Application Software for Analyzing the Dynamic Stability of Hydro-aggregates

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Summary
At the Hydro-electric Power Plant Râul Mare-Retezat from Romania, there occurred a short-circuit to the poles 1 and 12 which caused the increase of the vibrations and temperatures above the allowed limit and the hydro-aggregate was taken out of function for investigation. To identify the increase of the vibrations during the commencement and to locate the defect there was used an acquisition equipment National Instruments and absolute vibration transducers type accelerometer. The technical status of the hydro-aggregate is considered accurate when the energetic level of the rotor assembly in motion is steady. The energetic steadiness of the rotor in motion is depending on the steadiness of the motor couple of the turbine which is given only by the actuating tangential forces. Under the terms when besides the useful tangential forces there also occur disturbing forces on radial direction, these shall disturb the rotor dynamics with consequences which sometimes may lead to very serious damage. The measurements, checking and analysis of the results, during a magnetic unbalance appeared to the hydro-generator from the Hydro-electric Power Plant Raul Mare Retezat, are described in this paper.

Key words:
Application software, Performance analysis, Stability analysis, Turbogenerators, Dynamic echilibru

1. Introduction

The rotary parts of the hydro-generators are characterized by both big weights and increased spinning speeds. It results that the rotor in motion shows a high mechanical kinetic energy and a magnetic potential energy. The technical status of a hydro-generator is considered corresponsive only when the energetic level of the rotor assembly in motion is steady. [8]. The energetic steadiness of the rotor in motion is conditioned by the steadiness of the motor coupling of the turbine which is only given by the actuating tangential forces. Under the terms when besides the useful tangential forces there occur also disturbing forces on radial directions, these shall disturb the rotor dynamics introducing instability of the rotation axis of the rotor with severe consequences which sometimes lead to very serious damage. The disturbing forces can be mechanical or magnetic. This issue of the dynamic steadiness of the rotor may lead to inaccurate operation of the hydro-generator and to the worst at its damaging. The circular oscillations of the shaft and the movement of the hydro-aggregate's shaft by one angle from the ideal axis of the hydro-generator are caused by the unsteady spreading of the revolving masses in axial and radial plans, the construction of the rotor poles with weight deviation (the unsteady spreading and arrangement on the polar crown), deviations from verticality of the thrust bearing's skate, the unsteady vertical deformation of the support spider, breaking of the shafts' line in the area of the coupling flanges of the shafts which cannot be reduced to zero.

2. Dynamics of the rotary parts

The assembly in motion (weight of the turbine rotor and of the hydro-generator produce one momentum which seeks to bring the shaft back in its balanced position. Their measure and direction depend on the constructive type of hydro-generator and has the formula:

$$M_G = (G_1 L_1 \pm G_2 L_2)\sin\varphi \approx (G_1 L_1 \pm G_2 L_2)\varphi , \qquad (1)$$

where: G1-rotor weight, G2-weight of the rotor's generator; $\sin\varphi \approx \varphi$, because they have very small value; (+)- in the case of suspended type (Fig.1).

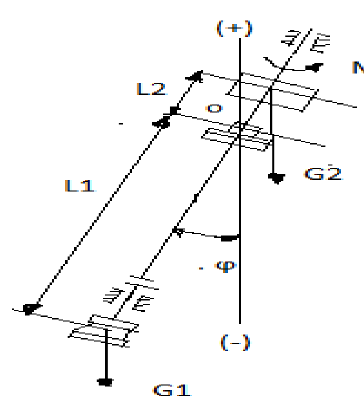

Fig. 1 Explanatory -concerning the deviation of the rotary assembly from the ideal axis

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The support of the thrust bearing and the supporting spider, due to its rigidity, causes a momentum of reaction which has the formula:

$$M_r = C_s \varphi, \tag{2}$$

where the rigidity of the thrust bearing's support and of the supporting spider has the formula

$$C_s = \frac{1}{2\left(\frac{\lambda_s}{N_s R_s^2} + \frac{\lambda_b}{N_b R_b^2}\right)}, \tag{3}$$

where: λ_s -resilience of the thrust bearing's support and of the dogging pins; λ_b-resilience of the supporting spider arm; R_s -arrangement radius of the pins; R_b -arrangement radius of the fixing pins of the spider arms; N_s -number of thrust bearing segments, respectively, of the dogging pins.

Consequently to the actions of the 3 categories of external forces during the rotation of the mass in motion there occur circular oscillations of the shaft which cause the vibrations form the radial bearings of the generator especially in that of the turbine [2]. In the case when the angle determines the exceeding of the clearances between the shaft's axle pin and the segments of the guiding radial bearings, the radial vibrations have unacceptable values, dry friction occurs and at the end, the seizure of the radial bearing and the taking out of operation of the hydro-aggregate. For a good operation of the hydro-aggregate, the condition that follows is imposed:

$$M_r > M_G . \tag{4}$$

And replacing with (1) and (2) is obtained:

$$C_s > G_1 L_1 \pm G_2 L_2 . \tag{5}$$

It results that in order to limit the circular oscillations and respectively the vibrations from the radial guiding bearings is necessary that the support of the thrust bearing and the support spider show accurate rigidity meaning it must be over dimensioned in comparison to the allowed limits of the mechanical stresses. Therefore, the thrust bearings on hydraulic and elastic support are not indicated out of this point of view, and the thrust bearing on a rigid support is the accurate solution. The vibrations of the radial bearing, which is in the support spider are worsen especially by the fact that the support spider takes over through the thrust bearings the entire vertical stress of the entire rotor of the hydro-aggregate. The simple amplitude of the vibrations from the guiding radial bearing which is located in the support spider has the formula:

$$y = K R_r \lambda kd = K' R_r \frac{1}{c_0}, \tag{6}$$

where: R_r -simple amplitude of the radial exterior force; λ-resilience of the supporting spider; c_0 -rigidity of the support spider; K_d -coefficient of amplification; K'–global factor.

It results that the vibration of the guiding radial bearing is determined by the insufficient roughness of the supporting spider which has the formula:

$$C_o = \frac{P_v}{f_{ts}}, \tag{7}$$

where: P_v -total vertical stress applied to the spider including to its weight; f_{ts} –complete arrow of the spider produced by the load P_v.

In order to ensure the allowed values of the vibrations it is recommended that C_o=350÷ 450 $[t/m]$, for $P_v \leq 1500$ $[t]$. The total roughness of the supporting spider together with the thrust bearing is:

$$C_o^{'} = \frac{C_o\, C_L}{C_o + C_L}, \tag{8}$$

where: $C_L = \frac{N_S\ P_{v1}}{f_{bs}}$ $[t/mm]$, rigidity of the thrust bearing; N_s- number of the dogging pins of the thrust bearing; P_{v1}-stress on one dogging pin; f_{bs} -total arrow of the pin and of the thrust bearing segment, produced by P_{v1}.

In order to avoid the taking out of operation of the hydro-generator because of the not allowed vibrations of the radial guiding bearing it is necessary to ensure an accurate roughness of the support spider and of the support of the thrust bearing, therefore it is necessary to over dimension accordingly, above the allowed limits of he mechanical stresses. [9]. Failure to ensure the *optimum* moment of gyration (GD^2)optimum, causes an increased revolution shock to the sudden projection of the nominal stress, which produces the besiege of the radial guiding bearings with vibrations shocks caused by the centrifugal forces of imbalance influencing subsequently the period of their operation [6]. The constructive solution to wedge the poles in the checked and indicated polar crown is that with tilted wedges arranged over the entire length of the blades by triggering the blades towards the rotor center, with disposition of the entire surface of the pole on the polar crown, and the number of the hammer shafts of the poles are indicated to be 1-2. The consolidation of the poles with short wedges applied only on the segment of the fastening plates or on a reduced length of the pole to push the pole towards outside, having the dogging of the pole on the polar crown only by means of the hammer shafts is not indicated. This solution does not have mechanical resistance to the tangential forces, it causes high vibrations on the pole which produce the damaging of the welds between the bars of the damping cage and cage ring, as well as damage of the connections between the excitation bobbins. The inner diameter of the polar crown increases by the revolution square once with its increase from the pre-set value to the accurate value which corresponds to

the sudden projections of nominal stress which, for the hydro-generators realized in the country is (1,3-1,5)$n_{n.}$. As the expansion of the polar crown is bigger than that of the rotor spider on which it is installed, there appears clearance between the two which triggers a mechanical and electro-magnetic unbalance influencing the functionality of the generator [4]. To avoid these clearances, the polar crown is installed with raising/shrinkage on the rotor's spider which is realized by radial and tangential wedging with 4 contact surfaces. By wedging, the polar crown remains stuck on the rotor's spider until the adopted detachment revolution, the polar crown does not become loose as it also has the tangential blocking, and through the number of the locking seats which is bigger than 4, the polar crown cannot be moved eccentrically.
Another important factor in the dynamic stability of the rotary parts is the magnetic unbalance which occurs subsequently to a short-circuit between the rotor bobbin spires to one or several poles. The putting in short between the spires of the bobbins of one pole of the hydro-generator's rotor creates unevenness of the rotor field which causes the increase of the vibrations and temperatures levels on the bearings at starting.

3. Application Software

The measurements have been carried out using an acquisition equipment National Instruments Compact RIO, with the following properties: 8 absolute vibration channels, 4 relative vibration channels, 1 revolution channel, 2 tension tri-phase scanners, current and powers, 8 channels for process parameters, simultaneous sampling, resolution 24 bits, acquisition speed 50 kS/s/ch, adjustable source by 300 V, 400 A. The equipment has been provided with Fast view acquisition software, analysis, recording and permanent display of the signals, the analysis of the vibrations and of the technical parameters, dynamic balancing through the method of the influence parameters [5]. The absolute vibration transducers type accelerometer HS100Fau sensitivity 500 mV/g and working frequency 0.2 Hz-10 kHz. Relative vibration transducers type proximately PRS04, have: sensitivity 4 mV/µm, measurement field 0-4 mm, working frequency 0-10 kHz. Revolution transducer QS30LDQ is laser type, having the working frequency 0-2 kHz. Magnetic field transducer CCHAPT has the output 0-300. The name and description of the parameters is given in Table 1. In Romania, at present this system isn't homologated [1, 3], although its usefulness is obvious [7, 10, 11].

TABLE 1. NAME AND DESCRIPTION OF THE PARAMETERS

Name	Description
AbsLRA-X	Absolute vibrations radial-axial direction minus X
AbsLRA-Y	Absolute vibrations radial-axial direction minus Y
AbsLR-X	Absolute vibrations radial bearing minus X
AbsLR-Y	Absolute vibrations radial bearing minus Y
RelLRA-X	Relative vibrations radial-axial direction minus X
RelLRA-Y	Relative vibrations radial-axial direction minus Y
RelLR-X	Relative vibrations radial bearing direction minus X
RelLR-Y	Relative vibrations radial bearing direction minus Y
Uflux	General tension flux transducer
UfluxE	General tension flux transducer undergoing the demodulation operation (analysis of the blanket)
Tacho	Signal generated by the revolution and phase transducer

Or the measurements carried out drily, without excitation tension, the revolution sensor located at minus Y, the reflecting sound quality fixed on the generator of pole 7, and the results have been graphically represented in Figures 2, 3, 4 and 5, out of which it can be noticed that the level of the vibrations, in the non-excited regime is reduced and corresponds to a normal operation condition.

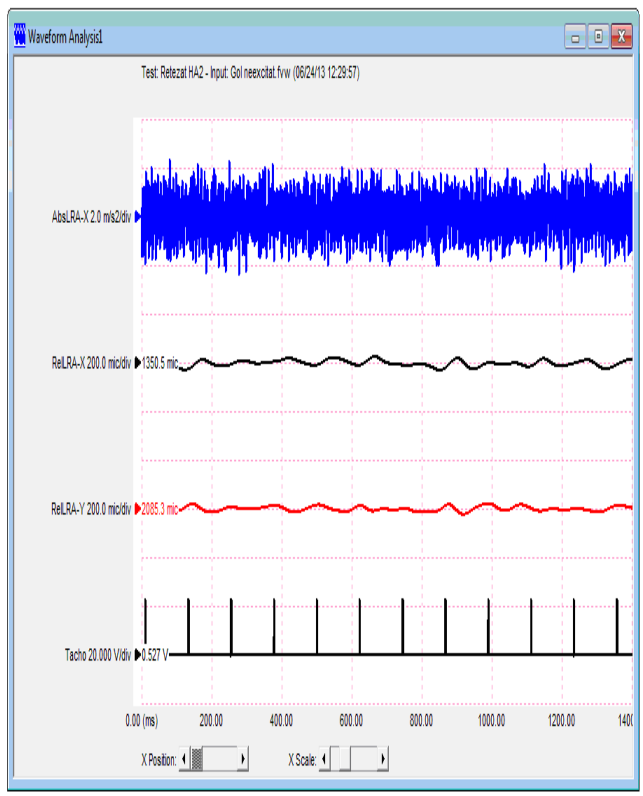

Fig. 2 Wave shapes of the vibrations, radial-axial bearing, drily, without excitation tension

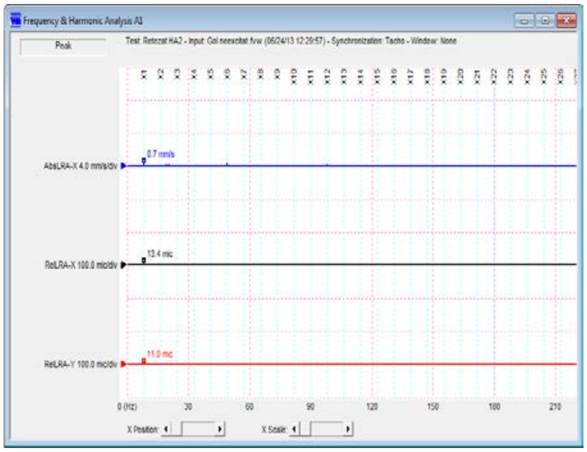

Fig. 3 Frequency specters of the vibrations, radial-axial bearing, drily, without excitation tension

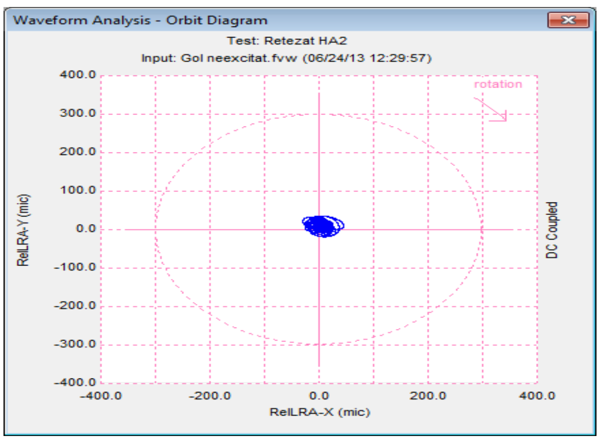

Fig. 4 Orbit diagram of the vibrations, radial-axial bearing, drily, without excitation tension

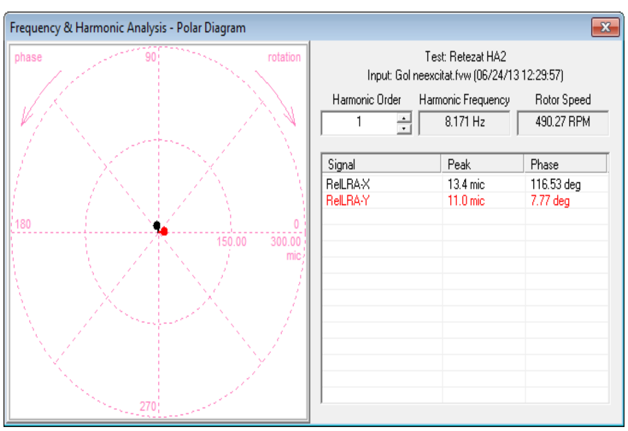

Fig. 5 Polar diagram vibrations, radial-axial bearing, drily, without excitation tension

For the measurements carried out drily, with the excitation tension according to the regime 16.3 kV, the sensor of revolution is located at minus Y, the reflecting sound quality fixed on the generator of pole 7, and the results have been graphically represented in Figures 6, 7, 8 and 9, out of which it is noticed the fact that the level of the vibrations in the excited regime exceeds by far the allowed limits. The phase of the vibrations is steady, which is characteristic to a steady angular position force. The trajectory is circular, which is characteristic to a movement without shock or friction, generated by a steady amplitude force and variable direction with a frequency which is equal to the revolution frequency.

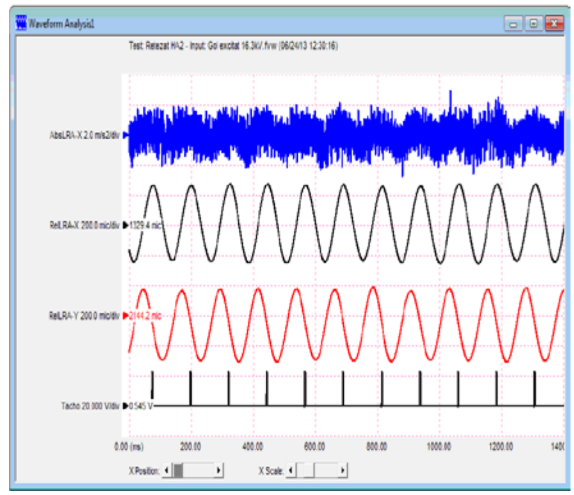

Fig. 6 Wave shapes of the vibrations, radial-axial bearing in dry, excited.

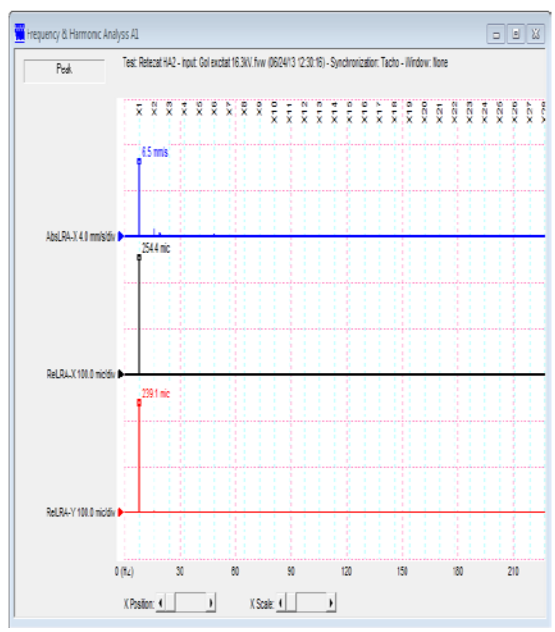

Fig. 7 Frequency specters of the vibrations radial-axial bearing in dry, excited

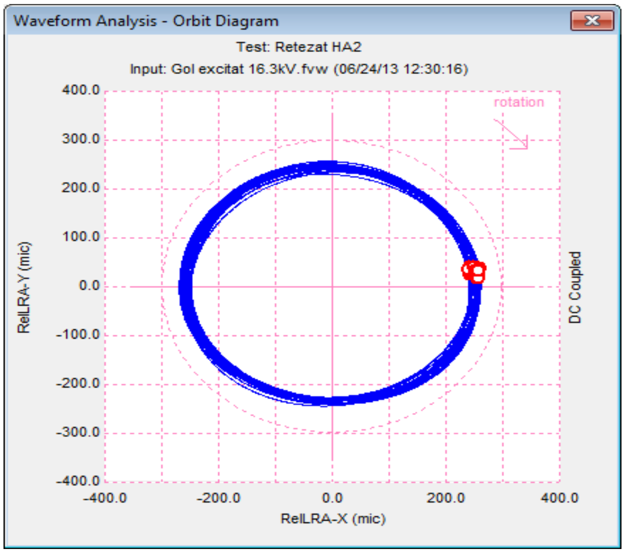

Fig. 8 Orbit diagram of the vibrations radial-axial bearing in dry, excited

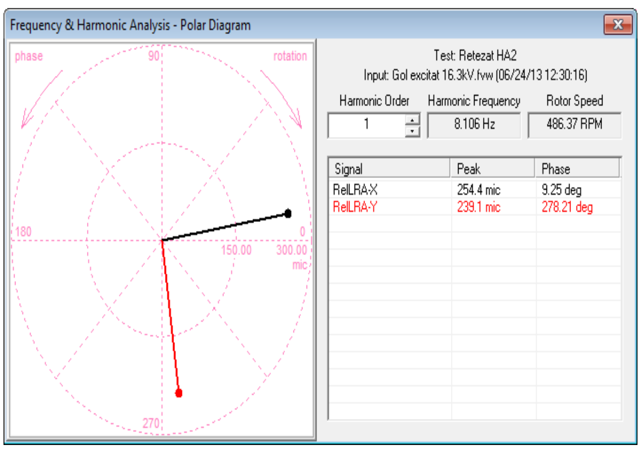

Fig. 9 Polar diagram of the vibrations radial-axial bearing in dry, excited

For the measurements carried out drily, excited, the monitoring of both of the bearings, the sensor of revolution is located at minus X, the reflecting sound quality fixed on the generator of pole 7, and the results have been graphically represented in Figures 10, 11, 12 and 13, out of which it is noticed the fact that the level of the vibrations of the two bearings are of amplitudes which are equal in phase which is characteristic to a disturbing force spread even over the entire height of the rotor.

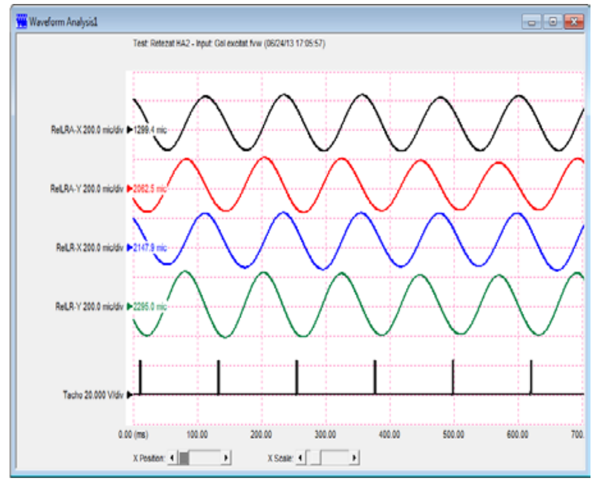

Fig. 10 Wave shapes of the vibrations, radial-axial bearing and radial bearing, dry, excited

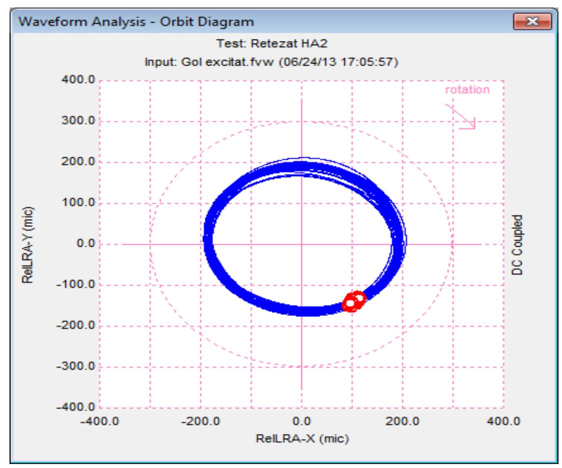

Fig 11 Orbit diagram of the vibrations, radial-axial bearing dry, excited.

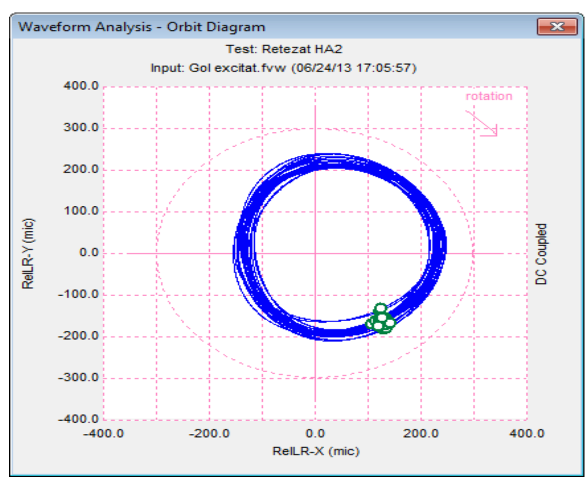

Fig. 12 Orbit diagram of the vibrations radial bearing, dry, excited

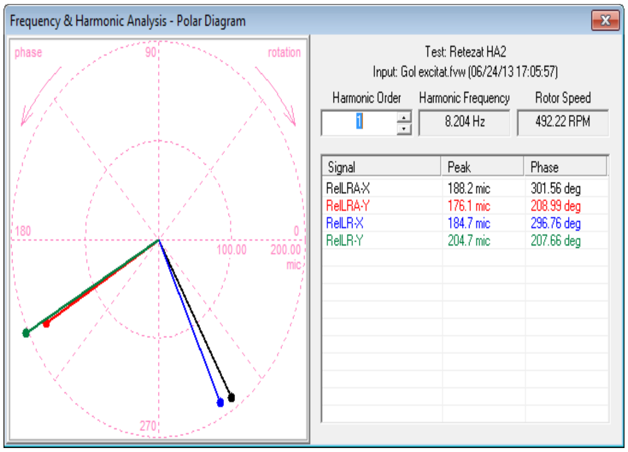

Fig. 13 Polar diagram of the vibrations radial-axial bearing and radial bearing, dry, excited

In order to check the dynamic air gap, a current of 100 A was applied to the rotor circuit, in stationary regime and the evolution of the axis in the bearing was monitored, according to several angular positions, and the results have been graphically represented in the Figures 14, 16, out of which it can be concluded that the rotor has a direction of movement independent from the angular position, which is characteristic to the presence of a stationary air-gap and lack of a dynamic air gap.

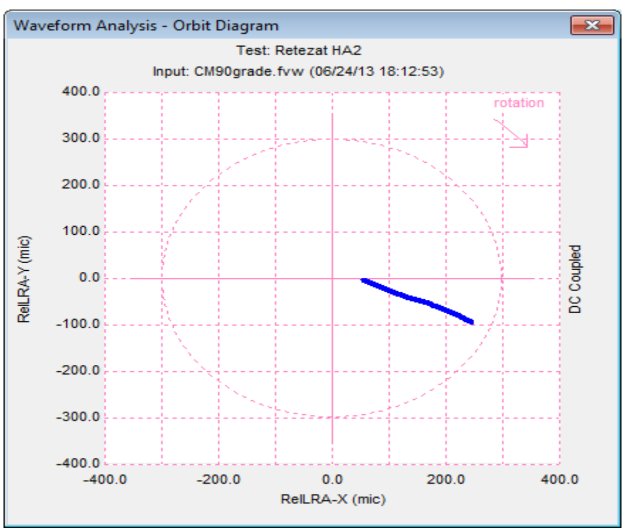

Fig. 14 Movement of the axis in the bearing under the action of the stationary magnetic field (rotor position 90 degrees)

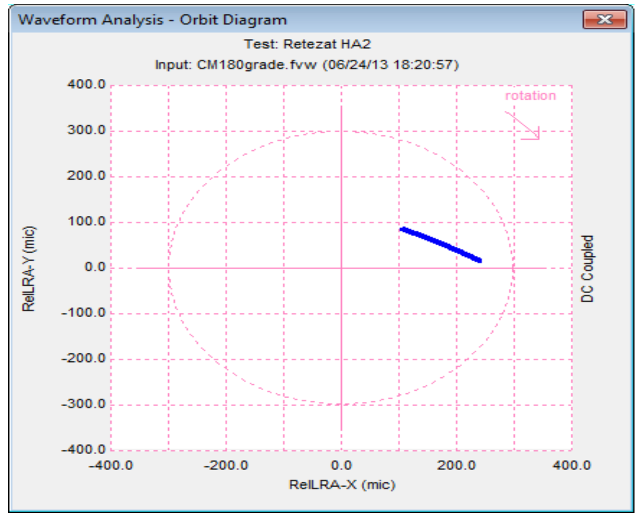

Fig. 15 Movement of the axis in the bearing under the action of the stationary magnetic field (rotor position 180 degrees)

For the measurements carried out drily, excited, the usage of the magnetic flux transducer, the revolution sensor is located at minus X, the reflecting sound quality fixed on the generator of pole 7, the magnetic field transducer is located in the quadrant minus Y plus X, at 30 degrees from minus Y and the results have been graphically represented in Figures 16, 17, 18 and 19, out of which it is noticed the fact that the signal generated by the magnetic flux transducer indicates an unevenness of the rotor magnetic field. The analysis of the angular position of this unevenness led to the detection of the short-circuit at the level of the poles 1 and 12.

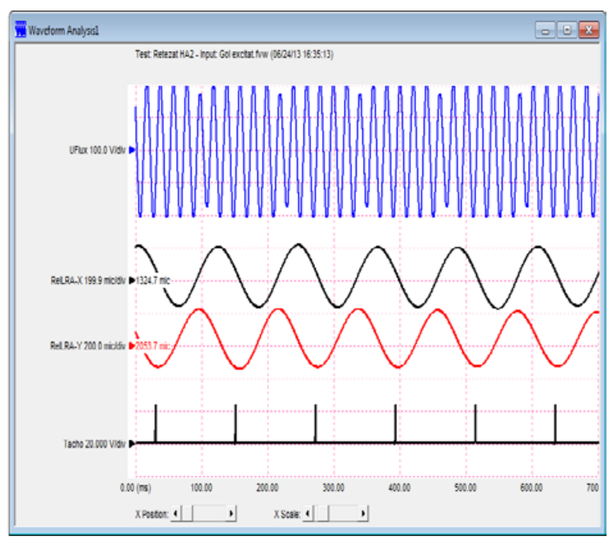

Fig. 16 Wave shapes flux signal transducer and vibrations axial-radial bearing, dry, excited

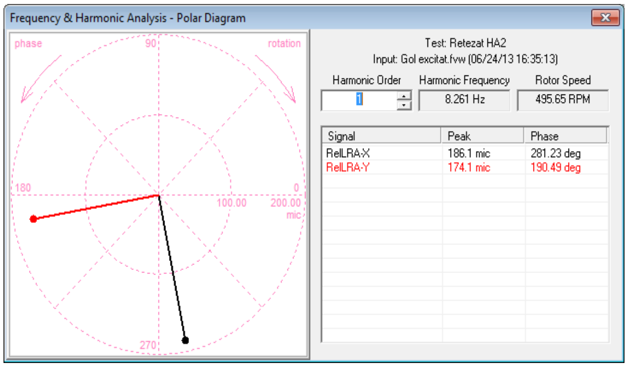

Fig. 17 Polar diagram and vibrations axial-radial bearing, dry, excited

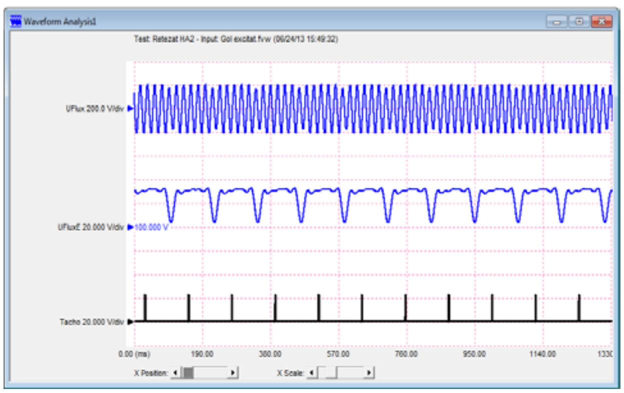

Fig. 18 Wave shapes signal magnetic flux transducer and blanket magnetic flux transducer signal, dry, excited

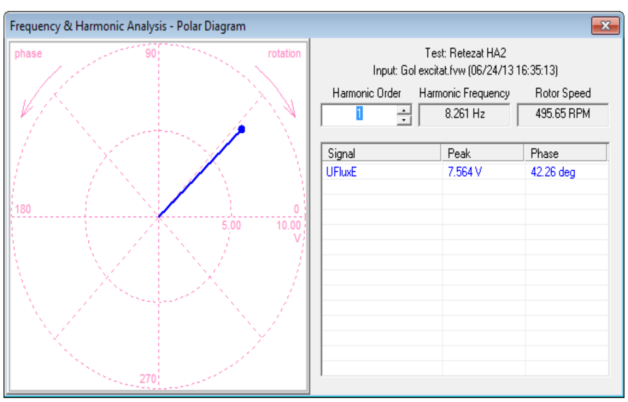

Fig. 19 Polar diagram of the blanket of the magnetic flux transducer's signal at a preset revolution dry, excited

For the measurements carried out drily, with the excitation tension according to the regime of 16.3 kV, after the removal of the vibration causes, the results have been graphically represented in the Figures 20 and 21, out of which it is obvious that the level of the vibrations, in excited regime, has been reduced and corresponds to a normal operation condition.

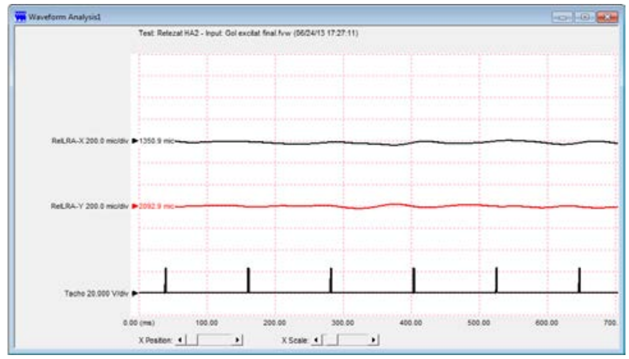

Fig. 20 Wave shapes of the vibrations, radial-axial bearing, dry, excited

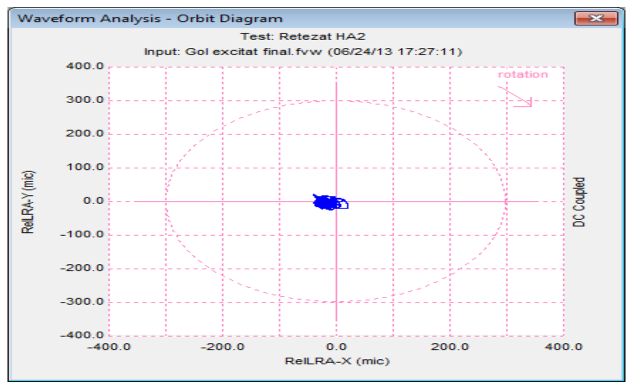

Fig. 21 Orbit diagram vibrations, radial-axial bearing, dry, excited

Consequently, the increase of the vibrations by far above the allowed limits was caused by the presence of two short-circuits at the level of the spires of pole 1 and respectively pole 12. The short-circuits had been produced over a certain revolution which prevented their detection through electric measurements or through inspecting the dynamic air gap. After the removal of the short-circuits, the installation can be operated normally.

4. Conclusion

An important factor in the dynamic steadiness of the rotary parts, of the hydro-generators is represented by the magnetic unbalance which occurs subsequently to a short-circuit between the spires of the rotor bobbin, to one or several poles. The putting in short between the bobbins' coils of one pole of the hydro-generator's rotor creates unevenness of the rotor field which triggers the increase of the level of vibrations and temperature on the bearings at starting. It is imposed to install a data acquisition system and its analysis in operation, to monitor the parameters which characterize the technical status of the rotor in motion, respectively the excitation current through the rotor winding, supply voltage of the excitation coils of the rotors, temperature of the excitation coils of then rotor bobbins and the absolute vibrations on the three rotor axes.

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