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Thermal interaction between main engine body and ship hull

Lech Murawski

Gdynia Maritime University, ul. Morska 81-87, 81-225 Gdynia, Poland

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ABSTRACT

The paper presents analysis of displacement of a propulsion system shaft line and a crankshaft axis caused by temperature of marine, slow-speed main engine. Detailed information of thermal displacement of a power transmission system axis is significant during a shaft line alignment and a crankshaft springing analysis and during designing of structural health monitoring (SHM) system. A warmed-up main engine is a source of deformations of an engine body as well as a ship hull in the area of an engine room and hence axis of a crankshaft and a shaft line. Engines' producers recommend the model of parallel displacement of the crankshaft axis under the influence of an engine heat. This model may be too simple in some cases (especially for SHM systems).

The paper presents computations of MAN B&W K98MC type engine mounted on 3000 TEU container ship. 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 correct model of the boundary conditions plays a key role during the analyses. Presented numerical analyses are based on temperature measurements of the main engine body and the ship hull during a sea voyage. Numerical analyses were performed using Nastran software based on Finite Element Method. The FEM model of the engine body comprised over 800 thousand degree of freedom (dof); the model of the ship hull contains over 200 thousand dof. Both models are analysed separately; the mutually interaction between them is taken into account by heat transfer and special model of boundary conditions. The specialized SHM system dedicated to marine propulsion systems working in very bad environmental conditions is the future aim of the research.

1. Introduction

Reliability of a marine propulsion system is closely correlated with the safety of navigation at sea. Two-stroke, slow speed main engines have been installed mostly on merchant ships since the late 70s. The engines are connected to a directly driven propeller by a relatively short shaft line (Murawski, 2003). In that propulsion systems there are no gears or flexible couplings. Structural health monitoring systems of the power transmission system is one of the main ways for improving safety at sea. Especially, reliable SHM systems working in heavy sea state need detailed data and characteristics of the propulsion system.

Power transmission system (crankshaft plus shaft line) is loaded by strongly unsymmetrical perpendicular forces. Especially stern tube bearing is loaded by very heavy propeller only from one side. Proper shaft line alignment and crankshaft springing are one of the most important parameters during marine propulsion system reliability evaluation. Shaft line alignment consist in appropriate bearings moving (usually vertical) with the aim of its proper loads. Crankshaft springing is a standard method of checking loads of engine main bearings. The crank deformation is measured during slow crankshaft revolution with a turning gear usage. Shaft line alignment is closely related with crankshaft springing. During the shaft line alignment and crankshaft springing analyses, knowledge of the thermal displacement of the crankshaft axis is essential (Murawski, 2016). Engines' producers suggest the model of crankshaft axis thermal displacement but it is very simple. They recommend the model of parallel displacement of a crankshaft axis under the influence of engine heat. Such a model gives us one number - a value of crankshaft displacement between cold and hot propulsion system (in steady state condition), different for each type of an engine. Sometimes, the displacement value is depended on the temperature difference between not running and running propulsion system. In such a case the model might be too simple. Couplings between thermal (in some cases with influence of electromagnetic field) and mechanical loads of a marine propulsion system are very complicated and cannot be omitted during analyses of a main engine mounting (Song and Moon, 2016). Problems with shaft line alignment and crankshaft springing are befalling especially for high powered engine mounted on optimised (light - with elastic hull) ships (Fontea et al., 2015; Simm et al., 2016; Yang et al., 2011).

E-mail address: lemur@wm.am.gdynia.pl.

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Practically, a relatively simple model of the thermal-mechanical coupling is sufficient: the engine temperature is a source of its body (main bearings and crankshaft) and ship hull displacement. There's no need to take into account the full coupling effects (occur with big elastic-plastic deformations with high temperature) between deformation and temperature fields because the temperature level is relatively low. The temperature of the main engine body is less than 70 °C (see Fig. 8) and ship hull has no more than 42 °C (see Table 2). Similarly, material (steel) properties was assumed as constant owing to low temperature level. More important is mutual interaction between the engine body and the ship hull. The heat transfer is a source of the ship hull deformation (especially in the engine room area) and the hull deformation is a source of added deformation of the engine body. Finally, the displacements are a source of additional mechanical loads of propulsion system's bearings and can be measured by the SHM system.

The target of the presented analyses is evaluation of displacements of the crankshaft and shaft line axis in the propulsion system's multiple and steady-state thermal working conditions (Germanisher Lloyd, 2012; American Bureau of Shipping, 2004; Det Norske Veritas, 2013). The hull deflections caused by buoyancy can be omitted because final shaft line alignment has to be done after ship launching (still water case). Sea waves and ship loading condition (cargo) have also an influence on shaft line alignment and crankshaft springing. But their characteristics and type of influence is different; those problems have been discussed in other author's work (Murawski, 2003, 2005a). Up to now in the shaft line alignment and crankshaft springing analyses methodology an interaction of the crankshaft and shaft line was analysed in a simplified way (Murawski, 2005a). The crankshaft displacements were modelled due to working temperature and its foundation stiffness. Those data were evaluated based on a simple data supplied by the producer. The data did not address the type of the ship (boundary conditions) on which the engine is mounted (Im et al., 2017; MAN B&W Diesel A/S, 1995). A better mathematical model of the boundary conditions of the marine power transmission system is the aim of presented research. 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 (Fontea et al., 2015; MAN B&W Diesel A/S, 2012). One of the causes might be the imprecise mathematical model of a crankshaft proposed for the shaft line alignment analysis.

There have been carried out a number of analyses of MAN B&W K98MC type engine mounted on a 3000 TEU container ship. Marine power transmission systems are usually modelled as isolated from ship hull and engine body (Andersen and Jensen, 2014; Iijima et al., 2008; Murawski and Charchalis, 2014; Senjanović et al., 2014a). There are several reasons for this methodology. Difficulties with the proper oil film calculation (Reynolds's equation) and the need for a detailed crankshaft, an engine body and a ship hull FEM model are one of the most important. This methodology (separately computation of ship hull and engine) was applied in the analyses. The computation of the engine's body deformation due to the gravity and its natural eigenvectors has been performed as well as the analysis of its thermal deformation in nominal work conditions. Also the computation of the ship hull eigenvectors and local stiffness has been performed as well as the analysis of its thermal deformation with special emphasis on main engine room area. The ship hull static and dynamic stiffness characteristics were used for proper modelling the engine boundary conditions. Natural vibrations (eigenvectors) analyses were performed for checking models consistency. The thermal analyses require an accurate temperature distribution on the engine's body and also on the ship hull (for checking numerical calculations). Wide temperature measurements on the ship and her main engine supplied the proper data. The temperature measurements were performed during a sea voyage.

2. Analysis methodology

All analyses were performed on the base of Finite Element Method (Zienkiewicz and Taylor, 2000). The Finite Element Method is a numerical technique that gives approximate solutions to differential equations that model problems arising in physics and engineering. As in simple finite difference schemes, the finite element method requires a problem defined in geometrical space (or domain), to be subdivided into a finite number of smaller regions (a mesh). In finite elements, each subdivision is unique and need not be orthogonal. For example, triangles or quadrilaterals can be used in two dimensions, and tetrahedra or hexahedra in three dimensions. Over each finite element, the unknown variables (e.g. temperature, velocity etc.) are approximated using known functions; these functions can be linear or higher-order polynomial expansions in terms of the geometrical locations (nodes) used to define the finite element shape. The governing equations in the finite element method are integrated over each finite element and the contributions summed over the entire problem domain. As a consequence of this procedure, a set of finite linear equations is obtained in terms of the set of unknown parameters over the elements. Solutions of these equations are achieved using linear algebra techniques. Eight steps of the FEM calculation can be distinguished:

- Discretize and Select the Element Types.
- Select a Displacement Function.
- Define the Strain/Displacement and Stress/strain Relationships.

In the case of one-dimensional deformation, in the x direction, we have strain $\boldsymbol{\varepsilon_x}$ related to displacement $\boldsymbol{u}$ described by:

$$\varepsilon_x = \frac{du}{dx} \tag{1}$$

Equation (1) applies to small strains. In addition, the stresses must be related to the strains through the stress/strain law, generally called the constitutive law. The ability to define the material behavior accurately is most important in obtaining acceptable results. The simplest of stress/strain laws, Hooke's law, which is often used in stress analysis, has the form:

$$\sigma_x = E\varepsilon_x \tag{2}$$

where σ_x is the stress in the x direction, and E is the modulus of elasticity.

- Derive the Element Stiffness Matrix and Equations.
- Assemble the Element Equations to Obtain the Global or Total Equations and Introduce Boundary Conditions.

In this step, the individual element nodal equilibrium equations are generated. After that, the characteristic matrix of finite elements is assembled into the global nodal equilibrium equations. The implication of the direct stiffness method is the concept of continuity, or compatibility, which requires that the structure remains together. The finally assembled, global equation for dynamic problems is written in matrix form as follows:

$$M\ddot{u} + C\dot{u} + Ku = F \tag{3}$$

where $\boldsymbol{F}$ is the force vector, $\boldsymbol{K}$ is the stiffness matrix, u is the displacement vector, $\boldsymbol{C}$ is the damping matrix, and $\boldsymbol{M}$ is the mass matrix.

- Solve for the Unknown Degrees of Freedom.

Equation (3) can be simplified to static strength analyses by expunction of elements with acceleration vector (mass matrix) and speed vector (damping matrix). Simplified equation modified by the boundary conditions (limits of some of the displacements), is a set of simultaneous algebraic equations that can be written in expanded matrix form as follows: