Influence of clearance model on numerical simulation of centrifugal pump

Z Wang1, B Gao, L Yang and W Q Du
Jiangsu University, Zhenjiang, Jiangsu, China
E-mail: initial_wz@sina.com

Abstract. Computing models are always simplified to save the computing resources and time. Particularly, the clearance that between impeller and pump casing is always ignored. But the completer model is, the more precise result of numerical simulation is in theory. This paper study the influence of clearance model on numerical simulation of centrifugal pump. We present such influence via comparing performance, flow characteristic and pressure pulsation of two cases that the one of two cases is the model pump with clearance and the other is not. And the results show that the head decreases and power increases so that efficiency decreases after computing with front and back cavities. Then no-leakage model would improve absolute velocity magnitude in order to reach the rated flow rate. Finally, more disturbance induced by front cavity flow and wear-ring flow would change the pressure pulsation of impeller and volute. The performance of clearance flow is important for the whole pump in performance, flow characteristic, pressure pulsation and other respects.

1. Introduction
Clearance that between pump block and impeller comprises two parts, one is front cavity and the other is back cavity. Mostly, it is convenient for mesh generation and numerical simulation to simplify the 3-D model. And front cavity enable a little of fluid with high energy pass through wear-ring to inlet of impeller, which would generate a disturbance for inlet flow situation. According to some literatures [1,2], The flow conditions in these small axial gaps are of significant importance for a number of effects such as disk friction, leakage losses or hydraulic axial thrust to name but a few. The front cavity between the impeller and the casing is set to achieve an acceptable pump efficiency.

The whole flow field model showed a higher accuracy than the simple model, and the flow pattern of two models were also different [3]. If this clearance is allowed to grow, either through corrosion or erosion, the pump will lose efficiency [4]. The size and shape of the wear-ring clearance directly affects the impeller sealing effect, and have a major impact on the overall performance of the pump [5]. According to [6], the effect of the clearance size change is mainly concentrated in the front and rear chamber, also in the exit of the clearances, while the impact is not significant in other parts of the pump.

1 E-mail: initial_wz@sina.com
Through theoretical analysis and test, the fluid force of the annular clearance seal of centrifugal pump is studied, which is important for vibration of centrifugal rotor [7]. The head and total efficiency of the centrifugal pump increase as the clearance value of wear-rings narrows, which is induced by three reasons. Such as the energy dissipation decrease within the impeller, the impact of secondary flow at the inlet of impeller on the mainstream weakens slowly, the eccentric whirl of centrifugal pump is dampened; the front shroud leakage diminishes [8].

The size of the clearance would have a greater influence on the distribution of flow pressure and velocity in pump chamber. As the value of the clearance increased, the pressure coefficient would decrease and the velocity increase [9]. Combined with formulas of hydraulic efficiency and volume efficiency, it can be seen that the pump efficiency decreases with the increasing clearances of wear-rings even though the mechanical losses decrease with the increasing of leakage rate [10]. The modification effects manifest in the front and rear chambers, also in the outlet of clearances, while insignificant in other sections of the pump. As the clearance widens, a low pressure area in the chamber tends to expand toward the volute and a high-pressure zone near the clearance outlet has a tendency to expand toward the impeller inlet [11]. There are many reasons for difference of forecasting losing, and especially the simplifying of prototype pump is considered as the dominant factor of large deviation in numerical simulation results [12]. The structure of front and back cavities is very complex on centrifugal pumps, especially the size of front and back wear-rings and balancing hole is tiny so that there are few literatures about numerical simulation of full flow field [13, 14].

2. Numerical simulation

2.1 Model pump
We choose a centrifugal pump with low specific speed, closed impeller and spiral volute to study the effect of clearance to internal and external characters of pump. The part parameters of model pump is shown as follows.

| parameters                  | value |
|-----------------------------|-------|
| Head (m)                    | 20    |
| Flow rate (m³/h)            | 55    |
| Speed (r/min)               | 1450  |
| Impeller type               | closed|
| Blade number Z              | 6     |
| Specific speed ns           | 69    |
| Tongue angle (°)            | 25    |
| Area ratio of import and export of pump | 1     |

Model pump is built by three dimension software, and there are two cases that case 1 match with the front cavity and back cavity but case 2 is without clearance. The components of model pump is shown in Figure.1 as follow.
2.2 The method of numerical simulation

The structural meshes of all components are generated by ICEM which is a part of ANSYS software. Based on the non-compressible Reynolds time averaged N-S equations and the RNG $k-\varepsilon$ turbulence model, the finite volume method is applied to the equation discretization. And second-order central difference scheme is applied to the diffuse term discretization, second order upwind discretization is used for the convection term. RNG model is a semi-empirical turbulence model, and through solution of two transport equations of the turbulence kinetic energy $k$ and turbulence dissipation $\varepsilon$ separately to decide the turbulent velocity and turbulence scale. Separated implicit method is used for the calculation, and SIMPLE algorithm is adopted for the pressure-velocity coupling.

Then LES (large eddy simulation) is used for the transient computing in order to gain pressure pulsation at different monitoring points. Rotor properties is changed to mesh motion, time step is calculated by speed. Four monitoring points are assigned to different place of model pump as follow.

| Monitoring points | Position          |
|-------------------|-------------------|
| P1                | Inlet of impeller |
| P2                | Outlet of impeller|
| P3                | Tongue of volute  |
| P4                | Outlet of volute  |

Table 2 Distribution of monitoring points.

The midline of span which is from shroud to hub (channel midline) is significant for pressure pulsation to analyse the influence of clearance to impeller, so P1 and P2 monitoring points locate at channel midline. It is one of the dominant factors for impeller operating that the state of inlet flow, which is the reason of placing the monitoring point P1. And the pressure pulsation of impeller outlet is an
important basis for the power capacity of impeller. In addition to presenting energy transmitting, the monitoring point P2 could present the rotor-stator interaction. Thus the monitoring point P3 is used for obtaining the influence of the front and back cavities to the tongue region. Finally, the monitoring point P4 could present the hydrodynamic characteristic of volute outlet.

3. Analysis of performance
For both case 1 and case 2, nine condition points are computed that range from 0.2 to 1.8 Q/Qv in order to present performance of model pump more completely. Thus the head curve, power curve and efficiency curve of two cases are gained, and they are shown in Figure.2. In this Figure, an obvious disparity could be found in all curves. Hump phenomenon appears in head curve with clearance in the small flow rate condition, but it doesn’t appear in head curve without clearance. Because front cavity enable a little of fluid with high energy pass through the wear-ring to inlet of impeller and merge with low energy fluid so that fluid came from outlet of impeller loss a part of energy. The head of both case 1 and case 2 is basically identical in design condition, but case 1’s head is obviously lower than case 2’s in large flow rate condition. The energy losing phenomenon appears in large flow rate condition too.

![Figure 2. Performance of model pump with or without clearance.](image)

The power of case 1 is lower than case 2’s in every condition. The reason similarly is extra loading generated by clearance flow in addition to energy losing at impeller inlet region. Impeller needs more energy to ensure that the fluid of outlet carry enough energy due to the energy losing. And extra loading would consume power so that shaft power of case 1 is higher. The result of these circumstances is the enormous efficiency difference of two cases in all condition. The efficiency of model pump with clearance is obviously lower than another. Thus it is an important factor for numerical simulation that whether or not there is clearance model.
4. Analysis of flow characteristic

The projection of absolute velocity on meridional plane could demonstrate the difference of flow characteristic and it could present the velocity evolution of fluid from inlet to outlet. As shown in Figure.3, each group of velocity vector is in same flow section, and each group include ten velocity vectors that express the projection of absolute velocity on meridional plane. And we choose three conditions to demonstrate the changing of velocity in different conditions, so the influence of front and back cavity to flow characteristic is presented. Case 1, which is the model pump with clearance, includes Figure. (a), (b), (c) expressing the flow character in 0.6, 1.0 and 1.4 Q/Qv conditions. And case 2 is similar to case 1.

It can be seen in Figure. (a), (b), (c), velocity magnitude increases with the increasing of flow rate in every vector group on meridional plane. The chaos of velocity direction in outlet of impeller in small flow rate condition changes to gather until in large flow rate condition. The first three group, which from inlet to outlet, have similar laws that the velocity magnitude of shroud is bigger than that of central or hub. But beginning from the fourth group, velocity magnitude changes slightly from shroud to hub. The axial velocity is dominant for first three group, but the radial velocity is major component for any other group. It is induced by fluid coming from front wear-ring with high energy that the velocity vector nears to the shroud in group 0-0 is not horizontal. And as the flow rate increases, this velocity vector get horizontal since the increasing of kinetic energy in inlet of impeller.

Case 1:     (a) 0.6 Q/Qv          (b) 1.0 Q/Qv         (c) 1.4 Q/Qv
Case 2: (d) 0.6 Q/Qv  (e) 1.0 Q/Qv  (f) 1.4 Q/Qv

Figure 3. Projection of absolute velocity on meridional plane.

A same law appears in case 2 and the velocity direction in outlet of impeller in small flow rate condition is more disorder. It is obvious that the velocity magnitude in all group, especially in 0.6 Q/Qv condition of case 2 is smaller than that of case 1. Because leakage of wear-ring would reduce flow quantity and it is necessary for fluid in impeller to carry more kinetic energy and guarantee the flow quantity. It could be deduced from velocity triangle that the decrease of the meridional velocity decreases the head.

5. Analysis of pressure pulsation

Pressure pulsation that induced by rotor-stator interaction and other hydrodynamic characteristics is one of the dominant factors for vibration in centrifugal pumps. 4 monitoring points is used for obtaining the pressure pulsation. As shown in Figure 4, there is pressure pulsation at different monitoring points. And it can be seen in Figure. (a), the pressure pulsation amplitude at blade passing frequency (BPF) of case 1 is obviously higher than that of case 2. And at low frequency region, more signals are obtained by numerical simulation in case 1, which demonstrates the pressure pulsation at low frequency region is more complex after add the clearance model. The similar law appears in Figure. (b). As the picture shows, the pressure pulsation amplitude at BPF and its harmonic of case 1 is higher than that of case 2. And the signals of low frequency region is more complicated after add the clearance model. The complex flow region which comprises impeller outlet, volute inlet and front cavity inlet influence the flow characteristic of impeller outlet. So the pressure pulsation of impeller outlet is not affected only by rotor-stator interaction but also by the flow characteristic of front cavity.

Figure 4. Pressure pulsation at different monitoring points.
The tongue region is an important area in which the fluid in volute and impeller flow out to diffuser. And impeller-volute interaction generate in tongue so that the pressure pulsation in this place is very complicated and messy. As the Figure.(c) shows, although the amplitude on the BPF of case 2 is higher than that of case 2, the peaks of case 2 is far more than that of case 1 in low frequency region. And in high frequency region which is bigger than 1500 Hz the amplitude of case 1 is bigger than that of case 2. Because of more messy flow in tongue area that computed by full flow field with clearance, the effect of impeller to tongue is weaker.

The flow situation of pump outlet is expected to be uniform, so the pressure pulsation of this region is expected to be small and smooth. The Figure. (d) shows the difference of pressure pulsation of pump outlet between case 1 and case 2. The amplitude on BPF of case 1 is 3 times as big as that of case 2, and the amplitude on high frequency region of case 1 is higher than that of case 2. Therefore the front cavity could influence the flow characteristic of pump outlet. And the energy recovery of case 1 is lower than that of case1, which is one of the reasons for lower head and efficiency.

6. Results
Relevant numerical simulation were performed in comparing performance, flow characteristic on meridional surface and pressure pulsation at different monitoring points. So the conclusions as follow:
(1) The efficiency curve of the model pump with front and back cavities is lower than that of model pump without front and back cavities. And the power curve of the model pump with front and back cavities is higher than that of model pump without front and back cavities. The head gap of two cases is not obvious nearby the design condition, but the model without front and back cavities does not present the hump phenomenon in small flow rate condition.

(2) The projection of absolute velocity on meridional surface in impeller outlet is chaotic in small flow rate condition whether the model pump have or not front and back cavities. With the increasing of flow rate, the velocity become more agglomerate. And the velocity magnitude of the model pump with front and back cavities is lower than that of the model pump without front and back cavities.

(3) The pressure pulsation amplitude of the model pump with front and back cavities is higher than that of the model pump without front and back cavities at impeller inlet, blade outlet and volute outlet. But the law of pressure pulsation amplitude at tongue is opposite to the others.

The clearance flow characteristics would influence the many aspects of centrifugal pump, such as the performance, absolute velocity on several surfaces, pressure pulsation and other internal and external characteristics. It is necessary for numerical simulation of full flow field to compute the front and back cavities, wear-rings and other clearances, which could improve the accuracy. This paper provides a reference for no-clearance numerical simulation to forecast the results via comparing the difference of two cases.

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