Boundary conditions modelling method of marine main engine body

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Abstract

The paper presents numerical analysis of dynamic stiffnesses and thermal deformations of marine, slow-speed main engine body. 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 boundary conditions of that kind of model is essential during the analyses. During shaftline 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 paper 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 preformed 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 shaftline 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 shaftline should be displaced (mainly in vertical direction) to the proper position [5, 8, 9]. Usually, the crankshaft axis is a baseline for shaftline alignment.

The target of the presented research is evaluation of displacements of the crankshaft and shaftline axis in the propulsion system's 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 Figure 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.

Figure 1: FEM model of engine body

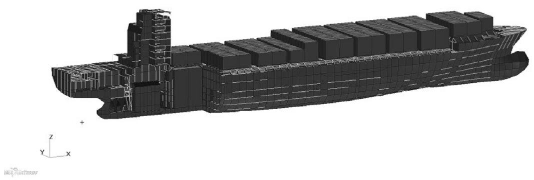

Figure 2: FEM model container ship

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 0.468x0.5 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 $E=9.2 \times 10^9$ 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.

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 paper. 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. A thermal deformation of the main engine's body is presented in Figure 3. 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.

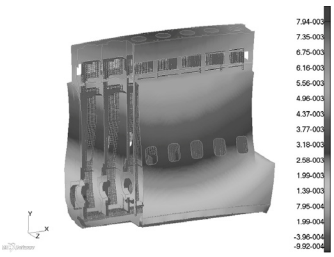

Figure 3: Thermal deformation of engine body

This direction of research looks very promising. It may allow improvement in 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 ship's hulls.

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