

Original Paper

Multiple size group modeling of gas–liquid flow instability and energy conversion in a multiphase pump: Effects of inlet gas volume fraction



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ABSTRACT

The variation of inlet gas volume fraction (IGVF) intensifies bubble breakup and coalescence behaviors, and the resulting evolution of gas pockets may directly impact the energy performance and stability of multiphase pumps. In this work, effects of IGVF on performance parameters, energy characteristics and flow instability are investigated by ANSYS CFX software. Bubble breakup and coalescence dynamics, along with their associated gas–liquid interfacial entropy production under high IGVF conditions are considered. Results show that predicted values of pressure increment, pressure fluctuation, and gas distribution characteristics agree well with experimental data. With the increase of IGVF, the pressure increment decreases while gas volume fraction (GVF) in the impeller and diffuser increases. When the IGVF rises from 5% to 15%, gas pockets form on the diffuser suction sides, accompanied by a sharp decline in pressure increment and rapid increase in GVF and average bubble diameter, ultimately triggering surge phenomena. The increasing IGVF accelerates bubble coalescence, leading to a higher proportion of large bubbles in the flow passages. Consequently, both the number and size of gas pockets increase, resulting in enhanced gas–liquid interfacial entropy production. Furthermore, pressure fluctuation characteristics are thoroughly analyzed, which can provide a reference for the energy loss analysis and optimal design of multiphase pumps.

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1. Introduction

Over the past decade, the energy consumption proportion of oil and gas in the Asia-Pacific region, especially in China, has continued to increase. However, the shortage crisis of onshore oil resources and the increasing exploitation difficulty also increase. As a key focus area in renewable energy development, the advancement of deep-sea offshore wind power brings new momentum to deep-sea oil–gas extraction. It not only reduces reliance on traditional energy sources and eliminates the need for

prolonged fuel transportation, but also cuts greenhouse gas emissions, making it environmentally friendly. This situation has driven the exploration and development of deep-water oil and gas, which can help achieve the carbon peaking and carbon neutrality goals and ensure national energy security (Wang and Gao, 2022; Si et al., 2025; Skvortsov, 2025; Rostamian et al., 2025). As the core equipment of deep-water oil and gas resource exploitation and long-distance transportation in Fig. 1, the research and development of multiphase pumps is of great significance (NTGD Pump, 2021; iStock, 2018). The increase of inlet gas volume fraction (IGVF) may lead to complex flow phenomena such as secondary flow and flow separation in the pump, accompanied by energy loss and instability. Therefore, enhancing both pressurization capacity and operation stability of a multiphase pump is critical under high IGVFs.

Different from the single-phase pump, there is transient gas–liquid two-phase complex flow in the multiphase pump. The

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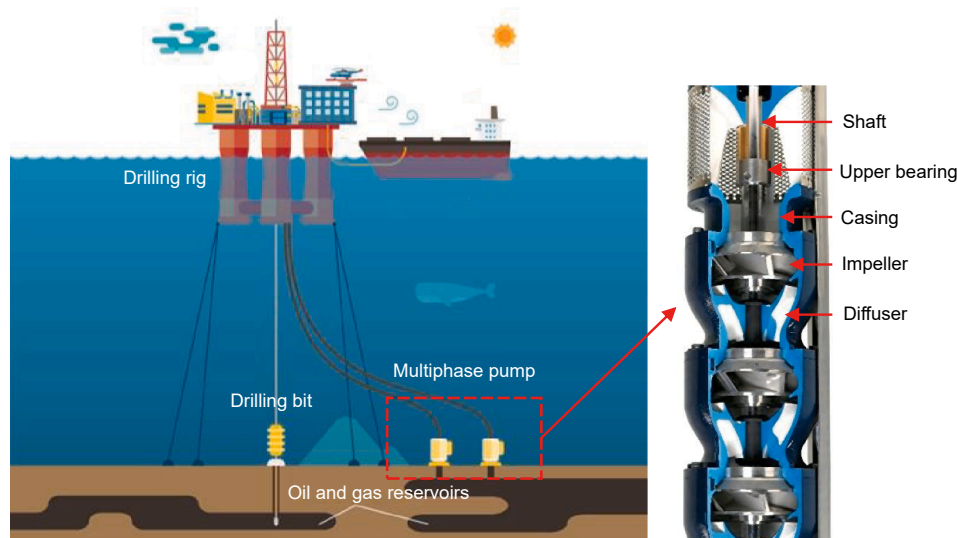


Fig. 1. Multi-phase pump for deep-water oil and gas exploitation (NTGD Pump, 2021; iStock, 2018).

pump exhibits not only secondary flow, tip leakage flow, wake flow, and flow separation, but also bubble growth, development, and collapse due to gas–liquid interaction. These phenomena lead to the transition of different flow patterns and instability in the pump. Experimental methods are employed to investigate the performance and flow characteristics of multiphase pumps. The experimental research includes performance tests and visualization tests. While performance tests measure parameters such as pressure increment, torque, pressure fluctuation, and vibration, they cannot reveal internal flow mechanisms. Visualization tests address this limitation by replacing the impeller shroud and diffuser shroud with transparent acrylic materials (Verde et al., 2017; Luo et al., 2024) or replacing all components, including the impeller and diffuser blades, with organic glass materials (Ge et al., 2020). Studies have investigated the effects of IGVF, inlet pressure, liquid flow rate, and rotational speed on the pump's external characteristics, with the IGVF being the primary influencing factor (Pessoa and Prado, 2001; Beltur et al., 2003; Gamboa and Prado, 2012; Zhu et al., 2017). The pump's pressure increment initially declines slowly before dropping sharply with increasing IGVF. Further, Xu et al. (2019) investigated the influence of IGVF on the pressure fluctuation characteristics inside the pump and found that the amplitude of pressure fluctuation increases with higher IGVF.

Regarding bubble dynamics and flow pattern identification, Murakami and Minemura (1974) firstly conduct visualization studies on gas–liquid two-phase flow in pumps. Researchers have used various techniques, gamma-ray computed tomography (Schäfer et al., 2015), electrical resistance tomography (Pirouzpanah et al., 2017), and high-speed imaging (Verde et al., 2017), to identify the flow patterns inside the pump. With increasing IGVF, the flow patterns in the multiphase pump can be classified into four main types: bubbly flow, agglomerated bubble flow, gas pocket flow, and segregated flow (Verde et al., 2017; Shao et al., 2018; Zhang et al., 2015, 2016a; Li et al., 2024). The evolution of flow patterns directly affects the pump's pressurization performance. When bubbles accumulate to a certain extent in the pump, forming a gas mass or gas pocket, occupying most of the flow channel, which may hinder the flow of water. In particular, the gas pocket flow significantly exacerbates performance degradation (a sharp drop in pressure increment) and operational instability, namely the surge phenomenon. However, the current

experimental setup allows for the collection of relatively limited parameters. Key factors such as bubble distribution, void fraction, and pressure distribution on impeller and diffuser blades cannot be accurately obtained. Additionally, due to the limitations of the organic glass material properties, internal flow characteristics can only be captured under off-design operating conditions.

With the rapid development of computer technology and computational fluid dynamics, numerical simulation methods for multiphase pumps have gradually emerged. These methods offer advantages such as low cost, short development cycles, internal flow mechanism analysis, and performance prediction under complex boundary conditions and special operating scenarios. Eulerian–Eulerian model is a widely used multiphase flow model for predicting gas–liquid two-phase flows (ANSYS, Inc., 2018), in which the inhomogeneous model can effectively simulate complex gas–liquid flow behaviors within multiphase pumps (Yu et al., 2013; Zhu et al., 2019). Using unsteady numerical simulations, Zhang et al. (2016b) investigated gas pocket motion characteristics in multiphase pumps, including size and location. Further studies have explored pressure fluctuation characteristics (Zhang et al., 2019b) and tip leakage vortex dynamics in multiphase pumps (Xiao et al., 2022).

Early studies on gas pockets predominantly used fixed bubble diameters, typically using fixed diameters of 0.1 mm and 0.2 mm, or selecting a single bubble diameter for each IGVF condition. However, as the IGVF varies, bubbles undergo transient behaviors such as coalescence and breakup within the pump, leading to continuous bubble size variations. This phenomenon becomes particularly pronounced under high IGVF conditions. The bubble diameter plays a pivotal role in determining multiphase pump performance characteristics (Caridad et al., 2008). The conventional approach of using constant bubble diameters in simulations proves inadequate, as it underestimates IGVF effects and fails to capture surge phenomena. To address this limitation, Li et al. (2019) and Zhang et al. (2019a) extended the Eulerian–Eulerian model with an inhomogeneous bubble model, while Ge et al. (2020) and He et al. (2020) employed the population balance model (PBM) with 10 bubble size groups to simulate bubble coalescence and breakup behaviors within the pump. Additionally, Si et al. (2022) adopted an alternative PBM approach, the multiple size group (MUSIG) model. These advanced modeling approaches collectively account for dynamic bubble size variations, enabling

more accurate prediction of internal flow field evolution in gas–liquid pumps.

In summary, the numerical simulation methodology based on the Eulerian–Eulerian model for internal flow analysis in multiphase pumps has reached relative maturity, enabling substantial research progress in performance evaluation, internal flow mechanisms, and phase distribution characteristics. Multiphase flow models that account for bubble breakup and coalescence through multi-size bubble groups have been increasingly applied to performance prediction studies of multiphase pumps under high IGVF conditions (Fall et al., 2025; Guo et al., 2024; Zhou et al., 2025). From the inhomogeneous bubble model and the PBM to its variant, the MUSIG model, these multi-size bubble models have been continuously refined and advanced by researchers, and are now one of the key multiphase flow models receiving major scholarly attention. Nevertheless, existing studies frequently overlook the critical effects of bubble size variation and distribution, particularly during high IGVF conditions, where bubble coalescence and breakup phenomena receive insufficient attention. This oversight leads to underestimation of gas–liquid interactions and consequently results in overpredicted pump performance. Moreover, the influence of dynamic bubble size changes on pressure fluctuation and energy characteristics remains significantly understudied.

In this work, effects of IGVF on energy losses and flow stability in a mixed-flow multiphase pump are investigated using the MUSIG model based on the Eulerian–Eulerian multiphase model. The numerical model and methodology are first validated through experimental test. Detailed analyses are subsequently performed to reveal internal flow characteristics, pressure fluctuation characteristics within the impeller, and entropy generation distribution features.

2. Numerical model and method

2.1. Geometric model

In this work, the research object is a single-stage mixed-flow multiphase pump. The main parameters are shown in Table 1. The pump model primarily consists of an inlet pipe, an impeller, a diffuser, and an outlet pipe, as illustrated in Fig. 2. The impeller and diffuser have 4 and 7 blades, respectively. The inlet and outlet pipes measure 4 times and 6 times the inlet shroud diameter of the impeller, respectively.

2.2. Mesh generation

The computational domain of the mixed-flow multiphase pump is discretized using structured meshes generated with the commercial software ANSYS ICEM and ANSYS TurboGrid. ANSYS ICEM is used to mesh the inlet and outlet pipes, while ANSYS

Table 1
Main parameters of the pump.

/	Parameter	Symbol	Value	Unit
Impeller	Number of blades	Z_1	4	/
	Inlet shroud diameter	D_{s1}	108.76	mm
	Outlet shroud diameter	D_{s2}	135.75	mm
	Blade outlet width	b_2	13.84	mm
	Tip clearance	δ	1	mm
Diffuser	Number of vanes	Z_{DV}	7	/
	Outlet shroud diameter	D_{s3}	108.97	mm
	Design flow rate	Q_d	75	m ³ /h
Operating conditions	Design head	H_d	17	m
	Rotational speed	n_d	3500	rpm

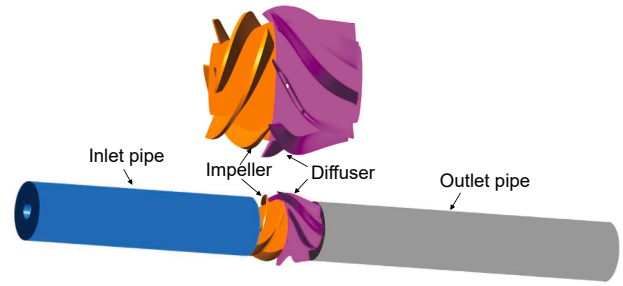


Fig. 2. Geometric model.

TurboGrid is employed for the impeller and diffuser. Fig. 3 shows the mesh of main components in the pump, where (a) is the impeller, (b) is the diffuser, (c) is the impeller blade, and (d) is the diffuser vane.

A mesh independence verification is conducted using four sets of meshes to eliminate the influence of mesh quantity on the computational results. The mesh numbers for each component are listed in Table 2. The verification targets are the head (H) and efficiency (η) of the multiphase pump under pure water conditions ($n_1 = 1500$ rpm, $Q_1 = 33$ m³/h; $n_2 = 3500$ rpm, $Q_2 = 78$ m³/h).

Fig. 4 illustrates the variation of the targets with increasing mesh numbers, where the values represent the relative change in the target values compared to those obtained from the previous set of mesh. When the total mesh number (N) exceeded 3.34 million, both the head and efficiency stabilize with relative variation rates below 1%. Considering the balance between computational cost and accuracy, Mesh 3 with a total of 3.347×10^6 elements is selected for subsequent simulations, comprising 0.481×10^6 for the inlet pipe, 1.213×10^6 for the impeller, 1.438×10^6 for the diffuser, and 0.215×10^6 for the outlet pipe. Furthermore, a single-channel computational strategy is adopted to address the substantial computational demands of transient gas–liquid flow simulations. This approach maintains the original mesh number for inlet and outlet pipes while utilizing single-channel meshes for both impeller (0.303×10^6 elements) and diffuser (0.205×10^6 elements), thereby reducing the total mesh number to 1.205×10^6 elements and significantly decreasing computation time.

2.3. Multiphase model

The multiphase pump's internal flow involves transient gas–liquid complex flow, where both the gas phase (air) and liquid phase (water) can be treated as incompressible, viscous fluids. The two-phase flow undergoes no heat exchange but exhibits strong interfacial interactions. The flow phenomena inside the pump include secondary flows, tip leakage flows, wake flows, and flow separation, coupled with dynamic processes of bubble growth, development, and collapse. These complex interactions lead to continuous evolution of different flow patterns within the pump. Therefore, this work employs the Eulerian–Eulerian inhomogeneous model in ANSYS CFX, which establishes separate continuity in Eq. (1) and momentum equations in Eq. (2) for each phase as follows (ANSYS, Inc., 2018):

$$\frac{\partial}{\partial t} (\beta_q \rho_q) + \nabla \cdot (\beta_q \rho_q \mathbf{w}_q) = 0 \quad (1)$$

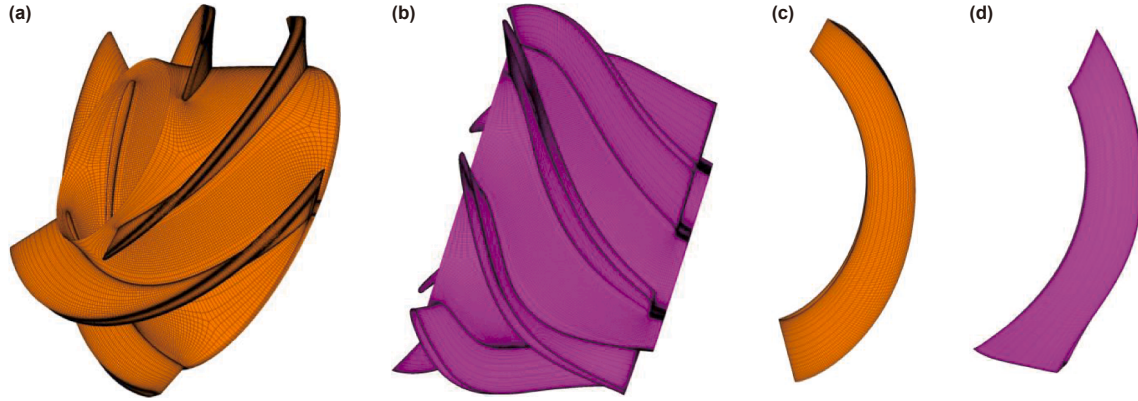


Fig. 3. Mesh generation. (a) Impeller. (b) Diffuser. (c) Impeller blade. (d) Diffuser vane.

Table 2
Mesh number of each component.

Component	Mesh number $N, \times 10^5$			
	1	2	3	4
Inlet pipe	0.54	1.65	4.81	12.78
Impeller	1.52	4.32	12.13	34.48
Diffuser	1.89	5.04	14.38	40.95
Outlet pipe	0.23	0.69	2.15	5.70
Total	4.18	11.70	33.47	93.91

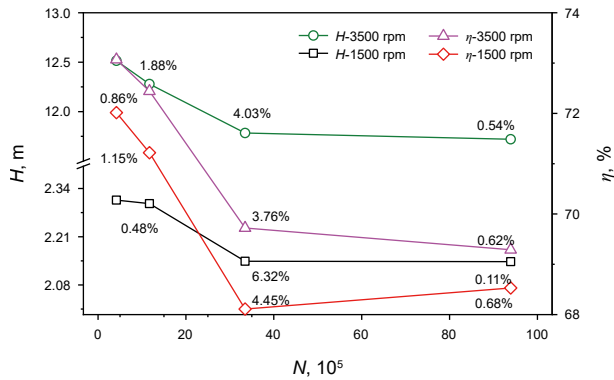


Fig. 4. Mesh independence analysis.

$$\frac{\partial}{\partial t} (\beta_q \rho_q \mathbf{w}_q) + \nabla \cdot (\beta_q \rho_q \mathbf{w}_q \otimes \mathbf{w}_q) = -\beta_q \nabla p_q + \nabla \cdot (\beta_q \mu_q (\nabla \mathbf{w}_q + (\nabla \mathbf{w}_q)^T)) + \beta_q \rho_q \mathbf{f}_q + \mathbf{M}_q \quad (2)$$

where β represents the volume fraction; ρ denotes density; the subscript q indicates either the gas or liquid phase; $\mathbf{w}$ stands for velocity; p is pressure; μ represents dynamic viscosity; $\mathbf{f}$ accounts for the mass force induced by impeller rotation; and $\mathbf{M}$ signifies the interphase force between gas and liquid phases, primarily including drag force ($\mathbf{F}_D$), lift force ($\mathbf{F}_L$), and virtual mass force ($\mathbf{F}_V$).

The sum of volume fractions of the gas (β_g) and liquid phases (β_l) is equal to 1, and the conservation equation is expressed in Eq. (3):

$$\beta_g + \beta_l = 1 \quad (3)$$

The gas volume fraction (GVF) β_g (including IGVF) is calculated using the following equation in Eq. (4):

$$\beta_g = \frac{Q_g}{Q_g + Q_l} \quad (4)$$

where the subscripts g and l denote the gas and liquid phase, respectively; Q is flow rate.

The drag force $\mathbf{F}_D$ is the key interphase force affecting bubble motion, caused by surrounding pressure distribution. The lift force $\mathbf{F}_L$ refers to the shear-induced force acting on the dispersed phase when the continuous phase rotates, perpendicular to the relative motion direction of the two phases. The virtual mass force $\mathbf{F}_V$ is proportional to the relative acceleration and the density of the continuous phase. The drag force $\mathbf{F}_D$ in Eq. (5), lift force $\mathbf{F}_L$ in Eq. (6), and virtual mass force $\mathbf{F}_V$ in Eq. (7) are expressed as follows (ANSYS, Inc., 2018):

$$\mathbf{F}_D = \frac{3}{4} C_D \frac{\rho_l \beta_g}{d_b} |\mathbf{w}_g - \mathbf{w}_l| (\mathbf{w}_g - \mathbf{w}_l) \quad (5)$$

$$\mathbf{F}_L = \beta_g \rho_l C_L (\mathbf{w}_g - \mathbf{w}_l) \times (\nabla \times \mathbf{w}_l) \quad (6)$$

$$\mathbf{F}_V = \beta_g \rho_l C_V \left(\frac{d\mathbf{w}_g}{dt} - \frac{d\mathbf{w}_l}{dt} \right) \quad (7)$$

where C_D represents the drag coefficient, d_b denotes the Sauter mean bubble diameter, C_L stands for the lift coefficient, and C_V indicates the virtual mass force coefficient.

Considering the transient behaviors such as bubble coalescence and breakup in multiphase pumps under high IGVEs, the surface tension effects cause bubble deformation, resulting in irregular shapes (e.g., elliptical or spherical-cap shapes). The original Schiller Naumann model is defined for the spheroidal particle at a low volume fraction and thus unsuitable for the bubble behavior including coalescence and breakup at high IGVEs (Schiller and Naumann, 1933). A modified drag coefficient model is adopted to account for bubble deformation and size variations (Ge et al., 2023). During computation, the higher drag coefficient derived from the following two equations is automatically selected as the interphase drag coefficient for gas–liquid interactions in Eqs. (8)–(13):

$$C_D = \begin{cases} C_{D1}, & C_{D1} \geq C_{D2} \\ C_{D2}, & C_{D2} > C_{D1} \end{cases} \quad (8)$$

$$C_{D1} = \frac{24}{Re} \left(1 + 0.15 Re^{0.687} \right) \quad (9)$$

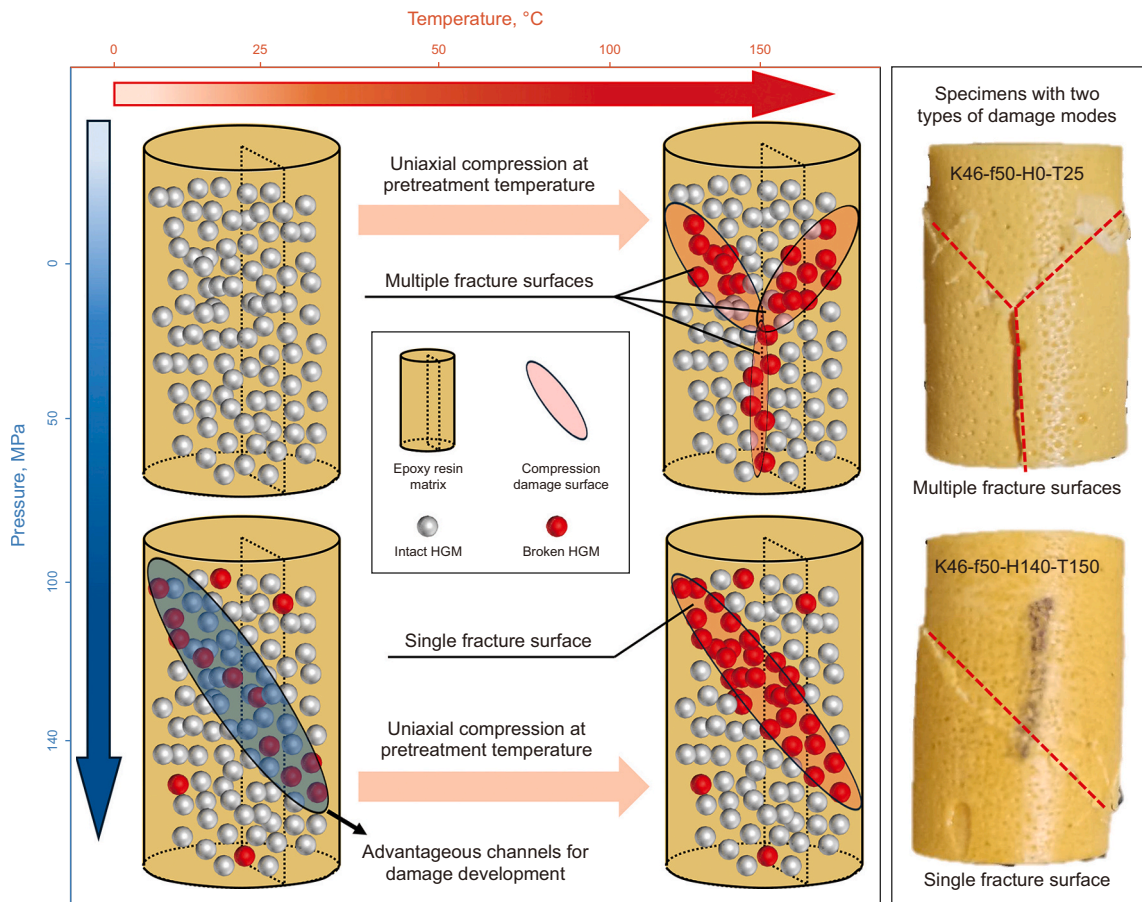


Fig. 5. The experimental system.

$$\text{Re} = \frac{\rho_l |\mathbf{w}_g - \mathbf{w}_l| d_b}{\mu_l} \tag{10}$$

$$C_{D2} = \frac{8}{3} \frac{\text{Eo}}{\text{Eo} + 4} \tag{11}$$

$$\text{Eo} = \frac{g(\rho_l - \rho_g) d_b^2}{\sigma} \tag{12}$$

$$d_b = \frac{\sum (N_i d_i^3)}{\sum (N_i d_i^2)} \tag{13}$$

where Re represents the Reynolds number, Eo denotes the ratio of gravity to surface tension, g stands for gravitational acceleration, σ indicates the surface tension, and N_i refers to the number count of bubble group i.

The lift coefficient model is calculated using the widely-cited Legendre–Magnaudet model (Legendre and Magnaudet, 1998), while the virtual mass coefficient is set to 0.5.

The MUSIG model is adopted as the bubble size distribution model to account for the polydisperse bubble sizes resulting from

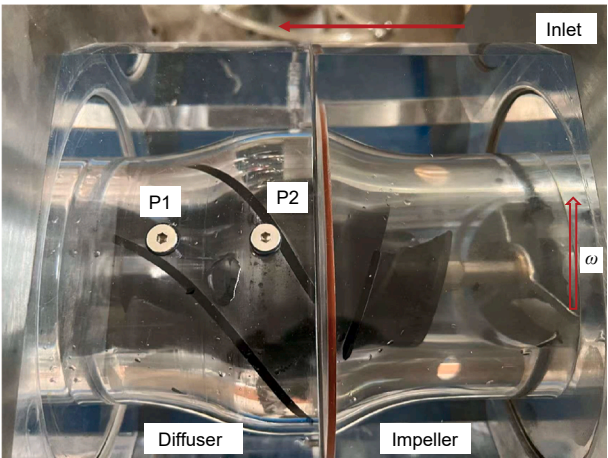


Fig. 6. The model pump with pressure fluctuation monitoring points.

Table 3 Main instrument parameters in the acquisition system.		
Instrument	Range	Uncertainty
Gas laminar flow mass flowmeter (large)	0–3000 SLPM	±1% RD
Gas laminar flow mass flowmeter (small)	0–200 SLPM	±0.5%
Liquid mass flowmeter	0–100 t/h	±0.2%
Dynamic torque sensor	0–5 N·m	0.1%
Pressure transmitter	–0.1–0.5 MPa	0.075%
Differential pressure transmitter	–0.1–1 MPa	0.075%
Data acquisition	48 input channels, 80 ks/s	16 bits
High frequency pressure fluctuation sensor	0.07–345 kPa	±1%

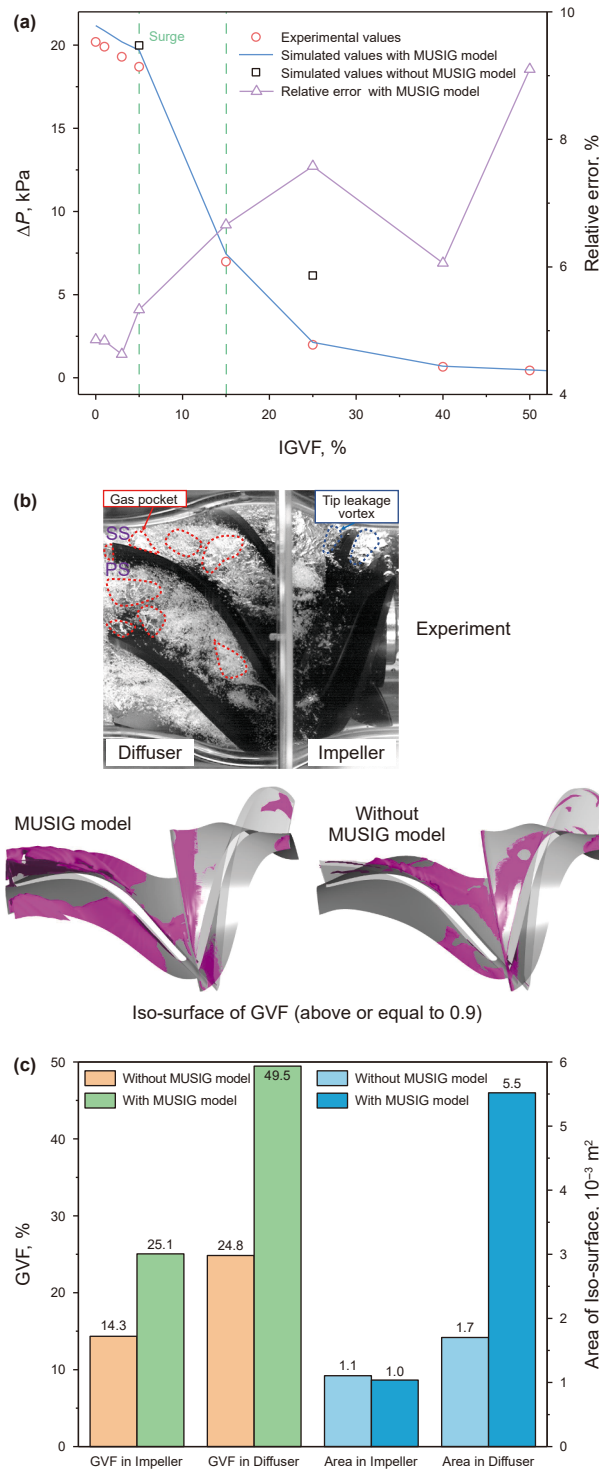


Fig. 7. Comparison of pump pressure increment and gas distribution characteristics between numerical simulations and experimental measurements ($n = 1500 \text{ rpm}$; $Q_{\text{BEP}} = 33 \text{ m}^3/\text{h}$). (a) Pressure increment. (b) Gas distribution at IGVF = 25%. (c) GVF and area of Iso-surface (GVF above or equal to 0.9).

coalescence and breakup phenomena in multiphase pumps. Specifically designed for polydisperse phases and intermediate-sized bubbles, it is capable of tracking discrete phases across multiple size groups where each group represents bubbles of a specific

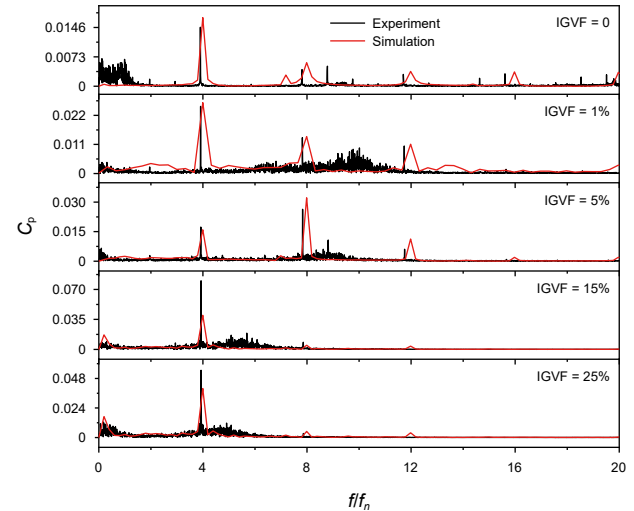


Fig. 8. Comparison of frequency spectrum analysis between experimental measurements and numerical simulations for pressure fluctuations in the diffuser under varying IGVF.

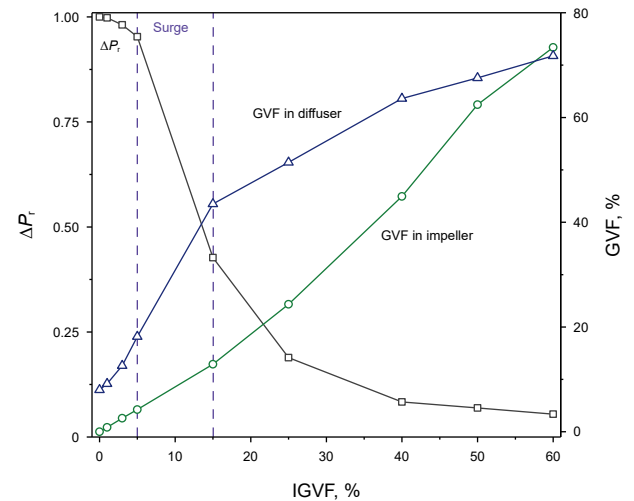


Fig. 9. Pressure increment and gas volume fraction of the impeller and the diffuser under different IGVFs.

diameter. This model is particularly suitable for bubbles in a distorted particle regime where their gradual slip velocity is sufficiently small relative to inertial timescales, thereby justifying the assumption that all bubble size groups share a common velocity field. The model dynamically adjusts the size distribution among groups by updating each group's volume fraction through population balance equations while maintaining a unity sum across all size groups.

Assuming identical density and velocity fields for all size groups, the model simplifies to the homogeneous MUSIG formulation, represented by the following governing equation in Eqs. (14)–(19) (ANSYS, Inc., 2018):

$$\frac{\partial}{\partial t} (\rho_g \beta_g f_i) + \frac{\partial}{\partial x^i} [\rho_g \beta_g \mathbf{w}_g^i f_i] = S_i \quad (14)$$

$$S_i = B_{Bi} - D_{Bi} + B_{Ci} - D_{Ci} \quad (15)$$

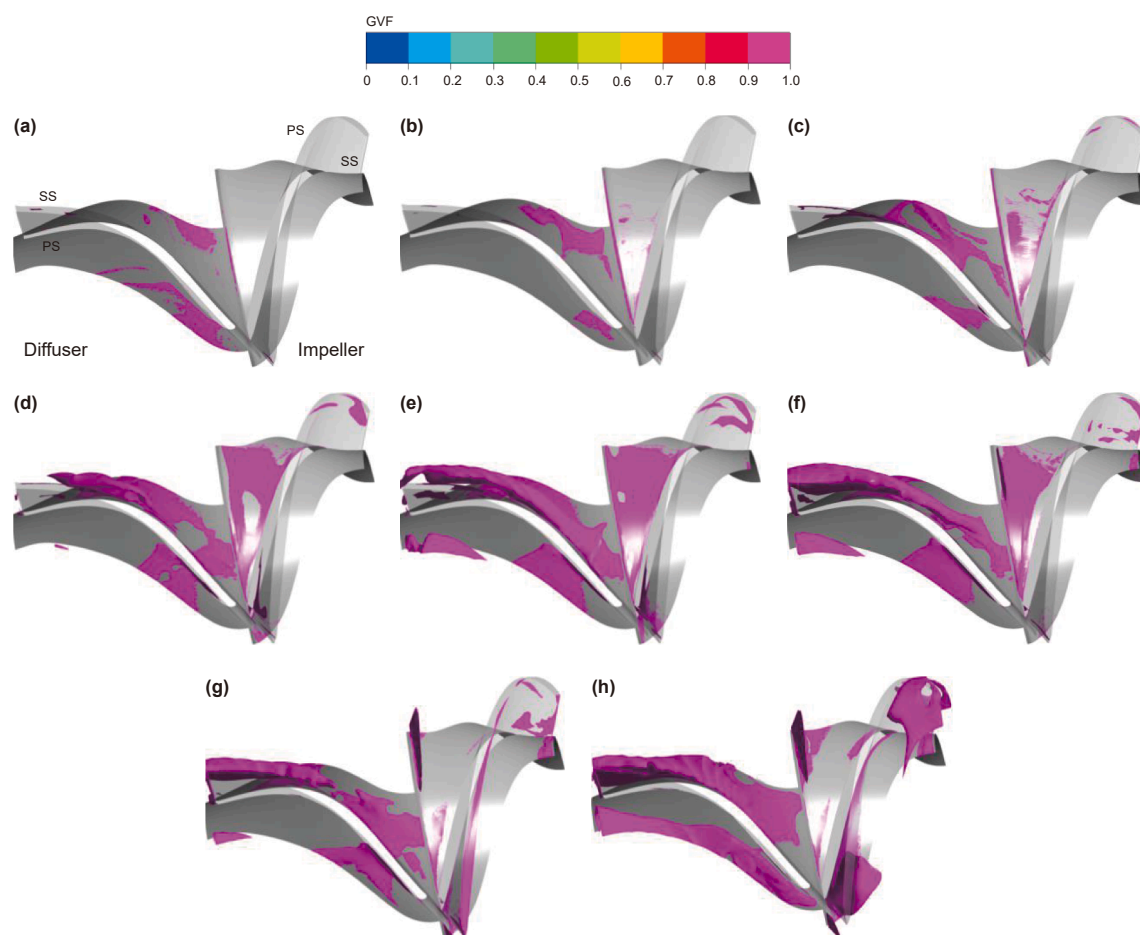


Fig. 10. Gas pocket distribution in the impeller and diffuser (iso-surface of GVF with the value of above or equal to 0.9). (a) IGVF = 1%. (b) IGVF = 3%. (c) IGVF = 5%. (d) IGVF = 15%. (e) IGVF = 25%. (f) IGVF = 40%. (g) IGVF = 50%. (h) IGVF = 60%.

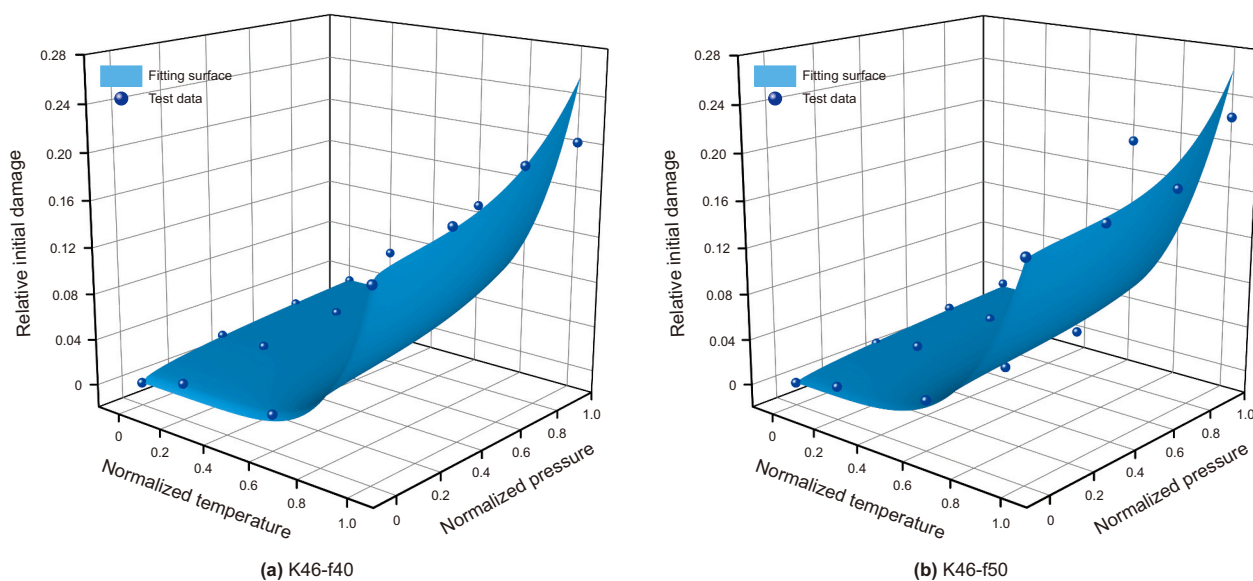


Fig. 11. Bubble distribution of different sizes in impeller and diffuser under different IGVFs.

$$B_{Bi} = \rho_g \beta_g \left(\sum_{j>i} g(m_j; m_i) f_j \right) \quad (16)$$

$$D_{Bi} = \rho_g \beta_g \left(f_i \sum_{j<i} g(m_i; m_j) \right) \quad (17)$$

$$B_{Ci} = (\rho_g \beta_g)^2 \left(\frac{1}{2} \sum_{j \leq i} \sum_{k \leq i} Q(m_j; m_k) X_{jki} f_j f_k \frac{m_j + m_k}{m_j m_k} \right) \quad (18)$$

$$D_{Ci} = (\rho_g \beta_g)^2 \left(\sum_j Q(m_i; m_j) f_i f_j \frac{1}{m_j} \right) \quad (19)$$

where f_i represents the size fraction, defined as the proportion of bubble group i in the total GVF ($f_i = \beta_i / \beta_g$); β_i denotes the product of bubble number density and volume for size group i ; B_{Bi} indicates the birth rate caused by breakup of larger bubbles; D_{Bi} represents the death rate due to fragmentation of larger bubbles into smaller ones; B_{Ci} stands for the birth rate from coalescence of smaller bubbles; D_{Ci} signifies the death rate resulting from merging of smaller bubbles into larger ones; $g(m_i; m_j)$ describes the breakup rate of a bubble with mass m_i into two bubbles of masses m_j and $m_i - m_j$; $Q(m_j; m_k)$ denotes the coalescence rate between bubbles of masses m_j and m_k forming a new bubble with mass $m_j + m_k$; X_{jki} is the mass matrix, quantifying the mass fraction transferred to group i from coalescence between groups j and k at time t .

To discretize the size groups, the mass (or diameter) of each size group must be defined. In this work, an equal-mass discretization method is employed, as expressed by the following equation in Eq. (20):

$$m_i = m_{\min} + \frac{m_{\max} - m_{\min}}{N} \left(i - \frac{1}{2} \right) \quad (20)$$

where m is the bubble mass, with subscripts min and max denoting the minimum and maximum values respectively, and N is the total number of size groups.

The minimum bubble diameter for the size groups is set at 0.1 mm, while the maximum is 10 mm. It is basically consistent with the setting in references (Si et al., 2022; Ge et al., 2020; Chen et al., 2019). A total of 10 bubble size groups are established, specifically: 0.1, 0.17, 0.28, 0.46, 0.77, 1.29, 2.15, 3.59, 5.99, and 10 mm. In the calculation, the medium-sized bubble is selected as the inlet bubble size (eg., 0.77 and 1.29 mm). The MUSIG model employs the Luo and Svendsen model (Luo and Svendsen, 1996) for bubble breakup and the Prince and Blanch model (Prince and Blanch, 1990) for coalescence, which can adapt to a variety of working conditions (different pump structures and different operating conditions such as IGVF, rotational speed, and flow rate).

2.4. Boundary conditions

In the Eulerian-Eulerian heterogeneous model for multiphase flow simulations, distinct turbulence models are applied to the gas and liquid phases. The liquid phase is modeled with the shear stress transport (SST) $k-\omega$ model, whereas the gas phase employs the dispersed-phase zero-equation turbulence model. Gas and liquid phase are incompressible fluid. By comprehensively considering the computational complexity, the minimal variation of gas-related parameters inside the pump, and previous research findings, the

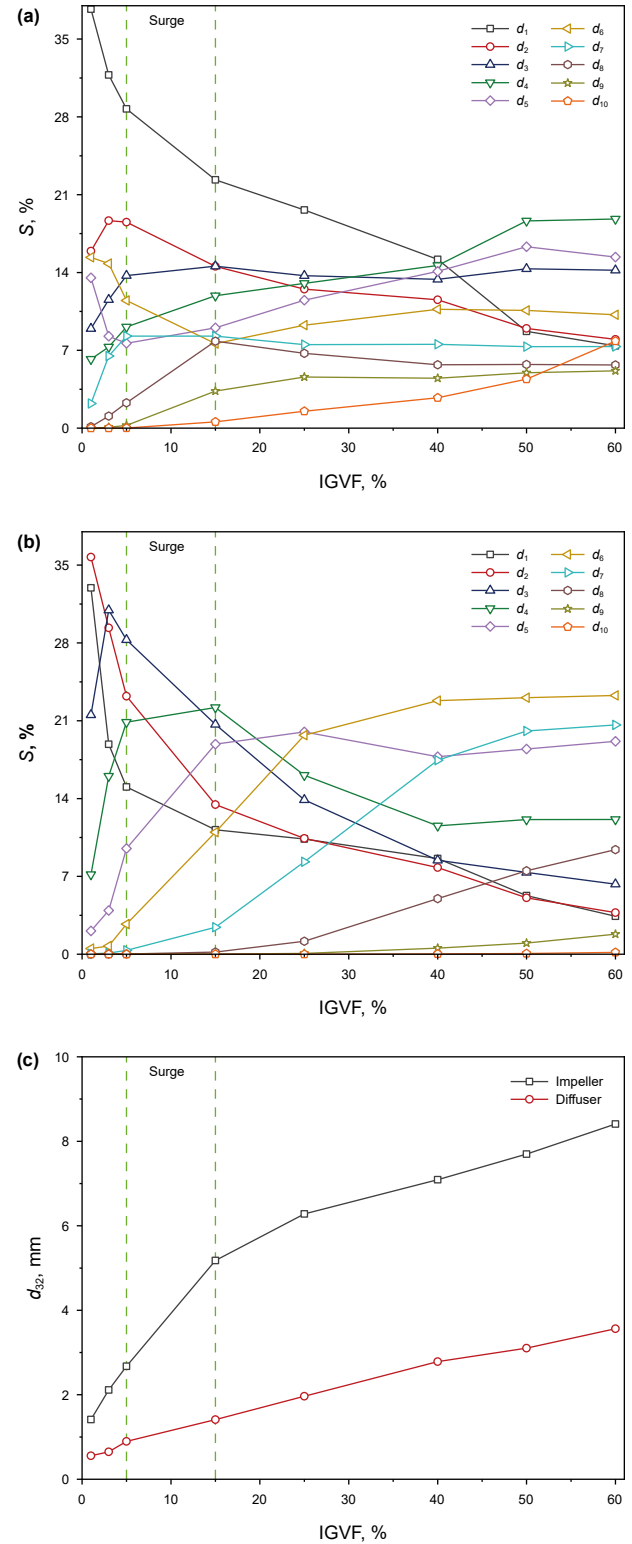


Fig. 12. Comparison of bubble distribution of different sizes in impeller and diffuser under different IGUVs. (a) Bubble distribution in the impeller. (b) Bubble distribution in the diffuser. (c) Average bubble diameter.

compressibility of the gas is not taken into account (Si et al., 2022; Fall et al., 2025; Guo et al., 2024; Zhou et al., 2025). Furthermore, the wake effects behind bubbles lead to increased liquid turbulence induced by larger bubbles in the dispersed phase, which is commonly predicted using the Sato Enhanced Eddy Viscosity model.

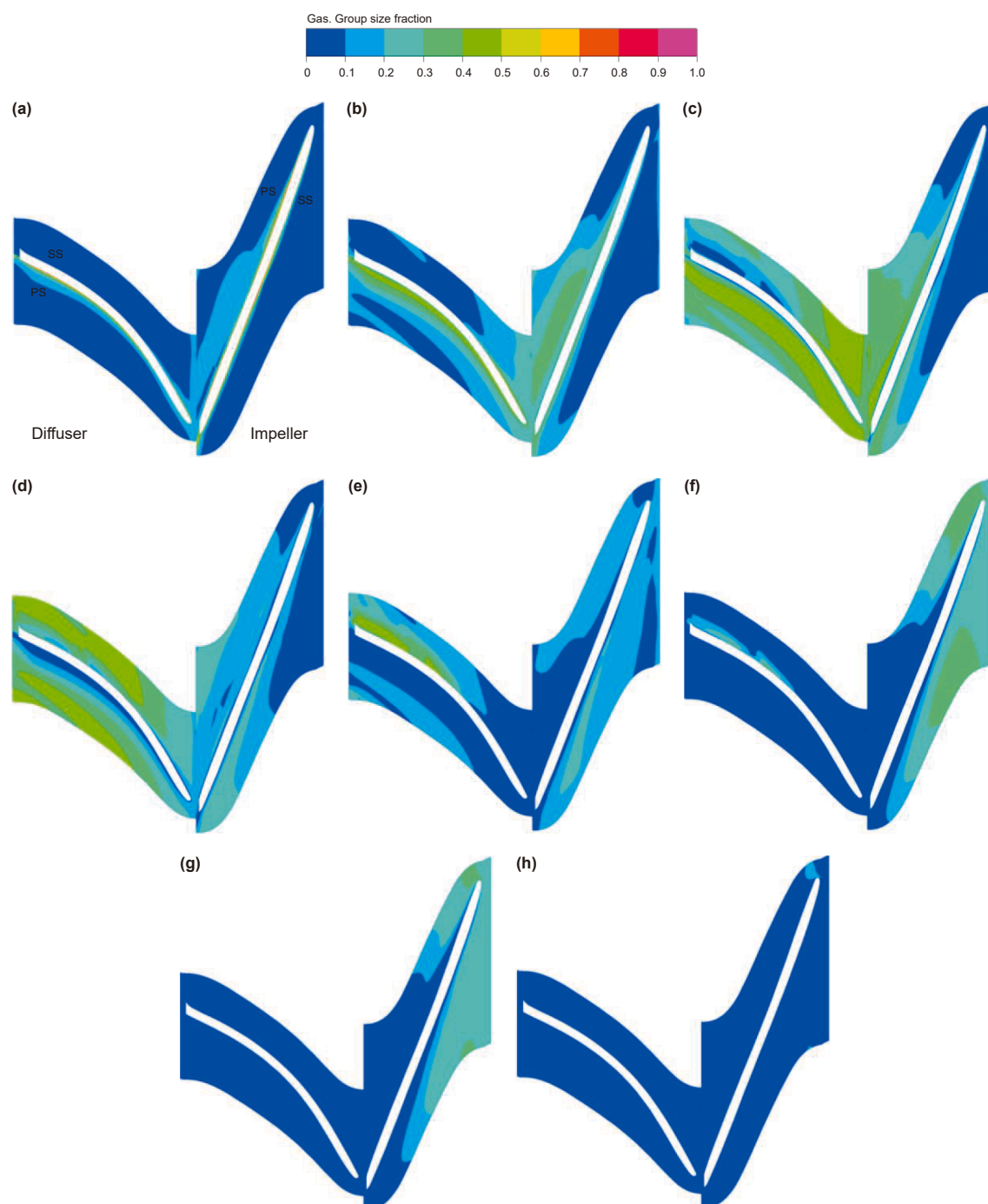


Fig. 13. Bubble distribution of different sizes in the impeller and diffuser when IGVF = 5%. (a) d_1 . (b) d_2 . (c) d_3 . (d) d_4 . (e) d_5 . (f) d_6 . (g) d_7 . (h) d_8 .

The inlet is specified as total pressure with prescribed IGVF and bubble size distribution, whereas the outlet is governed by mass flow. The rotor-stator interface is treated with a Transient Rotor Stator model. Convergence is achieved when maximum residuals fall below 10^{-4} with maximum iteration steps of 10. An unsteady scheme is used, with a time step size of $1/2n$ (where n is rotational speed), corresponding to 3° of impeller rotation per step. The total calculation time covers $900/n$ steps, equivalent to 15 complete impeller revolutions. The convection term is high solution, the turbulence term and the time term are discretized by the first order and the first order backward respectively. Prior to conducting unsteady multiphase simulations, steady-state calculations are first

performed under pure water conditions, with the steady results serving as initial values for subsequent two-phase unsteady computations. The computational conditions comprise both experimental validation cases ($n = 1500$ rpm; $Q_{BEP} = 33$ m³/h; IGVF = 0, 1%, 3%, 5%, 15%, 25%, 40%, 50%) and investigation cases ($n = 3500$ rpm; $Q_{BEP} = 78$ m³/h; IGVF = 1%, 3%, 5%, 15%, 25%, 40%, 50%, 60%).

2.5. Experimental verification

The experiment is conducted on the gas-liquid two-phase integrated test system with adaptive modifications in the State Key Laboratory of Water Engineering Ecology and Environment in Arid

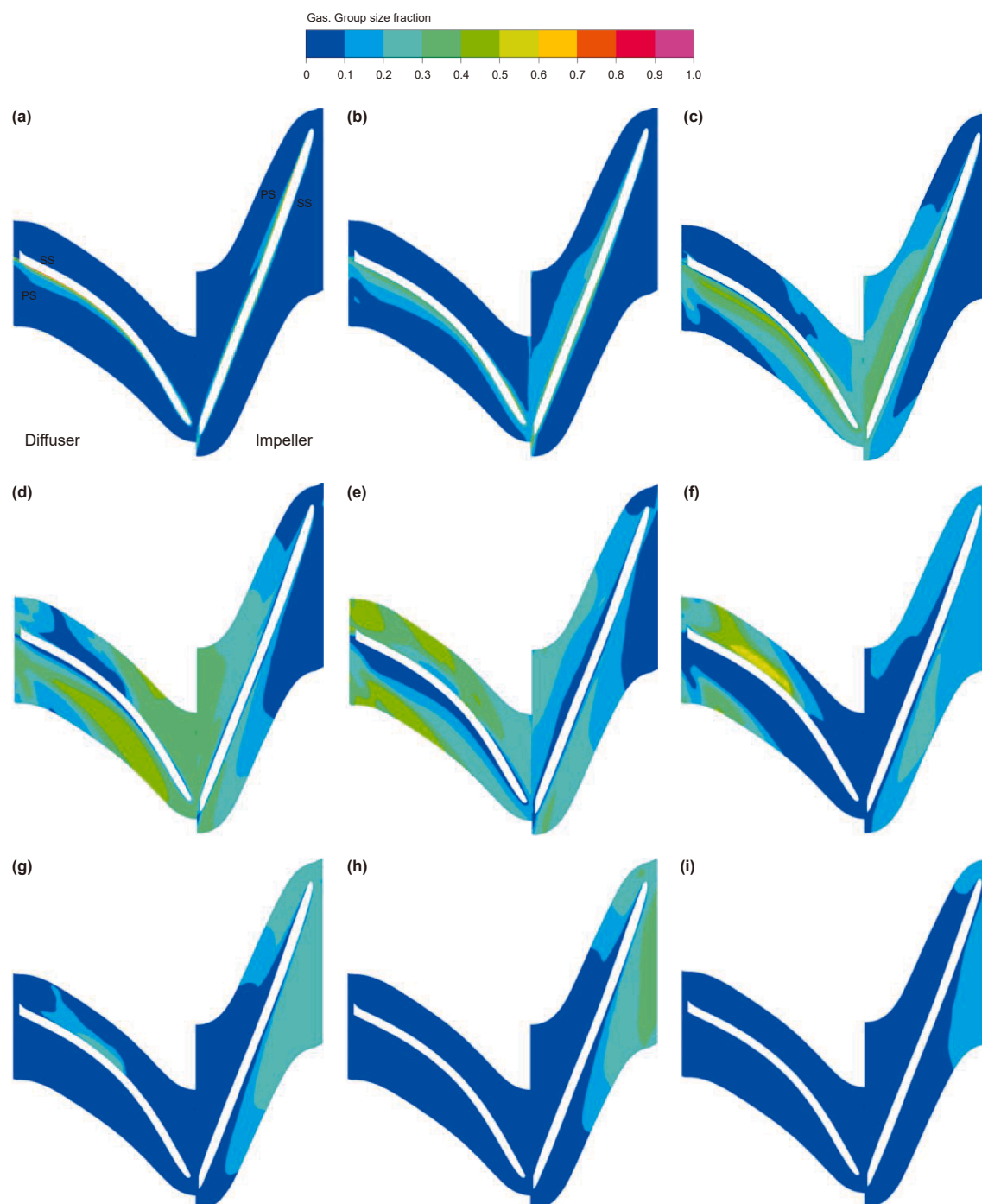


Fig. 14. Bubble distribution of different sizes in the impeller and diffuser when IGVF = 15%. (a) d_1 , (b) d_2 , (c) d_3 , (d) d_4 , (e) d_5 , (f) d_6 , (g) d_7 , (h) d_8 , (i) d_9 .

Area. A self-designed multiphase pump visualization test rig is employed to conduct external characteristic tests, flow visualization experiments, and pressure fluctuation measurements. The experimental system layout is shown in Fig. 5. Fig. 6 presents both the physical model pump and the arrangement of pressure fluctuation measurement points. During the experiments, water from the open water chamber and air from the air compressor are mixed in the mixer before entering the model pump through a long-section pipeline. The discharged flow subsequently returns to the

open water chamber. Table 3 shows the uncertainty of the main instrument in the test.

Fig. 7 compares the numerically simulated and experimentally measured pump pressure increment (ΔP) and gas-phase distribution under gas–liquid conditions at 1500 rpm. As shown in Fig. 7(a), the numerical predictions agree well with experimental data in capturing the decreasing trend of pressure increment with increasing IGVF, particularly the sharp ΔP drop (surge phenomenon) when IGVF increases from 5% to 15%. The flow characteristics

differ significantly before and after surge occurrence. ΔP under pre-surge conditions exhibits only a moderate decrease compared to that under pure water conditions, with relatively similar internal flow field characteristics. In contrast, under post-surge conditions, the internal flow becomes extremely complex with dominant gas-phase effects, causing ΔP to abruptly decrease to merely 50% of the value under pure water conditions.

Numerical errors increase with flow complexity, ranging from about 5% under pre-surge conditions to 6%–10% under post-surge conditions. In addition, the simulation successfully predicts gas pockets in diffuser passages and gas-entrained tip leakage vortex in the impeller at 25% IGVF (Fig. 7(b)). In the numerical results, gas pockets are approximated using an iso-surface with GVF values above or equal to 0.9. The minor discrepancies in gas pocket shape and location contribute to an 8% ΔP error, which is considered acceptable given the extreme complexity of gas–liquid flows (Shi et al., 2018; Zhang et al., 2017), thereby validating the numerical methodology. The simulated results are compared with those of the single bubble size model (generally 0.1 mm, without the MUSIG model). In comparison, the simulation values of both the single bubble size model and the MUSIG model are close to the experimental values when the IGVF is 5% (before the surge phenomenon). When the IGVF is 25% (after the surge phenomenon), the simulation value of the single bubble size model has a large deviation from the experimental value, and the error value is much higher than that of the MUSIG model. The gas phase distribution is also different. The gas phase basically does not accumulate on the suction surface of the diffuser vane. The GVF calculated by the MUSIG model in Fig. 7(c) is higher than that of the fixed bubble size model, particularly inside the diffuser. Compared to the MUSIG model, the fixed bubble size model shows a 43% reduction in the calculated GVF in the impeller and nearly a 50% reduction in the diffuser. Moreover, the area of the iso-surface within the diffuser calculated by the MUSIG model is significantly larger than that of the fixed bubble size model, approximately 3.3 times larger. This is mainly because the bubble breakage and coalescence are not considered in the prediction of fixed bubble diameter.

Fig. 8 presents a comparative analysis of spectral characteristics between numerical simulations and experimental measurements of pressure fluctuations at monitoring point P1 in the diffuser of the multiphase pump under various IGVBs at 1500 rpm. Pressure fluctuation values are normalized using the following dimensionless equation in Eq. (21):

$$C_p = \frac{p - \bar{p}}{\frac{1}{2}\rho_l U_{Ds2}^2} \quad (21)$$

where p represents the instantaneous pressure at the monitoring point; $\bar{p}$ denotes the average pressure over one impeller rotation period; and U_{Ds2} is the circumferential velocity at the impeller outlet shroud, calculated as $U_{Ds2} = \pi n D_{s2} / 60$.

Overall, the numerical results demonstrate good agreement with experimental data in frequency distributions at identical IGVB conditions, with both methods identifying blade passing frequency (BPF, $4f_n$) and its harmonics as dominant frequencies. This phenomenon is likely caused by rotor–stator interactions between the impeller and diffuser. This consistency holds across different IGVBs, further validating the numerical methodology. However, discrepancies are observed between the experimental and numerical results in the amplitude characteristics of pressure fluctuations, particularly in terms of the dominant frequency magnitude and low-frequency amplitude variations. The experimental data show significantly higher fluctuation amplitudes than numerical predictions, possibly due to multiple factors: abundant interference sources during testing (e.g., vibration noise),

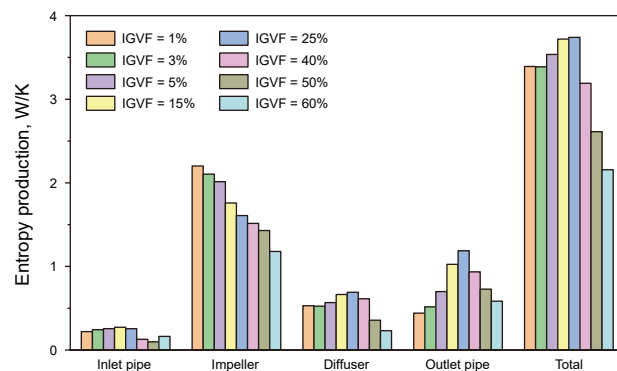


Fig. 15. Comparison of entropy generation of each component at different IGVBs.

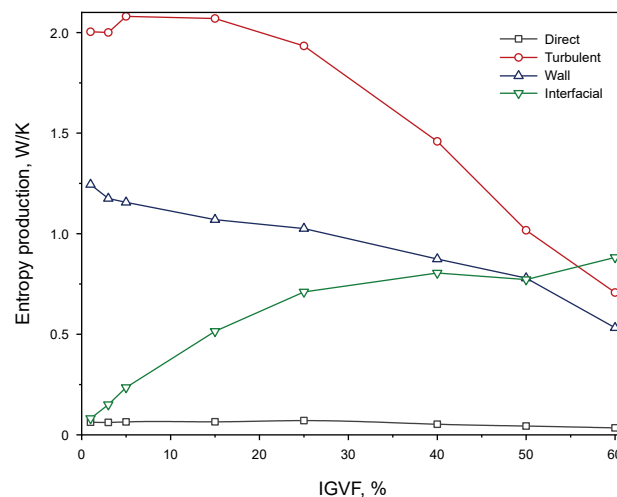


Fig. 16. Change of different types of entropy generation under different IGVBs.

rotational speed fluctuations, and fluid leakage effects in the experimental system. Additionally, the discrepancies could stem from geometric simplifications and modeling assumptions in the computational approach.

3. Results

The IGVB is a critical factor affecting the performance of multiphase pumps. Fig. 9 illustrates the variation of pump pressure increment with different IGVBs at a rotational speed of 3500 rpm and a total flow rate of 78 m³/h. Here, the pressure increment (ΔP_r) is uniformly normalized as: $\Delta P_r = \Delta P / \Delta P_0$, where ΔP_0 represents the pressure increment of the multiphase pump under pure water conditions.

The results show that ΔP_r decreases as IGVB increases, with a particularly dramatic reduction when IGVB rises from 5% to 15%, dropping to merely half of the value under pure water conditions. Consistent with other vane-type multiphase pumps, a distinctive surge phenomenon occurs in this mixed-flow pump when IGVB rises from 5% to 15%. Concurrently, the GVF within flow passages increases with rising IGVB. At low IGVB (1%–5%), the GVF in both impeller and diffuser increases gradually, exerting minimal impact on pumping capacity. However, when IGVB increases from 5% to 15%, the GVF in the diffuser rises to over three times the IGVB value, likely due to the emergence of gas pockets, which severely obstruct water flow and degrade pump performance. At 60% IGVB, the GVF within the impeller exceeds that in the diffuser, resulting

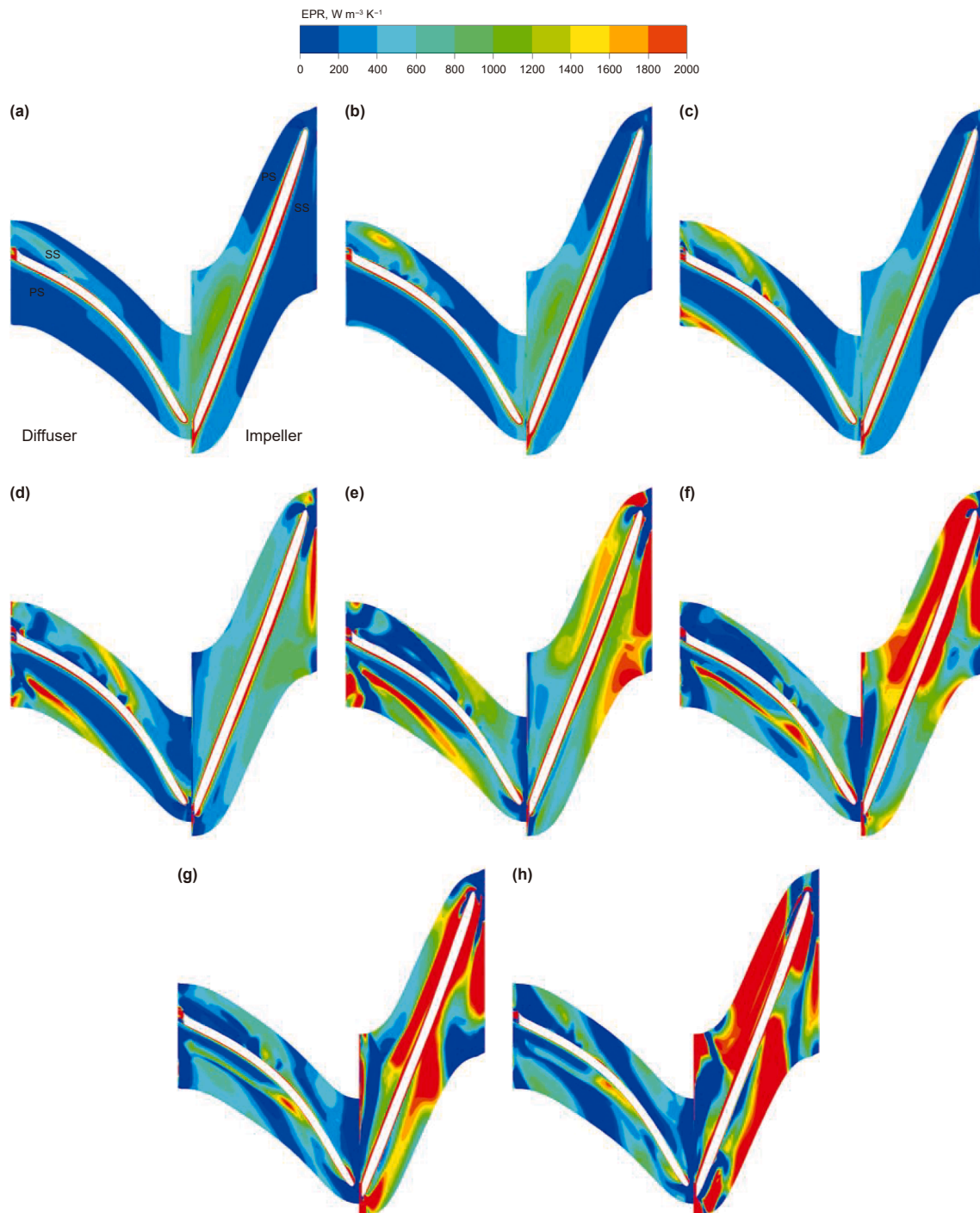


Fig. 17. EPR distribution at the middle section of impeller and diffuser under different IGVPs. (a) IGVP = 1%. (b) IGVP = 3%. (c) IGVP = 5%. (d) IGVP = 15%. (e) IGVP = 25%. (f) IGVP = 40%. (g) IGVP = 50%. (h) IGVP = 60%.

in a substantial reduction of effective flow area. This flow path constriction leads to severe performance degradation of the multiphase pump, as evidenced by a progressive decline in ΔP_r to merely 10% of the value under pure water conditions.

3.1. Effect of IGVP on the gas distribution characteristics

Fig. 10 presents the iso-surface of GVF above or equal to 0.9 in both impeller and diffuser to further analyze the influence of IGVP on gas pocket distribution within the multiphase pump. During

pre-surge conditions (IGVP lower than 5% in Fig. 10(a)–(c)), gas pockets are minimal and primarily form near the diffuser hub, impeller hub, and impeller outlet regions, with small-scale gas pockets also appearing near the tip clearance near the impeller inlet. These gas pockets have a negligible impact on flow capacity. At 15% IGVP in Fig. 10(d), gas pockets predominantly accumulate along the diffuser suction side, significantly reducing the effective flow area, and triggering abrupt deterioration in pressure increment in Fig. 9. When IGVP reaches 40% in Fig. 10(e) and (f), the gas pockets expand substantially, spanning from diffuser inlet to

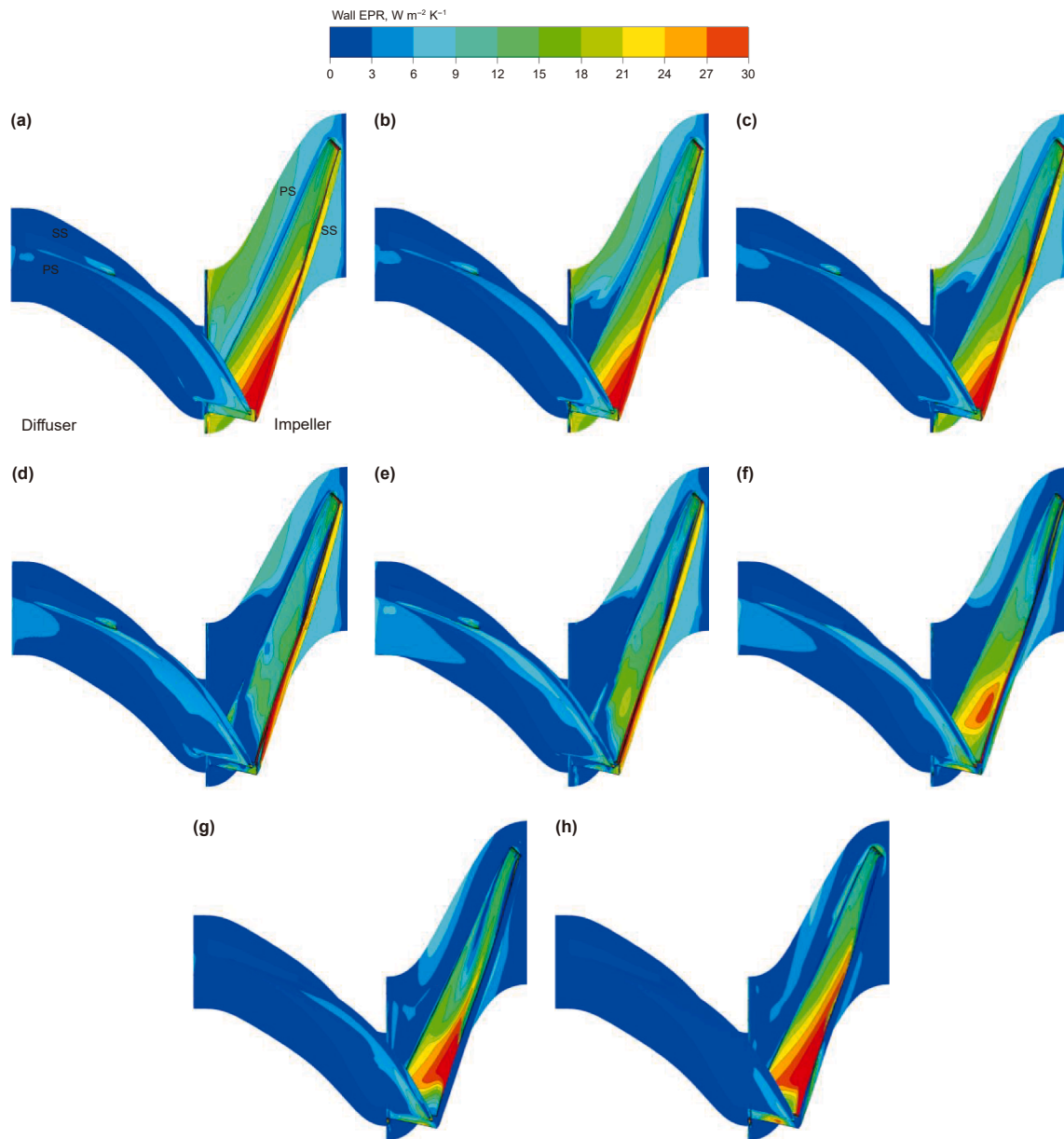


Fig. 18. Wall EPR distribution of impeller and diffuser under different IGUVs. (a) IGUV = 1%. (b) IGUV = 3%. (c) IGUV = 5%. (d) IGUV = 15%. (e) IGUV = 25%. (f) IGUV = 40%. (g) IGUV = 50%. (h) IGUV = 60%.

outlet while extending from suction surfaces toward the middle passage. Further IGUV increase drives gas pocket migration toward pressure surfaces, with further constriction of the diffuser flow area. And the gas pocket appears near the impeller outlet with the gas pocket at the tip clearance enlarging in Fig. 10(g) and (h). Notably, the reduced flow area in the diffuser promotes gas pocket development within the impeller. Consequently, the GVF in the impeller exceeds that in the diffuser, with enlarged gas pockets severely obstructing the flow path, leading to further deterioration of both the pump's pressure-boosting capacity and delivery performance.

The size and shape of gas pockets are closely related to bubble coalescence and breakup behavior. Fig. 11 presents the distribution of different bubble sizes in the impeller and diffuser to further investigate the influence of IGUV on bubble coalescence and breakup, where S represents the volume fraction of bubble size

groups in the flow channel, and d_1 – d_{10} represent the 10 bubble size groups used in the MUSIG model (see Section 2.3). The size proportion of bubble generally presents a normal distribution, and gradually presents an asymmetric distribution with the change of IGUV, which is biased towards large-size bubbles or small-size bubbles. As IGUV increases, the proportion of small bubble groups (d_1 – d_2) in the impeller decreases continuously, while medium (d_3 – d_5) and large bubble groups (d_7 – d_{10}) increase. In contrast, the proportion of small bubble groups (d_1 – d_3) within the diffuser decreases with increasing IGUV, while large bubble groups (d_4 – d_{10}) become more prevalent. Here, this work defines inlet bubble size groups d_5 and d_6 as the threshold between coalescence and breakup, with groups d_1 – d_4 representing bubble breakup and d_7 – d_{10} representing coalescence. Consequently, higher IGUV promotes coalescence while suppressing breakup, leading to an increased proportion of large bubbles in the flow passages which

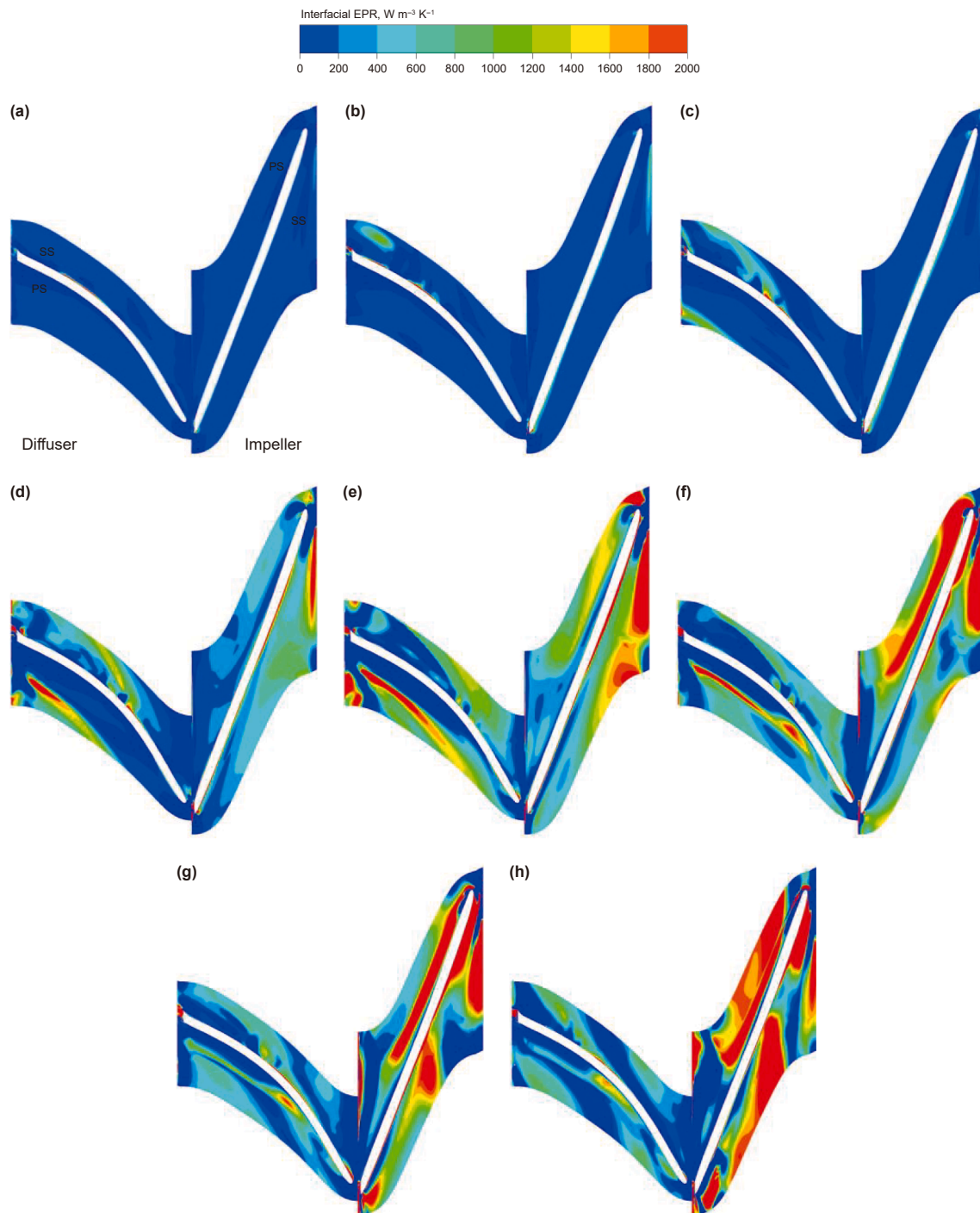


Fig. 19. Interfacial EPR distribution at the middle section of impeller and diffuser under different IGUVs. (a) IGUV = 1%. (b) IGUV = 3%. (c) IGUV = 5%. (d) IGUV = 15%. (e) IGUV = 25%. (f) IGUV = 40%. (g) IGUV = 50%. (h) IGUV = 60%.

readily form gas pockets (see Fig. 10). These gas pockets reduce the effective flow area, ultimately causing the pump's pressure increment to decrease with increasing IGUV.

To more clearly compare the influence of different IGUVs on bubbles of various sizes, Fig. 12 quantitatively illustrates the evolution of both the proportion of different bubble sizes and the average bubble diameter with increasing IGUV. It can be observed that near the surge phenomenon, the proportions of most bubble sizes exhibit significant increases or decreases, particularly for d_4 , d_6 , d_8 , d_9 , and d_{10} in the impeller in Fig. 12(a),

where changes exceed 30%, far exceeding those observed under other IGUV conditions. Similarly, in the diffuser in Fig. 12(b), the proportions of bubbles in groups d_5 – d_{10} also showed a significant increase, with growth rates substantially exceeding 100%. Given the numerous bubble size groups and the varying changes in their proportions under different IGUV conditions, it is challenging to precisely characterize bubble size variations. Therefore, the average bubble diameter (d_{32} , $d_{32} = \Sigma(S_i(d_i)^3) / \Sigma(S_i(d_i)^2)$) is introduced to quantitatively evaluate changes in bubble size. Fig. 12(c) compares the average bubble diameters in the

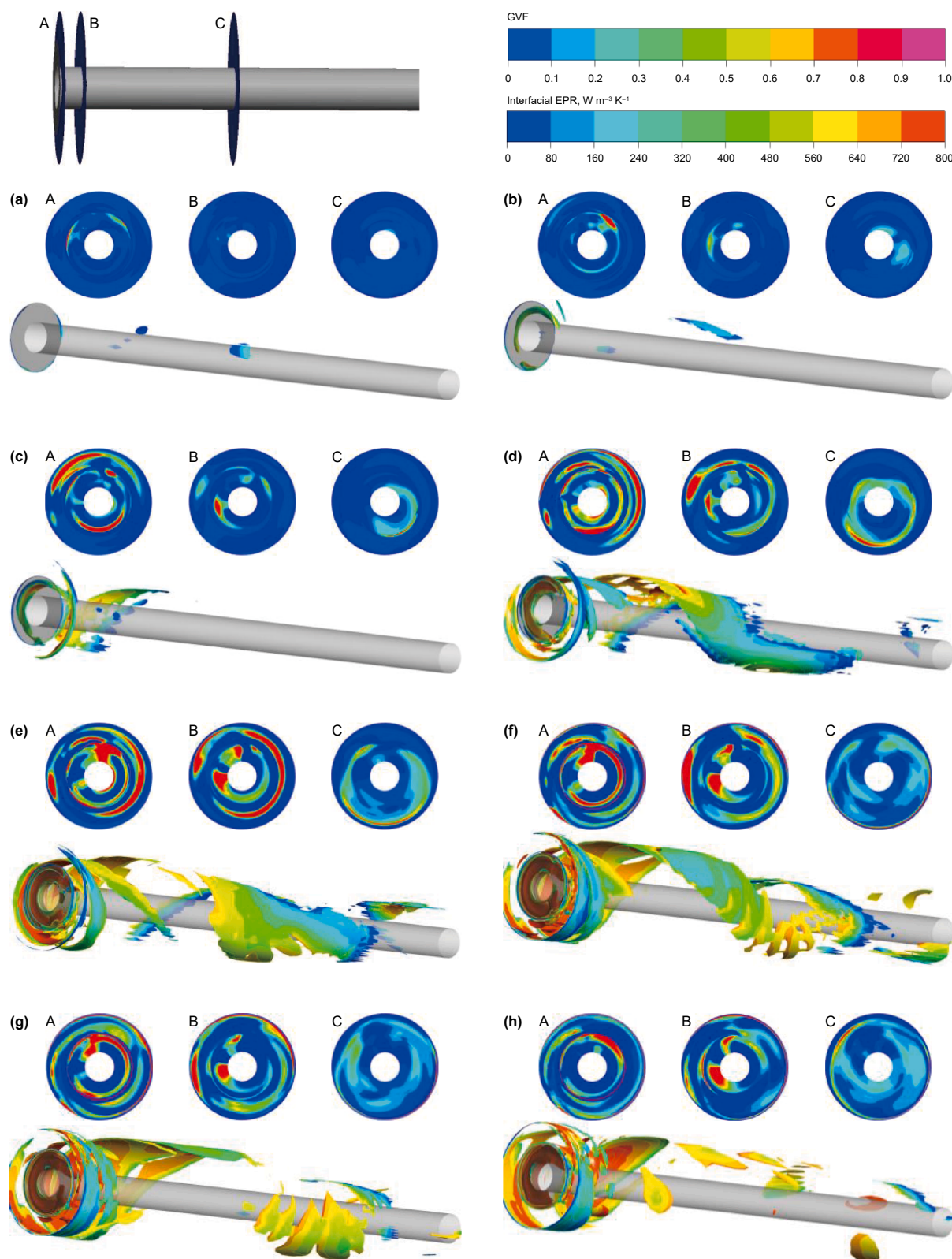


Fig. 20. Interfacial EPR distribution at the outlet pipe under different IGVPs. (a) IGVP = 1%. (b) IGVP = 3%. (c) IGVP = 5%. (d) IGVP = 15%. (e) IGVP = 25%. (f) IGVP = 40%. (g) IGVP = 50%. (h) IGVP = 60%.

impeller and the diffuser. As the IGVP increases, the average bubble diameter in both the impeller and diffuser continues to rise. Notably, when the IGVP increases from 5% to 15%, the

average bubble diameter in the impeller and diffuser increases significantly, from 2.7 mm to 5.2 mm and from 0.9 mm to 1.5 mm, respectively. These represent increases of 94% and 58%,

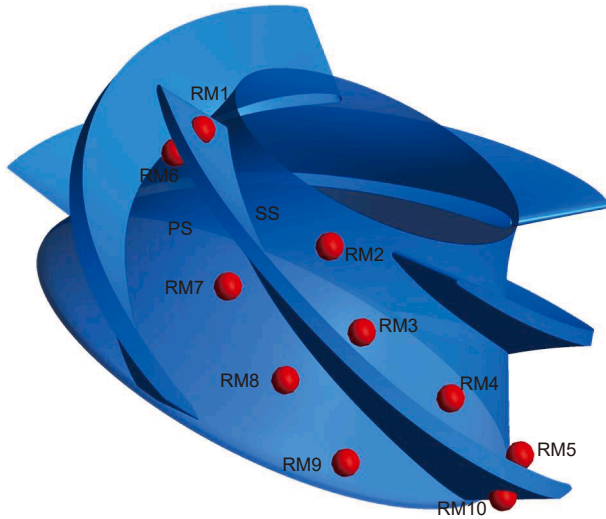


Fig. 21. Monitoring points in the impeller.

far exceeding the changes observed under other IGVF conditions. This finding can provide a microscale qualitative analysis of the emergence of surge.

Figs. 13 and 14 compare bubble size distributions before and after surge occurrence (at IGVF = 5% and 15%) to further analyze the spatial distribution of different bubble sizes within the flow passages. At 5% IGVF, large bubble groups (d_9 – d_{10}) constitute an insignificant proportion (no contour generated), while the d_{10} group similarly demonstrates negligible presence at 15% IGVF. At 5% IGVF, small bubbles (d_1 – d_5) in Fig. 13(a)–(e) are distributed throughout the entire flow passage, while large bubbles (d_6 – d_8) in Fig. 13(f)–(h) concentrate mainly in the impeller without accumulating near diffuser vanes, exhibiting negligible performance impact. When IGVF increases to 15%, the flow passage exhibits distinct bubble size distribution migration patterns. The spatial arrangement of bubble groups (d_2 – d_9) at higher IGVF conditions in Fig. 14(b)–(i) demonstrates remarkable similarity to the distribution of groups (d_1 – d_8) at 5% IGVF in Fig. 13(a)–(h). This observation confirms that the proportion of large-size bubble groups increases with the increased IGVF. Overall, the increased proportion of medium and large bubble groups (d_5 – d_{10}) within the flow passage exhibits a direct correlation with enhanced gas pocket formation propensity. This trend is particularly evidenced by the progressively growing concentration of d_5 – d_7 bubble groups near the diffuser suction sides at higher IGVF. This finding demonstrates complete consistency with the gas pocket development described in Fig. 10.

3.2. Effect of IGVF on entropy generation characteristics

Entropy production theory enables precise quantification of energy loss distribution within pumps and has been successfully applied to single-phase flows in rotating machinery (Chang et al., 2019). Entropy production, which typically exhibiting positive values, consists of two components: thermal entropy generation due to heat transfer and viscous dissipation entropy generation. In hydraulic machinery applications, the thermal contribution is generally negligible; therefore, calculations focus exclusively on dissipation-induced entropy production.

The entropy production rate (EPR), defined as entropy generation per unit mass, must first be determined before obtaining entropy production. Based on Reynolds-averaged equations, the EPR S_D'' can be divided into two components according to Kock in

Eq. (22): the direct EPR caused by mean velocity gradients S_D''' in Eq. (23), and the turbulent EPR induced by velocity fluctuations S_D'' . Since the fluctuating velocity components cannot be directly computed, Kock approximated the turbulent EPR S_D'' in the SST k - ω turbulence model as in Eq. (24) (Kock and Herwig, 2005):

$$S_D'' = S_D''' + S_D'' \quad (22)$$

$$S_D''' = \frac{\mu}{T} \left[\left(\frac{\partial \bar{u}}{\partial y} + \frac{\partial \bar{v}}{\partial x} \right)^2 + \left(\frac{\partial \bar{u}}{\partial z} + \frac{\partial \bar{w}}{\partial x} \right)^2 + \left(\frac{\partial \bar{v}}{\partial z} + \frac{\partial \bar{w}}{\partial y} \right)^2 \right] + 2 \frac{\mu}{T} \left[\left(\frac{\partial \bar{u}}{\partial x} \right)^2 + \left(\frac{\partial \bar{w}}{\partial z} \right)^2 + \left(\frac{\partial \bar{v}}{\partial y} \right)^2 \right] \quad (23)$$

$$S_D'' = 0.09 \frac{\rho k \omega}{T} \quad (24)$$

where $\bar{u}$, $\bar{v}$, and $\bar{w}$ represent the mean velocity components in the x , y , and z directions respectively; T denotes the temperature; ω signifies the turbulent vortex frequency; and k indicates the turbulent kinetic energy.

The wall-induced EPR must also be accounted for, calculated as in Eq. (25) (Kock and Herwig, 2004):

$$S_W'' = \frac{\tau_w \cdot \mathbf{u}_p}{T} \quad (25)$$

where τ_w represents the wall shear stress and $\mathbf{u}_p$ denotes the mean velocity in the near-wall region.

For gas–liquid flows, the EPR calculation can be performed by treating the flow as a homogeneous medium, though this approach requires modifications in Eq. (26) (Wang et al., 2019):

$$\begin{aligned} \mu_m &= \beta_g \mu_g + \beta_l \mu_l \\ \rho_m &= \beta_g \rho_g + \beta_l \rho_l \\ v_m &= \frac{\beta_g \rho_g v_g + \beta_l \rho_l v_l}{\rho_m} \end{aligned} \quad (26)$$

where m represents the mixture of gas and liquid.

Considering the complex interphase force interactions between gas and liquid phases, the interfacial EPR S_I'' is given as in Eq. (27) (Sun, 2014):

$$S_I'' = \frac{2H_c \sigma}{T} \nabla \cdot (\beta_g (1 - c_g) v_{lg}) \quad (27)$$

where H_c represents the mean curvature of bubble surfaces, c_g denotes the gas mass fraction, and v_{lg} signifies the slip velocity.

Integration of the four EPR yields the corresponding entropy generation terms: direct, turbulent, wall, and interfacial entropy production. The total entropy production is then obtained by summing these contributions, as in Eq. (28) (Chang et al., 2019):

$$S = S_D + S_D' + S_W + S_I = \int_V S_D''' dV + \int_V S_D'' dV + \int_A S_W'' dA + \int_V S_I'' dV \quad (28)$$

Fig. 15 compares the entropy production of different flow components across varying IGVFs. The total entropy production initially increases with rising IGVF, peaking during surge conditions at IGVF = 15% and IGVF = 25% due to gas pockets formation. Beyond this point, total entropy production decreases as IGVF further increases to 40%, coinciding with stabilized gas pocket development. The impeller exhibits the highest entropy

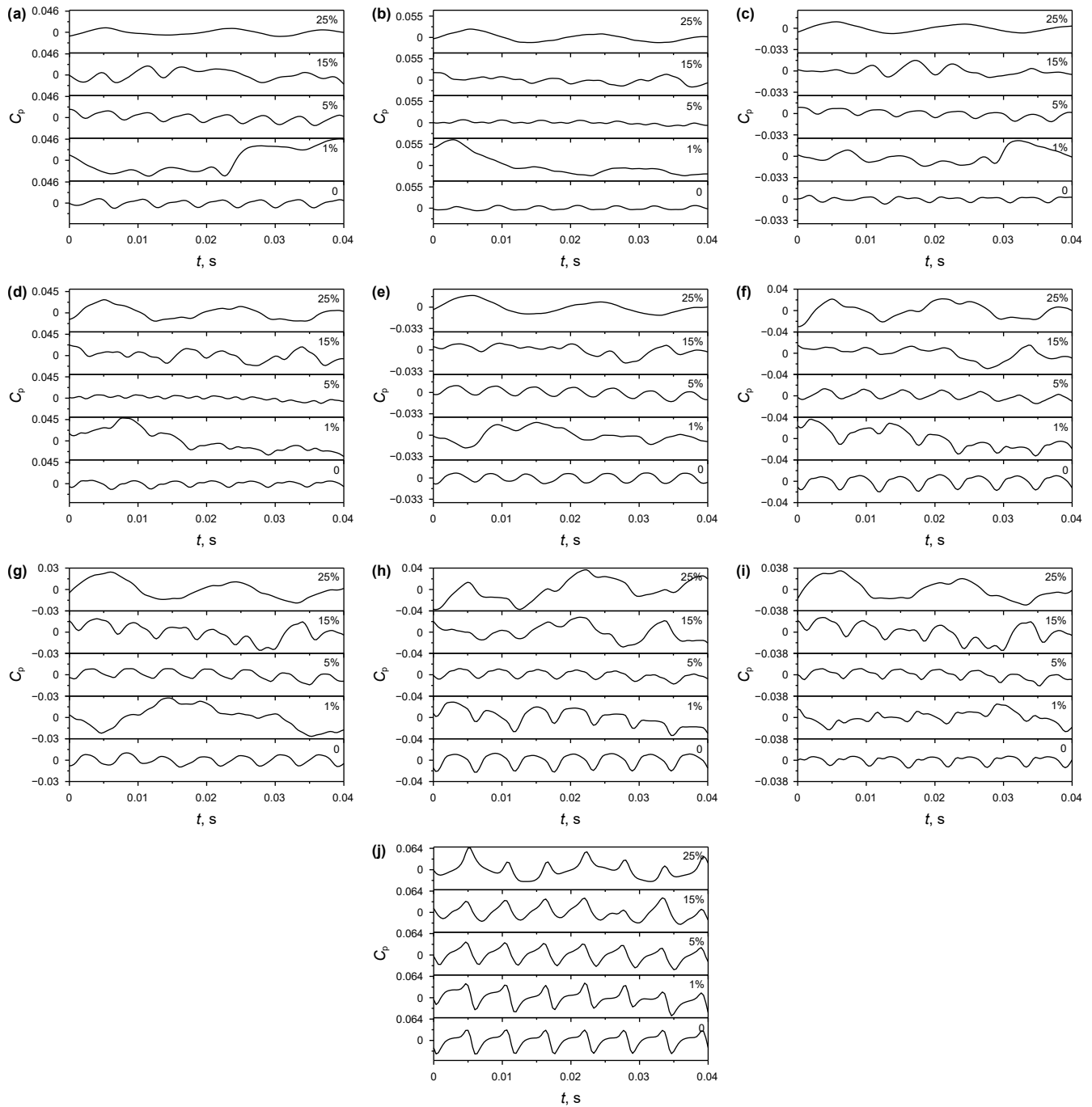


Fig. 22. Time domain characteristics of pressure fluctuation in the impeller under different IGVFs. (a) RM1. (b) RM6. (c) RM2. (d) RM7. (e) RM3. (f) RM8. (g) RM4. (h) RM9. (i) RM5. (j) RM10.

production among all components under all IGVF conditions, while the inlet pipe shows the lowest values, reflecting the impeller's complex internal flow dynamics. Notably, entropy production in the impeller gradually decreases as IGVF increases, whereas that in diffuser and outlet pipe display a progressively increasing trend. This behavior is attributed to flow blockage effects caused by gas pocket accumulation in these components under high IGVF conditions.

Fig. 16 presents the variation of four entropy production types under different IGVF conditions. The observed trends show similarities with single-phase flow, where turbulent entropy production remains dominant in the multiphase pump, followed by wall entropy production, while direct entropy production contributes the least. As IGVF increases, both turbulent and wall entropy production decrease, whereas interfacial entropy production accounting for gas-liquid phase interactions exhibits continuous

growth. A critical transition occurs at 60% IGVF, where interfacial entropy production surpasses both turbulent and wall entropy production, indicating that phase interaction effects become the dominant energy dissipation mechanism. As evident from Eq. (25), the interfacial entropy production explicitly incorporates both the IGVF and gas–liquid slip velocity. Consequently, interfacial entropy production becomes increasingly significant in gas–liquid flows, demonstrating persistent growth with rising IGVF due to progressively intensified phase interactions.

Fig. 17 presents the EPR distributions (excluding wall EPR) at mid-span (span = 0.5) of the impeller and diffuser under different IGVFs, revealing the spatial characteristics of energy dissipation within the flow passages. The results show that EPR in both components increase continuously with IGVF, with consistently higher values observed in the impeller compared to the diffuser. This finding is consistent with the entropy generation of each component in Fig. 15. At low IGVF conditions (1%–5%) in Fig. 17(a)–(c), high EPR values primarily occur near

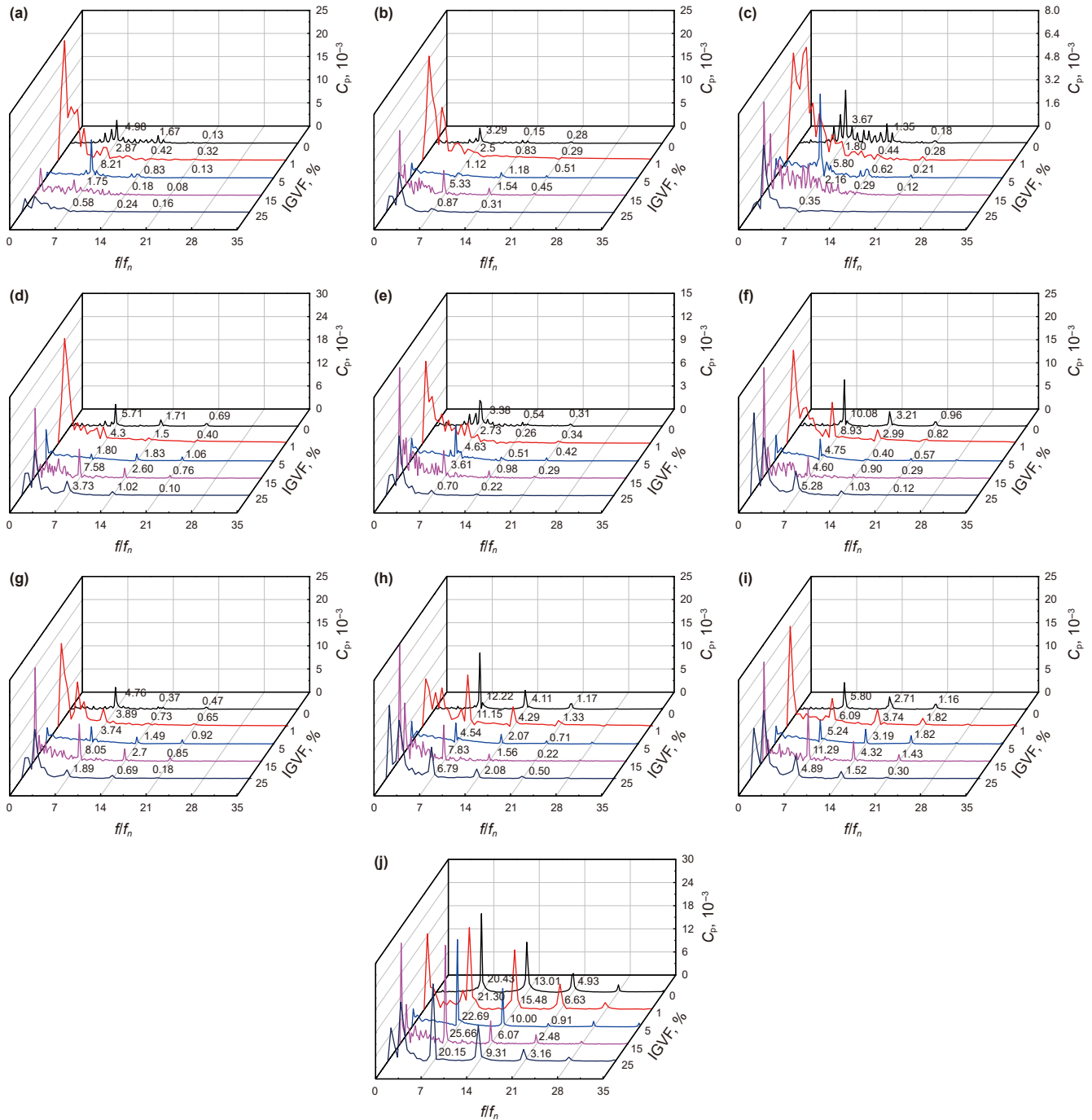


Fig. 23. Frequency spectrum characteristics of pressure fluctuation in the impeller under different IGVFs. (a) RM1. (b) RM6. (c) RM2. (d) RM7. (e) RM3. (f) RM8. (g) RM4. (h) RM9. (i) RM5. (j) RM10.

the impeller or diffuser outlet and along the diffuser suction sides, spatially coinciding with regions of bubble accumulation. As IGVF increases to 15%–40% in Fig. 17(d)–(f), high EPR values appear along diffuser pressure surfaces, as well as at inlet and mid-blade regions of the impeller. The formation of gas pockets that obstruct diffuser passages and subsequently develop in the impeller leads to progressively expanding high EPR zones in Fig. 17(g) and (h) due to flow complexity, reflecting intensified energy losses.

Fig. 18 displays the wall EPR distributions in the impeller and diffuser under different IGVFs. Consistent with Fig. 15, the impeller exhibits significantly higher wall EPR values, particularly near the blade trailing edges and tip regions in Fig. 18(a)–(h). The high wall EPR at blade trailing edges likely stems from rotor-stator interaction effects between the impeller and diffuser, where complex flow patterns at the interface promote gas pocket formation. Furthermore, strong interactions between tip leakage flows and the gas phase contribute to the pronounced wall EPR in tip regions.

Fig. 19 presents the interfacial EPR distributions under different IGVFs to further analyze the energy transfer characteristics at gas–liquid interfaces induced by velocity slip. Exhibiting spatial patterns similar to the total EPR shown in Fig. 17, the interfacial EPR shows stronger correlation with gas–liquid slip velocity. At low IGVF in Fig. 19(a)–(c), interfacial EPR values remain minimal, with slightly high values localized near the outlet of diffuser suction sides. As IGVF increases in Fig. 19(d) and (e), the interfacial EPR intensifies significantly. There are high interfacial EPR areas near the impeller inlet, the middle region and trailing edge of the impeller blades and the pressure surface of the diffuser vanes. Notably, when the IGVF exceeds 40% in Fig. 19(f)–(h), high interfacial EPR zones become dominant throughout the impeller. This phenomenon results from the combined effects of diffuser passage blockage and fully developed gas pockets within the impeller, which enhance phase interaction and velocity slip.

As shown in Fig. 15, the entropy production in the outlet pipe ranks second only to the impeller under high IGVF conditions, indicating progressively destabilized flow with increasing IGVF. Fig. 20 presents the interfacial EPR distribution in the outlet pipe at different IGVFs, visualized using iso-surfaces set at $800 \text{ W m}^{-3} \text{ K}^{-1}$ and color-mapped by GVF. Three representative cross-sections (A, B, C) are analyzed along the pipe from inlet to outlet. At low IGVF (1%–5%) in Fig. 20(a)–(c), the outlet pipe exhibits minimal GVF and correspondingly low interfacial EPR. When IGVF reaches 40% in Fig. 20(d)–(f), the volume of iso-surfaces expands progressively. The iso-surface develops axially from the inlet toward the outlet and radially from the central axis to the wall, forming a distinct spiral pattern. The inlet region maintains both high GVF and interfacial EPR values. At even higher IGVFs in Fig. 20(g) and (h), while the high interfacial EPR zone remains concentrated near the pipe inlet, its intensity gradually diminishes from the middle region toward the outlet.

3.3. Effect of IGVF on pressure fluctuation characteristics

Increasing IGVF promotes gas pocket development within the impeller and elevates flow complexity, resulting in dominant entropy production contribution, continuous expansion of high EPR zones and progressive escalation of energy losses. As the core power component of multiphase pumps, the impeller's pressure fluctuation characteristics fundamentally determine operational stability under gas–liquid flow conditions.

Ten monitoring points are arranged at mid-span within the impeller flow passage shown in Fig. 21 to analyze the pressure fluctuation characteristics in the impeller of the multiphase pump under different IGVFs. Monitoring points RM1 to RM5 are positioned along the blade suction sides, whereas RM6 to RM10 are located on the pressure sides. Furthermore, the monitoring points are uniformly distributed along streamlines covering the leading edge, middle region, and trailing edge of the impeller blades to enable comprehensive analysis of flow characteristics throughout the entire impeller passage. In the calculation, the pressure data of the monitoring point is saved once at each time step. In this way, the actual monitoring time step is $3.33 \times 10^{-4} \text{ s}$, and the corresponding actual monitoring frequency is $1/(3.33 \times 10^{-4}) = 3000 \text{ Hz}$.

Fig. 22 presents the time-domain characteristics of pressure fluctuations at various monitoring points within the multiphase pump impeller under different IGVFs. Analysis of monitoring point locations reveals that pressure fluctuations near the impeller outlet (at the impeller–diffuser interface, RM5 and RM10 in Fig. 22(i) and (j)) exhibit distinct periodicity in the time domain, with seven cycles corresponding exactly to the number of diffuser vanes. This phenomenon results from rotor-stator interactions between the impeller and diffuser (note that monitoring points rotate with the impeller during simulations). For the same streamline positions, the pressure fluctuation coefficients (C_p in Eq. (21)) on the pressure side (RM6–RM10 in Fig. 22(b)–(d), (f), (h) and (j)) consistently exceed those on the suction side (RM1–RM5 in Fig. 22(a)–(c), (e), (g) and (i)), likely due to bubble accumulation near the blade pressure sides. Furthermore, increasing IGVF significantly alters the time-domain characteristics at individual monitoring points. As IGVF rises from 0 to 5%, pressure fluctuation coefficients demonstrate a decreasing trend, potentially because minimal bubble presence in impeller passages at low IGVFs exerts a drag-reducing effect on unsteady flows, thereby reducing fluctuation amplitudes. This trend aligns with experimental observations of pressure fluctuations in the diffuser (Luo et al., 2024). However, the coefficients show marked elevation with fluctuation amplitudes substantially exceeding those under low IGVF or pure water conditions when IGVF increases from 5% to 25%. The time-domain periodicity of pressure fluctuations becomes disrupted, exhibiting cycles that deviate from the seven diffuser vanes (either higher or lower), while the curves demonstrate pronounced asymmetric distributions. These changes conclusively demonstrate the highly unstable flows caused by intensified bubble breakup and coalescence at high IGVF (above or equal to 15%, corresponding to surge conditions in multiphase pumps). Such complex flow patterns not only amplify pressure fluctuation coefficients but also degrade the periodicity and symmetry of pressure fluctuation characteristics, ultimately compromising the operational stability of the pump.

Fig. 23 displays the frequency spectrum characteristics of pressure fluctuations at various monitoring points within the multiphase pump impeller under different IGVFs. Similar to the time-domain behavior, monitoring points closer to the impeller outlet (at the impeller–diffuser interface in Fig. 23(i) and (j)) exhibit more pronounced dominant frequencies (BPF) and their harmonics (specifically at $7f_n$, $14f_n$, and $21f_n$), indicating stronger interactions in this region. For identical streamline positions, the amplitude of dominant frequencies and harmonics on the pressure side (RM6–RM10 in Fig. 23(b)–(d), (f), (h) and (j)) is consistently higher than that on the suction side (RM1–RM5 in Fig. 23(a)–(c), (e), (g) and (i)), accompanied by more significant low-frequency fluctuations. This phenomenon is attributed to complex flow

patterns near the blade pressure sides. Furthermore, increasing IGVF significantly alters the frequency spectrum characteristics. When the IGVF increases from 0 to 5%, amplitudes both at dominant and secondary frequency decrease, consistent with the time-domain trend. However, when IGVF increases from 5% to 25%, pressure fluctuation coefficients show an upward trend due to enhanced bubble breakup and coalescence effects at higher IGVFs. Notably, the low-frequency pulsation becomes more pronounced, with high amplitude at low frequencies, which adversely affects the stability of the pump.

4. Conclusions

In this work, numerical simulations are employed to predict the effect of IGVF on the energy characteristics and flow stability of a mixed-flow multiphase pump. The numerical model and methodology are validated experimentally. A comprehensive analysis is performed on the pump's performance, gas distribution, entropy generation and pressure fluctuation characteristics under varying IGVF conditions. The key findings are summarized as follows:

- (1) As the IGVF increases, the pressure increment of the mixed-flow multiphase pump decreases continuously, while the GVF in both the impeller and diffuser rises. Notably, surge phenomenon occurs when the IGVF increases from 5% to 15%. The GVF in the diffuser surges to over three times the IGVF, accompanied by a sharp decline in pump pressure increment, which reduces to merely half of that under pure water conditions. The average bubble diameter in the impeller and diffuser increases significantly, representing increases of 94% and 58%.
- (2) The increase in IGVF promotes bubble coalescence while suppressing bubble breakup, leading to an increased proportion of medium- and large-sized bubble groups (d_5-d_{10}) that readily form gas pockets. When the IGVF rises to 15%, gas pockets emerge on the suction side of diffuser vanes, expanding in size with further increases in IGVF. When the IGVF exceeds 50%, gas pockets develop within the impeller.
- (3) The impeller exhibits the highest entropy production, with total entropy production peaking during surge occurrences. Turbulent entropy production dominates at low IGVFs but decreases progressively with increasing IGVF. In contrast, interfacial entropy production grows continuously, becoming the primary contributor at 60% IGVF. The pump demonstrates significant energy dissipation, with the iso-surface volume of high interfacial EPR in the outlet pipe expanding persistently in a spiral morphology.
- (4) Near the impeller outlet, pressure fluctuations show distinct periodicity in the time domain, with dominant frequencies ($7f_n$) and their harmonics. As the IGVF rises from 5% to 25%, the amplitude rises sharply, accompanied by a loss of periodicity in the time domain. Low-frequency pulsations intensify with higher IGVF, exhibiting high-amplitude fluctuations that severely degrade the pump's operational stability.

CRediT authorship contribution statement

Zhen-Guo Ge: Writing – review & editing, Writing – original draft, Software, Investigation. **Ren-Bin Qiu:** Writing – review & editing, Software. **Jian-Jun Feng:** Writing – review & editing, Methodology, Funding acquisition, Conceptualization. **Lin-Zhi Zhai:** Validation, Methodology. **Guo-Jun Zhu:** Validation, Formal analysis. **Xing-Qi Luo:** Supervision, Funding acquisition.

Conflict of interest

The authors declare no potential conflicts of interest with respect to the research, authorship, and (or) publication of this article.

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Nomenclature

b_2	blade outlet width, mm
BPF	blade passing frequency
C	force coefficients
C_p	pressure coefficients
d	bubble diameter, mm
D_{s1}	impeller inlet shroud diameter, mm
D_{s2}	impeller outlet shroud diameter, mm
D_{s3}	diffuser outlet shroud diameter, mm
EPR	entropy production rate
F	interphase force, N
f	frequency, Hz
f_n	rotating frequency, Hz
g	gravitational acceleration, m/s ²
GVF	gas volume fraction
IGVF	inlet gas volume fraction
H	head, m
k	turbulent kinetic energy, m ² /s ²
MUSIG	multiple size group
M	force, N
n	rotational speed, rpm
N	mesh number, $\times 10^5$ total number of bubble size groups
p	pressure, Pa
P1, P2	pressure fluctuation monitoring points
PBM	population balance model
PS	pressure side
Q	flow rate, m ³ /h
RM1–RM10	monitoring points in the impeller
S	volume fraction of bubble size groups total entropy production, W/K
SS	suction side
SST	shear stress transport
t	time, s
T	temperature, K
$\bar{u}$, $\bar{v}$ and $\bar{w}$	average velocity along the x , y and z directions, m/s
v_{lg}	slip velocity, m/s
Z	number of blades

Greek symbols

ΔP	pressure increment, kPa
ΔP_r	normalized pressure increment
β	volume fraction
σ	surface tension, N/m
δ	impeller tip clearance, mm
ρ	density, kg/m ³
ω	angular speed, rad/s turbulent vortex frequency, s ^{−1}
μ	dynamic viscosity, Pa·s
η	efficiency, %

Subscripts

BEP	best efficiency point
d	design point
D	drag force
DV	diffuser vane
g	gas phase
I	impeller
l	liquid phase
L	lift force
V	virtual mass force

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