Numerical Simulation of Pressure Fluctuation around the Tongue Region in a Centrifugal Pump

L L Zheng, H-S Dou*, X P Chen, Z C Zhu and B L Cui

Key Laboratory of Fluid Transmission Technology of Zhejiang Province, Zhejiang Sci-Tech University, Hangzhou, 310018, China

E-mail: *huashudou@yahoo.com

Abstract. Pressure fluctuation near the tongue is one of the primary sources of pump vibration and noise. In order to investigate the effect of pressure fluctuation near the tongue, the RANS equations and the RNG k-epsilon turbulence model are employed to simulate the flow in the pump. The SIMPLE algorithm is applied to couple the solutions of the system of equations. Flow field within the centrifugal pump under different flow rates are obtained by simulation. The simulation results are compared with the experimental data to verify the reliability of the calculation model. It is found that the pressure fluctuation at each monitor point is a periodic wave but non-uniform under small flow rate. When the flow rate is larger than the design flow rate, average pressure and standard deviation at monitor points is relative uniform. The dominate frequency of pressure fluctuation is the blade passing frequency and the amplitude of pressure fluctuation is regular. At small flow rate, complex unstable flow makes average pressure and standard deviation at monitor points increasing obviously. Amplitude of pressure fluctuation is larger than that of design flow rate conditions and the maximum amplitude of pressure fluctuation in frequency domain exists at the monitor point just behind the tongue along the impeller rotation direction.

1. Introduction
Centrifugal pump has been widely applied in petrol chemical industry, aerospace industry, power energy, etc. Improvement of the hydraulic characteristics of centrifugal pump is desired in order to satisfy the requirement of science and technology. Complex unsteady flow has large effect on the performance of centrifugal pump and different kinds of the unsteady phenomena [1-8] exist in the pump, including rotor-stator interaction, separating vortex, clearance flow, etc. Small clearance between the tongue and impeller result in the strong pressure fluctuation in this region and it plays a dominant role for internal flow characteristics and external performances. Qu et al. [9] used the large eddy simulation method with sliding mesh technology to investigate the pressure fluctuations in a double suction centrifugal pump at design flow rate. Results show that the blade passing frequency dominated the pressure fluctuations in the volute, while rotating frequency dominated in the impeller. Cai et al. [10] studied the pressure fluctuations near the tongue region through experiments. Tan et al. [11] analyzed the unsteady flow around the tongue region in a centrifugal pump through numerical simulation and results show that the flow fields are even in the tongue region with small pulsations of pressure and velocity at the design flow rate. Majidi [12] simulated the unsteady flow in the centrifugal pump and found that the pressure fluctuation achieve large value at the outlet of the impeller and the tongue region. Spence et al. [13] carried out numerical simulation of the unsteady...
flow field in the centrifugal pump and analyzed the pressure fluctuation near the tongue region. Ren et al. [14] used PIV technique in order to discover the onset and development rule of unstable vortex within the impeller. Yuan et al. [15] studied the pressure fluctuation of unsteady flow in screw-type centrifugal pump with small blade and obtained the pressure fluctuations data in the volute at different flow rates. Cong and Wang [16] investigated the pressure fluctuation of turbulent flow near the volute tongue in a double suction centrifugal pump by using large eddy simulation method with sliding mesh technology.

In this paper, unsteady numerical investigation is carried out in a volute type centrifugal pump in order to elucidate the effect of pressure fluctuation on flow stability of the tongue region. The parameters of the whole flow field are obtained through computation by using the commercial CFD code ANSYS-Fluent. The numerical results are compared with experiment data to validate the accuracy of CFD results. The pressure fluctuation in time domain and frequency domain near the tongue are obtained at the monitor points under different flow rates.

2. Numerical methods and geometry

2.1. Governing equations and numerical methods
The governing equations are the unsteady three-dimensional incompressible Reynolds-averaged Navier-Stokes equations and the RNG k-ε turbulence model, and standard wall functions is applied in the simulation. The finite volume method (FVM) is used to solve the system. The coupling between velocity and pressure is achieved by the SIMPLE algorithm.

The boundary conditions are considered to have more physical meaning for turbomachinery flow simulations. The length of inlet duct is three times of impeller inlet diameter. The boundary conditions are set as that the velocity is given at inlet and Neumann boundary condition is given at outlet of the pump.

Unsteady calculation is based on the results of the steady simulation. The steady numerical simulation is carried out with a multiple frame of reference (MRF) approach. The computational data of the steady flow simulation in the pump is taken as the initial condition of the unsteady computation using sliding mesh model. For each impeller revolution, calculation is performed in a time sequence of 360 time steps and the time step is set as $\Delta t=1.7\times10^{-4}$ s.

2.2. Parameters of the centrifugal pump
It is important to note that the pump analyzed in present research is designed for engineering application. The pump contains three computational domains: inlet, volute and impeller as is shown in figure 1. Table 1 shows the main performance and geometry parameters of the pump.

*Figure 1. Prototype of centrifugal pump.*
2.3. Mesh generation
The quality of grid has a great influence on the accuracy of numerical simulation and the structure mesh is generated in the computation domain throughout the flow passages of the pump. The centrifugal pump is divided into three regions: inlet, impeller and volute. Each region is discretized independently. The cells for the three regions are 835826 cells, 2546314 cells and 1846573, respectively. In total, the model has 5228713 cells. This mesh size can obtain satisfactory simulation accuracy, give correct result for the pump performance and allow details of the main flow pattern involved to be analyzed.

2.4. Mesh independence test
The non-dimensional head coefficients are defined as:

$$\psi = \frac{gH}{N^2 r^2}$$  \hspace{1cm} (1)

where $\psi$ is the head coefficient, $H$ is the head of the centrifugal pump, $N$ is the rotate speed and $r$ is the outer radius of the impeller. The power coefficient $P_c$ is defined as:

$$P_c = \frac{P_{sh}}{\rho N^2 \pi r^4}$$ \hspace{1cm} (2)

$$P_{sh} = M \times N$$ \hspace{1cm} (3)

where $\rho$ is the density, $P_{sh}$ is the consumed shaft power, and $M$ is the measured torque. The mesh independence of the computation has been validated with five sets of meshes. Table 2 gives head coefficient and power coefficient of centrifugal pump under five different mesh numbers, which is 1.89 million, 2.85 million, 5.22 million, 6.50 million and 10.57 million, respectively. In the present study, taking the calculation accuracy and computational resource into consideration the case with 5.22 million cells is selected for the final simulation. The mesh is fine enough to satisfy $y^+<200$ near the wall and to obtain the detail pressure data.

### Table 1. Performance and geometry parameter.

| Performance parameter | Flow rate $Q$ (m$^3$·h$^{-1}$) | Head $H$ (m) | Rotate speed $N$ (min$^{-1}$) | Efficiency $\eta$ (%) | Specific speed $n_s$ (m$^3$·s$^{-1}$, m, min$^{-1}$) |
|-----------------------|--------------------------------|-------------|-------------------------------|----------------------|---------------------------------------------------|
| Impeller inlet diameter $D_0$ (mm) | 230 |
| Impeller outlet diameter $D_1$ (mm) | 450 |
| Blade inlet width $b_1$ (mm) | 64 |
| Blade outlet width $b_2$ (mm) | 124 |
| Outlet diameter $D_2$ (mm) | 200 |
| Blade number $Z$ | 4 |

### Table 2. Mesh independence test.

| Condition | Mesh (million) | $\psi$ | $P_c$ |
|-----------|---------------|-------|-------|
| 1         | 1.89          | 5.378E-3 | 8.086E-04 |
| 2         | 2.85          | 5.305E-3 | 8.034E-04 |
| 3         | 5.22          | 5.263E-3 | 7.953E-04 |
| 4         | 6.50          | 5.257E-3 | 7.933E-04 |
| 5         | 10.57         | 5.253E-3 | 7.896E-04 |
3. Results and discussions

3.1. Validation of CFD results
Experiments for the performance of the pump are conducted in the laboratory of Zhejiang Sci-Tech University [14]. The head, flow and rotation speed at design condition are $H_d=25$ m, $Q_d=551$ m$^3$·h$^{-1}$, $N_d=980$ min$^{-1}$, respectively. The test rig is composed of a centrifugal pump, a water tank and circulation line system. A three-phase alternating current asynchronous motor is used as driving motor to drive the pump. The torquemeter is installed between motor and pump to obtain the torque. Electromagnetic flowmeter is used to measure the instantaneous volume flow rate in the circulation line system.

Figure 2 shows the performance curves of the head, which is compared with the experimental data. It is seen from figure 2 that the head-flow rate ($H-Q$) curve of simulations is basically in agreement with the experiments. The results of simulation is slighter larger than the experimental data. The maximum of the relative error is within 5%. Figure 3 shows the relationship between flow rate at the outlet and time. In the unsteady computation, it is found that the flow rate reach its steady values after a time interval of 0.42 s. The pressure record is started from 0.42 s. The flow rate of the experiment is 551 m$^3$·h$^{-1}$ and the relative error between the experimental data and the predicted flow rate is about 2.5%. This indicates that the computational model, the mesh system and the boundary conditions in the present computation are valid to predict the performance of the centrifugal pump.

![Figure 2. Head curve of the pump.](image1)

![Figure 3. Relationship between flow rate $Q$ and $t$.](image2)

3.2. Pressure fluctuation under different flow rates
The strong pressure fluctuation in the tongue region plays a dominant role in the stability of the internal flow field. In order to obtain an improved understanding of the internal flow characteristics in the tongue region of the centrifugal pump, monitor points are arranged, as is shown in figure 4.
Figure 4. Distributions of monitor points.

Figure 5. Average pressure at the monitor points under different flow rates.

Figure 5 shows distributions of the average pressure at the monitor points under different flow rates. With the increase of the flow rate, the average pressure difference at the monitor points is decrease. When $Q=1.3Q_d$ flow rate, average pressure at the monitor points is almost the same. However, when $Q$ is not equal to $1.3Q_d$ flow rate, the value of average pressure at the monitor points no longer remain the same value. On the whole, average pressure at monitor points 4, 5 is smaller than other points of the flow rate except $1.3Q_d$ flow rate. It is indicated that the average pressure at the monitor point
which locates at the right side of the tongue is larger than that of the left side of the tongue in a period of impeller rotation, especially at the small flow rate. At small flow rate, the average pressure difference at the monitor points is relative large, especially at the $0.4Q_d$ flow rate. The complex flow makes the distribution of the average pressure at the monitor points in the tongue cause great difference distribution under different flow rates and it is also indicates that it is necessity to explore the complex flow near the tongue region.

Figure 6 shows distributions of the standard deviation at the monitor points under different flow rates. The value of standard deviation at the monitor points is relative large and non-uniform at $0.4Q_d$ flow rate, which reveal the strong pressure fluctuation. The largest value of standard deviation is obtained at monitor point 5. At $0.7Q_d$ flow rate, standard deviation difference at the monitor points is narrowed and the value is relative small when compared with $0.4Q_d$ flow rate. At the design flow rate, standard deviation at each monitor point reaches its minimum value, which reveals the good flow stability with low vibration and high efficiency. The standard deviation is increased at the $1.3Q_d$ flow rate and the value of standard deviation at the monitor points is almost the same. The minimum standard deviation is obtained at the design flow rate, which is corresponding to the stable internal flow field. When the flow rate deviates from the design flow condition, the internal flow situation will become worse and finally result in large standard deviation. Standard deviation at the monitor points are increased apparently under $0.4Q_d$ flow rate, which was affected the complex flow around the tongue region.

![Figure 6. Standard deviation at the monitor points under different flow rates.](image-url)
3.3. Prediction of pressure fluctuation around the tongue region

Figure 7. Pressure fluctuation and frequency distributions at monitor points.
Figure 7 shows pressure fluctuation and frequency spectrum distributions at the monitor points around the tongue region under different flow rates. Frequency, position at monitor points and amplitude are respectively along $x$, $y$, $z$ direction in frequency distribution, as is shown in figure 7. Frequency and amplitude of the pressure fluctuation are obtained by fast Fourier transformation (FFT). The pressure fluctuation at the monitor points in figure 7(d) reveals the similar pressure fluctuations around the tongue region. It is found that the pressure fluctuation has four large pressure fluctuations amplitude and it is caused by the interaction between four blades and the volute in a period of impeller rotation.

Distribution of pressure fluctuation amplitude in frequency domain is irregular at small flow rate. The $f_{BPF}$ become the dominate frequency, which reveals the regular distribution of pressure fluctuation amplitude with the increase of the flow rate. The dominate frequency is about 65 Hz which is equals to the blade passing frequency ($f_{BPF}$) except the small flow rate. The dominate frequency is four times of the impeller rotating frequency. At $0.4Q_d$ flow rate, the pressure fluctuation amplitude in time domain and frequency domain at the monitor points is irregular and non-uniform. The pressure fluctuation amplitude in frequency domain appears several peak values. The amplitude of pressure fluctuation in frequency domain reaches its maximum at monitor 5. The increased frequency value is caused by the complex pressure fluctuation distribution at the monitor points. With the increase of the flow rate, pressure fluctuation amplitude in both time domain and frequency domain are decreased at $0.7Q_d$ flow rate. The pressure fluctuation amplitude in both time domain and frequency domain obtain its minimum value under $1.0Q_d$ flow rate, which indicated the stability internal flow field. With further increase of the flow rate, distribution of pressure fluctuation at the monitor points is similar, regular and uniform, which indicates the similar pressure fluctuation at the monitor points around the tongue region.

Pressure difference at the monitor points is slightly increased and so is the pressure fluctuation amplitude in frequency domain at $1.3Q_d$ flow rate. Extrusion process and expansion process are corresponding to the process of blade approaching to the tongue and leaving the tongue, respectively. When the blade is approaching to the tongue, the fluid in this region will be extruded. When the blade is leaving the tongue period, the fluid in this region will be expanded. During the blade approaching to the tongue and leaving the tongue period, it will result in large pressure fluctuation. With the increase of the flow rate, pressure fluctuation amplitude in time domain and frequency domain becomes relative regular and uniform.

3.4. Distribution of pressure on the centre plane
Figure 8 shows distribution of pressure on the centre plane under different flow rates. At the small flow rate, distribution of pressure is worse and non-uniform, as is shown in figure 8. The non-uniform distribution of pressure is easily to form the flow separation which makes the flow more easily tending to be unstable. Furthermore, it increases instability of the internal flow field. The pressure fluctuation amplitude in time domain and frequency domain at the monitor points are increased obviously under the small flow rate. With the increase of the flow rate, the flow of the internal flow field become stable and distribution of pressure is relative uniform. At $0.4Q_d$ flow rate and $0.7Q_d$ flow rate, the pressure around the tongue region remains large difference. As the flow rate increase to $1.0Q_d$ flow rate and $1.3Q_d$ flow rate the pressure difference around the tongue region decrease. It is noticed that there is a small region low pressure at the tailing edge of the blade. At $1.3Q_d$ flow rate, the low pressure region at the tailing edge of the blade is increased, when compared with $1.0Q_d$. During a period of impeller rotation, the low pressure region at the tailing edge of the blade and increased flow rate will finally result in the increase the pressure fluctuation amplitude in frequency domain. The pressure distribution on the centre plane at different flow rate above is found to be in agreement with the pressure fluctuation amplitude in frequency domain.
4. Conclusions
Numerical simulation is performed for the three-dimensional in a centrifugal pump in order to study the pressure fluctuation around the tongue region. The flow is governed by the unsteady incompressible Navier-Stokes equations coupled with the RNG $k$-$\varepsilon$ turbulent model. The numerical simulation results are found to be in agreement with the experimental data, which proof that those results are reliable. Detail pressure fluctuation amplitude in time domain and frequency domain are obtained in order to analysis the pressure fluctuation around the tongue region. The conclusions obtained are as follows:

1. With the increase of the flow rate, the pressure frequency around the tongue region first decrease and then increase, the minimum of which appears at the design condition.
2. When the flow rate is larger than the design flow rate, average pressure and standard deviation at monitor points is relative regular and uniform. The dominate frequency of pressure fluctuation is the blade passing frequency and the amplitude of pressure fluctuation is regular.
3. At small flow rate, amplitude of pressure fluctuation in frequency domain is larger than that at design flow rate condition. The maximum amplitude of pressure fluctuation in frequency domain exists at the monitor point just behind the tongue along the impeller rotation direction.

Acknowledgements
This work is supported by Zhejiang Province Key Science and Technology Innovation Team Project (2013TD18), the Natural Science Foundation of Zhejiang Province (LQ16E090005) and the National Natural Science Foundation of China (51579224).

Nomenclature

- $\psi$ = head coefficient, dimensionless
- $P_c$ = power coefficient, dimensionless
- $\rho$ = density, kg·m$^{-3}$
- $f_{BPF}$ = blade passing frequency, Hz
- $r$ = outer radius of the impeller, m
- $P_{sh}$ = consumed shaft power, N·m·s$^{-1}$
- $M$ = measured torque, N·m
References

[1] Sano T, Yoshida Y, Tsujimoto Y, Nakamura Y and Matsushima T 2002 J. Fluids Eng. Numerical study of rotating stall in a pump vaned diffuser 124 363-370

[2] Westra R W, Broersma L, Van Andel K and Kruyt N P 2010 J. Fluids Eng. PIV measurement and CFD computations of secondary flow in a centrifugal pump impeller 132 061104

[3] Jia X Q, Cui B L, Zhang Y L and Zhu Z C, 2015 Int. J. Turbo. Jet. Eng. Study on internal flow and external performance of a semi-open impeller centrifugal pump with different tip clearances 32 1-12

[4] Zhang N, Yang M G, Gao B, Li Z and Ni D 2014 Adv. Mech. Unsteady pressure pulsation and rotating stall characteristics in a centrifugal pump with slope volute Artical ID710791 11

[5] Li X J, Yuan S Q, Pan Z Y, Li Y and Liu W 2013 J. Appl. Math. Dynamic characteristics of rotating stall in mixed flow pump Artical ID104629 12

[6] Ran H J, Luo X W, Zhu L, Zhang Y, Wang X and Xu H Y 2012 Chin. J. Mech. Eng. Experimental study of the pressure fluctuations in a pump turbine at large partial flow conditions 25 1205-09

[7] Spence R and Amaral-Teixeira J 2009 Comput. Fluids A cfd parametric study of geometrical variations on the pressure pulsations and performance characteristics of a centrifugal pump 38 1243-57

[8] Wu Y L, Li S C, Liu S H, Dou H S and Qian Z D 2013 Vibration of hydraulic machinery (Berlin: Springer)

[9] Qu L X, Wang F J, Cong G H and Gao J Y 2011 Transactions of the Chinese Society for Agricultural Machinery Effect of Volute Tongue-impeller Gaps on the Unsteady Flow in Double-suction Centrifugal Pump 42 50-55(in Chinese)

[10] Cai J C, Pan J, Guzzomi A 2015 Transactions of the Chinese Society for Agricultural Machinery Pressure fluctuations around volute tongue of centrifugal pump 46 92-96(in Chinese)

[11] Tan L, Wang Y C, Cao S L and Zhu B S 2014 Transactions of Beijing Institute of Technology Characteristics of unsteady flow around the tongue region in a centrifugal pump 34 670-675(in Chinese)

[12] Majidi K 2005 ASME J. Turbomach. Numerical study of unsteady flow in a centrifugal pump, 127 363-371

[13] Spence R and Amaral-Teixeira J 2008 Comput. Fluids Investigation into pressure pulsation in a centrifugal pump using numerical methods supported by industrial tests 37 690-704

[14] Ren Y, Wu D H, Liu H L and Jiang L F 2015 Transactions of the Chinese Society for Agricultural Machinery PIV experiment on flow instabilities in centrifugal pump 46 46-51(in Chinese)

[15] Yuan S Q, Zhou J J, Yuan J P, Zhang J F, Xu Y P and Li T 2012 Transactions of the Chinese Society for Agricultural Machinery Characteristic analysis of pressure fluctuation of unsteady flow in screw-type centrifugal pump with small blade 43 83-87(in Chinese)

[16] Cong G H and Wang F J 2008 Transactions of the Chinese Society for Agricultural Machinery Numerical investigation of unsteady pressure fluctuations near volute tongue in a double suction centrifugal pump 39 60-67(in Chinese)