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AN EXPERIMENTAL STUDY OF TRANSIENT HEAT TRANSFER CHARACTERISTICS IN A POROUS LAYER ENCLOSED BETWEEN TWO OPPOSING VERTICAL SURFACES WITH DIFFERENT TEMPERATURES

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NOMENCLATURE

a^*, thermal diffusivity of porous layer, $\lambda^*/(\rho c_p)_f$;
c_p, specific heat at constant pressure;
d, diameter of solid particle;
g, gravitational acceleration;
H, height of rectangular cavity;
H/W, aspect-ratio;
k, permeability, $\varepsilon^3 d^2/[150(1-\varepsilon)^2]$;
Nu^*, modified Nusselt number, $\alpha W/\lambda^*$;
q, heat flux;
Ra^*, modified Rayleigh number, $g\beta_f \Delta T k W/\nu_f a^*$;
T, temperature;
ΔT, temperature difference between hot and cold walls, $(T_h - T_c)$;
t, time;
W, width of rectangular cavity;
X, distance from bottom of rectangular cavity;
X^*, non-dimensional distance, X/W;
Y, distance from hot wall;
Y^*, non-dimensional distance, Y/W.

Greek symbols

α, mean heat transfer coefficient of porous layer, $q/\Delta T$;
β, cubical thermal expansion coefficient;
λ, thermal conductivity;
λ^*, thermal conductivity of porous medium (without convection), $\lambda_f^* + \lambda_s^*$;
ν, kinematic viscosity;
ρ, density;
ε, porosity;
τ, non-dimensional time, $\lambda^* t/[\varepsilon(\rho c_p)_f W^2]$.

Subscripts

c, h, cold and hot walls, respectively;
f, s, fluid and solid, respectively.

1. INTRODUCTION

RECENTLY, from the viewpoint of saving energy, attention in the industrial field has been paid to the insulation of buildings by using porous insulating material in which solar or geothermal energy can be stored. In addition, research of heat and mass flows in the porous layer has a wide application in the field of geophysics concerning the movement of geothermal hot water into the porous medium and the spreading of a pollutant based on convective movement. From these applications, it is necessary to obtain information about the heat transfer characteristics in the porous layer, especially the transient behavior of convective heat transfer. Many researchers [1–10] have reported on heat transfer in the porous layer in the steady state. However, only a few studies [11–13], which have focussed mainly on convenient analysis, have been carried out on transient heat transfer characteristics in the porous layer, but its behaviour has not been adequately clarified by experiment.

The present study deals experimentally with the effect of each factor considered on transient heat transfer characteristics in a rectangular cavity packed with porous medium. These cavities have two opposing vertical boundary surfaces which are kept at uniform (but different) temperatures. That is, the present experiments aim to examine the influence of cavity dimension (aspect-ratio, H/W), spherical solid particle diameter (porosity, ε) and the physical properties of the porous medium on transient heat transfer characteristics, especially the natural convective characteristics.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The present experiments were carried out using a rectangular cavity packed with porous medium, whose two opposing vertical wallls were kept at uniform (but different) temperatures while the other walls were thermally insulated. The main parts of the experimental apparatus consisted of the heating part, test section and cooling part as depicted in Fig. 1. Four kinds of test sections were constructed by inserting the acrylic resin frame (15 mm thickness) of 571 mm (height H) × 22, 40, 57 and 116 mm (variable width W) section area and 480 mm depth between the heating and cooling parts. In order to obtain two opposing vertical boundary conditions with a uniform temperature, the vertical hot and cold walls (copper plate of 5 mm thickness) were divided into five parts by bakelite frames. The surface temperature of the hot wall was maintained at uniform temperature using five independently controllable main mica electric heaters. The guard mica electric heaters were mounted on the rear side of the main heaters across a bakelite plate (5 mm thickness) to minimize the heat-loss from the main heaters to the environment. The surface temperature of the cold wall was maintained at uniform temperature by introducing temperature-controlled coolant (brine) into each of five separate cooling

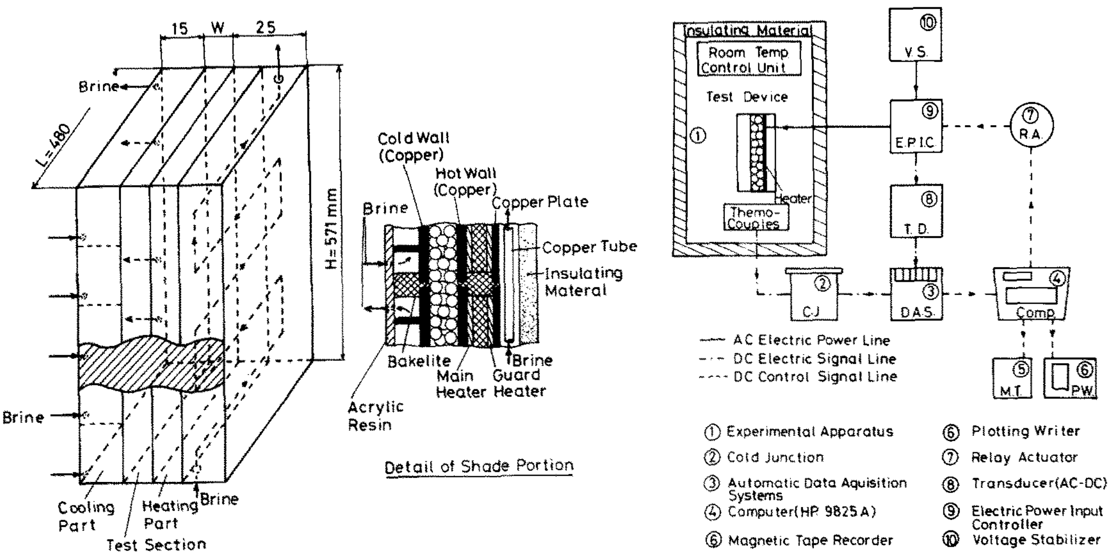

FIG. 1. Schematic view of experimental apparatus and schematic diagram of automatic temperature control system.

chambers attached to the rear side of the cold wall. The temperature control of the hot wall was performed using an automatic temperature control system as shown in Fig. 1. Consequently, it was possible to control the surface temperature distribution of the hot wall within ± 0.3°C. The temperature at every measuring point was measured with Cu–Co thermocouple (0.1–mm dia.). In order to measure the temperature distribution of fluid, five traversing small proves (stainless steel pipe of 0.8 mm dia.) with Cu-Co thermocouple (0.1 mm dia.) were set at five measuring points of $Y^*(Y/W)$ = 0.1, 0.25, 0.5, 0.75 and 0.9 at the fixed height of X^* (X/W) = 0.5 H/W.

The total heat-loss from the experimental apparatus was estimated within ± 5%. Before the test run, the effective thermal conductivity of the porous medium without natural convection, λ^*, was measured in the horizontal porous layer heated from above. The values obtained of λ^* agreed within ± 6% with those calculated from the experimental correlation equation proposed by Kunii and Smith [14]. After the temperatures at measuring points were confirmed as uniformly maintained at the cold-wall-temperature, the test run was started by heating the hot wall. The present experiments were performed using the four kinds of cavity width mentioned above: water and transformer oil as a working fluid, and three kinds of glass beads (d = 1.01, 5.03 and 16.4 mm) and iron balls (d = 11.0 mm) as spherical solid particles.

3. EXPERIMENTAL RESULTS

One would expect a complicated change in the behaviour of heat transfer in the porous layer to be brought about by the balancing of the time variation of the energy stored in the porous layer (q_{st}), energies transferred by fluid movement (q_{cv}) and conduction through the porous layer (q_{cd}). The total thermal energy q at the transient state from the hot wall to the cold wall through the porous layer can be expressed as follows:

$$q = q_{st} + q_{cv} + q_{cd}.$$

Typical variations of fluid temperature distribution with the combination of water–glass beads (d = 5.03 mm, and ε = 0.38) to the real time are shown in Fig. 2 for H/W = 4.9 (W = 116 mm) under the vertical temperature conditions of T_h=40°C and T_c=5°C. The solid circle shows the fluid temperature distribution of the horizontal porous layer heated from above without natural convection. From this figure, it can be seen that it takes about 90 min to reach to a steady state and the distributions of fluid temperature at measuring points vary complicatedly with time. In other words the increase of T_f to time at the position of Y^* = 0.1 stops after about 20 min from the test run, after which the value of T_f increase again. This re-increase of T_f means the onset of natural convection at this measuring point. Comparing the results with natural convection (open circle) with ones without convection (solid circle), the thermal steady-state of the porous layer with convection is reached faster than that without convection. Furthermore, the distribution of fluid temperature in the porous layer with convection becomes small compared to that without convection (except near the hot and cold walls). From these results, it might be said that the fluid movement has the effect of thermal homogeneity on the porous layer and also that this homogenizing effect helps the porous layer to reach to the thermal steady-state in a shorter time.

Figure 3 shows the relationship between Nu^* and t. These transient characteristics of the heat transfer in the porous layer can be explained as follows. For a short time after starting the heating, much heat is transferred into the porous layer from the hot wall because of a large temperature difference between the latter and the porous medium near it. This is stored in the porous medium and consequently Nu^* decreases sharply as the amount of heat stored, q_{st}, decreases with time. Then, Nu^* increases with time due to heat transmission by natural convection q_{cv}. Finally, Nu^* becomes constant when the heat transferred by convection rises to a maximum and the thermal steady-state is reached in the porous layer. The values of Nu^* (solid line) at the steady state (indicated at the right-hand side of Fig. 3) are derived from the experimental correlation proposed previously by the authors [2]. Judging from the variation of Nu^* to t in Fig. 3, it might be said that the influence of the convection on the transient heat transfer through the porous layer is decreased gradually as H/W is increased, due to the decrease of the increasing trend of Nu^* to t; in particular the result for H/W = 26 does not show the increase of Nu^* to t. However, as H/W increases under a fixed height of the cavity, the time period when the

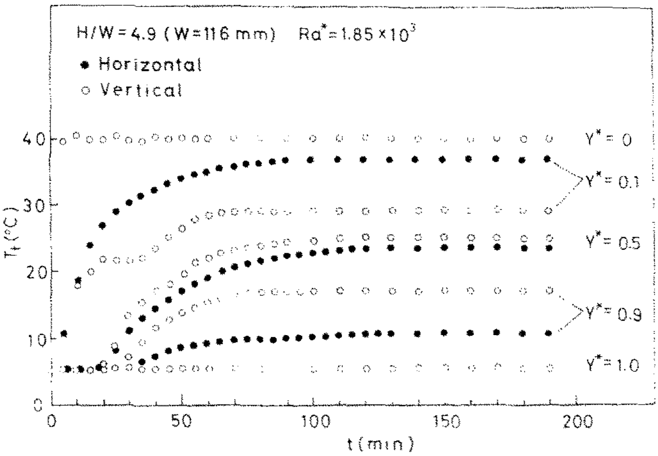

FIG. 2. Variation of T_f to t at $X^* = 0.5\ H/W$ for $H/W = 4.9$ ($W = 116$ mm), water–glass beads ($d = 5.03$ mm) and $Ra^* = 1.85 \times 10^3$.

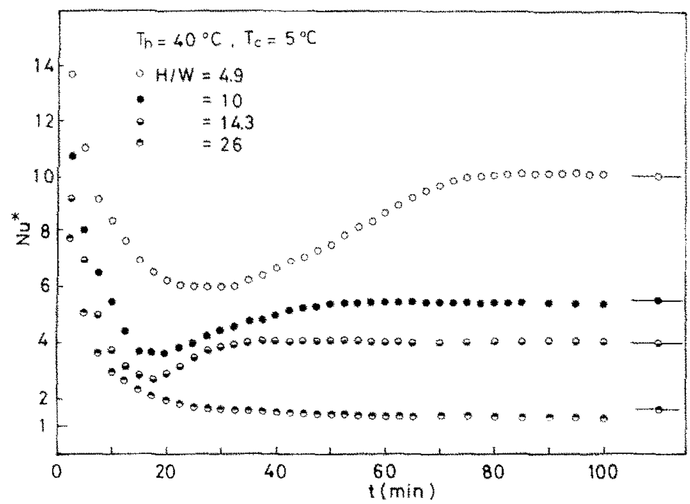

FIG. 3. Relationship between modified Nusselt number, Nu^*, and t for various aspect-ratio H/W, and $T_h = 40°C$, $T_c = 5°C$; $H/W = 4.9$ ($Ra^* = 1.85 \times 10^3$), $H/W = 10$ ($Ra^* = 1.02 \times 10^3$), $H/W = 14.3$ ($Ra^* = 7.28 \times 10^2$), $H/W = 26$ ($Ra^* = 1.86 \times 10$).

steady state is achieved in the porous layer becomes shorter because of the smaller heat capacity of the porous layer, in spite of the small effect of natural convection.

Figure 4 demonstrates the relationship between Nu^* and τ using porosity ε or diameter of the solid particle as a parameter. From this figure, it can be seen that the variation of Nu^* to τ becomes sharp as ε increases because the contribution of natural convection increases in the porous layer, i.e. a steep decrease in Nu^* after starting the test run appears and then the value of Nu^* increases sharply until reaching the steady state. It is interesting that this peculiar variation of Nu^* to τ disappears gradually as ε decreases.

Figure 5 shows the variations of Nu^* to τ with the various combinations of fluid and spherical solid particles. Comparing the results for a combination of transformer oil–glass beads ($d = 16.4$ mm) with those for a combination of water–glass beads ($d = 16.4$ mm) in Fig. 4, τ ($= 0.75$) (until reaching a steady state) for the former is twice that of the latter ($\tau = 0.35$). This difference might be caused by the difference of the modified Rayleigh number Ra^* (especially in dependence on the viscosity of fluid) due to the same permeability k which is determined by the geometric texture of porous layer between both combinations. Comparing the combination of transformer oil–iron balls with two other combinations of transformer oil–glass beads in Fig. 5, it is of interest that the former reaches the steady state in a smaller τ in comparison with the latter because the thermal diffusivity for a combination of transformer oil–iron balls ($a^* = 3.2 \times 10^{-3}$ m^2/h at 20°C) is larger than that for a combination of transformer oil–glass beads ($a^* = 5.8 \times 10^{-4}$ m^2/h).

4. CONCLUSION

The transient characteristics of heat transfer in a rectangular cavity packed with porous media were examined experimentally. It was concluded that the natural convection occurred which promoted the thermal homogenizing effect in

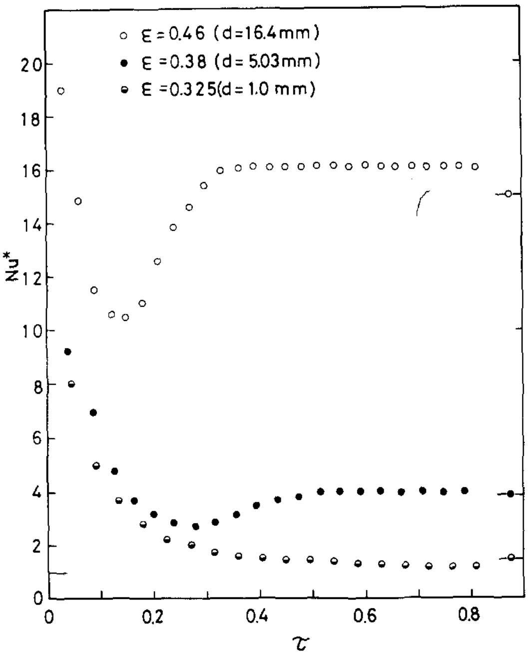

FIG. 4. Relationship between Nu^* and non-dimensional time τ for various porosities, $H/W = 14.3$, and $T_h = 40°C$, $T_c = 5°C$; $\varepsilon = 0.325$ ($Ra^* = 3.4 \times 10$), $\varepsilon = 0.38$ ($Ra^* = 7.28 \times 10^2$), $\varepsilon = 0.46$ ($Ra^* = 1.3 \times 10^4$).

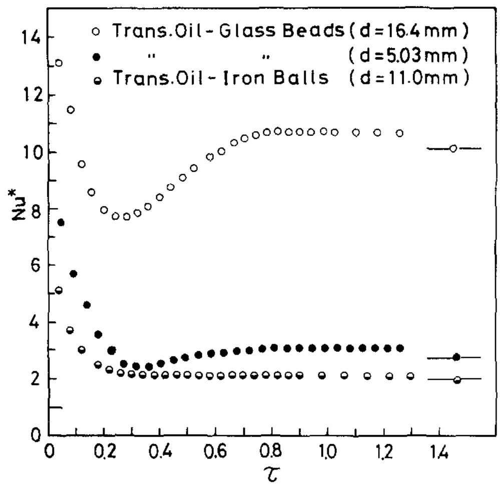

FIG. 5. Relationship between Nu^* and τ for various combinations of fluid and solid particles, and $H/W = 14.3$; transformer oil–glass beads ($d = 16.4$ mm, $Ra^* = 2.22 \times 10^3$), transformer oil–glass beads ($d = 5.03$ mm, $Ra^* = 1.13 \times 10^2$), transformer oil-iron balls ($d = 11.0$ mm, $Ra^* = 8.2 \times 10$).

the porous layer, and consequently the steady state was obtained in a shorter time by the contribution of natural convection. According to the purpose of the porous layer, it was possible to control both the time taken to reach a steady state and the amount of heat stored in the porous layer by varying the combinations of fluid and solid particles, diameter of solid particle and dimension of the porous layer.

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