Experimental Investigation on Unsteady Pressure Pulsation in a Centrifugal Pump With Special Slope Volute

Rotor–stator interaction, a major source of high amplitude pressure pulsation and flow-induced vibration in the centrifugal pump, is detrimental to stable operation of pumps. In the present study, a slope volute is investigated to explore an effective method to reduce high pressure pulsation level, and its influence on flow structures is analyzed using numerical simulation. The stress is placed on experimental investigation of unsteady pressure pulsation inside the slope volute pump. For that purpose, pressure pulsations are extracted at nine locations along the slope volute casing covering sensitive pump regions. Results show that distinct pressure pulsation peaks at f_{BPF}, together with nonlinear components are captured. These peaks are closely related to the position of pressure transducer and operating conditions of the pump. The improvement of rotational speed of the impeller results in rapid increase of pressure fluctuation amplitude at f_{BPF} and corresponding root mean square (RMS) value within 10–50 Hz. A comparison with conventional spiral volute pump is implemented as well, and it is demonstrated that slope volute contributes significantly to the decline of pressure pulsation level.

Keywords: centrifugal pump, slope volute, unsteady pressure pulsation, experimental validation

1 Introduction

Unsteady pressure pulsation induced by rotor–stator interaction has a direct impact on stable operation of the centrifugal pump. Even at nominal flow rate, flow discharged from the impeller matches the geometry of volute to a favorable extent, but pressure pulsation still occurs due to the interaction between the impeller and the volute [1]. Consensus has been attained that flow field distribution is not circumferentially uniform in the volute, and the unevenness is closely associated with not just the geometrical design of pump components but also operating conditions. Some studies have been dedicated to unsteady pressure pulsation by either experimental or numerical method, and most of them laid their emphasis upon the influence of geometrical parameters on pressure pulsation characteristics [2]. In essence, the principal purpose of probing into unsteady pressure pulsation is to open the possibility of modeling the intense rotor–stator interaction between the impeller and the volute tongue. Furthermore, an effective approach should be developed to control pressure pulsation during pump design, which is sorely necessitated.

Spence and Aaral-Teixeira [3] numerically investigated the effect of four geometric parameters, namely blade tip clearance, vane arrangement, shroud-to-casing radial clearance, and sidewall clearance, on pressure pulsation in a centrifugal pump. And the contribution of the four parameters is thereby ranked, as is remarkably important for pump design. Yao et al. [4] analyzed time-frequency characteristics of pressure pulsation in a double-suction centrifugal pump using fast Fourier transform (FFT) algorithm and time-frequency representation methods. Under off-design conditions, some unexpected flow phenomena occurring inside pumps have considerable effect on pressure pulsation. Barrio et al. [5,6] predicted flow pulsations associated with rotor–stator interaction using numerical simulation for different flow rates. The study facilitated a relation between geometrical parameters of blade passage and pressure pulsations. And both tangential and radial velocity components at reference locations in near-tongue region were taken into account as well. Parrondo et al. [7] also investigated the effect of operation point on pressure pulsation in a centrifugal pump, and particular attention was paid to pressure amplitude at blade passing frequency. Increasing blade tip clearance serves as an effective measure for reducing pressure pulsation, as illustrated by Yang et al. [8,9]. Besides, to lower pressure pulsation in the centrifugal pump, specifically devised impeller and volute structure such as the splitter blade and double volute have been presented [10,11]. However, relative position of the volute tongue with respect to the impeller has rarely been examined. Therefore, the major aim of this study is to seek an effective method to alleviate rotor–stator interaction by using the slope volute and then to investigate the unsteady pressure pulsation characteristics under various working conditions. Similar volute structure has been employed in hygienic pumps [12], but relevant pressure pulsation has not been addressed.

In present study, unsteady pressure pulsation characteristics in a centrifugal pump with slope volute are investigated from shutdown to maximum flow rates. Pressure pulsation signals are obtained with nine fast-response pressure transducers mounted on the slope volute casing. Detailed analysis of pressure spectrum is performed, and special attention is paid to pressure pulsation peaks at blade passing frequency and some nonlinear interaction frequency components. Meanwhile, the influence of rotational speed on pressure spectrum is investigated. Finally, to validate the low pressure...
pulsation characteristic of the slope volute, pressure pulsation in a centrifugal pump with conventional spiral volute is measured for comparison.

2 Experimental Setup

The single-stage model pump with an axial suction is equipped with an impeller incorporating six backward curved blades. Figure 1 presents a structure comparison of the slope volute and conventional spiral volute. For the diffusion part of the slope volute, a deviation of 15 deg from the vertical axis is adopted based on optimal results [13]. In Fig. 2, radial dimensions of the slope volute casing are identical, and cross-sectional area of the volute passage develops along axial direction. Eight cross-sectional areas of the two volutes are compared in Fig. 3, which shows small difference between the two volutes. Meanwhile, the inlet and outlet areas of the two volutes are equivalent. And the two impellers are identical as well. Besides, the same blade tip clearance is adopted for the two pumps to ensure comparable experimental results. In Fig. 1, it also can be seen that the tongue of the slope volute locates at the right side of the impeller, so the flow emanating from the impeller does not interact directly with the volute tongue. But the tongue of the spiral volute faces the impeller immediately. Therefore, it is inferred that intense rotor–stator interaction in the slope volute pump would be attenuated obviously relative to the spiral volute pump, which had been validated in previous study through numerical simulation [14].

Experiments were conducted in a closed-loop test rig to guarantee measurement accuracy, as shown in Fig. 4. Flow rate of the model pump was measured by electronic flowmeter with an absolute accuracy of 0.2%. The head of the model pump was measured with an uncertainty of 0.1%. Flow rate of the model pump could be regulated by means of a gate valve installed at the inlet of the tank. The pump was driven by an alternating current (motor, and a frequency inverter was used to adjust the rotational speed of the model pump continuously. The rated rotational speed of the model pump is 1450 r/min, which is kept invariant from shutoff to maximum flow rate conditions. But at some special tests (variable rotational speed experiment), the rotational speed would be adjusted.
casing, as shown in Fig. 5. All the transducers are located at the midspan of impeller outlet. From p1 to p7, the angle between two consecutive transducers is 15 deg, and from p7 to p9, this angle increases to 45 deg. Frequency tailoring and high resonant frequency (≥500 kHz) enable an extremely wide available frequency range (beyond 0–100 kHz) of the pressure transducers. The response time of the transducer is less than 1 μs, and the uncertainty of measured signals is usually lower than ±0.2%. In the centrifugal pump, frequencies excited by hydrodynamic phenomena usually fall into a low frequency range, which is normally narrower than 1 kHz. During signal sampling process, the sampling frequency was set as 10 kHz to satisfy Nyquist sampling theorem, so the detailed information associated with the original ones is well retained in obtained signals. Sampling resolution was set as 1 Hz, and Hanning-window was applied during sampling process. Time domain signals were transformed into frequency domain signals using auto-power spectrum algorithm, as presented in the following equation:
\[ S_1(\omega) = \int_{-\infty}^{\infty} R_x(t)e^{-j\omega t}dt \]  

(1)

\[ R_x(\tau) = \lim_{T \to 0} \frac{1}{T} \int_0^T x(t)x(t+\tau)dt \]  

(2)

where \( T \) represents observation time of \( x(t) \).

3 Results and Discussions

3.1 Performance of the Model Pump. Figure 6 shows experimental performance results of the slope volute pump from shutoff to 1.45 \( \Phi_d \) conditions. Nominal flow rate and head of the model pump is 48 m\(^3\)/h and 7.5 m, respectively. The best efficiency point of the model pump is near the nominal flow rate. At low flow rates, positive slope (curve instability phenomenon) occurs on head performance curve of the model pump, which indicates that unsteady rotating stall phenomenon develops inside the impeller channels under these working conditions [15]. So a large-scale separate-flow region may occur in the impeller channels, which would cause global instability within the model pump. Therefore, unsteady pressure pulsation characteristics may be affected significantly. From comparison with the spiral volute pump, it is found that the efficiency of the slope volute pump is low at high flow rates. But at low flow rates, the discrepancy is rather small, and the two pumps almost have equivalent efficiency.

To clarify the reason underlying low efficiency of the slope volute pump, Fig. 7 presents velocity distributions on the sixth cross section of the two volutes. And the numerical simulation method can be found in our preceding study [13]. At nominal flow rate, it is evident that two counter-rotating vortices of nearly the same scale are captured in the spiral volute, and the vortex centers keep almost symmetrical with respect to the midspan plane of the impeller. In the slope volute, similar flow pattern is obtained, but the scales of the two vortices are obviously different. The small-scale vortex faces the impeller outlet, while the large vortex locates at the left of the impeller. But with flow rate decreasing, at

![Speed (m/s)]

Fig. 8 Comparison of flow structures of two pumps under \( \Phi_d \): (a) relative velocity at impeller middle section, (b) absolute velocity at the sixth cross section, and (c) absolute velocity around the volute tongue
0.6Φₐ, the small vortex facing the impeller disappears, and only a large-scale vortex dominates the whole flow passage. The reason for low efficiency of the slope volute pump is probably related to flow structures in volute. At high flow rates, partial flow in the small-scale vortex region is suppressed by large-scale vortex, which could not be transported synchronously by the main stream toward volute outlet. Therefore, high energy loss in this region would be expected. But this phenomenon does not occur at low flow rates and in the spiral volute, all fluid would flow through volute smoothly causing low energy loss. So it is inferred from numerical results that low efficiency of the slope volute pump may be caused by vortex motion in the volute. To reduce the negative effect of small vortex, changing the shape in the small vortex region may be an effective way to improve the performance of the slope volute pump, which would be analyzed in the future study.

3.2 Influence of Slope Volute on Flow Structures in the Centrifugal Pump. It is essential to analyze the influence of slope volute on flow structures inside the model pump. Figure 8 shows a comparison of flow distributions in slope and spiral volute pumps at nominal flow rate. Relative velocity distributions on middle section of the impeller passage are presented in Fig. 8(a). It is observed that flow patterns of the two pumps are similar and almost the same in different blade channels. Therefore, it is concluded that the slope volute has little influence on flow structures inside impellers compared with the spiral volute. Figure 8(b) shows absolute velocity distributions at the sixth cross section of two volutes. According to the results indicated in Fig. 7, different flow structures in volutes may be a reason causing low efficiency of the slope volute pump at high flow rates. Figure 8(c) shows flows near volute tongues. In the conventional centrifugal pump, it is well accepted that the interaction between flows discharged from the impeller with jet-wake pattern and the volute tongue causes high amplitude pressure pulsation. However, in the slope volute pump flow pattern changes. It is evident that flow is divided into two parts by the volute tongue, one at the left of the volute tongue and the other at the right of the volute tongue. Flows discharged from the impeller do not interact immediately with the volute tongue but travel smoothly along the circumference of the volute toward diffuser outlet. And the interaction with the volute tongue occurs at the inlet of the diffuser. But with flow moving for a long distance, the pulsation level of the unsteady flow would be attenuated significantly. So the interaction effect with the volute tongue would be reduced rapidly. Therefore, it is inferred that pressure pulsation inside the slope volute is much lower compared with that in the spiral volute pump.

3.3 Unsteady Pressure Pulsation of the Model Pump. Unsteady pressure pulsation experiments of the model pump were conducted at different flow rates. Low frequency signals in 0–10 Hz frequency band were filtered during signal processing. To evaluate pressure pulsation energy in particular frequency band, RMS method was applied to deal with discrete pressure signals as presented in Eq. (3)

\[
RMS = \frac{1}{0.5mu} \sqrt{\sum_{n=1}^{N} (A_n - \bar{A})^2}
\]

(3)

\[
\bar{A} = \frac{1}{N} \sum_{n=1}^{N} A_n
\]

(4)

where \(A_n\) represents pressure amplitudes at different frequencies, and \(\bar{A}\) is mean amplitude.

In the centrifugal pump, the gap between the rotating impeller and the stationary volute is small, leading to intense pressure fluctuating [16,17]. Figure 9 shows time history of unsteady pressure pulsation signals of sensor p7 at four typical flow rates. As observed, the pressure signals fluctuate obviously. Under off-design conditions, especially at low flow rate of 0.2Φₐ, pressure pulsating amplitude is much larger than that at nominal flow rate, which is associated with the separate-flow structure developing in the model pump. According to Fig. 6, unsteady rotating stall phenomenon occurs within the impeller channels around 0.2Φₐ. The global instability inflow, of course, influences the overall hydrodynamic performance characteristics of the model pump and induces rather high amplitude pressure pulsation.

To investigate the influence of operating condition and the position of transducer on pressure pulsation characteristics, pressure spectra at sensors p3 and p7 are presented in Fig. 10 at four flow rates, namely 0.1Φₐ, 0.6Φₐ, 1.0Φₐ, and 1.3Φₐ. In the centrifugal pump, rotor–stator interaction is a primary reason for high amplitude pressure pulsation. And blade passing frequency \(f_{BPF}\) and its higher harmonics \(2f_{BPF}, 3f_{BPF}, \) and so on are expected thereby. From Fig. 10, it is easy to identify \(f_{BPF}\) and its higher harmonics at different flow rates except the pressure sensor p3 at 0.1Φₐ due to its small amplitude. At extremely low flow rate of 0.1Φₐ, some prominent discrete components at \(f_{BPF}, 2f_{BPF}, 3f_{BPF}, \) and \(4f_{BPF}\) can be identified for sensors p3 and p7, but peaks at other higher harmonic of \(f_{BPF}\) are not evident. Generally, it is accepted that pressure pulsation amplitudes at higher harmonics are lower than that at basic frequency. But this argument is not always suitable for the present case. As observed, at low flow rates of 0.1Φₐ and 0.6Φₐ, predominant components in pressure spectra locate at \(f_{BPF},\) and the corresponding amplitudes are larger than \(f_{BPF}.\) Especially for p3 at 0.1Φₐ, it is difficult to identify discrete peak at \(f_{BPF}.\) Besides, a large number of high amplitude components at low frequencies are generated, which are probably caused by unsteady separate-flow inside the model pump.

With flow rate increasing, pressure amplitude at \(3f_{BPF}\) decreases rapidly at high flow rates, exhibiting an opposite tendency relative to...
to that at f_{BPF}. Especially at 1.3 \phi_d, pressure amplitude at 3f_{BPF} is much smaller than that associated with f_{BPF}. Also, it is easy to find that pressure spectrum is sensitive to the position of pressure transducer. At 1.0 \phi_d and 1.3 \phi_d, predominant components correspond to f_{BPF} at p3, but spikes at f_R remain predominant at p7. From the above analysis, it is concluded that component at f_{BPF} does not always dominate pressure spectrum, and its magnitude is often less than other frequencies. The reason for low f_{BPF} magnitude probably lies in the fact that energy at f_{BPF} is often redistributed among other higher harmonics and nonlinear interaction components hence lowering the f_{BPF} intensity \cite{18}.

As illustrated in Fig. 10, at some operation conditions, the amplitude at component f_R is often larger than f_{BPF}, so f_R is a critical basic frequency. A large number of nonlinear components would be generated due to nonlinear interaction between f_R and f_{BPF}, as well as their higher harmonics, having the form of mf_{BPF} + nf_R, where m and n are integers. At low flow rate of 0.1 \phi_d, several nonlinear frequencies could be identified corresponding to higher harmonic of f_{BPF} and to its sum and difference frequencies with higher harmonic of rotating frequency f_R. Table 1 presents some evident nonlinear components.

According to Fig. 10, frequencies associated with remarkable peaks in pressure spectra are usually lower than 4f_{BPF}. To illustrate the relationship between pressure pulsation energy and flow rate, RMS method is used to process pressure signals ranging from 10 to 500 Hz. Figure 11 presents RMS values as a function of flow rate for different pressure transducers. It is seen in Fig. 11 that the position of pressure transducer influences not only pressure spectrum but also energy distribution of discrete signals in particular frequency band. In Fig. 11(a), RMS values of different pressure sensors almost have similar variation tendencies. When the model pump operates near nominal flow rate, pressure pulsation energy reaches a minimum level. At off-design conditions, either high or low flow rates, RMS values increase rapidly, especially for sensors p5, p6, and p7. Generally, at partial load unsteady flow phenomena would be easy to develop inside the impeller passage. These flow phenomena are represented by flow separation from blade suction side, vortex shedding at the trailing edge of the impeller, which would result in a sharp increase of pressure pulsation energy. But at extremely low flow rates from 0.1 \phi_d to shut-off conditions, pressure pulsation energy decreases. Also it is noted that RMS values of different sensors show apparent difference at low flow rates, especially at 0.2 \phi_d. RMS value of transducer p7 is almost 1.6 times of its counterpart associated with p3.

However, in Fig. 11(b), pressure pulsation energy exhibits a significant difference compared with that shown in Fig. 11(a).

<table>
<thead>
<tr>
<th>Value (f/f_{BPF})</th>
<th>Nonlinear component</th>
</tr>
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<tbody>
<tr>
<td>2.83</td>
<td>3f_{BPF} - f_R</td>
</tr>
<tr>
<td>3.17</td>
<td>3f_{BPF} + f_R</td>
</tr>
<tr>
<td>3.83</td>
<td>4f_{BPF} - f_R</td>
</tr>
<tr>
<td>3.66</td>
<td>4f_{BPF} - 2f_R</td>
</tr>
<tr>
<td>3.49</td>
<td>4f_{BPF} - 3f_R</td>
</tr>
</tbody>
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Fig. 10 Pressure spectra of pressure sensors p3 and p7 at different flow rates

Table 1 Some identified nonlinear components

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Pressure sensors p1 and p2 have similar variation tendencies, and RMS values increase first, then experience a local maximum value around 0.5Φₐ, finally fall as flow rate increases. At high flow rates, rapid increase in RMS value is not found. For pressure sensors p8 and p9, the trends change again. As observed, at low flow rates, sharp increase in RMS value does not occur. At p9, RMS value almost keeps increasing from shutoff to maximum flow rate.

From the above analysis, it is clear that different positions of the slope volute are associated with different pressure fluctuation characteristics. In the centrifugal pump, due to the asymmetric geometry of the volute, circumferential pressure distribution inside the volute is not uniform. As demonstrated, even at nominal flow rate, the unevenness of static pressure distribution along the volute casing exists, as also applies to unsteady pressure pulsation. So it is concluded that pressure pulsation energy does not have the same tendency along the slope volute casing due to nonuniform flow field distribution.

Angle distributions of pressure amplitudes at f_{BPF} on periphery of the slope volute for various flow rates are presented in Fig. 12. As observed, from 75 deg to 108 deg, pressure amplitudes have similar variation tendencies with respect to flow rate. Usually, rotor–stator interaction is intense near the slope volute tongue, namely around 45 deg, where maximum pressure amplitudes could be detected for flow rates ranging from 0.8Φₐ to 1.4Φₐ. But an opposite trend is observed at low flow rates ranging from 0.2Φₐ to 0.6Φₐ. And at 45 deg, pressure amplitude is smaller than those at other measuring points. At low flow rates, the exit angle of flow discharged from the impeller deviates significantly from that at nominal flow rate, which has an obvious influence on flow distribution near the slope volute tongue. According to our previous work [13], partial fluid flows back from the diffusion section passing the volute tongue, hence the interaction pattern between jet-wake flow structure and the volute tongue would probably be disturbed due to backflow structure. According to Akin and Rockwell [19], pressure amplitude may be decreased as a result of turbulent mixing in the centrifugal pump. So it is inferred that the decrease of pressure amplitude around the slope volute tongue at low flow rates is probably related to turbulent mixing of backflow structure and rotor–stator interaction.

As illustrated in Fig. 10, some evident peaks occur in pressure spectrum at given flow rate, and Fig. 13 shows two distinct peaks at impeller rotating frequency f_R and higher harmonic of blade passing frequency 3f_{BPF}. It is noted that pressure amplitudes at f_R and 3f_{BPF} fluctuate slightly at flow rates lower than 1.0Φₐ. At high flow rates (Φ > 1.0Φₐ), pressure amplitude at f_R increases gradually, while an opposite trend is seen for 3f_{BPF}. Furthermore, pressure amplitude drops rapidly as flow rate increases. Situations at f_R and 3f_{BPF} support the energy redistributing conjecture among discrete components in pressure spectrum. As observed, energy increase at f_R is at the expense of rapid energy decrease at 3f_{BPF}. 

![Fig. 11 RMS trends of different measuring points versus flow rate](image1)

![Fig. 12 Angle distributions of pressure amplitudes at f_{BPF} along the slope volute casing for different flow rates](image2)

![Fig. 13 Pressure amplitudes at distinct peaks f_R and 3f_{BPF} versus flow rate](image3)
3.4 Influence of Rotational Speed on Pressure Pulsation Characteristics. In the centrifugal pump, strong rotor–stator interaction, together with intense pressure pulsation, is excited as the impeller blades pass the volute tongue successively. So pressure spectrum characteristics, such as the amplitude of discrete component and energy distribution in particular frequency band, are closely related to rotational speed of the impeller [21]. For the model pump operating at nominal flow rate, Fig. 14 presents pressure spectra of pressure sensor p3 at three rotational speeds, namely 1450 r/min, 1200 r/min, and 1000 r/min. At nominal rotational speed of 1450 r/min, some discrete spikes corresponding to \( f_R, f_{BPF} \), and their harmonic frequencies can be identified. As observed, peaks at \( f_{BPF} \) always dominate pressure spectra at p3 for different rotational speeds. When rotational speed decreases to 1200 r/min, amplitudes at \( f_{BPF} \) and \( 3f_{BPF} \) decrease evidently. With rotational speed decreasing further, at 1000 r/min, amplitude at \( f_{BPF} \) decreases continuously, while the component at \( 3f_{BPF} \) could not be identified easily. But pressure amplitude at \( 4f_{BPF} \) increases rapidly as rotational speed decreases.

For the model pump operating at different rotational speeds, Fig. 15 shows both RMS values and amplitudes corresponding to \( f_{BPF} \) at p1 and p3. It is seen that both RMS value and amplitude at \( f_{BPF} \) increase as rotational speed increases. At nominal flow rate, from 1000 r/min to 1480 r/min, RMS value has an increment of 103% at p1 and 62% at p3. Under off-design conditions, especially at low flow rate \( 0.8A_d \), the increment increases rapidly, and RMS value increases by 150% at p1 and 113% at p3. Amplitude at \( f_{BPF} \) shows a similar trend relative to RMS value. At nominal flow rate, from 1000 r/min to 1480 r/min, amplitude increases by 183% at p1 and 46% at p3.

In the centrifugal pump, low frequency vibration signals induced by pressure pulsation have a great influence on mechanical components of the pump. As shown in Fig. 15, pressure pulsation energy decreases rapidly as rotational speed of the impeller decreases. So it is concluded that adjusting rotational speed of the impeller is an effective measure for reducing vibration energy. Finally, the negative impact of vibration on mechanical components of the pump and even the pumping system would be weakened significantly.

3.5 Validation of Low Pressure Pulsation Level Characteristic of the Slope Volute. To relieve rotor–stator interaction in the centrifugal pump, a special slope volute is investigated. Based upon theoretical analysis of flow structures in Fig. 8, the slope volute is characterized by low pressure pulsations, especially at \( f_{BPF} \). To validate our conjecture, pressure pulsation inside a centrifugal pump with conventional spiral volute was measured for comparison. The same impeller was used for the two pumps. In the presence of the size of the spiral volute, only three pressure transducers were mounted on periphery of the spiral volute for comparison, as shown in Fig. 16. The transducers were located at midspan of impeller outlet, as was implemented for the slope volute pump. And rotational speed of the impeller was kept invariant 1450 r/min for all examined flow rates.

In the centrifugal pump with the conventional spiral volute, \( f_{BPF} \) often dominants pressure spectrum, which is also an important forcing frequency inducing low frequency vibration signals in
fluid machineries [22–24]. To illustrate the influence of slope volute on \( f_{\text{BPF}} \), Fig. 17 presents a comparison of amplitude at \( f_{\text{BPF}} \) between the slope volute and the spiral volute pumps. As observed, experimental results agree with our conjecture very well. It is found that pressure amplitude of the spiral volute pump is much larger than that in the slope volute pump from shutoff to maximum flow rate conditions. Besides, variation of pressure pulsation for the slope volute pump is relatively smooth, as implies more uniform compared with that inside the spiral volute pump. For \( p7 \), evident discrepancy is present between the two pumps especially at low flow rates. At 0.4\( D_4 \) amplitude for the spiral volute pump nearly reaches 4.9 times of that for the slope volute pump, and 4.3 times at 0.5\( D_4 \). At nominal flow rate, there is still a difference of 1.9 times. For measuring points \( p8 \) and \( p9 \), similar results are observed for all flow rates in consideration. Additionally, amplitude of the spiral volute pump is three times of that for the slope volute pump for \( p8 \) and 3.2 times for \( p9 \).

To evaluate the influence of the slope volute on low frequency signals in 10–500 Hz band, a comparison of RMS values between the two pumps is performed in Fig. 18. It is noted that RMS value of the spiral volute pump is much larger than that of the slope volute pump at \( p7 \) for all measured flow rates. In particular, under off-design conditions, the difference is salient. For pressure sensors \( p8 \) and \( p9 \), similar trends can be found at low flow rates, and RMS value of the spiral volute is larger than that of the slope volute pump. But at nominal flow rate, the difference between the two pumps is not significant, and RMS values of the two pumps are pretty close.

The main purpose to investigate unsteady pressure pulsation characteristics is to seek an effective way to reduce pressure amplitude in the centrifugal pump. In most relevant literatures, attentions are often paid to the influence of geometric parameters on pressure amplitude. In this paper, by using a special slope volute, the intense interaction between flow discharged from the impeller and the volute tongue would be attenuated obviously. According to the comparison results, the slope volute has an evident impact on reducing pressure energy in the centrifugal pump, especially for the amplitude at \( f_{\text{BPF}} \).

4 Conclusions

An experimental investigation on unsteady pressure pulsation features of a centrifugal pump with slope volute is presented. The analysis focuses on pressure pulsations at nine different measuring positions on periphery of the slope volute. Pressure pulsation signals measured from shutoff to maximum operating conditions are analyzed using auto-power spectrum algorithm and RMS method. Pressure spectrum characteristics, both discrete component and RMS value, are easily affected by operating conditions and measuring position on the slope volute casing. At low flow rates, the predominant component in pressure spectrum often deviates from \( f_{\text{BPF}} \) to its high harmonic frequency. And some nonlinear components are captured due to nonlinear interaction between \( f_R \) and \( f_{\text{BPF}} \). With flow rate increasing, both the number and amplitude of nonlinear components decrease obviously.

Meanwhile, the influence of rotational speed on pressure pulsation is investigated. It is found that pressure amplitude at \( f_{\text{BPF}} \) and RMS value increase rapidly as rotational speed increases. Generally, low vibration signals are caused by pressure pulsation, which are critical to stable operation of the pump. Thus, low rotational speed of the impeller may be more appropriate during pump design process considering low vibration characteristic of the centrifugal pump.

To assess low pressure pulsation level of the slope volute, pressure pulsation of a conventional spiral volute pump is measured for comparison. It is observed that pressure pulsation inside the slope volute pump is much smaller than that inside the spiral volute pump, especially for amplitude at \( f_{\text{BPF}} \). So it is confirmed that intense rotor–stator interaction in the centrifugal pump is significantly attenuated by employing the slope volute.

Finally, it is expected that the present work will provide a different view considering lower pressure pulsation requirement of the centrifugal pump and contribute to a better understanding of unsteady pressure pulsation. In further study, experimental investigation on optimal design and unsteady flow field associated with the slope volute will be conducted to further lower pressure pulsation level and to improve high pump efficiency as well.

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Nomenclature

\[ A = \text{pressure amplitude (Pa)} \]
\[ b_2 = \text{impeller outlet width (17 mm)} \]
\[ d_2 = \text{impeller outlet diameter (172 mm)} \]
\[ c_p = \text{pressure coefficient (} \frac{A}{0.5 \rho u_2^2} \) \]
\[ f_R = \text{rotating frequency of impeller (24.2 Hz)} \]
\[ f_{\text{BPF}} = \text{blade passing frequency (145 Hz)} \]
\[ \eta = \text{efficiency} \]
\[ n_3 (\omega) = \text{rotational speed} \ (1450 \text{r/min} \ (151.8 \text{rad/s})) \]
\[ n_s = \text{specific speed} \ (n_s = 3.65n_3\sqrt{Q_0^2/H_0^2 \beta \lambda}) \]
\[ n_z = \text{tangential velocity at blade outlet} \ (13 \text{m/s}) \]
\[ Z = \text{blade number} \ (6) \]
\[ \rho = \text{water density} \ (1000 \text{kg/m}^3) \]
\[ \Phi_d = \text{nominal flow rate coefficient} \ (Q_d/(\omega d^2 b_z^2)) \]
\[ \Psi_d = \text{nominal head coefficient} \ (gH_d/(\omega^2 d^2)) \]

References


