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Forced convective heat transfer in porous medium of wire screen meshes

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Abstract The hydrodynamic and heat transfer characteristics of a porous medium consisting of 20 wire screen meshes are examined theoretically and experimentally. The hydrodynamic experiments are conducted for the range of Reynolds number based on mean velocity and wire diameter from 1.5 to 12. The Ergun's constants and thermal dispersion coefficients are calculated in this range. Nusselt number variation is determined in both thermally developing and fully developed flows by the help of forced convection heat transfer experiments conducted for the uniform heat flux boundary condition. Correlation functions of Nusselt number in the range of fully developed and thermally developing, and of thermal entrance length are obtained from experimental data. Solutions of momentum and energy equations simulating the experimental model are obtained numerically with variable porosity and the anticipated thermal dispersion coefficients. The thermal dispersion coefficients well-adjusted to the experimental data are determined by numerical solution of the energy equation.

List of symbols

A	cross section area
A_s	surface area of wires in a screen layer
A_E, B_E	Ergun's constants
a, b	dimensionless coefficients in Eq. (1)
c	specific heat of fluid
Da_L	Darcy number $(= L/(K_v)^{1/2})$
d	wire diameter
h	heat transfer coefficient
K_v	viscous permeability
k_f, k_s	heat conduction coefficients of fluid and solid, respectively
k_T	total heat transfer coefficient $(= k_o + k_d)$
k_o	effective heat conduction coefficient
k_d	dispersion conductivity
$L \times W \times H$	length $\times$ width $\times$ height
ℓ	length of the arc AA' in Fig. 3
ℓ_e	length of a mesh
l_v	mixing length
m, n	dimensionless coefficients in Eq. (8)
N	layer number
Nu_b	fully developed Nusselt number $(= hH/k_f)$
Nu_x	local Nusselt number $(= hH/k_f)$
P	pressure
Pe_m	Peclet number $(= \mathrm{PrRe}_m)$
Pr	Prandtl number $(= \nu/\alpha)$
q''	uniform heat flux
Re_m	Reynolds number $(= u_m d/\nu_f)$
S_b	specific surface area
s	coefficient of shrinkage
T	temperature
T_{in}	inlet fluid temperature
T_w	heated wall temperature
$\langle T \rangle_b$	mean flow temperature
u_m	mean velocity
u	velocity
U	dimensionless velocity
V	volume
V_s	wire volume of a screen
x_g	thermal entrance length
x, y	Cartesian coordinates
X, Y	dimensionless Cartesian coordinates
α	thermal diffusion coefficient of fluid
γ	dimensionless dispersion coefficient in Eq. (11)
ε	porosity
ε_m	mean porosity
ε_c	core region porosity
θ	dimensionless temperature
θ_w	dimensionless heated surface temperature
θ_b	dimensionless flow temperature
μ	dynamic viscosity of fluid
ν	kinematic viscosity of fluid
ρ	density of fluid
ω	coefficient of mixing length
$\langle . \rangle$	volume average within the REV
$\langle . \rangle^f$	volume average with respect to the fluid volume within the REV
c	core region
in	inlet
f	fluid
s	solid

1 Introduction

The porous media have been used widely in many engineering fields such as heat storage, solid matrix heat exchangers, thermal insulation of buildings, cooling of electronic equipment, chemical reactors, space researches. Therefore, momentum and heat transfer through porous media have been the interest of many researches.

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Depending up the application, natural, forced and mixed convection through some porous media have been examined both theoretically and experimentally.

The continuum model of a porous medium, the need for a continuum approach, the representative elementary volume (REV) and its selection are described theoretically in Bear-Bachmat [1]. The conservation equations of porous medium have been found by integrating the conservation equations of fluid and solid phases according to the REV [1–3]. This approach of obtaining the conservation equations of porous media is widely accepted by many researchers because the motion in porous media can be described in terms of differentiable quantities, and the measurable macroscopic quantities can be used in solving field problems.

In momentum and energy equations obtained by the notion of continuum, there are some terms which are defined by the microscopic quantities such as pore velocity and pore temperature. These terms which show the drag on solid-fluid interfaces, thermal conductivity of the medium and thermal dispersion arising from flow must be defined by the mean quantities prevailing in the REV. Therefore, some models developed by using constitutive theory and experimental studies are needed.

In momentum equation, the drag term defined by averaged velocity is considered as Darcy or Ergun's equation obtained experimentally [3]. Three flow regimes are observed normally during the flow of fluid through porous medium. The first regime is Darcy flow regime in which the relationship between pressure drop and flow rate is a linear function. The second one is Forchheimer flow regime in which pressure drop is modeled as a second order polynomial function of flow rate such as Ergun's equation. And the third one is turbulent flow regime in which this relationship is also a second order polynomial function but the constants are different from the second one [4, 5]. Three dimensional forms of these experimentally obtained functions are widely assumed to describe the drag term of the momentum equation.

The dispersion term within the energy equation of porous medium can be viewed as a diffusive process. Therefore, it is related to the overall temperature gradient, and the proportionality coefficient between dispersional heat flux and temperature gradient is called thermal dispersion conductivity [6–12]. Hunt-Tien [6] consider that the thermal dispersion conductivity is a function of fluid density, fluid heat capacity, permeability and volume-averaged velocity. The dispersion coefficient of this function is determined theoretically for packed beds of spheres at both wall region and core region by Kuo-Tien [7]. In Cheng et al [8–12], dispersion conductivity is considered as a function of fluid density, fluid heat capacity, volume-averaged velocity and Van Driest's mixing length.

The presence of wall introduces a number of effects on fluid flow and heat transfer in a porous medium. These effects are considered by porosity function near the wall. Most of the investigators [8–10, 13–16] have assumed that the porosity variation may be approximated by an exponential function of distance.

In literature, there are many investigation on the packed-sphere beds, the studies on the other type of porous media such as fibrous media, foammetals, wire screen meshes are very few. Due to the high porosity of the medium made of wire screen mesh layers, the specific surface area is decreased, consequently the resistance against flow is also decreased. On this reason, the characteristics of forced heat convection through porous medium of wire screen layers are examined in this study. Viscous and inertial permeabilities, dispersion and heat convection coefficients of this medium are examined.

From hydrodynamic experiments, the relationship between pressure drop and flow rate is obtained in the range of Reynolds number from 1.5 to 12. The Reynolds number is based on wire diameter and mean velocity. Then, the viscous and inertial permeabilities of the medium are calculated by the help of this relationship.

Local Nusselt number variation based on the flow direction and Peclet number is determined experimentally. Then, some correlation functions of Nusselt numbers and the length of thermal entrance region are derived from experimental data. On the other hand, the fluid temperatures are measured in cross-section perpendicular to flow direction.

After this experimental study, the momentum and energy equations obtained by the notion of continuum model of porous media, are arranged to simulate the experimental setup. Then, they are solved numerically to determine the thermal dispersion coefficients of the medium. The solutions obtained by using these dispersion coefficients are compared with experimental data.

2
Experimental study

The apparatus shown in Fig. 1 is prepared to perform momentum and heat transfer experiments. This apparatus is made of 5 and 10 mm thick plexyglass except the upstream reservoir. This reservoir which has dimensions of $520 \times 520 \times 1200$ mm (length, width, height respectively) and made of fiberglass, has series of adjustable overflow dividers to provide a constant pressure head. The downstream reservoir measures $180 \times 180 \times 300$ mm. Water which is held at a constant level in the upstream reservoir flows through porous medium in the channel into the downstream reservoir. Water level in the downstream reservoir varies according to flow rate. When water column comes to a certain level at a flow rate adjusted by the exit valve, the exit pressure attains a fixed value, also.

The test section of the experimental apparatus is in the form of a channel which has dimension of $180 \times 50 \times 500$ mm (width, height, length respectively). The upper portion of the test section is removable as shown in Fig. 2. Porous block consisting of 20 wire screen meshes is placed under this upper portion outside the test section, and its height is adjusted by four brass bolts. The adjusted heights are 17.08 and 15.60 mm for the porous medium of 12 and 14 meshed screens, respectively. Hence, the heights of the entrance and exit channels of the test section are decreased to the heights of the porous media by placing plexyglass plates inside the channel. Thus, the channel from upstream to downstream reservoirs has same dimensions with porous media, approximately.

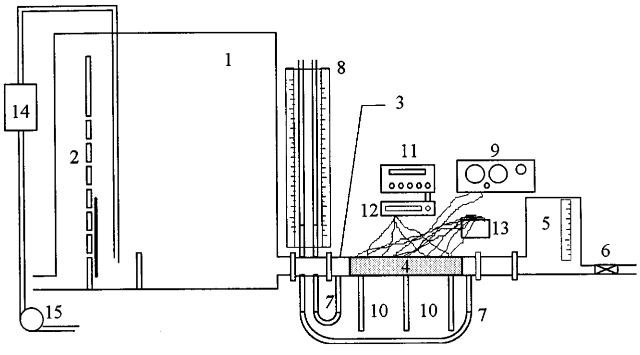

Fig. 1. Experimental Apparatus. *1* upstream reservoir; *2* dividers; *3* test section; *4* porous medium; *5* downstream reservoir; *6* valve; *7* pressure taps; *8* manometer; *9* DC source; *10* temperature probes; *11* millivoltmeter; *12* scanner; *13* ice bath; *14* water filter; *15* pump

There are two pressure taps at the inlet and outlet of the test section as shown in Fig. 1 and Fig. 2. They are connected to a vertical manometer. Thus, the pressures that can be determined by the water levels in the upstream and downstream reservoirs, are measured precisely by the help of the manometer. The set flow rate is determined by measuring the volume of water drained for a given period of time.

The medium is heated from the top in order to eliminate natural convection effects. Heating is supplied by five identical strip heaters put on the copper plate which is 15 mm thick. Each heater provides 200 watts maximum. These electrical heaters are supplied by an adjustable DC source. The uniform heat flux boundary condition can be maintained by supplying each heater with same voltage and current. The test section is insulated with asbestos cloth, asbestos plate and fiberglass.

The surface and fluid temperatures are measured by NiCr-Ni thermocouples. Twelve thermocouples were embedded in the copper plate close to the upper surface of the medium in order to measure the surface temperatures. Fifteen thermocouples were installed on the bottom surface. There are five thermocouples on the upper surface of the heaters and five at inner and upper parts of the insulation, also.

There are three taps of thermocouple probes in order to measure temperature profile in porous medium at the cross-section perpendicular to flow direction. Porous medium is drilled at the positions of these taps to move the probes. The sensitive point of the temperature probe that has same kind of thermocouple is surrounded with a small copper ring. This copper ring is placed on top of a ceramic tube which houses the thermocouple leads. An ice bath is used as the cold junction for all thermocouples. A millivoltmeter is used to measure the voltage of the signals from thermocouples.

The objective of these experiments is to obtain Ergun's constants by the help of the relationship between the measured pressure drop and flow rate. But the Ergun's equation valid for the porous medium consisting of 20 wire screen layers is needed. For this, the following general statement

$$\frac{\Delta P}{L} = \frac{\mu u_m}{a\varepsilon_m^3/S_b^2} + \frac{\rho u_m^2}{b\varepsilon_m^3/S_b} \tag{1}$$

obtained by a little modification on the Ergun's equation [17] of the packed beds of particles, can be used to define Forchheimer flow regime for all kinds of porous media. Here, S_b is the specific surface area of the medium, a and b are the dimensionless coefficients which need to be determined experimentally. $a = 36/150$ and $b = 6/1.75$ for the packed beds of spheres [17].

The volume and surface area of the solid phase inside the medium should be known for determination of S_b. The wire volume of a wire screen layer which is L in length and W in width is $V_s = \pi d^2 sLW/2\ell_e$ where $s = \ell/\ell_e$, the coefficient of shrinkage [18]. As shown in Fig. 3, ℓ_e is the distance between two adjacent pores and ℓ is the length of the arc AA'. The surface area of wires A_s is found as $2\pi dsLW/\ell_e$ easily. Thus, the specific surface area of the medium consisting of N layers is

$$S_b = \frac{NA_s}{V} = \frac{A_s}{V_s}(1-\varepsilon_m) = \frac{4}{d}(1-\varepsilon_m) \ . \tag{2}$$

If Eq. 2 is replaced in Eq. 1 and if it is written in dimensionless form,

$$P' = \left(-\frac{\Delta P}{L}\right)\frac{d^2}{\mu u_m} = A_E\frac{(1-\varepsilon_m)^2}{\varepsilon_m^3} + B_E\frac{1-\varepsilon_m}{\varepsilon_m^3}\mathrm{Re}_m \tag{3}$$

is obtained where $A_E = 16/d$ and $B_E = 4/d$ show the Ergun's constants, and $\mathrm{Re}_m = \rho u_m d/\mu$.

At the heat transfer experiments conducted in the condition of constant heat flux, the local Nusselt number of the heated surface is calculated as

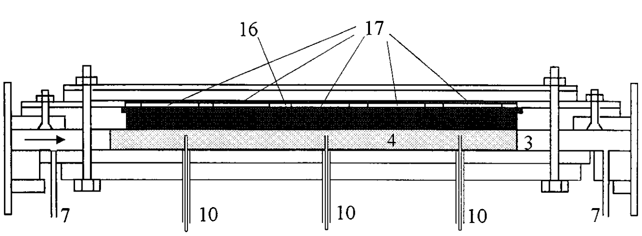

Fig. 2. Test section and upper portion *3* Test section; *4* porous medium; *7* pressure taps; *10* temperature probes; *16* copper plate; *17* heaters

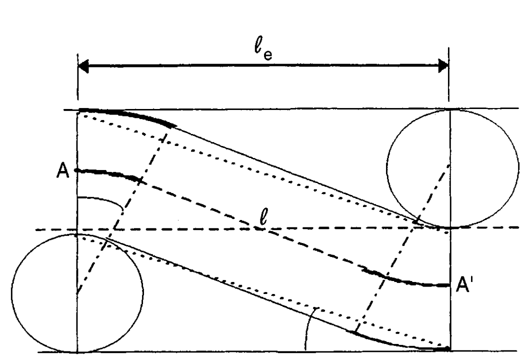

Fig. 3. Structure of a screen.

$$\mathrm{Nu}_x = \frac{hH}{k_f} = \frac{q''H}{k_f\left(T_w - \langle T\rangle_b\right)} \tag{4}$$

where $\langle T\rangle_b$ is the mean flow temperature, and it is defined as

$$\langle T\rangle_b = \frac{1}{u_m A}\iint_A \langle u\rangle\langle T\rangle \, \mathrm{d}A \tag{5}$$

$\langle T\rangle_b$ is obtained by the help of the measured inlet-outlet temperatures and the integration of the overall heat balance equation (lumped formulation) in the cross-section perpendicular to flow direction.

3
Governing equations

The experiments on momentum and heat transfer through porous medium prepared by adding layers on top of the others and by compressing altogether are considered as a flow problem in a rectangular channel as shown in Fig. 4. The diffusive process on z-direction can be ignored by comparing it to the y-directional diffusion because according to the channel dimensions, $H/W \approx 0.095$. Thus, this problem can be examined as a flow through porous medium between two parallel plates. The momentum and energy equations can be written as

$$0 = -\frac{\mathrm{d}\langle P\rangle}{\mathrm{d}x} + \mu\frac{\partial^2\langle u\rangle}{\partial y^2} - \frac{\mu\varepsilon}{K_v}\langle u\rangle - \frac{\beta\rho}{\sqrt{\varepsilon K_v}}\langle u\rangle^2 \tag{6}$$

$$\rho c\langle u\rangle\frac{\partial\langle T\rangle}{\partial x} = +\frac{\partial}{\partial y}\left[(k_o + k_d)\frac{\partial\langle T\rangle}{\partial y}\right] \tag{7}$$

where ρ, μ, c, k_o and k_d are density, dynamic viscosity and specific heat of fluid and effective thermal conductivity, thermal dispersion conductivity, respectively. The boundary conditions are

$$\langle u\rangle(0) = 0,\ \langle u\rangle(H) = 0$$

$$\frac{\partial\langle T\rangle(x,H)}{\partial y} = 0$$

$$-\left[k_o(0) + k_d(0)\right]\frac{\partial\langle T\rangle(x,0)}{\partial y} = q'', \quad \langle T\rangle(0,y) = T_g$$

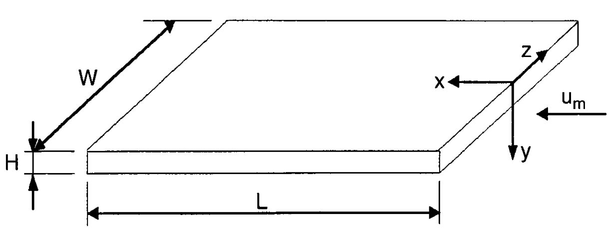

Fig. 4. Shape of the porous channel

The porosity variation near the wall region of porous medium is considered with the following porosity function. The values of ε_c, n and m are presented in Table 1.

$$\varepsilon = \varepsilon_c\left[1 + n\exp\left(\frac{-my}{d}\right)\right], \quad 0 \le y \le H/2$$
$$\varepsilon = \varepsilon_c\left[1 + n\exp\left(\frac{-m(H-y)}{d}\right)\right], \quad H/2 \le y \le H \tag{8}$$

K_v viscous permeability and β coefficients are obtained as

$$K_v = \frac{a\varepsilon^3}{S_b^2} = \frac{\varepsilon^3 d^2}{A_E(1-\varepsilon)^2}, \quad \beta = \sqrt{a}/b = B_E/\sqrt{A_E} \tag{9}$$

from Eq. 1 and Eq. 2.

The effective thermal conductivity of the porous medium of wire screen meshes at stagnant condition is considered as

$$\frac{k_o}{k_f} = \varepsilon + (1-\varepsilon)\lambda_{sf} - \frac{\varepsilon(1-\varepsilon)(1-\lambda_{sf})^2}{2 - \varepsilon(1-\lambda_{sf})}, \quad \lambda_{sf} = \frac{k_s}{k_f} \tag{10}$$

according to the Rayleigh-Ivanovski model [19, 20]. This statement obtained for medium of cylindrical particles gives the value of effective thermal conductivity of the medium of non-sintered screen meshes with 3% error in the range $\lambda_{sf} < 30$ [19].

Thermal dispersion conductivity k_d is considered as

$$\frac{k_d}{k_f} = \gamma\frac{\langle u\rangle}{\alpha}\ell_v(y/d) \tag{11}$$

by comparing with turbulent flow analogicaly. Here, ℓ_v

$$\ell_v = 1 - \exp\left(\frac{-y}{\omega d}\right) \tag{12}$$

shows the mixing length [8–12] where γ and ω are dimensionless dispersion coefficients.

Eq. 6 and Eq. 7 can be written in non-dimensional form as

$$A_o A(\bar{\varepsilon}) + B_o B(\bar{\varepsilon})\frac{\mathrm{d}^2 U}{\mathrm{d}Y^2} - U - C_o C(\bar{\varepsilon})\mathrm{Re}_m U^2 = 0 \tag{13}$$

$$U\frac{\partial\theta}{\partial X} = \frac{\partial}{\partial Y}\left(\frac{k_T}{k_f}\frac{\partial\theta}{\partial Y}\right) \tag{14}$$

using the following definitions:

$$X = \frac{x}{d\mathrm{Pe}_m}, Y = \frac{y}{d}, U = \frac{\langle u\rangle}{u_m},$$

$$u_c = -\frac{K_{vm}}{\mu}\frac{d\langle P\rangle^f}{\mathrm{d}x}, \theta = \frac{\langle T\rangle - T_{\mathrm{in}}}{q''d/k_f}$$

$$\overline{K} = K_v/K_{vm}, \bar{\varepsilon} = \varepsilon/\varepsilon_m, \quad A_o = u_c/u_m, \quad A(\bar{\varepsilon}) = \overline{K}$$

Table 1. Characteristics of the media consisting of 20 wire screen layers (wire diameter = 0.5 mm) [21]

	Mesh No	Mean Porosity ε_m	Core region porosity ε_c	n	m	REV's height δ/d	Total height H [mm]
1	12	0.7832	0.7812	0.2801	1.90	6.771	17.08
2	14	0.7320	0.7280	0.3763	2.15	6.148	15.60

$$B_o = \frac{K_{vm}}{\varepsilon_m d^2},\ B(\bar{\varepsilon}) = \overline{K}/\bar{\varepsilon},\ C_o = \frac{\beta\sqrt{K_{vm}}}{d\varepsilon_m\sqrt{\varepsilon_m}},$$
$$C(\bar{\varepsilon}) = \sqrt{B(\bar{\varepsilon})}/\bar{\varepsilon},\ k_T = k_o + k_d$$

The non-dimensional boundary conditions are

$$U(0) = 0,\ U(H/d) = 0$$
$$\frac{\partial\theta(X,H/d)}{\partial Y} = 0, \quad -\frac{k_T(0)}{k_f}\frac{\partial\theta(X,0)}{\partial Y} = 1, \quad \theta(0,Y) = 0$$

The momentum equation given by Eq. (6) is a non-linear differential equation with variable coefficients. After the second order differential term of this equation is written in the form of finite difference, Eq. (6) is solved iteratively using Newton-Raphson method. The solution well-adjusted with experiments is found out by comparing the flow rate calculated by numerically in 800 grids, with experimental value.

The numerical formulation of the energy equation given Eq. (7) is obtained by control volume method. The energy equation is solved with the anticipated values of dispersion coefficients, γ and ω. The numerical solution is repeated to find out the agreeable values of γ and ω by comparing the experimental data of Nusselt number, heated surface temperature and temperature profiles.

4
Results and discussion

The highest mean velocity, u_m, measured in the hydrodynamic experiments was 1.4654×10^{-2} m/s. If the pore velocity of fluid is calculated as u_m/ε by Dupuit-Forchheimer relation, Reynolds number based on the pore velocity and wire diameter is found as 10.7 approximately. Thus, it is clear that flow inside pores is laminar according to the explanations of Fand et al. [5]. Disjointing of fluid from solid surfaces and passing to the turbulent flow regime are not expected at these velocities.

The experiments done in the range of $1.5 < \mathrm{Re}_m < 12$ cover the beginning and developing regions of Forchheimer flow regime. Darcy regime in which $\Delta P/L$ changes linearly with u_m could not be observed in this study because of high error in measurement of pressure. Running experiments at high velocities were not needed because the working range was limited with the range of Reynolds numbers in heat transfer experiments.

In Fig. 5, the data of hydrodynamic experiments presented with dimensionless parameters are shown for two media. The curve-fit equation of experimental data is

$$P' = \left(-\frac{\Delta P}{L}\right)\frac{d^2}{\mu u_m} = 3.0972(\pm 0.1432)\mathrm{Re}_m + 10.8462(\pm 0.9924) \tag{15}$$

for first medium, and

$$P' = \left(-\frac{\Delta P}{L}\right)\frac{d^2}{\mu u_m} = 3.1493(\pm 0.1859)\mathrm{Re}_m + 58.8837(\pm 1.4754) \tag{16}$$

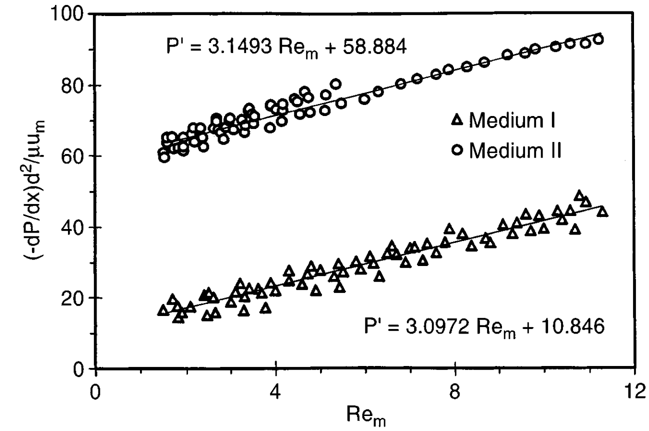

Fig. 5. Variation of dimensionless pressure gradient with respect to Reynolds number

for second medium. When these equations are compared with Eq. 3, the Ergun's constants are found easily as follows.

$A_E = 110.8585 \pm (\%9.15),\ \ B_E = 6.8632 \pm (\%4.62)$

for first medium,

$A_E = 321.5592 \pm (\%2.51),\ \ B_E = 4.6091 \pm (\%5.90)$

for second medium.

The values of mean porosity presented at Table 1 have been used in the calculation of Ergun's constants. Viscous permeability K_v and β coefficients of each medium can be found easily by using these constants. Velocity domain calculated by the numerical solution of Eq. 13 with variable porosity and with Ergun's constants, is used in the solution of energy equation.

The variation of local Nusselt number which is found out experimentally for the constant heat flux boundary condition is shown in Fig. 6 and Fig. 7 for the first and second media, respectively. The theoretical solution well-adjusted to the Nu_x variation which is found out experimentally at three different flow rates in the first medium, is provided by considering the values of the thermal dispersion coefficients γ and ω as 0.08 and 1.5, respectively.

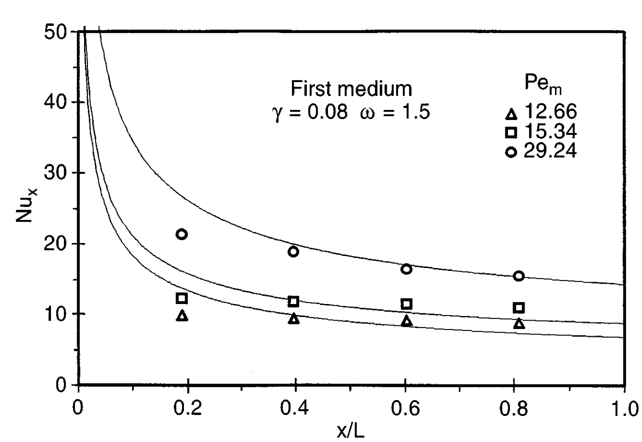

Fig. 6. $\mathrm{Nu}_x - x/L$ variation calculated by considering $\gamma = 0.08$ and $\omega = 1.5$ for the first medium

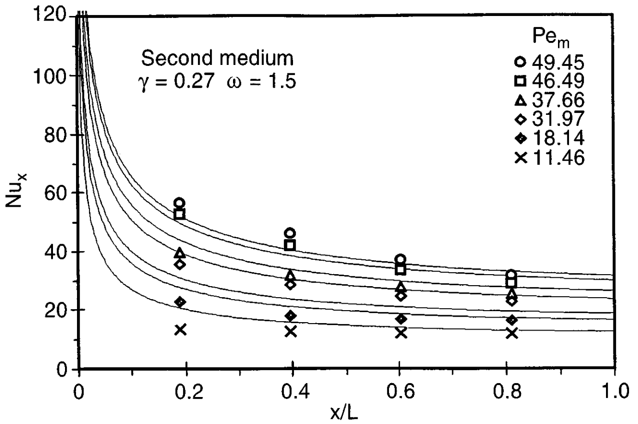

Fig. 7. $\mathrm{Nu}_x - x/L$ variation calculated by considering $\gamma = 0.27$ and $\omega = 1.5$ for the second medium

For the second medium, the theoretical solution by taking γ and ω as 0.27 and 1.5, respectively and the experimental data are shown in Fig. 7.

The variation of heated surface temperature obtained by the solution of energy equation with these thermal dispersion coefficients is shown in Fig. 8 and Fig. 9 together with experimental data for the first and second media, respectively.

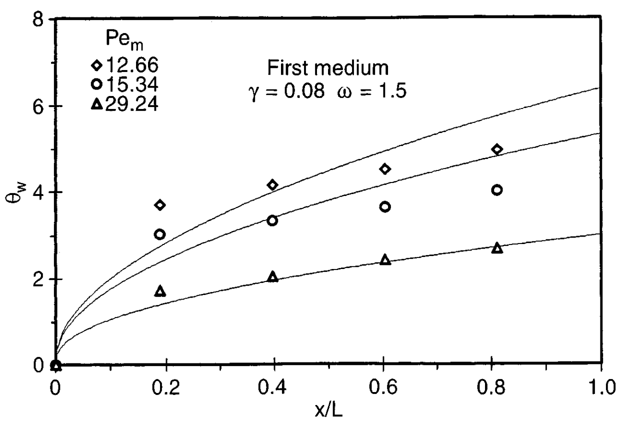

Fig. 8. The variation of $\Theta_w - x/l$ calculated with $\gamma = 0.08$ and $\omega = 1.5$ for the first medium

Temperature profiles of the second medium at the cross-section perpendicular to flow direction were measured by temperature probes for three different flow rates. One turn of the probe corresponds to the linear displacement of 1.4 mm and temperature measurement is recorded at every quarter of a turn. Approximately eight measurements were made within a REV. Thus, the mean was the average of the measurements in the REV. As shown in Fig. 10, temperature values are not present in the region of the half REV near the wall as a result of averaging the temperature data with respect to the REV. In fact, temperature values measured beginning from the copper surface are represented in the mean temperature value nearest to the wall. The theoretical solutions of temperature profiles obtained by considering γ and ω as 0.27 and 1.5 are shown in Fig. 10 for $x = 245$ mm. γ and ω coefficients which give better agreement of theoretical results with experimental data of temperature profiles at $x = 245$ could be found. But, at this time, theoretical Nu_x and T_w would deviate from their experimental data.

At the downstream locations of 45 mm, 145 mm and 245 mm, fluid temperature distribution across the medium are shown in Fig. 10, Fig. 11 and Fig. 12 for the Peclet numbers of 46.49, 18.14 and 11.46, respectively. In these figures, it is seen that the theoretical results are in reasonable agreement with experimental data except the results for $x = 245$ mm. At lower Peclet numbers, the agreement between theoretical and experimental data is not as good. At the downstream location where thermal entrance effects prevail, the agreement of experimental and theoretical data was not also as good for all the Peclet numbers considered, while the agreement gets better downstream for the mentioned Peclet numbers.

Thermal entrance effect prevails up to the channel exit at high flow rates as seen by the variation of Nu_x with respect to x/L in Fig. 6 and Fig. 7. Fully developed flow conditions do not occur and the boundary layer flow continues along the length of the channel. Consequently, the fully developed Nusselt number can not be determined

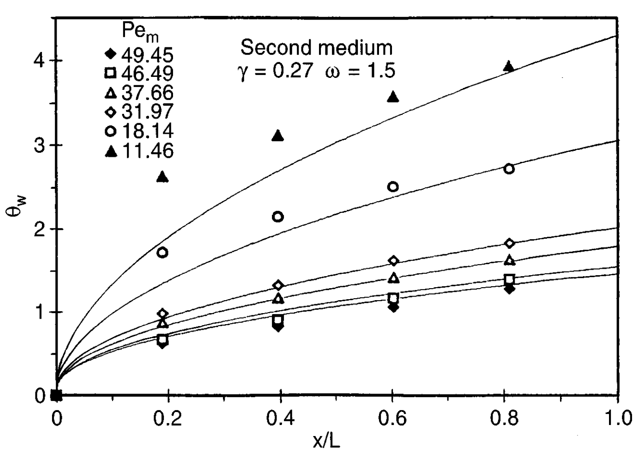

Fig. 9. The variation of $\Theta_w - x/l$ calculated with $\gamma = 0.27$ and $\omega = 1.5$ for the second medium

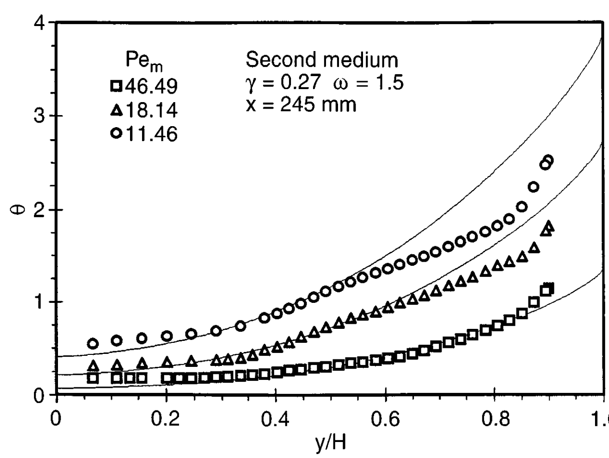

Fig. 10. Fluid temperature profiles at the location $x = 245$ mm for different Peclet numbers. $\gamma = 0.27$ and $\omega = 1.5$

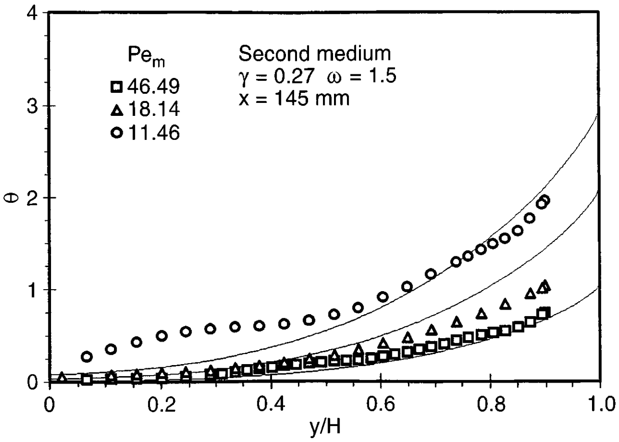

Fig. 11. Fluid temperature profiles at the location $x = 145$ mm for different Peclet numbers. $\gamma = 0.27$ and $\omega = 1.5$

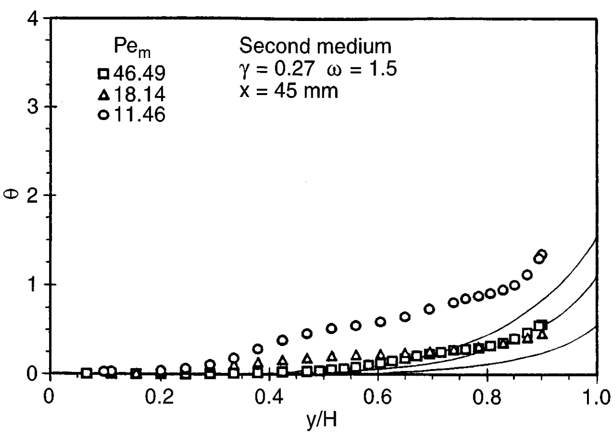

Fig. 12. Fluid temperature profiles at the location $x = 45$ mm for different Peclet numbers. $\gamma = 0.27$ and $\omega = 1.5$

directly by experimental data, so that, a curve fit is applied to the heated surface temperature data, and Nusselt numbers are calculated up to $2L$ by using the extrapolated surface temperature data. If Darcy number of the porous medium is defined as $\mathrm{Da}_L = L/(K_{vm})^{1/2}$ and if the variation of fully developed Nusselt number data with respect to Pe_m are reduced using $\mathrm{Pe}_m(\mathrm{Da}_L)^{1/2}$, the data collapse to a single curve in Fig. 13. A correlation function for this data can be obtained as

$$\mathrm{Nu}_b = 0.2211(\mathrm{Pe}_m\sqrt{\mathrm{Da}_L})^{0.5992}, \quad 10 \le \mathrm{Pe}_m \le 50 \tag{17}$$

as seen in Fig. 13.

The thermal entrance length is defined as the axial distance required to achieve a value of local Nusselt number, which is 1.05 times the fully developed Nusselt number value. A correlation function between the thermal entrance length and Peclet number is obtained as follow for the second medium experimental data as seen in Fig. 13.

$$\frac{x_g}{L} = 0.0402\mathrm{Pe}_m^{0.8368}, \quad 10 < \mathrm{Pe}_m < 50 \tag{18}$$

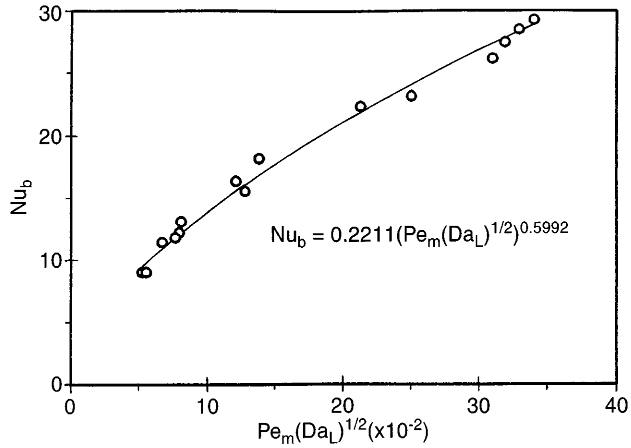

Fig. 13. Variation of Nusselt-number with respect to $\mathrm{Pe}_m(\mathrm{Da}_L)^{1/2}$

On the other hand, thermally developing Nusselt number data are correlated with respect to the axial distance and Peclet number as

$$\mathrm{Nu}_x = 1.4842\mathrm{Pe}_m^{0.7884}\,(x/L)^{-0.2751}, \quad 10 \le \mathrm{Pe}_m \le 50 \tag{19}$$

5 Conclusion

Hydrodynamic and forced heat convection characteristics of the porous media of wire screen layers have been investigated for a limited range of parameters in this study. Ergun's constants have been obtained by the relationship between pressure drop and flow rate measured in the range of Re_m form 1.5 to 12. The curve-fit equations of hydrodynamic experiments are compared with Eq. 3 which is obtained from Eq. 1, in order to find Ergun's constants. Eq. 1 can be considered as a general statement for all porous media. If the specific surface area S_b of a medium can be determined from geometrical structure of the medium, the Ergun's equation belonging to that medium is obtained, easily.

The definition of Eq. 1 with respect to the REV, and putting it in three dimensional form can be done by the method explained in Vafai-Tien [3]. Thus, K_v and F coefficients of the formulation of Vafai-Tien are found as $K_v = a\varepsilon^3/S_b$ and $F = (\sqrt{a}/b)/(\varepsilon\sqrt{\varepsilon})$.

In this range of Re_m, the thermal dispersion coefficients which give approximate agreement of theoretical results with experimental data of local Nusselt number, heated surface temperature and temperature profiles across the medium have been determined. In addition, correlation functions have been obtained for the experimental data of the fully developed Nusselt number, local Nusselt number and thermal entrance length.

It is hopped that this study will be useful guide for future studies of wire screen mesh or similar media. However, the present study could be extended to cover a wide range of independent parameters such as mesh numbers, compression ratios, wire diameters etc. in order to determine hydrodynamic and heat transfer characteristics of the medium.

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