FLOW MALDISTRIBUTION IN A SIMPLIFIED PLATE HEAT EXCHANGER MODEL - A Numerical Study

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***Abstract*− The performance of a plate heat exchanger (PHE) is severely influenced by non-uniform distribution of flow among its channels. Not only the PHEs, but many other process equipment needs uniform flow distribution for their optimum performance. Flow maldistribution (non-uniform distribution) is a common design problem which always puzzles process equipment designers. Being important design parameters, it has been investigated by several researchers and case based solution has been proposed and documented. Present numerical work is intended to target this aspect of the problem of PHEs but starts with a general investigation with simple multichannel geometry. The numerical setup consists of two headers having multiple channels for U- and Z- turn flow configuration under multichannel geometry and a simplified PHE for plate heat exchanger simulation. The problem has been investigated from hydrodynamic and thermodynamic view point. For hydrodynamic study, flow has been varied for Reynolds number 120 to 17600. It has been found that channel flow goes on reducing along downstream side. In thermal study the effect of wall temperature on air flow mal distribution has been investigated. Numerical results have been validated with the experimental results. Investigation reveals new features of flow mal-distribution which is helpful in better understanding of associated mal-distribution physics.**

Keywords- Flow maldistribution, Multi-Channels, U and Z-turn flow, Reynolds number, Plate type heat exchanger.

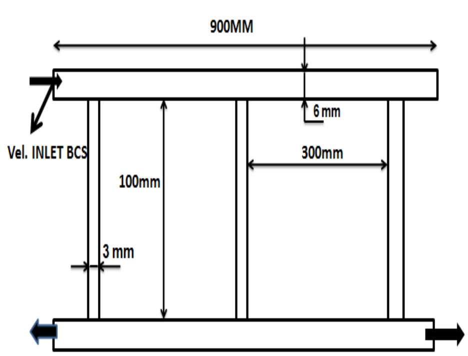
I. INTRODUCTION

In the multi channel flow distribution which observed in much industrial application, one of the common assumptions adopted in channel design is the uniform distribution of the fluid among channels. In many cases this leads to non- uniform flow velocity which may have direct impact on hydrodynamic behavior and thermal performance of the system. Plate heat exchanger (PHE) is one of the very common process equipment where flow maldistribution may have severe impact. PHE finds a wide application in pharmaceutical and dairy industries where efficient heat exchange is desired. But impact of maldistribution is not limited to PHE only. Its impact on the performance of process equipment has been realized by the equipment designers and researchers long back. A comprehensive review of flow maldistribution was first presented by Mueller at el. (1988). Most of the earlier works were concentrated on the effect of flow non-uniformity on the deterioration of the heat exchangers' performance with an assumption of parabolic inlet flow velocity profile Fleming (1967) set up a flow maldistribution model in paired-channel heat exchangers and investigated the effect of flow maldistribution on the performance deterioration. London (1970) developed an analytical method in order to obtain the influence of passage-to-passage maldistribution on heat exchanger performance. Shah and London (1980) extended London's work and set up an n-size passages model. Chiou (1978, 1985) presented experimental data obtained from wind tunnel experiments and set up a continuous flow distribution model. He also investigated the thermal performance deterioration in cross-flow heat exchangers based on this continuous flow distribution model. Ranganayakulu et al. (1997) presented a numerical analysis using FEM where the effects of inlet flow non-uniformity on thermal performance and pressure drop in cross flow plate fin heat exchangers was investigated. They found that the performance deterioration and variation in pressure drop are quite significant. Ranganayakulu and Seetharamu (1999) studied the combined effects of wall longitudinal heat conduction, inlet flow non-uniformity, and temperature non-uniformity on compact tube-fin and plate fin heat exchangers.

The summery of literature review shows the quantum and multi-dimensionality of work done on the subject of flow maldistribution. Most of the works are focused on developing theoretical model or experimental work where one kind of problem has been targeted at a time. Few numerical works are present, but they also focus on similar aspect of the problem. Geometry induced maldistribution is the most common type and is controllable at the designing level provided fluid flow features is known before executing designing process A computational flow analysis can be a great tool to the designers to handle the problem. Present work is dedicated to illustrate and bring out some local flow features which may be significant to the designers. The work starts with a simplified model of multi channel flow maldistribution in U and Z turn flow configuration and ends with presenting flow features among channels of simplified PHE. The objective of the problem is to study the affect of Reynolds number and wall temperature on the geometry induced maldistribution for both flow configurations.

II. PROBLEM DEFINATION AND ITS MATHEMATICAL FORMULATION

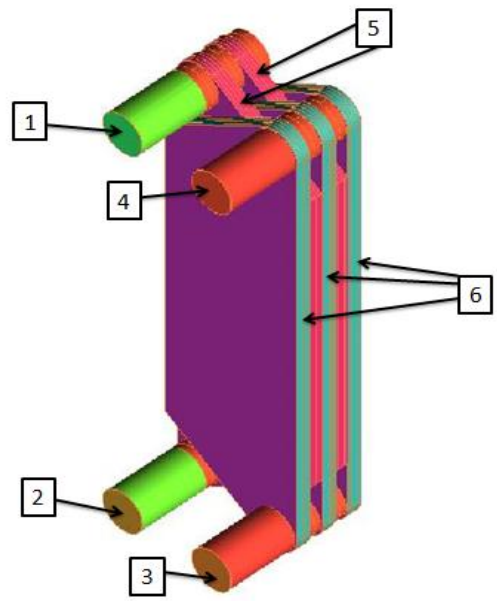
To analyze the problem of flow maldistribution fundamentally and to bring out the microscopic facts, the flow is assumed to be distributed among several parallel channels of equal size creating U- and Z- turn flow configuration as depicted in Fig.1 with its dimension. These flow configurations are very common in plate type heat exchangers. While analyzing the channel flow U- type configuration represents right side of bottom header closed and left side of bottom header is closed for Z- type configuration. With an intension to obtain a more realistic idea about geometry induced flow maldistribution a simplified PHE model is used whose solid model is shown in Fig.2 with its geometrical description. The plate geometry with its geometrical parameter is illustrated in Fig.3. Table 1 contains geometrical data pertaining to PHE plate.



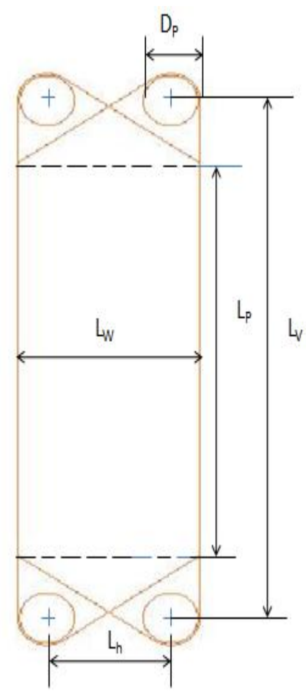
“Fig.1” U- and Z- turn configurations of multichannel

TABLE I. PLATE GEOMETRY

Port Diameter (Dp)	62 mm
Vertical Port Distance (Lv)	640 mm
Horizantal Port Distance (L_h)	140 mm
Effective channel width (Lw)	202 mm
Lp= Lv - Dp	578 mm



“Fig.2” Simplified solid model of PHE, (1) Hot inlet port, (2) Hot outlet port, (3) Cold inlet port, (4) Cold outlet port, (5) Hot channels and (6) Cold channels



“Fig.3” Simplified channel plates geometry

To carryout numerical investigation the flow is assumed to be three dimensional steady, incompressible, laminar and viscous where inertial and viscous forces are dominant. The effect of body force is assumed to be negligible. In the thermal analysis only convective and diffusive effect of heat transfer has been accounted.

Governing Equations:

For steady incompressible Newtonian fluid flow,

Continuity Equation,

$$\nabla.\mathbf{V} = 0 \quad (1)$$

Momentum Equation,

$$\rho\frac{\partial \mathbf{V}}{\partial t} + \rho\mathbf{V}.\nabla\mathbf{V} = -\nabla p + \mu\nabla^2\mathbf{V} \quad (2)$$

Energy Equation,

$$\rho C_p\frac{\partial T}{\partial t} + \rho\mathbf{V}.\nabla\left(C_p T\right) = k\nabla^2 T \quad (3)$$

Where **V**, T, ρ and p represents velocity vector, temperature, density and pressure respectively.

III. SOULATION METHODOLOGY

A. Multi channel Geometry:

The computation domain of the present problem consists of top and bottom headers of equal cross section connected with three flow channels of equal sized depicted in Fig.1. Water and air is used as a working to investigate the problem whose thermo-physical properties are illustrated in Table II. Following flow boundary conditions found to suitable and it has been implemented in present simulation work.

Inlet: Constant velocity inlet,
Outlets: Constant pressure,
Wall: No - slip.
The thermal study uses isothermal wall condition.

In order to get easy and robust pre-processing of the computational domain, ICEM has been used for geometry creation and meshing work where the technique of multi-block hexa meshing has been used. The simulation work has been carried out using FLUENT 6.3 solver. Some key features of present simulation work are,

Solver algorithm: SIMPLE
Interpolation scheme: FOU
Turbulence model: κ-ε with 2% turbulence intensity
6mm hydraulic diameter

B. Simplified model of plate type heat exchanger:

The preprocessing of simplified model of PHE has been done using ICEM as was done for multichannel geometry. The solid model of the computation domain generated in ICEM is illustrated in Fig.2. The unstructured meshing is selected over the structured meshing due high order of geometrical complexity involved in it.

Water is used as a working fluid on both side of the plate. The gaskets which allow entry of fluid in alternate channels is treated as adiabatic wall boundary condition Velocity inlet and respective entry temperature of fluid is used as boundary condition at both cold and hot inlets, whereas outlet boundary condition is set to pressure outlet. Since there are two different domains (hot and cold) separated by plate this is a conjugate heat transfer problem, hence all plates have a shadow of it, which has boundary condition of wall and coupled.

Solver algorithm: SIMPLE
Interpolation scheme: FOU
Turbulence model: κ-ε with 1% turbulence intensity
62mm hydraulic diameter

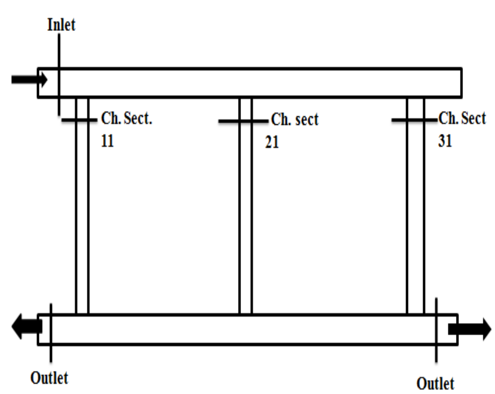
TABLE II. THERMO-PHYSICAL PROPERTIES OF FLUIDS

Properties	Water	Air
Thermal Conductivity	0.6 W/m-k	0..0242 w/m-k
Density	998.2 kg/m^3	1.225kg/m^3
Viscosity	0.001003 J/kg-k	1.7894×10^{-5}J/kg-k.
Specific Heat	4182 kg/m-s	1006.43 kg/m-s

IV. RESULTS AND DISCUSSIONS

A. Multi channel Geometry:

This section deals with the results and its analysis of the problem under investigation. The geometrical parameters of the computational domain are illustrated in Fig.1. This is a simple 3-D geometry. Analysis of the result has been presented based on flow configuration either U-turn or Z-turn. For precise illustration of the results, the geometry has been designated in specific manners which is shown in Fig.4.



“Fig.4” Designation of multi channel geometry

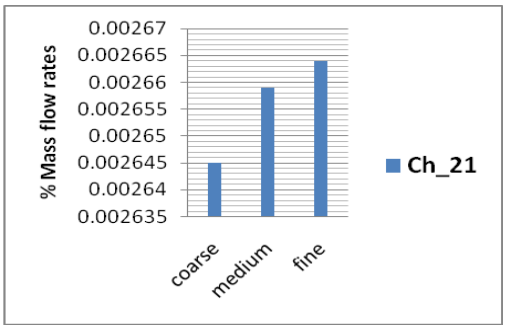
In present investigation Reynolds number has been varied from 1750 to 17600.

Grid Independence:

In order to check the grid sensitivity of the results, various grid sizes has been tried before selecting an optimum grid size. Mass flow rate obtained at section 21 has been used to as a parameter to represent the grid independence. Results for three grid sizes have been presented here for illustration which is depicted in Fig.5.

Mesh size considered are,

Coarse mesh: 325200 cells,
Medium mesh; 510336 cells
Fine mesh: 1364064 cells

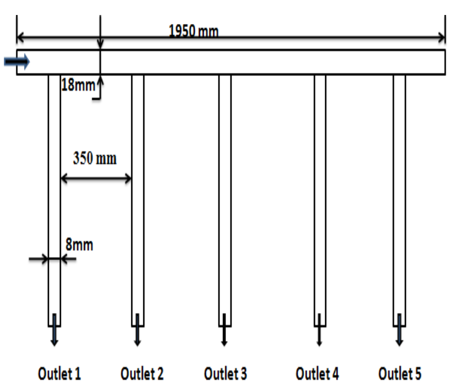


"Fig.5" % mass flow rates at Re=1750

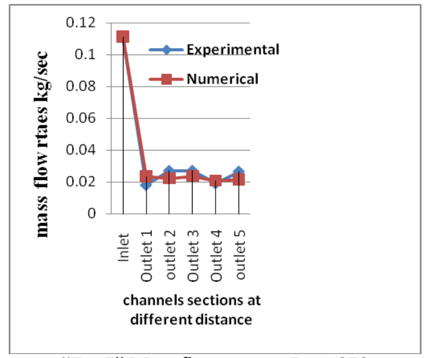
As it can be seen from the Fig.5, there is a little deviation (less than 5%) in the result predicted for medium and fine mesh sizes. Therefore medium mesh size has been selected to carry out remaining simulation work.

Validation:

As such there is no experimental or analytical work available in literature for the validation of the present work. So the present work has been validated by collecting experimental data on a simple setup specifically made for this purpose. Water has been used as medium for experimentation. The sketch of the setup is illustrated in Fig.6.



"Fig.6" Experimental set up for validation



"Fig.7" Mass flow rates at Re=1950

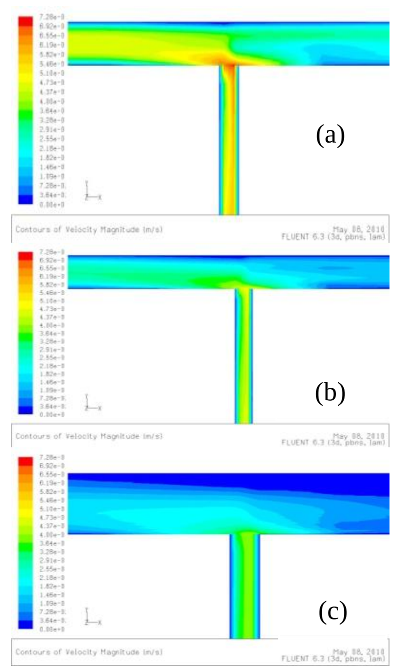
"Fig.7" presents experimental and simulated result for the case mentioned above. A close match between numerical and experimental work can be observed which is good for validation purpose.

1) Effect of Re on maldistribution:

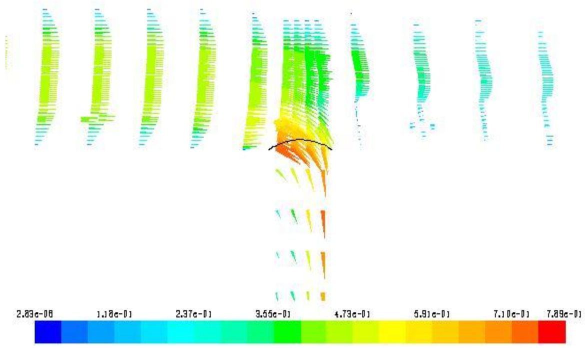
Effect of Re on flow maldistribution for U- and Z- turn flow configuration for water has been illustrated in following section with help of several plots and graphs. The results for U and Z turn flow configuration is expected to vary due to difference in energy potential (pressure head + kinetic head + datum head) available across each flow channel.

(a) U-turn flow configuration:

"Fig.8" (a), (b) and (c) shows the velocity contours of water at section 11, 21 and 31 respectively at Re=1759. "Fig.9" shows the velocity vectors of water at section 11, at Re=1759.

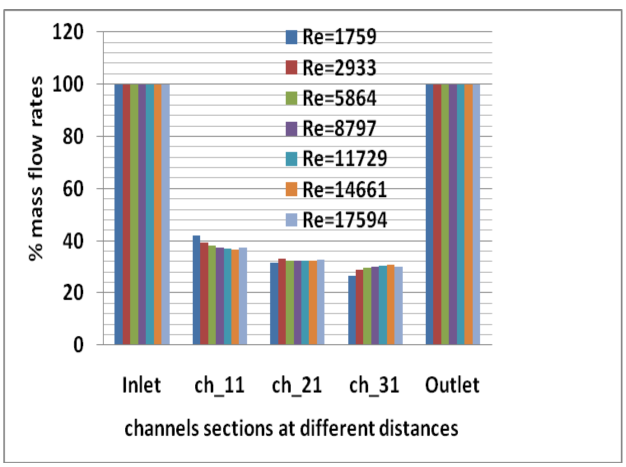


"Fig.8" Velocity contours (Re=1759) at section (a) 11, (b) 12 and (c) 13



"Fig.9" Zoomed view of velocity vectors at Ch_11 (Re=1759)

The difference in flow distribution at several channel section of equal size can be observed. The order of velocity magnitude is highest for the section 11 and goes on reducing along the downstream. Also a certain flow separation and recirculation at the first section can be observed which generally leads to flow obstruction and reduction in mass flow rate in the channel. At high Re this effect is expected to be more which leads to a significant change in flow distribution among channels. Existence of flow recirculation depends on local flow condition. Another important feature which can be observed is reducing magnitude of velocity which is because of lesser available mass flow rate at downstream side of the header which a natural reason for reducing flow in downstream channels.

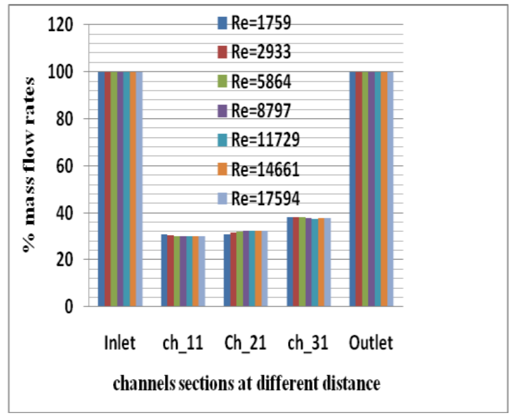


"Fig.10" Effect of Re on % mass flow rate for U-turn configuration

It can be observed from "Fig.10", the percentages mass flow rates at channel sections 11 decreases with the increase in Re. This feature can be attributed to increasing dominance of flow separation at higher Re where inertial forces become much stronger than viscous forces. A very mild increase in percentages mass flow rates with Re at channel sections 21 and 31 can be observed. This peculiar feature is probably due to reducing effect of flow reversal which is due to reduced local Re which the function of local velocity.

(b) Z-turn flow configuration:

In Z-turn flow configuration, first channel experiences highest energy potential difference and this difference goes on reducing in downstream side.

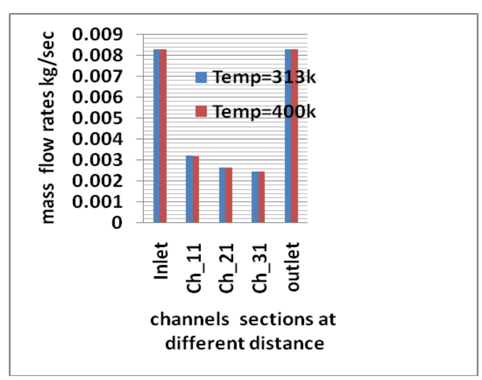


"Fig.11" Effect of Re on % mass flow rate for Z-turn configuration

Fig.11. shows the effect of Re on percentage mass flow rate at various sections. At a given Re, the percentage mass flow rate is found to be increasing in downstream side. This is reverse trend as obtained in U-turn flow configuration. This difference can be attributed to increased order of flow separation which goes on reducing in downstream due to weaker energy potential. Also it can be observed from Fig.6; percentages mass flow rates at all channel sections almost remain constant. This effect may be due to negligible impact increased Re compared to energy potential across a channel.

2) Effects of wall temperature on flow maldistribution:

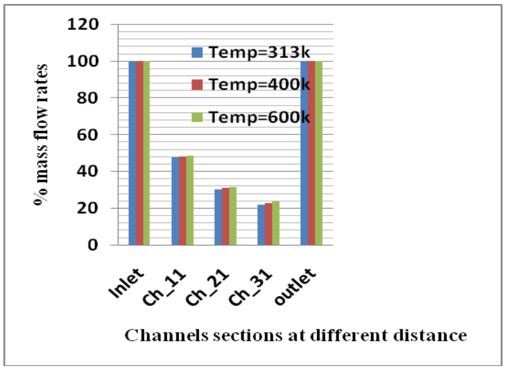
This section deals with the effect of wall temperature on flow maldistribution which has been investigated for water and air for several Re.



"Fig.12" Mass flow rates variation Re=1750, Temp=313k, 400k

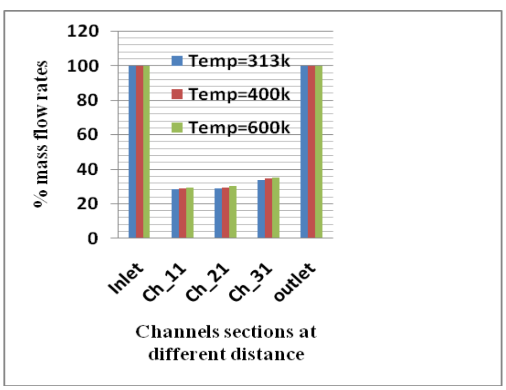
It can be observed from Fig.12 that flow maldistribution is independent of temperature variations. This may be because of independence of density of working medium (water) with temperature in the range considered for investigation. But in the case of working medium air, variation of density with temperature is going to occur. So a maldistribution is expected under such cases. To verify this understanding the flow maldistribution problem was investigated with air where temperature variation is involved. Results obtained under various cases have been presented in following section.

(a) U and Z-turn flow configuration:



"Fig.13" % Mass flow rates at Re=122 and temp=313k, 400k, 600k (U- turn flow configuration)

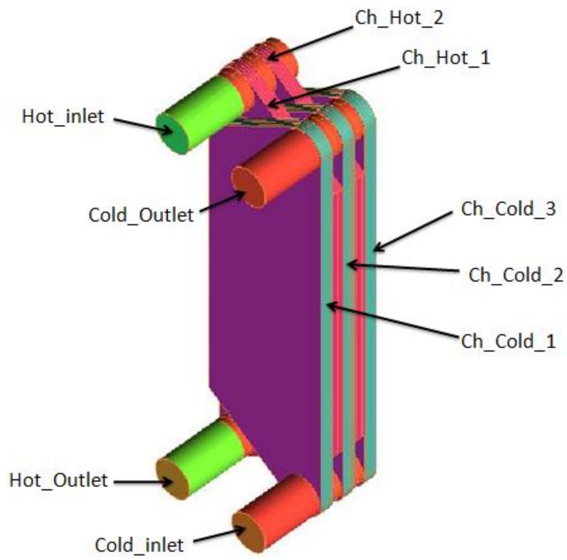
Figure 13 and 14 shows the effect of temperature on air flow distribution at different sections for U and Z-turn flow configuration at various Re. It is obvious from the figures that section 11 does not get influenced by temperature, but downstream channels experience some positive deviation with rising temperature. This feature can be attributed to bigger effect of temperature of air density variation. So it can be concluded form this analysis that flow maldistribution due to temperature is independent of flow configurations.



"Fig.14" % Mass flow rates at Re=122 and temp=313k, 400k, 600k (Z- turn flow configuration)

B. Simplified model of plate type heat exchanger:

As mentioned earlier chapter the simplified model of plate heat exchanger consists of three cold and two hot channels where surface corrugation has been neglected for the sake simplicity in simulation. The nature of flow expected in the flow domain to be either laminar or turbulent depending on inlet flow conditions and channel flow sections. In the absence of corrugated turbulence promoter, present analysis does not give actual flow features but it is capable of predicting flow behavior to some extent which is difficult to anticipate otherwise.

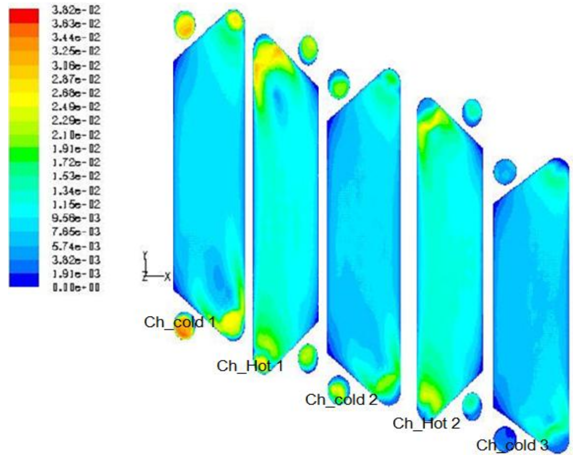


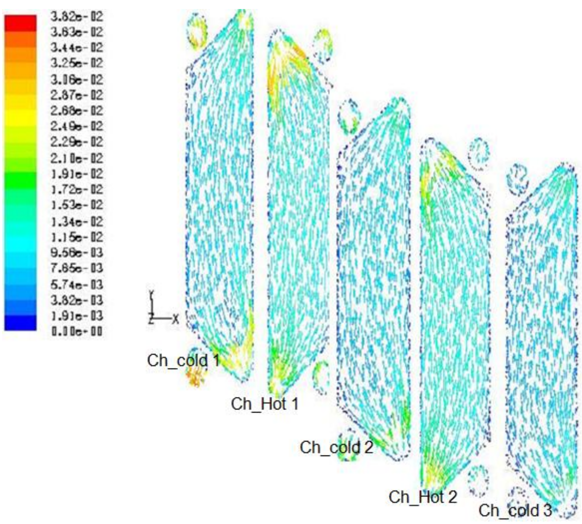
"Fig.15" Designation of channel sections for simplified model of PHE

a) Flow behaviour among channels:

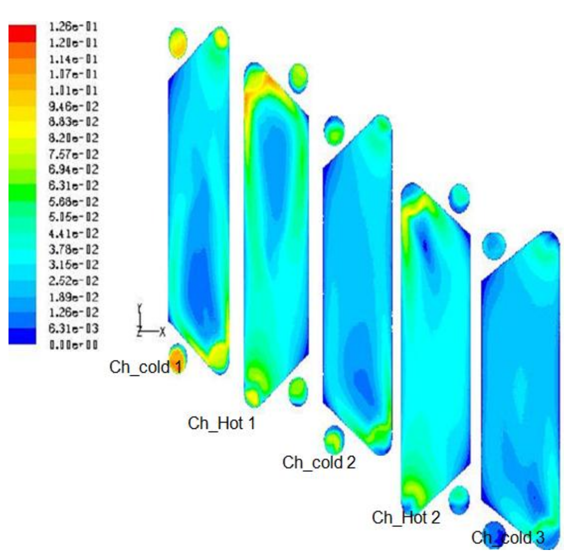
The flow behaviour among PHE channel plays a significant role in deciding their heat transfer characteristics. The simulated result is depicted in the form of contour and vector plots at Reynolds number considered for the investigation.

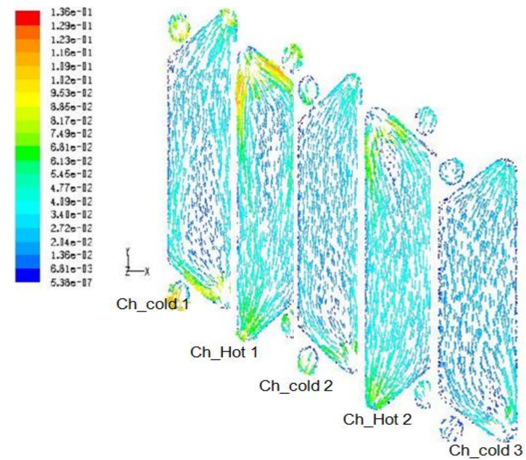
"Fig.16" (a), (b) and (c) shows the velocity contours and vectors for different channels section at Re=1500, 5000 and 10000 respectively. For cold and hot channels, it can be observed from the plots that at low Reynolds number i.e. 1500, stagnant zone appears at top and bottom left corners and it becomes stronger in downstream channels. With increasing Re, left bottom corner velocity stagnation disappears but at top right corner it still exists and become stronger in downstream side. Another important feature which these plots reveal is the existence of another velocity stagnation zone which appears just at entry port of cold channels. At low Re the domain of influence of this stagnation zone shrinks towards downstream sides of channel. But it becomes bigger with increasing Re values. These features can be attributed to the nature of flow geometry and increasing inertial effect at higher Re for existence of stagnation zone at top and bottom left corners. Entry side stagnation zone appears due to generation of vortex which becomes stronger with Re. All these changing features occurring in downstream flow channels probably leads to flow maldistribution. Percentage variation of mass flow rate is several downstream channels is depicted and described in following section.



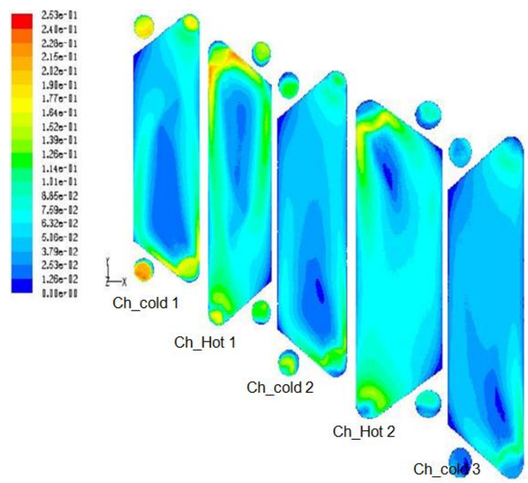


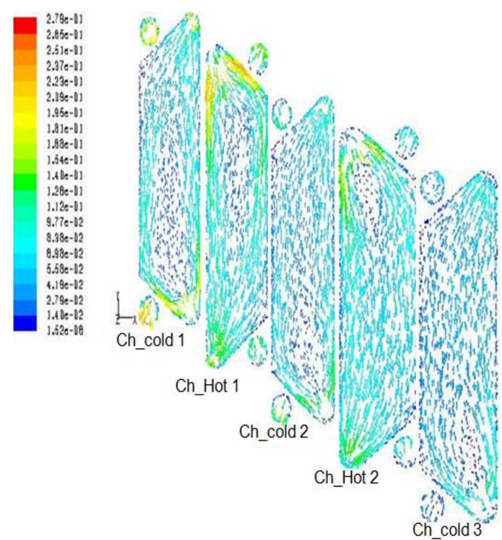
"Fig.16" (a) Velocity contours and vectors at Re=1500



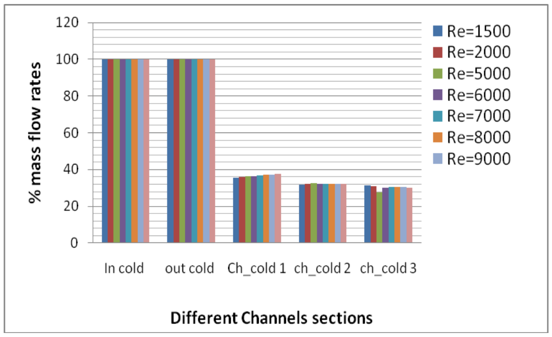


"Fig.16" (b) Velocity contours and vectors at Re=5000

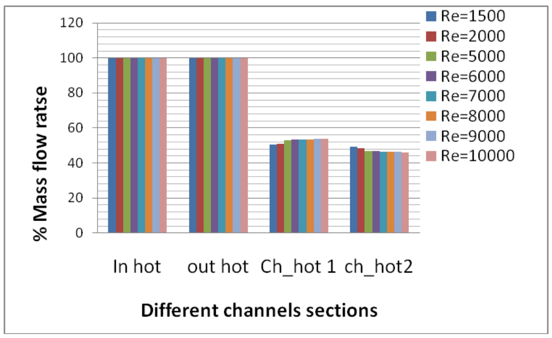




"Fig.16" (c) Velocity contours and vectors at Re=10000



"Fig.17" Percentage mass flow rates variations in different cold channels.

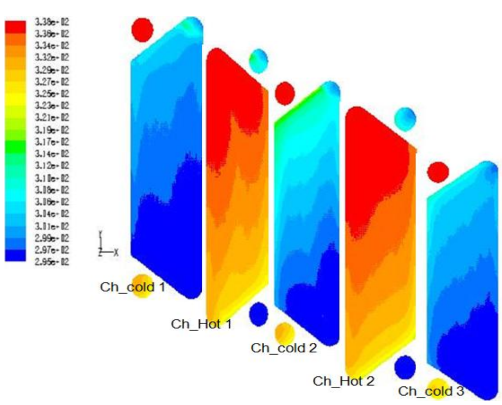


"Fig.18" Percentage mass flow rates variations in different hot channels.

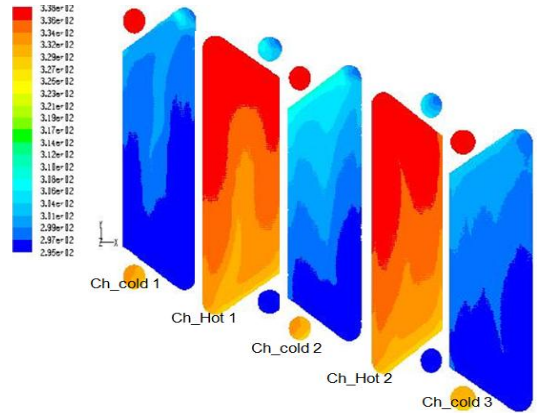
It can be observed from "Fig. 17" and "Fig.18" that at a given Re, the percentage mass flow rate is decreasing in downstream side. This difference can be attributed to increasing order of vortex occurring just after the entry of flow in channels from the port which goes on reducing in downstream due to weaker energy potential. At a given channel, percentage mass entering upstream channel is increasing and at downstream channels it decreases. This variation can be concluded form the velocity pattern observed earlier.

b) Thermal behaviour among channels:

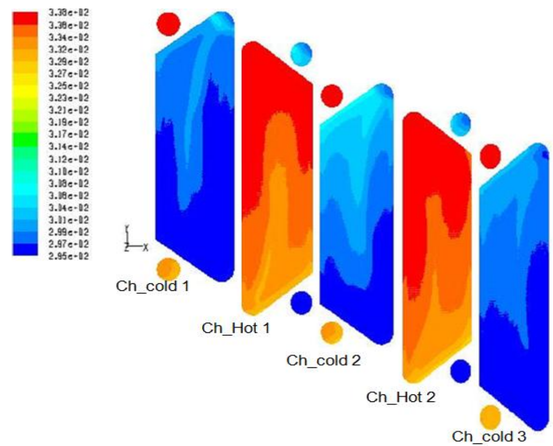
To study the thermal behavior of simplified model of plate heat exchanger the hot fluid is assumed to be at 338 K and cold fluid to be at 295K.



(a) Re=1500



(b) Re= 5000



(c) Re=8000

"Fig.19" Static temperature contours at different channels sections.

Qualitative results in the form of developed static temperature contours are shown in Fig. 19 at the Re considered for the investigation. In a channel the variation temperature can be seen because of heat exchange between hot and cold fluid. At downstream side temperature distribution is more uniform compared to upstream side. Since present simplified model of PHE consist of 5 channels, a better heat transfer can be observed from cold channel 2. Other two cold channels i.e. channel 1 and 3, are effected by surrounding boundary condition, therefore the temperature distribution in them is not as good as of channel 2.

V. CONCLUSION

The following conclusions can be drawn from the present analysis.

- Flow maldistribution is an important parameter which needs to be accounted while designing a device where distribution a flow is involved.
- Flow maldistribution is a function of Reynolds number and thermal condition of wall.

- Effect of flow maldistribution is more severe in Z-turn flow configuration compared to U-turn flow configuration.
- In U-turn flow configuration, downstream channels receive lesser flow compared to upstream channels but Z-turn flow configuration show opposite trends.
- In the simplified PHE model at a given Re, the percentage mass entering upstream channel is increasing and at downstream channels it decreases this is because increasing in order of vortex occurring just after the entry of flow in channels from the port.
- In the simplified PHE model, at low Re the domain of influence of this stagnation zone shrinks towards downstream sides of channel but it becomes bigger with increasing Re values. Entry side stagnation zone appears due to generation of vortex which becomes stronger with Re.

NOMENCLATURE

C_p, C_v	Heat capacity at constant pressure, volume (J/kg)
Ch	channel sections
L	length(m)
m	Mass (kg)
Re	Reynolds number
T	Temperature(K)
V	Velocity (m/sec)
PHE	Plate Heat Exchanger

Greek symbols

μ	Dynamic viscosity (Pa-s)
ρ	Density (kg/m^3)

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