Analysis of Cooling Performance of a Marine Air-cooled Sliding Intermediate Bearing Based on Fluid-Structure Interaction

Shao Yong
China Ship Development and Design Center, Wuhan 430064, China

Email: shaoyong1111@163.com

Abstract. The intermediate bearing is an important component in marine propulsive system. The transmission efficiency, reliability and service life of propulsion shafting are significantly affected by the lubrication performance of intermediate bearing. Based on the theory of CFD, the FSI heat transfer numerical model of a marine air-cooled sliding intermediate bearing was established. The hydrodynamic lubrication numerical model of intermediate bearing was coupled with the heat transfer model by UDF. The temperature fields of intermediate bearing and oil in tank were obtained under the maximum speed condition. The simulation results agree well with the experiment and the accuracy of numerical model is also accepted at the same time. As the results shown, the highest temperature of bearing bush is 77.88°C and the average temperature of oil in tank is 70.14°C. The results indicate that the cooling performance of intermediate bearing meets the design requirements.

1. Introduction
The propulsive shafting is the important equipment in marine power system and its stability and reliability have a direct impact on ship’s dynamic performance. The intermediate bearing is mainly used to support the rotation of propulsive shafting, which endures heavy loads in the working conditions. The lubrication performance of intermediate bearing is affected by many factors. Among them, the temperature of oil is one of the primary factors [1]. Therefore, the study on the cooling performance of intermediate bearing plays a significantly important role in the improvement of its service performances.

In this paper, the hydrodynamic lubrication model and the FSI (Fluid-Structure Interaction, FSI) heat transfer model of intermediate bearing were established respectively [2-3]. In order to take account of the interactive effects, the above two numerical models were coupled by UDF (User Define Function, UDF) supported by the commercial CFD (Computational Fluid Dynamics, CFD) software FLUENT. The study on the cooling performance was carried out under the maximum speed condition.

2. Basic Theory

2.1. Lubrication Theory

2.1.1. Lubrication Governing Equation. The basic governing equation of hydrodynamic lubrication model is the Reynolds equation. Based on the different assumptions, the Reynolds equation can be
obtained in different forms. In this paper, the hydrodynamic lubrication models are established on the average Reynolds equation with the consideration of surface roughness [4-5]. The basic lubrication governing equation is shown in equation (1).

\[
\frac{\partial}{\partial x} \left( \phi \frac{\rho h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi \frac{\rho h^3}{\mu} \frac{\partial p}{\partial y} \right) = 6U \frac{\partial \phi}{\partial x} \frac{\partial h}{\partial x} + 6U \sigma \frac{\partial (\rho \phi)}{\partial x}
\]

(1)

Where, \( h \) is the nominal oil film thickness. \( p \) is the oil film pressure. \( x \) is the circumferential direction of the bearing. \( y \) is the axial direction of the bearing. \( U \) is the relative velocity of journal and bearing bush in the circumferential direction. \( \mu \) is the viscosity of oil. \( \rho \) is the density of oil. \( \phi \) and \( \phi \) are the pressure flow factors in \( x \) and \( y \) direction on the lubrication surface respectively. \( \phi \) is the stress flow factor.

2.1.2. Oil film thickness equation. Figure 1 shows the oil film thickness \( h \) of a sliding bearing [6]. \( O_1 \) and \( O_2 \) are the center of the bearing and journal separately. \( R_1 \) and \( R_2 \) are the radius of bearing and journal. \( \phi \) is the offset angel. \( \theta \) is the circumferential coordinates of the film thickness, which take the point at the maximum oil film thickness as the origin.

![Figure 1 Oil film thickness of a sliding bearing](image)

According to the above geometrical relations, the oil film thickness \( h \) can be deduced as equation (2), shown as below.

\[
h = c + e \cos \theta
\]

(2)

Where, \( c \) is the radius clearance defined by \( c = R_1 - R_2 \). \( e \) is the eccentricity defined by \( e = R_1 - R_2 - h_0 \). \( h_0 \) is defined as the minimum oil film thickness.

2.2. CFD Basic Theory

The law of mass, momentum and energy conservation must be obeyed in fluid dynamics [7]. To describe different flowing phenomena, several essential flow equations should be added.

2.2.1. The continuity equation. The continuity or mass-conservation equation must be obeyed in any flowing situation as Equation (3) shows.

\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_x)}{\partial x} + \frac{\partial (\rho u_y)}{\partial y} + \frac{\partial (\rho u_z)}{\partial z} = 0
\]

(3)

Where, \( u_x \), \( u_y \) and \( u_z \) are the velocity component in the direction of \( x \), \( y \) and \( z \) respectively. \( t \) is the time.

2.2.2. The momentum equation. The momentum equation is based on Newton’s second law, which means as a differential unit, the changing rate of momentum with respect to time is equal to the total forces acting on its surfaces.

\[
\frac{\partial (\rho u_x)}{\partial t} + \nabla \cdot (\rho u_x \vec{u}) = - \frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + \rho f_x
\]

(4)
\[
\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = -\frac{\partial p}{\partial t} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + \rho f_x \tag{5}
\]

\[
\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = -\frac{\partial p}{\partial t} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + \rho f_x \tag{6}
\]

Where, \( p \) is the pressure on the unit. \( \tau_{xx}, \tau_{xy}, \) and \( \tau_{xz} \) are the component of viscous stress. \( \tau \) resulting from the viscosity effect among molecules. \( f_x, f_y, \) and \( f_z \) are the unit mass force in three directions.

2.2.3. **The energy equation.** The law of energy conservation which based on Newton’s first law must be obeyed by the flowing systems including heat transfer. According to the law, the rate of energy increasing per unit is equal to the sum of net heat flux added into the unit and the work that the mass force and surface force has done. The energy equation is shown as below.

\[
\frac{\partial \rho E}{\partial t} + \nabla \cdot \left[ \bar{u} (\rho E + p) \right] = \nabla \cdot \left[ k_{\text{eff}} \nabla T - \sum h_j J_j + \rho f + \mathbf{S} \right] \tag{7}
\]

Where, \( E \) is the total energy including the internal energy, kinetic energy and potential energy. \( h_j \) is the enthalpy of the component \( j \). \( k_{\text{eff}} \) is the effective heat conductivity. \( J_j \) is the diffusion flux of component \( j \). \( \mathbf{S} \) is the source term including the reaction heat and other heat defined by users.

3. **Numerical Modeling**

3.1. **Geometry Model**

The air-cooled sliding intermediate bearing composes of different parts. The main components include bearing cap, bearing block, upper/lower bearing bush, upper/lower oil tank, oil flinger, oil collector, connecting bolts and so on. The three dimensional models of above parts are established and assembled according to the containment relationships. The assembly model of intermediate bearing is shown as Figure 2.

![Figure 2: Geometrical models of intermediate bearing](image)

1-bearing cap; 2-bearing block; 3-upper bearing bush; 4-lower bearing bush; 5-upper shell of oil tank; 6-lower shell of oil tank; 7-oil flinger; 8-oil collector

3.2. **Mesh Generation**

Two oil tanks are set on the left and right side of the intermediate bearing, in which an amount of oil have been stored. The oil gathers on the bottom of oil tank and forms the oil pool with a certain depth. The oil flinger is equipped in each oil tank and its bottom immerses in the oil pool. According to the geometrical configuration, the geometry model of oil pool in tank can be obtained.
Based on the CFD software FLUENT, the solid structures of intermediate bearing and the fluid zone of oil pool in tank have been meshed respectively with the unstructured tetrahedral mesh.

3.3. **Boundary Conditions**

The oil flinger of intermediate bearing is fixed on the propulsive shaft and rotates along with the shaft. A certain amount of oil in tank will be thrown up when the oil flinger rotates. Most of them is collected by the oil collector and then transported to the bearing bush. The oil in bearing bush not only can lubricate bush and reduce the friction force, but also can take away most of the friction heat generated at the bearing bush, which finally is taken to the oil pool in tank. The rest of oil thrown up by the oil flinger directly returns to the oil pool. The heat convection in the pool and the heat transfer between oil pool in tank and the lower shell of tank are effectively enhanced by the stirring of oil flinger. The heat imported from the internal walls of oil tank and bearing bush transfers among the parts of intermediate bearing in the way of heat conduction. The heat dissipation is achieved by the heat convection and the heat radiation on the exterior surfaces of intermediate bearing. The thermal equilibrium of intermediate bearing is reached by the comprehensive effects of above heat transfer methods in working condition.

Considering the features of heat transfer listed above, the following simplifications are introduced in FSI heat transfer model of intermediate bearing.

1) The heat convection and radiation exist on the exterior surfaces of intermediate bearing. Since the maximum temperature of the exterior surfaces is below 65°C, the heat radiation is neglected. The external flow model of intermediate bearing is established and the convection heat transfer coefficients of each exterior surfaces are obtained. The third boundary conditions composed of the external temperature $T_0$ and the convection heat transfer coefficients $\alpha$ are applied on the exterior surfaces of intermediate bearing and the heat convection between the intermediate bearing and environment is correspondingly considered.

2) Since the intermediate bearing is installed on the bearing pedestal, the bottom of the bearing block is defined as the constant environmental temperature $T_0$.

3) The heat conduction exists among all the parts of the intermediate bearing. Hence, the contact surfaces of the parts are defined as the thermal-coupling wall.

4) The heat flux $q$ is applied on the inner surface of bearing bush accordingly [8]. In this way, the friction heat is considered.

5) The process of oil thrown up from the oil pool by oil flinger is ignored. In order to consider the effects caused by oil flinger on the heat convection of oil pool, the contact surfaces between the oil pool and oil flinger are defined as a wall with the rotational velocity in working condition.

6) The interfaces between the oil pool and the tank are defined as the thermal-coupling walls, which realize the coupled heat transfer between solid and fluid zones.

7) As Figure 3 shown, one outlet and two inlets of oil are set on the upper surface of oil pool. The outlet, defined as a pressure outlet, indicates the oil thrown up from the pool by oil flinger. The inlets are defined as mass-flow inlets. The inlet 1 indicates the oil which is collected and transported to the bearing bush and finally flows back to the oil pool. The inlet 2 indicates the rest of oil which is thrown up by the oil flinger and directly falls down to the oil pool.

![Figure 3 Boundary conditions of the oil pool model](image)
According to simplifies and assumptions, the FSI heat transfer model of the air-cooled sliding immediate bearing is established. Table 1 shows the main boundary conditions in this numerical model.

| Zone                                | Boundary type | Specifications          |
|-------------------------------------|---------------|-------------------------|
| Exterior surfaces                   | wall          | HTC α, Environment Temp. $T_0$ |
| Bottom surfaces                     | wall          | Constant Environment Temp. $T_0$ |
| Loading surface of friction heat    | wall          | Heat flux, $q$           |
| Contact surfaces among solid constructs | coupled-wall | Coupled heat transfer |
| Oil inlet 1 of oil pool             | mass-flow inlet | Mass flow rate, $Q_{m1}$, $T_1$ |
| Oil inlet 2 of oil pool             | mass-flow inlet | Mass flow rate, $Q_{m2}$, $T_2$ |
| Oil outlet of oil pool              | pressure outlet | Environment pressure, $P_0$ |
| Contact surfaces between oil pool and oil flinger | coupled-wall | Rotational angular velocity, $\omega$ |
| Interfces between oil pool and oil tank | coupled-wall | Coupled heat transfer |

### 3.4. Coupled Model Based on UDF

The thermal load applying to the intermediate bearing, which result from the friction heat generated in bearing bush, is affected by variable factors, such as rotational speed, exterior load and oil properties. Especially, the friction heat is significantly affected by oil temperature. The interactions between the oil temperature and thermal load applying to the intermediate bearing must be taken into account in numerical model.

Based on UDF function applied by commercial software FLUENT, the FSI heat transfer model of the sliding intermediate bearing is coupled with the hydrodynamic lubrication model. Figure 4 shows the framework of the coupling model.

![Figure 4](dummy.png)

**Figure 4** Framework of coupling model based on UDF

As Figure 3.3 shown, data transfer between two models are realized by UDF, which acts as the bridge between FSI heat transfer model and lubrication model. In the process of calculation, the oil temperature of the oil pool outlet is firstly read from the FSI heat transfer model and then transferred to the lubrication model by UDF, which is defined as the temperature of oil supplied to the bush. According this oil input temperature for lubrication model, the lubrication performance is analyzed and the friction heat is calculated. Based on the friction heat, the temperature of oil inlet 1 defined as $T_1$ and the heat flux loaded to the inner surface of bush are updated. Then, the next cycle of iterative calculation begins until the results are convergent.

### 3.5. Material Parameters

Several materials are involved into the simulation of the cooling performance, including solid materials of each part in intermediate bearing and fluid material of the oil. The parameters of solid materials are obtained from the material manual. The physical parameters of the oil are obtained from test data. The details are specified in Table 2 and Table 3.
4. Results and Discussion

4.1. Lubrication Performance
The lubrication performance of the intermediate bearing under the maximum speed condition was investigated in the case that the oil temperature is 65°C. The calculation results are listed in Table 4.

Table. 4 Results of lubrication performance

| Item                        | Unit | Value |
|-----------------------------|------|-------|
| Min. oil film thickness     | μm   | 57.20 |
| Min. oil film thickness ratio| —    | 63.95 |
| Offset angle                | °    | 33.34 |
| Friction force              | N    | 443.95|
| Friction coefficient        | —    | 0.00222|
| Total friction power loss   | kW   | 2.3478|

4.2. Cooling Performance
The cooling performance of intermediate bearing under the maximum speed condition was investigated. The results are listed as follow. Figure 5 shows the temperature fields of main solid parts and cross-section of temperature and velocity of oil pool in tank.

Figure. 5 Contours of cooling performance
As the above results shown, the maximum temperature of the lower bearing bush is 77.88°C and the average temperature of oil pool in tank is 70.14°C under the maximum speed condition, which satisfied the technical requirements.

5. Comparison with Experimental Results
The cooling performance test platform of intermediate bearing has been set up as figure 6. The exterior load applied on the intermediate bearing is controlled by the hydraulic system and the adjustable-speed motor is used to control the rotational speed of the journal. During the test, the adjustment of the working condition is utilized by the central console.

![Figure 6 Test platform of intermediate bearing](image)

The test of cooling performance was accomplished under the maximum speed condition. The results of test points in the intermediate bearing obtained from the simulation and test are listed in Table 5. From the comparison, it can be seen that the results from simulation agree well with the test data. The maximum error is below 5%, which indicates the accuracy of the model is acceptable.

| Test Point            | Exp. Value | Sim. Value | ΔT | Error |
|-----------------------|------------|------------|----|-------|
| Bearing bush          | 70.5       | 70.31      | 0.19| 0.27  |
| Oil pool in tank      | 69.9       | 66.61      | 3.29| 4.71  |

6. Conclusions
In this paper, the cooling performance of an air-cooled sliding intermediate bearing was analyzed. The conclusions can be drawn as follows:
(1) Based on the relative theory, the FSI heat transfer model and lubrication model are established separately. Two models are coupled based on UD. Hence, the lubrication performance is taken account into the heat transfer model.
(2) The cooling performance test platform of intermediate bearing has been set up. By contrast, the results from simulation agree well with test data. The maximum error is below 5%, which indicates the accuracy of the model is acceptable.
(3) As the results shown, the maximum temperature of bearing bush is 77.88°C and the average temperature of oil pool in tank is 70.14°C under the maximum speed condition, which meet the technical requirements. The cooling performance of intermediate bearing is satified.

References
[1] F Wlassow, F Duchaine, G Leroy and N Gourdain 2010 3D Simulation of Coupled Fluid Flow and Solid Heat Conduction for the Calculation of Blade Wall Temperature in a Turbine Stage ASME Turbo Expo (Power for Land, Sea, and Air) pp 723–734
[2] Q Liu, E A Luke and P Cinnella 2015 Coupling Heat Transfer and Fluid Flow Solvers for Multidisciplinary Simulations Journal of Thermophysics & Heat Transfer vol 19(4) pp 417–427
[3] G Chen and J B Heywood 1987 Numerical Analysis of Unsteady Heat Transfer in Coupling System of Combustion Chamber Journal of Huazhong University of Science & Technology vol 4 pp 29–36
[4] X Ye 2004 Numerical Investigation and Application of Three-Dimensional Lubrication
Performance in Piston Ring Pack *Huazhong University of Science & Technology*

[5] R Santhosh, J L, Hee, Simmons, K G Johnson, D Hann and M Walsh 2017 Experimental Investigation of Oil Shedding From an Aero-Engine Ball Bearing at Moderate Speeds *ASME Turbo Expo (Turbomachinery Technical Conference and Exposition)* pp V07AT34A018

[6] Q Lin, Z Wei, W Ning and W Chen 2013 Analysis on the lubrication performances of journal bearing system using computational fluid dynamics and fluid–structure interaction considering thermal influence and cavitation *Tribology International* vol 64 pp 8–15

[7] N P Starostin, A S Kondakov and M A Vasilieva 2013 Identification of friction heat generation in sliding bearing by temperature data *Inverse Problems in Science and Engineering* vol 21.2 pp 298–313

[8] N Tala-Ighil and M Fillon 2015 A numerical investigation of both thermal and texturing surface effects on the journal bearings static characteristics *Tribology International* vol 90 pp 228–239