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Advanced Methods for Static and Dynamic Shafting Calculations

Review paper

This article introduces a computer program developed by *Wärtsilä Switzerland* which provides a three-dimensional model of the shafting for the calculation of coupled vibrations, alignment and whirling in a ship propulsion plant. Based on the finite-element method the program covers both dynamic and static problems. Each node has six degrees of freedom. The following features are included: calculation in frequency range and time domain, linear and nonlinear bearing characteristic, consideration of variation of running gear inertia, and optimization of engine parameters. The mathematical model of the shaft line with all associated parameters and boundary conditions are represented by calculation results and validated by measurement. An everyday example for the calculation of coupled axial vibrations with the associated measured data is shown in this paper. Furthermore the influence of the variation of torsional inertia is demonstrated and a possibility for reduction of torsional stress in the crankshaft by injection timing optimization is explained. As an example for the static capabilities the reverse bearing offset calculation based on measured web deflections, bearing loads and bending moments is demonstrated.

Keywords: *bearings, coupled vibrations, shafting, ship propulsion plants, torsional stress*

Napredne metode za statički i dinamički proračun osovinskog voda

Pregledni rad

U članku je opisan program za elektroničko računalo razvijen u tvrtki *Wärtsilä Switzerland*, koji koristi trodimenzionalni model osovinskog voda za proračun spregnutih vibracija, centracije i vitlanja u brodskom pogonskom postrojenju. Program se temelji na metodi konačnih elemenata te pokriva statičke i dinamičke probleme. Svaki čvor konačnih elemenata ima šest stupnjeva slobode. Uključene su značajke: proračun u frekvencijskom i vremenskom području, linearne i nelinearne značajke ležaja, uzimanje u obzir promjene inercije rotirajućih zupčanika kao i optimizacija parametara stroja. Matematički model osovinskog voda sa svim pridruženim parametrima i rubnim uvjetima predstavljen je pomoću rezultata proračuna i provjeren mjerenjem. Prikazan je primjer proračuna spregnutih uzdužnih vibracija iz prakse s odgovarajućim podacima s mjerenja. Nadalje, pokazan je utjecaj promjene inercije na uvijanje, te je objašnjena mogućnost smanjenja naprezanja zbog uvijanja u koljeničastoj osovini pomoću optimizacije trenutka ubrizgavanja. Kao primjer mogućnosti statičke analize pokazan je proračun repozicioniranja ležaja na temelju izmjerenih pomaka, opterećenja u ležaju i momenata savijanje.

Ključne riječi: *brodsko pogonsko postrojenje, ležajevi, osovinski vod, spregnute vibracije, torzijsko naprezanje*

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1 Introduction

The problems in analysing vibrations for installations with two-stroke diesel engines have been accentuated since new engine generations have been introduced on the market with high efficiency for minimum fuel consumption, involving lower running speeds, larger stroke/bore ratios and higher combustion pressure. For these engine types, the vibration aspect has become more important.

A typical arrangement of a two-stroke diesel engine installation is shown in Figure 1. The engine is directly coupled to the propeller by one or more intermediate shafts and the propeller shaft. The integrated thrust bearing at the aft side of the engine transmits the axial forces from the propeller to the ship structure. The pads of the thrust bearing are mainly arranged below the ro-

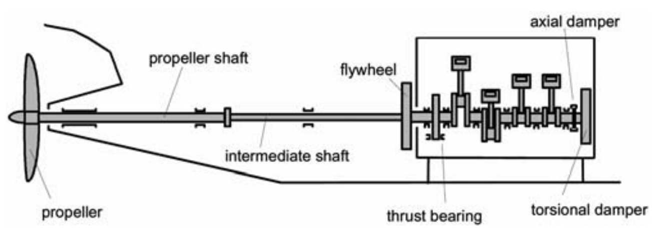


Figure 1 **Arrangement of shafting system**
Slika 1 **Raspored sustava osovinskog voda**

tational axis, so there is also a bending moment to accommodate. At the free end of the crankshaft an axial damper is integrated. The axial damper consists of two oil chambers which are connected to each other. Oil flows from one chamber to the other with the axial displacement of the crankshaft so leading to the needed damping effect.

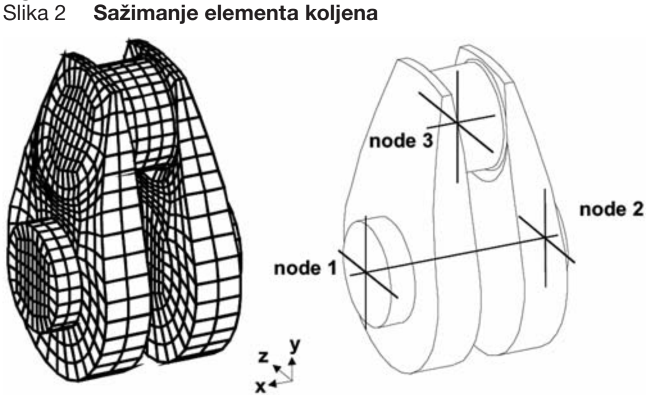
When displacement or reaction forces of the shafting system are of interest, then the shafting system including the bearing stiffness and offsets, but without the ship structure, can be considered.

In the past torsional and axial vibrations were calculated separately to determine the optimum shaft line arrangement (flywheel, intermediate shaft diameter and ultimate tensile strength, free-end disc or torsional damper, etc.). The same one-dimensional mathematical model was used for both kinds of calculations. Coupling effects between the axial and torsional vibrations could only be approximated by merging the results of both. Although the calculated and measured torsional amplitudes are usually in a very good accordance, the comparison of the calculated and measured axial amplitudes looks different. Radial excitation forces do not influence the torsional amplitudes, but tangential excitation forces influence the axial displacements [2], [3]. For the calculation of axial vibrations the mathematical model for the shaft line has to be modified. Instead of separating the shaft line into discrete 1-D elements, a 3-D model has to be used. The advantage of this model lies in much better representation of the real system. The following reasons exist for the use of a 3-D model for static and dynamic shaft analysis:

- Among the torsional–axial coupling coefficients for the crankshaft there are other coupling coefficients which may influence the calculation results, for example torsional-bending coefficients for the crankshaft or a torsional–axial coupling coefficient for the propeller.
- Radial and tangential forces act on the crank pin. For a realistic calculation of internal load in the crankshaft this is a requirement. For the classical 1-D torsional vibration calculation the torque between 2 cylinders is constant. In reality, the torque in the journal pin is different from the torque in the crank pin due to tangential shear forces in the crank pin.
- Calculations of main bearing forces (static & dynamic) are only reasonable with a 3-D model. This is particularly true when nonlinear bearing characteristics are used.

The first step for creating a 3-D model of the crankshaft is done with a finite element program. FE models for cranks of all

Figure 2 **Condensation of crank element**
Slika 2 **Sažimanje elementa koljena**



engines have to be created. For harmonic analysis the number of nodes for one crank has a limited influence on the lowest modes. Owing to the many single calculation steps for dynamic calculations, and only the results on some few points of the crankshaft being of interest, the finite element structure is reduced to a matrix structure with at least two main bearing nodes and one crankpin node (Figure 2). Symmetric mass and stiffness matrices are the result of the node reduction for the crank.

Mass and stiffness matrices of other cranks are calculated from the first under consideration of the crank sequence. Based on this a crankshaft element is created (Figure 3). The following properties are included: rotating and oscillating mass for each cylinder, main bearing stiffness and damping, crankpin damping, and stiffness and damping of the axial damper and thrust bearing.

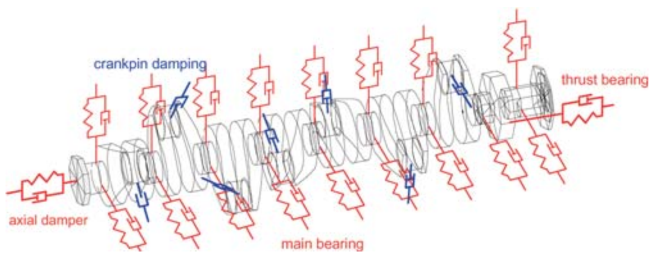


Figure 3 **3-D model of crankshaft**
Slika 3 **Trodimenzionalni model koljeničaste osovine**

The accuracy of a vibration calculation depends on the mathematical model and the associated parameters. The stiffness and damping coefficients of the bearing elements are crucial for the calculation of natural frequencies and forced vibrations.

2 Calculation package EnDyn

Some years ago, the coupled axial vibration calculations of installations with two-stroke Wärtsilä RTA diesel engines were carried out with a commercial FE program [3]. However, this way of working is not suitable for daily calculations with a minimum of effort and time. For this reason it was decided to start the development of a user friendly calculation program, known as EnDyn. It is based on the finite element method [1]. All needed engine data are included in crankshaft elements; only the characteristics of the flywheel, front-end disc, torsional damper (if any), the intermediate and propeller shaft, and the propeller have to be defined by the user. The first stage of development was achieved when the standard output of a coupled axial vibration calculation was presented to and accepted by the few Classification Societies which requested such calculations.

If a realistic mathematical model of the crankshaft and shaft line with six degrees of freedom (DOF) at each node exists, and then it is possible to carry out other kinds of calculations.

This led to the next extensions of this program: whirling and alignment calculations (based on the same input file used for the axial vibration calculation). The latest extension is the integration of an optimization algorithm, which can be used for many different kinds of calculations.

The mathematical model for the axial bearing elements in EnDyn is linear. The coefficients are determined by comparing measured and calculated axial vibrations. It is assumed that radial bearings are isotropic. The calculation of forced vibrations in the frequency domain has to be applied with linear spring damper elements without bearing clearance. Mean stiffness and damping coefficients are needed, the parameters in vertical and horizontal direction are the same. The structure stiffness is calculated by FEM, damping values are determined from measurement results, since reliable methods for calculating damping parameters with sufficient accuracy are not yet available.

More sophisticated calculations such as the calculation of orbits should be carried out including clearance and oil film stiffness and damping. The nonlinear bearing characteristics for radial bearings in EnDyn are checked by comparing the result with the output of advanced elasto hydrodynamic bearing calculations including consideration of the FE structure of the engine.

For alignment calculations, without consideration of the oil film, but including the bearing clearance, the 3-D model of the crankshaft enables to calculate realistic jack load diagrams and web deflections.

3 Example for coupled axial vibrations

This example is carried out for a 12-cylinder two-stroke Wärtsilä diesel engine with torsional damper driving a six-bladed fixed-pitch propeller. The results are checked by comparison of measured and calculated axial displacements at the free end of the crankshaft, and at the forward flange of the propeller shaft.

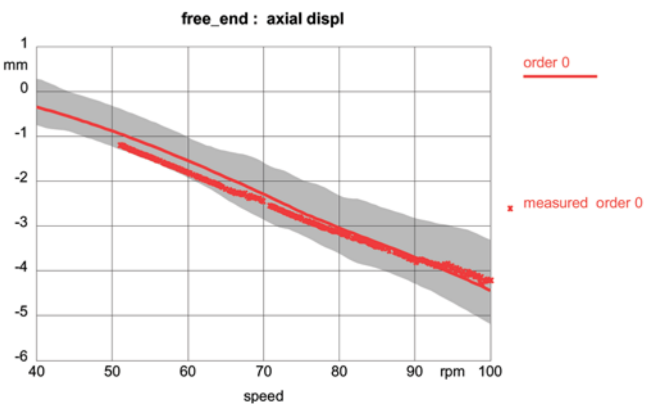


Figure 4 **Mean values and dynamic range of axial displacement at free end**
Slika 4 **Srednje vrijednosti i dinamički doseg uzdužnih pomaka na slobodnom kraju**

At the lowest speed the free end of the crankshaft moves in the forward direction due to the propeller thrust and the radial excitation forces but, by increasing the speed, the rotating mass forces become more dominant and the free end moves in the aft direction (Figure 4). This phenomenon can be observed very clearly in the two-stroke diesel engines of stationary power plants. These engines are operated at nominal speed and at various loads. In the idling condition, the length of the crankshaft is much longer compared to the full load condition. The grey filled area indicates the dynamic range. The ratio of the dynamic amplitudes and the mean value is quite small.

The analysis of the dynamic range is shown in Figure 5, namely the synthesis as well as three different orders with the largest amplitudes in the speed range together with the corresponding measured values. Owing to the torsional vibration damper and the axial damper fitted on this engine, the dynamic axial displacement is well below the axial limit for this crankshaft. The sixth order of the axial displacement is slightly influenced by propeller thrust variation.

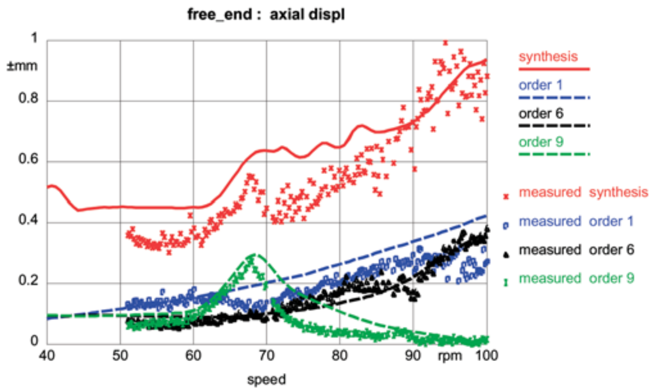


Figure 5 **Analysis of axial displacement at free end**
Slika 5 **Analiza uzdužnih pomaka na slobodnom kraju**

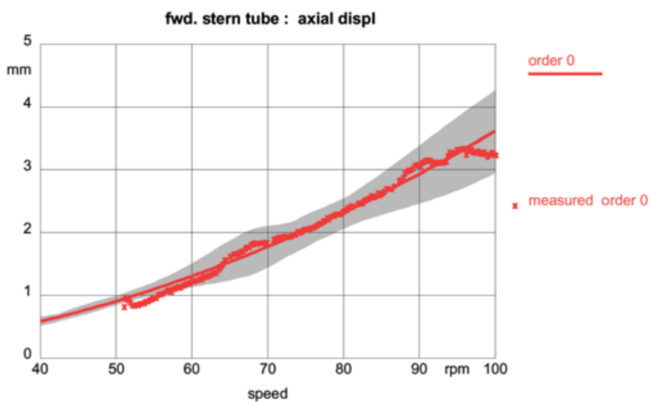
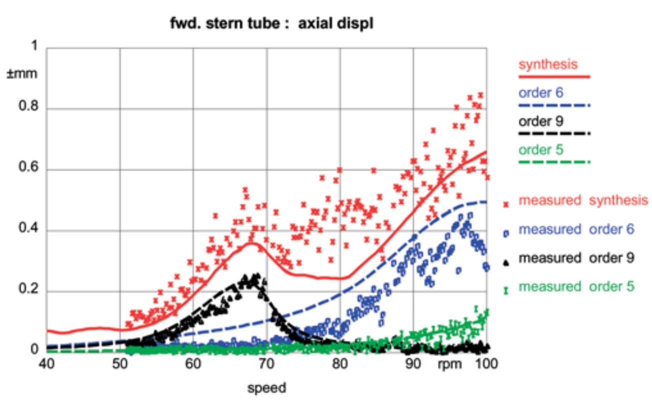


Figure 6 **Mean values and dynamic range of axial displacement at forward flange of propeller shaft**
Slika 6 **Srednje vrijednosti i dinamički doseg uzdužnih pomaka na prednjoj prirubnici osovine brodskog vijka**

Figure 7 **Analysis of axial displacement at forward flange of propeller shaft**
Slika 7 **Analiza uzdužnih pomaka prednje prirubnice osovine brodskog vijka**



Owing to the thrust, the propeller shaft moves in the forward direction (Figure 6). The mean value depends on the static thrust; the dynamic range is influenced by the length of the shaft line and the firing order of the engine. The analysis of the dynamic range is shown in Figure 7. The calculated amplitudes for order 6 are dominant owing to the propeller blade number. This may be influenced by the phase angle of the propeller.

4 Example for torsional stress reduction by optimization of injection timing

The following example is carried out for an eight-cylinder two-stroke Wärtsilä diesel engine without torsional damper driving a six-bladed fixed-pitch propeller.

4.1 Secondary order resonance

The equations of motion for the calculation of forced torsional vibrations are described by the derivation of the angular momentum and the difference of the external torque and the torque owing to damping and stiffness of the shafting system. The derivation of angular momentum contains expressions for the torsional inertia as well as the derivation of the torsional inertia. For torsional vibration calculations of rotating shaft systems including running gear, it is common to reduce the oscillating mass to a torsional inertia. For the classical approach, a set of linear differential equations is generated, taking into consideration the mean value of the inertia (Figure 8).

Angular momentum:

$$D = \Theta\dot{\varphi}$$

-> $$\frac{dD}{dt} = \Theta\ddot{\varphi} + \dot{\Theta}\dot{\varphi}$$

Principle of conservation of angular momentum:

$$\frac{dD}{dt} = T - B\dot{\varphi} - C\varphi$$

Differential equations: $\Theta = \Theta(\varphi)$

$$\boxed{\Theta\ddot{\varphi} + \dot{\Theta}\dot{\varphi} + B\dot{\varphi} + C\varphi = T}$$

Classic calculation: $\Theta = cst$

-> $$\Theta\ddot{\varphi} + B\dot{\varphi} + C\varphi = T$$

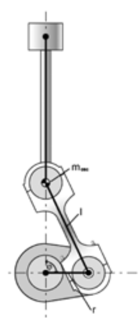
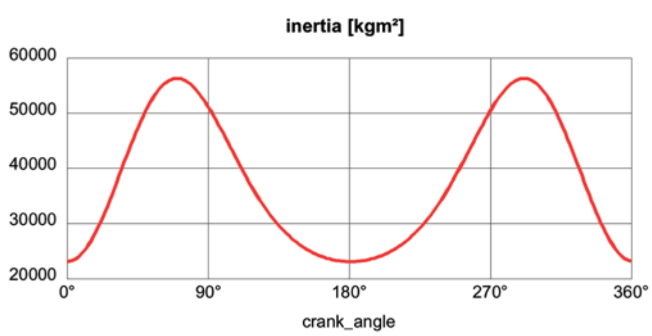


Figure 8 **Differential equations for forced torsional vibrations**
Slika 8 **Diferencijalne jednadžbe prisilnih torzijskih vibracija**

Figure 9 **Torsional inertia for a cylinder**
Slika 9 **Inercija pri uvijanju za cilindar**



The torsional inertia, calculated by equating the kinetic energy for the oscillating mass system and the kinetic energy for the model with the reduced torsional inertia, is a function of the crank angle (Figure 9). In addition to the mean value, a dominant amplitude of order 2 is noticeable for the inertia. This may influence torsional vibrations in installations without torsional vibration damper as is shown in the following example. This phenomenon, known as secondary resonance, was first observed and described by P. Draminsky [5].

Taking into consideration the periodic function of the inertia as well as the derivation of inertia the parameters for the mass and damping matrix are periodic. This concerns classical torsional vibration calculations, but is also applicable for calculations with a three-dimensional model of the shafting system. An iterative method of solving the nonlinear differential equations is used for calculations in the frequency domain; otherwise the calculations have to be solved in the time domain.

4.2 Calculation results with and without variation of torsional inertia

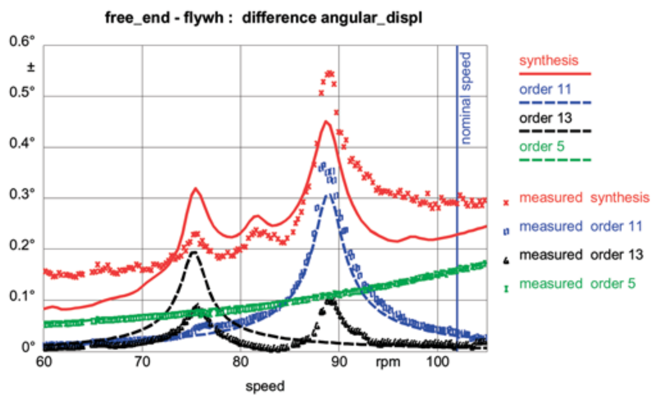
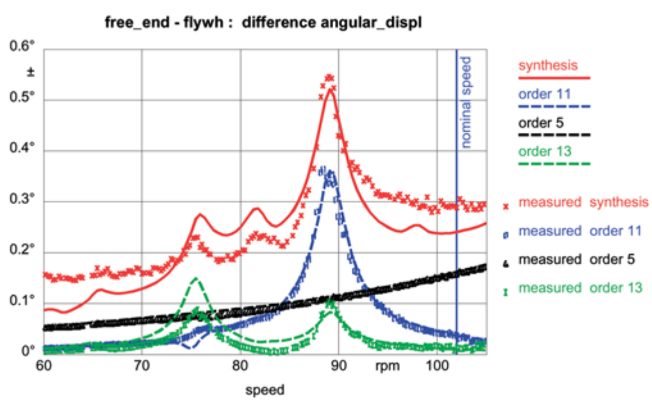


Figure 10 **Twist in crankshaft, calculated with mean torsional inertia**
Slika 10 **Smicanje u koljeničastoj osovini, proračunato s prosječnom inercijom pri uvijanju**

The comparison between the original calculated and measured amplitudes for order 13 for the twist in the crankshaft (Figure

Figure 11 **Twist in crankshaft, calculated with variation of torsional inertia**
Slika 11 **Smicanje u koljeničastoj osovini, proračunato s promjenom inercije pri uvijanju**



10) demonstrates the secondary resonance effect: the calculated amplitude at the torsional critical speed (75 rpm) is much higher than the measured one, and at 89 rpm (the critical speed for order 11) there is a second peak for order 13. As a result, the calculated synthesis is different from the measured one. The original calculation was carried out with the mean torsional inertia. After taking into account the variation of torsional inertia, there is much better agreement between the measured and calculated data (Figure 11).

Also the torsional stress was calculated again, and the result shows that the maximum stress is above the limit (Figure 12).

4.3 Reduction of the torsional stress level

As a countermeasure for reducing the stress level it was decided to install a torsional vibration damper. Another feasible solution is modifying the engine tuning to reduce the harmonic excitation order 11. A different approach, applicable for diesel engines with a common rail injection system such as Wärtsilä RT-flex engines, is an individual variation of the cylinder pressure. Owing to the fact that order 11 is not the main order, the harmonic excitation forces of order 11 have different phase angles for each cylinder.

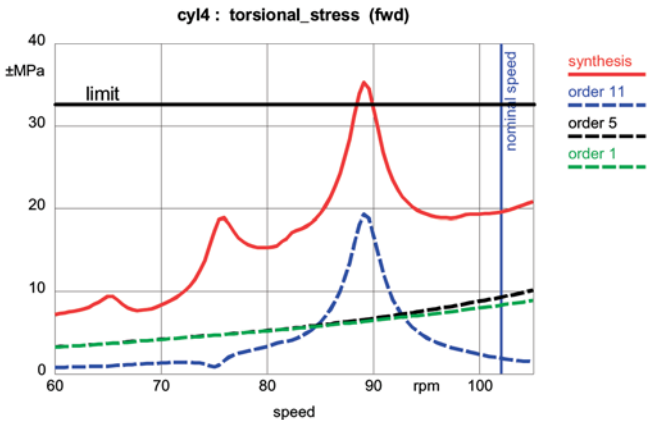


Figure 12 **Torsional stress calculated with variation of torsional inertia**
Slika 12 **Naprezanje na uvijanje, proračunato s promjenom inercije pri uvijanju**

An elegant option of varying the cylinder pressure histories among cylinders is provided by using different injection patterns [6]. Pre- or sequential injection can be used either on all cylinders if sufficient for reducing the harmonic excitation order 11 at the critical speed or on selected cylinders only, applying an optimization routine for determining the most appropriate selection of the respective injection parameters: timing and duration of the pre-injection pulse as well as the duration of the pause between pre- and main injection pulses in the case of pre-injection or the delay between the actuation of the individual injectors and the actuation sequence in the case of sequential injection.

For this example an individual variation of injection timing was considered. Based on gas excitation sets for different injection timings and with the optimization mode included in EnDyn, the solution for the lowest torsional stress was calculated. It was assumed that the optimization is only valid in a small speed range around the critical speed, and the range for injection timing variation is as small as needed for a moderate reduction of torsional stress. Figure 13 shows the result for the injection timing. Cylinders 3 and 5 have an earlier beginning of injection timing; all other cylinders have an injection timing delay. The torsional stress calculated after optimization is shown in Figure 14. The reduction applies mainly to a reduction of the torsional stress for order 11.

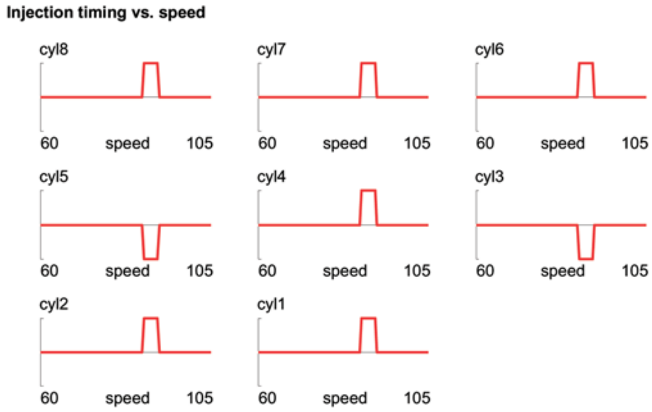


Figure 13 **Injection timing for reducing maximum torsional stress**
Slika 13 **Podešavanje trenutka ubrizgavanja radi smanjenja najvećih naprezanja uslijed uvijanja**

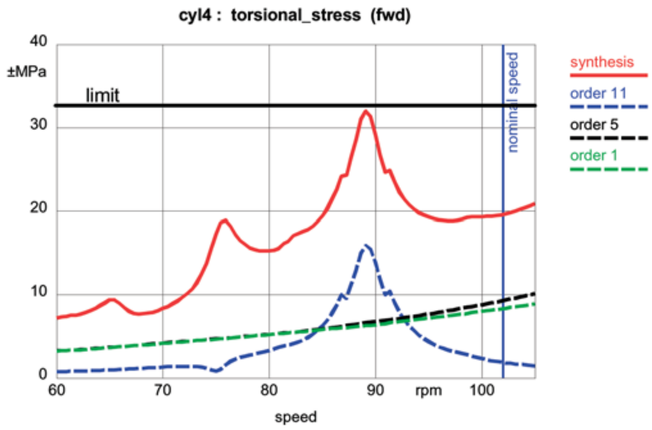


Figure 14 **Torsional stress calculated after optimization of injection timing**
Slika 14 **Naprezanje na uvijanje, proračunato nakon optimizacije trenutka ubrizgavanja**

5 Example for alignment: reverse bearing offset calculation

Bearing forces and bending moments in a shafting system depend on speed and power, on modal parameters like natural frequencies, mode shapes and modal damping, on weight and misalignment of the shafting system. The misalignment is influenced by the given bearing offsets and the static deformation of the ship hull depending on the ballast condition of the ship and the sea waves.

Practical methods to determine the misalignment in a given installation are web deflection measurement and jack load test. Sometimes strain gauge measurements are also carried out. Based on these data the analysis of the bending line of the shafting system is possible. The following example is carried out for a bulk carrier with six-cylinder diesel engine.

5.1 Crank web deflection

The web deflection for a crank is the difference of the distance between webs for the loaded and unloaded crank. Web deflections, depending on the crank sequence and the bending line of the crankshaft, have a different characteristic for each cylinder (Figure 15).

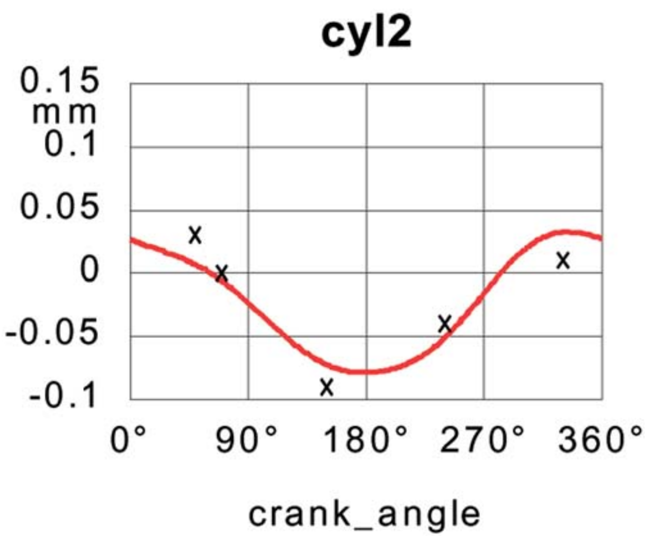


Figure 15 **Calculated and measured crank web deflection**
Slika 15 **Proračunati i izmjereni pomaci koljena**

Web deflections are measured at the top dead centre and around the bottom dead centre in the vertical plane, and at fuel pump side and exhaust side in the horizontal plane. Because of the connecting rod it is not possible to measure the web deflection continuously. Near the bottom dead centre the deflection has to be measured at either side of this position (exhaust side and fuel pump side of the bottom dead centre).

5.2 Jack load

Static bearing loads are checked by the jack load measurement, which allows the determination of the bearing load in the vertical plane. The principle of the jack load is simple: when the shaft is jacked up from the shell, the bending stiffness of the shafting system changes. Jack load measurements may be carried out for the main bearings of the engine, for intermediate bearings and the forward stern tube bearing. As a result of the measurement, the displacement values are plotted against the jack load.

When jack load measurements are carried out at the main bearings of the engine, the jack is positioned under the web. The crankshaft has to be turned to a position where the crank pin and journal pin are in the horizontal plane.

The main bearing load depends on the crankshaft angle (Figure 16). With a simplified model of the crankshaft, which

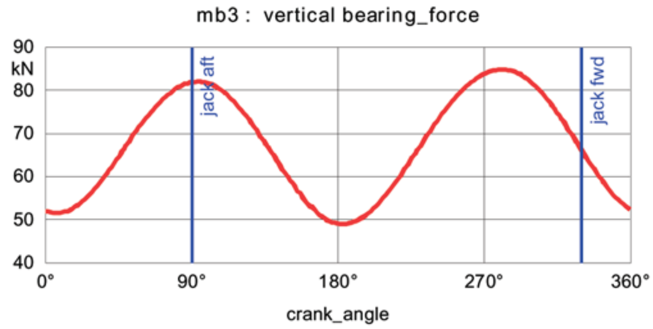


Figure 16 **Main bearing load for one revolution**
Slika 16 **Glavno opterećenje ležaja tijekom jednog okretaja**

is usually used for alignment calculations, the vertical bending stiffness is fixed. In fact the bending stiffness depends on the crankshaft angle, and therefore the main bearing load also depends on the crankshaft angle. This leads to the following conclusions:

- Even for static calculations the main bearing loads have to be carried out for one revolution to know the minimum and maximum values.
- For the analysis of jack load measurements it is crucial to know the crankshaft angle and the location where the jack is placed.
- The sum of all measured jack loads is not the same as the weight of the shafting system, when the crankshaft is part of the system. jack load

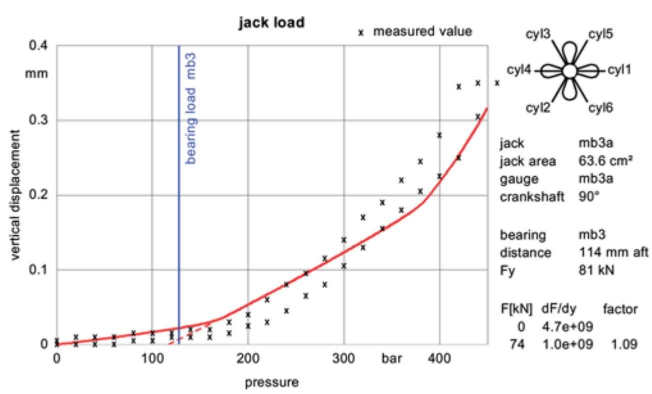


Figure 17 **Calculated and measured jack loads**
Slika 17 **Proračunata i izmjerena opterećenja**

Figure 17 shows the result of a jack load calculation, carried out including bearing clearance. In contrast to the measurement, the calculated jack load is without any hysteresis. There are two kinks in the curve of the jack load. The first kink leads to the unknown jack load. This is not in general true. When the bearing load is very high, and the neighbour bearing has only a small load, then the second kink can lead to the real jack load. Therefore, it is beneficial to carry out a reverse offset calculation to avoid a wrong analysis of the measured jack load data.

Part of the result of the jack load calculation is the jack factor. The analysed jack load multiplied by this correction factor gives the estimated bearing load.

5.3 Result of reverse offset calculation

Vertical web deflection [1/100 mm]

node	measured	calculated	difference
cyl6	-2.5	-3.2	0.7
cyl5	-8.0	-7.4	-0.6
cyl4	3.5	3.5	0.0
cyl3	-1.0	-1.5	0.5
cyl2	-5.5	-5.1	-0.4
cyl1	3.0	2.8	0.2

error = 0.5

Horizontal web deflection [1/100 mm]

node	measured	calculated	difference
cyl6	1.0	1.2	-0.2
cyl5	3.0	3.1	-0.1
cyl4	3.0	2.5	0.5
cyl3	0.0	0.8	-0.8
cyl2	-10.0	-10.6	0.6
cyl1	-20.0	-19.8	-0.2

error = 0.5

Bearing load fy [kN]

node	measured	calculated	difference
ibear	52.4	55.2	-2.8
fwd_bush	36.2	34.7	1.5

error = 2.2

Bending moment mz [kNm]

node	measured	calculated	difference
sg1	-50.0	-46.6	-3.4
sg2	-31.0	-36.4	5.4
sg3	12.0	14.5	-2.5

error = 4.0

Figure 18 **Comparison between measurement and calculation**
Slika 18 **Usporedba između mjerenja i proračuna**

Based on a best-fit procedure EnDyn enables a reverse offset calculation to be carried out for all bearings. Figure 18 shows the result of the reverse bearing offset calculation as a comparison of measured and calculated value for each location, and the difference. From the differences a mean error is calculated. The number of the measured data is bigger than the number of unknown bearing offsets, therefore the error will be always greater then zero. More important, this value is a good indication for the accordance between measurement and calculation. As a result of the reverse bearing offset calculation the displacement and bending stresses are shown over the shaft line in Figure 19.

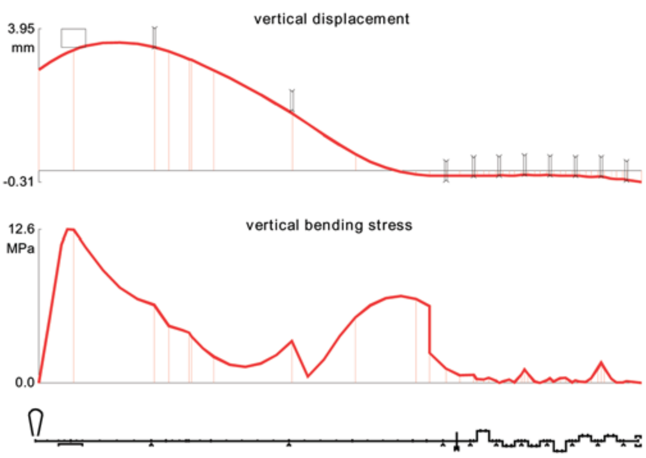


Figure 19 **Vertical displacement and bending stress**
Slika 19 **Vertikalni pomaci i naprezanja uslijed savijanja**

The main bearing forces are shown in Figure 20. The grey filled area indicates the bearing load over one revolution, the horizontal lines correspond to the load when cylinder 1 is at the top dead centre. The bearing loads are all within the limits for service condition.

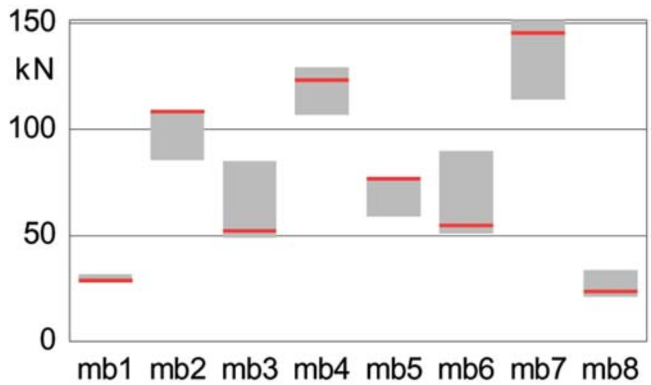


Figure 20 **Main bearing load distribution**
Slika 20 **Raspodjela opterećenja glavnih ležaja**

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