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Heat transfer in subcooled jet impingement boiling at high wall temperatures

W. Timm [a], K. Weinzierl [b], A. Leipertz [a,*]

[a] *Universität Erlangen, Lehrstuhl für Technische Thermodynamik, Am Weichselgarten 8, D-91058 Erlangen, Germany*

[b] *Siemens I&S MP TC, Postfach 3240, D-91050 Erlangen, Germany*

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Abstract

An analytical approach for heat transfer modelling of jet impingement boiling is presented. High heat fluxes with values larger than 10 MW/m^2 can be observed in the stagnation region of an impinging jet on a red hot steel plate with wall temperatures normally being associated with film boiling. However, sufficiently high degrees of subcooling and jet velocity prevent the formation of a vapor film, even if the wall superheat is large. Heat transfer is governed by turbulent diffusion caused by the rapid growth and condensation of vapor bubbles. Due to the high population of bubbles at high heat fluxes it has to be assumed that a laminar sublayer cannot exist in the immediate vicinity of a red hot heating surface. A mechanistic model is proposed which is based on the assumption that due to bubble growth and collapse the maximum turbulence intensity is located at the wall/liquid interface and that eddy diffusivity decreases with increasing wall distance.

Keywords: Jet impingement; Subcooled flow boiling; Turbulent mixing; Boundary layer

1. Introduction

In engineering applications like cooling of hot metals in steel processing high heat transfer rates have to be realized. As an effective cooling method subcooled water jets impinging on the heated surface can be used. This type of cooling has been subject of numerous investigations, however, due to the complexity of the process, this type of heat transfer mechanism is still not fully understood.

Ishigai et al. [1] investigated jet impingement heat transfer for wall temperatures up to ~1300 K and subcoolings up to 55 K. Characteristic points of the boiling curve were shifted to higher heat fluxes and wall temperatures when subcooling and jet velocity were increased. They reported that at subcoolings larger than 55 K no film boiling could be established within the range of the experiment with a jet velocity of about 2 m/s. Boiling curves for the nucleate boiling regime were given by Miyasaka et al. [2]. Their range of operating conditions covered jet velocities of up to 15.3 m/s and wall superheats of up to ~1100 K at a subcooling of 85 K with which heat fluxes of more than 40 MW/m^2 could be obtained. The same order of magnitude for heat transfer rates in subcooled flow boiling was reported by Gunther [3] who also measured bubble growth rates by means of high speed photography. He found that increasing subcooling and surface heat flux led to decreasing mean bubble diameters and increasing bubble populations and frequencies. Opposed to saturated pool boiling, in which bubbles generated at the heating surface detach from the wall when buoyancy forces exceed inertial and surface tension forces, Gunther found that in highly subcooled boiling

* Corresponding author. Tel.: +49-9131-85-29900; fax: +49-9131-85-29901.
E-mail address: sek@ltt.uni-erlangen.de (A. Leipertz).

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Nomenclature

a	laminar thermal diffusivity
c_p	specific heat capacity
c_ε	model constant, Eq. (11)
C	velocity gradient in jet impingement zone
$\overline{C}$	constant for evaluation of C
F	fraction of wall covered with bubbles
H	dimensionless stream function
I	dimensionless wall velocity function
N	dimensionless eddy diffusivity function
p	pressure
p_0	pressure in stagnation line
Pr_t	turbulent Prandtl number
$\dot{q}_{wall}$	average wall heat flux
$\dot{q}_x$	local wall heat flux
r_B	time averaged maximum bubble radius
t_B	mean bubble lifetime
T	water temperature
T_{sat}	water saturation temperature
T_w	wall temperature
T_∞	free stream temperature
u	flow velocity component, x-direction
u_∞	free stream u-velocity
v	flow velocity component, y-direction
v_j	jet velocity
v_s	velocity of moving heating wall
v_∞	free stream v-velocity
w	jet width
x	cartesian coordinate, parallel to wall
y	cartesian coordinate, normal to wall

Greek symbols

δ	height of boundary layer
ε_h	eddy diffusivity of heat
ε_m	eddy diffusivity of momentum
$\varepsilon_{m_{max}}$	maximum diffusivity near wall
ε^+_{max}	dimensionless diffusivity
η	dimensionless wall distance
ρ	specific density

vapor bubbles collapse due to condensation shortly after formation without leaving the wall. Typical orders of magnitude for lifetimes of 0.1 ms and for maximum radii of 0.1 mm were reported. A similar study was performed by Del Valle and Kenning [4] who reported similar results.

The mechanism of bubble growth and collapse itself was subject of many studies (see, e.g., [5,6]) and to date it is commonly accepted that in highly subcooled boiling the inertia of the surrounding liquid is the governing factor for bubble lifetimes and diameters. Nordmann [7] investigated this process by means of holographic interferometry. He found that the condensation rates of such bubbles lead to rapid implosions, resulting in heated liquid being erupted into the core of the flow.

The high level of turbulent mixing associated with this observation may serve as the basis of a physical model to explain the heat transfer rates of water jets as they are used, e.g., in hot strip mill runout table cooling. In another investigation [16] we found that heat fluxes between 15 and 45 MW/m^2 could be obtained by water jets with a flow velocity of 7 m/s and a subcooling of about 75 K impinging on moving hot steel strips with surface temperatures being in the range between 820 and 1220 K. Heat transfer rates were found to be almost independent of the velocity of the moving surface, but increased with surface temperature, indicating a kind of nucleate boiling. In this paper we develop an analytical model which describes this behaviour quite well.

2. Problem description

The flow profile in the stagnation zone of an impinging jet with a velocity v_j and temperature T_∞ on a non-moving surface is depicted in Fig. 1. Heat transfer and velocity distributions in this region are well known for the laminar flow where no boiling takes place. Solutions of the transport equations for this case can be obtained by means of similarity analysis and boundary layer theory. Zumbrunnen [8] extended this procedure for the case in which the impinging surface is moving with velocity v_s. In a more general approach Fujimoto et al. [9] solved the time-dependend Navier–Stokes and heat conduction equations for the entire domain including a k–ω turbulence model and appropriate boundary conditions at the free upper surface. All these investigations were restricted to situations in which the wall temperature was kept below boiling temperature. Boiling would create additional contributions to the turbulence within the liquid, requiring a highly sophisticated model including two-phase flow and boiling-induced mixing effects for which it would be difficult to find experimental data which are sufficient for model validation. However, the special type of boiling considered in this paper shows some features which suggest a further extension of Zumbrunnens approach to derive a physical model for subcooled jet impingement boiling with high heat transfer rates. As pointed out before, there is strong indication that the actual boiling process is limited to a small zone adjacent to the heating wall. According to Gunther's [3] observations the void frac-

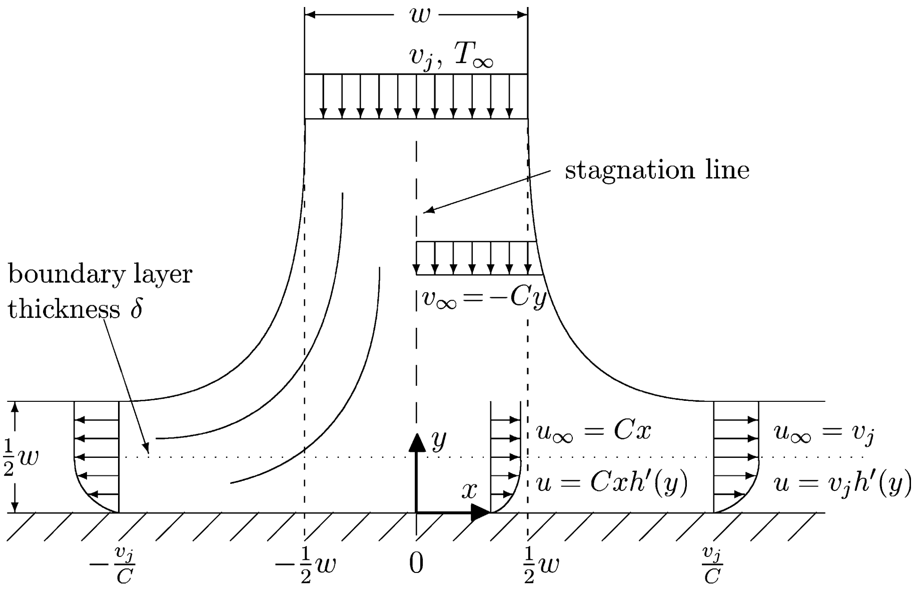

Fig. 1. Stagnation flow profile on a stationary surface.

tion F of the heating surface which is only covered with small bubbles significantly depends on the wall heat flux. He reported that more than 10% of the heating surface was covered with bubbles when a heat flux of 10.63 MW/m^2 was applied to a flow of mean velocity of 3 m/s parallel to the surface. The subcooling was 86 K. When extrapolating these results to heat fluxes of about 30 MW/m^2 the value of F would reach the range of 0.2–0.6 which is reasonable for the nucleate boiling regime as it is analysed here in this paper. The maximum heat flux will occur when F reaches a critical value at which the wall bubbles coalesce, forming a stable vapor film. According to Kwon and Chang [10] this critical value of F is about 0.82 and beyond the scope of this paper. In that investigation turbulent interaction between a bubbly wall layer and the core of the flow was proposed to be the primary mechanism of heat exchange between the heating wall and the subcooled fluid. The wall layer was treated as an additional surface roughness and the universal velocity profile consisting of laminar sublayer, buffer layer and fully turbulent layer was used.

The use of the universal velocity profile is a feature which has commonly been used in several models for highly subcooled flow boiling (see, e.g., [11,12]). The existence of a laminar sublayer adjacent to a wall with temperature T_w being significantly larger than the saturation temperature T_{sat} is, however, highly questionable, hence the use of the universal velocity profile is inappropriate for the case considered here.

The proposed mechanism for both the heat and the momentum exchange of an extremely superheated wall and an impinging subcooled water jet is illustrated in Fig. 2. A layer of alternately growing and collapsing bubbles with average radius r_B and lifetime t_B of about 10^{-4} m and 10^{-4} s, respectively, in the immediate vicinity of the wall is acting as a source of turbulent mixing. The high degree of subcooling of the surrounding liquid and the impinging flow prevent the bubbles from detaching from the wall, limiting the entire boiling and condensation process to a boundary layer phenomenon. It can be assumed that a single bubble, while growing, will replace a hot fluid volume near the wall that is subsequently shifted into colder regions of the flow. Momentum and heat exchange of such a deplaced fluid volume with the main flow will result in a temperature drop in the fluid layer surrounding the bubble, leading to its collapse immediately after it has reached its maximum radius. Cold fluid will then be transported back to the heating wall and the process repeats. As shown by Nordmann [7] this effect is of quite vigorous nature, implying that the mechanism of heat and momentum transfer between a superheated wall and a subcooled liquid is highly diffusive, resulting in turbulent temperature and velocity profiles (see also [6]).

Since it can also be assumed that large parts of the wall are covered with these bubbles causing micro-explosions and implosions, the maximum turbulence intensity and hence the eddy diffusivity is located at the wall. This kind of turbulence is a quality of the fluid and of the wall superheat rather than a quality of the flow. One can therefore suggest the use of mixing-length theory as a sufficient first order approach to obtain eddy diffusivities. Due to dissipation the turbulent motions fade away in an asymptotic manner as the wall distance

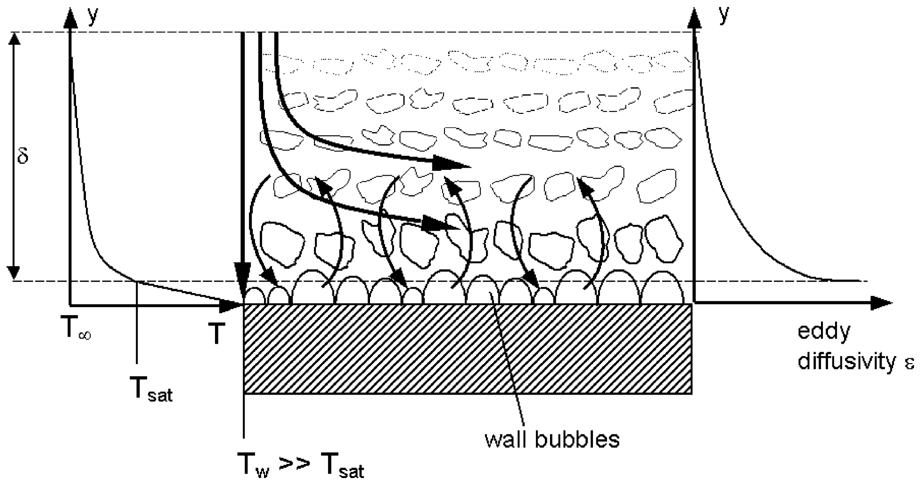

Fig. 2. Turbulence generation at the heating wall.

is increased leaving the outer regions of the flow undisturbed.

3. Physical model

The following assumptions and initial or boundary conditions were used to derive the proposed model:

- The flow is steady, heat transfer rates are independent of time.
- Net vapor generation is negligible, the problem is treated as an incompressible single-phase flow.
- The wall temperature is constant and several times larger than the saturation temperature. A laminar fluid layer in the direct vicinity of the heating wall cannot exist because of too high superheat.
- The fluid layer adjacent to the bubbly wall layer is at saturation temperature and is set to be the lower boundary of the computational domain.
- Temperature distributions within the superheated vapor/bubble zone near the wall are not considered. The influence of wall temperature is taken into account by an increased turbulent diffusivity near the wall.
- The no-slip condition is applied at the lower boundary, zero-gradient conditions at the left and right boundary of the domain (see Fig. 1).
- The flow outside of the boundary layer is laminar.
- Boiling-induced turbulence is a local effect restricted to a zone of thickness δ near the wall in which the eddy diffusivities ε_m for momentum and ε_h for heat are at least one order of magnitude larger than their respective molecular values, v and a (see Eqs. (2) and (3)).

The main equations to be solved are the well known boundary layer equations for momentum and heat transfer.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial y}\left[(v + \varepsilon_m)\frac{\partial u}{\partial y}\right] \tag{2}$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \frac{\partial}{\partial x}\left[(a + \varepsilon_h)\frac{\partial T}{\partial x}\right] + \frac{\partial}{\partial y}\left[(a + \varepsilon_h)\frac{\partial T}{\partial y}\right] \tag{3}$$

These equations are solved subsequently. As mentioned before, the flow profile given by Eqs. (1) and (2) can be obtained by a similarity approach. Once the components of the velocity vector in cartesian x and y-directions, u and v, are known, they can be introduced into Eq. (3) which is then solved numerically. Outside of the boundary layer the velocities are given by

$$u_\infty = Cx \tag{4}$$

$$v_\infty = -Cy \tag{5}$$

Here, the velocity gradient C can be expressed in terms of a constant $\overline{C}$, geometrical jet width before impinging the wall w and and jet velocity v_j (see Fig. 1) by

$$C = \overline{C}\frac{v_j}{w} \tag{6}$$

The width of the impingement region, in which a pressure gradient exists, is governed by the velocity gradient C, hence the constant $\overline{C}$ has to be known. Typical values are of the order of unity, e.g., Zumbrunnen [8] used $\overline{C} = 0.7854$. In this paper we assume a value of 1.0 for all calculations. The difference between the static pressure p_0 at the stagnation line and the pressure p at a certain point (x, y) is given by Bernoulli's equation

$$p_0 - p = \tfrac{1}{2}\rho C^2[x^2 + y^2] \tag{7}$$

Inside the boundary layer the effect of a moving surface of velocity v_s and boiling-induced turbulent mixing have to be included. This is done by introducing a dimensionless boundary layer coordinate $\eta \sim y/\delta$,

$$\eta = y\sqrt{\frac{C}{\varepsilon_{m_{max}}}} \tag{8}$$

Here, in the denominator of Eq. (8) the maximum eddy diffusivity $\varepsilon_{m_{max}}$ was chosen instead of the normally used molecular viscosity ν. This was done to obtain a reasonable dimensionless coordinate η because the high level of boundary layer turbulence and the proposed absence of a laminar sublayer suggest velocity and hence temperature profiles that are not affected by molecular diffusion in regions near the wall. As shown by Zumbrunnen [8] the flow velocity components can then be expressed in terms of a stream function H and an additional function I to incorporate the effect of surface motion, both of which are solely dependent on η, by

$$u = Cx\frac{\partial H}{\partial \eta} + v_s I(\eta) \tag{9}$$

$$v = -\sqrt{C\varepsilon_{m_{max}}}H(\eta) \tag{10}$$

For the diffusive term in Eq. (2) an expression has to be chosen that reflects the nature of the problem. The effect that a bounding wall has on turbulence is normally of damping nature. Velocity fluctuations are generally believed to decrease with decreasing wall distance and are supposed to approach zero at the wall inside a very thin viscous sublayer in which molecular friction is the dominant effect. This behaviour has been taken into account by Van Driest's damping function in connection with the mixing-length concept of Prandtl (see, e.g., [13]). The basic idea of this approach with some modifications can be used for the problem considered here. It can be assumed that the flow is subject to strong oscillatory motions at the heating wall because of an extremely superheated two-phase zone from which vapor bubbles are generated with high frequency causing turbulent velocity fluctuations. Viscous dissipation will diminish these fluctuations with distance from the wall, therefore they are absent in the laminar main flow outside of the turbulent boundary layer. Hence the diffusive term can be expressed in a dimensionless form by

$$N \equiv (\nu + \varepsilon_m)/\varepsilon_{m_{max}} = e^{-c_\varepsilon \eta} + \nu/\varepsilon_{m_{max}} \tag{11}$$

Opposed to the original approach that describes a turbulent flow profile with a laminar sublayer, the case investigated here is a laminar flow with a bubble-induced turbulent sublayer. Hence the velocity fluctuations are not dependent on the main flow and might rather be correlated with characteristic boiling parameters by

$$\varepsilon_{m_{max}} \propto Fr_B^2/\left(\tfrac{1}{2}t_B\right) \tag{12}$$

This correlation can be seen as the product of a characteristic length $\sim r_B$ and a velocity $\sim r_B/\left(\frac{1}{2}t_B\right)$. The parameter F accounts for the fact that the bubble coverage of the heating wall also is likely to have a strong effect on turbulent agitation within the fluid. Analytical boiling models based on a bubble-induced turbulence have already been proposed (see, e.g., [14,15]), but these mostly dealt with moderate wall temperatures and used analytical expressions for bubble parameters. Those expressions are usually based on buoyancy and surface tension force equilibrium, hence the use for high heat flux applications which are inertia controlled seems inappropriate.

With the above transformation, Eq. (2) can be rearranged as a system of ordinary differential equations

$$N'H'' + NH''' + HH'' - (H')^2 + 1 = 0 \tag{13}$$

$$N'I' + NI'' + HI' - IH' = 0 \tag{14}$$

The terms with primes denote derivatives with respect to η.

The conditions that have to be satisfied by the solution to this system are

$$\eta = 0: \quad H = H' = 0; \quad N = I = 1 \tag{15}$$

$$\eta \to \infty: \quad H' \to 1; \quad N \to 0; \quad I \to 0 \tag{16}$$

for cases in which $\varepsilon_{m_{max}} \gg \nu$.

In order to solve this system, one has to know the function N. Since it can be assumed that $\varepsilon_{m_{max}} \gg \nu$ for the problem considered here, an appropriate value for c_ε must be found. For $N \equiv 1$ $(c_\varepsilon \to \infty; \varepsilon_{m_{max}} = \nu)$ this system reduces to the laminar case which was originally solved by Zumbrunnen [8], the solution is plotted in Fig. 3. From this graph it can be seen that a value of $\eta = 4.0$ is sufficient for the estimation of the dimensionless boundary layer thickness δ $(H' \to 1)$ with hardly any significant changes in the zone where $\eta > 2.5$. The function I for wall motion effects is of special interest, because it incorporates an effect quite similar to the dimensionless eddy viscosity N which accounts for oscillatory motions at the wall/liquid interface and hence should behave in a similar way. The asymptotical nature in which the effect of wall motions diminish is reflected by the exponential form of Eq. (11). From the laminar solution one can see that surface motion effects, represented by I, are essentially diminished for $\eta > 3.0$. Assuming that bubble-induced fluctuations fade away in a similar way, a value of $c_\varepsilon = 2.0$ appears to be reasonable, because no measured data are available for this kind of problem that would allow an exact determination of

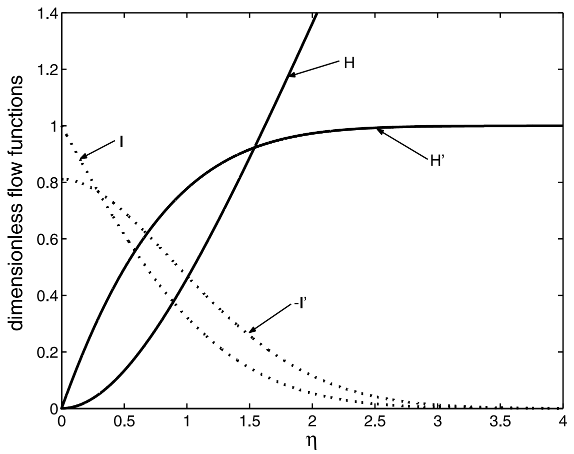

Fig. 3. Solution of the differential equations for a laminar flow profile with $N \equiv 1$ (see [8]).

the turbulent velocity distributions. The system can then be solved, the result is plotted in Fig. 4. Opposed to the laminar solution, the influence of a moving wall on the velocity profile has a pronounced turning point at $\eta \approx 0.681$ as can be seen from the slope of the first derivative of I. The reason for this is the increased influence of diffusive effects near the wall. The edge of the boundary layer can be located at $\eta \approx 2.0$ and strongly coincides with the shape of the function N which essentially determines the height of the boundary layer implying that boiling-induced diffusion is the dominating mechanism.

The general solution depicted in Fig. 4 is independent of $\varepsilon_{m_{max}}$, however, in order to obtain sufficient solutions for engineering calculations, appropriate values for the maximum eddy diffusivity according to Eq. (12) have to be determined. Both Gunther [3] and Del Valle and Kenning [4] found that in subcooled flow boiling at high heat fluxes the bubble population density increased significantly when heat transfer rates were increased. A moderate decrease in average maximum bubble radius and lifetime was reported by Gunther within a range of 5–10 MW/m^2. Data of these quantities for heat fluxes above 10.63 MW/m^2 (which are of special interest for this paper) could not be given, because of limited photographical resolution. However, the basic trends seem to indicate innumerable quantities of extremely small and short-lived bubbles all over the heating surface. In any case the results at the highest heat flux that was investigated can serve as a basis for estimating the typical order of magnitude for $\varepsilon_{m_{max}}$ ($r_B = 1.71 \times 10^{-4}$ m, $t_B = 1.42 \times 10^{-4}$ s, $F = 0.1579$).

To compare turbulent bubble diffusion with the typical molecular value, a dimensionless maximum diffusivity

$$\varepsilon^+_{max} = \frac{\varepsilon_{m_{max}}}{\nu_{ref}} \tag{17}$$

with $\nu_{ref} = 5 \times 10^{-7}$ m^2/s can be introduced. With Gunther's data for F, r_B and t_B mentioned above, the evaluation of Eq. (12) gives a value of $\varepsilon^+_{max} \approx 130$. As one can see from this example the assumption $\varepsilon_{m_{max}} \gg \nu$ is certainly satisfied. For heat fluxes larger than 10 MW/m^2 even higher values for ε^+_{max} can be expected.

Another question of interest is, how thick the turbulent boundary layer is in comparison to the chosen characteristic mixing length i.e. average maximum bubble radius. Assuming that the turbulent Prandtl number Pr_t equals unity ($\varepsilon_m = \varepsilon_h$) Eq. (3) can be solved, the temperature profiles for various values of ε^+_{max} are plotted in Fig. 5 for the case of constant $T_{sat} = 373$ K ($T_\infty = 298$ K, $v_j = 7$ m/s, $w = 1.5$ cm).

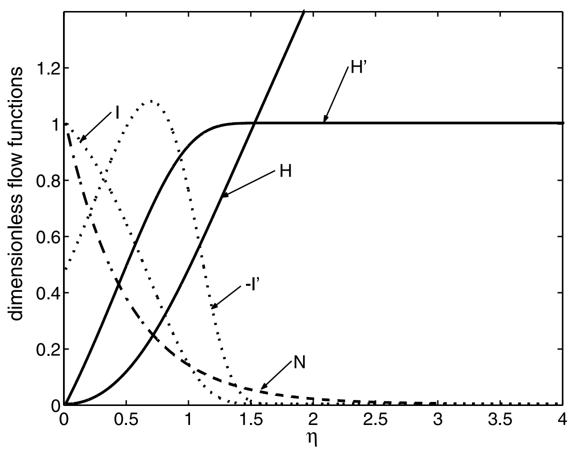

Fig. 4. Solution to differential equations for bubble-induced turbulent flow profile with $c_\varepsilon = 2.0$.

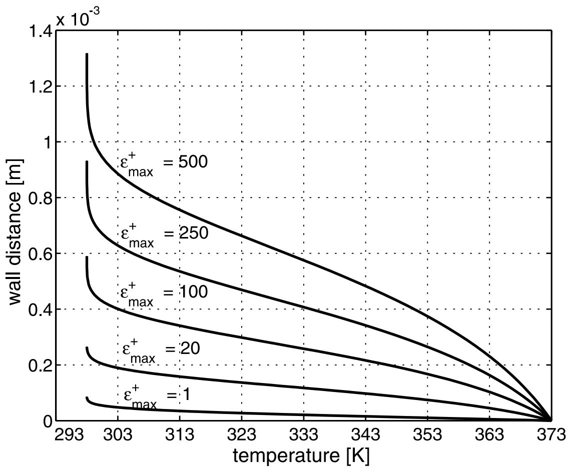

Fig. 5. Turbulent temperature profiles as a function of maximum eddy diffusivity ($\varepsilon^+_{max} = \varepsilon_{m_{max}}/\nu_{ref}$, $Pr_t = 1$).

4. Model calculations

Values for the effective boundary layer thickness can be obtained from Fig. 5 by simply taking the wall distance at which the water temperature approximately equals T_∞. The problem is treated as a single-phase flow, because there is hardly any net vapor generation, but there is a strong likelihood of a superheated two-phase zone with an average thickness that is similar to the mean surface roughness. From this superheated film heat is transferred to outer regions of the flow mainly by bubble-induced mixing. Therefore an imaginary plane through the wall bubble layer can be taken as the lower boundary of the domain at which the water is at saturation temperature. Hence the thickness of the boundary layer should be several times larger than half the value of the estimated thickness of the bubble layer ($\delta > \frac{1}{2}r_B$) to justify the similarity approach being applied in this paper. For the given example this condition is satisfied for values of $\varepsilon^+_{max} > 50$. Bubble radii and lifetimes of about 10^{-4} m and 10^{-4} s can be seen as an upper limit, because the impinging flow and high heat fluxes considered here will most likely shift these parameters to smaller values. Looking at Fig. 5 one can see that for high diffusivities of about $\varepsilon^+_{max} > 500$ the boundary layer thickness exceeds 1 mm, but is still about 10 times smaller than the original jet width of $w = 1.5$ cm which is another precondition for the applicability of boundary layer theory. Hence, the assumption of eddy diffusivities that are several hundred times larger than their molecular counterparts in conjunction with a turbulent boundary layer seem to be reasonable in impinging jet configurations, as long as the jet width itself is significantly larger than the corresponding mixing length (r_B). For the cases considered in this paper this corresponds to a jet width of about 1 cm and above.

As mentioned before, an engineering application in which heat transfer conditions of this kind occur is the cooling of steel strips on the runout table of hot strip mills. The initial supposition that the surface temperature is several times larger than the boiling temperature of water is of special importance in this case. With strip velocities v_s typically being in the range between 2 and 20 m/s the residence time of the hot steel surface in the impingement zone of a water jet lasts only a few milliseconds. During this time the cooling water is in direct contact with the steel strip when subcooling and jet velocity are sufficiently high, whereas an insulating vapor film is formed between water and strip outside of the impingement zone. While passing this zone of direct liquid/solid contact, the strip surface temperature typically drops about 100–200 K, but remains at temperatures of above ~750 K.

A statistical analysis of hot strip mill operating data (see [16]) lead to the results that are plotted in Fig. 6. Here the average heat flux transferred to the cooling water is plotted for different strip velocities versus an arithmetic mean temperature of the strip surface derived from the values shortly before and after it has passed the impingement zone of estimated width of 2 cm. Heat flux is increasing with increasing surface temperature but almost independent of the moving surface speed. As shown by Zumbrunnen [8] the latter aspect is a typical feature in stagnation flow heat transfer when the temperature of the fluid layer adjacent to the heating wall is constant. In this case one might expect that saturation temperature is the appropriate condition for the lower boundary which is a function of the system pressure. The maximum pressure is located in the stagnation line, the pressure distribution can be calculated by Eq. (7). The corresponding temperature distribution $T_{sat}(x)$ was used to calculate local heat fluxes in the impingement region for different ratios v_s/v_j of surface to jet velocity. The local heat flux $\dot{q}_x$ is given by

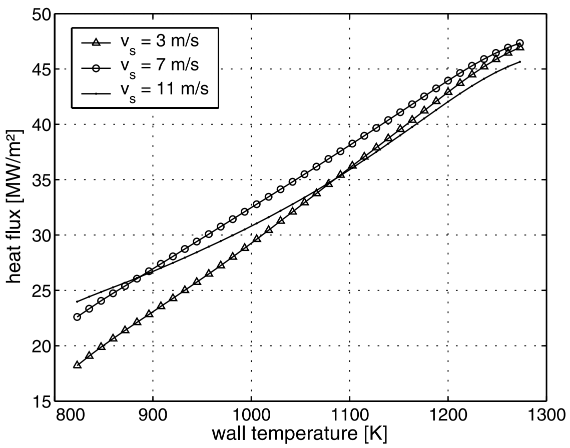

Fig. 6. Statistically determined heat flux as a function of wall temperature and velocity ($T_\infty = 298$ K, $v_j = 7$ m/s).

$$\dot{q}_x = v_{ref}\varepsilon^+_{max}\rho c_p \left|\frac{\partial T}{\partial y}\right|_{y=0} \tag{18}$$

The results for an eddy diffusivity near the wall of $\varepsilon^+_{max} = 250$ and surface velocities in the range between 0 and 14 m/s, which give surface to jet velocity ratios between 0 and 2.0, are plotted in Fig. 7. On a stationary surface the maximum heat flux is located at the stagnation line ($x = 0$), which obviously is a result of an increased saturation temperature in the stagnation line (approximately 378 K in this case). The model implies that heat flux rates are very sensitive with respect to subcooling, a drop of just 5 K from the stagnation line to the left and right boundaries of the domain result in a decrease of approximately 3.4 MW/m^2.

The strong dependence of heat transfer rates on subcooling was reported by several investigators (see, e.g., [2]) and hence adds to the model assumption of an

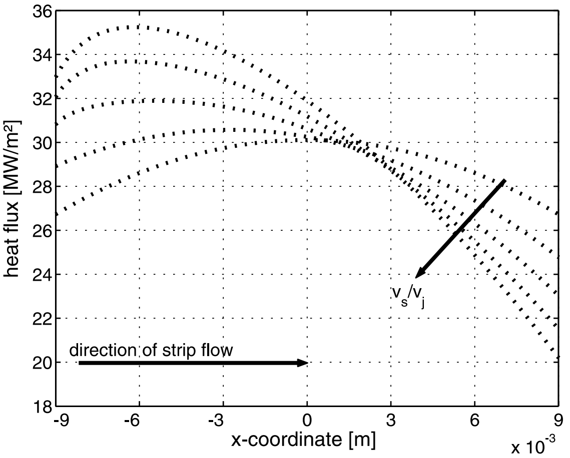

Fig. 7. Calculated heat transfer rates for various surface velocities ($\varepsilon^+_{max} = 250$, $v_j = 7$ m/s, $T_\infty = 298$ K).

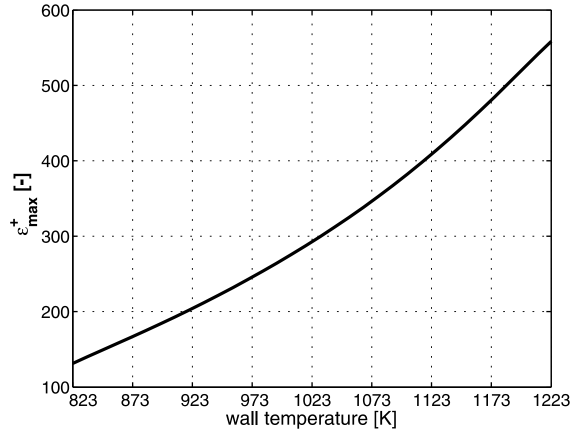

Fig. 8. Maximum eddy diffusivity as a function of wall temperature according to Fig. 6.

increased diffusivity, because the subcooling is essentially the governing temperature difference for heat transport within the fluid. As the surface speed (in x-direction) is increased, the maximum of local heat flux is shifted upstream because of a larger velocity difference between heating wall and cooling medium in counter flow than in downstream parallel flow. These effects almost compensate each other because the average wall heat flux which is given by

$$\dot{q}_{\mathrm{wall}} = \frac{C}{2v_{\mathrm{j}}} \int_{-v_{\mathrm{j}}/C}^{v_{\mathrm{j}}/C} \dot{q}_x \,\mathrm{d}x \tag{19}$$

only slightly increases with increasing v_s. For this example values for $\dot{q}_{\mathrm{wall}}$ between 29.4 and 30.5 MW/m^2 are obtained. These values are well in the range of those that are given in Fig. 6. In order to obtain eddy diffusivities, suitable to reproduce actual cooling conditions the data from Fig. 6 were recalculated by varying $\varepsilon_{m_{max}}$ until the desired model value for $\dot{q}_{\mathrm{wall}}$ at a given wall temperature was achieved. The wall temperature is assumed to be the governing factor for mean bubble radius and frequency. Surface motion might also have an effect on these parameters, but this is not revealed by the given statistical data. Therefore the influence of a moving wall on turbulent diffusivity is neglected. The resulting function is plotted in Fig. 8.

In Fig. 9 the corresponding values for heat fluxes are plotted that were calculated by the model. As one can see the results are in good agreement with those given in Fig. 6. The additional convective heat transport associated with increasing surface velocities is almost negligible in the range investigated here. The obtained eddy diffusivities cover values between $\varepsilon^+_{max} \approx 131$ and $\varepsilon^+_{max} \approx 558$. Pan et al. [14] investigated the transition boiling regime in saturated pool boiling and reported

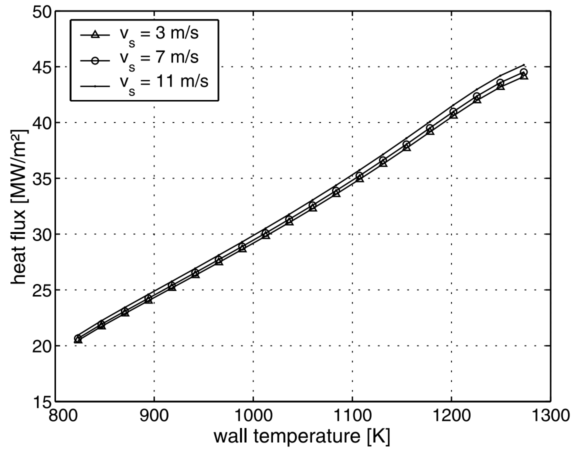

Fig. 9. Heat transfer rates—model calculation with ε^+_{max} from Fig. 8.

increased turbulent diffusivities of about $\varepsilon^+_{max} = 75$ for wall temperatures of 600 K. Hence the obtained eddy diffusivities in this study seem to have a reasonable order of magnitude and can very well be explained by bubble agitation according to Eq. (12).

5. Conclusions

An analytical model for subcooled jet impingement boiling was presented for cases in which heat flux and wall temperature are several times larger than in saturated nucleate pool boiling. Velocity and temperature profiles are obtained by means of a similarity approach that includes boiling-induced turbulent mixing.

The proposed model is able to reproduce the principal behaviour of this type of jet cooling as it can be observed, e.g., on runout tables of hot strip mills. The general trends are in good agreement with previous investigations.

The subcooled boiling process near the heating wall is of special interest. The heat transfer mechanism is assumed to be highly diffusive due to bubble growth and collapse. Therefore it is of special interest to obtain more information about bubble characteristics under extreme conditions i.e. high subcooling, wall superheat and flow velocity. This will be subject of further investigations.

It can be assumed that increasing surface temperature, subcooling and jet Reynolds number result in increasing heat fluxes due to higher turbulence, but to date no information about bubble frequencies and dimensions in this kind of problem are known.

In order to develop an advanced numerical model on basis of direct numerical simulation detailed information about velocity fluctuations near the wall have to be given. Hence, future research work will also focus on highspeed photographical studies to clarify the effects of all involved relevant parameters on boiling characteristics and heat flux in the jet impingement zone.

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