Effect of Inlet Flow-Guide on Hydraulic Loss of Seawater MHD Propeller with Helical Channel

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Abstract. Inlet flow-guide is very important to the hydraulic loss of the whole channel of marine MHD thruster with helical channel. In this paper, variable helix is used as the line-type of flow-guide blade, and helical loop number (N), blade number (n) and blade height (H) (helix height) are established as the main geometric parameters. $L_{16}(4^3)$ orthogonal table was used to design the inlet flow-guide, which contains three factors with four levels of N, n and H. The whole flow filed of helical channel with inlet flow-guide was numerical simulated with Fluent. The results show that the hydraulic loss of the helical channel can be greatly reduced by the inlet flow-guide; The results of orthogonal experimental show that the hydraulic loss of the helical channel is sensitive to the line-type of the guide vane, and the total hydraulic loss increases with the increase of n, increases first and then decreases with the increase of N, and decreases first and then increases with the increase of H.

1. Introduction  
The pressure loss of the channel system is an important factor affecting the thrust force and efficiency of the propulsion performance. When the effective channel size and electromagnetic parameters are fixed, the smaller the hydraulic loss of the channel, the better the propulsion performance [1]. The inlet flow-guide of the helical channel prewhirls the axial inflow into the effective section of the electromagnetic force area. It is urgent to study the influence of the inlet flow-guide on hydraulic loss of seawater MHD propeller.

Zuowu Tan et al. used one-dimensional mathematical model to analyze and calculate the hydraulic characteristics of helical channels with different pitches, inner and outer electrode diameter ratios, and pointed out the role of inlet flow-guide and the necessity of its optimal design [1]. Yang Peng explored the hydraulic characteristics of helical channels with different pitches through cyclic loop experiments, but did not study the hydraulic characteristics of inlet flow-guide. Finally, Peng pointed out that the next step would be to optimize the design of inlet flow-guide by using 3D numerical calculation method [2]. Lingzhi Zhao first proposed a method for calculating the three-dimensional flow field and hydraulic loss of the helical channel separately. Four types of inlet flow-guide and rectifiers with different blade heights and blade numbers were used for comparative analysis. The results show that the flow-guide can greatly
reduce the hydraulic loss, but no further systematic analysis was made on the specific influencing factors of the flow-guide [3].

In this paper, computational fluid dynamics (CFD) technology is used to carry out orthogonal experiments on three factors affecting the hydraulic characteristics of the helical channel of the inlet flow-guide. Firstly, the overall change of the hydraulic characteristics of the helical channel caused by the inlet flow-guide is analyzed. Secondly, the influence of the factors of the inlet flow-guide on the hydraulic characteristics of the helical channel is analyzed, which provides useful guidance for the design of the inlet flow-guide.

2. Research method of hydraulic loss

The flow channel components of the helical channel are mainly composed of an inlet flow-guide, a helical blade (corresponding effective section) and an outlet rectifier, as shown in Figure 1. For pipeline fluid machinery, the hydraulic performance of each component can be studied separately from local concepts [5, 6]. In this paper, only the hydraulic characteristics of the inlet flow-guide are studied. In order to simplify the modeling and reduce the calculation, we cancel the use of the rectifier.

Due to the fact that electromagnetic field is added to the effective section of MHD thruster in actual operation, the effect of electromagnetic field will change the flow field of fluid. However, the study shows that the hydraulic loss of the propeller in actual work is equivalent to that of the propeller driven only by hydrostatic pressure at the same flow rate [3].

Therefore, the dimensionless hydraulic loss coefficient \( \beta \) can be defined to characterize the hydraulic loss when only hydrostatic pressure is driven.

\[
\beta = \frac{P_{\text{loss}}}{0.5 \rho u^2} = \frac{P_1 - P_2}{0.5 \rho u^2}
\]  

(1)

Where \( P_1, P_2 \) are the total pressure of the average mass of a section; \( \rho \) is the fluid density; \( u \) is the average velocity of the effective section.

3. Orthogonal experimental scheme

For the inlet flow-guide with variable helix, the parameters of variable helix are: helix pitch at the end of helix, loop number of helix, helix height (blade height). Other parameters of the flow-guide include the number of blades and the thickness of blades. Because the inlet flow-guide is directly connected with the helical blade of effective section, the helix pitch at the end of helix is the same as that helical blade of the effective section; The flow-guide blade thickness is fixed at 6 mm, and the size of the guide cap at the front of the flow-guide with different structural parameters is the same. Therefore, the helical loop number (N), blade number (n) and blade height (H) (helix height) are determined as the main structural parameters. In order to better explore the influence of flow-guide structure parameters on the hydraulic loss of helical channel, 16 flow-guide structures at 4 levels were selected for each factor. The
values of each factor and level are listed in Table 1 below. The orthogonal experimental scheme determined by L16 (4³) orthogonal table is shown in Table 2.

**Table 1. Factor and levels**

| level | factor |          |          |          |
|-------|--------|----------|----------|----------|
|       | A      | B        | C        |
| 1     | 0.1    | 1        | 0.1      |
| 2     | 0.2    | 2        | 0.2      |
| 3     | 0.3    | 3        | 0.3      |
| 4     | 0.4    | 4        | 0.4      |

**Table 2. Orthogonal experimental configuration scheme**

| experiment number | code | experiment number | code |
|-------------------|------|-------------------|------|
|                   | A    | B | C |        | A | B | C |
| 1                 | A₁   | B₁ | C₁ | 9      | A₃ | B₁ | C₃ |
| 2                 | A₁   | B₂ | C₂ | 10     | A₃ | B₂ | C₄ |
| 3                 | A₁   | B₃ | C₃ | 11     | A₃ | B₃ | C₁ |
| 4                 | A₁   | B₄ | C₄ | 12     | A₃ | B₄ | C₂ |
| 5                 | A₂   | B₁ | C₂ | 13     | A₄ | B₁ | C₄ |
| 6                 | A₂   | B₂ | C₁ | 14     | A₄ | B₂ | C₃ |
| 7                 | A₂   | B₃ | C₄ | 15     | A₄ | B₃ | C₂ |
| 8                 | A₂   | B₄ | C₃ | 16     | A₄ | B₄ | C₁ |

4. **Numerical calculation**

4.1. **Physical Model**

The basic calculation model is shown in Figure 2. The main structural parameters of the effective section are shown in Table 3. The effective section is 500 mm in length, and the maximum axial height of the flow-guide is not more than 470 mm when the flow-guide is placed in the flow-guide section.

![Figure 2. Basic computing physical model](image)

**Table 3. Basic parameters of helical blade of effective section**

| parameters                        | value |
|-----------------------------------|-------|
| Diameter of outer electrode $D_{ou}$ (mm) | 166   |
| Diameter of inner electrode $D_{in}$ (mm) | 43.6  |
| Climber angle $\theta$ (°)         | 24.17 |
| Pitch $t$ (mm)                     | 125   |
| Length of effective section $L_{es}$ (mm) | 500   |
| Thickness of blade (mm)            | 6     |

4.2. **Numerical scheme**

Fluent is used to calculate the three-dimensional flow field in the helical channel. The working fluid is simulated seawater with a density of 1024 kg/m³ and a dynamic viscosity of 0.00103 N.m/s². The inlet
velocity of the flow field is 0.9626 m/s (flow rate is 75 m$^3$/h), and the outlet condition is outflow. The k-ε standard model and the standard wall function are adopted. The meshing method and rules are the same as those in Zhao’s [3]. The correctness of the numerical method has been verified by experiments [3].

5. Result and discuss
Combining with Fig. 2 and Formula 1, the determination indexes of orthogonal experiments, i.e. the resistance coefficients, are determined as follows:

\[ \beta_1 = \frac{P_{\text{Plane}2} - P_{\text{Plane}3}}{0.5 \rho u^2} \]  

\[ \beta_2 = \frac{P_{\text{Plane}3} - P_{\text{Plane}4}}{0.5 \rho u^2} \]  

\[ \beta_3 = \beta_1 + \beta_2 \]  

Where $\beta_1$, $\beta_2$ and $\beta_3$ represent the flow-guide section, the effective section and the overall hydraulic loss coefficient, respectively.

If there is only friction loss along the effective section, that is, the flow-guide completely converts the axial inflow into helical flow, then the equivalent hydraulic loss coefficient of the effective section in figure 2 is as follows:

\[ \beta_{2,\text{ideal}} = \lambda \frac{L}{D_{mhd}} \]  

Where $L$ and $D_{mhd}$ are the length and equivalent hydraulic diameter of the effective section of the spiral channel after expansion, respectively [2]; $\lambda$ is frictional coefficient which can be obtained by looking up tables [4]. Therefore, we get $\beta_{2,\text{ideal}} = 0.6223$.

5.1. Calculation results without flow-guide
The hydraulic characteristics of the helical channel without a flow-guide are calculated by using Figure 2 as the calculation model. The typical flow field distribution is shown in Figure 3. The calculation results show that $\beta_{2,\text{no flow-guide}} = 1.32$. As can be seen from Fig. 3, when there is no flow-guide, the direct contact between the axial inflow and the helical blade of effective section leads to a relatively large local hydraulic loss. The comparison between $\beta_{2,\text{ideal}}$ and $\beta_{2,\text{no flow-guide}}$ also illustrates this problem.

![Flow field distribution without flow-guide](image)  

**Figure 3.** Flow field distribution without flow-guide
5.2. Results of orthogonal experiments

Table 4. Results of orthogonal experiments

| Experiment number | $\beta_1$ | $\beta_2$ | $\beta_3$ | Experiment number | $\beta_1$ | $\beta_2$ | $\beta_3$ |
|-------------------|-----------|-----------|-----------|-------------------|-----------|-----------|-----------|
| 1                 | 0.029     | 0.73      | 0.759     | 9                 | 0.03      | 0.86      | 0.89      |
| 2                 | 0.0344    | 0.7057    | 0.7401    | 10                | 0.05      | 0.88      | 0.93      |
| 3                 | 0.0697    | 0.682     | 0.7517    | 11                | 0.11      | 0.85      | 0.96      |
| 4                 | 0.077     | 0.629     | 0.706     | 12                | 0.07      | 0.82      | 0.89      |
| 5                 | 0.03      | 0.84      | 0.87      | 13                | 0.03      | 0.83      | 0.86      |
| 6                 | 0.067     | 0.886     | 0.953     | 14                | 0.053     | 0.873     | 0.926     |
| 7                 | 0.06      | 0.90      | 0.98      | 15                | 0.07      | 0.83      | 0.9       |
| 8                 | 0.075     | 0.882     | 0.957     | 16                | 0.15      | 0.842     | 0.992     |

5.2.1. Overall Characteristic Analysis. The hydraulic loss characteristics of different flow-guide structures in orthogonal experiments are shown in Fig. 5. The typical flow field distribution with code $A_4B_3C_2$ is shown in Fig. 4.

Comparing $\beta_2$ and $\beta_{2\text{-no flow-guide}}$, we can see that the hydraulic loss coefficient of the effective section with flow-guide is greatly reduced, because the flow-guide makes the axial inflow flow turn into rotational flow, avoiding direct impact of the inflow on the helical blade of the effective section. Comparing $\beta_2$ and $\beta_{2\text{-ideal}}$ we can see that the hydraulic loss coefficient of the effective section with flow-guide is still larger than that of the ideal helical flow. This is because the diverter can not completely convert the axial inflow into helical flow, and there is still a part of the fluid impacting the helical blade to produce hydraulic losses.

![a. Velocity distribution](image1)

![b. Static pressure distribution](image2)

Figure 4. Flow field distribution of combination code $A_4B_3C_2$

5.2.2. Analysis of orthogonal experimental results. The results of range analysis are shown in Table 5, in which $k_i$ is the average factor level and $R$ is the extreme difference. The variation of the average values of each factor in Table 5 is shown in Figure 6.

The larger the range, the greater the influence of the selected level on the index. As can be seen from Table 5, the order of influence on hydraulic loss in diversion section is $BCA$, which shows that $n$ has the greatest influence on hydraulic loss in flow-guide section, followed by $H$ and $N$. The order of influence on hydraulic loss in the effective section is $ABC$, which indicates that the hydraulic loss in the effective section is sensitive to the change of $N$. When considering the influence of the flow-guide on the hydraulic loss of the whole channel, it can be seen that the primary and secondary factors affecting the hydraulic loss are $N$ and $H$, which indicates that the number of helical coils and the height of blades have a greater impact on the overall hydraulic loss. $N$ and $H$ determine the helical shape of the guide vane, so the helical blade shape has a great influence on the overall hydraulic loss. It is necessary to select $N$ and $H$ reasonably to select the appropriate blade profile to reduce the overall hydraulic loss.

From Fig. 6, we can intuitively see the influence of different factors on different targets at different levels. It can be seen that $\beta_1$ increases with the increase of $n$ and $N$, and decreases with the increase of...
$H$, $\beta_2$ increases with the increase of $n$, increases first and then decreases with the increase of $N$, and decreases first and then increases with the increase of $H$. Because $\beta_1$ accounts for a small proportion, the variation law of $\beta_3$ is basically the same of $\beta_2$. The best combination is $A_1B_1C_2$.

Table 5. Range analysis of orthogonal experiments

|      | $\beta_1$ |      | $\beta_2$ |      | $\beta_3$ |
|------|-----------|------|-----------|------|-----------|
|      | $k_1$ | $k_2$ | $k_3$ | $k_4$ | $R$ |
| $A$  | 0.053 | 0.063 | 0.065 | 0.076 | 0.023 |
| $B$  | 0.03 | 0.051 | 0.062 | 0.093 | 0.063 |
| $C$  | 0.069 | 0.051 | 0.057 | 0.059 | 0.038 |

Figure 5. Results of orthogonal experiments  
Figure 6. Influencing trends of various factors

6. Conclusion

Through 3D numerical simulation, the influence of flow-guide on channel hydraulic loss characteristics is explored. Firstly, the hydraulic loss characteristics of channel with and without flow-guide are compared and analyzed. The results show that flow-guide can effectively reduce the hydraulic loss in the effective section, that is, flow-guide is necessary. By comparing the results of orthogonal calculation with $\beta_{2,\text{ideal}}$, it can be concluded that the flow-guide can not completely change the effective flow into ideal flow. In the design calculation, the actual effective section’s frictional coefficient is 1.3 times of ideal frictional coefficient. The range analysis of the orthogonal experimental results shows that $N$ and $H$ have the greatest influence on the hydraulic loss of the channel. $\beta_3$ increases with the increase of $n$, increases first and then decreases with the increase of $N$, and decreases first and then increases with the increase of $H$. The line-type of guide vane has a great influence on the hydraulic loss of the channel. In order to obtain better hydraulic characteristics of the channel, the next step is to explore and optimize the design of other guide vane line-types.

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