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Theortical and Exprimental Study of Forced Convection Heat Transfer from a Horizontal Cylinder Embeded in Porous Medium

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Abstract

Theoretical and experimental study of heat transfer by forced convection from the cylinder in cross flow embedded in a saturated porous media was carried out. The theoretical part of the work includes the derivation of the

governing momentum and energy equation by using Darcy flow model. The two equations are solved by finite difference method with constant cylinder surface temperature, with a Peclet numbers ranged between (1< Pe<10).

The experimental part of this work included the construction of an experimental model composed of cupper cylinder with a (13 mm) in diameter heated internally by an electrical source. The cylinder was embedded in a packing of glass spheres with diameter 12 mm placed in a low velocity wind tunnel. Both the theoretical and the experimental results revealed that the average heat transfer increased when the Peclet number increased.

Keywords: Forced convection, porous media, Peclet number

Nomenclature

A_C	*Cross sectional area of test section (m^2)*
Api	*Cross sectional area of pipe (m^2)*
Ao	*Cross sectional area of orifice (m^2)*
A	*Radius of the cylinder (m)*
Cp	*Specific heat (J/kg.°C)*
D_p	*Diameter of particle (m)*
D	*Diameter of cylinder (m)*
G	*Gravitational acceleration (m/s^2)*
H	*Average heat transfer coefficient ($W/m^2.°C$)*

I	*Electrical current (A)*
K	*Thermal conductivity (W/m.°C)*
Q	*Power input to the electric heater (W)*
Q_{loss}	*Heat loss from heated cylinder to the wall (W)*
Q	*Local heat flux (W/m²)*
r,θ	*Radial and angular coordinate (m,deg)*
R	*Dimensionless radial coordinate*
T	*Temperature (°C)*
U	*Flow velocity (m/s)*
u,v	*Darcian dimension radial and transverse components of velocity (m/s)*
V	*Voltage (V)*
Δh	*Different between two levels of manometer (m)*
V_r, V_θ	*Darcian dimensionless radial and transverse components of velocity*
	Dimensionless Group
Nu	*Average Nusselt number*
Pe	*Peclet number*
Ra	*Rayleigh number*
Re	*Reynolds number*
	Greek Symbol
Ξ	*Logarithmic radial function*
Ψ	*Dimensionless stream function*
ϕ	*Dimensionless temperature*
Ψ	*Dimensional stream function (m²/s)*
E	*Porosity*
P	*Density (kg/m³)*
A	*Thermal diffusivity (m²/s)*
N	*Kinematic viscosity (m²/s)*
B	*Coefficient of thermal expansion (K^{-1})*

$\in$	*Emissivty*
Σ	*Stefan-Boltzman constant (W/m^2K^{-4})*
	Subscripts
R	*Radial direction*
S	*Surface condition or solid*
F	*Fluid*
Θ	*Transverse direction or local Nusselt number*
∞	*Free stream condition*
P	*Based on particle diameter*
M	*Stagnant or arithmetic mean temperature*
K	*Kerosene*

Introduction

(Pop,Ingham and Cheng , 1993) used the method of matched asymptotic expansions is employed for investigating the growth of the free convection boundary-layer on a horizontal circular cylinder which is embedded in a porous medium. It is assumed that the Rayleigh number is large, but finite, and the time of investigation is short. It is shown that the solution contains terms that are absent from the solution based on the boundary-layer approximation and that vortices form at both sides of the cylinder. The development of the plume region near the top of the cylinder, as well as the local and average Nusselt numbers, are evaluated and presented in graphical form.

(F.M.Hardy,F.S.Ibrahim and M.R.Eid.,2008) presented a numerical study of heat transfer in a conical annular cylinder fixed with saturated porous medium is presented .The heat transfer is assumed to take place by natural convection and radiation. The governing partial differential equations are non-dimensionalised using suitable non-dimensional parameters and the solved by using finite element method.

(F.M.Hardy. *etal*., 2008) studied the effect of chemical reaction on mixed convection fluid a long non-isothermal horizontal surface embedded in a saturated porous medium has been studied .

(A.Y.Bakier,2009) presented an analysis to investigate the effect radiation on mixed convection from a vertical flat plat in a saturated porous medium. Both a hot surface facing upward and a cold surface facing downstream are considered in the analysis .

(Vadim.N, and Amable.L, 2001) presented An analysis for the steady, two-dimensional, free convection around line sources of heat and heated cylinders in unbounded saturated porous media. It is extended to account also for the effects of forced convection. The study is based on the Boussinesq equations, with the velocities calculated using Darcy's law.

(Aydin.M, 2006) studied the heat transfer in a square cavity filled with clear fluid or porous medium is numerically investigated. To change the heat transfer in the cavity a rotating circular cylinder is placed at the center of the cavity. The ratio of cylinder diameter to cavity height is chosen 0.8.Depending on the angular velocity of the cylinder the convection phenomena inside the cavity becomes natural, mixed, and forced. To keep the number of data low the Grashof number,Gr,is set to 10^6 ,while the parameter defining the convection regime in the cavity ,Gr/Re^2 ,is changing from 0.0625 to 10^2. The Darcy number in the cavity is set to 10^{-2},10^{-3},and 10^{-4} .

Khalil,K. and Ali,j. (2003) presented a numerical investigation of mixed convection in a horizontal annulus filled with a uniform fluid –saturated porous medium in the presence of internal heat generation is carried out. The inner cylinder is heated while the outer cylinder is cooled. The forced flow is induced by the cold outer cylinder rotating at a constant angular velocity.

(Atalah.J, 2009) presented a Theoretical and experimental study of heat transfer by mixed convection heat transfer from the cylinder in cross flow embedded in a saturated porous media was carried out with a Peclet and Rayleigh numbers ranged between ($0.001< Pe<40$) and ($5<Ra<100$) respectively .

The purpose of the present study is to exzanine the influence Reynolds bunber on forced convention from a horizontal cylinder in saturated porous nedia.

Formulation and Numerical Method

Consider the problem of forced convection about a horizontal cylinder embedded in a porous media with constant surface temperature. In the mathematical formulation of the problem, assumed that(i)the convective fluid and the porous medium are everywhere in local thermodynamic equilibrium, (ii)the temperature of the fluid is everywhere below boiling point,(iii)properties of the fluid and the porous medium are homogeneous and isotropic, and (iv)the Boussinesq approximation is invoked. Under these assumptions the physical and the mathematical models are sketched in fig. (1). Based on the Darcy low the governing equations for the problem are .

Continuity Equation

$$\frac{\partial}{\partial r}(ru)+\frac{\partial v}{\partial \theta}=0$$

(1)

Momentum Equation

$$u=\frac{-K}{\mu}\left(\frac{\partial p}{\partial r}\right)$$

(2)

$$v=\frac{-K}{\mu}\left(\frac{\partial p}{r\partial \theta}\right)$$

(3)

It is convenient to eliminate the pressure term between the equation (2) and (3) by differentiate the first w.r.t (α) and the other w.r.t (r) thus momentum eq. becomes.

$$(4)\ \frac{\partial u}{\partial r}-r\frac{\partial v}{\partial r}-v=0$$

(4)

Energy Equation

$$u\frac{\partial T}{\partial r}+v\frac{\partial T}{r\partial \theta}=\alpha_m\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}\right]$$

(5)

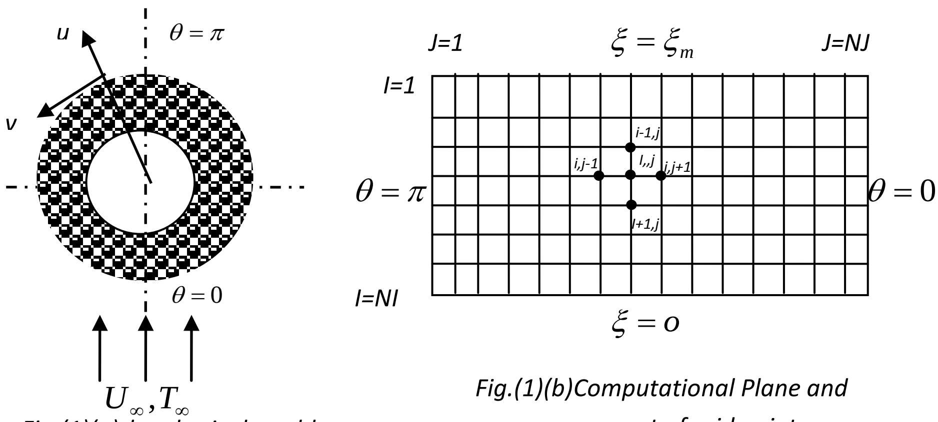


Fig.(1)(a)the physical problem and coordinate system

Fig.(1)(b)Computational Plane and arrangement of grid points

Dimensionless Formulation

The study of the governing equation will be carried out in non-dimensional form. This has the advantages of reducing the number of parameters in the result and making the transformation of the equations into finite differences form easier. The eqs. (4) and (5) are further transformed to their non-dimensional form by introducing the following dimensionless quantities Atalah,J.(2009).

$$R = \frac{r}{a}, \quad V_r = \frac{u}{U_\infty}, \quad V_\theta = \frac{v}{U_\infty}, \quad \Psi = \frac{\psi}{aU_\infty}, \quad \phi = \frac{(T - T_\infty)}{(T_s - T_\infty)}$$

Since the stream function (ψ) is related to the velocity components by

$$V_r = \frac{1}{r}\frac{\partial \psi}{\partial \theta}, \quad V_\theta = -\frac{\partial \psi}{\partial r}$$

Using the above variables , eqs. (4 and 5) becomes.

$$\frac{\partial \ \Psi}{\partial \theta^2} + R^2 \frac{\partial^2 \Psi}{\partial R^2} + R\frac{\partial \Psi}{\partial R} = 0$$

(6)

$$\frac{\partial \Psi}{\partial \theta}\frac{\partial \phi}{\partial R}-\frac{\partial \Psi}{\partial R}\frac{\partial \phi}{\partial \theta}=\frac{2}{Pe}\left(R\frac{\partial^2 \phi}{\partial R^2}+\frac{\partial \phi}{\partial R}+\frac{1}{R}\frac{\partial^2 \phi}{\partial \theta^2}\right) \quad (7)$$

The modified polar coordinates (ξ,θ) are employed, where $\xi=\ln R$ and accordingly the equations (6) and (7) are transformed to.

$$\frac{\partial^2 \Psi}{\partial \xi^2}+\frac{\partial^2 \Psi}{\partial \theta^2}=0 \quad (8)$$

$$\frac{\partial \Psi}{\partial \theta}\frac{\partial \phi}{\partial R}-\frac{\partial \Psi}{\partial R}\frac{\partial \phi}{\partial \theta}=\frac{2}{Pe}\left(\frac{\partial^2 \phi}{\partial \xi^2}+\frac{\partial^2 \phi}{\partial \theta^2}\right) \quad (9)$$

The appropriate boundary conditions of equations (8) and (9) are Atalah,J.(2009) .

$$V_r=0 \quad \textit{and } T=T_s \textit{ at } R=1 \quad \textit{(10 a)}$$

Where the dimensionless radial and transverse velocity components V_r and V_θ become

$$V_r=\frac{1}{R}\frac{\partial \Psi}{\partial \theta}=e^{-\xi}\frac{\partial \Psi}{\partial \theta} \quad \textit{(10b)}$$

$$V_\theta=-\frac{\partial \Psi}{\partial R}=-e^{-\xi}\frac{\partial \Psi}{\partial \xi} \quad \textit{(10c)}$$

Boundary conditions (10a) can be written as

$$\Psi=\frac{\partial \Psi}{\partial \theta}=0 \;\;,\;\; \phi=1 \;\; at \;\; \xi=0 \quad \textit{(11 a)}$$

$$e^{-\xi}\frac{\partial \Psi}{\partial \theta}\to\cos\theta \;\;,\;\; e^{-\xi}\frac{\partial \Psi}{\partial \xi}\to\sin\theta$$

$$as \quad \xi \to \infty \quad and \quad \phi \to 0$$

(11 b)

$$\frac{\partial \Psi}{\partial R} = 0 \quad for \quad \theta = 0, \pi, \qquad \xi \geq 0$$

(11c)

Boundary condition (11b) can not be applied, therefore $\xi = \xi_m = 2$.

Is taken as an outer boundary on which approximations to the condition at infinity, Badr(1988)

Since

$$e^{-\xi} \frac{\partial \Psi}{\partial \theta} \to \cos\theta$$

Becomes $\Psi = e^{-\xi_m} \sin\theta$

(11 d)

For engineering applications, we are often concerned with the effect of fluid motion on the heat transfer from the cylinder surface. This can be evaluated by computed the local Nusselt number in transverse direction Nu_θ and the mean Nusselt number Nu averaged over the cylinder surface which are given by

$$Nu_\theta = \frac{2ah_\theta}{K_m}$$

(12a)

$$Nu = \frac{2ah}{K_m}$$

(12 b)

Where

$$h_\theta = \frac{q_s}{(T_s - T_\infty)}$$

(13 a)

$$h = \frac{1}{2\pi} \int_0^{2\pi} h_\theta d\theta$$

(13 b)

Where

$$q_s = -K_m \left(\frac{\partial T}{\partial r} \right)_{r=a} , \quad h_\theta = -\frac{K_m}{(T_s - T_\infty)} \left(\frac{\partial T}{\partial r} \right)_{r=a}$$

(14)

And in term of ξ and θ

$$\left.\frac{\partial T}{\partial r}\right)_{r=a} = \frac{T_s - T_\infty}{a} e^{-\xi} \left.\frac{\partial \phi}{\partial \xi}\right)_{\xi=0}$$

$$h_\theta = -\frac{K_m}{a} e^{-\xi} \left.\frac{\partial \phi}{\partial \xi}\right)_{\xi=0}$$

(15)

Then the local Nusselt number

$$Nu_\theta = -2e^{-\xi} \left.\frac{\partial \phi}{\partial \xi}\right)_{\xi=0}$$

(16)

And the average heat transfer coefficients from $\theta = 0$ to $\theta = \pi$ becomes

$$h = \frac{K_m}{2a\pi} \int_0^\pi Nu_\theta d\theta$$

(17)

And the average Nusselt number becomes

$$Nu = \frac{1}{\pi} \int_0^\pi Nu_\theta d\theta$$

(18)

Nu is determined by using the Simpson's one-third rule

$$Nu = \frac{\Delta\theta}{3} \{ Nu_\theta(1) + Nu_\theta(n) + 4[Nu_\theta(2) + Nu_\theta(4) + \cdots] + 2[Nu_\theta(3) + Nu_\theta(5) + \cdots] \}$$

(19)

Experimental apparatus and procedure

The experimental set-up is explained in Fig.(2.a) which consists basically of the following elements.

1. **The Blower**

   The air forced to flow through the tunnel by centrifugal blower. It is an A.C. electrical motor, constant speed model with its speed is 2900 rev/min. as a maximum. It connected with a 50 mm diameter pipe followed by regulate gate valve and an orifice flow meter.

2. **The Duct**

   The pipe is followed by (0.16x0.14) m^2 wooden duct. Its length 1 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 at connection of the pipe and the duct, a gradual diffusion of flow was designed as shown in Fig.(2.b).

3. **Test Section**

   The wooden duct was followed by a wooden test section, it has the same section area of wooden duct, while the length of its 170mm . The three sides of it where fixed with the duct, whereas the top one was movable to help in filling of the porous elements. A mesh wire of (1.5x1.5)mm2 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 (14 mm) diameter to hold the cylinder in place. Two pieces of a law conductivity rubber where supported to insulate the cylinder ends as shown in Fig.(2.c). A thermometer was placed just above the test section and used to measure the temperature of flowing of air.

4. **Heated Element**

   The cylinder was heated internally by (8 mm) outside diameter electrical heater inserted in thermal glass tube and powered by A.C. power supply as shown in Fig.(2.c) .

   The small air gap between the glass tube and inner surface cylinder is filled with sand in order to reduced influence of convective current in this space. To measure the temperature of surface cylinder was used three copper-constantan thermocouples (0.2 mm) at the location of simi-cylinder with 90^o intervals from the top of cylinder , these thermocouples were attached to the outer surface of cylinder inside small holes were drilled to insert the thermocouples in the proper place. These thermocouples were fixed by using epoxy steel and it was calibrated before being to measure the surface temperature of the cylinder by using ice-water bath to give the calibration curve as shown in Fig.(3).

5. **Flow Measurements**

The air flow velocity was measured by thin-plate orifice meter. The recommended installation for concentric thin-plate orifice the inlet pressure tap is located one pipe diameter upstream, and outlet pressure tap is located one half diameter down stream of the orifice as measured from the upstream face of the orifice ,(Holman,1977), since the diameter of the orifice was (35.5 mm). The orifice plate was made according to the previous specification and clamped between two flanges on the line of pipe. The pressure drop a cross the orifice was measured using a U tube manometer by utilizing kerosene as a fluid. (Holman,1977) .

Experimental Procedure

In this experimental study, the porous media used in the experiments were packed beds of solid spheres made of glasses with a diameter (4,8,12)mm the test section was filled packing material, the spheres were poured randomly in the test section. At first the blower is turn on and left for minutes to reach the steady state operation, then a flow rate is adjusted to give a suitable Peclet number. In that instant, the electrical A.C. power is supplied to the electrical heater inside the cylinder and left to reach a steady state. a steady state is reached when the surface temperature remains constant this about (15-30) minutes, after that the readings of surface temperature, inlet air temperature, input voltage and current, and manometer are recorded.

Method of calculation

Calculation of Air Velocity.

The velocity of air in pipe (u_{pi}) is measured from Δh of the manometer by using the equation (Holman,1977).

$$u_{pi} = \overline{K}\,\frac{A_o}{A_{pi}}\sqrt{\frac{2g\rho_k \Delta h}{\rho_{air}}}$$
(20)

Where ρ_k is the density of the Kerosen using in the manometer and $\overline{K}$ is the flow coefficient taken Holman(1977);.

Then u_{pi} converted to velocity of air in test section (U_∞) by using continuity equation

$$U_\infty A_o = u_{pi} A_{pi}$$
(21)

Calculation of The Average Heat Transfer Coefficient.

The average heat transfer coefficient also

$$h = \frac{Q - Q_{Loss}}{A_s\left(T_s - T_\infty\right)}$$
(22)

Calculation The Effective Thermal Conductivity.

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

$$k_m = k_f\left\{1 - \sqrt{(1-\varepsilon)} + \frac{2\sqrt{(1-\varepsilon)}}{(1-\lambda B)}\left[\frac{(1-\lambda)B}{(1-\lambda B)^2}\ln\left(\frac{1}{\lambda B}\right) - \frac{(1+B)}{2} - \frac{(B-1)}{(1-\lambda B)}\right]\right\}$$
(23)

Where $B = 1.25\left[(1-\varepsilon)/\varepsilon\right]^{\frac{10}{9}}$, $\lambda = k_f / k_s$

Thermal conductivity of glasses is taken from Holman(1977) ;

Calculation of The Nusselt Number and Peclet Number

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

$$T_{mean} = \frac{(T_s - T_\infty)}{2}$$
(24)

And the Nusselt and Peclet numbers are defined as follows .

$$Nu = \frac{hD}{k_m}$$
(25)

$$Pe = \frac{U_\infty D}{\alpha_m}$$
(26)

where $\alpha_m = \frac{k_m}{\rho_f Cp_f}$

Heat Losses

The heat losses from the two ends of the cylinder to the duct can be estimated from the difference between the total heat input to the cylinder and the heat output by the radiation and the convection.

The radiation heat transfer was calculated from the following formula ,Atalah,J.(2009).

$$Q_{radiation} = \sigma A_S \in \left(T_S^4 - T_\infty^4\right)$$
(27)

While the convective heat transfer was calculated as follow .

$$Nu = 0.48 Ra^{0.25}$$
(28)

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

(29)

And $h = \frac{Nuk_f}{D}$

(30)

$$Q_{convection} = A_s h (T_s - T_\infty)$$
(31)

Then the heat lost from the ends of the cylinder is estimate as

$$Q_{loss} = Q_{input} - \left(Q_{radiation} + Q_{convection}\right) \qquad (32)$$

Fig.(2.a) Picture Explain Experimental Set-Up.

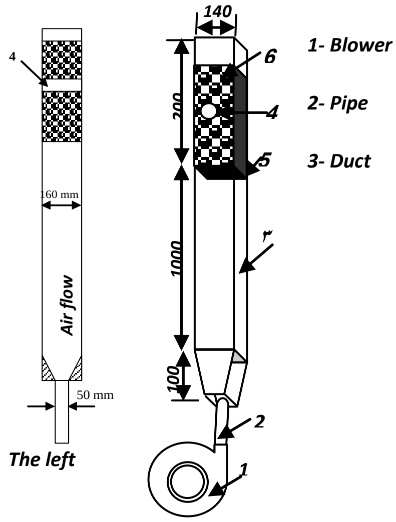
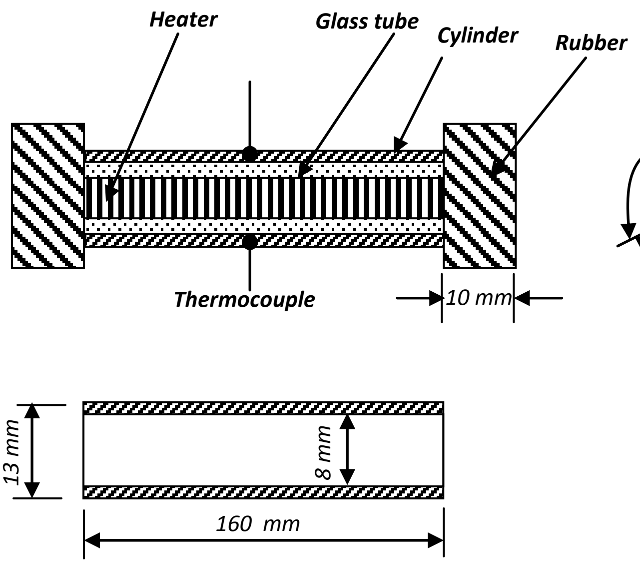


Fig.(2.b) The schematic of the duct and the test sectio



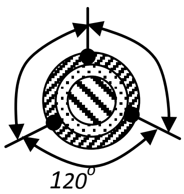


Fig. (2.c) Detailed construction of heated cylinder

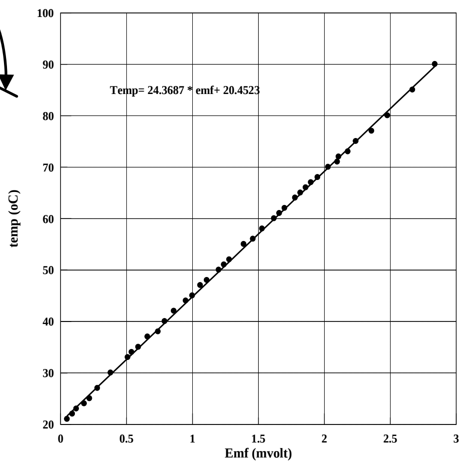


Fig.(3) Calibration of thermocouple

Discussion of the results

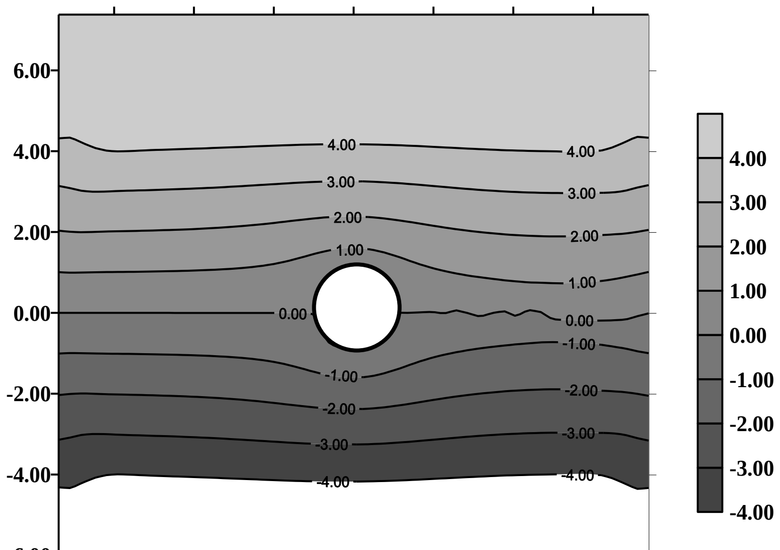
The numerical solution of the steady flow heat transfer equation over the cylinder is obtained for constant Prandtle number (0.75) at various values of Peclet numbers ranging from (1 to 10) .The stream lines pattern for the cases of Pe (10) is represented in Figs.(4). This figure shows that the streamline pattern is not variation for different values of Peclet number because absence the effect of the bouncy force of the flow.

The isothermal pattern for the cases for Pe(1 to 10) are represented in Figs. (5-8). These figures show that near the line angle=0 the thermal plum exist at the rear of the cylinder the width of the plume decreases as Peclet number increases while at the front of the cylinder the width becomes very thinning as Peclet number increases because the warm of the fluid therefore will cause a decrease of heat transfer rate. i.e. low gradient temperature in the downstream of the cylinder region.

Fig.(9)shows that the temperature distribution at angle=10^{o} as a function of radial distance (r) for Pe values 1 and 10. We note from this fig. that the temperature decreases with the radial distance.

Fig. (10) the variation of the predict local Nusselt number with angle for Pe=1 and 10 over the cylinder surface this Fig. shows that the Nusselt decreases monotonically from highest value at angle=180^{o} and reaches a minimum at angle=0^{o}.

Fig.(11) compares the experimental results with prediction of the theoretical solution for (12) mm nominal size particles. From this figure we observe the experimental results are agreement with the theoretical result in low Peclet number because the thermal boundary layer is large and is worked as a resistance to decrease the Nusselt number but in high Peclet numbers the experimental data converges to the theoretical results because the thermal boundary layer becomes very thin since our assumption for the theoretical equation was valid.



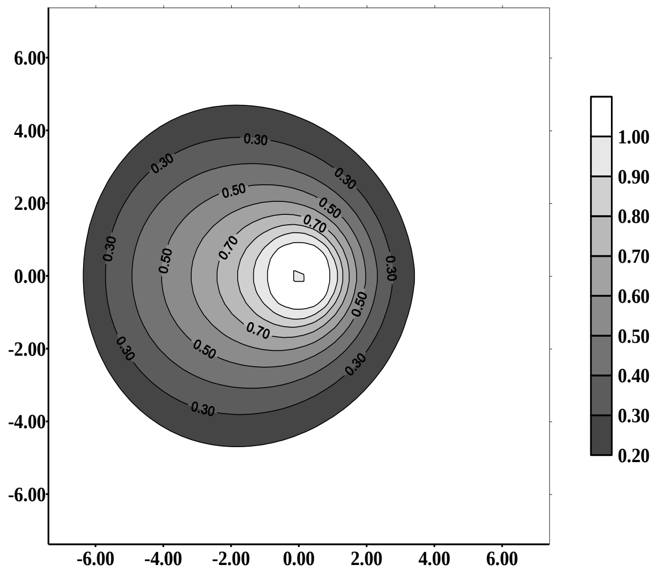
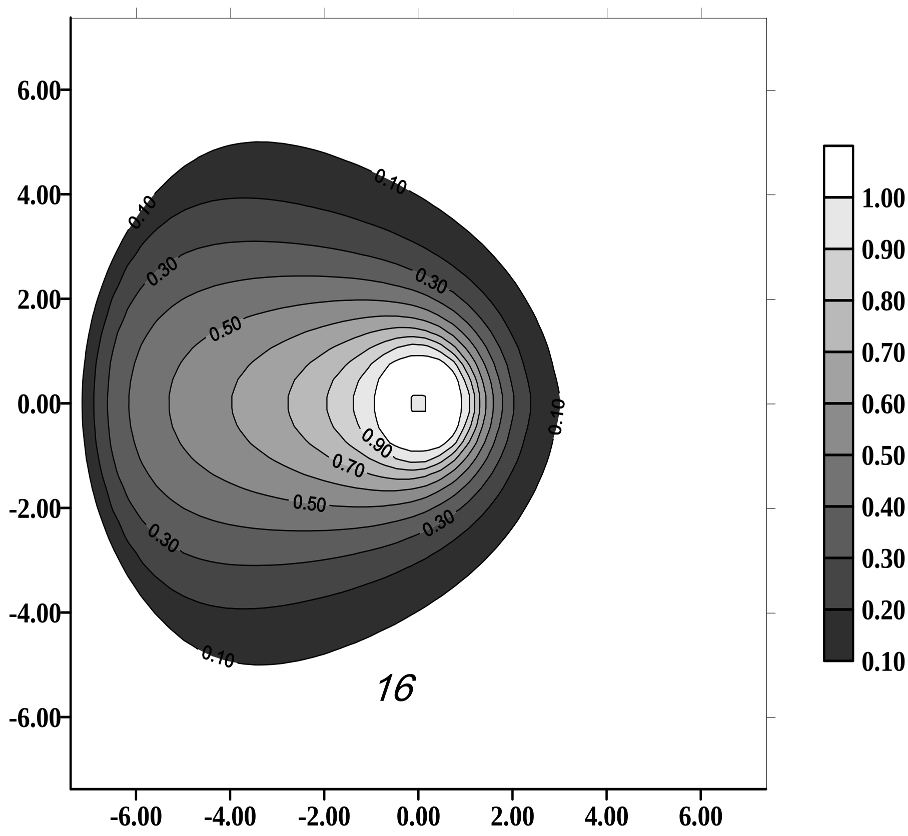


Fig.(5)Isothermal lines for Pe= 1



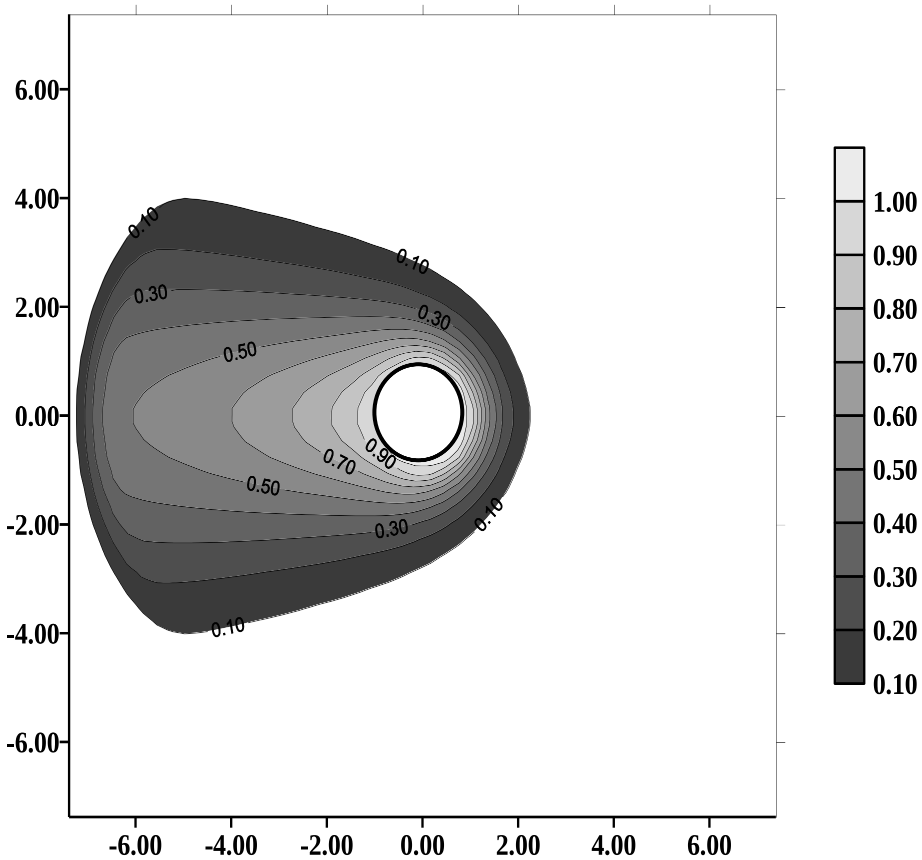


Fig.(7)Isothermal lines for Pe= 5

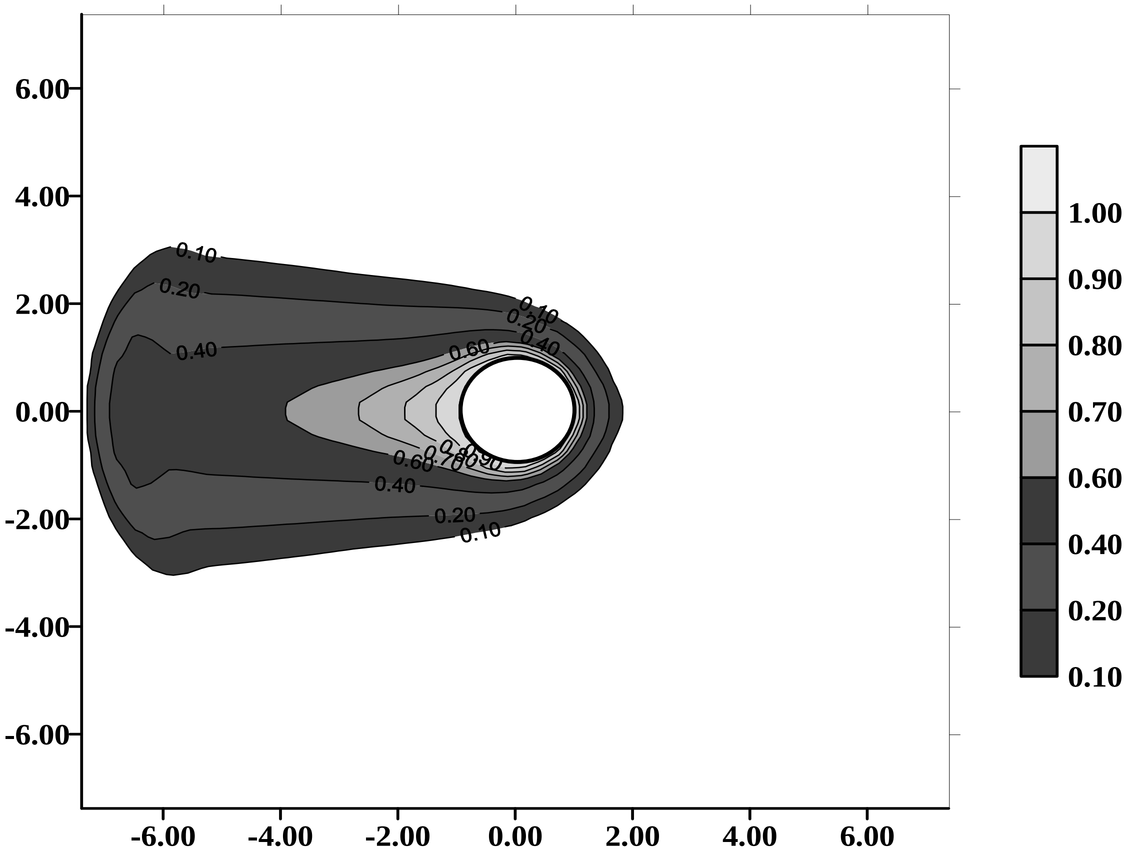


Fig.(8)Isothermal lines for Pe= 10

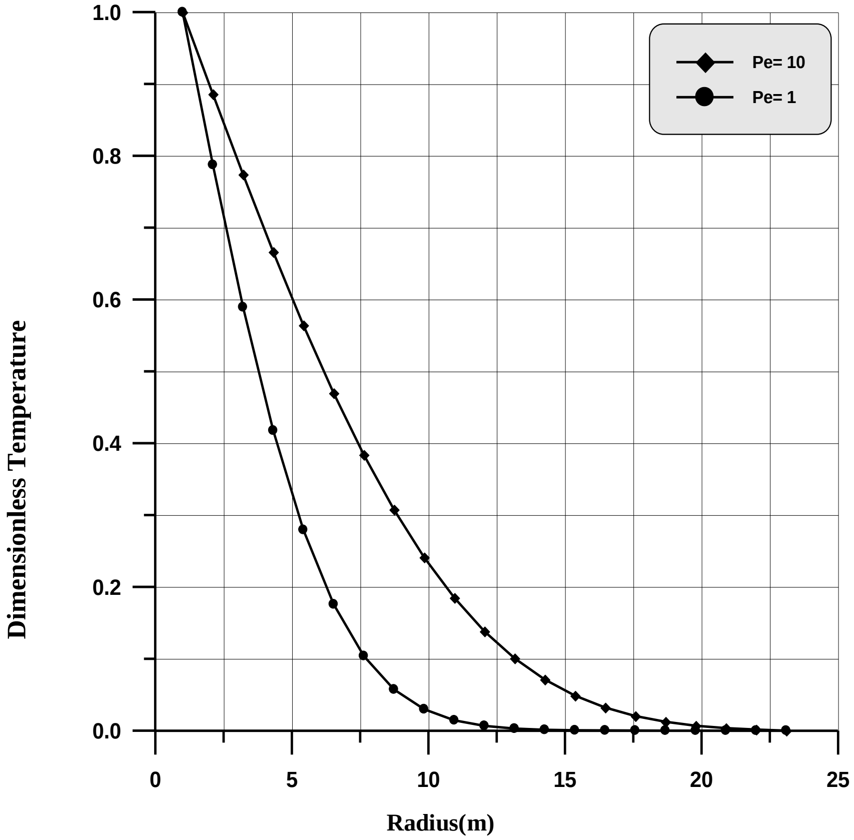


Fig. (9) Variation dimensionless temperature with radial distance for various Peclet number

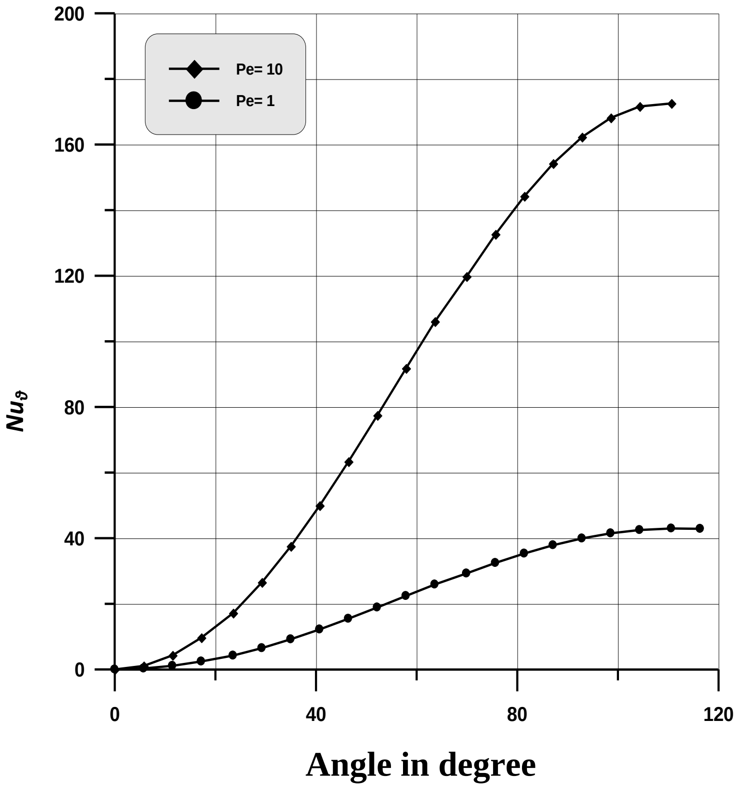


Fig.(10) Variation local Nusselt number with angle for various Peclet number

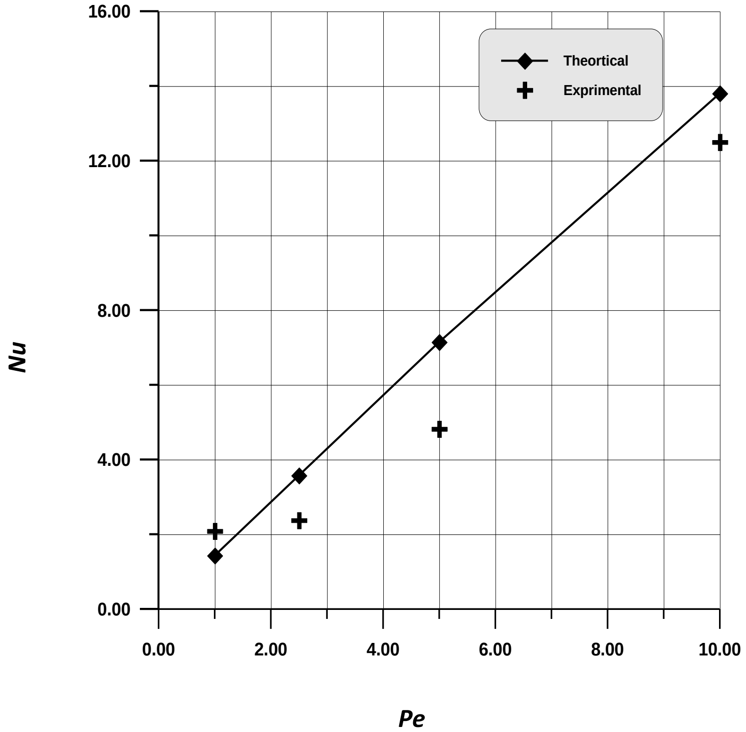


Fig.(11) Comparison Betwwen Theoretical and Experimental data

Conclusion

Forced convection from a horizontal cylinder embedded in a saturated porous media has been numerically and experimentally investigated.

1- The local Nusselt number increase with increase Peclet number.
2- The local Nusselt number decrease from highest value at $\theta=180^o$ and reaches a minimum at $\theta=0^o$.
3- The results have been compared with the experimental data, and the comparison shows good agreement.

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دراسة نظرية وعملية لانتقال الحرارة بواسطة الحمل القسري من اسطوانة أفقية مغموسة في وسط مسامي

محمد أسماعيل عليوي (مدرس مساعد)

جامعة تكريت / كلية الهندسة / قسم الهندسة الميكانيكية

الخلاصة

أجريت دراسة نظرية وعملية لانتقال الحرارة بالحمل القسري من اسطوانة في جريان متعامد مغموسة في وسط مسامي مشبع. تضمن الجانب النظري اشتقاق العادلات الحاكمة للزخم والطاقة وباستخدام نموج دارسي للجريان ان المعادلتين الناتجتين هما من نوع القطع الناقص تم حلهما بطريقة الفرو قات المحددة مع انتظام درجة حرارة سطح الاسطوانة كشرط حدي ولأعداد بكلت تتراوح بين (١> Pe >١٠) .

الجانب العملي فقد تضمن بناء نموذج تجريبي متآلف من اسطوانة من النحاس قطرها (13 ملم) ملم مسخن من الداخل كهربائيا ومغموسة في حشوه مؤلفة من كرات زجاجية قطرها (12 ملم) وضعت في نفق هوائي واطئ السرعة. أوضحت النتائج النظرية والعملية على السواء إن معدل انتقال الحرارة يزداد بازدياد عدد بكلت .

الكلمات الدالة : حمل قسري ، وسط مسامي ،عدد بكلت