Research on the characteristics of quasi-steady cavitation in a centrifugal pump

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Abstract. With the pressure decreasing, the process of cavitation in a centrifugal pump could be summarized as incipient cavitation, quasi-steady cavitation and unsteady cavitation. Quasi-steady cavitation is the condition that is between the incipient cavitation and unsteady cavitation in a centrifugal pump. Under this condition, the intensity of cavitation is relatively weak, and the head of the pump almost remains unchanged, but the cavitation exists, causing damage to the impeller by pitting and erosion. So it is important to investigate the quasi-steady cavitation. In this paper, both the numerical and experimental methods had been carried out to investigate the characteristics of quasi-steady cavitation. The internal flow in the pump, the performance of cavitation and the inlet and outlet pressure pulsation of the pump measured through experimental method have been studied under different \(NPSHa\) conditions. It was found that the head decreases about 0.77%-1.38% from non-cavitation condition and it could be regarded as the quasi-steady cavitation. Little change has been found from the internal flow between non-cavitation condition and quasi-steady cavitation condition. The period of inlet pressure pulsation changes from the time that the blade passes by to the period of shaft rotating with the development of cavitation. The dominant frequency of the inlet pressure pulsation is two times of shaft frequency whose amplitudes decrease firstly and then increase to a peak value, followed by a decrease to a low value in quasi-steady cavitation conditions. The dominant frequency of the outlet pressure pulsation is blade passing frequency whose amplitudes increase firstly and then decrease gradually with the decrease of \(NPSHa\).

1. Introduction
Cavitation first occurs locally at the low pressure region in the impeller and develops in accordance with the decrease of suction pressure in a pump [1]. As an important energy transforming device and fluid conveying equipment, centrifugal pumps are widely used in various fields [2]. However, the life of the pump and the stability of the system are greatly affected by cavitation. Not only the noise, vibration and instability of a system can be induced, but also the erosive damage with the surface of flow channels, and the loss of performance in pumps are also produced due to cavitation [3-9].

The main processes of cavitation in a centrifugal pump could be summarized as incipient cavitation, quasi-steady cavitation and unsteady cavitation with the decrease of pressure [10]. Inception of cavitation occurs when the static pressure of the liquid at any location is lower than the saturated vapor pressure of that liquid at the prevailing temperature in the presence of nuclei [1].

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Unsteady cavitation include the cavitation surge (CS), rotating cavitation (RC), asymmetric cavitation (AC) and high-order instabilities [11, 12], which is responsible for the undesirable noise and vibration generation. Quasi-steady cavitation is defined as the condition that is between the occurrence of incipient cavitation and unsteady cavitation in a centrifugal pump. Under this condition, the intensity of cavitation is weak relatively, and the head of the pump almost remains unchanged, but the cavitation has been existed and damaged to the impeller by pitting and erosion. So it is difficult from the cavitation performance to find that the pump is running in the condition of quasi-steady cavitation. Therefore, it is important and necessary to investigate in detail the characteristics of the quasi-steady cavitation.

In this paper, both the numerical and experimental methods have been carried out to investigate the characteristics of quasi-steady cavitation in a centrifugal pump. The internal flow in the pump, the performance of cavitation and the inlet and outlet pressure pulsations of the pump have been studied under different NPSHa conditions.

2. Numerical method

2.1. Research model

Figure 1 shows the computational model for this study. The researched target is a centrifugal pump which is composed of a shrouded impeller with six blades, a single suction, and a spiral volute casing. The impeller is designed to operate at 2910 r/min. The pump’s designed flow rate is \( Q_d = 50.6 \text{ m}^3/\text{h} \), and the designed head is \( H_d = 20.2 \text{ m} \). The specific speed of the pump is \( n_s = 132.2 \) at the designed point \( (n_s = 3.65 \frac{nQ^{0.5}}{H^{0.75}}) \). The shaft frequency is \( f_0 = 48.5 \text{ Hz} \) and the blade passing frequency \( f_d = 291 \text{ Hz} \) of this model pump at the designed point, where \( f_0 = \frac{n}{60} \), \( f_d = Z \left( \frac{n}{60} \right) \). Main geometric parameters of the pump are shown in Table 1.

Table 1. Main pump parameters

| Parameter                  | Sign | Value | Unit |
|----------------------------|------|-------|------|
| Inlet diameter of impeller | \( D_1 \) | 79    | mm   |
| Outlet diameter of impeller| \( D_2 \) | 140   | mm   |
| Inlet diameter of pump     | \( D_3 \) | 65    | mm   |
| Outlet diameter of pump    | \( D_d \) | 50    | mm   |
| Outlet width of blade      | \( b_2 \) | 15.5  | mm   |
| Basic circle of volute     | \( D_b \) | 149   | mm   |
| Total blade wrap angle     | \( \varphi \) | 140   | °    |
| Volute width               | \( b_3 \) | 32    | mm   |

2.2. Computational condition

In this study, the simulations were performed with the commercial software Ansys CFX 14.5, and the \( k-\varepsilon \) turbulence model was selected as turbulence model. In order to simulate the rotational effect, the multiple frames of reference were involved. The internal and external impeller surfaces were chosen as a rotating wall, the speed equals the nominal rotating speed, while all the other walls were stationary. A smooth wall condition was used for the near wall function. The grid interfaces between two stationary components were set as general grid interface (GGI). Between the rotational and stationary components GGI was also used as internal component connection. The boundary conditions of steady flow simulation were almost the same under both cavitation and non-cavitation conditions.
The total pressure at the inlet of the suction pipe was specified, while the outlet boundary condition was set up as mass flow rate. The reference pressure was set to 0 Pa. The calculations continued until the RMS (root mean square) residual dropped below $5 \times 10^{-5}$.

3. Experimental methods

The predictions from the numerical model were subsequently validated with experimental data measured in a closed hydraulic test rig which was shown in figure 3. At the same time, the pump inlet and outlet pressure fluctuations were tested in the condition of quasi-steady cavitation. The pump in the rig was driven at a constant speed of 2900 r/min by an electromotor, with a blade passing frequency $f_d = 290$ Hz. In the experiment, a turbine flow meter installed at the downstream pipe was used to measure the flow rate. The mean static pressure was tested by two pressure transmitter with their ranges were from 0 to 4 bar installed at the outlet of the pump and -1 to 0.6 bar installed at the inlet of the pump. Two transient pressure sensors with the ranges of -100 to 100 kPa and 0 to 0.6 MPa were respectively installed at the inlet and outlet of the pump to monitor the pressure pulsations. The inlet pressure and flow rate were adjusted by the vacuum pump and the flow control valve, respectively. A PXI 4472B data acquisition module which was made by National Instruments Company was applied to capture the electric signals and convert them to digital signals.

4. Results and discussion

4.1. Quasi-steady cavitation

Figure 4 shows the power spectral density (PSD) signals of pump inlet pressure pulsations which were measured through experiment at designed flow rate $(Q_d=50.6 \text{ m}^3/\text{h})$. Obvious changes in pump inlet pressure pulsations PSD signals have occurred starting from the $NPSHa=6.94$ m as the result of
development of cavitation. The amplitudes of inlet pressure pulsation PSD signals decrease sharply at 5 times (about 250 Hz) and 6 times (blade passing frequency) of shaft frequency. However, the magnitudes increase suddenly at 3 times (about 150 Hz) and 4 times (about 200 Hz) of shaft frequency, and obvious discrete pulsations occur near the frequency. So \( NPSHa = 6.94 \text{ m} \) can be regarded as the point of cavitating inception. According to the reports of Y. Tsujimoto and T. Kimura, et al [13, 14], \( NPSHa = 3.07 \text{ m} \) can be regarded as the starting point of unsteady cavitation in the pump. Therefore, the pump is running under the condition of quasi-steady cavitation when \( NPSHa \) decrease from 6.94 m to 3.47 m.

Figure 4. Waterfall plot of pump inlet pressure pulsation at \( Q_d \).

4.2. Cavitation performance of quasi-steady cavitation

Figure 5 shows a comparison between experimental and numerical results of the suction performance curves at \( Q_d \). The horizontal axis shows the Available Net Pressure Suction Head which is defined as \( NPSHa = (p_{inlet}-p_v)/\rho g \), where \( p_{inlet} \) is the inlet pressure and \( p_v \) is the vapor pressure. The vertical axis shows the head of the pump. A rational agreement between the experiments and the predictions can be found. The predicted accuracy was estimated to be within 5% of express for the flow regimes calculated numerically. The existed deviation between the experimental and numerical results may caused by the unavoidable testing error in the experiment. On the other hand, the roughness of the wall were not taken into account in the process of calculation may also played a critical role to the deviation. Compared with non-cavitation condition, the head of the pump decreases about 0.77% to 1.33% under the condition of quasi-steady cavitation. Hence, the head of the pump almost remains unchanged under the quasi-steady cavitation and it is difficult to find out the condition according to the suction performance.

Figure 5. Suction performance curves of the pump at \( Q_d \).
4.3. Analysis of internal flow characteristics of quasi-steady cavitation

Figure 6 has compared the distribution of water turbulence kinetic energy between non-cavitation and quasi-steady cavitation conditions in middle cross section of impeller at $Q_d$. It is clear that almost few changes can be found from the distribution of water turbulence kinetic energy between non-cavitation condition ($NPSHa=9.87$ m) and quasi-steady cavitation condition ($NPSHa=3.47$ m). Some turbulent kinetic energy distributes at the surface of each blade suction side which is near the outlet of the impeller, but the intensity of it in the impeller is weak. Owing to the influence of high speed rotating blades, the turbulent kinetic energy which is distributed at the interface between the blades and the volute is strong relatively. In the outlet of the volute and the tongue region, there is strong water turbulent kinetic energy existing, which indicates that the flow is relatively complex there under both non-cavitation and quasi-steady conditions.

![Figure 6. Distribution of water turbulence kinetic energy in middle cross section of impeller at $Q_d$.](image)

4.4. Pump inlet pressure pulsation characteristics of quasi-steady cavitation

Figure 8 has shown the time domain of pump inlet pressure fluctuations under the quasi-steady cavitation condition at $Q_d$. The horizontal axis shows the time that the impeller rotates 1 revolution. The vertical axis shows the amplitude of pump inlet pressure pulsation. It is obvious that with the process of cavitation developing from non-cavitation to quasi-steady cavitation condition, not only the period of pump inlet pressure fluctuations change from the time that the blade passed by to the period of shaft rotating, but also the intensity of the fluctuations gradually become weak. It is considered that vapor bubbles in the pump play an important role in enhancing the buffering effect to the pump inlet liquid.

Figure 9 shows the PSD signals of pump inlet pressure fluctuation under the quasi-steady cavitation condition at $Q_d$. It is clear that the dominant frequency of pump inlet pressure fluctuation is 2 times of shaft frequency (about 97 Hz) whose magnitudes decrease first and increase to a peak value ($NPSHa=4.39$ m), then decrease to a low value. Meanwhile, the magnitudes of PSD signals of pump
inlet pressure fluctuations also increase to the maximum value first at $NPSHa=4.39$ m and then decrease to a low value owing to the development of cavitation.

![Figure 7](image1.png)
(a) $NPSHa=9.87$ m  
(b) $NPSHa=3.47$ m  
**Figure 7.** Distribution of water velocity in middle cross section of impeller at $Q_d$.

![Figure 8](image2.png)
**Figure 8.** Time domain chart of pump inlet pressure pulsation at $Q_d$.

![Figure 9](image3.png)
**Figure 9.** Power spectral density chart of pump inlet pressure fluctuation under quasi-steady cavitation condition.
4.5. Pump outlet pressure pulsation characteristics of quasi-steady cavitation

Figure 10 has shown the time domain of pump outlet pressure fluctuations under the quasi-steady cavitation condition at $Q_d$. The horizontal axis shows the time that the impeller rotates 1 revolution. The vertical axis shows the amplitude of pump outlet pressure pulsation. It is clear that in both non-cavitation condition and quasi-steady cavitation condition, the pump outlet pressure fluctuations present a law of sine function. It can be found that the six blades passing through the tongue cause six cycles in each impeller revolution, which is a typical pressure fluctuation distribution caused by rotor-stator interaction [10]. Compared with the cavitation-free condition, the intensity of the fluctuations gradually become strong in the quasi-steady cavitation condition due to the development of cavitation.

![Figure 10. Time domain chart of pump outlet pressure pulsation at $Q_d$.](image1)

Figure 10. Time domain chart of pump outlet pressure pulsation at $Q_d$.

Figure 11 shows the PSD signals of pump outlet pressure fluctuation under the quasi-steady cavitation condition at $Q_d$. It can be found that the dominant frequency of the outlet pressure pulsation is blade passing frequency (about 290 Hz) whose amplitudes increase to the maximum value first at $NPSHa=4.39$ m, and then decrease with the decrease of $NPSHa$. At the same time, not only some discrete signals of the PSD exist, but also the subpeak occurs near the shaft frequency when $NPSHa=4.39$ m. The magnitudes of the PSD signals of outlet pressure pulsation decrease gradually near the 2 times of shaft frequency with the development of cavitation.

![Figure 11. Power spectral density chart of pump outlet pressure fluctuation under quasi-steady cavitation condition.](image2)

Figure 11. Power spectral density chart of pump outlet pressure fluctuation under quasi-steady cavitation condition.
5. Conclusion
The internal flow characteristics and pump inlet and outlet pressure fluctuations characteristics of the centrifugal pump under the condition of quasi-steady cavitation have been investigated through experimental method and numerical simulation. Several conclusions can be drawn as following:

1) The head of the pump decreases about 0.77% to 1.33% from non-cavitation condition can be considered as the quasi-steady cavitation condition, corresponding to the $NPSHa$ decreases from 6.94 m to 3.47 m.

2) Nearly few distinctions can be found from the characteristics of internal flow between non-cavitation and quasi-steady cavitation condition.

3) Not only the period of pump inlet pressure fluctuation changes from the time that the blade passed by to the period of shaft rotating, but also the intensity of the fluctuations gradually become weak with the process of cavitation developing from non-cavitation to quasi-steady cavitation condition. In contrast, the intensity of pump outlet pressure fluctuations become strong due to the decrease of $NPSHa$.

4) The dominant frequency of pump inlet pressure fluctuation is 2 times of shaft frequency whose magnitudes decrease first and increase to a peak value at $NPSHa=4.39$ m, then decrease to a low value. The dominant frequency of pump outlet pressure pulsation is blade passing frequency whose amplitudes increase to the maximum value first at $NPSHa=4.39$ m, and then decrease gradually with the decrease of $NPSHa$.

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