



# Experimental Study on the Impact of External Geometrical Shape on Free and Forced Convection Time Dependent Average Heat Transfer Coefficient during Cooling Process

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(Received 20 December 2011; accepted 11 March 2012)

## Abstract

In this research, an experimental study was conducted to highlight the impact of the exterior shape of a cylindrical body on the forced and free convection heat transfer coefficients when the body is held in the entrance of an air duct. The impact of changing the body location within the air duct and the air speed are also demonstrated. The cylinders were manufactured with circular, triangular and square sections of copper for its high thermal conductivity with appropriate dimensions, while maintaining the surface area of all shapes to be the same. Each cylinder was heated to a certain temperature and put inside the duct at certain locations. The temperature of the cylinder was then monitored. The heat transfer coefficient was then calculated for forced convection for several Reynolds number (4555-18222). The study covered free convection impact for values of Rayleigh number ranging between (1069-3321). Empirical relationships were obtained for all cases of forced and free convection and compared with equations of circular cylindrical shapes found in literature. These empirical equations were found to be in good comparison with that of other sources.

**Keywords:** *heat transfer, free convection, forced convection, circular cylinder, square cylinder, triangular cylinder.*

## 1. Introduction

Heat transfer to and from a bank of tubes in cross flow is relevant to numerous heat exchanger applications, such as steam generator in a boiler or air cooling in the coil of an air conditioner

In these applications, one fluid moves over the tubes, while a second fluid at a different temperature passes through the tubes and hence, heat is exchanged between the fluids based on the convection heat transfer coefficient. The tube rows of the bank are either arranged in staggered or aligned configuration in the direction of flow.

The flow conditions within the bank are dominated by boundary layer separation effects and by wake interactions, which in turn influence the convection heat transfer. Hence, the heat transfer coefficient associated with a tube is determined by its configuration and position of the bank. The heat transfer coefficient of a tube

with staggered configuration is higher than that associated with the aligned one.

The modeling of these relationships has been the concern of many researchers. Generally, the average heat transfer coefficient for the entire tube bank is evaluated empirically based on the maximum fluid velocity. Different forms of empirical correlations were proposed for air flow across tube bank with different geometry and configurations the applicability of these empirical models is limited to a confined range of flow conditions due to the complexity of the relationships. In this sense, therefore, artificial neural networks (ANNs) were applied in modeling heat transfer phenomena of different heat exchanger applications because of providing better and more reasonable solutions (Islamoglu, 2003)[1]. Jambunathan et al. (1996), [2] applied neural network model for prediction of convective heat transfer. More recently, Varshney and Panigrahi (2005) [3] developed a neural network

based control for a heat exchanger in a closed flow air circuit. Fatona (2008) [4] studied the application of Artificial Neural Network (ANN) in modeling the heat transfer coefficient of a staggered multi-row, multi-column, cross-flow, tube-type heat exchanger. Heat transfer data were obtained experimentally for air flowing over a bank of copper tubes arranged in staggered configuration with 5 rows and 4 columns. The Reynolds number and the row number were used as input parameters, while the Nusselt number was used as output parameter in training and testing of the multi-layered, feed-forward, back-propagation neural networks. The results show that the ANNs model were developed for the prediction of the convection heat transfer coefficient of air flowing over a staggered, multi-row, multi-column, cross-flow, copper tube-type heat exchanger. The model has high prediction performance with mean relative error (MRE) less than 1% for the training data set and less than 4% for the testing data sets respectively. The ANNs model can therefore be used as a modeling tool for preliminary design of heat exchangers.

Dhiman and Chhabra (2007) [5] studied the effects of cross-buoyancy and of Prandtl number on the flow and heat transfer characteristics of an isothermal square cylinder confined in channel. The numerical results have been presented for a range of conditions as:

$1 \leq Re \leq 30$ ,  $0.7 \leq Pr \leq 100$  (the maximum value of Peclet number being 3000) and  $0 \leq Ri \leq 1$  for a fixed blockage ratio of 0.125. The drag coefficient is found to be less sensitive to the Richardson number than the lift coefficient. The combined forced and free convection heat transfer from circular, square or triangular shapes has numerous industrial applications such as cooling tower, oil and gas pipelines, tubular and compact heat exchangers, cooling of electronic components, and flow dividers in polymer processing applications and so on.

Hashemian M. Rahnema (2008) [6] studied the turbulent heat transfer, in a three dimensional channel flow, in the presence of a square cylinder, was investigated numerically. The existence of a square cylinder in a channel, compared to a plain. One, changes the heat transfer rate from the walls of the channel. A Large Eddy Simulation (LES) of a turbulent flow was performed to simulate flow behavior in a channel for Reynolds numbers in the range of 1000 to 15000. The results obtained for the Nusselt number distribution along the wall of the channel, at  $Re = 3000$ , followed those of experimental data with good accuracy. It was

observed that the existence of a square cylinder makes the attached wall boundary layer separate, with a subsequent recirculation zone downstream of the cylinder. The Nusselt number distribution along the wall of the channel shows an increase, with a relative maximum, slightly downstream of the reattachment point. Heat transfer from the wall of the channel increases With increasing Reynolds number. A correlation was obtained for the variation of the mean total Nusselt number with the Reynolds number.

The present study is aimed at high leisters the effect of the external shape of a cylindrical body on the convection heat transfer coefficient when the body is subjected to free or forced convection heat transfer by cross flow air stream. The Body Locations can also vary inside the air duct to determine its effect on the heat transfer rates.

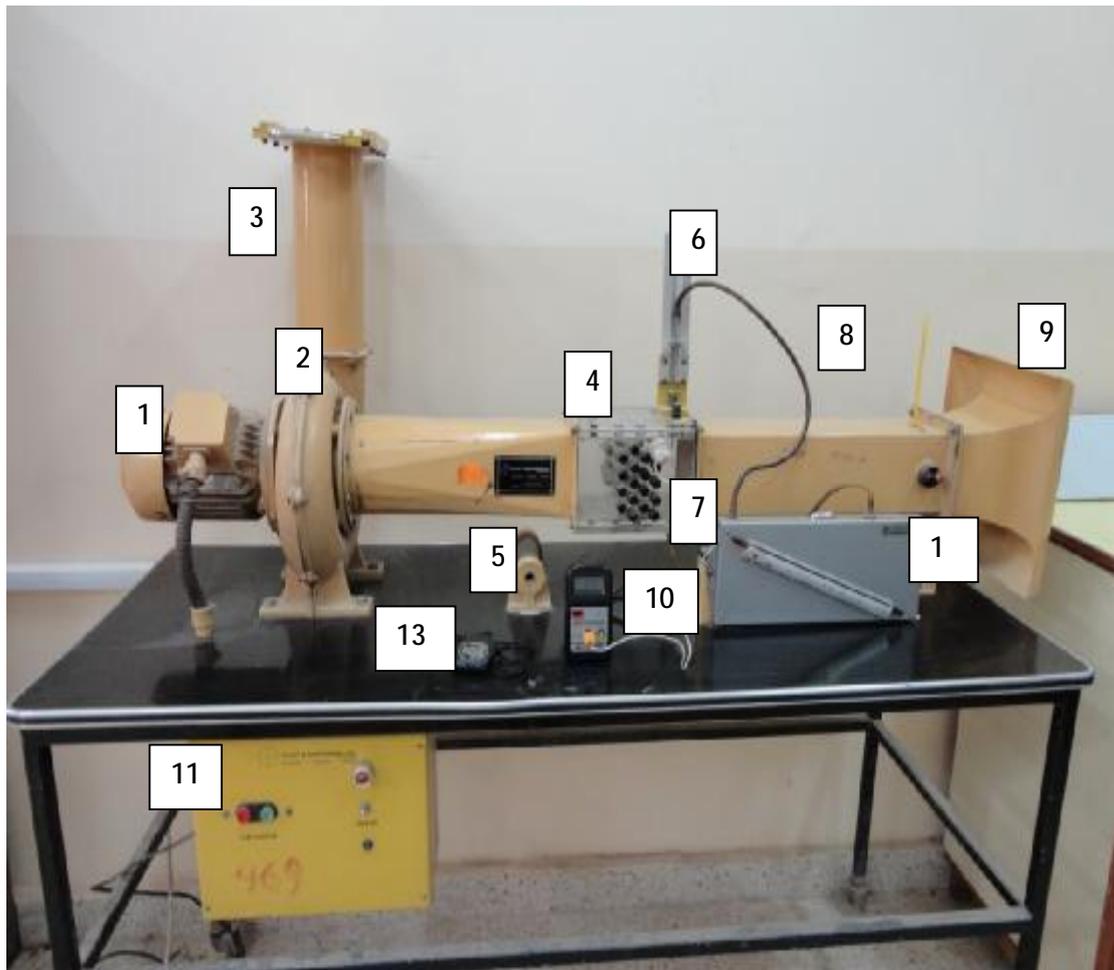
## 2. Experimentation

The cross-flow heat exchanger apparatus (Model TE.93/A, Plant Engineers, England) was used for the purpose of data gathering. The experimental setup is shown in Figure (1).

Air at ambient temperature (working fluid), driven by a centrifugal fan powered by a 1 hp electric motor at a constant speed of 2,500 rpm, is blown perpendicularly over a cylindrical copper bar. The air flow rate over the cylinder is regulated by a throttle valve attached to the discharge end of the centrifugal fan. The Copper bar was manufactured in the form of a cylinder with three different cross sections, while maintaining the proven surface area constant the three cross section tested shapes were circular, triangular and square as shown in Figure(2) . Table (1) shows the form of cylinders with dimensions and properties.

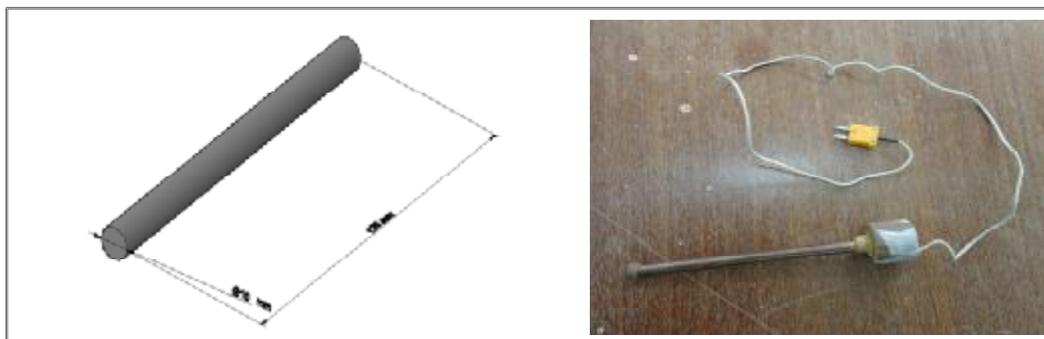
**Table 1,**  
**Physical and Geometrical Data for the Three Tested Cylinders.**

<b>Circular cylinder</b>	m= 0.0875 kg D= 10 mm Cp= 380 J/kg k
<b>Square cylinder</b>	m=0.7 kg L <sub>1</sub> = 7.8 mm Cp= 380 J/kg k
<b>Triangular cylinder</b>	m= 0.057 kg L <sub>2</sub> = 10.2 mm Cp= 380 J/kg k

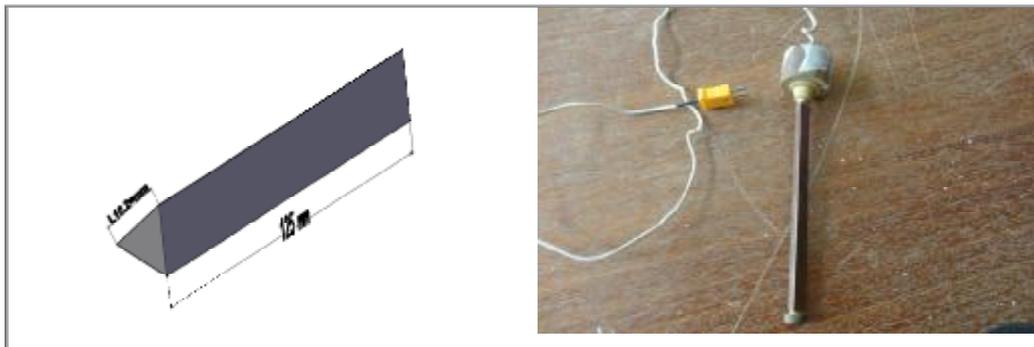


1	Electric motor	6	Total head tube	11	Control panel
2	Fan	7	Test element	12	Inclined manometer
3	Throttle opening	8	Thermometer	13	watch timer
4	Working section	9	Air inlet		
5	Electric heater	10	Digital thermometer		

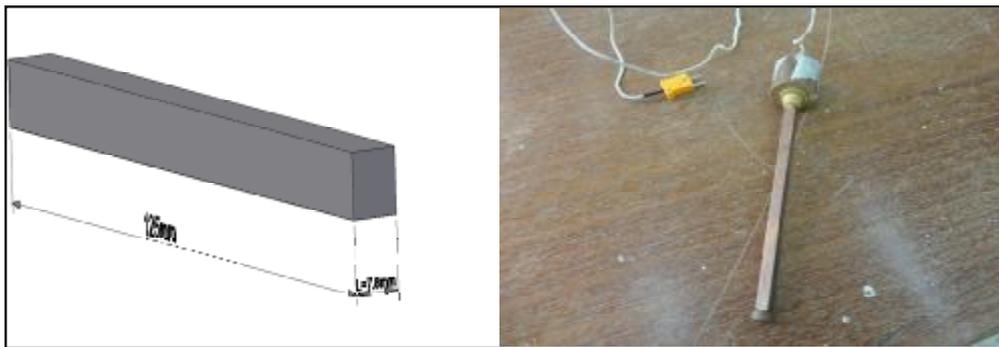
Fig. 1.The Experimental Setup.



Circular Cylinder



Triangular Cylinder



Square Cylinder

Fig. 2. Three Cross Section Al Tested Shapes (i.e. Circular, Triangular and Square)

The copper cylinder is heated to a maximum of about 85C with an electric heater, A K-type; 0.2 mm diameter thermocouple was imbedded at the centre of the cylinder to measure the temperature of the cylinder. The thermocouple voltage output

is wired to digital thermometer to measure temperature. The heated cylinder is then inserted into the spaces provided in the working section at different locations (Y) is the vertical distance of working section, see Figure (3).

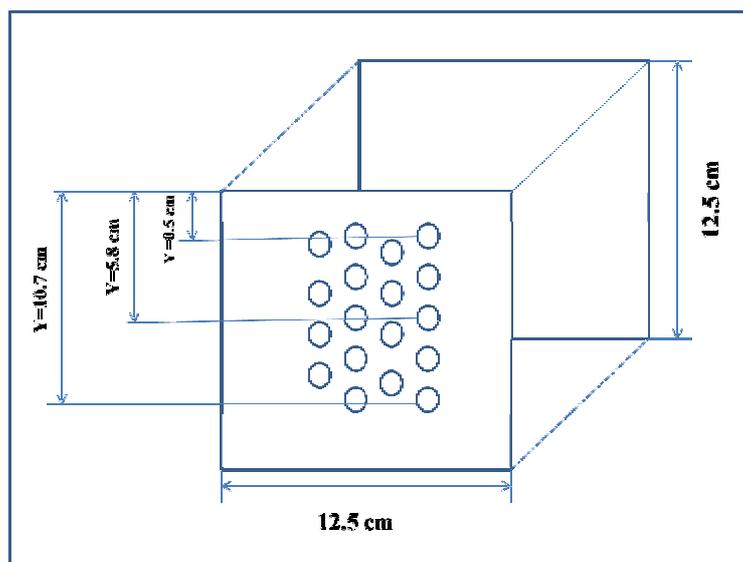


Fig. 3. The Different Locations at the Test Duct.

At each columns position, the cooling rate of the cylinder as indicated by a thermocouple embedded at its centre was recorded at the rate of 1 data per second by digital multimeter for 4 different flow rates with throttle valve at ( 20, 40, 60 ,80%) openings. The air velocity over the tube bank was measured with a total head tube connected to an inclined water manometer. By measuring the time of cooling. The heat transfer

coefficient of forced convection and the Nusselt number are calculated. After that empirical relationships were found several free convection tests were also carried out and the empirical were forced. Figure (4) show the position of thermocouple used to measure the temperature in the circular cylinder.

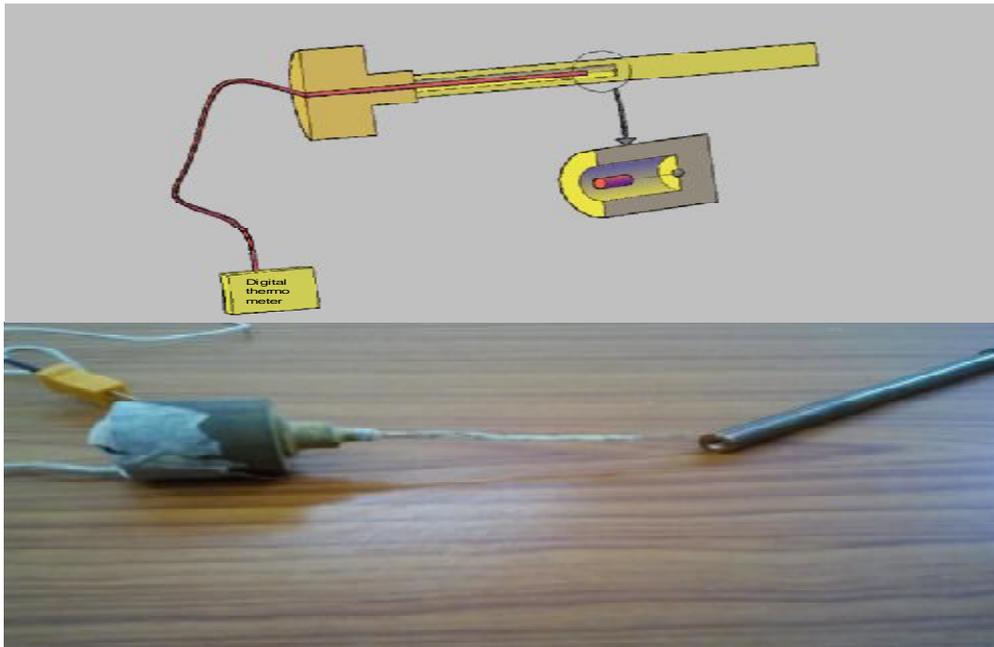


Fig. 4. The Position of Thermocouple Used to Measure the Temperature in the Circular Cylinder.

### 3. Theoretical Analysis

For Forced convection the Reynolds number ( $Re_{Dh}$ ) of air flow is determined using the following relation [7].

$$Re = \rho_a u D_h / \mu_a \quad \dots(1)$$

$$D_h = 4 * A / P \quad \dots(2)$$

The Prandtl number is determined using the equation:

$$Pr = \mu_a * cp_a / k_a \quad \dots(3)$$

And determined ( $T_{film}$ )

$$T_{film} = (T_s + T_a) / 2 \quad \dots(4)$$

For the purpose of estimating the heat transfer coefficient, it is assumed that all heat lost from the cylinder is transferred to the air flowing past it. It is also assumed that temperature gradient within the cylinder thickness is negligible, so that the thermocouple embedded at the centre in the inner

diameter gives a true indication of the effective surface temperature of the cylinder. The rate of heat loss from Cylinder to air is given by [4]:

$$q^0 = h A_1 (T - T_a) \quad \dots(5)$$

In a period of time (dt) the temperature drop (dT) is given as:

$$-q^0 dt = m cp dT \quad \dots(6)$$

Combining Equations (5) and (6) and eliminating  $q^0$  give the following:

$$-dT / (T - T_a) = (h * A_1 / m * cp) dt \quad \dots(7)$$

Integrating Equation (8) gives:

$$\ln (T - T_a) - \ln (T_0 - T_a) = (-h * A_1 * t) / m * cp \quad \dots(8)$$

Where:

$T_0$  = cylinder temperature at time ( $t$ ) = 0.

The plot of  $\ln (T - T_a)$  against  $t$  yields a straight line of slope (M) as:

$$M = (-h * A_1) / (m * cp) \quad \dots(9)$$

From which the heat transfer coefficient (h) is calculated as:

$$h = (-m \cdot c_p / A_1) \cdot M \quad \dots(10)$$

The fully developed Nusselt number (Nu) is evaluated by

$$Nu = (h \cdot D_h) / k_a \quad \dots(11)$$

The experimental Nusselt number is calculated as [8]:

$$Nu = c \text{Re}^m \text{Pr}^{0.3} \quad \dots(12)$$

The values of the constants (c & m) can be obtained from the relationship between the graphs  $\ln(Nu/\text{Pr}^{0.3})$  and  $(\ln \text{Re})$ .

For Free convection the empirical relationship for free convection may be determined from [8]:

$$Nu = c (\text{Ra})^n \quad \dots(13)$$

Where: Ra is Rayleigh number and is given by:  
 $\text{Ra} = \text{Gr} \cdot \text{Pr}$

The Grashof Number is given by:

$$\text{Gr} = (\beta \cdot (\rho_a)^2 \cdot g \cdot \Delta t \cdot (L_c)^3) / (\mu_a)^2 \quad \dots(14)$$

Where:

The ( $\beta$ ) coefficient of volumetric expansion of air is calculated from the following relationship

$$\beta = 1/T_{\text{film}} \quad \dots(15)$$

Natural convection heat transfer coefficients are typically very Low compared to those for forced convection. Therefore, radiation is usually disregarded in forced convection problems, but it must be considered in natural convection problems that involve gas. This is especially the case for surfaces with high emissivities. The total rate of heat transfer is determined by adding the convection and radiation components,

$$Q_{\text{total}} = Q_{\text{conv}} + Q_{\text{rad}} \quad \dots(16)$$

Radiation heat transfer from a surface at temperature  $T_s$  surrounded by surfaces at a temperature  $T_1$  (both in absolute temperature unit K) is determined from:

$$Q_{\text{rad}} = \epsilon \sigma A_s (T_s^4 - T_1^4) \quad \dots(17)$$

$$Q_{\text{conv}} = h \cdot A_s \cdot (T_s - T_1) \quad \dots(18)$$

$$h_{\text{equivalent}} = Q_{\text{total}} / (A_s \cdot (T_s - T_1)) \quad \dots(19)$$

$$Nu = (h_{\text{equivalent}} \cdot L_c) / K_a \quad \dots(20)$$

Where:

$L_c$  is the characteristic length. It is determined using the following relation.

$$L_c = A_s / P \quad \dots(21)$$

#### Sample of calculation:

The Nusselt number is calculated by taking the model of circular cylinder at  $y = 5.8$  cm, the value of (h) is obtained

$$h = (m \cdot c_p \cdot M) / A_1$$

$$h = (0.0875 \cdot 380 \cdot 0.00938) / 0.003925 = 79.46 \text{ J/m}^2 \cdot \text{s} \cdot \text{K}$$

$$\text{The } Nu = (h \cdot D_h) / k_a = (79.46 \cdot 0.01) / 0.027 = 29.43$$

This procedure is used for square and triangular cylinders for all cases.

## 4. Result and Discussion

Figure (1) shows the change in the Nusselt number with Reynolds number for circular cylinder. Nusselt number increases with increasing Reynolds number. The values of Nusselt number increases with distance vertical to the section of the test (Y) where the values of Nusselt number when ( $y = 5.8$  cm) are larger than those at ( $y = 0.5$  cm). Figure (2) shows the variation of Nusselt number with Reynolds number for triangular cylinder it can be seen that Nusselt number increases as Reynolds number increases and also as (Y) increases. A similar effect can be seen in Figure (3) for the square cylinder. However the reason of this is the separation of flow behind these bodies which increases the heat transfer due to the mixing effect inside the generated vortices and turbulent boundary layer development.

Figure (4) shows the effect of changing the face of the triangular cylinder on flow and heat transfer coefficient and thus on the Nusselt number. Nusselt numbers around the perimeter of the cylinders are observed to decrease at the beginning up to the separation points and then increase in the transition regime up to the turbulent limit where they decrease again [9]. Overall Nusselt numbers are correlated with the Reynolds numbers for the two positions of the cylinders in cross flow using the side length of the triangular cylinders. On the other hand, the overall Nusselt numbers are correlated with Reynolds number when the length of the cylinders is used as a characteristic length. Comparisons with circular and square cylinders in cross flow of air show that using triangular cylinder enhances the heat transfer at large Reynolds number.

Figure (5) shows the effect of changing the face of square cylinder on Nusselt number, It can be seen that when the square cylinder is facing the flow with its vertex the Nusselt number is large than the other cases where the flow hit the side of the square. It was observed that the existence of a square cylinder makes the attached wall boundary layer separate, with a subsequent recirculation zone downstream of the cylinder. The Nusselt number distribution on the wall of the square cylinder shows an increase, with a relative maximum, slightly downstream of the reattachment point. Heat transfer from the wall of the square cylinder increases with increasing Reynolds number. A correlation was obtained for the variation of the mean total Nusselt number with the Reynolds number.

Figure (6) shows that the heat transfer coefficient from forced convection and thus the Nusselt number of triangular cylinder is better than the cylinder of square and circular shape. This is due to flow separated behind the triangle shape is stronger than the other cylinder.

As for free convection Figure (7) shows the effect of face direction on Nusselt number for square cylinder at free convection that the square cylinder (tilted angle  $45^{\circ}$ ) has better heat transfer from the normally positioned square cylinder because the buoyancy forces generated by the base of  $45^{\circ}$  surface ( $\blacklozenge$ ) was found to have more effect than that of a horizontal surface ( $\blacksquare$ ).

Figure (8) shows the impact of changing the face of the triangular cylinder on Nusselt number for free convection. When the triangle cylinder faces down ward ( $\blacktriangledown$ ) the heat transfer coefficient is better than the Triangle cylinder that faces right ( $\blacktriangleright$ ) and faces up ward ( $\blacktriangle$ ) because of the buoyancy effect.

A comparison between the three forms of the cylinder (circular, triangular and square) of free convection is given in Figure (9). The heat transfer coefficient of the square cylinder is better than the that of triangular cylinder and circular cylinder. This may be attributed to the buoyancy forces developed by the heated surfaces of each of the three cylinders. The friction forces that retard the buoyancy forces are the smallest for the  $45^{\circ}$  tilted square cylinder.

The relationship between the experimental Nusselt number, Reynolds number and Prandtl number were obtained from drawing the relation between  $\ln(Nu/Pr^{0.3})$  &  $(\ln Re)$  of forced convection as shown in Figure (10) of the circular cylinder for Reynolds numbers ranging (4555-18222). The empirical relation obtained for

Nusselt number with Reynolds number and Prandtl number was as follows:

$$Nu = 0.247 Re^{0.4381} Pr^{0.3} \quad \text{at } y = 0.5 \text{ cm}$$

$$Nu = 0.2880 Re^{0.5406} Pr^{0.3} \quad \text{at } y = 5.8 \text{ cm}$$

As well as for the case of the square cylinder in Figure (11), the empirical relationships were:

$$Nu = 0.1346 Re^{0.375} Pr^{0.3} \quad \text{at } y = 0.5 \text{ cm}$$

$$Nu = 0.151 Re^{0.5825} Pr^{0.3} \quad \text{at } y = 5.8 \text{ cm}$$

For the square cylinder (tilted angle of  $45^{\circ}$ ) in Figure (12), the empirical relationships were as follow:

$$Nu = 0.275 Re^{0.569} Pr^{0.3} \quad \text{at } y = 5.8 \text{ cm}$$

For the triangular cylinder the empirical relationship as shown in Figure (13) & (14) is:

$$Nu = 0.4168 Re^{0.6748} Pr^{0.3} \quad \text{at } y = 5.8 \text{ cm} (\blacktriangle)$$

$$Nu = 0.472 Re^{0.587} Pr^{0.3} \quad \text{at } y = 5.8 \text{ cm} (\blacktriangleright)$$

These relations were found to be in good comparison with other researchers such as those given by Zukauskas and Jakob [11] whose equations are:

$$\text{For circular cylinder (4000-40000) } Re \ Nu = 0.1945 Re^{0.592} Pr^{0.3}$$

$$\text{For square cylinder (4000-40000) } Re \ Nu = 0.102 Re^{0.674} Pr^{0.3}$$

$$\text{For square cylinder (tilted angle } 45^{\circ}) \text{ (4000-40000) } Re \ Nu = 0.246 Re^{0.588} Pr^{0.3}$$

It was found that the concordance between the equations derived from this research and the equations obtained from the source is as shown in table (2):

**Table 2,**

Type of cylinder	Nu Present case	Nu Zukauskas and Jakob [11]	Difference %
circular cylinder	33.174	35.532	6.6 %
square cylinder	34.885	36.854	5.3 %
Square cylinder (tilted angle $45^{\circ}$ )	55.866	59.8	6.5 %

Figures (15-17) show the variation of surface temperature of the cylinder with time for circular, triangular and square cylinders, respectively. The higher Reynolds number is the less time is required to lower the temperature of the surface of

the cylinder due to the speed increase of air flowing, so more heat is removed from the specimen.

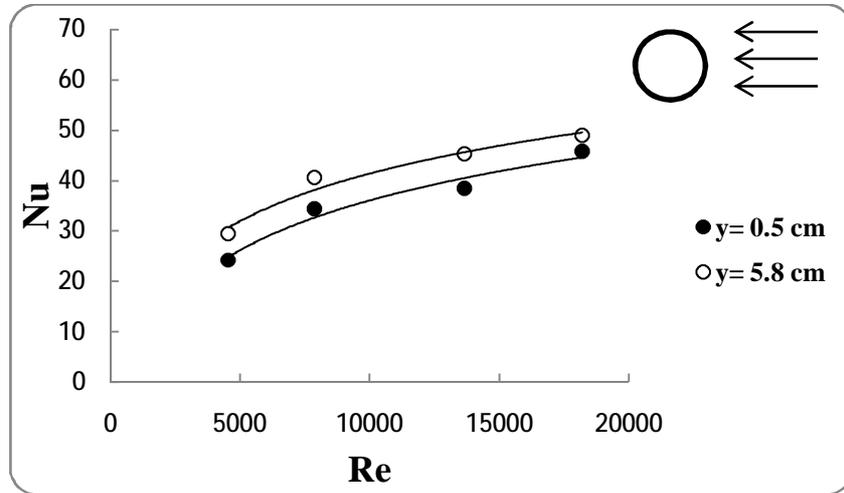


Fig. 1. Nusselt Number Variation with Reynolds Number for Circular Cylinder of Forced Convection Case.

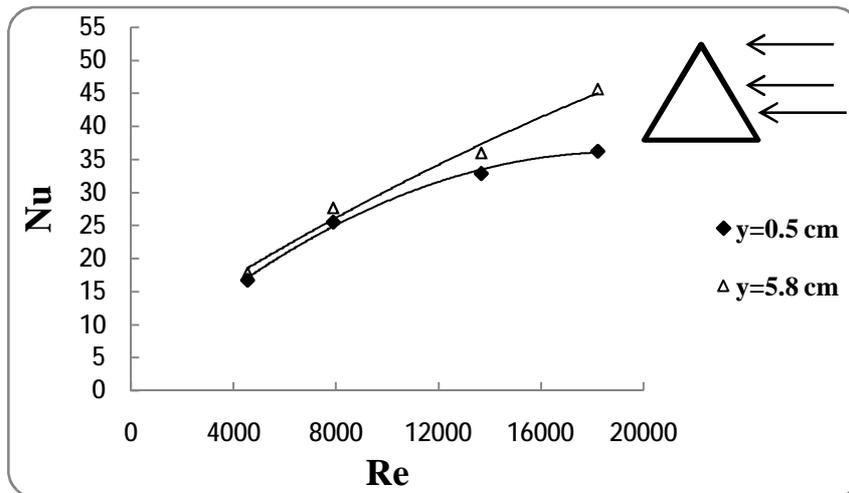


Fig. 2. Nusselt Number Variation with Reynolds Number for Triangular Cylinder of Forced Convection Case.

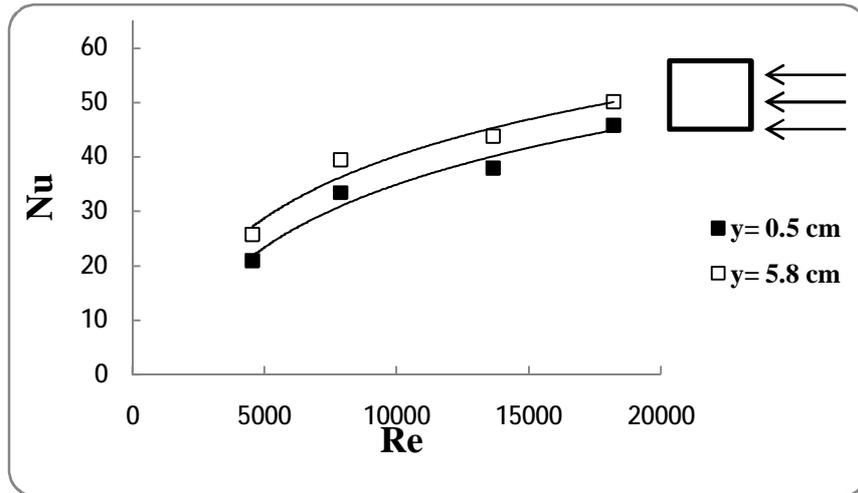


Fig. 3. Variation Nusselt Number with Reynolds Number of Square Cylinder of Forced Convection.

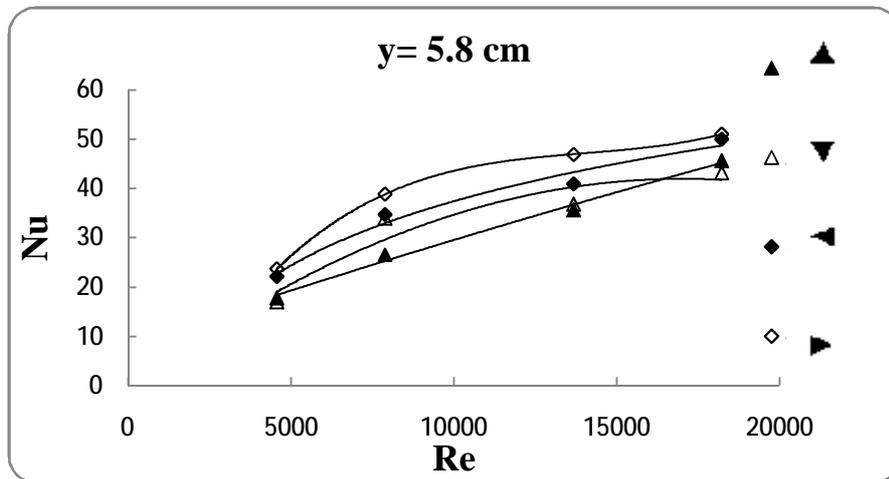


Fig. 4. The Impact of Changing the form of the Cylinder on a Number of the Cylinder Nusselt Triangular Forced Convection.

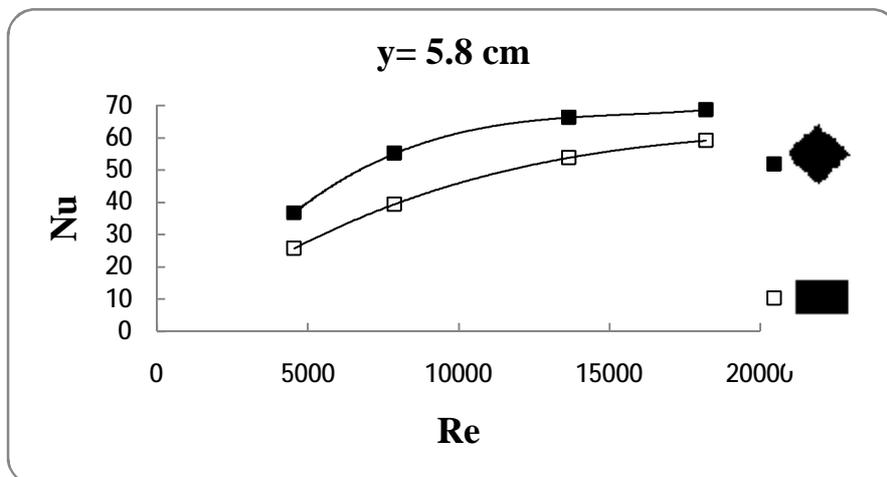


Fig. 5. The Impact of Changing the Face of Square Cylinder on Nusselt Number for Forced Convection.

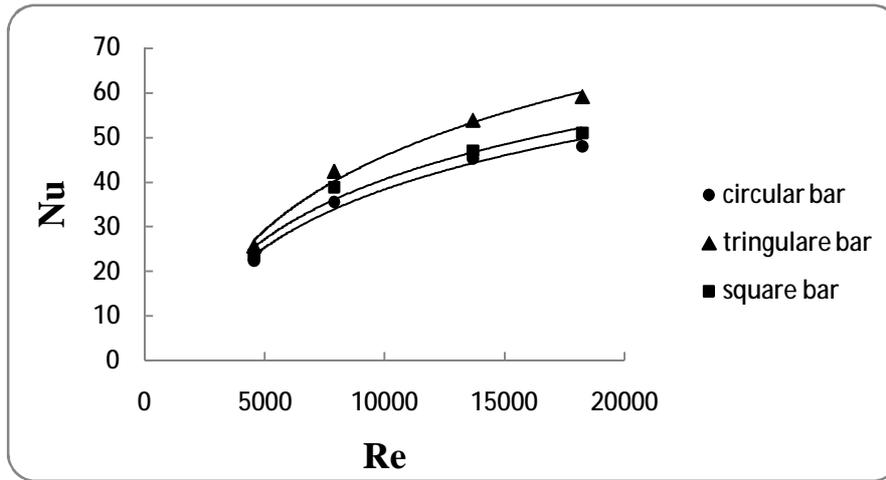


Fig. 6. The Impact of Changing the Face of Cylinder on Nusselt Number for Forced Convection at  $y = (5.8 \text{ cm})$ .

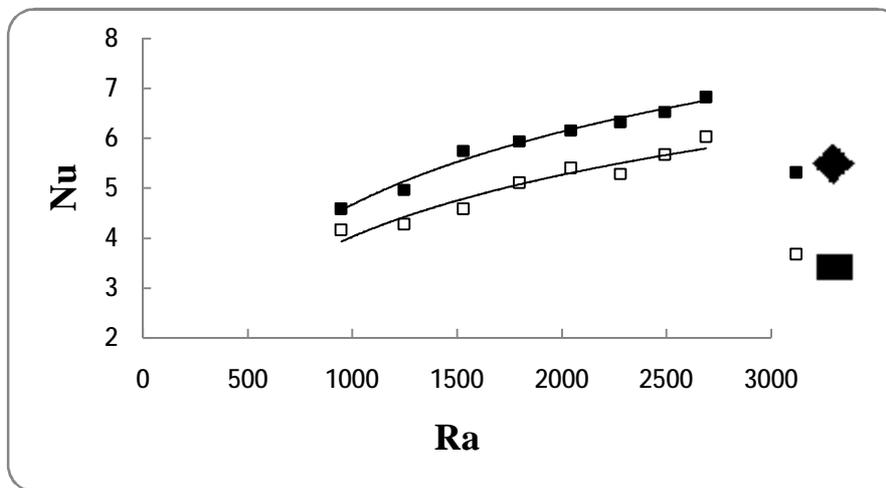


Fig. 7. The Impact of Changing the Face of the Square Cylinder on Nusselt Number for Free Convection at ( $y=5.8 \text{ Cm}$ ).

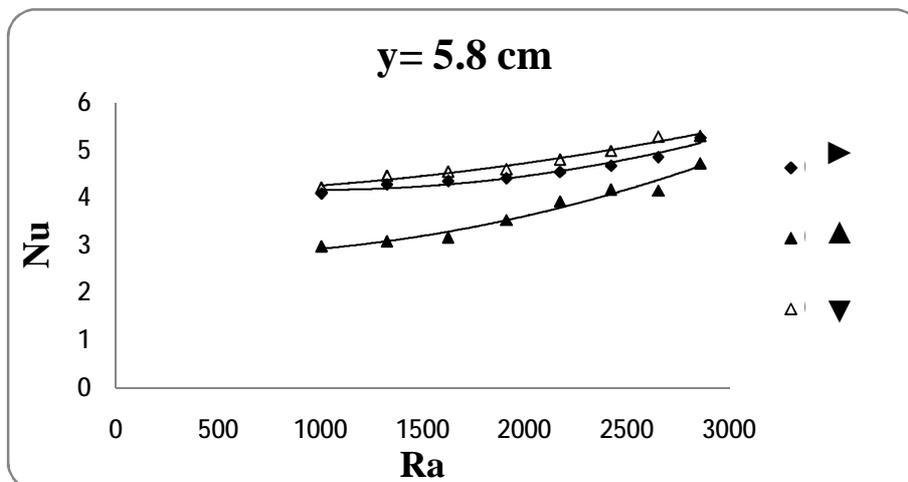


Fig. 8. The Impact of Changing the Face on Nusselt Number for Triangular Cylinder at Free Convection at ( $y=5.8 \text{ cm}$ ).

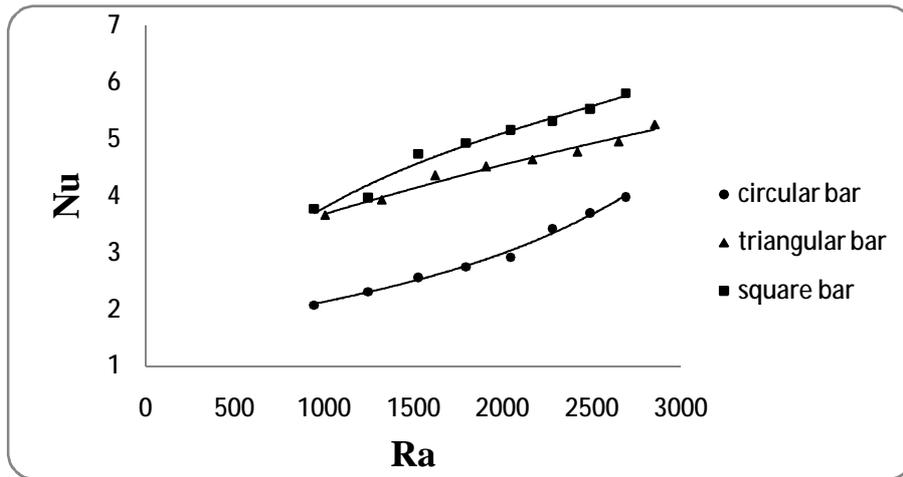


Fig. 9. The Impact of Changing the Face of Cylinder on Nusselt Number for Free Convection at  $y = (5.8 \text{ cm})$ .

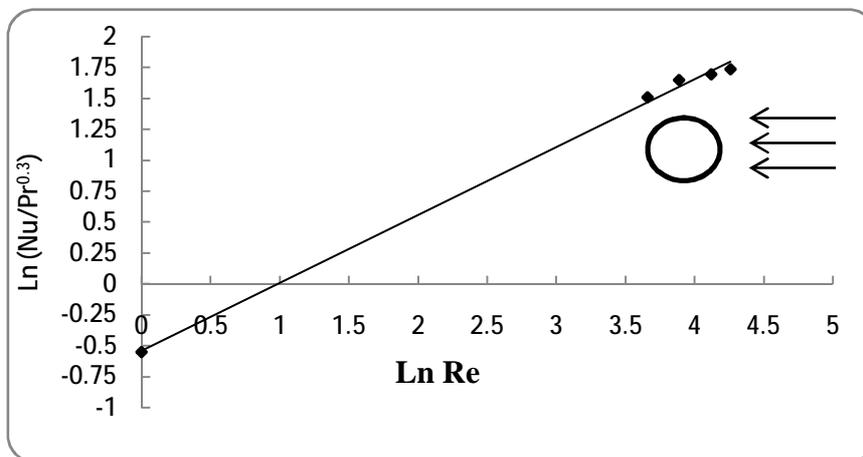


Fig. 10. The Empirical Relationship between the  $(\ln \text{Re})$  and  $(\ln \text{Nu}/\text{Pr}^{0.3})$  of Forced Convection upon Circular Cylinder ( $y = 5.8 \text{ cm}$ ).

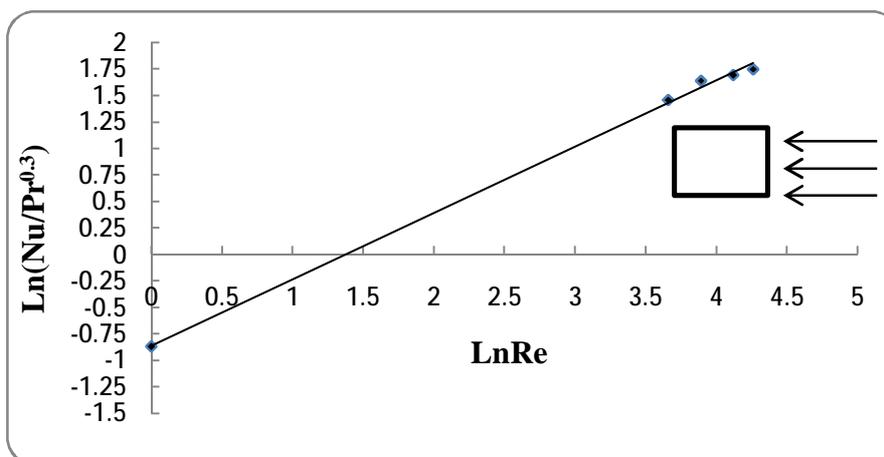


Fig. 11. The Empirical Relationship between the  $(\ln \text{Re})$  and  $(\ln \text{Nu}/\text{Pr}^{0.3})$  of Forced Convection upon Square Cylinder ( $y = 5.8 \text{ cm}$ ).

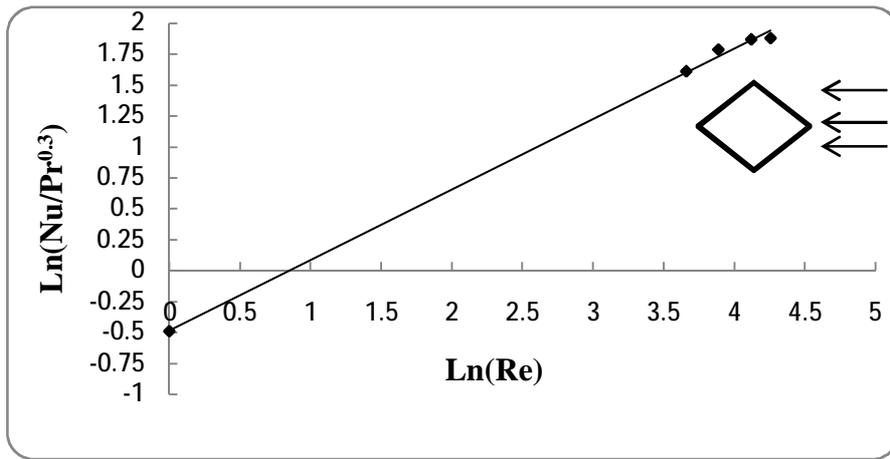


Fig. 12. The Empirical Relationship between the ( $\ln Re$ ) and ( $\ln Nu/Pr^{0.3}$ ) of Forced Convection upon Square Cylinder ( $y= 5.8$  cm).

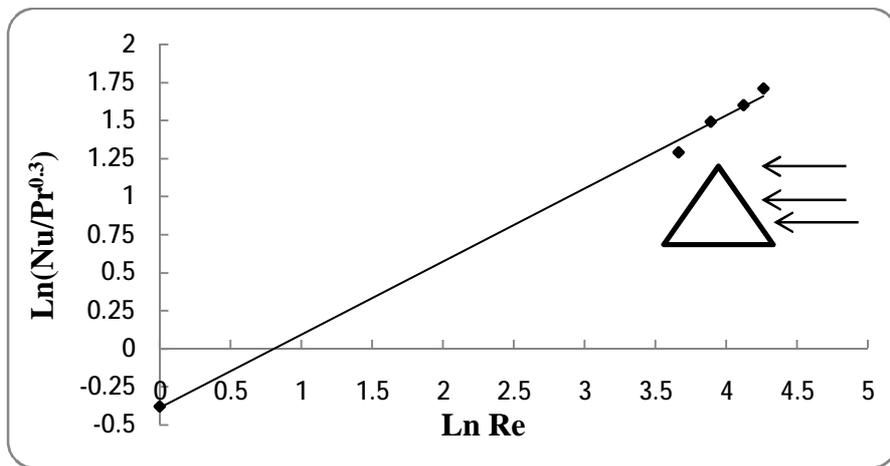


Fig. 13. The Empirical Relationship between the ( $\ln Re$ ) and ( $\ln Nu/Pr^{0.3}$ ) of Forced Convection upon Triangular Cylinder ( $y= 5.8$  cm).

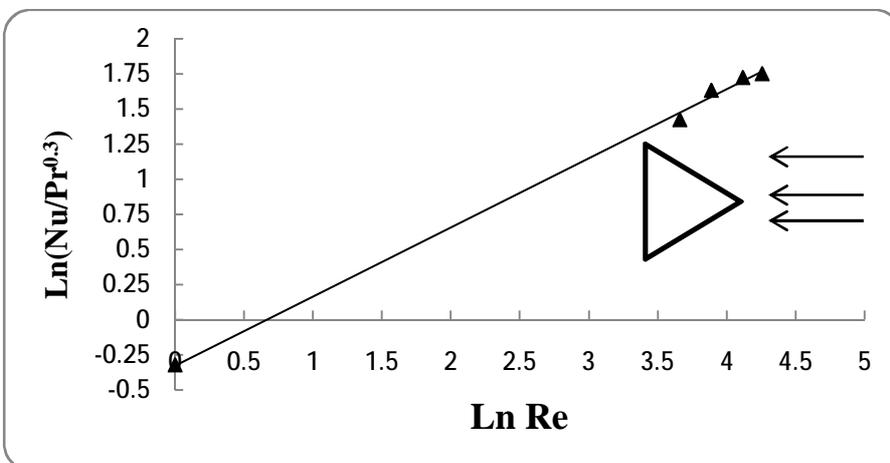


Fig. 14. The Empirical Relationship between the ( $\ln Re$ ) and ( $\ln Nu/Pr^{0.3}$ ) of Forced Convection upon Triangular cylinder ( $y= 5.8$  cm).

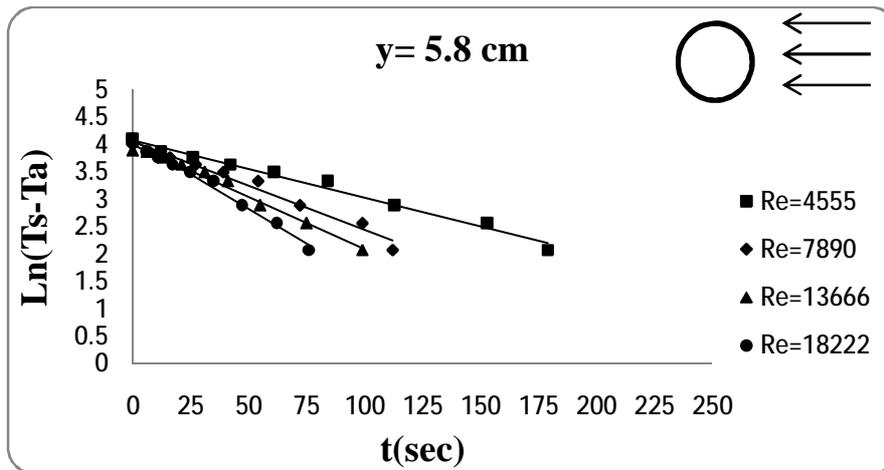


Fig. 15. The Impact of Changing Reynolds Number on Time for Circular Cylinder at ( $y = 5.8\text{cm}$ ) of Forced Convection.

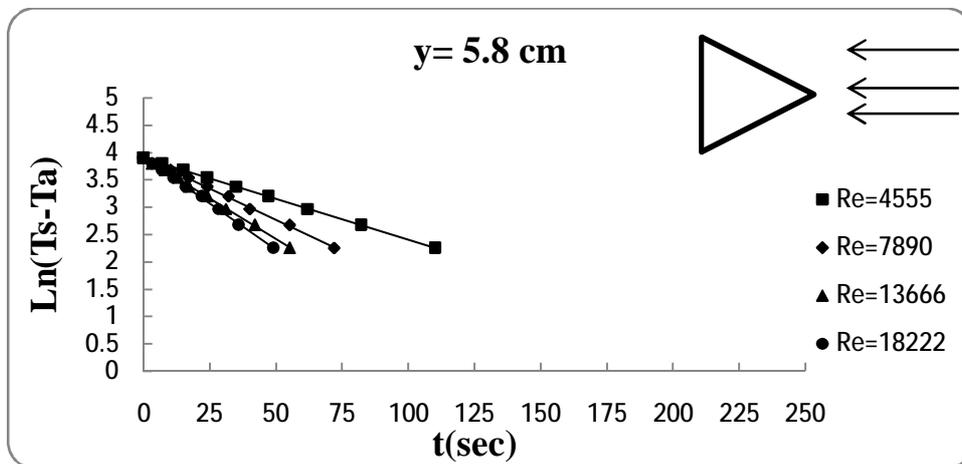


Fig. 16. The Impact of Changing the Reynolds Number on the Time of Low Temperature Triangular Cylinder at ( $y = 5.8\text{cm}$ ) of Forced Convection.

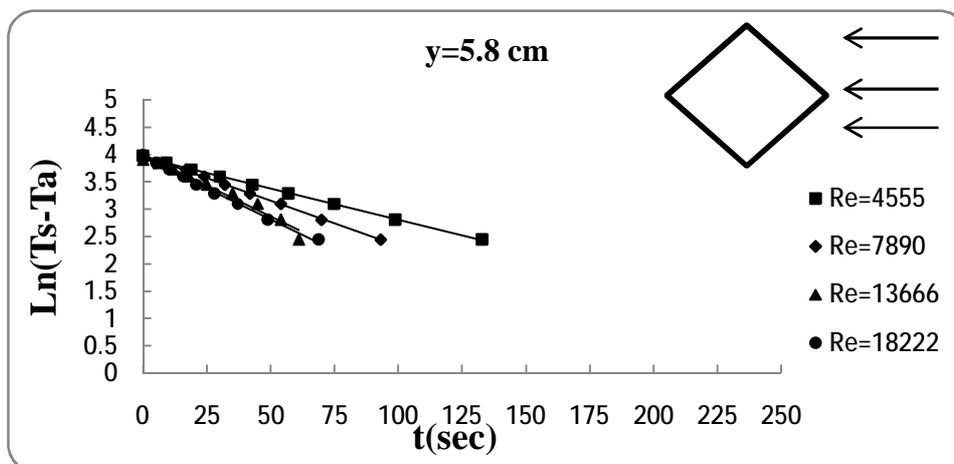


Fig. 17. The Impact of Changing the Reynolds Number on the Time of Low Temperature Square Cylinder at ( $y = 5.8\text{cm}$ ) of Forced Convection.

## 5. Conclusion

Experimental study was made on free and forced heat transfer from three cylinders of different cross-sections circular, triangular and square in cross flow of air. The three cylinders were manufactured from copper for its high conductivity. The three cylinders were made to have equal surface area to compare the effect of their shape on heat transfer coefficient. It was found that in the case of forced convection, the heat transfer of the triangular cylinder is better than that of the square and circular cylinder. For the triangular cylinder it was found ,the best case of heat transfer was when the triangle cylinder faces right ( $\blacktriangleright$ ). The square cylinder (tilted angle  $45^\circ$ ) has better heat transfer than the square cylinder has when it lies on one of its width. In the case of free convection the best position of the triangular cylinder was found where the triangle cylinder faced down ward ( $\blacktriangledown$ ). Several empirical relationships were obtained for the case of forced convection, for all cylindrical shapes, as well as for free convection. These relations were found to be in good comparison with those of others such as those of Zukauskas and Jakob [11].

### List of symbols

Symbol	Meaning of the symbol	Unit
$A_1$	effective surface area of cylinder	$m^2$
$A_s$	the surface area of heat transfer	$m^2$
$A$	the cross section area	$m^2$
$cp$	specific heat of copper cylinder	J/kg K
$cp_a$	specific heat of the air	J/kg K
$D_h$	is the hydraulic diameter	M
$D$	outside diameter of circular cylinder	M
$h$	coefficient of heat transfer	$J/m^2s K$
$H_w$	head tube connected to an inclined water manometer	M
$k_a$	thermal conductivity of air	J/ms K
$L_1$	Side length of square cylinder	M
$L_2$	Side length of triangular cylinder	M
$m$	mass of cylinder	Kg
$P$	Perimeter of duct	M
$q^0$	rate of heat loss	J/s
$T$	temperature of cylinder	K

$T_a$	temperature of air	K
$T_1$	Surrounded temperature	K
$u$	mean velocity of air	m/sec

### List of Greek letters

Greek letters	Meaning of the Greek letters	Unit
$\rho_a$	density of air	$kg/m^3$
$\rho_w$	density of water	$kg/m^3$
$\mu_a$	viscosity of air	kg/ms
$\epsilon$	is the emissivity of the surface	
$\sigma$	is the Stefan–Boltzmann constant= $5.67 \times 10^{-8}$	$W/m^2 .K^4$

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## دراسة عملية لتأثير تغيير الشكل الخارجي للجسم على المعدل الزمني لمعامل انتقال حرارة للحمل الحر والقصري خلال عملية التبريد

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### الخلاصة

تم في هذا البحث اجراء دراسة عملية عن تأثير الشكل الخارجي لجسم اسطواني على معامل انتقال الحرارة الناتج عن الحمل القصري والحمل الحر عند وضع هذا الجسم داخل مجرى هوائي وكذلك لبيان تأثير تغير موقع الجسم داخل مجرى الهواء وسرعة الهواء على قيم معامل انتقال الحرارة بالحمل القصري والحمل الحر مع تغير الشكل الخارجي . حيث تم تصنيع اسطوانات (دائرية ومثلثة و مربعة المقطع ) من معدن النحاس لموصليته الحرارية العالية بابعاد مناسبة مع المحافظة على ثبوت المساحة السطحية لجميع الاشكال وتسخينها الى درجة حرارة معينة ووضعها داخل مجرى الهواء بمواقع محددة ومراقبة انخفاض درجة حرارة الاسطوانة وتم حساب معامل انتقال الحرارة بالحمل القصري لقيم عدد رينولدز تتراوح بين (18222-4555) وبيان تأثير تغير السرعة على معامل انتقال الحرارة بالحمل القصري واجريت كذلك دراسة الحمل الحر وبيان تأثيره على معامل انتقال الحرارة لقيم عدد راييلي تتراوح بين (1069.320-3321.682). وتم الحصول على علاقة تجريبية في حالة الحمل القصري لجميع اشكال الاسطوانة وتم مقارنتها مع المعادلات التجريبية الموجودة في مصادر الاشكال الاسطوانية الدائرية وكانت قيم الثوابت مقاربة بشكل جيد للقيم الموجودة في المعادلات التجريبية في المصادر الاخرى.