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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, Hasselbrink 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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Nomenclature

D_h	hydraulic diameter, m	u_{lm}	lateral local median velocity, pixel
D_p	particle diameter, m	u_{rms}	lateral RMS-velocity (cartesian), m/s
F	F-statistic, –	u'_ξ	lateral RMS-velocity (curvilinear), m/s
G	steam mass flux, kg/m^2 s	u_v	steam velocity, m/s
M	magnification factor, –	v	streamwise velocity (cartesian), m/s
P_k	turbulent production, m^2/s^3	v_{rms}	streamwise RMS-velocity (cartesian), m/s
Re_b	Reynolds number based on v_b and D_h, –	v'_ξ	streamwise RMS-velocity (curvilinear), m/s
S	standard error, –	v_b	bulk velocity of liquid cross flow, m/s
T	temperature, °C	x	lateral coordinate (cartesian), m
We	Weber number, –	y	streamwise coordinate (cartesian), m
d	steam nozzle diameter, m	ε	turbulent energy dissipation rate, m^2/s^3
d_s	diffraction limited image diameter, m	η	lateral coordinate (curvilinear), m
d_τ	particle image diameter, m	λ	wave length, m
f	focal length of lens, m	ρ	mass density, kg/m^3
p	pressure, Pa	σ	surface tension, Pa m
r	effective velocity ratio, –	ξ	streamwise coordinate (curvilinear), m
r^2	correlation coefficient, –		
u	lateral velocity (cartesian), m/s		
u_j	initial jet velocity, m/s		

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 Hinze [14], the break-up of such particles has been related to turbulent stresses and turbulent dissipation in particular. As for example Husseinet 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 solid 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 mm^2 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

$$J = \rho_v u_v^2 / (\rho_L\ v_b^2) \tag{1}$$

with ρ_V and ρ_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_v 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_v and pressure p_v measured directly upstream of the injection point. The steam velocity u_v is calculated from the measured steam mass flux G, the mass density ρ_v and the area of the injector ($\pi\ (0.002)^2/4$ m^2). 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 Q_L by the cross-sectional area of the duct ($(0.03)^2$ m^2). The Reynolds number Re_b, based on v_b and D_h = 0.03 m, is varied between