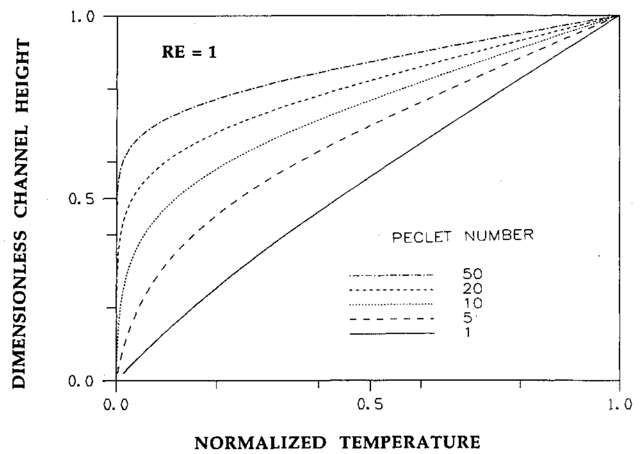

Fig. 2 Temperature profiles.

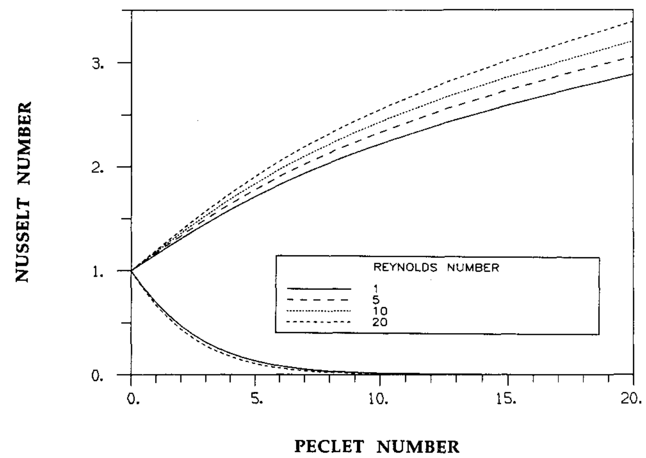

Fig. 3 Heat transfer rates at upper and lower walls.

The work presented here reveals several interesting conclusions about flow and heat transfer in a channel with injection at one wall. At low-blowing Reynolds numbers, the streamwise velocity profiles resemble Poiseuille flow. As the blowing increases, the profiles become more skewed with reduced velocity gradients at the injected wall. The pressure gradients along the channel decrease almost linearly with length, which reflects the increased pressure drop required to accelerate the injected fluid to the continually increasing velocities required by mass conservation.

The constant property assumption used in this work allows the momentum and mass conservation equations to be solved independently of the energy equation. Similarity solutions to the boundary-layer approximation of the energy equation with viscous dissipation neglected were found for several values of the Reynolds and Peclet numbers. The effects of Reynolds number on these solutions are not strong. However, very significant effects of the Peclet numbers are found. At Peclet numbers much less than unity ordinary conduction effects dominate and the temperature profile was found to be linear across the channel. As blowing increases, the profiles become increasingly nonlinear, with the temperature gradients (heat transfer rates) at the blown wall becoming smaller. This behavior is similar to that of a boundary layer on a transpiration cooled wall. For Peclet numbers greater than 3, the conduction to the blown wall is small. The opposite behavior occurs at the impermeable wall where heat transfer rates continually increase with increased blowing. These results can be used to approximate the transfer processes in thermal protection systems using a wick containing an evaporation material. The results show that high-evaporation rates can reduce heating of the substructure. Further, the increased convection at the upper wall can also be effective at reducing radiative heat transfer to the wick.

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Experimental Study of Natural Convection in Horizontal Porous Layers with Multiple Heat Sources

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Nomenclature

c = specific heat of fluid at constant pressure, J/kg-K
Da = Darcy number, K/H^2
d = diameter of glass bead, m
g = acceleration of gravity, m/s^2
H = height of the porous layer, m
h = average heat transfer coefficient on the heated surface, W/m^2-K
K = permeability of the saturated porous medium, m^2
k = effective thermal conductivity of the porous medium, W/m-K
Nu = average Nusselt number, hH/k
q = uniform heat flux, W/m^2
Ra = Rayleigh number, $Kg\beta qH^2/\nu\alpha k$
T = temperature, K
α = thermal diffusivity of porous medium, $k/(\rho c)_f$, m^2/s
β = isobaric thermal expansion coefficient of fluid, K^{-1}
θ = dimensionless temperature
ν = kinematic viscosity of fluid, m^2/s
ρ = density of fluid, kg/m^3
ϕ = porosity

Subscripts

f = fluid phase
s = solid phase

I. Introduction

OVER the past four decades, natural convection in saturated porous media has received considerable attention for its important applications in engineering and geophysics problems. However, most of the previous studies have considered the case of a horizontal porous layer uniformly heated from below, and emphases have been placed on the establishment of the criteria for onset of convection. Very few results have been reported for the case of a discrete heat source, although problems of this type are encountered more fre-

Presented as Paper 90-0257 at the AIAA 28th Aerospace Sciences Meeting, Reno, NV, Jan. 8-11, 1990; received Dec. 12, 1989; accepted for publication May 4, 1990.

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quently in the applications. Recently, Prasad and Kulacki[1,2] have studied the problem of natural convection in horizontal porous layers partially heated from below. In their studies, extensive results have been presented for the case of a single heat source. However, for the case of multiple heat sources, the results have only been reported by the present authors.[3] As a continuing effort toward a complete understanding of transport phenomena in porous media, we consider in this Note the effects of multiple sources on flow and temperature fields. The primary goal of this study is to validate our numerical results via controlled laboratory experiments.

II. Experimental Apparatus

To verify the numerical results, an experimental apparatus has been carefully designed (Fig. 1). The major component of the apparatus is a long rectangular box made of Plexiglas. Strip heaters are mounted on the bottom wall of the test section to serve as heat sources. The length of the heater runs along the width of the box. Twenty-one heaters make up for the complete length of the test section. The heater is designed in such a way that it provides a uniform heat flux. Each of the heaters is individually controlled and thus the heat flux can be varied in segments along the length of the test section. Sixty-three thermocouples (30 gauge, type T) are placed underneath the heaters with their junctions in direct contact with the heaters. Another 63 thermocouples are installed on the bottom surface in a "one-on-one" correspondence. With this arrangement, each heater is monitored by at least six thermocouples, and heat loss, if any, in both lateral and longitudinal direction can be accurately calculated.

Since the calculation of average Nusselt number requires the knowledge of surface temperature of the heater, its accuracy strongly depends on the precision of these measurements. An additional 20 thermocouples are attached to the surface of five heaters in the central portion of the test section. To minimize the interference to the flowfield, fine-gauge thermocouple wire (nominal size 0.56 × 0.96 mm) was used. By applying high-temperature, highly conductive adhesive to the beads, the thermocouple junctions are attached to the heater surface. The beads are made as small as possible to avoid interference and to provide accurate measurement and fast response.

To maintain effectively an isothermally cooled top surface, a cooling chamber is provided. The chamber is a rectangular box whose bottom wall is a copper plate, whereas the rest of the walls are made of Plexiglas. When it rests on top of the apparatus, the gap between the chamber and the test section is 5.08 cm (2 in.), the width of a heater. Twelve thermocouples are mounted on the surface of the copper plate to monitor the temperature variation.

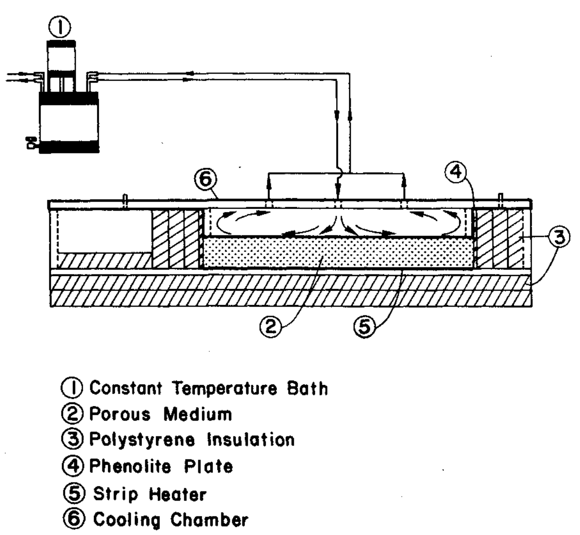

Fig. 1 Experimental setup for the present study.

Two sheets of polystyrene insulation are placed underneath the apparatus. The side walls are also insulated with three layers of polystyrene. It is expected that heat losses will be greatly reduced by these heavy insulations and the adiabatic condition is thus ensured.

III. Experimental Procedure

Experiments were performed using 3-mm-diam glass beads and water as the porous medium. The five heaters in the central portion of the test section were used to comprise the multiple heat sources. The porous layer has an aspect ratio of 21:1 and a packing parameter H/d of 16.9. The reason for choosing such a long shallow bed is that the assumption of a two-dimensional flow is nearly met and the effects of side walls can be minimized. In the beginning of the experiments, the test section was filled up to the desired height with glass beads and water. To produce an isotropic and homogeneous porous medium, the apparatus was shaken gently to let the glass beads settle and to release any trapped air. The apparatus was then entirely sealed.

Voltages supplied to the heaters were carefully controlled. For most of the experiments, the maximum voltage differences among heaters were controlled within ±0.1 V; on very few occasions, it was found that the difference went up to ±0.3 V. Sufficient time was allowed to let the system reach a steady state. The time required to reach a steady state was dependent on the Rayleigh numbers. In general, the higher the Rayleigh number, the less time was required to reach a steady state. At the steady state, the data acquisition system was activated to read the potential differences from each of the thermocouples. These voltage signals were then converted to temperatures. The process was repeated four more times at intervals of 2 min for each set of data. Therefore, each data point presented in this study is the average of five readings taken over approximately a 10-min period.

To perform an analysis on experimental errors, the actual uncertainty in the permeability and the effective thermal conductivity is unknown. These values are the subject of much heated debate in the literature and are often corrected in postexperiment analysis. Therefore, the uncertainties in these quantities have been neglected in the error analysis, and confidence in the chosen correlations for these values stems from their successful use by many investigators. Based on this analysis, the estimated uncertainties for the Rayleigh and Nusselt numbers are 5 and 3%, respectively. The uncertainty in Nusselt number is maximum when the input power is small (i.e., the temperature difference is small).

For the present study, the Reynolds number based on the glass beads diameter is calculated to be of the order of unity, which implies that the inertial effects are negligible. In addition, the Darcy number for the present case is on the order of 10^{-5}, which further implied that the Brinkman effects are insignificant.[4] Therefore, the applicability of Darcy's law in the present analysis is theoretically confirmed.

IV. Results

In the presentation of experimental results, the permeability used in the calculation of the Rayleigh number is based on the Kozeny-Carman equation. Except for highly porous, fibrous, or fissured media, it has been reported that this equation gives a very reasonable estimate of permeability.[5,6] For the calculation of the Nusselt and Rayleigh numbers, the effective thermal conductivity is estimated by the mixture rule:

$$k_e = \phi k_f + (1 - \phi) k_s \tag{1}$$

This conductivity model, although simple in form, gives very reasonable values of the stagnant thermal conductivity as long as there is no significant difference between k_s and k_f. The thermophysical properties used in the calculation of experimental results were evaluated at the average temperature of the copper plate (heat sink) and the heaters (heat source).

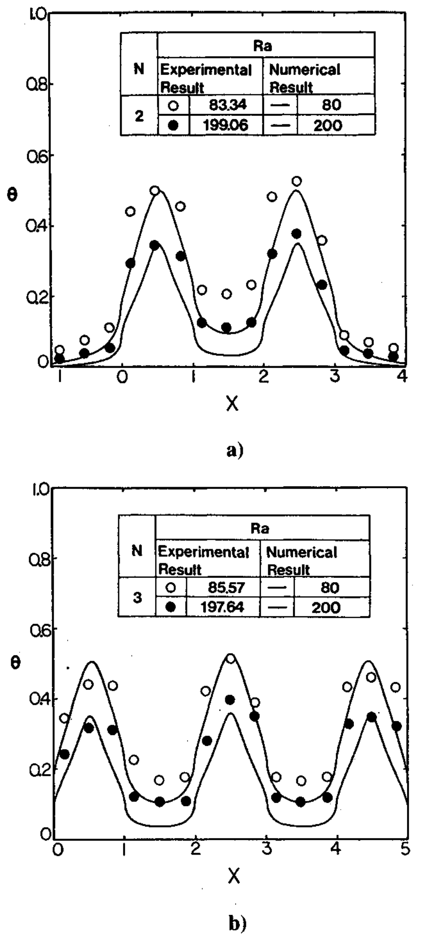

Fig. 2 Temperature distribution on the heated surface: a) $N = 2$; and b) $N = 3$.

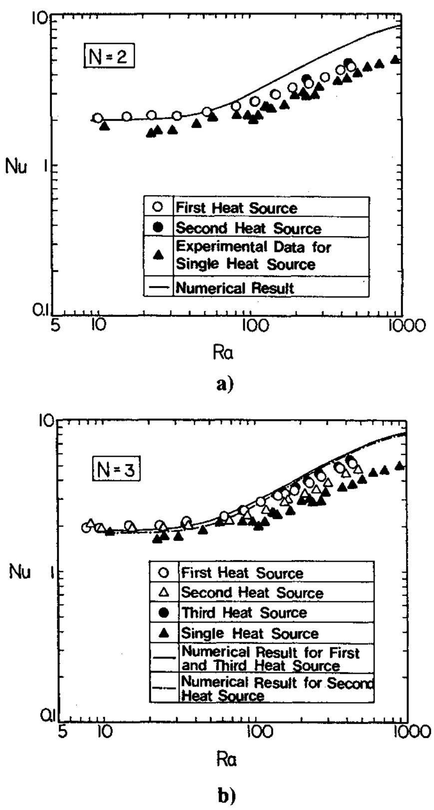

Fig. 3 Heat transfer results for the present study: a) $N = 2$; and b) $N = 3$.

Although we are without the aid of flow visualization, the flowfield was carefully monitored through temperature recordings. Numerical results have also been used to get a better understanding of the convective heat transfer process. The numerical method has been outlined in our previous study.[3] The only difference is in the thermal boundary condition, i.e., the constant-flux boundary condition is employed for the present case. As reported in our numerical study,[3] multiple recirculating cells are always present in the flowfield due to the nonuniform heating. This observation can be verified by the experimental results. The recorded temperature variation on the heated surface is compared with the numerical solution in Fig. 2. In general, the agreement between these two results is very good.

As the Rayleigh number (input power) increases, the flow and temperature fields remain stable. The instability reported by other investigators[7,8] has not been observed in the present study (at least, in the range of the parameters considered). Horne and O'Sullivan[7] have concluded that oscillatory convection in the flowfield originates from a combined process of a "triggering" mechanism and the instability of the thermal boundary layer. As reported in an experimental study by the present authors,[9] it is found that the "triggering" mechanism can be significantly weakened if one allows a sufficiently large unheated region in the layer, which provides additional capacity for dissipating small thermal disturbances generated in the heated region. This is apparent if one compares the geometry studied by the previous authors to the present case.

The Nusselt number for the individual heat source is defined by

$$Nu = \frac{hH}{k} = \frac{1}{\theta} \tag{2}$$

which is the reciprocal of the average temperature of the heat source. The Nusselt numbers for the individual heat source are shown in Fig. 3. For comparison, the experimental data for a single heat source[9] are also included. It is observed that the Nusselt number for the multiple heat source is always higher than that for a single heat source, which clearly indicates that the recirculating cells created by the multiple sources effectively brought down the average surface temperature. In addition, it is observed that the second heat source for $N = 3$ has the lowest Nusselt number, which has also been reported by the previous study.[3]

At the smaller Rayleigh numbers ($Ra < 100$), the experimental data agree very well with the numerical results. However, at the higher Rayleigh numbers, the experimental data for the present case are always less than those predicted by the numerical calculation. The discrepancy between these two results can be attributed to two factors: 1) There is some conduction heat transfer between two neighboring heat sources. Although the equipment is well insulated, heat is conducted through the bottom wall (Plexiglas plate), which can be identified from Fig. 2 by the comparison of the temperature variations. This factor, although it can be minimized, is due to the difference between the mathematical model and the physical model. For the present case, it is calculated that the maximum heat transfer by conduction through the bottom wall is 8% of the total power input, but mostly it is within 5%. 2) The thermal conductivity in the energy equation is not the stagnant thermal conductivity (i.e., based on the mixture rule) but the conductivity of the porous medium when the fluid is flowing. Since the fraction of energy transported by the fluid increases with convection, the effective thermal conductivity is expected to change and approach that of the fluid as the Rayleigh number increases. Although the concept of the effective thermal conductivity, which was proposed by Prasad et al.,[10] has greatly improved the agreement between the experimental and numerical results,[9,10] its theoretical justification awaits conclusive experimental proof. Despite all of these factors, the maximum difference between the experimental and numerical

Table 1 Correlations for natural convection in horizontal porous layers

	Correlation	Standard error
Single heat source[9]	$Nu = 0.269Ra^{0.451}$	0.637
Multiple heat source		
$N = 2$	$Nu = 0.575Ra^{0.334}$	0.237
	$Nu = 0.378Ra^{0.436}$ (for 1st and 3rd heat sources)	0.136
$N = 3$	$Nu = 0.372Ra^{0.403}$ (for 2nd heat source)	0.074

results is about 25%, but mostly they are within 10% of the numerical values. The correlations of heat transfer results are provided in Table 1.

Acknowledgment

The authors are grateful for the support from the Computer Center at the Colorado State University and the U.S. Nuclear Regulatory Commission.

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