Influence of inlet flow field uniformity on the performance of the nuclear main pump

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Abstract. A test loop system with adjustable spoiler function and visual structure was built in this paper. A PIV test system was adopted to investigate the influence of the uniformity of the inlet flow field on the performance of a nuclear main pump scale model, and the NUMECA software was used to numerically simulate the test cases. The research results show that when the rotation direction of the spoiler blade is opposite to that of the impeller, the greater the deviation of the spoiler blade from the axial angle, the more the pump efficiency and head drop; proper positive pre-rotation will not reduce the pump efficiency, but will increase the pump head; for the same spoiler blade deflection angle, when its structure produces irregular flow field disturbances under non-centrosymmetric conditions, the flow field downstream of it will not produce clear swirling flow, and it will not have more impact on the performance of the pump; the axial velocity distribution uniformity δ and the velocity weighted average angle θ, θ' have obvious direct effects on efficiency, and in the δ interval of each working condition in this paper (the difference between the maximum value and the minimum value of δ is 0.093), the speed weighted average angle has a greater impact on efficiency.

Nomenclature.

Q Rate of flow
δ Axial velocity distribution uniformity
θ Weighted average angle parameter of velocity
θi Absolute axial velocity angle
θ' Weighted average angle parameter of velocity
η Efficiency
H Head

1. Introduction

With the rapid development and construction of nuclear power in China, the market demand of nuclear power pump, which is the transport heart of hundreds of large and small systems, is also increasing day by day. Different from pumps used in petroleum, chemical, thermal power and other industrial fields, nuclear power pumps should not only meet the normal operation of the whole main and auxiliary systems, but also strictly ensure the safety, reliability, seismic behavior and durability of the equipment, in particular, as the heart of nuclear safety and related systems, nuclear safety grade pumps must meet stringent safety and quality requirements.

Some scholars have done some research on the influence of inlet flow field uniformity on the flow field and performance in the pump. By directly solving the three dimensional Reynolds average N-S
equation and the standard k-ε turbulence model equations, Lu Guanglin simulated the flow field in the inlet channel of large and medium-sized pumping stations, and then performed hydraulic optimization calculation for the inlet channel with complex geometry [1]. Tang Fangping introduced the research progress of the inlet flow field in China, discussed the research methods of the inlet flow field, and analyzed the advantages and disadvantages of various methods [2]. Liu Haijian designed three kinds of flow field uniformity devices for the inhomogeneity of water pump inlet flow field. Numerical simulation and experimental study show that the resistance coefficient of rectangular grid type is more than 3 times that of the diversion type and the concentric circle type uniform device, and the axial velocity angle of the section is the most important factor affecting the uniformity of the flow field and all the three flow field devices can significantly increase the axial velocity angle of the inlet section [3]. In view of the influence of elbow inlet channel on cavitation characteristics of large axial flow pump unit, Yan Hao derived the relationship between pump cavitation allowance and inlet velocity uniformity based on inlet velocity triangle theory, and simulated the external characteristic curve of large axial flow pump with CFD technology, and compared with the test results [4]. Yang Minguan designed 5 kinds of water guide cones, and used Fluent software to calculate the three-dimensional flow field of each type of water guide cones, analyzed the flow characteristics of the water guide cones, and summarized the three flow parts of the water guide cones flow field and the velocity distribution law of the axial surface of the flow field. The variation of axial velocity distribution uniformity and weighted average deflector angle with the deflector of water conducting cone are summarized [5].

The interactive combination of numerical simulation and experimental research is an important means to study the flow law in pump and guide the design of pump hydraulic model [6-13]. However, the accuracy of the numerical simulation results must be verified by a large amount of objective and reliable data, and the appropriate testing method is the key to ensure the reliability of the test results [14-20]. PIV particle velocimetry system can record the relevant information of the entire smooth surface at the same time to obtain the instantaneous velocity field of flow, which has become an advanced measurement method for studying the flow field inside turbomachinery [21-28].

At present, there is a lack of visual experimental research on the uniformity of flow field at pump inlet. In this paper, a visualization test system of flow field was established, and PIV produced by TSI was used as a test method to investigate the influence of flow field uniformity at the inlet of a nuclear main pump model on pump performance.

2. Research methods

2.1. Test system

In this paper, the flow field visualization experiment and numerical simulation are combined to study the shrinkage model of a nuclear main pump. Build the test system with adjustable spoiler function and visual structure, hollow polystyrene glass beads with average particle size of 20-30 μm were used as tracer particles for flow field measurement and PIV system was used for flow field measurement. At the same time, NUMECA software is used for numerical calculation of corresponding operating cases. The computational domain includes inlet extension section, spoiler section, impeller, guide blade, volute and outlet extension section, and the calculated number of grid nodes is 42,096 million.

In order to achieve adjustable flow uniformity at the pump inlet, a spoiler section is set upstream of the inlet. Four spoiler blades are evenly arranged in the inlet spoiler section, and each one can swing freely independently. In order to carry out PIV visualization experiment, the pump inlet was designed as a visual structure. The structure consists of plexiglass tubes and a transparent square water tank on both sides to reduce the distortion of light due to different refractive indices [29-31]. At the same time, only the two sides of the water tank that need to project laser and shoot are made into transparent structure, while the other four sides are all opaque, which can effectively reduce the interference of external light. The test system is shown in figure 1.
The test speed was selected as 896rpm, and six inlet flow cases were designed for the spoiler angle of 4 blades 0°, 4 blades +15°, 4 blades +30°, 2 blades +30°&2 blades -30°, 4 blades -15°, and no spoiler section. In order to investigate the influence of flow rate on the unevenness of velocity distribution, three flows (0.8 design flow rate, design flow rate and 1.2 design flow rate) were selected for each inlet. The arrangement of test cases is shown in table 1.

Table 1. Test cases.

| Cases | 1   | 2   | 3   | 4   | 5   | 6   |
|-------|-----|-----|-----|-----|-----|-----|
| inlet condition (blade angle × number of blades) | 0°×4 | 0°×4 | 0°×4 | 15°×4 | 15°×4 | 15°×4 |
| Q [m³/h] | 657 | 821 | 985 | 657 | 821 | 985 |

(b)

| Cases | 7   | 8   | 9   | 10  | 11  | 12  |
|-------|-----|-----|-----|-----|-----|-----|
| inlet condition (blade angle × number of blades) | 30°×4 | 30°×4 | 30°×4 | -30°×2 | -30°×2 | -30°×2 |
| Q [m³/h] | 657 | 821 | 985 | 657 | 821 | 985 |

(c)

| Cases | 13  | 14  | 15  | 16  | 17  | 18  |
|-------|-----|-----|-----|-----|-----|-----|
| inlet condition (blade angle × number of blades) | -15°×4 | -15°×4 | -15°×4 | no spoiler | no spoiler | no spoiler |
| Q [m³/h] | 657 | 821 | 985 | 657 | 821 | 985 |

2.2. Uniformity parameters

In order to quantitatively measure the uniformity of velocity distribution of each fluid element in the flow field at the pump inlet, the axial velocity distribution uniformity parameter of the flow field is introduced and defined as:
\[
\delta = 1 - \frac{1}{u_a} \left( m^{-1} \times \sum_{i=1}^{m} (u_{ai} - \overline{u}_a) \right)^{1/2} \times 100\% 
\]

Here, \( \overline{u}_a \) is the average axial velocity of the inlet section, \( u_{ai} \) is the axial velocity of each element in the inlet section, \( m \) is the number of measuring points on the inlet section. The axial velocity distribution uniformity of section flow field \( \delta \) reflects the design quality of the inlet flow passage of the pump, the closer to 100\%, the more uniform the axial velocity distribution of the inlet flow passage section is.

In addition, in order to quantitatively measure the uniformity of velocity angle distribution of each fluid element in the flow field at the pump inlet, a weighted average angle parameter of velocity in the flow field at the section \( \theta \) is introduced, which is defined as:

\[
\theta = \left( \sum_{i=1}^{m} u_{ai} \right)^{-1} \times \sum_{i=1}^{m} u_{ai} \left[ 90^\circ - \arctan \left| \frac{u_i}{u_{ai}} \right| \right] 
\]

Here, \( u_{ai} \) is the radial velocity of each element in the inlet section. The weighted average angle of cross section flow field velocity \( \theta \) reflects the radial velocity of each element in the cross section of the flow field at the pump inlet. The closer the \( \theta \) value is to 90\(^\circ\), the smaller the radial velocity is, and the higher the perpendicularity between the flow direction at the pump inlet and the cross section of the pump inlet is.

According to the definition of \( \theta \), it can be seen that this coefficient cannot reflect the direction of radial velocity, that is, it cannot describe the positive or negative character of the included angle between radial velocity and axial velocity. Therefore, we define the absolute axial velocity angle of each unit on the pump inlet cross section as follows:

\[
\theta_i = \arctan \left| \frac{u_{ai}}{u_{ai}} \right| 
\]

And another velocity weighted average angle parameter \( \theta' \) is proposed, which is defined as:

\[
\theta' = \left( \sum_{i=1}^{m} u_{ai} \right)^{-1} \times \sum_{i=1}^{m} u_{ai} \left[ 90^\circ - \arctan \left| \frac{u_i}{u_{ai}} \right| \right] 
\]

3. Results and analysis

3.1. Analysis of the influence of turbulence conditions on pump performance

Comparing the flow field velocity distribution of each case obtained by the test and CFD, it can be seen that the two are roughly similar. For the performance test, if the inlet pressure measuring device is installed downstream of the turbulent section, it is impossible to obtain a stable and accurate pressure value because the flow field here will be in a turbulent state. If the inlet pressure measuring device is installed upstream of the turbulence section, it is also impossible to obtain accurate pump performance because the measured efficiency will include the turbulence section piping. Therefore, this test only performed performance tests on 9 cases under the condition of no turbulence, 4 blades 0\(^\circ\) and 4 blades -15\(^\circ\) turbulent flow conditions with a relatively uniform inlet flow field velocity distribution. The test results are shown in table 2. Here we make case 17 the base case (\( \eta \) and \( H \) of which are 1), \( \eta \) and \( H \) values of other cases are ratios to this. The numerical calculation objects of pump performance include all cases, and the calculation results are shown in table 3.
Table 2. Results of pump performance tests.

| Cases | 1 | 2 | 3 | 13 | 14 | 15 | 16 | 17 | 18 |
|-------|---|---|---|----|----|----|----|----|----|
| inlet condition (blade angle \× number of blades) | 0°×4 | -15°×4 | no spoiler |
| \(\eta\) [%] | 1.04 | 1.01 | 0.76 | 1.07 | 1.00 | 0.74 | 1.06 | 1.00 | 0.71 |
| \(H\) [m] | 1.40 | 1.06 | 0.60 | 1.47 | 1.09 | 0.64 | 1.38 | 1.00 | 0.57 |

Table 3. Results of pump performance numerical calculation.

(a)

| Cases | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|-------|---|---|---|---|---|---|---|---|---|
| inlet condition (blade angle \× number of blades) | 0°×4 | 15°×4 | 30°×4 |
| \(\eta\) [%] | 0.99 | 0.99 | 0.80 | 0.98 | 0.96 | 0.73 | 0.95 | 0.93 | 0.61 |
| \(H\) [m] | 1.25 | 1.00 | 0.61 | 1.18 | 0.90 | 0.51 | 1.10 | 0.80 | 0.36 |

(b)

| Cases | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 |
|-------|----|----|----|----|----|----|----|----|----|
| inlet condition (blade angle \× number of blades) | 30°×2, -30°×2 | -15°×4 | no spoiler |
| \(\eta\) [%] | 0.95 | 0.93 | 0.71 | 0.99 | 0.98 | 0.80 | 0.99 | 1.00 | 0.81 |
| \(H\) [m] | 1.21 | 0.97 | 0.54 | 1.30 | 1.07 | 0.69 | 1.26 | 1.00 | 0.62 |

Figure 2 and figure 3 show the comparison of pump efficiency and head calculation results under various turbulent flow conditions. It can be seen from the figure that the pump efficiency under the condition of no turbulence section, 4 blades 0°, and 4 blades -15° turbulence are similar at each flow point. The efficiency of 4-blade 0° and 4-blade -15° cases are not decrease, on the contrary, the head of 4-blade -15° case is slightly higher than other cases. Because under the condition of 4 blades -15°, the fluid passing through the turbulence section will produce the same rotating flow as that of the impeller, which is equivalent to applying a positive pre-rotation at the pump inlet, so it will not cause disturbance to the flow field in the pump. At the same time, for uniform flow, the spoiler blade added in this experiment will not have a significant impact on the flow field when it is at the 0° position.

For the case of 4 blades at 15°, the efficiency and head are significantly reduced, and for the case of 4 blades at 30°, the efficiency and head are further reduced significantly. This shows that the disturbance generated by the spoiler blades that is opposite to the impeller rotation will have a non-negligible impact on the pump performance, and the greater the disturbance, the more performance degradation. For the cases of 2 blades 30° & 2 blades -30°, the efficiency in medium and small flow is similar to that of 4 blades 30°. In the case where the efficiency of large flow is higher than that of 4 blades 30°, the head at each flow point is higher than 4 blades 15°. This shows that the flow field disturbance generated by the spoiler blades under non-centrosymmetric conditions is not as regular as symmetrical, and will not produce a swirling flow with a clear direction, but it will not bring greater adverse effects on the pump performance.
Figure 2. Comparison of pump efficiency under various turbulent flow conditions calculated.

Figure 3. Comparison of pump heads under various turbulent flow conditions calculated.

Figure 4 and figure 5 show the test and calculation results of the efficiency and head under the case of no turbulence section, 4 blades 0°, and 4 blades -15°. It can be seen that the test efficiency is slightly lower than the calculated value under small flow conditions. With the increase of the flow rate, the test efficiency decreases rapidly and is significantly lower than the calculated value. This shows that the disturbance in the flow field except for the spoiler blades (there is a three-way structure upstream of the pump inlet) has a more obvious impact on the performance of the pump, and the impact is strengthened with the increase of the flow rate. Overall, the test head is lower than the calculated head. The head of the 4-blade -15° case is the highest, which is consistent with the calculation trend.

For the test results, the efficiency values under the three turbulent conditions are similar, and the 4-blade 0° and 4-blade -15° cases are even higher than the case without turbulent blades at large flow.
Similarly, the head without turbulence blades is the smallest under the three turbulence conditions. This is because the $0^\circ$ and $-15^\circ$ spoiler blades not only have no adverse factors on the flow field, but also weaken and rectify the inherent external turbulence of the system.

![Comparison of pump efficiency obtained by experiment and calculation.](image1)

**Figure 4.** Comparison of pump efficiency obtained by experiment and calculation.

![Comparison of pump heads obtained by experiment and calculation.](image2)

**Figure 5.** Comparison of pump heads obtained by experiment and calculation.

3.2. Analysis of uniformity of velocity distribution

In order to study the flow field under various turbulent flow conditions more intuitively and in-depth, the cross section of the pipeline flow field was taken as the research object in the experimental observation section. The diameter of the cross-section is 1.18 times that of the inlet pipe and is 414 mm from the pump inlet. Figure 6 to figure 8 show the axial velocity distribution on the inlet section of each flow rate and each turbulent flow condition. The Y coordinate value from small to large is the bottom-up direction of the pipe in the figure.
Figure 6. Axial velocity distributions on the inlet sections of each turbulent flow case under 0.8 times design flow: (a) 4 blades 0°, (b) 4 blades -15°, (c) 4 blades +15°, (d) 4 blades +30°, (e) 2 blades +30° & 2 blades -30°.
Figure 7. Axial velocity distributions on the inlet sections of each turbulent flow case under design flow: (a) no spoiler section, (b) 4 blades 0°, (c) 4 blades -15°, (d) 4 blades +15°, (e) 4 blades +30°, (f) 2 blades +30° & 2 blades -30°.

Figure 8. Axial velocity distributions on the inlet sections of each turbulent flow case under 1.2 times the design flow: (a) 4 blades 0°, (b) 4 blades -15°, (c) 4 blades +15°, (d) 4 blades +30°, (e) 2 blades +30° & 2 blades -30°.

By comparison, it can be seen that, in general, for all cases except 2 blades +30° & 2 blades -30°, the axial velocity of the upper part of the cross-sectional flow field is small, and the axial velocity of the lower part is large. This shows that the disturbance generated by the upstream elbow presents the radial velocity distribution characteristics here. Among them, the distribution characteristics of the
three cases of 4 blades 0°, 4 blades -15°, and 4 blades 15° are relatively similar. However, due to the different angles of the spoiler blades, the effect on the elbow disturbance is not exactly the same, so the characteristics of the high-speed zone are slightly different. The radial gradient of the cross-sectional velocity of the undisturbed section and the 4-blade 30° case is smaller than other cases, that is, the axial velocity distribution of these two cases is more uniform. For the cases of 2 blades +30° & 2 blades -30°, the axial velocity distribution not only has a gradient in the longitudinal direction of the pipeline, but also has an obvious velocity gradient in the transverse direction. As the flow rate increases, the axial velocity of the cross section will increase correspondingly, but the distribution trend is consistent, and the radial velocity gradient has no obvious regular changes.

Figure 9 and figure 10 show the comparison of the axial velocity value and the absolute axial angular distribution at the inlet section of the 4-blade 0° and the 4-blade 30° at the design flow. Through comparison, it can be found that for these two cases, both have the same characteristics that the position with a high axial velocity value has a small absolute axial angle, and the position with a low velocity value has a large absolute axial angle. But there are obvious differences under the same characteristics. Due to the guiding effect of the spoiler blades, the absolute axial angles of the speeds at various locations within the case of the 4 blades 30° are quite different. For the case of 4 blades 0°, the total velocity values are similar everywhere in the pipeline. The fluid energy in each micro-element flow tube remains basically constant, indicating that the attenuation of the flow field energy by the spoiler blades is small. For the case of 4 blades 30°, it can be seen from the comparison of axial velocity and angle that the spoiler blade obviously has a certain influence on the momentum of each micro-element flow tube, so the flow field energy loss is greater.

![Figure 9](image)

**Figure 9.** Axial velocity value and angle distributions of inlet section of 4 blades 0° under design flow: (a) axial velocity value, (b) absolute axial angular of velocity.
Figure 10. Axial velocity value and angle distributions of inlet section of 4 blades +30° under design flow: (a) axial velocity value, (b) absolute axial angular of velocity.

Table 4 is a summary of the axial velocity distribution uniformity $\delta$ and the velocity-weighted average angle $\theta$, $\theta'$ for each case obtained from the test. Figure 11 shows the comparison of the uniformity $\delta$ of axial velocity distribution for each cases. Figure 12 and Figure 13 show the comparison of the weighted average angles $\theta$ and $\theta'$ of speeds for each cases. For the uniformity $\delta$, the closer its value is to 1, the better the uniformity. It can be seen that the undisturbed section is the highest, followed by the 4 blades 30° case, and then the 4 blades -15° case. The cases of 4 blades 0°, 4 blades 15°, 2 blades +30° & 2 blades -30° are relatively low. There is no obvious trend law between uniformity $\delta$ and flow.

For the speed-weighted average angles $\theta$ and $\theta'$, the closer their values are to 90°, the better the uniformity. It can be seen that the cases of 4 blades 0° and 4 blades 15° are the closest to 90°, followed by no turbulence and 2 blades +30°&2 blades -30°. The $\theta$ value of the 4 blades 15° case is lower, and the 4 blades 15° case is the lowest. The distribution of the weighted average angle and the flow rate also have no obvious trend law, and there is no direct relationship with the uniformity $\delta$, it only depends on the incoming flow conditions and the turbulent flow conditions.

Table 4. $\delta$, $\theta$ and $\theta'$ on the entrance section of each case in the test.

(a)

| Cases | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|-------|---|---|---|---|---|---|---|---|---|
| inlet condition (blade angle × number of blades) | $0^\circ \times 4$ | $15^\circ \times 4$ | $30^\circ \times 4$ |
| $\delta$ | 0.876 | 0.866 | 0.899 | 0.868 | 0.884 | 0.888 | 0.863 | 0.921 | 0.942 |
| $\theta$ [$^\circ$] | 88.538 | 89.032 | 90.283 | 83.828 | 85.980 | 85.938 | 75.321 | 83.627 | 80.533 |
| $\theta'$ [$^\circ$] | 87.905 | 88.170 | 88.178 | 82.031 | 82.661 | 84.352 | 74.298 | 76.419 | 75.161 |

(b)

| Cases | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 |
|-------|----|----|----|----|----|----|----|----|----|
| inlet condition (blade angle × number of blades) | $30^\circ \times 2$, $-30^\circ \times 2$ | $-15^\circ \times 4$ | no spoiler |
| $\delta$ | 0.849 | 0.881 | 0.909 | 0.890 | 0.903 | 0.902 | - | 0.935 | - |
| $\theta$ [$^\circ$] | 81.672 | 87.135 | 92.331 | 90.233 | 90.657 | 90.690 | - | 88.050 | - |
| $\theta'$ [$^\circ$] | 81.129 | 85.962 | 86.425 | 87.410 | 87.092 | 85.331 | - | 83.405 | - |
Combine the previous comparison of pump performance with the three figures of speed uniformity, we could get some results: the axial velocity distribution uniformity $\delta$ is the highest under the no spoiler case, the weighted average angles $\theta$ and $\theta'$ are not low either, therefore, its efficiency is high; $\delta$ of the 4 blades $0^\circ$ case is low, while $\theta$ and $\theta'$ are closest to $90^\circ$ in this case, therefore, its efficiency is also high; $\delta$ of the 4 blades $-15^\circ$ case is in a medium level, while $\theta$ and $\theta'$ are very close to $90^\circ$ which are just after the 4 blades $0^\circ$ case, therefore, its efficiency is high too; $\delta$ of the 4 blades $15^\circ$ case is low, and $\theta$ and $\theta'$ are also low in this case, therefore, its efficiency is obviously lower than the previous cases; $\delta$ of the 4 blades $30^\circ$ case is high which is just after the no spoiler case, while $\theta$ and $\theta'$ of this case are the lowest, so its efficiency is the lowest; $\delta$ of the 2 blades $+30^\circ$ & 2 blades $-30^\circ$ case is low, and $\theta$ and $\theta'$ are in a medium level, therefore, its efficiency is low too which is just higher than $\delta$ of the 4 blades $30^\circ$ case. In conclusion, $\delta$, $\theta$ and $\theta'$ all have obvious direct effects on the efficiency of the pump. In the range of $\delta$ of cases in this paper (the difference between the maximum value and the minimum value is 0.093), the effects of $\theta$ and $\theta'$ on efficiency are higher than $\delta$ obviously.

Figure 11. Comparison of $\delta$ under various cases.

Figure 12. Comparison of $\theta$ under various cases.
4. Conclusions
A PIV test system was adopted to investigate the influence of the uniformity of the inlet flow field on the performance of a nuclear main pump scale model, and the NUMECA software was used to numerically simulate the test cases. The conclusions can be drawn as follows:

1) When the rotation direction of the spoiler blade is opposite to that of the impeller, the greater the deviation of the spoiler blade from the axial angle, the more the pump efficiency and head drop.
2) Proper positive pre-rotation will not reduce the pump efficiency, but will increase the pump head.
3) For the same spoiler blade deflection angle, when its structure produces irregular flow field disturbances under non-centrosymmetric conditions, the flow field downstream of it will not produce clear swirling flow, and it will not have more impact on the performance of the pump.
4) The axial velocity distribution uniformity $\delta$ and the velocity weighted average angle $\theta$, $\theta'$ have obvious direct effects on efficiency, and in the $\delta$ interval of each working condition in this paper (the difference between the maximum value and the minimum value of $\delta$ is 0.093), the speed weighted average angle has a greater impact on efficiency.

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