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Numerical investigation on a double suction twin-screw multiphase pump

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Abstract: Based on the dynamic mesh technology, the moving grids of the double-suction screw multiphase pump were generated by the software SCORG. Then the three-dimensional transient simulation model was established to investigate the two-phase flow mechanism inside the screw pump under different inlet gas volume fraction (IGVF). The results show that the pressure inside the working chamber increases step by step from inlet to outlet and is symmetrical. The pressure drops sharply at the inlet of circumferential clearance and then decrease linearly in the clearance. The curved flow in suction and discharge flow passage causes the gas separation and high gas volume fraction (GVF) area in suction and discharge chamber. Four vortices in cross section of discharge pipe cause four high GVF areas due to the centrifugal force produced by the vortices. The high GVF areas scatter in working chambers and change slightly with the rotation angle except the last chamber under the IGVF 10%. The two-phase leakage flow shows layered flow where the gas concentrates at the bottom of the gap. Thus, the high GVF areas on the surface of the rotors mainly exist at the position of gaps. The work will provide instructions for improving its performance under high IGVF.

Key words: Twin-screw pump; multiphase flow; transport characteristics; numerical simulation; dynamic mesh

1. Introduction

With the rapid development of the petroleum industry, to develop onshore sight oilfields, satellite oilfields, and desert oilfields, especially offshore oilfields, has received more and more attention. Traditional phase-separated transportation technology requires complicated oil, gas and water separation equipment, compressors and heat exchangers, and costs a large amount of investments. The multiphase transportation technology sends the mixture of oil and gas directly and many instruments are saved. It can also lessen wellhead rewind pressure; extends well life, and increase oil and gas production, so it is received more and more attention in the industrial field [1-3].

Multiphase pump are the core equipment in multiphase transportation systems. Twin-screw multi-phase pumps have attracted much attention because of their good performance of transporting gas-oil mixture. The twin-screw pumps have the advantages of stable flow rate, simple and compact structure, small pulsation, long life, and reliable operation [4]. K. Egashira of Kyushu University, etc. [5] had carried out research on twin-screw pumps since 1993 and studied the relationship between the internal flow rate and the screw parameters; and the relationship between the backflow and the pressure difference, IGVF, and speed were analyzed using a mathematical model of return flow established by authors. K. Rabiger [6,7] established a thermodynamic model for the performance and working process of multiphase pump, and studied the performance of the pump under high IGVF(90% ~ 99%). Tang and Zhang [8] established a numerical simulation model for a twin-screw pump to analyze the distribution law of pressure field and velocity field. Yan Di et al. [9] developed a numerical simulation model of a screw pump and simulated the transient flow field inside the screw pump; the flow rate, torque, fluid velocity distribution and pressure establishment process, and the influence of viscosity of the working medium on the pump performance were studied. The research team from Xi'an Jiaotong University has also done a lot of theoretical and experimental research on the theory and technology of twin-screw multiphase pump. Cao Feng et al. [10] established a mathematical model of the internal working mechanism of a twin screw mixed pump. The model was used to analyze the flow rate, power, efficiency, and internal pressure changes and their relationships with gas content, pressure difference, inlet pressure, and speed; the tests on the transportation characteristics of twin-screw multiphase pumps at different inlet gas volume fraction conditions had also been done to validate the model [11]. Xia et al. [12] studied the single screw expander and pointed out that the increase of the inlet liquid fraction can reduce the gas phase leakage at the tip clearance. The team from Texas A & M University has done a lot of work on the numerical and experimental research of twin-screw multiphase pump [13-16]. Among them, Partil [13] considered the effect of screw rotor rotation in the steady state CFD numerical calculations using the method of translational reference frame and conducted the calculation of three-dimensional flow field of a twin-screw multiphase pump under the condition of 50% IGVF; the accuracy of CFD model was verified by experimental test; and the effect of bubble size and inlet pressure on tip leakage, tooth root leakage and wall pressure distribution was studied and the results showed that the tooth root leakage flow is mainly gas flow. Shiv [14] established a performance prediction model of twin-screw multiphase pump and indicated that multiphase pumps perform better at high speeds, high IGVF, and low suction pressures. Muhammed [16] analyzed the rotor force of twin-screw multiphase pump based on CFD results.

Due to the movement and deformation of working chambers in screw pumps, it is difficult to generate dynamic meshes for transient numerical simulation. The above scholars mostly use simplified one-dimensional models or steady-state numerical simulation models to calculate the flow field of screw multiphase pump. Although the pressure and speed distribution in the screw pump can be obtained, it cannot predict the flow rate and pressure pulsation accurately. The double-suction screw multiphase pump has two symmetrical pairs of rotors, and its dynamic mesh generation is very complicated. At present, there are no reports on its transient numerical simulation model.

In this paper, a set of moving mesh of a twin-screw multiphase pump was generated based on a dynamic mesh technology, the three-dimensional transient simulation model was established to investigate the two-phase flow mechanism inside the screw pump under different inlet gas volume fraction (IGVF) The work aims to improve the performance of double suction twin-screw multiphase pump.

2. Geometric models and numerical methods

2.1. Geometric models and basic parameters

Figure 1 is a structural diagram of a double suction screw pump. There are two sets of screw rotors on the left and right, which are symmetrical to each other. The fluid flows in from the lower inlet and enters the rotor from the two ends of the pump body in two ways in the casing. Then the pressure of fluid increases as the fluid was conveyed from the two sides to the middle of the pump. After that, the fluid mixes in discharge chamber and flows out through the discharge port perpendicular to the paper. Geometric parameters of single screw rotor are shown in table 1.

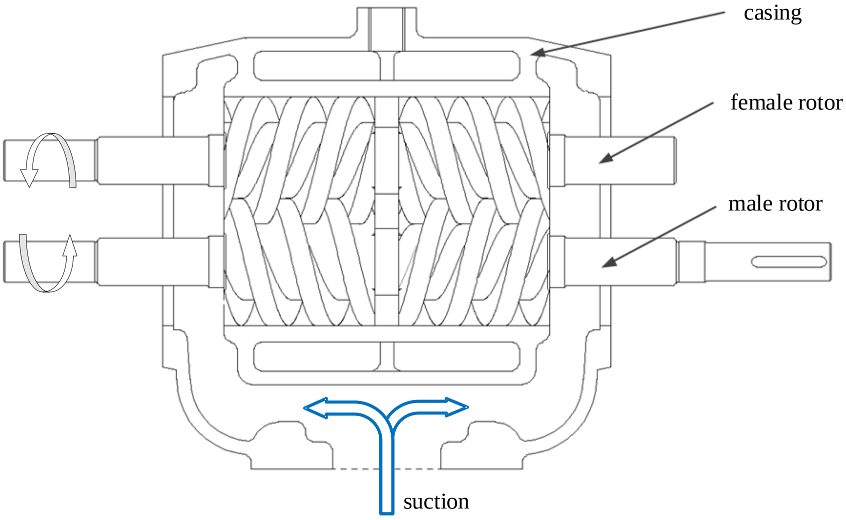

Figure 1. Structure of double suction screw pump

Table 1. Geometric parameters of single screw rotor

parameter	value
Tooth circle radius/*mm*	47
Tooth root radius/*mm*	28
Center distance/*mm*	75
Rotor length/*mm*	111
Theoretical volume/*L*	0.2664

2.2. Mesh generation and import

Screw pump is a positive displacement pump. Its fluid domain moves with the rotation of the rotor. Dynamic mesh technology has to be used to accurately capture the transient changes of the fluid domain.

In addition, the size of the gap between male and female rotors is usually only one thousandth of the diameter of the screw, which needs to be considered to ensure the quality of the gap grid.

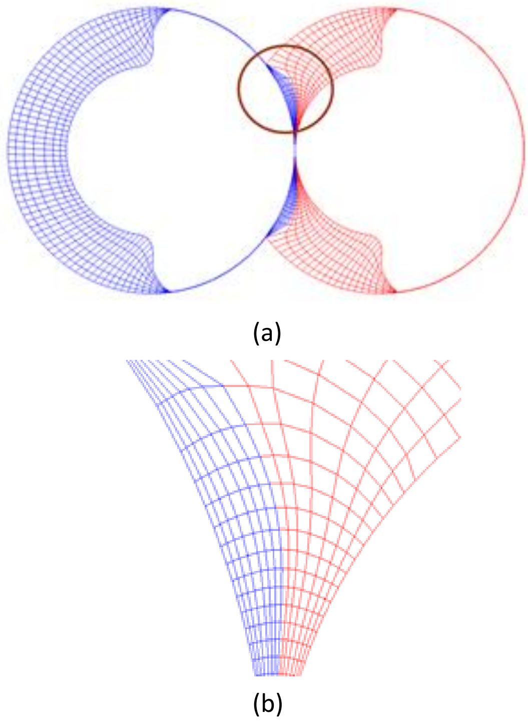

Figure 2. Axial section meshing of screw rotors

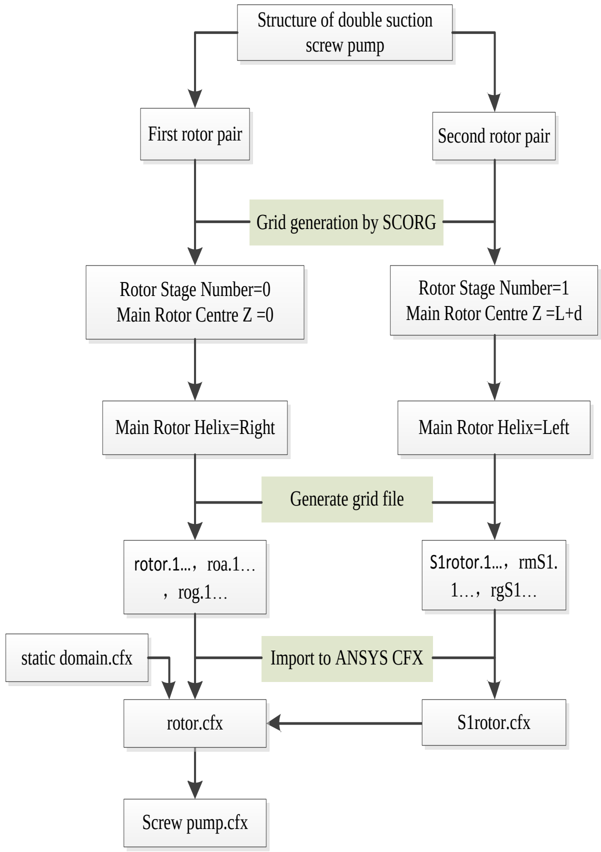

Figure 3. Grid generation method of double suction screw pump

SCORG software [17] can generate single fluid domain dynamic mesh for the screw rotor domain. Figure 2(a) is the mesh division of the axial section of the screw rotor using the rack line method, and Figure 2(b) is a partial enlarged view. This method can generate high-quality grids including the grids in the gaps. In addition, for the inlet and outlet of the twin-screw pump and the area connected to the rotor, ICEM was used for non-structural meshing. The number of rotor grids is 1.596 million and the number of grids in the stationary area is 0.945 million. Because the double suction screw pump has a symmetrical structure, the pump has two sets of rotors with different rotation directions. The grids for both sets of rotors need to be generated at the same time in SCORG to ensure their relative position when the grids are imported in to CFX-PRE. The method is different from that of the single-suction screw pump. Figure 3 shows the specific steps for grid generation and import in CFX.

2.3. Boundary conditions and model settings

Water and air are used as working medium in this paper. When the IGVF and the outlet pressure are low, the gas can be assumed incompressible to simply the solution of the CFD model. For numerical calculation, the SST k-ω turbulence model was selected. The particle model in the Euler-Euler heterogeneous flow model was used to analyze the distribution of gas and liquid phases. The bubble diameter is set to 0.2mm. The model inlet and outlet are set to opening. The reference pressure is 0.1Mpa. The wall surface adopts non-slip boundary conditions, and the inlet turbulence intensity is 5%. The calculation cases are shown in Table 2. The RMS residuals of main parameters are lower than 1e-4 when the solution is convergent. In addition, as the mass flow rate fluctuates with the timestep, so the relative difference between inflow and outflow average mass flow rate should be lower than 3% when the solution finished.

Table 2. Calculation cases of screw pump

Working condition	rotate speed /*r/min*	IGVF /%	Inlet pressure /*MPa*	Outlet pressure /*MPa*
1	1450	0	0	0.8
2	1450	2.5	0	0.8
3	1450	5	0	0.8
4	1450	7.5	0	0.8
5	1450	10	0	0.8

3. Results and analysis

3.1. Pressure distribution

Figure 4 shows the pressure distribution on the surface of the screw rotor and the pressure change of the rotor in the axial direction when the IGVF is 10%. There is almost no change in the pressure in each working chamber of the twin-screw pump. At the tip clearance, the pressure decreases linearly from the high-pressure chamber to the low-pressure chamber. From the two ends of the screw to the middle, the pressure increases step by step and is symmetrical from the left to the right. Therefore, the axial force of the double-suction twin-screw pump can be balanced by itself.

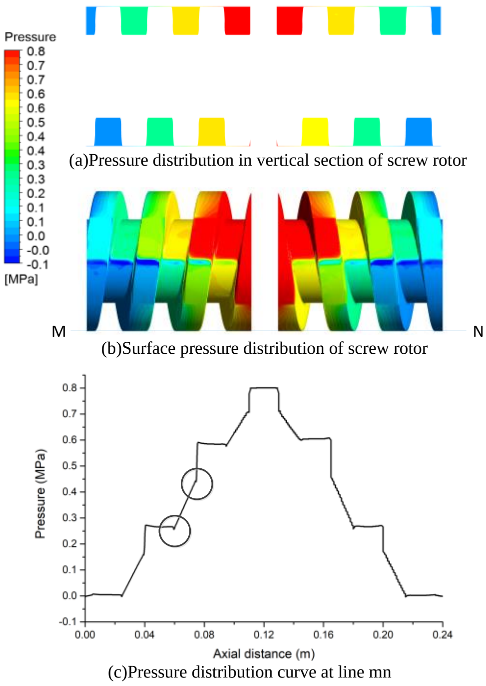

Figure 4. Pressure distribution in screw rotor under 10% IGVF

At the inlet of the screw pump, the pressure is lower than 0 MPa. This is because the suction process of the pump causes a sudden pressure decrease in fluid at the inlet. The same phenomenon occurs at the meshing point. When the working fluid flows through the engagement clearance, a jet generates under the pressure difference between the upstream and downstream, which causes the speed to increase sharply and the pressure to decrease. However, because the cavitation model is not considered, its pressure continues to decrease. Figure 4(c) shows the pressure distribution along the line MN and corresponds to the position in Figure4(b). It can be seen that the pressure change in the working chamber is very slight. From the high-pressure chamber to the entrance of the gap, the pressure drops sharply, and the pressure in the gap decreases linearly. At both ends of the tip space, the curve has two small sharp troughs. This is because the fluid velocity is accelerated quickly when it passes through the gap, and the pressure decreases temporarily.

3.2. Gas-liquid distribution

Figure5 shows the Gas phase distribution in working chamber at 10% IGVF. The mixture fluid firstly flows into the suction chamber through the suction pipe at the middle of the pump. Then the mixture fluid flows toward the two sides of the pump, as shown in Figure1. Gas separation happens along the flow path as shown in Figure5. The flow loss along the suction path will increase when gas separation happens. The gas separation in suction passage influences the gas distribution in working chamber. The high GVF area scatters in the working chamber and is not located near the root of the rotors, which is different from VETTER [18] and PRANG [19] for the assumption of gas-liquid distribution in screw multiphase pumps. One reason for this is the IGVF in the case is low, higher IGVF case needs to be studied. The gas distribution inside the working chamber does not change when the two-phase fluid are transported to the middle of the pump except the chamber next to the discharge chamber. In the last working chamber, the high GVF area increases. From the enlarge view in Figure5, the two-phase leakage shows layered flow. The liquid is on the top under the rotational inertial force while the gas concentrates on the surface of the rotor.

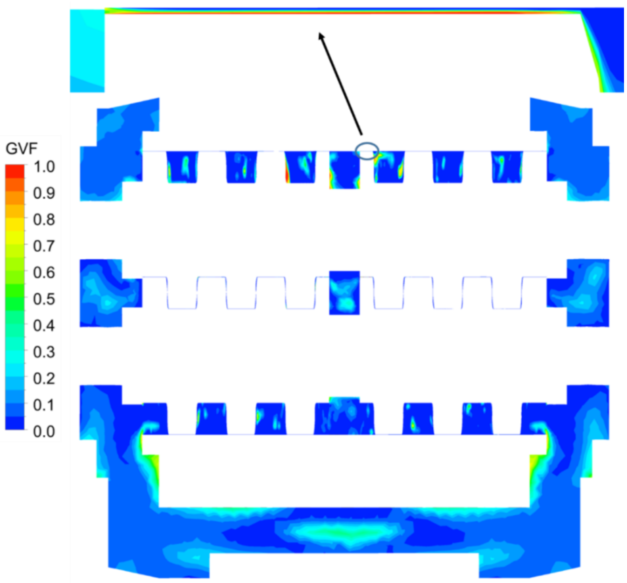

Figure 5. Gas phase distribution of horizontal section at 10% IGVF

Figure6 shows the GVF distribution on rotor surface under different IGVF. The high GVF area mainly locates on the rotor surface of the last working chamber and on the position of leakage gap. As illustrated in Figure5, when the two-phase leakage flow happens, the gas concentrates on the surface of rotor. Therefore, the high GVF area can be seen at the position of leakage gap in Figure5. When the IGVF increases, the high GVF area increase but the distribution law does not change.

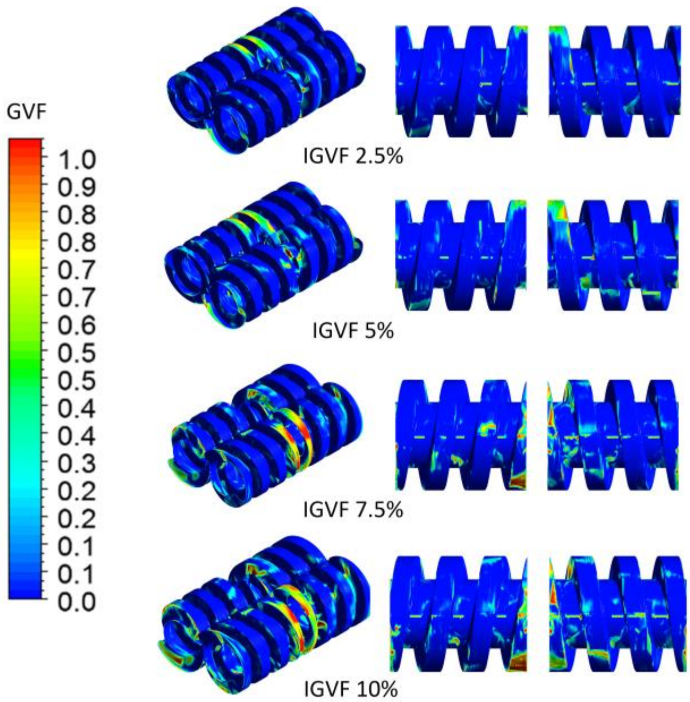

Figure 6. Gas phase distribution on screw rotor surface under different IGVF

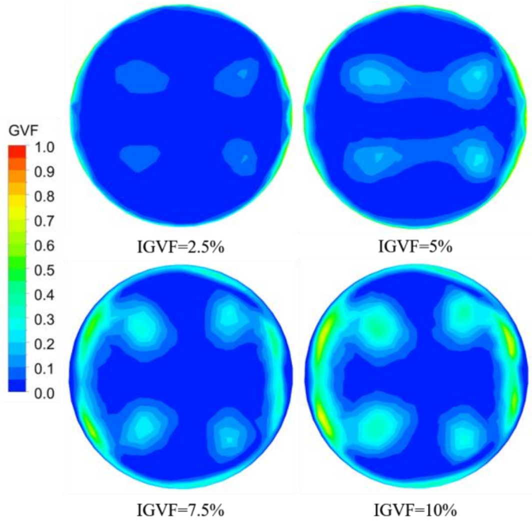

Figure 7. Gas phase distribution in discharge pipe under different IGVF

As discussed above, high GVF area exists in discharge chamber, which indicates that gas separation happens in the discharge chamber. And the liquid cannot bring out the gas fully in discharge pipe, which leads to the local high GVF volume in discharge chamber. Therefore, the GVF distributions in discharge pipe under different IGVF are calculated out, as shown in Figure7. Gas accumulation occurs near the pipe wall, and four high GVF areas appear inside the pipe. Because the discharge pipe is vertical to the plane in Figure1, the two-phase fluid that flows from the discharge chamber to the pipe needs to turn 90 degree. Therefore, the gas accumulates at the outside of the pipe flow. With the increase of IGVF, the gas accumulation is more obvious.

Figure 8 shows the gas velocity vector of outlet section at 10% IGVF. There are four vortices inside the pipeline, and the locations of vortices are the same as that of the high GVF areas inside the pipeline at 10% IGVF in Figure 7. The centrifugal force produced by the vortex makes gas gather in the center of the vortex.

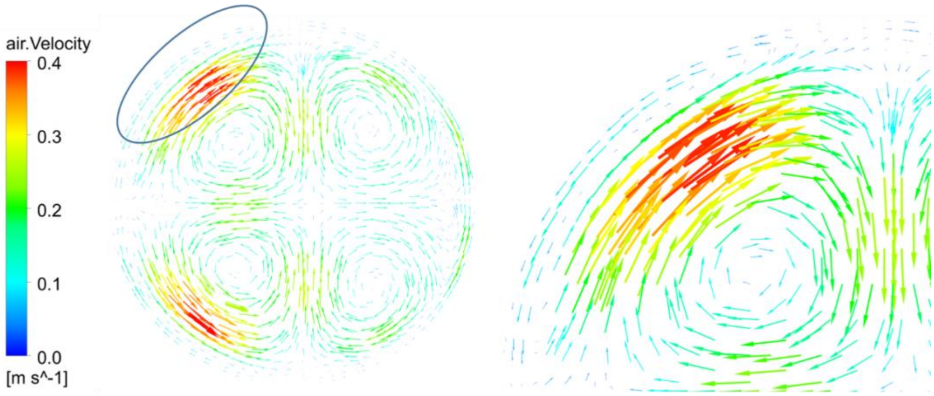

Figure 8. Gas velocity vector of outlet section at 10% IGVF

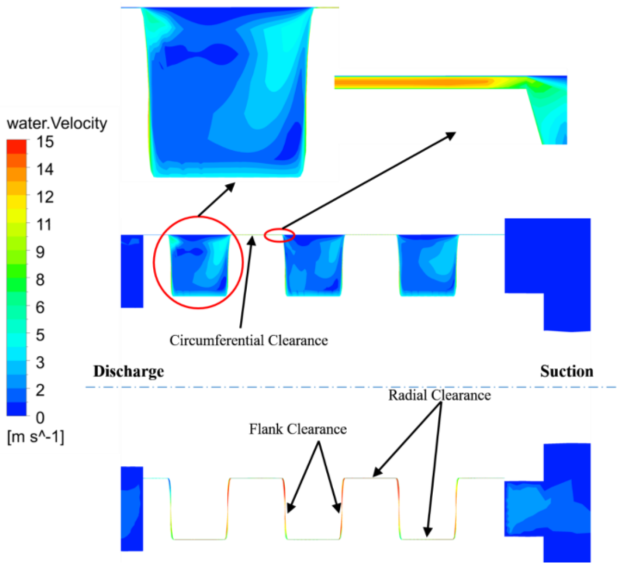

Figure 9. Contour of water velocity of horizontal section at 10% IGVF

3.3. *Velocity distribution*

Figure 9 shows the contour of water velocity of horizontal section at 10% IGVF. The velocity of working chamber is much lower than that at circumferential clearance, radial clearance and flank clearance. And velocity of fluid changes in gradient from wall to the center of the chamber. There are two reasons for the phenomena. One is the wall velocity of rotors, which cause the bigger velocity on the wall. The other is the high-speed leakage flow that happens in gaps and causes velocity gradient in the working chamber. The enlarge view of the velocity field in tip gap is also provided. The maximum velocity is at the center of tip gap.

4. Conclusion

This paper established numerical models of the double-suction screw pump and carry out numerical transient simulations for two-phase flow of the pump. The internal flow field of the twin-screw pump was analyzed. The main conclusions are as follows：

(1) From both ends to the middle of the rotors, the pressure increases step by step and is symmetrical. The pressure distribution in a single working chamber is uniform, the pressure at the entrance of the tip gap decreases rapidly, and the pressure in the gap decreases linearly.

(2) The high GVF areas scatter in working chambers and change slightly with the rotation angle except the last chamber under the IGVF 10%. The two-phase leakage flow shows layered flow where the gas concentrates at the bottom of the gap. Thus, the high GVF areas on the surface of the rotors mainly exist at the position of gaps.

(3) The curved flow in the suction and discharge flow passages causes the gas separation from the fluid. The high GVF area in discharge chamber leads to the severe two-phase leakage flow between the discharge chamber and its next working chamber. Four vortices in cross section of discharge pipe cause four high GVF areas due to the centrifugal force produced by the vortices.

(4) The fluid velocity in working chamber is lower than that near the wall or gap due to the rotor speed and the leakage flow.

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