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The effect of impeller slot jet on centrifugal pump performance *

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Abstract: To improve the cavitation performance of the centrifugal pump, a new kind of centrifugal pump impeller with slot is proposed. The slot is on the impeller shroud near the suction side of the blade leading edge. So, the fluid with high energy in the impeller front side chamber is drained to the lowest pressure area. The jet flow would compensate the blade inlet flow with a certain energy. With numerical simulations, the pump's inner flows for 5 different slot sizes and their hydraulic performances are compared with those of the prototype pump. The slot jet would result in the increase of the impeller flow rate and the volute area ratio, leading to the head decrease within the whole flow rate range. The slot jet can suppress the reverse flow remarkably at a low flow rate, and the pump efficiency is improved. The slot jet improves the pump cavitation performance effectively, especially at a low flow rate. To some extent, the smaller the slot size is, the better the pump cavitation performance is. The cavitation performance of the slot impeller with a size of 2 mm×1.75 mm is much better than that of the prototype impeller. Compared with the prototype impeller, the lowest pressure near the slot impeller blade inlet and even the pressure in all forepart regions are improved significantly. The slot impeller is shown to be effective in suppressing the cavitation, and the available net positive suction head is improved.

Key words: Centrifugal pump, slot impeller, slot jet, cavitation performance, circulation flow

Introduction

The cavitation is a very complicated harmful phenomenon, which usually occurs in hydraulic machineries. It not only reduces the hydraulic performance, but also causes noise and vibration, and even gives rise to the impeller material loss, shortens the operating life of the hydraulic machineries[1-4]. The lowest pressure area is located on the suction side of the centrifugal pump blade leading edge near the shroud, where the cavitation and the erosion would most easily occur. The cavitation inception occurs when the lowest pressure is lower than the corresponding saturation pressure. The vaporized liquid forms a series of minute bubbles, which gradually grow, develop and eventually collapse[5-10]. The cavitation would result a dramatic degradation of the centrifugal pump hydraulic performance. How to improve the cavitation performance of the centrifugal pump becomes a hot topic in the hydraulic machinery field in recent years, with various interesting studies[11-26], mainly including the passive control methods and the active control methods. With the passive control methods[11-20], the impeller and the inducer are shape optimized to suppress the cavitation inception. Balasubramanian et al.[11] investigated the influence of the centrifugal pump blade leading edge profiles on the cavitation behavior. Kang et al.[12] introduced the inducer to suppress the cavitation instabilities and examined the effect of the inducer blade geometry on the cavitation performance. With the active control methods[21-26], the jet is introduced to optimize the flow near the lowest pressure area. Susan-Resiga et al.[21] proposed a water jet from the tip of the crown cone to mitigate the draft tube instability. Shimiya et al.[22] tried to suppress the cavitation instability by a J-groove on the casing wall near the inducer inlet. With the J-groove the prerotation and the backflow are also suppressed in the upstream. The water jet at the impeller inlet drained the water from the outside source, and the positively-sloped head flow characteristics of the mixed-flow pump were successfully eliminated. Wu et

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al.[23] proposed to add an outside jet equipment before the inducer of the high-speed centrifugal pump to improve the suction performance. The jet drained the water from the impeller outlet to the inlet by a pipe. Chen et al.[24] proposed to design a groove between the blade pressure side and the suction side. The boundary layer separation was effectively suppressed by the jet flow through the groove. The gap drainage blade design method[25-26] was also proposed. A small sub-blade was added to the leading of the main blade. The separation of the boundary layer was effectively inhibited by the leakage flow through the gap. The active control methods can effectively improve the pump cavitation performance without modifying the blade profile. In this research, the slot is directly introduced on the impeller shroud to form the jet, and the leakage flow with a high energy drains the water to the low pressure area. The effect of the slot on the centrifugal pump hydraulic performance and the mechanism of the slot jet are analyzed in detail.

1. Configuration of slot on impeller shroud

A typical centrifugal pump model is used to investigate the effect of the slot jet on the hydraulic performance of the centrifugal pump. The key parameters are shown in Table 1.

Table 1 Main hydraulic parameters of centrifugal pump

Description	Value
Design flow rate, Q_d /m^3·h^{-1}	100
Design head, H_d /m	56.5
Rotation speed, n /r·min^{-1}	2 900
Specific speed, n_s	85.6
Number of blades, Z	6
Impeller outlet diameter, D_2 /mm	210
Width of blade outlet, b_2 /mm	16

For the jet to drain the water to the area of vapor bubbles when they first appear, the slot is made directly on the impeller shroud near the suction side of the blade leading edge. The number of slots is the same as the blade number. The section view of the pump is shown in Fig. 1, and the circumferential position of the slot is shown in Fig. 2. With the pressure gradient between the two sides of the slot, the jet from the impeller front side chamber can suppress the cavitation effectively.

The slot extends along the intersection of the shroud and the blade suction side. Its geometry size has a great effect on the pump energy and the cavitation performance. Here, the jets from the slots of 5 different sizes are investigated to optimize the pump hydraulic performance. The slot is oblong in shape, and its geometry sizes are expressed as $a \times b$. The width of the slot is referenced to the blade thickness δ, and the length is 12 mm, 8mm, 4 mm and 2 mm, respectively. The developed view along the blade is shown in Fig. 3, where ps and ss represent the blade pressure side and suction side, respectively.

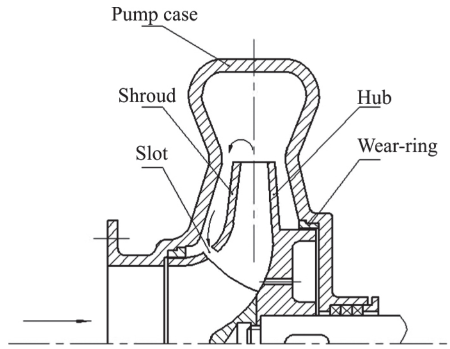


Fig. 1 Section view of centrifugal pump

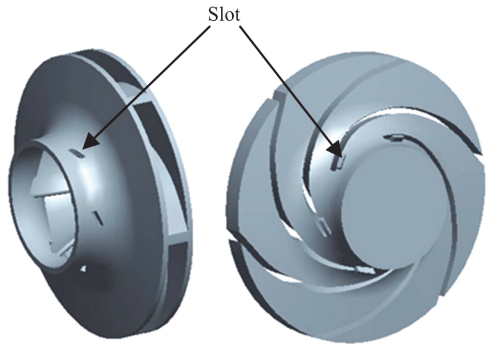
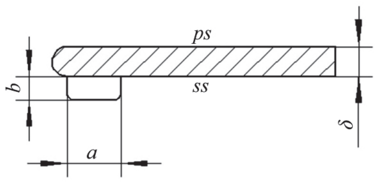


Fig. 2 3-D model of impeller and slot



$a \times b$/mm×mm
12×3.5
8×3.5
4×3.5
2×3.5
2×1.75

Fig. 3 Geometry sizes of the slot

2. Flow model, numerical methods and their validation

2.1 *Computational model and grid generation*

The computational domains, as shown in Fig. 4, are generated according to the model pump mentioned in Section 1, including the suction, the impeller, the volute, the impeller front side chamber and the discharge pipelines. The leakage flow through the clearance of the wear-ring and the back side chamber is neglected. The flow domains are discretized using the tetrahedral mesh and the mesh is refined locally near the slot. In view of the computation accuracy and the computational cost, the grid independence test is carried out, and the total grid number is 1.7×10^6.

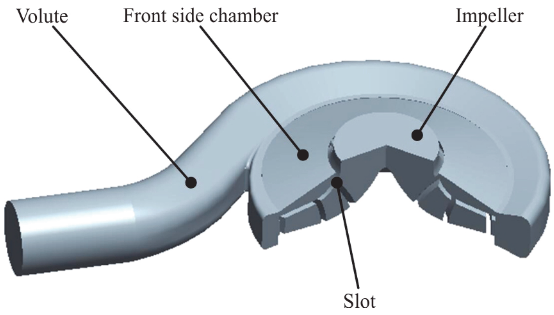
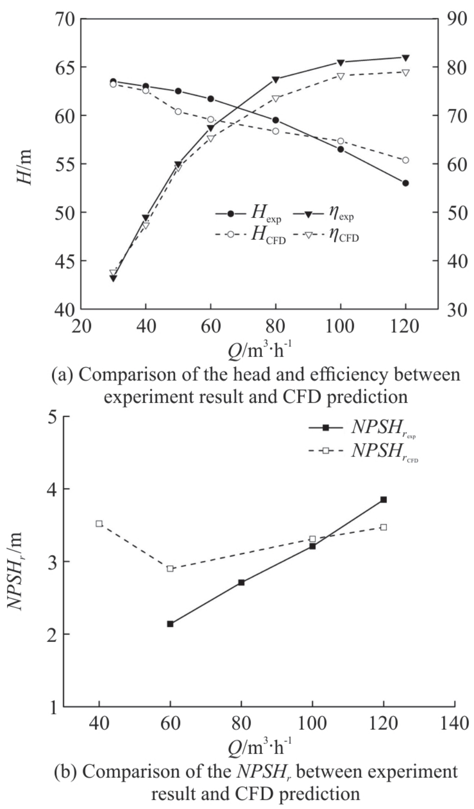


Fig. 4 Computation domains



(a) Comparison of the head and efficiency between experiment result and CFD prediction

(b) Comparison of the $NPSH_r$ between experiment result and CFD prediction

Fig. 5 Comparison of hydraulic performance between experiment result and CFD prediction

2.2 *Computational model and grid generation*

The inner flow simulations are conducted using the CFD code ANSYS Fluent. The flow is simulated based on the RANS equations, and the SIMPLEC algorithm is used to couple the pressure and the velocity. The turbulence effects are modeled by using the RNG k-ε turbulence model with the standard wall functions. The Reyleigh-Plesset model is applied as the cavitation mode. The rotating and stationary domains are coupled by using multiple reference frames. The inlet total pressure and the outlet mass flow rate boundary conditions are specified. The water temperature is set at 25°C. To improve the convergence speed, the steady non-cavitation flow simulation result is taken as the initial value in the cavitation flow simulation. The available net positive suction head ($NPSH_a$) is reduced gradually to evaluate the cavitation performance by reducing the inlet total pressure.

The hydraulic performance comparison between the experiment result and the CFD prediction result is shown in Fig. 5, and the experimental data are provided by the Shenyang Pump Research Institute. As we can see, the prediction errors are within 3.5% in most of the flow range under the design condition.

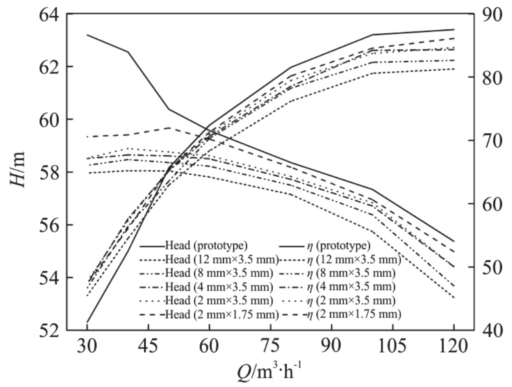


Fig. 6 Comparison of pump hydraulic performance curves for different slot sizes

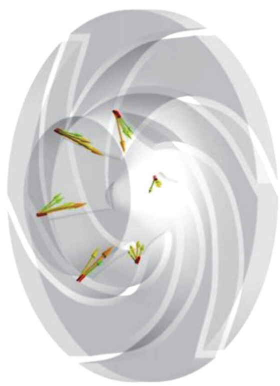


Fig. 7 (Color online) Localized 3-D vector plot of relative velocity

3. Results and discussions

3.1 *Effect of slot jet on pump energy performance*

The comparison of the pump hydraulic performances without and with the slots of different geometry sizes is shown in Fig. 6. It is shown that the

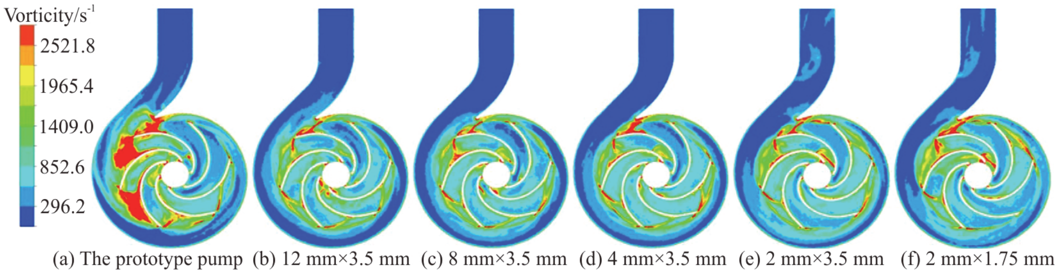


Fig. 8 (Color online) Comparison of vorticity distributions in the middle section of the pump

impeller head gradually decreases with the increase of the slot size within the whole flow rate range. The impeller flow rate is the sum of the pump inlet flow and the slot circulation flow, so the slot jet will increase the impeller flow rate. The pump head decreases with the increase of the impeller flow rate. The leakage from the impeller also means the increase of the volute area ratio, and that is another reason for the head decreases. The localized 3-D vector of relative velocity from slot is showed in Fig. 7.

It can be seen that the leakage flows from the front side chamber to the impeller inlet. The pump efficiency increases along with the reduction of the slot size within the most range of large flow rates. The circulation flow from the impeller front side chamber to the slot is accompanied with a certain energy loss. But the circulation flow also can improve the pump efficiency at some low flow rates. The reverse flow in the pump is a very serious issue at the low flow rate, which is the main reason for the low efficiency. The slot jet can effectively inhibit the reverse flow, so the efficiency is improved. The comparison of the vorticity distribution for slots of different sizes at $0.3Q_d$ is shown in Fig. 8. We can see that the vorticity is very high near the impeller outlet in the prototype pump, and it is greatly reduced with the action of the jet flow. The vorticity changes very slightly with the change of the slot size.

3.2 *Effect of slot jet on pump cavitation performance*

The cavitation simulations are performed with different slot sizes to improve the pump cavitation performance. The $NPSH_r$ corresponding to the 3% head drop is obtained by reducing the inlet total pressure gradually. The $NPSH_r$ varies with the slot size and the flow rate, as shown in Fig. 9.

As we can see, the slot jet can improve the pump cavitation performance effectively. Especially at the low flow rate, the cavitation performance for the slot impellers is greatly improved. When the flow rate is higher than that under the design condition, about $1.2Q_d$, the cavitation performance of the prototype pump is better than that of the slot impeller. Generally speaking, the smaller the slot size is, the better the pump cavitation performance is. Under the design condition, the cavitation performance of the impeller with the slot size of 2 mm×1.75 mm is better than that of the prototype pump. The lower the flow rate is, the better the cavitation performance is. The head drop curves of the pump with the slot size of 2 mm×1.75 mm at different flow rates are shown in Fig. 10.

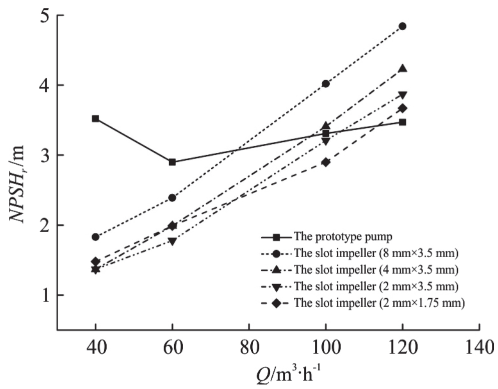


Fig. 9 Comparison of the $NPSH_r$ curves at various slot sizes

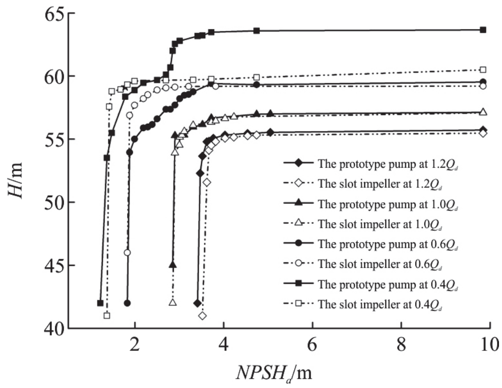


Fig. 10 Comparison of head drop curves at different flow rates for the pumps with the slot size of 2 mm×1.75 mm and the prototype pump

As we can see that the $NPSH_r$ significantly reduces at the low flow rate. At the design flow rate and a large flow rate, the head drop curves of the slot

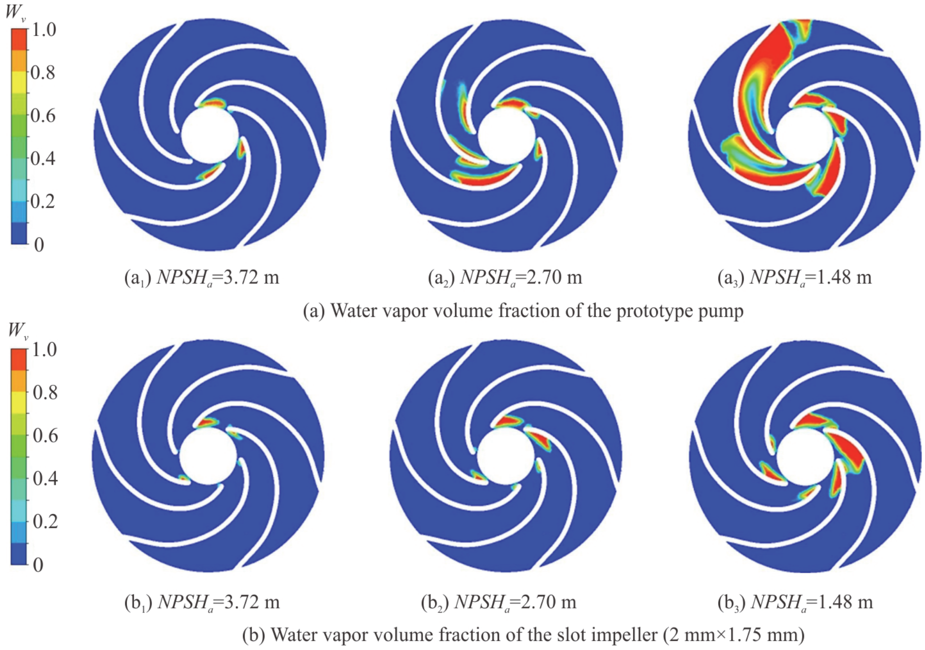


Fig.11 (Color online) Distributions of water vapor volume fraction at middle section of the impeller

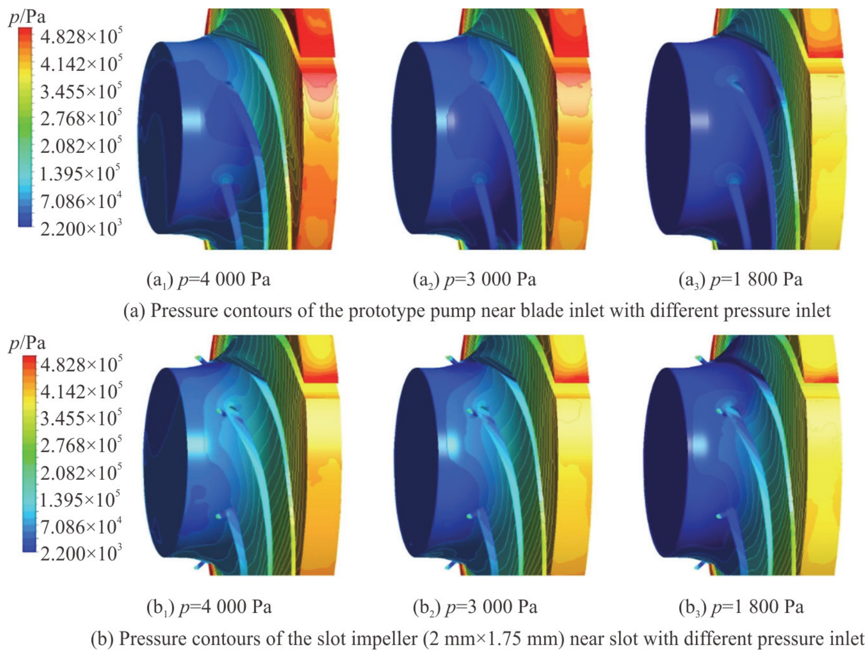


Fig.12 (Color online) Comparison of the pressure contours near the slot with different pressure inlets

impellers are almost the same as that of the prototype pump. The distributions of the water vapor volume fraction at the middle span for the prototype impeller and the slot impeller are compared in Fig. 11. The slot size is 2mm×1.75mm and the flow rate is $0.4Q_d$. As shown in Fig. 11, with the decrease of the inlet pressure, the vapor volume fraction increases gradually, and the volume distribution is significantly asymmetric. When the $NPSH_a$ is 3.72 m, some small and tiny vapors develop on the blade suction side. With the decrease of the pump inlet pressure, the cavities expand gradually. As the $NPSH_a$ drops to 1.48 m, the cavities of the prototype pump are much enlarged and elongated on the blade channel near the tongue of the volute, which is probably due to the interaction between the impeller and the volute. Obviou-

Table 2 Jet flow parameters

Pump flow rate, Q /(m^3/h)	Jet flow rate Q_k /(m^3/h)	Flow ratio $Q_k \cdot Q^{-1}$ /(%)	Jet head/m	Pressure difference/kPa
40	1.11	2.78	76.01	251.42
60	1.12	1.87	74.42	251.46
100	1.18	1.18	71.57	263.76
120	1.19	0.99	69.37	266.35

sly, the cavitation intensity in the pump with the slot jet is less than that of the prototype centrifugal pump, which is the role played by the slot jet flow.

3.3 *Mechanism of jet flow*

The slot jet flow provides an active control to improve the pump cavitation performance. The mechanism is that the jet flow compensates the blade inlet flow with a certain energy, and the available net positive suction head is improved. The slot jet can effectively suppress the intense reverse flow in the pump at a low flow rate, and the flow becomes more uniform. To analyze the mechanism of the jet flow, the pressure contours near the slot are compared with the prototype pump as shown in Fig.12. Compared with the prototype impeller, the lowest pressure near the slot impeller blade inlet and even the pressure in all forepart regions are improved significantly.

The jet flow parameters of the slot impeller (2 mm× 1.75 mm) are shown in Table 2. The jet flow rate slowly increases with the increase of the pump flow rate, while the jet head decreases. So, the power of the jet changes little with the pump flow rate. Although the pressure of the impeller outlet decreases with the increase of the pump flow rate, the pressure drop near the blade inlet increases with the increase of the pump flow rate, with the increase of the pressure difference on both sides of the slot.

The flow ratio listed in Table 2 decreases with the increase of the pump flow rate, but the jet power varies little with the pump flow rate. So, the slot jet flow at a lower flow rate plays a more important role in suppressing the cavitation.

4. Conclusions

In this paper, an active flow control method is proposed to suppress the pump cavitation. The slot is made directly on the impeller shroud near the suction side of the blade leading edge. With the action of the pressure difference on both sides of the slot, the slot jet is formed. The slot jet plays an important role in the suppression of the cavitation and the reverse flow. Based on the numerical reliability simulation, the effect of the slot jet on the centrifugal pump cavitation performance and the energy performance are investigated. The following conclusions can be drawn:

(1) The slot leakage increases the impeller flow rate and the volute area ratio. The pump head decreases to a certain degree. The circulation flow from the impeller front side chamber to the slot will reduce the pump efficiency in most of the flow range. The slot jet also can effectively suppress the reverse flow in the pump at a low flow rate, and then the pump efficiency is improved.

(2) The slot jet can improve the pump cavitation performance effectively, especially at a low flow rate. To some extent, the smaller the slot size is, the better the pump cavitation performance is. The cavitation performance of the slot impeller with a size of 2 mm× 1.75 mm is much better than that of the prototype impeller.

(3) The slot impeller mechanism is that the jet flow compensates the blade inlet flow with a certain energy, and the available net positive suction head is improved. The lowest pressure near the slot impeller blade inlet and the pressure of all forepart regions are improved significantly.

This work explores a new direction for improving the cavitation performance of the centrifugal pump. In future, the manufacture and the orientation of the slot, as well as the stress concentration problem for the slot will be explored.

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