Journal of Engineering and Development

www.jead.org
Vol. 19, No. 06, November 2015
ISSN 1813-7822

EXPERIMENTAL INVESTIGATION OF ORIENTATION EFFECTS ON HEATED CYLINDER WITH DIFFERENT DIAMETERS INSIDE A SQUARE-SECTION SHIELD

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(Received:8/4/2015 ; Accepted:21/6/2015)

Abstract: In this paper, experimental investigation has been done for laminar steady-state free-convection heat transfer from heated hollow cylinder to ambient air inside an open-ended square-section shield with orientation and cylinder diameter effects. Four aluminum heated cylinders of different diameters are employed. The cylinders are heated with various supplied power levels. Six inclinations of heated cylinder with the vertical position, namely 0^o, 15^o, 30^o, 45^o, 60^o and 75^o are used . The effects of orientation on local surface temperatures profile along the heated cylinder , convection heat transfer coefficient , Nusselt number and Rayleigh number at different diameters and heat inputs to the heating element have been plotted and explained. The experimental results show that the local heat transfer coefficient is maximum at inclination angle 75^o and at cylinder base .The present experimental results is compared with previously works shows the good agreement .

Keywords: *Orientation , Heated Cylinder , Shield , Free Convection.*

تحقيق عملي لتأثير الميل على أسطوانة مسخنة بأقطار مختلفة داخل مغلف مربع المقطع

الخلاصة : في هذا البحث أجري تحقيق عملي لأنتقال الحرارة بالحمل الحر الطباقي المستقر من أسطوانة مجوفة مسخنة الى الهواء المحيط داخل مغلف مربع المقطع مفتوح النهايات بزوايا ميل وأقطار مختلفة. تم أستخدام أربع أسطوانات مسخنة من الألمنيوم بأقطار مختلفة. أستخدمت عدة مستويات للقدرة المجهزة للأسطوانات. كما أستخدمت ست زاويا ميل للأسطوانة المسخنة مقاسة مع العمود هي ٠^o و ١٥^o و ٣٠^o و 45^o و 60^o و 75^o . تم تمثيل البيانات برسوم بيانية توضح تأثير ميل الأسطوانة على منحني درجات حرارة السطح الموضعية على طول الأسطوانة المسخنة و معامل أنتقال الحرارة بالحمل ورقم نيسلت ورقم ريليه مع تغيير قطر الأسطوانة والقدرة المجهزة لعنصر التسخين . بينت النتائج العملية أن أقصى قيمة لمعامل أنتقال الحرارة الموضعي عند زاوية ميل 75^o وعند قاعدة الأسطوانة. تم مقارنة البيانات العملية للبحث الحالي مع بحوث سابقة وكان التطابق جيد.

1. Introduction and Literature Review

Free-convection heat transfer from heated vertical and inclined cylinder occurs in many physical problems and engineering applications. The most important of these applications are oil cooling systems , domestic convectors , radiators , refrigerator condensers , compact heat exchangers , electronic systems , thermal fluid systems , nuclear reactors and ground thermo-siphons . Any small increase in heat transfer rate from these systems or equipments decreases the power consumption and increases the life of them . Lin and Armfield [1] numerically studied the transient processes of cooling down and stratification the fluid in a vertical cylinder by free-convection.

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Takhar et al. [2] investigated the natural-convection heat transfer on a vertical cylinder embedded in a thermal stratified high porous medium with effects of non-Darcy including convection , boundary and inertial . They used an implicit finite difference method to solve non linear partial differential equations governing the flow and perturbation method for small and moderate values of axial distance . They found good agreement of the results obtained using perturbation method with the results obtained using implicit finite difference method .

Jarall and Campo [3] experimentally investigated the natural convective heat transfer from heated vertical cylinder to air in laminar and steady-state conditions. They used three stainless-steel cylinders with different diameters and lengths and developed empirical correlation between the local Nusselt number and local modified Rayleigh number .

Roul and Nayak [4] performed an experimental study for natural convection through vertical tube insulated from external and open ended. They discussed the effects of aspect ratio , wall heat flux , tube thickness and tube length on the convection heat transfer behavior.

Mohammed et al. [5] presented an experimental investigation of natural-convection heat transfer over cylindrical heater inside an enclosure. They deduced an empirical correlation between Nusselt and Raylieh numbers from experimental data using least square method and found the heat transfer depended on cylinder diameter , cylinder length and inclination angle of cylinder.

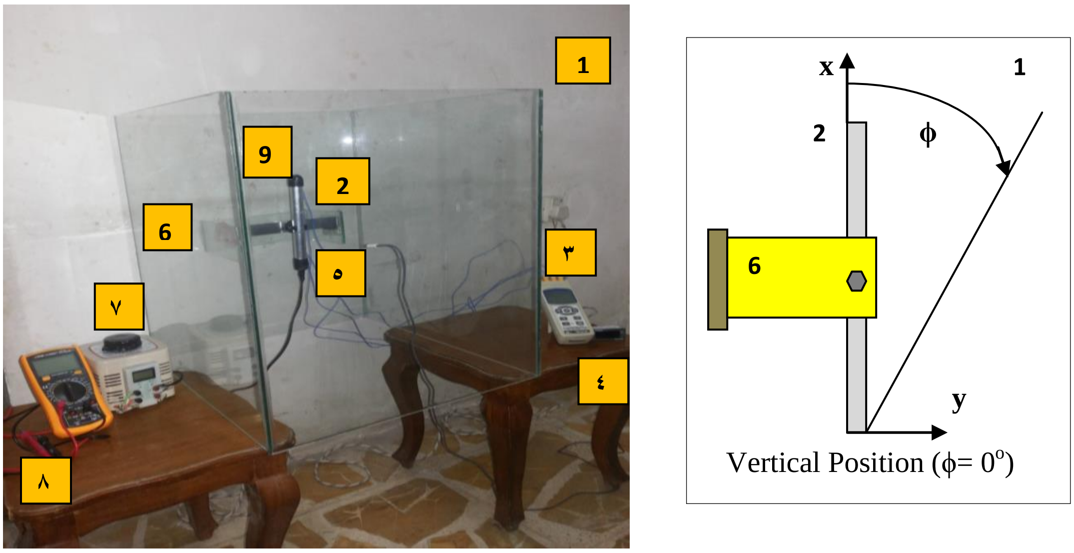
Totala et al. [6] presented an experimental study for natural-convective phenomena through vertical cylinder. They experimentally determined the local convection heat transfer coefficient along the cylinder and compared them with theoretical results obtained using appropriate governing equations and they found very good agreement.

Reddy et al. [7] experimentally studied the free-convection heat transfer over array of vertical cylinders with inclination angle and Rayleigh number effects on temperature distribution. They showed the temperature differences from cylinders to ambient air decreases for higher values of inclination angles and Rayleigh numbers . The aim of the present work is to investigate experimentally the influence of orientation of the heated cylinder inside opened shield on the free-convection heat transfer behavior and thermal properties in a laminar condition , Nusselt number and Rayleigh number. The current work will be carried out for a wide range of the inclinations at different cylinder diameters and input powers.

2. Experimental Work

2.1. Experimental Rig

The experimental rig shown in Fig.1 consists of aluminum hollow cylinder. The outside surface of cylinder is highly polished to minimize the radiation heat transfer loss . The hollow cylinder is heated from the internal surface using a coiled wire electrical heater of 250 *W* capacity. The electrical heater is coiled around a helical ceramic shaft to kept it inside and along the cylinder. The small gap between the ceramic shaft and internal surface of cylinder is filled by sand. Ends of the heated cylinder are insulated using two pieces of the low conductivity rubber. The electrical power input to the heating element inside the cylinder is changed using a contact type voltage regulator with digital reader and the current input is measured by a portable digital multi-meter type VICTOR VC890C^{+}. The assembled heated cylinder is supported in a vertical position with an adjustable support allowing different angles of inclinations in a square-section shield constructed of cast acrylic sheet of thickness 6 *mm* , with dimensions (500 *mm* length × 500 *mm* width × 600 *mm* height) opened from top and bottom faces. The heat loss from heated cylinder to ambient air is by free-convection . Four diameters of the tested hollow cylinder 20 , 25 , 30 and 35 *mm* are used with 2 *mm* thickness. The cylinder length is kept fixed at 200 *mm* . Different heater input levels of 25 , 50 ,75 and 100 *W* are used. The surface temperature of heated cylinder is measured by four K-type calibrated thermocouples are inserted into the hollow cylinder and along the cylinder length axially with a spacing of 50 *mm* and the distances between the heated cylinder ends and the first and fourth thermocouples are 25 *mm* . The thermocouples are connected to four channels data logger thermometer model TM-947SD. Another digital electronic K-type calibrated thermocouple is used to measure the ambient temperature inside the shield.



1.Opened Shield **2.**Heated Cylinder **3.**Data Logger Thermometer **4.**Digital Electronic Thermocouple **5.**Thermocouple Wires **6.**Carrier **7.**Voltage Regulator **8.**Digital Multi-Meter **9.**Rubber

Fig. 1 Photograph and Side Schematic of the Experimental Rig

2.2. Experimental Procedure and Calculations

The surface temperatures of heated cylinder are measured along the cylinder for six inclination angles with the vertical position, namely 0^o , 15^o, 30^o , 45^o, 60^o and 75^o. The readings of all thermocouples have been recorded every 30 minutes approximately under steady-state condition .

The total heat power lost from the heated cylinder is equal to the electrical power supplied to heating element (Q_{in}):

$$Q_{in} = I\,V \qquad (1)$$

The total heat input (Q_{in}) by heating element transform to heat and it transfers by free-convection (Q_C) from heated cylinder surface to ambient air in addition of , the heat transfer lost by radiation (Q_R) and conduction (Q_{cond}) , then :

$$Q_{in} = Q_C + Q_R + Q_{cond} \qquad (2)$$

By substituting Eq. (1) in Eq.(2) , the convection heat transfer becomes :

$$Q_C = I\,V - Q_R - Q_{cond} \tag{3}$$

The radiation heat loss (Q_R) is [8-11] :

$$Q_R = \varepsilon\sigma A_S\left(T_{Sav}^4 - T_A^4\right) \tag{4}$$

The radiation heat transfer loss (Q_R) from the heated cylinder is small because the emissivity (ε) of aluminum highly polished cylinder is 0.05 and it was found less than 4% of the heat input (Q_{in}) . The loss by heat conduction (Q_{cond}) is very small because the ends of heated cylinder are good insulated and it can be neglected.

According to Newton's law of cooling, the heat transfer from the heated cylinder by free-convection (Q_C) is given as [8-11] :

$$Q_C = h_{av}\, A_S(T_{Sav} - T_A) \tag{5}$$

Hence , the average convection heat transfer coefficient (h_{av}) is:

$$h_{av} = \frac{Q_C}{A_S(T_{Sav} - T_A)} \tag{6}$$

where , (T_{Sav}) is the average surface temperature :

$$T_{Sav} = \frac{1}{L}\int_0^L T_{Sx}\, d_x \tag{7}$$

The local convection heat transfer coefficient (h_x) is calculated as :

$$h_x = \frac{Q_C}{A_S(T_{Sx} - T_A)} \tag{8}$$

The local Nusselt number (Nu_x) at the heated cylinder is computed as [8-11] :

$$Nu_x = \frac{h_x x}{k} \tag{9}$$

and the average Nusselt number (Nu_{av}) is :

$$Nu_{av} = \frac{h_{av}\, D}{k} \tag{10}$$

The local Grashof number (Gr_x) is calculated as [8-11] :

$$Gr_x = \frac{g\beta(T_{Sx} - T_A)x^3}{\nu^2} \tag{11}$$

and the average Grashof number (Gr_{av}) is :

$$Gr_{av} = \frac{g\beta(T_{Sav} - T_A)D^3}{\nu^2} \tag{12}$$

Define local Rayleigh number (Ra_x) as the product of the local Grashof number and Prandtl number [8-11] :

$$Ra_x = Gr_x\, Pr \tag{13}$$

and the average Rayleigh number (Ra_{av}) is :

$$Ra_{av} = Gr_{av}\, Pr \tag{14}$$

All properties of air are taken at film temperature (T_f) . It calculated as a average of the ambient temperature (T_A) and surface temperatures (T_{Sav}) along the heated cylinder as :

$$T_f = \left(\frac{T_A + T_{Sav}}{2}\right) + 273.18 \tag{15}$$

and replaced T_{Sav} by T_{Sx} when local parameters are calculated .

3. Results Analysis

Local temperature difference ($\Delta T_x = T_{Sx} - T_A$) against heat flux ($q = Q_{in} / A_S$) for various axial distance (x) along the heated cylinder at diameter (D= 25 *mm*) and inclination angle ($\phi = 0^o$ vertical position) is shown in Fig. 2.

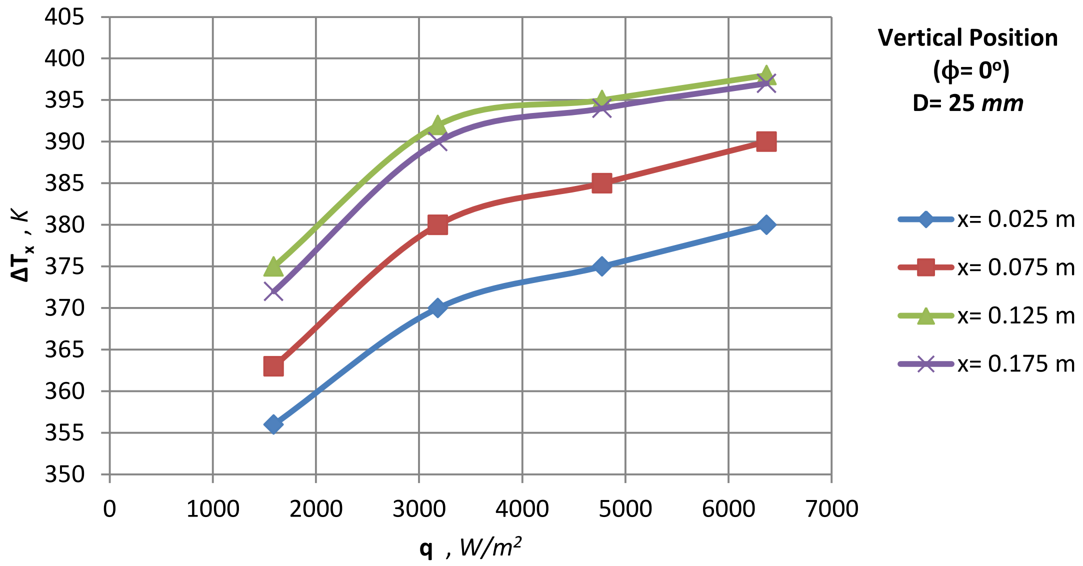


Fig. 2 Local Temperature Difference Against Heat Flux at Various Axial Distance.

Local temperature difference (ΔT_x) profiles along the heated cylinder for different orientations from $\phi = 0^o$ (vertical position) to $\phi = 75^o$ at heat input ($Q_{in} = 50$ *W*) and diameter (D= 25 *mm*) is shown in Fig. 3.

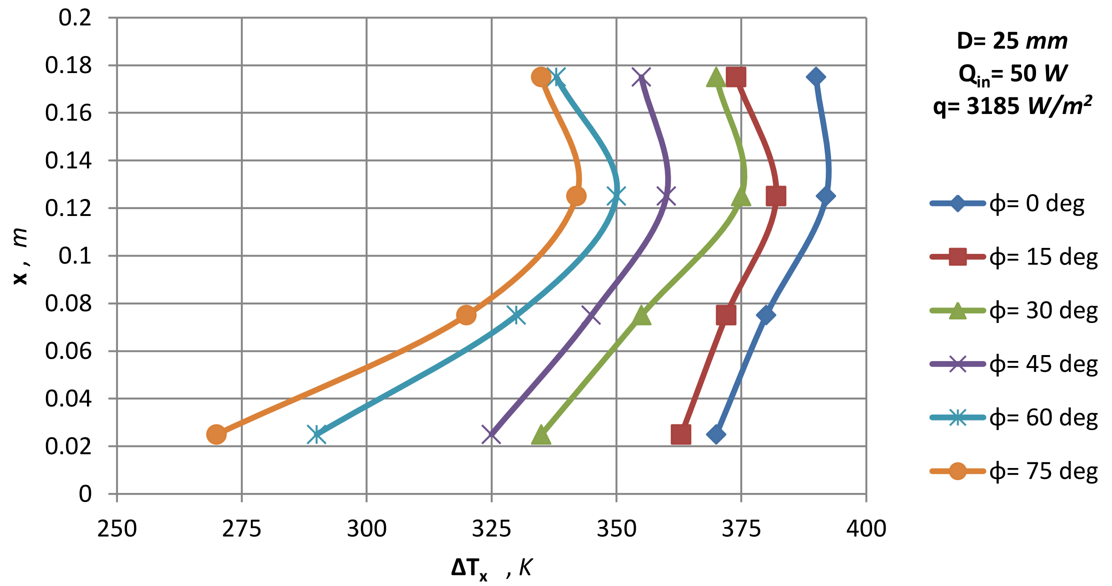


Fig. 3 Local Temperature Difference Profiles along the Heated Cylinder at Different Orientation.

Figs. 4 , 5 and 6 show local convection heat transfer coefficient (h_x) profiles , variation of local Nusselt number (Nu_x) and Rayleigh Number (Ra_x) along the heated cylinder for different inclination angles (ϕ) at heat input (Q_{in}= 50 *W*) and diameter (D= 25 *mm*) respectively .

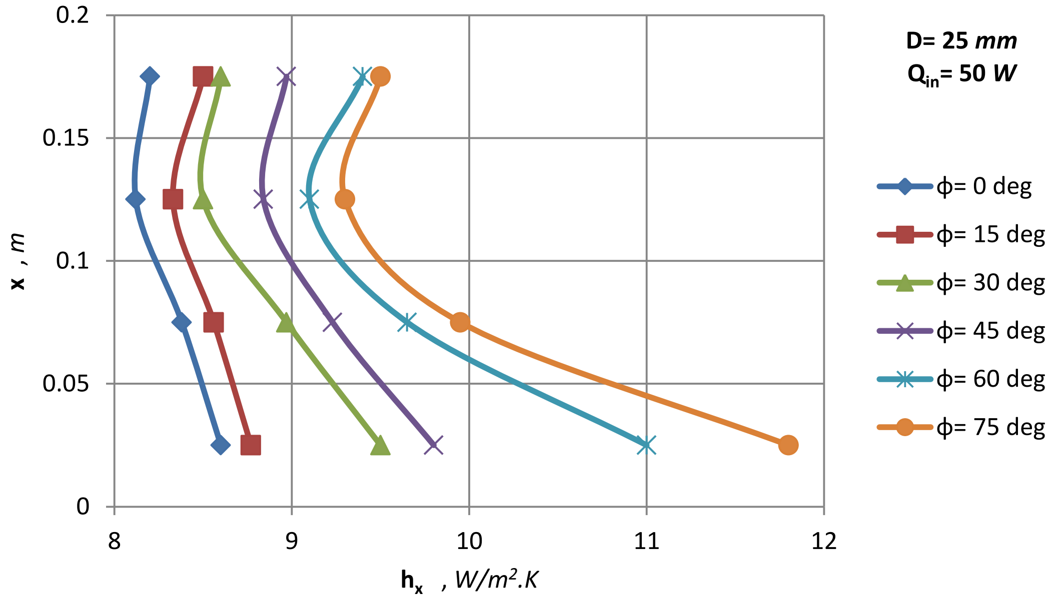


Fig. 4 Local Convection Heat Transfer Coefficient Profiles along the Heated Cylinder at Different Orientation.

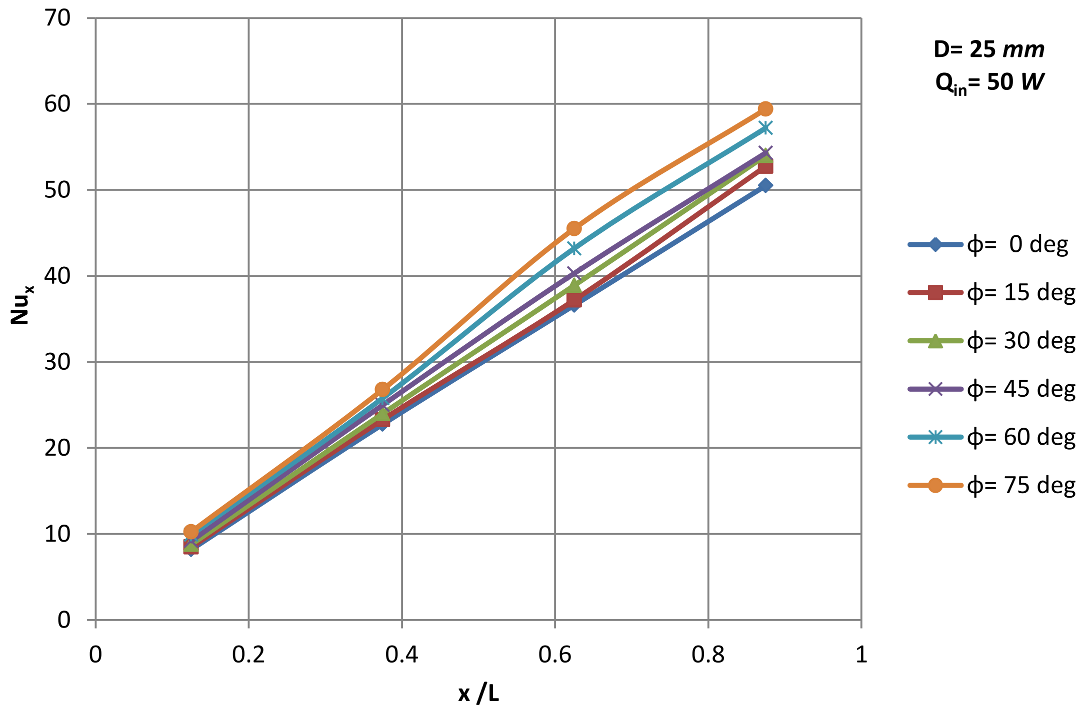


Fig. 5 Local Nusselt Number along the Heated Cylinder for Different Orientation.

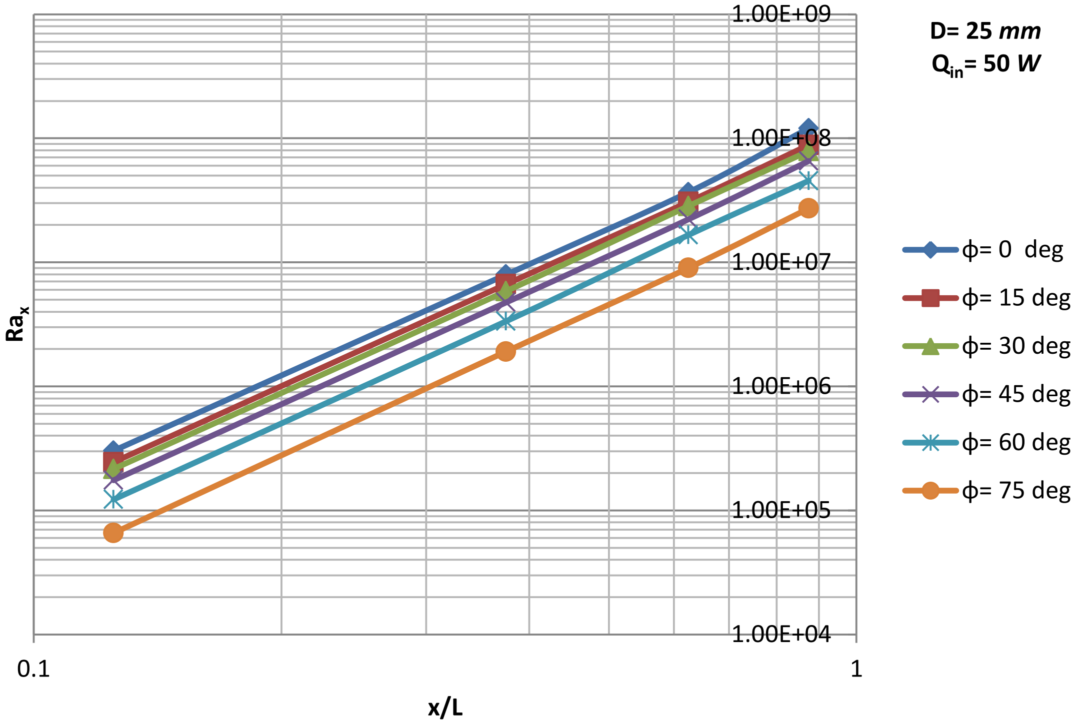


Fig. 6 Local Rayleigh Number (Ra_x) along the Heated Cylinder for Different Orientation .

Variation of Local Nusselt Number (Nu_x) with Local Rayleigh Number (Ra_x) for various inclination angles (ϕ) at heat input (Q_{in}= 50 *W*) and diameter (D= 25 *mm*) are shown in Fig. 7 .

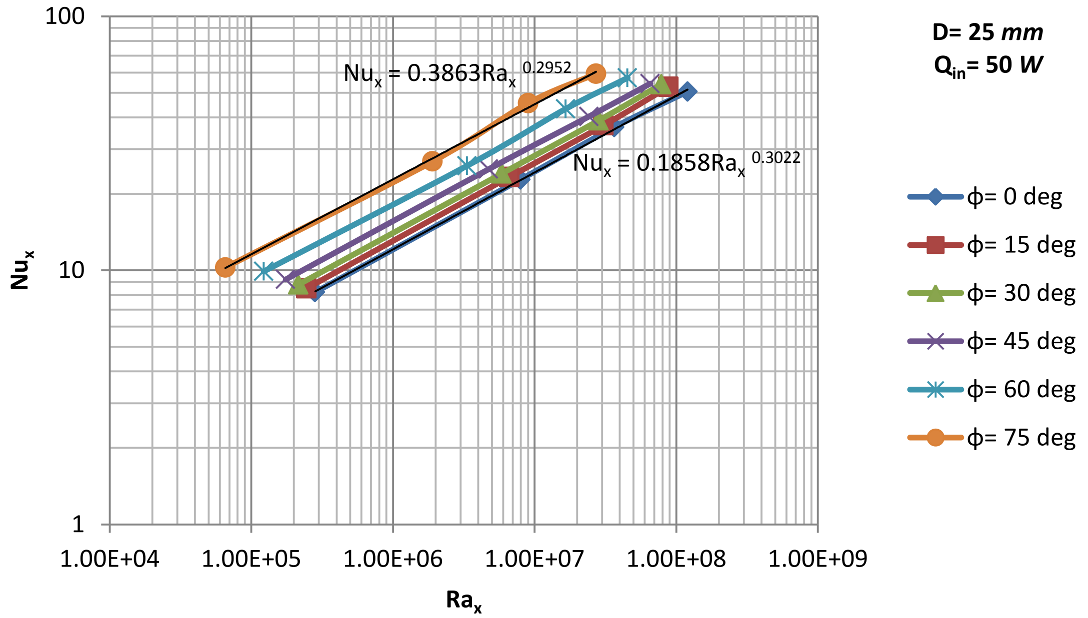


Fig. 7 Local Nusselt Number (Nu_x) versus Local Rayleigh Number (Ra_x) for Various Orientations

Figs. 8 , 9 and 10 show variation of average convection heat transfer coefficient (h_{av}) with inclination angle (ϕ) , average Nusselt Number (Nu_{av}) with diameter (D) and average Rayleigh number (Ra_{av}) with inclination angle (ϕ) at heat input (Q_{in}= 50 *W*) respectively .

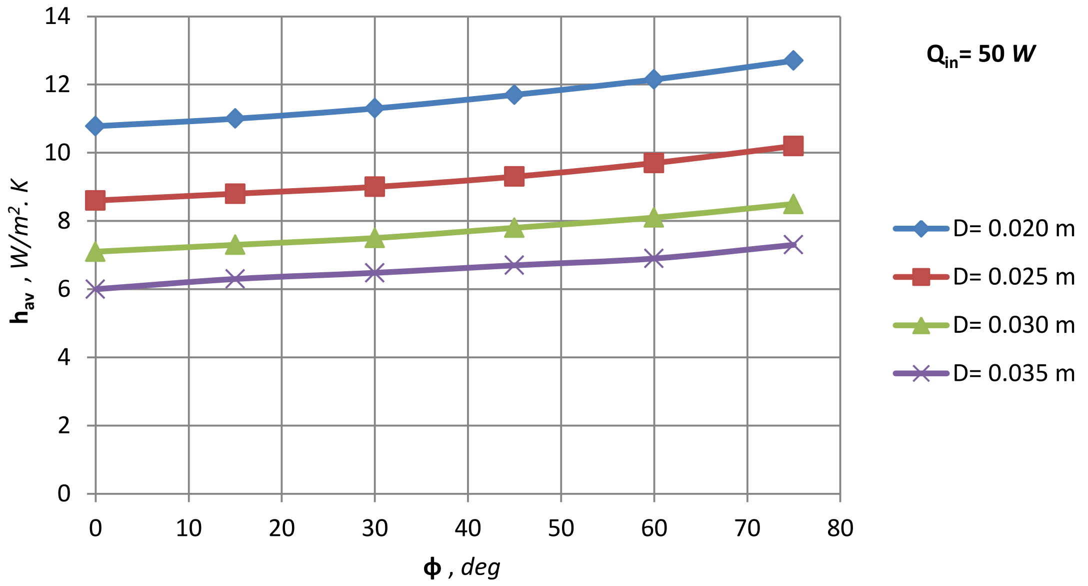


Fig. 8 Average Convection Heat Transfer Coefficient (h_{av}) versus Orientation for Different Diameters .

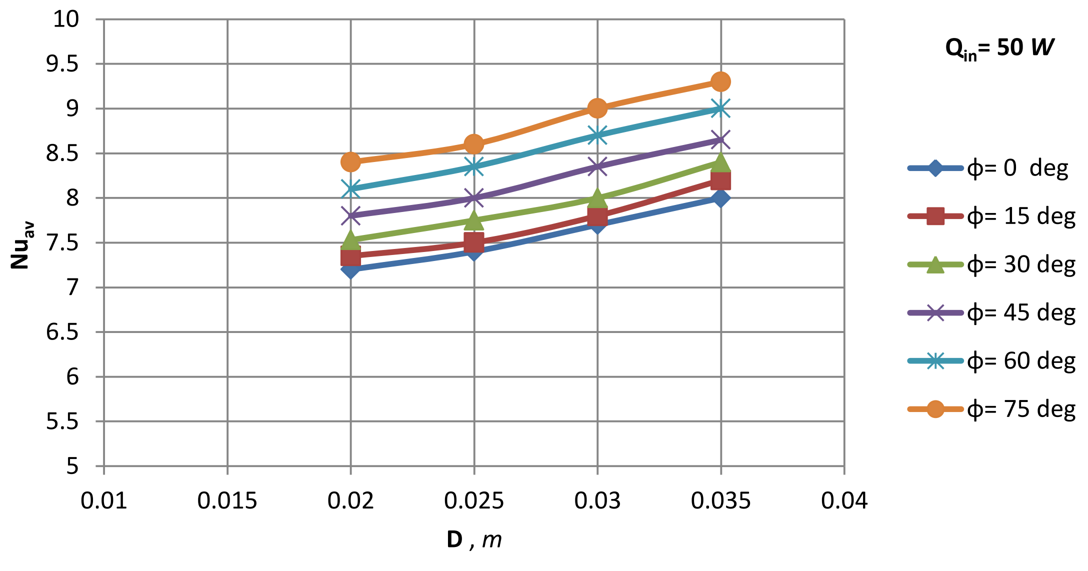


Fig. 9 Average Nusselt Number (Nu_{av}) versus Diameter of Heated Cylinder (D) for Various Orientations.

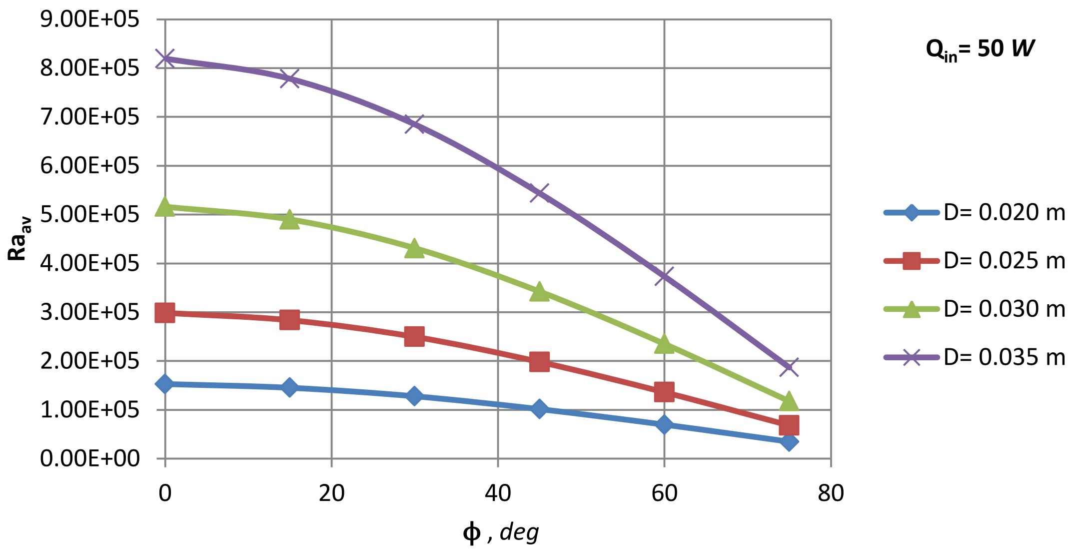


Fig. 10 Average Rayleigh Number (Ra_{av}) Versus Orientation (φ) for Various Diameters .

Experimental Results of the local Nusselt number (Nu_x) for present work are compared with Jarall and Campo's empirical correlation [3] of free-convection heat transfer from heated cylinder at vertical position ($\phi = 0^o$) :

$$\mathrm{Nu_x} = 1.2849(\mathrm{Ra_x^*})^{0.1651} \qquad (16)$$

where, Ra_x^* is modified local Rayleigh number . It is equal :

$$\mathrm{Ra_x^*} = \mathrm{Ra_x}\left(\frac{\mathrm{x}}{\mathrm{D}}\right) \qquad (17)$$

The experimental results are correlated to a formula between local Nusselt number (Nu_x) versus modified local Rayleigh number (Ra_x^*) for vertical position of heated cylinder as :

$$\mathrm{Nu_x} = 0.4685(\mathrm{Ra_x^*})^{0.2283} \qquad (18)$$

It is compared with a Jarall and Campo's work . The absolute percentage relative of error (ε%) between the present work and Jarall and Campo's empirical correlation is (ε% < 12 %) . The squared correlation coefficients (R^2= 99%) as shown in Fig. 11 and are found to be in good agreement .

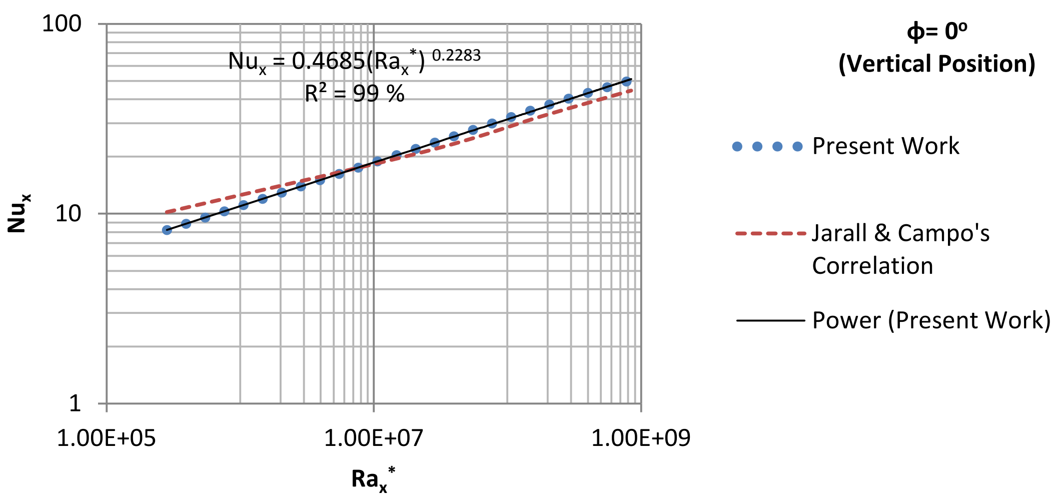


Fig. 11 Local Nusselt Number (Nu_x) Versus modified Local Rayleigh Number (Rax*) of Present Work and Jarall and Campo's Correlation.

Also the illustrated experimental results in Fig. 7 are correlated to a formulas between local Nusselt number (Nu_x) and local Rayleigh number (Ra_x) for heated cylinder at various of inclination angles as follows :

$$Nu_x = 0.1858Ra_x^{0.3022} \qquad \phi = 0^o \qquad (19)$$

$$Nu_x = 0.1826Ra_x^{0.309} \qquad \phi = 15^o \qquad (20)$$

$$Nu_x = 0.2023Ra_x^{0.3068} \qquad \phi = 30^o \qquad (21)$$

$$Nu_x = 0.2485Ra_x^{0.2994} \qquad \phi = 45^o \qquad (22)$$

$$Nu_x = 0.3Ra_x^{0.2979} \qquad \phi = 60^o \qquad (23)$$

$$Nu_x = 0.3863Ra_x^{0.2952} \qquad \phi = 75^o \qquad (24)$$

The absolute percentage relative of error ($\varepsilon\% < 2\%$) between these correlations and data of experimental work are ($\varepsilon\% < 2\%$) and the squared correlation coefficients are ($R^2 \simeq 99\%$) .

4. Conclusions

Steady state free-convection heat transfer from heated cylinder to ambient air inside opened square-section shield is investigated experimentally and focused on the effects of orientation on variation of the local surface temperatures , local Nusselt number and local Raleigh number along the heated cylinder with various diameters and heat inputs. The obtained experimental data and results in the present work indicate that :-

- The local convection heat transfer coefficient (h_x) increase as inclination angle moves from vertical position towards a horizontal position with percentage reach to 27 % at ϕ= 75^o compared with inclination angle ϕ= 0^o. The maximum value of local heat transfer coefficient (h_x) is at base of heated cylinder because the boundary layer development is starting from the base and it decreases in the upward direction because the boundary layer became thicker along the heated cylinder.

- The local Nusselt number (Nu_x) and local Rayleigh number (Ra_x) increase along the length of heated cylinder at different orientations .
- Average heat transfer coefficient (h_{av}) increases with increasing orientation and decreasing cylinder diameter.
- Average Nusselt number (Nu_{av}) and average Rayleigh number (Ra_{av}) increased by increasing diameter of the heated cylinder .

Abbreviations

A_S	Surface area of heat transfer , (m^2)
D	Diameter of heated cylinder , (*m*)
g	Gravitational acceleration , (m/s^2)
Gr	Grashof number , (-)
h	Convection heat-transfer coefficient , ($W/m^2.K$)
I	Input current intensity , (*A*)
k	Thermal conductivity (*W/m.K*)
L	Length of heated cylinder , (*m*)
Nu	Nusselt number , (-)
Pr	Prandtl number , (-)
q	Heat flux , (W/m^2)
Q_C	Convection heat transfer , (*W*)
Q_{cond}	Conduction heat transfer , (*W*)
Q_{in}	Total heat input by heating element , (*W*)
Q_R	Radiation heat transfer , (*W*)
Ra	Rayleigh number , (-)
T_A	Ambient temperature inside the enclosure , (^oC)
T_f	Film temperature , (^oC)
T_S	Surface temperature , (^oC)
ΔT	Difference of the surface and ambient temperature , (^oC)
V	Voltage supplied , (*V*)
x	Axial distance (distance measured from cylinder base to the position of thermocouple) along the heated cylinder , (*m*)

Greek Letters :

β	Volumetric coefficient of thermal expansion , (*1/K*)
ε	Emissivity of the surface , (-)
σ	Stefan-Boltzmann constant , ($\sigma = 5.67 \times 10^{-8}$ $W/ m^2.K^4$)
ϕ	Angle of inclination of the cylinder with vertical position , (*deg*)
$\boldsymbol{\nu}$	Kinematic viscosity , (m^2/s)

Subscript Symbols :

$_{av}$	Average
$_x$	Local

5. References

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