Prediction of flow-induced dynamic stress in an axial pump impeller using FEM

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Abstract. Axial pumps play an important role in water supply and flood control projects. Along with growing requirements for high reliability and large capacity, the dynamic stress of axial pumps has become a key problem. Unsteady flow is a significant reason which results in structural dynamic stress of a pump. This paper reports on a flow-induced dynamic stress simulation in an axial pump impeller at three flow conditions by using FEM code. The pressure pulsation obtained from flow simulation using CFD code was set as the force boundary condition. The results show that the maximum stress of impeller appeared at the joint between blade and root flange near trailing edge or joint between blade and root flange near leading edge. The dynamic stress of the two zones was investigated under three flow conditions (0.8Qd, 1.0Qd, 1.1Qd) in time domain and frequency domain. The frequencies of stress at zones of maximum stress are 22.9Hz and 37.5Hz as the fundamental frequency and its harmonics. The fundamental frequencies are nearly equal to vane passing frequency (22.9Hz) and 3 times blade passing frequency (37.5Hz). The first dominant frequency at zones of maximum stress is equal to the vane passing frequency due to rotor-stator interaction between the vane and the blade. This study would be helpful for axial pumps in reducing stress, improving structure design and fatigue life.

1. Introduction
Axial pump are used in a wide range of applications. However, many of these pumps are suffered from vibration and stress which induced by complex flow conditions more or less. The life of the pumps would be reduced due to the flow. The failure of axial pump is becoming more and more usual along with the growing requirements for high speed, high reliability and large capability.

Unsteady flow is a significant reason that causes the structural failure of pump. There were a lot of examples about flow-induced failure of centrifugal pump. In order to improve the stability of pump operation for pump, the research on flow characteristics of internal pump [1-2] is relatively more, while the study on the pump structural components for the stress characteristics is relatively less. The stress characteristic is helpful for prevention of fatigue damage and improvement the stability of pump.

Traditional stress calculation method [3] is to simplify the impeller blade force and calculate the stress using the theoretical formula. With the development of computational fluid dynamics, it is an effective method which use CFD technology to simulate and evaluate the surface of the water pressure on the impeller. This simulation method would provide a strong technical support to stress analysis of the impeller.
Some studies was developed on stress by CFD-FEM. But most of these studies are only on the impeller static stress distribution and static stress strength \cite{3-5}. The research on dynamic stress on pump was seldom. Relative to the pump, the dynamic stress study on turbine runner is more \cite{6-9}. And these researches for the runner dynamic stress would give an example to learn.

The paper reports on a structural simulation of flow-induced dynamics stress in an axial pump impeller. The loads on the impeller came from a simulation of the whole pump interior flow using CFD method \cite{10}. The loads, which varied in time domain and space domain, were set as force boundary based on coupled fluid-structure interaction \cite{10}. The sequential coupled fluid-structure interaction assumes that the influence of the structural deformation on the flow field is negligible in the calculations. The dynamic stress of impeller is investigated under three flow rates.

2. Features of the axial pump impeller

In this paper, the dynamic stress of a axial pump impeller was predicted. The structure of this axial pump is shown in Figure 1. The design capacity $Q_d$ is 33.4$m^3$/s. The design head $H$ is 7.5m. The rotating speed is 125rpm under design flow condition. The diameter of impeller $D_2$ is 2950mm. The number of impeller blades is 6. The number of guide vanes is 11. This investigation was carried out under three flow conditions (0.8$Q_d$, 1.0$Q_d$, 1.1$Q_d$).

![Figure 1. Structure of an axial pump](image)

There are four components of impeller, namely blade, root flange, hub and pivot. The blade, root flange and pivot were select as the simplified model. The simplified geometrical model of impeller is shown in Figure 2.

![Figure 2. Simplified geometrical model of impeller blades](image)
3. Structural simulation method

The governing equation for dynamic elastic stress of structure is\cite{11}:

\[
M\ddot{u} + C\dot{u} + Ku = \{F_s\} + \{F_i\}
\]

(1)

\[
\sigma = DBu
\]

(2)

Where \(M\) is the mass matrix, \(C\) is the damping matrix, \(K\) is the stiffness matrix, \(u\) is the node displacement, \(\dot{u}\) is the node velocities, and \(\ddot{u}\) is the node accelerations. \(F_s\) is the force on the interface between the fluid and the solid of the impeller, and \(F_i\) is the inertia force caused by the impeller’s rotation and gravity. \(D\) is elastic matrix determined by material elastic modulus and Poisson ratio, \(B\) is strain matrix based on unit shape function, \(\sigma\) is node stress.

The value of \(C\) is usually given by the Rayleigh’s theory\cite{11}:

\[
C = c_kK + c_M\text{ }M
\]

(3)

Where \(c_k\) and \(c_M\) are two constants. The values of \(c_k\) and \(c_M\) are calculated from modal damping ratios.

Before structural calculation, the fluid-structure interaction should be considered. There are two strategies for the solution of the fluid-structure interaction. One is strongly coupling method. Another is weakly coupling method. In this paper the structural simulation was carried out based on weakly coupling method, which is also called the sequential coupled fluid-structure interaction\cite{10}. The effect of structural deformation to the flow field is ignorant. And the calculations of structure field were done by using the results of the whole pump interior flow simulation as the force boundary.

Equation (1) is solved by using Newmark time integration method. Newmark time integration method is an implicit algorithm. After solved Equation (1), the time histories of nodes displacement would be obtain from the results. And the velocity of nodes is obtained by using a central-difference scheme, which is expressed as\cite{11}:

\[
\ddot{u}_{n+1} = \frac{u_{n+2} - u_{n+1} -\frac{1}{2}u_{n+1} -\frac{1}{2}u_{n}}{t_{n+2} - t_n} - \frac{u_{n+1} - u_{n}}{t_{n+1} - t_n}
\]

(4)

Where \(u_n\) is displacement at the time instant \(t_n\), \(\dot{u}_{n+1}\) is velocity at the time instant \(t_{n+1}\).

4. FE model and boundary conditions

The structural domain was discreted by using tetrahedral elements. The mesh model consists of 46355 elements. The model is shown as Figure 3.

![Mesh model for structural simulation](image)

Figure 3. Mesh model for structural simulation

(1) Boundary conditions for flow simulation

The flow simulation is carried out using Fluent6.3. Turbulence was modelled with a RNG k-\(\varepsilon\) model. The inlet boundary condition is velocity inlet at pump inlet. The outlet condition is set as outflow at pump outlet. The solid wall is set as no-slid wall. The simulation time is 10 rotational revolutions, time step is 0.012s. And the result data of last 5 rotational revolutions were extracted for structural simulation.
Boundary conditions for structural simulation

The structural simulation time is 2.4s, 5 rotational revolution. The time step is 0.012s, which is consistent with the CFD time step. The constraint for the impeller is pivot. The loads on the impeller include the inertia force and the surface forces. The inertia force includes the impeller’s own weight and the rotational inertia force. The surface force is pressure pulsation which caused by unsteady flow. The pressure pulsation was obtained from the whole flow passage simulation by CFD \cite{10}. The Newmark time integration method is used to solve Equation (1).

5. Results and discussion

The structural dynamic analysis of impeller was performed. The stress of impeller under unsteady flow would be obtained. The typical stress distributions under different flow conditions at $t=2.016$ are shown in Figure 4.a- Figure 4.c. Assuming the time of a rotational revolution is $T$; the time $t=2.016$ is $1/5T$ in latest revolution.

Figure 4.a. Stress distribution in impeller at $0.8Qd$ ($t=2.016$s)

Figure 4.b. Stress distribution in impeller at $1.0Qd$ ($t=2.016$s)

Figure 4.c. Stress distribution in impeller at $1.1Qd$ ($t=2.016$s)
As shown in Fig. 6, the characteristic of the stress on impeller would be got. At the $1.0Q_d$ and $0.8Q_d$ flow conditions, the zone of maximum stress appears at joint between blade and root flange near trailing edge. At the $1.1Q_d$, maximum stress is localized at joint between blade and root flange near leading edge. The zones of maximum stress are consistent with the crack position of impeller using Dongshen water supply project. As a result, the fatigue of axial pump impeller should take the zones of maximum stress into account as hot points. In a revolution, the stress distribution features at other times are similar to features at $t=2.016s$ under three flow conditions. The main difference is the value of stress.

Dynamic stress is significant to fatigue damage. For the axial pump impeller in this study, there are two zones where the crack would occur. So the time and frequency domain analysis of the hot zones should be investigated. Two locations of the stress maximum positions were adopted. The result data of these two locations were recorded. RA was located near trailing edge at joint between blade and root flange. RB was located near leading. Two locations were shown in Figure 5.

![Figure 5. Locations of recording points](image)

The dynamic stress of impeller is normalized to stress coefficient $C_s$, which is defined \[^{[10]}\] as:

$$C_s = \frac{\Delta \sigma_t}{0.5 \rho \omega^2}$$

(5)

Where, $\Delta \sigma_t$ is difference between transient value and average value, $\rho$ is the density of water, $\omega$ is the circumferential velocity of impeller outlet.

Time histories of stress coefficient $C_s$ at RB under different flow condition are shown in Figure 6.a-Figure 6.c. Using technology of FFT, the frequency spectra of stress coefficient $C_s$ at RB is obtained, shown in Figure 7.

![Figure 6.a Time history of stress coefficient Cs at RB under 0.8Q_d](image)
Figure 6.b Time history of stress coefficient $C_s$ at RB under $1.0Q_d$

Figure 6.c Time history of stress coefficient $C_s$ at RB under $1.1Q_d$

Figure 7 Frequency spectra of stress coefficient $C_s$ at RB

The time histories and frequency spectra of stress coefficient $C_s$ at RA are shown in Figure 8.a-Figure 8.c and Figure 9.

Figure 8.a Time history of stress coefficient $C_s$ at RA under $0.8Q_d$
As shown in Figure 6 and Figure 9, the time history of stress coefficient at RA and RB is random. That is to say, the stress in impeller varied randomly with time. Under a same flow condition, the peak to peak value (p-p value) of stress coefficient at RB is lager than p-p value at RA.

In frequency domain, the first dominant frequency at RA and RB is 22.9Hz. The number of vane is 11, and the rotating speed is 125rpm. Then the vane passing frequency is 22.9Hz. So The first dominant frequency at RA and RB is equal to the vane passing frequency due to rotor-stator interaction between the vane and the blade. The frequency 37.5Hz is clearly found, which is equal to 3 times the blade passing frequency. Under 0.8$Q_d$ flow condition, some low level frequencies of stress were found. The amplitudes of these low level frequencies became smaller with the flow rate increasing. The possible cause is that there are back flow, secondary flow, separate flow, et al.

6. Conclusion
The flow-induced dynamics stress for an axial impeller was predicted using FEM. The stress of two zones on impeller (trailing edge and leading edge of blade) was investigated under three flow conditions. The results indicate that:

1. The distribution of stress in impeller is uneven. But the features of dynamic stress distribution are similar to each other under different flow conditions. At the $1.0Q_d$ and $0.8Q_d$ flow conditions, the zone of maximum stress appears at joint between blade and root flange near trailing edge. At the $1.1Q_d$, maximum stress is localized at joint between blade and root flange near leading edge. In addition, the dynamics stress of impeller changed periodically with the impeller rotating. As a result, the fatigue may be happened. The probability of fatigue should be noticed in using.

2. The frequencies of stress at zones of maximum stress are 22.9Hz and 37.5Hz as the fundamental frequency and its harmonics. The fundamental frequencies are nearly equal to vane passing frequency (22.9 Hz) and 3 times blade passing frequency (37.5Hz).

3. The first dominant frequency at RA and RB is 22.9Hz. The number of vane is 11, and the rotating speed is 125rpm. Then the vane passing frequency is 22.9Hz. So the first dominant frequency at RA and RB is equal to the vane passing frequency due to rotor-stator interaction between the vane and the blade.

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