Analysis of Shaft Alignment Taking Oil Film Characteristics of Stern Tube Bearing into Consideration *

(Part 1, Theoretical analysis)

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In order to achieve energy savings, ships are now being designed with larger propellers operating at a lower speed. Such designs impose a heavier load on the stern tube bearing. The shaft alignment calculations have been made taking oil film characteristics of stern tube bearing into consideration.

The oil film characteristics have been analyzed applying the finite width hydrodynamic theory to the curved shaft in the stern tube bearing. This theoretical analysis has made it possible to elucidate the oil film characteristics of the stern tube bearing. The optimum width-diameter ratio of stern tube bearing at lower speed has been also obtained by this analysis.

Key words: Stern Tube Bearing, Shaft Alignment, Finite Width Bearing Theory, Oil-Film Characteristics, Low Speed, Optimum L/D, Edge Pressure

1. Introduction

In order to achieve energy savings, ships are now being designed with larger propellers operating at a lower speed through the use of reduction gear systems. Stern tube bearings are subject to more severe operating conditions than other types of bearings because they support the shafts that carry heavy propellers on their ends. Accordingly, the designs of these energy saving ships impose a heavier load on the stern tube bearings. Owing to this, it has become necessary to evaluate the minimum speed in the port when designing.

Theoretical analyses assuming the rigid propeller shafts to be inclined linearly in the stern tube bearings are reported(1,2). An analysis taking arbitrary shaft deflection within the stern tube bearing into consideration, however, has never been reported.

This study contributes to the theoretical analysis of the propeller shaft alignment and the oil film characteristics of the stern tube bearing.

It has made clear the oil film characteristics of the stern tube bearing in which the propeller shaft rotates at an extremely low speed with arbitrary deflections.

2. Method of Theoretical Analysis

2.1 Reynolds equation of stern tube bearing

Pressure distribution of the stern tube bearings can be obtained from the following Reynolds equation with Reynolds boundary condition (3):

$$\frac{\partial}{\partial y}\left[\frac{h}{\eta} \frac{\partial p}{\partial y}\right] + \frac{\partial}{\partial z}\left[\frac{h}{\eta} \frac{\partial p}{\partial z}\right] = \frac{\partial}{\partial y}\left[\frac{h}{\eta} \frac{\partial p}{\partial y}\right]$$

(1)

where, $p$ : oil film pressure, $h$ : film thickness, $\eta$ : film viscosity, $\omega$ : angular velocity, $\theta$ : circumferential coordinate from the minimum film thickness position, $z$ : axial coordinate from the bearing edge, $D$ : bearing diameter=2R, and $L$ : bearing width.

The stern tube bearing of the analysis is shown in Fig.1. The deflection curve of the propeller shafts within the stern tube bearing is schematically shown in Fig.2. As the deflection curve is assumed to be an arbitrary function of $z$ on the hmax-hmin plane, it is represented by the following equation.

$$h(z)=C \left[1-F(z)\cos \theta \right]$$

(2)

where, $F(z)=f(z)/C$, $C$ : bearing radial clearance.
Nondimensional film reaction force components $F_r$ and $F_\theta$ are obtained from Eq. (3) by using the nondimensional film thickness $H/h, C$, nondimensional film pressure $\Pi = Pf_0/\mu^0$, and clearance ratio $\phi = C/R$.

$$
F_r = \frac{1}{4\lambda_0} \int_0^{2\pi} \int_0^1 \Pi \cos \theta \ d\theta \ d\xi
$$

$$
F_\theta = \frac{1}{4\lambda_0} \int_0^{2\pi} \int_0^1 \Pi \sin \theta \ d\theta \ d\xi
$$

(3)

where, $\lambda_0 = L/2R$, $\xi = z/R$.

Nondimensional bearing load carrying capacity $F_0$ or Sommerfeld number $S_0$ and attitude angle $\gamma$ can be calculated by Eq. (4) and Eq. (5).

$$
F_0 = \sqrt{F_r^2 + F_\theta^2}
$$

(4)

and

$$
S = \frac{1}{(2\Pi F_0)}
$$

(5)

Nondimensional frictional force $F_T$ can be calculated from the following equation.

$$
F_T = \int_0^{2\pi} \int_0^1 \left[ \frac{1}{H} + \frac{1}{2} \frac{\partial^2 \phi}{\partial \theta^2} \right] d\theta \ d\xi
$$

(6)

Accordingly, the frictional coefficient is obtained from Eq. (7).

$$
\frac{S_\phi}{S_0} = \frac{F_T}{2F_0}
$$

(7)

Fig. 1 A schematic diagram of the stern tube bearing

Fig. 2 Shaft deflection within the stern tube bearing

2.2 Linear calculation for the deflection

The deflection of the propeller shaft which is schematically represented in Fig. 3 is calculated by means of the transfer matrix method.

In the calculation, it is assumed that the shaft is supported rigidly on all the bearings except the stern tube bearing. The distributed reaction force component in the axial direction which is obtained by integrating the film pressure in the circumferential direction is used as the supporting condition of the stern tube bearing for calculating the shaft deflection by means of the transfer matrix method.

2.3 Numerical calculation

Fig. 4 shows a flow chart of the analysis. The calculating process is composed of two parts, the calculation of oil film characteristics and that of shaft deflection.

The former is to obtain the film reaction force distribution within the stern tube bearing from $f(z)$, and the latter is to calculate the shaft deflection from the film reaction force distribution obtained in the former part. The calculation is repeated until both deflections become equal.

As it is likely that little fluctuation of propeller external force occurs in static conditions at lower speeds, it can be assumed that the direction of the external force acting on the stern tube bearing coincides with the direction of gravity.

The criterion of convergence in numerical calculation for shaft deflection is given in the following equation,

$$
\varepsilon \int_0^L \left[ (f_0 - \delta) - e - h \cos \left[ \gamma - \frac{1}{2} \gamma_1 \right] \right] dz = 0
$$

(8)

where, $\varepsilon$ is a small value of criterion, and $f_0$ is a small deflection of the shaft center line in the film thickness direction from the standard deflection line. $\delta$ is the eccentricity of the bearing center from the standard deflection line.

Fig. 3 Arrangement of shafting
3. Calculation Results of Bearing Characteristics

3.1 Calculation results from the assumed deflection curves

The length of diameter ratios of almost all stern tube bearings ranges from 1 to 2.5. Calculations are performed for two kinds of L/D ratios, and for two kinds of arrangements of supply oil grooves. Two oil grooves with twelve degrees each are arranged horizontally.

The deflection figures of the shaft center line are assumed to be an inclined line or a 2nd order curve. It is found that the result, which is calculated by using the second order curve, corresponds with the measured results in a real ship moving at a very low speed.

3.1.1 The bearing load carrying capacity

The non-dimensional film thickness and attitude angle are calculated against Sommerfeld number in Fig.5 for the case in which the bearing and journal are parallel with each other. They are the results obtained when L/D ratios are 1 and 2. The results agree with those which have been hitherto reported.

The results of the cases in which the bearing and journal are not parallel are shown in from Fig.6 to Fig.9. The value of the non-dimensional film thickness hc/C is calculated in the middle of the bearing width. It is assumed that the deflection shapes of the journal are the inclined line and the 2nd order curve. The inclined angle Z is defined as the angle between the bearing center line and the line which is linked together with the journal centers at both ends of the bearing in the attitude angle plane. The deflection w is defined as the distance between the deflection curve and the inclined line in the middle position of the bearing width in the attitude angle plane. The values of the inclined angle corresponding to the symbols in these figures are tabulated in Table 1. The value of the non-dimensional film thickness hc/C at the bearing end is identical on one dotted line.

Fig.5 The film thickness and attitude angle for plane journal bearing

Table 1 Angle of inclination

<table>
<thead>
<tr>
<th>NOTATIONS</th>
<th>ANGLE OF INCLINATION (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>L/D=1</td>
</tr>
<tr>
<td></td>
<td>x10^-6</td>
</tr>
<tr>
<td>□</td>
<td>0</td>
</tr>
<tr>
<td>○</td>
<td>90</td>
</tr>
<tr>
<td>△</td>
<td>190</td>
</tr>
<tr>
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<td>390</td>
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</tr>
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<td>□</td>
<td>690</td>
</tr>
<tr>
<td>□</td>
<td>790</td>
</tr>
<tr>
<td>□</td>
<td>890</td>
</tr>
</tbody>
</table>

Fig.6 The film thickness and attitude angle for the 2nd order curve (L/D=1, W/D=4.5x10^-5)
The nondimensional load carrying capacity decreases with an increase of the inclined angle \( \alpha \). When the Sommerfeld number is constant, the film thickness \( h_e/C \) increases as the inclined angle \( \alpha \) decreases and it reaches its maximum value when the inclined angle is zero or when the bearing and journal are in parallel condition. It is found that the load carrying capacity obtained when the deflection figure is assumed to be a 2nd order curve is less than the one which is obtained by assuming the deflection figure to be an inclined line for the same \( L/D \) ratio. Comparing with the results obtained by using the 2nd order curve, the load carrying capacity decreases as the deflection \( w \) increases.

3.1.2 Reaction force distribution in the axial direction

Figure 10 shows the reaction force distribution in the axial direction, \( R_x/R_{max} \), and nondimensional film thickness \( h_e/C \) against the same Sommerfeld number for the cases of the inclined line and the 2nd order curve. The solid line represents the case when the inclined line is used. The chain line and dotted line both represent the cases when the 2nd order curve is used. The former is the case of \( w/D = 4.5\times10^{-5} \), the latter is that of \( w/D = 9.5\times10^{-5} \). The axial location at which the reaction force reaches the maximum value is in the neighborhood of the bearing end for both cases when the inclined line and the second order curve are used. This axial location becomes much closer to the bearing end as the deflection \( w \) increases in case of the second order curve.

Two peaks of the reaction force distribution are found near both edges of the bearing in the case of the second order curve. The degree of protrusion becomes remarkable with a decrease in the inclined angle \( \alpha \) and in the nondimensional film thickness \( h_e/C \).
This indicates that the approximation of the deflection figure as an inclined line is not appropriate for long width bearings such as the stern tube bearings.

3.1.3 Frictional coefficient

The frictional coefficient is shown for the cases of the inclined line and the 2nd order curve from Fig.11 to Fig.14. The meanings of the symbols and the dotted lines are the same as explained in 3.1.1.

For the constant Sommerfeld number, the frictional coefficient increases with an increase of the inclined angle. The degree of the increase is found remarkable against large Sommerfeld number.

The frictional coefficients become almost equivalent to each other regardless of the variation of the inclined angle when \( \frac{h_e}{C} \) reaches a value more than 0.06. This is attributable to the fact that the degree of variation of the load carrying capacity becomes greater as the film thickness decreases. Some studies pointed out that the frictional coefficients are not variable with the variation of the inclined angle. On the contrary, it is made clear from the authors' calculation that the frictional coefficients varied to some degree with the variation of the inclined angle. For the case in which the journal is inclined within the bearing, the frictional force increases in the vicinity to one bearing edge and likewise decreases on the other side. The frictional coefficient is not variable because the frictional force of the bearing does not vary in total as a result of averaging. When the film thickness is thin, the total frictional force somewhat differs because the force can not be averaged. Accordingly, the frictional coefficients vary with the variation of the inclined angle when the film thickness is thin.

3.2 Calculation results for the ships in operation

In order to investigate the degree of the film thickness formed in the stern tube bearings of ships in operation, the calculation was performed on A-maru (400,000 ton BW tanker, in operation) and B-maru (diesel engines with lower speed propellers, in stage of designing).
Table 2 shows the particulars of A-maru and B-maru.

3.2.1 The film thickness

The solid lines in Fig. 18 represent the relationship between the rotational speed and the nondimensional film thickness at the bearing ends in A-maru. According to the calculation, the film thickness at the aft end is 15.1 $\mu$m at 25 rpm. Although this is under the critical value of the film thickness which has been accepted, no problems have occurred when A-maru operates at 25 rpm.

In the vicinity of 30 rpm, the film thicknesses of the aft and fore ends reversed themselves. This indicates that it would be possible to rotate the minimum operating speed in port an appropriate inclined angle at which the film thicknesses of the aft and fore ends are equal. This inclined angle can be obtained from the analysis.

3.2.2 The optimum length to diameter ratio of the stern tube bearing

<table>
<thead>
<tr>
<th>Table 2</th>
<th>The particulars of the stern tube bearing</th>
</tr>
</thead>
<tbody>
<tr>
<td>DIAERET (mm)</td>
<td>A-maru</td>
</tr>
<tr>
<td>EFFECTIVE LENGTH (mm)</td>
<td>2350</td>
</tr>
<tr>
<td>RADIAL CLEARANCE (mm)</td>
<td>800</td>
</tr>
<tr>
<td>ANGLE OF OIL GROOVE (HORIZONTAL DEGREES)</td>
<td>14 DEGREES</td>
</tr>
<tr>
<td>OIL VISCOSITY (cSt)</td>
<td>110</td>
</tr>
<tr>
<td>AVERAGE PRESSURE BEARING AREA (MPa)</td>
<td>0.38</td>
</tr>
</tbody>
</table>

The length to diameter ratio of the stern tube bearing is recommended to be within the range from 1.5 to 4, though it is variable according to the lubricating systems. The L/D ratio of 2 is applied to ships for commercial use whose stern tube bearings are made of white metal material.

It is necessary for the film thickness formed within the stern tube bearing to be thick in order to avoid direct contact between the journal and the bearing surface. Furthermore, the uniforming of film thickness is also required in the stern tube bearing.

In this analysis, the optimum L/D ratios are defined as those at which the film thicknesses of the fore and aft ends are equal to each other.

The film thicknesses at the fore and aft ends are represented against the L/D ratio in Fig.15 and Fig.16 for the cases of A-maru and B-maru, respectively. The revolution speed is 30 rpm. The solid line represents the case when the slope angle $\alpha_2$ to the standard line of the deflection (the slope of the bearing) is constant. The dotted line represents the case when the adjoining support bearing is constant. The dotted line represents the case when the adjoining support bearing is set 100 $\mu$m above the standard line of the shaft with a constant slope angle $\alpha_2$.

The chain line in Fig.15 represents the case when the slope angle is varied so that the deflection curve is almost parallel to the bearing.

Fig. 15 The effect of the L/D ratio on the film thickness (A-maru)

Fig. 16 The effect of the L/D ratio on the film thickness (B-maru)
In the case of A-maru, the film thickness at the aft end increases with an increase of L/D ratio though it decreases at the fore end. The same tendency as in the case of A-maru can be recognized in the case of B-maru.

The film thicknesses at the fore and aft ends become equal in both cases when the L/D ratio is about 2. The film thickness at the fore end reaches the maximum value against the L/D ratio of approximately 1.5 in the case of A-maru though it reaches the maximum when the L/D ratio is approximately two in the case of B-maru. This is due to the fact that the height of the adjacent support bearing above the foundation and the degree of deflection within the stern tube bearing are different between A-maru and B-maru.

It is clear that the film thickness at the fore end would not increase even if the L/D ratio were greater than two. This indicates that little effect is expected from the thickening of the film with an increase of the L/D ratio when the L/D ratio becomes greater than a certain value.

Moreover, the film formed within the adjacent support bearing is effective for the decrease of the film thickness at the aft end and it causes the optimum L/D ratio to increase.

The value of optimum L/D ratio found to be in a range from 2 to 2.5 according to the calculation result with variation of the slope angle $\theta_s$. As shown by the chain curve in Fig.15, the film thicknesses at the fore and aft ends reach the maximum value when the L/D ratio is between 1.5 and 2 with the slope angle being varied for every L/D ratio. When the slope angle $\theta_s$ is constant, the film thickness increases with a decrease of the inclined angle, though the deflection $w$ increases with an increase of L/D ratio and it reduces the film thickness.

From the consideration above, it is seen that the optimum L/D ratio for ships in low speed operation is about two. This confirms theoretically the fact that the L/D ratio of two is recommendable from past experiences for the stern tube bearings with white metal material.

Fig.17 shows the reaction force distribution in the axial direction with the L/D ratio being varied for B-maru. The reaction force distributes uniformly in the axial direction in the case of L/D =2 compared with those of other L/D ratios.

3.2.3 The location of oil grooves

The film thickness and Sommerfeld number are represented against the revolutionlal speed in Fig.18. The solid line represents the case when oil grooves of twelve degrees are set in the horizontal direction case (i). The dotted line is the case when the oil grooves are set thirty degrees above the horizontal plane case (ii).

The load carrying capacity in the latter case increases to some degree compared with that of the former case. This is due to the fact that the bearing area on which the film pressure generates becomes larger in case (ii).

The film thickness at the aft end increases in both cases to almost an equivalent degree, within the range of 25 rpm to 40 rpm. The ratio of the film thickness in case (ii) compared to that in case (i) is 1.61 at 25 rpm and 1.36 at 30 rpm. The rate of increase effect of the film thickness becomes larger as the revolutionlal speed decreases. Finally, the effect of the oil groove location appears remarkably in case (ii). In the stern tube bearing at very low speed of

![Graph showing the effect of L/D ratio on reaction force distribution](image)

![Graph showing the effect of shaft speed on film thickness and load carrying capacity](image)
operation, the value of the minimum film thickness is important in designing. Changing the oil groove location would be effective to increase the minimum film thickness.

Figure 19 shows the normalized reaction force $R_z/R_z \text{max}$ against the nondimensional axial coordinate $z/L$. The solid line represents a result of the bearing with two oil grooves set in the horizontal direction (case (i)). The dotted line represents a result of the bearing with two oil grooves set thirty degrees above the horizontal plane (case (ii)). Reaction force $R_z$ of the latter bearing distributes more uniformly than that of the former one. This indicates that the film thickness at the aft end increases in the case using the latter bearing instead of the former one.

4. Conclusions

Shaft alignment of ships with a very low speed of operation is analyzed theoretically taking the oil film characteristics of the stern tube bearings into consideration.

The deflection curve of the propeller shaft is obtained by means of the transfer matrix method by using the film characteristics of the stern tube bearing. The film characteristics are calculated by using the deflection curve. The results obtained after repeating these steps are shown in Fig.15 to Fig.19.

Furthermore, the film characteristics are also calculated by using the deflection curves which are assumed as an inclined line and a second order curve. The results are shown in Fig.15 to Fig.14.

Conclusions from this paper are summarized in the following.

(1) When the deflection figure is assumed as a second order curve, the film thickness decreases as the degree of convex of the second order curve becomes remarkable when the inclined angle $\phi$ is constant. The reaction force distribution has two peaks in the axial direction in the vicinity of bearing ends.

(2) The frictional coefficient cannot be represented by a single curve against Sommerfeld number in the case of very thin films at the fore and aft ends.

(3) The optimum $L/D$ ratio is approximately two for the stern tube bearing with a very low speed of operation.

(4) Increase of the height of the adjacent support bearing above the foundation has an effect to decrease the film thickness at the aft end.

(5) The film thickness varies with the locations of the oil grooves. The degree of the effect increases with a decrease of the rotational speed.

An experimental study will be reported in the next paper.

References


(2) I. Matsumoto et al., Pre-prints of the Japan Society of Shipbuilding Engineers, (1981-11)