Effects of Hull Deformation on the Static Shaft Alignment Characteristics of VLCCs: A Case Study

George Korbetis, BETA CAE Systems S.A., Thessaloniki/Greece, ansa@beta-cae.gr
Orestis Vlachos, Anastasios G. Charitopoulos, Christos I. Papadopoulos, NTUA, Athens/Greece

Abstract

A typical VLCC driven by a two-stroke Diesel engine is studied. A detailed finite element model of the hull structure is generated. The propulsion shaft of the ship is modelled as a statically indeterminate multi-supported beam. First, a reference shaft alignment plan is assumed, and the static equilibrium of the shaft is calculated using matrix analysis. Next, different loading conditions (laden/ballast) of the ship are assessed. For each loading condition (a) hydrostatic equilibrium of the ship is computed, (b) the corresponding hull deformations are calculated using Finite Element Analysis, (c) the relative vertical displacements at the bearing locations are determined and (d) the static shaft equilibrium is re-evaluated. The computed bearing loads are compared to those of the reference case.

1. Introduction

The propulsion system of conventional cargo ships typically consists of a two-stroke Diesel engine, and a shafting system, which transmits the engine power to the propeller, Fig.1. Radial shaft loads (propeller/shaft/engine weights) are supported by journal bearings (stern tube bearings, line bearings, crankshaft bearings). Proper design, installation and alignment of the shafting system of a ship is crucial for stable, efficient and reliable operation, *ABS (2004), NKK (2006)*. Primarily, shaft alignment is concerned with the determination of proper longitudinal and vertical bearing positions, aiming at equi-distribution of bearing loads. The successful application of a static shaft alignment plan is essential for trouble-free dynamic operation of the propulsion system, aiding in decreasing bearing wear, increasing bearing expected lifetime and decreasing maintenance and replacement costs.

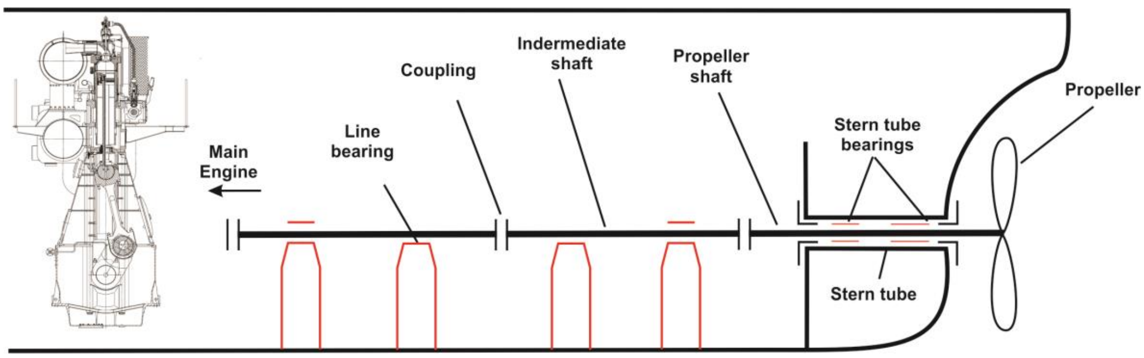


Fig.1: Typical arrangement of a ship shafting system

In operation, shaft alignment may be considerably influenced by hull deflections, due to different loading and environmental conditions. The effect of hull deflections on shaft alignment is more pronounced in very long ships, with relatively flexible hulls and stiff shafts. In such cases, the robustness of shaft alignment at different loading conditions of the ship, taking hull deflections into account, should be carefully assessed. In this respect, the use of detailed Finite Element Analyses for the calculation of hull deflections is imperative.

Recently, the subject of shaft alignment has gained increasing attention. *Devanney and Kennedy (2003)* underlined the drastic deterioration of tanker newbuilding standards in the last decade, and the corresponding effect on the reliability of the shafting system. Specifically, emphasis was put on the severity of stern tube bearing failures in modern VLCCs and ULCCs, which may lead to loss of propulsion and vessel immobilization. The authors claimed that the main reason of this failure is the design of propulsion shafts with decreased diameters, followed by improper shafting alignment. They suggested that (a) hull deflections should be thoroughly taken into account for a range of loading conditions of the ship, (b) the engine room structure should be reinforced, to minimize additional offset of the bearings, and (c) time varying loads on the stern tube bearing and heat dissipation in the lubricant domain should be taken into account.

Šverko (2003) highlighted several design concerns in propulsion shafting, especially for VLCC and large bulk carrier vessels. In such vessels, shaft alignment is very sensitive to hull deflections; this behavior was attributed to the increased hull flexibility of such ships (due to scantling optimization and increased ship lengths) and to the increased stiffness of the propulsion shaft (due to the demand for higher propulsion power and, consequently, larger shaft diameters). If the hull deformations can be predicted accurately, an optimal set of bearing offsets for the vessel on even keel may exhibit a reasonably good performance at other loading conditions of the vessel; however, since hull deflections cannot be easily calculated accurately, a practical solution could be to complete the alignment at dry dock conditions, and make provisions to correct (if needed) bearing vertical offsets when the reactions are verified afloat. *Šverko (2006)* addressed the problem of predicting hull deflections through analysis of series of collected real life data. Hull deflections were estimated by measurement of shaft deflections using bending gauges. The goal of this study was to find appropriate dry dock bearing offsets that will result in acceptable alignment performance over a wide range of vessel loading conditions. *Murawski (2005)* also utilized a FEM model of a large containership, and introduced a new parameter to be considered: the stiffness characteristics of the bearing foundations. He concluded that, in a holistic approach to the shaft alignment problem, bearing stiffness and oil film characteristics of each bearing should be taken into account in the design stage. *Dahler et al. (2004)* reported the results of an joint industrial project between DNV, MAN B&W and DAEWOO concerned with the numerical and experimental study of shaft deflections and bearing loads in large ships propelled by two-stroke Diesel engines. They utilized a complete FEM model of the ship, which exhibited a fine mesh at the aft end of the ship hull (engine room). Focus was given on engine and crankshaft deflections and on the corresponding bearing loads. To this end, FEM analyses were performed taking into account the real crankshaft geometry, and the results were compared with simulations using simplified crankshaft models. Simulation results were also compared to experimental measurements. They concluded that FEM-hull analyses can capture the general trend of hull deflections reasonably well, but fail to account for local variations in the curvature of the shaft, leading to inaccurate predictions of bearing loads. Finally, they suggested that by applying the final shafting plan after vessel launch, possible errors due to wrong estimation of hull deflections could be avoided. *BV (2013)* released Rule Note NR 592, concerned with Elastic Shaft Alignment (ESA) of ships. The proposed methodology of shafting alignment calculations takes into account hull deformations, oil film characteristics and stiffness of the bearings' foundation. The rule is mainly applicable to ships characterized by a propeller shaft diameter greater than 750 mm, or between 600 mm and 750 mm, but with propeller weight greater than 30 tones or a prime mover with power output greater than 20 MW.

In the present work, a typical VLCC vessel, driven by a two-stroke Diesel engine, is studied. The vessel has a propeller shaft diameter of 815 mm; therefore it is within the scope of the ESA Rule of BV. Here, a detailed finite element model of the hull structure of the ship, complying with the meshing requirements set by Classification Societies, is generated with the use of the ANSA pre-processor. The propulsion shaft of the ship is modeled as a statically indeterminate multi-supported beam; the bearing stiffness and clearance are taken into account, and the static equilibrium of the shaft is calculated using matrix analysis. Considering the undeformed hull of the vessel, a reference shaft alignment plan is assumed, and the static equilibrium of the shaft is calculated. Next, different loading conditions (laden/ballast) of the ship are assessed. For each loading condition (a) hydrostatic equilibrium of the ship is computed, (b) the corresponding hull deformations are calculated, (c) the relative vertical displacements at the bearing locations are determined and (d) the static shaft equilibrium is re-evaluated. The computed bearing loads (reaction forces) are compared to those of the reference case.

2. Problem Definition

2.1. Finite element analysis of the ship hull

The main characteristics of the studied VLCC are presented in Table I. Finite Element Analysis is performed to calculate the hull deformations of the vessel, at different loading conditions. Of particular importance are the deformations at the bearing locations of the propulsion shafting system. The static analyses are conducted with the aid of the ANSA pre-processor the MSC/NASTRAN solver. Here, thermal loads from the engine or the environment are not taken into consideration. First, a FEM model of the ship structure is generated. The whole structure of the ship is represented by first-order shell elements; at the stern tube region, solid tetrahedral elements are used. A coarse mesh is generated for the whole structure (element length of 0.95 m), except from the engine room floor, where finer mesh (element length of 0.2 m) ensures better accuracy results, Figs.2 and 3.

Table I: Main characteristics of the VLCC vessel of the present study

Type	Crude Oil Tanker
Deadweight	320000 t
Length betw. Perp. L_{PP}	320.00 m
Breadth B	60.00 m
Depth D	30.50 m
Scantling draft T	22.50 m
Service speed V_s	15.9 kn
Main engine	Wärtsilä 7RT-FLEX84T-D
Keel laid	April 2010

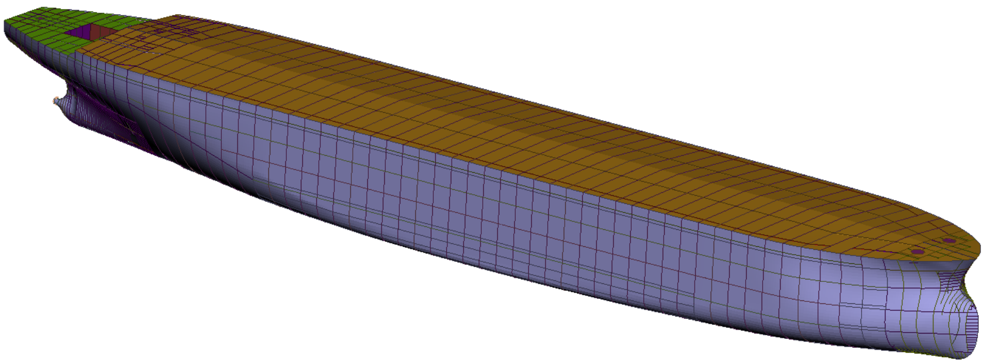


Fig.2: Global FEM model of the vessel of the present study

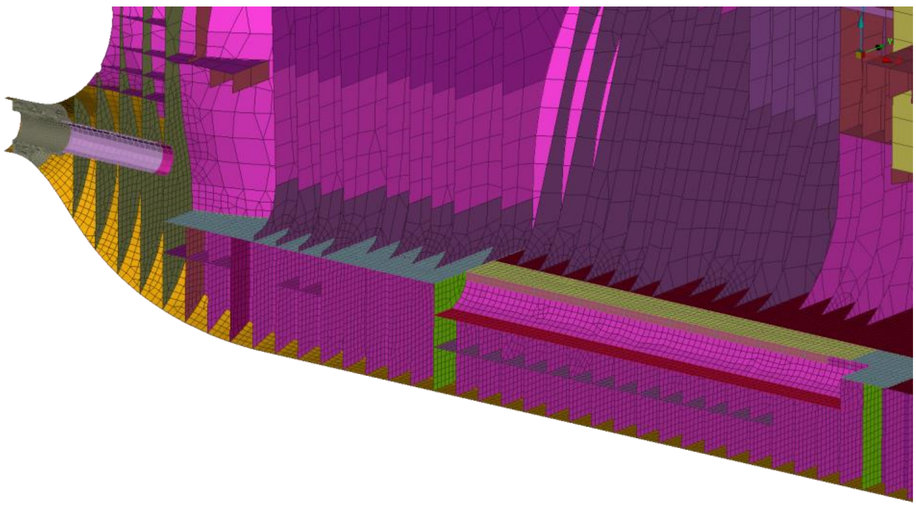


Fig.3: Detail of the generated FEM mesh at the engine room region of the vessel

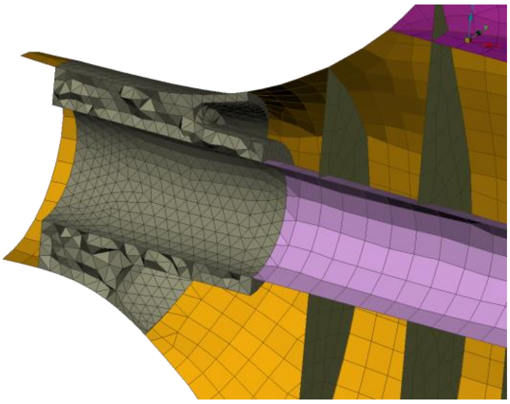


Fig.4: Detail of the generated FE mesh at the stern tube region of the vessel

The mesh generation is an automated process performed by the ANSA Batch Meshing Tool. Meshing parameters and quality criteria are defined in two meshing scenarios (fine mesh for the engine room floor and coarse mesh for the rest of the structure), Table II. Re-meshing algorithms act on areas with poor mesh quality until the predefined quality criteria are fulfilled. The final model comprises of about 402.000 shell elements, 143.000 beams and 17.000 solid tetrahedrals, Figs.2 to 4.

Table II: Meshing parameters and quality criteria.

Global Meshing Parameters (Scenario I)	
Element length	0.95 m
Filling openings with diameter	< 1m
Engine room floor Meshing Parameters (Scenario II)	
Element length	0.2
Filling openings with diameter	< 0.5m
Quality Criteria	
Skewness (Nastran)	30°
Aspect ratio (Nastran)	3
Angle (Quads)	45-135°
Angle (Trias)	30-120°
Minimum Element Length	0.01 m
Maximum Element Length	1.5 m

Stiffeners are represented by beam elements pasted on the shells. This method simplifies the model by avoiding the generation of very small shell elements. The properties of the beam elements are calculated in accordance with the cross section of each stiffener.

Machinery, auxiliary structures and small constructions that do not contribute to ship strength are not modeled in the present FEM model. Their mass is applied to the model as non-structural mass. This mass is appropriately distributed over the FEM model, so as to reach the prescribed lightship weight and the corresponding center of gravity. The mass of the present structural model is 34442 t, while the lightship weight is 43938.7 t and its center of gravity L.C.G. at 151.338 m. Thus, 9496.7 t of lumped masses are appropriately distributed in holds, stern and bow by the automatic process of the ANSA Mass Balance Tool, Fig.5. The engine mass is represented by a lumped mass of 990 t distributed to the engine foundation positions by RBE3 elements. Bearing positions where measurements will take place are represented by single nodes on the bearing axis, connected to the engine room floor with RB2 elements, Fig.6.

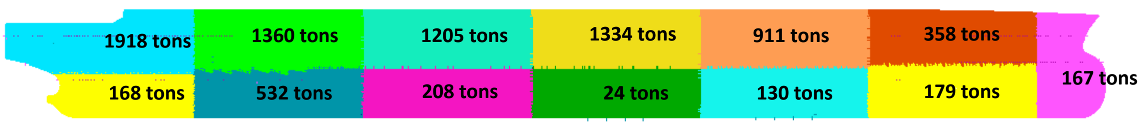


Fig.5: Distribution of non-structural mass in the present FEM model

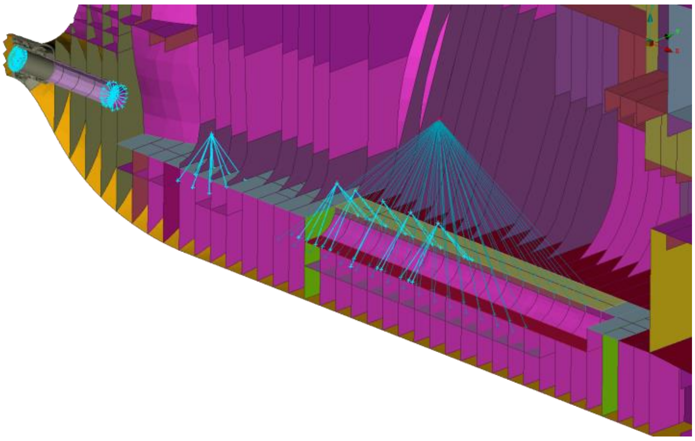


Fig.6: Engine and bearings representation in the present FEM model

Three representative loading conditions of the vessel, namely full-load departure, ballast arrival and departure with partial load, are considered in the present analysis, Fig.7. The contents of the tanks are represented by lumped mass connected to the each hold bottom with RBE3 elements. The ship is positioned on steel water considering the vessel's total displacement and center of gravity. Buoyancy is applied as pressure at the hull underneath the waterline using PLOAD4 entities, Fig.8. Finally, the vessel is trimmed in order to achieve static equilibrium between weight and buoyancy, which makes the model able to run without the need of displacement constraints (SPCs), which would lead to high local stresses. A NASTRAN keyword for inertia relief (INREL) is added for this solution.

Ballast arrival condition (L.C. 1)
Displacement: 145647 tones
Draft: 9.69 m
Trim: 2.12 m

Full-load departure condition (L.C. 2)
Displacement: 364074 tones
Draft: 22.52 m
Trim: 0.11 m

Departure with partial load (L.C. 3)
Displacement: 229276 tones
Draft: 14.78 m
Trim: 3.05 m

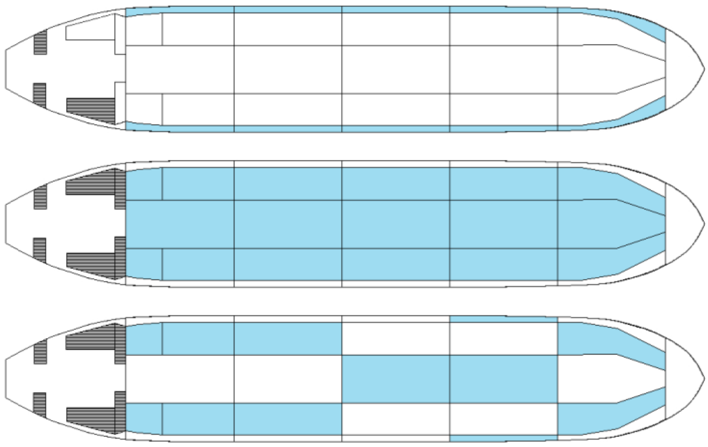


Fig.7: Representative loading conditions of the vessel

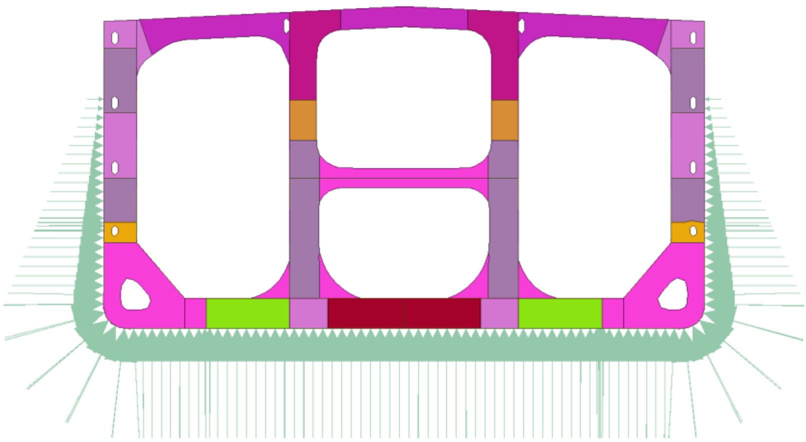


Fig.8: Application of hydrostatic pressure due to buoyancy in the FEM model

2.2 Calculation of Static Shaft Alignment

As noted in the introduction, a successful application of a static shaft alignment plan is important for trouble-free operation of the ship in the anticipated service conditions. The propulsion shaft of the ship is supported by the stern tube bearings, the line bearing(s) and the crankshaft bearings of the main engine. At first, a reference line can be defined as the one passing through the centers of the aft and fore stern tube bearings. Shaft alignment is concerned with the determination of the proper vertical offset of the center of the remaining bearings from the reference line, that result in even pressure distribution amongst all of the bearings of the system. This should stand both for static and dynamic conditions of the vessel.

In the static conditions of the vessel:

- The main engine (M/E) is not running - i.e. it is in cold condition.
- The eccentric thrust produced by the propeller is not considered and, likewise, any resulting bending moments are also not taken into account. The propeller contributes to the static loading of the shaft by its gravitational force.

2.2.1 Modeling of the shafting system

In the present paper, the propulsion shaft is represented by an assembly of two-node beam elements subjected to purely flexural deformations. External loads and deformations are applied at the beam nodes. All internal loads (e.g. the distributed weight of a beam) can be expressed in terms of equivalent nodal generalized loads, through the application of basic principles of mechanics, *Hughes and Paik (2010)*. The degrees of freedom allowed for each node are three rotations about each axis of a Cartesian 3D coordinate system and three displacements along each axis of the same system. We shall, from now on, refer to all parameters related to each simple beam element as "local" parameters and similarly, we shall denote all parameters related to the whole assembly of beams as "global" parameters.

For a single beam element, we may consider a vector **f**, containing the values of external and internal nodal loads of each of the six degrees of freedom (DOFs) of each node, and a vector **u** with the corresponding generalized displacements (i.e. displacements and rotations). A linear relationship between the nodal generalized displacements and nodal forces is assumed, namely **f = ku**. Matrix **k** represents the stiffness of each beam; the elements of matrix **k** are a function of the geometric and material properties of the beam (length, moment of inertia, Young's modulus).

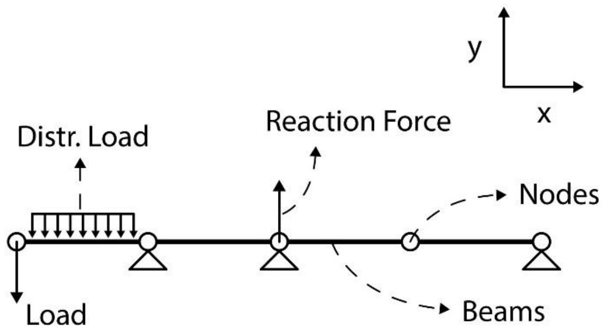
In Fig.9(a), a simple shaft consisting of four beam elements is presented. Using vectors **f**, **u**, and matrix **k** of each beam element, a global linear relationship between generalized forces and displacements of the system can be defined. To this end, vectors **F** and **U** are defined, which hold the values of all nodal DOFs, the total number of which evaluates to six times the number of the system nodes. The corresponding stiffness matrix of the system **K** (global stiffness matrix) is produced by appropriately utilizing the local stiffness matrix of each beam. The global problem can now be defined as **F**=**KU**, and can be solved for **F** or **U**, *Hughes and Paik (2010)*.

2.2.2 Modelling bearing supports

The propulsion shaft of a ship can be modeled as a multi supported beam. A simple type of support is that presented in Fig.9(a), denoted as a small triangle below the constrained node of the beam. Those idealized supports allow zero displacements of the shaft in the radial direction. In practice, the shaft is supported by hydrodynamically lubricated journal bearings. Geometrically, a journal bearing is a hollow cylinder, which encloses a solid shaft that rotates about its axis. The radius of the bearing is slightly larger than that of the shaft; the difference between the bearing and the shaft radius is called clearance. Therefore, the shaft may undergo a small displacement before contact with the bearing surface. Further, the journal bearing foundation is also deformable, therefore it will elastically deform when a load is applied. The elasticity of the bearing foundation can be taken into account by introducing an appropriate bearing stiffness coefficient. In practice, several of the bearings of propulsion shafts will be deliberately shifted in the y direction by an appropriate vertical offset, described in the shafting plan of the vessel. Therefore, a predefined displacement of the shaft at the bearing position should be taken into account. The above concepts are presented in Fig.9(b).

In summary, at bearing of the shafting system, an initial y-offset may be imposed (as displayed in Fig.9(b)); at equilibrium, the shaft will either "float" within the boundaries of the bearing clearance, without interacting with the bearing, or it will come into contact with the bearing upper or lower inner surface. In the latter case, the "spring-like" behavior of the bearing foundation will cause an additional deformation of the bearing support, to such a degree, that the resulting reaction force will balance the shaft weight that this bearing was meant to support.

(a)



(b)

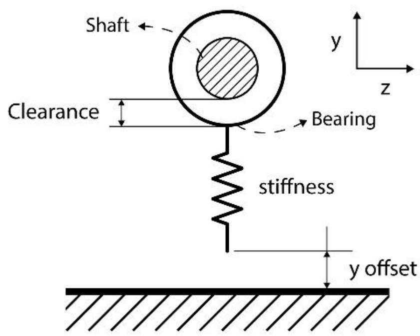


Fig.9: (a) Simplified model of a shafting system. (b) Sketch of a bearing; clearance, offset and foundation stiffness.

The above considerations become even more important as we examine the overall behavior of a given shafting system under different ship loading conditions. The deflections of the ship hull due to the action of load and buoyancy, directly affect the vertical position of the bearings, Fig.10. This additional disturbance can be taken into account through the application of an additional vertical offset to each bearing, relative to the reference line.

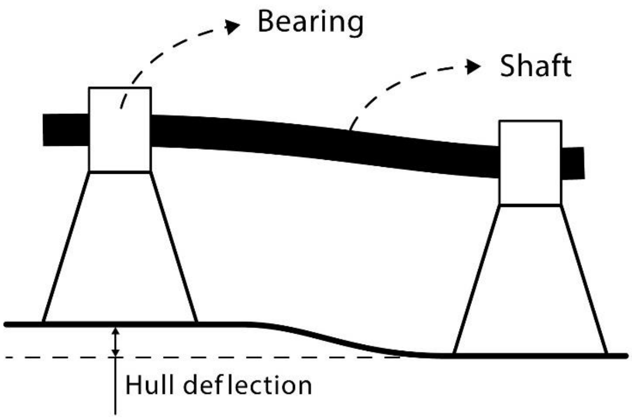


Fig.10: Vertical offset of a bearing due to hull deflections

2.3. Calculation of Static Shaft Alignment

Fig.11 shows a model of the shafting system studied. The propeller shaft, the intermediate shaft and part of the crankshaft of the main engine are considered. The propeller shaft is supported by two stern tube bearings, the intermediate shaft by a line shaft bearing and the engine crankshaft by the crankshaft bearings (five of them are included in the present calculations). Bearing details are presented hereinafter:

- Stern tube bearings: Aft bearing: L/D =2.22, foundation stiffness of 4×10^9 N/m. Fore bearing: L/D=0.53, foundation stiffness of 5×10^9 N/m. Both bearings have a radial clearance of 0.55 mm.
- Line shaft bearing: L/D =0.78, radial clearance of 0.425 mm, foundation stiffness of 10^9 N/m.
- M/E crankshaft bearings: Foundation stiffness of 6×10^9 N/m, radial clearance of 0.345 mm.
- Density of the shaft material: 7850 kg/m^3, Young's modulus of elasticity: 2.06×10^{11} N/m^2.
- The shaft is discretized with 39 beam elements. The geometry of each beam and load details are presented in Table III.

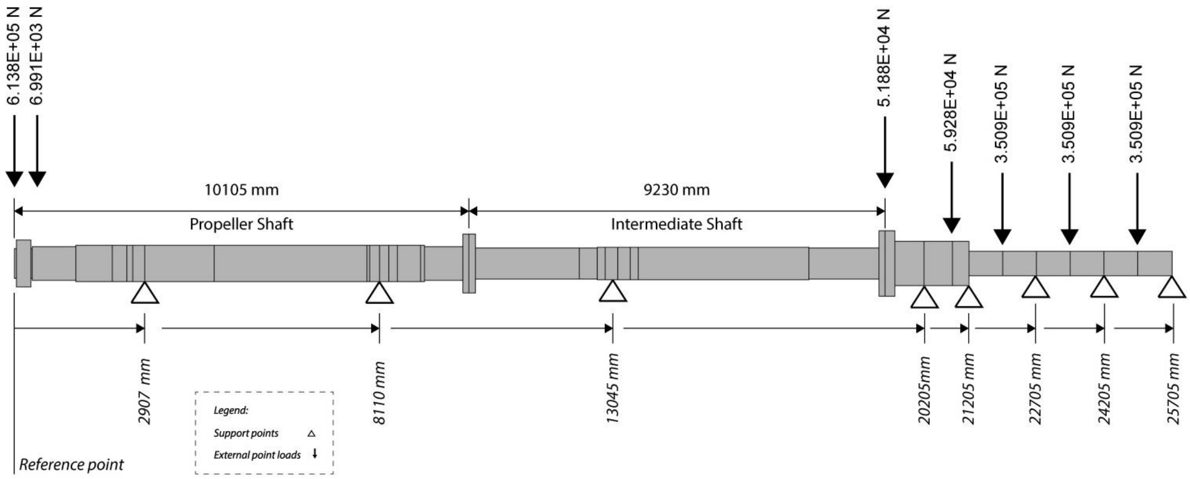


Fig.11: Model of the shafting system of the present study

Initial static shaft alignment plan

A reference line of the shafting system is defined as that passing through the centers of the aft and fore stern tube bearings. Initially, no hull deformations are considered (this case resembles dry-docking conditions of the ship). The line shaft bearing and the engine crankshaft bearings are appropriately offset from the reference line. In Table IV, the corresponding vertical offsets of each bearing are presented, accompanied by the properties of each bearing and by the calculations of reaction forces at each bearing support location.

Table IV: Initial shaft alignment plan: Bearing properties, vertical offsets and reaction forces.

Bearing No.	**Bearing**	**Radial Clearance (mm)**	**Stiffness (N/m)**	**L/D**	**Offsets (mm)**	**Reactions (kN)**	**Mean Pressure (MPa)**
1	Aft S/T	0.550	4.0×10^9	2.221	-0.06	997	0.676
2	For S/T	0.550	5.0×10^9	0.528	0.00	84.4	0.241
3	Intermediate	0.425	1.0×10^9	0.780	-3.90	165	0.426
4	M/E 1	0.345	6.0×10^9	-	-6.60	181	-
5	M/E 2	0.345	6.0×10^9	-	-6.60	301	-
6	M/E 3	0.345	6.0×10^9	-	-6.60	406	-
7	M/E 4	0.345	6.0×10^9	-	-6.60	396	-
8	M/E 5	0.345	6.0×10^9	-	-6.60	161	-

Table III: Discretization details of the shafting system of the present study.

Element		Dist. to right end of element (m)	Length (m)	Diameter (m)		External Load (N)
No.	Type			Left	Right	
1		0.050	0.050	0.650	0.650	
2	Load at right end	0.375	0.325	1.035	1.035	6.99E+06
3		0.405	0.030	0.650	0.650	
4	Load at right end	1.372	0.967	0.726	0.775	6.14E+05
5		2.175	0.803	0.775	0.815	
6		2.505	0.330	0.815	0.815	
7		2.635	0.130	0.815	0.815	
8	Bearing at right end	2.907	0.272	0.815	0.815	
9		4.445	1.538	0.815	0.815	
10		7.835	3.390	0.815	0.815	
11		7.895	0.060	0.817	0.817	
12	Bearing at right end	8.110	0.215	0.817	0.817	
13		8.325	0.215	0.817	0.817	
14		8.505	0.180	0.817	0.817	
15		9.020	0.515	0.817	0.817	
16		9.120	0.100	0.817	0.817	
17		9.970	0.850	0.817	0.705	
18		10.105	0.135	1.320	1.320	
19		10.240	0.135	1.320	1.320	
20		12.555	2.315	0.705	0.705	
21		12.955	0.400	0.705	0.705	
22		13.130	0.175	0.710	0.710	
23	Bearing at right end	13.405	0.275	0.710	0.710	
24		13.680	0.275	0.710	0.710	
25		13.855	0.175	0.710	0.710	
26		17.655	3.800	0.710	0.710	
27		19.200	1.545	0.705	0.705	
28		19.335	0.135	1.458	1.458	
29	Load at right end	19.336	0.001	1.458	1.458	5.19E+04
30		19.555	0.219	1.458	1.458	
31	Bearing at right end	20.205	0.650	0.980	0.980	
32	Load at right end	20.840	0.635	0.980	0.980	5.93E+04
33	Bearing at right end	21.205	0.365	0.980	0.980	
34	Load at right end	21.955	0.750	0.552	0.552	3.51E+05
35	Bearing at right end	22.705	0.750	0.552	0.552	
36	Load at right end	23.455	0.750	0.552	0.552	3.51E+05
37	Bearing at right end	24.205	0.750	0.552	0.552	
38	Load at right end	24.955	0.750	0.552	0.552	3.51E+05
39	Bearing at right end	25.705	0.750	0.552	0.552	

3. Computational Results

3.1. FEM analyses for different loading conditions

Hull deformations have been computed for the three different loading conditions presented in Section 2.1. For loading condition 1 (ballast arrival condition), the hull exhibits a hogging behaviour, which causes considerable displacements at the bearing positions. Figs 12 and 13 show distributions of Von Misses stresses at the ship hull and at the engine room region. For loading condition 2, the hull is bending towards the opposite direction (sagging), whereas for loading condition 3 a hogging at aft and sagging at fore behaviour is exhibited. In Table V and Fig.14, the bearing offsets from the reference line are presented for all loading conditions considered in the present study.

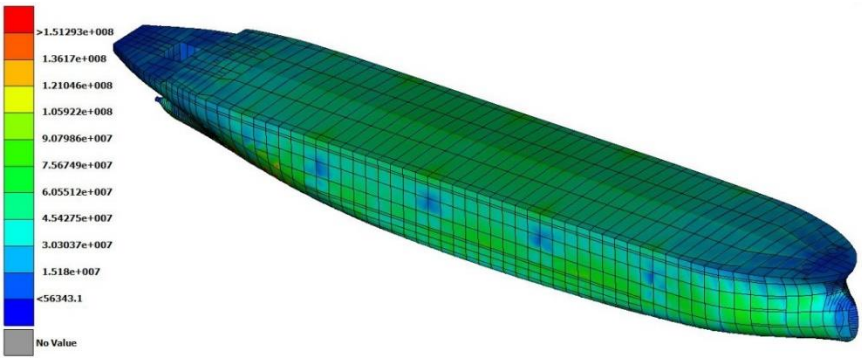


Fig.12: Loading condition 1 (ballast arrival condition): Distribution of Von Misses stresses on hull.

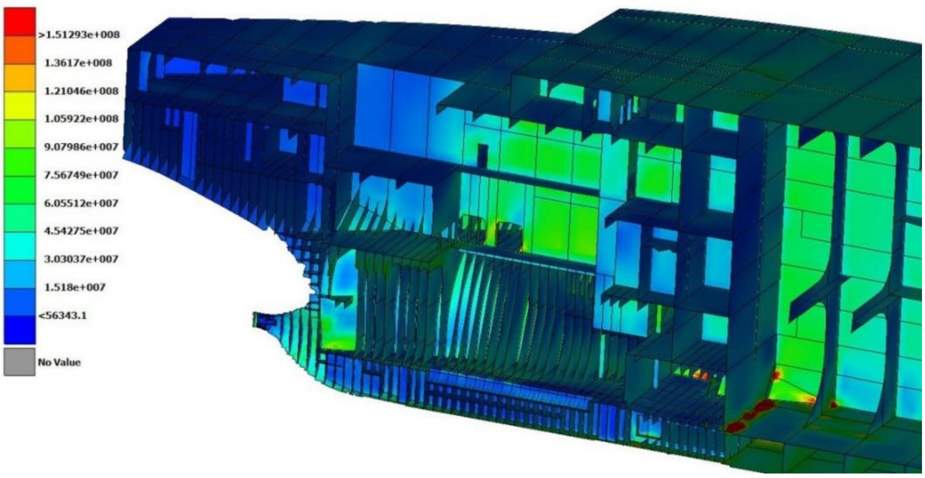


Fig.13: Loading condition 1 (ballast arrival condition): Distribution of Von Misses stresses at engine room region.

Table V: Bearing vertical offsets at different loading conditions of the ship (distance from a reference line passing through the centers of the aft and fore stern tube bearings)

Bearing	Initial case (even keel)	Loading Condition 1	Loading Condition 2	Loading Condition 3
Aft S/T	-0.06	-0.06	-0.06	-0.06
For S/T	0.00	0.00	0.00	0.00
Intermediate	-3.9	-1.89	-4.66	-3.12
M/E 1	-6.60	-2.27	-8.99	-2.44
M/E 2	-6.60	-1.96	-9.32	-2.37
M/E 3	-6.60	-1.50	-9.85	-2.28
M/E 4	-6.60	-1.06	-10.44	-2.20
M/E 5	-6.60	-0.63	-11.07	-2.14

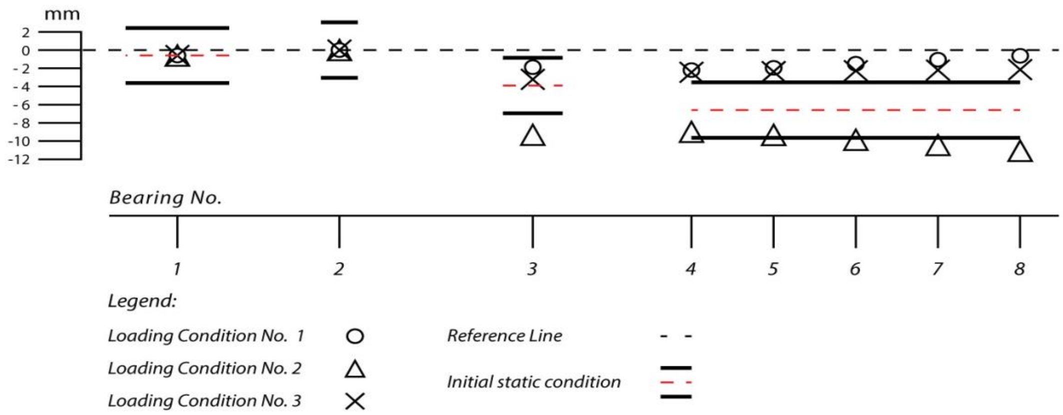


Fig.14: Bearing vertical offsets at different loading conditions of the ship.

3.2. Effects on shaft alignment

In Table VI and Fig.15, the calculated bearing reaction forces are presented for the three different loading conditions studied in the present work.

Table VI: Calculated bearing reaction forces for different loading conditions of the vessel

Bearing No.	**Loading Condition 1**			**Loading Condition 2**			**Loading Condition 3**		
	Bear-ing Offsets (mm)	**Reac-tions (kN)**	**Differ-ence from Initial (%)**	**Bear-ing Offsets (mm)**	**Reac-tions (kN)**	**Differ-ence from Static (%)**	**Bear-ing Offsets (mm)**	**Reac-tions (kN)**	**Differ-ence from Static (%)**
1	-0.06	1030	3%	-0.06	976	-2%	-0.06	995	0%
2	0	25.6	-70%	0	132	56%	0	138	64%
3	-1.89	193	17%	-4.66	127	-23%	-3.12	28.3	-83%
4	-2.27	185	2%	-8.99	204	13%	-2.44	468	159%
5	-1.96	300	0%	-9.32	295	-2%	-2.37	180	-40%
6	-1.50	407	0%	-9.85	406	0%	-2.28	326	-20%
7	-1.06	402	2%	-10.44	416	5%	-2.20	397	0%
8	-0.63	153	-5%	-11.07	137	-15%	-2.14	159	-1%

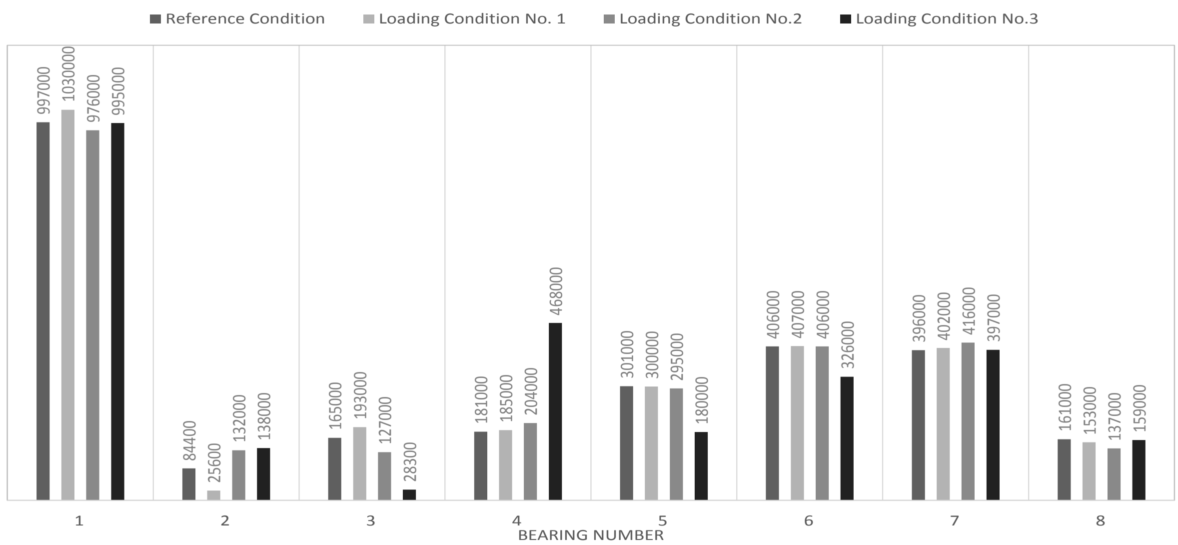


Fig.15: Calculated bearing reaction forces for different loading conditions of the vessel

Overall, although the vertical offsets of certain bearings are of the order of 10 mm, the differences in bearing reaction forces are not very pronounced. In particular, the reaction force of the aft stern tube bearing ranges from 976 kN to 1030 kN (maximum deviation of approximately 5%). Bearing 4 (aft engine bearing) exhibits the maximum deviations in reaction forces, ranging from 181 kN (even keel condition) to 468 kN (L.C. 3). Bearings 7 and 8 display the least amount of deviation.

4. Conclusions

A preliminary study of shaft alignment in a typical VLCC vessel was conducted. A detailed finite element model of the hull structure of the ship was generated; a very fine mesh was utilized at the engine room region of the ship. The propulsion shaft of the ship was modeled as a statically indeterminate multi-supported beam and solved using matrix analysis. Bearing clearance and the stiffness of the bearing foundation were taken into account. First, considering the undeformed (even-keel) hull of the vessel, a reference shaft alignment plan was assumed, and the static equilibrium of the shaft was calculated, yielding the reaction forces at the shaft bearings. Next, three representative loading conditions of the vessel, corresponding to full-load, partial load and ballast conditions were simulated. The corresponding hull deflections were computed, the offset of the bearings due to hull deflections were determined, and the bearing reaction forces were calculated.

In general, the differences in bearing reaction forces at different loading conditions are not very pronounced. At the aft stern tube bearing, the reaction force exhibits a maximum deviation of approximately 5%. The bearing 4 (aft engine bearing) exhibits the most pronounced deviations in reaction forces. The results support, for this specific case and vessel, conclusions drawn by other researchers in recent literature: An appropriate even-keel shaft alignment plan exhibits reasonably good performance at other loading conditions of the vessel. This study could be further extended to account for (a) hot conditions of the engine / application of eccentric thrust loads, (b) the full range of loading conditions of the ship and (c) detailed behaviour of the oil film at each bearing (solution of the Reynolds equation in the lubricant domain).

References

ABS (2004), *Guidance notes on propulsion shafting alignment*, American Bureau of Shipping

BV (2013), *Elastic shaft alignment (ESA)*, Bureau Veritas

DAHLER G.; BRODIN, E.; VARTDAL, B.J.; CHISTENSEN, H.W.; JAKOBSEN, S.B.; OK, Y.K.; HEO, J.H.; PARK, K.R. (2004), *A study on flexible hulls, flexible engines, crank shaft deflections and engine bearing loads for VLCC propulsion machinery*, CIMAC Congress, Kyoto

DEVANNEY, J.; KENNEDY, M. (2003), *The down ratchet and the deterioration of tanker newbuilding standards*, Center for Tankship Excellence

HUGHES, F.O.; PAIK J.K. (2010), *Ship structural analysis and design*, SNAME

MURAWSKI, L. (2005), *Shaft line alignment analysis taking ship construction flexibility and deformations into consideration*, Marine Structures 18, pp. 62–84

NKK (2006), *Guidelines on Shafting Alignment*, Nippon Kaiji Kyokai

SVERKO, D. (2003), *Design concerns in propulsion shafting alignment*, ABS Technical Papers

SVERKO, D. (2006), *A solution to robust shaft alignment design*, ABS Technical Papers