Multiphase performance and internal flow pattern of helico-axial pumps

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Abstract. While the performances of pumps are highly affected by the gas content, the growing demand for transportation of liquid and gas mixtures in many industrial fields (e.g. oil and gas upstream, chemical plants, food processing, etc.) pushes the research and development of multiphase pumps. In the present work, two types of multi-stage helico-axial gas-liquid multiphase pumps (without and with splitter blade) were developed and experimentally tested both in single and multiphase conditions. The efficiency of the pumps is satisfactory and the head of the pump with splitter blades exceeds the design target. Both pumps accept high gas volume fractions (up to 50%), which is outstanding among the rotodynamic pumps. The differences between the two types give guidance for the further design and optimization of helico-axial pump. Additional insight in the internal flow is gained using spectral analysis of the pressure fluctuations in the impeller.

1. Introduction

Although pumps are used in a variety of industrial applications often including multiphase flows, most pump types are designed mainly for pumping pure water. Recently, increasing demand from the food processing, chemical and oil and gas industries for improved efficiency of equipment pushed the development of specifically designed multiphase pumps, able to accept large variation of gas liquid ratio. The main difficulties for the design of multiphase pump are the blocking of pump passage by the separated two phases and the complex inner flow field.

Various study has demonstrated the inability of centrifugal pumps, (traditionally the most common pump type) to handle gas-liquid two-phase conditions. In the research of Caridad et al. [1], the pump head drops as the gas volume fraction (GVF) increases over the whole operating range, and the relative flow angle also increases with the GVF enlargement. Some multiphase investigations in cavitating, low GVF two-phase flow conditions were also carried out in centrifugal pumps. The results show that the centrifugal pump could only operate on very low gas component conditions, which has GVF lower than 4% [2]. Some other traditional pump types, like the mixed-flow and axial pumps have also been studied under two-phase conditions. In these pumps, the two-phase condition is usually caused by cavitation or a small amount of bubbles in the working material.

In order to achieve the transportation of specific material, the positive displacement pumps (e.g. twin-screw pump) have been widely used in different industrial fields, especially oil and gas [3]. Although the reliability of twin-screw pump under very high GVF makes them the dominant type in the gas-liquid mixture pumping fields, they also have higher costs, low abrasion resistance and low frequency, that limit their improvement [4].
Considering the respective disadvantages of the traditional rotodynamic pumps and positive displacement pumps, a new-designed pump type was developed: the helico-axial pump, which is based on axial rotodynamic pump and compressor. The pump is usually in multi-stage structure, with a rotor and a stator in each stage. As a rotodynamic pump, the axial flow direction avoids the separation of the gas and liquid to a large extent and the larger wrap angle affords more working area and better control capacity in the impeller blades.

Zhang et al. [5-7] have studied the helico-axial pump design experimentally, reaching efficiency up to 40% and a commendable GVF range, while Tremente et al. [8] investigated the influence of attack angle on the two-phase distribution in the helico-axial pump using simulations and experiments.

The orthogonal optimization, neural network, genetic algorithm and direct inverse iteration [9,10] methods have been used to optimize the design of helico-axial pumps. However, the investigation on the helico-axial pump is still insufficient compared to traditional pump types because of the relatively short development time and also complexity of the multiphase phenomena observed.

In the present work, two helico-axial pumps were designed and tested in two-phase test system. The performance curves and pressure fluctuations of the two pumps have been compared to investigate the effect of the design method and the phenomena involved with multiphase pumping.

2. Experimental setup

2.1. Pump structure

Two 3 stages helico-axial pumps (denoted A and B) with the same design flow rate and head \( Q_d \) and \( H_d \), respectively were designed and manufactured.

Pump A was designed based on quasi-three-dimensional design method [11] and the experimental estimation of the pump efficiency. Based on the performances analysis of Pump A, the diameter of Pump B was increased by 7% to improve the pump head the hub curve was also adapted to alleviate the flow separation at impeller inlet and outlet.

Since helico-axial pumps have a narrower passage compared to other traditional pumps, splitter blades are used to increase the flow control and pressure rise without decreasing too much the passage area. For this reason, the splitter blades were added from 40% to 100% of the impeller axial length, with the same blade angle of the main blades, as shown in Figure 1.

Figure 1. Stage structure.
2.2. Test rig

In order to perform the air-water two-phase test, the test rig for the helico-axial multiphase pump should be designed for the two-phase conditions. Based on the traditional water pump test rig, the gas line was added into the two-phase test rig. The test rig is mainly composed of water circling system, gas supplement system, pump section, high speed shooting system, and measurement system. The structure of the test rig is shown in Figure 2. Unlike the traditional test rig, the two-phase test rig has a two-tank structure in the water circling system. The additional tank was designed to help venting gas phase from the mixture. The gas supplement system was driven by a compressor to supply gas phase material in the two-phase tests. The flowmeters, pressure gauges were set on the liquid line and gas line to measure the corresponding parameters of the two phases.

Compared with the single-phase pumps, the inner flow pattern of two-phase pumps is much more complicated. And the flow pattern of the two phases would directly determine the pump performance and stable operation. In order to study the inner flow pattern of the two phases in the pump passages, the pressure distribution was also monitored during the test. Five monitoring points of pressure fluctuation were designed at the third stage impeller, which are distributed in different axial and circumferential positions as shown in Figure 3. The five points are named as P1 to P5, and the order is from the inlet side to outlet side. By the monitoring of transient pressure, the inner flow conditions could be described, and the two-phase behavior and mechanism could be further discussed.

![Figure 2. Structural diagram of the water-air test rig.](image)

![Figure 3. Pressure holes and the installation of pressure fluctuation sensors on the third-stage impeller.](image)
3. Performances parameter

Six main parameters can be measured directly: the liquid flow rate \( Q_l \), gas flow rate \( Q_g \), pressure at pump inlet and outlet \( p_{in} \) and \( p_{out} \), pump rotation speed \( n \), and shaft torque \( M \). The gas volume fraction (GVF) of the mixture could be obtained by the flow rate values of the two phases, as equation (1). Due to the compressibility of the gas phase, the volume flow rate would change as the pressure rises, therefore the inlet GVF is used to characterize the working condition.

\[
GVF = \frac{Q_g}{Q_g + Q_l} \times 100\%
\]

The pressure rise is determined as the pump pressure difference between pump inlet and outlet, equation (2). In single-phase, the pressure rise is usually converted into pump head \( H \), as equation (3).

\[
\Delta p = p_{out} - p_{in}
\]

\[
H = \frac{p_{out} - p_{in}}{\rho g} + \left( z_{out} - z_{in} \right) + \frac{v_{out}^2 - v_{in}^2}{2g}
\]

where \( \rho \) is the liquid density; \( g \) is the gravity acceleration; \( z \) is the height of the corresponding location; \( v \) is the velocity; and the subscript \( in \) and \( out \) denote the pump inlet and outlet respectively.

However, in multiphase flow conditions the density of mixture would change along with the GVF, flow rate and pressure. So, the pump head cannot be processed directly. Considering the Bernoulli equation for compressible flow, the pump head can be calculated as:

\[
H = \int_{p_{in}}^{p_{out}} \frac{1}{\rho g} dp + \frac{v_{out}^2 - v_{in}^2}{2g} + z_{out} - z_{in}
\]

(4)

The \( \rho \) in function (4) is the density of the mixture which is determined as follows:

\[
\rho = \frac{\rho_l Q_l + \rho_g Q_g}{Q_l + Q_g}
\]

(5)

where subscripts \( l \) and \( g \) denote the liquid and gas respectively. For the air-water mixture, the mass flow rate of water is much larger than the flow rate of air \( Q_g \rho_g \ll \ll Q_l \rho_l \), so the pump head can be simplified to function (6)

\[
H = \int_{p_{in}}^{p_{out}} \frac{Q_g}{\rho g} dp + \frac{p_{out} - p_{in}}{\rho g} + \frac{v_{out}^2 - v_{in}^2}{2g} + z_{out} - z_{in} = H_g + H_l
\]

(6)

where the \( H_g \) and \( H_l \) are defined as function (7) and (8)

\[
H_l = \frac{p_{out} - p_{in}}{\rho_l g} + \frac{v_{out}^2 - v_{in}^2}{2g} + z_{out} - z_{in}
\]

(7)
\[ H_g = \int_{p_{in}}^{p_{out}} \frac{Q_s}{\rho g Q_i} dp \]  

(8)

Assuming air as an ideal gas which obeys the ideal gas state equation, \( H_g \) can becomes:

\[ H_g = \frac{p_{out}}{\rho g Q_i} \ln\left(\frac{p_{out}}{p_{in}}\right) \]  

(9)

By the deduction and definition of the two-phase pump head, the power characteristics could be described and compared in different working conditions.

4. Performances of the pumps

4.1. Single phase performance

The two pumps were tested with pure water and water-air mixtures.

The pump head and efficiency curves are shown Figure 4, where \( Q_d \) and \( H_d \) are the design flow rate and head respectively. Pump A has the best efficiency performance at 0.95\( Q_d \) while the total head under design flow rate is 0.65\( H_d \), which could not reach to design target. Compared with Pump A, the Pump B has a higher head level in whole operation range. At design flow rate, the pump head is 1.12\( H_d \), and the best efficiency point is at design flow rate \( Q_d \). Its efficiency is lower than Pump A under partial load and exceeds it at overload conditions. The performance result shows that the design of Pump B could preferably match the design target.

![Figure 4. Performances of Pump A and Pump B.](image)

The use of splitter blades largely increases the power capacity of the impeller and also broaden the high efficiency range of the pump, important characteristics in the downsizing of the multiphase pump. However, splitter blades also increase the friction losses, which decrease the pump efficiency.

4.2. Multiphase performances

The multiphase performances of the two helico-axial pumps are shown in Figure 5 and Figure 6 as a function of the dimensionless liquid flow rate for different GVF.
The highest GVF is about 50% in the two pumps and during the test the pump could sustain a momentary high GVF condition of 100%. It was found that a minimum amount of liquid phase is necessary for stable and continuous operation of the rotodynamic pumps (see also 4.2.2).

The head curves of the mixture under different GVF appear to show the tendency to shift in parallel toward the lower-left corner compared with the single-phase curve. In Pump B, the head curve shows a hump shape in single-phase, and the multiphase curves also inherit the shape characteristics when the GVF is lower than 15%, which means that under low GVF conditions, the mixture is in a quasi-homogeneous state. When the gas component keeps increasing, the phase separation becomes severe, and the head curve shows a different shape.

The shape of the efficiency curves with changed GVF is similar to that of single-phase flow. The best efficiency under multiphase conditions decreases with the increase of gas component.

5. Transient pressure field in the helico-axial pump
For further and thorough understanding of the two-phase mechanism, the transient pressure in the passage was monitored in the test of Pump B. In the test procedure, the pressure fluctuations in five different monitoring points were recorded under all tested working conditions. The sampling frequency...
of the pressure fluctuation sensors is 10240Hz, and the sampling time is 30s for each working condition. Various interesting flow phenomena could be observed by spectral analysis of these signals.

5.1. Pressure fluctuation under single-phase conditions
The single-phase frequency domain of the pump is shown for three typical flow rate values (0.8Q_{BEP}, Q_{BEP}, 1.2Q_{BEP}) under the design rotation speed. in Figure 7 to Figure 9, for the five different measurement points (P1 to P5, in the order from the impeller inlet to outlet) as a function of the dimensionless frequency (normalized by impeller rotation frequency $f_i$).

![Figure 7. Frequency field chart under 0.8Q_{BEP}.](image1)

![Figure 8. Frequency field chart under 1.0Q_{BEP}.](image2)
While the blade passing frequencies of the blades (and their harmonics) and splitter blades are dominant in all the spectra, substantial differences can be observed. Under partial load and at best efficiency point the harmonics of the blade passing frequencies are observed to a much higher frequency, while some nonsynchronized low-frequency fluctuations also emerge at overload condition. The low-frequency noises reflect the disordered flow phenomenon.

The dominant frequency moves to $2f_{BPF}$ and the fluctuation under higher frequency region becomes more obvious. This phenomenon indicates that the high frequency fluctuation dominates the inner flow pattern to a certain extent. The high-frequency fluctuation under larger flow rates may be stimulated by the attack of the flow to the flow components. Apart from the synchronized fluctuations

5.2. Pressure fluctuation under multiphase conditions

5.2.1. Pressure fluctuation under best efficiency points of different GVF conditions. In order to have a comparison of the multiphase fluctuation performance, the pressure fluctuation conditions were analyzed under the test conditions near best efficiency points in different GVF curves (10.3%, 18.5% and 38.5% GVF) as shown in Figure 10 to 12.

![Figure 10. Frequency field chart under GVF=10.3%.]
Since the main components of the signal are still the blade passing frequencies at their harmonics, no acute flow structure or process like the separation of two phases or bubble fall-off is observed in the best efficiency multiphase conditions.

However, the main difference between the single-phase conditions and the multiphase conditions is that the low-frequency noises are more obvious in every monitoring point under multiphase working conditions. This means the two-phase flow generates some irregular vibrations in the pump section.

Apart from the dominant frequency of $f_{BPF}$, another frequency peak is noticeable. The frequency value is $44f_i$, which is the frequency of the impeller-diffuser interaction (diffuser blade number times impeller inlet blade number. In the single-phase conditions, this small peak is not obvious while in multiphase conditions, the peak is much more noticeable, revealing a larger interaction between the flow in the diffuser and impeller.

**Pressure fluctuation under unstable multiphase condition.** Apart from the stable operation conditions, the multiphase pump has some unstable working conditions. In case of moderate liquid flow and GVF=12.5% the pump is under partial load condition with mixture. The pressure fluctuation curves are
shown in Figure 13. It is obvious that the low-frequency noises are much more severe compared with the cases in the previous section. The low-frequency noises are the reflection of unstable flow structure and disorder at pump inlet and outlet region, at where the pressure fluctuation doesn’t depend on the rotation of the pump impellers. The spectrum of the instable condition can be used to estimate the minimum liquid flowrate and explain the instability under lower flow rate conditions.

6. Conclusions
Two helico-axial pumps, Pump A and Pump B, were designed and manufactured. The two pumps mainly differ in the impeller diameter and the usage of splitter blades. A two-phase test rig was established to test the pumps performances, with a two-tank liquid system, gas supplement system and pressure fluctuation monitoring system.

The single-phase performance proves that the introduction of the splitter blades could largely increase the pumping capacity while keep a similar efficiency level. This conclusion gives guidance for the design and downsizing technique of the helico-axial pump.

As working under multiphase conditions, the two pumps both show improved performances. The stable operation range is from 0 to 50% GVF and the highest temporary GVF value could reach to 100%. The pump head and efficiency would decrease as the GVF value increases. The performance curves indicate that the introduction of gas phase would reduce the working range of the multiphase pump and the range would keep decreasing as GVF increases.

Moreover, the pressure fluctuations were monitored and analyzed at five points in the third-stage impeller of Pump B. Under single-phase conditions, the fluctuation amplitude peaks appear at multiples of the impeller rotation frequency, especially at $f_{BPF}$ and $2f_{BPF}$. The three single-phase working conditions have different amplitude distribution in the frequency field diagram.

Under stable multiphase pumping conditions, the main fluctuation peaks still emerge at multiples of impeller rotation frequency $f_i$, which indicates that the main fluctuation is still caused by rotation. The main difference is that the low frequency noises are more severe in the two-phase conditions and the distribution of small peaks is different from single-phase conditions.

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