NR592 DT R02 JULY 2025

ELASTIC SHAFT ALIGNMENT(ESA)

NR592 - JULY 2025

RULE NOTE

BUREAU VERITAS

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RULES, RULE NOTES AND GUIDANCE NOTES

NR592 DT R02 July 2025 takes precedence over previous revision.

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These rules are provided within the scope of the Bureau Veritas Marine & Offshore General Conditions, enclosed at the end of Part A of NR467, Rules for the Classification of Steel Ships. The latest version of these General Conditions is available on the Bureau Veritas Marine & Offshore website.

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NR592

ELASTIC SHAFT ALIGNMENT (ESA)

July 2025

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Section 1 General

1 Scope

1.1 General

1.1.1 The present Rule Note provides specific requirements and methodology for shaft alignment assessment onboard large ships, guidelines for calculation in view of hull deflection, aft steelwork elasticity, oil film behaviour and shaft bearing stiffness as well as the related requirements.

1.1.2 The present Rule Note applies for ships fitted with oil lubricated shaft line bearings only.

1.1.3 The present Rule Note does not apply to ships designed with azimuthal thrusters or non conventional shaft lines intended for main propulsion.

1.2 Definitions

1.2.1 Elastic Shaft Alignment (**ESA**) means the iterative calculation method for shaft alignment described in this Rule Note.

1.2.2 FE stands for Finite Element.

1.2.3 Sea swell is defined as global regular and long period waves not created by local weather conditions.

2 Application

2.1 Class notation

2.1.1 Ships complying with the requirements of the present Rule Note may be assigned the additional class notation **ESA**, in accordance with NR467, Pt A, Ch 1, Sec 2, [6.25.8].

Note 1: NR467 Rules for the Classification of Steel Ships.

2.2 Assignment and Maintenance of ESA additional class notation

2.2.1 Initial survey

Prior to the assignment of **ESA** additional class notation, it is to be demonstrated that the onboard shaft alignment matches the system description and input parameters listed in the approved calculation report.

2.2.2 If a modification implemented onboard a ship assigned **ESA** additional class notation has an influence on the input parameters or values of the requested results listed in Sec 2 of this Rule Note, **ESA** notation may be suspended until a detailed description of the performed actions and an updated calculation report showing compliance with applicable requirement are submitted to, and approved by, the Society

3 Documentation

3.1 Documentation to be submitted

3.1.1 A calculation report is to be submitted for approval. It is to contain items listed in Tab 1.

The Society may require additional drawings or documentation if deemed necessary.

3.1.2 Details of input parameters to be considered and required technical data within the scope of **ESA** notation are listed in Sec 2.

Table 1 : Data to be submitted

No.	Item
1	General description of calculation method
2	Assumptions
3	List of investigated calculation conditions
4	Input parameters
5	Detailed results
6	Conclusions

No.	Item
7	Shaft line model
8	Hull flexibility matrix
9	Hull relative deformations
10	View of complete ship FE model
11	Detailed views of FE model of aft part of ship structure
12	Detailed alignment procedure (see Sec 2, [6])

Section 2 Elastic Shaft Alignment (ESA)

1 Overall methodology

1.1 General

1.1.1 Scope and objective of shaft alignment

Shaft alignment for main propulsion of ships mainly refers to rigid and low speed parts of their line shafting. Studied systems therefore depend on propulsion type installed onboard, Tab 1 presents common main propulsion types found on large ships and related shaft alignment scope.

Stern bearings machining and alignment, as well as other bearing offsets, are to be optimized in order to reach the most favourable load distribution for relevant operating conditions.

Prime mover or gearbox is to be positioned to get acceptable loads on each support, and to anticipate thermal expansion and hull deformation effects.

Table 1 : Propulsion types and shaft alignment systems

Propulsion type	Prime mover	Alignment system
Direct drive installation	Low-speed diesel/gas engine	from propeller to crankshaft
	Electric motor	from propeller to rotor shaft
Geared drive installation	Medium-speed diesel/gas engine	from propeller to main gearbox output shaft
	Steam/gas turbine	
	Electric motor	

1.1.2 Definitions

a) Elastic alignment calculations consider the relevant line shafting system and its supports (see [1.1.1]). For each declared operating condition, the offsets of supports and the loading of the line shafting are investigated.

b) Offsets are the initial vertical and horizontal positions of a bearing fixed by the alignment procedure and modified by the flexibility of the structure, the loading deformations and the thermal expansion.

c) The equilibrium of flexible beam elements subjected to the external forces and supported by bearings is calculated in three dimensions. This means that vertical and horizontal displacements are coupled.

d) These elastic alignment calculations to be submitted are necessary to optimize the aft stern tube bearing slope and the partial slope, if needed. It should be possible to ensure correct oil film build-up by investigating shaft location with respect to the oil grooves for declared running conditions.

1.1.3 Vibration

These elastic alignment calculations could be supplemented by a global whirling calculation of line shafting and ship structure which are connected through the oil film stiffness and damping.

The scope of the present Rule Note does not cover vibratory assessment of ship design and its propulsion line(s).

The Society may require the above whirling calculation on a case by case basis.

1.2 Calculations

1.2.1 Required input parameters

The conditions of calculation should be as close as possible to the real operating conditions. The following input parameters are to be considered:

- deformations of the ship structure with respect to the declared loading conditions of the ship: light ship, ballast, full load, etc. (see [3.1])
- aft hull structure flexibility matrix calculated in way of each supporting point (see [3.2])
- propeller hydrodynamic efforts (forces and moments) in vertical and transverse directions in straight course
- thermal expansion of supports (seat, sleeve, antifriction material)
- deformation of prime mover or gearbox foundation: in case of low-speed engine, pre-sag of main bearings is to be considered.

1.2.2 Additional input parameters

Additional parameters could be considered for more precise calculations:

- deformation of ship structure with respect to the sea swell
- propeller hydrodynamic efforts in turning conditions, including rudder effects
- temperature effects on lubrication, by a calculation of local and global dissipation.

1.2.3 Calculation methods

The following calculation principles may be applied for elastic alignment studies:

- In static conditions: bearing reactions may be calculated with the Hertz contact theory.
- In running conditions: bearing reactions may be calculated by integration of the oil film pressure which is given by the Reynold's equations for a journal bearing.

Guidance on these two methods is given in App 1.

Other possible methods may be considered by the Society on a case by case basis.

Submitted report should precisely detail calculation process and the assumptions made.

1.2.4 Assessment

Assessment of the alignment conditions is based on approval of the output results listed in Tab 2, evaluated against the criteria listed in Tab 4 and Tab 6.

In addition, results are to be in compliance with the applicable manufacturer limits.

For new ships, guidance on alignment procedure is given in Article [6].

Table 2 : Results to be submitted

In static conditions:	In running conditions:
Reaction distribution between shaft bearings	
Reaction distribution along effective length of aft bush bearing	
Shaft location inside bearings	
Static contact pressure on anti-friction material	Oil film pressure
Squeezing of anti-friction material (for information)	Oil film thickness

2 Models

2.1 Structural Finite Element model

2.1.1 General

Structural FE model of the ship under study is to be used for preliminary calculations necessary to perform elastic alignment analysis:

- hull deformations (see [3.1])
- hull flexibility matrix (see [3.2]).

2.1.2 Drawings

The FE model is to be performed using at least the following drawings:

- structural parts
- steelwork of engine room and double bottom (if any).

2.1.3 Model for hull deformations

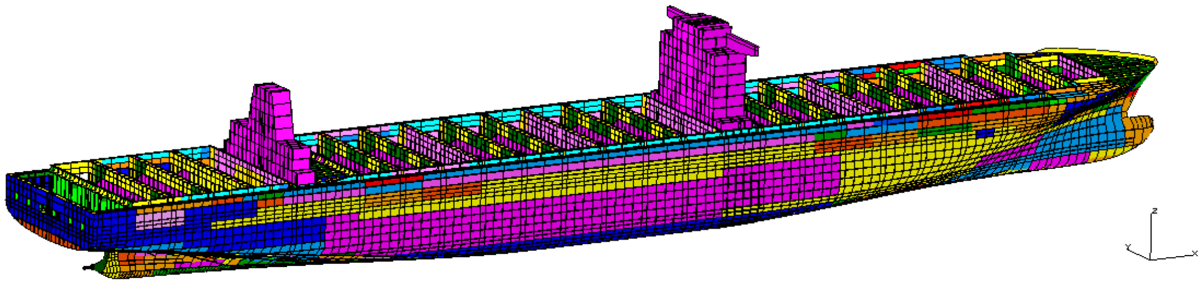
In order to compute hull deflections according to the relevant ship loading conditions, a complete ship FE model is to be used.

In addition to loading influence, hull deformations due to sea swell may be also considered.

A view of ship FE model for each relevant loading condition is to be submitted, as shown in Fig 1.

Requirements for calculations of hull deformations are given in [3.1].

Figure 1 : Complete ship FE model



2.1.4 Model for hull flexibility matrix

The terms of the hull flexibility matrix are to be calculated at each shaft line support (see [3.2]). For that purpose, the system is to be limited to the aft part of the ship.

The FE model to be used may be extracted from the model of the whole ship. It should extend from the aft end up to the forward watertight bulkhead of the engine room.

FE model is to be precisely refined and developed according to the following guidelines:

a) Nodes should be restrained in displacement and rotation in way of the forward transverse section.

b) Longitudinal secondary stiffeners are to be modeled in order to ensure a sufficiently refined mesh of the ship structure. As a consequence, standard size of the finite elements used is to be based on the secondary stiffener spacing.

c) The structural model should be built on the basis of the following criteria:

- Webs of primary members are to be modeled with at least three elements on their height
- Plating between two primary supporting members is to be modeled with at least two element stripes
- The ratio between the longer side and the shorter side of the elements is to be less than 3 in the areas expected to be highly stressed
- Holes for the passage of ordinary stiffeners may be disregarded.

d) Cast part of bossing as well as forward stern tube bush steelwork should be modeled with solid elements (8 nodes bricks), as shown on Fig 3.

e) Longitudinal position of the equivalent supporting points (see [2.3]) is to be exactly the same on the line shafting and on the structure.

Views of aft hull structure FE model showing modeling details are to be submitted. See examples shown on Fig 2 and Fig 3.

Figure 2 : Aft hull structure FE model

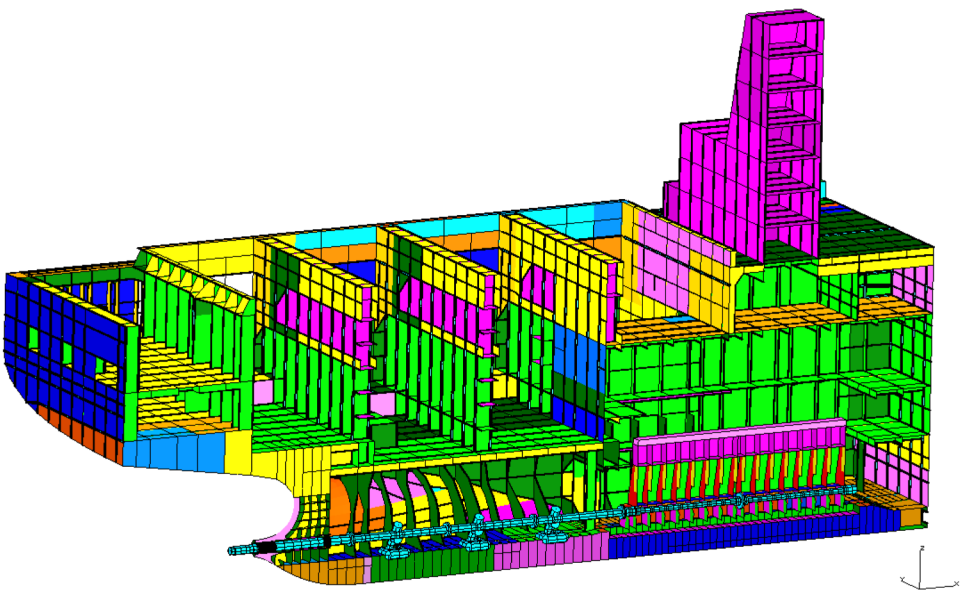
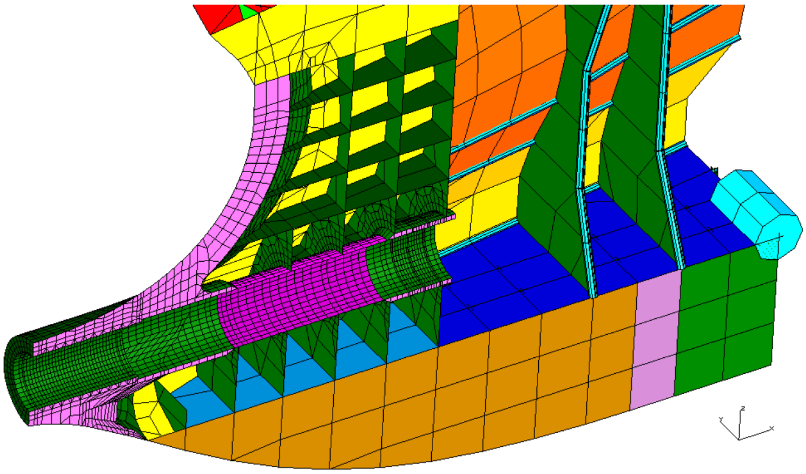


Figure 3 : Stern bossing details



2.2 Line shafting

2.2.1 General

The shaft line model is to be based on the exact geometry and characteristics of shafts and masses which are part of the mechanical system defined in Tab 1.

2.2.2 Shafts

Shafts are to be modeled using circular or conical beam elements.

For installations directly driven by low-speed diesel/gas engines, the crankshaft equivalent beam model is to be used when the exact stiffness matrix of crankshaft is not available.

Local masses (i.e. propeller, wheels, gears, couplings, etc) are to be considered in addition to the shaft weight.

Where applicable, buoyancy effects on shaft sections operating in water or oil are to be included in the model.

External loads on shafts are to be considered. The following efforts are listed for reference:

- geared installations: tooth forces and moments in each direction
- direct coupled low-speed engines: chain forces, cylinder weights.

2.2.3 Propeller

Propeller should be modeled by application of additional mass on shaft model, in way of propeller centre of gravity.

Buoyancy effect in water is to be considered.

Depending on the considered ship loading condition, propeller mass is to be adapted, taking into account the exact immersion ratio.

Mean values of hydrodynamic propeller efforts are to be applied in each relevant operating condition, in vertical and transverse directions.

2.3 Bearings

2.3.1 General

For hull flexibility matrix calculation, bearings are to be modeled with supporting points connected to the structure (see [3.2]).

For elastic shaft alignment calculation, line shafting model should include the following bearing particulars:

- effective contact length
- oil groove angular location
- clearances
- mechanical properties of sleeve and anti-friction materials
- machining of slope and partial slope, if any.

Axial location of supporting points should match exactly for FE model and shaft line model.

2.3.2 Aft bush bearing

The aftermost bearing is to be modeled with at least five supporting points in order to have detailed results at each section of the bearing.

The aft bearing is to be considered as a long bearing for the chosen elasto-hydrodynamic calculation method.

2.3.3 Other bearings

Each other bearing (forward bush, intermediate bearings, gearbox or main engine bearings) is to be modeled with one or more supporting points.

3 Preliminary calculations

3.1 Hull deformations

3.1.1 General

Calculation of hull steel work deformation is required for determination of the relative displacements of the line shafting supports as a function of loading and operating conditions.

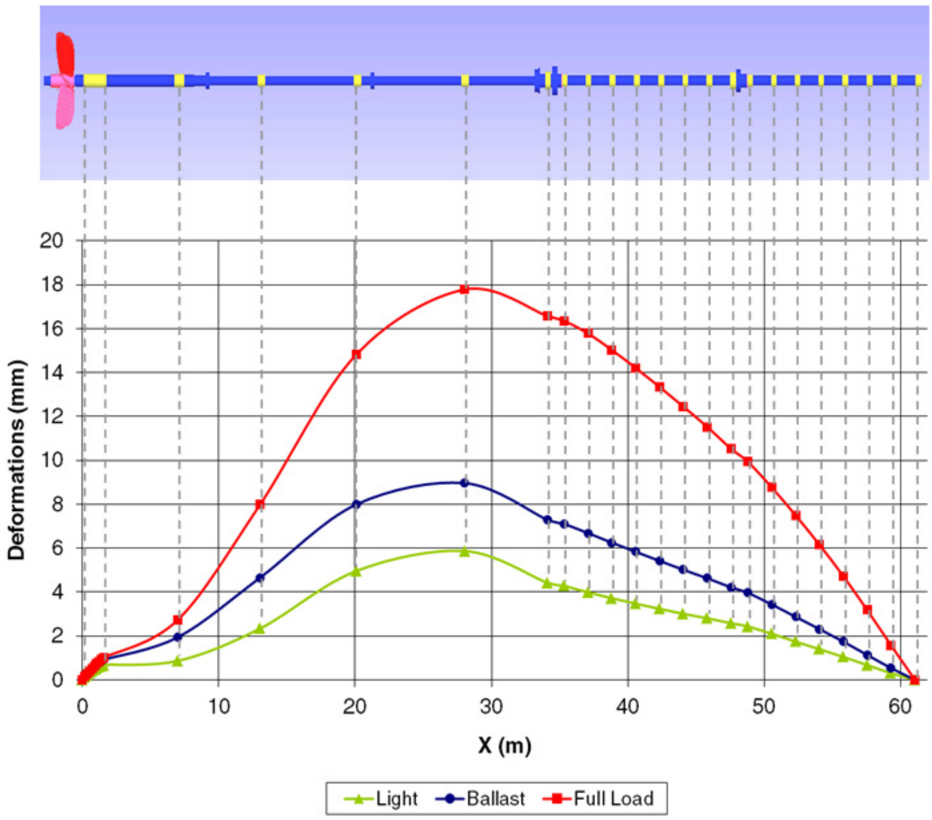
3.1.2 Principle

Calculations are to be performed using the FE model of the whole ship, as mentioned in [2.1.3].

Since practical alignment operations are generally performed in light ship condition, relative deformations between light ship and any relevant operating conditions (ballast and full load in particular) are to be calculated in order to add the corresponding relative displacements of bearings to their initial offset values for alignment analysis.

The hull relative displacements previously defined are to be obtained at each supporting point. Moreover, values are to be computed according to the reference line defined by the aftermost support (aft end of stern bush) and the forward most support of the shaft line, as shown on Fig 4.

Figure 4 : Hull deformations



3.1.3 Deformations due to sea swell

In addition to the deformations of hull structure as a function of loading conditions (see [3.1.2]), the deformations of hull structure due to sea swell could be included in elastic alignment calculations, based on the following requirements:

a) Influence of sea waves caused by local wind should not be considered.

b) Sea swell wave characteristics are to be defined for maximizing the double-bottom relative deformation in way of shaft line supports. Wave parameters (direction, height H and wave length λ) are to be chosen as follows:

- Couple (H, λ) is to be physically realistic (i.e: the wave should not break with the chosen values).
- Only head sea condition should be investigated.

c) Loading due to wave defined by (H,λ) should be applied, considering two sinusoidal equivalent profiles: maximum pressure (wave crest) and low pressure (wave trough) located in way of the aft peak.

d) Relative displacements of shaft supporting points between the two load cases defined in item c) are to be considered in elastic alignment calculations as additional bearing offsets included in vector U_b^0 (see definition in App 1). The calculated displacements due to sea swell are to be given at the same points where hull displacements due to ship loading have been calculated (see [3.1.2]).

3.2 Hull flexibility matrix

3.2.1 General

The hull flexibility matrix is to be calculated with the model of aft hull structure described in [2.1.4].

3.2.2 Definition

In way of supports, the displacements in transverse and vertical directions induced by a transverse or vertical unit force applied on one support determine a line of the flexibility matrix.

The flexibility matrix may be written as follows:

$$\begin{bmatrix} d_{1,T1} & d_{1,V1} & \cdots & d_{1,Tj} & d_{1,Vj} & \cdots & d_{1,Tn} & d_{1,Vn} \\ \vdots & \vdots & \ddots & \vdots & \vdots & \ddots & \vdots & \vdots \\ d_{i,T1} & d_{i,V1} & \cdots & d_{i,Tj} & d_{i,Vj} & \cdots & d_{i,Vn} & d_{i,Vn} \\ \vdots & \vdots & \ddots & \vdots & \vdots & \ddots & \vdots & \vdots \\ d_{2n,T1} & d_{2n,V1} & \cdots & d_{2n,Tj} & d_{2n,Vj} & \cdots & d_{2n,Tn} & d_{2n,Vn} \end{bmatrix}$$

where:

d : Displacement in transverse or vertical direction

n : Total number of supporting points

i : Row index for the load case reference (i ∈ [1, 2n]). For instance:

- i = 1: first load case defined by a unit force applied on the first support in transverse direction
- i = 2: second load case defined by a unit force applied on the first support in vertical direction
- i = 3: third load case defined by a unit force applied on the second support in transverse direction.

j : Index for the considered support (j ∈ [1, n]). For each support j, two columns are built for transverse and vertical displacements (column indexes Tj and Vj).

Each term of the hull flexibility matrix is noted as follows:

$d_{i,Tj}$: Transverse displacement of support j due to load case i

$d_{i,Vj}$: Vertical displacement of support j due to load case i.

The hull flexibility matrix size is to be 2n x 2n.

3.2.3 Data to be submitted

The following data are to be submitted, in electronic format, as attachment to the calculation report:

- hull flexibility matrix
- coordinates of the supporting points considered for calculation of the hull flexibility matrix.

3.3 Line shafting stiffness matrix

3.3.1 General

Stiffness matrix of line shafting is to be computed, and reduced if necessary, in way of the supporting points, for vertical and transverse directions, with a suitable calculation method which should be specified in the submitted report.

The supporting points considered in this calculation shall match the supporting points considered in calculation of the hull flexibility matrix (see [3.2]).

The line shafting stiffness matrix size is to be 2n x 2n.

Influence coefficients are defined as vertical and transverse variations of the reactions on the supporting points when a unit displacement is successively applied to each point in vertical and transverse directions. Calculation of these coefficients is based on the line shafting stiffness matrix.

3.3.2 Data to be submitted

Table of the influence coefficients is to be submitted for information.

4 Static calculations

4.1 Input data and assumptions

4.1.1 General

The aim of static alignment calculations is to check that the parameters are adjusted in order to reduce the risk of failure or excessive wear down in stopped, start-up and slow down conditions.

Bearing offsets and stern bush machining data are to be optimized at design stage for static conditions.

Since shaft alignment operations are performed in static condition, the corresponding calculations are to be made as accurately as possible, in compliance with the requirements listed in [4.2]. Static reactions measured by load tests are to be used for the correlation between calculations and measurements.

The relevant static cases are to be investigated, considering the possible combination of the following influence parameters:

- ship's loading condition
- ambient temperature in engine room (cold/warm)
- shaft line bolting (connected/opened).

4.1.2 Input data

The input data listed in Tab 3 are to be considered for calculation of the static alignment.

Table 3 : Input data for a static alignment calculation

No.	Input data
Calculation data	
1	Initial squeezing of antifriction material
Bearings	
2	Offsets of supports taking into account thermal expansion, pre-sag and structural deformation **(1)**
3	Effective length **(2)**
4	Diameters of shell sleeves and antifriction material layer
5	Young's modulus and Poisson's ratio of shell and antifriction material layer
Shafts	
6	Outer and inner diameters of shafts in way of supporting points
7	Young's modulus and Poisson's ratio of shafts in way of supporting points
8	Stiffness matrix of shaft line in way of supporting points **(3)**
General	
9	External forces and moments **(4)**
10	Flexibility matrix of steel work **(5)**

(1) Hull structural deformations are to be calculated according to [3.1].
(2) Effective length is the active part of the bearing, e.g. chamfers are not considered.
(3) Stiffness matrix of shafts is to be calculated according to [3.2].
(4) External loads listed in [2.2] are to be included.
(5) Flexibility matrix is to be calculated according to [3.2].

4.2 Alignment analysis

4.2.1 Methodology

The analysis should be performed with the models described in Article [2] and the input data listed in [4.1].

An acceptable method is described in App 1.

4.2.2 Output data

For the relevant calculation cases, the results to be submitted for the approval of static calculations are the following:

- maximum contact pressure on each bearing
- distribution of reactions
- squeezing of anti-friction layer
- shaft deflection and slope

- shaft bending moment
- shaft shear force
- shaft bending stress.

4.2.3 Acceptance criteria

The submitted results are to comply with the acceptance criteria listed in Tab 4.

Table 4 : Acceptance criteria for static alignment calculations

No.	Result	Limit	
1	Maximum local pressure on stern bushes, P_B	$P_B < 110$ bars	
2	Bearing loads **(1)**	Reaction values and distribution are to be within the applicable manufacturers' requirements	
3	Specific pressure on stern bushes, P_S **(2)**	Antifriction material type	
		White metal	Others
		$P_S < 0{,}8$ MPa	$P_S < 0{,}6$ MPa
4	Mean relative shaft slope in aftmost bearing, θ_S **(3)**	The relative slope between shaft and stern bush inner axes is to be less than the ratio of radial clearance divided by the bearing effective length	
5	Shaft bending stress and moment	Calculated bending stress and moment are to be in compliance with the manufacturers' requirements	

(1) For static conditions, recommended load distribution on aft bush bearing is: 2/3 of reaction on aft part, 1/3 on forward part.
(2) Specific pressure P_S (in MPa) is defined as follows:

$$P_S = \frac{R_V}{L_{eff} \cdot D_O}$$

where:
R_V : Total vertical reaction on the considered bearing, in N
L_{eff} : Effective length of the considered bearing, in mm
D_O : Outer diameter of shaft in way of the considered bearing, in mm.
(3) Mean relative shaft slope is defined as the difference between the slope values calculated at ending nodes of the considered bearing.

5 Running calculations

5.1 Input data and assumptions

5.1.1 General

The aim of running alignment calculations is to check that the parameters are adjusted in order to reduce the risk of oil film break-up or excessive pressure on the antifriction material when the ship is sailing.

Bearing offsets, stern bush machining data and oil groove location are to be optimized at design stage for running conditions.

Calculations in straight course are to be performed as defined in [5.2] for each relevant operating condition. The following influence parameters are to be considered to define the calculation cases:

- ship's loading condition
- ambient engine room temperature (cold/warm)
- shaft speed (low/mid/maximum).

As specified in [1.2.2], turning phases are generally critical for the oil film and bearing behaviour. If realistic propeller effort values are available for those conditions, it is recommended to perform the corresponding calculations.

5.1.2 Input data

The input data listed in Tab 5 are to be considered for the alignment calculation in running conditions.

Table 5 : Input data for an alignment calculation in running conditions

No.	Input data
Calculation data	
1	Initial position of shaft in its bearings
Bearings	
2	Offsets of supports taking into account thermal expansion, pre-sag and structural deformation **(1)**
3	Effective length **(2)**
4	Diameters of shell sleeves and antifriction material layer
5	Young's modulus and Poisson's ratio of shell and antifriction material layer
Shafts	
6	Outer and inner diameters of shafts in way of supporting points
7	Young's modulus and Poisson's ratio of shafts in way of supporting points
8	Stiffness matrix of shaft line in way of supporting points **(3)**
General	
9	Oil viscosity
10	Rotational speed of shafts
11	External forces and moments **(4)**
12	Flexibility matrix of steel work **(5)**

(1) Hull structural deformations are to be calculated according to [3.1].
(2) Effective length is the active part of the bearing, e.g. chamfers are not considered.
(3) Stiffness matrix of shafts is to be calculated according to [3.2].
(4) External loads listed in [2.2] are to be included.
(5) Flexibility matrix is to be calculated according to [3.2].

5.2 Alignment analysis

5.2.1 Methodology

The analysis should be performed with the models described in Article [2] and the input data listed in [5.1].

A proposed method is described in App 1.

5.2.2 Output data

For each relevant calculation case, the results to be submitted for approval of the elastic calculations are the following:

- maximum local oil film pressure
- relative position of shaft centres with respect to oil grooves
- minimum oil film thickness
- distribution of reactions
- squeezing of anti-friction layer
- shaft deflection and slope
- shaft bending moment
- shaft shear force
- shaft bending stress.

5.2.3 Acceptance criteria

The submitted results are to comply with the acceptance criteria listed in Tab 6.

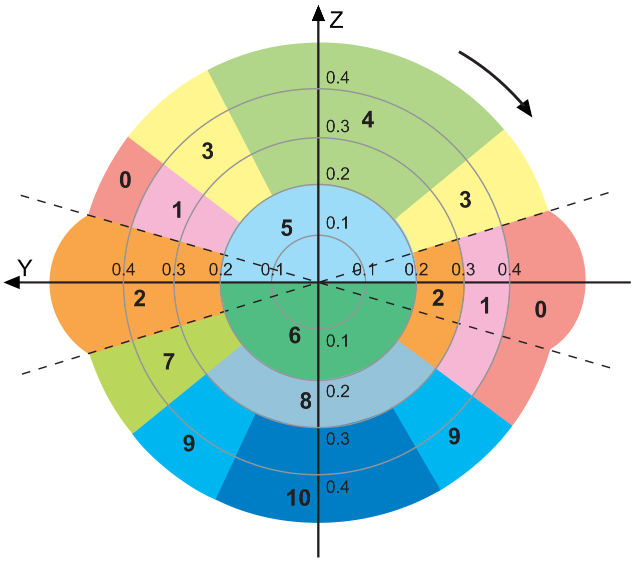
Table 6 : Acceptance criteria for alignment calculations in running conditions

<table>
<tr><th>No.</th><th>Result</th><th colspan="2">Limit</th></tr>
<tr><td>1</td><td>Maximum local oil film pressure, P_O</td><td colspan="2">P_O < 80 bars</td></tr>
<tr><td>2</td><td>Bearing loads</td><td colspan="2">Reaction values and distribution are to be within the applicable manufacturers' requirements</td></tr>
<tr><td rowspan="3">3</td><td rowspan="3">Specific pressure on stern bushes, P_S (1)</td><td colspan="2">Antifriction material type</td></tr>
<tr><td>White metal</td><td>Other</td></tr>
<tr><td>P_S < 0,8 MPa</td><td>P_S < 0,6 MPa</td></tr>
<tr><td>4</td><td>Shaft position in aft bush with respect to oil grooves</td><td colspan="2">The shaft centre is to be located in safe areas (see Fig 5)
Zone 0 is forbidden, zone 10 is optimum</td></tr>
</table>

(1) See definitions in Tab 4.

No.	Result	Limit
5	Minimum oil film thickness, h_{min}	$h_{min} > 30$ µm
6	Mean relative shaft slope, θ_S **(1)**	The relative slope between shaft and stern bush inner axes is to be less than the ratio of radial clearance divided by the bearing effective length
7	Shaft bending stress and moment	Calculated bending stress and moment are to be in compliance with the manufacturers' requirements
(1) See definitions in Tab 4.		

Figure 5 : Scale of shaft location severity zone



y and z axes refer to bearing radial clearance, in mm.

6 Alignment procedure

6.1 General

6.1.1 Data to be submitted

The detailed shaft alignment procedure should be submitted for approval. It should include the corresponding calculations and description of each step that will be performed onboard: sightings, shaft installation fitting procedure and applied measurement tolerances.

The final report of measurements performed onboard showing compliance with the approved alignment procedure is to be submitted to the Society.

6.1.2 Propeller immersion

For the calculation cases corresponding to the alignment operations, the propeller immersion is to be adjusted in relation to the foreseen ship loading condition during these operations.

6.1.3 Recommendations

Sub-articles [6.2] and [6.3] are recommendations about shaft alignment procedure, including measurement steps and related tolerance.

These recommendations are only guidance to designers and shipyards.

6.2 Ship in dry-dock

6.2.1 Ambient conditions

Relative alignment of stern bushes should be carried out when weldings of the neighbouring steel work of the aftbody of the ship are completed.

As far as possible, laser or optical centering checks should be performed at night in order to avoid undesirable light or temperature disturbance.

6.2.2 Sightings

Once the sloping and fitting of stern bushes have been realised, exact position of their centres should be precisely checked by optical or laser sightings. Measured vertical and horizontal offsets of stern bushes should be in accordance with the theoretical alignment. Maximum deviation between measurements and optimized offsets should be typically 0,05 mm or less.

6.2.3 Gearbox/prime mover prepositioning

In addition to stern bushes centering, preliminary positioning of gearbox or prime mover should be performed (depending on the shaft alignment system, see [1.1.1]).

Vertical and transverse offsets of flywheel or gearwheel centre should be measured and adjusted if necessary, according to the values determined by elastic alignment study.

6.3 Ship afloat

6.3.1 Floating conditions

Final alignment operations should be performed with the ship afloat in order to take into account the hull deformations due to the hydrostatic pressure. The following steps should be performed in floating conditions:

- intermediate bearing positioning
- prime mover positioning
- shaft bolting
- load tests.

6.3.2 Positioning of bearings

Position of the intermediate bearings and prime mover/gearbox should be adjusted by an accurate method.

If gap/sag method is used to adjust the shaft bearings, the requirements hereafter should be followed:

- gap/sag values should be calculated and submitted in the calculation report
- tolerance for gap/sag attained onboard should be typically within a range of 0,05 mm or less compared to the calculated values.

6.3.3 Bearings load tests

After shaft bolting, final load checkings should be performed on the accessible bearings with jack-up test method. If applicable, it should be applied on:

- forward bush
- intermediate bearings
- aft gearbox bearings (journal type only)
- three aftmost main engine bearings.

Calculation of correction factors and jacking procedure should be submitted in the report.

6.4 Sea trials

6.4.1 Test procedure

For running-in of stern bearings, the relevant ship operating conditions should be tested, including the following courses:

- straight
- zigzag
- turning.

Test sequences should be sufficiently spaced in time in order to permit necessary dissipation of heat generated by the succession of severe loadings, thus avoiding overheating of stern bearings.

6.4.2 Monitoring

The following points should be monitored during sea trials:

- lubricating oil flows
- bearing temperatures
- vibration signs (noises, unusual vibrations in engine room, etc).

Appendix 1 Shaft Alignment Calculation Methods

1 General

1.1 Introduction

1.1.1 The objective of this Appendix is to give general guidelines on an acceptable method for elastic alignment calculations in static and running conditions.

2 Methodology

2.1 Hertz contact theory

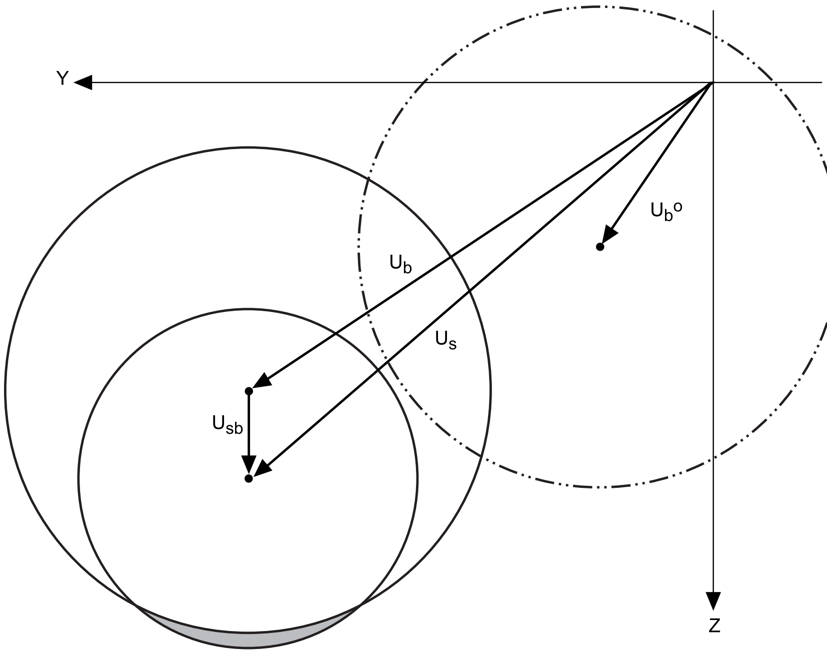
2.1.1 When the shaft line is laying on bearings without rotation, the Hertz contact theory is applicable to describe the characteristics of the contact: stiffness, reaction, length of contact, maximum pressure, squeezing.

This calculation is a part of the global resolution of equilibrium (see [2.3]).

2.1.2 The Hertz contact theory is considering a cylinder in a finite length cylindrical socket with a load applied on the cylinder. The Hertz law leads to the maximum static pressure and reaction in the contact basing on the mechanical properties as well as the geometry of the cylinder and the socket.

2.1.3 For the application of the Hertz theory on shaft alignment calculations, the cylinder is the shaft and the socket is the bearing at the supporting point. The displacement of the shaft inside the bearing is known (see Fig 1) and the contact pressure and the load have to be calculated considering that it has the same direction as the displacement U_{sb}.

Figure 1 : Bearing and shaft equilibrium in static condition



2.2 Oil film calculation

2.2.1 When the shaft rotates at a sufficient speed, a flow of oil is induced by its viscosity and the shaft speed creates a lift of the shaft. There is no more contact between the shaft and the bearing: the oil film is built-up. The calculation of this oil film is to be carried-out on the basis of two equations:

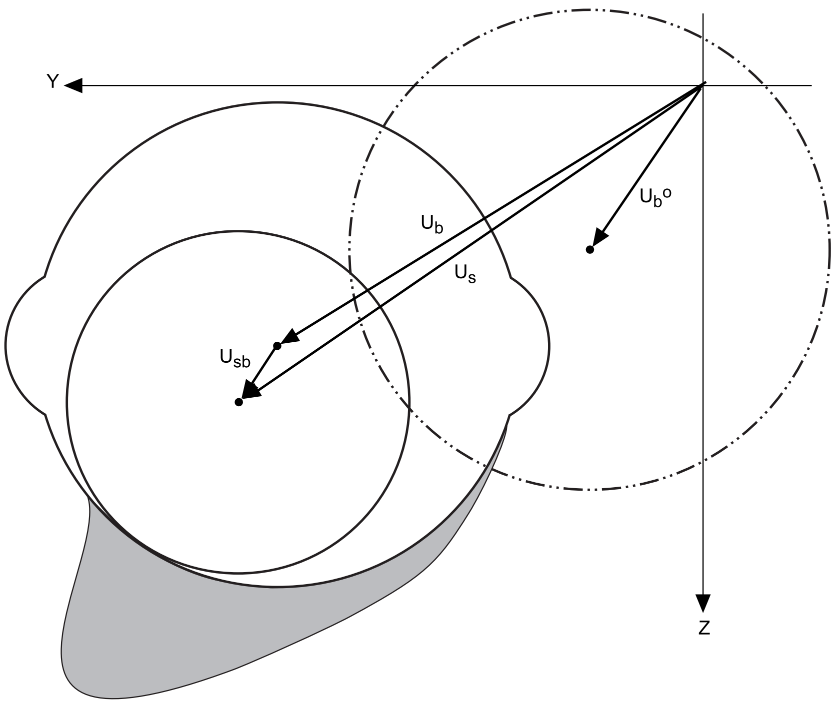
- a hydrodynamic differential equation which determines the behaviour of a thin and viscous fluid (Reynolds equation)
- a geometric equation which determines the height of the oil film according to the relative position between the deformed journal of the shaft and its machined profile.

These equations lead to the characteristics of the oil film: stiffness, reactions, oil pressure, damping.

This calculation is a part of the global resolution of equilibrium (see [2.3]).

2.2.2 For the application of the oil film theory on the shaft alignment calculation, the initial shaft displacement inside the bearing is known (see Fig 2) and the load has to be calculated by integration of pressure along the bearing circumference.

Figure 2 : Bearing and shaft equilibrium in running condition



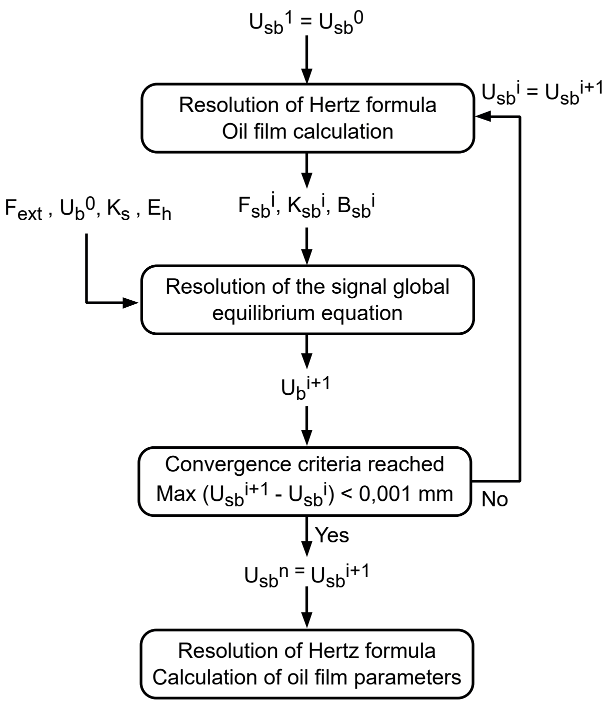
2.3 Global equations

2.3.1 The global equations are based on the quasi-static equilibrium of the shaft with the structure, the bearings and the external forces.

The aim is to reach, by an iterative process, the equilibrium position of the shaft line inside the bearings and to obtain, with a final Hertz or oil film calculation, the characteristics of the shaft behaviour in bearings (see Fig 3).

2.3.2 The equations are to take into account the mechanical parameters of all bearings. The problem is then reduced in way of the supporting points.

Figure 3 : Flow chart of iterative process



2.3.3 General equilibrium equation may be written as follows:

$$[K] \cdot U + B_{sb} + F_{ext} = 0$$

where:

[K] : Global stiffness matrix, being a combination of the following partial matrices:

- $[K_s]$: Shaft line stiffness matrix, see Sec 2, [3.3]
- $[K_{sb}]$: Stiffness matrix of contact in static contact or running oil lubricated contact, see [2.1] and [2.2] respectively
- $[E_h]$: Hull flexibility matrix as defined in Sec 2, [3.2]

U : Vector of displacements, including the following components, see Fig 1 and Fig 2:

- U_b^0 : Vector of initial bearing centre position with reference to shaft centre (without gravity), depending on the loading condition, temperature and alignment procedure
- U_s : Vector of shaft centre displacements in way of the supporting points relatively to the reference line
- U_{sb} : Vector of shaft centre relative displacements in way of the supports

B_{sb} : Vector considering the non-linearity of the contact conditions:

$$B_{sb} = F_{sb} - ([K_{sb}] \cdot U_{sb})$$

where F_{sb} is the contact force vector

F_{ext} : External load vector, including gravity and other external efforts (propeller, engine, gearing), reduced in way of the supports. See Sec 2, [2.2].

2.3.4 The resolution of this equation is based on an iterative process using an initial displacement U_{sb}^0 close to the equilibrium solution in order to calculate the main contact characteristics: K_{sb}^0, F_{sb}^0 and B_{sb}^0.

Then a calculation of the global equation, see Fig 3, determines vector U_{sb}^1.

2.3.5 The convergence criteria for this iterative process is calculated using the maximum absolute difference between the terms U_{sb}^i and U_{sb}^{i+1}. This value is to be less than 0,001 mm.

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