ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1213578

STIFFNESS CHARACTERISTICS OF MAIN BEARINGS FOUNDATION OF MARINE ENGINE

Lech Murawski

Gdynia Maritime University, Faculty of Marine Engineering
Morska Street 81/87, 81-225 Gdynia, Poland
tel.:+48-58-6901 481
e-mail: lemur@wm.am.gdynia.pl

Abstract

During shaftline alignment and crankshaft springing analyses, knowledge of stiffness of radial bearings is essential. Dynamic stiffness's characteristics of marine main body are difficult to estimate because of lack of available documentation and complicated shape of the body. In the literature, there is a lack of the detailed data on the stiffness of the crankshaft foundation in the frame of marine main engine. Those parameters are crucial for the shaft line alignment analysis as well as for the analysis of interactions between the shaft line and the crankshaft. Especially for the high power engines, the correct model of the boundary conditions plays a key role during the analysis. 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 mounted on a ~3000 TEU container ship. The model of main engine body is relatively big – it contains over 800 thousand degree of freedom. The elasticity of ship hull has been estimated and taken into account during analysis. It has been found, that static stiffness parameters recommended by the producer for the shaft line alignment are evaluated correctly, however they represent only the flexibility of the engine's body, not taking into account the flexibility of the ship's hull. What is more, the dynamic magnification of vibration amplitudes is not taken into account.

Keywords: *dynamic stiffness of main engine bearings, shaft line alignment, crankshaft springing, marine propulsion system numerical analysis*

1. Introduction

The target of the presented research is evaluation of displacements of the crankshaft and shaftline axis in the propulsion system multiple working conditions [6, 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 [1]. 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. The data did not address the type of the ship on which the engine is mounted [7]. 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 [4, 5].

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. 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 main data of the analysed ship and its propulsion system are as follows:

Hull of the container ship:

- Total length 291.66 m,
- Length between perpendiculars 271.20 m,
- Width 32.20 m,

- Design draught 12.00 m,
- Maximal draught 13.20 m,
- Maximal caring capacity 58000 t,
- Container No. 4546 TEU,
- Ship's speed with 90% MCR 24.0 knots.

Main engine:
- Type 7 K 98 MC,
- Power 40040 kW,
- Nominal revolutions 94 rpm,
- Mean indicated pressure 19.2 bar,
- Stroke 2660 mm,
- Cylinder bore diameter 980 mm,
- Oscillating mass per cylinder 18149 kg,
- Connecting rod ratio 0.413,
- Flywheel 13629 kgm^2,
- Crankshaft journal diameter 1062 / 400 mm,
- Crankshaft pin diameter 1062 / 531 mm,
- Firing order 1725436.

Shaft line:
- Intermediate shafts diameter 735 mm,
- Propeller shaft diameter 845 mm.

Propeller:
- Propeller diameter 8.20 m,
- Number of blades 5,
- Pitch ratio (mean) 1.045,
- Blade area ratio 0.8505,
- Mass in air 72200 kg,
- Moment of inertia in air 236000 kgm^2.

2. Analyses methodology – model verification

All analyses were performed on the base of Finite Element Method [3]. Commercial software: Patran – Nastran was used for modelling and numerical calculations. The FE model of the B&W K98MC main engine's body has been presented in Fig. 1. Fig. 2 presents a part of the model with details of the engine's main bearing foundation. 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 812 thousands degrees of freedom and over 170 thousands elements. Engine model is 8 times (!) greater than model of the ship hull (see Fig. 3). It is a main reason that calculations of engine and ship hull stiffness have to be performed separately.

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. FEM model of the container ship for those analyses is presented in Fig. 3.

It is obvious that detailed model of engine body have to be analysed as separated from 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 do not take into account couplings between supporting points of ship hull (ship hull is a continuous beam). In the third method, the foundations' arms are modelled by continuous cuboid (with cross section 0.468x0.5 m) with special material properties. During separate calculations, the properties were determined in that way that the local stiffness of the cuboid is equal to local stiffness of the ship hull with engine foundation. The Young's modulus was determined as $E=9.2x10^9$ Pa.

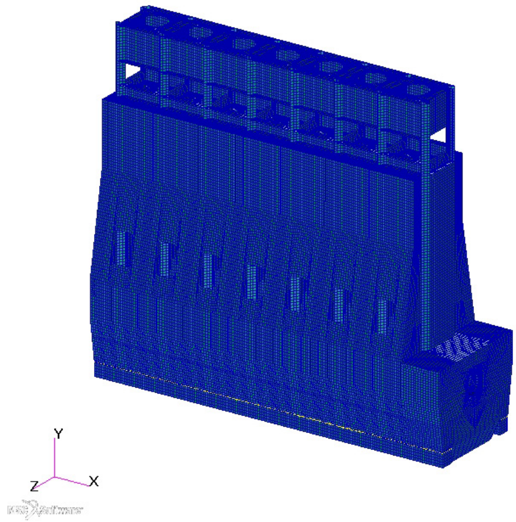

Fig. 1. FEM model of the B&W K98MC main engine's body

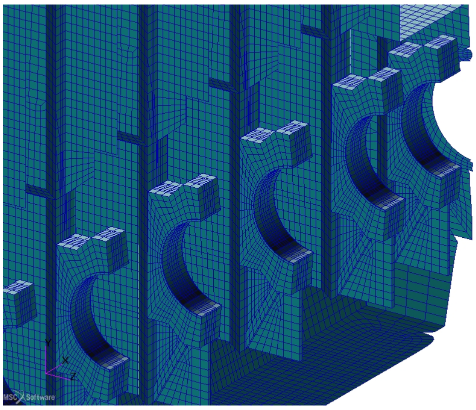

Fig. 2. Details of the engine's main bearing foundation

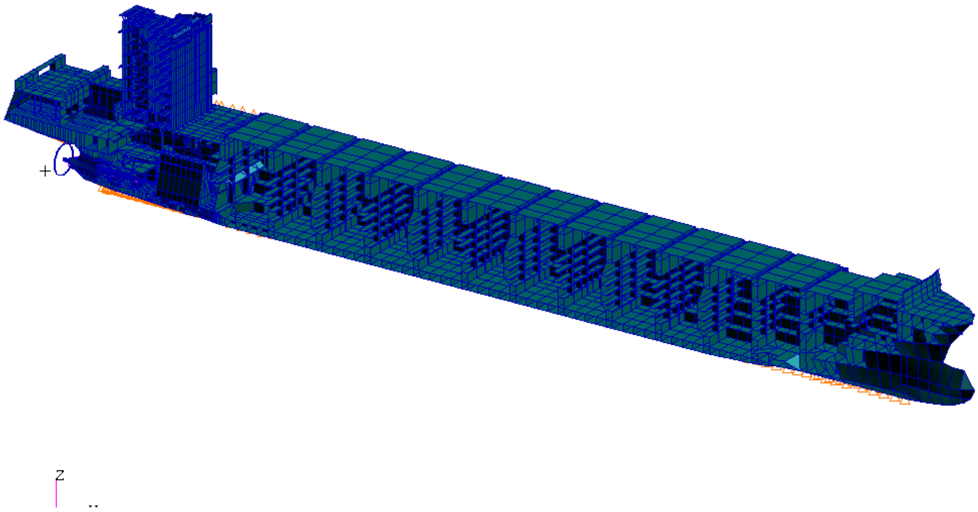

Fig. 3. FEM model of 4500 TEU container ship

The model of the engine body was verified by eigenvalue vectors determinations. 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. Examples for most interesting natural modes are presented in Fig. 4-6.

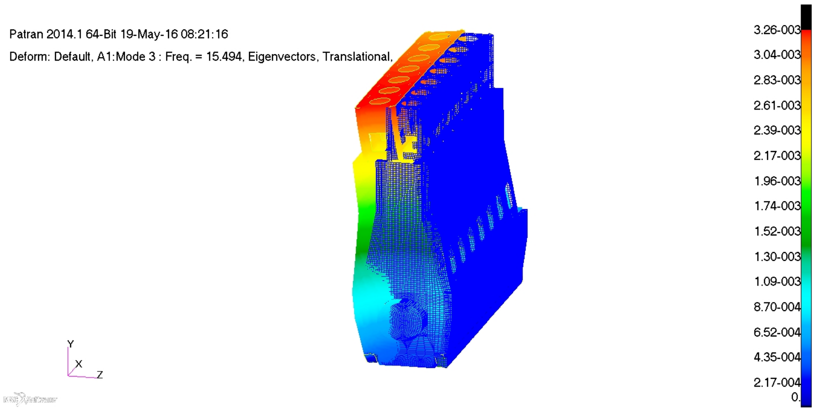

Fig. 4. H-mode of engine body natural vibrations

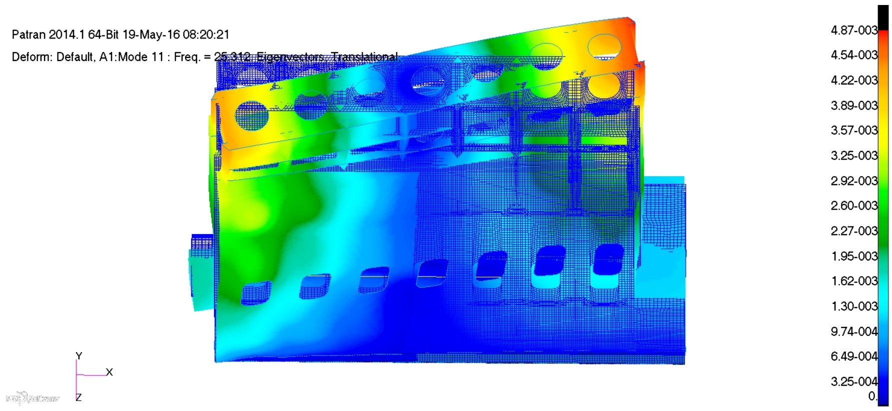

Fig. 5. X-mode of engine body natural vibrations

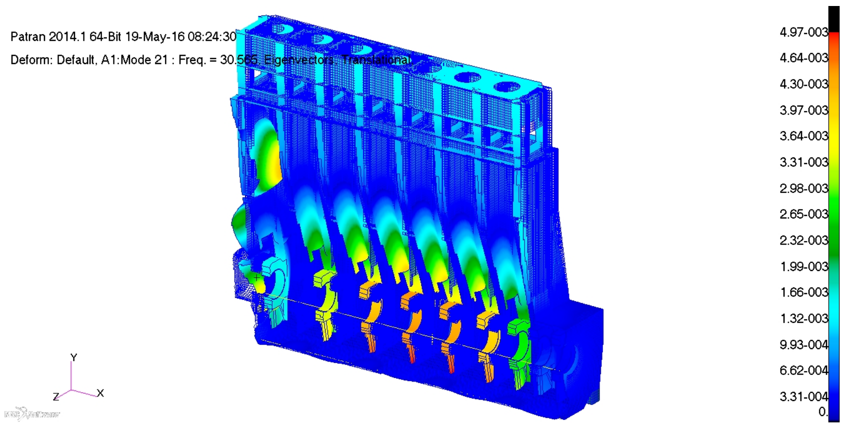

Fig. 6. Mode of main bearings foundation natural vibrations

Values of natural frequencies are presented in Tab. 1 for each type of boundary conditions (modelling method of ship hull and engine foundation). While the boundary conditions have not very important influence on natural frequencies of main bearings foundations, the global engine eigenvalues are completely different. The modelling method of boundary conditions (ship hull stiffness with engine foundation) is essential during engine body analyses. Fixed nodes in the foundation arms area gives 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 experience (measurements on-board). During further calculations, cuboid model will be analysed. On the base of natural vibrations analyses, it may be observed that stiffness of the engine body (especially main bearings foundations) is high. It is much higher than primary excitation frequencies of propulsion system. Therefore, dynamic stiffness of the engine bearings should not be much differs from static stiffness.

Tab. 1. Natural frequencies of engine body for different boundary conditions

Eigenvector type	Fixed foundation arms	Ship stiffness modelled by beams	Ship stiffness modelled by cuboid
H	22.6 Hz	12.4 Hz	15.5 Hz
X	31.3 Hz	24.1 Hz	25.3 Hz
Main bearings	31.4 Hz	29.1 Hz	30.6 Hz

3. Stiffness analyses of main bearings foundation

Determination of the static stiffness consists in applying unitary forces equal to mass forces and radial gas forces (750 kN), to each of main bearings one by one, first in vertical and then in horizontal direction. The acquired displacements serve to calculate the local static stiffness. The main engine producers give the crankshaft's foundation stiffness (on request, but it is not included in official documents), but without subdivision to engine's body and ship's hull stiffness. Sometimes ago this parameter was assumed as infinitely great, now it is considered as grade of 6.0×10^9 N/m. The author has vast experience with the stiffness of multiply multiple types of the hull [2].

In the first place, the deformation of the main engine body under only gravity load has been computed. It has been observed that the deformation from gravity was several times smaller (it is of the grade of 0.01 mm), than the deformation from the thermal load and cylinder mass and gas forces. Thus, the influence of gravity may be neglected in further analysis.

Next the static analysis has been carried out consisting in application of the load in vertical and then in horizontal direction (18 load cases for computation). Estimated radial forces have been applied in continuous way on the surface of the main bearing. Selected deformations under vertical and horizontal static forces are presented in Fig. 7, 8. Tab. 2 contains static stiffnesses for particular main bearings. The stress level in the structure of main bearings is not high – it does not exceed 15 MPa for horizontal load and 22 MPa for vertical one.

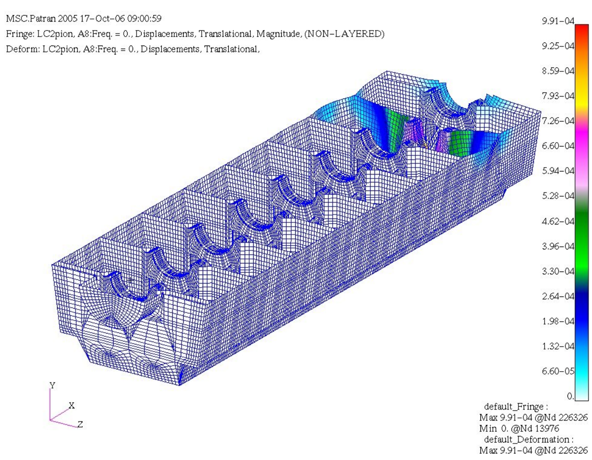

Fig. 7. Deformation of main bearing foundation under vertical force

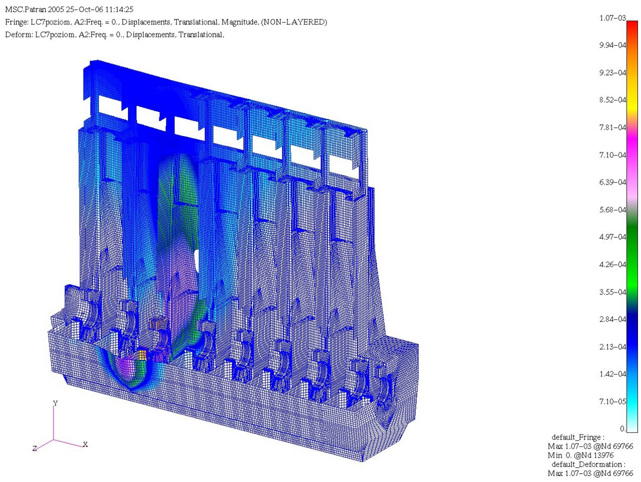

Fig. 8. Deformation of main bearing foundation under horizontal force

Tab. 2. Static stiffnesses of main bearings foundations

Main Bearing No.	*Vertical Stiffness* $\times 10^{10}$ *[N/m]*	*Horizontal Stiffness* $\times 10^{10}$ *[N/m]*
1	1.06786	1.18757
2	1.12869	1.15801
3	1.09631	1.03682
4	1.09969	1.03182
5	1.10082	1.03382
6	1.10082	1.03382
7	1.09969	1.03282
8	1.09631	1.02786
9	1.06893	1.09631

Next step was the computation of forced vibration, applying vertical and then horizontal load to each main bearing of the engine one by one. The analyses have been run in a frequency range of 0-30 Hz. The nominal rotation speed of the examined engine is 94 rpm. The engine has seven cylinders, while the propeller has five blades. For such a configuration, the basic excitation frequencies are 7.83 Hz and 10.97 Hz. Horizontal dynamic deformations under force acting on 4th main bearings are presented in Fig. 9 and 10. In the first case, the excitation frequency is equal to 10.97 Hz in the second 30 Hz.

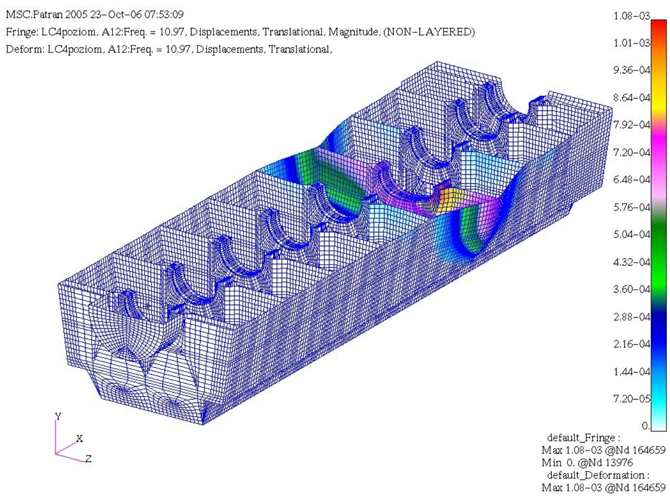

Fig. 9. Deformation of main bearing foundation under excitations with frequency 10.97 Hz

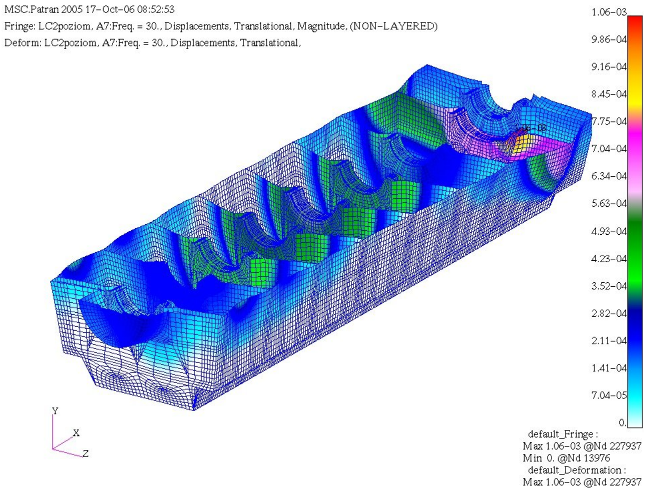

Fig. 10. Deformation of main bearing foundation under excitations with frequency 10.97 Hz

The dynamic stiffness values for the excitation frequency of 0 Hz are almost identical with the static stiffness (the differences are at the 3rd decimal place). It speaks well for the correctness of the dynamic analysis. As it was foreseen after the normal modes analysis, the dynamic stiffnesses do not differ significantly from the static ones. The stiffness decrease may be observed of 2% in vertical and down to 7% in horizontal direction.

4. Conclusions

On the basis of performed analysis it can be stated, that the static stiffness specified by the producers are properly evaluated, however they represent only the flexibility of the engine's body, not taking into account the flexibility of the ship's hull. In the author opinion, in the analysis of shaft line alignment and crankshaft springing the sum of both flexibility parameters have to be taken into account. It may be observed, that the stiffness of the engine's body is very high and the cylinder mass and gas forces acting on one cylinder have little influence on other main bearing's displacement. It means that it is a very good structure, in which it is not required to take into account the coupling between particular bearings (for main harmonic component, for higher frequencies the couplings may be observed; compare Fig. 9 and 10) – there is no need of determination of equivalent stiffness reflecting the continuity of the engine's structure.

Important is the fact, that first significant normal modes have natural frequencies above the range of excitation frequencies of the propulsion system. What's more, the significant normal modes are quite few and those of interest are of the whole engine's body. No significant normal modes have been found in the region of the engine's main bearings. It speaks well for the well-designed – rigid structure of the engine's body. In such case, the resonance type discontinuities of main bearings' flexibility are not expected and the characteristics should be close to linear.

The dynamic stiffnesses do not differ from the static ones. Such a change of dynamic stiffnesses cannot have a significant effect on the analysis of shaft line's lateral (whirling) vibration. In further commercial analyses, stiffness evaluation may be limited to static value, which can be assumed as constant in the domain of excitation frequency.

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. As further step the propulsion system analysis methodology should be elaborated, which incorporates more complex crankshaft representation including in full its 3D characteristics. The effect of crankshaft's springing on the shaft line alignment should also be examined further.

While the boundary conditions have not very important influence on natural frequencies of main bearings foundations, the global engine eigenvalues are completely different. The modelling method of boundary conditions (ship hull stiffness with engine foundation) is essential during engine body analyses. Fixed nodes in the foundation arms area gives us too stiff model but hull stiffness modelled by beams gives us too elastic model. Model with cuboid foundation is the best. During further calculations, cuboid model should be used.

References

[1] Murawski, L., *Shaft line alignment analysis taking ship construction flexibility and deformations into consideration*, Marine Structures No 1, Vol. 18, pp. 62-84, 2005.
[2] Murawski, L., Szmyt, M., *Stiffness characteristics and thermal deformations of the frame of high power marine engine*, Polish Maritime Research, No 1(51), Vol. 14, pp. 16-22, 2007.
[3] Zienkiewicz, O. C., Taylor, I. R. L., *The Finite Element Method. Fourth edition, V. 1/2*. McGraw-Hill, London 1992.
[4] *Bearings,* MAN B&W Diesel A/S, Copenhagen 2000.
[5] *Elasto-hydro-dynamic evaluation of main bearing performance,* MAN B&W Diesel A/S, Copenhagen 2002.
[6] *Guidance notes on propulsion shafting alignment*, American Bureau of Shipping, Houston 2004.
[7] *Shafting alignment for direct coupled low-speed diesel propulsion plants,* MAN B&W Diesel A/S, Copenhagen 1995.
[8] *Sulzer RTA-C, Technology Review*, Wartsila, Helsinki 2003.