Experimental and numerical study of heat transfer in a bubbly turbulent flow in an abrupt pipe expansion

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Abstract. Results from the experimental and numerical simulations on the flow structure and heat transfer in polydisperse bubbly flows in a sudden pipe expansion are presented. The addition of air bubbles results in a significant increase in the heat transfer rate (up to 300%). These effects are augmented by increasing the gas volumetric flow rate ratio. The largest heat transfer enhancement was observed in flow relaxation zone after the reattachment point.

1. Introduction
Two-phase bubbly flows are frequently used in chemical technologies, power engineering, and many other practical applications. A lot of new technologies are presented, as example the advanced thermal management of internal combustion engines. Typically, such flows are turbulent with an interfacial interaction between the fluid carrier phase and the gas bubbles. They may be complicated by flow separation at sharp edges, polydisperse bubbles, or its break up and coalescence, and interfacial heat transfer. Recirculation flow, occurring from flow separation at a sharp edge, largely determines the structure’s turbulent flow, and has a significant impact on the intensity of the transfer for momentum, mass, and heat as shown in [1]. The correct simulation of bubble distributions over a channel or pipe section is of great importance for safe operation, and the prediction of various emergency scenarios in heat generators connected to various energy elements.

To our knowledge, the first experimental study on two-phase bubbly flows behind a sudden expansion is [2,3]. These experimental works examined bubbly flows behind a sudden expansion of a horizontal channel. The pressure loss, size, and gas content of the bubbles, as well as the averaged and pulsation velocities of the phases, were measured in these investigations. These works reported a reduction in bubble size along the channel length due to the growth in the longitudinal pressure gradient and the fragmentation of bubbles. The length of the recirculation zone was approximately 7H, where H is the height of a step. The presence of a secondary corner eddy was also shown.

Later studies of separated bubbly flows, without interfacial heat transfer, were continued in the works of [4,5]. A detailed theoretical and experimental study on the upward separated bubbly flow over a sudden pipe expansion was conducted in [4]. The mean and fluctuating axial bubble velocities, their void fraction, the bubble size distribution, the pressure drop, and the wall friction were all measured in this work. The intensity profile for the bubble’s velocity fluctuations were qualitatively similar to the carrier liquid phase pulsations. An experimental study on the influence of bubbles on the...
structure of turbulent flow was made in [5]. Small bubbles were involved in the separated flow and had a negative velocity in the flow separation zone. They were present almost over the entire pipe section, while large bubbles mostly moved through the flow core and the shear mixing layer. The distribution of the mean axial liquid velocity strongly depended upon the size of the gas inclusions.

The only work devoted to the numerical simulation of bubbly flows behind a sudden pipe expansion without heat transfer was [6]. The main purpose of this work was to develop a model for describing gas-liquid flows in the presence of separation regions, and to test the model by comparing it to proper authors’ measurements. The paper presents a mathematical model for a polydisperse flow in a vertical pipe. Separation of the two-phase flow occurs due to obstacle positioning, which occupies half of the pipe’s cross section. The liquid phase turbulence was modeled using the SST k-ω model. The dynamics of the bubbles were determined using the inhomogeneous MUltiple S1ze Group (h-MUSIG) model, which accounts for bubble break-up and coalescence processes. The predictions for bubbly pipe flows with an internal obstacle demonstrated the complicated relationship and interference between size-dependent bubble migration, bubble coalescence, and breakup effects in real flows [6].

Many important questions regarding the fluid mechanics and thermophysics in two-phase bubbly flows downstream from a sudden pipe expansion remain open. These primarily concern is the influence of the gas volumetric flow rate ratio on the heat transfer. We have not yet found any experimental works, that investigate heat transfer in separated bubbly flows. Ref. [7] presented a mathematical model and the results for flow structure and heat transfer simulations for a vertical bubbly flow downstream from a sudden pipe expansion. The model was based upon the Eulerian method. The turbulent characteristics of the carrier fluid were predicted using the Reynolds stress transport model. The authors accounted for the break-up, coalescence, and the expansion of bubbles as their densities changed. These changes influenced the gas volumetric flow rate ratio, and the inlet liquid temperature and its velocity, while these effects were studied for the flow structure and heat transfer. The addition of the gas phase in turbulent fluid resulted in more than a factor of 2.5 increase in the heat transfer.

The aim of this work is the experimental and numerical study of a gas volumetric flow rate ratio on the turbulent structure of fluid flow and the heat transfer intensification during two-phase bubble flow downstream of a sudden pipe expansion.

2. Introduction
The experimental setup consisted of a closed circuit in a liquid, and an open circuit in the gas phase (see Fig. 1a). The test liquid, i.e., distilled water, was pumped from the main tank $l$ to the test section using the centrifugal pump $2$ to pump the test liquid through the water supply line $3$. The volume of liquid in the main tank was 40 liters. The temperature of the liquid was automatically controlled and maintained at $25±0.1^\circ C$. The Grundfos CHI 4-40 ($Q_l ≤ 4.5$ m$^3$/h) was used to pump the liquid. The liquid flux was directed upstream, while the liquid flow rate was controlled using a valve $4$. The flow rate was monitored using an ultrasonic flow meter $5$.

Bubbles were generated by feeding air into the liquid flow through 12 capillaries, where each capillary was 50 mm in length and possessed an internal diameter of 0.7 mm. The capillaries were uniformly distributed across the pipe channel’s cross section, which had a diameter of 14 mm. The outer faces of the capillaries were mounted into a cylindrical body with an inner diameter of 200 mm and a height of 40 mm. Atmospheric air was supplied to this volume by a compressor. The gas flow rate was determined using a gas flow controller $7$ produced by Bronkhorst ($Q_g ≤ 1$ l/min). The measurement error of the liquid and gas flow rates were ±2% of the measured value. Then, a gas-liquid mixture was fed into the test section. The setup’s test section produced flow separation was a small diameter pipe $8$ ($2R_1 = 14$ mm) that was inserted through the adapter and into a large diameter pipe. The inner diameter of the pipe, after expansion $16$, was $2R_2 = 42$ mm, and the step height was $H = 14$ mm. The expansion ratio defined as $ER = (R_2/R_1)^2 = 9$. The end of the small pipe was mounted flush with the plane of the adapter located in the flow separation zone. To make ensure a fully
developed, gas-liquid flow in the test section’s inlet, the length of the flow-stabilization area before the measurement region was 140R.

Measurements of bubbles sizes were collected using video with shadow illumination, prior to expansion location. To reduce the optical distortion, a box with a square cross-section 10 was filled with immersion liquid. The video camera 11 had a frame rate up to 400 fps, which was used to obtain images of the two-phase flow. Bubble sizes were calculated using image processing procedure in Matlab. For nearly spherical bubbles, the measurement uncertainties depended upon the square of temperature deviation did not exceed ±0.15ºC within the IR range. For comparison, three resistive temperature detectors (RTDs) were installed to avoid the output effects on the results of measurements. This pipe had a diameter of 42 mm and a length of 1 m. Then the liquid went to the main tank through the downcommer 17.

The measuring method employed here was infrared thermography. To increase the infrared emissivity for these measurements, the external surface of the heat transfer measurement unit was painted black. We used a thermal imager 14 (Fluke Ti32) to collect measurements. The thermal imager’s temperature ranged from -20 to +600ºC. The resolution of the radiation receiver was 320x240 pixels, while the thermal sensitivity was less than 0.045ºC. The spectral infrared (IR) range was from 7.5 to 14 μm (long-wave). The len’s spatial resolution was 1.25 mrad, while according to the manufacturer’s information, the temperature-dependent, measuring error was ± 2%.

For comparison, three resistive temperature detectors 15 (type Pt1000) were installed on opposite side of the heat transfer measurement unit related to the IR camera field of view at distances of x = 68, 230, and 405 mm (x/H ~ 5, 16.5 and 29). The thermal resistors, which were placed in a thin layer of thermal paste, were pressed, using straps, against the pipe surface. Around the resistors, insulation was used to reduce heat loss into the surrounding environment (Fig. 1b). The data from these resistive temperature detectors (RTDs) were recorded on the computer’s hard disk using an E14-440 instrument board.

Two preliminary calibration tests were performed for the thermal resistors mounted onto the heat transfer measurement unit’s surface. In the first test, working fluid with different temperatures was pumped through the channel. The temperature deviation did not exceed ±0.15ºC within the IR camera’s entire temperature field. The temperature of 25ºC was selected as the base temperature. From 25 to 35ºC, the differences between the temperatures of the setup’s wall and base temperature were obtained. This procedure was repeated several times. The differences of the values obtained using IR thermography and RTDs did not exceed ±8%. In the second test, the heat transfer measurement unit was mounted onto a channel consisting of a round pipe with an inner diameter of 42 mm. The liquid was then pumped across the pipe, which allowed the single-phase, liquid flow in the pipe to be modeled. Then, the test section was heated using electric current before data was collected and compared with the well-known, Dittus-Boelter correlation. The signal for the downstream temperature sensor was monitored to avoid entrance effects. The results obtained using the different methods were
in a good agreement with the correlation and with each other. The differences of the values obtained using IR thermography and RTDs did not exceed ±6% in this case.

The experiments were conducted as follows: (1) the required liquid flow rate was set; (2) five thermograms were recorded in the absence of heating; (3) the gas phase was added into the flow, and voltage was supplied to the heat transfer measurement unit, while 20 thermal images were recorded after the flow stabilized; and (4) the heat was turned off for two minutes before 5 thermograms were recorded. At the same time, temperatures at the test section’s inlet and outlet, as well as signals from the temperature sensors mounted onto the heated wall’s surface, were recorded. The temperatures of the liquid in inlet and outlet sections were measured by additional RTDs, mounted before and after the test section. Then, the cycle of measurements were repeated. Given the axial symmetry of the flow, thermal images were processed to average vertical regions that correspond to the test section’s height. For all modes, the wall temperature was determined. Applying the known heat flux to the wall, temperatures at the test section’s inlet and outlet and within its temperature distribution field were calculated, as well as the heat transfer coefficient $h$ and the Nusselt number $Nu$, which were the main result of our experimental studies.

3. Problem statement and governing equations

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. 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 (fluid) phase. In order to account for the interaction between phases, that is, momentum transfer and heat and

![Figure 1](image-url). The scheme of the experimental setup (a): 1 – main tank; 2 – pump; 3 – supply line; 4 – control valve; 5 – ultrasound flow meter; 6 – gas-liquid mixer; 7 – gas flow controller; 8 – the pipe of small diameter (before expansion); 9 – expansion area; 10 – box filled by water; 11 – video recorder; 12 – test section; 13 – copper conductors; 14 – infrared (IR) camera; 15 – resistive temperature detectors (RTDs); 16 – the pipe of large diameter (after expansion); 17 – downcomer; (b): cross section of heat transfer measurement unit: 1 – thin stainless steel wall; 2 – inner pipe; 3 – two-phase bubbly flow; 4 – thermal insulation; 5 – RTDs.
mass transfer, the conservation equations have to be extended by appropriate source/sink terms. The Eulerian approach is based on kinetic equations for a one-point PDF of bubbles coordinates, velocity, and temperature in the turbulent Gaussian fluid flow fields [7,9]. Properties such as the mass of particles per unit volume are considered as a continuous property and the particle velocity is the averaged velocity over an average control volume. The mean liquid flow is treated as a steady-state, incompressible and axisymmetrical flow and it is described by mass conservation continuity, two-momentum and energy equations with taking into account the effect of bubbles presence. The two-equation turbulence models have some well-known shortcomings. These models do not describe turbulent stress anisotropy, which it leads to considerable errors in modeling strongly non-equilibrium flows. One way to treat partially this anisotropy is to apply the model of Reynolds stress transport or second-moment closure (SMC). These models are more complex from the point of view of computations, compared to $k$–$\varepsilon$ models, and SMC models are more detailed than the two-equation models and have a lower computational cost than the large-eddy simulation (LES) and direct numerical simulation. The SMC model predicts the turbulent Reynolds stresses directly from partial differential equations and allows us to compute the anisotropic flow. In the present study, the low-Reynolds number elliptic blending second-moment closure of [10] is employed. It is modified for the presence of bubbles by [11]. The interfacial force is usually divided into several components, for example: drag, virtual mass, gravity, lift, turbulent dispersion and wall lubrication [7].

3.1. Numerical realization

The mean transport equations for both gas and dispersed phases and the SMC model are solved using a control volumes method on a staggered grid. The QUICK scheme is used to approximate the convective terms, and the second-order accurate central difference scheme is adopted for the diffusion terms.

The velocity correction is used to satisfy continuity through the SIMPLEC algorithm, which couples velocity and pressure. At the inlet all velocity components, temperatures of the phases and turbulence levels are uniform. The symmetry conditions are set on the channel axis for gas and dispersed phases. No-slip conditions are set on the wall surface for the carrier phase. At the outlet edge, the computational domain condition $\partial \rho / \partial r = 0$ is set for all variables. The first cell was located at a distance $y_+ = y U_*/\nu = 0.3 – 0.5$ from the wall, where $U_*$ is the friction velocity obtained for the single-phase flow in the inlet pipe. At least 10 control volumes were generated to ensure resolution of the mean velocity field and turbulence quantities in the viscosity-affected near-wall region. Grid sensitivity studies are carried out to determine the optimum grid resolution to give the mesh-independent solution. Grid convergence was verified for three grid sizes: 128×50, 256×100, and 400×150 control volumes. For all numerical investigations performed, the basic grid with 256×100 control volumes, in the longitudinal and transverse directions, was used.

4. Results and discussion

The distribution of the wall friction coefficient at the wall, $C_f = 2\tau_w / (\rho U_{\text{m}1}^2)$, and along the pipe length are shown in Fig. 2 for various gas volumetric flow rate ratios. Line $l$ is the result of the wall friction coefficient prediction for single-phase liquid flows. The increase in the volumetric gas flow rate ratio leads to a significant increase in the absolute value of the drag coefficient (almost three times that of the single-phase flow mode). Note that the minimum value of friction on the wall, located in the recirculation area, slightly shifts toward the edge cross section of the flow separation with increasing concentration.

Figures 3 show distributions of the local heat transfer (a) and the parameter for heat transfer enhancement (b) in a bubbly flow downstream of a sudden pipe expansion with various Reynolds numbers. Here $U_{\text{m}1}$ is the mean fluid velocity.
Figure 2. The distributions of friction coefficient along the axial coordinate at various gas volumetric flow rate ratios. 1 – single-phase flow (β = 0), 2 – 3.5%, 3 – 5.2%.

The Nusselt number was calculated according to the following relationship for the case \( q_W = \text{const} \):

\[
\text{Nu} = H q_W \left[ \lambda (T_{m2} - T_{m1}) \right].
\]

Here \( H \) is the step height, \( q_W \) is the heat flux density for the pipe wall, \( \lambda \) is the fluid’s coefficient of heat conductivity, and \( T_{m2} \) and \( T_{m1} \) are mean liquid temperatures at the pipe axis in the outlet and inlet sections respectively, while the mean liquid temperatures are calculated using the formula

\[
T_m = \frac{2}{U_1 R_1^2} \int_0^{R_2} T U/rdr.
\]

The increasing Reynolds number lead to a decrease in the heat transfer enhancement in the flow relaxation region, and when \( \text{Re}_H = 3.15 \times 10^4 \) and \( \beta = 3.5\% \) (see Fig. 3), there was only a slight increase in heat transfer intensity (less than 10%) compared to single-phase flows with otherwise identical parameters. In the flow recirculation zone and in the initial portion of the relaxation zone, the experimental results were almost no uninfluenced by gas bubbles contributing to the heat transfer (up to a distance of \( x/H = 10–15 \)), and the heat transfer rate was almost identical to that for single-phase flows when the volumetric gas flow rate ratio is \( \beta = 0 \). This was confirmed by the existence of relatively large bubbles for large values of the Reynolds number, and because they practically do not penetrate into the region of separated flow, and mostly pass through the flow core and the shear mixing layer. Recent work [7] showed that small bubbles \( (d < 1.5 \text{ mm}) \) caused heat transfer intensification over the entire length of the recirculation zone, while the larger ones caused intensification mostly in the flow relaxation region. For numerical calculations using \( \text{Re}_H = 2.09 \times 10^4 \) and \( 3.15 \times 10^4 \), we obtained similar values for heat transfer change, and for \( \text{Re}_H = 1.02 \times 10^4 \), there was an almost 30% increase in the heat transfer enhancement by two-phase bubbly flows compared to single-phase flows, which significantly differed from the measured data. Overall, there was fair agreement between measured data and our numerical predictions both in the region of flow recirculation and in its relaxation zone. Maximum heat transfer in the experiments and the computations occurred over distance \( x/H = 7–10 \) for the investigated range of Reynolds numbers and the air bubble concentrations. Note that the position for the point of maximum heat transfer roughly coincided with the attachment point of the flow [1]. For measured and predicted distributions, the Nusselt numbers along the pipe length are characterized by an increase in the heat transfer intensity approximately up to the flow reattachment point; then it sharply decreases in the area of the flow relaxation. The maximum differences between the measured and predicted values of the heat transfer were up to 25% in the flow recirculation zone and up to 15% in flow relaxation region for the investigated Reynolds numbers.
Figure 3. Nusselt numbers (a) and heat transfer enhancement ratios (b) distributions along the pipe length for various Reynolds numbers. Points and are authors’ measurements and computations respectively. $\beta = 3.5–3.6\%$, $a_1 = 1.7$ mm. 1 – $Re_H = U_{m1}H/\nu = 1.02 \times 10^4$, 2 – $2.09 \times 10^4$, 3 – $3.15 \times 10^4$.

5. Conclusion

The turbulent flow structure and the heat transfer for a bubbly turbulent flow downstream of a sudden pipe expansion were experimentally and numerically investigated. An experimental study on bubbly flow was carried out using shadow photography and infrared thermography. The experimental and numerical investigation was performed using the following range of Reynolds numbers: $Re_H = (1–3.2) \times 10^4$ and the gas volumetric flow rate ratio $\beta = 0–10\%$. In general, the structure of the mean and fluctuating two-phase bubbly flow with small values for $\beta \leq 10\%$, was qualitatively similar single-phase fluid flow.

An increase in the gas volumetric flow rate ratio leads to a significant increase (up to 30\%) in the wall friction coefficient compared to single-phase fluid flow downstream a sudden pipe expansion. It was experimentally and numerically shown that the addition of air bubbles caused a significant increase in the heat transfer rate (up to 3 times), and these effects increased with increasing gas volumetric flow rate ratios. The main increase in heat transfer was observed in the flow relaxation region after the point of flow reattachment. The significant decrease in the intensification ratios is characteristic for heat transfer in the flow relaxation region as the Reynolds number increased. Distributions for the Nusselt numbers along the pipe length were qualitatively similar values for single-phase and droplet-laden mist turbulent separated flows.
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