Preliminary Design Study of certain Computer Water-Cooling Pump

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Abstract. According to the producer’s requirement to the performance of computer water cooling pump, the structure of the pump is confirmed according to the basic theory of vane pump and fan. Based on the consideration of performance and efficiency, the basic parameters of pump are chosen rationally; and then, implicit function derivation and coordinate transformation are used to design and calculate the profile of impeller and volute of water pump. The anti-leakage device for front and rear disc of water pump is designed according to the experience to finish the theoretical design of water pump. The primary experiment shows that the performance of pump basically meets the design requirement but the parameters of relevant structure should be in further optimization.

1. Introduction
With the increasing requirements of the users for the computer performance, the CPU water-cooling heat dissipation has been gradually valued by the users due to its excellent dissipation effect. However, the existing circulating pump still has the problem of mismatching between the capacity and water-cooling heat dissipation system. The capacity of circulating pump corresponds to the increased motor energy consumption, while the small capacity fails to meet the heat dissipation requirements, causing damage to CPU. Therefore, it is necessary to study the computer water-cooling circulating pump.

The hydraulic design of traditional centrifugal pump mainly adopts the similar conversion algorithm and velocity coefficient method [1]. These two methods are mature in application, but they are not suitable for the micro pump design due to the limitations of the traditional hydraulic model library. The circulating pump was preliminarily designed based on actual calculation, so as to meet the needs of manufacturer.

2. Design of water-cooling pump

2.1. Basic parameter design of impeller

2.1.1. Basic design requirements.

| Parameter                        | Design requirements |
|----------------------------------|---------------------|
| Maximum flow $q_v$              | 500L/h              |
| Maximum revolution $n$          | 4000r/min           |
| Volute inlet/outlet diameter    | 8                   |
| Motor input power $P$           | 25W                 |
| Maximum span of Volute          | 50mm                |
The above parameters in design shall be as small as possible to maximize the pump lift.

2.1.2. Basic design parameter setting. The specific speed of the pump $n_s$ [2] can reflect the characteristics of similar pump, which can reflect the geometry of pump, and also be calculated based on the known design parameters $q_V, H$ and $n$. The model needed can be selected based on the specific speed calculated. The formula for specific speed is as follows:

$$n_s = \frac{3.65 n \sqrt{q_V H}}{H^4}$$  \hspace{1cm} (1)

According to the known parameters $q_V$ and $P$, the lift of infinite blades can be preliminarily calculated, $H_T \approx 18m$ so $n_s = 20$ and the pump is determined as a low specific-speed centrifugal pump.

The impeller of the low specific-speed centrifugal pump [2] is the cylindrical blade as shown in Figure 1. The dimension ratio $D_2/D_0 = 3$ and set $D_1/D_0 = 1$, and the closing lift is 1.1-1.3 times that of the working condition designed, and the lift increases with the decrease of flow with slow change.

Impeller diameter $D_2 = \frac{6000}{\pi n}$

![Figure 1. Impeller shape](image)

The blade outlet incidence angle $\beta_{2y_{\infty}} = 30^\circ$ and the number of blades $z = 5$ according to the experience [3] [4], so as to ensure no hump in the pump performance curve and stable operation.

The basic design parameters are shown in Table 2.

### Table 2. Basic design parameters.

| Parameter               | Parameter value | Parameter               | Parameter value |
|-------------------------|-----------------|-------------------------|-----------------|
| Outlet incidence angle  | $30^\circ$      | Diameter of impeller $D_2$ | 48 mm         |
| Number of blades $z$    | 5               | Blade width $b$         | 3 mm          |
| Impeller inlet diameter | 16mm            | Blade thickness $s$     | 1 mm          |
| Blade inlet diameter $D_1$ | 16mm            |                          |                |

2.1.3. Basic design parameter setting. The solution of optimal lift under the basic design requirements is the key to the pump design. After optimal lift solution, the impeller shape and performance curve of the pump can be determined based on the specific speed. The optimal lift is solved as follows[5]:

Theoretical lift
Theoretical lift in finite blades

\[ H_{T00} = \frac{p}{\rho g q_v} \]  \hspace{1cm} (2)

Lift

\[ H_T = KH_{T00} \]  \hspace{1cm} (3)

Effective power

\[ H = H_T \cdot \eta_h \]  \hspace{1cm} (4)

Shaft power

\[ P_e = P_{sh} \cdot \eta \]  \hspace{1cm} (5)

Slip factor

\[ K = \frac{1}{1 + \frac{2\pi}{3} \cdot \frac{1}{1 - \left(\frac{D_1}{D_2}\right)^2}} \]  \hspace{1cm} (7)

Overall efficiency

\[ \eta = \eta_V \cdot \eta_h \cdot \eta_m \]  \hspace{1cm} (8)

Volume efficiency

\[ \eta_V = \frac{1}{1 + 0.68n_x} \]  \hspace{1cm} (9)

Flow efficiency

\[ \eta_h = 1 + 0.0835 \frac{1}{\sqrt{n}} \quad \text{lg} \]  \hspace{1cm} (10)

Mechanical efficiency

\[ \eta_m = 1 - 0.07 \cdot \frac{1}{\left(\frac{n}{100}\right)^{\frac{1}{6}}} \]  \hspace{1cm} (11)

Formulas (1)–(11) are integrated, and the equation (12) about lift H for the flow q_v and speed n can be obtained.

\[
\frac{H}{1 + 0.0835 \frac{1}{\sqrt{n}} \cdot \left(\frac{\sqrt{q_v}}{n}\right)^{\frac{3}{2}}} = \frac{1}{1 + \frac{2\pi}{3} \cdot \frac{1}{1 - \left(\frac{D_1}{D_2}\right)^2}} \cdot \frac{1}{1 + \frac{\rho g q_v}{P_{sh}}} \cdot \frac{1}{1 + 0.68 \left(\frac{3.65n\sqrt{q_v}}{H^2}\right)^{\frac{1}{3}}} \left(1 + 0.0835 \frac{1}{\sqrt{n}} \cdot \left(\frac{\sqrt{q_v}}{n}\right)^{\frac{3}{2}}\right) \left(1 - 0.07 \cdot \frac{1}{\left(\frac{3.65n\sqrt{q_v}}{100H^2}\right)^{\frac{1}{3}}} \right)
\]  \hspace{1cm} (12)

Assuming that the flow rate q_v is constant, the values are assigned to the speed n within 3000 r/min ~ 5000 r/min. The lift obtained is small and can be ignored. Therefore, the speed of pump is determined n = 4000 r/min. This problem is simplified into a one-dimensional problem about the changes in lift H with the flow q_v.

Assign values to the flow q_v within (0, 500L/h) to solve the implicit function of the lift H, combined with the efficiency of pump. Select H = 8.19m and q_v = 300L/h as the design operating point. The calculation results of each parameter under the design operating point are shown in Table 3.
Table 3. Calculation results.

| Parameter                              | Parameter value | Parameter                | Parameter value |
|----------------------------------------|-----------------|--------------------------|-----------------|
| Theoretical lift $H_{T\infty}$         | 15.33 m         | Effective power $P_e$    | 12.52 W         |
| Theoretical lift in finite blades $H_T$| 10.42 m         | Shaft power $P_{sh}$     | 25 W            |
| Lift $H$                               | 8.19 m          | Overall efficiency $\eta$| 0.501           |
| Flow $q_v$                             | 300 L/h         |                          |                 |

It can be seen from the comparison in Table 1 that the pump structure and performance parameters designed meet the basic design requirements. Under the design operating point, the specific speed $n_s \approx 30$ proves that the pump model is selected correctly.

2.2. Blade profile design

The blade profile can be designed with equal-pitch spirals according to experience.

Establish the polar coordinate system and a function $\rho = ae^{k\theta}$

According to the empirical formula: $k = \frac{1}{\tan(90 - \beta_{2\infty})}$

Boundary conditions: $\theta = 0^\circ, \rho = \frac{D_b}{2}$

Blade profile: $\rho = 0.008e^{0.5774\theta}$

The diagram of blade profile is shown in Figure 2.

![Blade profile](image)

Figure 2. Diagram of blade profile

2.3. Basic parameter design of impeller

2.3.1. Volute initial angle. Set the initial angle $\gamma$ as shown in Figure 3[5]

![Volute initial angle](image)

Figure 3. Volute initial angle of circulating pump
Pump outlet flow rate: $v_0 = \frac{4q\nu}{\pi d^2} = 1.6572\text{m/s}$

According to the empirical formula, the laryngeal flow rate: $v' = \frac{q\nu(2\pi - \frac{\pi y}{180})}{2\pi d\left[0.0225\cos\left(\frac{\pi y}{180}\right) + d - 0.0215\right]}$

According to the calculation, when the initial angle is 21 degrees, the laryngeal flow rate is 1.65 m/s, closest to the outlet flow rate. At this moment, the flow loss is the lowest, so the initial angle is 21 degrees[6].

2.3.2. Volute profile design. The log spiral can be used in the volute profile design [5].

Set the volute profile formula under polar coordinates as $R = be^{c(j - 21)}$

Boundary conditions: $R = R_1 = 0.025\text{m}$ when $j = 21^\circ$

Maximum opening: $R = \frac{D_2}{2} = A = 0.0313\text{m}$ when $j = 360^\circ$

Volute profile obtained: $R = 0.025e^{0.000668(j - 21)}$ $j \in (21, 36)$

2.4. General drawing of circulating pump structure design

According to the above steps, the structural diagram of circulation pump is obtained as shown in Figure 4.

![Figure 4. Structural diagram of circulation pump](image)

Figure 4. Structural diagram of circulation pump

Considering the leakage of pump, the leak-proof design is realized in both front and rear discs. The structure design of front and rear discs is shown in Figure 5, and the related structural parameters are shown in Table 4.

![Image of front and rear discs and leakage prevention](image)

(a) Front and rear discs  (b) Leakage prevention in front disc
3. Conclusion

According to the actual experience combined with efficiency, the design parameters are selected reasonably, and the profile functions of blade and volute are obtained using the polar coordinate system, so a centrifugal pump meeting the manufacturer’s requirements is designed preliminarily. However, the numerical simulation study was not performed for the design results due to the limitation of time and other objective conditions. At the same time, further research remains to be done about the optimization of pump.

References

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