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STEAM-TURBINE, GAS-TURBINE, AND COMBINED-CYCLE POWER PLANTS AND THEIR AUXILIARY EQUIPMENT

Some Features Relating to the Occurrence of Low-Frequency Vibration in Large Steam Turbines and Methods for Removing It

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Abstract—The article addresses general matters concerned with the occurrence of low-frequency vibration (LFV) in turbine units. It is pointed out that, despite the level of knowledge that has been achieved in regard to LFV, it still arises from time to time in power plant turbine units. Along with LFV caused by aero- and hydrodynamic excitation, LFV can also bear a subharmonic pattern. It is emphasized that the measures taken to remove LFV depend on the LFV occurrence origin. The article presents LFV occurrence and removal examples, including those relating to the use of honeycomb seals in high-pressure cylinders. With honeycomb seals, decreased sizes of channels and an increased channel component of overshroud forces caused by aerodynamic excitation are typically observed. It is pointed out that, in some cases that involve rotor rubbing against the stator, a multicomponent LFV with subharmonic and self-oscillation components is observed. It is shown that the regulatory documents do not contain criteria for estimating a multicomponent vibration in the low-frequency band. It is pointed out that multicomponent LFV can be a diagnostic indicator pointing to rubbing of the rotor against the babbit or seals. Recommendations on removing LFV of various origins are suggested. A diagnostic table that helps determine factors causing the LFV and that produces recommendations on increasing the turbine units operational reliability is given. It is stated for the first time that the turbine thrust bearing can behave as a source of oil excitation. It is also noted that the conditions under which a self-oscillation type LFV occur and its suppression methods should differ from the methods for suppressing self-excited LFV of a subharmonic nature.

Keywords: plain bearing, turbine unit, low-frequency vibration, subharmonic vibration, honeycomb seals, recommendations on reducing vibration, contradictions in the regulatory framework

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Despite the advances achieved in the control of low-frequency vibration in steam turbines, it still occurs from time to time both in the course of long-term operation of turbine units and immediately after completion of their repairs and modernizations. In addition, the tendency toward increasing the initial steam conditions upstream of the turbine also can have an influence on the LFV occurrence likelihood. Low-frequency vibration primarily gives rise to the danger of its quick sudden growth with destruction of the babbit lining of bearing inserts and wear of seals with degradation of economic efficiency. Therefore, much attention is paid to LFV at enterprises. There are special individual standards for the permissible level of low-frequency vibration.

Low-frequency vibration may be caused by various factors, and LFV suppression methods may differ from one another. In view of this circumstance, it is important at the initial stage to reveal the LFV origin and the factors that caused its occurrence for the possibility to work out optimal and efficient ways of its removal. The factors causing loss of rotor motion stability on oil film were considered for the first time in [1]. After that, the first attempts were made in [2] to explain this phenomenon theoretically, to identify the forces causing self-oscillations at the lowest natural frequency, and find a criterion determining the self-oscillation occurrence conditions. At that time, the notion of boundary conditions had not been introduced as yet. Thus, the author of [3] says that there was even no word in the German language that would denote this phenomenon. In the 1960s, the notions of nonconservative disturbing forces in the oil layer appeared in the former Soviet Union's technical literature on vibration. They were introduced by P.L. Kapitsa and received further development in [4–8].

It should be noted that matters relating to ensuring the stability of rotors in response to aerohydrodynamic excitation of so-called "oil" and "steam" LFV were essentially resolved in the 1980s–1990s in the theoretical and experimental works carried out at the Moscow Power Engineering Institute, Central Boiler and Turbine Institute, and the All-Russia Thermal Engineering Institute [4, 9], and domestic manufacturers currently produce vibration-stable machinery as a rule. Nonetheless, the need to address this topic still remains of issue for the following reasons:

1. There is a shortage of modern literature sources in which these matters would be addressed in a correct way and at the modern level. The surge of articles on this topic subsided long ago, and specialists who are beginners in vibration are not able to get familiar with the experience gained in the control of it by the researchers of two preceding generations.

2. The way the material is presented in the handbooks has been simplified so that it does not make it possible to solve practical problems with taking into account the technological tolerances of the turbine generator shaft line support parameters in the course of turbine unit repair and operation.

3. There is a lack of regulatory framework for designing turbine units with correct stability margins.

4. In some modern books and articles, the rotor vibration problem has been outlined incorrectly or with insufficient clarity.

5. In the course of equipment repairs, errors are made or changes accumulate with time in the "rotor–supports" system, which entail the occurrence of LFV even in the turbine units in which it did not occur before.

6. At some time, it was believed that the use of pad-type bearings would help solve the LFV problem cardinally because hydrodynamic forces do not theoretically participate in it. Initially, pad-type bearings were used in the turbines manufactured at the Kharkov Turbine Works for reducing friction loss with lubrication individually supplied to pads. However, field experience gained at some thermal power plants has shown that the drawbacks of such bearings resulting from real deviations in assembling in view of a low damping level exceed their advantages and, with all other things being equal, elliptical bearings are more reliable, cheaper, and simpler.

7. Despite the significant experience specialists have gained in control of LFV, some issues, such as subharmonic resonances and nonlinear properties of complex multisupport rotor systems on plain bearings still remain insufficiently understood and require carrying out of additional numerical analyses and experiments.

8. The modern machinery construction industry faces new objectives for development of turbines for extrasupercritical and ultrasupercritical steam conditions, due to which matters concerned with evaluation of the disturbing forces relating to the turbine unit's hot part are becoming increasingly more relevant.

Recently, additional factors causing LFV to occur have been determined: disturbing forces on the turbine thrust bearing collar have been revealed. Currently, a new problem in predictive analytics arises: prediction of stability margins and dangerous modes given the technological tolerances of the turbine unit assembling and alignment parameters in the course of repair and operation. In addition, it is necessary to know how to evaluate the stability margins of high-speed rotors of turbine units for various purposes.

LOW-FREQUENCY VIBRATION OCCURRENCE ORIGIN

Self-Exciting Low-Frequency Vibration

The danger of self-exciting LFV (self-oscillations) is not only because it increases in an avalanche-like manner under certain conditions causing the need to shut down the turbine, which entails corresponding economic consequences. In frequent cases, it proceeds without being noticed, with "soft" rubbing and wear of seal combs, which occurs for a short period of time. The power unit economic efficiency degrades, which can only be restored during the next labor-consuming repair. Self-exciting low-frequency vibration is caused by the action of nonconservative position forces, which arise when there is a linear or angular shift of the rotor axis with respect to the stator axis in the wheel space seals and in the plain bearings. It is necessary to distinguish the following main kinds of exciting nonconservative forces [4–6, 8]:

1. Excitation forces and low stiffness anisotropy in the oil layer of support and thrust plain bearings may become a cause of low-frequency vibration in any rotor and cause loss of motion stability in oil film: a so-called "oil LFV." The loss of stability occurs in one of the lowest modes of rotor vibration in the shaft line.

2. Circumferentially variable forces in rotor blades caused by the nonuniformity of working fluid leaks in the stage tip area and diaphragm seals or blade row forces were discovered in the 1950s by Lomakin [10] (Leningrad Metal Works) in pumps and then by Thomas in steam turbines [11].

In the 1970s, A.G. Kostyuk not only described blade row forces but also developed the theory of overshroud and labyrinth forces in seals caused by the nonuniform pressure profile about the shroud, and the overshroud forces in the seals turned out to be significantly higher than the blade row forces [4, 6].

The total forces, including the blade row and overshroud ones, are frequently a cause of so-called "steam LFV"; however, this notion is not quite correct, because blade row forces act simultaneously with oil disturbance forces, which are counteracted primarily by damping in the oil film. The highest forces arise in the turbine's first high-pressure stages. In the intermediate pressure stages, they are noticeably lower, and these forces are negligibly low in the low-pressure sections.

LOW-FREQUENCY VIBRATION OF A SUBHARMONIC ORIGIN

A subharmonic resonance at frequencies with values fractional with respect to the operating frequency occurs in nonlinear systems like rotors on plain bearings, with disturbances caused by rubbing against babbit or seals, nonlinearity in connection with a growth of reactions at the babbit walls or loosened and lifted-off parts in the support system [12]. Such phenomenon in a nonlinear system can be caused by any noticeable disturbance, including an ordinary imbalance. The frequency of this vibration depends on how close the system resonance is to the subharmonic frequencies equal to 1/2, 1/3, ..., $1/n$ of the rotor rotation frequency n. For example, if the resonance is near 1/3 or 1/2 of the rotation frequency, the system is pooled into this resonance. It should be noted that the occurrence of subharmonics with possible pooling into resonance is mainly typical for the external first turbine bearings and less frequently for generator bearings. Most probably, this is because their pedestals are more prone to become lifted off as a consequence of loss of uniformity in expansions and biting if there are small clearances in key joints.

In view of the fact that many rotors of large turbines have critical and, accordingly, natural frequencies close to the above-mentioned values, especially if there is a deviation from the design mutual position of supports, a so-called subharmonic resonance can occur. Since the lowest natural frequencies of turbine rotors are close to the subharmonic resonance frequencies, the power plant personnel often do not see the difference between LFV of self-oscillatory and subharmonic types, whereas the conditions under which self-oscillatory LFV occurs and the methods to control it should, of course, differ from the methods to control self-excited LFV of the subharmonic nature.

In addition, a so-called multicomponent LFV often occurs (Fig. 1). Once the boundary low-frequency vibration reaches the rotation frequency of real turbine machinery units, its values can grow smoothly or very rapidly until they reach the permissible limits. Figure 1 shows the spectral characteristics of support vibration in the low-frequency band from 0 to 50 Hz, which is called a multicomponent LFV, because vibrations at several lowest natural frequencies of rotors in the shaft line can be excited simultaneously. The components of different frequencies are seen. If we add all these harmonic components with taking their frequencies into account, the resulting displacements will be large. The vibration level often becomes significant, up to rubbing of the rotor against the stator, which generates the need to shut down the turbine. In this case, LFV is multicomponent in nature and is a diagnostic indicator of rubbing, because vibrations of different rotors are excited at their natural frequencies in this case, and there are also subharmonic frequencies [9, 13, 14].

The relevant regulatory documents do not contain the notion of a multicomponent LFV nor do they identify LFV of a subharmonic nature. Moreover, the assessments of low-frequency vibration level given in [15, 16] for an ordinary LFV of the self-oscillatory type differ, as in the old versions of the relevant state standard (GOST) by a factor of two. It should also be noted that the criteria written in [15, 16] are not suitable for such multicomponent LFV. This matter needs special consideration. As a rule, such multicomponent LFV occurs when the rotor rubs against babbit or seal combs. The nature of the occurrence of this vibration is complex and is connected with nonlinear effects entailing vibration of rotors on the oil film. Subharmonic vibration at frequencies far away from the subharmonic resonance is stable in nature and, as a rule, small, and often do not pose a danger. Their components may grow with a growth of rotational vibration or decrease in the rotor vibration damping. However, in coming closer to the subharmonic resonance, they can become destructive.

Vibration Caused by Rubbing of Rotor Parts against the Stator

Increased vibration, including LFV, may arise as a consequence of various kinds of rotor parts rubbing against the stator. In the case of LFV to occur, the force action scheme is similar to the action of oil layer and steam flow circulation forces. Under such conditions, a component transverse to the rotor displacement occurs. However, it acts only in the periods of time when the rotor comes into contact with the stator. In the majority of cases, rubbings are unstable in nature and so is the LFV. Low-frequency vibration may be accompanied by both superharmonic components and components multiple of the frequency of LFV proper, e.g., $2f_{\text{LFV}}$, $3f_{\text{LFV}}$, $4f_{\text{LFV}}$, etc. However, this instability of disturbances is sufficient also for subharmonic components to occur, which is clearly seen in Fig. 1.

Rubbings themselves may be of "soft" and "hard" nature. Relatively soft rubbings are observed when the shaft "match-strikes" in the bearing bore (when there are journal misalignments) against the babbit surface when there are oil particles on it, due to which the wear becomes not so significant; in this case, however, the babbit becomes locally heated, and the oil temperature becomes somewhat higher at the discharge. Rotor rubbings against the combs of springed diaphragm or end seals are also low noticeable and often not revealed by diagnostics during single or casual measurements. Such rubbings occur at relatively low vibration as a consequence of misaligned wheel space or high vibration. In the case of soft rubbings, there are insignificant deviations in vibration velocity, but they clearly manifest themselves in time trends, and the rotor rotation trajectories can transform into a closed elliptical cloud. In some cases, in the case of a high rotational vibration, rubbing occurs after several tens of cycles. This entails a characteristic sequential scatter

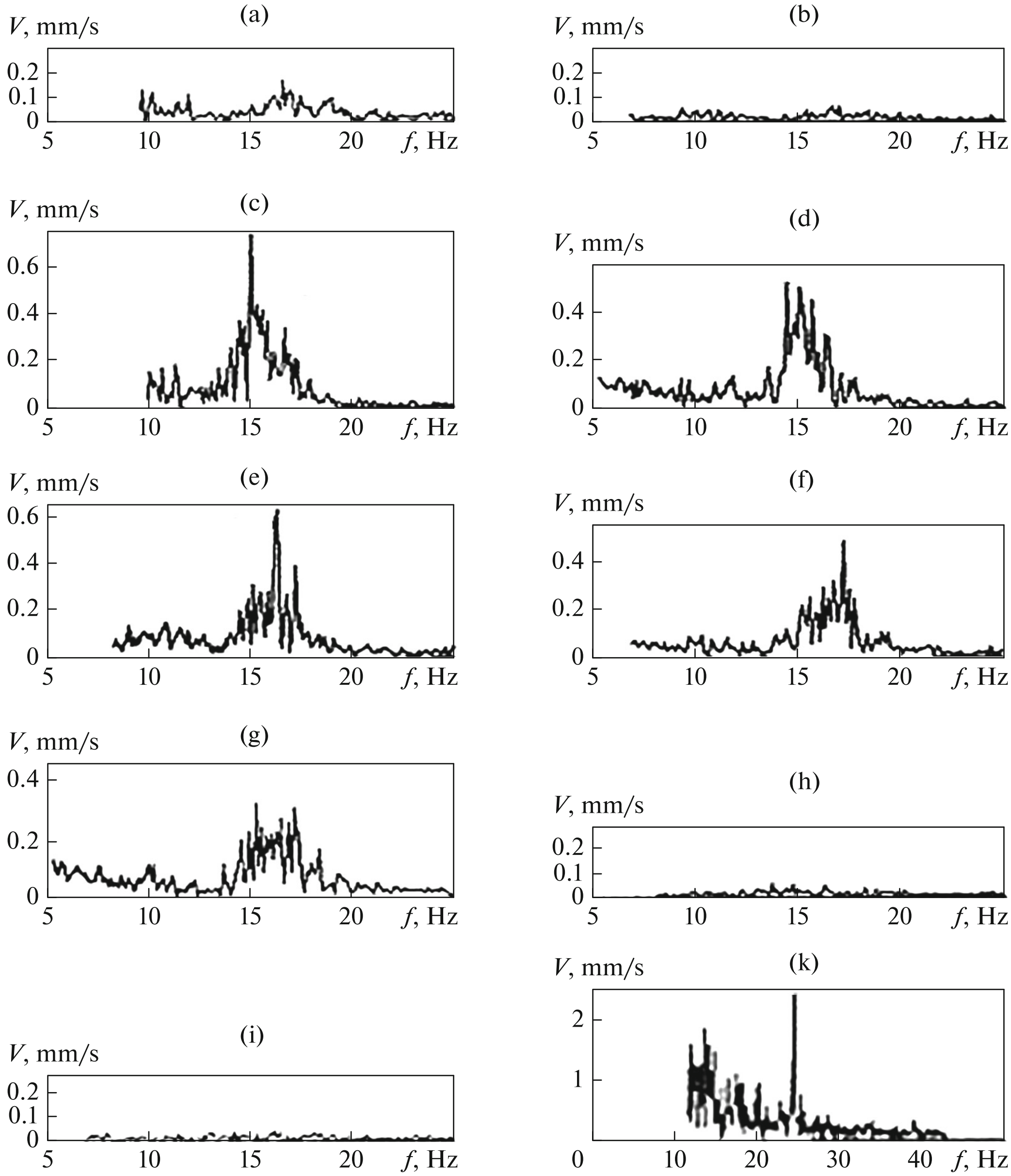

Fig. 1. Vibration velocity V of multicomponent LFV versus the rotor rotation speed f of the 800-MW turbine unit on supports. Support number: (a) 3; (b) 4; (c) 5; (d) 6; (e) 7; (f) 8; (g) 9; (h) 10; (i) 11; (k) 12.

and restoration of the main rotation vibration peak in the rubbing place.

If there is high vibration with one or several frequencies, rubbing in the wheel space or on the casing may occur. Under such conditions, the entire spectrum of the shaft line lowest natural frequencies can be excited with simultaneous existence of subharmonic frequencies.

Influence of Oil Temperature on the Occurrence of LFV

If there is a change of oil temperature, complex mutually contradicting processes occur in the oil layer, which are connected with a change and ratio of damping and disturbing forces. For example, as the oil temperature at the bearing inlet decreases to 39.2–39.5°C, traces of LFV with a frequency of 25 Hz appear on the supports of a 800-MW turbine unit with pad-type bearings and on supports with elliptical bearings (Fig. 2), with the vibration velocity rms value reaching V = 0.70–0.75 mm/s (according to the results of tests carried out by MPEI specialists in 1996 on the Surgut-2 state-owned district power plant's 800-MW turbine unit of station no. 3). As the LFV decreases further, a multicomponent LFV occurs with an avalanche-like growth of vibration velocity with 4–5 mm/s and higher

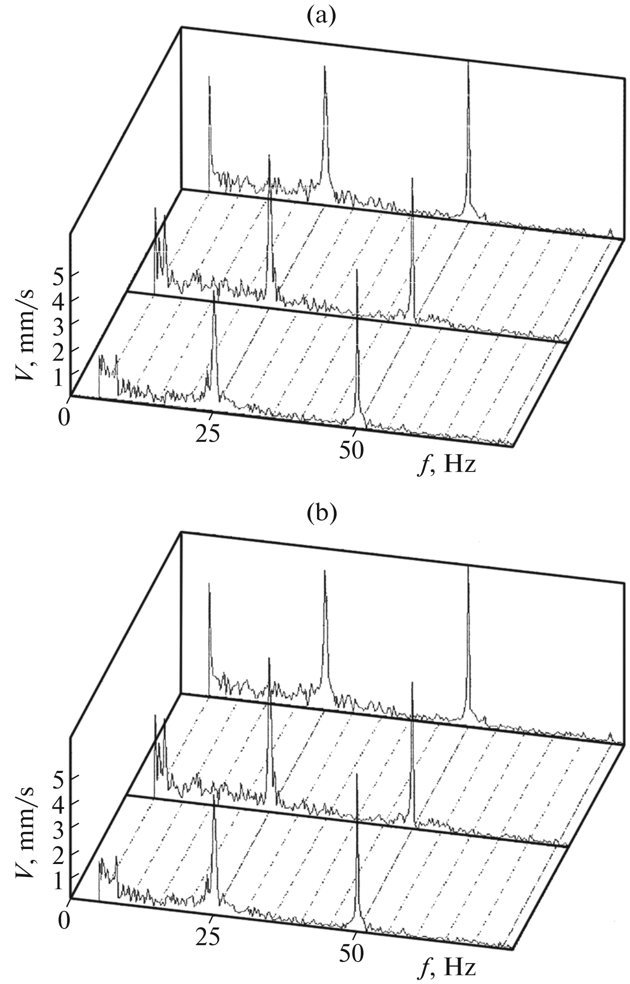

Fig. 2. Low-frequency vibration at 25 Hz frequency when the oil temperature decreases to 39.5°C. (a) Vertical support no. 2, channel 9, sensor 6; (b) horizontal support no. 6, channel 10, sensor 15.

components. Thus, the obtained data confirm that a decrease of oil temperature at the bearing inlet below 40°C is a gross violation of the operation manual.

As a rule, the following circumstances should be taken into account if the oil temperature decreases. The dynamic coefficient of oil viscosity at 40°C ($\mu = 0.02$ Pa s) is a factor of 1.5 higher than it is at 50°C and almost a factor of 2.0 higher than it is at 60°C; the journal loading decreases, the uplift increases, and the work of damping forces becomes smaller than the work of excitation forces. The motion trajectory approaches a circular one because the oil film anisotropy decreases, and the stability boundaries in both the rotation frequency and steam flowrate decrease [4, 9]. For some designs of bearings, this rule may not be valid if the work of damping forces exceeds the work of excitation forces. Thus, as the temperature increases, the viscosity decreases, and the damping and disturbing oil forces decrease; the work of damping forces may become higher than the work of excitation forces, and self-oscillations will not occur. However, for some combinations of support misalignments and other factors, a growth of oil temperature may also entail a decrease of stability margins. For this reason, each turbine unit must be studied individually.

Apart from the factors mentioned above, LFV in a turbine unit may occur as a consequence of low-frequency pulsations of medium, oscillations, and pulsations of medium in pipelines if there are paired vortices in their bends, vortex flows in the exhaust parts, and kinematic impacts on the base during a seismic excitation. These aspects are not considered in this article.

SPECIFIC FEATURES OF THE OCCURRENCE OF LOW-FREQUENCY VIBRATION IN HIGH-CAPACITY STEAM TURBINES AND METHODS FOR REMOVING IT

In 50–200-MW turbines produced by Leningrad Metal Works (LMZ), low-frequency vibration occurred and occur as a consequence of errors made during repairs, operation, or if there are essential deviations from the mutual position of supports and misalignments of rotors in the course of operation. In frequent cases, insufficiently heated oil (below 38–39°C) is fed to the bearings for unclear reasons. The experiments carried out by specialists of NRU MPEI on 800-MW turbines have shown the following: if the oil temperature at the bearing inlet is below 40°C, individual low-frequency bursts occur; at temperatures below 39°C, LFV is observed in almost all 12 bearings [13].

Sometimes, as a consequence of gross misalignments, LFV occurred in a 200-MW turbine unit in which there is no high level of steam excitation. In some turbines, as a consequence of gross misalignments of rotors at half-couplings, the initial pressure had to be temporarily decreased from 13.0 to 10.0 MPa or lower to avoid LFV. The phenomenon was removed by correctly aligning the rotor half-couplings, primarily of the high- and intermediate pressure rotors (HPR and IPR). Sometimes, LFV occurred as a consequence of incorrectly decreased upper clearances in the turbine or generator bearings. Drafts in the turbine building caused overcooling of oil in the heat exchangers along the exciters of TGV-200M generators, which entailed LFV in the exciter supports. For modern turbines, the specific load to ensure stable rotor motion on the oil film of elliptical bearings does not exceed 2 MPa.

In addition, the experiments carried out on the Shenk speeding up-and-balancing setup have confirmed that LFV may appear in highly loaded inserts of steam turbines [14] in view of the fact that, in the case of thinning the minimal oil layer, especially if there are small misalignments of journals in the bearing bores, rubbings and LFV in the transverse direction occur. In this connection, LMZ specialists have developed and implemented a second spherical hydraulic jack for such turbines, for example, for the turbine unit at the Bushehr NPP [14]. During the startup, such hydraulic jack ensures the absence of misalignment and coaxial position of rotor journals in the bearing bores.

Along with bearings with rigid half-inserts, support pad-type bearings with each pad resting on a sphere or ribs were developed and installed in some turbines in the 1970s. The number of pads could be from three to six. Specialists of the Kharkov Turbine Works (KhTZ) installed such bearings with the purpose to achieve essentially lower friction loss [9, 17]. As regards the stabilizing properties, theoretically, these bearings do not produce a hydrodynamic disturbance, and the stability margins of the rotors on them with respect to rotation frequency should be equal to infinity. However, the use of such bearings on some turbines did not help avoid LFV in them. Thus, in the K-500-23.5 turbines of KhTZ installed on pad-type bearings, LFV was removed only by correctly aligning the high- and intermediate-pressure rotors. Unsuccessful use of pad-type bearings stemmed from the fact that, with a low damping in them, these turbines were equipped with four-pad bearings with the worst (lowest) anisotropy of oil film stiffness. Even from the simple formula for the stability criterion [4], it is seen that the stability boundary for both rotation frequency and steam flowrate becomes essentially smaller with a low anisotropy. In addition, if there are off-design loads on the supports and if there are other disturbance sources, the work of disturbing forces seems to become higher than that of damping forces.

The stability boundary can be determined with respect to any system parameter whose the variation causes a growth of disturbing nonconservative forces. As a rule, the vibration form in the case of loss of rotor stability in the shaft line corresponds to one of the lowest natural frequencies. The vibration form itself is a spatial curve. It should be noted that, depending on the technological tolerances during assembling and operation, for the turbine rotors with a high anisotropy of support properties (if highly elliptical bearings are used), the stability can be lost predominantly in the horizontal or vertical direction in one of the shaft line rotors. The shaft motion orbits have the shape of elongated ellipses in this case. For rotors resting on supports with a low anisotropy of stiffness and damping (cylindrical bearings), the orbits have the shapes of precession closer to a circular one with approximately the same level of vibration in both the vertical and horizontal planes.

In other supports and rotors, a dynamic response with the same frequency usually occurs, and this frequency depends on the dynamic properties of the specific part of the structure. In some cases, the supports of one rotor may fall in resonance with the frequency at which the stability of another rotor is lost. In that case, it is only the knowledge of the dynamic properties of system components that allows one to determine the main cause of vibration and remove it in a reliable manner [9].

With various changes in the mutual positions of supports, transformation may occur in the shaft line vibration forms, and the corresponding frequencies can exchange their locations as they move to each other. As the natural frequencies of some parts of the structure come close to each other, all of its resonating parts are involved in intensive vibration, although the disturbance source can be localized, e.g., in the high-pressure rotor (if there is a high level of steam excitation) or in any other rotor with a support in which intensive oil excitation takes place.

A typical case that occurs after the turbine unit has been in operation for a certain period of time is a small wear of the bearing saddle, which results in that the bearing becomes less elliptical, and LFV occurs. In this case, an increased upper vertical clearance is usually revealed during the repair, and the bearing is "brought in line with the data sheet requirements" by scraping the horizontal joint. As a rule, this ends with "continuation" of the oil LFV, although it can be removed by scraping the bearing saddle at the worn place with checking the surface by means of a gauge.

In turbine units of the old design with high static transverse aerodynamic forces in the control stage, like the first versions of the K-800-23.5-5 turbine, there is a direct link between the oil and steam LFV because the exciting and damping forces in the oil film depend essentially on the transverse loads in the bearings and misalignments of the supports. In view of this circumstance, it is difficult to separate LFV in a turbine into oil and steam LFV. However, with a low level of disturbing aerodynamic forces in the high- and intermediate-pressure rotors of the K-800-23.5-5 turbine, owing to the use of so-called trough-shaped[1] axial-radial seals proposed in the 1970s by A.G. Kostyuk and later on, variable-pitch multifin seals, the low-frequency vibration is sooner of an "oil" nature. Nowadays, variable-pitch multifin seals can be recommended for use instead of trough-shaped ones [5].

As was already noted, rubbing may occur if there is loss of alignment of the rotors in the wheel space, also as a consequence of excessive mutual displacements of supports, which are accompanied by shaft misalignments in the bearing bore, or if there are decreased upper clearances in the bearings. During rubbings, the low-frequency components are unstable in nature. If there are hard rubbings, the vibration is, as a rule high, and the turbine unit must be urgently shut down. If the startup began with rubbings, the vibration main cause in all likelihood lies in inadequate assembling or in misalignment of the wheel space.

[1] Trough-shaped axial-radial seals proposed by MPEI specialists were implemented for the first time by TMZ Engineer V.I. Vodichev in the T-250/300-23.5 turbine. After that, these seals were tested at the TsKTI. This was the way in which the TMZ-TsKTI seals proposed by A.G. Kostyuk appeared in the literature.

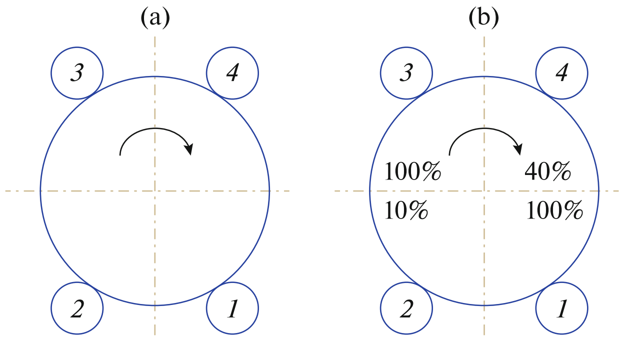

Fig. 3. (a) Initial and (b) altered steam admission schemes of the turbine unit HPC. (*1–4*) Valve numbers.

The occurrence of LFV with the lowest natural frequency or several natural frequencies in the corresponding directions can be a characteristic indicator of hard rubbings. Thus, in a 200-MW turbine, if there is rubbing in the control stage canopy upper part, vibration with the natural frequency of predominantly horizontal vibration of the high- and intermediate-pressure rotors equal to around 35–37 Hz may occur.

LOW-FREQUENCY VIBRATION REMOVAL EXAMPLES

Below, we present practical examples of LFV manifestation and removal in a few similar K-800-23.5-5 turbine units.

Example 1. After modernization of the high-pressure cylinder (HPC) involving the replacement of the inner cylinder and installation of honeycomb seals in the third to eighth turbine stages, problems arose, which were connected with increased buckling of the high-pressure rotor and occurrence of insignificant LFV near 25 Hz in the turbine unit first and second supports. No changes were made in the steam admission as a result of modernization (Fig. 3a). The numbering of valves in the figure corresponds to their opening sequence.

The redistribution of clearances in the wheel space did not yield a positive effect, and it caused rotor buckling in some cases. Attempts to remove LFV were met with success only by additionally loading the first pad-type bearing through decreasing the vertical clearance. After 4 years of operation, honeycomb inserts showed a uniform wear, and it was decided to permit over-shroud honeycomb seals for further operation without taking repair and restoration measures and with adopting the recommendation to carry out selective replacement of the worn inserts during the next scheduled overhaul. The honeycomb inserts showed a uniform wear at the top and bottom with a tendency to more significant wear on the right in comparison with that on the left side.

Example 2. In the course of overhaul in the HPC in the turbine stages from the third to tenth, honeycomb sealing inserts were installed. The clearances in the wheel space were in line with their data sheet values. As a result of repair, LFV occurred. The replacement of inserts in bearing nos. 1 and 2 by inserts of another design did not yield a substantial effect. As a result of changing the steam admission scheme by the version shown in Fig. 3b, it became possible to achieve the turbine unit operation in all modes without LPF (the numbering of valves corresponds to their opening sequence; the valve opening degrees are given in percentage).

However, with using such steam distribution scheme, bearing no. 1 began to be heated up gradually; its temperature reached 100°C; as a result, the turbine generator continued to operate until the overhaul with the hydraulic jack switched on. During the overhaul, the bearing was lowered and oil clearances were increased, due to which the babbit temperature returned to its normal range, but LFV again manifested itself during transients at around 25 Hz; i.e., the LFV was subharmonic in nature.

Example 3. During an overhaul, honeycomb sealing inserts were installed in the 800-MW turbine unit's wheel space. The clearances in the wheel space turned out to be increased as a consequence of the cylinder bore having an elliptical shape. After the overhaul, LFV around 25 Hz occurred. For removing the LFV, bearing nos. 1 and 2 were raised, and measures were taken (axial clearances in the seals were decreased) in order to decrease the high-pressure rotor's relative expansion. In regard to steam admission, the opening characteristic of high-pressure control valve no. 4 was shifted. With the valve opened by more than 10% (by opening the valve main cup), a growth in the vertical vibration component of insert no. 1 occurred. It was decided to shift the characteristic to exclude the valve cup being opened by more than 10%. As a result of the measures taken, in reaching the nominal power output at a low loading rate, there was no need to open the high-pressure control valve no. 4; LFV did not occur, and stable motion of the rotor was ensured. The steam admission was done as is shown in Fig. 2a.

ASSURANCE OF THE MAXIMALLY EFFICIENT PERFORMANCE OF TURBINE BEARINGS

With taking into account the experience gained from the operation and adjustment of turbine units, general recommendations for ensuring the most efficient performance of turbine bearings can be formulated as follows.

For the bearings of LMZ turbines, the optimal elliptical degree should be equal to 0.60–0.65. For the bearings of the Elektrosila plant (the last support of

Table 1. Measures for removing low-frequency vibration of turbine generators

LFV type	LFV removal measure
“Steam” LFV in combination with “oil” LFV It depends on the valve opening degrees, misalignments, and parameters of bearings and seals	Check the wear of inserts, remove the wear. Check the clearances and restore the optimal clearances (with the elliptical degree equal to 0.60–0.65). Increase the lateral clearance. Increase (and sometimes decrease) the oil temperature. Change the valve opening sequence if the vibration is in the high-pressure rotor. Realign the rotors at half-couplings. Replace the type of bearings, including the removal of the “chiller.” Deswirl the flow (by using stabilizing devices). Upgrade the seals. Shift the cylinder with respect to the rotor (a weak measure)
“Oil” LFV caused by a change of support reactions as a consequence of misalignments in combination with transverse forces from steam admission	Check the wear of inserts, remove the wear. Realign the rotors at half-couplings. Check the clearances and restore the optimal clearances (with the elliptical degree equal to 0.6–0.7). Increase the lateral clearance. Increase (and sometimes decrease) the oil temperature. Change the valve opening sequence if the vibration is in the high-pressure rotor. Replace the type of bearings, including the removal of the “chiller”
The LFV is unstable and multifrequency, caused by rubbing at increased vibration and/or rubbing caused by mutual thermal and force displacements of the rotor and stator	Remove increased vibration, remove rubbing. Check the possibility of rubbing, remove rubbing. Realign the rotors at half-couplings. Remove the wear of inserts. In the case of weak rubbing, a decrease of oil temperature sometimes decreases the vibration and decreases the likelihood of rubbing (a temporary measure). Change the valve opening sequence. Remove the nonlinearity like a support lift-off
Subharmonic resonance at frequencies equal to 1/2, 1/3, 1/4, 1/5 n	Realign the rotors at half-couplings. Remove the nonlinearity like a support lift-off. Check the wear of inserts and remove their wear. Increase or decrease the lateral clearance. Change the valve opening sequence if the vibration is in the high-pressure rotor and if there are significant transverse forces due to steam admission

the generator) with a deep grinding, to enhance stability, the lateral clearance should be increased to 0.0040–0.0045.

Axial-radial or variable-pitch multifin seals [5] should be used; in addition, increased tip and decreased axial clearances for seals should be assigned to avoid “steam” LFV.

For decreasing the aerodynamic excitation in honeycomb inserts, the depth of channels must be increased in designing the seals. Honeycomb inserts result in a decreased channel depth and in an essentially increased channel component of the overshroud forces.

Experience gained from the operation of turbine units has shown that increasing the elliptical degree of bearings by decreasing the vertical clearance is inadvisable because it results in a decreased work of damping forces in small displacements of the journal under the conditions of a highly stiff oil wedge; in addition, this may entail the possibility of rubbing against the lower or upper insert with the occurrence of LFV of another nature.

According to the studies reported in [9, 14], detuning of the first (lowest) natural frequencies of the rotors in the shaft line from the value $1/n$ of the shaft working rotation frequency is of much importance. In fact, this means that, with taking the scatter into account, the critical frequencies shall not be in the zone (0–0.575) f at all. If the shaft line is detuned insufficiently, or if its detuning becomes degraded under the conditions of misalignments, there appears a danger of "oil" low-frequency vibration of a subharmonic nature to occur with the frequency equal to $1/n$ of the nominal rotor rotation frequency.

In the case of LFV and rubbing, it is very important to determine their primary cause. It is necessary to determine whether a high vibration (rotation, LFV, or other) occurred followed by the occurrence of rubbing with a rich and unstable vibration spectrum or rubbing occurred initially as a consequence of incorrect assembling with a subsequent growth of vibration. It is clear that the recommendations will be completely different in these cases. If all problems began from imbalance, the turbine generator shall be balanced or shut down to check whether or not a blade has separated. If an intense LFV occurred first and then rubbing occurred, the factors causing its occurrence should be clarified: the wear of the babbit saddle, change of oil temperature, excessive oil cooling, violation of the bore shape during the repair (insufficient elliptical degree), misalignment, subharmonic resonance, etc. The abovementioned and other recommendations and measures are given in Table 1.

CONCLUSIONS

(1) Low-frequency vibration may have different nature, and it is important to correctly define it at the initial stage. It can appear both as a result of long-term operation and as a consequence of improper repairs and modernizations.

(2) With increasing the initial steam conditions, it is necessary to consider first of all "steam" excitation of low-frequency vibration.

(3) The amplitude of self-excited vibration can feature a very fast unlimited growth; therefore, it is extremely important to take efforts for preventing its occurrence.

(4) A decrease of oil temperature at the bearing inlet below 40°C is a gross violation of the operation manual.

CONFLICT OF INTEREST

The authors declare that they have no conflicts of interest.

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