Int. J. Heat Mass Transfer. Vol. 32, No. 11, pp. 2099–2112, 1989
Printed in Great Britain

0017–9310/89 $3.00+0.00

Analysis on film boiling heat transfer of impacting sprays†

SUGATO DEB
Libbey–Owens–Ford Co., Technical Center, 1701 E. Broadway, Toledo, OH 43605, U.S.A.

and

S.-C. YAO
Mechanical Engineering Department, Carnegie–Mellon University, Pittsburgh, PA 15213, U.S.A.

(*Received* 18 *October* 1988 *and in final form* 20 *March* 1989)

Abstract—The analysis on film boiling of impacting sprays is separated into two cases—a dilute spray (negligible heat transfer interaction among droplets) and a dense spray (significant interaction). The heat transfer of an impacting dilute spray is analyzed by dividing the mechanisms into three identified subprocesses—drop contact heat transfer, bulk air convective heat transfer and radiative heat transfer. Both horizontal and vertical sprays are modeled. The predictions are very satisfactory. For the dense spray film boiling, the asymptotic approach is successful. The low end asymptote corresponds to the dilute spray situation while the high end asymptote is adequately represented by the pool boiling situation for most spray conditions. The liquid mass flux is the most dominant parameter in both cases. Dense spray film boiling is found to show very little dependence on droplet parameters; however, dilute spray film boiling is significantly influenced by droplet parameters.

INTRODUCTION

SPRAY cooling heat transfer has been widely applied in many practical systems because of the high heat transfer rate it provides. In particular, it is popular in the metallurgical industry and in nuclear reactor safety devices for cooling of very hot materials at beyond the Leidenfrost temperature. At this condition, film boiling occurs such that the liquid is usually separated from the solid by a vapor cushion.

A typical example of film spray cooling would be the cooling of alloy strips in the continuous casting processes. Although such applications have been employed for a couple of decades, the fundamental understanding of this heat transfer process is still very limited resulting in the general inability to predict the heat transfer of impacting sprays. This has been the main reason why other less effective but more predictable cooling methods are often preferred in practice. In fact, the insufficient fundamental knowledge of the spray film-boiling process has hampered its widespread application in the industry.

The previous investigations related to film boiling spray heat transfer have been largely experimental in nature. These studies can be broadly classified in two categories—individual droplet impacting heat transfer and spray heat transfer. As a first step in the understanding, the dynamics of a single droplet impinging on a hot plate was studied by various investigators [1, 2]. The next step of complexity was the case of a single stream of droplets [3–6]. These droplet impacting experiments did help to provide some basic understanding of what processes might be involved in spray cooling. However, the direct extension of such information to sprays could not be realized. Therefore, other researchers [7–9] have investigated spray heat transfer directly. Most of these studies used commercial full cone nozzles to produce dense sprays which had wide ranges of droplet sizes, with interdependent droplet impinging velocity and liquid mass flow rate. While the reported experimental information are of great value, the results cannot be generalized because many parametric effects are not separable. Furthermore, there is a great deal of discrepancy among the published experimental data of various investigators. The very limited usefulness of these existing experimental results is mainly due to the lack of a suitable spray generator which allows independent control over the spray parameters.

Recently, a multiorifice spray generator is developed using the impulse-jet technique [10]. This device is capable of producing uniform size liquid droplets in large quantities in the form of a spray, with independent control over the spray parameters, namely the liquid mass flux, the droplet size, and the impinging velocity.

With this available spray generator, a systematic experimental study was undertaken [11, 12]. Experiments were conducted for both horizontally and vertically impinging sprays at film boiling conditions.

†The work was performed at Carnegie–Mellon University as the Ph.D. Thesis of the first author.

NOMENCLATURE

b	air jet radius
B	wall superheat parameter, defined in equation (2)
C_p	specific heat
CHF	critical heat flux
D	diameter
d	average diameter of droplet
E	surface effusivity factor, defined in equation (5)
F	amplification factor for vertical spray
G	liquid mass flux at the wall surface
Δh	latent heat
K_d	dimensionless vapor parameter at T_{sat}, defined in equation (3)
Nu	Nusselt number
Pr	Prandtl number
q	heat flux
R	experimental-to-model ratio of heat flux
Re	Reynolds number
SF	dimensionless surface material parameter, defined in equation (4)
T	temperature
U	normal free-stream velocity
V	normal droplet velocity
We	Weber number, defined in equation (1)
x	distance along the surface from the stagnation point.

Greek symbols

ϵ	heat transfer effectiveness
ε	emissivity
λ	conductivity
μ	dynamic viscosity
ν	kinematic viscosity
ρ	density
σ	Stefan–Boltzmann constant in equation (12), or surface tension of droplet in expression for K_d
τ	time characteristic.

Subscripts

a	air
b	bulk air convection
c	contact
d	droplet
hy	hydrodynamic
l, liq	liquid
max	maximum
r	radiative
s	wall surface material
sat	saturation
St	steel
sub	subcooling
sup	superheat
th	thermal
v	vapor
w	wall, i.e. surface of hot plate
∞	free stream.

Results show that under the experimental conditions used, the drop–wall contact heat transfer is the most dominant mechanism in the spray cooling process as compared with bulk air convection and thermal radiation. In dilute sprays, the contact heat transfer depends strongly on the impacting droplet dynamics, and in turn, the droplet dynamics depend on the droplet size and the droplet impinging velocity. In dense sprays, however, droplet dynamics apparently do not have a substantial effect on the contact heat transfer, possibly because of the significant interruption by other droplets during impaction. The contact heat transfer is clearly affected by the liquid mass flux alone. It is also observed that vertical sprays impinging downwards tend to have a higher heat transfer than horizontally impacting sprays in the film boiling region because of the possible existence of secondary impactions from rebounding droplets.

The dilute and dense sprays mentioned above are, however, not categorized in a clear-cut manner. Qualitatively speaking, a dilute spray is one in which the number density of droplets is small so that interaction among the droplets during impactions on the surface is not strong. On the other hand, in a dense spray, the number density is high enough that this may not be true. Therefore, the heat transfer process of a dense spray is very much different from the extrapolation of single impacting drops. Although the general characteristics of dilute and dense sprays are so different, a quantitative categorization, however, is still not available.

On the analytical modeling of spray heat transfer, hardly any work has been successful. Models developed [13, 15] have been essentially of a single sessile (stationary) droplet on a hot surface with minor modifications for an impinging droplet. These models are simplistic in nature and involve unrealistic assumptions. Consequently, they are limited in their application and, further, their predictions do not agree well with experiments. Bolle and Moureau [16] have discussed the evolution of spray cooling heat transfer, both analytically and experimentally, in their review essay.

An exploratory modeling of the heat transfer from impacting dilute sprays that has been successful to some degree is given in ref. [14]. The total process was divided into subprocesses based on the heat transfer mechanisms involved and the contribution of each to the overall heat transfer of impacting sprays was analyzed. However, the analytical results could not be completely tested with experimental evidence because in most data the detailed spray information required is unavailable. Also, the parametric effects studied by the model are mostly unavailable in the experimental data.

The present approach uses a similar overall strategy in modeling dilute sprays; however, the division into subprocesses have been restructured. The present analysis has shown some of the mechanisms in ref. [14] to be relatively unimportant while other important mechanisms have not been considered earlier. In particular, non-dimensional analysis has been carried out and surface material effects have been incorporated into contact heat transfer. Convective heat transfer has been completely reanalyzed and free-stream turbulence effects which are very dominant but were not considered earlier have been included. In addition to dilute sprays, the predictive capability has been extended to the case of dense sprays as well which requires a very different approach because of the higher complexity of the problem. The data base of comparison has been expanded significantly to include the very recent parametric spray data of refs. [11, 12] as well as the data in the literature of commercial spray experiments.

The objective of this research effort is to develop a comprehensive analytical model for the spray cooling process in general, i.e. given the characteristics of the impacting spray and the properties of the surface being cooled, the model should be able to predict the profile of heat flux vs surface superheat in the film boiling region.

Since no earlier attempt at analyzing film boiling spray heat transfer has been very successful, the majority of the present research effort has been concentrated on the establishment of the mechanisms controlling the models. As a first attempt, the model is limited to dilute sprays impacting normally on a surface beyond the Leidenfrost temperature. First, the overall heat transfer process has been broken down into its basic component processes. The next stage involves identifying the important heat transfer mechanisms comprising each component process. Finally, each of the mechanisms is modeled.

Dense sprays are characterized by interaction among the impinging droplets on the surface. It is extremely difficult to quantify this interaction. The approach taken here is to analyze and predict the limit or asymptotic conditions of a dense spray. Then the intermediate region could be predicted as a combination of the two asymptotic extremes.

DILUTE SPRAY ANALYSIS

The heat transfer process of spray cooling on surfaces beyond the Leidenfrost temperature can be generally divided into five basic identifiable mechanisms as shown pictorially in Fig. 1. Impact heat transfer occurs when the droplet is separated from the surface by a thin film of vapor (or it may come in contact with the hot surface momentarily if its momentum is high enough), and last for as long as the drop remains adjacent to the surface (at most, of the order of milliseconds) and is shown in Fig. 1(a1). Impact heat transfer is essentially characterized by conduction heat transfer between the drop and the surface. Further, whenever the drop is in the near vicinity of the wall, it starts vaporizing. The flow of vapor on the surface provides local convective cooling shown in Fig. 1(a2). In addition, every drop locally entrains some air behind it due to its relative velocity and the resultant 'local' air circulation also provides local convection heat transfer as seen in Fig. 1(a3). Kendall and Rohsenow [3] claimed that the last effect, that of local air circulation, is of the same order of magnitude as the previous two effects combined, namely impact heat transfer and vapor convection effect. Since all the above-mentioned effects are due to the presence of the droplets alone and are very intimately related, they have been grouped together as contact heat transfer in the present modeling as represented in Fig. 1(a). Besides, all experimental works have these effects lumped together in their data. In addition to the droplets themselves, 'bulk' air may sometimes be supplied to the spray by external means or be simply entrained from the environment by the fast moving spray. The convective cooling due to the impingement of the bulk flow of air is then considered separately, shown in Fig. 1(b). Finally, radiative heat transfer from the heated surface to the environment should also be considered as in Fig. 1(c).

The complete model is constructed by adding the representations for the three subprocesses though, in reality, the combining process may be more complicated than a simple addition. The present additive is acceptable as the subprocesses can be considered decoupled for a dilute spray.

Subprocess analysis

As a first attempt, the model would be for an ideal, normally impacting spray, i.e. a spray with all droplets of the same diameter and impacting normally on the surface. This would also enable comparison with the well controlled experimental work in refs. [11, 12] which are mainly for dilute, normally impacting sprays. These are the only reported results in the literature for experiments conducted with monodispersed sprays. For practical sprays having a spectrum of droplet sizes and velocities, representative values may be used for the model prediction. Each heat transfer subprocess of a dilute, normally impacting spray is described in the following sections.

Contact heat transfer. During droplet impaction, the droplet–wall contact time period ranges in the order of microseconds to milliseconds. This includes, as mentioned earlier, the droplet–wall impact heat transfer and the local convective effects of the vapor and the droplet-entrained air. Because of the complexity of the problem, attempts by the present researchers as well as by other previous investigators

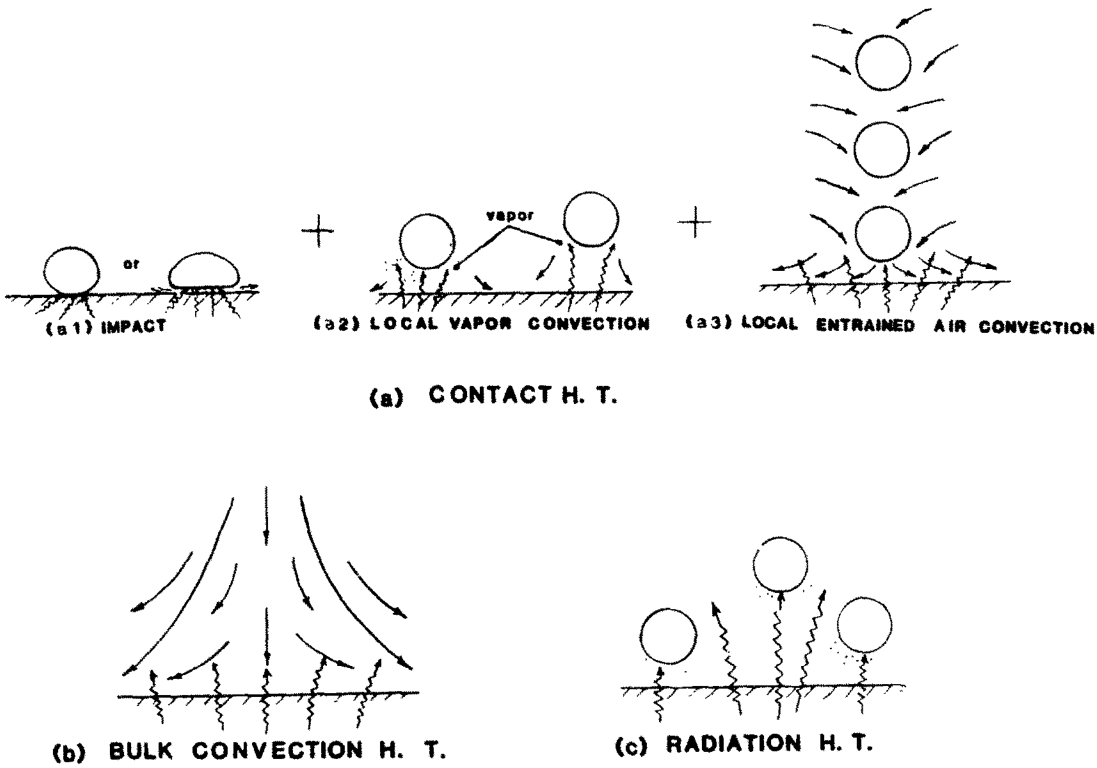

FIG. 1. Components of spray heat transfer.

to model the detailed heat transfer process of impacting droplets onto a hot surface have not been very successful. Hence, a semi-empirical correlation is attempted instead, based on the experimental data in the literature for the heat transfer to a single stream of droplets impinging on a hot surface. The result is extended to the case of a dilute spray for which it is still valid as there is negligible interaction among the droplets in such a spray.

Dimensional analysis is applied. The resulting non-dimensional parameters are:

the Weber number,

$$We = \rho_1 d V^2/\sigma \tag{1}$$

the wall superheat parameter,

$$B = C_{p,\mathrm{v}}(T_\mathrm{w}-T_\mathrm{sat})/\Delta h_\mathrm{v} \tag{2}$$

and the dimensionless vapor parameter,

$$K_\mathrm{d} = \lambda_\mathrm{v}/(C_{p,\mathrm{v}}\mu_\mathrm{v}). \tag{3}$$

The surface material effect is brought in through the surface factor, SF

$$SF = (E_\mathrm{s}/E_\mathrm{St}-1) \tag{4}$$

which is proportional to the thermal effusivity [16], E, as defined below

$$E_\mathrm{s} = \sqrt{(\lambda_\mathrm{s}\rho_\mathrm{s}C_{p,\mathrm{s}})}. \tag{5}$$

Usually, the heat transfer rate is indicated by droplet heat transfer effectiveness, ϵ. The heat transfer effectiveness is defined as the ratio of actual heat transferred from the hot surface during the impaction of one drop, to the total heat transfer which is required for complete evaporation of the droplet. Physically, it reflects the efficiency of use of the droplet. It is given by [5]

$$\epsilon = \frac{q_\mathrm{c}}{G[\Delta h_\mathrm{v}+C_{p,1}(T_\mathrm{sat}-T_\mathrm{liq})]}. \tag{6}$$

Liquid–solid impact heat transfer and local vapor convection effects are expected to be linearly proportional to the liquid mass flux. It is assumed that the convective effect of the locally entrained air also grows linearly with the mass flux since the spray is dilute. Using the data of Kendall and Rohsenow [3] ($d \sim 260$ μm, $V = 1$–2 m s^{-1}) and Pederson [4] ($d = 200$–400 μm, $V = 2$–10 m s^{-1}) for a single stream of droplets impinging on hot target surfaces of approximately 6.5 mm diameter in both cases, the following non-dimensional expression has been statistically arrived at:

$$\epsilon = 0.027\exp[0.08\sqrt{(\ln(We/35+1))}/(B+SF/60.5)^{1.5}] + 0.21K_d B\exp[-90/(We+1)]. \tag{7}$$

The above equation for the effectiveness has been obtained to fit the data for the film boiling region mainly, and to some extent the adjacent region of

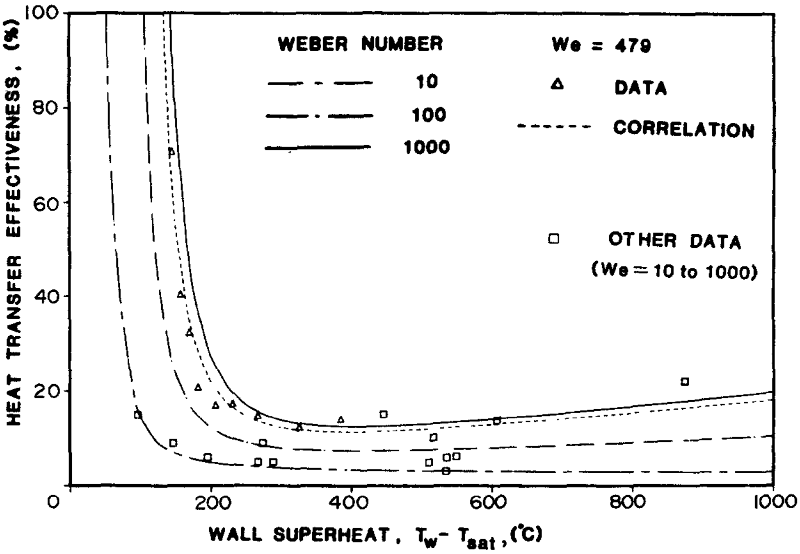

FIG. 2. Effectiveness variation with superheat and Weber number.

transition boiling and the Leidenfrost condition. All liquid and vapor properties are evaluated at T_{sat}. The first term in equation (7) represents the liquid–solid impact heat transfer which decays with the increase in the wall superheat. The second term is the local convective effects of the vapor flow and entrained air that increases with the wall superheat. (This variation is graphically shown in the lower part of Fig. 4). Figure 2 shows the correlation for the effectiveness at a representative Weber number of 479. On average, the error is around 15%, with the best fit to experimental data in the medium Weber number region (50–150). Figure 2 also shows the trend of the variation of effectiveness with the Weber number and the wall superheat. Effectiveness increases with *We* at all temperatures as both terms in equation (7) do the same. However, it tends to saturate at higher *We*.

Bulk-air convective heat transfer. This term represents the heat transfer to the bulk flow of air in the spray as shown earlier in Fig. 1(b). It is basically the convective heat transfer that would result if one was able to shut off the liquid content in the spray and let the rest of the air in the spray flow onto the surface. The heat transfer solution for such a case, i.e. a laminar jet impacting normally on a surface, can be obtained as a special case of the closed form solution for wedge flows using similarity transformations [17]. However, experimental observation [11] of this subprocess showed that for pure air flow normal to the heated surface, like that in a spray but without the droplets, the heat transfer was significantly higher than the theoretical prediction. After detailed investigation by the present authors, the discrepancy was traced to the fact [18] that surface heat transfer is enhanced significantly, especially at and around the stagnation point, by the presence of free-stream turbulence. Free-stream turbulence needs to be distinguished from the conventional high Reynolds number turbulence. Free-stream turbulence is due to large-scale eddies which are present in such laminar flows. They penetrate the boundary layer and lead to significant surface heat transfer enhancement.

In order to incorporate the enhancement of free-stream turbulence into the model in a simple manner, an enhancement curve shown in Fig. 3 has been constructed based on the experimental data of McCormick *et al.* [18] which is for a rectangular plate of 4 : 1 aspect ratio, inclined to a parallel flow at a 40° angle of attack. It shows that for a flow that is laminar according to the Reynolds number, low levels of free-stream turbulence (intensity $\sim 10\%$) tend to enhance the surface heat transfer by about 50% over the Smith–Spalding laminar solution [17], equation (8). At higher free-stream turbulence levels, one would expect the enhancement to increase, but using the laminar solution as the base shows a wide scatter of data. On the other hand, using the Ambrok turbulent solution [17], equation (9), as the base for the averaged enhancement over the surface at high free-stream turbulence shows good correlation among the data such that for every 1% increase in free-stream turbulence, the enhancement increases by approximately 5%. The higher free-stream turbulence somehow creates effects similar to that of the turbulent boundary layer, possibly due to better mixing within the boundary layer, though the bulk flow Reynolds number is still in the laminar range. Heat transfer in the stagnation point region gets enhanced much more than the average over the rest of the surface. As shown in Fig. 3 the enhancement factor around the stagnation point is more than double the averaged enhancement over the surfaces in the experiment of ref. [18]. Since the available data are very limited, the understanding in this subject area is still not complete.

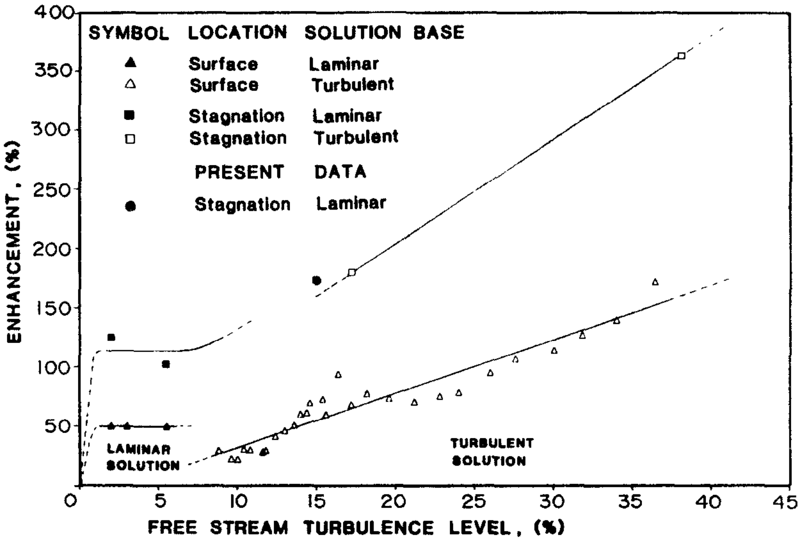

FIG. 3. Enhancement of convective heat transfer by free-stream turbulence. (Enhancement = Experimental/Theoretical − 1). Data from ref. [18].

For the case of a normally impacting air jet on a flat surface, the local laminar solution [17] is given by

$$Nu_L = 0.76 Re_x^{0.5} Pr^{0.5} \tag{8}$$

while the local turbulent solution [17] is given by

$$Nu_T = 0.033 Re_x^{0.8} Pr^{0.6}. \tag{9}$$

If the radius of the jet is b and the free-stream velocity is U_∞, then

$$U_x = \frac{U_\infty \pi x}{4b} \tag{10}$$

and

$$Re_x = \frac{U_x x}{\nu}. \tag{11}$$

The above equations give the local value of the parameters as a function of x, the distance from the stagnation point. The averaged parameters, if desired, can then be calculated using a characteristic length based upon the geometry of the problem such as the dimension of the hot surface.

In order to use Fig. 3 in a predictive manner, the free-stream turbulence present in the flow needs to be known. It can be easily measured using a hot-wire anemometer. The free-stream turbulence level in the flow at the experimental conditions of ref. [11] was measured to be about 15%. This places it in the ambiguous region between the laminar and turbulent solution base of Fig. 3. However, from the heat transfer data measured at the stagnation point, the enhancement appears to be about 175% with the theoretical laminar solution as the base. Plotting this value in Fig. 3 shows that it is consistent with the other laminar based stagnation data. It is noticed that using the Ambrok turbulent solution as the base does not work exactly at the stagnation point since the calculated heat transfer coefficient is zero. Instead, it should be averaged over the stagnation region, i.e. over a circular area around the stagnation point of the jet radius.

In order to investigate the potential interaction of droplets with the free-stream turbulence, a solid particle spray heat transfer experiment was carried out [11]. The solid particles used were soda-lime glass beads of roughly the same diameter as the liquid droplets and in the same volume flux. Since the solid particles will impact the heated target with small contact areas and for very short duration, the contact heat transfer would be insignificant. Therefore, the difference between the overall heat transfer with and without solid particles would give the effect of the presence of droplets on bulk convective heat transfer.

The results of the solid particle spray experiments [11] indicate that the addition of solid particles in the air flow increases the convective heat transfer only slightly ($<10\%$) for a particle size in the range of 0.175–0.42 mm. This increase is almost negligible compared to the large enhancements due to free-stream turbulence effects. Hence, for the sake of simplicity, this particle loading effect has not been incorporated into the present model.

Another concern is that liquid droplets present in the thermal boundary layer tend to create an effect of a distributed heat sink [14]. The conventional idea of heat sinks in flows which are free from free-stream turbulence, is as follows. At the instant of droplet impaction, rapid vaporization of the droplet occurs. The generated vapor appears very close to the wall and may stay in the thermal boundary layer. Since the generated vapor, which is at the saturation temperature, is much lower in temperature than the air adjacent to the hot wall, it will act as a heat sink in the thermal boundary layer and, therefore, enhance the heat transfer.

In the present situation, though the heat sink effect may exist, it is likely to be less significant because of the following reason. The effect of the free-stream turbulence in an impacting spray is to promote mixing due to the large-scale pulsations penetrating the boundary layer [19]. These and also the particle induced pulsations tend to wash away the vapor from the hot wall and, therefore, dilute the previously mentioned heat sink effect. Hence, it was considered acceptable to drop the heat sink effect from the model. As will be shown later on, the model predictions agree fairly well with experimental data which further backs up the expectation that the heat sink effects of droplets is not significant.

Radiative heat transfer. One can model the high temperature wall as an infinite gray plate and the opposite zone of water droplets as another infinite plate. Air is almost transparent to radiation. The radiative heat flux between the wall and the droplet zone can then be easily established. Since a large amount of tiny droplets are randomly distributed in the environment and the thickness of the droplet zone is relatively large, the amount of radiative heat returning to the wall is negligible. Further, since the emissivity of deep water is 0.96 [20], the total emissivity of the droplet zone could be simply approximated as unity, i.e. the droplet zone can be treated as a black sink. The radiative heat flux is hence simply given by the Stefan–Boltzmann law

$$q_r = \sigma\varepsilon_s(T_w^4 - T_\infty^4). \tag{12}$$

Overall predictive strategy

The strategy for predicting the overall surface heat flux as a function of the surface temperature for the case of a dilute spray is outlined below in a concise manner.

(1) Contact heat transfer. From the given conditions of the particular application, i.e. the spray and surface conditions along with the relevant liquid and solid material properties, calculate the non-dimensional numbers using equations (1)–(5). Then use the correlation developed, equation (7), to calculate the effectiveness, ϵ. Finally, use the definition of effectiveness in equation (6) to calculate the predicted contact heat transfer contribution q_c. For the case of a vertical spray, use Fig. 6 to amplify the contact heat transfer as calculated above in order to account for the multiple contact contribution.

(2) Bulk air convective heat transfer. The following represents the procedure for a normally impacting air stream. Knowing the free-stream turbulence level in the bulk air in the spray, use Fig. 3 to decide on the theoretical solution needed—the laminar solution (equation (8)), or the Ambrok turbulent solution (equation (9)), and calculate the heat flux. Use Fig. 3 also to obtain the enhancement of the supplied free-stream turbulence level. Select the stagnation curve or the surface curve for the impacting air jet depending on which better represents the given condition. Finally, use this enhancement value along with the definition of enhancement to calculate the predicted bulk convective heat transfer, q_b.

(3) Radiative heat transfer. Use equation (12) with $T_\infty = T_{sat}$ to calculate q_r. The value of surface emissivity is needed for this calculation.

The overall heat transfer from the surface or wall at a selected T_w used in the calculations is given by $q_w = q_c + q_b + q_r$. One can generate the spray cooling curve, i.e. a plot of q_w vs $(T_w - T_{sat})$ by carrying out calculations for different values of T_w.

Results and discussions

The qualitative comparisons of each heat transfer mechanism are revealed in Fig. 4 for a typical case wherein the contribution of each subprocess to the total spray heat transfer can be observed. No previous work has been able to provide this kind of relative information. Contact heat transfer has a significant contribution throughout, especially at lower temperatures. Bulk air convective heat transfer grows with increasing temperature. Its magnitude depends directly on the flux of air in the spray and may become dominant at sprays of low liquid mass flux when the contact heat transfer contribution drops. Radiative heat transfer has a negligible contribution over the temperature range considered. This is partly because the present heated surface is chrome plated and the surface emissivity is low.

For a long time, there has been contradictory arguments on the dominant heat transfer mechanism in a spray being the liquid impaction or the air convection. Based upon the present modeling in terms of the three mechanisms, it appears that different mechanisms may become dominant at different working conditions of sprays. When the liquid flux is high, that is for a dense spray, the liquid impaction is definitely the controlling mechanism in heat transfer. For a dilute spray where liquid flux is low, the liquid droplet impaction heat transfer is the key heat transfer mechanism in film boiling when the surface temperature is low. At high surface temperatures, the air convection becomes dominant in heat transfer because the convection is proportional to the temperature difference while droplet contact heat transfer reduces at high surface temperatures.

Next, the quantitative information is examined. The present model is restricted to the situation of single contact between any drop and the wall during impaction. This is true, in general, for horizontal sprays impinging on a vertical surface. Comparison with the horizontal spray experimental results of ref. [11] will be most meaningful as their parametric effects are well separated. The surface used in this experiment was copper with a chrome finish ($\varepsilon_s = 0.15$). Figure 5 provides a comparison between the model prediction and

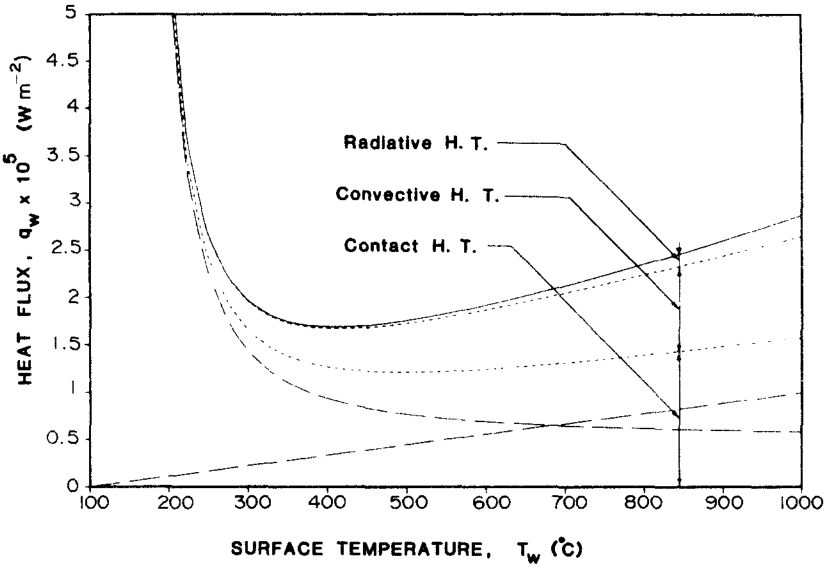

FIG. 4. Decomposition of overall heat transfer into components.

the experimental results at different mass fluxes with all other spray parameters held constant. The particular interest is at the film boiling region. The experimental results indicate that the heat flux tends to increase with the liquid mass flux when the mass flux is low, but does not increase proportionally at higher mass fluxes. The comparison between the data and the model shows that the model predicts reasonably well for the film boiling region, but it is not as good in the transitional boiling region. This may not be too much of a disadvantage since most practical applications are designed for the film boiling region rather than the transitional boiling regime. This is because the film boiling region, characterized by its low heat transfer rate, is usually the controlling factor in the cool down process in spray applications. At the highest liquid mass flux, the model tends to overpredict. This will be explained later. The drastic dip in heat transfer around the Leidenfrost temperature for higher liquid mass fluxes, as experimentally observed,

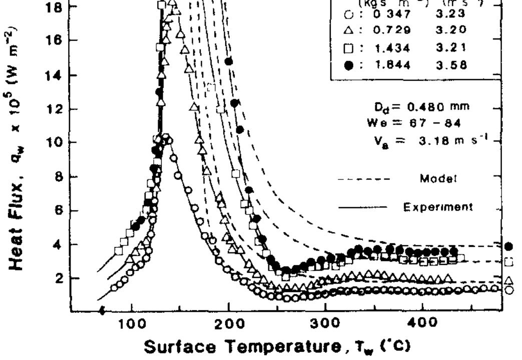

FIG. 5. Overall heat transfer profiles at different mass fluxes: model prediction vs experimental results [11].

has not been successfully explained by the original researchers and is still under further investigation.

For the case of a vertical spray flowing downwards, multiple contacts between drops and the surface may occur due to the repeated impaction of fragmented droplets as also observed by Bolle and Moureau [16] and many others. Multiple impactions of a drop increase the contact heat transfer but have negligible effects on the other two subprocesses. It is difficult to model this enhancement exactly because of the randomness and complexity of the repeated contacts. Instead, the extension of the present model to vertically downward impinging sprays has been accomplished by using a factor F to amplify the contact heat transfer. The F factor has been deduced from the experimental data of refs. [11, 12] and is dependent on the mass flux. At low liquid mass fluxes, the secondary contacts due to fragmented droplets contribute significantly to the overall contact heat transfer as the number density of the primary contacts are relatively small. As the mass flux increases, this contribution becomes less significant, and finally a saturation effect is observed. A simple form of F is obtained as shown in Fig. 6. Thus, vertical sprays can be modeled as well.

A point to be noted here is that there are no tuning parameters in this model but there may exist a limit of application. The model works well for a dilute spray when compared with all the experimental data of ref. [11] which are for an ideal monodispersed spray. To further check the model, the predictions are also compared with previously published experimental results [7–9, 21], which are mostly at high liquid flux (dense spray) and high Weber number conditions utilizing practical spray nozzles. The surface material is either copper or stainless steel and the liquid used is water. In most cases, the spray parameters are not accurately reported, except for the liquid mass flux. Therefore,

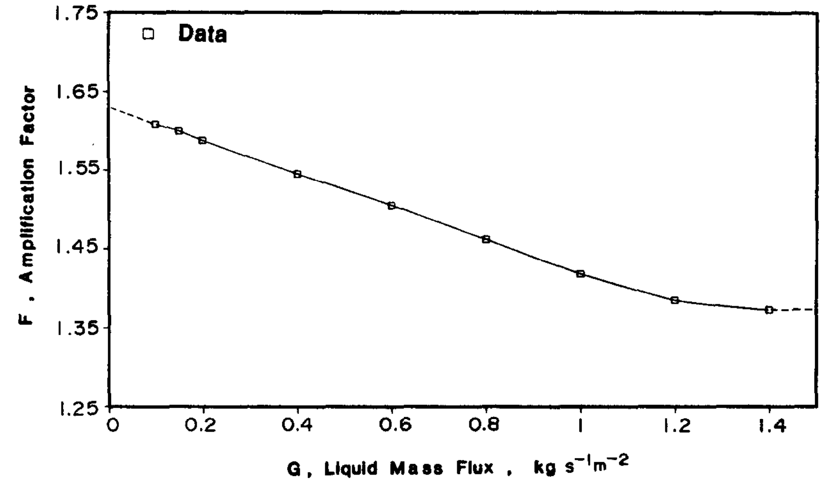

Fig. 6. Vertical contact heat transfer amplification factor [11].

some best estimated parameters have been used by the present authors in this comparison. All the results are presented in Fig. 7 which is a plot of the ratio of the experimental data to the model prediction of the heat flux in the film boiling region, against the liquid mass flux which has the strongest influence on the heat transfer. This plot shows a clear trend that the model prediction breaks down beyond 2 kg m^2 s^{-1} for water on a copper or stainless steel surface.

It is apparent that the prediction indicates an ever increasing heat transfer due to the increase of mass flux; however, the experimental data shows a tendency of saturation. As the mass flux increases, i.e. as the spray becomes dense, droplet-to-droplet interaction in the heat transfer processes can be significant. The heat transfer of each droplet in a dense spray will be less effective than that of a single droplet impacting the surface alone which is typical for the dilute spray situation. The consequence of this is that the present model of a dilute spray overpredicts at high mass flux as shown in Fig. 7 since it is based on the assumption that droplet interaction is negligible. This also accounts for the overprediction in Fig. 5 for the case of the highest mass flux. Consequently, the identified water mass flux of 2 kg m^{-2} s^{-1} can provide a comprehensive guideline for the categorization of sprays as dilute or dense at the present moment. Since water is the liquid used almost universally for film boiling

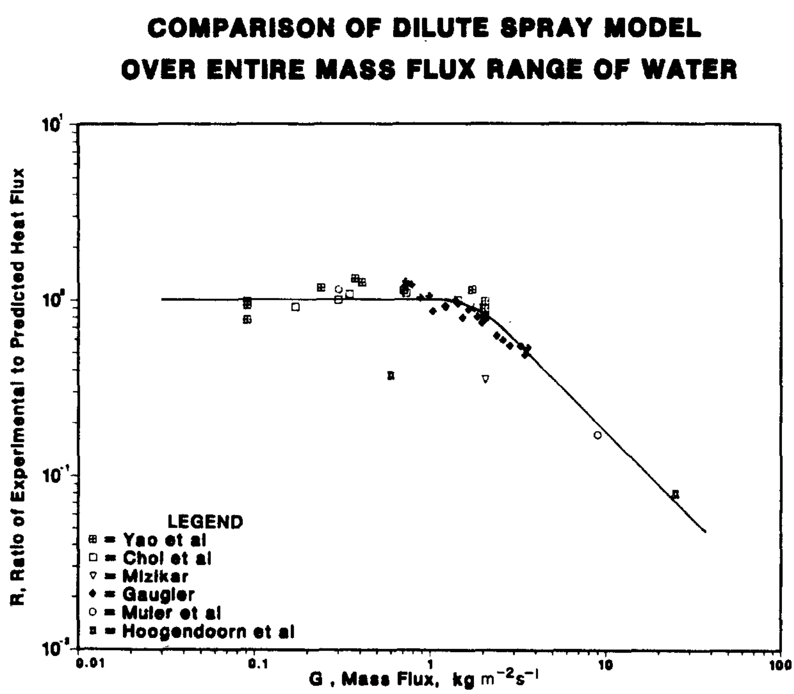

Fig. 7. Comprehensive plot of experimental to model heat flux ratio vs the liquid mass flux.

heat transfer, and most metallic surfaces can be expected to behave very similarly, this suggested value can be fairly general in its applicability.

DENSE SPRAY ANALYSIS

As mentioned above, in a dense spray droplet interactions effectively decrease the heat transfer to each drop. The reason for the reduction in heat transfer can be explained qualitatively as follows. When a spherical drop impinges on a surface, it starts flattening out as a disk first because of its incoming kinetic energy. After it reaches a maximum diameter it starts contracting and may bounce off or shatter depending on how dominant the surface tension forces are. A drop on contact, therefore, has an influence zone around it hydrodynamically. Besides, the impaction of the drop causes a local drop in temperature in the solid material. A thermal wavefront will travel into the material and then retract as the drop moves away. Thus there is also a thermal influence zone extending radially away from the drop into the material. Each of these influence zones, hydrodynamic and thermal, are characterized by a maximum diameter of influence on the surface and a characteristic time of the phenomenon.

For a dilute spray, the drops are far apart so that their influence zones, both hydrodynamic and thermal, do not interact with that of another drop both in space and time. As the number flux of drops increases, these influence zones are no longer independent and start influencing each other. This affects both the spreading process of the drop on the surface which results in a smaller surface area to extract the heat from, as well as the surface temperature seen by the drop which may not have recovered completely from the previous impaction. Both these effects work to reduce the heat transfer and hence we do not see a proportional increase with the mass flux once the demarcating mass flux between dilute and dense sprays is exceeded. (It is noticed that the factor which distinguishes the dilute and dense sprays is the liquid flux but not the ratio of liquid to gas volume fluxes.) Beyond this condition, this interaction increases monotonically with the mass flux, thereby reducing the heat transfer increase. As the mass flux increases further, eventually a flooded condition of the surface results, especially for vertical sprays impacting on horizontal surfaces. Further increase in the mass flux would not change the heat flux much as the additional mass flux would be impacting on the water layer instead of on the hot surface. This accounts for the final saturation effect observed in the laboratory as well as in practice in metallurgical cooling processes for very high mass fluxes as a pool of liquid then forms on the surface.

Predictive approach

Photographs have been taken to try to quantify the number of visible impactions at typical values of the mass flux to compare it with simple theoretical analysis that was carried out. The results were fairly satisfactory. However, the follow up calculations to characterize the interaction among the droplets based on probability analysis along with the influence zone information was not very successful. This is partially due to the uncertainty of some parameters used in analysis; however, it is also likely to indicate that the interaction may have a much higher level of complexity. Details of calculations are available in ref. [26].

The approach taken for dense sprays, then, is to try to analyze and predict the limiting or asymptotic conditions of a dense spray. Then the intermediate dense region could be evaluated as a combination of the two extreme conditions. The low mass flux asymptotic conditions correspond to the case of a dilute spray which has been modeled satisfactorily in the previous section. At the high mass flux asymptotic condition the large amounts of liquid spray would cause flooding and result in the formation of a shallow pool on the surface. Hence, pool boiling heat transfer may best represent the situation. This can be generally true for most spray conditions. However, if the spray is concentrated and has a large impacting velocity, then the droplet momentum could be large enough to penetrate the pool and reach the surface. A similar situation occurs when a liquid jet is used. Such situations can be represented by forced convection boiling instead of pool boiling. However, the present study will be limited to conventional spray conditions which lead to pool film boiling. Figure 8 shows a schematic of the postulated breakup process.

High end asymptote. Pool film boiling of water has been studied by various investigators [22–24], mostly using thin wires or tubes as the heated surface. The heat transfer is dependent on the liquid subcooling and on the surface superheat. The approach taken here is to decouple the two effects as a first-order approximation. As is shown later, it seems to hold rather well. Of the two, pool film boiling is very strongly affected by the subcooling of the liquid in the pool, and to a much lesser extent on the surface superheat. There is very limited data available for the subcooling effect, none for a flat plate configuration. Consequently, the best available data is used. Figure 9 shows the pool heat flux variation as a function of the subcooling for water at 555°C wall superheat. The curve has been constructed from the data of Bradfield [23]. The test specimen was a solid sphere of pure copper 2.35 in. in diameter. The heat transfer data shows a very steep slope at high subcooling and tends to flatten out at low subcooling.

The dependence of pool heat flux on the surface superheat is shown in Fig. 10 for saturated water on a horizontal plate [22]. The heat flux increases with the surface superheat but the increase is very mild compared to the almost one order of magnitude variation of the heat flux with water subcooling. This data extrapolated for the surface superheat of 555°C seems

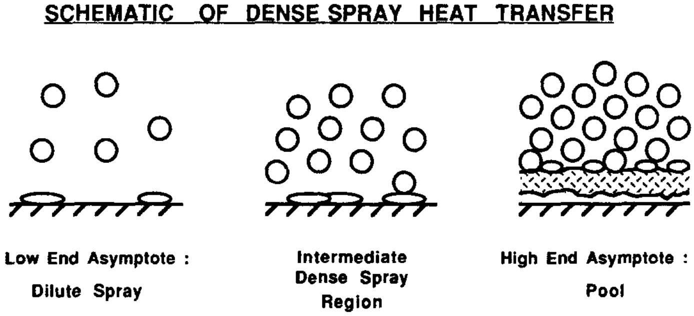

FIG. 8. Schematic of the breakup of the dense spray region.

reasonably consistent with the 0°C subcooling heat flux value in Fig. 9, indicating that the data for the two types of heating specimen used are fairly compatible.

In order to obtain the pool heat flux for a given subcooling and surface superheat, Fig. 9 is first used to get the predicted heat flux at a superheat of 555°C. This can then be scaled down to the specified superheat by using the slope of the curve in Fig. 10 which can be written as follows:

$$q_{\mathrm{pool},T} = q_{\mathrm{pool},555^{\circ}\mathrm{C}} 10^{-0.0026(555-\Delta T_{\mathrm{sup}})}. \quad (13)$$

It should be noticed that the pool temperature is supposed to be higher than the sprayed liquid temperature due to the finite mass of the pool. However, when the liquid mass flux is high, this temperature difference will diminish because the pool is replenished quickly. In the present modeling of dense spray, due to the lack of well reported data, the sprayed liquid temperature is used for the pool temperature.

A comparison of the predicted pool heat flux with the experimental spray heat flux data at very high mass flux conditions shows good agreement between the two. This can be checked in the comprehensive plot of Fig. 11. Thus, the high end asymptotic condition can be handled quite satisfactorily.

Intermediate region. The intermediate dense region has characteristics of both the low and the high end asymptotic conditions. In order to predict the heat transfer for a given spray condition in this region, a suitable combination of the asymptotic predictions needs to be developed. Churchill and Usagi [25] have

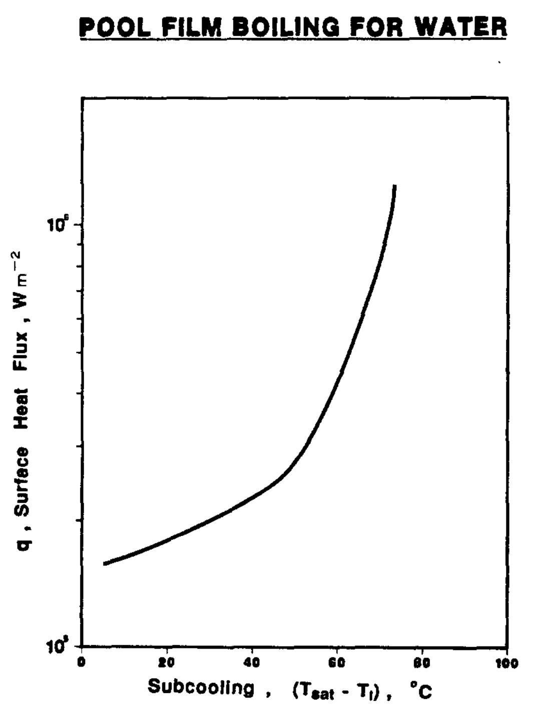

FIG. 9. Effect of water subcooling on pool film boiling heat flux at 555°C superheat [23].

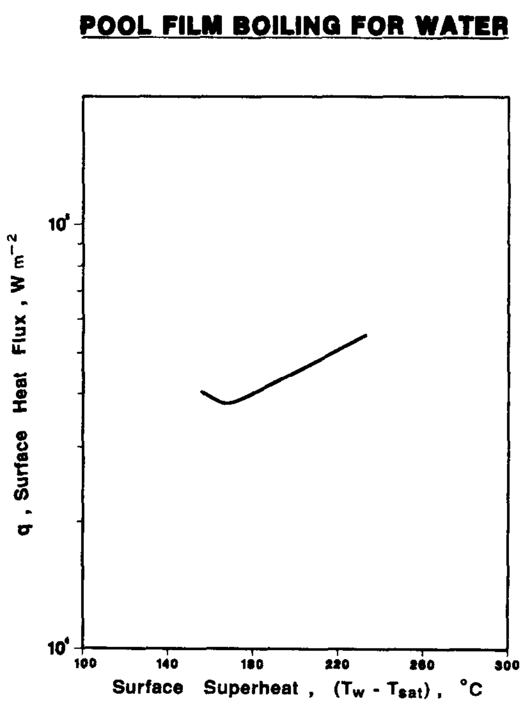

FIG. 10. Effect of surface superheat on saturated pool film boiling heat flux [22].

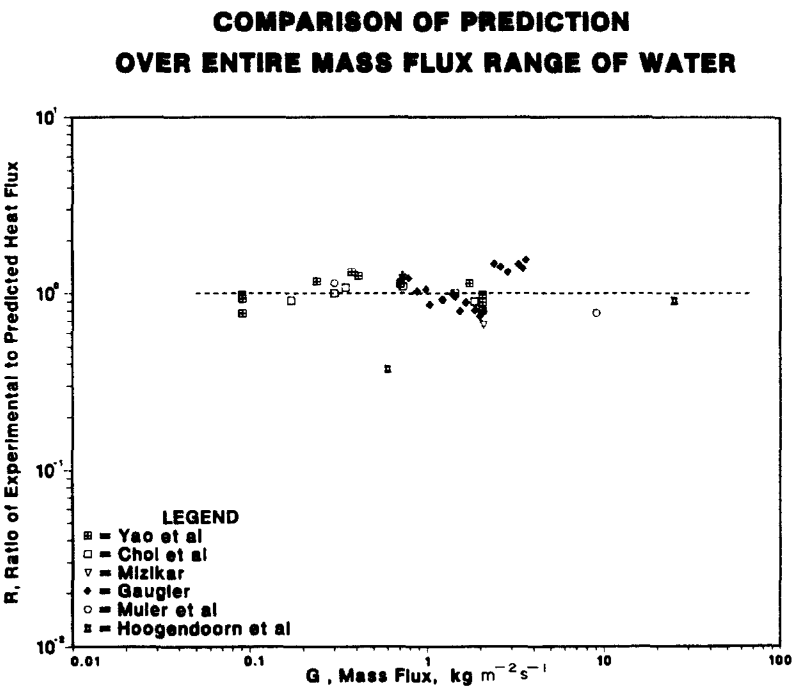

FIG. 11. Comprehensive plot of experimental to predicted heat flux ratio vs the liquid mass flux.

proposed a general form of an empirical equation for the correlation based on limiting (asymptotic) solutions. Using their general form, a correlation for the entire dense region can be established as

$$q_{\text{dense}}^{-0.86} = q_{\text{dilute model}}^{-0.86} + q_{\text{pool}}^{-0.86}. \tag{14}$$

This equation gives the true formulation at the extreme conditions of dilute and very dense sprays. It provides a smoothed correlation at an intermediate range of the spray flux.

OVERALL PREDICTIONS

Dense spray results do not show very much dependence on the droplet diameter and velocity as opposed to dilute spray results which indicate significant effects. This is possibly because the strong droplet interaction effects suppress the droplet dynamics in a dense spray. In the limit of reaching the pool boiling condition, the resulting heat transfer is completely independent of droplet characteristics.

A comprehensive plot of the prediction for the entire data base of both dilute and dense sprays is presented in Fig. 11. (It is similar to the earlier plot of Fig. 7 which compared the dilute spray model prediction only.) As can be seen, the prediction is satisfactory in the entire dense spray region, especially at high mass fluxes. The average error associated with the predictive equation (14) is 29% in the dense spray region. The largest errors occur in the intermediate dense region. Possible causes are limitation of the fitting scheme used for this region as well as the fairly large scatter present in the experimental data base itself.

Subcooled pool film boiling heat transfer seems to be the maximum heat flux obtainable for conventional impacting sprays. Impacting liquid jets and concentrated, high velocity impacting sprays are not constrained by the pool boiling limit and may reach higher heat fluxes corresponding more closely to forced convection boiling.

The present correlations are developed for water spray on metallic surfaces. It is expected that similar approaches will be valid for other fluids, and specific correlations may be developed from the data base of its liquid–surface combination.

CONCLUSIONS

The conclusions for the dilute spray modeling are as follows.

(1) Heat transfer of impacting dilute spray on surfaces beyond the Leidenfrost temperature can be divided into three subprocesses—contact, bulk air convective and radiative heat transfer. Contact heat transfer is handled by a semiempirical correlation. Bulk air convective heat transfer may be enhanced by the free-stream turbulence effect. The radiative heat transfer prediction can be simplified due to negligible radiation reflection.

(2) Contact heat transfer appears to be the dominant mechanism especially in the lower temperature range, while convective heat transfer becomes significant at higher temperatures. Radiative heat trans-

fer is usually negligible but can become important if surface emissivity is high. Figure 4 shows the overall heat transfer trend with temperature and the individual contributions for a typical situation of a dilute impacting spray.

(3) The model prediction is restricted to dilute sprays and is very satisfactory for the film boiling region. Vertical sprays, involving possible multiple droplet–wall contacts, enhance contact heat transfer and can be correlated by use of the factor F as shown in Fig. 6.

(4) A liquid mass flux of approximately 2 kg m^{-2} s^{-1} can be regarded as a boundary to categorize sprays as dilute or dense, for the case of water on a copper or stainless steel surface. The present model of a dilute spray overpredicts the heat transfer of a dense spray due to the existence of significant interaction among the heat transfer processes of individual droplets in a dense spray. This can be seen in Fig. 7.

The conclusions for the dense spray region are detailed below.

(1) Dense sprays do not show much dependence on droplet size and velocity, especially at high mass fluxes.

(2) Dense spray heat transfer has been handled in terms of the asymptotic conditions with respect to the mass flux. The high mass flux asymptote can be adequately represented by the pool boiling condition and can be predicted from equation (14).

(3) Pool boiling heat flux is strongly influenced by the liquid subcooling and to a lesser degree by the surface superheat. The maximum heat transfer obtainable using a conventional impacting spray appears to be that of the corresponding subcooled pool boiling condition.

(4) Figure 11 shows the predictive capability across the entire mass flux range. The prediction is generally good with further study recommended for the range of intermediate mass flux.

Acknowledgement—This research is supported by the National Science Foundation under Grant CBT-8420701.

REFERENCES

1. M. H. Shi and J. C. Chen, Behavior of a liquid droplet impinging on a solid surface, 83-WA/HT-104, ASME (1983).
2. V. Groendas and R. Mesler, Measurement of transient surface temperature beneath Leidenfrost water drops, Int. Heat Transfer Conf., Munich, pp. 131–136 (1982).
3. G. E. Kendall and W. M. Rohsenow, Heat transfer to impacting drops and post critical heat flux dispersed flow. Heat Transfer Laboratory 85694-100, Massachusetts Institute of Technology, March (1978).
4. C. O. Pederson, The dynamics and heat transfer characteristics of water droplets impinging upon a heated surface, Ph.D. Thesis, Carnegie–Mellon University, October (1967).
5. C. O. Pederson, An experimental study of the dynamic behavior and heat transfer characteristics of water droplets impinging upon a heated surface, *Int. J. Heat Mass Transfer* **13**, 369–381 (1970).
6. K. Takeuchi, J. Senda and K. Yamada, Heat transfer characteristics and the breakup behavior of small droplets impacting upon a hot surface, *ASME–JSME Thermal Engng Joint Conf.* (1983).
7. R. E. Gaugler, Experimental investigation of spray cooling of high temperature surfaces, Ph.D. Thesis, Carnegie Institute of Technology (1966).
8. I. Mizikar, Spray cooling investigation for continuous casting of billets and blooms, *Iron Steel Engng* 53–70 (June 1970).
9. H. Muller and R. Jescher, Untersuchung des Wärmeubergangs an einer simulierten sekundarkuhlzone beim Stranggiessverfahrer, *Arch. EisenhüttWes.* **44**(8), 589–594 (1973).
10. N. Ashgrizzadeh and S. C. Yao, Development of multiorifice impulsed spray generator for heterogeneous combustion experiments, *ASME/JSME Thermal Engng Joint Conf. Proc.*, Vol. 2, pp. 429–433 (1983).
11. K. J. Choi and S. C. Yao, Mechanisms of film boiling heat transfer of normally impacting spray, *Int. J. Heat Mass Transfer* **30**, 311–318 (1987).
12. S. C. Yao and K. J. Choi, Heat transfer experiments of mono-dispersed vertically impacting sprays, *Int. J. Multiphase Flow* **13**, 639–648 (1987).
13. L. H. J. Wachters and N. A. Westerling, The heat transfer from a hot wall to impinging water drops in a spheroidal state, *Chem. Engng Sci.* **21**, 1047–1056 (1966).
14. L. Liu and S. C. Yao, Heat transfer analysis of droplet flow impinging on a hot surface. In *Heat Transfer, 1982*, pp. 161–166. Hemisphere, Washington, DC (1983).
15. B. S. Gottfried, C. J. Lee and K. J. Bell, The Leidenfrost phenomenon: film boiling of liquid droplets on a flat plate, *Int. J. Heat Mass Transfer* **9**, 1167–1187 (1966).
16. L. Bolle and J. C. Moureau, Spray cooling of hot surfaces. In *Multiphase Science and Technology*, pp. 1–97. Hemisphere, Washington, DC (1982).
17. W. M. Kays and M. E. Crawford, *Convective Heat and Mass Transfer* (2nd Edn). McGraw-Hill, New York (1980).
18. D. C. McCormick, F. L. Test and R. C. Lessman, The effect of free-stream turbulence on heat transfer from a rectangular prism, *J. Heat Transfer* **106**, 268–275 (May 1984).
19. H. Miyazaki and E. M. Sparrow, Analysis of effect of free-stream turbulence on heat transfer and skin friction, *ASME J. Heat Transfer* **99**, 614–619 (November 1977).
20. R. Siegel and J. R. Howell, *Thermal Radiation Heat Transfer*. Hemisphere, Washington, DC (1981).
21. C. J. Hoogendorn and R. den Hond, Leidenfrost temperature and heat transfer coefficients for water sprays impinging on a hot surface, *Proc. Fifth Int. Heat Transfer Conf.*, Vol. 4, pp. 135–138 (1974).
22. E. R. Hosler and J. W. Westwater, Film boiling on a horizontal plate, *ARS J.* **1**, 553–558 (April 1962).
23. W. S. Bradfield, On the effect of subcooling on wall superheat in pool boiling, *J. Heat Transfer* **89**, 269–270 (1967).
24. L. A. Bromley, Heat transfer in stable film boiling, *Chem. Engng Prog.* **46**, 221–227 (1950).
25. S. W. Churchill, *The Interpretation and Use of Rate Data: the Rate Concept*. McGraw-Hill, New York (1974).
26. S. Deb, Boiling heat transfer of impacting liquid sprays on solid surfaces, Ph.D. Thesis, Carnegie–Mellon University (May 1988).

ANALYSE DU TRANSFERT THERMIQUE PAR EBULLITION EN FILM DE BROUILLARD IMPACTANT

Résumé—L'analyse de l'ébullition en film d'un liquide pulvérisé impactant est faite selon les deux cas: brouillard dilué (interaction thermique négligeable entre gouttelettes) et brouillard dense (interaction sensible). Le transfert thermique d'un brouillard dilué impactant est analysé en divisant les mécanismes en trois sous-mécanismes identifiés: transfert thermique par contact des gouttes, transfert thermique par convection d'air et transfert radiatif. On modélise les brouillards verticaux et horizontaux. Les prédictions sont très satisfaisantes. Pour l'ébullition en film de brouillard dense, l'approche asymptotique est suivie favorablement. L'asymptote basse correspond au cas du brouillard dilué tandis que l'asymptote haute est relative au cas de l'ébullition pour la plupart des conditions de brouillard. Le flux de masse liquide est le paramètre le plus important dans les deux cas. L'ébullition en film de brouillard dense est trouvée être très dépendante des paramètres de gouttelette; l'ébullition en film de brouillards dilués est aussi significativement influencée par ces paramètres.

ANALYSE DER WÄRMEÜBERTRAGUNG BEI DER FILMVERDAMPFUNG VON AUFTREFFENDEN SPRÜHSTRÖMUNGEN

Zusammenfassung—Die Untersuchung der Filmverdampfung einer auftreffenden Sprühströmung wird in zwei Fälle unterteilt—eine Sprühströmung geringer Dichte (vernachlässigbare Wärmeübertragung zwischen den Tropfen) und eine Sprühströmung hoher Dichte (merkliche Wechselwirkung zwischen den Tropfen). Der Wärmeübergang bei einer Sprühströmung geringer Dichte wird analysiert, indem der Vorgang in drei getrennte Unterprozesse eingeteilt wird—Wärmeübertragung durch Tropfenkontakt, konvektive Wärmeübertragung aus der strömenden Luft und Strahlungswärmeübertragung. Sowohl horizontale als auch vertikale Sprühströmungen werden behandelt. Die Vorhersagen sind sehr zufriedenstellend. Für Sprühströmungen hoher Dichte ist der asymptotische Ansatz erfolgreich. Die untere Asymptote entspricht der Situation der Sprühströmung geringer Dichte während die obere Asymptote für die meisten Strömungsbedingungen gut mit Ansätzen für das Behältersieden dargestellt werden kann. Der Flüssigkeitsmassenstrom ist in beiden Fällen der dominierende Parameter. Filmsieden von Sprühströmungen hoher Dichte zeigt eine sehr geringe Abhängigkeit von den Tropfenparametern, wogegen das Sieden von Sprühströmungen geringer Dichte deutlich von den Tropfenparametern beeinflußt wird.

АНАЛИЗ ТЕПЛОПЕРЕНОСА ПРИ ПЛЕНОЧНОМ КИПЕНИИ УДАРЯЮЩИХСЯ СТРУЙ

Аннотация—Анализ пленочного кипения ударяющихся струй проведен для двух случаев—разбавленной струи (взаимодействие между каплями пренебрежимо мало) и плотной струи (значительное взаимодействие). Теплоперенос в ударяющейся разбавленной струе анализируется методом разделения на три подпроцесса: теплоперенос при контакте капель, конвективный теплоперенос в воздухе и радиационный теплоперенос. Моделируются горизонтальная и вертикальная струи. В случае пленочного кипения плотной струи успешно применяется асимптотический метод, при этом одна из асимптот соответствует случаю разбавленной струи. В обоих случаях основным параметром является массовый поток жидкости. Найдено, что пленочное кипение плотной струи в очень малой степени зависит от параметров капель, в то время как для разбавленной струи наблюдается сильная зависимость.