Temperature fields induced by direct contact condensation of steam in a cross-flow in a channel

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Abstract The temperature fields in the center plane of a channel with a square cross-section have been measured. Steam injected at relatively low mass fluxes through a small hole in one of the walls of the channel condensed intermittently in a small area close to the inlet. The upstream temperature of the liquid cross-flow, T_L , the momentum ratio, J , and the Prandtl number proved to be important for the single-phase temperature field induced in the jet further away from the steam inlet. Jet centerlines of velocity and temperature are measured and positions are compared. Different locations for $J < 100$ and low T_L are explained from dependencies on Reynolds and Prandtl numbers. Next to the jet centerline a second high-temperature zone was found to occur, close to the wall and downstream of the steam inlet. The importance of capillary forces is investigated with the aid of 3D CFD computations.

List of symbols

$C_{am,j}$	Added mass coefficients ($j = 1, 2$) (–)
C_p	Specific heat at constant pressure (J/kg K)
d	Diameter, $2R_a$ (m)
F	Force (N)

G	Steam mass flux (kg/m ² s)
h	Distance of steam pocket centre to injection wall (mm)
J	Momentum ratio, Eq. (1) (–)
k	Heat conduction coefficient (W/m K)
p	Pressure (0.1 MPa)
Pe_T	Péclet number, LV/α (–)
Pr	Prandtl number, ν/α (–)
Q	Flow rate (m ³ /s)
r	Velocity ratio, $\sqrt{J}$ (–)
r_{foot}	Bubble foot radius (mm)
R	Radius of steam pocket (mm)
Re	Reynolds number, LV/ν (–)
$(R + h)$	Penetration depth (mm)
R_a	Radius of steam inlet (mm)
$\dot{R}$	Growth rate of steam pocket radius (m/s)
t	Time (s)
T	Temperature (°C)
U	Wall normal steam pocket velocity (m/s)
V	Typical velocity (m/s)
$\forall$	Steam pocket volume (m ³)
x	Wall-normal coordinate (Fig. 4) [m]
y	Wall-parallel coordinate (Fig. 4) [m]

Greek letters

α	Heat diffusivity, $k/\rho C_p$ (m ² /s)
μ	Dynamic viscosity (Pa s)
ν	Kinematic viscosity, μ/ρ (m ² s)
ρ	Mass density (kg/m ³)
σ	Surface tension coefficient (N/m)
ξ	Coordinate along jet centerline (Fig. 4) [m]

Subscripts

$Cond$	Anisotropic condensation
L	Water, liquid
T	Thermal

Dedicated to Prof. Dr.-Ing. Dr.-Ing. E.h. mult. Franz Mayinger on the occasion of his 80th birthday.

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x	In wall-normal direction
y	In wall-parallel direction
V	Steam, vapor

1 Introduction

Direct steam injection is a very effective way to rapidly and homogeneously heat fluids [15]. A well-known industrial application is the sterilization process of milk. To improve the taste of the milk it is necessary to decrease the heating time and increase the process temperature during sterilization. Turbulent mixing and heating phenomena induced by the condensation of steam in a cross-flow of water have in the past 5 years been investigated by us in various ways. This paper focuses on experimental results regarding temperatures and heat transfer in connection to topology changes of the injected steam. In addition, results of 3D-numerical simulations with the Volume-Of-Fluid method (VOF) in Fluent are presented, highlighting the validity of an assumption which was made in an analytical model to predict the penetration depth of steam [6].

When steam is injected into sub-cooled water direct contact condensation occurs. Depending on process conditions like steam mass flux, G , bulk water temperature, T_L , and on the injection configuration (direction of injection, nozzle diameter and shape), different regimes of direct contact condensation can be found. At high steam mass fluxes (near sonic and sonic), the steam forms either an oscillatory or a stable vapor jet that ends at a certain distance from the injector. Stable steam jets require choked injector flow and have been studied extensively, see Weimer et al. [16] and Chen and Faeth [4]. At lower steam mass fluxes, the condensing steam forms a vapor pocket that continuously grows and collapses at the steam injection hole. At very low steam fluxes and high water sub-cooling, the steam-water interface moves periodically in and out of the injection hole and this regime is indicated as chugging. For stagnant pools, both regimes have been investigated by Nariai and Aya [9], Aya and Nariai [1], Youn et al. [17] and Chan and Lee [3].

The topology of the steam-liquid interface during direct-contact condensation is not smooth. On the contrary, waves and interface instabilities cause an apparent interface roughening that undoubtedly must affect heat and mass transfer. In the case of relatively small injection Reynolds numbers, required spatial and temporal resolutions in measurement and simulation is less than at high injection Reynolds numbers. For this reason, low Reynolds injection numbers have been chosen in this study of direct contact condensation.

Previous studies of unstable steam condensation at low mass fluxes of steam merely focused on pressure

oscillations and regime transitions. The present paper describes an experimental study of unstable direct steam condensation in channel flow to determine time-averaged temperature fields in relation to interface topology. In most applications temperatures induced by the steam in the liquid should be as homogeneous as possible. The present paper investigates the importance of various dimensionless parameters for homogeneity of the temperature field induced in a cross-flow of water. Particular new features of the experiments are the effects of a liquid cross-flow and of temperature of the approaching liquid.

The topology history of steam pockets at low mass fluxes of injected steam is reminiscent of that of bubble growth during convective boiling at high heat fluxes. For this reason a model to predict growth, total time of growth and maximum steam pocket size was developed [6] based on existing knowledge of forces occurring during detachment of boiling bubbles [13]. Capillary forces are assumed to be unimportant in this model. The validity of this assumption is checked in the present paper with the aid of 3D VOF-simulations with Fluent^{TM}.

2 Experimental

Figure 1 schematically shows the experimental set-up with the steam supply system on the right. A steam generator delivers saturated steam at a maximum pressure of 10 bar (absolute) and a maximum flow rate of 0.33 kg/s. A reduction valve enables control of the pressure downstream of the main supply line. Before injection into the test section the steam is fed to a condenser, to heat up the supply lines and prevent condensation during actual injection. Only a small steam flow rate (0.8 kg/h maximum) is needed in the test section during an experimental run. At a distance of two meters upstream of the steam injection point, the diameter of the steam supply line gradually decreases from DN 40 (ϕ 40 mm) to DN 4 (ϕ 4 mm). The steam flow rate after this convergence is measured by a Coriolis mass flow meter (accuracy 1% of measured value) and controlled by a PID-actuated pneumatic valve. Further downstream, 150 mm upstream of the injection point, a pressure transducer (accuracy 0.1% full scale range which is between 0 and 10 bar absolute) and a Pt-100 element monitor the injection conditions of the steam. This DN 4-section of the steam line as well as the mass flow controller is covered with an electrical heating wire and insulation to avoid condensation.

The measurement loop of Fig. 1 is a closed circuit containing approximately 50 l of demineralized water. The flow is upward in the test section and is provided by a frequency controlled centrifugal pump. The volumetric flow rate of the water is measured with an ultrasonic flow

Fig. 1 Schematic of test rig

meter (accuracy 0.25% full scale which is $0\text{--}9 \times 10^{-4} \text{ m}^3/\text{s}$). The closed loop can be pressurized up to 8 bar absolute via an expansion vessel whose upper half is connected to a pressurized air supply. The expansion vessel also minimizes pressure fluctuations. Four Pt-100 elements monitor the water temperature at various locations in the loop. The system pressure and the water temperature are constant during the measurements despite the injection of steam. A PID-actuated bleed valve that is connected to a pressure transducer, accuracy 0.1% full scale which is 0–7 bar absolute, is located at the inlet of the measurement section. It controls the pressure level. Water temperature is kept constant during steam injection with the aid of a heat exchanger and a 17 kW electrical heater whose output power is controlled by a PID-controller. Both are positioned downstream of the test section.

The measurement section has a square inner cross-section of $30 \times 30 \text{ mm}^2$ and is optically accessible at the location where steam is injected. The steam is injected at saturation conditions (in some cases slightly superheated) through a flush-mounted wall injector with a circular hole with a diameter of 2 mm. The whole set-up is thermally insulated with a 20 mm thick foam layer.

Velocity fields were measured with PIV [5]. For the temperature measurements described below three Class A PT-100 temperature sensors with a diameter of 1 mm have been introduced at various positions in or near the center plane of the channel in a way depicted in Fig. 2. The probe tips are not precisely in a plane in order to minimize mutual interaction. Three temperatures are recorded simultaneously during 2 min. Only time-averaged values of temperatures are deduced. Temperature TI-104, see Fig. 2, is

Fig. 2 Schematic of thermocouple mounting and (re)-positioning in test section

measured 2 m upstream of the steam injection point and yields the bulk temperature T_L . Positions of the thermocouples have been measured with a camera and with accuracy better than $\pm 0.5 \text{ mm}$. All PT-100's have been calibrated to provide corrected data with an uncertainty of 0.1°C . PIV measurements were performed with and without the intruding PT-100's. It was found that the sensors do not affect the upstream velocity field and that mutual interaction is minimal.

Temperature differences ΔT are defined as the average of 120 independent temperature measurements at a certain place, minus T_L . Error propagation has carefully been analyzed [14] which yielded a typical error in ΔT of $\pm 0.06 \text{ K}$.

Mass flow rate of steam has been given 3 values: 500, 1,000 and 1,500 g/h, corresponding to fluxes, G , of 44, 88 and $132 \text{ kg}/(\text{m}^2\text{s})$, respectively (steam injection diameter is 2 mm). Approach temperature T_L has been given the values 25, 65, 75°C . Volume flow rate of the water, Q_L , has been varied (0.48, 0.95, $1.41 \text{ m}^3/\text{h}$ through area $30 \times 30 \text{ mm}^2$) such that three momentum ratios, J , were measured: $J = 15$, $J = 57$ and $J = 125$. Here,

$$J = \rho_V U_V^2 / (\rho_L U_L^2) \quad (1)$$

with ρ_V the mass density of the injected steam, ρ_L that of the approaching liquid and U_V and U_L are the corresponding mean velocities. The velocity ratio, r , is defined as the square root of J .

3 Experimental results

3.1 General observations

Velocities and temperatures were such that steam injection was in the intermittent regime, see Fig. 3. A steam plume is formed at the inlet and it grows until the entire vapor pocket suddenly condenses totally. Thereafter, the growth of a new vapor plume commences. The closer the temperatures of inlet steam and approaching liquid are, the longer the maximum penetration depth of the steam is. In all conditions steam penetration was confined to a distance of 10–20 times the steam inlet radius, R_a . Away from this condensation area a single-phase liquid jet is created. It is in this jet that temperatures and velocities have been measured. The intermittency of the steam injection, which will be further analyzed below, causes temperature and velocity fluctuations that necessitate fast, instantaneous measurement techniques if fluctuations are to be recorded. For velocity fluctuations Particle Image Velocimetry (PIV) could be applied but only time-averaged temperatures have been measured.

A schematic of the single-phase jet measured is given in Fig. 4. A jet centerline can be defined as the places where the magnitude of the velocity is highest or the places where the temperature is highest. This paper focuses on temperatures but will also compare positions of the temperature centerline with that of the velocity centerline. The jet velocity centerline as depicted in Fig. 4 is the coordinate axis ξ of a curvilinear coordinate system (ξ, η).

Fig. 3 Examples of injected steam bubble shapes at two arbitrary instants of time (LHS). The bubble grows until sudden total collapse occurs. Steam plume predictions with VOF and Fluent™ are magnified on the RHS

Fig. 4 Schematic of Cartesian coordinate system and jet center line with coordinate ξ along the jet axis

Temperature difference ΔT , defined above, decreases with increasing coordinate ξ (Fig. 4) and generally decreases with increasing distance to the centerline. Quantifications and an interesting exception will be detailed below. Temperature difference ΔT generally decreases with increasing approaching liquid temperature, as expected. The higher the momentum ratio, J , is, the larger the penetration depth. For a constant J of 55 the temperature field has been measured for steam inlet mass flow rates of 500, 1,000 and 1,500 g/h. The temperature fields were found to be practically the same. This will be further investigated below.

3.2 Centerline values

Each temperature sensor is translated along a line at an angle of 40° with the horizontal. The location of the highest temperature, determined with an interpolation spline, is at the temperature center line. Figure 5 shows typical results for three momentum ratios and a liquid approach temperature of 65°C . The errors indicated are safe 95% probability errors and account for interpolation, positioning inaccuracy and distance of ΔT -variation in the jet corresponding to the inaccuracy of 0.06 K in ΔT .

Figure 5 clearly shows that a higher J corresponds to a steam penetration further away from the inlet and that centerlines of a certain J -value coincide within measurement accuracy.

Fig. 5 Jet temperature center line positions, with error bars, for three momentum ratios and for $T_L = 65^\circ\text{C}$. Dotted lines for $J = 57$ are given to guide the eye, as are the solid lines for $J = 15$ and the dashed ones for $J = 125$

Fig. 6 Jet temperature center line positions in dimensionless coordinates, with error bars, for three momentum ratios and for $T_L = 65^\circ\text{C}$ (red) and $T_L = 25^\circ\text{C}$ (blue); $d = 2R_d$; $r = \sqrt{J}$

Figure 6 shows that scaling the axis with $rd = 2R_d\sqrt{J}$ causes all centerlines to practically coincide for the dashed $J = 125$ -values. The correlation

$$x/(rd) = 1.3\{y/(rd)\}^{0.3} \quad (2)$$

contains coefficients 1.3 and 0.3 which have been determined by least-square fitting. The values for single-phase jets found by Smith and Mungal [12], 1.5 and 0.27, respectively, are that close that differences can be attributed to differences in geometries of steam inlet and channel. The single phase crossflow jets are usually a free crossflow while in the present research the channel is confined to $3 \times 3 \text{ cm}^2$.

The temperature decrease along the centerline for all measurements is given in Fig. 7. The lines on top are for the lower liquid approach temperature, 25°C , and the lower ones for 65°C . Another trend which is obvious is that at constant momentum ratio, for the solid lines for example,

Fig. 7 Average value of $(T - T_L)$ for 120 measurements at the centerline for $T_L = 65^\circ\text{C}$ (red) and $T_L = 25^\circ\text{C}$ (blue). Lines for n -powers $-2/3$ and -1.3 are given to guide the eye, as are the other lines

Fig. 8 Positions of velocity center lines (red, unconnected symbols) and temperature center lines (blue, connected triangles) for three momentum ratios; $T_L = 25^\circ\text{C}$ and $G = 0.88 \text{ g/(m}^2\text{s)}$

an increase in steam mass flow rate causes an increase in ΔT . An increase in J at constant steam mass flow rate also increases ΔT .

Smith and Mungal [12] find that concentrations in single-phase cross-flow jets decay proportional to $\xi^{-1.3}$ in the so called near-field zone and proportional to $\xi^{-2/3}$ in the far-field zone. The findings of Smith and Mungal are also depicted in Fig. 7. The present findings appear to lie mostly in the far-field zone. Agreement with the transition criterion of Smith and Mungal for the two regimes, $\xi = 0.3 r^2 d = 0.6 R_d J$, is good.

The positions of the temperature center lines are compared with the velocity center lines, measured with PIV, in Fig. 8. Details about these PIV measurements are given by Clerx [5]. Figure 8 clearly shows that for $T_L = 25^\circ\text{C}$ the temperature center lines do not penetrate as deeply in the channel as the velocity center lines at momentum ratios less than 100. Kamotani and Greber [7] measured a similar trend for single-phase jets in free cross-flow. The present findings are further discussed below.

Fig. 9 Time-averaged temperature fields measured at the *black dots* and interpolated. Each *horizontal axis* is wall-normal *x*-position. Each *vertical axis* is *y*-position. *Top row* is for momentum ratio $J = 15$, *middle row* for $J = 55$ and *bottom row* for $J = 125$. Average values of $(T - T_L)$ for 120 measurements for $T_L = 25^\circ\text{C}$ are indicated by *colors* according to the *color bar* in K

3.3 Temperature fields

Figures 9, 10, 11 show temperature fields measured for steam mass fluxes, G , of $44 \text{ kg}/(\text{m}^2\text{s})$ (each left column), $88 \text{ kg}/(\text{m}^2\text{s})$ (each middle column) and $132 \text{ kg}/(\text{m}^2\text{s})$ (each column on the right).

As expected, a high-temperature zone occurs near the center of the velocity jet, see Fig. 9 for a liquid approach temperature of 25°C . The position of this zone is coupled to the position of the centerline, of course, and all trends regarding these positions have been discussed in the previous section.

An interesting property of the temperature field is revealed by Figs. 10 and 11, for approaching liquid temperatures of 65 and 75°C , respectively. It is the occurrence of a high-temperature zone above the steam inlet near the wall, on the LHS of each plot in these figures. This zone will be called ‘hot wall plume’ to distinguish it from the normal hot zones near the center lines of Fig. 9.

4 Discussion

The hot wall plume is probably the wake area of the hot ‘cylinder’ of steam that is present most of the time in the near wall area. Steam penetration is intermittent, so maximum penetration is reached less frequently than small

steam penetrations. Under some process conditions a small steam bubble is even always visible at the steam inlet. Heat transfer from the steam pocket near the wall is on a time-averaged basis apparently that efficient that a hot wall plume is established. With decreasing temperature difference between inlet steam and approaching liquid the maximum penetration depth of steam increases. This has a clear effect on the position of the jet centerline, as shown in Figs. 9, 10, 11 and of course in Figs. 6 and 7. The position of the hot wall plume, on the contrary, is independent of this temperature difference, see Figs. 9, 10, 11. Comparison of these figures shows that the strength of the hot wall plume depends on the temperature difference between approaching liquid and injected steam. The hot wall plume is in a wake zone with low velocities due to the presence of the wall. If the steam would be injected through a slit, planar 2D computations show that the entire downstream volume of the jet wake is at high temperature. With the actual circular jet, a hot wall plume is created with in first order approximation a linear temperature profile which is high at the wall, say at temperature T_w , and which is at T_L further away from the wall. The higher $(T_v - T_L)$ is, the higher is $(T_w - T_L)$. More research is required to fully explain the occurrence and characteristics of the hot wall plume.

The error bars in Figs. 6 and 8 are conservative, so the observed trends in position of the center line are

Fig. 10 Time-averaged temperature fields measured at the *black dots* and interpolated. *Left column* is for $G = 4.4$, *middle* for $G = 8.8$ and *right column* for $G = 13.2 \text{ g/(m}^2\text{s)}$. See caption Fig. 9 for legends of *axis* and *rows*. Average values of ΔT for $T_L = 65^\circ\text{C}$ are indicated by *colors* according to the *color bar* in K

Fig. 11 Time-averaged temperature fields measured at the *black dots* and interpolated. See caption Fig. 9 for legends of *axis* and *rows*, and caption Fig. 10 for legend of *columns*. Average values of ΔT for $T_L = 75^\circ\text{C}$ are indicated by *colors* according to the *color bar*

unmistakable. The differences in velocity and temperature center lines are interesting but difficult to explain. We will show in another publication that self-similarity in the velocity field of the jet exists, if the approaching velocity field is subtracted and the curved centerline is employed as a reference. The spreading rate, a constant, is related to the turbulent viscosity which is therefore also practically constant [15]. The influence of molecular thermal diffusivity and the Prandtl number on scalar transport in a self-similar circular turbulent jet is discussed by Pope [11], p. 165). The thermal diffusivity, α , is defined as $k/(\rho C_p)$, the thermal Péclet number, Pe_T , is defined as LV/α with V the centerline velocity, the Prandtl number is ν/α and the Reynolds number, Re , is based on V and on the half-width of the jet. In water at Prandtl numbers exceeding 1, temperature transport is affected by α if Re is less than 10,000 [11]. In our experiments, Pr is about 6.3 at 25°C and about 2.8 at 65°C while the jet Reynolds number is less than 10,000. It is noted that intermittency is high in the jet due to the intermittent steam injection and that conditional sampling to discriminate between the turbulent flow of the approaching liquid and the actual induced jet flow would yield other results. Present values are total-time-averages and indicate that Pr , Pe_T and Re might be relevant for the temperature dependencies of jet characteristics.

That turbulent diffusion of a scalar (temperature) in a jet may depend on T_L does not directly explain the different positions of the velocity and temperature centerlines. There is nevertheless a striking relation between these positions and the corresponding Pr -values. The closer Pr is to 1, the closer the values of Pe_T and Re and the closer the velocity and temperature centerlines are expected to lie. The high value of Pr in Fig. 8 (about 6.3) and the different Reynolds numbers might explain the different positions of the velocity and temperature centerlines at the lower J -numbers. The lower value of Pr at 65°C (2.8) and the different Reynolds numbers might explain the shift away from the injection wall of the temperature centerlines in Fig. 6 for the lower J -numbers. The velocity center lines practically do not move so the temperature and velocity center lines come closer by the increase of T_L from 25 to 65°C. The connection between Pr and centerline positioning is reminiscent to the connection between Pr and heat transfer from a heated cylinder to a cross-flow of water. Many correlations describe the latter connection with a factor $Pr^{1/3}$. The downstream temperature wake of a heated cylinder is enlarged if J is decreased by increasing Q_L , as is done here. This extension would explain the finding that the biggest differences in position of the velocity and temperature center lines occur at lowest J -value. Despite the strongly intermittent condensation and steam penetration process, heat and flow characteristics of the jet are therefore reminiscent of those of the wake downstream of a heated

cylinder. The counter-rotating vortex pair found by Smith and Mungal [12] for a steam jet enhances the similarity of steam jet in a cross-flow and wake of a heated cylinder.

5 Prediction of steam injection

It is nowadays customary to model unit operations with CFD packages, even if phase transition occurs and if multiple scales are present in the flow. Such modeling may aid our understanding of the physical phenomena involved, but with a proper understanding of these phenomena it should be possible to capture the physics in mechanistic modeling based on a simplified set of governing equations. Design rules, rules of thumb and correlations for practical use are usually based on such mechanistic models. In the following, a mechanistic model for intermittent steam penetration is shortly described and some of its main assumptions, or simplifications, are validated with the aid of CFD computations.

The flow regime of intermittent steam injection resembles bubble injection through a needle and vapor bubble generation at a cavity. High-speed recordings as those of Figs. 3 and 12 yield bubble shapes with numerous small scale deformations but with a global shape which is close to those observed in bubble growth and detachment. Only the final stage is in the case of steam injection controlled by heat transfer: the vapor pocket suddenly coincides and disappears. It is instructive to try to understand why this does not happen in an earlier stage and it is useful to try to model earlier growth with simplified physics. In the case of air bubble injection such modeling is usually based on a momentum balance normal to the wall. In the case of steam injection this can also be done, as shown below, but velocities are higher. This implies that the rate of momentum release at the inlet of the steam is considerable as compared to air injection. This momentum release will be seen to make the bubble growth process that fast that inertia forces are controlling the growth. Thus, the main physical phenomenon which should be treated carefully in the bubble growth model is inertia. As is the case in air bubble injection, the momentum balance is valid at all times during bubble growth. In order to predict detachment-collapse of a bubble an additional criterion is needed. In the case of air bubble injection such a criterion is usually a geometrical criterion based on empirical findings of bubble shapes at detachment; see for example Brucato [2]. In the case of intermittent steam injection, an empirical correlation via the Jacob number related to heat transfer can be provided [5]. This criterion as well as a detailed description of the model will be published shortly elsewhere. Here, a general description of the governing momentum balance, including the main assumptions is

Fig. 12 Comparison of measurement and prediction of vapor plume collapse. Asymmetry is not observed in the experiment because of the low approach velocity of 0.15 m/s, the small volume (each figure measures $8 \times 8 \text{ mm}^2$) and the short time of growth (ms-scale, see times indicated)

given, followed by some results of 3D CFD computation to validate these assumptions.

With the mass density of the liquid, ρ_L , a constant, the following equation for the velocity component normal to the wall of the vapor pocket center, U , is derived:

$$\begin{aligned} (C_{am,1}\rho_L \nabla) \frac{dU}{dt} = & (-C_{am,1}U - C_{am,2}/2)\rho_L d\nabla/dt \\ & - C_{am,2}\rho_L \nabla d/dt/2 - 32\pi\mu_L RU \\ & + 20\pi\mu_L R + G^2\pi R_a^2/\rho_v - F_{cond} \end{aligned} \quad (3)$$

The last term on the RHS of (2), F_{cond} , accounts for anisotropic momentum loss due to condensation and is found to be negligible. Also drag, in (2) overestimated with $32\pi\mu_L RU$, is found to be negligible; R is instantaneous radius of the bubble which has time rate of change $\dot{R}$, μ_L is the dynamic viscosity of the liquid. Main contributions in (2) stem from inertia terms proportional to instantaneous volume of the bubble, ∇ , or to its time rate of change, $d\nabla/dt$. The evaluation of the instantaneous added mass coefficients, $C_{am,1}$ and $C_{am,2}$, is therefore important; the values for spheres and truncated spheres close to a plane wall given by van der Geld [13] have been adopted here. Last but not least, injected steam momentum is accounted for by the term $G^2\pi R_a^2/\rho_v$.

Equation (2) is supplemented with the equation $dh/dt = U$ for distance h of the center point to the wall, and both equations are integrated in time with a second order Adams–Bashforth method. This is the only numerical step involved. Penetration depths and times of growth predicted with this set of equations have been compared favorably with measurements [6].

An important assumption of the above model is that capillary forces are balanced by a higher pressure inside the vapor plume and do not play an essential role in steam

penetration. This assumption has been validated by simulations with the VOF method in Fluent^{TM} in two ways: a study of predicted and measured steam plume shapes and variation of the surface tension coefficient. Figure 12 shows that histories of interface topologies can be reproduced by such simulations reasonably well. The intermittency of steam penetration as revealed by the penetration depth was clearly observed with time-dependent axisymmetric simulations, both with laminar flow and with the RANS method to model turbulence. Also full 3D, i.e. not axisymmetric, simulations exhibited the intermittency of steam intrusion; see the example of Fig. 1. More details about numerical approach as well as more simulation results are given by Liew [8] and Pecenko [10].

The simulations of Fig. 12 show that the foot of the plume moves independently of the remainder of the gas–liquid interface. This is a clear indication that the capillary forces of the foot are not constraining motion of the vapor–liquid interface. Further evidence of the unimportance of capillarity was obtained by raising the value of the surface tension coefficient, σ , by a factor 3. The resulting variation in intermittency in plume penetration was negligible while the maximum penetration depth decreased by only 5%. The reason for this reduction is of course the increasing tendency to maintain a spherical shape with increasing σ .

It is concluded that the assumption of unimportance of capillary forces in the momentum balance (2) is justified.

6 Conclusions

Steam has been injected in a cross-flow of water in a channel of $30 \times 30 \text{ mm}^2$. Steam injection was found to be intermittent, implying a continuously alternating sequence of growth and collapse of the injected steam plume. The

present study focused on mean temperature fields in relation to interface topology, temperature of the approaching steam, T_L , momentum ratio, J , and Prandtl number, Pr . Local temperature measurements were done by moving three intrusive thermocouples to various positions downstream of the steam injection point. The different positions of velocity and temperature center lines for $J < 100$ and $T_L = 25^\circ\text{C}$ have been explained with dependencies on Reynolds and Prandtl numbers.

Although the expectation was that highest temperatures would occur merely in the liquid jet induced by the steam flow, this turned out not to be true. In the vicinity of the wall and downstream of the steam injection point a second high temperature zone has been measured.

Since the momentum balance of steam injection is controlled by inertia, a simple mechanistic model for the prediction of steam penetration depths and corresponding growth times could be derived. The success of this model has been substantiated in this paper by proving the validity of a simplifying assumption with the aid of 3D VOF computations with Fluent^{TM}.

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References

1. Aya I, Nariai H (1987) Boundaries between regimes of pressure oscillation induced by steam condensation in pressure suppression containment. *Nucl Eng Des* 99:31–40
2. Brucato A (2008) Bubble formation at single nozzles. In: Proceedings of 11th international conference on multiphase flow in industrial plants, #455
3. Chan CK, Lee CKB (1982) A regime map for direct contact condensation. *Int J Multiph Flow* 8(1):11–20
4. Chen LD, Faeth GM (1982) Condensation of submerged vapor jets in subcooled liquids. *Trans ASME* 104:774–780
5. Clerx N (2010) Experimental study of direct contact condensation of steam in turbulent duct flow. Ph.D. Dissertation, ISBN: 978-90-386-2370-2, Eindhoven University of Technology, The Netherlands
6. Clerx N, van der Geld CWM (2009) Experimental and analytical study of intermittency in direct contact condensation of steam in a cross-flow of water. In: Passos J et al. (ed) Proceedings of ECI international conference on boiling heat transfer, Florianopolis, Brazil, pp 1–8, on CD
7. Kamotani Y, Greber I (1972) Experiments on a turbulent jet in a cross flow. *AIAA J* 10:1425–1429
8. Liew R (2009) Phosphor thermometry and simulation of direct steam injection. Master Science Dissertation, Report WPC200903, Eindhoven University of Technology
9. Nariai H, Aya I (1986) Fluid and pressure oscillations occurring at direct contact condensation of steam flow with cold water. *Nucl Eng Des* 95:35–45
10. Pecenko A (2010) Numerical simulation methods for phase-transitional flow. Ph.D. Dissertation, Eindhoven University of Technology, The Netherlands
11. Pope SB (2000) Turbulent flows. Cambridge University Press, Cambridge
12. Smith SH, Mungal MG (1998) Mixing, structure and scaling of the jet in crossflow. *J Fluid Mech* 357:83–122
13. van der Geld CWM (2009) The dynamics of a boiling bubble before and after detachment. *Heat Mass Transf* 45:831–846
14. van Deurzen LGM (2010) Temperature measurements on steam injection in cross-flow and a non-isothermal diffuse-interface model for two-phase flows. Master Science Dissertation, Report WPC201006, Eindhoven University of Technology
15. van Wissen RJE, Schreel KRAM, Van der Geld CWM (2005) Particle image velocimetry measurements of a steam-driven confined turbulent water jet. *J Fluid Mech* 530:353–368
16. Weimer JC, Faeth GM, Olsen DR (1973) Penetration of vapor jets submerged in subcooled liquids. *AIChE J* 19(3):552–558
17. Youn DH, Ko KB, Lee YY, Kim MH, Bae YY, Park JK (2003) The direct contact condensation of steam in a pool at low mass flux. *J Nucl Sci Technol* 40(10):881–885