Numerical Study on Hydrodynamic Characteristics in a Centrifugal Pump at off Design Conditions Based on a Cycling-System Model

Wen-Hui Yin\textsuperscript{1}, Xi Wang\textsuperscript{1}, Tao Yu\textsuperscript{1}, Jie Jian\textsuperscript{1}, Xiang-Yuan Zhang\textsuperscript{1}, Zhi-Jun Shuai\textsuperscript{1,*}, Wan-You Li\textsuperscript{1} and Chen-Xing Jiang\textsuperscript{2,*}

\textsuperscript{1} College of Power and Energy Engineering, Harbin Engineering University, Harbin 150001, China
\textsuperscript{2} College of Energy, Xiamen University, Xiamen 361005, China

E-mail: \textsuperscript{1}shuaizhun98@163.com; \textsuperscript{2}jiangchx@xmu.edu.cn

Abstract. In this paper, the unsteady hydrodynamic characteristics of a centrifugal pump handled at low-flow-rate conditions are numerically investigated. A cycling-system model is established to capture the influence of off-design operation on the internal flow field and the pressure pulsation. The preliminary results show that the cycling-system model can predict the flow field of the pump more conforming to the practical situation than that of the pump alone; besides, the generation of stall clusters in the impeller will induce low-frequency pressure pulsation.

1. Introduction

The performance of a centrifugal pump depends on its internal flow characteristics. The smoother the internal flow is, the more regularly the velocity and pressure distribute, and the more stable operating state is [1]. With the development of computational fluid dynamics (CFD) techniques, many numerical studies have been performed to investigate the internal flow characteristics inside the pumps, and most researchers consider pumps as a single component, with fixed boundary conditions in their numerical simulations [2-9]. However, when the pump is handled at off-design conditions, the internal flow field changes greatly and subsequently causes unstable operation, such as backflow, vortex, stall, and flow separation [10-17]. When stall clusters are generated inside the pump, the uniformity of the internal flow field will be destroyed. In special, when the rotation frequency of the stall cluster is the same as or close to the natural frequency of the pump system, resonance occurs [18-21], which will seriously affect the efficiency and stability. In the circumstances, fixed boundary conditions are no longer applicable. In this paper, to capture the influence of the off-design operation on internal flow field and pressure pulsation, a simplified version of the cycling pipeline model is established. It does not need to fix the inlet and outlet boundary conditions. Based on the cycling-system model, the unsteady simulation of the centrifugal pump at rated and off design conditions are performed.

2. Physical Model
The whole cycling system is composed of three parts: a centrifugal pump, a valve, and a water tank. Since the valve is mainly used to control the flow rate, it is simplified to a converging-diverging pipeline. The cycling-system model is shown in Figure 1.

![Figure 1. 3D Perspective of the Cycling-system Model.](image1)

![Figure 2. Flow Field of the Model Pump.](image2)

The main structural parameters of the research pump are shown in Table 1, and the whole flow field of the model pump is shown in Figure 2.

| Parameter                           | Centrifugal pump |
|-------------------------------------|------------------|
| head (m)                            | 325              |
| inlet diameter (mm)                 | 150              |
| outlet diameter (mm)                | 125              |
| number of the long impeller blade   | 5                |
| number of the short impeller blade  | 5                |
| number of volute guide vanes        | 7                |
| number of inlet guide vanes         | 3                |

Considering the complex flow characteristics inside the pump, an unstructured tetrahedral grid is selected to generate the mesh system. After grid-independence analysis, the total cell number of the mesh is 4 million shown in Figure 3. SST is used as the turbulent model.

![Figure 3. Grid Independence Verification.](image3)

3. Governing equations and computations procedures
The unsteady incompressible Reynolds-averaged Navier-Stokes (RANS) equations are solved for the flow passing through the centrifugal-flow pump. The basic equations in a manner of RANS are the following continuity and momentum equations, respectively.
3.1. Mass conservation equation

\[
\frac{\partial}{\partial t} (\rho) + \frac{\partial}{\partial x_i} (\rho u_i) = 0
\]  

(1)

3.2. Navier-Stokes Equations

\[
\frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = - \frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_j} [\mu (\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i})] + \rho g + F_i
\]  

(2)

3.3. Simulation set-up

The relevant settings of the two models, a single pump and a cycling-system, are shown in Table 2.

| Boundary Conditions       | Single Pump | Cycling-System |
|---------------------------|-------------|----------------|
| Pressure-inlet            | 2×10^5 Pa   | /              |
| Rated condition           | 44.36 kg/s  | /              |
| Low-flow condition        | 7.5 kg/s    | /              |

3.4. Monitoring points set-up

The layout of the main monitoring points is shown in Figure 4.

**Figure 4.** monitoring points set-up. (a) Monitoring Points in Inducer. (b) Monitoring Points in Diffuser and Volute.

4. Results and Discussions

4.1. External Characteristics

In unsteady simulations, the pump head and efficiency also fluctuate within a certain range. The average value is taken in the characteristic analysis (see Figure 5).

**Figure 5.** Pump Characteristics Curves. **Figure 6.** Flow Rate in Time Domain.
For a centrifugal pump with guide vanes, the flow range is mainly divided into partial load swirl and full load instability due to the rotating stall [22]. It can be seen from the characteristics curves that the head has a partial depression at a low flow rate (less than 0.6 \( Q_d \)), and this unstable form is called a partial load swirl; as the flow rate increases, around 0.8 \( Q_d \), the depression appears in both head and efficiency curves, that unstable form is called full-load instability.

The generation of the rotating stall in impeller of diffuser will induce the flow instability and fluctuation, where a fixed boundary value does not match the physical truth. However, the cycling-system model does not need to impose the inlet and outlet boundary conditions and is self-coupled to solve, and the parameters of the flow field inside the pump are obtained through internal calculation. The flow rate simulated by the cycling-system model is monitored at 0.17 \( Q_d \) condition, and the flow rate changes at time domain is shown in Figure 6. It is clear that the outlet flow rate is not a fixed value but fluctuates with time.

4.2. Analysis of internal flow field
The centrifugal pump pressure contours are shown in Figures 7 and 8. The overall pressure at low-flow-rate condition is higher than that at rated condition, which is caused by the reduced velocity. In the middle section of the impeller and volute, there are a low-pressure area at the root of the blade and a high-pressure area around at both conditions due to the increase of the flow velocity. At rated condition, the isobaric line is vertical to the blade surface, indicating that the pressure gradient is small at the pressure surface, which is not easy to cause flow separation; while at low-flow-rate condition, the isobaric line is no longer vertical to the blade surface. The increased surface pressure gradient will cause unstable flow structures such as flow separation.

![Figure 7](image_url)  
*Figure 7. Pressure Contour at Rated Condition. (a) Overall pressure distribution (b) Pressure distribution at middle section.*
Figure 8. Pressure Contour at 0.17 $Q_d$ Condition. (a) Overall pressure distribution (b) Pressure distribution at middle section.

Figure 9 illustrates the streamline diagrams of the impeller-volute section at rated and 0.17 $Q_d$ conditions. At the junction of the diffuser guide vane and the volute, flow separation happens. In the flow channel of the volute, there is a vortex caused by unreasonable structural design. At low-flow rate condition, a vortex, in the opposite direction of the impeller rotation, is generated on the suction side of the impeller short blade and block the flow channel. Besides, the unstable structures in the guide vanes of the diffuser causes backflow in the flow channel, and some flow channels are completely blocked by the vortex, which seriously affects the performance of the pump. The streamlines in the flow path of the entire volute are no longer smooth, which are caused by the off-design operation.

Figure 9. Streamlines inside the Pump at Two Conditions. (a) Rated condition (b) Low-flow-rate condition.

Figure 10 shows the velocity streamline of the volute section at two conditions. It can be seen that at rated condition, there is a pair of reverse vortices in the volute section and they maintain symmetrical distribution proximately; while at low-flow-rate condition, on one side of the volute cross-section, there aren’t any secondary flow structures, while the other side, two vortices of the secondary flow structure are generated with asymmetrical distribution. The dominant vortex almost fills the entire volute flow path. The off-design operation influences the flow field inside the volute.
Figure 10. Volute Secondary Flow at Two Conditions. (a) Rated condition (b) Low-flow-rate condition.

4.3. Pressure fluctuation characteristics

Figure 11 shows the pressure fluctuation in the region of inlet guide vane in a cycle. At rated condition, the pressure near impeller (seeing locations Points 5 and 6 presents obvious regular fluctuation with five peaks within a cycle, which is related to the rotor-stator interaction between impeller and inlet guide vane. The other locations have no obvious regularity. In the case of low-flow-rate condition, only the location Point 6, closest to the impeller, has obvious regularity.

Figure 11. Pressure Fluctuation in Time Domain at Inducer. (a) Rated condition (b) Low-flow-rate condition.

For further processing, Fast Fourier transform (FFT) is employed to obtain the frequency domain information (Figure 12). At rated condition, the locations Points 5 and 6 have higher pressure amplitude at 5 times the shaft frequency, corresponding to the blade passing frequency (BPF). While at low-flow-rate condition, only the location Point 6 has a higher peak at BPF. Besides, compared the two working conditions, the pressure amplitude at BPF is higher at low-flow-rate condition near the area of rotor-stator interaction.
Considering the similar results of 7 diffuser guide vanes, only one group data is shown to explain the pressure fluctuation characteristics of the two working conditions. Figure 13 shows the pressure fluctuation of the diffuser in time domain. Point S1 near the impeller has 10 peaks within a cycle, corresponding to the blade number of the impeller. Besides, the sudden increase and decrease of the pressure are sharper at location Point S1 due to 5 long blades and 5 short blades swept. The difference is that at low-flow-rate condition, the average pressure amplitude and fluctuation amplitude are higher.

The frequency domain information at diffuser is shown in Figure 14. Both working conditions have obvious peaks at 5, 10, and 15 times the shaft frequency. While at low-flow-rate condition, many random pressure pulsations are generated, which is due to the stall clusters generation at low-flow-rate condition.
Figure 15 shows the pressure fluctuation at the volute in time domain. The pressure fluctuations have no obvious regularity at two working conditions, which is caused by asymmetry of volute structure. The difference is that the amplitude at low-flow-rate condition is higher than that at rated condition. According to results in frequency domain (Figure 16), both two conditions have obvious pressure amplitude at 5,10,15 times the shaft frequency. But the amplitude is higher at low-flow-rate condition than that at rated condition. Besides, 7 and 14 times the shaft frequency are generated under low-flow-rate condition corresponding to the blade number of the diffuser.

**Figure 15.** Pressure Fluctuation in Time Domain at Volute. (a) Rated condition (b) Low-flow-rate condition.

**Figure 16.** Pressure Fluctuation in Frequency Domain at Volute. (a) Rated condition (b) Low-flow-rate condition.

5. Conclusions

This paper establishes a cycling-system model to numerically investigate the unsteady hydrodynamic characteristics and the pressure pulsation of a centrifugal pump handled at low-flow-rate conditions. The conclusions are as follows:

1. The numerical method of the cycling-system model does not impose fixed inlet and outlet boundary conditions, and the parameters are obtained through internal calculations, which are more conforming to practical flow especially at off design conditions.

2. At low-flow-rate condition, the generation of stall clusters in impeller and diffuser domains will influence the pressure distribution and flow pattern inside the pump; the existence of the diffuser worsen the pulsation, which needs further investigation.

3. Low-frequency random pressure pulsations are generated at low-flow-rate condition, which is dangerous if the rotation frequency of the stall cluster is the same as or close to the natural frequency of the pump system. In other words, if a pump must be handled at some off design conditions, we have to consider the possibility due the low-frequency excitation in advance.

Acknowledgment

This research work was supported by the National Science Foundation of China (Grant number 51806042)
Nomenclature

\( u, v, w \)  three Eulerian velocity components

\( Q_d \)  rated flow rate

\( Q_s \)  low flow rate

\( T \)  one cycle

\( f \)  frequency

\( f_n \)  shaft frequency

References

[1] Zhang L, Zhang W and Wang C 2018 Effect of impeller blade leading-edge chamfer on internal flow field of centrifugal pump Power System Engineering 34 1-5

[2] Li X J, Yuan S Q, Pan Z Y, Yuan J P and Si Q R 2013 Numerical simulation of the whole flow field for centrifugal pump with structured grid Transactions of the Chinese Society for Agricultural Machinery 44 50-54

[3] Li Z F 2009 Numerical simulation and experimental study on the transient flow in centrifugal pump during starting period Zhejiang University

[4] Tang H and He F 2002 Numerical simulation of flow field in centrifugal pumps Pump Technology 3:8+14

[5] Wang X, Rao J, Li Z S, Zhu X N and Chang C 2019 Numerical study of the centrifugal compressor impeller with gurney flaps Chinese Journal of Turbomachinery 61 37-40

[6] Wang J L, Yang J G, Liu Y and Zhang T C 2019 Multi-objective optimization design of a centrifugal impeller Chinese Journal of Turbomachinery 61 26-31+4

[7] Liu Y Y, Wang W J and Li T L 2019 Reach on cavitation flow of centrifugal pump based on slotting jet flow Chinese Journal of Turbomachinery 61 18-22

[8] Yao H Y, Wu D Z, Cao L L and Wu P 2019 Numerical investigation of internal flow in cavitation resistant centrifugal pump Chinese Journal of Turbomachinery 61 10-17+4

[9] Fang Y J, Ji C J, Sun Q, Liu J N and Zhang X W 2020 Validation of numerical simulation for centrifugal compressor under different inlet directions Chinese Journal of Turbomachinery 62 15-21

[10] Yao Z F, Wang F J , Qu L X, Xiao R F, He C L and Wang M 2011 Experimental investigation of time-frequency characteristics of pressure fluctuations in a double-suction centrifugal pump Journal of Fluids Engineering 133

[11] Kaupert K A and Staubli T 1999 The unsteady pressure field in a high specific speed centrifugal pump impeller-part 1: influence of the volute Journal of Fluids Engineering 121

[12] Lu Y J 1992 Rotating stall and surge phenomenon and separation vortex flow in fans and blade machines Water Conservancy & Electric Power Machinery 5 20-44

[13] Jia X Q 2017 Investigation on unstable flow and vibration characteristics of a centrifugal pump Zhejiang University

[14] Zhou P J 2015 Investigation of characteristics in centrifugal pumps China Agricultural University

[15] Qu L X, Wang F J, Cong G H and Yao Z F 2011 Pressure Fluctuations of the impeller in a double suction centrifugal pump Transactions of the Chinese Society for Agricultural Machinery 42 79-84+78

[16] Zhou P J, Wang F J and Yao Z F 2015 Investigation of pressure fluctuation in centrifugal pump impeller under rotating stall conditions Transactions of the Chinese Society for Agricultural Machinery 46 56-61
[17] Cong G H and Wang F J 2008 Numerical investigation of unsteady pressure fluctuations near volute tongue in a double-suction centrifugal pump *Transactions of the Chinese Society for Agricultural Machinery* 60-63+67
[18] Zhou P J, Wang F J and Yao Z F 2016 Study on effects of blade number on stall characteristics for centrifugal pump impeller *Journal of Mechanical Engineering* 52 207-212
[19] Wang Z B 2020 Analysis of pressure pulsation under rotating stall of pump-turbine in pump mode Xi'an University of Technology
[20] Huang K L 2003 Internal flow and induced vibration and noise characteristics of cooling pump in internal combustion engine Jiangsu University
[21] Wang H and Tsukamoto Hiroshi 2003 Experimental and numerical study of unsteady flow in a diffuser pump at off-design conditions *Journal of Fluids Engineering* 125
[22] Yang J, Yuan S Q, Pei J and Zhang J F 2015 Overview of rotating stall in centrifugal pump with vaned diffuser *Journal of Drainage and Irrigation Machinery Engineering* 33 369-373