KSME

Springer

Journal of Mechanical Science and Technology 34 (8) 2020

Original Article
DOI 10.1007/s12206-020-0725-z

Keywords:
· Anti-cavitation performance
· Axial force
· Balance hole
· High-speed centrifugal pump

Correspondence to:
Xiaorui Cheng
cxr168861@sina.com

Citation:
Cheng, X., Chang, Z., Jiang, Y. (2020). Study on the influence of the specific area of balance hole on cavitation performance of high- speed centrifugal pump. Journal of Mechanical Science and Technology 34 (8) (2020) 3325~3334. http://doi.org/10.1007/s12206-020-0725-z

Received January 14th, 2020
Revised May 13th, 2020
Accepted June 10th, 2020

† Recommended by Editor Yang Na

Study on the influence of the specific area of balance hole on cavitation performance of high-speed centrifugal pump

Xiaorui Cheng[1,2], Zhengbai Chang[1] and Yimeng Jiang[1]

[1]College of Energy and Power Engineering, Lanzhou University of Technology, Lanzhou, China, [2]Key Laboratory of Fluid machinery and Systems, Lanzhou, China

Abstract In this paper, the influence of the specific area of balancing hole on the cavitation performance of high-speed centrifugal pump is studied by numerical method. The results show that in the initial cavitation stage, with the increase of the specific area, the head and efficiency of the pump decreases, and the shaft power increases in a small range. The specific area of the balance hole can change the magnitude and direction of the rotor axial force. With the increase of the specific area, the anti-cavitation performance of the pump is weakened, especially when the specific area reaches a certain value, the vortex flow appears in the balance hole, which causes serious distortion of the flow condition at the inlet of the centrifugal impeller. Meanwhile, cavitation also occurs in the balance chamber and is mainly concentrated near the hub of the centrifugal impeller.

1. Introduction

The high-speed centrifugal pump is mainly used in the field of aerospace, using high speed to improve the linear speed of impeller outlet, to achieve the purpose of high head. In order to balance the rotor axial force, the balance hole is usually designed in the back cover plate of the centrifugal impeller. However, the balance hole has a certain impact on the performance of the centrifugal pump [1-3]. On the other hand, the balance hole introduces high pressure fluid into the inlet of the centrifugal impeller, which changes the flow condition of the inlet and will have a certain impact on the anti-cavitation performance of the centrifugal impeller.

As for the flow characteristics in the balance hole, relevant scholars have done some researches. Cao et al. [4] studied the pressure pulsation characteristics of low specific speed centrifugal pump with radial return balance hole, and found that the pressure pulsation at the separating tongue of volute can be improved by using radial balance hole. Liu et al. [5, 6] pointed out that the liquid leakage coefficient of the balance hole decreased with the increase of the balance hole specific area. For the cavitation problem of centrifugal pump, relevant scholars have also done some researches. Dong et al. [7] studied the influence of the balance hole diameter on the performance of centrifugal pump and the pressure of balance chamber, and pointed out that when the specific area of balance hole $k \geq 2.645$, the cover force of balance chamber was basically balanced. Sha et al. [8] studied the effect of balance hole on the performance of high temperature and high pressure centrifugal pump, and pointed out that the balance hole has little effect on the performance of centrifugal pump, but it can effectively balance the axial force; compared with the centrifugal pump without balance hole, the centrifugal pump with balance hole can reduce the axial force by about 15 %. Li et al. [9] proposed a joint cavitation vibration analysis algorithm, which can effectively detect the generation and development of cavitation. Cui et al. [10] put forward a method to improve the cavitation characteristics of centrifugal pump by using jet device. Results shows that the anti-cavitation effect of centrifugal pump is the best when the jet flow is 6 %. Liang et al. [11] studied the change characteristics of cavitation bubble in the flow field during the cavitation process of centrifugal pump, and proposed that cavitation bubble can absorb the energy of vortex core to a certain extent and

Table 1. Main parameters of rotor components of high-speed centrifugal pump.

Model	Main parameters				
Inducer	Inlet hub diameter d_{h1}/mm	Outlet hub diameter d_{h2}/mm	Rim diameter D_y/mm	Screw pitch P/mm	Number of blades Z
	10	18	40	13.93	2
Centrifugal impeller	Flow Q/(m^3/h)	Head H/m	Inlet diameter D_1/mm	Outlet diameter D_2/mm	Outlet width b_2/mm
	16.3	1200	40	95	3

increase the volume of vortex core. Guo et al. [12, 13] studied the bubble evolution mechanism and anti-cavitation performance of high-speed centrifugal pump with split blade induction, and the effect of blade number on anti-cavitation performance of high-speed centrifugal pump. Zhang et al. [14] proposed a method to improve the cavitation characteristics of centrifugal pump with slot centrifugal impeller. Rakibuzzaman et al. [15] analyzed the cavitation behavior of the multi-stage centrifugal pump numerically and experimentally, they found that the simulation results can truly represent the development of the cavitation phenomenon attached to the impeller. Kim et al. [16] took the mixed flow pump as the research object, they analyzed the effect of blade thickness on hydraulic performance and cavitation, and found that the increase in blade thickness will increase the steam volume of NPSH. Jiang et al. [17] studied the effect of temperature change on anti-cavitation performance of a high-speed centrifugal pump with variable pitch inducer with annular nozzle.

The current researches focus on the influence of the balance hole on the flow field characteristics and axial force of the centrifugal pump, while there are few researches focusing on the influence of the balance hole on the cavitation characteristics of the centrifugal pump. In this study, a high-speed centrifugal pump with inducer was taken as the research object, the influence of balance hole on anti-cavitation performance of high-speed centrifugal pump and the influence of balance hole on the performance of high speed centrifugal pump in the process of cavitation are studied. It provides a theoretical reference for the design of balance hole of high-speed centrifugal pump.

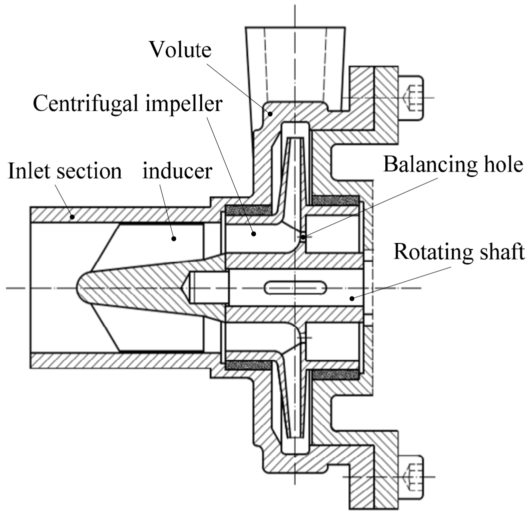


Fig. 1. Structural drawing of high-speed centrifugal pump.

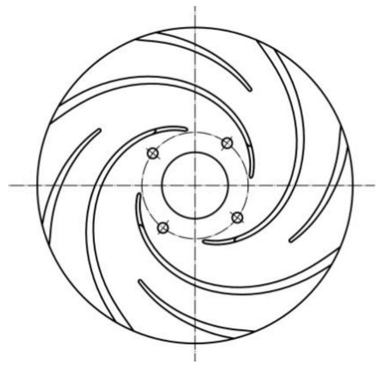


Fig. 2. Structural drawing of centrifugal impeller.

2. Research object and scheme

2.1 Structure of high speed centrifugal pump

The rotor part of high-speed centrifugal pump adopts the structure of equal pitch inducer and centrifugal impeller, and its pressure chamber adopts volute. Fig. 1 shows the structure of high-speed centrifugal pump. The structure of centrifugal impeller adopts cylindrical splitter blades. At the inlet of centrifugal impeller, the number of blades is 4, at the outlet, the number of blades is 8 (as shown in Fig. 2). The balancing mode of the rotor axial force adopts the structure which rear ring is equipped with a balance hole. Considering the number of blades at the inlet of the centrifugal impeller is 4, in order to ensure the excellent matching between the balance hole and the centrifugal impeller, the number of balance holes is 4 as well. Table 1 shows the main parameters of rotor components of high-speed centrifugal pump.

2.2 Research scheme

In this study, four schemes (as shown in Table 2) are designed to study the influence of the balance hole diameter on the internal and external characteristics and the rotor axial force of high-speed centrifugal pump. In order to reasonably describe the balance hole diameter, the ratio of the total area of the balance hole to the flow area of the back ring is used to define the characteristic size of the balance hole diameter, which is called the specific area k of the balance hole.

Table 2. Parameters of balance holes in different schemes.

Scheme	k	Φ/mm	D_m/mm	δ/mm	D/mm
Scheme 1	0.5	2.2	62	0.15	32
Scheme 2	1	3.1	62	0.15	32
Scheme 3	3	5.3	62	0.15	32
Scheme 4	5	6.8	62	0.15	32

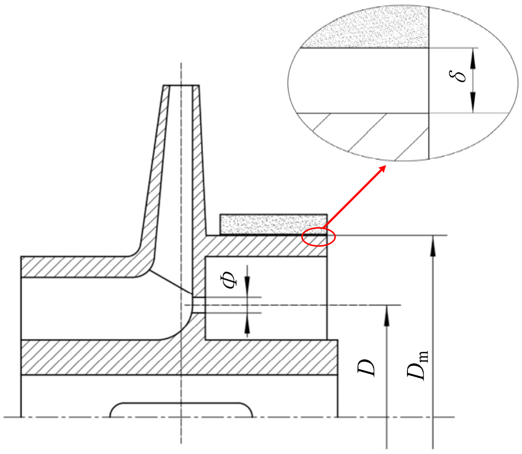


Fig. 3. Schematic diagram of the scheme.

$$k = \frac{N\frac{1}{4}\pi\Phi^2}{\frac{1}{4}\pi((D_m+2\delta)^2 - D_m^2)} = \frac{N\Phi^2}{4\delta(D_m+\delta)} \tag{1}$$

where N is the number of balance holes, Φ is the diameter of balance holes, D_m is the diameter of back ring, and δ is the gap size of back ring. Because of $\delta << D_m$, the specific area k of the balance hole can be expressed as:

$$k = \frac{N\Phi^2}{4\delta D_m}. \tag{2}$$

Under the condition that the position and area of the back seal ring and the diameter D of the balance hole are constant, the specific area of the balance hole of each scheme is determined by changing the diameter of the balance hole. Table 2 shows the corresponding relationship of geometric parameters of balance hole under different schemes, and Fig. 3 is the schematic diagram of scheme.

3. Numerical calculation

3.1 Geometric model of fluid domain

Using the Pro/E three-dimensional modeling software to establish the three-dimensional model of each flow-passing part of the high-speed centrifugal pump. Fig. 4 shows the three-dimensional calculation model of the whole flow field of the high-speed centrifugal pump. Its main flow-passing components include: inlet section, inducer, centrifugal impeller, volute, front chamber, back chamber, balancing chamber and outlet section. In order to ensure the stability of the flow field structure in the numerical calculation, the inlet and outlet sections are extended.

Table 3. Number of mesh elements in fluid domain.

Fluid domain	Number of mesh elements (×10^4)
Inlet section	201.0
Inducer	153.7
Centrifugal impeller	158.6
Volute and outlet section	131.4
The whole fluid domain	1039.1

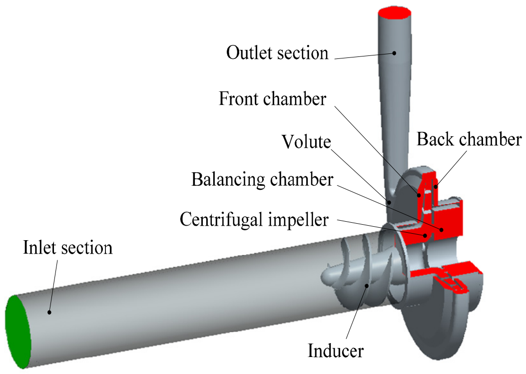


Fig. 4. The fluid domain three-dimensional model diagram.

3.2 Meshing of fluid domain

Gambit is used for the meshes generation for whole flow field calculation model of high-speed centrifugal pump. Considering the complex spatial structure of the calculation domain geometric model, the tetrahedral unstructured mesh with good adaptability is adopted. For the inlet edge of the blade, the tongue of volute and other large curvature fluid domain, the grid is densified. Checking the grid independence for the calculation model of scheme 2, the results are shown in Fig. 6. The final grid number is shown in Table 3. In other schemes, the mesh number of flow-passing components is kept as shown in Table 3. Fig. 5 shows the mesh structure of main flow-passing components of high-speed centrifugal pump.

3.3 Governing equations

Cavitation flow can be regarded as an ideal uniform gas-liquid mixed flow, so in this study, a two-phase flow model [18] is built by using the mixed model. The control equations of this uniform and balanced flow can be expressed as follows:

$$\frac{\partial\rho}{\partial t} + \frac{\partial(\rho u_j)}{\partial x_j} = 0 \tag{3}$$

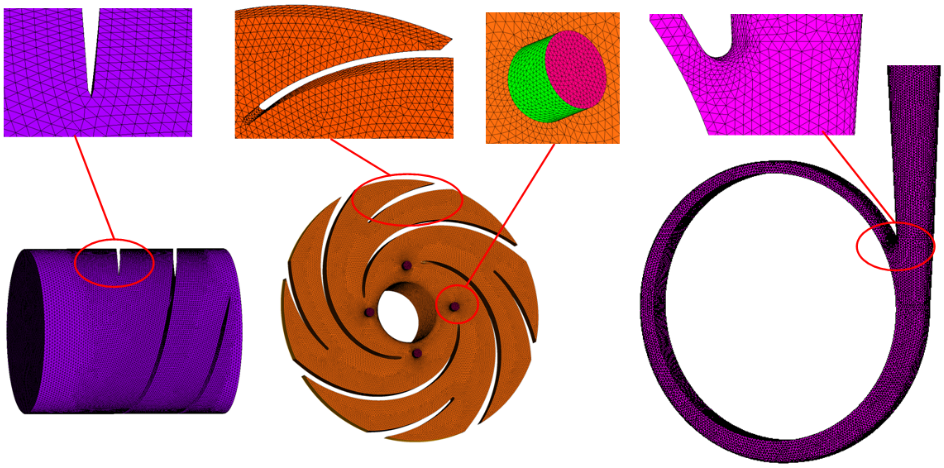


Fig. 5. Fluid domain meshes of main flow-passing parts.

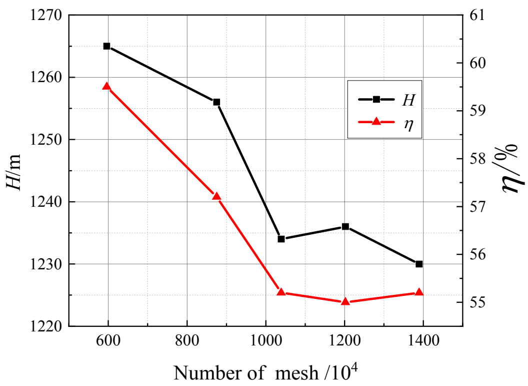


Fig. 6. Meshes independence check.

$$\frac{\partial(\rho u_j)}{\partial t}+\frac{\partial(\rho u_i u_j)}{\partial x_j}=\rho f_i-\frac{\partial \rho}{\partial x_i}+\frac{\partial}{\partial x_i}\left[(\mu+\mu_t)\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial u_i}-\frac{2}{3}\frac{\partial u_k}{\partial x_k}\delta_{ij}\right)\right] \quad (4)$$

$$\rho=\rho_v\alpha_v+\rho_l(1-\alpha_l) \quad (5)$$

where ρ is the density, u_i and u_j are the velocity components, μ and μ_t are the dynamic viscosity and turbulent viscosity of the mixed medium respectively, δ_{ij} is the Kronecker number, and α is the volume fraction, where the subindices v and l represent gas and liquid, respectively.

3.4 Turbulence model

Considering the vortex characteristics of turbulent flow in centrifugal pump, RNG k-ε turbulence model is adopted to better deal with high strain rate and strong streamline bending flow and improve the prediction accuracy of fluid domain calculation [19]:

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho k u_i)}{\partial x_i}=\frac{\partial}{\partial x_i}\left[\alpha_k\mu_{\text{eff}}\frac{\partial k}{\partial x_j}\right]+G_k+\rho\varepsilon \quad (6)$$

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial(\rho\varepsilon u_i)}{\partial x_i}=\frac{\partial}{\partial x_i}\left[\alpha_\varepsilon\mu_{\text{eff}}\frac{\partial \varepsilon}{\partial x_j}\right]+\frac{C_{1\varepsilon}}{k}G_k-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \quad (7)$$

$$\mu_{\text{eff}}=\mu+\mu_t \quad (8)$$

$$\mu_t=\rho C_\mu\frac{k^2}{\varepsilon} \quad (9)$$

where C_μ, α_k and α_ε are empirical coefficients, ε is turbulence dissipation rate, G_k is turbulent kinetic energy generation term, $C_{1\varepsilon}$ and $C_{2\varepsilon}$ are empirical constants, whose values are 1.42 and 1.68, respectively [20].

3.5 Cavitation model

Cavitation flow is regarded as a homogeneous equilibrium flow, that is, regardless of the velocity slip between the liquid and the vapor, the fluid medium is regarded as a homogeneous fluid composed of gas-liquid mixture with variable density [18]. In order to describe the process of cavitation development and collapse more accurately, the cavitation model use the idealized basic equations of cavitation dynamics without empirical coefficients proposed by Rayleigh- Plesset [21]. Namely,

$$R\frac{d^2R}{dt^2}+\frac{3}{2}\left(\frac{dR}{dt}\right)^2=\frac{1}{\rho}\left[P_R-P_\infty(t)-\frac{4\mu}{R}\frac{dR}{dt}\right] \quad (10)$$

where $P_R=P_B-\dfrac{2\sigma *}{R}$.

R : Cavitation radius
ρ : Liquid density
μ : Liquid viscosity of movement
P_B : Cavitation internal pressure
P_∞ : The static pressure of the liquid
σ* : Surface tension coefficient.

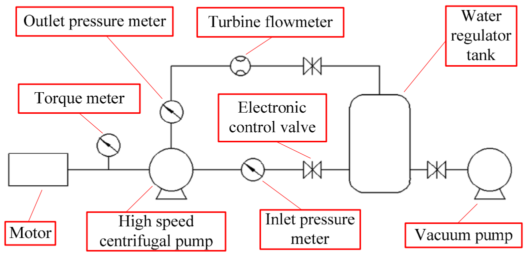


Fig. 7. Test system of high-speed centrifugal pump.

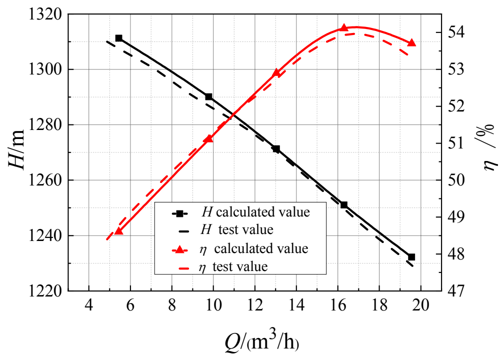


Fig. 8. Comparisons between calculated and tested external characteristic of high-speed centrifugal pump.

3.6 Boundary conditions

In this study, the whole flow field of high-speed centrifugal pump is numerically calculated. The inlet of the pump is set as the pressure inlet and the outlet as the mass flow outlet. The solid wall is selected as the wall condition without sliding. The fluid medium is 25 °C (constant temperature) aviation kerosene. Its saturated vapor pressure is 1329 Pa, and the system reference pressure is set to 0. The results of non-cavitation steady calculation was set as the initial value of cavitation calculation. The total pressure at the pump inlet was gradually reduced to make cavitation happen in the pump. By observing that the outlet pressure tends to be stable or the convergence residual value is less than 10^{-5}, it is determined that the solution has converged.

3.7 Validation of numerical calculation method

In order to verify the reliability of numerical calculation results, a high-speed centrifugal pump with balancing hole, whose specific area is k = 1, was tested. Fig. 7 shows the test system of high speed centrifugal pump. The external characteristic test is carried out on an open test bench. Fig. 8 is the comparison curves between the numerical calculation results and the test results of the external characteristics of the high-speed centrifugal pump. It is shown that the test values of head and efficiency are basically in agreement with the calculated values. The numerical calculation method is reasonable in the prescribed error range.

4. Results and discussion

4.1 Effect of specific area of balance hole on external characteristics of high-speed centrifugal pump during cavitation

Fig. 9 shows the change curve of the head of high-speed centrifugal pump with cavitation number under different specific area of balance hole. The calculation formula of cavitation number is defined as the following equation:

$$\sigma = \frac{p_{in} - p_v}{0.5\rho U_1^2} \tag{11}$$

where p_{in} is inlet pressure of the pump, p_v is vaporization pressure, ρ is fluid density, and U_1 is circumferential velocity at the intersection of inlet edge of blade of centrifugal impeller and front cover plate.

When the head H (energy index) of high-speed centrifugal pump drops by 3 %, the cavitation number $\sigma_{3\,\%}$ is used to measure the anti-cavitation performance of high-speed centrifugal pump, which is called critical cavitation number of pump [10, 11, 14]. It can be seen from Fig. 6 that when the cavitation number $\sigma > \sigma_{3\,\%}$, the pump head keeps basically unchanged with the increase of cavitation number; but when the cavitation number $\sigma < \sigma_{3\,\%}$, the pump head reduces sharply with the decrease of cavitation number. This is mainly because that with the constant decrease of cavitation number, the inlet pressure of high-speed centrifugal pump continues to decline. When the inlet pressure is less than the liquid saturated vapor pressure, the number of bubbles in the flow passage keeps increasing, and the bubbles break after encountering the high-pressure area in the flow passage, and the increasing number of bubbles constantly precipitates and collapses, which destroys the velocity distribution in the flow passage and consumes a lot of energy, resulting in a sharp decrease of the head.

At the same time, the critical cavitation number $\sigma_{3\,\%}$ rises with the increase of the specific area of the balance hole. Thus, the anti-cavitation performance of the high-speed centrifugal pump is weakened. The reason is that with the increase of specific area, the leakage q of balancing hole increases, the main flow at the inlet of the centrifugal impeller is strengthened by the impact of the leakage flow of the balance hole (see Fig. 10), which induces the vortex at the inlet of the centrifugal impeller blade and forms the local low pressure area. Hence, the cavitation in the pump increases.

When there is slight cavitation in the flow passage of high-speed centrifugal pump, the head hardly drop with the decrease of cavitation number, and the corresponding cavitation number is called σ_0. As shown in Fig. 9, when $\sigma = \sigma_0$, with the increase of the specific area of the balance hole, the head of the high-speed centrifugal pump decreases by 5.15 % of its maximum. This is mainly because that a part of the high-pressure fluid at the outlet of the centrifugal impeller flows into the balance chamber through the back ring, and then leaks to the inlet of the centrifugal impeller through the balance hole,

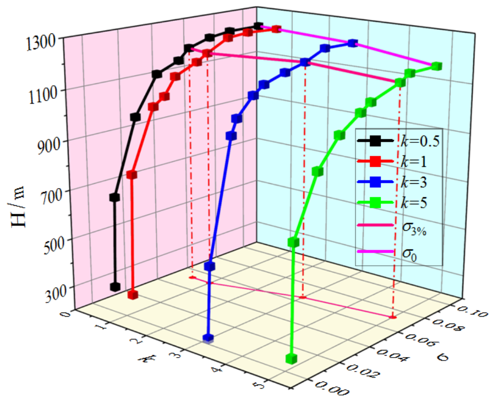


Fig. 9. Curve of head with cavitation number and specific area of balance hole.

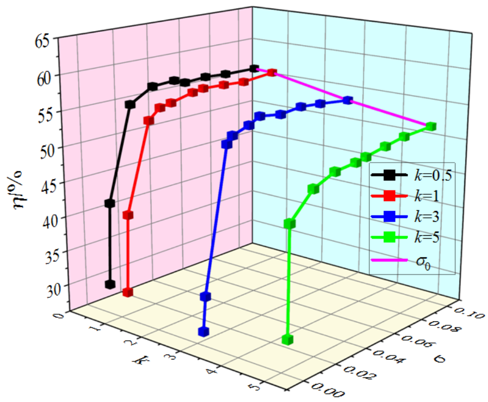


Fig. 11. Curve of efficiency with cavitation number and specific area of balance hole.

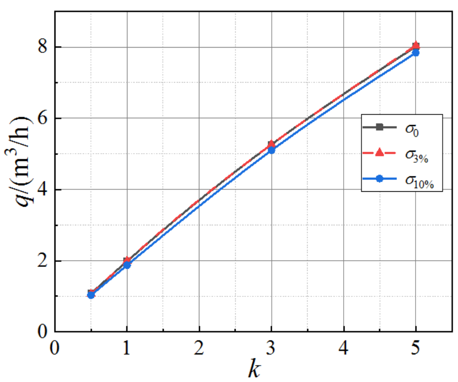


Fig. 10. Curve of the leakage of the balancing hole with the specific area under different cavitation numbers.

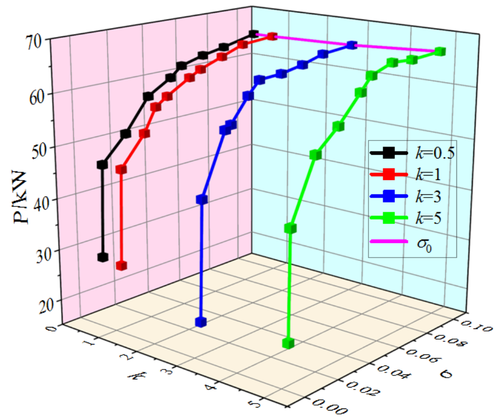


Fig. 12. Curve of shaft power with cavitation number and specific area of balance hole.

forming a secondary return flow, which continuously circulates in the pump cavity, consuming part of the energy. With the increase of the specific area of the balance hole, the cross-section area of the flow through the balance hole increases, thus the leakage increases, and the energy consumed by the circulation of the leakage flow in the pump cavity increases, thus the head decreases.

As shown in Fig. 11, when the cavitation number $\sigma \geq 0.04$, slight cavitation occurs in the pump, and the efficiency of the pump decreases slightly with the decrease of the cavitation number; when the cavitation number $\sigma < 0.04$, the efficiency declines sharply with the decrease of the cavitation number. This is because when slight cavitation occurs, fewer bubbles are generated. In the process of bubble formation and collapse, the fluid energy consumption is less, and the hydraulic loss is smaller; however, with the further reduction of cavitation number, serious cavitation occurs in the flow passage of high-speed centrifugal pump, resulting in a large number of bubbles. In the process of continuous generation and collapse of the bubbles, a large amount of fluid energy is consumed, and the hydraulic loss is also increased greatly. At the same time, the generated bubbles block the flow passage, resulting in the decrease of pump outlet flow and the sudden drop of efficiency.

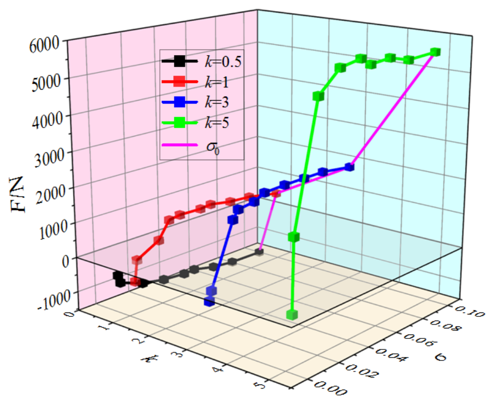
When the cavitation number $\sigma = \sigma_0$, the efficiency decreases by 4.13 % with the increase of the specific area of the balance hole. This is mainly because with the increase of the specific area of the balance hole, the large vortices in the balance hole change the inflow state of the blade inlet of the centrifugal impeller. So, the hydraulic loss increases and the efficiency decreases.

Fig. 12 shows the variation curves of shaft power under different specific area of balance hole. It can be seen that with the decrease of cavitation number, the shaft power decreases gradually. This is because when cavitation becomes more and more serious, the passage is blocked, the flow rate in the passage of centrifugal impeller decreases, the load of blade under the action of fluid declines, and the shaft power decreases.

When the cavitation number $\sigma = \sigma_0$, the shaft power increases to 2.42 % of the minimum value with the increase of the specific area of the balance hole. This is mainly because with the increase of the specific area of the balance hole, a large vortex is generated in the balance hole, which induces the fluid and destroys the inflow conditions at the inlet of the centrifugal impeller, makes the flow in the channel more complex, increases the load of the centrifugal impeller blade under the action of the fluid, and the shaft power is increased [5].

4.2 Effect of specific area of balance hole on axial force of rotor of high speed centrifugal pump

Fig. 13 shows the curves of the rotor axial force in different cavitation number with different specific area of balance hole. It can be seen that when the specific area of the balance hole k = 0.5, the axial force of the rotor points to the inlet of the centrifugal impeller, and when the specific area of the balance hole $k \geq 1$, the axial force of the rotor deviates from the inlet of the centrifugal impeller, and with the decrease of the cavitation number, the rotor axial force decreases gradually. This is because the area of the rear cover plate of the centrifugal impeller is larger than the area of the front cover plate, but the pressure difference between the front and rear cavities is very small. If there is no pressure relief function of the balance hole, the pressure exerted by the fluid in the rear cavity and the balance chamber on the rear cover plate is larger than the pressure exerted by the fluid in the front cavity on the front cover plate, so the axial force of the rotor points to the inlet of the centrifugal impeller. When the specific area of the balance hole k = 0.5, the pressure relief capacity of the balance hole is insufficient, the pressure of the fluid in the balance chamber is not sufficiently reduced, the pressure of the fluid acting on the rear cover plate is still greater than the pressure acting on the front cover plate, and the rotor axial force still points to the inlet of the centrifugal impeller. When the specific area of the balance hole $k \geq 1$, the pressure reduction degree of the balance hole is large, the fluid in the balance chamber is fully relieved, the pressure of the fluid acting on the rear cover plate is less than the pressure acting on the front cover plate, and the rotor axial force deviates from the inlet of the centrifugal impeller. In addition, with the decrease of cavitation number, the cavitation at the inlet of the centrifugal impeller becomes more and more serious. The head drops sharply and the outlet pressure of centrifugal impeller drops, which leads to the pressure on the front and rear cover plates of the centrifugal impeller decreases, and the rotor axial force decreases.



Note: "-" only means that the axial force of the rotor points to the inlet of the centrifugal impeller, while the value represents its magnitude.

Fig. 13. Curve of axial force with cavitation number and specific area of the balance hole.

When the cavitation number $\sigma = \sigma_0$ and the specific area of the balance hole $k < 1$, the axial force of the rotor decreases with the increase of the specific area of the balance hole; when the specific area of the balance hole $k > 1$, the axial force of the rotor increases with the increase of the specific area of the balance hole; when the specific area of the balance hole k = 1, the axial force of the rotor is small. This is because, when the specific area of the balance hole $k < 1$, the leakage of the balance hole is relatively small, so the pressure in the balance chamber is high, which makes the pressure on the rear cover plate greater than that on the front cover plate. At this time, the rotor axial force points to the inlet of the centrifugal impeller. However, with the increase of the specific area of the balance hole, the pressure relief capacity of the balance hole increases, and both the balance chamber pressure and the pressure on the rear cover are reduced, the axial force direction of the rotor deviates from the inlet of the centrifugal impeller, and increases with the further increase of the specific area of the balance hole.

4.3 Effect of specific area of balance hole on flow field in high speed centrifugal pump

Fig. 14 shows the flow field distribution of the high-speed centrifugal pump under the condition of critical cavitation in the axial plane. It can be seen that the pressure distribution in the balance chamber is uneven and there is a large pressure gradient. The pressure increases with the raise of the radial distance from the rotating shaft, but the average pressure in the balance chamber decreases with the increase of the specific area of the balance hole. This is due to the increase of the

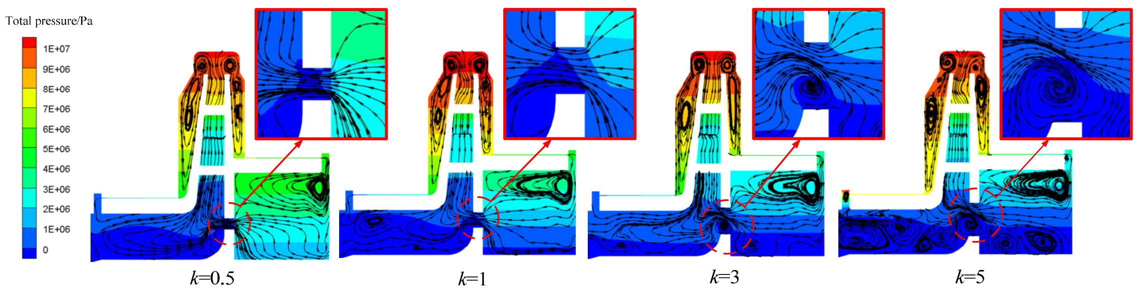


Fig. 14. Distribution of flow field of high-speed centrifugal pump under different specific area of balance hole in axial plane.

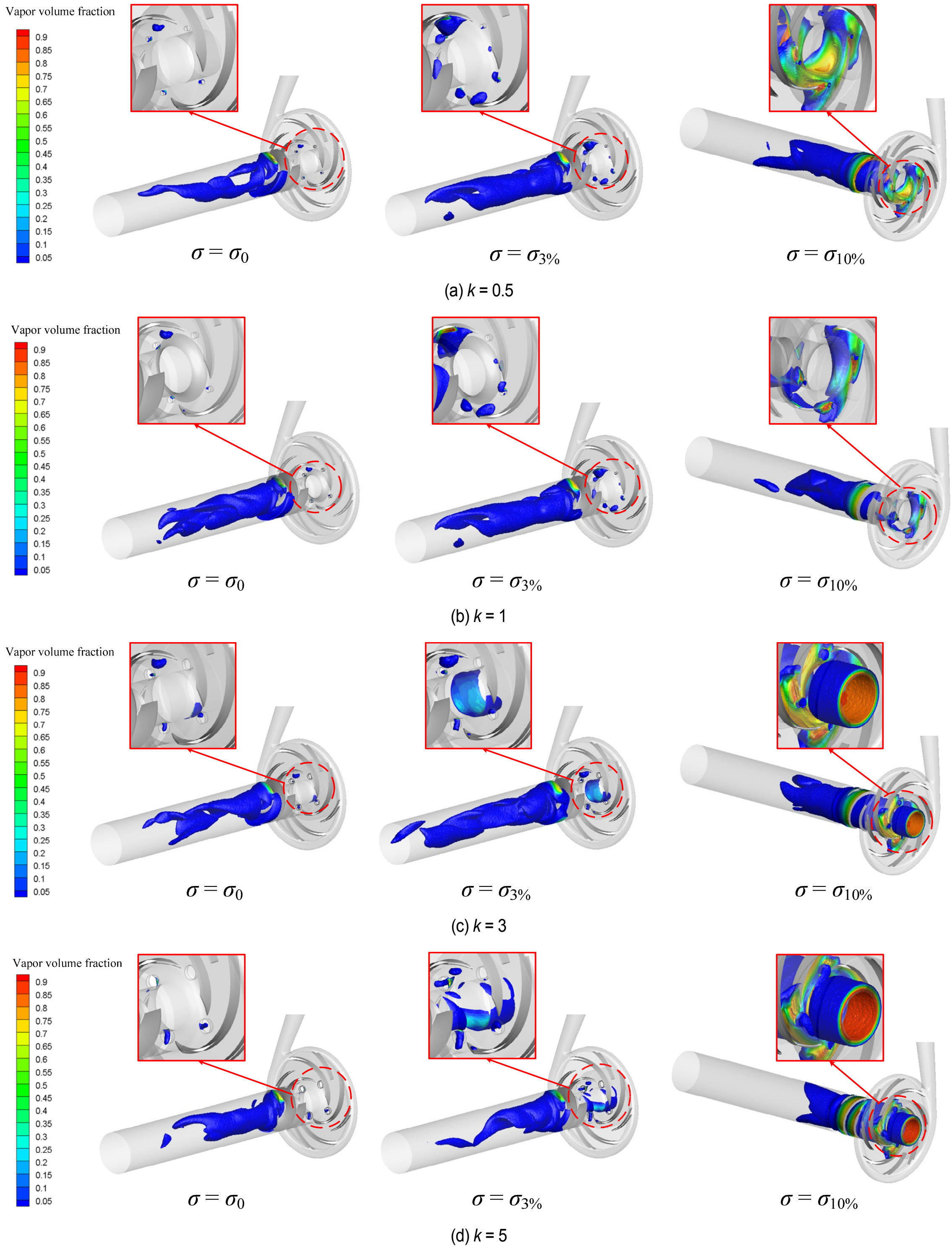


Fig. 15. Bubble volume distribution of high-speed centrifugal pump with different specific area of balance hole.

specific area of the balance hole, resulting in the increase of the pressure relief capacity of the balance hole.

As shown in Fig. 14, when the specific area of the balance hole is k = 0.5 and k = 1, the flow in the balance hole is smooth, the high-pressure jet leaking from the balance hole is mixed with the main stream at the inlet of the centrifugal impeller evenly, and the flow is relatively stable, and there is no obvious local low-pressure area at the inlet of the centrifugal impeller blade, so the anti-cavitation ability of the centrifugal pump is strong. When the specific area of the balance hole is k = 3 and k = 5, there is an obvious vortex in the balance hole. This vortex leads to more vortices at the inlet of the centrifugal impeller, and the flow is more disordered, resulting in a local low-pressure area at the inlet of the centrifugal impeller blade. It can be seen that the increase of the specific area of the balance hole will reduce the anti-cavitation performance of the pump.

4.4 Effect of specific area of balance hole on cavitation state of high speed centrifugal pump

Fig. 15 shows the distribution of bubble volume in the flow passage of high-speed centrifugal pump under different cavitation conditions with different specific area of balance hole.

When the cavitation number $\sigma = \sigma_0$, the distribution of bubbles in the pump passage remains unchanged with the increase of specific area of the balance hole, there is slight cavitation near the rim of the inducer inlet and the local position of the balance hole, but the pump head does not change at this time.

When the cavitation number $\sigma = \sigma_{3\%}$, there are obvious cavitation in many places of the pump, and the bubble distribution in the high-speed pump flow passage is quite different in different balance hole specific area. With the increase of the balance hole specific area, the inlet bubble of the centrifugal impeller blade increases, and the cavitation is getting serious. The head of high-speed centrifugal pump decreased to 3 %. The main cavitation areas respectively include the inlet edge of inducer near the rim, the inlet edge of impeller blade, the balance hole, the transition section between inducer and centrifugal impeller, and the position of balance chamber near the hub wall of centrifugal impeller. This is due to as the increase of the specific area of the balance hole, the pressure relief capacity of the balance hole and the flow velocity in the balance hole increase, and the pressure in the balance chamber decreases.

When the cavitation number $\sigma = \sigma_{10\%}$, the head of high-speed centrifugal pump decreases to 10 %. The distribution of bubbles in the flow passage of high-speed centrifugal pump is quite different under different specific area of balance hole. With the increase of specific area of balance hole, the inducer is filled with bubbles, and the bubbles in the flow passage at the inlet of centrifugal wheel are also increased. The high-speed centrifugal pump is seriously cavitating, at this time, the pump can not work normally. Especially when the specific area of the balance hole $k \geq 3$, as shown in Figs. 15(c) and (d), except for the inlet of inducer and centrifugal impeller blade, serious cavitation bubbles appear in the balance chamber near the hub wall of the centrifugal impeller. However, it should be noted that with the increase of radial distance, the volume fraction of bubble decreases. This is mainly due to the low flow velocity near the hub wall, and as the fluid flow in the balance chamber increases with the radius, the circumferential velocity of the fluid increases, while the fluid pressure increases.

5. Conclusion

In this study, the whole flow field of high-speed centrifugal pump is calculated. The key research contents respectively include the influence of the specific area of the balance hole on the anti-cavitation performance of the high-speed centrifugal pump, the variation law of the external characteristics, the internal flow field, the rotor axial force and the bubble distribution under different cavitation conditions. The following conclusions are drawn:

(1) In the initial cavitation stage of high-speed centrifugal pump, with the increase of the specific area of the balance hole, the head and efficiency of the pump are decreased, and the shaft power is increased, but the change range is small.

(2) With the increase of the specific area of the balance hole, the anti-cavitation performance of the high-speed centrifugal pump is weakened, especially when the specific area of the balance hole $k \geq 3$, the anti-cavitation performance of the high-speed centrifugal pump drops sharply.

(3) The specific area of the balance hole of high-speed centrifugal pump not only affects the value of the axial force of the pump rotor, but also affects the direction of the axial force.

(4) In the process of increasing the specific area of balance hole, cavitation is easy to occur at the inlet of inducer of high-speed centrifugal pump near the rim, balance hole and the inlet of centrifugal impeller blade near the back; at the same time, when the specific area of balance hole increases to a certain extent, cavitation will also occur in balance chamber.

Acknowledgments

This work is financially supported by the National Natural Science Foundation of China (Project No. 51469013).

Nomenclature

Q	: Flow rate (m^3/h)
k	: Specific area of the balance hole
H	: Head (m)
η	: Efficiency (%)
P	: Shaft power (kW)
F	: Rotor axial force (N)
p_{in}	: Pressure of pump inlet
p_v	: Pressure of fluid vaporization
σ	: Cavitation number
σ_0	: Cavitation number of pump with slight cavitation
$\sigma_{3\%}$	: Critical cavitation number of pump
$\sigma_{10\%}$	: Cavitation number of pump with serious cavitation
q	: Leakage of balancing hole

References

[1] K. Majidi, Numerical study of unsteady flow in a centrifugal pump, *Journal of Turbomachinery,* 127 (2) (2004) 805-814.

[2] J. Gonzalez, J. Femandez and C. Santolaria, Numerical simulation of the dynamic effects due to inpeeler-volute interaction in a centrifugal pum, *Journal of Fluids Engineering,* 124 (5) (2002) 348-355.

[3] S. H. Li, *Non-design Working Conditions and Optimal Design of Vane Pumps*, Machinery Industry Press (2006).

[4] W. D. Cao, X. D. Zhang and W. D. Shi, Pressure fluctuation performance of low specific-speed centrifugal pump with radial reflux balance hole, *Advances in Science and Technology of Water Resources*, 31 (5) (2011) 23-26.

[5] Z. L. Liu, X. C. Chen, D. W. Wang and Y. H. Hou, Experiment and analysis of balance hole liquid leakage in centrifugal pump, *Transactions of the Chinese Society of Agricultural Engineering,* 33 (7) (2017) 67-74.

[6] Z. L. Liu, D. W. Wang and S. Liang, Fluid leakage characteristic test on balance aperture of centrifugal pump impeller, *Transactions of the Chinese Society of Agricultural Engineering,* 43 (7) (2012) 85-88.

[7] W. Dong and W. L. Chu, Influence of balance hole diameter on performance and balance chamber pressure of centrifugal pump, *Transactions of the Chinese Society of Agricultural Engineering,* 46 (6) (2015) 73-77.

[8] Y. J. Sha, S. H. Liu, Y. L. Wu and B. G. Wang, Influence of balancing holes on performances of the hightemperature and high-pressure centrifugal pump, *Journal of Hydroelectric Engineering*, 31 (6) (2012) 259-264.

[9] Y. Li, G. W. Feng, X. J. Li, Q. R. Si and Z. C. Zhu, An experimental study on the cavitation vibration characteristics of a centrifugal pump at normal flow rate, *Journal of Mechanical Science and Technology*, 32 (10) (2018) 4711-4720.

[10] B. L. Cui, K. C. Zhu, Y. L. Zhang and P. F. Lin, Experimental and numerical study of the performance and cavitation flow of centrifugal pump with jetting device, *Journal of Mechanical Science and Technology*, 33 (10) (2019) 4843-4853.

[11] L. Dong, H. H. Shang, Y. Q. Zhao, H. L. Liu, C. Dai and Y. Wang, Study on unstable characteristics of centrifugal pump under different cavitation stages, *Journal of Thermal Science*, 28 (4) (2019) 608-620.

[12] X. M. Guo, Z. C. Zhu, B. L. Cui and G. P. Shi, Effects of the number of inducer blades on the anti-cavitation characteristics and external performance of a centrifugal pump, *Journal of Mechanical Science and Technology*, 30 (7) (2016) 3173-3181.

[13] X. M. Guo, L. H. Zhu, Z. C. Zhu, B. L. Cui and Y. Li, Numerical and experimental investigations on the cavitation characteristics of a high-speed centrifugal pump with a splitter-blade inducer, *Journal of Mechanical Science and Technology*, 29 (1) (2015) 259-267.

[14] R. H. Zhang, L. C. Yun and J. Li, Investigation on the effect of impeller slot jet on centrifugal pump performance, *Journal of Hydrodynamics*, 31 (4) (2019) 733-739.

[15] M. Rakibuzzaman, K. Kim and S. H. Suh, Numerical and experimental investigation of cavitation flows in a multistage centrifugal pump, *Journal of Mechanical Science and Technology*, 32 (3) (2018) 1071-1078.

[16] Y. I. Kim, S. Kim, H. M. Yang, K. Y. Lee and Y. S. Choi, Analysis of internal flow and cavitation characteristics for a mixed-flow pump with various blade thickness effects, *Journal of Mechanical Science and Technology*, 33 (7) (2019) 3333-3344.

[17] J. Jiang, Y. H. Li, C. Y. Pei, L. L. Li, Y. Fu, H. G. Cheng and Q. Q. Sun, Cavitation performance of high-speed centrifugal pump with annular jet and inducer at different temperatures and void fractions, *Journal of Hydrodynamics*, 31 (1) (2019) 93-101.

[18] Z. Y. Pan and S. Q. Yuan, *Foundation of Pump Cavitation*, Jiangsu University Press (2013).

[19] H. L. Liu, Y. Wang and D. X. Liu, Assessment of a turbulence model for numerical predictions of sheet-cavitating flows in centrifugal pumps, *Journal of Mechanical Science and Technology,* 27 (9) (2013) 2743-2750.

[20] F. J. Wang, *Computational Fluid Dynamics Analysis - Principle and Application of CFD Software*, Tsinghua University Press (2011).

[21] S. S. Pan and X. M. Peng, *Cavitation Mechanism*, National Defense Industry Press (2013).

Xiaorui Cheng is a Professor in the College of Energy and Power Engineering, Lanzhou University of Technology. He obtained a Ph.D. His research interests include analysis and optimization of internal flow of hydraulic machinery.