Zivoslav Adamovic[1],
Ljiljaan Radovanovic[2]

THE RESEARCH OF VIBRATION LEVEL EFFECT ONTO THE DAMAGE OF THE TURBOGENERATOR ENGINE ROTORS

The amplitude of vibrations and the phase angles and their changes when the rotor of turbogenerator has cracks are researched in this paper. This paper deals with two kinds of generator rotor's damage. In the first case of damage of turbogenerator rotor, the cracks appeared at the end of the cogs of the rotor's active part. In the second case cracks occurred in the middle of the cross section of the shaft, on the exciting side at the end of the shaft.

Introduction. Two kinds of the damage that have appeared on the generator rotors in a thermoelectric power plant, are analysed in this paper.

The signs of such damages are drawn from the vibration analysis and loading is also pointed out at which it is necessary to pay special attention to these signs during exploitation process of turbogenerator.

The arrangement of the turbogenerator rotors is presented in Fig. 1. There are totally nine rotor places; the nominal number of revolutions is 3000 1/min. The damages have appeared on the exciting side of the generator rotor.

The vibration analysis. For the usual vibrations analysis there is a vibrations acceptor for vertical direction at each of nine bearings, and with its help the maximal amplitudes of the vibration path are measured. To get more information about vibration behaviour in the initial period and to compare the measuring results with calculations, there are installed the vibrations acceptors for horizontal and vertical direction, as follows:

— at every bearing;

— at both shaft ends of the generator turbine of high pressure, as well as of four generator turbines of low pressure (Fig. 2).

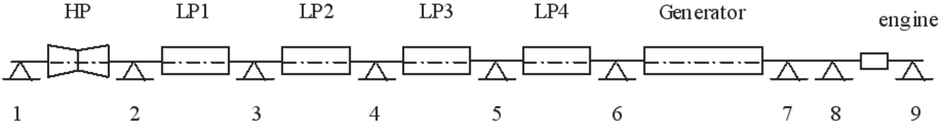

Fig. 1.The arrangement of the turbogenerator rotors:
HP rotor with high pressure turbine, LP rotor with low pressure turbines

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[1]Zivoslav Adamovic — Professor of University of Novi Sad Technical faculty “Mihajlo Pupin”;

[2]Ljiljaan Radovanovic — Teaching Assistant of University of Novi Sad Technical faculty “Mihajlo Pupin”.

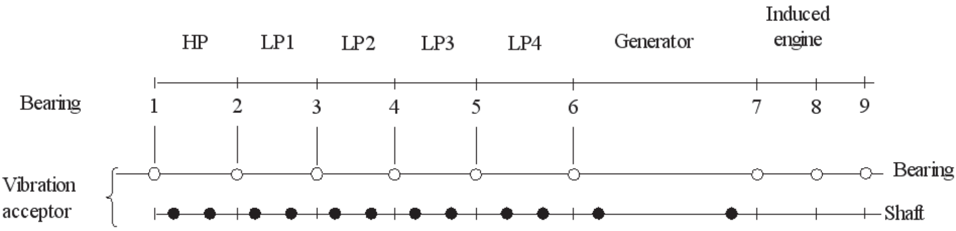

Fig. 2.The additional measuring points for starting (into operation) the bearing vibrations (horizontally and vertically); the shaft vibrations (horizontally and vertically) HP — the rotor of high pressure turbine; LP — the rotor of low pressure turbines

The cracks have been found at the cog ends at the active parts of the rotor 1 after the operation for 3200 hours and of the rotor 2 after 7600 hours. Since we could expect with great probability that on the new rotor some cracs would appear, there was applied a vibrations control system in order to ensure further work. The system has the following characteristics:

— the signals from the measuring points sketched in Fig. 2 can be registered and processed in computer. Those signals are analysed by determining amplitude and the phase angle of vibrations n and $2n$;

— the measuring values can be preset on the screen, i.e. registered as the table of measuring values. The tables contain the effective values as well the amplitude values of the vibrations n and $2n$ at any measuring point. So received abundance of information comprises the values of 20 vibrations of bearings and 32 vibrations of shafts (see Fig. 2);

— the amplitude values or phase angles in xy-presentation can be obtained as the function of revolutions number or time;

— the amplitude value and phase angle of vibrations n or $2n$ in polar diagrams are determined with the help of point P (Fig. 3). The distance of the pole is the measure for the X amplitude, and the polar angle is identical to the phase angle φ, the revolutions number or time are the parameters in this diagram. The normal state is determined on the ground of the operation experience and marked with P_0. The concentric circles around that point indicate when there is the state of alarm or when the machine needs to be stopped (Fig. 4).

The value $\Delta A/\Delta t$ is set as a trend where $\Delta A = A(t_2) - A(t_1)$ and marks amplitude difference during the time difference $\Delta t = t_2 - t_1$.

For short time and long time trends the differences of 1 hour and 6 days are chosen.

The vibrations of the rotor with cracks as function of anglular speed. Here, we shall use a simple model according to Fig. 4 (Lavall's shaft).

The rotor is loaded by its weight G and non-balanced force $F = m_e\Omega^2$. If k_0 is the stiffness of the rotor without a crack, then the weight produces static bending $x_s = G/k_0$. The non-balanced force produces the circular motion of the shaft middle around the point of its static bending with

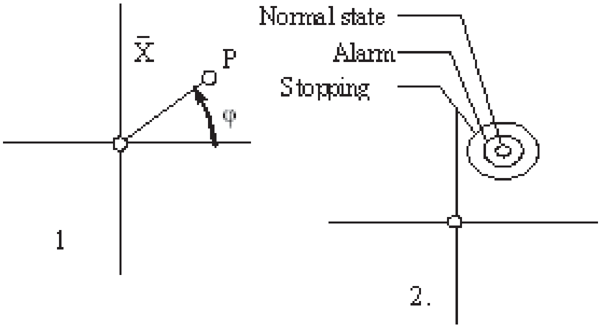

Fig. 3. The polar diagram of the vibration n and $2n$:
1 — the principle sketch; 2 — work areas

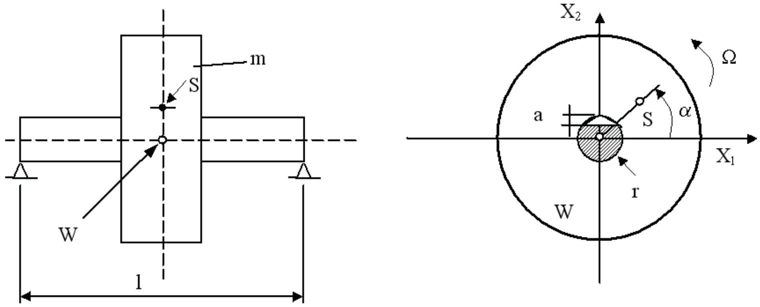

Fig. 4. Shaft with crack at $l/2$

the frequency Ω and the radius y. Here a vibrations acceptor registers harmonical motion with the frequency Ω and the amplitude y. Further the main moments of inertia are different, $L_2 \neq L_1$. At the initial, still small cracks stiffness reduces only a little, depending on crack growth. So, e.g. the hardness of the shaft sketched in Fig. 4 at the supposed depth of the cracks $t = 0.20r$ in the directions 1 and 2 is $k_1 = 0.999k_0$ and we can deduce that the influence of the crack on vibration behavior is still small.

If the surface of the crack is affected by tractive power then the effect of the crack is maximum because its surface doesn't endure tractive powers. In that case it is the crack which "breaths" unlike "open" crack, which is always like that. The "breathing" of the crack depends on the forces conditioned by the mass of unbalanceness.

Theory with the time dependant coefficients shows that the shaft's center describes circular track, which has similarity with movement Ω and partially with 2Ω and 3Ω (angular speed $\Omega = 2n$, n — number of revolutions).

For example, in Fig. 5 is shown one flow with the supposition that depth of the crack and angular speed $\Omega = w_k 3$, where w_k stands for the first own frequency of the shaft without crack.

Shaft's gravity is for $0{,}030x_i$, eccentric in direction of the axis $2\alpha = 90°$. The middle movement is characterized by the point P with coordinates 0 and $1.010x$. Conditioned to this point shaft's center has been moved for $z_1(t)$ and $z_2(t)$. With the assumed degree of muffle $D = 0.05$ maximum movement is $0.0430x_i$ (Fig. 5).

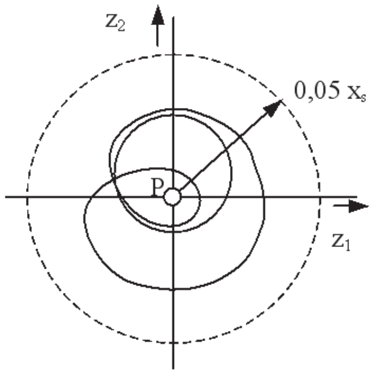

Fig. 5. Shaft's central track, depth of the crack $t = 0,2r$, eccentric $e = 0,030x$ and $2\alpha = 90°$

In Fig. 6 maximum vertical movement of the shaft is shown as the function of the angular speed, where eccentric of the gravity appeared first on the side of the crack, and then on the opposite side. Compared to the shaft without crack there are following differences:

— near ω_k as its first frequency, amplitude maximum is dependant on the position of the eccentrics;

— there are additional resonant spots in frequency scope at $\omega_k/2$ and $\omega_k/3$ (ultraharmonical resonance).

Movement is similar in horizontal direction. At any angular speed the movement, therefore, consists of harmonics with frequencies Ω, 2Ω and 3Ω:

— first harmonic at $\Omega = \omega_k$,

— second harmonic at $\Omega = \omega_k/2$,

— third harmonic at $\Omega = \omega_k/3$.

The phase angle of harmonics changes fairly fast for 180° inside the frequency scope where the harmonics prevail (see Fig. 6).

Cracks expansion normally increases as well as the moving amplitude of the first harmonics, but it is also possible the amplitude to be decreased, when eccentricity appears in a definite scope on the side opposite to the crack ($\alpha \approx 270° \pm 45°$). As soon as the crack appears, or it increases, the phase angles of harmonics are changing.

After the cracks appeared on the exiciting side of the generator rotor, the measured amplitude values of the shaft vertical vibrations were observed near the bearing 7 (Fig. 7). Characteristically, n vibration only near the first own frequency presents a little resonant place. Since the moments of (mass) inertia of the generator rotor are not totally equal, so without any cracks, there appears the vibration $2n$ with one resonant place at $n_1/2 = 340\,\text{min}^{-1}$ and at $n_2/2 = 910\,\text{min}^{-1}$. There are no resonant places at the nominal revolutions number of $3000\,\text{min}^{-1}$, and the amplitudes of $16\,\mu$m, i.e. $10\,\mu$m, are very low.

If the shaft had a crack, according to the theory it would follow that

— the amplitudes of the oscilations n and $2n$ had to be increased respectively, i.e. to be reduced,

— the phase angle had to be changed, and

— the oscilation $3n$ had to appear especially at $n_1/3$ (Fig. 7).

The measuring of cracks and vibrations. After longer work in the active part of the rotor, on the cogs different cracks appeared.

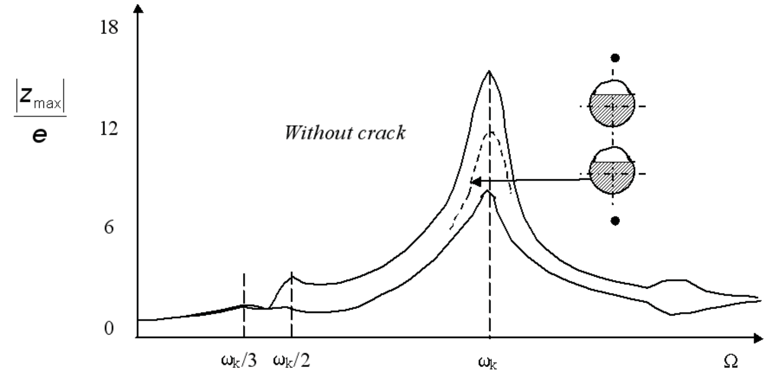

Fig. 6. Maximum vertical amplitude of vibrations as the function of the shaft's anglular speed (Laval-shaft with $t = 0.20r$, $e = 0.030x$, degree of muffle $D = 0.05$)

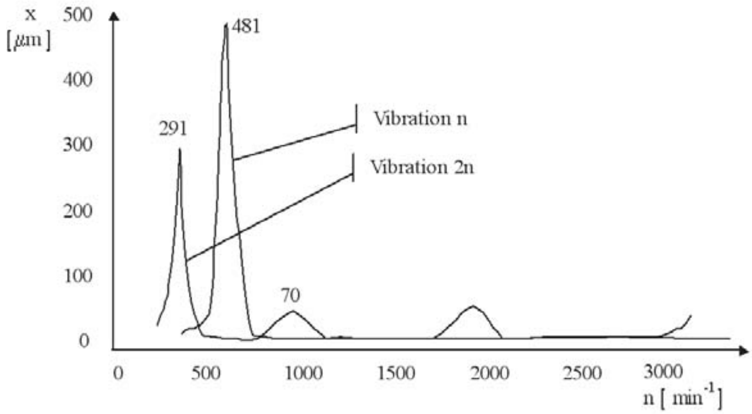

Fig. 7. Vibration amplitude n and $2n$ in vertical direction, at the induced side of the generator

At first in corners, then they expanded in the axis direction, and after that they appeared in the middle of the engine.

The cracks apeared because of fatigue. They appeared first in corners because of tension high concentration formed since the groove ends for a length of about 30 mm were fairly roughly worked because of better isolation attaching. It is apparent that the allowed tension values with the rotor were exceeded. Those damages were repaired so, that the material was scraped, and then the cracks disapeared.

To protect at least partial running operation, there needed to be used the spare rotor. It had then up to 700 working hours. There weren't any cracks on it. It was expected to find similar cracks on the rotor like at two others, after some time. It was decided to be stopped after 45 days due to security reasons and to be checked. The found condition of the rotor would be authoritative for the further working periods. Besides that, special steps used in checking were discovered and this allowed finding cracks and their spreading without disassembling the rotor. To find out how deep cracks are, without endangering reliability at work, some mechanical calculations were done. During the process of starting and stopping at the end of the cogs a certain difference is appearing in the tension $\Delta\sigma = 700\ldots760$ MPa in radial direction (yield, strain border > 700 MPa, resistance to tearing > 800 MPa). This calculation has shown that with taking into consideration small radii at the bottom of the groove, some cracks may appear and their sizes are measured by tenth of millimeter. At the critical area, during the work at 3000 min^{-1}, there is radial tension of the values $\sigma_r = 85 \pm 15$ MPa at zero load and $\sigma_r = 125 \pm 15$ MPa at maximum load. This calculation with some of the assumptions has shown that zero load crack is growing when its depth is 0.7 mm; at full load this value is 0.12 mm. During the first inspection, after 55 days, first cracks could have been noticed. After fourth stopping, cracks existed on 10 cogs. Depth of the cracks was between 24 and 42 mm (Fig. 8). After that, this rotor was put out of work because two other were repaired.

In Fig. 8 you can see the depth enlargement of the cracks as the function of work duration (time). Therefore we consider that the crack growth was very little during the work lasting 12000 hours. The measured crack depth was 52. . . 58 mm. The vibration behaviour of the turbogenerator was good, the influence of normal changes of the drive parameters onto the vibration behaviour was known.

Figure 9 presents the state of vibrations at the measuring places where the biggest changes have been recorded. They are the vertical vibrations on the exciting side and the found changes could be normally considered as non-substantial, but taking into consideration the situation, it has been decided that the rotor should be put out of oreration. The vibrations during the machine halting slightly differed from those of the previous cases, among them of course there were differences too.

The checking after the disconnection however showed that some cracks very fast expanded during the last working period. Once it has been found that some of them have the depth up to 230 mm (see Fig. 8), this rotor is thrown away as an inapplicable one.

The alarm trend of vibrations. The cogs cracks in the rotor 1 were removed when its active part shortened and after that the rotor was again switched on. The vibration behaviour of the turbogenerator was controlled with the same attention as earlier when we had worked with the rotor 3. During the next months the turbogenerator worked under full loading.

It has been noted that the generator vibrations constantly change in one direction, in spite of the constant working conditions. It is important especially for vertical vibrations on the turbine side (Figs. 10 and 11). The same tendencies have also the other vibration values of the generator. Without taking into consideration the fact that the vibration level was not high, the changes of the amplitude and phase angles were extreme and decision was made to put turbogenerator out of work. Its behaviour during the process of stopping confirmed this decision, because the machine was vibrating while it was passing through different phases of number of

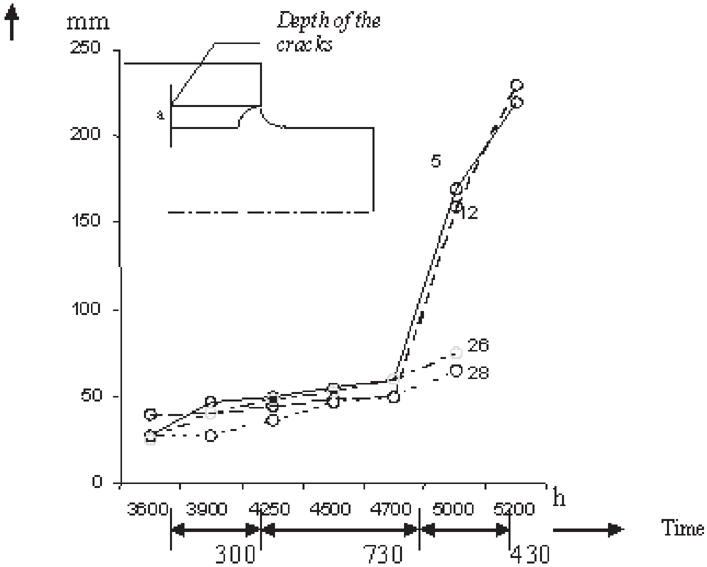

Fig. 8. Depth of the cracks as the function of work duration

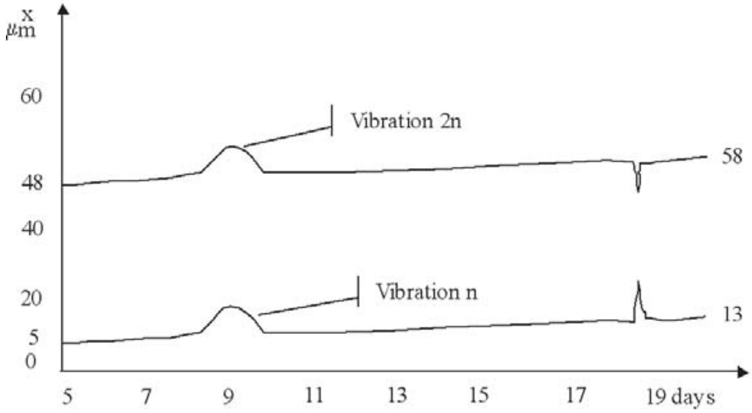

Fig. 9. The vertical vibrations of generator shaft on the exciting side

revolutions. Rotor grazed several gaskets. Besides, that rotor could revolve for further 30 hours but it was observed that it was distorted for 2.5 mm. When it was stopped, a crack was found, which was about 98 mm from the bearing impulsive side and expanded for half of the perimeter. When the surface of the crack was opened, it was found that half of the cross section was broken (Fig. 12).

As a result it is was confirmed that vibrational behaviour shown in Figs. 10 and 11 is characteristic of the rotor with cracks but there is no assurance that cracks exist in the rotor if it behaves like that. Further informations are given by the curve of shutting down (Fig. 13). They show behaviour of this turbogenerator when it was shut down after 50 days and 12 days before it was put out of work. It was observed that for the last 12 days critical number of revolutions $n_2/2$ moved from 950 to 960 and that

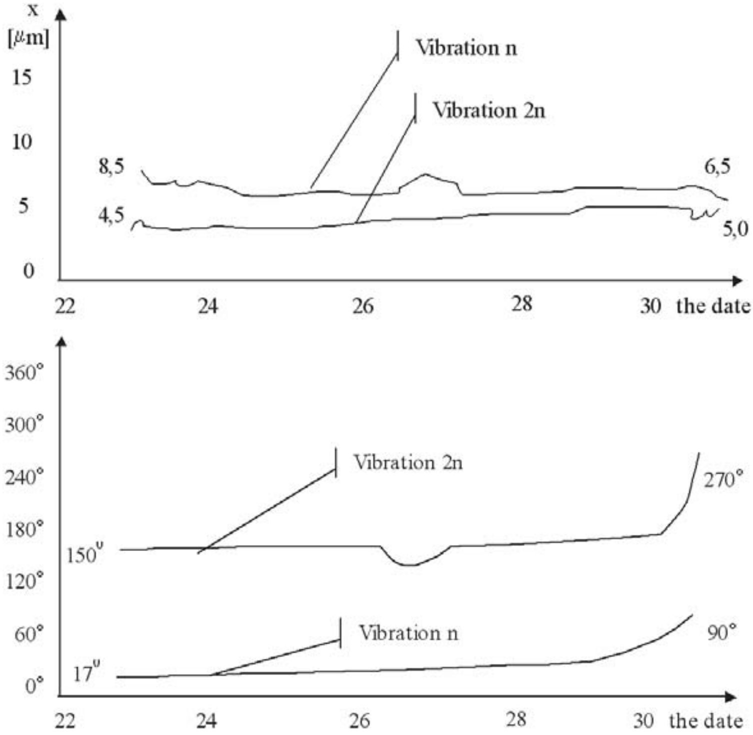

Fig. 10. The vertical vibrations of generator bearings bedplate on the turbine side

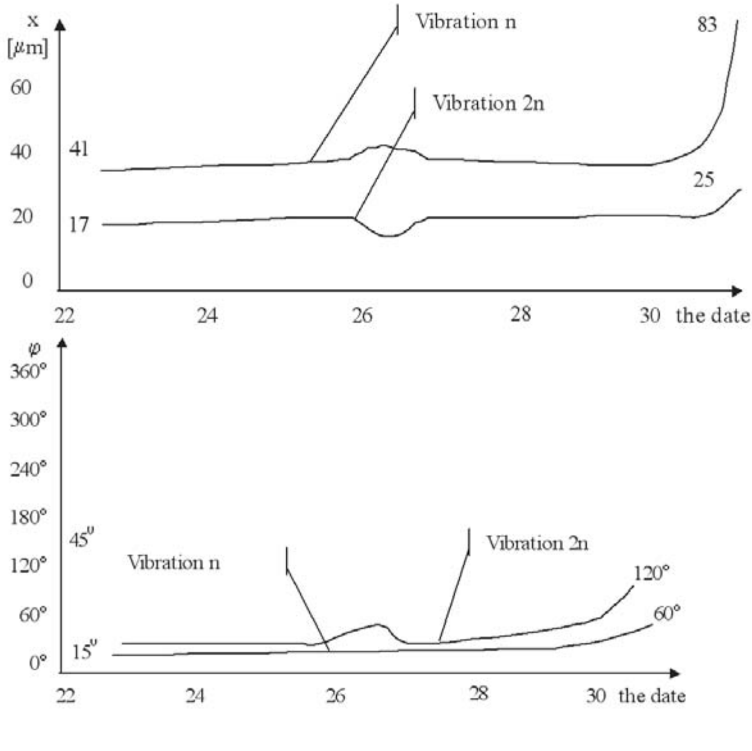

Fig. 11. The vertical vibrations of the generator shaft on the turbine side

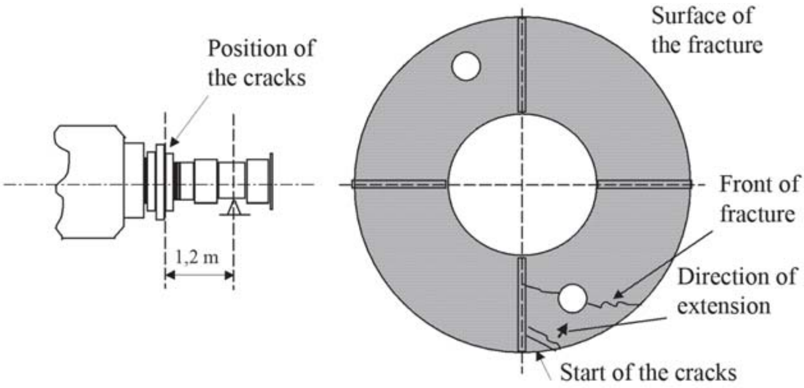

Fig. 12. Position (left) and surface of the cracks (right)

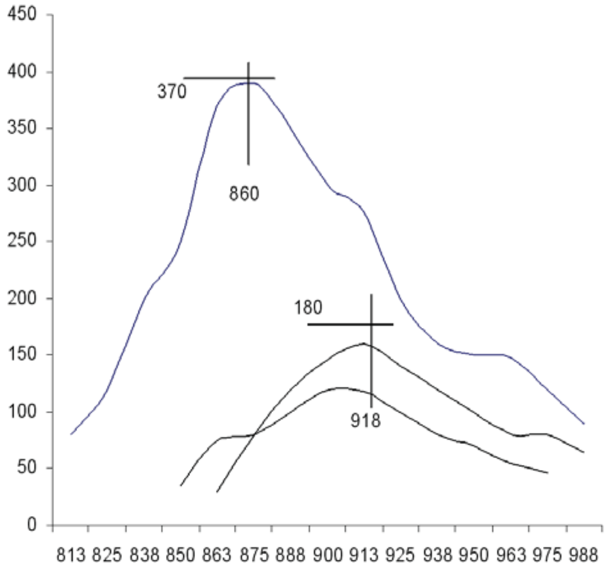

Fig. 13. The curve of shutting: vibration $2n$ of the shaft on the vertical turbine side, with resonance on $n_2/2$

maximum value of vibrational amplitude from previous 180 increased to 370.

No reason was found for this crack. Material of the shaft (dynamical resistance) didn't have any faults. At the exit point of the crack notch factor was calculated and it was 3.7. If we take into consideration this and decrease it due to roughness of the surface and effect of the dimensions and probability of lower values of the material, then we allow alternating tension of 52 MPa. With nominal alternating tension $a = 28$ MPa, factor of security is 52/28.

The crack has not appeared at the spot of the highest nominal tension, which is 270 mm in axial direction toward the rotors center, and it has nominal tension at 33 MPa and factor of security is calculated at 1.8.

Conclusion. In this paper two kinds of damage of the same type of generator's rotors are considered. In the first case of damage of turbogenerator rotor, the cracks appeared at the end of the cogs of the active part of rotor. In the other case on the exciting side at the end of the shaft cracks appeared in the middle of lateral there cracks.

When the first mentioned cracks became visible, it was necessary to put in work the third rotor.

It was expected that the cracks would become visible in short or longer period of time. The vibrations were carefully watched.

The amplitude of vibrations and the phase angles changed when the rotor of turbogenerator had cracks.

Mechanical inquires at the breaking point have shown that the starting crack with depth of 0.60 mm at the above conditions spreads during the work. Possible explanation for the mentioned crack is that it existed mechanically while it was still in the production of the rotor.

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