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

K. CORNWELL, B. G. NAIR and T. D. PATTEN
Department of Mechanical Engineering, Heriot-Watt University, Edinburgh EH1 1HX, Scotland

(*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

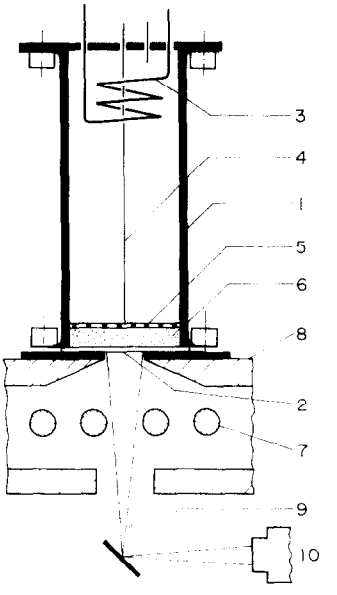

FIG. 1. Visualisation rig.

1 mm of the glass base and heat reached the wick chiefly by conduction.

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.

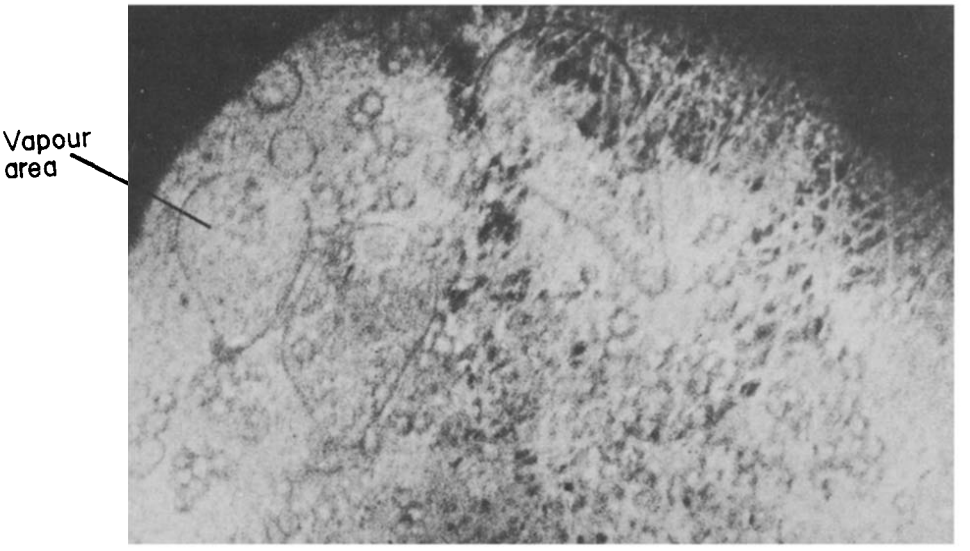

FIG. 2. Photograph showing liquid and vapour covered areas.

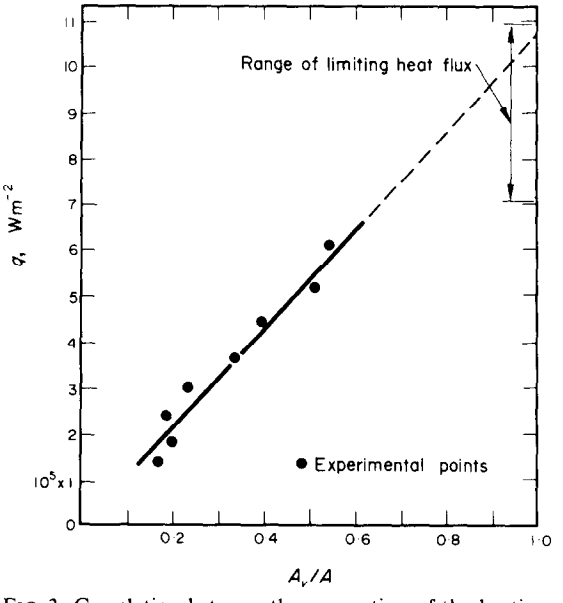

FIG. 3. Correlation between the proportion of the heating surface covered with vapour and the heat flux.

The general correlation for flow through a restriction such as a porous media is given by:

$$f = k_0 Re^m \tag{1}$$

where the friction coefficient

$$f = \frac{(\Delta P_v/1)d_p}{\rho c^2/2}$$

and Reynolds number

$$Re = \frac{\rho c d_p}{\mu}.$$

In these expressions ΔP_v is the vapour pressure drop across the wick of thickness l, the terms ρ, μ and c refer to the vapour density, viscosity and bulk velocity and d_p is a characteristic dimension of the wick material (conventionally the mean pore diameter). In the present case the pressure drop is additionally dependent upon whether the wick is dry or saturated with water. When the wick is saturated the

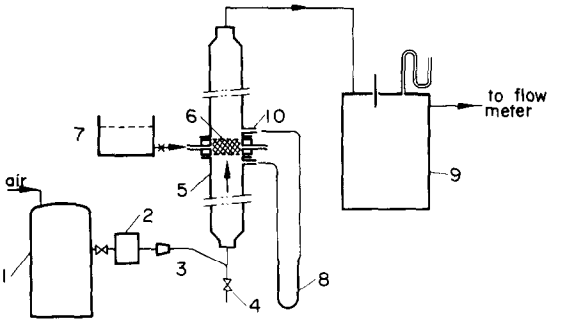

FIG. 4. Pressure drop rig. 1. Pressure vessel; 2. pressure regulator; 3. nozzle; 4. liquid drain; 5. test section (25 mm dia.); 6. wick held between two perforated plates; 7. constant level water tank for saturating the wick; 8. micromanometer; 9. receiver; 10. pressure taps.

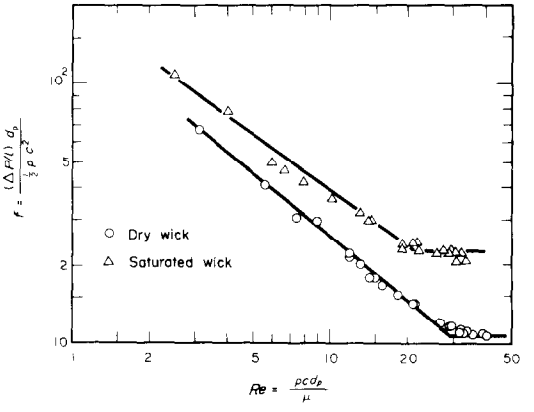

FIG. 5. Pressure drop characteristics of dry and saturated wicks (for $d_p = 0{\cdot}59$ mm and a wick diameter of 25 mm).

vapour formed at positions on the boiling surface forces water out of the neighbouring pores within the wick in order to flow out of the wick. The rig sketched in Fig. 4 was designed to give some indication of the difference in pressure drops in these two conditions. Air rather than steam was used as the working fluid to avoid problems of condensation

in the wet wick and it was assumed that the correlation of equation (1) would be similar for steam. The results presented in Fig. 5 indicate a flow regime where f decreases with increasing flow and a regime where it is almost constant. Vapour flow through the wet wick under boiling conditions is within the former regime and $m = -0{\cdot}71$.

As the flow rate of vapour is high and the wick is thin it is assumed that the vapour flow area A_v does not vary through the wick thickness. Under this condition the mass flow rate of vapour G_v (per unit vapour area A_v) is given by:

$$G_v = \rho c$$

and substitution into equation (1) yields on rearrangement

$$G_v = k_1 \left(\frac{\Delta P_v}{l}\right)^{1/(2+m)} \tag{2}$$

where k_1 is a constant involving ρ, μ, k_0, d_p and m. The heat flux based on total area A is given by

$$q = \left(\frac{A_v}{A}\right) G_v h_{fg}$$

where h_{fg} is the enthalpy of vaporisation. Substitution in equation (2) yields

$$q = k_1 h_{fg} \left(\frac{A_v}{A}\right) \left(\frac{\Delta P_v}{l}\right)^{1/(2+m)}. \tag{3}$$

The experimental results given in Fig. 3 show a linear relationship between q and A_v/A which indicates that for this wick ΔP_v is constant in the nucleate boiling heat flux range. Physically this implies that the increase in vapour flow rate (due to the increase in heat flux) is accommodated entirely by the increase in vapour flow area and a limiting condition is reached when this area is maximum. Substitution of the appropriate properties together with the slope $q/(A_v/A)$ from Fig. 3 into equation (3) yields a value of 42 N/m^2 or 4·3 mm of water for ΔP_v. Since the correct value for ΔP_v must be rather greater than the water head of 6 mm above the boiling surface, this value is low; but is nevertheless sufficient to show that the assumptions made in the theory are reasonable. Attempts to measure ΔP_v using a micro-probe were unsuccessful owing to the interference with the boiling mechanism.

In conclusion the study to date has shown that: (i) vapour and liquid areas co-exist at the heating surface over a wide range of heat flux, (ii) the extent of these areas may be correlated with the heat flux, and (iii) the limiting heat flux may be roughly estimated by extrapolating the A_v/A vs q curve.

Acknowledgement—The authors wish to thank the National Engineering Laboratory, East Kilbride, Glasgow, (U.K.) for financial support and the Heriot-Watt University for providing research facilities. This note is published with the permission of the Director, National Engineering Laboratory and is Crown copyright.

REFERENCES

1. J. K. Ferrell and J. Alleavitch, Vaporization heat transfer in capillary wick structures, *Chem. Engng Progr. Symp. Ser.* **66**, No. 102, 82–91 (1970).
2. J. K. Ferrell and H. R. Johnson, The mechanism of heat transfer in the evaporator zone of a heat pipe, ASME Paper 70-HT/Spt-12, American Society of Mechanical Engineers, New York (1970).
3. R. A. Moss and A. J. Kelly, Neutron radiographic study of limiting planar heat pipe performance, *Int. J. Heat Mass Transfer* **13**(3), 491–502 (1970).