rates in terms of local Nusselt numbers for the convergent nozzles were significantly greater than for the orifices away from the stagnation region. Since greater heat transfer rates occurred in regions adjacent to the stagnation region, local heat transfer distributions with the convergent nozzles were also more uniform. Such improved uniformity is highly desirable in air jet applications to mitigate hot, cold, or wet spots. The greater heat transfer rates were larger for $H/d = 2$ in comparison to those for $H/d = 6$ and occurred along all three axes. It has been observed in steady jets that secondary vortices on the impingement surface resulting from incident coherent flow structures do not form for $H/d > 6$ (Fox et al., 1993). Evidence of incident coherent flow structures in the flows of this study have been documented by Sheriff (1997). Consequently, the larger increases for $H/d = 2$ and more uniform distribution may indicate the presence of more effective secondary vortices near the boundary layer. For $H/d = 2$ and $0.6 < x_1/l_1 < 1.0$, local Nusselt numbers were between 34 percent and 38 percent larger than those for the orifice array. The largest difference at all separation distances occurred for $x_1/l_i > 0.3$. The spatially averaged heat transfer for the nozzle array was 26.2 percent higher than the results for the orifice array and 36 percent higher than given by the correlation of Martin (1977) for arrays of short straight tubes. It should be noted that these tubes were referred to as nozzles by Martin (1977), but differ from the convergent nozzles of this study. For $H/d = 6$, the spatially averaged heat transfer with the convergent nozzles was 16 percent and 48.9 percent higher than values for the orifice and from the short tube correlation, respectively.

Small off-center peaks in Nusselt number are evident in Fig. 2 on either side of the stagnation point at $x_1/l_1 = x_2/l_2 = x_3/l_3 \approx 0.2$. Locations agreed with the locations of off-center peaks observed experimentally in jet arrays with orifices (Huber and Viskanta, 1994) and numerically within laminar jet arrays (Chen et al., 1994). Since both of the jet flows in these cited studies did not contain coherent flow structures, it can be discerned that the peaks are related to boundary layer development in the stagnation region and not to the interaction of incident flow structures with the surface. Reductions in stagnation point Nusselt numbers due to pulsating flow conditions are also evident in Fig. 2, where $S_d = 0.0044$. Reductions became greater with increasing pulse magnitude and have been theoretically explained by a nonlinear dynamics model of the hydrodynamic and thermal boundary layer responses to a periodic incident flow velocity (Mladin and Zumbrunnen, 1995). Flow pulsations were found to effectively increase the time-averaged thermal boundary layer thickness and lead to reduced heat transfer in a stagnation region. Convective heat transfer distributions thereby became still more uniform. However, at higher pulsation frequencies within the frequency range of this study, a more abundant succession of incident vortices was produced in the pulsed jet so that nonlinear dynamical boundary layer effects became secondary to improved mixing (Sheriff, 1997). Nusselt numbers for the nozzle array were as much as ten percent larger as a consequence than those for the corresponding steady flow case. When orifices were used, a higher initial turbulence level was generated in the discharging jet and coherent flow structure development was impeded even under pulsating flow conditions. Measurable enhancements in Nusselt numbers were therefore not obtained with the orifice arrays even at the highest Strouhal number considered.

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Enhancement of Saturated Flow Boiling Heat Transfer on Cylinders Using Interference Sleeves

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Introduction

Many techniques have been developed in recent years to enhance boiling heat transfer. Bergles (1988), Thome (1990), and Webb (1994) have reviewed most of these enhancement techniques for both pool boiling and flow boiling. Abhat and Seban (1974) studied the effect of copper mesh around a cylinder in water. Hasegawa et al. (1975) studied various woven meshes for boiling water on a plate. Asakavicius et al. (1979) examined surfaces with several layers of copper screen for boiling water, R-113 and ethanol. Typical enhancements were not very large, about 100 percent at low heat fluxes and much less at high heat fluxes. A recent experimental study on pool boiling by Shimada et al. (1991) showed that an interference plate (a plate with small holes located on a given pitch) placed with a small clearance over a copper heating surface traps the vapor bubbles stably. These holes in the interference plate facilitates smooth the exchange of vapor-liquid and thereby higher heat

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Table 1 Sleeve geometries used in the present study

Sleeve no.	Hole diameter d, mm	Pitch p, mm	Clearance δ, mm	Wall thickness t, mm	No. of holes per m^2 (N/A)
1	1	4	0.36	2.78	8.8×10^4
2	1	8.3	0.36	2.78	2.0×10^4
3	1	14.3	0.36	2.78	7.0×10^3
4	1, 2 (comb.)	4, 8 (comb.)	0.36	2.78	5.8×10^4 2.7×10^4
5	1	4	0.92	3.58	9.1×10^4
6	1	4	2.3	2.25	9.1×10^4
7	1, 2 (comb.)	14.3, 24.8 (comb.)	0.36	2.78	4.5×10^3 1.6×10^3
8	2	4	0.36	2.78	8.8×10^4

fluxes at lower wall superheats between the heating surface and the liquid are achieved. Shimada et al. (1991) claim that the performance of this arrangement was comparable and sometimes even better than some of the commercially available high-performance boiling heat transfer surfaces. In their study, a 6-mm thick polycarbonate sheet was used as the interference plate with varying hole geometries (size and pitch) and clearance above the heated surface. The Shimada et al. (1991) study showed that the "best" boiling curve was obtained when combination holes of 1 mm and 4 mm diameter were used with 8.1 mm and 14 mm pitch (respectively) and with 0.12 mm clearance. The primary advantage of using an interference plate over a boiling surface as a heat transfer enhancement technique over other enhancement techniques is the easy cleaning of the heat transfer surface of fouling deposits. This is particularly important in certain process applications such as desalination. Secondly, preparation of the enhanced surfaces does not require any complicated mechanical processing.

Rao and Balakrishnan (1997) studied subcooled flow boiling over a horizontal heated tube (flow in the axial direction) covered with an interference sleeve having holes in a triangular pitch. The sleeves were made of aluminum. The liquid boiled was distilled water at atmospheric pressures. They found that the various parameters which influence the boiling heat transfer performance are sleeve geometry (hole geometry and the clearance between the heating cylinder and the interference sleeve), mass velocity of the boiling liquid and the liquid inlet subcooling. Furthermore, it was found that the heat transfer enhancement factor is greatly reduced with higher subcoolings. In the present study, experiments have been carried out at saturated conditions.

The details of the experimental setup, the test section, and the arrangement of the sleeve on the heating cylinder have been presented elsewhere (Rao and Balakrishnan, 1997). In the experiments, the applied heat flux was varied and the temperatures of the heating cylinder and the liquid were measured. Since the vapor qualities encountered in the present study are small (up to a maximum value of 0.01), the results are presented in a format similar to those used in subcooled boiling and in pool boiling. Eight perforated sleeves were used to examine the sleeve geometries. The geometrical details of these sleeves are given in Table 1.

Results and Discussion

Figure 1(a) shows the effect of hole pitch on boiling heat transfer at the inlet liquid mass velocity of 577 kg/m^{2}s. The hole diameter and the gap clearance are also maintained constant at 1 mm and 0.36 mm, respectively. At all heat flux levels the boiling curve for sleeve 1 gives lower wall superheats compared to plain tubes. In the case of sleeve 2 (with a hole pitch of 8.3 mm), there is no enhancement in the heat flux range from 1.3×10^4 W/m^2 to 3×10^4 W/m^2. Increasing the heat flux to a value above 1.4×10^5 W/m^2 results in the wall temperature shooting up which leads to film formation. For sleeve 3 (with a hole pitch of 14.3 mm), the enhancement in heat flux is several times that of a plain tube. Transition to film boiling is observed at a heat flux of about 4.6×10^4 W/m^2. As the pitch of the holes is increased from 8.3 mm to 14.3 mm, the film transition heat flux decreases from 1.4×10^5 W/m^2 to 4.6×10^4 W/m^2. The boiling curve obtained with sleeve 4 (with combination holes of 1 mm and 4 mm diameter on 4 mm and 8 mm pitch

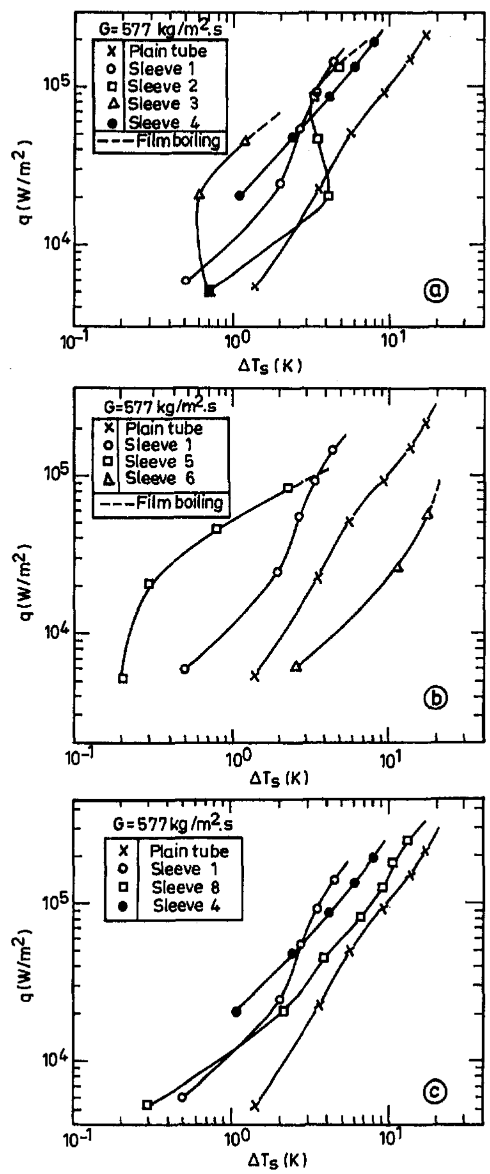

Fig. 1 Effect of sleeve geometry on the boiling curve (a) effect of hole pitch; (b) effect of gap clearance; (c) effect of hole diameter

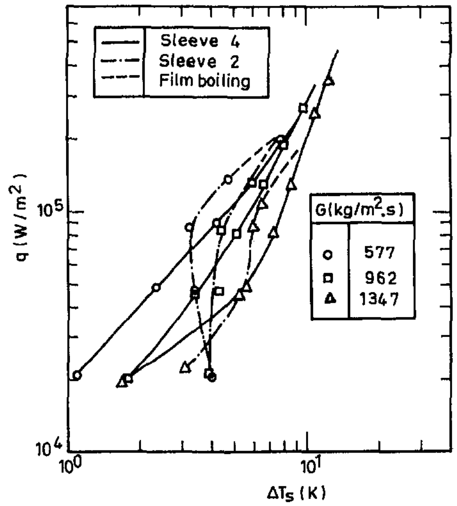

Fig. 2 Effect of liquid inlet mass velocity on the boiling curve using sleeve 2 and sleeve 4 geometries

respectively) are also shown in the figure. This shape of this curve is typical of standard boiling curves and there is no tendency towards transition to film boiling.

Figure 1(*b*) shows the effect of gap clearance at an inlet mass velocity of 577 kg/m^{2}s. Three gap clearances 0.36, 0.92, and 2.3 mm were used. The hole diameter and the pitch of the holes are the same at 1 mm and 4 mm, respectively. The results show that at all heat fluxes sleeve 5 (0.92 mm clearance) gave better enhancement compared to sleeve 1 (with 0.36 mm clearance). This is most likely due to the larger coalescence of the vapor bubbles. According to Ishibashi and Nishikawa (1969), the coalesced bubble regime gives rise to higher heat transfer coefficients than the isolated bubble regime. For sleeve 5, transition to film boiling occurred at a heat flux of 9×10^4 W/m^2. This transition occurs because as the heat flux is increased, the number of holes may be insufficient to discharge the large amount of vapour generated and it accumulates in the large gap clearance causing the vapor-liquid exchange equilibrium to be disturbed. In the case of sleeve 6 (gap clearance of 2.3 mm) the situation is quite different. Sleeve 6 always gave larger wall superheats compared to the plain tube. This is due to the existence of a vapor layer in the large gap. Visual observations showed that at low heat fluxes, vapor was coming from the top part of the sleeve intermittently in slugs. At higher heat fluxes the vapor slugs are seen all around the sleeve. At a heat flux of 5.6×10^4 W/m^2, the gap clearance was completely filled with vapor.

Figure 1(*c*) shows the boiling curves for three sleeves, one with 1 mm diameter holes (sleeve 1), one with 2-mm dia. holes (sleeve 8) and one with combination holes of 1 mm and 2 mm on 4 mm and 8 mm pitch, respectively (sleeve 4). It can be observed that the boiling curve with 2-mm diameter holes shifts towards the left of that obtained with a sleeve of 1-mm diameter holes until a heat flux value of 1.2×10^4 W/m^2. This is due to lower incipient heat fluxes required with larger hole diameters. Film boiling and unusual boiling behavior (decreasing wall superheat with increase in heat flux) are not observed with sleeves with 2-mm diameter holes. The boiling behavior obtained with sleeve 4 (combination holes) is even superior to sleeves with 2 mm holes at heat flux levels beyond 1.2 $\times 10^4$ W/m^2.

Three inlet mass velocity 577 kg/m^{2}s, 962 kg/m^{2}s, and 1347 kg/m^{2}s were used to obtain data. Figure 2 shows the effect of mass velocity on the boiling curve using sleeve 2 geometry. It can be observed that as the mass velocity increases from 577 kg/m^{2}s to 1347 kg/m^{2}s the boiling curve shifts towards the larger ΔT_s values. It can be also observed that the film boiling occurs at a heat flux value of about 1.3×10^5 W/m^2 for the three mass velocities considered. Similar results were noticed for sleeve 3 also. It can therefore be concluded that the film transition heat fluxes are not affected by the mass velocity at least for sleeve 2 and sleeve 3 geometries. This is different from the observations on subcooled flow boiling over interference sleeves (Rao and Balakrishnan; 1997) where it was seen that the film transition heat fluxes are delayed with increasing mass velocity. Figure 2 also shows the boiling curves obtained with sleeve 4 (combination holes geometry). The effect of mass velocity is similar to that obtained with sleeve 2, but the shape of the boiling curves are similar to standard boiling curves and there is no tendency towards transition to film boiling.

Enhancement Factors. Figure 3 shows the enhancement factors of the eight sleeves used in the present study at different heat flux values where the enhancement factor "E" is defined as the ratio of the excess temperature of the plain tube to that of a sleeve at a given heat flux and mass velocity. The figure shows that sleeve 3 has enhancement factors ranging from 8 to as high as 13. But it has the disadvantage of film formation at a heat flux of 4.6×10^4 W/m^2. The enhancement factors seen with sleeves 5 and 6 are not more than 3. In addition, there is the disadvantage of film formation. Film formation was not

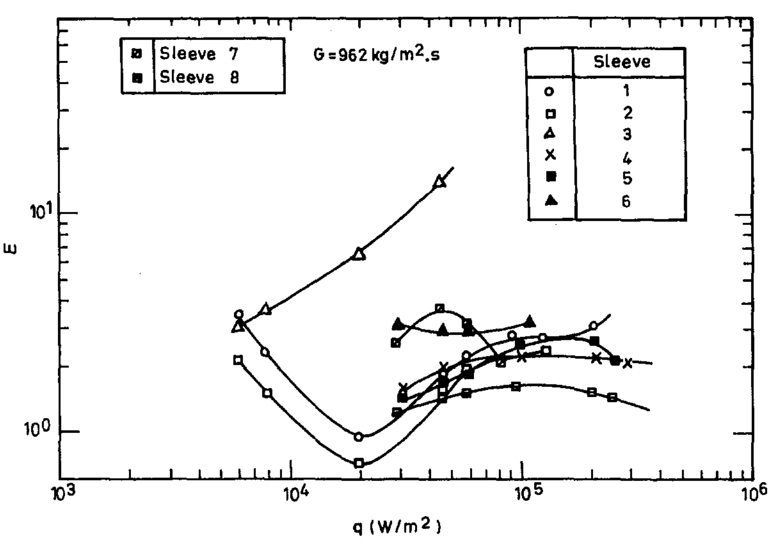

Fig. 3 Enhancement factors at various heat flux values at a mass velocity of 962 kg/m^{2}s for the eight sleeves used in this study

encountered with sleeves 1 and 4. Sleeve 4 has enhancement factors ranging from 1.5 to 2.2 and these values are almost constant with heat flux. Sleeve 2 shows a maximum enhancement factor of 2.2 just before film boiling occurs. Sleeve 8 gives a maximum enhancement factor of 1.6 without film formation. All the data in Fig. 3 are at a mass velocity of 962 kg/m^2s. At these flow rates, the maximum enhancement factors encountered under subcooled conditions were about two (Rao and Balakrishnan, 1997). In fact, at higher subcoolings (around 20 K), the enhancement factors were less than one indicating poorer performance than with plain tubes.

Conclusions

The perforated interference sleeve, as a boiling heat transfer enhancement technique, gives larger enhancement factors at saturated flow conditions compared to subcooled flow conditions. High enhancement factors were encountered with large hole pitch sleeves but with the disadvantage of an early transition to film boiling conditions. This transition heat flux is not influenced by the liquid mass velocity. Sleeves with large gap clearance of 2.3 mm gave poor performance at 577 kg/m^2s mass velocities; i.e., at all heat flux levels they gave high ΔT_s values. The "best" boiling curve with high enhancement factors and with high film transition heat fluxes at low heat flux levels, was obtained with the sleeves with 2 mm holes. The best boiling curve at higher heat fluxes was obtained using sleeves with 1 mm, 2 mm combination holes on 4 mm, 8 mm pitch, respectively.

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A New Method for Tracking Radiative Paths in Monte Carlo Simulations

B. M. Shaughnessy[1,3] and M. Newborough[2,3]

Nomenclature

c = elements of matrix $\mathbf{C}$
$\mathbf{C}$ = matrix of cumulative density functions
F_{ij} = exchange factor from surface i to surface j
n = number of surfaces
N = number of energy bundles tracked
p_{ij} = transition probability from surface i to surface j
p_{ijk} = transition probability from surface i to surface k via a single reflection from surface j
$\mathbf{P}$ = matrix of transition probabilities
R = uniformly generated (pseudo) random number ($0 \leq R \leq 1$)
s = specularity parameter (Eq. (5))
$z_{\alpha/2}$ = percentage point of the Standard Normal Distribution
ϵ = emissivity
θ = polar angle, rad

Subscripts

e = enclosure
i, j, k = surface indices
I = incident
o = obstruction
R = reflected

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Introduction

The Monte Carlo (MC) approach is a powerful means of solving radiative heat transfer problems involving realistic geometries and properties. As described elsewhere (e.g., Siegel and Howell, 1972; Modest, 1993), radiation exchange factors, F_{ij}, between surfaces in an enclosure are calculated by ray-tracing (or tracking) the paths of a large number of "energy bundles" according to probability functions that describe emission, reflection, and absorption. Unfortunately, the time penalty associated with computing ray-surface intersections is high, and this inhibits high-accuracy solutions. This article presents a new method for tracking the reflected paths of energy bundles, which is much faster than traditional ray-tracing approaches (Shaugh-

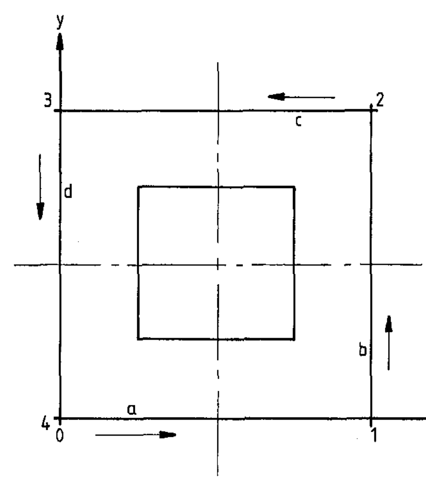

Fig. 1 Two-dimensional enclosure with central obstruction. The faces *a*, *b*, *c*, and *d*, and the dimensionless wall location around the enclosure perimeter are indicated.