ANALYSIS METHODS OF CRANKSHAFT'S STIFFNESS CHARACTERISTICS

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Abstract

Development of on-line diagnostic (monitoring) method of marine propulsion system working parameters is the authors' target. Crankshaft springing characteristics are one of the most important from the ships' main engines reliability point of view. Planned monitoring system will be able to verify crankshaft springing characteristics by continuous measurements of the crankshaft free-end's axial deformations. Development of the analysis methods of crankshaft's stiffness characteristics is the first step of planned SHM system. The main purpose of research is method developing of the springing analysis for the marine crankshafts in the high-power engines. Crankshaft modeling method, by Finite Element Method, has been discussed. Short overview of the crankshaft boundary conditions is presented. Bearings' oil film stiffness characteristics, ship hull stiffness characteristics and temperature deformation of the ship hull and main engine body are taking into account. Influence of the crankshaft's foundation stiffness on springing values is analysed. During the analyses it was proved that flexibility of engine foundation has a big influence upon the value of crankshaft springing. The authors' method of cylindrical mass and gas forces decompositions has been presented. Analysis of modeling precision of piston-crank system's forces has been performed. Results of calculations are well compatible in the terms of quality with the measurements data.

Keywords: *marine engine, power transmission system, marine propulsion system, crankshaft springing, shaft line boundary conditions*

1. Introduction

Marine structures like: ships and offshore structures surrounded by a dangerous environment are exposed to long-term cyclic loadings comes from continuously acting sea waves and short-term extreme loads such as severe storms, seaquakes or even collisions. Other source of marine structures excitations is propulsion system and other ship machinery. Any damage of a marine structure can results in endanger human life, ecologic and economical catastrophe. Therefore, all kind of improved diagnostic and monitoring systems are still desirable in shipbuilding industry and shipowners.

Structural Health Monitoring (SHM) is a multidisciplinary research topic devoted to development and implementation of methods and systems that realize inspection and damage detection by integration of sensing systems with structures [1]. It also includes a variety of techniques related to diagnostics and prognostics. SHM systems also potentially allow the reduction of maintenance costs directly connected to the effectiveness of the non-destructive techniques which are used for the monitoring of important structures, and can be used to conduct non-destructive inspections for areas which have been traditionally difficult to access. The idea of the marine SHM is to build a system that is able to evaluate a condition of a monitoring structure in different environmental and exploitation conditions. A typical structural health monitoring system is composed of a network of sensors that measure the parameters relevant to the state of the structure and its environment. One of the most important information is strain/stress distribution in the structure.

Development of on-line diagnostic (monitoring) method of propulsion system working parameters is the authors' target. Crankshaft springing characteristics are one of the most important from the point of view of main engine reliability. But, crankshaft springing cannot be measured during normal ship exploitation; usually, crankshaft springing is checked rarely (e.g. during major repairs). Planned monitoring system will be able to verify crankshaft springing characteristics by continuous measurements of the crankshaft free-end axial deformations. Development of the analysis methods of crankshaft's stiffness characteristics is the first step of planned SHM system.

The main purpose of research is developing of the method used in springing numerical analysis for the marine crankshafts in the high-power engines. In the current computational methods of the shaft line alignment, interactions between the crankshaft and the shaft line are very simplified: the crankshaft was modeling by the linear system of some cylindrical beam elements [9, 10]. Using of some high-power engines (above 30 000 kW), which are applying more and more often by now, gives a many of some failures for the main bearing of engine and for the aft bearing [7]. Actually there are a many signals about some defects of the main bearing in the literature (mainly for first three from the drive end of the engine).

2. Crankshaft modelling method

According to main engines' producers, crankshaft should be modeled as linear continuous beam, during shaft line alignment analysis. The model should be sufficient for determining bending moment and shear forces acting on crankshaft via shaft line and should be sufficient for proper main bearings' reaction determining. In author opinion, so simple model is debatable. The producers' model was verified by stiffness characteristics' comparison with detailed crankshaft Finite Element Method (FEM) model. The crankshaft model of the main engine type Sulzer 8 RTA 96 C is shown in Fig. 1. The crankshaft body is modeled by 8-nodal, 3-D solid elements, and the bearings - by beams. The FEM model contains about 650 thousands degrees of freedom. Two kinds of characteristics are analysed: stiffness of the crankshaft end and stiffness of the each crank. The characteristics were determined as a function of crankshaft rotating angle.

Fig. 1. Crankshaft FEM detailed model with one crank view

In order to complete mapping of the crank deformation under gravity and the crankshaft-piston system interactions (springing of crankshaft) there were the calculations for every 22.5° in rotation of the shaft resolved. In this case 16 variants of calculations for ever variant of analyses is necessary. Therefore there was made the reducing assumption in order to lessen number of the mathematical models that replaced rotation of gravity force direction by rotation of the same crankshaft.

Undoubted advantage of this approach is fact, that only 1 MES crank model is needed to the analyses. On the other hand there is difficulty in properly mapping of piston-crank interactions. The system forces should have some changeable value and point of application for an ever calculations variant. The used method, taking into account changeable values of the contact force for both systems is described in the next units. In the first approximation this effect is omitted.

The problem with changeability of the application point for the contact forces between the connecting-rod and the crankshaft was resolved by theirs modeling as some mass forces; mass of the central part in the crank-pin was increase by mass of piston-crank system.

3. Crankshaft's boundary characteristics

During numerical calculations, mathematical model of a marine power transmission system is usually isolated from ship's hull. Therefore, determining correctness of boundary conditions is one of the most important, difficult and debatable during marine power transmission systems' static and dynamic calculations [6]. In the subject-matter literature detail data on stiffness of the crankshaft foundation connected with the frame of marine main engine are still lacking. In author's opinion, oil film stiff and damping characteristics of propulsion system's bearings', as well as flexibility of ship hull, main engine body and bearings' frames should be taken into account. Thermal deformations of main engine, in the different loading condition should be also analysed.

Oil film stiffness is usually higher than the ship hull stiffness [8]. The oil film's stiffness and damping characteristics calculation are nevertheless important [3]. The characteristics' longitudinal distribution (especially for a stern tube bearing) determines the supporting points of the shaft line. What is more, only the oil film has got strong non-linear characteristics in the power transmission system [2]. Beside added water property, only the oil film has got significant damping characteristics. Not taking them into consideration may cause serious errors.

The marine journal bearings of the power transmission system are relatively wide with a slow rotation. During a shaft line alignment analysis, proper distribution of the oil film's pressure should be checked. Especially during a slow journal rotation, the shaft's support might be pointwise and the local oil pressure excessively high [4]. The relative deformation of the bearing shaft and tube should be taken into account especially for the stern tube bearing (not the central loading by propeller's forces).

Specialised software, based on the Finite Difference Method, was made by one of the authors to determine non-linear lube oil stiffness and damping characteristics of journal bearings. Reynolds and Stefan's principle was applied in the algorithm [3]. Analysis was performed taking the relative deformation of the bearing journal and the tube into consideration. The ship hull and the shaft line's bending line have to be known as well as the static and dynamic bearings reactions' distribution. Therefore, the shaft line with crankshaft alignment and lateral vibration calculations should be performed as an iterative process. The algorithm determining the bearings' characteristics is based on the finite bearing's length theory.

The stern tube bearing lube oil film from a container ship was analysed, as an example. Apart from propulsion's revolutions, shaft line with crankshaft alignment has a strong influence on a propulsion system bearing's pressure distribution. Especially, stern tube bearing should be always modeled as a continuous, elastic support. The lube oil film's vertical stiffness distribution along shaft line alignment (a bearing's relative vertical shift) dependence is shown on Fig. 2.

A ship hull beam is often more elastic than a shaft line [4]. Therefore the proper determining of the stiffness characteristics of the bearings mounting places is very important. Stiffness of the ship hull (analysed together with the main engine body) is the main component of the propulsion system's boundary conditions [4]. An example of a container ship's (4400 TEU) dynamic, vertical stiffness characteristics of the stern tube bearing foundation (and response of the other bearings) are shown in Fig. 3.

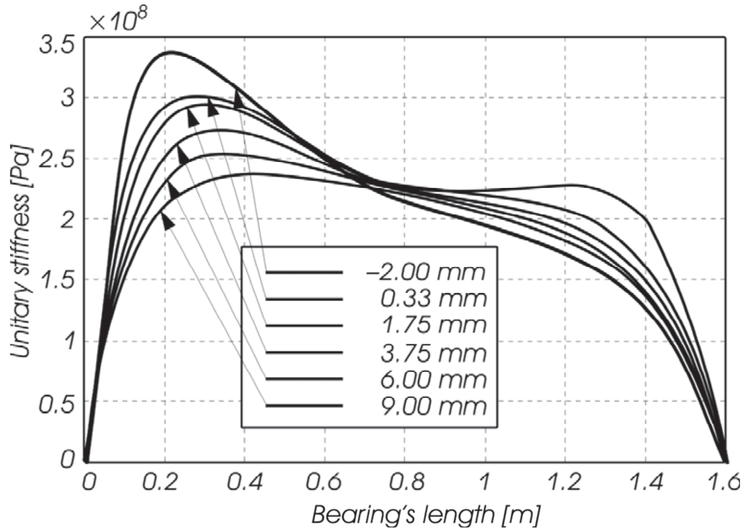

Fig. 2. The stern tube bearing's oil film stiffness distribution as an alignment function

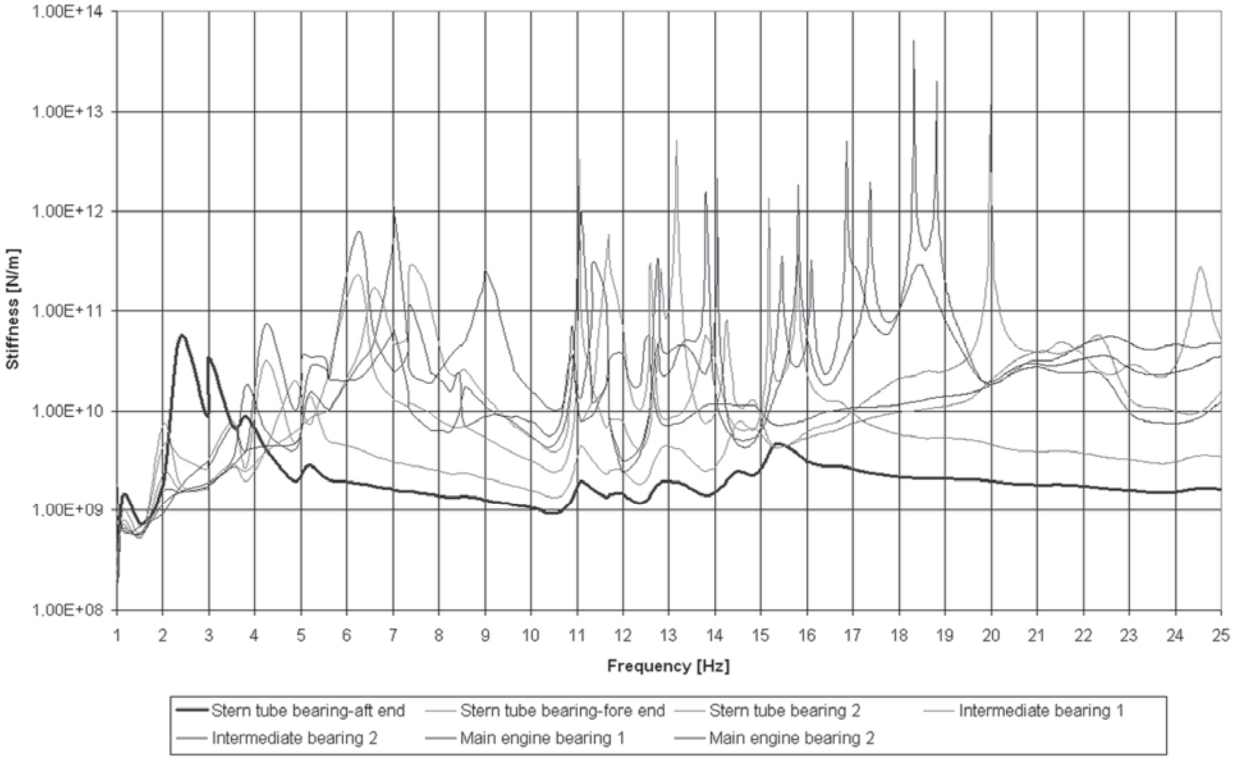

Fig. 3. The stern tube bearing's hull stiffness distribution

The aim of the last example of boundary condition is thermal displacements analysis of the crankshaft axis in the propulsion system's multiple working conditions. Up to now the crankshaft was modeled 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. This analysis goal has been better representation of the boundary conditions of the marine power transmission system. A thermal deformation of the main engine's body (MAN B&W 7 K 98 MC) has been analysed by one of the authors [5]. From a point of view of the main engine (crankshaft) – shaft line cooperation, the most important are the displacements of the main bearings of the engine. Ship hull thermal deformation should be also taken into account. The resultant thermal deformation of the main engine body and ship hull is presented in Fig. 4.

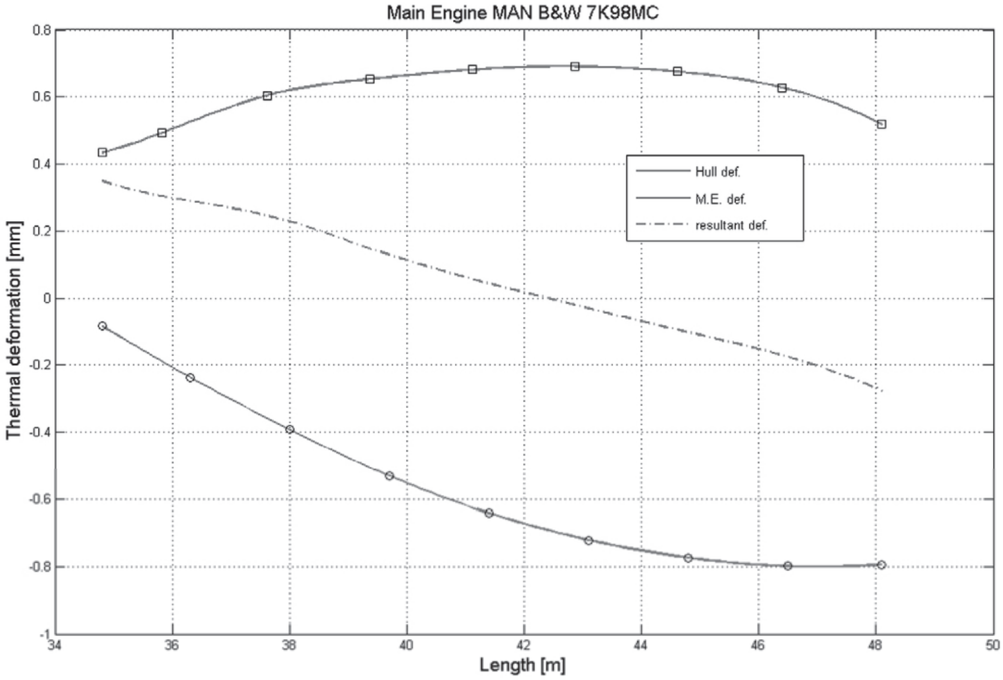

Fig. 4. The resultant thermal deformation of the main engine body and ship hull

4. Foundation stiffness analysis

In this part of research, the analysis influence of foundation crankshaft rigidity upon its springing values is presented [3, 4]. Two groups of calculations were made. In the first case only rigidity of the main engine body was taken into account; it equals $k_{SG} = 7.1\times10^{9}$ N/m. That corresponds to the engine tests which were conducted by the producer. The second group of the analyses introduces into account rigidity of the ship hull (especially doubled bottom of a ship’s hull), incorporating an equivalent of the foundation crankshaft rigidity, $k_z = 1.4\times10^{9}$ N/m, what corresponds to investigations of the engine conducted after installation of the power transmission in the ship (after its launching). One of the possibly deformations for the analyzed crankshaft, in the case of location of the first cylinder in GMP position, is presented in Fig. 5. Comparison of deformations for the rigid foundation of the crankshaft in the engine body and for some elastic shaft foundation in ship hull, have been analysed.

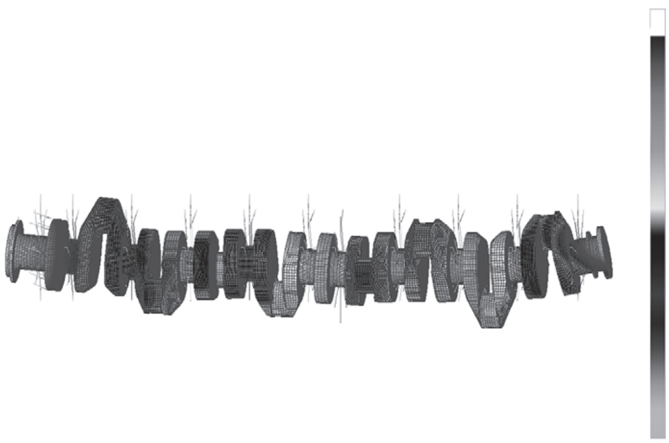

Fig. 5. Crankshaft deformation for elastic foundation

The measurement of springing consists in a measure of the differences in the distances between the crank webs, along shaft axis, and the ones which are made during its rotation. The change of springing value, arising upon the influence of rigid foundation change for crankshaft is shown in Fig. 6. Admissible values of springing are depended on the angular position of the crank, thus they are given for vertical and horizontal deformation separately. The difference of springing value (Fig. 6) was related to the admissible values.

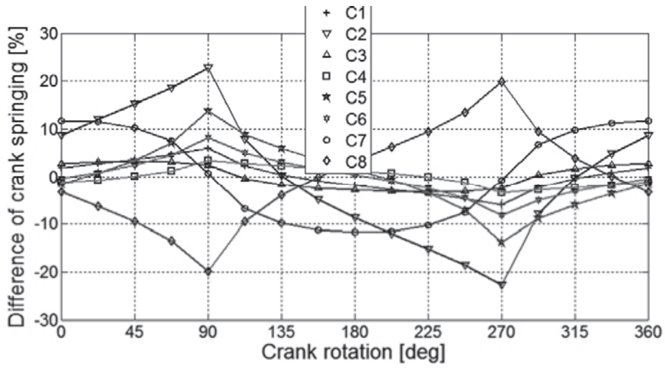

Fig. 6. Influence of foundation elasticity on crank springing

According to the analysis the crankshafts springing value differs considerably for every cylinder. During the analyses it was proved that the flexibility of the engine foundation exerts a large influence upon the crankshaft springing value. Although, the influence is different for every cylinder and for angular position of the crankshaft. Some significant differences in springing, above 10% the admissible values, one can observe for the cylinder No 2, 5, 7 and 8. Maximal differences reach the values above 20% of the admissible one. Therefore this is the reason of frequently appearing problems in practice, with reach correct springing in the ship, in spite of suitable results reached by the producer.

5. Influence of modeling precision for forces of piston-crank system

Below we present some analyses which are made on the assumption that in the rigidity of crankshaft foundation are taken into account both, main body and hull of a ship. The interaction force presented above is evaluated approximately, its value is precise for two extreme centers of the piston only: in GMP and DMP, i.e. when the axes of the piston with the rod, the connecting-rod and the inside crank, all of them lie in one line. According to change of the angle of rotation for the crankshaft, both, value of force and its direction are changeable. Classical distribution of the forces in the piston-crank system is shown in the left side of Fig. 7. That leads to equivalent weight of the crank-pin by radial and tangential forces (γ=90°) to the inside crank in the crankshaft. Radial forces have deciding influence upon the springing value in the crankshaft.

Computations within the presented method have two significant disadvantages. First, they are extraordinary labour absorbing – it would be necessary to build a new mathematical model of the crankshaft for every angle of rotation, additionally examined for every cylinder. In the considered case it means the computations for 128 variants! More, values of radial forces should be evaluated and applied for every variant separately. Second disadvantage is fact that tangential forces participate in the crankshaft springing (analyzed in total). In the case of an analysis for one crank only, tangential force not causes longitudinal deformation, measured along the main pin line. However, the force transmitting itself to the other cranks is a tangential force any more, by reason of angular dislocation for the particular inside cranks (which is connected with sequence of ignition) and causes additional springing of the next inside cranks.

In connection with inconveniences of the classical method of analyses the authors developed own method of computations. It consists in using the calculations of deformations for crankshaft being upon influence of modified gravity force with changeable direction. Next, the results of analyses are transformed with taking into account the changes of the value and direction of the real piston-crank forces. The transformation must be made for two variants of computations for every point in the crankshaft springing diagram. For the new method the new distribution of forces in the crankshaft were developed, with decomposition of the resultant force into a radial force (radial to inside crank) and force acting according to the gravity force. The new distribution of forces for two characteristic positions of the crank is presented in the right side of Fig. 7.

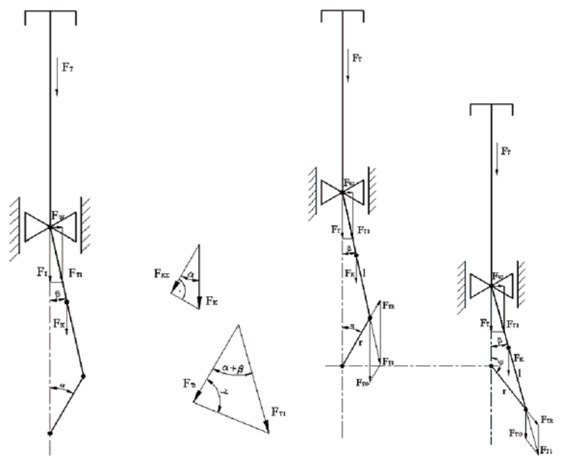

Fig. 7. Classical and new method of distribution of piston-crank system forces

On the basis of the force distributions (in the Fig. 7) one can find some relations, which allow to evaluate summary coefficients of correctness, for the forces acting along radius of the inside crank and for the forces acting in according with the gravity force. Applying the mentioned correction coefficients, it is possible to evaluate precisely some springing values for the crankshaft, by using simplified computational analyses. During analyses of the crankshaft springing it is necessary to make vector summation of the value reached from a computational variant with “pure radial forces” (piston in GMP), and founded by variants with taking into account gravity forces (for given angle of the crankshaft rotation). It must be underlined that the values of the correction coefficients are changing by rotation for every cylinder, adequately to sequence of ignition. Values of the crankshaft springing, found after taking into account the correction coefficients (precisely modeling of the value and direction of the piston-crank system forces) are shown in Fig. 8. Change of the springing value before and after precise mapping was related to admissible one.

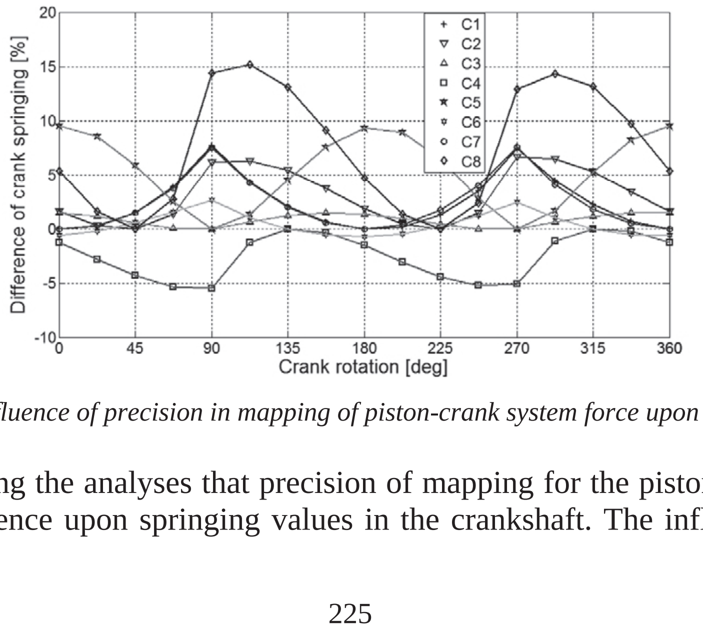

Fig. 8. Influence of precision in mapping of piston-crank system force upon springing

It was shown during the analyses that precision of mapping for the piston-crank system forces have significant influence upon springing values in the crankshaft. The influence is different for some different cylinders, and for different radial positions of crankshaft. Some considerable differences in springing, above 5% of admissible values, were observed for majority of the cylinders, except for cylinder no 3 and 6. Maximal differences reach 15% admissible values.

6. Conclusions

The results of analyses of computations for the crankshaft springing characteristics were positively verified by comparison to measurements (which were made for three real ships). Results of calculations are well compatible in the terms of quality with the measurements data. Quantitative comparison is difficult because of significant measurements data dispersion (even 35%) for the mentioned ships. According to the experiments values of the inside cranks in the crankshaft are different for different cylinders. During the analyses it was proved that flexibility of engine foundation has a big influence upon the value of crankshaft springing. Because of that there are often some problems in practice with finding of properly springing in the ship, nevertheless the tests made by the producer showing some acceptable values of springing.

The influence of piston-crank interactions with the inside cranks, together with the angle of rotation for the crankshaft is enough significant. This effect is not negligible during the precise springing analyses of the crankshaft. In the connection with some shortcomings of the classical method of computations, the authors developed the new method, which uses the calculations of the crankshaft deformations, arising upon influence of modified gravity force (the force with changeable direction of acting). During the analyses it was proved that the precision of mapping of the piston-crank system forces has the fundamental influence upon the crankshaft springing value.

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