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# Numerical study of the flow and heat transfer in a turbulent bubbly jet impingement

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ABSTRACT

This contribution presents the results of modeling of the flow structure and heat transfer enhancement obtained by adding air bubbles into a turbulent liquid (water) impinging jet. A set of axisymmetrical steady-state RANS equations for the two-phase flow is utilized. The dispersed phase (bubbles) is modeled by the Eulerian approach. Liquid-phase turbulence is computed with the Reynolds stress model, taking into account the effect of bubbles on the carrier phase. The effect of changes in the gas volumetric flow rate ratio and bubble size on the flow structure, wall friction, and heat transfer in a gas–liquid impinging jet is numerically studied. The predictions demonstrate the significant anisotropy of the turbulent fluctuations in the axial and radial directions for the bubbly impinging jet. The addition of a gas phase into the turbulent liquid increases the wall friction (by up to 40% in comparison with the single-phase liquid jet) and heat transfer (by up to 50%).



## 1. Introduction

Jet impingement is a very effective method for enhancing heat transfer rate between a solid impinging surface and a jet flow in various applications (see comprehensive reviews [1–5]). Potential applications where high heat fluxes should be dissipated while the surface is kept at a relatively low temperature include high-power equipment, glass tempering, cooling of surfaces and pre-forms, spray painting, rotor blades, and nuclear reactors during emergency conditions [1–5]. The main parameters that affect the heat and mass transfer processes include the distance between the nozzle orifice (or pipe outlet) and the impingement (target) surface, the temperatures of the jet and wall surface, the velocity magnitude of the jet, and the turbulence intensity.

Many attempts to modify flow characteristics and increase heat transport in impinging jets have been made in the last decades. Impinging two-phase jets with droplets or bubbles are employed to enhance heat transfer between the flow and the solid surface. Gas-droplet jet impingement on a hot surface removes large amounts of heat because of the latent heat of evaporation [6,7]. One of the ways of increasing the heat transfer rate is by adding a gas (or vapor) phase to an impinging liquid jet [8–13]. The enhancement in heat transfer is attributed to the high turbulence level in the near-wall region accompanying bubble-induced agitation in a bubbly impinging jet [8].

Serizawa et al. [8] experimentally studied the heat transfer in an impinging air–liquid bubbly jet. The heat transfer coefficient was increased by a factor of 2–4 at a gas volumetric flow rate ratio of $\beta \leqslant 0.53$. The enhancement was attributed to the high turbulence level caused by bubble-induced agitation and to acceleration of the liquid phase by the more rapidly moving air. Measurements in a slot impinging bubbly jet by boiling on the target surface were carried out in [9]. The gas volumetric flow rate ratio was $\beta = 0$–$0.86$ and the Reynolds numbers varied Re = 3700–21,000. Significant enhancement of the heat transfer (up to 2.2-fold in comparison with single-phase water impinging jet) was obtained. The experimental investigation of single and multiple confined two-phase jets using Freon R-113 in the presence of boiling was carried out in [10,11]. The regimes of single-phase forced convection and boiling heat transfer were studied in [10,11]. Heat transfer of the liquid–vapor impinging jets was augmented by a factor of 1.2 than that of the single-phase jet impingement [10]. The experimental investigation of heat transfer and pressure drop an in the air bubbles-water impinging jet was performed in [12,13]. The heat transfer enhancement (up to 2 times) in bubbly impinging jet at $\beta = 20$–$30\%$ was shown. The following increase of the gas volumetric flow rate ratio led to a decrease in the heat transfer enhancement.

The turbulent structure in a round bubbly impinging jet without interphase heat transfer was studied experimentally under

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## Nomenclature

| | |
|---|---|
| $C_D$ | drag coefficient of bubbles |
| $C_P$, $C_{Pb}$ | specific heat capacity of carrier liquid (water) and dispersed (gas bubbles) phases respectively (J kg$^{-1}$ K$^{-1}$) |
| $D$ | diffusion coefficient (m$^2$ s$^{-1}$) |
| $d$ | bubble diameter (m) |
| $\mathrm{Eo}_b$ | $\mathrm{Eo} = g(\rho - \rho_b)d_H^2/\sigma$ modified Eotvos number |
| $h_c$ | convective heat transfer coefficient between liquid and bubble (W K$^{-1}$ m$^{-2}$) |
| $J$, $J_b$ | superficial velocity of liquid (water) and bubbles respectively (m s$^{-1}$) |
| $k$ | $k = \langle u_i u_i \rangle/2$ turbulent kinetic energy (m$^2$ s$^{-2}$) |
| Nu | $\mathrm{Nu} = h2R/\lambda$ Nusselt number |
| $P$ | pressure (Pa) |
| Pr | $\mathrm{Nu} = \mu C_p/\lambda$ Prandtl number |
| $R$ | pipe radius (m) |
| $\bar{R}$ | specific gas constant (J kg$^{-1}$ K$^{-1}$) |
| Re | $2RU_{m1}/\nu$ Reynolds number |
| $\mathrm{Re}_b$ | $\mathrm{Re}_b = \lvert U_R \rvert d/\nu$ Reynolds number of bubbles, based on the sleep velocity |
| $r$ | radial coordinate (m) |
| $T$ | temperature (K) |
| $\lvert U_R \rvert$ | $\lvert U_R \rvert = \lvert U - U_L \rvert$ sleep (interphase) velocity (m s$^{-1}$) |
| $U_i$ | components of mean velocity (m s$^{-1}$) |
| $U_{m1}$ | bulk mean velocity of the liquid flow at the pipe edge (m s$^{-1}$) |
| $U_*$ | friction velocity (m s$^{-1}$) |
| $\langle u' \rangle$, $\langle v' \rangle$ | intensity of velocity fluctuations in axial and radial directions respectively (m s$^{-1}$) |
| $\langle u_j' t \rangle$ | turbulent heat flux (m K s$^{-1}$) |
| $\langle u' v' \rangle$ | turbulent Reynolds stress (m$^2$ s$^{-2}$) |
| We | $\mathrm{We} = \rho \lvert U_R \rvert^2 d/\sigma$ Weber number |
| $x$ | axial coordinate (m) |
| $y$ | distance normal from the wall (m) |

*Greek letters*

| | |
|---|---|
| $\Phi$ | volume fraction of the dispersed phase |
| $\alpha$ | local void fraction of bubbles |
| $\beta$ | $\beta = J_b/(J_b + J)$ gas volumetric flow rate ratio |
| $\varepsilon$ | dissipation of the turbulent kinetic energy (m$^2$ s$^{-3}$) |
| $\lambda$ | heat conductivity (W K$^{-1}$ m$^{-1}$) |
| $\mu$ | dynamic viscosity (kg m$^{-1}$ s$^{-1}$) |
| $\nu$ | kinematic viscosity (m$^2$ s$^{-1}$) |
| $\rho$ | density (kg m$^{-3}$) |
| $\sigma$ | surface tension (N m$^{-1}$) |
| $\tau$ | $\tau = \frac{4\rho_b d^2}{3\mu \mathrm{Re}_b C_D}$ dynamic relaxation time of bubbles (s) |
| $\tau_W$ | wall friction (Pa) |
| $\tau_\Theta$ | $\tau_\Theta = C_{Pb}\rho_b d^2/(12\lambda_b Y)$ thermal relaxation time of bubbles (s) |

*Subscripts*

| | |
|---|---|
| 0 | single-phase flow |
| 1 | parameter under initial conditions |
| $L$ | liquid |
| $T$ | turbulent parameter |
| $W$ | parameter on the wall condition |
| $b$ | bubble |
| $m$ | mean-mass parameter |

*Acronym*

| | |
|---|---|
| CV | control volume |
| RANS | Reynolds averaged Navier–Stokes equations |
| SMC | second moment closure |

the superposition of external periodic excitation in [14]. The shear stress was measured on the target surface at a change of $\beta$ = 0–12%. The effect of suppression of the large-scale structures was observed for high gas contents. It was found out that an addition of the gas phase at $\beta$ < 12% leads to a significant increase in the wall shear stress (up to 40%) [13]. Recently, an experimental study of isothermal bubbly impinging turbulent jet by a combination of optical planar fluorescence for bubble imaging (PFBI) and particle image velocimetry (PIV) and particle tracking velocimetry (PTV) methods was carried out in [14]. The bubbly jet was investigated at different gas volume flow rate ratios of $\beta$ = 0–4.2% at a fixed Reynolds number, Re = 12,000. Distributions of local void fraction $\alpha$ and mean and fluctuating velocities of both phases were measured. Bubbles increase the liquid phase turbulence at the nozzle edge when there is an additional supply of bubbles into the liquid jet. Then, they suppress the turbulence fluctuations compared to the single-phase flow in the downstream direction at $x/(2R) > 0.3$. The bubbles significantly intensify turbulence fluctuations due to an increase of the slip velocity between the phases close to the impingement surface [15].

The above mentioned studies were focused on the influence of gas volumetric flow rate ratio on heat transfer and turbulent structure in impinging jet. The effect of the size of dispersed phase on these characteristics was poorly studied. The authors found no studies on numerical study of flow structure and heat transfer in the impinging jets. A numerical study of an isothermal bubbly flow under plunging free round jet conditions using mono- and polydispersed approaches was performed in [16]. The turbulence is modeled by a $k$–$\varepsilon$ model with taking into account the bubble-induced turbulence. The bubble break-up and coalescence processes were considered in the polydispersed air–water flow by using the inhomogeneous MUlti SIze Group (MUSIG) model [17]. Several studies of heat transfer in bubble flow in pipes have been presented in the literature [18–20]. The complexity of modeling bubbly turbulent flows with interphase heat transfer is associated with the large number of different physical multiscale phenomena involved, such as heat transfer, bubble coalescence and break-up processes.

The aim of this work is to carry out a numerical simulation of the effect of bubbles size and gas volumetric flow rate ratio on flow structure and heat transfer modification in the turbulent bubbly jet impingement. This study may be interesting for scientists and engineers dealing with the problem of heat transfer enhancement in power equipment. The paper is organized as follows. In Section 2, the governing equations employed in this study for the simulation of a two-phase turbulent bubbly impinging jet are introduced. The numerical methods are described in Section 3. In Section 4, a numerical study of bubbly jet impingement with heat transfer is presented. Section 5 summarizes the main findings and conclusions.

## 2. Problem statement and governing equations

### 2.1. Physical model

The modeling of dispersed phase is accomplished by the Eulerian two-fluid approach that treats the particulate phase as a continuous medium with properties analogous to those of a liquid [21]. In the two-fluid approach, both phases are considered as interacting continua. This technique involves the solution of a second set of Navier–Stokes-like equations in addition to those of the carrier (liquid) phase. Furthermore, the significance of liquid–par-