WORKING BLADES DEVELOPMENT FOR THE LAST STAGES OF STEAM TURBINE LOW PRESSURE CYLINDER

Riza SHERFEDINOV[1], Mykhailo ISHCHENKO[1], Liubov SLASTON[1] and Svitlana ALYOKHINA[2,3]

t[1]JSC "Ukrainian Energy Machines", Heroiv Kharkiv avenue, 199, Kharkiv, 61037, Ukraine

[2]A. Podgorny Institute of Mechanical Engineering Problems of the National Academy of Sciences of Ukraine, 2/10 Pozharsky str., Kharkiv, 61046, Ukraine, E-mail: alyokhina@ipmach.kharkov.ua

[3]National University of Radio Electronics, Nauky Ave. 14, Kharkiv, 61166, Ukraine

***ABSTRACT**: Paper presents the case study of the design of titanium alloy working blades of the last stage of low pressure cylinder of powerful steam turbines conducted by the lead Ukrainian enterprise JSC "Ukrainian Energy Machines". The key aspects of the design are in the increasing degree of reactivity in the blade root zone, the "reverse twisting" of the guide blades, the gas-dynamic optimization of the stage at a partial load and using titanium alloy for the blades. Paper highlights results of the stress state and vibration characteristics of the new blades of last stage of low pressure cylinder. The presented work is a part of the low-cost modernization carried out for the power installations with long-term operation time.*

***KEYWORDS**: steam turbine, low-pressure cylinder, working blade, strength, vibration*

1 INTRODUCTION

Turbines of K-1000-60/1500, K-220-44, and K-1000-60/3000 types operate at nuclear power plants of Ukraine (Virchenko et al., 1997). By now, considerable experience in the operation of steam turbine units K-1000-60/3000 has been accumulated; it indicates that working service of a significant part of their elements is practically finished according to the normative documentation. As a result, the operational reliability of the entire turbine unit is reduced, which requires taking measures to improve the level of safety (Kupetz et al., 2014), (Levchenko et al., 1997). However, it is not possible fully replace the existing equipment due to economic and technological reasons. Therefore this situation have required to search balance between new and existing technologies which could be implemented on the already operated power plants in relatively short terms and with the lowest cost as possible.

One of the elements that has a significant effect on the level of operational reliability are the working blades of the last stages of low pressure cylinders (LPC), which experience heavy loads of static, dynamic and cyclic nature. For example, perturbing forces, acting on the blade during the rotation of the turbine rotor, cause tangential vibration in the plane of the disk, which is complicated by axial and torsion vibrations, which in its turn can result in the damage to the blade system. The blades operate in an aggressive wet steam environment and are subject to droplet impact erosion, which results in damage to the leading edges with a subsequent decrease in reliability characteristics. So, this element of the steam turbine was choosing as perspective for the modernization.

The working blades of the LPC turbine of the K-1000-60/3000 type with an operating time of more than 100,000 hours and significant erosive wear, which is determined by the chord size of the active part, are subject to replacement in accordance with the manufacturer's regulatory documentation. Due to the fact that this type of turbine was not produced in Ukraine, the replacement of their structural elements is associated with a number of economic difficulties. It should be noted that the replacement of the entire turbine plant with modern turbine units from the world's leading manufacturers is not economically feasible, and the replacement of the most worn parts is impossible due to the difference in geometric characteristics and design operating modes of turbine. In order to solve this problem, an import substitution program was initiated, within the framework of which JSC "Ukrainian Energy Machines" started developing blades for the turbines mentioned above.

The purpose of the publication is to highlight main design studies of JSC “Ukrainian Energy Machines” when creating a working blade made of titanium alloy with an active part length of 1200 mm of the last stages of the LPC of turbines K-1000-60/3000 as part of the import substitution program for the South-Ukraine, Rivne and Khmelnytsky power stations.

2 ANALYSES AND FORMULATION OF THE PROBLEM

A number of research works are devoted to the issues how to create blades from titanium alloys in modern scientific literature (Brotzu et al., 2017) – (Ma et al., 2016). All of them describe in sufficient detail physical processes occurring in the blade system, based on numerical or experimental methods. A number of well-known works are devoted to the study of the characteristics of the blade system of gas turbines (Brotzu et al., 2017), (Vaferi et al., 2019). The results presented in these works cannot be fully used for the blades of steam turbines, since there are some differences in physical processes. Another class of research works either covers only narrow issues related to the design and development of blades (for example, gas dynamic (Rajamurugan et al., 2022) and vibration characteristics (Ma et al., 2016) of blades), or concerns a completely different type of blade system (Rajamurugan et al., 2022), (Mazarbhuiya & Pandey, 2017). One of the most complete descriptions of works in the direction of studying processes in the last stages of steam turbines can be considered (Tanuma, 2016). The author focuses on the active usage of the commercial software systems and careful separate studies of aerodynamic and mechanical designs issues.

However, modern developments in the field of turbine engineering can rarely be used in the modernization of turbines with a long service life due to the existence of a number of technological limitations. No works in this direction have been found in the open literature. In this regard, the adaptation of existing methods was required, and in some cases the development of new ones, which made it possible to modernize the existing steam turbines operated at Ukrainian power plants.

3 KEY ASPECTS OF THE DESIGN METHODOLOGY

The design of the working blade, which will be operated in turbine that already has long operation time, involves a comprehensive solution to a number of complex design issues. Performing calculations when designing a working blade involves the determination of thermal and gas-dynamic characteristics, stress-strain state, and vibration characteristics. The last two issues are the most important because of the theoretical possibility of turbine elements destruction at the safety parameters overriding.

Gas-dynamic processes play almost the most important role in the operation of the flow part of the turbine, therefore, designers and researchers pay the most attention to them when designing high-power steam turbines. The flow in the flow part of turbo machines is spatial and non-stationary. Such phenomena as secondary currents significantly affect the structure of the flow, given that their influence on the formation of the general picture of the flow increases even more. The general structure of the flow is also significantly affected by outflows into regeneration, heating and technological selections, overflows in overbandage and diaphragm seals. In order to take into account all these features, a special software complex was created, which effectively allows three-dimensional calculations of the flow in the last stages of the turbine (Rusanov et al., 2021), (Rusanov et al., 2020). The results of gas-dynamic calculations made it possible to more qualitatively form the boundary conditions for calculating the vibration characteristics of the turbine stages.

The observed tendency of the raise of the cooling water design temperature has made it interesting to increase the specific steam load of the last stage. The performed computational and experimental studies indicate that the main parameter determining the gas thermodynamics of the stage is the volumetric steam flow rate at the outlet from it. Therefore, an increase in the temperature of the cooling water and, as a consequence, the pressure in the condenser can be accompanied by an increase in the mass steam flow rate through it and, accordingly, by the specific steam load of the exhaust (Slaston et al., 2020).

Based on the experience in the design and operation of high-power turbines (Zaitsev et al., 1996), a number of fundamental provisions have been formulated:

– increased degree of reactivity in the root zone, which excludes the occurrence of its negative values even with a significant decrease in the volumetric steam consumption; the upper limit of the degree of reactivity is chosen at a level determined by the strength of the working blades.

– significant design heat drop (not less than 190 kJ/kg) permissible on grounds of strength and acceptable for the selected radial dimensions of the stages as well as a moderate degree of reactivity (50-60%) at the average radius; with a decrease in the volumetric flow rate, this contributes to a later transition of the stage to the ventilation mode, characterized by increased dynamic effects of the flow on the blades.

– "reverse twisting" of the guide blades with a decrease in the angle of outflow from them from the root to the periphery, providing a combination of reactivity at the root and at the average radius, indicated above, which is usually unattainable at

– stages of such a fanning according to traditional swirling laws (Davidenko et al., 2010).

– gas-dynamic optimization of the stage at a partial (about 80%) volumetric flow rate at the outlet of the stage to prevent the early occurrence of the root separation of the flow; at the same time, it is possible to ensure the optimal shape of the meridional streamlines and a satisfactory uniformity of the distribution of the output speed of the stage in the nominal mode as well (Shubenko et al., 2021).

For a working blade of the maximum size, if it is necessary to use ready-made and tested connection structures, it is necessary to introduce restrictions on the value of centrifugal forces of the blade or on the value of bending forces in the mathematical model.

The material of the working blades of the last stages installed in the already operated K-1000-60/3000 turbines is TC-5 titanium alloy. This alloy has a lower density compared to steels, which significantly affects the stress values under static loading from the centrifugal effect. With purpose to save general operation characteristics of the turbine-installation it is logically to choose material for the new blades from the same material but with improved characteristics. As it follows from the analysis of existing titanium alloys and alloys used in turbine construction, titanium alloy Ti6Al4V was chosen to improve reliability, erosion resistance and wear resistance (Rusanov et al., 2020).

The safe and long term operation of the modernized turbine cannot be started without stress and vibration analysis. The mathematical aspects of this problem simulation were in detail described in (Vorobiov et al., 2019). The methodology proposed by authors consists of combination of the finite-elements method and existing empirical methods (RTM, 1977).

4 RESULTS AND DISCUSSION

With purpose to demonstrate the key results obtained at the case of the low-cost modernization of the existing turbines the some calculations are presented. The focus is made on the stress and vibration simulation as the most important part of the safe operation of the whole turbine installation.

The new blade design provides for two holes for intermediate ties at a distance of 520 mm and 870 mm from the root section, respectively, as well as a shroud tie does at the periphery for vibration damping.

The assurance of the structural strength of long blades is largely determined by the degree of the uneven distribution of stresses in the zones of sharp deformation (Shi et al., 2021), (Vorobiov et al., 2019). The determination of the stress-strain state of the blade and tail connection was carried out according to the methods of JSC "Ukrainian Energy Machines" (RTM, 1977) and the finite element method (FEM).

As a result of the strength calculation, tensile stresses from the centrifugal force (Fig. 1), bending stresses from steam thrust (Fig. 2), and total tensile stresses along the blade height (Fig. 3) are determined.

The FEM strength calculation provides for the correct determination of the boundary conditions, the mesh of the finite elements of the turbine profile part, and cyclosymmetry in the circumferential direction. The finite element mesh and the results of calculating the blade under static loading are presented in the form of a distribution of equivalent stresses – Fig. 4.

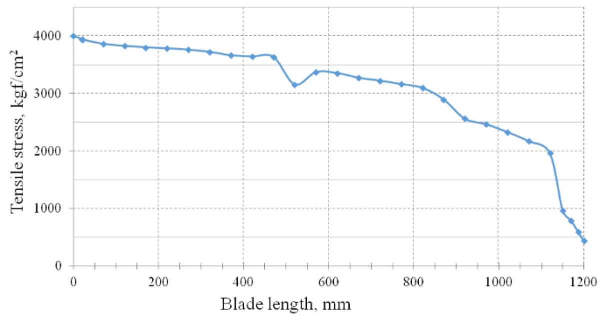


Fig. 1 Tensile stress by centrifugal forces

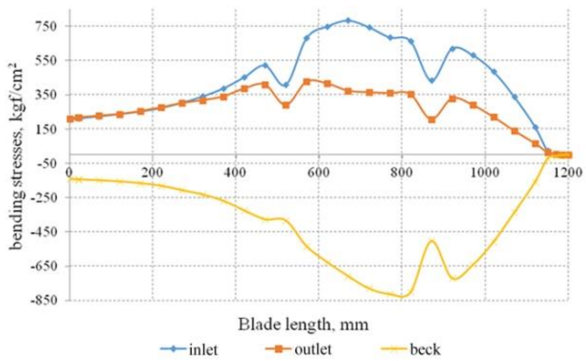


Fig. 2 Bending stress by steam thrust

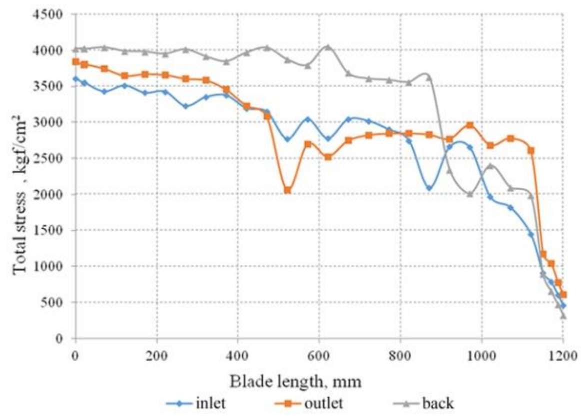


Fig. 3 Total stresses

The maximum stresses of 403 MPa occur at the back of the working blade closer to the root part, which indicates a correctly selected blade installation in the radial direction.

The calculation results show that each tooth of the tail connection is loaded unevenly. Equivalent stresses for the third tooth are 242 MPa, for the second tooth they are 268 MPa, and for the first tooth they are 365 MPa. The bending and bearing stresses for the second and third teeth are close to each other and do not exceed the permissible values. The calculation of the adopted design of the tail connection showed a symmetrical distribution of the load on the left and right teeth, and low stress concentrations in the radius transitions.

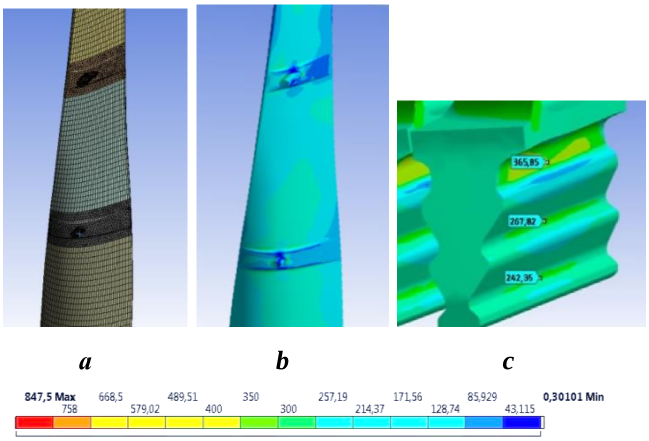


Fig. 4 Stress state of the blade: a – finite element mesh; b – profile part; c – blade root section

Geometric dimensions and structural elements of the profile part and tail connection of the calculated working blade of 1200 mm satisfy the strength conditions.

According to RTM 108.021.106-77 "Calculation of the static strength of steam turbine blades" (RTM, 1977), the safety margin for the total stresses in the blades are $n_t = \sigma_{0,2}/\sigma_{tot} \geq 1.5$, where $\sigma_{0,2}$ is the yield stress of the blade material at operating temperature; σ_{tot} is the maximum calculated total centrifugal stress and steam load.

In accordance with the results obtained, the safety margin for total stresses is $n_t = 2.14$ (with an acceptable $n_t \geq 1.5$), which meets the requirements of RTM 108.021.106-77.

Irregularity of the steam flow around the circumference of the flow path, rotor vibration and other reasons result in the emergence of disturbing forces. The vibration of the blades caused by these forces leads to their failures (Vorobiov et al. 2019).

Vibration calculations for a single blade of the adopted design were performed according to the method of JSC "Ukrainian Energy Machines" and FEM.

In the total displacements of the blade, along with displacements in the tangential direction, there is a large axial component. However, a comparison of this mode of vibration with subsequent ones allows speaking about the predominant nature of tangential vibrations and classifying the frequency of 27.198 Hz as the first form of tangential vibrations (Fig. 5*a*). Fig. 5 shows total displacements during oscillations with a frequency:

a) 27.198 Hz – *1* is eigenfrequency, *1* is the mode of tangential vibrations;

b) 54.418 Hz – *2* is eigenfrequency, *1* is the mode of tangential vibrations;

c) 133.32 Hz – *3* is eigenfrequency, *2* is the mode of tangential vibrations;

d) 185.46 Hz – *4* is eigenfrequency, *1* is the mode of tangential vibrations.

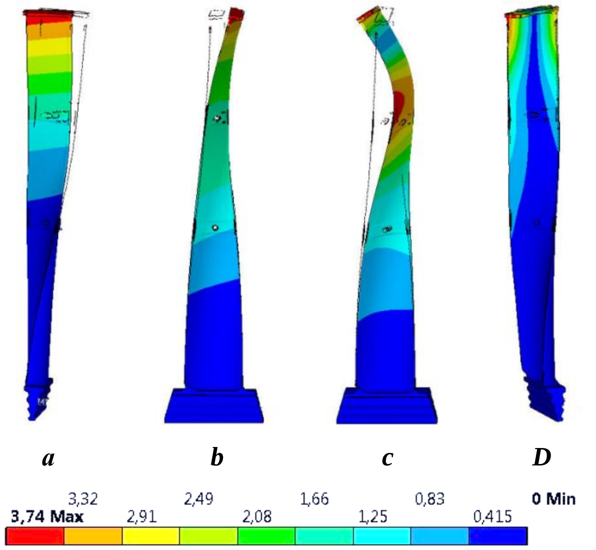
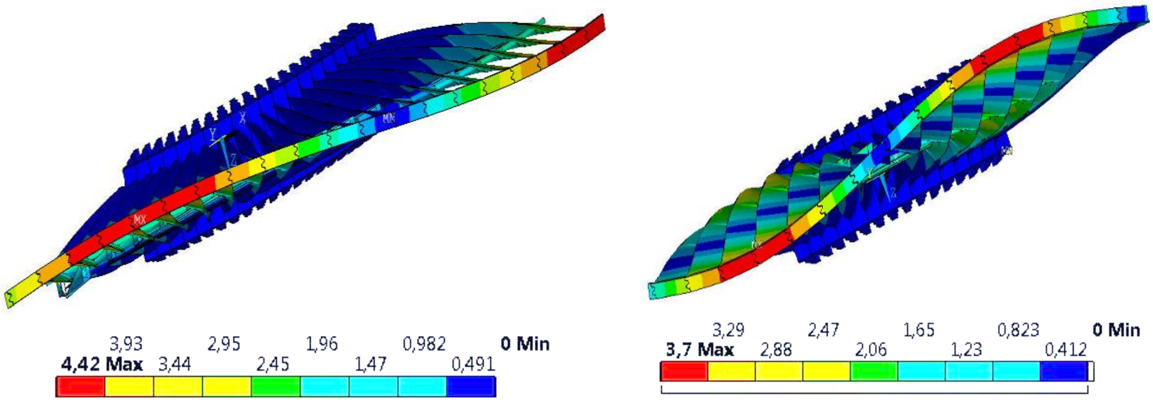


Fig. 5 Transfer at different values of eigenfrequency

The first mode of critical vibrations of the bladed disk is located between the beams of the 2nd and 3rd multiplicities to the revolutions and must be detuned from resonances with the corresponding multiplicities. The reserves between the operating and critical velocities are: 68 % (with the required 10 %) for the 2nd multiplicity and 27 % (with the necessary 7 %) for the 3rd multiplicity. The type of vibrations of the first mode at a critical velocity of 36.55 s^{-1} with 3 nodal diameters is shown in Fig. 6a.

The second mode of critical vibrations of the bladed disk is located between the beams of the 5th and 6th multiplicities to revolutions and must be detuned from resonances with the corresponding multiplicities. The reserves between the operating and critical velocities are: 24 % (with the required 5 %) for the 5th multiplicity, 7.5 % (with the required 4 %) for the 6th multiplicity. The type of vibration of the second mode at a critical rotational speed of 46.25 s^{-1} with 6 nodal diameters is shown in Fig. 6b.



a *b*

Fig. 6 Vibrations of the LPC stage rotor: a – 1-st vibration form; b – 2-nd vibration form

According to the results of the performed vibration calculations of blading, the blade L = 1200 mm has satisfactory vibration characteristics for the shroud. In the tunable range (up to 310 Hz), there are two modes of critical vibrations. The first mode of critical vibrations of the bladed disk is located between the beams of the 2nd and 3rd multiplicities to the revolutions, the second mode of critical vibrations of the bladed disc is located between the beams of the 5th and 6th multiplicities to the revolutions. Both modes are tuned from resonances with the corresponding multiplicities in accordance with the requirements of RTM 108.021.03-77 "Standards for vibration tuning of steam turbine blades" (RTM, 1977).

In accordance with the performed calculations of the working blade made of titanium alloy Ti6Al4V with a length of 1200 mm, a complete set of design documentation has been developed.

5 CONCLUSIONS

The low-cost modernization of the longterm operated high power steam turbines have required effective combination of the modern developments with existing elements of the power installations as well as formulating key aspects of the scientific and technological methodologies. These key aspects are in the increasing degree of reactivity in the blade root zone, the "reverse twisting" of the guide blades, the gas-dynamic optimization of the stage at a partial load and using other material for the blades.

The main results of design calculations for a titanium blade of a low-pressure cylinder with an active part length of 1200 mm, which is produced by JSC “Ukrainian Energy Machines”, are presented. The focus was made on the stress and vibration calculations. The correspondence of the indicated blades to the technical requirements for such objects is shown.

The working blades made of titanium alloy Ti6Al4V produced by JSC “Ukrainian Energy Machines” are successfully operated on turbines of the K-1000-60/3000 type in the South-Ukraine and Khmelnitsky nuclear power plants.

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