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The Forced Convection For The Horizontal Cylinder Embedded In A Porous Medium Application

Ahmed Adil Rahmah

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Abstract

Theoretical and experimental study was carried out of heat transfer by forced convection from the cylinder in cross flow embedded in a saturated porous media. The theoretical part of the work includes solving the standard energy transport equation in porous media regions by FLUENT. There are maximum temperature and minimum velocity at the rear of cylinder q = 0^0 but minimum temperature and maximum velocity at the front of cylinder q = 180^0. The stagnant area at q = 0^0 and the separation at

*□ = 90^0 andq = 270^0. The experimental part of this work included the construction of an experimental model composed of cast iron cylinder with a (18 mm) inner diameter and (20mm) outer diameter with a length of(200mm) heated internally by an electrical heater. The cylinder was embedded in a packing of glass ball with diameter (12mm) placed in across flow wind tunnel. The experimental results revealed that the average heat transfer increased when the Peclets number and Reynolds number increased for steady state condition. The relationship between **Na&Re** and **Na&Pe** for experimental gives us **Nu=alnRe-b** and **Nu=2Pe-2.666** respectively when **a** and **b** are constants*

*depending on **Re** and **Q** for 20 £ Q £ 120*
*(**Q** in watt) and 10.087 £ a £ 27.61 , 66.33*
£ b £183.6 .

2000 £ Re £ 3000

Introduction

Intensive interest has been shown in recent years in the area of convective heat transfer in fluid-saturated porous media. This is quite natural because of the numerous and wide-ranging engineering applications of convective flow through porous media. For example this class of phenomena is encountered in industrial and geophysical contexts such as petroleum reservoir modeling, chemical and nuclear engineering, solar power collectors, regenerative heat exchangers containing porous materials, also the corresponding problem of forced convection in a porous medium has important applications in geothermal reservoirs where pressure gradients are generated as a result of withdrawal or re-injection of geothermal fluids (Bejan, 1984). Theoretical studies of the problem have been presented by (Badr & Pop, 1988), they used truncate Fourier series for following series expansion of the stream function and temperature, involved the use of Darcy's law but did not require the boundary layer approximation.

It was in good agreement with the boundary layer solutions. Eventually (Fand & Pan,1987); studied forced convection using paced bed of 3 mm glass spheres with water as a fluid medium. They advanced several hypothetical arguments to develop an empirical correlation of their data in the form.

$$Nu = 1.48(0.255 + 0.699 ReD^{0.5})Pr^{0.29}\, ReP^{0.179}; ReP > 100$$

(Beckerman et al.,1987); studied forced convection in packed bed for a wide range of thermal conductivity (from 200w/m.K for aluminum to 0.23 w/m.K for nylon) and three nominal particle size (3 mm, 6 mm, and 13 mm). These and other potential applications of the concept indicated the need for information on heat transfer coefficients associated with convection from surface embedded in porous media. The heat transfer augmentation produced by the porous matrix attributed to a combination of effects, including thinning of the hydrodynamic and thermal boundary layers around the cylinder, increasing mixture (i.e. thermal dispersion), and direct conduction through the porous matrix. The purpose of the present study is to examine the influence variation of particles diameters on the buoyancy inducing mixed convection from a horizontal cylinder embedded in saturated porous media.

A theoretical study was introduced the problem by employing the boundary layer approximation of the energy equation by Darcy's law. The porous medium treated as a continuum by volume averaging the properties of the solid and fluid phases ,were previous by (Cheng,1982). A theoretical study of the problem used truncate Fourier series for the following series expansion of the stream function and temperature presented by (Huny,1986) which involved the use of Darcy's law without the required of the boundary layer solutions.

(Sherzad, 2000) studied the forced convection from cylinder in saturated porous media. An experimental part includes measuring temperatures around cylinder **Pe** based on the cylinder diameter and the velocity of the flow that changed from 3 to 50m/s ,and a theoretical part includes momentum ,energy equations using Darcy flow model were solved by finite difference method with a constant surface temperature .

In this work, a theoretical model of forced convection heat transfer from cylinder embedded in porous medium was achieved by FLUENT program (tolarski,2006), to get information concerning the nature of flow and heat transfer about a horizontal cylinder embedded in a porous medium.

The experimental work studied forced convection in packed bed for a wide range of heat flux and velocity include the measure of temperatures and velocity around cylinder to calculate Nu,Pe,and Re numbers.

Mathematical model:

Although Finite Element methods have gained recently popularity in the field of CFD applications, most of the commercial CFD software is still based on the Finite Volume method. In this technique, the equations of motion are treated in balance form of a finite sized control volumes (CV). A benefit of writing the governing equation in this particular form is that the conservation laws are fulfilled implicitly. Numerical implementations appear to be very robust and (depending on the applied discretization scheme) economical regarding the CPU time (Holman,1977) .

Transient three-dimensional convection-diffusion of a general property φ in the velocity field is governed by:

$$\frac{\partial(\rho\phi)}{\partial t} + \frac{\partial(\rho u\phi)}{\partial x} + \frac{\partial(\rho v\phi)}{\partial y} + \frac{\partial(\rho w\phi)}{\partial z} = \frac{\partial}{\partial x}\left(\Gamma\frac{\partial\phi}{\partial x}\right) + \frac{\partial}{\partial y}\left(\Gamma\frac{\partial\phi}{\partial y}\right) + \frac{\partial}{\partial z}\left(\Gamma\frac{\partial\phi}{\partial z}\right) + S \qquad \ldots(1)$$

The cylinder which equipped with porous media in this paper had been modeled by using a commercial computational fluid dynamics package, Fluent, and its accompanying mesh generation software, Gambit. In FLUENT, the full set of mass, momentum and energy equations solved two- or three-dimensions. FLUENT allow us to include heat transfer within the fluid / or solid in our model. When heat is added to a fluid and the fluid density varies with temperature, flow could be induced due to the force of gravity acting on the density variations. Such flows are termed natural-convection flows and can be modeled by FLUENT.

Porous media are modeled by the addition of a momentum source term to the standard fluid flow equations. The source term (S) is composed of two parts; The first term in the right side of equation (2) is a viscous loss term (Darcy term), while the second term in the right side of equation (2) is an inertial loss:

$$S = -\left(\frac{\mu}{\alpha} v_i + C2\,\frac{1}{2}\rho\,|v|\,v_i\right) \qquad \ldots(2)$$

Where α is the permeability, this permeability was obtained by using the following equation:

$$\alpha = \frac{d_p^2\,\phi^3}{172.8(1-\phi)^2} \qquad \ldots(3)$$

FLUENT solves the standard energy transport equation (Equation in porous media regions using an effective conductivity in conduction flux. The effective thermal conductivity in the porous medium is computed by FLUENT as the volume average of the fluid conductivity and the solid conductivity:

$$k_{eff} = k_f\,\phi + (1-\phi)k_s \qquad \ldots(4)$$

Mesh generating:

Gambit program was used for building the grid shown in Fig. (1). the type of element used was Quadrilateral element. The model used a segregated solver with implicit formulation. The momentum and energy equations are solved using the first order scheme. The standard discretization is recommended to solve the forced convection flows. Therefore, standard discretization used to decretive the pressure term [8]. Under relaxation factors applied in order to control the changes of variable values between successive iterations and avoiding divergence of the solutions. During the resolution of the equations, the typical values used under relaxation parameters were approximately 0.3 for pressure, 0.7 for momentum and 0.8 for energy and density.

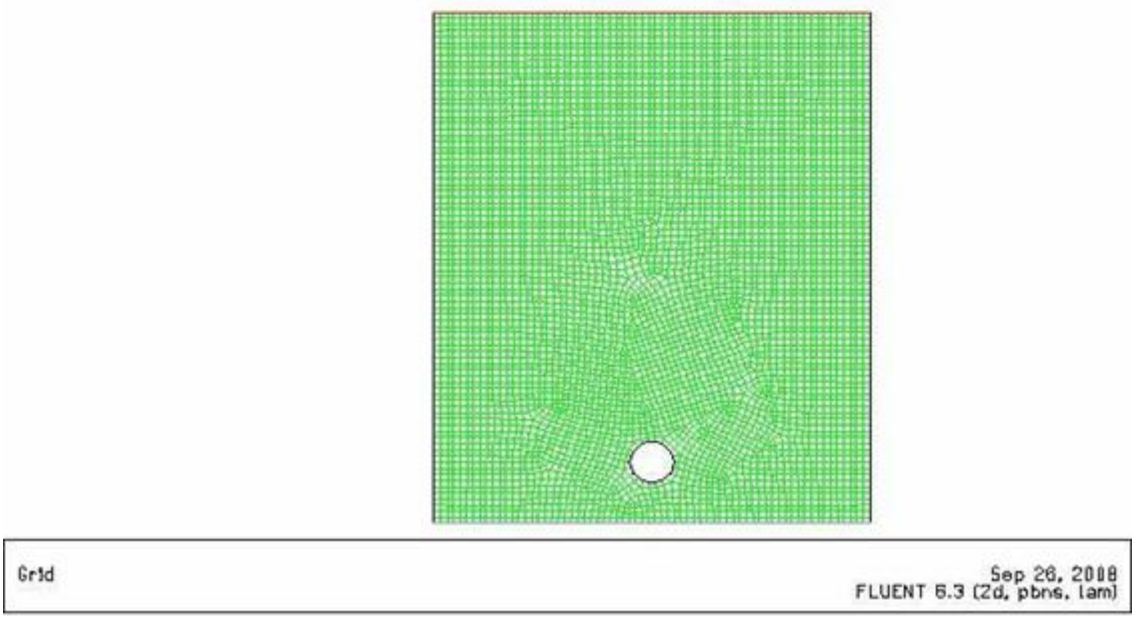

Fig. (1) Mesh generated of the problem by GAMBIT software

<u>Experimental Apparatus and Procedure</u>

The experimental set-up consists basically the following elements.

1. The Blower:

The air forced to flow through the tunnel by centrifugal blower. It is an A.C. electrical motor, model 3394m with constant speed of 2750 rev/min. as a maximum. It connected with a 100 mm diameter pipe followed by regulating gate valve.

2. The Duct:

The pipe is connected by (0.2x0.2) m^2 iron plate duct. Its length 0.4 m, this length can be expected to give uniform flow at the test section, to avoid high turbulence level and the flow separation phenomena that can take place by connecting the pipe and the duct, a gradual diffusion of flow was designed as shown in Fig.(2).

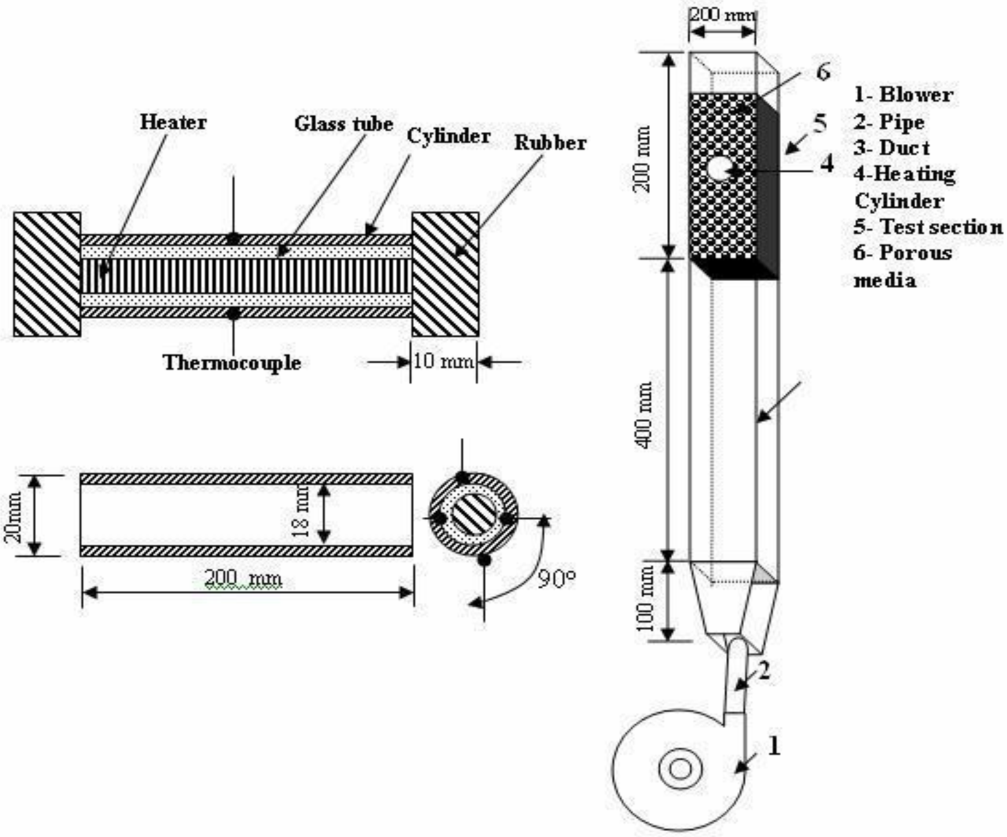

Fig (2) The schematic of the duct and the test

Section 3.Test Section:

The duct was followed by an iron plate test section; it has the same section area of iron plate duct, where the length of it is 120mm. The three sides fixed to the duct, whereas the top one was movable used to fill the porous elements. A mesh wire of (5x5) mm^2 was used to close the ends of the test section and to fix the spheres in position. The side walls of test section are provided by two holes of (22 mm) diameter to hold the cylinder in place. Two pieces of a low conductivity rubber where supported to insulate the cylinder ends. Thermostor were placed just above the test section and used to measure the temperature of air flow.

4. Heated Element:

The cast iron cylinder (L=200mm,Din=18mm,D out=20mm) was heated internally by electrical heater(Ni -Cr alloy) 21Ω resistance inserted in thermal insulation and powered by A.C. power supply .
To measure the temperature of cylinder surface, four thermostor (2 mm) tip diameter used at the location of Semi-cylinder with 90^o intervals from the top of cylinder, another two thermostor were placed at the ends of cylinder test, these thermostor were attached to the outer surface of cylinder ,small holes were drilled to insert the thermostor in the proper place. These thermostor were fixed by using epoxy steel and it was calibrated before measurement using ice-water bath which gives the calibration curve as shown in Fig.(3).

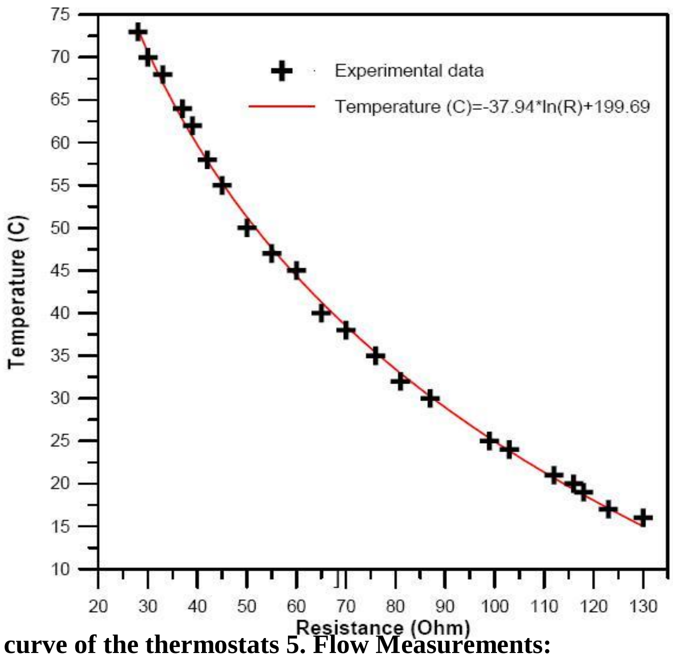

Fig. (3): The calibration curve of the thermostats 5. Flow Measurements:

The air flow velocity was measured by using velocity measurement device LM-8000 (0.1 m/s resolution).

Electrical devices:

1. The voltage regulator type (HSN 1.25 KVA (0-250 V)) was used to control the supply voltage to heater, and clamp meter mode no.266 (100 mA+2%) was used to measure the current.

2. Multi-meter was used to measure the resistance of the thermostor which used to calculate the temperature according to calibration curve.

Experimental Procedure:

In this experimental study, the porous media used in the experiments were packed beds of solid spheres made of glasses with (12) mm diameter, the test section was filled with packing material, the spheres were poured randomly in the test section. At first the blower is turned on for four minutes to reach the steady state operation, then a flow rate is adjusted to give a suitable Peclet number. The electrical A.C. power is supplied to the heater inside the cylinder and left to reach a steady state by noting the temperature. A steady state has been reached when the surface temperature remains constant for (20-30) minutes depending on input voltage and the air flow velocity, after that the readings of surface temperature, inlet air temperature, input voltage, current, and the air flow velocity were measured.

<u>Method of calculation</u>

1-Calculation of Air flow rate:

The air flow velocity was measured by velocity measurement device (LM-8000(0.1 m/s resolution). Then used in continuity eq.

$$m = \rho U\infty\, A \qquad ...(5)$$

2-Calculation of The Average Heat Transfer Coefficient:
The total heat transfer from the cylinder is estimated as

$$Q = Q_t - Q_{loss} \qquad ...(6)$$

$$Q_t = I * V \qquad ...(7)$$

The average heat transfer coefficient also

$$h = \frac{Q}{A_s (T_s - T_\infty)} \qquad ...(8)$$

$$Ts=(T1+T2+T3+T4)/4 \qquad ...(9)$$

Calculation The Effective Thermal Conductivity:

Several theoretical and empirical models for the evaluation of this parameter have been reported recommended the model for the stagnant effective thermal conductivity.

$$k_{eff} = k_f \phi + (1 - \phi) k_s \quad ...(10)$$

Thermal conductivity of glasses and air with another property have been taken from (Holman, 1977).

3-Calculation of Nuselts, Reynolds, and Peclets Number:

All the properties used are evaluated at the film temperature, calculated as

$$T_{mean} = \frac{(T_s + T_\infty)}{2} \quad ....(11)$$

And the Nusselt, Reynold and Peclet numbers are defined as follows:

$$Nu = \frac{hD}{k_f} \quad ...(12)$$

$$Pe = \frac{U_\infty D}{\alpha_m} \quad ...(13)$$

where $\alpha_m = \dfrac{k_f}{\rho_f \, Cp_f}$

$$\mathrm{Re} = \frac{U_\infty \rho D}{\mu} \quad ...(14)$$

4-Heat Losses:

The heat losses from the two ends of the cylinder can be estimated from the heat output by the radiation and the convection taken from (Yanes, 1990). The radiation heat transfer calculated from the following formula:

$$Q_{radiation} = \sigma A_c \in (T_s^4 - T_\infty^4) \quad ...(15)$$

$$Nu_N = 0.48 Ra^{0.25} \quad ...(16)$$

Where $Ra = \dfrac{\beta g (T_s - T_\infty) D^3}{\nu^2} \Pr$ ...(17)

...(18)

And $h_N = \dfrac{Nu_N \, k_f}{D}$ ...(19)

$$Q_{convection} = A_c h_N (T_s - T_\infty) \quad ...(20)$$

Then the heat lost from the ends of the cylinder is estimated as:

$$Q_{loss} = (Q_{radiation} + Q_{convection})$$

Results and Discussions

Temperature distributions around cylinder were presented in Figs. (4, 5, and 6). These figures show thermal plume exists at the rear of the cylinder and the maximum temperature was found at the rear of cylinder, the width and the length of the plume decreased as Reynolds number increased. Also the levels of temperature distribution were decreased with the increase of Reynolds number because the high level remove the heat from the cylinder at the Reynolds high number. The effect of the boundary layer diminished gradually as Reynolds number increases till it disappeared at Reynolds high number.

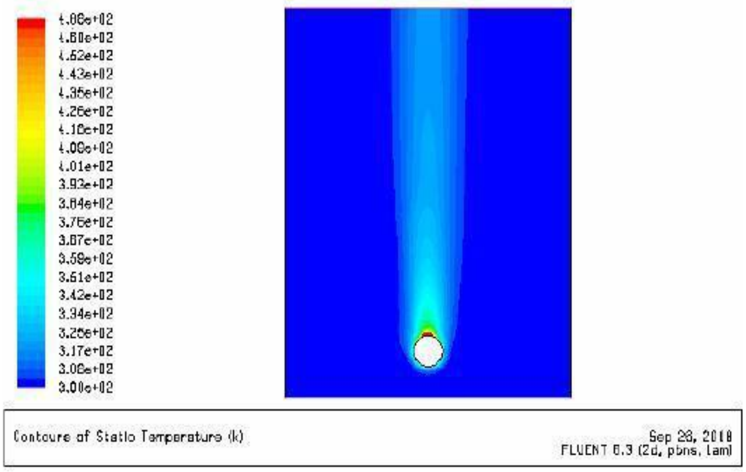

Fig.(4) Temperature distribution around the cylinder at Re =2274

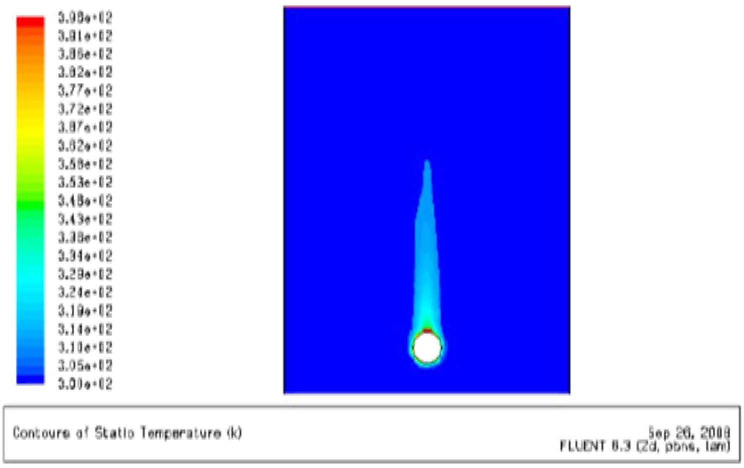

Fig.(5) Temperature distribution around the cylinder at Re =2497

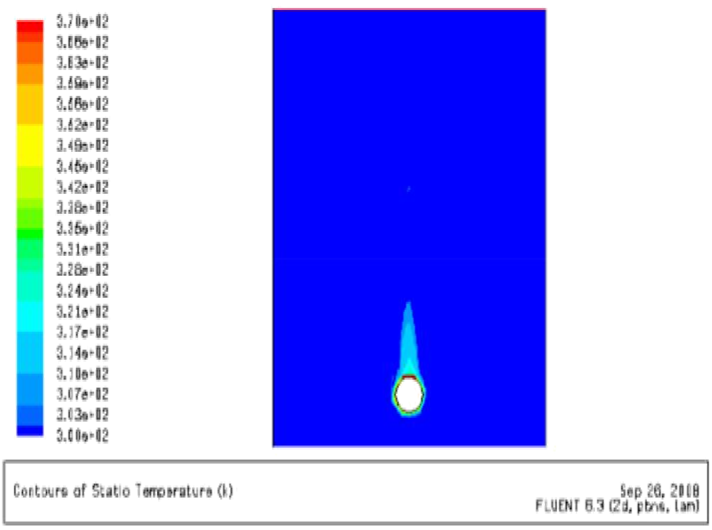

Fig.(6) Temperature distribution around the cylinder at Re =2746

Contours of velocity presented in Fig. (7, 8, and 9) showed that maximum velocity was occurred at $\theta = 900$ and $\theta = 2700$ because of tangential flow and minimum velocity at front and back of the cylinder due to stagnant, this behavior was the same for all cases, therefore the velocity pattern was independent on the Reynolds number.
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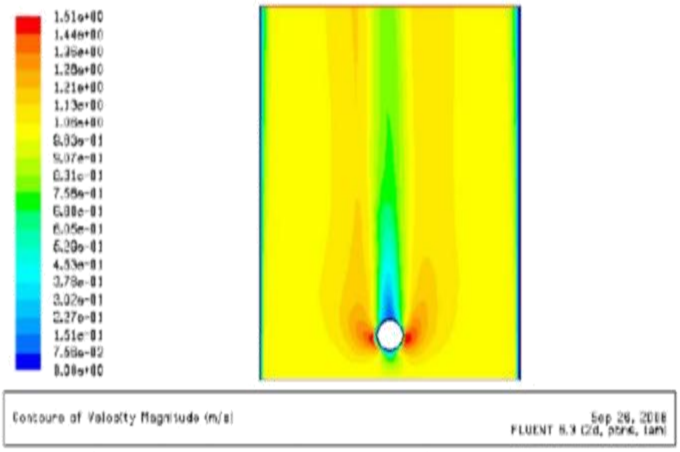

Fig.(7) Velocity distribution around the cylinder Re = 2274

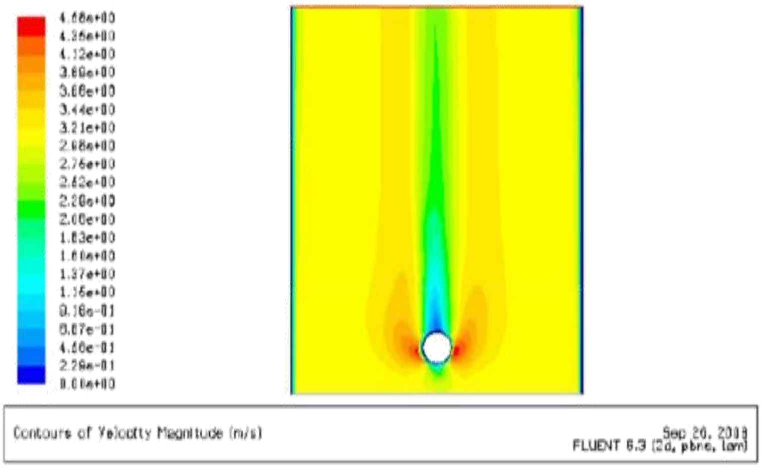

Fig.(8) Velocity distribution around the cylinder at Re = 2497

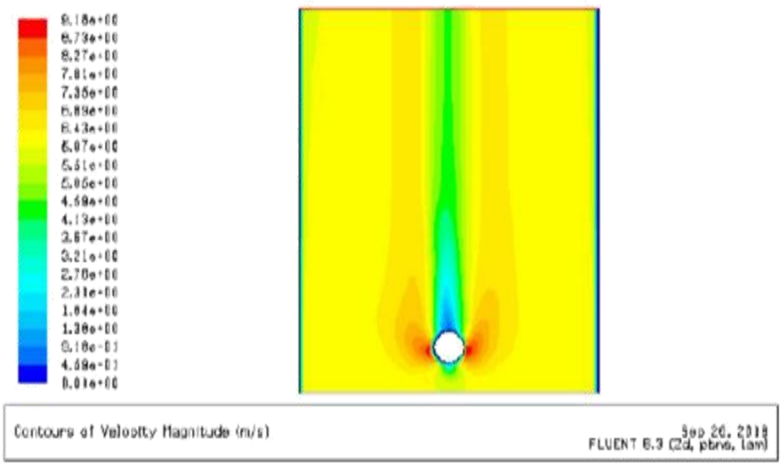

Fig.(9) Velocity distribution around the cylinder at Re = 2746

The relation between Nu and time in fig.(10,11,and 12),Nusslet number decreased with time where the porous medium is cold at the beginning, so it absorbs the heat unit it stabilized, and the only effect left of Re number, so the heat with drawl by forced convection.

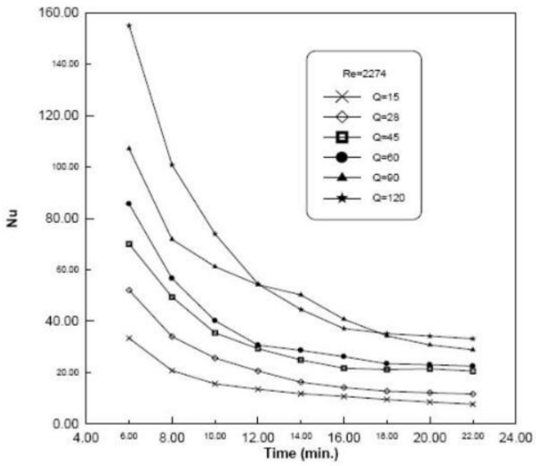

Fig. (10) Variation Nusselt number with time at Re=2274

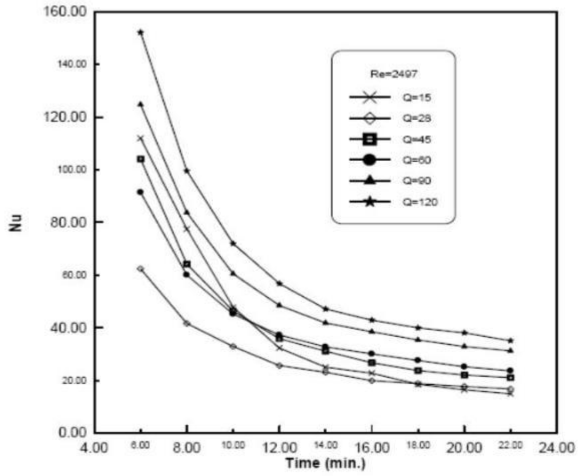

Fig. (11) Variation Nusselt number with time at Re=2497

\ **Fig. (10) Variation Nassslt number with time at Re= 2274**

Fig. (11) Variation Nassslt number with time at Re= 2497

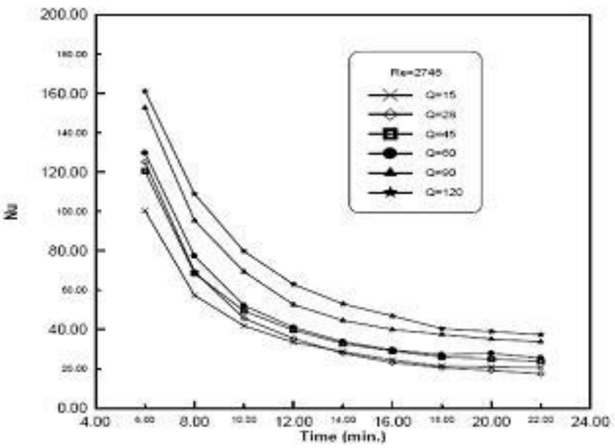

Fig. (12) Variation Nassslt number with time at Re= 2746

figure (13) represents the relation between Na&Re steady status for a different values of heat supplied. The experimental results revealed that the average heat transfer increased when the Reynolds number increased . The relationship between Na&Re for experiment give us Nu=alnRe-b when a and b are constants depending on Re and Q for $20 \le Q \le 120$ (Q in watt) and

$2000 \le Re \le 3000$,$10.087 \le a \le 27.61$, $66.33 \le b \le 183.6$.

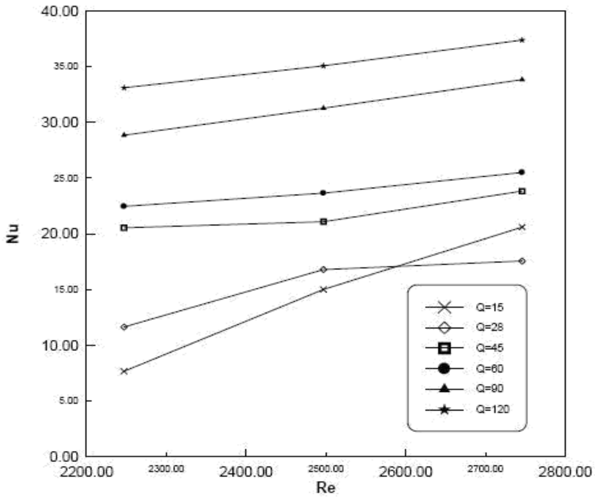

Fig. (13) Variation Nassslt number with Re For various heat flux steady state condition

Figure (14) represents the relation between *Nu* & *Pe* . The experimental results revealed that the average heat transfer increased when the Peclet number increased . The relationship between Na&Pe for experimental give : Nu=2Pe-2.666 for the same condition in figure (13)

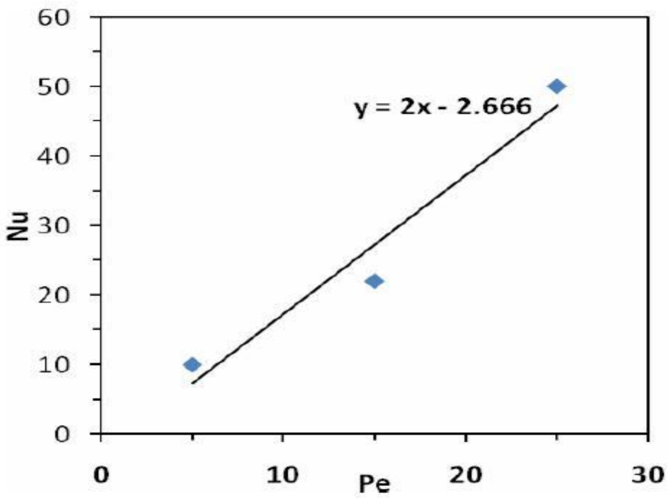

Fig (14) Variation Nassslt number with Pe For various Re steady state condition

Conclusions

1. There is maximum temperature and minimum velocity at the rear of cylinder □ □ 00 but minimum temperature and maximum static pressure at the front of cylinder □ □ 180 ^{o}.
2. The stagnant area at □ □ 00 and the separation at □ □ 900, 2700.
3. There is increasing Nu with Re& *Pe* for different heat flux.

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Author Information

Ahmed Adil Rahmah
Missan Oil Company/ Ministry of Oil