Turbulent stresses in a direct contact condensation jet in cross-flow in a duct with implications for particle break-up

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A B S T R A C T

An experimental study has been conducted to investigate the turbulent mixing and heating caused by a (superheated) steam jet injected into a turbulent cross-flow of water in a square channel. The velocity field in the mid plane of the channel has been measured by means of particle image velocimetry for several different values of the ratio of the momentum fluxes of steam and water and various bulk temperatures of the approaching water flow. Condensation is rapid and the single phase jet created is strong, turbulent and with a self-similar velocity profile. The focus of the present paper is an analysis of the three components of the Reynolds stress tensor in a curvilinear coordinate system aligned with the curved centerline of the single-phase jet downstream of the condensation region. It is found that both measured diagonal components of the Reynolds stress tensor exhibit a maximum value at the jet centerline. Scaling laws for the decay of the turbulence intensities along the centerline have been formulated. Moreover, consequences for the break-up of particles in this flow are discussed and compared with the case of a steam jet injected into a stagnant fluid.

1. Introduction

Jets in cross-flow can be found in a wide variety of industrial applications, like pipe tee mixers. In a number of those applications condensing jets of a saturated or superheated vapor are injected into a flowing liquid. Such condensing jets in cross-flow can be applied for heating purposes when also a high mixing rate is needed. A turbulent jet that is injected normal to a cross-flow is an example of a free turbulent shear flow. It is inherently more complex than jets entering a quiescent medium, also referred to as free turbulent jets. This complex nature is exemplified by the intensive interaction between the jet and the cross-flow and several types of vortical structures that arise at various locations in the flow field. Visualization studies performed by Fric and Roshko [1] and Kelso et al. [2] give a profound insight in the dominant vortical structures and separation regions appearing in the near and far-field regions of the jet. Smith and Mungal [3] conducted an extensive set of concentration measurements and related the scalar mixing to the vortex structures occurring in the flow field. Experimental investigations that are of more relevance to the present work deal with velocity and temperature distributions of jets in cross-flow. The earliest of these studies focused on the mean centerline trajectories of jets and the evolution of the flow along these trajectories (Keffer and Baines [4]; Pratte and Baines [5]; Kamotani and Greber [6]). To facilitate scaling, Keffer and Baines [4] introduced what can be considered as ‘natural’ coordinates, with a streamwise axis along the centerline trajectory and a spanwise axis perpendicular to the centerline. Kamotani and Greber [6] conducted similar experimental work, but explored flow regions further downstream as well as heated jets in cross-flow. Trajectories of the centerline based on the maximum jet temperature appeared to penetrate less far into the cross-flow than trajectories based on the maximum jet velocity. Kamotani and Greber [6] also studied the spanwise temperature profiles along the jet centerline, in the same natural coordinate system as Keffer and Baines [4]. Based on a similarity theory with intermediate asymptotic behavior of the jet, Hassellbrink and Mungal [7] derived scaling laws for the centerline position, centerline velocities and scalar concentration, for both the near-field and the far-field region of the jet. The scaling laws were verified by velocity and concentration measurements, using particle image velocimetry (PIV) and laser-induced fluorescence [8].

Studies on steam injection found in the literature deal with steam jets injected in a quiescent pool of liquid. The main focus of these studies was to obtain expressions for the condensing steam jet length and the mean steam-water heat transfer coefficient (Weimer et al. [9]; Kerney et al. [10]; Chen and Faeth, [11]).

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The only investigation of far-field properties of such jets appears to have been carried out in our laboratory by van Wissen et al. [12]. In that work, PIV measurements downstream the condensation zone yielded an axisymmetric single-phase jet with self-similarity properties similar to that of a non-condensing free turbulent jet of identical jet Reynolds number (defined in the usual way for a jet, see van Wissen et al. [12]).

Until recently, condensing steam jets issuing into a confined liquid cross-flow were not studied yet. In the present research an experimental study has been conducted by means of PIV measurements in the region downstream of a steam injection point in a duct with a square cross section to investigate the turbulent mixing and heating phenomena induced by the condensation of steam in a cross-flow of water. The velocity profile downstream of the limited condensation region is self-similar (Clerx, [13]). This paper presents experimental results of turbulent stresses and scaling analysis of such stresses at the centerline of the confined liquid jet. In mixing applications the stresses exerted by the fluid on small-sized particles play an important role. Since Hinzé [14], the break-up of such particles has been related to turbulent stresses and turbulent dissipation in particular. As for example Hussein et al. [15] showed, away from the origin of turbulence turbulent dissipation is equal to turbulence production. Turbulence production and turbulent stresses have therefore been measured for various mass flow rates of water and steam and for various bulk temperatures, as an extension of the work of van Wissen et al. [13,16]. In the present cross-flow, particles have a finite residence time in the jet. The implications of this finite residence time for particle break-up are evaluated.

2. Experimental

2.1. Test rig

The experimental set-up, shown in Fig. 1, is a pressurized flow loop of demineralized water. The flow is driven by a frequency controlled centrifugal pump. An ultrasonic flow meter (accuracy: 0.25 % of the full scale range which is 3.24 cubic meters per hour) measures the volumetric flow rate. The closed loop can be pressurized up to 0.8 MPa via a membrane connection to a pressurized air supply. Four calibrated Pt-100 elements (accuracy: 0.1 °C) monitor the water temperature. A PID-actuated bleed valve that is connected to a pressure transducer (accuracy: 0.1% full scale which is 0.7 MPa) controls pressure. The water temperature is kept constant during steam injection with the aid of a heat exchanger and a 17 kW electric heater whose output power is controlled by a PID-actuated solenoid state relay. The whole set-up is thermally insulated with a 20 mm thick foam layer. The system pressure and the water temperature are constant during the measurements even though steam is being injected at a constant flow rate (Fig. 2).

The measurement section, indicated in grey in Fig. 1, has a square inner cross-section of 30 × 30 mm² and is optically accessible at the location where steam is injected. Before arriving at the transparent walls, the water flows through a channel with a length of 1200 mm (40 times the hydraulic diameter D_h) to obtain fully developed turbulent flow at the steam injection point. The steam is injected through a flush mounted wall injector with a circular hole with a diameter of 2 mm. The amount of injected steam is measured by a Coriolis mass flow meter (accuracy: 1 % of measured value) and controlled by a PID-actuated pneumatic valve. At 150 mm upstream of the steam injection point, a pressure transducer (accuracy: 0.1 % full scale range of 1 MPa) and a Pt-100 element monitor the inlet conditions of the steam.

An important experimental parameter is the ratio of momentum fluxes of injected steam and that of the approaching liquid flow, J. It is defined by

\[ J = \frac{\rho_s u_s^2}{(\rho_l u_l^2)} \] (1)

with \( \rho_s \) and \( \rho_l \) the mass densities of steam and water, respectively. Horizontal and vertical velocity components are denoted by \( u \) and \( v \) respectively. The steam velocity is denoted by \( u_s \) and \( v_b \) is the approaching water bulk velocity which is the time-averaged component in vertical direction; components of the water flow in a cross-section, so-called secondary motion, are at most 5% of \( v_b \). The steam momentum flux comprises the mass density of steam at temperature \( T_s \) and pressure \( p_s \) measured directly upstream of the injection point. The steam velocity \( u_s \) is calculated from the measured steam mass flux \( \dot{G} \), the mass density \( \rho_s \) and the area of the injector (π × 0.002²/4 m²). The mass density of water in (1) corresponds to the measured loop pressure \( p_l \) and temperature of the water measured at the inlet of the measurement section. The bulk velocity \( v_b \) is determined by dividing the measured volumetric flow rate \( \dot{Q_b} \) by the cross-sectional area of the duct ((0.03)² m²). The Reynolds number \( Re_b \), based on \( v_b \) and \( D_h = 0.03 \) m, is varied between

Nomenclature

\[ \begin{align*}
D_h & \text{ hydraulic diameter, m} \\
D_p & \text{ particle diameter, m} \\
F & \text{ F-statistic, } \sim \\
G & \text{ steam mass flux, kg/m²s} \\
M & \text{ magnification factor, } \sim \\
P_k & \text{ turbulent production, m²/s³} \\
Re_b & \text{ Reynolds number based on } \nu_b \text{ and } D_h, \sim \\
S & \text{ standard error, } \sim \\
T & \text{ temperature, °C} \\
We & \text{ Weber number, } \sim \\
d & \text{ steam nozzle diameter, m} \\
d_s & \text{ diffraction limited image diameter, m} \\
d_l & \text{ particle image diameter, m} \\
f & \text{ focal length of lens, m} \\
p & \text{ pressure, Pa} \\
r & \text{ effective velocity ratio, } \sim \\
r^2 & \text{ correlation coefficient, } \sim \\
u & \text{ lateral velocity (cartesian), m/s} \\
u_i & \text{ initial jet velocity, m/s} \\
u_{lm} & \text{ lateral local median velocity, pixel} \\
u_{rms} & \text{ lateral RMS-velocity (cartesian), m/s} \\
u'_x & \text{ lateral RMS-velocity (curvilinear), m/s} \\
u_r & \text{ steam velocity, m/s} \\
u' & \text{ streamwise velocity (cartesian), m/s} \\
u'' & \text{ streamwise RMS-velocity (curvilinear), m/s} \\
u_b & \text{ bulk velocity of liquid cross flow, m/s} \\
x & \text{ lateral coordinate (cartesian), m} \\
y & \text{ streamwise coordinate (cartesian), m} \\
\varepsilon & \text{ turbulent energy dissipation rate, m²/s³} \\
\eta & \text{ lateral coordinate (curvilinear), m} \\
\lambda & \text{ wave length, m} \\
\rho & \text{ mass density, kg/m³} \\
\sigma & \text{ surface tension, Pa m} \\
\zeta & \text{ streamwise coordinate (curvilinear), m} \\
\end{align*} \]
3,000 and 58,000. Measurements have been performed at Reynolds numbers between 10,000 and 30,000 which show [13] that the duct flow is fully developed near the steam injection point.

### 2.2. Experimental conditions

A total of 18 different process conditions have been measured: one series at a liquid bulk temperature of 25°C and one at 65°C. Each series comprises three essentially different momentum flux ratios \( J \). The absolute loop pressure at the steam injection point is about 28 MPa for all runs. The steam temperature (saturated or slightly superheated) is around 133°C. Typical measured histories of \( T_{L} \) and \( p_{L} \) are given in Fig. 2. For a period of at least 15 min of steam injection, the fluctuations in \( p_{L} \) are negligible while \( T_{L} \) varies within 0.2 °C.

The process conditions of selected runs for which experimental results will be given below are listed in Table 1. The tabulated values represent values time-averaged over a period of 3 min, the total time of measurement allocated to each experimental run. Within each run, four sets of 250 image pairs have been recorded with a video-camera at a frequency of 15 Hz, which is equivalent to an actual measuring time of 67 s. The way these recordings are made is described in the next section.

### 2.3. Optical technique

Particle image velocimetry is used to measure instantaneous velocity fields of the steam jet injected in the cross-flow. Extensive descriptions of PIV can be found in Raffel et al. [17] and Westerweel [18]. The measurements are carried out in the center plane of the duct. The center plane is illuminated by a frequency-doubled Nd:YAG laser (Spectra-Physics PIV-200) that generates two pulses of 200 mJ at 15 Hz. To obtain optimal cross-correlation of two subsequent PIV-recordings, the delay time between the two laser pulses is chosen such that the mean displacement of the particles is around 8 pixels, which is one-fourth of the interrogation window size (Keane and Adrian, [19]). This limit is applied to the displacements of particles inside the jet, because there the velocity is highest. An HR 532/45° mirror directs the laser beam towards the measurement section. With a positive cylindrical lens a laser sheet with a thickness of 1 mm is created which is subsequently stretched and parallelized by a negative cylindrical lens and a positive cylindrical lens, resulting in a vertical laser sheet with a height of 60 mm.

Recordings are made with a Kodak Megaplus ES 1.0 CCD camera with a resolution of 1008 × 1018 pixels and a dynamic range of 10 bit. A Nikkor AF 50 mm f/1.4 D lens is focused at an area of 38 × 39.9 mm². The fluid is seeded with spherical fluorescent particles with a mean diameter of 30 μm. The seeding particles are injected in the circulated water which creates the cross flow in the duct. They remain there until they are filtered out of the fluid. The particles are of a melamine resin based polymer coated with Rhodamine B and have a mass density of 1500 kg/m³. The seeding concentration yields at least 20 particle images per interrogation window, which satisfies the particle density criterion for accurate cross-correlation as defined by Keane and Adrian [19].

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![Fig. 1. Schematic of the set-up and photograph of test section. Inner cross-section AA is a square of 30 × 30 mm².](image1)

![Fig. 2. Typical histories of loop pressure, \( p_{L} \), and liquid temperature, \( T_{L} \), measured at the entrance of the measurement section.](image2)

<table>
<thead>
<tr>
<th>Run number</th>
<th>( u_{b} ) (m/s)</th>
<th>( u_{v} ) (m/s)</th>
<th>( J ) [-]</th>
<th>( r ) [-]</th>
<th>( Reb ) [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>J13</td>
<td>0.29</td>
<td>26.41</td>
<td>13.0</td>
<td>3.6</td>
<td>9816</td>
</tr>
<tr>
<td>J57</td>
<td>0.29</td>
<td>54.83</td>
<td>57.5</td>
<td>7.6</td>
<td>9854</td>
</tr>
<tr>
<td>J123</td>
<td>0.29</td>
<td>78.86</td>
<td>123.5</td>
<td>11.1</td>
<td>9818</td>
</tr>
</tbody>
</table>
The conversion of pixel-coordinates to physical coordinates is carried out by an in situ calibration. For this purpose a grid with small dots (diameter 0.3 mm) at a pitch of 2 mm is positioned in the center plane of the duct and photographed with the same camera in fixed position before PIV recordings. The grid is mounted in a square frame which enables positioning of the grid inside the duct with an accuracy of less than 0.1 mm. The calibration grid is also used to focus the camera lens.

The PIV-recordings are evaluated with the software package PIVview (version 2.4), developed by PivTec GmbH [20]. Each image is subdivided into interrogation windows of 32 × 32 pixels. An overlap over 50 % is chosen to increase the spatial resolution. The mean displacement vector in an interrogation window is estimated by applying a discretized cross-correlation function on the local intensity patterns. Cross-correlation is carried out by using a multi-pass interrogation method. With this method the interrogation of the image sample (interrogation window) is repeated at least once more, using a window offset in the following pass equal to the local integer displacement determined form the preceding pass. This method results in a higher signal-to-noise ratio due to a higher amount of matched particle images and also leads to a reduction of the uncertainty in the displacement (Westerweel et al., [21]). The location of the maximum correlation peak is detected with sub-pixel accuracy using a Gaussian peak fit.

2.4. Error analysis

Each instantaneous velocity field contains spurious vectors whose magnitude and direction differ significantly from their neighboring vectors. Validation of the raw displacement data is carried out by detecting the spurious vectors by means of local median filtering, as proposed by Westerweel [22]. With our settings, the filter compares a velocity component with the median values of particles. Fig. 3 also shows that spurious data have significant high-relation due to an in-plane loss of pairs and out-of-plane motion of the gas–liquid interface near the steam injection point. This is presumably caused by random motion of the gas–liquid interface near the steam injection point. This is presumably caused by random motion of the gas–liquid interface near the steam injection point.

It appears that spurious vectors are not evenly distributed over the displacement field. They are most likely to occur in the region close to the steam injection point. This is presumably caused by rapid motion of the gas–liquid interface near the steam injection point, which increases the chance of mismatches in the cross-correlation due to an in-plane loss of pairs and out-of-plane motion of particles. Fig. 3 also shows that spurious data have significant higher values of \(|v|/r\), typically exceeding 5 pixels, which makes it easy to distinguish them from correctly estimated displacements. A maximum allowable \(|v|/r\) of 4 pixels is found to result in proper detection of outliers in the major part of the observed domain. This maximum value varied from run to run. In the analysis of all experimental runs detected spurious vectors are eliminated without being replaced by an interpolated vector. Close to the steam injection point this results in an area with a large amount of unresolved displacements and with highly fluctuating velocity components. This ‘steam pocket’ region is determined for each experimental run separately, and a boundary designated, using the following criterion. All positions where more than 30% of the time (of the 1000 instantaneous measurements) a spurious vector was detected are part of the ‘steam pocket’ area. This region will be clearly indicated, by a dashed line, in contour plots of the average velocity field in the next section.

Strain rates are calculated with the mid-point difference, but the influence of measurements errors is decreased by a second order digital filter as proposed by Lanczos [23] and Hamming [24]:

\[
\frac{\partial u_j}{\partial x_i} \approx \frac{2u_{i,j+1} + u_{i,j+2} - u_{i,j-2} - 2u_{i,j-1}}{5(x_{j+1} - x_{j-1})}
\]

Errors in the instantaneously measured velocity components \(v(t)\) and \(u(t)\) can occur due to timing errors, calibration errors and errors in the estimation of the displacement vectors. Timing errors are typically smaller than \(10^{-6}\) s and calibration errors are smaller than 0.1 mm absolute and 0.01 mm relative and can be considered negligible. Errors in the estimation of the displacement vectors originate from the cross-correlation procedure. Possible causes are: smoothing of the displacement vector due to the size of the interrogation window, false peak detection due to a high noise level and inaccurate peak detection. In general, the estimation accuracy of the position of the correlation peak is in the order of 0.1 pixel. In some of the experiments, however, displacements have been found to be biased towards integer values; this is commonly referred to as ‘peak-locking’. This peak locking is caused by particle images having about the same size as a pixel. That this might occur is easily seen as follows. The image diameter, \(d_p\), of a particle with diameter \(d_p\) is estimated [20] from

\[
d_p \approx \left\{ M^2 d_p^2 + d_p^2 \right\}^{1/2}
\]

with \(M\) the image magnification factor and \(d_p\) the diffraction limited image diameter. According to PivTec [20]

\[
d_p \approx \frac{2.44(M + 1) f D}{\lambda},
\]

where \(\lambda\) is the wavelength of the light and \(f\) and \(D\) are the focal length and diameter of the lens. In some of the present experiments \(M = 0.24, d_p = 30 \mu m, \lambda = 532 nm, f/D = 5.6\) and \(d_p = 9.7 \mu m\). This results in a particle image diameter of 12.1 \(\mu m\) which is in only 30% larger than the pixel size (9 × 9 \(\mu m^2\)). The occurrence of peak-locking in these cases is confirmed by histograms of estimated particle displacements, as the example of Fig. 4. When particles can only appear as a single pixel on the CCD, the absolute error in the instanta-
neously measured velocity components corresponds to a localization error of about 0.5 pixel. However, time averaging will make these individual error estimates irrelevant, as will now be discussed.

In the following sections, only time-averaged values of local velocity components like the streamwise component $\overline{u}$ and mean turbulent intensities like the streamwise intensity $\overline{\nu^2}$ with $\nu' = \overline{u} - \overline{u}$, will be considered. The error of these time-averaged values is estimated without the above error estimates of instantaneous velocity components but with the aid of the so-called standard error. For a quantity which is measured $n$ times, with instantaneous results $o_i$ and mean $\overline{o}$, the standard error is given by

$$S_o = \left\{ \frac{\sum_{i=1}^{n} (o_i - \overline{o})^2}{n} \right\}^{1/2} \sqrt{n - 1}. \quad (4)$$

For local velocities, LES-computations [Clerx, [13]] show that the correlation time is small as compared to the time interval between subsequent velocity measurements taken at 15 Hz. This implies that all velocity data are independent and that the standard error in velocity components is given by (4) with $o$ replaced by component $\nu$ or component $\nu$. Similarly, the standard error in the mean streamwise turbulent intensity, or velocity fluctuation correlation, $\overline{\nu'\nu'}$, follows with $o$ replaced by $\nu'\nu'$, with $n$ equal to 1000.

Typical values for the standard errors are given in Fig. 5 in terms of error percentage relative to the magnitude of the local velocity. Outside the jet it is typically 0.2 to 0.3 % for the horizontal, spanwise $u$-component and 0.1 % more for the vertical, streamwise $\nu$-component. Inside the jet this percentage varies from 1.7 near the injection point to 0.5 % in the far-field. It is concluded that the measurement accuracy is sufficiently high to permit comparison of various process conditions.

3. Results

3.1. Mean velocity field

When hot steam is injected into a cross-flow of water, rapid condensation occurs in a small region near the steam injection point. The condensation regime in the present range of steam mass fluxes and approaching liquid temperatures is known to be intermittent [Clerx and van der Geld, [25]; Clerx et al. [26]]. A highly fluctuating flow in the vicinity of the steam injection point results (Fig. 6). A turbulent single-phase jet arises further downstream of the injection point. This jet is deflected by the approaching liquid flow and by the solid wall opposite to the steam injection point. The differences between the velocity fields of Fig. 6 prove that the correlation time of the velocity fluctuations in the jet is less than 1/15 s and that all velocity data of the jet are independent while the flow surrounding the liquid jet appears to be undisturbed.

Jet-cross-flow interaction is nicely shown in Fig. 7. The bulk Reynolds number is around 9850 and the momentum flux ratio is 57.5. In this plane in the center of the channel the jet fully penetrates the flow. Of course, the actual jet is 3D and might be thought of as axisymmetric about a curved axis through the core of the jet visible in Fig. 7.

Fig. 8 and similar figures (Clerx, [13, appendix C]) prove that the penetration depth merely depends on $J$. More details showing that...
the lateral turbulence is noticeably higher. Typical relationships between turbulence intensity components at the centerline, $v_{n0}$, are similar. The correlations $u_{n0}v_{n0}$ and $u_{n0}u_{n0}$ outside the jet region are in the range of 0.1 to 0.3, increasing with increasing $J$. Relative fluctuations inside the jet are decaying in streamwise direction of the jet.

3.2. Spatial distributions of stresses and turbulence intensities

Fig. 7. Vector field for run J57 averaged over 1000 measurements in time. Contour colors represent the velocity magnitude normalized with the bulk velocity (0.29 m/s). The dashed contour encloses the region where velocity data is unreliable. The number ‘2.2’ indicates the position of the maximum occurring value of $|v|/v_b$.

Fluctuations of the $u$ and $v$ velocity components for time-averaged vector fields such as those shown in Figs. 7 and 8 are given in Figs. 10 and 11. The contour plots display the RMS-values of both velocity components, scaled with the local velocity magnitude $|v|$. Both RMS-components are of the same size outside the jet region, while in the jet region $u_{rms}$ is noticeably higher. Typical relative values for $u_{rms}$ and $v_{rms}$ outside the jet region are in the range of 0.1 to 0.3, increasing with increasing $J$. Relative fluctuations inside the jet are decaying in streamwise direction of the jet.

The distributions of three components of the Reynolds stress tensor in the jet flow are determined by calculating the fluctuations of the velocity components $u_i$ and $v_i$ in the rotated frame, ($\xi, \eta$). Let $\overline{u_iu_i'}$ be defined as the ensemble average of $(u_i - u_i')$; the definition of $\overline{u_i'v_i'}$ and $\overline{u_i'u_i'}$ are similar. The correlations $\overline{u_i'u_i'}$, $\overline{u_i'v_i'}$ and $\overline{u_i'u_i}$ are scaled with the square of the streamwise centerline velocity, $v_{cl}$, analogous to the scaling performed in free turbulent jets (Pope, [27]; van Wissen et al., [12]). Note that $\overline{u_i'u_i}$ is the streamwise turbulence intensity and $\overline{u_i'v_i'}$ the lateral turbulence intensity.

3.2. Spatial distributions of stresses and turbulence intensities

The lateral profiles of the turbulence intensity components at successive streamwise coordinates for $J = 57.5$ (run J57) are presented in Fig. 12. The standard errors in the mean turbulence intensities, estimated with Eq. (4), are determined relative to the displayed values and are at maximum 10 % in case of $\overline{u_i'u_i}$ but up to 30 % in some cases of $\overline{u_i'v_i'}$. Turbulent intensities show a rapid decrease with increasing $\xi$. As compared to jets at the lower value of $J = 13.0$, the levels of turbulence appear to be significantly increased. Typical values for this increase are 60 % for the streamwise centerline fluctuation at $\xi = 12.3$ mm and 50 % for $\overline{u_i'u_i}$ at the
same streamwise coordinate. This increase was also observed by Kamotani and Greber [6], who measured the RMS-fluctuation of the streamwise velocity of a non-condensing jet in a plane parallel to the centerline of the jet. A second difference with respect to the jet at the lower momentum flux ratio is that the centerline values of the streamwise and lateral turbulence intensity components are almost of equal size at \( f = 57.53 \). More details are given by Clerx [13].

The centerline decrease of both the streamwise and lateral turbulence intensities are now compared with the centerline variation of the RMS-fluctuations for an air jet issuing into a cross-wind, as reported by Hasselbrink and Mungal [8]. In another article of Hasselbrink and Mungal [7], scaling laws based on a similarity theory with asymptotic behavior for the jet in cross-flow were derived. The streamwise variation of the lateral RMS-fluctuation (in \( x \)-direction) along the centerline in the near-field region (for \( x/(rd) \ll 1 \)), \( u_{rms}' \), of the jet was found to be given by

\[
u_{rms}'/u_j = (c_{uv}/c_{q})^{-1/3}(\rho_j/\rho_\infty)^{1/2}(x/d)^{-1}
\]  

with \( u_j \) the initial jet velocity, \( c_{uv}/c_{q} \) the ratio of profile and entrainment coefficients, \( \rho_j/\rho_\infty \) the mass density ratio of the jet fluid and cross-flow fluid and \( x/(rd) \) the ratio of the lateral distance to the nozzle and the nozzle diameter (see Hasselbrink and Mungal [7,8]). The relation for the far-field region (for \( x/(rd)^{1/2} \)) of the jet found by Hasselbrink and Mungal [8], is

\[
u_{rms}'/u_j = c_{uv}/(9c_{qw})^{-1/3}(\rho_j/\rho_\infty)^{1/2}(y/(rd))^{-2/3}
\]  

with \( c_{uv} \) a profile coefficient and \( c_{qw} \) the far-field entrainment coefficient. Typical values reported for the leading coefficients in the equations above are 1.35 in case of the near-field case and 0.4 for the far-field region. Hasselbrink and Mungal [8] give similar scaling laws for the streamwise RMS-fluctuations with only the leading coefficient being different.

For the present study, the streamwise and lateral RMS-fluctuations of \( v_n \) and \( u_n \) at the centerline of the jet are defined by

\[
u_n = \sqrt{(u_n'v_n')/(u_n)^2}
\]  

\[
u_n = \sqrt{(u_n'v_n')/(u_n)^2}
\]  

It is obvious that the above analysis is to be replaced by \( v_n' \) and \( u_n' \). The initial jet velocity \( u_j \) is replaced by \( v_n \), which is the streamwise centerline velocity of the jet in the \( (\xi, \eta) \)-coordinate system. The initial jet velocity, \( u_j \), used by Hasselbrink and Mungal [7] to scale the RMS-velocity, is of course constant along the centerline. The variation of \( v_n \) along the centerline is small (Clerx, [13]) and can therefore be used to scale the RMS-velocities \( v_n' \) and \( u_n' \).

Application of the aforementioned scaling laws to the RMS-fluctuations of the condensing jet in cross-flow, projected onto the \( (\xi, \eta) \)-axis, shows that both RMS-components of run 113 (\( J = 13.0 \) or \( r = 3.6 \)) satisfy the centerline decay relation for the far-field region given by Eq. 7:

\[
u_{rms}' = 1.72 \pm 0.01(y/(rd))^{-0.64 \pm 0.01}
\]  

\[
u_{rms}' = 1.43 \pm 0.01(y/(rd))^{-0.65 \pm 0.01}
\]  

The experimental values for \( v_n' \) and \( u_n' \) and the resulting fits are plotted in Fig. 15. It appears that the exponents for both components are close to \(-2/3\) but that the leading coefficients are larger than those reported by Hasselbrink and Mungal [8]. This difference can result from scaling the RMS-velocities with \( v_n \) and from different
entrainment rates. It is noted that the lateral component exceeds the streamwise component. This is in agreement with the observation made by Hasselbrink and Mungal [8] that $v_0^{\text{rms}} / \langle u \rangle \approx C_{\text{cud}}$, with $c$ equal to 0.7. For the present case $c$ was found to be 0.8. For run $J_{13}$, the observed $x/(rd)$-range is $1.4 < x/(rd) < 2.6$. According to the definition of the ‘far-field region’ given by Hasselbrink and Mungal [8], i.e., $x/(rd) > 1$, the observed range can be considered to be far-field indeed.

The centerline RMS-fluctuations of run $J_{57}$ ($J = 57.5$ or $r = 7.6$) is proportional to $(x/d)^{-1}$, hence obeys the scaling law for the near-field jet region given by Eq. (6):

$$u'_{\text{c}} = 3.63 \pm 0.49 (x/d)^{-0.97 \pm 0.06}$$

$$v'_{\text{c}} = 4.31 \pm 0.76 (x/d)^{-1.07 \pm 0.06}$$

Measured values of $v'_{\text{c}}$ and $u'_{\text{c}}$ with the resulting fit are shown in Fig. 14. In this case the leading coefficients are also larger than those for the single-phase jet given by Hasselbrink and Mungal [8]. Note that $v_0^{\text{rms}} / u_0^{\text{rms}}$, which is consistent with the observations made above. The observed $x/(rd)$-range for the jet of run $J_{57}$ is $0.99 < x/(rd) < 1.4$, which does not strictly comply with the near-field criterion $x/(rd) \ll 1$ of Hasselbrink and Mungal [8]. Apparently, the near-field definition of Hasselbrink and Mungal [8] does not apply precisely to the case of a confined condensing jet in cross-flow.

4. Prediction of particle disintegration, a generic approach

The jet initiated by direct-contact condensation in a channel studied in this paper is an efficient means of heating up mixtures. However, many mixtures in the process industry contain small particles, like starch, that have a chance to break up if exposed to high fluid stresses. The prediction of the limiting particle size in turbulent flows is usually based on the pioneering article by Hinze [14]. If particle sizes are in the inertial subrange,
Fig. 12. Lateral profiles of turbulence intensities for run J57 in (a) streamwise and (b) lateral direction and (c) cross-term of streamwise and lateral fluctuations at three streamwise positions $\zeta$. The dashed lines in sub-figure (c) are plotted to guide the eye.

Fig. 13. Lateral and streamwise RMS-fluctuations along jet centerline versus $y/(rd)$ for run J13 together with the far-field scaling law of Hasselbrink and Mungal [8] fitted to the new data. The fit uncertainties given are for a 95% confidence interval. The fit statistics for $u'_i$ are $r^2 = 0.99$ and $F = 4223$ and $r^2 = 0.99$ and $F = 4848$ for $v'_i$.

Fig. 14. Lateral and streamwise RMS-fluctuations along jet centerline versus $x/d$ for run J57 together with the near-field scaling law of Hasselbrink and Mungal [8] fitted to the new data. The fit uncertainties are for a 95% confidence interval. The fit statistics for $u'_i$ are $r^2 = 0.95$ and $F = 247$ and $r^2 = 0.94$ and $F = 176$ for $v'_i$. 
The par-...is the mass density of the continuous phase, \( \rho \) the particle diameter, \( \sigma \) surface tension coefficient and \( \epsilon \) turbulent dissipation. In regions away from the source of turbulence, \( \epsilon \) can be estimated from the turbulence production: \( \epsilon \approx \rho \nu \). The production of kinetic energy in the macro scales is given by

\[
P_k = \sum_i \sum_j u_i u_j \frac{\partial u_i}{\partial x_j}
\]

in a Cartesian coordinate system \( \{x_i\} \). Production (15) encompasses a major contribution of \( \nu \partial u_i / \partial x \). Typical turbulent intensity profiles are given in Fig. 12 and at the centerline by Eqs. (10)-(13), while mean velocity gradients are easily computed from Eq. (5). This gradient must be multiplied at each place with the local turbulent intensity (Fig. 12) to get the major contribution to the production. The resulting production value can be set equal to the dissipation rate because the other production contributions are negligible. It follows that turbulent dissipation is inhomogeneous both in streamwise and lateral direction. Figures similar to Fig. 13 for other process conditions show that turbulence production depends strongly on the turbulence level of the approaching cross-flow. Turbulence intensities increase with increasing momentum flux ratio. The centerline decay of the RMS of streamwise velocity fluctuations is for \( J = 13.0 \) and \( T_L = 25^\circ \text{C} \) found to satisfy the following scaling law (\( \gamma \) is height above the steam injection point and \( d \) the diameter of the steam inlet, 2 mm in the present study, see (12) and (13) for definitions of \( u_i^* \) and \( \nu \))

\[
\sqrt{\langle \nu_i^* \rangle} = 1.72 \pm 0.01 \gamma / (d \sqrt{J})^{0.64 \pm 0.01}
\]

\[
\sqrt{\langle \nu_i \rangle} = 1.43 \pm 0.01 \gamma / (d \sqrt{J})^{1.65 \pm 0.01}
\]

Similar scaling laws have been found to prevail for the lateral velocity fluctuations and for \( J = 57.5 \) and \( T_L = 25^\circ \text{C} \).

From data presented in the above Results section and the approach summarized in the preceding section the stresses exerted on a particle can be determined. As with steam injection in a stagnant pool of liquid (Van Wissen et al., [12]), particle break-up is most probable in a small flow region close to the nozzle but there are two important differences. Due to the approaching liquid flow the particle residence time is limited and the response time of the particle becomes important. In addition, turbulence production and stresses in the present cross-flow jet are inhomogeneous in axial direction, as shown for example by the above correlations. This limits the chance of break-up even further.

5. Conclusions

When steam is injected in a turbulent duct flow of water with a temperature difference of at least several tens of degrees (here \( 70^\circ \text{C} - 100^\circ \text{C} \)), condensation occurs in a small region near the steam injection point. Depending on the injection flow rate of steam the topology of the steam possesses a certain degree of intermittency. A turbulent single-phase jet arises further downstream that is deflected under the action of the cross-flow. The ratio of injected steam momentum to cross-flow momentum, \( J \), is found to largely govern the resulting flow field and self-similarity in the flow field has been demonstrated.

Turbulence intensity profiles have been investigated in the ‘natural’ frame of reference for two momentum flux ratios. The distributions in lateral direction show that both the RMS-values of streamwise and lateral velocity fluctuations exhibit maximum values at the centerline of the jet. The corresponding intensities decrease rapidly in lateral direction to the turbulence level of the approaching cross-flow. Turbulence intensities increase with increasing momentum flux ratio. The centerline decay of the RMS of streamwise velocity fluctuations is for \( J = 13.0 \) and \( T_L = 25^\circ \text{C} \) found to satisfy the following scaling law \( \gamma \) is height above the steam injection point and \( d \) the diameter of the steam inlet, 2 mm in the present study, see (12) and (13) for definitions of \( u_i^* \) and \( \nu \)).

\[
\sqrt{\langle \nu_i^* \rangle} = 1.72 \pm 0.01 \gamma / (d \sqrt{J})^{0.64 \pm 0.01}
\]

\[
\sqrt{\langle \nu_i \rangle} = 1.43 \pm 0.01 \gamma / (d \sqrt{J})^{1.65 \pm 0.01}
\]

Similar scaling laws have been found to prevail for the lateral velocity fluctuations and for \( J = 57.5 \) and \( T_L = 25^\circ \text{C} \). From data presented in the above Results section and the approach summarized in the preceding section the stresses exerted on a particle can be determined. As with steam injection in a stagnant pool of liquid (Van Wissen et al., [12]), particle break-up is most probable in a small flow region close to the nozzle but there are two important differences. Due to the approaching liquid flow the particle residence time is limited and the response time of the particle becomes important. In addition, turbulence production and stresses in the present cross-flow jet are inhomogeneous in axial direction, as shown for example by the above correlations. This limits the chance of break-up even further.

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References