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CFD Simulation for Manifold with Taperedlongitudinal Section

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Abstract- **The flow distribution highly depends on the header shape and the total flow rate. Therefore it is important to design manifold able to guarantee satisfactory flow equipartition within the manifold at different flow rates. In this study, numerical models were employed to study the uniformity of the flow distribution from manifold with various configurations. The numerical model consists of manifold with uniform longitudinal section having five laterals formsharp-edged junctions at right angles to the manifold axis.A modified manifold design with tapered longitudinal section was employed and the flow and pressure distributionswere investigated using the CFD model. Different inlet flows were tested for two manifold configurations and the values of these flows are (500, 750 and 1000 L/min). The results show that the manifold with tapered longitudinal section gives a relatively equal distribution compared with the manifold with uniform longitudinal section**

Keywords- **Computational fluiddynamics, Manifold, Distribution manifold, uniformity flow.**

I. INTRODUCTION

Flow in manifold is of great importance in many industrial processes when it is necessary to distribute a large fluid stream into several parallel streams and then to collect them into one discharge stream. The applications in which manifolds play a major role extend from traditional situations such as municipal water distribution systems and automobile engines to very recent, high-technology devices such as microchannel heat sinks and critical biological systems such as blood circulation in the human body [1]. Manifolds can usually be categorized into one of the following types [2-5]: dividing, combining, parallel, and reverse flow manifolds .Parallel and reverse flow manifolds

are those which combine dividing and combining flow manifolds and are most commonly used in plate heat exchangers. In a parallel flow manifold, the flow directions in dividing and combining flow headers are the same which is generally referred as a Z-manifold. In a reverse flow manifold, the flow directions are opposite is referred as a U-manifold.

A uniform flow distribution requirement is a very common issue in many engineering applications because it has significant influence on the performance of fluidic devices such as plate-type heat exchangers, a variety of piping system, heat sinks for cooling of electronic devices, fuel cells, chemical reactors, solar thermal collectors, flow distribution systems in treatment plant, the piping system of pumping stations. Therefore, for most applications, the goal of manifold design is to achieve the uniform flow distribution through all of the outlets. A great number experimental, analytical and numerical studies deal with flow distribution in manifold. Bajura [6] developed the first general theoretical model for investigation of the performance of single-phase flow distribution for both intake and exhaust manifolds. Primary emphasis is placed on configurations in which the lateral tubes form sharp-edged junctions at right angles to themanifold axis. A mathematical model was formulated in terms of a momentum balance along the manifold. Bajura and Jones [7] extended the previous model and prediction for the flow rates and the pressures in the headers for the dividing, combining, reverse and parallel manifold configurations.. Majumdar [8] developed a mathematical model with one-dimensional elliptic solution procedure for predicting flows in dividing and combining flow manifolds. The mathematical model has been further used by Datta and Majumdar [9] for numerically investigation of flow distribution in parallel and reverse flow manifolds. In both the studies, the authors have found two non-dimensional parameters (area ratio and friction parameter) which affect the flow distribution. Pigford et al. [10] studied analytically and experimentally the flow distribution in parallel and reverse flow manifolds using air. They found that, for the same geometrical and operating conditions, reverse flow manifold provides more uniform flow distribution as compared to that in a parallel flow manifold. Mueller and Chiou [11] presented in a review article the factors influencing maldistribution in heat exchangers.

The flow distribution from manifold has become of interest in predicting the heat transfer performance of compact heat exchangers. Often, flow rates through the channels are not uniform and in extreme cases, there is almost no flow through some of them, which result in a poor heat exchange performance. Choi et al. [12] studied numerically the effect of the area ratio on the flow distribution in manifolds of a liquid cooling module for electronic packaging. Results showed that the flow rate in the last channel was 2.75 times that in the first channel. It is concluded that the area ratio is one of the most important parameters affecting the coolant distribution and should be carefully examined in the design of a liquid cooling module.Kim et al. [13] numerically investigated the effects of header shapes and the Reynolds number on the flow distribution in a parallel flow manifold of a liquid cooling module for electronic packaging, 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. Anjun et al. [14] investigated experimentally the effect of inlet header angle and mass flow rate on flow maldistribution to optimize the design of plate fin heat exchanger. The result showed that optimum performance can be obtained for an inlet angle of 45˚.V.V. Dharaiya, et al. [15] studied numerically the effect a tapered header configuration to reduce flowmaldistribution in minichannels and microchannels. Jiao et al. [16] investigated experimentally the effect of the inlet pipe diameter, the first header's diameter of equivalent area and the second header's diameter of equivalent area on the flow maldistribution in plate-fin heat exchanger. For reducing flow maldistribution in the header, the authors were suggested modified header configuration. They installed a second header (B or C) after the first header to study the flow velocity distribution. Tonomura et al. [17] developed a computational model to optimize the rectangular manifold. Simulation results show that longer branched channels enable fluid to be distributed equally into each channel. Also demonstrated is the fact that the magnification of the outlet manifold area makes the flow distribution uniform.. Wen et al. [18] investigated flow characteristics in the entrance region of plate-fin heat exchanger by means of particle image velocimetry (PIV). The experimental results indicate that flow maldistribution in the conventional header is very serious, while the improved header configuration with punched baffle can effectively improve the uniformity.

Tong et al. [19] investigated numerically number of the strategies capable of perfecting manifold design to achieve the same rate of mass outflow through each of the exit ports of a distribution manifold. The results show that are most effective for the attainment of the goal of outflow uniformity, These are: (a) enlargement of the cross-sectional area of the distribution manifold, (b) variation of the cross-sectional areas of the outflow channels, (c) linear tapering of the cross-sectional area of the distribution manifold, and (d) non-linear tapering of the cross-sectional area of the manifold by means of quarter-elliptical contouring of the manifold wall. Pan, Zeng, et al. [20] performed a three-dimensional computational fluid dynamics (CFD) model to calculate the velocity distribution among multiple parallel microchannels with triangle manifolds. The simulation results showed that the velocity distribution became more uniform with larger microchannel length, depth or smaller width. Andrew and Sparrow [21] present a method to investigate the effect of the geometric shape of the exit ports on mass flow rate uniformity effusing from a distribution manifold. The results of the per-port mass effusion normalized to the average mass-flow rate of the manifold demonstrate that the single continuous slot provides the best performance. Tong et al. [22] applied a logic-based systematic method of designing manifold systems to achieve flow rate uniformity among the channels that interconnect a distribution manifold and a collection manifold. The method was based on tailoring the flow resistance of the individual channels to achieve equal pressure drops for all the channels.Hanfei Tuo and Pega Hrnjak [23] investigation an experimentally and numerically the flow maldistribution caused by the pressure drop in headers and its impact on the performance of a microchannel evaporator with horizontal headers and vertically oriented tubes. Experimental results show that the flash gas bypass method almost eliminates the quality induced maldistribution.

In general, all previous studies for manifolds with different application have show that typical manifold design does not give a uniform flow distribution at outlets. Therefore, the objective of the study was to predict the flow distribution through each outlet manifold with uniform longitudinal section and to develop an optimized manifold with tapered longitudinal sectiondesign having a better flow distribution through outlets.

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II. Problem Formulation And Numerical Simulation

A schematic diagram illustrating the two manifolds configuration system being considered is presented in Figure 1. The diagram depicts a two-dimensional array of five parallel laterals form sharp-edged junctions at right angles to the manifold axis. As seen in the figure, the distribution manifold is fed from one end with fluid and the fluid discharge to atmospheric pressure at five outlets. The previous studies with different applications have shown that traditional manifold design does not give a uniform flow distribution between outlets. It may be expected that in the geometry pictured in Figure 1a. , there will be a nonuniform mass flow distribution such that the smallest mass flowrate will occur in the outlet closest to the inlet and the highest flowrate will be encountered in the outlet farthest from the inlet. Correspondingly, the end-to-end pressure drops in the respective laterals will also be nonuniform. To counteract this problem [18], it appears logical to taper the manifold so that its cross-sectional area decreases in the streamwise direction. Because the flow in the distribution manifold is gradually depleted as mass is extracted at each outlet.If the cross-sectional area of the distribution manifold were to be constant all along the length of the manifold, the axial momentum would gradually decrease. This momentum decrease would give rise to an increase in the static pressure. Such an increase in static pressure should favor a higher efflux through the downstream outlets. The taper of manifold considers as a reasonable solution where cross-sectional area decreases in the streamwise direction. Therefore, we suggested a new design for manifold geometry, which is the manifold with tapered longitudinal section. The geometry of manifold with tapered longitudinal section with modifications in the dead end diameter is shown in Fig. 1b. The diameter ratio (D_1/D_2) is varied parametrically to estimate the optimal tapered distribution manifold. The manifold with tapered longitudinal section was comparable to those of the manifold with uniform longitudinalcross-section.Table 1 shows the models dimension of the two manifolds and Table 2 shows the diameter ratio (D_1/D_2) of the tapered manifold.

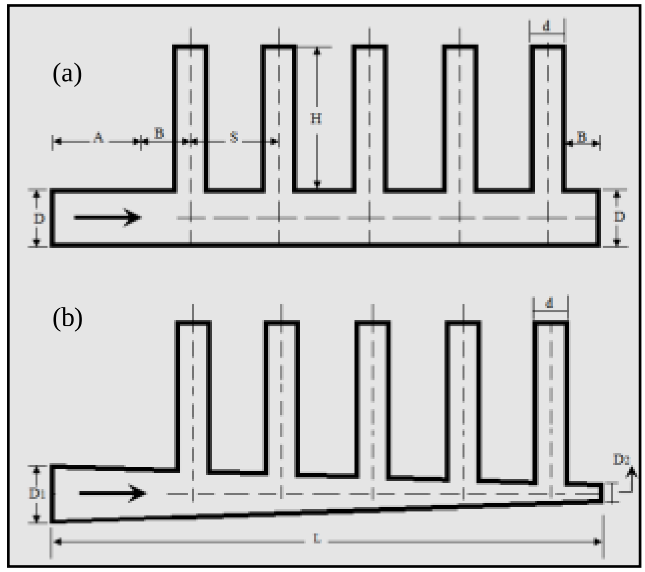

Figure 1. Geometric depiction of work (a) manifold with uniform longitudinal section, (b) for manifold with tapered longitudinal section

Table 1
Dimensions For Manifold With Uniform Longitudinal Section

Length of manifold, cm	127.0
Diameter of main pipe, cm	10.16
Diameter of lateral pipe, cm	5.080
Length of lateral pipe, cm	25.00
Developing length, cm	340.0
Distance after last lateral, cm	[illegible]

TABLE 2
Parametrically Varying The Diameter Ratio Of The Manifold With Tapered Longitudinal Section

Case	**1**	**2**	**3**	**5**	**4**	**6**
***D_1* (in)**	4	4	4	4	4	4
***D_2* (in)**	3.6	3.2	2.8	2.4	2.0	1.6
D_1/D_2	1.11	1.25	1.42	1.6	2	2.5

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III. GOVERNING EQUATIONS

In the present problem, the fluid flow is three-dimensional; that is, all three possible velocity components (x, y, and z) exist, and all three components depend on the three coordinates of Cartesian geometry. The statement of the governing equations for the fluid flow being considered here amounts to writing a set of four partial differential equations.

Conservation of mass:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \quad --- (1)$$

x-momentum

$$P\left[\frac{\partial}{\partial x}(u^2)+\frac{\partial}{\partial y}(uv)+\frac{\partial}{\partial z}(uw)\right] = -\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial u}{\partial z}\right) --- (2)$$

y- momentum

$$P\left[\frac{\partial}{\partial x}(vu)+\frac{\partial}{\partial y}(v^2)+\frac{\partial}{\partial z}(vw)\right] = -\frac{\partial p}{\partial y}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial v}{\partial z}\right) --- (3)$$

z-momentum

$$P\left[\frac{\partial}{\partial x}(wu)+\frac{\partial}{\partial y}(wv)+\frac{\partial}{\partial z}(w^2)\right] = -\frac{\partial p}{\partial z}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial w}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial w}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial z}\right) --- (4)$$

Where u, v, w are the velocity components in three dimensions respectively, ρ is the fluid density and the effective viscosity, μ_{eff}, is defined as$\mu_{eff}=\mu+\mu_t$. The turbulent viscosity is depends on the selected turbulence model as well as on the specific application. in the present study the Realizable k–e model was chosen for application here[21][24]

The simulation of two geometries was done using acommercial CFD software FLUENT. The design, meshing andboundary definition of the geometries were done using thepresolver software, GAMBIT. Tet/Hybrid T-grid scheme wasused for the mesh generation [15].

The grid elements in each geometrical model were approximately 1,000,000 elements.. Grid independence test was carried out to determine the best mesh spacing for the geometrical model. The solutions are considered to be converged when all of the residuals for the continuity and momentum equa-tions are less than or equal to 10^{-6}.The boundary condition used for the simulation are shown in table 3.

Table 3
Boundary Condition For Two Manifolds

	Test,1	Test,2	Test,3
Reynolds number	$10x10^4$	$15x10^4$	$20x10^4$
Inlet vol. rate L/m	500	750	1000
Inlet water temp.˚C	20	20	20
Outlet gage pressure	Zero	Zero	Zero

IV. THE NON-UNIFORMITY FLOW COEFFICIENT, Φ

To estimate the flow distribution among the parallel tubes, the dimensionless variables, β_i and **Φ**, are used to evaluate the flow distribution. The definition is given below

$$\beta_i=\frac{Q_i}{Q} \quad \dots\dots\dots (5)$$

Where β_i represents the flow ratio for the i^{th} outlet, Qi indicates the volume flow rate for the i^{th} outlet (L/min), and Q is the total flow rate (L/min) of the manifold. The concept of standard deviation was used by Chiou [26] to define the non-uniformity, Φ, and characterize the influence of flow ratios in the system:

$$\Phi=\sqrt{\frac{\sum_{i=1}^{n}\left(\beta_i-\bar{\beta}\right)^2}{N}} \quad \dots\dots (6)$$

Where*N* is the total number of outlet in the manifold and $\bar{\beta}$is the average flow ratio for the total number of outlets and can be calculated by using the following equation

$$\bar{\beta}=\frac{\sum_{i=1}^{n}\beta_i}{N} \quad \dots\dots\dots (7)$$

A larger value of Φ indicates decreased uniformity. For this reason, to design a manifold with a uniform volumetric

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flow rate, Φ must be minimized by using optimal values of manifold.

V. Results And Discussions

A numerical model was developed in this study to:

1. Determine the flow distribution and pressure drop among the parallel pipe.
2. Find the optimum design for tapered distribution manifold, this achieve through varied the diameter ratio (D_1/D_2) parametrically.
3. Compare the result of the manifold with uniform longitudinal section with the result obtained from manifold with taper longitudinal section model.

CFD simulation was first performed on manifold with uniform longitudinal section model having circular of diameters 10.16 cm (4 in), and straight flow with outlet of constant cross-sectional areas. The axial momentum would progressively decrease. This would give rise to the static pressure from the entrance to the manifold dead end. Such an increase in static pressure should favour a higher efflux through the downstream outflows. Figure 2 represents the static pressure contour for circular cross-section header (D=10.16 cm) with Reynolds number (Re=150,000).). It can be clearly seen from Fig. 5 that the pressure along the manifold is increasing which resulting in non-uniformityflow.

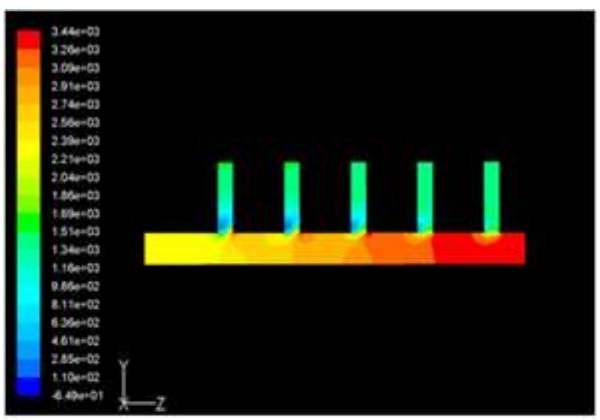


Figure 2. Pressure contour for flow in manifold with uniform longitudinal section

Figure 3 show the flow distribution plots for manifold with uniform longitudinal section (diameter 10.16 cm) for three value of Reynolds number (100,000, 150,000, 200,000). The flow through the first outlet was found to be very small compared with the last outlet as shown by pressure contours (Fig. 2).

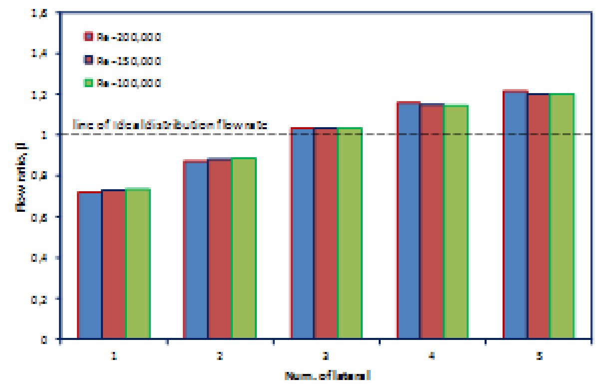


Figure 3. Flow distribution plot for manifold with uniformlongitudinal section (Re = 100,000, 150,000 and 200,000).

From the figure 4, the value of non-uniformities (Φ) of tapered manifold are 0.035, 0.0226, 0.0222, 0.0201, 0.019, 0.014 and 0.0182 at diameter ratio of 1, 1.1, 1.25, 1.42, 1.66, 2, and 2.5, respectively. The (Φ) values are lower for the manifold with circular cross-section and the corresponding value of 0.0345 as shown in Figure 4. Then the dead end diameter is reduced gradually from 10.16 cm (4 inch) to 5.08 cm (2 inch), the flow distribution is generally improved. The non-uniformities (Φ) decreases until it reaches a minimum value (optimal design) then it start to increase although the diameter ratio was increased too as shown in figure.The results of flow ratio for the tapered distribution manifold at different diameter ratio are given in Figure. 5.

Table 4 shows the non-uniformity coefficient for circular and tapered manifold cross-sectionsof three values of the Reynolds number, 100,000, 150,000, and 200,000. It can be seen that the flow distribution was severe in case manifold of circular cross-section. The flow was more evenly distributed for the manifold with tapered cross-section.

TABLE 4
THE NON-UNIFORMITY FLOW (Φ) FOR TWO MANIFOLDS

Manifold cross section	The non-uniformity flow coeff. (Φ)	
	Circular	Taper
Re=100,000	0.0367	0.0142
Re=150,000	0.0345	0.0140
Re=200,000	0.0340	0.0139

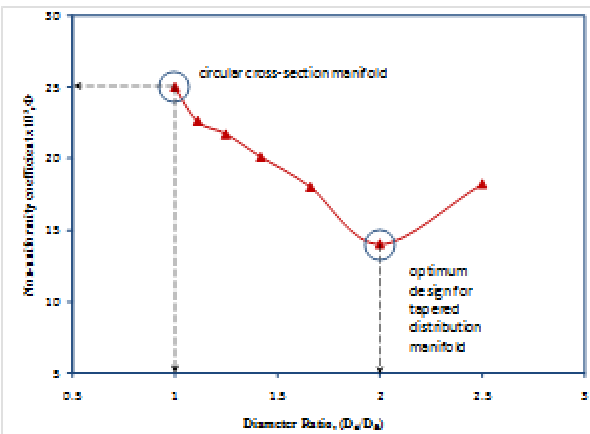


Figure 4. The non uniformity coefficient (Φ) for different D_1/D_2

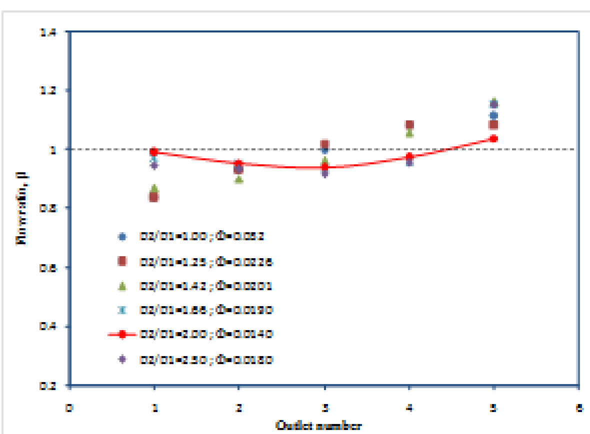


Figure 5. The flow ratio (β) for different D_1/D_2

From the results shown figures 4 and 5 the optimum configuration of distribution manifold can bedetermine using diameter ratio (D_1/D_2) of (2 inch). Figure 6 shows the pressure contour for tapered distribution manifold. The pressure along the manifold was found to be nearly uniform which resulted in a better flow distribution through the outlets.

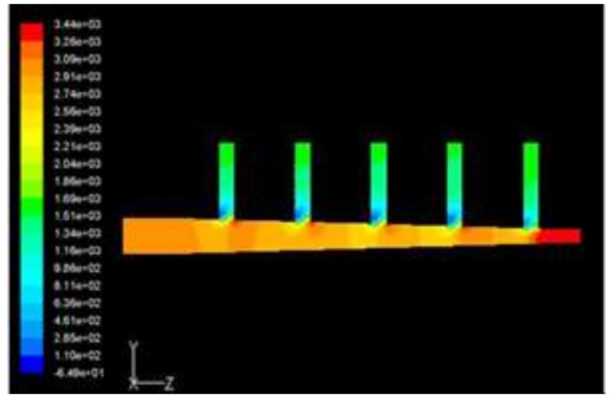


Figure 6. Pressure contour for flow in manifold with tapered longitudinal section.

Figure 7 represent flow distribution from manifold with tapered longitudinal section having inlet diameters of 10.16 cm (4 inch) and dead diameter of 5.08 cm (2 in). The improvement of flow distribution through the outlets is compared to that obtain from circular cross-section manifold as shown in Figure. 7.

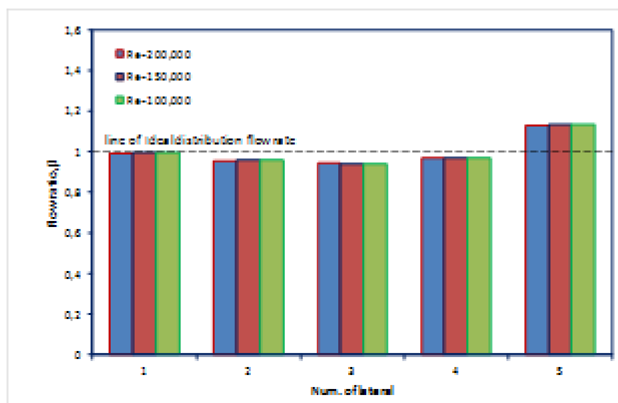
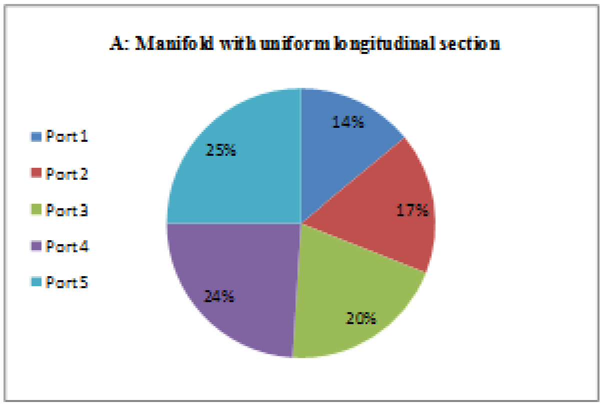


Figure 7: Flow distribution plot for manifold with tapered longitudinal section (Re = 100,000, 150,000 and 200,000).

Figure11(A and B) show that the flowrate fraction of each outlet (which is the rates of outlet to the total flow rate in the manifold).

For circular cross-section manifold, the result shows a nonuniform flow smallest flowrate occurred in found the outlet closest to manifold inlet and the highest flowrate found in the last manifold outlet. Let the respective outlets be numbered as (1) which is the first outlet while the last is outlet (5).The discharge from outlet (1) is lower by 44% than theoutlet (5). While for the tapered cross-section manifold,the percentage is reduced from 44% to 13%.



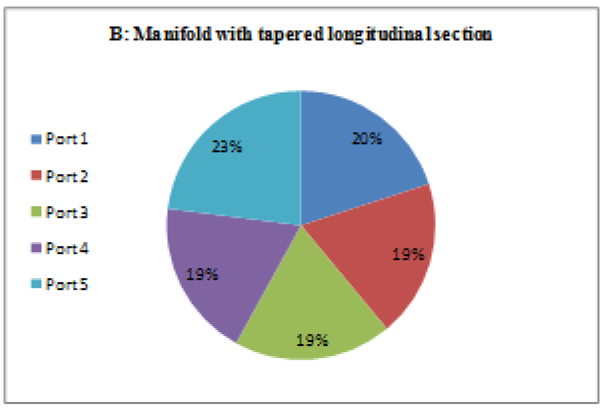


Figure 11. Flowrate fraction Percentage for two manifold configurations (Re=150,000, (Q=750 L/m)

VI. CONCLUSIONS

The goal of this investigation has been to evaluation of a tapered header configuration to achieve the same rate of mass outflow through each of the exit ports of a distribution manifold. The CFD simulation and experimental data for different header configurations namely; circular and tapered cross-section headers were carried out.

Severe maldistribution was found for theheader with Circular cross-section whereas the flow through the channels was nearly uniform in the case of tapered header configuration a numerical model was used to predict the flow across each lateral for three different Reynolds numbers (i.e., 100,000, 150,000 and 200,000) and the results were found to have the same trend as compared to those obtained from experimental data. the flow distribution of the outlet flow rates are independent of the Reynolds number because of the Reynolds number was found to have only a slight effect on the uniformity of the mass effusion from the outlet.

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