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Natural Convection from Heated Rough Surface at the Bottom of Vented Rectangular Enclosure

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ABSTRACT

Natural convection heat transfer from tilted rectangular enclosure heated at the bottom rough surfaces wall and vented by uniform slots opening at top wall experimentally investigated. Rough surfaces of roughness 0.002 m are used to study their effect on the heat transfer characteristics. The experiments are carried out to study the effects of venting ratio, enclosure's tilt angle and Rayleigh number on the cooling of rough surface inside the enclosure. The experiments are carried out at a Rayleigh number ranging from 2 × 108 to 1.52× 109 for enclosure tilt angles ranging from 0º to 90. Top venting arrangement is studied at different venting ratios of 1, 0.75, 0.5 and 0.25. Roughness shows a large effect on heat transfer for the rectangular enclosure where the average Nusselt number increases with the increase of venting ratio and decrease enclosure's tilt angle at the same Rayleigh number. This can be attributed to the roughness may increase the blockage effect on the flow that can cause the buoyancy force to decrease, but on the other hand it increases the turbulence intensity resulting in a higher heat transfer. The results are compared with a smooth rectangular enclosure of the same surface area to study the effect of roughness on heat transfer. The average Nu of rough surface in rectangular enclosure is higher than that of smooth surface by the range from 12 % to 21% depending on Ra. Correlations are developed for the top venting arrangement to predict the average Nusselt number of the enclosure in terms of the Rayleigh number, venting ratio and enclosure tilt angle.

Keyword: Natural convection; Enclosure; Rough surface; venting arrangement.

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Nomenclature

q''	input heat flux, W/m^2
Ra	modified Rayleigh number based on the input heat flux
T	local temperature of the heated surface, ºC
T_s	average surface temperature of the heated surface, ºC
T_∞	ambient temperature, ºC
t	thickness of enclosure's wall, m
ΔT	temperature difference across enclosure wall thickness, ºC
θ	tilt angle of enclosure.
α	thermal diffusivity, m^2/s
β	coefficient of volumetric thermal expansion at constant pressure, 1/K
ε	emissivity
σ	Stefan-Boltzman constant.
υ	kinematic viscosity, m^2/s
A	area of a wall of the enclosure, m^2
A_s	area of the heated surface, m^2
F	view factor, dimensionless
g	acceleration of gravity, m/s^2
H	depth of the enclosure, m
h	local value of heat transfer coefficient, W/m^2 K
$\bar{h}$	average value of heat transfer coefficient, W/m^2 K
I	electric current, amp.
k_a	thermal conductivity of air, W/m K
k_w	thermal conductivity of wall, W/m K
L	length of enclosure, m
Nu	local Nusselt number, dimensionless.
$\overline{Nu}$	average Nusselt number, dimensionless.
VR	venting ratio, dimensionless.
q	rate of heat transfer by free convection, W
q_c	rate of heat losses by conduction,W
q_r	rate of heat losses by radiation, W

I. INTRODUCTION

Natural convection from heated surface inside rectangular enclosure with different parameters was receiving growing interest in the last few decades because of its employment in many practical fields in the areas of energy conservation. Natural convection in open cavities or vented enclosures is getting more attention due to the importance of such geometry in solar thermal central receiver system, where heat losses affect the performance of the system. Further, applications of cavities include aircraft-brake housing system, pipes connecting reservoirs of fluids with different temperatures, refrigerators, electronic cooling, energy saving household refrigerators, energy conservation in building micro electronic equipment, and many others. Roughness surfaces are proven to be a good technique to increase the heat transfer mechanism. However, in free convection blockage effect induced by the existence of rough surface is essential since the convection mechanism is weaker than that of forced convection. Roughness also increases the turbulent intensity on the surface, which causes heat transfer to increase.

Experimental and numerical works were carried out to better understand the heat transfer mechanism inside cavities. The studies are classified according to the type of the enclosure whether it was with or without venting. Adams et al. [1] and Yu and Joshi [2] studied numerically and experimentally the process of passive cooling of discretely heated enclosures by combined conduction, convection and radiation. They found that both conduction and radiation affect the rate of cooling of the enclosure, and neither can be neglected. Nada [3] studied natural convection heat transfer in horizontal and vertical closed narrow enclosures with heated rectangular finned base plate. The study was experimentally investigated at a wide range of Rayleigh number (Ra) for different fin spacing and fin length values. The results show that increasing fin length increases Average Nusselt number and finned surface effectiveness (ε); also increasing Ra increases Average Nusselt number for any fin-array geometry. Mahmoud et al [4] studied natural convection heat transfer inside smooth and rough surfaces of vertical and inclined equilateral triangular channels of different inclination angles with a uniformly heated surface are performed. The inclination angle is changed from 15 to 90. Smooth and rough surface of average roughness (0.02mm) are used and their effect on the heat transfer characteristics are studied. The results show that The average Nusselt number of rough channel is higher than that of smooth channel by about 8.1% for inclined case at $\theta = 45$ and 10% for vertical case. Showole and Tarasuk [5] studied numerically the heat transfer from fully open (OR=1) square cavities at different tilt angles. In these studies the four walls of the enclosure were at the same temperature. The studies were carried out for different ranges of Grashof number (Gr) between 10^3 and 10^7 and at different aspect ratios. The studies showed that the aspect ratio and the tilt angle affect the flow field within the cavity and the heat transfer coefficient.

Chan and Tien [6, 7], Angirasa et al. [8], Mohamad [9], Balaji and Venkateshan [10] and Lin and Xin [11] investigated numerically the heat transfer from fully open square horizontal cavities. In these studies the bottom horizontal wall of the cavity was isothermal at high temperature and the other walls were adiabatically insulated. Different ranges of Grashof number were studied. Miyamoto et al. [12] presented a two dimensional numerical study of the natural convection heat transfer from fully and partially open tilted cavities. Two opening ratios, 0.5 and 1, were studied. All the walls of the cavity were isothermal at the same temperature. Chakroun et al. [13] investigated experimentally the effect of the opening ratio of the cavity on the heat transfer coefficient in a tilted partially open square cavity. In this study the bottom surface of the cavity was heated with a constant heat flux, and the other side walls were adiabatically insulated. The opening in the cavity was one opening at the center of the top wall. The study showed the increase of the heat transfer coefficient with increasing the opening ratio.

Elsayed and Chakroun [14] studied experimentally the effect of the place of the opening on the passive cooling process of tilted partially open square cavities. The bottom surface of the cavity was heated at a fixed constant heat flux and the other four side walls were adiabatically insulated. Four positions (centered slit, high wall slit, low wall slit and uniform wall slots) of the opening slit, which was at the top wall, were studied. Stasiek [15] carried out experimental studies on the heat transfer and fluid flow across corrugated-undulated heat exchanger surfaces. Piao et al. [16, 17] investigated experimentally natural forced and mixed convection heat transfer in a cross-corrugated channel solar air-heater.

Ruhul [18] studied the effect of placing adiabatic roughness elements at the bottom of an enclosure cooled at the top and heated at the bottom. It was found that the presence of roughness elements on the hot horizontal wall increases heat transfer rate across the enclosure in comparison with a corresponding smooth walled enclosure. Most of the previous studies were carried out on rectangular cavities with different heating wall configurations and at different inclination angles. Shakerin et al. [19] have observed that the influence of the roughness is mainly localized to within about two roughness heights above and below the roughness location. Nevertheless they have not observed from dye visualization the presence of a flow-separation bubble inside the cavities between the ribs. The effects the influence of transverse roughness elements on transient natural convection from both dynamical and thermal viewpoints studied by Guillaume and Jacques [20]. Three different ribbed geometries have been tested. Both flow visualizations and heat transfer distributions indicate that the flow and heat transfer characteristics in the vicinity of the roughness are significantly affected by the protrusions. The flow features show that the separation of the viscous layer occurs in the ribbed region with the generation of complex eddy structures. The effects of roughness on heat transfer for semi-cylindrical cavity studied by Walid et al. [21]. Heat transfer measurements are performed on smooth and rough surfaces for different tilt angles. Roughness shows a large effect on heat transfer for the semi-cylindrical cavities. Roughness may increase the blockage effect on the flow that can cause the buoyancy force to decrease, but on the other hand it increases the turbulence intensity resulting in a higher heat transfer. Roughness elements may significantly affect the free convection heat transfer but how this modification is generated has not been fully examined. Yousaf and Usman [22] presented a study on natural convection heat transfer in a square cavity with sinusoidal roughness elements. Two-dimensional square cavity in the presence of roughness on vertical walls was studied numerically. Numerical study was performed for a range of the Rayleigh number from 10^3 to 10^6 for a Newtonian fluid of the Prandtl number 1.0. The sinusoidal roughness elements were located on a hot, and both the hot and cold walls simultaneously with varying number of elements and the dimensionless amplitude. Hydrodynamic and thermal behavior of fluid in the presence of roughness was analyzed in form of isotherms, velocity streamlines, and the average heat transfer. Results based on this numerical study showed that the sinusoidal roughness considerably affect the hydrodynamic and thermal behavior of fluid in a square cavity. Dwesh and Singh [23] studied the combined free convection and surface radiation in tilted open cavity. A numerical investigation has been performed for steady, incompressible, laminar, free convection with surface radiation in a tilted 2D open cavity in which the wall facing the opening is heated by a constant heat flux source. The effects of emissivity, tilt angle and Rayleigh number on the heat transfer in the tilted open cavity are analyzed. The results showed that the heat transfer increased first and then decreased with decreasing the cavity tilt angle for all Rayleigh number and emissivity.

It is seen from the literature that there is a short of the data on natural convection within a rectangular enclosure for the roughness bottom heated wall at varies venting ratio. Therefore, the present work aims to present experimental investigation to the heat transfer characteristics in a rectangular enclosure heated at the roughness bottom wall at varies venting ratio and enclosure tilt angles. The experiments are carried out at tilt angle θ (measured from the vertical axis) that varies from 0º to 90º, at venting ratio from 0.25 to 1.

II. EXPERIMENTAL SET UP

The schematic diagram of the experimental setup is shown in Fig.1. The rectangular enclosure is mounted on a stand and supported by two arms. The arms and the stand are designed to minimize the disturbance to the air flow and to ensure good physical stability. The rectangular enclosure can be tilted about its longitudinal axis. The tilt angle is measured with respect to the vertical axis and can be read from a protractor. The rectangular enclosure has an internal dimensions of W=200 mm, H=160 mm and L=320 mm. Bottom side of the enclosure, where the heat source is made of copper sheet (320 mm x 200 mm) of 1 mm thickness. The cross sectional view of the test section and the distribution of thermocouple along the tested surface are shown in Figs. 2 (a) and (b). The tested surface plate is heated electrically by means of the heater, which consisted of nickel-chromium heating wire wound around a threaded sheet of mica and sandwiched, also between two sheets of mica. Each mica sheet has a 0.5 mm thickness as shown in Fig. 2(c). The rough surface composed of 0.002 m diameter copper rods placed 0.01 m apart along the heated sheet length as shown in Fig. 2(c). The rods are fixed to the surface such that they are in full contact with the surface along the upper heated surface. The test plate is insulated from down surface by three layers of glass wool.

The other walls of the enclosure are made of Plexiglass of 8 mm thick and having emittance of 0.9. All walls of the enclosure, except the upper wall having the slots venting, are thermally insulated with a 5-cm thick glass wool to minimize heat losses from the enclosure.

The electric heater is connected with a DC power supply to control the power input to the heater. The voltage and the current supplied to the heater are measured by a digital voltmeter and an ammeter of accuracy 0.025 percent. The surface temperature distribution of the heated surface is measured using 11 thermocouples (type k) distributed equally spaced along the axes of the heated surface as shown in Fig 2 (b). The thermocouples are fixed on the heat surface using fine tin solders. To estimate the heat losses across the walls of the enclosure, two thermocouples (type k) are fixed across the thickness of each blind wall of the enclosure. The ambient temperature outside the enclosure is measured by a separate thermocouple (type k) placed at the bottom of the plastic box containing the set up. All the thermocouples are calibrated in a constant temperature path and a measurement accuracy of ± 0.2 ºC is obtained.

III. UNCENTIRITINITY ANALYSIS

The uncertainty $\Delta \overline{Nu}$ in the value of $\overline{Nu}$ is estimated based on the procedure of Holman and Gajda [24] and is expressed as follows

$$(\Delta \overline{Nu})^2 = \sum_{i=1}^{i=n} \left(\frac{\partial \overline{Nu}}{\partial x_i} \Delta x_i\right)^2$$

where Δx_i is the uncertainty in the x_i variable. The uncertainty in the various variables used in the determination of the Nusselt number are: 0.23 % for the electric current *I*, 0.2% for the electric volt *V*, 0.2 C for any temperature measurement, 0.001 m for any distance measurement, 0.45% for the thermal conductivity of air, 2% for the thermal conductivity of plexiglass, and 5% for the emittance of the wiring board and the plexiglass. It is found that the uncertainty for the data of $\overline{Nu}$ ranges from 6 percent to 10 percent.

IV. DATA REDUCTION

The average Nusslet number from the rectangular enclosure is defined as:

$$\overline{Nu} = \frac{\bar{h}H}{k_a} \qquad (1)$$

where H is the height of the enclosure and k_a is the thermal conductivity of the air taken at $(T_s + T_\infty)/2$. Where the local Nusselt number is given by the equation:

$$Nu = \frac{q}{A(T_S - T_\infty)} * \frac{H}{K_a} \qquad (2)$$

where A is the area of the board, q is the rate of heat transfer by convection from the heated surface, T_s is the local temperature of the surface and T_∞ is the ambient temperature measured at the bottom of the surface containing the experimental set up. Temperature measurements showed that the variation of the temperature of the heated surface was very negligible along the x-axis and varies slightly along the y-axis in a manner depending on the top venting arrangement and the tilt angle. Therefore, in calculating the average heat transfer coefficient the heated surface was divided into 7-equal sections along the y-axis as shown in Fig. 2. The average heat transfer coefficient can be expressed as:

$$\bar{h} = \frac{q}{6A}\sum_{i=1}^{i=6}\frac{1}{(T_{s,i} - T_\infty)} \qquad (3)$$

To calculate q, an average energy balance for the heated plate gives:

$$IV = q + q_c + q_r \qquad (4)$$

Where I and V are the electric current and the voltage input to the electric surface, q is the heat transfer by free convection from the heated surface through the venting of the enclosure, q_c is the heat losses by conduction through the walls of the enclosure and q_r is the heat transfer by radiation from the electric surface to the surroundings as seen through the venting of the enclosure. The conduction heat losses through the walls of the enclosure are the sum of the heat losses through the bottom heated wall of the enclosure and the heat losses through the other blind walls of the enclosure. This is expressed as:

$$q_c = \sum k_w \; A_j \Delta T_j / t \qquad (5)$$

where j is the wall identification number, k_w is the thermal conductivity of the wall of the enclosure, A is the area of a side wall of the enclosure, t is the thickness of the walls of the enclosure, and ΔT_j is the temperature difference between the inner and outer surfaces of the j the wall, respectively.

The radiation was incorporated in the losses based on the radiosity /irradiation formulation. The six interior surfaces of the enclosure are assumed to be opaque, diffuse, isothermal and gray. The surfaces containing the venting were assumed to be black at the ambient temperature. The radiation heat loss q_r from the cavity is the net rate at which radiation leaves the surfaces containing the venting (q_i) and is given by

$$q_r = q_i = \varepsilon_i A_i(\sigma T_i^4 - G_i) \qquad (6)$$

where the irradiation G_i is given by

$$G_i = \sum_{j=1}^{N} F_{ij} J_j \qquad (7)$$

where F_{ij} is the view factor and J is the radiosity given by

$$J_i = \varepsilon_i \sigma T_i^4 + (1 - \varepsilon_i)\sum_{j=1}^{j=N} F_{ij} J_j \qquad (8)$$

The view factors F_{ij} between parallel and perpendicular surfaces are calculated based on the graphs and expressions given by Incropera and De Witt [25] and Suryanarayana [26]. For each venting arrangement Eqns. 6-8 are solved together to find the radiation heat losses in terms of the surfaces temperatures of the enclosure.

The conduction heat losses through the enclosure walls and the radiation heat losses are within 11% and 5 % of the input heat, respectively.
The dimensionless Rayleigh number (*Ra*) is calculated from the measured quantities using the definition:

$$Ra = \frac{\beta g H^4 q''}{k_a \nu\alpha} \qquad (9)$$

Where q″ is the input heat flux, *H* is the height of the enclosure, g is the acceleration of gravity and β , υ , α , k_a are the coefficient of volume expansion, kinematics viscosity, thermal diffusivity, and thermal conductivity of the air, respectively. All the properties of the air in the last equation are taken at $(T_s + T_\infty)/2$.

V. RESULTS AND DISCUSSION

Figures 3 to 6 show, the effect of the tilt angle (θ) on the local Nusselt number (*Nu*) vs x/L at varies of Rayleigh number (*Ra*) at constant venting ratio (*VR* = 0.25) from roughened heated surface. All the figures give approximately the same general shape. The *Nu* increase with the increase of *Ra* and there is symmetry of data around the center of roughened heated surface. The *Nu* increases with the increase of *Ra* because of the increase of the buoyancy force acting on the air with the increase of the *Ra*. Increasing the buoyancy force increases flow driving force and consequently causes an increase of flow intensity and enhances the mixing within the air layer which leads to higher heat transfer rate. The *Nu* number at the ends of roughened heated surface is higher than that of the middle. This can be attributed to the ends effects and the natural circulation of air due to buoyancy force which make a hotter zone near the center of roughened heated surface more than that at the ends. Also, the same results and the same general shape are shown in Figs from 7 to 18 which show the variation of local *Nu* with X/L at different of each Ra, tilt angle (θ) and venting ratio (*VR*).
Figure 19 and 20 show the effect of changing θ and *VR* on the average Nusselt number ($\overline{Nu}$) at Ra = 3.7×10^8. The $\overline{Nu}$ increases with increase of *VR* but it decreases with the increase of θ. The increase of *VR* leads to increase of circulation of quantity of air due to buoyancy effect and causes more cooling to the heated surface. When the *VR* increases, the resistance to the circulation motion of the air in the enclosure (i.e. the resistance to the convection currents to escape from the enclosure) decreases, and this leads to faster replacement of the hot air by cold air. On the other hand increase of θ makes the top venting face changed from vertical to incline and then to horizontal which leads to decrease the quantity of circulating air due to buoyancy effect which causes a less cooling of heated surface.

VI. COMPARISON WITH LITRATURE

To the author's knowledge, there are no previous studies on natural convection heat transfer in a tilted rectangular enclosure heated at the bottom rough surfaces wall and vented by uniform slots venting at top wall. The only available studies that can be compared with the present work are those established by Nada and Moawed [27] for a similar enclosure with a smooth surface as shown in Fig 21. As can be seen from this figure, there is a fair agreement between present $\overline{Nu}$ of smooth surface results in the rectangular enclosure with that of smooth surface of Ref. [27] but the present results of roughness surface shows higher values for $\overline{Nu}$ than that of smooth surface in the rectangular enclosure. The increases in $\overline{Nu}$ for the rectangular enclosure with roughness surface over the smooth surface ranges from 12% to 21% depending on Ra. Highest values for the heat transfer for the rough surface in rectangular enclosure is believed to be caused by the presence of roughness. Roughness surface effects via vortices generation and flow instability are not present in the smooth surface in rectangular enclosure. Roughness helps to trip the boundary layer and adds more turbulence to the flow resulting in increase in heat transfer.

VII. EMPIRICAL CORRELATIONS

Empirical correlations are developed to fit the experimental data for top venting arrangements. The expression for the correlation is as follows:

$$\overline{Nu} = 16.13\,(Ra/10^8)^{0.64}\,(VR)^{0.103}\,(1+\cos\theta)^{0.0114} \qquad (10)$$

$$2\times10^8 \le Ra \le 1.52\times10^9\ ;\ 0^\circ \le \theta \le 90^\circ \text{ and } 0.25 \le VR \le 1$$

The calculated average Nusselt number ($\overline{Nu}_{cal}$) from Eq. 10 is plotted against experimental average Nusselt number ($\overline{Nu}_{exp}$) as shown in Fig. 22. As shown from this figure the maximum deviation between the experimental data and the correlation is 10 %.

VIII. FIGURES

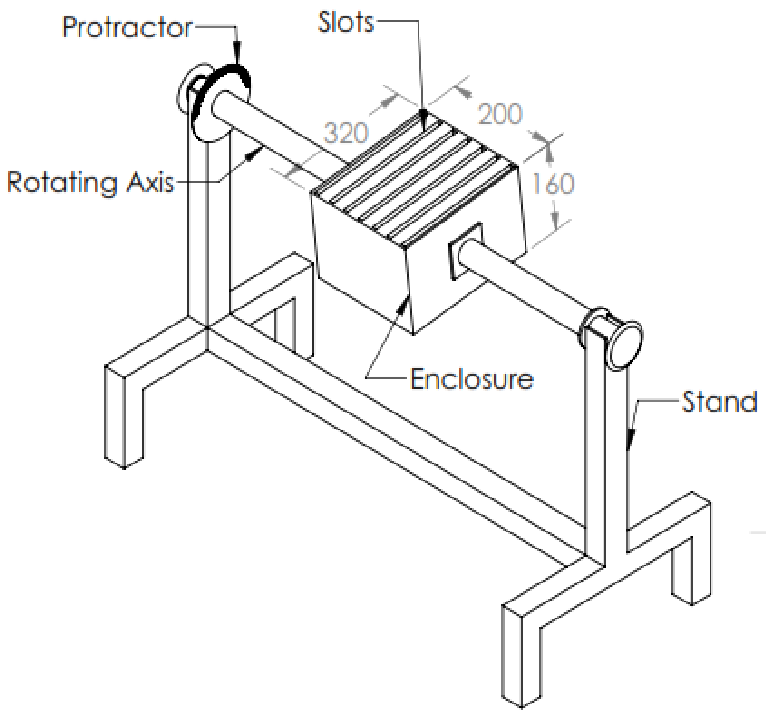

Fig.1. Schematic diagram of the experimental set up.

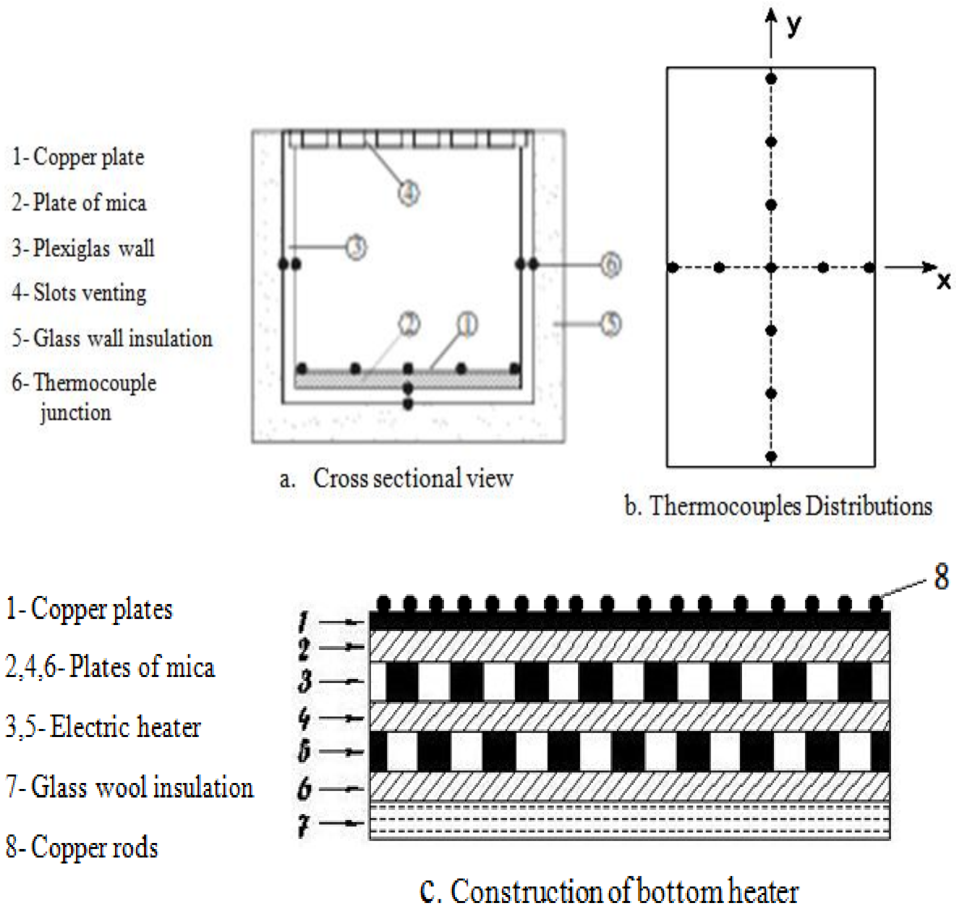

Fig. 2 Test section arrangement

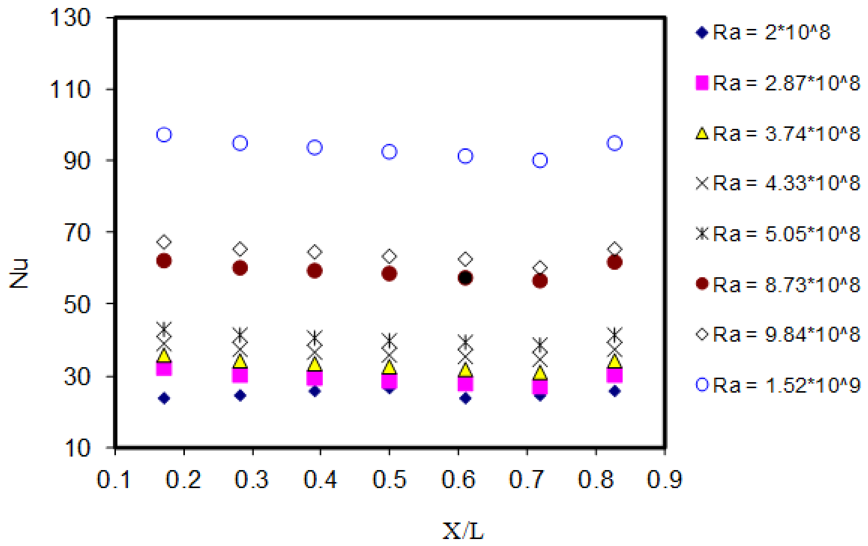

Fig.3. Variation of Nusslet number with X/L at VR = 0.25 and θ = 0

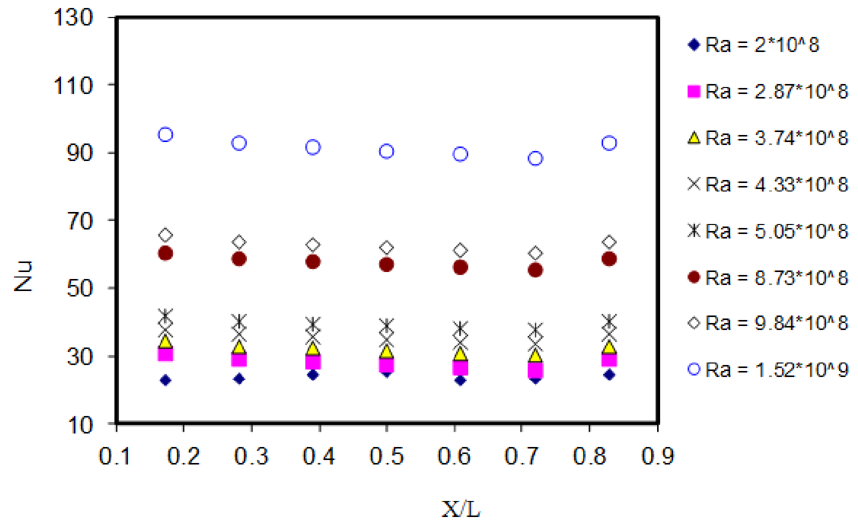

Fig.4. Variation of Nusslet number with X/L at VR = 0.25 and θ = 30

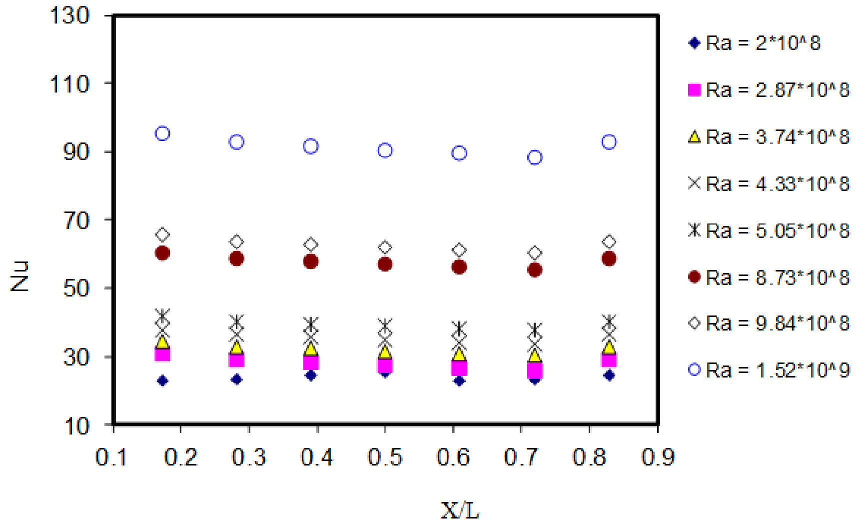

Fig.5. Variation of Nusslet number with X/L at VR = 0.25 and θ = 60

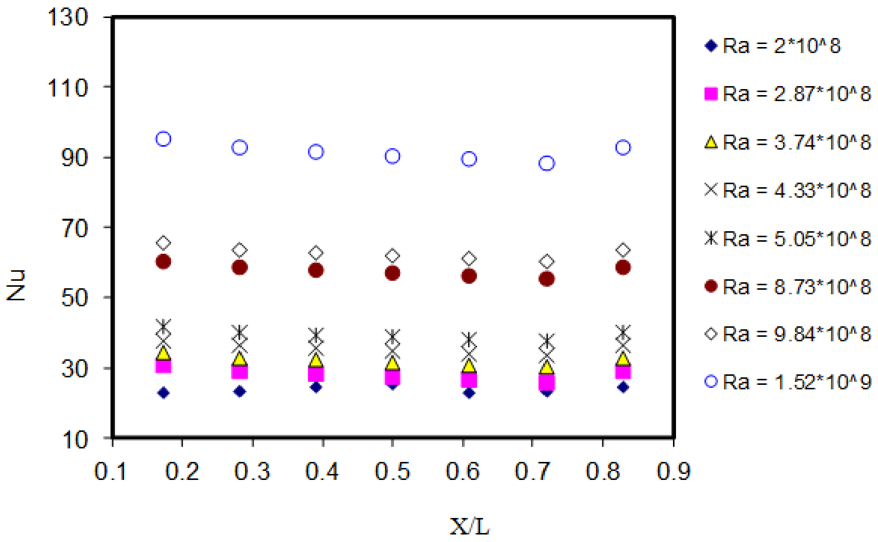

Fig.6. Variation of Nusslet number with X/L VR = 0.25 and θ = 90

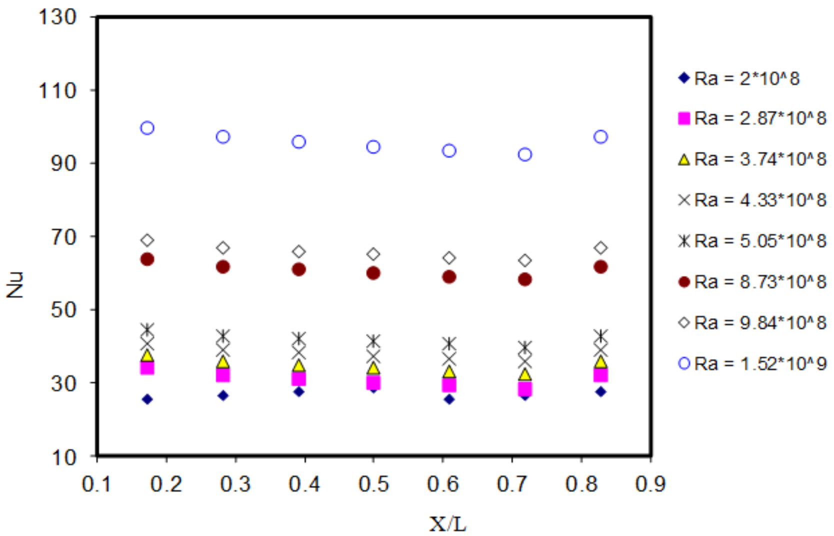

Fig.7. Variation of Nusslet number with X/L at VR= 0.5 and θ = 0

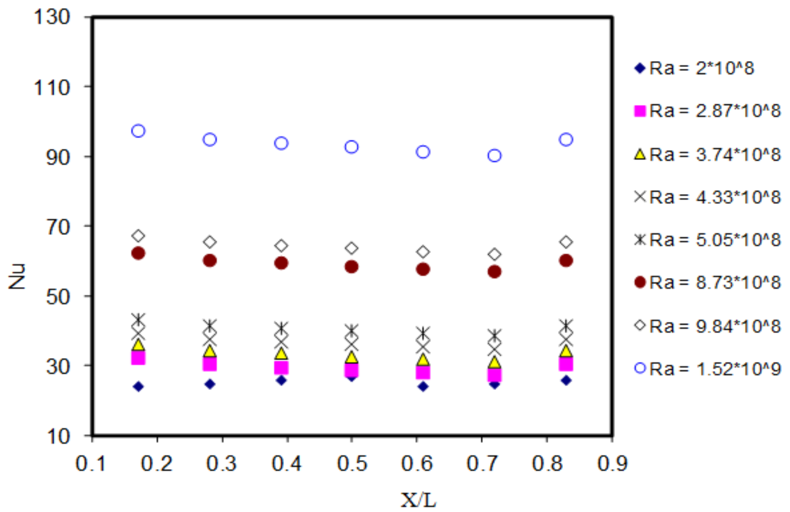

Fig.8. Variation of Nusslet number with X/L at VR = 0.5 and θ = 30

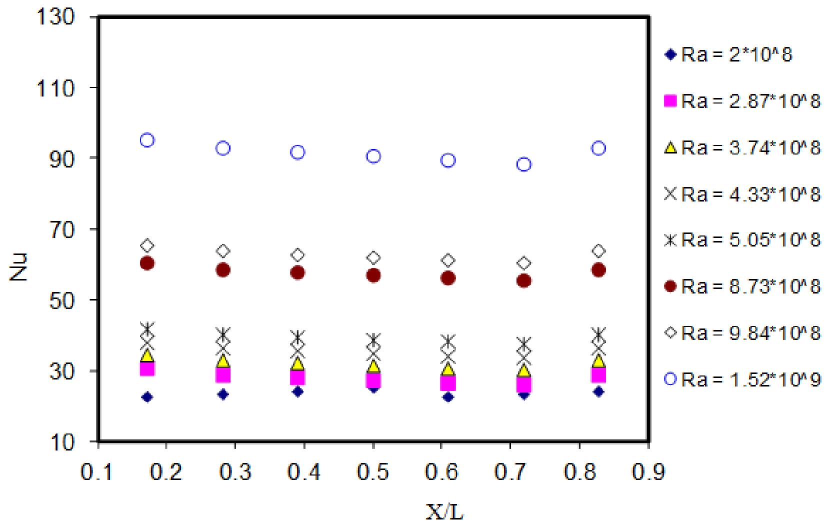

Fig.9. Variation of Nusslet number with X/L VR = 0.5 and θ = 60

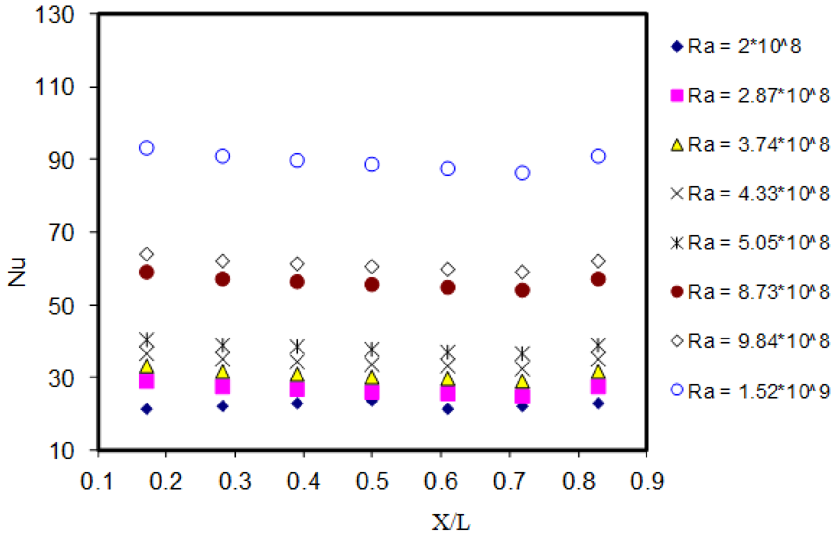

Fig.10. Variation of Nusslet number with X/L at OR = 0.5 and θ = 30

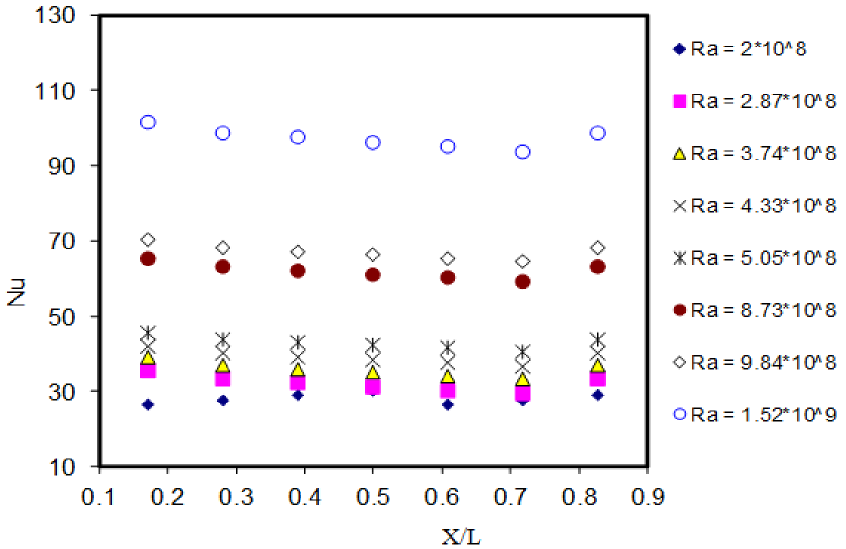

Fig.11. Variation of Nusslet number with X/L at VR=0.75 and θ = 0

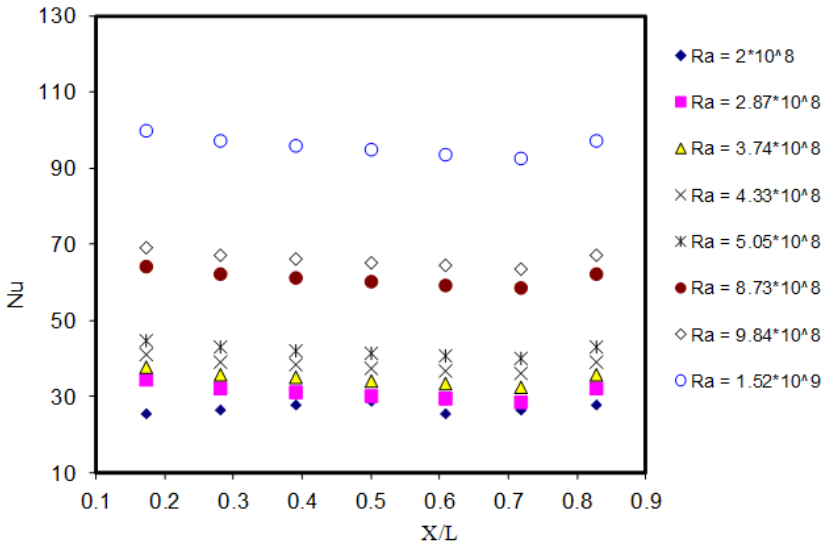

Fig.12. Variation of Nusslet number with X/L at VR = 0.75 and θ =30

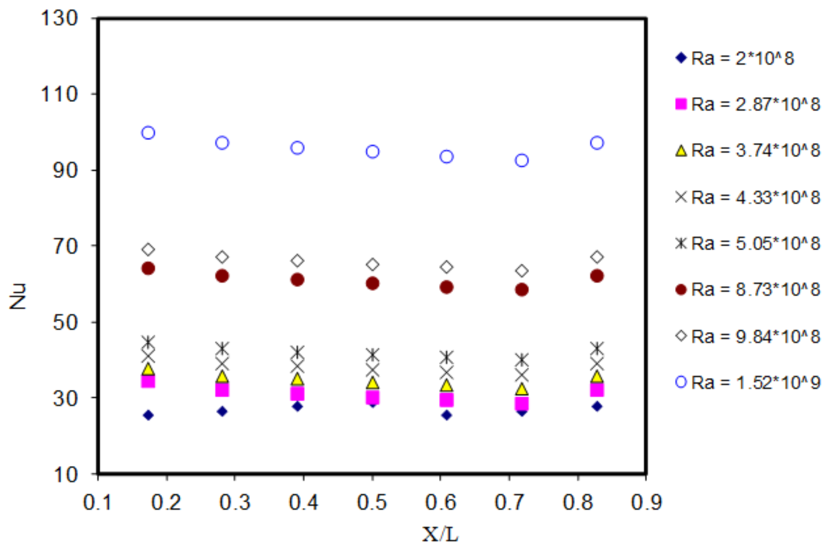

Fig.13. Variation of Nusslet number with X/L at VR=0.75 and θ = 60

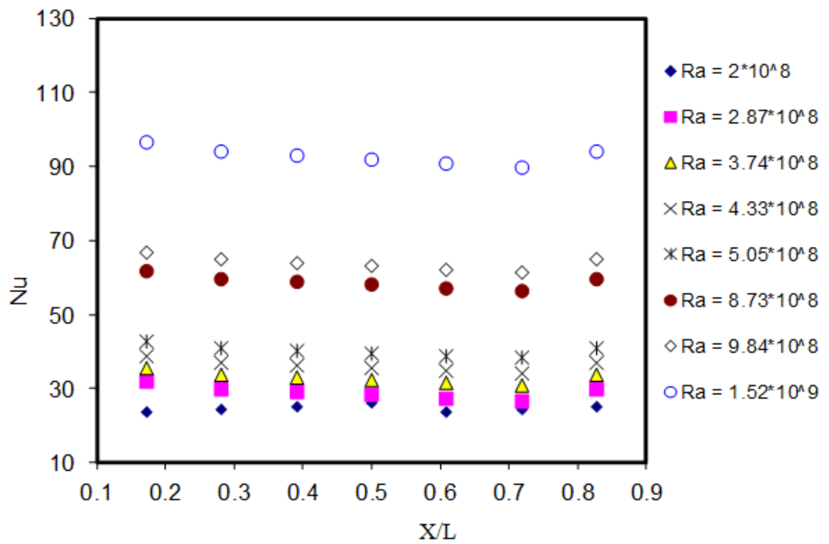

Fig.14. Variation of Nusslet number with X/L at VR= 0.75 and θ = 90

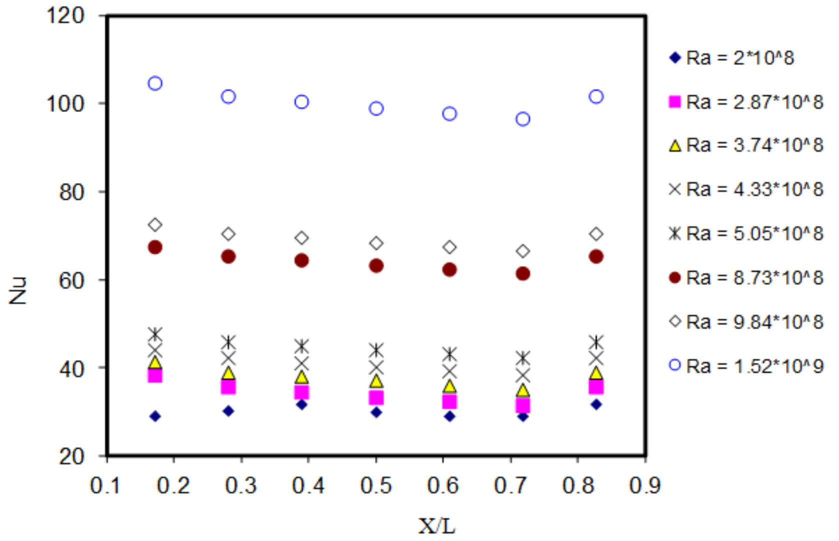

Fig.15. Variation of Nusslet number with X/L at VR= 1 and θ = 0

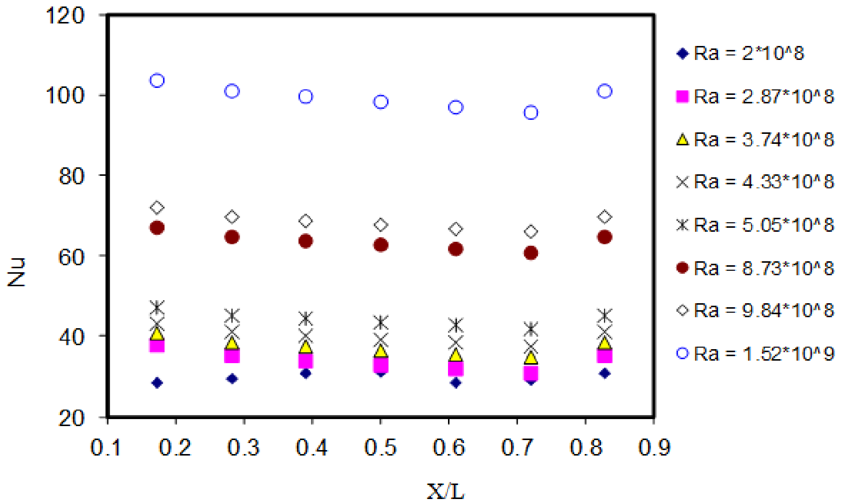

Fig.16. Variation of Nusslet number with X/L at VR = 1 and θ = 30

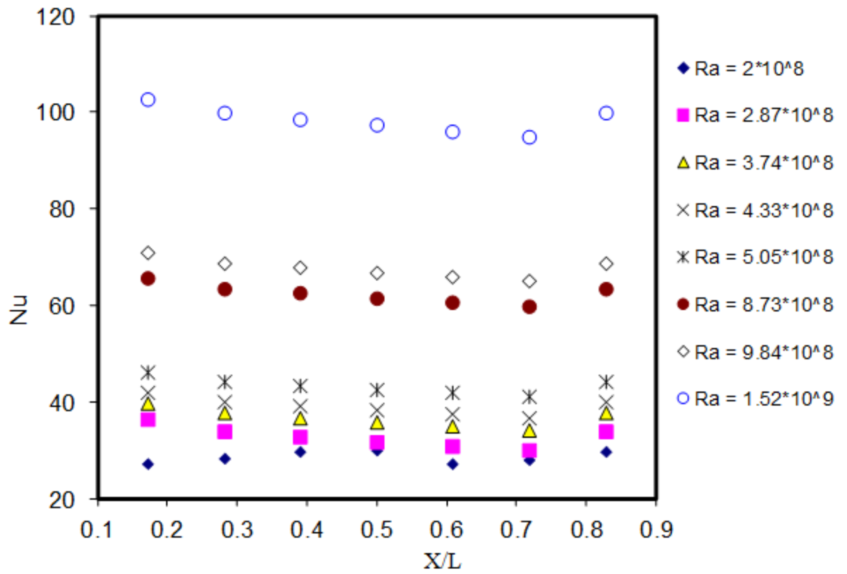

Fig.17. Variation of Nusslet number with X/L and VR = 1 and θ = 60

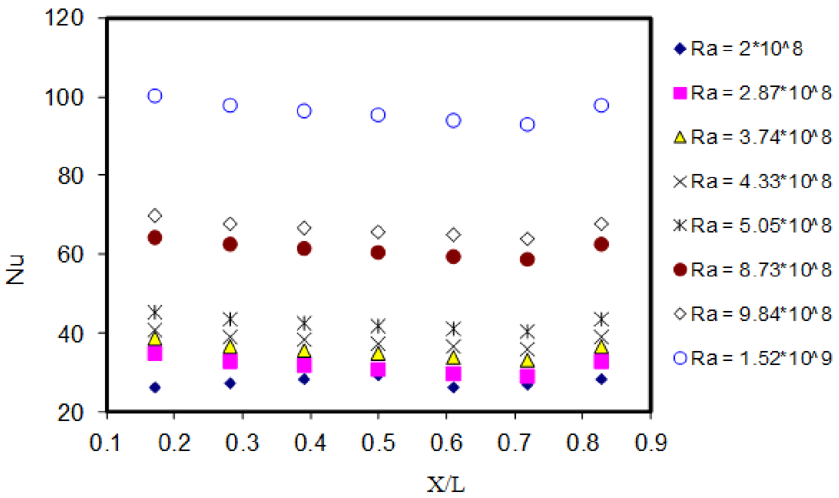

Fig.18. Variation of Nusslet number with X/L and VR = 1 and θ = 90

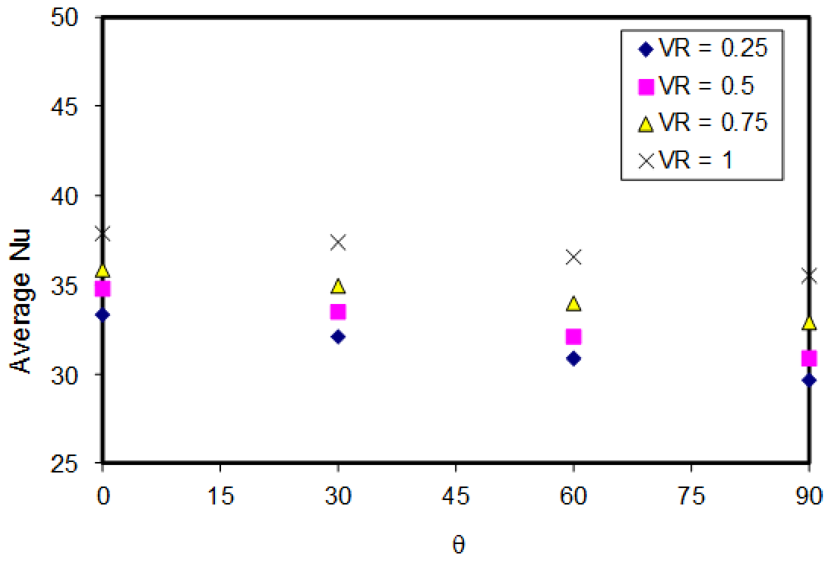

Fig.19. Variation of $\overline{Nu}$ with θ at Ra = 3.7×10^8 and different VR

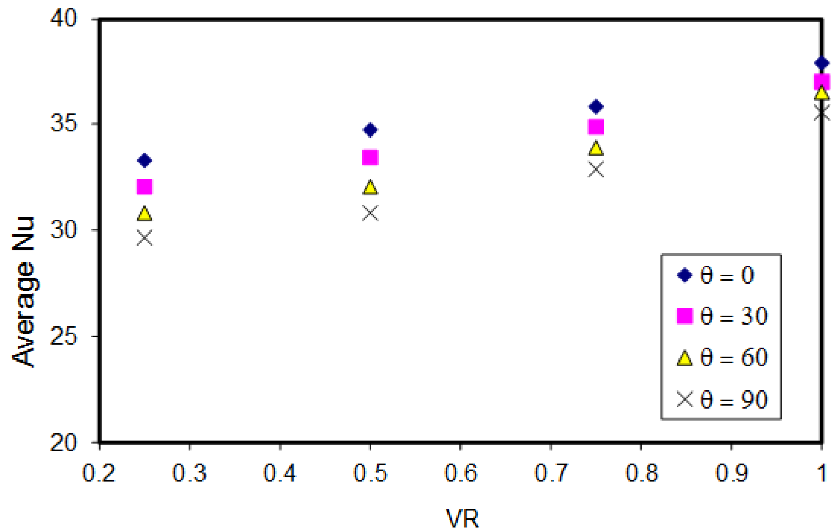

Fig.20. Variation of $\overline{Nu}$ with VR at different θ and Ra = 3.7×10^8.

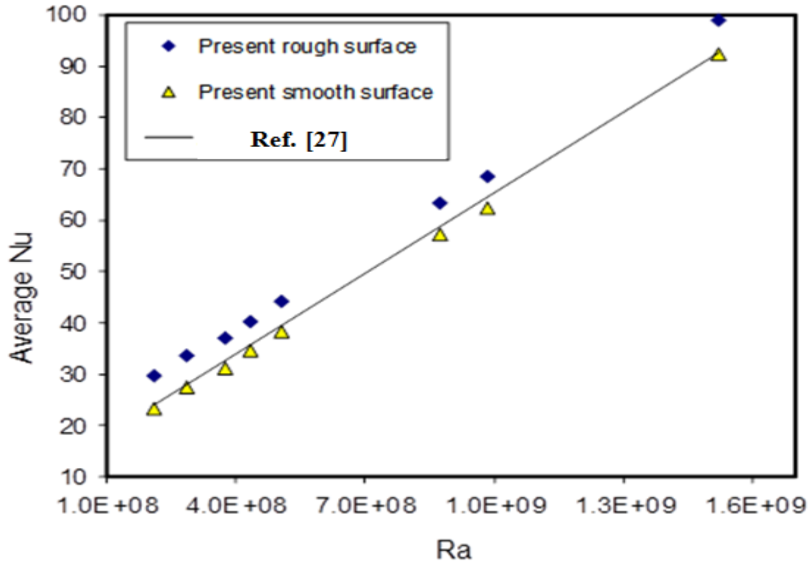

Fig.21. Comparison between present experimental data and Ref.[26] at θ = 0º and VR=1

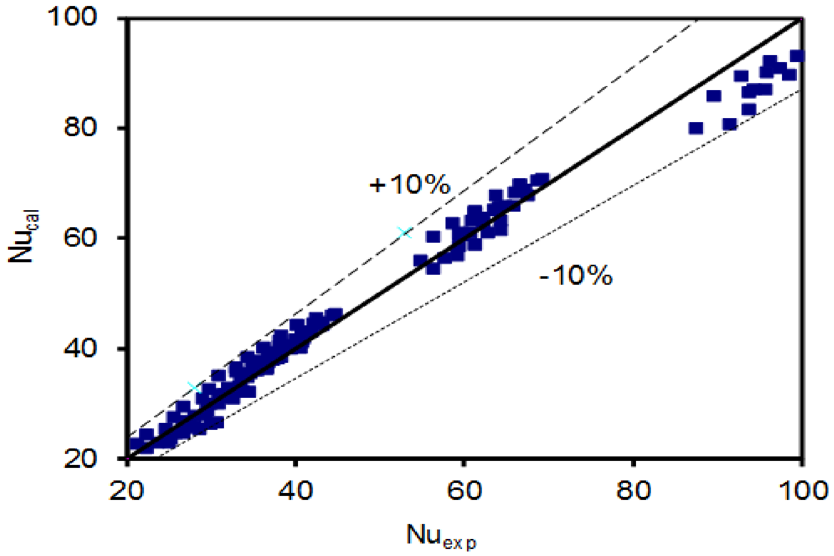

Fig.22. $\overline{Nu}_{cal}$ from Eq. 10 against $\overline{Nu}_{exp}$

IX. CONCLOUSION

The study of natural convection heat transfer from tilted rectangular enclosure heated at the bottom rough surfaces wall and vented by uniform slots opening at top wall is investigated experimentally. Rough surfaces of roughness 0.002m are used to study their effect on the heat transfer characteristics. The effects of venting ratio, enclosure's tilt angle and Rayleigh number on the cooling of rough surface inside the enclosure are investigated. The experiments are carried out at a Rayleigh number ranging from 2×10^8 to 1.52×10^9 for enclosure tilt angles ranging from 0º to 90 and venting ratios of 1, 0.75, 0.5 and 0.25. The results showed that:

1- Local *Nu* increase with the increase *Ra* and *VR* but it decreases with the increase of tilting angle.
2- Roughness shows a large effect on heat transfer for the rectangular enclosure where the average Nusselt number increases with the increase of venting ratio and decrease enclosure's tilt angle at the same Rayleigh number.
3- The results are compared with a smooth rectangular enclosure of the same surface area to study the effect of roughness on heat transfer.
4- The increase of Nusselt number of rough surface in rectangular enclosure over smooth surface in ranges from 12% to 21% depending on Ra.
5- Correlations are developed to predict the average Nusselt number of the enclosure in terms of the Rayleigh number, venting ratio and enclosure tilt angle.

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