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INFLUENCE OF MARINE MAIN ENGINE FOUNDATIONS ON THE RESULTS OF VIBRATION CALCULATIONS

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Abstract

The article presents an influence of foundations of slow-speed main engine body on the results of numerical analysis of the engine dynamic stiffnesses and thermal deformations. The engine body is much stiffer than its foundation pads and ship hull (double bottom) – boundary conditions of the engine. Especially for the high power, marine engines, the correct model of the boundary conditions plays a key role during the analyses. Therefore, modelling method of engine foundation (boundary conditions) of that kind of model is essential during the analyses. During shaft line alignment and crankshaft springing analyses, knowledge of dynamic stiffnesses characteristics and thermal displacements of radial (main) bearings is significant. Those data of marine main engine body are difficult to estimate because of lack of available documentation and complicated shape of the engine and ship hull. The article presents the methodology of the characteristics determination of the marine engine's body as well as the example of computations for a MAN B&W K98MC type engine (power: 40000 kW, revolutions: 94 rpm) mounted on a 3000 TEU (twenty-foot container equivalent unit) container ship (length: 250 m). Numerical analyses were performed with usage of Nastran software based on Finite Element Method. The FEM model of the engine body comprised over 800 thousand degree of freedom.

Keywords: *marine engines, shaft line alignment, crankshaft springing, boundary conditions of propulsion system*

1. Introduction

Slow-speed (60-180 rpm) main engine connected directly by shaft line (intermediate shafts and propeller shaft) with propeller is typical for merchant ships [4]. In that, propulsion system there is no gears or flexible couplings. Efficiency is a main reason for so simple propulsion system. Usually main engine is powerful – above 20000 kW.

Power transmission system (crankshaft plus shaft line) is loaded by strongly unsymmetrical perpendicular forces. Especially stern tube bearing is loaded by very heavy propeller from one side. Proper shaft line alignment and crankshaft springing is one of the most important procedures during marine propulsion system designing, installation, and exploitation. The axis of journal bearings of shaft line should be displaced (mainly in vertical direction) to the proper position [5, 8, 9]. Usually, the crankshaft axis is a baseline for shaft line alignment.

The target of the presented research is evaluation of displacements of the crankshaft and shaft line axis in the propulsion systems multiple working conditions [8]. Up to now in the shaft line alignment and crankshaft springing analyses methodology an interaction of the crankshaft and shaft line was considered in a simplified way [3]. The crankshaft was modelled as a linear system of cylindrical beam elements, while it's displacements due to working temperature and it's foundation stiffness were evaluated based on a simple data supplied by the producer. For example, the data did not address the type of the ship (boundary conditions) on which the engine is mounted [9]. Better mathematical model of the boundary conditions of the marine power transmission system is the aim of presented part of the research. Accurate analyses (with detailed boundary conditions) are especially important for the high power propulsion systems. In the literature there may be found numerous examples of the damage of the first three (counting from the driving end) main bearings of the main engine [1, 7].

Within the research there have been carried out a number of analyses of MAN B&W K98MC type engine mounted on a big container ship and modelled as a boundary conditions. The computation of the engine's body deformation due to the gravity has been performed as well as the analysis of its natural dynamic characteristics. The static and dynamic stiffness (horizontal and vertical) of each of the main bearings have been evaluated. The displacements of the crankshaft axis under a steady-state thermal load have been also determined.

2. Model of boundary conditions of the engine body

All analyses were performed on the base of Finite Element Method [6]. Commercial software: Patran-Nastran was used for modelling and numerical calculations. The FEM model of the B&W K98MC main engine's body has been presented in Fig. 1. Foundation of crankshaft in the main bearings is the most important region in presented type of analysis. FEM model of main bearings is realised by 3-D solid elements (8-nodes), other part of engine body is modelled by 4-nodes plate elements. The whole FEM model of the engine' body has over 800 thousands degrees of freedom. Engine model is 8 times (!) greater than model of the ship hull (see Fig. 2). It is the main reason for separate calculations of the engine temperature deformations and stiffnesses and the ship hull characteristics.

Fig. 1. FEM model of engine body

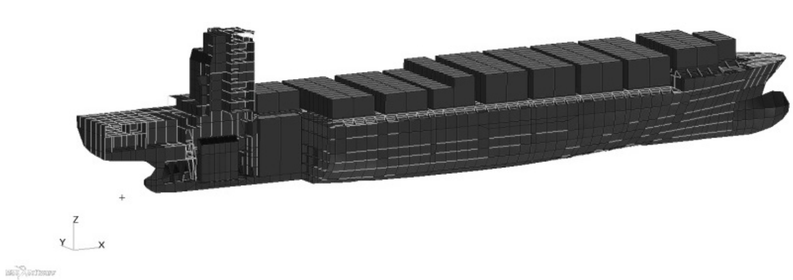

Fig. 2. FEM model container ship

The analysis consists of the following steps:

1. Ship hull stiffnesses determination with particular focus on foundations of the propulsion system.
2. Preparation of simplified models of stiffnesses of a ship hull in order to use them in engine body calculations.
3. Verification of a ship hull simplified models by dynamic analyses of an engine body with different boundary conditions.
4. Determination of thermal deformation of an engine body (with selected boundary conditions) performed on the base of measured temperature distribution.
5. Determination of a ship hull deformation (with special emphasis on propulsion system foundation) performed on the base of steady state transient thermal (coming from an engine body) analysis.
6. Repeated calculations of thermal deformation of an engine body carried out on the base of more detailed boundary conditions, taking into account thermal deformation of a ship hull in an engine room area.

Stiffness of the ship hull is essential during presented analyses. On the base of the separate analyses, stiffness of ship hull in the engine room area (with fundaments) was estimated and its value is equal to 1.1×10^9 N/m. The detailed model of the engine body has to be analysed as separated from the ship hull. Three types of engine foundation model (boundary conditions) were analysed. First one is classical – known from literature: foundation arms are completely blocked (fixed deformation). In the second way, the ship hull stiffness was modelled by beam elements. This method does not take into account couplings between supporting points of the ship hull (the ship hull is treated as a continuous beam). In the third method, the foundation arms are modelled by continuous cuboid (with the cross section 0.468×0.5 m) with special material properties. Area of all propulsion systems bearings' foundation was distinguished and loaded by unitary pressure [3]. Displacements of the bearings give me the value of the ship hull local stiffness. During separate calculations, the properties of the vicarious cuboid were determined in the way that the local stiffness of the cuboids was equal to the local stiffness of the ship hull with the engine foundation. The Young's modulus of the cuboids was determined as $E = 9.2\times10^9$ Pa.

The model of the engine body was verified by natural vibrations determinations. The main target of that kind of analysis is model coherence checking. In the author opinion, each FEM model (even made up for static type analysis) should be checked by natural modes analysis. It was assumed that dynamic stiffness of engine main bearings will be performed in the range of 0-30 Hz (engine's main force harmonic component is equal to 10.97 Hz and the propeller's is equal to 7.83 Hz). As the analysis of forced vibration has been performed with the use of modal superposition method, the prior determination of the natural frequencies and eigenvalues in the range of 0-70 Hz has been necessary. First 50 (up to 14 Hz) normal modes are determined for the ship. The first normal mode of the ship hull (1.59 Hz), the superstructure (6.57 Hz) and the main engine body (8.77 Hz) are presented in Fig. 3-5.

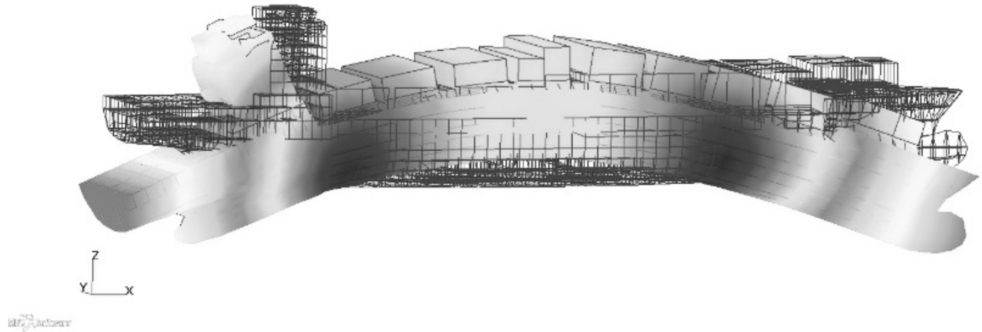

Fig. 3. The first natural mode of a ship hull (vertical 2-nodes)

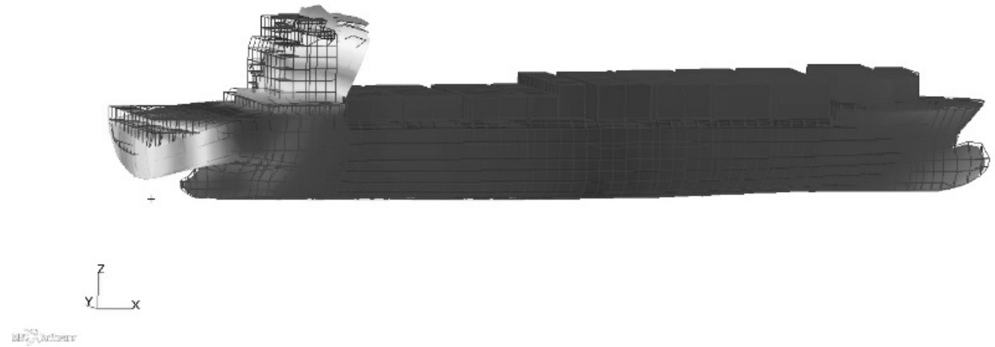

Fig. 4. The first natural mode of a ship superstructure

Fig. 5. The first natural mode of a main engine body

Values of natural frequencies for each type of boundary conditions (the modelling method of the ship hull and the engine foundation) were compared. While the boundary conditions have not very important influence on natural frequencies of the main bearings foundations, these conditions affect the global engine eigenvalues very much. The modelling method of the boundary conditions (the ship hull stiffness with the engine foundation) is essential during engine body analyses. Fixed nodes in the foundation arms area give us too stiff model but hull stiffness modelled by beams gives us too elastic model (because of not taking into account couplings between hull areas). Model with cuboid foundation is the best and it is consistent with author's experience.

3. Thermal analysis of main engine body

As an example, analysis of thermal deformation of main engine body is presented in the article. Determination of crankshaft axis deformation is the main target of the calculations [2]. Before the start of the thermal deformation analysis of the engine body it is necessary to determine the engine temperature distribution. The temperature map has been created on the basis of the measurements carried out on a marine main engine during sea trials. The temperature determination on the base of measurements is much more accurate in comparison to calculation analysis of heat transfer.

In the analysis, determination of thermal deformation of the engine body was performed, taking into account the ship hull stiffness and thermal deformations of the engine foundations. The model of the part of the hull was simplified by a 3-D cuboid beam with special material properties, but in the bottom part deformation of the ship hull was added. Thermal deformation of the engine body was performed on the base of measured on-board temperature distribution. The results for the whole engine and the foundations of the main bearings are presented in Fig. 6 and 7. The numerically computed average value of the translation of the crankshaft's axis (0.46 mm) is greater than the one recommended by the producer (0.37 mm). The difference is not particularly big (bellow 20%), but the displacement is of a hogging type. It seems that the producer's assumption about the parallel translation of the crankshaft's axis is incorrect.

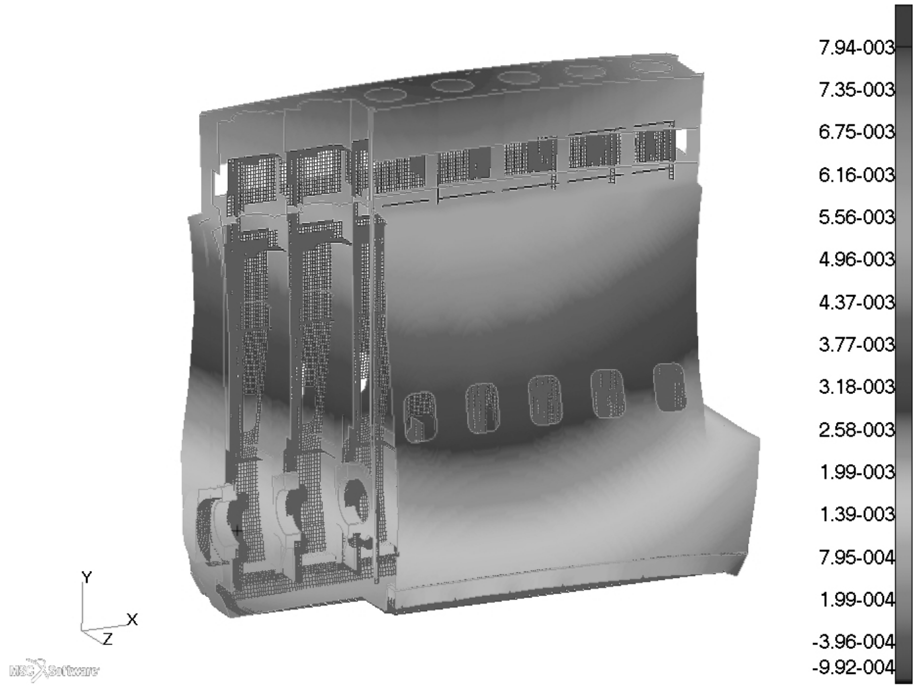

Fig. 6. Thermal deformation of engine body

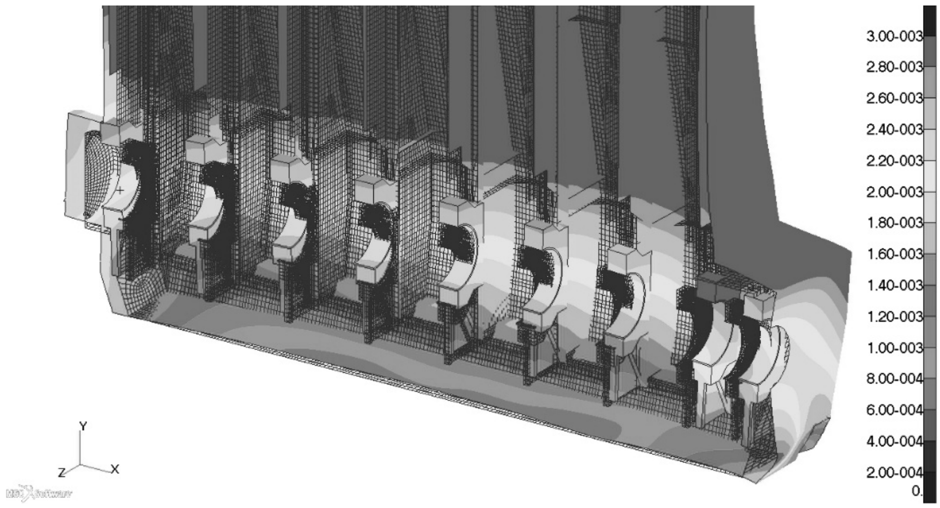

Fig. 7. Thermal deformation of the main bearings with the influence of the ship hull thermal deformation

From a point of view of the propulsion system and the main engine – shaft line cooperation, the most important are the displacements of the main bearings of the engine. A diagram of the vertical and horizontal thermal displacement of the crankshaft axis is presented in Fig. 8. The circles show the places of the main bearings. Horizontal deformations of the crankshaft axis under the heating are much smaller in comparison to the vertical. It confirms that the shaft line alignment could be performed only in the vertical plane.

Deformations presented in Fig. 8 are calculated according to the absolute coordination system. From the engine exploitation point of view, the most important are relative deformation of the crankshaft axis against the ship hull (deformed by temperature). Therefore, vertical deformations of double bottom (the engine foundation) are subtracted from absolute vertical deformations of the main bearings foundations. Relative thermal displacement of the crankshaft axis is presented in Fig. 9. In the diagram, the lines of total vertical thermal displacement of the engine bearings in hot condition, according to MAN B&W (he = 0.37 mm) as well as mean value of the calculated vertical displacement are drawn.

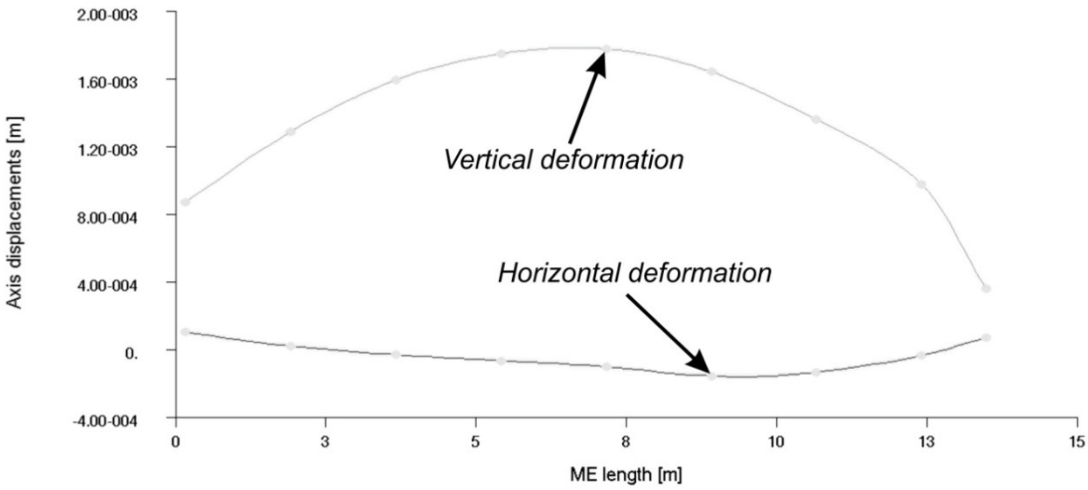

Fig. 8. Diagram of the thermal displacement of the crankshaft axis with the influence of the ship hull thermal deformation

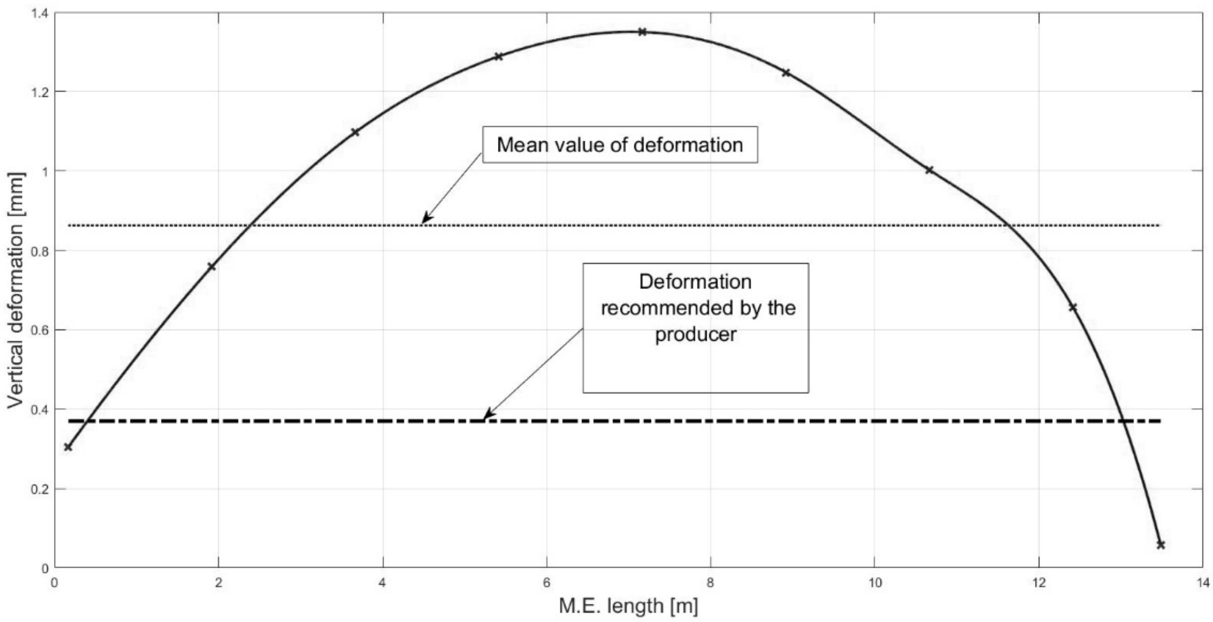

Fig. 9. Diagram of relative thermal displacement of the crankshaft axis with the influence of the ship hull thermal deformation

4. Conclusions

Finite stiffnesses of a foundation of a shaft line mating with its bearings' reactions are the cause of local deformations of a ship hull structure. The boundary conditions can be modelled as a single, linear spring. Foundation of an engine body should not be modelled by local supports because 1-D spring does not take into account couplings between supporting points of a ship hull and an engine body. The author proposed modelling method of engine foundation arms. The boundary conditions should be modelled by a continuous cuboid with special material properties. The property (Young modulus) should have local stiffness equal to local stiffness of an analysed ship hull (double bottom) with the engine foundation.

Horizontal deformations of the crankshaft axis (and also shaft line axis) under the heating are negligible, in spite of the temperature differences between left and right sides of the engine body. This is in accordance with the marine engines manufacturers' recommendations. They recommend that the shaft line alignment be performed only in a vertical plane.

The numerically computed average value of the translation of the crankshaft's axis (taking into account thermal deformation of the main engine body and the ship hull) is greater than the one recommended by the producer. However, the difference (between producer's data and the calculated by the author) of displacement of the driving end of the crankshaft is in the acceptable error band (17.5%). It is important that the calculated displacement of the crankshaft axis be of a hogging type. It seems that the producer's assumption about the parallel translation of the crankshaft's axis is only a rough approximation. The hogging type of deformation can have significant influence on the loads values coming from the shaft line (additional bending moment and shear force acting between the crankshaft and the shaft line). The effect seems to be considerable in the precise shaft line alignment analysis. Presented conclusion should be treated with caution because connection between the shaft line and the crankshaft is smooth. Inflection point is located few meters away from the coupling. Detailed shaft line alignment calculations will show the influence of the presented deformation line of the crankshaft axis. If a ship hull is elastic, the engine's main bearings should be placed accordingly to the relative thermal displacement (the first 3-4 bearings should be placed higher and higher) but it depends on the ship hull type. The thermal deformation of the power transmission system axis may have also big influence on the crankshaft springing, especially for the first cranks. It also means that the loading value of the first three main bearings may be higher than in the theory, in the classical shaft line alignment analyses.

The presented direction of research looks promising. It may allow improving installation of high power propulsion systems and avoiding failure of the engine's main bearings. The worked out methodology may be used for more advanced and complete numerical computations for multiple main engine types together with specific ships' hulls. As a further step, the propulsion system analysis methodology should be elaborated, which incorporates more complex crankshaft representation including its full 3D characteristics. The effect of a crankshaft's springing on the shaft line alignment should also be examined further.

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