Proceedings of Fifth International Conference on **Enhanced, Compact and Ultra-Compact Heat Exchangers: Science, Engineering and Technology**, Eds. R.K. Shah, M. Ishizuka, T.M. Rudy, and V.V. Wadekar, Engineering Conferences International, Hoboken, NJ, USA, September 2005.

**CHE2005 - 25**

# AN EXPERIMENTAL INVESTIGATION OF THE PORT TO CHANNEL FLOW AND PRESSURE DISTRIBUTION OF THE SMALLER AND LARGER PLATE PACKAGE HEAT EXCHANGERS

**Prabhakara Rao Bobbili [1], Bengt Sunden [2,*] and Sarit Kumar Das [3]**

[1] Division of Heat Transfer, Department of Heat and Power Engineering, Lund Institute of Technology, Lund University, P.O. Box 118, SE-22100 Lund, Sweden. Email: Prabhakara_Rao.Bobbili@vok.lth.se

[2,] Division of Heat Transfer, Department of Heat and Power Engineering, Lund Institute of Technology, Lund University, P.O. Box 118, SE-22100 Lund, Sweden. Email: bengt.sunden@vok.lth.se

[3] Heat Transfer and Thermal Power Laboratory, Department of Mechanical Engineering, Indian Institute of Technology-Madras, Chennai – 600 036, India. Email: sarit_das@hotmail.com; skdas@iitm.ac.in

## ABSTRACT

An experimental investigation has been carried out to find the flow and the pressure difference across the port to channel in plate heat exchangers for a wide range of Reynolds number, 1000 to 17000. In the present study, low corrugation angle plates have been used for different number of channels, namely, 20 and 80. Water has been used as working fluid for both hot and cold fluid sides. The pressure probes are inserted through the plate gasket into both the inlet and exit ports of the channel. The pressure drop is recorded at the first, middle and last channels for each plate package of the heat exchanger. Also, the overall pressure drop has been measured for various flow rates. This overall pressure drop is a function of the flow rate, the cross-sectional area ratio of channel to port and number of channels per fluid. The results indicated that the flow maldistribution increases with increasing overall pressure drop in the plate heat exchangers. The experimental results are verified with Bassiouny and Martin (1984a) analytical results.

* Corresponding author: Division of Heat Transfer, Department of Heat and Power Engineering, Lund Institute of Technology, Lund University, P.O. Box 118, SE-22100 Lund, Sweden.
Phone: +46-46-2228605 ; Fax: +46-46-2228612
E-mail: bengt.sunden@vok.lth.se

## 1. INTRODUCTION

The usage of Plate Heat Exchangers (PHE) in industries around the world has increased considerably in recent years. They have been used in the process industry for many years due to a number of advantages such as compactness, flexibility, ease of maintenance and ability to recover heat at extremely small temperature differences. The energy crisis in the 1970's opened a field of application for highly efficient heat exchangers of the compact type which could utilize a maximum of waste heat through enhanced surfaces and devices. However, the demands of the process industries include mainly reliable operation and most often combination of temperature levels, pressure drop, fouling resistance and ease of maintenance, which have been excellently met by the plate heat exchangers making them the fastest growing member of the heat exchanger family.

Heat Exchangers in general and PHEs in particular undergo deterioration in performance due to flow maldistribution. The common idealization in the basic plate heat exchanger design theory is that the fluid is distributed uniformly at the inlet of the exchanger on each fluid side throughout the core. However, in practice, flow maldistribution is more common and significantly reduces the idealized heat exchanger performance. Flow maldistribution can be induced by the heat exchanger geometry, operating conditions (such as viscosity or density-induced maldistribution), multiphase flow, fouling phenomena etc. Geometry-induced flow maldistribution can be classified into gross flow maldistribution, passage - to - passage flow maldistribution and manifold-induced flow maldistribution. Port-to-channel flow maldistribution belongs to the manifold-induced flow maldistribution and it mainly depends on the port and channel geometries and channel friction coefficient $\zeta_c$. This parameter can be calculated as reported by Bassiouny and Martin (1984a). For PHEs, the ports acting as manifolds may induce maldistribution in the flow channels which is dependent on the flow configuration (U or Z type) and pass arrangement. The influence of flow maldistribution on heat transfer equipment performance was clearly mentioned by Kitto and Robertson (1989). A good review of the work devoted to the problems associated with maldistribution has been compiled by Mueller and Chiou (1988).

The analytical model for flow distribution in manifolds has been described and presented as closed form equations using the general flow channeling and unification concept by Bajura and Jones (1976). Using this concept, the plate heat exchanger flow distribution was analytically modeled for flow and pressure distribution for both U and Z type arrangements by Bassiouny and Martin (1984a, 1984b). They presented a derived maldistribution parameter from the momentum equations through an energy balance between the inlet and outlet ports of the PHE. This parameter indicates the magnitude of flow maldistribution for a given plate package. This parameter is a function of the number of channels, the cross-sectional area ratio of the plate channel and the port and the flow resistance in the channel.

Based on the same theory, Huang (2001) reported that Heat Transfer Research Institute (HTRI) has developed a model and carried out an experimental study by using different corrugated plates (mixed corrugated plates) for predicting port pressure distributions and channel flow rate distributions in a single pass plate heat exchanger based on port maldistribution data. However, they have not carried out experiments by varying the number of channels in a plate heat exchanger extensively. Guidelines were presented to minimize port flow maldistribution. They reported that higher port flow maldistribution occurs in plate heat exchangers with Z arrangements than those with U arrangements, which also is according to practical experience. The flow maldistribution due to port to channel flow has a severe effect on the heat exchanger thermal performance which has been investigated analytically by Rao et al. (2002) and Rao (2004), and experimentally by Rao and Das (2004) and Rao et al. (2005). Rao and Das (2004) have conducted an experimental study on the port to channel flow maldistribution in a small package of a 37 channel plate heat exchanger by creating the flow maldistribution while inserting a reduced cross-sectional area of a wooden mandrel in both inlet and outlet ports of the PHE. These investigations have been carried out based on the analytical models for a single phase flow distribution from port to channel in a PHE conduit by Bassiouny and Martin (1984a, 1984b).

Thus, the investigators have reported results for a small plate package to study the effect of the port flow maldistribution in a plate heat exchanger by changing the port and plate geometry. The present experimental study aims to bring out a clear picture about how the pressure drop is affected by the flow maldistribution in an actual U-type PHE in presence of a large number of channels per fluid and for a wide Reynolds number range. The emphasis of the study is the application of a simplified flow channelling theory to the engineering of PHEs for practical applications.

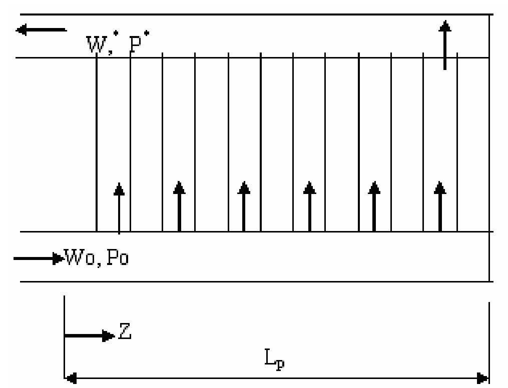


Fig. 1 Flow arrangement for a U type plate heat exchanger

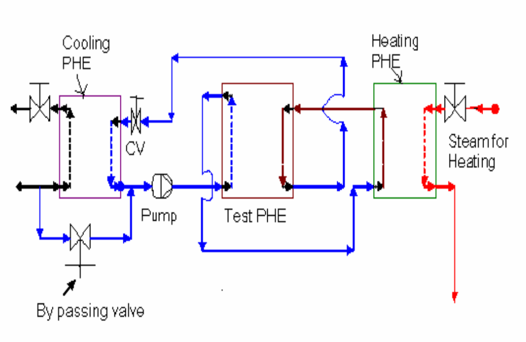


Fig. 2 The detailed circuit of cold and hot fluids

## 2. EXPERIMENTAL SETUP AND PROCEDURE

The test plate and frame heat exchanger consists of a maximum of 80 (40/40) channels. It has been chosen to investigate the flow distribution in a small (21 plates) and large (81 plates) plate package from the first channel to the last channel of a U type arrangement. The U-type PHE flow arrangement has been shown in Fig. 1. A line diagram of the detailed fluid flow circuit through various components like a test plate heat exchanger, hot and cold water circuits, circulation pump, two auxiliary heat exchangers for cooling the exit of hot water and for heating cold water are shown in Fig. 2. The specifications of the used corrugated plate are given in Table 1. The magnetic flow meter has been used to record both the mass flow rates of the two fluid streams. It has been calibrated with a standard flow meter and the maximum deviation is about ±3.0%. Three differential pressure transmitters (range from 0 to 400 kPa) are used to record the pressure drop across the channels and for both inlet/outlets of the two streams. The two differential pressure transmitters are used to read the pressure drop across the test plate heat exchanger at the inlet and outlet of the cold and hot fluid streams, which are connected to a computerized data-acquisition unit and another pressure transmitter is used to measure the pressure drop along the channels in the plate

**Table: 1 Geometric characteristics of a plate**

| S.No | Particulars | Dimensions |
|---|---|---|
| 1 | Port diameter, $D_p$ | 32 mm |
| 2 | Equivalent diameter of the channel, $d_e$ | 4.8 mm |
| 3 | The vertical distance between ports, $L_p$ | 357 mm |
| 4 | Plate width (gasket to gasket),w | 100 mm |
| 5 | Chevron angle, $\beta$ | $30^o$ |
| 6 | Corrugation pitch, $\delta$ | 14 mm |
| 7 | Amplitude of corrugation, b | 2.4 mm |
| 8 | The plate material | SS |
| 9 | The gasket material | EPDM |

package by using small pressure steel tube taps having a 2 mm internal diameter. The two small tubes are fixed to the corrugated plate by welding through plate gasket at the inlet and outlet of the each channel port. The arrangement of the fixed pressure taps in relation to the particular channel is shown in Fig. 3. The fixed pressure taps are connected using flexible pipes and these flexible pipes were fixed to the pressure control value panel, where the inlet and outlet of the measured channel were connected to the pressure transmitter. A multi-meter is used to record the pressure signal from the pressure transmitter, which gives a signal in form of current, mA.

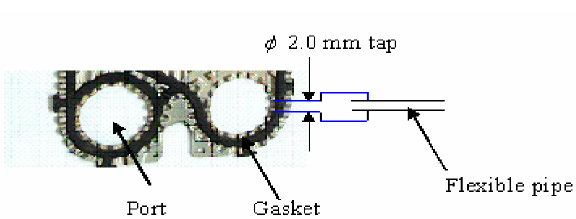


Fig. 3 Arrangement of the pressure tap of the channel in the tested plate heat exchanger.

In addition to these, two mobile static pressure tubes are used to track the static pressure drop in both the inlet and outlet ports at different locations of the channels. These were connected to the pressure difference switch value panel where the pressure difference transmitter is fixed, see Fig. 4. With these probes, the static pressure drop in each channel can be measured. The deviation of the static pressure drops between the fixed probes and the mobile probes at the particular location of the channel has been found to be about maximum ± 2.0 %. All three pressure difference transmitters have been calibrated with a standard pressure meter and the maximum uncertainty is about ±1.0%. Eight PT-100 thermometers are used to measure the inlet/outlet temperatures of the cold and hot fluids in the plate heat exchanger as shown in Fig. 4. At each inlet and outlet of the fluid steams, 2 PT-100 thermometers are located to record the average temperature of each fluid. These thermometers are placed close to the PHE ports in the well-insulated pipe sections. All eight PT-100 thermometers were connected to the data-acquisition unit to record the fluid mean temperatures. These thermometers are calibrated with a standard thermometer and the deviation from the standard thermometer is about maximum ± 0.15%. The experiments have been carried out for isothermal conditions at T= $20^oC$ and a non-isothermal test at $T_m$ = $40^oC$. Both the tests have been carried out simultaneously for the small and the large plate package. All the test points are repeated at the same conditions to check the repeatability and consistency was been found.

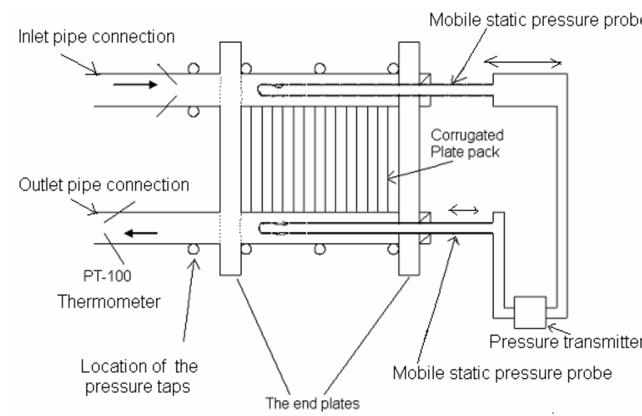


Fig. 4 Various locations of the fixed the pressure taps, the differential pressure transmitter and PT-100 thermometer across the tested plate heat exchanger.

## 3. DATA REDUCTION AND EVALUATION OF FLOW MALDISTRIBUTION PARAMETER

The total pressure drop between the inlet and outlet of the two fluid steams have been recorded with the help of the data logger at steady state conditions. The measured total pressure drop across the PHE for the two steams under both isothermal and non-isothermal conditions consists of three components: the pressure drop in inlet/outlet ports of the channels, the pressure drop due to reduction in cross-sectional area between connection and port, the pressure drop in the short length of the connecting pipe (ID 36 mm) due to friction, the pressure drop in the main corrugated flow passage of the plate package. Hence, the mean channel pressure drop can be obtained from

$$\Delta P_{chm} = \Delta P_{tp} - \Delta P_{port} - \Delta P_{ec} - \Delta P_{f,pipe} \tag{1}$$

The port pressure drop is calculated based on an empirical equation (Shah and Focke, 1988)

$$\Delta P_{port} = 1.5\rho \frac{V_p^2}{2} \quad (2)$$

The pressure drops due to sudden contraction and sudden expansion at inlet and outlets respectively, have been calculated by using the following formula.

$$\Delta P_{ec} = K_{ec}\rho \frac{V_p^2}{2} \quad (3)$$

where, $K_{ec}$ is the total pressure loss coefficient of the sudden contraction and expansion at the inlet and outlet of the ports. This has been calculated using the procedure of Miller (1990).

The pressure drop due to friction in the small steel connecting pipe at the inlet and outlet of the PHE was estimated on the basis of the smooth steel tube friction factor and pipe flow velocity as

$$\Delta P_{f,pipe} = 4 f_{pipe} \frac{L_{pipe}}{d_{pipe}} \rho \frac{V_{pipe}^2}{2} \quad (4)$$

The pressure drop due to elevation has been neglected because of the small difference in elevation between inlet and outlet. For the flow resistance due to shear in the corrugated passage, the Fanning friction factor, $f_{ch}$ is introduced according to

$$\Delta P_{chm} = 4 f_{ch} \frac{L_{ch}}{d_h} \rho \frac{V_{chm}^2}{2} \quad (5)$$

All the pressure drop measurements have been carried out at isothermal conditions, and the fluid properties were calculated at the mean flow temperatures.

In this analysis, the flow maldistribution parameter, $m^2$, as introduced by Bassiouny and Martin (1984a, 1984b) has been taken as the principal parameter to designate the port to channel maldistribution. The physical significance of the maldistribution parameter is that, it is an index of deviation of the flow from the mean channel velocity in a plate heat exchanger. It is a function of the momentum correction factor, the plate channel flow geometry and the resistance to flow in the channel which results in the overall pressure drop from inlet to outlet. The value of $m^2$ approaches zero when the flow is uniformly distributed among the channels. The more flow maldistribution, the higher is the value of $m^2$.

The flow maldistribution was calculated from the experimental data using the following non-dimensional overall pressure drop equation, which is taken from the Bassiouny and Martin (1984a) model for a U -type PHE (flow arrangement shown in Fig. 1).

$$\left(\frac{\Delta P_{first}}{\rho V_p^2}\right)_{z=0} = \left(\frac{m^2}{\tanh^2 m}\right)\left(\frac{A_p}{nA_c}\right)^2 \frac{\zeta_c}{2} \quad (6)$$

Here, $\Delta P_{first} = \Delta P_{chm} + \Delta P_{port}$

In the above equation, the only unknown parameter is 'm' and the other parameters are known for a given plate heat exchanger. The maldistribution parameter can be found by using an iterative method for the measured overall channel pressure drop and the flow rate. The obtained $m^2$ value has been used to obtain the pressure distribution from the first channel to the last channel by using the following equation given by Bassiouny and Martin (1984a).

$$\frac{\Delta P_{first}}{\rho V_p^2} = \left(\frac{A_p}{nA_c}\right)^2 \frac{\zeta_c}{2} m^2 \left(\frac{\cosh^2 m(1-z)}{\sinh^2 m}\right) \quad (7)$$

The value of the obtained maldistribution parameter $m^2$ can be compared with the following theoretical value given for identical inlet and exit port dimension,

$$m^2 = \left(\frac{nA_c}{A_p}\right)^2 \frac{1}{\zeta_c} \quad (8)$$

Here $\zeta_c$ is the overall frictional resistance of the channel and is equal to $\zeta_c = 4 f_{ch} \frac{l_{ch}}{d_h}$. The theoretical values of $m^2$ calculated from Eq. (8) for the present PHE are shown in Table 2.

**Table: 2 The flow maldistribution parameter, $m^2$ for the flow distribution in PHE**

| Re | n=10 | n=20 | n=30 | n=40 | n=100 | n=200 |
|---|---|---|---|---|---|---|
| 1000 | 0.30 | 1.22 | 2.74 | 4.87 | 30.44 | 121.8 |
| 2000 | 0.33 | 1.34 | 3.03 | 5.39 | 33.66 | 134.6 |
| 3000 | 0.36 | 1.43 | 3.21 | 5.71 | 35.70 | 142.8 |
| 5000 | 0.38 | 1.54 | 3.46 | 6.15 | 38.44 | 153.8 |
| 10000 | 0.43 | 1.70 | 3.83 | 6.80 | 42.51 | 170.0 |
| 15000 | 0.45 | 1.80 | 4.06 | 7.21 | 45.08 | 180.3 |

## 4. THE UNCERTAINTY IN MEASUREMENTS

The uncertainty analysis for the derived quantities was carried out following the procedure given in Moffat (1988). The uncertainty of the flow rate measurement was estimated to be maximum ± 3.0 percent. The uncertainty in the measurement of pressure was found to be ± 1.0 percent maximum. The maximum errors in measurements of the primitive variable ΔP and m were ± 2.5%, and ± 2.1%, respectively. Using the above values of measured quantities, the maximum uncertainty in the values of Re and f, were calculated to be ± 3.5% and ± 4.0 %, respectively.

## 5. RESULTS AND DISCUSSION

Theoretically, the port flow maldistribution is caused by port pressure variation in a given pass so that the flow distribution among channels is determined by pressure profiles at the inlet and outlet ports and by the hydraulic resistance in the channels. Two factors, namely, fluid friction and momentum change, affect the pressure profiles in PHE manifolds. Figure 1 shows a U-type arrangement with inlet and outlet ports on the same side of the exchanger. The inlet port pressure increases along the port because the momentum gain from the decrease in flow rate is higher than the sum of friction and turn around losses. On the other hand, the pressure in the outlet port decreases due to both friction and momentum losses. As a result of this, the channel pressure drop, that is, the pressure difference from inlet port to outlet port decreases along the ports for the U-arrangement, which in turn causes the channel flow rate to decrease along the ports. The channel pressure drop decreases in the direction of the inlet port for a U-arrangement. The channel velocity distribution for the U arrangement also reflects the channel pressure drop distribution. From a basic principle, the inlet and outlet static pressure profiles in the ports are parallel to each other for uniform flow distribution while in case of non-uniform flow the profiles are convergent.

The experiments were designed for 200 kPa, 100 kPa, 50 kPa and 25 kPa at the connections for both the small and the large plate heat exchanger. To observe the flow phenomena in the small and the large plate packages of the plate heat exchanger, the experiments have been carried out under isothermal and non-isothermal conditions for the range of Reynolds number from 1000 to 17000 for both corrugated plate packages. As stated earlier, the total/overall pressure drop has been recorded at various fluid flow rates of each stream. The mean channel pressure drop has been calculated by using Eq. (1) and the obtained value is substituted in Eq. (5) to calculate the Fanning friction factor. The procedure has been repeated for different flow rates and corresponding mean channel pressure drops to develop a correlation between the Fanning friction factor and Reynolds number by using a regression method, see Fig. 5. The Reynolds number is based on hydraulic diameter of the corrugated channel which is equivalent to twice the pressing depth of the channel. The following correlation has been obtained for the range of fluid flow rates at isothermal conditions.

$$f = 1.059\, Re^{-0.145} \qquad for\ \ 900 < Re < 10{,}000 \tag{9}$$

Eq. (9) has also been checked at higher Reynolds number and the maximum deviation has been found to be about ± 2.9 %, where the Reynolds number achieved is about 17,500.

To compare the experimental pressure drop at the inlet and outlet connections and the mean channel pressure drop data obtained from Eq. (1), both the data sets have been non-dimensionalised with the port dynamic pressure as follows.

$$\Delta p_{cc} = \frac{\Delta P_{tp}}{\Delta P_{port}} \tag{10}$$

$$\Delta p_{ch} = \frac{\Delta P_{chm}}{\Delta P_{port}} \tag{11}$$

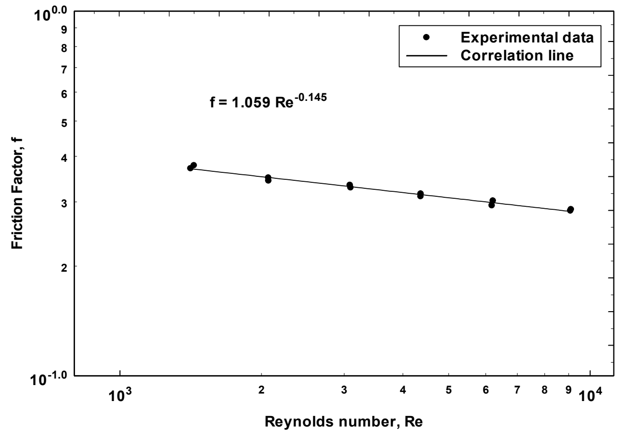

Fig. 5 Flow friction characteristics of chevron plate (the corrugation angle, $\beta = 30°$)

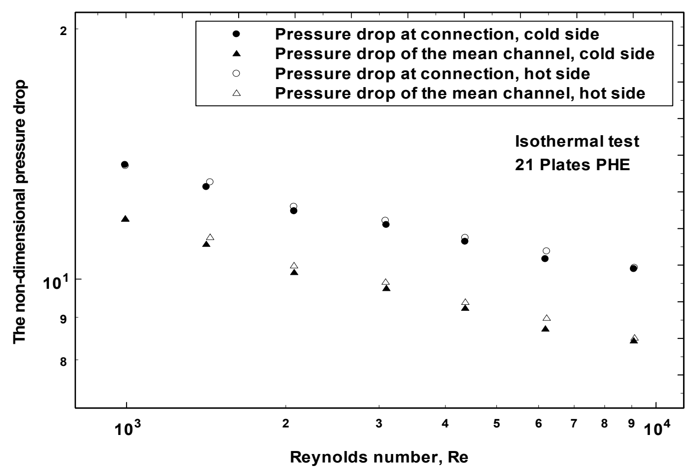

Fig. 6 Comparison of the experimental pressure drop between the connection and the mean channel for a 21 plate package plate heat exchanger.

The obtained non-dimensional pressure drop at the connection and the mean channel have been compared at isothermal conditions for a 21 plate package PHE as shown in Fig. 6. It is observed that a constant difference between the mean channel pressure drop and the total pressure drop exits at the connections. The cause of this pressure drop difference is due to the sudden contraction and expansion at inlet and outlet of the ports. Hence, this pressure loss will affect the first few channels and there will be flow maldistribution from the first channel even for the small package plate heat exchangers due to flow separation at the inlet and outlet connections. The diameter of the connections at inlet and outlet of the ports should be the same to minimize the pumping power of the two fluid streams. The same trend can be observed for the non-isothermal tests as shown in Fig. 7.

To observe any influence of the properties variation in the mean channel pressure drop, the non-dimensional mean channel pressure drop for the cold and hot fluid streams for isothermal and non-isothermal conditions were plotted versus

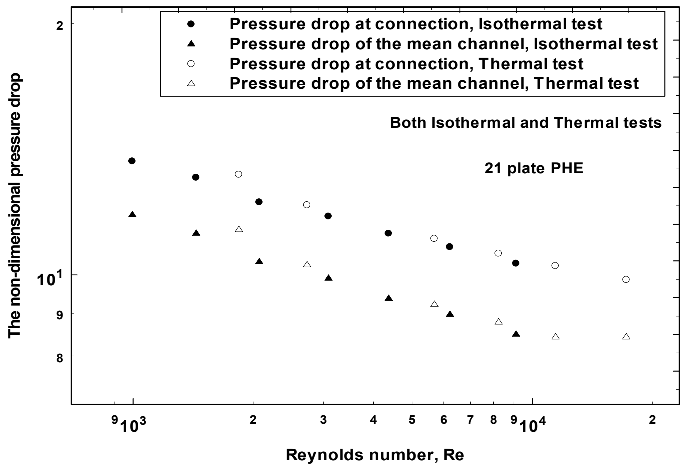


Fig. 7 Comparison of the experimental pressure drop of both isothermal and thermal tests for the connection and the mean channel for a 21 plate package plate heat exchanger.

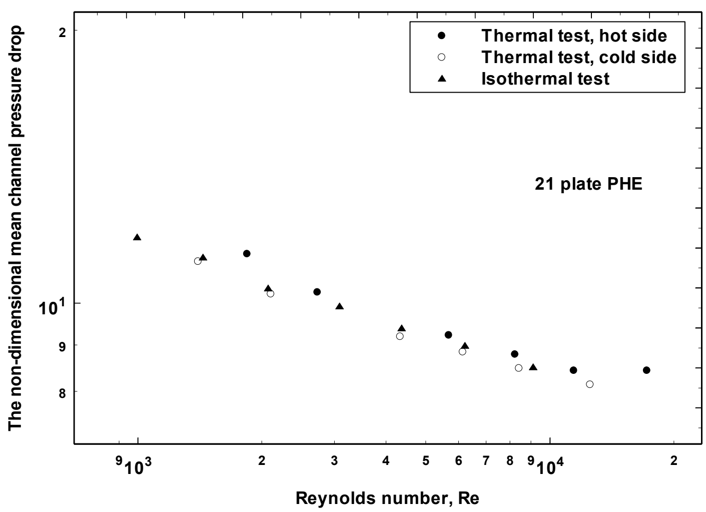


Fig. 8 Comparison of the mean channel pressure drop of isothermal and non-isothermal tests.

Reynolds number as shown in Fig. 8. The hot fluid stream has a higher pressure drop than the cold fluid steam as the dynamic viscosity changes with temperature. Hence, the isothermal mean channel pressure drop falls between the cold and hot fluid streams, see Fig. 8.

The channel pressure drop has been recorded for the first, the middle and the last channels of a 21 plate package and an 81 plate package plate heat exchanger. The obtained data has been non-dimensionalised based on the mean channel pressure drop of the plate package of the plate heat exchanger as follows.

$$\Delta p^{1} = \frac{\Delta P_{ch}}{\Delta P_{chm}} \tag{12}$$

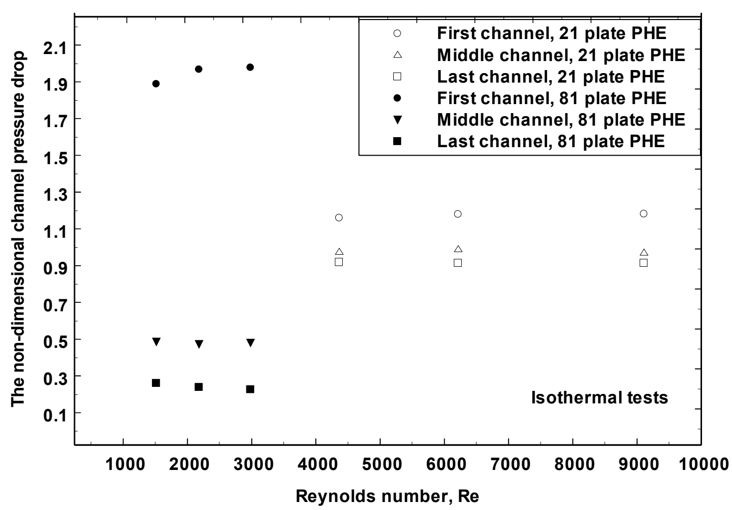


Fig. 9 Comparison of the channel pressure drop between a small and a large plate package plate heat exchanger.

The non-dimensional pressure drops of the first, middle and last channels of the 21 plate package and 81 plate packages versus Reynolds number at isothermal conditions have been depicted in Fig. 9. In Fig. 9, it is shown that the channel pressure drop varies from the first channel to the last channel. These changes are very severe in a large number of plates in a package whereas for a small number of channels, the flow maldistribution is very small. The measured channel pressure drops in the first and the last channel in an 81 plate package were found to be about 68.79 kPa and 7.81 kPa, respectively, while a 95.98 kPa pressure drop was found at the connection. For the 21 plate PHE, the corresponding pressure drops were 96. 22 kPa and 74.58 kPa, respectively, and 98.98 kPa was found at the connection. The magnitude of the non-dimensional mean channel pressure drop in the large plate heat exchanger is ranging from 0.25 to 1.99 whereas in the small plate heat exchanger it is from 0.92 to 1.2. The cause of the difference between the first and the last channel in a small plate heat exchanger might be due to the change of cross-section at the inlet and outlet of the ports and the connections. Also, the mean channel pressure drop of the larger plate package is smaller than the port pressure drop for a given mass flow rate. The port pressure drop is about twice the mean channel pressure drop in this case.

With the help of the mobile static probe, five channel pressure drops have been measured and compared with the calculated pressure drop of the Bassiouny and Martin (1984a). The results show good agreement with the analytical model for large package heat exchanger whereas for the smaller package plate heat exchanger deviation is found. In the analytical model, it is needed to calculate the flow maldistribution parameter, $m^2$, to compare with the measured data for the first channel to the last channel. The flow maldistribution parameter has been calculated by using Eq. (6). The brief calculation procedure was outlined in section 3. The calculated flow maldistribution parameter is about 0.62 in the small plate heat exchanger whereas for the large package heat exchanger it is about 3.54. With the calculated flow maldistribution parameter, the pressure drop in the channels have been computed by using Eq. (7) and compared with the measured experimental data. Figure 10 shows the pressure variation from the first channel to the last channel for a small and a large plate package. The deviation for the small plate heat exchanger is due to a large dynamic pressure which is calculated from Eq. (2). The constant in Eq. (2) has a higher value but in a small plate heat exchanger it will be around 1.2. This has been observed from the experimental data in a small package rather than in a large package plate heat exchanger. Hence, the results have confirmed the flow maldistribution in the channels at the same mass flow rate for a high number of channels and the effect on the hydraulic performance of the plate heat exchanger.

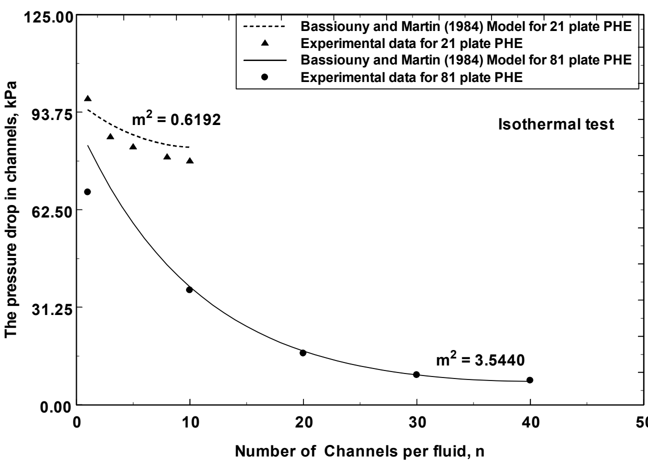


Fig. 10 Comparison of the experimental channel pressure drop with Bassiouny and Martin (1984a) analytical model in a small and a large package plate heat exchanger.

## 6. CONCLUSIONS

An experimental study has been carried out to bring forward the effect of the number of channels on the hydraulic performance in a corrugated channel plate heat exchanger. The experiments have been conducted for a 21 plate package and a 81 plate package of a plate heat exchanger for isothermal and non-isothermal conditions at steady state. The results show that the pressure distribution from the first to the last channel is decreasing more severely for the larger plate package than for the smaller plate package. Hence, the designers should consider the effect of number of channels on the hydraulic performance in the larger plate package of PHE to minimize the pumping power to the process fluids. From the present study, the effect of the reduction of the cross-sectional area at inlet and outlet ports to the connection of PHE was also observed and the measurement data showed that flow maldistribution will occur even for a smaller plate package by the reduction in cross-sectional area between the connection and ports. Hence, the size of the connector and the channel port should be the same to minimize the port flow maldistribution in a smaller plate heat exchanger.

## 7. ACKNOWLEDGEMENTS

The authors gratefully acknowledge the financial support from SIDA-Swedish Research Links and Swedish Research Council (VR) for the project under which the present collaborative research was carried out. Additional funding was provided from the Swedish Energy Agency (STEM). Alfa-Laval is also acknowledged for co-operation and funding the project. The authors wish to express gratitude to Mr. Leif Hallgren of Alfa-Laval Lund AB, Sweden for fruitful discussions with him during the course of the present work.

## 8. NOMENCLATURE

| | |
|---|---|
| $A_c$ | cross-sectional area of the channel, $m^2$ |
| $A_p$ | cross-sectional area of the port, $m^2$ |
| $d_h$ | hydraulic diameter of the corrugated channel, m |
| $d_{pipe}$ | connection pipe diameter, m |
| $f$ | Fanning friction factor, eq. (9) |
| $f_{ch}$ | channel friction factor |
| $f_{pipe}$ | Fanning friction factor in pipe (connection) |
| $K_{ec}$ | total pressure loss coefficient due to sudden expansion and contraction |
| $L_{ch}$ | length of the channel, m |
| $L_p$ | length of the port in PHE, m |
| $L_{pipe}$ | length of the pipe connection, m |
| $m^2$ | flow maldistribution parameter, Bassiouny and Martin (1984a) |
| $n$ | number of channels per fluid |
| $Po$ | pressure in the inlet port, Pa |
| $P^*$ | pressure in the outlet port, Pa |
| $\Delta p_{cc}$ | non-dimensional pressure drop at the connection, eq. (10) |
| $\Delta p_{ch}$ | non-dimensional pressure drop of the mean channel, eq. (11) |
| $\Delta p^1_{ch}$ | non-dimensional pressure drop in the channel, eq. (12) |
| $\Delta P_{chm}$ | pressure drop in the mean channels, Pa |
| $\Delta P_{ec}$ | pressure drop due to sudden expansion and contraction, Pa |

| | |
|---|---|
| $\Delta P_{first}$ | pressure drop at the first channel, Pa |
| $\Delta P_{f,pipe}$ | pressure drop due to friction in the pipe connection, Pa |
| $\Delta P_{port}$ | pressure drop in the inlet and outlet of ports in PHE, Pa |
| $\Delta P_{tp}$ | pressure drop in the inlet and outlet of ports at the connection (test point), Pa |
| *Re* | Reynolds number based on the mean channel velocity in PHE |
| $V_{chm}$ | mean channel velocity, m/s |
| $V_p$ | velocity in the port, m/s |
| $V_{pipe}$ | velocity in the pipe connection, m/s |
| $W^*$ | velocity in the outlet port, m/s |
| *Wo* | velocity in the inlet port, m/s |
| $z$ | non-dimensional axial location in the channel, $(Z/L_p)$ |
| $Z$ | axial location in the channel, m |
| $\beta$ | corrugation angle of the plate |
| $\rho$ | density of the fluid, kg/m$^3$ |
| $\zeta_c$ | overall friction coefficient of the corrugated channel |

## 9. REFERENCES


Bajura, R.A. and Jones, Jr. E.H., 1976, Flow Distribution Manifolds, *ASME, J. Fluids Engineering*, Vol. 98, pp. 654-666.

Bassiouny, M.K. and Martin, H., 1984a, Flow Distribution and pressure drop in plate heat exchangers-I, U-type arrangement, *Chemical Engineering Science*, Vol. 39,No.4, pp. 693-700.

Bassiouny, M.K. and Martin, H., 1984b, Flow Distribution and Pressure Drop in Plate Heat Exchangers-II, Z-type Arrangement, *Chemical Engineering Science*, Vol. 39, No.4, pp. 701-704.

Huang, L., 2001, Port Flow Distribution in Plate Heat Exchangers, *Proc. 3rd Int. Conference on Compact Heat Exchangers and Enhancement Technology for the Process Industries*, Davos, Switzerland, July, pp. 259-264.

Kitto, Jr. J. B., and Robertson, J.M., 1989, Effects of Maldistribution of Flow on Heat Transfer Equipment Performance, *Heat Transfer Engineering*, 10(1), pp.18-25.

Miller, D.S., 1990, *Internal Flow Systems*, 2nd ed., BHRA (Information Services), Bedford, UK.

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

Mueller, A.C. and Chiou, J.P., 1988, Review of Various Types of Flow Maldistribution in Heat Exchangers, *Heat Transfer Engineering*, Vol. 9, No. 2, pp. 36 - 50.

Rao, P.B., Theoretical and Experimental Study on the Effect of Flow Maldistribution on the Thermal Performance of Plate Heat Exchangers, Ph.D Thesis, Dept. of Mechanical Engineering, Indian Institute of Technology Madras, Chennai, India, 2004.

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

Rao, P.B., Kumar K. P., and Das, S.K., 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.

Rao, P.B., Sunden, B. and Das, S.K., 2005, An Experimental and Theoretical Investigation of the Effect of Elow Maldistribution on the Thermal performance of Plate Heat Exchangers, ASME, J. Heat Transfer, Vol. 127, (in Press).

Shah, R.K and Focke W.W., 1988, Plate Heat Exchangers and Their Design Theory, in *Heat Transfer Equipment Design*, R.K Shah et al., eds., Hemisphere, Washington, pp. 227-254.