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DIAGNOSIS OF GAS TURBINE ENGINES ROTORS SYSTEM IN NONSTATIONARY STATES

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Abstract

Vibration tests of marine gas turbine engines are performed as research of on-line and off-line types. On-line Systems generally monitored one or two vibration symptoms, which asses the limited and/or the critical values of parameters and they, potentially, can warn and/or shutdown engines. Off-line Systems are usually used for vibration analysis during non-steady state of work. The paper presents comparison of different methods of analysis of vibration symptoms measured under run-up and shut-down processes of marine gas turbine engines. Results of tests were recorded on gas turbine engine DR76 type of the COGAG type propulsion system. Main goal of the research was qualified on helpfulness and unambiguous result, from synchronous measurement, order tracking and auto tracking. All vibration symptoms were chosen from the methodology of the diagnosing gas turbine engines operated in the Polish Navy, called Base Diagnosing System. Second purpose of the paper was the estimation of the possibility of usage those analysis methods of gas turbine engines for on-line monitoring systems.

Ke words*: dynamics, gas turbines, rotor vibration, run–up process*

1. Introduction

Operation of marine propulsion systems is a complex issue due to the specific characteristics of the marine environment and the need to maintain a high level of readiness for service and reliability of ships. The use of diagnostic procedures off-line or on-line allows you to use them according to their current condition. This is particularly important in the case of turbine engine, hourly plan and annual plan of technical services is the main usage criteria. This strategy of exploitation makes scheduling maintenance, logistics and security simpler and easier to implement, but also contributes to a significant increase in costs due to the need for replacement of components (often more technically efficient). Furthermore, operating such a exploitation policy makes it impossible for the early detection of other primary causes of faults that occur between appointing terminals.

Diagnostics of gas turbine engines includes a wide range of parameters, controls and maintenance procedures [1]. One of them is the control of unacceptable balance of rotors. Identification of different unbalanced states, determining its value and the accurate placements of corrective masses is commonly known. Such procedures are carried out on Polish ships for over 20 years. Prepared and used test equipment ensures the implementation of diagnostic tests on four types of turbine engines in service. In the case of naval propulsion diagnostic procedures these are limited for several reasons. The most important of these is the need to maintain a constant readiness to start the engine, associated with the tactical requirements. In addition, due to the fact that the engines are foreign construction, there is a lack of information on the structural parameters of the engine, reducing warranty, no spare parts readily available, etc. The use of vibration diagnostics, makes the use of the engine more rational; from a technical point of view, especially towards vitality of service, which in effect will not withdraw, even a technically efficient ship, from service. Measurements and analysis of vibration parameters of marine gas turbine engines can be divided into:

- off-line (measurements performed in free-run mode, periodically);
- on-line (real-time monitoring).

Both methods have their advantages and disadvantages. Off-line Systems are usually offered as a very simple analyzers - data collectors. Measurement path is determined in the collector interface, with preset measuring settings, so that the measurement could be performed by an average technical staff, whose main task is a precise procedure. The analysis of measurement results is carried out of the ship, sending the results to the coast laboratory. Currently, there is not many off-line data collectors, who would engage in that precise diagnostic evaluation. The main advantage of such devices is their price. It should be emphasized that the data collectors are useful mainly to assess the go-state of vibrations of turbine engines.

On-line diagnosis of vibrations provides continuous surveillance of the technical condition of gas turbine engines, including registration, analysis, forecasting and alarming. It allows you to recognize the basic signs of changes in the technical condition with the possibility of analyzing the trend of selected symptoms. On-line vibration systems usually work as part of a complex and symptomatic diagnosis of marine propulsion systems. Proper diagnosis of such structures, for example, turbine engine, depends on various issues, including how the measurement and processing of vibration signals was taken. Important in the further analysis is the fact that internal combustion engines in gas turbine propulsion ships do not run at a constant speed with compressor and turbine rotors.

This is the main reason for synchronizing the processing of selected displacements (of the signals) i.e. the rotational frequency of one or both of the engine rotors [2,3]. This method allows you to identify the most common groups of rotor systems, which allows you to identify their failure. Damages to operating gas turbine engines can be categorized as follows:

- damage or crushing of first-stage compressors' blades or power turbine blades (rare);
- the appearance of unbalance, originating from heating or salinity;
- cracks sealing systems and leakage of lubricating oil to the inside of the drum rotor;
- lack of alignment between the gas-dynamic gas generator and power turbine;
- thermal damage to the combustion chambers – torsion of power turbine rotor;
- damage to the auxiliary engine mechanism.

Some failures can be resolved in the recorded spectra as a change in vibration frequency of rotating engine components, hence the introduction of a synchronous sampling of the transient engine operation, eg in the boot process or in the run.

The occurrence of non-stationary effects, typical for residual unbalance may be due to small, incremental damage whose symptoms may be poorly recognized in the early stages of development. The results of the identification of such phenomena is exemplified in the article comparing the various methods of synchronous signal processing method such as PLD or Order Tracking [7]. The presented method for identification of defects can be introduced into the turbine engine monitoring systems as a tool for early identification of unbalance.

2. The aim and test methods

Monitoring of vibration signals from rotating machinery is a well-known diagnostic procedure, known throughout the world [2,5,7]. Most of rotating machinery and marine gas turbine combustion engines are designed as a supercritical machines, hence, in steady states, are diagnostically limited. Therefore it was decided to analyze the dynamics of rotors of gas turbine engines, using a method of off-line measurements of the unknown states It was expected that the results would yield information on the following areas: unbalance of rotors, lack of concentricity of the rotors, changes in their vibration frequency and changes in the speed of rotor system critical.

Marine gas turbine combustion engines mounted on a DR76 type of propulsion system for ships COGAG class Tarantula Polish Navy were studied using this method. Longitudinal cross section of rotor system is shown in Figure 1.

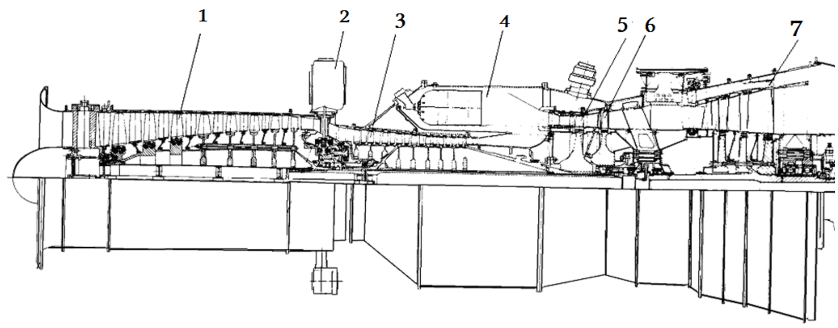

Fig. 1. Longitudinal section of rotor system gas turbine engine DR76 type, where: 1 – low pressure compressor (LPC), 2 – auxiliary drives, 3 – high pressure compressor (HPC), 4 – burning chambers, 5 –high pressure turbine (HPT), 6 – low pressure turbine (LPT), 7 – power turbine (PT)

The study included analysis of the vibration parameters during start-up and run of rotors. Comparison of the results of modeling of dynamic loads using FEM (Final Elements Methods) and measurements of on the real object makes it possible to take correct decisions and give the proper diagnosis

3. Model of the unbalanced rotor

Application of computer simulation to diagnose the condition of turbine engine rotors should be used already during the process of calculation and design, which it is currently implemented. The problem begins when the manufacturer does not provide this kind of know-how in the technical specification for the user. Such a situation arises in the case of exported warships equipped with turbine engines. While placing the engine, rotating parts are assembled with great care. Main objective is to reduce unbalance in rotors. But even the best procedures are not able to prevent factors, such as the inadequacy of heat treatment or the difference of thermal expansion of materials which may cause slight unbalance in rotor, mentioned as residual. Problems in the dynamics of Marine Gas Turbine Engines (MGTE) are associated with the following elements of the engine: rotors, bearings, bearing brackets (bearing struts), engine block, the type of construction, the terms of hydro-meteorological and during sea trials and the aerodynamic parameters inside the engine. Proper and stable work of MGTE engine is mainly connected with these parameters. Loss of energy in rotating machinery is manifested in the form of loss of torque, a decrease in rotor speed, exhaust temperature increase or intensity in vibrations. Vibration energy dissipation is related to: unbalancing of rotors, oversize tolerated shaft misalignment, abrade of blade tips with the inner roller, wear of axis and radial bearings, asymmetry of elasticity and damping asymmetry of the rotor and the gas-dynamic processes anomaly. Emission of vibration yields a lot of information, including the ability to diagnose the technical condition of rotors. Vibration measurement, identification, classification, mathematical analysis, including the use of trend function, give information on the actual technical state and allow the prediction of the wear process in the future.

In the identification an important factor is to compare the results of modeling with the results of the measurements. Each rigid body has six degrees of freedom, whereas the deformable objects have an unlimited number of degrees of freedom. Rotating machinery such as MGTE have a number of degrees of freedom equal to the sum of all degrees of free parts of the engine, minus the number of rigid nodes connecting these elements. Each part of the engine can be described by physical characteristics such as stiffness and damping, obtained from vibration measurements the actual object or model or the modeling of the geometry and properties of materials (the use of rigidly connected structures). The use of a certain type of rigid object model allows the use of the motion ordinary differential equations. Deformable objects require the use of partial differential equations. This second assumption is much more complicated, but can help to achieve to the actual object, especially when it's in a wide range of engine speeds. This was the reason for the choice of the second type of model turbine engine. Diagram of diagnosis using the MGTE model shown in Figure 2.

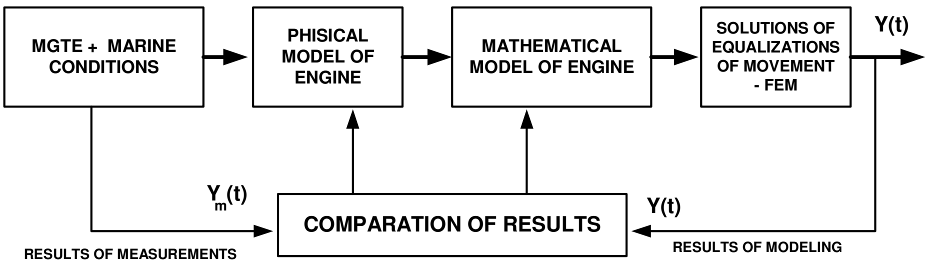

Fig. 2. Scheme of diagnostics model MGTE

Residual unbalance may appear in all sections of the rotor, however, two vectors of unbalance, at both ends of the shaft, may represent the replacement model. These vectors vary in values and phase shifts. Such an FE model allows for dynamic response to unbalance which in effect allows you to compare modeling results with the reports of vibration measurement. The most sensitive point in the unbalance of GT rotor, with respect to vibrations, is the measuring point on the front of the generator exhaust bracket bearing the vertical direction. This is the effect of the minimum thermal expansion of the rigid support used for measurement of radial vibrations at this point. The model is linear so it is clear that response is directly proportional to the value of unbalance The rotor is loaded dynamically and statically from various sources [4].

Identification of the sources and their calculations of the loads were a major problem during the modeling and evaluation of the actual object's vibration. Damage in the objects such as blades, have an impact on changes in the moments of inertia of rotating parts. This results in a shift of the main axis of inertia, which is not parallel to the axis of rotation. It is the main source of unbalance in the form of vibrations of rotor. Implementation of the mathematical model is difficult, mainly due to the problems of determining the stiffness and damping of supports and bearings at different temperatures - Figure 3. Shape of the axis deflection is defined as discrete sets:

- Set of static deflections – u_s;
- Set of dynamic deflections – u_d.

Both sets depend on actual technical state of rotor and geometry, which can change through cracks and wanes of engine parts.

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

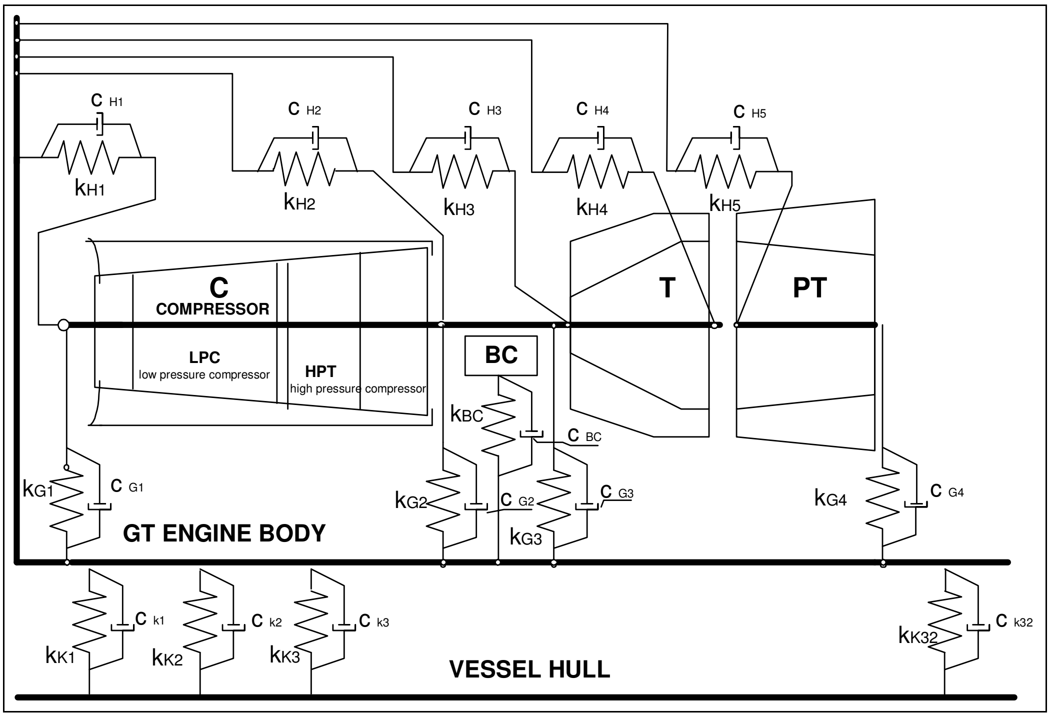

Fig. 3. Axi – symmetric lumped mass inertia model of the MGTE, where: LPC – low pressure compressor, HPT – high pressure compressor, T – turbines (low & high pressure), PT – power turbine, BC – burning chambers, k – stiffness, c – dumping

This equation is a discrete set of points of axis movement of the rotor. Taking into account the damping and stiffness of the support bearings, we can demand that they are functions of temporary positions, namely:

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

For the simplification it is assumed that, for a constant speed, these values are constant. Using FEM modeling can provide a three-dimensional discrete model. Rotors MGTE, in the circular symmetry, have been described by one-dimensional, two-beam bar having a symmetrical six degrees of freedom. All parts of the model have geometric and physical properties of the elements. Discrete model of traffic parameters have been obtained by solving the equation:

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

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

This can be solved as a linear problem, but in MGTE rotor must allow for changes in stiffness and damping, which are functions of motion parameters. In this case equation (3) should be expressed as:

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

Equation (4) indicates that the rotor motion should be described as a nonlinear dynamic problem, and therefore should expect more than one harmonic in both measured and modeled spectrum. [8]

4. Non – steady states vibration signals analysis

To obtain the measurements of the real object Bruel & Kjaer 3560B analyzer was used. Namely, it was used during the collection and processing of measurement data using the PULSE(v.12). Two transducers (accelerometers ICP) have been fitted to the steel girders, situated on the flanges, on the front and on central pillar of the LPC. The fixing cantilevers are characterized by vibration resonance frequency value differing from harmonic frequencies due to rotation speed of the given rotors. Measurements were made perpendicular to the axis of rotation of the rotor. Such a choice was made on the basis of theoretical analysis of unbalance and as a result of analysis of the results of preliminary research on the subject.

Common assessment of the unbalance of rotors was developed through the concept of dimensionless coefficients of diagnosis. Using theoretical analysis of dynamic interactions, as well as using the results of initial diagnostic tests, the following symptoms were selected as the most sensitive to changes in balancing rotors [2]:

- First harmonic of amplitude of the corresponding velocity of the rotor,
- Second harmonic of amplitude of the corresponding velocity of the rotor,
- S 1 - the ratio of the average amplitude of vibration corresponding rotor speed (and harmonic) and the second harmonic of the corresponding rotor,
- S 2 ratio of the average amplitude of vibration corresponding rotor speed (and harmonics) corresponding and the third harmonic of the rotor.

These symptoms can confirm the theoretical assumption of nonlinear rotor dynamics.

5. Vibration analysis of the run-up process

The first test was to analyze the process of starting the engine. The characteristic changes in LPC rotor speed is shown in Figure 4.

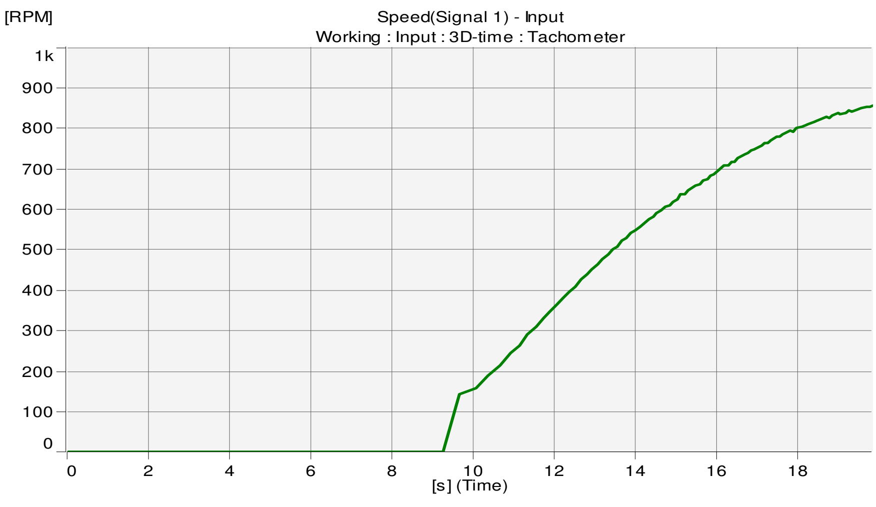

Fig. 4: Rotors LPC rotational speed characteristics during run-up process

Synchronous signal measured by a tachometer connected with the auxiliary drive gear box where the transmission ratio averaged on *i*=0,125, so the LPC rotor was 8 times greater (in speed) than that shown in Figure 4 The main objective of the analysis of synchronous oscillations in the boot process was to determine the dynamics of the disorder. The impact of "other" signals is shown in Figure 5.

The boot process started at the point t = 7 seconds (see Figure 5), so all recorded vibration signals recorded from the start point contained the signals coming from other sources, i.e. non-rotating motor or frequency of its vibrations or a combination thereof. This allows to identify the main "other" signals, such as: f_1 = 305 Hz, f_2 = 600 Hz, f_3 = 1.6 kHz, and f_4 = 2 kHz. which are associated with sources outside the engine.

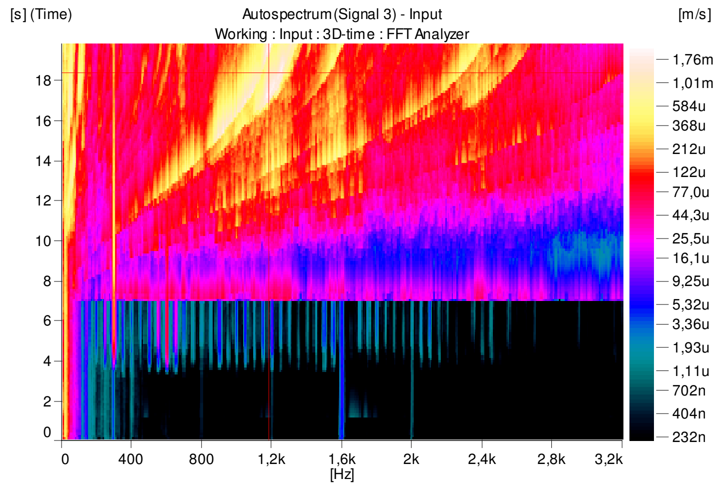

Fig. 5. Synchronous spectra of the velocity of vibration during run-up process with using the band – pass filter of 0,1Hz – 3,2 kHz range

The highest signal during the boot process is the rotor speed and harmonic vibrations, but in Figure 5 it is not clearly visible due to the lack of a synchronous signal tracking.

6. Vibration analysis of the shut-down process

Next test was associated with the analysis of vibration parameters and related to the process runs the motor rotor. Figure 6 shows autospectrum of the velocity measured over the middle LPC bearing using the order tracking procedure. Changes of parameters are presented in the domain of time function, in contrast to the boot process ,where the dominant energy range of vibration signal was 1 / 2 harmonic - seen as a 4th order. The pressure drop of the lubricating oil in the bearing caused an increase in values ranging from displacement and slope between the HPC and LPC rotor (rotating shafts each other, while the shaft rotates within the LPC HPC shaft - see Figure 1) and the typical dominance of the subharmonics .

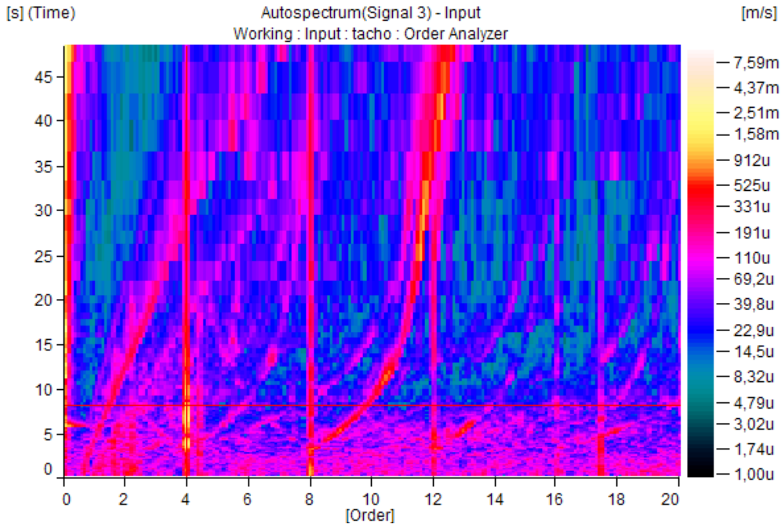

Fig. 6. Autospectrum of velocity of vibration in the shut-down process with the use of order tracking procedure, in the domain of time function

The increase in stiffness of the bearing system confirms the existence of the harmonic “right-hand branches” at the point where t (time) is equal to 4 seconds for the following rows: 4, 8 and 12, which is associated with a pressure drop of lubricating oil in the bearings.

Analysis of the dynamics of the turbine engine rotor in transient states of a system PULSE should be applied in both processes, ie start-up and run. The start-up process helpsto recognize the "other" signals, but the definition of dynamic functions is very difficult due to the significant acceleration of the rotors. Identifying characteristics of rotor system dynamics is much more recognizable in the process runs through the analysis of orders - Figure 7 and 8.

Analysis of the first harmonic (8th order) allows to observe changes in dynamics as trends. Application of the rotational speed function as a field of analysis is the most important factor in the study of the use of the Order Tracking procedure. This allows you to detect changes in the natural frequency, ignoring interference from the signals originating from the thermodynamics processes of turbine engines.

Subharmonics signal analysis is very useful in the diagnosis of rotating machinery. Autospectrum of ½ subharmonic's velocity range (considered in the LPC rotor) indicates the individual characteristics of particular rotors. The nature of changes in order values in the rotor speed can be thought of as an individual fingerprint of each rotor.

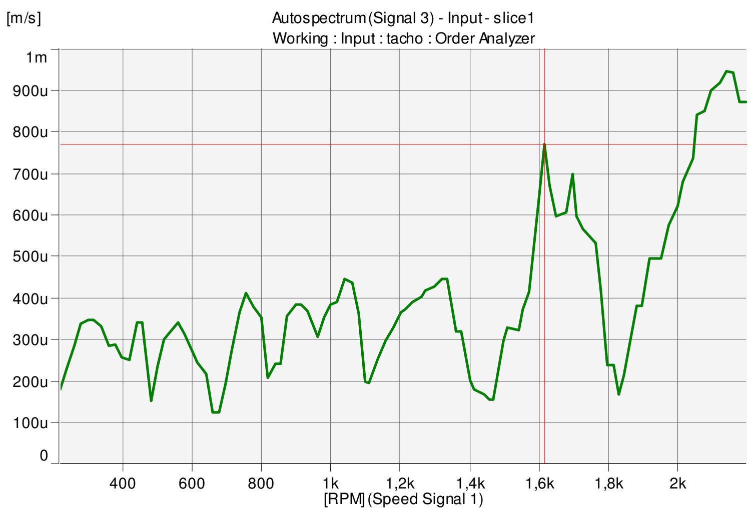

Figure 7. Autospectrum of 8 order (I harmonic) of velocity of vibration in the shut-down process of LPC rotor stoppage

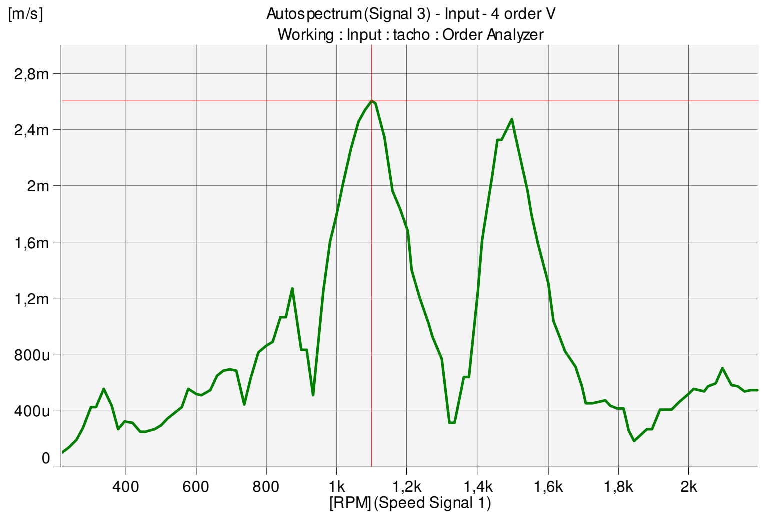

Fig. 8. Autospectrum of 4 order (subharmonic) of velocity of vibration in the shut-down process of LPC rotor stoppage

All changes to the technical condition of rotor system, such as changes in stiffness and damping parameters of alignment, or unbalance result in changes in characteristics of subharmonics - Figures 7 and 8.

Conclusions

All statistical analysis performed on the available population of engines clearly show that the selected parameters analyzed in the non-stationary processes are the basis for predicting changes in the technical condition of rotor system. Implementation of this research turns out to be a credible verification of the technology. Conclusions presented below have been incorporated into operational diagnostics of marine gas turbine engines:

- synchronous measurement of vibration signals during the boot and run processes enables us to recognize symptoms of damage, including the formation of resonance and changes in natural frequencies and unbalanced rotors
- symptoms of S1 and S2 do not have sufficient sensitivity for use in transient states due to the instability of the processes and the need for averaging the results,
- application of auto tracking and monitoring the turbine engine rotor systems can identify a wide range of typical damages, confirmed by the vibro-acoustic diagnostics.

Application of the proposed methods of analysis allows for the rational management of engine life time even in the developed processes of consumption. The analysis of test results obtained gives the following conclusions:

- the approach to the assess the technical condition of gas turbine engines rotor system allows to quickly detect changes in the permitted unbalance and the maintained database enables easier identification of the studied group of engines
- studies on trends in chosen parameters make it possible to reliably detect changes in the value of sensitive operational parameters during the operation of the engine and to evaluate its capabilities.

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