VIBRATION DIAGNOSTICS OF MARINE GAS TURBINE ENGINES

Andrzej Grządziela

Polish Naval Academy, Mechanical – Elelctrical Faculty
Śmidowicza Street 69, 81-103 Gdynia, Poland.
phone: +48 58 626 27 24
e-mail: a.grzadziela@amw.gdynia.pl

Adam Charchalis

Maritime University of Gdynia, Mechanical Faculty
Morska Street 81 - 87, 81- 225Gdynia, Poland
phone: +48 58 6218-997, fax: +48 58 6218-997
e-mail: achar@am.gdynia.pl

Abstract

Vibration tests of marine gas turbine engines are performed as researches on-line and off-line types. The paper presents analyses of both methods. Results of tests were received on three types of gas turbine engines operated in the COGAG type propulsion systems. The application of periodical diagnostic procedures or on-line monitoring systems makes it possible to operate ship propulsion systems in accordance with their current technical state. In the case of ship gas turbines the hourly period of scheduled maintenance or repair surveys is presently the criterion for maintenance time determination. The Finite Element Analysis (FEA) is used for confirmation actual technical state. FEA are used successfully for a wide range of problems and it may also be used for the modelling and analysis of rotor system. Presently, the diagnostics team uses FEA and the modelling of rotordynamics in conjunction with vibration analysis for detection and identification of unbalancing. Main goal of researches was qualified of helpfulness and unequivocally results received by methods of 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 BDS (Base Diagnosing System). This element of BDS is accepted and used in all ships of Polish Navy, which are powered by the COGAG power plant. Second purpose of researches was estimation of the possibility of implementation presented methods of vibration analyses of gas turbine engines fornew, modern on-line monitoring system. The proposed diagnostics method makes it possible to determine the limiting value of vibration symptoms witch, if exceeded, indicates the inadmissible axis slope value between rotated machines of gas turbine engine and, more-over, it provides an unambiguous relationship between the value of symptom of rotors unbalance and the rotors rotational speed.

Keywords: *Dynamics, Gas Turbines, Rotor Vibration*

1. Introduction

Diagnosing the ship gas turbine engines covers a wide range of service parameters and maintenance operations [1]. One of them is the supervising of allowable unbalance of rotors. Identification of various unbalance forms, its values and fitting places of correction masses, is commonly known. Such tests have been carried out on Polish Navy ships for more than 20 years. The Diagnostic Unit for Gas Turbine Engines has realized the tests on the engines of three types. In the case of combat ship's power plants, the diagnostic procedures are limited for a few reasons. The most important is the necessity of being still ready to start the engine, connected with tactical technical demands. The other ones are associated with a lack of information on structural parameters, guarantee limitations, incomplete amount of spare units etc.

Application of periodical or on-line diagnostic procedures makes it possible to operate ship propulsion systems in accordance with their current technical state. Especially, in the case when gas turbines engine maintenance schedule is a criterion 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.

Application of the vibration diagnosing makes managing the engine's operation times much more rational, especially at its end and does not require taking the ships out of service. It is possible to divide vibrations measurements of marine gas turbine engines in practice on:

- *off-line* type (measurements made as a free run or synchronous sampling),
- *on-line* type (real time monitoring).

Both methods have their own advantages and disadvantages. *Off-line* type systems are usually offered as very simple analyzers – data collectors. Defined measuring path allows in a ships power plant, with programmed setups of registration, use of average technical personnel, whose the main task is precision in measuring procedure. The main advantage of this kind of devices is the prize, but it is necessary to underline that data collectors are mainly useful only for assessing the vibro-activity of gas turbine engines. Structure of vibration diagnosing GTE, including both methods, is presented in the Fig. 1.

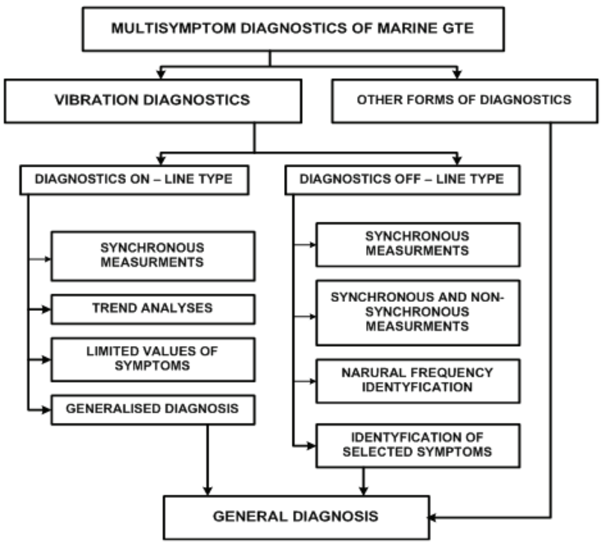

Fig. 1. Structure of vibration diagnosing GTE

The vibration, diagnostics systems *on-line* type, often called the monitoring module, ensure permanent supervision of the technical state of gas turbine engines, including recording, analyzing, alarming and predicting. It allows recognizing primary symptoms of the permanent, technical state changing with possibility of the trend remarks.

2. Simulation of dynamics GTE rotors system

Dynamic model of rotor system should be as simple as possible, and can be used to simplify the time when it is still sensitive to interesting phenomena. The main factors are taken into account turns out: damping, flexibility and torsional stiffness of supports. Inclusion of damping effect is necessary when the model to the classical diagnostic tasks, a continuous like: *mapping ↔ symptom state* in a quantitative manner and not only qualitative. A consideration of the elastic supports is not always necessary because it is observed the object vibration components or enclosures elements in the interested direction. In addition, the stiffness of these elements is usually many times greater than the stiffness of the rotating elements and inner damping is so small enough that the vibrations move without significant changes in frequency domain. The problem of mapping observations is to find spectral transmittance in the frequency domain [4].

The necessity to take into account the flexibility of mounting elements occurs only when it is so large that it affects in a qualitative way on the dynamics of the system.

Likewise, the torsional rigidity of a single of rotor system. The introduction of this parameter is necessary, when the model of GTE is a multi – masses system. Therefore it is observed difference in angular velocities of each rotors disks, and in other cases, it needs the torsional stiffness of connection enough. In such a case can be considered a model of the rotor as shown in Fig. 2.

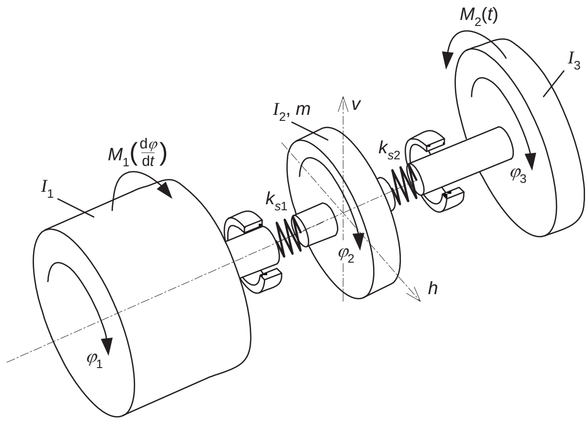

Fig. 2. Exemplary dynamic model of rotor system: I_1 – reduced moment of inertia of the LPC; I_2 – moment of inertia transmission system; I_3 – reduced moment of inertia of the LPT; m – mass of unbalanced system; k_{s1}, k_{s2} – couplings stiffness; $\varphi_{1,2,3}$ – angles of rotation; $M_{1,2}$ – drive and anti - drive torques

The equations of motion of the present system are presented in the form:

$$
\begin{aligned}
m\ddot{h} + kh &= me(\ddot{\varphi}\sin\varphi + \dot{\varphi}^2\cos\varphi)\,,\\
m\ddot{v} + kv &= me(-\ddot{\varphi}\cos\varphi + \dot{\varphi}^2\sin\varphi),\\
I_1\ddot{\varphi}_1 + k_{31}(\varphi_1 - \varphi_2) &= M_1(\omega)\,,\\
I_2\ddot{\varphi}_2 + k_{31}(\varphi_1 - \varphi_2) + k_{32}(\varphi_2 - \varphi_3) &= ke(v\cos\varphi_2 - h\sin\varphi_2)\,,\\
I_3\ddot{\varphi}_3 + k_{32}(\varphi_3 - \varphi_2) &= M_2(t)\,.
\end{aligned}
\tag{1}
$$

Acceptance of the rotor as a deformable body will use the partial differential equations of motion. The solution, although quite difficult to model, can significantly closer to the real object, which is important for machines working in a wide range of rotational speed like GTE. Schematic diagnostic model adopted in the diagnosis is shown in Fig. 3 [3]. Visualisation of modelled results of rotors movement is transformed from time waveform domain to the spectral domain using FFT and it is final, expected result of modelling – see Fig. 3.

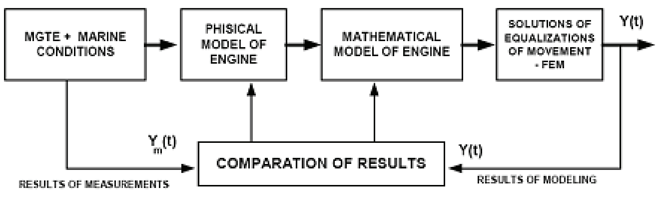

Fig. 3. Schematic model of diagnostic of the gas turbine engine

3. Module of monitoring

Diagnosing such machines like gas turbine engines depends on measurement and processing of vibration signals. It is an important fact that marine gas turbine engines do not work with a constant rotational speed of compressors and power turbines rotors. That is the reason to synchronize processing method with selected reference magnitude that is rotational frequency of one or both rotors [2, 3]. It allows recognizing most of the typical multifunction of rotors systems. Basic, operational inefficiency of gas turbine engines include following question:

- in result of damage or crushing of the compressors' blade(s) or power turbines blade(s) – more rarely,
- in result emerged unbalancing coming from the carbon deposit or the salinity,
- in result of seizing of rotors sealing system and leaking the lubrication oil inside the rotor,
- in result of aero-dynamical misalignment between gas generator and power turbine,
- in result of thermal damages of combustion chambers – torsional vibration of the power turbine,
- in result of damage auxiliary mechanisms of the engine.

The vibration monitoring system usually synchronously measures vibrations being the information source on rotor unbalance, which are generated over the front or middle bearing of the gas generator and sometimes over the bearing of the power turbine. The maximum (*peak – to – peak)* value of vibration velocity or displacement amplitudes within the range of the middle-pass filter controlled by synchronizing signal taken from rotational speed of rotor is a diagnostic parameter. The set values of *warning* signals and *shutdown* ones react only to a surpass of tolerated values. The preliminary trend analysis of vibration parameters, for purposes of the mentioned diagnostic system, was based on the parameters provided by the engine producer. On the basis of available documents, an analysis of history of changes of the parameters was performed by accounting for trends of changes of symptoms respective to their limiting levels.

On the basis of usefulness analysis of vibration parameters, the following were used as useful signals for the relation „*defect - symptom*":

- value of the 1st harmonic of vibration velocity (displacement) amplitude associated with the gas generator compressor,
- value of the 1st harmonic of vibration velocity (displacement) amplitude associated with the propulsion turbine
- to elaborate uniform procedures for unbalance assessment of rotors of gas turbine engines having service wear of various degrees, a concept of finding dimensionless parameters characterizing that state, were applied. With the use of a theoretical analysis of excitations as well as diagnostic test results, the following symptoms were selected as the most sensitive in practice and analyzed as trends [2]:
    - S 1 - ratio of the average value of vibration velocity amplitude of the relevant rotor (1st harmonic) and the component which corresponds with 2nd harmonic of excitation frequency of the relevant rotor, and
    - S 2 - ratio of the average value of vibration velocity amplitude of the relevant rotor (1st harmonic) and the component, which corresponds, with 3rd harmonic of excitation frequency of the relevant rotor – Fig. 4.

4. Off – line module of vibration measurements

Some of inefficiency can be recognized in the vibration spectra as a change of natural frequency of rotated elements of engine, so it is a reason to introduce the synchronous sampling under variable engine condition, e.g. run-up or shut-down processes. The occurrence of non-stationary effects, typical in the case of small unbalancing, can also be the result of progressive forming of damages, even though their intensity is slight in the early stages. The presented method can be introduced to the gas turbine monitoring systems as an early predictive diagnostics tool of unbalance identification.

For realization off – line type of measurements, multichannel analyzers are used. Measuring transducers are usually fixed to steel cantilevers located on the flange over the front or middle struts of compressors rotor. The fixing cantilevers characterized of a natural vibration resonance frequency value different enough from harmonic frequencies due to rotation speed of the given rotors so it is an important fact in the recording procedure [2, 4, 7].

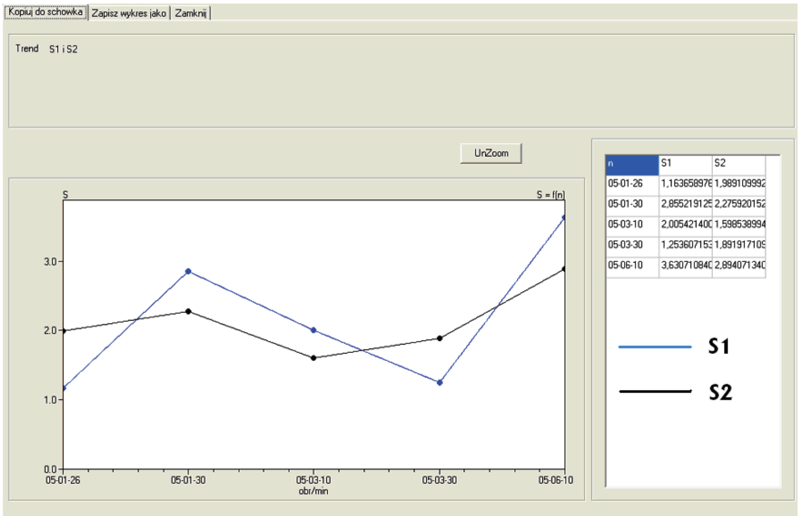

Fig. 4. Example of trend analyses of S1 and S2 symptoms

The test, connected with analyzes of vibration parameters, was the shut-down processes in the turning of engine process. The Order Tracking lies in the fact that tacho signal delivering the tracking reference must be selected from rotating shaft. Tachometer is inserted in a measurement setup. They can be used for triggering measurements and starting, updating and/or stopping storage of time signals and/or spectra to multi-buffers. Order analyzers can measure time (revs) signals, autospectra for signals and the cross-spectra of selected pairs of signals.

Upgrowth of stiffness of bearing system confirm the "right-hand branches" of harmonics appeared from the time point equal 4 second from 4, 8 and 12 orders when the pressure of lubrication oil fall down – Fig. 5. Analyze of the I harmonic (8 order) enables to observe changes of natural frequency calculated to the *rpm* domain – Fig. 6.

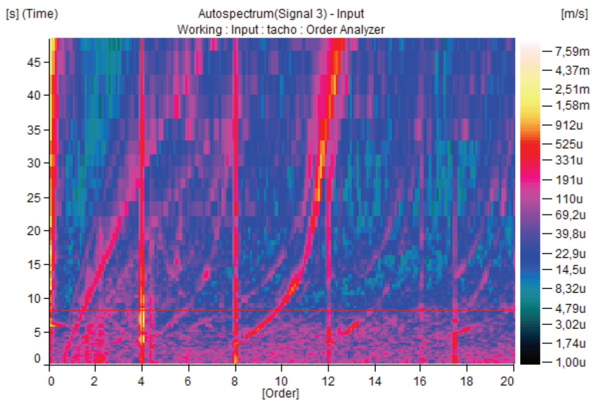

Fig. 5. Order analysis of shut down process of gas turbine compressor

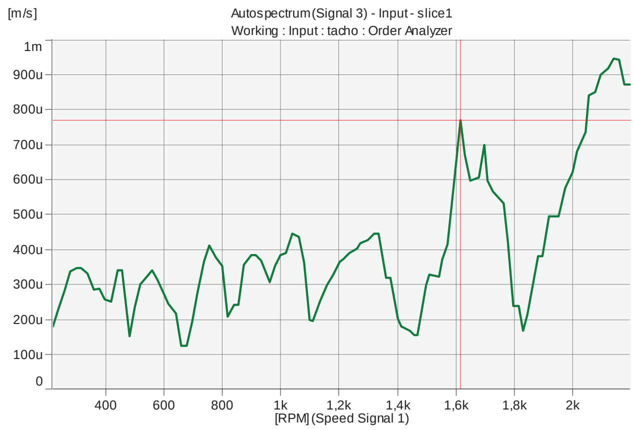

Fig. 6. Autospectrum of 8 order (I harmonic) of velocity of vibration in the shut-down process of LPC rotor stoppage

4. Conclusions

The realization of the investigations with the use of presented procedures made reliable verification of the investigation results possible. The following detail conclusions were drawn for further diagnostic inference of compared method:

- both procedures – *on – line* and *off – line*, have their own advantages and disadvantages but they can fulfill all technical requirements for vibro diagnosing rotors systems of marine gas turbine engines,
- the synchronized measurement of vibration signals allows to recognize specified symptoms of resonance and changes of natural frequencies during processes of wqstoppage of rotor systems.

Application Order Tracking procedures for monitoring system of gas turbine engines enables recognize changes of technical state characteristic for modes changes. Identification of foundation stiffness changing or unbalancing coming from impurities of the rotor system implicates natural frequencies changing which are observed in Order Tracking waterfall charts.

References

[1] Cioch, W., Jamro, E., *Digital signal acquisition and processing in FPGAs*, Przegląd Elektrotechniczny, Stowarzyszenie Elektryków Polskich, Main topics: Electrical measurements, R. 85, Nr 2, 2009.
[2] Charchalis, A., Grzadziela, A., *Diagnosing of naval gas turbine rotors with the use of vibroacoustic parameters*, The 2001 International Congress and Exhibition on Noise Control Engineering. The Hague, pp. 268, The Netherlands 2001.
[3] Downham, E., Woods, R., *The rationale of monitoring vibration on rotating machinery*, ASME Vibration Conference, Paper 71 - Vib - 96, Sept. 8 - 10, 1971.
[4] Grządziela, A., *Vibroacoustic method of shafting coaxiality assessment of COGAG propulsion system of a vessel*, Polish Maritime Researches, No. 3, pp. 29-30, 1999.
[5] Grządziela, A, *Diagnosing of naval gas turbine rotors with the use of vibroacoustics parameters*, Polish Maritime Researches, No. 3, pp. 14-17, Gdansk 2000.
[6] Grządziela, A., *Vibration analysis of unbalancing of marine gas turbines rotors*, Mechanika, T. 23, Z. 2, pp. 187-194, 2004.
[7] Krzyworzeka, P., Adamczyk, J., Cioch, W., Jamro, E., *Monitoring of nonstationary states in rotating machinery*, ITE, Krakow – Radom 2006.