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VIBRATIONAL STRESSES OF DAMAGED STEAM TURBINE BLADES AFTER RENOVATION REPAIR

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The last-stage blades of K-1000-60 / 3000 steam turbines operate in a humid steam environment, which causes erosion damage in the blades and reduction in their residual life. The relevance of this work is related to the need to continue the safe operation of such turbine blades. A number of variants of the finite-element models of individual blades and last-stage blades in the disk-blade systems of the above turbines are considered. Results of the numerical study of the influence of blade part removals in erosion damage zones after renovation repair on the vibration characteristics of individual blades and blades in the disk-blade system are presented. An analysis of the stress-strain state under the conditional load from the steam flow during the forced oscillations of individual blades and blades in the disk-blade system is carried out. The loads are given as evenly distributed and linearly variable on blade surfaces. The dependence of the maximum equivalent vibration stresses on excitation frequency is determined. It is assumed that the physical and mechanical properties of the blade material are preserved (as for the original version) after the renovation repair of blades and processing of their surfaces. There is a significantly greater reduction in the vibration stresses of blades in the disk-blade system than in the stresses of individual blades. Graphs of the dependence of the maximum stresses on excitation frequency both for undamaged individual blades and blades in the disk-blade system after their renovation repair are given. Various variants of blade part removals in areas of blade leading and trailing edges are considered. It is shown that with decreasing chords of blades after renovation repair, frequency regions of increased vibration may appear in lower blade parts. In the lower parts of individual blades and blades in the disk-blade system, the maximum stresses increase in comparison with their values in undamaged blades. With the change in the stress-strain state of rotor blades in comparison with the original version of undamaged blades, the possibility of extending their safe lifetime in case of multi-cycle fatigue is assessed. The safe lifetime of the considered blades with a chord of at least 150 mm after their renovation repair can be extended according to their stresses, if the cyclic symmetry of the disk-blade system is not violated, and the physical and mechanical properties of the material are preserved after the processing of damage removal zones on blade trailing edges.

Keywords*: blade, vibrations, erosion damage, renovation repair, lifetime extension, three-dimensional finite element model, disk-blade system, forced oscillations, amplitude-frequency characteristic.*

Introduction

The disk-blade system of modern turbines is their most responsible and stressful part. Rotor blades are subjected to intense static and dynamic loads. Blade oscillations significantly affect the reliability of turbines.

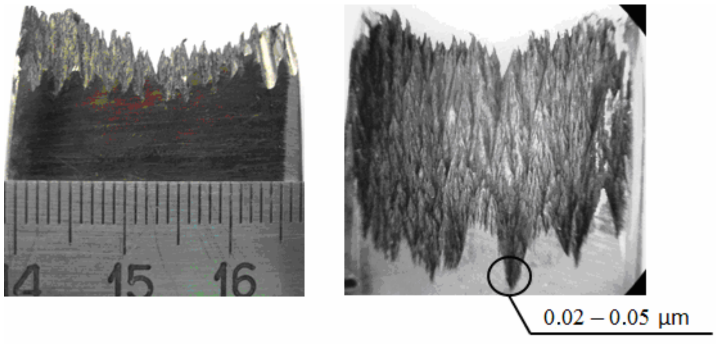
The last-stage Ti-alloy blades of a K-1000-60/3000 turbine are considered in the work.

The blades of the low-pressure cylinder stages of these turbines operate in a humid steam environment, where droplet impact erosion occurs. After more than 180 thousand operating hours, there are noticeable erosion zones on a number of blades [1]. Under the surface of the developed erosion damages, there are sometimes traces of localization of plastic deformations, which can lead to the emergence of microcracks, which in turn leads to emergency situations. The erosion damages in the last-stage blades of turbines operating in wet steam conditions are located quite densely in the upper thirds of blade leading edges and in the lower thirds of blade trailing edges [1].

The above erosion damages are sawtooth-like in nature, and have 400 to 700 μm spaced and 400 to 500 μm deep caverns. In their mouths, the radii of rounding are about 0.02 to 0.05 μm, which also causes a concentration of equivalent vibration stresses (hereinafter referred to as stresses), but not as significant as in the case of a fatigue crack (Fig. 1) [1]. In this case, the nature of damage is close to small nicks and dust erosion, which are observed, for example, in the Ti-alloy compressor blades of gas turbine engines [2].

For such blades it is recommended to carry out the smoothing of the whole zone of damages during preventive stops of turbines by various types of mechanical processing: turning, milling, grinding, polishing, etc. [1, 3]. This changes the geometric dimensions and shape of blade cross-sections.

In [3], the authors consider the finite element model of vibrations in individual rotor blades and results of the numerical analysis of frequencies and forms of natural vibrations, as well as distribution of relative stresses with account taken of centrifugal forces.



a b

Fig. 1. General view of erosion damage on the sides of a blade airfoil:
a – on the back; b – on the front

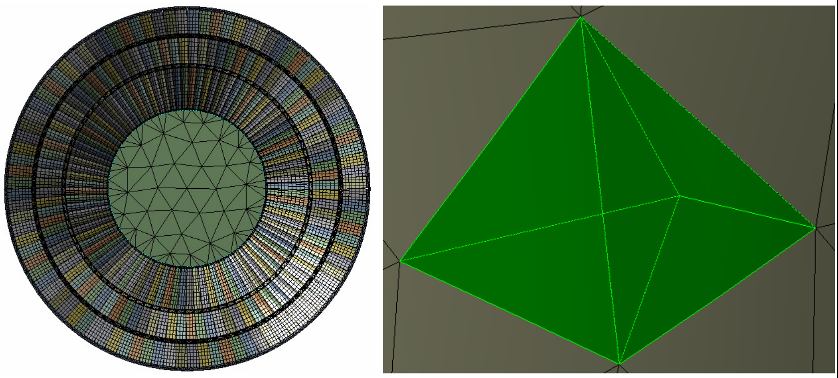


Fig. 2. Finite element model of the disk-blade system and a finite element

Forced Oscillations of Individual Blades After Renovation Repair

In this work, considered are the forced oscillations of individual blades and undamaged blades in the blade-disc system, as well as after the removal of erosion damages and the necessary processing of removal zones in the upper and lower blade parts after the repair. Several options for the possible removal of the most severe erosion damage zones in the blades are considered, and the impact of changes in blade geometry on vibration characteristics and blade stress state is evaluated. The dimensions of the cutouts in the lower and upper blade parts on the disk are considered to be the same for all the blades, which does not violate the cyclic symmetry of the system. Finite element models of blades with removed erosion damage zones have been developed.

The finite element model of an individual blade and the model of a blade in the disk-blade system consist of almost 15 and 60 thousand elements and more than 38 and 175 thousand nodes, respectively. Prismatic and tetrahedral curved finite elements were used. Fig. 2 shows a general view of the finite element model of the disk-blade system and a finite element.

In the erosion damage zones in the lower blade thirds, near blade trailing edges, three variants of material cutouts are considered (Fig. 3): a cutout up to 8 mm deep (Fig. 3, a); a cutout up to 28 mm deep (Fig. 3, b); a cutout up to 48 mm deep (Fig. 3, c).

Note that in undamaged blades the chord is 175 mm.

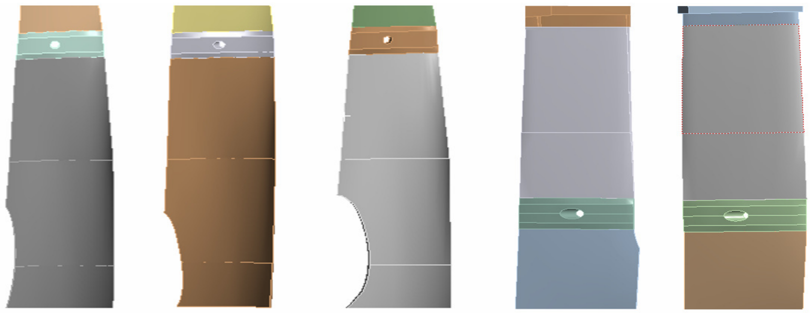
In the model of each individual blade, in the upper third of its leading edge, the change in geometry was taken into account compared to that in an undamaged blade, namely, the material cutout in the area of maximum stresses (maximum cut-out depth is 18 mm), the cutouts on the blade suction and pressure surfaces at angles to the profile chord, Fig. 3, d, e). Note that with this variant of the cutout in the upper blade part, the values of maximum stresses are 10% higher compared to those in the same part of an undamaged blade. For all the cases, it was assumed that with the removal of the damaged blade material after renovation repair, its physical and mechanical properties are the same as those in undamaged blades. The conditional load from the steam flow on a blade is given as evenly distributed (1 MPa) on the blade pressure surface and linearly variable from 1 MPa from the blade airfoil to the blade root, where the zero value of the load is taken [4].

The calculation of the stress-strain state of an individual blade (rigidly fixed at the root) under the action of a variable conditional linear load from zero to 1 MPa with a frequency of 2,100 Hz shows that the maximum values of stresses in the material cutout zone, after the repair of an individual blade, decrease. In the upper blade thirds (cutouts in blade leading edges), the following values of stresses were obtained (Fig. 4): 12.24 MPa (this area is located on the blade suction surface) (Fig. 4, a); 2.15 MPa (Fig. 4, b); 2.21 MPa (Fig. 4, c); 2.1 MPa (Fig. 4, d).

The zones of maximum stresses in all the variants are located similarly, to the left of the centers of the upper blade thirds. It should be noted that with decreases in chords in the lower blade parts (cutouts in blade trailing edges), the zones of maximum stresses in the upper blade parts increase.

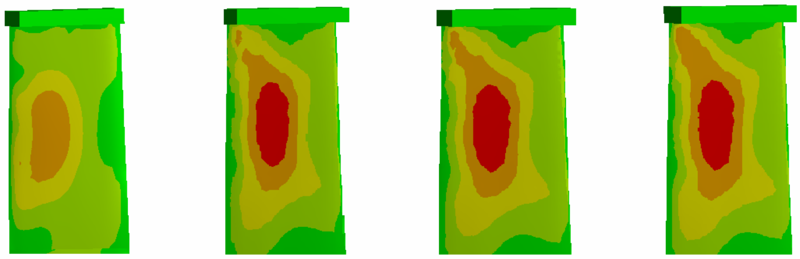
For the lower blade thirds, the following values of maximum stresses were obtained (Fig. 5): 0.44 MPa (Fig. 5, a); 0.38 MPa (Fig. 5, b);), 0.36 MPa (Fig. 5, c); 0.44 MPa (Fig. 5, d).

The maximum values of stresses in all the variants of material cutout are located at blade trailing edges near blade attachment areas (Fig. 5). It can also be noted that with a decrease in blade chord length, the zone of increased stresses increases, but the maximum stress values decrease as the blade chord length decreases due to the linear decrease in load.



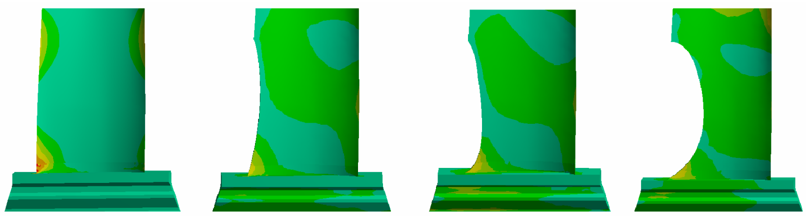
a b c d e

Fig. 3. Models of a blade (after material cutout):
a – with a of 170 mm chord; b – with a 150 mm chord; c – with a 130 mm chord; g – with a cutout in the upper third; e – without damage



a b c d

Fig. 4. Distribution of stresses in the upper part of a blade:
a – without damage; b – with a 170 mm chord; c – with a 150 mm chord; d – with a 130 mm cord, in the lower part after processing



a b c d

Fig. 5. Distribution of stresses in the lower part of a blade:
a – without damage; b – with a 170 mm chord; c – with a 150 mm chord; d – with a 130 mm chord, in the lower part after processing

Investigations of the forced oscillations of blades with a conditional load from the steam flow allow us to estimate the concentration factors of stresses. They are equal to the ratio of the stress in a damaged blade to the maximum stress in an undamaged blade, and have the following values: in the upper blade part: the concentration factor for the first option of processing in the lower blade part is $K_{1\text{в}}$=2.15/12.24=0.2; for the second processing option, $K_{2\text{в}}$=2.21/12.24=0.18; for the third processing option, $K_{3\text{в}}$=2.1/12.24=0.17, whereas in the lower blade part, the concentration factor for the first processing option is $K_{1\text{н}}$=0.38/0.44=0.86; for the second processing option, $K_{2\text{н}}$=0.36/0.44=0.82; for the third processing option, $K_{3\text{н}}$=0.44/0.44=1. It should be noted that at a uniform load of 1 MPa throughout the blade pressure surface, the values of concentration factors do not change significantly.

According to the results of calculations of the amplitude-frequency characteristics for the stresses of individual blades under linear loading, resonant frequencies and stresses in material removal zones at these frequencies are determined. A sharp increase in stresses in the lower blade parts is observed twice in the range from 100 to 200 Hz, a blade with a 130 mm chord having the largest values in this range. Also, a sharp increase in stresses is observed at a frequency of 345 to 355 Hz, and for blades with 170, 150 and 130 mm chords, also at a frequency of 541 to 550 Hz (Fig. 6).

The resonant frequencies and stresses in material removal zones at these frequencies for the upper blade parts are shown in Fig. 7. At frequencies of 100 and 200 Hz, there is an increase in stresses in undamaged blades, and at a frequency of 350 Hz, for all other variants of the blades. The next increase in stresses occurs in the ranges from 508 to 552 Hz and from 791 to 800 Hz. Starting from the frequency of 1,020 Hz, stresses in undamaged blades increase to 41.2 MPa at a frequency of 2,258 Hz. Stresses in the lower thirds of blades with removed damage zones also increase, but insignificantly. The maximum value of stresses for these blades is observed at a frequency of approximately 2,400 Hz, and is equal to 10.83 MPa.

At a uniform load of 1 MPa, the amplitude-frequency characteristic differs insignificantly in shape in comparison with the linear-variable load. The values of the maximum stresses also increase insignificantly.

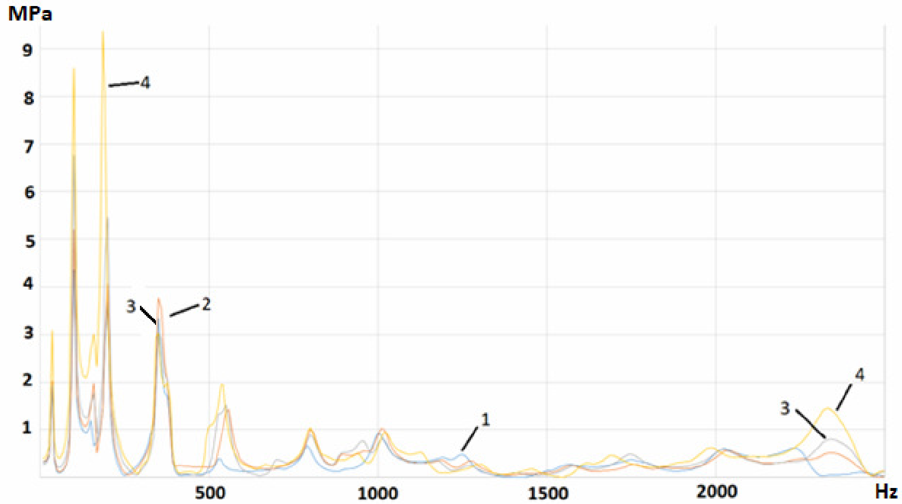


Fig. 6. Amplitude-frequency characteristic for the stresses in the lower part of a blade:
1 – without damage; 2 – with a 170 mm chord; 3 – with a 150 mm chord;
4 – with a 130 mm chord, under linearly variable loading

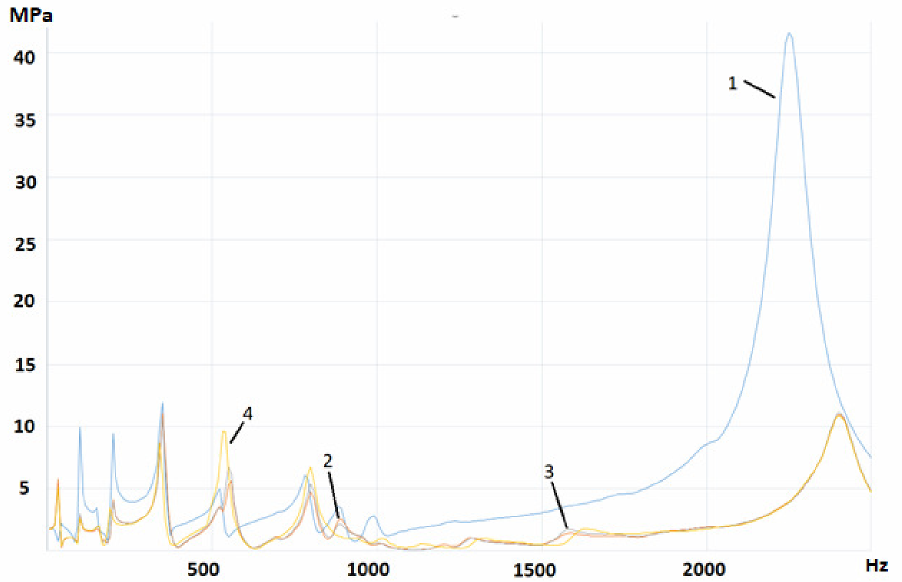


Fig. 7. Amplitude-frequency characteristic under linearly variable loading for stresses in the upper part of a blade:
1 – without damage; 2 – with a 170 mm chord; 3 – with a 150 mm chord;
4 – with a 130 mm chord, in the lower part after processing

Forced Blade Oscillations in the Disk-Blade System After Renovation Repair

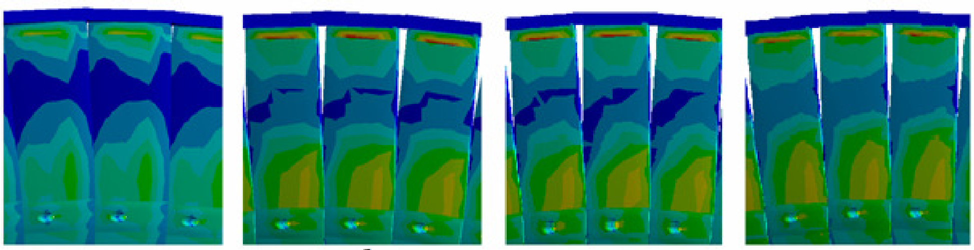
To study blade oscillations in the disk-blade system, a finite element model of this system was created (Fig. 2). Blades in the upper part of the system have shroud shelves. The possible displacement of adjacent shroud shelves and the influence of the damping wire were not taken into account. The question of the possible influence of these factors on blade oscillations is considered in [4].

The investigations of forced oscillations of blades in the disk-blade system with loads from the steam flow allow us to estimate the concentration factors of stresses. To do this, it is assumed that in the operating mode, there is an excitation force caused by the steam flow from guide vanes, the steam frequency being an aliquot of the rotation frequency of 50 Hz and the number of guide vanes (42), and equaling to 2,100 Hz. The load from the 2,100 Hz steam flow is assumed to be equal to 1 MPa in the upper blade part, decreasing linearly to zero at the blade root and is evenly distributed on the blade surface [4]. These loads are conditional, they are taken in order to estimate their stresses compared to stresses in undamaged blades. The damping in the disk-blade system was taken into account according to Feucht, using the appropriate factors. The calculations made it possible to determine the concentration factors of stresses in blades after renovation repair, which are equal to the ratio of the above stresses.

First, we present the results obtained under the distributed load that decreases linearly to zero near the blade root.

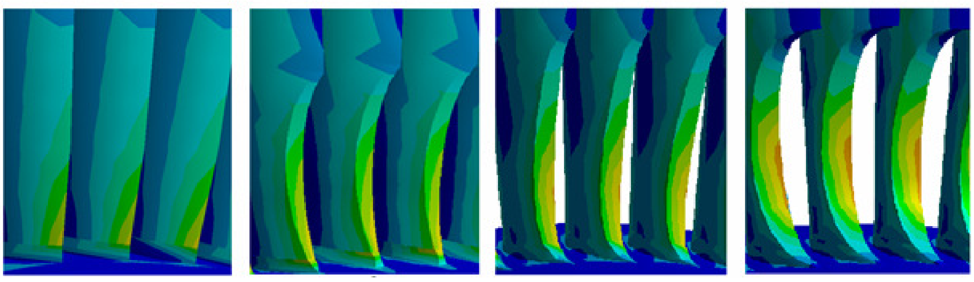
In undamaged blades, the maximum stresses are 2 kPa in the upper part of the upper blade third, and the minimum ones are 0.2 kPa in the center of the upper blade third. In blades with a 170 mm chord, stresses do not exceed 2.3 kPa and are located near the shroud shelves where the minimum value of stresses is 1 kPa. In blades with a 150 mm chord, stresses reach 2.5 kPa, while their lowest value is 1.8 kPa. In blades with a chord of 130 mm, stresses are in the range from at least 1.8 kPa to 2.2 kPa. In all of the above cases, the maximum values of stresses are observed nearer to shroud shelves and near blade leading edges, and the minimum values are nearer to the middle parts of the upper blade thirds. Note that the size of the zone of maximum stresses in the upper blade third increases by 2–4 times when the damaged material is removed in the lower blade third, and the value of minimum stresses decreases (Fig. 8).

For the root part of a blade, the following results were obtained: in undamaged blades, the maximum stresses are 2.2 kPa near the root of the blade trailing edge, and the minimum stresses of 1.3 kPa are observed



a b c d

Fig. 8. Distribution of stresses in the upper part of a blade (for the case of processing the lower part of its trailing edge):
a – without damage; b – with a 170 mm chord; c – with a 150 mm chord; d – with a 130 mm chord



a b c d

Fig. 9. Distribution of stresses in the lower part of a blade (for the case of processing the lower part of its trailing edge):
a – without damage; b – with a 170 mm chord; c – with a 150 mm chord; d – with a 130 mm chord

nearer to the middle part of the lower blade third; in blades with a 170 mm chord, stresses reach 1.7 kPa and are farther from the root than the maximum value of stresses in undamaged blades, their minimum value being 1 kPa; in blades with a 150 mm chord, the values of stresses are in the range from 0.8 kPa to 1.9 kPa, reaching their maximum from the side of the blade pressure surface and slightly lower than the middle part of the cutout; in blades with a 130 mm chord, the maximum stresses reach 2.4 kPa, and are observed in the same zone as the maximum stresses in blades with a 150 mm chord, their minimum value being 1.3 kPa (Fig. 9).

If the damaged material is removed in the lower blade third, the zone of maximum stresses increases and moves from the blade root to the middle part of the cutout. It is possible to note the relative similarity of the distribution of stresses for these variants, but at the same time the levels of stresses differ significantly.

The concentration factors of stresses that are equal to the ratio of stresses in damaged blades to the maximum stresses in undamaged blades are as follows: in the upper blade parts, the concentration factor for the first processing option is $K_{1в}$=2.3/2=1.15; for the second option, it is $K_{2в}$=2.5/2=1.25; and for the third one, it is $K_{3в}$=2.2/2=1.1. In the lower blade parts, the concentration factor for the first processing option is $K_{1н}$=1.7/2.2=0.77; for the second option, it is $K_{2н}$=1.9/2.2=0.86; and for the third one, $K_{3н}$=2.4/2.2=1.09. Note that the concentration factors of stresses in the blade roots near the disk, at an excitation frequency of 2,100 Hz, for blades with 170, 150 and 130 mm chords are 1.3, 1.32, and 1.6, respectively.

The presence of a cutout in the lower third of a blade with a 150 mm chord affects the stresses in the upper third of the blade, and the value of stresses in this area increases by 10–25%. At the same time, there is a decrease in the level of stresses in the lower thirds of the trailing edges of blades with 170 and 150 mm chords to the values that are smaller than those in undamaged blades.

The decrease in the values of stresses in some cases can be explained by the fact that with the change in blade geometry, the vibrational characteristics of the system change as well, which is confirmed by the study of the amplitude-frequency characteristics according to the maximum stress values of blades in the disk-blade system for different options of damage removal. The dependence of the maximum stresses in the lower thirds of blades in the disk-blade system on excitation frequency is shown in Fig. 10. The results are given for frequency values of up to 175 Hz, because for higher frequencies there is a gradual decrease in stresses.

Based on the results of the calculations of the amplitude-frequency characteristics of the disk-blade system, it is possible to determine the resonant frequencies and stresses in the material removal zones at these frequencies. For all the variants of the disk-blade system, the maximum of amplitudes is observed in the range from 30 to 32.5 Hz. In this frequency range, for undamaged blades, the maximum value of stresses is 1.08 MPa at a frequency of 32.5 Hz; for blades with a 170 mm chord, it is 1.65 MPa at a frequency of 31 Hz; for blades with a 150 mm chord, it is 1. 51 MPa at a frequency of 31.25 Hz; for blades with a 130 mm chord, it is 3.12 MPa at a frequency of 30 Hz.

The next sharp increase in stresses is in the range from 45 to 57.5 Hz. In this frequency range, the maximum value of stresses for undamaged blades is 0.245 MPa at a frequency of 57.5 Hz; for blades with a 170 mm chord, it is 0.313 MPa at a frequency of 56 Hz; for blades with a 150 mm chord, it is 0.269 MPa at a frequency of 53.75 Hz; for blades with a 130 mm chord, it is 0.351 MPa at a frequency of 45 Hz. It should be noted that when the chord decreases from 150 to 130 mm, the resonant frequencies decrease (for example, from 53.75 to 45 Hz and from 136.5 to 121.25 Hz), and the maximum stresses increase, their significant increase being observed at a frequency of 50 Hz (Fig. 10).

Another sharp increase in stresses is observed in the range from 121.25 to 143.75 Hz. The maximum value of stresses for undamaged blades is 0.18 MPa at a frequency of 143.75 Hz; for blades with a 170 mm chord, it is 0.329 MPa at a frequency of 141 Hz; for blades with a 150 mm chord, it is 0.14 MPa at a frequency of 136.25 Hz; for blades with a 130 mm chord, it is 0.295 MPa at a frequency of 121.25 Hz. As the frequency of the external force increases, the values of stresses gradually decrease for all the variants of blades.

At a blade vibration excitation frequency of 2,100 Hz, the maximum value of stresses for option 1, without damage, is 2.2 kPa; for option 2, it is 1.7 kPa; for option 3, 1.9 kPa; for option 4, 2.4 kPa.

For blades with a 130 mm chord, a significant effect on the stress-strain state is exerted by the stress of centrifugal forces. There is an increase in stresses near blade roots. In addition, in the frequency range from 0 to 2,100 Hz, the maximum stresses in blades with a 130 mm chord at individual frequencies are 2–2.5 times higher than in blades with a 150 mm chord.

The amplitude-frequency characteristic at the maximum stresses for the upper blade thirds was also obtained for different types of material removal in the lower blade thirds.

A sharp increase in the values of maximum stresses is observed in undamaged blades at a frequency of 31 Hz, the maximum stress value being 0.87 MPa. For blades with a 170 mm chord, the maximum stress value is 0.816 MPa at a frequency of 31 Hz. For blades with a 150 mm chord, the maximum stress value is 1.47 MPa at a frequency of 30.9 Hz. For blades with a 130 mm chord the maximum stress value is observed at a frequency of 30 Hz and is 2.7 MPa.

The next sharp increase in stresses is in the frequency range from 45 to 57 Hz. In this range, the maximum stress value for undamaged blades is equal to 0.19 MPa at a frequency of 57 Hz, for blades with a 170 mm chord, it is 0.2 MPa at a frequency of 56.6 Hz; for blades with a 150 mm chord, it is 0.264 MPa at a frequency of 54.8 Hz; and for blades with a 130 mm chord, it is 1.12 MPa at a frequency of 45 Hz.

Another sharp increase in stresses is observed in the range from 121 to 144 Hz. The maximum stress value for undamaged blades is 0.36 MPa at a frequency of 141 Hz; for blades with a 170 mm chord, it is 0.36 MPa at a frequency of 140 Hz; for blades with a 150 mm chord, it is 0.43 MPa at a frequency of 137 Hz; for blades with a 130 mm chord, it is 0.94 MPa at a frequency of 121 Hz. As the frequency of the external force increases, the values of stresses gradually decrease for all the variants of blades (Fig. 11).

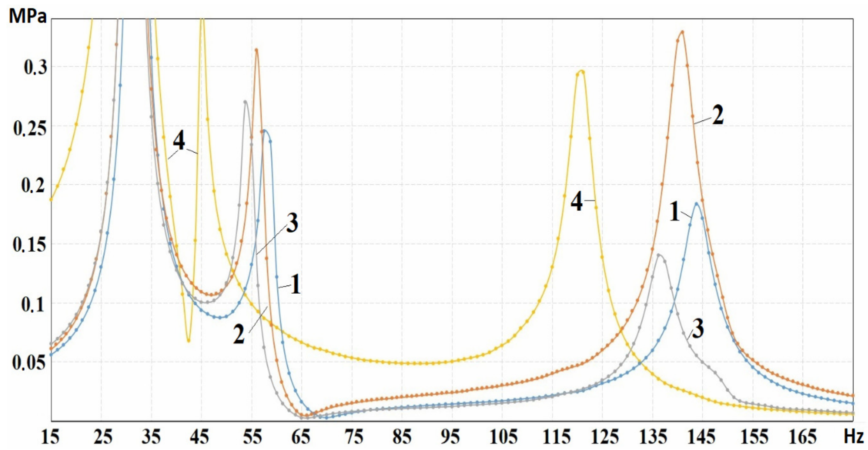


Fig. 10. Amplitude-frequency characteristic at the maximum stresses in the lower part of a disk blade:
1 – without damage; 2 – with a 170 mm chord; 3 – with a 150 mm chord;
4 – with a chord of 130 mm, in the lower third of the blade

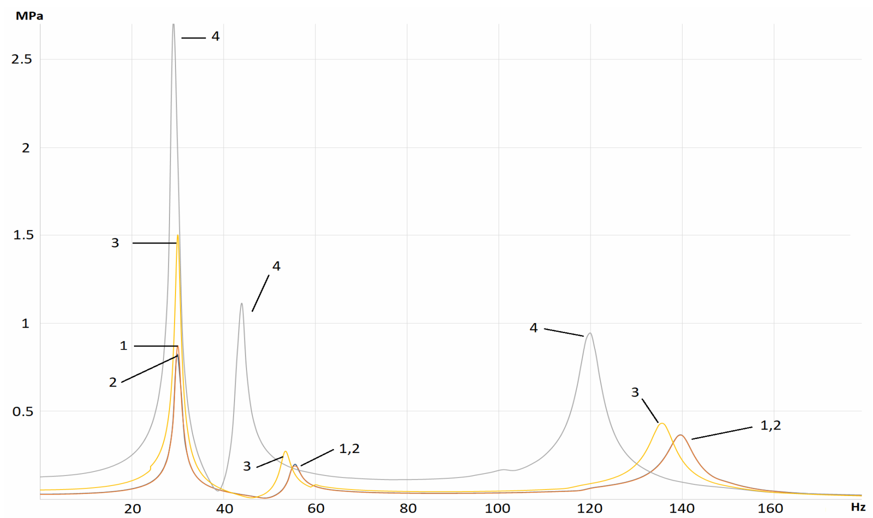


Fig. 11. Amplitude-frequency characteristic at the maximum stresses in the upper third of a disk blade:
1 – without damage; 2 – with a chord of 170 mm; 3 – with a 150 mm chord;
4 – with a 130 mm chord, in the lower blade thirds

Fig. 12 shows the amplitude-frequency characteristic at the maximum stresses in the upper blade thirds during the removal of the damaged material in the upper and lower blade thirds. It should be noted that the presence of a cutout only in the upper blade third (the lower third is not damaged) significantly increases the maximum stresses in the upper blade part compared to the same part in undamaged blades (curve 1) and with blades with cutouts in their lower and upper thirds (curve 4). This increase in stresses is observed at a frequency of 30.8 Hz. The maximum stress values are: 0.67 MPa for undamaged blades, 0.816 MPa for blades with cutouts in their upper and lower thirds, 5.16 MPa for blades with cutouts only in their upper parts. Also, a significant increase for this type of cutouts is observed at frequencies of 58.8 and 145.4 Hz. At an operating rotational speed (50 Hz), the maximum stress value for undamaged blades is 25 kPa, and for blades with cutouts in their upper thirds, it is 62 kPa.

The above results were also obtained for the uniform loading over the entire area of the blade pressure surface to assess the effect on the results of this type of loading. The results of comparing the amplitude-frequency characteristics at maximum stresses are shown in Fig. 13.

Thus, for blades with a uniform load in the disk-blade system, three resonant modes were detected in the frequency ranges from 29 to 31 Hz, from 55 to 58 Hz, and from 136 to 146 Hz. For undamaged blades, the first resonance maximum stress value is observed at a frequency of 30.83 Hz at a uniform load of 1 MPa (curve 1), whereas for the load from 0 to 1 MPa (curve 2) the frequency is 30.8 Hz. The values of stresses are 4.8 and 4.1 MPa, respectively. The second resonance maximum stress value is observed at frequencies of 57.5 and 56.7 Hz for curves 1 and 2. The values of stresses are 0.2 and 0.53 MPa, respectively. The third resonance maximum stress value is observed at frequencies of 144.17 and 145.8 Hz. The values of stresses are 0.73 and 0.51 MPa for curves 1 and 2, respectively. The change in the maximum stresses, depending from the load frequency for different calculation options, is explained by the change in the forms of forced oscillations on which the formation of the response (reaction) of the system depends.

For blades with a 150 mm chord, the first resonance maximum stress value is observed at a frequency of 30.86 Hz for a uniform load of 1 MPa (curve 3), for the variant with a variable load (curve 4) the frequency is 31.25 Hz. The values of stresses are 1.54 and 1.5 MPa, respectively. The second resonance maximum stress value is observed at frequencies of 53.71 and 53.75 Hz, and the values of stresses are 0.2 and 0.27 MPa for curves 3 and 4, respectively. The third resonance mode is observed at frequencies of 136 and 136.25 Hz, and the values of stresses are 0.2 and 0.14 MPa for curves 3 and 4, respectively.

The above results show that with an evenly distributed and linearly variable loading on blades, there is (Fig. 11) a qualitative correspondence of the amplitude-frequency characteristic at the maximum values of stresses with insignificant changes in their magnitude.

Evaluating the results of the study of the stress-strain state of individual blades and blades in the disk-blade system, we can note the following.

The stress-strain states of individual blades and blades in the disk-blade system differ significantly in shape and in the values of maximum stresses, because the disk-blade system is more rigid. Therefore, to address their reliability, it is more appropriate to study the results of the stress-strain state of blades in the disk-blade system. Such a system has dynamic properties different from those in individual blades.

The research results show that when the cross-sectional area near the root part of a blade with a 150 mm chord decreases after renovation repair, the maximum stresses at 2,100 Hz change insignificantly due to the change of the amplitude-frequency characteristic of the system. With a further decrease in the cross-sectional area, when the blade chord reaches 130 mm, there is an increase in stress by 10% relative to its value for the original version of the blade cross-section. At the above frequency ranges, there are maximum values of stresses for a blade with a 130 mm chord, which are 2–3 times higher than those for an undamaged blade.

To assess the influence of the change in the stress level on the limit of endurance of the blade material under multi-cycle loading, it is advisable to use the results of the study of fatigue endurance of TC-5 Ti-alloy blades, given in [1]. According to them, the endurance limit exceeds 350 MPa, and according to [5], it is equal to 350–460 MPa at a temperature of 20–25 °C. It is known that with increasing temperature, the endurance limit of Ti alloys decreases significantly [2]. In the steam turbines of thermal power plants, the temperature in high- and medium pressure cylinders reaches 540 °C and more. In this case, plastic deformations appear in stress concentration zones and the lifetime of such elements is determined by the method of accumulation of damage in different modes of operation [6, 7]. For elements of the last stages of low-pressure cylinders of turbines in thermal and nuclear power plants, where the temperature does not exceed 30–65 °C, the change in the endurance limit will be insignificant relative to the temperature range of 20–25 °C. Therefore, in order to estimate their lifetime after the repair we can use the fatigue endurance values for Ti alloys [1, 2, 5], which are in the range of 350–460 MPa.

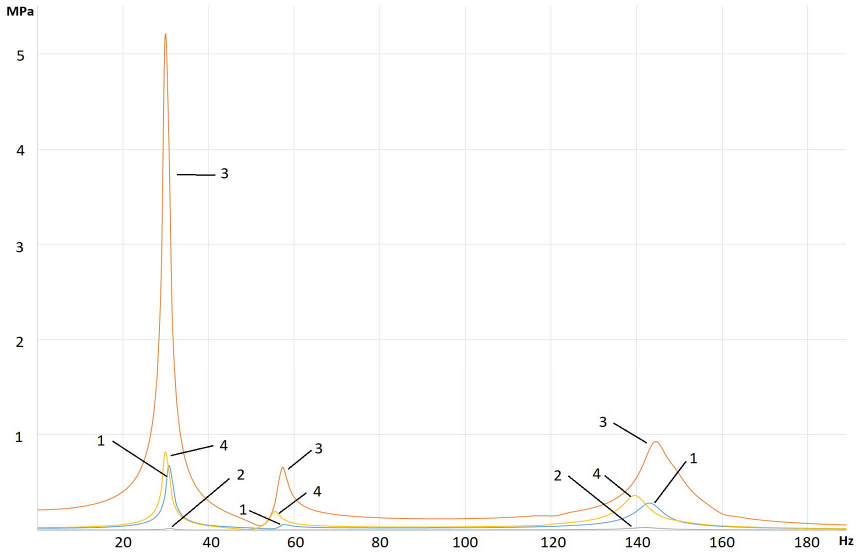


Fig. 12. Amplitude-frequency characteristic at the maximum stresses in the upper third of a disk blade:
1 – without damage in the upper and lower thirds; 2 – with a 170 mm chord, the upper part is without damage;
3 – the lower third is without damage, the upper one has a cutout; 4 – with cutouts in the upper and lower thirds

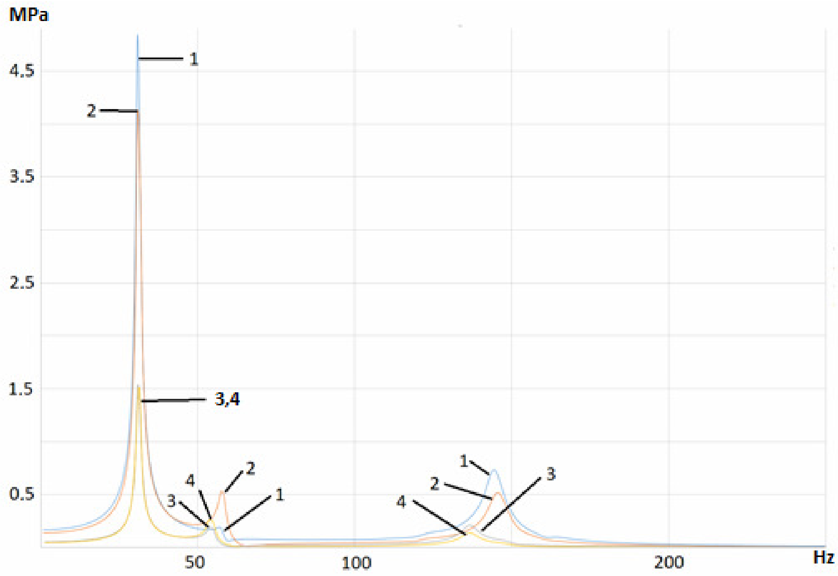


Fig. 13. Amplitude-frequency characteristic at the maximum stresses of a disk blade:
1 – without damage with a uniform load; 2 – with a linearly variable loading;
3 – with a 150 mm chord in the lower third, with a uniform load; 4 – with a linearly variable loading

If we assume that stresses in observable undamaged blades are smaller than 300 MPa [1], then with a reduction in blade chord length from 150 mm in the lower part of the blade trailing edge to 130 mm, stresses can increase 2–2.5 times, which significantly reduces the residual number of cycles before destruction. The obtained calculation data show that stresses at an excitation frequency of 30 Hz increase almost 3 times and 1.6 times at a frequency of 121 Hz relative to stresses in undamaged blades. The maximum stresses in blades with a 150 mm chord and a 130 mm chord are, respectively, 1.1 and 1.43 times greater at a frequency of 45 Hz, compared with stresses in undamaged blades. With a change in the cross-section of a blade when its chord decreases from 150 to 130 mm, the maximum stress values can be observed at a frequency of 50 Hz. The oscillations of these blades during the rotation of the disk will be kinematically excited by the vibration of the shaft line with a frequency of 50 Hz from the residual imbalance. The normalized warning values of the scope of vibration displacement of the shaft line in the plain bearings for such units are 165, and for emergency ones, 260 μm.

As the blade chord length decreases starting from 150 mm, areas of increased blade vibration may appear. As mentioned above, at certain frequencies up to 2,100 Hz for blades with a 130 mm chord, stresses are 2–2.5 times higher than for blades with a 150 mm chord, and for some frequencies they are 2–3 times higher for undamaged blades. All this indicates a possible increase in vibrational stresses in blades with a shorter than 150 mm chord. Given that after the repair, such blades must perform a significant number of load cycles added to those performed for 180 thousand hours of operation (about 1,012), their further use in view of the above is impractical [8].

In addition, it should be borne in mind that a greater change in geometry of blades leads to a significant deterioration of their aerodynamic properties, which adversely affects the efficiency of the stage, and can generate additional vibration loads on the disk-blade system.

Conclusions

The above results make it possible to conclude that the safe lifetime of Ti-alloy blades, with a chord of at least 150 mm, of the last stage of the low pressure cylinder of a K-1000-60/3000 turbine after the repair can be extended according to stresses, if the cyclic symmetry of the disc-blade system is not violated and the physical and mechanical properties of the blade material are preserved after the processing of removal zones.

In practice, there is uneven erosion of different last-stage blades, so it is important to assess the influence of uneven changes in the cross-sections of different blades after the repair on their stress-strain state in the disk-blade system.

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Вібронапруженість пошкоджених лопаток парової турбіни після відновлювального ремонту

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Лопатки останніх ступенів парових турбін типу К-1000-60/3000 працюють в умовах вологого парового середовища, що призводить до ерозійних пошкоджень і зниження їх залишкового ресурсу. Актуальність даної роботи пов'язана з необхідністю продовжити безпечну експлуатацію робочих лопаток таких турбін. Розглядається декілька варіантів скінченно-елементних моделей окремих лопаток та лопаток в системі диск-лопатки останнього ступеня турбіни зазначеного типу. Наводяться результати чисельного дослідження впливу видалення частин лопаток в зонах ерозійних пошкоджень після відновлювального ремонту на вібраційні характеристики окремих лопаток та лопаток в системі диск-лопатки. Проведено аналіз напружено-деформованого стану за умовного навантаження від паропотоку при вимушених коливаннях окремих лопаток та лопаток в системі диск-лопатки. Навантаження задаються рівномірно розподіленими та лінійно змінними по поверхнях лопаток. Визначається залежність максимальних еквівалентних вібраційних напружень від частоти збудження. Приймається, що фізико-механічні властивості матеріалу лопаток зберігаються (як для вихідного варіанта) після ремонту лопаток і обробки їх поверхонь. Спостерігається значно більше зниження вібронапруженості лопаток в системі диск-лопатки, ніж на окремих лопатках. Наводяться графіки залежності максимальних напружень від частоти збудження для непошкоджених окремих лопаток і лопаток в системі диск-лопатки після їх відновлювального ремонту. Розглядаються різні варіанти видалення частин лопаток в зонах їх вхідних та вихідних кромок. Показано, що зі зменшенням хорд лопаток після ремонту у їх нижніх частинах можуть з'являтися частотні області підвищеної вібрації. В окремих лопатках та лопатках в системі диск-лопатки в нижніх частинах значення максимальних напружень збільшуються в порівнянні зі значеннями в лопатках без пошкоджень. Зі зміною напруженості робочих лопаток в порівнянні з вихідним варіантом лопаток без пошкоджень оцінюється можливість продовження їх ресурсу безпечної експлуатації за багатоциклової утоми. Цей ресурс розглянутих лопаток з хордою не менше 150 мм після відновлювального ремонту може бути продовжений за даними напружень, якщо не порушується циклічна симетрія системи диск-лопатки та зберігаються фізико-механічні властивості матеріалу після обробки зон видалення пошкоджень на вихідних кромках лопаток.

Ключові слова: *лопатка, коливання, ерозійні пошкодження, відновлювальний ремонт, продовження ресурсу, тривимірна скінченно-елементна модель, облопачений диск, вимушені коливання, амплітудно-частотна характеристика.*

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