Effect of variation in channels' area on the performance of counter flow microchannel heat exchanger

Mushtaq I. Hasan[a] , Rafid M. Hannun[a] , Abdullah K. Eqab[a]

[a] Mech. Eng. Dept.- College of Engineering – Thi-Qar University

Abstract:

In this paper the counter flow microchannel heat exchanger (CFMCHE) is numerically investigated by solving 3D Navier Stock equations in two fluids and 3D energy equation in two fluids and walls. The axial conduction in the walls and entrance region are taken into consideration. And a new technique is used by using variable cross-sectional area microchannels instead of constant area microchannels to improve the performance of this type of heat exchangers.

The variation in channels area is made by inclined only the walls separating the channels which lead to changing the height of channels and keeping its width constant. The idea of using variable cross-sectional area microchannels applied on three channels geometry (square, rectangular and trapezoidal).

The results obtained show that, using of variable cross-sectional area channels lead to improve both the hydrodynamic and thermal performance of CFMCHE since it lead to reduce the pressure drop and increase the effectiveness of heat exchanger.

تأثير التغير في مساحه القنوات على أداء المبادل الحراري المتعاكس المايكروي

م.د. مشتاق إسماعيل حسن، م. د. رافد معلك حنون، م.م. عبد الله خزعل عكاب

قسم الهندسة الميكانيكية – كلية الهندسة – جامعة ذي قار

الخلاصة:

في هذا البحث تم دراسة المبادل الحراري المتعاكس المايكروي عدديا من خلال حل معادلات نافير ستوك ثلاثيه الأبعاد في كلا المائعين وحل معادلة الطاقة ثلاثية الأبعاد في كلا المائعين بالإضافة إلى الجدران. كذلك تم اخذ كل من التوصيل المحوري خلال الجدران ومنطقة الدخول في كلا المائعين (الحار والبارد) بنظر الاعتبار خلال الحل العددي. وتم اقتراح أسلوب جديد يؤدي إلى تحسين أداء هذا المبادل وذلك باستخدام قنوات مايكرويه بمساحة مقطع متغيره على طول المبادل بدل القنوات المايكروية ثابتة المساحة.

إن تغيير مساحة المقطع للقنوات تم من خلال إمالة الجدار الفاصل بين القنوات بزاوية معينة مما يؤدي إلى تغير ارتفاع القوات فقط والحفاظ على عرض القنوات ثابت. فكرة استخدام قنوات ذات مساحة مقطع متغيرة تم تطبيقها على قنوات مايكروية بأشكال مختلفة هي (المربع، المستطيل و المعين).

النتائج التي تم الحصول عليها تشير إلى إن استخدام قنوات متغيرة المساحة بدل القنوات ثابتة المساحة يؤدي إلى زيادة كل من الأداء الهيدروديناميكي والأداء الحراري لهذا المبادل من خلال تقليل هبوط الضغط وزيادة فعالية المبادل.

Nomenclature:

Quantity	Symbol	Unit (SI)
Specific heat	C_p	J/(kg K)
Channel height	H	m
Thermal conductivity	k	W/m K
Channel length	L	m
Mass flow rate	m	kg/s
Pressure	P	Pa
Heat transfer rate	q	W
Maximum heat transfer rate	q_{max}	W
Temperature	T	K
Separating wall thickness	t	m
Fluid x-component velocity	u	m/s
Fluid y-component velocity	v	m/s
Volumetric flow rate	V	m^3/s
Average velocity	v_{in}	m/s
Fluid z-component velocity	w	m/s
Channel width	W_{ch}	m
Axial coordinate	x	m
Vertical coordinate	y	m
Horizontal coordinate	z	m
Pressure drop across heat exchang	ΔP	Pa

Greek symbols

Performance index	η	1 / Pa
t transfer rate to pumping power ratio	η^*	-
Heat exchanger effectiveness	ε	-
Dynamic viscosity	μ	Pa/s
Density	ρ	kg/m^3
Inclination angle	θ	degree

Subscripts

Quantity	Symbol
Cold fluid	c
Hot fluid	h
Inlet	I

Maximum value	**max**
Outlet	**o**
Solid	**s**

1-Introduction:

The importance of microchannels increased due to their high heat transfer coefficient and decreasing size. Microchannels offer an increase of heat transfer surface and large surface area to volume ratio providing a much higher heat transfer per unit volume than channels of conventional sizes. The development in fabrication of small devices (micron sized) in now days increasing the need for better understanding of thermal and hydrodynamic characteristics of flow in microchannels and micro tubes.

Microchannel heat exchangers are of interest because they can remove large amount of heat over a small volume. This ability makes it well suited for highly specific applications that required compact high heat energy removal solutions such as, biomedical processes, metrology, telecommunications, cooling of high heat flux high density microelectronics, automotive industries, nuclear reactor barriers, fuel processing, aerospace and chemical industries [1 and 2]. The size of channels is the factor of interest in microchannels and it also affect both the pressure drop and effectiveness of (CFMCHE). Since the hydraulic diameter has similar effect on the heat transfer coefficient and pressure drop i.e. decreasing the hydraulic diameter lead to increase both the heat transfer and pressure drop, therefore it must be deal carefully with this factor in design of this type of heat exchangers. In addition to the size of channels its shape also plays an important role in the hydrodynamic and thermal performance of a CFMCHE. There are a large number of researches in the literature in which the fluid flow and heat transfer in microchannels were studied experimentally, numerically and analytically also the performance of microchannel heat sinks and its affecting parameters were studied extensively. But there are limited researches related to the performance of two fluids microchannel heat exchangers and more research works are needed to investigate this type of heat exchangers.

Al-Bakhet and Ahmad (2005) [3] numerically investigated the flow and heat transfer in parallel flow microchannel heat exchanger. They used the idea of a hybrid approach, in which the nonlinear momentum equations for one or two channels were solved by using CFD codes. The velocity field was inputted into a user developed code for solving the energy equation.

They studied the heat transfer for thermally developing laminar flow in two parallel rectangular channels which represent the complete parallel flow microchannel heat exchanger. From the results in the entrance region the developing velocity profiles leads to higher values of the overall heat transfer coefficient.

Brandner et al (2006) [4] studied experimentally various microstructures cross flow heat exchanger and compare between them. Several microchannel devices with different hydraulic diameters compared with respect to their heat transfer capabilities. Also two devices in which the microchannels replaced by different two- dimensional arrays of columnar micro fins were compared to each other and to microchannel device. They found that the heat transfer in micro heat exchanger can be enhanced by decreasing the hydraulic diameter of microchannels. Also if the design is based on stacked foils comprising microchannels, two different layouts of micro column arrays (aligned and staggered) were compared with respect to their heat transfer capabilities at a given mass flow. A staggered array of micro columns maximized the heat transfer in a given volume.

Foli et al (2006) [5] presented two methods for determining the optimal design parameters of the microchannels in microchannel heat exchanger MCHE that maximize the heat transfer rate or heat flux subjected to specified design constraints. The first approach that is combines CFD with the analytical solution of a simplified transport equation for momentum and heat transfer. This approach optimized the dimensions of a microchannel with predetermined geometry to find the optimum values of aspect ratio of channels. The second approach involves the usage of multi-objective genetic algorithms in combination with CFD. It has been demonstrated that the performance of MCHE depends on the operating conditions and aspect ratio of the microchannels that make up the flow passages.

Ngo et al (2007) [6] Investigated numerically the thermal hydraulic characteristics of a new microchannel heat exchangers with s-shaped and zigzag fins using FLUENT code. They found that the microchannel heat exchanger (MCHE) with s-shaped fins provided 6-7 times lower pressure drop while maintaining heat transfer performance that was almost equivalent to that of a conventional MCHE with zigzag fins. Correlations of Nusselt numbers and pressure drop factors for new MCHE with s-shaped fins and for a zigzag fins as a function of Reynolds number and Prandle number were developed.

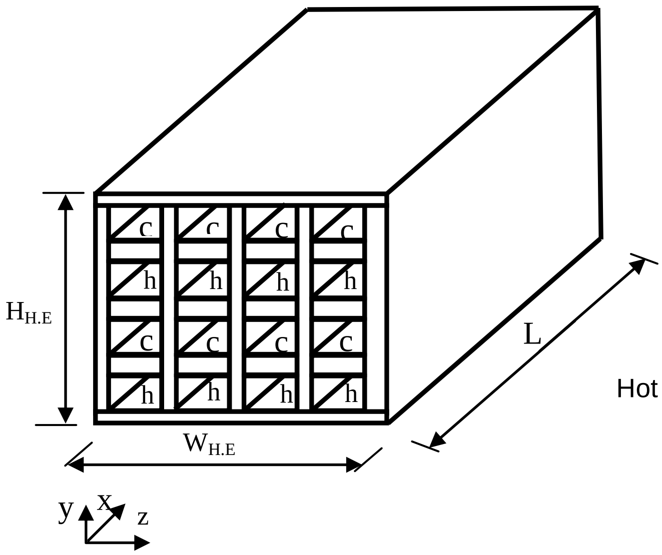
kang et al (2007) [7] developed a theoretical model to predict the thermal and fluidic characteristics of micro cross-flow heat exchanger. This model describes the interactive effect between the effectiveness and pressure drop in cross flow microchannel heat exchanger. The flow in channels was assumed fully developed laminar flow. They found that under the same effectiveness, the heat transfer rate increase with rising average temperature of working fluid in hot and cold flow sides. The pressure drop decreases because of the temperature influence especially on the cold flow side. And the higher average temperature situation has the larger heat transfer rate. Under the different effectiveness the heat transfer rate and pressure drop decreased with the increase in effectiveness.

Hrnjak et al (2008) [8] used the microchannel heat exchanger as air cooled condenser in ammonia chiller to show the benefits of this heat exchanger in charge reduction and excellent heat transfer to volume, mass and surface area ratio. The charge, heat transfer and pressure drop measurements were taken for two types of condensers: a serpentine flat macro tube (hydraulic diameter Dh= 4.06 mm) and a parallel flow microchannels tube (Dh = 0.7 mm). Overall condenser performance was quantified in terms of heat capacity, refrigerant and air side pressure drop and overall heat transfer coefficient. Comparisons between the two similarly sized condensers show the superiority of the microchannel design. They found that the microchannel charge is an average of 53 % less than for the serpentine; the microchannel condenser charge per capacity ratio is around 76 % less than for the macro channel serpentine condenser. Also the overall heat transfer coefficient for a given face velocity is 60 - 80 % higher than for serpentine condenser.

Mushtaq et al (2009) [9] investigated numerically the effect of channels geometry (the size and shape of channels) on the performance of counter flow microchannel heat exchanger and used liquid water as a cooling fluid. They found that with decreasing the size of channels both the effectiveness of heat exchanger and pressure drop were increased and they claimed that the decision of increasing or decreasing the size of channels depends on the application in which this heat exchanger is used. Also they found that the circle is the best shape for the channels of this type of heat exchangers since it gives higher overall performance (including both the hydrodynamic and thermal performance).

2. Mathematical model:

Mushtaq et al in their paper [7] studied CFMCHE with different channels shapes and sizes with constant cross-sectional area. In this paper the CFMCHE has been studied and an improvement is made by using variable cross-sectional area channels instead of constant cross section to improve both the hydrodynamic and thermal performance. Schematic structure of a counter flow microchannel heat exchanger with square channels is shown in Fig.1.



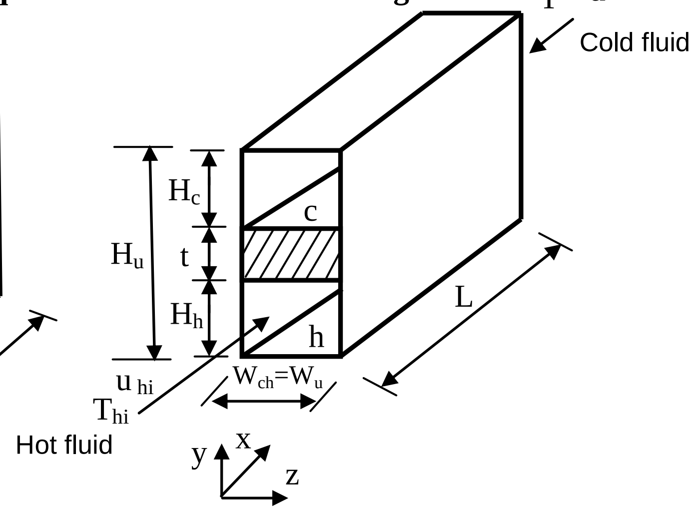


Fig. 2 A schematic of heat exchange unite

Fig. 1 A schematic model of the counter flow MCHE

To study the entire CFMCHE numerically, it is complicated and needs huge CPU time. Due to geometrical and thermal symmetry between channels rows an individual heat exchange unit which consists of two channels (hot and cold) and a separating wall will be considered as shown in Fig. 2 to represents a complete CFMCHE and gives an adequate indication about its performance. Heat is transferred from hot fluid to cold fluid through the thick wall separating them.

The governing equations and its boundary conditions in Cartesian coordinates for 3D, laminar steady incompressible flow with constant properties of fluids are: Continuity equation

$$\frac{\partial u_j}{\partial x}+\frac{\partial v_j}{\partial y}+\frac{\partial w_j}{\partial z}=0 \qquad \textbf{(1)}$$

Momentum equations

$$u_j \frac{\partial u_j}{\partial x} + v_j \frac{\partial u_j}{\partial y} + w_j \frac{\partial u_j}{\partial z} = -\frac{1}{\rho_j}\frac{\partial P}{\partial x} + \frac{\mu_j}{\rho_j}\left(\frac{\partial^2 u_j}{\partial x^2} + \frac{\partial^2 u_j}{\partial y^2} + \frac{\partial^2 u_j}{\partial z^2}\right)$$

$$u_j \frac{\partial v_j}{\partial x} + v_j \frac{\partial v_j}{\partial y} + w_j \frac{\partial v_j}{\partial z} = -\frac{1}{\rho_j}\frac{\partial P}{\partial y} + \frac{\mu_j}{\rho_j}\left(\frac{\partial^2 v_j}{\partial x^2} + \frac{\partial^2 v_j}{\partial y^2} + \frac{\partial^2 v_j}{\partial z^2}\right)$$

$$u_j \frac{\partial w_j}{\partial x} + v_j \frac{\partial w_j}{\partial y} + w_j \frac{\partial w_j}{\partial z} = -\frac{1}{\rho_j}\frac{\partial P}{\partial z} + \frac{\mu_j}{\rho_j}\left(\frac{\partial^2 w_j}{\partial x^2} + \frac{\partial^2 w_j}{\partial y^2} + \frac{\partial^2 w_j}{\partial z^2}\right)$$

Energy equation

$$u_j \frac{\partial T_j}{\partial x} + v_j \frac{\partial T_j}{\partial y} + w_j \frac{\partial T_j}{\partial z} = \frac{k_j}{\rho_j Cp_j}\left(\frac{\partial^2 T_j}{\partial x^2} + \frac{\partial^2 T_j}{\partial y^2} + \frac{\partial^2 T_j}{\partial z^2}\right) \qquad \textbf{(5)}$$

Where j=h and c for hot and cold channels respectively
The energy equation for the solid wall is

$$\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} = 0 \qquad \textbf{(6)}$$

The boundary conditions are:

Lower channel (Hot fluid) ($0< y <H_h$)

at x = 0	$u_h = u_{hi}$, $v_h = w_h = 0$, $T_h = T_{hi}$		**(hot fluid inflow)**
at x = L	$\frac{\partial u_h}{\partial x} = v_h = w_h = 0$,	$\frac{\partial T_h}{\partial x} = 0$	**(hot fluid outflow)**
at z = 0	$u_h = v_h = w_h = 0$,	$\frac{\partial T_h}{\partial z} = 0$	**(adiabatic wall)**
at z = W_{ch}	$u_h = v_h = w_h = 0$,	$\frac{\partial T_h}{\partial z} = 0$	**(adiabatic wall)**
at y = 0	$u_h = v_h = w_h = 0$,	$\frac{\partial T_h}{\partial y} = 0$	**(adiabatic wall)**
at y = H_h	$u_h = v_h = w_h = 0$,	$-k_h \frac{\partial T_h}{\partial y} = -k_s \frac{\partial T_s}{\partial y}$	**(Solid –fluid interface)**

Upper channel (Cold fluid) ($H_h + t < y < H_h + t + H_c$) :

at x = 0	$\frac{\partial u_c}{\partial x} = v_c = w_c = 0$,	$\frac{\partial T_c}{\partial x} = 0$	**(cold fluid outflow)**
at x = L	$u_c = u_{ci}$, $v_c = w_c = 0$, $T_c = T_{ci}$		**(cold fluid inflow)**
at z = 0	$u_c = v_c = w_c = 0$,	$\frac{\partial T_c}{\partial z} = 0$	**(adiabatic wall)**
at z = W_{ch}	$u_c = v_c = w_c = 0$,	$\frac{\partial T_c}{\partial z} = 0$	**(adiabatic wall)**
At y = ($H_h + t + H_c$)	$u_c = v_c = w_c = 0$,	$\frac{\partial T_c}{\partial y} = 0$	**(adiabatic wall)**

at y = (H_h + t) **$u_c = v_c = w_c = 0$,** $-k_c \frac{\partial T_c}{\partial y} = -k_s \frac{\partial T_s}{\partial y}$ **(Solid - fluid interface)**

Solid wall B.C. **(H_h < y < H_h + t):**

at x = 0 , L $\frac{\partial T_s}{\partial x} = 0$ **(adiabatic wall)**

at z = 0 , W_{ch} $\frac{\partial T_s}{\partial z} = 0$ **(adiabatic wall)**

at y = H_h $-k_h \frac{\partial T_h}{\partial y} = -k_s \frac{\partial T_s}{\partial y}$ **(Solid - fluid interface)**

at y = H_h + t $-k_c \frac{\partial T_c}{\partial y} = -k_s \frac{\partial T_s}{\partial y}$ **(Solid - fluid interface)**

By solving the above governing equations numerically the velocity, pressure and temperature distributions are determined in the fluid and solid domains. From these distributions one can determine heat exchanger effectiveness, heat transfer rate, pressure drop, pumping power required, overall heat transfer coefficient and overall performance index.

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

$$\varepsilon = \frac{q}{q_{\max}} \quad \varepsilon = \frac{q}{q_{\max}} \tag{7}$$

Where

C_{min} (T_{hi} – T_{ci}) **(8)**

Where $C_h = mC_{p_h}$ **and** $C_c = mC_{p_c}$

$$\varepsilon = \frac{C_h(T_{hi} - T_{ho})}{C_{\min}(T_{hi} - T_{ci})} = \frac{C_c(T_{co} - T_{ci})}{C_{\min}(T_{hi} - T_{ci})} \tag{9}$$

The pumping power required to circulate hot and cold fluids in a CFMCHE is [11]:

$$P.P = V_h \Delta P_h + V_c \Delta P_c \tag{10}$$

Where V is the volumetric flow rate (m^3 /s)

$$V = v_{in} A \tag{11}$$

$$\Delta P_t = \Delta P_h + \Delta P_c = (P_{hi} - P_{ho}) + (P_{ci} - P_{co}) \tag{12}$$

Where ΔP_t is total pressure drop in heat exchange unit.

To calculate the overall performance of a counter flow MCHE, performance index is determined, which is represented by the ratio of CFMCHE effectiveness to the total pressure drop [10]. Performance index link the thermal and hydrodynamic performance to obtain an indication about the overall performance.

$$\eta = \frac{\varepsilon}{\Delta P_t} \qquad (13)$$

A different form of performance index defined as the ratio of the heat transferred between the fluids to the total pumping power can also be used for determining overall performance of a CFMCHE [9].

$$\eta^* = \frac{q}{P.P} \qquad (14)$$

The aspect ratio of rectangular channels used is the ratio of channel height to the channel width.

$$AR = \frac{H_{ch}}{W_{ch}} \qquad (15)$$

Both of the η and η^* used to obtain an indication about the overall performance, since it link the thermal and hydrodynamic performances. The trends of variation of η and η^* are the same but its values are different.

2-1. Variable cross – sectional area:

The aim of the new idea presented in this paper is to improve the overall performance of a CFMCHE by making a variation in cross-sectional area of channels along the heat exchanger to study its effect on the pressure drop and thermal performance (effectiveness) since both microchannels in the heat exchange unit will be diverge. The variation in cross-sectional area of microchannels is done only by inclined the wall separating two channels with

a certain small angles (the increment value in the inclination angle is $\Delta\theta = 0.5^o$). The schematic of this situation is shown in the figures 3 and 4 below:

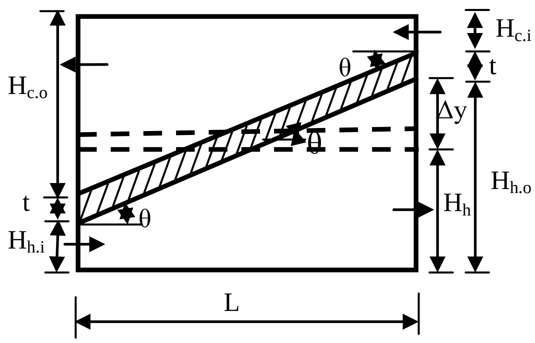


Fig. 3 Two-dimensional schematic of the heat exchange unit with variable area channels.

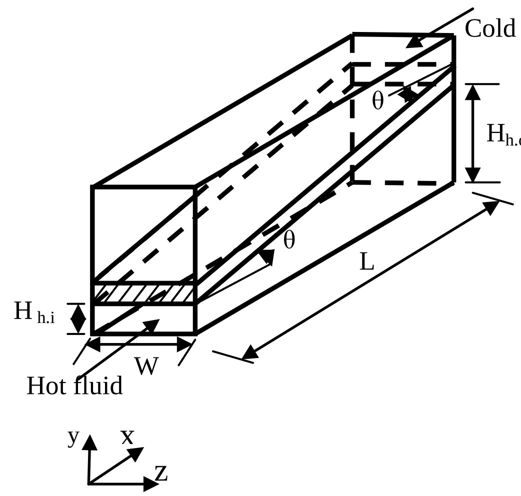


Fig. 4 3D schematic of the heat exchange unit with variable cross-sectional area channels

To continue in using individual heat exchange unit consists of two channels and separating wall as a model to represent the complete CFMCHE due to numerical solution constraints (to conserve the symmetry between different units in the same heat exchanger) to fulfill the requirements of the present numerical model. The volume, shape and the external dimensions of the heat exchange unit in addition to the width of the channels are remaining constant and the variation is done in the channels height only.

For variable cross-sectional area channels:

Cross – sectional area $A_{ch}(x) = W_{ch}(Hi + x \tan(\theta))$ **(16)**

At the channel inlet:

$$H_i = H - \Delta y = H - \frac{L}{2}\tan\theta \qquad (17)$$

$$A_{ch,i} = W_{ch} H_i \qquad (18)$$

At the channel outlet:

$$H_o = H + \Delta y = H + \frac{L}{2}\tan\theta \qquad (19)$$

$$A_{ch,o} = W_{ch} H_o \qquad (20)$$

Where: Δy **: Increment in the channels height.**

H_i **: Channels height at the inlet.**

H_o **: Channels height at the outlet.**

The approach of using channels with variable cross-sectional area is applied to (square, rectangular and trapezoidal) channels. The figures and equations explained above were for

the square and rectangular channels. The change in the area of the trapezoidal channels was done in the same way by moving the separating wall by a certain angle:

3. Numerical Solution

A computational fluid dynamic code is used to calculate flow velocity, pressure and temperature in the channels of a CFMCHE. Finite volume method (FVM) was used to convert the governing equations to algebraic equations accomplished using an "upwind" scheme. The SIMPLE algorithm was used to enforce mass conservation and to obtain pressure field [12]. The segregated solver was used to solve the governing integral equations for the conservation of mass, momentum and energy.

A mesh was generated by descretizing the computational domain (two channels and the separating wall) with hexahedral elements and a mesh refinement is made. The convergence criteria used to control the solution for momentum and energy equations were set to be less than 10^{-6}.

4. Results and discussion:

The geometry of microchannels has a considerable effect on both the hydrodynamic and thermal performance of a CFMCHE and it's important to investigate this parameter extensively to optimize the overall performance of this heat exchanger (maximizing the effectiveness and minimizing the pressure drop). All observed studies in literature as the author's knowledge were carried out for microchannel heat exchangers with constant cross-sectional area of microchannels.

The comparison between the different cases for different values of inclination angle is made with the same volume and outer dimensions of heat exchange unit, Re, Pr, wall thickness and thermal boundary conditions. The simulation is made first for ordinary heat exchange unit (constant cross-sectional area channels) and then it repeated for variable cross-sectional area channels with number of angles to find out the influence of inclination angle for different studied shapes.

For square channels:

Fig. 5 shows the variation of effectiveness with inclination angle for two values of Re for square channels. From this figure one can see that the effectiveness increased with increase the angle of inclination due to increase the area of heat transfer between two fluids with

increase the angle of inclination. Also this is may be due to the effect of decreasing the velocity of flow with increase the cross-sectional area in the flow direction to conserve the continuity which leads to increase the time taken to exchange the heat between two fluids. In addition to the effect of the entrance region which is located in the first half of channels with smaller area and higher velocity. The trend of variation of the effectiveness for the two values of Reynolds numbers is similar. Also it can be find from this figure that, the values of effectiveness corresponding to Re = 50 are larger than its values at Re = 100 due to decreasing the velocity of flow with decreasing the value of Reynolds number.

Fig. 6 indicates the variation of total pressure drop across heat exchange unit with inclination angle for two values of Re for square channels. From this figure it can be seen that the pressure drop decreased with increase inclination angle until it reach an optimum value which gives minimum pressure drop beyond it the pressure drop increased with the increment in the angle of inclination. This behavior of pressure drop can be explained as follow.

As mentioned before the variation in cross-sectional area is made by moving the separating wall by certain angle from mid point to produce two diverging channels. Therefore each of the new variable cross-sectional area channels has two parts. In The first part the cross-sectional area is smaller than that of ordinary channel without inclination while in the second part the cross-sectional area become larger. The entrance region is located in the first part for both channels. The pressure drop across channels is decreased due to expansion in channels cross-sectional area. Increase in the angle of inclination produce extra decrease in the area of the first part including the entrance region which is results in maximizing its effect. This is leads to extra increase in the pressure drop. This increase in the pressure drop will exceed the reduction occurred due to expansion in area. Therefore, the overall pressure drop across the channels will increase with increase the angle of inclination. Also the trend of variation of pressure drop with inclination angle is similar for two values of Re.

Variation of performance index with inclination angle for two values of Re for square channels is shown in Fig. 7. From this figure it can be seen that, the performance index increased with increase the angle until reach an optimum angle at which the performance index has its maximum value then it will be decreased with increasing the angle due to increase the pressure drop. The angle $\theta = 1.5^o$ is the optimum angle for all studied Re for

square channels. The percentage of modification in performance index at θ = 1.5^o compared to θ = 0^o is shown in table 1 below for different values of Reynolds number.

Table 1 percentage modification in performance index for square channels

Re	$(\frac{\eta_{\theta=1.5} - \eta_{\theta=0}}{\eta_{\theta=0}} x\,100\%)$
50	23.87 %
100	27.96 %
300	35.84 %
500	41.63 %

From this table the modification in performance index increased with increase Re due to increase both the percentage of modification in effectiveness and percentage of reduction in pressure drop with increase Re.

Fig. 8 illustrate the variation of (η*) with inclination angle for two values of Re for square channels. From this figure the trend of variation in η* is similar to that for the performance index as discussed before. Also θ = 1.5^o is the optimum angle which is the same as for the performance index. The modification in (η*) ratio at θ = 1.5^o for different values of Re indicated in table 2 below.

Table 2 modification in (η*) with Re

Re	$[\frac{(\eta^*)_{\theta=1.5} - (\eta^*)_{\theta=0}}{(\eta^*)_{\theta=0}}]x\,100\%$
50	22.47 %
100	26.84 %
300	36.39 %
500	43.69 %

Also it can be seen from this table the modification in (η*) increased with increase Re.

Rectangular channels with (AR = 0.5):

Fig. 9 indicates the variation of effectiveness with inclination angle for rectangular channels initially with aspect ratio 0.5. It's noted that the effectiveness increased with the inclination angle due to increase the surface area of heat transfer with angle. Also this is may be due to decrease the velocity of flow along the channels which lead to increase the temperature difference due to increase of the heat transfer period as explained with Fig. 5.

Fig. 10 shows the variation of total pressure drop with inclination angle. It's clear that the total pressure drop decreased with increase inclination angle until reach its optimum value at

which the pressure drop is minimum, then the pressure drop increase with increase the angle as explained with Fig. 6 for square channels.

Fig. 11 shows the variation of performance index with inclination angle. From this figure the performance index increase with increase inclination angle until reach its maximum value and then decreased. The angle $\theta = 0.5^o$ is the optimum angle at which the performance index has its maximum values and the modification in performance index at $\theta = 0.5^o$ is 9.71 % compared with channels of constant area.

Variation of heat transfer rate to the pumping power ratio (η^*) with inclination angle is illustrated in Fig. 12. It's clear that the (η^*) ratio has similar trend as for performance index and also the optimum angle is $\theta = 0.5^o$ at which the maximum modification in (η^*) is 8.77 % compared with constant area channels $\theta = 0^o$.

Rectangular channels with (AR = 1.2):

Fig. 13 indicates the variation of performance index with inclination angle for rectangular channels with AR=1.2. It's clear from this figure that the variation of the performance index corresponding to the AR = 1.2 is similar as that for AR = 0.5. The optimum angle corresponding to AR = 1.2 is $\theta = 3^o$ at which the modification in performance index is 28.57 % compared to ordinary channels with $\theta = 0^o$ at Re = 50.

Variation of heat transfer rate to the pumping power ratio with inclination angle is illustrated in Fig. 14. It's shown that the variation of (η^*) with angle is similar for the trend of performance index in the previous figure. The optimum angle is also $\theta = 3^o$ and the percentage of modification in (η^*) at this angle compared with the constant area channels $\theta = 0^o$ is 28.72 %.

Rectangular channels with (AR = 1.5):

Fig. 15 shows the variation of performance index with inclination angle for rectangular channels with AR=1.5. From this figure one can see that, the performance index in this case has similar trend as for all previous discussed cases and the optimum angle is $\theta = 4^o$ at which a maximum modification of 47.32 % in performance index is obtained at Re = 50. Which is higher than the modification in all previous cases.

Fig. 16 represent the variation of heat transfer rate over pumping power ratio with inclination angle. The variation of (η^*) with angle has similar trend compared with the

previous discussed cases. Also θ = 4^{o} is the optimum angle at which a modification in (η^{*}) of 46.98 % compared with constant area channels θ = 0^{o} is obtained.

Aspect ratio of channels:

Fig. 17 shows the variation of optimum inclination angle with aspect ratio for rectangular channels. From this figure it's clear that, the optimum angle which gives the maximum overall performance increased with increase aspect ratio of channels. This is due to increasing the height of channels with increase the aspect ratio which leads to increase the ability of extra expansion in cross-sectional area of channels since the varying in area is made with fixed value of width.

Fig. 18 shows the variation of the percentage modification gained in performance index at the optimum angle [enhancement factor $\left(\dfrac{\eta_{(optimum\ \ angle)} - \eta_{\theta=0}}{\eta_{\theta=0}}\, x\, 100\%\right)$**] with aspect ratio for rectangular channels. From this figure the enhancement in performance index is increased with increase the aspect ratio of channels due to increase the effectiveness and decrease the pressure drop at the specified aspect ratio. Therefore it's important to take this enhancement into account and consider the channels with variable cross-sectional area as main choice in design a CFMCHE and trying to obtain maximum allowable modification.**

Trapezoidal channels:

Fig. 19 indicates the variation of effectiveness with inclination angle for trapezoidal channels. From this figure the effectiveness increased with increase of the angle of inclination as for square and rectangular channels which discussed before.

Fig. 20 shows the variation of performance index with inclination angle for trapezoidal channels. Its clear from this figure the performance index increased with increase the angle of inclination until it reach the optimum angle which is θ = 0.5^{o}. Increasing the inclination angle larger than this value responsible for decreasing the value of performance index due to increase the pressure drop. The maximum enhancement in performance index at the optimum angle is 7.2 %.

Variation of heat transfer rate over pumping power ratio (η^{*}) with inclination angle is illustrated in Fig. 21. From this figure it's clear that the value of (η^{*}) increased with increase angle up to maximum at θ = 0.5^{o}. After θ = 0.5^{o} the value of (η^{*}) decreased with increase

angle. The maximum modification in (η^*) is 6.24 % obtained at $\theta = 0.5^o$ compared with constant cross-sectional area channels.

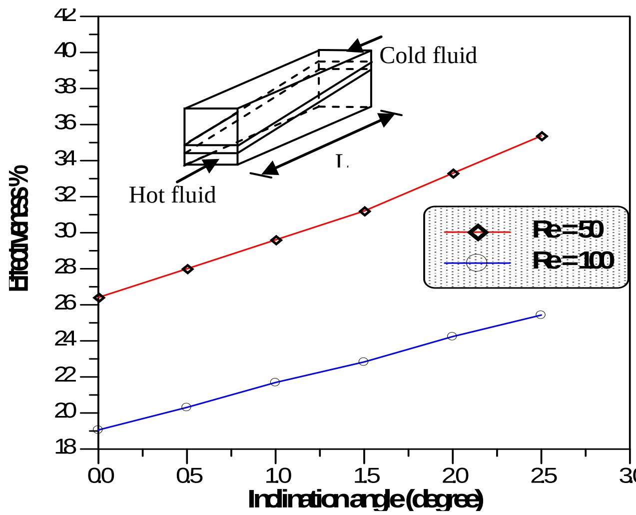


Fig. 5 Variation of effectiveness with inclination angle for square channels

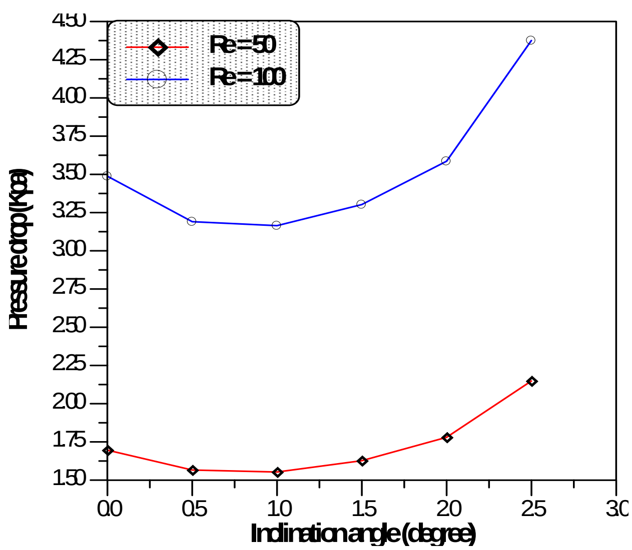


Fig. 6 Variation of pressure drop with inclination angle for square channels

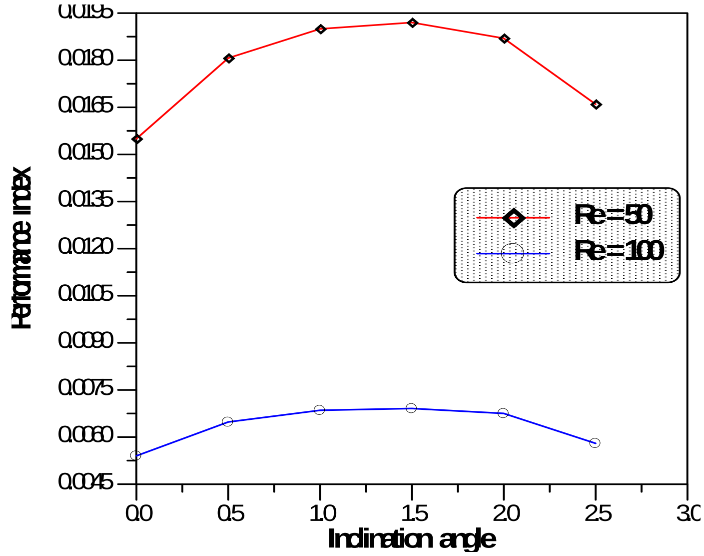


Fig. 7 Variation of performance index with inclination angle for square channels

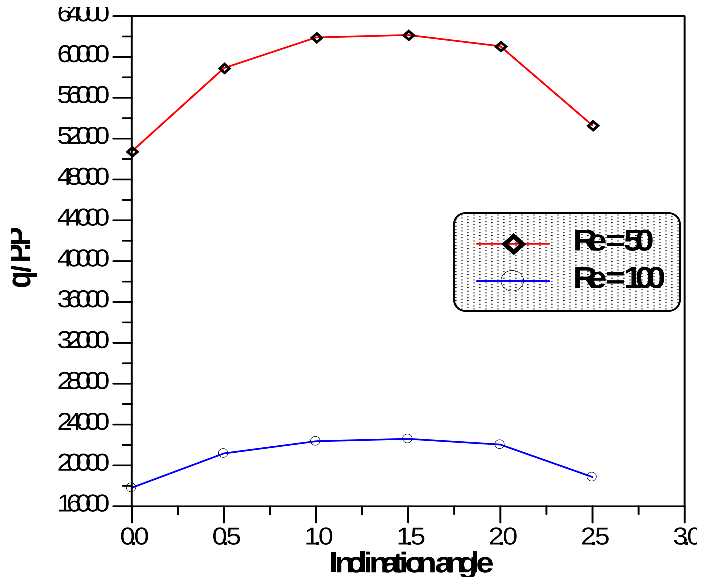


Fig. 8 Variation of heat transfer rate over pumping power ratio with inclination angle for square channels

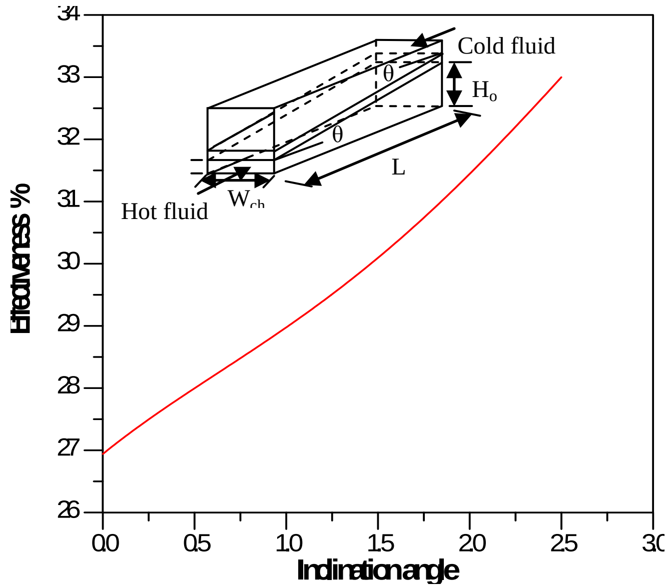


Fig. 9 Variation of effectiveness with angle for rectangular channels (AR=0.5 and Re =50)

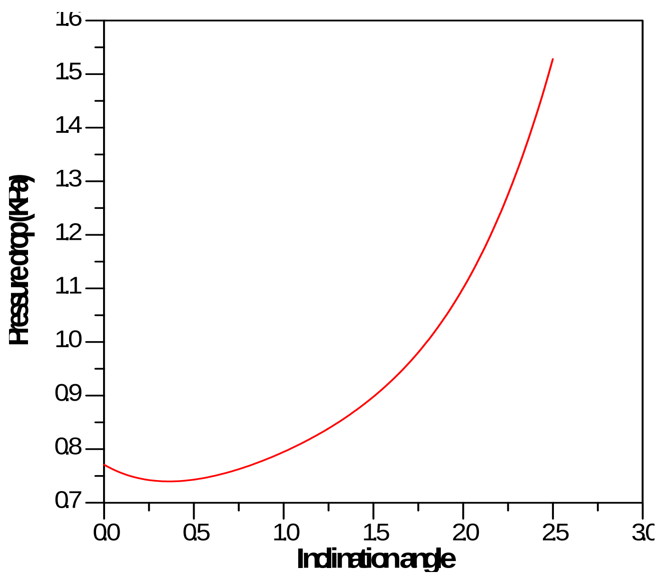


Fig. 10 Variation of pressure drop with angle for rectangular channels (AR=0.5 and Re = 50)

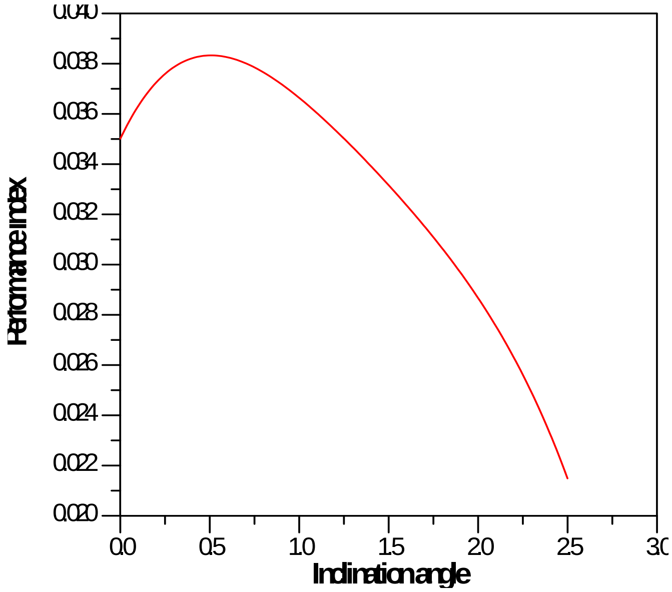


Fig. 11 Variation of performance index with angle for rectangular channels (AR=0.5 & Re =50)

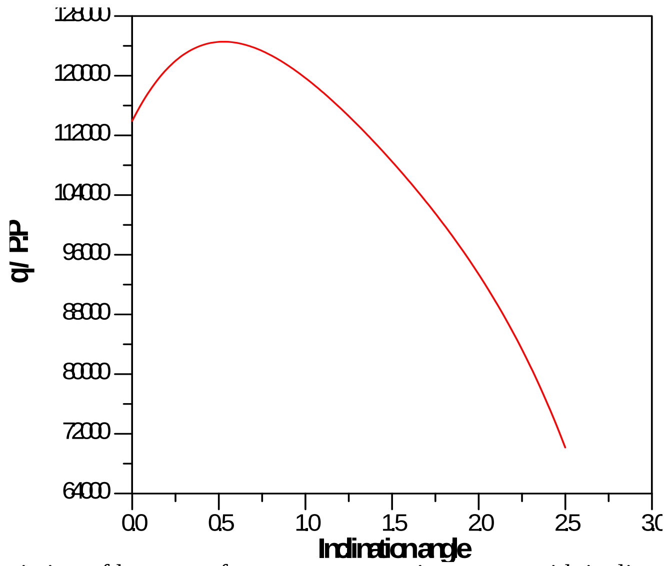


Fig. 12 Variation of heat transfer rate over pumping power with inclination angle for rectangular channels (AR=0.5 & Re = 50)

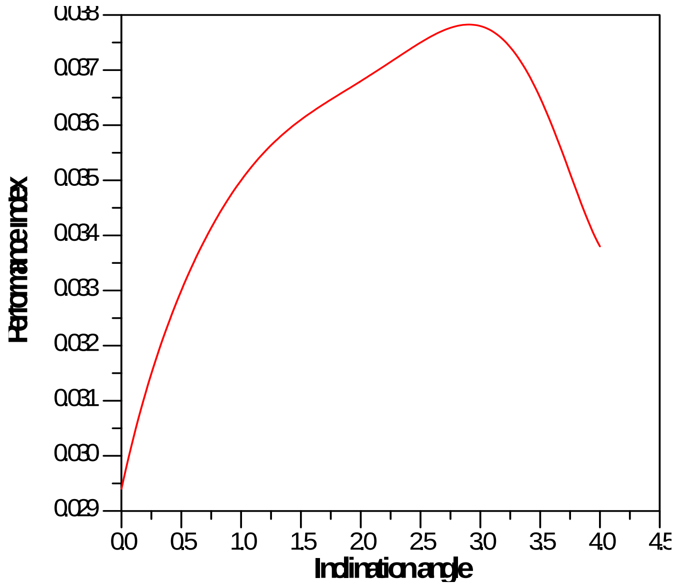


Fig. 13 Variation of performance index with angle for rectangular channels (AR=1.2 & Re = 50)

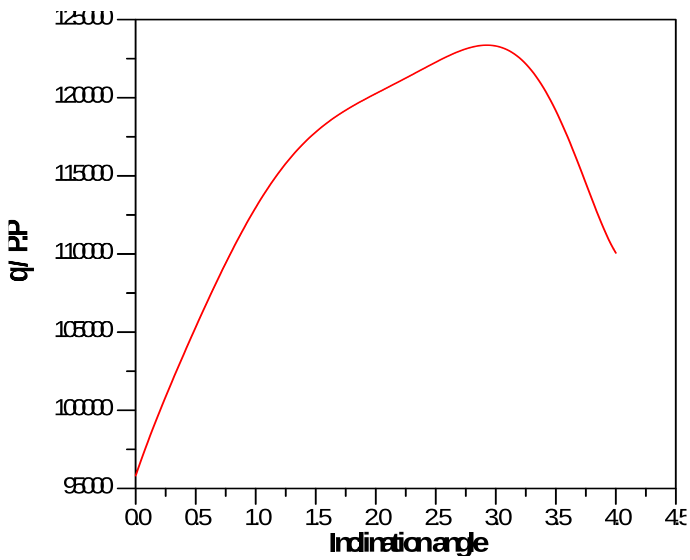


Fig. 14 Variation of heat transfer rate over pumping power with Angle for rectangular channels (AR=1.2 & Re = 50)

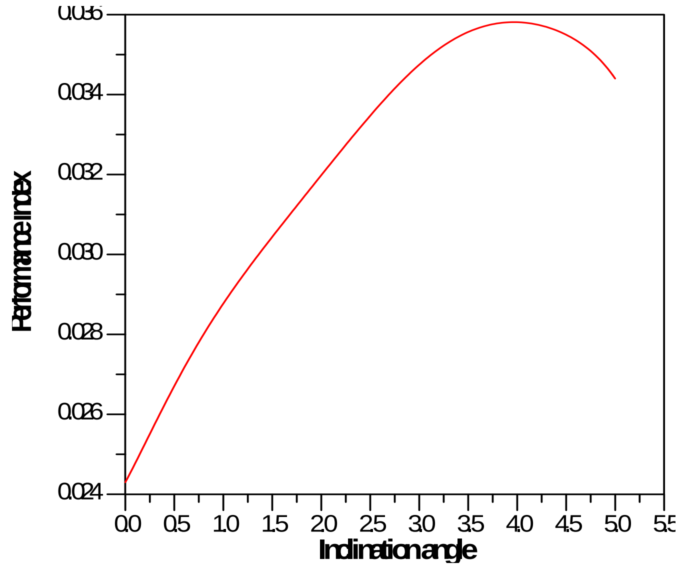


Fig. 15 Variation of performance index with angle for rectangular channels (AR=1.5 & Re=50)

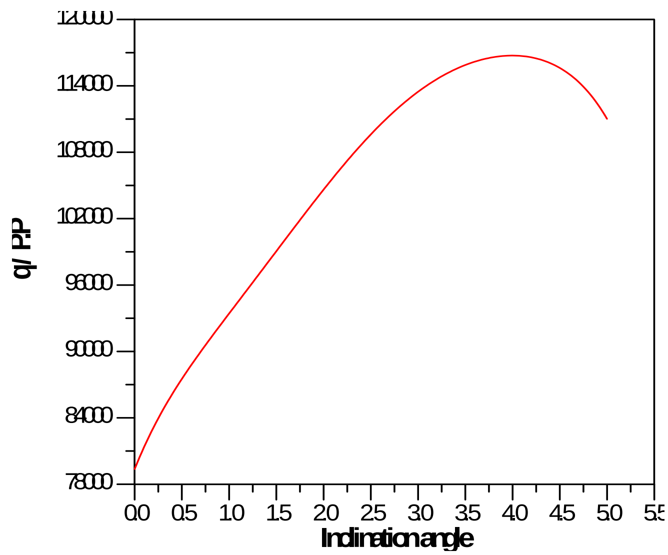


Fig. 16 Variation of heat transfer rate over pumping power with angle for rectangular channels (AR=1.5 & Re = 50)

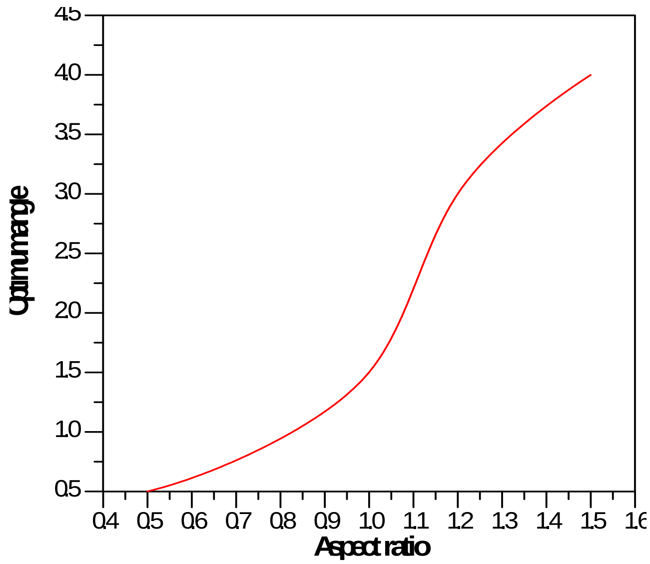


Fig. 17 Variation of optimum angle with aspect ratio for rectangular channels

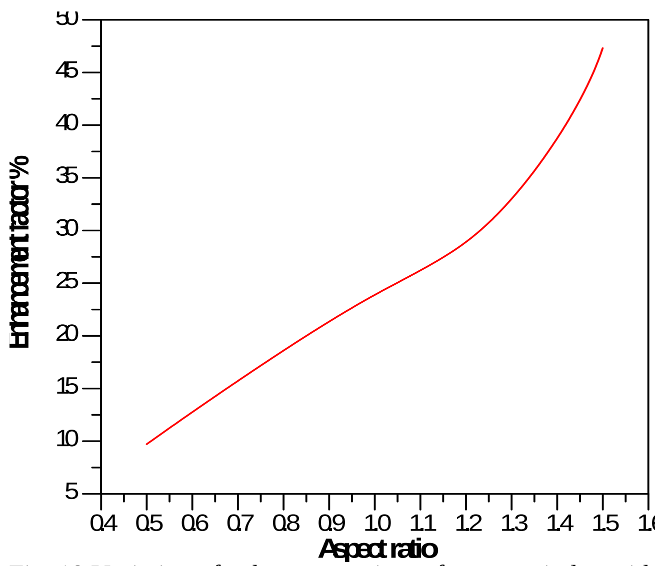


Fig. 18 Variation of enhancement in performance index with aspect ratio for rectangular channels

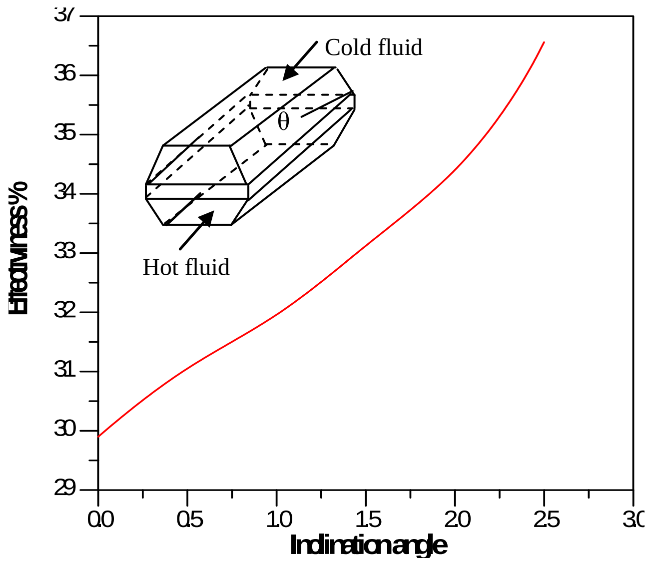


Fig. 19 Variation of effectiveness with inclination angle for trapezoidal channels (Re = 50)

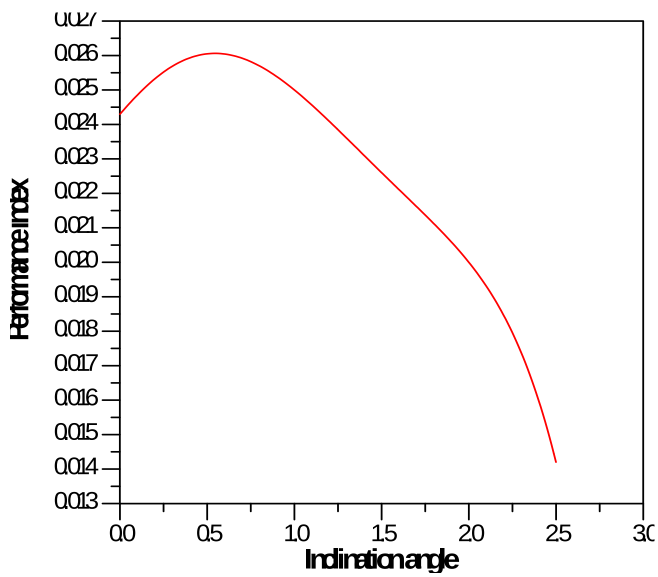


Fig. 20 Variation of performance index with inclination angle for trapezoidal channels (Re = 50)

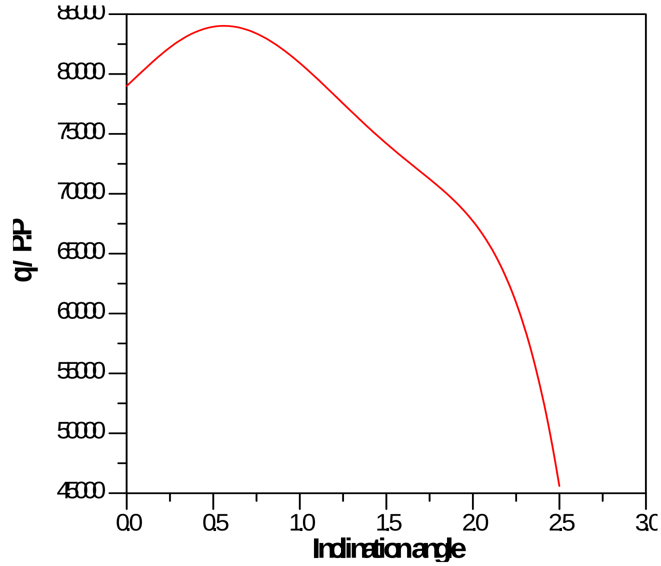


Fig. 21 Variation of heat transfer rate over pumping power with inclination angle for trapezoidal channels (Re = 50)

5. Conclusions:

From the results obtained the following conclusions can be drawn:

1- **Using of variable cross- sectional area microchannels instead of constant cross- sectional area microchannels lead to modify both the hydrodynamic and thermal performance of a CFMCHE since it lead to increase the effectiveness and reduce the pressure drop.**

2- **For each channel geometry there is a best inclination angle at which the performance reach its maximum value.**

3- **For rectangular microchannels the value of optimum angle found to be increased with increase the aspect ratio of the channels under consideration.**

4- **The value of enhancement in overall performance of a CFMCHE with rectangular microchannels fount to be increased with increase the aspect ratio of channels.**

6. References:

[1] Tatsuya Okabe, Kwasi Foli, Markus Olhofer, Yaochu Jin, Bernhard Sendhoff , " Comparative studies on micro heat exchanger optimization " IEEE, 2003.

[2] Sanjay K. Roy, Branko L. Avanic, "A very high heat flux microchannel heat exchanger for cooling of semiconductor laser diode arrays ", IEEE transactions on components, Packaging, and manufacturing technology - Part B, 19 (2), May 1996.

[3] Hussien Albakhit, Ahmad Fakheri, "A hybrid approach for full numerical simulation of heat exchangers", ASME Heat Transfer Summer Conference, San Francisco, CA, USA, 2005.

[4] Brandner J. J., Anurjew E., Bohn L., Hansjosten E., Henning T., Schygulla U., Wenka A., Schubert K., " Concepts and realization of microstructure heat exchangers for enhanced heat transfer ", Experimental Thermal and fluid Science, 30 (8), pp. 801 - 809, 2006.

[5] Kwasi Foli, Tatsuya O., Markus O., Yaochu J., Benhard S., " Optimization of micro heat exchanger : CFD, analytical approach and multi - objective evolutionary algorithms " Int. Journal of Heat and Mass Transfer, 49 (5), pp. 1090 - 1099, 2006.

[6] Tri Lam Ngo, Yasuyoshi K., Konstantin N., Takao I., "Heat transfer and pressure drop correlations of microchannel heat exchanger with S-shaped and zigzag fins for carbon dioxide cycles", Experimental Thermal and Fluid science, 32 (2), pp. 560-57-, 2007.

[7] Shung - Wen Kang, Shin - Chau Tseng, " Analysis of effectiveness and pressure drop in micro cross - flow heat exchanger ", Applied Thermal Engineering, 27 (5), pp. 877 - 885, 2007.

[8] Pega Hranjak, Andy D. Litch, "Microchannel heat exchangers for charge minimization in air - cooled ammonia condensers and chillers", Int. Journal of Refrigeration, 2008.

[9] Mushtaq I. H., A. A. Rageb, M. Yaghoubi, Homayon H., " Influence of channel geometry on the performance of counter flow microchannel heat exchanger", Int. Journal of thermal sciences, Vol. 48, 1607–1618, 2009.

[10] Incropera .Frank P., Dewitt David P., " Fundamentals of heat and mass transfer " 4th Edition, Jhon Wiley and sons, 1996.

[11] Walshaw A. C., Jobson D. A., "Mechanics of fluids" Second edition SI Units, Longman Group Limited, London, 1972.

[12] Patankar, S.V., "Numerical heat transfer and fluid flow", Hemisphere Publishing Corporation, 1980.