ABSOLUTE AND RELATIVE VIBRATION INCONSISTENCIES IN DYNAMIC ANALYSIS OF HIGH POWER TURBOGENERATOR ROTATING SYSTEM

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Summary

The paper presents the research results of machine condition monitoring and fault diagnostics of the High Power Steam Turbines and Electric Generator rotating system. The inconsistencies of journal bearings absolute vibration parameters of the rotating system and journal bearings shafts relative vibration parameters have been shown. In the low frequency range the shafts relative vibration displacements are decisive informative in defects diagnostics. The journal bearings supports absolute vibration velocities of low frequencies are being strongly damped by massive bearings supports, etc. Journal bearings absolute vibration parameters supply the information erroneously attributed to the rotor. High frequency absolute vibration acceleration, however, make it possible to provide monitoring of coupling.

Keywords: steam turbine, electric generator, journal bearings, absolute and relative vibration

1. INTRODUCTION

More than 25 turbo generators whose power exceeds 20 MW are operating in Lithuanian Thermal Power Stations. Most of these machines operation period is more than 20-40 years as through the world. But perspective requirements to those machines are to operate not only safely under long – period mode but to provide higher efficiency. Only after modernization used in automatic control and mechatronic systems this task will be achieved. In this presentation the modernization of 60 MW power turbogenerator have been studied. Permanent monitoring system provides vibration, temperature, geometric parameters and process variable information.. The experimental research has been provided to evaluate technical condition of the turbogenerator as a whole machine, journal bearings and tooth wheeled coupling. The inconsistencies between absolute and relative vibration parameters are examined in this study.

2. 60 MW POWER TURBOGENERATOR

The rotor of a turbogenerator unit with some data formats of on line monitoring system is shown in Fig.1.

Measuring system with eddy current transducers (proximity probes) serve the measurement of mechanical quantities: peak-to-peak values of shafts vibration displacements S_{p-p} and radial clearances Δr between the shafts and the bearings. Due to the contact less measuring principle, small dimensions, a rugged construction and the endurance against aggressive media, this type of proximity probes is successfully used in turbo machines – Steam turbines Generator sets.

The monitoring results (S_{max} , S_p , S_{0-p} , Δr , etc.) are given in steady state and transient data formats and adapted for the analysis with diagnostic software DDS 2000 [1-3]. The diagrams of the maximum values of shaft displacement from time integrated mean position of the 1st and 2nd bearings shafts S_{max1}, S_{max2} under the varying loads are shown in Fig. 2.1a and Fig. 2.2a. According the recommendations of ISO 7919-2:2001 standard the steam turbine HPR technical condition is not acceptable for continuous long term operation - the S_{p-p} exceeds permissible safe operation values – zone boundaries C/D is S_{p-p}=260 μm.

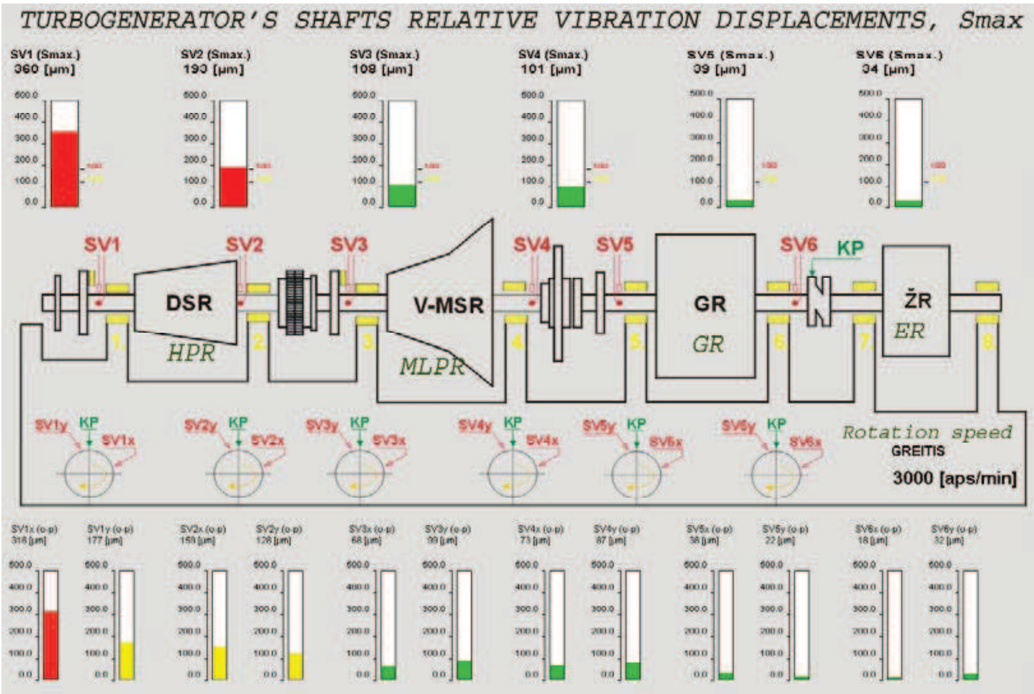


Fig.1 The schematic plot of 60 MW turbogenerator with monitoring system results

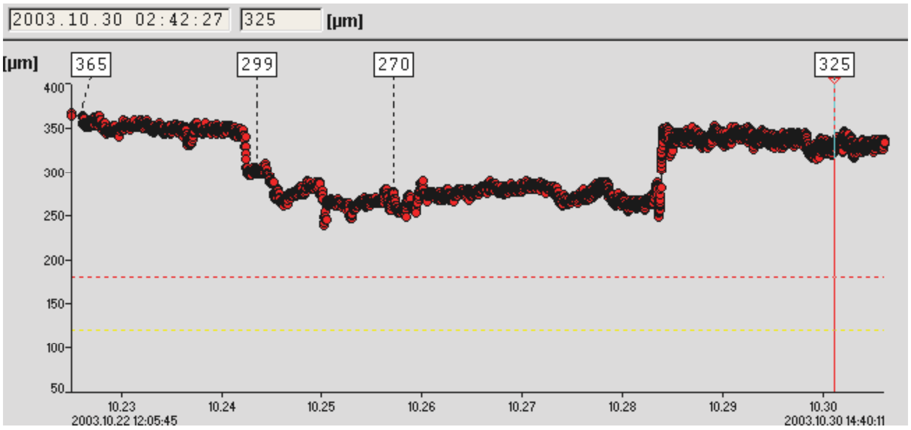


Fig. 2.1a HPR 1st bearing maximum value of shaft displacement S_{max1} plot at varying load: 365 µm/41 MW, 299 µm/49 MW, 270 µm/55 MW and 325 µm/43.5 MW

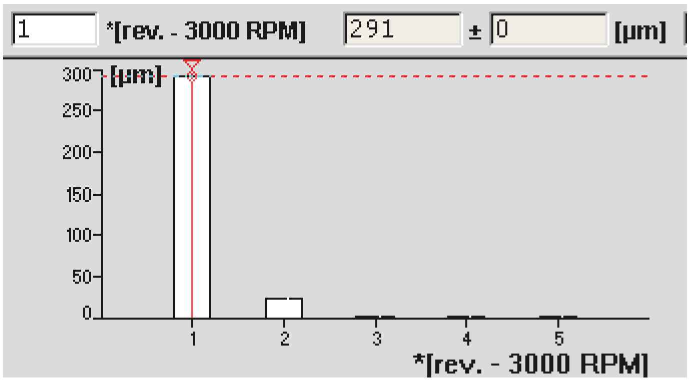
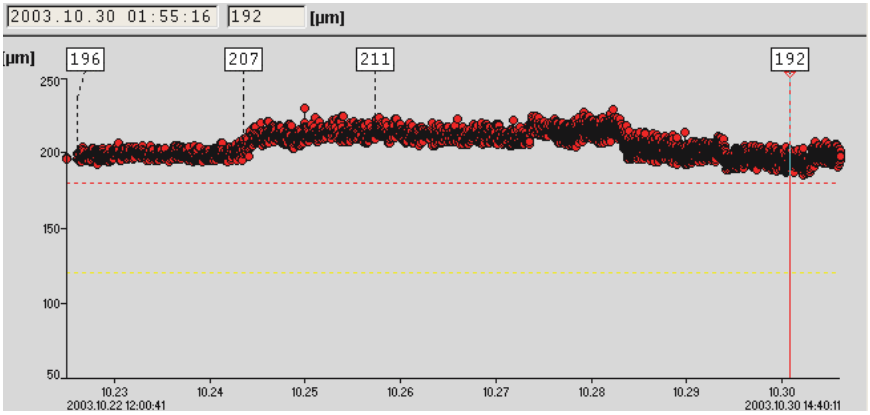


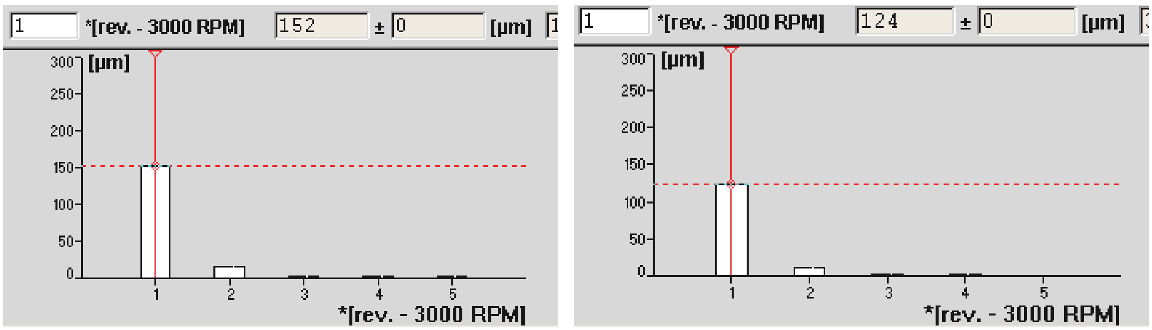
Fig. 2.1b HPR 1st bearing shaft relative vibration displacements spectra in horizontal direction at 41 MW load, S_{max1}=365 µm



2nd bearing shaft with two proximity probes

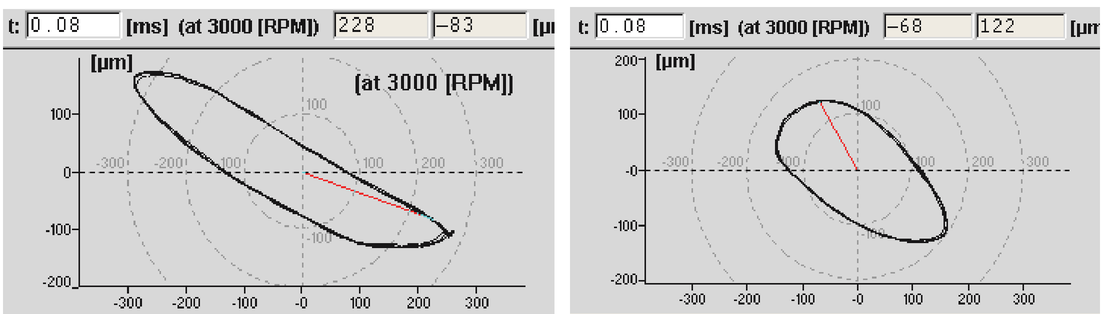
Fig. 2.2a HPR 2nd bearing maximum displacement S_{max2} plot at varying load:196 μm/41MW, 207 μm/49MW, 211 μm/55MW and 192 μm/43.5 MW

The shafts relative vibration displacements spectra shows that 1X frequency vibration displacements amplitudes are prevailing in 1st, 2nd , 3rd and 4th bearings. The maximum vibration displacement amplitudes of 1X frequency of 1st and 2nd bearings are dominant in horizontal (and vertical) plane as shown in Fig. 2.1b and Fig. 2.2b but in horizontal plane vibration severity is higher than in vertical. The 2nd bearing shaft is less sensitive to load changes in comparison with 1st bearing shaft.



Horizontal Vertical

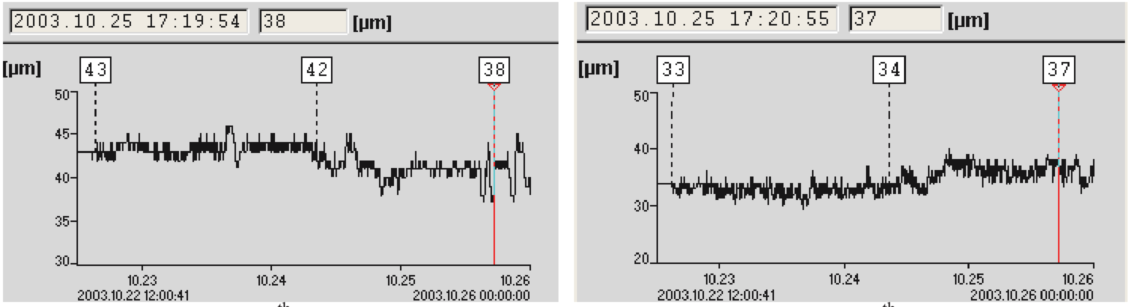
Fig. 2.2b HPR 2nd bearing shaft relative vibration displacements spectra in horizontal and vertical directions at 41 MW load, S_{max2}=196 μm



1st 2st

Fig. 2.2c HPR 1st and 2nd bearings shafts relative displacement orbits at 41 MW load

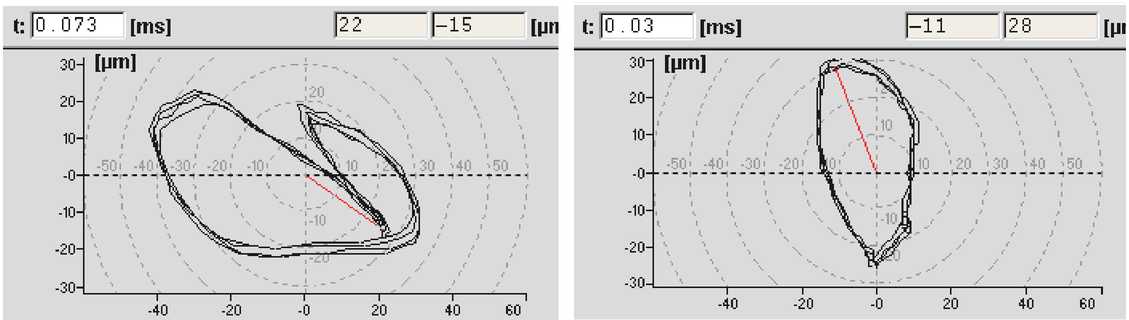
The orbits shapes of 1st and 2nd bearings are very different in values, but both are close to elliptical orbit shape as orbits of 3rd,4th bearings shafts (not shown) and 6th bearing [1]. The maximum values of shaft displacements S_{max} plots and orbits indicate that the most vibroactive are the 1st and 2nd bearings shafts and that vibration severity is independent of the load level of turbogenerator. The orbits of the 1st and 3rd bearings shafts have the same shape but differ more from the ellipse shape because these two bearings are radial-axial type. The orbits of 2nd, 4th and 6th bearings shafts are close to elliptical shape because these bearings are radial and provide the similar dynamic motion.



5th 6th

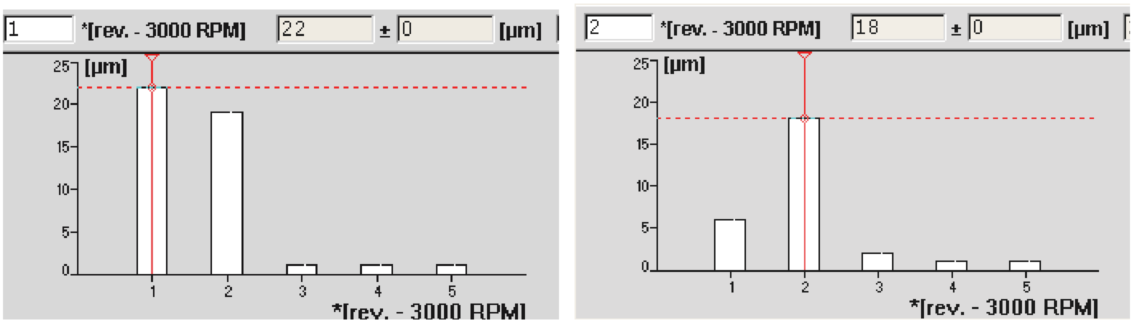
Fig. 2.3a GR 5th and 6th bearings maximum values of shafts displacements S_{max5} ,S_{max6} plots at varying load: 43 μm/41MW, 42 μm/49MW, 38 μm/55MW and $\boldsymbol{S_{max6}}$ **:** 33 μm/41MW, 34 μm/49MW, 37 μm/55MW

The generators rotor maximum values of shafts displacements of the 5th bearing S_{max5} and of 6th bearing S_{max6} under varying load are shown in Fig. 2.3a. Evaluating vibration severity according ISO 7919-2 recommendations the technical condition of generators bearings is acceptable for long-term operations – peak-to-peak values of shaft displacements 60 μm. Despite low vibration displacement values the kinetic orbit of 5th bearing shaft is very differs from the ellipse shape despite the radial, not radial-axial design of this bearing as shown in Fig. 2.3b. The vibration displacement spectra of 5th bearing is shown in Fig.2.3c and indicates that 1X and 2X frequency vibration displacement amplitudes have the same values in horizontal direction, but 2X is dominant in the vertical direction.



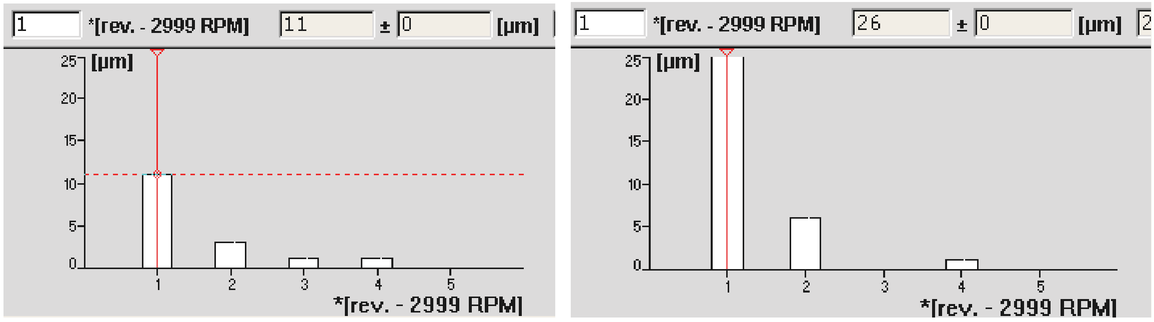
5th 6th

Fig. 2.3b GR 5th and 6th bearings shafts relative vibration displacements kinetic orbits at 41 MW loading



Horizontal Vertical

Fig. 2.3c GR 5th bearing shaft relative vibra-tion displacements spectra in horizontal and vertical directions at 41 MW load, S_{max5}=43 μm



Horizontal Vertical

Fig. 2.3d GR 6th bearing shaft relative vibration displacements spectra in horizontal and vertical directions at 41 MW loading, S_{max6}=33 µm

Table 1 Bearings shafts relative vibration displacements parameters: $S_{0\text{-}px}$, $S_{0\text{-}py}$, S_{max} and harmonics S_{p1x}, S_{p2x} in X and S_{p1y}, S_{p2y} in Y directions (1X=50 Hz frequency) **41 MW load**

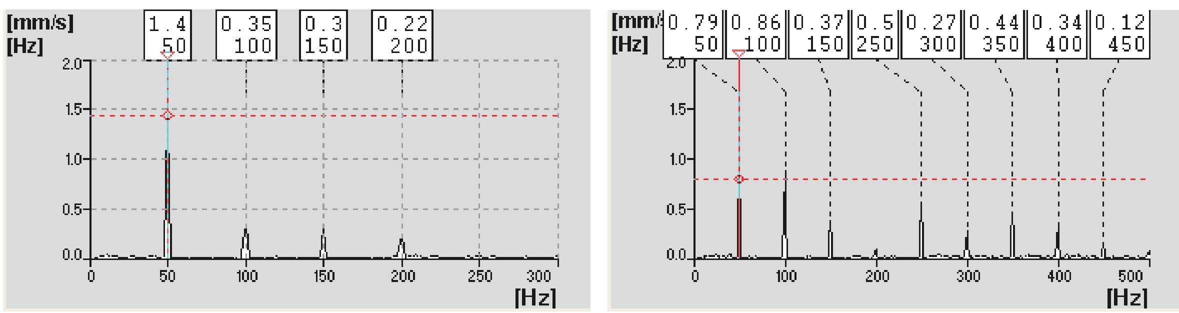
Rotors	HPR		MLPR		GR	
Bearings shafts	**1**	**2**	**3**	**4**	**5**	**6**
Maxima displacement of total vibration, S_{max}, µm						
Smax	**365**	**196**	123	100	43	33
Peak values of vibration displacements $S_{0\text{-}px}$ in X and and $S_{0\text{-}py}$ in Y directions, µm						
$S_{0\text{-}px}$	**324**	**161**	84	70	42	17
$S_{0\text{-}py}$	**173**	**129**	110	89	22	32
1X and 2X frequency amplitudes S_{p1x}, S_{p2x} in X and S_{p1y}, S_{p2Y} in Y directions, µm						
Sp1x	**291**	**152**	68	65	**22**	11
Sp1y	**152**	**124**	103	81	**6**	26
Sp2x	23	15	3	1	**19**	3
Sp2y	31	11	1	6	**18**	6

The vibration displacement spectra of 6th bearing shown in Fig.2.3d indicates that 1X frequency vibration displacement amplitudes are prevailing in comparison with 2X vibration amplitudes in both horizontal and vertical directions.

The summary of relative vibration displacements parameters is shown in Table 1. Despite the fact that GR vibration severity is low, it is important to establish main sources of these vibrations that caused kinetic orbit shape changes from ellipse shape.

3. HPR AND MLPR BEARINGS ABSOLUTE VIBRATION

Turbo generator's bearings absolute vibration velocities and accelerations were measured with two different kinds of seismic transducers. The vibration velocity measuring transducer that operates on the inertial mass-moving case principle and provide measurements directly in velocity units and the vibration acceleration measuring transducer that operates on piezoelectric crystal physics and provide measurements directly in acceleration units and after integration – in velocity units. These measurements signals have been analyzed with Vibration Signal Analyzers for diagnostics purposes [1-3]. The absolute vibration velocities parameters are shown in Fig. 3.1, Fig.3.2 and Table 2.

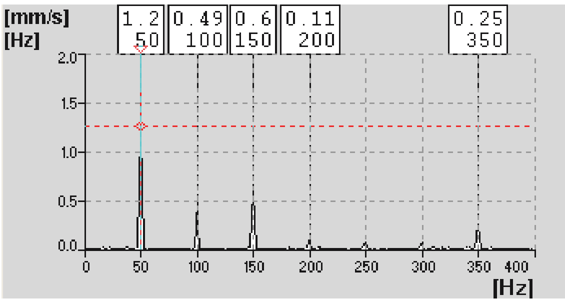


Horizontal, V_{rms}=2.0 mm/s Vertical, V_{rms}=1.6 mm/s

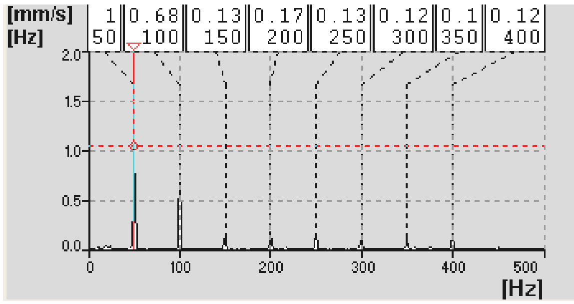
Fig.3.1 HPR 1st bearing absolute vibration velocity spectra in horizontal and vertical at 44 MW load

According absolute vibration velocities root mean square values V_{rms} and ISO 10816-2 recommendations the techical condition of HPR both bearings are good, because vibration velocity V_{rms} value is in A vibration evaluation zone. According ISO 10816-2 A/B vibration severity evaluation zone boundary is V_{rms}=3.8 mm/s (page 6, Annex A). This result is to the contrary with HPR bearings shafts relative vibration displacements peak-to-peak values. Vibration severity values for 1st bearing shaft S_{max1}=365 μm and for 2nd bearing shaft S_{max2}=196 μm, and S_{p-p}>400 μm has been reached not acceptable evaluation zone D.

Only 1X=50 Hz frequency vibration velocities amplitudes dominant in HPR 2nd and 1st horizontal vibration spectra and in MLPR 3rd and 4th bearings vibration spectra. In vertical direction vibration velocities amplitudes of 2X frequency is slightly greater than 1X component.



Horizontal, V_{rms}=1.6 mm/s



Vertical , V_{rms}=1.3 mm/s

Fig. 3.2 HPR 2nd bearing absolute vibration velocity spectra in horizontal and vertical directions at 44 MW load

The higher V_{rms} value of MLPR 3rd and 4th bearings as shown in Table 2 reached V_{rms}=3.7 mm/s when measured in vertical direction on 4th bearing support. The 1X frequency harmonic vertical velocity amplitude reached V_p=3.6 mm/s. The technical condition of HPR and MLPR bea-rings is acceptable according ISO 10816-2 recom-mendations. This result is in contrary with MLPR bearings shafts relative vibration displacements values for 3rd bearing shaft S_{max3}=123 μm and for 4th bearing shaft S_{max4}=100 μm that S_{p-p} .260 μm reaches not acceptable evaluation zone D. This result is in contrary with MLPR bearings shafts relative vibration displacements values as with HPR too.

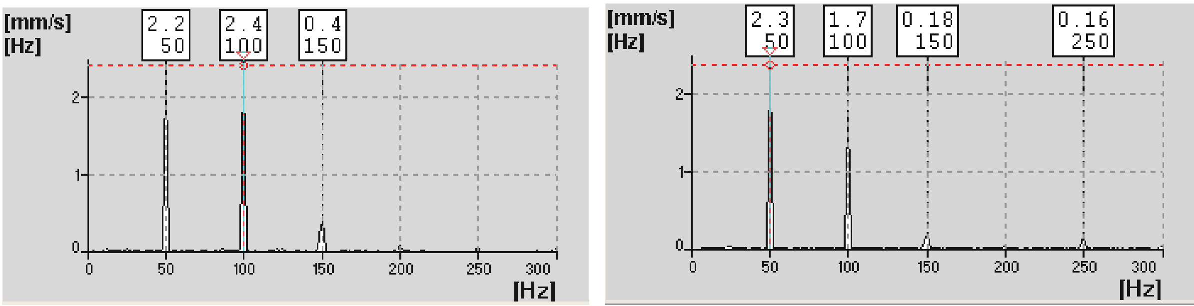
Table 2 Turbogenerator's bearings absolute vibration velocities root mean square values V_{rms} (mm/s) at 44 MW load

Rotors	**HPR**		**MLPR**		**GR**		**ER**	
Bearings	***1***	***2***	***3***	***4***	***5***	***6***	***7***	***8***
Direction V,vertical	1.6	1.3	1.5	**3.7**	3.0	1.8	2.6	1.7
1X	*0.79*	*1.0*	*1.3*	***3.6***	*2.3*	*1.6*	*2.4*	*1.6*
2X	*0.96*	*0.68*	*0.35*	*0.61*	*1.7*	*0.21*	*0.98*	*0.46*
3X					*0.18*	*0.13*		
H horizontal	**2.0**	1.6	1.3	2.4	**3.3**	0.97	0.54	1.4
1X	***1.4***	*1.2*	*0.87*	*2.3*	***2.2***	*0.52*	*0.49*	*0.56*
2X	*0.35*	*0.49*	*0.17*	*0.22*	***2.4***	*0.73*		*1.2*
3X					*0.40*	*0.25*		*0.13*
A, axial	0.91	1.2	-	1.6	3.2d	1.8	-	2.2
1X	*0.35*	*0.77*	-	*1.4*	***3.1***	*1.3*	-	*1.7*
2X	*0.37*	*0.78*	-	*0.51*	*1.1*	*1.1*	-	*1.3*
4X			-			*0.18*	-	

4. GR BEARINGS ABSOLUTE VIBRATION VELOCITIES AND STATOR VIBRATION

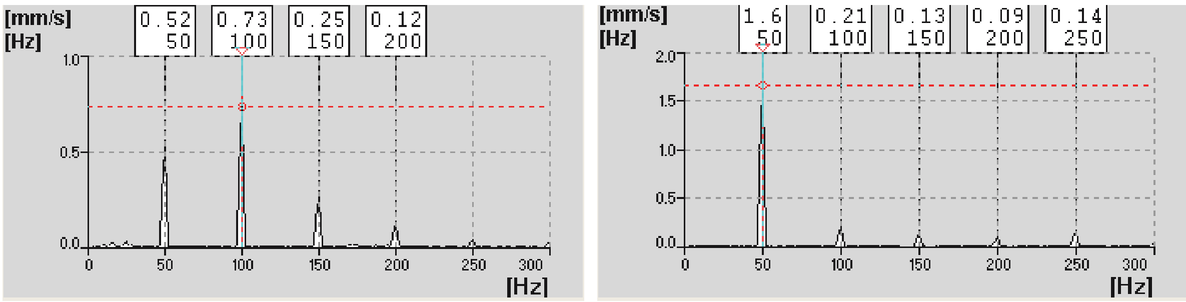
The generator's bearings absolute vibration velocity spectra have been measured in horizontal and vertical directions and results are shown in Fig. 4.1 and 4.2.

The 1X and 2X vibration velocity amplitudes are prevailing in both bearings vibration velocities spectra. The absolute vibration velocity of 1X fre-quency in vertical direction is greater than 2X frequency of 5th bearing.



Horizontal, V_{rms}=3.3 mm/s Vertical, V_{rms}=3.0 mm/s

Fig 4.1 GR 5th bearing absolute vibration velocity spectra in horizontal and vertical directions at 44 MW load



Horizontal, V_{rms}=1.76 mm/s Vertical, V_{rms}=0.97 mm/s

Fig 4.2 GR 6th bearing absolute vibration velocity spectra in horizontal and vertical directions at 44 MW load

The vibration velocity amplitudes in horizontal direction of 1X and 2X frequencies are approximately equal as shown in Fig.4.1. The main difference between relative and absolute vibration spectra are shown in Fig. 4.1 and Fig. 2.3c in vertical direction. The vibration severity is higher according to the absolute vibration parameters in comparison with relative vibration parameters. The 1X and 2X frequency bearing absolute vibration data is in contrary with 5th bearing shaft relative vibration displacement data as shown in Fig. 2.3c.

The vibration severity of 6th bearing is lower than of the 5th according to the absolute vibration velocity V_{rms} values. The 5th and 6th bearings shafts relative vibration displacements values are approximately equal. But difference between vibration parameters spectra is accented as shown in Fig. 4.2 and Fig. 2.3 d. Only 1X frequency vibration displacement amplitude is in horizontal direction (Fig.2.3d.) in comparison with horizontal vibration velocity spectrum in Fig. 4.2. The 6th bearing support has large mass in comparison with 5th bearing support mass and this phenomenon provides intensive damping of absolute vibrations.

The differencies in absolute and relative vibration spectra of 5th and 6th bearings have been explained after generator's stator vibration monitoring that shown in Fig 4.3.

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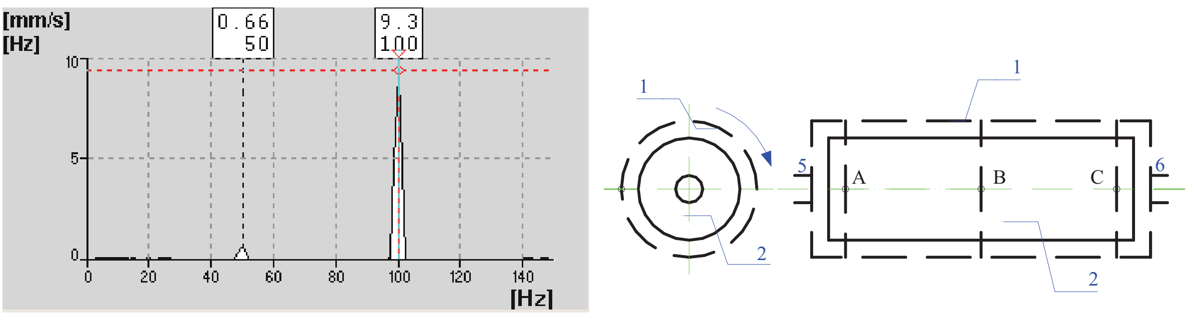

Point B vibration Measurement points A, B, C

Fig 4.3 Generator's Stator vibration velocity spectrum measured at the middle point B in horizontal direction at 44 MW load

In Table 3 the Stator's points A, B and C absolute vibration velocities measured in horizontal direction are shown. The midle point B provides the highest vibration severity. The 2X frequency vibration velocity amplitude 9.30 mm/s is excited by electromagnetic energy transformation mechanism in the electric generator –unbalance magnetic and centrifugal forces magnify the initial eccentricity, both static and dynamic

Table 3 Absolute vibration velocities 1X and 2X frequency amplitudes

Harmonics	Measurement points on Stator's case		
	A Turbine side	B	C Exciter side
1X=50 Hz	1.7 mm/s	0.66 mm/s	1.3 mm/s
2X=100 Hz	0.52 mm/s	**9.30 mm/s**	0.63 mm/s

The Stator vibration 2X frequency excites 5th bearing support 2X frequency vibration. This 5th bearing's support vibration is accting on proximity probe's attached point. The proximity probes SV5xy and SV6xy mounting locations are in the bearing support. This absolute vibration displacements acted on the Dual Radial Vibration proximity probes and 2X frequency displacements are changing 5th bearing orbit as shown in Fig. 2.3b.

5. CONCLUSIONS

1. The HPR bearings shafts provide the highest, not acceptable relative displacements peak-to-peak values S_{p-p} and maximum value of shaft displacement S_{max} that caused by the unbalance of the rotor.
2. The 1X=50 Hz frequency shafts relative vibration displacements amplitudes predominate in vibration displacement spectra of all bearings shafts except 5th bearing shaft.
3. The HPR and MLPR bearings absolute vibration velocities values V_{rms} are low. The absolute vibration values of HPR bearings supports are very low and are direct opposite in comparison with high relative vibration displacement values.
4. The vibration severity of GR bearings is high according to the absolute vibration velocities and this issue is in contrary with maximum values of shaft relative displacements S_{max5} and S_{max6}.
5. High power turbogenerator stationary machine condition monitoring system at first must provide shafts relative vibration measurements and after that - bearings absolute vibration measurements.

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