Heat Transfer Enhancement in MCHS using Al_2O_3 /Water Nanofluid

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Abstract

In the present paper, Numerical simulation is performed to investigate the laminar force convection of Al_2O_3/water Nanofluid in a flow channel with different constant heat flux 50, 90, 150 W/cm^2 respectively and two values of mass flow rate of fluid. The heat sources are placed on the bottom wall of channel which produces much thermal energy that must be discarded from the system. The remaining surfaces of channel are kept adiabatic to exchange energy between Nanofluid and heat sources. The effects of Reynolds number Re < 1000), the volume fraction of nanoparticles of nanofluid have the percentages of 0, 1, 3 and 5%. on the average heat transfer coefficient (h) , pressure drop (ΔP), surface Nusselt number and wall temperature (T_w) are evaluated. The use of Nanofluid can produce an asymmetric velocity along the height of the channel. The results show that the wall temperature decreases remarkably as Re and volume of fraction increase. It is also observed that there is an enhancement of average heat transfer *coefficient* and it's observed also that the use of Nanofluid improves MCHS performance by reducing fin (conductive) thermal resistance.

Keywords: Nanofluid ; Laminar flow; Pressure drop; Heat transfer enhancement ; Forced Convection .

تحسين انتقال الحرارة في القنوات الميكروية باستخدام جسيمات النانوية الناتجة من خلط اوكسيد الالمنيوم مع الماء

في الدراسة المقدمة تم جراء محاكاةٍ عدديةٍ لدراسة إنتقالِ الحرارة بالحمل القسري لجريان طباقي لمائع (نانو) ناتج من خلط الماء مع جسيمات اوكسيد الالمنيوم يجري داخل قناة ميكروية مستطيلة الشكل محاطة بمادة السليكون الحراري وعرضه لفيض حراري ثابت وبمقادير مختلفة 50 ، 90 ، 150 واط / سنتيمتر المربع على التوالي مع نسبةِ تدفقٍ من السائل النانوي وبمقدار $1x10^{-5}$ و5.1×10^{-5} كغم بالثانية على التوالي. تم تسليط الفيض الحراري عند السطع السفلي للقناة . اما السطوحَ الباقيةَ للقناةِ تَبْقى معزوله. تم دراسة اتاثير كل من سرع الجريان للسائل ولأعداد رينولد اقل من 1000،والنسب الحجمية للجسيمات في الماء 0، 1، 3 و5 % على معاملِ انتقال الحرارةِ بالحمل ، فرق الضغطِ ، عدد نوسيلت ودرجة حرارة الجدران . بينت النتائج المستنبطه من الدراسة ان استخدام الجسيمات تنتجَ سرعةَ لا مُتَناظِرةَ على طول ارتفاع القناةِ اضافة الى هبوط في درجة حرارة الجدار مع زياده السرع والنسب الحجمية للجسيماتِ. كما بينت النتائج تحسين معاملِ انتقال الحرارة بالحمل من خلال انخفاض في المقاومة الحرارية للتوصيل الحراري.

الكلمات الدالة: المائع النانوي، جريان طباقي، انخفاض الضغط، تحسين انتقال الحرارة، الحمل القسري

Introduction

In last decade, primary research begins in the quest of using the nanoparticles in the various fields of applications. In the heat transfer problems values the quest for a good heat extraction in various ways of either using many fluids that have promising heat extraction efficiency. The maximum of a base fluid with Nano particle shows marked heat transfer enhancement[1]. Further investigation regarding an increases in the surface area of the radiative /convective surfaces for which a micro channels being introduced in the problems of that transfer enhancement[2].

As far as contribution of the friction factor and pressure drop for a laminar flow in the range up to 2000 Reynold number have a linearly variation in a micro chanel hydraulic geometry [3,4].

Concerning the geometrical shape with regarding its aspect ratio to the hydraulic diameter where the friction factor found to have some dependence [5,6,7 and 8], and this aspect ratios has no noticeable effect as far as the flow in laminar [9].

The Nanofluid can its maintained its enhancement properties in the range of up to 5% of nanoparticles to the base fluid[10]. While the channel geometrical shape affect the heat conductive properties by the geometrical size criterion and its volume proposed by the presence of thermal resistance [11,12,13 and 14]

Gupta et al. [15] studied fully developed laminar flow and heat transfer in equilateral triangular cross-sectional ducts with constant heat flux. Limited studies regarding the use nanofluids as coolants in micro and minichannel heat sinks exist in literature. Nguyen et al. [16] have investigated the usage of nanofluids in cooling electronic devices with nanofluids as the coolant marked reduction in the junction temperature was observed, especially at higher flow rates and higher particle loading percentage.

Micro channel cooling is a promising way to solve the cooling problem for computer chips. In micro channel cooling a fluid is flowing through a micro channel in close contact with the electronic chips . Due to the large area to volume ratio of the micro channel, the heat removal is larger than by conventional air cooling. Heat transfer by forced convection for gases is in the range 25-250 KW/m^2[17], whereas experimental micro channels with water have shown a heat flux of 500 KW/m^2 [18].

The aim of this work is to study the heat transfer enhancement for Al_2O_3 nano particle with water of distinct intrinsic thermal properties in a rectangular micro channel that exposed to different heat flux 50, 90 and 150 W/cm^2 respectively that give surface temperature of near 80 degree. Two mass flow rate of 1 $x10^{-5}$ and 5.1 $x10^{-5}$ kg/sec.

Numerical Solution and Mathematical Modeling for the case study

In this case study a micro channel heat sink of silicon configured in the figure 1.a,b,c and the flowing nanofluid of Al_2O_3/ water in the micro channel of rectangular cross-sectional area. The assumed flow of the nanofluid in question has to have a laminar flow. This configuration of the heat sink of the fluid performs a 3D conjugate heat transfer in the nanofluid. The channel dimensions given in table.1

In this configuration the heat flux subjected from the base and both silicon fins in a uniform heat flux which is dissipated in the nanofluid coolant. In order to control the flow of the fluid a forced flow in the micro duct with laminar flow being assumed of up to 1000 Reynold. Thus fluid at 20 C were supplied to the duct with two mass flow rate of 1 x10^{-5} and 5.1 x10^{-5} kg/sec and the heat flux assumed at 50,90 and 150 w/cm^2 exposed from the lower surface.

In order to get the requirements of the numerical solution, a discretization of the system of governing equations are given in appendix 1. A mesh generation in Gambit[19] was performed as shown in Figure 2.a,b,c and table (2). These numbers were optimized after many attempts to have a study solution and reduction in the number of iteration required for convergence.

1. Governing equations.

The Microchannel heat exchanger (MCHE) considered in this case is shown schematically in Figure(1). Heat is transferred between the fluids through the sink wall separating them. Several assumptions were made on the operating conditions:

1. The MCHE operates under steady-state conditions with Constant heat flux and negligible radiation heat transfer.
2. The fluids remain in single phase along the channel and the flow is laminar.
3. Thermophysical properties of the fluids and MCHE material are temperature-independent.
4. Flow mis-distribution and external heat transfer effects are neglected.
5. The outer walls of the MCHE are considered insulated (adiabatic).

The system of equations with other parametric relations is in good processing with fluent Code [20].

2. Thermo physical properties

The various thermal physical properties of Nanofluids can be evaluated as follows:

Density and Viscosity

The base-fluid considered in this work is water. Thermo-physical properties were obtained as polynomial functions of temperature found some were else [21] as :

$$\rho_w = -3.57\times10^{-3}T^2 + 1.88T + 753.2 \quad ..(1)$$

while the water viscosity is given by :

$$\mu_w = 2.591\times10^{-5}\times10^{2383/(T-143.2)} \quad .. (2)$$

The specific heat of water is considered constant at Cp_w = 4200. The effective density of the Nanofluid containing suspended particles can be evaluated through the following equation:

$$\rho_{nf} = \frac{\rho_{bf}V_{bf} + \rho_p V_p}{V_{bf} + V_p} = (1-\phi)\rho_{bf} + \phi\rho_p \quad ..(3)$$

For typical Nanofluids with Nanoparticles at a value of volume fraction less than 1%, a- minor change of less than 5% in the fluid density is expected. With respect to viscosity : the viscosity of mixture particles/fluid have two model invalued in calculation the Maxwell model which is introduced for sub millimitric size particles in suspension in fluid and that of Enstien Model for nano particles in suspension in fluid with volume fraction less than 0.01, where the surface interaction between particles and fluid being neglected. The Enstien Model [22] expressed as :

$$\mu_{nf} = \mu_{bf}(1+2.5\phi) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(4)$$

However other correlated models for Al_2O_3/water have been considered:

$$\mu_{nf} = (123\phi^2 + 7.3\phi + 1)\mu_f \quad [23] \ldots\ldots\ldots (5)$$

$$\mu_{nf} = (1 + 39.1\phi + 533.9\phi^2)\mu_f \quad [24] \ldots\ldots(6)$$

$$\mu_{nf} = (6.2\phi^2 + 2.5\phi + 1)\mu_f \quad [25] \ldots\ldots\ldots\ldots(7)$$

$$\mu_{nf} = (150\phi^2 + 2.5\phi + 1)\mu_f \quad [26] \ldots\ldots\ldots\ldots (8)$$

This correlation adapted for the present calculation as taking into consideration the size of the particles of less than 36 nm which can accommodate the agglomeration that can exist in real fluid under investigation.

Specific Heat

The specific heat of Nanofluid can be determined on assumption that a thermal equilibrium between the Nanoparticles and base fluid maintained as follows:

$$Cp_{nf} = \frac{(1-\phi)\rho_f Cp_f + \phi\rho_p Cp_p}{\rho_{nf}} \quad .(9)$$

Using this relation a prediction of small decrement in specific heat for 3% Al_2O_3/water within a range of 7-8% to that of water. Due the size dependent that being noticed [26-29] a relation adapted of Yang and Zhang [31] :

$$(Cp)_{bf} = \frac{(1-\phi)(\rho Cp)_{bf} + \phi(\rho Cp)_p}{(1-\phi)\rho_{bf} + \phi\rho_p} \quad .(10)$$

Thermal Conductivity

The effective thermal conductivity of Nanofluids based on an empirical model of . Maxwell [31] on assumption that a spherical particles shape dispersed a supporting field:

$$\frac{k_{nf}}{k_f} = 1 + \frac{3(\alpha-1)\phi}{(\alpha+2)-(\alpha-1)\phi} \quad , \quad \alpha = \frac{k_p}{k_f} \quad ....(11)$$

And $k_{nf} = k_{bf}\left(4.97\phi^2 + 2.7\phi + 1\right)$

for Water / γ-Al_2O_3

For two other effecting physical parameter the shape of the particles and the effect of the particle motion in the fluid. A brownian type random movement being introduced by Hamilton and Crosser[32]. The Maxwell correlate becomes:

$$\frac{k_{nf}}{k_f} = \frac{\left[k_p + (n-1)k_f - (n-1)\phi(k_f - k_p)\ \right]}{\left[k_p + (n-1)k_f + \phi(k_f - k_p)\ \right]} \quad (12)$$

Where the parameter n is the 'shape factor' defined as : $n = \frac{3}{\psi}$, where ψ called the (Sphericity), is defined as the ratio of the surface area of the sphere to that of the particle for the same volume. For spherical particles ψ = 1, and for the cylinders ψ = 0.5. Jang and Choi [33] found that the Brownian motion of Nanoparticles at the molecular and Nano scale level is a key mechanism governing the thermal behavior of Nanofluids .

3. Pure and Nanofluid Temperature

In The Nanofluid and pure fluid cooling MCHS, the energy absorbed by the working fluid can be written as :

$$Q_{nf} = (\rho Cp)_{nf}\,\dot{V}_{nf}\left(T_{nfo} - T_{nfi}\right) = (\rho Cp)_{nf}\,\dot{V}_{nf}\,\Delta T_{nf} \quad ..(13)$$

$$Q_f = (\rho Cp)_f\,\dot{V}_f\left(T_{fo} - T_{fi}\right) = (\rho Cp)_f\,\dot{V}_f\,\Delta T_f \quad ..(14)$$

noting that the thermal capacity of the Nanofluid is usually smaller than that for the pure fluid because of the low specific heat of Nanoparticles .

From Newton's law of cooling, the local heat transfer coefficient for both Nanofluid and pure fluid cooled MCHS can be written as:

$$q'' = h_{nf}(x)\left(T_{snf}(x) - T_{nf}(x)\right) = h_f(x)\left(T_{sf}(x) - T_f(x)\right) \quad ....(15)$$

$$h_{nf}(x) = \frac{q''}{\left(T_{snf}(x) - T_{nf}(x)\right)}, \quad h_f(x) = \frac{q''}{\left(T_{sf}(x) - T_f(x)\right)} \quad ....(16)$$

in order to established a heat transfer enhancement by the Nanofluid If the , h_{nf} (x) > h_f (x) and Q_{nf} >Q_f conditionally $\dot{V}_{nf}$ = $\dot{V}_f$ and T_{nfi} = T_{fi} , we observed that :

$$\left(T_{nfo} - T_{nfi}\right) > \left(T_{fo} - T_{fi}\right) \quad ....(17)$$

$$\left(T_{snf}(x) - T_{nf}(x)\right) < \left(T_{sf}(x) - T_{f}(x)\right) \quad (18)$$

by Combining Eqs. (27) and (28), it can be found that :

$$T_{snf}(x) < T_{sf}(x) \quad ....(19)$$

This result indicates that Nanofluid can cool MCHS better that pure fluid.

Results and Discussions

The calculation being carried by using the FLUENT code (Ansys 12) and the mesh that has been established as mentioned earlier. These calculations established for two mass flow rates that maintained a laminar flow through the channel. Then three heat flux imposed on the channel accordingly there mixture of nanofluid and pure water being used.

The temperature distribution across the cross sectional area and respectively with its length has been estimated as shown in Figure (3) to (8) for are the temperature contours of the outlet of the channel a mentioned conditions. While the profile along the channel presented in Figure (9) and (10) and table (4). As noted on each the related condition.

The contucted results from the contours of the temperature distribution in highest in contact with channel wall and concaving toward the center of the channel, while the temperature profile along the channel decreasing with respect to either increasing nanofluid concentration or the increasing velocity. Then the velocity profile for each at the exit shown in figures (11-16).

Concerning the heat transfer coefficient a three regions along the channel being noticed for the pure water to that of the mixture nanofluids so that a rapid slope ends at 1 mm for water and the mixtures but the intermediate range for water ends at 6 mm while the mixtures ends at 7 mm, furthermore length being steady sate meaning a better heat transfer coefficient for the mixtures from that of pure water Figure(17). similarly the Nusselt number confirm this type of behavior but Nusselt resolve very clearly the transfer enhancement from the lower concentration to the higher one as shown in figure 18a,b,c. Never the less, to confirm this results with [34] shows that a close similarity in the result concerning Nusselt number Figure 19. In comparing to heat flux with both Nusselt number and Reynold number shows the higher the heat flux have its highest and a linearly variation with the increasing of both number and shows a lower value for a lower heat flux as given in figure(20).

Conclusions

The simulated flow inside a micro channel for Al_2O_3/water mixture with different concentration to conduct heat flux exposed from bottom side of the channel to be constant at 50 W/cm^2, 90w/cm^2 and 200 W/cm^2. The flow of the fluid assumed laminar in range of less than 1000 Reynold number, for concentrations of 1%, 3% and 5%. The outlet flow temperatures obtained from the calculations for both of flow rate which is mentioned previous and heat fluxes (50 W/cm^2 and 90 W/cm^2) is less than 80 degree centigrade while greater than 80 degree centigrade at 150 W/cm^2 for the same conditions**.** Also From the review of the Nanofluid thermal conductivity study in the literatures, it is seen that there is a significant discrepancy in experimental data. This discrepancy may be due to some specific parameters of Nanofluids such is clustering of Nanoparticles, particle size distribution of Nanoparticles, duration and severity of ultrasonic vibration applied to the

Nanofluid, and pH value of the Nanofluid.

The analysis performed, provides a-fundamental understanding of the combined flow and conjugate convection–conduction heat transfer in the three-dimensional Microchannel heat sink. Therefore, the results of the analysis as well as the conclusions can be considered as quite general and applicable to any three-dimensional conjugate heat transfer problems.

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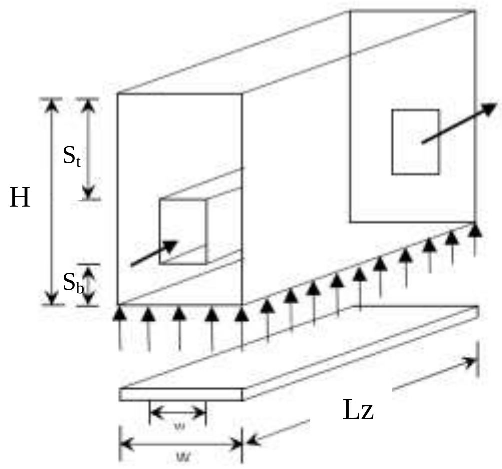

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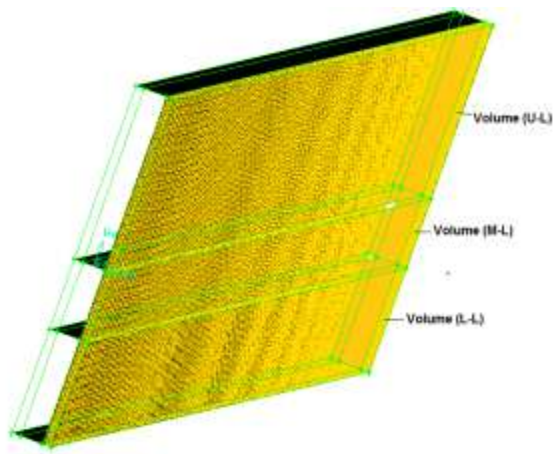

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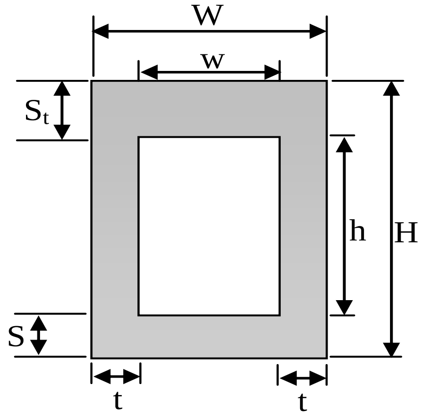
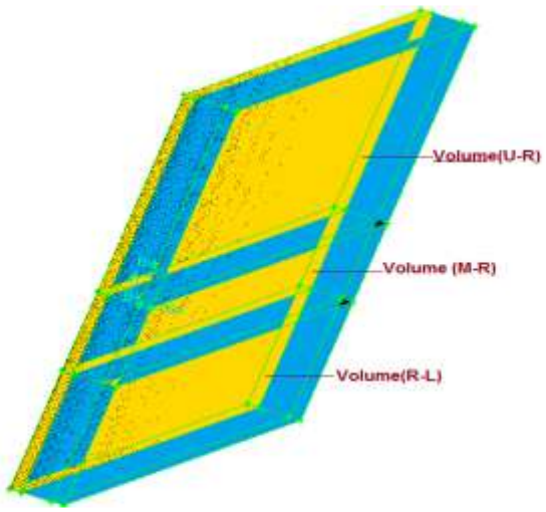


(c)

Figure(2)Schematic diagram of Single Microchannel in GAMBIT.



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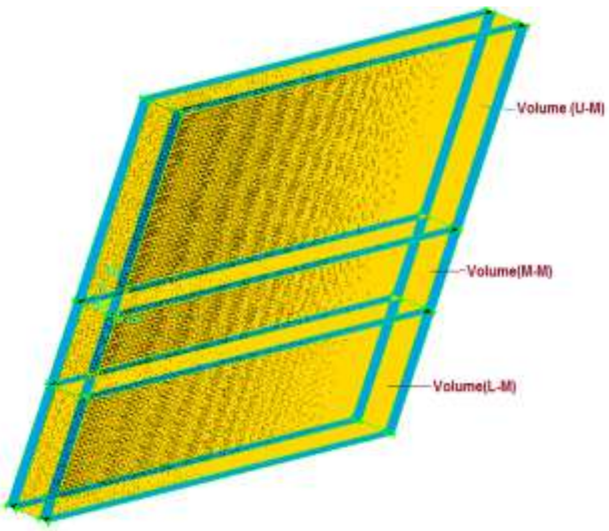

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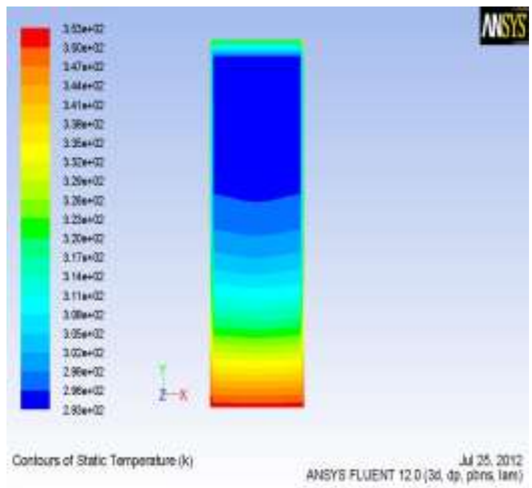


Fig. 3 Temperature contour of Outlet at Φ=0% and at 50W/cm^2, $\dot{m}=1x10^{-5}kg/s$.

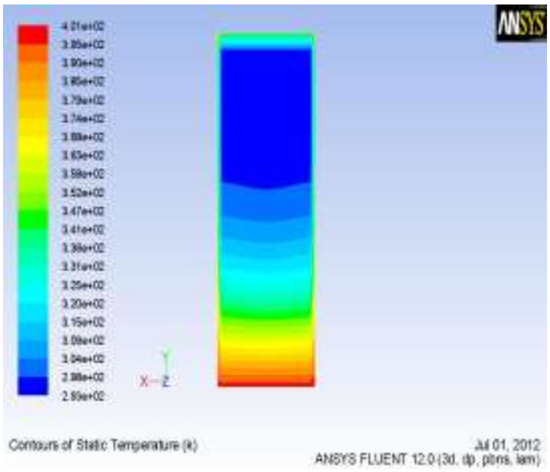


Fig. 4 Temperature contour of Outlet at Φ=0% and at 90W/cm^2 , $\dot{m}=1x10^{-5}kg/s$

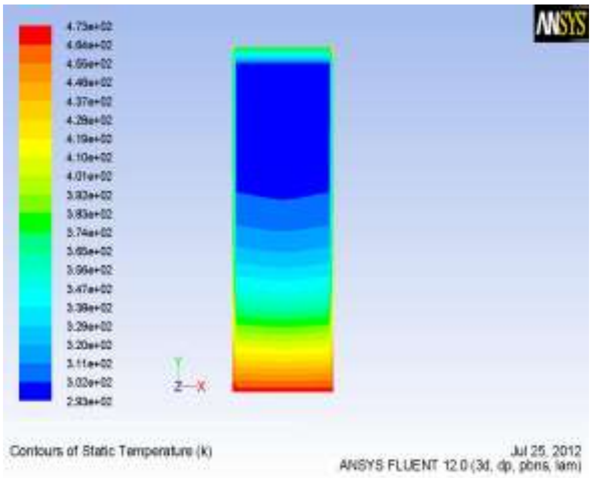


Fig. 5 Temperature contour of outlet at Φ=0% and at 150w/cm^2 , $\dot{m}=1x10^{-5}kg/s$.

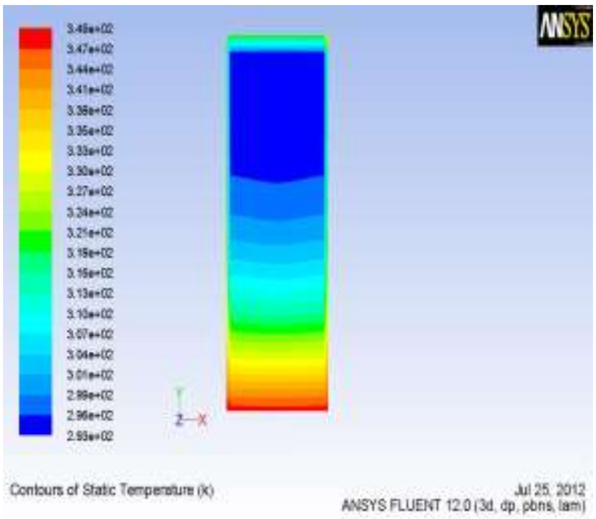


Fig. 6 Temperature contour of Outlet at Φ=5% and at 50w/cm^2 , $\dot{m}=1x10^{-5}kg/s$.

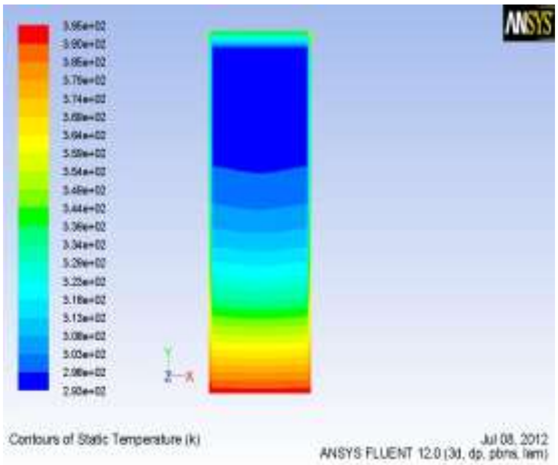


Fig. 7 Temperature contour of Outlet at Φ=5% and at 90W/cm^2 , $\dot{m}=1x10^{-5}Kg/s$.

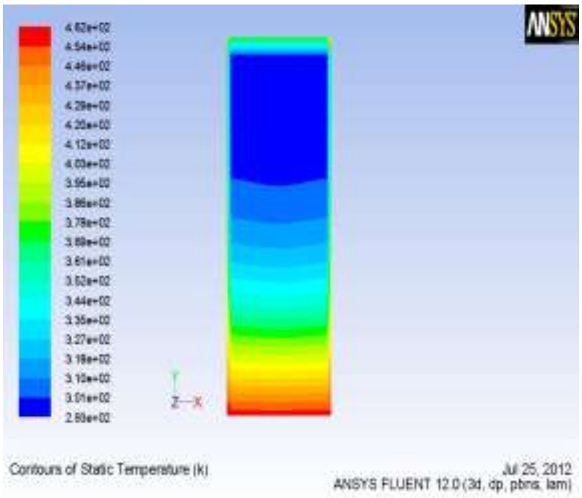


Fig. 8 Temperature contour of outlet at Φ=5% and at 150w/cm^2 , $\dot{m}=1x10^{-5}kg/s$.

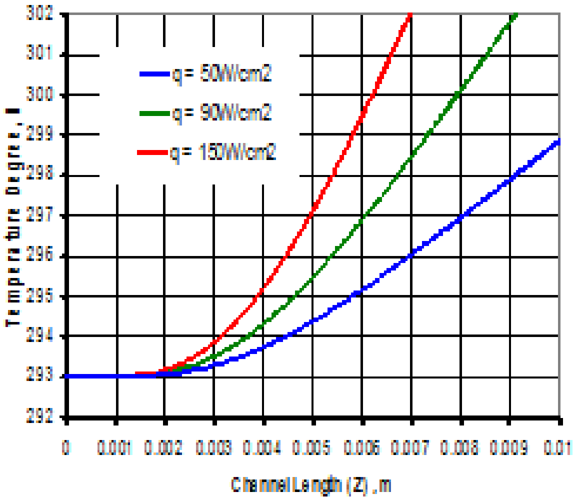


Fig. 9 Temperature profile of fluid at at 5% Alumina , $\dot{m}=1.10x10^{-5}kg/s$.

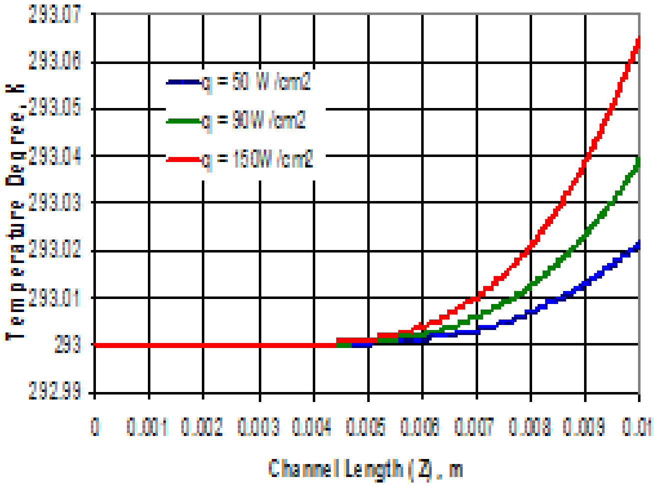


Fig. 10 Temperature profile of fluid at at Φ= 5% Alumina , $\dot{m} = 5.12x10^{-5} kg/s$.

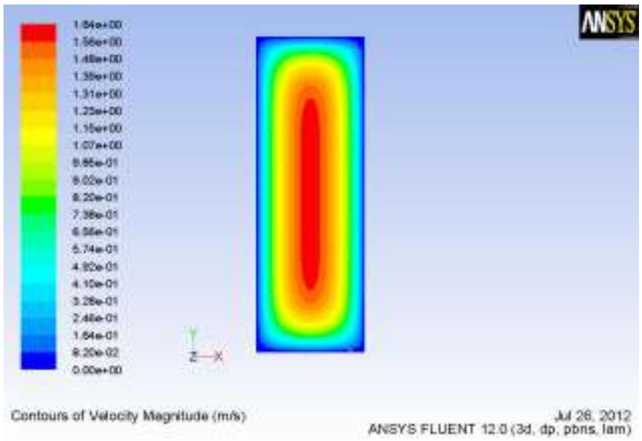


Fig. 13 Velocity contour of Outlet at V= 0% and at 90w/cm^2 , $\dot{m} = 1x10^{-5} kg/s$.

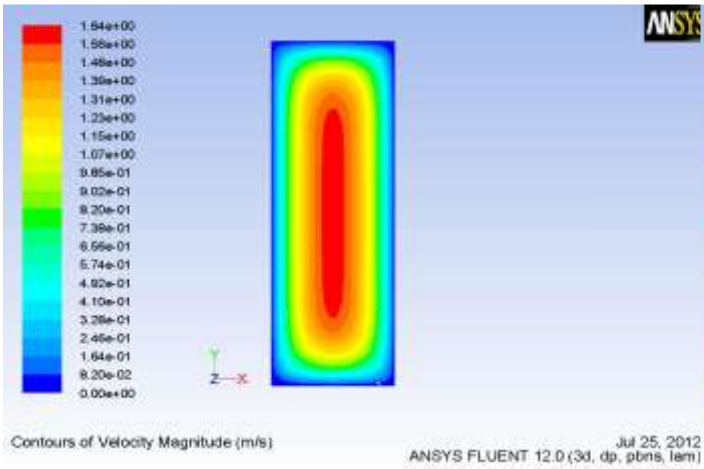


Fig. 11 Velocity contour of outlet at Φ= 0% and at 50W/cm^2 , $\dot{m} = 1x10^{-5} kg/s$.

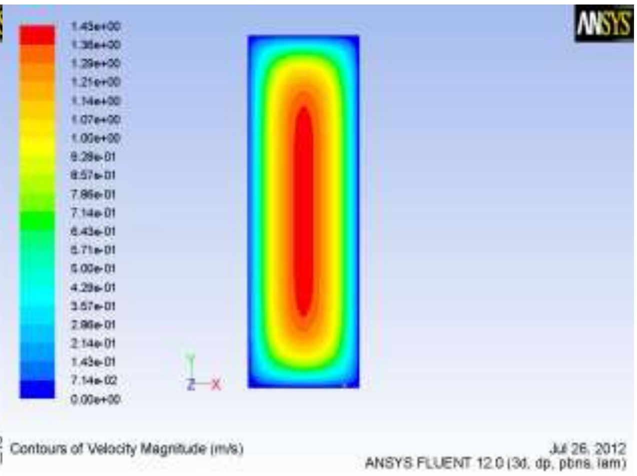


Fig. 14 Velocity contour of Outlet at Φ= 5% and at 90w/cm^2 , $\dot{m} = 1x10^{-5} kg/s$.

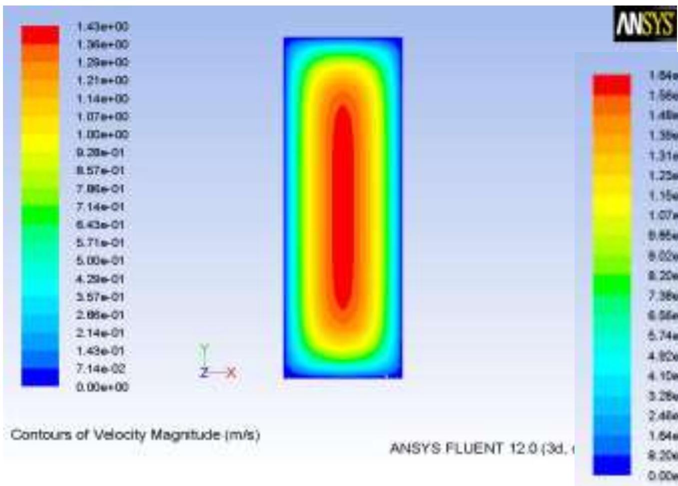


Fig. 12 Velocity contour of Outlet at, Φ= 5 at 50w/cm^2 . $\dot{m} = 1x10^{-5} kg/s$

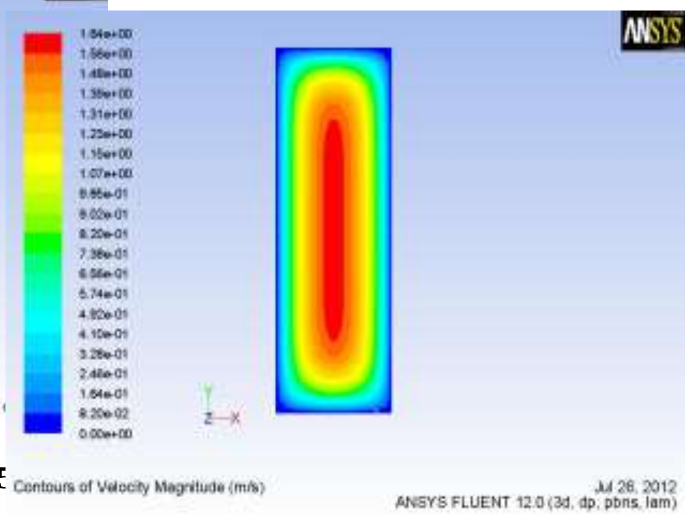


Fig. 15 Velocity contour of Outlet at Φ= 0% and at 150w/cm^2 , $\dot{m} = 1x10^{-5} kg/s$.

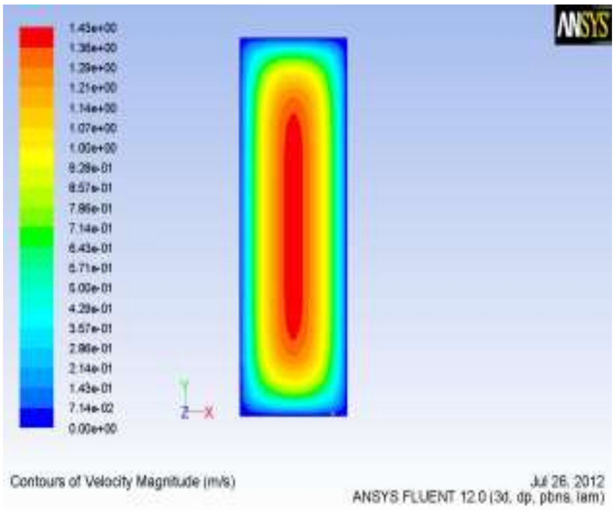


Fig. 16 Velocity contour of Outlet at Φ= 5% and at 150w/cm^2 , $\dot{m}=1x10^{-5}kg/s$.

Table (1): Dimensions of the of the designed cooling model

W	100µm
w	57µm
H	900µm
h	180µm
t	21.5µm
S_t	450µm
S_b	270µm
L_z	10000µm
D_h	86 µm

Table(2).Volume and Node distribution

Volume	L-L	L-M	L-R	M-M	M-R
Node	43416	188136	43416	109746	25326
Element	35000	175000	35000	100000	20000
Volume	M-L	U-L	U-M	U-R	
Node	25326	61506	266526	61506	
Element	20000	50000	250000	50000	

Table (3).Estimate Parameters for Nanofluid Concentration

$\dot{m}=1\times10^{-5}kg/s$	Heat Flux=50 w/cm^2		
property	*Φ=0*	*Φ=3*	*Φ=5*
Temp.(K)	353	351	349
Re	83.57	62.83	49.95
***V*(m/sec)**	0.9764	0.8945	0.8482

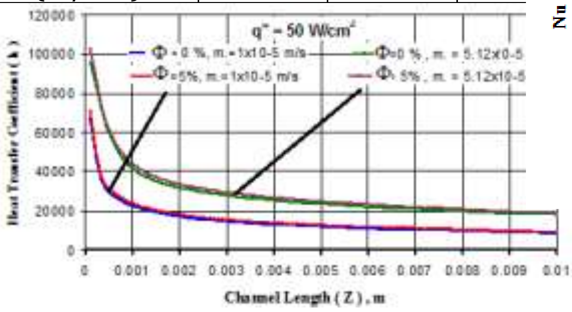
$\dot{m}=5.21\times10^{-5}kg/s$	Heat Flux=50 w/cm^2		
property	*Φ=0*	*Φ=3*	*Φ=5*
Temp.(K)	325	324	323
Re	427.9	321.7	8255.
***V*(m/sec)**	5	4.58	4.34

$\dot{m}=1\times10^{-5} kg/s$	Heat Flux=90 w/cm^2		
property	*Φ=0*	*Φ=3*	*Φ=5*
Temp.(K)	401	397	395
Re	83.57	62.83	49.95
***V*(m/sec)**	0.9764	0.8945	0.8482

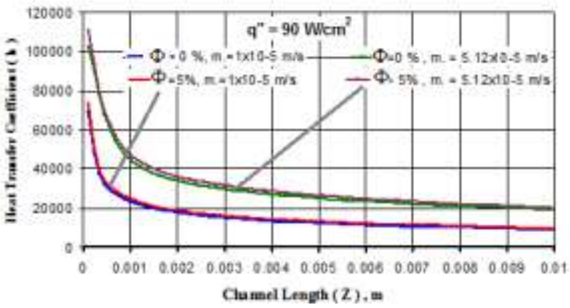
$\dot{m}=5{,}12\times10^{-5} kg/s$	Heat Flux=90 w/cm^2		
property	*Φ=0*	*Φ=3*	*Φ=5*
Temp.(K)	350	348	347
Re	427.9 4	321.7	255.
***V*(m/sec)**	5	4.58	4.31

$\dot{m}=5.12\times10^{-5} kg/s$	Heat Flux=150 w/cm^2		
property	*Φ=0*	*Φ=3*	*Φ=5*
Temp.(K)	388	385	383
Re	427.	321.	255.
V(m/sec)	5	4.5	4.3

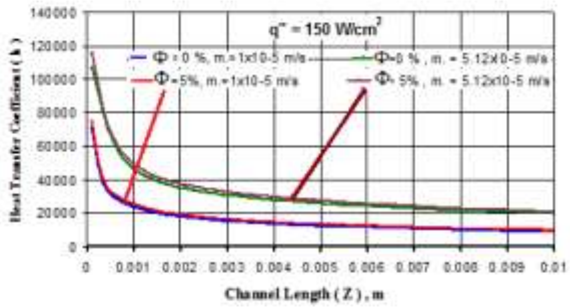
$\dot{m}=1{,}0\times10^{-5} kg/s$	Heat Flux=150 w/cm^2		
Property	*Φ=0*	*Φ=3*	*Φ=5*
Temp.(K)	473	467	462
Re	83.5	62.83	49.9
***V*(m/sec)**	0.97	0.89	0.85



(a)

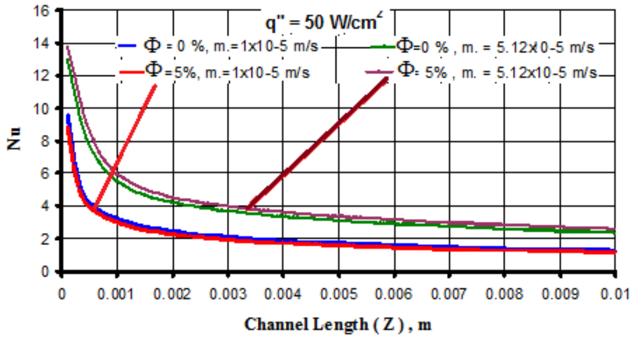


(b)

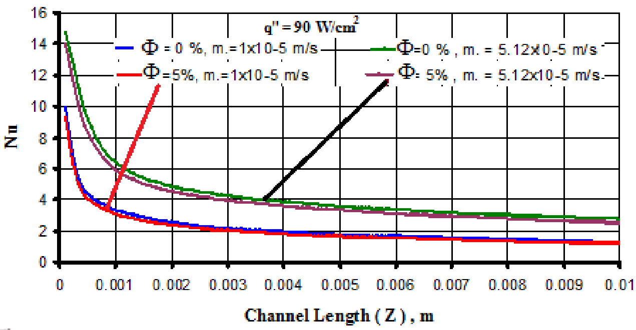


(c)

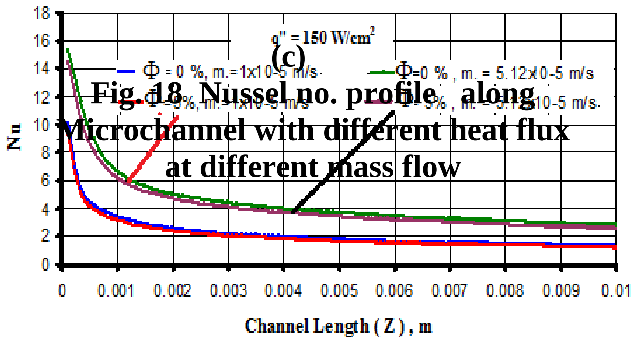
Fig. 17 Heat transfer coefficient profile along Microchannel with different at flux at different mass flow



(a)



(b)



(c)

Fig. 18 Nussel no. profile along Microchannel with different heat flux at different mass flow

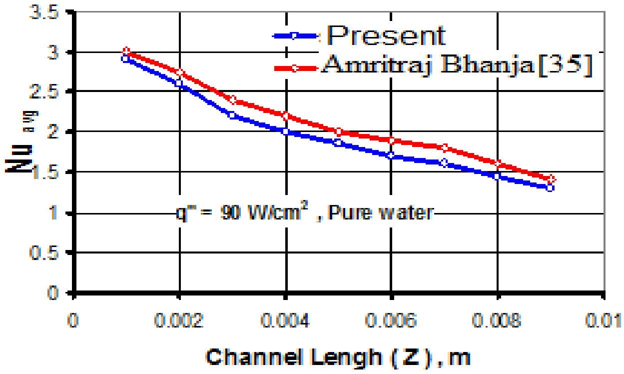


Fig. 19 Comparison of Nusselt Number profile along Micro channel

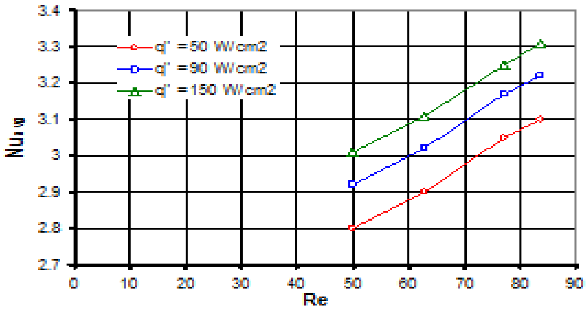


Fig. 20 Variation of average Nu with Re at different heat flux and $\dot{m} = 1x10^{-5} kg/s$.