A novel approach to investigate temperature field evolution of water lubricated stern bearings (WLSBs) under hydrodynamic lubrication

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Abstract
Water lubricated stern bearings (WLSBs) are the critical component of ship propulsion system and have important effect on navigation safety. Operating temperature plays a main role on the performance of WLSBs. This paper aims to investigate the effect of operating conditions on bearing temperature characteristics under hydrodynamic lubrication. A novel CFD simulation method developed to improve calculation accuracy. Finite difference method was used to decrease the error of geometric modeling while the experiment and experimental correction formula were exploited to obtain improved boundary conditions. Based on the new method, the effects of operating conditions on temperature characteristics for two typical WLSBs were studied, and mechanisms of bearing temperature field evolution were discussed. Results show that the max friction coefficient appears when bearings are in low velocity and low load condition. Total heat flux density is a function of linear velocity, pressure and friction coefficient. Max temperature of bearing at 0.4 MPa decrease along with increased velocity, while decrease first and then increase at 0.2 MPa. Moreover, peak temperature appears at eccentric side and beyond minimum water film thickness position about 4–40\textdegree. High temperature area mainly located at the position of 80–140\textdegree in circumferential direction and 0.2–0.13 m in axial direction. With the increase of inlet water velocity, the max temperature of bearing changes slightly. It is appropriate to set the inlet velocity at 2 m/s to obtain better cooling performance. This work can provide theoretical basis for the operation monitoring of WLSBs and the development of new materials.

Keywords
Water lubricated stern bearing, temperature characteristics, simulation

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Introduction
The interests in environment-friendly technologies have stimulated by the strict regulations and the enhancement of environmental awareness, particularly in oil-lubricated bearings replaced with water lubricated ones. Water lubricated bearings have been widely used in hydroelectric turbine,\textsuperscript{1} nuclear main pump,\textsuperscript{2} machine tool\textsuperscript{3} and other industrial equipment with the advantages of low cost, simple structure, environmental friendly and satisfied tribological properties. It is also popular in ship propulsion system due to its good adaptability to shaft misalignment caused by...
manufacturing error, improper installation and hull deformation. Polymer materials are the typical bush materials used in WLSBs due to wear resistance, vibration damping and anti-corrosion properties. To improve the properties of WLSBs, some scholars carried out a number of distinguished researches in polymer materials. However, most of the current studies launched at room temperature without considering the influence of temperature variation. Researches show that the minimum water film thickness of bearing will decrease with the increase of temperature, so do the hardness and elastic module of bush materials. However, the wear rate, friction coefficient, vibration and noise will increase along with the temperature. 

Researches indicated that the average friction coefficient of UHMWPE increases by 65.96% under 720h, 80°C aging condition compared with the original sample. Moreover, most bush materials of WLSBs are semi-crystalline polymer, and their operating life will decrease exponentially with the increase of temperature. The accelerated aging under high temperature can cause a fracture of molecular chains and chemical oxidation reaction that correlate with the deterioration of materials. The water intake of bearings under high operation temperature should also be considered. Luca found that the weight of UHMWPE composites continued to increase after 112 days of immersion into 80°C water, while the samples were approximately saturated since the seventh exposure day into room temperature water. The excessive water swelling of bush materials can decrease the running clearance, leading to the lock between bush and shaft. Some researches considering temperature variation have conducted but the temperature range under study is diversify. The working conditions have a significant effect on the bearing temperature characteristics. Studies show that the surrounding water temperature increment of WLSBs at 0.4 MPa can reach 50°C after an hour operation while it is less than 20°C at 0.2 Mpa. In a word, it is essential to study the temperature characteristics of WLSBs under different operation conditions.

The full circumferential temperature distribution usually gained by numerical calculation which can be achieved by two methods, one is to solve Reynolds equation and energy equation, and the other is to solve N–S equation and energy equation directly. The former is only applicable to laminar flow of lubricants and ignores the effect of inertial force, volume force and heat conduction in the direction of film thickness. For WLSBs, the bearing clearance is larger than that of traditional bearings. Besides, a number of water grooves usually arranged on the lining. These issues make the cooling water in turbulent state and the effect of inertial force could not be ignored. Therefore, the latter is more suitable to study the temperature characteristics of WLSBs. The commercial software FLUENT is widely used in fluid flow simulation, which can obtain the three-dimensional temperature distribution of water film and is more accurate than the solution of Reynold equation and energy equation simultaneously.

It is very important to set boundary conditions during the calculation. However, some cases simplified the rotating shaft as heat source with constant temperature, which ignored the heat taken away by the lubricants. There are also some scholars simplified the shaft surface as adiabatic surface with all the heat are absorbed by lubrication film. In fact, both the heat conduction and heat convection should not overlooked because the shaft sleeve of WLSBs is usually made of copper alloy with a high thermal conductivity and the cooling water may be in turbulent state. Besides, geometry modeling is essential to the calculation, but it is difficult to obtain the geometry parameters of water film. In conclusion, it is crucial to develop a valid method to set boundary conditions and establish water film model.

For safety and reliability reasons, WLSBs usually designed in hydrodynamic lubrication. It should admitted that the excessive friction heat occurred due to the severe asperity contact under some extreme conditions. Nevertheless, these situations are not the ordinary state and the excessive friction heat will take away quickly when the hydrodynamic lubricate film formed. In this study, a novel approach proposed to investigate temperature field evolution of WLSBs under hydrodynamic lubrication. Two common applied WLSBs, Thordon XL, and SF-1, were used in this research. The structure diagram of whole research work can see in Figure 1. Firstly, the friction coefficient obtained on the test bench, and Gnielinski formula modified to calculate the heat transfer resistance of cooling water. Besides, the distribution model of bearing heat flux was established. Based on the experiment results and mathematical model, the effect of operation conditions on the heat flux density can be evaluated. In addition to this, Reynold equation, solved by finite difference method (FDM), was used to establish the geometry models of water film. Finally, the influence of operating conditions on the bearing temperature characteristics can be investigated. This research can provide theoretical basis for operation monitoring of bearings and development of new bush materials.

**Experiments**

**Test bearing and shaft**

WLSBs are quite different from traditional journal bearings, especially its structure and lining material. The sleeve material of two test bearings is brass and the lining is SF-1 and Thordon XL respectively. Eight grooves arranged in the lining to improve the cooling
and lubricated performance. The test shaft paired to the bearing made with 45 steel, and the shaft journal covered by ZQSn10-2 sleeve, as shown in Figure 2(a). Test bearing schematic and parameters can see in Figure 2(b), Tables 1 and 2.

Experimental apparatus and procedures
Experiments conducted on the marine stern bearing device SSB-100, which designed by Wuhan University of Technology as shown in Figure 2(c). During test, middle journal loading method is conducted to ensure the load evenly exerted on the test bearing and the force sensor is applied to indicate the loading force $N$ of hydro cylinder. Cooling unit can adjust the flow of tap water to ensure good lubrication and cooling. All the parameters were transmitted to a computer for display and further analysis by the multichannel analyzer. Finally, friction coefficient $f$ and nominal pressure $p_b$ can be calculated by equation (1)

$$
\begin{align*}
    p_b &= \frac{N - W_b}{D L} \\
    f &= \frac{F_f}{N - W_b} \\
    F_f &= \frac{M - M_0}{D/2}
\end{align*}
$$

Table 1. Material properties.

| Material     | Tensile strength (MPa) | Elastic modulus (MPa) | Density (g/cm$^3$) | Heat conductivity W/(m·K) |
|--------------|------------------------|-----------------------|--------------------|--------------------------|
| SF-1         | 67                     | 8080                  | 1.4                | –                        |
| Thordon XL   | 35                     | 850                   | 1.21               | –                        |
| ZQSn10-2     | 240                    | 108,000               | 8.83               | 100.8                    |

Table 2. Structure parameters.

| Bearing number | Lining material | Lining thickness (mm) | Sleeve thickness (mm) | Inner diameter (mm) | Running clearance (mm) | Bearing length (mm) | Groove number | Groove width (mm) | Groove depth (mm) |
|----------------|-----------------|----------------------|-----------------------|---------------------|-----------------------|--------------------|---------------|------------------|------------------|
| 1              | SF-1            | 18.38                | 26                    | 153.24              | 1.09                  | 150                | 8             | 8                | 6                |
| 2              | Thordon XL      | 18.45                | 26                    | 153.06              | 0.98                  | 150                | 8             | 8                | 6                |
Further, the friction coefficient $f$ can be expressed as equation (2)

$$f = \frac{2(M - M_0)}{p_0L D^2}$$

In order to keep dimensional stability, all the test bearings submerged into water for two months before experiment. The room temperature is 301 K constantly and the cooling water properties shown in Table 3.

Experimental procedure was as follow: First, the applied nominal pressure set to 0.2 MPa and the shaft velocity established at 0.48 m/s. The cooling water flow $Q$ set to 30 L/min according to the Thordon Instruction. This working condition maintain about 6 h for each bearing to ensure enough running in. After running in period, the shaft velocity set to 0.8, 1, 2, 3, 4, 5, 6, 7, 8, 9, 10 m/s when the nominal pressure set to 0.05, 0.1, 0.2, 0.3, 0.4, 0.5, 0.6 MPa respectively. All the working condition last 15 min before record parameters.

### Geometric modeling

**Fundamental equation**

Take cooling water as incompressible fluid and ignore the water properties change, $\rho$ and $\eta$ can be seen as constant, we obtain modified Reynolds equation (3) when WLSBs are in hydrodynamic lubrication condition.

![Friction pair schematic and experiment apparatus](image-url)
The boundary condition was set to Reynold boundary condition.

The bearing capacity of water film is equation (5):

\[
\begin{align*}
F_h &= \int \int \rho \sin \phi dxdy \\
F_v &= \int \int \rho \cos \phi dxdy \\
F &= \sqrt{F_h^2 + F_v^2}
\end{align*}
\]  

Water grooves ignored in the FDM calculation since grooves mainly distributed on the upper part, which is not the main working area. According to the Winkler assumption, plastic liner under load is equivalent to many parallel springs, which fixed between the bearing sleeve and water film. Dimensionless elastic deformation of lining can be written as equation (6):

\[
\delta h = \frac{2\eta \omega (1 + v)(1 - 2v)P}{(1 - v)cE\omega^2}
\]  

So \( H \) can be obtained by equation (7):

\[
H = 1 + \epsilon \cos \phi + \delta h
\]  

**Numerical scheme**

The inner surface of bearing can evenly divide into \( m \times n \) grids; \( m \) and \( n \) are the number of circumferential and axial grids respectively. Reynolds equation discretized and solved by SOR. So the dimensionless water film pressure expressed as equation (8):

\[
P_{i,j}^k = \beta \left( A_{i,j}P_{i+1,j}^{k-1} + B_{i,j}P_{i-1,j}^{k-1} + C_{i,j}P_{i,j+1}^{k-1} + D_{i,j}P_{i,j-1}^{k-1} - F_{i,j} - P_{i,j}^{k-1} \right) + P_{i,j}^{k-1}
\]

Dimensionless coefficients \( A_{i,j} \sim F_{i,j} \) expressed as equation (9):

\[
\begin{align*}
A_{i,j} &= H_{i+1/2,j}^3 \\
B_{i,j} &= H_{i-1/2,j}^3 \\
C_{i,j} &= \left( \frac{D}{L} \frac{\Delta \phi}{\Delta \lambda} \right)^2 H_{i,j+1/2}^3 \\
D_{i,j} &= \left( \frac{D}{L} \frac{\Delta \phi}{\Delta \lambda} \right)^2 H_{i,j-1/2}^3 \\
E_{i,j} &= A_{i,j} + B_{i,j} + C_{i,j} + D_{i,j} \\
F_{i,j} &= 3\Delta \phi (H_{i,1/2,j} - H_{i-1/2,j})
\end{align*}
\]

The pressure iterate procedure will stop when pressure convergence met equation (10).

\[
\sum_{i=2}^{n-1} \sum_{j=1}^{n} \left| P_{i-1,j} - P_{ij} \right| \leq \delta
\]  

The procedure stops when bearing capacity reached according to equation (11).

\[
\left| \frac{F - W}{W} \right| \leq \delta
\]  

**FDM validation and geometry modeling**

The water film thickness \( h_{\text{min}} \) calculated by the finite difference method has been compared to the experimental results obtained by Gao et al., as shown in Table 4. Water film thickness \( h \) of No. 1–No. 4 are the measured values by four displacement sensors installed in different position of the tested bearing. It is noted that the maximum error is about 9.7%. This is probably due to the influence of turbulent. In a word, the calculated results can meet the requirement of geometry modeling.

The calculated pressure distribution curves also validated. We can see from Figure 3 that the numerical
results match well with the literature. It is believed that the numerical procedure is credible.

Based on the calculated results, the geometry models of water film can establish.

**Improved boundary conditions and CFD model validation**

**Improved boundary conditions**

When the bearings are in hydrodynamic lubrication, the friction force is equal to the shear force between water molecules, so the friction heat generated in the water film. Because of the low heat conductivity of polymer lining, all the friction heat exerted on the interface between water film and shaft sleeve.

Assumed all the power consummation were converted to friction heat, we can obtain the total friction heat flux density $q$ by equation (12)

$$q = \frac{2V_2(M - M_0)}{\pi d^2 L}$$  \hspace{1cm} (12)

The heat conductivity through shaft sleeve can be simplified as heat conduction of cylindrical wall, so thermal resistance $X_1$ can express as equation (13)

$$X_1 = \frac{\ln \left( \frac{d_2}{d_1} \right)}{2\pi \lambda_1 L}$$  \hspace{1cm} (13)

The cooling principle of WLSBs is differ from the law of heat convection for circular tube because of the rotate shaft. However, we can make a correction of the cooling water velocity to meet the requirements. Assumed the cooling water is Newtonian fluid, ignored the influence of gravity and boundary slip, the average velocity in the circumferential direction is $V_2 \div 2$ because the circumferential velocity of inside water film is equal to $V_2$ and the circumferential velocity of outside water film is zero. Therefore, the correct velocity $V$ of cooling water can be obtained by equation (14) and the schematic is shown in Figure 4

$$V = \sqrt{V_1^2 + (V_2/2)^2}$$  \hspace{1cm} (14)
The inaccuracy of experimental correlations formula for turbulent heat transfer is often ±20% or even ±25%, while the max deviation between Gnielinski formula and experiment results is within ±10% in most cases. Therefore, the Gnielinski formula was chosen to calculate turbulent heat flow capacity of water film, as shown in equation (15)

\[
Nuf = \frac{(\frac{f}{8})(Re - 1000)Prf}{1 + 12.7 \sqrt{Prf^3}} \left[ 1 + \left( \frac{1}{L} \right)^{2/3} \right] c_t
\]

\[
c_t = \left( \frac{Prf}{Prw} \right)^{0.11}
\]

\[
Re = \frac{\rho Vl}{\eta}
\]

\[
l = 4A
\]

\[
f' = (1.82 \log Re - 1.64)^{-2}
\]

\[
h_w = \frac{Nuf \lambda}{l}
\]

\[
X_2 = \frac{1}{\pi D L h}
\]

The total friction heat will dissipate by cooling water and shaft sleeve, so the heat flux density \( q' \), which should be exerted on the water film inside wall can calculate by equation (16)

\[
q' = \frac{X_1}{X_1 + X_2} q
\]

The lining and shaft sleeve were separated by water when bearing is in hydrodynamic lubrication. Due to the high heat conductivity, shaft sleeve temperature can see constant. Water film inside surface was set to rotating wall. Besides \( q' \) exerted on it. Water film outside surface was set to stationary and isothermal wall. Water film inlet was set to velocity inlet, while the outlet was set to pressure outlet. Inlet and outlet temperature were set to room temperature. Turbulent parameters were set to hydraulic diameter and turbulence intensity when the cooling water was in turbulent condition. The cavitation effect ignored because the high temperature areas distribute in the lower half where the cavitation will not occur.

**CFD model validation**

In order to validate the simulation model, a comparison between literature and calculated results had been conducted, as shown in Figure 5. It is notable that the error of partial grooved bearing is smaller than the full grooved bearing, especially when the inlet velocity is high. This may cause by the definition of flow condition. As we know, the total heat exchange of groove region is much higher than the non-groove region because the heat transfer coefficient of turbulent flow is bigger than laminar flow. The model developed in this study treat the groove and non-groove region as a whole to calculate Reynold number, so the total heat exchange is smaller than actual condition. The more groove number of bearing, the bigger error between the actual condition and simulate results. Generally, whatever the partial grooved bearing or the full grooved bearing, the max temperature calculated by this new model match well with the results of literature.

**Results and discussion**

**Effects of linear velocity and nominal pressure on the friction coefficient**

Figure 6(a) and (b) show that the \( f - V_2 \) curve is similar to Stibbeck curve. WLSBs are often in mix lubrication when the velocity is low. Friction coefficient \( f \) drop dramatically first because the lubricate condition will improve along with linear velocity \( V_2 \). When the bearing is in hydrodynamic lubrication, the friction force determined by inner shear force between water molecules and the shear force related with velocity positively. Therefore, friction coefficient \( f \) will increase along with linear velocity \( V_2 \). Besides, the max friction coefficient appears when bearings are in low velocity and low load condition. One reason is the friction force in boundary or mix lubrication which is bigger than that in hydrodynamic lubrication. Another is the viscoelastic behavior which is often occurs at the low load condition due to the bigger clearance.

As shown in Figure 6(c) and (d), friction coefficient \( f \) decrease along with nominal pressure \( p_b \) and go to
stabilize when the pressure is high. The actual pressure decrease with the increase of elastic deformation. However, when the nominal pressure increase to a certain value, the elastic deformation will reach to maximum, so $f$ is in steady state. It also can see that the effect of nominal pressure on the friction coefficient of Thordon bearing is bigger than that on SF-1 bearing. This may cause by the different resistance to deformation. The elastic modulus of SF-1 is much higher than Thordon, so the friction coefficient curve of SF-1 is more moderate.

**Effect of linear velocity and nominal pressure on the total heat flux density of bearing**

Based on the equations (12) and (2), we can obtain equation (17)

$$q = \frac{p_b V_2 D^2}{\pi d^2}$$  \hspace{1cm} (17)

It is obvious that total heat flux density $q$ is the proportional function of nominal pressure $p_b$, linear velocity $V_2$ and friction coefficient $f$. However, the friction coefficient determined by nominal pressure and linear velocity, so it is necessary to study the effect of linear velocity and nominal pressure on the total heat flux density of bearing.

Figure 7(a) and (b) show that total heat flux density $q$ decrease first and then increase along with linear velocity $V_2$, with a turn point at 1 m/s. The variation of $q$ is the result of competition between $V_2$ and $f$. When the velocity is lower than 1 m/s, the major factor is $f$. While the influence of $V_2$ will in domination when it is beyond 2 m/s. It also can see that the increase of heat flux density along with velocity for Thordon bearing is more significantly than SF-1 bearing. This is because the matrix of SF-1 is PTFE, which is a distinguished self-lubricated material.

Figure 7(c) and (d) show that the variation of $q$ along with $p_b$ for Thordon and SF-1 are different. For Thordon bearing, $q$ is stable when the pressure is lower. The effect of friction coefficient on heat flux density is counteract with pressure. However, when the pressure is higher, $q$ shows an approximately linear increase along with pressure. Friction coefficient will be stable.
when the pressure increase to a certain level, so the heat flux density is proportion to pressure. For SF-1 bearing, \( q \) increase first, then decrease and finally increase along with the pressure. When the pressure is between 0.2 MPa and 0.5 MPa, the effect of \( f \) on \( q \) is bigger than pressure. While pressure is the major factor when the pressure lower than 0.2 MPa or bigger than 0.5 MPa. We can conclude that Thordon bearing is more suitable to low load condition while SF-1 bearing is the better choice at high load condition.

Effect of linear velocity, nominal pressure on the bearing temperature characteristics

The results of FDM show that minimum water film thickness \( h_{\text{min}} \) increase along with linear velocity \( V_2 \) and decrease along with nominal pressure \( p_b \). Besides, when the film thickness ratio is bigger than 3, bearings can be seen in hydrodynamic lubrication. Therefore, it can see that the bearing is in hydrodynamic lubrication when the velocity is greater than 4m/s and pressure is 0.2 MPa or the velocity is greater than 5m/s and pressure is 0.4 MPa.

In this paper, it regarded as light working condition when the bearing is at 0.2 MPa and 4–10 m/s. Meanwhile it studied as heavy working condition when the bearing is at 0.4 MPa and 5–10 m/s. Based on the Thordon Instruction, cooling water inlet velocity \( V_1 \) was set to 1m/s in this section. The working temperature of Thordon bearing is usually lower than 323 K and seawater temperature can reach 313 K at equatorial area in summer, so the bearing temperature rise should not exceed 10 K. In this paper, the region over 310 K is defined as high temperature.

The max temperature obtained by CFD analysis and the average temperature calculated by Equation (18).

\[
\begin{align*}
\dot{m} = \rho V_1 [\frac{\pi (D^2 - d^2)}{4} + n'b'h'] \\
\bar{T} = \frac{\pi d_1 q}{c_{w,m}} + T_0
\end{align*}
\]

Figure 8(a) shows that the max temperature at 0.4 MPa decrease along with \( V_2 \), while decrease first
and then increase at 0.2 MPa. The max temperature usually appears at the min water film thickness region, besides both the heat flow and minimum water film thickness $h_{\text{min}}$ will increase along with $V_2$, which can see from Figure 8(c) and (d). Hence, max temperature variation is the result of competition between the two factors. $h_{\text{min}}$ plays a dominate role at heavy working condition. When the bearing applied on low load, $h_{\text{min}}$ is predominating at lower velocity while heat flow is predominating at higher velocity. As shown in Figure 8(b), the average temperature increase slightly along with $V_2$, because the heat flow increase along with $V_2$ and cooling water flow is constant.

Figure 9(a) and (b) present the axial temperature distribution of min water film area for SF-1 and Thordon respectively. It can see that cooling water temperature increased sharply in inlet, then maintain at a high level and finally decrease in outlet rapidly. This attributed to the entrance effect of heat transfer. Meanwhile high temperature area decrease along with $V_2$, because the predominate effect of water film thickness. Figure 9(c) and (d) show circumferential temperature distribution at the middle ring of SF-1 and Thordon respectively. We can see that temperature increase first and then decrease from the horizontal direction of non-eccentric side (circumferential angle is 0). Besides the high temperature area located at the position of 80–140° in circumferential direction and 0.02–0.13 m in axial direction. The peak temperature appears at eccentric side and beyond $h_{\text{min}}$ position about 4–40°, which is verified by the results published recently.

**Effect of cooling water inlet velocity on the bearing temperature characteristics**

The temperature rise is the biggest for SF-1 bearing when it is at 0.2 MPa pressure, 4 m/s velocity. While Thordon bearing is in the worst cooling condition at 0.4 MPa pressure, 5 m/s velocity. The range of 1-5m/s studied to evaluate the effect of cooling water inlet velocity on the bearing characteristics.

As shown in Figure 10, max temperature decrease first, then increase and finally decrease along with
cooling water inlet velocity $V_1$, but the variation is slight which match well with literature. It is notable that max temperature rise is lowest when $V_1$ is 2 m/s. The average temperature decreases along with $V_1$ and decreases the most when $V_1$ is 2 m/s. Max temperature variation is the result of competition between heat transfer capacities and cooling water flow. But the influence of inlet velocity is small in min water film thickness area because the min water film thickness is usually dozen micrometers or even a few micrometers. This is why the variation of max temperature is little. For the reason that cooling water flow increase rapidly and heat transfer capacity increase slightly along with $V_1$, the average temperature decrease dramatically. In conclusion, it is appropriate to set the inlet velocity at 2 m/s.

Figure 11(a) and (b) present that the temperature increased sharply in the inlet, then maintain at a high level and finally decrease in the outlet rapidly. The temperature peak and high temperature area in axial direction move to the outlet side with the increase of $V_1$. Besides the temperature peak of SF-1 bearing appears at the middle part while it appears at the outlet side for Thordon bearing. Figure 11(c) and (d) show that the high temperature area in the middle part will reduce along with $V_1$.

**Conclusion**

A novel CFD simulation method developed based on the improved geometry model and boundary conditions. Effects and mechanisms of working conditions on temperature characteristics for two WLSBs were evaluated. From the results, following conclusions can summary:

1. With the increasing velocity, friction coefficient drops dramatically first and then increase slowly. For given operating conditions, increasing nominal pressure will decrease friction...
coefficient and it will go to stabilize when the pressure is high. The friction coefficient curve of SF-1 is more moderate due to the high elastic module. Besides, the max friction coefficient appears when bearings are in low velocity and low load condition.

(2) The variation of total heat flux density is the result of competition among linear velocity, friction coefficient and nominal pressure. With the increasing velocity, total heat flux density decreases first and increases after with a turn point at 1 m/s. For SF-1 bearing, the heat flux density increases first, then decreases and finally increase along with the pressure. For Thordon bearing, heat flux density is stable when the pressure is lower, while shows an approximately linear increase along with pressure when the pressure is higher. Thordon bearing is more suitable to low load condition while SF-1 bearing is the better choice at high load condition.

(3) Bearing temperature characteristics determined by the heat transfer capacity and water film thickness. Max temperature of water film at 0.4 MPa decrease along with increased velocity, while decrease first and then increase at 0.2 MPa. Besides the peak temperature appears at eccentric side and beyond min water film thickness position about 4–40°. The high temperature area mainly located at the position of 80–140° in circumferential direction and 0.2–0.13 m in axial direction.

(4) With the increase of inlet water velocity, max temperature of bearing changes slightly. It is appropriate to set the inlet velocity at 2 m/s to obtain better cooling performance. Temperature peak and high temperature area in axial direction move to the outlet side with the increasing cooling water inlet velocity. Besides, high temperature area will decrease along with the cooling water inlet velocity.

Figure 10. Effect of inlet velocity on max temperature, average temperature, heat transfer capacity of water film: (a) max temperature, (b) average temperature, and (c) heat transfer capacity.
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Figure 11. Temperature variation in axial direction and in circumferential direction of bearings: (a) axial temperature variation of SF-1 bearing, (b) axial temperature variation of Thordon bearing, (c) circumferential temperature variation of SF-1 bearing, and (d) circumferential temperature variation of Thordon bearing.
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**Appendix**

**Notation**

- $M$: friction torque
- $n$: rotary speed
- $Q$: cooling water flow
- $N$: loading force of hydro cylinder
- $f$: friction coefficient
- $P_b$: nominal pressure
- $W_b$: bearing weight
- $D$: bearing inner diameter
- $d$: shaft diameter
- $L$: bearing length
- $F_f$: friction force
- $M_0$: shaft friction torque without load
- $h_{\text{min}}$: min water film thickness
- $\Theta$: attitude angle
- $x$: circumferential coordinate
- $y$: axial coordinate
- $\rho$: water density
- $\eta$: water dynamic viscosity
- $h$: water film thickness
- $U$: relative velocity of water
- $c$: bearing clearance
- $R$: bearing inner radius
- $F_h$: horizontal component of the bearing capacity
- $F_v$: vertical component of the bearing capacity
- $F$: bearing capacity
\( \Omega \)  
hydrodynamic lubrication domain

\( \beta \)  
relaxing factor, \( 1 < \beta < 2 \)

\( k \)  
iteration number

\( A_{i,j} \sim F_{i,j} \)  
dimensionless coefficients

\( \delta \)  
pressure convergence accuracy, here is 0.001

\( \vartheta \)  
bearing capacity accuracy, here is 0.01

\( q \)  
total friction heat flux

\( q' \)  
friction heat flux exerted on water film

\( X_1 \)  
heat flow resistance of shaft sleeve

\( X_2 \)  
heat flow resistance of water film

\( V_1 \)  
cooling water inlet velocity

\( E \)  
elastic module of lining

\( v \)  
Possion’s ratio of lining

\( V_2 \)  
linear velocity of shaft sleeve

\( Nu_f \)  
Nusselt Number of water film

\( h_w \)  
convective heat transfer coefficient

\( c_t \)  
correction factor

\( A \)  
cross-sectional area of water pass way

\( S \)  
circumference of water pass way

\( R_e \)  
Reynold number

\( l \)  
characteristic length of flow pass way

\( f' \)  
Darcy resistance

\( Pr_f \)  
Pr number of cooling water at average temperature

\( Pr_w \)  
Pr number of cooling water which closed to the wall

\( d_1 \)  
shaft sleeve inner diameter

\( d_2 \)  
shaft sleeve outer diameter

\( \lambda_1 \)  
heat conductivity of shaft sleeve

\( \lambda \)  
heat conductivity of water

\( \omega \)  
shaft angular velocity

\( \varepsilon \)  
eccentricity ratio

\( \varphi \)  
dimensionless circumferential coordinate, \( \frac{\varphi}{2\pi} \)

\( \lambda \)  
dimensionless axial coordinate, \( \frac{\lambda}{L} \)

\( H \)  
dimensionless water film thickness, \( \frac{h}{L} \)

\( \rho \)  
dimensionless water film pressure, \( \frac{\rho \rho_f}{\rho_w} \)

\( m \)  
cooling water flow per second

\( \bar{T} \)  
cooling water average temperature

\( n' \)  
water groove number

\( c_w \)  
water specific heat capacity

\( T_0 \)  
water inlet temperature

\( b \)  
water groove width

\( h' \)  
water groove depth

\( d \)  
thickness of lining

\( \delta h \)  
dimensionless elastic deformation of lining