Int. J. Heat Mass Transfer Vol. 16, pp. 1652-1655 Pergamon Press 1973. Printed in Great Britain

TRANSIENT AND FORCED CONVECTIVE FILM BOILING IN SUBCOOLED WATER AT LOW REYNOLDS NUMBERS

S. J. BOARD and R. B. DUFFEY

Central Electricity Generating Board, Berkeley Nuclear Laboratories, Berkeley, Gloucestershire, GL13 9PB, England

(*Received* 13 *December* 1972)

NOMENCLATURE

C,	constant of proportionality;
c,	liquid specific heat;
D_i,	diameter of vapour region;
D_w,	diameter of wire;
h,	heat transfer coefficient;
k,	thermal conductivity of liquid;
Nu_i,	Nusselt number hD_i/k;
Nu_w,	Nusselt number hD_w/k;
n,	constant in heat transfer correlation;
Pr,	Prandtl number of liquid, $c\rho v/k$ (evaluated at the film temperature);
$(Q/A)_i$,	heat flux per unit area from vapour/liquid interface;
$(Q/A)_w$,	heat flux per unit area of wire;
Re_i,	Reynolds number, $(U_\infty D_i/v)$ (evaluated at the film temperature);
Re_w,	Reynolds number, $(U_\infty D_w/v)$ (evaluated at the film temperature);
t,	time;
T_{sat},	saturation temperature;
T_∞,	bulk liquid temperature;
U_∞,	bulk liquid velocity;
α,	thermal diffusivity of liquid, $k/\rho c$;
v,	kinematic viscosity of liquid;
ρ,	liquid density.

1. INTRODUCTION

In the development of a thermal explosion, a molten material becomes dispersed in a liquid coolant, and rapid energy transfer occurs. If the molten material is at a high temperature, a vapour layer will cover the energy transfer surface, but the bulk of the liquid will still be subcooled. A systematic study of the energy transfer and vapour generation processes in a stagnant liquid under these conditions is at present in progress (e.g. [1]). However it is likely that at certain stages in the dispersal process there will be significant relative velocities between the hot metal and the coolant, leading to the possibility of additional convective heat transfer, and it is therefore necessary to determine the rate of energy transfer to a flowing subcooled liquid.

This problem has been investigated elsewhere for large bodies at high Reynolds numbers: Walford [2] used a 0·6 cm sphere at $Re = 6 \times 10^3$–2×10^4 in water, and Stevens and Witte [3] a 1·9 cm sphere at $Re = 1{\cdot}2 \times 10^5$ $2{\cdot}4 \times 10^5$ in water. Relevant information is also available from the work of Motte and Bromley [4] using 1–1·7 cm cylinders at $Re = 10^4$–2×10^5 in various organic liquids.

For the typical debris representative of a thermal explosion, however, the effective particle diameter will be $\sim 10^{-3}$ cm [5]. From the data of Wright [6] the velocities characterising the mixing and expansion stages of thermal explosions can be deduced to be 300–1000 cm s^{-1}. Therefore the relevant range of Re for thermal explosion studies is 30–300. The present paper reports experiments which have been performed with small bodies at low Reynolds numbers to examine the heat fluxes and energy transfer processes under these conditions.

2. EXPERIMENTAL METHOD

The method was developed from that used earlier for stagnant conditions [1]. Wires of both molybdenum and tungsten of $2{\cdot}5 \times 10^{-3}$ cm dia were immersed in a tank of distilled water and heated by a pulsed ruby laser to peak temperatures between 660°C and 900°C in a time ~ 300 μs. The wire temperature was measured by a resistivity technique, and the heat flux was determined from the cooling rate over the period of 300 μs after the end of the heating pulse. Conduction along the length of the wire was negligible. Some 2 cm from the wire in the tank was a 0·4 cm dia nozzle supplying a water flow at up to 600 cm/s (see Fig. 1). The experimental errors were estimated to be $\sim \pm 7$ per cent in temperature, mainly due to 0·03 cm uncertainty in the heated length, and ± 10 per cent random error in heat flux arising from the measurement of the slope of the cooling

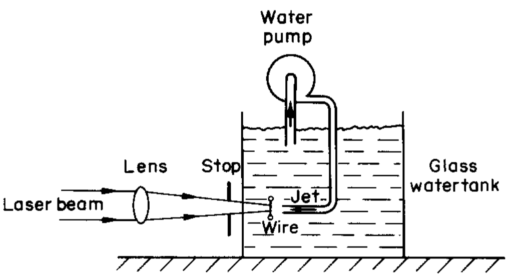

FIG. 1(a). Schematic diagram of apparatus.

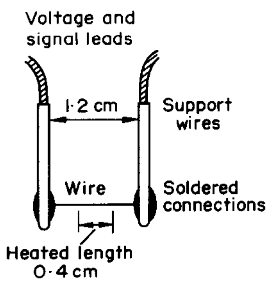

FIG. 1(b). Detail of wire assembly.

curve. The mass flow rate was determined by volume collection to within ± 5 per cent. A Hycam cine camera (9000 pps) was used to photograph the boiling on the wire.

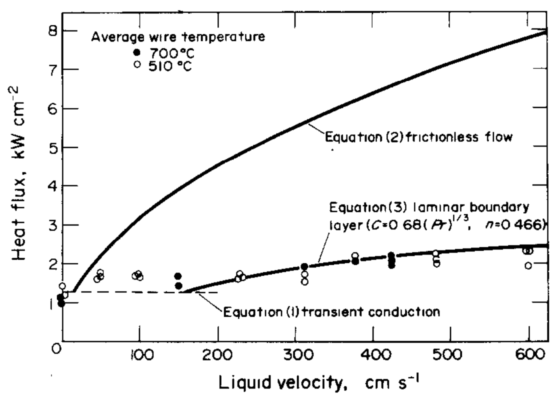

FIG. 2. Forced convection heat fluxes from $2{\cdot}5 \,.\, 10^{-3}$ wires to 30°C water.

3. RESULTS

As for the transient heating of foils in stagnant water [1] the heat flux from the wire was found to be not strongly dependent on the wire temperature as the wire cooled from 900°C to 400°C. In Fig. 2, it can be seen that the effect of water velocity is to increase the heat fluxes by a factor of of about two as the velocity increases to 600 cm s^{-1} ($Re_w \sim$ 280).

The photographs showed that a vapour layer grew rapidly on the wire, attaining a thickness that remained approximately constant during the period of heat flux measurement. A typical photograph of a tungsten wire at $\sim$600°C in 30°C water flowing at $\sim$400 cm s^{-1} is shown in Fig. 3. This shows that upstream of the wire the vapour film was thin ($< 10^{-3}$ cm) compared with the wire diameter but downstream there were irregularly spaced bubbles of typically 5×10^{-3} cm radius. Such bubbles would be expected from surface tension forces acting on a vapour cylinder (capilliary instability e.g. [7]). The equivalent uniform vapour film thickness giving the same surface area was deduced by measurement at a series of equispaced points along the wire, and it was found that the ratio of vapour to wire diameters (D_i/D_w) was $\sim 2{\cdot}0$. Under zero flow conditions similar bubbles existed but remained symmetric with respect to the wire and the value of (D_i/D_w) was found by a similar process to be $\sim 2{\cdot}3$.

4. DISCUSSION

(a) *Stagnant liquid*

The heat flux result at zero flow rate may be related to the transient conduction flux from the vapour/liquid interface into the bulk liquid if it is assumed that the whole of the visible interface is maintained at the saturation temperature. Under these conditions, for negligible net vapour generation, the following approximate relation, neglecting any shape factors, has been shown to agree with experimental data [8]

$$\left(\frac{Q}{A}\right)_w = \left(\frac{D_i}{D_w}\right)\frac{k(T_{sat} - T_\infty)}{(\alpha t)^{\frac{1}{2}}}. \tag{1}$$

This relation may be expected to still apply at low flows and is therefore shown by the broken horizontal line at 1·26 kW cm^{-2} on Fig. 2 obtained for $t = 450\,\mu$s, the time from the beginning of the heating pulse. There is good agreement with the present experimental data at zero flow (see later for $0 < U < 100$ cm^{-1}).

(b) *Forced convection*

For finite liquid velocity, slip may be expected to occur at the vapour/liquid interface. Under these conditions, an approximate (lower limit) flux assuming a transient conduction boundary layer as derived by Motte and Bromley [4] and Witte [9] is:

$$\left(\frac{Q}{A}\right)_w = \left(\frac{D_i}{D_w}\right)^{\frac{1}{2}}\frac{k(T_{sat} - T_\infty)}{(\alpha D_w/U_\infty)^{\frac{1}{2}}}. \tag{2}$$

This relation, plotted as the upper curve in Fig. 2, predicts an order of magnitude increase in flux over the experimental range.

However, it has been found [10–12] that in the presence of small amounts of surface active agents (impurities in the water), the drag on very small gas bubbles ($\gtrsim 10^{-2}$ cm) tends to approach that for solid spheres in which the fluid velocity reduces to zero at the interface. Hence the wire surrounded by the vapour layer may be treated as a solid cylinder of diameter equal to the effective diameter of the vapour/liquid interface. Assuming that the interface is maintained at the saturation temperature, the heat conduction to the subcooled liquid may be approximated by considering a uniform temperature gradient across a viscous shear boundary layer adjacent to the interface so that neglecting any shape factor and for negligible nett vapour generation,

$$\left(\frac{Q}{A}\right)_i = \frac{k(T_{sat} - T_x)}{(\nu D_i/U_\sigma)^{\frac{1}{2}}}.$$

This may be expressed in non-dimensional terms as

$$Nu_i = Re_i^{\frac{1}{2}}.$$

The recommended correlations for the average heat flux from solid cylinders (including fine wires) in non-boiling cross flow [13] are of the same form, i.e.

$$Nu_w = CRe_w^n \tag{3}$$

but take account of the reduction in heat transfer due to the effects of the shape factor and boundary layer separation, giving $C = 0{\cdot}683(Pr)^{\frac{1}{3}}$ and $n = 0{\cdot}466$ for Re 40–4 . 10^3. The predictions of this correlation (Fig. 2) are in good agreement with the data for flow velocities above about 200 cm s^{-1}, provided it is assumed that $Re_i = Re_w$. It appears therefore that the presence of the bubbles on the downstream face of the wire does not significantly increase the overall heat transfer, despite increasing the interfacial area. This may at least partly be explained by the fact that at low Reynolds numbers most of the heat transfer from a cylinder occurs from the upstream face, ~80 per cent of the total being lost between 0° and 120° from the upstream stagnation point at $Re < 500$ (14).

For velocities <200 cm s^{-1} the average flow boundary layer (~10^{-3} cm) becomes comparable in thickness to the transient conduction boundary layer for the timescale of the present experiment, and hence the flux is greater than predicted by either convection or conduction alone.

The above analysis can be shown to significantly underpredict the film boiling fluxes obtained for $Re \gtrsim 10^4$ [2–4] and in fact the fluxes are several times greater than that predicted even assuming slip at the vapour/liquid interface (equation (2)). However it is not necessary to make the physically difficult assumption of liquid/surface contact [9] since Motte and Bromley [4] have shown that such fluxes may be attributed to turbulent diffusion of heat occurring at the interface at high Reynolds numbers.

5. CONCLUSIONS

Experimental data have been obtained on subcooled transient film boiling and forced convective film boiling from fine wires at the low Reynolds numbers (30–300) relevant to the mixing and expansion stages of a thermal explosion.

The transient boiling data are well predicted by assuming that the vapour/liquid interface is maintained at the saturation temperature against transient conduction to the water. The forced convection film boiling results may be similarly predicted by assuming that the interface at the saturation temperature loses heat across a laminar flow boundary layer in the liquid, where the liquid velocity reduces to zero at the interface.

ACKNOWLEDGEMENT

This paper is published by permission of the Central Electricity Generating Board.

REFERENCES

1. S. J. Board, A. J. Clare, R. B. Duffey, R. S. Hall and D. H. Poole, An experimental study of energy transfer processes relevant to thermal explosions, *Int. J. Heat Mass Transfer* **14**, 1631–1641 (1971).
2. F. J. Walford, Transient heat transfer from a hot nickel sphere moving through water, *Int. J. Heat Mass Transfer* **12**, 1621–1626 (1969).
3. J. W. Stevens and L. C. Witte, Transient film and transition boiling from a sphere, *Int. J. Heat Mass Transfer* **14**, 443–450 (1971).
4. E. L. Motte and L. A. Bromley, Film boiling of flowing subcooled liquids, *Ind. Engng Chem.* **49**, 1921–28 (1957).
5. A. R. Davey, G. James and K. Morgan, unpublished work (1970).
6. R. W. Wright, A. F. Firstenberg and G. H. Humberstone, Kinetic studies of heterogeneous water reactors, USAEC Report STL-32-50, 24–36 (1966).
7. S. Chandrasekhar, *Hydrodynamic and Hydromagnetic Stability*. Oxford University Press (1961).
8. R. B. Duffey, A. J. Clare, D. H. Poole, S. J. Board and R. S. Hall, Measurements of transient heat fluxes and vapour generation rates in water, to be published in *Int. J. Heat Mass Transfer* **16**, 1513–1525 (1973).
9. L. C. Witte, Heat transfer from a sphere to liquid sodium during forced convection, USAEC report ANL-7296 (1967).
10. W. N. Bond and D. A. Newton, Bubbles, droplets and Stokes' law, *Phil. Mag.* **5**, 794–800 (1928).
11. F. N. Peebles and H. J. Garber, Studies on the motion of gas bubbles in liquids, *Chem. Engng Prog.* **49**, 88–97 (1953).

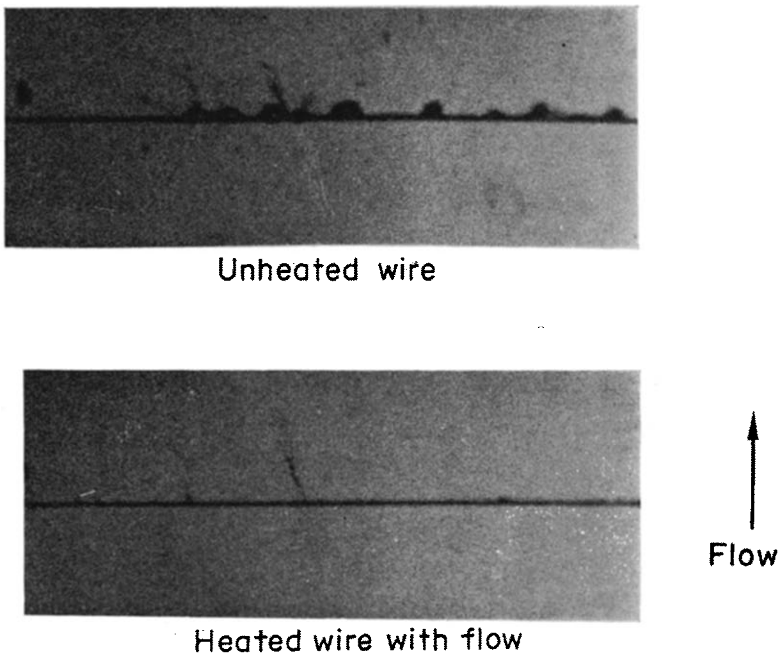

FIG. 3. Vapour film on a 2·5. 10^{-3} cm dia 600°C tungsten wire in subcooled flowing water.

12. G. K. BATCHELOR. *An Introduction to Fluid Dynamics*. Cambridge University Press (1967).
13. J. G. KNUDSEN and D. L. KATZ, *Fluid Dynamics and Heat Transfer*. McGraw-Hill, New York (1958).
14. E. R. G. ECKERT and E. SOEHNGEN, Distribution of heat transfer coefficients around circular cylinders in cross flow at Reynolds numbers from 20 to 500, *Trans. Am. Soc. Mech. Engrs* 74, 343–347 (1952).

Int. J. Heat Mass Transfer. Vol. 16, pp. 1655–1657. Pergamon Press 1973. Printed in Great Britain

HEAT TRANSFER TO GAS–LIQUID MIXTURES IN A VERTICAL TUBE FITTED WITH TWISTED-TAPES

A. F. NOORUDDIN* and P. S. MURTI
Regional Research Laboratory, Hyderabad–9, India

(*Received* 25 *May* 1972 *and in revised form* 19 *February* 1973)

NOMENCLATURE

D, i.d. of the tube [m];
h_L, single phase heat transfer coefficient for liquid flow [kcal/hm² °C];
h_{tp}, two-phase heat transfer coefficient [kcal/hm² °C];
N_{ReL}, liquid Reynolds number, $DV_L\rho/\mu$;
V_G, velocity of gas [m/h];
V_L, velocity of liquid [m/h];
ρ, density [kg/m³];
μ, viscosity [kg/hm];
λ, number of pipe diameters per 360° tape rotation.

THE introduction of twisted-tapes in the single-phase and two-phase gas–liquid streams were shown to have greatly increased the heat transfer rates and this enhancement was of the order of three times at critical heat flux and constant pumping power [1]. The aim of the present investigation is mainly to obtain data on heat transfer under swirl flow conditions employing tapes of different sizes and different flow rates of the fluid streams.

EXPERIMENTAL SET-UP

The test section was described in detail elsewhere [2]. Briefly, it consisted of a 20 cm high, 1·28 cm i.d. and 2·22 cm o.d. stainless steel tube, surrounded by a steam jacket. Fourteen 26-gauge copper–constantan thermocouples were attached to the tube at seven different levels to measure the wall temperature. The main test section was flanked by upstream and downstream calming sections of 1 m and 1·5 m lengths, respectively. Four different tape twist ratios ranging from 3·47 to infinity were investigated. The tape fitted in the test section was separated by a short distance from the tapes on each end of it to minimize the heat conduction along the tape. Steam, after passing through a water-separating tee, an entrainment separator and a distributor, was introduced into the assembly at the inlet section.

* Chemical Engineering Department, Regional Engineering College, Srinagar, India.

RESULTS AND DISCUSSIONS

In all the runs, the heat flux was calculated from the amount of condensate collected. The heat balance was checked and was found to be within ±10 per cent for most of the runs. The fitness of the experimental set up was tested by comparing single-phase data, with the values predicted by the well known Sieder–Tate equation. The agreement was within ±6 per cent, proving the worth of the set up.

TWO-PHASE SWIRL FLOW RESULTS

Heat transfer studies in the presence of twisted-tapes were carried out using air and water as the two-phase fluids at 5 different flow rates of liquid, varying the superficial liquid Reynolds number from 4060 to 27500. Figures 1–3 show some of the results obtained in this work with the variation of heat-transfer coefficient ratios plotted against the gas to liquid volumetric flow rate ratios (V_G/V_L). The plots show a general similarity in the variation of heat transfer results