THE EFFECT OF SPACE VOLUME ON FREE CONVECTION HEAT TRANSFER FOR ONGITUDINAL FINNED CYLINDER WITH DIFFERENT SLOPE NGLES

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ABSTRACT:

This study deals with free convection heat transfer for the outer surface of two cylinders of the shape of (Triangular & Rectangular fined cylinders with 8-fins), putted into two different spaces; small one with dimension of (Length=1.2m, height=1m, width=0.9m) and large one with dimension of (Length=3.6m, height =3m, width=2.7m). The experimental work was conducted with air as a heat transport medium. These cylinders were fixed at different slope angles (0^o, 30^o, 60^o and 90^o) .The heat fluxes were (279, 1012, 1958, 3005, 4419) W/m^2, where heat transferred by convection and radiation. In large space, the results show that the heat transfer from the triangular finned cylinder is maximum at a slope angle equals to 90^o and is minimum at the slope of 0^o angle whit the heat transferred from the triangular finned cylinder is maximum at slop angle of 30^o and minimum at the slop of angle 60^o in the small space with following range of the Raleigh number ($1.68*10^7$-$3.46*10^8$).

الخلاصة:-

يتناول البحث دراسة عملية لانتقال الحرارة بالحمل الحر من أسطوانتين ذات زعانف طولية مستطيلة المقطع ومثلثة المقطع بعدد(٨ زعنفة) وذلك باستخدام الهواء كوسط ناقل للحرارة موضوعتين داخل حيزين مختلفين في الحجم أحداهما كبيربأبعاد (3m*3.6m*2.7m) والاخر صغير الحجم بأبعاد (1m*1.2m*0.9m)، ولمعرفة الخواص الحرارية تم وضع النموذجين اعلاة بزوايا ميل (0^o,30^o,60^o,90^o) عن الأفق ولمستويات فيض حراري(4419, 3005, 1958, 1012, 279)W/m^2 في حيز مفتوح إلى الهواء الخارجي حيث تنتقل الحرارة بالحمل والإشعاع. وقد وجد أن أعلى مقدار للحرارة المنتقلة بالحمل من الاسطوانة ذات الزعانف المثلثة المقطع في الحيز الكبيروبزاوية ميل مقدارها (90^o) ولكلا النموذجين،أما بالنسبة للحيز الصغير فقد وجد أن أعلى مقدار للحرارة المنتقلة بالحمل يكون من الاسطوانة ذات الزعانف المثلثة المقطع وبزاوية ميل قدرها(30^o)ولكلا النموذجين ولمدى عدد رايلي ($1.68*10^7 – 3.46*10^8$) .

Keyword: - Free Convection Heat Transfer , Longitudinal Finned Cylinder, Different Slope angle.

INTRODUCTION: -

The longitudinal fins with variable cross-sectional area are used in electronic and electrical devices and heat exchangers and in most applications of air-conditioning (condensers, and air heaters). The longitudinal fins have different cross-sectional areas like circular, triangle or other areas. Hence, cost factor should be considered especially if the product used commercially. It can be achieved by decreasing the heat exchanger size and weight by increasing its efficiency to maximize heat exchange.

There are many factors that increase the heat exchanger efficiency:-

- Decrease the size of fins by increase the surface area.
- Using high conductivity metals like Aluminum.
- Using forced convection heat transfer or counter current flow.
- Changing heat exchanger geometrical parameters.

Many researchers studied the free convection heat transfer and concluded empirical and numerical relations between Nusselt number and Raleigh number.

(Atipoang et al, 2008) presented an experimental study for the effect of slope angle on finned tube heat exchanger performance in free convection. The number of rows were with in the range of (1-4), with an inlet water temperature of (40, 60, and 80 ^{o}C), volumetric flow rate of (1.5 L/min), ambient temperature is (27^{o}C) and slope angle with in the rang of (0^{o} – 90^{o}). They found that the best performance can be achieved for slope angle between (30^{o}-45^{o}) were heat exchanger efficiency decreased with increasing the number of rows.

***(Lkuo* 1983)** presented experimental studies for two cylinders of variable diameter at constant wall temperature. The Grashoff number (4*10^4 – 4*10^5) depended on outside diameter of cylinder with different geometry ratios as (s/d=2-∞) and (x/d=2 – 10). The optimum ratios were (x/d=3) when (s/d= ∞), besides ($\frac{Nu_i}{Nu_o}$) equals one for all furthermore; an obvious enhancement obtained (x/d>3). In case of walls, the best geometrical ratio is (x/d=7).

The effect of the air space volume was studied upon the affects the free convection heat transfer coefficient. Also it was required to fined the optimum position of the fin and the appropriate space volume due to wide the application of fin tube in compact power plant, accurate electronic devices and heat exchangers design.

***(Sparrow* and Chrysler 1981)** studied the free convection heat transfer from a short cylinder conducted to a flat plate in constant heat flux condition and Raleigh number in the range of (1.4*10^4 – 1.4*10^5). Three positions were investigated. They developed the following equation for Nusselt number for all cases:-

$$Nu = CRa^{m} \qquad (1)$$

(Yassen 1978) presented an experimental study on inclined flat cylinder. He used two cylinders of (38mm, 47mm) diameter. The slope angle was in the range of (0^{o} – 90^{o}). Constant heat flux was considered and Raleigh number was between (0.28*10^6 – 3.44*10^6). He found that:-

$$Nu = \left[0.665 - 0.4885(\sin\theta)\right] Ra^{[1/4+(1/(12\sin\theta))]1.74} \qquad (2)$$

He also found that the vertical position is better and heat transfer coefficient value increased to a certain value and then decreased.

(Young et al, 2005) presented a theoretical study for heat transfer for rectangular cross-section fin using one dimension analytical method and finite difference method. A comparison between two methods was done. This comparison deals with heat loss and temperature which was represented as equations interims of Biot number and non-dimensional length of fin. They found that finite difference gave better results with low error ratio by increasing number of nods.

TEST MODELS:-

There are two models used in the experimental work which were designed in standard same used for heaters. These models were made from hard aluminum (D_o=48mm, D_i=16mm), where the heater was positioned inside. Figure (1) shows triangular and rectangular longitudinal fins of dimensions (height =13mm, Length =300mm, number of fins=8). Electrical heater of 1kW was used the models were mounted on moving frame in order to obtain a range of angles (0^o – 90^o). The test model was positioned inside two spaces (inside thermally insulated wood) (Length=1.2m, height=1m, width=0.9m) and, then in room with dimensions (Length=3m, height=3.6m, width=2.7m).

CALCULATIONS:-

General equation ($Nu = CRa^m$) to calculate the Nusselt number and Rayleigh number as follows:-

$$\left.\begin{aligned} &Nu = h * D_o / k \\ &and \\ &Ra = \frac{\beta * g * D_o * (T_s - T_a)}{\nu^2} * \Pr \end{aligned}\right\} \tag{3}$$

The equations are used to calculate the heat generated and transferred by radiation and free convection

$$\begin{aligned} Q_g &= I * V \\ Q_g &= Q_{conv} + Q_{rad} \\ Q_{conv} &= h * A_t * (T_s - T_a) \\ Q_{rad} &= \sigma * A_t * \varepsilon * (T_s^4 - T_a^4) \end{aligned} \tag{4}$$

and the film temperature and volumetric expansion coefficient are also calculated as follows:-

$$T_f = \frac{T_a + T_s}{2} \tag{5}$$

$$\beta = \frac{1}{T_f} \tag{6}$$

RESULTS AND DISCUTION:-

Rayleigh and Nusselt numbers were calculated for ($Ra=1.68*10^7$-$3.46*10^8$), variable heat flux (279, 4419 W/m^2). And slope angle (0^o, 90^o). (Fig. 2, 3, 4, and 5) respectively, the result shows that:-

1- (Fig. 2 , 3) show that the free convection heat transfer increased in large space with increasing, Rayleigh number and reach maximum value for slope angle of 90^o and reach minimum value at $\theta = 0^o$. That is due to the longitudinal rectangular cross-section of the fin allowing the hot air with low density to flow upward uniformly. Moreover, the maximum lifting force for air is achieved, proportional with sin (θ) (upward effect), in the case of slope angle ($\theta = 30^o$ and 60^o). For the case of $\theta = 0^o$, the fins work as obstacles to block air flow which causes retardation of flow to be staffed. In addition, it adds thermal resistance on so, the loss from fin decreases. The heat lost from the test model of triangular fins is higher than rectangular fin, because of the large area of triangular cross – section.

2- (Fig. 4 , 5) show that the free convection heat transfer is increased in the small space by increasing the Rayleigh number. In cases the maximum value occurs when the slope angle equals to $\theta = 30^o$ and minimum value when $\theta = 60^o$.

For the case of test model inclination, the lifting force of air is proportional to sin(θ) and the normal component of buoyancy will be the effective force and reach the maximum value. In case of $\theta = 90^o$, the longitudinal rectangular and triangular fins of the test model in vertical position allow the hot air to flow uniformly (maximum lifting force at $\theta = 90^o$).

Due to this position in the small space, the distance between the fin and the upper surface of the space will be minimum at ($\theta = 90^o$) and will equal to 25 cm. In this case, it causes an increase in the ambient temperature because of the thermal warming inside the small space. This will lead to lower temperature difference and heat transfer.

3- A comparison for free convection heat transfer of large space and small space are shown in (fig. 6, 7, 8, 9, 11, 12, 13, and 14) for different slope angles (0^o, 30^o, 60^o and 90^o) for both models. It was found that the free convection heat transfer will be much more than large space as compared with small space for different slope angles. That is due to the increase in the ambient air temperature which results from thermal warming in case of small space. As a result, the lower temperature difference and heat transfer will decrease, and vice versa for large space, large air draughts and higher heat transferred.

4-The relationship between Rayleigh and Nusselt numbers represented logarithmically and gave acceptable results according to the boundary conditions of each model.

Test model Triangular for small space

$\theta = 0^o \quad y = 0.03902 + 0.2008 \quad x$

$\theta = 30^o \quad y = 0.377 + 0.165 \quad x$

$\theta = 60^o \quad y = -1.024 + 0.282 \quad x$

$\theta = 90^o \quad y = -0.3512 + 0.214 \quad x$

Test model Rectangular for small space

$\theta = 0^o \quad y = 0.24097 + 0.1894 \quad x$

$\theta = 30^o \quad y = -0.21475 + 0.2601 \quad x$

$\theta = 60^o \quad y = -0.3075 + 0.249 \quad x$

$\theta = 90^o \quad y = 0.01697 + 0.22315 \quad x$

Test model Triangular in large space

$\theta = 0^o \quad y = 1.40399 \quad x - 2.25759$

$\theta = 30^o \quad y = 1.8083 \quad x - 2.40391$

$\theta = 60^o \quad y = 1.49397 \quad x - 2.42405$

$\theta = 90^o \quad y = 1.50816 \quad x - 2.44565$

Test model Rectangular in large space

$\theta = 0^o \quad y = 1.76686 \quad x - 3.03874$

$\theta = 30^o \quad y = 1.76417 \quad x - 3.0181$

$\theta = 60^o \quad y = 1.67471 \quad x - 2.82292$

$\theta = 90^o \quad y = 1.64845 \quad x - 2.75683$

Where:-

$$y = LogNu \quad , \quad x = LogRa$$

The deviation of this curved were treated by plotting the constant value C with their angles as shown in figs (10 and 15) where:-

$$C = m\,\theta^{n} \qquad (٧)$$

The general equation for test model after substituting the value of constant C is ($1^{o} \le \theta \le 90^{o}$)

a- Test model for rectangular cross – section small space:-

$$Nu = 1.828195 * \mathrm{Ra}^{0.230413} * \theta^{-0.1195} \qquad (8)$$

b- Test model for triangular cross – section small space:-

$$Nu = 1.491094 * \mathrm{Ra}^{0.144825} * \theta^{-0.133} \qquad (9)$$

C-Test model for rectangular cross – section in large space:-

$$Nu = 0.1477245 * \mathrm{Ra}^{1.713} * \theta^{0.46923} \qquad (10)$$

d- Test model for triangular cross – section in large space:-

$$Nu = 0.380609 * \mathrm{Ra}^{1.471} * \theta^{0.59543} \qquad (11)$$

The angle effect is included in the Nusselt number equation above. In addition, substituting the value of angle (θ), the error ratio does not exceeds 10%.

CONCLUSIONS:-

These results show that the free convection heat transfer coefficient of triangular cross – section is higher than the one with rectangular cross – section (for the same number of fins). The free convection heat transfer will be maximum at ($\theta=90^{o}$) and minimum at ($\theta=0^{o}$) in large space. In addition, the free convection heat transfer will be maximum at ($\theta=30^{o}$) and minimum at ($\theta=60^{o}$) in small space.

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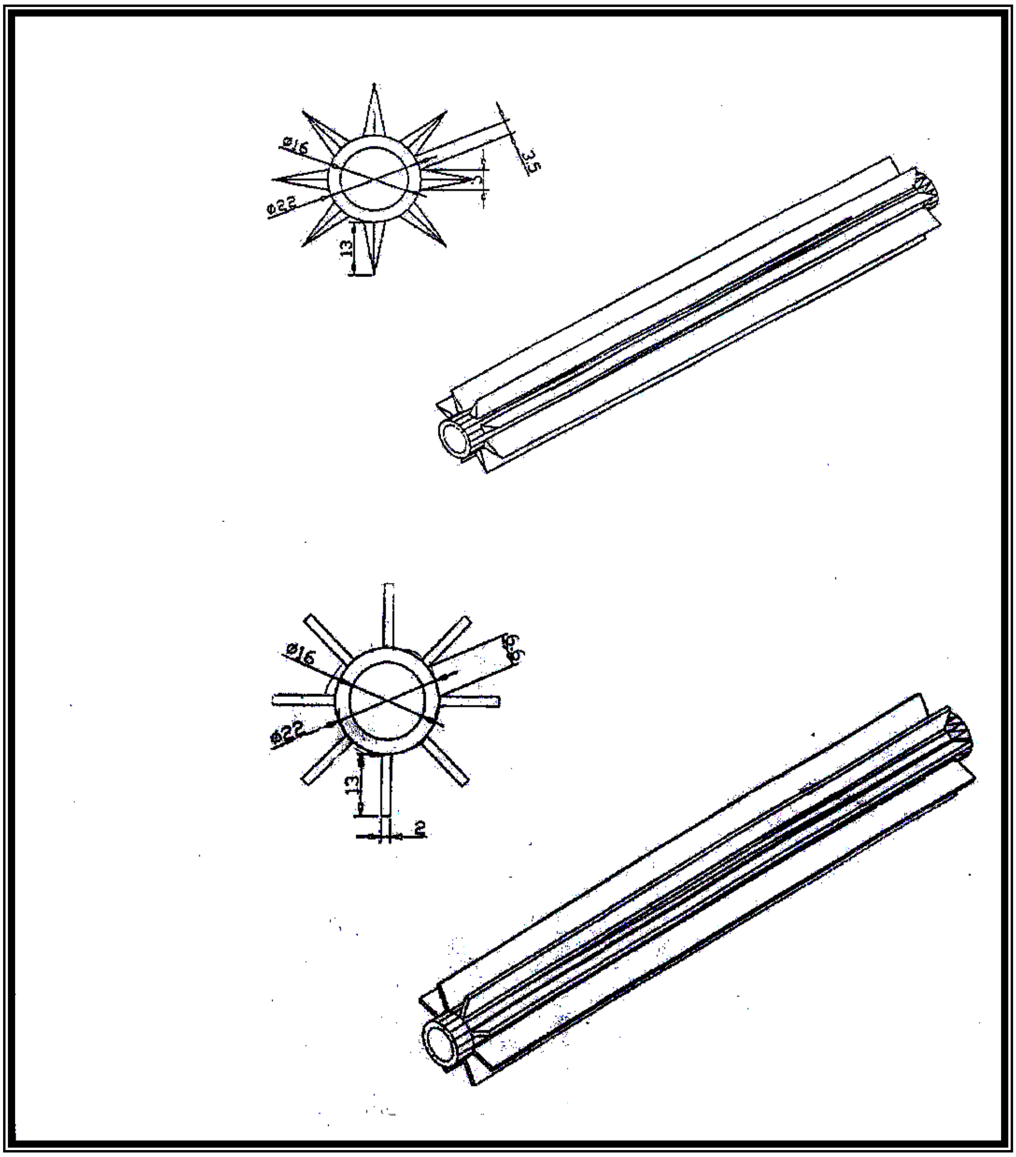

Fig (1) Isometric testing rig.

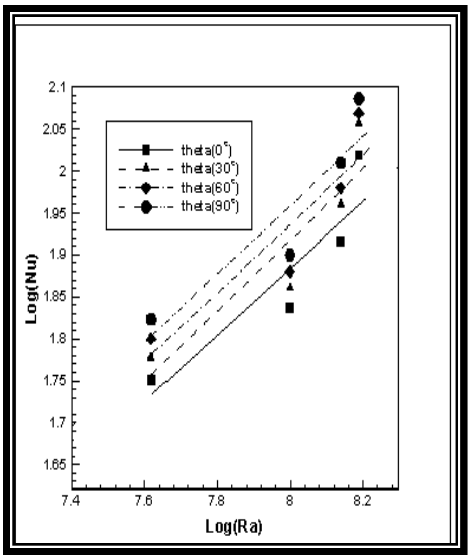

Fig (3) Relationships between Nu and Ra for longitudinally finned cylinder with different slope angles for rectangular cross – section in large space.

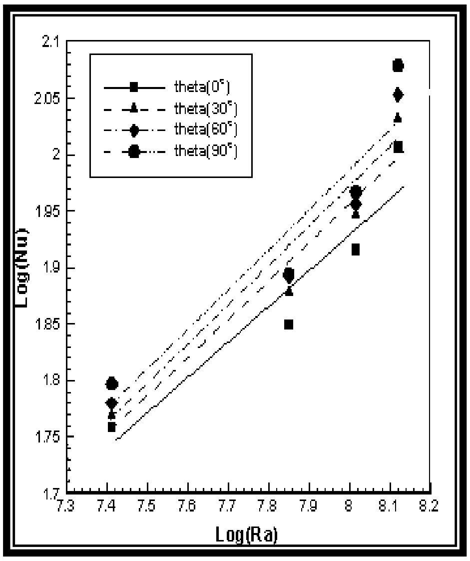

Fig (2) Relationship between Nu and Ra for longitudinal finned cylinder with different slope angles for triangular cross – section in large space.

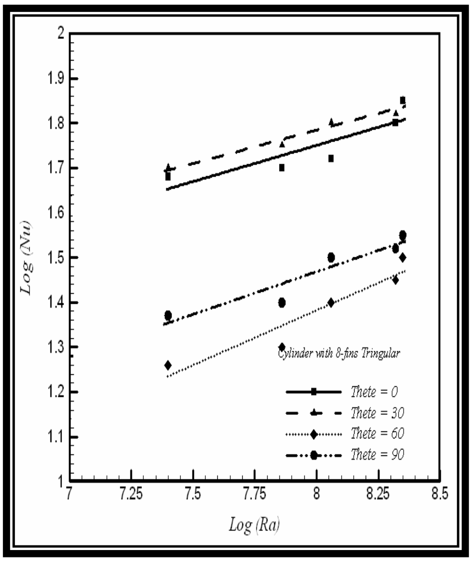

Fig (5) Relationship between Nu and Ra for longitudinally finned cylinder with different slope angles for rectangular cross – section in small. space

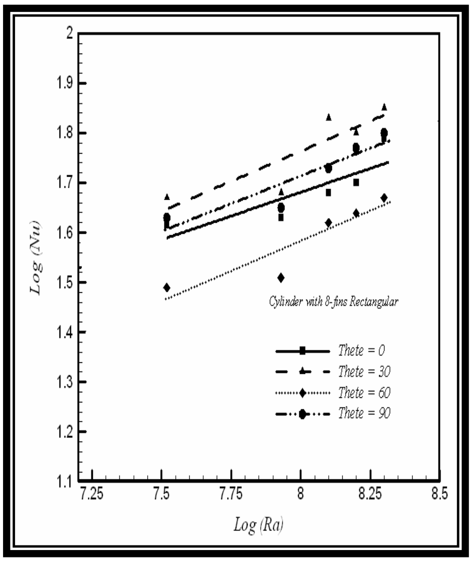

Fig (4) Relationship between Nu and Ra for longitudinally finned cylinder with different slope angles for triangular cross – section in small space.

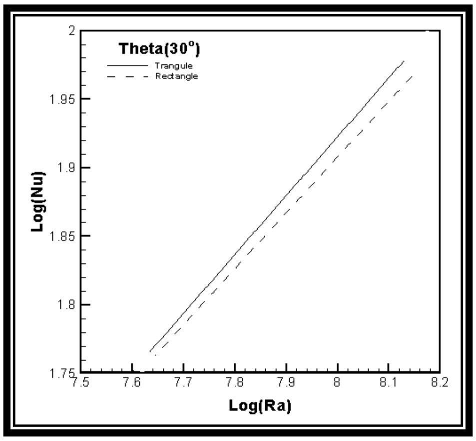

Fig (7) Relationship between Nu and Ra for test section at angle of 30^o in large space.

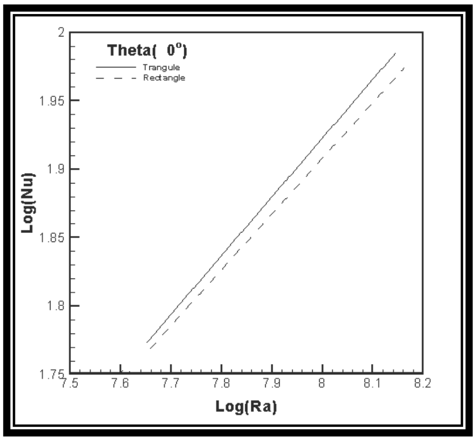

Fig (6) Relationship between Nu and Ra for test section at angle of 0^o in large space.

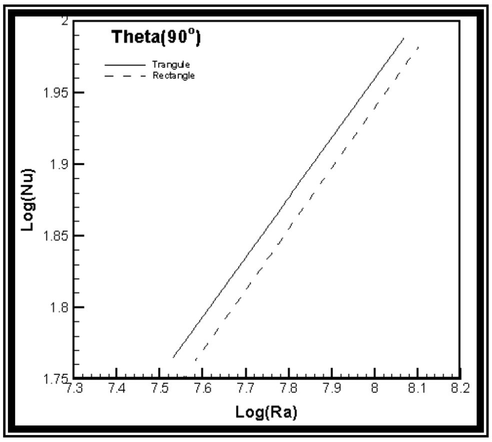

Fig (9) Relationship between Nu and Ra for test section at angle of 90^o in large space.

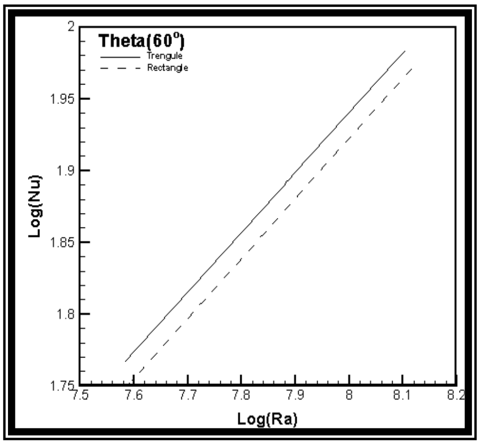

Fig (8) Relationship between Nu and Ra for test section at angle of 60^o in large space.

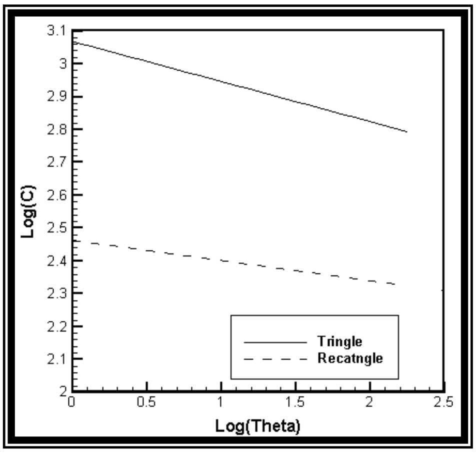

Fig (10) Relationship between slope angle and constant value (C).

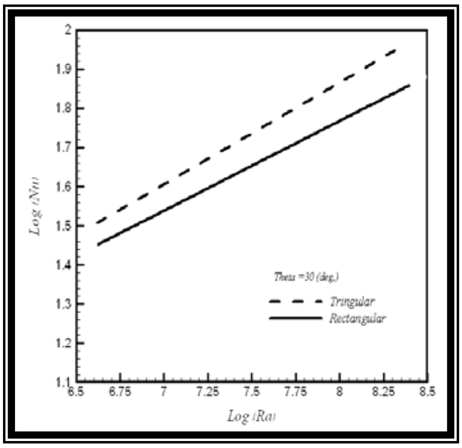

Fig (12) Relationship between Nu and Ra for test section at angle of 30^o in small space.

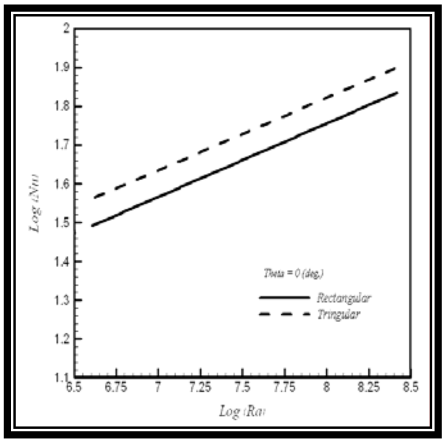

Fig (11) Relationship between Nu and Ra for test section at angle of 0^o in small space.

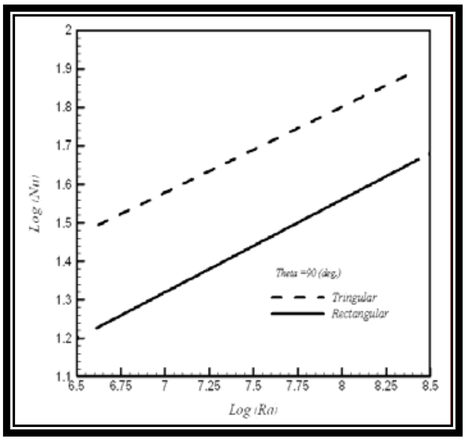

Fig (14) Relationship between Nu and Ra for test section at angle of 90^o in small space.

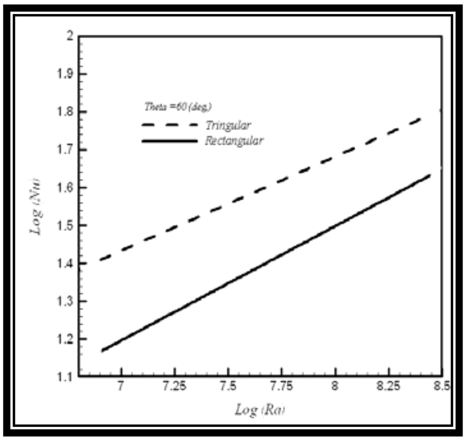

Fig (13) Relationship between Nu and Ra for test section at angle of 60^o in small space.

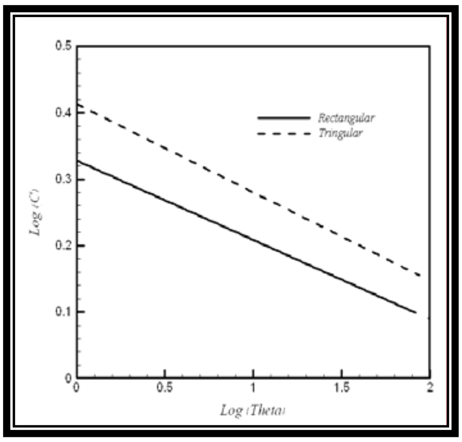

Fig (15) Relationship between slope angle and constant value (C).

NOMENCLATURE:-

A_t Total area in (m^2)
D_o Outer diameter in (m)
D_i Inner diameter in (m)
D_p Outer diameter tube with out fins in (m)
g Gravity acceleration in (m/s^2)
h Convection heat transfer coefficient in ($W/m^2.K$)
k Thermal conductivity in (W/m.K)
L Length of the cylinder in (m)
m Power of Rayleigh number
n The power of slope angle
Nu Nusselt number
Pr Parendtl number
Q_{conv} Heat transfer rate by convection in (W)
Q_q Heat generation in (W)
Q_{rad} Heat transfer rate by radiation in (W)
Ra Rayleigh number
T_a Ambient temperature in (^oC)
T_f Film temperature in (oC)
T_s Surface temperature in (^oC)
T_o Base temperature in (^oC)
T Tip of fin in (mm)
Lc Characteristic length in (m)
s distance between the cylinder and the wall (m).
x distance between two cylinder (m).

Roman symbols:-

β Volumetric expansion coefficient
ε Emmisivity.
σ Stefan - Boltzman constant in (W/m^2K^4)
υ Kinematic viscosity in (m^2/s)