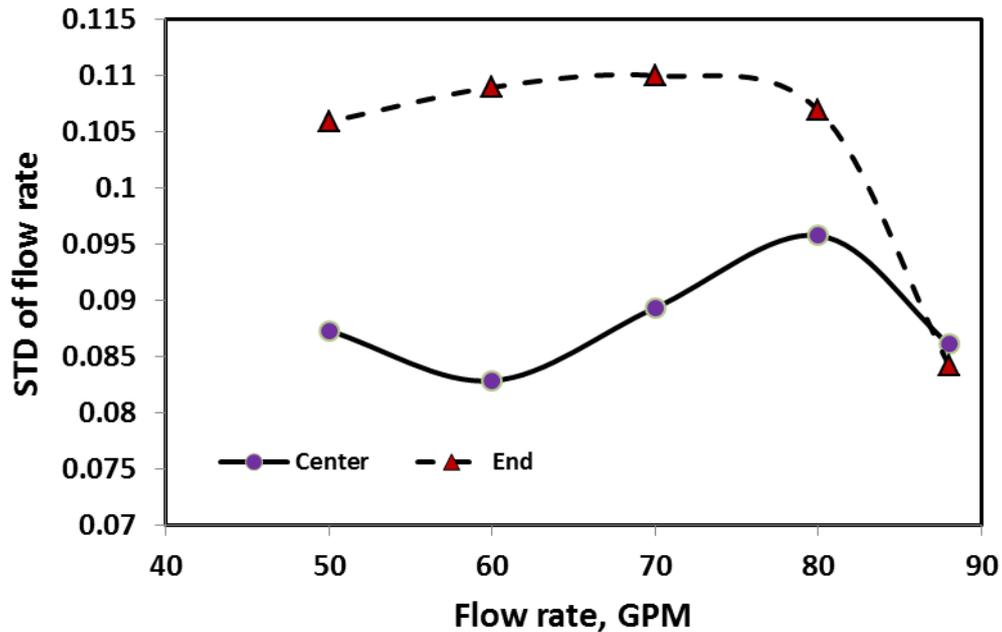


Fig.7 Influence of nozzle location and inlet total flow rate on the standard deviation of the flow rate distribution and for the case of only 8 tubes are opened.

3.4 Influence of the Number of Nozzles

Two cases were considered while opening all of the 16 tubes, one case using single inlet nozzle (92 mm diameter and located at the center of the inlet header) and the other case using two inlet nozzles spaced apart by a distance of 500 mm. The diameter of each of the two nozzles was calculated (each nozzle is of 70 mm diameter) in order to keep the same inlet flow rate like the case of single nozzle operation for the same inlet volume flow rate. The header center is placed in the center distance between the two nozzles. Figure 8a presents the volume flow rate distribution in the tubes for the case of two nozzles at 45, 60, 70, 80, and 93 GPM (corresponds to Reynolds numbers of 25820, 34426, 40164, 45902, and 53361, respectively).. The figure indicates similar distribution for high flow rates above 45 GPM ($2.8395 \times 10^{-3} \text{ m}^3/\text{s}$) where the distribution is quite nonuniform. These results assure that maldistribution is less at high flow rates. It is clear from the figure that the maldistribution behavior is again inversed especially on tube number 5 as compared to that of single nozzle operation. However, due to the reduced pressure losses because of the reduction in the distance from the inlet nozzle to the header side, the flow maldistribution in the side tubes was reduced as compared to the case of using single nozzle. The low pressure region has moved to the center of the header between the two inlet nozzles. Figure 8b shows a comparison between the values of the STD of the flow rates for different numbers of operating nozzles. The figure indicates decay in the STD as the flow rate increases for both cases. The influence of inlet flow rate or Reynolds number on the standard deviation is exhibited at any number of operating nozzles. However, it is shown that the levels of STD in case of two nozzles operation are lower than those for the single nozzle case at lower values of inlet volume flow rates. The behavior is reversed at higher flow rates and the single nozzle operation gives slightly lower standard deviation. The figure indicates insignificant influence of increase of the number of nozzles on the maldistribution. At high flow rates of excess of around 5.7 kg/s, the case of two nozzles provide reduced maldistribution than the case of single nozzle. This supports the numerical results of Habib et al [12, 14] which show similar

conclusions for flow rate of around 12 kg/s. Table 1 (second row of data) summarizes the results of the nozzle number effect on the standard deviations (STD) in the mass flow rates inside the tubes of the header. The table also assures that when the number of nozzles is increased from one to two nozzles with keeping the same inlet flow velocity, slight changes in the standard deviation in the mass flow rate inside the exchanger tubes were encountered.

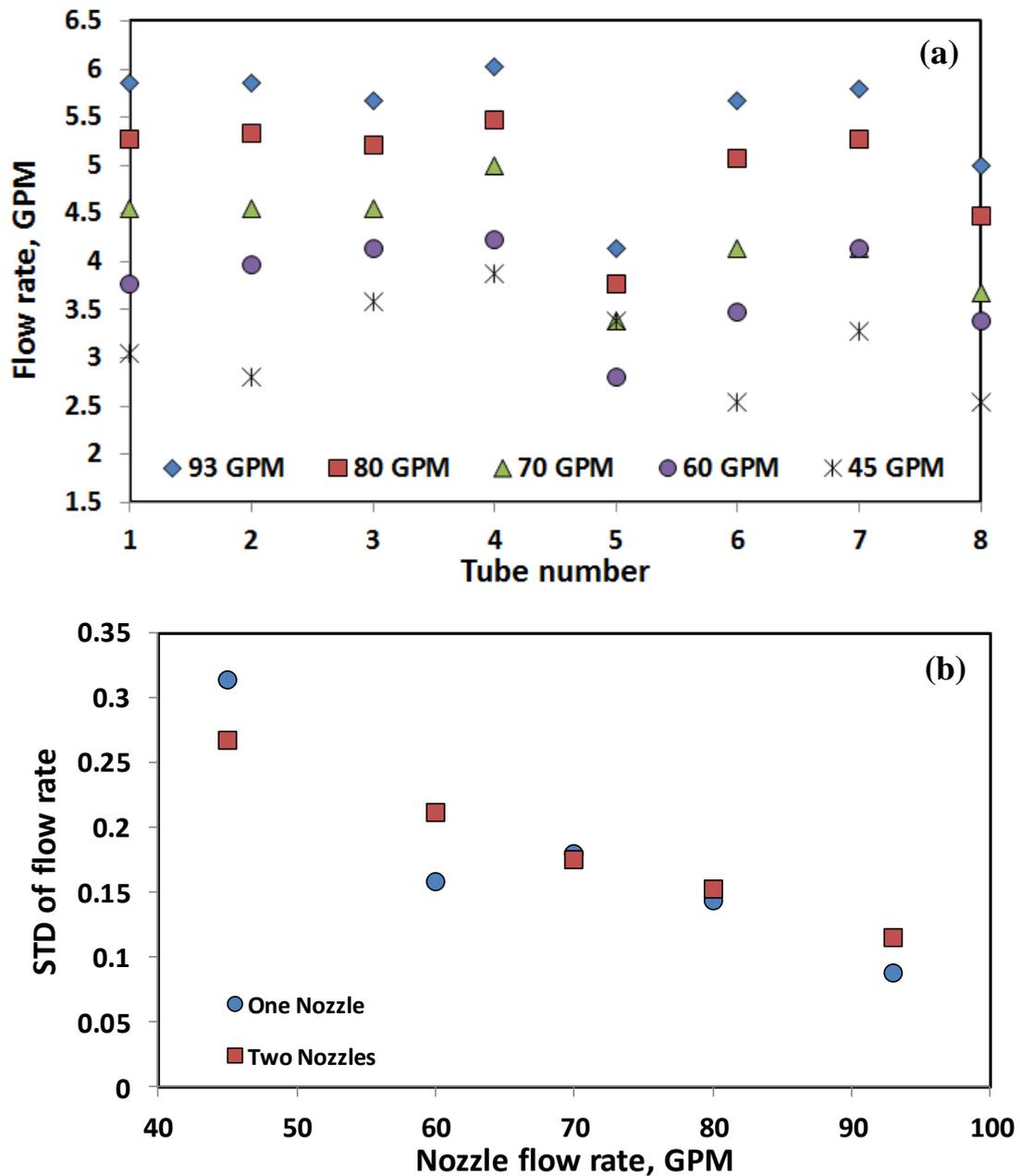


Fig.8 Maldistribution of flow rate using two nozzles, (a), standard deviations of the flow rate distribution for different number of nozzles, (b), at different inlet total flow rates and for the case of all tubes are opened.

3.5 Influence of the Nozzle Diameter

The influence of the nozzle diameter was examined in the present work considering the case of only 8 opened tubes. The considered nozzle diameters were 2, 3 and 4 inches and total volume flow rates of 60, 70 and 80 GPM (corresponding to 3.786×10^{-3}

³, 4.417×10^{-3} , and 5.048×10^{-3} m³/s, respectively) with fixing each nozzle at the center of the inlet header. The corresponding Reynolds numbers at 60, 70, and 80 GPM are 47720, 55673, and 63626 for the 4 inches nozzle; 63251, 73793, and 84334 for the 3 inches nozzle; and 94876, 110689, and 126502 for the 2 inches nozzle. Figure 9a presents the variations of the percent standard deviations in the volume flow rate distribution in the tubes for the three considered cases of the inlet volume flow rate.

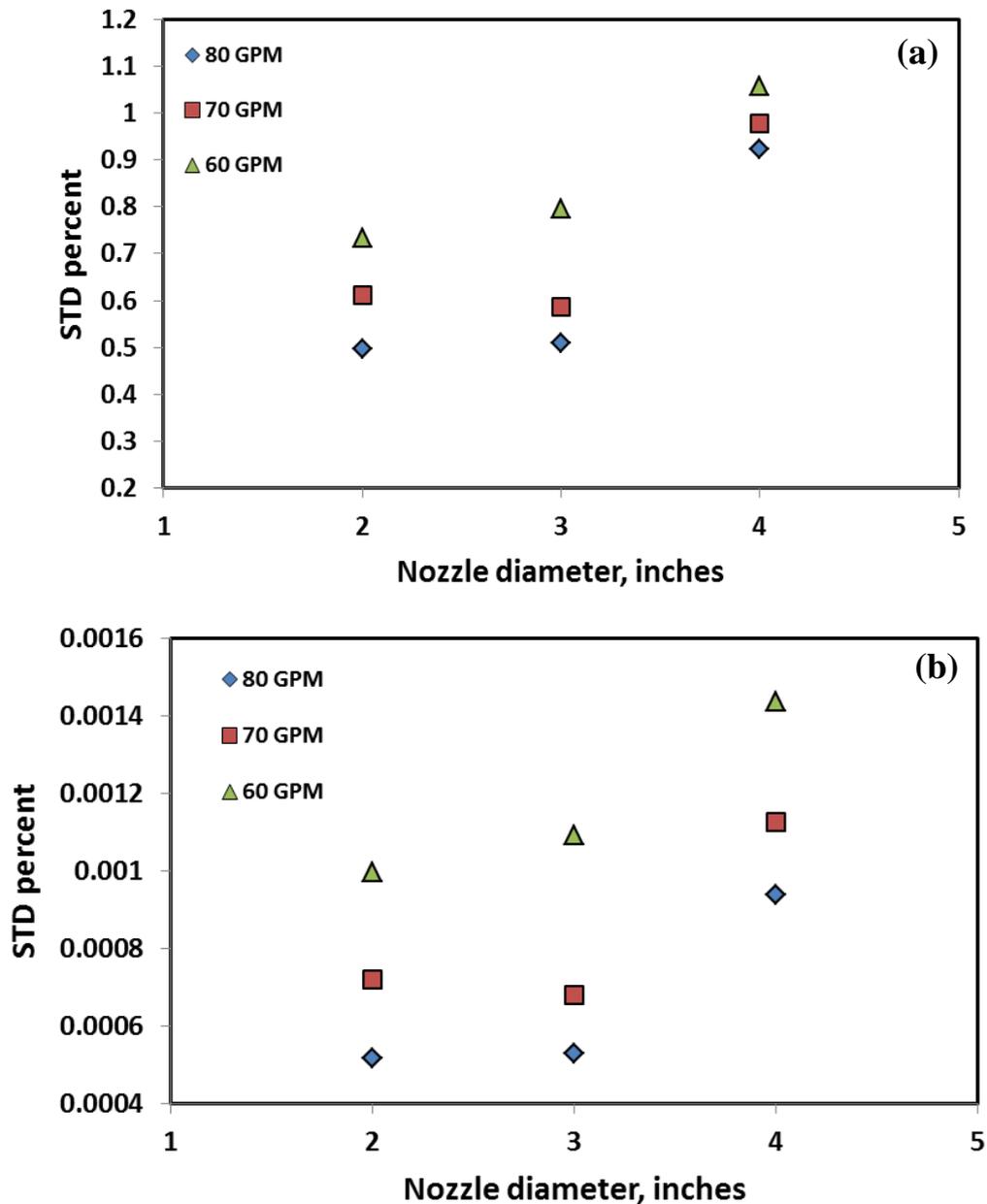


Fig.9 Influence of nozzle diameter on: (a) maldistribution of flow rate and (b) maldistribution of pressure drop, in the air cooled heat exchanger tubes at different values of inlet total flow rate for the case of only 8 tubes are opened.

The figure indicates a slight change in the standard deviation in the volume flow rate distribution among the tubes of the air cooled heat exchanger as the diameter increases from 2 to 3 inches. This is followed by sharp increase as the diameter changes from 3 to 4 inches. In order to explain the influence of the nozzle diameter,

the variation in the standard deviation in the pressure drop across the tubes at different nozzle diameters was calculated and is presented in Fig.9b. The figure confirms the results of Fig.9a. The pressure rise across the tubes is correlated with the high maldistribution in flow rates as it reflects the non-uniformity of the pressure inside the main inlet header. When the inlet nozzle diameter increases, this should result in reducing the inlet velocity. This effect is equivalent to reducing the inlet volume flow rate at fixed nozzle diameter which should result in an increase in the flow maldistribution as discussed above. This may justify the increase in the flow maldistribution when the diameter is increased as shown in Fig.9a. As a result of this flow behavior, the pressure fluctuations will be also increased as shown in Fig.9b. The present experimental results do not match the conclusions of Habib et al [12, 14] where the effect of nozzle diameter provide contradicting results. The differences may be attributed to differences in the heat exchanger geometry, number and location of nozzles and tube geometrical distribution. As well, the flow rates considered in the present study are much lower than the cases of Habib et al [12, 14].

4. Conclusions

The flow distribution mechanism of the header is essential for the design needs to be studied. There is a lack of data for the maldistribution in heat exchangers. Maldistribution of flow in the header is influenced by many parameters such as inlet flow velocity and the number, location and diameter of nozzles. The work presents experimental investigation of the influence of inlet flow Reynolds number, the number and location of nozzles on the maldistribution of the isothermal flow in the air-cooled heat exchangers. It is shown that the Reynolds number has significant influence at low flow rates and small influence on flow maldistribution at high flow rates. It was found that locating the nozzle at the center region of the header may result in 25-30% reduction in STD of the flow rate. When the number of nozzles was increased while keeping the same inlet flow velocity, slight changes in the standard deviation of the mass flow rate were encountered. The results indicated an increase in the standard deviation in the volume flow rate distribution among the tubes and in the pressure drop across the tubes when the nozzle diameter was increased.

Acknowledgement

The authors wish to acknowledge the support received from King Fahd University of Petroleum and Minerals for funding this work through Project No. IN090016.

References

- [1] J. Anjun, R. Zhang, J. Sangkwon, Experimental investigation of header configuration on flow maldistribution in plate-fin heat exchanger, *Applied Thermal Engineering* (2003), Volume 23, Issue 10, Pages 1235–1246.
- [2] R.B. Prabhakara, S. Bengt, K.D. Sarit, An experimental investigation of the port flow maldistribution in small and large plate package heat exchangers, *Applied Thermal Engineering* (2006), Volume 26, Issue 16, Pages 1919–1926.
- [3] V. Hoffmann, J. Neale, M. Walmsley, Flow Profiles on the Fin Side of a Plate Fin-And-Tube Heat Exchanger Experiencing Gross Flow Maldistribution, *Proceedings of 7th World Conference on Experimental Heat Transfer, Fluid Mechanics and Thermodynamics* (2009), on CD, Kraków, Poland, June 28 - July 3.

- [4] V. Hoffmann, J. Neale, M. Walmsley, The Effect of Inlet Conditions on the Air Side Hydraulic Resistance and Flow Maldistribution in Industrial Air Heaters, *International Journal of Heat and Fluid Flow* (2011), Vol. 32 (4), 834- 845, ISSN 0142-727X.
- [5] N. Pipatpaiboon, S. Rittidech, P. Meena, Experimental Study of a Thermosyphon Heat Exchanger (TPHE) in a Bio-diesel Factory in Thailand, *Arab J Sci Eng* (2012) 37:2047–2060.
- [6] X. Luo, W. Roetzel, The single-blow transient testing technique for plate-fin heat exchangers, *International Journal of Heat and Mass Transfer* (2001), 44 (19), 3745-3753.
- [7] C. Ranganayakulu, K.N. Seetharamu, The combined effects of wall longitudinal heat conduction and inlet fluid flow maldistribution in cross-flow plate-fin heat exchangers, *Heat and Mass Transfer* (2000), 36 (3), 247-256.
- [8] P. Yuan, Effect of inlet flow maldistribution on the thermal performance of a three-fluid cross-flow heat exchanger, *International Journal of Heat and Mass Transfer* (2003), 46 (20), 3777-3787.
- [9] A.J. Jiao, Y.Z. Li, C.Z. Chen, R. Zhang, Experimental investigation on fluid flow maldistribution in plate-fin heat exchangers, *Heat Transfer Engineering* (2003), 24 (4), 25-31.
- [10] A.J. Rao, R. Zhang, S.A. Jeong, Experimental investigation of header configuration on flow maldistribution in plate-fin heat exchanger, *Applied Thermal Engineering* (2003), 23 (10), 1235-1246.
- [11] B.P. Rao, P.K. Kumar, S.K. Das, Effect of flow distribution to the channels on the thermal performance of a plate heat exchanger, *Chemical Engineering and Processing* (2002), 41 (1), 49-58.
- [12] M.A. Habib, R. Ben-Mansour, S.A.M. Said, J.J. Al-Bagawi, and K.M. Al-Mansour, Correlations of flow maldistribution parameters in an air cooled heat exchanger, *Int. J. Numerical Meth. Fluids* (2007), 56 (2), 143-165.
- [13] P. Bhramara, V.D. Rao, K.V. Sharma and T.K.K. Reddy, CFD Analysis of Two Phase Flow in a Horizontal Pipe – Prediction of Pressure Drop, *World Academy of Science* (2008), *Engineering and Technology* 40.
- [14] M.A. Habib R. Ben-Mansour, S.A.M. Said, M.S. Al-Qahtani, J.J. Al-Bagawi, K.M. Al-Mansour, Evaluation of flow maldistribution in air-cooled heat exchangers, *Computers & Fluids* (2009), 38, 677–690.
- [15] S.D.S. Josedite, P.M.F. Fabiana, S. Ramdayal, R.F.N. Severino, G.B.L. Antonio, Non-Isothermal Separation Process of Two-Phase Mixture Water/Ultra-Viscous Heavy Oil by Hydrocyclone, *Advances in Chemical Engineering and Science* (2011), 1, 271-279.
- [16] X. Luo, W. Roetzel and U. Lüdersen, The Single-Blow Transient Technique Considering Longitudinal Core Conduction and Fluid Dispersion, *International Journal of Heat and Mass Transfer* (2011), 44 (1), 121-129, ISSN 0017-9310.
- [17] C.J. Meyer and D.G. Kröger, Plenum Chamber Flow Losses in Forced Draught Air-Cooled Heat Exchangers. *Applied Thermal Engineering* (1998), 18 (9-10), 875-893, ISSN 1359-4311.
- [18] A.A. Aganda, J.E.R. Coney, P.E. Farrant, C.G.W. Sheppard and T. Wongwuttanasatian, A Comparison of the Predicted and Experimental Heat Transfer Performance of a Finned Tube Evaporator. *Applied Thermal Engineering* (2000), 20 (6), 499-513, ISSN 1359-4311.

- [19] G. Hetsroni, D. Mewes, C. Enke, M. Gurevich, A. Mosyak, R. Rozenblit, Heat transfer to two-phase flow in inclined tubes, *International Journal of Multiphase Flow* (2003) 29, 173–194.
- [20] K. Mohammadi, M.R. Malayeri, Parametric study of gross flow maldistribution in a single-pass shell and tube heat exchanger in turbulent regime, *International Journal of Heat and Fluid Flow* 2013, 44, 14-27.
- [21] H. Jonas, N. James, W. Michael, The effect of inlet conditions on the air side hydraulic resistance and flow maldistribution in industrial air heaters. *International Journal of Heat and Fluid Flow* 2011, 32, 834-845.
- [22] G.F. Holman, G.L. Chires, T,R, Bott, *Process heat transfer*, CRC Press LLC, New York, (1994).