Static characteristic analysis on a hydrodynamic bearing of a hydraulic suspended micropump

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Abstract. A hydraulic suspended micropump was designed based on the conception of double suction impeller. To study the running performance of the micropump, three dimensional simulation of the static characteristic of the hydrodynamic bearing at rated condition was performed. Parameters such as the eccentricity and rotational speed, having effect on the bearing capacity, were analyzed. Results show that the eccentricity and rotational speed have a certain effect on the bearing capacity. The region of the maximum static pressure and the positive pressure changes as the eccentricity increases. The bearing capacity improves as the rotational speed increases. All the results can provide a academic basis for the improvement and application of a hydrodynamic bearing of a micropump.

1. Introduction

Blood pump is a medical device for blood transportation. It can partly substitute the function of human organs to assist blood circulation and is the main medical treatment of organ failure. At present, most pumps use mechanical bearings as a shaft support, which not only cause great frictional loss but also easily break down. Besides, the heat generated during the runtime may lead to clotting and hemolysis. Hence, more and more people in recent years start to research on blood pumps using maglev or fluid dynamic as a shaft support. The fluid dynamic suspended blood pump has the advantages of low noise, less friction loss, and long serving life. Relative to the axial pump, centrifugal pump is less likely to cause the hemolysis because of the low rotary speed. However, further improvements are needed for this fluid dynamic bearing to be used in a blood pump. A shaftless miniature turbopump with a 40-mm outer impeller diameter was developed by Luo et al. [5-8]. The pump rotor was suspended by a hybrid bearing. The double suction design gave much better inflow upstream of the pump impeller with less mechanical damage of the blood which can lead to coagulation and hemolysis if the pump was used for blood transport.

In this paper, a hydrodynamic bearing of a hydraulic suspended micropump was designed. Numerical simulation of the static characteristics with radial support was performed by computational fluid dynamics (CFD) methods. The effect of different design parameters on the pressure distribution and the bearing capacity were analyzed.

2. Structural design
2.1. Pump design
As shown in Figure 1, the structure of the blood pump consists of two parts: the drive components and the flow components. The drive components contain a controller and a motor made up by armature windings, silicon steel. Brushless DC permanent magnet type motor used and the rotor position sensor is installed, via a dedicated controller to adjust the motor speed. The liquidity components including the impeller, pump casing and the outlet pipe.

![Figure 1. Schematic diagram of the miniature pump.](image)

1, Volute casing; 2, Pump cover; 3, “O” ring; 4, Screw; 5, Sealing seat; 6, Rotor; 7, Impeller; 8, Permanent magnetic; 9, Stator winding; 10, Silicon steel

There is a gap of liquid film between the impeller and volute casing which can be used as the pressure film to support and lubricate the impeller rotor. As the liquid flows through the impeller, the pressure in the outlet is higher than it in the inlet, so in the gap, the liquid flows from the outlet to the inlet, which can take away the heat generated by the rotation and prevent the flow dead zone in the channel. This program can effectively prevent clotting phenomenon resulting from long stagnation and high temperature. Meanwhile hemolysis cause by mechanical bearing can be avoided by using the fluid dynamic suspended supporting system.

Considering the physiological compatibility of the material, the whole blood pump is made up by Gr.Ti-6AL-4V(ASTM F-136). It has a 40-mm outer impeller diameter and weighs 420g. The geometrical parameters for the miniature pump are listed in Table 1. The pump inlet and outlet diameters are the same, 10 mm, for better interchangeability of the connecting tubes. The impeller exit diameter is 24 mm and the diameter at the impeller vane leading edge is 10 mm.

| Parameter                   | Symbol | Value |
|-----------------------------|--------|-------|
| Impeller exit diameter      | \(D_e\) (mm) | 24    |
| Impeller inlet diameter     | \(D_i\) (mm) | 10    |
| Pump inlet pipe diameter    | \(D_i\) (mm) | 10    |
| Pump outlet pipe diameter   | \(D_e\) (mm) | 10    |
| Trailing edge vane width    | \(b_2\) (mm) | 3     |
| Leading edge vane width     | \(b_1\) (mm) | 5     |
| Trailing edge vane angle    | \(\beta_2\) (°) | 29    |
| Leading edge vane angle     | \(\beta_1\) (°) | 22    |
| Number of vanes             | \(Z\) | 4     |
| Volute casing base diameter | \(D_3\) (mm) | 25    |
| Volute casing section       |          | rectangular |

2.2. Fluid dynamic bearing design
A pair of fluid dynamic bearings is used to make the design more compact. Figure 2 shows half of the fluid dynamic bearing. The radial bearing is formed by the rotor outer surface as the journal and the inner surface of the pump cover as the bearing bushing. If the pump is horizontal, there is only radial
force acting on the rotor because the pump is balanced in the axial direction due to the symmetrical inflow from the two pump inlets. The main geometrical parameters for the fluid dynamic bearing are listed in Table 2.

| Parameter                                                | Symbol | Value |
|----------------------------------------------------------|--------|-------|
| Rotor diameter                                           | $D_2$ (mm) | 24    |
| Average clearance between the bushing and the journal    | $c$ (μm)  | 100   |
| Ratio of width to diameter                                | $k$    | 0.5   |
| Minimum liquid film depth for the radial bearing          | $h_{\text{min}}$ (μm) | 65    |
| Thrust bearing bushing width                              | $d$ (mm) | 7     |
| Number of bearing pads in the thrust bearing              | $B$    | 6     |
| Minimum liquid film depth in the thrust bearing           | $h_{\text{max}}$ (μm) | 24    |

3. Calculation method
The computational model of the hydrodynamic bearing is shown in Fig.3(a). Figure 3(b) shows the local mesh of the computational domain. In order to calculate the pressure distribution and the bearing capacity accurately, the scale of the mesh at the thickness direction is set by 10μm. Structured mesh are used for the calculation, and the total number of nodes is 762 880.

![Computational model](image1)

![Local mesh](image2)

Boundary conditions were shown as follows,
1) Total pressure was set at the inlet. The total pressure was specified according to the calculation result of the whole pump including the hydrodynamic bearing with less mesh. The total pressure at the inlet was -1990 Pa when the flow rate of the pump is 5L/min.
2) Static pressure was used at the outlet. The static pressure at the inlet was -10243 Pa when the flow rate of the pump is 5L/min.
3) All the solid walls were calculated using no slip boundary conditions. The blade wall in the impeller rotated as the same speed as the impeller.
4. Results and discussion

For qualitative analysis of the liquid film, static pressures on three lines alone the inner surface of the casing from the inlet to the outlet of the hydrodynamic bearing are analyzed. The position of three lines is shown in Fig.4(a). Is denotes the dimensionless scale along axial direction. Figure 4 (b) shows the profile of the rotor and stator. The impeller rotates at counter-clockwise, and the initial position is at X axial direction which is marked as 0°.

4.1. The effect of eccentricity on the bearing capacity

Figure 5 shows the distributions of static pressure on the liquid film when the flow rate \( Q = 5L/\text{min} \), rotational speed is \( n = 5000\, \text{r/min} \) and the angle \( \phi = 30^\circ \). \( \varepsilon \) is the eccentricity. \( \theta \) denotes the circumferential direction, \( h \) denotes the height of the liquid film. Static pressures on four kinds of eccentricities are investigated. Three dimensional distributions of static pressure on the liquid film are shown in Fig.6.

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**Figure 4.** Position of three lines and the profile of amplified clearance

(a) Position of three lines (b) Profile of the rotor and stator

**Figure 5.** Distributions of static pressure
When $\varepsilon=0.8$, the maximum static pressure $p_s$ on the liquid film occurs before the appearance of minimum clearance. The maximum static pressure is about 11kPa. The minimum static pressure occurs after the appearance of minimum clearance. The maximum static pressure locates at the inlet along the axial direction, and the minimum static pressure locates at the outlet. At both side of the minimum clearance, there have a high pressure region and a low pressure region. The high pressure region is the drive source for supporting the rotor.

When $\varepsilon=0.6$, the maximum static pressure decreases compared with the result of $\varepsilon=0.8$. The bearing capacity is reduced due to the static pressure at the inlet has a uniform distribution.

When $\varepsilon=0.4$ and $\varepsilon=0.2$, no peak value of positive pressure can be found on the liquid film, which is to say that the bearing capacity is small. Only a small effort will make the liquid film shake, affecting the stable operation of the system. This may lead to friction between the rotor and the stator, even destroy the blood cell.

It can be summarized that the eccentricity has great effect on the distribution of the static pressure. The increase of the eccentricity will change the maximum static pressure and the region of positive pressure. The uniform clearance changes as the eccentricity changes, which affects the convergence of the clearance of the liquid film, so the peak pressure and the region of positive pressure changes.

The bearing force ($F$) of the liquid film can be calculated by the integral of the static pressure, then the relationship between $F$ and $\varepsilon$ can be obtained, which is shown in Fig. 7. When $\varepsilon=0.6$, the bearing force produced by the hydrodynamic liquid film can reaches 5.4N. The bearing force increases as $\varepsilon$ increases. Small value of $\varepsilon$ can maintain the running of the impeller.

**Figure 6.** Three dimensional distributions of static pressure

![Figure 6](image)
4.2. The effect of rotational speed on the bearing capacity

According to the fluid lubricate theory, the bearing capacity of the hydrodynamic liquid film has a relationship with the relative velocity at the rotor surface.

When $Q=5L/min, \varphi=30^\circ, \varepsilon=0.8$, distributions of static pressures on the liquid film with different rotational speed ($n=6000r/min, 5500r/min, 5000r/min, 4500r/min, 4000r/min$) are shown in Fig.8. At the inlet of the hydrodynamic bearing, the maximum static pressure increases as the rotational speed increase, while the minimum static pressure doesn’t change obviously. At the outlet, in the region of the perigon which is $50^\circ \sim 280^\circ$ has the same distribution of static pressure. The maximum static pressure become larger and the minimum pressure become smaller as the rotational speed increases, leading to the difference between the high pressure region and low pressure region.
When Q=5L/min, φ=30°, ε=0.8. The relationships between the bearing force and the rotational speed with different eccentricities are shown in Fig.9. At the same eccentricity, the bearing force increases as the rotational speed increases. The larger the eccentricity is, the change of amplitude of bearing force will increase as the eccentricity increases. When ε=0.8, n=6000r/min, the bearing force can reach 8.5N, which is 2.4 times as the result of n=4000r/min. During the hydraulic design of a micropump with high rotational speed, the clearance of the liquid film can be enlarged at the certain ratio.

![Figure 9. The relationship between the bearing force and the rotational speed](image)

5. Concluding remarks

   (1) The maximum static pressure occurs before the appearance of minimum clearance, while the minimum static pressure occurs after the minimum clearance. A high pressure region accompanied with a low pressure region at each side of the minimum clearance. The high pressure region is the drive source to form the bearing force.

   (2) Eccentricity has great effect on the bearing capacity. The maximum static pressure, the region of positive pressure and the bearing force increases as the eccentricity increase.

   (3) At the same eccentricity, the bearing capacity improved as the rotational speed increase. The clearance of the liquid film can be enlarged at the certain ratio during the optimal design for a high rotating micropump.

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