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Experimental Study on Port to Channel Flow Distribution of Plate Heat Exchangers

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ABSTRACT

In the present study, experiments have been conducted to analyze the flow and pressure distribution in a plate heat exchanger. Unlike previous studies, here experiment on local port pressure distribution has been carried out in a commercial plate heat exchanger rather than an idealized manifold. Flow rate in each channel and channel pressure drops are evaluated by measuring the pressure inside the inlet and exit ports at different locations for different port dimensions. In these experiments the measurement of pressure is done without disturbing the fluid flow inside the port. This technique also offers the option of manipulating port size and geometry without changing the plate characteristics. Direct experimental measurement also provides the scope for eliminating other effects such as gasket and end losses or improper wetting of the channels from the port to channel flow maldistribution effect. Measurement carried out indicates the existence of non-uniform flow distribution which increases with flow rate and decreases with port diameter. The results clearly indicate that it is important to take the flow maldistribution into account for the better design of plate heat exchangers..

INTRODUCTION

Plate heat exchangers (PHEs) are widely used in the power and process industries due to its multi-fold advantages. Initially PHE were designed for the dairy, brewery and food processing industries where cleaning and maintenance is of prime importance. Later, it has been found that PHEs are having advantages like high heat transfer coefficient, compactness, flexibility, and less fouling etc. In recent years the working range of temperature and pressure has been enhanced due to the advances in material technology by using new temperature and pressure resistant materials for gaskets. As a result this type of exchangers is also used in power and chemical process industries today. Initially, modeling thermal performance of plate heat exchanger was carried out, based on the assumption of equal flow rate in all the channels [1, 2], which is an ideal case of no flow maldistribution. In reality the flow is distributed non-uniformly to channels affecting both thermal and hydraulic performance of the heat exchanger. Therefore, for better design, there is a need for a good knowledge of flow distribution and the effect of this distribution on the thermal and hydraulic performance.

In the area of flow distribution in manifold systems, the analytical model developed by Bajura [3] explained about the flow and pressure distribution inside the different manifold designs having different area ratios and flow resistances. and derived closed form equations from flow channeling and unification concept. In the recent past, the literature shows many numerical and analytical studies to find out pressure drop and flow distribution along the inlet and exit ports, which in turn are useful to predict the flow and thermal behavior of the exchanger. Effect of unequal flow distribution in parallel and reverse flow manifold systems was analyzed by Datta and Majumdar [4], and expressed the distribution in the channels in the form of closed form equation using the general flow channeling and unification concept by Bajura and Jones [5]. The PHEs can be mainly classified into two categories, U-type and Z-type configurations based on their flow arrangement. Flow and pressure distribution are different for these two kinds of heat exchangers. An analytical study on flow distribution and pressure drop in PHEs for both U-type and Z-type presented by Bassiouny and Martin [6, 7], gives velocity and pressure distribution in both intake and exit conduits. In their analysis, a general characteristic parameter m^2, which determines the flow maldistribution, has been derived using the mass and momentum balance formulations. These results are compared with the experiments on dividing flow of water in a manifold system, but not validated with the actual flow behavior in a real PHE. An experimental work by Rao and Das [8] presented the influence of flow distribution on the pressure drop across a PHE. Here the maldistribution was predicted from overall pressure measurement but not confirmed by local measurement inside the port.

From the above discussion it is clear that the experimental data is not available for measurement of flow maldistribution of plate heat exchangers, since in-port measurement is difficult and intrusive. However some experimental studies on flow distribution in manifolds with simple parallel flow channels are usually used for supporting the analysis of flow distribution of plate heat exchangers. Hence, there is a need to investigate the actual flow distribution in the channels and pressure distribution along the ports of the non-simplified plate heat exchanger geometry by conducting proper experiments. This is the main motivation for the present work in carrying out the present experiments to determine the real fluid behavior inside the ports. This will also be useful to analyze the thermal performance. Which was shown by an analytical study made by Rao et al. [9]. It is important to mention here that detail flow distribution data is available with PHE designers like HTFS and manufacturer like Alfa Laval but their data are proprietary in nature and not made public due to commercial reasons.

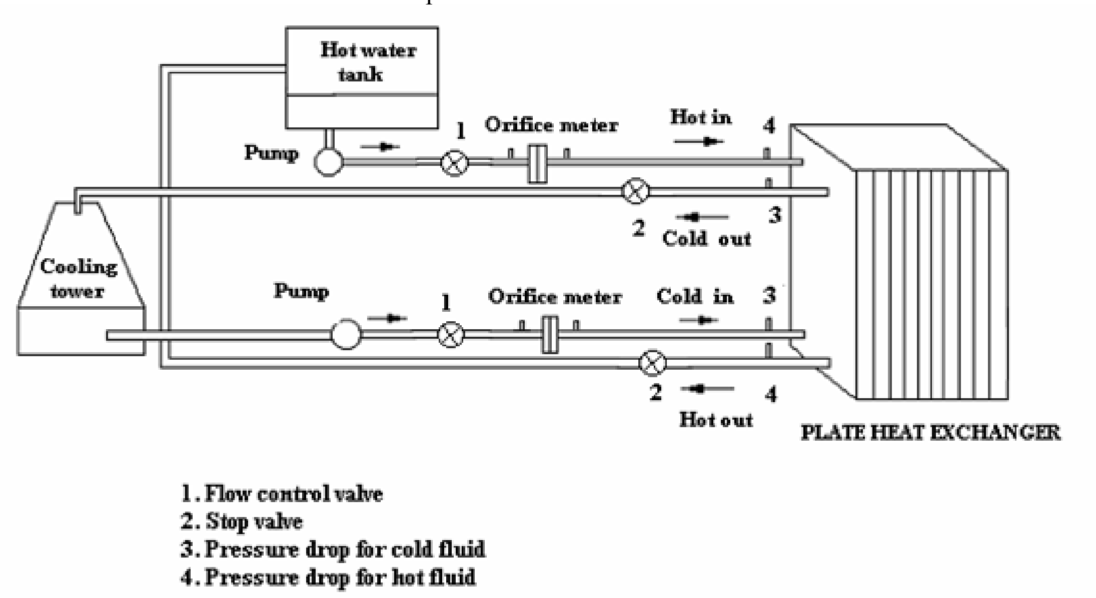

Fig 1. Schematic view of the experimental setup

EXPERIMENTAL SETUP

The schematic diagram of the experimental setup shown in Fig. 1. In this study double distilled water is used as the working fluid, stored in a storage tank and circulated with the help of pumps of 3HP capacity. The control valves are provided in the pipelines to control the flow rate. Orifice meters are provided to measure the flow rates. Pressure taps are provided at the inlet and exit lines to measure the total pressure drop across the heat exchanger. U-tube mercury manometers are used to measure the pressure drop across the heat exchanger. Calibration of the orifice plate has been done as per ASME standards and it is found that the average coefficient of discharge is 0.5942. The plate heat exchanger consists of 26 plates, which forms 25 number of cold channels used in this experiments, and all the plates are closely packed with the help of two thick cast iron end plates by tightening bolts. Heat exchanger plates are made of stainless steel, having corrugated surface, and its geometrical features are shown in the Fig. 2. For conducting these experiments, mandrels for inlet and exit ports are designed specially to change the diameter of the port, as shown in Fig.3. It must be noted here that the mandrel is designed in such a way that its inner diameter acts as the port diameter.

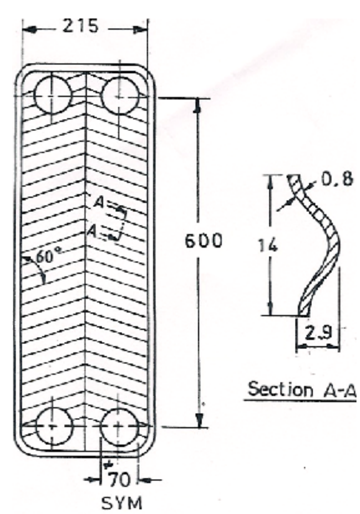

Fig 2. Geometrical features of the plate

Holes are drilled through the mandrel to measure pressure at different axial locations inside the port. The location of each hole is chosen in such a way that it is possible to measure the pressure distribution over the entire plate stack length. Pressure at each location can be measured by connecting the corresponding hole to one limb of U-tube manometer keeping the other limb open to atmosphere. This kind of mandrel design enables the pressure measurement without disturbing the fluid flow inside the port. Each mandrel is provided with five holes to measure the pressure at five different equally spaced distances along the port. Two different mandrels of sizes 34mm, and 24mm as port diameters are used to study the effect of port diameter on flow distribution. This enables to study the port diameter effect without changing the geometrical features of the PHE.

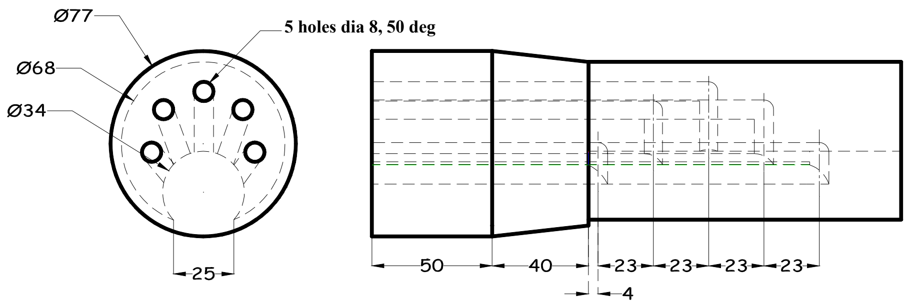

Fig 3. Schematic view of the mandrel and locations of pressure measurement inside the port

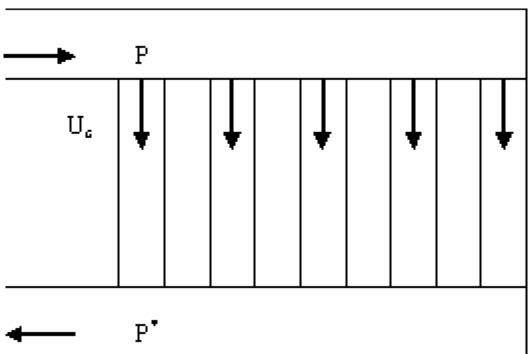

Fig 4. U- type configuration

EXPERIMENTAL PROCEDURE AND DATA REDUCTION

The two mandrels, one in the inlet port and the other in the outlet port were inserted. In this study, cold experiments were conducted to analyze the flow distribution along the channel. Water was circulated through the heat exchanger with the help of a pump and water flow rate adjusted by a control valve. The experiments were conducted with both increasing and decreasing flow rates to reduce the hysterysis effects and mean value was taken for the flow rate under consideration. Also, the total pressure drop across the heat exchanger was measured.

The uncertainty of the measurement of orifice diameter was 3%, and from the calibration curve, uncertainty of discharge coefficient was found to be 2%. The uncertainty of the pressure drop measurement was found to be 4.5% (maximum) for mercury manometer. The uncertainty in flow measurement was calculated to be 3.1% by using the procedure outlined by Moffat [10].

The experiments were conducted at steady state condition for the different flow rates such that it covers the full range of Reynolds number and also for different port diameters. All the experiments were carried out for U- Type configuration as shown in Fig 4. Initially experiments were conducted for finding the correlation for the friction factor as shown in Fig. 5 to know the flow resistance in a channel for varying flow rates. The following correlation for friction factor was obtained.

$$f = 21.4\,\mathrm{Re}^{-0.3} \quad \text{for } 500<\mathrm{Re}<5000 \tag{1}$$

The Reynolds number is defined on the basis of twice the plate spacing b, as

$$\mathrm{Re} = \frac{U_c(2b)}{\nu} \tag{2}$$

In PHE transition from laminar to turbulent takes place in the range of 400 to 500 Reynolds number. The experiments were conducted in the turbulent regime. The relationship between pressure drop in the channel and channel velocity can be written as

$$P - P^* = \zeta_c \rho \frac{U_c^{\,2}}{2} \tag{3}$$

where $\zeta_c = 1 + C_{Td} + f\frac{l_c}{d_e} + C_{Td}^{\,*}$

Turning pressure losses at the inlet and exit ports are small as compared to the pressure loss due to friction inside the corrugated channel. By substitution of friction factor correlation into Eq.3, channel pressure drop can be written in terms of channel velocity as follows

$$P - P^* = 21.4\left[\frac{U_c(2b)}{\nu}\right]^{-0.3} \frac{l_c}{D_e}\rho\frac{U_c^{\,2}}{2} \tag{4}$$

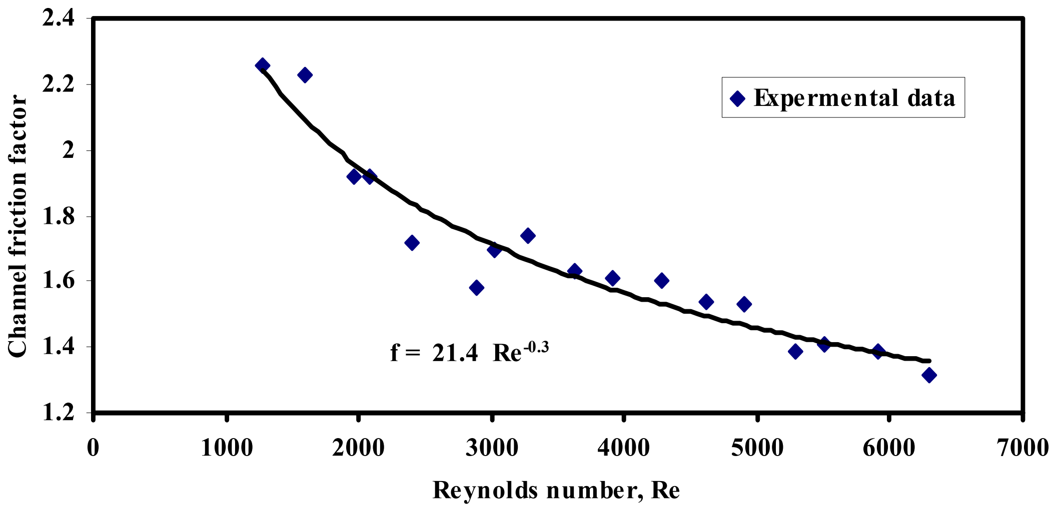

Fig.5 Flow friction characteristics of a single channel

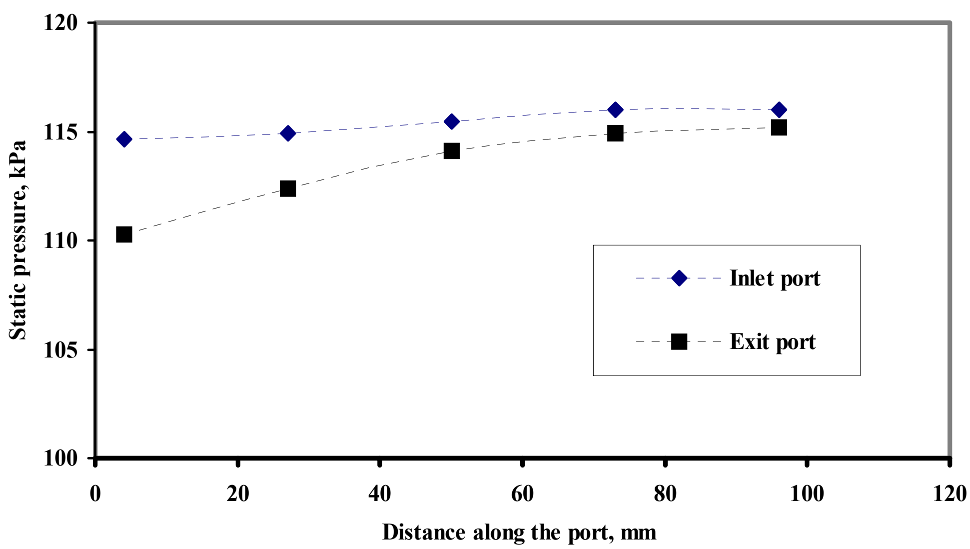

Fig.6 Port pressure profile for 24 mm diameter with flow rate of 1 kg/s

By substituting the measured pressure drop, channel velocity at a given port location has been calculated. Non-dimensional channel velocity has been calculated using the velocity profiles as follows.

$$u_c = \frac{U_c}{U_m} \tag{5}$$

RESULTS AND DISCUSSION

A number of experiments have been carried out for the range of Reynolds number 500 to 4000 and different port sizes of 24 mm and 34 mm diameter, for U-type configuration. From the channel pressure drops channel velocity has been calculated. The port pressure profiles and velocity profiles have been obtained using these results.

Figure. 6 indicates pressure profiles in the inlet and exit ports for flow rate of 1 kg/s with 24 mm port diameter. Variation in the pressure is due to momentum change as a result of flow branching in the channel. Pressure rises in the intake conduit due to the decrease of fluid velocity as the fluid flows out to the channels, and the pressure falls in the exit port due to the increase of fluid velocity as fluid flows into the port from the channels.

The channel pressure drop decreases gradually, which indicates the decrease in flow rate in the channel as the fluid flows through the port. This in turn shows the non-uniform distribution of the fluid from the port to the channels. For the same port size, with flow rate of 2 kg/s, pressure profiles

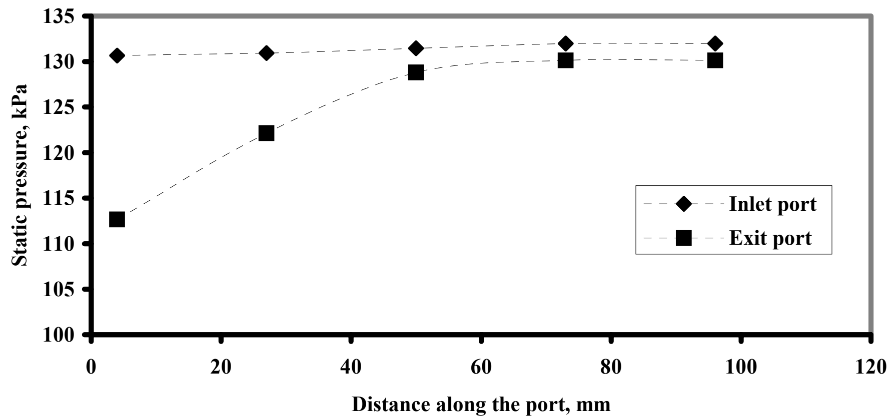

Fig.7 Port pressure profile for 24 mm diameter with a flow rate of 2 kg/s

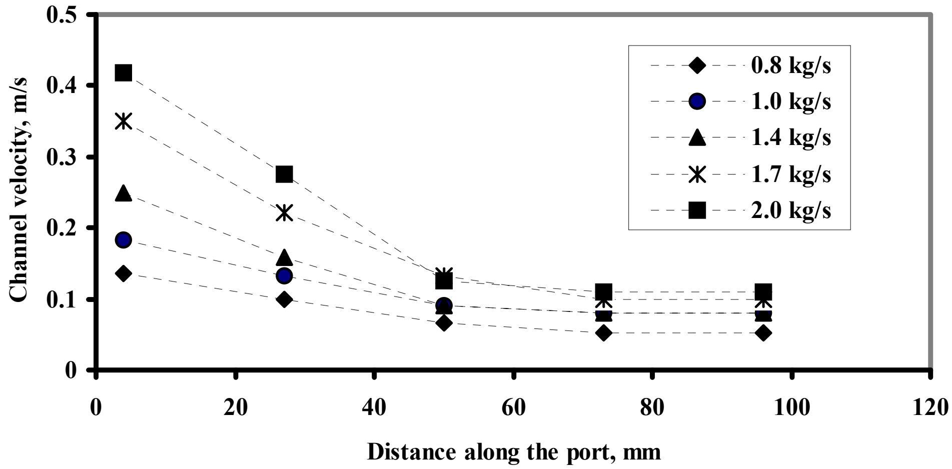

Fig.8 Channel velocity variation along the port of 24 mm diameter.

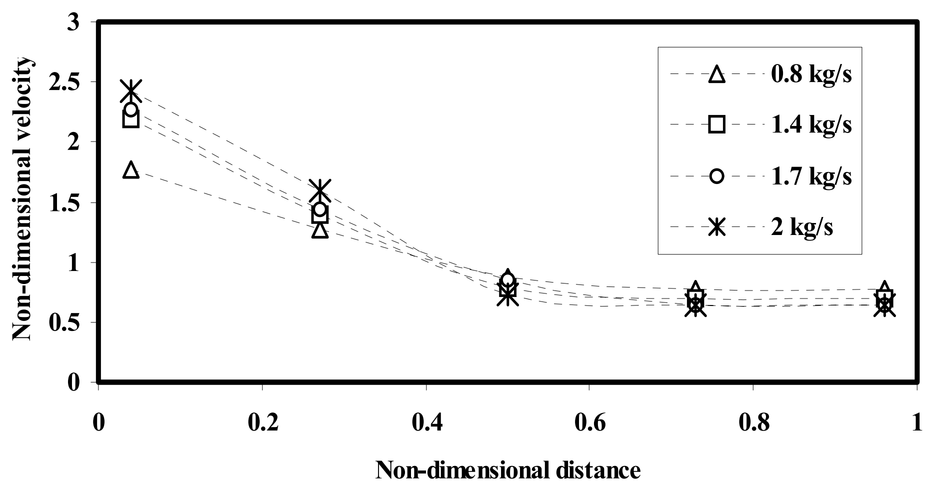

Fig.9. Non-dimensional channel velocity variation along the port of 24 mm diameter.

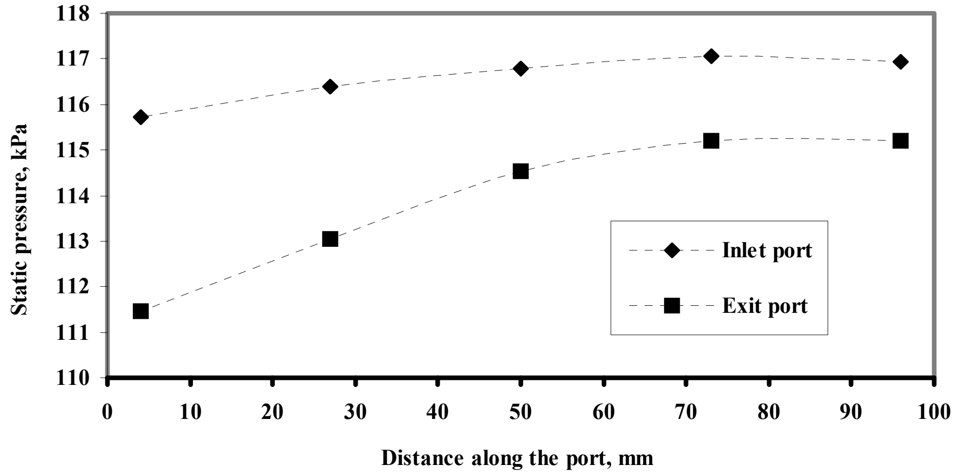

Fig.10 Port pressure profile for 34 mm diameter with a flow rate of 2 kg/s.

are shown in Fig.7. When compared to flow rate of 1kg/s, at the entrance pressure drop is about four times, and as a result the variation in channel pressure drop is more. This shows that the flow non-uniformity increases with the flow rate.

Channel velocity variation along the port for different flow rates is shown in Fig. 8, flow non-uniformity increases with flow rates. Channels near the port entrance have higher velocities and farther the channel from the entrance, lower is its velocity. This shows that channels closer to the entrance carry more fluid and flow in end channels is minimum. The non-dimensional velocity profiles are shown in Fig. 9. Bassiouny and Martin [6,7] developed an equation for flow
distribution from port to channel, represented by a parameter ‘m^2’ known as the maldistribution parameter, as given below

$$m^2 = \frac{1}{\zeta_c}\left(\frac{nA_c}{A}\right)^2 \tag{6}$$

This equation is applicable for the identical inlet and exit ports. As the flow rate increases the channel friction coefficient decreases, which in turn increases flow maldistribution. The maldistribution also increases with the decrease of port diameter. The present experimental findings agree with this.

Figure. 7 and Fig. 10 indicate that at the entry to the heat exchanger pressure drop for 24mm diameter is about five times compared to in 34 mm diameter, and it is observed that non-uniform flow distribution decreases with the increase of port diameter. This observation also agrees with Eq. (6), which indicates that the parameter m^2 is inversely proportional to port cross-sectional area.

CONCLUSIONS

The flow maldistribution for port to channel flow in a plate heat exchanger has been determined experimentally. Direct local in-port pressure measurement with artificial port size reduction has been utilized for this purpose. The resuls indicate that the port size and fluid flow rate has major influence on the flow distribution.

It is evident from the results that the usual practice of designing PHEs considering uniform flow distribution particularly for smaller number of plates is questionable. Yet present experiment agrees with the physical features of port pressure distribution given by the flow channeling theory proposed by Bassiouny and Martin [6,7].

NOMENCLATURE

A	Cross-sectional area of the port, m^2
A_c	Cross-sectional area of the channel, m^2
b	Plate spacing, m
C_{Td}	Coefficient of turning loss from the inlet port to the channels
C_{Td}^*	Coefficient of turning loss from the channels to the exit port
d_e	Equivalent channel diameter, m
f	Friction factor for channel flow
m^2	Maldistribution parameter
n	Number of channels
P	Pressure in intake port, Pa
P^*	Pressure in exit port, Pa
Re	Reynolds number
U_c	Channel velocity, m/s
u_c	Non-dimensional channel velocity
U_m	Mean velocity, m/s
ζ_c	Channel frictional coefficient
ν	Kinematic viscosity of the fluid, m^2/s
ρ	Density of the fluid, kg/m^3

REFERENCES

Watson, E. L., McKillop, A.A., Dunkley, W.L., and Perry, R.L., 1960, Plate Heat Exchanger - Flow Characteristics, *Industrial and Engineering Chemistry,* Vol. 52, pp. 733-744.

Buonopane, R.A., Troupe, R.A., and Morgan, J.C., 1963, Heat Transfer Design method for Plate Heat Exchangers, *Chemical Engineering Progress*, Vol. 59, pp. 57-61.

Bajura, R.A., 1971, A Model of Flow Distribution in Manifolds, *Journal of Engineering for Power*, pp. 7-12.

Datta, A.B., and Majumdar, A.K., 1980, Flow Distribution in Reverse Flow Manifolds, *International Journal of Heat Fluid Flow*, Vol. 2, pp. 253-262.

Bajura, R.A., and Jones Jr., E.H., 1976, Flow Distribution Manifolds, *Journal of Fluid Engineering Transactions,* ASME 98, pp. 654-666.

Bassiouny, M.K., and, Martin, H., 1984, Flow Distribution and Pressure Drop in Plate Heat Exchangers-I, U-type, *Chemical Engineering Science,* Vol. 39, pp. 693-700.

Bassiouny, M.K., and, Martin, H., 1984, Flow Distribution and Pressure Drop in Plate Heat Exchangers-I, Z-type, *Chemical Engineering Science,* Vol. 39, pp. 701-704.

Prabhakara Rao, B., and Sarit K. Das, 2004, An Experimental Study on the Influence of Flow Maldistribution on the Pressure Drop across a Plate Heat Exchanger, *Transactions of the ASME*, Vol. 126, pp 680-691.

Prabhakara Rao, B, Krishna Kumar, P, and Sarit K. Das, 2002, Effect of Flow Distribution to the Channels on the Thermal Performance of a Plate Heat Exchanger, *Chemical Engineering and Processing*, Vol. 41, pp. 49-58.

Moffat, R.J., 1988, Describing the Uncertainties in Experimental Results, *Experimental Thermal and Fluid Science*, Vol. 1, pp 3-17.