GUIDANCE NOTES ON

PROPULSION SHAFTING ALIGNMENT

SEPTEMBER 2019

American Bureau of Shipping
Incorporated by Act of Legislature of
the State of New York 1862

1701 City Plaza Drive
Spring, TX 77389 USA

Foreword

ABS identified the need to provide a more detailed explanation of alignment design and practices, which resulted in the development of the *Guidance Notes on Propulsion Shafting Alignment*. Their primary purpose is to provide clarification for ABS Surveyors and plan review engineers to verify consistency of the survey and plan approval processes. Regarding the shaft alignment design and implementation efforts by the shipbuilding industry, these Guidance Notes are valuable to further advance its approach towards shaft alignment analyses and procedures.

Additionally, ABS has developed state-of-the-art analytical tools primarily for the purpose of engineering analysis and design. The ABS shaft alignment program, combined with alignment optimization software, is capable of analyzing complex propulsion installations and, when used as a design tool, may provide an optimized solution to the alignment problem.

This 2019 edition is arranged to include additional topics on shaft alignment condition monitoring and a section dedicated to alignment problems and their solutions. Shaft alignment survey requirements are summarized in the introduction and are addressed further under a separate section on shaft alignment procedures and practices. Discussion on alignment calculation and measurement is expanded to include the stern tube bearing clearance measurement. Additional clarification is provided on the application of the hull deflections, alignment optimization, propulsion systems with no forward stern tube bearing, and reverse engineering calculation.

These Guidance Notes become effective on the first day of the month of publication.

Users are advised to check periodically on the ABS website www.eagle.org to verify that this version of these Guidance Notes is the most current.

We welcome your feedback. Comments or suggestions can be sent electronically by email to rsd@eagle.org.

Terms of Use

GUIDANCE NOTES ON

PROPULSION SHAFTING ALIGNMENT

CONTENTS

This Page Intentionally Left Blank

SECTION 1 Introduction to Propulsion Shafting Alignment

1 General

Propulsion shafting alignment is a process that consists of:

- The design and analysis
- The alignment procedure
- Measurements and verification

The terminology and requirements for shafting alignment vary depending on the type of the system analyzed (propulsion, gen-set, etc.), the powertrain size, and the alignment process itself.

The following terminology is used in these Guidance Notes.

Propulsion Shafting: A system of revolving rods that transmit torque and motion from the prime mover to the propeller. The shafting is supported by bearings, whose number and position is determined based on *allowable bearing loads* and *lateral vibration (whirling) requirements*. Static shafting alignment analysis criteria defines the acceptable load distribution and contact condition between shafts and bearings.

These Guidance Notes distinguish between *"alignment process"* and *"alignment procedure"*. The alignment process encompasses all shafting alignment activity starting with analyses and reviews, to installation procedures, condition verification and trials, while the alignment procedure in itself refers only to shafting alignment production work activities on site.

Propulsion Shafting Alignment: A static condition observed at the bearings supporting the propulsion shafts. To suitably define propulsion shafting alignment, a minimum set of parameters are defined in the design stage and subsequently confirmed acceptable on board:

- Bearing vertical offsets
- Bearing reactions
- Bearing to shaft contact condition (i.e. misalignment angles)
- Crankshaft web deflections
- Gear mesh misalignment.

Alignment is considered satisfactory when these parameters can be controlled in the static and dynamic condition and maintained within the required limits under all operating conditions of the vessel for all service drafts and service temperatures.

A change in the vessel's draft due to altered loading conditions affects hull girder deflections. This in turn influences bearing offsets, causing a redistribution of the load among the bearings. Achieving a satisfactory alignment design for the loading conditions included in the vessel's loading manual requires several repeated analyses.

A temperature rise or drop also affects bearing offsets. However, unlike the hull deflections which affect all bearings in the system simultaneously, effect of temperature changes tend to be local to a particular bearing, or set of bearings, such as in the main engine and gear-box.

Although shafting alignment analysis and procedure are conducted for a static condition, the industry often uses the term *dynamic alignment* to describe the running condition of the propulsion shafting. This terminology raises some controversy when the dynamic or ”running” condition is defined by utilizing dynamic propeller loads in an essentially static analysis. This “quasi-static” approach potentially draws misleading conclusions about aft stern tube bearing loading, slope boring and overall performance.

The propeller loads accounted for in a quasi-static analysis must be defined for a service condition that is deemed critical for propulsion system performance, namely maneuvering at high speeds with large rudder angles. Merely accounting for dynamic propeller loads for steady straight-ahead runs does not contribute to a safer or better alignment design and may result in misleading conclusions and a substandard aft stern tube bearing slope boring design.

The quasi-static approach can only be considered appropriate when all relevant loads are accounted for, including also the most critical service load condition for those components evaluated in the design. Moreover, the quasi-static analysis of the aft stern tube bearing contact entirely neglects the hydrodynamic loads components acting on the bearing.

Descriptions, processes, procedures and proposed solutions for shafting alignment issues outlined in these Guidance Notes are based upon industry-wide accepted best practices and may not necessarily fully reflect ABS Rule requirements.

ABS propulsion shafting alignment requirements and limits are defined in the relevant part of the applicable set of ABS Rules and must be referred to for Classification purposes.

2 Objectives

These Guidance Notes provide a comprehensive approach to the propulsion shafting alignment process and examine in detail the design and the design review process, and also address production practices, inspection and verification of the shafting alignment condition.

These Guidance Notes clarify the position of ABS on emerging propulsion shafting alignment procedures in excess of or not presently addressed by current ABS Rule requirements, with the intent to achieve a more unified approach on shaft alignment design, analysis and production practices throughout the industry.

A reduction in damage to propulsion shafting, bearings, seals and coupling bolts, main engine bearings and propulsion gear meshes as well as a reduced risk of environmental pollution are some of the immediate benefits of an appropriate and correct approach to shafting alignment.

These Guidance Notes primarily focus on alignment issues in alignment-sensitive propulsion systems, such as those on VLCCs, ULCCs, large bulk carriers and large container vessels. The following designs are specifically addressed:

- Direct drive propeller installations
- Low speed diesel installations
- Systems with relatively short and rigid shafting
- Installations with no forward stern tube bearing
- Vessels with a relatively flexible hull structure
- Twin screw propulsion

Insight is also offered into alignment issues with turbine or electric gear-driven installations that fall into any of the above categories.

Although the focus of these Guidance Notes is on vessels with alignment-sensitive installations, the same criteria and approach equally apply to all propulsion shafting installations.

These Guidance Notes elaborate on the following:

- Static shaft alignment design, analysis and review
- General alignment considerations

- Robust alignment design
- Aft stern tube bearing problems
- Diesel engine bearing problems
- Designs with no forward stern tube bearing
- Intermediate shaft bearing offset adjustment
- Crankshaft equivalent model
- Bearing clearances
- Gear meshes
- Acceptance criteria
- Quasi-static shaft alignment analysis
- Propeller dynamic load application
- Dynamic shaft alignment design, analysis and review
- Whirling and lateral vibrations
- Fluid-structural dynamic interaction (FSI)
- Shaft alignment procedure
- Shipbuilder practices
- Construction tolerances
- Construction priorities
- Stern tube slope boring or stern tube bearing inclination
- Alignment measurement and monitoring
- Sighting through
- Sag and Gap
- Bearing load measurements
- Bearing condition monitoring
- Effects of hull girder deflections
- Hull girder deflection measurement
- Hull girder deflection prediction
- Alignment optimization
- Alignment problem prevention

3 The Shaft Alignment Challenge

3.1 Modern Vessel Design

Most modern vessels are state of the art designs, with highly efficient propellers and high-powered main drives with low fuel consumption. Their hull structure is optimized to reduce deadweight and maximize cargo capacity. Unfortunately, most modern ship design improvements can adversely affect propulsion shafting alignment if this is not correctly addressed.

Slow speed installations make for an efficient propeller and low engine fuel consumption. This in turn though requires a large and heavy propeller to produce the required thrust. The result is a heavy cantilever load on the aft stern tube bearing, making it difficult for the bearing to maintain an adequate lubrication film at low revolutions and during maneuvering at higher speeds.

High-powered engines require large diameter shafting to transmit torque. This makes shafting less compliant to hull and structural deflections and the mismatch between the flexibility of the ship's structure and its propulsion shafting becomes a concern for the designers and engineers. A rigid shafting system is more prone to bearing unloading, which can result in bearing damage, overload of adjacent bearings, and may potentially lead to dynamic instability of the propulsion system.

Consequently, alignment of the propulsion system becomes increasingly sensitive to small deviations in bearing vertical offsets and angular misalignment. This sensitivity presents difficulties in analyzing the alignment and the accurate conducting of shaft alignment on board, and may become the source of bearing performance problems and possible damage.

In order to minimize the possible occurrence of alignment-related problems, it is important to verify that:

- The shaft alignment analysis provides the necessary, accurate and applicable data to support each stage of the actual alignment process, and
- The alignment procedure is conducted with the required accuracy in accordance with Class Rule requirements and good marine practices.

3.2 Shaft Alignment Design

The objective of the alignment process is to verify that calculations and onboard verification, conducted in the static condition, will result in the problem-free operation of the propulsion shafting. It has been proven that an acceptable static shafting alignment is a prerequisite for a trouble-free dynamic operation of the propulsion shafting.

Propulsion shaft alignment design is based on assumptions and simplifications applied in the analytical model. The final alignment of the propulsion system on board is contingent on shipyard procedures, prior experience and production practices. Enhanced shipyard production efficiency and modern work methods do not always support the possible impact of improved ship designs on propulsion shafting alignment accuracy. For example, it is more difficult to maintain control of stern tube bearing misalignment when the stern tube bore sighting is conducted in the early stage of block erection, rather than when all hull structural works are completed.

Shafting alignment design has to adapt to these changes. Engineers are obliged to evaluate the impact that particular shipyard practices may have on the final alignment condition. Designers need to consciously account for all relevant parameters. Parameters that were once insignificant, or had a minor impact on the overall alignment condition, may now be required to be accounted for in the design stage. The design must consider not only the final alignment service condition but as well the intermediate stages specific to the alignment production process and provide the necessary information for verification of alignment in those different stages.

The shaft alignment design challenges can be summarized as follows:

- High sensitivity of the shaft alignment to small disturbances in the bearing vertical position
- Disparity between highly flexible hull girder structure and the rigid propulsion shafting
- Appropriately evaluating hull girder deflections
- Difficulties in obtaining critical service propeller loads during maneuvering
- Problems in maintaining the desired accuracy of the shaft alignment analysis
- Inconsistencies and inaccuracies in the shaft alignment production process

There are several reasons for the challenges of shafting alignment in modern vessels.

Allowable shaft alignment tolerances are measured in fractions of millimeters, while acceptable hull construction margins are measured in tens of millimeters. This poses a problem if alignment is not conducted in the very latest stages of ship construction, because even small structural corrections may significantly affect alignment.

Ideally, a shaft alignment analysis should be performed for maximum allowable alignment tolerances, resulting in acceptable bearing reactions and misalignment angles under all operating conditions of the vessel: loaded, ballast, hot and cold. This implies that hull deflections are initially accounted for in the analysis. Because hull girder deflections are not simple to predict, calculate or measure, it is difficult for the industry to comply with the requirement that effects of hull deflections be accounted for in alignment calculations.

The aft stern tube bearing is a very sensitive part of the installation and is more affected by the way the propeller loads are implemented in the slope boring design, rather than by changes in bearing offsets. Appropriately addressing the slope boring is crucial to prevent aft stern tube bearing damage and failure. Slope boring or the bearing inclination are used in marine industry to achieve the required level of alignment tolerances and more uniform load distribution over the bearing length. Aft stern tube bearings with double slope designs are becoming more widely accepted as they provide an improved bearing performance margin for critical service periods and is thus preferred over a single slope design.

It is also recognized in the industry that a gradual exposure of the aft stern tube bearing to heavier loads, through a controlled process called bedding-in, improves bearing performance by increasing the contact area between the shaft and the bearing.

The alignment procedure is not consistent across the industry; shipbuilders have adopted diverse vessel construction practices, resulting in a range of different approaches to shafting alignment. These Guidance Notes cannot investigate and address all approaches, but instead focuses on some of the more extreme cases that could potentially lead to propulsion installation damage and failure.

Alignment sensitivity is traditionally measured by the bearing reaction response to a bearing offset change. On sensitive installations such as VLCCs, ULCCs, and other large vessel types mentioned above, a fraction of a millimeter disturbance in vertical bearing offset may cause a significant change in bearing reaction. Most damages to and failures of propulsion shafting bearings are related to an improperly executed alignment.

Crankshaft deflections and propulsion gear misalignment can also cause significant problems when the engine maker's and the gearbox manufacturer's requirements are not adhered to in the shafting alignment design and subsequent installation and measurements on board.

Stern tube seals may be adversely affected due to excessive stern tube bearing wear, or when an inappropriate alignment design results in vibration of the propulsion shafting system.

Sleeve couplings, clutches and shaft coupling bolts should be evaluated for any potential impact on the shaft alignment condition but will generally be less susceptible to alignment issues if alignment related bending effects and displacements are accounted for in the shaft alignment design.

3.3 Solution to the Alignment Challenge

Conventional propulsion shafting alignment design is based on static forces and moments. This approach may be considered an acceptable industry practice, since the primary purpose of the analysis is to support the static shafting alignment procedure. Evaluation of the impact of dynamic loads is however still desired, particularly on alignment sensitive installations where the aft stern tube bearing performance may be a cause of concern.

These *Guidance Notes* primarily refer to a static shaft alignment with the focus on evaluation of propulsion shafting bearings performance. The preferable condition for conducting the propulsion shafting alignment procedure is in the dry dock with the vessel's structural work nearly completed, and close to launching. Subsequent construction works once afloat are generally not expected to disturb the shafting alignment condition.

However, to comfortably rely on shafting alignment conducted in dry dock, hull deflections need to be predicted with relatively high confidence. Hull deflections can be either calculated or obtained through measurements. Since the hull deflections we are interested in are those that occur in the stern structure of the vessel where the propulsion shafting is located, the calculations mostly require relatively fine finite element models to capture all relevant structure for desired accuracy.

Employing a simplified approach based on the beam theory is an alternative analytical solution that is practical and less time consuming. However, the beam theory approach requires additional calibration through a comparison with finite element models or actual measurements (Section 6).

The beam theory approach, combined with data obtained from measurements, is utilized in the hull deflection estimation process embedded in the ABS shaft alignment software. The software is used for shaft alignment design optimization as is discussed further in these Guidance Notes.

The knowledge and expertise acquired by ABS from extensive field research and theoretical development is applied for improving ABS Rules and the services that ABS offers to clients. Services offered include the following:

- Optimization of alignment offset design by taking into account hull girder deflections under different loading condition of the vessel
- Full scale dynamic investigation (Fluid Structure Interaction – FSI) into the interaction between the propeller shaft and the stern tube bearing for slope boring optimization
- Onsite assistance to ship builders and ship owners to resolve alignment design and alignment-related bearing damages and failures
- Performance monitoring of stern tube bearings and intermediate shaft bearings

3.4 Analytical Support

The objective of shaft alignment analysis is to define and provide a set of alignment parameters the shipyard can use for the purpose of aligning the propulsion system in accordance with design values, and verification that the alignment is conducted within acceptable tolerances.

The industry offers a number of computer programs that can accurately and comprehensively address basic shaft alignment design and produce the desired parameters to assist in the acceptable execution of shaft alignment on board.

In comparison, ABS shaft alignment software has several key distinguishing features. In addition to producing the commonly required shaft alignment calculation features such as bearing reactions, misalignment angles, sag and gap data, and an influence coefficient matrix, ABS alignment software also provides:

- Estimation of hull deflections for a given ship type and basic ship parameters
- Alignment optimization for a given hull deflection range including thermal growth
- Evaluation of stern tube bearing contact and misalignment condition
- Jack-up simulation diagrams

The basic ABS shaft alignment analysis software package is available for distribution to eligible third parties and includes shaft alignment model samples and a user’s manual. Software application training is provided on request.

More specialized services such as a full-scale FSI analysis of the aft stern tube bearing for critical service conditions, specifically for high speed maneuvering and with a partly immersed propeller may be performed upon client request. In addition, 3D, six-degree-of freedom shaftline model analyses (i.e., whirling calculations, strength and fatigue calculations for shaftlines of ice class vessels, et al.) can be performed using the ABS Integrated Powertrain Tool (IPT), an ANSYS-based ABS customized software.

4 Rule Requirements

In the continuously improving shipbuilding environment, the role of the Class Society in addressing and preventing shafting alignment problems has become significantly more acute. ABS assumes a proactive approach by continuously reviewing and adapting Classification Rules and requirements for propulsion shafting alignment to stay abreast of current industry practices, and accordingly mitigate the risk of alignment related damages and failures.

A brief overview of ABS Rule requirements for propulsion shafting alignment calculations is described below. The reader should refer to the most current ABS Rules for more details and recent Rules updates.

Shafts and associated components used for the transmission of power essential for the propulsion of the vessel are to be designed and constructed to withstand the maximum working stresses to which they may be subjected in all service conditions. Consideration in the design should also be given to additional stresses in the shafting system resulting from the installation and shaft alignment as well as from vibrations.

In general for all ships, shaft alignment calculations and a shaft alignment procedure should be submitted to ABS for reference. However, for propulsion shafting of diameters equal to or larger than 300 mm (11.81 in.), and propulsion shafting with no forward stern tube bearing, they are to be submitted for review by ABS for compliance with ABS Rule requirements.

The alignment calculations should include bearing loads and bearing reactions, shear forces and bending moments along the shafting, slope boring details as applicable, and a detailed description of the alignment procedure.

The alignment calculations are carried out for theoretically aligned cold and hot conditions of the shafting and for the maximum allowable alignment tolerances. They are to show that:

- Bearing loads under all operating conditions are within the acceptable limits specified by the bearing manufacturer
- Bearing reactions are always positive (i.e., supporting the shaft), except as determined acceptable in accordance with current ABS Rule requirements
- Shear forces and bending moments on propulsion equipment are within the limits specified by manufacturers
- Shear forces and bending moments at the crankshaft flange are in accordance with the engine manufacturer's limits
- The designed relative misalignment slope between the shaft and the aft stern tube bearing is to be positive, and not to exceed $0.3*10^{-3}$ *[rad]*

5 Shaft Alignment Survey

When alignment calculations are required to be submitted in accordance with ABS Rules, the calculated data should be verified and recorded by appropriate measurement procedures in the presence and to the satisfaction of an ABS Surveyor.

To prepare for propulsion shafting installation and alignment surveys, the surveyor reviews ABS Rule requirements and possesses a copy of and is familiar with the ABS approved shaft alignment calculations and the builder's Inspection and Test Plan (ITP). The surveyor also references the ABS *Propulsion Shafting Alignment Guidance Notes* to acquire in depth technical information regarding design and review, the shaft alignment procedure and measurements.

During survey preparation, the surveyor collects information from relevant documents, plans, and publications in support of onboard verification.

5.1 Required Information

Surveyor needs to be provided with the ABS approved shaft alignment calculation, alignment procedure and the associated review approval letter and be familiar with the calculations, hot and cold static conditions, and the jack-up method described therein.

Systems, machinery, and components are verified based on ABS approved plans and calculations.

Manufacturers provide the shipyard with drawings and technical documents relevant to shafting installation and alignment. The surveyor may request additional technical data of interest from the shipyard to support required inspections and surveys.

Typically, the Surveyor would familiarize with the following information:

- Prescribed bearing offsets
- Bearing reactions
- Design slope and slope boring requirement of the aft stern tube bearing
- Sag and gap table for the drydock and afloat condition
- Prescribed jack-down force of the tail shaft and location of temporary supports for sag and gap measurements
- Shaft deflection in way of forward seal for systems without forward stern tube bearing
- Positions of the hydraulic jack, jack correction factors and theoretical jack-up diagrams
- Influence coefficient matrix
- Main engine manufacturer's crankshaft deflection tolerances for the drydock and afloat condition
- Centering tolerance of aft and forward stern tube seals
- Push-up length and designed fitting force of stern tube bearings at their corresponding temperatures
- Clearance tolerance for tail shaft and stern tube bearings
- Tail shaft bearing clearance calculation
- Shafting arrangements in relation to sighting and measurement inspections
- Pull-up length and design fitting force of propeller at its corresponding temperature
- Dimension and fitting tolerance of shafts coupling bolts
- Approved alignment procedure.

5.2 Attendance

The ABS Rules identify the procedures required for shaft alignment and the specific shaft alignment steps and records the surveyor must witness and verify to confirm that alignment is carried out in agreement with the approved calculations and procedures.

The Inspection and Test Plan (ITP) is a shipyard production document that lists the inspection points for installing and testing components and parts. These inspections take place throughout the shaft alignment process. The surveyor reviews the ITP and verifies that it covers all of the ABS Rules required attendance points for shaft alignment, propulsion shafts, and associated parts. The surveyor rates each ITP point for review, witness, or for monitoring.

Surveyor's presence is sometimes requested by the ship owner and shipyard for activities where ABS Rules do not mandate attendance. What follows is a list of typical inspection items where surveyor's attendance is either required or advisable:

- Equipment calibration records
- Pre-shaft-assembly survey
- Shaft run-out
- Sighting through
- Stern tube bearing fitting
- Slope boring or bearing inclination

- Stern tube bearing clearance verification
- Sag and gap measurement verification
- Shafts installation
- Bearing reaction measurement verification
- Shaft bearing horizontal clearance verification
- Engine bedplate deflections
- Crankshaft deflections measurement verification
- Gear contact verification
- Gear shaft bearing reactions.
- Run-in procedure

6 Commitment to Safety

ABS personnel will follow the client's or facility's established safety procedures when on premises, provided that the established safety procedures are not less effective than those contained in the ABS Safety Manual and its associated procedures. If ABS personnel encounter conditions or procedures that may compromise their safety, they may stop their work immediately until corrective actions are taken. Nothing in the latest revision of the ABS Safety Manual is intended to replace or supersede any governmental or local authority's regulations or requirements for the implementation of or content of a premises safety plan, provided such plan is not less effective than the safety policies contained in the ABS Safety Manual.

SECTION 2 Shaft Alignment Design and Review

1 General

This Section denotes issues relevant to shaft alignment design and review. In principle the designer confirms, and the reviewer verifies, that the strength and performance of all parts constituting the propulsion system (e.g., bearings, shafts, coupling bolts, couplings, clutches, and gears) are not adversely impacted by the shaft alignment design.

In particular, the alignment design must address and satisfy the following:

- Acceptable bearing reaction loads
- Acceptable bearing misalignment limits
- Uniform load distribution all bearings
- Shaft strength limits
- Satisfactory crankshaft deflections
- Acceptable gear contact condition
- Satisfactory coupling bolts strength
- Acceptable misalignment tolerances for clutches and flexible couplings

2 Review vs. Design

Analytical models do not always represent the propulsion systems accurately and don't necessarily offer sufficient information to provide an error free alignment production procedure. This Section addresses the potential incompatibility between the analysis and the shaft alignment on board, discusses solutions and provides guidance where possible.

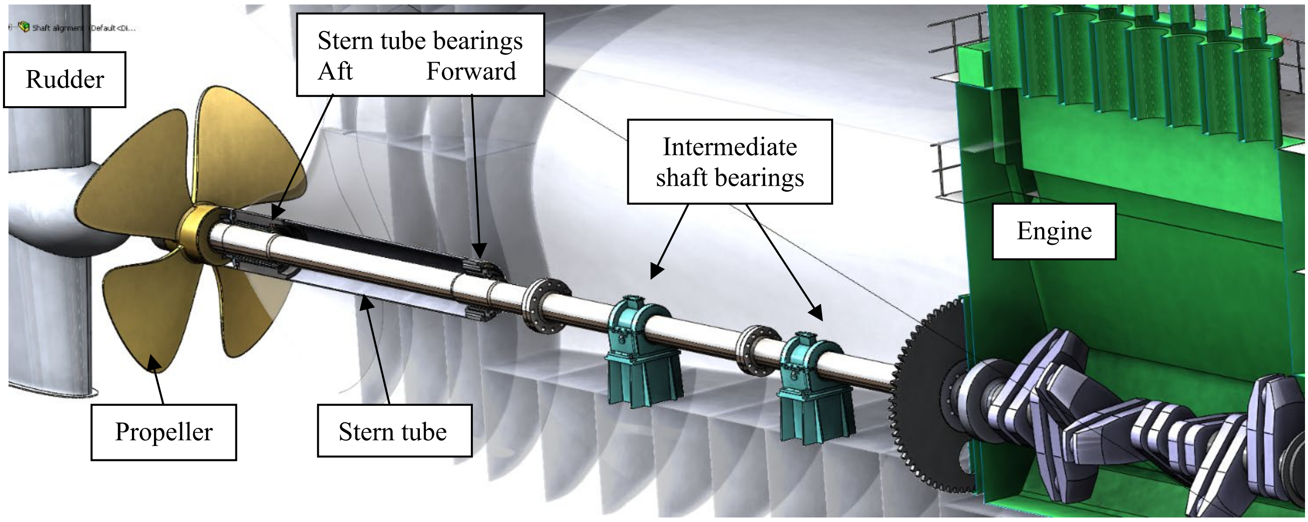
Firstly, the differences between review and design process are defined. The review process serves to verify soundness of the design that as minimum has to thoroughly follow the alignment requirements and guidelines defined in ABS Rules. These requirements are performance and strength related, with specific concerns for safety of life, equipment and the environment.

The design process is more complex and requires experienced engineers. It is a time-consuming effort with a goal to define a satisfactory set of parameters that comply with all relevant alignment requirements. The primary parameters are the bearing vertical offset, bearing longitudinal position, and the slope-boring angle of the aft stern tube bearing. The design must likewise verify that the alignment complies with shaft geometry, material properties, installation constraints and other requirements related to propulsion shafting interaction with surrounding systems.

A correctly executed design process essentially optimizes propulsion shafting alignment for the given parameters. This not only implies the use of an optimization tool or software, but also a skilled engineering process of combining the design parameters into an acceptable solution that complies with imposed Rules, regulations and requirements.

The design and review requirements in these Guidance Notes are described primarily for installations involving direct drive propulsion systems such as shown in Section 2, Figure 1. Other designs may have slightly different requirements but, in general, a similar approach applies.

FIGURE 1
Directly Coupled Propulsion Shafting – Example



3 Review

Plan review during and after construction is conducted by ABS, to verify that a vessel, structure, item, equipment or machinery is in compliance with the Rules, Guides, standards or other applicable criteria.

Engineers verify that they received all the required plans and particulars:

- Shaft alignment model
- Scope of submitted calculation
- Analysis results
- Shaft alignment procedure
- Run-in procedure

The Engineer conducting the review of the submitted shaft alignment documents the review results.

In reviewing the submitted shaft alignment analysis and procedure, the engineer inspects the alignment analysis results and conducts a check analysis using the ABS shaft alignment software.

The reviewer pays particular attention to installations with no forward stern tube bearing (NFSB) where the alignment procedure is specifically customized to verify that after completion of the bore sighting the misalignment on the aft stern tube bearing will remain within Rule limits and will remain so after completion of all shaft alignment work.

The following list of plans and particulars present what is considered an industry-wide general set of best practices in addressing and proposing solutions for shaft alignment issues.

3.1 Plans and Particulars

The engineer is to be provided the following plans and documentation during plan review of shaft alignment calculations and procedures.

3.1.1 Propulsion Shafting

- Shafting arrangement
- Rated power of main engine and shaft rpm
- Thrust shafts, line shafts, tube shafts and tail shafts, as applicable

- Couplings – integral, demountable, keyed, or shrink-fit – coupling bolts and keys
- Line shaft bearing details
- Stern tube bearings detailed drawings
- Main engine, and if applicable gearbox bearings
- Allowable bearing loads
- Stern tube seal arrangements
- If applicable, power take-off to shaft generators, propulsion boosters, or similar equipment, rated 100 kW (135 hp) and over
- Material properties of the shafts and the bearings (modulus of elasticity and density)
- Propeller mass and material properties

3.1.2 Cardan Shafts

- Dimensions of all torque-transmitting components and their material properties
- Rated power of main engine and shaft rpm
- Engineering analyses

3.1.3 Calculations and Procedures

- Propulsion shaft alignment calculations
- Detailed shaft alignment procedure
- Run-in procedure for alignment sensitive installations

3.1.4 Diesel Engine

- Crankshaft equivalent model
- Location and mass of the following equipment, when applicable: flywheel, chain drive tightening load, cam-drive gear, torsional vibration damper
- Detailed drawings, location and material properties of the main engine bearings
- Allowable bearing loads

3.1.5 Reduction Gear

- Detailed drawings of gear shafts and main gears with material properties
- Mass and location of the gears
- Location and material properties of the gear shaft bearings
- Allowable bearing loads

3.1.6 Stern Tube Lubrication Arrangement and Oil Analysis

The stern tube lubrication arrangement is preferably also submitted to ABS for review. Of particular interest for alignment sensitive installations is consideration of the fresh lubrication oil inlet at the aft end of the aft stern tube bearing.

Lubricating oil analysis is to be conducted after sea trials of the vessel.

3.1.7 Run-in Procedure

A shaft alignment run-in procedure with details on vessel speeds, rudder angles, and duration of each run-in sequence is to be submitted for review for installations with no forward stern tube bearing and for shaft installations with a double slope stern tube bearing design.

3.2 Sign Convention

Sign convention may differ among shipyards, design companies, and Class societies. It is therefore important that alignment calculation includes the sign convention applied.

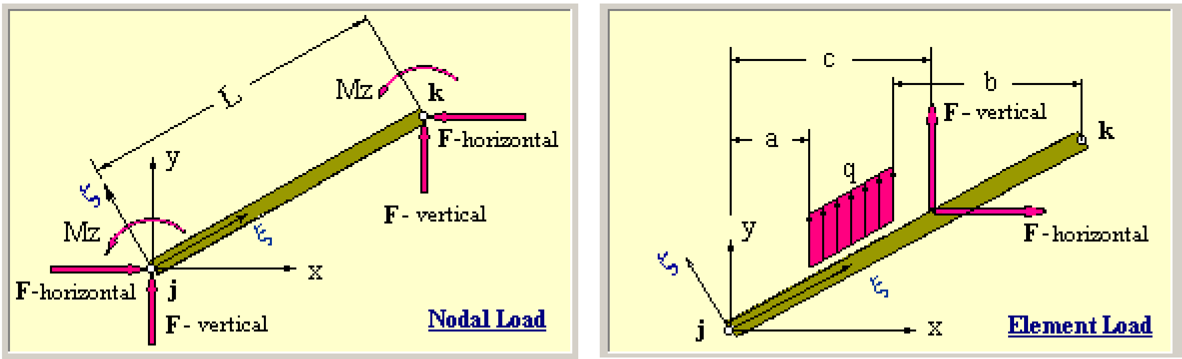
ABS shaft alignment software uses the following sign convention:

3.2.1 Forces and Moment

The right-handed Cartesian system is used as a referenced coordinate system. Loads (forces, moments) and displacements (linear and rotations) are positive when complying with Cartesian system convention.

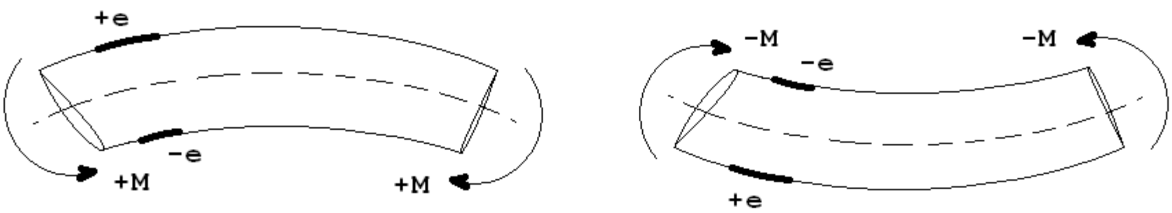
As per Section 2, Figure 2, vertical forces applied in the global *'x, y'* Cartesian coordinate system are positive if acting in positive direction of *'y'* coordinate axis.

FIGURE 2
Load Application – Sign Convention



Moments and horizontal forces are defined by the coordinate axis direction, as well as by the side (left or right) of the element's cross section that they act upon (Section 2, Figures 2 and 3).

FIGURE 3
Moment Sign Convention in ABS Shaft Alignment Program



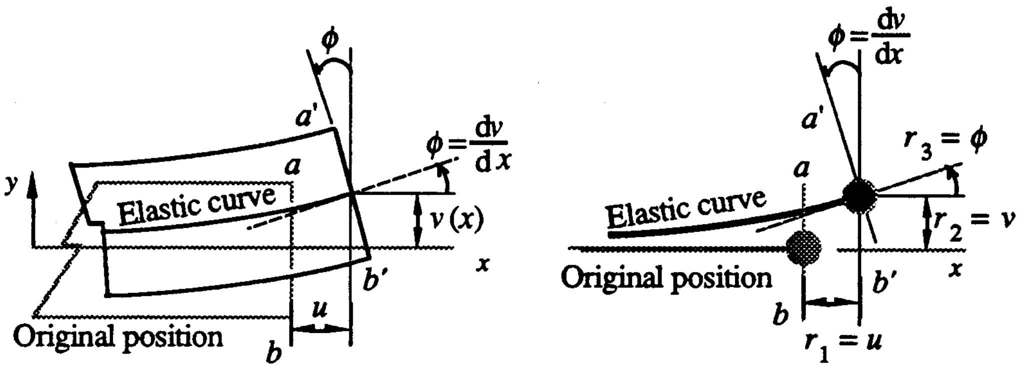
3.2.2 Displacements and Angular Deflection

Section 2, Figure 4 shows what ABS software considers to be positive nodal displacements in *x* (axial) and *y* (vertical) directions and positive rotation about the z-axis:

- *u* represents horizontal displacement of the node measured from its original undeformed location, and it is equal to displacement vector variable *dx*
- *v(x)* represents vertical displacement (*y* axis) of the node measured from its original undeformed location, and it is equal to displacement vector variable *dy*, and

- *F* represents nodal rotation measured from the original global horizontal axis, and it is equal to displacement vector variable *Rot(z)*. Rotation about z axis is the actual slope at the node and is considered positive when the rotation is counterclockwise.

FIGURE 4
Axial, Vertical and Rotational Beam Displacements



3.2.3 Shaft Centerline vs. Bearing Center Line

It is important to recognize which approach is used for the offset definition when different computational approaches are compared.

ABS shaft alignment software defines bearing offset based on the shaft centerline as a reference. When dealing with journal bearings, the advantage of this approach is that there is no need to know correct value of the bearing clearance to accurately complete the shaft alignment analysis. However, when sighting trough information is provided to the shipyard, the bearing offset is to be corrected to the bearing centerline for correct bearing positioning. The applied correction value is one-half of the bearing clearance.

Software that utilizes the bearing centerline to determine offset will require precise knowledge of the bearing clearance for correct shaft alignment calculation. This poses a problem as a final bearing design, including the clearance, may not be known until the later stages in the project.

Note: The above discussion applies to journal bearings only.

3.3 Shaft Alignment Model

By exercising sound judgment, the engineer verifies that the submitted shaft alignment model represents the actual propulsion system with sufficient accuracy.

The engineer verifies that the line shaft model and if applicable, the reduction gear model, correspond to the respective design drawings. The diesel engine equivalent model is evaluated by confirming compliance with engine design particulars: engine type, diameters, location of the timing gear, etc.

Bearing offsets are to be verified for both hot and cold conditions. External loads are to be confirmed to be appropriately accounted for and applied at the proper location.

3.4 Scope of Calculation

The shaft alignment calculation provides a robust shafting alignment design and is expected to account for all relevant parameters influencing the propulsion system. The calculation report is to be customized to provide the information and data that shipyard personnel need in order to achieve satisfactory alignment under all service conditions of the vessel, i.e. from ballast to fully loaded. Accordingly, the calculations submitted for review are to include information for conducting and verifying alignment procedures for:

- Vessel in dry dock with cold engine and gear box
- Vessel afloat with cold and hot engine and gear box

Per ABS Rules, alignment calculations are performed for the maximum allowable alignment tolerances and include bearing reactions, shear forces and bending moments along the shafting. The analysis should demonstrate that:

- Bearing loads under all operating conditions are within the acceptable limits specified by the bearing manufacturer
- Bearing reactions are always positive, i.e. the bearings support the shafts
- Shear forces and bending moments on propulsion equipment and at the crankshaft flange are within the limits specified by manufacturer

The analysis should also include the following:

- For geared systems, the calculated misalignment between main gear and pinion is to be less than $0.1*10^{-3}$ *[rad]*,
- The designed relative misalignment slope between the shaft and the aft stern tube bearing is to be positive
- The stern tube bearing fitting calculation based on the actual interference fit tolerances, including fitting pressure and push-in distance
- A clearance calculation on aft and forward stern tube bearings with alignment model showing only the propeller shaft supported on two stern tube bearings. Installations with no forward stern tube bearing may be subject to specific requirements.
- Sag and gap values and the location of temporary supports
- Jack-up locations for the verification of bearing reactions

Because the alignment procedure starts in the dry dock with sighting, positioning of bearings, and slope boring, the alignment calculation needs to provide sufficient information for shipyard personnel to accurately perform the shaft alignment procedures in dry dock. It may be beneficial to also conduct sag and gap and initial bearing reaction load verifications while the vessel rests on blocks in the dry dock when alignment is not affected by hull deflections. As such, the dry dock calculation values can be more accurately verified before launching the vessel, but this implies that hull deflections are already accounted for in the alignment design.

It is important to evaluate the alignment's sensitivity to hull deflections with the vessel afloat, even if the hull deflections are not known. When hull deflections are not available, the assessment of alignment sensitivity to hull deflections may be considered by consulting the bearing influence coefficient matrix as explained later in the Guidance Notes.

This on its own is insufficient to assess sensitivity to hull deflections. Some knowledge of the vessel's basic structural data is also required, and at least one analysis is expected to be provided for the vessel's afloat condition and with a hot engine. This condition implies a fully or partially submerged propeller. Measurements for this condition should confirm that all bearing reactions are positive, and within reasonable tolerances compared with the corresponding calculated values. This confirms that the dry-dock condition values can provide the expected acceptable results.

Analytical data for sag and gap is normally provided either for the dry dock condition or for the afloat condition. The effects of hull deflections are neglected in this part of analysis. Sag and gap measurements alone are insufficient to accept a shaft alignment condition, which must be validated with subsequent bearing reaction measurements after the shafting is assembled, either in the dry dock or afloat.

3.5 Results Verification

For the purpose of calculation results verification, the calculation should as a minimum include:

- Influence coefficient matrix
- Bearing reactions and allowable loads on all bearings
- Stern tube bearing clearance

- Deflection curvature
- Stern tube bearing slope boring requirements
- Angular inclination at the main gear wheel
- Shear forces and bending moments

3.5.1 Influence Coefficient Matrix

The influence coefficient matrix tabulates the bearing reaction change respective to the unit offset change at a particular bearing (Section 2, Table 1). Accordingly, the influence coefficient matrix can be used to evaluate shafting sensitivity to possible disturbances in bearing offsets and assess the impact of this on the bearing reactions. The bearing offset disturbances of concern are:

(a) Hull deflections

(b) Thermal deviations

(c) Bearing offset adjustment

3.5.1(a) Hull Deflections.

ABS has developed a methodology to account for hull deflection impact on the shafting alignment for several categories of ships, such as tankers, bulk carriers and container vessels. The established approach allows designers using ABS software to estimate hull deflections of those vessels with relatively high confidence.

Where the hull deflections are not available, the influence coefficient matrix can be used to assess the impact of hull deflection on the propulsion shafting. One can obtain information about the sensitivity of the system to potential disturbances affecting the bearing offsets, but not the actual magnitude of those disturbances. The ratio between the stiffness of the shafting and the hull structural stiffness can be assessed, and this information can be of assistance when establishing bearing offsets during the alignment design process.

The following three scenarios demonstrate a possible application of the influence coefficient matrix in estimating the impact of the hull deflections on the propulsion shafting. Only a static condition and structural deflections resulting from a change in the vessel's draft are considered.

i) *Scenario 1 – Compliant System*: Proportionally rigid shafting and hull:

If the stiffness of the shafting is proportional to the stiffness of the hull structure, theoretically, it is expected for the shafting to follow the flexing of the hull without unloading any of the bearings.

Rigid shafting will have a large influence coefficient and will be sensitive to very small deviations in the bearing offsets. However, at the same time, the proportionally rigid hull structure will not adversely disturb the bearing offsets because deflections of the vessel's stern part are also expected to be of a similar magnitude.

This scenario is typical of a smaller vessel, for example below *30,000 DWT*.

ii) *Scenario 2 – Noncompliant Shafting*: Rigid shafting and elastic structure:

Rigid shafting and a relatively flexible structure, although not desired, is very common on modern commercial vessels.

Stiff shafting and a relatively elastic structure will most probably have as a consequence a propulsion system that is highly sensitive to hull deflections.

Stiff shafting has large influence coefficients and cannot accommodate hull deflections. Bearing reactions may change significantly through flexing of the relatively elastic hull. As the hull bends, the relatively rigid shaft may not deform enough and may unload some of the bearings.

This scenario applies to large vessels with relatively short, large diameter shafting. Although not desirable from the point of view of alignment, these types of arrangements are very energy efficient and are common on VLCCs, ULCCs, large bulk carriers and the container vessels.

A sample influence coefficient matrix that corresponds to the sensitive alignment of a large bulk carrier is shown in Section 2, Table 1.

iii) *Scenario 3 – Compliant Shafting*: Elastic shafting and rigid structure:

This is the preferred condition. Elastic shafting and a relatively rigid hull structure will result in a very submissive shafting, which regardless of hull deflection maintains contact with the bearings.

3.5.1(b) Thermal Deviations.

In contrast to hull deflections which affect the entire propulsion system and all the bearings simultaneously and are represented by a smooth continuous curve, the effect of thermal deviations is typically local to a particular bearing or equipment such as the engine and the reduction gearbox.

When the structure – for example an oil tank – below a particular bearing expands or contracts due to a temperature change, the offset at the affected bearing changes as well. The influence coefficient matrix can be directly applied as an investigative tool to assess how this local bearing offset change affects the overall load distribution among the other bearings in the system.

Temperature changes in the diesel engine and the reduction gear foundation will result in simultaneous offset changes in all of the engine and gearbox bearings. It is often assumed in alignment calculations that all engine bearings are affected by thermal changes equally. This is not completely true. The midsection of a diesel engine expands more due to lesser heat dissipation than the forward and the aft engine structure. The bedplate pre-sag is commonly applied to compensate for this thermal discrepancy of the diesel engine.

3.5.1(c) Bearing Offset Adjustment.

Correction of the alignment condition may be necessary when:

- The bearing reaction measurements show a large deviation from the calculated values, or
- The diesel engine crankshaft deflections do not meet the engine designer's requirements, or
- Gear tooth contact area is smaller than that required by the Rules.

The bearing reactions are often fine-tuned by offset adjustment at the intermediate shaft bearing(s). This adjustment has a local effect similar to the thermal changes explored above.

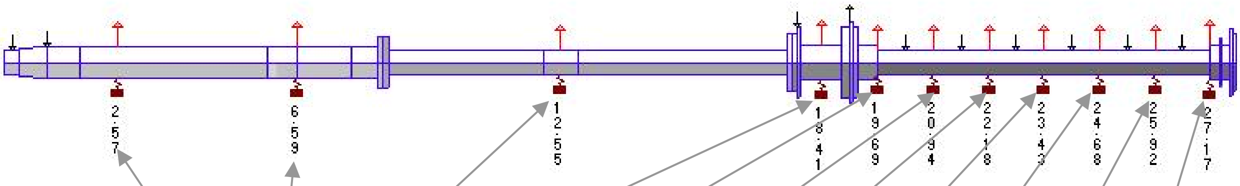
The assessment of alignment condition by application of the influence coefficients can be very useful and fast in cases like these. Read further in the Guidance Notes how bearing reactions are adjusted for installations with no forwards stern tube bearing.

3.5.2 How to Read Influence Coefficients

The influence coefficient matrix tabulates the relationship among relative bearing reactions resulting from a unit offset change at each particular bearing. A sample influence coefficient matrix is shown below.

The sample propulsion shafting system has *11* bearings.

TABLE 1
Influence Coefficient Matrix



RELATIVE BEARING REACTIONS [kN] -> R[0.1-offset]-R[0-0ffset]
Due to 0.1[mm] OFFSET relative to the ZERO bearing Offset

Node	Supp.	< 7> 1	< 14> 2	< 27> 3	< 41> 4	< 46> 5	< 48> 6	< 50> 7	< 52> 8	< 54> 9	< 56> 10	< 58> 11
< 7>	1	4.289	-7.896	4.768	-3.222	2.120	-0.069	0.014	-0.003	0.001	-0.000	0.000
< 14>	2	-7.896	15.425	-11.112	9.951	-6.546	0.215	-0.044	0.009	-0.002	0.000	-0.000
< 27>	3	4.768	-11.112	13.421	-25.421	18.858	-0.618	0.125	-0.025	0.005	-0.001	0.000
< 41>	4	-3.222	9.951	-25.421	166.594	-250.824	123.820	-25.141	5.104	-1.035	0.202	-0.028
< 46>	5	2.120	-6.546	18.858	-250.824	511.825	-379.402	125.081	-25.395	5.148	-1.004	0.141
< 48>	6	-0.069	0.215	-0.618	123.820	-379.402	461.940	-295.733	108.086	-21.910	4.273	-0.599
< 50>	7	0.014	-0.044	0.125	-25.141	125.081	-295.733	389.320	-280.955	104.921	-20.460	2.870
< 52>	8	-0.003	0.009	-0.025	5.104	-25.395	108.086	-280.955	386.156	-279.504	100.648	-14.120
< 54>	9	0.001	-0.002	0.005	-1.035	5.148	-21.910	104.921	-279.504	381.883	-259.045	69.538
< 56>	10	-0.000	0.000	-0.001	0.202	-1.004	4.273	-20.460	100.648	-259.045	281.235	-105.848
< 58>	11	0.000	-0.000	0.000	-0.028	0.141	-0.599	2.870	-14.120	69.538	-105.848	48.046

Each column provides bearing reactions change corresponding
to the relating bearing rise of 0.1[mm] at respective bearing
(e.g. column #2 are bearing reactions to 0.1[mm] offset at bearing #2)

Support designation for this particular model:
#1 Aft S/T brg; #2 Fwd. S/T brg; #3 I/S bearing; #4 to #12 M/E bearing;

The larger the influence coefficient number, the more sensitive the particular bearing is to an offset change. More sensitive bearings experience higher reaction load changes for the same amount of offset change.

The relationship between the offset and the reaction load is linearly proportional. Therefore, to obtain a similar matrix for *1 mm* offset change (instead of *0.1 mm* as in the case above) all influence coefficients would be multiplied by a factor of *10*.

Experience shows that a typical shafting model, as in Section 2, Table 1, represents a propulsion shafting system that is relatively sensitive to an offset variation (resulting for example from the hull deflections or thermal deviations). Sensitivity of the alignment is judged relative to the stiffness of the ship structure below each bearing in the system. Flexibility of the double bottom structure is expected to be higher below the intermediate shaft bearings than below the main engine bearings, or within the stern tube block. The interface between the shafting and the main engine, as well as the stern tube block and the double bottom in front, are particularly sensitive areas, because of the sudden change in the stiffness of the supporting structure between them.

Influence coefficients determined for the engine bearings are very high, which leads to the conclusion that the engine is much more sensitive to bearing offset variations than the shafting. Although this is a correct statement, the structure supporting the engine block and crankshaft is also relatively rigid compared to the structure of the double bottom below the shafting. Thus, the deflections within the engine are relatively small and therefore the reactions will not change substantially. This is true for all but the three aftmost engine main bearings which remain sensitive to hull deflections because of the influence of the shafting resting on the more flexible double bottom structure aft of the engine.

Similarly, while the influence coefficients for the shafting are small, a higher load change is expected due to a more flexible double bottom structure.

The reason why the engine is still considered very sensitive to hull deflections is not because of the engine itself, but rather because of the discrepancy in stiffness between the engine structure and the double bottom structure aft of the engine, below the shafting.

Per example in Section 2, Table 1 the two aft most engine main bearings - #4 and #5 - are being very sensitive to an offset variation of the intermediate bearing - #3. The table indicates that raising the offset at the intermediate bearing by *1 mm* unloads engine main bearing #4 by *254 kN* and adds an additional load of *188.6 kN* to main bearing #5.

3.5.3 Bearing Reactions

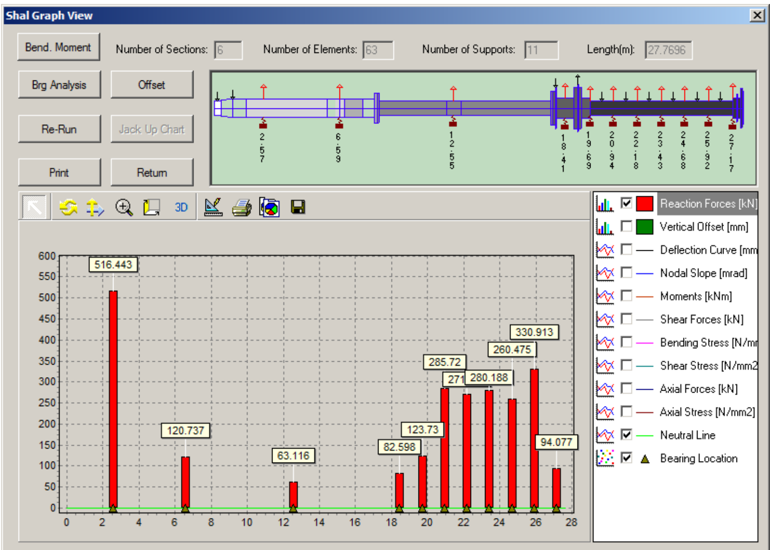
ABS Rules stipulate that the primary criterion for acceptance of alignment is satisfactory bearing reactions. An alignment condition is acceptable as long as the bearing reactions remain positive under all service drafts, and no bearing is unloaded. In general, any positive static load is therefore acceptable.

If the measured bearing reaction for a particular loading draft of the vessel deviates more than *20%* of the calculated value, ABS Rules require that the alignment condition is reassessed, and the cause of the deviation investigated and accounted for in a revised calculation. Section 2, Figure 5 gives an example of calculated bearing loads or reaction forces.

For practical reasons and to prevent unloading or overloading of the bearings due to unaccounted-for disturbances, the following is preferred:

- At least *10%* of the allowable bearing load is desired on each bearing
- Measured bearing reactions may not exceed *80%* of the manufacturer's maximum allowable load limit

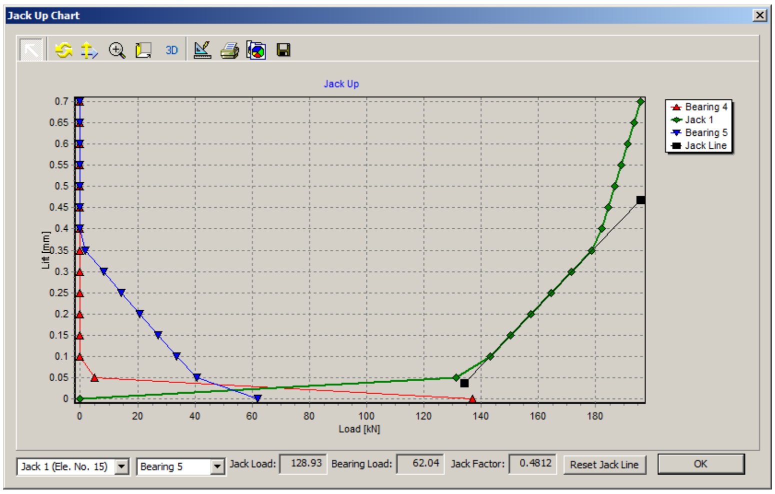
FIGURE 5
Calculated Bearing Reactions



Simulated jack-up diagrams as shown in Section 2, Figure 6 provide valuable information for evaluating the actual bearing load-confirmation. It is suggested that the owner or the shipyard request the calculation of jack-up loads and have the respective diagrams available for the Surveyor.

A more in-depth explanation about the jack-up procedure is provided in Section 4.

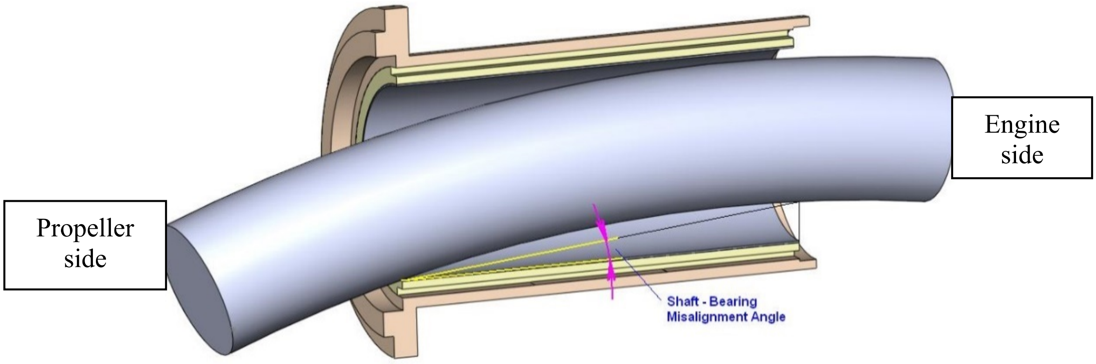
FIGURE 6
Simulated Jack-Up Diagram



3.5.4 Misalignment - Deflection Curve

Besides bearing reactions, the relative angular misalignment between the tail shaft and the aft stern tube bearing is another significant parameter to be evaluated for the alignment acceptance.

FIGURE 7
Stern Tube Bearing Misalignment



Balancing the load distribution on the aft stern tube bearing is much more challenging than on any other bearing in the propulsion system, because of the large cantilever load exerted by the propeller which directly and mostly affects the aft stern tube bearing.

The relative misalignment angle of the shaft inside the bearing is defined by the shaft deflection curvature as shown in Section 2, Figure 7. The angle is measured from the theoretical zero alignment line, as established in shaft alignment calculation.

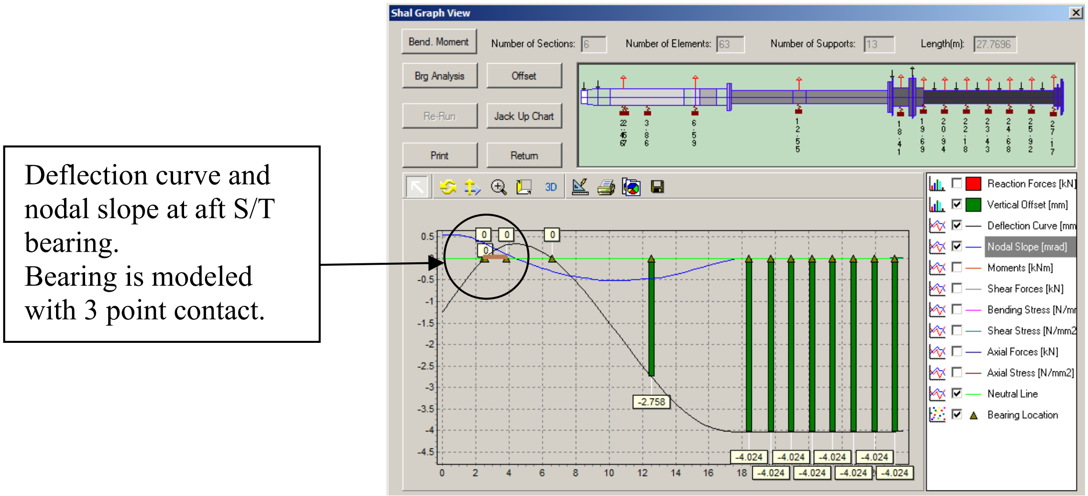
The relative misalignment between shaft and bearing is zero when the shaft is in contact with both the forward and the aft bearing edges.

Because of the relatively high stiffness of the stern block, hull deflections do not significantly affect the aft stern tube bearing misalignment when both the forward and aft stern tube bearings are installed.

However, in installations with no forward stern tube bearing, hull deflections may have a substantial impact on aft bush misalignment. This must be accounted for in the alignment design and when proposing the slope boring and the bore sighting procedures.

If hull deflections are not available, the review engineer can evaluate the potential sensitivity of the misalignment in the stern tube bearing by consulting the influence coefficient matrix as explained earlier.

FIGURE 8
Deflection Curve and Nodal Slope



Note: The ABS shaft alignment software also calculates the nodal slopes but the slope at the nodes which represent bearings should not be mistaken for the bearing misalignment slope.

3.5.5 Slope Boring/Bearing Inclination

Slope boring or bearing inclination is widely adopted as a marine industry practice to prevent excessive edge loading of the aft stern tube bearing.

ABS Rules specify that in installations where the stern tube bearing misalignment angle exceeds 0.3×10^{-3} *rad*, slope boring or bearing inclination slope boring is normally conducted.

The ABS shaft alignment software provides an interactive routine for aft stern tube bearing analysis that includes the slope boring evaluation tool (Section 2, Figure 10).

Conventionally, a single bearing slope is applied throughout the length of the bearing for which the above criteria apply. Recently, it is increasingly accepted that the aft stern tube bearing performance can improve by a double slope application.

3.5.6 Angular Inclination at the Main Gear Wheel

In installations with a reduction gear, an important part of the investigation is the calculation of the shaft bending curvature and definition of the contact condition between the main gear wheel and the pinion. More details on reduction gear alignment are provided later in the Guidance Notes.

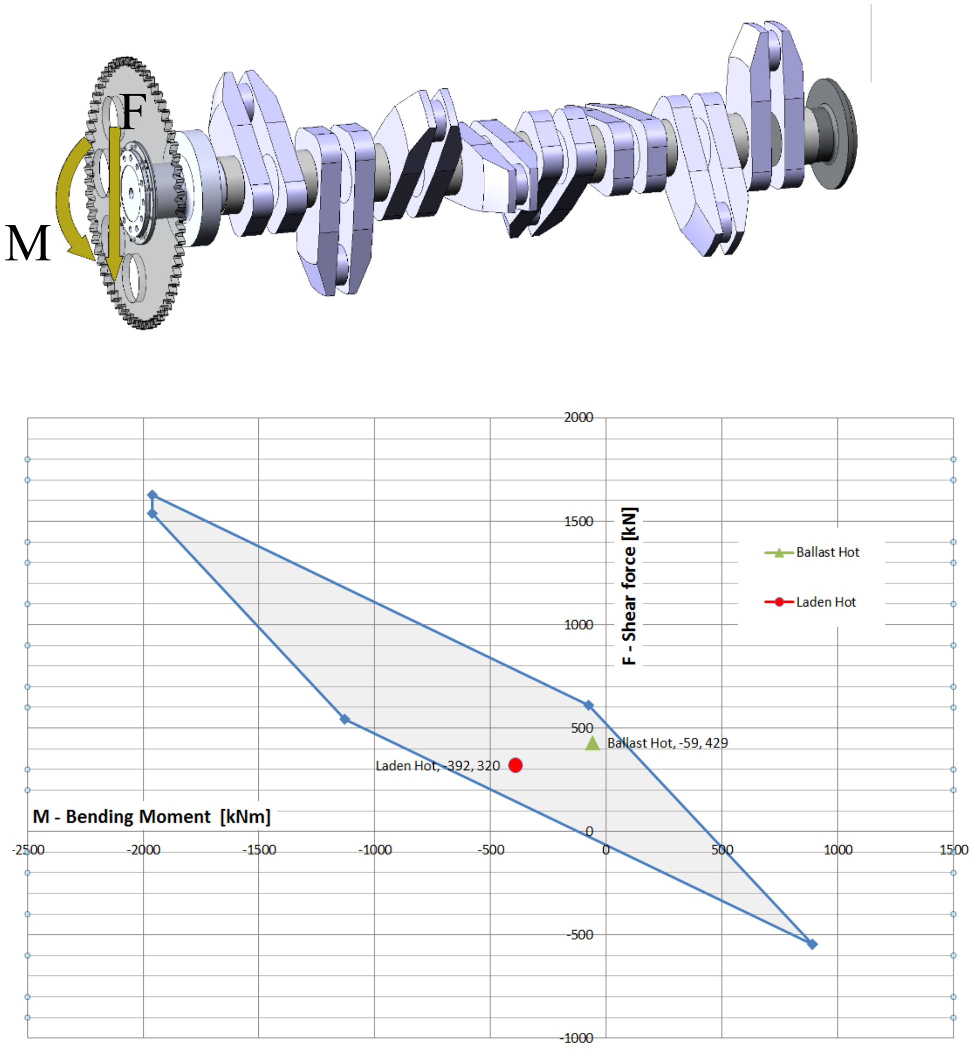
3.5.7 Shear Forces and Bending Moments

Shear forces and bending moments must be kept within acceptable limits, in association with other loads and stresses in the shaft.

Forces and moments on propulsion machinery are to be within the limits specified by the equipment manufacturers.

Some diesel engine manufacturers require bending moments and shear forces at the main engine aft flange to be within the required boundaries represented by the shaded area in Section 2, Figure 9 to protect the engine from eventual harmful misalignments.

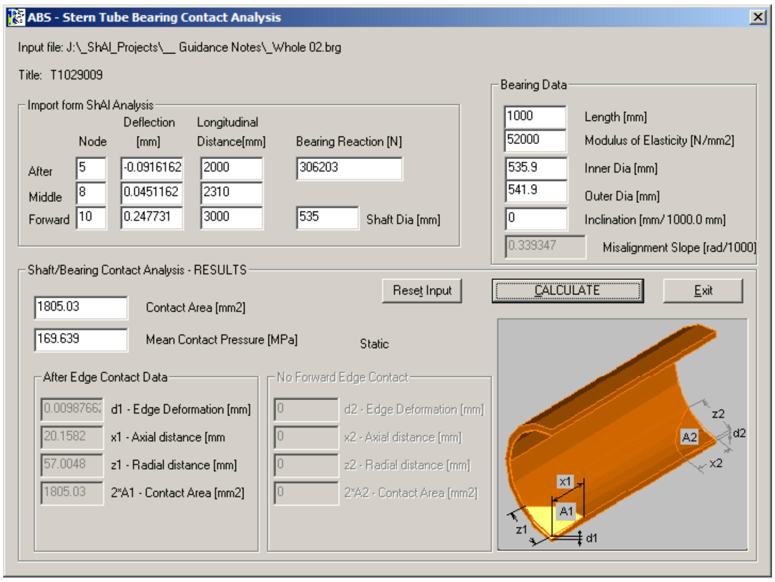
FIGURE 9
Example of Diesel Engine Output Flange Allowable Shear Force and Bending Moment Chart



3.5.8 Allowable Bearing Load

The allowable bearing load correlates to the acceptable compressive stress levels in the bearing material. The current ABS Rules provide the stern tube bearing acceptability criteria of 0.8 N/mm^2 for metallic and 0.6 N/mm^2 of the permissible load for oil-lubricated synthetic bearing materials. These acceptability criteria consider compressive pressure estimated from the bearing load over the projected area of the bearing. The criteria may be insufficient if misalignment between the bearing and the shaft is excessive, resulting in bearing edge loading. Slope boring of the bearing or bearing inclination may be a solution to the edge loading problem. The ABS Shaft Alignment software provides a tool for the aft stern tube bearing static loading analysis and slope boring evaluation (Section 2, Figure 10).

FIGURE 10
Aft Stern Tube Bearing Evaluation Software



3.5.9 Aft Stern Tube Bearing Clearance

ABS Rules require verification of the aft stern tube bearing clearance prior to propeller fitting. A separate clearance measurement calculation is to be submitted to ABS for review and confirmation for that purpose.

The acceptable clearance range is based on its impact on the bearing misalignment. Verification by measurement of the aft stern tube bearing clearance is acceptable if the difference between measured and calculated clearance results in bearing misalignment smaller than *+/- 0.1 mrad*.

There are two distinct stern tube bearing designs, one with and one without a forward stern tube bearing. Each design requires a different approach for calculating and verifying the aft stern tube bearing clearance.

3.5.9(a) Both stern tube bearings installed

In this design, the aft stern tube bearing clearance calculation is to be submitted for:

- The propeller shaft only, supported on the aft and forward stern tube bearings
- Unrestrained shaft at both ends
- Only gravity forces acting on propeller shaft; propeller weight is not considered
- Aft and forward stern tube bearings are modeled using multiple supporting points

3.5.9(b) No forward stern tube bearing installed

In this design, the aft stern tube bearing clearance calculation is to be submitted considering the following:

- Propeller shaft and intermediate shaft are connected and supported only on the aft stern tube bearing and the intermediate bearing.
- Propeller shaft is unrestrained at the aft end
- Intermediate shaft is unrestrained at the forward flange
- Aft stern tube bearing is designed with a double slope and modeled using multiple supporting points
- Intermediate shaft bearing is modeled with multiple supporting points

- The stern tube lubrication arrangement is designed with the fresh oil inlet located aft of the aft stern tube bearing (further details are provided in Subsection 6/4)
- The calculation contains details needed for clearance verification and its comparison with the bore sighting measurements.
- A sag and gap calculation or procedure on the propeller-shaft connection with the intermediate shaft is not required. Should this be included in calculation, the review engineer should include a review comment, and the surveyor should verify that the sag and gap procedure does not foresee any information required to conduct shaft alignment corrections.

Alternative approaches may be considered where required. For example:

- If the propeller shaft and intermediate shaft assembly results in an excessive load on the intermediate bearing or in excessive shaft stresses, the clearance calculation is to be conducted by adding an additional temporary support on the forward end of the intermediate shaft.

3.6 Shaft Alignment Procedure

A shaft alignment procedure should be submitted for review. Alignment verification is based on the submitted shaft alignment calculations. As a minimum, the shaft alignment procedure includes:

- Bore sighting before and after installation of stern tube bearings
- Stern tube bearing fitting pressure verification
- Tail shaft bearing clearance measurement
- Sag and gap measurements
- Bearing load measurements

Installations with no forward stern tube bearing require a specific shaft alignment procedure that addresses the issue of maintaining control over tail shaft misalignment, i.e. the bore sighting and aft stern tube bearing clearance measurement.

3.7 Documenting the Review

The engineer conducting the review of the submitted shaft alignment calculations is to document the results of the alignment review in a format which includes, but is not limited to, the following contents:

Descriptions of the propulsion system, including details of:

- Shafting, reduction gear, equivalent crankshaft model and propeller
- Masses of all attached equipment such as flywheel, turning gear wheel, shaft generator, and gears
- Location of the permanent bearings and temporary bearings
- Location of the jacking points for bearing reaction measurements
- Bearing offset data for hot and cold conditions
- External static loads

The analyses completed by the designer should consider:

- Hot and cold condition
- At least one afloat condition
- Sag and Gap

Where deemed necessary, the engineer may apply the optimization routine to investigate the propulsion installation sensitivity to disturbances originating from hull deflection, thermal deviations and bearing offset adjustments and consider this when evaluating the submitted calculations and prescribed displacements.

The engineer comments on observed differences between the submittal and the check analyses. Plan review comments about significant differences which may affect propulsion installation and ship safety should be stated in the review letter.

4 Design

4.1 General

Particular attention is to be given to the following when designing the shaft alignment:

- Equivalent model of the actual propulsion system
- Application of static and dynamic loads
- Slope boring design
- Consequence of the intermediate shaft bearing offset adjustment
- Alignment design with no forward stern tube bearing
- Main engine bedplate pre-sag application
- Sag and Gap procedure
- Bearing clearance
- Bearing elasticity
- Bearing material
- Gear meshes misalignment

Conditions to be satisfied are primarily:

- Bearing reactions
- Bearing load distribution
- Crankshaft deflections
- Gear meshes misalignment angle

4.2 Aft Stern Tube Bearing

The aft stern tube bearing is undoubtedly the most critical part of the propulsion system installation. Heavy weight and the dynamic loads from the overhanging propeller are primary reasons for aft stern tube bearing problems.

4.2.1 Design

The aft stern tube bearing design is significantly different from other bearings in the propulsion shafting system. It is considerably longer and with a regularly applied slope or inclination.

As mentioned earlier, the conventional aft stern tube bearing design is based on the static loads. The parameter of primary concern is the slope boring angle, which is defined with the aim to even the load distribution over the bearing length.

Conventionally, a single slope is applied, however there is strong evidence that further improvement in bearing performance and consequently a reduction of bearing damage can be achieved with a double slope design.

Moreover, a single or a double slope bearing design can be further optimized by conducting the fluid-structural-interaction (FSI) analysis for a dynamic turning condition.

Applying a bearing slope has a twofold benefit:

- It provides a smooth transition from the static to the running condition and provides the conditions for the quick creation of an oil wedge, minimizing the time the bearing is exposed to direct metal to metal contact with the shaft.
- It maintains sufficient oil film thickness during the maneuvering of the vessel when propeller dynamic downward forces are at their largest.

4.2.2 What is Preferred?

For aft stern tube bearings, as for the other journal bearings in the system, the contact pressure in the static condition may exceed the bearing load limit permitted by bearing manufacturer.

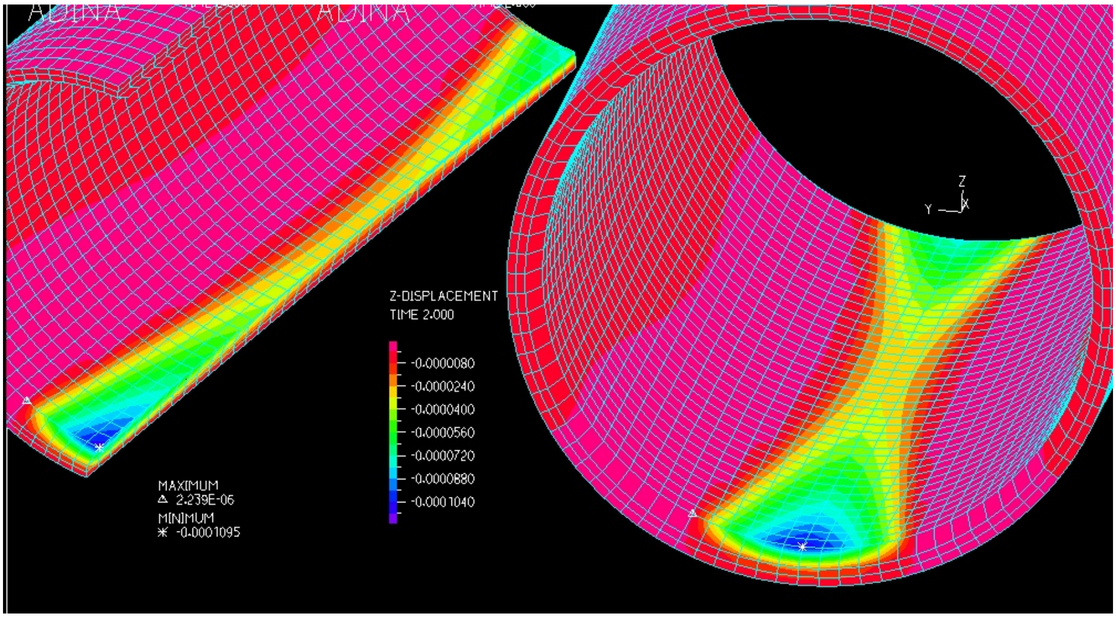
The actual static contact pressure, although exerted only by gravity forces, is in most cases much higher than the pressure exerted in a running condition, where lubricant distributes the load over much larger area in the bearing.

It is preferred to maintain minimum contact pressure, and this is best accomplished by a low misalignment angle between the shaft and the bearing. The ideal condition is to have no angular misalignment. This is achieved when the shaft is sitting on both the aft and the forward bearing edges, thus maintaining maximum contact area and minimum contact pressure with the bearing shell, as illustrated in Section 2, Figure 11.

Under ideal circumstances, the contact area is symmetrically distributed between the forward and aft edge of the bearing. A low misalignment angle provides a larger static contact area between the shaft and the bearing, smaller contact stresses exerted on the bearing by the shaft, and a faster oil film development, all resulting in an extended bearing life.

The above approach is somewhat different when double slope design is applied (more details are provided 2/4.4).

FIGURE 11
Desired Contact Between the Shaft and Bearing
(No Misalignment)



4.2.3 Class Requirements

The maximum acceptable misalignment between the shaft and the bearing is 0.3×10^{-3} *rad* and is widely accepted and applied as a marine industry practice. When this limit is exceeded in the design, a reduction of the misalignment angle is to be considered either by slope boring or bearing inclination. The safety margin of this limit is not fully explained yet, and this criteria should not be blindly applied. The issue is complex, as the misalignment angle directly influences bearing hydrodynamics. A dynamic analysis with fluid to structure interaction is required to determine the acceptability of the said limit.

ABS Rules also require a bearing length of minimum 1.5 times the shaft diameter for oil-lubricated bearings, and minimum 4 times the shaft diameter for water-lubricated bearings.

4.2.4 Design Strategies

The shaft alignment modeling process consists of:

- Finding an adequate bearing offset to suit all operating conditions
- Defining the location of the stern tube bearing contact point
- Defining the bearing contact area and load
- Accounting for disturbances from hull girder deflection and the thermal expansion of the structure

The contact area between shaft and bearing is directly correlated to the approach taken in designing shafting alignment. The three alignment solutions indicated below are analyzed on a diesel engine driven VLCC installation to investigate their impact on the load distribution throughout the stern tube bearing:

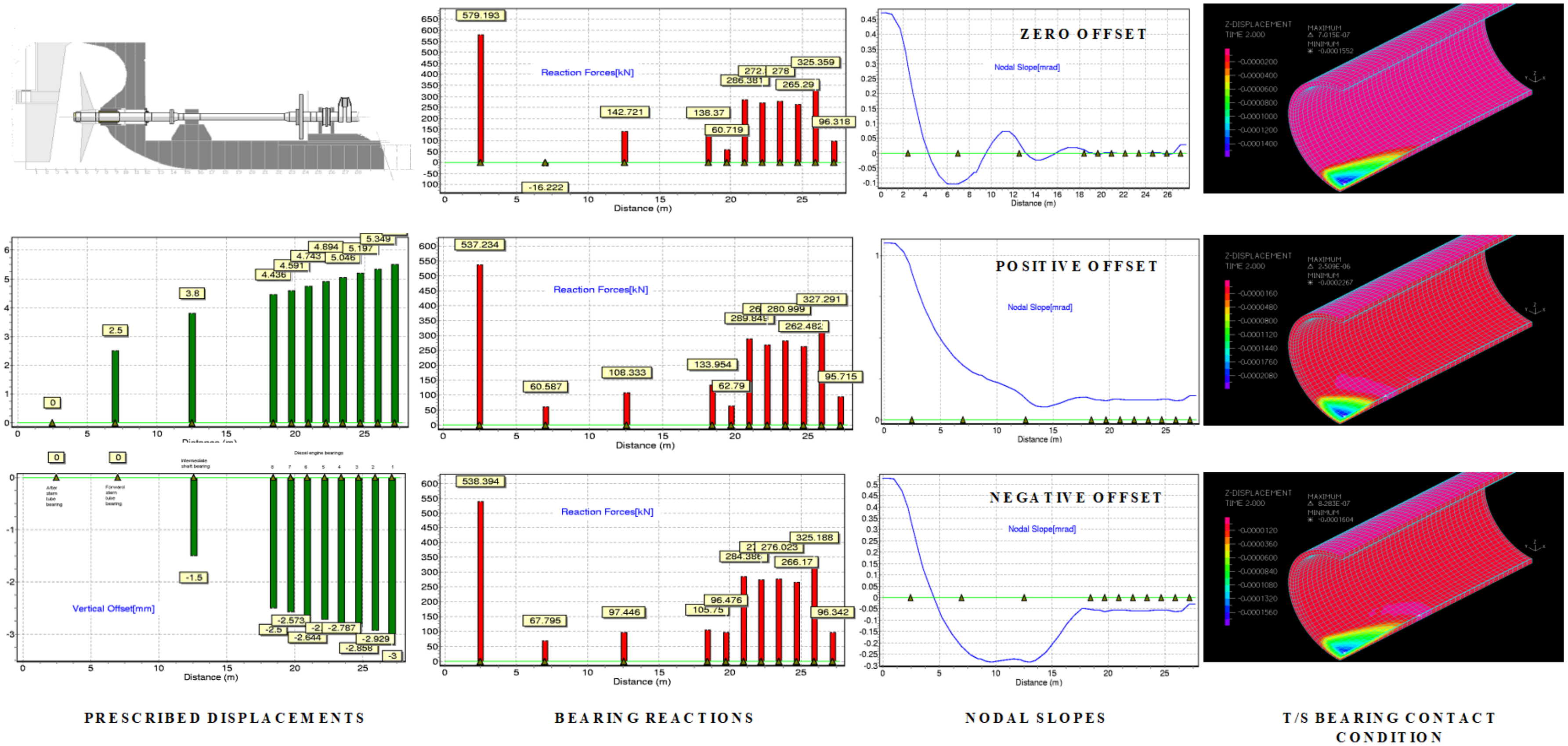
- Zero offset alignment
- Positive offset alignment
- Negative offset alignment

Zero Offset Alignment: This refers to the position of the bearings which corresponds to ideally-straight shafting, the centerline of which is horizontal and undeformed. The results of the alignment analysis conducted with zero-offset bearings are often not satisfactory. In the example shown below (Section 2, Figure 12 – row 1), the forward stern tube bearing becomes fully unloaded, however, the load distribution along the aft stern tube bearing appears very good, with a relatively large contact area between shaft and bearing.

Positive Offset Alignment: This is characterized by a shafting design where most of the bearings in the system are located *above* the after stern tube bearing position. The bearing offset is selected to satisfy the alignment criteria. In this example (Section 2, Figure 12 – row 2), the bearing reactions are all acceptable, however, the contact area between the bearing and the shaft is not optimal. The relative misalignment slope between shaft and bearing is estimated to be *0.855 mrad*, which exceeds the required limit set by ABS Rules.

Negative Offset Alignment: This is characterized by the shafting design where most of the bearings in the system are located *below* the after stern tube bearing position. The bearing offset is selected to satisfy the alignment criteria. In this example (Section 2, Figure 12 – row 3), the bearing reactions are all acceptable and the contact area is relatively large (approximately twice as large as in the Positive Offset approach). The relative misalignment slope between shaft and bearing is estimated to be *0.213 mrad* which is within the required limit set by ABS Rules.

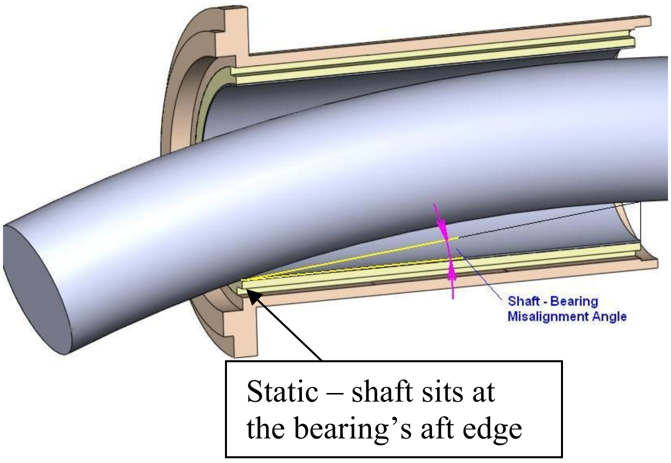
FIGURE 12
Aft Stern Tube Bearing Contact as a Function of Alignment Design



4.3 Aft Stern Tube Bearing – Single Slope Design

The slope boring angle calculation is to be based on the static afloat conditions with hot engine, and where plausible, optimized for dynamic turning condition utilizing FSI approach. Transition from static to dynamic is desired to be as quick as possible and slope boring assists this process. Minimizing angular misalignment provides a better bearing performance and a longer bearing life.

FIGURE 13
Misalignment Angle



ABS's proposed approach to slope boring of the aft stern tube bearing is summed up below.

- In general the slope boring is to be defined based on the static loads only. The pseudo-dynamic analysis, where dynamic loads are utilized in a static calculation is not deemed a suitable approach for slope boring design.
- The dynamic loads are acceptable in slope design evaluation and optimization when a full-scale dynamic analysis is conducted using the fluid-structure interaction (FSI) software.
- The propeller dynamic loads applied for the aft stern tube bearing optimization should be for the worst service condition that the bearing is exposed to, which is with the vessel turning hard-over at a full speed.
- Slope boring is to satisfy an entire range of vessel loading conditions, between ballast and fully laden. This is particularly relevant in installations with no forward stern tube bearing, where effect of hull deflections on the aft stern tube bearing misalignment is significant.

Further clarifications are offered as follows:

- A key objective of bearing design is to provide an early hydrodynamic lift of the shaft on the lubricant. This can be achieved by minimizing the misalignment angle between shaft and bearing for the static condition, or for low revolutions, before an oil wedge develops.
- The basic slope design is based on the static loads only and slope boring is commonly applied only on the bottom part of the bearing. The static design is in most cases proven satisfactory for dynamic service conditions also.
- Since the most critical dynamic condition for the bearing is during hard-over turns when propeller dynamic loads exert a heaviest downward push on the bearing, the static slope design can be adjusted and optimized by considering these critical dynamic loads utilizing the FSI approach.
- The basic slope design issue can be easily addressed with the ABS Stern Tube Contact Analysis interface in the ABS Shaft Alignment software utilized for modeling and evaluating contact between shaft and aft stern tube bearing.

4.3.1 Suggested Procedure for Contact Evaluation

When the ABS Shaft Alignment program is applied to single slope bearing designs, the following approach is suggested for the aft stern tube bearing contact evaluation:

i) Two contact points are to be created in the shaft alignment model, at a minimum - one each at the aft and forward edge of the bearing

ii) Define bearing offsets and calculate the aft stern tube-bearing slope in accordance with ABS Rules alignment requirements

iii) Repeat the analysis with a single point contact by removing the forward contact point on the aft stern tube bearing

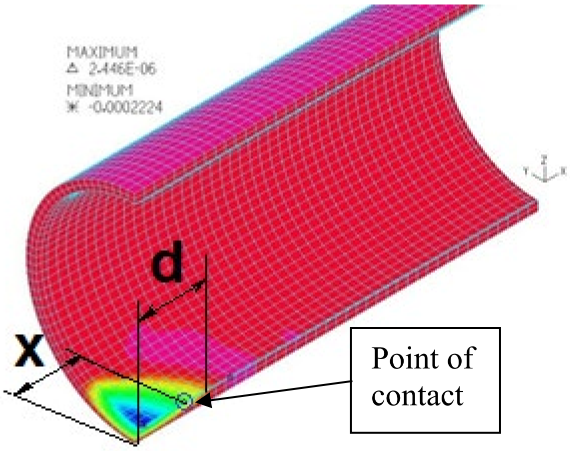
iv) Apply the same bearing offsets used earlier for the two-point analysis

v) Evaluate bearing contact

vi) Adjust the longitudinal distance of the aft point and recalculate.

vii) Repeat v) and vi) until satisfactory results are obtained.

The slope boring angle is now defined for applied contact point corrections. The results of this procedure for defining more accurate static point of contact between the shaft and the bearing may be considered satisfactory when the longitudinal coordinate *x* of the selected single point of contact is equal or slightly smaller than calculated distance *d*, as shown on the bearing model below.



4.3.2 Single Point Contact vs. Multi Point Contact

In conventional shaft alignment analysis, the contact between shaft and bearing is conveniently modeled with a single point. The contact point represents the position of the assumed bearing reaction. Location of the contact point will define reaction intensity and, even more importantly, the misalignment slope between the shaft and the bearing. This approach is proven sufficiently accurate when the requirement for slope boring does not have to be applied.

When a two-point or a multiple-points contact approach is applied, the procedure is to first conduct the analysis with a single point contact to obtain the shaft bending curvature. Based on the shaft bending curve the zero misalignment bearing slope can be established. The same bearing slope is then utilized to define the offsets on selected points along the bearing length. The selected points are the aft and the forward bearing edge, and the assumed point of maximum pressure is selected between *D/3* and *L/4* from the aft bearing edge; where *D* is shaft diameter and *L* is bearing length.

Considering the number of contact points, neither the single nor the two-point contact approach is correct, because the bearing-to-shaft contact is actually established over an area of a bearing that the shaft penetrates into. ABS established practice is to initially assume either the combined approach, both the single and two-point contact, or the single point contact only.

Designers who use the ABS software and take advantage of the stern tube bearing evaluation interface should consider combining the two-point and the single-point approach. The software has the ability to evaluate the contact area and provide a relatively accurate prediction of the bearing loads for the static condition and before the oil film develops.

A designer using the ABS shaft alignment software:

- Takes the initial bearing contact at the aft edge of the bearing, between *D*/3 and *L/4* from the after edge as required by ABS Rules
- Evaluates the contact area and if necessary, corrects the initially estimated bearing location and
- Repeats the calculation until desired results are obtained

Correction of the initial prediction of the bearing contact is recommended when the initially applied contact point falls outside the calculated contact area. If the analysis indicates contact at both edges of the bearing, the initially applied point of contact should be moved in accordance with the findings. Several iterations may be needed in order to stabilize the results.

Alternatively, from the point of view of the load redistribution, the analysis may start from the premise of an ideal two-point contact alignment, by appropriate selection of the bearing offset. As a next step, a more realistic case can be evaluated by considering a single point contact. By removing one point of contact, load and slopes will distribute differently. Doing so provides a more realistic slope boring condition and load distribution along the bearing length.

Note that amending the bearing contact position to more accurately represent the actual contact area is particularly important when bearings are of composite material, rubber or wood. Although for these the area of contact is relatively large, white metal bearings usually have a relatively small static contact area. Therefore, if the contact point is set at no more than *D/3* from the aft edge, omitting the correction will have less influence on the overall load distribution results.

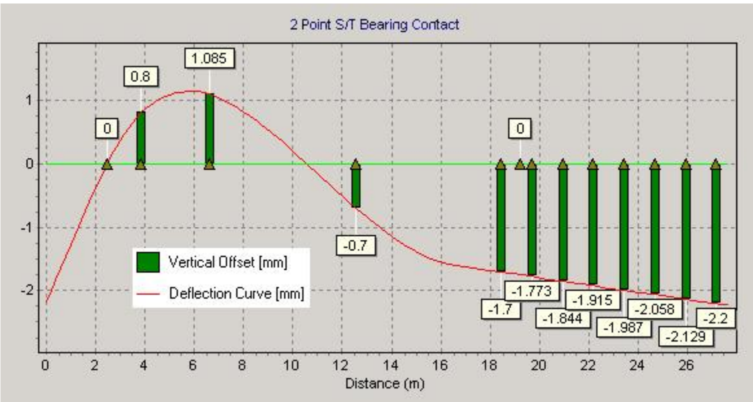
4.3.3 Analysis Example

Both solutions, one with a single and one with a two-point contact approach, were analyzed. The following procedures are suggested for each case.

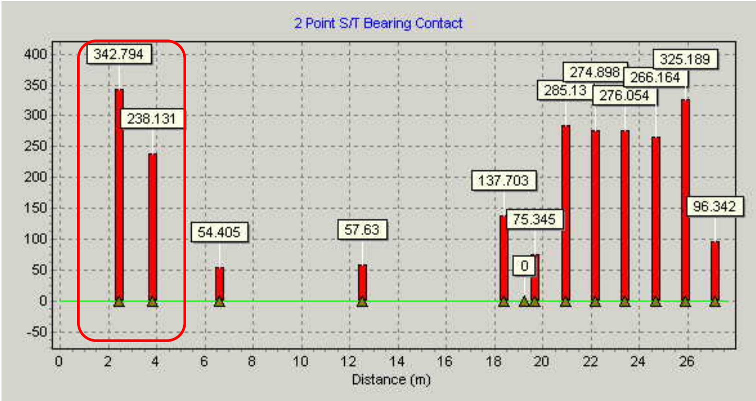
4.3.3(a) Two-point Contact

If the two-point contact approach is used, it is considered only as a transient case until the single-point contact is verified to be satisfactory as well. In a two-point contact analysis, the misalignment slope is zero.

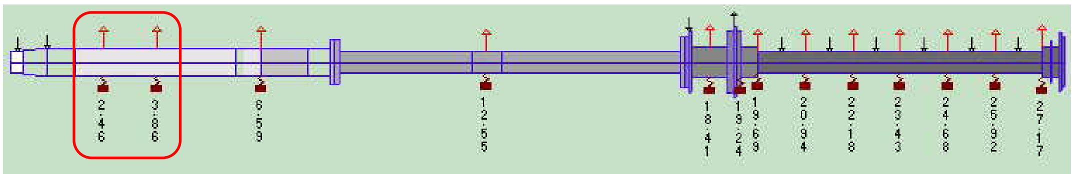
i) The bearing offset is selected to satisfy bearing reactions. It must first be confirmed that the selected offset at the bearings results in all positive bearing reactions as is required.

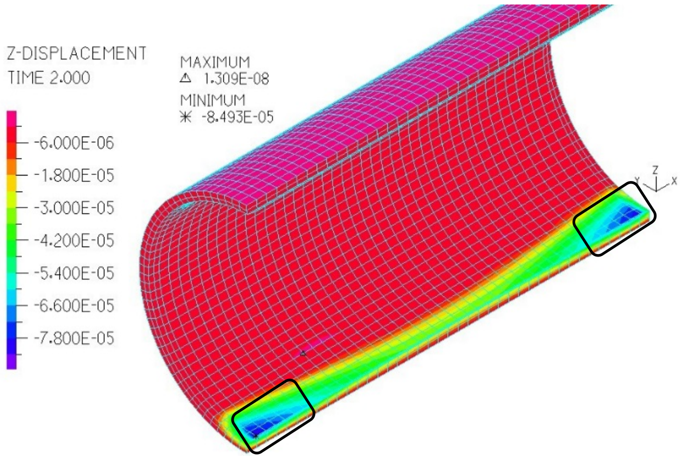


ii) Bearing reactions are verified to be satisfactory



iii) The contact points are selected at the aft and forward edges of the stern tube bearing. In this approach, it is not suggested to start with $D/3$ from the edges



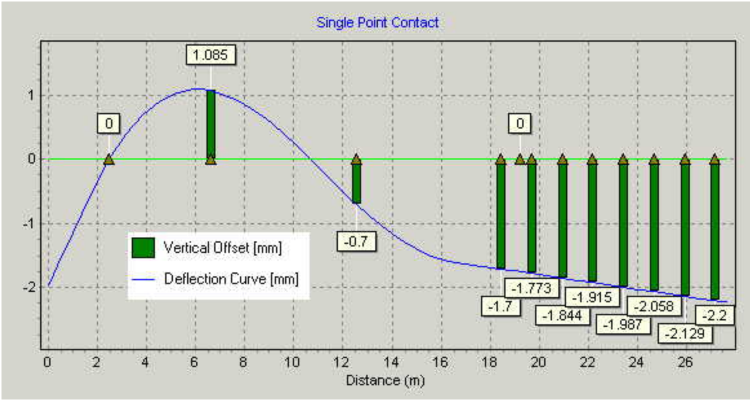


iv) The alignment analysis is repeated, by considering single contact points

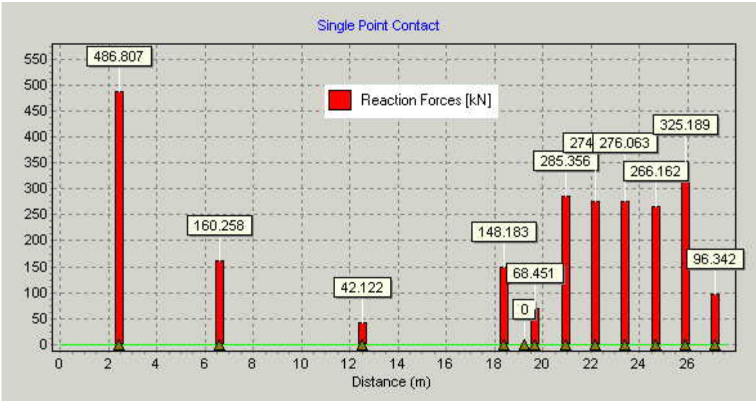
Provided that the foregoing resulted in acceptable solutions, the final step is to remove the front-edge contact point and investigate the single point contact only. If the single contact is not satisfactory, the analysis should not be accepted as valid.

4.3.3(b) Single Point Contact

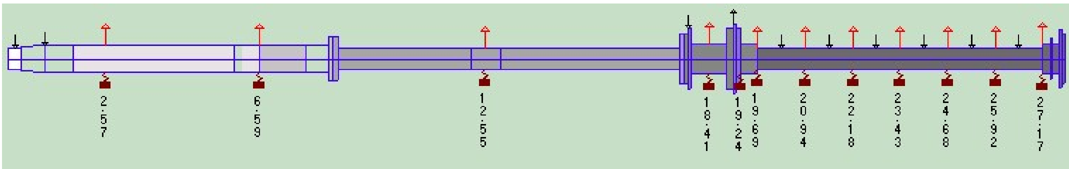
i) The bearing offset is selected to satisfy bearing reactions

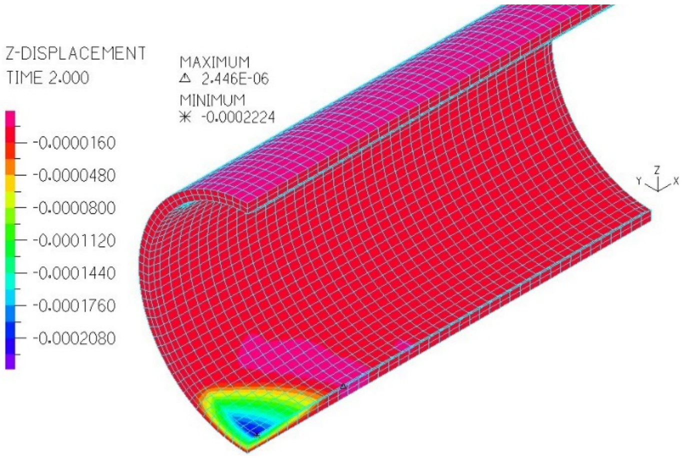


ii) Bearing reactions are verified to be satisfactory



iii) Initially, the contact point is selected at the aft edge of the stern tube bearing or, if preferred, at $D/3$ from after edge.

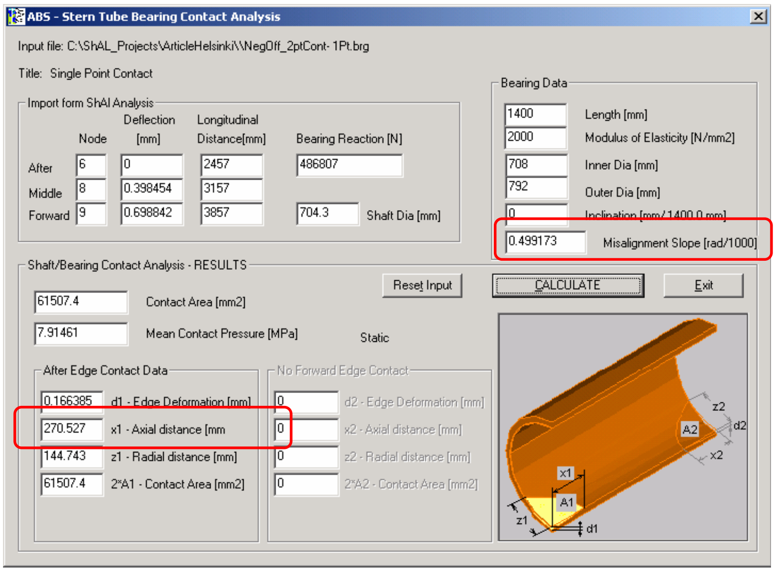




iv) The bearing misalignment angle and contact position are recorded and analyzed:

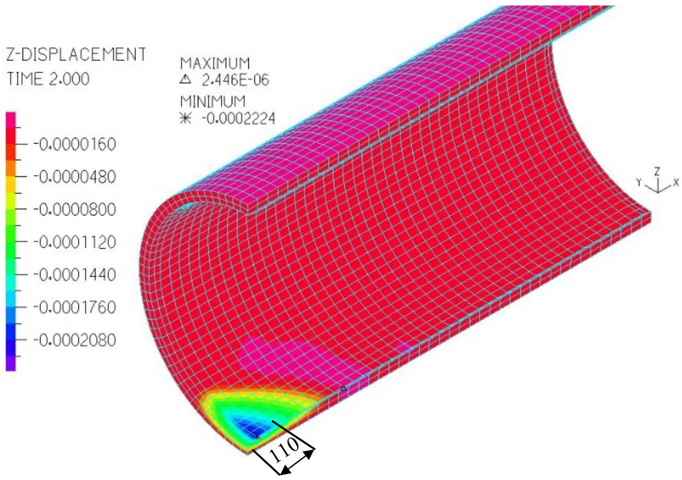
In this alignment analysis, the contact between bearing and shaft is arbitrarily placed at the aft edge of the bearing.

However, *x1-axial* distance [mm] in the screen shot below, taken from the bearing contact evaluation routine, shows that the contact area is established up to *270 mm* from the bearing's aft edge.

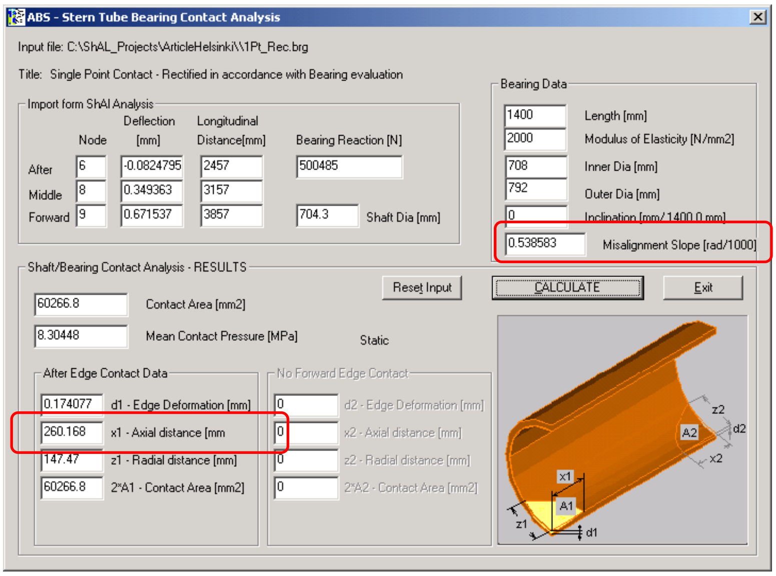


Therefore, the initially assumed contact point can be adjusted and moved forward.

v) The alignment analysis is repeated with the adjusted bearing contact point



The shaft alignment analysis is now conducted with an adjusted bearing contact point location, which is moved *110 mm* forward from the aft edge. This distance is arbitrarily selected to be approximately half the distance of the contact area. The shaft alignment model needs to be amended, and a node is generated as selected.



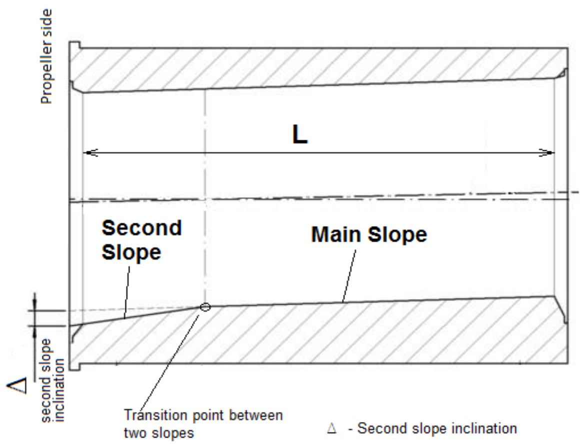
Results are as expected: per analysis with contact point placed at after bearing edge, the misalignment angle increased from *0.499 mrad* to *0.538 mrad*, thus reducing contact length (x1-axial distance [mm]) to *260 mm* from the bearing's aft edge.

The conclusion from the above analysis is that the contact point between the shaft and the stern tube bearing plays a significant role in defining the misalignment angle and proposing the slope boring condition. The contact area should be analyzed this way for several conditions of the vessel until an "optimal" slope boring angle is achieved.

4.4 Aft Stern Tube Bearing – Double Slope Design

Typical double slope design is given in Section 2, Figure 14. Slope angles are designed to result in the heaviest reaction load acting at the transition between the two slope angles ("knuckle point") and an approximately zero load at the aft and forward edge of the bearing. Where possible, the double slope design is to be optimized for running conditions by considering dynamic propeller loads. The optimization parameters include the location of the transition point as well as the two slope angles.

FIGURE 14
Double Slope



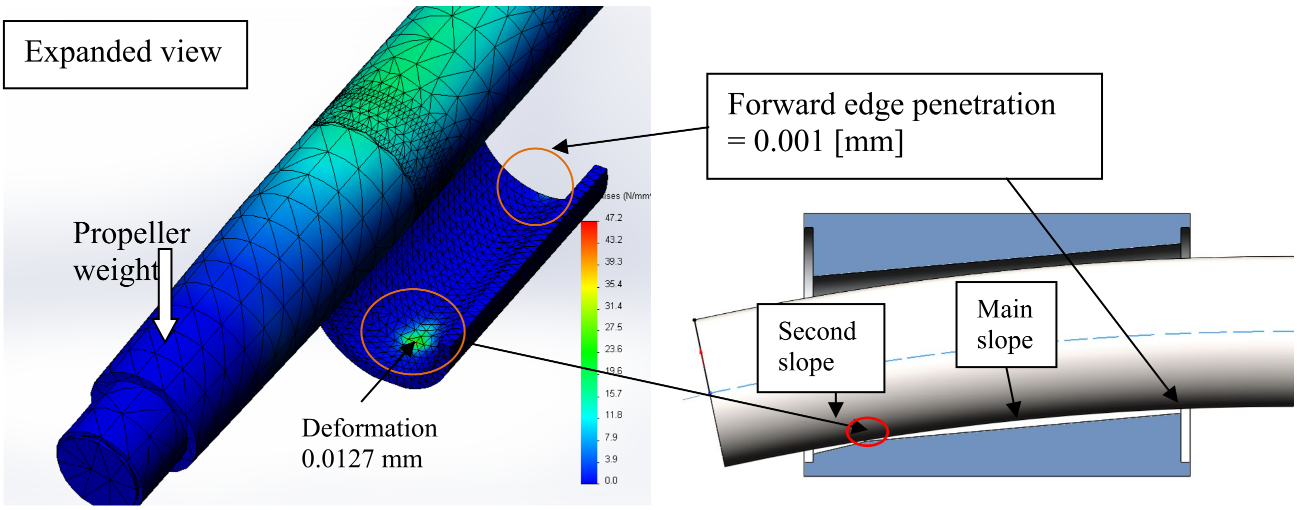
ABS approach is to position the transition point between *D/3* and *L/4* distance from the aft bearing edge. Double slope designs that deviate from these requirements are given consideration, provided the supporting analysis is based on the full-scale fluid dynamic analysis (FSI) with dynamic propeller loads defined for the most critical service condition, namely for the vessel turning at maximum service speed and maximum rudder angle.

The ABS approach to the double slope design considers the propeller shaft siting hard on the transition point in the static condition. With such a design, the bedding-in process between the shaft and the bearing is largely localized to the area around the transition point. During the bedding-in process, the area of contact between the shaft and the bearing expands in the forward and the aft direction of the slope transition point, thus improving bearing performance as the contact area enlarges and the contact pressure drops.

In the example of the double slope aft stern tube bearing design shown in Section 2, Figure 15, shaft bearing contact is evaluated using a 3-D solid model of the propulsion system.

Section 2, Figure 15 on the left shows the cross-section of a double slope aft stern tube bearing design. The main slope is applied throughout the bearing, and the second slope is usually applied only at the aft end on the bearing bottom.

FIGURE 15
Example: Double Slope Design



The finite element (*FE*) calculation shows that hard contact between shaft and the bearing is established at the transition between two slopes. The shaft plastically deforms the bush by *0.0127 [mm]*.

As desired, the shaft is in very light contact with the forward edge bush. In this example the deformation of the bush is *0.001 [mm]*.

Shaft is almost touching the aft edge but is not in contact.

For installations with no forward stern tube bearing, ABS Rules explicitly require a double slope design solely, and endorses the following design approach for a double slope.

- The transition ("knuckle point") between two slopes is selected between *D/3* and *L/4* distance from the aft bearing edge.

  D = actual shaft diameter

  L = length of aft stern tube bearing

- The static bearing slopes are defined to minimize, or just barely avoid aft and forward edge contact between the shaft and the bearing. The contact load at the bearing edges is kept at or close to zero, resulting in a very small or zero misalignment angle between the shaft and both bearing slopes.

- This design strategy forces the shaft to sit hard at the transition point in the static condition, which gives better control over the location of the acting load, thus enabling a higher accuracy of the shaft alignment calculation.

Notwithstanding ABS requirements, a double slope is preferred to a single slope design, because of several benefits:

- With the shaft sitting hard at the transition point and with very light or initially no contact between the shaft and bearing edges, two oil wedges are created when the shaft starts rotating: one forward and one aft of the transition point. This minimizes metal-to-metal contact and improves shaft lift.
- The bedding-in process causes wear and plastic deformation of the bearing around the area of maximum pressure, which in this case is the slope transition area. This is desired as long as wear and plastic deformation remain limited and ceases when the contact area enlarges sufficiently to significantly reduce the contact pressure.
- The double slope design is less sensitive to stern tube bore machining and production errors during bearing installation.

4.5 Aft Stern Tube Bearing Dynamic Analysis using FSI

Conventionally, the single slope is applied throughout the bearing, and in the double bearing slope design the second slope is applied only on the aft bottom of the bearing. The slope design is based on the static action only, which in most cases is a satisfactory approach for dynamic service as well.

A dynamic analysis performed for the straight runs only is not sufficient for aft stern tube bearing performance evaluation because this operational condition is not critical for the bearing.

Since the most critical dynamic condition for the bearing is during hard turning at full speed, when propeller dynamic loads push the shaft down on the bearing, the static slope design can be improved and optimized by accounting for these critical dynamic loads. Fluid-structure-interaction (FSI) software is to be utilized in dynamic optimization of bearing slopes.

Propulsion installations which are potentially sensitive to alignment problems, such as large twin screw and large single screw vessels, or installations with no forward stern tube bearing, may particularly benefit from design optimization utilizing FSI.

FSI is a demanding analysis because it requires the dynamic propeller loads to be known. Dynamic propeller loads can be determined through a computational fluid dynamics (CFD) analysis, by simulating the transient turning condition of the vessel. The CFD analysis adds to the overall complexity of a dynamic bearing analysis, because of the time and resources spent to obtain required inputs for FSI.

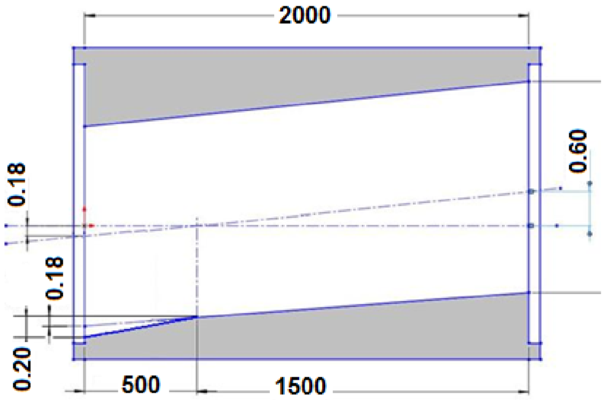
ABS proprietary software for FSI analysis of the aft stern tube bearing, takes into account bearing geometry (single or double slope, location of longitudinal oil grooves, clearance), dynamic propeller loads, static loading on shafting and bearing offsets.

The engineer conducting FSI analysis needs to be aware of limitations and assumptions applied when particular FSI software is used. Namely, if no bearing elasticity and deformation is accounted for in FSI calculation the results are more conservative and may indicate shaft contact with the bearing, where in practice this may not be the case.

The FSI calculation is normally performed for the full ballast and the laden condition, for which a different set of input parameters need be defined. The parameters utilized in FSI analysis are propeller loads, vessel speed and shaft RPM. Those time varying parameters change as the ship progresses from a straight-ahead course into the turn.

4.5.1 Example

An example with performance evaluation of a double sloped stern tube bearing on the single screw installation is shown below.



The initial double slope design consists of

- main slope of *0.6/1500 mm/mm*
- second slope of *0.2/500 mm/mm*

To briefly show how the FSI process is conducted, the three basic cases are extracted here from the several analyses conducted during the bearing optimization process:

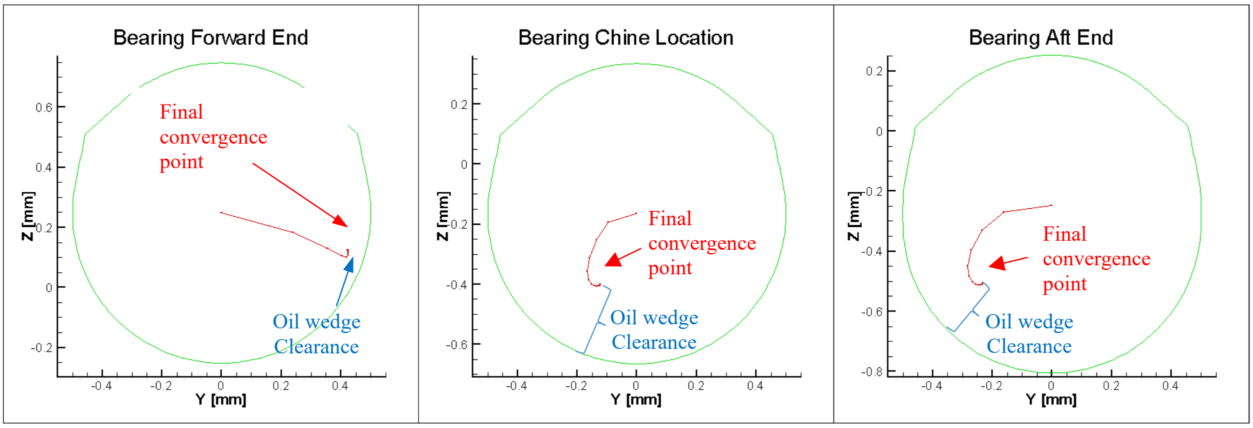
- The first case (Section 2, Figure 16) is for the straight ahead run at MCR with a zero rudder angle.
- The second case (Section 2, Figure 17) represents a simulated starboard turn with maximum rudder angle and MRC condition at the beginning of the turn. This case appears most critical for this particular installation because of the dynamic down forces from the propeller.
- The third case (Section 2, Figure 18) is for a simulated port turn with maximum rudder angle and MRC condition at the beginning of the turn. Although in this case the shaft appears to be getting in contact with the forward top of the bearing, the condition is potentially less serious because gravity forces counteract the upward dynamic propeller loads.

4.5.1(a) Simulation Case 1 – Straight Ahead Run, Zero Rudder

The straight ahead run of the vessel at MCR condition and zero rudder angle before entering the turn is shown in Section 2, Figure 16. Position of the shaft in the bearing is indicated at three characteristic locations: the aft bearing edge, the transition point between two slopes (chine), and the forward bearing edge. For this condition the oil film thickens at the final converged bearing position plot indicates stable bearing performance with good clearance margin between the shaft and the bearing.

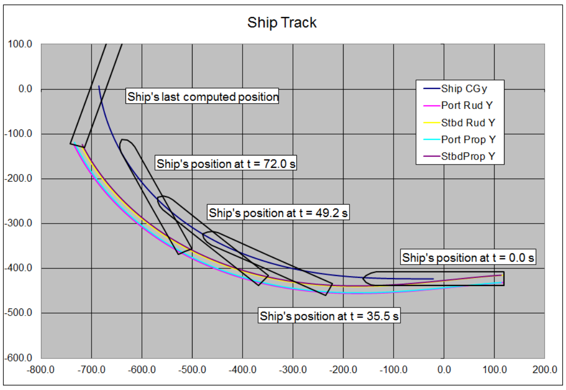
Propeller loads for the straight running condition are easily obtained with little effort and time required. This case is however not critical for the bearing performance. Accordingly, although the evaluation of the bearing performance for the straight MCR run is desired, Industry best practice is not to utilize the results of such a calculation in bearing design optimization. For design optimization, the bearing performance at turning condition should be investigated.

FIGURE 16
Ahead Run at MCR at Zero Rudder



4.5.1(b) Turning Simulation

On single screw vessels with a clockwise-rotating propeller, starboard turning is potentially more critical for the bearing performance than port turning.



The starboard turning circle simulation is shown on the chart on the left. The port turn would be the mirror image of the same chart.

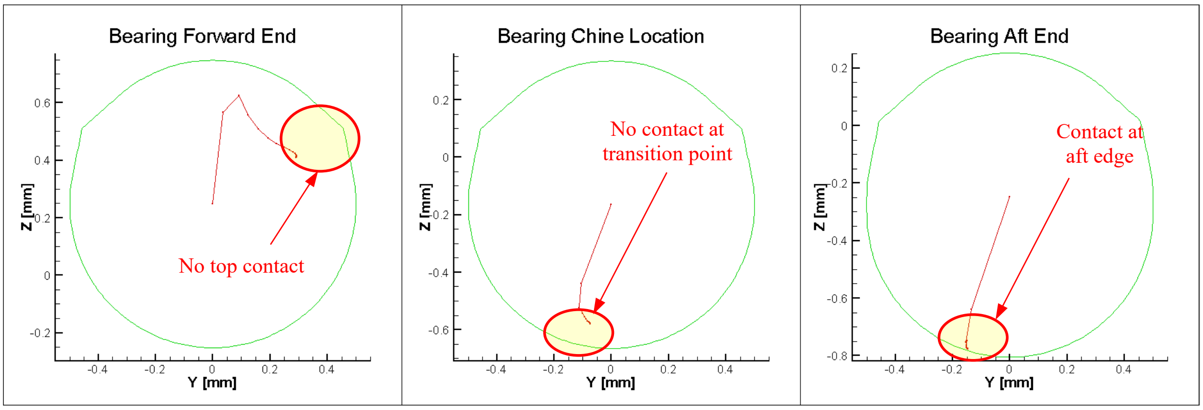
The chart indicates several assumed vessel positions for which propeller loads, vessel speed and shaft rpm are estimated.

The FSI calculation is then performed and the bearing condition is evaluated at each position in the turn.

In Section 2, Figure 17 and 18 the shaft position in the bearing is simulated at approximately *50 seconds* into the starboard and the port turn respectively.

4.5.1(c) Simulation Case 2 – Starboard Transient Turn

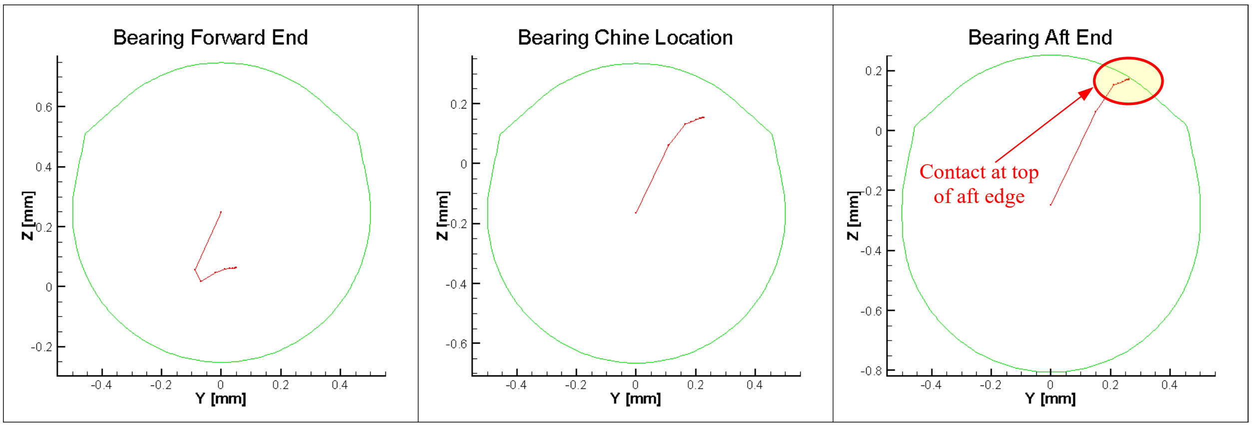
FIGURE 17
Starboard Transient Turn Simulation



The starboard turning high-speed test simulation (Section 2, Figure 17) shows possible metal-to-metal contact at the aft bearing's bottom edge. When contact persists for most of the turn, the redesign of the initial bearing slopes and new FSI analysis may be required.

4.5.1(d) Simulation Case 3 – Port Transient Turn

FIGURE 18
Port Transient Turn Simulation



The port turning the high-speed test simulation (Section 2, Figure 18) shows possible metal-to-metal contact at the bearing's aft top edge. Although top contact is not as critical as when the shaft is in contact with the bottom of the bearing, if contact persists for most of the turn, redesign of the initial bearing slopes and new FSI analysis may be required. One solution to mitigate top contact may be the application of the second slope throughout the bearing circumference, instead of only in the bottom bearing part.

4.6 No Forward Stern Tube Bearing Installations

ABS Rules require that for installations with no forward stern tube bearing, the aft stern tube bearing is to be of the double slope design. A single slope design may be specially considered where adequate technical documentation is submitted, to justify that a single slope design provides an equivalent satisfactory performance.

Advantages of installations with no forward stern tube bearing**:**

- The most significant benefit of installations with no forward stern tube bearing is possibility to verify and correct eventual misalignment in the aft stern tube bearing after the propeller shaft is fitted in the stern tube. The misalignment can be easily adjusted and practically eliminated by simple offset adjustment of the intermediate shaft bearing, while using the clearance at the aft edge of the stern tube bearing as a control parameter. An appropriate stern tube bearing clearance calculation without the propeller fitted should be prepared for these adjustments.
- The alignment production procedure can be simplified by bolting the propeller shaft and intermediate shaft immediately after the propeller shaft is fitted in the stern tube. Sag and gap measurements are thus unnecessary to be performed at the connection between the propeller shaft and intermediate shaft flanges.
- The system is more elastic and thus less susceptible to the adverse effects that hull deflections may have on the main engine bearings.

Disadvantages of installations with no forward stern tube bearing:

- The benefit of the direct correlation between the aft stern tube bearing misalignment and the intermediate bearing offset change can easily turn into its biggest disadvantage if proper alignment procedure is not followed. If the stern tube bearing clearance is not used to verify and correct the aft stern tube bearing misalignment, and if the intermediate bearing position is not precisely defined by sighting, control over stern tube bearing misalignment would not be possible.

- Treating it the same as a conventional system with both stern tube bearings and allowing the intermediate bearing offset adjustment to correct the bearing reactions will result directly in an uncontrolled change of the aft astern tube bearing misalignment.
- The forward stern tube bearing seal may be affected by shaft bending due to greater distance between the intermediate bearing and the stern tube, making the shaft seal more susceptible to eventual whirling vibrations and shaft run-out.
- The intermediate shaft bearing requires to have an upper and the lower bearing shell fitted.
- Additional effort is required to precisely define the position of the intermediate bearing by sighting, and to set it exactly back in its place after the propeller shaft is inserted in the stern tube.

From the point of view of shaft alignment design, it is beneficial to have a more elastic system with a larger span between bearings to minimize possible bearing unloading as the vessel loading condition changes. Commonly applied criteria for an acceptable span between bearings is $400\sqrt{d}$ or larger, with "*d*" being the shaft diameter. Shaft run-out and whirling margins, however, impose a limit on the maximum permissible span.

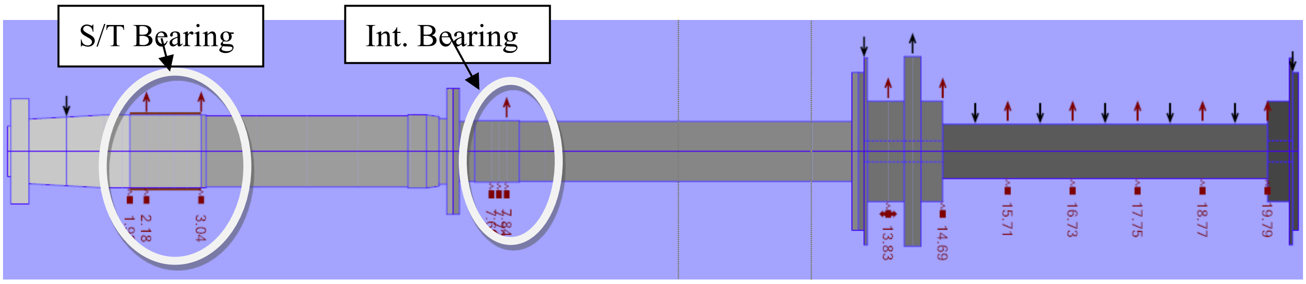
Considering this, it may be advantageous to design propulsion system with no forward stern tube bearing for ships were the distance between main engine and propeller is relatively short, as is common on tankers or bulker carriers. Removing the forward stern tube bearing is not without consequences, and it advisable to have the alignment procedure amended as explained above under "Advantages of installations with no forward stern tube bearing".

To minimize potential shaft alignment issues on installations with no forward stern tube bearing, the following requirements are to be implemented:

- Stern tube bearing to be of double slope design
- Lubricant inlet in the stern tube to be located aft of the stern tube bearing
- Intermediate bearing is to be chocked after bore sighting and clearance verification
- Run-in procedure to be conducted before commencing sea trials

For the equivalent shaft alignment model with no forward stern tube bearing, it is a good practice to apply a multiple support approach when modeling the aft stern tube bearing and intermediate bearing.

FIGURE 19
Equivalent Model – No Forward Stern Tube Bearing



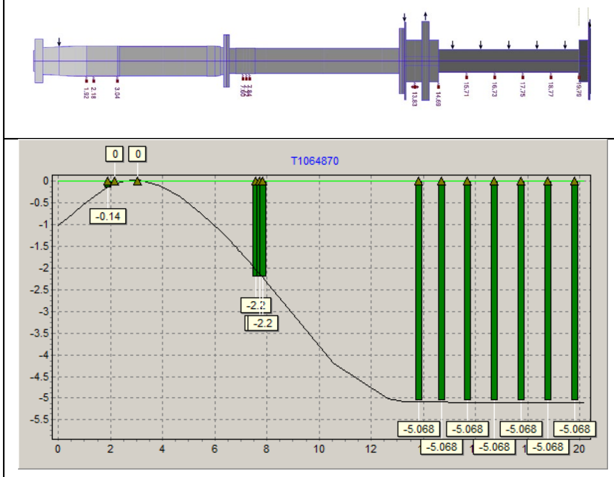
At least three points of support are needed for proper representation of the double-sloped bearing, one at each bearing edge and one at the transition point between two slopes as shown in Section 2, Figure 19.

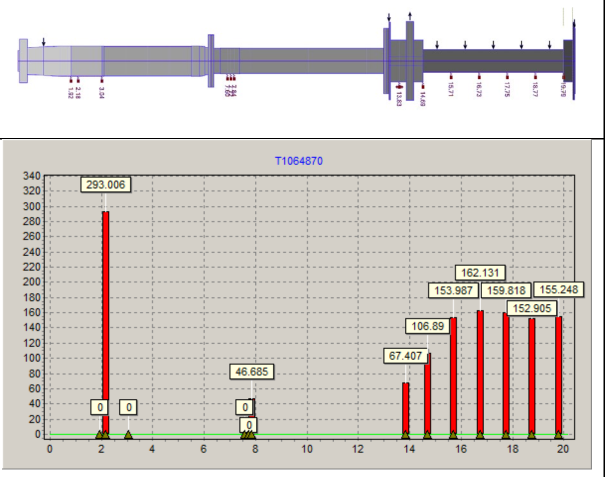
A three-point support is desired for the intermediate bearing to obtain a more accurate position of the reaction load on the bearing and a more precise misalignment evaluation at the aft stern tube bearing.

The following example shows the effect of hull deflections on bearing reactions. There is a substantial impact of the load redistribution within the aft stern tube bearing as the draft of the vessel changes. This, in contrast with the systems with both stern tube bearings, where impact of hull deflections on the aft stern tube bearing load change and misalignment is not significant, as explained earlier.

DRY-DOCK

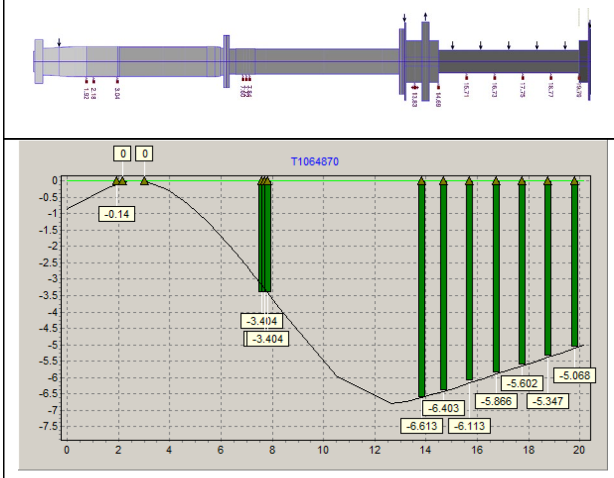
Offset | Reactions

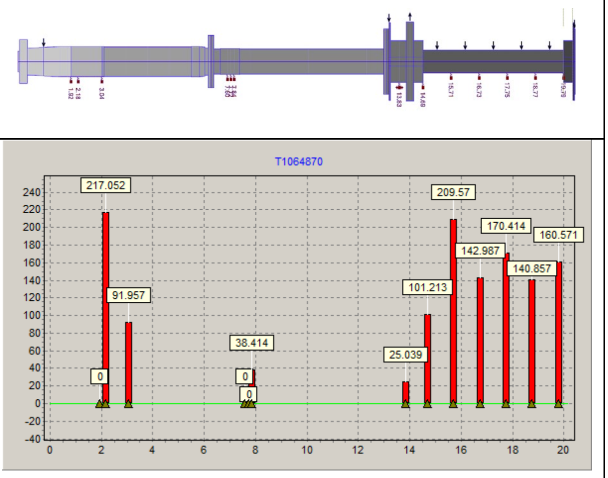




BALLAST

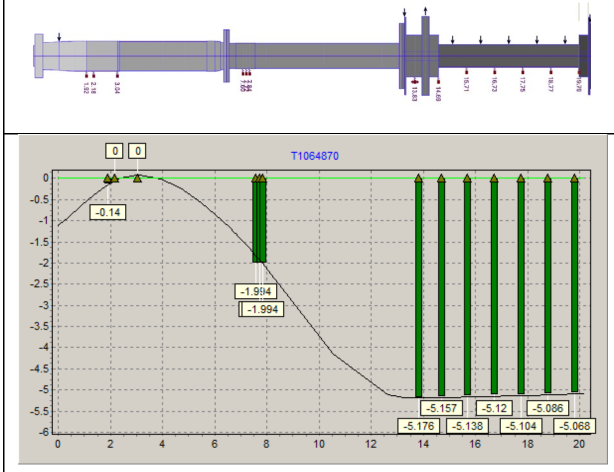
Offset | Reactions

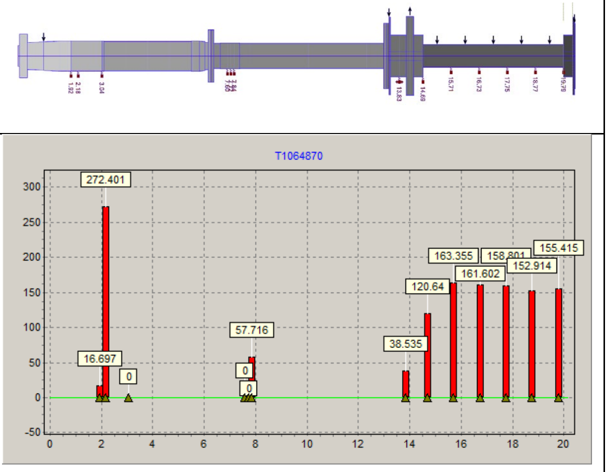




FULLY LADEN VESSEL

Offset | Reactions





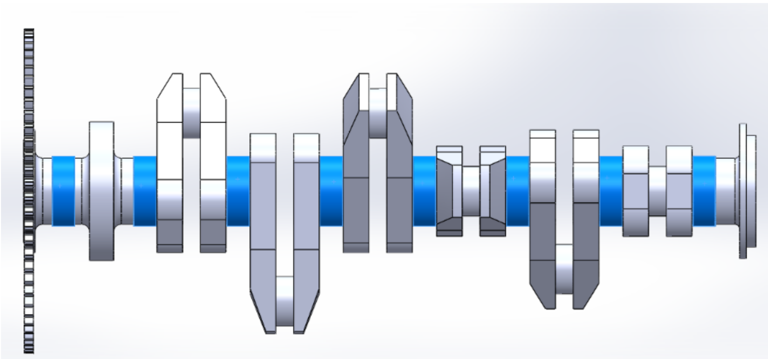
4.7 Crankshaft Modeling

Bearings of large two-stroke crosshead diesel engines are sensitive to the alignment condition, primarily because of the following:

- The proximity of the engine bearings to each other
- The flexibility of the engine foundation and the engine structure
- The flexibility of the crankshaft

Most commercial shaft alignment software follows basic beam theory and does not support complex shaft structures such as crankshafts. This is why the crankshaft utilized in the shaft alignment software is to be represented by an equivalent beam model. The engine maker often provides the equivalent crankshaft model.

FIGURE 20
Crankshaft FE Model



Where an equivalent crankshaft model, Section 2, Figures 21, is not available it can be generated by first creating a 3-D finite element (FE) model of one crank-throw and a comparable beam model which will result in the same bending deflection at the crank pin (Section 2, Figures 22 and 23).

FIGURE 21
Crankshaft – Equivalent Model for Shaft Alignment

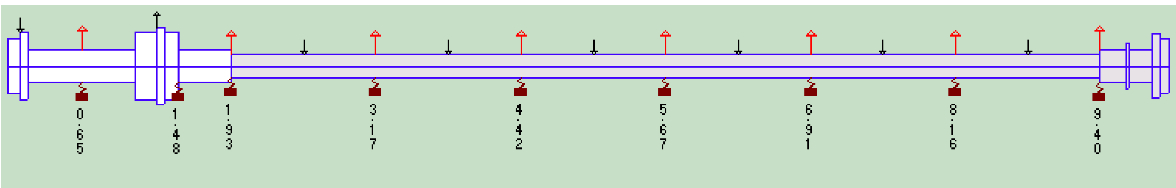
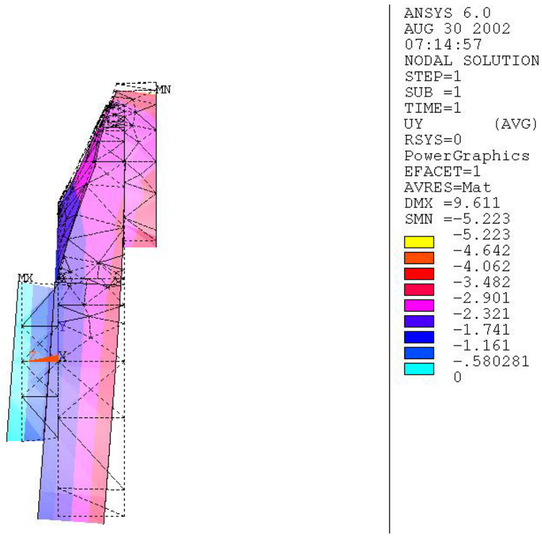
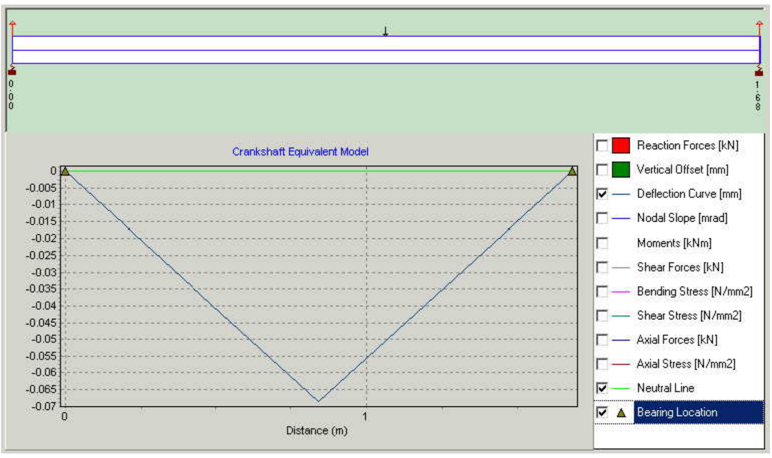


FIGURE 22
FE Model of Half of the Crank



Note: Figure 22 shows only half of the single crank 3-D FE model which will not provide an accurate equivalent model. Whole crankshaft modeling is necessary to establish interaction among cranks.

FIGURE 23
Defining Equivalent Model



Inaccuracy in the alignment calculation as a result of crankshaft geometry simplification is normally not very significant.

The larger impact on calculation accuracy is associated with the load variation on the crankpins as a result of the crankshaft angular position. This is important when comparing calculation results with measurements.

Normally, the assumption in beam-model crankshaft calculation is that loads at the crankpins are constant, as if all the pistons are in the top dead center (TDC) and no load variation with changing crankshaft angular position is considered. However, when measurements are conducted on the engine bearings, the crank-throws are turned *90º* to port and starboard and the piston cannot remain in the TDC. This results in a difference between measured and calculated reactions inside the engine. The following example explains the issue further.

4.7.1 Example:

When cylinder No. 1 is in the TDC, the bending deflections of the node in the middle of the crank journal will be different than for the 90º crank throw position and as a consequence, different bearing reactions for different crankshaft angular positions. The difference in crankshaft deflections and bearing reactions depends on the crankshaft design.

The variation of the bearing reactions due to the stiffness variation of the crankshaft as the crankshaft's angular position changes from *0º– 360º* is normally expected to be:

- Up to *10%* at the two aft most main engine main bearings
- Up to *20%* on the main bearings within the engine

Designers should be aware of those differences to verify a sufficient margin is foreseen in the alignment analysis to prevent bearing unloading or overloading.

4.8 Applying a Partial Equivalent Model of the Crankshaft

When a partial crankshaft model is applied, sometimes only two or three aft most crankshaft bearings are considered in the alignment analysis. Sometimes the crankshaft model is reduced to as few as two bearings, and in some cases, even the load from the crank mechanism, i.e. piston, piston rod, crosshead, etc., is completely neglected. With a stripped crankshaft model like this, the load in diesel engine bearings cannot be properly evaluated and the model can help only to minimize the reaction evaluation error on the foremost line shaft bearings.

Such simplification is not advisable. Experience shows the consequences of not evaluating the loading condition of at least the two aft most main engine main bearings using a crankshaft model with a minimum of four bearings: engine damage may eventually result along with the subsequent costly engagement by all involved parties to restore the damage. Accordingly:

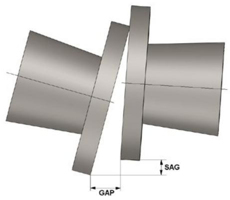
- The equivalent model cannot provide completely accurate information on main engine main bearing loading. The difference between calculation utilizing the equivalent crankshaft model and the actual measure reactions may be greater than 20%.
- For the direct-coupled main engine the alignment design must not result in the unloading of main engine main bearings. The three aft most main engine main bearings are particularly sensitive to unloading and are most affected by position adjustments at the intermediate shaft bearing.
- Industry best practice is not to apply an equivalent crankshaft model with less than four bearings, because the result of the bearing reactions, as well as the sag and gap data, will most likely be inaccurate.

4.8.1 Example

Partial crankshaft modeling (Section 2, Figures 24 and 25) may result in an incorrect prediction of the crankshaft aft-flange sagging. The calculation accuracy may be particularly affected when the model includes less than four bearings and when the loads acting on the flange and aft most crank-journals are not taken into account.

- Difference in estimated sag is *0.03 mm*
- Difference in estimated gap is *0.067 mm*

FIGURE 24
Reduced Crankshaft Model – 2 M/E Bearings Only



Reduced crankshaft model
Sag and gap @ intermediate shaft & M/E flange:
Sag = -1.261 mm
Gap = 1.239 mm
Aft most M/E bearing reaction: *173.7 [kN]*

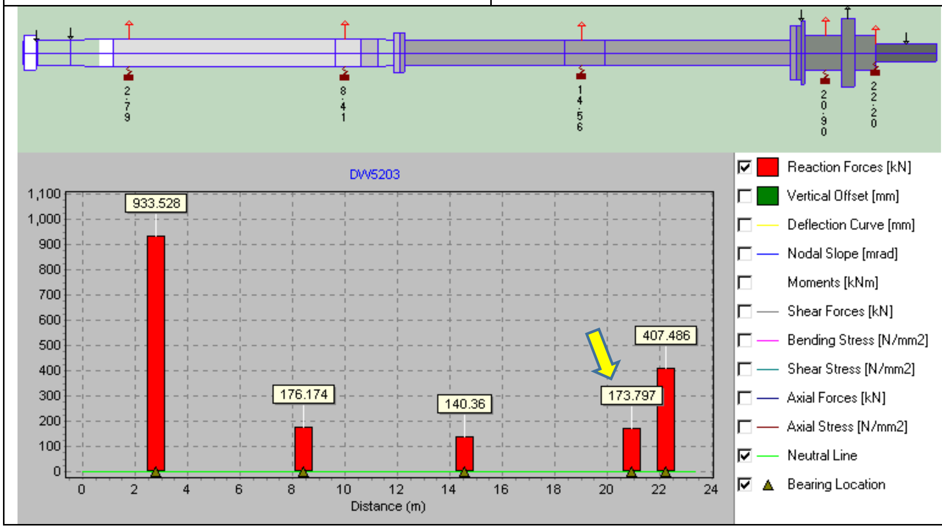
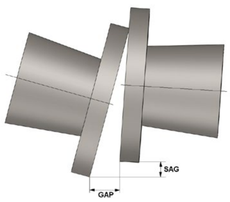
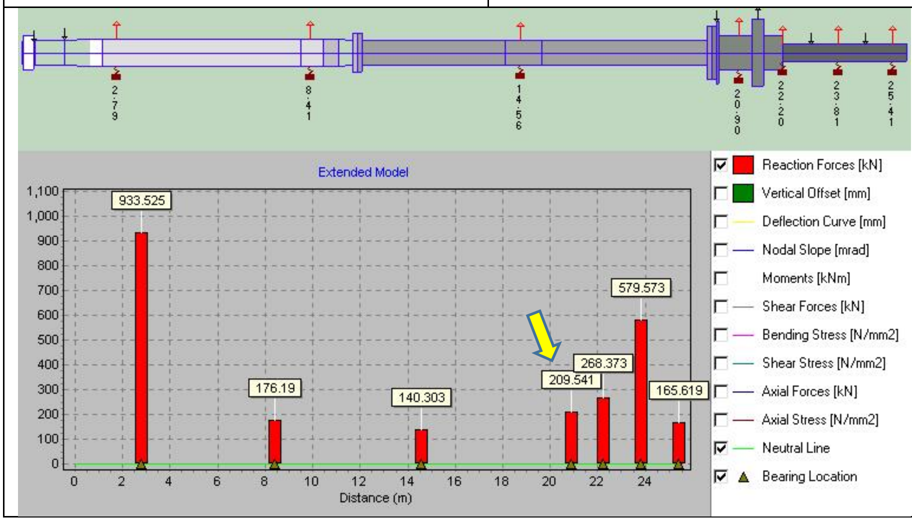


FIGURE 25
Reduced Crankshaft Model – 4 M/E Bearings



Extended model crankshaft
Sag and gap @ intermediate shaft & M/E flange:
Sag = -1.230 mm
Gap = 1.306 mm
Aft most M/E bearing reaction: *209.5 [kN]*



The line shaft bearings and the engine positioning using the sag and gap method are to be conducted prior to the shafting assembly. In this example, the actual sag and gap values are expected to be closer to the results of the second model with the extended crankshaft and it is interesting to consider what could happen to shaft alignment when only the first model is applied.

Should the personnel conducting the alignment be given the data from the first model, the actual alignment, which might well be correctly executed, will unnecessarily be ‘readjusted’ to coincide with the sag and gap values of this inadequate model. Since the sag and gap adjustment is normally conducted by changing the offset of temporary supports and thus also the intermediate shaft bearing offset, the consequence after the shaft assembly would be bearing reactions that do not match the analytical predictions for both the engine bearings and the line shaft bearings.

As mentioned earlier, for this reason ABS does not recommend sag and gap method as an accurate method for shaft alignment, and requires bearing reactions to be verified by the jack up measurements.

4.9 Engine Bearing Misalignment

The line shaft bearings, the forward stern tube bearing, and the main engine main bearings are normally not evaluated for angular misalignment with the shaft. The load carrying requirements are not as critical as for the aft stern tube bearing. Furthermore, the length of those bearings is two to three times shorter than the length of the aft stern tube bearing.

Moreover, with all the modeling simplifications of the main engine crankshaft in commercial shaft alignment software, it may be difficult to accurately predict the engine bearing misalignment. The possibility of a bearing unloading inside of the main engine, which signals a potential misalignment of adjacent engine bearings can nevertheless be evaluated.

An error of *10%* to *20%* is likewise expected in predicting a possible misalignment between the crankshaft journal and the main bearings.

The possible cause of main engine bearings edge loading is almost solely the result of disturbances originating from the shaft line. On rare occasions, incorrect engine pre-sag, bearing shell and housing machining errors may also cause the internal engine bearing misalignment.

4.10 Bearing Elasticity

Bearing stiffness is not simple to define because it is not solely linked to the bearing material properties and the load but is also a function of the contact area. The contact area changes with bearing load redistribution, which may result from bearing offset adjustment and external disturbances such as:

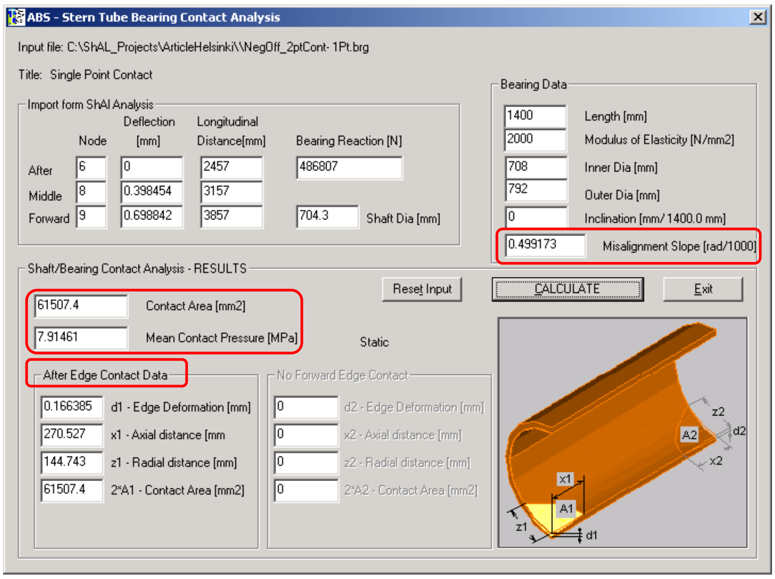
- Hull deflections, and
- Thermal effects on the bearing offset

Modifying the bearing offset affects the misalignment angle between bearing and the shaft. This consequently results in a changed contact area between shaft and bearing. The extent the shaft penetrates into the bearing material is proportional to the area of contact.

In analytical modeling it is necessary to find an equilibrium solution that balances the load at the bearing, the misalignment angle, the contact area and the shaft’s penetration into the bearing. To the designer’s discretion, because of the nonlinearity of this process, an iterative method can provide a simple way to find the equilibrium solution.

ABS Shaft Alignment software provides an interface to evaluate the bearing contact condition.

FIGURE 26
ABS Bearing Evaluation Interface



4.11 Bearing Wear Down

Bearing wear does not immediately affect the alignment condition but should be considered for a good and robust shaft alignment design. Alignment optimization is particularly helpful to account for the effects of bearing wear.

Bearing wear progresses with time and is greatly influenced by the misalignment angle between shaft and bearing. The misalignment angle determines the amount of static contact pressure exerted by the shaft on the bearing, and accordingly the shaft rotational speed (rpm) needed for the shaft to lift off the bearing as the main engine starts.

In general wear is not desired, but slow and controlled bearing wear can have a positive impact on bearing performance, specifically during the initial bedding-in process where the contact area enlarges by friction and wear. As the contact area increases, wear slows down and the contact pressure drops.

Excessive wear of the tail shaft bearing is of primary concern and is mostly the result of heavy edge loading by the overhung propeller. The propeller cantilever effect results in significant tail shaft bending. Aft stern tube bearings are often slope bored to counteract shaft bending.

It is difficult to verify the in-service condition of the tail shaft bearing, without installing costly monitoring systems. A commonly applied practical method is the verification of bearing wear by measuring the amount the tail shaft drops over time. Typically, stern tube bearing wear-down measurements are taken by using a poker gauge or wear-down gauge. This tool is commonly supplied by the shaft seal manufacturer.

4.12 Aft Stern Tube Bearing Clearance Calculation

Before the propeller is fitted, measurement of the aft stern tube bearing clearance can be utilized as an indirect method to verify misalignment between the aft stern tube bearing and the propeller shaft. The calculation of the aft stern tube bearing clearance, as well as the verification procedure, is different for propulsion installations with and without forward stern tube bearings.

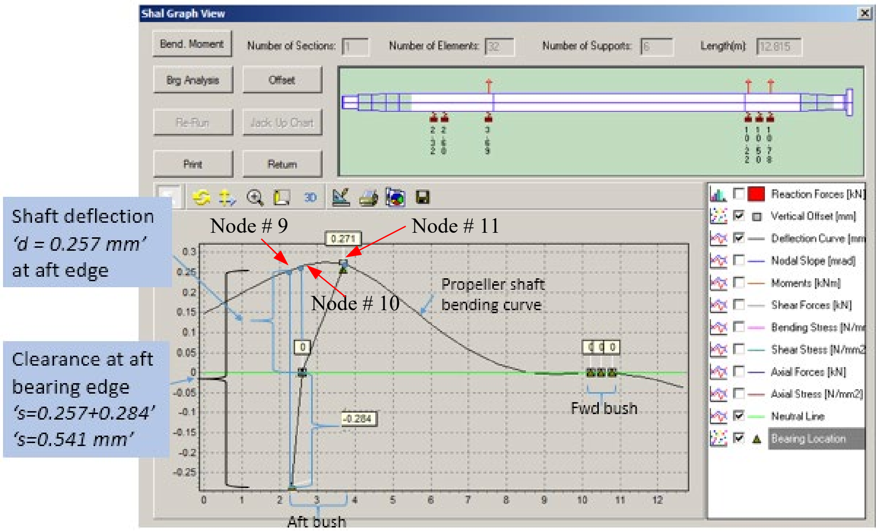
As explained earlier:

- Double sloped stern tube bearings are modeled with at least three multi-point supports, where one of the nodes is placed at the transition point between the two slopes, one node at the aft bearing edge and one at the forward bearing edge
- For the purpose of the bearing clearance calculation, the analytical model consists only of the propeller shaft sitting on the aft and the forward stern tube bearings
- No load from the propeller or any external mass at the aft end and no restraints at the forward end

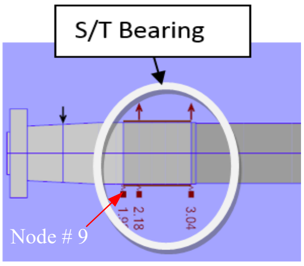
4.12.1 Installation with Forward and Aft Stern Tube Bearings

In conventional stern tube designs where both stern tube bearings are present, the bearing clearance is calculated at the aft edge of the aft stern tube bearing as shown in Section 2, Figure 27. After propeller shaft is placed in the stern tube, the calculated clearance is compared with the measured value. How tail shaft clearance is measured is explained in more detail in Subsection 3/4 of these Guidance Notes.

FIGURE 27
Installation with Forward and Aft Stern Tube Bearing



Example:



SYSTEM DISPLACEMENT VECTOR - IN GLOBAL COORDINATES

Node No	X coord. [m]	dx [mm]	dy [mm]	Rot(z) [mrad]	Restrained Direction (Support Clearance [mm])
1	0.000	0.0000	0.1468	0.0523	
2	0.025	0.0000	0.1481	0.0523	
3	0.395	0.0000	0.1674	0.0521	
4	0.415	0.0000	0.1684	0.0520	
5	0.764	0.0000	0.1865	0.0515	
6	1.113	0.0000	0.2043	0.0501	
7	1.439	0.0000	0.2203	0.0478	
8	1.765	0.0000	[illegible]	0.0442	
9	2.320	0.0000	0.2574	0.0344	Support: Y-Unloaded (1.10) Stiffness:N/A
10	2.546	0.0000	[illegible]	0.0285	
11	2.604	0.0000	0.2661	0.0267	Support: Y-Unloaded (1.10) Stiffness:N/A
12	2.663	0.0000	0.2676	0.0249	
13	3.176	0.0000	0.2756	0.0047	
14	3.690	0.0000	0.2710	-0.0241	Support: Y (1.10) Stiffness:N/A
15	3.800	0.0000	0.2680	-0.0310	
16	4.589	0.0000	0.2280	-0.0660	
17	5.379	0.0000	0.1698	-0.0781	
18	6.168	0.0000	0.1091	-0.0733	
19	6.957	0.0000	0.0571	-0.0572	
20	7.747	0.0000	0.0203	-0.0356	
21	8.536	0.0000	0.0009	-0.0142	
22	9.326	0.0000	-0.0037	0.0011	
23	10.115	0.0000	-0.0005	0.0047	
24	10.225	0.0000	0.0000	0.0039	Support: Y (1.10) Stiffness:N/A
25	10.505	0.0000	0.0007	0.0004	Support: Y-Unloaded (1.10) Stiffness:N/A
26	10.785	0.0000	0.0000	-0.0056	Support: X Y (1.10) Stiffness:N/A
27	11.415	0.0000	-0.0078	-0.0177	
28	11.515	0.0000	-0.0096	-0.0189	
29	11.940	0.0000	-0.0186	-0.0232	
30	12.365	0.0000	-0.0292	-0.0262	
31	12.620	0.0000	-0.0360	-0.0269	
32	12.680	0.0000	-0.0376	-0.0269	
33	12.815	0.0000	-0.0413	-0.0269	

Aft bearing edge Node #9

For propeller shaft in Section 2, Figure 27, the displacement vector table can be used to extract calculated shaft deflection value at the aft bearing edge (node #9):

$d = 0.257\ mm$

Deflection *'d'* is determined with reference to the zero offset line.

To define value of the total clearance at the same node, we have to account for the bearing slope at the same location:

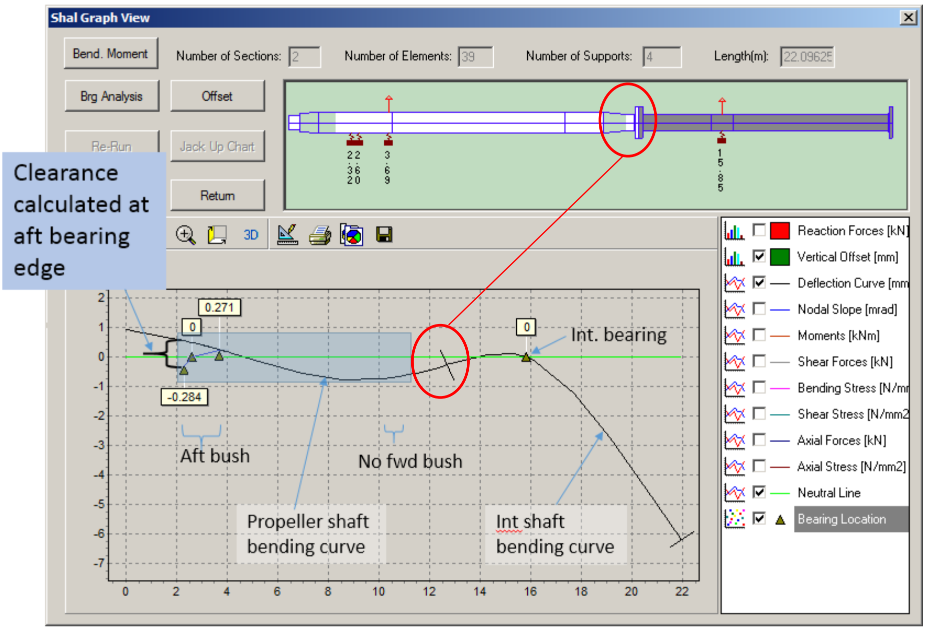
$s = -0.284\ mm / 226\ mm$

The required clearance at the aft edge will be:

$c = d + |s| = 0.541\ mm$

4.12.2 Installation with No-Forward Stern Tube Bearing

FIGURE 28
Installation with No Forward Stern Tube Bearing



For the aft stern tube bearing clearance calculation in installations with no forward stern tube bearing, the propeller shaft and the intermediate shaft are bolted together as shown in Section 2, Figure 28. The reason is that two bearings are needed for the calculation. The second bearing in this case is the intermediate shaft bearing, used instead of the forward stern tube bearing. The aft stern tube bearing clearance shown here is *0.284 mm*.

In practice, the intermediate shaft bearing is precisely positioned in place during the bore sighting procedure. It is important to confirm that the intermediate bearing does not overload when calculating the clearance using the above procedure. A temporary support needs to be added to relieve the intermediate bearing when the calculated bearing reaction is over the manufacturer's allowable limit. When a temporary support is used, its vertical offset is set during the bore sighting procedure simultaneously with that of the intermediate bearing.

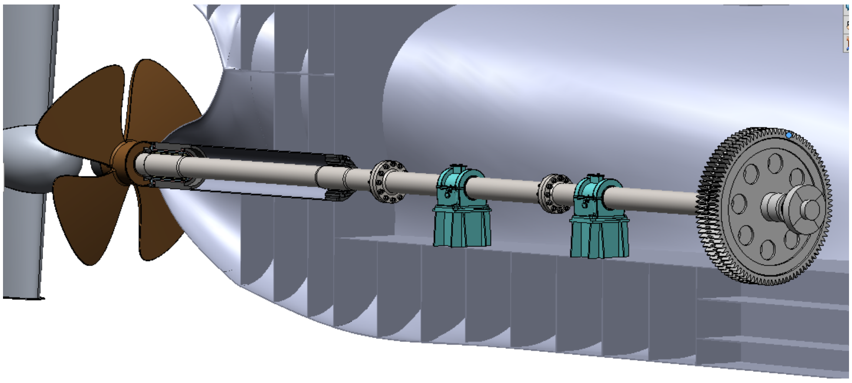
4.13 Gear Meshes

Gear driven propulsion installations, where the propeller is directly connected to the gearbox as in Section 2, Figure 29, may be significantly affected by the shafting alignment condition, resulting in:

- Gear mesh misalignment
- Gear shaft bearing misalignment and overloading

ABS Rule requirements imposed on gear contact are very stringent. Uniform contact across *90%* of the effective face width of the gear teeth is required. Considering this, it is crucial to know how shaft alignment affects gear-pinion misalignment. The gearbox manufacturer defines the misalignment tolerances. Therefore, to confirm *90%* contact between the gear teeth, gear-pinion misalignment must be maintained within these tolerances.

FIGURE 29
Gear Driven Propulsion Systems



The marine industry may use other methods to indirectly control misalignment and gear tooth contact. One commonly applied approach is to investigate gear shaft bearings reaction difference and maintain that within *20%*. This approach may be acceptable only when a zero moment and shear force are maintained at the connecting flange to the line shaft. Failing this, there is no assurance that the misalignment angle is within acceptable limits. The bearing reactions on their own cannot guarantee it.

FIGURE 30
Gear Driven Propulsion – Equal Gear Shaft Bearing Reactions
0.21 mrad Gear Misalignment Angle

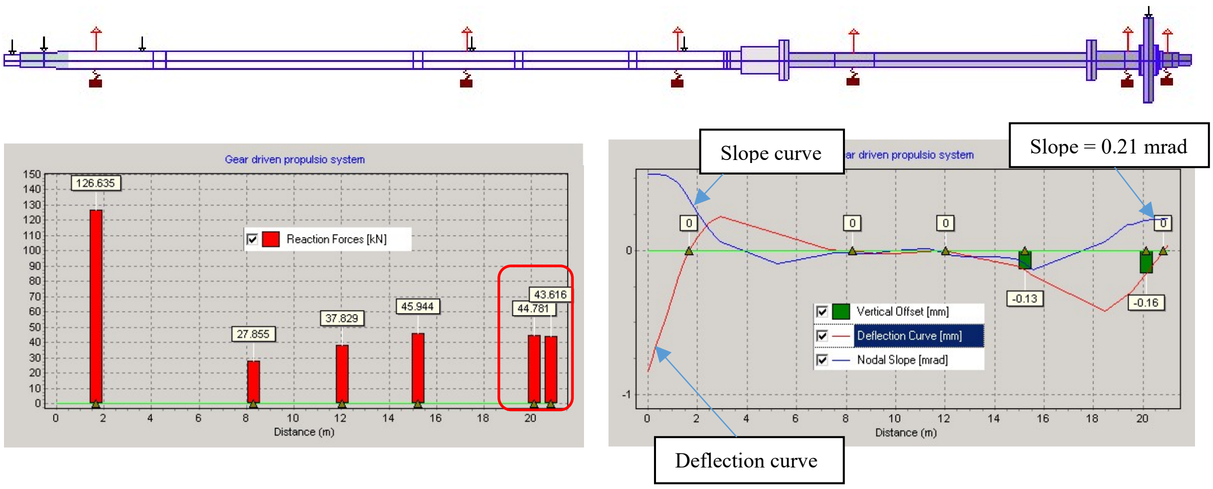
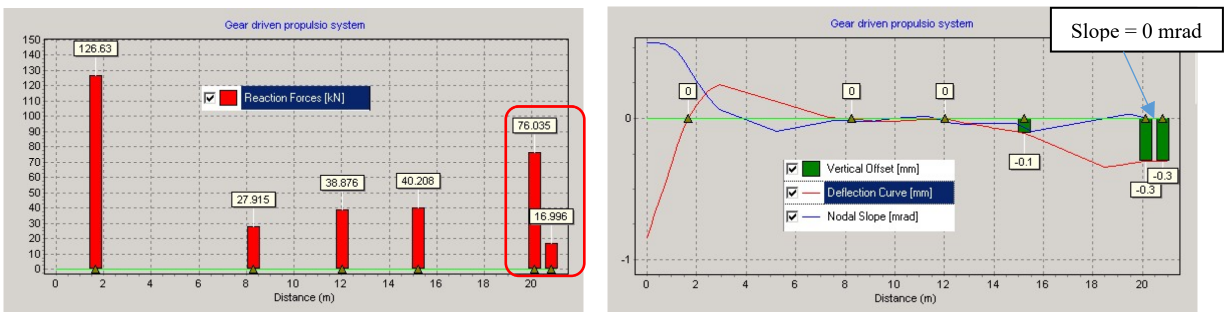


FIGURE 31
Gear Driven Propulsion – Uneven Gear Shaft Bearing Reactions
Zero Misalignment Angle at Gear Wheel



Section 2, Figures 30 and 31 show that an appropriate selection of bearing offsets can bring the gear misalignment angle to almost zero. However, when focusing on a desirable misalignment angle, the gear-shaft bearing reactions will likely be more than *20%* apart. Ultimately the difference in bearing reactions does not matter as long as all bearings remain loaded. ABS considers the approach where a misalignment angle between gear meshes is controlled and maintained as close to zero as possible to be an acceptable method to address gear alignment.

4.13.1 Hybrid or Pseudo-Dynamic Analysis

In the case of geared installations, it may also be advisable to perform a pseudo-dynamic analysis with dynamic gear load applied in addition to an ordinary static investigation. Here, steady acting dynamic loads at gears are incorporated in the static shafting alignment model on the top of the existing static forces. In contrast to the pseudo-dynamic approach where dynamic propeller loads are applied to evaluate the stern tube bearing (where a significant hydrodynamic component of the load is completely neglected), in the gear application the pseudo-dynamic method has much more validity when a steady-state dynamic loads are applied. Nevertheless, the results obtained by a pseudo-dynamic approach should be considered with caution and used primarily to investigate dynamic load impact on gear bearing loads and gear misalignment condition.

Depending of the gearbox design and its operating condition, the reaction load at the gear shaft bearings may overcome the gravity forces and result in negative bearing reactions. Accordingly, the particular ABS Rule requirement for all bearing loads to be positive may not be met in pseudo-dynamic analyses.

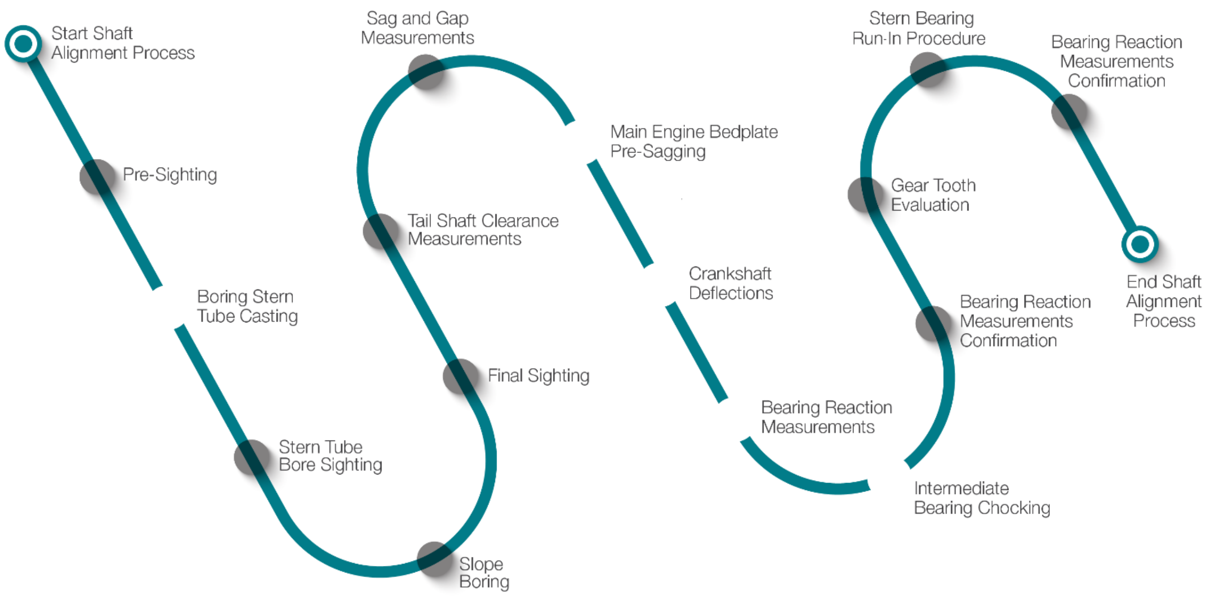
SECTION 3 Shaft Alignment Procedure

1 General

Shaft alignment can be a lengthy process and, in some shipyards, starts as soon as the stern blocks are assembled in drydock. ABS Rules require that major welding work at the stern is completed and recommends that all the heavy stern structure is in place, including the main engine, before shaft alignment procedure commences.

The alignment procedure is the executable part of the shaft alignment calculation prepared by the designer or the builder and is conducted based on parameters defined therein.

FIGURE 1
Shaft Alignment Procedure



The shaft alignment procedure starts while the vessel is in the drydock or on a slipway (Section 3, Figure 1). All sightings are carried out, and sag and gap measurements may be taken at this stage. The measured results in drydock must satisfy the calculated results for the drydock condition.

Experienced shipyards may carry out bearing reaction measurements while in drydock when hull deflections are considered in shaft alignment calculations.

After launching the vessel, bearing reaction measurements and crankshaft deflections are typically taken again and correlated with the approved calculations and the engine manufacturer's requirements. Some hull deflections are expected to exist between the drydock and the afloat phase.

During the sea trial stage, bearing reactions should be verified for specific vessel loading conditions, as stated in ABS Rules. Main engine crankshaft deflections and reduction gear tooth contact are verified for a ballasted vessel with the main engine in hot static condition.

The bearing run-in procedure is required to be carried out for the shaft installations identified by ABS Rules as soon as the vessel is placed in a ballasted condition and before commencing sea trials. Shaft bearing temperatures are monitored and recorded during the run-in procedure and monitored throughout sea trials.

Shaft alignment is considered acceptable when all the measured results are within the acceptance criteria and tolerances of the approved shaft alignment calculations. Acceptance criteria include ABS Rule requirements and manufacturer limits.

The propulsion shafting alignment procedure can be summarized in the following activities:

- Pre-sighting
- Boring of the stern tube casting
- Stern tube bore sighting
- Slope boring
- Final sighting
- Tail shaft clearance measurement
- Sag and gap measurement
- Main engine bedplate pre-sagging
- Crankshaft deflections
- Bearing reaction measurements
- Gear tooth evaluation
- Main engine chocking
- Intermediate bearing offset adjustment and chocking
- Stern tube bearing running in procedure
- Sea trial

Section 4 of the Guidance Notes is dedicated to a more detailed explanation of the alignment measurements and monitoring.

2 Sighting Through (Bore sighting)

The process of establishing the reference line through the center of the bore is often called sighting-through or the bore sighting.

The purpose of the bore sighting is to set the propulsion shafting bearings, the engine or the gearbox, to required position, as well as to define and verify the stern tube bearing slope boring angles.

Sighting is commonly conducted by one of three methods: piano wire, optical instrument, or laser instrument. These methods are examined in more detail in the following paragraph.

The bore sighting procedure will depend on the method utilized for installation of the stern tube bearings. There are two commonly applied methods, the press-fitting and the epoxy resin chocking. Each method will be explained in detail below.

Bore sighting equipment:

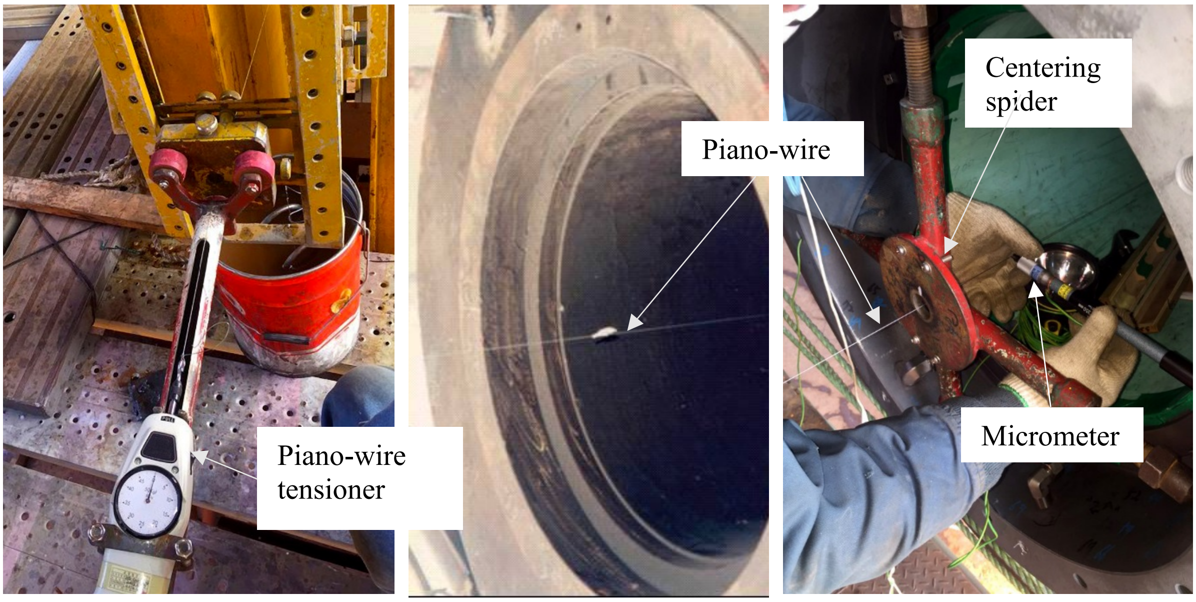
- Piano wire
- Optical telescope
- Laser

2.1 Piano-wire

The piano wire method uses a *0.5 mm* to *0.7 mm* diameter steel wire to represent the reference line. The wire extends from outboard of the stern tube's aft end through the full length of the stern tube.

It continues to the aft end of the main engine flywheel or to a temporary support that represents the main engine's future location, if the main engine has not been installed yet. The wire is threaded through centering spiders positioned at the stern tube casting aft and forward ends and is pre-tensioned with a known force using push-pull scales or a certified weight (Section 3, Figure 2).

FIGURE 2
Bore sighting – Piano Wire



Top, bottom, port, and starboard measurements are taken between the piano wire and the bore of the stern tube casting in way of the bearing locations. The measurements are used to establish the bearing and main engine positions relative to the reference line. It is crucial to remember that piano wire measurements must be adjusted to compensate for the natural wire sag.

The surveyor verifies the measurement records and can request that the shipyard re-measure any questionable results.

Advantages of the method:

- Piano wire sighting is dependable, low-cost, and easy to understand. Shipyards have been using this method for many years.
- Analog micrometers are accurate up to ±*0.01 mm*. The measurement accuracy of piano wire readings may be improved somewhat using digital micrometer gauges.

Disadvantages of the method:

- A skilled worker is needed to understand the piano wire method limitations and to take true measurements with an inside micrometer.
- Micrometers fitted with a visual or an audible indicator can remedy errors somewhat. It is desired to have the same shipyard workers taking all the measurements to maintain consistency.
- Casting surface irregularities, cleanliness, and wire vibrations may influence measurement accuracy.

2.2 Optical Telescope

Optical instrument sighting employs a precision telescope to project an optical reference line (Section 3, Figure 3). The instrument is positioned on a base or tripod and is leveled in the horizontal plane before sighting begins. A reference target, which is utilized to establish a reference line, is fixed at one end and not touched throughout the measurement process. Transparent targets are set in the physical center of the casting bore at several positions. The deviation of each target center is recorded relative to the reference line.

The shaft alignment procedure submitted to ABS for review should include a sighting target arrangement. However, ABS Rules do not specify the number of targets required. The surveyor follows the approved target arrangement. Typically, four to six target locations are selected for the stern tube sighting.

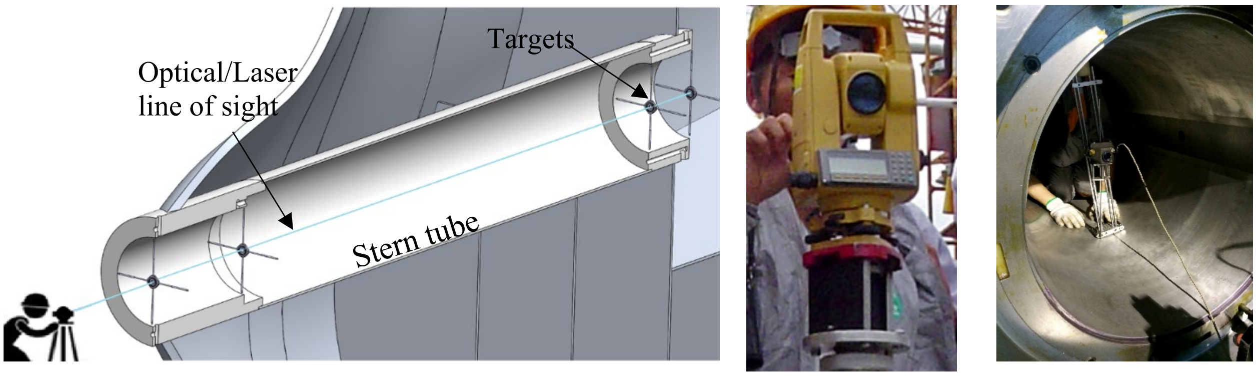
Advantages of the method:

- The optical instrument sighting method is relatively easy to set up and operate. The shipyard also uses the same equipment for other applications, such as for the sighting of hull block accuracy. The instrument's multiple uses make it cost-effective.
- The results of optical instrument sighting are easily read directly from the instrument's internal Vernier scale. Measurement accuracy is *±0.01 mm*.

Disadvantages of the method:

- Sighting accuracy greatly depends on the target's setting, centering precision, and telescope leveling along the reference line.

FIGURE 3
Bore sighting – Optical Instrument & Laser



Optical/laser pre-sighting principle Telescope Laser

2.3 Laser

Laser instrument sighting uses a laser source set in the center of one reference diameter, such as the recess at the aft side of the stern boss (Section 3, Figure 3). Two reference targets are defined. The receiver target is positioned inside the bearing at a location specified in the shaft alignment calculation and a reference reading is taken. The receiver is then moved to selected measuring positions along the reference line and additional readings are taken. The results are digitally recorded. The bearing offsets are automatically calculated in relation to the reference line and may be presented in the form of a graph or table.

Advantages of the method:

- Laser instrument sighting equipment has an accuracy of *±0.005 mm*. In addition, the laser method provides readings and results much faster than the piano wire or optical instrument methods.
- Laser instrument sensors are typically wired to a computer and allow for continuous data acquisition. Laser instrument sighting software also records sensor inclination and automatically calibrates the sensor unit, effectively eliminating human reading error.

Disadvantages of the method:

- Laser instrument sighting equipment is high tech and costly. Shipyards often employ specialized contractors to carry out laser instrument sighting. Accuracy largely depends on the correct setting of the receiver and transmitter along the reference line.
- Laser instrument sighting is sensitive to temperature changes, noise, and vibrations from the surrounding environment.

2.4 Pre-sighting

Pre-sighting is a sighting through the stern tube before the bearings are installed and is primarily used to establish tolerances for machining of the stern tube casting.

Pre-sighting establishes a centerline through the stern tube casting, which may also be projected forward to the front end of the main engine foundation.

ABS Rules do not specifically require the surveyor to witness pre-sighting. However, witnessing this inspection is a good practice.

Pre-sighting is not required if bearings are installed by epoxy resin chocking, because then the inside of the stern tube casting is machined to an oversized diameter that can accommodate the stern tube bearings and the surrounding epoxy that holds them in place.

For more details see Section 4, “Measurements and Monitoring”.

FIGURE 4
Horizontal and Vertical Stern Tube Boring

Horizontal boring

Vertical boring

2.5 Machining of Stern Tube Casting

Once pre-sighting is completed, and the boring machine is aligned with the established reference line, the inner diameter of the stern casting is step-bored for the bearing interference fitting. The stern tube casting is bored in situ in the building dock using a horizontal boring bar or by vertical boring in the block stage (Section 3, Figure 4).

2.6 Stern Tube Bore Sighting and Final Sighting

2.6.1 Stern Tube Bore Sighting before Bearing Machining

The centerline established at the pre-sighting provides a reference line to the builder to confirm the amount of vertical and horizontal stern bore misalignment that may affect shaft alignment, as well as the eccentricity of the stern tube bore.

At this time, the builder also measures the final stern tube inner dimensions with an inside micrometer. This information allows the bearing manufacturer to machine the bearing outer diameter to the correct interference fit and to make any other corrections to compensate for deviations found during the bore sighting, Section 3, Figure 5.

FIGURE 5
Aft Bush Prepared for Machining

The shipyard conducts a confirmatory sighting in the dry dock before the stern tube bearings are fitted to define as-bored inner dimensions and to determine the tolerances for machining the stern tube bearing outer diameter for the required interference fit.

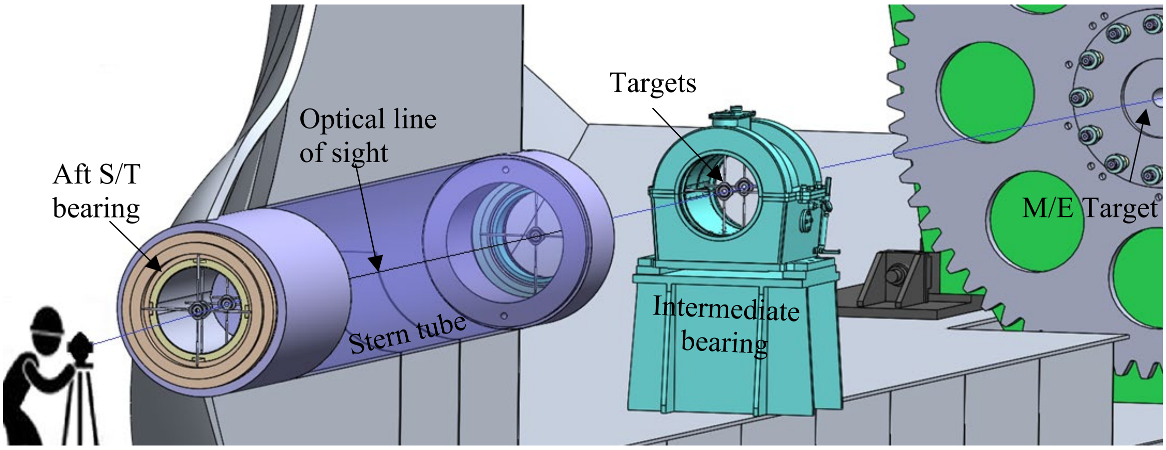
ABS Rules require the surveyor to witness sighting of the stern tube before bearings are fitted.

See also bearing inclination, Subsection 3/3.

2.6.2 Final Sighting after Stern Tube Bearing Fitting

Preferably, when all major steel works are completed and the main engine and other heavy machinery are installed, the final sighting is performed. At this stage the as-fitted position of the stern tube bearings is verified in relation to the engine position and other shaft line bearings (Section 3, Figure 6). The slope boring misalignment angle relative to the center reference line is confirmed. The horizontal misalignment of the stern tube bearings and the main engine are also confirmed. Intermediate bearings are not positioned by sighting as these are removed for propeller shaft fitting and subsequently reinstalled. The exception to this practice is installations with no forward stern tube bearing. These are to be specially considered, as explained below.

FIGURE 6
Final Bore Sighting – No Forward Stern Tube Bearing



Section 3, Figure 6 represents the case with no forward stern tube bearing, where the intermediate bearing is fixed as a second reference point for the aft stern tube bearing slope verification.

2.7 No Forward Stern Tube Bearing Systems

For shafting systems with no forward stern tube bearing, the aft most intermediate bearing is used to control the aft stern tube bearing misalignment, and its position becomes important for the sighting accuracy.

The builder reconfirms the vertical distance from the top of the intermediate bearing pedestal to the shaft centerline established during the pre-sighting. The measured value is compared with the calculated value. The final vertical position of the intermediate bearing is later established during a final sighting.

More about installation with no forward stern tube bearing in Section 4, “Measurements and Monitoring”.

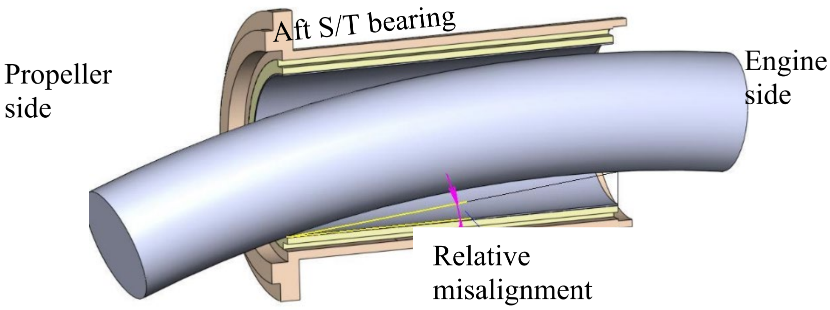
3 Slope Boring or Bearing Inclination

Conditions inside the stern tube bearing change when the tail shaft starts rotating. The transition from a static shaft condition at zero RPM to a dynamic shaft condition and the development of the oil film need to be as swift as possible.

The slope boring or the bearing inclination is applied to maximize contact area and facilitate a fast transition from a static to a dynamic condition. It assists the hydrodynamic lift of the shaft on the bearing lubricant during starting of the engine and minimizes metal-to-metal contact and oil film breakage when the engine comes to a stop.

ABS and other classification societies established a misalignment angle limit of *0.3 mrad*. The bearing misalignment angle is measured from the centerline through both the aft and forward stern tube bearings. A slope boring tolerance or deviation of *±0.1 mrad*, or approximately *1 millimeter per meter ±0.1 mm*, is acceptable.

FIGURE 7
Bearing Misalignment



Slope boring is the preferred method for reducing relative misalignment and is mostly applied to press-fitted stern bearings. During slope boring, the bearing's inner bore is machined in a lathe to the desired angle. The inner diameter centerline is offset at an angle with the outer diameter centerline. The alternative to the slope boring is bearing inclination, where the bearing is chocked under the required angle with epoxy resin, and machining is not required.

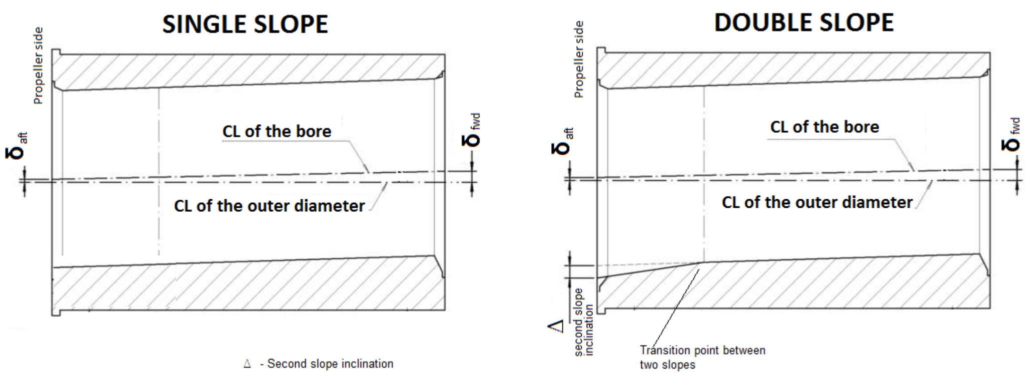
When the propeller-shaft curvature inside the bush is closely matched by two differently angled slopes the design is considered a double slope configuration. The double slope further improves the contact area between the shaft and the bottom of the bearing, resulting in better bearing performance and a longer bearing life. The ABS approach to the double slope design is as follows:

- The transition point between two slopes is defined by ABS Rules and is to be located between *D/3* and *L/4* distance from the aft bearing edge (*D* – bearing diameter; *L* – bearing length).
- The design of bearing slopes is to be such so that in static condition the heaviest load occurs at the transition point between two slopes and the lowest possible load occurs at the bearing aft and forward edges.

Installations with no forward stern tube bearing, are required by ABS Rules to have a double-sloped aft stern tube bearing.

The slope boring procedure depends on the stern tube bearing design (single or double slope) and the method used for the bearing installation (press fitting or inclination). The difference between methods is explained below.

FIGURE 8
Single and Double Slope Bearing Designs



δ - Slope boring or inclination offset between centerline of the bore and the centerline of the outer diameter.

△ - Second slope

The single bearing slope can be accomplished by slope boring or by bearing inclination.

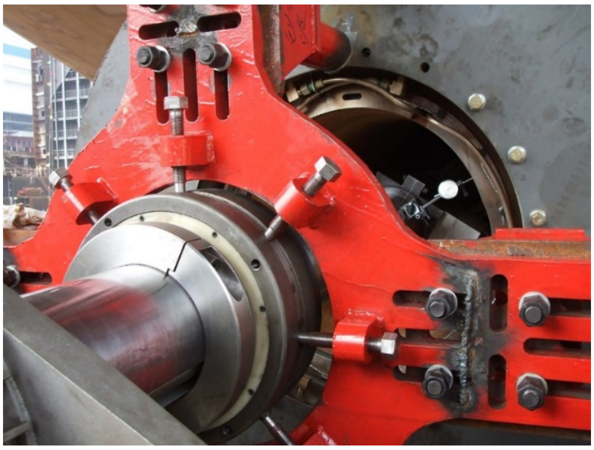
2.6.3 Slope Boring

Slope boring is normally applied by bearing manufacturer, based on parameters defined in shaft alignment calculation information. Commonly, the slope is applied on the inner diameter of the bearing. The outer diameter slope is seldom applied, typically when corrections are required after the stern tube casting machining.

Depending on a design requirement, the manufacturer can supply the bearing with a double or a single slope. A worn or damaged bearing is easier and less costly to replace with this method, because the outer diameter is kept cylindrical.

Alternatively, the bearing slope can be machined in the shipyard with bearing fitted in the stern tube (Section 3, Figure 9). This is often the case when non-metallic bearings are used, or when corrections are required after bearing fitting, or refurbishing.

FIGURE 9
Slope Boring Machine



The onsite slope boring is however a very slow and sensitive process which requires specialized equipment, and machining precision may be affected by environmental conditions such as temperature changes and vibrations.

2.6.4 Bearing Inclination and Epoxy Resin Chocking

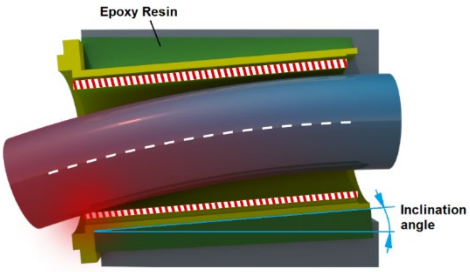
An alternative approach to the pressure fitting is the chocking of the bearings with epoxy resin.

When bearings are chocked the application of a single slope can be achieved by bearing inclination before epoxy casting. The bearings are inclined and fixed under the required angle during the final sighting process (see 3/2.4). This allows precise bearing setting and elimination of relative misalignment between the forward and aft stern tube bearings, which is otherwise difficult to completely eliminate with pressure fitted method.

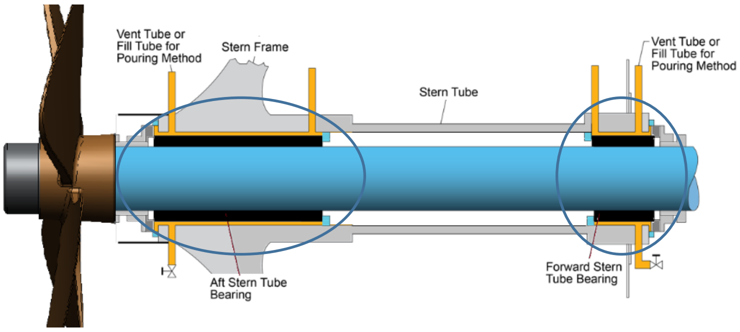
Where double slope design is required, the slope boring is normally conducted by bearing manufacturer.

The preferred bearing inclination and chocking procedure consists of individual chocking of the aft and the forward stern tube bearing (Section 3, Figure 10).

FIGURE 10
Bearing Inclination and Epoxy Resin Chocking



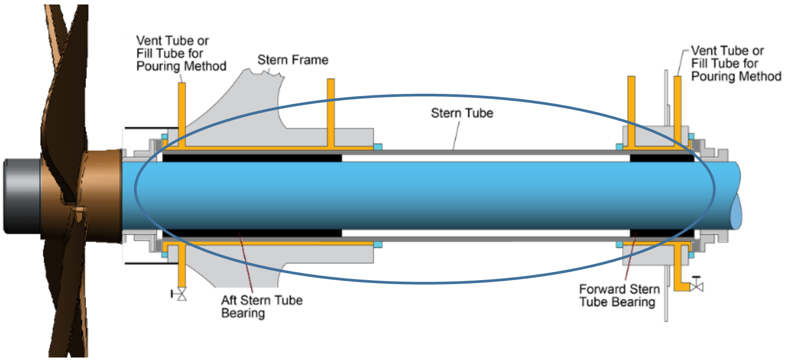
Bearing inclined and chocked in the stern tube.



Each stern tube bearing is set by sighting and individually chocked in the stern tube with epoxy resin. Misalignment can be entirely eliminated.

Alternatively, the whole stern tube can be chocked into the stern block, together with pre-fitted slope bored bearings (Section 3, Figure 11). This approach is no different from the pressure fitting approach explained before and provides no benefits of eliminating bearing misalignment while chocking is performed.

FIGURE 11
Stern Tube Chocking



The stern tube is chocked together with pre-fitted bearings. No benefits of eliminating bearing misalignment by chocking

Inclining and epoxy resin chocking of the bearings in the stern tube casting is a less demanding procedure than the pressure fitting and has the following benefits:

- Pre-sighting is not required
- Stern tube bore does not require precise machining
- Stern tube bearing outside diameter machining is not needed
- Very precise installation of bearings based on bore sighting only

Disadvantage of the bearing chocking is:

- Reduced heat conductivity due to epoxy resin layer between bearing and the stern structure.

Stern tube outboard end design must include a special arrangement to maintain water tightness at the boundary of the epoxy resin with the stern tube casting

The Surveyor's presence is normally required for the final slope boring or bearing inclination verification, which is often conducted as a part of the sighting through process.

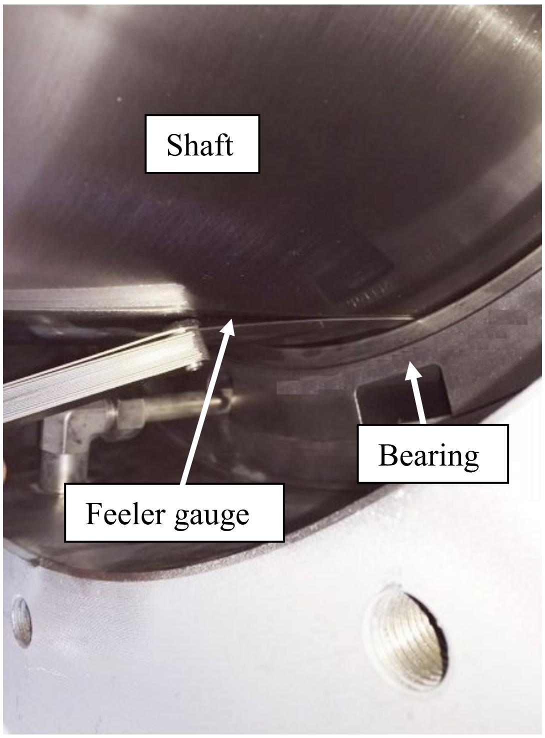
4 Bearing Clearance and Slope Verification Procedure

The propeller shaft is inserted once the stern tube bearings are fitted. The tail shaft bearing clearances are measured as a final verification of the shaft to bearing misalignment before the propeller is fitted (Section 3, Figure 12). With propeller not fitted, the bottom bearing clearance is expected to be greater than zero. The measured values are compared with the calculated clearance values for this condition.

For confirmatory purposes, the tail shaft clearances are measured again after the propeller is fitted. When propeller is fitted the bottom clearance is always zero.

The Surveyor is required to attend the tail shaft clearance measurements before the propeller is installed.

FIGURE 12
Clearance Measurement



The aft stern tube bearing clearance calculation and the measurements procedure will be different for propulsion systems with or without the forward stern tube bearing.

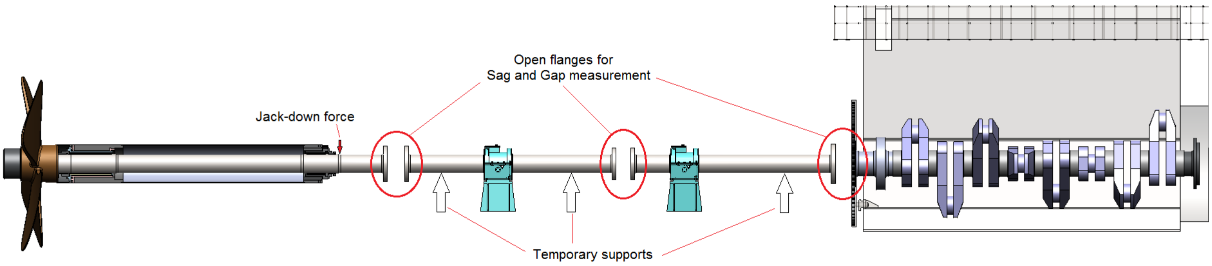
Additional information about clearance measurements is provided in Subsection 4/7.

5 Sag and Gap

The sag and gap procedure is commonly performed prior to the shafting assembly and is simultaneously applied on all open flanges of the system.

The procedure sets up the shafting system to bring the measured sag and gap values close to the calculated values.

FIGURE 13
Sag and Gap – Open Flanges



The conditions and sequence of adjusting the sag and gap values between particular flanges is to be specified in the approved shaft alignment procedure.

Sag and gap is only to be utilized for setting up the shafts, main engine, and intermediate bearings. The procedure is relatively inaccurate, and as such should never be used as a standalone alignment method to modify bearing offsets and engine or gearbox position. Namely, if bearings, engine, or gearbox offsets are set in place by bore sighting, sag-and-gap is not to be used to change their position. Moreover, additional limitation in sag and gap procedure application are to be considered in installation with no forward stern tube bearing, as explained in Subsection 4/8.

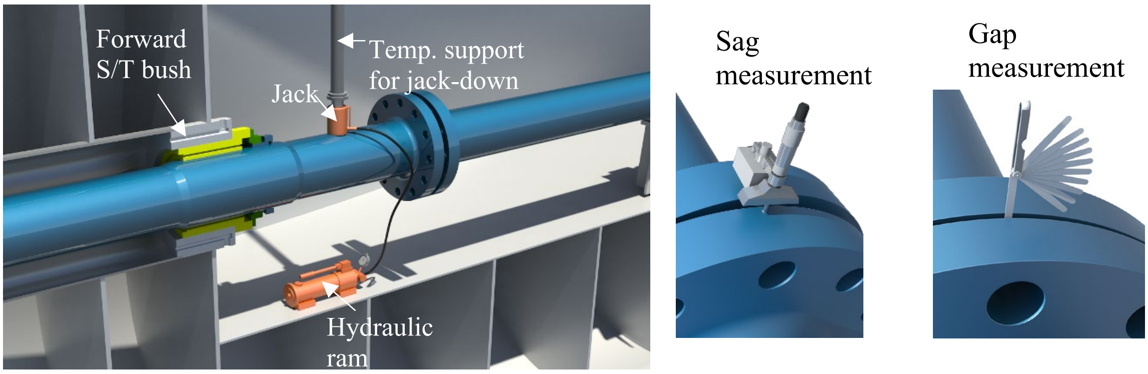
Sag and gap can be only used as a cursory verification method and should not be considered as a final alignment confirmation. For this, jack-up tests are needed to fine-tune bearing offsets and bring bearing reactions within specified limits.

Rigid propulsion shafting systems in certain vessels, such as tankers and bulk carriers, have small shaft deflection values, resulting in bearings being susceptible to small vertical position variations. The sag and gap values are controlled by lifting or lowering of lineshaft supports, as well as adjusting the engine height and inclination until required sag and gap tolerances are achieved. Since measurement is conducted on an open-flange system, the procedure is usually assisted with required number of temporary supports, which are installed in addition to, or instead of the permanent supports (Section 3, Figure 13).

With assistance of the temporary supports the sag and gap has to be simultaneously adjusted at all open flanges until sag and gap values are brought within the acceptable tolerance of ±*0.1 mm* of the corresponding calculated values.

In some instances when there is the possibility of the propeller shaft not sitting on the bottom of the forward stern tube bearing, a jack down load is applied at the forward end of the shaft to maintain bottom contact during the sag and gap verification (Section 3, Figure 14). The required jack-down load is defined in the shaft alignment calculation and applied at the location specified in calculation.

FIGURE 14
Jack Down Force Application



For the purpose of enabling comparison between calculated and measured sag and gap values, the alignment calculation needs to identify how the values of sag and gap are calculated, which sign convention is employed, and how and where the calculated values are to be verified.

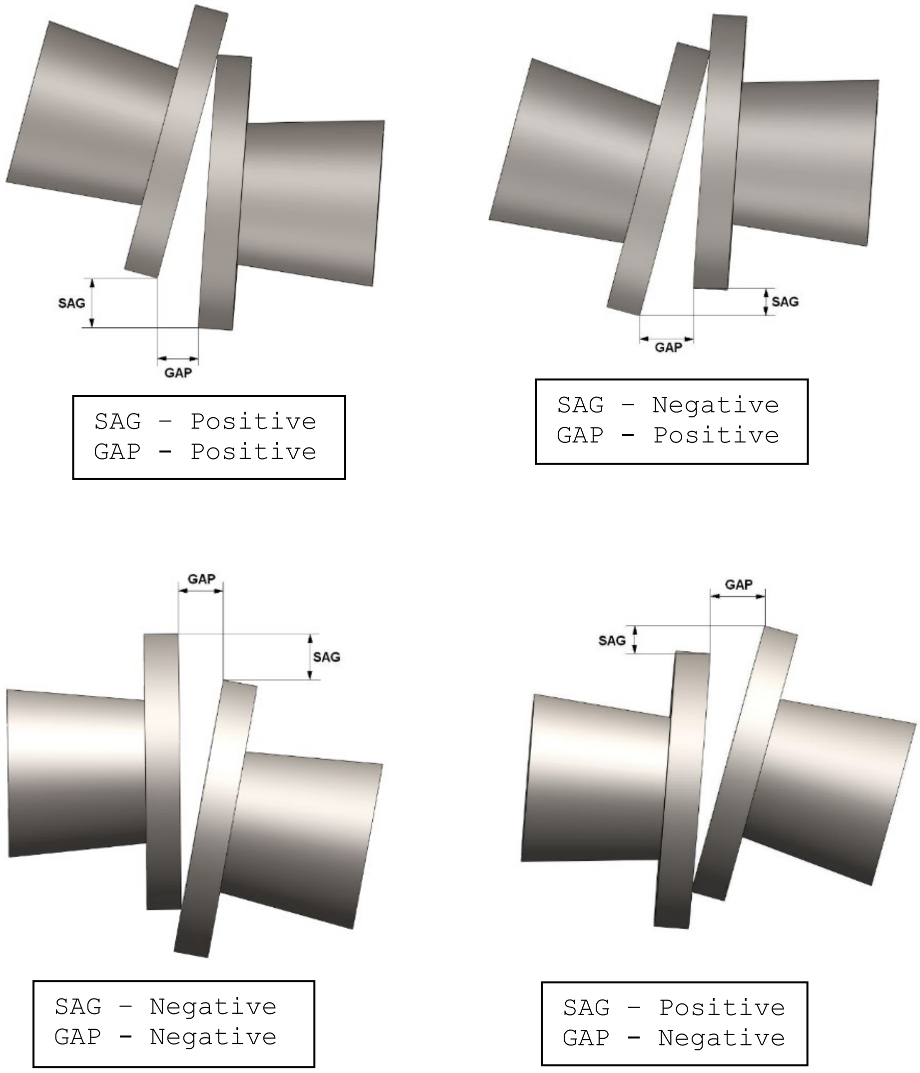
The sag and gap is based on the same beam theory applied for the assembled system. The procedure is defined as follows:

- An acceptable alignment design is defined and calculated first for an assembled system.
- Longitudinal positions of temporary supports are extracted from respective nodes of an assembled system.
- The assembled system is uncoupled at the flanges and each shaft is analyzed separately.
- Displacements and slopes at each end of the shaft flange connection are calculated.
- Sag is calculated by taking the bending displacement at each flange location and subtracting that from the deflection of the mating flange.
- Gap is defined as a difference in distance between the top and the bottom edges of the uncoupled flange pair. Gap at each flange is calculated from the angular inclination of the shaft at flange location and the flange diameter. Total gap is obtained by linear summation of the gaps at both flanges.

ABS defines four distinct positions of interaction between mating flanges for the purpose of uniquely identifying the sag and gap sign convention (Section 3, Figure 15).

Details on sag and gap measurement are given in Subsection 4/6.

FIGURE 15
Flange Arrangement in Sag and Gap Analysis



ABS Rules require the alignment analysis to identify sag and gap data and temporary support location corresponding to the conditions in which they will be measured.

ABS Rules state that sag and gap should be measured in drydock, or after launching in a light afloat condition, unless agreed to otherwise by ABS.

ABS Rules further state that sag and gap measurements must be verified in the presence of a surveyor, unless otherwise agreed to by ABS. The surveyor verifies that sag and gap measurements at all open flanges are brought within ±*0.1 mm* of the corresponding calculated values.

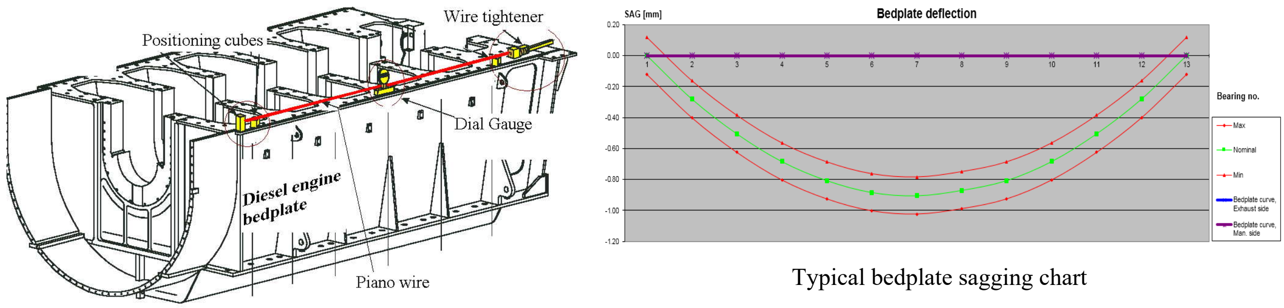
When there is no forward stern tube bearing, the sag and gap is not required to be verified between intermediate and the propeller shaft. Further explanation is provided in 3/9.2.

6 Engine Bedplate Pre-sagging

Hull deflections and thermal expansion of an engine in operation cause the engine bedplate to hog, thereby changing the vertical offsets of engine crankshaft bearings. These changes ultimately affect the bearing load distribution in the propulsion system as well as the crankshaft deflections. The main engine bedplate is pre-sagged to compensate for this hogging effect.

Pre-sagging is applied based on the engine maker's recommendations and procedures. The pre-sag compensation is typically applied to engines with more than 6 cylinders and a cylinder bore larger than *600 millimeters*.

FIGURE 16
Bedplate Sagging Measurement Using Piano Wire



Typical bedplate sagging chart

Some engine makers recommend pre-sag application and verification to be conducted with vessel afloat. Piano wire is one of the methods used for verification (Section 3, Figure 16).

Results of bedplate pre-sag are normally confirmed for each installation by the engine manufacturer.

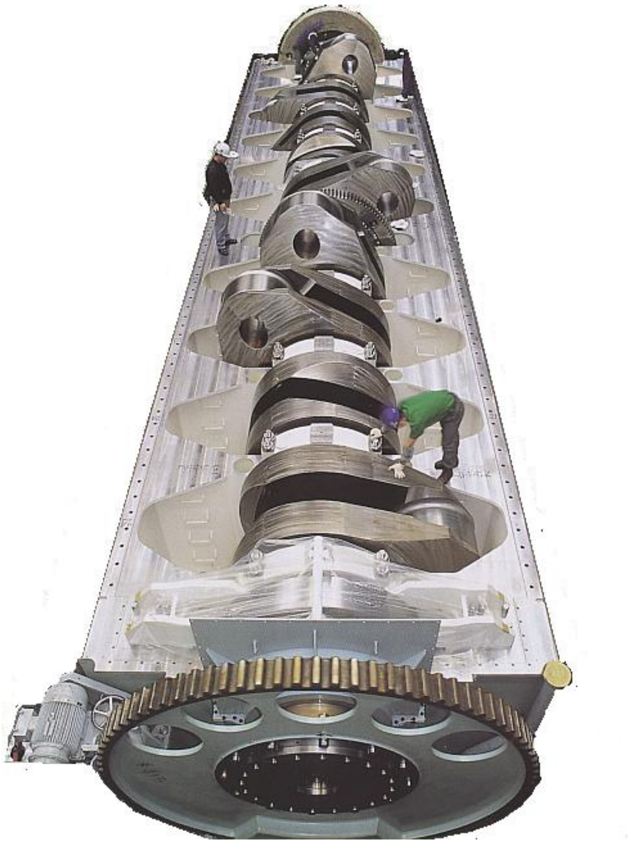
Surveyor's attendance during the pre-sag procedure although not required can be arranged on client request.

7 Crankshaft Deflections

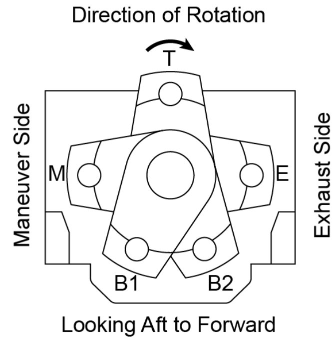
Crankshaft deflection measurement is an indirect method of verifying that the crankshaft stress levels remain within the manufacturer's recommended limits. Crankshaft deflection values indicate how much the adjacent webs open and close during one crankshaft revolution.

Readings are taken at prescribed angular locations of the crank webs. A different location naming convention may be used depending on the engine manufacturer. The first measurement is where the piston is at its lowest point, also known as its bottom dead center, or *BDC* (bottom dead center), point. Refer to Section 3, Figure 17. The deflection gauge is inserted at *B1* with the crank rotated just enough to install the deflection gauge without interference from the crank pin. On the way down, the last measurement is taken at *B2*, just before the deflection gauge would touch the crank pin again at the other side. Measurements taken in between are located at the port side (*M*), top (*T*) and starboard side (*E*).

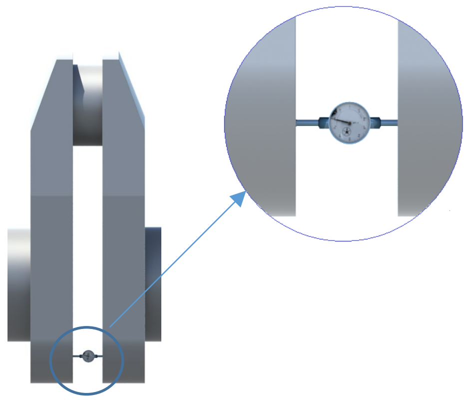
FIGURE 17
Crankshaft Installation in the Engine



Crankshaft installation in the engine



Five crankshaft angles where deflection readings are normally taken at each cylinder.



Dial gauge records crank web opening and closing deflections.

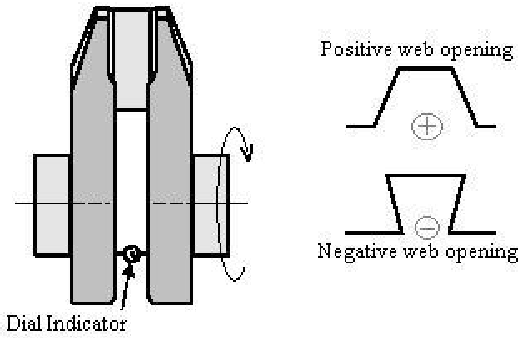
Satisfactory crankshaft deflection measurements indicate that the engine position and engine bearing offsets are within the engine manufacturer's acceptance criteria.

Engine makers recommend crankshaft deflections to be as close to zero as possible for the in service condition with warm engine and a loaded vessel. Each engine maker sets acceptable deflections limits for crankshaft. Deflections limits are typically different for new and old engines.

Crankshaft deflection verification is an important part of the shaft alignment process. The shear force and the bending moment at the flange connected to the intermediate shaft influence crankshaft deflection values of the three aft most crank throws. The main engine manufacturer verifies the main engine crankshaft deflections during bearing reaction measurements.

When bearing offsets are adjusted during shaft alignment, the crankshaft deflections must be reconfirmed to be within the engine manufacturer required tolerances.

The engine manufacturer supplies an engine tool set with a deflection gauge and proposes a sign convention for measuring. For some engine manufacturers, opening the webs is represented by the plus sign, and closing the webs is represented by the minus sign.



When attending crankshaft deflection measurements, the surveyor compares the measured deflections against the engine manufacturer's specified tolerances. Crankshaft deflections provide a first indication of a correctly installed and aligned engine and are taken before and after the main engine chocking and during or after sea trials in hot static condition.

ABS Rules do not require the surveyor to witness crankshaft deflections during the bearing reaction measurements. However, on client request the surveyor presence during the procedure can be arranged as a part of the ongoing shaft alignment process.

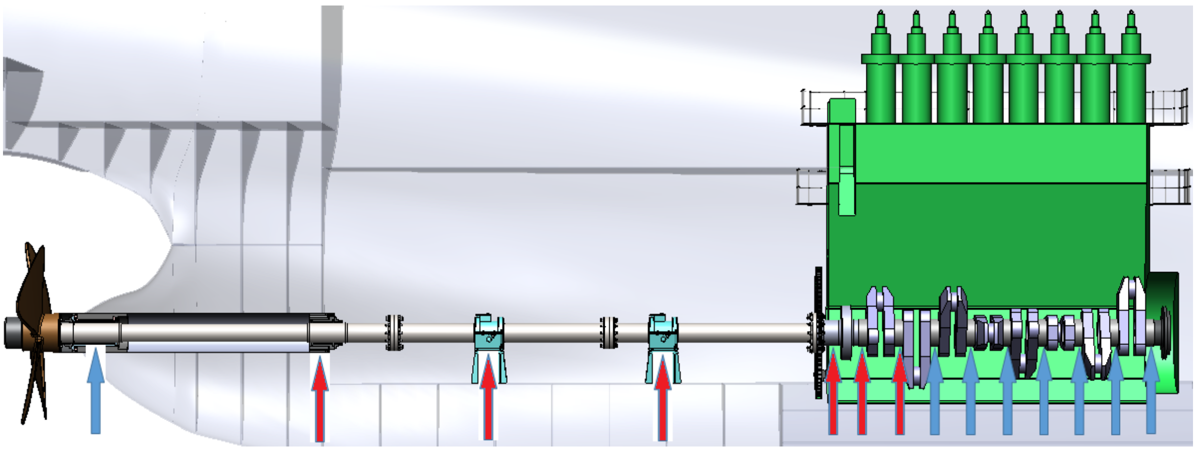
8 Bearing Reactions

Satisfactory bearing reaction measurements are the primary confirmation of shaft alignment acceptability. Bearing reaction measurements indicate whether bearing offsets deviate from the calculated values in the shaft alignment model. Bearing reactions are in direct correlation with the bearing offset, and changing the vertical bearing position results in a proportional variation of bearing loads.

The shipyard conducts bearing reaction measurements with the vessel in the light afloat condition to verify that the alignment condition corresponds to the approved calculations. Bearing reactions are typically confirmed at all accessible bearings with the jack-up method.

In Section 3, Figure 18, the red arrows indicate bearing reactions that are typically verified on the vessel. The aft stern tube bearing is not accessible for jack up reaction measurement. Although accessible, the main engine bearing reactions are typically confirmed on the three aft most main engine main bearings only.

FIGURE 18
Bearing Reactions



The measured values are plotted to generate a jack-up curve from which the bearing reaction can be derived and compared with calculated value. Bearing reaction measurements are re-confirmed during sea trials.

Bearing offset adjustment, hull deflections, bearing foundation flexibility, and thermal deviations have a significant effect on bearing reactions and make it difficult to predict accurate acceptability margins.

Reaction measurements of the main engine main bearings are particularly sensitive to these external influencing factors because of closely located bearings.

ABS Rule requirements in relation to bearing reaction measurements specify that bearing loads under all operating conditions should be within the acceptable limits specified by the bearing manufacturer and as indicated in the calculation of shaft alignment document. Confirmatory bearing load measurements are taken during sea trials in hot static condition, as applicable.

ABS Rules also define minimum acceptability criteria for the other bearings in the system.

If hull deflections are considered in the design, measured bearing reactions must be within ±*20%* of the calculated values and must not exceed *80%* of the manufacturer's maximum allowable limit.

If hull deflections are not considered in the design, measured bearing reactions may not exceed *80%* of the manufacturer's maximum allowable limit.

Before evaluating jack-up measurement results, the surveyor consults with the builder to establish if hull deflections were considered in the analysis, which bearings indicate a deviation from the approved calculations and by how much, and what the expected bearing load distribution is when the draft is increased to fully ballasted and fully laden conditions.

The builder must confirm that bearing reaction results close to zero or close to the upper limit will not adversely affect alignment under increased draft conditions.

8.1 Jack-up

Bearing reaction measurements are either conducted by the jack-up method, using hydraulic jacks or by strain gauge measurements, another accepted, but less frequently utilized method.

Jack-up is the only direct and most widely used method for bearing reaction verification. A hydraulic jack is installed to lift the shaft and measure the load at a jacking location close to the particular bearing.

The jacking load is indicated by the hydraulic pressure applied during lifting and lowering of the shaft. At the jacking location, a dial gauge is fixed to a magnetic stand mounted on the bearing housing, with the gauge spindle contact point touching the shaft. The measurement indicates the amount of shaft displacement. Shaft displacement and applied jacking load are recorded and plotted on a graph, from which the bearing load can be calculated.

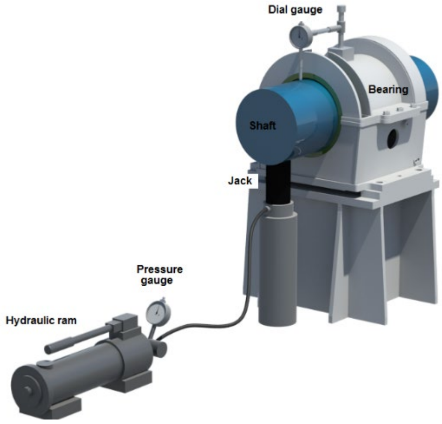
The jack-up method requires the same preparation time for each repeated measurement, making it time-consuming when repeated measurements are required. Misalignment of the hydraulic jack and dial gauge with the shaft center can result in a reduced accuracy of the lifting and lowering curves and a wide hysteresis.

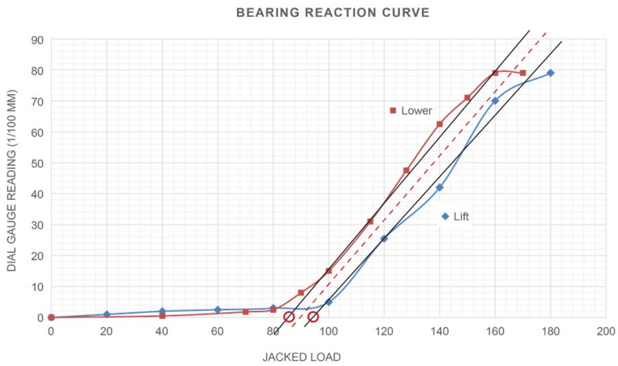
Accuracy of the measurement can be improved and hysteresis minimized by using load cells.

The jack-up method measures the jacked load close to the bearings instead of in the bearing center and pre-calculated correction coefficients must be applied to calculate the actual bearing reaction.

Jacking curves for each bearing type have a different appearance. The builder and surveyors should be trained to evaluate and interpret results correctly.

FIGURE 19
Jack-up Reaction Measurement





Typical jack-up chart

8.2 Strain Gauge

The strain gauge method is particularly valuable in systems where bearings are not accessible for jacking up, such as at strut bearings. It is an indirect method which utilizes the reverse engineering approach to evaluate bearing loads and offsets from the measured bending strains (Section 3, Figure 20).

Strain gauges are installed on the surface of the shafts at multiple locations and can measure vertical and transverse bearing loads. The measured strain is converted into bending moments and stress and that are recalculated into bearing loads and shaft deflections.

Typically, two pairs of strain gauges are glued to the propulsion shaft in pre-selected locations *180 degrees* apart. The strain gauges are hooked up to an input/output interface unit that is connected to a computer. The computer monitors, records, stores, and processes the incoming data.

A minimum of three strain gauge locations are recommended for installations with one intermediate bearing.

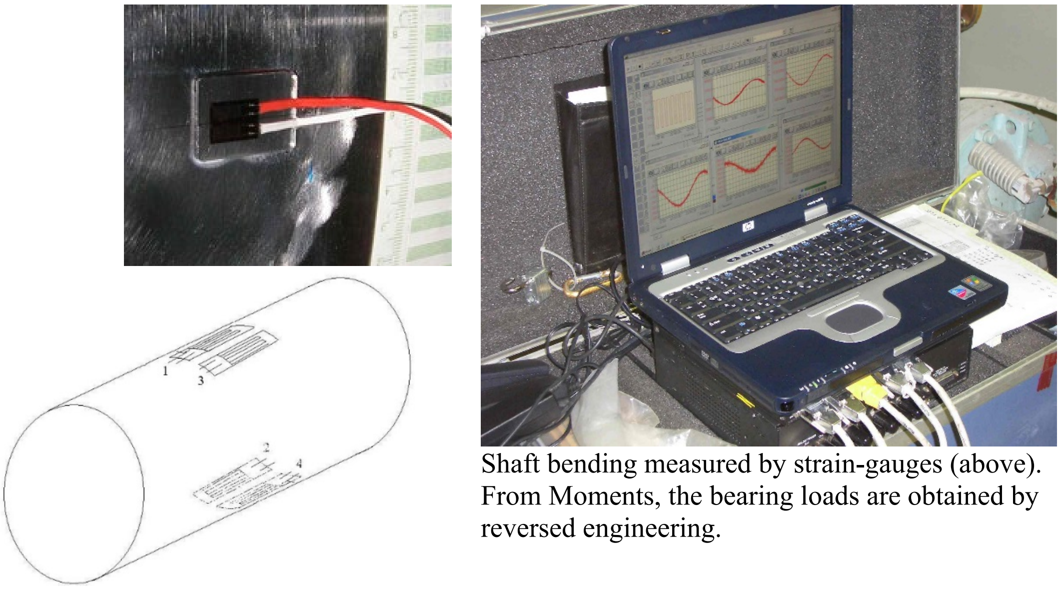
Installation of strain gauges takes much time and requires specialized personnel to install the sensitive equipment and conduct measurements. However, once installed, measurements are easily repeatable.

When strain gauges are installed with a telemetry system, the performance of the shafting system can be evaluated with the vessel in operation.

The accuracy of the results obtained by the strain gauge method depends on the precision of the strain gauge installation, quality of the reverse engineering model, and the accuracy of parameters utilized in analytical model. Bearing loads obtained by the strain gauge method should be reconfirmed on accessible bearings with the jack-up method.

The jack-up and the strain gauge bearing reaction measurement procedures are described in detail in Section 4.

FIGURE 20
Strain Gauge Measurement



Shaft bending measured by strain-gauges (above). From Moments, the bearing loads are obtained by reversed engineering.

9 Intermediate Bearing Offset

Commonly, when corrections in the alignment system are required, it is the intermediate bearing offset which is fine-tuned to obtain desired outcome.

The shaft alignment procedure is conducted differently on a system with no forward stern tube bearing. In latter case, the particular concern is to verify that the aft stern tube bearing misalignment is not adversely affected by inappropriate adjustment of the intermediate shaft bearing.

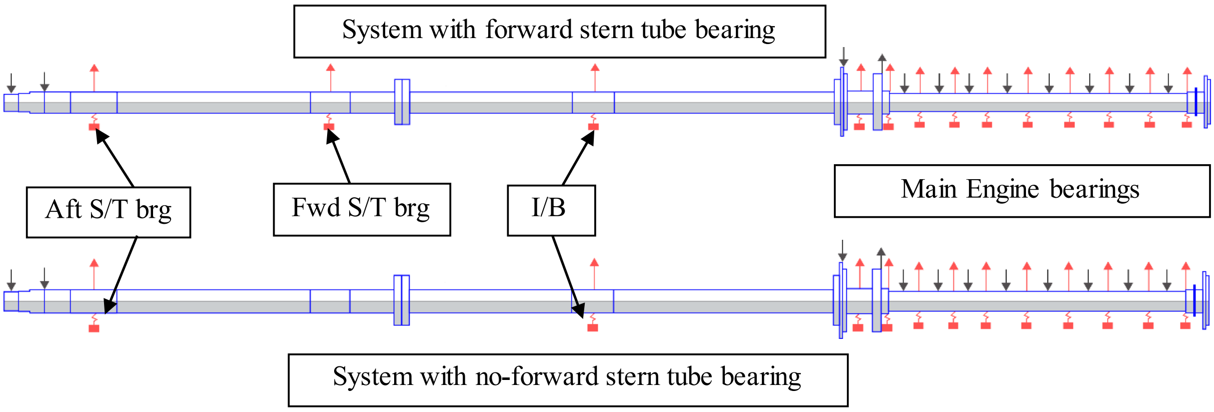
In conventional alignment design, with both stern tube bearings installed, the intermediate-shaft bearing offset is commonly adjusted to:

- Correct and fine-tune the main engine bearing reactions.
- Bring crankshaft deflections within manufacturer's limits.
- Adjust the forward stern tube bearing reaction load, when bearing is unloaded or its reaction is too low.
- On the alignment sensitive installations, with only one intermediate bearing and short and rigid shafting, such as on tankers and the bulk carriers, applying the corrections to resolve an issue on one bearing may- and often does - worsen the other two adjacent bearings. The intermediate bearing offset adjustment is often a fine balancing act that can greatly benefit from the application of specialized optimization software.

Note: Crankshaft deflections and main engine bearing reactions are correlated. Adjustment of the one directly influences the other. Accordingly, any adjustment of the intermediate bearing influences the condition of both. One should be aware that correcting one parameter may result in worsening another.

To demonstrate the impact of intermediate bearing adjustment on the overall alignment condition, shaft alignment analysis was carried out for both a system with and without the forward stern tube bearing, per Section 3, Figure 21.

FIGURE 21
Propulsion System With and Without Forward Bush



Both designs are analyzed for their sensitivity to intermediate bearing offset variations as shown Section 3, Figures 22 and 24. For the purpose of this analysis, the intermediate bearing position is kept in the same location for both designs.

We investigated the impact of intermediate bearing offset adjustments of *0.1 mm*, *0.2 mm*, *0.5 mm*, and *1.0 mm* from the initially prescribed baseline on:

- The change in misalignment slope of the aft stern tube bearing
- The reaction change at the main engine aft most bearing
- The reaction change at the main engine second aft most bearing

Unloading or misalignment of these three bearings is of particular concern because of the severity of the consequences thereof.

Both analyses, with and without forward stern tube bearing, resulted in the aft most main engine bearing unloading when the intermediate bearing is raised by more than *0.5 mm*. We can draw this conclusion by consulting the respective influence coefficients matrices.

An important part of the observation was the notable impact on the bearing reactions at two aft most main engine bearings as the offset at the intermediate bearing gradually changed from *–1 mm* to *+1 mm*. The change in gradient of the plotted curves provides valuable information about load transfer from one bearing onto the other.

The blue line in Section 3, Figures 22 and 24 correlates the misalignment angle of the stern tube bearing with the intermediate bearing offset change.

9.1 System with Forward Stern Tube Bearing

In systems with both stern tube bearings shown in Section 3, Figure 22, the following is observed when changing the intermediate bearing offset from *+1 mm* to *-1 mm*:

- The maximum variation of the aft stern tube bearing misalignment is less than *0.1 mrad* (acceptable misalignment variation in ABS Rules is *+/-0.1 mrad*).
- When the intermediate bearing lifts for more than *0.2 mm*, the aft stern tube bearing misalignment gradient changed direction, indicating that the forward stern tube bearing becomes unloaded.
- With a further increase of the intermediate bearing offset, the forward stern tube bearing unloads more and more, until the system behaves in a similar manner as a system with no forward stern tube bearing.
- The aft most main engine bearing gradually unloads as the intermediate bearing offset increases.
- The second aft most main engine bearing behaves in an opposite manner to the aft most main engine bearing. Its reaction load increases with an increasing intermediate bearing offset until the aft most main engine bearing unloads. After that point a slightly decreased bearing load is observed as lift on *I/B* progresses.

Advantages:

- A system with a forward stern tube bearing is preferred because misalignment of the aft stern tube bearing is less sensitive to offset adjustments at the intermediate bearing.

Disadvantages:

- This system is more rigid, therefore less accommodating to hull deflections, and more susceptible to unloading of the bearings. This means that hull deflections have a more pronounced impact on the alignment.

FIGURE 22
Aft and Forward Stern Tube Bearing – Sensitivity to Intermediate Bearing Offset Change

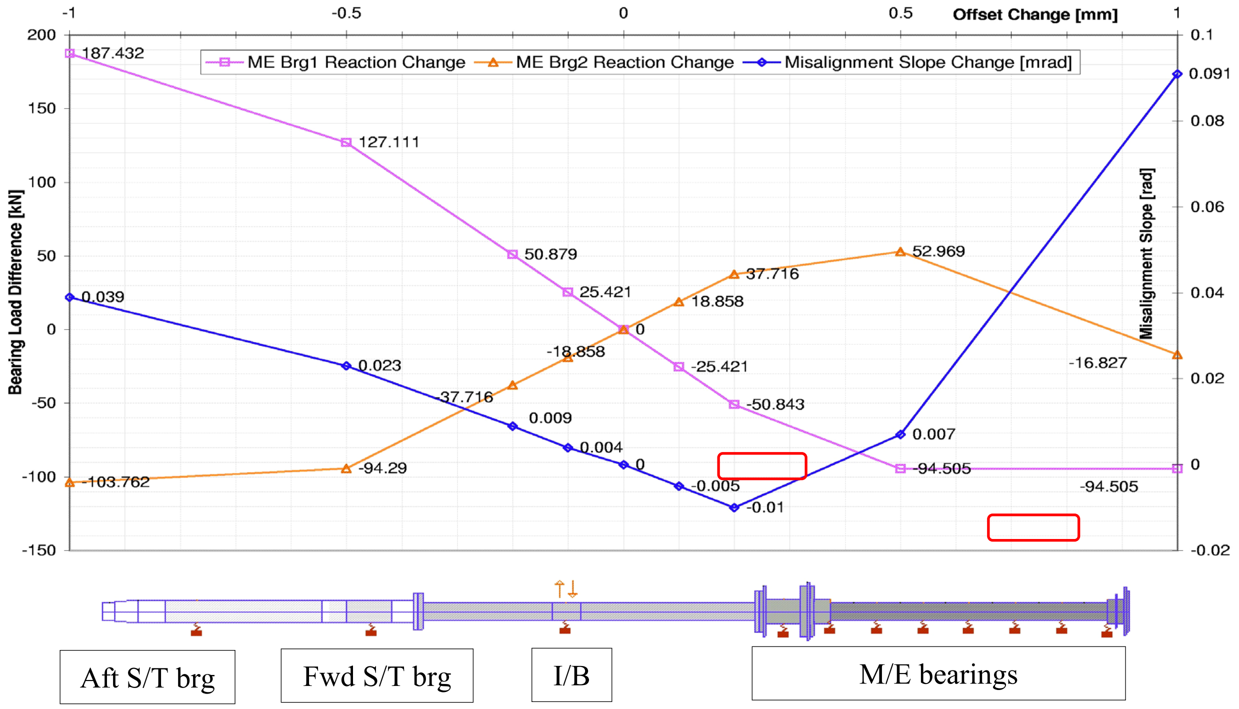


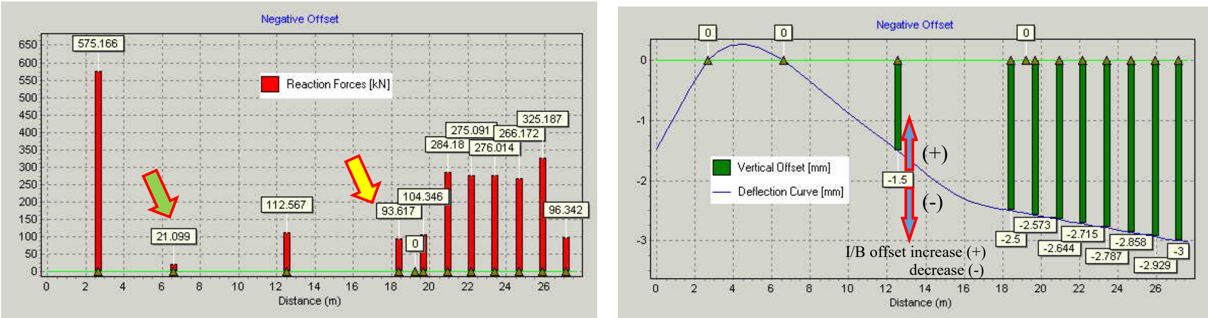
TABLE 1
Influence Coefficient Matrix – System With Forward Stern Tube Bearing

RELATIVE BEARING REACTIONS [kN] -> R[0.1-offset]-R[0-0ffset]
Due to 0.1[mm] OFFSET relative to the ZERO bearing Offset

Node	Supp.	< 7> 1	< 14> 2	< 27> 3	< 41> 4	< 46> 5	< 48> 6	< 50> 7	< 52> 8	< 54> 9	< 56> 10	< 58> 11
< 7>	1	4.598	-8.354	***4.963***	-3.354	2.206	-0.072	0.015	-0.003	0.001	-0.000	0.000
< 14>	2	-8.354	16.063	***-11.351***	10.113	-6.653	0.218	-0.044	0.009	-0.002	0.000	-0.000
< 27>	**3**	***4.963***	***-11.351***	***13.478***	***-25.460***	***18.883***	***-0.619***	***0.126***	***-0.026***	***0.005***	***-0.001***	***0.000***
< 41>	4	-3.354	10.113	***-25.460***	166.620	-250.841	123.821	-25.141	5.104	-1.035	0.202	-0.028
< 46>	5	2.206	-6.653	***18.883***	-250.841	511.836	-379.403	125.081	-25.395	5.148	-1.004	0.141
< 48>	6	-0.072	0.218	***-0.619***	123.821	-379.403	461.940	-295.733	108.086	-21.910	4.273	-0.599
< 50>	7	0.015	-0.044	***0.126***	-25.141	125.081	-295.733	389.320	-280.955	104.921	-20.460	2.870
< 52>	9	-0.003	0.009	***-0.026***	5.104	-25.395	108.086	-280.955	386.156	-279.504	100.648	-14.120
< 54>	9	0.001	-0.002	***0.005***	-1.035	5.148	-21.910	104.921	-279.504	381.883	-259.045	69.538
< 56>	10	-0.000	0.000	***-0.001***	0.202	-1.004	4.273	-20.460	100.648	-259.045	281.235	-105.848
< 58>	11	0.000	-0.000	***0.000***	-0.028	0.141	-0.599	2.870	-14.120	69.538	-105.848	48.046

Column #*3* and row #*3* in the above influence coefficient matrix represent the reaction load variation at all system bearings, as the offset at the intermediate bearing changes by *0.1 mm*.

FIGURE 23
Bearing Reactions for Design Offset – With Forward Stern Tube Bearing



The influence coefficient matrix Section 3, Table 1 indicates that an offset increase by *0.1 mm* at the intermediate bearing (Section 3, Figure 23, chart on the right, blue arrows) results in a *-25.4 kN* reaction change at the aft most main engine bearing (row *#3* column *#4*). The same intermediate bearing offset raised by *0.5 mm*, reduces the reaction at the aft most main engine bearing by five times as much, to *-25.4kN x 5 = -127.3 kN*.

Section 3, Figure 23 shows the calculated load of the aft most main engine bearing to be *93.6 kN*. It is clear then that a *0.5 mm* offset increase on the intermediate bearing, producing a load change greater than the calculated load of *93.6 kN* (Section 3, Figure 23, chart on the left, yellow arrow), will completely unload that bearing. This unloading is indicated by the sudden change in gradient and flattening out of the bearing's loading line, represented by the purple line in Section 3, Figure 22.

A similar analysis conducted on the forward stern tube bearing shows that it will also unload as the intermediate bearing is lifted. Section 3, Figure 23, chart on the left, green arrow, shows the calculated load of the forward stern tube bearing to be *21.1 kN*. Unloading of the forward stern tube bearing happens faster than at the aft most main engine bearing. The influence coefficient matrix shows that raising the intermediate bearing offset by *0.2 mm* will completely unload the bearing. Because its influence coefficient is *-11.3 kN*, and a *0.2 mm* offset change at the intermediate bearing produces a reaction change of *-11.3 kN x 2 = -22.7 kN* which is greater than the calculated load of *21.1 kN*. This means that the forward stern tube bearing unloads.

Unloading of the forward stern tube bearing in turn causes another gradient change of the aft stern tube bearing misalignment angle after the main engine aft most bearing unloads. This happened when offset at the intermediate bearing increases beyond *0.5 mm*. The gradient of the misalignment curve (Section 3, Figure 22) changed sharply, and in this case, it had a positive effect as it reduced the stern tube bearing misalignment angle. The maximum change in misalignment at the bearing was *0.101 mrad* (which is significantly less compared to system with no forward stern tube bearing).

9.2 System with No Forward Stern Tube Bearing

In systems designed with no forward stern tube bearing, the following is observed in an example when changing the intermediate bearing offset from *-1 mm* to *+1 mm* (Section 3, Figure 24):

- The aft stern tube bearing misalignment changes proportionally with the intermediate bearing offset change.
- The maximum variation of the aft stern tube bearing misalignment is approximately *0.2 mrad.*
- The aft most main engine bearing gradually unloads as the intermediate bearing offset increases
- The second aftmost main engine bearing reaction load increases with an increasing intermediate bearing offset but begins to unload as soon as the aft most main engine bearing becomes fully unloaded.

Advantages:

- The most significant advantage of a system with no-forward stern tube bearing is that aft stern tube bearing misalignment corrections are possible before the propeller fitting, by indirect measurement of the bottom bearing clearance. The correlation between the clearance and misalignment in aft stern tube bearing is explained in more detail in Section 4.
- A system with no-forward stern tube bearing is more flexible, making line shaft and main engine bearings less susceptible to the hull deflections. The aft stern tube bearing though, does remain sensitive to the impact of hull deflections.

Disadvantage:

- The aft stern tube bearing misalignment is directly affected by an intermediate bearing offset change. This may become critical if intermediate bearing is not correctly positioned by sighting, or if its offset was amended for a purpose other than misalignment correction, such as for example during the sag and gap adjustments. Refer to tail shaft bearing clearance measurement for more details.
- The aft stern tube bearing misalignment angle is more sensitive to propeller loads and therefore, careful multi-slope design is critical to reliable operation. ABS measurements have shown that during vessel heavy maneuvering, the intermediate bearing load may be also affected in terms of becoming more readily unloaded or overloaded.

FIGURE 24
No-Forward Stern Tube Bearing – Sensitivity to Intermediate Bearing Offset Change

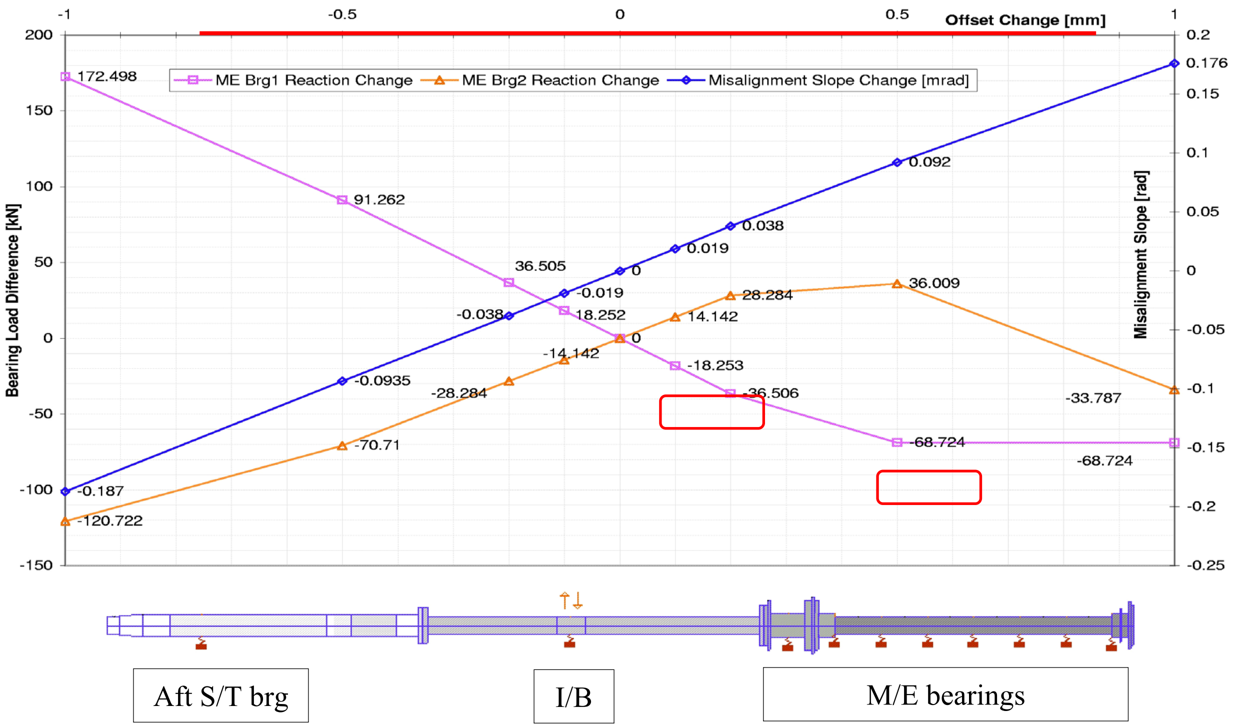


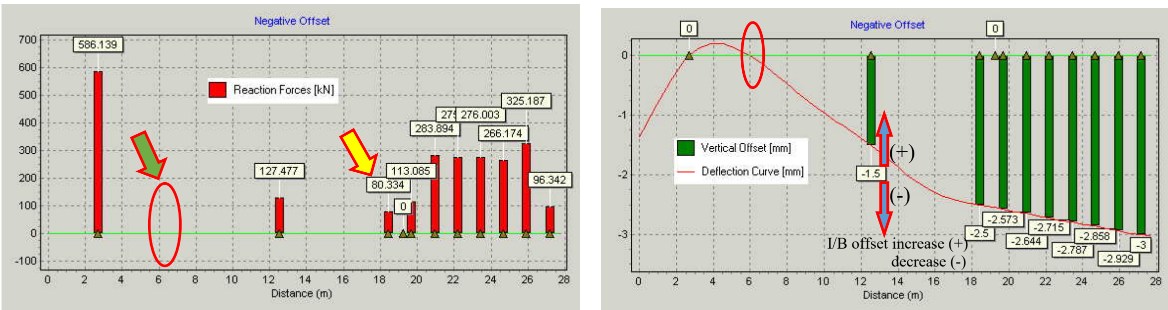
FIGURE 25
Influence Coefficient Matrix – System Without Forward Stern Tube Bearing

RELATIVE BEARING REACTIONS [kN] -> R[0.1-offset]-R[0-0ffset]
Due to 0.1[mm] OFFSET relative to the ZERO bearing Offset

Node	Supp.	< 7> 1	< 27> 2	< 41> 3	< 46> 4	< 48> 5	< 50> 6	< 52> 7	< 54> 8	< 56> 9	< 58> 10
< 7>	1	0.254	-0.940	1.905	-1.253	0.041	-0.008	0.002	-0.000	0.000	-0.000
< 27>	2	-0.940	5.458	-18.314	14.182	-0.465	0.094	-0.019	0.004	-0.001	0.000
< 41>	3	1.905	-18.314	160.253	-246.653	123.683	-25.113	5.099	-1.034	0.202	-0.028
< 46>	4	-1.253	14.182	-246.653	509.081	-379.313	125.063	-25.391	5.147	-1.004	0.141
< 48>	5	0.041	-0.465	123.683	-379.313	461.937	-295.732	108.085	-21.910	4.273	-0.599
< 50>	6	-0.008	0.094	-25.113	125.063	-295.732	389.320	-280.955	104.921	-20.460	2.870
< 52>	7	0.002	-0.019	5.099	-25.391	108.085	-280.955	386.156	-279.504	100.648	-14.120
< 54>	8	-0.000	0.004	-1.034	5.147	-21.910	104.921	-279.504	381.883	-259.045	69.538
< 56>	9	0.000	-0.001	0.202	-1.004	4.273	-20.460	100.648	-259.045	281.235	-105.848
< 58>	10	-0.000	0.000	-0.028	0.141	-0.599	2.870	-14.120	69.538	-105.848	48.046

Column # 2 and row #2 in the above influence coefficient matrix represent the reaction load variation at all system bearings as the offset at the intermediate bearing changes by 0.1 mm.

FIGURE 26
Bearing Reactions for Design Offset – without Forward Stern Tube Bearing



Red circle and green arrow on Section 3, Figure 26, chart on the left, points to location where forward stern tube bearing is removed.

The influence coefficient matrix Section 3, Table 2 indicates that an intermediate bearing offset increase by *0.1 mm* (Section 3, Figure 25, chart on the right, blue arrows), results in a *–18.3 kN* reaction change at aft most main engine bearing. Raising the intermediate bearing offset by *0.5 mm* reduces the reaction at the aft most main engine bearing by five times as much, to *–18.3 kN x 5 = –91.5 kN*.

Section 3, Figure 25, yellow arrow, chart on the left, shows the calculated load of the aft most main engine bearing to be *80.3 kN*. It is clear then that a *0.5 mm* offset increase on the intermediate bearing, producing a load change greater than the calculated load of *80.3 kN*, will completely unload that bearing. This unloading is indicated by the sudden change in gradient and flattening out of the bearing's loading line, represented by the purple line in Section 3, Figure 24.

With no forward stern tube bearing in between, the ratio between the intermediate bearing offset and the aft stern tube bearing misalignment is a constant number. This proportional relationship illustrates that the aft stern tube bearing is directly impacted by bearing offset changes. Some of these may be undesirable offset changes resulting from the effects of hull deflections, temperature changes and production errors during installation and alignment work.

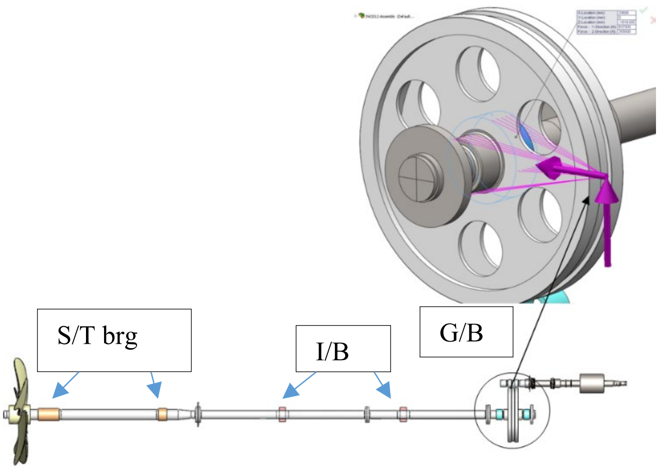
The surveyor needs to be aware of the impact of intermediate bearing offset adjustments, in particular for installations with no forward stern tube bearing.

10 Gear Misalignment

When propulsion gear units are installed, the builder conducts gear tooth contact inspections after completing offset adjustments and before chocking the gearbox to verify that the shafting alignment does not adversely affect gear and pinion misalignment.

The propulsion gearbox consists of the main gear wheel and shaft supported by two bearings. The thrust bearing is normally incorporated in the gearbox. Gearbox bearings are a sensitive part of the propulsion system because of their proximity to each other.

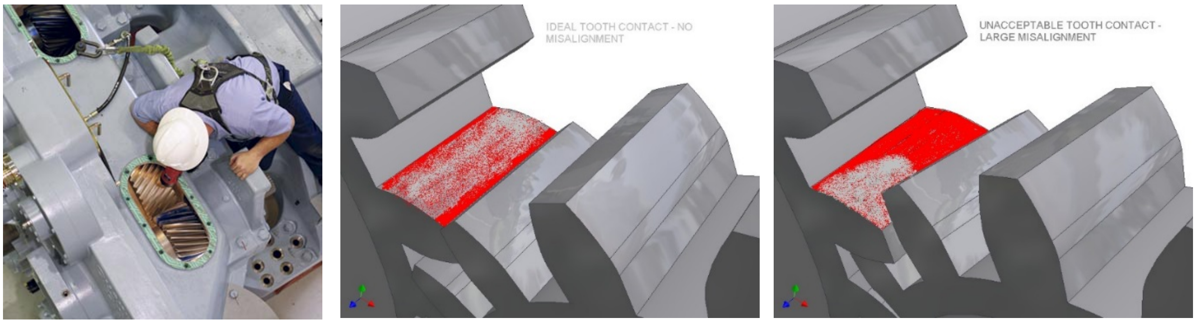
FIGURE 27
Gear Alignment



Propulsion shafting alignment contributes to gear shaft bending and affects the misalignment condition between the main gear and the pinion. ABS Rules require that the calculated relative misalignment between the gear and pinion be less than *0.1 mrad*.

It is difficult to verify the misalignment angle by direct measurement.

FIGURE 28
Gear Alignment



Gear contact evaluation with dye

Case A) Good contact (over 90%) - No misalignment

Case B) Unacceptable contact – Large misalignment

Gearbox misalignment can be verified by coating the pinion teeth with a contrasting dye or copper (Section 3, Figure 28). The gears are turned over a full revolution by engaging the main engine turning gear.

An acceptable misalignment shows visible markings across 90% or more of the face width of the gear teeth. When gear mesh misalignment shows contact impression to one side of the teeth, adjustments to bearing offsets must be made to correct the gearbox alignment.

The method and limitations of gearbox adjustment are incorporated in the shaft alignment calculation.

ABS Rules require a surveyor to attend and record gear tooth contact for propulsion gear units larger than 1120 kW, or 1500 HP at sea trials. On client request surveyor’s attendance can be arranged to also witnesses gear tooth contact between pinion and gear for units smaller than 1120 kW, or 1500 HP.

11 Main Engine Chocking

Main engine chocking is carried out with the vessel in light afloat condition. Main engine manufacturers generally do not consent to chocking a main engine in drydock unless the shipyard has proven prior experience.

ABS Rules do not require the surveyor to witness main engine chocking. However, on client request the surveyor attendance to monitor the procedure can arranged as a part of the shaft alignment process.

12 Quayside Trials

Once the propulsion train has been brought into operational condition, main engine quayside trials, also referred to as dockside trials or mooring trials, are conducted.

Propulsion systems with new bearings are susceptible to damage. The surveyor should keep this in mind when attending main engine quayside trials. The trials should be conducted with great care and limited to testing the main engine safety devices, remote control system, and main engine controls.

13 Run-in Procedure

ABS Rules require that a surveyor witness a stern bearing run-in procedure for all installations with no forward stern tube bearing and for installations with a double-sloped aft stern tube bearing.

The run-in procedure needs to be conducted as early as possible, before starting full operational sea trials. The procedure progressively exposes the stern bearing to an increased load and assists with bedding in the tail shaft with the stern tube bearing in a controlled manner to create proper contact with the bearing bottom. It prepares the stern tube bearing to endure varying service loads without sustaining damage.

Before commencing the run-in procedure, the vessel is brought to deeper waters in open sea, anchored, and ballasted to the approved sea trial draft. If the vessel proceeds from the shipyard for sea trials in a light ballast condition with a partially immersed propeller, then low engine RPM and minimum rudder angle movements are recommended.

A typical run-in procedure is described below:

	Vessel speed	*Rudder angle*	*Direction*
1	Slow	5°	Port
2	Slow	5°	Starboard
3	Slow	15°	Port
4	Slow	15°	Starboard
5	Slow	25°	Port
6	Slow	25°	Starboard
7	Slow	35°	Port
8	Slow	35°	Starboard

	Vessel speed	*Rudder angle*	*Direction*
9	Half	5°	Port
10	Half	5°	Starboard
11	Half	15°	Port
12	Half	15°	Starboard
13	Half	25°	Port
14	Half	25°	Starboard
15	Half	35°	Port
16	Half	35°	Starboard
17	Full	5°	Port
18	Full	5°	Starboard
19	Full	15°	Port
20	Full	15°	Starboard
21	Full	25°	Port
22	Full	25°	Starboard
23	Full	35°	Port
24	Full	35°	Starboard
25	Nav. Full	5°	Port
26	Nav. Full	5°	Starboard
27	Nav. Full	15°	Port
28	Nav. Full	15°	Starboard
29	Nav. Full	25°	Port
30	Nav. Full	25°	Starboard
31	Nav. Full	35°	Port
32	Nav. Full	35°	Starboard

Although not required, ABS suggests conducting the procedure for every installation.

Rudder movements are critical to the in-service load condition of the aft stern tube bearing. This load-adding effect is further emphasized with increasing main engine RPM. The load on the stern tube bearing is gradually raised by widening the rudder angle and increasing the vessel speed. As the rudder is turned to port or starboard, the load on the stern tube bearing is either reduced or increased.

Example:

The vessel proceeds at a slow speed on a straight ahead or *0-degree* course. After *5 minutes*, the rudder is moved *5 degrees* to one side and this course is held for *5 minutes*. The rudder goes back to *0 degrees* for *2 minutes* and then to starboard for the next *5 minutes*. The rudder angle is gradually increased until a maximum of *25 degrees* is reached. The sequence is repeated for half speed and navigation full speeds, each lasting approximately *40 minutes* to *1 hour*.

Stern tube bearing and lube oil temperatures are monitored throughout the run-in procedure. The surveyor must pay particular attention to the stern tube bearing temperature, or the temperature rate change, when the vessel makes turns at increasing speed.

If the aft stern tube bearing temperature rises more than *5°C per minute*, or approaches the temperature alarm level, the engine RPM is to be immediately reduced to slow and the rudder angle is returned to *0 degrees*, until the bearing temperature drops back to an acceptable level.

If a bearing high temperature alarm initiates, the surveyor requests the builder to examine the stern tube lube oil filters. The run-in procedure may be resumed if no bearing contaminants are noted in the stern tube lube oil filters and the aft stern tube bearing temperature has returned to an acceptable level.

14 Sea Trial

Sea trial testing is the final stage of the shaft alignment process. All shafting and components are installed, and the functionality of the whole system can be tested and confirmed.

In addition to measurements conducted during drydock and in the lightship condition, ABS Rules require confirmatory bearing reaction measurements for specific conditions. The engine manufacturer finally validates engine alignment by taking crankshaft deflection measurements of all cylinder units before sea trials. When a propulsion gearbox is installed, surveyors witness gear tooth contact confirmation.

The surveyor should review the vessel's pre-sea trial transit draft condition and the corresponding amount of propeller immersion. In addition, the surveyor also confirms the operational functionality of shaft bearing temperature, stern tube lube oil, and cooling sea water sensors.

14.1 Confirmatory Crankshaft Deflection Measurements

The surveyor should request that the shipyard provide main engine crankshaft deflection tolerances for the hot condition. The surveyor verifies the vessel's ballast condition for the sea trial draft or other service condition, as determined acceptable by ABS.

14.2 Confirmatory Bearing Reaction Measurements

Confirmatory bearing reaction measurements could in some cases indicate bearing offset adjustments may be needed to correct bearing load distribution or to improve crankshaft deflections. The shipyard conducts an assessment to determine the severity of the problem before taking corrective action.

14.3 Corrective Actions

In most situations, the shipyard can perform an intermediate bearing offset adjustment. However, such adjustments are not permitted on systems with no forward stern tube bearing.

In a worst-case situation, un-chocking the main engine and re-aligning the system is the only alternative to performing offset corrections to improve bearing load distribution.

Such drastic corrective measures must be completed after sea trials, when the vessel has returned to the shipyard.

14.4 ABS Rule Requirements

Shaft alignment designs that do not consider hull deflections rely heavily on the builder's experience with similar vessels and similar propulsion system designs.

If vessels have shaft alignment designs that do not consider hull deflections, ABS Rules require confirmatory bearing reaction measurements for at least one additional service draft condition, such as full ballast draft or fully laden draft, provided it is included in the vessel's loading manual. The measured bearing reactions may not exceed *80%* of the manufacturer's maximum allowable limit for any draft condition.

14.5 ABS Rule Requirements for First Vessel in Series

For the first vessel of a series or for a vessel that is not part of a series, ABS Rules require taking confirmatory bearing reaction measurements at the drydock or lightship condition and in a selected service draft condition, as outlined in the approved calculation of shaft alignment document.

Where hull deflections are accounted for in the shaft alignment analysis, the measured bearing reactions at the drydock or lightship condition must be within *±20%* of the calculated value and may not exceed *80%* of the manufacturer's maximum allowable load limit to be acceptable.

When measured bearing reactions at the drydock or lightship condition are found outside *±20%* of the corresponding calculated values, the offset adjustment of intermediate bearings and of the main engine may be considered as possible corrective action. In addition, the builder is to submit to ABS a revised shaft alignment calculation, which is to be in compliance with Rule requirements.

When builder cannot comply with *±20%* margin, the alignment acceptance criteria for designs where hull deflections are not accounted for will apply.

14.6 ABS Rule Requirements for Subsequent Vessels in Series

On subsequent same-series vessels, and assuming the shaft alignment design accounts for hull deflections, bearing reaction measurements at the drydock or light afloat condition become the benchmark to determine which ABS Rule requirements apply.

When the measured values for that condition fall within ±20% of the calculated values, ABS Rules do not require additional confirmatory bearing reaction measurements at a selected service draft during sea trials.

However, when bearing reactions at the drydock or light afloat condition are found outside ±20% of the corresponding calculated value, the ABS Rule requirements for the first vessel in a series must be applied.

Note however that additional bearing reaction measurements for at least one selected service draft condition are always required for shaft alignment designs that do not account for hull deflection effects.

14.7 Bearing Reaction Loads Acceptance

In addition to the above, this paragraph addresses alignment condition acceptance from the viewpoint of bearing reaction verification, whether or not hull deflections accounted for in the alignment calculation.

In general, the alignment reaction load acceptance criteria are based on whether:

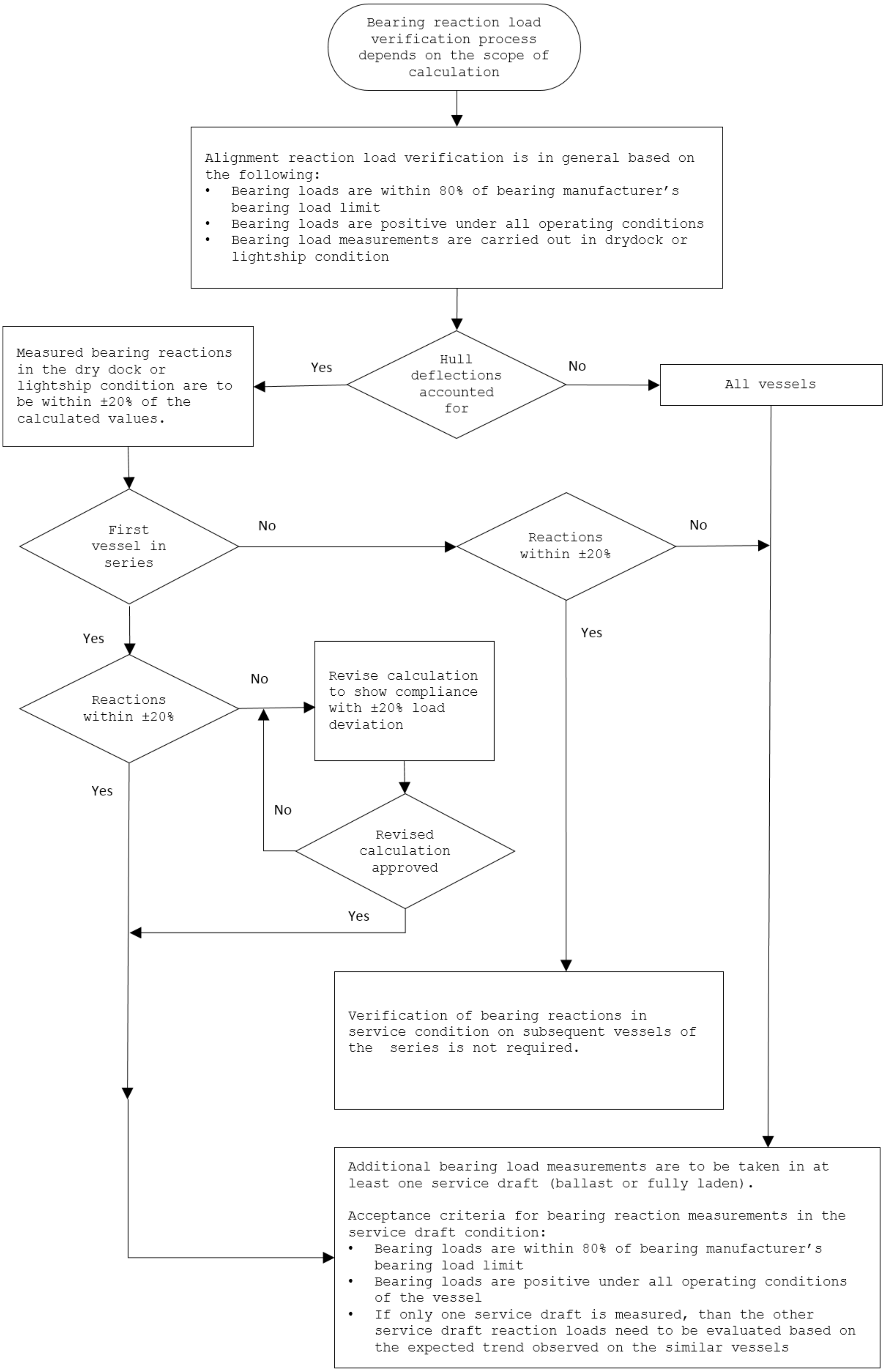
- Bearing loads may not exceed *80%* of the bearing manufacturer's maximum allowable limit, and
- Bearing loads are positive under all operating conditions of the vessel

As long as these criteria are proven satisfactory through a suitable verification method (e.g., jack-up), the shaft alignment is considered acceptable.

Note: The misalignment condition in bearings and reduction gears, as well as the crankshaft deflections need to be confirmed acceptable for final acceptance of the shaft alignment.

When a shaft alignment calculation is conducted employing advanced techniques that can consider hull deflections, the alignment verification procedure may be relaxed for subsequent vessels in a series.

The flow chart summarizes the reaction measurement procedure for cases where hull deflections are and are not accounted for.


Bearing reaction load verification process depends on the scope of calculation
Alignment reaction load verification is in general based on the following:
• Bearing loads are within 80% of bearing manufacturer's bearing load limit
• Bearing loads are positive under all operating conditions
• Bearing load measurements are carried out in drydock or lightship condition
Hull deflections accounted for
Yes
No
Measured bearing reactions in the dry dock or lightship condition are to be within ±20% of the calculated values.
All vessels
First vessel in series
No
Reactions within ±20%
No
Yes
Yes
Reactions within ±20%
No
Revise calculation to show compliance with ±20% load deviation
Yes
No
Revised calculation approved
Yes
Verification of bearing reactions in service condition on subsequent vessels of the series is not required.
Additional bearing load measurements are to be taken in at least one service draft (ballast or fully laden).
Acceptance criteria for bearing reaction measurements in the service draft condition:
• Bearing loads are within 80% of bearing manufacturer's bearing load limit
• Bearing loads are positive under all operating conditions of the vessel
• If only one service draft is measured, than the other service draft reaction loads need to be evaluated based on the expected trend observed on the similar vessels

14.8 Partially Immersed Propeller

When the vessel is operated with a partially immersed propeller, the propeller forces exerted on the aft stern tube bearing are significantly higher than when propeller is fully immersed. Turning at higher vessel speeds is particularly critical. To prevent bearing damage, it is advisable to restrict maneuvering and reduce the service speed as much as possible for safe navigation.

More about partially immersed propeller can be found in 6/1.5.1.

14.9 Monitoring During the Sea Trials

The aft stern tube bearing temperature must be closely monitored at sea trials during any test requiring maneuvering at higher rudder angles, such as the zig-zag maneuver, steering gear trials and turning circle. Starboard maneuvers on systems with a clockwise-rotating propeller, or port turns for counterclockwise propellers, should be given particular concern. During these maneuvers, the surveyor should pay particular attention to the stern tube and intermediate bearing temperatures.

With the increased usage of automation and monitoring systems onboard, it is possible to plot and print trends for specific systems of interest, such as bearing temperatures. Customarily, it is designated shipyard personnel who record temperatures and other parameters during the course of the sea trials and commissioning programs.

The surveyor should verify sea trial records applicable to class items to confirm compliance with ABS Rule requirements.

SECTION 4 Alignment Measurements and Monitoring

1 General

Propulsion shaft alignment is defined as a static condition, where bearing loads, bearing offsets, and misalignment between the shaft and the bearing are observed at the bearings supporting the propulsion shafts. Acceptability of alignment is verified by confirmation that the following minimum set of parameters are acceptable:

- Bearing reactions
- Bearing vertical offset
- Misalignment angles
- Crankshaft's web deflections
- Gear misalignment

Previous Sections have addressed the above listed parameters from the viewpoint of alignment review and design, and the production process. This Section approaches these same topics from the perspective of alignment condition confirmation during and after construction of the vessel, along with long term shaft alignment condition monitoring.

Long-term shaft alignment monitoring, in particular monitoring of the aft stern tube bearing performance, is an optional precautionary measure that contributes to vessel's safety. Moreover, with stern tube bearing monitoring, bearing maintenance can be established based on the bearing condition rather than on a fixed scheduled. This Section addresses some of the advantages of stern tube bearing monitoring.

2 Bearing Reaction Measurements

The industry employs two methods for the verification of bearing reactions:

- The jack-up method, utilizing hydraulic jacks
- Strain gauges in conjunction with the jack-up method

2.1 Jack-up Method

Due to its simplicity, the jack-up method is the most widely applied method in the industry. Hydraulic jacks are placed in close proximity to the bearing whose reaction is verified. A compression load cell can be utilized in combination with the hydraulic jacks to improve measurement accuracy.

The advantages of the jack-up method are:

- It is the only method that provides reaction load directly
- It uses simple and ready to use measuring equipment
- Ease of installation
- Low cost

The disadvantages of the jack-up method are:

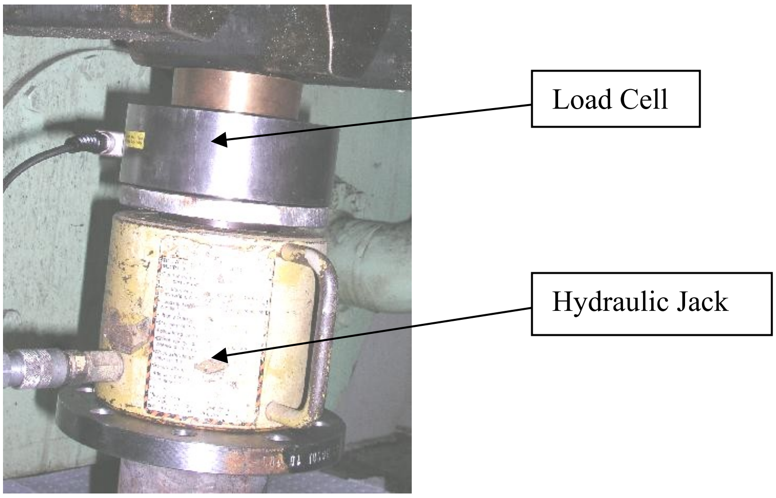
- Measurement results in wide hysteresis if load cell is not used.
- Installation inaccuracies due to:
  - Misalignment of the hydraulic jack
  - Misalignment of the dial gauge
- Correction factors need to be applied to obtain the actual bearing load

2.1.1 Equipment

The jack-up method requires the following equipment:

- Hydraulic jack
- Dial gauge with magnetic stand
- Load cell (optional)

FIGURE 1
Hydraulic Jack with Load Cell

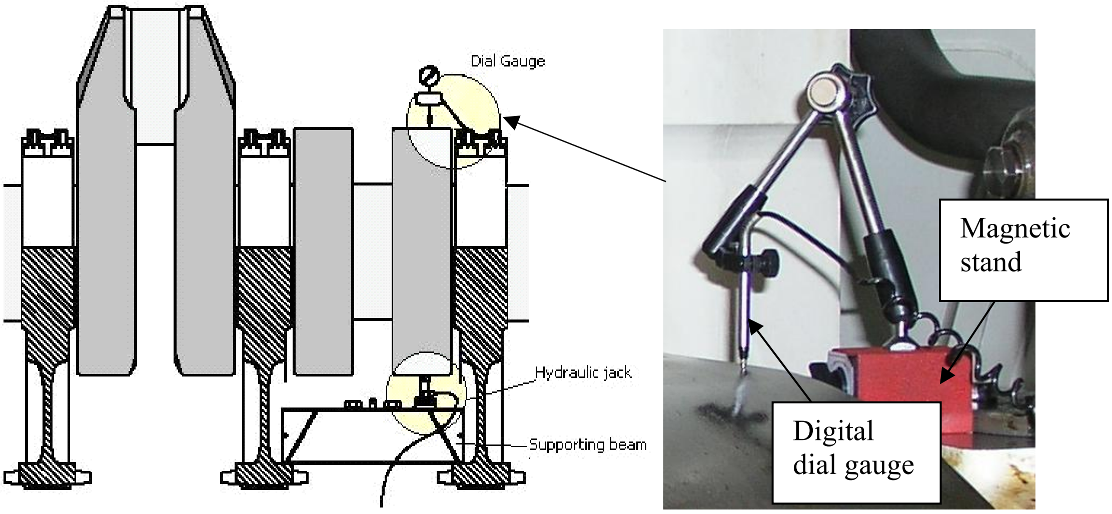


The hydraulic jack should be located as close to the bearing as possible and seated on a stiff support, such as an internal stiffener of the double bottom structure underneath the shaft.

When the main engine aft most bearing reaction is measured, attention should be paid that the turning gear is declutched from the flywheel. An engaged turning gear introduces a significant side load which results in incorrect bearing load interpretation.

A load redistribution problem may also be caused by turning gear lock-up, whereby some portion of the reaction is locked in at the contact point between gears.

FIGURE 2
Jack-up Measurement of the Bearing Reactions Inside Diesel Engine



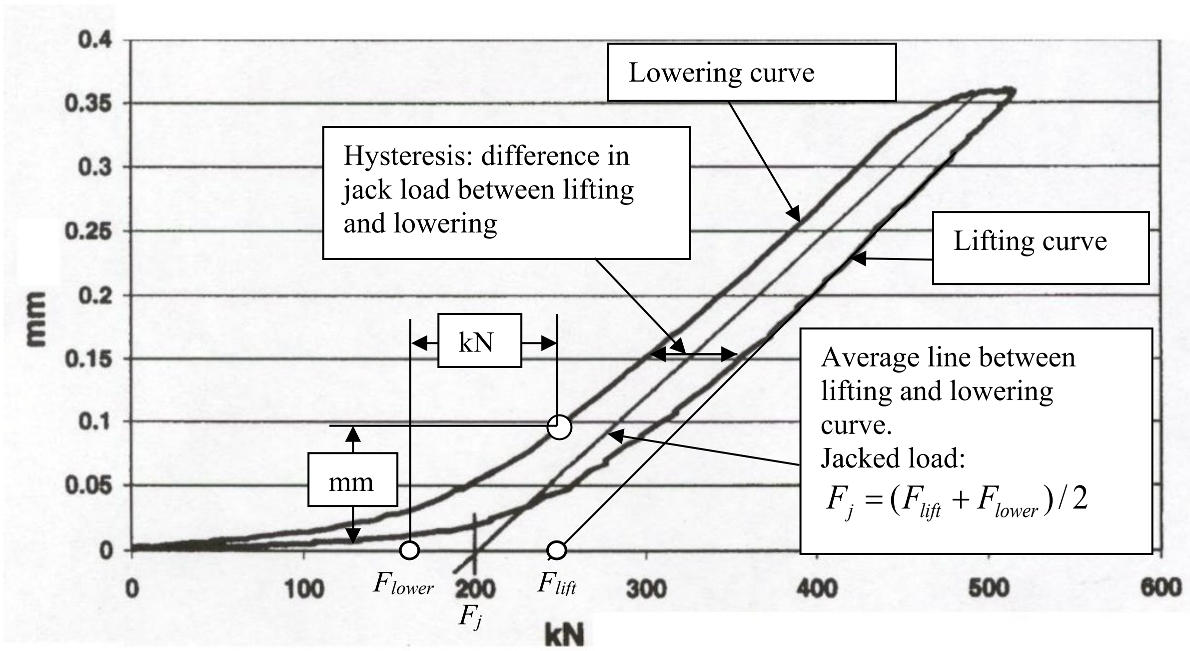
The dial gauge should be affixed to a structure that is not too flexible, to avoid relative movement and erroneous readings.

Shaft lifting and lowering during jacking is controlled by reading the dial gauge at selected intervals and recording the corresponding jacked load.

By following this procedure, a smoother jacking curve may be generated, resulting in a more accurate bearing reaction calculation.

2.1.2 Procedure

FIGURE 3
Measured Jack-up Curve

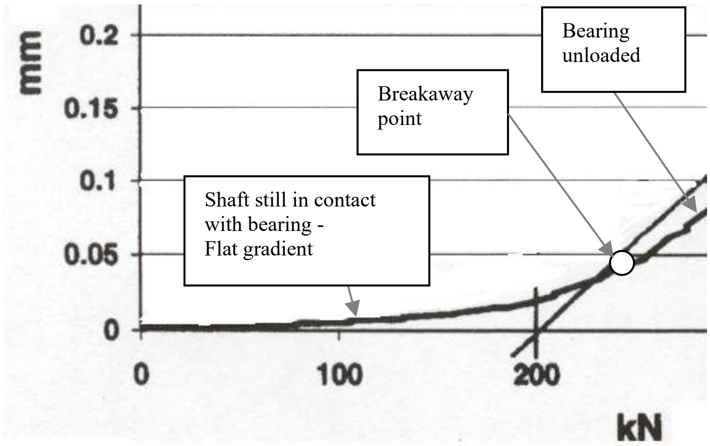


The objective of the jack-up procedure is to generate a lifting and lowering curve that can be used to find the jacked load and calculate the bearing reaction and the gradient of the average line between the lifting and lowering curve.

2.1.2(a) Jack-up process – Lifting

The following behavior is experienced during jacking-up of the shaft:

FIGURE 4
Jack-up Lifting Curve



Initially, as the shaft lifts, the bearing continues to carry some of the load while the hydraulic jack takes over. (Section 4, Figure 4). This stage is represented by a relatively flat curve.

The lifting curve gradient increases slowly until the load is fully transferred from the bearing onto the hydraulic jack.

As the lifting continues, the lifting curve becomes much steeper with a relatively constant gradient.

The breakaway point is where the jack has fully lifted the shaft off the bearing and the load is completely transferred onto the jack.

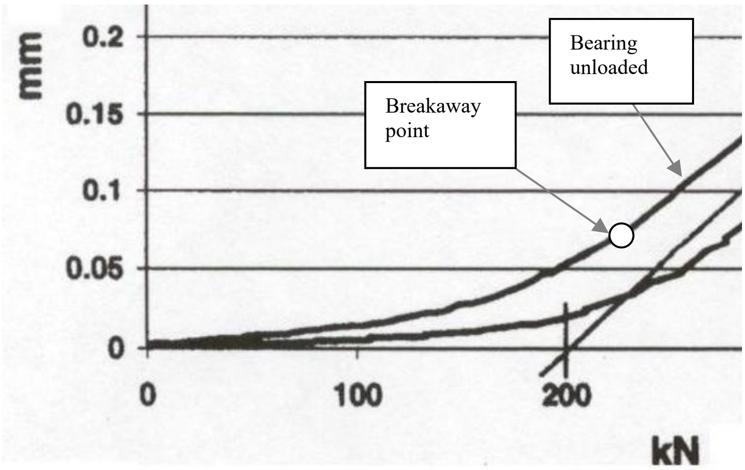
Jacking stops when a sufficient number of points are recorded to define the gradient of the jacking curve. The process then reverses, and the shaft is lowered back down as pressure is released from the hydraulic jack.

2.1.2(b) Jack-up process – Lowering

At the start of unloading of the hydraulic jack, the lowering is not the same as that for lifting. The reason for this difference is due to friction in the hydraulic jack and internal resistance in the shaft. The lowering shaft curve is similar in shape to the lifting curve and the same number of points are recorded as during lifting.

The following behavior is experienced during lowering of the shaft back onto the bearing.

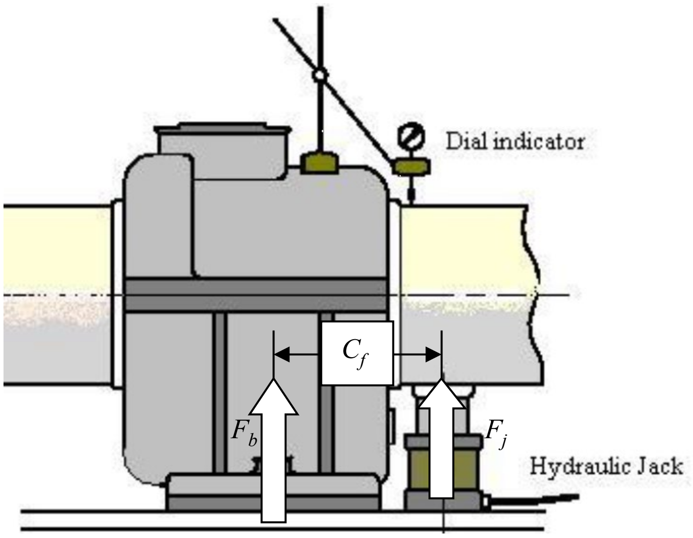
FIGURE 5
Jack-up Lowering Curve



The load reduces almost linearly as the jack is lowered and the gradient of the lowering curve remains constant until the point is reached where the bearing starts picking up the load again (Section 4, Figure 5). This is the breakaway point, from where the gradient of the lowering curve changes.

2.1.3 Calculation of Bearing Reaction

FIGURE 6
Reaction Measurement at Intermediate Shaft Bearing



The hydraulic jack records the load by direct measurement at the location of the jack. The jack-measured load will not be exactly the same as the bearing load, because the jack is placed close to the bearing and not in its center (Section 4, Figure 6).

The actual bearing reaction (F_b) is correlated to jacked load (F_j) by applying a correction factor (C_f):

$$F_b = C_f \cdot F_j$$

The shaft alignment analysis is to provide the jack-up correction factor for each jacking location.

2.1.4 Confirmation by Using the Gradient Line

Besides comparison with calculated reactions, the jacked load can be additionally verified by examining the jack-up curve gradient as shown in Section 4, Figure 3. This is the slope of the lifting and lowering curve and is normally expressed as change in lifting or lowering force over a change in vertical offset.

For the purpose of jack-up measurements, the average line between the lifting and lowering curve from Section 4, Figure 3 is used to calculate the gradient:

$$\text{Gradient} = \frac{\text{Force Change}}{\text{Displacement Change}} \; \frac{\text{kN}}{\text{mm}}$$

Lifting and lowering gradients are commonly called "influence coefficients". In the shaft alignment calculation these are presented in the form of a matrix that correlates bearing vertical offset with bearing load.

The influence coefficient matrix can be used to verify jack-up procedure accuracy because it provides information about the gradient, namely the bearing reaction change per unit lift applied to a particular bearing.

The influence coefficient matrix in Section 4, Table 1 below is generated by ABS shaft alignment software for a propulsion system with five bearings.

TABLE 1
Sample Influence Coefficient Matrix

RELATIVE BEARING REACTIONS [kN] -> R[0.1-offset]-R[0-0ffset]
Due to 0.1[mm] OFFSET relative to the ZERO bearing Offset

Node	Supp.	< 7> 1	< 14> 2	< 27> 3	< 41> 4	< 46> 5
< 7>	1	4.289	-7.896	4.768	-3.222	2.120
< 14>	2	-7.896	15.425	-11.112	9.951	-6.546
< 27>	3	4.768	-11.112	13.421	-25.421	18.858
< 41>	4	-3.222	9.951	-25.421	166.594	-250.824
< 46>	5	2.120	-6.546	18.858	-250.824	511.825

Each column provides bearing reactions corresponding to a bearing offset increase of 0.1[mm] at a particular bearing. The red encircled value of *13.421 kN* in Section 4, Table 1 represents the gradient of the jack-up curve when bearing *#3* is raised for *0.1 mm*. The other values in the same column represent the bearing reaction change happening at the other four bearings in the system as a result thereof.

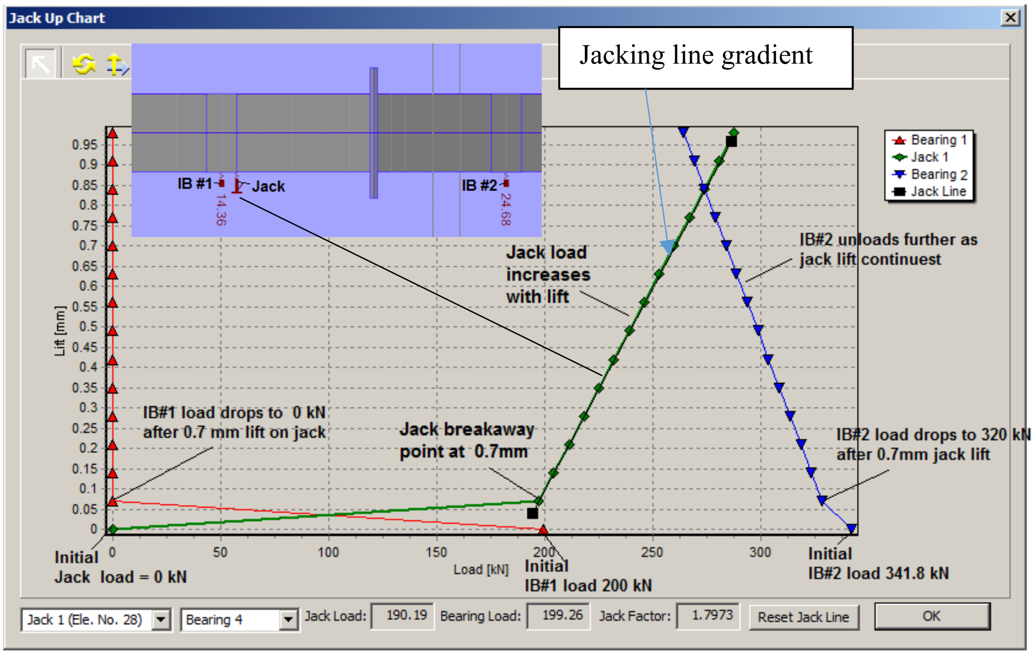
The gradient calculated from the jack-up curve is easily compared with analytical values in the matrix.

For example, if the reaction of bearing *#3* is measured, the expected jack-up gradient can be found from the influence coefficient matrix (Section 4, Table 1) at the intersection of row *#3* and column *#3*.

The value obtained by reaction measurement should be close to the value calculated from the jack-up curve.

In commercially available software, the gradient values are normally given as a regular output of the shaft alignment analysis for the actual bearing location only. However, ABS software provides gradients for the jack-up locations as well, i.e. at the point of measurement. An example of ABS software simulation of the jacking process is given on Section 4, Figure 7.

FIGURE 7
Calculated Jack-up Curve



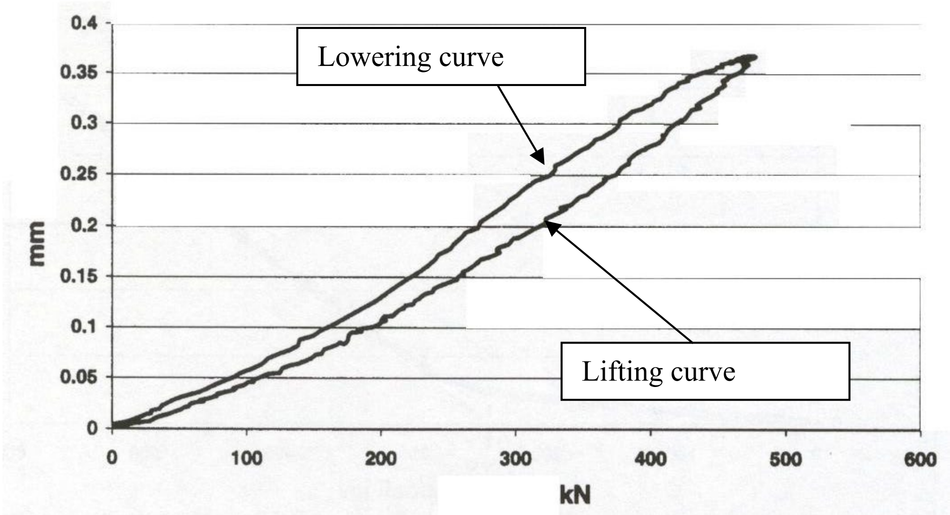
The simulated jack-up line for both the lifting and lowering is identical and represents the average line obtained by measurements on the same bearing. When the simulated gradient line differs substantially from measured average gradient line, the measured jack-up diagram should be reevaluated, the calculation input data checked again and reconfirm that the correct jack location is used in the simulation.

2.1.5 Unloaded Bearing

When the jack-up procedure is performed on an unloaded bearing, the jack immediately picks up the load and there will be no visible transition, i.e. no breakaway point as the jack gradually lifts the shaft. The same trend continues on the downward slope. The appearance of this jack-up curve in Section 4, Figure 8 compared with that in Section 4, Figure 3 is very different and is readily recognizable as that of a bearing carrying no load.

A bearing clearance measurement will confirm that shaft is not sitting on the bush.

FIGURE 8
Jack-up Curve for Unloaded Bearing



2.2 Strain Gauge Method

The advantages of the strain gauge method are:

- It can provide relatively accurate information on the loading condition of bearings that are inaccessible for performing jack-up measurements.
- Once the strain gauges are mounted, measurement can be easily repeated within a short time.
- It can provide data about vertical and horizontal load at the bearings.
- It can provide simultaneous information on more than one bearing load.
- Measured strains and the corresponding bending moments provide useful information on shaft bending curvature that can be further utilized for "reverse calculation" of actual bearing positions.

The disadvantages of the strain gauge method are:

- Equipment installation requires a relatively long time (approximately one hour per measurement point).
- Accuracy of recalculated bearing loads depend on analytical model utilized in reverse calculation of bearing reactions.
- It requires sophisticated and relatively expensive equipment.
- The method is always to be used in conjunction with the jack-up procedure.

The strain gauge technique for shaft bending moment measurement is based on the basic beam theory, where moments are defined per the formula:

$$M = E \cdot W_p \cdot \varepsilon$$

where

E = Young's modulus

ε = strain

W_p = section modulus (for circular shape $= \pi \dfrac{D^3}{32}$)

Strain gauges are firmly bonded on the shaft's surface. Section 4, Figure 9 shows how a pair of uniaxial gauges is installed on the shaft to measure tension in the longitudinal direction.

Bending of the shaft deforms the strain gauges and thus changes their resistance. These changes can be observed on a voltmeter. Accordingly, the strain can be calculated:

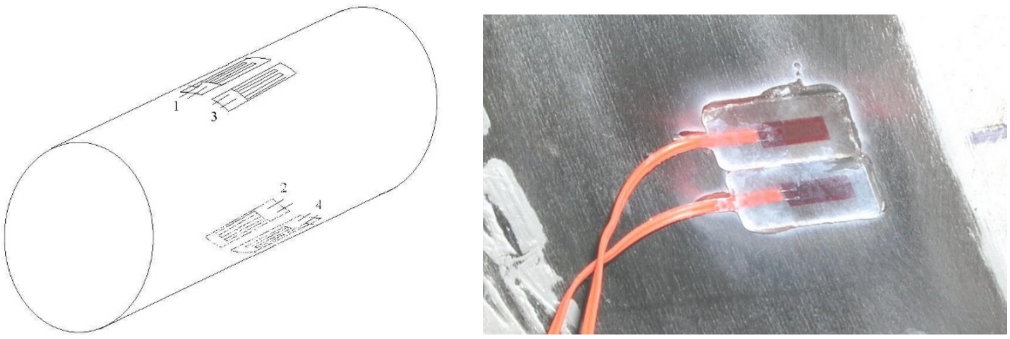
$$\varepsilon = \frac{V_o}{V_{ek}} \cdot \frac{1}{k}$$

$$\varepsilon = \frac{\Delta R}{R} \cdot \frac{1}{k}$$

where

R	=	is bridge resistance, in Ω
ΔR	=	change in bridge resistance, in Ω
k	=	bridge factor (a common value for bridge factor is 2)
V_o	=	measured voltage
V_{ek}	=	excitation voltage

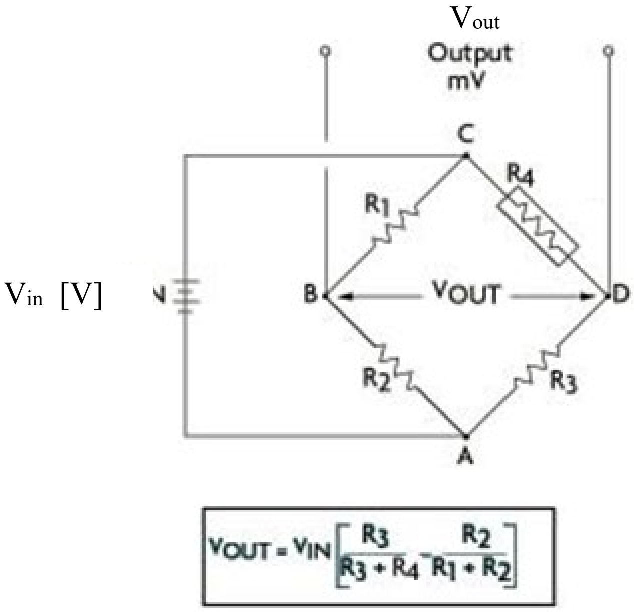
FIGURE 9
Strain Gauge Installation



To increase the precision of the measurements and allow for error correction, more than one strain gauge is to be installed to measure the strain at the same location. Section 4, Figure 9 illustrates how four gauges are installed in two pairs at 180° opposed to each other and connected in a Wheatstone bridge configuration as indicated in Section 4, Figure 10.

When a fixed voltage V_{in} is applied on two opposite corners A and C of the bridge, the output voltage V_{out} changes as the shaft flexes and the strain gauges contract and expand accordingly. The change in voltage is proportional to strain and from this the strain moment can be recalculated based on formulas shown below.

FIGURE 10
Wheatstone Bridge



Voltage and resistance are related as follows:

$$\frac{\Delta R}{R} = \frac{V_{out}}{V_{in}}$$

Combining the above relationships, the shaft bending moment can be determined by applying the following equation:

$$M = E \cdot W_p \cdot \varepsilon = E \cdot W_p \cdot \frac{\Delta R}{R} \cdot \frac{1}{k} = E \cdot W_p \cdot \frac{V_{out}}{V_{in}} \cdot \frac{1}{k}$$

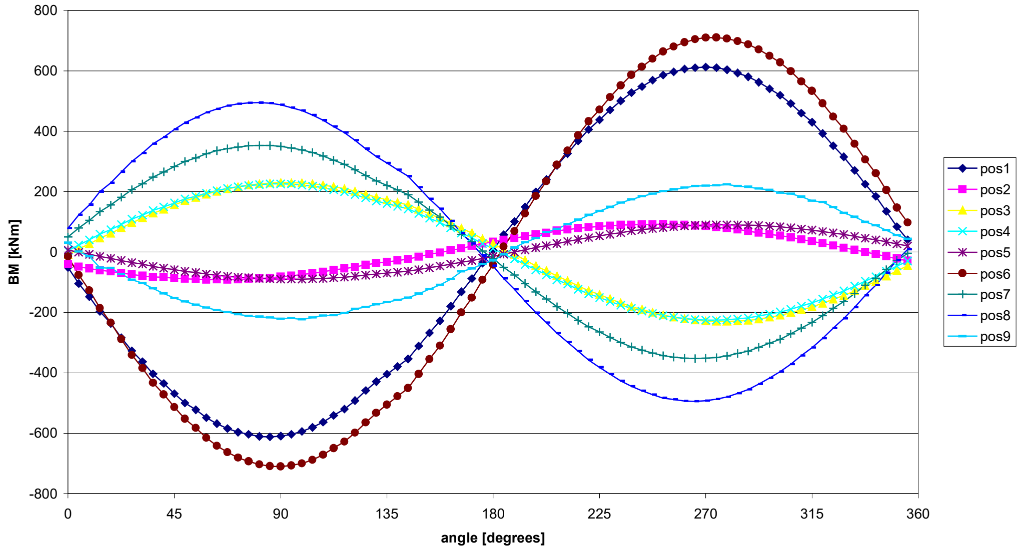
When the shaft is rotated for a full circle, each of the strain gauges is exposed to deformation proportional to two times the bending moment. A simple analysis of continuous full circle measurements will separate vertical and horizontal bending moments.

2.2.1 Example

Bending moment curves taken at nine different locations along the line shaft are shown in Section 4, Figure 11. Each measurement location is placed along the same imaginary line perpendicular to the centerline of the shaft. At each location, four uniaxial gauges are placed in pairs 180° apart.

It may be noted from Section 4, Figure 11 that maximum values of the bending moment curves do not line up at the same phase angle. This indicates a possible existence of the horizontal load acting on the shaft and/or possible gauge misalignment.

FIGURE 11
Bending Moments Measured at Nine Different Locations Along the Shaft Line



2.2.2 Mathematical Representation

Once bending moments are measured, the bearing reactions can be calculated using the reverse engineering approach.

To achieve this, a system of linear equations is established that defines the relationship between reaction loads and bending moments at each measured location.

$$M_{mi} - M_{0i} = K_{i1}\Delta R_1 + K_{i2}\Delta R_2 + K_{i3}\Delta R_3 + \ldots + K_N\Delta R_N, \quad \text{and} \quad i = 1\ldots N_m$$

where

N_m	=	number of bending moment's measurement points
i	=	measurement point number $i = 1\ldots N_m$
M_m	=	measured bending moment
M_0	=	initial bending moment (calculated or measured)
K	=	bending moment's influence factor as calculated
R_m	=	bearing reaction

Using the matrix notation, an expression is then written as:

$$[M_m] - [M_0] = [K] * [\Delta R]$$

If the number of equations is equal to the number of variables, the above system of equations could be solved by applying simple matrix inversion. For cases where number of variables is not the same as number of equations, the solution to the above system can be obtained using a best fit approach such as for example the least square method.

Solving the equation for ΔR:

$$[\Delta R] = [K]^{-1} \cdot ([M_m] - [M_0])$$

In reality, the number of measurement locations could be less than the number of bearings. In that case analytically obtained data must be combined with information obtained by strain gauge measurement.

3 Bearing Vertical Offset Measurements

The bearing vertical position is either set during the bore sighting or during the sag and gap procedure. The final bearing offset tuning and adjustments are performed during the bearing reaction measurements.

After completing the final bearing offset adjustments, either in drydock or immediately after launching the vessel, the intermediate bearings and main engine are to be chocked. It should be noted however that added weight from unfinished construction works, hull deflections and thermal influence can further affect bearing offsets.

The actual bearing offsets after final chocking can only be evaluated by the following methods:

- Optical
- Laser
- Hydraulic jacks
- Strain gauges
- Crankshaft deflections
- Combined method, hydraulic jacks, strain gauges, and crankshaft deflections

Optical and laser: Optical and laser methods are restricted to systems where visual contact can be established. These methods can be applied to crankshafts with hollow journals. The methods are relatively inaccurate and do not provide information on the whole system, but rather on the segment of the propulsion train that is optically accessible.

Hydraulic Jack-up: The jack-up method based on bearing reactions can be used to recalculate the bearing offset by utilizing the reverse engineering approach. This method though, cannot uniquely define shafting curvature and therefore may provide erroneous results. The jack-up method is also restricted by bearing accessibility and for example, the aft stern tube bearing cannot be reached. This problem may be resolved by using the jack-up method in combination with the strain gauge method.

Strain Gauge: The strain gauge method can provide information about the bearings that are normally not accessible for evaluation by other methods and also on the shafting bending curvature. Strain gauges are not easy to apply on the crankshaft and to improve reverse analysis accuracy, the method is combined with jack-up measurements and crankshaft deflection measurements.

Crankshaft Deflections: Crankshaft deflections provide information about crankshaft bending curvature. Crankshaft deflections can be utilized in reverse analysis if the *3D* crankshaft model is available.

Combined Measurements: The combined method is a preferred approach when bearing offset is to be recalculated from other measured parameters. This method utilizes measured data obtained from strain gauges, bearing reactions and crankshaft deflections and uses all of it in recalculating the bearing offsets (see also Subsection 5/3).

3.1 Reverse Shafting Alignment Calculation of the Bearing Offsets

The reverse shafting alignment is an analysis where bearing offsets are recalculated from measured bearing reactions, bending moments and crankshaft deflections. It is an inverse approach to the regular shaft alignment calculation.

Regular shaft alignment calculation solves a system of equations to obtain system loads from prescribed bearing displacements or offsets. The matrix representation used is:

$$\{R\}_{Nx1} = [K]_{NxN}\,\{r\}_{Nx1}$$

In reverse approach the inverse equation is used:

$$\{r\}_{Nx1} = [K]^{-1}_{NxN}\,\{R\}_{Nx1}$$

where

{R} - nodal load vector (moments and forces)

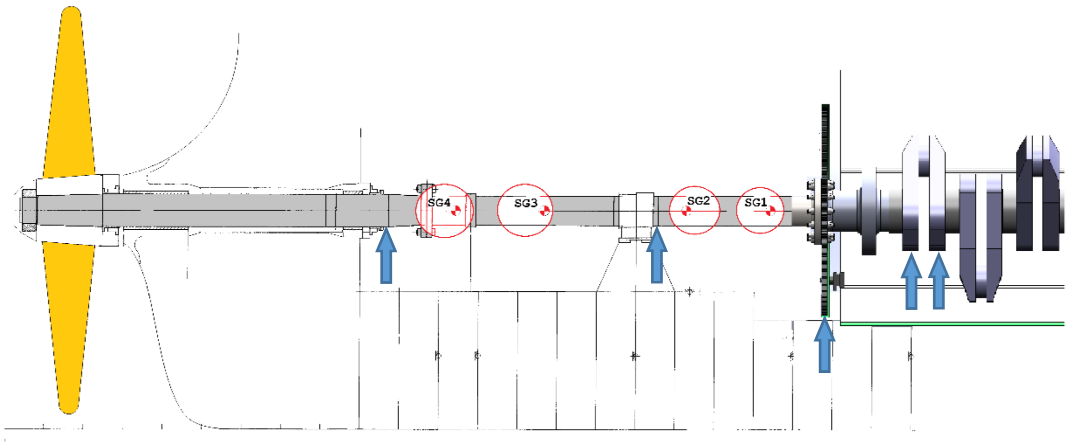
{r} - nodal displacement vector

[K] - system stiffness matrix

Computationally, the reverse approach is no different from the regular alignment calculation if the number of required input variables is equal to the number of equations being solved. However, this is seldom the case with the propulsion shafting alignment where input variables, such as bearing reactions and bending moments, are obtained by measurements. The shortcoming is that not all of the bearings in the system are accessible. Unless their reaction loads are assumed, some of the required variables will be missing, hereby preventing a unique solution.

When measurements are conducted for the purpose of reverse calculation, it is desired to have bending moments measured in addition to bearing reactions. Bending moments are obtained from strain gauge measurements, and as explained before, have the advantage of recording information on the condition of the system simultaneously at several locations. This produces measurement consistency, especially when measurements are repeated. Section 4, Figure 12 shows one such installation.

FIGURE 12
Moments and Bearing Reactions Measurement



In the shipbuilding environment, strain gauge measurements are limited to the shaft line only and bearing reactions are measured only on accessible shaft bearings and on the three aft most main engine bearings. That implies the reverse calculation is confronted with an excess of data collected along the line shaft and a lack of information from the inside the stern tube and the main engine. When a reverse calculation uses all of the measured data, the system is either over-defined (having more variables than required) or under-defined (having less variables than required). Considering this, the best fit approach may be the most appropriate way to conduct a reverse shaft alignment calculation.

Measured parameters are not entirely accurate due to associated measurement errors. It is therefore good practice to assign a confidence factor to each measured parameter and conduct reverse analysis accordingly.

Applying strain gauge measurements for obtaining hull deflections is explained in the Section 5 of these Guidance Notes.

4 Bearing Misalignment Measurements

Bearing misalignment is defined as the angular difference between shaft and bearing centerlines. Ideally, when centerlines match, that angle is zero.

Misalignment can be easily measured with feeler gauges on intermediate shaft bearings, because of their fore and aft accessibility. Misalignment is defined by subtracting the bearing clearance measurements fore and aft (vertical misalignment), and port and starboard side (horizontal misalignment) of the bearing. (Section 4, Figure 13).

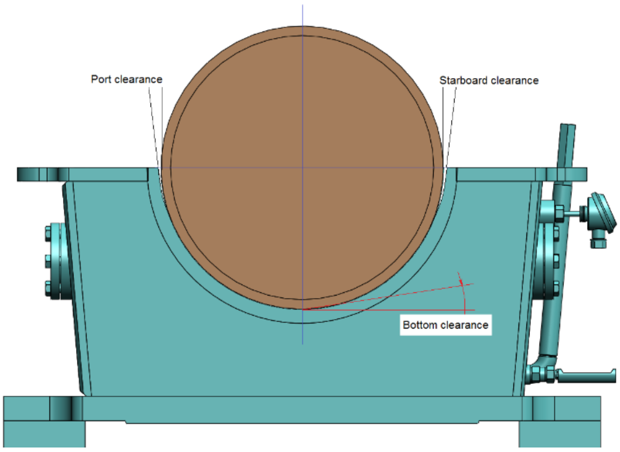
It is advisable to rotate the shaft and conduct the measurement several times within one shaft revolution. This will indicate whether the measured values are the result of shaft run-out or bearing misalignment.

Intermediate bearing misalignment is readily corrected by appropriately fitted chocks.

Epoxy resin chocks are seldom used for intermediate shaft bearings. Unlike steel chocks, epoxy resin chocks cannot be easily adjusted should such be required during bearing reaction verification in one of the service conditions later in the process.

It is a good practice to conduct final intermediate bearing chocking only after bearing reactions are confirmed acceptable on all bearings, and the intermediate bearing misalignment is verified and confirmed acceptable.

FIGURE 13
Intermediate Shaft Bottom Clearance and Runout Measurement



Main engine bearings misalignment verification is conducted in a similar manner as the intermediate bearing.

Misalignment measurement on the stern tube bearings is possible after the propeller shaft is installed and before the propeller is fitted. Here the clearance measurement is used to indirectly evaluate misalignment. The paragraph related to tail shaft bearing clearance measurement provides more detailed information about the procedure.

5 Crankshaft Deflection Measurement

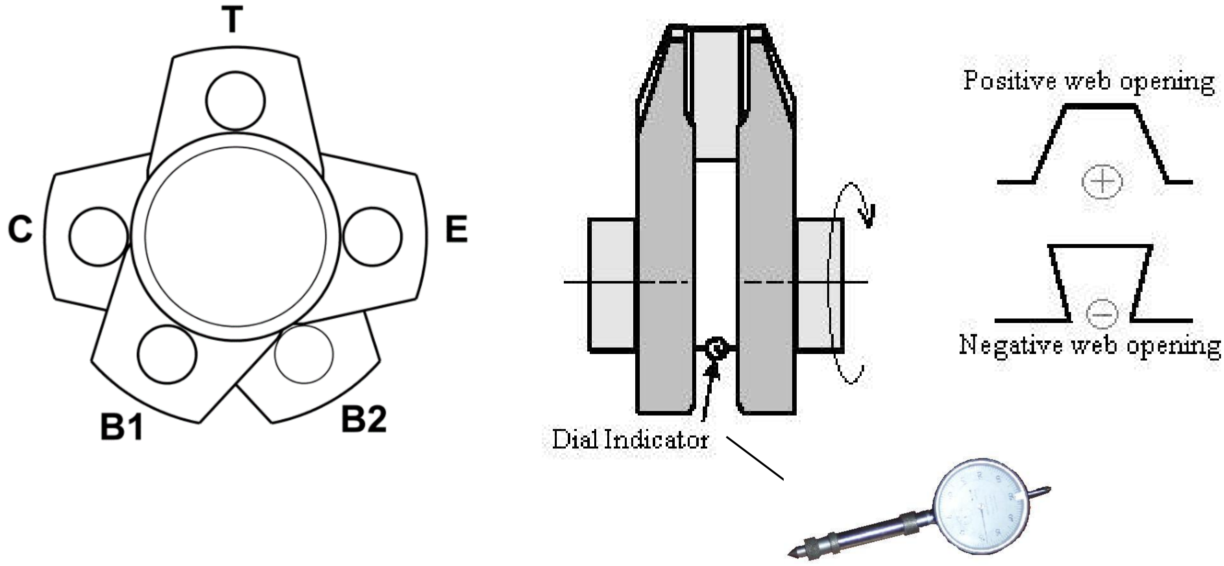
Crankshaft deflections are commonly measured at several stages: with the engine on the test bench, before connecting the crankshaft to the line shaft, after connecting, in the cold and hot condition, and in at least one service draft.

Crankshaft deflection limits and tolerances are defined by the engine manufacturer for each particular engine.

The engine manufacturer supplies an engine tool set with a deflection gauge and proposes a sign convention for measuring. For some engine manufacturers, opening the webs is represented by the plus sign, and closing the webs is represented by the minus sign.

Crankshaft deflection measurement is conducted with a dial indicator being placed at a predefined location between crank webs. The crankshaft is then rotated using the turning gear and the readings are taken at the prescribed angular locations. It is common practice to record deflections for each crank throw at five different crankshaft angles, as indicated in Section 4, Figure 14.

FIGURE 14
Crankshaft Deflection Measurements



Readings are taken at prescribed angular locations of the crank webs. A different location naming convention may be used depending on the engine manufacturer. The first measurement is where the piston is at its lowest point, also known as its bottom dead center, or *BDC* point. The deflection gauge is inserted at *B1* with the crank rotated just enough to install the deflection gauge without interference from the crank pin. On the way down, the last measurement is taken at *B2*, just before the deflection gauge would touch the crank pin again at the other side. Measurements taken in between are located at the camshaft side (*M*), top (*T*) and exhaust side (*E*).

Crankshaft deflection measurement procedures may differ depending on the engine manufacturer and engine size.

6 Sag and Gap Measurement

Although the sag and gap procedure is not endorsed as an accurate method, it still has its purpose as it may indicate possible problems in the preassembly stage of propulsion shafting setup.

ABS Rules state that the sag and gap values are to be simultaneously set to appropriate values with assistance of the temporary supports at all open flanges, until values are brought within acceptable tolerances of ±0.1 mm from the corresponding calculated values.

Advantages of the sag and gap method are that it:

- Uses simple measuring equipment such as dial gauges and feeler gauges.
- Brings the bearings and the engine/gearbox offsets close to the design values thus saving time required for the jack-up reaction verification.

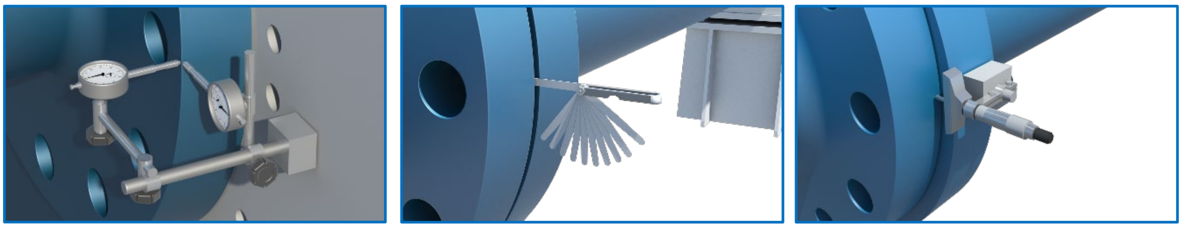
The disadvantages of the sag and gap method are:

- It cannot verify condition of the assembled system.
- Its accuracy is limited because of the large diameters of flanges and shafts (over 300 mm), and very small sag and gap values (measured in just few millimeters or fractions of millimeters).

The measurement tolerances will depend on the following:

- Dial gauge location
- Condition of the flanges where the gauges are placed
- Environmental condition (temperature, vibrations, et al.).

FIGURE 15
Sag and Gap Measurement Tools



Dial Gauge Feeler Gauge Depth Micrometer

Common measuring tools for sag and gap are the dial gauge, the feeler gauge and the depth micrometer. Dial gauge accuracy is up to *0.001[mm]*, and feeler gauge accuracy cannot be obtained greater than *0.05 [mm]*. Analog depth micrometers reach an accuracy up to *0.01[mm]*, and digital versions can go up to *0.001[mm]*.

Measurements are taken as illustrated in Section 4, Figure 15. It is often difficult to measure precisely with feeler gauges, and gap measurement errors often exceed *0.1 [mm]*. Even with the much more precise dial gauges, measurements may be hampered by inaccurate positioning of the gauges, difficulties in leveling the equipment, machining inaccuracies of the flanges, cleanliness of the surface, etc. Skilled workers are required to perform sag and gap measurements.

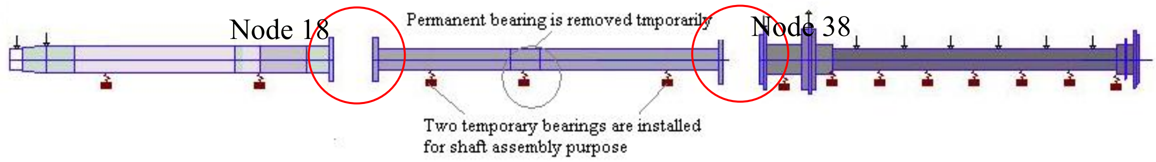
Example:

The table below is a standard sag and gap calculation output of the ABS shaft alignment software. It provides all the necessary data to verify the preassembly condition of the shafting.

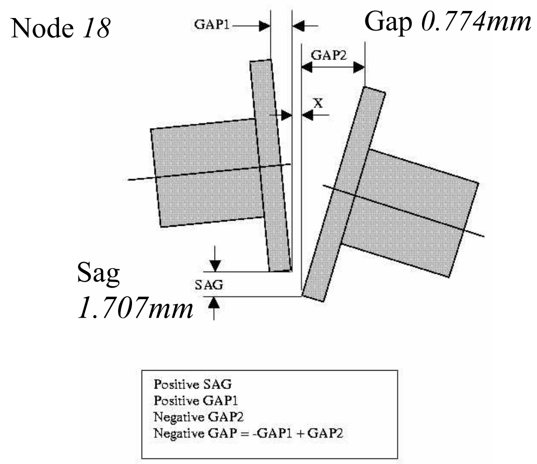
SAG and GAP ALIGNMENT DATA

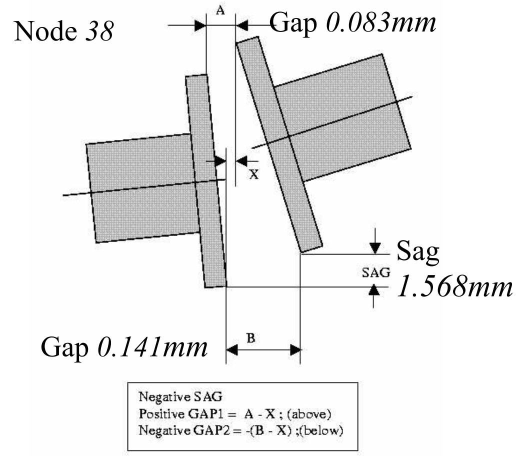
Node No	Between Shafts Shaft No	Flange Dia [m]	Shaft No	Flange Dia [m]	SAG [mm]	Gap1 [mm]	Gap2 [mm]	GAP [mm] Gap1+Gap2
18	1	1.1100	2	1.1100	1.7078	0.6997	-0.0745	Above flanges -0.7742
38	2	1.2600	3	1.2600	-1.5681	Above 0.0837	Below 0.1414	N/A

FIGURE 16
Preassembly Shafting Setup – for Sag and Gap Measurement



Node *18*: Propeller shaft and intermediate shaft flange positions. Calculated sag value: *1.707 mm.* Calculated gap: -*0.774 mm* (negative value indicates that gap is to be measured at the top of the flanges)	Node *38*: Intermediate shaft and crankshaft flange positions Calculated sag value: -*1.568mm* (negative value indicates that left flange is lower than right flange) Calculated gap: *0.083 mm* at top and *0.141 mm* at bottom of flanges.



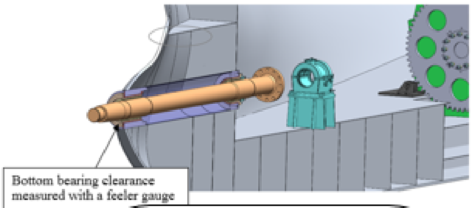


7 Stern Tube Bearing Clearance Measurement

Clearance measurement Flow Chart

Propeller not installed
Shaft(s) not restrained

Stern tube design

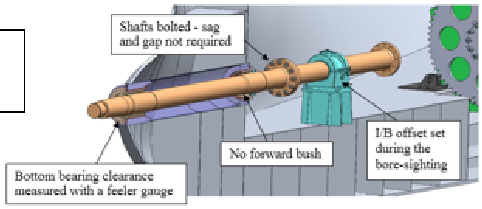


Installations with both S/T Bearings

- The shaft is unrestrained at each free end and supported only by both stern tube bearings.
- No additional supports or restraints are permitted

- Clearance measurement serves as an indirect verification of the aft S/T bearing misalignment.
- Misalignment corrections are not possible by offset adjustment because both S/T bearings are fixed in place.

- I/B offset can be adjusted to fine tune bearing loads on forward S/T bearing and the M/E bearings.
- I/B is chocked only after M/E chocking.



Installations with no fwd S/T bearing

- Propeller shaft and intermediate shaft are bolted together before clearance is verified
- Shafts are unrestrained at each free end and supported only by aft S/T bearing and I/B.
- No additional supports or restraints are permitted

- The procedure is simpler and faster than the conventional approach with two S/T bearings
- Aft stern tube bearing bottom clearance measurement can be used as a control mechanism to verify and fine-tune the misalignment slope angle by adjusting the intermediate bearing offset.

- The stern tube bearing clearance is precisely set to match the calculated value by adjusting the intermediate bearing offset
- Sag and gap at the flanges connecting the propeller shaft and the intermediate shaft do not need to be confirmed

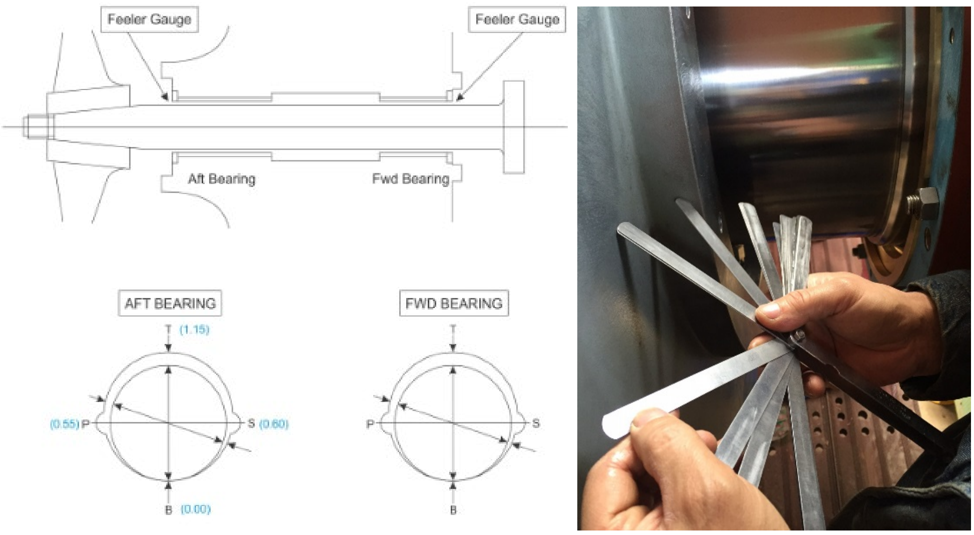
- I/B is chocked immediately after verification of S/T bearing clearance.
- Bearing loads on M/E and I/B are fine tuned by M/E offset and inclination adjustment.

After installation of the shafting system is completed and before floating of the vessel, bearing clearances are measured and recorded at four positions along the shaft circumference: top, bottom, port, and starboard.

The total horizontal clearance is calculated by adding port and starboard clearance measurements. The vertical clearance is calculated by adding top and bottom clearance measurements.

The resulting total clearance must be within the sum of the design clearance, including the manufacturer's tolerances specified in the bearing and tail shaft drawings.

FIGURE 17
Feeler Gauge Clearance Measurement



This paragraph however focuses on the bottom clearance measurement at the aft stern bush before the propeller is installed, because this measurement can be utilized to verify and adjust stern tube bearing misalignment.

The tail shaft clearance calculation translates the calculated bearing misalignment slope angle into a measurable bottom clearance value. The slope angle can thus be confirmed by measuring the bottom clearance with the tail shaft resting unrestrained on two bearings.

The slope angle is acceptable when the measured clearance is within the calculated allowable bearing clearance range established in the calculation sheet.

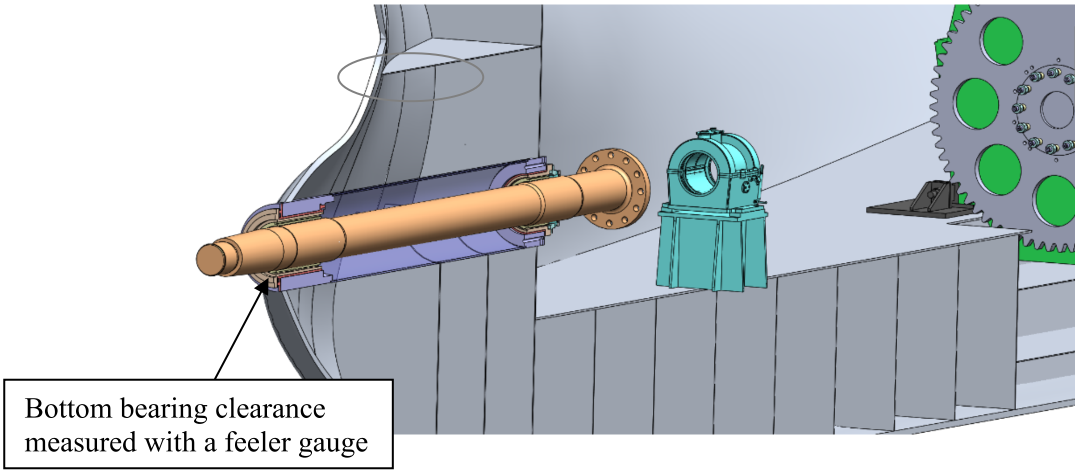
The shipbuilder verifies the tail shaft clearance by inserting a feeler gauge between the lower part of the tail shaft and the aft stern tube bearing. Only the aft edge clearance is required to be verified.

The transition point on stern tube bearings with a double slope may be also measured from the bearing aft edge.

7.1 Installation with Both Stern Tube Bearings

In case of stern tubes with both stern tube bearings (Section 4, Figure 18), the aft stern tube bearing clearance measurement is conducted with the propeller shaft installed in the stern tube with no propeller fitted. The shaft is kept unrestrained at each free end and supported only by both stern tube bearings. No additional supports or restraints are permitted on installations with a forward and an aft stern tube bearing.

FIGURE 18
Clearance Measurement – Both Stern Tube Bearings



Generally, the bearing condition is deemed acceptable if measured clearance does not deviate more than *0.1 mm* from calculated value.

7.2 Installation with no Forward Stern Tube Bearing

The shaft alignment process for installations with a forward stern tube bearing differs from the process for installations with no forward stern tube bearing.

For installations with no forward stern tube bearing, the recommended procedure for measuring the bottom clearance is to bolt the propeller shaft and intermediate shaft together immediately after the bore sighting is completed and the intermediate bearing is set to required vertical position (Section 4, Figure 19).

Only in installations with no forward stern tube bearing can the aft stern tube bearing bottom clearance measurement be used as a control mechanism to verify and fine-tune the misalignment slope angle by adjusting the intermediate bearing offset. This is only effective when the propeller is not attached, and the tail shaft and the intermediate shaft are coupled and kept unrestrained at their free ends. The shaft assembly sits freely on the aft stern tube bearing and the intermediate shaft bearing. The intermediate shaft bearing acts as the second support instead of the forward stern tube bearing.

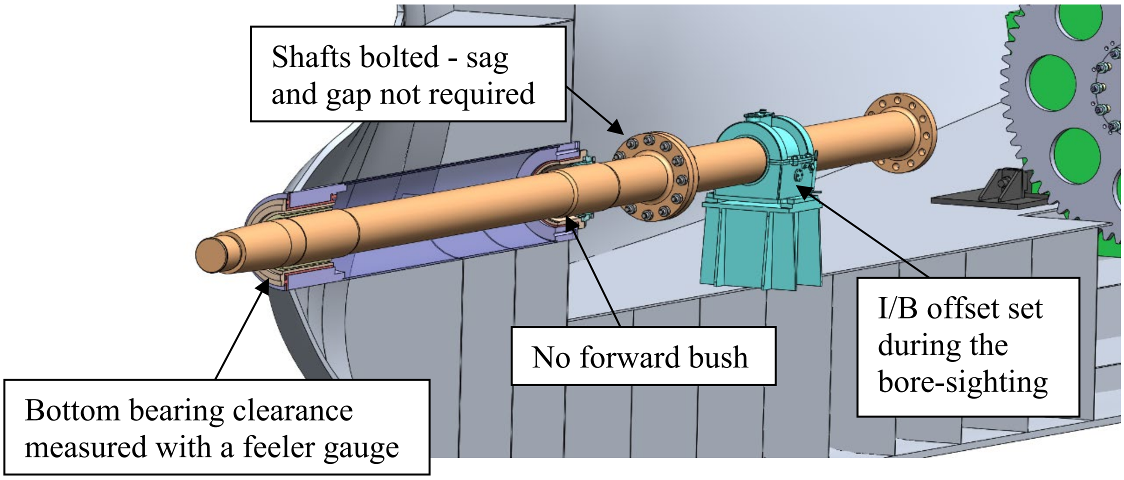
The advantages of this method are:

- The procedure is simpler and faster than the conventional approach.
- Sag and gap at the flanges connecting the propeller shaft and the intermediate shaft do not need to be confirmed.
- The stern tube bearing clearance can be precisely set to match the calculated value by adjusting the intermediate bearing offset.

When the clearance is measured, attention should be paid to the following.

- Once the vessel is afloat, hull deflections impacting the intermediate bearing offset may still influence the stern tube bearing misalignment angle. It is advised to be particularly aware of this when the clearance measurement is conducted in an early block stage and with the main engine not yet in place.
- The alignment calculation should be consulted to prevent potential overloading of the intermediate bearing and excessive bending of the forward end of the intermediate shaft.

FIGURE 19
Clearance Measurement – No Forward Stern Tube Bearing



8 Eccentricity (Run-out) Measurement of the Shaft

Excessive shaft eccentricity or "run-out" may result in alignment problems, because the bearing load and direction of the load may vary significantly as the shaft rotates, resulting in horizontal loads on the bearing that may consequently lead to bearing damage or even failure. Excessive run-out may additionally generate vibration and whirling problems.

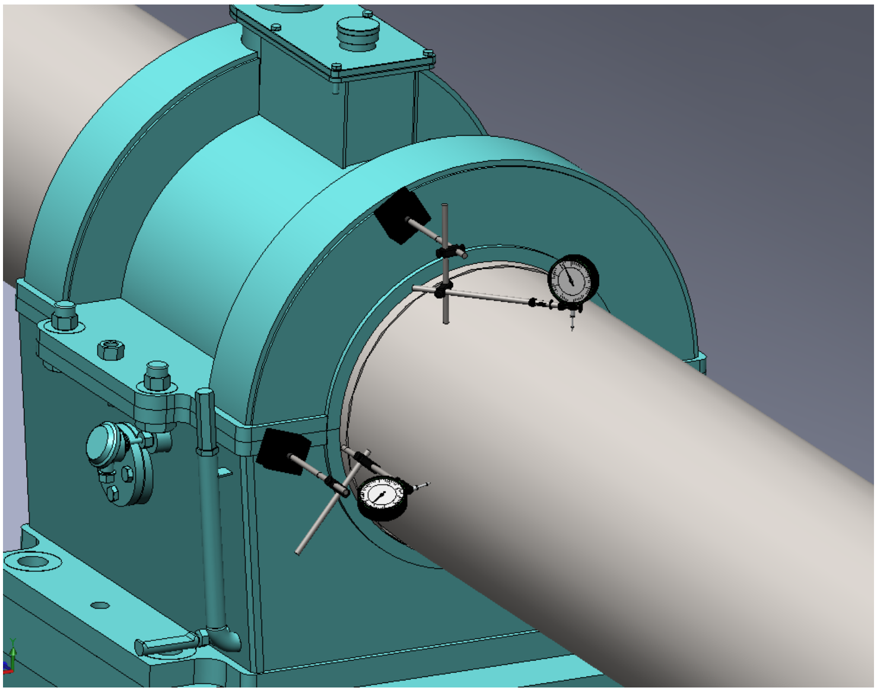
As a preventive measure, while the shaft alignment procedure is conducted shafting run-out can be measured directly with a dial indicator or evaluated indirectly by recording the change in bearing reaction or the strain in the shaft. Run-out measurements can only be performed with a slow rotating shaft.

Onboard run-out measurements are performed for the purpose of shaft alignment, but occasionally shafts may be removed for run-out verification on a lathe.

8.1 Dial Gauge Run-out Measurements

The measurement utilizes two dial indicator gauges (Section 4, Figure 20), one placed in the vertical plane and the other in the horizontal plane. As the shaft rotates, under ideal alignment conditions and no run-out, the indicators will not deviate from the initially set value, usually set to zero. However, such an ideal condition is not probable in practice and some deviation will always be recorded. This may not necessarily be the result of run-out, but may also be attributed to hysteresis in the shaft, caused by internal friction with the bearing.

FIGURE 20
Dial Gauge Run-out Measurement



9 Stress Measurements

9.1 Stress in the Shafting

The stress level in propulsion shafting is seldom affected by shaft alignment. However, in some cases where the level of the stress in the shafting is already at the limit, the bending and shear stress introduced by the alignment condition may be a contributing factor.

Bending and shear stress in shafts can be easily measured using strain gauges.

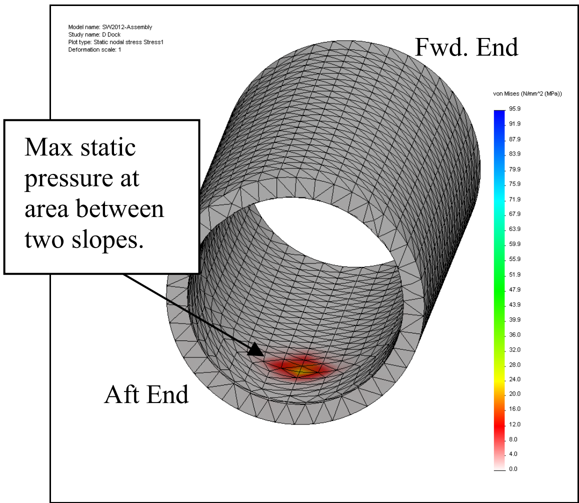
9.2 Stress in the Bearing

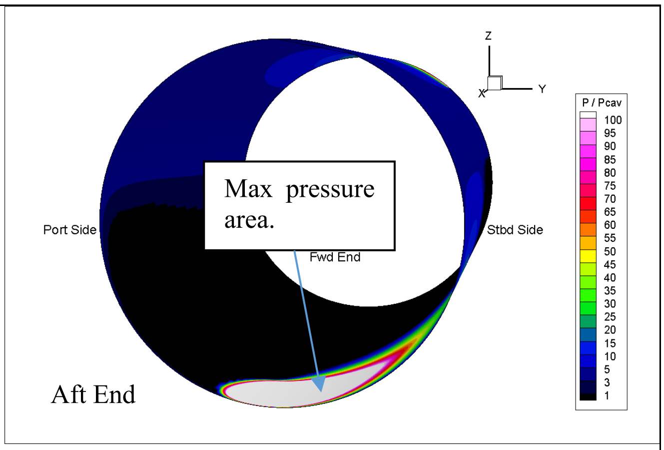
In contrast, the stress level in the bearings is dominated by the shaft alignment condition. Bearing-to-shaft misalignment is a particularly important factor that defines stress distribution. The misalignment angle is directly proportional to the contact area and, accordingly, to the stress as well.

Bearing stress measurement is a near impossible task and can only be indirectly evaluated by contact area examination when the bearing shell is removed for inspection.

Section 4, Figure 21 illustrates a simulated condition of static, metal-to-metal contact between the shaft and the bearing, and the dynamic evaluation of bearing performance through a starboard turn. The evaluated aft stern tube bearing is of double slope design. Slopes are designed so that in a static condition the shaft exerts maximum pressure at the transition point between two slopes.

FIGURE 21
Bearing Stress Evaluation





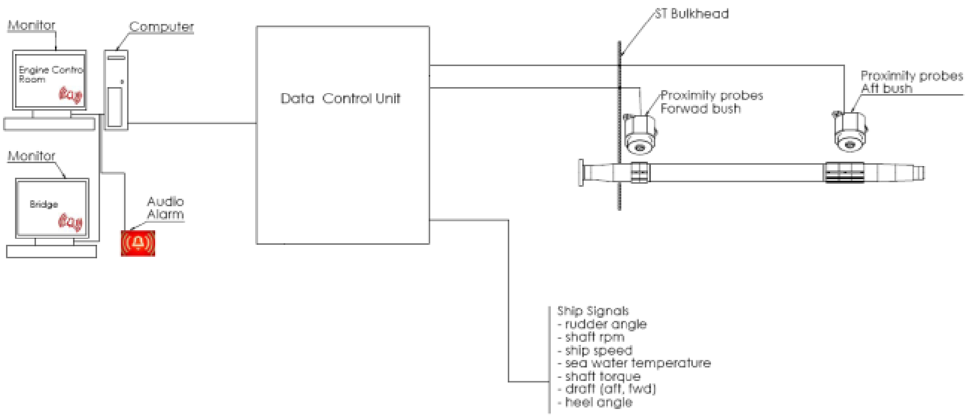
Static: Shaft imprint on the aft stern tube bearing. Simulated static bearing contact – Light ship Condition. Shaft in hard contact with a bearing at the transition area between two slopes.	Dynamic: Fluid structure interaction software is utilized to evaluate dynamic performance of the bearing during turning. Maximum pressure is found to be exerted at the aft bearing bottom edge.

10 Stern Tube Bearing Monitoring

Ocasionally vessels are equipped with a stern tube bearing monitoring system.

Stern tube bearing monitoring can minimize the risk of a possible bearing damage, thus enhancing vessel safety, reducing likelihood of environmental pollution and avoiding unnecessary maintenance.

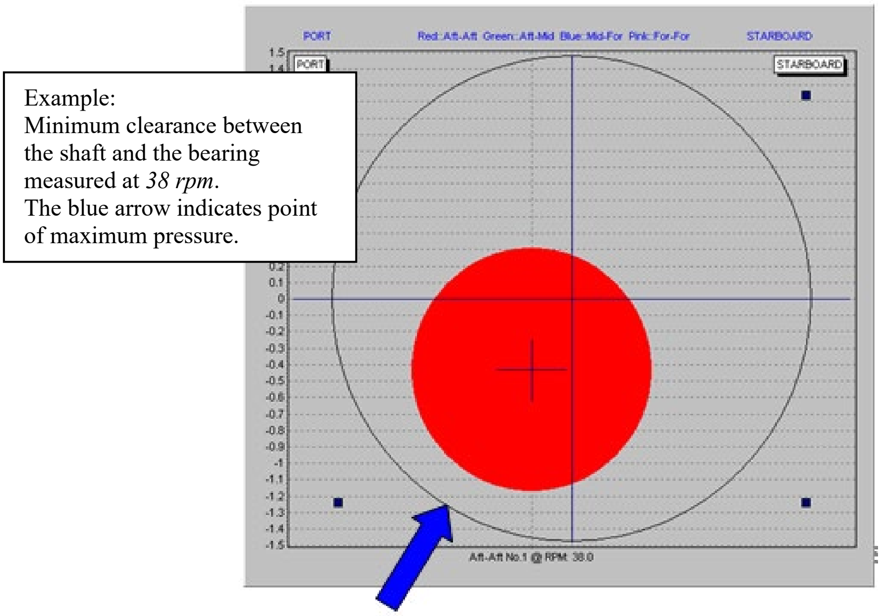
FIGURE 22
Schematic of a Monitoring System Hardware Arrangement



Fitting proximity sensors inside the stern tube bearing is one possible way to monitor stern tube bearing performance on an operational propulsion system. Appropriate selection of sensors and their location along the bearing length allows measuring and recording the shaft position inside the bearing, and the distance between the shaft outer diameter and the bearing shell inner surface can also be defined. In combination with bearing temperature and other monitored parameters such as RPM, vessel speed and rudder angle, advanced alarm settings can be established to alert the ship's crew of possible adverse conditions developing in the bearing. A schematic of one such system is presented in Section 4, Figure 22.

The following example shows a real time monitored interaction between the tail shaft and aft stern tube bearing.

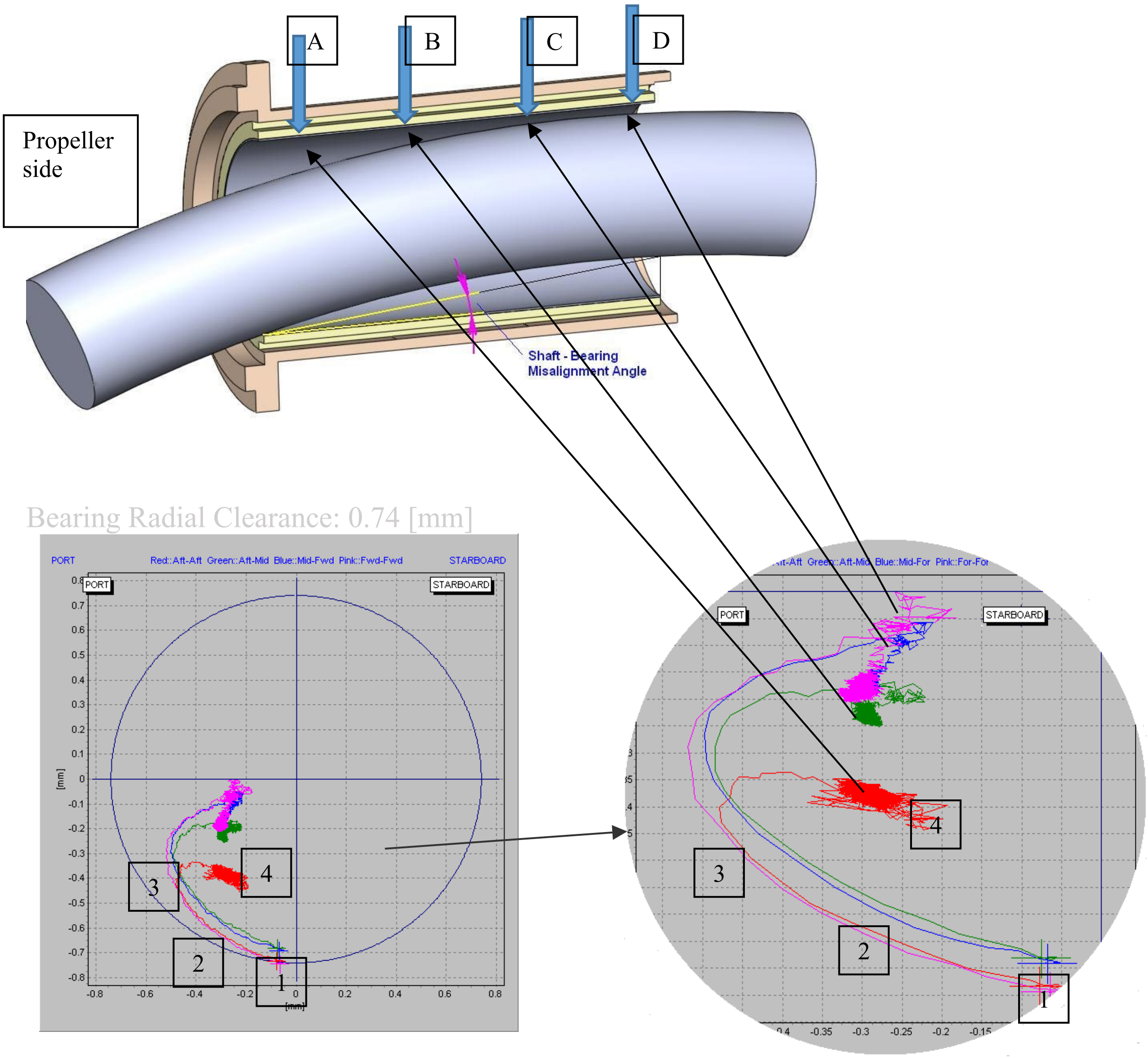
FIGURE 23
Stern Tube Bearing – Real Time Monitoring Example



When the engine starts and the shaft begins rotating, an oil wedge develops and the lubricant lifts the shaft. Section 4, Figure 23 is a screenshot taken during sea trials on a container vessel that has a stern tube bearing monitoring system installed. The red circle shows the shaft position inside the bearing after engine start up, and the blue arrow indicates the point of minimum clearance. The screenshot shows the shaft in equilibrium: a stable shaft run with a relatively large minimum clearance.

Section 4, Figure 24 provides the history plot of the same event recorded on Section 4, Figure 23. It shows the shaft orbit inside the aft stern tube bearing transiting from idle to a steady equilibrium at 38 rpm.

FIGURE 24
Aft Stern Tube Bearing Monitoring Arrangement

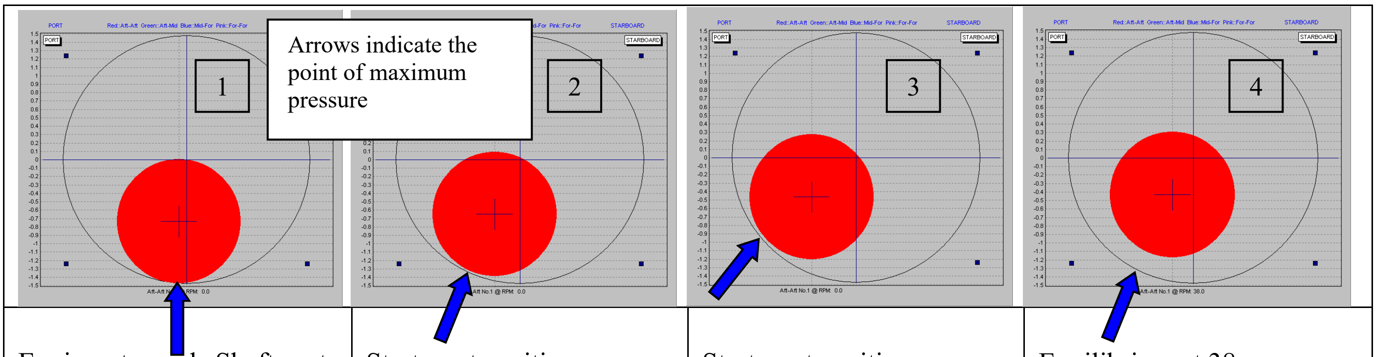


Shaft orbits are recorded at four locations inside the stern tube bearing and identified by different colors:

- *A - Red*: aft bearing edge
- *B - Green*: middle aft part
- *C - Blue*: middle forward part
- *D - Pink*: forward bearing edge

The same transition event from static to steady dynamic equilibrium at 38 rpm is given on four charts in Section 4, Figure 25. Charts capture the aft most monitored plane (A) at the aft edge of the stern tube bearing, showing the shaft position during starting of the engine and the hydrodynamic lift of the shaft.

FIGURE 25
Transition from Static to Steady Dynamic Run



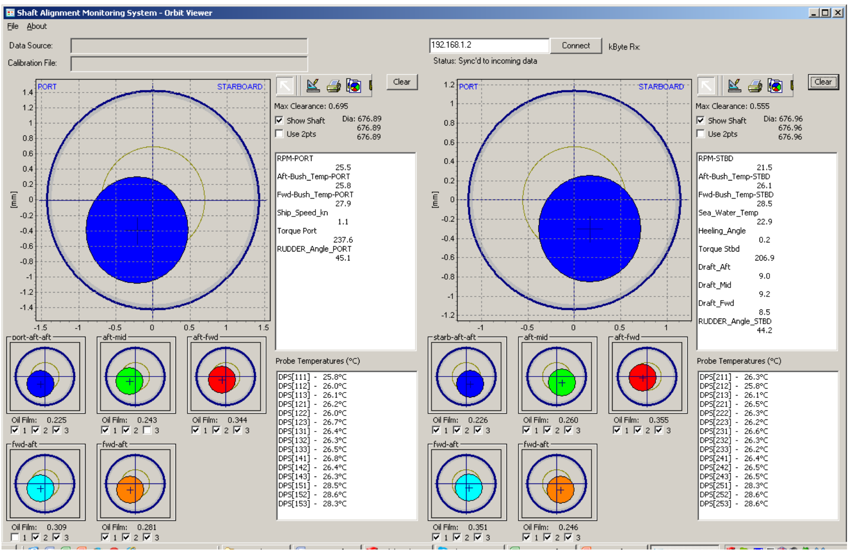
Engine stopped. Shaft rests at the BDC. Static metal to metal contact.	Start up: transition Shaft rotating; oil film developing.	Start up: transition Shaft rotating; oil film developing.	Equilibrium at 38 rpm. Shaft afloat.

Another stern tube bearing monitoring example in Section 4, Figure 26, shows the twin-screw vessel equipped with proximity sensors on the port and the starboard stern tube bearings.

On this particular installation the shaft position in the bearing is monitored at three locations in the aft bush and two locations in the forward bush.

Measurements are shown in real time on the screen in ECR and combined with other parameters collected from the ship controls (rudder angle, ship speed, rpm, etc.). Alarms and warnings are set to indicate potentially critical service.

FIGURE 26
Twin-Screw Stern Tube Bearing Monitoring Example



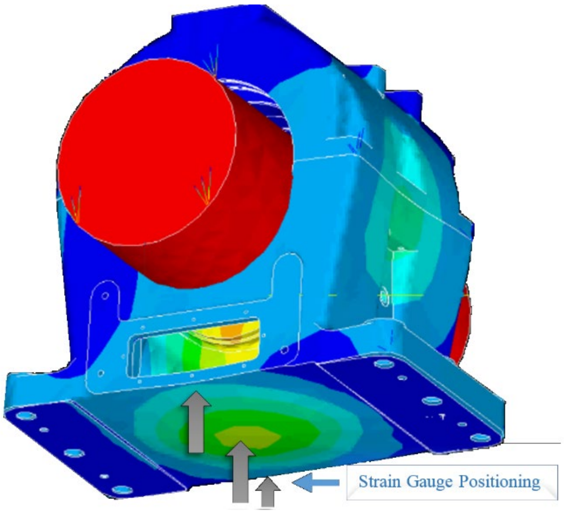
An adequately designed and correctly implemented propulsion shafting alignment is expected to deliver a bearing performance that complies with ABS Rules. Alignment is however carried out with shafting in a static condition and conventional temperature monitoring and lube oil sampling are mostly the only tools available to gauge operational bearing performance. The results are “after the fact” and preventing bearing damage requires quick and skilled action from ship’s personnel. Real time monitoring of stern tube bearings in a dynamic condition provides immediate useful information that can be analyzed and offer ship’s personnel ample time to take corrective or preventive action to mitigate bearing damage.

11 Intermediate Bearing Monitoring

Jack-up tests, typically used as a bearing load verification method, can only be accomplished under static shaft conditions. Sensors based on strain gauge technology can be utilized to measure the bearing load and misalignment angle through the bearing housing's displacement-induced strain in static and dynamic condition.

Strain gauges are typically installed on the bottom of the bearing casing, as indicated with arrows on the FE model of the bearing in Section 4, Figure 27.

FIGURE 27
Strain Gauge Location

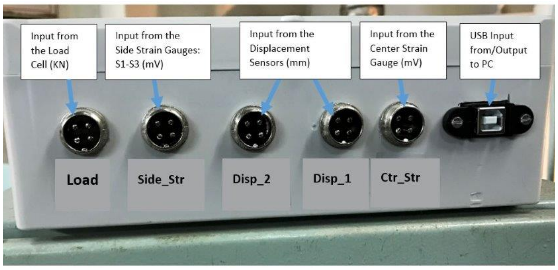


Unlike temperature sensors, this technology may allow the early identification of shaft alignment related problems, such as bearing unloading, bearing overloading or excessive shaft-bearing misalignment.

The sensor consists of two units, a data acquisition unit and a display unit, which works either wirelessly or through cable; see Section 4, Figure 28. The sensor output can also be integrated into the vessel's engine control room multi-function monitoring and display system.

FIGURE 28
Intermediate Bearing Load Variation During Vessel Maneuvering

Wireless display



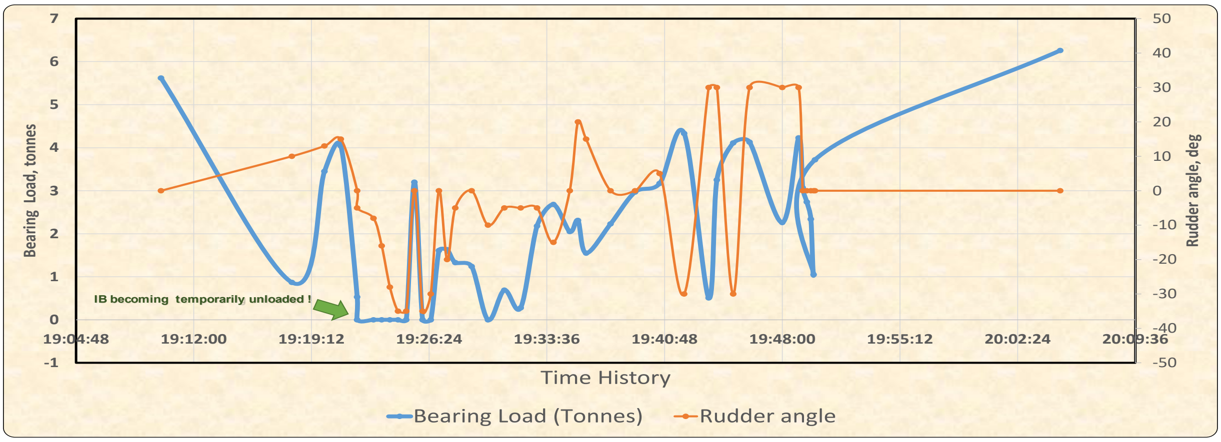
Data acquisition unit

Hard-wired display

The sensor data acquisition software provides bearing load, the misalignment angle and a bearing status indication, as "OK" or as "Alarm". The "Alarm" options can be presented as "overload", "unload" and "excessive misalignment angle". All the measured values are recorded in the sensor's internal memory card and can be retrieved for onward processing, such as for the purposes of a reverse shaft alignment calculation, which can provide a suitable diagnosis of the shaftline system.

Section 4, Figure 29 shows an example of a typical sensor output from a vessel maneuvering in a busy harbor area. The specific example is a stern tube arrangement without the forward S/T bearing. It is noted that during the recorded vessel move, where "ahead" and "astern" directions are actually combined with full rudder turns, the intermediate bearing becomes unloaded three (3) times. This provides unique visibility and assists in understanding of the shaft dynamic behavior under various operating conditions.

FIGURE 29
Intermediate Bearing Load Variation During Vessel Maneuvering



For the same example, in Section 4, Table 2 the results of reverse calculation are presented, where the intermediate bearing load is measured by the Smart Bearing Sensor during the above vessel sea trial maneuvering .Reverse engineering using the shaft alignment model allows us to investigate the bearing load re-distribution based on the Smart Bearing Sensor measured load of the intermediate bearing. The results are compared with the hot static case of the originally approved calculations for the same vessel loading condition.

TABLE 2
Intermediate Bearing Load Re-distribution During Vessel Maneuvering

	Bearing Reactions [kN]				Redistributed Load [kN]	
	Hot static– Shipyard Calculation Report		Maneuvering			
	Calculated	Measured	Calculated	Measured	Estimate	Measured
Aft S/T bearing	613.1	-	652.4	-	38.3	-
Intermediate bearing	52.0	-	-	0.0	-	-52.0
M/E bearing 8 – aft most	82.4	-	132.4	-	49.1	-
M/E bearing 7	95.2	-	57.9	-	-37.3	-
M/E bearing 6	268.8	-	269.8	-	1.0	-

At a certain stage during the maneuvering, the intermediate bearing appeared completely unloaded. Unloaded I/B may be critical for aft stern tube bearing performance when like in this installation there is no forward S/T bearing.

SECTION 5 Hull Girder Deflections and Alignment Optimization

1 Hull Girder Deflections

The alignment process is normally completed with the vessel in a relatively light draft condition, where the impact of hull defections remains relatively small. At this stage bearing offsets are tuned to achieve an acceptable load distribution among all bearings. When the vessel's draft increases to the ballast or to the laden condition the bearing reactions will change.

In case when hull deflections are not accounted for in calculation, predicting how bearing loads start being affected when the vessel's draft is increased and changed for different operational service conditions relies solely on shafting models and empirical data from previous experience with similar propulsion systems.

Some undesirable in-service consequences of the aforementioned may be:

- Distortion of the sighting line established in dry dock, thereby changing the bearing offsets and redistributing bearing loads, possibly resulting in unloaded bearings and excessive misalignment
- Hogging and sagging of the engine bedplate, thereby changing the offsets of engine crankshaft bearings, resulting in unloaded main engine bearings and excessive crankshaft web deflections that may lead to bearing or even crankshaft damage

It is recognized that even today many if not most shaft alignment designs do not take hull deflections into account. Under these circumstances, relying on the experience of shipyard personnel is the only means of controlling the alignment condition.

ABS Rules then require satisfactory in-service alignment to be substantiated by conducting additional bearing load verification through jack-up measurements at selected vessel service drafts and with the engine in a hot static condition.

Having the ability to account for hull deflections in an early design stage minimizes dependence on the experience of shipyard personnel, which could be of particular concern when implementing alignment on new ship hull designs. Accounting for hull deflections in the alignment design allows bearing reactions to be quite accurately assessed and confirmed for all vessel service drafts. The alignment calculation will provide a different set of confirmatory values for each evaluated condition when hull deflections are considered in the design stage.

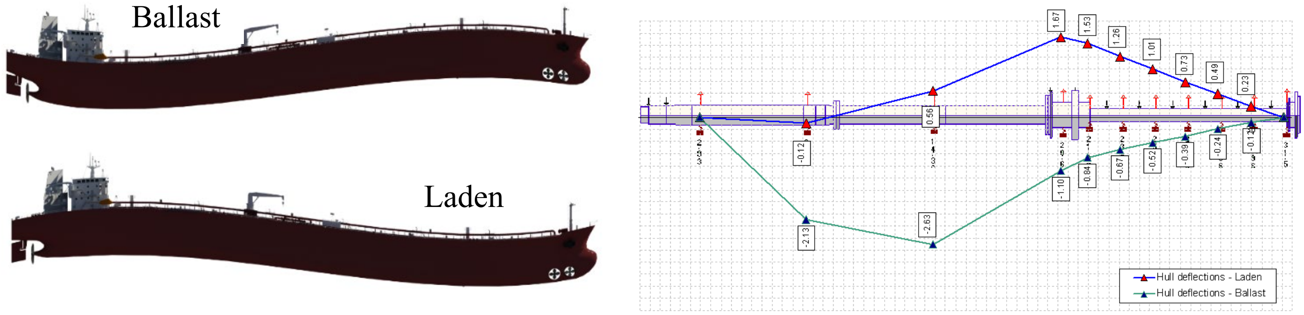
- Sighting data and bearing clearance verification information is provided for the vessel in the dry dock
- Sag and gap values are provided either for the dry dock or for the lightship condition
- Bearing load verification is defined not only for dry dock or the light ship condition but also for ballast and fully laden drafts.

ABS acknowledges the impact hull deflections have on shaft alignment, and developed a calculation tool to estimate hull deflections and optimize alignment design for all considered service drafts of a vessel. The ABS optimization software utilizes hull deflections that are a combination of analytically obtained and measured deflection values.

ABS conducted hull girder deflection measurements on a number of vessels of different types and sizes and applied these results to calibrate the analytically obtained values.

Section 5, Figure 1 shows the behavior of a typical tanker and bulk carrier under two extreme loading cases: ballast and laden. In ballast condition the vessel hogs at midship and sags at the stern. In the laden draft the opposite is true and the vessel sags at midship and hogs at the stern. A similar behavior is observed on other vessel types that have a high block coefficient. Large vessels with a smaller block coefficient, such as container carriers, will have high midship hull deflections, but for a typical design the stern structure will mostly hog in both the ballast and the laden condition.

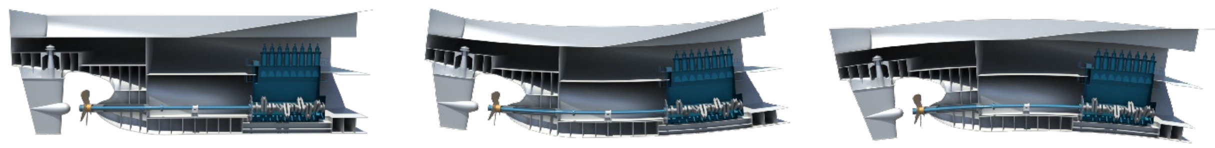
FIGURE 1
Hull Girder Deflections Influence on Propulsion System



Typical hull girder deflections of a VLCC under ballast and laden conditions.

Offset change on the propulsion shafting bearings under ballast and laden conditions.

FIGURE 2
Stern Section Hull Deflections



Dry dock Ballast Laden

Predominantly, hull girder deflections in the stern section have an impact on shaft alignment. Section 5, Figure 2 illustrates how the stern structure deflects on the typical tanker or bulk carrier.

1.1 The Effect of Hull Deflections

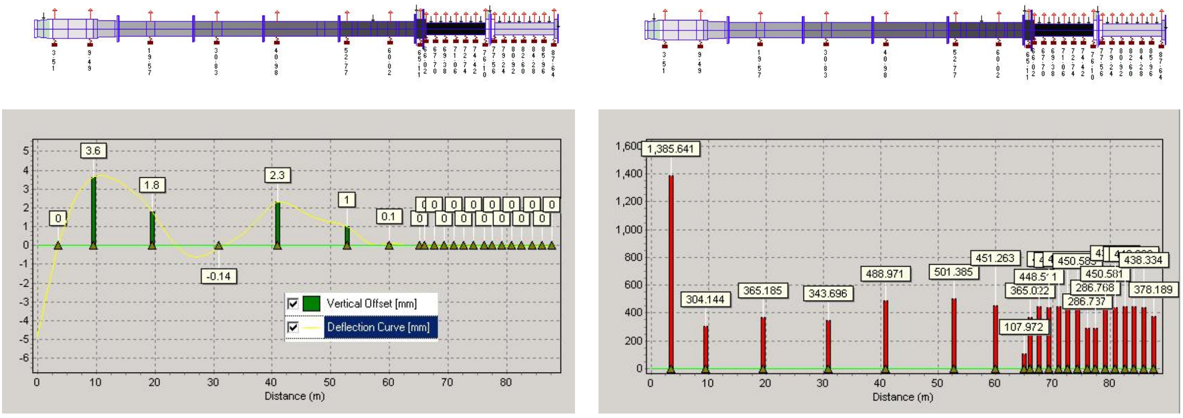
In conventional alignment designs hull deflections are normally not accounted for. The consequences of such a design, and accordingly the importance of considering hull deflections in alignment design is illustrated in the following example.

1.1.1 Conventional Design – No Hull Deflections

For prescribed bearing offsets in dry dock illustrated by the chart on the left of Section 5, Figure 3, the reaction loads on the bearings are almost ideally defined, as can be seen in the chart on the right of Section 5, Figure 3.

However, the calculated results will be valid for only this alignment condition. That poses a problem, because alignment must be satisfactory for all vessel loading drafts.

FIGURE 3
Shaft Alignment Design with No Hull Deflections Considered



Section 5, Figure 4 illustrates that when implementing the design for the dry dock or the light ship condition from Section 5, Figure 3, the alignment in the laden draft will be unacceptable.

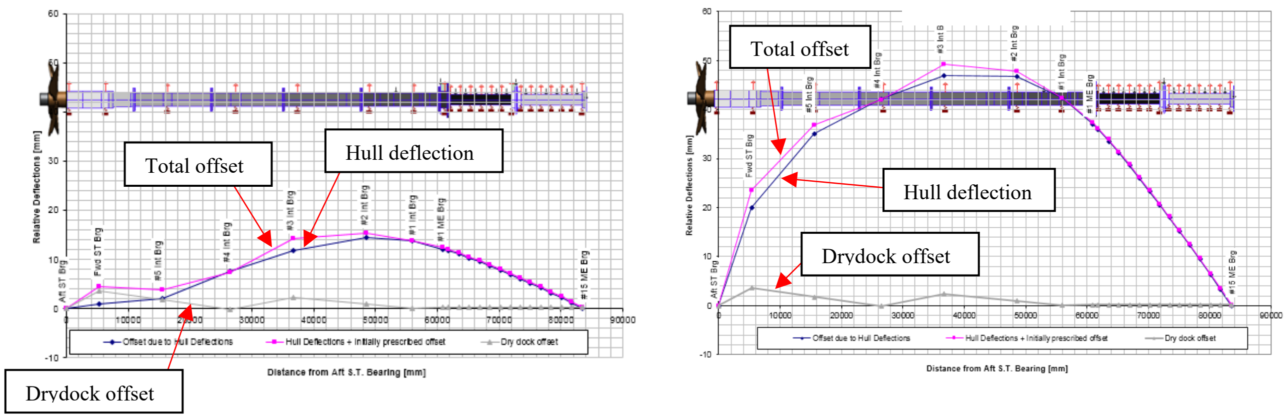
To demonstrate this potential problem, hull deflections for ballast and laden vessel conditions are applied to the design from Section 5, Figure 3.

1.1.2 Conventional Design Evaluated with Hull Deflections

Charts in Section 5, Figure 4 show three curves each, representing dry dock bearing offset, hull deflections, and the total offset. The total bearing offset is defined as a sum of the dry dock offset and respective hull deflections. Hull deflections in the chart on the left are for ballast and on the right for laden draft of the vessel.

FIGURE 4
Total Offset and Hull Deflections Only

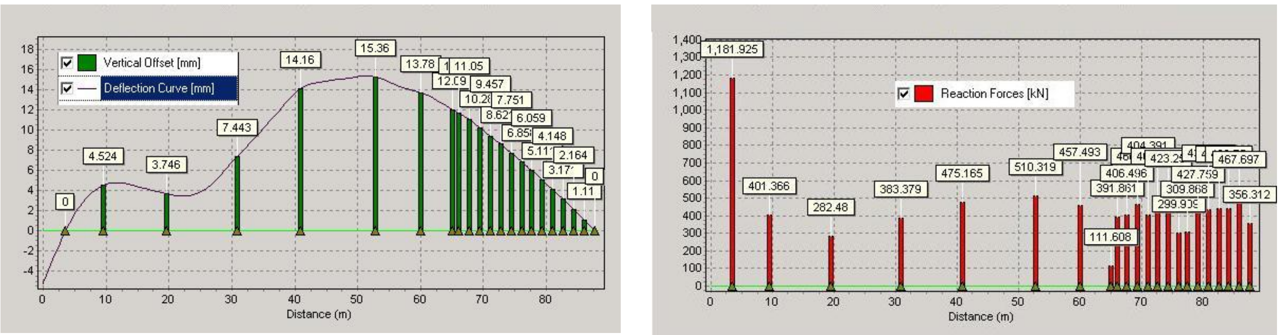
Ballast *Laden*



Hull deflection data and bearing offsets for each condition shown in Section 5, Figure 4 are used in the analysis. The produced results for the ballast and laden conditions are illustrated by Section 5, Figure 5 and 6 respectively.

FIGURE 5
Still-water Hull Deflections – Ballast

Ballast

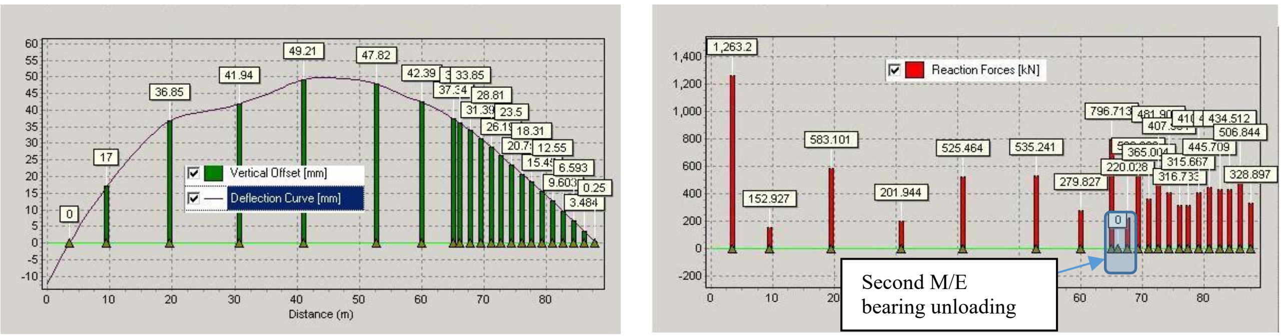


Bearing offset: Still water hull deflections – Ballast

Bearing reactions: Still water hull deflections – Ballast

FIGURE 6
Still-water Hull Deflections – Laden

Laden



Bearing offset: Still water hull deflections - Laden

Bearing Reactions: Still water hull deflections - Laden

Section 5, Figure 5 indicates that the ballast condition still appears satisfactory when hull deflections are considered. However, Section 5, Figure 6 suggests that in the laden draft the second aft most main engine bearing may unload. Such a result is unacceptable and may lead to bearing damage.

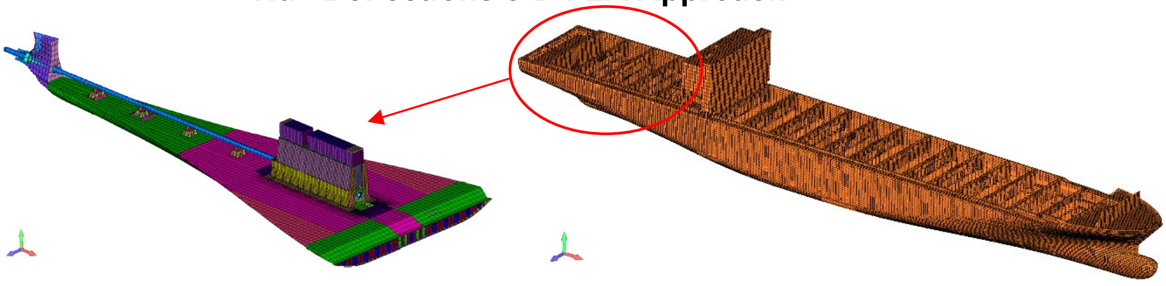
This scenario is common for short, stiff shafting systems installed in a relatively flexible hull structure and is typical for vessels with only one intermediate bearing such as tankers and bulk carriers. It is therefore beneficial that a designer acquires reliable hull deflection data when developing a shaft alignment design. Hull deflections are typically obtained through either an analytical or numerical process or through onboard measurements, or both.

2 Analytical/Numerical Approach

The 3D analytical method for the hull deflection calculation is a very accurate but time consuming and expensive approach. It typically requires detailed finite element modeling of the vessel, in particular the stern part with a comprehensive model of the engine room, the engine and the shafting. The analytical approach is seldom undertaken solely for the purpose of investigating the effect of hull deflections on shaft alignment. It is a more common practice to take advantage of the full-scale vessel modeling performed for dynamic loading analysis or similar, and extract from it the hull deflection data that may impact the shafting alignment condition. Refer to Section 5, Figures 7, 8 and 9.

Section 5, Figures 7, 8 and 9 show two container vessel models used for hull deflection calculation.

FIGURE 7
Hull Deflections 3-D FEA Approach



Detailed FE model of the shafting, engine and a double bottom.

Whole vessel FE model.

FIGURE 8
Hull Deflections Calculated Using FEA

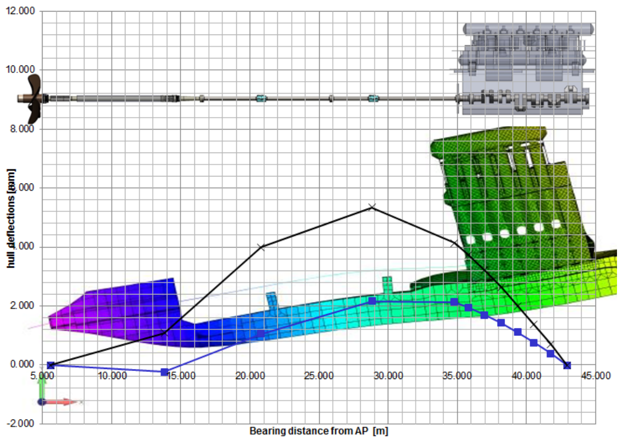
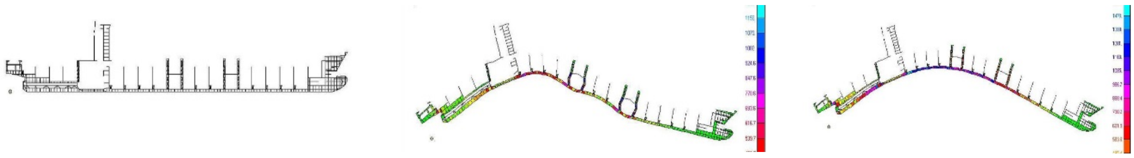


FIGURE 9
Vessel Deflections Change with Loading Condition



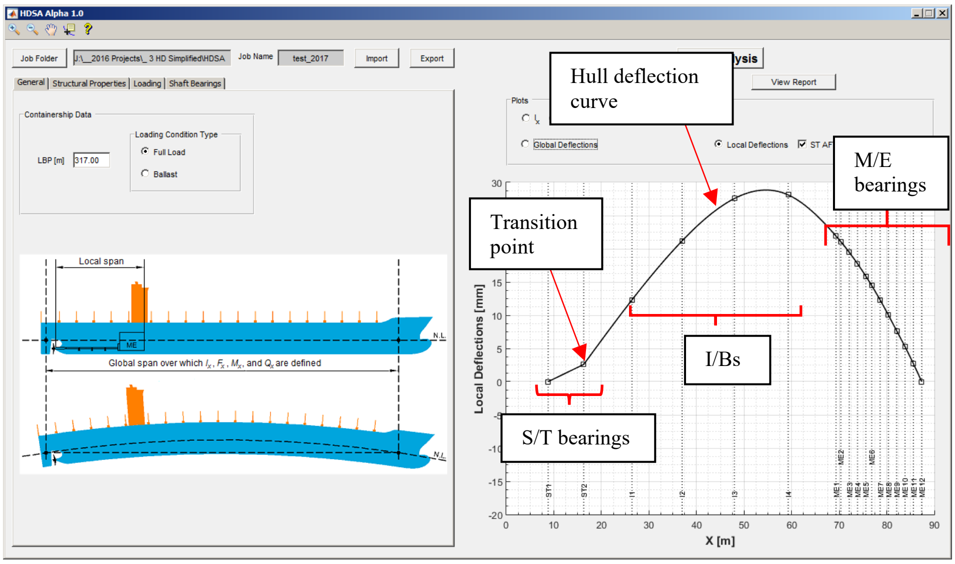
Dry dock – no hull deflections

Ballast – still water deflection

Laden – still water deflections

Another analytical method based on the *1D* beam theory can also be utilized for hull deflection evaluation (Section 5, Figure 10). When information about sectional modulus inertia and shear area are provided, the beam model may produce high accuracy hull deflections results that match the *3D* model. However, performance of the *1D* model is hindered by the very high stiffness of the stern tube structure. The *1D* model cannot capture this abrupt change in deflection gradient and therefore calibration of the *1D* model is required. Calibration is performed by measurements and by comparing the *1D* results with the *3D FE* models.

FIGURE 10
Hull Deflection 1-D Beam Approach



3 Hull Girder Deflection Measurements

From the point of view of shaft alignment, the only hull deflections of interest are those manifested in the stern section of the ship, where the propulsion shafting is located.

Bearings that support the propulsion shafting experience changes in their offset when the vessel's draft changes. This is caused by the varying buoyancy effect on the vessel's hull. By measuring the change in bearing offset for two different vessel conditions, hull deflections can be simply defined as a difference between those two measured conditions.

Direct measurement of the bearing offset is not possible at this time, because the optical sighting-through line is obstructed by the already-installed shafting. Measurements thus rely on indirect methods that are primarily based on measuring bending of the shafting. Bending measurements can be utilized to reverse calculate bearing offsets. During this process, bearing reactions and crankshaft deflections are concurrently measured and applied in the same calculation to improve reverse calculation accuracy.

Particulars about reverse calculation and measurements of the bending moment, bearing reaction and crankshaft deflection for the purpose of obtaining the hull deflections are covered in 5/3.3.

A minimum of five measurements sets at different vessel draft conditions must be conducted to obtain the necessary data for defining the hull deflections:

- Dry-dock
- Light draft immediately after launching <u>and</u> before any bearing adjustment
- Light draft after bearing adjustment
- Ballast
- Fully laden

Note: Both measurements at light draft after the launching are needed because the bearing offset is normally readjusted after launching.

3.1 Bending Moment Measurements

Bending moments are measured using strain gauges. A typical set up for strain gauge measurements is shown in Section 5, Figure 11. An advantage of using the strain gauge method is the consistency of results for repeated measurements. Moreover, measurement errors can be minimized or even fully eliminated when consecutive readings are compared.

The bending curvature of the shafting is essential information needed for obtaining the bending moments and the actual position of the propulsion shafting bearings. The more strain gauge locations used, the better the accuracy of the estimated deflection curve will be.

The number of strain gauges installed on the propulsion shafting depends on its design. For shafting with one intermediate bearing, a minimum of three strain gauge locations are needed. Each strain gauge location ideally consists of four strain gauges wired into a Wheatstone bridge.

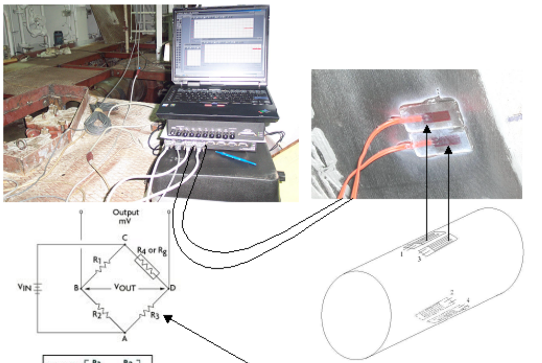
Strain measurements are recorded in each location while slowly rotating the shaft for two to three revolutions. The shaft is rotated first in the ahead direction and then astern and the resulting moments are the average between the two readings as illustrated by the graphs in Section 5, Figure 12. This procedure helps relax the shafts and reduces the hysteresis error arising from internal shaft friction.

The strain gauge signal is conditioned as it passes through the strain gauge data acquisition hardware and is then filtered and converted from an analog into a digital signal in the A/D converter.

From the A/D converter, the digital information is transmitted to a portable computer, where the acquired data is further processed before being logged onto the computer's hard drive.

Data processing software on the PC controls the acquisition process and further samples, filters and maps the collected information into bending moments. During the acquisition, data is continuously monitored on the computer screen and simultaneously stored on the drive.

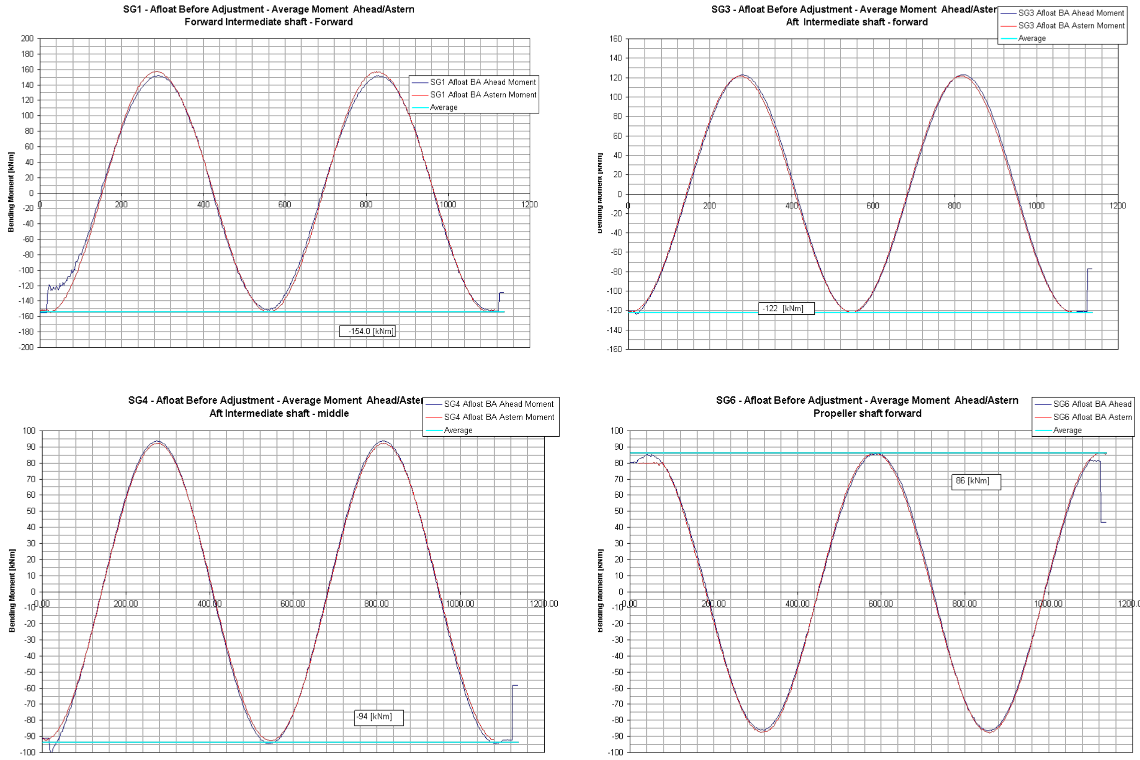
FIGURE 11
Strain Gauge Measurement Schematics



Strain gauge schematic – full bridge is apply at each measurement location

Two pairs of strain gauges are located 180 degrees apart and connected to form Wheatstone bridge

FIGURE 12
Measured Moment Examples



The above graphs are moment readings collected at four locations along the line shaft. At each location two measurements were taken while rotating the shaft in the ahead and astern direction. The sine curves represent variation of the moment during the two shaft rotations. The amplitude of the recorded sine curves represents the actual bending moment.

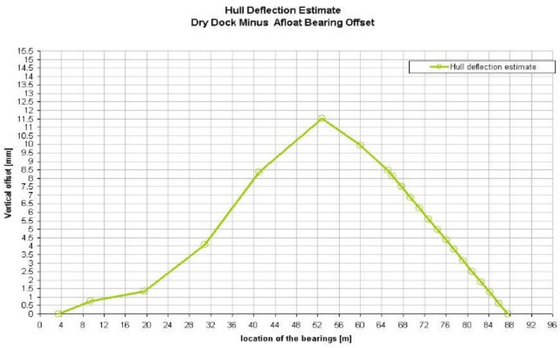
3.2 Hull Girder Deflection Measurements Example

In this example the same container vessel from 5/1.1 is used to evaluate hull deflection measurements.

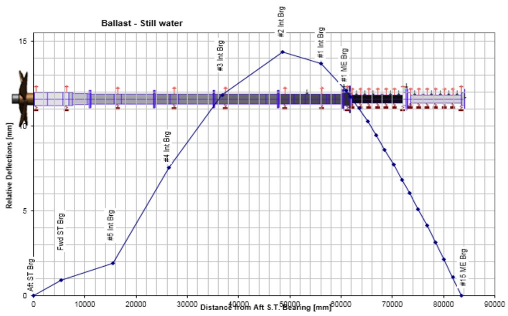
Strain gauges were placed at nine locations along the line shaft and the bending moments were measured. Engine crankshaft deflections, main engine bearing reactions, line shaft bearing reactions and forward stern tube bearing reactions were measured immediately before or after the strain gauge measurements were performed. Reverse analyses were subsequently conducted to determine the bearing offset from the measured parameters.

The obtained results for the ballast and the laden condition are shown in Section 5, Figure 13 and 14 respectively and compared with the analytically obtained hull deflections.

FIGURE 13
Hull Deflections – Dry-dock to Ballast



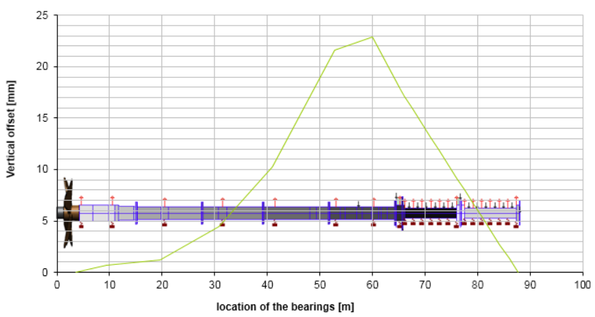
Measurement: Hull deflection change from dry dock to ballast condition



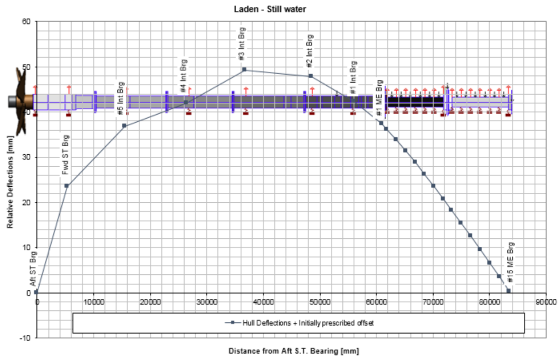
Calculated: Hull deflection – ballast condition

The shape and intensity of the analytically predicted deflection curve of the vessel in ballast condition closely matches the measured deflections.

FIGURE 14
Hull Deflections – Dry-dock to Laden



Measurement: Hull deflection change from dry dock to light laden condition



Calculated: Hull deflection – fully laden condition

The analytically predicted hull deflections for the fully laden condition was not fully comparable with measurement as the condition measured was for the partially laden ship.

3.3 Bearing Offset Measurement - Reverse Calculation Example

The bearing offset measurement is the same process utilized in obtaining the hull deflections. The hull deflections which are affecting the shaft alignment are obtained by simply subtracting the dry dock measured bearing offsets from the waterborne measured offsets.

Ship details

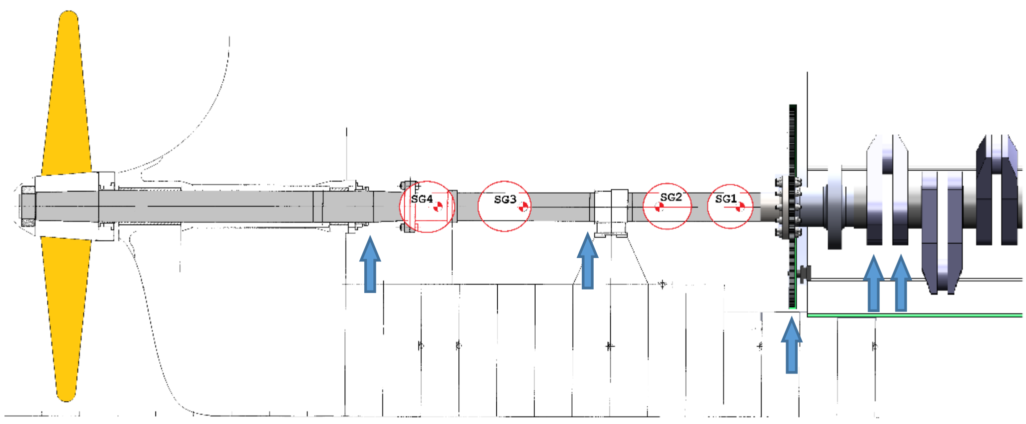
Type & Displacement of the ship	VLCC; 320,000 DWT
Shafting length	19.8 [m]
Propeller	9.9 [m] diameter; single screw
Engine	6 Cylinder; MCR 30,000 kW @ 76 rpm

In this example, a VLCC vessel is used to illustrate the bearing offset measurement through the reverse calculation process.

Since bearing offsets cannot be directly measured after the line of sight through the bearings is obstructed by installed shafts, an indirect process needs to be applied. The offset is indirectly obtained from measured bending moments, bearing reactions and crankshaft deflections. This process is called a ‘reversed calculation’ because it is opposite to the regular shaft alignment analysis where based on bearing offset we calculate reactions and bending moments.

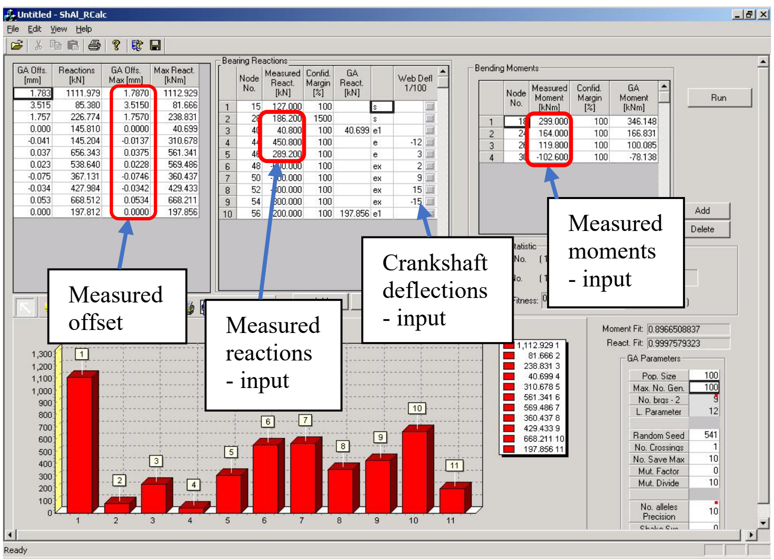
Bending moments are obtained through the strain gauge measurements. In this example, the strain gauges are installed at four locations along the line shaft, as indicated by the red circles in Section 5, Figure 15. Bearing reactions are measured at the forward stern tube bearing, the intermediate bearing and the three aft most main engine bearings, as indicated by the blue arrows. Data is recorded in the dry dock, and at ballast and fully laden drafts.

FIGURE 15
Moments and Bearing Reactions



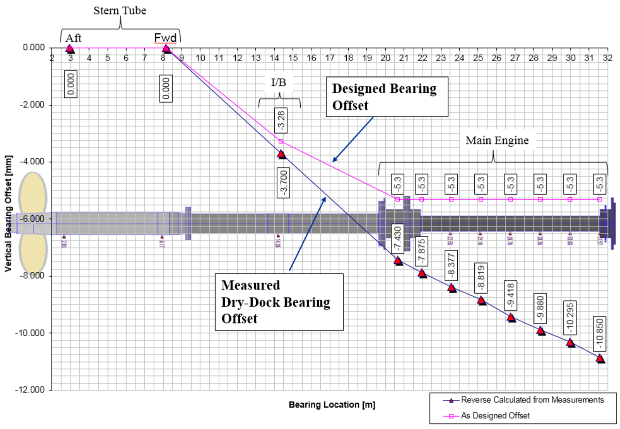
The reverse analysis graphical interface for the dry dock condition is shown in in Section 5, Figure 16. Similar analyses are performed for the ballast and laden conditions.

FIGURE 16
Reverse Calculation GUI

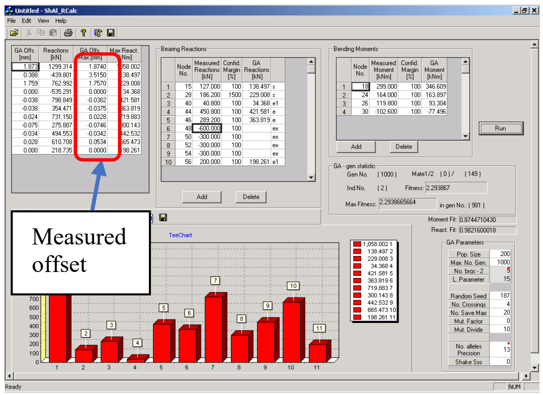


Section 5, Figures 17, 18 and 19 show the measured offset results of the reverse analysis conducted for the dry dock, ballast draft and the laden draft conditions of the vessel. On each chart the reverse calculated offsets are presented along with designed offsets.

FIGURE 17
Reverse Calculated Offset – Dry Dock Condition



Bearing offsets

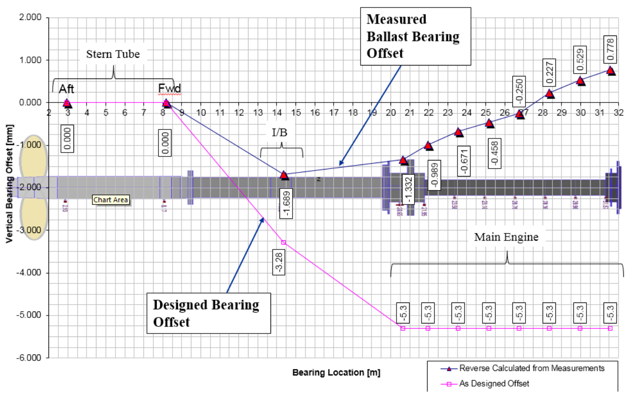


Input data and results

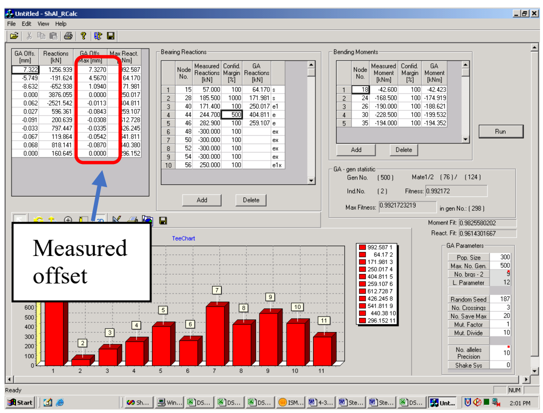
Although the strain gauge measurements were conducted to obtain hull deflections, the obvious advantage of the process is that the actual bearing offsets can also be obtained for the condition in which the measurements were conducted.

The dry dock results in Section 5, Figure 17 indicate a significant difference between the reverse calculated (measured) offsets and the designed offsets. This was expected, because the aft most main engine bearing reaction was readjusted in the dry dock per main engine maker's recommendation to a very light load as seen in the chart on the right of Section 5, Figure 17. In doing this, the original offset for this bearing, which was set by sighting and the sag and gap method, was changed to accommodate the engine maker's requirements.

FIGURE 18
Reverse Calculated Offset – Ballast Condition



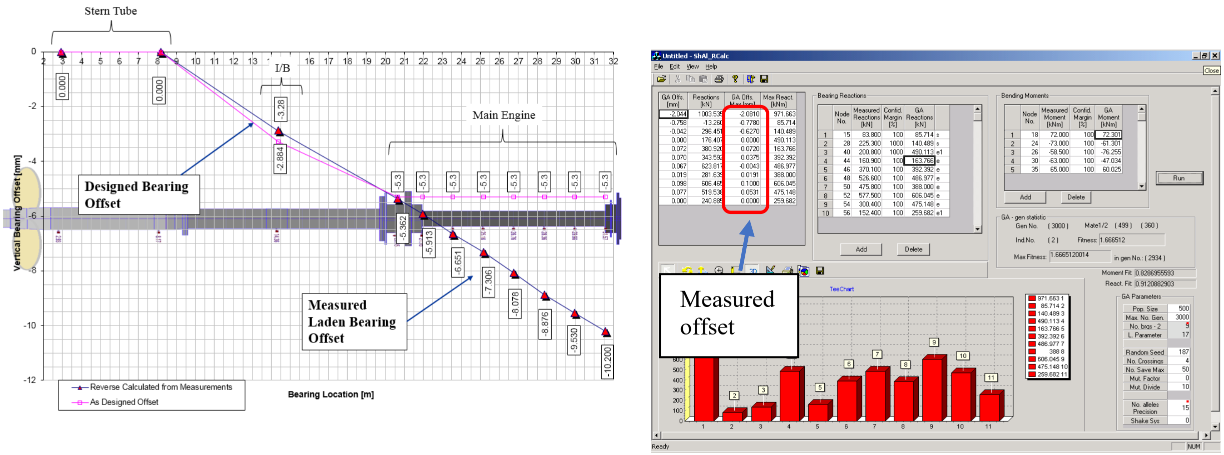
Bearing offsets



Input data and results

The measured (reverse calculated) intermediate bearing and main engine bearing offsets for the ballast draft condition are shown above the design offsets in the graph on the left of Section 5, Figure 18. The measured bearing loads seen in the chart on the right in Section 5, Figure 18 appear to be in compliance with the requirement that none of the bearings are unloaded.

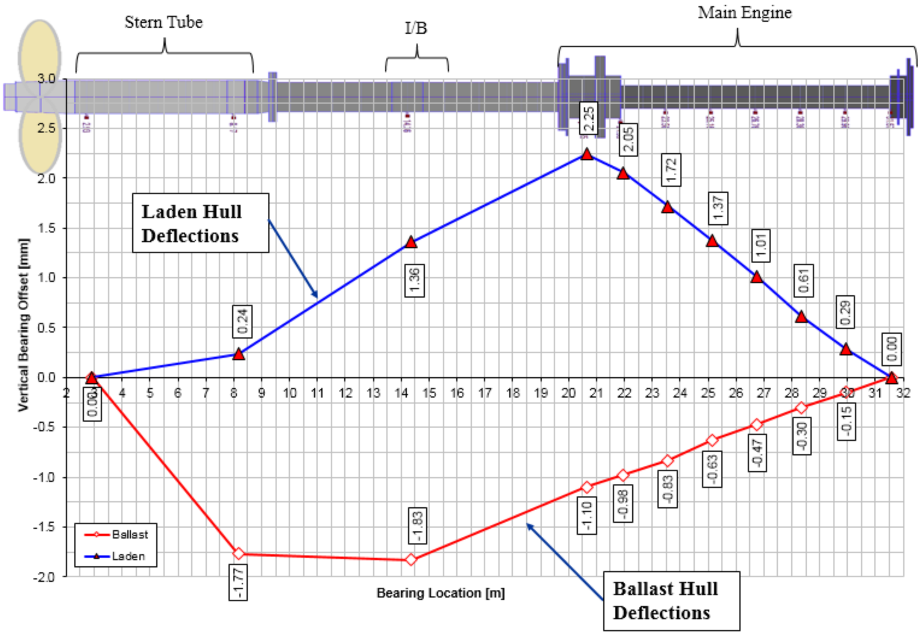
FIGURE 19
Reverse Calculated Offset – Laden Condition



Bearing offsets Input data and results

The measured (reverse calculated) bearing offsets for the laden draft shown in the graph at the left of Section 5, Figure 19 appear closer to the design (dry dock) values in Section 5, Figure 17. This behavior is expected for large tanker and bulk carrier type vessels. The measured bearing loads appear satisfactory, with all bearings positively loaded.

FIGURE 20
Measured (Reverse Calculated) Hull Deflections



Hull deflections for this vessel are now obtained by simply subtracting the dry dock bearing offsets in Section 5, Figure 17 from the ballast offsets in Section 5, Figure 18 and the laden offsets in Section 5, Figure 19. The results are shown in Section 5, Figure 20.

4 Design Optimization

Knowledge of hull deflections beforehand provides information to designers to optimize shaft alignment offset design for in service vessel conditions.

When a designer accounts for hull deflections in the shaft alignment calculation for the drydock, light afloat, fully ballasted, and fully laden conditions, it becomes possible to define alignment parameters, such as bearing offsets and slope angles for the cold static condition, that will produce satisfactory alignment results for in-service vessel conditions. This process is called alignment optimization.

Bearing offsets optimized to provide satisfactory in-service vessel conditions may not always strictly satisfy ABS Rule requirements for other conditions, such as drydock or light afloat because those are not normally considered service conditions but are used during the alignment setup and verification process.

4.1 Theoretical Background

The shaft alignment problem is stochastic, with an infinite number of bearing offsets that may be analyzed but not precisely predicted to satisfy the imposed requirements. The goal of the shaft alignment optimization is to provide a robust set of acceptable solutions to bearing offsets that all satisfy imposed constraints, alignment parameters and criteria.

Multiple solutions are necessary because it is often an imperative to have the engineering evaluation as the final decisive factor in selecting the desired alignment design. Providing multiple solutions is a relatively simple task for the Genetic Algorithm (*GA*) software applied as the optimization tool in the ABS shaft alignment program.

The Genetic Algorithm's ability to conduct parallel search throughout the solution space is ideally suited for shaft alignment tasks. The *GA* optimization engine simultaneously provides multiple sets of satisfactory bearing offsets and proposes the one that best fits the imposed criteria. The GA software optimizes within constraints imposed on the solution space that are defined by hull deflection curvatures, which normally represent the still water ballast and the laden vessel drafts.

Parameters and alignment criteria that may be considered in optimization process are:

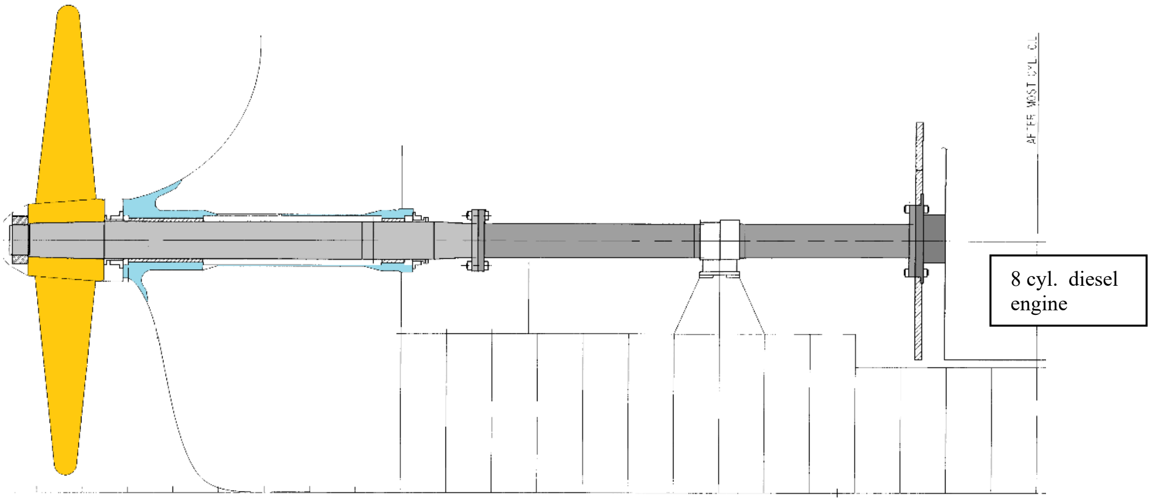
- Thermal expansion
- Diesel engine bedplate prescribed sagging
- Bearing wear down
- Main engine flange allowable moment and shear force.

4.2 Optimization Example

VLCCs are particularly sensitive to alignment problems because of the vessel's large size, short and rigid shafting and relatively flexible hull structure.

The VLCC used in this example is different from the VLCC used in example 5/3.3 for which the hull deflections were measured. Refer to Section 5, Figure 20. The VLCC in this optimization example uses hull deflection measurement data together with the analytical approach to predict actual hull deflections on the vessel.

FIGURE 21
Shafting Arrangement



The shafting system in Section 5, Figure 21 was originally designed with bearing offsets and bearing reactions per Section 5, Figure 22 and 23 respectively.

To illustrate importance of hull deflections, we first investigate a conventional shaft alignment design where hull deflections are not considered. The design results are satisfactory for the considered condition, namely a hot main engine and a *100%* immersed propeller.

FIGURE 22
Bearing Offset; Shaft Deflection Curve; Nodal Slopes

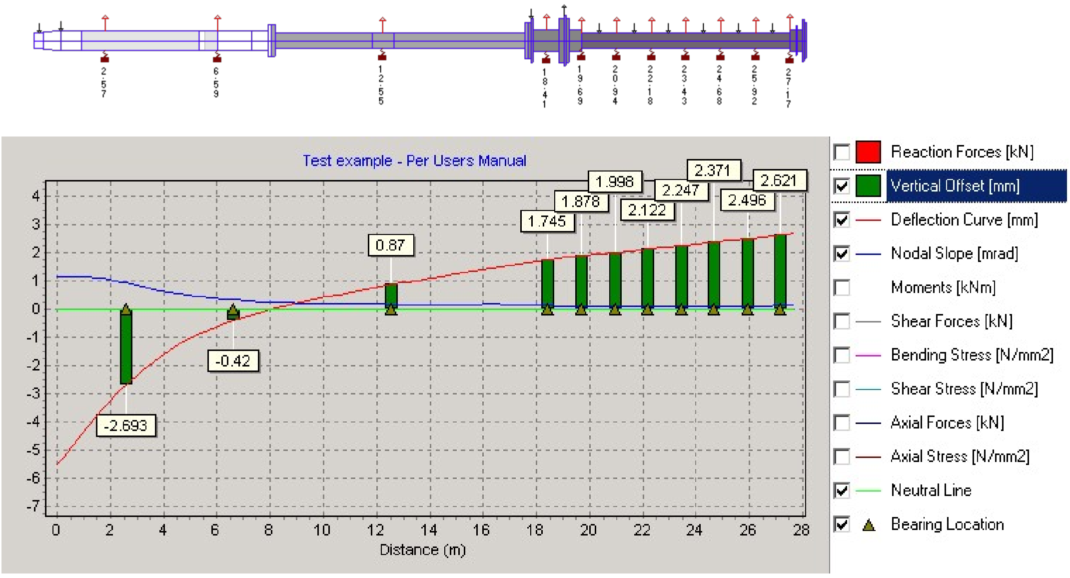
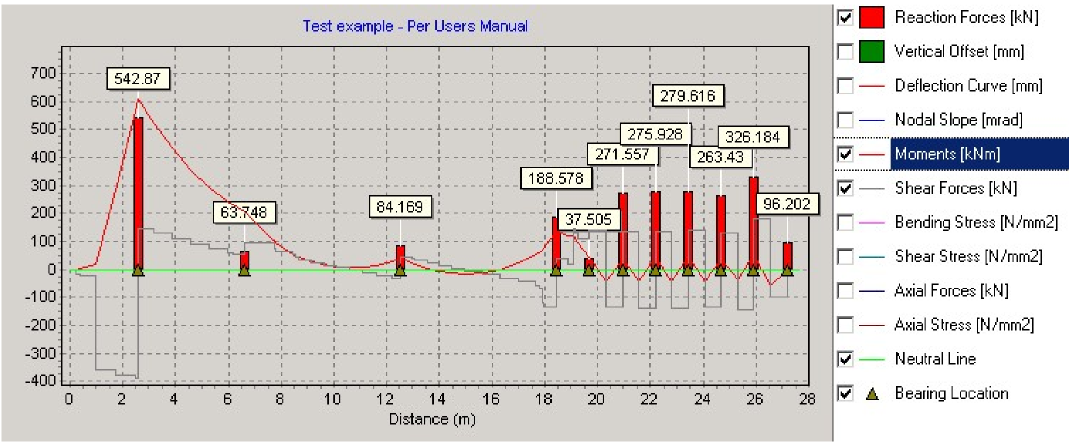


FIGURE 23
Bearing Reactions; Bending Moment; Shear Forces



As was done in the earlier container vessel example, we verify the proposed conventional design by applying the hull deflections defined in Section 5, Table 1.

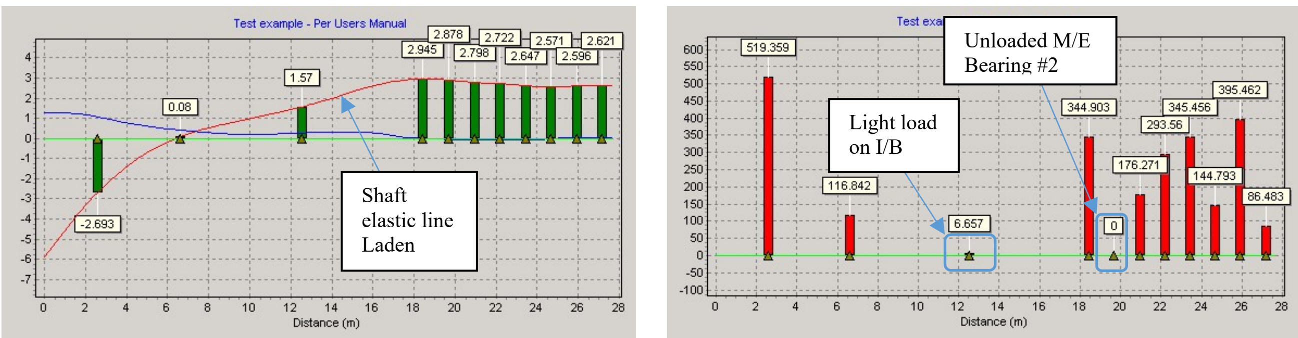
TABLE 1
Estimated Hull Girder Deflections

	Hull Deflection Estimate [mm]										
Brg #	1	2	3	4	5	6	7	8	9	10	11
Laden	0	0.5	0.7	1.2	1	0.8	0.6	0.4	0.2	0.1	0
Ballast	0	-0.05	-0.07	-0.12	-0.1	-0.08	-0.06	-0.04	-0.02	-0.01	0

The aim of this excise is to first check if the originally proposed offset design will be satisfactory for the vessel's service drafts as well. Secondly, we will conduct design optimization and propose a bearing offset which will be optimized for the vessel in service (for all drafts between the ballast and the fully laden vessel).

FIGURE 24
Laden – Bearing Offset Disturbed by Hull Deflections

Laden vessel hull girder deflections from Section 5, Table 1

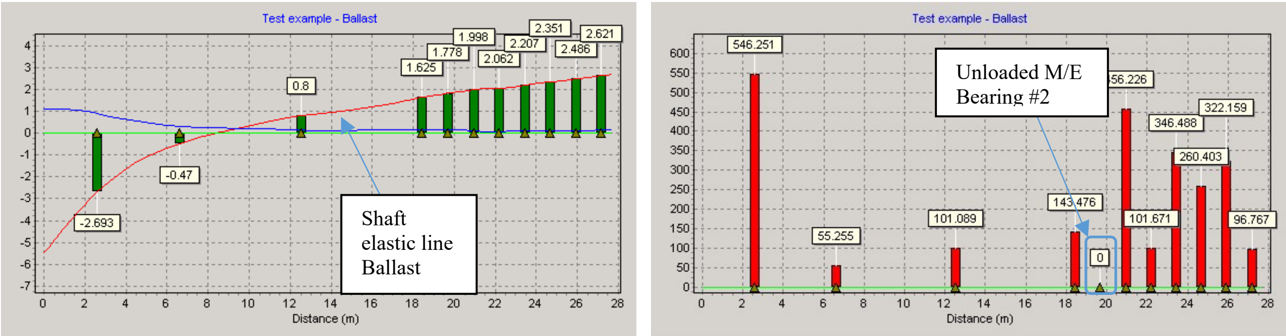


Total bearing offset

Bearing reactions: M/E second aftmost bearing unloaded, intermediate shaft bearing very lightly loaded

FIGURE 25
Ballast – Bearing Offset Disturbed by Hull Deflections; Bearing Reactions – Unloaded M/E Bearing #2

Ballast vessel hull girder deflections from Section 5, Table 1



Total bearing offset

Bearing reactions: M/E second aftmost bearing unloaded

As can be seen in Section 5, Figures 24 and 25, this analysis demonstrates that the initially prescribed offsets now no longer satisfy the alignment condition. The second aftmost main engine bearing appears unloaded both for the ballast and the laden condition. Static unloading of the second aftmost main engine bearing is critical for the engine's performance and may result in bearing failure due to crankshaft pounding load from cylinder combustion forces.

4.3 Optimization

The above analysis suggests that a different set of initially prescribed offsets is required to satisfy alignment requirements for in-service vessel conditions; the dry dock bearing offsets do not satisfy alignment when vessel is at ballast or laden draft. The reason for this discrepancy is the effect of hull deflections caused by a ballasted or laden vessel.

The Genetic Algorithm (*GA*) is utilized to optimize bearing offsets for given hull deflections as illustrated in Section 5, Figures 26 and 27.

FIGURE 26
Genetic Algorithm Optimization Results

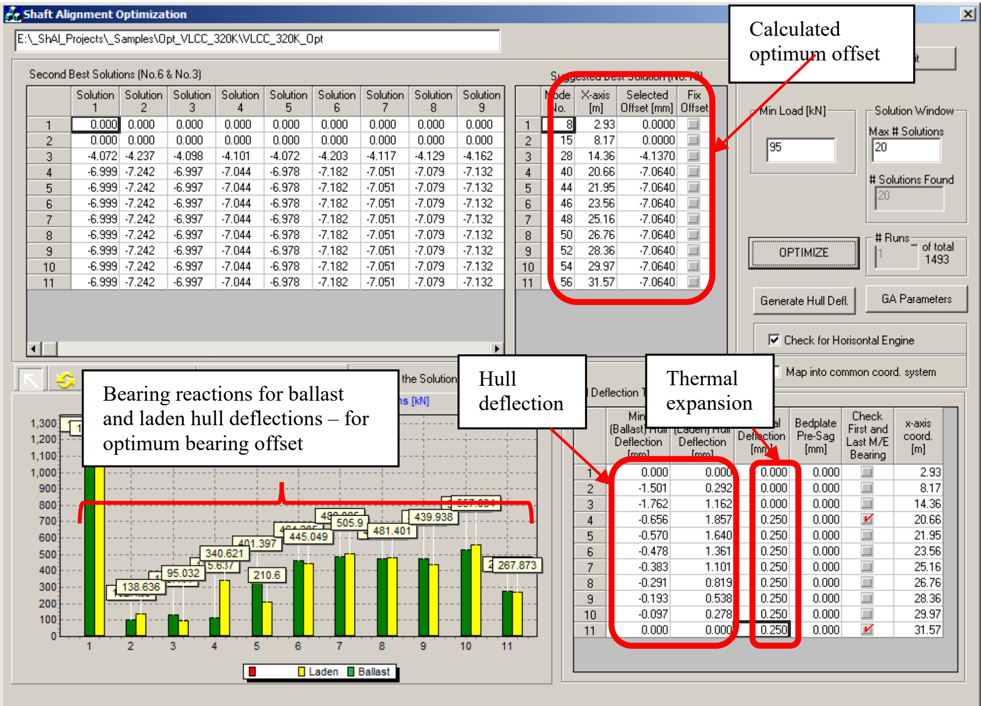
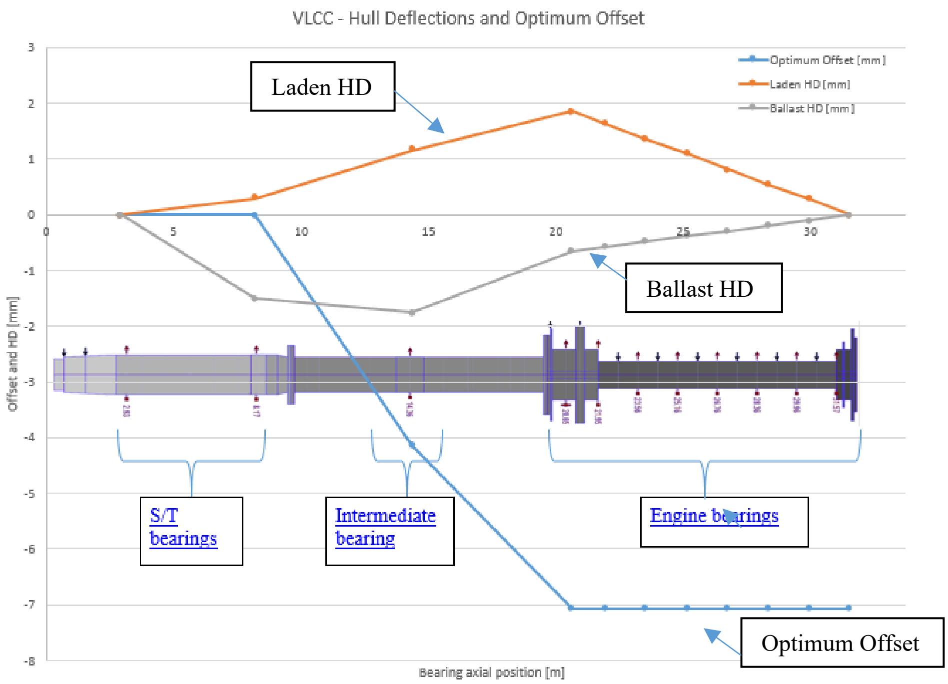


FIGURE 27
Genetic Algorithm Optimization Results Chart



Selected optimum offset and reactions are summarized in Section 5, Table 2, along with additional useful information for design evaluation.

The orange curve (Laden HD – Max) in Section 5, Figure 27 represents the hull deflections for a laden condition, while the grey curve (Ballast HD – Min) represents those for a ballast condition. The blue curve (Optimum Offset) represents the offset for the dry dock condition, which is used as a benchmark to calculate the bearing offsets for the laden and the ballast condition

Laden bearing offset = *Total Offset Max* + *Min Hull Deflection*

Ballast bearing offset = *Total Offset Min* + *Max Hull Deflection*

Section 5, Tables 3, 4 and 5 illustrate the optimized bearing offsets and bearing reactions for the dry dock, ballast and laden conditions respectively. The values shown correspond with those summarized in Section 5, Table 2.

TABLE 2
Optimal Solution

Tabulated optimization results: best solutions selected from the pool of 20 satisfactory solutions

Optimization with Genetic Algorithm
SOLUTION No.(13) Generation: 186 String: 18 FITNESS: 1.100000

Sup. No	Node No	SUPPORT REACTIONS Ry[0] [kN]	delRy [kN]	Ry (Max.Offs) [kN]	Ry (Min.Offs) [kN]	Ry (dy) [kN]	Total Offset Max. [mm]	Total Offset Min. [mm]	GA defined dy [mm]	Max Hull Deflect. [mm]	Min Hull Deflect. [mm]	Thermal Offset [mm]	Engine PreSag [mm]
1	< 8>	1235.188	-138.104	1103.041	1129.257	1097.084	0.000	0.000	0.000	0.000	0.000	0.000	0.000
2	< 15>	-150.541	274.509	138.636	102.426	123.968	0.292	-1.501	0.000	-1.501	0.292	0.000	0.000
3	< 28>	282.802	-35.290	95.032	133.491	247.511	-2.975	-5.899	-4.137	-1.762	1.162	0.000	0.000
4	< 40>	390.951	-686.902	340.621	115.637	-295.952	-4.957	-7.470	-7.064	-0.656	1.857	0.250	0.000
5	< 44>	107.866	588.184	210.600	401.397	696.051	-5.174	-7.384	-7.064	-0.570	1.640	0.250	0.000
6	< 46>	486.794	-2.917	445.049	464.205	483.877	-5.453	-7.292	-7.064	-0.478	1.361	0.250	0.000
7	< 48>	476.312	0.633	505.900	488.085	476.945	-5.713	-7.197	-7.064	-0.383	1.101	0.250	0.000
8	< 50>	481.445	-0.137	481.401	472.261	481.308	-5.995	-7.105	-7.064	-0.291	0.819	0.250	0.000
9	< 52>	467.161	0.030	439.938	473.320	467.191	-6.276	-7.007	-7.064	-0.193	0.538	0.250	0.000
10	< 54>	530.945	-0.006	557.034	528.119	530.939	-6.536	-6.911	-7.064	-0.097	0.278	0.250	0.000
11	< 56>	276.203	0.001	267.873	276.926	276.204	-6.814	-6.814	-7.064	0.000	0.000	0.250	0.000

Explanation of the parameters in Section 5, Table 2.

- *Ry[0]* - Bearing reactions for zero offset (even keel)
- *delRy = Ry[0] – Ry(dy)* - Reaction difference between zero offset and the dry dock offset
- *Ry(Max.Offs)* - Laden bearing reactions
- *Ry(Min.Offs)* - Ballast bearing reactions
- *Total Offset Max.* - Laden bearing offset
- *Total Offset Min.* - Ballast bearing offset
- *GA defined dy* - Optimization-software generated bearing offset
- *Max Hull Deflect.* – Hull deflection for laden vessel
- *Min Hull Deflect.* – Hull deflection for ballast vessel
- *Thermal Offset* – change in bearing offset caused by thermal expansion
- *Engine PreSag* – change in offset of the main engine bearings due to applied bedplate presag.

TABLE 3
Dry Dock – Bearing Reactions for Prescribed Offset

	Reactions	Dry-Dock Offset
	Ry (dy) [kN]	GA Defined Dy [mm]
1	1097.084	0
2	123.968	0
3	247.511	-4.137
4	0	-7.064
5	696.051	-7.064
6	483.877	-7.064
7	476.945	-7.064
8	481.308	-7.064
9	467.191	-7.064
10	530.939	-7.064
11	276.204	-7.064

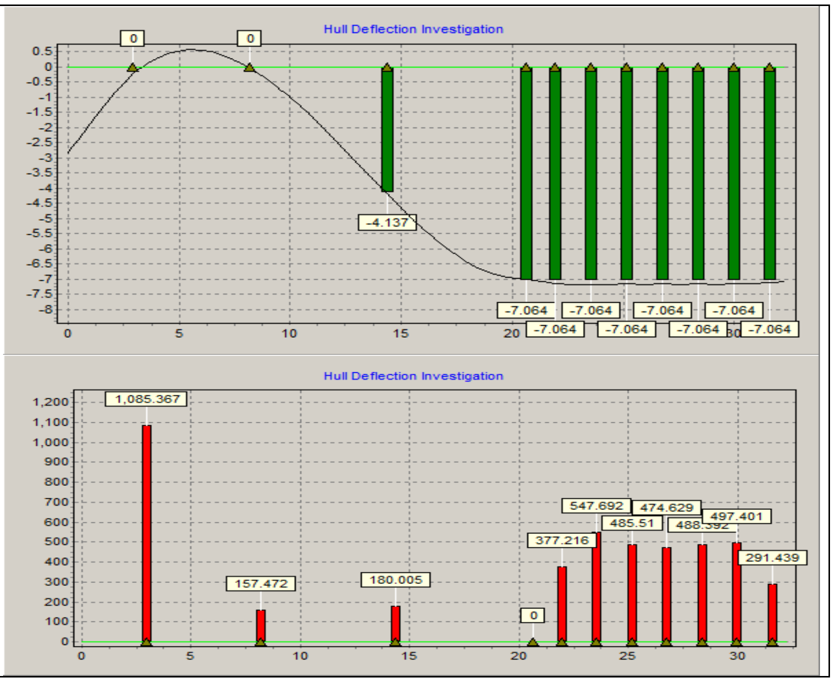


TABLE 4
Ballast – Bearing Reactions and Total Bearing Offset

	Reactions	Total Offset
	Ry (Min.Offs) [kN]	Ballast Hot [mm]
1	1129.257	0
2	102.426	-1.501
3	133.491	-5.899
4	115.637	-7.47
5	401.397	-7.384
6	464.205	-7.292
7	488.085	-7.197
8	472.261	-7.105
9	473.32	-7.007
10	528.119	-6.911
11	276.926	-6.814

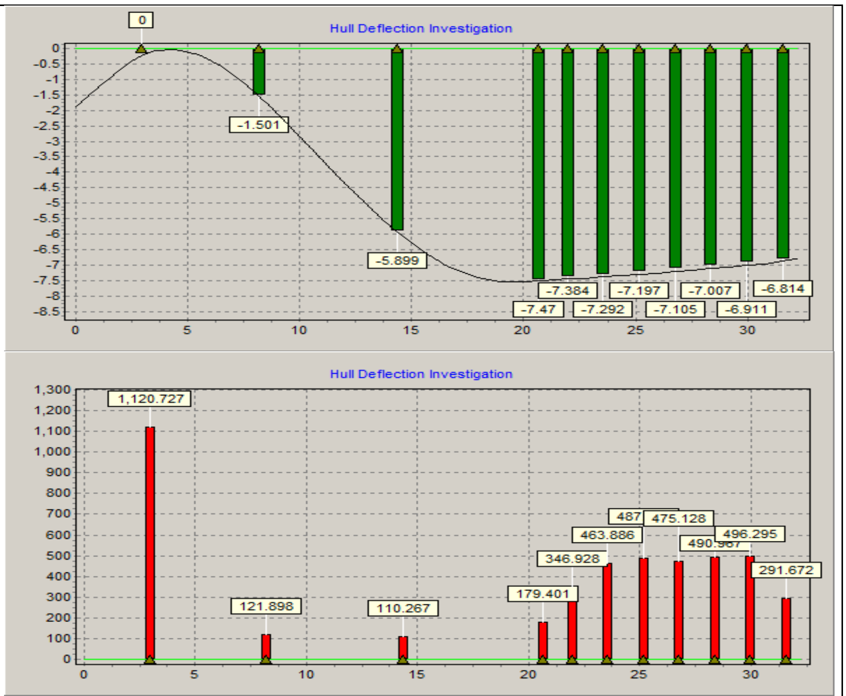
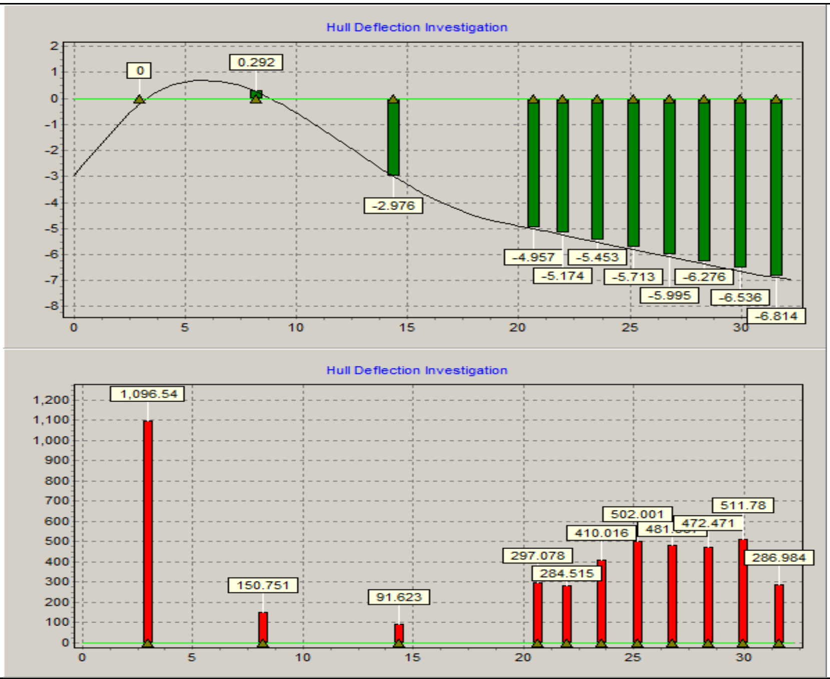


TABLE 5
Laden – Bearing Reactions and Total Bearing Offset

	Reactions	Total Offset
	Ry (Max.Offs) [kN]	Laden Hot [mm]
1	1103.041	0
2	138.636	0.292
3	95.032	-2.975
4	340.621	-4.957
5	210.6	-5.174
6	445.049	-5.453
7	505.9	-5.713
8	481.401	-5.995
9	439.938	-6.276
10	557.034	-6.536
11	267.873	-6.814



The optimized results in Section 5, Tables 4 and 5 indicate satisfactory bearing loads for the ballast and laden draft condition; the alignment for service drafts is acceptable and robust enough to prevent bearing unloading.

On the other hand, when reviewing the design for the dry dock condition in Section 5, Table 3, it is observed that the aft most main engine bearing is unloaded. This though, appears to be exactly in line with the engine maker's recommendation. For the dry dock, or the light ship afloat condition, before the engine is chocked, engine makers recommend adjustment of the intermediate bearing and the main engine offset, to unload or very lightly load the aftmost main engine bearing. This recommendation is based on the practical knowledge that the engine aft most bearing will pick up load as vessel draft increases. Our calculation with GA-optimized offset confirms the engine maker's recommendations are in line with design expectation when hull deflections are accounted for.

The benefits of conducting shaft alignment optimization are immediately obvious: the optimization algorithm applied in this analysis proves highly effective in finding a robust alignment solution in relatively short time.

SECTION 6 Alignment Best Practices

1 General

This Section addresses some less commonly applied industry practices, and reviews some already mentioned alignment design procedures that may benefit shafting performance and prevent shafting alignment problems:

- Shaft alignment condition monitoring
- Double slope design
- Stern tube bearing lubrication arrangement
- Adopting advanced techniques for calculation and verification
- Propeller loads and propulsion efficiency

1.1 Shaft Alignment Condition Monitoring

The industry standard practice is to monitor bearing performance by continuous temperature measurements and periodical lubricant analysis.

Conventionally, only one temperature sensor is installed on each bearing. For the aft stern tube bearing, being the most sensitive part of the propulsion system and inaccessible when in service, temperature measurement would benefit from more than one temperature sensor for redundancy reasons. The temperature measurement system is preferably designed with a power failure alarm and an alarm that indicates an open circuit, a short circuit, or an earth fault in the temperature sensing circuit.

Regular lubrication oil sampling is another important measure for bearing performance evaluation. It is a good practice to take oil samples under service conditions with a rotating shaft and at service temperatures and analyze these monthly for water content by using a special test kit.

Additionally, it may be advisable to analyze oil samples every six months for the following:

- Free water, *Mg* & *Na*
- Metallic elements (*Pb, Fe, Cu, Cr, Sn, Si, Ni*)
- Viscosity at 40°C

An increase in stern tube lubricant consumption is yet another indicator that something is amiss with the bearings and it should be monitored and recorded at regular intervals.

Record keeping is important for evaluating a change in the bearing condition. Comparing successive oil samples can indicate potential problems or trends when parameter values significantly change.

Although continuous temperature measurement is a very good indicator of bearing performance, it is an indirect bearing condition evaluation method that cannot always give ample warning to prevent bearing damage or failure. More direct methods that for example measure the clearance between the bearing and the shaft in real time (see Subsection 4/7), further enhance bearing performance monitoring capabilities. In combination with the temperature data, the direct measurement of bearing clearances provides a faster warning system that responds when clearance and temperature trends slide below the pre-established safety margins.

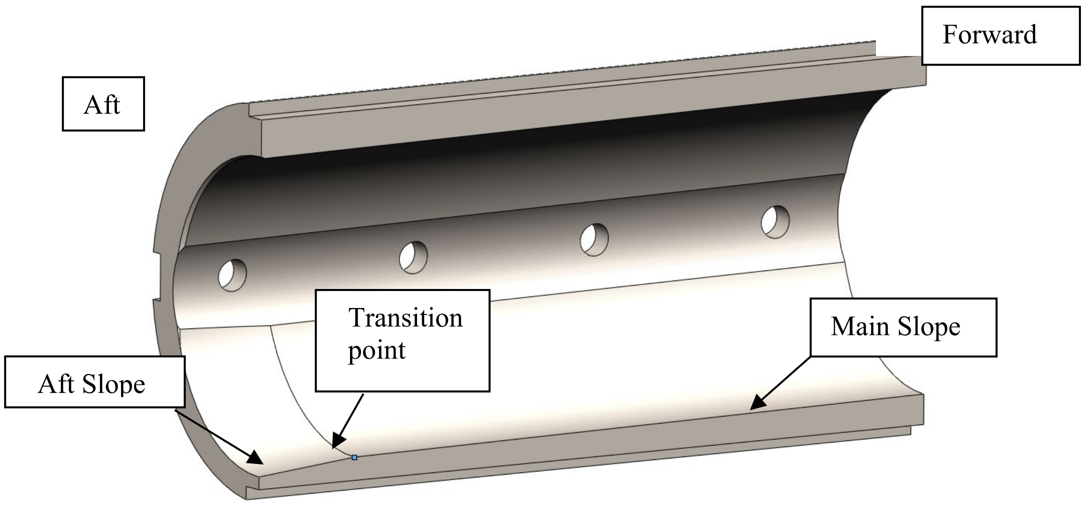
1.2 Double Slope Design

The aft stern tube bearing double slope design is a recognized improvement for some sensitive alignment installations, such as those with no forward stern tube bearings or twin-screw propulsions. This design was elaborated on in previous sections but, due to its recognized advantages, this section briefly summarizes once again the major benefits of the double slope design.

When a double slope design is such that in a static condition the shaft is sitting on the transition point between two slopes (the “knuckle”, Section 6, Figure 1) and very lightly contacting the forward and aft bearing edges, it can then present the following benefits:

- Alignment calculation can be more precisely conducted, because the resting point of the shaft in the bearing is precisely known
- Initial high bearing load on the transition point results in shaft bedding-in whereby the contact area expands forward and aft of the transition point
- Two oil wedges support the shaft in its transition from the static to the running condition

FIGURE 1
Example of a Double Slope Design

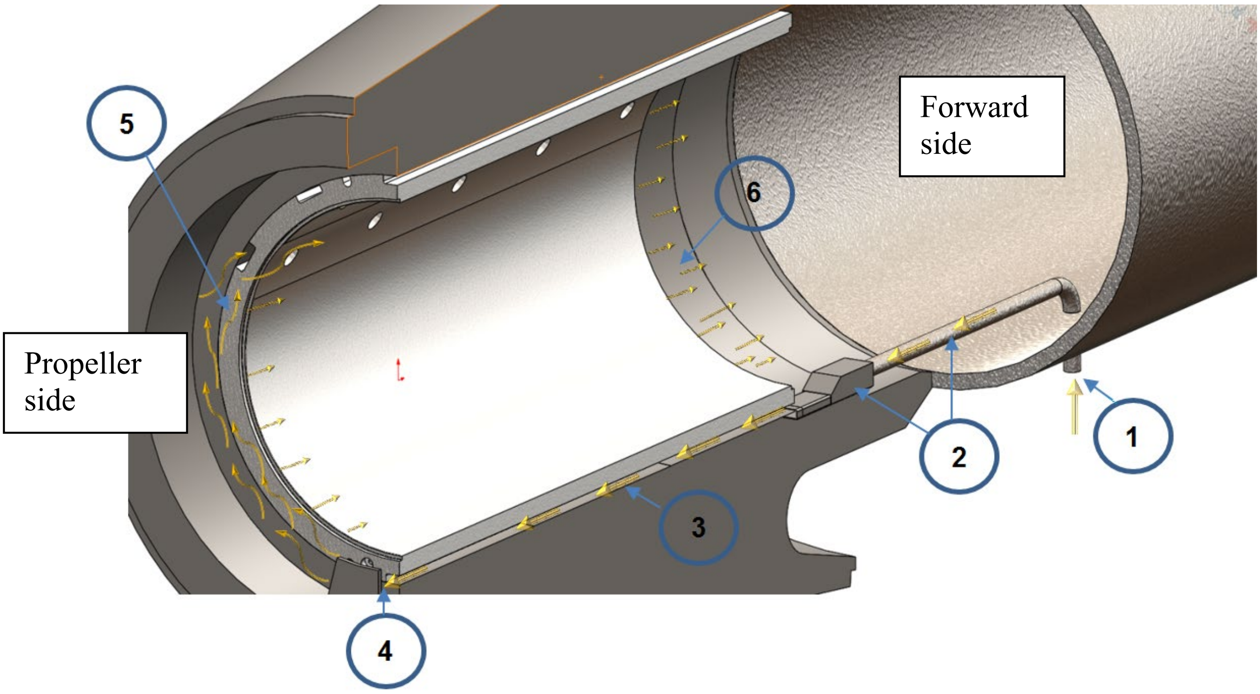


1.3 Stern Tube Bearing Lubrication Arrangement

In the conventional stern tube bearing lubrication design, the oil enters the stern tube between the aft and the forward bearing. To cool the aft stern tube bearing, especially its hottest area at the aft edge, fresh lubricant has to flow though the bearing from front to aft, and then back from aft to front to exit the bearing. It is obvious that the cooling effect in such an arrangement may not be as efficient as when fresh oil is continuously streamed through the bearing.

Stern tube bearing lubrication can benefit from a design where the fresh oil inlet is located at the aft end of the aft stern tube bearing (Section 6, Figure 2).

FIGURE 2
Example of Lubricant Inlet at the Aft End of the Stern Tube Bearing



Section 6, Figure 2 shows the cross section of the aft bush with the fresh lubricant inlet at the aft end of the bearing. The only difference from the conventional oil inlet between the aft and the forward bearing is that an extra pipe with a converter box (2) is installed. This modification provides the way for fresh oil to flow into the bottom groove of the bearing, and further to the aft end of the bush.

The oil flow in Section 6, Figure 2 is depicted by yellow arrows. For clarity the shaft is not shown. Oil flow is only possible through the clearance between the shaft and the bearing and through the oil grooves. As fresh oil enters the stern tube (1), it is distributed to the bottom oil groove (3) through an appropriately sized pipe and a converter box (2). Oil exits at the bottom aft end of the bearing bush (4), which is normally the hottest area in the bearing. A deflecting plate (4) may be welded in front of the oil entry in the recess between the bearing bush and the shaft seal (5), to prevent exposing the seal to the direct oil stream. Fresh oil is dragged by the rotating shaft and streamed through the bush from the hottest area in the aft towards the cooler front (6) and then further to the forward bush.

This arrangement provides a more efficient heat exchange and can reduce possible overheating of the lubricant by a continuous influx of fresh oil, flushing the bearing through from the aft towards the front.

1.3.1 Lubricant Properties

Stern bearings are lubricated either by fresh or sea water or by high viscosity oils. The properties of the applied lubricant are crucial for bearing performance and the lubricant should be identified in very initial stage of the alignment design.

Water

Water is the only fully environmentally friendly lubricant. This permits water lubricated bearings to be of the open type without seals. The application of water as a lubricant is normally limited to vessels outfitted with composite material or rubber bearings.

Because water has a very low viscosity, it flushes through the bearing faster and therefore has a higher cooling rate. The downside of this low viscosity, though, is that water may not provide a sufficient hydrodynamic lift and thus not effectively support the shaft under high propeller loads.

Abrasion and corrosion on open type bearings are of concern for their operational life expectancy. This can be addressed by adequate abundant flushing and utilizing non-metallic bearing materials. Special arrangements with closed, treated water circuits are sometimes considered. However, in general the life span of water lubricated bearings is mostly less than that of oil lubricated ones.

Oil

Oil is the most commonly used lubricant. Oils of different types and composition are manufactured to suit specific applications.

Oil viscosity is essential for bearing performance and is therefore one of its most important characteristics to consider. Selection of the most suitable viscosity is crucial. Higher viscosity oils have a better carrying capacity and can sustain higher loads, but they have a higher internal friction that may generate more heat.

A different oil viscosity may be considered for vessels operating in colder waters than those operating in warmer waters. Overheating or water diffusion can compromise viscosity and reduce the oil's carrying capacity. Water is therefore the foremost contaminant ship operators must closely monitor with scheduled lube oil tests.

Oils are considered pollutants. Some countries demand special stern tube lubrication arrangements or the application of environmentally acceptable lubricants - so called bio-oils - on vessels trading in their coastal waters.

1.4 Advanced Design Techniques

It may be advisable to conduct a more elaborate and detailed alignment analysis for new ship designs that are expected to be alignment sensitive (e.g., a large twin-screw propelled vessel).

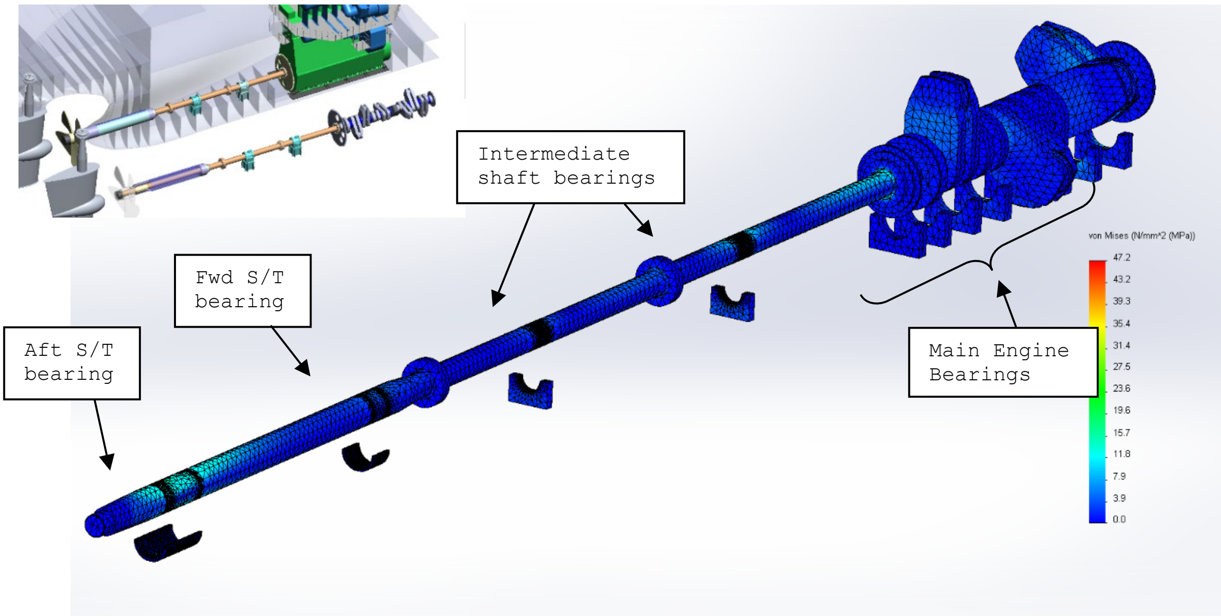
The application of a 3D FE model would be the preferred approach for alignment sensitive installations in addition to the regular shaft alignment calculations. An FE model enables a detailed evaluation of the contact and misalignment condition of all bearings in the system, as well as crankshaft deflections, and in the case of geared systems, also gear mesh misalignment.

Normally, a dynamic condition evaluation of the stern tube bearing performance is conducted by employing CFD calculations of the propeller loads and rudder turning actions, and an FSI analysis.

The propeller's wake field becomes increasingly asymmetrical during wide rudder movements, causing a difference in water velocities on either side as well as at the top and bottom of the propeller. These, in turn, have an effect on the moments and hydrodynamic forces acting on the tail shaft. Localized loads at the aft end of the stern tube and toward one side will temporarily increase and could cause bearing damage or failure. A clockwise-rotating propeller, for example, undergoes the worst loading on the aft stern tube bearing with a hard-over starboard rudder turn and full vessel speed at maximum RPM.

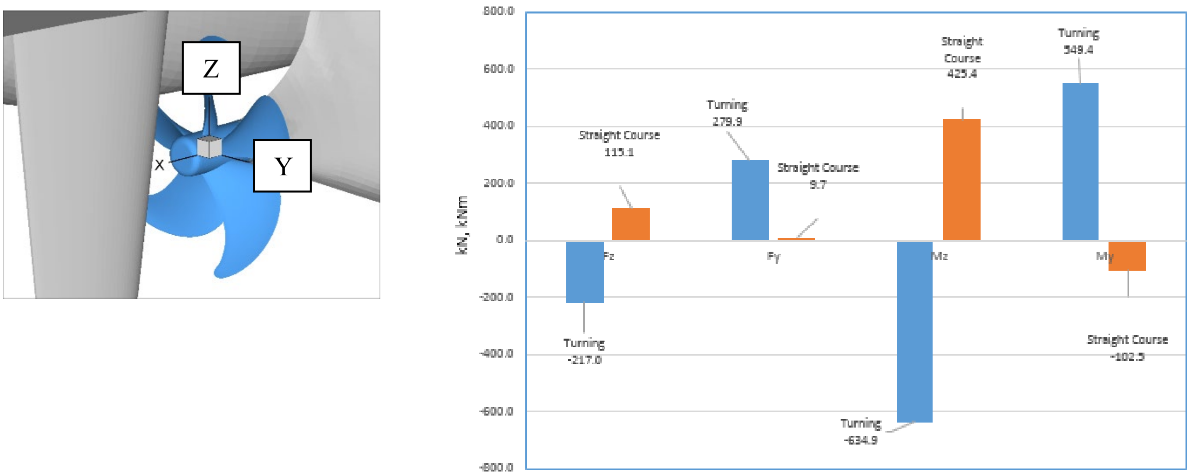
An example of the 3D FE model is illustrated in Section 6, Figure 3. This particular model is created for a twin-screw vessel. Twin-screw vessels are known to be more sensitive to alignment issues due to the much greater compounded propeller load, particularly when the vessel is turning.

FIGURE 3
FE Model for Crankshaft Deflection and Stern Tube Bearing Contact Evaluation



By taking advantage of the symmetry of the structure, the investigation can be performed by modeling only one propulsion shafting assembly and then applying appropriate loads for the twin propulsion.

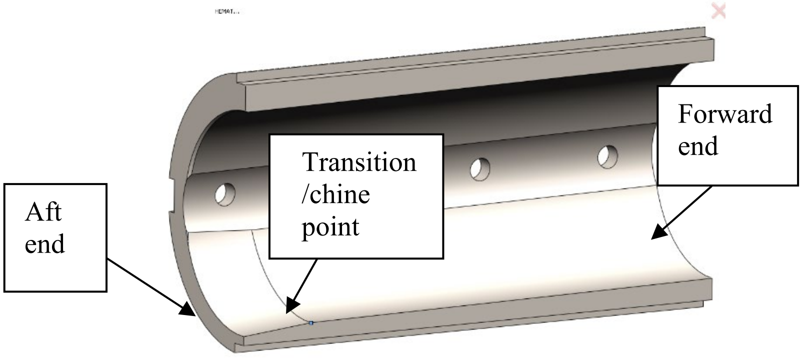
FIGURE 4
Propeller Load Comparison: Straight Ahead vs. Turning



Section 6, Figure 4 shows an example of the propeller load comparison between a straight ahead run and a turning maneuver on a twin-screw vessel.

CFD calculation of propeller loads indicate that the Z-axis vertical loads and bending moment around the transverse Y-axis are significantly higher for the turning condition. More significantly, the turning loads result in a downward push adding to the gravity loads exerted by the combined weight of shaft and propeller. In twin-screw installations with counter rotating propellers, one shaft is always pushed down during turning, regardless of turning direction, to either port or starboard.

FIGURE 5
Example of a Double Slope Design

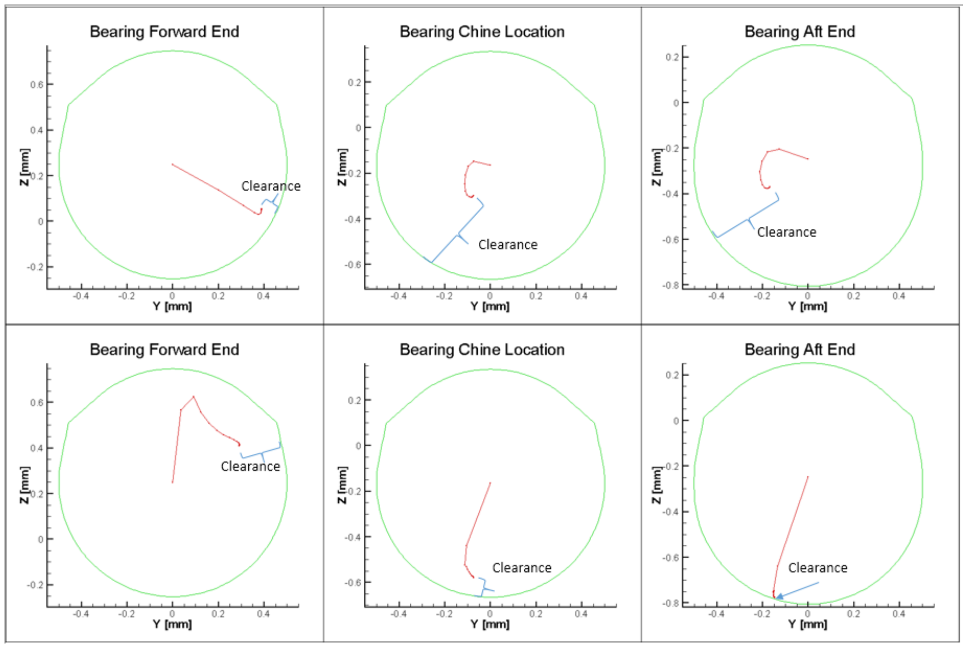


Section 6, Figure 6 shows FSI dynamic performance analysis results conducted with propeller loads for turning and a straight ahead run on a twin-screw vessel with a double slope bearing design similar to that shown in Section 6, Figure 5. The results clearly illustrate why the turning condition may be potentially critical. FSI calculation also facilitates optimization of a static double slope design. For additional details on FSI optimization, see 2/4.5.1.

FIGURE 6
FSI Calculation Results for Port Shaft for Straight Run and Starboard Turn

Straight ahead run

Starboard turning



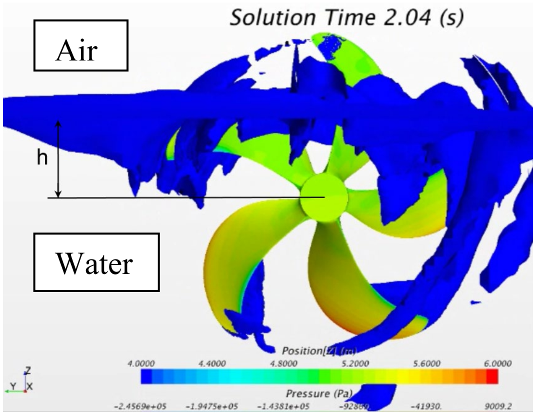
1.5 Propeller Loads and Propulsion Efficiency

1.5.1 Partly Immersed Propeller

The service condition of the vessel is defined by vessel's loading manual. The ship's propeller is fully immersed under normal service conditions. Occasionally when transiting through shallow waters such as rivers and canals, the vessel is de-ballasted and trimmed to prevent grounding. This is likely to result in the propeller not being fully immersed.

A partly immersed propeller may be unfavorable to shaft alignment and in particular to the aft stern tube bearing. Propeller loads in such cases are significantly greater and much more unstable when compared with a fully immersed propeller. Therefore, vessel operation with a partly immersed propeller is recommended to be conducted at low vessel speeds and with maneuvering limited to the minimum rudder angles necessary for safe navigation. Particularly sensitive installations are those with no forward stern tube bearing and the twin-screw designs.

FIGURE 7
Partly Immersed Propeller



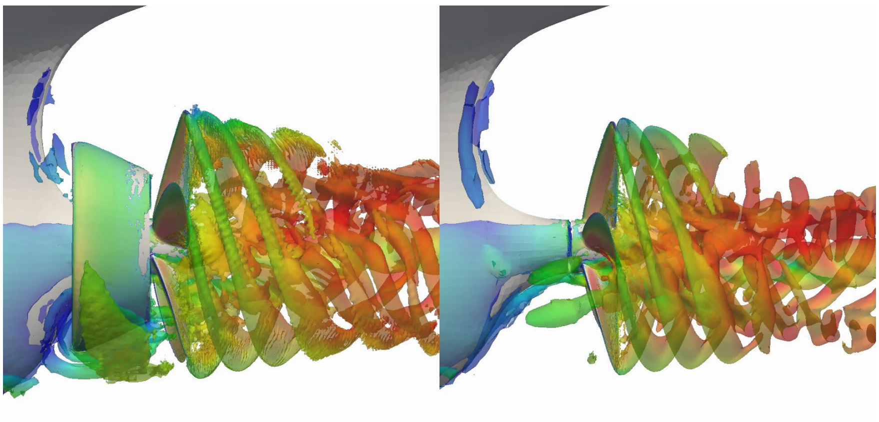
	Fully Immersed Draft 11.5	Partly Immersed Draft 6.31
Fx -thrust	1.877e6	1.606e6
Fy –transverse	-4.945e4	-7.120e4
Fz – vertical	1.713e5	1.154e5
Mx – torque	-2.539e6	-2.249e6
My – vertical	-3.213e5	-1.008e6
Mz - transverse	4.567e5	-6.837e4

The Table in Section 6, Figure 7 provides a comparison of CFD calculated propeller loads for a fully immersed and a partly immersed propeller. Although the calculated vertical load appears to be approximately *48%* smaller, the moment in vertical plane of a partly immersed propeller is calculated to be over three times larger than for a fully immersed propeller.

1.5.2 Energy Saving Devices

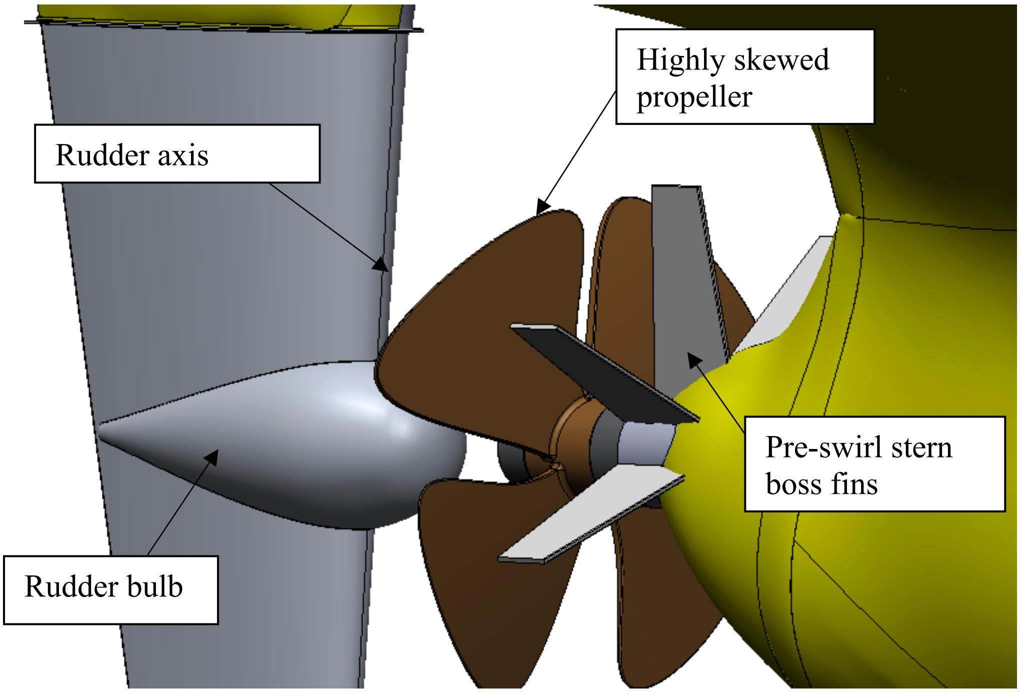
Energy saving devices (ESD) are often introduced to increase propulsion efficiency. Some ESD designs such as the addition of a nozzle shown at the left of Section 6, Figure 8, may result in beneficial alignment performance, with smaller down forces on the aft stern tube bearings. Although propeller load reduction may not be the primary purpose of ESD, it may be good practice to also investigate the impact of ESD on the alignment when deciding on particular design.

FIGURE 8
Energy Saving Devices



In conclusion, Section 6, Figure 9 illustrates possible design improvements that can be implemented on the propeller, stern boss, and rudder to increase propulsion efficiency.

FIGURE 9
Efficiency Improvements



SECTION 7 Glossary

1 Abbreviations

ABS	American Bureau of Shipping
A/D	Analog/Digital converter
CFD	Computational Fluid Dynamics
C/S	Crankshaft
ESD	Energy Saving Device
FE	Finite Element
FEM	Finite Element Model
FSI	Fluid Structure Interaction
G/B	Gearbox
GA	Genetic algorithm
GUI	Graphical users interface
I/B	Intermediate shaft bearing
I/S	Intermediate shaft
ITP	inspection and test plan
M/E	Main engine; implies diesel engine if not stated differently
NFSB	No Forward Stern tube Bearing; refers to a propulsion system that does not have a forward stern tube bearing
S/T	Stern tube
TDC	Top dead center – defines position of the piston in the engine cylinder.
T/S	Tail shaft

2 Definitions

Analytical model		A mathematical representation of the physical powertrain system created for the purpose of shaft alignment analysis and calculation.
Beam		A structural component designed to support loads acting upon it, primarily by resisting against bending.
Beam theory		A special case of the general theory of elasticity, defining a mathematical basis for analysis of beam strength; based on the beam material properties, geometry, support location, and loads acting upon the beam.
Bearing clearance	*mm (in.)*	Radial gap between the shaft and the surrounding bearing shell.

Bearing inclination	*rad*	The process of installing a stern tube bearing under a prescribed angle and fixing it in place by means of epoxy resin. An alternative to slope boring.
Bearing load	*N (lb)*	Originates from the static and dynamic forces and moments acting upon a bearing.
Bearing offset	*mm (in.)*	Vertical displacement of the contact face of the bearing from the optically established centerline of the shafting.
Bearing reaction	*N (lb)*	The resisting force to applied action; the force measured to confirm bearing load as part of the shaft alignment process.
Bearing slope	*rad*	The angle defined in the shaft alignment calculation under which the bearing is machined or inclined. A bearing slope is applied primarily on aft stern tube bearings. The angle is measured with reference to a centerline. Some shaft alignment designs may result in zero or no slope on the stern tube bearing.
Bearing span	*m (ft)*	The horizontal distance between the centers of two adjacent bearings.
Bedplate pre-sagging	*mm (in.)*	Process applicable to large engines by which the engine is supported on four corner points and is allowed to sag freely between these points; the vertical deformation introduced on the engine's bedplate compensates for the engine's uneven thermal rise and counteracts alignment problems resulting from hull deflections.
Bore sighting		See "sighting-through"
Builder		Shipyard
Bureau		American Bureau of Shipping (ABS)
Chine		See "transition point"
Class		Classification society
Clearance	*mm (in.)*	The gap between two adjacent objects. For the purpose of shaft alignment see bearing clearance definition.
Cold static condition		A vessel's condition with the main engine and shafting components at ambient engine room temperature and not rotating.
Coupling bolts		Fasteners that hold together the flanges of the propulsion shafting.
Crankshaft deflections	*mm (in.)*	Change in distance between crank webs, measured during one rotation of the crankshaft.
Dial gauge		A tool used for measuring small displacements and shaft run-out; available in analog or digital models.
Double slope		A bearing design with two distinct adjacent slopes. The point of change between two slopes is called a knuckle point, chine or a transition point.
Dynamic load		A load that exerts varying amounts of force upon a structure or object. When in motion, the propeller for example exerts dynamic loads on the propulsion shafting.
Equivalent crankshaft model		A simplified representation of the actual crankshaft that produces equivalent bending deflections under the same load applied.
Feeler gauge		A measuring tool consisting of a number of thin blades of calibrated thickness used to measure narrow gaps or clearances.

Final sighting		Verification of the slope boring angle, the main engine position, and the offset target points of the intermediate shaft bearing relative to the stern tube center line.
Hogging	*mm (in.)*	Upward or convex bending of a beam.
Horizontal offset	*mm (in.)*	Undesirable deviation of the bearing centerline in the horizontal (transverse) plane.
Hot static condition		A vessel's condition with the main engine shut down after it has been in operation. Machinery temperatures and shafting components are close to the running condition temperature.
Hull deflections	*mm (in.)*	Deformation of the hull girder, measured from the even keel line, under varying vessel operational conditions.
Hysteresis		Displacement between the lifting and the lowering line of a plotted jack-up curve. Caused by friction in the hydraulic jack and in the shaft material.
Influence coefficients	*N/mm (lb/in)*	Values defined by the ratio between the bearing reactions change and the unit offset change at particular bearing. Representing the 'stiffness' at a particular bearing or a jacking point on the shaft. Provides an understanding of system sensitivity, e.g., bearing reaction response attributed to changes in bearing offsets.
Intermediate bearing		A bearing, typically a plummer block type, designed to support the intermediate shafts.
Intermediate shaft		Shaft that connects the propeller shaft to the prime mover.
Jack-up procedure		Procedure that uses a hydraulic jack to measure bearing reactions; the result is a jack-up curve from which the bearing load is derived.
Journal bearing		A sliding bearing of cylindrical shape in which the journal of a cylindrical shaft, the "journal", slides and floats on a thin layer of lubricant.
Knuckle point		In bearing design, the transition between two slope angles.
Lifting/lowering line gradient	*mm/N (in/lb)*	Slope of the plotted jack-up line. See influence coefficients.
Lightship		The displacement of a bare structure of the vessel; no cargo, fuel, lubricating oil, ballast water, fresh water or feed water in tanks, consumable stores, and passengers or crew and their effects.
Line Shaft		Another term for intermediate shafting
Load cell		Equipment based on strain-gauge principle that converts a load acting upon it into a digital signal.
Main drive		Equipment where torque driving the propulsion shafting is generated (diesel engine, electric motor, or turbine).
Main engine bearing		A bearing supporting the engine's crankshaft.
Measurement accuracy		Closeness of a measured value to a standard or known value. Depends on measuring instrument accuracy and skill of the user.
Measurement tolerance		A predefined range within which a measurement is considered acceptable.
Misalignment angle	*rad*	See relative misalignment slope.

Negative offset	*mm (in.)*	Bearing vertical position below the referenced (zero) line.
Negative slope	*rad*	A bearing slope in which the forward end of the slope is lower than the aft end.
No forward stern tube bearing		A propulsion system that does not have a forward stern tube bearing; see single stern tube bearing installation.
Offset	*mm (in.)*	The distance by which an object is displaced from its referenced position. In shaft alignment, the offset is defined by the distance a bearing is raised or lowered respective to the established reference line; see bearing offset.
Poker gauge	*mm (in.)*	A wear-down gauge to measure propeller drop and aft stern tube bearing wear. Tool is provided by the manufacturer of the shaft seal assembly.
Positive offset	*mm (in.)*	Bearing's vertical position above the reference or zero line.
Powertrain		A set of revolving elements transmitting torque from the source (prime mover) to the consumer (propeller).
Prescribed displacements	*mm (in.)*	Desired bearing offset prescribed by designer to obtain satisfactory alignment.
Pre-sighting		A sighting through the stern tube before the bearings are installed to define the center reference line of the stern tube bearings.
Prime mover		On a ship, usually a diesel engine, turbine, or electric motor that generates the power to move the ship.
Propeller shaft		The aftmost located shaft of the propulsion system that penetrates the vessel's hull and to which the propeller is attached.
Propulsion shafting		Solid or hollow line of shafts that provide connection between the prime mover and the propeller.
Relative misalignment slope	*rad*	The difference between the shaft angle (defined by two points on the shaft's centerline – located above the two end points of the bearing), and the slope of the bearing.
Rule of thumb		A method established, or a procedure derived entirely from practice or experience, without any basis in scientific knowledge; a rough practical method.
Rules		ABS *Rules for Building and Classing Steel/Marine Vessels* are implied if not stated differently.
Sag and Gap	*mm (in.)*	A verification process of the alignment condition before shafting assembly.
Sagging	*mm (in.)*	Downward or concave bending of a beam.
Service draft	*m (ft)*	A draft between the ballasted and the fully laden vessel for which its structure and machinery are designed.
Shaft alignment procedure		The method used during installation of bearings, shafts, propeller, prime mover and other equipment, in accordance with the designer's requirements.
Shaft alignment process		The combination of design and analysis, shaft alignment procedures, and measurement.
Shaft run-out	*mm (in.)*	A measurement of a maximum radial displacement of the shaft during one full shaft rotation.

Sighting through	Optical procedure by which bearings are offset to values prescribed in the calculation, and slope bored or inclined, if required.
Single stern tube bearing installation	See no forward stern tube bearing.
Slope boring	Procedure by which the stern tube or the stern tube bearing is machined under an angle to comply with calculated misalignment requirements.
Static condition	For the purpose of shaft alignment, it refers to the shafts being in a stationary mode to allow shaft alignment related activities to be conducted.
Static load	A constant force applied to a stationary object, for example, the weight of a shaft on a bearing.
Stern tube bearing	Also known as tail shaft bearing or stern tube bush. Installed inside the stern tube to support the tail shaft. Some configurations have an aft and a forward stern tube bearing, while others may not have a forward stern tube bearing installed.
Stern tube bearing run-in procedure	A test to progressively expose the stern tube bearing to an increased load and accelerate the bedding-in process of the tail shaft in the bearing in a controlled manner.
Stern tube boring	The process of machining the stern tube casting in situ using either a horizontal or a vertical boring rig.
Stern tube bush	See stern tube bearing.
Straight alignment shafting	Propulsion shafting supported by the bearings which are positioned on a straight centerline of the un-deformed shafting. Also called zero offset alignment.
Strain-gauge method	Method used to measure strain change in the shafting, which is then converted into shaft deflection, bending moments and stress.
Tail shaft	See propeller shaft.
Tail shaft bearing	See stern tube bearing.
Tail shaft bottom clearance	Measured between the propeller shaft and the bottom of the aft S/T bearing. For clearance verification the propeller must not be fitted and shaft should be unrestrained on the front end.
Transition point	In shaft alignment with double or multiple sloped stern tube bearing, it refers to the location inside the bearing where one slope ends and a next one begins; also called a knuckle point or chine.
Undeformed shafting	Shafting with a straight central line. This assumes that no gravity and no external forces or moments are acting on the propulsion shafting system.
Vertical offset	See bearing offset.
White metal	Also known as Babbitt metal. A tin-based composite alloy used for journal bearings.
Zero offset alignment	See straight alignment shafting.

Note: All definitions are based on the ABS adopted sign convention (2/3.2).