

# A Study on the Application of New Shaft System for VLCC

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**Abstract.** Some of the most frequent damages in the ship-mounted machinery occur in the propeller shaft system, and the most frequent damage is due to the exothermic accident occurs in the after stern tube bearing (STB). Recently, engine power and shaft diameter of very large crude-oil carriers (VLCCs) tend to increase whereas the distance between forward and after STB becomes shorter. In this study, the applicability of new shaft system for VLCCs adopting single STB system was investigated. The study results show that single STB system is quite possible by applying proper shaft arrangement and partial slope at the white metal. Also, a considerable economic effect is expected by simplifying related shaft system.

## 1. Introduction

The stern tube is mounted at the place where propeller shaft is penetrated through the hull. It plays a role in supporting propeller axis as well as preventing sea water from entering to the inside of the ship. Generally, two stern tube bearings (STBs) are installed. A bearing installed at the forward of the stern tube is called forward STB, and a bearing installed at the rear side of the stern tube is called after STB.

Some of the most frequent damages in the ship-mounted machinery occur in the shaft system, and the most frequent damage occurs in the after STB [1]. Exothermic accidents at the after STB whose length is long are caused by excessive local compressive pressure at the rear part as propeller load squeezes the propeller shaft.

In recent design of very large crude-oil carriers (VLCCs), the engine horsepower and shaft diameter tend to increase, whereas the shaft length becomes shorter and shaft diameter tends to increase as the location of engine is moved to a stern. Furthermore, as the distance between forward and after STBs becomes relatively shorter, shaft system flexibility decrease. This can aggravate the state of exothermic problem according to hull deflection.

In this study, the applicability of the shaft system where only after STB was installed while removing the forward STB was investigated as a measure to solve above mentioned problem. If the forward STB can be removed, additional benefit such as simplification of building the complex stern tube which takes much time can be achieved.

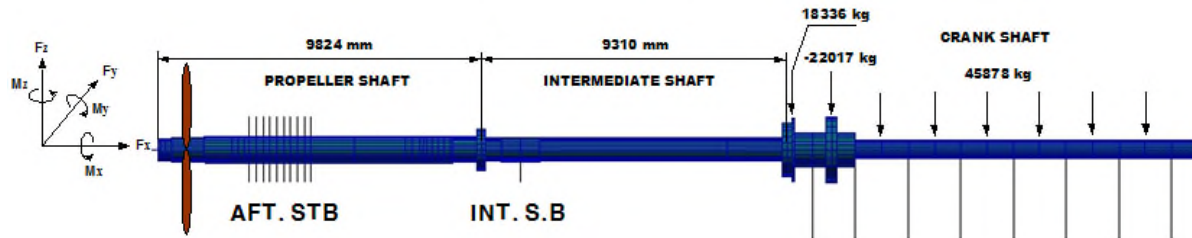
## 2. Shaft system arrangement

The proposed shaft system arrangement for 318K VLCC after removing forward STB is shown in figure 1. An intermediate shaft bearing is moved to the stern side compared to that in the existing ship to reduce a load burden on the after STB what is increased due to the removal of the forward STB.

In the past, the forward and after STBs were installed to have a rake to be set to "0", but recently, a front part of the after bearing is raised and a rear part is lower. The reason for this is to prevent excessive local pressure occurred at the rear part of the bearing via squeezing the propeller shaft by propeller load.



Length of the after bearing in the proposed design is 2000 mm, which is 200 mm longer than that of the existing ship. By doing this, a load burden on the after bearing can be alleviated and lakes are arranged similarly between them.



**Figure 1.** Proposed shaft system arrangement for VLCC.

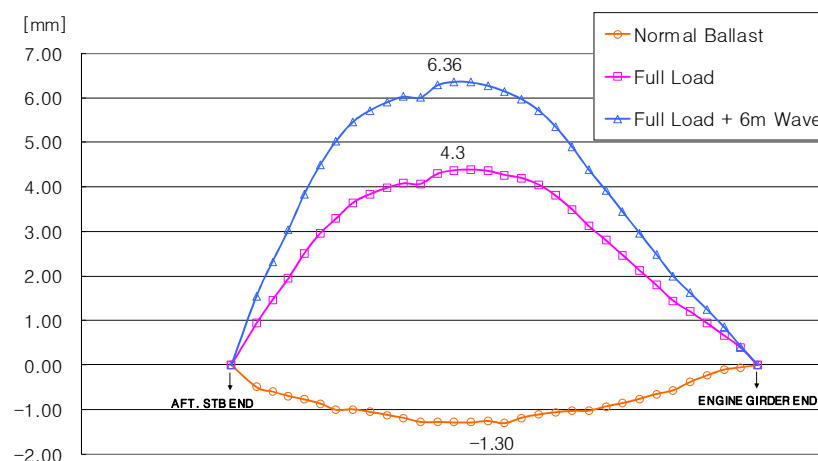
### 3. Shaft system analysis

The exothermic problem in the shaft is generated during engine running state. Thus, propeller load, heat effect, and hull deflection at the engine running state are considered. For ship loading conditions, besides the typical loading conditions i.e. normal ballast condition and full load condition, full load with 6m wave condition is considered additionally.

#### 3.1. Hull deflection

There have been a number of studies on hull deflection and shaft analysis as follows: a study on hull deflection using the finite element analysis results [2], a study on construction and application of hull deflection database [3], and a study on approximate curve of hull deflection [4]. Recently, most studies on shaft analysis consider hull deflection. In this study, shaft system analysis was conducted considering hull deflection using the finite element analysis (FEA).

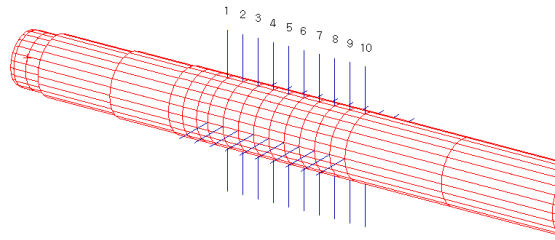
The relative hull deflection calculated using FEA is shown in figure 2. This relative deflection is obtained by fixing the end location of the rear part of the after STB and the location of girder in the main engine to “0”.



**Figure 2.** Relative hull deflection.

#### 3.2. Bearing contact model

Gap element (contact element of NASTRAN) was applied to calculate the compressive pressure in the after STB. As shown in figure 3, total numbers of 40 elements, 10 each in the vertical and horizontal directions of the ship, were arranged. The location of the element in the stern side was set to No. 1 followed by assigning a number sequentially to distinguish the elements.



**Figure 3.** Gap element for contact simulation.

#### 4. Calculation of pressure

The Hertzian contact condition assuming half-elliptical pressure distribution along contact width was adopted in this study to calculate the local pressure, as used [2]. 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. In (2),  $\nu_1$  and  $\nu_2$  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),  $D_1$  is bearing inner diameter and  $D_2$  is shaft diameter.

$$Q = \frac{\pi b^2 E^*}{2R} \quad (1)$$

$$E^* = \left( \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right) \quad (2)$$

$$R = \frac{D_1 D_2}{D_1 - D_2} \quad (3)$$

The maximum pressure is expressed as in (4).

$$P_{\max} = \frac{bE^*}{R} = \sqrt{\frac{2}{\pi}} \sqrt{\frac{QE^*}{R}} \quad (4)$$

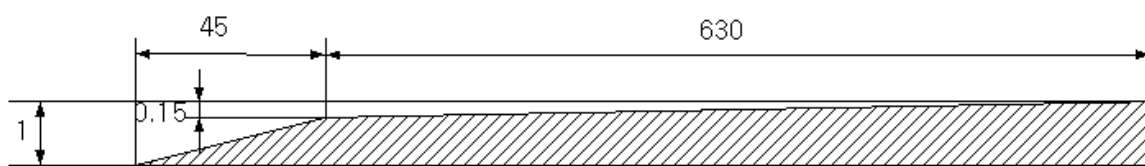
The reaction forces computed in the shaft system analysis are substituted in (4) to get the maximum local compressive pressure. If the obtained value does not exceed 100 bar, which is the maximum allowable value of the material, a shaft system that installs only the after STB can be applicable. Table 1 summarizes the compressive pressure distribution of the after STB at each loading condition according to the shaft system analysis.

**Table 1.** Pressure distribution of after STB.

Station	Maximum pressure [bar]		
	Ballast	Full load	Full load with 6m wave
1	119.6	116.8	112.9
2	94.3	91.1	80.2
3	69.6	51.5	55.9
4	45.6	0	0
5	0	0	0
6	0	0	0
7	0	0	0
8	12.2	27.4	23.5
9	28.7	64.0	71.9
10	64.0	84.9	98.7

The calculation results showed that all pressures at the rear part in the after STB exceeded 100 bar at all loading conditions, and the pressure at full load with 6m wave condition was also close to the maximum allowable value at the front part of the bearing. The shaft analysis was conducted with some possible offsets and only the result under the full load with 6m wave condition did not satisfy the goal. Although there was some relieving effect of load burden on the after STB by moving the intermediate shaft bearing into the stern side, it did not obtain the effect that lengthened the length of the after STB. This was because the center of the after STB and the shaft did not contact with each other. To solve this problem, a measure that builds a partial slope was applied [2, 5]. This method aims to reduce the local pressure by distributing contact evenly with the shaft by means of a slope in the white metal.

The rear part of the bearing was assumed to have a partial slope and the shaft analysis was conducted. Currently, shipbuilding yards can process up to 0.01 mm precision, and 0.15mm of partial slope height was assumed.



**Figure 4.** Partial slope of white metal in after STB.

Table 2 summarizes the compressive pressure distribution of the after STB at each loading condition when 0.15 mm partial slope is applied. The calculation results showed that the maximum local pressure of the after STB did not exceed 100 bar at all loading conditions. The maximum pressure (94.5 bar) was generated at the ballast condition, and large pressure was generated at No. 4 location overall. This was due to the high height of the partial slope so that load burden was significantly exerted on the center part of the bearing.

**Table 2.** Pressure distribution of after STB with partial slope height of 0.15 mm.

Station	Maximum pressure [bar]		
	Ballast	Full load	Full load with 6m wave
1	68.6	63.2	56.1
2	68.7	64.8	48.7
3	77.9	63.0	67.3
4	94.5	83.2	80.5
5	57.2	55.0	57.9
6	27.8	26.9	14.1
7	0	0	18.7
8	0	24.5	20.0
9	0	52.4	61.8
10	35.4	68.9	85.4

The analysis results showed that even if the forward STB was removed, a problem of shaft heating can be solved by making the contact surface between the after STB and the shaft distribute evenly, and a good flexible shaft system can be constructed. Furthermore, lengthening the length of the after STB by 11% compared to that of existing ship, and moving the intermediate shaft bearing into the stern side to lessen the loading burden on the after STB contributed also to the possibility of removal of the forward STB.

## 5. Conclusions

This study aimed to ensure shaft flexibility against hull deflection with regard to the proposed shaft system for VLCC. The applicability of the shaft system by removing the forward STB and installing only the after STB was investigated, and the following conclusions were made.

The movement of the intermediate shaft bearing into the stern side contributed to the effect of relieving the load burden on the after STB, which was increased due to the removal of the forward STB.

Only lengthening the length of the after STB did not achieve the reduction effect of local compressive pressure but the additional construction of partial slope achieved even distribution of compressive pressure over the entire bearing length.

The several additional measures proposed in this study verified that the forward STB was able to be removed from the shaft system of existing VLCC and a significantly visible effect can be expected due to the simplification of the system.

## 6. References

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