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DECREASE IN THE STRESSES ON THE TAIL JOINT OF A STEAM TURBINE BLADE BY CHANGING THE SHAPE OF THE SUPPORT PADS

Introduction. *The herringbone tail joints of long working blades in steam turbines, which function as multi-support structures, operate under complex stress conditions characterized by uneven distributions of both general and local stresses. Improving the geometry of the tail joint enhances the safety margin and service life of this critical component.*

Problem Statement. *At present, the uniformity of stress distribution across the support teeth of the tail joint is compromised by the required technological gap of 0.02 mm. This gap prevents full contact along part of the teeth, producing an uneven load distribution across the remaining teeth.*

Purpose. *The aim of the research is to develop an original design in which a specially shaped supporting surface of the teeth compensates for technological gaps and increases the uniformity of stress distribution in multi-support herringbone-type tail joints.*

Materials and Methods. *Finite element methods for modeling power-generation equipment have been applied. The calculation algorithm used here has served as the basis for a methodology for designing high-stress, multi-support mounts in large steam turbines.*

Results. *An original design featuring a specially contoured supporting surface of the teeth has been proposed. This design has increased the uniformity of stress distribution in multi-support tail joints of working blades in high-power steam turbines. It has significantly reduced the load on the upper teeth, redistributing these loads toward the lower teeth, which have previously been insufficiently engaged.*

Conclusions. *This research has proposed a novel approach to modifying stress conditions in herringbone tail joints of powerful steam turbines through the use of specially shaped (cylindrical) bearing surfaces that compensate for technological gaps. The resulting design has improved load uniformity across the teeth and has enhanced the strength and service-life characteristics of the entire assembly.*

Keywords: steam turbine, tail joint, stress-strain state, process gap.

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The herringbone tail joints of steam turbines, which are multi-support structures, operate under complex stress conditions, which are characterized by uneven distribution of general and local stresses [1].

Ensuring reliable operation of such tail joints is largely determined by the uniformity of load distribution across the teeth, which is disrupted by inevitable technological gaps within the tolerance field (up to 0.02 mm), causing a lack of contact on some teeth, and an uncalculated increase in load on others that have come into contact.

Recommendations for specifying tooth steps cause significant difficulties, for example, performing tooth steps with an accuracy of 0.005 mm.

The study [2] has shown the inappropriateness of searching for solutions aimed at ensuring uniform distribution of load on the teeth using differentiated gaps due to their extreme smallness. Thus, on the first pair of teeth it is necessary to provide a gap of 0.01 mm, on the second — 0.006 mm, which is unattainable with existing processing technologies. The assessment that is used to in the present date is the assessment of the stress state when selecting the dimensions of the tail joint, carried out under the condition of contact of all mating support surfaces. In this case, the assumptions are made that the gaps within the technological tolerances (up to 0.02 mm) [3] during turbine operation are selected due to the undermining of the support surfaces in contact, due to their plastic deformation [4]. The picture is aggravated by the fact that the initial technological gaps within the tolerance field cannot decrease during operation only due to micro-roughness of the support surfaces.

The choice of the contact surface size during design [4] was made by based on the average bearing stresses, although for a number of reasons — the absence of contact in certain areas of the supporting surface of the teeth due to a violation of the plane or the absence of contact over the entire surface of some teeth (in the presence of gaps), a local increase in bearing stresses often occurs.

According to some estimates, a stress state close to all-round compression is realized near the contact area [5], and the crushing stresses can reach the yield point of the material. The zones of plastic deformation of the material are small, they practically do not increase the compliance of the teeth and do not affect the distribution of the load between them.

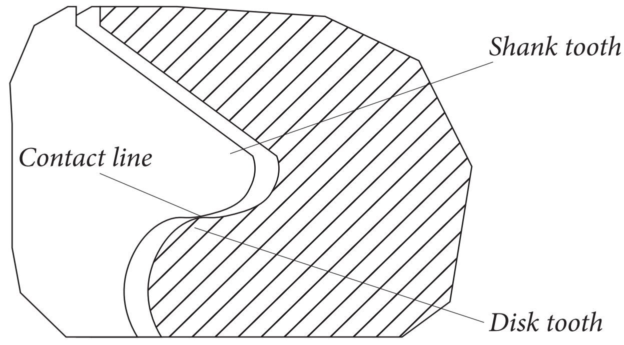

Fig. 1. Tail connection contact unit

Due to the special shape of the supporting surface of the teeth, this ensures compensation of technological gaps, to increase the uniformity of the distribution of stresses in the herringbone tail joints. The proposed design, shown in Fig. 1, is original and is protected by a patent for a utility model [6].

The proposed shape of the support surfaces of the contacting teeth (using the example of a 1030 mm blade of the last stage of the turbine K-325-23.5 MW) (Fig. 2) allows us to consider their interaction as a problem of elasticity theory during contact of two cylinders with parallel axes, the interaction diagram of which is presented in (Fig. 3) (Hertz problem [7]).

Let us estimate the magnitude of the convergence of the cylinders Δ.

Under the action of a force equivalent to the reactive force on one support surface, based on the total centrifugal force acting on the tail joint, we have [7]:

$$\Delta = 2p\frac{\theta}{\pi}\left(0.815 + \ln\frac{4R_1R_2}{a^2}\right), \qquad (1)$$

where $p = 707 \cdot 10^3$ N/m is the force per linear meter of tooth; coefficient of the ratio of longitudinal and transverse deformation of cylinders:

$$\theta = \frac{1-\nu^2}{E},$$

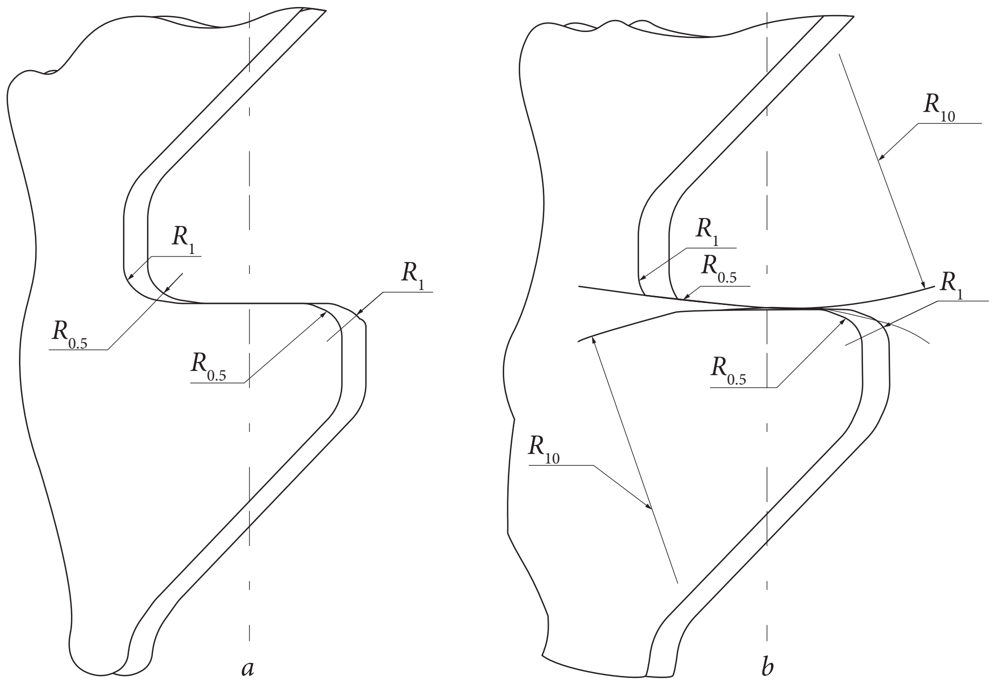

Fig. 2. Formation of contact surfaces of teeth: *a* — existing design; *b* — proposed design

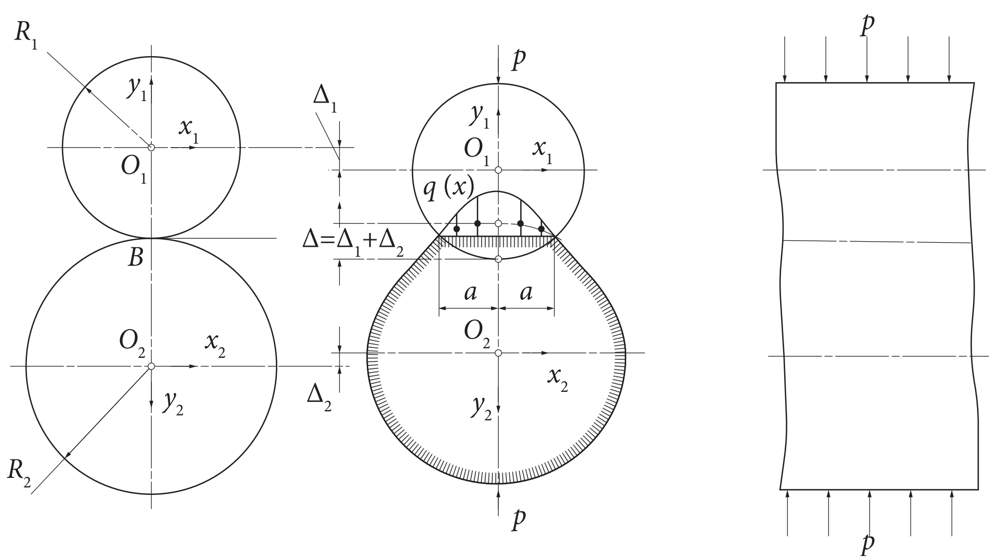

Fig. 3. To the calculation of contacting cylinders according to the Hertz problem

ν is Poisson's ratio; E is the modulus of elasticity of the first kind; $R = R_1 = R_2 = 0.01$ m — radius forming the support surfaces of the teeth of the tail joint of a 1030 mm blade (Fig. 2).

Half the width of the contact area formed under the load: $\alpha = 0.798\sqrt{p\dfrac{2R^2}{R}\theta} = \sqrt{2\rho R\theta} =$ $= 0.2 \cdot 10^{-3}$ (Fig. 3).

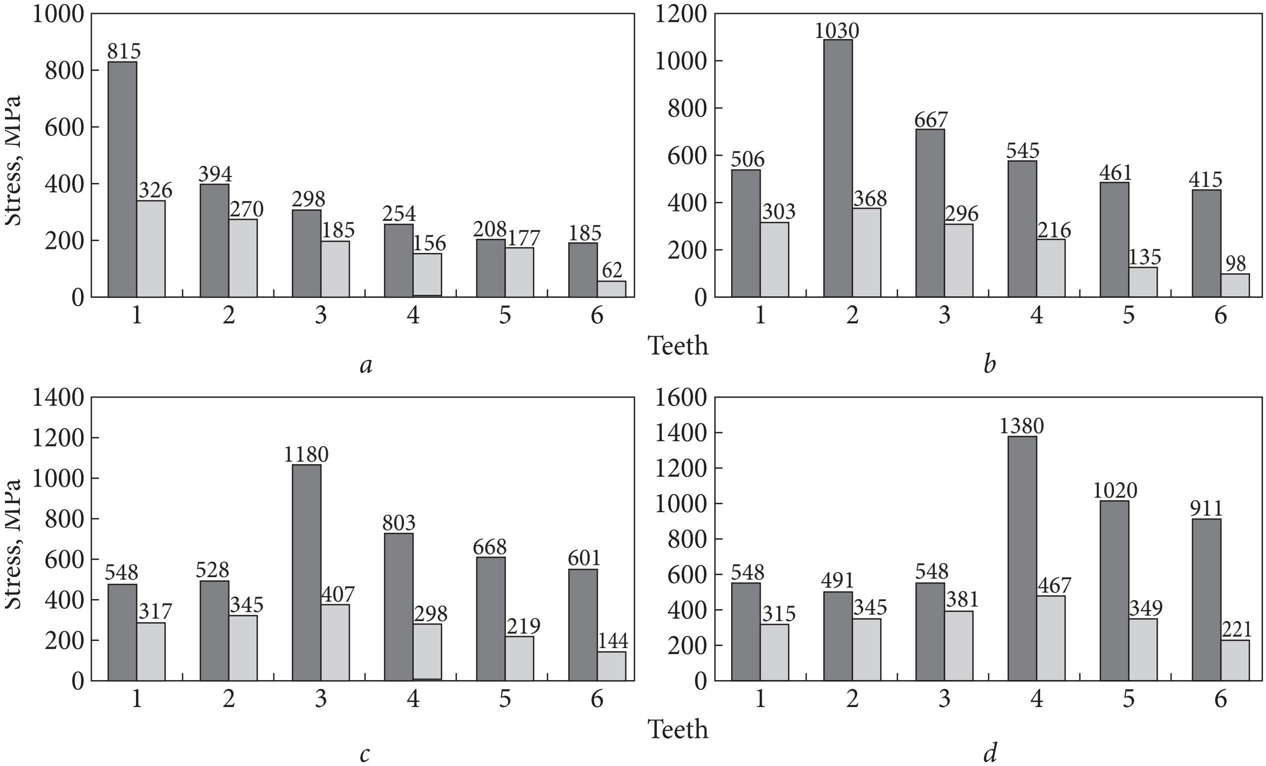

Fig. 4. Distribution of stresses across the calculated sections: *a* — option No. 1; *b* — option No. 2; *c* — option No. 3; *d* — option No. 4

As a result of the calculation, we have that even under the condition of contact of all teeth of the tail joint, the proposed form of the support surfaces will lead to the movement of the tail in the radial direction under the action of centrifugal forces according to (1) by a value of about 0.02 mm. At the same time, taking into account that in reality only a part of the teeth are in contact, the deformations can exceed the value of 0.02 mm, since the load on the contacting teeth is higher than when all the teeth are in contact.

One of the main features of contact problems is the presence of significant stresses in the contact zone, especially in cases where the area of the initial contact is zero (contact at a point or along a line). Contact zones are characterized by the occurrence of all-round compressive stresses, which allows the material to withstand high surface stresses without destruction [7—8]. Let us determine the value of the maximum stress (pressure) in the contact zone for the example under consideration:

$$q_{\max} = 0.42\sqrt{pE\frac{R_1 + R_2}{R_1 R_2}} =$$

$$= 0.42\sqrt{\frac{2\rho E}{R}} \approx 2.3\cdot 10^4 \text{ kg/cm}^2.$$

In this case, the average stress in the contact zone on the area bounded by lines equidistant from the initial contact line will be:

$$q_{cp} = \frac{p}{2a} \approx 17.6\cdot 10^3 \text{ kg/cm}^2.$$

The assessment of the permissibility of the value of contact stresses is based on the assumption that the contacting cylindrical (spherical) surfaces do not move relative to each other and the load is applied statically, increases gradually from zero

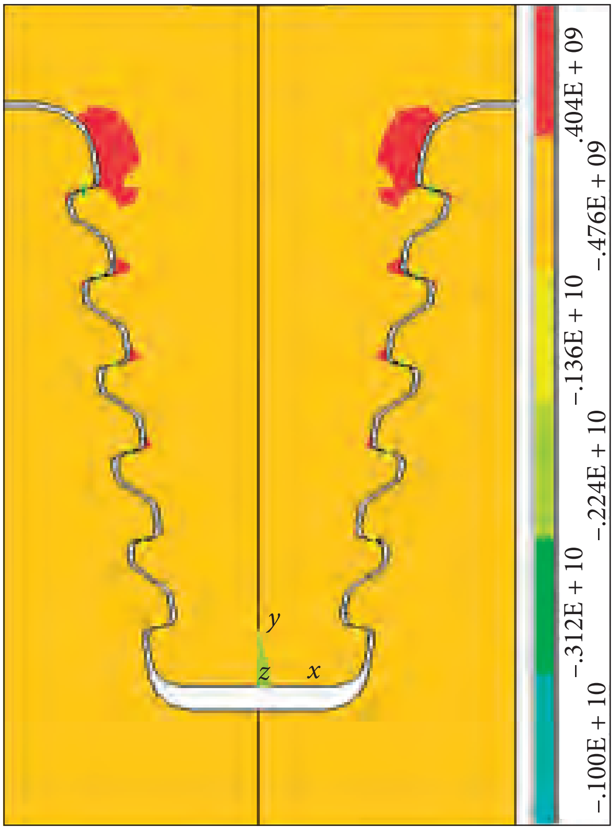

Fig. 5. Distribution of stresses in the tail joint when all cylindrical teeth are in contact

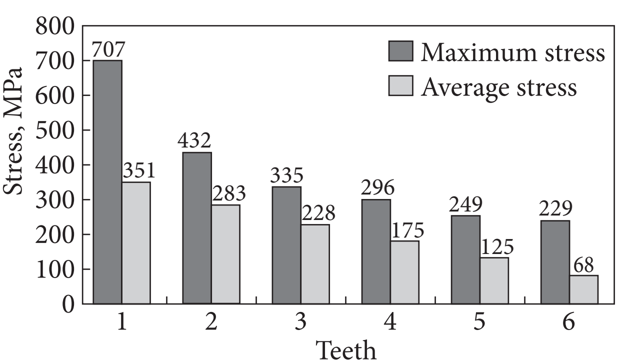

Fig. 6. Distribution of stresses over the calculated sections of the tailpiece during contact over all support areas cylindrical shape

to the final value. In particular, in gear transmissions and ball bearings, despite significant differences in the conditions of loading the working surfaces from those indicated above, contact stresses of up to 30.000 kg/cm^2 and more are permitted. The selection of the specified permissible values of stresses on the working surfaces has been verified by the practice of operating gear transmissions [8].

Considering, that in our case the load on the tail joint increases from zero to the working level gradually and without impacts, the obtained values of $q_{\max}$ and $q_{average}$ should be assessed as satisfying the conditions of operability.

NUMERICAL STUDIES

The conducted assessment of the deformability of a single support surface of the proposed cylindrical shape allows us to expect a positive effect in a multi-support joint.

The stress-strain state (SSS) of the herringbone tail joint is considered for the following support contact options:

No. 1 — contact of all pairs of teeth;

No. 2 — gap on the first pair, counting from the root section of the blade;

No. 3 — gaps on the first and second pairs of teeth;

No. 4 — gaps on the first, second and third pairs.

The calculations of these variants are performed under the condition that the contact areas of the tooth have a flat surface and deviations in the step sizes within the manufacturing tolerances cause the presence of gaps, which cannot be eliminated when the shank is loaded.

The nature of the stress-strain state of the shanks of the above-listed variants indicates that the zone of the upper teeth is the most loaded, regardless of the presence or absence of contact on it. At the same time, the average stresses in the section remain significant, and the maximum, in the absence of contact on the first, second and third pairs of teeth, in the cross sections fall and the maximum shifts to the section corresponding to the pair of teeth that first enters into contact [9].

This is explained by the fact that in the absence of a load on the teeth, the bending moment disappears, causing an increase in maximum stresses, the level of which, in the presence of contact, also increases significantly due to the presence of concentrators at the base of the tooth. The concentration of stresses in the transition from the tooth to the body of the shank is weakly displayed in the absence of contact between the teeth [10, 11].

The decoding of the stress state pattern of the tail joint with contact of all pairs of teeth with supporting surfaces in the form of planes obtained by calculation is presented in Fig. 4 in the form of graphs of average and maximum stresses in six calculated sections. Considering that the dependencies shown in Fig. 4, *a* were obtained assuming contact on all supports, this variant should be considered purely theoretical.

The diagrams of average and maximum stresses shown in Fig. 4 reflect several of the possible combinations of contacts of the tail joint teeth. In the absence of contact on individual supports or on several of them in total, the loads on the teeth are redistributed with a significant increase in stresses in the cross sections of the tail joint.

It is necessary to emphasize an important feature associated with the absence of contact on one, together on two or three pairs of teeth, counting from the root section of the blade. In this case, the length of the blade to the seal section actually increases, which leads to a change in the natural frequency of oscillations of the blade.

According to the proposed solution for making the supporting surfaces of the tail joint cylindrical, Fig. 5 shows the results of the calculation, obtained by using the finite element method, of the stress-strain state of the tail joint, in which the contact areas are reduced to a line along the teeth.

A more detailed comparison of the stress diagrams of the above-mentioned variants in Fig. 4 and 6 allows us to conclude that even with practically unachievable contact of all teeth with flat support pads, the variant with cylindrical supports wins, since in this variant the difference between the maximum and average stresses in the most loaded first section of the root is reduced from 2.5 to 2 times due to the favorable redistribution of the load between the supports, and the uncertainty of the actual length of the blade, which affects the vibration characteristics of the blade, is eliminated [10—12].

CONCLUSIONS

1. The labor proposes an original approach to solving the problem of weakening the stress-strain state of the herringbone-type tail joints of powerful steam turbines by using a special (cylindrical) shape of the tooth support surfaces, which ensures compensation for technological gaps.

2. Using the example of the last-stage working blade of the K-325-23.5 turbine, estimates of the radial displacements of the proposed design rootstock, as well as stresses in the contact zone, are made based on the solution of the elasticity theory problem for contact between two cylinders with parallel axes (the Hertz problem). The obtained values of the maximum and average contact stresses do not exceed the permissible ones, which allows them to be considered as satisfying the operability conditions.

3. For a comparative quantitative assessment of the stress values for pairs of rootstock teeth, numerical studies were performed using a software package based on finite element methods for rootstocks with flat support surfaces for different support contact options and the proposed design with cylindrical contact surfaces. Comparison of stress diagrams showed that the option with cylindrical supports wins in all the cases considered, since the ratio of maximum to average stress in the most loaded first section of the tail joint is reduced from 2.5 to 2 times due to a more favorable redistribution of the load between the supports.

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ЗНИЖЕННЯ НАПРУГИ НА ЗУБЦЯХ ХВОСТОВОГО З'ЄДНАННЯ
ЛОПАТКИ ПАРОВОЇ ТУРБІНИ ЧЕРЕЗ ЗМІНУ ФОРМИ ОПОРНИХ ПЛОЩИН

Вступ. Хвостові з'єднання ялинкового типу довгих лопаток парових турбін, які є багатоопорними конструкціями, працюють у складних напружених умовах, що характеризуються нерівномірним розподілом загальних і місцевих напружень. Вирішення проблеми покращення геометрії хвостового з'єднання довгих робочих лопаток дозволяє збільшити запас міцності та термін служби цього відповідального елемента.

Проблематика. Наразі рівномірність розподілу напружень по опорних зубцях хвостового з'єднання порушується вимушеним виконанням технологічного зазору у 0,02 мм, що ліквідує контакт по частині зубців, утворюючи нерівномірний розподіл навантаження в цьому вузлі.

Мета. Розробка оригінальної конструкції, де за рахунок спеціальної форми опорної поверхні зубців, що забезпечує компенсацію технологічних зазорів, можна підвищити рівномірність розподілу напруг у багатоопорних хвостових з'єднаннях ялинкового типу.

Матеріали й методи. Використано методи моделювання енергетичного обладнання з використанням методів кінцевих елементів. Розглянутий алгоритм розрахунку став базою для методики конструювання високонапружених багатоопорних кріплень у потужних парових турбінах.

Результати. Розроблено оригінальну конструкцію, де за рахунок спеціальної форми опорної поверхні зубців, що забезпечує компенсацію технологічних зазорів, збільшується рівномірність розподілу напруг у багатоопорних хвостових з'єднаннях робочих лопаток потужних парових турбін. Значно зменшено навантаження на верхніх зубцях, яке перенесено на нижні зубці, що раніше не були достатньо задіяні.

Висновки. Запропоновано оригінальний підхід до вирішення задачі зменшення напруги у хвостових з'єднаннях ялинкового типу потужних парових турбін за рахунок використання спеціальної (циліндричної) форми опорних поверхонь зубців, що забезпечує компенсацію технологічних зазорів. Така конструкція сприяє більш рівномірному розподілу навантажень по зубцях, збільшуючи ресурс та характеристики міцностні всієї конструкції.

Ключові слова: парова турбіна, хвостове з'єднання, напружено-деформований стан, технологічний зазор.