The Effect of Fins Geometry on Longitudinal Trihedron Cylinder Forced Convection Heat Transfer Coefficient

Dr.Abbas Jassem Jubear, Department of Mechanical Engineering, University of Wassit,
Email:Abbaskut72@yahoo.com.au

Received 30 October 2014 Accepted 13 January 2015

ABSTRACT

This study aims to perform an experimental study of heat transfer by forced convection for three cylinders (with 12 triangular, rectangular shape fins and without fins) made of aluminum. The inner and outer diameters of these cylinders considered in this study ware (16 mm) and (48 mm) respectively .The study has been performed for different,geometry, slope angles ($0° - 45°$) and range of air velocities (0.5-4m/s). The three cylinders ware heated by a constant heat flux generated by passing an alternative current (AC) through a resistance placed inside the internal space for each cylinder . The effects of air velocity , inclined angle and geometry of fins have been studied in this study for a range of heat fluxs (13-247) W. The results show that the rate of heat transfer increases as the air velocity increases but it decreases with increasing inclined angle. The heat transfer coefficient increases generally with increasing Reynolds number (Re) and the heat transferred from triangular test sample (12) fins is greater than that from rectangular test sample and without fin type. An empirical relationships between Reynolds number (Re) and Nusselt number (Nu) is concluded:

- Test sample (12) fins with triangular section : $\mathrm{Nu} = 0.02474\ \mathrm{Re}^{0.939}$
- Test sample (12) fins with rectangular section: $\mathrm{Nu} = 0.047\ \mathrm{Re}^{0.8353}$
- Test sample without fins $\mathrm{Nu} = 0.06\ \mathrm{Re}^{0.71}$

Keywords: Fins, Force convection, Heat transfer, fin tube, fin geometry.

تاثير شكل الزعانف على معامل انتقال الحرارة ألقسري لاسطوانة مزعنفة طولي

الخلاصة

يهدف البحث الحالي إلى إجراء دراسة عملية لانتقال الحرارة بالحمل ألقسري من اسطوانة ذات زعانف طولية مستطيلة المقطع واسطوانة ذات زعانف طولية مثلثة المقطع واسطوانة بدون زعانف مصنوعة من الألمنيوم ولمدى سرع مختلفة (0.5 - 4 م/ثا).ولمستوى فيض حراري ثابت وذلك بإمرار تيار كهربائي متناوب خلال مقاومة مثبتة داخل الحيز الداخلي لكل اسطوانة. تم وضع النموذج المستطيل المقطع بزوايا ميل مختلفة عن الأفق (0° – 45°). استعمل في هذه الدراسة نموذج الاسطوانة المزعنفة طوليا بقطر داخلي مقداره (16 ملم) وبقطر خارجي مع الزعنفة مقداره (48 ملم) , وتمت دراسة تأثير كل من سرعة الهواء وزاوية الميل والشكل الهندسي للزعانف ولمعدلات فيض تراوحت بين (13.4-247) واط. توصلت الدراسة إلى ان كمية الحرارة المنتقلة تزداد بزيادة سرعة جريان الهواء ولجميع نماذج الاختبار ووجود نقصان في كمية الحرارة المنتقلة بزيادة زاوية الميل عن الأفق. إن معامل انتقال الحرارة يزداد بشكل عام مع زيادة عدد رينولدز وان كمية الحرارة المنتقلة من نموذج اختبار (12) زعنفة

مثلثة المقطع اعلى من النموذج المستطبل المقطع على الرغم من زيادة المساحة السطحية. تم استنباط علاقة تجريبية تربط كل من عدد رينولدز(Re) وعدد نسلت (Nu) ولنموذجي الاختبار كما يلي:

- نموذج اختبار (12) زعنفة مثلثة المقطع : $Nu = 0.02474\ Re^{0.939}$
- نموذج اختبار (12) زعنفة مستطيلة المقطع : $Nu = 0.047\ Re^{0.8353}$
- نموذج اختبار بدون زعانف $Nu = 0.06\ Re^{0.71}$

Nomenclature

Symbol	Explanation	Unit
A_{cy}	Un finned area of the cylinder.	m^2
A_f	Total area of the fins.	m^2
A_{scl}	Side area of fin has different section.	m^2
A_{si}	Inner surface area of cylinder.	m^2
A_t	Total area of cylinder.	m^2
C_P	Specific heat at constant pressure.	kJ/kg.K
D_P	Diameter of cylinder at the fin base.	m
D_i	Inner diameter of cylinder.	m
D_{oc}	Outer diameter of cylinder.	m
Gr	Grashof number.	_
h	Convection heat transfer coefficient.	W/m^2K
I	Electrical current.	Amp.
K	Conduction heat transfer coefficient of air.	W/m.K
K_f	Conduction heat transfer coefficient of aluminum.	W/m.K
L	The length of fin.	m
L_S	Inclined length of the different cylinder.	m
L_f	Height of fin.	m
n	Number of thermocouple position along the finned	_
N	Number of fins.	_
Nu	Nusselt number.	_
Pr	Prandtl number	_
Q	Heat flux.	W/m^2
$Q_{conv.}$	Heat transferred by forced convection.	W
$Q_{gen.}$	Generated heat by electrical current.	W
$Q_{rad.}$	Radiation heat transfer.	W
R_a	Rayleigh number.	_
R_e	Reynolds number.	_
S_{sur}	The shape factor equal (1).	_
T_{air}	Ambient air temperature.	K
T_f	Average film temperature.	K
T_{sav}	Average surface temperature of cylinder.	K
t_1	Fin thickness at the top.	m
t_2	Fin thickness at the base.	m
U	Maximum air velocity inside the duct.	m/s
V	Voltage.	Volt

Greek symbols

Symbol	Explanation	Unit
ε	Emissivity surface factor and equal (0.04).	-
θ	Inclined angle of cylinder.	degres
μ	Dynamic viscosity .	kg/m.s
ν	Kinematic viscosity .	m^2/s
π	The constant ratio and equal (3.14).	_
σ	Stefan Boltzmann constant.	_

1- INTRODUCTION

Finned heat sinks are commonly used devices for enhancing heat transfer from central air-conditioning devices**. Kenneths et al., 2001,** air-cooled engines of motor cycles and automobiles , tubes of liquid-gas heat exchangers used in the refrigeration industry. **Ravi et al. 2008**, solar collectors , nuclear reactor, cooling of electrical and electronic components and under ground electric transmission cables using pressurized gas and others**, Sakr et al., 2008**. When the principles of heat transfer are used to used to design important engineering equipment such as heat exchanger. The designer work for important aim which is the developing the production to improve the economy and reducing the consumption of energy . For all that, economy with other technical specification play an important role in design and selecting the equipment of heat exchanger .The designer must take into account the weight and volume of heat exchangers which are specially used in aerospace and aeronautics and in this case will be a secondary factor. Other factors such as efficiency and volume of heat exchanger are also considered important economic factors in these applications**, Fady R. S., 2009.** There are many studies deal with the fluid flow and heat transfer in free and forced convection from plate surface and semi-cylindrical surface in (theoretical and experimental) methods .Also there are many studies which deal with the heat transfer from the pulse and corrugated surface in the free and forced convection . In **1984, Moon S.,** presented an experimental study [Analysis of combined natural and forced convection around cylinders and spheres] using cylindrical shapes and spherical shape. The equations for free and force convection ware:

$$\mathrm{Nu} = 1.233\mathrm{Gr}^{0.1735} \qquad \text{for } 1 < \mathrm{Gr} < 10^6 \qquad (1)$$

And accuracy about (93.5%)

While the equation for forced convection was:

$$\mathrm{Nu} = 0.5422\ \mathrm{Re}^{0.5139} \qquad \text{for } 70 < \mathrm{Re} < 3200 \qquad (2)$$

Teertstra et al. 1999., performed an analytical modeling of forced convection in slotted plate fin heat sinks. An experimental measurements have been performed for a range of slot configurations, $\frac{\mathbf{s}}{\mathbf{P}}$ ~0.5, 0.059≤ $\frac{\mathbf{P}}{\mathbf{L}}$ ≤0.44,for the range of Reynolds number, 40≤Re_b^* ≤180. An approximatate model is proposed that predicts the experimental results for the average heat transfer rate within a 12% RMS difference . **Fady R.S., 2009.,** presented an experimental study of the effect of forced convection heat transfer coefficient from circumferentially finned cylinder. The results of this study showed In the case of stationary cylinder for zero degree inclination and heat flux equal to (Q=500w/m^2),the formula was

$$h = [(2 \times 10^{-3}) - (2 \times 10^{-5}\mathrm{Re})]\theta + 0.004\ \mathrm{Re} + 3.9 \tag{3}$$

and For the same constants but for oscillating cylinder was:

$$\mathrm{Nu} = 0.003 \times \mathrm{f} + 0.77 + (0.001 \times \mathrm{Re}) \tag{4}$$

Golnoosh M., 2012. Investigated the steady-state external natural convection heat transfer from vertically-mounted rectangular interrupted finned heatsinks. A systematic numerical, experimental, and analytical study is conducted on the effect of the fin array and single wall interruption. FLUENT and COMSOL Multiphysics software are used in order to develop a two-dimensional numerical model for investigation of fin interruption effects. Results show that adding interruptions to vertical rectangular fins enhances the thermal performance of fins and reduces the weight of the fin arrays, which in turn, can lead to lower manufacturing costs. **Yoav Peles et al., 2005.** Investigates heat transfer and pressure drop phenomena over a bank of micro pin fins. A simplified expression for the total thermal resistance has been derived, discussed and experimentally validated. Geometrical and thermo-hydraulic parameters affecting the total thermal resistance have been discussed. It has been found that very low thermal resistances are achievable using a pin fin heat sink. In many cases, the increase in the flow temperature results in a convection thermal resistance, which is considerably smaller than the total thermal resistance. **Kavita H. Dhanawade et al., 2014.** Using an experimental study to investigate the heat transfer enhancement over horizontal flat surface with rectangular fin arrays with lateral square and circular perforation by forced convection. The cross sectional area of the rectangular duct was 200 mm x 80 mm. The data used in performance analysis were obtained experimentally for fin arrays of material aluminum, by varying geometry and size of perforation as well as by varying Reynolds number from 21* 10^4 to 8.7* 10^4. It is observed that the Reynolds number and size perforation have a larger impact on Nusselt number for the both type of perforations. **AIEssa, et al, 2004,2008,2009,20012.** Studied the heat dissipation from a horizontal rectangular fin embedded with square perforation, rectangular perforations with aspect ratio of two, equilateral triangular perforations of bases parallel and towards its fin tip, by using finite element technique under natural convection. They compared the results of the perforated fin with its external dimensionally equivalent solid fins. They showed that perforation in the fins enhances heat dissipation rates. Also, the heat transfer of perforated fin enhances with increase in the fin thickness

2- DATA REDUCTION

The convective heat transfer rate from electrically heated test surface is calculated by using a relation. The length of finned region which is subjected to constant and steady heat flux is (300mm) and the total quantity of heat generated in the electrical heater ($Q_{gen.}$)is converted to heat. This heat transferred across the fin by conduction and from the cylinder to the ambient by forced convection ($Q_{conv.}$)in addition to the heat loss by the radiation ($Q_{rad.}$).

$$\mathrm{Q_{gen.}} = \mathrm{Q_{conv.}} + \mathrm{Q_{rad.}} \tag{5}$$

The total quantity of generated heat is calculated as follow:

$$Q_{gen.} = V \times I \tag{6}$$

The heat transferred by radiation is calculated as follows: **Yunus A., 1998.**

$$Q_{rad.} = \sigma \times \varepsilon \times S_{sur} \times A_t\left(T_{s_{av}}^4 - T_{air}^4\right) \tag{7}$$

and, the heat transferred is calculated by convection as follows:

$$Q_{conv.} = Q_{gen.} - Q_{rad.} \tag{8}$$

Therefore, the heat transfer coefficient in forced convection was calculated from the equation which is known as Newton cooling law as follows:

$$h = \frac{Q_{conv.}}{A_t\ \Delta T} \tag{9}$$

where :A_t − represents the surface area which is subjected to convection and equal to the finned area in addition to the un finned area which are taken into consideration.

$$A_t = A_f + A_{cy} \tag{10}$$

$$A_{cy} = \pi\, D_b\ L_{cor} - \pi\, D_b\ t_2\, N \tag{11}$$

The surface area was calculate for the test sample as follow:

$$A_f = A_{sd} \times 2\, N \tag{12}$$

$$A_{sd} = L_s\, P \tag{13}$$

$$p = \pi(D_b + D_{oc}) \tag{14}$$

$$L_s = \sqrt{L_{cor}^2 + \left(\frac{t_2}{2}\right)^2} \tag{15}$$

$$L_{cor} = L_f + \frac{t_1}{2} \tag{16}$$

Where:

L_{cor} is corrected fin height. (ΔT) in equation (9) represents the difference between the average temperature for the cylinder and the supplied air temperature and then the average temperature for the cylinder is calculated as follows:

$$T_{s_{av.}} = \frac{(T_1 + T_2 + \cdots + T_n)}{n} \quad (17)$$

While the average film temperature (T_f) is calculated as follows:

$$T_f = \frac{T_{s_{av.}} + T_{air}}{2} \quad (18)$$

This temperature is taken to calculate the physical properties for the working fluid (air).According to the values of temperature and physical properties taken from the tables in **J. P. Holman, 2008.** In order to calculate the heat flux exerted on the finned cylinder this requires calculation of the generated power due to passing the electrical current in a heating resistance by applying the equation (6).Where the surface area which is subjected for this power is the internal area for the finned cylinder and is calculated as follows:

$$A_{si} = \pi\, D_i\, L_c \quad (19)$$

$$Q = \frac{Q_{gen}}{Asi} \quad (20)$$

3- EXPERIMENTAL SETUP DETAILS

The aim of this study is to know the thermal behavior for a longitudinal finned cylinder with triangular ,rectangular section and cylinder without finns inside air (for in compression fluid) for a velocity range between (0.5 – 4 m/s) ,and the inclined angle was (0° and 45°), Also the steady is at constant heat flux between (13 – 247w/ m^2) which can be obtained from the device. In order to study the air velocity effect (U) and the inclined angle on the overall heat transfer coefficient and the dimensionless parameter value which represent Reynolds number (Re) and Nusselt number (Nu) , this section includes an explanation for the method of system design has been used to achieve this aim.

A suitable area for heat exchanger surface for measuring the temperature along the surface to calculate the overall heat transfer coefficient and the dimensionless parameter. Using a cylinder contains a longitudinal fins with triangular, rectangular section as shown in the **Fig.1**. The Heating was under constant heat flux condition with the possibility of heat flux change using variable transformer. The test device components include the air duct ,which has been manufactured from plywood material , has a square section (60cm×60cm) and length(2 m), the duct entrance is manufactured in quadrilateral pyramid shape. Pyramid crown contains a circular opening in order to fix the air blower. In way to prevent the existence of any obstruction inside the air duct to make the air flow easy and reduce effects of the obstruction . Also in one of the air duct walls a glass gate is manufactured at distance (30cm) from the open end of the duct . Its length (40cm) and it's high (56cm) in order to reach to the sample in case of fixing it or change the inclined angle. Also the air duct is supported by angled iron support in order to fix the air duct as shown in the **Fig.2.** Air blower which used is operating according to centrifugal principle with power (350 W). It has an opening to come at the air at circular shape with diameter (0.0635 m). Also this blower contains a gate in order to select the air quantity at the entrance point and from it we can select the velocity inside the air duct .Three positions are selected to open this gate in order to give three values of different velocities between (0.5 - 4m/s) inside the air duct.

Three test samples, made from aluminum with cylindrical shapes, longitudinal finned (12) fins and triangular, rectangular section and without finns have been used . These samples are manufactured by milling machine from one piece, with the total length about (320 mm) and the finned length about (300mm), the external diameter about (48 mm) and the fin height (13 mm). The internal diameter of the cylinder (16 mm) which is used for containing the heating mechanism under constant heat flux condition as shown in **Fig.1**.The support was manufactured from cast iron material with angle may be changed to different position (0° and 45°) **as shown in Fig.3.**

The electrical circuit consists of heating coil power is (1000 W). The heating coil was put inside a glass tube from the Pyrex and insulated by ceramics insulation at the ends. Regavolt transformer was used of type (BRISTOL CONN 15 Amp.). From it the electrical power supplied to the electrical coil can be controlled through changing the voltage with a range (20 - 100 V).Because of the changing which happens in the electrical energy from the main source , voltage stabilizer was used of type (Gold source SVC 1500Watt) to maintain the stability of supplied energy for the test device and the measurement devices . The device was of type (HEME ANALYST 2050) . The supplied electrical power on the heating coil can be measured directly . Using this device contains the voltage and current measurements can be obtained from this device and then multiplying these parameters to results in the supplied power at high accuracy .

Thermocouples were used of type **k** which were fixed on the longitudinal finned cylinder using the Epoxy material which has heat resistance and high conductivity , six thermocouples were used distributed at equal distances along the cylinder .These thermocouples were connected to the selector switch (15 switches) and then connected to the digital thermometer. The selector switch was manufactured using plastic box .Micro switch was fixed on it's face, and then the outlet points were connected to all switches with each other by thermocouple wires of type **K** and it were collected in one point . This point was connected to the temperature measurement device and when one switch is operated we can read the temperature at the selected point. Digital thermometer used in this work is of type [TM-6862] and has ability to measure the temperature for a range [40 - 1200℃]. This device is fit with the thermocouple of type **K** only . It contains two inlets for thermocouples (T_1 and T_2) and the selector switch can be changed from T_1 to T_2 as shown in. This device was used to measure the temperature from the finned cylinder surface by the thermocouples which are fixed on the finned cylinder surface ,at the fin base. Digital anemometer used in this work is of type [e.schiltknechting .sia ch-8625 gossan zh,Switzerland], shows the digital anemometer device which was used.

Basic processes are followed in the test experiments and the resulting are recorded in order to depend it on the analysis of effect of the inclined angle on the heat transfer coefficient in forced convection . The test sample and the support are fixed inside the air duct and then the test cylinder is heated according to the selecting heat flux, 13, 52, 109, 168 and 247 W/m^2 which is selected by the current and voltage and waiting for some time until reaching the steady state temperature case which take (3-4 hr) and recording the six thermocouples reading and calculating the average of it. Operating the air blower system according to the selecting velocities in this and recording the temperatures. Choosing another heat flux by changing of the input power.

4- RESULT

Lab experiments have been performed using Reynolds number $(4.16 \times 10^3 - 7.22 \times 10^4)$, heat flux $(13 - 247)$ W and inclined angle $(0°)$ for the three samples in addition to inclined angle $(45°)$ for test sample (12) rectangular fins to study the effect of inclined angle on heat transfer . Nusselt number (Nu) and Reynolds number (Re) have been calculated for all cases and the graphs has been drawn for each test sample as shown later where the results has showed the following.

The relation between Nusselt number (Nu) and Reynolds number (Re) can not be represented by a straight line and as a result logarithmic equations have been used and acceptable results have been obtained.

Test sample (12) fins rectangular and for inclined angle $(\theta = 0°)$ with different ranges of velocity:

$U = 0.5\ m/s$

$\mathrm{Y = 0.7471\ X \pm 0.7591}$

$\mathrm{U = 2.5 \frac{m}{s}}$

$\mathrm{Y = 1.1129\ X \pm 2.689}$

$\mathrm{U = 4\ m/s}$

$\mathrm{Y = 0.6458\ X \pm 0.5316}$

Test sample (12) fins triangular and for inclined angle $(\theta = 0°)$ with different ranges of velocity :

$\mathrm{U = 0.5\ m/s}$

$\mathrm{Y = -0.5972\ X \pm 0.1411}$

$\mathrm{U = 2.5\ m/s}$

$\mathrm{Y = 1.02\ X \pm 2.54}$

$\mathrm{U = 4\ m/s}$

$\mathrm{Y = 1.202\ X \pm 2.137}$

Test sample without fins rectangular and for inclined angle $(\theta = 0°)$ with different ranges of velocity:

$\mathrm{U = 0.5\ m/s}$

$\mathrm{Y = 0.4743\ X \pm 0.1035}$

$\mathrm{U = 2.5 \frac{m}{s}}$

$\mathrm{Y = 0.6379\ X \pm 0.7949}$

$U = 4\ m/s$

$Y = 1.037X \pm 2.776$

Test sample without fins and for inclined angle ($\theta = 45°$) with different ranges of velocity:

$U = 0.5\ m/s$

$Y = 0.961\ X \pm 1.4244$

$U = 2.5\ m/s$

$Y = 3.412\ X \pm 13.425$

$U = 4\ m/s$

$Y = 2.894\ X \pm 11.57$

Where :

$Y = \mathrm{Log\ Nu}$

$X = \mathrm{Log\ Re}$

From the general equation of heat transfer by external forced convection: $Nu = CRe^{n}$

The constants are obtained and the equations will be as follows:

The test sample (12) fins with rectangular, tringular section and test sample without fins was:

$Nu = 0.047\ Re^{0.8353}$

$Nu = 0.02474\ Re^{0.939}$

$Nu = 0.06\ Re^{0.71}$

5- DISCUSSION

The relation between Nusselt number (Nu) and Reynolds number (Re) for different geometry (triangular, rectangular and without fins) with inclined angle (0°) was **shown in figure (4 to 6).** It is shown that heat transfer by forced convection increases as Reynolds number increases for the three samples . Heat transfer from test sample with 12 fins (rectangular ,triangular) is higher than sample without fins due to increasing surface area and better heat transfer mixing due to the exisisting fins. Also the heat transferred from test sample 12 fins triangular shape is greater than rectangular shape because the most fins is constant thickness encountered in practice, the fin thickness t is too small relative to the fin length L, and thus the fin tip area is negligible. The fins with triangular profiles contain less material and are more efficient than the ones with rectangular profile, and thus are more suitable for application requiring minimum weight such as space application.
The effect of the different air velocity on the rate of heat transfer with inclined angle (0°) for different geometry are **show in figures (7 to 9).** Its show that the heat transfer increases with increasing the velocity of air flow because of increasing in heat transfer coefficient by forced

convection as a result heat transferred increases from fin where as air velocity is maximum , the cooling rate for fin increases.

Figures (10 to 12) show the relations between Reynolds number and Nusselt number for rectangular shape at different velocities and for inclined angles (0º and 45º). It is shown that heat transfer at inclined angle (0°) is greater than at (45°) because when the longitudinal grooves for test sample is in horizontal plane , it make the heating air to flow in uniform shape and as a result the rate of heat transfer increases . In the case of placing the sample at inclined angle (45°), the fins work as an obstacles to air flow and hence the flow will be turbulated and a thermal resistance will be added and due to all that the rate of heat losses from tube decreases.

Figure (13) shows the effect of the velocity of air flow on the rate of heat transfer for different velocities and inclined angle (45°) for rectangular test sample (12) fins .The Figure shows that the heat transfer increases with increasing the velocity of air flow because of more heat is removed by forced convection.

6- CONCLUSIONS

Based on the analysis of the practical results obtained from the lab experiments and according to the graphs related to the effects of inclined angle , heat flux and Reynolds number (Re) on heat transfer , the following can be concluded:

1- Heat transfer decreases with increasing the inclined angle.

2- Heat transfer increases with increasing Reynolds number (Re).

3- Heat transferred from triangular test sample 12 fins is greater than that from rectangular test sample and that without fin types.

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Figure (1) The Two Finned cylinder

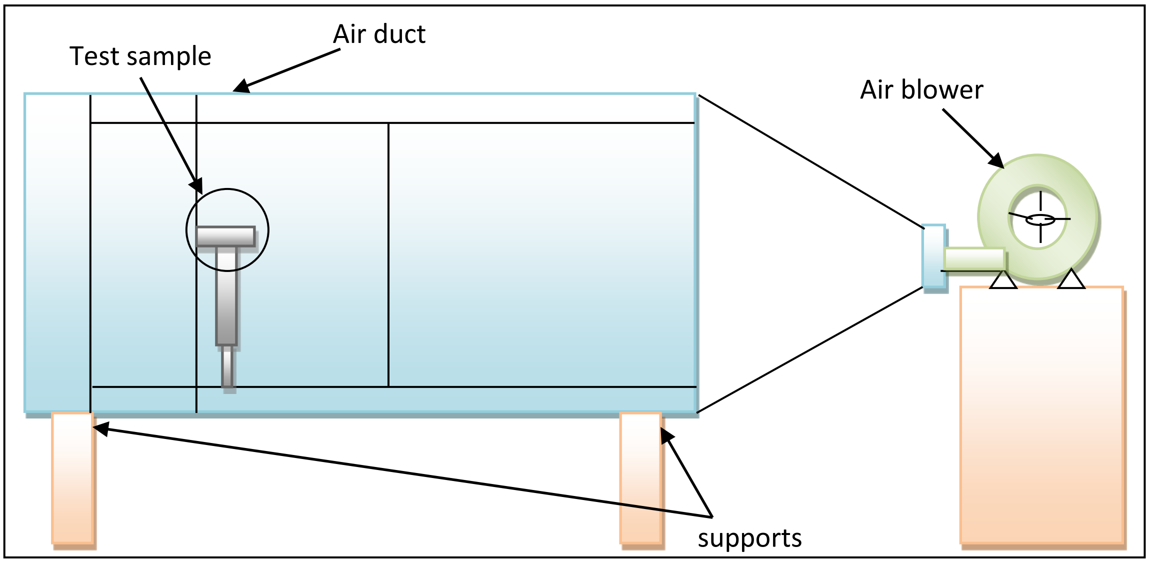

Figure (2) The Air Duct.

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Figure (3) The Support of fins

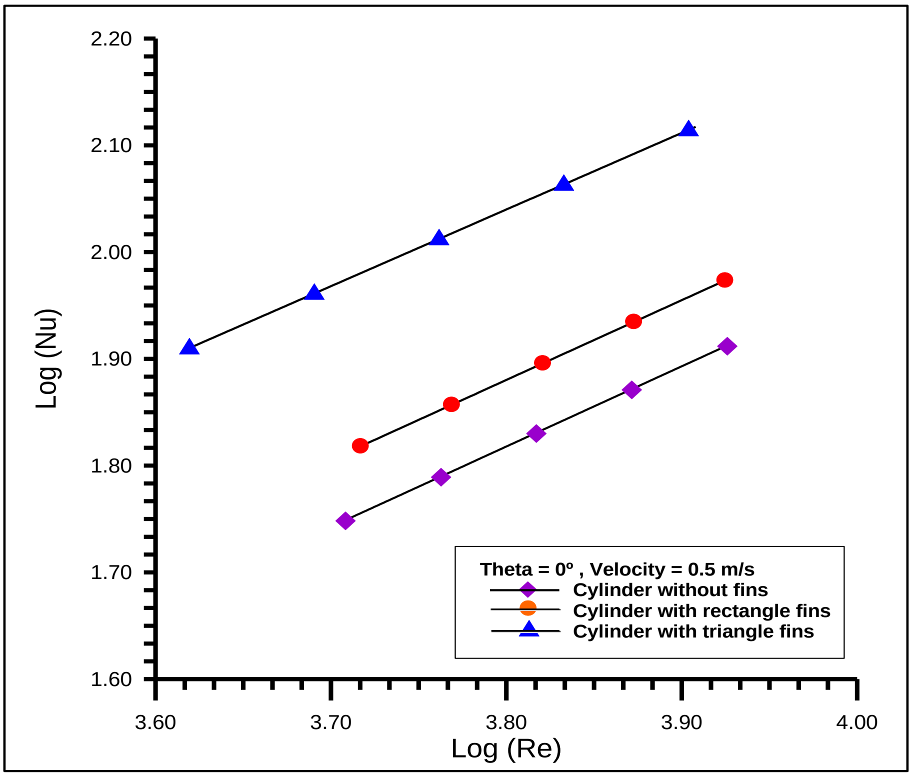


Figure (4) The effect of different geometry of fins on heat transfer rate s

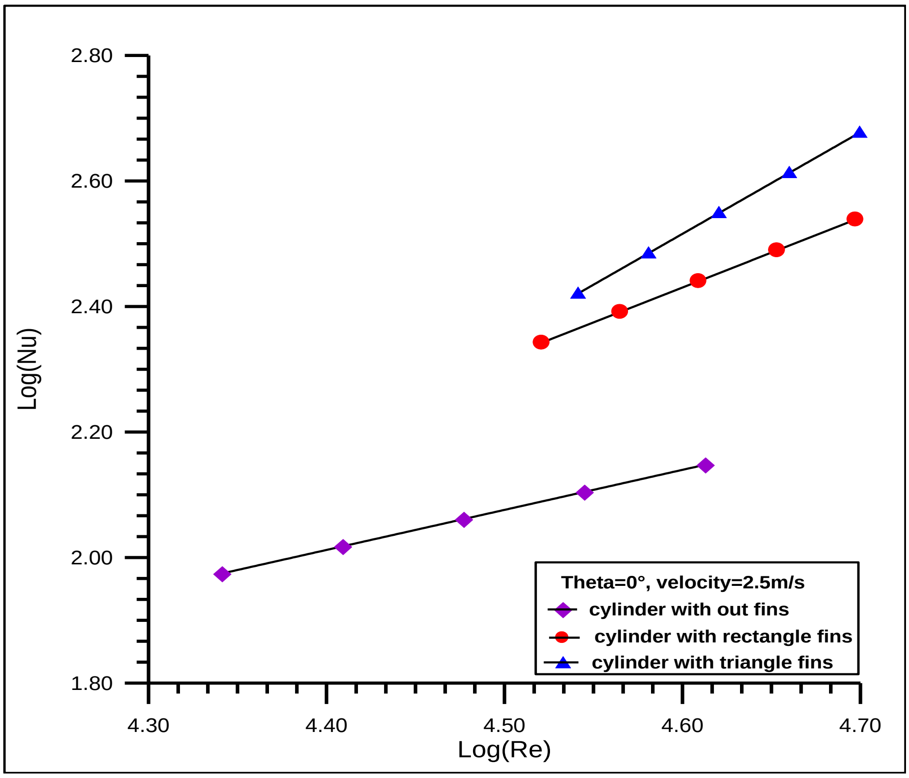


Figure (5) The effect of different geometry of fins on heat transfer rate

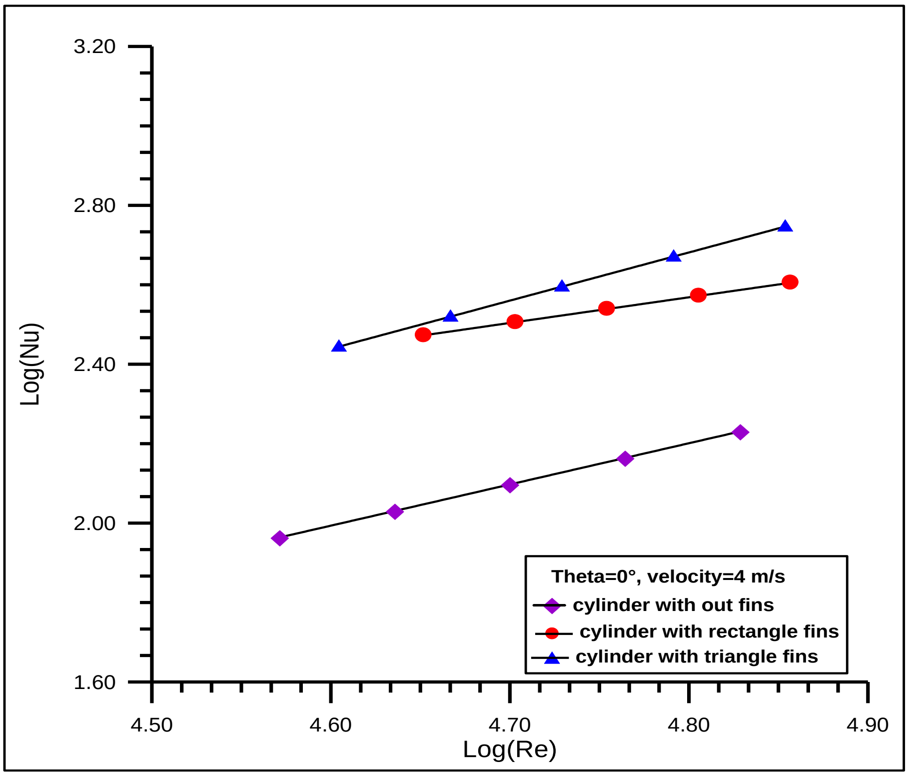


Figure (6) The effect of different geometry of fins on heat transfer rate

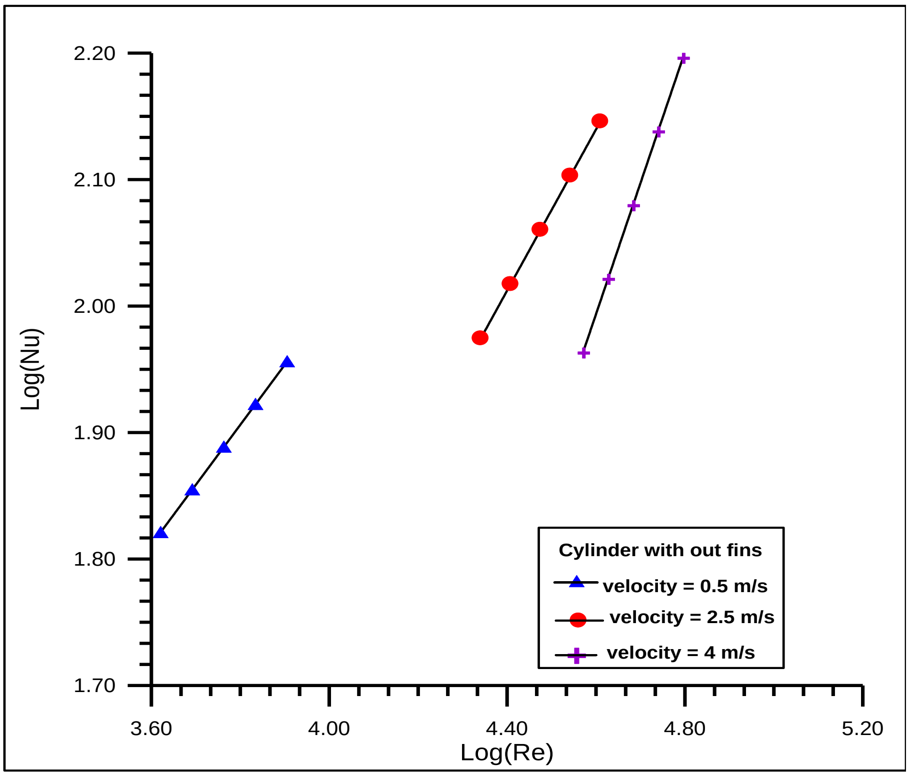


Figure (7) The effect of air flow velocity on heat transfer rate without fin test sample

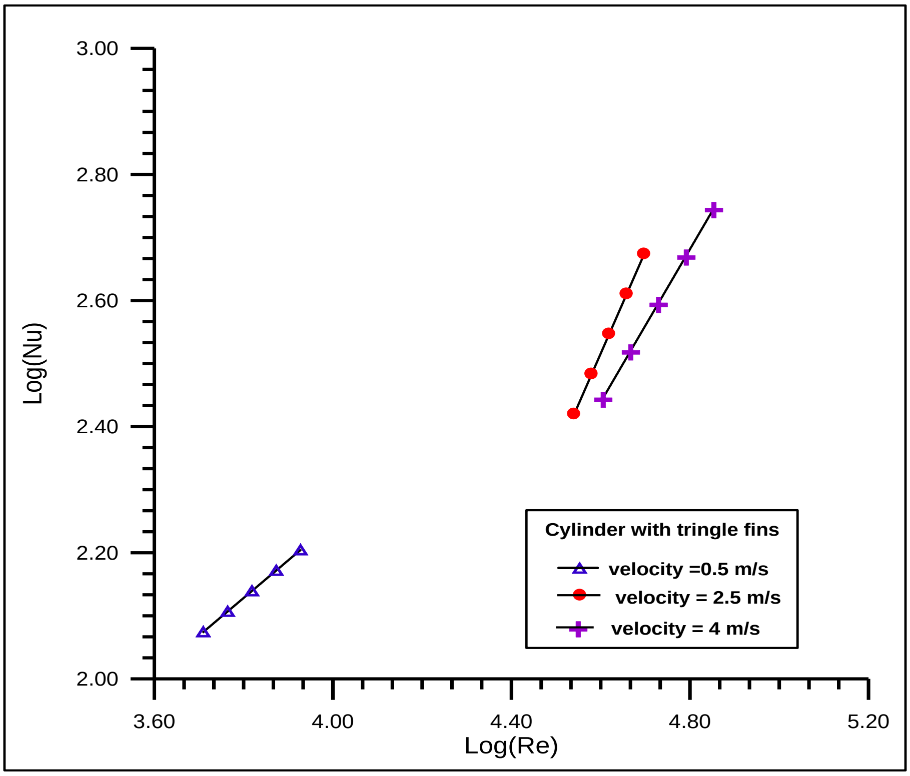


Figure (8) The effect of air flow velocity on heat transfer rate triangular test sample

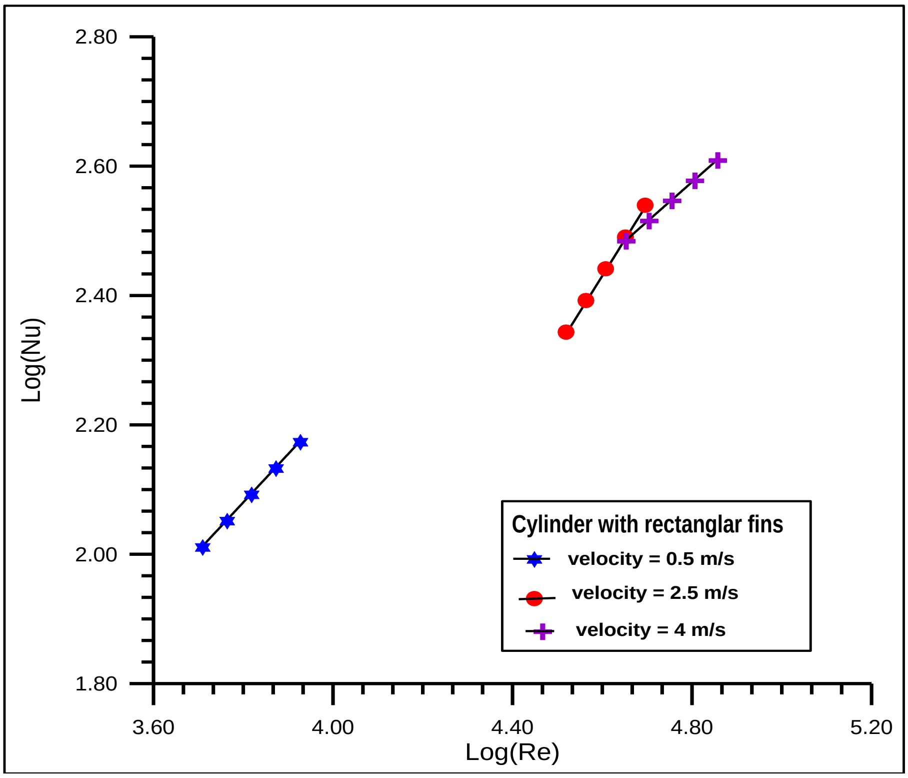


Figure (9) The effect of air flow velocity on heat transfer rate rectangular test sample

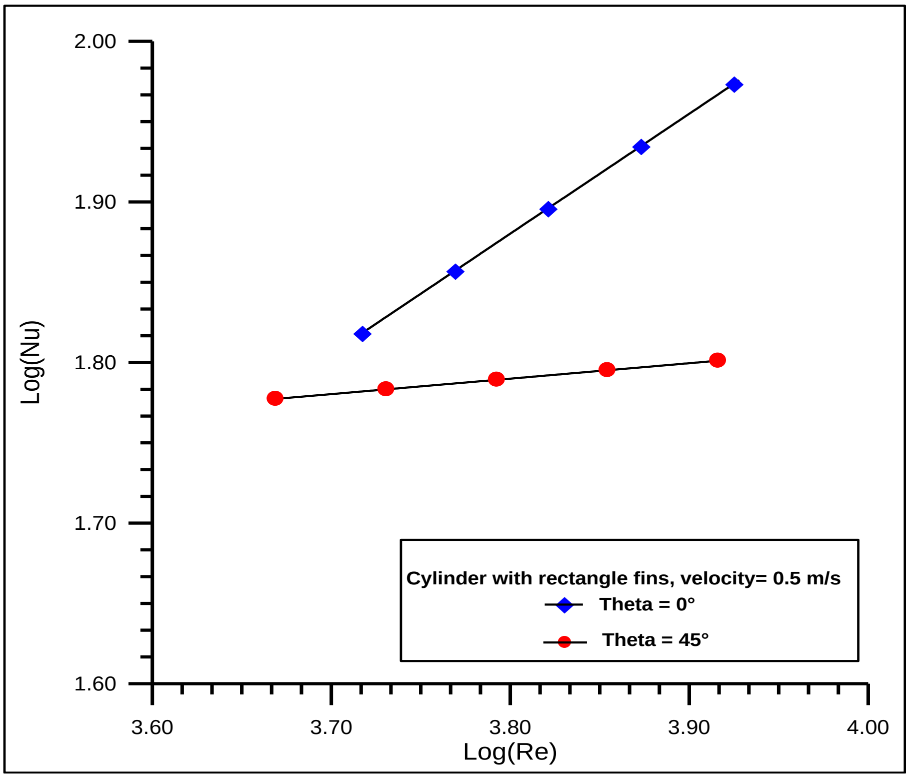


Figure (10) The effect of inclined angle on heat transfer rate

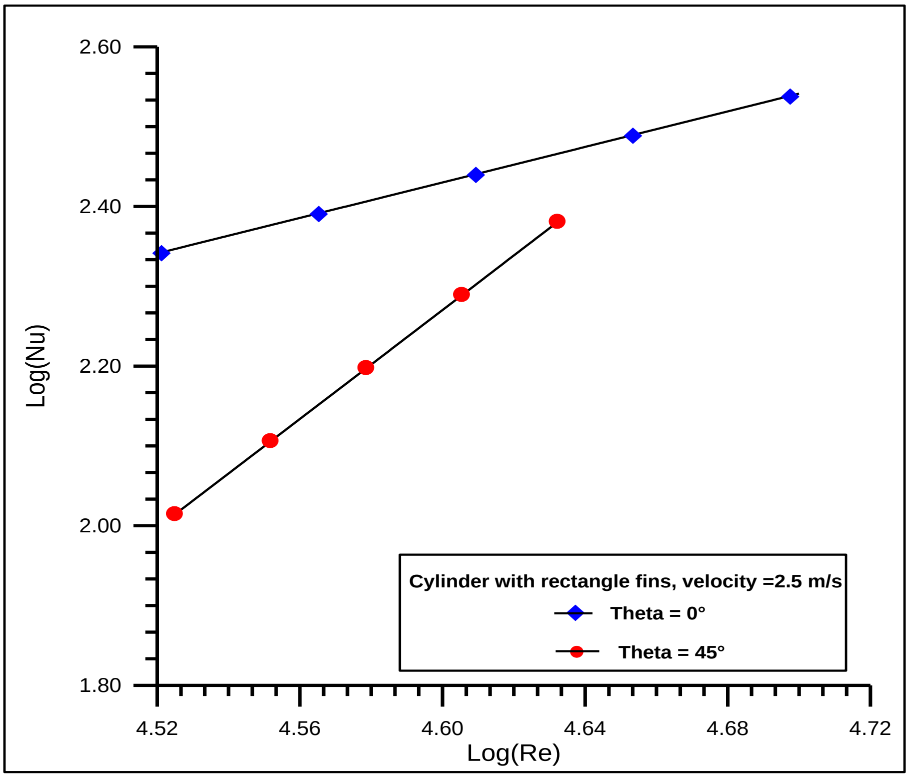


Figure (11) The effect of inclined angle on heat transfer rate

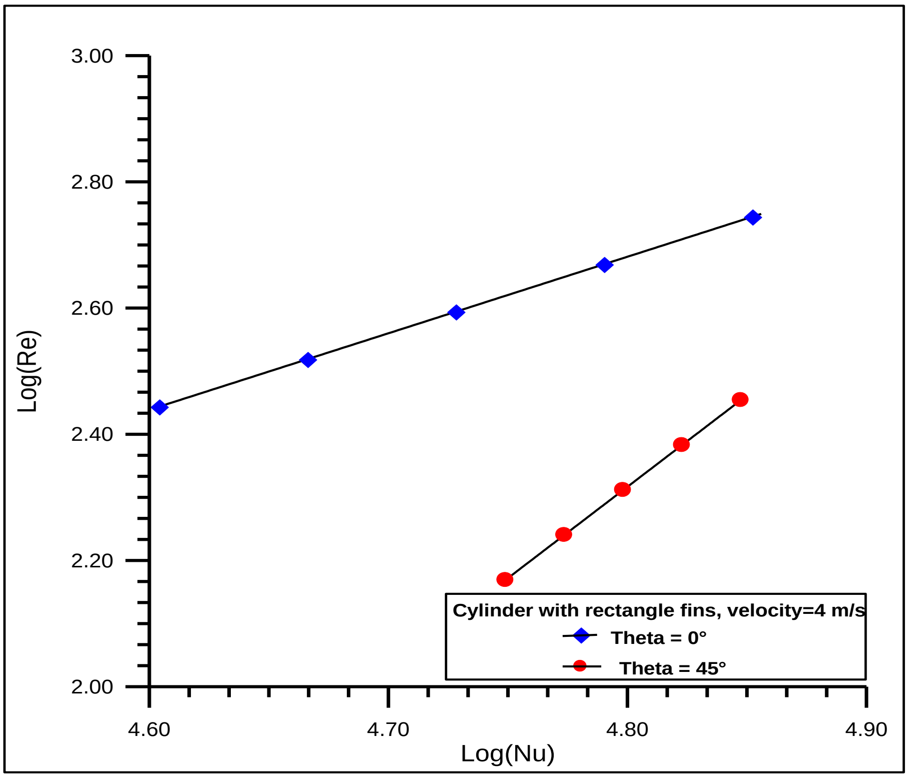
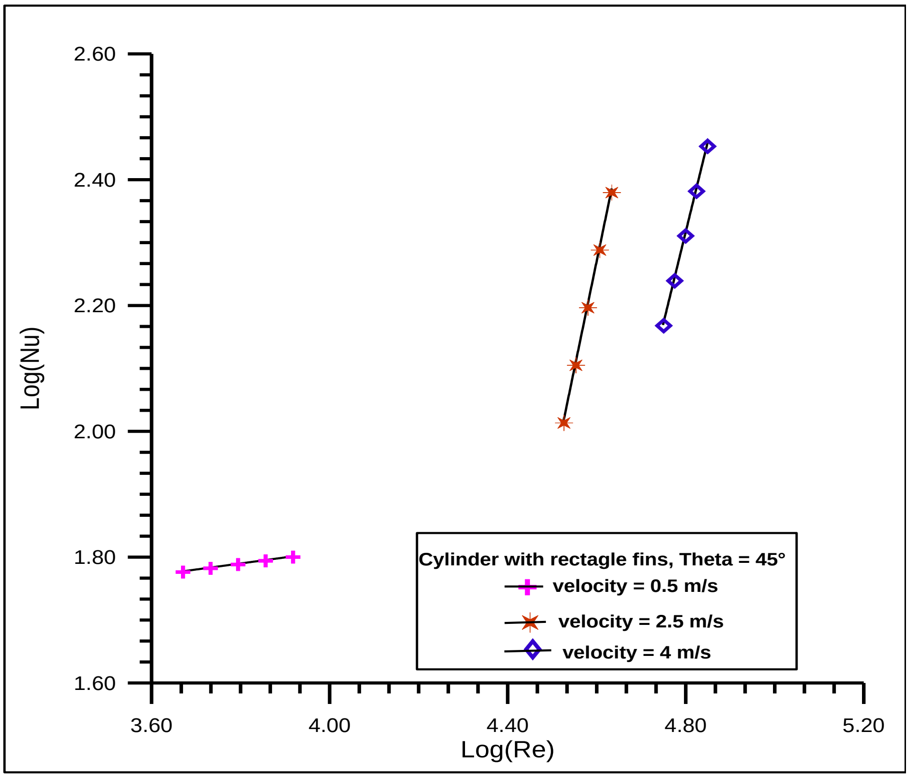


Figure (12) The effect of inclined angle on heat transfer rate



Figure(13) The effect of air flow velocity on heat transfer rate at inclined angle 45°