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Axial vibrations of the ship power transmission system: propulsion shaftline - engine crankshaft

SUMMARY

A computerized calculation method of the axial vibrations of a ship power transmission system is presented. The vibrations may cause increasing failure frequency of the engine crankshaft and be a source of dangerous ship hull and supresstructure forced vibrations. The shaftline axial vibrations are coupled with the torsional-bending vibrations through the engine crankshaft and are forced by the propeller induced hydrodynamic forces (if torsional - axial coupling is taken into account). Several computer algorithms for analyzing the problem have been elaborated by the author.

Nonlinear algorithms for determining the dynamic characteristics of thrust bearing, engine main bearing and axial detuner are also presented.

A crankshaft deformation analysis was carried out to determine coupling coefficients of the torsional, bending and axial vibrations. Sample calculations of ship axial vibrations were performed and compared with experimental results.

INTRODUCTION

Coupled axial vibrations of the mechanical system: propulsion shaftline - engine crankshaft are one of the important sources of ship mechanical vibrations. They endanger the engine crankshaft strength. Moreover, the vibrations are transferred to different regions of the ship hull structure through the thrust bearing (and possibly axial detuner) and ship double bottom. Excessive superstructure vibrations worsen ship's crew working conditions and may detrimentally influence maritime safety. Seriousness of this problem is stressed by all international ship classification societies.

The torsional - bending - axial coupling action of the system should be accounted for while considering its dynamics. Determination of a coupling degree of the axial vibration with other vibrations, crankshaft deformation analysis and torsional vibration calculations are necessary in order to appropriately determine the axial vibrations of a power transmission system. The determination of mutual interference of system vibrations and its boundary conditions is also necessary. In order to obtain it the performance analysis of the engine main bearings, thrust bearings and axial detuners should be carried out. Appropriate determination of the dynamic characteristics of those elements of the propulsion system may be crucial for suitable estimation of its axial vibrations.

CALCULATION METHOD OF AXIAL VIBRATIONS

Main exciting forces of the propulsion system axial vibrations are originated from the following sources [5, 15]:

- crankshaft deformations due to gas and radial mass forces which are applied to particular cranks
- crankshaft deformations due to shaftline torsional vibrations
- unsteady axial hydrodynamic forces induced on screw propeller
- coupling effect of the torsional and axial vibrations through screw propeller.

The DSLW computer program elaborated by the author contains a calculation algorithm of axial vibrations of the mechanical system in question, in which all earlier enumerated coupling effects of the axial vibrations are accounted for. The program is based on the finite element methods [3, 6, 7]. A part of the program for torsional vibration calculations was elaborated with the use of the standard calculation procedures of the vibrations [18], adapted and modernized by the author. The calculation procedures of the coupled torsional, bending and axial vibrations are an important novelty.

Mathematical model equations of the propulsion system motion are of the following form [6]:

$$\mathbf{M}\frac{d^2\mathbf{q}(t)}{dt^2}+\mathbf{C}\frac{d\mathbf{q}(t)}{dt}+\mathbf{K}\cdot\mathbf{q}(t)=\mathbf{h}(t) \qquad (1)$$

where:

M - inertia matrix
C - damping matrix
K - stiffness matrix
h(t) - generalized excitation column matrix
q(t) - generalized displacement column matrix
t - time

The computing sequence of the algorithm is as follows

- determination of characteristic matrix elements
- calculation of propulsion system torsional vibration amplitudes

- calculation of radial forces
- determination of axial force column matrix values on the basis of coupling coefficients
- calculation of natural and forced axial vibrations of the system
- determination of total vibrations for selected values of engine rotational speed.

The motion equation matrices and vectors are formed for torsional and axial vibrations separately. The gyroscopic force influence as well as geometric stiffness matrix is not taken into account in the mathematical model. This assumption is acceptable due to low rotational speed of the shaftline (of about 90 rpm) and low initial loading [5].

The following elements can be distinguished in a real ship power transmission system:

★ crankshaft
★ shaftline
★ screw propeller
★ couplings and torsional vibration dampers
★ engine main bearings
★ thrust bearing and
★ axial detuner.

Their stiffness, inertia or damping values may differ by several orders. Dynamic characteristics of some elements (e.g. of detuners or bearings) can change with engine speed, loading or longitudinal vibration amplitude. In the calculation algorithm each element of the motion equation matrices is formed separately. Two determination methods of the characteristic matrix elements are employed. The magnitudes are determined, according to the first of them, on the basis of producer's technical specifications or empirical relationships from literature sources. In the cases where reliable data were lacking own calculation algorithms were elaborated. This was applied to determining the stiffness and damping of engine main bearings, thrust bearing and axial detuner.

The relative torsional deformation of the crankshaft crank causes its axial deformation. Similarly the radial forces cause an axial deformation of the crankshaft [5]. Crankshaft deformation calculations are aimed at the determination of the ratio of the radial force (relative torsion amplitude) and axial force with the assumption that the axial deformation of the crankshaft due to both loadings is the same. It makes the determination of equivalent axial forces, i.e. shaftline axial vibration excitations, possible. The crankshaft seating boundary conditions, i.e. the dynamic stiffness characteristics of the engine main bearings, should be determined in advance.

CALCULATION ALGORITHM OF MAIN BEARING CHARACTERISTICS

Reliable data on the dynamic characteristics of ship engine main bearings are lacking in the available literature sources. However the substantial influence of the appropriate modelling of the crankshaft seating in main bearings is stressed in many works [8, 9, 16]. Therefore an own algorithm for the performance analysis of main bearings of low-speed marine engines was elaborated, which was aimed at the determination of the nonlinear dynamic stiffness of crankshaft seating in engine frame. It is hereafter used to model crankshaft boundary conditions more precisely and to more exactly determine stiffness values of the crankshaft loaded by radial forces or torque.

The narrow bearing theory was employed in the calculation algorithm, elaborated by Ocvirk for hydrodynamic lubrication and by Michell for lubricant squeezing-out. Gümbel's boundary conditions were applied in it [1]:

$$p \neq 0; \quad \varphi \in (0, \pi)$$

where:

p - lubricating oil film pressure
φ - angular coordinate of crankshaft journal position

The Reynolds equation which describes the lubricating oil film pressure distribution in a bearing is based on the Navier - Stokes liquid motion equations and flow continuity equations [1]. The calculation procedure of the program is based on the finite difference method [2] selected due to its simplicity and complexity of the program algorithm [4].

Main bearing dynamic characteristics

The calculations were carried out for the main bearing of the Sulzer 6 RTA-76 low-speed engine. The main bearing is of the following parameters:

- crankshaft journal diameter - 790 mm
- bearing width - 375 mm
- radial clearance - 0.58 mm
- engine rated speed - 87 rpm

The main bearing performance parameters and journal centre motion trajectory at the engine rated speed, and the average radial clearance value (0.58 mm) and the average lubricant viscosity (0.1 Ns/m) taken into account, are presented in Fig. 1 and 2 on the basis of the calculations in which the time increment is equal to 1/400 of shaft rotation period. Fig. 1 presents the following parameters: the exciting force value, shaft journal centre displacement and angle between the displacement and direction of force versus crankshaft rotation angle. Zero rotation angle corresponds to the top dead centre of the first engine cylinder piston. In Fig. 2 the relative motion trajectory of one crankshaft journal rotation is shown. In the figure the displacement equal to one (r = 1 in polar coordinate system) corresponds to the fully reduced clearance. The bearing performance parameters are presented in the crank-related system. The motion trajectories (Fig. 2.) are drawn in the absolute coordinate system.

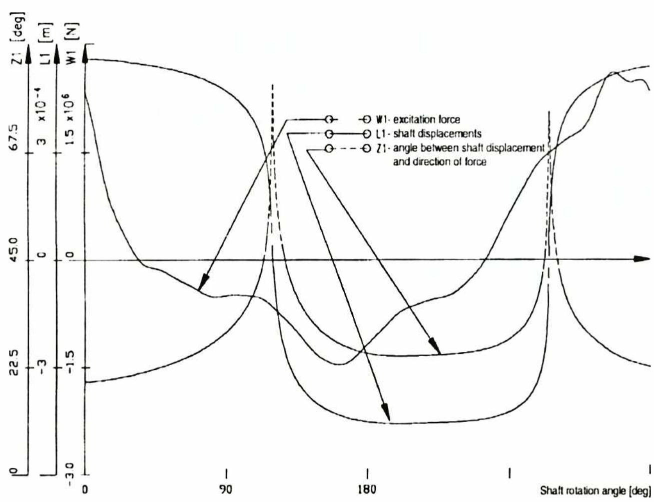

Fig. 1. Main bearing performance parameters at 87 rpm

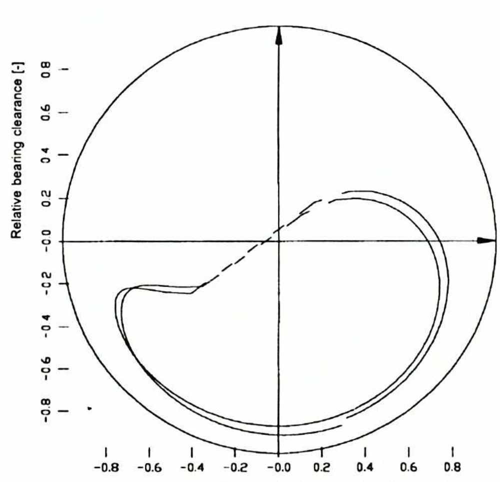

Fig. 2. Motion trajectory of crankshaft journal centre at 87 rpm

The calculated minimum thickness of oil film was 45 μm at the rated speed and 36 μm at 50 rpm. The values are in compliance with those given in the literature (50 μm acc. to [19]).

The main bearings of the engine in question are intensively loaded; the shaft journal displacements are highly nonlinear (see Fig.1). The crankshaft seating should be modelled in such a way, when analyzing crankshaft kinematic deformations, as to obtain a displacement of the journal at a real load of the main bearing.

Crankshaft deformation analysis

The k_r coefficient which reflects the coupling effect of the radial forces (F_r) and axial forces (F_l) was defined as follows:

$$k_r = \frac{F_r}{F_l} \qquad (2)$$

The coefficient k_t which reflects the coupling effect of the crankshaft torsional deformation ($\Delta\phi$) and axial forces (F_l) was defined in a similar way:

$$k_t = \frac{F_l}{\Delta\phi} \qquad (3)$$

Crankshaft deformations were calculated by means of the Finite Element Method based ADINA and NASTRAN software systems [17, 20] which make fully nonlinear analysis possible. The bending-torsional -axial vibration coupling coefficients were determined on the basis of the deformation analysis of the crankshaft loaded by axial forces, radial forces and torques.

The sample crankshaft model of B&W 6S70MC engine is shown in Fig. 3. Its deformations due to the radial forces applied to the fourth crank is depicted in Fig. 4. The deformed mesh scale has been expanded in order to make the image more clear.

Fig. 3. Crankshaft model of B&W 6S70MC engine

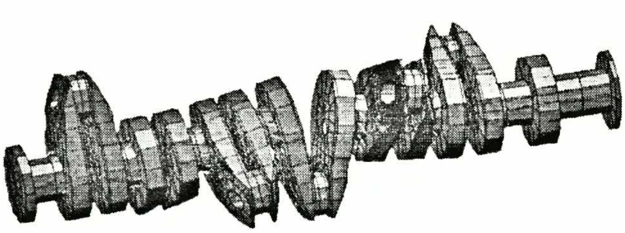

Fig.4. Dimensionless deformation of the crankshaft due to the radial force applied to the fourth crank

Influences of the crankshaft deflection in the main bearings on crankshaft axial stiffness and on values of the k_r and k_t coefficients were analyzed. The crankshaft deflection in the main bearing accounts for the lubricating oil film dynamic stiffness and stiffness of the engine frame founded on the ship double bottom. The influences calculated for the 6 RTA-76 engine are presented in Fig. 5, 6 and 7. The real crankshaft deflection value in the main bearings of 6 RTA-76 engine is 0.65mm.

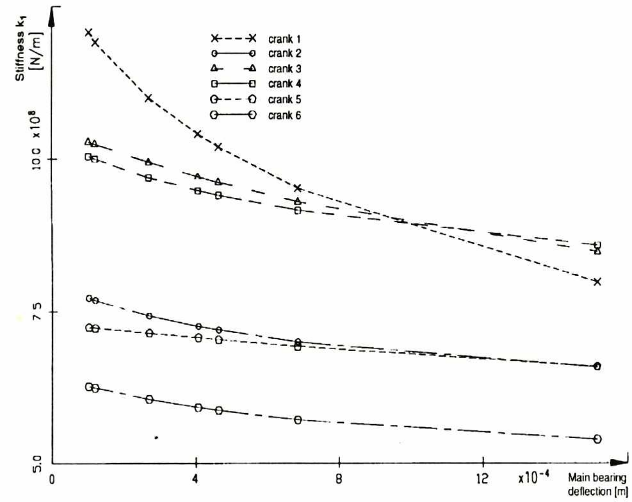

Fig. 5. Influence of crankshaft deflection in the main bearings on its axial stiffness

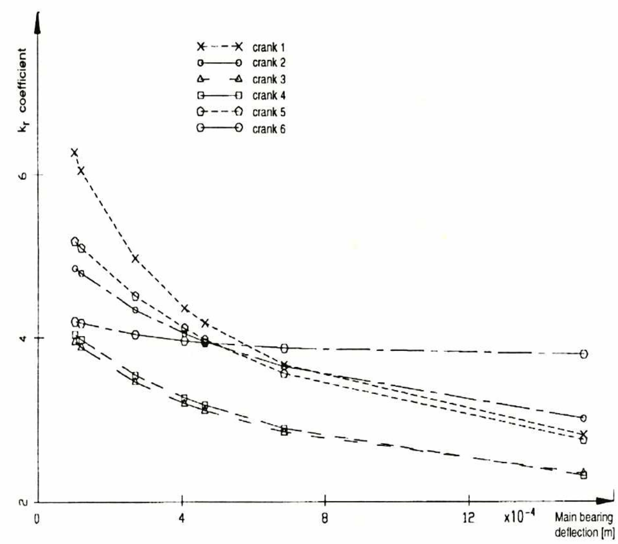

Fig. 6. k_r coefficient versus crankshaft deflection in main bearings

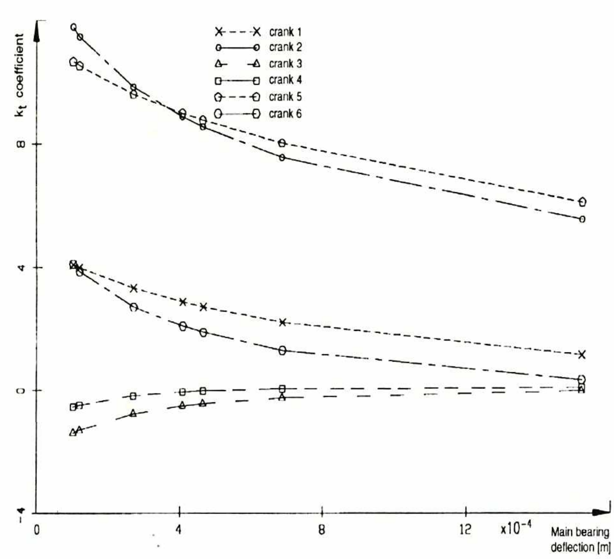

Fig. 7. k_t coefficient versus crankshaft deflection in main bearings

CALCULATION ALGORITHM OF THRUST BEARING CHARACTERISTICS

The ship thrust bearings transfer the hydrodynamic thrust of ship screw propeller. The static thrust bearing load is caused by the constant thrust component. The main dynamic thrust bearing load is due to the axial vibrations of the shaftline-engine crankshaft system [11].

It was assumed that:

- steel structure stiffness of the bearing together with its seating is linear and independent of shaftline rotation speed [14]
- sliding surfaces are undeformable
- shaftline rotation speed is steady.

Additionally, the following assumptions connected with the Reynolds theory were adopted:

- lubricating oil flow is laminar
- liquid is incompressible and of zero inertia
- lubricating oil viscosity is constant
- temperature influence on bearing performance characteristics is neglected [12].

The Reynolds equation which describes pressure distribution in lubricating oil film can be written as follows [12]:

$$\frac{\partial}{\partial R}\left(R\,\frac{h^3}{\eta}\frac{\partial p}{\partial R}\right)+\frac{1}{R}\frac{\partial}{\partial \varphi}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial \varphi}\right)=6R\,\omega\frac{\partial h}{\partial \varphi}+12R\,\frac{\partial h}{\partial t} \tag{4}$$

where:

R - radius
φ - angular coordinate
h - lubricating oil thickness
p - pressure in lubricating oil film
η - lubricating oil viscosity
ω - thrust disc rotation speed
t - time

The expession (4) which describes oil pressure distribution on a thrust bearing block , used in the author's calculation algorithm , was solved after substitution of the set of differential equations by that of finite difference equations. The Gauss - Seidl iterative method was applied to solve the algebraic linear equation set [2].

Bearing dynamic characteristics

Calculations and test measurements were carried out for a bulk carrier of 163 000 dwt and 283 m length. The thrust bearing of Sulzer 6 RTA-76 main engine installed on the ship was characterized by the following parameters:

- number of blocks - 6
- block external radius - 880 mm
- block internal radius - 445 mm
- block contact angle - 35.5 °

Axial vibration amplitudes of the power transmission system are a dynamic load of the thrust bearing in the algorithm in question. It is necessary to know preliminary values of shaftline axial vibrations of a ship to determine the bearing dynamic characteristics. The values were calculated by using the algorithm described in [11]. The torsional vibration resonance appears at the engine speed of n = 51.2 rpm, which causes an elevated axial vibration level (due to torsional-axial vibration coupling in the system).

The bearing performance parameters at the rated speed (87 rpm) of the propulsion system of the ship in question are shown in Fig. 8. At this speed the 6-th harmonic component of axial vibrations of the power transmission system is dominant. In the figure the change of lubricating oil film thickness and block inclination angle due to the system axial vibrations are presented.

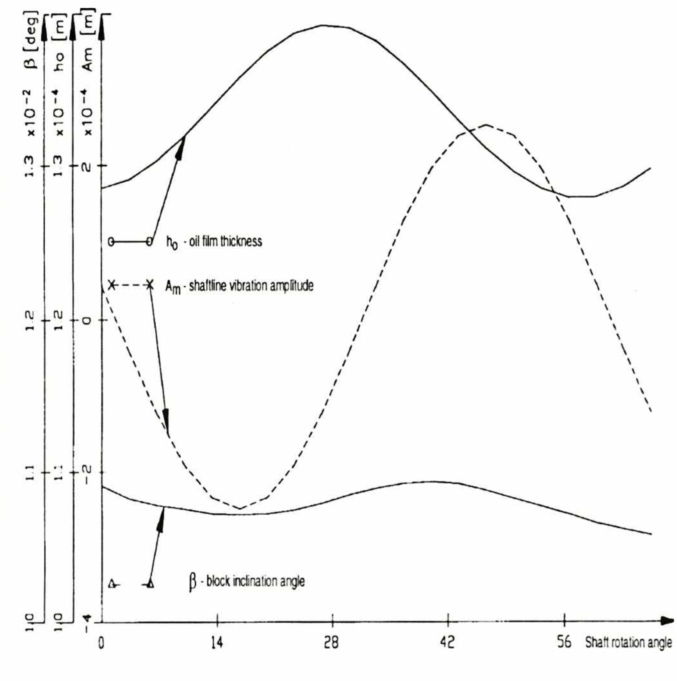

Fig. 8. Influence of the propulsion system axial vibrations on the bearing performance parameters at n = 87 rpm

The thrust bearing performance parameters at the engine resonance speed 51.2 rpm (increased axial vibrations coupled with torsional vibrations) are presented in Fig.9. At this speed the maximum dynamic load of the thrust bearing appears. The bearing performance parameters are presented in the analogical way as those at the rated speed.

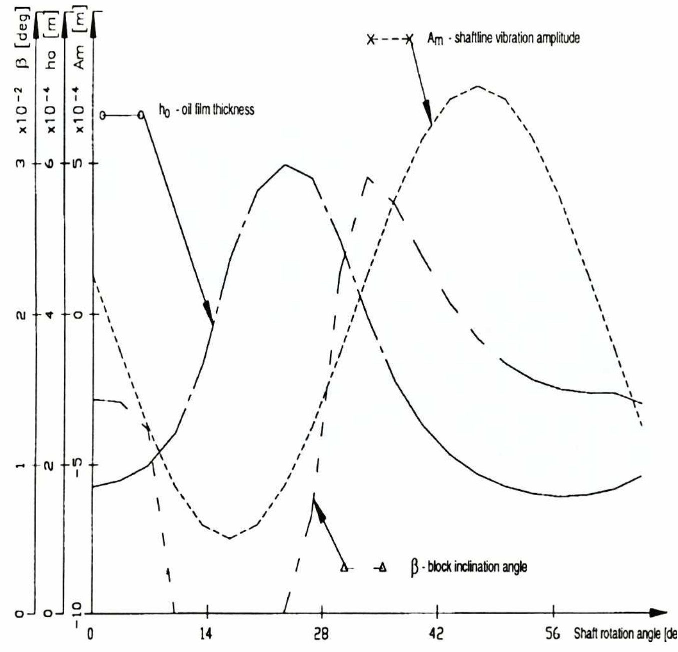

Fig. 9. Influence of the propulsion system axial vibrations on the bearing performance parameters at n = 51.2 rpm

The sample oil pressure distribution on a block of the thrust bearing loaded by axial dynamic forces at 51.2 rpm is shown in Fig. 10. The calculated dynamic load of the bearing load-carrying segments is greater than the static ship propeller thrust. The dynamic load amplification factor is 1.5 at the engine rated speed and 4 at the torsional vibration resonance. The lubricating oil pressure on the astern running blocks reaches the value equal to that due to propeller thrust. The dynamic characteristics of the analyzed thrust bearing is highly nonlinear (Fig. 9) at the engine resonance speed (51.2 rpm).

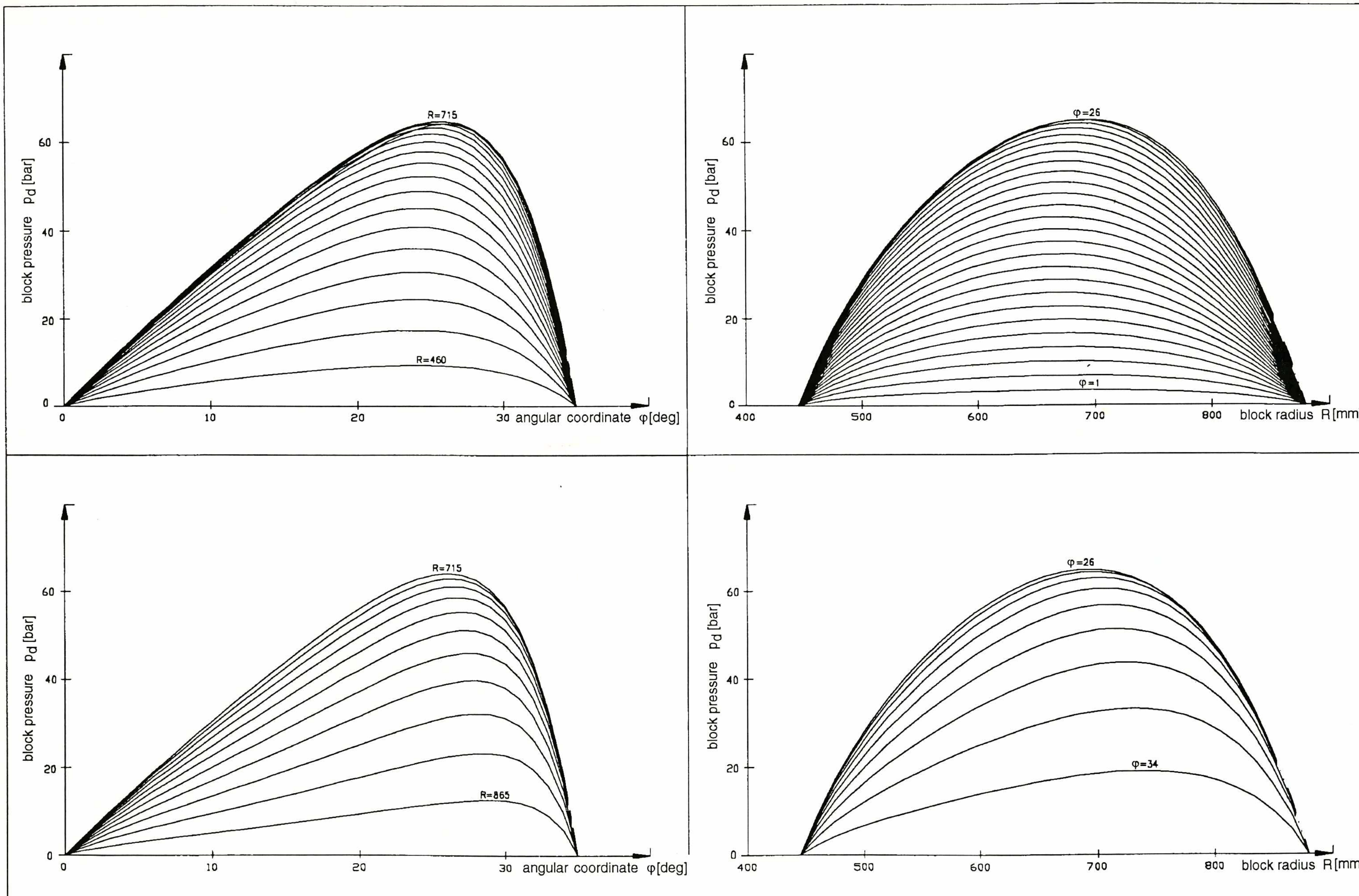

Fig. 10. Distribution of the maximum blok pressure under dynamic load at n = 51.2 rpm

Axial vibration analysis of the bulk carrier

A calculation procedure and results of the coupled axial vibrations of the power transmission system of the earlier described bulk carrier is presented here. The calculations and test measurements were carried out by the methods given in [10] on the basis of the above discussed calculation results of crankshaft deformations and thrust bearing dynamic characteristics. The coupling coefficients of the torsional- bending- axial vibrations of the propulsion system were determined on the basis of the crankshaft deformation analysis with boundary conditions estimated from the main bearing calculations.

The experimental verification was performed by the research team of the Structural Mechanics Division, CTO, during sea trials of the ship. Axial vibration amplitudes were measured close to the thrust bearing (at the flywheel) and on the opposite side of the crankshaft (at the „ free end " of it).

The following first natural frequencies of the system axial vibrations were obtained from calculations: 11.07 Hz, 19.39 Hz, 33.38 Hz. Results of the axial vibration analysis of the system in question justified by the sea trial measurements are presented in Fig. 11 to 14.

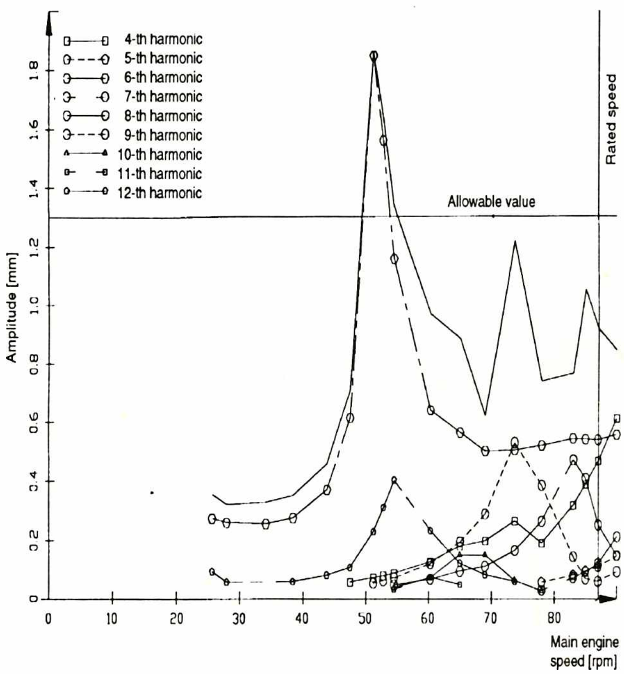

Fig. 11. Calculated vibration amplitudes of the crankshaft „ free end "

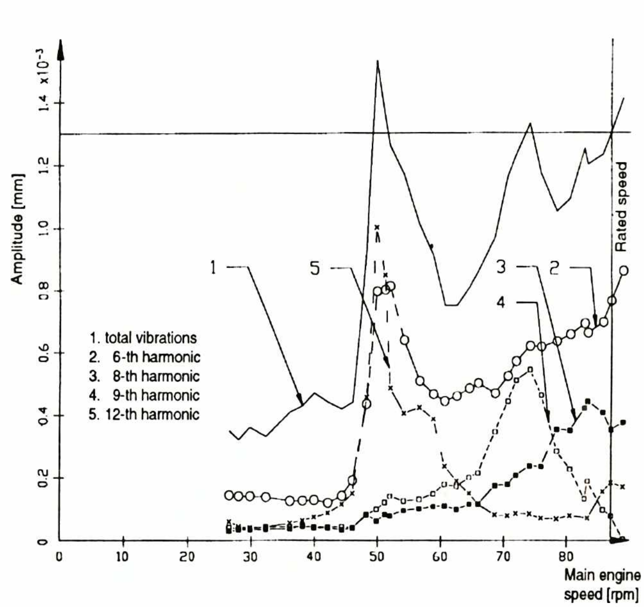

Fig. 12. Vibration amplitudes measured at the crankshaft „free end "

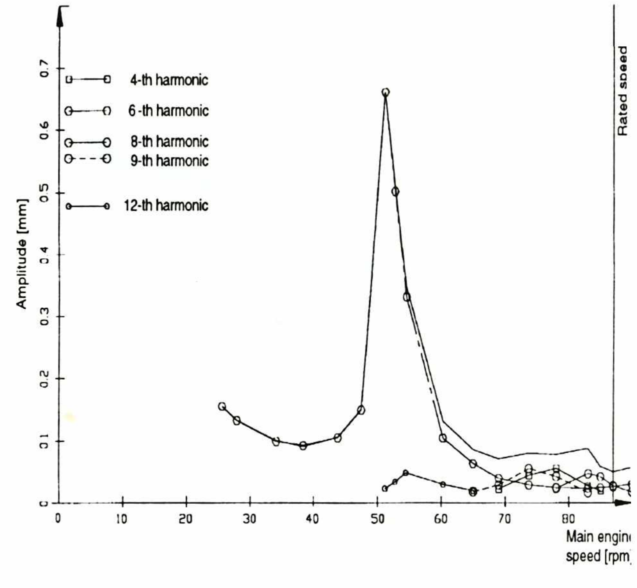

Fig. 13. Calculated vibration amplitudes of the thrust bearing

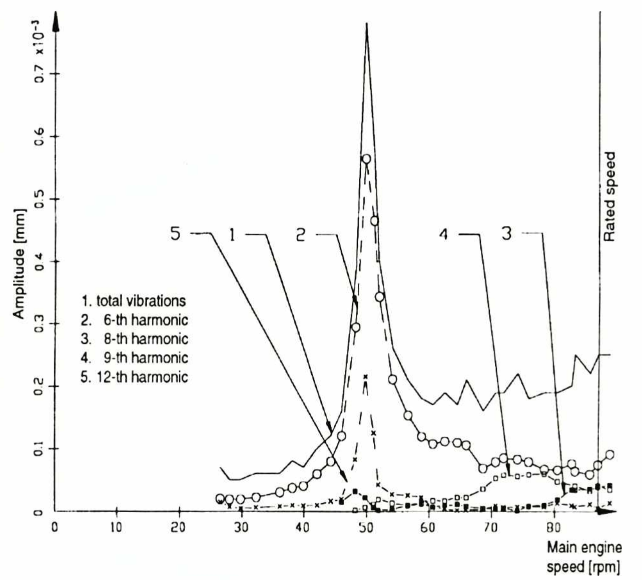

Fig. 14. Measured vibration amplitudes of the thrust bearing

CALCULATION ALGORITHM OF AXIAL DETUNER CHARACTERISTIC

The axial detuners are applied in the low-speed marine engir to mitigate axial vibration amplitudes of the crankshaft free er Excessive vibration amplitudes can cause ship's propulsion syste to fail. The axial detuner introduces, apart from mitigation vibration amplitudes, an additional excitation source of ship h and superstructure vibrations. The resultant excitation force whi appears in the detuner and thrust bearing can exceed the excitati force of such system without any detuner. The conditions of t vibration amplitude mitigation are often opposite to those of t minimization of excitations.

The Reynolds equation which describes pressure distributi in oil film is analogical to the expression (4). The term which accoun for oil wedge action is only neglected. The energy losses due to t drag of lubricating oil flow between detuner chambers are taken in account in the second part of the calculation algorithm. Values of t drag coefficients λ and ζ are assumed according to [13].

The mutual interaction of the axial vibrations of the bulk carri and detuner performance parameters was accounted for during th analysis. Calculation results at the shaftline resonance speed (51.2 rpr are presented in Fig. 15. Different ratios of control valve openin were analyzed. A change of the ratio causes the detuner stiffness an reaction applied to detuner casing as well as shafline axial vibratio amplitudes to change.

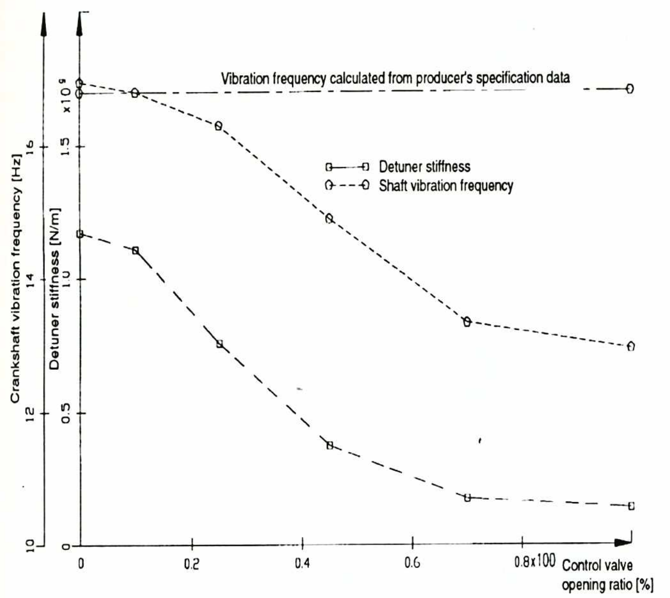

Fig. 15. Detuner stiffness and power transmission system vibration frequency at 51.2 rpm

25% opening ratio of the detuner control valve was hereafter assumed for calculations. The detuner stiffness at this regulation setting is in accordance with producer's specification at engine rated speed. The detuner performance analysis connected with axial vibration calculations was performed over full range of engine speed. The detuner stiffness and shaftline vibration frequency versus engine speed is presented in Fig. 16.

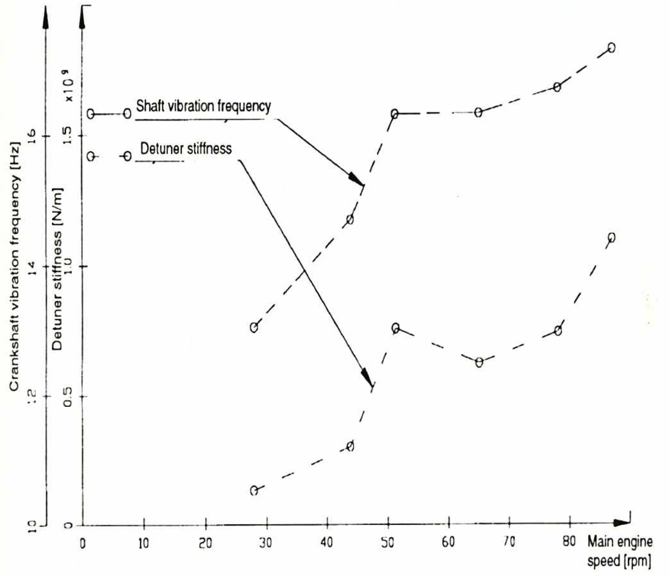

Fig. 16. The detuner stiffness and power transmission system vibration frequency at 25% control valve opening ratio

Axial vibration analysis of a tanker

The propulsion system axial vibration calculations with the earlier determined detuner characteristics taken into account were performed in order to verify correctness of the assumed calculation methods of axial detuner characteristics. The verification which consisted in the comparison of results of calculations and real ship measurements was carried out for the tanker of 90 000 dwt and 246.9 m length.

The shaftline torsional vibration resonance of the considered propulsion system appears at the engine speed of n = 44 rpm. It causes an elevated level of axial vibrations.

For calculation purposes the system dynamics is considered as that in linear range but with nonlinear boundary conditions (due to detuner and thrust bearing). The natural vibration frequencies and forms of the power transmission system were determined in the first step of calculations.

Results of the axial vibration amplitude analysis are shown in Fig. 17 to 19 .In Fig. 19 only the 6-th harmonic component is presented. The cause was a low level of the harmonic components in relation to the stochastic ones.

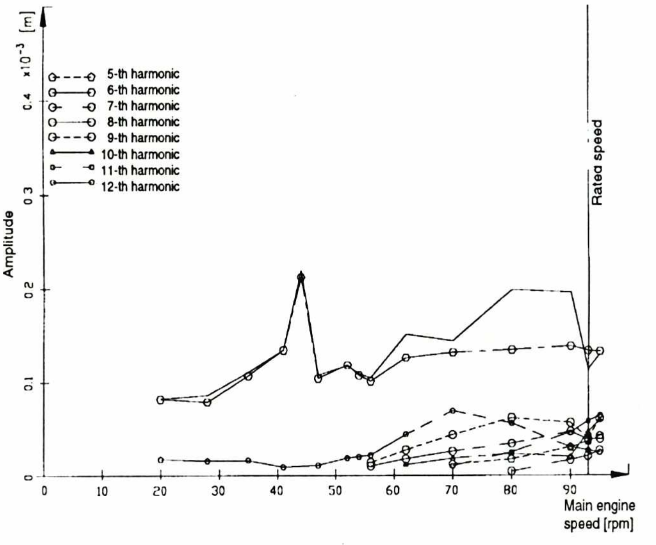

Fig. 17. Calculated axial vibration amplitudes of the crankshaft „free end"

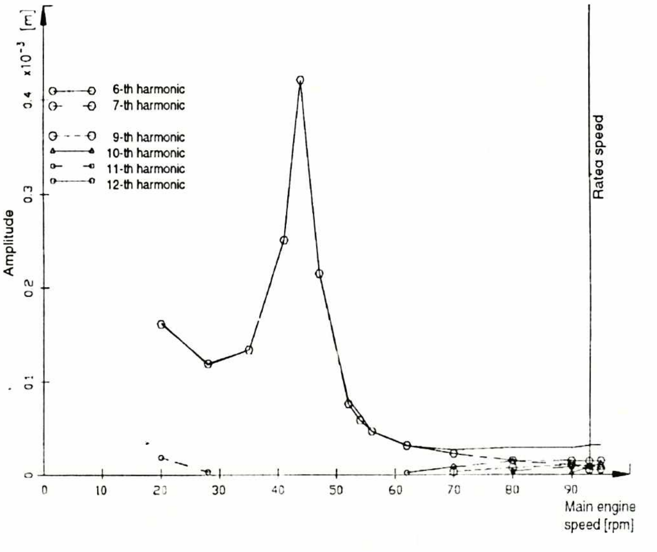

Fig. 18. Calculated vibration amplitudes of the thrust bearing

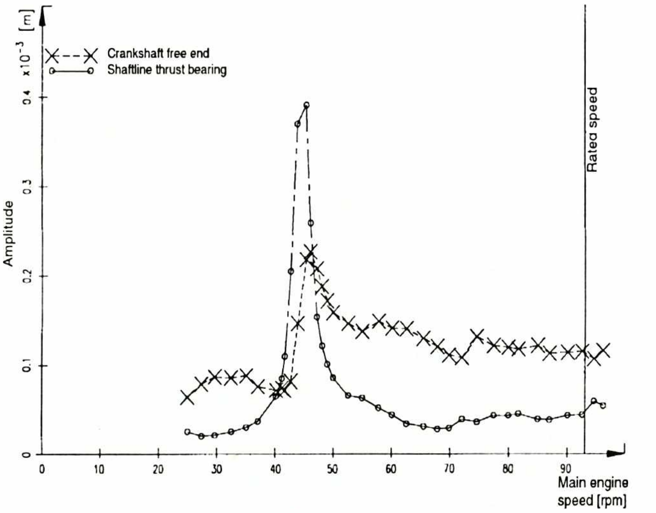

Fig. 19. Measured vibration amplitudes of the tanker propulsion system

CONCLUSIONS

● The experimental verification confirmed that the assumed mathematical models of the marine power transmission systems and their boundary conditions were correct. The calculated crankshaft vibration amplitudes were well correlated with those measured both at the rated speed and that where torsional vibration resonance appeared.

● The performance characteristics of the analyzed thrust bearing is highly nonlinear (Fig. 2.) at the engine resonance speed. The thrust bearing stiffness and damping change as the crankshaft axial vibration amplitudes and frequencies change. Its loading due to static propeller thrust also substantially influences the stiffness-damping characteristics. Application of a constant stiffness of the bearing within full operation range, which has been a usual practice till now, does not make obtaining exact results of such analyses possible.

● The axial detuner stiffness changes as the crankshaft axial vibration amplitudes and frequencies change. The change of engine speed or of the detuner regulation setting have also an important influence on the detuner stiffness-damping characteristics. The detuner applied in a propulsion system can be regulated in such a way as to obtain its stiffness of the same value as that provided solely for a single engine speed in producer's specification.

● The axial detuner effectively mitigates vibration amplitudes of the crankshaft „free end " ,but it is rather ineffective in decreasing vibrations at other points of the propulsion system. The total value of external excitations (at the detuner and thrust bearing) would not always be lowered after application of the detuner. Similarly, the vibration level of ship's hull and superstructure would not always be improved due to the introduced axial detuner.

● The crankshaft - shaftline system should be considered as a linear mechanical system with nonlinear boundary conditions. Both stiffness and damping of the thrust bearing and detuner should be determined at several engine speeds. Such parameters as vibration amplitude, bearing loading, or detuner regulation setting as well as the mutual interaction of detuner's characteristics and axial vibrations should be also accounted for. The values of detuner and thrust bearing stiffness specified by the engine producers can be used for assessment calculations only.

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Appraised by Wiesław Ostachowicz, Prof.,D.Sc.,M.E.

Conferences

CTO

KORAB TOP

XVII Ship Technology Scientific Session

Jurata, a small summer resort and fishermen village at the Hel Penninsula, hosted the annual Ship Technology Scientific Session from 29 to 31 May 1996. Such sessions are, according to their program assumptions, meant to be the domestic meetings of scientists, designers, builders and representatives of managing staff of Polish shipbuilding and shiprepair industries, aimed at experience exchange, presentation of recent achievements and formulation of the most vital tasks for the future.

The reported Session was special due to its leitmotive :

„Polish shipbuilding industry 1996 - a challenge of the future"

Its plenary part was devoted to the following issues:

● assessment of today state of Polish shipbuilding industry and conditions for its further development in view of competitive environment of the world shipbuilding and shiprepair industries

● consequences of signing the OECD Agreement on abiding by the usual competitiveness terms and of entry of Poland to the European Union, for Polish shipbuilding industry.

The session was organized by the Ship Design and Research Centre (CTO) and the Society of Polish Naval Architects and Marine Engineers (TOP - „KORAB").

It gathered 220 participants who represented more than 60 Polish and foreign institutions, scientific centres and enterprises. They had an opportunity to be acquainted with 54 papers whose central idea was a role of innovativeness for further development of the Polish shipbuilding industry.

The presentation of the papers was carried out at the plenary session (13 papers) and topic sessions in the following two areas:

I - Design, engineering processes and organization in shipbuilding

II - Ship and marine environment safety in ship design, production and operation.

Problems presented in the papers met so high interest of the participants that an additional discussion session was needed. It was also possible to visit an exhibition where 11 firms dealing with computer science and welding presented their commercial offer and technical papers.

The session proceedings were published in 3 neatly edited volumes.