Research of fluid-induced pressure fluctuation due to impeller-volute interaction in a centrifugal pump

Q Z Liu, K Yang, D Y Li and R Z Gong
Harbin Institute of Technology, School of Energy Science and Engineering, Harbin 150001
E-mail: Gongruzhi@hit.edu.cn

Abstract. The fluid pressure fluctuation generated by unsteady flow is a very important factor to induce vibration of the centrifugal pump. The relative movement between impeller and volute generates an unsteady interaction which affects not only the overall pump performance, but is also responsible for pressure fluctuations. Pressure fluctuations interact with the volute casing or even with the circuit and give rise to dynamic effects over the mechanical parts, which are one of the most important sources of vibration and hydraulic noise. To investigate the flow characteristic in the centrifugal pump, the unsteady flow is simulated by CFD methods in this paper. Unsteady flow characteristic in the centrifugal pump is obtained considering the impeller-volute interaction in the whole flow field. Based on the unsteady flow simulation, amplitude-frequency characteristics of the pressure fluctuation in the centrifugal pump are obtained through setting up monitoring point at the impeller outlet. The research shows that the frequency component include the blade passing frequency as the main component, the multiplication of blade passing frequency, and the harmonic interference due to the unsteady flow.

1. Introduction
The fluid pressure fluctuation induced by hydraulic factors such as unsteady flow of fluid is an important factor cause vibration of the centrifugal pump. In the centrifugal pump, the fluid pressure fluctuation can be as large as all of the pressure increment [1]. Because the fluid pressure fluctuation is significant, the noise and vibration caused by the fluid pressure fluctuation can generate a large stress and reduce the fatigue life of components. The investigation on mechanism of the fluid pressure fluctuation helps to the steady running of the centrifugal pump system.

The flow analysis inside the centrifugal pump is highly complex mainly due to 3D flow structure involving turbulence, secondary flow, cavitations and unsteadiness [2]. Jose Caridad [3] presents the results of numerical simulations carried out in a centrifugal pump impeller of an electrical submersible pump (Ns = 2063) conveying an air-water mixture. G. Pavesi[4] puts up a study of time-frequency characterization of the unsteady phenomena in a centrifugal pump, R. Spence[5] performs a investigation in pressure pulsations in a centrifugal pump using numerical methods supported by industrial tests, Raúl Barrio [6] provides a numerical analysis of the unsteady flow in the near-tongue region in a volute-type centrifugal pump. R.Barrio [7] explores the use of a commercial CFD code to estimate the total radial load on the impeller of two test pumps. The unsteady flow in the centrifugal pump is simulated during one revolution by using CFD software [8].

In this paper, unsteady flow of the centrifugal pump was simulated based on the CFD method [9].
Through setting up pressure monitors [10], the characteristics of pressure fluctuation was obtained. A study by using FFT and Wavelet transform on time frequency characteristics of the pressure waves was also made.

2. The control formulation of flow field and discrete method

The Reynolds-Averaged momentum equation is used to describe the incompressible flow, and the tensor form of the equations is:

\[
\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} = \frac{1}{\rho} \frac{\partial p}{\partial x_i} - \frac{1}{3} \frac{\partial}{\partial x_i} \left( \frac{\partial p}{\partial x_i} \right) + \frac{2}{3} \frac{\partial}{\partial x_i} \left( \frac{\partial u_i}{\partial x_i} \right)
\]

(1)

Where, \( u_i (i = 1, 2, 3) \) is the velocity of flow, \( p \) is the pressure of flow, \( \mu \) is the kinematic viscosity, and \( -\rho u_i' u_j' \) is the Reynolds stress.

The momentum equation will be closed when adopting the RNG \( k - \varepsilon \) model and combining the continuity equation. The RNG \( k - \varepsilon \) model can be expressed as:

\[
\begin{align*}
\frac{\partial k}{\partial t} + \nabla \cdot (\rho u_k u_k) &= \frac{1}{\sigma_k} \frac{\partial}{\partial x_i} \left( \sigma_k \frac{\partial k}{\partial x_i} \right) + p_r - \varepsilon \\
\frac{\partial \varepsilon}{\partial t} + \nabla \cdot (\rho u_k \varepsilon) &= \frac{1}{\sigma_\varepsilon} \frac{\partial}{\partial x_i} \left( \sigma_\varepsilon \frac{\partial \varepsilon}{\partial x_i} \right) + \frac{C_\varepsilon}{k} p_r - C_{\varepsilon 2} \varepsilon^2
\end{align*}
\]

(2)

Where \( p_r \) denotes the generation rate of turbulent kinetic energy.

\[
p_r = \nu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} \nu_t = C_u \frac{k^2}{\varepsilon}
\]

(3)

The five parameters in the model can be abstained through theoretical analysis:

\( C_u = 0.0845 \), \( C_\varepsilon = 1.42 \), \( C_{\varepsilon 2} = 1.68 \), \( \sigma_k = 1 \), \( \sigma_\varepsilon = 1.3 \)

The segregated implicit solver is applied to solve the flow, and the SIMPLE pressure correction algorithm is used for solving the coupling of velocity and pressure. The turbulent kinetic energy, the turbulence dissipation and the momentum equation are discrete through second-order upwind scheme. The convergence is determined when the maximum residual reaches \( 10^{-4} \) [11].

Figure 1. The positions of the sliding surface in the computing model

Multiple coordinate systems are applied to simulate the unsteady flows in rotating region and static region of centrifugal pump. Flows in the two regions are solved respectively, and the data is transferred by the sliding interface between the regions. There are two sliding interfaces in centrifugal pump showed as Fig.1, one is between impeller and volute, the other is between the inlet of the impeller and the entrance region of centrifugal pump.

3. Unsteady flow and Frequency spectrum analysis of pressure fluctuation in impeller

There is pipe system at the outlet of most centrifugal pumps, which means there is a load in most centrifugal pumps, and the pipe system has an important influence on the characteristics of centrifugal pump. In this paper, the pressure at the outlet of the centrifugal pump is considered a constant. Tab.1
shows geometries dimension and condition parameters of the centrifugal pump. The parameters of flow rate and head are determined by optimum operating condition and similarity criterion.

| Name                          | Value              | Name                | Value     |
|-------------------------------|--------------------|---------------------|-----------|
| Diameter of blade outlet      | 162mm              | Specific speed      | 0.04      |
| Diameter of impeller cover plate | 170mm          | Rotating speed      | 3000r/min |
| Width of impeller outlet      | 16mm               | Flow rate           | 0.065m³/s |
| Number of blades              | 5                  | Head                | 40m       |

In the unsteady flow simulation, the rotational speed is set on 3000 r/min, so the rotation frequency is 50Hz, and there are 5 blades of the impeller, so the blade-passing frequency is 250Hz. Pressure monitoring points is emplaced at the interface between impeller outlet and volute inlet, and the position of monitoring points is describe as inscribed angle at impeller outlet. The inscribed angle corresponding to tongue is set on 0°.

In Fig.2, the coordinate surface of X and Z shows that the pressure rises with the increasing of inscribed angle, and there is an obvious drop of pressure nearby the position of tongue. The coordinate surface of Y and Z shows that the pressure fluctuation presents similar sine rule along with the increase of the impeller rotation angle. At any monitoring points, the pressure rises and falls five times in a revolving cycle, which is equal to the number of blades. This means the unsteady pressure distribution changes under a blade-passing frequency and dues to the rotor-stator interaction between blades and volute.

![Figure 2. Unsteady pressure distribution on the inscribed angle at impeller outlet](image-url)

Fig.3 shows the waterfall plot of the pressure fluctuation on the inscribed angle at impeller outlet. It can be seen that the dominant frequency is the blade-passing frequency (250Hz), all the other frequencies appear as times of the blade-passing frequency, and the amplitude of all the other frequencies are very small.

![Figure 3. Waterfall plot of the pressure fluctuation on the inscribed angle at impeller outlet](image-url)
There are five peak values of the pressure fluctuation under blade-passing frequency around the impeller outlet. Tab.2 shows the five peak value and their inscribed angles. The maximum pressure fluctuation is at the position of 13.5° instead of the position of tongue (0°).

| location number | I1    | I2    | I3    | I4    | I5    |
|-----------------|-------|-------|-------|-------|-------|
| Inscribed angle at impeller outlet | -108° | -50°  | 13.5° | 69.5° | 135.5°|
| Amplitude (MPa^2/Hz) | 13.9  | 40.3  | 61.5  | 8.5   | 13.1  |

I1~I5 are the position of the peak value in Fig.7, and they are also shows in Fig.4.

Fig.5 shows the power spectrum of the pressure fluctuation in position I3. The dominant frequency appears the blade-passing frequency.

FFT analysis gives us the frequency components during the whole time, but it can ignore the important frequency components when the time window length is different. In this paper, five layers of Wavelet Analysis are applied to make up the deficiency of FFT analysis. It can be seen in Fig.6 that there is a frequency component of 375Hz (one and half harmonic generation of the blade-passing frequency), and the amplitude of this frequency component is big. This may be caused by the complex unsteady flow in impeller, but the mechanism is not been understood.

From FFT and Wavelet analysis above, it can be seen the dominant frequencies of pressure fluctuation include blade-passing frequency, multiplication of blade-passing frequency, and the
frequency caused by the complex unsteady flow. The entire frequency components may produce resonance with the natural frequency of centrifugal pump rotor system.

4. Analysis of radial force of the impeller

The radial force of impeller due to the pressure fluctuation can reflect to the vibration of the rotor system. Fig.7 (a) shows the unsteady radial force distribution when \( \omega = 3000 \text{r/min} \). It can be seen that \( F_x \) and \( F_y \) changes periodically. Fig.7 (b) shows the vector diagram of the radial force. It can be seen that the transient force distribute as an ellipse.

![Fig.7. The unsteady radial force distribution](image)

From the FFT analysis result of \( F_x \) and \( F_y \) in Fig.8, it can be seen that the dominant frequency components are also blade-passing frequency and times of blade-passing frequency.

![Fig.8. Power spectrum of \( F_x \) and \( F_y \)](image)

From the wavelet analysis result of \( F_x \) and \( F_y \) in Fig.9, besides blade-passing frequency component and times of blade-passing frequency component, there is also a frequency component of 375Hz (one and half harmonic generation of the blade-passing frequency).

![Fig.9. Power spectrum of \( F_x \) and \( F_y \) wavelet expansion](image)
5. Conclusion
Research above shows that the maximum pressure fluctuation at the impeller appears nearby the
tongue, and the dominant frequency of pressure fluctuation is the blade-passing frequency. Besides the
blade-passing frequency and the multiplication, there is a one and half harmonic generation of the
blade-passing frequency due to the unsteady flow. The radial force on the impeller presents the similar
frequency characteristics with the pressure fluctuation and the transient force distribute as an ellipse.

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