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INCREASING THE ACCURACY OF MEASURING THE FLUTTER OF STEAM TURBINE BLADES IN SERVICE

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ПОВЫШЕНИЕ ТОЧНОСТИ РЕГИСТРАЦИИ АВТОКОЛЕБАНИЙ ЛОПАТОК ПАРОВЫХ ТУРБИН В УСЛОВИЯХ ЭКСПЛУАТАЦИИ

Flutter of turbine blades may be a reason of blade damage. Appearance of power steam turbines last stages shrouded blades flutter, realized with disk natural forms and in-phase natural forms, was registered in service by help of discrete-phase method modern variant. In this paper peculiarities of power steam turbines last stages shrouded blades flutter, realized in service with in-phase natural forms, are considered. Advantages and defects of registration of these natural forms with help of modern variant discrete-phase method induction gauges are studied. Appearance of additional errors during registration of flutter, realized with in-phase natural forms, is marked. These additional errors are called because of torsion and longitudinal vibration of turbine and generator rotors assembly and axial vibration of stator details, in which induction gauges are fastened. Measures, concerning of errors reduction, are recommended.

STEAM TURBINE; BLADE; FLUTTER; DISCRETE-PHASE METHOD; MEASUREMENT; INDUCTION GAUGE; VIBRATION RELIABILITY.

Автоколебания лопаток последних ступеней мощных паровых турбин могут стать причиной их усталостного разрушения. Для успешной борьбы с автоколебаниями необходимо определить собственные формы лопаточных венцов, с которыми автоколебания реализуются в процессе эксплуатации. В настоящей статье рассмотрены особенности автоколебаний бандажированных лопаток последних ступеней мощных паровых турбин, реализующихся в процессе эксплуатации с синфазными собственными формами лопаточных венцов. Проанализированы преимущества и недостатки регистрации подобных колебаний с помощью модернизированного варианта дискретно-фазового метода. Показано, что дополнительные погрешности измерений могут быть связаны с крутильными и продольными колебаниями валопровода, а также осевыми вибрациями деталей статора, в которых закреплены индукционные датчики, вызванными автоколебаниями лопаток с синфазными формами. Рекомендованы мероприятия по минимизации погрешностей измерений.

ПАРОВАЯ ТУРБИНА; РАБОЧАЯ ЛОПАТКА; АВТОКОЛЕБАНИЯ; ДИСКРЕТНО-ФАЗОВЫЙ МЕТОД; ИЗМЕРЕНИЯ; ИНДУКЦИОННЫЙ ДАТЧИК; ВИБРАЦИОННАЯ НАДЁЖНОСТЬ.

It is impossible to determine the dynamic stress levels in turbomachinery blades by calculation, and experimental studies are therefore required for ensuring their fail-safe performance.

Frequency detuning is performed for the blades fabricated for the last stages of high-power steam turbines, i.e., sufficient margins are provided between the operational rotations and the rotations where resonances with the most excitable natural forms occur. In some cases, vibration studies in experimental model and full-scale turbines are not carried out to the full extent. Furthermore, it should be noted that a number of modes that are potentially dangerous for fatigue strength of the blades can be inspected only when the turbine is operating. In this regard, systems that can continuously monitor the state of vibration in the blades under operating conditions are being developed. Virtually the only measurement technique ensuring the continuous operation of such systems is the so-called discrete phase method (DPM), in which the vibrational state of the blades is assessed by the indications of fixed sensors mounted against the tips of the rotating blades [1]. In recent years, due to the evolution of computer technology, the capabilities of the DPM have increased extremely.

It should be borne in mind that the DPM can be used to measure the amplitudes of peripheral sections and their oscillation rates or the mutual blade displacements, rather than the dynamic stress values that are of practical interest. In order to substantiate the transition to these values, it is necessary to analyze the experimental data, i.e., to determine the oscillation frequency, the ratio of the tangential and the axial amplitude components, the distribution of amplitudes over the wheel periphery. It is rather important to evaluate and minimize the measurement errors in order to reasonably assess the hazards of the registered oscillations. This paper discusses the errors occurring when registering blade flutter in the last stages of high-power steam turbines using the DPM, and the methods for reducing the influence of these errors.

The last-stage blades of the more high-powered steam turbines are typically shrouded, i.e., a "fully constrained" blade ring is a particular case of a cyclically symmetric system whose natural forms have different values of nodal diameters and nodal circles. Two-types of oscillations can be identified with respect to the turbine blades, the in-phase forms (without nodal diameters) where the amplitudes and phases of all blades on the wheel are identical, and the out-phase forms (with different values of nodal diameters and nodal circles) where the amplitudes vary circumferentially by the sine law. The first group of out-phase forms is commonly called the disc ones.

The last-stage blades in operation may experience resonance or stall oscillations, and sometimes flutter.

Methods for calculating the frequencies of individual and constrained blades have been developed for reducing the risk of resonance oscillations [2–4]. Since the main source of resonant oscillations is the time-constant non-uniformity of the flow parameters on the wheel circumference, the danger of the majority of natural forms for the shrouded blades (and any fully constrained blades) is theoretically equal to zero because the natural forms are orthogonal to the disturbing loads. In particular, the work of the disturbing forces turns out to equal zero for all forms of in-phase and most forms of out-phase oscillations [2]. The only dangerous modes are those for which the equality $m = k$ is satisfied, where m is the number of nodal diameters, and k is the oscillation multiplicity, i.e., the number of blade oscillations per rotor revolution. This fact is taken into account in the design of standards, with only the disc oscillations for which $k = 2–4$ (and sometimes $k = 5–6$) detuned. The peripheral sections move almost strictly in the axial direction during disc oscillations, because the tangential component of deflection is highly limited due to high tensile and compressive stiffness of the shroud ring.

Stall oscillations of last-stage blades occurs under light or no loads, when the flow around the blades exhibits an off-design behavior due to a sharp decrease in the volumetric flow rate of steam. Stall oscillations happens in disc modes whose frequencies are not multiples of the rotation frequency and whose amplitudes are unstable.

The theoretical possibility of various types of blade flutter was discussed in [4, 5]. The last-stage blade flutter occurring in steam turbines in operation was discovered relatively recently, after a number of power stations installed control systems based on using DPM sensors. Flutter with disc modes with a relatively large number of nodal diameters was the first type registered [6, 7]. In some cases, flutter was the greatest risk factor for blade fatigue strength.

When developing methods for controlling the oscillations in the shrouded blades, the fact that all of the above-described types of the most dangerous oscillations occurred with disc modes was taken into account, i.e., the axial component of the deflection of the peripheral section of the blade had to be measured.

The standard version of the DPM recording the displacements of the peripheral section of the blade could not be used for controlling the oscillations of the shrouded blade, as its tip was 'enclosed' by the shroud platform. An upgraded version of the DPM was designed [8, 9] to control these blades. The essence of this version is that a small-diameter magnet is placed within the shroud platform, and the cross-section of the induction DPM sensor is shaped as an elongated rectangle whose minimal inertia axis makes an angle β with the turbine axis. As the magnet moves past the sensor, the magnetic flux changes, and an EMF whose value reverses sign when the magnet moves past the core is induced in the sensor coil. A pair of sensors located in one axial plane at a small distance S from each other is used to best measure the axial component of the oscillations. The minimal inertia axes of the cross-sections of the first and the second sensor make a $+\beta$ and a $-\beta$ angle with the turbine axis, respectively. If the blade deflects by the value x in the axial direction, the distance between the cross-sections of the sensor cores in the plane of the magnet's rotation changes by ΔS, which must be measured with a high degree of precision.

The number of time pulses generated by the measuring equipment at a frequency of 40 MHz is counted in order to precisely measure the time intervals between the passage of the magnet past the first and the second sensor of the pair. The readings from the revolution sensor located near the half-coupling of the turbine rotor are used to determine the relationship between the time intervals and their corresponding linear values. A slot made in the cylindrical surface of the half-coupling is used as a keyphasor. Using a high frequency of 40 MHz allows to determine both the amplitudes and the revolutions with a very high precision. Even when the rotational speed of the peripheral section is equal to 660 m/s, the amplitude is determined with an error of 0.016 mm, and the number of revolutions per minute, equal to 3000 rpm, with an error of about 0.00375 rpm (the errors are reduced with a decrease in speeds and revolutions).

The proportionality factor k_n between the x and ΔS values depends not only on the angle β, but also on the radial clearance between the magnet and the sensor and on the axial displacement of the magnet relative to the sensor center; it is therefore determined on the calibrator. As the flutter frequencies are not multiples of the rotary speed, the blade moves past the sensors with an arbitrary phase, and therefore the value $\Delta S_{\max}$ which is proportional to the oscillation amplitude x_0 can be measured during a short period of time:

$$\Delta S_{\max} = \frac{1}{k_n} x_0 . \quad (1)$$

To assess the level of dynamic stress, it is necessary to know the oscillation frequency f in addition to the amplitude x_0. Since the DPM sensors do not measure the entire oscillation process, but only its discrete values once per revolution and, besides, the blade oscillation frequency f is higher than the rotary speed n, it is fundamentally impossible to determine the true oscillation frequency using a single pair of sensors.

A most detailed description of the measurements using a upgraded version of the DPM is presented in [10]; we adopted a number of the formulae given below from this study. In particular, it was established that the following relation exists between the true frequency f and the frequency f_{meas} "measured" using one pair of sensors:

$$f = kn \pm f_{meas}, \quad (2)$$

where k is an integer.

To determine the frequency f, it is necessary to use the readings from two pairs of sensors located at an angular distance Δφ from each other, and use the following formula:

$$\cos\left(\frac{\Delta\varphi f}{n}\right) \approx \frac{\sum_{i=1}^{M} \Delta S_{1i} \Delta S_{2i}}{\sqrt{\sum_{i=1}^{M} \Delta S_{1i}^2 \sum_{i=1}^{M} \Delta S_{2i}^2}}, \quad (3)$$

where ΔS_{1i} and ΔS_{2i} are the deviations from the mean values for the first and the second pair of sensors at an i-th measurement, M is the total number of measurements. The accuracy of formula (3) increases with an increasing number of measurements, but it becomes practically accurate when measurements are performed for several seconds.

After finding the frequency f when registering flutter with disc modes, it proved possible to determine the number of nodal diameters m, as well as to clarify the features of the oscillations influencing the possibility of supplying the energy from the flow to the blades [7, 10].

The need to change the measurement technique became clear after flutter with in-phase modes was registered, occurring simultaneously with the same frequency in four stages (the blades of the last and penultimate stages of a double-flow low-pressure rotor of a high-power turbine) [10, 11]. Three forms of flutter relatively close in frequency of oscillation were registered, slightly different from the first in-phase frequency of the penultimate-stage blades and the second in-phase frequency of the last stage blades whose calculated values were close to each other.

The peripheral section of the blade in in-phase modes has not only an axial (x_0), but also a tangential (y_0) component, with the inequality $y_0 > x_0$ satisfied for the first mode. Ref. [10] revealed that the following formula holds true with both axial and tangential components present:

$$\Delta S_{max} = \sqrt{\left(\frac{1}{k_n} x_0 \cos\frac{fS}{2nR}\right)^2 + \left(2 y_0 \sin\frac{fS}{2nR}\right)^2}, \quad (4)$$

where R is the radius in which the DPM sensors are installed.

At $y_0 \approx 0$ and $S << R$ formula (4) naturally becomes (1). It follows from formula (4) that the 'normal' pairs of DPM sensors with small bases S are ineffective for measuring the tangential component of the oscillation. Provided that $y_0 > x_0$, a significant increase in the useful signal can be achieved by composing 'additional' pairs from the sensors that are already part of various 'normal' pairs.

If the angles β for the sensors making up the 'additional' pair with the base S_{add} are the same, the measurement results are described by the formula:

$$\Delta S_{\max} = (\pm \frac{1}{k_n} x_0 + 2y_0)\sin\frac{fS_{add}}{2nR};. \quad (5)$$

From now on we are going to use *a* + sign, which can be always achieved by selecting the sensors with angles +β or –β.

It is noted in [10] that the highest useful signal can be obtained if the mutual blade displacement with the angular coordinates φ_j and φ_k are determined using the same DPM sensor. By introducing the notations $\varphi_k - \varphi_j = \psi$, $\varphi_k + \varphi_j = \psi + 2\varphi_j$, it is possible to determine the mutual displacements at an i^{th} measurement, when the oscillation phase of the j^{th} blade moving past the sensor is equal to α_i:

$$\Delta S_i(jk) = \left(\frac{1}{k_n} x_0 + 2y_0\right) \times$$
$$\times \sin\frac{\psi f}{2n} \cos\left[\alpha_i + \frac{(\psi + 2\varphi_j)f}{2n}\right]. \quad (6)$$

Because the flutter frequency is not a multiple of the rotary rate, the maximum value of the mutual displacements is equal to:

$$\Delta S_i(jk) = \left(\frac{1}{k_n} x_0 + 2y_0\right)\sin\frac{\psi f}{2n}. \quad (7)$$

The condition $\sin(\psi f/2n) = 1$ corresponds to the measurement of the mutual displacements of the blades moving past the DPM sensor in antiphase. By changing the value of ψ (this possibility is incorporated in the software for processing the measurement results), it is possible, independent of the location coordinates of the DPM sensors, to determine not only the maximum value of the mutual displacements, but also to refine the oscillation frequency *f*.

Using the modified measurement technique with formulae (3)–(7) allowed to determine the ratio between the axial and the tangential amplitude components of for the peripheral sections of the blades of different stages with three different forms of in-phase oscillations, the phase shift between the oscillations of different blade rings, the dependence of flutter intensity on the operation mode of the turbine.

A number of errors absent when registering flutter with disc modes were revealed when analyzing the measurement results. For example, the frequencies calculated by formula (3) did not precisely satisfy condition (2). Certain patterns in the variation of amplitudes and oscillation frequencies over the wheel circumference were "registered", despite the fact that the excited normal mode was in-phase. The x_0/y_0 ratios were significantly different for the blades of two stages of the same type oscillating with equal frequencies and approximately equal total amplitudes.

Let us determine the causes for these discrepancies and consider the possibilities of minimizing the errors detected.

The fundamental difference of flutter with in-phase and disc modes is that in the first case, the principal vector and the principal moment of the forces acting on the rotor from the blade ring are not equal to zero. As a result, these axial forces and torques can cause longitudinal and torsional oscillations of rotors assembly, and the oscillations of the stator components where DPM sensors are mounted.

The experimentally observed equality of the flutter frequencies of all four blade rows is specifically connected to the torsional and longitudinal flexibility of the rotors assembly components, because, due to the inevitable manufacturing deviations, the frequencies of different sets of blades made from the same technical drawing should be slightly different.

The presence of torsional oscillations of the rotors assembly in one of the modes of in-phase flutter is confirmed by the experimental data presented in fig. 1.

Fig. 1,*a* shows the readings of the RPM sensor located near the rotor's half-coupling. Fig. 1б shows the readings of the sensor pair with a small base S recording almost exclusively the axial component of the peripheral section of the deflection. It can be seen that the oscillations of the blades and the rotor occur with the same 'measured' frequency of about 15 Hz. Comparing the readings of two pairs of sensors using formula (3) revealed that the true frequency was $f = 115$ Hz, i.e., relation (2) is satisfied provided that $k = 2$.

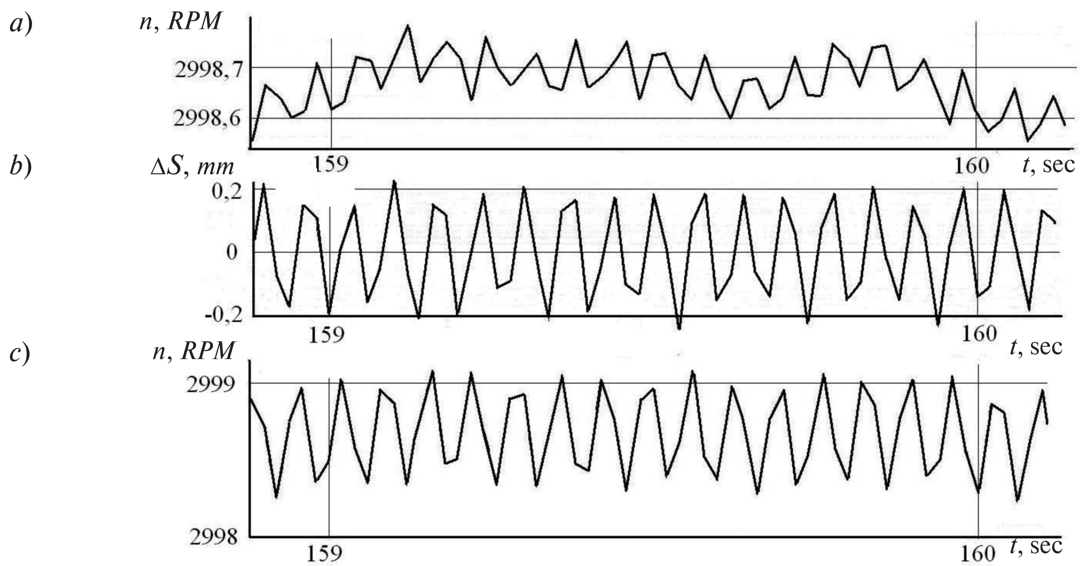

Fig. 1. Data read from: RPM sensor (*a*); pair of sensors registering the axial component of the oscillations (*b*); blade sensor used for RPM readings (*c*), shown versus the time of the readings

The oscillations of the rotor (i.e., the oscillations of the keyphasor) result in errors in finding both the frequency and the intensity of the oscillations. In order to qualitatively and quantitatively assess the effect of rotor vibration on the measurement results, it is necessary to find the amplitude of rotor oscillations, and the possible phase shift between the oscillations of the rotor and the blades (a phase shift may occur because the flutter frequencies are higher than the fundamental frequencies of the torsional and longitudinal oscillations of the rotors assembly).

The data on the connection between the keyphasor and the blades can be obtained if one of the blade sensors is used for RPM readings, and one of the blades is used as the keyphasor. This possibility is incorporated in the software for processing the results, with the signal from only one of the blades with magnets installed used to determine the revolutions. The measurement results are shown in fig. 1,*c*. It can be seen that the “change in the revolutions” also occurs with $f_{meas.}$ = 15 Hz, but with a much greater intensity, as the blade flutter amplitudes are substantially greater than those of the torsional oscillations of the rotor in the half-coupling area.

If the blade used to determine the number of revolutions oscillates with the frequency f and the amplitudes x_0 and y_0 in the axial and tangential directions, and the phase angle at the moment when the blade passes the sensor at an i^{th} measurement equals α_i, its deviations from the equilibrium position for the i^{th} and the $(i + 1)^{st}$ measurements are equal to:

$$\Delta S_i = \left(\frac{1}{2k_n} + y_0\right)\sin\alpha_i;$$

$$\Delta S_{i+1} = \left(\frac{1}{2k_n} + y_0\right)\sin\left(\alpha_i + \frac{2\pi f}{n}\right). \quad (8)$$

Taking into account expressions (8), it is easy to see that the error in determining the circumference using the blade RPM sensor at the i^{th} metering is equal to:

$$\delta S_i = \Delta S_{i+1} - \Delta S_i = \left(\frac{1}{k_n} + 2y_0\right)\sin\frac{\pi f}{n}\cos\left(\alpha_i + \frac{\pi f}{n}\right). \quad (9)$$

Thus, the distance $2\pi R + \delta S_i$ (and not $2\pi R$) can be measured in the i^{th} revolution; the length of the i^{th} revolution can change, causing the ‘measured’ rotary rate to change as well (see fig. 1,*c*).

Knowing the time dependence of the keyphasor oscillations, it is possible to assess their effect on the accuracy of the blade amplitudes and their frequencies by calculation, and then compare the results with the experimental data.

Due to the error in determining the circumference, the error in determining the distance between the j^{th} and the k^{th} blades for the i^{th} measurement is found to be:

$$\delta S_i(jk) = -\frac{\psi}{2\pi}\left(\frac{1}{k_n}x_0 + 2y_0\right)\cdot\sin\frac{\pi f}{n}\cos\left(\alpha_i + \frac{\pi f}{n}\right). (10)$$

Thus, the 'measured' mutual displacement of the j^{th} and the k^{th} blade is given, instead of formula (6), by the expression:

$$\Delta S_{i\Sigma}(jk) = \Delta S_i(jk) + \delta S_i(jk). \quad (11)$$

Let us transform expression (11) to the following form:

$$\Delta S_{i\Sigma}(jk) = (\frac{1}{k_{\bar{i}}}x_0 + 2y_0)(C_1\cos\alpha_i - C_2\sin\alpha_1), \quad (12)$$

where

$$C_1 = \sin\frac{\psi f}{2n}\cos\frac{(2\varphi_j + \psi)f}{2n} - \frac{\psi}{4\pi}\sin\frac{2\pi f}{n};$$

$$C_2 = \sin\frac{\psi f}{2n}\sin\frac{(2\varphi_j + \psi)f}{2n} - \frac{\psi}{2\pi}\sin^2\frac{\pi f}{n}. \quad (13)$$

Because the flutter frequencies are not multiples of the rotary rate, the maximum value of $\Delta S_{\Sigma max}$ is determined by the formula:

$$\Delta S_{\Sigma max} = (\frac{1}{k_n}x_0 + 2y_0)\sqrt{C_1^2 + C_2^2}. \quad (14)$$

In contrast to formula (7), in this case the value of $\Delta S_{\Sigma max}$ depends not only on the angular distance between the blades ψ, but also on the position of the blade on the wheel circumference, because the C_1 and C_2 coefficients depend on the angular coordinate φ_j.

Let us compare the calculated and the experimental dependences of the effect of the vibration of the keyphasor (in this case, the blade acting as the keyphasor) on the accuracy of determining the flutter intensity. Let us take into account the fact that the margin of error does not depend on the oscillation intensity of the vibrations when using the blade as the keyphasor, because the multiplier $(1/k_n)x_0 + y_0$ is included in the expression both for ΔS_i and for δS_i.

To reduce the effect of random errors in formulae (7) and (14), let us consider the case $\sin(\psi f/2n) = 1$ allowing to obtain the maximum measurement results. In this particular case, let us define the calculated dependence of the oscillation intensity on the angular coordinate φ_j, i.e., the angular distance of the j^{th} blade from the first one determining the start of a revolution.

The relative calculated value of $\Delta S_{\Sigma max}/\Delta S_{max}$ is in this case determined by the expression:

$$\Delta S^{rel}_{\Sigma max}(\varphi_j) = \sqrt{1 + \frac{n^2}{4f^2}\sin^2\frac{\pi f}{n} + \frac{n}{f}\sin\frac{\pi f}{n}\sin\frac{f}{n}(\varphi_j - \pi)}. \quad (15)$$

The maximum and the minimum values are achieved provided that $\sin[(f/n)(\varphi_j - \pi)] = \pm 1$ and be equal to $1 \pm (n/2f)\sin(\pi f/n)$. For flutter with a frequency of 115 Hz this means that the maximum and the minimum values differ from ΔS_{max} by ± 17,6%. At the same time, the average $\Delta S_{\Sigma max}$ value for all the blades on the wheel $\Delta S_{\Sigma max}$ only slightly differs from ΔS_{max}. In fact, for φ_j varying from 0 to 2π, the sum of the last terms in the radicand of formula (15) is equal to zero and the relative difference, approximately equal to $(n^2/8f^2)\sin^2(\pi f/n)$, is only 1,55%.

Let us compare the calculated dependence obtained by formula (15) to the experimental found using the blade RPM sensor for the case $\sin(\psi f/2n) \approx 1$. We are going to perform the comparison not only for the ΔS_{max} values but also for the more representa tive RMS values, which are determined by the results of all measurements rather than the individual points corresponding to the maximum measured values. Furthermore, let us move onto relative values by dividing the calculated and the experimental values into the corresponding averages for the set. The dependence of the calculated and the experimental values on $\varphi_{rel} = \varphi_j/2\pi$ is shown in fig. 2.

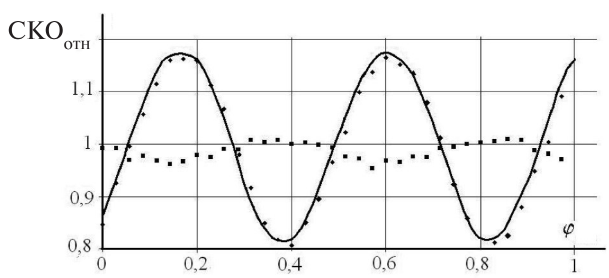

Fig. 2. Oscillation intensity over the wheel circumference versus the keyphasorvibration: calculated (–) and experimental (♦) values using the blade RPM sensor; experimental values (■) using the conventional RPM sensor.

It can be seen that there is a good agreement between the experimental and the calculated results with the blade RPM sensor used. This indicates that the 'measured' differences in oscillation intensity over the wheel are fully explained by the vibration of the keyphasor rather than the actual differences in blade amplitudes. Fig. 2 also shows the dependence of RM-S_{rel} (denoted as СКО$_{отн}$ in the figure) on φ_{rel} using a conventional RPM sensor. Evidently, there is also a certain dependence of the oscillation intensity on the angular coordinate. Naturally, the dependence is much weaker, because the amplitude of the torsional oscillations of the rotors assembly near the RPM sensor is smaller than the blade flutter amplitude (see figs. 1, *a* and 1, *в*).

The weak dependence of the average RMS_{rel} value in the set on the vibration of the keyphasor was also confirmed experimentally: the RMS_{rel} value obtained using the blade RPM sensor was only 1,6% higher than the one obtained using the conventional sensor. The differences for the registered flutter with the frequencies of about 108 Hz and 98 Hz were 0,4% and 0,05%, respectively, and did not exceed the measurement and calculation errors (it was previously noted that the error magnitude was proportional to $\sin^2(\pi f/n)$).

The effect of the keyphasor vibration on the errors in determining the oscillation frequency can be revealed by using the transformations similar to the ones previously performed. As before, let us consider virtually the most important case allowing to obtain the maximum results: $\sin(\psi f/2n) = 1$. To find the frequency, we need to calculate the sums that are a part of the right-hand side of formula (3); in this case and provided that $[(1/k_n)x_0 + 2y_0] = 1$ these sums take the following form:

$$\sum_{i=1}^{M} S_{1\Sigma}^2 = \frac{M}{2}[1 + \frac{n^2}{4f^2}\sin^2\frac{\pi f}{n} + \frac{n}{f}\sin\frac{\pi f}{n}\sin\frac{f}{n}(\varphi_j - \pi)]\,; \quad (16)$$

$$\sum_{i=1}^{M} \Delta S_{2\Sigma}^2 = \frac{M}{2}[1 + \frac{n^2}{4f^2}\sin^2\frac{\pi f}{n} + \frac{n}{f}\sin\frac{\pi f}{n}\sin\frac{f}{n}(\varphi_j + \Delta\varphi - \pi)]\,; \quad (17)$$

$$\sum_{i=1}^{M} \Delta S_{1\Sigma}\Delta S_{2\Sigma} = \frac{M}{2}[\cos\frac{\Delta\varphi f}{n} + \frac{n^2}{4f^2}\sin^2\frac{\pi f}{n} + \frac{n}{f}\sin\frac{\pi f}{n}\cos\frac{\Delta\varphi f}{2n}\sin(\varphi_j + \frac{\Delta\varphi}{2} - \pi)], \quad (18)$$

where $\Delta S_{2\Sigma}$ are the readings of the second sensor located at an angular distance $\Delta\varphi$ from the first one, also registering the mutual displacements.

The calculations and the experiments show that in this case the 'change' of the frequencies is also much stronger if using the blade RPM sensor than if using the conventional one, and is fully explained by the keyphasor vibration (see fig. 3).

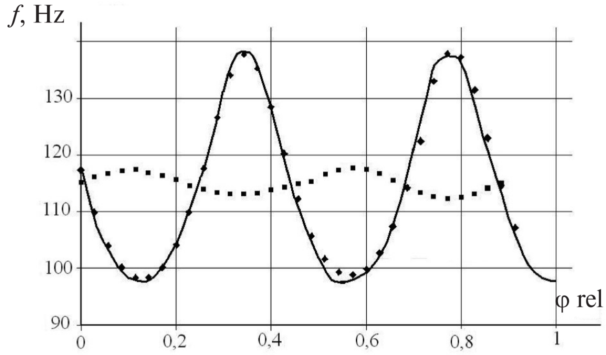

Fig. 3. Oscillation frequency over the wheel circumference versus keyphasor vibration: calculated (—) and experimental (◆) values obtained using the blade RPM sensor; experimental values (■) obtained using the conventional RPM sensor.

The results of the calculations and the experiments also confirm that the average frequencies in the set practically do not depend on the keyphasor vibration. For all three forms of flutter with the frequencies of about 115, 108 and 98 Hz the difference in the average set frequencies when using the blade and the conventional RPM sensors did not exceed a few tenths of percent and was smaller than the measurement and calculation errors.

It also follows from comparing the experimental data presented in figs. 2 and 3 that the torsional oscillations of the rotors assembly near the RPM sensor and the flutter of the stage examined occurred in antiphase.

When using the readings from the sensor pair with small bases *S*, i.e., when mainly the axial component of the amplitude of the peripheral section is registered, the longitudinal oscillations of the rotors assembly can have a significant effect on the measurement accuracy. The same goes for the axial oscillations of the stator components where the blade DPM sensors are installed, related to the longitudinal oscillations through the thrust bearing. For example, when registering the in-phase oscillations of the right and left blades of the penultimate stage with the frequency f_{meas}= 7,6 Hz, it was found that the average total oscillation amplitudes of these stages differed by 15%, while the axial components of the amplitudes differed by 1,5 times. Naturally, the natural forms at the same frequencies should be very close, and such a large difference in the x_0/y_0 ratios could be explained by the influence of some additional factors. Furthermore, when using formula (3), it was found that the average frequencies were equal to 111,5 Hz for one of the stages, and to 103,4 Hz for the second one, i.e., condition (2) was not satisfied. The measurement results for the average values in the set completely coincided when using the conventional and the blade RPM sensors.

The most likely cause for the significant differences in the x_0/y_0 ratios for the left and right stages are the longitudinal oscillations of the rotors assembly that for one stage were summed with the axial movements of the peripheral sections of the blades, and subtracted from them for the second stage. The differences in the 'measured' frequencies most likely depended on the difference in the levels of axial vibration of two pairs of sensors used to determine the frequency and the presence of the phase shift between the vibration of the stator components and the blade flutter. For example, calculations show that for the vibration in one of the pairs of sensors equal to 10% of the axial movement of the blades and the phase shift between the axial vibration and the flutter equal to 0.15π, the 'measured' flutter frequency is 103,6 Hz for one stage and 112,4 Hz for another (with the true frequency equal to 107,6 Hz). Approximately the same differences in the frequencies can be obtained provided that the axial vibration is only 5%, but the phase shift is 0,3π.

In some cases, determining the flutter frequencies by formula (3) based on measuring only the axial amplitude component of the peripheral section is impossible. For example, when registering flutter with f_{meas}=2,2 Hz using formula (3), it was obtained that f = 103,4 Hz for one of the penultimate stages, and 99,9 Hz for another. Since in view of relation (2) the true frequency could equal either 97,8 or 102,2 Hz, it was necessary to use another, more reliable way to determine the frequency.

The true value of the flutter frequency was determined using formula (7) when determining the dependence of the mutual displacement values on the angular distance ψ between the blades. The results of the calculations for the frequencies of 97,8 and 102,2 Hz are shown in aig. 4. The experimental curve for the mutual displacement values versus ψ is also plotted in that figure. The relative values of the mutual displacement $A_{rel} = A(\psi)/A_{max}$ are presented versus $\psi_{rel} = \psi/2\pi$.

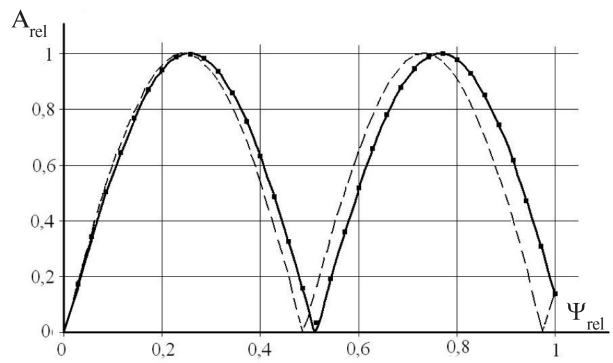

Fig. 4. Mutual displacement values versus angular distance between the blades: the calculated (—) and the experimental (■) values at a frequency of 97,8 Hz; the calculated values (---) at a frequency of 102,2 Hz.

It can be seen that the experimental points practically coincide with the calculated curve for f = 97,8 Hz and are significantly different from the calculated curve for f = 102,2 Hz (especially for large values of ψ_{rel}).

The described measurement technique allows to obtain not only the data on blade oscillations (including blade flutter with in-phase modes) that is comprehensive from a practical standpoint, but also some interesting data on the state of some other components of the turbine unit, especially when it is operating in the transition modes. For example, data can be obtained on the torsional and longitudinal oscillations of the rotors assembly, as well as on the elastic spin-up of the rotors assembly, the axial shift of the rotor and the static deformation of the blades during rapid ceasing of power; on the change of the relative rotor expansions in the spots where the DPM sensors are installed during a change in the operation mode of the turbine unit. Determining the possibility of a phase shift between the axial and the tangential components of the blade oscillations due to torsional and longitudinal oscillations of the rotors assembly may be of interest for assessing the magnitude of the air damping (or air excitation) when calculating the non-stationary flow effect on the oscillating blade row.

The following conclusions can be formulated based on the above.

1. Using a modified measurement technique allowed to determine the total amplitude, as well as the sum and the difference of the axial and tangential

components with the help of the DPM sensors designed to register only the axial component of oscillations.

2. When registering the flutter with in-phase modes using the upgraded version of the DPM, we have found additional errors that were absent when registering the flutter with disc modes. The appearance of the additional errors can be attributed to the fact that the in-phase blade flutter causes torsional and longitudinal vibrations of the rotors assembly and may cause axial vibration of the stator components where the DPM sensors are installed.

3. The consequence of the keyphasor vibration is the 'natural change' in the amplitudes and frequencies of the blade oscillations over the wheel circumference. However, it should be kept in mind that the average amplitudes and frequencies in the set virtually do not depend on the keyphasor vibration.

4. The axial vibration of the stator components and the longitudinal oscillations of the rotors assembly mainly affect the accuracy of determining the axial amplitude component of the peripheral section of the blade and can lead to errors in determining the oscillation frequency. Therefore, the level of dynamic stresses in the blades must be assessed by the value of the tangential component of the oscillation amplitude.

5. The most accurate value of the flutter frequency can be determined if the experimental dependence of the mutual blade displacements on the angular distance between them is compared with the calculated value obtained using formulae (2) and (7).

6. In order to improve the operational reliability of turbine units, systems for monitoring the vibrational state of the blades, based on the upgraded version of the DPM, should be installed both in new high-power turbines, and in the turbines with fatigue damage of the blades.

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