The finning itself was found to be highly enhancing compared to an unfinned tube, yielding a $7\frac{1}{2}$-fold increase in the heat transfer rate.

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Effect of cross-flow on boiling heat transfer of refrigerant-12

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NOMENCLATURE

- b coefficient used in equation (5)
- c_{sf} constant in Rohsenow equation (2)
- D outside diameter of tube [m]
- D_b bubble diameter [m]
- G mass velocity [kg s^{-1} m^{-2}]
- h heat transfer coefficient [W m^{-2} K^{-1}]
- h_{fg} latent heat of vaporisation [J kg^{-1}]
- k thermal conductivity [W m^{-1} K^{-1}]
- Nu Nusselt number
- n index in the Kutateladze equation (11)
- Pr Prandtl number
- q heat flux [W m^{-2}]
- Re Reynolds number
- T temperature [K]
- ΔT wall superheat [K]
- V velocity [m s^{-1}].

Greek symbols
- μ dynamic viscosity of fluid [kg s^{-1} m^{-1}]
- ρ density [kg m^{-3}].

Subscripts
- b pool boiling, bubble
- F condition at mean film temperature
- f forced convection
- fb forced convection boiling
- l liquid
- s saturation condition
- su superficial vapour
- w heated surface condition.

INTRODUCTION

VERY little published work exists in the literature for cross-flow boiling of refrigerants. With R-12 in particular, such studies are almost non-existent. The purpose of this investigation was to determine experimentally the cross-flow nucleate boiling characteristics of R-12 at small velocities that are important for the design of flooded evaporators. This work is an extension of a previous study conducted by the authors on water to investigate the effect of cross-flow on boiling heat transfer [1].

Following the method of Rohsenow [2], a semi-empirical model has been proposed on the basis of superimposition of cross-flow effects over pool boiling. The experimental data of R-12 is seen to be successfully predicted by this model.

EXPERIMENTAL WORK

Figure 1 shows the schematic of the experimental apparatus used in the investigation. The test vessel was a rectangular steel container of 124 × 80 × 265 mm size, to hold the pool of boiling R-12. Stainless steel tubes with outside diameters of 16.0, 12.5 and 9.6 mm and each 11.4 mm long were successively employed as three test sections. One tube at a time was held horizontally in the test vessel by means of two copper strips. The tube was heated electrically; the power was supplied to it through the copper strips.

Multiple entry of the liquid was provided at the bottom of the test vessel through a distributor and two 100 sieve in^{-2} (16 sieve cm^{-2}) wire mesh baffles were placed in the flow path. Multiple outlets of the unevaporated liquid through a header, 50 mm above the test section level, were also provided in order to reduce the non-uniformity in the liquid flow across the test section. The vapour separator was provided to allow only liquid flow to the test vessel and the liquid level indicator showed the liquid level in the vessel. A bypass line was used to bypass most of the liquid and allow only the desired amount of refrigerant for make-up in the test vessel during the pool boiling runs. The vapour from the test vessel passed through a back pressure regulating valve to the compressor. The bypass liquid was sent to the auxiliary evaporator to ensure its complete evaporation before reaching the compressor. The compressed refrigerant then flowed through an oil separator, water- and air-cooled condensers, filter and a subcooler and finally through the rotameter to the expansion valve.

Test surface temperatures were measured by means of calibrated 32 gauge copper–constantan thermocouples carefully spot-welded on the outside surface at the top side and the bottom at two sections on each tube located 29 mm from the ends. The bulk temperature was measured by a thermocouple dipped in the liquid. All the tests were performed at a boiling temperature of 5°C. The stabilized power supply to the test section was made through an autotransformer and measured by means of a precision grade voltmeter and ammeter.

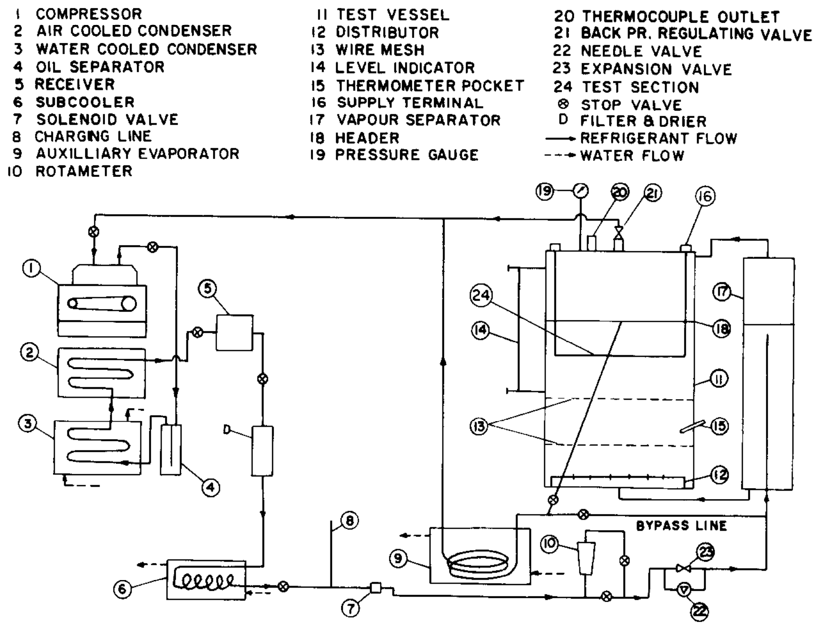

FIG. 1. Schematic diagram of experimental setup.

The experiments were performed under steady state and the following heat flux and velocity ranges were covered:

Heat flux: 15,000–86,000 W m^{-2}
Velocity: 0–0.31 cm s^{-1}.

HEAT TRANSFER MODEL

It is well known that the high rate of heat transfer in pool boiling is mainly due to bubble agitation and mixing of the liquid near the heated surface. The small bubbles adhering to the heating surface act in the manner of surface roughness and may increase the turbulence. In flow boiling the velocity of flow interacts with the growth and motion of individual bubbles, resulting in an increase of turbulence. Thus the agitation of the liquid caused by the bubbles will be more in the case of flow boiling than in pool boiling.

Rohsenow [2] added the pool boiling and convective components to suggest that the in-tube flow boiling heat transfer coefficient is given as in equation (1)

$$h_{fb} = h_b + h_f. \tag{1}$$

In the proposed model, it is assumed that similar superimposition of cross-flow occurs over the pool boiling at relatively low flow velocity and heat flux. It is also assumed that the bulk flow of the boiling liquid causes additional agitation of the liquid by the bubbles, which is responsible for augmentation of pool boiling component. It is further assumed that the heat transfer effect of bulk flow can be added to the thus augmented pool boiling heat transfer.

The model is a modification of the pool boiling correlation of Rohsenow [3], given by equation (2)

$$Nu_b = \frac{1}{c_{sf}}(Re_b)^{0.67} \times (Pr)^{-0.7} \tag{2}$$

which may be rewritten as

$$h_b = \frac{1}{c_{sf}}\frac{k}{D_b}\left(\frac{G_b D_b}{\mu}\right)^{0.67} \times (Pr)^{-0.7} \tag{3}$$

where G_b is given by q/h_{fg} for saturated liquids.

For a particular fluid and constant properties, equation (3) yields

$$h_b \propto G_b^{0.67}.$$

Since the superficial vapour velocity, V_{su}, is given by G_b/ρ_v,

$$h_b \propto V_{su}^{0.67}. \tag{4}$$

When cross-flow of liquid occurs the superficial vapour velocity is augmented by the bulk velocity and it influences the pool boiling component. If the flow velocity is small such that there is no suppression of nucleation and the departure diameter could be assumed to be invariant with velocity, the effective vapour velocity, V_{fb}, can be written as

$$V_{fb} = V_{su} + bV \tag{5}$$

where b is an, as yet, undefined variable.

Thus the bubble Reynolds number in the pool boiling increases in the ratio of $(V_{su}+bV)/V_{su}$ and the first term in equation (1) increases to

$$\left(1+\frac{bV}{V_{su}}\right)^{0.67} \times h_b.$$

Thus, on the basis of equation (1), the total heat transfer coefficient for cross-flow boiling is given by

$$h_{fb} = \left(1+\frac{bV}{V_{su}}\right)^{0.67} \times h_b + h_f. \tag{6}$$

It may be noted that in the above relation V_{su} appears in the denominator suggesting that as the heat flux increases, contribution due to convection decreases which is a common observation in flow boiling.

RESULTS AND DISCUSSION

Figure 2 shows a typical plot of heat flux as a function of wall superheat for pool boiling and cross-flow velocities. It is observed that the heat transfer coefficient increases with increase in flow velocity. The almost equal slope of the lines suggests that there is a power law relation between q and ΔT; the index is independent of velocity.

The heat transfer coefficients h_{fb}, h_b and h_f, used in the development of the proposed correlation, are as follows:

$$h_{fb} = \frac{q}{(T_w - T_s)_{fb}} \tag{7}$$

$$h_b = \frac{q}{(T_w - T_s)_b} \tag{8}$$

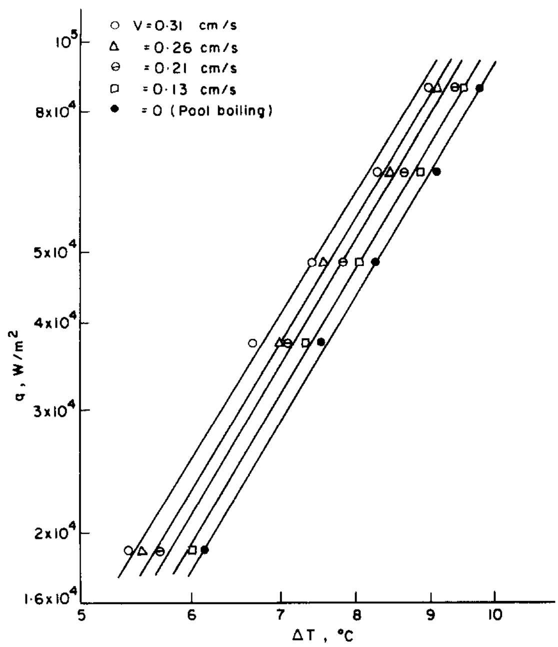

FIG. 2. Effect of velocity on boiling heat transfer of R-12 for tube diameter = 9.5 mm.

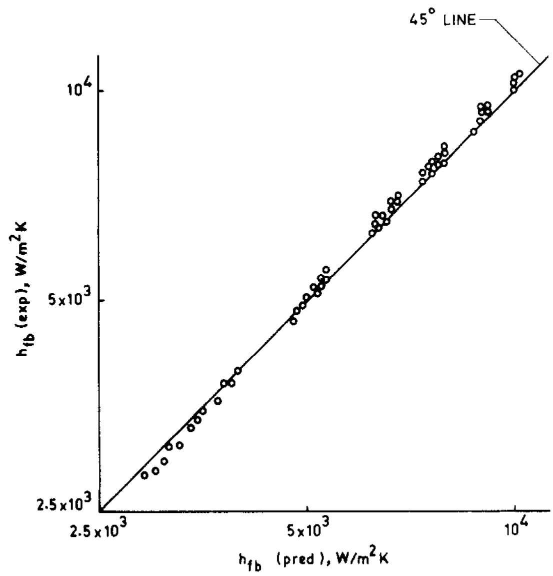

FIG. 3. Comparison of experimental and predicted heat transfer coefficients for crossflow boiling (for R-12) by modified superposition method.

and

$$h_f = \frac{k_F}{D}(Pr_F)^{0.3}\left[0.35+0.56\left(\frac{DG}{\mu_F}\right)^{0.56}\right] \tag{9}$$

$$1 < \frac{DG}{\mu_F} < 1000.$$

The values of h_{fb} and h_b were experimentally determined and equation (9) is an empirical equation used to calculate the heat transfer coefficient for forced convection normal to a single cylinder [4].

The value of b in equation (6) was found to be 0.4 to best represent the experimental data. Thus the developed correlation is given by

$$h_{fb} = \left(1+\frac{0.4V}{V_{su}}\right)^{0.67} h_b + h_f. \tag{10}$$

Figure 3 shows the comparison of experimental and predicted values of h_{fb}. It is clear from this graph that the agreement is very good.

The experimental values of heat transfer coefficients were also compared with those obtained from the Rohsenow correlation, equation (1), and Kutateladze [5] equation (11)

$$\frac{h_{fb}}{h_f} = \left[1+\left(\frac{h_b}{h_f}\right)^n\right]^{1/n} \tag{11}$$

Kutateladze recommended a value of n equal to 2, but the deviation of values obtained from equation (11) with $n = 2$ was 7.35% and unacceptable. A similar observation was made in the earlier study [1]. The best fit value of n was determined numerically for this study and was found to be 0.69. The percentage standard deviation of heat transfer coefficient values obtained from equations (1), (10) and (11) from those of experimental values were found to be 6.182, 2.718 and 2.817, respectively.

This suggests that the experimental data is well correlated by the proposed model.

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