Proceedings of the ASME 2010 3rd Joint US-European Fluids Engineering Summer Meeting and 8th International Conference on Nanochannels, Microchannels, and Minichannels
FEDSM-ICNMM2010
August 1-5, 2010, Montreal, Canada

FEDSM-ICNMM2010-30407

ANALYSIS OF FLOW-INDUCED VIBRATION OF THE VOLUTE OF A CENTRIFUGAL PUMP BASED ON FINITE ELEMENT METHOD

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ABSTRACT

During the operation of centrifugal pumps, radial hydraulic force is generated due to non-uniform flow within pumps, which is one of the main sources of the vibration of the centrifugal pump volute. In this paper, based on CFD and finite element method, it was calculated and analyzed that the volute vibration of a centrifugal pump caused by radial hydraulic force. The reason of the occurrence of radial force was analyzed, and by simplifying the theoretical formulas the force was calculated. Then the unsteady flow field of a centrifugal pump was simulated and analyzed under different running conditions by CFD method. Based on the simulation results, the radial hydraulic force of the pump was calculated. Finally, vibration response of the pump volute due to the hydraulic radial force was obtained. By analyzing the vibration response datum, vibration parameters were got such as the displacement, velocity and acceleration of vibration. It was obtained that the main vibration frequencies of the pump volute which is caused by unsteady flow are blade frequency and its harmonic frequencies. The pump volute has a minimum vibration under design flow rate condition, and it has a maximum vibration at the 1.5 times design rated flow whilst the vibration frequency is the integral multiple of the blade frequency. This study is helpful to understand the flow-induced vibration of pump volute and to improve the hydraulic design of the centrifugal pump.

Key words: centrifugal pump, finite element method, flow-induced vibration, frequency response

INTRODUCTION

Pumps are widely used in many fields as universal machines. With the development of the technology, pump is required to have not only high efficiency but also low noise and vibration. Many factors cause the noise and vibration of the centrifugal pump. The most important one is the hydraulic radial force which is caused by complex inner flow, so the research of the influence of the hydraulic radial force on the noise and vibration of the centrifugal pump has made great sense.

Many investigations of the hydraulic radial force and vibration of the centrifugal pump have been done by many researchers. Chen Hongxun [1] did numerical simulations of the inner flow of the pump and calculated the surface pressure of the blade through setting up the mathematical model and using potential flow and boundary element solution in order to obtain the hydraulic radial force. The influence of the hydraulic design on the vibration of the centrifugal pump was studied based on experiments by He Xijie [2]. Specifically, the influence of two factors, the gap between the tongue and the impeller and the laying angel of the tongue, was analyzed in detail. Wu Renrong [3] made comparisons of several hydraulic design methods, which can help to reduce the vibration and the noise of the pump systematically and summarized the factors resulting in the vibration and noise. Also some optimum design methods were proposed. Some international researchers were engaged in the noise and vibration generating mechanism by setting up numerical models and measuring pressure pulsations [4-8]. Langthjem believed that the main noise-generating mechanism was the unsteady impeller blade surface forces [9,10]. However, the research of the vibration of the pump volute caused by hydraulic radial force has not been carried out yet. How the hydraulic radial force was generated and then affected the vibration of the pump volute have not been clarified in detail.

In this paper the numerical simulation of the inner flow of the centrifugal pump was presented. The cause and the character of the radial force were analyzed. The frequency response of the pump body under the hydraulic radial force was calculated using finite element method. The research in this paper can predict the dynamic character of the pump, so it has made great sense in understanding the mechanism of the vibration resulted from the inner unsteady flow in the centrifugal pump.

NOMENCLATURE

b Impeller outlet width, [mm]
B_2 Impeller outlet width including shroud, [m]
D Impeller outside diameter, [m]
F Radial force, [N]
H Pump head, [m]
n Node number, [-]
K_r Empirical coefficient, [-]
p Pressure on the casing inner surface, [Pa]
Q Volume flow rate, [m^3/h]
Q_d Designing flow rate, [m^3/h]
S Area of the casing inner surface, [m^2]

Greek symbols

α Angle between the force and normal direction, [°]
θ Angle between the force and the coordinate axis X, [°]

ANALYSES AND CALCULATIONS OF HYDRAULIC RADIAL FORCE

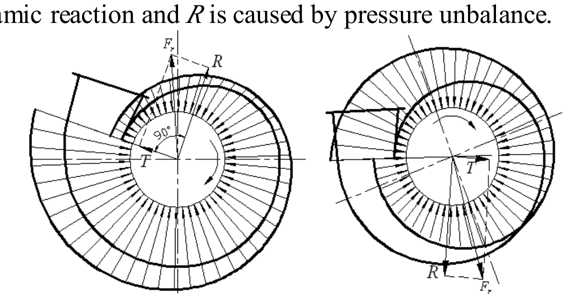
Many factors can cause vibration and noise in centrifugal pumps, one of the most important factors is the hydraulic radial force. The generation of the hydraulic radial force is mainly due to the inhomogeneous distribution of the circumferential flow field.

Causes of hydraulic radial force

Inhomogeneous distribution of the pressure is one of the most important causes of the hydraulic radial force. The spiral volute was designed according to the flow rate. Under the design condition, the velocity and pressure of the fluid in the pump chamber is uniform and symmetric, so there is no hydraulic radial force theoretically.

The unsymmetric distribution of the stress under off-design condition, however, leads to generation of the radial force. When the pump runs below design flow rate condition, the velocity in the pump volute is becoming increasingly small from the tongue and the pressure increases. But the absolute velocity at the impeller outlet increases and the direction of it changes. So when this flow meets fluid in the pump volute, there will exist collision because of different velocities and directions. That will lead to great energy loss and part of the kinetic energy will be transferred to pressure energy which will increase the pressure of the fluid in pump volute. As a result, from the tongue to the outlet of pump volute, the fluid is impacted by the flow out of the impeller continuously, the pressure increases and the pressure in the pump volute is increasingly high from the tongue along the inner surface of the casing. And when the flow rate is over design discharge, the flow velocity in pump volute will increase, so the pressure will decrease continuously from the tongue.

Another factor contributing to the radial force is the dynamic reaction imposed on the impeller by the outflow. The pressure water around the impeller impedes the outflow, of which the velocity is non-axisymmetrical because of the shape of the pump volute. The velocity of the flow varies as the pressure changes and the direction is opposite to the direction of absolute velocity at the impeller outlet, approximately being tangent with the impeller tip circle, so does the direction caused by the dynamic reaction. The direction points to the tongue at low flow rate while at large discharge the direction is opposite. As is shown in figure 1, the total radial forces are composed of T and R. Both of them are radial force, while T is caused by dynamic reaction and R is caused by pressure unbalance.



（a） $Q < Q_d$ （b） $Q > Q_d$

Fig.1 Causes of hydraulic radial force

Calculation methods of hydraulic radial force

The radial force is generally calculated by the following formula [11]:

$$F = 9.81 K_r H D B_2 \times 10^3 \quad \text{(N)} \tag{1}$$

As the formula shows, once the operating condition determined, the radial force of the impeller can be calculated. The radial force will be only a function of the flow rate as long as the centrifugal pump structure parameters remain unchanged according to the traditional radial force theory. The non-uniform circumferential distribution of flow rate and static pressure lead to the change of the radial force and energy loss in the flow channel. But the research shows that the circumferential distribution of the static pressure in centrifugal pumps varies with the change of relative position of the tongue and impeller, and then the magnitude and direction of radial force change all the time. As a result, the pump volute and impeller are subject to alternating the radial force.

The pressure distribution on the coupling face of the impeller and casing is discrete, so the force imposed on every node of the coupling face should be calculated firstly and then be decomposed in Cartesian coordinates system, component forces in X direction and Y direction. According to the composition theory, the total force can be solved.

In the calculation process, firstly the circumferential coupling face should be divided into 36 small faces with the same area, the force acted on the center of each small face can be monitored in the monitoring panel in the Surface→Point menu of the calculation software FLUENT. Then these forces need to be analyzed to calculate the component forces in the direction of coordinate axis, the magnitude and the direction of the resultant force can be obtained by superposition method. The calculation process listed as follows:

$$S = \pi Db \tag{2}$$

$$\Delta s = \frac{S}{36} = \frac{\pi Db}{36} \tag{3}$$

$$F_i = p_i \cdot \Delta s \quad (i=1, 2, \cdots, 36) \tag{4}$$

$$a_i = \frac{2\pi(i-1)}{36} = \frac{(i-1)\pi}{18} \tag{5}$$

$$\begin{cases} F_{x,i} = F_i \cdot \cos\alpha_i = p_i \cdot \Delta s \cdot \cos\frac{(i-1)\pi}{18} \\ F_{y,i} = F_i \cdot \sin\alpha_i = p_i \cdot \Delta s \cdot \sin\frac{(i-1)\pi}{18} \end{cases} \tag{6}$$

$$\begin{cases} F_x = \sum_{i=1}^{36} F_{x,i} = F_{x,1} + F_{x,2} + ... + F_{x,36} \\ F_y = \sum_{i=1}^{36} F_{y,i} = F_{y,1} + F_{y,2} + ... F_{y,36} \end{cases} \tag{7}$$

All the equations are combined to calculate the component force in the direction of the coordinate axis as follows:

$$\begin{cases} F_x = \frac{\pi D b}{36}(p_1 \cos 0 + p_2 \cos\frac{\pi}{18} + ... + p_{36} \cos\frac{35\pi}{18}) \\ F_y = \frac{\pi D b}{36}(p_1 \sin 0 + p_2 \sin\frac{\pi}{18} + ... + p_{36} \sin\frac{35\pi}{18}) \end{cases} \tag{8}$$

Both the magnitude and the component forces and the resultant forces under different operating conditions can be obtained considering the diameter and the width of the coupling face and the static pressure at each point.

The radial force can be calculated as follows:

$$F_r = \sqrt{F_x^2 + F_y^2} \tag{9}$$

And the angle between the radial force and the coordinate axis X is written as follows:

$$\theta = \arctan(\frac{F_y}{F_x}) \tag{10}$$

Calculation results and analyses of hydraulic radial force

The impeller outside diameter of the centrifugal pump in this paper is 165mm, and the impeller has 6 blades. The performance parameters of the pump are listed in table1.

Table1 parameters of the centrifugal pump

Flow rate (m^3/h)	Head (m)	Rotation speed (r/min)	Specific speed (n_s)	Inlet diameter (mm)	Outlet diameter (mm)
12.5	32	2900	47	50	32

Based on the above formula, the magnitude and the direction of the radial force can be calculated under three different operating conditions, which are $Q = 0.6Q_d$、$Q = 1.0Q_d$ and $Q = 1.5Q_d$. The distribution of the force can be obtained with the different relative position between the impeller and casing.

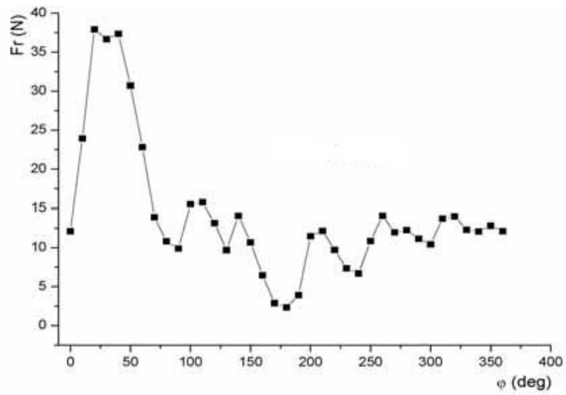
Some rules can be derived from the calculation results shown in figure 2 and 3

(1) The angle between the direction of the radial force and the positive x axis direction tend to be at the range from 225 to 360 degrees. It means that the direction of the radial force does not change a lot.

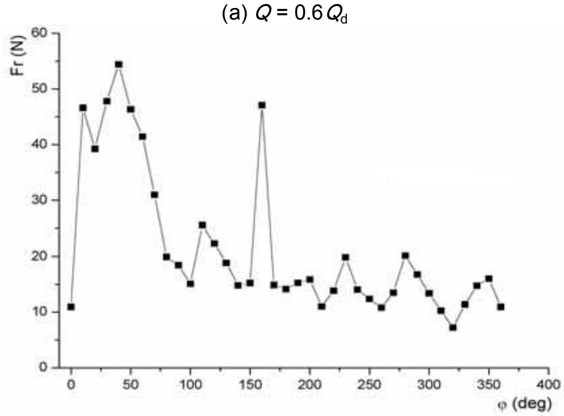
(2) All the radial forces get to maximum values when the blades are near the tongue. It can be proved that the interaction between the impeller and tongue is the strongest when they get close.

(3) There appear six maximum values of the force, during one period, which coincide with the fact that the centrifugal pump has six blades.

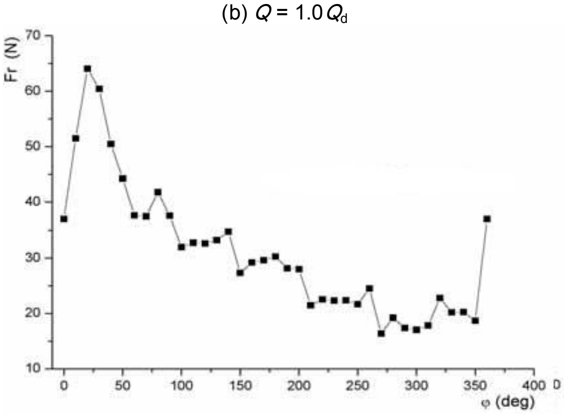
Through Fourier Transform, the radial force in time-domain can be transformed to that in the frequency domain. That transforming function could be the boundary condition of the pump volute frequency response analysis.



(a) $Q = 0.6Q_d$

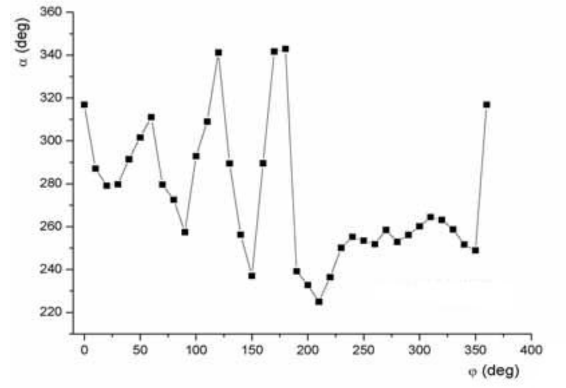


(b) $Q = 1.0Q_d$

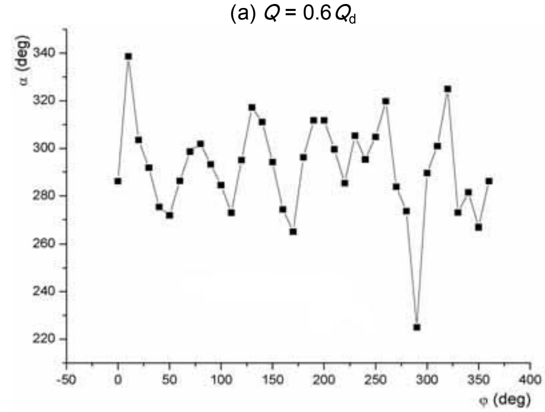


(c) $Q = 1.5Q_d$

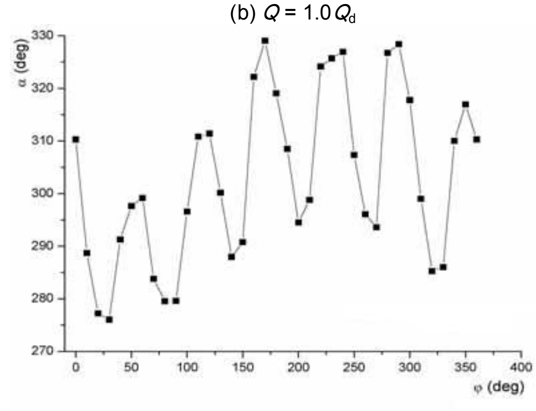
Fig.2 The distribution of the radial force as the relative position between the blade and the baffle tongue change



(a) $Q = 0.6Q_d$



(b) $Q = 1.0Q_d$



(c) $Q = 1.5Q_d$

Fig.3 The distribution of the direction of radial force as the relative position between the blade and the baffle tongue change

ANALYSES OF PUMP VOLUTE RESPONSE

Theory of Frequency Response Analysis

The frequency response analysis is used to calculate dynamic response of the pump structure in each calculation frequency with the external excitation. The calculation result consists of two parts, the real part which represents the response amplitude and the imaginary part which is the response phase angle. The motion equation of forced damped vibration in harmonic excitation is

$$[M]\{\ddot{x}(t)\}+[B]\{\dot{x}(t)\}+[K]\{x(t)\}=\{P(\omega)\}e^{i\omega t} \quad (10)$$

A harmonic formal solution should be assumed to simple harmonic vibration

$$\{x\}=\{u(\omega)\}e^{i\omega t} \quad (11)$$

In this solution, $\{u(\omega)\}$ is a complex displacement. The first order and the second order derivatives are

$$\{\dot{x}\}=\mathrm{i}\omega\{u(\omega)\}e^{i\omega t} \quad (12)$$

$$\{\ddot{x}\} = -\omega^2\{u(\omega)\}e^{i\omega t} \quad (13)$$

With combination of equations (11) , (12) and (13), we can get

$$-\omega^2[M]\{u(\omega)\}e^{i\omega t} + i\omega[B]\{u(\omega)\}e^{i\omega t} + [K]\{u(\omega)\}e^{i\omega t} = \{p(\omega)\}e^{i\omega t} \quad (14)$$

It can be simplified to be

$$-\omega^2[M]\{u(\omega)\} + i\omega[B]\{u(\omega)\} + [K]\{u(\omega)\} = \{p(\omega)\} \quad (15)$$

This equation can represent complex equation system. Every motion equation of input exciting frequency can be solved as static problem using complex algorithms.

Finite Element Model

The material of the pump volute is grey cast iron, the material properties are shown in the following table2.

Table2 Material properties of the pump body

Density kg/m^3	Elastic Modulus GPa	Poisson Ratio
7000	150	0.25

Tetrahedral mesh is used because of its good adaptability to complex boundary. The 3-dimensional finite element model is shown in figure 4.

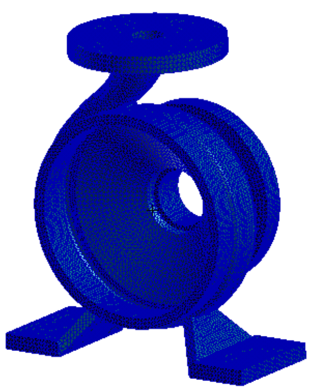


Fig.4 Centrifugal pump mesh

Boundary Condition and Load Condition infliction

Only the base of the pump is fixed without considering other constrains in order to simplify the analysis. The forces imposed on the pump volute are very complex but it is the radial force that contributes most to the vibration of the pump, so in this paper the analysis mainly concentrates in the force-induced vibration. The schematic diagram of radial force is shown in figure 5.

The frequency domain response of the centrifugal pump can be calculated based on the finite element model. The node close to the inner flow channel is chosen for the analysis in order to reflect the vibration characteristic of the pump volute accurately due to the radial force, so the influence of other factors, such as axial force, on the vibration characteristic can be neglected.

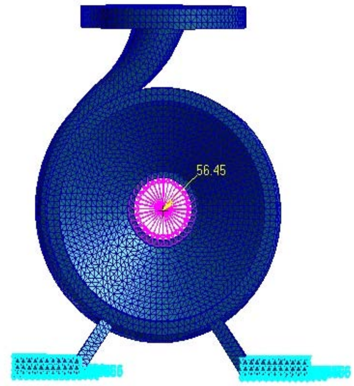


Fig.5 Radial force acted on the pump volute

Frequency Response Analyses

The results of frequency response analysis are the force vibration acceleration, velocity and displacement of the finite element node. The displacement, velocity, acceleration and stress amplitude frequency of the node on the pump volute under design condition is shown in the following figures, from figure 6 to figure 9 respectively.

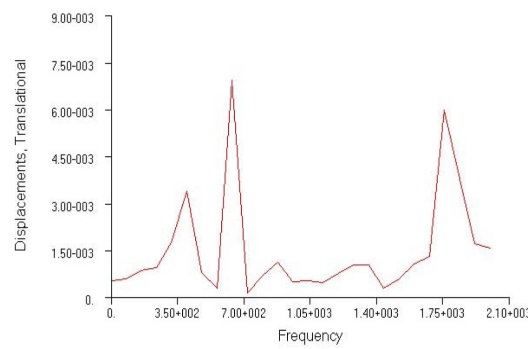


Fig.6 Displacement amplitude-frequency map at Q=1.0Q_d

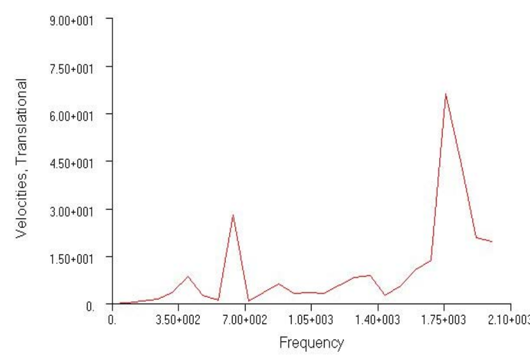


Fig.7 Velocity amplitude-frequency map at Q = 1.0Q_d

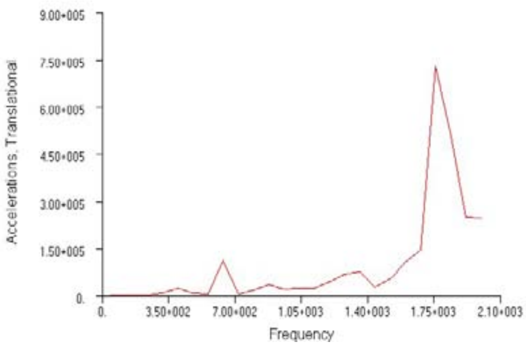


Fig.8 Acceleration amplitude-frequency map at $Q = 1.0Q_d$

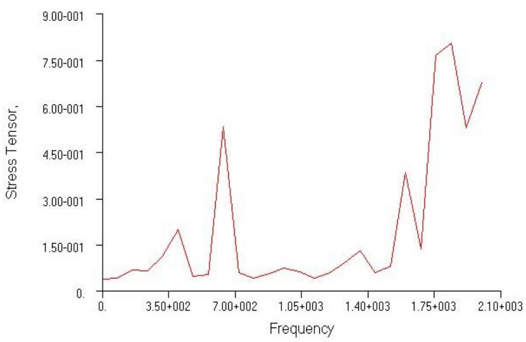


Fig.9 Stress amplitude-frequency map at $Q = 1.0Q_d$

Conclusions can be drawn from the figures that there are two peak values in all of the figures, which appear at the frequency of 630Hz and 1740Hz. The rotational speed of the pump is 2900r/min, so the shaft frequency of the pump is 48.33Hz. The peak frequencies are approximately 13 and 36 times of the fundamental frequency. The vibration of the pump volute below design condition mainly attribute to the high-order harmonic frequency of the shaft frequency.

The frequency responses in low and high flow rate operating conditions, $Q = 0.6Q_d$ and $Q = 1.5Q_d$ are also calculated in this paper. The amplitudes of displacement, velocity, acceleration and stress in three operating conditions, $0.6Q_d$、 $1.0Q_d$ and $1.5Q_d$ are shown in the figures, from figure 10 to figure 13 respectively。

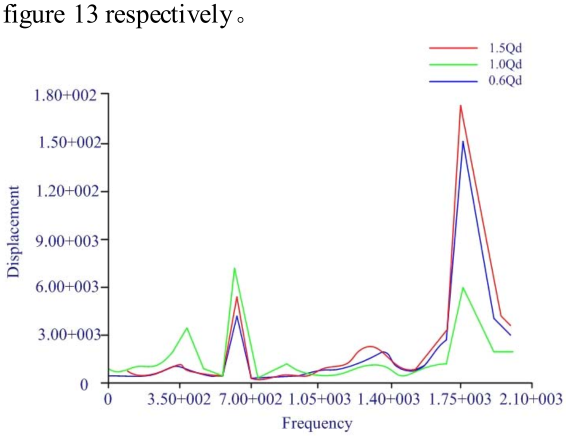


Fig.10 Displacement amplitude-frequency under three operating conditions

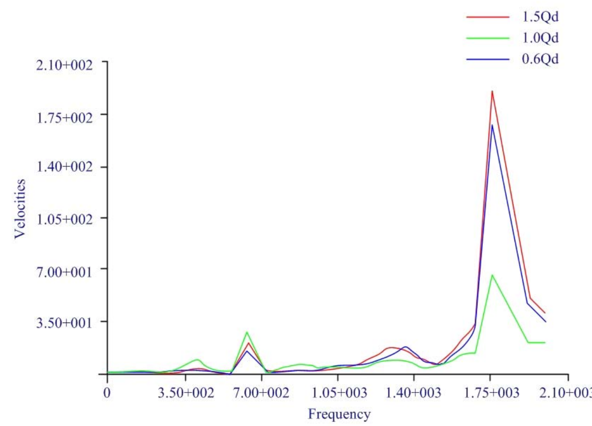


Fig.11 Velocity amplitude-frequency under three operating conditions

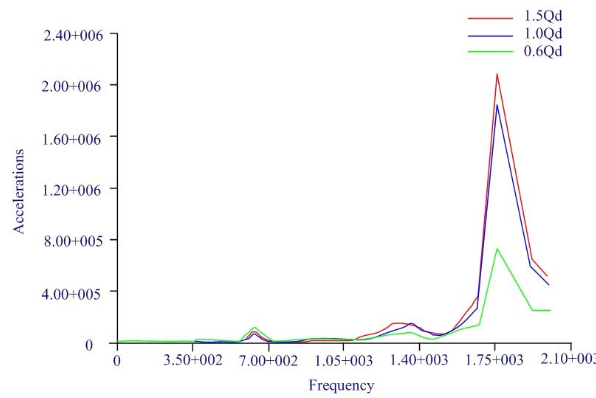


Fig.12 Acceleration amplitude-frequency under three operating conditions

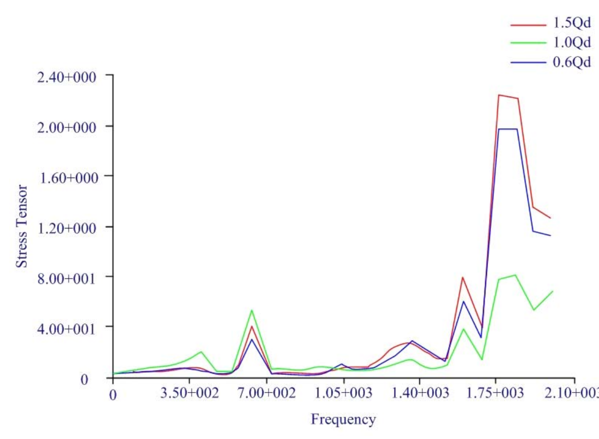


Fig.13 Stress amplitude-frequency under three operating conditions

These figures show that:

(1) There are two obvious peak values in all the figures at the frequency of 630Hz and 1740Hz, while at the frequency of 1595Hz (about 33 times of the shaft frequency), another peak also appears. All of these frequencies are integral multiple of the shaft frequency.

(2) The amplitudes of displacement, velocity, acceleration and stress reach the minimum value at design condition while at

low flow rate condition, the amplitude value will increase, at large flow rate condition, the amplitude will reach the maximum value.

CONCLUSIONS

The vibration characteristic of the pump volute caused by hydraulic radial force was studied in this paper. The software Fluent was used to simulate the inner flow of the pump. The distribution of the hydraulic radial force acted on the coupling face between the impeller and casing was calculated. The simulation results were used to analyze the finite element model of the pump. The research shows that:

(1) At the integral multiple shaft frequency (13 times and 36 times), the pump has very severe vibration, which mainly derives from the shaft frequency and harmonic frequency.

(2) The pump has least vibration at design condition, while at 0.6 times flow rate operating condition, the vibration will be strengthen and at 1.5 times flow rate operating condition, the vibration will reach the highest level.

All these can not only provide theoretical fundaments for the research of flow-induced noise and vibration, but also support the vibration analysis of other kinds of fluid machinery.

ACKNOWLEDGMENTS

This project is supported by China National Funds for Distinguished Young Scientists (No.50825902), National Natural Science Foundation of China (No. 50979034), and Natural Science Fund for Colleges and Universities in Jiangsu Province (No.BK2009218).

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