Forced Convection Heat Transfer around Heated Inclined Cylinder

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ABSTRACT
Experimental and numerical investigations are conducted to study heat transfer coefficient around cylindrical shape bluff body subjected to constant heat flux at its outer surface and cooled by air steam flowing around it. The study covered positioning the cylinder in vertical, horizontal and 45o inclined angle with respect to the free air stream, with its axis in horizontal position parallel to the flow and Reynolds numbers ranging (8,900 - 48,000) for vertical, inclined and horizontal positions.
Numerical approach is realized by conducting mathematical model of the problem and solving it numerically using a CFD Code FLUENT 6.3.26 after describing the mesh model using the Gambit 2.2.30. Three dimensional Cartesian coordinate system is considered in this study. The studied geometry is generated by using GAMBIT with dimensions of rectangular box L=127mm, W=130mm and H=129mm. The dimensions of heated cylinder is set identical to the experimental test H=90mm and R=17mm. Proper assumptions are based to solve the governed equations. The experiments are based on air as cooling fluid to investigate heat transfer coefficient around the heated cylinder. Constant heat flux is generated on the surface of the cylinder using proper electrical heater. The heat flux around the cylinder equal (5217, 5201 and 5284 W/m2) for vertical, inclined and horizontal positions respectively. Air flow is ensured around the heated cylinder at different velocities using proper test rig.
Local heat transfer coefficients around the cylinder are investigated based on the heat flux measured value and the temperature differences between local cylinder surface temperature and the air stream temperature measured by thermocouples, and standard Pitot-static tube with curved junction (N.P.L standard) was used to measure air free stream velocity.
The conducted comparison between the recent experimental results and those obtained from previous work showed reliable agreement. Where maximum error obtained from this comparison was 6.4% for vertical position.

Keywords
Forced convection, Circular cylinder, Cross flow, Experimental and Numerical Study.

1. INTRODUCTION
Many experimental investigations involved flow around bluff bodies initiated by cross flow have shown that the convective heat transfer from a bluff body is dependent
upon many factors such as material type, size of the body, temperature difference between body and flow, fluid properties, velocity, blockage effect, surface roughness of the body and others.
The investigations of previous works are classified in three parts depending on cylinder position vertical, inclined by 45^0 and horizontal position.
Most early studies were concerned with overall heat transfer to different fluids flowing across cylinders in vertical positions. The overall convective heat transfer from smooth circular cylinders was reviewed by A.A. Zukauskas in Heat transfer from tubes in cross-flow[1], and V.T. Morganin overall convective heat transfer from smooth circular cylinders in [2], and Davis in (1942) [3] measured heat transfer from wires to water, liquid paraffin, and three different transformer oils over the range of $0.14<Re<170$ and $3<Pr<1.5\times10^3$. The wires were electrically heated, and heat loss to the fluids was measured. It was found that heat transfer is strongly affected by the fluid properties. Additionally, a higher heat transfer rate was observed with a larger temperature difference between wall temperature and free stream temperature (ΔT), Kramers in [4] correlated Davis' data for liquids and those of several investigators for Reynolds number (5 - 10^3) in air. Van Mell [5] studied experimentally the local convective heat transfer from a cylinder with inside diameter 20.04 mm, for the Reynolds number from 5000 to 40000. Perkins and Leppert in [6] conducted a study on the overall heat transfer from a circular cylinder to water and ethylene glycol covering $40<Re<105$, $1<Pr<300$ and $2.5<\Delta T<60$ °C. With a thermal boundary condition of uniform heat flux, Fand in [7] measured the overall heat transfer with a thermal boundary condition of uniform wall temperature in water covering $104<Re<105$ and the $2<\Delta T<6$ °C. A correlation was proposed, All fluid properties were determined at the film temperature, Chun and Boehm [8] studied forced flow and convection heat transfer over a circular cylinder in cross flow that having two cases either a constant heat flux or an isothermal wall temperature. Used a numerical solution and governing equations are integrated for Reynolds number up to 3480. S. Sanitjai, R.J. Goldstein [9] studied the local and average heat transfer by forced convection from a circular cylinder to air and liquids for Reynolds number from 2×103 to 9×104 and Prandtl number from 0.7 to 176. For subcritical flow, the local heat transfer measurement indicates three regions of flow around the cylinder R.P. Bharti, R.P. Chhabra [10] studies the forced convection heat transfer from an unconfined circular cylinder in the steady cross-flow regime has been studied using a finite volume method (FVM) implemented on a Cartesian grid system in the range as $10 \le Re \le 45$ and $0.7 \le Pr \le 400$. S. Sarkar, A. Dalal, G. Biswas [11] work on the effect of Prandtl number on forced convective flow and heat transfer

characteristics past a circular cylinder in unsteady regime has also been explored recently, The results of forced convection with the Prandtl number range $0.7 \leq Pr \leq 100$ reveal functional dependence on the average Nusselt number with the Reynolds $60 \leq Re \leq 200$ and Prandtl number.

The first investigation for this incline case was studied by D'Alesfsio and Dennis [12]. The study covered steady laminar forced convection from an elliptic cylinder. Both the steady state and unsteady cases have been considered for moderate Reynolds numbers, Re, in the range $40 < Re < 70$ with Prandtl number Pr = 1 to 25. The numerical results show that there is good agreement between the steady and limiting unsteady heat transfer quantities Nu_{ave} and Nu which are the average and local Nusselt numbers respectively, H.M. Badr [13] has numerically investigated the two-dimensional laminar forced and mixed convection heat transfer from an isothermal elliptic cylinder to air ($Pr = 0.7$). He reported the influence of Reynolds number (20–500), angle of inclination (0–90) and aspect ratio (0.4–0.9) on heat transfer, and E.H. Ahmad and H.M Badr [14] studied the effect of fluctuations in the free-stream velocity on the mixed convection heat transfer from an elliptic cylinder for three values of the Reynolds numbers of 50, 100 and 150 and Grashof numbers as 20,000, 30,000 and 50,000, P. Sivakumar, R.P. Bharti [15] studied momentum transfer characteristics of the power-law fluid flow past an unconfined elliptic cylinder numerically by solving continuity and momentum equations using FLUENT in the two-dimensional steady cross-flow regime. The Reynolds number ($0.01 \leq Re \leq 40$) and the aspect ratio of the elliptic cylinder ($0.2 \leq E \leq 5$) on the local and global flow characteristics have been studied. R.P. Bharti1, P. Sivakumar, R.P. Chhabra [16] Forced convection heat transfer to incompressible power-law fluids from a heated elliptical cylinder in the steady, laminar cross-flow regime has been studied numerically. In particular, the effects of the power-law index ($0.2 \leq n \leq 1.8$), Reynolds number ($0.01 \leq Re \leq 40$), Prandtl number ($1 \leq Pr \leq 100$) and the aspect ratio of the elliptic cylinder ($0.2 \leq E \leq 5$) on the average Nusselt number (Nu) have been studied. The average Nusselt number for an elliptic cylinder shows a dependence on the Reynolds and Prandtl numbers and power-law index, which is qualitatively similar to that for a circular cylinder.

And for horizontal position investigate by T.S. Sarma [17] Local measurements have been made of the heat transfer from a horizontal circular cylinder to air in cross flow in forced convection and mixed convection. The studies have been conducted with the constant heat flux boundary condition at low Reynolds numbers ranging from 500 to 4700 and at modified Grashof numbers ranging from 0.8×107 to 3.3×107. R.A. Ahmad [18] A numerical analysis study for forced-convection heat transfer from a horizontal circular cylinder dissipating a uniform heat flux in a cross flow of air is conducted by solving the full two-dimensional steady-state Navier-Stokes and energy equations in the range of the Reynolds numbers from 100 to 500. And Y.A. Çengel [19] studied an empirical approximation which represents the average Nusselt number for forced convection over a circular cylinder for a range of Reynolds number between 4000 and 40,000 and $Re\,Pr > 0.2$. and C. Sak, R. Liu, D.S.-K. Ting [20] studied the convective heat transfer rate from a heated circular cylinder in cross flow of air. An aluminum cylinder of diameter 50.8 cm (2″) with uniform surface temperature was placed horizontally in a wind tunnel. The cylinder was subjected to a homogeneous, isotropic turbulent flow; the cylinder surface temperature was monitored and measured with five embedded thermocouples. Tests were conducted at a Reynolds number of 27,700, relative turbulence intensity (Tu), from 2.9% to 8.3% and turbulence integral length to cylinder diameter ratio, L/D from 0.50 to 1.47.

2. DEFINITION OF THE PROBLEM

The geometry of the problem is shown in figure (2.1). The geometry consists of a rectangular box contains the heated cylinder.

3. NUMERICAL SIMULATION

In order to analyze the flow field and heat transfer over heated cylinder in cross flow, and a solution of the Navier-stokes and energy equations is required. In the present work, the problem was solved numerically using a CFD Code FLUENT 6.3.26 after describing the mesh model using the Gambit 2.2.30, The system geometry in the present work basically consists of a box (which represents the flow tunnel) and this box contains heated cylinder in the middle of wind tunnel, The geometry is generated by using Gambit 2.2.30, interconnecting them by some interrelationships prepared for meshing and boundary conditions specifications. A very smooth mesh, a higher order element type tetrahedral / hybrid in 3D is used for mesh generation to approximate precisely the geometry interfaces

4. EXPERIMENTAL WORK

The experimental rig was designed and constructed in the Heat Transfer Lab, at the Mechanical Engineering Department University of Baghdad, were the experiments were carried out. A suction type, low speed wind tunnel with solid steel walls was used in the present work, the fan: driven by a one phase (AEI) A.C motor with a speed of 2850 rpm, the valve by which the flow rate of air can be controlled through opening from 0 % to 100 %. The velocity of air values are (4 m/s), (8.4m/s), (13.6 m/s), (18.3 m/s) and (21.7 m/s). The test section consists of a rectangular box with (length 127 mm x width 130 mm x height 129 mm with thickness 19 mm), both sides of the box are made of glass fiber, and the dimension of heated cylinder (height 90 mm x radius 17 mm) and made of thermal Teflon FEP (fluorinated ethylene propylene) 100 and the heater is placed on piece of wood help to change the angle of cylinder. The heater that used was (KANTHAC) with 7.1 Ω/m resistance, 1.3 mm thickness and made from aluminum surrounding around a cylinder to give the heater a constant heat flux in all regions of the cylinder. There are three ring of Alumel-Chromel (type K) thermocouples, these rings were separated by a uniform horizontal distance of a (38 mm) from the center ring which every ring have eight thermocouples at angles (0, 45, 90, 145, 180, 225, 270, 315), were fixed at the surface of the cylinder. See Figure (4.2), The following parameters were recorded during the test: heat flux and velocity. The ranges of measured variables are shown in **table (1).**

Table 1

Parameters	Values
Velocity Of Air Through Wind Tunnel	(4 , 8.4 , 13.6, 18.3 ,21.7 m/s) 20% , 40% , 60%, 80%, 100%
Power	49.5 w to 51 w
Number of Tests	20 Test each position
Time of testing	7:30 A.M to 12 A.M in April, May , June Month

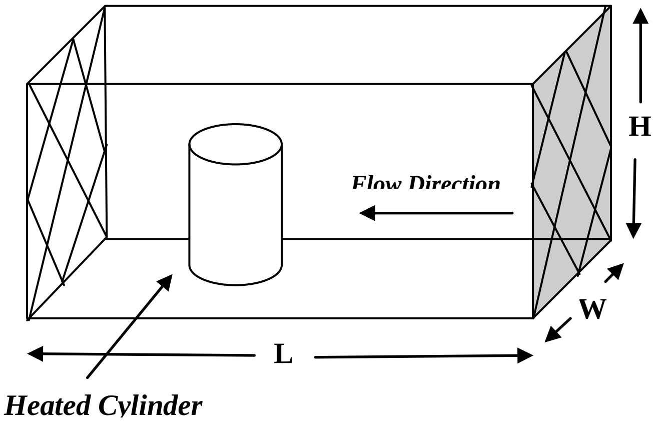
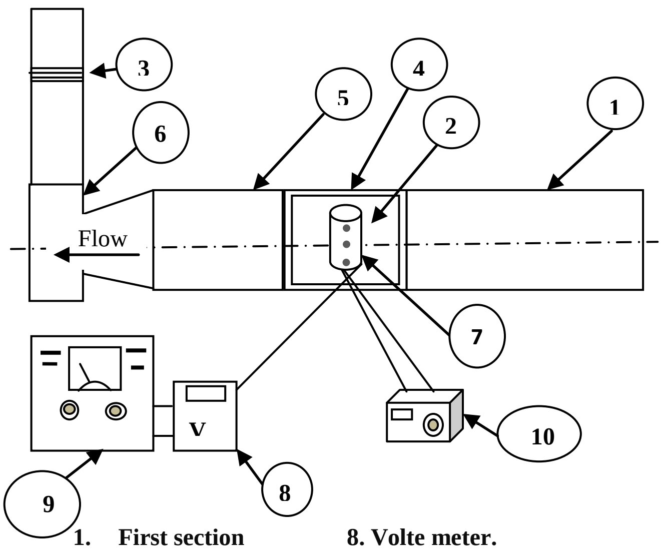


Fig. (2.1): Model Geometry



1. **First section**
2. **Heated cylinder**
3. **Valve.**
4. **Test Section.**
5. **Second section.**
6. **Fan.**
7. **Thermocouple.**
8. **Volte meter.**
9. **D.C Power Supply.**
10. **Selector Switches & Reader**

Fig. (4.2): Diagram of Experimental Apparatus

5. RESULTS AND DISCUSSION

5.1 Experimental Results

The temperature profile has been measured with the different coordinates at different velocities for position of cylinder. The direct reading of temperature profile from the surface of cylinder and different velocities values of (V= 4 m/s (20% opening), 8.4 m/s (40% opening), 13.6 m/s (60% opening) , 18.3 m/s (80% opening), 21.7 m/s (100% opening)) is plotted in figure (4) for three position. It is seen that, The variation of local heat transfer coefficient for vertical, incline and horizontal position depend on place of thermocouple around heated cylinder (h_θ $W/m^2.c^0$) , the velocity profile around the cylinder, and find the local Nusselt number is depending on the temperature profile around cylinder coordinate. The Figures (5.1), (5.2) and (5.3) represent the temperature distribution at any point on the surface of the cylinder and comparison of local Nusselt number with different velocities at constant heat flux. The experimental results show that the local Nusselt number increase with the increase heat transfer coefficient on this position because of the product of heat transfer coefficient at any location and depend on temperature distribution for vertical and incline position and for horizontal position added the local axial distance of heated cylinder. The local Nusselt number increase with increasing the Reynolds number at constant heat flux because the temperature Profile decreases with increasing the velocity at constant heat flux take power (49.5 - 51 W), $(h = \frac{q}{A(T_w - T_\infty)})$, The Figures (5.4) (5.5) and (5.6) represent. The experimental results show that the local Nusselt number and the heat transfer coefficient decreases with increasing surface temperature at constant heat flux, and the variation of local Nusselt number ($Nu_{\theta,X}$) depend on local heat transfer coefficient ($h_{\theta,X}$) and the path of flow rate will across it, to find the characteristic length in this position that the flow rate take it and the Nusselt number equal $Nu_{\theta,x} = \frac{h_{\theta,x} \times D_H}{k_{air}}$ As shown in different position, and the maximum value of average Nusselt number increases with increase of Re. The average Nusselt number was calculated from an integration of the local Nusselt number (Nu_x) and has been plotted with the Reynolds number for different values of velocity of air. and the average Nusselt number increases with increasing Reynolds number for three position, The high value of average Nusselt number due to the low difference between the air and heated cylinder temperature and the thin thermal boundary layer formed. and the correlation of average Nusselt number (Nu) with Reynolds number (Re) of experimental results in the form:

$$Nu = c\, Re^{n} \qquad \text{Eq. [1]}$$

Where c and n are empirical constants presented in **Table (2)** for the three positions.

Table (2): Correlation Equation for Experimental Results

Cylinder position	C	n	Reynolds Number
Vertical	0.8158	0.5518	8900 < Re < 48000
Incline	0.1686	0.6971	8900 < Re < 48000
Horizontal	4.2644	0.4026	8900 < Re < 48000

5.2 Numerical Results

Fluent program has been used to verify the experimental results where all experimental cases have been simulated in the fluent program. And the temperature distribution in the figures (5.7), (5.8) & (5.9) show the temperature distribution at different position of cylinder, which express the heat transfer from a surface cylinder to air. These figures show how the temperature distribution along the cylinder. And the result for these figure take the heat flux (5217, 5201, 5284 W/m2) for vertical, incline and horizontal respectively, The temperature distribution, which has been found by Fluent Program, agrees with behavior which has been depicted in experimental work. And for velocity distribution in the Figures (5.10), (5.11) & (5.12) show air velocity distribution around the cylinder for vertical, incline and horizontal positions, respectively. The figures show the stream lines for the velocity before and after crossing the cylinder and the distribution are based on (13.6 m/s) velocity of air. The figures are helpful to predict and justify the temperature distribution around surface of cylinder.

5.3 Verification

The experimental and numerical temperature profile results for Vertical, incline and horizontal positions have been compared with each other, as shown in Figures (5.13), (5.14) & (5.15) show the temperature distribution differences between experimental and theoretical results versus its characterized dimension at vertical ,incline and horizontal cylinder positions respectively inside the air tunnel and take the same velocity of air with magnitude (13.6 m/s) the medium velocity of air in this work for three positions. And the results related to these figures are based on the same heat fluxes in experimental and enter its to Fluent Program (5217, 5201 and 5284 W/m2) for vertical, inclined and horizontal positions respectively. The figures show that there are some differences between these temperature results at some points due to the effect of flow separation on the experimental values. The figures reveal that theoretical temperature profile follows the same behavior as the present experimental results. And the different between theoretical and experimental results in temperature about of 7.5%, 7.9% and 6.3% for vertical, incline and horizontal positions respectively. the difference between the experimental and numerical results is natural because of errors in the temperature measurements and because the Manufacturing of cylinder that don't give smooth 100% and the helical wire of heater on surface this will effect on velocity stream line around the cylinder in experimental work. And verify the results obtained from present study, a comparison is made with the results achieved from previous studies. The present results for Nusselt number in vertical position shown in figure (5.16) has the same behavior related to (Richardson 1963) experimental results, (Matsui (1970) experimental results and (S. Sanitjai, R.J. Goldstein (2004) results. the results related to local Nusselt number for inclined position which reveals same behavior of (D'Alesfsio and Dennis (1995) results. Figure (5.17) show the present results related to Nusselt number in horizontal position which agrees with the experimental results of (Smith and Kuethe (1966) , (Y.A. Çengel (1998)), and (R. Liu, D.S.-K. Ting, (2007)).

6. CONCLUSIONS

The results of the present work indicate that the local wall temperature in the radial direction for vertical and incline position and the axial direction in horizontal position decreases as Reynolds number increases and the heat flux decreases. for the same heat flux, the local Nusselt number increases as the Reynolds number increases for all position of cylinder, for the same velocity, the local Nusselt number increases as the changing the position of cylinder, and for the constant heat flux, the average heat transfer coefficient increases as the Reynolds number increases. And from this research find the best position is vertical; it's give a good heat transfer with flow rate, and the comparison gives a good agreement between the present and previous works.

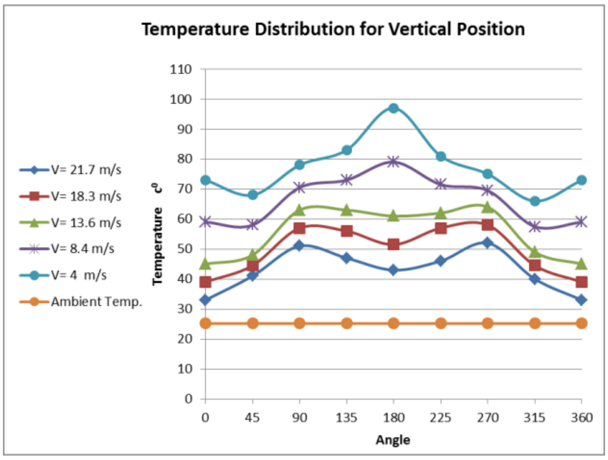


Figure (5.1) Temperature Distribution for Vertical Position

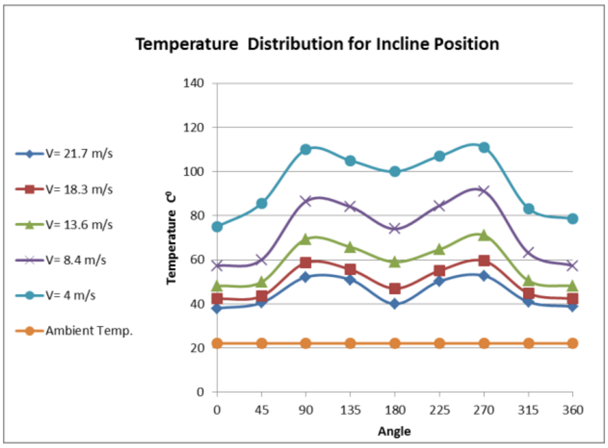


Figure (5.2) Temperature Distribution for Incline Position

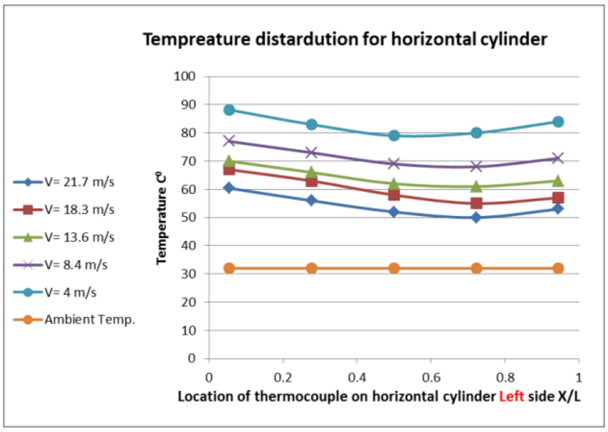


Figure (5.3) Temperature Distribution for Horizontal Position

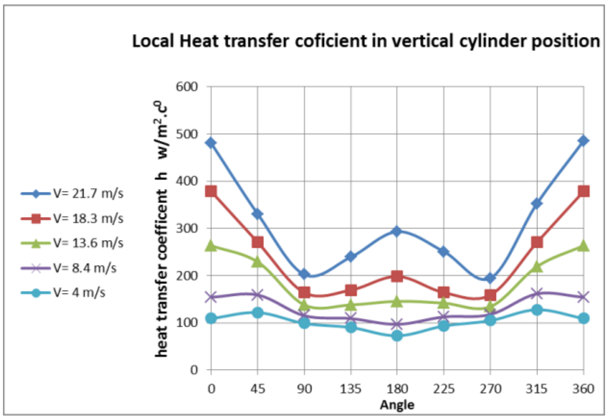


Figure (5.4): local Nusselt Number in Vertical position

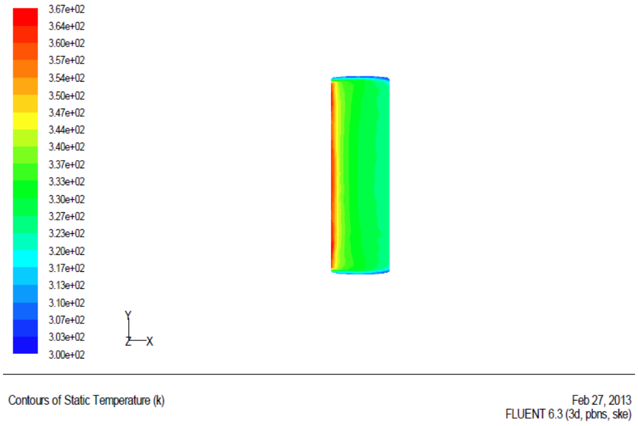


Figure (5.7): Temperature Distribution for Vertical Position

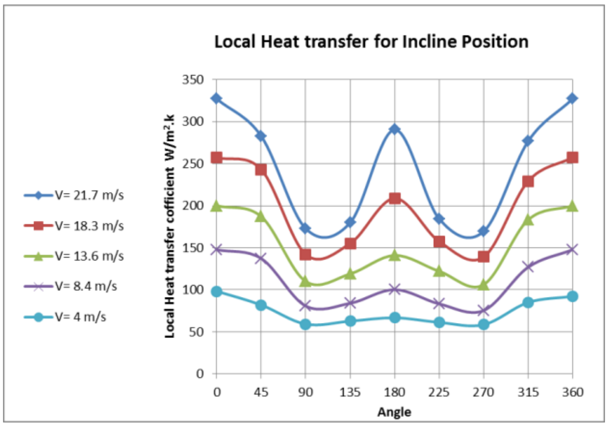


Figure (5.5): local Nusselt Number in Incline position

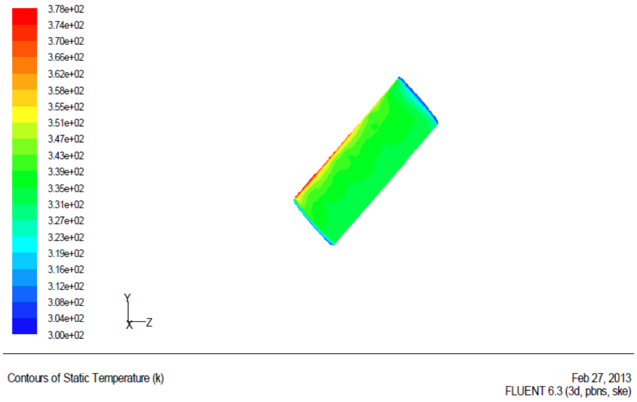


Figure (5.8): Temperature Distribution for Incline Position

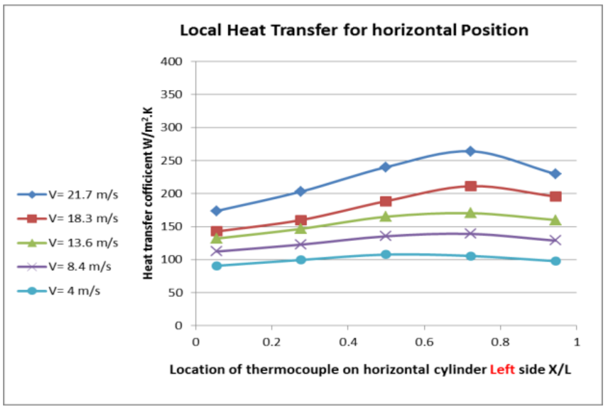


Figure (5.6): local Nusselt Number in Horizontal position

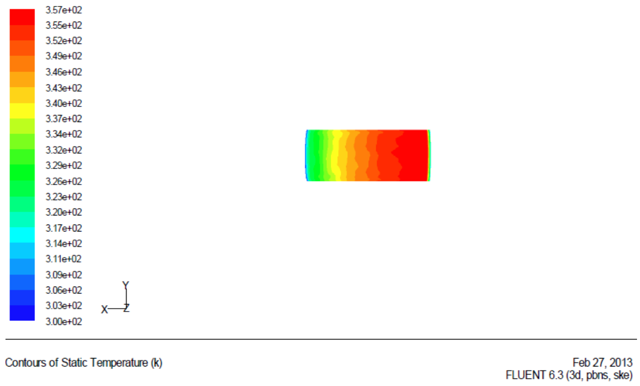


Figure (5.9): Temperature Distribution for Horizontal Position

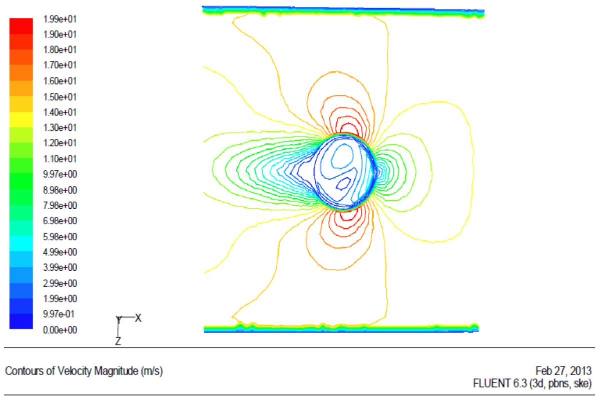


Figure (5.10) Velocity distribution for Vertical position

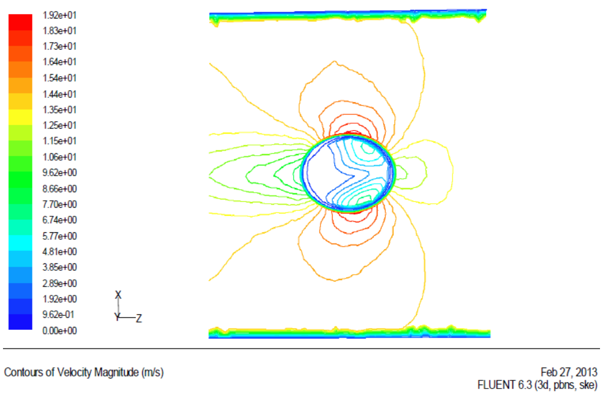


Figure (5.11): Velocity distribution for Inclined position

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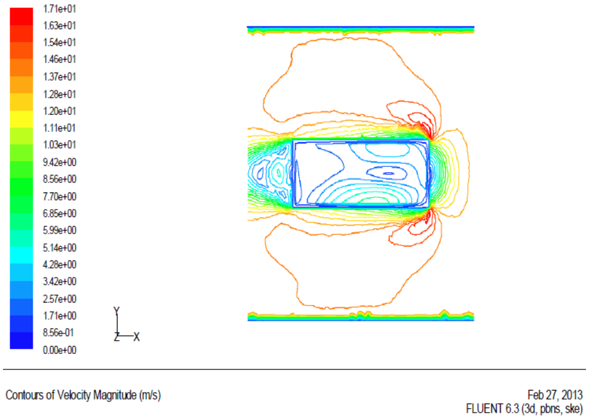

Figure (5.12): Velocity distribution for Horizontal Position

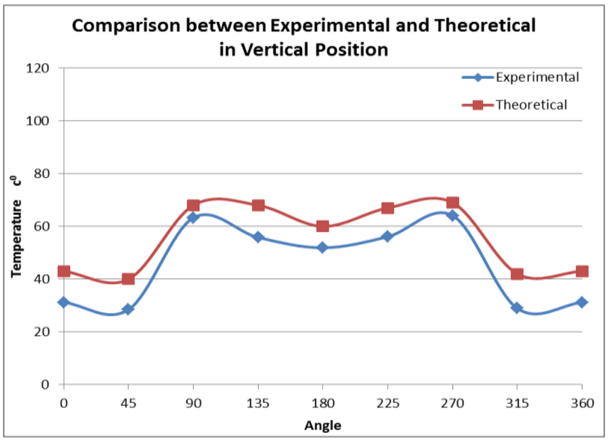

Figure (5.13): Comparison between Experimental and Theoretical in Vertical Position

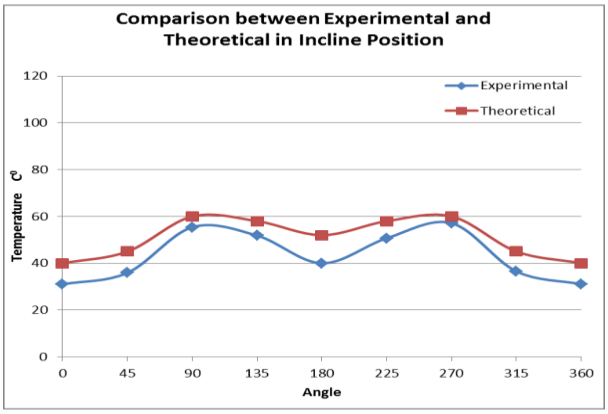

Figure (5.14): Comparison between Experimental and Theoretical in Incline Position

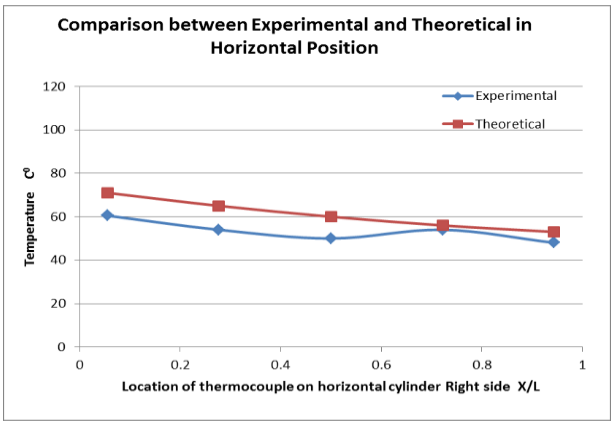

Figure (5.15): Comparison between Experimental and Theoretical in Horizontal Position

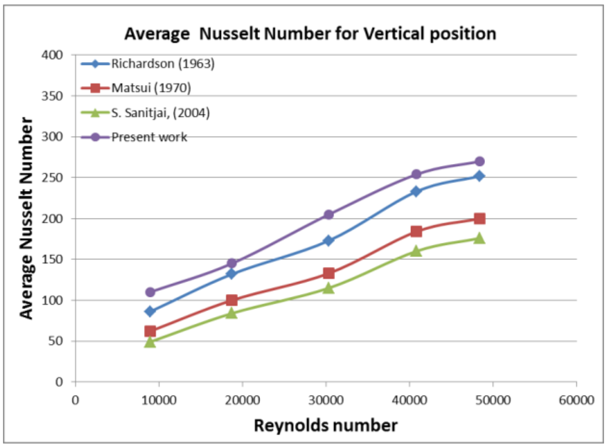

Figure (5.16): Comparison of Previous and Present Results in Vertical Position

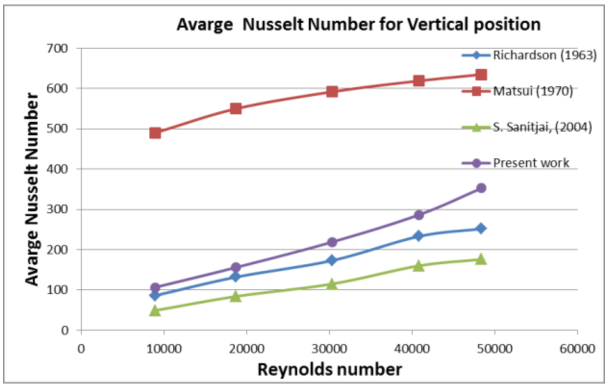

Figure (5.17): Comparison of Previous and Present Results in Horizontal Position

7. NOMENCALTURE

h Heat Transfer Coefficient $W/m^2.K$
L,D Characteristic Length of cylinder
Nu Nusselt Number
q'' Heat Flux W/m^2
V Velocity of air m/s
A_S Surface Area m^2
T_∞ Ambient Temperature ^0C
T_S Surface Temperature ^0C

8. ACKNOELEDGMENTS

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