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Multiple Jet Impingement Cooling

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Experiments were performed to study the effect of nozzle rotation, flow rate, degrees of subcooling, number of jets, and velocity on the heat flux in jet impingement cooling using deionized distilled water. The heat flux is a strong function of the flow rate and degree of subcooling, attaining a maximum value of the order of 600 W/cm² at the highest flow rates. The heat flux shows no significant dependency on velocity and number of jets for fixed flow rates. The rate of rotation shows no significant effect on the heat flux within the forced convective and early nucleate boiling region. At temperatures close to the critical heat flux the rotation retards the heat flux (compared with a stationary jet) diminishing it by 20%. Comparisons with published results are presented.

Nomenclature

d = diameter of heated surface, m
d_n = diameter of jet, m
G = mass velocity, ρu, kg/m²/s
H = latent heat, J/kg
k = thermal conductivity, W/m/K
q = heat flux, W/m²
T = temperature, °C
u = jet velocity, m/s
w = uncertainty
x = Cartesian coordinate, m
Δ = difference
ρ = density, kg/m³
σ = surface tension, N/m

Subscripts
co = critical heat flux
cu = copper
fg = vaporization
l = liquid
n = nozzle
surf = surface
v = vapor
1 = top thermocouple
ε = subcooling

Introduction

RESEARCH in the field of high heat-flux rejection is gaining importance with the rapidly advancing technological developments incurring concentrated heat loads as a result of higher efficiency, miniaturization, and extended thermal capabilities. Hence, methods of enhanced heat removal must be devised which can provide safe operation for devices such as computer components, gas turbine blades, and reentry vehicle nose tips.

Advances in spray cooling have shown that heat fluxes higher by an order of magnitude of two are feasible when compared to forced convective cooling technology. This is because the energy is removed in the form of latent heat for phase change as opposed to sensible heat in convection. Consequently, surface temperatures are limited to the region of the boiling point, which can be significantly lower than that required for convection for the same heat removal capability. Droplet spray cooling,[1-3] and jet impingement cooling[4-8] have proven to be far more efficient than the conventional pool boiling method of heat removal.[9-12] This is partially because the hydrodynamic instabilities inherent in pool boiling in the region of the critical heat flux (CHF) are precluded by the deposition of a thin film of liquid[13,14] on the surface, expediting vapor escape and promoting droplet-surface interaction. Empirical correlations derived from experiments performed using droplet impingement do not agree well, primarily due to the difficulty of producing controlled configurations of droplet sizes and velocities. Pressure nozzles[3] and gas atomized nozzles[15] produce varying spectrums of droplet sizes and velocities based on nozzle design, gas and liquid flow rates. On the other hand, jets can be characterized and reproduced as long as impingement is prior to jet breakup.[21] This is because the coolant flowfield is determined by the diameter of the jets, their configurations, and the coolant flow rate.

Surface cooling can be made efficient if the hot surface is wetted with a thin film[15] of continuously replenished coolant, i.e., at any instant there exists a thin layer of liquid on the hot surface. One way of achieving this is by using an array of rotating jets. A close study of droplet impingement on a surface (see Fig. 1), will reveal that it is analogous to a surface being impinged by an array of rotating jets. Thus, the water impinges on a part of the surface for an instant, delivering a quantity of liquid which can be determined by knowledge of the jet diameter, normal, and tangential velocities. A stationary jet impinging on a stationary surface exhibits a disc-shaped stagnation flow with an annular hydraulic jump at the periphery of the disc.[22] When this jet is rotated about a radius arm, the disc stagnation flow traverses a circular path, covering an annular area. By rotating the multiple jets, the immediate jet flowfield is imposed on the whole surface.

A number of experiments have been performed using single stationary jets[4-8] of water under various conditions of velocity, and surface-to-nozzle characteristic length ratios. Katto et al.[16] gave a review of the research in saturated jet cooling in the CHF region. Experiments were performed with R12, R113, and water under conditions of direct impingement of a jet on a surface at various angl›s, horizontal and upside down. Ruch and Holman[7] determined that the nucleate boiling heat flux was a function of the degree of superheat. Cho

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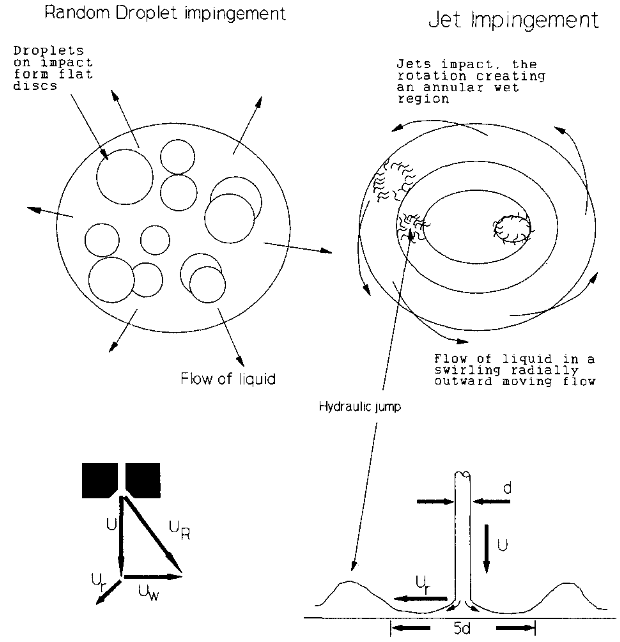

Fig. 1 Droplet impingement vs rotating jet impingement.

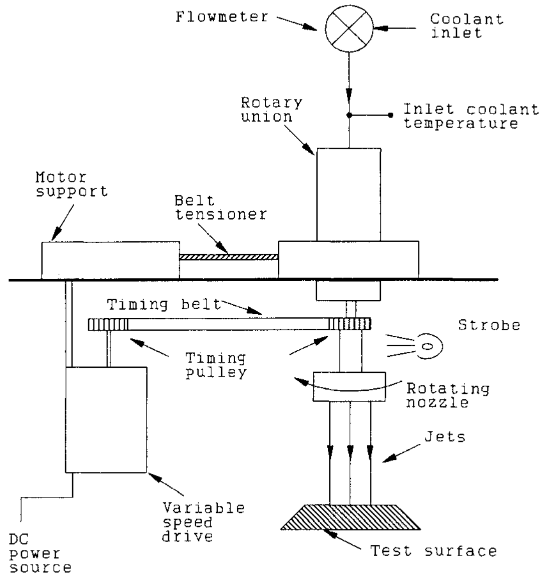

Fig. 2 Rotating jet setup.

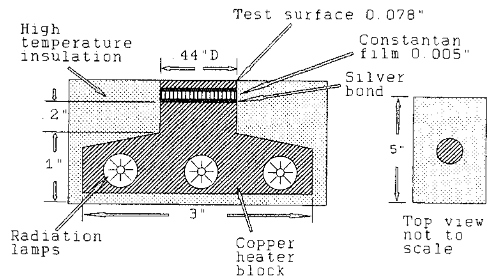

Fig. 3 Test surface design.

and Wu[17] in a limited study showed that the burn-out heat-flux was dependent on the liquid jet velocity. They also gave a comparison of the physical phenomenon occurring in both jet impingement and spray cooling. High heat fluxes have been obtained in forced convective subcooled boiling in internal flows as well as external flows.[18] However, such surfaces have been maintained at high degrees of superheat. This study endeavors to 1) examine the effect of rotation of an array of jets about a radius arm on the heat flux; and 2) examine the effect of the number of jets and their configuration on the heat flux.

Development of Apparatus

A schematic of the rotating nozzle assembly is shown in Fig. 2. A rotary union facilitates the supply of coolant to a jet nozzle which is rotated by a timing belt-pulley arrangement powered by a variable speed dc motor. Since the jet is rotating on a radius arm (see Fig. 1), a tangential velocity is imposed which would cause the jet to impinge the surface at an angle; the angle, if not large, does not influence the heat transfer process.[19]

The components of the rotary jet system are shown in Fig. 2. Using a variable flow gear pump, distilled, deionized, and filtered (5 μm) water is pumped at a predetermined flow rate measured using a calibrated orifice flowmeter. The predefined inlet temperature (to maintain degree of subcooling) to the jet nozzle is maintained by a temperature controller in conjunction with a preheater located in a vapor trap.

Owing to the large thermal gradients required to supply large heat fluxes (up to 1000 W/cm^2) to a surface, care must be taken in the design of the heater body[15] so that temperatures within the body do not harm its physical and chemical integrity. With reference to Fig. 3, three tungsten-in-quartz heat lamps (500 W each, temperature = 2500 K, $0.5 \leq \lambda \leq 4.5$ μm) are inserted into cylindrical chambers within the copper heater block. The radiating tungsten filament enclosed within the cylindrical chamber constitutes a black body. Therefore, all the radiative energy is absorbed into the copper base of the heated surface. These radiation lamps have a fast thermal response (99% rated power within 3 s), can withstand high temperatures by virtue of their sealed quartz envelope, and provide a high radiative heat flux. Using a finite element method, the heater body design was optimized assuming a uniform test surface heat flux of 1000 W/cm^2. The lower heater body was cuboid of dimensions $5 \times 3 \times 1$ in. The test surface was of cylindrical geometry and had dimensions of 0.44-in.-diam $\times$ 0.2 in. in length. The lower heater body and the test surface were machined from a single piece of copper (OHFC).

The heater body consists of two parts: 1) the lower copper body with inserted radiation lamps, and 2) the 1-cm^2 $\times$ 0.078-in. circular crown from which heat removal is desired. The whole heater body, excluding the top surface of the circular crown, and the thermocouple leads were well insulated with high temperature insulation. This is then placed in an aluminum box, from which the power leads for the lamps and thermocouples are accessed, and compressed air for cooling the lamp terminal ends is injected. The whole assembly is then covered with aluminum foil, such that only the 1-cm^2 top surface of the circular crown is exposed to the coolant. Any excess coolant flows off the aluminum shroud and into the sink.

A temperature controller[15] monitors the temperature of the thermcouple surface, maintaining it at some preset value by regulating the power input to the quartz lamps. In the event of an overshoot in the temperature of the surface, an alarm disconnects the power to the heaters by means of relays. The power to the lamps along with the temperatures are continuously indicated by the panel meters and recorded by the PC (personal computer) controlled data acquisition system. All process parameters are displayed and stored on the PC.

Given a system where the temperature gradient is one-dimensional and linear, the heat flux can be determined using Fourier's law of heat conduction

$$q = k(\Delta T/\Delta x) \tag{1}$$

At a desired heat flux of 1000 W/cm^2, in a copper body (k = 393.8 W/m/K) of uniform cross section, the temperature gradient is

$$(\Delta T/\Delta x) = 25.4°\text{C/mm} \tag{2}$$

This requires that the size of the thermocouple be of the order of 40 μm for a temperature measurement resolution to be within 1°C. Secondly, an uncertainty analysis

$$(w_q/q) = \sqrt{(w_x/\Delta x)^2 + (w_T/\Delta T)^2} \qquad (3)$$

implies that the prediction of the heat flux is also dependent on the accurate measurement of the distance Δx between the two thermocouples. Ideally, Δx is a parameter in the heat flux equation which should be optimized so as to be within the allowable error and simultaneously maintain system temperatures as low as possible.

The common method of temperature measurement using fine thermocouples is not recommended because the wire sizes required are too fine at the high temperatures expected, being more susceptible to standard wire errors,[20] corrosion and failure.

An alternative method of temperature measurement would be to silver solder a thin film of constantan, (selected because with copper it composes a thermocouple, type T), interleaved between a copper piece on the top (which makes up the test surface) and the heated copper block at the bottom, see Fig. 3. Using finite element thermal modeling a heat flux of 1000 W was imposed on the lamps (hence, modeling assumes a maximum heat flux of 1000 W). Assuming no losses from the block, and a test surface temperature of 150°C, the design of the crown was modified in the numerical model such that flat isotherms were predicted within the crown. This laminated thermocouple construction was then located in the region of the crown where flat isotherms were predicted. For a heat flux of 600 w/cm^2, using a constantan film (k = 21.12 W/m/K) of thickness Δx = 150 μm, a temperature difference of 42.6°C is obtained across the film. Such a temperature difference is easily measured. The thermocouple junctions existing are now those of copper-silver-solder and silver-solder-constantan. Independent calibrations of copper-silver-solder and silver-solder-constantan were performed against a standard copper-constantan thermocouple over the temperature range 0–400°C. These calibrations were used in the measurement of temperature. Having determined the heat flux from Eq. (1), and knowing the thickness of the top copper test surface (0.078 in.), the test surface temperature is extrapolated. This surface temperature is an average over the surface.

Experimental Procedure

Before each experiment, the test surface was cleaned lightly with 1/0 emery polishing paper, then swabbed with hydrochloric acid, and finally washed with deionized distilled water. Deionized, distilled water was used as the coolant in all experiments. All experiments were performed with the nozzle 11 ± 1 mm above the test surface. This distance was selected because it afforded clear visual access to the jets and the surface. Tests performed by the authors validate that the jets did not breakup prior to 14 mm.[21]

When the nozzle is stationary, the impinging jets spread out radially to form thin films in the region of impingement, coalescing with the other jet films in their outer regions, and bounded by a hydraulic jump[22] in the form of raised ridges (see Fig. 1). The height of the film measured using a traveling telescope attachment ranged between 0.5–1.5 mm. Because the film is of the order of a millimeter in thickness, conduction through the film and subsequent evaporation at the liquid surface is insignificant. This is true because the thermal conductivity of water is low (0.65 W/m/K) and would require high test surface temperatures for appreciable heat transport by way of conduction and evaporation at the surface. Hence, the only alternative for heat transport at low degrees of superheat are forced convection and subcooled boiling. In the case when the nozzle was rotated, swirling flows (the liquid impinging the surface has tangential momentum) were set up on the surface at a rotational speed much lower than that of the nozzle (see Fig. 1). Also the region where the jet impinged had a skewed disc profile, the film thickness still being of the order of 1 mm. The flow visualization was made possible with the aid of a strobe.

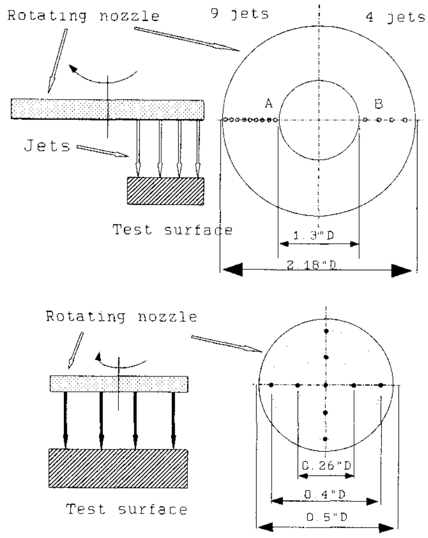

Fig. 4 Nozzle surface jet configurations.

Experiments were begun with the surface at room temperature and the temperature was ramped-up continuously until CHF conditions were imminent, at which time the heaters were automatically shut-off and the recorded data analyzed. Cooling cycle data is not taken, because after burnout conditions occurred a thin layer of oxide formed on the surface changing its characteristics. A copper surface was selected because of its high thermal conductivity and wettability with water.[2]

Surface Profile Information

The surface roughness was measured using a diamond tip profilometer with a resolution of 0.1 μm. Over six scans of the test surface were taken and the average roughness values obtained were R_q = 4.4 μm and R_p = 9.7 μm. Here, R_q is the geometric average roughness defined as the rms of the roughness heights about a mean line. R_p is the maximum height of the profile above the mean line within the assessment length.

Nozzle Construction

Various jet configurations were designed and used in the experiments. Figure 4 illustrates two methods of traversing the jets over the test surface. The axis of the rotary nozzle was aligned so as to have all the jets impinging as uniformly on the surface as possible. The uniformity of impingement is characterized by traversing each jet over equal test surface areas. This uniformity of impingement did not prove to be critical. The number of jets was varied from 4 to 9. The diameter of the orifices d_n varied from 127 to 368 μm. The specified flow rate is the amount of coolant actually impinging on the surface.

Results and Discussion

Experiments were performed with jet velocities ranging from u = 1.2–21 m/s, with a heated test surface of area 1 cm^2 and jets ranging in diameter d_n = 127–368 μm. The number of jets per nozzle ranged from n = 4–9, rotation rates range from 0 to 2100 rpm and flow rates ranged from 1.1 to 4.7 l/h. Experiments were also performed for different inlet coolant temperatures (measured at nozzle, see Fig. 2) ranging from 20 to 95°C. Uncertainty in the test surface temperature prediction is 2°C based on the uncertainty in the measurement of the constantan and silver solder films thicknesses and the calibration. The related uncertainty in the heat flux is of the order of 33 W/cm². The uncertainty in flow rate is 0.15 l/h and in $d_n = 25\ \mu$m.

Effect of Flow Rate on the Heat Flux

Figure 5 is a plot of the heat flux vs extrapolated surface temperature. The water flow rate impinging the surface was varied from 1.4 to 4.2 l/h. The inlet water temperature was held steady at 27°C. An orifice plate with nine jets (each of diameter 127 μm) was rotated at 2100 rpm for each of the cases. It is observed that the maximum heat flux increases with flow rate, the maximum heat flux increasing at a rate of approximately 87 W/cm²/l.

In the low heat flux region, i.e., less than 100°C, all the heat removed is through forced convection. A pronounced change in slope in the heat flux profile is noted around 102°C for the lowest flow rate of 1.4 l/h. Nucleation sites manifested by small bubbles on the surface begin to appear above 100°C, increasing in number density, size, and evolution with increase in surface temperature. This causes a steep increase in the heat flux with temperature, till a maximum is attained at $T_{surf} \simeq 113$°C. Owing to the dense nucleation, the liquid film thickness increases to about 2 mm (measured using a traveling telescope). Above 115°C, the surface is saturated with nucleation sites and the boiling is extremely violent. Owing to the thicker film and the violent boiling, the region of jet impingement (stagnation region as illustrated in Fig. 1) is totally enveloped and obscured. Above 140°C the surface temperature is not steady, but oscillates with a differential of 5–25°C. This is due to the intermittent blowing off of the liquid film and rewetting of dry spots. As the temperature is further increased (greater than 140°C) dry spots begin to appear on the surface, a thin oxide layer forming in the dry regions due to their higher temperature. Such dry spots increase in size, precipitating a sudden jump in the temperature. At this point the surface temperature has risen to over 215°C, the jets do not wet the surface any more (Leidenfrost region). The water jets on impinging the surface form large droplets, which roll off the surface.

Effect of Velocity and Number of Jets

Experiments performed at the same flow rate for different velocities, i.e., different number of jets or different jet diameters, indicate no appreciable change in the maximum heat flux attained. Figure 6 clearly illustrates that for a given flow rate, rotation speed, and inlet water temperature, no significant increase in heat flux (note uncertainty in the heat flux is ≃33 W/cm²) occurs for change in jet velocity (made possible by using different number of jets). Furthermore, experiments

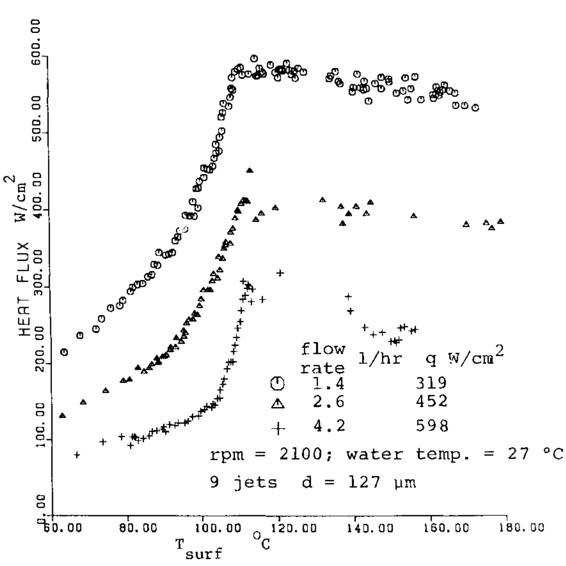

Fig. 5 Effect of flow rate on heat flux.

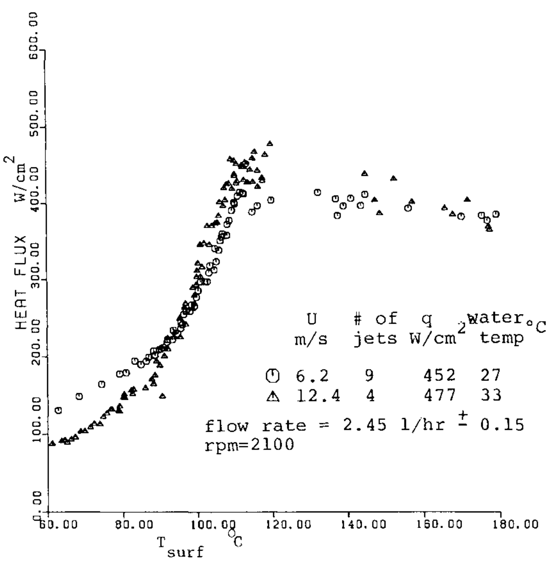

Fig. 6 Effect of velocity on heat flux.

performed (see Fig. 4 for different jet configurations) also indicate that the heat flux is not very sensitive to the placement of the jets, as long as the surface is uniformly wetted. A study of the flowfield on the surface indicates 1) a thick film (0.5–2 mm) of coolant in a swirling flow on the surface (rate of rotation of the swirling flow is much lower than that of the nozzle); and 2) the jet penetration to the surface through the film is limited to a width of approximately 5 jet diameters. Thus, based on the stagnation area imposed by the jets, at any instant only 3% of the surface experiences the direct influence of the jet stagnation flowfield, the rest being under the influence of thick liquid-film pool boiling. This suggests that most of the heat is removed by subcooled boiling, the coolant film thickness and the flowfield within governing the heat removal. Cho and Wu[17] using a single stationary jet indicated an increase in heat flux with increasing velocity, and ascribed this increase to the higher fluid momentum. Such would be the case if forced convective boiling is the dominant heat transfer mechanism. The use of a single jet precludes the interference with the flowfields of other neighboring jets (as would be the case with multiple jets), which leads to higher velocities within the flowfield on the surface.

Effect of Inlet Water Temperature

For this study an orifice plate with four jets, of diameter 127 μm was selected. The flow rate was set at 2.3 l/h and rotation at 2100 rpm. Figure 7 indicates that as the inlet water temperature is increased, the heat flux at any temperature decreases. For any two given situations, the difference in heat flux at any temperature remains the same up to the point of maximum heat flux. As explained earlier, due to the thickness of the film residing on the surface, the dominant modes of heat transfer available are forced convection and subcooled boiling. By increasing the water temperature, subcooled boiling is reduced, thus the decrease in heat flux. By using coolant at near saturation conditions (92°C), significant loss in heat removal capability is incurred.

Effect of Rate of Rotation

One of the objectives of this study was to examine the effect of rotating the jets about a radius arm. Figure 8 illustrates the influence of rotation on the heat flux. The flow rate was set at 2.3 l/h and an orifice plate having four jets of diameter 127 μm was used. Within the forced convective and incipient boiling region, no significant difference is noted between the three different rotation rates (0, 1200, and 2100 rpm). Experimental observations indicate a more stable heat removal capability is manifested by the stationary jets at higher tem-

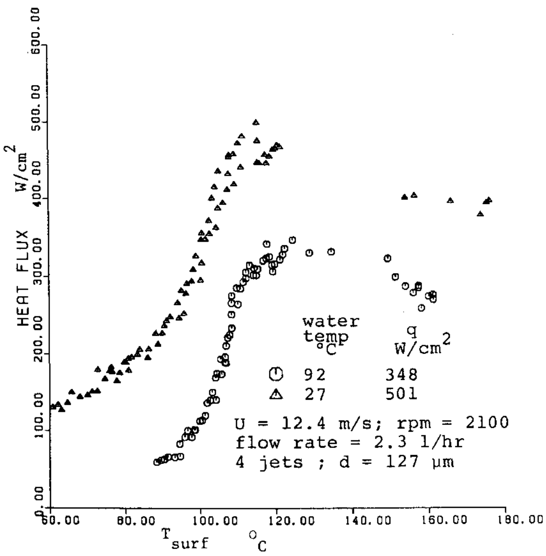

Fig. 7 Effect of inlet coolant temperature on heat flux.

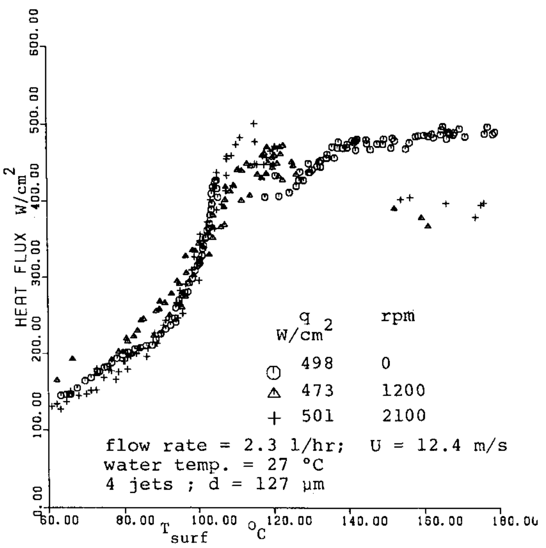

Fig. 8 Effects of rotation rate on heat flux.

peratures (>120°C). For the rotating jets, past the point of maximum heat flux, (T_{surf} = 110–120°C), the heat flux begins to decrease with increasing superheat. Such trends are perhaps a result of the coolant flow mechanics on the surface. In the case of the stationary jets the fluid after impingement flows out radially from the surface. However, when the orifice plate is rotated, an additional circumferential component is introduced. This induces a radially-outward-moving swirling flow (see Fig. 1) within the liquid film on the surface. As a consequence, the residence time of the coolant on the surface and the film thickness increases.

Katto and Yokoya[16] have analyzed CHF data for a single stationary jet under saturated temperature conditions, and arrived at a generalized correlation for the CHF covering the vapor to liquid density ratio ρ_v/ρ_l = 0.000624–0.189 and d/d_n = 3.9–53.9. Their analysis covers the range of jet diameter d_n = 0.7–4.1 mm, test surface diameter d = 10–60 mm and velocities of u = 0.3–60 m/s. The present experimental data falls within the same domain and, hence, a comparison was sought. Katto and Yokoya[16] correlated the CHF q_{co} to G, d, and d_n at saturation temperature and observed that the CHF q_{co} can be related to G, d, and d_n by the relation

$$\frac{q_{co}}{GH_{fg}} = K\left[\frac{\sigma(\rho_l - \rho_v)}{G^2(d - d_n)}\frac{1}{(1 + d/d_n)}\right]^m \tag{4}$$

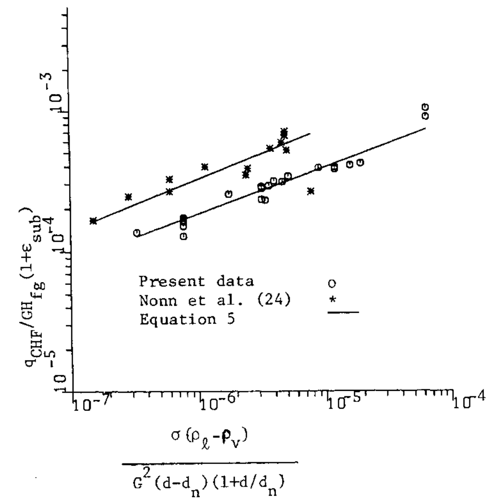

Fig. 9 Comparison of empirical relation with results.

where K = 0.0169 and m = 0.334,[16] for water at atmospheric pressure. The denominator of the left side of Eq. (4), GH_{fg}, can be conceived as the inherent heat removal capability of the flowfield. In the above relation the surface to nozzle diameter ratio d/d_n for multiple nozzles is defined as a function of the total areas of the jets, i.e.

$$(d/d_n) = \sqrt{\left(d^2 \Big/ \sum d_n^2\right)} \tag{5}$$

This is because the direct influence of the jet is experienced within the stagnation region which is proportional to the area of the jet. The contribution of the degree of subcooling to the CHF is introduced in the correlation Eq. (4) by accounting for the inherent sensible heat capacity of the fluid prior to phase change. Thus, the inherent heat removal capability is increased by a factor which can be included in Eq. (4) in the form[23]

$$\frac{q_{co}}{GH_{fg}(1 + \varepsilon_{sub})} = K\left[\frac{\sigma(\rho_l - \rho_v)}{G^2(d - d_n)}\frac{1}{(1 + d/d_n)}\right]^m \tag{6}$$

where

$$\varepsilon_{sub} = K_\varepsilon\left(\frac{\rho_l}{\rho_v}\right)^r\left[\frac{C_{pl}(T_{sat} - T_l)}{H_{fg}}\right]^s \tag{7}$$

Monde and Katto[6] obtained values of K_ε = 2.7, r = 0.5 and s = 2.0 for subcooling up to 30°C. Subcooling in the present study was extended to 78°C. Figure 9 is a comparison of all the data in the range d/d_n = 13.7–44.4, d = 11.3, d_n = 127–368 μm, and u = 3.3–20.5 m/s, with the above correlation given by Eq. (6). New constants

$$K_\varepsilon = 0.578, \quad r = 0.232, \quad s = 2.0 \tag{8}$$

were noted to provide the best fit. Experimental data from experiments performed with stationary multiple jets by Nonn et al.[24] using FC-72 have been presented using the new correlation, Eq. (8), in Fig. 9. The correlation agrees well with experiment.[24]

Conclusions

Experiments were performed using a system of rotating multiple jets to cool a surface under various conditions of d/d_n, flow rate, rotation rate, and subcooling.

The heat flux increased appreciably with increase in flow rate. No significant change with jet velocity, number of jets, or rate of rotation was observed at any particular fixed flow rate. The heat flux decreased appreciably with increase in inlet water temperature. Katto and Yokoya's[16] correlation was modified to include the effect of subcooling. This correlation showed good agreement with the present and published results.[24]

Further increase in heat flux can be realized if efforts are directed towards maintaining a very thin film on the surface (less than 3 μm). A thin film enhances evaporative cooling by conduction and reduces resistance to vapor escape. Methods of fabricating orifice plates with orifices of less than 25 μm and means of sweeping off the excess coolant to assist in the application of thinner films have to be developed. Only then can a rotating jet be envisaged to have an enhancing effect over stationary jet cooling.

Acknowledgments

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