A Study on the Application of New Lining Material for Ship Propulsion Shafting System

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Abstract. Exothermic accident at the rear part of the after stern tube journal bearing is mainly caused by an excessive local compressive pressure at the rear part as the propeller weight deflects the propulsion shaft. By using a material with lower stiffness than white metal, which is used for the lining of stern tube bearings, it would be possible to decrease the local compressive pressure by increasing the contact area with the shaft, eliminating the cause of exothermic accident. We analyzed the shafting system in order to investigate the compressive pressure distribution characteristics and allowable pressure when a new, less rigid material is applied. The analysis results confirmed that a low-stiffness product can significantly decrease the local compressive pressure compared to an existing product when applied to an actual ship. Furthermore, the pressure distribution changes were small even when the loading condition was changed, allowing for a robust design.

1. Introduction

Journal bearing supports the smooth rotation of a cylindrical rotating shaft and receives a load perpendicular to the rotating shaft. It has a long cylindrical shape to support large loads, and uneven loading to the bearing can lead to an exothermic accident. Stern tube is attached where the propeller shaft penetrates the hull, preventing seawater intrusion into the ship as well as supporting the propeller shaft. Generally, stern tube bearings are placed at the front and rear of the stern tube, transmitting torque to the propeller and supporting the shaft [1]. The bearing at the front of the stern tube is called a forward stern tube bearing, and the one at the rear is called an after stern tube bearing. Both are long cylindrical journal bearings.

The most frequent damages in ship machinery occur in propeller shaft bearing and main engine bearing, and stern tube bearings are damaged the most among them [2]. Most exothermic accidents occur at the rear part of the long after stern tube bearing. Their main cause is an excessive local compressive pressure at the rear part as the propeller weight deflects the propulsion shaft.

Recently, after stern tube bearings are installed with a consideration of rake, which slightly elevates the front part to distribute some of the rear load to the front. Load distribution by rake has limitations, in that the evenness of contact between the shaft and lining material for stern tube bearing (white metal) is not sufficient. After stern tube bearings whose lining materials have much lower stiffness than white metals are emerging as a solution, but most shipyards are still using stern tube bearings made of white metal.

This study aims to investigate the compressive pressure distribution characteristics and determine the allowable pressure for the new material through shafting system analysis before applying bearings made of materials with a low stiffness. The results could also become the performance test requirement.
criteria for guaranteeing product performance from the manufacturer. The Hertzian contact condition assuming half-elliptical pressure distribution along the contact width was adopted in this study to calculate local compressive pressure [3]. We chose an actual LNG carrier for the analysis, and the analysis data of this ship were used for the shafting system bearing offset and hull deflection. We introduced the gap element used in [4] to simulate bearing rake and calculate reactions.

2. Design criteria for shafting system alignment

Shafting system alignment design is a design process to set the bearing reaction to a desired level by adjusting the spacing, length, and offset of bearings supporting the shafting system [5]. The only design criterion in the past was a mean pressure of 8bar on the stern tube bearing in a shafting system alignment design, and there has been active research to add a local pressure design criterion to the mean pressure criterion in order to prevent exothermic accidents [6]. A number of shipyards are using local pressure under 100bar as the design criterion.

The Hertzian contact condition assuming half-elliptical pressure distribution along contact width was adopted in this study to calculate the local pressure, as used in [6]. The reaction per unit length (Q) can be expressed as a relationship between contact width (b) and material properties as shown in (1), where \( E^* \) and \( R \) are defined in (2) and (3), respectively.

\[
Q = \frac{nb^2E^*}{2R} \quad (1)
\]

\[
E^* = \left( \frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2} \right) \quad (2)
\]

\[
R = \frac{D_1D_2}{D_1+D_2} \quad (3)
\]

In (2), \( v_1 \) and \( v_2 \) are the Poisson's ratios of the bearing material and shaft material, respectively, and \( E_1 \) and \( E_2 \) are their respective Young’s modulus. In (3), \( D_1 \) and \( D_2 \) are bearing inner diameter and shaft diameter, respectively.

Because of the half-elliptical contact pressure distribution, the maximum pressure is expressed as in (4), and substituting (1) here yields (5).

\[
P_{\text{max}} = \frac{bE^*}{R} \quad (4)
\]

\[
P_{\text{max}} = \sqrt{\frac{2}{\pi}} \sqrt{\frac{QE^*}{R}} \quad (5)
\]

The reactions computed in the shafting system analysis are substituted in (5) to get the maximum local compressive pressure, and the value must not exceed the tolerance of the material.

This study refers to bearing material properties in order to calculate the local pressure and determine the allowable pressure to be used in the new product, and the properties of relevant materials are listed in table 1.

| Material              | Young’s modulus [N/mm²] | Poisson’s ratio |
|-----------------------|-------------------------|-----------------|
| Shaft                 | 206,900                 | 0.3             |
| White metal           | 53,000                  | 0.3             |
| New lining material   | 490                     | 0.45            |

3. Shafting system analysis

As heating problems in the shafting system occur during operation of the engine, propeller weight in an operating engine, thermal effect and hull deflection were taken into consideration. We chose the
representative ship loading conditions i.e. “ballast” condition, “full load” condition, and “full load with wave” condition. The model for shafting system analysis is shown in figure 1.

![Figure 1. FE model for shaft analysis.](image)

3.1. Hull deflections
There has been much research on hull deflection and shafting system analysis, using finite element analysis results to calculate hull deflection [6], constructing a hull deflection database and applying it [7], and using approximation curves for the hull deflection [8] for example. Most recent shafting system analyses consider the effect of hull deflection. In this study, the finite element analysis was used to analyze the shafting system considering hull deflection.

3.2. Bearing contact model and offset
The gap element was applied to calculate the compressive pressure in the after stern tube bearing. As shown in figure 2, 10 elements were placed in the left and right width directions and up and down directions of the ship, adding up to 40.

![Figure 2. Gap element for simulation of after stern tube bearing.](image)

The offsets in shafting system analysis of the actual ship were used with no modification, and the details for operating engine condition are outlined in table 2. Table 3 lists the rake values in the actual ship. There is no rake in the forward stern tube bearing, and the rake applies only for the after stern tube bearing.

![Table 2. Bearing offsets.](image)

| Aft. STB [mm] | Fwd. STB [mm] | Inter. Shaft Bearing [mm] | 2nd Reduction Gear Bearing [mm] |
|---------------|---------------|--------------------------|-------------------------------|
| 0.0           | 0.0           | -0.35                    | -2.70                        |
|               |               | -5.40                    | -5.40                        |

![Table 3. Rake of stern tube bearing.](image)

| Bearing      | Aft. Offset [mm] | Fwd. offset [mm] | Rake [10^-4 rad] |
|--------------|------------------|------------------|------------------|
| Aft. STB     | -0.12            | 0.13             | 1.42             |
| Fwd. STB     | 0.0              | 0.0              | 0.0              |
4. Analysis result
The total reactions on forward and after stern tube bearings determined through a shafting system analysis for the new product with lower stiffness are listed in table 4. The results of the new product are nearly identical to the existing actual ship results. Calculated mean pressure for new product are 6.0bar and 1.4bar for after and forward stern tube bearings, respectively.

Table 4. Bearing reaction at stern tube bearing.

| Bearing  | Bearing reaction force [ton] | New product / Existing ship |
|----------|-----------------------------|----------------------------|
|          | “ballast”                   | “full load”                |
| Aft. STB | 77.0 / 77.4                 | 76.9 / 77.3                |
| Fwd. STB | 5.8 / 5.2                   | 5.8 / 5.2                  |

Table 5. Maximum pressure distribution of stern tube bearing.

| Station | Maximum local pressure [bar] | New product / Existing ship |
|---------|-------------------------------|-----------------------------|
|         | “ballast”                     | “full load”                 |
|         | “full load with wave”         |                             |
| 1       | 6.8 / 79.9                    | 6.8 / 80.9                  | 6.8 / 76.3                  |
| 2       | 6.7 / 66.2                    | 6.7 / 67.4                  | 6.8 / 70.6                  |
| 3       | 6.6 / 52.3                    | 6.6 / 54.0                  | 6.7 / 55.0                  |
| 4       | 6.5 / 38.3                    | 6.5 / 41.2                  | 6.6 / 38.5                  |
| 5       | 6.5 / 47.2                    | 6.5 / 30.5                  | 6.5 / 43.4                  |
| 6       | 6.5 / 44.3                    | 6.5 / 46.3                  | 6.5 / 35.8                  |
| 7       | 6.5 / 47.6                    | 6.5 / 49.2                  | 6.5 / 52.2                  |
| 8       | 6.5 / 56.2                    | 6.5 / 57.3                  | 6.5 / 57.1                  |
| 9       | 6.5 / 68.4                    | 6.5 / 69.2                  | 6.5 / 66.5                  |
| 10      | 6.6 / 82.8                    | 6.6 / 83.8                  | 6.6 / 87.9                  |

The Hertzian contact condition assuming half-elliptical pressure distribution along the contact width was applied to calculate the local pressure, and table 5 shows the results.

Calculation results for the new product are largely different from the actual ship shafting system analysis results for the existing product. The biggest difference is the decrease of maximum local pressure to 1/10 of the actual ship in all loading conditions, possibly due to a significant increase in the contact width in each position. It also displays lower sensitivity to the loading conditions. This can be an advantage of robust design, since it means a very small change in pressure distributions even when the loading condition changes.

Figure 3. Pressure distribution under “full load with wave” condition.
The maximum local pressure for an actual ship was observed under “full load with wave” condition with a value of 87.9 bar, 14% lower than the allowable pressure. To apply the new product, manufacturers should guarantee with reliable experimental data the durability for local compressive pressure of over 7.8 bar, up 14% from 6.8 bar. If the new product cannot withstand a local compressive pressure of over 7.8 bar, it is desirable to use an existing product with proven material properties.

5. Conclusion
A shafting system analysis and local compressive pressure calculation were performed in this study for a journal bearing lining product with a low stiffness which enables a larger contact area between the shaft and the stern tube bearing, and the following results were obtained.

Bearing products made of materials with a low stiffness significantly lower the local compressive pressure by increasing the contact width with the shaft.

Products with a low stiffness are less sensitive to changes in loading condition, and provide the advantage of robust design because they show small changes in pressure distribution even when the loading condition changes.

When manufacturers guarantee a proper level of allowable pressure, the application of products with a low stiffness will be possible.

In order to apply the new product, it is mandatory to perform shafting system analysis and calculation of local compressive pressure and present the product performance specifications that the manufacturers need to guarantee.

6. References
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