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Study surface roughness on overall performance of parallel flow microchannel heat exchanger

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Abstract

The aim of this paper is to investigate numerically the surface roughness effects on the overall performance of parallel flow microchannel heat exchanger (PFMCHE) with different hydraulic diameters. Water has been used as working fluid flow velocities in one direction for hot and cold channels with square shape. Surface roughness considered in this study has been choiced as ratio with equal 0.14 of surface roughness to hydraulic diameter. The obtained results showed that, the pumping power increased with increasing effect of surface roughness and decreasing hydraulic diameter, also the thermal performance increased slightly with increasing surface roughness.

Keywords: surface roughness, Microchannel heat exchanger, Laminar flow, Numerical investigation, Thermal performance .

NOMENCLATURE:

PFMCHE parallel flow microchannel heat exchanger

C_p specific heat (J\kg.K)

D_h hydraulic diameter (m)

f friction factor

H height (m)

k height of the roughness element (m)

$\dot{m}$ mass flow rate (kg/s)

P total pressure (Pa)

P.P pumping power

T temperature (K)

U_k velocity at the top of roughness (m/s)

u_{avg} average velocity (m\s)

u fluid x-component velocity (m/s)

v fluid y-component velocity (m/s)

w fluid z-component velocity (m/s)

Re Reynold number

Re_k local Reynold number

RVM roughness viscosity model

t thickness of separating wall

W width (m)

x axial coordinate (m)

y vertical coordinate (m)

Z horizontal coordinate (m)

ΔP pressure drop across heat exchanger (Pa)

<u>Greek Symbols:</u>

ρ density (kg/m^3)

μ_R roughness viscosity (m^2/s)

μp apparent viscosity (m^2/s)

ε Heat exchanger effectiveness

η^* overall performance

<u>Subscripts</u>:

i inlet

o outlet

c cold

h hot

s fluid solid interface

1. Introduction

In recent years, the researches in the field of thermal – hydraulic at micro scale level have been increased constantly due to rapid growth of technology applications which required to transfer high heat rates in relatively small space and volume. Such applications spread as cooling system for computers CPUS, compact heat exchangers and microfluidic devices [1].The effect of surface roughness doesn't influence on the laminar regime of flow in ducts with conventional size, but in the micro size the surface roughness has high effect on thermal and hydraulic performance. The present work studying interesting the problem of surface roughness in (PFMCHE). There are many researchers in literature studied many parameters affecting the performance of (PFMCHE) including roughness effect. Below a review of most related papers.

Wu and Cheng (2003) [2] investigated experimentally the laminar flow to study the pressure drop and convective heat transfer of water in 13 channels have trapezoidal shape. They found in their results that, the Nusslet number and apparent friction factor depended on different geometric parameters where With increasing surface roughness in laminar flow Nusselt number increased while apparent friction is constant. The experimental results showed that, the Nusselt number increased almost linearly with Reynold number especially at low Reynold number ($Re \geq 100$). In their article, the authors found a new dimensionless correlations for apparent friction constant and Nusselt number of water flow in micro channels having different geometric parameters. There are an experimental study presented by using water flows in smooth and rough rectangular microchannel. They used experimental apparatuses enabled to explore a wide range of length (7mm to 300 μm) and Reynolds number (0.01 to 8000). They measured the local friction factor by using strain gages. They observed in their results that, the surface roughness induce decreasing hydraulic performance due to the increasing in friction factor with increasing surface roughness**[3]**. Also presented a numerical investigation to study the performances of parallel flow heat exchanger with rectangular ducts. The results obtained that, overall heat transfer coefficient is rapidly changed for $x / D_h Pe$ (Graetz number) below 0.03, therefor the assumption of constant overall heat transfer coefficient is not valid if the Graetz number based on heat exchanger length is of order of 0.03**[4]**. From literature studies discussed experimentally the water flows in rectangular microchannel. The roughness elements are saw-tooth in structure with height element are (107 - 117)μm with Reynold number varying from 210 to 2400. The pressure measurements are taken in sixty locations along the flow length to determine the local pressure gradients. In their result the friction factor and transition to turbulent flow are obtained and compared with other experiment paper, also they found that the critical Reynold number decreased with increasing relative roughness**[5]**. The effect of surface roughness on characteristics of flow studied experimentally , the author used glass and silicon micro tubes treated as smooth surface. The stainless steel micro tubes have (3% - 4%) relative roughness have to be treated as roughness surface. They concluded from experimental results in smooth glass and silicon that, the product of Darcy friction factor and (Re) remain approximately 64 which same value in macro tube. In rough tubes (stainless steel) the product of friction factor with Re is (15 – 37)% higher than 64**[6]**. the effectiveness of counter flow microchannel heat exchanger in isosceles right triangular microchannel with different values of Reynold number studied numerically. They used water as working fluid for both hot and cold fluids. By using fluent 16.0 to solve continuum, energy and momentum equations ,then they calculated for different Reynold number thermal and hydraulic performances. The results obtained show that, with increasing Reynold number decreased effectiveness, friction factor and performance index of heat exchanger where pressure drop and heat transfer increased as Reynold number increase **[7]**.

In this paper the effect of the surface roughness on the performance of parallel flow microchannel heat exchanger will be studied numerically by connecting the viscosity in roughness .In addition to study the effect of hydraulic diameter changing on the overall performance of (PFMCHE).

2. problem description :

The configuration shown in fig.1 explains physical model of parallel flow microchannel heat exchanger which has square shape. Due to thermal and geometric symmetry and for defaulting solving complete heat exchanger therefor will be selected one unit to studying in this work. Fig.2 shows one unit cell selected for this study which consists of two channels and separating wall between them with water as working fluid with aspect ratio is one and length is 500micro. the water enters in upper channel as cold fluid in uniform velocity u_{ci} and uniform temperature T_{ci} and the lower channel enters the hot water in uniform velocity u_{hi} and temperature T_{hi}

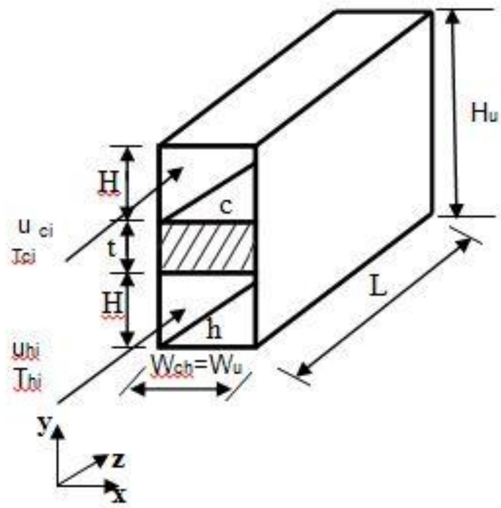


Fig.2 unit cell of (PFMCHE)

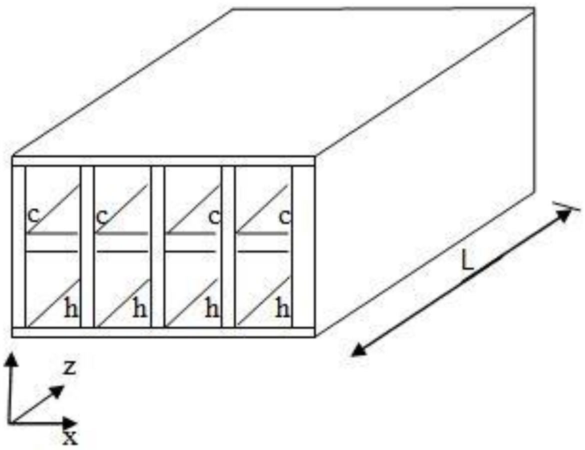


Fig.1 physical model

2.1 mathematical formulation **:**

The governing equations used in this paper based on the following physical and geometry assumptions :

- The flow is laminar and steady.
- The value of Knudsen number is small than 0.001 (no slip).
- Water used in this paper is incompressible, constant properties and Newtonian.
- There is no heat transfer to\from the ambient
- The energy dissipation is negligible.
- The pressure drop in axial direction only.
- Three dimensional the flow and heat transfer .

The governing equations and boundary conditions in Cartesian coordinate are written below [8] [9]

Continuity equation:

$$\nabla V = 0 \qquad (1)$$

Momentum equation:

$$\rho(V.\nabla V) = -\nabla P + \nabla.(\mu \nabla V) \qquad (2)$$

Energy equation:

$$\rho C_p (V \nabla T) = k \nabla^2 T \qquad (3)$$

Mala and Li [10] modified roughness viscosity model where surface roughness in laminar flow considered interms roughness – viscosity function. The apparent viscosity in momentum equations become viscosity of fluid sum roughness viscosity as equation (8) and the ratio of roughness to hydraulic diameter choice as 0.14.

$$\frac{\mu R}{\mu} = A\, Re_k \frac{r}{k}\left(1-\exp\left(-\frac{Re_k}{Re}\frac{r}{k}\right)\right)^2 \quad (4)$$

$$U_k = \left(\frac{\partial u}{\partial r}\right)_{r=k} k \quad (5)$$

$$Re_k = \left(\frac{\partial u}{\partial r}\right)_{r=R} k^2/v \quad (6)$$

$$A = 0.1306\left(\frac{R}{k}\right)^{0.3693} \exp\left\{Re\left(6\times 10^{-5}\frac{R}{k} - 0.0029\right)\right\} \quad (7)$$

The apparent viscosity of fluid is

$$\mu p = \mu + \mu R \quad (8)$$

2.2 Boundary condition

The boundary conditions used to complete the model are:

For lower channel (hot fluid)

Location	B.c		comments
At x=0	$u=u_{hi}$, $v_h = w_h = 0$	, $T_h = T_{hi}$	hot fluid in flow
At x= w	$\frac{\partial u_h}{\partial x} = v_h = w_h = 0$	, $\frac{\partial T}{\partial x} = 0$	hot fluid out flow
			(fully developed flow , End of channel)
At z =0	$u_h = v_h = w_h = 0$	, $\frac{\partial T_h}{\partial z}$	no- slip, adiabatic wall
At z = L	$u_h = v_h = w_h = 0$	, $\frac{\partial T_h}{\partial z}$	no- slip, adiabatic wall
At y = 0	$u_h = v_h = w_h = 0$	, $\frac{\partial T_h}{\partial y}$	no- slip, adiabatic wall
At y = H_h	$u_h = v_h = w_h = 0$	, $k_h \frac{\partial T_h}{\partial y} = -k_s \frac{\partial T_s}{\partial y}$,	fluid – solid interface
		$T_s = T_h$	(no –slip conjgat heat transfer)

For upper channel (cold fluid)

($H_h + t \le y \le H_h + t + Hc$)

Location	B.c		comments
At x = 0	$\frac{\partial u_c}{\partial x} = v_c = w_c = 0$	, $\frac{\partial T_c}{\partial x} = 0$	cold fluid outlet (fully developed
			Fully developed flow end of channel
At x= w	$u_c = u_{ci}$, $v_c = w_c = 0$	, $T_c = T_{ci}$	cold fluid inflow
At z = 0	$u_c = v_c = w_c = 0$	, $\frac{\partial T_c}{\partial z} = 0$	No –slip adiabatic wall
At z =L	$u_c = v_c = w_c = 0$	, $\frac{\partial T_c}{\partial z} = 0$	No –slip adiabatic wall
At y = $H_h + t$	$u_c = v_c = w_c = 0$	, $-k_c \frac{\partial T_c}{\partial y} = -k_s \frac{\partial T_s}{\partial y}$, $T_c = T_s$	Fluid –solid interface
			(no slip, conjugate heat transfer)
At y = $H_h + t + H_c$	$u_c = v_c = w_c = 0$,	$\frac{\partial T_c}{\partial y}$	(no slip adiabatic wall)

For solid wall separating two channels ($H_h \le y \le H_h + t$)			
Location	B.C		comments
At x = 0 and x=W	$\frac{\partial T_s}{\partial x} = 0$		adiabatic wall
At z=0	$\frac{\partial T_s}{\partial z} = 0$		adiabatic wall
At z=L	$\frac{\partial T_s}{\partial z} = 0$		adiabatic wall
At y= H_h	, $k_h \frac{\partial T_h}{\partial y} = -k_s \frac{\partial T_s}{\partial y}$	, $T_s = T_h$	fluid –solid interface
At y = $H_h + t$	$-k_c \frac{\partial T_c}{\partial y} = -k_s \frac{\partial T_s}{\partial y}$	, $T_c = T_s$	Fluid –solid interface

Where j= h, c for hot and cold fluids respectively.

By CFD code the above equations with their boundary conditions the values of velocity, pressure, effectiveness, pumping power, friction factor and overall performance are calculated

The inlet velocity is calculated based on Reynolds number

$$Re = \frac{\rho\, w_i D_h}{\mu}$$

Where D_h is the hydraulic diameter is 4A / p

.

From Darcy equation can be calculate friction factor [11]

$$f = \frac{\Delta p}{u_{avg}^2}\,\frac{2D}{\rho L} \quad (10)$$

And the pressure drop is the difference between the total pressure at the inlet and outlet.

$$\Delta P_u = \Delta P_{ch.h} + \Delta P_{ch.c} = (P_{h.i} - P_{h.o}) + (P_{c.i} - P_{c.o}) \quad (11)$$

Heat exchanger effectiveness is the ratio of the actual heat transfer to the maximum possible heat that can be transferred[10]:

$$q = m^{\cdot}\, C_c\,(Tc_{,out} - T_{C,in}) = m^{\cdot}\, C_h\,(Th_{,out} - T_{h,in})$$

$$\varepsilon = \frac{m^{\cdot}\, C_c\,(Tc_{,out} - T_{C,in})}{C_{min}\,(T_{h,in} - T_{c,in})} = \frac{m^{\cdot}\, C_h\,(Th_{,out} - T_{h,in})}{C_{min}\,(T_{h,in} - T_{c,in}} \quad (12)$$

For any heat exchanger the effectiveness is a function of NTU and the heat capacity ratio. ε = f (NTU, Cr)

Where NTU is the number of transfer units

$$NTU = \frac{U_A}{C_{MIN}}$$

And Cr is the heat capacity ratio

$$Cr = \frac{C_{min}}{C_{max}}$$

$$\varepsilon = \frac{1-e^{-NTU(1-Cr)}}{1-Cre^{-NTU(1-Cr)}}$$

and pumping power can be calculated from volumetric flow rate multiplying total pressure drop. [12]

$$P.P = \dot{V} \quad \Delta P \qquad (12)$$

To calculate the overall performance of heat exchanger the performance factor has been used to correlate the hydrodynamic and thermal performance as define below

$$\eta * = \frac{Q}{\Delta P} \qquad (13)$$

3. Numerical solution:

The above system of governing equations and boundary conditions are numerically solved by using finite volume method. The flow is developing, the 3D continuity and 3D Naiver stock equations are solved numerically by using ANSYS fluent. A mesh has been chosen in accepted size and a mesh refinement has been made to find out the appropriate mesh size that gives highly accurate solution as indicate in table (1).Table (1)shows that the solution becomes independent of grid size and from third mesh further increase in the grids will not have a significant effect on the solution and results of such arrangement are acceptable. Therefore the forth mesh size is used for all calculations.

Table 1 Grid independent study

Mesh size	Outlet temperature (K)
mesh1 (number of elements=1320743)	308
mesh2 (number of elements=1528356)	308.93
mesh3 (number of elements=1963380)	308.95
mesh4 (number of elements=2575633)	308.96

The convergence criteria used to control the numerical solution for both momentum and energy equations are10^{-6}.

Table 2 the properties of material [13]

Property	Material	
	Water	Aluminum
Thermal conductivity (k) (W/m^2.K)	0.643	273
Specific heat Cp (J / Kg .K)	4200	871
density(Kg / m^3)	1000	2791
Viscosity (m^2 / s)	0.001	-

4.Results and discussion:

The model was solved with pure water as a working fluid with constant properties selected according to the mean temperature across heat exchanger. Two cases have been solved for each size with considering surface roughness and with neglecting it to explore the effect of roughness on the performance of (PFMCHE)

To check the validity of the built numerical model, a verification was made by solving the experimental model presented in [14] and the results were compared. The experimental model presented in [14] is a rectangular microchannel with aspect ratio 1.92 µm. The width (265 μ m), height (510 μ m), (76 mm) and hydraulic diameter (348.8 μ m) where FCC770 as working fluid.

The surface roughness of aluminum measured was 3.7 μ m due to surface finish of machining.

Fig. 3 shows the comparison between results of present numerical model and the experimental data of [14] for variation friction factor with Reynolds number. From this figure it can be seen that, the agreement between numerical and experimental results is acceptable since the average error is 5% which may be due to difference in material properties and some assumptions in this model .

To check the behaver of thermal side, another validation has been made with reference [4] as shown in fig. 4 shows the validation between results of present model with data in Al- bakhit and Fakeri 2006[4]. The figure explains the variation of dimensionless axial distance $x^* = \frac{x}{Dh\ Re\ pr}$ with dimensionless mean temperature $T_m = \frac{T_{i} - T_{ci}}{T_{h,in} - T_{c,in}}$ for the cold and hot fluids flows in rectangular microchannel heat exchanger at Reynold number is 100 and $\frac{k_w\ D_h}{k_s\ t_s} = 100$. It can be noted in this figure a good agreement between the results of present model and that in [4] where the maximum error was 1% due to the difference in some assumptions between them

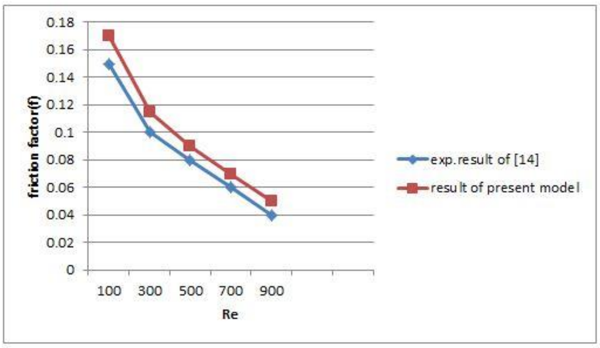
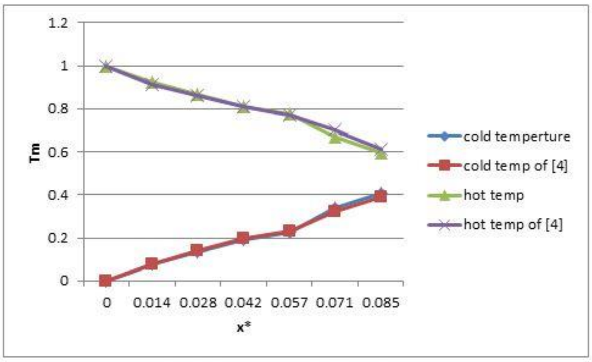
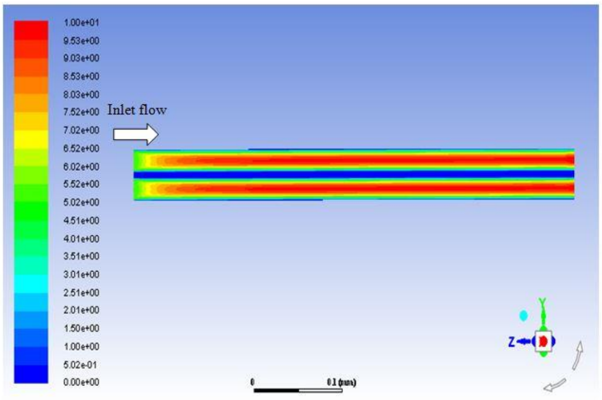


Fig (3) variation of friction factor with Re as comparison between results of present model and that of [14]



Fig(4) distribution of temperature along channel as comparison between results of present model and that of [14]

Fig.5and 6 illustrate the velocity contours at Reynold number 100 and hydraulic diameter of 20 μ m. Fig.5 shows plane drawing along of channel at center of heat exchanger (x = 0) μ m. Fig.6 explains the flow velocity at different planes distributed on along longitudinal axis. From both figures it can be seen that, maximum values of water velocity at inlet of channels and the region near inlet and this values decreased at toward wall due to increasing the effect of surface roughness and the velocity follow parabolic distribution



Fig(5) contour of velocity(m/sec) distribution along cold and hot cells of PFMCHE at Re= 100 and D_h = 20 um

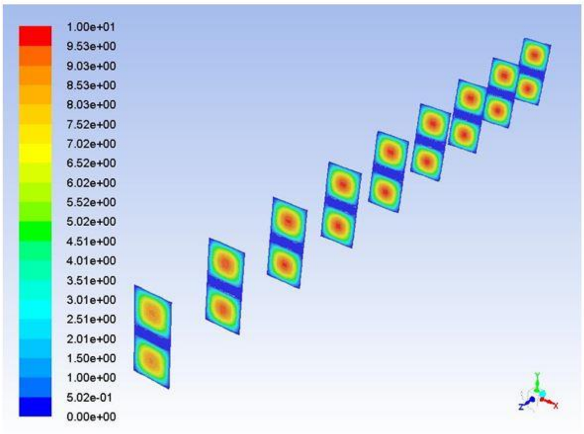
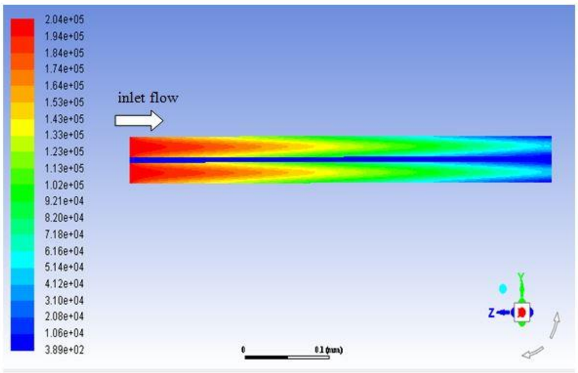
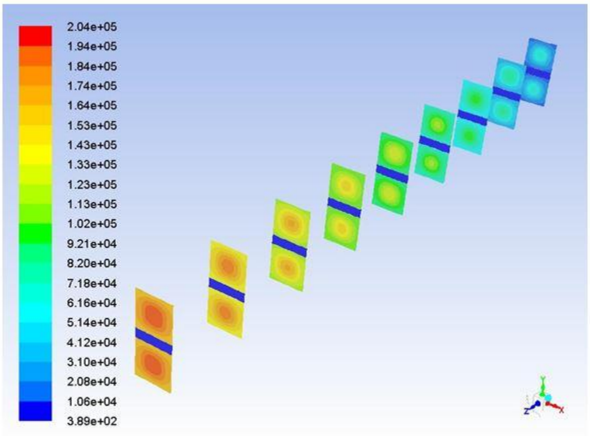


Fig (6) contours represent the velocity(m/sec) in cold and hot cells of PFMCHE at D_h =20 μm

Figures 7and8 display pressure drop contours in hot and cold channels of heat exchanger. Fig.7 represents contour plane drawing in center of channels with (y-z)axis. Fig.8 explains the pressure drop at different planes distributed on along channels of (PFMCHE).From both figeres7and8 it can be noted that, pressure drop has maximum values at inlet channel and decreased with flow toward the end of channel due to losses caused by friction factor.



Fig(7) pressure(pa) distribution contour along cold and hot channels of PFMCHE with D_h = 50 μm and Re = 100



Fig(8) contour of pressure(pa) along cold and hot cells of PFMCHE at D_h =20 μm and Re =100

Figers9and10 indicate temperature contours with square section shape at Reynold number of 100 and inlet water temperature of 303℃ in cold channel and 373℃ in hot channel. Fig.9 expresses contour plane drawing with (y-z) axis in center of channel. Fig.10 explains the temperature distribution on along of channel. From both figures. It can be seen that, increasing temperature difference and heat transfer with water flow and become maximum values in the end of channel due to heat transfer from hot fluid to cold fluid**.**

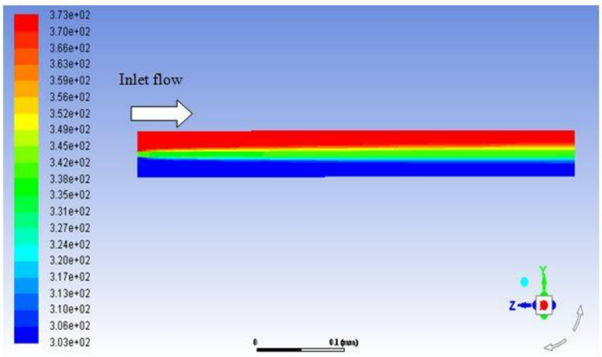
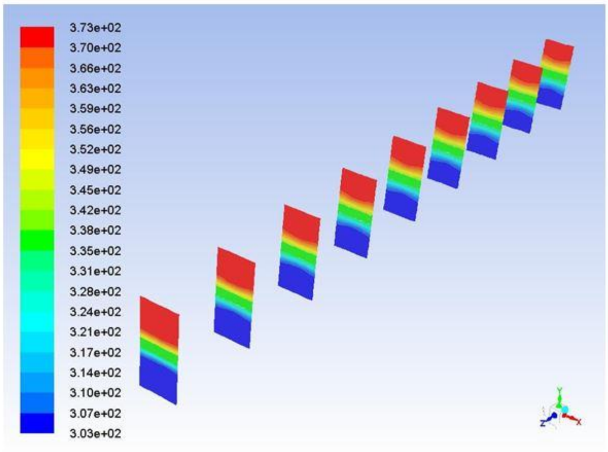


Fig (9) Temperature(k) distribution contour along hot and cold cells of PFMCHE with Re =100 and D_h = 20 μm



Fig(10) Temperature(k) contour along heat exchanger unit at D_h = 20 μm and Re =100

The relationship between effectiveness and Reynold number through hydraulic diameter selected rang (20, 50, 110, 150) μ m explained in fig.11 also studied in this figure the effect of surface roughness and neglect it. It can be noted from this figure that, increasing the effectiveness of (PFMCHE) with decreasing hydraulic diameter due to increase heat transfer rate which depend on temperature difference ($\Delta T_h = T_{hi} - T_{ho}$). Also it can be seen in fig.11 decreased the effectiveness with increasing Reynold number due to increase velocity of flow. the effect of surface roughness on effectiveness and generally thermal performance is very small due to don't direct relationship between surface roughness effect and thermal characteristics.

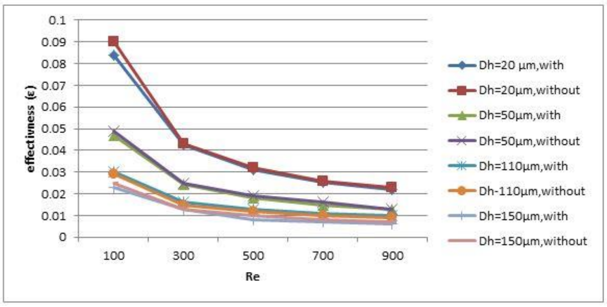


Fig (11) variation of effectiveness (ε) with Re number with and without considering roughness effect.

Fig.12 explains the variation of friction factor with Reynold number at considering surface roughness effect. It can be seen in this figure the friction factor decreased with increase Reynold number due to increasing velocity of flow according to Darcy equation

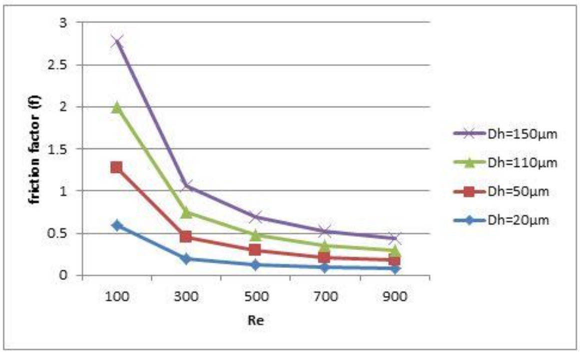
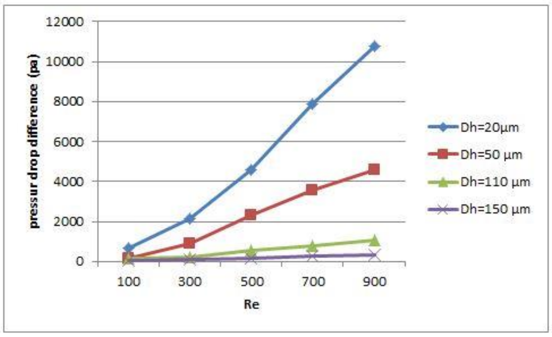


Fig (12) variation of friction factor with Re for different values of hydraulic diameter

Fig.13 illustrates the variation of pressure drop difference with Reynolds number for all selected diameter. pressure drop difference represents the difference between total pressure drop at considering surface roughness and with neglect it. It can be noted in this figure that, the pressure drop difference of (PFMCHE) increased with increasing Reynold number due to increasing the velocity which cause increasing in flow rate also it can be seen that, increasing hydraulic diameter causes decreased in pressure drop difference due to the small cross section area.



Fig(13) variation pressure drop difference with Re for different value of hydraulic diameter

fig.14 represent relationship between pumping power difference for all hydraulic diameter selected. Pumping power difference represents changing between pumping power with considering surface roughness effect and neglect it. It can be observed in this figure that, the pumping power difference increased with increase hydraulic diameter due to increase pressure drop. Also it can be seen from this figure that, Increasing Reynold number caused increased pumping power difference due to increasing velocity of water which cause increase in volumetric flow rate also increasing this effect with considering surface roughness due to the restriction between molecular of fluid because surface roughness effect as seen in fig.14.

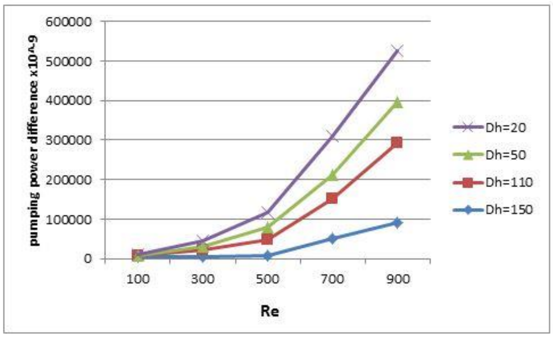
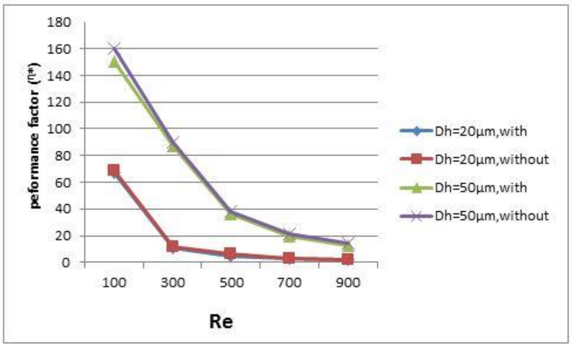
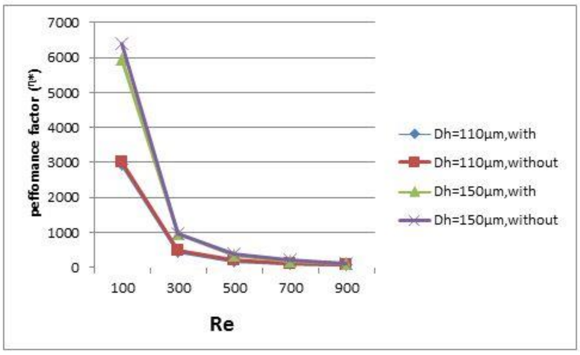


Fig (14) variation of pumping power difference with Re for different hydraulic diameter

Fig.15 shows the variation of performance factor with Reynold number for two cases, with considering surface roughness and with neglect it. Fig.15 divided to a and b to show clear difference of hydraulic diameter effect on performance factor. Fig.15.a deals with hydraulic diameters selected range (110, 150) μ m while fig.15.b deals with hydraulic diameters range (20, 50) μ m. It can be noted in both figures a and b increased performance factor of (PFMCHE) with increased hydraulic diameter due to increase flow velocity. Decreased performance factor with increase Reynold number due to the increasing in pumping power large than increase in heat transfer as seen in figures.15. a and b.



Fig(15-a) variation performance factor with Re for two cases with and without considering roughness effect for hydraulic diameter (20,50) µm



Fig(15-b) variation performance factor with Re for two cases with and without considering roughness effect for hydraulic diameter(110, 150) µm

Conclusions

In this paper numerical investigation is made to study the effect of roughness on the performance of PFMCHE with different sizes and water at constant properties as working fluid. From the results it can be conclude that

1. The surface roughness has considerable effect on hydraulic performance and effect is increased when hydraulic diameter and velocity of flow is small.
2. Surface roughness enhance the thermal performance ,but this enhancement is slight.
3. The roughness effect decreased with increasing Reynold number and hydraulic diameter.
4. The surface roughness cause extra increasing in pressure drop and pumping power.
5. In general the surface roughness lead to decrease the overall performance of PFMCHE.

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