Int. J. Heat Mass Transfer. Vol. 19, pp. 234–235. Pergamon Press 1976. Printed in Great Britain

AN EXPERIMENTAL STUDY OF THERMAL EQUILIBRIUM IN LIQUID SATURATED POROUS MEDIA

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(*Received* 18 *March* 1975 *and in revised form* 7 *July* 1975)

NOMENCLATURE

A, cross sectional area of test section [cm^2];

$\bar{p}$, characteristic wetted perimeter [cm];

C_p, constant pressure specific heat per unit mass of fluid [cal/(g °C)];

K_{eff}, effective thermal conductivity [cal/(s cm °C)];

L, length of test section [cm];

μ, viscosity [g/cm s];

$\bar{M}$, volume averaged mass flow rate [g/s];

Pe, Peclet number, $= \dfrac{\bar{M}C_pL}{K_{\text{eff}}A}$;

Re^*, Reynolds number, $= \dfrac{\bar{M}}{\mu\bar{p}}$.

Subscripts

f, fluid property;

eff, effective.

INTRODUCTION

HEAT-TRANSFER processes in porous media are important in many branches of engineering, for example, in geothermal power generation, heat pipe technology, and transpiration cooling. Many analytical models have been proposed to consider heat transfer in porous materials [1–6]. Some of the underlying assumptions of these models have not been verified. The purpose of this paper is to examine the assumption of thermal equilibrium that is made in the volume averaged energy conservation equation proposed by Drew *et al.* [5, 7], Dybbs *et al.* [2, 5], and Slattery [1] for porous media.

There appears to be some controversy over the question of thermal equilibrium. Some researchers assume that a temperature difference exists between the solid and the fluid of a porous medium and others, that no temperature difference exists. Bland [3] was one of the first to predict the temperature distribution within a gas saturated porous medium. He assumed that there was a temperature difference between gas and solid and that heat was exchanged between them. Subsequently several analytical and experimental techniques were devised to determine the heat-transfer coefficient between solid and fluid [4, 8, 9] Glaser *et al.* [10] measured temperature differences between gas and solid in a packed bed of spheres. Electric current was passed through the metallic particles of the packed bed which created within the particles a steady generation of heat. Turnacliff [10] measured the solid temperatures and predicted the gas temperatures to be different. Apparently no experimentalists have simultaneously measured both the gas and solid temperatures within a porous medium [9, 12]. Baldwin [13] measured temperature differences between the water and the metal spheres. However, he also generated heat within the solid and the existence of a heat transfer coefficient is not surprising.

Drew *et al.* [5, 7], Dybbs *et al.* [2, 6] and Slattery [1] have averaged the microscopic energy equations that apply in the pores of a porous medium to obtain macroscopic equations that can be compared to experiment. The basic assumption of these models is that in any local averaging volume, the average fluid temperature is the same as the average solid temperature. This assumption was also used by Jakob [14] and Eckert *et al.* [15]. Apparently, no attempts have been made to validate this assumption. Thus, the objective of the present work is to experimentally examine the limits of validity of this assumption for liquid saturated flows in porous media.

EXPERIMENTAL

The experimental study of thermal equilibrium in porous media requires the simultaneous measurement of the temperatures of the solid and fluid phases. To do this a scaled up model of a porous medium was designed so that thermocouples could be located solely in each phase. That the results obtained from the scaled up model can be extrapolated to other porous materials can be seen from dimensional analysis [16].

The model studied was a simple cubic packing of 1·27 cm dia brass spheres contained in the square cross-sectional test section shown in Fig. 1. A unit cube of the packing is shown in Fig. 2. To achieve uniform porosity throughout the test section, half spheres were glued to the walls, quarter spheres at the corners, and one-eighth spheres at the ends. The container walls were made of lexan.

Thermocouples were located at seven planes in the test section which was 19·05 cm long and 2·54 cm square. At four of these planes, eight thermocouples were placed; four in the fluid and four in the solid. Each one of the thermocouples in the solid was equidistant from the center of the section with a corresponding thermocouple in the fluid. See Fig. 3. At the central plane, a thermocouple was located in the center of the section and another was embedded in a half sphere at the wall. At the entrance and exit planes, one thermocouple was located in the solid and three in the fluid.

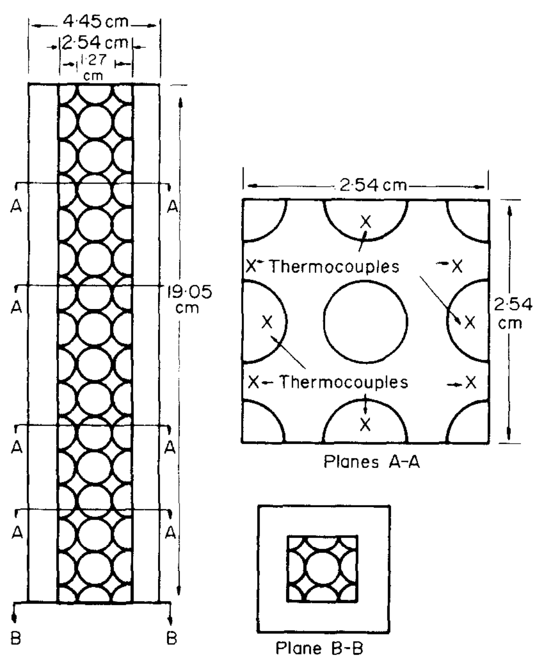

FIG. 1. Test section and thermocouple locations.

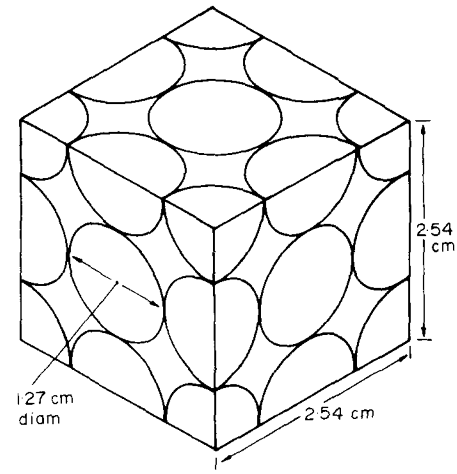

FIG. 2. Unit cube of test section.

The test section was in contact with a heater at one end and a cooler at the other. These heat exchangers were designed to maintain the ends at uniform temperature and also allow the fluid to flow with a minimum of obstruction. The heater and cooler were identical in construction. These were made of a 1·18 cm thick plexiglas plate which had a 7·62 cm hole at the center so it could slide over the flanges of the test section. Thin wall 0·33 cm copper tubes were inserted in holes drilled along the mid-section of the plexiglas plate. At both ends of the tubes there were two end pieces for the inlet or outlet of cooling or heating water. The tubes were 0·32 cm apart. They were placed in contact with the test section. To promote good thermal contact, the copper tube walls were machined flat where they contacted the test section.

The flow of distilled water through the test section was provided by a constant head tank and controlled by a fine metering valve. The temperature was recorded by a DIGITREND 210 multi-channel digital temperature recorder. Figure 3 shows the thermal boundary conditions used in the experiments. In all the experiments, the axial temperature drop $T_2 - T_1$, was from 7·3 to 73·3°C. For the

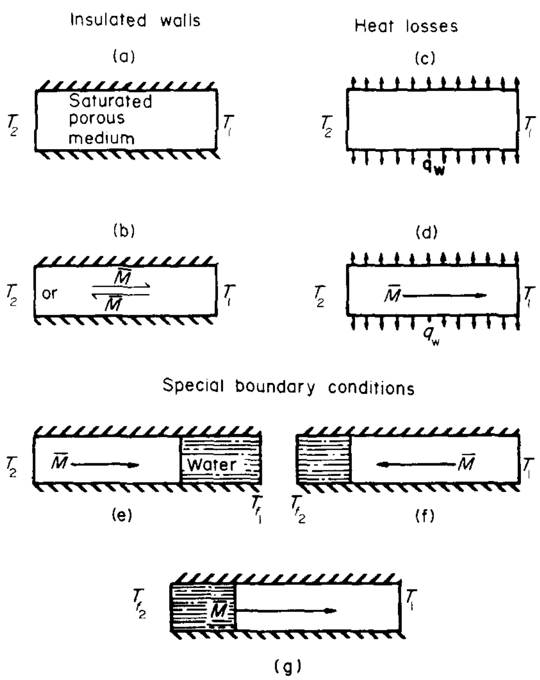

FIG. 3. Boundary conditions.

insulated wall cases the test section wall was wrapped with six Briskeat flexible heating tapes, each connected to a variable voltage transformer. The current passing through each tape was separately controlled so that the exterior wall temperature could be matched with the interior wall temperature. Under such conditions no heat was lost from the test section.

The cases investigated under insulated thermal conditions were: no flow, Fig. 3(a); flow and temperature gradient in the same direction, forward flow, Fig. 3(b); and flow and temperature gradient in the opposite direction, reverse flow, Fig. 3(b). In one forward flow case the hot bath temperature and the cool bath temperature were kept constant while the flowrate was varied. In another, cooling was done in a heat exchanger downstream of the specimen, Fig. 3(c). Heating was done upstream of the specimen in one reverse flow case, Fig. 3(f). Another case investigated was with hot water entering the test section, that is, the solid was not heated directly by the heat exchanger, Fig. 3(g).

The heat loss cases investigated were: no flow, Fig. 3(c); and forward flow, Fig. 3(d).

Temperatures were measured at steady state, which was established when three to five complete sets of temperature measurements showed no difference to within the experimental accuracy of ± 1°C.

CONCLUSIONS AND DISCUSSION

For all of the boundary conditions shown in Fig. 3, the solid and liquid temperatures at any one plane in the porous matrix were within the experimental accuracy of the thermocouples, ± 1°C. This was true for the special boundary conditions Fig. 3(e, f, g) in which the cooling was done downstream of the porous medium Fig. 3(e), and heating was done upstream of the porous medium, Fig. 3(f, g). Thus thermal equilibrium between solid and liquid existed for the range of flow conditions and heat-transfer rates studied. For the forward flow cases the Peclet number range was $0 \leqslant Pe \leqslant 610$ corresponding to a Reynolds number range $0 \leqslant Re^* \leqslant 38{\cdot}2$ and for the reverse flow cases $0 \leqslant Pe \leqslant 2112$ corresponding to $0 \leqslant Re^* \leqslant 117$. One should note that thermal equilibrium existed for flow rates beyond the Darcy flow regime, $Re < 10$.

These results have an important practical implication. For a given saturated porous medium under similar boundary conditions and range of Pe studied, thermal equilibrium should exist between solid and fluid phases. Hence the temperature measured by a thermocouple placed within the porous medium would be the temperature of the solid or the fluid, as well as the average temperature of the solid and the fluid.

Acknowledgements—Financial support for this work was provided by National Aeronautics and Space Administration through grant number 36-003-064.

REFERENCES

1. J. C. Slattery, *Momentum, Energy, and Mass Transfer in Continua*. McGraw-Hill, New York (1972).
2. A. Dybbs and S. Schweitzer, Conservation equations for nonisothermal flow in porous media, *J. Hydrology* **19**, 171 (1973).
3. D. R. Bland, Mathematical theory of the flow of a gas in a porous solid and of the associated temperature distributions, *Proc. R. Soc.* **A221**, 1144 (1954).
4. H. Littman, R. G. Barile and A. H. Pulsifier, Gas-particle heat transfer coefficients in packed beds at low Reynolds numbers, *I/EC Fundamentals* 7(4), 554 (1968).
5. D. A. Drew and L. A. Segel, Averaged equations for two phase flows, *Stud. Appl. Math.* **L**, 205 (1971).
6. A. Dybbs, Forced convection in a saturated porous medium, Ph.D. Thesis, University of Pennsylvania (1970).
7. D. A. Drew, Averaged field equations for two-phase media, *Stud. Appl. Math.* **L**, 133 (1971).
8. J. C. Y. Koh and E. del Casal, Heat and mass flow

through porous matrices for transpiration cooling, in *Proceedings of 1965 Heat Transfer and Fluid Mechanics Institute*. Stanford University Press (1965).

9. J. C. Y. Koh, J. L. Dutton and B. A. Benson, Fundamental study of transpiration cooling, NASA Cr-134523 (1973).
10. M. Glaser and G. Thodos, Heat and momentum transfer in the flow of gases through packed beds, *A.I.Ch.E. Jl* **4**, 63 (1958).
11. R. Turnacliff, An experimental study of local convective heat transfer, Ph.D. Thesis, University of Minnesota (1957).
12. R. S. Brodkey, D. S. Kim and L. E. Gates, A mathematical model for heat transfer in a packed bed and a simplified solution thereof, *A.I.Ch.E. Jl* **18**(3), 623–627 (1972).
13. D. Baldwin, Heat transfer in beds of oriented spheres, Ph.D. Thesis, Carneige I.T. (1961).
14. M. Jakob, *Heat Transfer*, Vol. 2. Wiley, New York (1957).
15. E. R. G. Eckert and R. M. Drake, Jr., *Heat and Mass Transfer*. McGraw-Hill, New York (1959).
16. K. F. Wong and A. Dybbs, An experimental study of thermal equilibrium in liquid saturated porous media, Department of Fluid, Thermal, and Aerospace Sciences, Case Western Reserve University Report FTAS/TR-75-109 (1975).

Int. J. Heat Mass Transfer. Vol. 19, pp. 236–238. Pergamon Press 1976. Printed in Great Britain

OBSERVATION OF BOILING IN POROUS MEDIA

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(*Received 9 October* 1974 *and in revised form* 11 *March* 1975)

NOMENCLATURE

- q, heat flux based on total area [W/m^2];
- A, total surface area [m^2];
- A_v, vapour covered surface area [m^2];
- f, friction factor;
- k_0, k_1, k_2, constants;
- Re, Reynolds number based on d_p;
- m, power index, equation (1);
- ΔP_v, vapour pressure difference [N/m^2];
- l, wick thickness [m];
- ρ, vapour density [kg/m^3];
- μ, vapour viscosity [Ns/m^2];
- c, bulk velocity [m/s^2];
- d_p, mean pore diameter [m];
- G_v, vapour mass flow rate per unit vapour area [kg/s – m^2];
- h_{fg}, enthalpy of vapourisation [kJ/kg – K].

THE EVAPORATIVE heat-transfer process in porous media leads to the high thermal conductance of heat pipe devices. The actual mechanism of heat transfer at the heating surface is not clearly understood. Ferrell and Alleavitch [1] and Ferrell and Johnson [2] investigated the heat transfer from porous beds formed of glass and Monel beads. They postulated from their heat-transfer results the existence of a thin layer of liquid next to the heating surface and that the heat transfer to the bulk liquid occurred by conduction across this layer and the beads. On the other hand Moss and Kelly [3] during their neutron radiographic study of a planar heat pipe found evidence of the existence of a vapour layer at the heating surface.

This communication presents the results of an investigation undertaken to observe the evaporative process in a thin porous medium. The porous medium used was a 6 mm thick layer of polyurethane foam and the working fluid was distilled water maintained at a level just above the foam. The rig designed for observing the process is shown in Fig. 1 and consisted essentially of a glass cylinder (1) of 75 mm internal diameter with a 3 mm thick Pyrex glass plate (2) at the base forming the heating surface and a reflux condenser (3). A rod (4) and a perforated plate (5) ensured a good contact between the foam wick (6) and the base. Four Carborundum heaters (7) enclosed in the furnace (8) supplied heat to an area 25 mm in diameter at the centre of the base. A mirror (9) facilitated observation and photography (10). The radiant energy from the heaters was absorbed within 1 mm of the glass base and heat reached the wick chiefly by conduction.

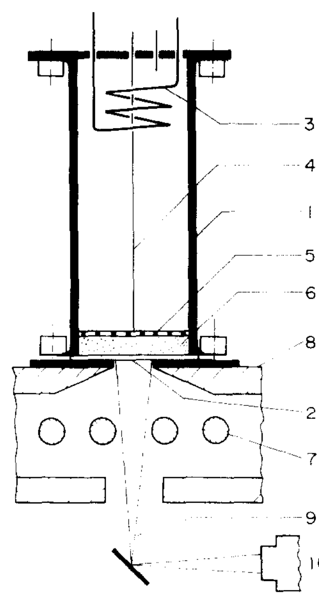

FIG. 1. Visualisation rig.

Observation indicated the simultaneous and continuous existence of vapour and liquid regions in the wick. These regions extended through the wick thickness and the boundaries between them were constantly undergoing minor adjustments. The mean vapour covered area at each heat flux was determined by measuring the required area on four photographs of the type shown in Fig. 2. The heat flux q (based on the total heat input over the plate) is found to vary with the proportion of vapour covered area to total area A_v/A as shown in Fig. 3. In the absence of boiling heat transfer at the surface this ratio is zero and the limiting heat flux is reached when the ratio is unity and the surface is covered with vapour. For comparison the range of limiting heat fluxes obtained under similar boiling conditions but in a number of separate experiments is also indicated in Fig. 3.

The boiling heat flux may be related to the vapour flow and therefore to the vapour pressure drop across the wick.