Generator vibration control model using fourier transform, Bode diagrams and Nuquist procedure

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Abstract

During the construction of new or in the course of revitalization of existing power plants, management systems of great capabilities are installed, as one of the most important parts of the equipment. Timely detection of faults can ensure the avoidance of significant damage, by planning maintenance activities of rotating machines, and especially vibration control using special methods, such as Fourier transforms, Bode diagrams and Nyquist procedures.

Key words: generator vibrations, Fourier transforms, Bode diagrams, Nuquist procedures

1. Introduction

In the last thirty years, technological-information development has had an impact on all areas of work, including power plant management systems. In the same period, modern diagnostic systems were developed, which use the same or similar technical solutions [1, 2, 3]. During the construction of new or in the course of revitalization of existing power plants, management systems of great capabilities are installed as one of the essential parts of the equipment [4].

Timely detection of faults can ensure the avoidance of significant damage, by planning maintenance activities of rotating machines, and especially vibration control using special methods, such as Fourier transforms, Bode diagrams and Nyquist procedures. Of course, it should be emphasized that for the implementation of such activities, expert knowledge of the mechanical, fluid, temperature and electrical characteristics of rotary machines is necessary [2, 5, 6, 7].

2. Materials and methods

2.1. Basic quantities monitored on the generator

The problem of vibrations is very important for the working life of the generator, but as it is a very complex and broad area, which requires in-depth expert knowledge, both during measurements and when analyzing the obtained results, diagnosing the condition and making recommendations on taking corrective measures, so alone in itself can be the subject of extensive special studies [8]. By the way, when measuring vibrations, three quantities are used as a standard [1, 7, 9, 10]:

- Displacement (mm, µm),
- Speed (mm/s) i
- Acceleration (mm/s^2, g= 9.81 m/s^2).

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Vibration measurements are performed on the machine in the process of work, in various machine states and operating modes, whereby the basic concepts related to vibrations are defined by ISO standards (which are equivalent to the American standard) [7, 11, 12].

Measurements on the machine during rotation are performed on stationary parts (measurement of absolute vibrations of bearing housings and generators, coil heads and magnetic circuit) and on rotating parts (examination of relative shaft vibrations) [9]. Tests are performed at steady and transient operating modes (starting up, braking at a stop and starting at a sudden load), as well as at different thermal states of the machine and different loads (mechanical rotation in an unstressed state, idling and different load modes). According to existing regulations, measurements are made in the frequency range from 2 to 1000 Hz, although sometimes the lower limit of the frequency range is required to be only 1Hz [6, 11, 13].

When examining vibrations on an unexcited generator, the sources of vibrations are of a mechanical nature, while when the generator is idling, these vibrations, which normally come from sources of an electromagnetic nature, are superimposed, while by comparing these measurement results, mechanical and electrical sources of vibrations can be separated [7, 14]. Otherwise, vibrations can be periodic and aperiodic, whereby periodic can be harmonic (represented by a simple sinusoidal function $x = XM \sin \omega t$), or non-harmonic (whose description requires knowledge of the mean and effective value of vibrations). In addition, when analyzing vibrations, one moves from the time domain to the frequency domain, using the Fourier transformation. A sinusoid in the time domain is mapped to a peak in the frequency domain, thus obtaining the frequency spectrum of vibrations.

In general, it can be said that amplitude, speed and acceleration are important for assessing the severity of vibrations, and for sample analysis, frequency of analysis [5, 15].

According to the practice of the world's leading manufacturers of hydro and thermal generators and their generator monitoring systems, it is recommended to monitor the following values and parameters [5, 8, 15]:

1. Temperatures of generator parts and assemblies and cooling fluids;
2. Vibrations of mechanical parts of the generator and relative movements (orbits);
3. Form of stator, rotor and gap of the generator;
4. Partial discharges in the coil;
5. Operating data of the generator;
6. Presence of water in the oil;
7. Speed of rotation;
8. Magnetic flux;
9. Flow - water pressure for cooling;
10. Oil levels in bearings;
11. Control of low revolutions;
12. Structural advent of sound;
13. Content of gases and moisture in the cooling air;

14. Degree of usefulness of the generator;
15. Rotor winding temperatures;
16. Quality of cooling water (distillate);
17. Measurement of individual losses i
18. Noise measurement in the generator barrel,

whereby, on the basis of such recommendations, the experimental part of work and vibration measurements on the Kostolac B - Drmno TPP turbogenerator (Block 1 and 2) would be started.

1.1. Measurement of vibrations on the turbogenerator of TPP Kostolac B - Drmno (Block 1 and 2)

The entire research is based on the experiment and measurement of vibrations, which occur on the turbogenerator of TE Kostolac B - Drmno (Block 1 and 2) 110 MW, where the disposition of the turbogenerator with measuring points is shown in figure no. 1 [8, 9, 11, 16, 17]:

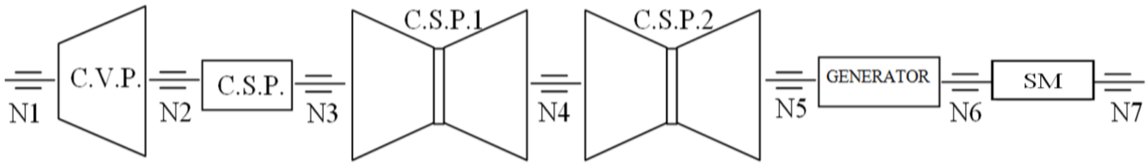

Figure no. 1 Arrangement and numbering of measuring points

where is: C.V.P. - high pressure turbine
C.S.P. - medium pressure turbine
C.N.P.1 - low pressure turbine 1
C.N.P.2 - low pressure turbine 2
SM - synchronous machine - excitation winding
N_1, N_2, N_3, N_4, N_5, N_6, N_7 - bearings (measuring points)

The experiment used built-in instrumentation on the machine, as well as state-of-the-art IRD instrumentation.

2. Results

The first part of the experiment refers to the measurement of the vibrations of blocks 1 and 2, and the obtained results are shown in table no. 1.

Measurements were performed according to the ISO10816 procedure, monitoring absolute vibrations (mm/s) in the horizontal (H), vertical (V) and axial (A) directions, at full load, during stable operation and normal speed and at 4580 and 4510 revolutions in a minute.

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Table No. 1 Vibration measurement results of blocks 1 and 2

rpm	Unit of measure	Direction	N_1	N_2	N_3	N_4	N_5	N_6	N_7
4580	SUM (µm ▲)	V	4.1	4.1	5.1	5.1	7.1	10.1	9.6
		H	17.1	7.1	6.6	12.1	8.1	5.5	5.0
		A	16.6	6.6	6.1	4.6	4.1	5.6	6.6
4580	SUM (mm/s)	V	0.8	0.8	1.8	1.2	1.3	1.2	1.1
		H	4.8	1.8	1.6	3.4	1.9	0.9	0.6
		A	4.6	1.5	1.6	0.9	1.5	0,8	.6
4510	SUM (µm ▲)	V	4.6	3.6	4.1	5.1	3.5	10.1	8.6
		H	16.6	7.1	6.1	11.0	8.0	6.5	6.0
		A	14.6	5.6	6.1	5.6	4.1	6.1	7.5
4510	SUM (mm/s)	V	0.8	1.6	1.1	0.9	1.3	0.8	0.8
		H	4.6	1.8	1.3	3.1	1.9	0.8	0.8
		A	4.2	1.2	1.7	1.3	1.4	0.8	0.6

It is recommended that fast Fourier transformation and Bode diagrams be used when measuring vibrations, and the tendency to imbalance is checked according to the Nyquist Bode procedure [6, 7, 9], which was also applied in the experiment.

Frequency scales on vibration vs. frequency graphs are usually linear, have values from zero to some maximum value, and are displayed on the left side of the vibration image on most FastFourierTransform analyzers. Amplitude scales can be linear or logarithmic, with a base of 10 (dB), and usually represent the actual values from zero to the peak of the curve [2, 3, 15]. Logarithmic scales are often used to improve the resolution of extreme values (peaks) of natural frequencies.

Picture No. 2 shows part of the graph (period of one year) of continuous monitoring of vibration speeds for sliding bearing No. 5, for a period of one month, during 24 hours:

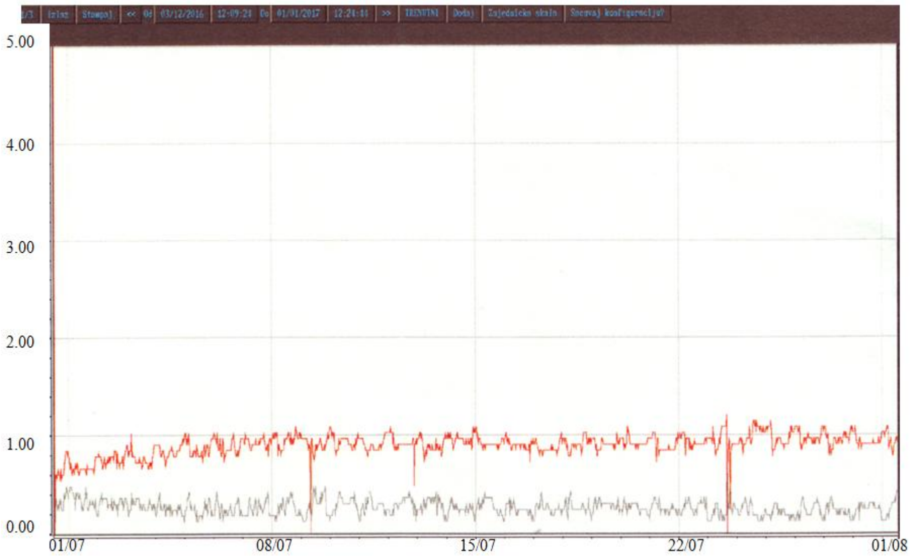

Figure no. 2 Monitoring of vibration speeds for sliding bearing no. 5

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As a very useful procedure in the procedures of measuring and observing the movement of the rotor [5, 9], the generation of the so-called of the polar diagram of the amplitude and phase of movement. It should be emphasized the possible unusual effects on the rotor response and on the movement phase, under the influence of shaft bending.

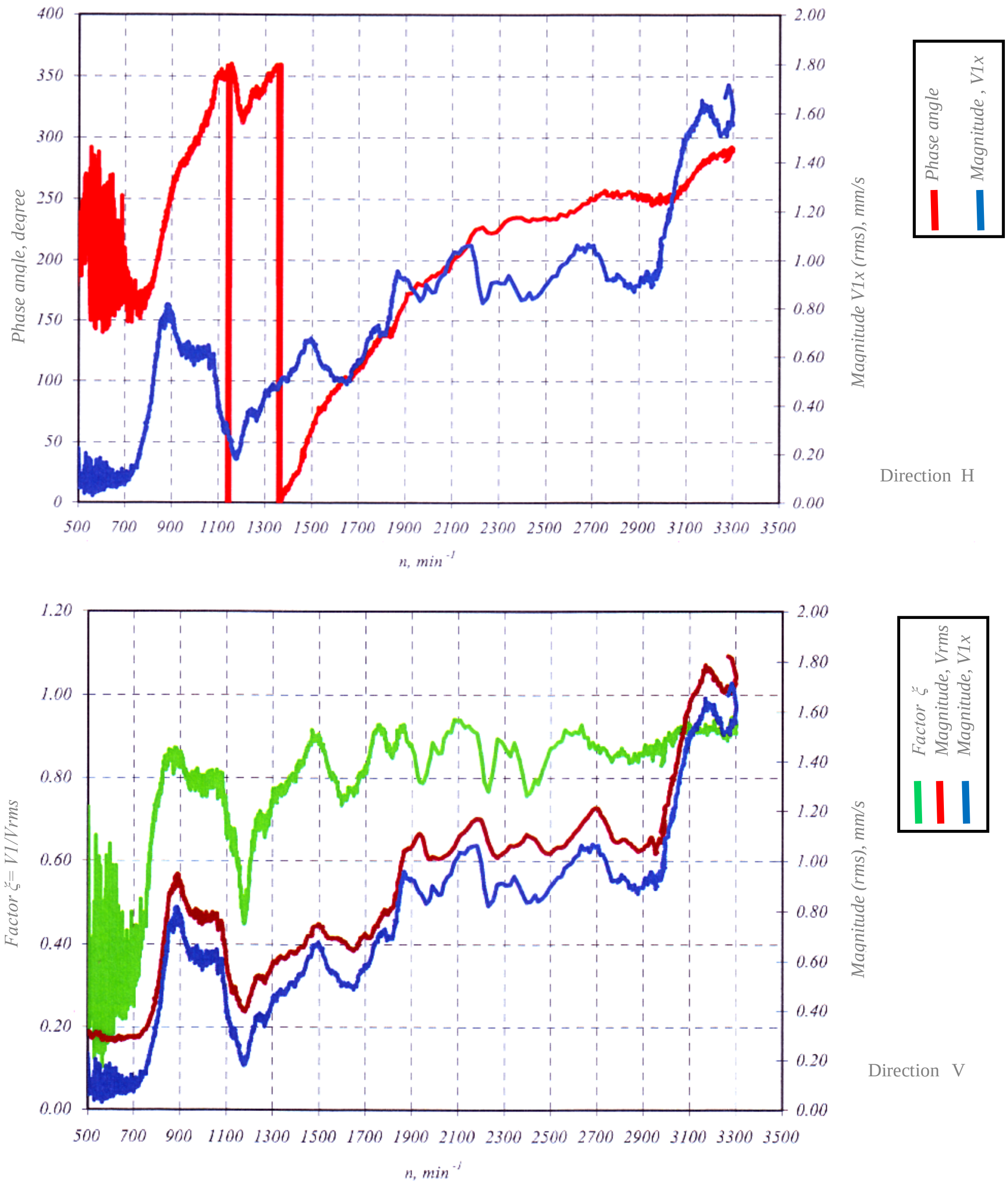

Figure no.3 Bode diagrams

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Figure no. 3 show Bode diagrams of the bearing housing, for measuring point N_3, directions H and V, while Figure 4 shows the spectral and temporal characteristics for measuring point N_5:

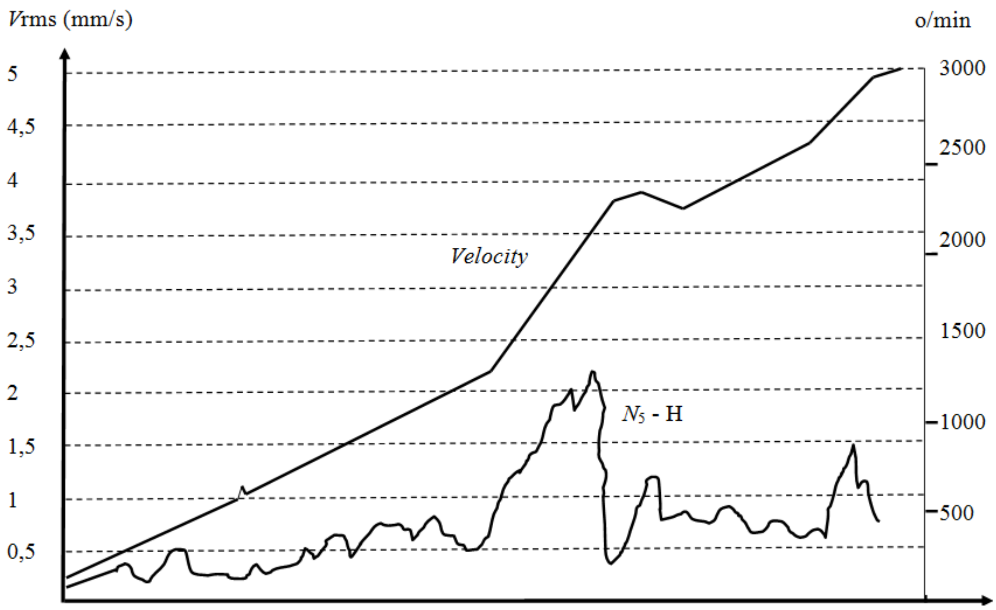

Figure no. 4 Cascade spectrum N_5 – H

Table no. 2 shows the dynamic parameters of the rotor (Block B2 - turbogenerator of TPP Kostolac, Drmno), i.e. the tendency of the rotor towards imbalance, according to the Bode procedure [7, 9]:

Table No. 2 Parameters of the rotor's tendency to imbalance according to the Bode procedure

direction	Drive part										Working part					Note
	Measuring points															
	N_1					N_2					N_3					
	ω_n min^{-1}	Ω_1 min^{-1}	Ω_2 min^{-1}	η_n	Q_n	ω_n min^{-1}	Ω_1 min^{-1}	Ω_2 min^{-1}	η_n	Q_n	ω_n min^{-1}	Ω_1 min^{-1}	Ω_2 min^{-1}	η_n	Q_n	
X	2.028	1.919	2.085	1.58	19.58	Θ	Ø	Ø	2.15	Θ	2.855	2.840	2.930	1.15	35.13	The parameter Qn is defined according to the standard ISO 10814 : 1996 : $Q_n=\omega_n/(\Omega_2-\Omega_1)$
Y	2.148	2.145	2.186	1.49	26.80	2.608	2.586	2.668	1.16	21.75	Θ	Ø	Ø	1.08	Θ	

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where is:

- Q_n - factor of increase
- ω_n – critical angular speed of the shaft
- η_n – tendency of the rotor towards imbalance
- Ω – operating speed of the shaft
- ʘ - data and/or parameter is not defined;
- Ø - lack of data and/or inaccessibility of the measuring point.

Table No. 3 shows the parameters of the inclination and sensitivity of the rotor (Block B2 - turbogenerator of TE Kostolac, Drmno) towards imbalance, according to the Nyquist procedure [7, 9, 17]:

Table No. 3 Parameters of the tendency of the rotor towards imbalance according to the *Nyquist* procedure

direction	Drive part											Working part	Note
	Measuring points												
	N_1				N_2				N_3				
	ω_n min^{-1}	Ω_{45} min^{-1}	η_n	Q_n	ω_n min^{-1}	Ω_{45} min^{-1}	η_n	Q_n	ω_n min^{-1}	Ω_{45} min^{-1}	η_n	Q_n	
X	2.030	⊗	1.45	⊗	1.420	⊗	2.15	⊗	2.880	2.815	1.08	33.88	The parameter *Qn* is defined according to the standard ISO
Y	2.125	⊗	1.45	⊗	2.610	2.840	1.16	22.21	2.886	2.476	1.08	3.28	

where is:

- Q_n - factor of increase
- ω_n – critical angular speed of the shaft
- η_n – tendency of the rotor towards imbalance
- Ω – operating speed of the shaft
- ʘ - data and/or parameter is not defined;
- Ø - lack of data and/or inaccessibility of the measuring point.

Note:

• According to the ISO 10814:1996 standard, point 5, the evaluation of the rotor's sensitivity and tendency to imbalance is performed in the parameter range $0.8 \leq \eta_n \leq 1.3$;

• Based on the previous paragraph and the working speed of the shaft of Ω =3000 min^{-1}, the test is performed in the area of critical shaft speeds of 2300 min$^{-1} \leq \omega_n \leq$ 3750 min^{-1}.

Table No. 4 shows the results of identification of relative damping parameters ζ_n, as well as modal sensitivity M_n of the rotor shaft:

Table no. 4 Relative damping parameters ζ_n, as well as modal sensitivity M_n of the rotor shaft

direction	Drive part													Working part		Note
	Measuring points															
	N_1					N_2					N_3					
	ω_n min^{-1}	Ω_1 min^{-1}	Ω_2 min^{-1}	ζ_n %	M_n	ω_n min^{-1}	Ω_1 min^{-1}	Ω_2 min^{-1}	ζ_n %	M_n	ω_n min^{-1}	Ω_1 min^{-1}	Ω_2 min^{-1}	ζ_n %	M_n	
X	2.030	1.979	2.085	2.58	1.85	1.418	1.388	1.506	3.56	1.28	2.880	2.835	2.916	1.45	9.8	
Y	2.150	2.105	2.185	1.88	2.05	2.615	2.546	2.666	2.36	4.06	2.830	2.316	>3000	⊙	⊙	

where is:

- ω_n – critical angular speed of the shaft
- Ω – operating speed of the shaft
- ⊙ - data and/or parameter is not defined;

Note:

• markings comply with ISO 10814:1996 standards
• symbols comply with the ISO 7000:1989 standard

3. Conclusion

Detecting the cause of vibrations in turbogenerators is in most cases very complex, because it is usually a matter of several different causes. If the turbine has been working with increased vibration for a long time, as a rule, the consequences of the vibrations are transformed into their complementary causes. The spread of vibrations is usually

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determined by individual parts and bearings of the turbine and generator, under normal load and normal operating parameters. It should always be kept in mind that the spread of vibrations does not have to be characteristic of the place of their origin. This means that it is necessary, of course, while monitoring the spread of vibrations and changes in their magnitudes, to carry out the following tests of the turbogenerator operating mode:

- Determine the spread of vibrations under normal load and normal parameters;
- To do test of idles, without generator excitation, in order to determine the dependence of the magnitude of the vibration amplitude on the number of revolutions;
- To do test of idles, with the transition to excitation of the generator, due to showing the change in vibrations when the generator voltage changes;
- To do test in long-term of idling for 3 to 4 hours, with the generator excited, during which the effect of heating of individual assemblies of the turbine body on the magnitude of vibrations is checked;
- To do turbine load test, where the load should be changed at a normal speed, and the magnitude of the vibration should be measured every ¼ to 1/5 of the load increase;
- To do test of long-term operation, at the maximum possible even load (according to the conditions of the size of the vibration), for a duration of 3 to 4 hours;
- To do trial of „removing“ the load and stopping the turbine, which allows judgment on the impact of residual thermal deformations on the magnitude of vibrations;
- In case of doubts about the causes of vibration, apart from the described tests, separate commissioning can also be carried out:
    - Turbines without generators, or
    - Generator with synchronous motor without turbine.

On the other hand, the quality status according to the requirements of ISO standard 10816-2:2001 is as follows:

1. Vibration state of the bearing housing during start-up until synchronization
Drive part (bearings N_1, N_2, N_3, N_4): the identified vibration condition meets the requirements of ISO 10816-2:2001.
Working part (bearings N_5 and N_6): the identified state of vibration meets the requirements of ISO 10816-2:2001.
2. Vibration state of the rotor during exploitation - cold state
Drive part (bearings N_1, N_2, N_3, N_4): the identified vibration condition meets the requirements of ISO 10816-2:2001.
Working part (bearings N_5 and N_6): the identified vibration condition meets the requirements of ISO 10816-2:2001.
3. Vibration state of the rotor during exploitation - warm state
Drive part (bearings N_1, N_2, N_3, N_4): the identified vibration condition meets the requirements of ISO 10816-2:2001.
Working part (bearings N_5 and N_6): the identified state of vibration meets the requirements of ISO 10816-2:2001.
4. Final remarks and conclusions

Despite the fact that the requirements of the ISO 10816-2:2001 standard were met in all tested operating modes of the turbo-aggregate, both during start-up and in operational modes, due to the reasons stated in the Final remarks and conclusions in the Report on the dynamic condition of the rotor, there are indications that supplement the conclusions presented in the Report on the dynamic condition of the rotor, which refer to the rotor line, which primarily refers to the appearance of higher harmonics in the vibration spectrum.

5. Suggestions for improvement

In order to increase the safety and reliability of the operation of the turbo-aggregate, as well as to better prepare the upcoming overhaul activities, it is suggested that in the following period, when the operating conditions permit, at least one more test of the vibration condition of the bearing housing in accordance with ISO 10816-2 :2001.

4. References

[1] Adamović Z., Josimović Lj., Vulović S., Ilić B., Spasić D. 2016. Vibrodijagnostičko održavanje tehničkih sistema, Društvo za tehničku dijagnostiku Srbije, Beograd

[2] Dimitrijević D., Adamović Ž. Adamovic M. 2024. Implementation of wear and tear products control procedures in hydraulic systems, Advanced technologies in education and economy, Conference with international participation, Vrnjačka banja

[3] Dimitrijević D., Adamović Ž. 2015. Komparativne metode u dijagnostifikovanju naprezanja tehničkih sistema, Tehnička dijagnostika, vol 14, br. 4

[4] Adamović Ž., Ivić M., Vuković V. 2021. Sistem kvaliteta, Univerzitet PIM, Banja Luka

[5] Zuber N., Šostakov R. 2012. Daljinski monitoring stanja rotirajućih mašina, Tehnička dijagnostika, vol 11, br.1, str. 9-16

[6] Adamovic Ž., Josimovic LJ. 2020. Tehnička dijagnostika, Društvo za tehničku dijagnostiku Srbije, Begrad

[7] Jevtic M., Radovanovic LJ., Adamovic Ž. 2011. Numerical and Eksperimental Aspects of Thermally Induced Vibration in Real Rotors, Thermal Science Journal, vol 15, issue 2, pp 545-558

[8] Studija: Procena stanja i preostalog radnog veka generatora proizvodnih jedinica EPS-1 faza, Elektrotehnički institut Nikola Tesla, Beograd, 2012.

[9] Ličen H., Zuber N. 2007. Prediktivno održavanje rotirajuće opreme na bazi merenja i analize vibracija, Tehnička dijagnostika, vol 6, br. 1, pp 13-20

[10]Curran K., King R. 2019. Predictive Maintenance for Vibraton-Rolated Failures in the Semi-Conductor Industry, Comput. Eng. Inf. Technol, 8:1, Juornal of Computer Engienering Information Technology, vol 8, issue 1

[11]Studija: Implementacija monitoringa parcijalnih pražnjenja kod generatora u elektranama EPS u jedinstveni dijagnostički centar, Elektrotehnički institut Nikola Tesla, Beograd, 2013.

[12]Adamovic Ž., Josimovic Lj. 2021. Hidraulika i pneumatika, Društvo za tehničku dijagnostiku Srbije, Beograd
[13] Lifson A., Simmons H., Smalley A. 1987. Vibration Limits for Rotating Machinery, Mechanical Engieneering, Southwest Research Institute San Antonio Texas, pp 60-65,
[14]Otic G., Jovanov G., Adamovic Ž. 2021. Proactive Maintenance Model, Based on the Law on change of Mechanical Vibration, Acta Polytechnica, Budapest, vol 18, no 4, pp 161-178
[15]Адамовић Ж., Илић Б. 2013. Наука о одржавању техничких система, Српски академски центар, Нови Сад
[16]Vulovic S., Otic G., Adamovic Ž. 2018. Vibrations of Turbo Rotor Aggregate Smederevo Iron Casual by Film in Bearings, Journmal of the Balkan Tribological Association, vol 24, no 3,
[17]Izveštaj o ispitivanju vibracija turbogeneratora TE Kostolac A, Blok A1, turboagregat 110 MW, NORTH Control, Subotica, 2008.