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DIAGNOSING ELEMENTS OF PROPULSION PLANT OF NAVAL VESSELS BY MEANS OF VIBRATION MEASUREMENT

SUMMARY

The application of vibro-acoustic analysis methods in naval technology has been presented in the paper. This element of Base Diagnosing System (BDS) is accepted and used in the ships, which are powered by the COGAG power plant. The paper presents investigations of permissible in-service unbalance and appropriate assemblage of turbine rotors on the basis of selected vibroacoustic parameters, and finally determination of their permissible operation time resources. Another element of BDS is vibration control of misalignment of propulsion shafts. The described conception concerns evaluative process of the centring state in a transmission shafts within powered, marine gas turbine system as a function of ships displacement. Some structural components of the gas turbine unit and reduction gearbox have been selected for the analysis. Some results of the vibro-tests have been presented as well. All tests had been worked out during sea trials. This described idea is based on the researches of the corvettes' 1241 type power plants.

***Keywords**: propulsion plant, gas turbine engines, diagnosing, vibration*

ZASTOSOWANIE WIBROAKUSTYCZNYCH METOD DRGANIOWYCH W DIAGNOSTYCE OKRĘTOWYCH SILNIKÓW TURBINOWYCH

W artykule przedstawiono zastosowanie wibroakustycznych metod drganiowych w diagnozowaniu okrętowych układów napędowych. Prezentowane metody są integralnym elementem Bazowego Systemu Diagnostycznego obsługującego okręty wyposażone w układy napędowe typu COGAG. Przedstawiono wyniki badań dopuszczalnego niewyważenia eksploatacyjnego wirników turbinowych silników spalinowych wraz z metodą oceny dopuszczalnego czasu eksploatacji do następnych czynności obsługowych. Innym elementem systemu zaprezentowanym w artykule jest kontrola współosiowości elementów układu napędowego. Przedstawiona metoda polega na ocenie stanu współosiowości jako funkcji wyporności kadłuba okrętowego. Analiza dotyczy wytypowanych elementów w układzie turbinowy silnik spalinowy – przekładnia redukcyjna. Przedstawiono również wyniki badań na obiektach rzeczywistych, wykonane w trakcie testów morskich. Prezentowane wyniki dotyczą badań zrealizowanych na układach napędowych korwet typu 1241RE.

1. BASE DIAGNOSING SYSTEM

Since 1984 the vessels equipped with gas turbine engines were operated in the Polish Navy. From the side of users, doubts are often expressed concerning maintenance times or making decision on further exploitation of engines. It is very important task in the case when all elements of propulsion system are foreign.

Application of periodical or on-line diagnostic procedures makes it possible to operate ship propulsion systems in accordance with their current technical state [3]. Especially, in the case when ships' gas turbines hourly period of scheduled maintenance is presently the criteria for maintenance time determination. Though such exploitation strategy makes early scheduling of maintenance operations and their logistic assurance possible, but it simultaneously contributes to increase of costs because of its replacement system of elements (technically often still serviceable ones) as well as it makes impossible to early detect primary symptoms of failures occurring before the end of maintenance time.

2. OBJECT OF INVESTIGATIONS

The Tarantula class, among other Polish Navy ships, are also subject to a permanent basic diagnostic system. They are fitted with COGAG gas turbine propulsion systems. To obtain reliable data on diagnostic parameters, investigations of the gas turbines installed in the presented propulsion system were carried out by means of the multi-symptom diagnostic model whose one of the main features is recording and analysing vibroacoustic signals. The investigations were aimed at determination of permissible in-service unbalance and appropriate assemblage of turbine rotors on the basis of selected vibroacoustic parameters, and – finally – determination of their permissible operation time resources. The investigations were based on the following assumption: if technical state degradation of gas turbine rotor sets is a function of their operation time (at a load spectrum assumed constant) then it is possible to select from the recorded vibration signal spectrum such parameters whose changes can be unambiguously assigned to the operation time [1, 2]. Second one important problem is shaft misalignment be-

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tween engines and reduction boxes and propeller and reduction box. Dynamic reactions, resulting from exceedance of allowable alignment deviations of the torque transmission elements are able to cause failure of the propulsion system and even lead to loss of movability of the vessel in a relatively short time [2]. Therefore diagnostic control of the gas turbine power plant in operation became necessary.

Appropriate assembling the main engines and the other torque transmission elements inclusive of propellers is practically determined by a set of tolerated dimension and geometrical location requirements, called geometrical dimension assembling chain [3]. Both typical and modular power plants are prone to coaxiality deviation from its permissible values and in consequence to possible failures one or more elements of the propulsion system. The excessive deviation can lead to the loads on bearings and gear teeth much higher than calculated and in result to their premature failures [4].

3. THEORETICAL ASSUMPTION OF ROTOR DYNAMICS

Application of computer simulation for diagnosing a technical state of gas turbines rotor sets should be applied during calculation and project process. In fact it is acted. A problem is started when the producer does not include this kind of know-how in the technical specification for user. Such situation steps out for the export objects like navy vessels equipped with gas turbine engines. During engine assembly, the rotating components are mounted with great care with the main objective of minimising shaft unbalance. However, even with the best of care, such factors as machining imperfection, differential thermal expansion etc cause a small residual unbalance of gas turbine rotor.

The dynamic problems of Marine Gas Turbine Engines (MGTE) are connected with such basic elements:

- rotors,
- bearings,
- struts of bearings,
- engine body,
- type of substructure,
- hydro- and meteorological conditions during sea trials,
- gas flown parameters inside the engine.

The quality of work process and stability of MGTE are connected with the state of such parameters as well. Dissipation of energy in rotating machines displays as a torque, revolutions, temperature, gas flown and vibration.

Vibrations are connected to:

- rotors unbalancing,
- oversize of tolerated axis slope of shafts,
- abrade of blade tips with the inner roller,
- wear of axis and radial bearings,
- asymmetry of springiness and damping characteristics of rotor and their parts and irregularity gas flown forces.

Emission of vibration brings a lot of information including opinion of the technical state. Measurements of vibration, their identification, classification, mathematical analysis, including trend function bring information on actual technical state and it allows predicting wear process in the future.

Every rigid body has six degrees of freedom, however a deformation body has unlimited degrees of freedom. Rotating machines like MGTE have an amount of degrees of freedom equal sum of all degrees of freedom engines' parts reduced by amount of rigid nodes connecting these elements of engine. Each part of engine can be represented by physical characteristics obtained from vibration measurements or from modelling of geometry and material – a rigidly joined structure. Application of specified model of rigid body gives ordinary differential equations. The deformation body needs partitive differential equations. The second assumption is much more complicated but it can approach to the real object, especially when it works in wide range of rotary speed. It was a reason of choice the second model. Scheme of diagnostics model MGTE is presented on Figure 1.

The residual unbalance occurs on each and all stages of rotor but two vectors of unbalance at both ends of shaft can represent effect of that. They have different values and phase shift. This FE individual and average model creates responses of unbalancing and it can be compare with vibration reports of measurements. The most sensitive unbalance response point at the GT engine is the front frame over a vertical strut. It is an effect of minimal thermal expansion

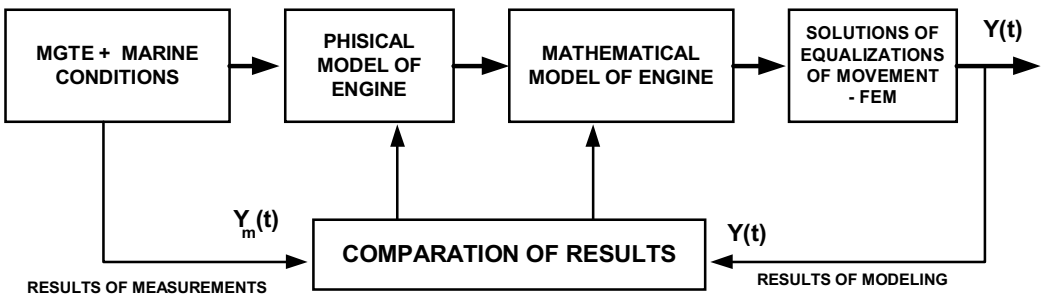

Fig. 1. Scheme of diagnostics model MGTE

for radial and axis vibration in this point. The model is linear so it is clear that response is directly proportional to the amount of unbalance [5]. It should be noted here that the real GT engine response is unlikely to be linear over wide range (revolution of shaft). Furthermore, the received effect can be accepted only as a statistical approximation of the dynamic engine response.

Construction of rotor is forced from different sources. There are not only unbalancing or gas-flown forces but vibration of vessel's hull, enclosure module, propeller, shaft and gearbox as well. List of potential sources is very long. Generally, there are axis slopes, crack of shaft, blade tip, crack or wane. These sources were focused during modelling and investigation on real object. Losses of material make an influence for changes of moments of inertia of rotated parts. They cause the displacement of main axis of inertia, which is not coincident to the axis of rotation. Finally, it is main sources of unbalancing – vibration of rotor. Mathematical model of this question is difficult because of assessments of damping and stiffness coefficients of struts and bearings.

Shape of the axis deflection is defined as discrete sets:

- set of static deflections – $\mathbf{u}_s$;
- set of dynamic deflections – $\mathbf{u}_d$.

Both sets depend on actual technical state of rotor and geometry, which are changed through cracks and wanes of engine parts

$$\mathbf{u}(\omega t) = \mathbf{u}_s + \mathbf{u}_d(\omega t) \tag{1}$$

This equation is a discrete set of displacement values points of axis of rotor. Taking into account damping and stiffness of bearing's supports, it can be posit that they are functions of the temporary positions, so:

$$k_{ik} = f(u) \quad c_{ik} = f(u) \tag{2}$$

Simplifying problem, it cans asses that for constant rotation these values are constant as well. Using FE method the model presents a three-dimensional discrete model. Rotors of MGTE because of circular symmetry have been described by one-dimensional, two hatches balk – rod symmetry FE which have six degrees of freedom. All of parts have geometrical and material characteristics.

Movement parameters of discrete model have been found by solution of following equation

$$\mathbf{Ku} + \mathbf{C}\dot{\mathbf{u}} + \mathbf{M}\ddot{\mathbf{u}} = \mathbf{F}(t) \tag{3}$$

where:

- $\mathbf{K}$ – matrix of structure's stiffness
- $\mathbf{C}$ – matrix of structure's damping
- $\mathbf{M}$ – matrix of structure's inertia
- $\mathbf{F}$ – vector of forces
- $\mathbf{u}, \dot{\mathbf{u}}, \ddot{\mathbf{u}}$ – displacement and their derivatives (velocity and acceleration)

The issue can be solved as a linear problem but in MGTE's rotor has to allow changes of stiffness and damping which are functions of movement's parameters. In this case equation (3) can be expressed as

$$\mathbf{K}(\mathbf{u}, \dot{\mathbf{u}})\mathbf{u} + \mathbf{C}(\mathbf{u}, \dot{\mathbf{u}})\dot{\mathbf{u}} + \mathbf{M}\ddot{\mathbf{u}} = \mathbf{F}(t) \tag{4}$$

Main purpose of researches had been found sensitive vibration symptoms represented residual unbalancing of rotors and forces from misalignment of shafts. FEA (Finite Element Analysis) are used successfully for a wide range of problems and it may also be used for the modelling and analysis of rotor system. Presently, diagnostics teams commonly use FEA and rotordynamics in conjunction with vibration analysis for detection and identification of unbalancing and misalignment of shafts. A linear model obeys the basic principle of linear superposition. Applied to a structure, this means that displacement resulting from a combination of structural loads is the sum of the displacements due to each individual load making up the combination. Unfortunately we had not enough structural information to create FEA model of marine gas turbine engines. It is a typical situation that the product of operation is made abroad. So, it was decided to apply passive method of investigation that to find reliable symptoms of unbalancing and misalignment using statistical methods and verified them by endoscopic examination. For initial analysis from first to fourth harmonics of amplitude of velocity of vibration, the dimensionless parameters S1 and S2 were taken as sensitive symptoms.

4. DIAGNOSING THE ROTORS UNBALANCING

The dynamic problems of marine gas turbine engines (MGTE) are connected with such basic elements: rotors, bearings, struts of bearings, engine body, type of substructure, hydro- and meteorological conditions during sea trials and gas flown parameters inside the engine. The quality of work process and stability of MGTE are connected with the state of such parameters as well. Dissipation of energy in rotating machines displays as a torque, revolutions, temperature, gas flown and vibration.

Vibrations are connected to:

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analysis, including trends, provide information on the actual technical state and it allows predicting wear process in the future.

Vibration analysis of MGTE during sea trials are accomplished by two different procedures:

1) on-line – in real time,
2) off-line – periodic or single measurements.

Both procedures have advantages and disadvantages. On-line system gives permanent control of vibration parameters in real time. It allows to monitor vibration parameters, holding memory and shutting down the engine in critical state. The data preview of memory can activate the trend functions and it shows changes of frequency or time parameters of vibration as a function of operational time. The disadvantages of this system are linked with costs because of software and hardware are stationary and sometimes individual. Our objects of researches did not have on-line monitoring systems and there are only four propulsion plants that it was a reason to apply periodical off-line diagnostic system.

For realisation of the investigations the measurement instruments: FFT-2148 analyser and PULSE v 9.0 software of Bruel & Kjaer, were used making it possible to collect and process measured data. Measuring transducers (accelerometers) were fixed to steel cantilevers located on the flange of the low-pressure (LP) compressor only. It was decided to carry out the investigations with the use of the transducer fixed to the LP compressor flange for lack of transducers and equipment suitable for measuring signals at the temperature as high as 200–300° C occurring on the high-pressure (HP) compressor flange.

The fixing accelerometers' cantilevers are characterised of a vibration natural resonance frequency value, differ enough from harmonic frequencies, due to rotation speed of the turbine rotors and their harmonics. The measurements were taken perpendicularly to the rotation axis of the rotors over main bearings. Such choice was made on the basis of theoretical consideration of excitations due to unbalanced shaft rotation, and results of preliminary investigations of the object [5]. As signals, usable for the „defect-symptom" relation, the following magnitudes were selected by the turbines' producer:

Ysnc – 1st harmonic RMS value of vibration velocity amplitude connected with the LP rotor of compressor;
Yswc – the same but connected with HP rotor of compressor;
Yrms – RMS value of vibration velocity amplitude within the range of 35–400 Hz.

The choice was justified by the time-between-repair values scheduled by the turbines' producer. For the purpose of these investigations a simplification was made consisted in assuming values of the after-repair turbine vibration symptoms as those of the new turbine. To make such assumption was necessary due to rather low number of the investigated objects (only eight turbines of each type). The turbines' producer specified the following limit values of RMS vibration velocity amplitude (Tab. 1).

In order to obtain uniform diagnostic procedures regarding unbalance assessment of the turbine rotors the dimensionless parameters characterising that states were applied.

On the basis of theoretical considerations as well as results of other diagnostic investigations carried out for some years the following parameters were selected as the most sensitive:

S1 – ratio of the mean vibration velocity amplitude of a given rotor (1st harmonic) and the velocity component relevant to 2nd harmonic excitation frequency of the rotor in question;
S2 – ratio of the mean vibration velocity amplitude of a given rotor (1st harmonic) and the velocity component relevant to 3rd harmonic excitation frequency of the rotor in question.

During operation the limited values of dimensionless parameters of low and high-pressure rotors were assessed (Tab. 2).

One of the most important elements of the off-line system is database. For further consideration idle load and full power (nominal load) were taken. Each spetrum were transfered as matrixes M_S and coppied like fingerprints to the database. Spectras were not synchronized to the revolutions of rotors, at the same loads in different air temperature conditions vary each other, it caused the preparation at the the

Table 1. Limit values of RMS vibration velocity amplitude

	Permissible value of Yrms	Permissible value of harmonics Y
DR 76 engines	24 [mm/s]	17 [mm/s]
DR 77 engines	30 [mm/s]	20 [mm/s]

Table 2. Values of dimensionless parameters of low – (SNC) and high-pressure compressor rotors (SWC)

	$S1_{SNC}$	$S2_{SNC}$	$S1_{SWC}$	$S2_{SWC}$
DR 76 engines	min 1.0	min 1.5	min 1.0	min 1.5
DR 77 engines	min 1.2	min 1.5	min 1.1	min 1.3

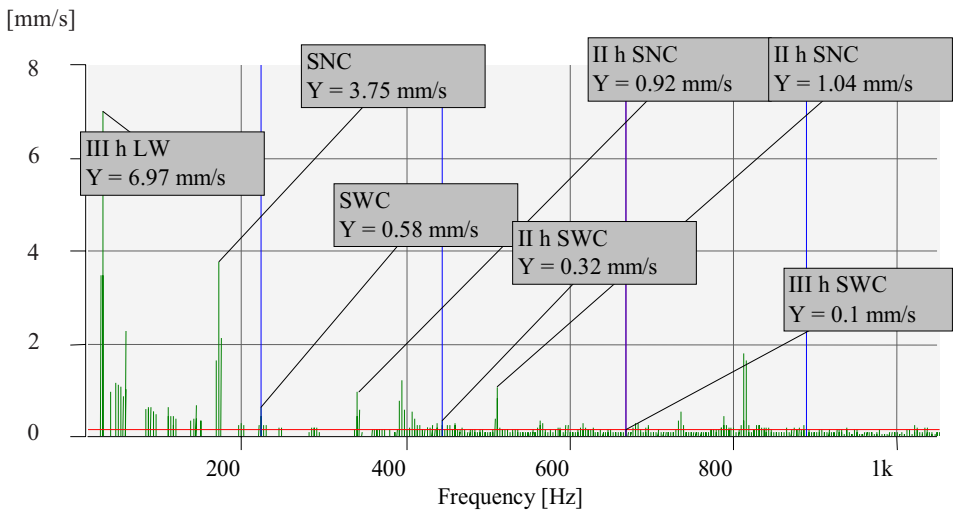

Fig. 2. Typical patern spectra DR 77 engines, where: SNC – low-pressure compressor rotor, SWC – high-pressure compressor rotor, h – harmonics

procedure of identyfication as follows: Measured spectras had 2 kHz probe frequency and 800 lines, so sensitivity was ±150 rpm. During tests changes of revolution of rotors varied ±230 rpm, so it was decided to make patern spectras using 400 lines for both loads. It appeared as an important point of analysis because of sensitivity of typical spectra (Fig 2).

Next, collected spectra were compared using a cross-coherent method for recognising difference between previous and present harmonics and/or bands of frequency. This procedure enables, in short time, to recognise changes of technical state and collect information to the database [6]. Typical analysis using cross-coherent method are presented in Figure 3.

For recognising the residual unbalancing only subharmonics and first four harmonics were taken into acount. When values of previous and next spectra coherention approach values under level of $\gamma = 0.8$ then founded symptom was verified as a trend function of harmonics – Figure 4a and S1 and S2 symptoms – Figure 4b.

Two-way realisation of the investigations made reliable verification of the investigation results possible.

The following detail conclusions were drawn for further diagnostic inference:

- for DR 76 engines: Ysnc vibroacoustic parameters are diagnostically susceptible at the engine load $N = 1.0$;
- for DR 77 engines: Ysnc vibroacoustic parameters are diagnostically susceptible at the engine load $N = 1.2$;

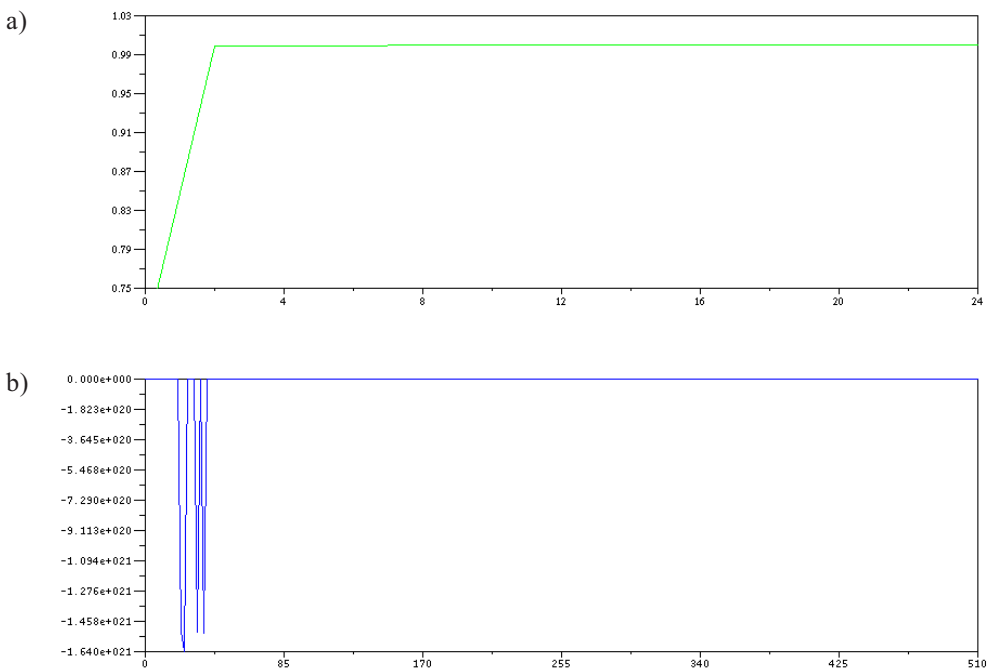

Fig. 3. Typical cross-coherent analysis two serial spectra: a) values of coherence; b) difference between first and second spectra

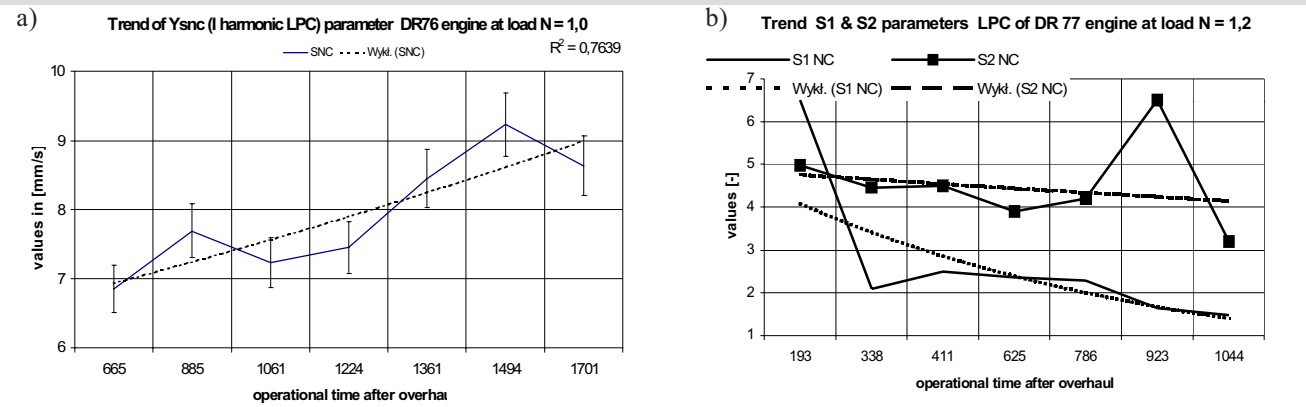

Fig. 4. Typical expotential trend of the I harmonic low pressure compressor, where NC – means LPC (a); typical exponential trend of the S1 and S2 parameters of LPC DR 77 engines, where NC – means LPC rotor (b)

– changes of 1st harmonic values connected with HP compressor rotors (Yswc) and LP ones (Ysnc) at the work of DR 76 and DR 77 engines at idle load are hardly noticeable in function of operation time therefore their operational susceptibility is too low.

DR 76 and DR 77 gas turbine engines are installed in the considered M – 15 E service propulsion system. The load $N = 1.0$ and $N = 1.2$ of engines was assumed the criterion for determining their maintenance time basing on exceedance of permissible values of the considered symptoms at normal engine operation.

5. DIAGNOSING OF SHAFTS MISALIGNMENT

Usual measurement methods of coaxiality parameters of the propulsion system require disassembling protection covers of shafting between engines and reduction gears. Measurement conditions make it necessary to suspend operation of the COGAG-system for about 8–10 days and it is of course intrusive method. A vibroacoustic method presented in this paper allows assessing permissible values of the alignment parameters without stopping exploitational use of the vessel. Moreover, the presented results are intended to form the database for elaboration of an on-line monitoring system of coaxial of torque transmission elements, applicable to the COGAG propulsion system in question.

Dynamic reactions, resulting from exceedance of allowable alignment deviations of the torque transmission elements are able to cause failure of the propulsion system and even lead to loss of movability of the vessel in a relatively short time [2]. Therefore diagnostic control of the gas turbine power plant in operation became necessary.

Appropriate assembling the main engines and the other torque transmission elements inclusive of propellers is practically determined by a set of tolerated dimension and geometrical location requirements, called geometrical dimension assembling chain [3]. Both typical and modular power plants are prone to coaxiality deviation from its permissible values and in consequence to possible failures one or more elements of the propulsion system. The excessive deviation can lead to the loads on bearings and gear teeth much higher than calculated and in result to their premature failures [4].

The usual control methods of the coaxiality deviations do not fulfil user's expectations in the case of the gas turbine propulsion system. Difficult access to flange connections, long control time, organisational difficulties and lack of qualified personnel create hazards of taking measurements with errors exceeding allowable values. The proposed vibroacoustic method instead of the usual coaxiality control methods was assumed the aim of this work on result of an analysis of the earlier mentioned operational hazards.

Application of the vibroacoustic diagnostics to technical maintenance makes its possible to lower operational cost of the vessel by basing its operational on its actual technical state and predicted failure states [1].

It was assumed that determination of a relationship between the coaxiality parameters and changes of the recorded vibration signals should bring about identification of the proposed diagnostic model consisting in:

– choice of geometrical parameters describing the position deviations, i.e. axis slope and displacement;
– choice of adequate parameters of the vibroacoustic signal;
– determination of a mutual relationships between sets of the coaxiality deviations and vibration diagnostic parameter values;
– sensitivity assessment of the symptoms in question;
– establishing the database for statistical analysis and operational decision making.

The research in question was limited to control of the axis slope only. This assumption was made to account for influence of the displacement hull deflection on the element position of the serial multi-shaft system. In this case the axis displacements are controlled solely during assembling the shafting system in the production and repair stages.

The energy emitted in result of a change of technical state of the flange connection was assumed to be reflected in the recorded vibration signal. By using experiments, position and fitting direction of accelerometers were chosen at

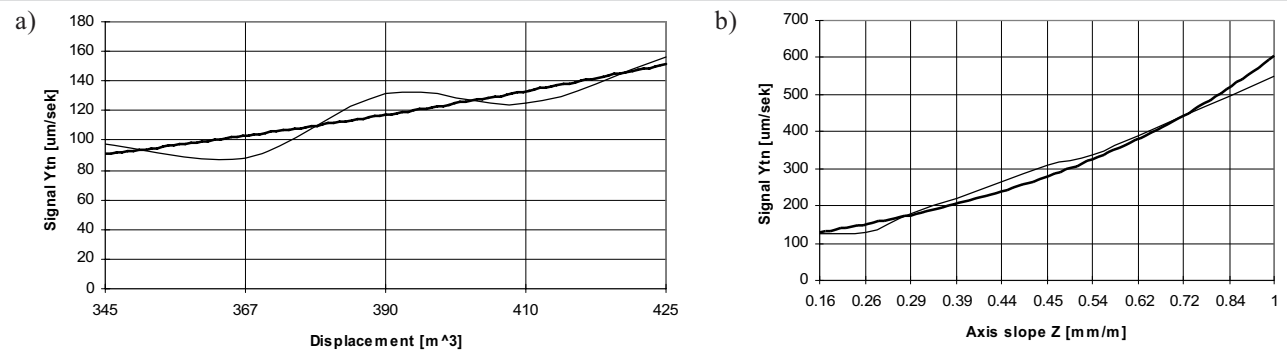

Fig. 5. Symptom change regression functions at the different displacement values of the vessel and different axis slope values (a), Symptom change regression functions at the different slope values of the vessel's displacement of D = 410 m^3 (b)

the transverse cross-section of the transmission gear just over the main, radial bearing of the input shaft. On the basis of detail identification of the main signal against disturbances the lateral measurement direction with respect to the rotation axis was determinated. The coaxiality measurements were carried out in the conditions of not transmitting the torque to the ship propeller. In Figure 5 examples of investigation results are presented in the form of symptom change regression functions at the different displacement values of the vessel and different axis slope values. The results were approximated with the use of the least square method. The results were analysed by assigning the mean value of the vibration signal, i.e. the harmonic connected with the excitation frequency of vibration velocity amplitudes, to the measured axis slope value. Archive charts of the investigation results are exemplified in Figure 6.

During the investigation no limit state of the axis slope of Z = 1 mm/m was found in the propulsion system in question. A regression function of the symptom changes at different vessel displacement was calculated to determine the limiting (tolerated) values of the symptom. In result three conclusions dealing with general measurement principles could be elaborated as far as the investigated vessel is concerned.

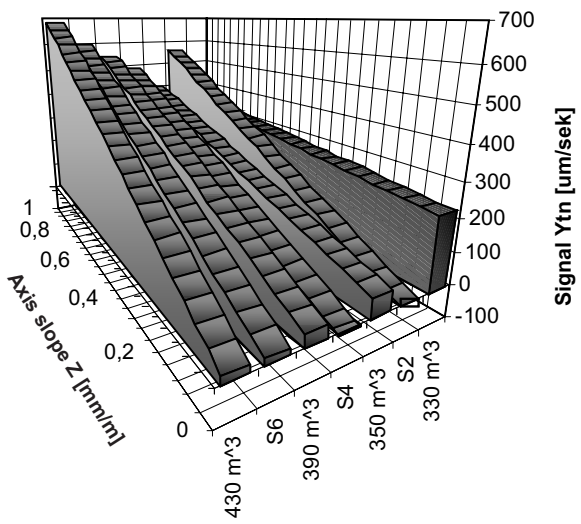

Fig. 6. Exemplified archive charts of the investigation results

It is important to determine the limiting values of the selected symptom at different vessel displacements. The proposed method can be applied in the entire range of the vessel's displacement except of D = 340 m^3. At this displacement the sensitivity of the symptom change in function of axis slope is too low.

Assessment of axis slope at the flexible coupling on the basis of the symptom in question is of a secondary importance.

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Assessment of axis slope at the flexible coupling on the basis of the symptom in question is of a secondary importance.

Selection of the accelerometer fitting position is the most important because of close vicinity of the propulsion (reduction) gear main radial bearing. Accelerometer cantilevers applied during the measurements did not influence measurement results, as their resonance frequencies were higher than that of the selected vibration symptom.

It was also stated, of the basis of analysis of the investigation result, that the mean signal value Ytn increases as the vessel's displacement increases, at the axis slope Z = const.

6. FINAL REMARKS AND CONCLUSIONS

- Application of the proposed approach makes managing the engine's operation time much more rational, especially at its end.
- The proposed approach is non-invasive and does not require taking the ships out of service.
- Realisation of investigations of the kind makes it possible to collect data for a database of the future monitoring system of ships wich is expected to improve their operational features.
- Experience gained during the investigations would be utilised for other power plants equipped with gas turbines.
- The proposed diagnostic method is a coherent element of Basic Diagnostic System used by Polish Navy for many years.

– The proposed exploitation method leads to important economical profits and especially to reliability improvement, a first-rate problem.
– Presented method is enough sensitive in operation of propulsion plant that it enables to find primary symptoms of changes technical state of rotordynamics.
– The regression equation of trend is good rule for verifying the sensitivity of symptoms.
– Cross-coherent procedure is useful tool for fast recognising differences between spectra in database.

1) The proposed measurement method makes it possible to determine the limiting value of Ytn symptom witch, if exceeded, indicates the inadmissible axis slope value and, more-over, it provides an unambiguous relationship between the axis slope and the vessel displacement values.
2) Implementation of the method to routine operation of the ship power plant could lead to important reduction of maintenance scope on the condition of extending the diagnostic control to all propulsion connections.
3) The proposed method could also adapted to diagnosing other torque transmission components of similar propulsion system.
4) At this identification stage of the object the postulated realisation of an on-line state transducer concept seems to be rather difficult and needs further research as dynamic loading of the ship hull and propeller shaft. Thermal stresses from the side of the propulsion gas turbine as well as other rotational speeds of the connecting shaft have not been accounted for so far.
5) The on-line monitoring system for early detecting the symptoms of exceedance coaxiality allowances of selected components of the ship gas turbine propulsion systems. It should be assumed the final results of further investigation in site of occurrence of many disturbances of measurement signals as well as the limited control dependability of the gas turbine propulsion systems.

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