

EXPERIMENTAL STUDY OF NATURAL CONVECTION HEAT TRANSFER THROUGH HORIZONTAL OPEN ENDED EQUILATERAL TRIANGULAR CHANNELS

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Experimental study of natural convection heat transfer inside smooth and rough surfaces of horizontal equilateral triangular channels with a uniformly heated surface are performed. The effect of smooth and rough surface of average roughness, $r_a = 0.02\mu\text{m}$, on the heat transfer characteristics are studied. The local and average heat transfer coefficients and Nusselt number are obtained for smooth and rough channel at different Rayleigh numbers from 6.45×10^5 to 4.45×10^6 . The findings show that the values of temperature difference between the inside surface and ambient air increase with increase of axial distance from both ends of the channel until a maximum value at the middle of the channel. The results show a higher values of local (N_{ux}) for rough channel along the axial distance compared with the smooth channel. The average Num of rough channel is higher than Num of smooth channel by about 7%. The results obtained are correlated using dimensionless groups for both rough and smooth surfaces of the equilateral horizontal triangular channels.

KEY WORDS: *Natural Convection, Constant heat flux, Horizontal Triangular Channels, Rough surface, Smooth surface*

NOMENCLATURE

A_c Cross-sectional area, m^2	q Heat flux, Q/A_s , W/m^2
A_s Surface area, m^2	Ra Rayleigh number, $Gr Pr$
C Constant	r_a Average roughness, μm
c_p Specific heat of air at constant pressure, $\text{J}/\text{kg K}$	Ri Richardson number, Gr/Re^2
D Hydraulic diameter of triangular channel, m	S Fin spacing, m
g Gravity acceleration, m/s^2	T_∞ Ambient air temperature, K
Gr Grashof number, $g\beta qD^4/k v^2$	T_s Inside surface temperature of triangular channel, K
H_f Fin height, m	T_{ms} Average inside surface temp. of triangular channel, K
h_x Local convective heat transfer coefficient, $q/\Delta T_s$, $\text{W}/\text{m}^2 \text{K}$	T_{mf} Mean film temperature, K

h_m	Average convective heat transfer coefficient, $W/m^2 K$	x	Axial distance measured from triangular channel entrance, m
k	Thermal conductivity, $W/m K$	ΔT_s	Temperature difference, $(T_s - T_\infty), C$
L	Triangular channel length, m	Greek Letters	
Nu_x	Local Nusselt number, hD/k	β	Volumetric coefficient of thermal expansion, K^{-1}
Nu_m	Average Nusselt number, $h_m D/k$	μ	Dynamic viscosity of air, $kg/m s$
Pr	Prandtl number, $c_p \mu/k$	ν	Kinematic viscosity of air, (μ/ρ) , m^2/s
Q	Electric power of main heater, W	ρ	Density of air kg/m^3

1. INTRODUCTION

Natural convection heat transfer has gained considerable attention because of its applications in many practical fields in the area of energy conservation, design of solar collectors, heat exchangers, nuclear engineering, cooling of electrical and electronic equipment and many others. The increasing interest in developing compact and highly efficient heat exchangers motivated researchers to study heat transfer from tubes of non-circular cross section, (elliptic, rectangular, square, ... etc). Shehata [1] studied natural convection inside vertical, inclined and horizontal annulus tube of radius ratio of 0.73 with heated inner tube and adiabatic outer tube. The results showed that the maximum heat transfer coefficient was obtained at the vertical position and it was decreased with decrease the inclination angle. Natural convection inside open ended horizontal and vertical annuli of tube with different annulus ratio was also studied experimentally by Sarhan et al. [2]. Correlations of the dimensionless group of average $Nu_m - Ra$ were obtained for the horizontal and vertical positions. The results showed that the cooling by natural convection inside the annuli decreases with the increase of radius ratio. Abdul-Aziz. [3] studied heat transfer by natural convection from the inside surface of a uniformly heated tube at different inclination angles. The experiments were conducted in the range of Ra from 1.44×10^7 to 8.85×10^8 , L/D from 10 to 31.4 and angle of inclination from 0 to 75. The maximum average Nu_m was recorded for vertical position. Omara et al. [4] studied experimental investigation of natural convection inside horizontal elliptic tube with different angle of attack at uniformly heated surface. The angle of attack (α) was varied from 0 to 90 with steps of 15. The experiments covered a range of Rayleigh number (Ra) from 1.45×10^6 to 1.78×10^7 . The results show that the angle of attack has a significant effect on the heat transfer coefficient. The values of temperature grow up along the axis of the tube from both ends until reaching to the maximum value at the middle of the tube. Also, the local (Nu) is increased with the increase of (α) at the same axial distance. The average Nu_m is increased with the increase of (α) at the same Ra .

Dogan and Sivrioglu [5] studied experimentally the mixed convection heat transfer from longitudinal fins in a horizontal rectangular channel. Reynolds number, Re , was always about 1500. Experiments were conducted for modified Rayleigh numbers $3 \times 10^7 < Ra < 8 \times 10^8$ and Richardson number $0.4 < Ri < 5$. Dimensionless fin spacing was varied from $S/H = 0.018$ to $S/H = 0.04$ and fin height was varied from $H_f/H = 0.25$ to $H_f/H = 0.80$. For mixed convection heat transfer, the results showed that the optimum fin spacing which yields the maximum heat transfer is $S = 8-9$ mm and

optimum fin spacing depends on the value of Ra. Nada [6] studied natural convection heat transfer in horizontal and vertical closed narrow enclosures with heated rectangular finned base plate. The study has been experimentally investigated at a wide range of Rayleigh number (Ra) for different fin spacing and fin length values. The results gave an optimum fin spacing for which Nusselt number (Nu_m) and finned surface effectiveness (ϵ) are maximum. It has been found that increasing fin length increases Nu_H and (ϵ); also increasing Ra increases Nu_m for any fin-array geometries. Achenbach [7] studied the effect of surface roughness on the heat transfer from a circular cylinder to the cross flow of air. In his study Reynolds number was varied from 2.2×10^4 to 4×10^6 . The variation of the roughness parameter was $0 < \epsilon/D < 900 \times 10^{-5}$. Particular attention has been paid to the transition from a laminar to a turbulent boundary layer as a function of Reynolds number and roughness parameter. Abdmonem and Michel [8] studied the effect of surface roughness on the average heat transfer of an impinging air jet. The roughness took the shape of a circular array of protrusions of 0.5 mm base and 0.5 mm height. The results indicated an increase up to 6.0% of the average Nusselt number due to surface roughness.

Varol et al. [9] studied natural convection in a triangle enclosure with flush mounted heater on the wall, the study of natural convection heat transfer in triangular enclosure was analyzed numerically at different parameters, including the aspect ratio of triangle ($AR = 1$ and 0.6), Rayleigh number and both length and position of heater. The results showed that the flow and temperature field are affected by the shape of enclosure and Rayleigh numbers play an important role on them. Both position and location of heater affect the flow circulation and heat transfer. Koca et al. [10] studied the effects of Prandtl number on natural convection in triangular enclosures with localized heating from below. The study has been analyzed by solving governing equations of natural convection in stream function–vorticity form with finite-difference technique. The results showed that computations were carried out for dimensionless heater locations ($0.15 \leq s \leq 0.95$), dimensionless heater length ($0.1 \leq W \leq 0.9$), Prandtl number ($0.01 \leq Pr \leq 15$) and Rayleigh number ($10^3 \leq Ra \leq 10^6$). It is observed that both flow and temperature fields are affected by variation of Prandtl number, location of heater and length of heater as well as Rayleigh number. Varol et al. [11] studied numerically analysis of the entropy production due to free convection in partially heated isosceles triangular enclosures from below and symmetrically cooled from sloping walls, having inclination angle of $30^\circ \leq \theta \leq 60^\circ$. The results showed that for Rayleigh number ($10^3 \leq Ra \leq 8.8 \times 10^5$). Dimensionless entropy production number increases with increasing of Rayleigh number and increasing of sloping wall inclination angle of the triangular enclosure. An increase in Rayleigh number decreases the Bejan number. Nusselt number increases with increasing of Rayleigh number and length of heater.

In the present work natural convection heat transfer through horizontal open ended equilateral triangular channel is experimentally studied. The effect of the inner surfaces roughness is taken into considerations.

2. EXPERIMENTAL SET UP AND TEST PROCEDURES

The experimental test rig is shown diagrammatically in Fig. 1. It consists of a horizontal equilateral triangular channel (test section) mounted on a frame.

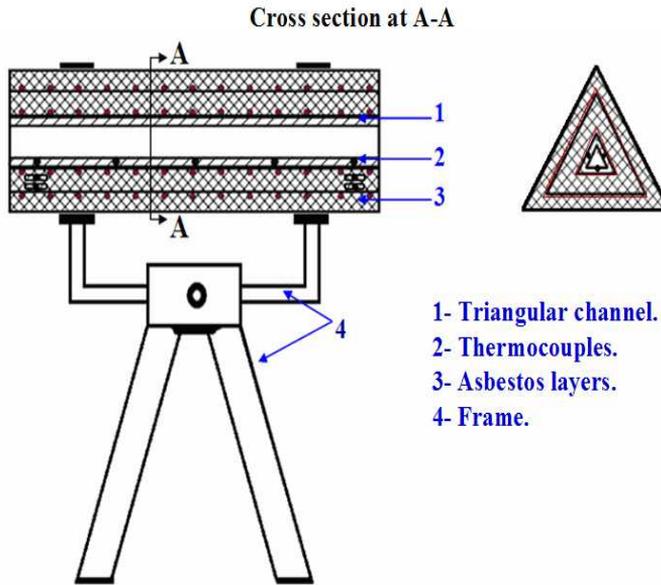


Fig. 1. Experimental setup

The equilateral triangular channel of length 500 mm is made from copper of 1.5 mm thickness. The outer surface of the triangular channel is completely covered with an electricity insulating tape as shown in Fig. 2. A nickel–chrome wire of 0.4 mm diameter is uniformly wound to form the main heater. The main heater is covered with an asbestos layer of 45 mm thickness surrounded by another nickel–chrome wire of 0.4 mm diameter was wound uniformly to form a guard heater. The guard heater is covered with a 30 mm thick asbestos layer. Two pairs of thermocouples are installed in the asbestos layer between the main heater and the guard heater. The thermocouples of each pair were fixed on the same radial line. The guard heater is adjusted so that, at steady state, the readings of the thermocouples of each pair became practically the same. Thus, all the energy generated by the main heater should be flown inward to the triangular channel.

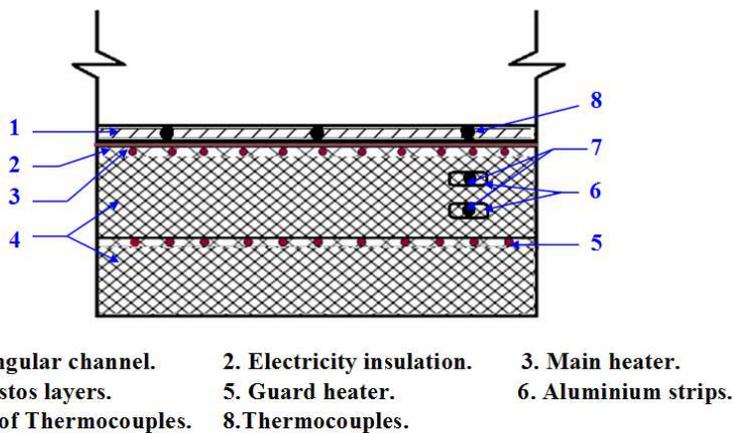


Fig. 2. Heaters arrangement

The inner surface temperature of the triangular channel is measured by 15 chromel-alumel thermocouples of 0.4 mm diameter soldered in slots milled along the axial and circumferential directions. The distribution of thermocouples is located for five measuring stations for each face at axial distances of 50, 150, 250, 350 and 450 mm from one end of the triangular channel as shown in Fig. 3.

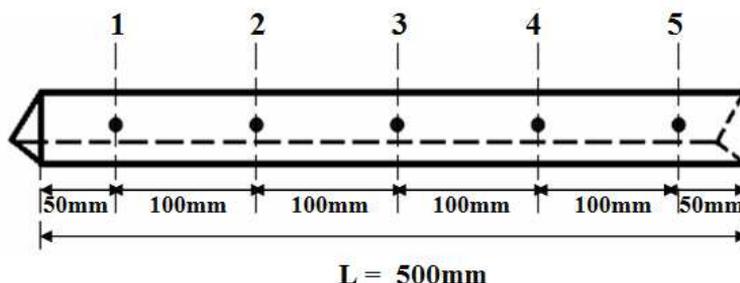


Fig. 3. Distribution of the thermocouples on the triangular channel

The ambient air temperature was measured by two thermocouples fixed at the triangular channel ends. All temperatures are measured by digital thermometer capable of reading ± 0.1 C. The input electric power was regulated by AC power variance and is measured by a digital wattmeter with a resolution of ± 0.01 W. The experiments were conducted with the test section in a closed room $2.5\text{ m} \times 2.5\text{ m}$ of plastic transparent walls to prevent currents of air. The measuring instruments are mounted outside of this room. The input electric power to the main heater is controlled and changed by the AC variac for each experiment. The steady state condition for each run was achieved after 3 – 4 h approximately. The steady state condition is considered to be achieved when the temperature reading of each thermocouple was not changed by more than ± 0.5 C within 20 min. When the steady state condition was established, the readings of all thermocouples, the input power, the inside surface temperatures of triangular channel and the ambient air temperature were recorded.

Two equilateral triangular channels one has a smooth surfaces and the second has a rough surfaces of $0.02\mu\text{m}$ roughness were used for the present study.

3. UNCERTAINTY ANALYSIS

Generally, the accuracy of the experimental results depends upon the accuracy of the individual measuring instruments and the manufactured accuracy of the triangular channel. Also, the accuracy of an instrument is limited by its minimum division (its sensitivity). In the present work, the uncertainties in both the heat transfer coefficient (Nusselt number) and Rayleigh number are estimated by using the differential approximation method. For a typical experiment, the total uncertainty in measuring the main heater input power, temperature difference (ΔT_s), heat transfer rate and the triangular channel surface area were $\pm 0.2\%$, $\pm 0.5\%$, $\pm 2.7\%$ and $\pm 0.23\%$, respectively. These are combined to give a maximum error of $\pm 3.5\%$ in heat transfer coefficient (Nusselt number) and a maximum error of $\pm 4.7\%$ in Rayleigh number.

4. DATA REDUCTION

In the present work, the local heat transfer coefficient, h_x , between the inner surface of the triangular channel and the air inside the triangular channel is calculated from Eq. (1).

$$h_x = q / (T_s - T_\infty) \quad (1)$$

The temperature difference between the inner surface of the triangular channel and the air inside the triangular channel is given by:

$$\Delta T_s = (T_s - T_\infty) \quad (2)$$

The corresponding local Nusselt number, Nu is calculated from Eq. (3).

$$Nu_x = h_x \times D/k \quad (3)$$

The average heat transfer coefficient between the inner surface of the triangular channel and the air inside the triangular channel is calculated from the following equation:

$$h_m = \frac{1}{L} \int_{x=0}^{x=L} h_x dx \quad (4)$$

Also, the average inner surface temperature is calculated from Eq. (5)

$$T_{ms} = \frac{1}{L} \int_{x=0}^{x=L} T_{sx} dx \quad (5)$$

The corresponding average Nusselt number Nu_m is calculated from Eq. (6).

$$Nu_m = h_m \times D/k \quad (6)$$

The Rayleigh number Ra is calculated from Eq. (7).

$$Ra = Gr \times Pr \quad (7)$$

The physical properties are evaluated at the mean film temperature, as in [12] as:

$$T_{mf} = (T_{ms} + T_\infty)/2 \quad (8)$$

5. RESULTS AND DISCUSSION

The results obtained in this work are the temperature distribution on the inner surface of the triangular channel, local Nu and average Nu_m at Ra values from 6.45×10^5 to 4.45×10^6 , for smooth and rough channel (average roughness $r_a = 0.02\mu\text{m}$).

5.1. Temperature Distribution

The measurements of the temperature on the inner surface of the triangular channel for all the experiments showed that the variation in temperature readings is only axially. Generally, the maximum difference in the temperature readings ΔT_s for a horizontal triangular channel occurs at the middle of the channel for different heat fluxes.

Figure 4 shows a parabolic variation of the temperature difference (ΔT_s) with axial distance (x) for inner smooth surface. The experiments were performed at different values of Ra from $6.45 \times 10^5 \leq Ra \leq 4.33 \times 10^6$. The surface temperatures are gradually increased from both ends of the triangular channel with a maximum value at

the middle point of the triangular channel. The general shape of all the curves gives symmetry around the middle point of the triangular channel. The reason for this shape of ΔT_s - x curves is contributed to growth of the boundary layer from both ends along the axial distance.

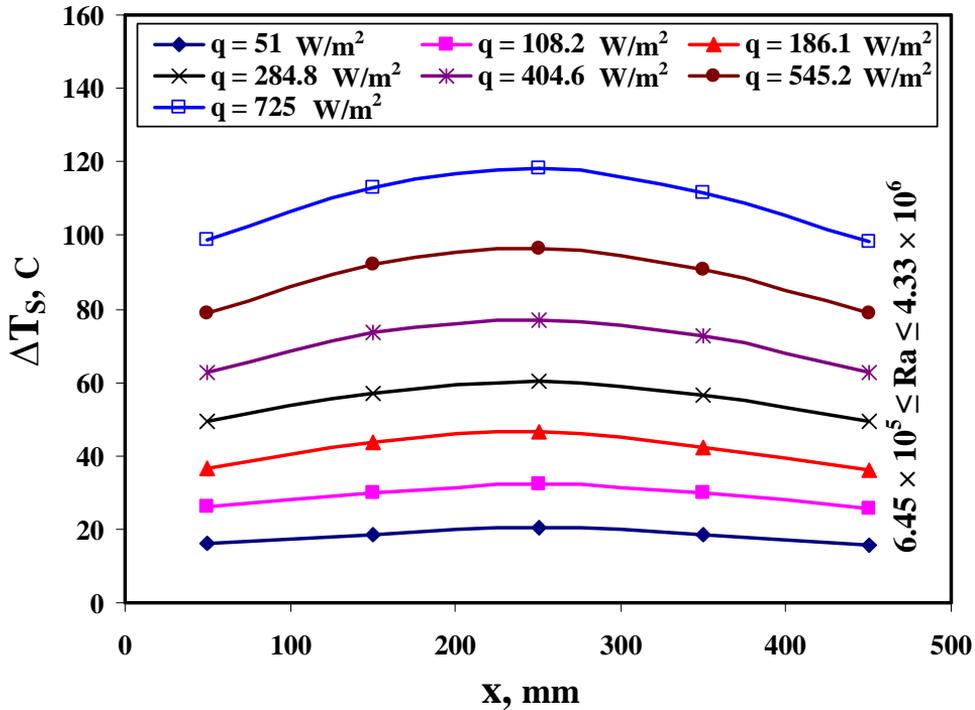


Fig. 4. Variation of surface temperature difference (ΔT_s) with axial distance (x) for smooth channel at different values of heat flux.

Figure 5 shows a comparison between the temperature difference (ΔT_s) for rough and inner smooth surfaces along the axial distance at $q = 404.6 \text{ W/m}^2$. The temperature difference variations along the axial distance represent similar parabolic curves with low values for rough surface. The lowest values of the temperature difference for the rough surface would be occurred due to increase in the effective inner surface area due to its roughness; or can be explained by the increase in the heat transfer coefficient while heat flux is constant.

5.2. Local Nusselt Number, Nu_x

Figure 6 illustrates the variation of local Nusselt number (Nu_x) along the axial distance of the triangular channel at the different heat fluxes. The local Nusselt number (Nu_x) has a maximum value at both ends of the triangular channel, and then is decreased gradually until it reaches to a minimum value at the middle of the triangular channel. This can be explained by the growth of the boundary layer from both ends toward the middle of the triangular channel causing the decrease of the convective heat transfer coefficient and consequently decreases the Nu_x towards the middle of the triangular channel.

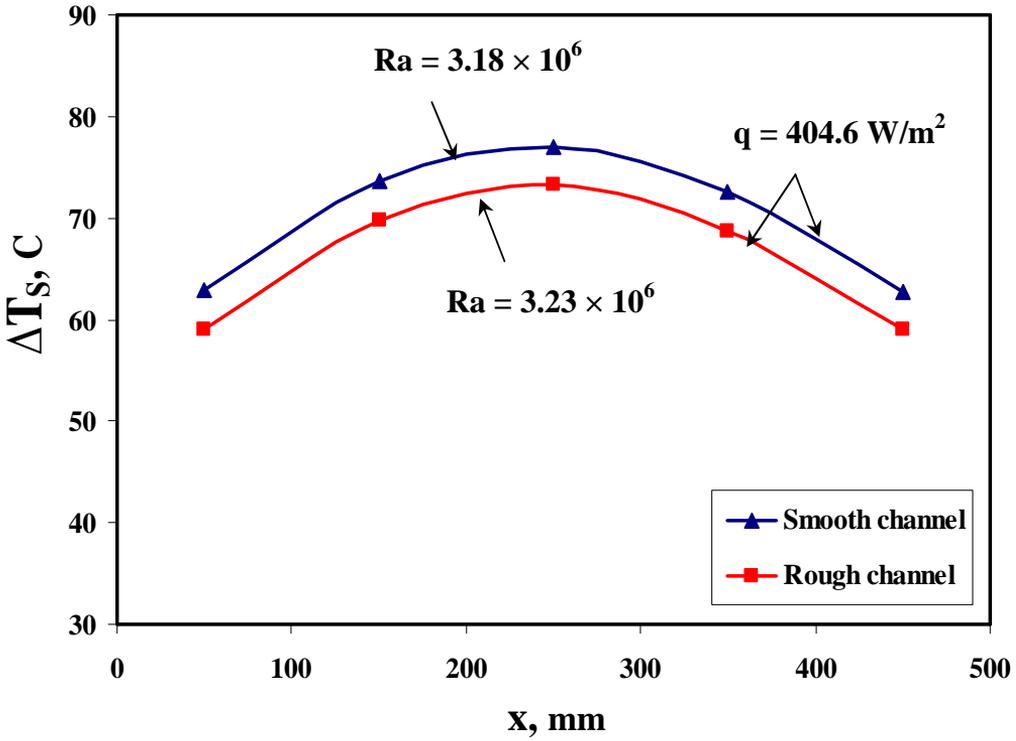


Fig. 5. Comparison between smooth and rough channels in (ΔT_s) with axial distance (x) at $q = 404.6 \text{ W/m}^2$.

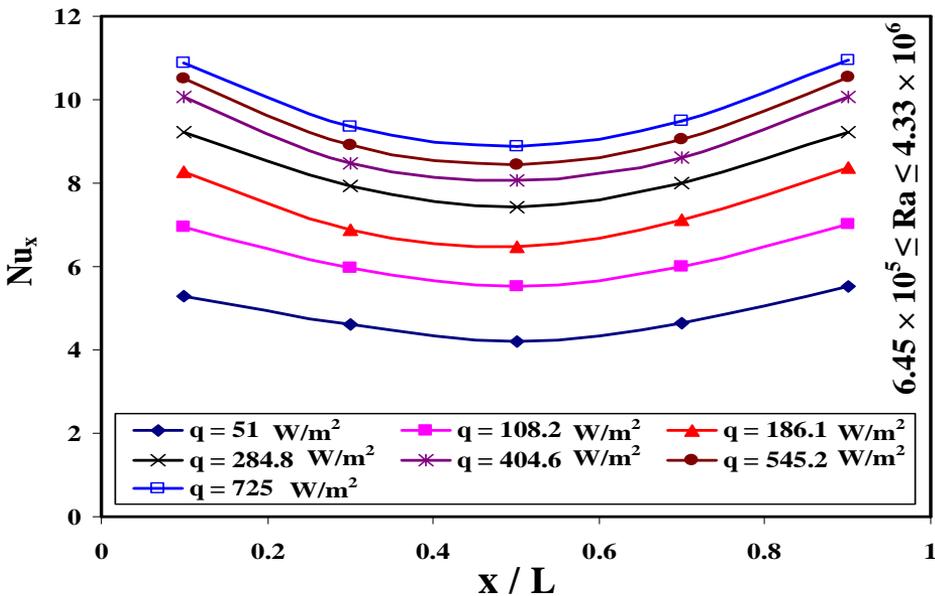


Fig. 6. Variation of the local Nusselt number (Nu_x) with axial distance along the channel (x/L) .

Figure 7 shows a comparison between the local Nu_x for inner rough and smooth surfaces along the axial distance at $q = 404.6 \text{ W/m}^2$. The local Nu_x values vary along the axial distance in a form of parabolic curves with higher values for the rough surface. The lower values of the local Nu_x number for the inner smooth surface compared to those values obtained for the rough surface are attributed again to the relative increase in the heat transfer coefficient along the axial distance.

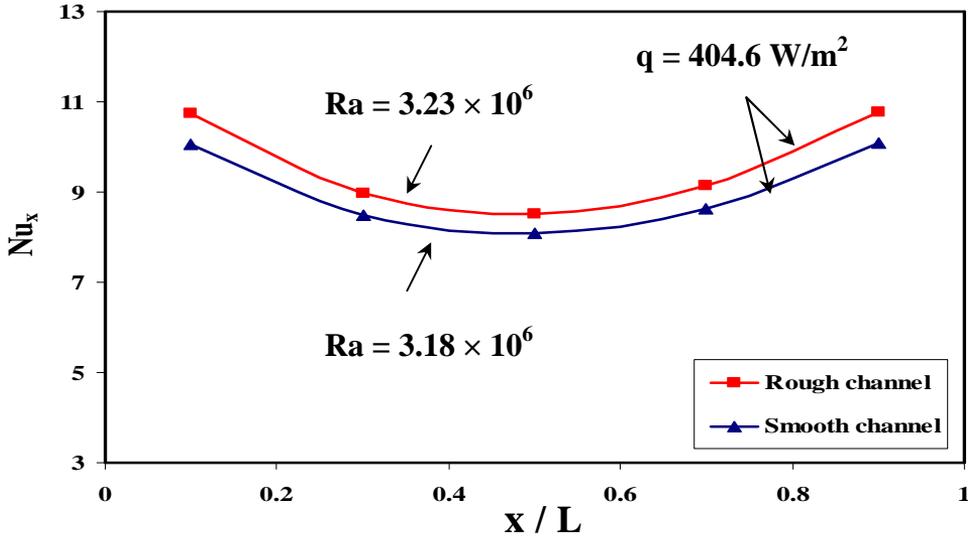


Fig. 7. Comparison between smooth and rough channel in (Nu_x) with (x/L) at $q = 404.6 \text{ W/m}^2$.

The variation of the average Nusselt number, Nu_m , with Ra for smooth and rough channels is shown in Fig. 8. The value of Nu_m increases with the increase of Ra for smooth and rough inner surfaces. Also, comparing Nu_m versus Ra for smooth and rough inner surfaces shows that Nu_m values for rough surface are higher than Nu_m values for smooth surface.

5.3. Correlation of the Results

The experimental results were fitted using power regression in the following form:

$$Nu_m = C Ra^n \tag{9}$$

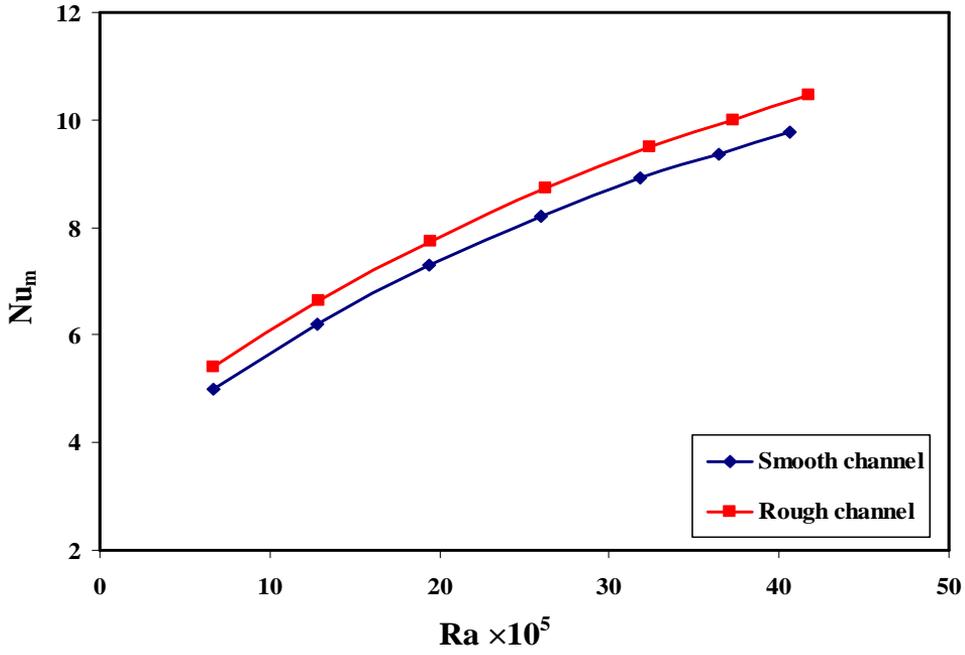
Where C and n are constants.

Evaluations of the empirical correlations for both smooth and rough surface channels are as followings:

For smooth channel $Nu_m = 0.014 Ra^{0.43}$, $6.45 \times 10^5 \leq Ra \leq 4.33 \times 10^6$ (10)

For rough channel ($r_a = 0.02\mu\text{m}$) $Nu_m = 0.015 Ra^{0.43}$, $6.51 \times 10^5 \leq Ra \leq 4.45 \times 10^6$ (11)

Equations (10) and (11) represent the general equation for natural convection inside horizontal equilateral triangular channels for $6.45 \times 10^5 \leq Ra \leq 4.45 \times 10^6$. The calculated data from Equations (10) and (11) of the average Nusselt number (Nu_{mCal}) are plotted against experimental data of the average Nusselt number (Nu_{mExp}) in Fig. 9 (a) & (b). As noted from figure, the maximum deviation between the experimental data and the correlated equations are $\pm 5.4\%$ for smooth channel and $\pm 8.2\%$ for rough channel.



Fig

. 8. Comparison between smooth and rough channels on the average Nusselt Nu_m for different values of Ra .

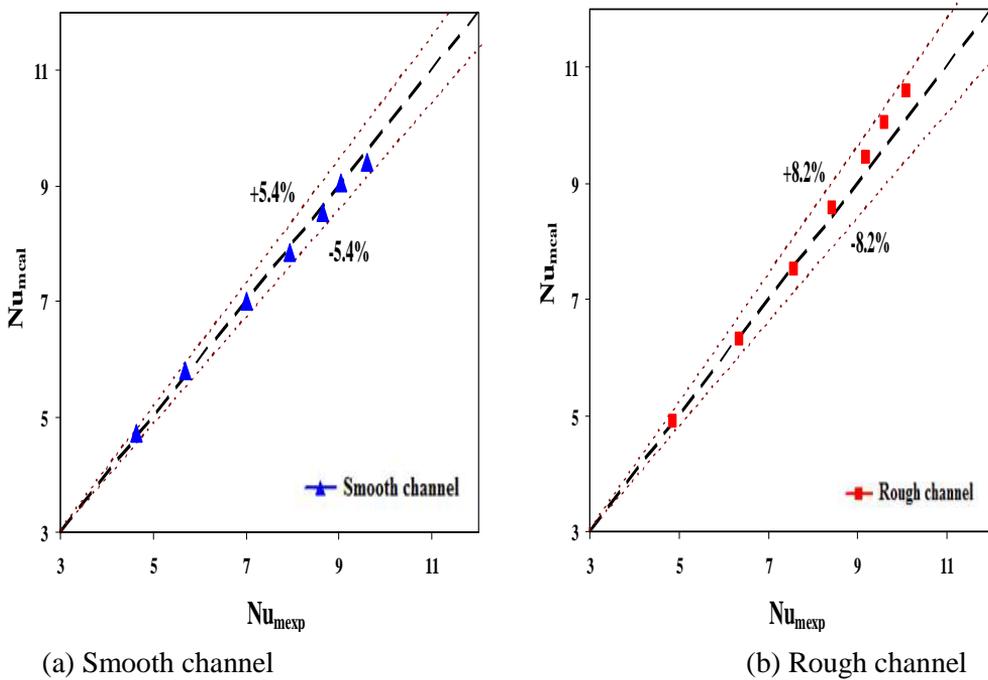


Fig. 9. Average Nu_{mcal} versus Average Nu_{mexp} for smooth and rough horizontal triangular channels.

6. COMPARISON WITH THE PREVIOUS WORK

Comparison of the present work with that of Sarhan et al. [3], Omara et al. [4] and Varol et al. [9] is shown in Fig. 10. The present results of natural convection inside horizontal equilateral triangular channels (smooth channel and rough channel) have lower values than the results of Sarhan et al. [3] for natural convection inside horizontal annuli of circular tube and Omara et al. [4] and for natural convection inside horizontal elliptic tube. But the results of Nu_m for present work (triangular channels) are higher than the results of Varol et al. [9] for natural convection in triangle enclosure with aspect ratios ($AR = 1$ and 0.6).

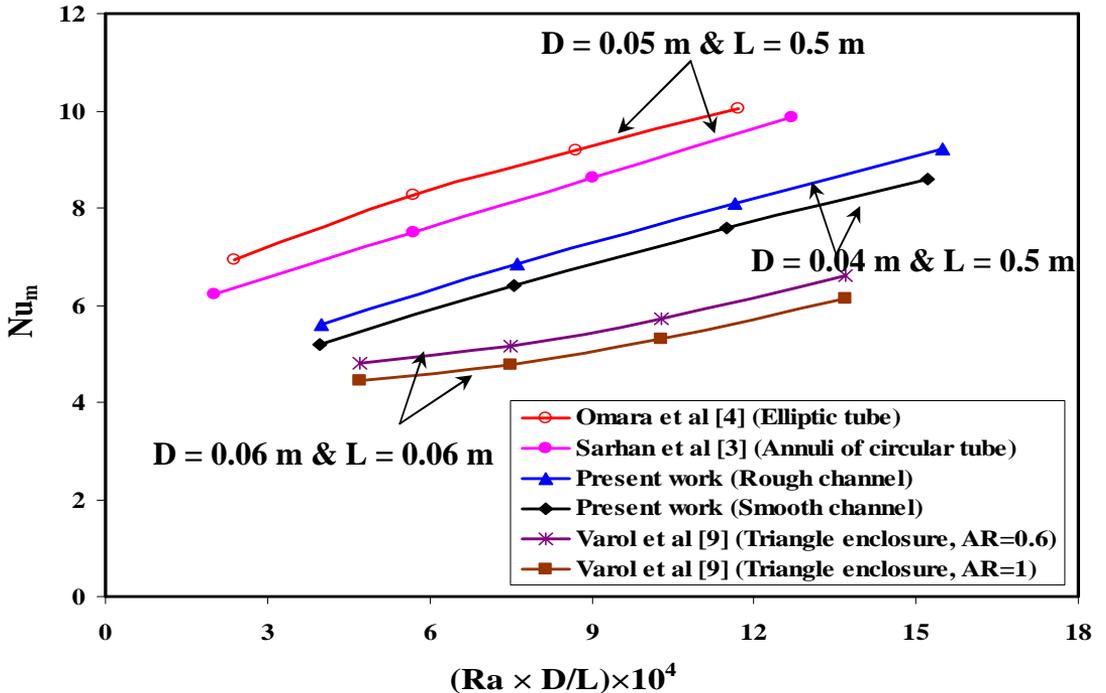


Fig. 10. Comparison of the present work versus Sarhan et al. [3], Omara et al [4] and Varol et al [9].

7. CONCLUSIONS

Natural convection heat transfer through the inside surface of a horizontal triangular channels with a uniformly heated surface is investigated experimentally. The average roughness of rough channel was ($r_a = 0.02\mu\text{m}$). The experiments covered a range of Rayleigh number, Ra from 6.45×10^5 to 4.45×10^6 . The local and average heat transfer coefficients and Nusselt number are estimated for smooth and rough channels at different values of Rayleigh numbers. The following main points can be drawn from this study:

- The value of the local Nu_x in rough channel is higher than Nu_x in smooth channel.

- The values of temperature difference between the surface and the ambient increase with the increase of axial distance from both ends of the channel until a maximum value at the middle of the channel.
- The average Nu_m increases with increasing Ra as expected.
- The average Nu_m of rough channel is higher than Nu_m of smooth channel and the results indicated an increase of up to 7% of the average Nusselt number due to surface roughness.
- Correlations of Nusselt number for natural convection inside open ended horizontal triangular channels (smooth channel and rough channel) are presented [Eqs 10 & 11] respectively.

8. REFERENCES

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دراسة عملية لانتقال الحرارة بالحمل خلال مجري مفتوحة الطرفين ذات مقطع مثلي متساوي الأضلاع

يقدم هذا البحث دراسة عملية لعملية إنتقال الحرارة بالحمل خلال مجري ذات مقطع مثلث متساوي الأضلاع بالتسخين المنتظم لسطح الأنبوية. تم دراسة تأثير نعومة وخشونة السطح الداخلي بدرجة خشونة تساوي 0.02 ميكرومتر علي خصائص إنتقال الحرارة. تم الحصول علي رقم ناسلت الموضعي والمتوسط علي الأسطح الناعمة والخشنة لعدد رايلي رقم من 6.45×10^5 إلي 4.45×10^6 . وضحت النتائج التجريبية تأثير خشونة السطح الداخلي علي قيم فرق درجات الحرارة بين سطح المجري إلي الهواء الجوي والذي تبين أنه يزيد مع زيادة المسافة المحورية من كلا النهايتين حتي تصل الي أعلى القيم عند منتصف المجري . وضحت النتائج التجريبية قيم مرتفعة لعدد ناسلت الموضعي للأسطح الخشنة خلال المسافة المحورية مقارنة بالأسطح الناعمة. حيث كانت قيم رقم ناسلت المتوسط للسطح الخشن أعلى من قيم رقم ناسلت المتوسط للسطح الناعم بمقدار 7% ، تم الحصول علي معادلة تجريبية باستخدام المجموعات اللابعدية لكل من الأسطح الناعمة والخشنة للمجري مثلث المقطع متساوي الأضلاع.