Int. J. Heat Mass Transfer. Vol. 21, pp. 985–989

0017-9310/78/0701-0985 $02.00/0

HEAT TRANSFER IN A CONFINED RECTANGULAR CAVITY PACKED WITH POROUS MEDIA

NOBUHIRO SEKI, SHOICHIRO FUKUSAKO and HIDEO INABA
Department of Mechanical Engineering, Faculty of Engineering,
Hokkaido University, Sapporo 060, Japan

(*Received* 25 *August* 1977)

Abstract—This paper presents an experimental investigation of convective heat transfer in a confined rectangular cavity packed with porous media, on the opposing vertical walls of which different temperatures are imposed. Measurements are made for each of two kinds of solid particles using three kinds of fluids, i.e. water, transformer oil and ethyl alcohol. The present experiments cover a wide range of Rayleigh number Ra^* between 1 and 10^5, Prandtl number Pr^* between 1 and 200 and geometrical aspect-ratio H/W between 5 and 26. The experimental results indicate that Nusselt number Nu^* is correlated by the following relationship:

$$Nu^* = 0.627\, Pr^{*0.130}\, (H/W)^{-0.527}\, Ra^{*0.463}.$$

NOMENCLATURE

- a_m, thermal diffusivity, $\lambda_m/(\rho c_p)_f$;
- c, constant, defined by equation (1);
- c_p, specific heat at constant pressure;
- d, diameter of solid particle;
- Da, Darcy number, k/W^2;
- g, gravitational acceleration;
- H, height of rectangular cavity;
- k, permeability;
- Nu^*, Nusselt number, $\alpha W/\lambda_m$;
- Nu, Nusselt number for common fluid layer, $\alpha W/\lambda_f$;
- Pr^*, Prandtl number, ν_f/α_m;
- Ra^*, Rayleigh number, $g\beta_f \Delta T W k/\nu_f \alpha_m$;
- T, temperature;
- ΔT, temperature difference between hot and cold walls, $(T_h - T_c)$;
- T^+, non-dimensional temperature, $(T-T_c)/(T_h-T_c)$;
- W, width of rectangular cavity;
- X, distance from hot wall;
- Y, distance from bottom of a rectangular cavity.

Greek symbols

- α, coefficient of heat transfer;
- β, cubical expansion-coefficient;
- λ, thermal conductivity;
- λ_m, thermal conductivity of porous medium (without convection);
- ν, kinematic viscosity;
- ρ, density;
- ε, porosity.

Subscripts

- c, refers to cold wall;
- f, refers to fluid;
- h, refers to hot wall;
- m, refers to mixing of solid particle and fluid;
- s, refers to solid.

1. INTRODUCTION

NATURAL convective heat transfer of fluid in a confined rectangular cavity has been studied both experimentally and analytically by various investigators so far. However, there are a few studies of the cavity packed with a porous medium, which are of practical importance in a process design of a nuclear power reactor core with multishield structure containing porous insulating materials and of the regenerative heat exchanger containing porous materials. Schneider [1] investigated experimentally the natural convective heat transfer through the granular material under the conditions of fixed height and width of a rectangular cavity. Bories and Combarnous [2] studied the effect of the dimension of the cavity on heat transfer in a vertical porous layer in the limited range of $10^2 \sim 10^3$ of Rayleigh number, Ra^*. However, both investigators did not clarify the effects of the dimension of a rectangular cavity and the physical properties of porous media on heat transfer through the vertical porous layer. Analytical studies of natural convective heat transfer in the vertical porous layer were reported by only several investigators [3–7] for laminar two-dimensional flow in the small range of $Ra^* < 10^3$ in which the Darcy's law can be available.

The purpose of the present study is to examine experimentally the effects of the dimension of a rectangular cavity and the physical properties of porous media on heat transfer in a confined rectangular cavity packed with porous media, whose opposing vertical walls have different but uniform temperature while upper and bottom ones insulated.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The experiments are performed by using three kinds of rectangular test-cells, that is, 22, 57 and 116 mm (width) × 571 mm (height) in section area by 250 mm in depth. The schematic diagram of the experimental apparatus is depicted in Fig. 1. The main parts of the

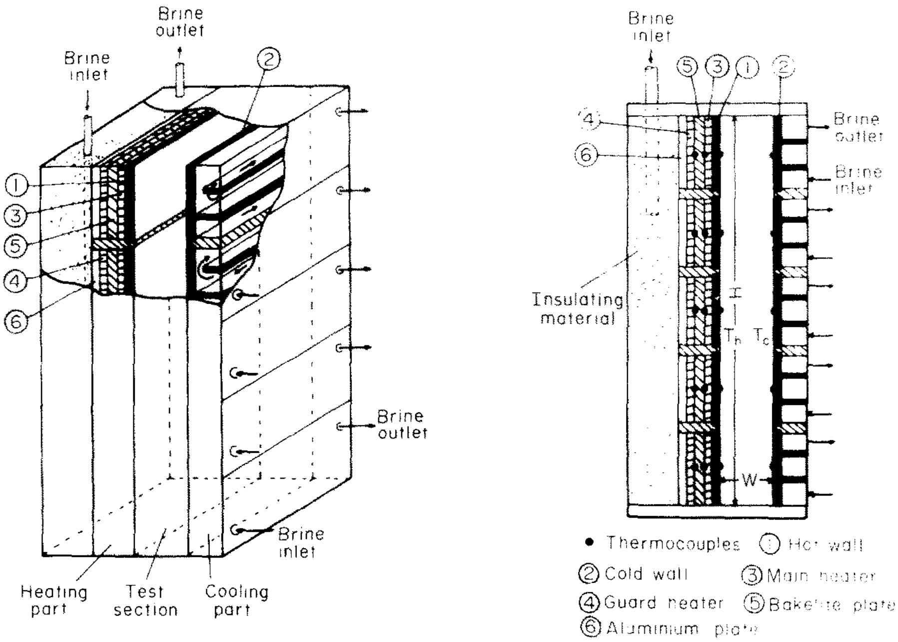

FIG. 1. A schematic diagram of experimental apparatus.

experimental apparatus consist of a test section and heating and cooling parts. The test section in which porous media are packed is indicated at the middle of the right side in Fig. 1.

Surface temperature of the hot wall (copper plate of 5 mm in thickness) is maintained at uniform temperature by using five controllable main heaters. The guard heaters are mounted outside the main heaters across a bakelite plate in order to minimize the heat-loss from the main heaters to the environment.

Surface temperature of the cold wall (copper plate of 5 mm in thickness) is kept at uniform temperature by temperature-controlled coolant (brine) in each of five separate chambers attached to the outside of the cold wall. Eighteen Cu–Co thermocouples (0.3 mm in diameter) are attached on the hot and the cold walls, moreover, additional ten thermocouples are inserted between the main and the guard heaters. Five small proves (stainless steel pipe of 0.8 mm in diameter) with Cu–Co thermocouple of 0.1 mm in diameter are set in the X-direction at the middle height of the porous layer in order to measure the temperature distribution.

Measurements are made for two solid particles (glass beads and iron balls) of various diameters with three kinds of fluids, that is, water, transformer oil and ethyl alcohol. The porosity ε obtained in the present study is in the range of $0.36 < \varepsilon < 0.41$. The liquid used is carefully injected from a nozzle of the bottom wall into the test-cell in order to avoid the mixing of undesirable air bubble. The experimental apparatus is covered with an insulating material (100 mm in thickness) and placed inside a temperature-controlled room to prevent a thermal disturbance from the environment. Measurements of heat transfer are carried out after the thermal condition of the test section has reached a steady state. It takes about $5 \sim 10$ h to reach the thermal steady state. The net heat transferred through the porous layer is estimated by subtracting the heat-loss from the total heat obtained by summing up the electrical inputs to five main heaters and the net mean heat flux q is evaluated by dividing the net heat by total heating area. To evaluate the heat-loss from the apparatus, careful preparatory experiments are performed by placing the apparatus filled with fluids of known thermal conductivity in a horizontal position. From the results of preparatory experiments, the heat-loss from the apparatus is estimated within $\pm 5\%$ of the input. Before the running, the thermal conductivity of the porous medium λ_m is necessarily measured. The physical properties of the porous medium adopted to compute the various parameters are determined by the average temperature of the surface temperatures of hot and cold walls.

3. EXPERIMENTAL RESULTS AND DISCUSSIONS

Temperature distributions in the X-direction are measured at the middle height of the porous layer in order to examine a behavior of heat transmission through the vertical porous layer. Figure 2 shows the typical non-dimensional temperature distributions for the aspect-ratio of $H/W = 10$ and the width of cavity of $W = 57$ mm. The temperature distributions for water–glass beads as shown in Fig. 2(a) are remarkably different from those for transformer oil–glass beads as shown in Fig. 2(b). These differences suggest that the physical properties of fluids have considerable effect on the heat transfer through the vertical porous layer.

The temperature distributions for ethyl alcohol–iron balls are shown in Fig. 2(c). It can be seen from the figure that the intensity of the convection is not so strong. It owes to the larger thermal conductivity of iron balls than that of glass beads. Judging from the temperature distributions as shown in Fig. 2, it is

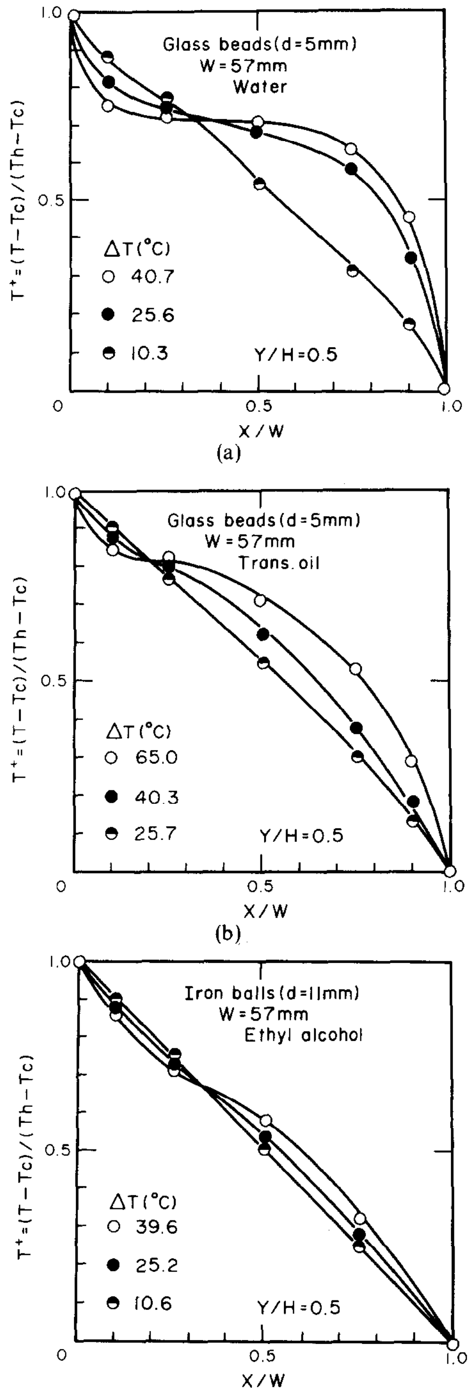

FIG. 2. Temperature distributions in a vertical porous layer for $H/W = 10$, $W = 57$ mm, (a) water–glass beads; (b) transformer oil–glass beads; (c) ethyl alcohol–iron balls.

clear that the thickness of thermal boundary layer developing near the hot wall is larger than that developing near the cold wall. This difference between the thickness of the thermal boundary layers might be explained by the fact that an ascending flow with a thin boundary-layer thickness near the hot wall comes into collision with the upper insulated wall and consequently the boundary-layer thickness becomes large in the vicinity of the upper insulated wall. The flow along the upper insulated wall is separated into several parts by solid particles near the cold wall, therefore, a descending flow with a thick boundary-layer thickness appears near the cold wall.

One of the interesting characteristics, as shown in Fig. 2, is that the position of intersection between the temperature distribution line having a large temperature difference and that having a small one varies depending on the combination of fluid and solid particle. That is, it is observed that the position of intersection for transformer oil–glass beads in Fig. 2(b) approaches to the hot wall ($X/W = 0$) closer than that for water–glass beads in Fig. 2(a). This difference of the position of the intersection in the present experiments might be caused by the fact that viscosity dependency of temperature for transformer oil (about 700% decrease from 20 to 80°C) is larger than that for water (about 230% decrease from 10 to 50°C). Moreover, the position of intersection for ethyl alcohol–iron balls in Fig. 2(c) approaches to the center portion of porous layer closer than that for the above-mentioned two cases. Such a behavior might be understood by the reason that viscosity dependency of temperature for ethyl alcohol is small (about 200% decrease from 10 to 50°C) and also the contribution of natural convection to heat transfer through the porous layer is small due to the large thermal conductivity of iron ball.

From the above-mentioned temperature distributions in the vertical porous layer, it is clear that the heat transfer through the vertical porous layer is remarkably affected by various kinds of fluids or solid particles as well as various dimensions of rectangular cavities. The following functional relationship given by a product of powers is derived from the result of the present dimensional analysis of heat transfer through a vertical porous layer;

$$Nu^* = c\,Pr^{*l}(H/W)^m Ra^{*n}. \qquad (1)$$

Where, Nu^* is Nusselt number defined as $Nu^* = \alpha W/\lambda_m$, Pr^* is Prandtl number ($Pr^* = \nu_f/a_m$) and Ra^* is Rayleigh number ($Ra^* = g\beta_f \Delta T W k/\nu_f a_m$).

All of the constant and exponents are determined by using the root-mean-square method, whose deviation is within $\pm 7\%$ in the present study. The experimental correlation obtained is:

$$Nu^* = 0.627\,Pr^{*0.130}(H/W)^{-0.527}Ra^{*0.463},$$
$$1 < Pr^* < 200,\ 5 < H/W < 26,\ 10^2 < Ra^* < 10^4. \qquad (2)$$

Figure 3 shows the present data and Schneider's data [1] together with the present experimental correlation, equation (2), indicated as a solid line. In this figure, one can see that the obtained results in the range of $10^2 < Ra^* < 10^4$ agree fairly well with the solid-line and when Ra^* is less than 10^2, the experimental data indicated as broken lines lie above the solid line. From these facts, it could be concluded that the conductive heat transfer is dominant only in the range of $Ra^* < 10^2$, because the value of $Nu^*/[Pr^{*0.130}(H/W)^{-0.527}]$ remains constant though Ra^* is varied in this range.

The point of intersection between the solid line and

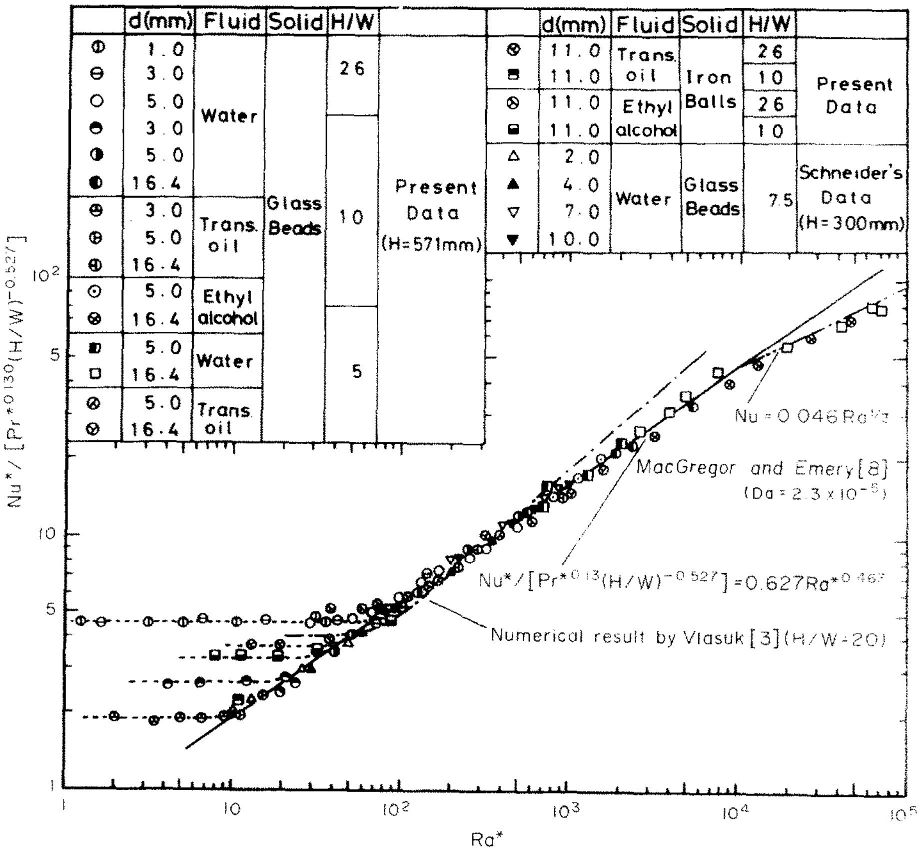

FIG. 3. The relationship between $Nu^*/[Pr^{*0.130}(H/W)^{-0.527}]$ and Ra^*.

the broken lines means a critical value for the onset of natural convection. It also should be noticed that such critical values of Ra^* increase with increasing aspect-ratio H/W under the same combination of fluid and solid particle, while, the critical value for ethyl alcohol–iron balls (□) is larger than that for transformer oil–glass beads (○) under the fixed aspect-ratio $H/W (= 10)$. From these results, it might be concluded that the variation of the critical value is closely related with the physical properties of fluid and solid particle as well as the dimension of a rectangular cavity.

One of interesting results as shown in Fig. 3 is that the experimental data are plotted below the solid-line in the range of $Ra^* > 10^4$, but they lie close to the experimental prediction by MacGregor and Emery [8] of the turbulent heat transfer for the common fluid layer without porous media, if their experimental prediction $[Nu = 0.046 Ra^{1/3}]$ is rearranged by the present functional relationship under $d = 16.4$ mm, $H/W = 5$, Darcy's number $Da = 2.3 \times 10^{-5}$ and $\lambda_m/\lambda_f = 1.13$ (as using water–glass beads). Such a behavior might be explained as follows: in case of using a spherical solid particle as a solid particle, the porocity ε near the wall surface would be almost unit due to one particle having only one contact point with the wall surface. From the porosity distribution in packed bed reported by Yagi *et al.* [9], it is clear that the porosity distribution from the wall surface to a distance of radius of a solid particle in the direction of core portion of porous layer is varied from 1 to 0.45 and that inside of porous layer becomes constant. In the present experimental study using solid particles having large diameter ($d = 16.4$ mm), the core with large porosity would be more extended to a long distance from the wall surface.

Moreover, the thickness of boundary layer developing near the vertical wall decreases with increasing heat flux, when the range where fluid flows is to be limited to the range having a large porosity near the vertical wall and the behavior of heat transfer in the porous layer might be similar to that in the common fluid layer without porous medium.

The numerical results of Vlasuk [3] are also shown in Fig. 3. From the comparison of the experimental data with the numerical result, as shown in Fig. 3, it is clear that the latter shows good agreement with the former for $Ra^* < 10^3$. This agreement might mean that his analytical prediction based on Darcy's law is effective in the range of $Ra^* < 10^3$. However, his analytical prediction becomes larger than the experimental value in the range of $Ra^* > 10^3$. This difference between the experimental data and his prediction might indicate that the Darcy's law could not be applied in the physical model of porous layer adopted in the range of $Ra^* > 10^3$. That is, for $Ra^* > 10^3$, the turbulent mixture of fluid or the viscous force which is proportional to a square of fluid velocity would become an important factor from the standpoint of evaluating the heat transfer in the porous layer.

4. CONCLUSION

The experiments are carried out by the rectangular cavities packed with porous media, whose opposing vartical walls have different but uniform temperature, while the upper and bottom walls are insulated. It is clear that the Prandtl number Pr^* as well as the aspect-ratio H/W has a significant effect on the heat transfer through the vertical porous layer. The following correlation is derived from the experiments:

$$Nu^* = 0.627\, Pr^{*0.130} (H/W)^{-0.527} Ra^{*0.463}$$

$$1 < Pr^* < 200, \quad 5 < H/W < 26, \quad 10^2 < Ra^* < 10^4.$$

REFERENCES

1. K. J. Schneider, Investigation of the influence of free thermal convection on heat transfer through granular material, 11th *International Congress on Refrigeration*, Munich, West Germany, Paper No. II-4 (1963).
2. S. A. Bories and M. A. Combarnous, Natural convection in a sloping porous layer, *J. Fluid Mech.* **57**, 63–70 (1973).
3. M. P. Vlasuk, Heat transfer with natural convection in permeable porous medium, 4th *All-Union Heat and Mass Transfer Conference*, Minsk, Paper No. 1–49 (1972).
4. B. K. C. Chan, C. M. Ivey and J. M. Barry, Natural convection in enclosed porous media with rectangular boundaries, *J. Heat Transfer* **92C**, 21–27 (1970).
5. J. E. Weber, The boundary-layer regime for convection in a vertical porous layer, *Int. J. Heat Mass Transfer* **18**, 569–573 (1975).
6. C. G. Bankvall, Natural convection in vertical permeable space, *Wärme- und Stoffübertragung* **7**, 22–30 (1974).
7. J. E. Weber, Thermal convection in a tilted porous layer, *Int. J. Heat Mass Transfer* **18**, 474–475 (1975).
8. R. K. MacGregor and A. F. Emery, Free convection through vertical plane layers—moderate and high Prandtl number fluids, *J. Heat Transfer* **91C**, 391–403 (1969).
9. S. Yagi and D. Kunii, Studies on heat transfer in packed beds, *Int. Dev. Heat Transfer* **4**, 750–759 (1962).