Influence of posterior cavity structure on axial force and pressure distribution of stamping pump

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Abstract. In order to study the influence of different posterior cavity structure on multistage stamping pump characteristics, pressure distribution and axial force, four different posterior cavity structures of arc-type, L-type, plane-type and molded line-type were designed and the full fluid domain models for single stage were established. The performance of the pump was analyzed and verified by experiment and numerical simulation respectively, so as to study the influence of posterior cavity on pressure distribution and flow state in the pump. The results showed that vortex appeared in the flow area between impeller and posterior cavity, it reacted on the fluid in impeller and guide vane, which could improve the single stage head and efficiency, but also increased the pressure pulsation and axial force. The ring cavity formed by bulkhead and shaft of L-type and molded line-type posterior cavity structure could act as balance room and reduce the total axial force, so as to reduce the wear and impact between rotor and pump body. L-type, plane-type and molded line-type posterior cavity could effectively suppress pressure pulsation and improve the operation stability of the pump.

1 Introduction

Since the 1970s, the technology of impeller, guide vane and pump shell of centrifugal pump has been developed through stamping and welding of stainless steel sheet. This type of pump is called stamping welding pump, or stamping pump. Stamping pump has small surface roughness, very thin blades, so the hydraulic performance should be much better than the casting pump in theory. But because of its design method is still not mature, it showed problems such as working condition deviation and low efficiency utilization in engineering practice. Therefore, the design and research of the stamping pump is a novel academic field [1, 2]. For example, Wang [3, 4] based on the test results, established the numerical simulation method of loss, systematically assessed the relationship between different types of energy loss, to explore the optimal angle of import and export and the guide vane width. Huang [5] analyzed the pressure distribution and velocity distribution of the flow field in
stamping pump, then calculated the pump characteristic curve, and validated with the experimental performance, to provide reference for optimal design of stamping pump. Three-dimensional turbulence numerical simulation was carried out by Liu [6, 7] on the internal flow channel of the impellers of a stamping and welding multistage centrifugal pump, and compared with the experimental results, it was proved that the numerical simulation results could predict the flow field distribution in the impeller more accurately. Ma [8] found that the static and static interference between the impeller and the guide vane was the cause of static pressure fluctuation. The mean of static pressure fluctuation gradually increased from the inlet of the impeller to the outlet.

Unlike the casting pump, a cavity is formed between the rear cover plate and the bearing baffle in the stainless steel stamping multistage pump, which is called the posterior cavity for short [9]. In the traditional manufacturing process, the structure of posterior cavity between the impeller and guide vane produces relatively large fluid domain, which led to a certain efficiency loss. At the same time the vortex and reflux of the internal flow of posterior cavity, will cause the pressure pulsation of the impeller flow passage, and destabilize the operation. Traditional multistage centrifugal pump reduces high axial force by setting the balance room and plate, while in stamping pump, it could reduce the axial force through the improvement of the structure of posterior cavity. To abate the high axial force and the pressure pulsation which caused by the imperfect cavity structure during operation of the stamping pump, and explore the optimized cavity structure to raise head and control the cavity fluid energy loss. We made the MDP64-20 multistage stamping pump as the research object, through the design of structure of posterior cavity in the multistage stamping pump, finally set up four different kinds of posterior cavity calculation model. And we made the numerical simulation and experiment comparison to the model, to study the rule about how the posterior cavity structure influence the axial force of stamping pump and pressure distribution of the impeller, and to provide a reference for the design of the stamping multistage pump.

2 parametric modeling and numerical methods.

2.1 Research model

Due to the same structure of per stage of the multistage stamping pump, so the first stage of the multistage stamping pump MDP64-20 was the research object, and the flow state at all stages could be analogized. The basic performance parameters of the model pump were: speed \( n=2900 \text{r/min} \), rated flow \( Q_{\text{m}}=64 \text{m}^3/\text{h} \), single stage rated head \( H=18 \text{m} \), the number of stages was 8, and the rated efficiency was 69\%. The structure parameters were: the blade number was 6, the inlet diameter of the impeller was \( D_1=80 \text{mm} \), the outlet diameter was \( D_2=142 \text{mm} \), and the guide blade number was 12.

The axial section and fluid domain model of the posterior cavity of the original pump were shown in Figure 1. It showed that the original pump posterior cavity fluid domain space was large. The fluid through the impeller passage into the cavity, caused a certain energy loss, and produced high axial force due to the defect structure. In order to reduce the posterior cavity volume and build a low-velocity high-pressure room, respectively designed four posterior cavity structure, arc-type, L-type, plane-type and molded line-type. The arc-type, L-type and plane-type structures were respectively welding a stainless steel plate whose section were arc, L, and plane, between the bearing
plate and bearing seat. The molded line-type structure was welding a stainless steel plate which was same as and parallel to the back shroud line, as shown in Figure 2.

![Diagram of the original pump](image1)

**Figure 1.** Axial plane diagram and fluid domain model of the original pump (a) axial plane diagram (b) fluid domain model.

![Diagram of four kinds of posterior cavity structures](image2)

**Figure 2.** Axial plane diagram of the four kinds of posterior cavity structure (a) arc-type (b) L-type (c) plane-type (d) molded line-type.

At the same time, the whole fluid domain model of four kinds of cavity structure, including impeller, pump cavity, guide vane, inlet and outlet were built. And based on the original model pump, respectively set up pump fluid domain model of arc, L, plane and molded line type. To better monitor the pressure distribution and flow state of fluid in different posterior cavity structures, the whole fluid domain modeling could be performed.

### 2.2 Mesh and numerical calculation method

Unstructured grid was used to divide the fluid domain of four different cavity structure respectively. Divided the fluid domain boundary layer between the shroud and shell, to ensure more accurate simulation results. Convergence inspection to the mesh at the same time. When grid number of arc-type and plane-type fluid domain model was more than 1.3 million, the change of calculated head was less than 1%. And the same for when the grid number of L-type and molded line-type fluid domain model was more than 1.4 million. So in order to meet the requirement of both calculation accuracy and efficiency, respectively, the grid number of four kinds of structure fluid domain model was about 1.4 million.

RNG k-ε model is based on low Reynolds number flow viscosity analytic formula mixed function,
to make the turbulence model smooth transition by the near wall area to the far wall area, so that has a good simulation effect and accuracy. So RNG $k$-$\varepsilon$ turbulent model was adopted in numerical calculation.

The water of flow field was set as $25^\circ \text{C}$. At this temperature, the saturated vapor pressure of water was $3610 \text{Pa}$. In order to compare the influence of the four kinds of structures under different working conditions, the flux of inlet boundary conditions were set as $Q=0.4Q_m, 0.6Q_m, 0.8Q_m, 1.0Q_m, 1.2Q_m$, and the outlet condition was free flow. Using the first order windward format to calculate the flow field; no slip conditions on each wall; defining atmospheric pressure as reference pressure; The convergence accuracy was $10^{-4}$. The unsteady numerical simulation step length was set as $1.149 \times 10^{-4} \text{s}$, namely the impeller rotated $2^\circ$ every time step. The impeller rotated one revolution every 180 time steps. The total time steps was 1080, and the impeller rotated six revolution.

To monitor different cavity structure on the influence of the pressure pulsation, set $P_1$ and $P_2$ monitoring between the impeller outlet and the guide vane inlet, monitoring the position as shown in Figure 3, $P_1$ located in the guide vane edge, and $P_2$ located between two guide vanes.

![Diagram](image)

**Figure 3.** Location diagram of monitoring point.

3. Test verification and external characteristic analysis.

3.1 Test equipment

The MDP64-20 vertical multistage stamping pump was set as test model, then welded a cambered surface between the rear cover plate and the bearing baffle, to process into arc-type cavity structure. And it was used to verify the reliability of numerical simulation and contrast, to analyze the differences between the results of numerical simulation and experimental values. The experiment was driven with 45 kw motor, and vacuum gauge and pressure gauge was installed in the inlet and outlet respectively, through the electric flow to adjust pipe flow. We measured the pressure difference between the inlet and outlet of the condition of $Q=0.4Q_m, 0.6Q_m, 0.8Q_m, 1.0Q_m, 1.2Q_m$. The head under various operating conditions was calculated by equation (1), and the efficiency by equation (2).

$$H = \frac{\Delta P}{\rho g}$$  \hspace{1cm} (1)

$$\eta = \frac{\rho gHQ_m}{3600N}$$  \hspace{1cm} (2)
Where $H$ is the head; $\Delta P$ is pressure difference between inlet and outlet; $\rho$ is the density of fluid medium, $\rho = 998 \text{kg/m}^3$; $g$ is the local acceleration of gravity, $g = 9.8 \text{m/s}^2$; $N$ is the shaft power and was obtained by the force transducer.

### 3.2 Analysis of external characteristics.

Figure 4 showed the flow-head curve obtained by experiment and numerical simulation. The head of the test pump in the standard condition reached 18.335m, which was 1.86% higher than the rated head of the prototype pump 18m. Compared the results of the numerical simulation and test value, the variation trend of the curve was almost consistent. The head of simulation value of arc-type cavity structure was higher than the test values in the whole traffic condition. This was because the test process of measurement was the head of the whole multistage pump, and when calculate the single-stage head, interstage leaking and the corresponding energy loss made the test value slightly lower than the simulation. Therefore, the numerical simulation could accurately predict the performance of four types of the posterior cavity stamping pump.

The numerical simulation results showed that the head value of the arc-type cavity was the largest under different working conditions, and the predicted value was 19.32m in the standard condition. It is indicated that the arc-type structure could reduce the energy consumption inside the flow channel of the flow parts, improve the pressure difference of the inlet and outlet, and raise the head. However, the effect of L-type, plane-type and molded line-type structure on the head was not obvious.

![Figure 4](image-url)

**Figure 4.** Quantity-head curves of test and numerical simulation results of four types.

Figure 5 showed the flow-efficiency curve obtained by experiment and numerical simulation. The efficiency of the test pump in the standard condition was 70.15%, and compared with the rated prototype pump efficiency of 69% increased by 1.15%. This was because, on the one hand, due to the installation of the welding surface, the volume of the posterior cavity was greatly reduced, so that the flow through the posterior cavity decreased, and the energy dissipation decreased, which meant that the consumption of the shaft power decreased. On the other hand, due to the improvement of the head, the efficiency must increase according to equation (2).

The efficiency trend obtained by numerical simulation was the same as the experiment. The
Simulation value of arc-type cavity structure under different conditions of efficiency was maximum, but L-type, plane-type and molded line-type structures had not raised obvious efficiency. This was because only reduced the energy dissipation of fluid in the cavity, the promoting of efficiency was limited. The efficiency of the plane-type cavity structure was the lowest in the four structures.

![Figure 5. Quantity-Efficiency curves of test and numerical simulation results of four types](image)

4. Pressure distribution analysis.

4.1 Radial pressure distribution analysis.

Because when under different flow, the external characteristics of the four kinds of posterior cavity structures showed the same trend. So based on the standard condition, the performance of the four posterior cavity structures under different working conditions could be predicted. Fig. 6 showed the pressure distribution of the cross section of impeller under standard conditions. Figure 6 showed that rather than volute, the guide vane showed little radial asymmetric structure, so the pressure distribution showed center symmetric. And between every two blades pressure distribution was almost the same. Pressure gradually increased in gradient from the inside to outside. Comparing the four different cavity structure stress distribution, in the impeller passage, pressure distribution were almost identical. That meant different cavity structure did not cause the change of the distribution of pressure in the impeller passage. The high pressure area of arc-type was more expansive than the other three kinds between the edge of impeller and the guide vane, which meant the improvement of the head.

![Figure 6. Radial plane pressure distribution of different kinds of posterior cavity structure.](image)
4.2 Analysis of pressure pulsation.

The characteristics of pressure pulsation can be expressed as pressure pulsation coefficient, as shown in equation (3).

\[ C_p = \frac{P - \overline{P}}{0.5 \rho u_s^2} \]  

(3)

Where \( C_p \) is the pressure fluctuation coefficient; \( P \) is the instantaneous pressure value, Pa; \( \overline{P} \) is the average pressure value, Pa; \( \rho \) is fluid density, kg/m\(^3\); \( u_s \) is the outlet velocity of the impeller, m/s\(^{-1}\).

The pressure pulsation of the last 2 revolutions was analyzed. Figure 7 showed the pressure pulsation time domain diagram of different monitoring points. It could be seen from Figure 7 (a), that the pressure at the blade of guide vane reached the maximum when the impeller blade swept to the blade of the guide blade. When the impeller blade reached the position of the most remote, the pressure reached the minimum. Because the number of blades was 6, there were 6 peaks and troughs in a cycle. It could be seen from Figure 7 (b) that the pressure pulsation in the middle area of the guide blade was weaker than that in the blade area. This was because the vane was center symmetrical structure, had no high pressure pulsation of spiral pressurized water chamber, and through the pressure balance of symmetry points, to reduce the radial force of the pump.

Compared different cavity structure of the numerical simulation results, the arc-type showed more severe pressure pulsation. This was because the arc-type cavity structure improved peak stress of the guide blade area, so led to the more serious pressure pulsation. In addition, after two high peaks, arc-type cavity structure could reach a lower peaks, it was also benefit from the symmetrical structure. The flow guiding effect of the guide vane could balance the high pressure pulsation in a cycle, gradually balance to a steady value. And pressure pulsation of L-type, plane-type and molded line-type structure was excellent, just showed a small amplitude, which could greatly reduce the vibration and noise. And the reduction of pressure pulsation could also reduce the relative displacement of the pump and shaft, improve the service life of pump and running stability.

![Figure 7](image-url)  

(a)  

Figure 7. Time domain diagram of pressure pulsation at different monitoring points (a) P₁ (b) P₂.

5 Axial force analysis
5.1 Axial pressure distribution analysis.

Figure 8 showed the pressure distribution of the axial section of four different posterior cavity structures in standard conditions. The amplitude of the pressure of the fluid in the posterior cavity of the four structures was very close. So this proved the head increase of arc-type cavity structure and the pressure pulsation reduction of L-type, plane-type and molded line-type cavity structures, were not due to change of pressure distribution in cavity. In addition, pressure amplitude of fluid in posterior cavity and the front cavity was very close, only the area between the impeller and cavity had a slight pressure surges due to the sudden change of flow direction and structure of the discontinuous. The pressure distribution was beneficial to improve the stability of the pump and the vibration of the bearing baffle and shaft. At the same time, the uniform distribution of pressure also inhibited the fluid exchange between impeller and cavity, so that reduced the energy loss and improved the efficiency of the pump.

![Figure 8. Axial plane pressure distribution of different kinds of posterior cavity structure.](image)

5.2 Axial flow analysis.

Figure 9 showed the velocity distribution and streamline of the axial section of four different posterior chamber structures in standard conditions. The velocity gradient of the axial decay formed in the four posterior cavity structures. In addition, there was a widespread vortex in front cavity, this was because the closed structure of front cavity caused the fluid back flow, and the vortex in the plane-type cavity structure was more apparent. It can be found from the axial section flow chart of the arc-type that the vortex formed in the area of the inlet of posterior cavity, which only existed in the arc-type. This was due to the effect of the arc-shaped partition, it caused a large part of the flow backflow to form a vortex. Such a vortex would react the flow in the impeller, to increase the fluid pressure in the impeller and the guide vane and raise the head of the pump. But at the same time, the existence of the vortex caused more severe pressure pulsation, which was consistent with the more serious pressure pulsation of the arc-type.

The diaphragm and shaft of the L-type and molded line-type posterior cavity formed the annular chamber. In the annular chamber, the fluid velocity was almost zero, and there was a small amount of vortex. The annular chamber and its internal low-velocity high-pressure fluid could play the role of balance chamber, so that to keep the balance of the axial force of centrifugal pump, avoid rotor axial displacement, reduce collision, friction and wear between the impeller and the pump shell, which greatly impacted the pump efficiency and the effect of operation reliability.
5.3 Axial force analysis.

The axial force of the multi-stage centrifugal pump is mainly divided into three parts: (1) the dynamic counterforce acting on the impeller caused by the fluid in the impeller inlet and outlet due to the change of the flow state; (2) the guide vane force acting on the guide vane caused by fluid on the guide blade inlet and outlet due to the change in the flow state; (3) the shroud force of the fluid acting on the impeller shroud [10]. The three parts axial force was the force caused by the pressure of the fluid acting on the overcurrent part of the pump. Therefore, when the pressure of the fluid increased, the three parts axial force will increase. It was assumed that the axial force was positive from the outlet to inlet, and Table 1 showed the numerical simulation results of various axial forces and the total axial force of the four different posterior cavity structures in the standard conditions. It could be found from the data in Table 1 that the guide vane force was far greater than the dynamic counterforce and the shroud force, and in the opposite direction.

Table 1. Axial force in single stage of four different posterior cavity structures

|              | dynamic counterforce (N) | guide vane force (N) | shroud force (N) | Total axial force (N) |
|--------------|--------------------------|----------------------|------------------|----------------------|
| Arc-type     | 201.032                  | -1698.88             | 239.026          | -1258.822            |
| L-type       | 188.73                   | -1649.72             | 252.077          | -1208.913            |
| Plane-type   | 190.518                  | -1658.66             | 227.961          | -1240.181            |
| Molded line-type | 189.517               | -1661.39             | 258.981          | -1212.892            |

The single stage total axial force of arc-type cavity structure was the biggest in four cavity structure. This was because the arc-type cavity structure could improve the pressure of the fluid of impeller, compared with the other three structure. When the head raised, it will cause greater dynamic counterforce, shroud force and guide vane force, and the guide blade force increased more than the sum of the shroud force and dynamic counterforce. The total axial force of L-type and molded line-type cavity structure were similar and smaller than arc-type and plane-type. The total axial force of L-type reduced about 3.9%, and molded line-type reduced about 3.7%. This was due to the annular chamber of L-type and molded line-type cavity structure. It played the role of the balance chamber,
raised the shroud force, and then reduced the total axial force. The effect could reduce the rotor axial impact, reduce the friction, wear and vibration of the shaft, impeller and pump body, so that improved the stability and longevity of the pump operation.

6 Conclusion
(1) The numerical simulation results showed the same trend of the curves of flow-head and flow-efficiency as that of the experimental results. The arc-type posterior cavity structure showed obvious increase of head and efficiency, and the test result showed that the increase of head was 1.86% and the efficiency was 1.15%.

(2) Because the symmetrical structure of guide vane, the pressure distribution in the impeller and the guide blade was very uniform. The high pressure area of arc-type posterior cavity was more expansive, with the increase of head, also showed a more severe pressure pulsation. L-type, plane-type and molded line-type posterior cavity structures could effectively suppress pressure pulsation and improve the stability of pump operation.

(3) The vortex of the arc-type posterior cavity reacted on the fluid in the impeller, which was the reason for the improvement of head, and also increased the total axial force. The annular chamber of L-type and molded line-type posterior cavity structures had the effect as the balance chamber, so that to reduce the single stage total axial force and reduce the impact and wear, finally improved the stability and service life of pump.

Conference

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