RESULTS AND DISCUSSIONS

We assume that, as in other cases $K_3(\tau)$ is much smaller than $K_2(\tau)$ in this case also, and thus we confined ourselves to the discussion on $K_2(\tau)$ only.

In the asymptotic case when $\tau \to \infty$ we have calculated the diffusion coefficient $[K_2(\tau)-Pe^{-2}]$ [for various values of the Hartmann number (M), radiation parameter (F) and Rayleigh number (Ra)] and have plotted them against F and $Ra^{1/4}$, respectively, in Figs. 1 and 2. It is interesting to note that these graphs coincide with the corresponding graphs of the effective Taylor diffusion coefficient calculated with the Taylor model by Mandal *et al.* [9]. This further strengthens the use of the Gill and Sankarasubramanian model and treats $K_2(\tau)-Pe^{-2}$ as the diffusion coefficient. Following Kay [11], and Grief *et al.* [12] we worked out all calculations taking $Ra^{1/4}$ in the range 0–10^2.

Figure 1 shows that the diffusion coefficient is very sensitive to any change in the value of $Ra^{1/4}$ independently of the value of F and M namely it decreases rapidly with a decrease in $Ra^{1/4}$.

Figure 2 shows that the diffusion coefficient has a minimum value at $Ra^{1/4} = r_m$ where r_m lies between 0 and 10 depending on the value of F. When $Ra^{1/4} > r_m$ a change in F drastically changes the diffusion coefficient. However, when $Ra^{1/4} < r_m$ the effect of F is not so significant. Thus in the experimentally observed range (10–10^2) as indicated by Kay [11] the role of the radiation parameter F is very important. However, it is clear from both Figs. 1 and 2 that with an increase in radiation the diffusion coefficient decreases. This is due to the fact that due to loss of energy by radiation the velocity decreases and thereby the diffusion coefficient also decreases. Figures 1 and 2 both show that the role of M is not very significant in comparison with those of $Ra^{1/4}$ and F.

In Figure 3 we have plotted $[K_2(\tau)-Pe^{-2}]$ against τ for various values of F, $Ra^{1/4}$ and M. The asymptotic value of $[K_2(\tau)-Pe^{-2}]$ is reached at a time τ of $O(1)$ and this attainment is independent of all parameters. In the initial stage the effects of the parameters are similar to those in the asymptotic case.

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EFFECT OF CROSSFLOW IN BOILING HEAT TRANSFER OF WATER

R. L. Singh, J. S. Saini and H. K. Varma

Department of Mechanical and Industrial Engineering, University of Roorkee, Roorkee 247672, India

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NOMENCLATURE

D	outside diameter of tube [m]
G	mass velocity [kg s^{-1} m^{-2}]
h	heat transfer coefficient [W m^{-2} K^{-1}]
k	thermal conductivity of fluid [W m^{-1} K^{-1}]
n	index in the Kutateladze equation (1)
Pr	Prandtl number of fluid
q	heat flux [W m^{-2}]
T	temperature [K]
ΔT	wall superheat [K].

Greek symbol

μ	dynamic viscosity of fluid [kg m^{-1} s^{-1}].

Subscripts

b	pool boiling
F	condition at mean film temperature
f	forced convection
fb	forced convection boiling
s	saturation condition
w	heated surface condition.

INTRODUCTION

Crossflow boiling has not been adequately investigated. A few investigators [1–4] who have worked on crossflow boiling have experimentally studied the phenomenon with water at atmospheric and superatmospheric pressures, being limited to the high velocity and high heat flux ranges. However, no method of analysis has been proposed by any one of the above workers on saturated boiling, except the work of Vliet and Leppert [3] who have analysed the case of crossflow for critical heat flux.

Several authors have, however, proposed methods of analysis for boiling inside tubes when forced convection and nucleate boiling components co-exist.

Kutateladze [5] has given the following relationship for the boiling curve:

$$\frac{h_{fb}}{h_f} = \left[1+\left(\frac{h_b}{h_f}\right)^n\right]^{1/n}. \qquad (1)$$

He found that with $n = 2$, the above equation correlates the results of several experimental studies of boiling water flowing inside tubes.

Rohsenow [6] suggested a simple method of superposition of the effects of pool boiling and forced convection as

$$q_{fb} = q_b + q_f. \qquad (2)$$

This work is intended to bridge the gap in knowledge related to the studies in crossflow boiling in the low velocity, low heat flux range at atmospheric pressure. The results may be important for applications to fire-tube boilers and qualitatively, for the flooded refrigerant evaporators.

EXPERIMENTAL SET-UP

Figure 1 shows the schematic diagram of the experimental set-up. Three stainless steel tubes with outer diameters of 15.9, 12.7 and 9.5 mm and a length of 0.90 m were employed independently as test sections. These were supported horizontally on the opposite walls of a rectangular M.S. tank which was adequately insulated. The tubes were heated electrically.

Water was fed in at the bottom of the tank through a distributor and the flow across the cross-section was made uniform by means of two baffles in the form of 100 seives/in.^{2} of wire mesh. The outlet of water was provided from all the sides of the tank through a header above the test section level.

The heating of the test section produced the boiling of water. The steam formed was removed from the top of the tank to a condenser and the condensate then passed to a receiver where it met the spillover water from the header.

The flow rate of water was measured by a precalibrated orificemeter located between the pump and the test tank.

Thermostatically-controlled immersion heaters were fitted in the tank and in the receiver to keep the water at the saturation temperature. An air vent was provided in the cover of the test tank for removing any dissolved air or gas in the water. Two glass windows, one in the front and the other at the back were provided to permit visual observation and control of the boiling phenomenon.

The temperatures of the tube surface and bulk water were measured by means of 32 gauge copper–constantan thermocouples. The thermocouples were fitted on the outside surface at three sections on each tube by means of spot welding. Starting from the rear stagnation point on each section, these junctions were located at 60° intervals on the 12.7 and 15.9 mm diameter tubes and at 90° intervals on the 9.5 mm diameter tube. For bulk water temperature, thermocouple junctions were dipped in water away from the tube. The thermocouple e.m.f. was measured by means of a digital microvoltmeter with a least count of 1 μV.

The experimental investigations were carried out in the following heat flux and mass velocity ranges:

heat flux: 11 000–45 000 W m^{-2};

mass velocity: 0 (pool boiling)–2.14 kg m^{-2} s^{-1}.

Heat transfer coefficients were calculated on the basis of the average temperatures over the tube surface.

RESULTS AND DISCUSSION

Heat transfer coefficients obtained from the experimental data of pool boiling were compared with those calculated from the correlations of refs. [7–9]. The predicted values of the heat transfer coefficient were generally lower than those observed. The mean deviations were about 10, 18 and 40% for the correlations of Forster and Greif [7], Rohsenow [8] and Kutateladze and Borishanskii [9], respectively. Similar deviations have been widely reported by many previous investigators.

Measurement of surface temperatures for pool and crossflow boiling shows that the temperature symmetry exists about the vertical diameter of the tube. The temperature decreases from the forward stagnation point to the rear one i.e. from the bottom to the top.

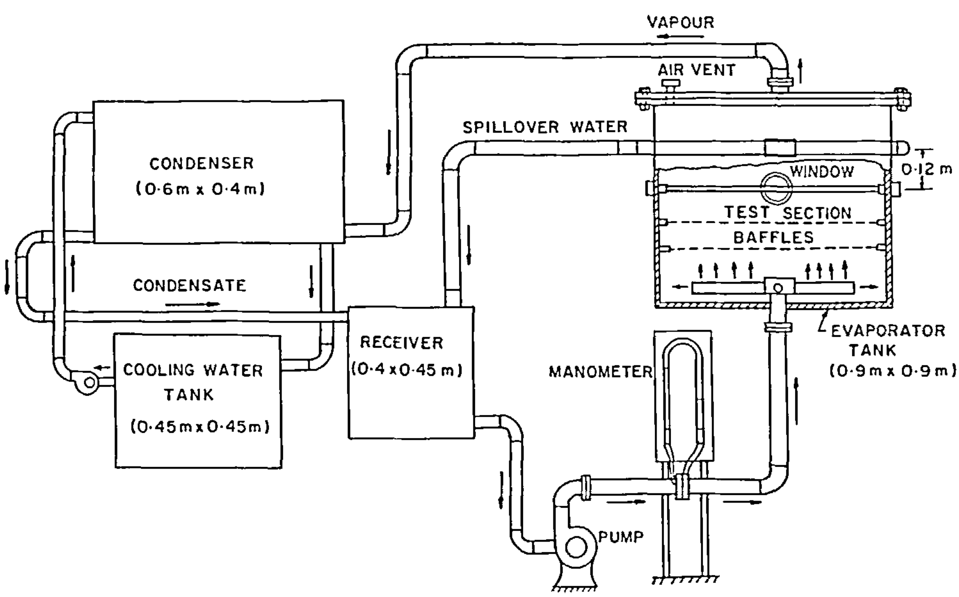

FIG. 1. Schematic diagram of the experimental set-up.

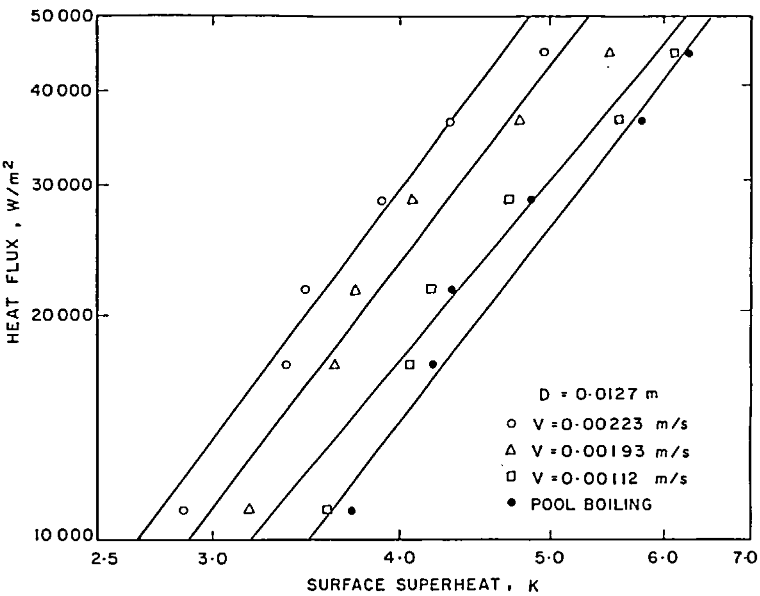

FIG. 2. Effect of velocity on boiling heat transfer.

Figure 2 shows a typical plot of heat flux as a function of wall superheat for pool boiling and three fluid velocities. It has been observed that the heat transfer coefficient increases with an increase in the fluid velocity.

An attempt has been made to correlate the experimental heat transfer coefficient by including the effect of crossflow on pool boiling. The heat transfer coefficients h_{fb}, h_b and h_f used in the development of the proposed correlation are defined as

$$\Delta T = (T_w - T_s), \tag{3}$$

$$h_{fb} = \frac{q}{\Delta T_{fb}}, \tag{4}$$

$$h_b = \frac{q}{\Delta T_b}, \tag{5}$$

and

$$h_f = \frac{k_F}{D}(Pr)_F^{0.3}\left[0.35 + 0.56\left(\frac{DG}{\mu_F}\right)^{0.52}\right]. \tag{6}$$

It may be noted that the h_{fb} and h_b values were experimentally determined; equation (6) gives the heat transfer coefficient for forced convection normal to a single cylinder [10].

On the basis of the experimental data the value of the

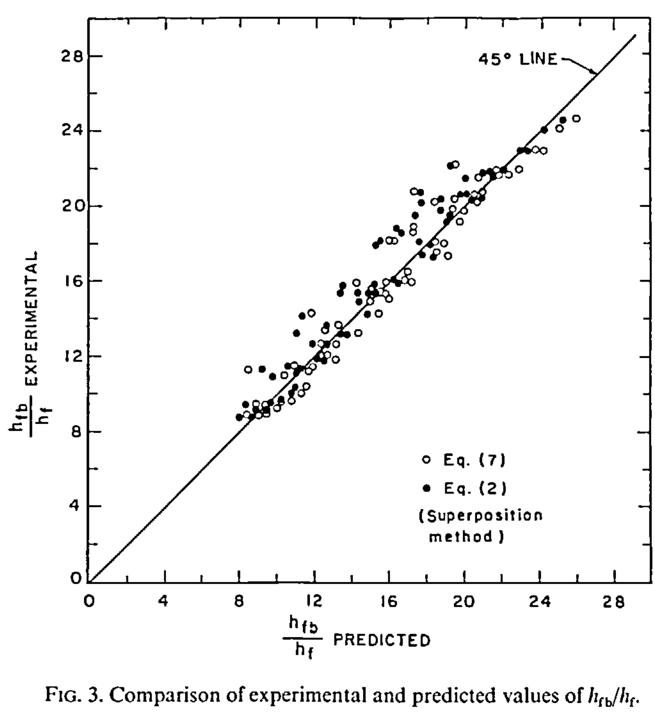

FIG. 3. Comparison of experimental and predicted values of h_{fb}/h_f.

Table 1. Standard percentage deviation of the experimental values of the heat transfer coefficient from the predicted values

Tube number	Outer diameter of tube (mm)	Percentage standard deviation from equation (7) (%)	Percentage standard deviation from equation (2) (%)
1	15.9	4.32	1.86
2	12.7	6.76	7.85
3	9.5	9.20	9.34
Overall		7.42	7.80

exponent has been determined as 0.86. Accordingly

$$\frac{h_{fb}}{h_f} = \left[1+\left(\frac{h_b}{h_f}\right)^{0.86}\right]^{1.163}. \tag{7}$$

Figure 3 shows the plot of the heat transfer coefficient h_{fb} obtained by the above correlation against the experimental data. On the same plot a comparison of h_{fb} has been made with the one obtained by equation (2). It is found from this plot that the values predicted by the two equations do not differ appreciably from the experimental values. However, the predicted 'h' values obtained by equation (7) show only marginal superiority. Table 1 has been prepared to show the standard percentage deviation of the values of the experimental heat transfer coefficient from those predicted by equation (7) as well as equation (2).

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THE EFFECT OF A MAGNETIC FIELD ON THE HEAT TRANSFER CHARACTERISTICS OF AN AIR FLUIDIZED BED OF FERROMAGNETIC PARTICLES

JULIE J. NEFF and BORIS RUBINSKY*

Department of Mechanical Engineering, University of California, Berkeley, CA 94720, U.S.A.

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NOMENCLATURE

- A_s effective area of the heat transfer probe [m^2]
- B magnetic flux density [gauss]
- d_p average particle diameter [m]
- h heat transfer coefficient [W m^{-2} K^{-1}]
- T_b bulk bed temperature [K]
- T_s probe surface temperature [K]
- u airflow velocity [m s^{-1}]
- u_{mf} minimum fluidization airflow velocity [m s^{-1}].

INTRODUCTION

A GAS–SOLID fluidized bed contains a bed of particulate matter through which a gas (air) is passed. As the airflow rate increases, four fluidized regimes can be observed; namely, the fixed bed, incipient fluidization, bubbling and slugging regimes [1–5]. It has been shown that the heat transfer is different in the four different fluidization regimes [2, 5]. The region of low air velocity corresponding to the fixed bed regime is characterized by low heat transfer coefficients. The incipient fluidization regime is characterized by a gradual increase in the heat transfer coefficients with maximal heat transfer occurring in the bubbling regime. The slugging regime is accompanied by a decrease in the heat transfer coefficient.

Recent investigations have shown that electrical fields can effect the fluid flow mechanics in fluidized beds of insulating and semi-insulating particles and consequently the heat transfer [6–8]. It has also been reported by Agbim *et al.* [9] that the fluidized bed regimes were changed when fluidizing magnetized iron shots. The onset of bubbling was suppressed and the bed stabilized when fluidizing magnetic iron shots as compared to the fluidization of iron shots of the same size but not magnetic.

This note documents the effect of an applied exterior magnetic flux on the flow regimes and heat transfer from a vertical flat surface in a fluidized bed of ferromagnetic

* To whom correspondence should be addressed.