Monitoring of rolling bearing failures as result of acceleration

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Abstract. The article discusses modern methods of diagnosing and technical diagnostics of irrigation pumping units to ensure reliability during operation. Using new designs of pumping power equipment and developing new operating modes provide for improving the mode of pumps based on improved diagnostics, significantly saving operating costs. Together with this provides detection, control, and prediction of multi-stage failure of the rolling bearing by monitoring the vibration trends of the centrifugal pump unit during operation.

1 Introduction

For pumping in large volumes, with different parameters of pressure and water flow, electric-driven pumping units of various designs and purposes are used.

In Uzbekistan and abroad, the production of pumping units is concentrated at large enterprises that produce the entire range of units in demand in the water industry. Of the variety of pumping units, the most common type can be distinguished: centrifugal pumping units.

The demand for this type of pump is due to its simple design, low metal consumption, and high efficiency. The theory and design of centrifugal pumping units have reached a high level of development thanks to the use of electric drives and rolling bearings.

The reliable operation of a centrifugal pump unit depends on many factors. The electric motor is a reliable link in the composition of a centrifugal pump unit, the operating conditions of which are debugged and normalized. Another important factor in the trouble-free operation of a centrifugal pump unit is the reliability of rolling bearings.

Centrifugal pumping units account for more than 70% of the total number of dynamic equipment units controlled by monitoring and diagnostic systems for water management. In contrast to plain bearings, features of the operation of rolling bearings are the presence of constant mechanical contact between its parts. Even though the resource of rolling bearings, incorporated in the manufacture, allows them to operate for a long time, practically without being subjected to significant wear, fatigue phenomena accumulate in the parts of the rolling bearing, ultimately leading to fatigue failures.

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Currently, in most cases, condition monitoring is carried out without considering the possible non-stationary nature of the development of faults, which creates the problem of ensuring control of the state of a centrifugal pump unit during operation to prevent sudden failures of the units. Therefore, developing new and existing methods for monitoring the condition, diagnosing, and monitoring a centrifugal pump unit, taking into account the identification, control, and prediction of damage accumulation processes in parts and multi-stage failure of rolling bearings, is an urgent task.

This work is devoted to the study of the processes of multi-stage failure of the parts of the rolling bearing of a centrifugal pump unit, i.e., repeatedly occurring self-eliminating failures of the same nature - intermittent failures, establishing the dependences of vibration changes in the degradation process, developing control methods and developing methods for identifying and predicting staged failure processes.

The failure of a centrifugal pump unit due to fatigue failure of a rolling bearing does not occur suddenly and not instantly. First, some signs of the approach of this process appear, the nature of the vibroactivity of the aggregate changes. Against the background of a stable vibration level, some outliers of vibration trends appear. The study of the sequence and intensity of these emissions is important information not only about the approach of the moment of destruction of the rolling bearing parts but also about the operating time that the operating personnel still have to take measures to eliminate an emergency and complete catastrophic destruction.

The work aims to study the process of changing the vibration of centrifugal pumping units in operation and identify the patterns of multi-stage failure processes of rolling bearing parts by vibration trend emissions. Use the patterns of vibration trend emissions, starting from the first stage of degradation, and then monitor the development of defects for a timely assessment of the technical unit state.

The operation of a centrifugal pump unit at oil refineries is inextricably linked with assessing its technical condition at all stages of its life cycle: past, present, and future.

Console pumps are a type of centrifugal pump with a one-way fluid supply to the impeller located at the end of the shaft, remote from the drive (Fig. 1).

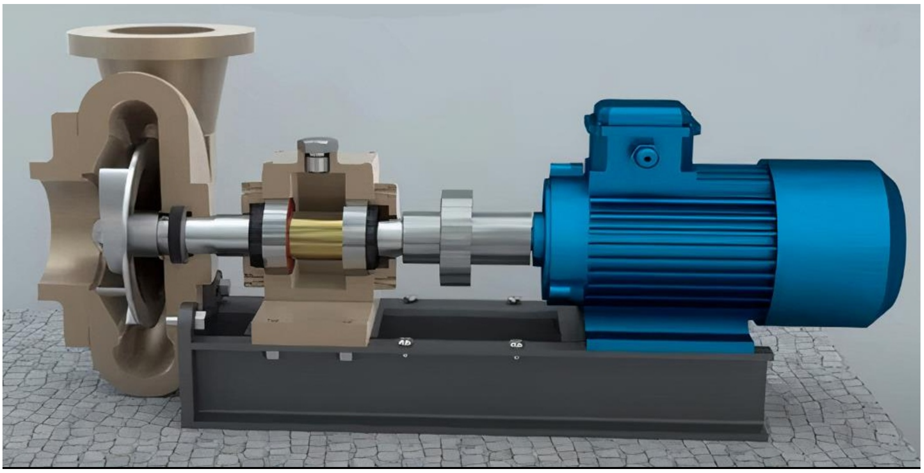

Fig. 1. Centrifugal pump type K (console)

Rolling bearings are the main supporting part of a centrifugal pump unit. Rolling bearings are bearing supports and fix the position of the shaft with the impeller, the motor rotor in the centrifugal pump unit. Rolling bearings use the rotational movement of the inner or outer rings, rolling elements (ball, roller), cage, and the complex movement of rolling

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elements.

The degradation of parts of the rolling bearings of a centrifugal pump unit during its operation is accompanied by an increase in the unit's vibration (Fig. 2) [1, 3, 4, 6, 7, 8].

Degradation of parts during fatigue failure occurs due to the development of cracks, surface wear, and shape changes. With contact fatigue failure, gaps in pairs of mating parts increase, and cracks, chips, pits, and shells appear. (Fig. 3, 4).

Suppose forces that cause a vibration of a centrifugal pump unit influence the process of fatigue failure. In that case, the nature of the change in vibration characterizes the process of fatigue damage to parts and assemblies.

With normally operating parts, during the incubation period for the development of fatigue defects, the change in vibration occurs at constant rates, and there is a monotonous increase in vibration parameters (Fig. 2) [9].

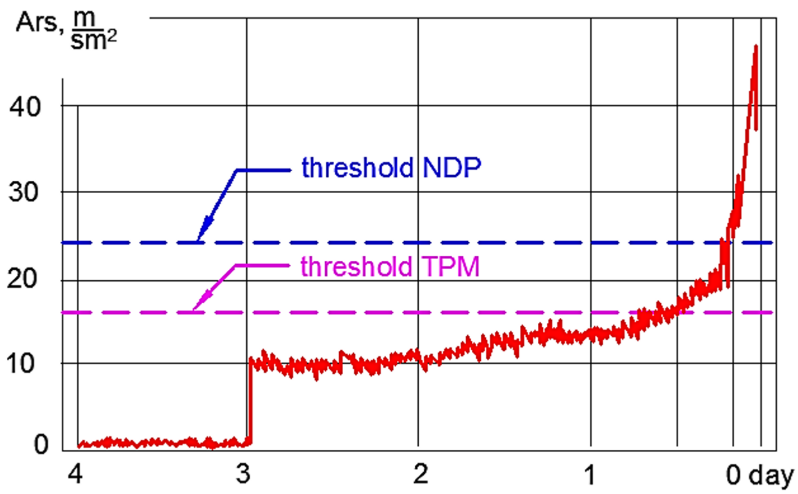

Fig. 2. Four-day trend of vibration growth as parts of bearing of centrifugal pump unit degrade

Fig. 3. Example of development of defect on outer ring of rolling bearing

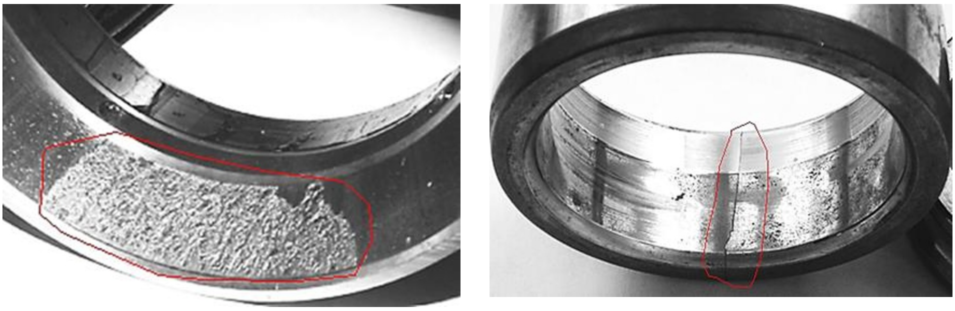

Fig. 4. Examples of development of defects on inner ring of rolling bearing.

Bearings are one of the most vulnerable elements in the design of a centrifugal pump unit. During the operation of rolling bearings, vibrations occur [2, 3, 4] due to variable reactions at the points of contact between the bearing parts: rolling elements, rings, and cage. Rolling bearing parts operate under conditions of constant, cyclic load changes.

2 Methods

Damage to any bearing element, for example, one rolling element in the bearing, can lead to serious damage to the entire centrifugal pump unit, up to a complete loss of its performance. Therefore, during the operation of a centrifugal pump unit, it is necessary to ensure control over the vibration of a working mechanical system to monitor and diagnose the condition of rolling bearings [2, 5, 6, 7, 8, 9].

Sensors are installed directly on the controlled centrifugal pump unit in certain places of the casing to obtain initial data on the unit's state. The sensors convert the measured parameters of various physical quantities (vibration, temperature, current strength, pressure, etc.) into an electrical signal. As vibration sensors, piezo accelerometers are used, the output signal of which is proportional to vibration acceleration. COMPACS® systems use vibration sensors of two designs, AB-311FRU and AB-321FK, which allow you to control the vibration of units of various designs. To mount vibration sensors on the diagnosed unit of a centrifugal pump, sensor holders are used, the installation of which is carried out without making changes to the unit's design (Fig. 5).

Fig. 5. Option to install vibration sensor on cantilever centrifugal pump NL125 / 200-72

The reliable operation of a centrifugal pump unit depends on many factors. The electric motor is a reliable link in the composition of a centrifugal pump unit, the operating conditions of which are sufficiently normalized. Another important factor in the reliable operation of a centrifugal pump unit is the operating conditions of the rolling bearings. Rolling bearings are widely used in pumping units. Operating problems arise when using rolling bearings in medium and large power pumping units.

Figure 6 shows the distribution of rolling elements in the working area of the bearing, in which there is nobody at the bottom point of the bearing with load P_o. Rolling bodies with loads $P`_1$ are located symmetrically with an angle of $0,5\gamma$ relative to the axis of symmetry OZ. An even number of rolling bodies is considered in the proposed calculation scheme. The traditional and proposed schemes of the loaded rolling elements of the bearing in the working area during the operation of the bearing are constantly repeated, replacing each other with a period multiple of half the step of $0,5\gamma$, the location of the rolling elements of the bearing.

There are 18 rolling elements in the working area of the ball bearing (Figure 6, 7), i.e., half of all rolling elements at $z=36$. This is a fundamental feature of the proposed calculation method, in contrast to the traditional approach, where 17 rolling bodies are involved in the work, i.e., less than half the total number of rolling elements.

The equilibrium condition for the system of converging forces, in this case, will take the following form

$$Q = 2\sum_{i=1}^{n_k} P_1^{/} \cos\left(i-0,5\right)\gamma; \tag{1}$$

The rolling element located at the lowest point of the bearing is excluded from the constraint equation for a ball bearing

$$\frac{\lambda_1^3}{\left(P_1^{/}\right)^2} = \frac{\lambda_2^3}{\left(P_2^{/}\right)^2} = \ldots = \frac{\lambda_i^3}{\left(P_i^{/}\right)^2}; \tag{2}$$

The elastic lowering of the inner ring λ_0 under the action of force Q) provided no gaps between the balls and the inner ring. The deformations of the rolling elements are considered as radial approaches λ_i of the rolling elements with the inner ring, which are related by the equations

$$\lambda_i = \lambda_0 \cos\left(i-0,5\right)\gamma; \tag{3}$$

where: λ_0 is vertical elastic lowering of the inner ring of the rolling bearing (Figure 7) under the action of vertical force Q, provided there are no gaps in the articulation of parts and the same dimensions of all rolling elements, $i = 1, 2, 3, ..., n_k$.

From (2)

$$P_i^{/} = \left(\frac{\lambda_i}{\lambda_1}\right)^{\frac{3}{2}} \cdot P_1^{/}; \tag{4}$$

where: i=1, 2, 3, …, n_k

The presented equations make it possible to establish the dependence of the total load Q on the force P_1

$$Q = 2P_1^{/}\left(\cos 0,5\gamma + \sum_{i=1}^{n_k} \frac{\cos^{\frac{5}{2}}(i-0,5)\gamma}{\cos^{\frac{3}{2}} 0,5\gamma}\right); \tag{5}$$

To determine the radial reaction force P_1, the expression

$$P_1^{/} = \frac{Q}{k_n}; \tag{6}$$

where: k_n is the constant geometric characteristic of the ball bearing, determined by the equation

$$k_n = 2\left(\cos 0,5\gamma + \sum_{i=1}^{n_k} \frac{\cos^{\frac{5}{2}}(i-0,5)\gamma}{\cos^{\frac{3}{2}} 0,5\gamma}\right); \tag{7}$$

The remaining radial reaction forces of the rolling elements are determined by the formula

$$P_i^{/} = P_i^{/} \frac{\cos^{\frac{3}{2}}(i-0,5)\gamma}{\cos^{\frac{3}{2}} \cdot 0,5\gamma}; \tag{8}$$

where: i= 2, 3, …, n_k

The resulting expressions and the initial design schemes make it possible to determine the radial reaction forces P_i` that perceive the vertical load Q in statics with greater accuracy than existing methods due to the accurate determination of the number of rolling elements n_k that perceive the load.

The comparison of the constant geometric characteristics of the bearing, according to the old and new (7) formulas (Table 1), showed that with a decrease in the number of rolling elements in the bearing, the difference in the application of approaches is clearly visible.

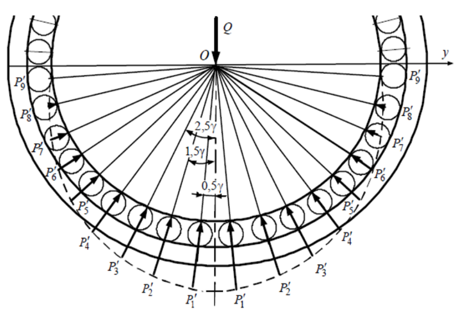

Fig. 6. Scheme of distribution of reaction forces between rolling elements of ball bearing with new approach

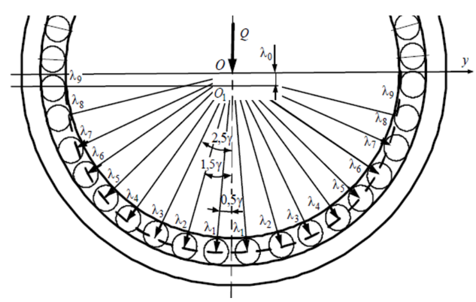

Fig. 7. Scheme of distribution of radial approach of rolling elements with inner ring in new approach

Table 1. Comparison of geometric characteristics of ball bearing depending on number of rolling elements

№	Number of rolling elements, z	Angular pitch, γ	Traditional method, k	New method k_n	Discrepancy %
1	36	10.00	8.2380	8.2849	0.6%
2	34	10.59	7.7799	7.8305	0.7%
3	32	11.25	7.3228	7.3755	0.7%
4	30	12.00	6.8646	6.922	0.8%
5	28	12.86	6.4076	6.4678	0.9%
6	26	13.85	5.9491	6.0157	1.1%
7	24	15.00	5.4925	5.5626	1.3%
8	22	16.36	5.0336	5.1128	1.6%
9	20	18.00	4.5775	4.6616	1.8%
10	18	20.00	4.1179	4.2159	2.4%
11	16	22.50	3.6629	3.7679	2.9%
12	15	24.00	3.43401	3.54614	3.3%
13	14	25.71	3.2016	3.3304	4.0%
14	13	27.69	2.97205	3.1117	4.7%
15	12	30.00	2.7495	2.8894	5.1%
16	11	32.73	2.52067	2.6737	6.1%
17	10	36.00	2.2836	2.4733	8.3%
18	9	40.00	2.05235	2.2675	10.5%
19	8	45.00	1.8409	2.0518	11.5%
20	7	51.43	1.61391	1.8566	15.0%
21	6	60.00	1.3536	1.7321	28.0%

Table 2. Distribution of radial forces depending on number of rolling elements in bearing for two operating states

№	Number of rolling elements, Z	Angular pitch γ	$P_0 = \frac{Q}{k};$	$P_1^{/} = \frac{Q}{k_n};$
1	36	10.00	0.1214 Q	0.1207 Q
2	34	10.59	0.1285 Q	0.1277 Q
3	32	11.25	0.1366 Q	0.1356 Q
4	30	12.00	0.1458 Q	0.1445 Q
5	28	12.86	0.1561 Q	0.1546 Q
6	26	13.85	0.1681 Q	0.1662 Q
7	24	15.00	0.1821 Q	0.1798 Q
8	22	16.36	0.1987 Q	0.1956 Q
9	20	18.00	0.2185 Q	0.2145 Q
10	18	20.00	0.2428 Q	0.2372 Q
11	16	22.50	0.2730 Q	0.2654 Q
12	15	24.00	0.2912 Q	0.2820 Q
13	14	25.71	0.3123 Q	0.3002 Q
14	13	27.69	0.3365 Q	0.3214 Q
15	12	30.00	0.3637 Q	0.3461 Q
16	11	32.73	0.3967 Q	0.3740 Q
17	10	36.00	0.4379 Q	0.4043 Q
18	9	40.00	0.4873 Q	0.4410 Q
19	8	45.00	0.5432 Q	0.4874 Q
20	7	51.43	0.6196 Q	0.5386 Q
21	6	60.00	0.7388 Q	0.5773 Q

The two schemes for calculating radial loads on rolling elements are a single general method for calculating radial loads for different angles of the rolling elements. The traditional and proposed methods do not need to be compared and opposed since they characterize each bearing in different design states.

Table 2, for the first time, presents the values of the coefficients k and k_n calculated by formulas old and (7) for ball bearings. Table 2 gives expressions for calculating the radial forces P_0 and $P`_1$ as a function of the vertical force Q for the traditional and proposed scheme.

Figure 8 shows calculation schemes for determining the radial forces on the rolling elements of a bearing 417 with the number of rolling elements $z = 7$. For the traditional calculation scheme, Figure 8 a, the radial reaction forces from the side of the bearing cage from Table 2 are shown $P_0 = 0.62\ Q$

Force $$P_1 = P_0 \cos^{\frac{3}{2}} \gamma = 0{,}62 \cos^{\frac{3}{2}} 51{,}43^0 = 0{,}305\,\mathrm{Q};$$

where: Q - is the vertical load on the bearing.

Figure 8 b for the proposed scheme shows the force $P`_1 = 0.539Q$ from Table 1. Force $P`_2$ is calculated by the formula (8)

$$P_2^{/} = P_1^{/} \frac{\cos^{\frac{3}{2}}(1{,}5)\gamma}{\cos^{\frac{3}{2}} \cdot 0{,}5\gamma} = 0{,}539\,\mathrm{Q} \frac{\cos^{\frac{3}{2}} 77{,}14^0}{\cos^{\frac{3}{2}} \cdot 25{,}71} = 0{,}066\,\mathrm{Q};$$

In Figure 8, load distribution functions $P_i = f(\gamma)$ are plotted on the outer ring of the bearing. The obtained curves reflect the qualitative dependence of the radial forces on the angle of the rolling element. Therefore, figure 9 for rolling bearing number 417 shows the dependence of the radial forces on the angle of rotation γ in a rectangular coordinate system, obtained in the calculation using both calculation schemes. The resulting diagram of changes in the radial forces of the rolling bearing has the following physical meaning. At point A_1, Figure 8, the rolling body enters the working area and takes on a radial load, which, at an angle of 1.5γ from the vertical axis of the shaft at point A_2, acquires the value $P_2`$, Figure 8 b. At point $A_3$, the radial force acquires the value $P_1$, and then at point $A_4$ - $P_1`$, and at point A_5, the radial force reaches its maximum value P_0.

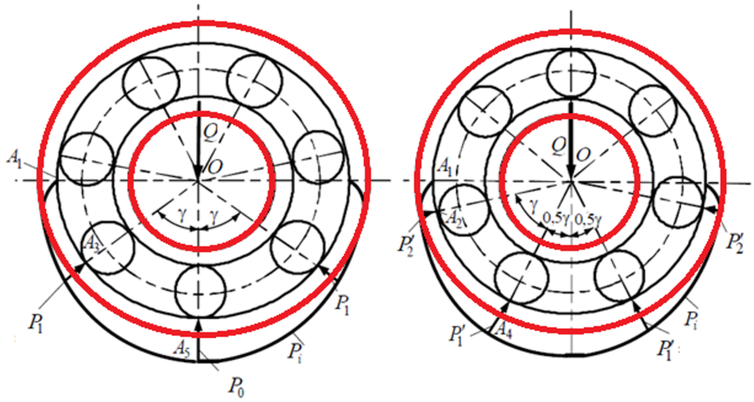

Fig. 8. Calculation schemes for determining radial forces on rolling elements of bearing number 417 (a is traditional, b is new)

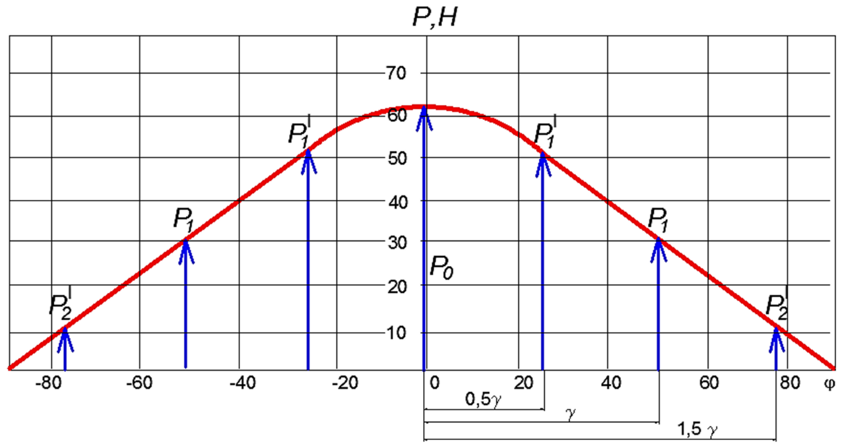

Fig. 9. Graph of change in radial force depending on angle of rolling element

From the graph in Figure 9, it can be seen that the radial force P_i changes according to a law close to a linear law. This quality is the advantage of a rolling bearing. After reaching the maximum value, the radial force decreases similarly. New knowledge about the physical processes of changing radial forces in rolling bearings has been obtained.

The maximum value of the radial force is determined according to the Shtribeck scheme when an odd number of rolling elements 3 perceives the vertical force Q. The proposed calculation scheme characterizes the operating state of the bearing with an even number of rolling elements 4. Thus, when the rolling bearing operates with a frequency of rotation angle $\varphi = 0.5\gamma$, there is a constant alternation of two calculated states, even and odd. For the previously considered rolling bearing with the number of rolling elements $z = 36$ (Figures 6, 7), the number of rolling elements according to the odd scheme is 17, and for the even scheme 18.

3 Results and Discussion

Under actual operating conditions of machines, the load dynamics on the rolling bodies are non-stationary. During the operation of a rolling bearing, there is a constant redistribution of loads over the rolling elements located in the lower half of the rolling bearing, taking into account the presence or absence of a rolling element at the lower point of the bearing. The shaft sags with the inner ring at high frequency. When the rolling elements roll in the bearing, the loads acting on these bodies change.

Figure 10 shows the change trend in RMS vibration acceleration during the operation of a centrifugal pump unit for 33 days recorded on the drive motor from the side of the front bearing. At the beginning of the period under consideration, a classic pattern of running-in of units and parts is observed (Figure 10, Section 1), and a decrease in the vibration level is observed. Further operation of the centrifugal pump unit proceeds at a constant degradation rate of components and parts (Figure 10, Section 2). Within 2 weeks, the vibration level changes insignificantly. Further, during the unit's operation, the running-in process again occurs with the transition to a constant degradation rate (Figure 10, Section 3). The vibration level decreases and stabilizes for a little more than a day. Then the process of defect development begins, and the vibration grows (Figure 10, Section 4), but having reached the level recorded earlier in section 2, it stabilizes on it for less than a day (Figure 10, Section 5). Then, during one and a half days, the running-in process occurs (Figure 10, Section 6). The vibration level decreases and stabilizes at some level for less than two days

(Figure 10, Section 7). Then there is a catastrophically fast development of defects in the rolling bearing (Figure 10, Section 8). The vibration level increases 2.8 times from 7.5 m/s^2 to 20 m/s^2. In this case, the development of the defect stops until the failure of the bearing, and then a drop in the vibration level is observed during the running-in of the defect. With further operation for 12.5 days (Figure 10, Section 9), an abrupt change in the vibration level is observed, visualizing the multi-stage failure processes of rolling bearing parts. In this example, the degradation process of a rolling bearing does not correspond to the classical degradation scheme. The change in the vibration state of a centrifugal pumping unit has a staged character - outbursts of vibration trends appear.

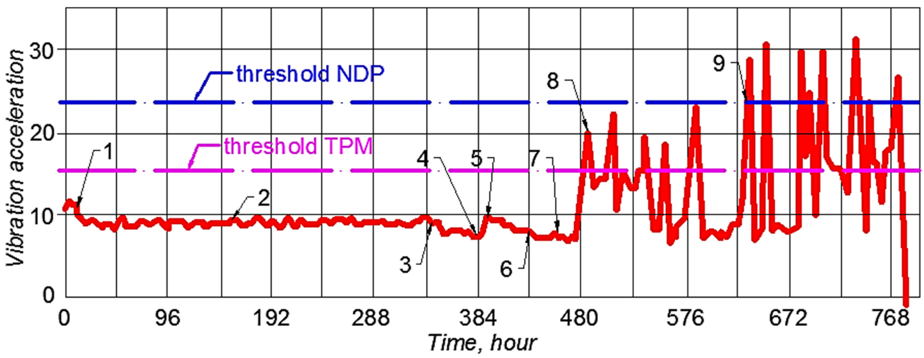

Fig. 10. Trend of change in vibration acceleration of centrifugal pump unit of technological position H-9 from front bearing (70-32320) of engine (VAO-500L-2, power 400 kW, rotation speed 2978 rpm)

A vibration surge or vibration trend surge is an abrupt change in the magnitude of the vibration parameter trend value, fixed at a time interval that exceeds the set threshold value and determines the occurrence of a surge.

The operation of bearings of a centrifugal pump unit is associated with a cyclic change in the loads applied to them; a change in loads leads to a cyclic change in their stress-strain state [8, 9, 10, 11]. Consequently, the process of degradation of parts, the development of cracks in them, is of a fatigue nature, while loops of elastic-plastic deformation - hysteresis - appear in the material itself under constant loads and unloading, which provides the parts with a certain resistance to fracture.

4 Conclusions

The traditional scheme for calculating the distribution of radial forces for an odd number of rolling bearing working bodies has been improved, and a new calculation scheme with an even number of rolling working bodies has been proposed. The proposed scheme made it possible to obtain new knowledge about the change in the geometric characteristics of the bearing and the redistribution of radial forces in the rolling bearing during its operation with a period multiple of half the pitch of the rolling bearing elements (0.5γ).

A new technique has been developed for isolating vibration emissions while measuring the vibration level. For the first time, a technique has been developed for controlling the staged degradation of parts by the amplitudes of vibration emissions. For the first time, a technique has been developed for controlling the staged degradation of parts by the duration of the intervals between vibration emissions.

Acknowledgments

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