Investigation on impeller radial force for double-suction centrifugal pump with staggered blade arrangement

Z C Zhang, F J Wang, Z F Yao, H F Leng and P J Zhou

College of Water Resources and Civil Engineering, China Agricultural University, Beijing 100083, China

E-mail: zhangzichao@sina.cn

Abstract. In order to find the effects of blade arrangement on impeller radial force, a double-suction centrifugal pump with two impeller configurations is investigated by using CFD approach. The two impeller have same geometry, same blade number, and different blade arrangement. One is staggered impeller in which the blades are arranged with half of blade phase angle staggered in circular direction, another is traditional symmetrical impeller with symmetrical blade arrangement. Results show that the radial force vector diagram for symmetrical impeller is a hexagonal, while it is nearly a circle for staggered impeller. The staggered impeller results no radial force saltation which exists in symmetrical impeller. The blade passing frequency dominates the radial force fluctuation in symmetrical impeller, while this frequency is almost not existed in staggered impeller. The results indicate that staggered blade arrangement can significantly reduce radial force fluctuation in double-suction centrifugal pump.

1. Introduction

Double-suction centrifugal pumps are widely used in various engineering fields. The operation stability of this kind pump is becoming an important issue with its capacity increasing fast. The rotor-stator interaction between impeller and tongue is the primary source of vibration. Many studies about centrifugal pump pressure fluctuations and radial force have been done [1-3]. Although axial forces of double-suction pumps are theoretically balanced, influence of radial force on vibration can not be ignored. High radial force can result in fatigue failure of pump shaft and low volume efficiency [4].

Centrifugal pump radial force can be influenced by many factors, for example the gap between impeller and tongue. Barrio R investigated radial force of impeller with different gaps between impeller and tongue by changing impeller diameter [5]. Solis M got different gaps between impeller and tongue by changing tongue angle to study pressure fluctuations [6]. Zhu Lei simulated the unsteady flow in centrifugal pump with three different types tongue and compared the radial forces [7].

Impeller radial force fluctuation is usually thought to be caused by pressure fluctuation in the impeller and the volute casing. Effects of impeller type on pressure fluctuations have been investigated by some researchers. Yang Min found that pressure fluctuations in double-suction centrifugal pump can be reduced by staggered blade arranging on both sides of the impeller [8]. Yao Zhifeng measured fluctuating pressure on the walls of suction chamber and spiral casing in five double-suction centrifugal pump impeller configurations. Results show that the amplitude of pressure fluctuations in the pump with staggered blade layout is lower [9]. But researches on influence of impeller type on
radial force of double-suction centrifugal pump are very few at present. So it is necessary to reveal the relation between the radial force and the blade arrangement for double-suction centrifugal pump. In this study, two kinds of impellers with different blade arrangement are designed. The unsteady flows in the double-suction centrifugal pump with different impeller are simulated by using CFD approach. Influence of blade arrangement on radial force of double-suction centrifugal pump is analyzed in detail.

2. Physical configuration and numerical model

2.1. Geometry and grid

The rated parameters of the investigated double-suction centrifugal pump are as follows: flow rate \( Q = 280 \text{m}^3/\text{h} \), Head \( H = 45 \text{m} \), speed \( n = 2950 \text{r/min} \), specific speed \( n_s = 125 \). In order to exam the effects of blade arrangement on radial force, the impeller blades are arranged in two different styles. Thus two impellers — symmetrical impeller and staggered impeller are obtained, as shown in Figure 1. Symmetrical impeller is that the blades in both sides are arranged symmetrically, staggered impeller is obtained by arranging blades with half of blade phase angle (30 degree in this case) staggered on both sides of the impeller. The two impellers have common parameters: impeller outlet width \( b_2 = 40 \text{mm} \), impeller inlet diameter \( D_1 = 130 \text{mm} \), impeller outlet diameter \( D_2 = 204 \text{mm} \), the number of blade for impeller \( Z = 6 \).

![Figure 1. Two impellers with different blade arrangement](image)

2.2. Setup of unsteady simulation

ANSYS 13.0 is applied and the unsteady flow in the pump is simulated by using the Reynolds averaged Navier–Stokes approach with k-ε turbulent model. The mass flow of inlet and the average static pressure of outlet are selected as the boundary condition, the volute casing and suction section walls are in stationary frame and modeled using a no-slip boundary condition. The time step is 0.0001695s, which is 1/120 of impeller rotational period. Initializing the unsteady calculation with the steady solution, a total 10 revolutions is set for the computations to achieve the periodic unsteady solution convergence. The maximum iteration loops is set to 10 in order to reduce all the root means
square (RMS) residual below 10E-5. During the calculations, the radial force is monitored and recorded at every time step.

3. Results and discussions

3.1. Pump general performance
According to effective operating range of the double-suction centrifugal pump, numerical simulations at four operating points \(0.6Q_n, 0.8Q_n, 1.0Q_n, 1.1Q_n\) for each impeller are conducted.

To verify credibility of the calculated results, an experimental measurement for pump performance of symmetrical impeller is made. The predicted time-average head and time-average efficiency in a calculating cycle under four operating points are compared with experimental results, as shown in Figure 3. It shows that the calculated efficiency and head coincide with experimental data. The agreement indicates that the calculation is reasonable and can be used to perform detail analysis.

![Figure 3. Performance prediction and experimental verification](image)

3.2. Outlet static pressure distribution of impeller
Radial force is caused by nonuniform distribution of static pressure surrounding the impeller. Thus the outlet static pressure of impeller is analysed. Outlet sections of impeller are shown in Figure 4, 1 and 1’ section are located in the middle of each side outlet of impeller respectively. Figure 5 shows impeller outlet static pressure distribution of two sections (1 and 1’) for two kinds of impellers along outlet periphery of impeller at four operating conditions. In the figures, abscissa \(\alpha\) is the angle between position of impeller outlet and tongue, tongue is located in 0 degree, ordinate \(p_j\) is static pressure.

![Figure 4. Impeller outlet sections](image)

Figure 5 shows that for two impellers, the static pressure increases from tongue at partial flow rates, while decreases from tongue at a large flow rate. The static pressure distribution is uniform at design condition. The periodic fluctuations are caused by the interaction between tongue and trailing edge of blade.

Under various operation conditions, static pressure distributions of two sides in symmetrical impeller are the same, while 30 degree phase difference exists in two sides of
staggered impeller. Also static pressure fluctuations of staggered impeller are lower than that of symmetrical impeller near the volute tongue under various operation conditions. The difference of static pressure distribution in two impellers may lead to different radial force distribution, because radial force and outlet static pressure are closely related.

Figure 5. Comparison of impeller outlet static pressure distribution

3.3. Radial force vectors
Rectangular coordinate system is established to show radial force vector, as shown in Figure 6. It gives the radial force vectors monitored in one rotational period under various operation conditions. In Figure 6, intersection of X-dotted line and Y-dotted line is origin of coordinates. Because of short and wide flow passage, the total radial force is the sum of radial force on the surfaces inside impeller flow passage and outside surfaces of shroud.

Figure 6 shows that amplitude and direction of radial force are changing at every time step. The vector diagram at every operation condition is on the whole circular. The centre of the circle is located in origin of coordinates at design condition, at partial flow rates the centre is located in the minus x direction, at a large flow rate the centre is located in the plus x direction. The distance between origin of coordinates and the centre of the circle becomes farther as the flow rate is far from design condition, the condition indicates the radial force increases as the flow rate is far from design point. The saltation of radial force of symmetrical impeller is more obvious than the staggered one. The staggered impeller almost eliminates the saltation of radial force. This phenomenon results from interaction of blade and tongue which
leads to different outlet static pressure distribution of two impellers. Radial force of staggered impeller is smaller than symmetrical impeller on the whole. Radial force direction in a rotational period changes against the direction of impeller rotation. Figure 6 shows that arrow in vector diagram and arrow in impeller structure sketch are opposite.

3.4. Radial force fluctuations
As treatment for pressure fluctuations, radial force fluctuations are introduced. Here, radial force fluctuations are normalized to radial force coefficient $C_F$, which is defined as

$$C_F = \frac{(F - \overline{F})}{0.5\pi D_b \rho u_z^2}$$

where $\overline{F}$ is the average value of radial force among all time steps for 10 rotor revolutions, $u_z$ is the impeller outlet revolution speed.

Figure 7 shows the frequency spectra of radial force fluctuations for the two type pumps under various operation conditions, where $f_r$ is the rotating frequency, $f_r = 49$Hz. The dominant frequency and corresponding amplitude of the radial force fluctuations are obtained by Fast Fourier transform (FFT).

Figure 7 shows some common features for the two impellers: (1) The dominant frequency is rotating frequency ($f/f_r=1$); (2) Radial force fluctuation at design condition is lower than that at off-design conditions; (3) The amplitude of radial force fluctuation becomes higher as the flow rate is far from design flow rate. However, some different features are also seen in Figure 7. The blade passing frequency ($f/f_r=6$) for symmetrical impeller is second dominant,
while it can not be found in staggered impeller. This indicates that staggered impeller can reduce vibration of impeller and fatigue failure of shaft causing by alternating load.

4. Conclusions
Effects of two impeller configurations on impeller radial force for double-suction centrifugal pumps are investigated by using CFD approach.

1. Phase difference exists in two side outlet static pressure distributions for the staggered impeller, while no difference is found for the symmetrical one. So two kinds of impeller have different radial force characters.

2. The radial force vectors are periodic whether for symmetrical impeller or for staggered impeller. Radial force direction in a rotational period changes against the direction of impeller rotation. The vector diagram for symmetrical impeller is a hexagonal, while it is nearly a circle for staggered impeller. This means that staggered impeller results no radial force saltation which exists in symmetrical impeller.

3. For any type of impeller, the amplitude of radial force fluctuation becomes higher when the flow rate gets far from design flow rate. The dominant frequencies for symmetrical impeller are rotating frequency and blade passing frequency. While for staggered impeller, dominant frequency is mainly rotating frequency, the blade passing frequency is almost eliminated totally. The staggered impeller gives better radial force fluctuation performance than the symmetrical impeller.

Acknowledgments
The authors would like to acknowledge the financial supports given by the National Natural Science Foundation of China (No.51139007).

References
[1] Zhang M and Tsukamoto H 2005. *Trans. ASME J. Fluids Eng.* 127(4) 743-51
[2] Yang M, Min Siming et al. 2009 *Trans. Chin. Soc. Agric.Mach.* 11 (40) 83-88
[3] Guo S J and Okamoto H 2003 *Int. J. of Rotating Mach.* 9(2) 135-44
[4] Zhao W Y, Zhang L et al. 2009 *Drain. Irrig. Mach.* 27(4) 205-9
[5] Barrio R, Blanco E et al. 2008 *Trans. of ASME J. of Fluids Eng.* 130(11) 111102-1-111102-11
[6] Solis M and Bakir F et al. 2009 Pressure fluctuations reduction in centrifugal pumps influence of impeller geometry and radial gap *Proc. ASME 2009 FEDSM* (Colorado, USA ,2-6 August 2009)
[7] Zhu L, Yuan S Q et al. 2011 *Trans. of the CSAE* 27 (10) 50-55
[8] Yang M, Wang F J et al. 2008 Performance improvement of double-suction centrifugal pump by using CFD *24th Symposium on Hydraulic Machinery and Systems* (Foz Do Iguassu, Brazil , 27-31 October 2008 ) 111
[9] Yao Z F, Wang F J et al. 2011 *J. Mech. Eng.* 47 (12) 133-43
[10] Gonzalez J, Parrondo J etc. 2006 *Trans. ASME* 128 454-62