Tikrit Journal of Engineering Sciences/Vol.19/No.3/September 2012, (58-67)

Thermally Developing Forced Convection in a Horizontal Equilateral Triangular Channel

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Abstract

An experimental investigation was carried out to study thermally developing fully developed laminar forced convection in a horizontal equilateral triangular channel where the channel surface was heated uniformly. The channel length was (1.5 m) long and constructed from three plane (100 mm) walls to form the equilateral triangular cross section. The experiments were conducted for three mass flow rates (1.91×10^{-3}, 2.54×10^{-3} and 3.03×10^{-3} kg/s) and four heat fluxes (91, 171, 272 and 406 W/m^2). Reynolds number range was ($1198 \le Re \le 1988$). It was found that local Nusselt number increasing as the heat fluxes and the mass flow rate increased. Also it was found that the greatest local Nusselt number value was in the beginning of the channel then it decreased accompanied by growing the thermal boundary layer along the channel. Present experimental results have a good agreement with previous results obtained for similarly configured channels.

Keywords: Equilateral triangular channel, Forced convection, Thermally developing

انتقال الحرارة بالحمل القسري لجريان متطور حرارياً في قناة أفقية مثلثة متساوية الأضلاع

الخلاصة

تم إجراء استقصاء عملي لدراسة انتقال الحرارة بالحمل القسري لجريان متطور حرارياً تام التشكيل هيدروليكياً في قناة أفقية مثلثة متساوية الأضلاع، وقد تم تسخين القناة تسخيناً منتظماً. كان طول القناة (1.5 m) وتكونت من ثلاثة أسطح بطول (100 mm) لتشكيل مقطع مثلث متساوي الساقين. أجريت التجارب لثلاث تدفقات كتلية (1.91×10^{-3} و2.54×10^{-3} و3.03×10^{-3} kg/s) وأربعة قيم للفيض الحراري (91 و171 و272 و 406 W/m^2)، وكان مدى عدد رايلي ($1198 \le Re \le 1988$). وجد أن قيمة عدد نسلت الموقعي تزداد بزيادة الفيض الحراري والتدفق الكتلي، كما وجد أن أعلى قيمة لعدد نسلت الموقعي تكون في بداية القناة وتقل بالتوازي مع نمو الطبقة المتاخمة الحرارية على طول القناة. توافقت النتائج العملية الحالية مع نتائج سابقة لقنوات مشابهة.

الكلمات الدالة: قناة مثلثة متساوية الأضلاع، حمل قسري، متطور حرارياً.

Nomenclatures

Ac   cross-sectional area of the triangular channel, m^2.
c_p specific heat at constant pressure, J kg^{-1} K^{-1}.
D_h hydraulic diameter of the triangular channel, m.
h   convection heat transfer coefficient for the heat transferred from the

inner surface of the test channel to the air in the channel, W m^{-2} K^{-1}.

k thermal conductivity of the air, W m^{-1} K^{-1}.

$\dot{m}$ steady state mass flow rate of the air flow through the triangular test channel, kg s^{-1}.

L the test channel length, m.

Nu Nusselt number based on hydraulic diameter of the test channel.

Q_c steady state rate of forced convection from the inner surfaces of the test channel to the air flow, W.

Re Reynolds number of the air flow based on hydraulic diameter of the test channel.

T_a Mean temperature of the airflow through the test channel, K.

T_{ai} Mean bulk temperature of the air flow at the inlet of the test channel, K.

T_{ao} Mean bulk temperature of the air flow at the outlet of the test channel, K.

T_w Mean surface temperature of the test channel, K.

U Mean air velocity in the triangular channel, m s^{-1}.

x Axial position, m.

x* Dimensionless axial position for thermally developed flow = $x/D_h RePr$

Greek Symbols

ν kinematic viscosity of air, m^2 s^{-1}.

ρ density, kg.m^{-3}.

Introduction

Forced convection is playing a great rule in many industrial applications such as automotive industry, air conditioning, electric cooling, spacecraft and aircraft applications [1]. Many of these applications use triangular cross section channels particularly compact heat exchangers.

The compact heat exchangers with triangular cross- sectional channels have two benefits: a high ratio of heat-transfer surface-area to core volume (i.e. usually excess of 700 m^2m^{-3}) and a considerably lower fabrication cost than the identical-duty shell-and-tube heat-exchanger [2].

The compact heat exchangers have short length so the entrance region where the most heat transfer can occur in this part of the channel is very important [3].

An experimental determination of the steady state fully developed hydraulically and thermally developed laminar and turbulent forced convection in an equilateral triangular cross section duct was undertaken by Leung and Probert [4]. The (2 m) duct long was oriented horizontally and was constructed of three plane (100 mm) walls, uniformly heated by an insulted electric heater wire which was wounded around each of the three external surfaces of the metal duct, the triangle corners were sharp. They suggested that when the flow is laminar (Nu=3.25) and when the flow is turbulent (Nu=0.012 $Re^{0.83}$).

Ahn[5] had conducted experimental investigations to study forced convection of fully developed turbulent flows in horizontal equilateral triangular ducts with different surface roughness pitch ratios (rib pitch to rib height) of (4, 8 and 16) on one side. The rib height-to-channel hydraulic diameter was fixed at (0.046). The ducts bottom walls were heated uniformly while the other surfaces were thermally insulted. Reynolds number varied from (10,000) to (70,000). He concluded that the increasing in friction factors and heat transfer can be seen in order of roughness pitch ratio (4, 8 and 16). Also he found that for roughness pitch ratio (8) the position of maximum velocity moved to the highest place in the range studied and it had the best performance.

Ahn et al.[6] had experimental investigations to study the forced convection of fully developed turbulent flow in a horizontal equilateral triangular duct fabricated with different surface roughness pitch ratios of (4, 8 and 16) on one side wall only. The entire bottom wall of the duct was heated uniformly and the other surfaces were thermally insulated. In order to understand the mechanism of the heat transfer enhancement, measurements of heat transfer were done to investigate the contributive factor of heat transfer promotion, namely, the fin effect. The rib height to channel hydraulic diameter ratio is fixed at (0.046). The flow Reynolds number varied from (10,000) to (70,000). They found that the friction factors in the roughened tubes showed much higher than in the roughened triangular duct with the roughness element on one side only. Also the concluded that pump power performance comparisons revealed that the ribbed duct with rib pitch to height ratio (8) in the entire Reynolds number range could perform superior heat transfer enhancement.

A forced convection and flow friction characteristics of air cooled horizontal equilateral triangular ducts was conducted experimentally by Luo et al.[7]. The internal surfaces had been fabricated with uniformly spaced square ribs. The range of the rib height to the hydraulic diameter was from (0.11) to (0.21), rib to rib spacing from (3.41) to (13.93) and Reynolds number from (4,000) to (23,000). They found that using ribs significantly enhance forced convection between the turbulent flow and the triangular duct. Optimum relative ribs height and relative rib to rib spacing corresponding to maximum thermal performance of this system were (0.18) and (7.22). Also they concluded that the flow friction in the duct increased linearly with the relative rib height and the relative rib height of (0.18) was proposed to be the optimum rib size. The optimum relative rib to rib spacing to enhance the forced convection was obtained at (7.22).

Ray and Misra [8] had presented a paper dealt with the evaluation of pressure drop and heat transfer characteristics of laminar fully developed flow through ducts of square and equilateral triangular cross sections with rounded corners for both H1 and H2 boundary conditions. The dimensionless radius of curvature (R_c) of both type of ducts was varied from zero to the maximum possible value (1 for square duct and $1/\sqrt{3}$ for triangular ducts). The solutions for velocity and temperature are considered in the form of a harmonic series. The constants of the series are evaluated by least square technique. From the velocity and temperature solutions, the friction factor (fRe) and Nusselt number are calculated. They concluded that (fRe) for both square and triangular ducts increased rapidly with the increased in (R_c), particularly at lower values of (R_c). For triangular duct, the (fRe) data showed maxima at ($R_c \approx 0.35$). Also they found that variation of local Nusselt number along the periphery of the duct showed the presence of non uniformity in local heat transfer coefficient. For triangular ducts the effectiveness of the rounded portion increased continuously with the increasing in R_c for H2 boundary condition.

The above literature review provided some information about the effect of surface roughness and instructions on - laminar and turbulent-fully developed hydraulically and thermally forced convection in triangular ducts.

However there is still insufficient information about laminar thermally developing forced convection in smooth equilateral triangular duct. In this experimental investigation an equilateral triangular duct is chosen because it gives the best convection heat transfer performance among the other triangular configuration [7].

Experimental Apparatus and Test Procedure

Experimental Apparatus

A typical experimental apparatus of the present investigation is shown schematically in figure (1). The test section is (1.5 m) long and of equilateral triangular cross section with sharp corners constructed of three plane (100 mm) walls made of aluminum. The (1 m) long hydraulically transition section is used to insure that the air entering the test section has a uniform velocity distribution (hydraulically fully developed flow). At the exit of the test section the air is exhausted into the atmosphere through (0.5 m) duct to avoid end effect.

Uniform heating is provided from an electric heater wire (D=0.74 mm) which was wounded uniformly around the external surfaces of the triangular channel. The electrical power supplied is monitored by a variac transformer which provided a controllable constant heat flux for the test section.

The entire test section is thermally insulated from its ambient environment by a (1.5 cm) glass wool. A (220 W) centrifugal blower is used to suck the air into the triangular channel. The blower is physically isolated from the channel using a flexible connecting nylon tube to inhibit the transmission of vibration. To measure the air speed inside the duct an anemometer- type EXTECH instruments- AN100- is used. The mass flow rate is controlled by a flow regulator valve located between the blower and the test section. Forty four T type 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.

Experimental Procedure

Tests were carried out for three different air speeds (0.4, 0.5 and 0.6 m/s)-those speeds were chosen to ensure having laminar flow since the lower limit of critical Reynolds number is considered to approximately 2000 in triangular channels [10] - which corresponding the air mass flow rates (1.91×10^{-3}, 2.54×10^{-3} and 3.03×10^{-3} kg/s) and for four heat fluxes (91, 171, 272 and 406 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

The average heat transfer coefficient for triangular duct is determined by the ratio of the constant heat flux to the difference between the duct surface and air temperatures [11]:

$$h(x) = \frac{Q_c}{A_c\left(T_w(x) - T_a(x)\right)} \quad \ldots\ldots\ldots\ldots..(1)$$

Where convection heat transfer is obtained from:

$$Q_c = \dot{m} c_p \left(T_{ao} - T_{ai} \right) \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

And the steady state bulk temperature of the air flow (T_a) is calculated based on the measured inlet temperature by imposing energy conservation along the axial (x) direction[11]:

$$T_a(x) = T_{ai} + \frac{Qc}{A_c \rho c_p UL} x \ldots\ldots\ldots\ldots (3)$$

The characteristic dimension to define Reynolds number and Nusselt number is chosen to be the hydraulic diameter of the triangular duct ($D_h = 4 \frac{\text{flow area}}{\text{primeter}}$):

$$Re = \frac{U.D_h}{\upsilon} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

$$Nu = \frac{h.D_h}{k} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

All the air properties are based on the mean fluid temperature $(T_{ai}+T_{ao})/2$ and interpolated from the tabulated thermo-physical properties of air [12].

Results and Discussion

To investigate steady state laminar forced convection from inner surface of a horizontal equilateral triangular channel at low Reynolds number ($1198 \leq Re \leq 1988$) under thermally developing and fully developed flow an experimental rig was constructed. The channel wall was (1.5 m) long and the walls were fabricated from three (10 cm) sheets made of aluminum with sharp corners and smooth surface.

The experimental results obtained for three mass flow rates and four heat fluxes.

Since the channel is heated uniformly the bulk air temperature has a linear distribution along the channel as shown in figure (2), which is conforming to Luo[7], this true for all mass flow rates and heat fluxes.

The temperature of the bottom wall (B) of the channel always has the lowest value comparing with the temperature of the left (L) and right (R) walls, as shown in figures (3,4 and 5). This is happen because the heated air always rises up, for the bottom wall case the heated air rises up away from the bottom wall then driven by the flowing air which will decrease, while for the left and the right walls case the heated air rises up along the inclined walls towards the top which will keep their temperatures higher than the bottom one.

Figure (6) shows that as the heat flux increases local Nusselt number increases sequently, this is because more heat will be convected from the inner surface of the triangular channel to the flowing air. The same is true at any mass flow rate as shown in figures (7 and 8).

From these figures it can also observed that local Nusselt number value is the highest in the beginning of the channel entrance region but then decreases as asymptotically approaches the fully developed region where the thermal boundary layer completed its growth.

Referring to figures (9, 10, 11 and 12) when the air flows slowly inside the channel, i.e. low mass flow rate, the air will spend more time in the channel and will get hotter- comparative with higher mass flow rates- this will decrease the temperature difference between the temperature of the flowing air and the temperature of the channel surface along the channel which means lower local Nusselt number compared with higher mass flow rates at the same location in the channel.

All the recent work results were compared with those in ref. [13], the maximum difference was (33%) at ($\dot{m}=2.54\times10^{-3}$ kg/s) and (406 W/m^2) which can be accepted since ref. [13] results are for constant wall temperature beside their results were theoretically obtained. Comparisons between the recent work and ref. [13] are shown in figures (13 to 16).

Conclusions

The recent experimental study has conducted for steady state laminar flow forced convection in a horizontal equilateral triangular smooth channel at low Reynolds numbers for different heat fluxes. The obtained results can be summarized as follows:

1. Since the channel is heated uniformly the bulk air temperature has a linear distribution along the channel.
2. The bottom wall always has the lowest temperatures compared with the other left and right walls since the heat is removed from above it by the flowing air.
3. Decreasing the mass flow rate will decrease the heat transfer between the channel surface and the flowing air due to increasing the temperature of the flowing air.
4. As the heat flux increased more heat is convected to the flowing air which means higher local Nusselt number.
5. Growing the thermal layer along the channel will decrease the local Nusselt number value from its greatest value at the beginning of the duct till it gets it's fixed value when the thermal layer is completely grown, i.e. thermally fully developed.

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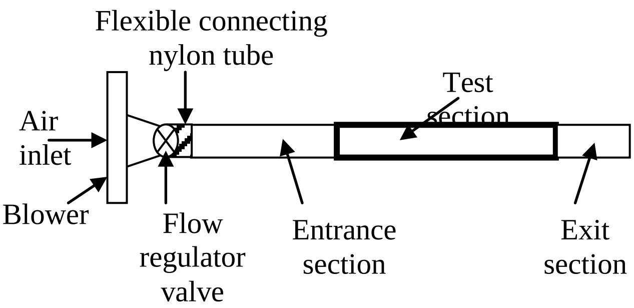

Figure (1) Schematics of experiment setup.

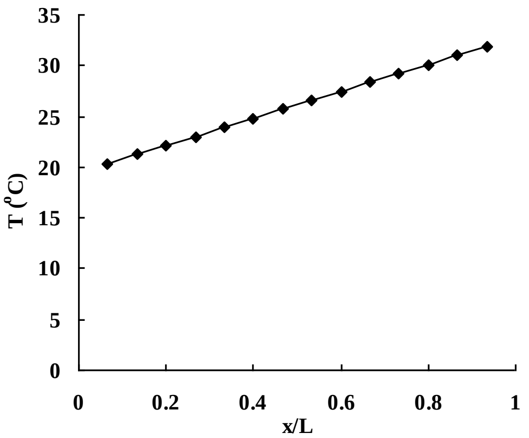

Figure (2) distribution of bulk air temperature along the channel at (2.54×10^{-3} kg/s) and (272 W/m^2).

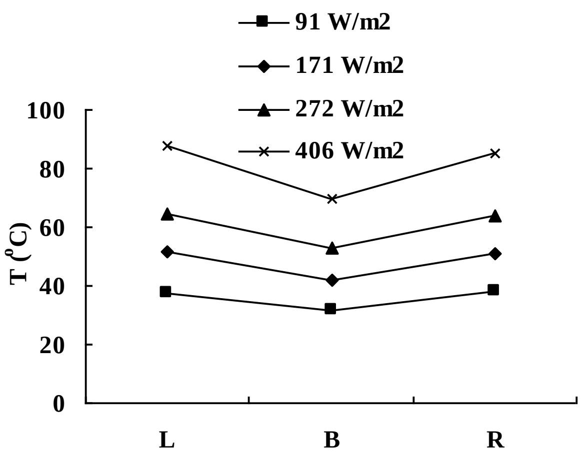

Figure (3) Variation of circumferential surface temperature at (x=110 cm) and ($\dot{m}=3.03\times10^{-3}$

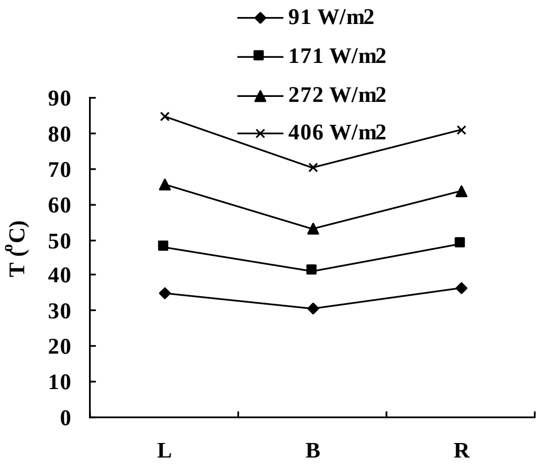

Figure (4) Variation of circumferential surface temperature at (x=110 cm) and ($\dot{m}=2.54\times10^{-3}$ kg/s).

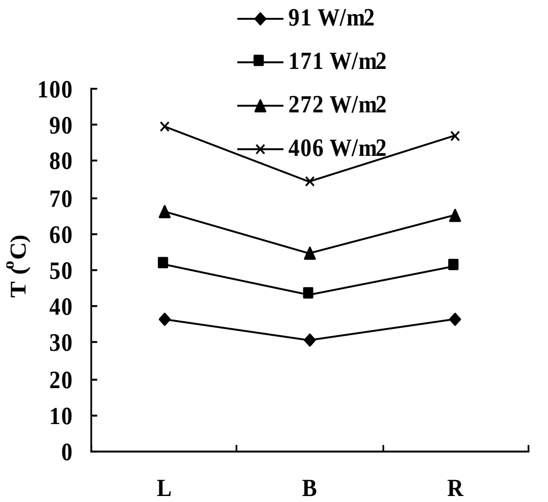

Figure (5) Variation of circumferential surface temperature at (x=110 cm) and ($\dot{m}=1.91\times10^{-3}$ kg/s).

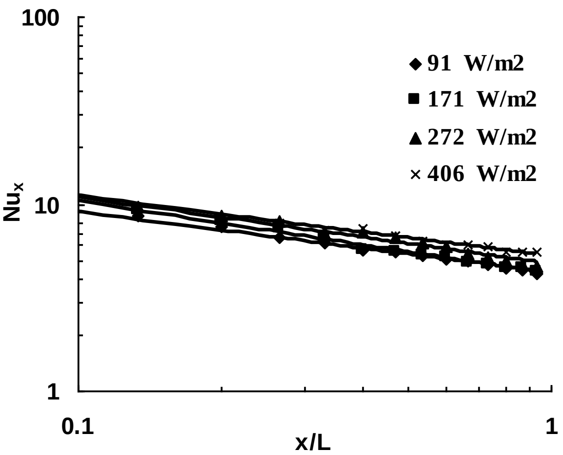

Figure (6) Effect of changing heat flux on local Nusselt number at ($\dot{m}=1.91\times10^{-3}$ kg/s).

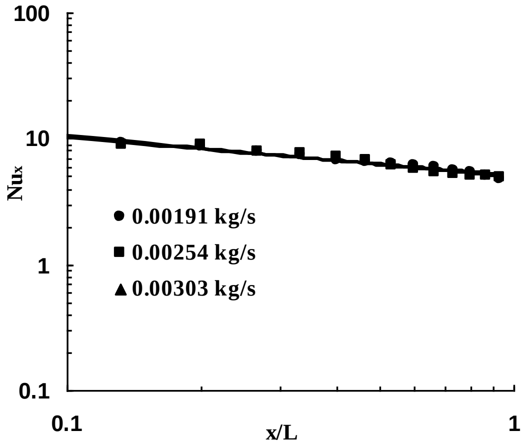

Figure (9) Effect of changing mass flow rate on local Nusselt number at (91 W/m^2).

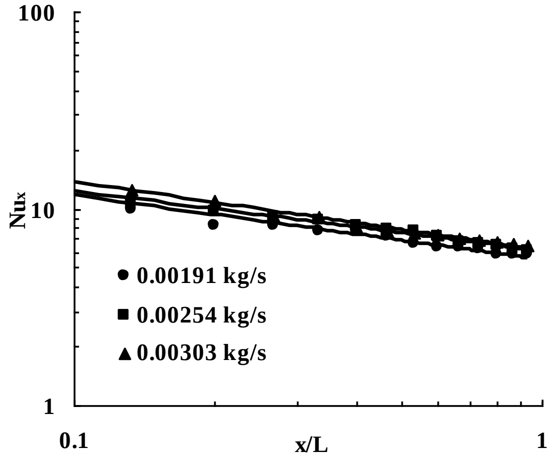

Figure (12) Effect of changing mass flow rate on local Nusselt number at (406 W/m^2).

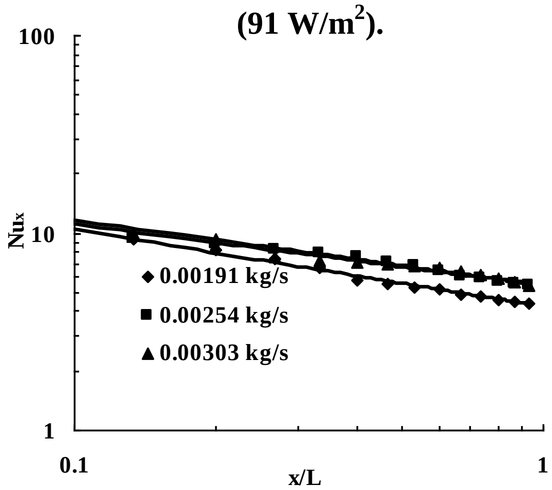

Figure (10) Effect of changing mass flow rate on local Nusselt number at (171 W/m^2).

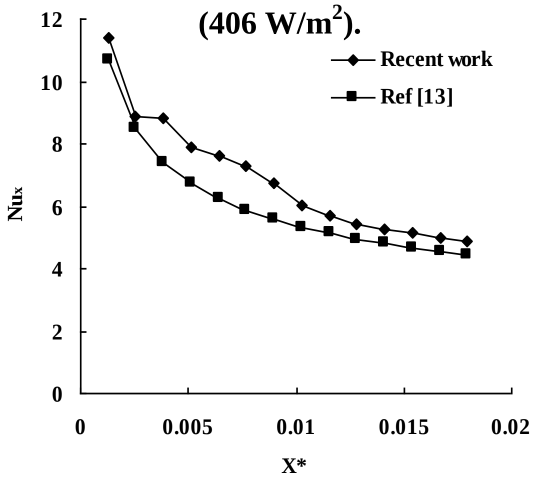

Figure (13) Comparison between recent work and ref. [13] at ($\dot{m}$ =2.54×10^{-3} kg/s) and (91 W/m^2).

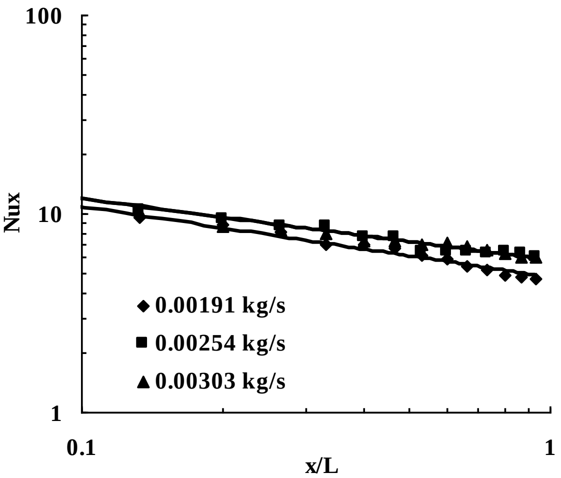

Figure (11) Effect of changing mass flow rate on local Nusselt number at (272 W/m^2).

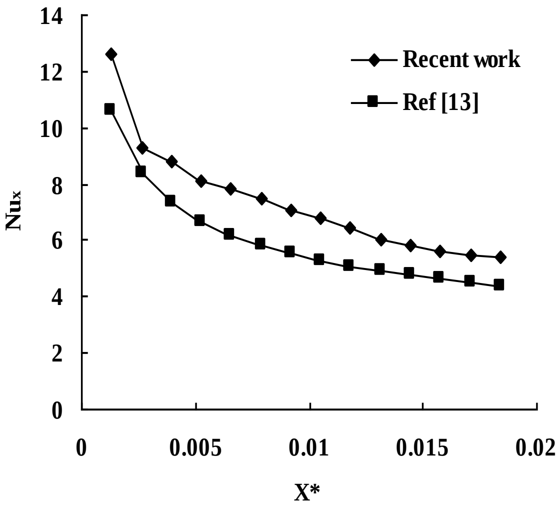

Figure (14) Comparison between recent work and ref. [13] at ($\dot{m}$ =2.54×10^{-3} kg/s) and (171 W/m^2).

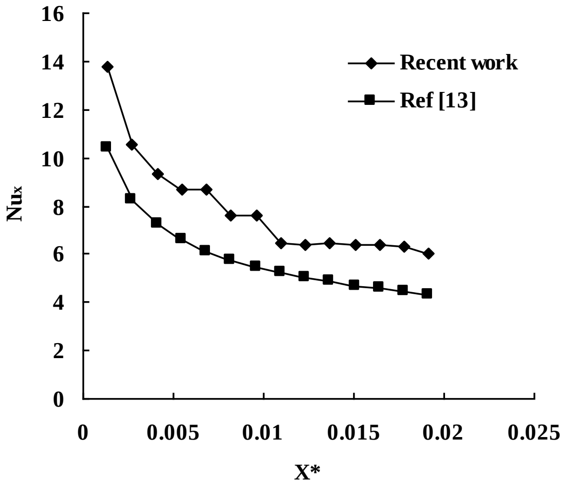

Figure (15) Comparison between recent work and ref. [13] at ($\dot{m}$=2.54×10^{-3} kg/s) and (272 W/m^2).

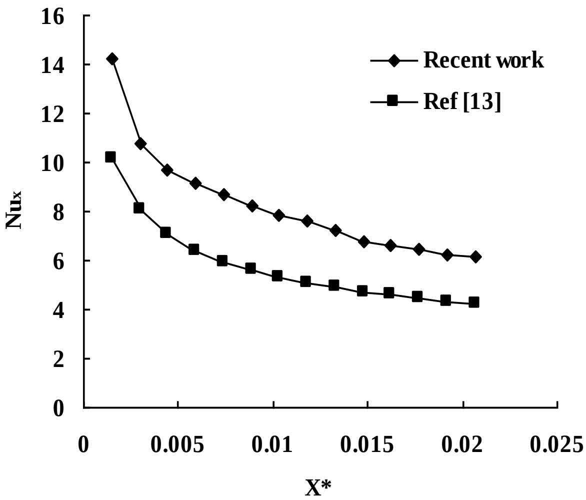

Figure (16) Comparison between recent work and ref. [13] at ($\dot{m}$=2.54×10^{-3} kg/s) and (406 W/m^2).