Surface Roughness and Its Effects on the Heat Transfer Mechanism of Spray Cooling[4]

R. Mesler.[5] These investigators have presented interesting results, which certainly reveal that surface roughness has an effect on the heat transfer in spray cooling. The exceptionally high heat fluxes that are achieved with spray cooling are remarkable. This discussion centers on two interpretations they make of their results. First, they only recognize heterogeneous nucleation as the source of nuclei for the nucleate boiling, which appears to account for the exceptionally high heat fluxes. Research has shown that another mechanism can operate to cause nucleation in liquid films, that is, secondary nucleation. Second, they postulate the existence of an ultrathin liquid film but provide no direct experimental support. An examination of the postulated ultrathin liquid film relative to their other observations reveals a contradiction that questions the suitability of the postulate.

First, the paper discusses only heterogeneous nucleation as the mechanism responsible for the onset of nucleation in their experiments, but research over the last 15 years has provided evidence for another mechanism that operates to produce nucleation in liquid films (Bergman and Mesler, 1981; Carroll and Mesler, 1981; Kim et al., 1983; Mesler and Mailen, 1977; Mesler, 1982; Udomboresuwan and Mesler, 1986). When liquid drops strike a liquid surface, they entrain gas or vapor bubbles. When the liquid is superheated these entrained bubbles serve as nuclei for more bubbles to grow. When nucleate boiling occurs in a liquid film, bubbles bursting from the film produce drops close to the liquid film. Some of the drops return to the film where they can entrain bubbles and thus produce a chain reaction. The bubbles bursting from the film thus sustain further nucleation. The process has been called secondary nucleation after the analogous process that produces nuclei in crystallization. By supplying a liquid film on a heated surface with a spray, as in the present study, abundant sources of nuclei are created that are independent of the surface temperature.

Their study includes results of a pool boiling experiment where it was observed that a surface temperature of about 105°C was necessary for the onset of nucleation. They observed that for spray-applied liquid films nucleation occurred at temperatures below this. This observation can be interpreted as an indication of nucleation initiated by drop-entrained bubbles.

Second, the authors do not examine a postulated ultrathin liquid film in light of all their other observations. These authors state in their conclusions that "For roughness greater than 1 μm, nucleation plays a major role in the heat transfer. However, for films of the order of 0.1 μm, heat is conducted through the (ultrathin liquid) film and evaporated on the surface." This implies that for a surface roughness less than 1 μm the spray-applied liquid film will be on the order of 0.1 μm thick. Apparently the basis for this statement is that it is conceivable that a postulated film 0.1 μm thick might exist on a polished surface with roughness less than 1 μm. The authors offer no direct evidence that a film of such a thickness exists. It appears significant that the portion of the heat flux versus surface temperature plots for surfaces polished with any of the abrasives studied look quite similar at surface temperatures above 100°C. The plots all level off at a high value for the heat flux but the value is higher the smoother the surface. The authors state, "Spray cooling of an ultrathin liquid film on a flat surface enhances heat removal by evaporation, nucleate boiling being enhanced by early bubble departure." This implies that nucleation occurred in the liquid film applied to the smoothest surface as well as in the films applied to the rougher surfaces. How can the authors be so sure that the mechanism for heat removal is so different for the smoothest surface at the high surface temperatures and high heat fluxes when the plot of heat flux versus surface temperature is so similar? How is it that nucleation is being enhanced by early bubble departure? Is this a new mechanism for nucleation?

It is useful to examine the suitability of the postulated ultrathin film by considering it with respect to the observation that an increase in liquid flow rate is accompanied by an increase in the heat flux. An increase in liquid flow rate must cause an increase in liquid film thickness. Yet an increase in film thickness should cause a decrease in the heat flux according to the logic applied elsewhere in the paper. There is therefore a contradiction that questions the suitability of the postulated ultrathin liquid film in explaining the situation. The logic applied here is the same logic as the authors use in arguing that the thinner film accounts for the increase in heat flux with an increase in air flow rate.

A couple of other comments would seem to be in order. The authors explain that heat removal can be enhanced by maximizing microlayer evaporation. However, they do not mention the crucial role of the liquid film in making this possible (Mesler, 1982). In a liquid film a bubble can escape very rapidly because the bubble needs only to break out of the film. Furthermore, the film very quickly re-establishes itself. Escape is so quick that the microlayer does not have time to dry out as when a bubble grows by itself in an abundance of liquid. In that case the microlayer drys out well before the bubble escapes, leaving the surface dry and delaying further microlayer evaporation until the bubble has departed and another bubble grows.

The investigators found that increasing either or both air flow rates and liquid flow rates increased the maximum values of heat flux. Increasing either the air flow rate or the liquid flow rate should act to supply more liquid to the film. It has been noted in previous research that taking additional steps to keep a surface wet with a liquid film increases the maximum heat flux (Mesler, 1982).

Spray applying a liquid film on a surface with a temperature above the saturation temperature of the liquid benefits heat transfer in two ways. First, droplets in the spray entrain bubbles to provide an abundant supply of bubble nuclei to sustain nucleate boiling in the film. Second, the film acts in concert with the nucleate boiling to promote microlayer evaporation where heat can be conducted from the surface to the vapor region through an extremely thin liquid film, the microlayer.

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Authors' Closure

The authors appreciate Professor Mesler's interest in our paper. We would like to respond by stating our current understanding on the heat transfer mechanisms in spray cooling.

The current knowledge of the thermophysics of spray cooling on surfaces is based on observations in experiments performed

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using primarily air atomizing nozzles using water (Sehmbey et al., 1992a, 1992b; Pais et al., 1992; Yany et al., 1993a, 1993b, 1993c; Tilton, 1989). Figure 1 of Pais et al. (1992) illustrates the spray impingement scenario. Within a nozzle a thin film of water is injected into an accelerating stream of air. The liquid film disintegrates to form a full-cone spray of droplets in the range 10 μm < diameter < 18 μm and 25 m/s < velocity < 58 m/s. The spray is configured so as to ensure droplet impingement on the entire test surface.

The test surface was of a solid copper (OHFC) block construction. The area was approximately 1 cm^2. The thickness of the copper block was greater than 6 mm. Tests with thin stainless steel (Kenning, 1992) and other resistor film surfaces (Gu et al., 1993) indicate that under pool boiling and forced convective spray, cooling temperature variations of the order of 30°C to 7°C, respectively, exist on the surface. The high thermal diffusivity of copper is assumed to assist in the equilibration of surface temperature.

The droplets that impinge on the surface flatten on impact into thin disks whose thickness can be much lower than the diameter of the drop, assuming a dry surface (Chandra and Avedisian, 1991). The droplet momentum is lost in the process of impact with the superjacent liquid film. At the instant of impact and within the region of impact the transient film velocities are high. However the average liquid film velocity on the surface is an order of magnitude less than that of the droplets. The existence of a superimposed gas stagnation flow field has the additional effect of squeezing the deposited liquid film further (Yang et al., 1993b). Under conditions of zero heat input, 1 liter/h flow rate, and 20 psig air pressure, the liquid film on the surface is 85 μm thick and almost flat (± 1 μm change in flatness) (Yang et al., 1993c).

Raising the temperature of the surface above that of the liquid forces heat into the liquid film. As the liquid film flows radially outward on the surface it picks up heat through forced convection and rises in temperature. Radial spokelike rivulets appear dividing the film into sectors. The formation of these rivulets has been ascribed to surface tension variations brought about by temperature variations within the liquid film (Chandra and Avedisian, 1991). Depending on surface conditions—roughness (Berenson, 1962; Pais et al., 1992), material (Sehmbey et al., 1992a), and the thermofluid flow field within the liquid film and the surface (Hsu, 1962)—nucleation will begin. Incipience of nucleation is first noted at the outer edge (Yang et al., 1993d), the liquid reaching saturation conditions here first. In the region of nucleation the film thickness is also noted to decrease significantly (Yang et al., 1993d). This annular region we will term the thin film. As the heat flux from the surface is increased this concentric thin film region recedes toward the center of the surface. Also the high momentum of the droplets causes droplet shattering and rebound (Tilton, 1989). When bubbles burst (Mesler and Bellows, 1988) or implode, secondary droplets are ejected into the ambient. This secondary droplet flow is swept away by the air stagnation flow field. Such secondary droplets can contribute significantly to the excess fluid loss from the surface (Tilton, 1989).

The presence of nucleation has unequivocally been observed in stationary thin films (Mesler, 1976) as well as in thin films produced by spray impingement (Yang et al., 1993d). The microlayer at the base of the bubble is said to be responsible for the high heat flux capability with nucleation. The microlayer is thin enough so that the heat can be conducted through it and then evaporate at its surface to cause the bubble to grow. When this microlayer is depleted, the bubble growth is through evaporation at the walls of the bubble. The thermal conductivity of water is very low (0.68 W/m·K). Hence, compared to the microlayer evaporation, the circuitous route of bubble wall evaporation is much slower. In thin film nucleation a bubble emerges out of the film and bursts, and the site is replenished with a new microlayer. Droplet impingement enhances heat transfer further through nucleation by dispatching the bubbles prematurely (pre-emptive puncturing of submerged bubble by impinging droplet) forcing a fresh microlayer to form. Thus, bubble wall evaporation is superseded by the more efficient microlayer evaporation. It must be noted that nucleation in water in an open container will occur only above 100°C.

The liquid next to the solid wall, the sublayer, is superheated. When a droplet impacts the liquid film, or a bubble bursts or implodes, it induces a lot of turbulence in the region of impact. The hot sublayer is supplanted by the cooler drop and supernatant liquid. Packets of the displaced superheated sublayer (Kays and Crawford, 1993) reach the surface of the film where they can evaporate directly into the atmosphere. Because of the existence of a secondary-gas stagnation flow above the liquid film, the partial vapor pressure at the surface will be much less than the superimposed ambient pressure. Hence, the hot liquid at the surface of the film is supersaturated and its evaporation is enhanced even further.

If the film on the surface is thin enough (of the order of a few microns) then direct conduction through the liquid film and evaporation at the film surface into the ambient is conceivable. A simple conduction analysis in water indicates a 1.5°C/μm/10^6 W/m^2 which implies that a liquid film of 10 μm thickness and a superheat of 15°C could support a heat flux of 10^6 W/m^2. In this event evaporative heat transfer could compete with nucleation.

Water droplets that exist in an air–steam–water ternary phase medium have adsorbed on their surface monolayers of air molecules. When these droplets impact the surface of the superjacent liquid film, the adsorbed layer of air is liberated as a number of microscopic bubbles within the liquid (Gopalen and Mesler, 1990; Longuet-Higgins, 1990; Huang and Hammitt, 1972). Also through a process of decompression in the region of impact, between the droplet and the liquid film, cavitation is possible (Chandra and Avedisian, 1991; Huang and Hammitt, 1972). A number of studies (Mesler and Bellows, 1988; Gopalen and Mesler, 1990) have illustrated that these bubbles will act as homogeneous nucleation sites if the superheat within the liquid is sufficient (5.5°C for water (Longuet-Higgins, 1990)). The ability of a bubble to grow within a liquid is also dependent on the ability of the liquid to transport the heat to the bubble walls within which evaporation is to occur. In water, a conduction analysis as presented above indicates that to support large heat fluxes the required temperature gradient within the liquid is high. It must be noted that the sensible heat capacity of water is negligible compared to its latent heat content. Hence, even though having superheated liquid is a condition for bubble existence (Hsu, 1962), it is not a sufficient condition for bubble growth, which leads to high heat fluxes. However, if the film is very thin, the possibility of a conduction-convection path through the liquid, from the hot surface to the secondary bubble, exists. Then such secondary nucleation will play a significant role.

Roughness has been shown to play an important part in pool boiling (Berenson, 1962) and in spray cooling (Pais et al., 1992). By providing cavities for trapping vapor it advances the incipience of nucleation at lower superheats. When the surface is of a mirror finish, few nucleation sites exist. Then the heat transfer is limited to forced convection, thin film evaporation, and secondary nucleation. Because of the low thermal conductivity of water the latter two processes can support high heat fluxes (Pais et al., 1992) only if the superjacent liquid film is ultra thin. Observations of the liquid flow field for the very smooth surface (Pais et al., 1992) were obscured by the billowing steam. A definitive contribution of nucleation, secondary nucleation, and evaporation was not feasible with the apparatus used. The minimum thickness of the liquid film is influenced by the average height of the roughness elements. Above a threshold liquid film thickness, the superheat required to drive heat transfer through conduction–convection at the surface is larger than that required for nucleation. In this case nucleation will predominate.

In conclusion the above observations indicate that several heat transfer mechanisms coexist and contribute significantly to the high heat fluxes dissipated during spray cooling. The studies performed so far have not had the capability to isolate and determine the individual contributions of the aforementioned mechanisms.

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Natural Convection With Radiation in a Cavity With Open Top End[6]

C. Balaji[7,9] and S. P. Venkateshan.[8,9] The subject paper reports the results of a numerical study of natural convection and surface radiation in a cavity with open top end. The objective of the present discussion is to bring to focus certain deficiencies in the modeling of the equations by Lage et al. Because of these, conceptual errors have been committed in reporting the effect of surface radiation in the combined mode problem. The details of the problem geometry and coordinate system for the natural convection problem and the simultaneous natural convection and radiation problem are given, respectively, in Figs. 2 and 8 of the paper. A cursory look at the figures exposes certain deficiencies. For the convection problem the bottom wall has been assumed to be adiabatic while for the combined mode problem the bottom wall has been assumed to be at the temperature T_2, which is taken to be the same as the temperature of the right wall. Specification of the first and second type of boundary conditions for the bottom wall is mathematically inconsistent. This is all the more jarring, because while solving the combined mode problem, free convection results obtained in the absence of surface radiation along with an adiabatic bottom wall have been used. Temperature T_2 has been evaluated (for the right and bottom wall) assuming a balance between convection and radiation as shown in Fig. 8. This assumption, apart from causing errors, is fundamentally incorrect, and the authors fail to mention the very important fact that the question of convection from the right wall does not arise at all if the right wall does not receive heat radiatively from the left wall. As a consequence, a conceptual error has been made in reporting the effect of radiation; this point will be elaborated a little later.

The correct way of handling the combined mode problem would be to assume that the right wall and bottom wall are in convective and radiant balance ($q_{\mathrm{conv}} + q_{\mathrm{rad}} = 0$), i.e., the walls are truly adiabatic (Balaji and Venkateshan, 1993a). With this condition the convection equations along with the radiosity equations have to be simultaneously solved. This results in a temperature distribution on both the bottom and right side walls. The above procedure has been used and elaborate results have been obtained for the problem considered by Lage et al. and communicated elsewhere by the authors (Balaji and Venkateshan, 1993b).

The CPU time reported for the pure convection problem was 1500 CPU seconds for Ra $= 10^5$, on a supercomputer facility. However, even with the computationally more involved procedure of solving radiation and convection equations simultaneously for the same Rayleigh number, our calculations consumed only 750 CPU seconds on a Siemens system (5 MIPS) for nonblack walls and only 150 CPU seconds for black walls, with a finite volume algorithm for the flow equations and Hottel's method for radiation. Use of this procedure not only reduces the errors that will be made by using simpler models but is also consistent with the physics of the problem under consideration.

Finally, in the section on the results reporting the effect of radiation on the overall heat transfer, the definition that has been used for the effect of radiation is

$$\text{Enhancement due to radiation} = \frac{q_{\mathrm{conv(left)}} + q_{\mathrm{rad(left)}}}{q_{\mathrm{conv(left)}} + q_{\mathrm{conv(right)}}} \quad (1)$$

where $q_{\mathrm{rad(left)}}$ is given as the sum of the radiative heat transfer from the left wall to wall 2 and the left wall to the ambient. It will now be shown that Eq. (1) is actually incorrect. If one considers the actual physics of the problem, the left wall, being an isothermal heat source, heats the right and bottom walls radiatively to a temperature above the ambient temperature. This gives rise to the development of boundary layers on the right and bottom walls and they lose heat convectively to the air. Under equilibrium the net radiation received by these walls, i.e., bottom and right walls, balances the convection heat transfer away from the two walls and there is no net heat transfer from the right and bottom walls. Thus, in the problem with no radiation ($\epsilon = 0$) the heat loss from the system is $q_{\mathrm{conv(left)}}$ only.

Hence Eq. (1) should be modified as

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