UDC 629.5.026:621.82

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Experimental Determination of Stern Tube Journal Bearing Behaviour

Preliminary communication

The most sensitive component in the propulsion shafting system is the aft stern tube bearing, which is exposed to heavy static and dynamic propeller loads exerted to the bearing surface by the propeller shaft.

The *Faculty of Mechanical Engineering and Naval Architecture*, *University of Zagreb*, and the *American Bureau of Shipping* intend to work on the development of the theoretical model of the stern tube journal bearing behaviour, which is proposed to be done in several stages. The initial stage is to consider nonlinear fluid structural interaction, and this is the theory shown in this article. Owing to the fact that the shaft journal in operation lies on a lubricating fluid film, which separates the journal in the bearing from the bearing liner, the theoretical nonlinear model of the journal bearing may provide more insight in the behaviour of the shaft inside the bearing and help us to determine a more realistic elastic line of the shafting. The *American Bureau of Shipping* (*ABS*) initiated a research project to measure actual interaction between the shaft and the stern tube bearing with the goal of applying obtained knowledge in development of an accurate analytical/numerical model for shaft-bearing analysis. The initial focus of the investigation was to capture the transient operation of the shaft inside the bearing and to observe the shaft's hydrodynamic lift during starting, stopping and reversing of the engine. Proximity probes were used to monitor the shaft bearing interaction with micron precision. The installed probes measure the distance between the shaft and the bearing at eight locations inside the aft stern bush and at four locations inside the forward stern bush, and utilize obtained data to define the shaft orbit and dynamics as it revolves in the bearing.

The results of presented measurements show that the shaft behaviour inside the stern bush is very different depending on the location inside the bush. Therefore, an analytical approach, which uses a simplified theory of shaft bearing interaction, provides no practical value since it neglects shaft dynamics and interaction between fluid and the structure.

Keywords: *propulsion shafting system, stern tube bearing, shaft alignment, proximity sensors, alignment condition monitoring, nonlinear model of radial journal bearing*

Eksperimentalno određivanje ponašanja rukavca vratila u statvenom ležaju

Prethodno pripoćenje

Jedna od najosjetljivijih sastavnica sustava vratilnog voda je stražnji statveni ležaj, koji je u radu izložen velikim statičkim i dinamičkim opterećenjima. Opterećenja potječu od brodskog vijka, a prenose se putem vratila brodskog vijka na kliznu površinu ležaja.

Fakultet strojarstva i brodogradnje, Sveučilišta u Zagrebu i *American Bureau of Shipping*, USA, namjeravaju zajednički raditi na razvoju teorijskog modela koji će opisivati ponašanje rukavca vratila u statvenom ležaju. Razvoj modela planira se u nekoliko faza. U prvoj fazi analizira se nelinearni model radijalnog ležaja i teorijske osnove su prikazane u ovome radu. Polazeći od činjenice, da u stabilnom radu rukavac vratila leži na sloju fluida za podmazivanje koji odvaja rukavac vratila od košuljice ležaja, pomoću teorijskog nelinearnog modela ležaja može se dobiti bolji uvid u ponašanje rukavca unutar ležaja i na taj način olakšati određivanje realne elastične linije vratilnog voda. *ABS* je započeo realizaciju istraživačkoga projekta s ciljem eksperimentalnog utvrđivanja stvarnih odnosa između rukavca vratila i stražnjega statvenog ležaja, s ciljem primjene rezultata mjerenja u razvoju računalnog modela za analizu odnosa vratilo-ležaj. Početno zanimanje u istraživanju su prijelazne pojave rada vratila u ležaju i promatranje djelovanja hidrodinamičkoga sloja fluida za vrijeme upućivanja, zaustavljanja i prekreta stroja. Senzori za mjerenje zračnosti korišteni su za praćenje odnosa vratilo-ležaj. Senzori mjere udaljenost između vratila i ležaja na osam mjesta unutar stražnjega statvenog ležaja i na četiri mjesta unutar prednjega statvenog ležaja i koristeći te podatke određuje se putanja vratila unutar ležaja.

Rezultati mjerenja pokazuju da je ponašanje vratila unutar statvenih ležajeva vrlo različito ovisno o mjestu unutar ležaja. Zbog toga analitički proračun koji se zasniva na pojednostavljenom linearnom modelu interakcije vratilo-ležaj ne daje rezultate koji odgovaraju stvarnom stanju, jer zanemaruje dinamičke pojave i interakciju između fluida i konstrukcije.

Ključne riječi: *vratilni vod, statveni ležaj, polaganje vratilnog voda, senzori zračnosti, nadzor stanja polaganja, nelinearni model radijalnoga kliznog ležaja*

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Received (Primljeno): 2009-02-09
Accepted (Prihvaćeno): 2009-09-30
Open for discussion (Otvoreno za raspravu): 2011-06-30

1 Introduction

The modern ocean going merchant vessels are highly efficient to the benefit of the whole industry (ship builders, ship operators, and the environment). In general, most of the vessels are:

- larger, with optimized, and therefore more flexible hull structure,
- have high-powered engines of very low revolution, thus requiring bigger shaft diameters,
- have optimized cargo-carrying capacity and minimized engine room space, making shafting short and rigid,
- optimized propulsion train and engines for low fuel consumption, resulting in low revolution, and large diameter of the propellers of very high efficiency.

Shipbuilding practices are modified to increase ship building efficiency, so that more vessels are built in mega-blocks, which minimizes production and dock occupancy time. Unfortunately, with all the benefits achieved by structural optimization, the shafting alignment bears the unfavourable consequences for the following reasons:

- with an optimized more flexible hull, the propulsion system becomes more sensitive to hull deflections,
- the low revolution and the heavy loads from a big shafting and the propeller results in more load exerted and more pronounced misalignment on the bearings, and
- the production of the vessel in mega-blocks makes it difficult to control the alignment accuracy, particularly the misalignment angle inside the aft stern tube bearing.

1.1 Propulsion system

Although most of the excitation on the propulsion shafting comes from the diesel engine, the aft stern tube bearing is more affected by loads from the propeller. In service, the propeller is rotating in a non-uniform wake field resulting in periodic hydrodynamic loads on the propeller blades. Further, this causes varying thrust loads, as well as dynamic forces and moments on the tip of the propeller shaft. Bearings supporting the shafting are directly exposed to those dynamic loads. Among bearings, the aft stern tube bearing is in the least favourable position. Experience shows that the primary reasons for the after stern tube bearing damage and failure are mainly related to:

- shaft alignment to excessive static misalignment (edge contact) between the shaft and the bearing,
- propeller dynamics; high propeller loads, unfavourable wake field, cavitation,
- shaft dynamics; whirling, bending vibration.

In order to prevent bearing damage, avoid failures and extend bearing life, it is imperative that our analytical model represents, as closely as possible, the actual service conditions of the propulsion system.

If our analytical models are not able to replicate the exact operating conditions of the propulsion system, and are unable to capture all relevant parameters which influence the propulsion system operation, our analytical effort may be wasted. The aft stern tube bearing is the most sensitive bearing in the propulsion train system. It is almost exclusively exposed to heavy propeller loads. Moreover, the aft stern tube bearing is the only bearing the access to which is limited and where there are no simple means to verify and monitor the bearing condition and the shaft bearing interaction.

In order to validate the analytical method, actual information obtained from the propulsion system in exploitation is needed. The data derived by measuring are an essential factor in development of theoretical calculation models. The *American Bureau of Shipping* (*ABS*) initiated the research project in order to investigate stern tube journal bearing behaviour. Measurements were customized to investigate the static misalignment condition between the shaft and the bearing, to correlate the static to dynamic transition with the misalignment angle, and to investigate the shaft bearing interaction under the service conditions of the vessel. The focus of the investigation was to capture the transient operation of the shaft inside the bearing and to observe the shaft's hydrodynamic lift during starting, stopping and reversing of the engine.

With a sufficient number of field samples, the measured information can be utilized to calibrate the developed software, compare theoretical and measured values and derive a conclusion on necessary improvements to achieve a more realistic theoretical model in the next stage of investigation.

2 Modelling of the radial journal bearings behaviour

Due to the complexity of the interaction between shaft dynamics and lubricant hydrodynamics, a theoretical investigation will be performed in several stages.

In the first stage, the theoretical model is simplified so that the angular velocity of the journal and the orientation of the magnitude of reaction force in the bearing are considered to be time-independent and the curvature of the journal within the bearing is neglected. The second stage will define a more complex theoretical/numerical model where the shaft bending curvature within the bearing will be accounted for. The third stage will attempt to apply transient propeller loads and investigate shaft transition from static to dynamic operation as a function of the misalignment angle between the shaft and the bearing. All of the above theoretical models will be calibrated based on the information collected by field measurements.

This article addresses the first stage only, and the theoretical background for the same is presented in detail in the text below.

2.1 Theoretical background

The theoretical model of the propulsion shafting is represented by a linear system of girders. The system is modelled using the finite element method, with beams describing the moving parts, bearings to define shaft journal supports, and fluid (oil) as the interface between the beam elements and the bearings. The bearing can be modelled in a few different ways:

- absolutely stiff,
- linear elastic,
- linear elastic radial journal bearings,
- nonlinear model of radial journal bearings.

The nonlinear model enables a more complex and more realistic description of the radial journal bearing behaviour, as it takes into account the properties of the lubricant which appears between the journal and the bearing in operation. This model is based upon the theory of the hydrodynamic lubrication in journal bearing. The analysis is performed on a model of the radial journal bearing schematically shown in Figure 1, with all basic geometry and load characteristics. The theory evolves from Reynold's differential equation (1) and its solutions. Reynold's differential equation describing stationary hydrodynamic lubrication in journal bearing is as follows:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{1}{r^2}\frac{\partial}{\partial \varphi}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial \varphi}\right)=6\omega\frac{\partial h}{\varphi} \tag{1}$$

where:
p - hydrodynamic pressure in the lubricant film,
h - thickness of the lubricant film,
x, r, ϕ - geometric characteristics of radial journal bearing model,
η - dynamic viscosity of the lubricating fluid,
ω - speed of revolution.

2.2 Basic characteristics and relationship of non-linear model numerical structure

The basic characteristics related together within this model are the following:

a) The Sommerfeld number represents the bearing load characteristic and is defined by the following expression:

$$S_O=\frac{R\psi^2}{bd_R\eta\omega} \tag{2}$$

where:
R = bearing reaction force R,
ψ = relative bearing clearance, defined as $\psi=\frac{d_L-d_R}{d_R}$,
η = dynamic viscosity of the lubricating fluid,
ω = journal revolution,

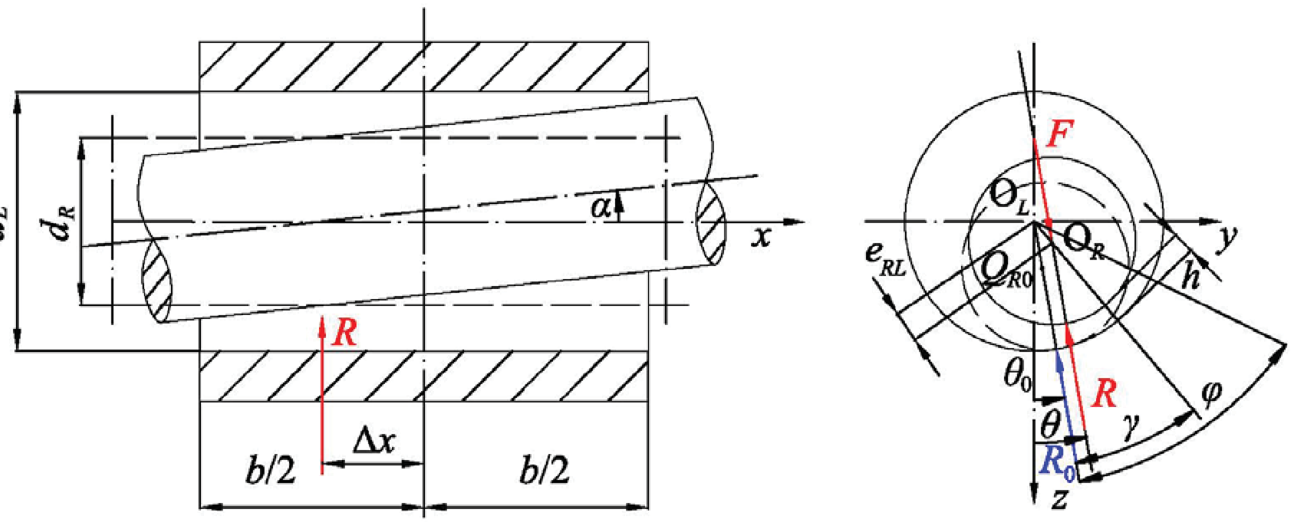


Figure 1 **Schematic view of the model of nonlinear radial journal bearing**
Slika 1 **Shematski prikaz modela nelinearnog radijalnog ležaja**

Equation (1) can be solved numerically by the finite-element or by the finite-differences method. The solutions describe the static elastic properties of the radial journal bearing, i.e., the behaviour of the shafting journal lying on a film of a lubricating fluid in a journal bearing, and are presented in [3,4].The nonlinear model of the radial journal bearing besides the hydrodynamic properties of the fluid film also takes into account the geometric properties of the bearing and alignment properties of the journal into the bearing liner. The model is based upon the following presumptions:

- the angular velocity of the journal is time independent,
- the absolute value and the direction of the reaction force in the bearing are time independent,
- the angle of inclination of the journal axis with respect to the bearing axis is taken into account,
- owing to the nonsymmetrical pressure distribution along the lubricating fluid film in the bearing, the bearing reaction is axially displaced from the midsection of the bearing,
- the curvature of the journal within the bearing is neglected,
- the component of the bearing reaction in the axial direction may be neglected.

b = bearing length,
d_R = journal diameter.

b) The angle of the bearing loading γ can be approximated by the following expression [3]:

$$\gamma=\arctan\left(\frac{\pi}{2}\frac{\sqrt{1-\varepsilon^2}}{\varepsilon}\right)\sum_{j=0}^{4}\left(B_{\gamma 0,j}+\lambda\cdot B_{\gamma 1,j}\right)\varepsilon^j \tag{3}$$

where:
$B_{\gamma 0,j}$ and $B_{\gamma 1,j}$ = approximation coefficients presented in [3].
The equation (3) is valid within the range $\lambda \geq 0.125$.

c) The relative axial shift of the bearing reaction, X can be approximated as:

$$X=\frac{\Delta x}{b} \tag{4}$$

where:
Δx = axial shift of the bearing reaction,

$$X = \sum_{i=0}^{4}\sum_{j=0}^{4-i}\left(B_X\right)_{i,j} A^i W^j \tag{5}$$

where:
W is estimated from:

$$W = \sum_{i=0}^{2}\sum_{j=0}^{2-i}\left(B_W\right)_{i,j} \lambda^i \cdot \log^j\left(2\pi \cdot So\right) \tag{6}$$

Coefficients B in equations (5) and (6) are based upon estimates presented in [3].

d) The relative inclination angle, A:

$$A = \frac{2\alpha \cdot b}{d_L - d_R} \tag{7}$$

where:
α = inclination angle of journal,
d_L = bearing diameter.

e) The relative bearing length, λ is defined as:

$$\lambda = \frac{b}{d_R} \tag{8}$$

f) The relative journal eccentricity in the bearing, ε is defined as:

$$\varepsilon = \frac{2e_{RL}}{d_L - d_R} \tag{9}$$

where:
e_{RL} – eccentricity of the journal in the bearing central plane.

According to (8) and (2), using expression (9), the following expression can be used to relate ε with So and λ, as presented in [8]:

$$\varepsilon = \frac{So}{\left(1 + \frac{1-\lambda}{15}\right)So + \frac{9\lambda - 1}{10}} \tag{10}$$

The equation (10) is valid within the following range:

$$0.15 \le \lambda \le 2 \quad \text{and} \quad 0.01 \le So \le 10$$

g) The relative thickness of the lubricant film H, can be approximated by the following expression:

$$H = \sum_{i=0}^{4}\sum_{j=0}^{4-i}\left(B_H\right)_{i,j} A^i \left(1-\varepsilon\right)^j \tag{11}$$

Coefficients B in equation (11) are based upon estimates presented in [3].

2.3 Implementation of non-linear numerical model of the radial journal bearing

In operation, the shafting journal floats on a lubricant which separates the journal from the bearing liner. The subject model will describe the behaviour of the shaft in the bearing and determine an elastic line of the shafting. The complication is that the model requires a nonlinear calculation procedure. The results of such calculations, and developed theory may be used as a starting point for the analysis of the shaft/bearing interaction, gradually improved as compared with experimental results from ships in service.

3 Measurements

Measurements were performed on the vessel specified in chapter 3.1 in order to investigate the static misalignment condition between the shaft and the bearing, and to correlate this information with shaft speed and hydrodynamic lift of the shaft. The focus of the investigation was to capture transient operation of the shaft inside the bearing, and to observe the shaft's hydrodynamic lift during starting, stopping and reversing of the engine.

The subject report compiles only one segment of the recorded information, namely:

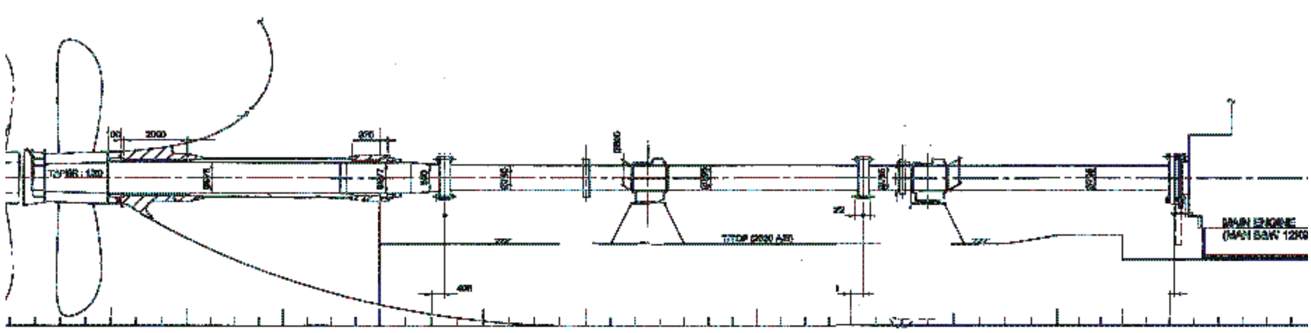
- Static measurements conducted in the dry dock,
- Static measurements with waterborne vessel - before the sea trials of the ship.
- Diesel engine start and stop test.
- Dynamic operation under various vessel operating conditions.

Due to a very large amount of recorded dynamic data (collected during the sea trial), only selected information is presented in this article.

3.1 Propulsion system data

Type of ship: 9600 TEU container carrier
Main Engine:
- Type: 2-stroke, slow-speed diesel engine, *MAN B&W* 12K98MC-C
- MCR : 68500 kW at 104 min^{-1}

Figure 2 **Shafting arrangement**
Slika 2 **Dispozicija vratilnog voda**



Propeller:
- Fixed Pitch Propeller, 6 Blades, 8800 mm in diameter, Mass 84000 kg

Shafting data (Table 1)

Table 1 **Propulsion shafting system data**
Tablica 1 **Podaci o propulzijskom vratilnom vodu**

Items	Dia x Length (mm)
Propeller shaft	Ø975 x 13079
No.1 Intermediate shaft	Ø795 x 13348
No.2 Intermediate shaft	Ø795 x 9870

The actual propulsion system drawing and its equivalent model for shaft alignment analysis are shown in Figure 2 and Figure 3 respectively.

Proximity probes were used to monitor shaft bearing interaction (Table 2).

Table 2 **Proximity probes data**
Tablica 2 **Podaci o senzorima zračnosti**

Bently Nevada: 3300 11mm XL Underwater Probe, 5/8-18 **3300 XL 11MM PROXIMITOR**	Measurement accuracy is expected to be within +/- 0.01 [mm] (probe accuracy is 0.001 mm)

For the considered vessel, the shaft position inside the stern tube bearing was measured in four equidistant planes, with four pairs of probes (Figure 4). Probes are 90 degrees apart (45 degrees port and starboard of the top of the bearing). Our point of interest is to find the shaft's position relative to the centre of the bearing. The proximity probe measures the distance from the tip of the probe to the closest distance on the shaft. Knowing the bearing/shaft clearance, we can define the shaft position relative to the bearing and observe the interaction between the shaft and the bearing at a particular plane (as in Figure 5).

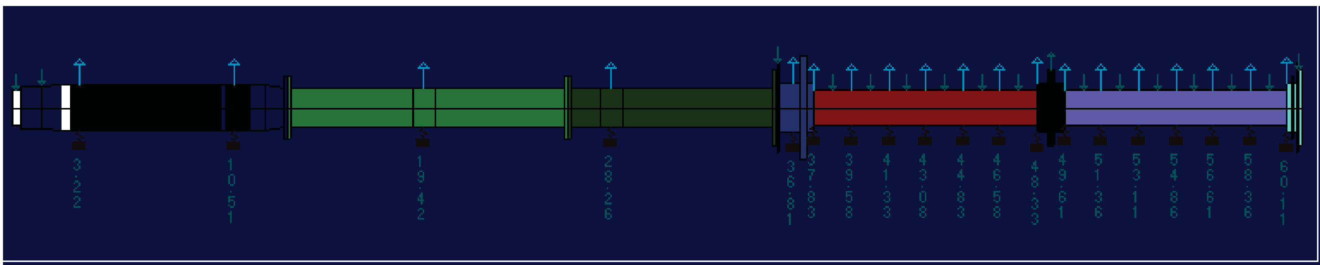
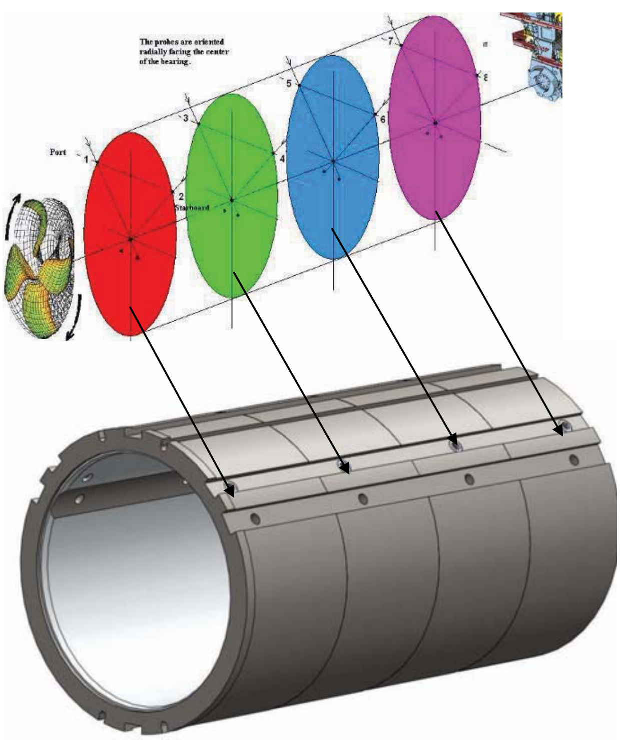


Figure 3 **Propulsion shafting – shaft alignment model**
Slika 3 **Propulzijski vratilni vod – dispozicijski model**

3.2 Measurement equipment and installation

Figure 4 **Location of proximity sensors in four planes of the aft bush**
Slika 4 **Prikaz razmještaja senzora zračnosti u četiri ravnine stražnje ležajne blazinice**



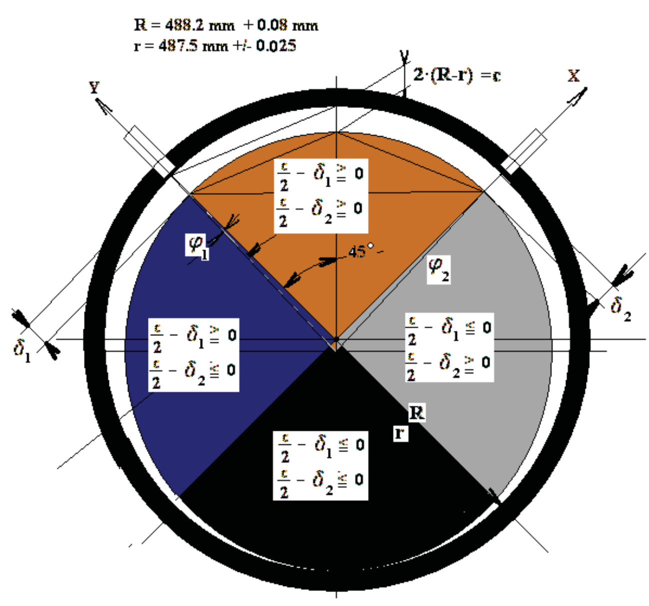


Figure 5 **Probe location in the bearing bush**
Slika 5 **Razmještaj senzora u blazinici ležaja**

4 Measurement procedure and results

4.1 Static measurement

Static measurements were conducted in the dry dock, and for the waterborne vessel, before the sea trials of the ship. The purpose of the static measurement was:

– to obtain a misalignment angle between the shaft and the bearing,
– observe how the construction process of the ship affects shaft bearing misalignment, and
– to capture the influence of different draught conditions on the shaft bearing interaction.

For the considered vessel, the shipyard conducted slope boring of the aft stern tube bearing in order to obtain acceptable shaft bearing contact. The slope boring angle designed by the yard was 0.46 mm over 1940 mm of the effective length of the bearing. The slope boring is conducted in order to minimize the misalignment angle. In the case of the considered vessel, the measurement has shown that the slope boring was very successful, as the misalignment angle was verified to be almost zero. For the purposes of comparison, we conducted the analysis using the *ABS* Shaft Alignment Software. The accuracy of the conducted calculations is presumed to be very close to the actual measurements when only the propeller shaft is put in place (the rest of the shafting is disconnected). Two such conditions were investigated and shown below:

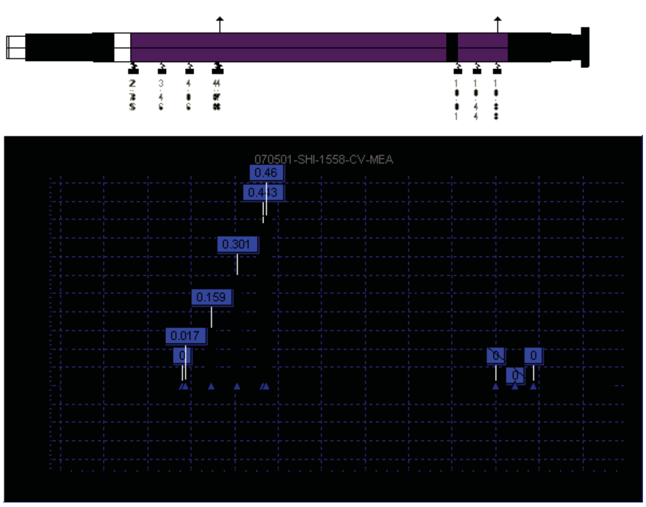
– propeller shaft only, no propeller attached (Figure 6, Figure 7)
– propeller shaft only with propeller attached (Figure 8, Figure 9).

4.1.1 Propeller shaft only (no propeller)

Analysis results

The analytical estimate of the bending curvature of the shaft inside the aft stern tube bearing is from the four points along the bearing (the four points coincide with the location of proximity probes). The maximum diametric clearance between the shaft and the bearing is 1.48 mm. Calculation is conducted with a bearing slope boring angle of 0.46 mrad. The forward stern tube bearing is not considered in the measurement; however, it is included in the analytical evaluation. In this case, the point contact is obtained on the aft edges of the aft stern tube bearing and forward stern tube bearing.

Figure 6 **Analytical investigation of the shaft/bearing interaction - propeller shaft only – no propeller**
Slika 6 **Analitičko određivanje međusobnog položaja između rukavca vratila i ležaja - samo propelersko vratilo – bez propelera**



Measured misalignment (Table 3)

Table 3 **Measured distances - propeller shaft only**
Tablica 3 **Izmjerene zračnosti – samo propelersko vratilo**

Measured: Shaft's distance from the bearing (measured):		
Location	Transverse offset of the shaft center from the bearing C/L[mm]	Vertical offset of the shaft center from the bearing C/L [mm]
Aft-Aft	-0.043	-0.305
Aft-Middle	-0.031	-0.423
Middle-Fwd	-0.022	-0.585
Fwd-Fwd	-0.018	-0.713

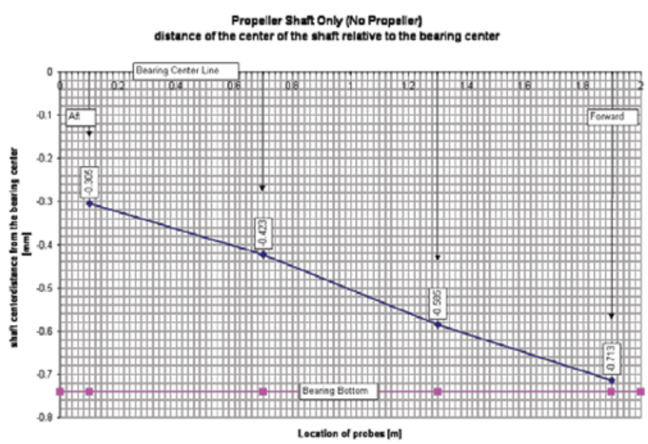


Figure 7 **Measured distances- propeller shaft only**
Slika 7 **Izmjerene zračnosti – samo propelersko vratilo**

It is observed that the shaft is sitting on the forward edge of the aft bearing and is raised up on the aft end, as expected when the propeller is not attached. It can also be noticed that there is a transverse offset of the shaft (when resting inside the bearing) relative to the centre of the bearing. Distances are compliant with the analytically predicted position of the shaft.

4.1.2 Propeller shaft with propeller

The propeller was installed; shafts were not assembled. The results of the analysis and measurements are illustrated in Figure 8 and Figure 9. As expected, the weight of the propeller resulted in shaft bending which deflected the shaft so that the contact between the shaft and the bearing was obtained on the aft end as well.

Analysis results

This represents the analytical estimate of the bending curvature of the shaft inside the aft stern tube bearing. The maximum

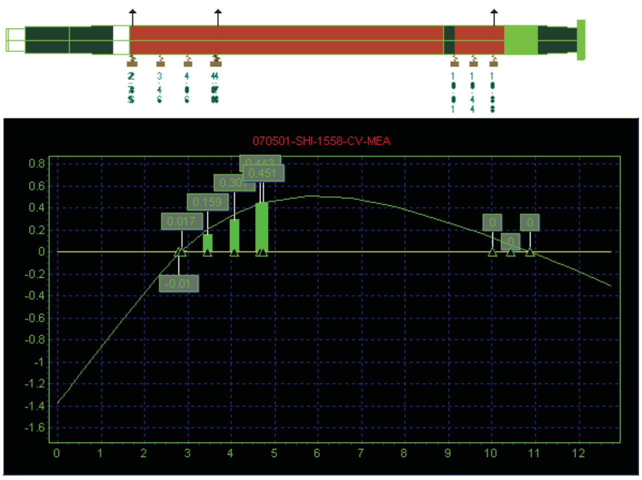


Figure 8 **Analytical investigation of the shaft/bearing interaction - propeller shaft with propeller attached**
Slika 8 **Analitičko određivanje međusobnog položaja između rukavca vratila i ležaja - propelersko vratilo spojeno s propelerom**

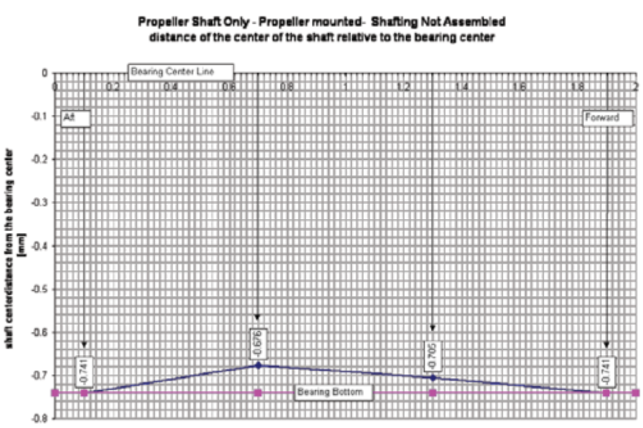
Measured misalignment (Table 4)

Table 4 **Measured distances - propeller shaft with propeller attached**
Tablica 4 **Izmjerene zračnosti – propeler spojen na propelersko vratilo**

Measured :Shaft's distance from the bearing (measured):		
Location	Transverse offset of the shaft center from the bearing C/L[mm]	Vertical offset of the shaft center from the bearing C/L [mm]
Aft-Aft	-0.011	-0.741
Aft-Middle	-0.000	-0.676
Middle-Fwd	-0.011	-0.705
Fwd-Fwd	-0.015	-0.741

diametric clearance between the shaft and the bearing is 1.48 mm. Calculation is conducted with a bearing slope boring angle of 0.46 mrad.

Figure 9 **Measured distances - propeller shaft with propeller attached**
Slika 9 **Izmjerene zračnosti – propeler spojen na propelersko vratilo**



The forward stern tube bearing was not considered in the measurement; however, it is included in the analytical evaluation. In this case, the point contact is obtained on the aft stern tube bearing: two points (forward and aft edge of the bearing), and on the forward stern tube bearing: aft edges of the bush. Distances between the shaft and the bearing are compliant with the analytically predicted position of the shaft.

4.1.3 Shafting assembled

After shafting assembly, analytical results are sensitive to the actual alignment condition. Since the analysis cannot match the required precision of the proximity probe measurements, analytical analysis is not performed for the rest of the cases presented in this paper.

Measured misalignment (Table 5)

Table 5 **Measured distances - shafting assembled**
Tablica 5 **Izmjerene zračnosti – vratilni vod spojen**

Measured: Shaft's distance from the bearing (measured):		
Location	Transverse offset of the shaft center from the bearing C/L [mm]	Vertical offset of the shaft center from the bearing C/L [mm]
Aft-Aft	-0.002	-0.743
Aft-Middle	-0.002	-0.667
Middle-Fwd	-0.001	-0.687
Fwd-Fwd	-0.001	-0.735

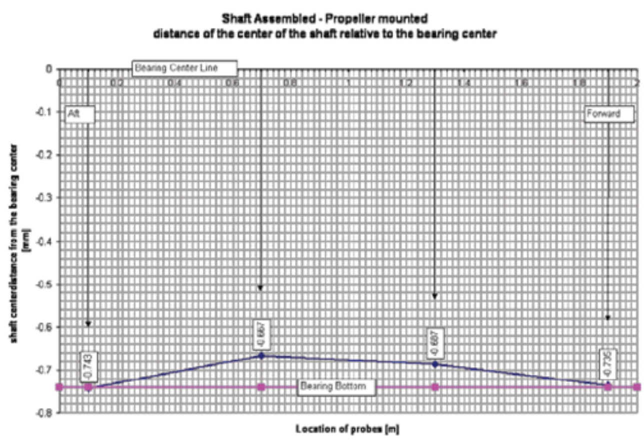


Figure 10 **Measured distances - shafting assembled**
Slika 10 **Izmjerene zračnosti – vratilni vod spojen**

4.2 Dynamic measurement

The second goal of our investigation was to correlate the shaft-bearing misalignment to the revolution corresponding to the hydrodynamic lift. We also used this opportunity to examine the interaction between the shaft and the bearing under other operating conditions, as mentioned below. Dynamic measurements were conducted during the preparation for sea trials and during the actual sea trials of the vessel. For shaft speed, an analog proximity switch was used. The switch was placed above the flywheel, and it counted flywheel teeth. The same information provides the speed of the shaft when combined with the time readings.

Before the sea trial, measurements were used to verify proper operation of the installed equipment and to set up the sea trial measurement schedule. We will show only selected readings which in our opinion capture the most interesting parts of the shaft behaviour:

– Engine start (Figure 11, Figure 12)
– Engine shut down (Figure 13, Figure 14, Figure 15)
– Manoeuvring: Astern – Stop -Astern – Stop (Figure 16)
– Running at 61 rpm - sea trials (Figure 17)
– Full Ahead - 112 rpm – Rudder 0 deg - sea trials (Figure 18)
- Rudder test – 111 rpm – Rudder 36 deg Port to 36 deg Stbd - sea trials (Figure 19)

4.2.1 Starting the engine

Draught Aft: 9.8 m, Fwd: 5.6 m; 34 °C
Ahead operation.
Ahead shafting revolution is clockwise.
Record is taken over a period of 207 seconds.

Figure 11 and 12 show the orbit of the shaft as it moves from the idle position (static) to the steady operation at 38 rpm. Figure 12 shows the shaft orbit inside the bearing during starting of the engine. It is observed that the shaft hydrodynamic lift occurs within one revolution (at 270° of one rpm). The shaft then stabilizes at 38 rpm. One can notice that the shaft is not sitting exactly at the bottom of the bearing. This is due to imperfections

Figure 11 **Starting the engine – from stop to 38 rpm**
Slika 11 **Pokretanje motora – od stop do 38 o/min**

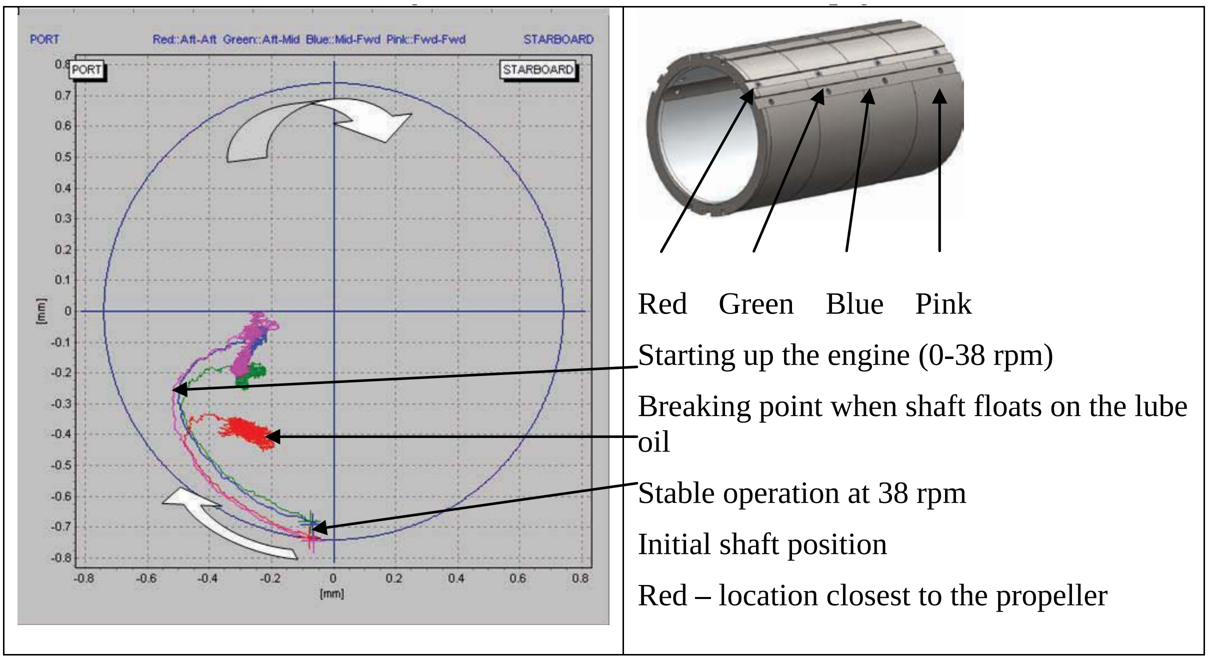
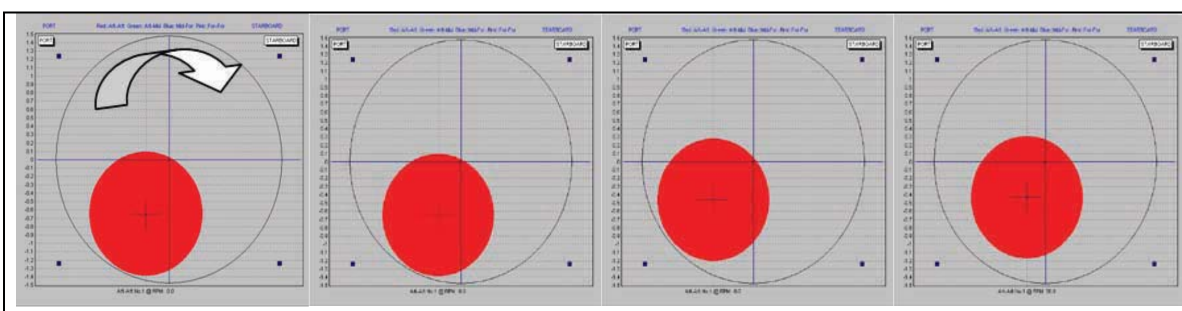


Figure 12 **Engine starting**
Slika 12 **Pokretanje motora**



Engine stopped. Shaft rests at the BDC.	Start up: transition	Start up: transition	Steady operation at 38 rpm

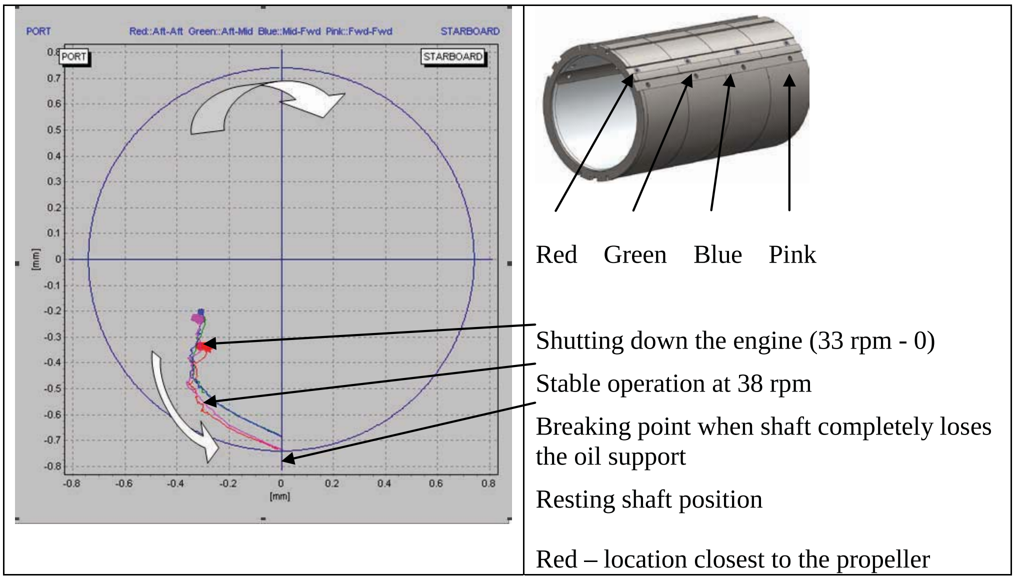
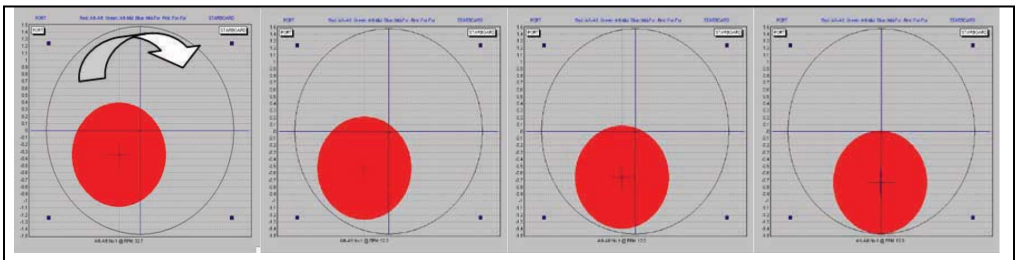


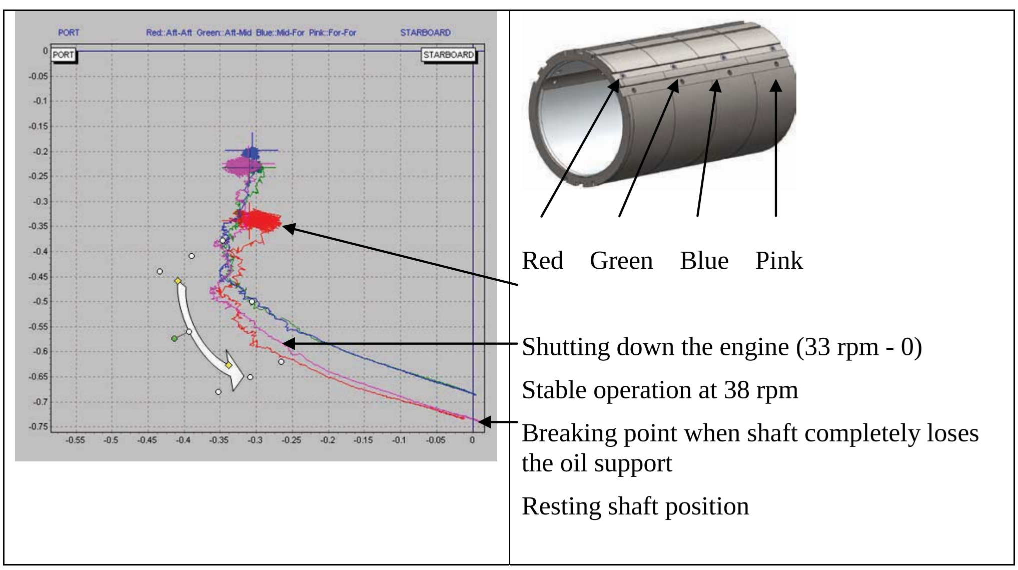
Figure 13 **Engine shutting down – from 33 rpm to stop**
Slika 13 **Zaustavljanje motora – od 33 o/min prema stop**



Steady operation at 33 rpm	Shut down: transition	Shut down: transition	Engine stopped. Shaft rests at BDC.

Figure 14 **Shutting down the engine**
Slika 14 **Zaustavljanje motora**

Figure 15 **Enlarged view of the engine shut down curves**
Slika 15 **Uvećani prikaz putanje središta rukavca pri zaustavljanju motora**



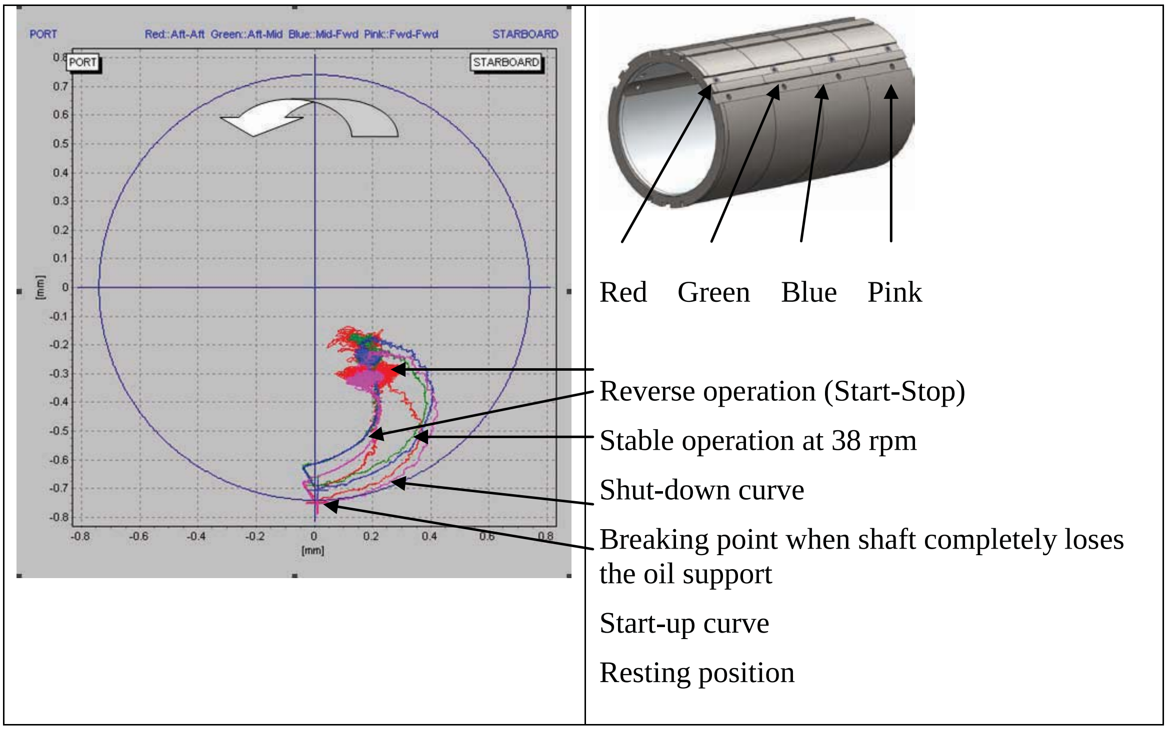


Figure 16 **Manoeuvring (astern: start-stop-start- stop)**
Slika 16 **Manevar (krmom: start-stop-start-stop)**

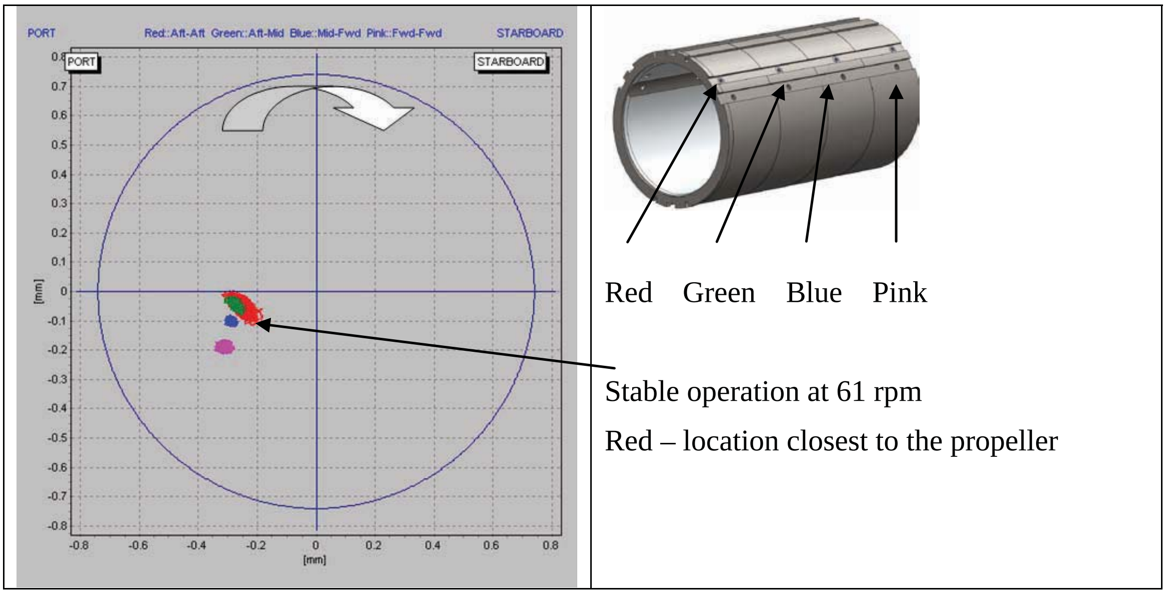


Figure 17 **Steady running ahead – 61 rpm**
Slika 17 **Jednolika vožnja pramcem – 61 min^{-1}**

in the bearing surface and the plastic deformation which results from pressure exerted from the shaft.

4.2.2 Engine shutting down

Ahead operation.
Draught Aft: 9.8 m, Fwd: 5.6 m; 34 °C
Record is taken over a period of 89 seconds.

The charts in Figure 13 and Figure 14 capture the snapshots of the shaft behaviour at the aftmost measured plane during the process of engine stopping. Figure 15 shows an enlarged view of the shutdown curves. The shaft bending inside the stern bush is obvious, as we see the shaft is sitting at both edges of the bearing and is detached from the bearing in between (see static measurements above).

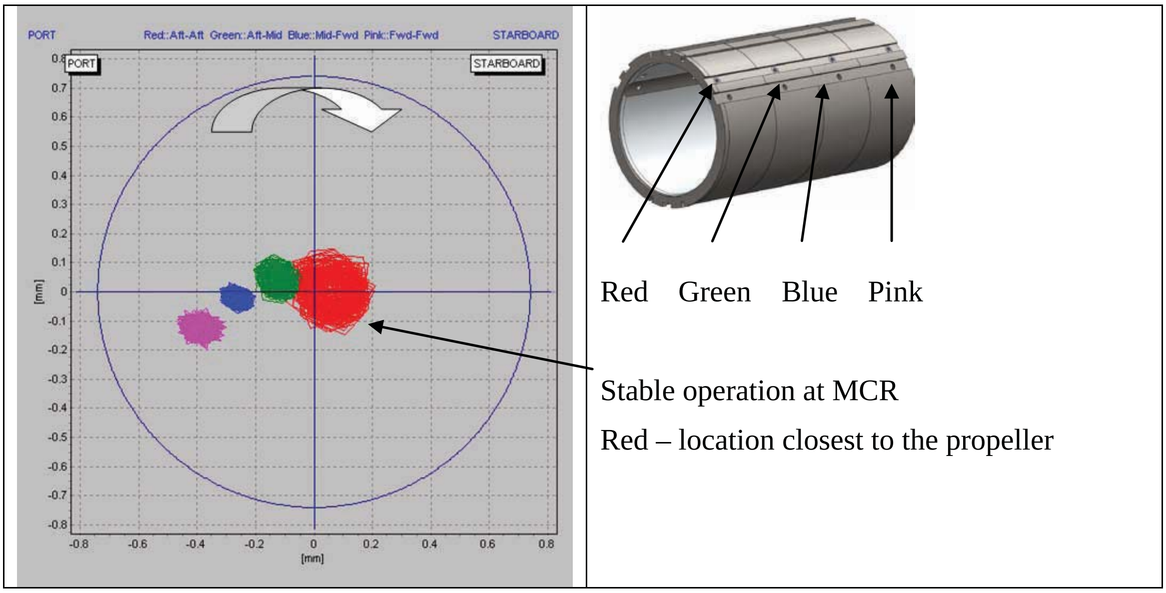


Figure 18 **Steady running ahead at rated speed – 112 min^{-1}**
Slika 18 **Jednolika vožnja pramcem s nominalnim brojem okretaja – 112 min^{-1}**

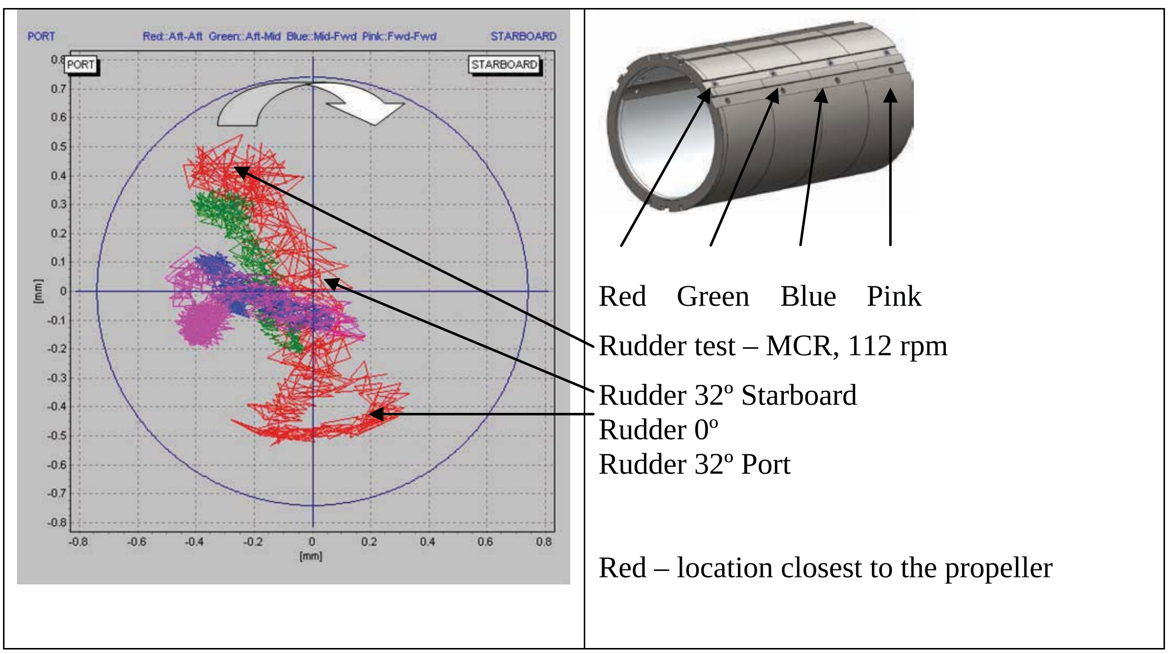


Figure 19 **Shaft behaviour inside the bearing during the rudder test**
Slika 19 **Ponašanje vratila unutar ležaja za vrijeme testa upravljivosti**

4.2.3 Manoeuvring (Astern operation)

Run Astern – Stop – Run Astern – Stop

Draught Aft: 9.8 m, Fwd: 5.6 m; 34 °C

Record is taken over a period of 35 seconds.

Ship manoeuvring is shown in Figure 16. Recording started at the 33 min^{-1} astern run for 35 seconds. Engine stopped for 8 seconds, then followed astern running for 18 seconds, and then engine stopped.

4.2.4 Steady operation ahead at 61 rpm

Draught Aft: 9.8 m, Fwd: 5.6 m; 34 °C

Running steady ahead at speed of 61 rpm. Rudder angle zero.

Shaft is closer to the forward edge of the bearing and raised up as we look towards the aft edge, Figure 17.

4.2.5 Steady operation ahead at rated speed

Draught Aft: 9.8 m, Fwd: 5.6 m; 34 °C

Running steady ahead at speed of 112 min^{-1}. Rudder angle zero.

Record is taken over 60 seconds

For the rated speed operation, with rudder angle zero, the shaft behaviour inside the shaft is as shown in Figure 18:

– Red - location aft (closest to the propeller) 100 mm from the bush edge: the shaft is hovering around the centre of the bearing.
– Green - at 700 mm from the aft bearing edge: the shaft is more stable but closer to the bearing.

As we move towards the forward edge of the bearing, the shaft exerts more pressure on the bearing. On the foremost location (Pink), 100 mm from the edge, the distance of the shaft from the bearing is still very satisfactory although the shaft moves half the clearance towards the edge (approximately 0.3 mm clearance).

4.2.6 Rudder test

Draught Aft: 9.8 m, Fwd: 5.6 m; 34 °C

Running at rated speed of 112 min^{-1}.

Comprehensive measurements were conducted during the rudder test as well (Figure 19). The rudder was tested for maximum inclination under full vessel speed of 112 rpm: the rudder moved from 36° port to 36° starboard. Instability of the wake field during turns results in the propeller loads which are:

– when turning portside: downward force
– when turning starboard; upward force.

The dynamics of the shaft is quite different in different positions inside the stern tube bearing. It shows that the shaft dynamics is much more stable when closer to the forward edge of the bearing. On the contrary, forces from the propeller are pushing the shaft extremely close to the bearing as we get close to the aft edge of the stern bush.

5 Conclusion

The *ABS* initiated the stern tube bearing misalignment measurement research project in order to investigate static bearing misalignment, to correlate static to dynamic transition with the misalignment angle, and to investigate shaft/bearing interaction under the service conditions of the vessel. The focus of the investigation is to capture the transient operation of the shaft inside the bearing and to observe the shaft's hydrodynamic lift during starting, stopping and reversing of the engine. In the case presented in this paper, the measurement revealed that the static misalignment angle between the shaft and the bearing is almost nil. Accordingly, in the static condition, a very low misalignment angle results in shaft contact at both edges of the after stern tube bearing (front and aft). Data recorded for dynamic operation did not indicate any anomalies of the shaft operation inside the stern tube under measured conditions.

Very important information that we obtained from the measurement is that the simplified analytical models (which are presently used in some shaft alignment analyses) are not capable of accurately representing the propulsion system operation which results in most critical shaft/bearing interaction. Namely, the critical interaction between the shaft and the bearing can be generally attributed to the following:

- Starting the propulsion system (transition from static to dynamic state),
- Operation is highly nonuniform, unsteady wake field during the rudder operation, inclement weather, shallow water operation, etc.

Conducted measurements enabled us to observe and record actual behaviour of the shaft inside the stern tube, and correlate this behaviour with other operational parameters (speed, rudder angle, draught/trim, shaft vibration, and in some cases structural vibration and cavitation). Therefore, if in the analytical approach the applied software does not account for operational condition which can result in damage and failure of the stern tube bearing, then the same software does not have significant practical value.

Accordingly, the *ABS* and the *University of Zagreb, Faculty of Mechanical Engineering* intend to collaborate on the theoretical model development. The theoretical model is proposed to be developed in several stages. The initial stage, as laid out in this article, is to consider nonlinear fluid structural interaction. Owing to the fact that the shaft journal in operation lies on a lubricating fluid film, which separates the journal in the bearing from the bearing liner, the theoretical nonlinear model of the journal bearing is expected to provide much closer compliance with field data than the analytical approach which uses a simplified theory of the shaft/bearing interaction.

The *ABS* will need a few more years to complete the investigation into shaft/bearing interaction. However, the results of the current investigations are already in use in the *ABS* in order to design software which will enable an accurate evaluation of the shaft/bearing interaction and to improve existing software for propeller loads prediction.

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