Applied Thermal Engineering 31 (2011) 3235–3242

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Characteristics of flow distribution in compact parallel flow heat exchangers, part II: Modified inlet header

Chi-Chuan Wang [a], Kai-Shing Yang [b], Jhong-Syuan Tsai [c], Ing Youn Chen [c,*]

[a] Deaprtment of Mechanical Engineering, National Chiao Tung University, Hsinchu 300, Taiwan
[b] Green Energy and Environment Research Labs, Industrial Technology Research Institute, Hsinchu 310, Taiwan
[c] Deaprtment of Mechanical Engineering, National Yunlin University of Science and Technology, 123 University Rd., Section 3, Douliu 640, Yunlin, Taiwan

ARTICLE INFO

Article history:
Received 23 February 2011
Accepted 2 June 2011
Available online 12 June 2011

Keywords:
Flow distribution
Modified header
Baffle plate
Baffle tube

ABSTRACT

This study presents the experimental results of liquid flow distribution in compact parallel flow heat exchanger through a rectangular and 5 modified inlet headers (i.e., 1 trapezoidal, one multi-step, 2 baffle plates and 1 baffle tubes header). The basic header has a rectangular shape with 9 × 9 mm cross section and 90 mm long header length having a 4 mm inlet tube for flow into the header and distributed to nine 3 mm parallel tubes with 400 mm length. A jet stream induced at the header inlet associated with vortexes affecting the flow distribution to the front tubes. The flow distribution in the header highly depends on the header shape and the total flow rate. Normally the first several tubes have the lowest flow ratios for the conventional headers and the flow distribution is significantly improved by lifting the jet stream using the modified header with baffle tube, followed by the baffle plate and multi-step header. The baffle tube yields the best flow distribution for it removed the vortex flow, and it is applicable for all the flow rates.

1. Introduction

Flow distribution from a header into parallel channels applicable to heat exchangers is frequently encountered in heat transfer devices, such as condensers, evaporators, and solar energy flat-plate collectors. However, the flow rates of single-phase distribution through the parallel channels are often not uniform. Occasionally, there is very low flow through some of them, and even reverse flow may occur which would reduce the heat exchanging performance. As a consequence, the issue of uniform flow distribution has recently received growing attentions for the heat exchanger design. The uneven distribution in parallel channels could be related to the stream velocity in the header (or manifold), size of the header, diameter of the parallel channels, location and size of inlet port to the header, flow direction, orientation of the channels and the headers. In addition, the flow mal-distribution effects have been generally associated with improper exchanger entrance configuration due to poor header design and imperfect passage-to-passage flow distribution [1]. Thus, for designing compact heat exchangers, it is very important to understand the flow distribution phenomena in the header and parallel channels. In part I of this study [2], the experimental results dealing with the effects of inlet flow condition, diameter of parallel tube, typical header size, area ratio, Z and U-type flow directions, as well as the gravity were presented and discussed. The numerical results also indicate that the jet flow generated at the header inlet with vortex flows circulated at the header inlet, as well as a small eddy flow formed at the inlet of the first tube. Both vortex flow and eddy flow would reduce the flow rate especially at the first several tubes. The flow ratio can be reduced more than 50% relative to the average flow ratio. This may lead to disaster to the thermal system. In view of the severe outcome, the objective of this investigation is to provide some simple and feasible designs experimentally and numerically that can provide significant relief of the mal-distribution.

2. Background

Kim et al. [3] numerically investigated the effect of outlet header shapes on the flow distribution with the same inlet velocity for three different header geometries (i.e., rectangular, triangular, and trapezoidal) with the Z-type flow direction. Their results indicated that the triangular shape provided the best distribution regardless the inlet velocity. They assumed a uniform velocity at the header inlet without considering the entrance effect. The flow distribution of trapezoidal shape is similar but is slightly worse than that of the

* Corresponding author. Tel.: +886 5 5342601; fax: +886 5 5312062.
E-mail address: cheniy@yuntech.edu.tw (I.Y. Chen).

1359-4311/$ – see front matter © 2011 Elsevier Ltd. All rights reserved.
doi:10.1016/j.applthermaleng.2011.06.003

Nomenclature

D	inner tube diameter (m)
f	Fanning friction factor
G_i	mass flux for ith tube, $G_i = Q_i\rho/A$ (kg/sm^2)
Q_i	volume flow rate for ith tube (L/min)
Q	total volume flow rate (L/min)
Greek symbols	
ρ	fluid density (kg/m^3)
β_i	flow ratio for ith tube
$\overline{\beta}$	average flow ratio for the total tubes
Φ	non-uniformity
ΔP	pressure drop (Pa, N/m^2)
Subscript	
f	friction
g	gravity
i	ith tube

triangular. For the rectangular header, the last tube has the highest flow rate, causing the flow mal-distribution. When the rectangular header was replaced with a triangular header, the flow distribution in the header was significantly improved. The reason is that the pressure along the flow direction in the outlet header is significantly reduced for the triangular shape due to the reduction of the cross section area. Thus, the distribution of the pressure difference across the tubes is more uniform for obtaining the better flow distribution.

Jiao et al. [4] experimentally investigated the header configuration for flow mal-distribution in plate-fin heat exchanger with central flow into the header. They installed a second header (B or C) after the first header to study the flow velocity distribution. The header configuration B has five holes in the connection part between the first and second headers while the header configuration C has seven holes. The flow velocity distribution of the configuration C gives the most uniform result among the cases considered in their paper. The flow distribution was effectively improved by the modified header configuration.

Luo et al. [5] experimentally examined the effects of constructed distributors or headers, and built on a binary pattern of pores, for flow equal distribution in a multi-channel heat exchanger. Thermal performance and pressure drop were determined with different assembly configurations of constructed headers, conventional pyramid distributors and a mini cross flow heat exchanger (MCHE). Experimental results showed that the integration of constructed inlet and outlet headers could improve the fluid flow distribution and consequently lead to a better thermal performance of the MCHE, but higher pressure drops were also encountered.

Wen and Li [6] utilized a baffle plate with small size holes installed in the inlet header for central flow direction to optimize the header configuration. The small holes are spotted in the baffle plate according to the velocity distribution and the punched ratio is gradually increasing in symmetry from the axial line to the boundary. By changing the flow resistance through various sizes of holes, the fluid flow is distributed uniformly before it reaches the header outlet and the uniform distribution is achieved.

Recently, Tong et al. [7] also numerically studied the flow distribution for three different header geometries assuming uniform velocity at header inlet (linear taper, concave-down quarter ellipse, and concave-up quarter ellipse) for the Z-type flow direction. These modified headers would reduce the cross sectional area along the flow direction in the header. The results show that increasing the taper angle is more beneficial as far as better flow distribution is concerned. The taper angle (relative to the horizontal) was varied from 0° (no taper) to approximately 10°. Also, the combination of concave-down and concave-up tapering for the inlet and outlet headers is advantageous for flow distribution, but is inferior to the simple concave-down tapering. Since the fabrication of a linear taper is comparatively easier to fabricate than that needed for non-linear tapering, they concluded that the use of linear tapering would be more cost effective.

The flow non-uniformity in heat exchangers is generally caused by the poor design of the flow inlet configuration [1]. More recently, Shi et al. [8] had placed a deflector in a proper location into the inlet header of the parallel flow heat exchanger for considering the low cost and easy installation. The CFD simulation is to place a deflector in different locations, then compare the flow distribution results and find the proper location. In their study, the 1st location is located at 5 mm deep inside the inlet channel, and the 2nd–5th location is located at from 10 mm deep to 25 mm deep and the interval is 5 mm. Also, the no deflector situation was also calculated as the benchmark. Air and hot water were utilized as working fluids for tests. The numerical results showed that the flow mal-distribution is very serious for no deflector. Also, comparing the numerical results of pressure drop and flow distribution for the deflector installing at 5 different locations in the inlet header, a deflector adding at the 3rd location was selected for heat transfer testing and it is called the “new type” for comparing the “old type” header without deflector. The experimental results indicated that the heat transfer performance of heater core could be improved from 1.03% to 3.98% for various combinations of air and water flow rates by adding the flow deflector into the 3rd location in the inlet header. However, the detail size of the flow deflector was not given in their paper.

From the above review, there are different methods to modify the inlet header. The first is to reduce the cross section area along the flow path of the inlet header, e.g., linear taper or multi-step header. The second is to install a baffle plate with different size holes on the plate. The third is to add a deflector at the proper location of the inlet header. The purpose of the above methods is to obtain the uniform flow distribution for each parallel tube as the flow resistance increases. Also, the concept of baffle tube had not been utilized in the inlet header of the parallel tube heat exchanger. Therefore, the fourth one is to test the baffle tube with different hole sizes for obtaining the uniform flow distribution. The objective of this study is to propose novel designs to improve the flow distribution. The novel designs, featuring trapezoidal taper and multi-step blocker, as well as baffle plate and baffle tube, are tested through experimental and numerical verifications for the superior flow distribution.

3. Experimental apparatus

3.1. Test rig

The test rig with the same schematic diagram as described in part I of this study [2] is shown in Fig. 1. Water is heated by a thermostat where water is maintained at 25 °C. A very accurate Yokogawa magnetic flow meter (AXF005G) is installed at the downstream of the gear pump for measuring the water flow rate. The accuracy of water flow meter is within ±0.045 L/min of the test span. Leaving the flow meter, the water temperature is measured by a resistance temperature device (Pt100 Ω) having a calibrated accuracy of 0.1 K. The pressure entering the test section is measured by a YOKOGAWA EJX pressure transducer with an accuracy of 0.025%. Also, a YOKOGAWA EJ110 differential pressure transducer having an adjustable span of 1300–13000 Pa installed across the

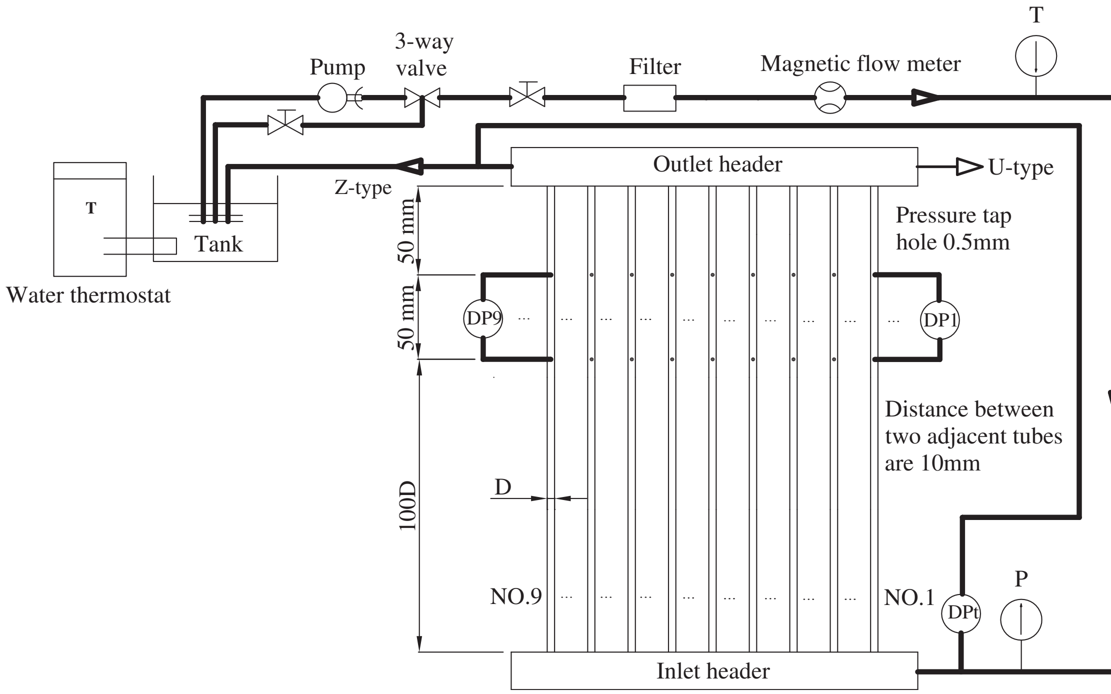


Fig. 1. Schematic diagram of the test apparatus.

inlet of the upstream header and outlet of the downstream header with a resolution of 0.3% of the measurements.

The test section includes an inlet and outlet headers along with 9 parallel tubes as shown in Fig. 1. The foremost tube to the inlet location of header is termed 1st tube, and the aftermost one is termed 9th tube. The pressure drop for each tube at the location with 50 mm from the outlet header is measured by a YOKOGAWA EJ110 differential pressure transducer. Also, a 100 diameter calming tube length from the inlet header is used to ensure fully developed flow. The pressure taps are vertically drilled with a diameter of 0.5 mm.

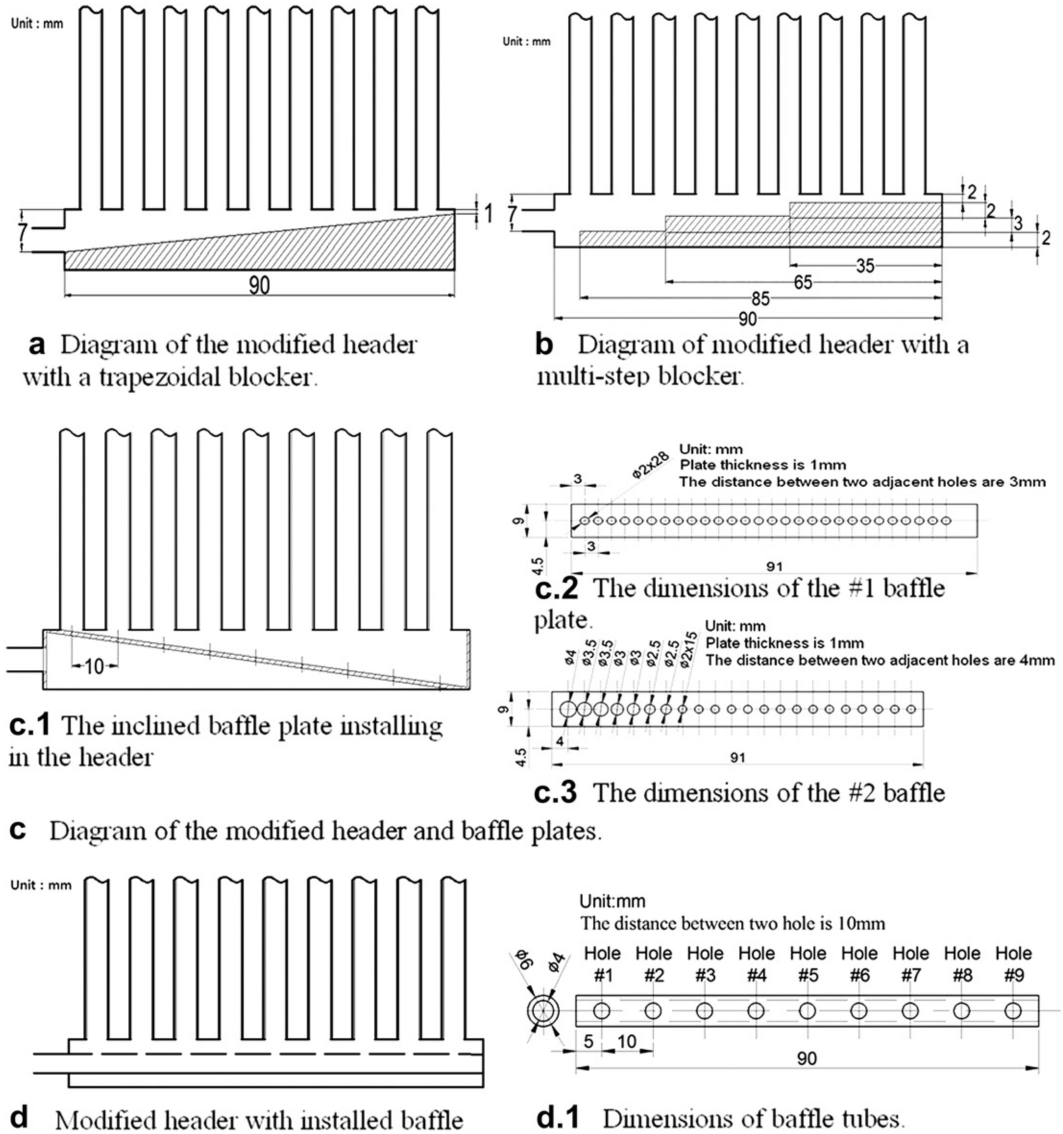
The measurement of pressure drop in each tube is used to calculate the flow rates among the tubes. The total pressure drop includes gravitational drop ($\Delta P_g = \rho gh$) and frictional drop (ΔP_f), i.e., $\Delta P_T = \Delta P_f + \Delta P_g$ where $\Delta P_f = 2f(\Delta L/D)(G^2/\rho)$ The deviation of the calculated volume flow rate (Q_i) from the pressure drop is estimated to be less than 1.34% of the actual flow rate. The derived uncertainties of the flow ratio β and non-uniformity Φ are 2% and 3.88%, respectively.

3.2. The modified headers

Normally, the diameter of the inlet tube to the header is much less than the cross section area of the header. Also, for most applications, the distance from the header inlet to the first tube is very short for space saving, and hence the entrance effect is very significant. As described in part I [2], jet flow was induced due to the sudden expansion from the 4 mm entering tube into the 9 mm × 9 mm cross section area of the header. The smaller inlet tube directly connected to the header is normally seen in the heat exchanger design. With the inlet jet stream, the vortex flow forms at the entrance. Thus, the flow rates into the first several tubes are severely reduced, and the flow stream is forced toward to the rear tubes of the header. Also, the parallel tubes with smaller diameter have a higher flow resistance, and hence the non-uniformity of 2 mm tube is much less than the 3 mm tube as reported in part I of this study [2], however, the total pressure drop across the heat exchanger with smaller parallel tubes would be much greater than that of larger parallel tubes. Therefore, the modified header is based on 9 mm × 9 mm cross section and 90 mm header length which has a much higher non-uniformity than that of the 120 mm header length [2]. The test section consists of a modified inlet header, 9 parallel tubes with 3 mm diameter and 400 mm tube length, a pitch of 10 mm between tubes, as well as a distance of 3.5 mm from the header inlet to the 1st tube. The inlet flow tube to the header has an inner diameter of 4 mm. The tests were conducted for Z-type and U-type flow directions in vertical-up flow orientation at various flow conditions.

The typical rectangular header with 9 mm × 9 mm cross sectional area and 90 mm header length is also tested as a benchmark. There are 5 modified headers made for experiments with 1 trapezoidal taper and 1 multi-step blocker, 2 baffle plates and 1 baffle tube installed in the basic header to increase the flow resistance for reducing the entrance effect and for improving the flow distribution. The fabrication of experimental baffle tube is based on the best numerical simulations pertaining to various baffle tubes with different hole sizes. The results of the modified headers are compared to the basic header. For a typical header with a uniform cross sectional area, the inlet velocity would gradually decrease along the header length. The decreasing velocity would increase the static pressure, resulting in a higher flow rate at the downstream. To offset this trend, it is reasonable to modify the header such that its cross sectional area is decreased along the flow path in the header.

The types of modified designs in this study consist of (1) trapezoidal taper; (2) multi-step blocker; (3) baffle plate; and (4) baffle tube. Fig. 2(a) is the modified header with the trapezoidal taper having a length of 90 mm with height being 1 mm and 8 mm at both ends. Fig. 2(b) denotes the modified header with multi-step blocker, and the installed blocker has 3 stepwise increases along the flow path in the header. In addition, 2 baffle plates are made and their dimensions as shown in Fig. 2(c). Both baffle plates, having a thickness of 1 mm, have folded tips at both ends for fixing the baffle plate at an inclined position in the header. The #1 baffle plate has 28 holes with the same 2 mm diameter and 3 mm pitch



Baffle tube	Hole #1 (mm)	Hole #2 (mm)	Holes #3~#9 (mm)
#1	4	3.7	3.2
#2	4	3.5	3
#3	3.7	3.2	2.8
#4	3.5	3	2
#5	3	2.2	1.5
#6	2.8	2	1.2
#7	3.8	2.2	1.5

e The hole diameters for 7 baffle tubes

Fig. 2. Diagrams of the modified headers.

between holes. The #2 baffle plate has the largest hole (4 mm) near the inlet, then the diameter of the followed holes is decreased to the 7th hole (2.5 mm), and the other 15 holes were kept with a constant diameter of 2 mm.

The schematic of baffle tube and the dimensions of 7 baffle tubes having different hole sizes are given in Fig. 2(d). The inner and outer diameters of the baffle tubes are 4 mm and 6 mm, respectively. Its length is 94 mm with 4 mm screwed length at the

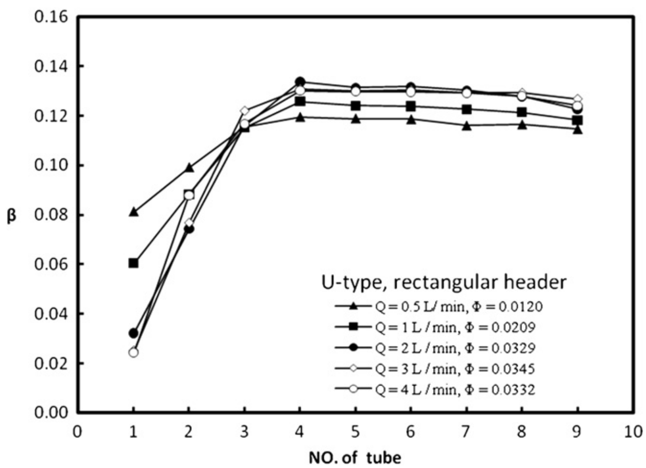


Fig. 3. Flow ratio of the rectangular header for U-type and Z-type flows.

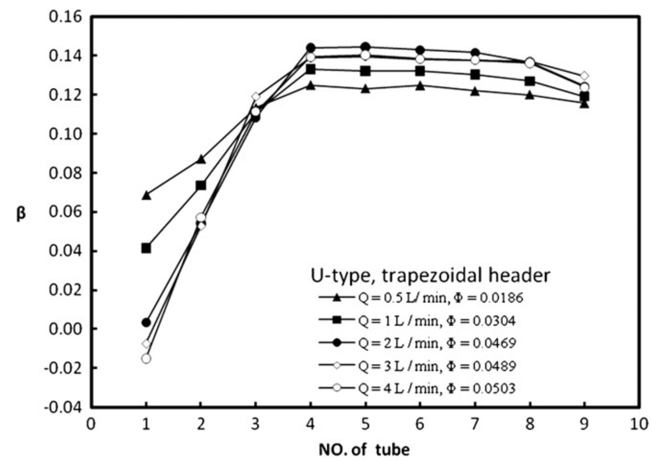


Fig. 4. Flow ratio of the modified header with trapezoidal blocker for U-type and Z-type flows.

front for installing at the center of the header inlet. All the baffle tubes have evenly distributed 9 holes with 10 mm pitch between the holes. However, the tube diameter is in consecutive decreasing from the 1st hole to the 2nd hole, and the diameters of 3–9 holes are kept at the same smallest diameter. The baffle tubes have various diameter sizes with the largest one being placed at the 1st hole, followed by the second larger at the 2nd hole and the other holes were kept at the same smaller diameter for the baffle tubes. This ideal is originated from the measured flow ratio of the rectangular header as shown in Fig. 3 which clearly indicates the lowest flow ratio occurring in the 1st tube and increases consecutively to a plateau around 0.12–0.14 and does not vary much subject to the change of a given total flow rate. The larger diameters of the 1st and the 2nd holes in the baffle tube implicate a smaller flow resistance and allow more flow entering into the 1st and the 2nd tubes near the header inlet, thereby improving the flow distribution. As shown in Fig. 2(e), a total of 7 baffle tubes were put into numerical simulations to screen the appropriate one for experimental verification. The experimental results obtained from the selected baffle tube are compared to its numerical solution to validate the effect of the optimization.

As seen, significant departure of uniformity is encountered for U-type or Z-type flow arrangement in typical inlet headers. In this regard, it is the purpose of this study is to propose some novel modified headers to relief the mal-distribution associated with the conventional Z-type and U-type arrangements subject to vertical-up orientation with various total flow rates (Q) of 0.5, 1, 2, 3 and 4 L/min. The U-type flow still has better flow distribution than Z-type flow which was also previously reported in the first part of the present study [2]. Therefore, only the results of U-type flow distribution are given in the following section for discussion.

4. Results and discussion

For evaluating the improvements of the flow distribution from the modified headers, the flow ratio of the typical rectangular header at different flow rates for U-type flow are given in Fig. 3. The flow ratio of the first tube is severely lower than the other tubes, followed by the 2nd tube with still much lower flow ratio than the average. The flow ratios of the 3rd to 9th tubes reveal a much less variation with a flow ratio ranging from 0.12 to 0.14. On the other hand, the flow ratios at the first several tubes become even ill-distributed when the total flow rate is further increased. This is because the entrance effect with a high speed jet flow is induced in the header inlet, yet the jet flow phenomenon becomes more pronounced with the increasing total flow rate. The non-uniformity is increased with the rise of the total flow rate and it reached a maximum at a flow rate of 3 L/min.

The results of flow ratio for the trapezoidal header at different flow rates for the U-type flow are given in Fig. 4. The non-uniformities (Φ) of trapezoidal header for U-type flow are 0.0186, 0.0304, 0.0469, 0.0489 and 0.0503 for U-type flow at flow rates of 0.5, 1, 2, 3 and 4 L/min, respectively. The Φ values are higher than the typical header with corresponding values of 0.012, 0.0209, 0.0329, 0.0345 and 0.0332 as shown in Fig. 3. Notice that a reversed flow even appears in the 1st tube when the total flow rate reaches 3 or 4 L/min. As a consequence its flow ratio (β) becomes negative. The results are quite surprising for the original idea with trapezoidal design is to direct more flow into the first several tubes by gradually decreasing the effective cross sectional area alongside the header. The flow ratio of this design is greatly increased till the 4th tube. This is because the violent interactions of the jet flow along the trapezoidal surface at the entrance that creates an intensified vortex, lowering the static pressure and leading to a very small pressure gradient or even reversed pressure gradient amid the intake conduit and exhaust conduit. And this phenomenon may become more pronounced with rising total flow rate. Therefore the reversed flow occurs when the total flow is increased over 3 L/min. Moreover, as the flow entering the trapezoidal header, the flow rate may still accelerate (or de-accelerate more moderately than that of rectangular design) despite flow is continuously branching along the header, implying a less pressure gradient difference and a more severe mal-distribution occurs at the first several tubes. Tong et al. [7] numerically studied the flow distribution for trapezoidal header with linear taper. Their results show that increases of the taper angle have a favorable effect on the flow distribution. However, in their calculation a uniform velocity profile was assumed at the inlet of the manifold system, the effects of jet stream and vortex flow were not taken into account which may lead to some unrealistic results.

The flow ratios of the multi-step header for U-type flow are shown in Fig. 5. With the first step being parallel to the inlet, the intensity of the inlet jet stream may be relaxed due to the drag/friction contribution caused by the step surface. Hence, the effective pressure gradient of the first several tubes amid intake conduit and exhaust conduit is increased, resulting in a better flow distribution accordingly. As seen in the figure, the flow ratios in the 1st and 2nd tubes are obviously higher than those in the rectangular

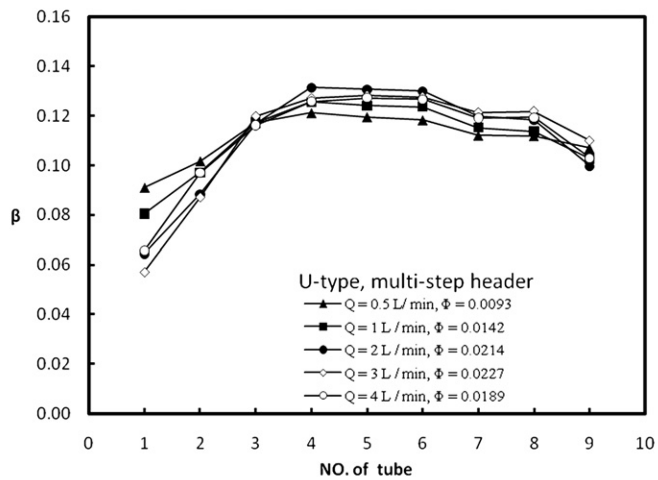


Fig. 5. Flow ratio of the modified header with multi-step blocker for U-type and Z-type flows.

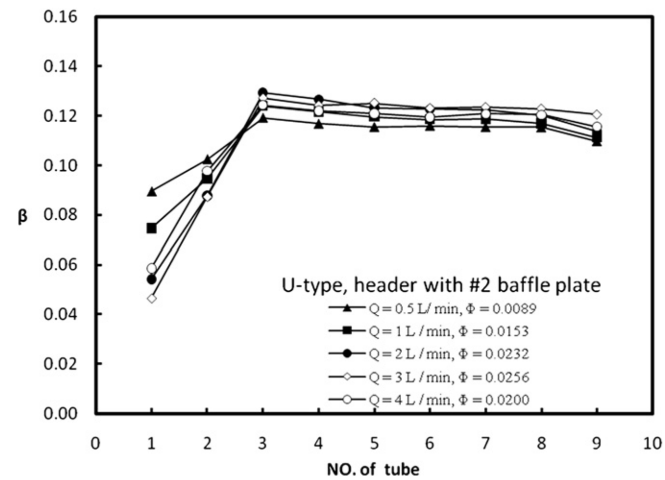


Fig. 7. Flow ratio of the modified header with #2 baffle plate for U-type and Z-type flows.

and trapezoidal headers. For example, the β values of the 1st tube for the multi-step header are 0.065 and 0.057 for $Q = 4$ and 3 L/min, respectively; which are higher than the corresponding β values of the rectangular and trapezoidal headers as shown in Figs. 3 and 4, respectively.

Though the flow ratio at the first several tubes in the multi-step header is improved by the presence of multi-step design, it is still much less than the average ratio of 0.11. Therefore, the concept of inclined baffle plate with multiple small holes installing in the header is considered for increasing the flow resistance and for reducing jet flow strength to improve the flow distribution. The flow ratios of #1 baffle plate for U-type flow are shown in Fig. 6. The resultant flow ratios of the 3rd to 9th tubes are nearly uniform in the range of 0.11–0.12 for the five tested total flow rates. Though the flow ratios for #1 baffle plate at the 1st tube with $Q = 4$ and 3 L/min are 0.085 and 0.065which are still higher than the corresponding values of rectangular, trapezoidal and multi-step headers. This improvement is due to the higher flow resistance by the small holes in the baffle plate, and it partially reliefs the jet flow effect.

Even though the flow distribution has been improved by the #1 baffle plate, the flow ratios of the 1st and 2nd tubes are still lower than the other tubes. The #1 baffle plate features a constant diameter and a total of 28 holes with constant pitch. With aforementioned gradually increase of flow rate in the first several tubes, the #2 baffle plate has the largest hole (4 mm) near the inlet, then the diameter of the subsequent holes is decreased to 2 mm for the last 15 holes. The flow ratios of #2 baffle plate for U-type flow are shown in Fig. 7, it appears that the results are worse than that of #1 baffle plate for lower flow ratio occurring at the 1st and 2nd tubes. The larger diameter of the 1st hole provides less restriction to the entering velocity. Therefore, the corresponding flow ratios in the first several tubes for the #2 baffle plate are lower than that of #1 baffle plate.

For considering the reliable installation using the foregoing baffle plate, it may not be cost-effective for its comparatively higher manufacturing cost for installing, assembling, and immobilization during operation. In the meantime, the flow ratios of the 1st and 2nd parallel tubes still suffer from mal-distribution. In this sense, the concept of baffle tube shown in Fig. 2(d) may be easier to install. To select an optimized baffle tube design, a prior numerical simulation using EFD.lab software is first conducted for the U-type flow with a total volume flow rate, $Q = 2$ L/min with 7 different baffle tube designs. The hole diameter for 7 baffle tubes are shown in

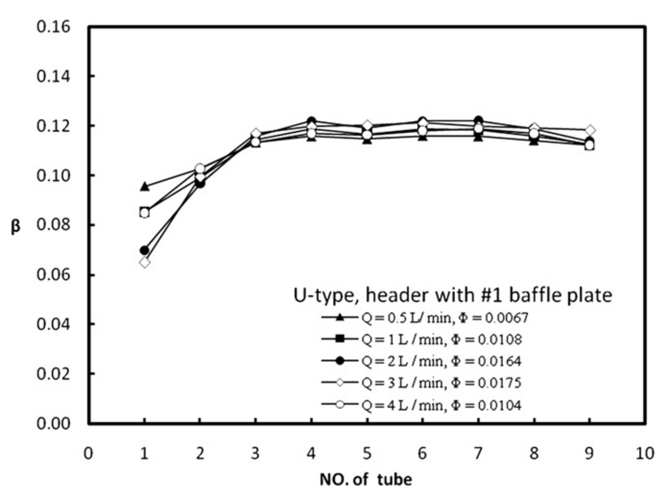


Fig. 6. Flow ratio of the modified header with #1 baffle plate for U-type and Z-type flows.

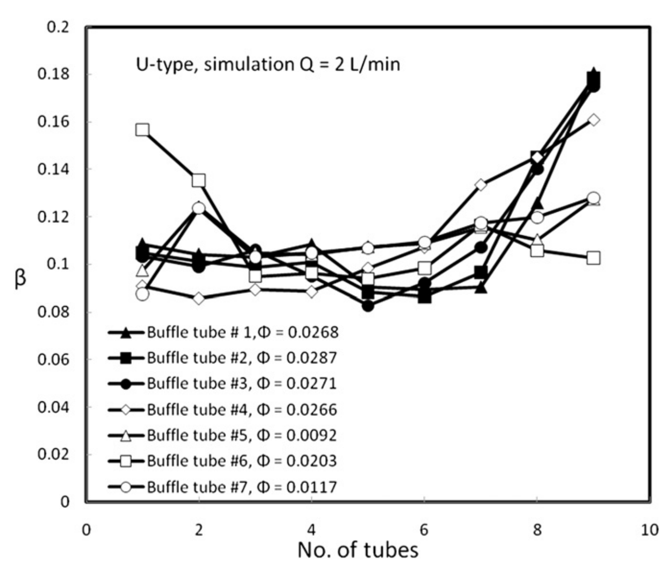


Fig. 8. The simulated flow ratios of the modified header for 7 baffle tubes.

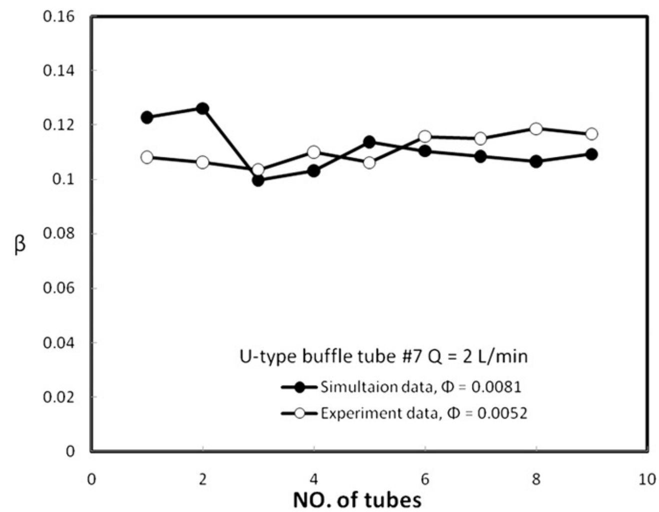


Fig. 9. Comparison of experimental and numerical flow ratios of the modified header with #7 baffle tube.

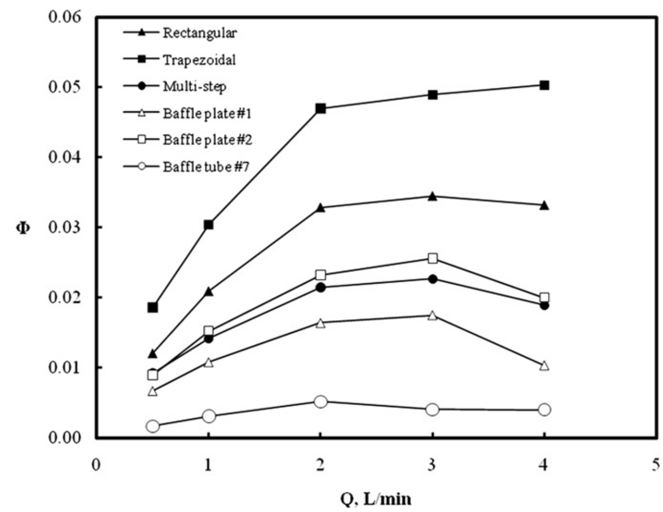


Fig. 11. Volume flow rate vs. Φ for U-type and Z-type flows with the tested headers.

Fig. 2(e). The generated hybrid Mesh 5 contains 195,993 cells for the simulation of the parallel flow heat exchanger.

Fig. 8 shows the simulated flow ratios and the non-uniformity of the modified header for the 7 baffle tubes. The improvement of flow ratio (β) on the 1st and 2nd tubes has been clearly seen with the β values varying from 0.09 to 0.11 for the 1st tube, and 0.085–0.12 in the 2nd tube. Except for the #6 baffle tube whose β is up to 0.16 and 0.14 for the 1st and 2nd tubes, the β values of the front 4 tubes for the #1, #2 and #3 baffle tubes are close to the average ratio, 0.11, or slightly less. However, the β values for the 8th and 9th tubes are significantly increased, and it exceeds 0.18 in the 9th tube. The reason is that the diameters of the rear holes of #1–#3 baffle tubes are only slightly smaller that of the front holes. The #6 baffle tube has higher β values in the 1st and 2nd tubes (0.16 and 0.135, respectively), and the values are about 0.1 or slightly less in the rear tubes because the diameter for the rear holes is too small (1.2 mm), hence forcing more flow into the 1th and 2nd tubes. The #4 baffle tube has β values about 0.9 in the first 4 tubes and then gradually increased to 0.16 for the 9th tube for the 2 mm diameter of the 3rd to 9th holes which are greater than the optimized diameter. The #5 and #7 baffle tubes have the best flow distribution with the non-uniformities of 0.0092 and 0.0117. The #5 and #7 baffle tubes have the same hole diameters starting from #2 to #9 holes, but the 1st hole diameter of #7 baffle tube is 3.8 mm which is greater than that of the #5 baffle tube (3 mm). Though the non-uniformity of #5 baffle tube is slightly less than the #7 baffle tube, the #7 baffle tube presents a lower pressure drop that is quite beneficial for a long term operation. Thus, the #7 baffle tube is selected from the numerical results for fabrication and verification. The experimental results and the simulations with U-type flow at 2 L/min are given in Fig. 9. As seen in the figure, the measured β values are nearly in line with the simulations.

The #7 baffle tube has 3.8 and 2.2 mm diameters in the 1st and 2nd hole, respectively, and the other holes are kept at 1.5 mm with higher flow resistance for guiding more flow into the 1st and 2nd tubes. The experimental results of the flow ratios for #7 baffle tube with U-type flow are respectively given in Fig. 10. As clearly seen in the previous figures showing the mal-distribution problem, especially those in the first several tubes had been significantly improved and the flow ratio of the #7 baffle tube with U-type flow are quite close to uniform distribution. The flow ratio of the 1st tube

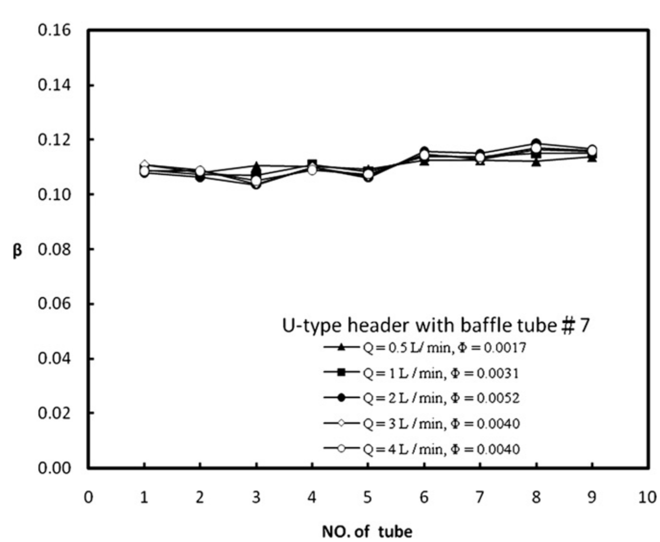


Fig. 10. Flow ratio of the modified header with #7 baffle tube for U-type and Z-type flows.

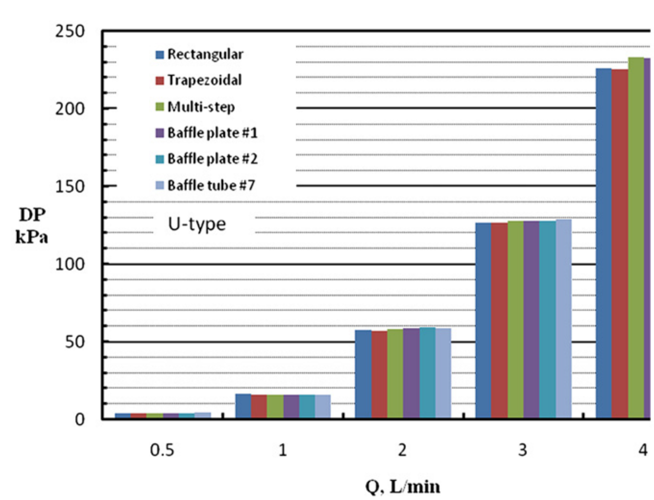


Fig. 12. Comparison of the total pressure drop across inlet and outlet headers.

is near 0.11, and the flow ratios for the 9 parallel tubes pertaining to various flow rates varied in a very narrow range of 0.105–0.12. The larger diameter of the 1st and 2nd holes for the #7 baffle tube suggests a comparatively small flow resistance, and hence the flow ratios of the 1st and 2nd tubes are comparable with other tubes as shown in Fig. 10.

For comparing the performance of flow distribution of the 6 tested headers, the results of non-uniformity (Φ) verse total flow rate are given in Fig. 11 for U-type arrangement. The value of Φ is increased with the rise of the total flow rate from 0.5 to 2 L/min. No consistent trend is observed for Φ with flow rate being 3–4 L/min, but the change of Φ for flow rate from 0.5 to 2 L/min is much greater than that for 2–4 L/min. The trapezoidal header has the highest non-uniformity, followed by the conventional header; both suffered from the entrance effect of jet stream at the inlet of the header. The headers with inclined #1 and #2 baffle plates reveal smaller Φ values than that of the rectangular header, but are comparable with the multiple step headers. The #1 baffle plate is marginally superior to the #2 plate for U-type flow with all flow conditions. The header with a baffle plate still suffers from jet flow into the header with entrance effect affecting the flow distribution as observed from numerical simulation. For the header with baffle tube, the flow is initially into the baffle tube. The flow is then distributed through the holes and passed to the parallel tubes, thereby lifting the entrance effect. As a result, the baffle tube has the lowest non-uniformity. The #7 baffle tube has the best uniform flow distribution for its non-uniformity being much lower than the other modifications.

Fig. 12 represents the measured total pressure drop verse the total flow rate with the present 6 tested headers for U-type flow. The results show that the total pressure drop increased with the total flow rate. For the same flow rate, the difference among the tested headers is quite small. At the highest flow rate of 4 L/min, the largest difference of pressure drop amid the tested headers is only 7 kPa (<3%), suggesting the increased pumping power for the modified header is virtually negligible.

5. Conclusion

This study experimentally investigates the liquid flow distribution in compact parallel flow heat exchanger through a rectangular and 5 modified inlet headers (i.e., 1 trapezoidal, one multi-step, 2 baffle plates and 1 baffle tubes header). The flow distribution highly depends on the header shape and the total flow rate. The higher flow rate is associated with a higher non-uniformity. Among the headers being tested, the proposed novel baffle shows substantial improvements of flow non-uniformity, and are major finding are given in the following:

1. Normally the 1st tube always has the least flow rate and followed by the 2nd tube due to the entrance effect. Most of the modifications except the baffle tube are unable to remove the mal-distribution in the first several tubes.
2. The multi-step header has a much higher flow ratio in the 1st and 2nd tubes than the trapezoidal and the rectangular headers because some flow was forced by the first and second steps toward the front tubes.
3. The flow distributions of the #1 and #2 baffle plates had been marginally improved comparing to the multi-step header. However, their flow ratios of the 1st and 2nd tubes are still much less than the other rear tubes due to the existing vortex flow near the header inlet.
4. The flow ratios to the 1st tube for the modified header with the #7 baffle tube have been significantly improved and is very close to the average flow ratio of 0.111. The numerical result shows that no vortex flow exists at the inlet header of the novel baffle tube design. Also, the modified header #7 with baffle tube flow shows the best flow distribution than the others. The proposed modified headers show only insignificant increase in pressure drop across the inlet and outlet headers than the typical rectangular header even at higher total flow rate.

Acknowledgements

The grant from National Science Committee (NSC 98-2221-E-224-049) of Taiwan is appreciated for supporting this study. Also, the authors are indebted to financial support from the Bureau of Energy of the Ministry of Economic Affairs, Taiwan.

References

[1] W.M. Kays, A.L. London, Compact Heat Exchangers, third ed. McGraw-Hill, New York, 1984.
[2] C.C. Wang, K.S. Yang, J.S. Tsai, I.Y. Chen, Liquid flow distribution characteristics in compact parallel flow heat exchangers, part I: typical inlet header, Applied Thermal Engineering 31 (2011) 3226–3234.
[3] S. Kim, E. Choi, Y.I. Cho, The effect of header shapes on the flow distribution in a manifold for electronic packaging applications, International Communications in Heat and Mass Transfer 22 (1995) 329–341.
[4] A. Jiao, R. Zhang, S. Jeong, Experimental investigation of header configuration on flow mal-distribution in plate-fin heat exchanger, Applied Thermal Engineering 23 (2003) 1235–1246.
[5] L. Luo, Z. Fan, H.L. Gall, X. Zhoub, W. Yuan, Experimental study of constructal distributor for flow equidistribution in a mini cross flow heat exchanger (MCHE), Chemical Engineering and Processing 47 (2008) 229–236.
[6] J. Wen, Y. Li, Study of flow distribution and its improvement on the header of plate-fin heat exchanger, Cryogenics 44 (2004) 823–831.
[7] J.C.K. Tong, E.M. Sparrow, J.P. Abraham, Geometric strategies for attainment of identical outflows through all of the exit ports of a distribution manifold in a manifold system, Applied Thermal Engineering 29 (2009) 3552–3560.
[8] J.Y. Shi, X.H. Qu, Z.G. Qi, J.P. Chen, Effect of inlet manifold structure on the performance of the heater core in the automobile air-conditioning systems, Applied Thermal Engineering 30 (2010) 1016–1021.