Investigation on Effective Support Point of Single Stern Tube Bearing for Marine Propulsion Shaft Alignment

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ABSTRACT

Over the last few decades, oil tankers and bulk carriers, which preferably adopting the concept of a single stern tube bearing, have been subjected to the mechanical damages on propulsion shaft bearings since such a concept overly requires a single bearing to support the full load of the propulsion shaft. This paper was to fundamentally address the problem of current practices of aligning a single stern tube bearing for commercial ships by determining the effective supporting point to balance the load of the propulsion shaft properly. Since selecting a case ship subjected to the damage on the stern tube bearing, it investigated the actual cause of the damage, thereby finding a practical way to enhance the sustainability of the propulsion shaft system using the single stern tube bearing. Computer-aided simulations using the finite element analysis as well as on-site measurements were carried out. Research findings revealed that the ignorance of the relative slope angle

between the propulsion shaft and the aft stern tube bearing had led to misalignment, which resulted in adverse effects on the stability of the shaft system, and consequently damaged the stern tube bearing. Results of the analysis suggested that the degree of slope be taken into account when estimating the effective supporting point of the bearing. Finally, this paper pointed out that the establishment of a shaft installation guideline considering the effect of the shaft slope, thereby preventing wiping damage of the aft stern tube bearing would be an urgent task.

Keywords: Shaft alignment, Gapsag method, Relative slope, Misalignment, effective support point

Symbol list

$i = 1, \cdots$

s_i	Sag [mm]
g_i	Gap [mm]
ξ_i	Deflection of flange end [mm]
θ_i	Angle of flange end [rad]
D_i	Flange diameter [mm]
PS	Propeller shaft
ASTB	After stern tube bearing
SW	Shaft weight
IB	Intermediate bearing
IS	Intermediate shaft
TS	Temporary support bearing
FW	Flywheel
σ_d	Dynamic yield stress
σ_y	Static yield stress
$\dot{\varepsilon}$	Equivalent plastic strain ratio

D and P constants in the Cowper-Symonds rate enhancement formula (D= 40.4 and P=5 for mild steel while D=3,200 and P=5 for HT steel).

1. Introduction

1.1. Background

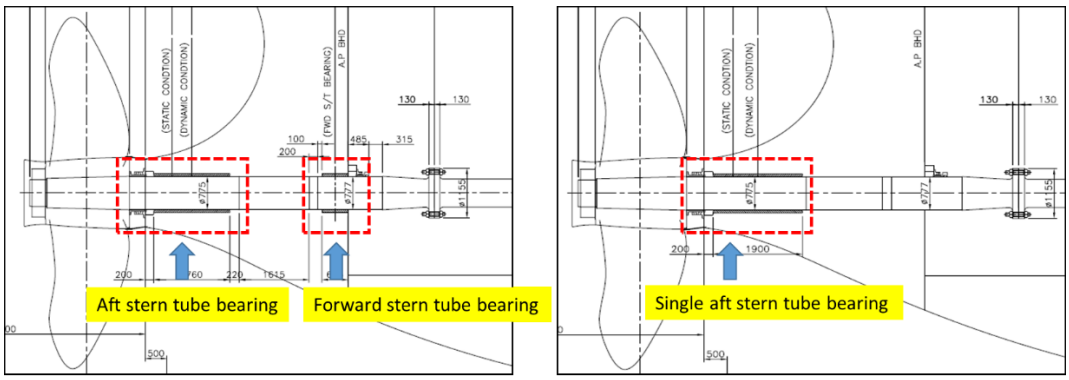
When installing the propulsion shafts on-board, needless to say, a correct alignment is crucial to ensure the stable and proper load distribution to the shaft supporting bearings. An error may cause an uneven load distribution on the bearings, consequently leading to abnormal abrasion, over-or-unloaded state, overheating of a specific bearing and/or breakage of the reduction gear [1].

Since last 1950s when the importance of the propulsion shaft alignment has been highly acknowledged particularly for the US Naval ships, there have been vigorous studies [2-3], especially between the late 1960s and early 1970s, through which a theoretical basis for calculating the optimal position of individual shaft bearings was established [4-8].

For conventional types of commercial ships, as shown in Fig. 1 (a), it was a common practice to place two shaft bearings on both sides - the forward and the aft - of the stern tube in order to distribute the mechanical load imposed by the weights of the propeller and the shaft to each bearing properly.

In this conventional practice, an excessive local load on the bearings can be prevented by calculating the relative minimum contact angle [9] that is used to evaluate the minimum contact area between the propulsion shaft and the supporting bearings. As following this way, the shaft stability can be checked simply by a visual inspection on whether the propulsion shaft is in proper contact with the forward stern tube bearing by jack down force.

As striving to achieve simpler shipbuilding process and higher shaft stability against hull deformation cause by ship draft change, shipyards have favourably adopted a new concept of a single stern tube bearing as shown in Fig. 1 (b), with which the forward bearing is removed and the aft bearing is only intended to support the propulsion load into modern commercial ships, particularly for oil tankers and bulk carriers.



(a) (b)

Fig. 1. Traditional two stern tube bearing design versus single stern tube bearing design (*by courtesy of Shipyard*).

This concept, however, may increase the risk of damaging the shaft and/or the bearing due to the over-reliance on the aft stern tube bearing that alone needs to support the overall weight of the propeller and shafting system. Several accidents associated with this issue have been reported [10-11].

Moreover, the case ship selected in this study was one of the 30 series vessels (all of them are identical) constructed in a single shipyard. When this study was initiated, 17 vessels were already delivered to service and there have been some issuable reports; 6 out of them were subjected to stern side noise and 2 out of them were reported for damages on stern tube bearings. In addition, oil sampling analysis data revealed that tin element had continually increased in the bearing main material composition.

Given that 11 out of 17 ships were subjected to the issues related to shaft alignment at the time of this study, it can be seen that incidental rate for single stern tube bearing system was as much as 64 %.

According to the accident reports, it appears the conventional practice of aligning propulsion shaft is inadequate for the single stern tube bearing system. Consequently, the demand for developing an appropriate shaft alignment method suitable for the single stern tube bearing system to prevent similar accidents has risen.

1.2. Limitation of Gapsag method

In order to obtain the correct values for the shaft alignment at the design stage, several conditions have been suggested to be taken into account; the change in the bearing reaction force at different cargo loading conditions, the mechanical deformation of the main engine

due to the long-term operation, the bending moment caused by the propeller thrust power and the elastic deformation of bearings imposed from vertical direction, etc. [12-18].

Meanwhile, the gapsag method has been commonly used to check whether the shaft is aligned at the exact position in accordance with the design values before assembling the shaft in the ship building process [19]. In this principle, when connecting departed pieces of the propulsion shaft together, the first piece is regarded to be the reference. Using the gapsag values derived from deflection and the angle of flange ends determined by the shaft weight and jack down force, the shaft pieces are to be connected one by one in serial order as shown in Fig. 2 and Eqs (1) and (2) [1].

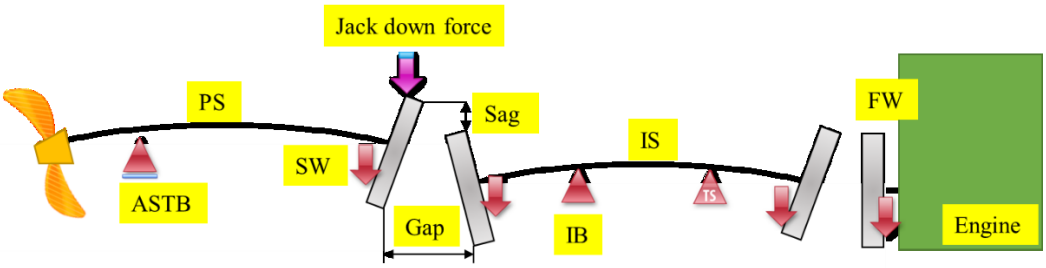


Fig. 2. Definition of gap and sag.

$$\left.\begin{aligned} s_1 &= \xi_2 - \xi_1 \\ s_2 &= \xi_4 - \xi_3 \\ &\vdots \\ s_N &= \xi_{2N} - \xi_{2N-1} \end{aligned}\right\} \tag{1}$$

$$\left.\begin{aligned} g_1 &= (\theta_2 - \theta_1)D_1 \\ g_2 &= (\theta_4 - \theta_3)D_2 \\ &\vdots \\ g_N &= (\theta_{2N} - \theta_{2N-1})D_N \end{aligned}\right\} \tag{2}$$

For the two stern tube system, the jack down force is a downward load imposed on the propeller shaft. Therefore, using the try and error method, the designers provide arbitrary loads on the position of the jack down force as shown in Fig. 2. An optimal jack down force

is determined at the point where the load imposed on the forward stern tube bearing (FSTB) is closest to zero. This ensures that the propeller shaft is in simple contact with the FSTB [1].

On the other hand, since the forward stern tube bearing (FSTB) does not exist in the single stern tube bearing system, the load value of the FSTB cannot be obtained. Instead, the optimal jack down force is determined at the value where the propeller shaft defection value at the position of the imaginary FSTB approaches zero.

Given that the shaft position values obtained from the gapsag method are determined by the positions of the two flanges to be connected, if the jack down force is different as shown in Fig. 3, the reference shaft (propeller shaft) is deemed to be established at an different angle. This case may influence on the alignment of the rest of the shaft pieces successively.

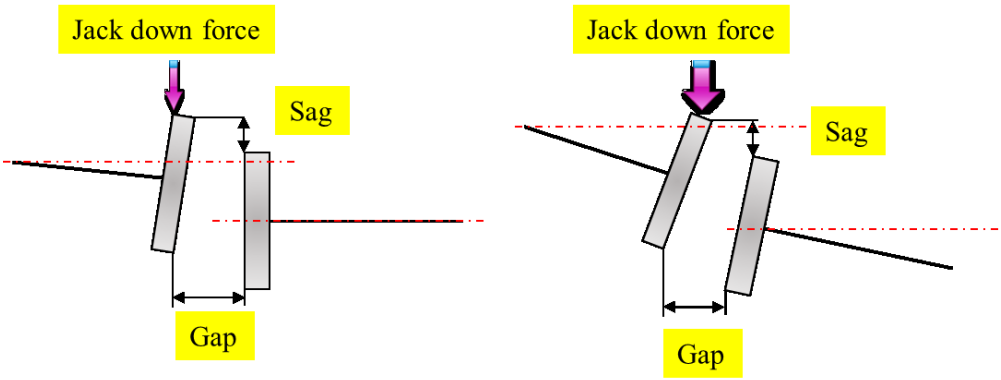


Fig. 3. Jack down force dependent deviation of shaft line under same gapsag value.

In this case, even if the reaction force of the bearing measured by the jack up method [20] after the completion of shaft assembly satisfies the design values, the change in the relative inclination angle between the shaft and the aft stern tube bearing cannot be detected and/or estimated. The jack up method has been widely used, thanks to its simplicity as a direct method of confirming the bearing reaction as shown in Fig. 4. To measure the bearing reaction, hydraulic jacks are placed in close proximity to the bearing. As a result, the whole shaft system may be subjected to an unstable condition.

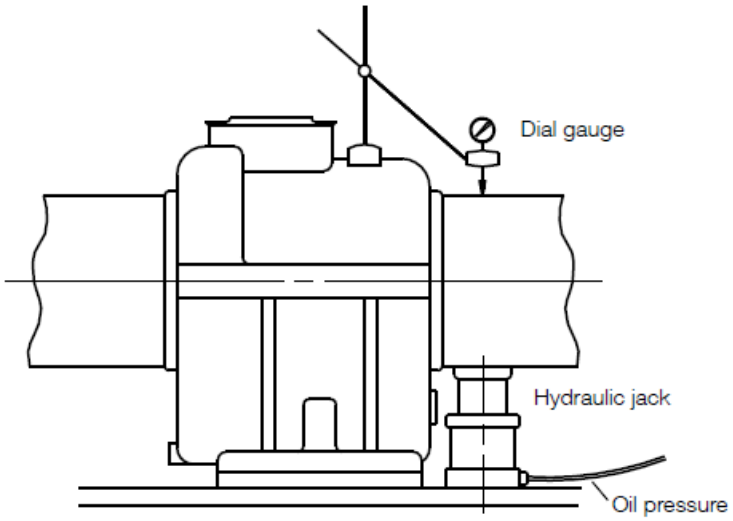


Fig. 4. Jack up example at intermediate shaft bearing [20].

1.3. Past research & objectives of this paper

Despite voluminous guidelines on shafting arrangement, a few academic research work in this field have been introduced. Shi et al. [21] and Murawski [22], have investigated the impact of hull deformations on shafting alignment, striving to find an optimized shafting alignment for marine vessels. Huang et al. [23-24] carried out vibration analysis of ship propulsion system by means of FEA in order to investigate the torsional and transverse vibration under idling and loaded conditions in accord with various speeds.

However, none of past research directly address the issues with which this paper attempt to deal.

In this context, this paper was to systemically investigate the adequacy or inadequacy of conventional practices of aligning the propulsion shaft for the concept of a single stern tube bearing as providing recommendations for ensuring the shaft alignment more stable, therefore, making sure of the proper load distribution on the shaft bearing. To achieve this, it investigated a case ship experiencing a stern tube bearing damage during the service and addressed the cause of the damage through the theoretical estimation of the effective points

of the stern tube bearings using finite element analysis (FEA). In addition, it examined the deviation between the initial shaft position values measured at the construction stage and the current values.

1.4. Approaches adopted

Fig. 5 shows the study outlines which mainly consist of four steps: investigation of incident on case ship, calculation of reaction force, calculation of effective support point and contact analysis. Each step uses different methods as presented in the figure.

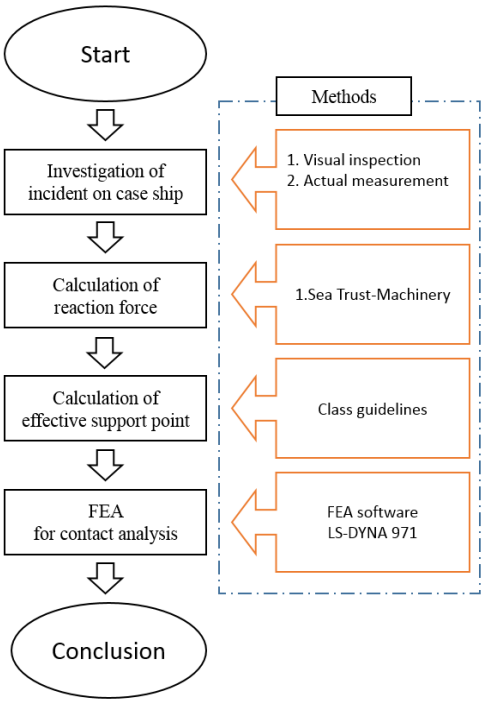


Fig. 5. Study outlines.

2. Case ship

As the case ship, this paper has selected an oil & chemical tanker with 50,000 dead weight tonnage (DWT) which is a common type of commercial vessels with numerous construction records particularly in Korean shipyards. However, such medium range tankers have been subject to various incidences during sea-trial or actual voyage. The

selected case ship was also reported for an incident on having an excessive heat on the stern tube bearing during the service just one year after delivery.

Table 1 presents a brief specification of the case ship, while Fig. 6 shows the damaged aft stern tube bearing; a wiped bearing surface where surface rubbing, melting and smearing is evident as shown on the figure is attributed to the overheating of the bearing that may be due to a variety of causes, such as; operational overload, loss of clearance and misalignment.

Table 1. Specification of case ship and shafting system *(by curtesy of shipyard)*.

50k DWT oil/chemical tanker	
L × B × D (m)	174.0 × 32.2 × 19.1
Main engine	Type: MAN Diesel & Turbo 6G50ME-B
	MCR: 7,700kW × 93.4rpm
	NCR: 5,344 kW × 82.7 rpm
Propeller	4 blade fixed pitch
	Diameter: 6,600 mm
	Material: Ni-Al-Bronze
	Mass: 18,200 kg
	Cap & nut mass: 1,538 kg
Flywheel	Mass: 11,207 kg

L: length, **B**: breadth, **D**: depth, **MCR**: maximum continuous rate, **NCR**: nominal continuous rate

Fig. 6. Wiping damage on aft stern tube bearing.

3. Calculation of reaction forces

In general, bearings are over-heated when the reaction force (load) to the bearing exceeds the allowable level. Given the fact, in order to reproduce from a trilateral view whether initial design values determined by shipyard would be adequate or inadequate, the bearing reaction forces at different shaft position were calculated independently using different shaft alignment calculation software, 'SeaTrust-Machinery', developed by Korean Register [25] as shown in Fig. 7. The analysis results are summarized in Table 2.

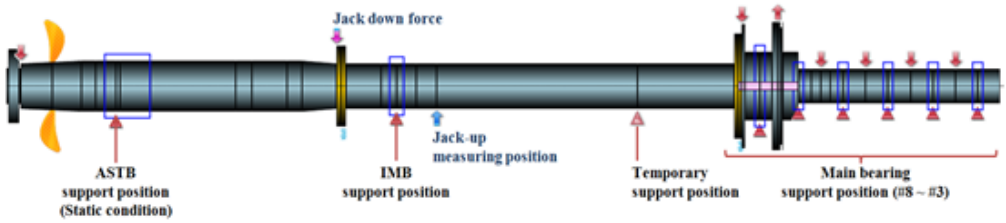


Fig. 7. Independent shaft alignment calculation model.

Table 2. Comparisons the result of bearing load hot static (100% propeller immersed).

Bearing	Reaction force (load) [kN]			Distance from propeller shaft left end [mm]
	Shipyard calculation	Independent calculation	Max. Permissible Load [kN]	
ASTB	248.23	256.94	368	2,192
IB	76.27	72.07	160.2	7,782
MB8	77.29	82.32	336	15,027
MB7	69.48	65.47	336	15,975
MB6	128.22	128.32	336	16,689

ASTB: after stern tube bearing, **IB**: intermediate shaft bearing, and **MB**: main engine bearing

The results of analysis reveal that the deviation of the bearing loads between the shipyard calculation and independent calculation using 'SeaTrust-Machinery' was negligibly small. It also confirms that there had been no calculation error at the design stage as all reaction forces to all shaft support bearings had been kept within the tolerable levels [20].

In order to examine whether there was an error during the shaft installation work, the propulsion shaft was opened up and the offset of the shaft supporting bearing was measured by laser sighting as shown Fig. 8. The results of the measurement were compared with the shipyard designed shaft deflection curve in Fig. 9 where the reference line (a virtual line) drawn in parallel between the rear end centre of the stern tube and the engine room side was used as a guideline for installing the shaft.

Fig. 8. Bearing offset measurement by laser sighting.

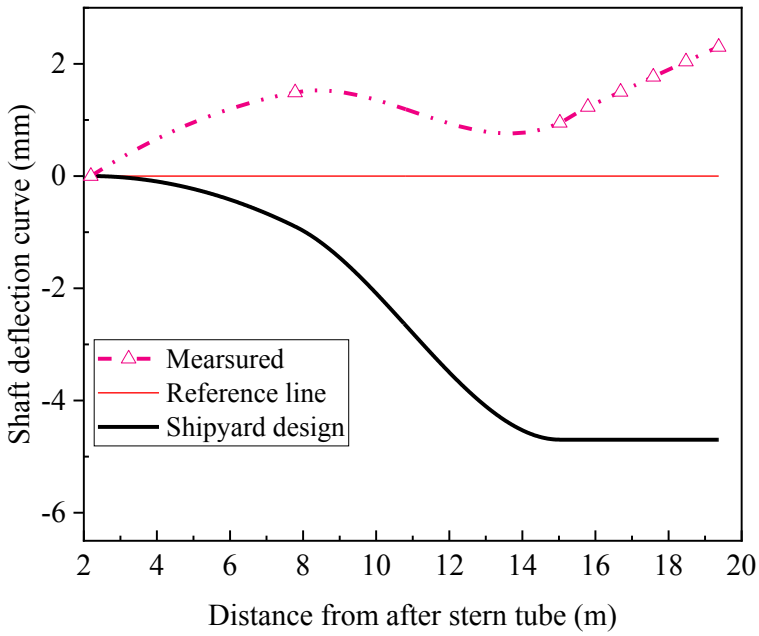


Fig. 9. Deviation between measured and designed shaft deflection curve against reference line.

It reveals that the offset of the IB was originally designed to be placed below the reference line, however, it was actually gauged at 1.49 mm above that line. Similarly, the offset of MBs was designed to be positioned straight 4.7 mm below the reference line, but they were

arranged tilt upwards from 0.95 mm up to 2.3 mm above. The bearing offsets are summarized in Table 3.

Table 3. Comparisons the bearing offset.

Bearing	Offset [mm]		Distance from propeller shaft left end [mm]
	Measured	Shipyard designed	
ASTB	0	0	2,192
IB	1.49	-0.9	7,782
MB8	0.95	-4.7	15,027
MB7	1.23	-4.7	15,975
MB6	1.5	-4.7	16,689
MB5	1.77	-4.7	17,853
MB4	2.04	-4.7	18,477
MB3	2.3	-4.7	19,371

ASTB: after stern tube bearing, **IB**: intermediate shaft bearing, and **MB**: main engine bearing

In practice, it is true that the MBs can be mounted with a slight tilt as an effort to optimize load distribution of the MBs to take into account of potential errors caused by several reasons; the deviation between the two-dimensional crankshaft model and the actual crankshaft shape, the engine-free sagging effect caused by deformation of hull before launching and the potential hull deformation by ship draft after launching [26].

Nevertheless, it was found that the level of tilt measured for the case ship far exceeded than practice, thereby it is perceived that the conventional guideline for the shaft alignment is somewhat inadequate. Given the fact that the change in the shaft line more or less affects the magnitude of the reaction force on the shaft support bearing, this paper re-calculated the bearing reaction force based and the results are presented in Table 4.

Table 4. Deviation of bearing load at designed and measured.

Bearing	Reaction force (load) [kN]		
	Measured	Designed	Permissible
ASTB	259.02	257.47	368
IB	62.44	68.07	160.2

MB8	56.06	64.84	336
MB7	134.22	77.94	336
MB6	73.29	128.02	336

Regarding the ASTB and the IB, the effect of the bearing offset change on the bearing reaction force was not significant. Although the effect of the offset change on the bearing reaction forces on the MBs were relatively high, their values were still placed within the acceptable levels as guided by the main engine manufacturer [20]. It was found that the results of the analysis were inconclusive to confirm the axial change to be the direct cause of stern tube bearing damage.

4. Calculation of effective support point of stern tube bearing

Conventional procedures of shaft alignment assume, as shown in Fig. 10 where the propulsion shaft is overall supported by up to three point rigid bodies of long journal bearings, such as the ASTB, as simply regarding the positions of the long journal bearings to be the effective support points.

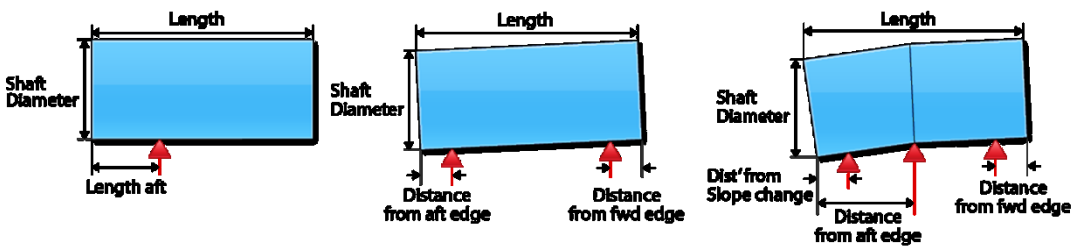


Fig. 10. Application of support point of long journal bearing in conventional procedures of shaft alignment.

However, in reality, it is very difficult to predict the appropriate bearing support points as they may depend on various parameters: the shaft weight and the elasticity of the shaft and bearing in static conditions, the propeller eccentric thrust power and the change in oil film distribution between the shaft and the bearing in dynamic conditions.

On the other hand, using the projected area method, as shown in Fig. 11. , the reaction force at the effective supporting point of the bearing is divided by the projected area with respect to the bearing length, the pressure applied to the bearing [27]. This method, however, has a tendency to over-estimate the stiffness of the bearing [28]. In addition, the method is not a reliable approach to estimate the contact area between the actual shaft and the bearing, as a result, it is difficult to determine whether excessive local load may possibly occur, leading to bearing damage.

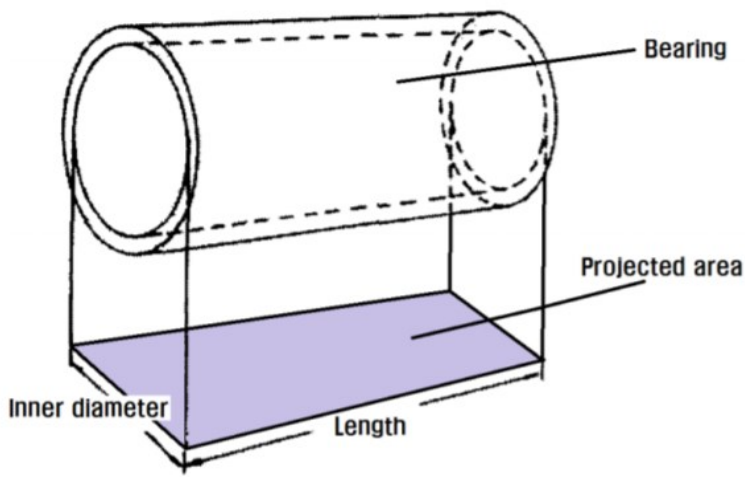


Fig. 11. Projected area for calculation of mean pressure of bearing [27].

To make up for the limitations of this conventional method, several approaches have been introduced. Especially, Lloyd Register (LR) [29] suggests effective support points of the shaft in static and dynamic conditions based on extensive database and shipyards also apply different approaches to determine effective support points in consideration of ship type and size based on their accumulated experience. As shown in Fig. 12. , LR, the first Classification Society to introduce this requirement, and other classification societies have provided safety guidelines recommending that the relative inclination angle between the propeller shaft and the stern tube rear bearing be not to exceed 0.3 mrad (a slope of 0.3 mm

per 1 m) in order to ensure the sufficient contact area between the propeller shaft and the bearing [30].

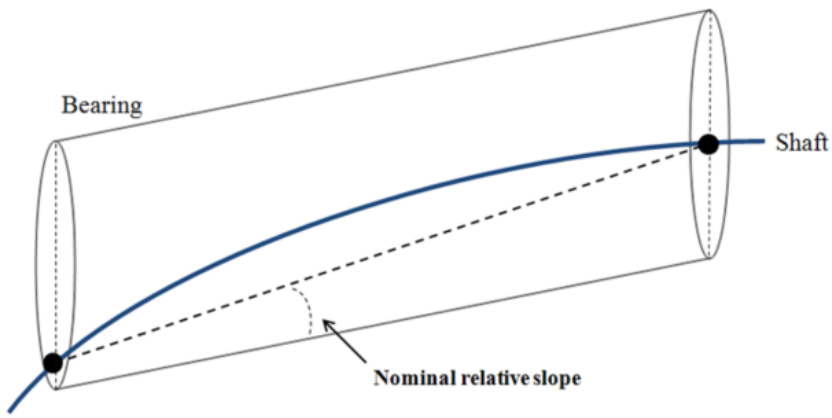


Fig. 12. Definition of the relative slope.

This paper applied these class guidelines to the case ship and carried out the on-site measurement. As a result, it was revealed that there was no significant difference in surface pressure calculated based on bearing reaction force and projected area. However, a remarkable fact was found that the relative inclination angle at the effective support point of the stern tube bearing was measured at 0.637 mrad, which far exceeds the reference value of 0.3 mrad.

The results imply that the reference shaft was raised to the upper right than the design value for some reason, thereby the relative inclination angle between the stern tube bearing and the propeller shaft was upward as well. Therefore, it can be assumed that there was a problem in the jack down process for placing the propeller shaft to the design position.

Furthermore, as mentioned earlier, in case of a ship without the forward stern tube bearing, there is a high possibility that a shaft alignment error occurs. Thus, this finding indicates that the shaft load had been overly concentrated on the ASTB under certain sea conditions.

5. FEA

In order to investigate the contact effects between the shaft and the bearings and to find a way to minimize the misalignment error in design and construction of propeller shaft, the contact analysis using FEA was carried out as illustrated in Fig. 13.

Fig. 14 shows 3D modelling of the propeller shaft and the after stern tube bearing.

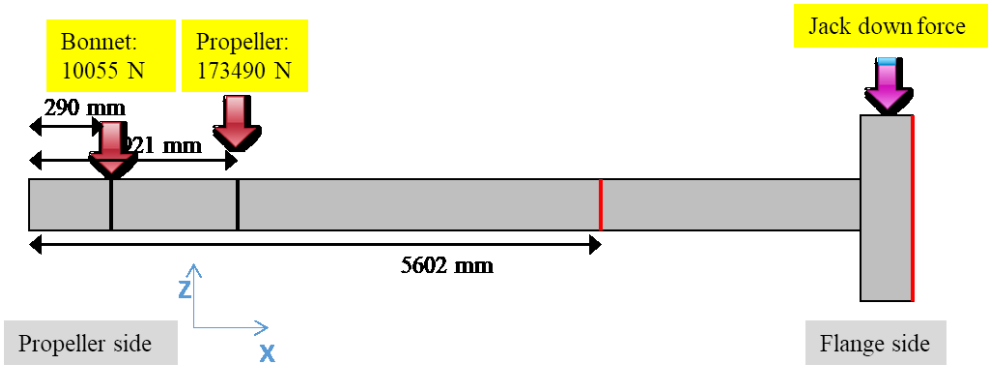


Fig. 13. Modeling of contact analysis.

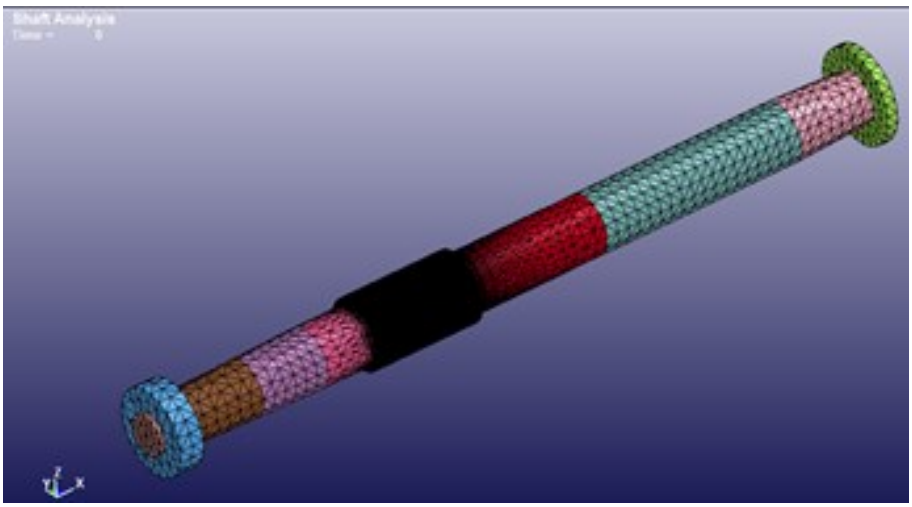


Fig. 14. Modeling of contact analysis.

The analysis was performed with LS-DYNA 971, a 3D FEA software, which is a multi-purpose explicit and implicit finite element program extensively used to analyse the nonlinear response of structures. In particular, it is an expert for fully automated contact analysis for a wide range of material models [31]. Given that explicit analysis is suitable

for large deformation analysis such as car crash analysis and dropping test, this case study adopted the implicit FE analysis which is suitable for the following case study conditions:

- Contact between the shaft and bearing occurs by gap at various times
- The shaft motion is not subjected to big deformation
- Long-term load is applied to the bearing

5.1. Coordinate system and unit systems

A conventional right-hand axis system was employed as a global coordinate system. The global coordinate system for this model was defined as follows:

- Origin : An intersection point of after end, centerline and bottom planes
- X-axis : Longitudinal direction, positive stern to bow
- Y-axis : Transverse direction, positive center to port
- Z-axis : Vertical direction, positive bottom to deck

Units applied to finite element analyses were defined as follows:

- Force : Newton [N]
- Length : millimeter [mm]
- Mass : tonne [ton]
- Time : second [sec]

Therefore, following physical quantities were defined as follows:

- Stress(Pressure) : MPa
- Mass density : ton/mm^3

5.2. Mechanical properties of steels

Mechanical properties for standard steels applied in this analysis are shown in Table 5. The values for mass density, elastic modulus and poisson ratio, hardening exponent and the material constant are commonly applied to all standard steels, whereas the higher grade steel has the higher initial yield stress. For material constant, steel 235 is significantly lower

than other steels.

Table 5. Elastic and plastic mechanical properties of steels.

Items	Nominal grade of steel			
	235	315	355	390
Mass density, ρ [ton/mm^3]	7.85E-09			
Elastic modulus, E [MPa]	206,000			
Poisson ratio, ν	0.3			
Hardening exponent, n	0.2			
Material constant, q	5.0			
Initial yield stress , σ_0 [MPa]	235	315	355	390
Material constant, D [/s]	40.0	3200.0	3200.0	3200.0

-. The friction coefficient of 0.3 was applied for contact with steel.

In a conservative way, the strain ratios leading to the rupture of the materials at different steel grade are determined based on the NORSOK Standards [32].

The dynamic yield stress in relation to strain ratio was used as the criteria to determine the occurrence of the plastic behavior. To calculate the dynamic yield stress, Cowper-Symonds rate enhancement formulae can be applied as shown below [33];

$$\frac{\sigma_d}{\sigma_y} = 1 + \left(\frac{\dot{\varepsilon}}{D}\right)^{1/P}$$

Table 6 shows the yield stress and rupture stain for steels while Fig. 15. presents the linearized stress-strain curves of the materials. As is generally known, changes in the strain rate of the mild steel are relatively sensitive to increased stresses, whereas HT 36 steels are reversed.

Table 6. Yield Stress and rupture strain (Units: MPa, %).

	Mild Steel	HT 32 Steel	HT 36 Steel
Yield stress (static)	235	315	355
Critical strain for rupture(*)	20.0	16.7(**)	15.0
Critical strength for rupture(*)	327	416	461

*) Strength and strain for rupture are calculated from fracture criteria of NORSOK standard N-004.

**) Rupture strain for HT 32 steel is not specified in NORSOK. Therefore, it is interpolated from yield stresses of mild steel and HT 36 steel. Since the plastic stiffness slope is also based on the yield stress in NORSOK, this approach is reasonable.

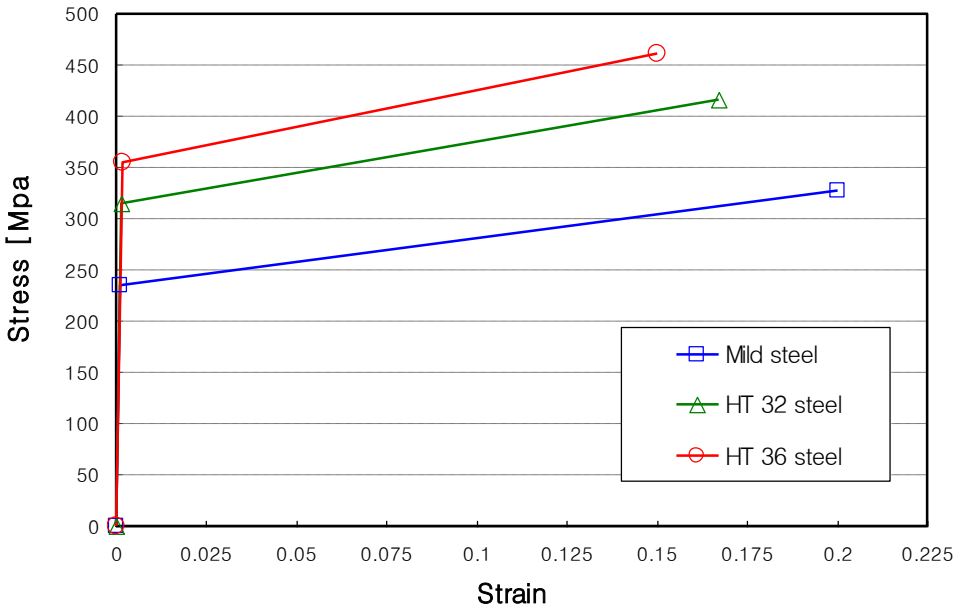


Fig. 15. Linearised Stress-strain Curve.

5.3. Analysis and discussion

The model of the stern tube bearings was prepared to analyse two conditions; (C1) where the slope was not applied to the bearing, equivalent to initial design theoretically calculated by shipyard. and (C2) where the inside of stern tube bearing was uniformly tapered with 0.3 mrad which exactly reflect the actual shaft condition in accordance with the approved drawing. In other words, the analysis was to point out the shipyard's improper practice where considering the slope of the bearing as a straight line in the theoretical calculation.

Given that fact that the shaft installation process is in a static condition, the influence of the oil film between the shaft and the bearing was disregarded. Fig. 16. shows the drawing of the single slope applied to after stern tube bearing.

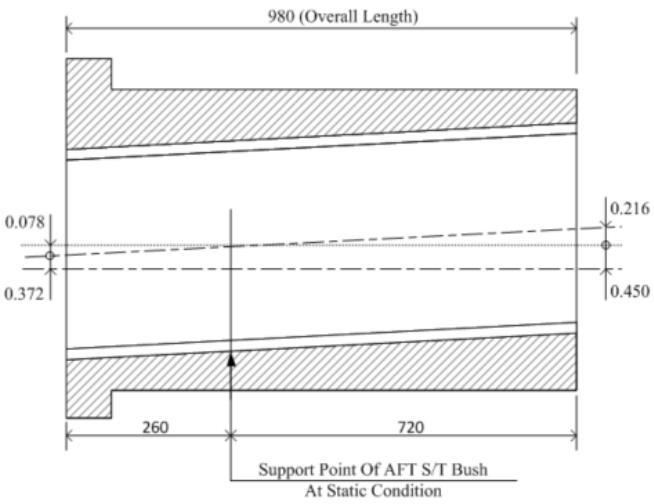


Fig. 16. Drawing of single slope applied to after stern tube bearing (*by curtesy of shipyard*).

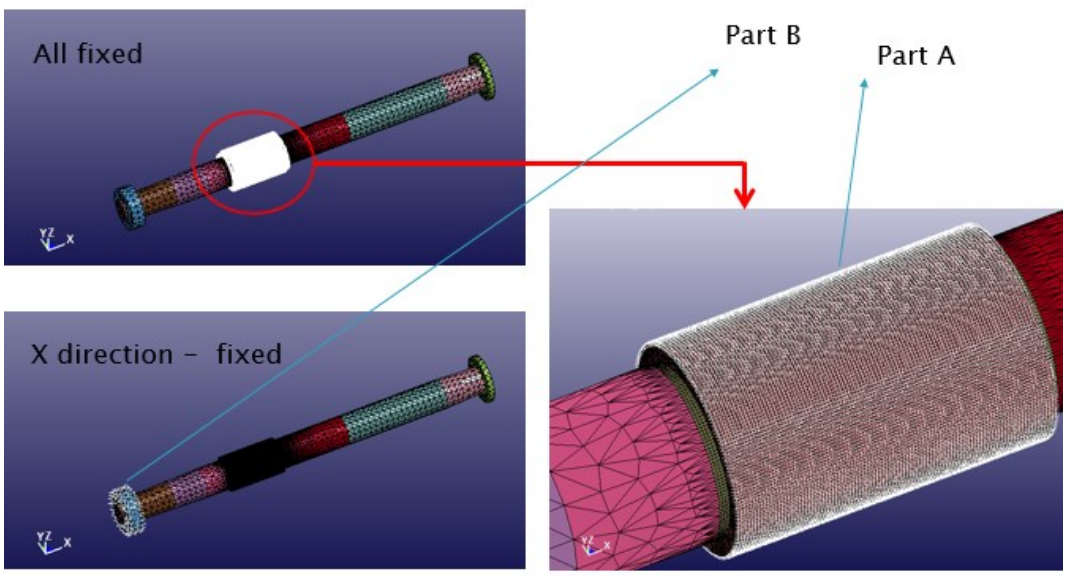


Fig. 17. The boundary condition of modelling.

Fig. 17 shows the applied boundary condition for the analysis. Here, the X direction represents the longitudinal direction (axial direction). Fixed constraint conditions were

applied to all surfaces of the bearing (Part A) which were physically rigid in order to prevent the bearing from rotating in any direction during the simulation.

The propeller model (Part B) was constrained only for the displacement of x direction. 3D solid element widely used for thick shell model was applied for this analysis. As shown in Fig. 18. , finer mesh was applied on the contact areas between the shaft and the bearings to investigate the contact effects precisely.

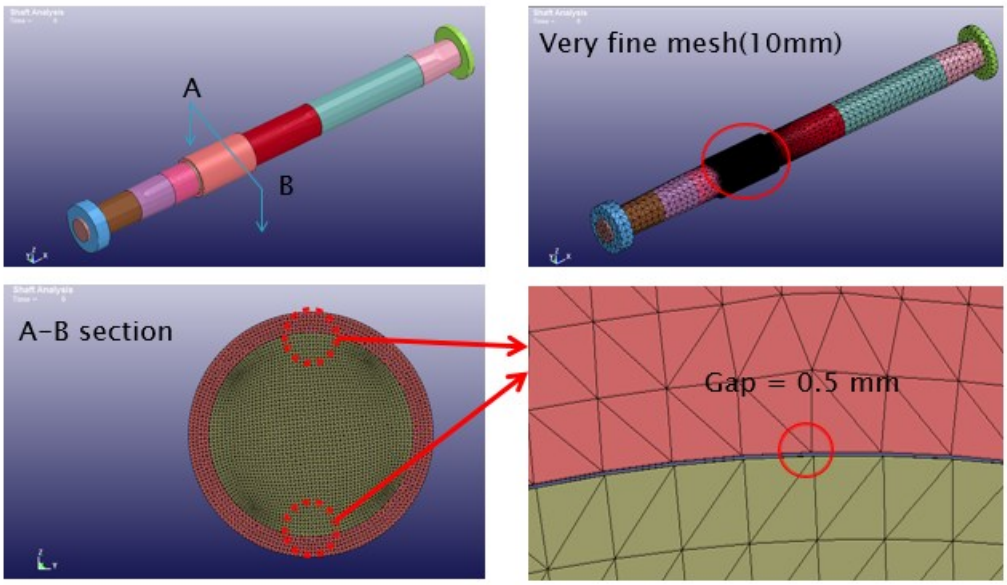


Fig. 18. Mesh condition of modelling.

Since the initial gap (clearance) between the shaft and the bearing was given to 0.5 mm, it is possible to estimate the change in the maximum pressure of the contact surface pressure and its stress over time under the different conditions. In addition, the jack down force to the flange of the propulsion shaft gradually increased from 0 up to 49 kN to make sure the load condition equivalent to the self-weight of the propeller, propeller bonnet and the propulsion shaft as shown in Fig. 19.

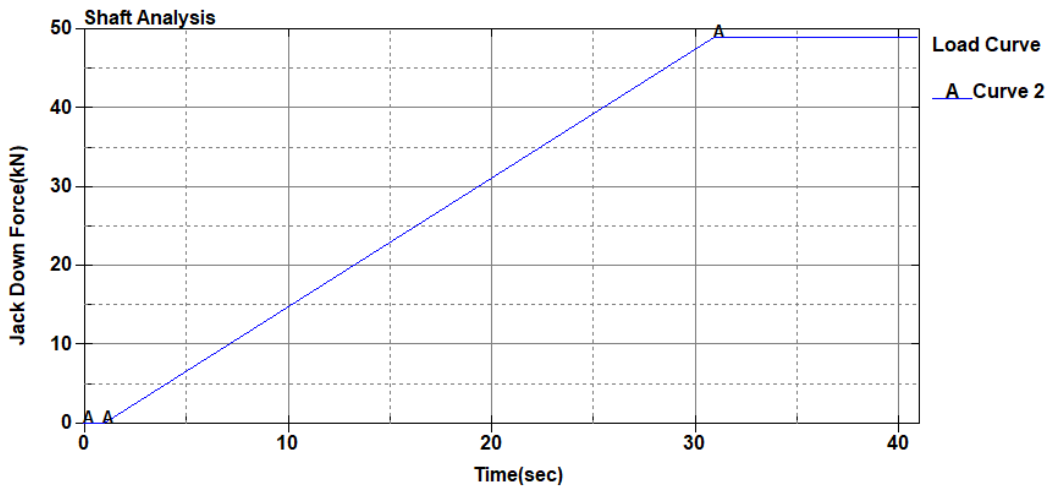


Fig. 19. Load curve at different time instance.

The results revealed that both C1 & C2 are in the same trend. In both cases of C1 and C2, before the start of the jack down, the load is concentrated on the rear end (propeller side) of the stern tube bearing due to propeller's self-weight. Once the jack down force was gradually increasing, the load became being distributed to both ends of the bearing. For the brevity of the paper, the results of C2 are only presented in Fig. 20. to Fig. 22. where the contour describes the von-Mises stress. In this case, the contact area between the shaft and the bearing became maximum as shown in Fig. 21. Then, when the jack down forces reached to the maximum level of 49 kN, the load has completely moved to the bearing fore end (flange side).

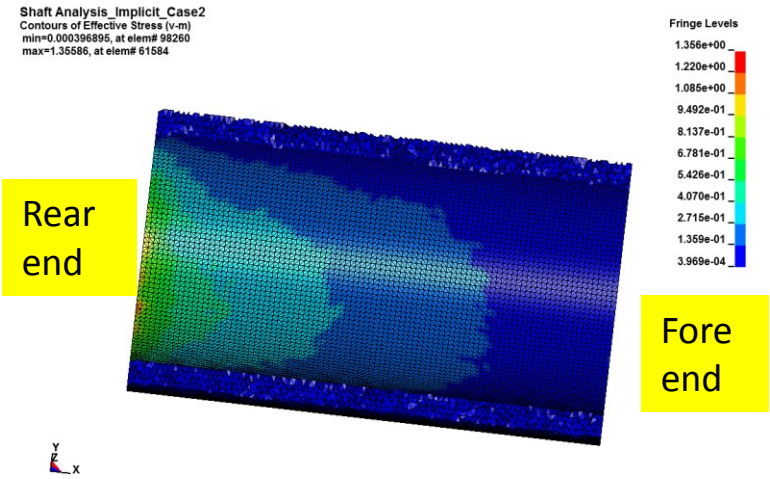


Fig. 20. Start condition; jack down of 0 kN.

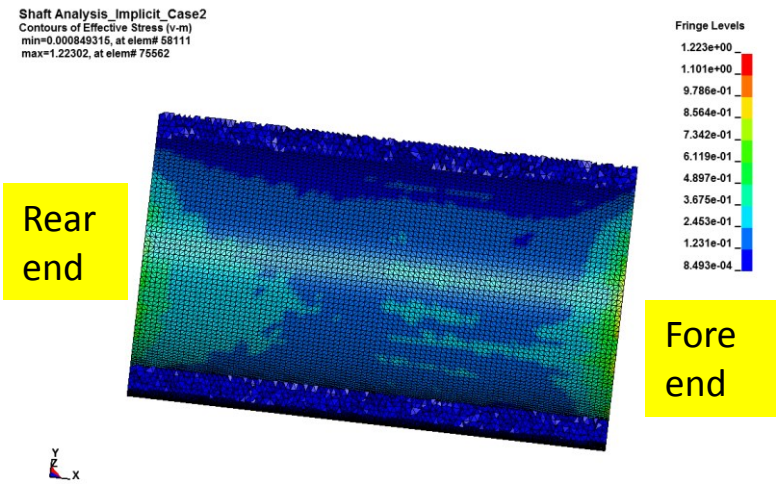


Fig. 21. Optimum condition during process of jack down forcing.

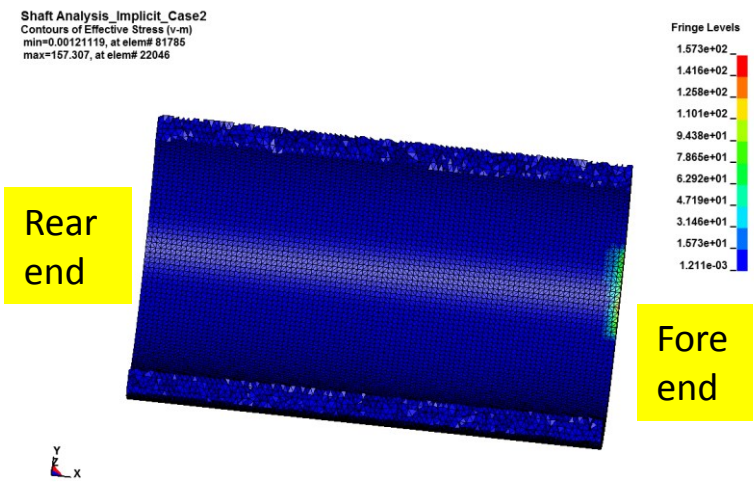


Fig. 22. Finish condition; jack down of 49 kN.

In terms of ensuring the integrity of bearings, it is ideal to distribute the load at both ends of the bearing equally. When using conventional shaft alignment method, it may be reasonable to consider calculating the optimal jack down force required to move the shaft from the both ends of the bearing (regarded as effective support points) to the position where the load is uniformly shared.

This technique can be an effective approach to prevent catastrophic conditions caused by excessive local load leading to bearing damage even if the machined tilt angle is slightly distorted in the process of fitting the stern tube bearing to the stern frame. This is because the effective contact area between the shaft and the bearing can be extended.

In the C2 condition (slope was applied), the propeller shaft when the jack down force of 49 kN was applied in the same manner as the initial shipyard calculation was used as a reference in order to predict the remaining axes mounted by the gapsag method. Then, the predicted shaft deflection curve was compared with the shipyard designed shaft deflection curve in Fig. 23.

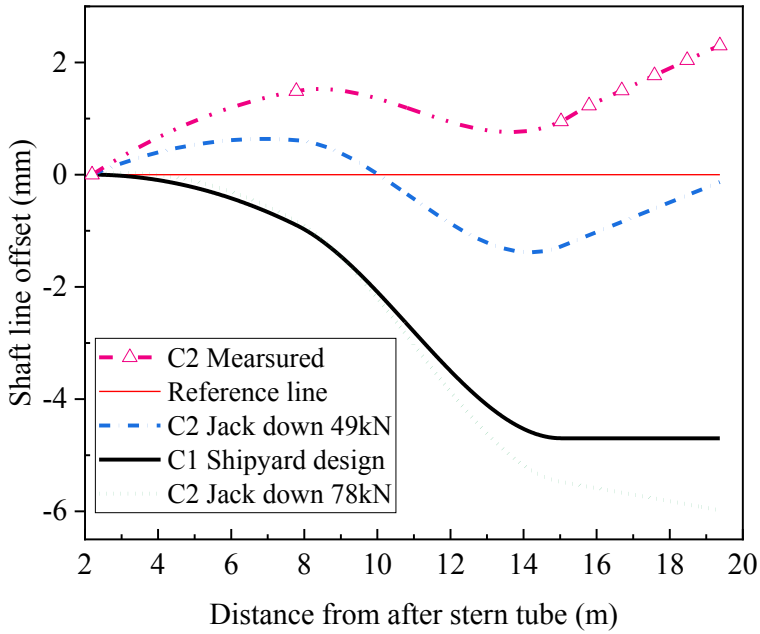


Fig. 23. Predicted shaft deflection curve against shipyard designed and measured.

As shown in the figure, regarding the trend of the predicted shaft deflection curve of C2, Jack down force of 49 kN is similar to that of the shaft deflection curve of C2 measured where the stern tube heating phenomenon. Therefore, it can be concluded that the slope of the inside of the stern tube bearing is directly associated with the axial change.

Moreover, it was found that the propeller shaft had been mounted, exceeding the estimated value (shaft deflection curve of C2 Jack down 49 kN) since the bearing machining tilt angle was over-increased by 0.06 mrad, compared with the designed angle (3.0 mrad), after pressurizing the stern tube bearing to fit into the stern tube.

As a single slope value of 0.3 mrad means a slope of 0.3 mm per meter, in consideration of the simple proportional conversion, the slope value in accordance with the propeller shaft length (6,682 mm) can be estimated 2 mm, thereby it can be perceived that the propeller shaft flange is inevitably placed higher than the case where there is no bearing inclination. This means that the C2 condition requires greater jack down force needs to be imposed for C2 than C1 when mounting the propeller shaft in place.

It should be noted that the axial deflection and deflection angle of the propulsion shaft flange increase in proportion to the jack down force when additional jack down force was applied, like C2 Jack down force of 78 kN, so that the shaft deflection curve could be rather downwardly directed relative to the design shaft deflection curve if the design gapsag value was directly used without correction.

In addition to calculating the jack down force for proper load sharing to set up the effective support points, research findings reveal that it is also important to apply the slope of the stern tube bearing to the shaft alignment calculation at the design stage.

The results of the analysis point out that the shaft alignment error between the design and the installation will be minimized if an appropriate procedure to check the position of the propeller shaft from the position of the forward seal box is established.

On the other hand, since the contact analysis was performed based on the case ship, it is still necessary to collect additional data obtained from a variety of case studies. Nevertheless, research findings provide ship designers and marine engineers with a significant insight into the safety of the propulsion shaft alignment. Therefore, it is believed that this paper will be used as a corner stone for the future regulatory framework.

It also needs to be mentioned that we considered the Cowper-Symonds formula on the assumption that the material may be subject to the nonlinear behaviour (large deformation) beyond the yield due to the influence of the contact (5 mm) gap at the time of the analysis. However, such behaviours were not observed, thereby it can be said that the possibility of the nonlinear behaviour for shafting systems can be ignored when performing a similar analysis in the future.

6. Concluding remarks

The present study has investigated the cause of the damage to the stern tube bearings. The detailed contact analysis using FEA and the on-site measurement were performed in respect

to obtain an insight into the effective support point of the ASTB. As results, the following conclusions were obtained.

(1) It was demonstrated that the existing verification process through the bearing reaction force measurement after the shaft assembly has a limitation as it is not capable of accurately estimating the change in shaft deflection curve. In addition, it was confirmed that the IB / MBs may be upwardly or downwardly arranged in a series if the remaining axes are mounted by the gapsag value without applying jack down force to position the propulsion shaft correctly.

(2) In terms of securing the stability of the shaft system, results of the FEA revealed that it may be a reasonable approach to regard the both ends to be the effective support points so that the uniform distribution of the load at both ends can be achieved.

(3) Based on the result of the shaft deflection curve estimation, when the jack down force had been applied it was found the propeller shaft could be arranged to the right-higher than the design value if the machining slope which might be changed after pressing the bearing is not considered.

(4) It is necessary to improve the design/construction method to secure the correct position of the propeller shaft and the intermediate shaft(s) through the complement of design / production. For this purpose, the establishment of a proper procedure to check whether the propeller shaft is located up/down/right/left in the position of propeller shaft forward seal box can be a practical means to minimize the misalignment the propeller shafts.

(5) Research finding reveals that, for the conventional method for propulsion shaft alignment, it is reasonable that the optimal jack down force required to move the shaft from the end of the support to the position where the load is uniformly shared can be calculated as regarding both ends of the bearing as effective support points. In this context, it can be believed that this paper has provided the insight for developing the proper procedure to check the adequate position of propeller shaft.

(6) For future work, it will be necessary to conduct further research on securing additional data for various ships in order to establish practical guidelines for ensuring the stability of shafts in a general way.

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