THERMALLY DEVELOPING MIXED CONVECTION IN A HORIZONTAL EQUILATERAL TRIANGULAR CHANNEL

Thamer Khalif Salem, **Raaid R. Jassem & Manar Salih Mahdi**
Mech. Engineering, College of Engineering, Tikrit University, Iraq.
Email: Thamer_khalif @yahoo.com, Raaid.rashad@yahoo.com, manar_salih@yahoo.com

ABSTRACT

An experimental study has been conducted to investigate thermal developing, mixed convection, in a horizontal equilateral triangular channel heated uniformly. An experimental work has been conducted for horizontal equilateral triangular channel, heated uniformly, to investigate thermally developing mixed convection. The channel was (1.5 m) long and constructed from three plane (100 mm) walls to form the equilateral triangular cross section. These experiments had been conducted for range values of ($115 \leq af \leq 237(W/m^2)$)) of heat flux and for the range ($0.0005 \leq m \leq 0.0015(kg/\text{sec.})$) of air mass flow rate.

The test apparatus has been manufactured first, Then a thermocouples have been stickled in proper positions. Then many readings, for temperatures, have been registered for each thermocouple. These readings have been taken for different speeds of the outlet air (from the Equilateral triangular channel).

Five tests for the heat flux have been conducted, for each one of them three values of air mass flow-rate have been used. It has been noticed that the surface temperature of test has a proportional relation with the channel length. And this is found for all values of the heat flux and air mass flow-rate .

It was found that the temperature ratio and local Nusselt number had increased with increasing the heat fluxes and the mass flow rate. Also it was found that the greatest local Nusselt number value was in the entrance of the channel then it decreased accompanied by growing the thermal boundary layer along the channel.

Keywords: *Equilateral triangular channel, mixed convection, Thermally developing.*

Nomenclature

A_c	Cross-sectional area	(m^2)	y	High of triangle cross section	(m)
C_p	Specific heat	($J.kg^{-1}. K^{-1}$)	I	Electrical current	(Ampere)
D_h	Hydraulic diameter	(m)	k	Thermal conductivity	($W.m^{-1}.^oC^{-1}$)
g	Acceleration due to gravity	(m/s^2)	$\dot{m}$	Mass flow rate	($kg.s^{-1}$)
Nu	Nusselt number		V	Heater voltage	(Volt)
Pr	Prandtl number			***Greek symbols***	
Re	Reynolds number		β	Volumetric ratio	(K^{-1})
Ri	Richardson number		ν	Kinematic viscosity	($m^2.s^{-1}$)
Q_{add}	Heat added	(W)	θ	Temperature ratio	
Q_{in}	Heat supply	(W)		***Subscripts***	
q_f	Heat flux	($W. m^{-2}$)	a	Air	
T	Temperature	(^oC)	b	Bulk	
u	Velocity of air	($m. s^{-1}$)	i	Inlet	

Gr	Grashof number		o	Outlet
h	Heat transfer coefficient	$(W.m^{-2}.^{o}C^{-1})$	s	Surface
x	Base of triangle cross section	(m)		
X	Non dimensional length(x/L)	-		

1. INTRODUCTION

Mixed convection heat transfer in channels characterized by non- circular cross sections is a fundamental issue in many fields such as research and industry fields. Because of its uses in many thermal applications such as compact heat exchangers, solar collectors and cooling of electrical and electronically devices. Different shapes of the cross-section area have been analyzed, like square, rhombic, rectangular, triangular, sinusoidal, elliptical ones, even with truncated corners. Thus in the last five decades the mixed convection had attracted many researchers[I. Uzun and M. Unsal 1997, Cengel,1998][1,2].

Zangana,2005 [3] studied the effect of delta– winglet type of vortex generator on a pressure head losses and heat transfer rate in an equilateral triangular duct experimentally. He studied a single, double, and triple pairs of generators embedded in the turbulent boundary layer. A (6 – 24) degree of angle of attack of generators was studied for two different sizes of generators. The study shows that the friction factor is directly proportional with the size of generators, number of generators and the angle of attack and inversely proportional with the flow rate. Moreover, he found that the friction factor increases slightly with the size of generators compared with Reynolds number as well as with the angle of attack of generator.

An experimental investigation was performed by (**Mohanad A. & Etals,2007**) [4] to study the behavior of friction factor and heat transfer coefficient in an asymmetrically heated equilateral triangular duct, by using delta–winglets vortex generators, which were embedded in a turbulent boundary layer. Two side walls of the heated test section were electrically heated with a constant heat flux, whereas the lower wall was indirectly heated. Reynolds number (Re) range from (23,000) to (58,000). Two sizes and three attack angles of vortex generators were studied for three cases; single, double, and treble pairs of generators. Each pair was supported in one wall of the test section at the various locations from the leading edge. The indicated results that friction factor (*f*)and Nusselt number (Nu) were relatively proportion with the size, number and the inclination angle of the generators. The (*f*) had decreased as airflow rate increased whereas Nu number increased.

Zhang,2007 [5], had reported Nusselt numbers for laminar hydrodynamically fully developed and thermally developing flow for a uniform wall temperature condition in isosceles triangular ducts with apex angles ranging from 30° to 120°.

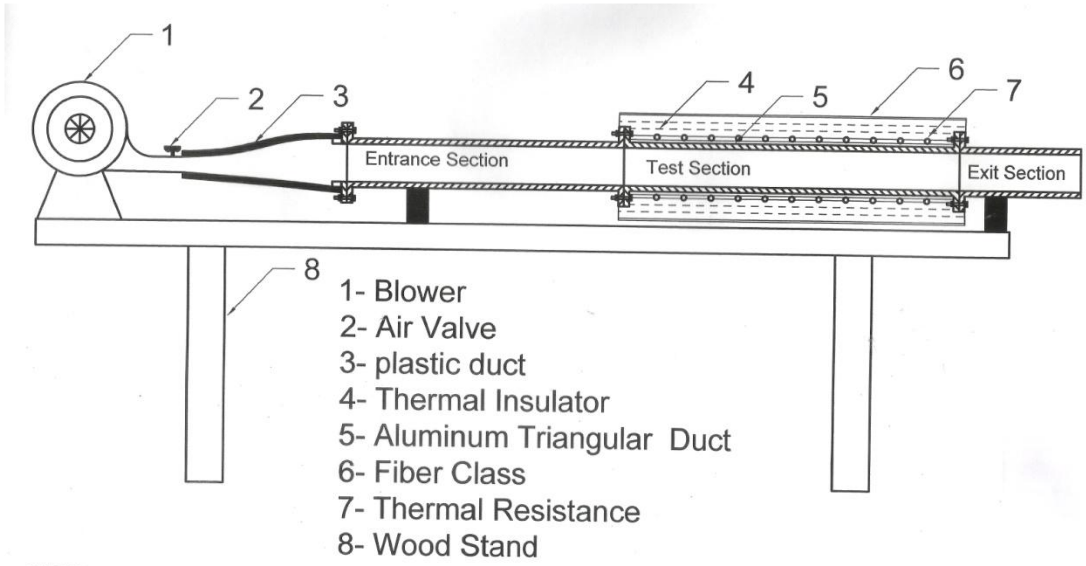
Talukdar and Shah,2008 [6] had studied the effect of Rayleigh number on bulk mean temperature and Nusselt number in triangular ducts with different apex angles. They pointed out the increase of these parameters for increasing Rayleigh numbers and the higher heat transfer rate from the bottom boundary.

Gupta et al.,2008 [7] studied fully developed laminar flow and heat transfer in equilateral triangular cross-sectional ducts following serpentine and trapezoidal path. Friction factors for fully developed flow in an equilateral triangular duct containing built-in vortex generators of delta wing had studied by (**Mohammed G,2010**)[8], rectangular wing, pair of delta winglets, and a pair of rectangular winglets had been investigated experimentally for Reynolds numbers ranging from (24500) to (75750). The ratio of the cross- sectional area of the test duct to that of the vortex generator (AD/AVG) was remaining constant during experiments. The variables parameters, vortex generator type, angle of attack, and Reynolds number.The variables parameters, vortex generator type, vortex generator angle of attack, and Reynolds number. The results showed that the friction factor was affected strongly by the wing greater than the winglet pair of vortex generators. The delta wing caused flow loss greater than the rectangular wing while the flow loss accompany with the existence of the pair of delta- winglets were less than that of the pair of rectangular winglet. It was also observed that the friction factor is affected remarkably by the angle of attack of vortex generator.

(Oronzio M. & etals, 2012)[9] had investigated the Convective heat transfer can be enhanced passively by changing flow geometry and boundary conditions or by improving the thermal conductivity of the working fluid, for example, introducing suspended small solid nanoparticles. They presented, a numerical investigation on laminar mixed convection in a water-Al2O3-based nanofluid, flowing in a triangular cross-sectioned duct. The duct walls were assumed at a uniform temperature, and the single-phase model had been employed in order to analyze the nanofluid behaviour. The hydraulic diameter was equal to 0.01 m. A fluid flow with different values of Richardson number and nanoparticle volume fractions had been considered. Results showed the increase of average convective heat transfer coefficient and Nusselt number for increasing values of Richardson number and particle concentration. However, also wall shear stress and required pumping power profiles grow significantly.
The study had investigates the effect of rotating horizontal single or multi cylinders on mixed convection heat transfer in an equilateral triangular enclosure filled with air by (**Ahmed K.,2012**)[10]. The governing equations were solved numerically by using the finite element method with Flex PDE soft package. for steady state, laminar flow, two dimensional only, incompressible flows with Boussinseq approximation, constant properties of the fluid. Three cases were performed: single rotating cylinder, three rotating cylinders at the same direction and three rotating cylinders at different directions. The main parameters were: Rayleigh number ($Ra = 10^2 - 10^5$), Prandtl number (Pr = 0.7), the dimensionless angular velocity (Ω=0-1000) (for both directions clockwise CW and counter clockwise CCW) and dimensionless radius of a rotating cylinder (R=0.1-0.25). It was found that the average Nusselt number for the single or multi rotating cylinder was increased with increasing Ra, R and Ω for all cases. Also, the average Nusselt number of single rotating cylinder was greater than the multi rotating cylinders for the same ratio of the solid cylinder or cylinders volume to total enclosure volume.
Accurate analytic solutions for the fully developed slip flow and H1 heat transfer in rectangular and equilateral triangular ducts were presented by (**C.Y.Wang ,2013)[11]**. Both velocity slip and temperature jump had significant influences on the Poiseuille and Nusselt numbers. These exact solutions served as benchmark cases for other methods, whether analytic, approximate, or numerical.
In the present paper, an experimental work was conducted to investigate thermally developing flow in a horizontal equilateral channel. While the hydraulic flow was fully developed.

2. EXPERIMENTAL WORK

Mixed convection heat transfer is one of the most influential convection transfer, so an experiment mixed convection in a horizontal equilateral triangular channel was conducted where the channel surface was heated uniformly. An aluminum duct was used to made the channel. The test section was (1.5 m) long and of equilateral triangular cross section with sharp corners constructed of three plane (100 mm) walls. The (1 m) long hydraulically transition section was used to ensure that the air entering the test section had a uniform velocity distribution (hydraulically fully developed flow). Length of the test section from the channel was (1.5 m)only. At the exit of the test section, the air was exhausted into the atmosphere through (0.5 m) duct to avoid end effect. A centrifugal blower was used to suck the air into the channel. The blower was physically isolated from the channel using a flexible connecting nylon tube to inhibit the transmission of vibration. An anemometer- type" EXTECH instruments- AN100" was used to measure the air speed inside the duct. The mass flow rate was controlled by a flow regulator valve located between the blower and the test section. The entire test section was thermally insulated from its ambient environment by a (1.5 cm) glass wool. The tube surface temperature was measured by (44) thermocouple (T type) which were positioned along the test section duct. These thermocouples were used to measure the inlet and outlet air temperatures and the surface temperature in fourteen locations, (10 cm) apart, along the axial length of the test channel for each wall. One thermocouple was used to measure the air temperature at the entrance of the duct which was located in the air flow another one was located after packed to measure the out flow air temperature, and it was positioned in the same pervious way. Another thermocouple measured the laboratory ambient temperature. Uniform heating was provided from an electric heater wire (D=0.74 mm) which was wounded uniformly around the external surfaces of the triangular channel. The test section was thermally insulated with fiber glass of (100mm) in thickness, to reduce the heat losses. The ends of the tube were insulated electrically and thermally by using two parts of tephlon. The voltage was stabilized by a stabilizer. The electrical power supplied is monitored by a variac transformer which provided a controllable constant heat flux for the test section. An A meter was used to measure the current passed through the heater with accuracy (10^{-4}A). Digital Anemometer AM4200 with accuracy of (0.1 m/s) is used to measure the velocity of entering air to the test section. A Fig. (1a,b) illustrates the schematic drawing of the test section.



EXPERIMENTAL PROCEDURE

Tests were carried out for three different air mass flow rates (0.5×10^{-4}, 0.1×10^{-3} and 0.15×10^{-3} kg/s), those mass flow rates were chosen to ensure having laminar flow and for five heat fluxes (115, 146, 171,204 and 237 w/m^2). During a typical experimental run, a constant power input was supplied, and the air velocity was adjusted by the valve until the specified value was achieved. The air and the channel surfaces temperatures were recorded when the steady state was reached after (40 to 60 minutes). Then the valve was adjusted to achieve a new air speed, and the above procedure was repeated.

DATA REDUCTION

There are many variables have been calculate some of them are locally indirectly measured such as Nusselt number, heat transfer coefficient and the other in between variables have been calculated e.g. Grashof number (Gr) and Richardson number (Ri). The local heat transfer coefficient was calculated using the following equation (Hussein & et als)[12]:-

$$h_z = \frac{q_f}{(T_s - T_b)_z} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

Where (q_f) is the heat flux for a unit of the area and can be calculated from the following equation:-

$$q_f = \frac{Q_{in}}{A_c} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Where:- $A_c = 0.5 * x * y$ & $Q_{in} = I * V$

The bulk temperature for the air was calculated using the following equation (J. P. Holman)[13]:-

$$\mathrm{T_b} = (\frac{\mathrm{T_i} + \mathrm{T_o}}{2}) + 273.18 \qquad \ldots\ldots\ldots\ldots\ldots (3)$$

Where the quantity of the added heat transfer (Q_{add}) to the air was calculated as follow:-

$$Q_{add} = mC_{pa}(T_{ao} - T_{ai}) \qquad \text{………… (4)}$$

The local Nusselt number was calculated using the following equation:-

$$Nu_z = \frac{h_z D_h}{k} \qquad \text{…………………(5)}$$

The characteristic dimension to define Reynolds number was chosen to be the hydraulic diameter of the triangular duct:

$$\mathrm{Re} = \frac{uD_h}{\upsilon} \qquad \text{………………………………(6)}$$

Where : $D_h = 4\dfrac{Flow.area}{primeter}$

Peclet number (Pe) was calculated as follow:-

$$Pe = \mathrm{Re}\,\mathrm{Pr}_a \qquad \text{………………...(7)}$$

Where : $\mathrm{Pr}_a = \upsilon / \alpha$

Grashof number (Gr) for uniform heat flux was calculated as follow[Holman,2012][14]:-

$$Gr = \frac{g\beta q_f D_h^{\,2}}{k\nu^2} \qquad \text{…………..…(8)}$$

Where $\beta = \dfrac{1}{T_b}$

Rayleigh number (Ra) was calculated using the following equation:-

$$Ra = Gr\,\mathrm{Pr}_a \qquad \text{…........(9)}$$

Where Richardson number (Ri) was calculated :-

$$\mathrm{Ri} = \mathrm{Gr}/\mathrm{Re}^2 \qquad \text{……………..…(10)}$$

The average value of Nusselt number was calculated as follow:-

$$Nu_{av} = \frac{\int_0^x Nu_x\, dX}{X} \qquad \text{…………...(11)}$$

RESULTS AND DISCUSSION

In this paper, an experimental work was conducted to investigate thermally developing mixed convection heat transfer in a horizontal equilateral triangular channel. Fifteen tests were done for three different mass flow rates (0.5 $\times$ 10^{-3}, 0.1$\times$ 10^{-2} and 0.15$\times$ 10^{-2} kg/sec) and five heat fluxes (115, 146, 171, 204 and 237 W/m^2).

Figures (2 and 3) show the left surface temperature at the minimum and the maximum heat fluxes. When the air is flowing slowly it will have more time to expose to heat which leads to increase its temperature, while when the air is flowing faster the period of its exposing to the heat will be less, and consequently its temperature will be lower. This was true for the right surface – figures(4 and 5)- and the bottom surface –figures (6 and 7).

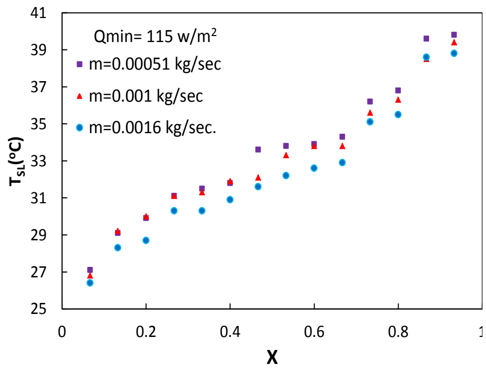


Figure (2) temperature of the left surface of the channel at Q=115 W/m^2.

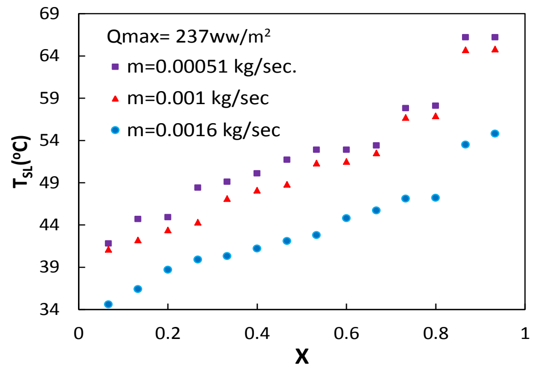


Figure (3) temperature of the left surface of the channel at Q=237 W/m^2.

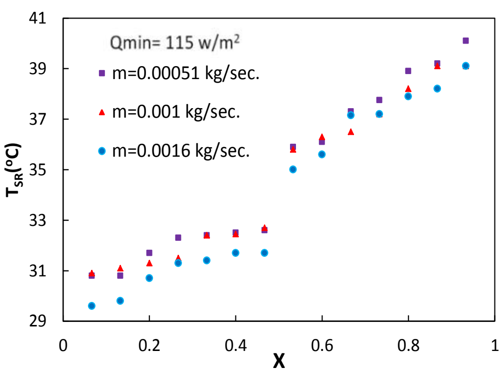


Figure (4) temperature of the right surface of the channel at Q=115 W/m^2.

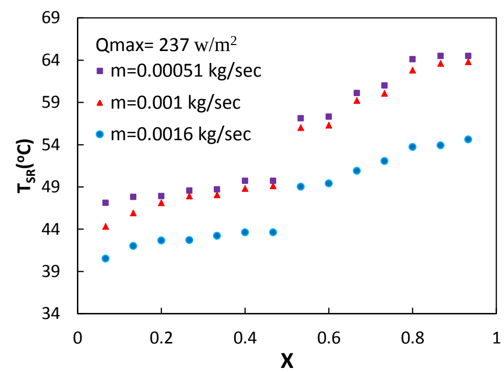


Figure (5) temperature of the right surface of the channel at Q=237 W/m^2.

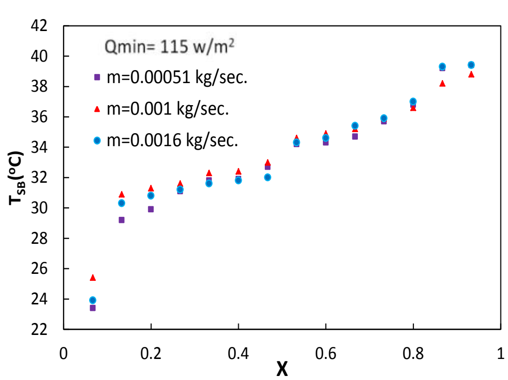


Figure (6) temperature of the bottom surface of the channel at Q=115 W/m^2.

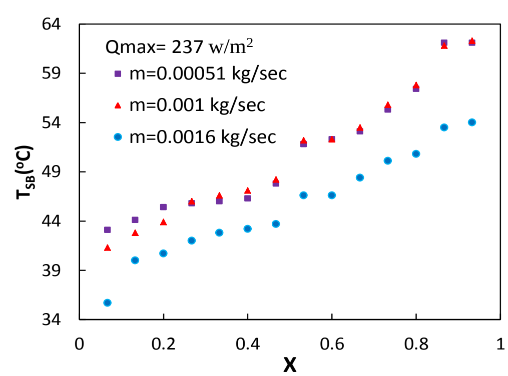


Figure (7) temperature of the bottom surface of the channel at Q=237 W/m^2.

Figures (8-10) show the distribution of θ $(\frac{T-T_S}{T_i-T_S})$ along the channel at fixed mass flow rate. At the entrance region of the channel the value of the temperature ratio (θ) is high, then it was decreases to get it is lowest value at the end of the canal, due to the continued canal heating, which leads to high air temperature until it becomes the difference between canal surface temperature and air temperature constant in this position the flow of heat up to the area fully developed. The same general behavior was achieved for different flow rates at the same heat flux.

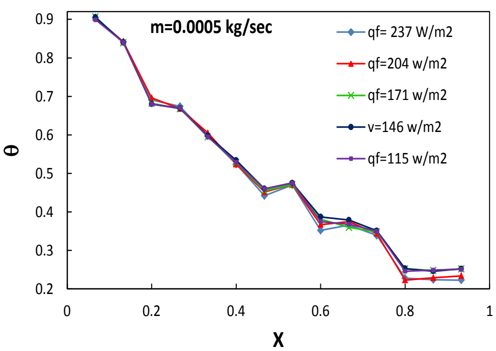


Figure (8) distribution of θ along the channel at $\dot{m} = 0.0005 kg/sec$

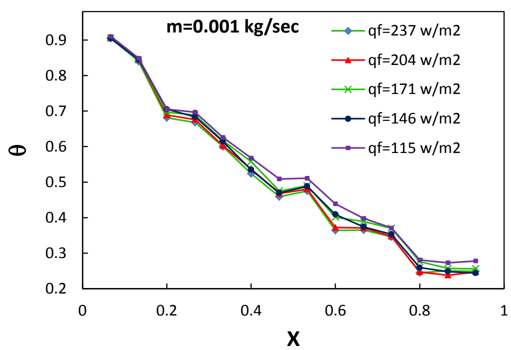


Figure (9) distribution of θ along the channel at $\dot{m} = 0.001 kg/sec$

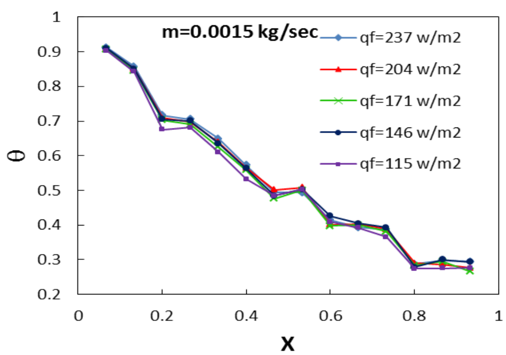


Figure (10) distribution of θ along the channel at $\dot{m} = 0.0015 \; kg/sec$

The heat flux increasing did not show any significant difference between them. Figures (11-15) show the effect of the mass flow rate on θ at the same heat flux. At the entrance of the channel, there was not a significant effect of the mass flow rate in the opposite at the exit of the channel. As mentioned before, when the mass flow rate increases, the air temperature will be lower and then the temperature difference between the air and the channel surface is greater, which means higher value of temperature ratio θ. Where the highest θ was 0.912 at $\dot{m}$= 0.15$\times$ 10^{-2} kg/sec and Q= 237 W/m^2, and the lowest value was 0.899 at $\dot{m}$= 0.5×10^{-3} kg/second Q= 115 W/m^2.

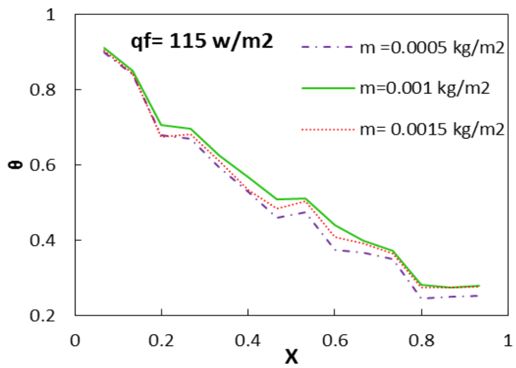


Figure (11) distribution of θ along the channel at Q= 115 w/m^2.

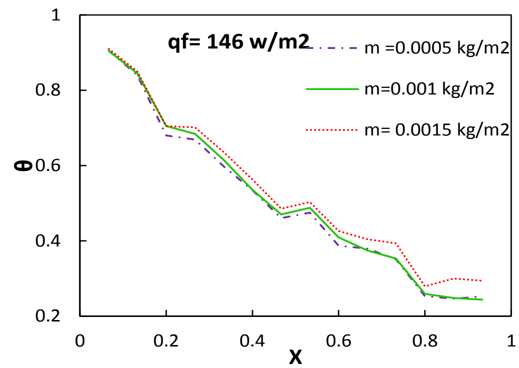


Figure (12) distribution of θ along the channel at Q= 146 w/m^2.

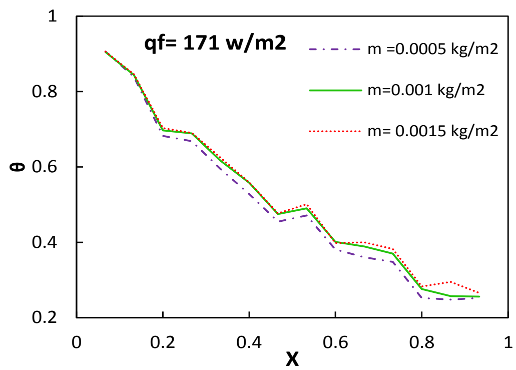


Figure (13) distribution of θ along the channel at Q= 171 w/m^2.

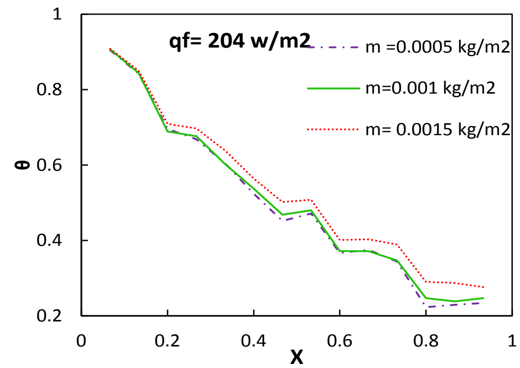


Figure (14) distribution of θ along the channel at Q= 204 w/m^2.

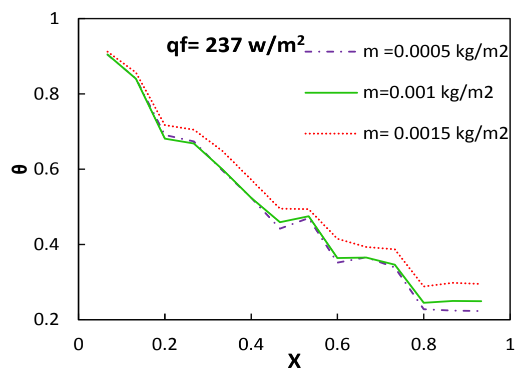


Figure (15) distribution of θ along the channel at Q= 237w/m^2.

Figures (16-18) show the distribution of Nu_x along the channel at different heat fluxes. the highest Nusselt number (Nux) is observed at the enterance of the channel. because there the maximum temperature difference between the cold air and the hot surface of the channel, there the thermal boundary is thin. the air temperature increasing along the channel means lower temperature difference and then lower Nusselt Number (Nux) and thick thermal boundary layer. As the heat flux increases, more heat will be transfer to the air as can be noticed.

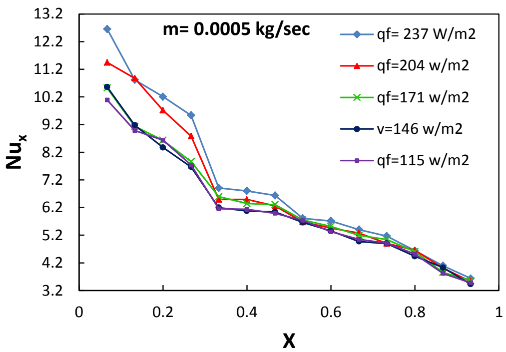


Figure (16) distribution of Nu_x along the channel at $\dot{m} = 0.0005 kg/sec$

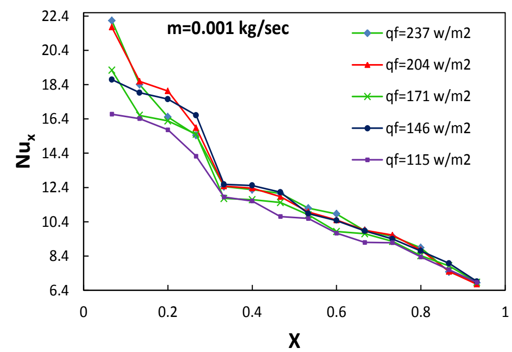


Figure (17) distribution of Nu_x along the channel at $\dot{m} = 0.001 kg/sec$

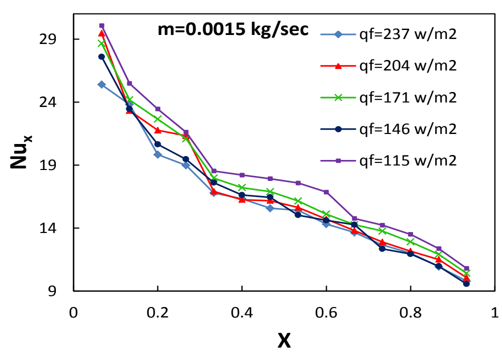


Figure (18) distribution of Nu_x along the channel at $\dot{m} = 0.0015\ kg/sec$

Figures (19-23) show the effect of mass flow rate on Nu_x along the channel. When the air is moving faster through the channel its temperature will be lower comparing with a slower air, so the temperature difference is greater than for the slower air, which means higher Nu_x. The highest Nu_x(=30.7) was observed at $\dot{m}$= 0.15× 10^{-2} kg/sec and Q= 115 W/m^2, and the lowest value was 10.523 at $\dot{m}$= 0.5×10^{-3} kg/sec and Q= 171 W/m^2.

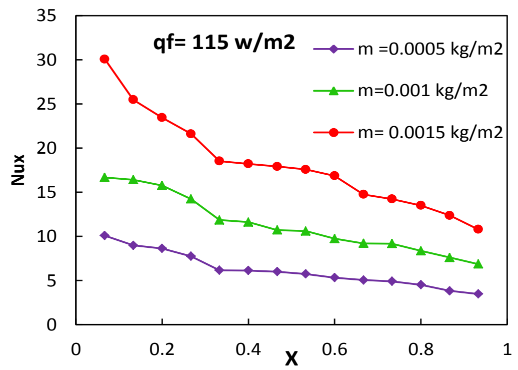


Figure (19) distribution of Nu_x along the channel at Q= 115 w/m^2.

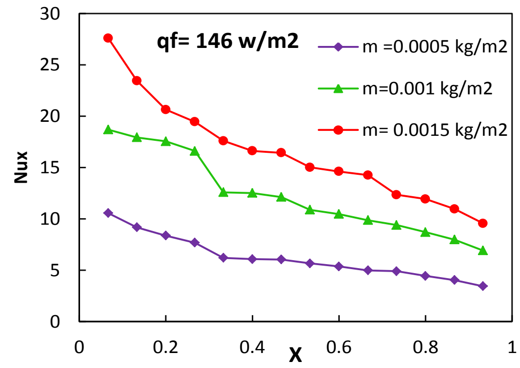


Figure (20) distribution of Nu_x along the channel at Q= 146 w/m^2.

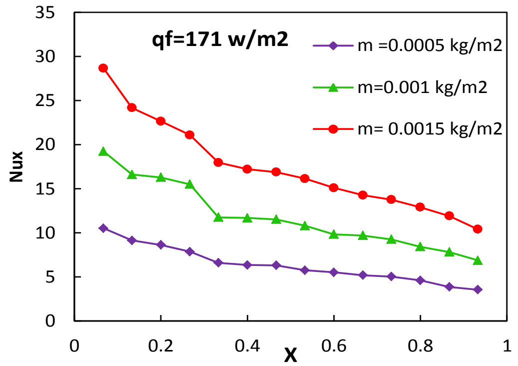


Figure (21) distribution of Nu_x along the channel at Q= 171 w/m^2.

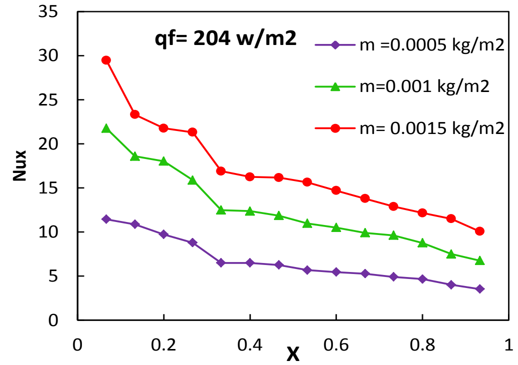


Figure (22) distribution of Nu_x along the channel at Q= 204 w/m^2.

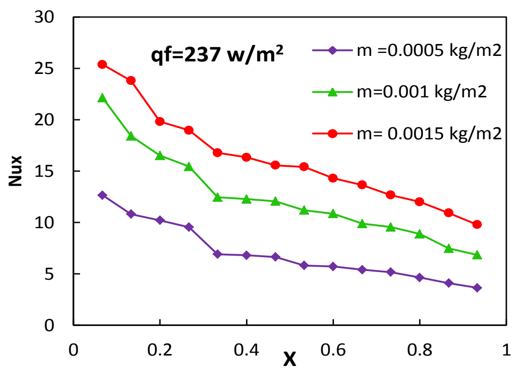


Figure (23) distribution of Nu_x along the channel at Q= 237 w/m^2.

CONCLUSIONS

In this paper, an experimental work was conducted to investigate thermally developing, fully developed steady laminar mixed convection in a horizontal equilateral triangular channel. Fifteen tests were done for three mass flow rates (0.5×10^{-3}, 0.1×10^{-2} and 0.15×10^{-2} kg/sec) and five heat fluxes (115, 146, 171, 204 and 237 W/m^2).
The results can be summarized as follows:

- For low flow rates, the air have more time to expose to heat, and the air bulk temperature will be higher than of the high flow rate.

- The highest θ can be noticed in the entrance of the channel where the air bulk temperature is low, and with continuous heating along the channel the air bulk temperature increased leading to decreasing θ which get its lowest value at the exit of the channel.
- The changing in heat flux not effected on temperature-ratio value(θ), but its value increased with mass flow rate increasing..
- Nu_x increased with increasing both the mass flow rate and the heat flux.

REFERENCES

[1]. I. Uzun and M. Unsal, "A numerical study of laminar heat convection in ducts of irregular cross-sections," International Communications in Heat and Mass Transfer, vol. 24, no. 6, pp. 835–848, 1997.

[2]. Cengel, Y. A.," Heat Transfer A practical Approach", 1st Edition, Mic Groohil com., Inc., USA, 1998

[3]. Zangana, H. E.,"Effect of Delta-Winglet Vortex Generators on a Forced Convection Heat Transfer in an Asymmetrically Heated Triangular Duct", Master Thesis, University of Anbar, Engineering College, Sep. 2005.

[4]. Mohanad A. Al-Taher, Adnan A. Abdul-Rassol and Hamid E.Zangana," Effect of Delta–Winglet Vortex Generators on a Forced Convection Heat Transfer in an Asymmetrically Heated Triangular Duct", Anbar Journal for Engineering Sciences © AJES / 2007.

[5]. Zhang LZ: Laminar flow and heat transfer in plate-fin triangular ducts in thermally developing entry region. Int J Heat Mass Transfer 2007, 50:1637-1640.

[6]. P. Talukdar and M. Shah, "Analysis of laminar mixed convective heat transfer in horizontal triangular ducts," Numerical Heat Transfer, Part A, vol. 54, no. 12, pp. 1148–1168, 2008.

[7]. Gupta RV, Geyer PE, Fletcher DF, Haynes BS: Thermo-hydraulic performance of a periodic trapezoidal channel with a triangular cross-section. Int J-Heat Mass Transfer 2008, 51:2925-2929.

[8]. Mohammed Ghanem Jehad," Experimental Study Of The Friction Factor In Equilateral Triangular Duct With Different Types Of Vorex Generators (OBSTACLES)", Al-Qadisiya Journal For Engineering Sciences Vol. 3 No. 2 Year 2010.

[9]. OronzioManca, Sergio Nardini, Daniele Ricci, and Salvatore Tamburrino," Numerical Investigation on mixed Convection in Triangular Cross-Section Ducts with Nanofluids", Hindawi Publishing Corporation Advances in Mechanical Engineering Volume 2012, Article ID 139370, 13 pages.

[10]. Ahmed K. M. Alshara," Effect Of Single Or Multi Rotating Horizontal Cylinders On The Mixed Convection Heat Transfer Inside A Triangular Enclosure", Al-Qadisiya Journal For Engineering Sciences Vol. 5 No. 1 Year 2012.

[11]. C.Y.Wang, "Benchmark Solutions for Slip Flow and H1 Heat Transfer in Rectangular and Equilateral Triangular Ducts ", J. Heat Transfer -- February 2013 -- Volume 135, Issue 2, 021703 (8 pages) http://dx.doi.org/10.1115/1.4007576.

[12]. Hussein A. Mohammed and Yasin K. Salman, "Experimental Mixed Convection for Downward Laminar Flow in the Thermal Entranc Region of Inclined and Vertical Circular Tubes", Experimental Thermal and Fluid Science, pp 1-12, February 2007, www.elsevier.com/locate/etfs.

[13]. J. P. Holman," Experimental Method for Engineers", McGraw- Hill Book company 5th Edition, 1977.

[14]. Holman, J. P., 2012. "Experimental Methods for Engineers", 8th. Edition, McGraw- Hill Book Company, New York, USA.