Forces exciting vibrations of ship's hull and superstructure

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ABSTRACT

The paper presents identification of main excitation forces of vibrations of ship hull and its superstructure, as well as investigation of influence of the analyzed excitation on ship vibration level. The following excitation forces were analyzed: dynamic pressure on the transom deck, propeller-generated hydrodynamic forces, as well as internal and external engine-generated forces. Rigidity and attenuation characteristics of lubricating oil film were taken into account in the analysis of the excitations transmitted by the propulsion system bearings (stern tube bearing, intermediate bearing, engine's main and thrust bearings). The dynamic characteristics of ship's power transmission system were also taken into account. Results of the analyzes were verified by comparing results of the computations and measurements.

Keywords : hull vibration, superstructure vibration, main engine vibrations, excitation forces, pressure pulses, propulsion system's reactions, crankshaft-propeller phasing

INTRODUCTION

The following main forces exciting vibrations of ship hull and superstructure (Fig.1) were considered [1,12] :

1. propeller-induced pressure pulses upon the ship transom deck

2. dynamic reactions of the thrust bearing to non-coupled longitudinal vibration of the power transmission system (generated by axial hydrodynamic force of the propeller)

3. dynamic reactions of the thrust bearing to coupled longitudinal vibration of the power transmission system (generated by kinematic couplings of the crankshaft and hydrodynamic couplings of the propeller)

4. dynamic reactions of the radial bearings (stern tube bearing, intermediate bearings and main bearings of the engine), generated by hydrodynamic radial forces and moments induced on the propeller

5. non-balanced moments of the main engine, generated by gas and inertia forces of piston-crank system.

The analysis was performed for the cargo ships (mainly for medium size containers) with typical propulsion system – a constant pitch propeller directly driven by a low speed main engine. In the paper an example analysis was made for a 2700 TEU container carrier. The ship particulars were as follows: ship length – 207 m, maximum deadweight – 35600 t, speed – – 22.7 knots. The ship was powered by MAN B&W 8S70MC-C engine of 24840 kW rated output at 91 rpm. The five-blade propeller was of 7.6 m diameter and 42400 kg in weight. All conclusions are valid for the ship loaded with containers from 1200 TEU up to 4500 TEU in number. For the analysis of the ship's hull and superstructure vibrations Patran-Nastran software was used, whereas the analysis of excitation forces (dynamic calculations of the power transmission system) was made mainly with the use of the author's DSLW FEM – based software dealing with coupled torsional-bending-longitudinal vibrations of power transmission system [7] and DRG one concerning shaft line whirling vibrations [8].
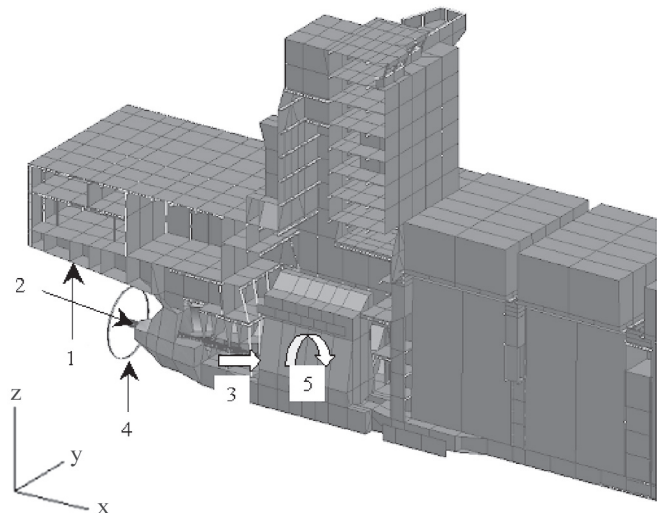
Fig. 1. Excitation forces of ship's hull and superstructure vibrations

Reactions of stern bearing, shaft line bearing and main bearings of the engine are generated by bending vibration of the shaftline. The reaction level and distribution depend on shaft line alignment [5,11] and on stiffness-damping characteristics of lubricating oil film in the bearings [6,8]. The stiffness-damping characteristics of the oil film were found with the use of the author's BRG software based on Finite Difference Method [6]. Dynamic rigidities of ship hull structure within areas of bearing foundations were determined with the use of Nastran software.

Analysis of bending vibration of the propeller shaftline showed a low level of bending vibration, nevertheless the level of dynamic reactions of the bearings (particularly the stern tube bearing) was significant. It may be a source of excessive vibrations of ship hull and superstructure.

Reactions of the thrust bearing are generated by longitudinal vibration of the power transmission system. [3,7]. Two basic sources of longitudinal vibration excitation forces may be distinguished : excitations generated by the main engine [4,7] (coupled torsional-bending-longitudinal vibrations) and excitations generated by the propeller (coupled and non-coupled hydrodynamic forces). During the analysis total reactions of the thrust bearing, generated by all kinds of kinematic couplings were found. These reactions result from longitudinal vibrations of the power transmission system, coupled with torsional-bending vibrations [3,7]. Non-linear characteristics of thrust bearing's oil film were taken into account during the calculation.

Reaction phases generated by non-coupled vibrations were determined with respect to propeller blade (phase „zero" was assumed for the upper position of the blade). Reaction phases generated by coupled vibrations were determined with respect to the first crank of the engine (phase „zero" was assumed for the top dead centre). The amount of total reactions (generated by coupled and non-coupled vibrations) depends on mutual phasing of propeller blades and crankshaft. With a suitable arrangement of angular position of the propeller (in relation to the crankshaft) the resultant vector of the reactions can be reduced to algebraic difference of modules of the summed reactions. The optimum setting angle of the propeller with respect to main engine cranks depends on rotational speed of the propulsion system, minimized harmonic component and load condition of the ship.

The reaction of longitudinal vibration damper should be applied to the axis of the crankshaft due to its full symmetry. To find the application point of thrust bearing reaction is more difficult. The thrust bearing (of Mitchell type) consists of pads supported at their edges and arranged at the perimeter of thrust disk. Generally the pads do not fill completely the full perimeter, being installed at the lower part of the thrust disk. In this case the axis of thrust bearing reaction should be lowered. Another solution consists in applying the thrust bearing reaction in line with the crankshaft axis and adding a suitable bending moment as a transverse dislocation of force does not change the loading system if adequate moment is added. The effect of neglecting this moment on the vibration level of ship hull and its superstructure should be investigated during further numerical analysis.

Regardless of the bearing forces, vibrations of ship hull and its superstructure are generated by pressure pulses induced on the plating above the propeller (transom), as well as by unbalanced internal forces of the main engine. The pressures, their distribution and phase shift are specified by designers of the propeller. The unbalanced forces and moments of the main engine are the last significant forces to be analyzed. In contemporary main engines all the external forces are generally balanced. The remaining moments which excite the following vibrations of engine body, should be investigated: vertical-longitudinal (ML), vertical-horizontal (MH) and torsional vibrations around the vertical axis (MX). For the engine in question, (i.e. MAN B&W 8S70MC-C), only the 1st, 3rd, 4th, 5th and 8th harmonic components of excitations are significant from the point of view of ship hull and superstructure vibrations.

Two main structural models of the 2700 TEU container carrier were used for the computations :

♦ the model for the ship in the design load condition (Fig.2)
♦ and that for the ship in the ballast condition [9,10].

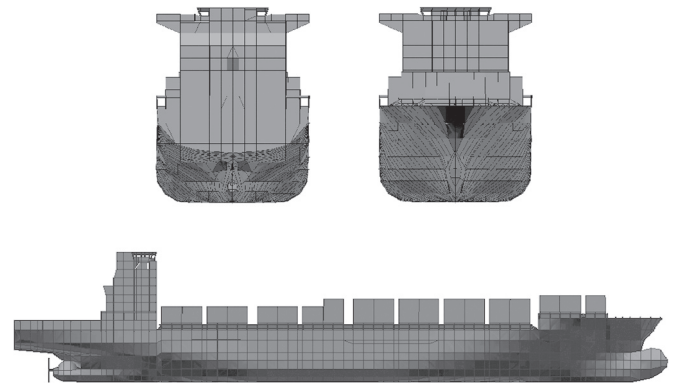Each of the models contained more than 42000 degrees of freedom.

Fig. 2. *Structural model of 2700 TEU container carrier in the design load condition*

Results of the computations are presented for the representative group of the points (all the points, except the bridge wings, are placed in the ship plane of symmetry) located as follows :

⇨ **BOW** – at hull bow on the main deck
⇨ **DTR** – at deck transom edge on the main deck
⇨ **SBF** – at fore bottom of the superstructure
⇨ **SBB** – at aft bottom of the superstructure
⇨ **MEF** – at main engine fore cylinder heads
⇨ **MEB** – at main engine aft cylinder heads
⇨ **STL** – at bridge wing (left)
⇨ **STF** – at fore top of the superstructure
⇨ **STB** – at aft top of the superstructure

KINDS OF EXCITATION

The analyses of forced vibrations were made for the container carrier in the design load condition (Fig. 1) without added mass of water. The wet model of ship hull was used to find the total forced vibrations (generated by complex sources of excitations) and to compare them with measurement results. Computations for the container carrier in the ballast condition were also carried out because it was the condition of the ship during the measurements.

The propeller induces a variable field of water pressure exciting vibration of ship transom deck. The vibrations are transmitted to all parts of the ship structure. Vibrations excited by the first-blade (5th harmonic) component of propeller-induced excitations were analyzed. The image of forced vibration velocities of the ship hull, superstructure and main engine is shown in Fig.3.

Hydrodynamic forces and moments are induced by rotating propeller. Only the variable hydrodynamic forces acting in line with propeller shaft and generating non-coupled axial vibration of the propulsion system are discussed. Phases of these vibrations are strictly associated with the propeller. The non-coupled axial vibrations of the power transmission system generate excitations to ship hull and superstructure through the
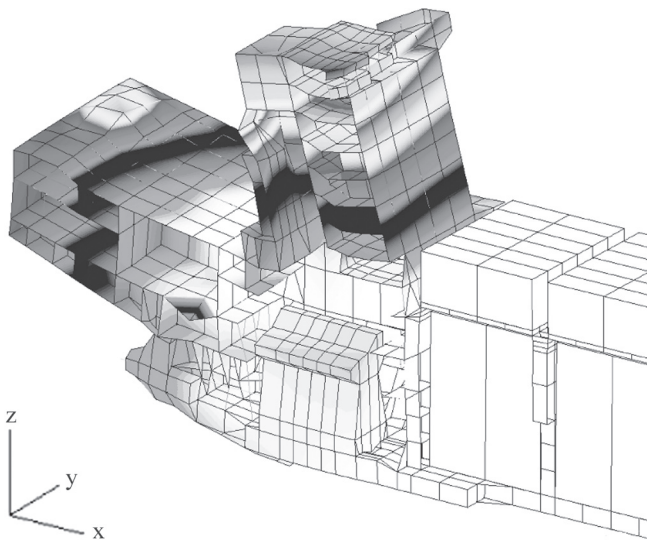
Fig. 3. *Vibration velocities of the container carrier,*
excited by water pressure field on the transom deck

thrust bearing and axial vibration damper. The forced vibration of ship hull, superstructure and main engine is presented in Fig.4.
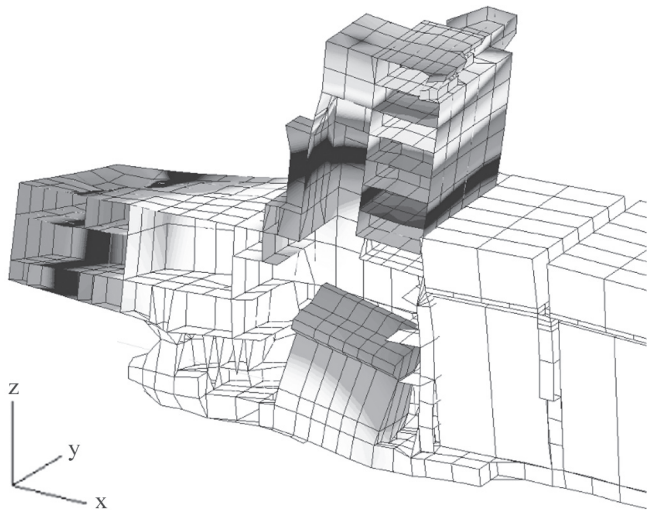

Fig. 4. *Vibration velocities of the container carrier,*
forced by axial vibrations of its propulsion system

Torsional-bending vibrations are generated by ship propulsion system. Bending of crankshaft cranks and torsional vibrations of the crankshaft and propeller generate coupled longitudinal vibrations of the power transmission system. The longitudinal vibrations induce dynamic reactions on the thrust bearing and axial vibration damper, generating excitations to ship hull and superstructure. Phases of the coupled longitudinal vibrations depend mainly on the position of main engine crankshaft.

The coupled longitudinal vibrations are mainly due to gas and inertia forces generated in the engine cylinders. Therefore a relatively wide spectrum of harmonic components may be of significant importance. In the considered case the first eight harmonic reactions (8-cylinder engine) are significant. Therefore eight variants of computation of the forced vibrations of ship hull and superstructure were carried out, i.e. for all significant harmonic components of exciting forces. Distribution of forced vibration velocities for the ship is similar to that shown in Fig.4.

The general image of forced vibration amplitudes
(velocities) for all the considered harmonic components
is shown in Fig.5.
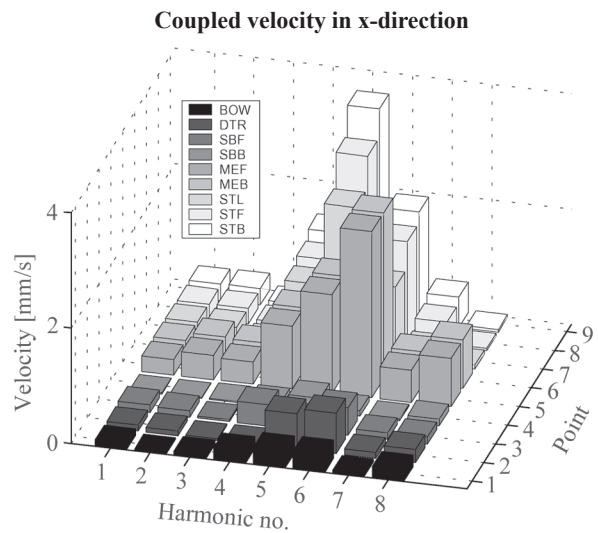
Fig. 5. *Longitudinal vibration velocities*
forced by coupled longitudinal vibrations

As in the case of other kind of exciting forces, the highest vibration level appears in the upper part of ship superstructure. The 5th harmonic component is dominant as in the case of excitations generated by the propeller. The levels of expected vibrations generated by the coupled and non-coupled longitudinal vibrations of the propulsion system are similar. Phases of vibrations generated by the non-coupled vibrations depend on a position of the propeller blades, whereas phases of vibrations generated by the coupled vibrations depend on a position of the crankshaft cranks. Therefore a mutual angular position of the propeller and crankshaft may significantly affect vibration level of ship superstructure. This problem is further investigated below. Side harmonic components (with respect to 5th component), i.e. 6th and 4th components may be dominating for vibrations of the main engine body and ship hull.

The transverse forces and moments induced by rotating propeller generate bending vibration of the shaftline. The bending vibrations generate dynamic reactions of radial bearings of the propulsion system, i.e. the stern tube bearing, intermediate bearing and main bearings of the main engine. The resulting bending vibrations of the shaftline generate forced vibration of ship hull and superstructure.

Fig.6 presents the vibration velocities of ship hull, superstructure and main engine, forced by transverse vibration of the propulsion system.
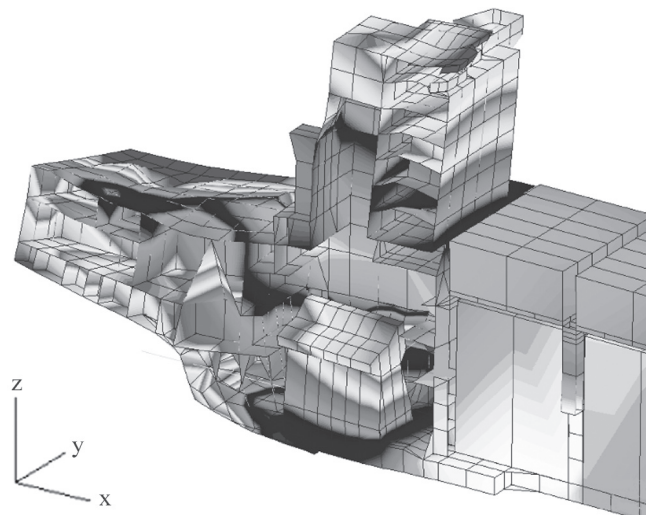
Fig. 6. *Vibration velocities of the container carrier,*
forced by transverse vibration of the propulsion system

Non-balanced external dynamic forces and moments appear as a result of operation principle of piston engines. In ship engines all external forces and high-order moments are balanced. However some of the moments remain still non-balanced. The moments generate longitudinal, transverse and torsional vibrations of the engine, transmitted to other structural parts of the ship. Values of the non-balanced moments are given in the engine documentation. For the considered engine the following harmonic components are significant: 1, 3, 4, 5, and 8.

Velocities of forced vibrations of the ship for: 1st harmonic (excitation of L and X type), 3rd harmonic (excitation of X type), 5th harmonic (excitation of X type) and 8th harmonic component (excitation of H type) for rated rotational speed of the propulsion system are shown in Figs.7 to 10.

During computation of the ship vibrations forced by the external non-balanced moments of the main engine, the excitation moments of the order 1., 3., 5. and 8. appear significant for the considered type of the main engine (8S70MC-C). The 1st harmonic component excites mainly vibration of the hull
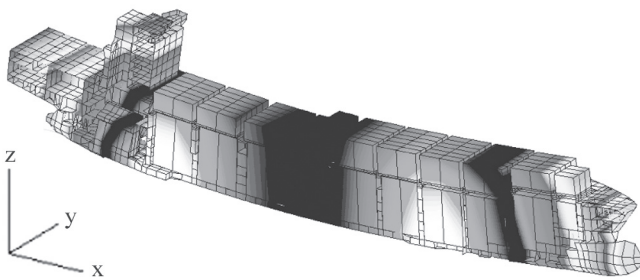

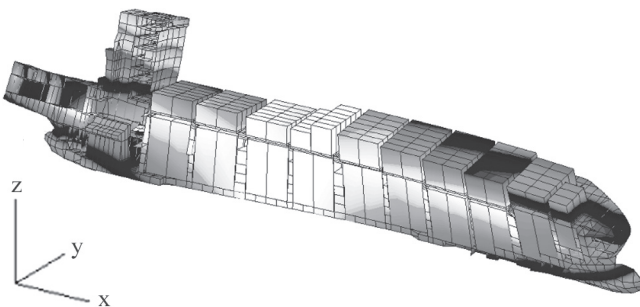**Fig. 7.** *Vibrations forced by 1st harmonic component (L and X type) of main engine's external moments*

Fig. 8. *Vibrations forced by 3rd harmonic component (X type) of main engine's external moments*
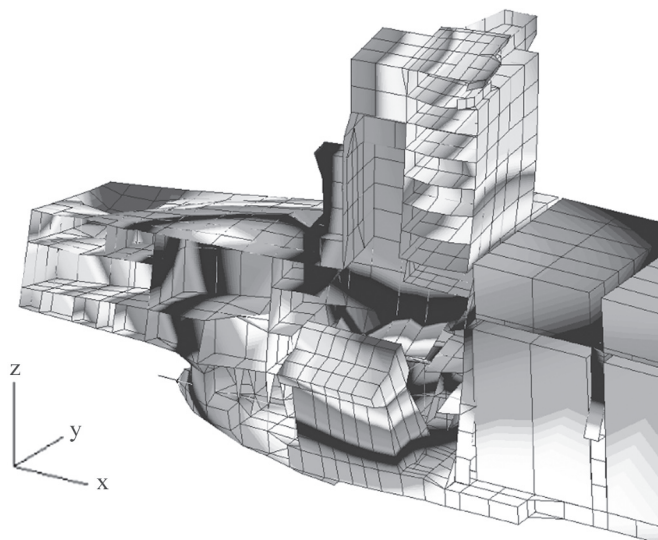

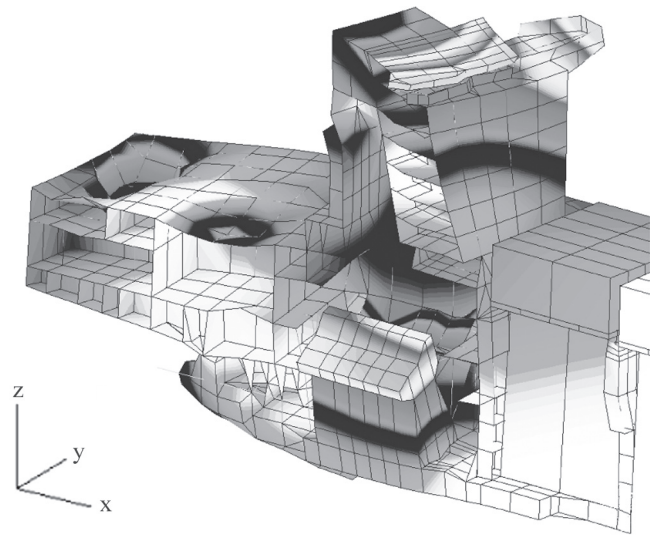**Fig. 9.** *Vibrations forced by 5th harmonic component (X type) of main engine's external moments*

Fig. 10. *Vibrations forced by 8th harmonic component (H type) of main engine's external moments*

structure, whereas the 8th component is responsible for vibration level of the superstructure and main engine (particularly in transverse direction). The excitation forces generated by non-balanced moments of the main engine are responsible particularly for increasing the level of transverse vibrations. For the remaining orientations of vibration other excitation forces are dominating.

OPTIMIZATION OF MUTUAL ANGULAR SETTING OF PROPELLER AND CRANKSHAFT

Two sources of excitations should be taken into account when analyzing the dynamic behaviour of ship hull and superstructure:

♦ the excitations generated by the propeller and
♦ those generated by the main engine.

For the propeller-induced excitations the 5th harmonic component is dominating. The engine-induced excitations are represented by full spectrum of harmonic components (8th harmonic component is main for the engine in question), but also the 5th harmonic component is here significant. According to the above presented analyses the levels of expected vibrations forced by the both excitation sources are similar. Phases of the vibrations generated by the propeller depend on a position of the propeller blades, whereas the phases of the vibrations generated by the main engine depend on a position of crankshaft cranks. Therefore mutual phasing of the propeller and crankshaft may considerably affect vibration level of ship superstructure.

Ship superstructure vibrations of each orientation (longitudinal - x, transverse - y, vertical - z) may be optimized (to reduce its level to a minimum). Fig.11 shows the optimum relative angles of angular position of the propeller blade against the first crank of the crankshaft.

Only one angular position of the propeller can be chosen. The highest vibration level is expected for longitudinal orientation of vibration (in x direction). The assumed optimum angle $\alpha = -20.3°$ gives the minimum level of longitudinal vibration at rated rotational speed of the engine.

Fig. 12 presents the total vibration level at the assumed mutual angular phasing of the propeller and crankshaft. It also presents the maximum vibration level (for the worst angular phasing of the propeller : $\alpha = +15.7°$), as well as the minimum one.
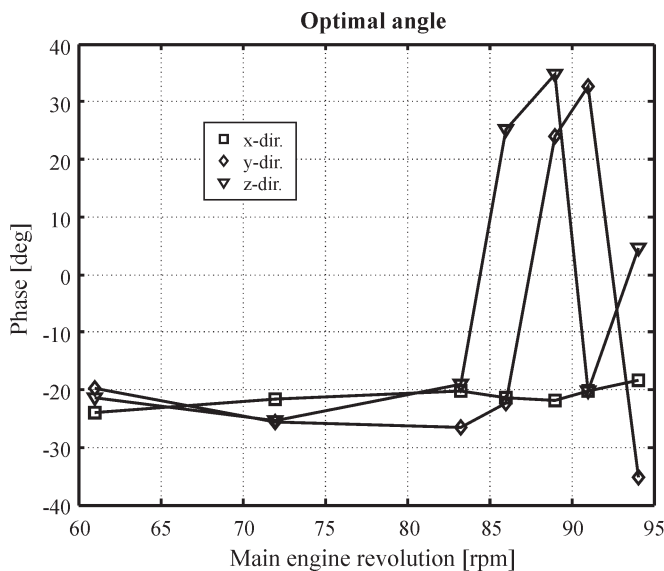
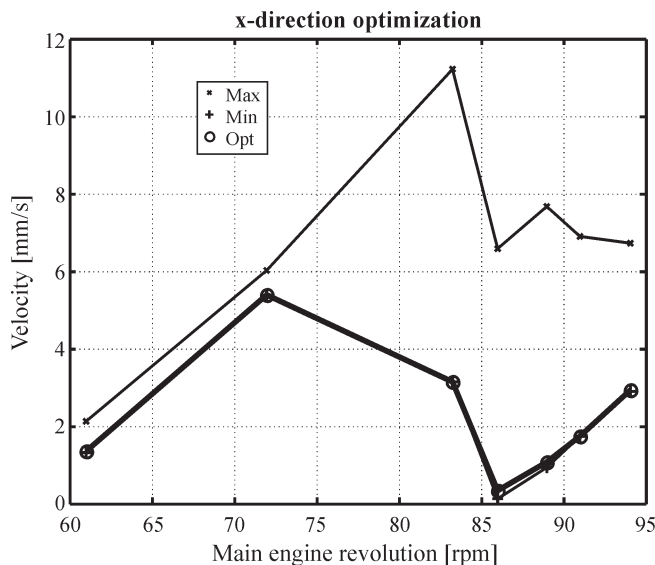**Fig. 11.** *Optimum phasing angles of the propeller against the main engine crankshaft*

Fig. 12. *Longitudinal vibration velocities of the superstructure (bridge wing) for the optimum phasing of the propeller and main engine crankshaft*

Minimum level of longitudinal vibration may be obtained within a wide range of engine rotational speeds for the optimum angle of propeller angular phasing with respect to the crankshaft. It could be achieved because the optimum angles for this orientation of vibration (x direction) were almost constant in function of engine rotational speed (Fig.11). At the same time the vibration levels for other orientations of vibration (e.g. transverse) may be not optimal, for instance vibration velocity values may be higher than the minimum ones theoretically possible.

The optimum angle value (α = -20.3°) is near to that obtained from another method based on analysis of vibration phases of deckhouse (α = -23.8°). This angle was found by optimizing phases of the forces exciting coupled and non-coupled longitudinal vibrations of the power transmission system. Changes in phases of structural response and other excitation sources were not taken into account. The correctness of the optimum angle was verified by the measurements carried out on the ship in question for different angular positions of the propeller. During sea trial an unacceptable vibration level of the ship superstructure was measured. After changing the relative propeller angle the vibration level measured during next sea trial was reduced as much as three times.

SUMMARY VIBRATIONS – – VERIFICATION BY MEASUREMENTS

The above presented analyzes are based on the hull structural model without added mass of water, for the ship in design load conditions. In order to verify the presented method the forced vibrations should be computed both for the ballast and load conditions by using the mathematical model of the ship with added mass of water. In such mathematical model all the formerly analyzed kinds of excitations were assumed to act simultaneously. Additional independent computations were carried out for the vibrations forced by the main engine and then for those forced by the propeller. The results of these computations can be directly compared with the measurement results [2].

The velocities of the forced vibrations of the ship at rated rotational speed of the propulsion system are shown in Fig.13.
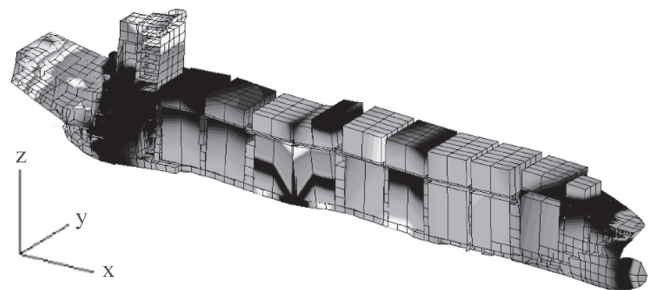

Fig. 13. *Total vibration velocities of the container carrier in design load condition*

Figs.14 and 15 present the diagrams of longitudinal vibration velocities of the superstructure for both considered ship's load conditions, separately for different types of excitation. Possible minimum and maximum vibration levels are also shown. The following notation was used: Prop –excitation by propeller, M.E. – excitation by main engine, Max –maximum vibrations, Min – minimum vibrations, SUM – expected summary vibrations.
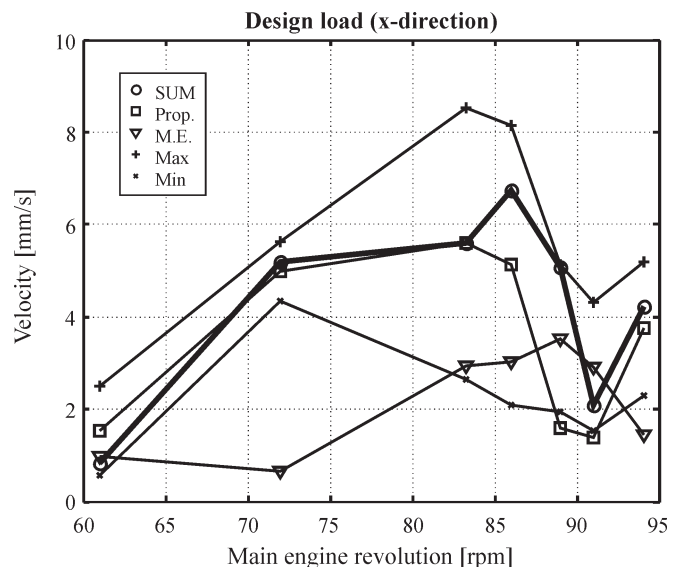

Fig. 14. *Total velocities of longitudinal vibrations of bridge wing (ship in design load conditions)*
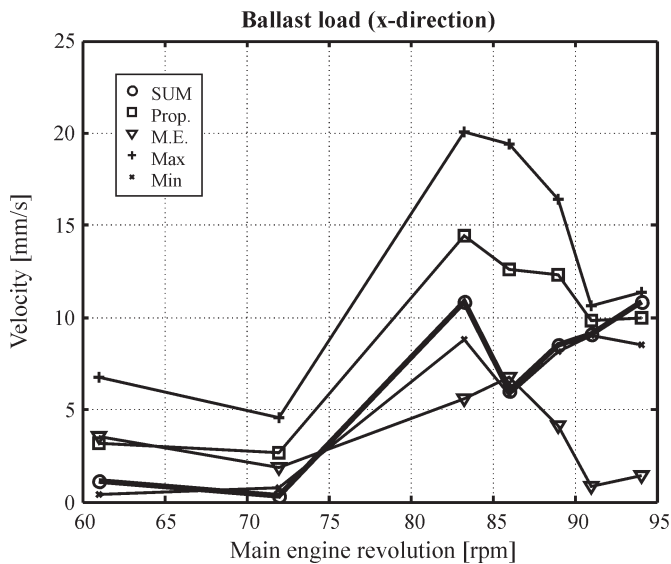
Ballast load (x-direction)

Fig. 15. *Total velocities of longitudinal vibrations of bridge wing (ship in ballast conditions)*

Figs. 16 and 17 present a comparison of the computation results and ship trial measurement results. The results were converted into rms values of vibration velocity to obtain the customary presentation. The comparison is presented for two essential reference areas : the bridge wing of the superstructure and the transom deck.
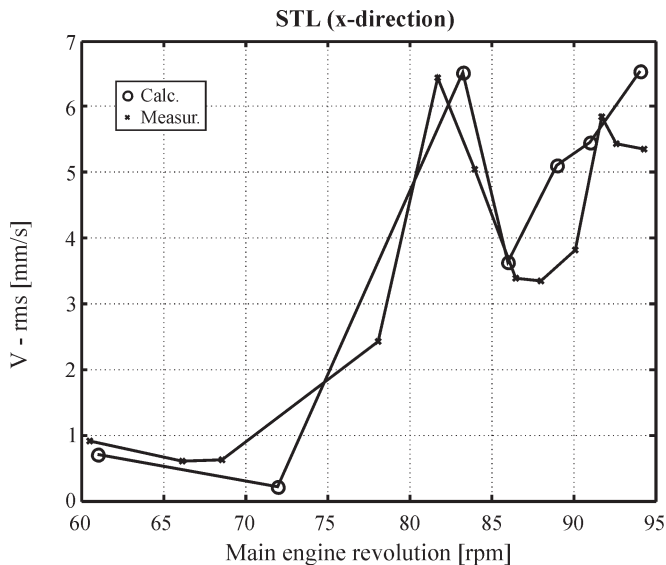
STL (x-direction)

Fig. 16. *Verification of the bridge wing vibration: computation results versus measurement results*

The forced vibration level is clearly higher for the ship in ballast conditions when compared with that for design load conditions. Sea trials are usually carried out in ballast conditions of the ship and therefore even a correctly designed ship may show excessive vibrations of the superstructure during service. Vertical - horizontal vibrations of the upper part of superstructure are dominating. Comparable vertical vibrations may be also observed for the transom deck.

Magnitude of excitation forces generated by the main engine is similar to that generated by the propeller. Therefore, when analyzing the hull and superstructure vibrations, at least the excitation forces generated by pressure pulses on the transom deck and the forces generated by longitudinal vibration of the power transmission system should be taken into account. Ships of an identical hull structure may show a high or relatively low level of vibration depending on whether its propulsion system

is suitably designed. Therefore the complete dynamic analysis of the ship should be carried out simultaneously with its design process – such analysis should be one of its elements.
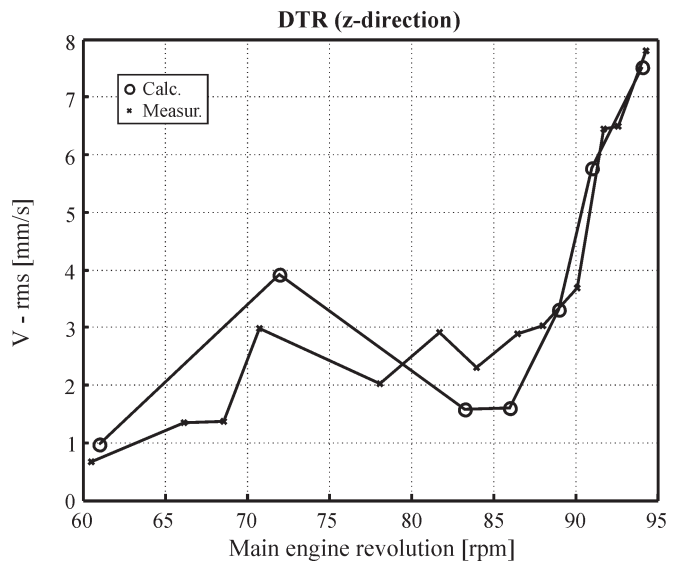
DTR (z-direction)

Fig. 17. *Verification of the transom deck vibration: computation results versus measurement results*

The verification presented in Figs 16 and 17 showing a satisfactory compliance of the numerical computations with measurement results, confirmed that the applied assumptions and computation methods were correct.

RECAPITULATION

❍ The excitation forces generated by non-balanced moments of the main engine are responsible particularly for increasing the level of transverse vibrations. In the case of dynamic analysis of the container carrier in question other excitation forces are dominating (in longitudinal-vertical direction).

❍ If levels of the total vibration generated by the propeller and main engine are similar to each other the vibration levels may be optimized by changing the relative angular position of propeller blades and crankshaft. Vibrations of each orientation may be optimized. The longitudinal vibrations expected to be of a high level (excited by main excitation sources: pressure pulses on deck transom and thrust bearing reaction) are usually optimized.

❍ Forced vibration level is clearly higher for the considered ship in ballast conditions than in design load conditions. The level of excitations generated by the main engine is similar to those generated by the propeller. Therefore, when analyzing the hull and superstructure vibrations, at least the excitation forces generated by pressure pulses on the transom deck and the forces generated by axial vibration of the power transmission system should be taken into account.

❍ Ships of an identical structure may show a high or relatively low level of vibration depending on whether their propulsion system is suitably designed or not. Therefore a complete dynamic analysis of the ship should be carried out in line with its design process, being one of its elements.

❍ Correctness of the used assumptions and computation methods was confirmed by the measurements (Figs 16 and 17) as the presented results of the numerical computations and measurements appeared to be in a satisfactory compliance.

ACRONYMS

BRG - bearing
DSLW - coupled vibration of shaftline
DRG - bending vibrations
FEM - Finite Element Method

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*C*onference

Diagnostics of machines

Faculty of Transport,
Silesian University of Technology, organized

32nd Domestic Symposium on Diagnostics of Machines

which took place on 28 February ÷ 05 March, 2005 at Węgierska Górka, a south-Poland mountain resort.

It was held within the frame of 60th anniversary of Silesian University of Technology and 35 years of activity of „Transport" education line.

Scientific program of the already traditional meeting of experts in diagnostics contained broad spectrum of topics covered by 60 papers prepared by representatives of 16 Polish universities and scientific research centres.

The greatest number of papers was presented by scientific workers from Silesian University of Technology (15) and Warsaw University of Technology (11). Maritime specialists offerred 12 papers :

❖ *Use of current signals for diagnosing bow thrusters* by T. Burnos (Technical University of Szczecin)

❖ *Diagnosing ship propulsion systems on the basis of measurement of operational parameters* – by A. Charchalis (Gdynia Maritime University)

❖ *Service investigations of starting and rundown processes of LM 2500 ship gas turbine* – by A.Charchalis (Gdynia Maritime University) and P. Wirkowski (Polish Naval University)

❖ *Making operational decisions with taking into account likelihood of diagnosis on technical state of machine* by J. Girtler (Gdańsk University of Technology)

❖ *Analysis of trends of vibration parameters of ship gas turbines* – by A. Grządziela (Polish Naval University)

❖ *Assessment of state of injectors of driving engine of synchronous electric generator on the basis of its electric energy parameters* – by G. Grzeczka (Polish Naval University)

❖ *Endoscopic diagnostics of ship engines* – by Z. Korczewski and B. Pojawa (Polish Naval University)

❖ *Subject-matter analysis of trend of changes of metallic impurities in lubricating oil of ship gas turbines* by M. Mironiuk (Polish Naval University)

❖ *Expertise of cause of a failure of piston-rod stuffing--box of a ship low-speed diesel engine* – by L. Murawski (Ship Design & Research Centre, Gdańsk)

❖ *Diagnostics of underwater objects by using ROV vehicles* – by A. Olejnik (Polish Naval University)

❖ *Technical state assessment of injection devices of self--ignition engine on the basis of indication diagram's course* – by R. Pawletko (Gdynia Maritime University)

❖ *Ship double-rotor gas turbine as an object of modelling* – by P. Wirkowski (Polish Naval University).