# EFFECT OF ARTIFICIAL CAVITIES ON HEAT TRANSFER AND FLOW CHARACTERISTICS MICROCHANNEL

**Suha A. Mohammed** **Ekhlas M. Fayyadh**
shawsuha@yahoo.cm 20084@uotechnology.edu.iq
**Mechanical Engineering Department, University of Technology, Baghdad, Iraq**

## ABSTRACT
An experimental investigation was conducted to study single-phase fluid flow and heat transfer in a copper micro channel. To investigate the effect of artificial cavities on fluid flow and single phase heat transfer in micro channel heat sink, two model of straight micro channel recognized as two models (model -1and model -2) were designed and manufactured ,where model-1 have smooth bottom surface while Model-2 have 47 artificial cavities distributed uniformly at the bottom surface along the micro channel length. The two models having the same nominal dimension of 300µm height and 300µm depth while the real dimension value are 367 µm for width and 296 µm for depth .De-ionized water was used as the working fluids. Experimental test was conducted at 30$^{o}$C inlet temperature with Reynolds numbers range from 700 to 2200 covering laminar flow conditions. The experiments were conducted with horizontal micro channel under both adiabatic (for friction factor calculation) and diabatic (for Nusselt number calculation) conditions. The results indicated that the experimental Darcy friction factor can be predicted well with conventional scale fanning friction factor correlations for developing flow in laminar region by shah and London (1978) correlation for two models. Also, the experimental Nusselt number Agree well with each correlation of shah and London (1978) and Mirmanto correlation in laminar region.

**Keywords: Micro channel, Pressure drop, heat transfer, de-ionized water, laminar flow**.

# دراسة تأثير الفجوات الافتراضية على خصائص الجريان وانتقال الحرارة في القناة المايكروية

**سهى عبد الاله محمد** **اخلاص محمد فياض**

## الخلاصة

تم في البحث الحالي اجراء دراسة تجريبية للجريان احادي الطور وانتقال الحرارة لقناة مايكرويه مفردة. لدراسة تأثير التجاويف الصناعية على الجريان وانتقال الحرارة أحادي الطور لقناة مايكروية، تم تصميم وتصنيع نموذجين للقناة المايكروية من النحاس النقي. النموذج الاول يمثل قناة مايكروية ذات سطح املس اما النموذج الثاني فيمثل قناة مايكروية يحتوي سطح قاعدتها على 47 تجويفًا اصطناعيًا موزَّعًا بشكل منتظم على طول القناة. كلا النموذجين لهما نفس الابعاد التصميمية بارتفاع (300) مايكرون وعمق (300) مايكرون بينما قيمة البعد الحقيقي هي (367) مايكرون للعرض و (296) مايكرون للعمق. تم استخدام المياه غير المؤينة كمادة للعمل التجريبي عند درجة حرارة دخول 30 درجة مئوية مع أرقام رينولدز تتراوح من (700 إلى 2200) تغطي ظروف التدفق الطباقي. بينت النتائج العملية أن معامل الاحتكاك دارسي يمكن التنبؤ به جيدًا من خلال العلاقات التقليدية للباحثين shah and London (1978) للجريان الطباقي في طور النمو وايضا كانت النتائج العملية لعدد ناسلت تتفق بشكل جيد النتائج الباحثين shah and London (1978) وايضا Mirmanto (2012.

**Received : 5-8-2019**
**Accepted :22-9-2019**

## INTRODUCTION

Rapid growth in micro-fluidic system leads to great attention in micro scale channels. The micro scale channel is demand for miniaturized, lightweight, compact, and high efficiency of heat transfer, becomes increasing necessary due to the increase of heat production, space limitation, and materials saving Mesbah et.al. (2011). The validity of the conventional flow characteristics and heat transfer theories in micro channel passage is one of the most important issue that deals in previous published literatures .Their published results have been discrepancy, where the measured friction factor and heat transfer coefficients have either well exceeded as presented by Choiet al (1991), or fallen far below as indicated by Peng, et al (1994) from the conventional scale channels correlation. Many factors cause the deviation from classical theory. Where Mala et. al. (1997) attributed the deviation reason to electrical double layer while the aspect ratio may effects these deviation as presented by Papautsky et. al.(2000). Also, Mirmanto et al. (2012) conducted sets of experiments in a single copper micro channel to examine the effect of aspect ratio and hydraulic diameter on pressure drop and heat transfer. Deionized water was tested at range of Reynolds number. Results show that the measured Nusselt number and friction factor were found to be higher than the conventional scale fully developed conditions. Also, they concluded that when entrance effects, experimental uncertainties, inlet and exit pressure losses, and departure from laminar flow were considered, the results indicated that equations developed for conventional scale flow are applicable for water flows in micro channels of that size. While Garimella et al. (2001) measured the effect of development flow on the Nusselt number of three different tube sizes and shapes. According to the geometry range studied, the effect of aspect ratio was not significant. Mala and Lee (1999) attributed the reason of the deviation from conventional scale channels correlation to the surface roughness. In addition to the surface roughness effect, the crosssectional shape of the channel and hydrophilic surface can have great influence on pressure drop and heat transfer as found by Wu and Cheng (2003). They tested 13 silicon trapezoidal microchannels of different hydraulic diameter, aspect ratio and relative roughness using deionized water as a working fluid. They found that the laminar Nusselt number and apparent friction factor increased with the increased of surface roughness and surface hydrophilic. At low Reynolds number ($Re < 100$) Nusselt number increased almost linearly with Reynolds number but increased slowly at Reynolds number greater than 100. Also Kandlikar et al. (2003) investigated the effect of surface roughness on pressure drop and heat transfer for stainless steel tubes with circular mini and micro tube using distilled water as working fluid .The roughness of tubes was changed by etching them with an acid solution to get three different roughness value of each tube .They found that the surface roughness results negligible effect on pressure drop and heat transfer for the larger diameter case. However, the surface roughness effect is significant for the smaller diameter. Also, it was found that the surface roughness and tube diameter have no effects on flow transition. Salem et al. (2013) Conduct set of experiment in circular stainless steel micro tube to investigate the effect of surface roughness on friction factor and heat transfer using distilled water and R134a as a working fluid .It was found that the result have shown good agreement with the conventional theory in the laminar region for both friction factor and Nusselt number. Although early transition has been observed. At turbulent region the results of friction factor can be predicted well with conventional theory while the measured Nusselt numbers are lower than those predicted. The researchers focused on single phase heat transfer enhancement in micro channel rather than flow boiling heat transfer due to the higher pressure drop implemented with two phase flow beside the flow boiling system would require a condensation step in the closed loop system Steink and Kandlikar(2006). In another study for Steink and Kandlikar(2004) reviewed and identified several possible techniques to enhance the heat transfer in micro channels .They discussed The possibility of applying conventional single-phase heat transfer enhancement techniques in micro channel and mini channel flow. Also, they listed passive and active techniques for single phase heat transfer enhancement. They mentioned that alter the surface characteristics with increasing surface roughness cause to reduce the thermal boundary layer thickness. Surface roughness effect was also studied by Xing et al (2016) in three stainless steel circular micro channels having different surface roughness but with the same diameter using air as working fluid. The authors observed that the high relative roughness results higher Poiseuille and Nusselt number. However, the friction factor and Nusselt number increases with increasing Reynolds number.

This study focus on using one of the passive technique method for enhance the heat transfer in conventional smooth micro channel by drilling artificial cavities .Where two square micro channel recognized as (model-1 and model-2) were used as a test sections to study the fluid flow and heat transfer characteristics of water at 30˚C Accordingly, the friction factor and Nusselt number were tested respectively with Reynolds number varied from 700 to 2200, covering the laminar region and compare the results with conventional scale correlation.

## EXPERIMENTAL APPARATUS AND DATA REDUCTION

### Experimental setup

The experimental facility consists of liquid tank, sub-cooler, peristaltic pump, turbine flow meter, pre-heater, test section, and inline filter. Schematic diagram and a plate of the experimental set-up are shown in Figs. 1 and 2, respectively. A chiller unit is used for cooling purpose in the condenser and sub-cooler. De-ionized water was used as a working fluid. The de-ionized water was degassed in the liquid tank by vigorous boiling for approximately one hour. The non-condensable gases were released to the ambient by opening the valve located on top of the condenser. 7 µm filters was installed before the peristaltic pump in the system to remove any particles in the water. The de-gassed water was then pumped to the test section and the fluid inlet temperature was controlled using a preheater. The micro channel test section is designed and machined from oxygen free copper blocks, hence, the block of the micro- channel test section has dimensions of 12 mm width, 25 mm height and 72 mm length .Single microchannel having length of 62 mm was cut into the top surface of the copper blocks between the 2 mm diameter inlet and outlet plenums using milling machine with feed rate of 10 mm/min. The nominal dimensions of the micro channel are 300 µm width and 300 µm depth. There dimensions were measured using an electro microscope and the actual values are (367 µm) for width and (269 µm) for height as shown in Fig.3. The surface roughness at the bottom of the microchannel test sections was measured with AA3000 Scanning Probe Microscope (Atomic force Microscope AFM Contact mode) which have multi–analysis: granularity and roughness. The average surface roughness value is 0.011µm. Fig. 4 shows the roughness values that were evaluated over sample areas of 2cm x 1cm.To provide the heating power to the test section, a cartridge heater of 250 W heating power was inserted at the bottom of the copper block in a direction parallel to the flow that is located in a drilled (8 mm) hole through the copper block. The local axial wall temperature was measured in the 1 mm inner diameter and 6 mm depth hole at the side of copper block. To measure the local heat transfer coefficient along the channel, six holes located along the axial direction of channel at 12.4 mm equidistance and 1 mm from the channel bottom to accommodate K – type thermocouples .Also , from entrance of channel with distance of 24.8 mm, two holes were located vertically and 5 mm below the axial holes with interval 5 mm .Additionally , an o-ring slot was machined on the top surface of the microchannels to upper surface of the copper block and the stainless steel cover plate. Then two holes were drilled at two locations in the top cover polycarbonate to accommodate thermocouple, one of them at inlet plenum while other at outlet plenum of the micro channel test section as well to accommodate differential pressure drop across the micro channel test sections Fig. 5 present the main part of the test section. All thermocouples were calibrated during the present study with an uncertainty of ± 0.5 K. The pressure drop was measured using a differential pressure transducer (26pcffT6D) which was calibrated with an uncertainty value of ±0.4 kPa. All data were recorded after steady state condition for 5 min using Applent AT4532x data acquisition system.

## MODELS OF THE MICRO-CHANNEL:

Two models of microchannel were manufactured and tested named model-1 which is the plain microchannel and model-2 the enhanced microchannel. The artificial cavities in model-2 have been drilled along the center line at the bottom of the micro-channel block with radius of 25 μm. The interval distance between center to center artificial cavities was regular with a value of 1 mm. Fig. 6 present microscope image for the two models

## DATA REDUCTION

The net pressure drop along the micro channel $\Delta P_{ch}$ is given by:

$$\Delta P_{ch} = \Delta P_{meas} - \Delta P_{loss} \qquad (1)$$

$\Delta P_{meas}$ is the overall pressure drop between the channel inlet and outlet plenums. It is measured directly using the differential pressure sensor. $\Delta P_{loss}$ defined by Eq.2 below, is the pressure loss due to the inlet and outlet plenum and the sudden contraction and enlargement.

$$\Delta P_{loss} = 2\left(\frac{1}{2}\rho_l V_p K_{90}\right) + \frac{1}{2}\left(\rho_l V_{ch}(K_c + K_e)\right) \qquad (2)$$

In Eq. (2), $V_p$ and $V_{ch}$ are the liquid velocities in the plenum and in the channel respectively. The flow enters and leaves the channel in a direction normal to the flow direction. Here, K90 is the loss coefficient due to 90° turns of the flow in inlet and outlet plenums which is determined as 1.2 by Philips (1987). Kc is the sudden contraction loss coefficient and Ke is the sudden enlargement loss coefficient for the channel inlet and outlet, respectively. Their values were interpolated from tabular information provided by Shah and London (1978). The Fanning friction factor based on the channel pressure drop is given by:

$$f_{ch} = \frac{\Delta P_{ch} D_h}{2 L \rho V_{ch}} \qquad (3)$$

The heat loss from the test section ($q_{loss}$) was estimated by applying an electrical power (P) to the test section when there is no fluid inside it. The temperature difference between the bottom wall and ambient was recorded for each heating power after attaining steady state. The applied power was then plotted versus this temperature difference and the data were fitted to obtain an equation to calculate the heat loss (qloss) this method was stated also by Lee and Garimella (2008). The rate of heat removal, $q_{rem}$, is determined as follows:

$$q_{rem}=P-q_{loss} \qquad (4)$$

$$P=V*I \qquad (5)$$

Where, V and I is the electrical voltage and electrical current respectively. The heat flux at the microchannel walls is defined as:

$$q''=q_{rem}/A_{ht} \tag{6}$$

$$A_{ht}=(2H+w)*L \tag{7}$$

Where H,W, and L represent the channel depth , width and length respectively. The local single-phase heat transfer coefficient ($h_{sp}(z)$) and average Nusselt number (Nu) are calculated as

$$h_{sp}(z) = \frac{q''}{T_w(z) - T_f(z)} \tag{8}$$

$$Nu = \frac{1}{L}\int_0^L \frac{h_{sp}(z)\, Dh\, dz}{k_l} \tag{9}$$

where the channel wall temperature ($T_{w(z)}$) at axial location z was corrected using the 1D heat conduction equation as given by Eq.10 below. $k_l$ is the liquid thermal conductivity and $T_f(z)$ is calculated by Eq.11 below based on an energy balance assuming uniform heat flux boundary conditions.

$$T_w(z) = T_{tc}(z) - \frac{q''\, t}{Kcu} \tag{10}$$

$$T_f(z) = T_i + \frac{q''\, wz}{m^{\cdot}\, C_p} \tag{11}$$

Where, $T_{tc}(z)$ is the local thermocouple reading, $k_{Cu}$ is the copper thermal conductivity and the distance from the channel bottom to the thermocouple location (t) is 1 mm. $T_i$ is the fluid inlet temperature and cp is the liquid specific heat. The uncertainties of the derived parameters are calculated using the method developed by Kline and McClintock (1953). Nucertainty in the heat transfer coefficient ($h_{sp}$) and Nusslet number ($Nu$) are estimated to be 2% and 2.3%, respectively.

## RESULTS AND DISCUSSIONS

### Friction factor

Experimental friction factors were obtained from the two models adiabatically (without heating applied to the test sections). As mentioned previously, the channel experimental pressure drop was obtained using Eq.1, by subtracting the inlet and exit losses from the total pressure drop measured between the inlet and exit plenums, and the corresponding friction factor was calculated using Eq.12. Figs. 7 and 8 represent experimental friction factors versus Reynolds number for model-1 and model-2 respectively at inlet fluid temperatures, T = 30˚C. The Figures show that increasing friction factor with increasing Reynolds number for both models. Also the figures indicated that model-2 have higher friction factor compared to model-1.because of increasing pressure drop due to flow resistance with the presence of artificial cavities. Also for each model the experimental friction factor were compared with a correlation of shah and London (1978) for developing flow as stated below:

$$f_{app.} = \frac{3.44}{Re\sqrt{L''_{sub}}} + \frac{f_{fd}Re + \frac{k}{4L''sub} - \frac{3.44}{\sqrt{L''sub}}}{Re\,(1 + C(L''^{-2}_{sub})} \tag{12}$$

$$L''_{sub} = \frac{L_{sub}}{Re\, D_h} \tag{13}$$

Also the experimental friction factor was compared with apparent friction factor for fully developed flow in laminar region for Shah and London (1987) as below:

$$f.f_d Re = 24(1 - 1.3553\beta + 1.9467\beta^2 - 1.7012\beta^3 + 0.9564\beta^4 - 0.2537\beta^5 \quad (14)$$

Where, β is the aspect ratio =1. Based on comparison with these conventional correlations, various conclusions have been drawn regarding their applicability to micro channel in predicting pressure drop in micro channels. Also, its seems that increasing surface roughness by drilling artificial cavities not cause to deviate the experimental friction factor result from the conventional correlation.

**Nusselt number**

The average Nusselt number are determined using Eq.15, Figs.9 and 10 represent the Reynolds number versus average Nusselt number for model-1 and model-2 respectively in the laminar flow regime. The figures show that the Nusselt number increases with increasing Reynolds number for each models. Figs. (9 and 10) also show the comparison of the experimental Nusselt number with the predictions from the correlations of Shah and London (1978) for developing and fully developed flow as stated below:

Fully developed laminar flow, Shah and London (1978):

$$\overline{Nu} = 8.235(1 - 10.6044\beta + 61.1755\beta^2 + 155.1803\beta^3 + 176.9203\beta^4 + 72.923\beta^5 \quad (15)$$

Developing laminar flow, Shah and London (1978):

$$\overline{Nu} = 0.775\ L^{*(-1/3)} Pr^{(1/3)} \quad (16)$$

Also, the experimental Nusselt number compare with correlation of Mirmanto (2012) in laminar region and Mehmed (2016) as stated below:

Developing laminar flow, Mirmanto (2012):

$$\overline{Nu} = Re^{0.283} Pr^{-0.513} L^{*-0.309} \quad (17)$$

$$L^* = \frac{L}{RePrD_h} \quad (18)$$

It is obvious that the experimental values show similar trend where Nusselt number increases with Reynolds number. However, the current experimental results agree very well with the experimental results of Shah and London (1978) for developing laminar flow. Its seems that the Nusselt number is higher for model-2 compare to model-1 which indicate to enhance the heat transfer rate with presence of artificial cavities due to increasing mixing and recirculation.

## CONCLUSIONS

Experimental study of single phase heat transfer and flow characteristics have been conducted at inlet temperature of (30 ˚C) for the friction factor and heat transfer of deionized water in two models of single copper micro channels of square cross-section. In the laminar region, the apparent friction factor is in reasonable agreement with the correlation of Shah and London (1978) for developing flow in laminar region. Also the experimental Nusselt numbers agree with developing flow of Shah and London (1978) in laminar region. The results indicate that the presence of artificial cavities not cause to deviate the experiments results of both flow characteristics and heat transfer from the conventional correlation.

1. The results indicated that the conventional theory is suitable to predict the single-phase flow friction factor for models of microchannels (1 and 2). Hence, Model-1(smooth surface) is predicted reasonably with the correlation of Shah and London (1978) for theory of developing flow with a Mean Absolute Error (MAE) of (13.6 %). While the model with the presence of artificial cavities has appeared a slight shift from the developing flow theory to the fully developed flow theory as Reynolds number is increased more than 1000.
2. The friction factor obtained from model-2 is higher than model-1 (smooth surface) with a percentage of (24.2%).
3. The average Nusselt number increases with Reynolds number for two models conversely to the fully developed heat transfer theory.
4. The correlations predicted reasonably the experimental data of Nusselt number for two models (1 and 2). Where, Shah and London (1978) correlations predicted the experimental data for Model-1 with a Mean Absolute Error (MAE=13.6%), and the Mirmanto (2012) correlation predicted the experimental data for model-2 with a Mean Absolute Error (MAE=9.5%).
5. There is enhancement of heat transfer performance for model with the presence of artificial cavities (Model -2 and Model-3), have higher Nusselt number values than Model-1(smooth) with value about (18%).

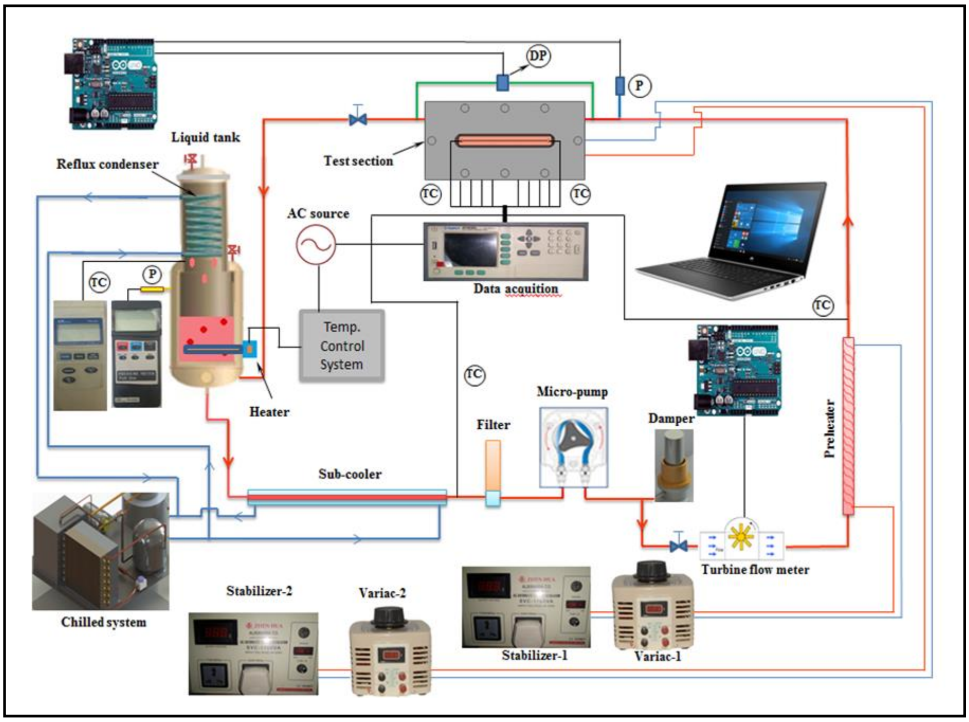


Fig.(1): Schematic diagram of the experimental facilities.

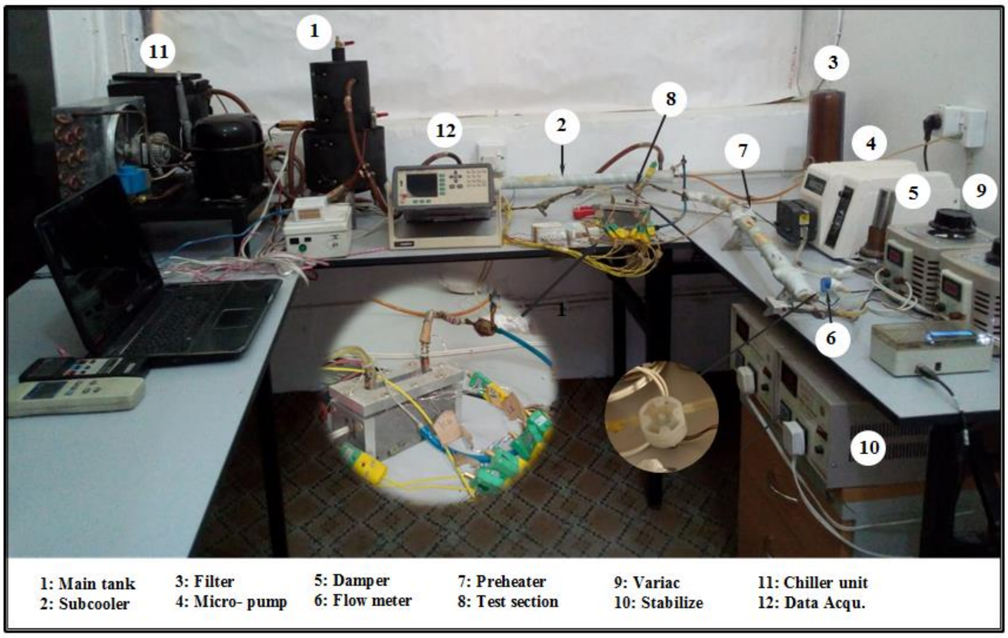


Fig.(2): The experimental facilities

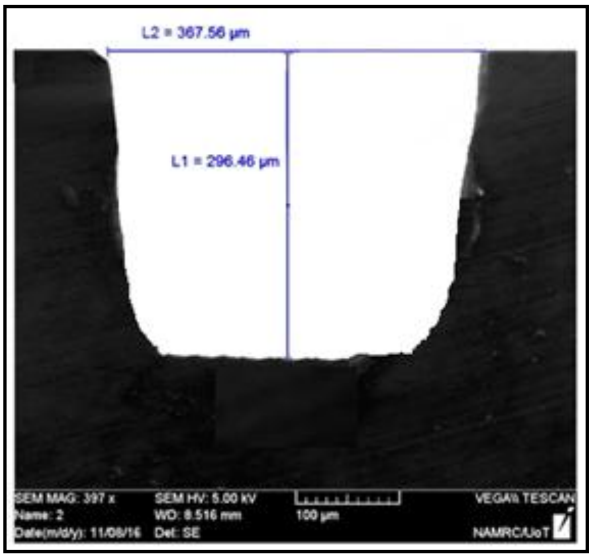


Fig. (3): Surface roughness measurements

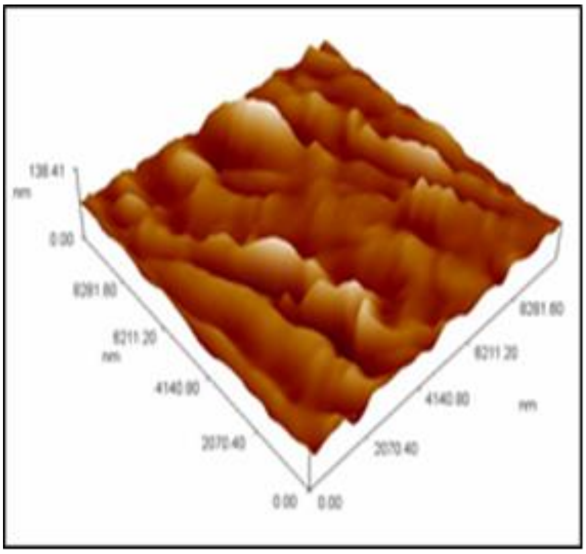

Fig.(4): Microscope picture of microchannel of test section.

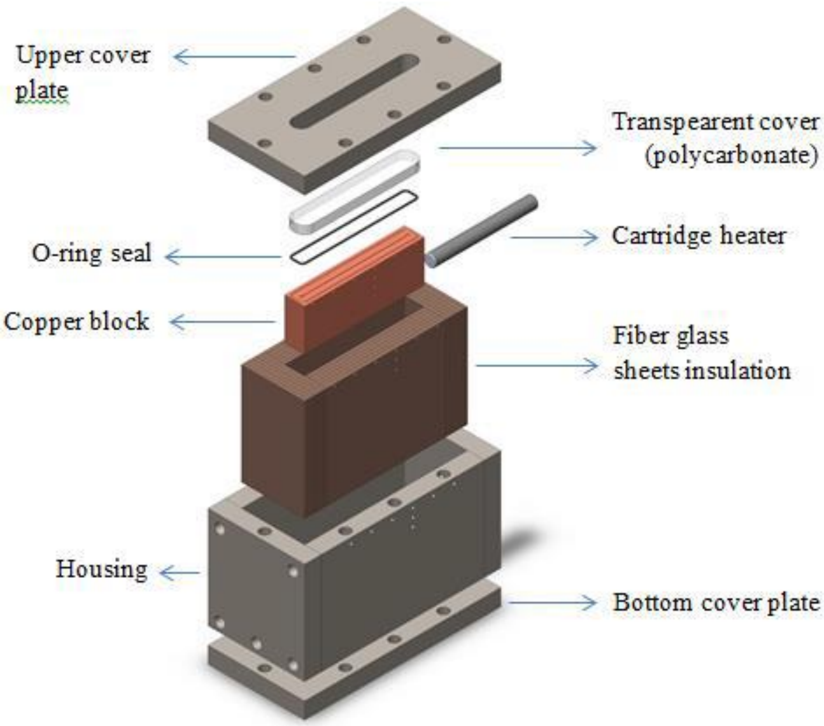


Fig. (5): Test section constructions showing the main parts

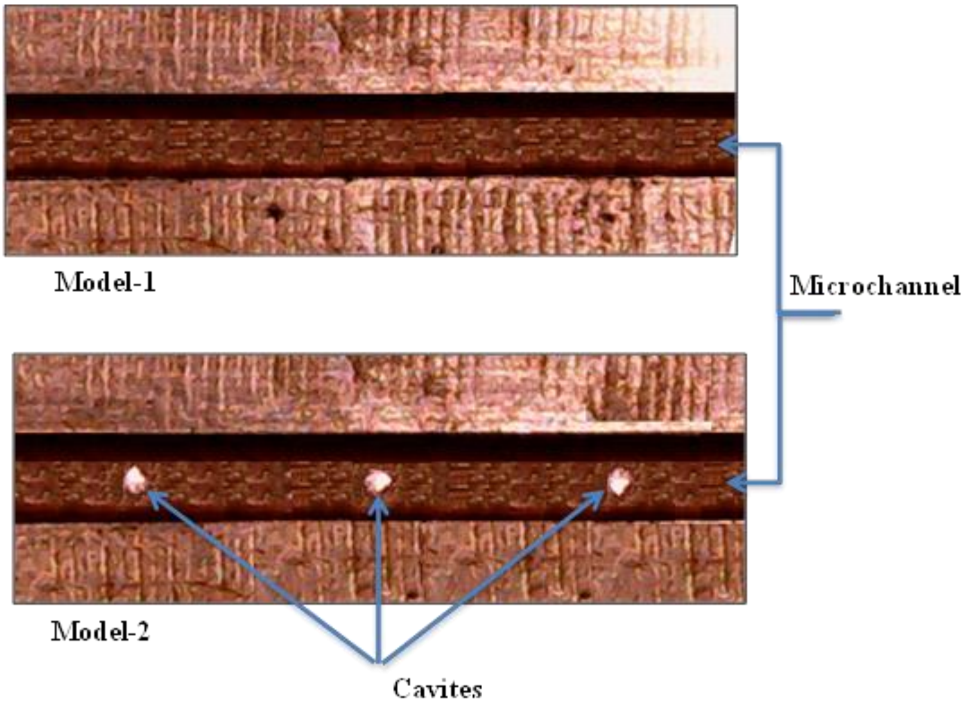


Fig.(6): Models of microchannel

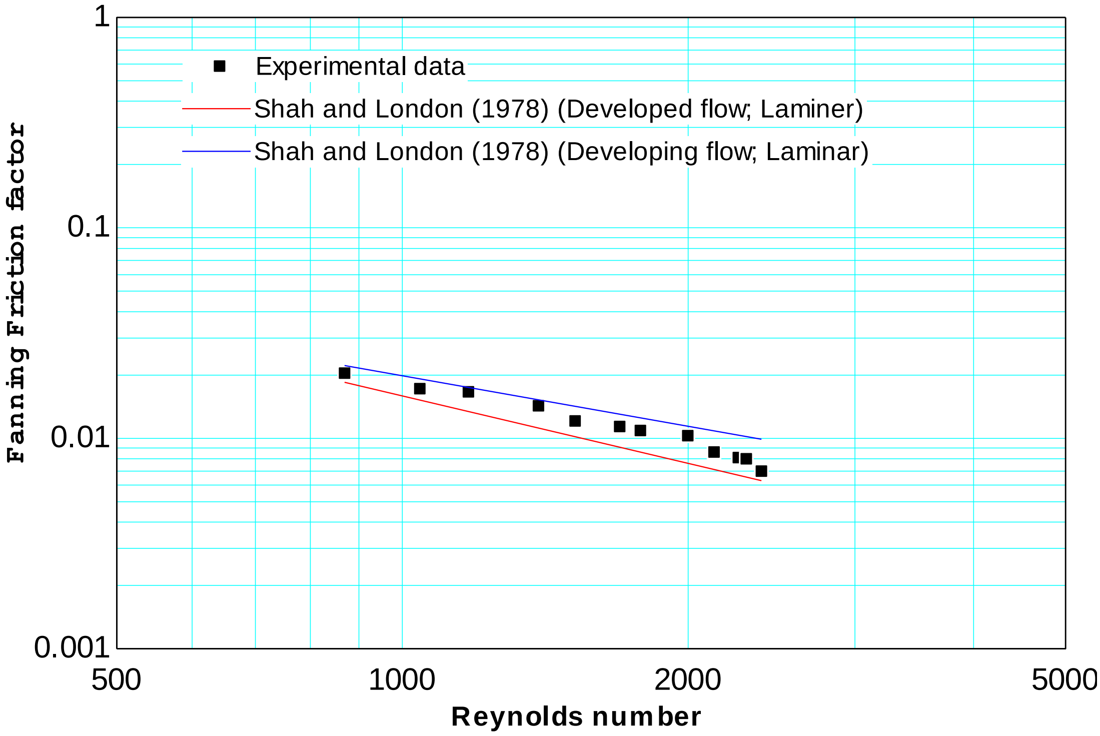


Fig.(7): Experimental fanning friction factor results compared with laminar flow correlations for a Model-1.

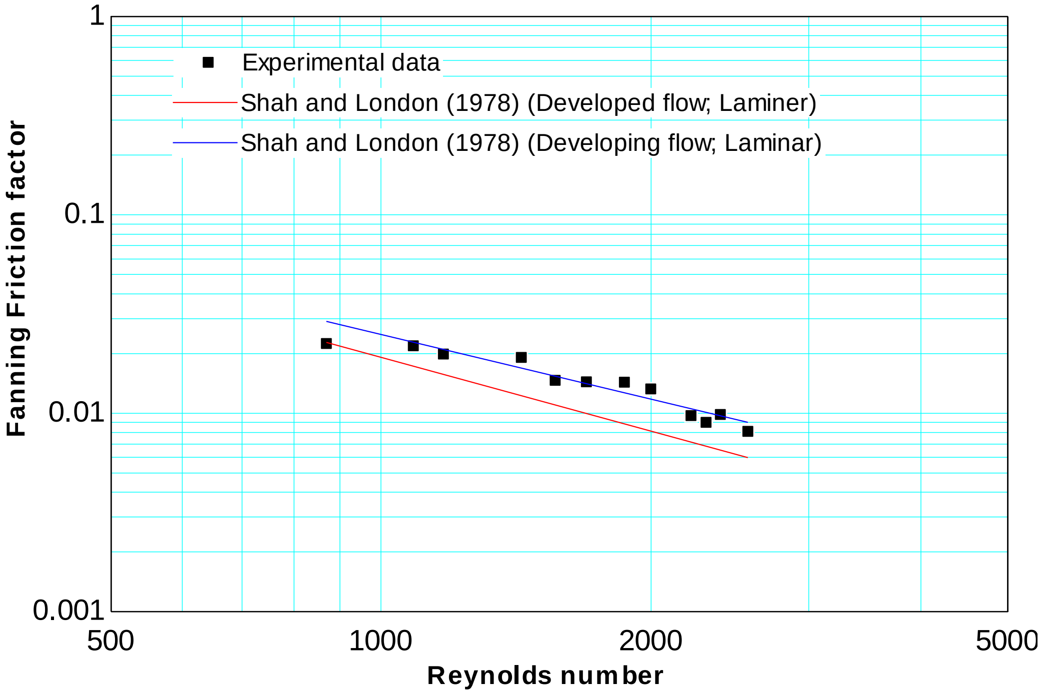


Fig. (8): Experimental fanning friction factor results compared with laminar flow correlations for a Model-2.

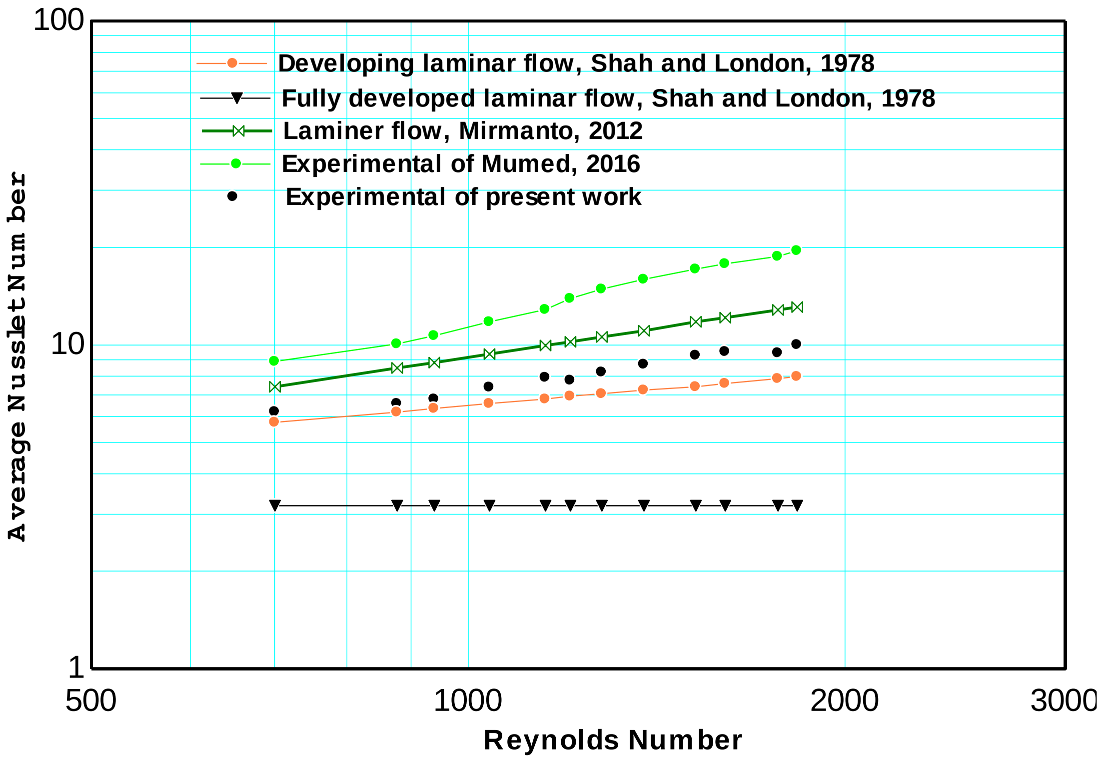


Fig.(9): Average Nusselt number comparisons with conventional and micro-scale laminar flow correlations for Model-1

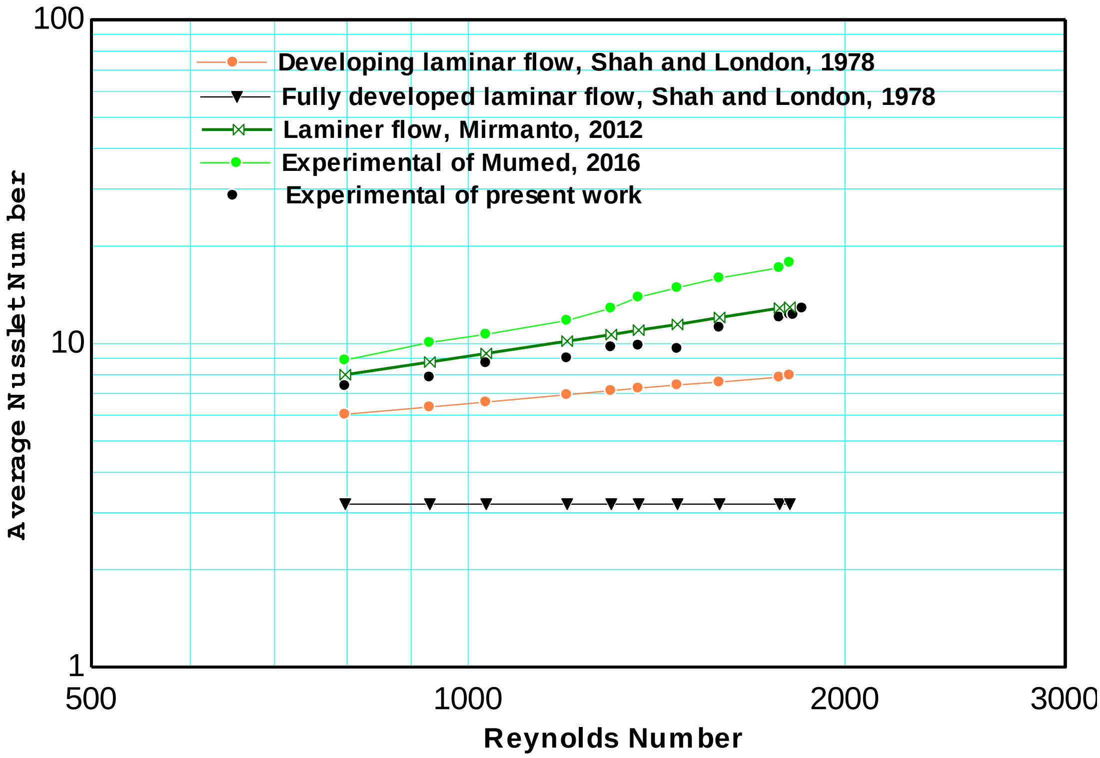


Fig. (10): Average Nusselt number comparisons with conventional and micro-scale laminar flow correlations for Model-2

## REFERENCES


Choi, S.B., Barron, R.F., Warrington R.O., Fluid flow, and heat transfer in microtubes, Micromech. Sensors Actuat. Syst., ASME DSC 32 (1991) 123–134.

G.M. Mala, M. Li, Flow characteristics of water in microtubes, Int. J. Heat Fluid Flow 20 (1999) 142-148.

G.M. Mala, D. Li, J.D. Dale, Heat transfer and fluid flow in microchannels, Int. J. Heat Mass Transfer 40 (1997) 3079–3088.

Kandlikar, S. G., Schmitt, D., Carrano, A. L., and Taylor, J. B., Characterization of Surface Roughness Effects on Pressure Drop in SinglePhase Flow in Minichannels, Phys. Fluids, 17(10), 100-106, 2006.

Kandlikar, S. G., Joshi, S., and Tian, S., Effect of Surface Roughness on Heat Transfer and Fluid Flow Characteristics at Low Reynolds Numbers in Small Diameter Tubes," Heat Transfer Eng., 24(3), pp. 4–16, 2003.

Kline, S. and McClintock, F.A., Describing uncertainties in single-sample experiments, Mech. Eng. 75(1), pp. 3-8, (1953).

Mesbah G. Khan, Amir Fartaj, (2011), A review on microchannel heat exchangers and potential applications, Int. J. Energy Res. 35 (2011) 553-582.

Mirmanto, M., Heat Transfer Coefficient Calculated Using Linear Pressure Gradient Assumption and Measurement For Flow Boiling in Microchannels, International Journal of Heat and Mass Transfer, vol. 79, pp. 269–278, 2014.

Mala, G.M., Li, D., Flow characteristics of water in microtubes, Int. J. Heat and Fluid Flow 20 (1999) 142-148.

Peng, X.F., Peterson, G.P., Wang, B.X., Heat transfer characteristics of water flowing through microchannels, Exp. Heat Transfer 7 (1994) 265–283.

Peng, X.F., Peterson, G.P., Convective heat transfer and flow friction for water flow in microchannel structures, Int. J. Heat Mass Transfer 39 (1996) 2599–2608.

Phillips R J 1987 Forced convection, liquid cooled, microchannel heat sinks MS Thesis (Department of Mechanical Engineering, Massachusetts Institue of Technology).

Papautsky, B.K. Gale, S. Mohanty, T.A. Ameel, A.B. Frazier, Effects of rectangular microchannel aspect ratio on laminar friction constant, 2000, unpublished, from authors_ website.

Qu, W. and Mudawar, I. Measurement and prediction of pressure drop in two-phase micro-channel heat sinks. International Journal of Heat and Mass Transfer, 46(15):2737-2753, 2003b.

Srinivas Garimella, William J. Dowling, Mark Van Der Veen, Jesse D. Killion, The effect of simultaneously developing flow on heat transfer in rectangular tubes, Heat Transfer Eng. 22 (2001) 12-25.

Salem M.M., Elhsnawi M.H., Mohamed S.B., Experimental Investigation of Surface Roughness Effect on Single Phase Fluid Flow and Heat Transfer in Micro-Tube, International Journal of Mechanical, Aerospace, Industrial, Mechatronic and Manufacturing Engineering Vol:7, No:9, 2013.

Steinke, M.E. and Kandlikar, S.G. "Review of SinglePhase Heat Transfer Enhancement Techniques for Application in Microchannels, Minichannels and Microdevices" Int. J. of Heat and Technology 22, no. 2 (2004) 3-11.

Xing Y., Zhi, T., Haiwang, L., Yitu, T., Experimental investigation of surface roughness effects on flow behavior and heat transfer characteristics for circular microchannels, Chinese Journal of Aeronautics (2016), doi:http://dx.doi.org/10.1016/j.cja.2016.10.006

Kays W M and London A L 1984 Compact Heat Exchangers Third edition (New York: McGraw-Hill).

Lee, P.-S. and Garimella, S.V. Saturated flow boiling heat transfer and pressure drop in silicon microchannel arrays International Journal of Heat and Mass Transfer, 51(3-4):789- 806, 2008.

Shah, R.K. and London, A.L., Laminar flow forced convection in ducts, Supplement 1 to Advances in Heat Transfer, Academic Press New York, 1978.