The two-phase flow analysis of helico-axial oil-gas multiphase pump for offshore oil fields

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Abstract. Mixed transportation technology of oil and gas is an efficient transportation method of offshore oil fields, in which oil and gas multiphase pump is the key equipment. In this paper, aimed at a single compression stage unit of a helico-axial multiphase pump prototype, a numerical simulation of internal flow field is carried out. It is assumed that the flow is steady and the two-phase flow pattern is bubble flow. Water is the main phase and incompressible air is the second phase. The change of external characteristics is compared in different inlet gas volume fraction (GVF) condition; the internal flow field characteristics such as velocity fields, pressure fields, and two-phase distributions are discussed as well. The result shows that the differential pressure and the hydraulic efficiency decrease with the increase of GVF, and there are some significant changes in internal flow fields. Further, under the condition of 10% inlet GVF, the changes of external characteristics and internal flow field of the pump when the air bubble diameter varies are compared. The result shows that the differential pressure and the hydraulic efficiency decrease with the increase of bubble diameter.

1. Introduction

As the demand for fossil energy grows in recent decades, especially for oil and gas, the development of offshore oil fields meets more opportunities and challenges. The transportation of oil and gas is a significant technology of the exploitation of offshore oil fields, which mainly includes separated transportation technology and multiphase transportation technology. Compared with the former, the latter has the advantages of smaller wellhead back pressure [1], simpler structure, and smaller investment [2], etc.

Oil and gas multiphase pump is the key equipment of multiphase transportation technology. According to the differences of working principle, it can be divided into vane type and displacement type. Helico-axial oil-gas multiphase pump is a kind of vane multiphase pump, a compression stage unit of which is constituted of a rotated impeller and a static diffuser. Since the long spiral vane makes the flow channel have large curvature radius, the working medium will almost flow along the axial direction, and the gas-liquid separation can be reduced or avoided efficiently as a consequence [3]. Besides, helico-axial multiphase pump has the advantages of large flow, high rotational speed, small size, and high lift [4].
Numerical simulation is a common method to research the helico-axial multiphase pump [5-9]. In this article, based on the Reynolds averaged N-S equations, two-fluid model and steady bubble flow assumption, aimed at a single compression stage unit of a helico-axial multiphase pump prototype, a numerical simulation of two-phase 3D internal flow field in the pump is carried out.

2. Numerical simulation methods

2.1. Governing equations
With water and air acting as the two-phase media, assumed that the two-phase flow pattern is bubble flow and air is incompressible, the continuity equation and momentum equation can be expressed as follows [10-15]:

Continuity equation
\[ \frac{\partial}{\partial t}(\alpha_k \rho_k) + \nabla \cdot (\alpha_k \rho_k \mathbf{w}_k) = 0 \]  

Momentum equation
\[ \frac{\partial}{\partial t}(\alpha_k \rho_k \mathbf{w}_k) + \nabla \cdot (\alpha_k \rho_k \mathbf{w}_k \mathbf{w}_k - \alpha_k \mathbf{r}) = -\alpha_k \nabla p + \mathbf{M}_k + \alpha_k \rho_k \mathbf{f} \]

In the formula, the subscript \( k \) represents any phase (\( k=l \) is liquid phase and \( k=g \) is gas phase); \( \rho_k \) is the k-phase density; \( p \) is the pressure; \( \alpha_k \) is the k-phase volume fraction and meets \( \alpha_l + \alpha_g = 1 \); \( \mathbf{w}_k \) is the k-phase relative velocity; \( \mathbf{r} \) is viscous stress tensor; \( \mathbf{M}_k \) is the phase force of k-phase; \( \mathbf{f} \) is the body force.

2.2. Model and mesh
As is shown in figure 1 and figure 2, the computational object of this paper is the fluid domain of a single compression stage unit of a helico-axial multiphase pump prototype designed by the laboratory, including a single stage impeller, a single stage diffuser, and front and rear extensions.

![Figure 1. Compression unit diagram](image1)
![Figure 2. Computational fluid domain diagram](image2)

For the convenience of mesh generation, unstructured mesh is adopted, which is generated by Meshing in Workbench software. In order to validate the grid independence, the trend how the hydraulic efficiency of the pump changes with the grid number is examined. As is shown in figure 3, when the grid number is larger than 2.67 million, the change of hydraulic efficiency is less than 0.11%. Therefore, this set of grid whose total number is 2.67 million is chosen, which can both ensure the accuracy of the calculation and save the time. The numbers of nodes and grids of each domain are shown in table 1.
2.3. Parameter setting  
Assumed that the flow is steady and the two-phase flow pattern is bubble flow, with water acting as the main phase and 25°C incompressible air acting as the second phase, the effects on external characteristics and internal flow fields of the two-phase flow in the pump of different inlet gas volume fraction (GVF) and different air bubble diameter conditions are examined in this article. The SST model is adopted for continuous phase water, and the zero-equation model is adopted for dispersed phase air. 

The operating parameters of the pump are listed in table 2.

Table 1. The numbers of nodes and grids of each domain

| domain          | nodes   | grids   |
|-----------------|---------|---------|
| Front extension | 188496  | 167188  |
| Rear extension  | 309960  | 275633  |
| impeller        | 333546  | 1329770 |
| diffuser        | 235741  | 896415  |
| total           | 1067743 | 2669006 |

Table 2. The operating parameters of the pump

| Inlet flow (m³/h) | Rotational speed (rpm) | Outlet static pressure (MPa) |
|-------------------|-------------------------|-----------------------------|
| 100               | 4500                    | 0.5                         |

Firstly, with the bubble diameter of 0.3mm remaining the same, the inlet GVF is changed into the values shown in table 3. The effects on external characteristics and internal flow fields of different inlet (GVF) conditions are analyzed in section 3.1.

Table 3. The change of the inlet GVF with the bubble diameter remaining the same

| No. | A.1 | A.2 | A.3 | A.4 | A.5 | A.6 |
|-----|-----|-----|-----|-----|-----|-----|
| Inlet GVF (%) | 0   | 10  | 20  | 30  | 40  | 50  |

Next, with the inlet GVF of 10% remaining the same, the bubble diameter is changed into the values shown in table 4. The effects on external characteristics and internal flow field of different bubble diameter conditions are analyzed in section 3.2 and section 3.3.

Table 4. The change of the bubble diameter with the inlet GVF remaining the same

| No. | B.1 | B.2 | B.3 | B.4 |
|-----|-----|-----|-----|-----|
| bubble diameter (mm) | 0.1 | 0.3 | 0.5 | 0.7 |

3. Analysis of numerical simulation results
3.1. Flow characteristics in different inlet gas volume fraction (GVF) conditions

3.1.1. External characteristics analysis. The differential pressure $\Delta p$ and the hydraulic efficiency $\eta$ are selected as the external characteristics parameters. The differential pressure $\Delta p$ is the total pressure difference between the outlet of the rear extension and the inlet of the front extension:

$$\Delta p = p_{\text{out}}^* - p_{\text{in}}^*$$  \hspace{1cm} (3)

In the formula, $p_{\text{out}}^*$ is the average total pressure of the outlet cross section of the rear extension; $p_{\text{in}}^*$ is the average total pressure of the inlet cross section of the front extension.

The hydraulic efficiency $\eta$ is the ratio between the work that the pump does to the fluid and the shaft work:

$$\eta = \frac{30 \Delta p Q}{\pi n T}$$  \hspace{1cm} (4)

In the formula, $Q$ is the flow of the pump, which is given as 100 m$^3$/h in the study; $n$ is the rotational speed of the pump, which is given as 4500rpm in the study; $T$ is the torque that the rotating components apply to the fluid.

As is shown in figure 4, the hydraulic efficiency and differential pressure decrease with the increase of inlet GVF, and the decreasing trend of hydraulic efficiency is severer than that of differential pressure.

![Figure 4](image_url)  \hspace{1cm} Figure 4. The change of hydraulic efficiency and differential pressure with the increase of inlet GVF

3.1.2. Internal flow fields analysis. According to figure 5, the static pressure gradually increases from inlet to outlet. Due to the influence of centrifugal force, the radial distribution of static pressure in the impeller section is also on an increasing trend, where the pressure at the hub is the lowest and that at the shroud is the highest. With the increase of inlet GVF, since the differential pressure decreases, the static pressure of the inlet cross section increases under the condition that the average static pressure of the outlet cross section is constant.

![Figure 5](image_url)  \hspace{1cm} Figure 5. Average static pressure distributions on meridional plane in different inlet GVF conditions
As is shown in figure 6, the GVF of the suction side is higher than that of the pressure side in the impeller and diffuser channels due to the influence of pressure distribution. As a result of fluid impact, there exists gas-liquid separation at the entrance of impeller, where the GVF increases. Due to the influence of centrifugal force, the GVF is on an increasing trend from shroud to hub in impeller channels. In diffuser channels, owing to the disappearance of centrifugal force and the rectification effect of the guide vanes, the gas and liquid remix.

![Figure 6. GVF distributions on blade-to-blade plane at 50%, 5%, and 95% spanwise in different inlet GVF conditions](image)

3.2. Effects on external characteristics of different air bubble diameter conditions

The differential pressure $\Delta p$ and the hydraulic efficiency $\eta$ are selected as the external characteristics parameters, as is discussed in section 3.1.1.

As is shown in figure 7, with the bubble diameter increasing, the differential pressure $\Delta p$ and the hydraulic efficiency $\eta$ are both on a decreasing trend and the trends are similar. When the bubble diameter increases from 0.1mm to 0.7mm, $\eta$ decreases from 0.416MPa to 0.241MPa, and $\Delta p$ decreases from 57.2% to 43.7%. This suggests that the increase of bubble diameter results in the decrease of the pump's work capacity.
3.3. Effects on internal flow field of different air bubble diameter conditions

3.3.1. Pressure fields analysis. When the diameter of the bubble is small (as is shown in figure 8(a)), the pressure gradient in the impeller and diffuser channels is quite uniform, and the static pressure gradually increases along the flow direction. With the increase of the bubble diameter (as is shown in figure 8), the pressure distribution becomes inhomogeneous, and there is a large area of high pressure region in the junction area of impeller and diffuser.

3.3.2. Velocity fields analysis. When the diameter of the bubble is small (as is shown in figure 9(a)), the flow in the channels is uniform, and the velocity vector is approximately parallel to the blade direction. With the increase of the bubble diameter (as is shown in figure 9), the backflow and vortex phenomenon in diffuser channels increase, which leads to the increase of the local flow loss and thus reduces the pumping capacity of the pump.

3.3.3. Two-phase distribution analysis. When the diameter of the bubble is small (as is shown in figure 10(a), 10(b), 10(c)), the distribution of CVF in the channels is uniform, though the law is satisfied that the GVF increases from shroud to hub. The accumulation of gas in the flow channels is not obvious. With the increase of the bubble diameter (as is shown in figure 10), the gas-liquid separation is more significant. For example, there is a large area of region whose GVF is more than 0.9 at 50% and 5% spanwise when bubble diameter is 0.7mm (as is shown in figure 10(j), 10(k)), while the GVF at 95% spanwise is quite low (as is shown in figure 10(l)). This suggests that the trend is more obvious that the
gas gathers near the hub and the liquid gathers near the shroud, which results in the decrease of external characteristics such as the differential pressure and the hydraulic efficiency.

4. Conclusions
The flow characteristics in different inlet GVF conditions are related to the working performance of the pump, and numeral simulation is a common method to study the flow characteristics. However, the bubble diameter has an influence on the result of the numeral simulation. In this article, the numeral simulation is carried out in different inlet GVF conditions and different bubble diameter conditions.
With the bubble diameter of 0.3mm remaining the same, the inlet GVF is changed. The result of simulation turns out that the hydraulic efficiency and differential pressure decrease with the increase of inlet GVF.

With the inlet GVF of 10% remaining the same, the bubble diameter is changed. The result of simulation turns out that in large bubble diameter condition, the gas-liquid separation is severer, the local flow loss is larger, and the pressure distribution is more inhomogeneous, which leads to the decrease of hydraulic efficiency and differential pressure. The selection of bubble diameter should be consistent with the experimental data.

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