Research the Influence of Hydraulic Model on Vibration of Pump

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Abstract. In this paper, the cooling water pump of centrifugal equipment is taken as the research object in ship system. The pump set is normally open. And, it is also one of the main vibration sources of ship. On the premise of meeting the requirement of hydraulic performance parameters, several different hydraulic model prototypes of this pump set are manufactured. On the same test bench, the vibration tests of this pump show that the measured values are quite different. Fluid software is used to analyse these hydraulic models under the same boundary conditions. It is found that there are obvious differences in hydraulic fluctuations and fluid vortices among impeller passage, vortex chamber, inlet and outlet passages. In order to verify the inference, vibration test and analysis are carried out on the whole pump set and the foots of installation. The corresponding relationship was found out between the fluid fluctuation of hydraulic components and the vibration characteristics. Based on the corresponding relationship, the current hydraulic model of pump is optimized and improved. The parameters of hydraulic model which can affect the vibration sensitivity of pump are optimized. Finally, the experimental results prove that this method can reduce the vibration effectively.

1. Introduction
Pump set is a widespread general machinery. Where there is liquid flow, there is always a pump at work. With the development of science and technology, it is no longer satisfied with the hydraulic performance of pump. The research on vibration suppression of pump is emphasized gradually. With the increasing requirement of vibration and noise, it is more and more strict in ship equipment system [1-2]. The pump unit researched in this paper belongs to the normally open equipment and is also one of the main vibration sources in the ship system.

At present, the hydraulic components of this pump in China are still using widely which were designed from 1970s to 1990s. In order to adapt to the low vibration performance of ships, the technology of vibration suppression is improved. It is urgent to find out the corresponding rules between different hydraulic models and vibration characteristics of pump set. And the targeted approach is adopted to suppress vibration. In view of such problems, the pulsation and vortex of hydraulic components were analyzed in this paper, such as impeller passage, vortex chamber, inlet and outlet passages. The corresponding relationship with the vibration of this pump set was found out. So, the vibration value of the pump set could be reduced effectively.

2. Hydraulic model calculation
2.1. Theoretical basis of hydraulic modelling
Mathematical models were established that could reflect the nature of engineering and physical problems. Specifically, the differential equation is necessary to be established which can reflect the relationship between variate and corresponding definite solution conditions in the problem. And this is first step for numerical simulation. The basic governing equations of fluid include mass conservation equation, momentum conservation equation, energy conservation equation, and the corresponding definite solution conditions of these equations [3].

According to the hydraulic characteristics of flow field, it can be regarded as an incompressible viscous flow in the pump. The ideal governing equation of conservation type is as follow:
• Mass conservation equation
\[
\frac{\partial \rho}{\partial t} + \text{div}(\rho \vec{u}) = 0
\]  
\[ (1) \]
• Momentum conservation equations which were also known as Naiver-Stokes equations or equations of motion
\[
\frac{\partial (\rho \vec{u})}{\partial t} + \text{div}(\rho \vec{u} \vec{u}) = \text{div}(\mu \text{ grad } u) - \frac{\partial p}{\partial x} + S_x \\
\frac{\partial \rho v}{\partial t} + \text{div}(\rho \vec{v} \vec{u}) = \text{div}(\mu \text{ grad } v) - \frac{\partial p}{\partial y} + S_y \\
\frac{\partial \rho w}{\partial t} + \text{div}(\rho \vec{w} \vec{u}) = \text{div}(\mu \text{ grad } w) - \frac{\partial p}{\partial z} + S_z
\]  
\[ (2) \]
• Energy conservation equation
\[
\frac{\partial (\rho T)}{\partial t} + \text{div}(\rho \vec{u} T) = \text{div}(k \text{ grad } T) + S_r 
\]  
\[ (3) \]

2.2. Design of hydraulic component
The hydraulic performance was calculated and analyzed based on the 3D models of hydraulic components. According to the analysis results, the hydraulic components were optimized. And the parameters which are closely related to the hydraulic performance were adjusted for optimization analysis [4]. The hydraulic components with low vibration were developed and verified by tests. Thus, an excellent hydraulic model would be obtained. And the vibration amplitude could be reduced from the vibration excitation source.

2.2.1. Design of inlet and outlet channel
The axial suction and radial discharge were adopted in the inlet and outlet channel. The suction of straight conical tube was adopted in the inlet channel. And the structure of single vortex chamber was adopted in outlet channel. The diameter of inlet and outlet pipe was adjusted to match the interface of system. The three-dimensional model of the pump set is shown as below.

![Three-dimensional Model of the Pump Unit](image)

2.2.2. Selection of the impeller
According to the performance of equipment cooling pump: Q=80m³/h, H=45m, n=2950r/min. It is concluded that ns=82.2. The hydraulic model with similar ns and high efficiency was selected in hydraulic model library. Similar conversion was carried out to ensure that the hydraulic performance of
the pump meets the requirements. In order to reduce the vibration caused by hydraulic pulsation, different number of blades (six and seven) were considered.

2.2.3. Design of vortex chamber
From the perspective of vibration, the vortex chamber of different base circles was designed. In order to optimize the appropriate diameter of base circle, the ratio which is the base circle diameter of vortex chamber to the outside diameter of impeller (D3/D2) was changed. And the improvement of the vortex chamber was analysed.

3. Inlet and outlet channel

3.1. Boundary conditions
The first step is to set the boundary conditions. These parameters need to be defined here such as import boundary, export boundary and wall boundary.

3.1.1. Inlet boundary
The inlet boundary is located at the inlet pipe. According to the rated flow of hydraulic parameters, it was set as the velocity inlet boundary. The turbulent kinetic energy and the dissipation rate are calculated according to the actual working parameters [5]. The formula is as follows:

\[
k = \frac{3}{2}(v \cdot I)^2
\]

\[
\varepsilon = C_u \frac{k^{3/2}}{l}
\]

Where \( I = 0.16 \Re^{(-1/8)} \), and \( I \) is turbulence intensity; \( \Re = \nu D \rho / \mu \), \( \Re \) is Reynolds number, \( \nu \) is fluid velocity, \( D \) is hydraulic diameter, \( \rho \) is fluid density, \( \mu \) is fluid viscosity; \( l \) is turbulence length scale, \( l = 0.07D \); \( C_u = 0.0845 \).

3.1.2. Outlet boundary
The outlet boundary is located at the end of the outlet pipe. Considering the length of the outlet pipe, the exit condition is set to free outflow.

3.1.3. Wall boundary
The boundary conditions of the wall surface are determined by the properties of the fluid moving in it. In this paper, the non-slip wall condition was selected. All the wall surfaces that move with the impeller were set as moving wall surfaces in the impeller area. And the rotation direction and speed were consistent with the impeller. Such as vortex chamber, inlet and outlet areas, these surfaces were set as static wall which do not change with the rotation of impeller.

3.2. Computational domain setting for hydraulic analysis
Through the analysis of previous data, the unsteady flow structure was the main factor that induces the vibration and noise in the centrifugal pump. The main factors included flow separation on blade surface, vortex, rotating stall, cavitation, dynamic and static interference of impeller-tongue, etc [6]. The fluid analysis on the hydraulic components was carried out. And the hydraulic model of the impeller was established accurately, as shown in figure 2.

In order to determine the influence of inlet and outlet pipelines on the model stress, the pressure chamber was combined with the original impeller, straight cone pipe suction chamber and discharge pipe respectively. In addition, it provided relatively accurate models such as velocity field and pressure field for mechanical research and even vibration analysis. Radial force was analyzed on these four kinds of computational domain grids.
Through the above four computational domains, the hydraulic components that affect the radial force and the changing factors were obtained. When the impeller was calculated separately in the computational domain 1st, there is basically no radial force. When the inlet pressure changes in computational domain 2nd consisted of impeller and pressure-chamber, the radial force was basically unchanged. When the computational domain connects with the inlet and outlet pipelines, the radial force decreased in last two computational domains. From the calculation results of the third and fourth domains, it could be seen that the position of the elbow at the inlet turning has little effect on the radial force.

3.3. Vibration test verification
The vibration acceleration level of the test environment is no more than 80dB, which is lower than 10dB of the device operation. It is considered that the test environment is reasonable. And the vibration test data need not to be modified. This type of vertical water pump unit was tested by acceleration level method.

By adjusting the condition of the turning elbow of the inlet pipe, the vibration test was carried out. It showed that the test results of the last two computational domains were almost the same, which were 121.8db and 122.1db respectively, as shown in figure 3 below. It is proved that the position of the elbow in the inlet has little influence on the radial force and the vibration.

4. Impeller passage
4.1. Force analysis of impeller
There are two kinds of fluid excitation forces on the impeller: One is the steady-state force that causes the eccentricity of the impeller-rotor system. And the other is the transient force that causes the vibration [7-9]. In this paper, the direct integration method was used to calculate the force on the impeller. And the pressure and viscous force on the impeller were integrated respectively by Fluent software. The boundary of the impeller region was selected in software. And the pressure and viscous force were calculated in three directions of X, Y and Z respectively. The resultant force was obtained by adding the two forces together. Then the frequency response function (FRF) of the resultant force on the impeller was obtained by FFT method.

The force of the impeller was analysed. When the transient calculation results converged, the calculation was carried out again. One set of results was recorded for each three degrees rotation of
impeller. And the results of impeller rotating were recorded in two entire cycles. The forces on impeller were exported in the three directions. The force condition was shown in figure 4 below.

**Figure 4. The Forces on Impeller in Three Directions**

While the impeller has rotated twice, the change of force in three directions was showed in figure 4. As can be seen from the figure 4, the impeller rotated once (i.e. 360°) as a cycle. The force on the impeller was basically a cyclical change after the stable operation of pump. After the curve of force & angle was obtained, it could be expressed in the frequency domain by FFT, as shown in figure 5 below.

**Figure 5. FRF of Impeller Force**

According to FRF analysis of pump, the vibration peak existed at the blade frequency. And the generation of blade frequency was related to the hydraulic components. In order to reduce the low frequency vibration from the vibration source, the hydraulic components of the pump set need to be optimized further.

### 4.2. Impeller fluid analysis

#### 4.2.1. Modelling analysis of hydraulic components

- Steady calculation, full multi-grid accelerated convergence.
- Impeller with full channel technology, rotor parts with the consolidation method.
- Turbulence model: S-A
- Inlet boundary conditions: total temperature, total pressure, speed direction.
- Exit boundary conditions: mass flow, initial pressure.
- The conditions of wall are no slip and adiabatic, impeller given design speed.
- Initial field: uniform field
- The volume loss and mechanical loss were not considered.

According to the design drawing, the grid models of six blades (scheme 1) and seven blades (scheme 2) were generated respectively, as shown in figure 6. The simulation results corresponding to the two hydraulic models of impeller were shown in table 1.

**Figure 6. Grid Structure of Impeller**
Table 1. Simulation Results of Two Schemes

| Scheme | Flow (m³/h) | Revolving speed (r/min) | Lift (m) | Efficiency (%) |
|--------|-------------|------------------------|----------|----------------|
| Scheme 1 | 80          | 2950                   | 45       | 80.1           |
| Scheme 2 | 80          | 2950                   | 49.6     | 83.7           |

4.2.2. Determination of simulation results
The inlet pressure of working point was set as 200000Pa. The calculated results of the two schemes were accurate and reliable through the analysis of y+ value and temperature field. Generally, the range of y+ values of most wall surfaces in the computational domain met the needs of S-A turbulence model as shown in figure 7. It could be found that the temperature changes very small in computational domain from figure 8. And it was consistent with the actual situation. So, the analysis result could be judged to be relatively true and accurate.

4.2.3. Fluid analysis results
The following figure shows that the distribution of flow line inside the impeller is generally uniform. Compared with the two schemes, the impeller with seven blades has reflux zones on the working face.

According to the results of fluid simulation, the hydraulic pulsation of scheme 1 is better than that of scheme 2.

4.2.4. Rotor balance mode for axial force
In order to compare and verify the influence on vibration, two balance methods were adopted. One was to balance the axial force of the rotor by setting front and rear seal rings and balance holes on impeller. The other was that the impeller does not need balance holes. In order to balance axial force, the external balance tube was installed which was connected to the high-pressure chamber behind the impeller and the inlet. The stress state of the rotor was improved. The residual axial force was afforded by the bearing of pump. The guide bearing of water-lubricated was arranged between the impeller and sealing device. As an auxiliary support, it suffered radial force of impeller and centrifugal force of rotor dynamic imbalance. The dynamic stability of rotor system was improved by reducing the cantilever length of...
rotor shaft in operation process. And it was conducive to improve the bearing in vibration and noise reduction.

4.3. Vibration measurement

According to the spectrum diagram of FRF obtained from tests, the prominent peak frequency might correspond to the blade frequency of hydraulic pulsation (blade frequency is axial frequency multiplied by blade number).

Compared the influence of scheme 1 (impeller of six blades) and scheme 2 (impeller of seven blades) on vibration of pump, it is found that the vibration value of scheme 1 is 120.7db, which is better than that of scheme 2 (122.8db). After changing the balance mode, the vibration value of scheme 1 is 116dB better than that of scheme 2 (118dB).

According to the results of vibration test, it could be concluded that the vibration of scheme 1 was better than that of scheme 2 in the same mode of balance holes. When balance holes were closed, the vibration results of both schemes were better than before. After changing balance mode, the vibration of plan 1 was still better than that of plan 2.

4.4. Section of this chapter

According to the results of fluid simulation, it could be obtained that the hydraulic pulsation of six blades was better than that of seven blades. Because the vortices were produced by impeller of seven blades. At the same time, the hydraulic pulsation was affected. And the vibration tests have the corresponding results with simulation. In the frequency range of 10Hz-320Hz, the vibration value of six-blades impeller was better than that of seven-blades impeller.

In addition, the external balance tube was used to connect the high-pressure chamber behind the impeller to the inlet without opening balance holes. From the perspective of theoretical analysis and vibration test verification, it was found that the vibration effect of impellers including six and seven blades was better than that of traditional impeller with balance holes. It could also be concluded that the balance hole produces certain disturbance to hydraulic pulsation. And it could affect hydraulic model on radiating vibration. At the same time, the improvement of balance mode on the impeller was better than the change of the number of blades in vibration reduction.

Therefore, the best choice was six-blades impeller in the comparison of these schemes. And the external balance tube was used to connect the high-pressure chamber behind the impeller with the inlet instead of the previous balance holes.

5. Design of vortex chamber

The vortex chamber with different base circle was designed. And the clearance between impeller and tongue was optimized to reach a better value. The hydraulic performance and hydraulic pulsation were analyzed to make the liquid flow smooth and uniform without vortex. The vibration value was decreased as much as possible from the excitation source caused by hydraulic flow [10-12].

In the past researching on vibration and noise reduction of centrifugal pump, it was proved that the vortex chamber of different base circle had influence on vibration. It means that the different clearance
value between impeller and tongue had different vibration value. Therefore, the appropriate base circle of the vortex chamber was researched to match with the impeller of six blades.

5.1. Calculation of hydraulic performance

Two kinds of base circle with the limit size of the vortex chamber were provided here. The scheme which the diameter of base circle was 208mm of the minimum limit size, could be defined as scheme 01. And the scheme which the diameter of the base circle was 222mm of the maximum limit size, could be defined as scheme 02. The vortex chambers of two base circles matched the same impeller of six blades were compared and analyzed. At first, the hydraulic performance of the two schemes was calculated.

Table 2. Analysis Results of Variable Working Conditions

| Scheme | Flow (m³/h) | Efficiency (%) | Lift (m) | Scheme | Flow (m³/h) | Efficiency (%) | Lift (m) |
|--------|------------|----------------|----------|--------|-------------|----------------|---------|
|        |            |                |          | Scheme 01 | 50          | 78.5           | 39.1    |
|        |            |                |          |        | 60          | 80.0           | 38.3    |
|        |            |                |          |        | 70          | 80.7           | 37.2    |
|        |            |                |          |        | 80          | 81.1           | 45.6    |
|        |            |                |          |        | 90          | 81.0           | 43.5    |
|        |            |                |          |        | 100         | 80.1           | 40.7    |
|        |            |                |          |        | 110         | 78.2           | 37.2    |
|        |            |                |          | Scheme 02 | 60          | 74.8           | 38.3    |
|        |            |                |          |        | 70          | 80.1           | 37.0    |
|        |            |                |          |        | 80          | 80.9           | 43.9    |
|        |            |                |          |        | 90          | 76.7           | 36.6    |

Through table 2, It could be found that the efficiency of scheme 01 is 81.1% as rated point. The head is 45.6m, and the efficient point is between 80-90m³/h. The efficiency of scheme 02 is 80.9% as rated points. The head is 43.9m, and the efficient point is around 80m³/h. There is no significant improvement in the efficiency, and the head is 1.7m lower in Scheme 02.

5.2. Fluid simulation

The vortex chamber of two basic circles matched the same impeller, were compared and analyzed [13]. Generally, both two flow diagrams of each scheme showed that the overall flow were well inside the pump, as shown in figure 12. In scheme 01, there exists a backflow area in impeller. In scheme 02, there is no obvious backflow area. And the streamline is basically consistent with the blade profile.

The velocity flow diagram of the central surface of the impeller was observed as shown in figure 12(a). The velocity flow from the center to the edge of impeller was relatively uniform. And there was only one section existing abnormal phenomena in the cavity. The detail of vortex was found in this area after magnifying it as shown in figure 13. When the vortex entered the subsequent pressure chamber, it would impact on the fluid and spread. Thus, it caused a large hydraulic pulsation.

The static pressure gradient line of radial distribution was analysed in vortex chamber of scheme 01. Under the influence of the impeller, periodic fluctuations occurred near the inner side. And the gradient line of static pressure formed an arc in the outlet cylinder. Compared with scheme 01, The range of static pressure gradient was small and the hydraulic pulsation is also small in scheme 02.
5.3. Vibration measurement
In the test, the effect of scheme 01 (base circle is 208mm) and scheme 02 (base circle is 222mm) on vibration were compared. It is found that the vibration value of scheme 02 is 110.7db, which is better than that of scheme 01, which is 113.3db.

5.4. Section of this chapter
By analysing the cloud image of velocity vector and its cross section in the vortex chamber, it could be clearly found that there were obvious velocity gradients in both horizontal and vertical directions near the tongue. The reason for this phenomenon was as follow. Caused by the interference between impeller and baffle tongue, the liquid which was accelerated by centrifugal force of impeller suffered local resistance at the tongue. And the tongue slows down the overall speed. While some of liquid was reaccelerated due to the different pressure between the two sides of baffle tongue. It flowed back to the vortex chamber in the left of baffle tongue and mixed with the high-speed fluid. Because of this part of the fluid backflow on the left side of baffle tongue, the high-speed area was affected. In the common influence area of impeller and baffle tongue, there would be a large alternating gradient of velocity and pressure. And the change of gradient was closely related to the rotation of impeller. This was the reason why the blade frequency of the hydraulic component was obvious in FRF curve [14-16].

From the fluid simulation results, it could be seen that the hydraulic components of this type of pump have such problems as vortex and backflow at the edge of impeller. Therefore, the flow field was not stable. And there also be certain pressure pulsation. This pulsation situation would be more complex after entering the vortex chamber. Based on analysis of pressure pulsation in flow field, it showed that the pressure pulsation was the main contributor to blade frequency peak in FRF.

On the finite element analysis platform, the fluid analysis which include matching different clearance between impeller and tongue was carried out. The structural parameters of hydraulic components with the smallest hydraulic pulsation were found out. So, the liquid flowed into the pump smoothly.

Based on the above, the analysis and calculation results of scheme 01 and scheme 02 were compared. Combined with the test results, scheme 02 (base circle diameter of 222mm) was selected as the base circle size of the new vortex chamber. That is to meet the requirements of hydraulic performance, but also to obtain better hydraulic pulse characteristics.

6. Hydraulic optimization and test

6.1. Calculation basis of hydraulic performance optimization
On the premise of maintaining the performance and efficiency of the pump unit, the hydraulic model was optimized. So, it was necessary to establish a mathematical model for the hydraulic model and performance.

The impeller model with similar specific speed and high efficiency was selected from the data basement of hydraulic model. The design parameters were substituted into formula, and the specific speed is as follows:

\[ n_s = \frac{3.65 \times n \times \sqrt[4]{Q}}{H^{3/4}} = 83.4 \]  \hspace{1cm} (6)

The similarity conversion is performed by the following formula.

\[ \frac{Q_s}{Q_n} = \lambda_Q^3 \times \frac{n_s}{n_n} \] \hspace{1cm} (7)
\[ \frac{H_s}{H_n} = \lambda_H^3 \left( \frac{n_s}{n_n} \right)^2 \] \hspace{1cm} (8)

From formula (5) and (6)

\[ \lambda_Q = \sqrt[3]{\frac{Q_s}{Q_n} \times \frac{n_s}{n_n}} = 0.85 \] \hspace{1cm} (9)
\[ \lambda_H = \sqrt[3]{\frac{H_s}{H_n} \times \frac{n_s}{n_n}} = 0.851 \] \hspace{1cm} (10)

According to \( \lambda = 0.852 \), all dimensions were converted. And the performance curve is shown in figure 16.

![Figure 16. Performance Curves](image)

According to the above analysis of the dynamic characteristics and flow field on this pump unit, reasonable optimization and improvement on the structure of fluid components have carried out. The purpose was to reduce the vibration and improve the efficiency. It included the local optimization and improvement of inlet angle, outlet angle and packing angle in blades [17].

Since optimized and improved impeller without opening balance hole, the vortex chamber of scheme 02 that basic circle is 222mm, was further optimized. So, the clearance between the impeller and the tongue could reached a better value. The liquid flow was stable and uniform without vortex. And the contribution of hydraulic pulsation to the vibration was decreased. So, the source of excitation was reduced greatly.

6.2. Test and verification
This new pump set which was optimized and improved was tested. And its vibration level of low-frequency was 108.5db (see figure 17). The influence of hydraulic pulsation on unit vibration was greatly reduced.
7. Conclusion
The range of these parameters was researched in the centrifugal pump, such as inlet form, impeller, tongue and vortex chamber. Under the premise of constant performance, the combination parameters of typical key components were optimized to obtain the minimum pulsating pressure.

The scheme of the new pump set was researched. The hydraulic model of the pump was optimized and improved according to the test and analysis results. This new hydraulic model has broken through the one used since the 1970s. Based on the method of vibration reduction, the design scheme has been successfully completed.

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