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CALCULATION OF THE T-SHAPED SHANK OF THE STEAM TURBINE ROTOR BLADE

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The broad possibilities of the developed approach [8, 9, 11, 12] are illustrated by the solution of a new practically important task related to specific design developments.

The most important load-bearing elements of the rotors of steam turbines are the blade shanks. This paper presents the calculation of the T-shaped shank, which is affected by the forces caused by the rotation of the rotor. They consist of the surface load blade distributed over the area of the root section of the blade and mass forces distributed over the volume of the shank. but along some of its central parts. This leads to an uneven distribution of contact forces along the axis along the shank. The results of the calculation allow us to draw the following conclusions that in the case of a uniform load, stress determination can be carried out within the framework of a flat setting, since this leads to a relatively small (five, eight percent) error compared to the spatial setting. Taking into account the uneven nature of the load distribution along the length of the shelf allows you to significantly clarify the level of maximum stresses in comparison with a flat problem. Thus, the value of the intensity of tangential stresses increased by more than thirty percent and exceeded the yield strength of the material. The results of the calculation of the shank beyond the elastic properties of the material are presented in this work and reflect the development of the zone of plastic deformations in the plane $Z^{3'=0}$. By studying the influence of the geometric parameters of the shank, it was found that a decrease in the level of plastic deformations can be achieved by simultaneously increasing the radius the gap and the width of the shelf.

Keywords: finite element method (ITU), semi-analytical finite element method (NMSE), stress-strain state (SSS), elastic deformation, rotary device fastening part, cylindrical body, sampling, elastic-plastic setting.

Entry. Blade shanks are one of the most important load-bearing elements of steam turbine rotors. The level of stresses arising in the shank significantly depends on the durability of the blade and the operating modes of the turbine as a whole. Therefore, when studying such objects, special attention is required to the selection and justification of calculation schemes and problem statement [1-5, 7, 10]. At present, their calculation, as a rule. It is performed as part of a flat setting. However, in real conditions, the parameters of the stress-strain state of the shank under the influence of a number of factors can change in all three coordinates. These include the uneven nature of the distribution of loads over the surface of the industrial part and the conditions for the interaction of the shank and rim of the disc.

To find out the effect on the stress-strain state of these factors, consider one of the design options for the T-shaped shank, shown in Fig. 1. The shank is made of steel, the mechanical characteristics of which in the form of dependencies of the intensity of tangent stresses T on the intensity of deformations ε_i^p obtained at a temperature of 300°C, are given in Table 1. Modulus of elasticity of the material $E = 2,06 \cdot 10^5\,\text{MPa}$, Poisson's coefficient ν=0,3. The forces acting on the shank are due to the rotation of the rotor with angular velocity $\omega = 314,16\text{s}^{-1}(3000\,\text{rpm})$. They consist of surface load with intensity distributed over the area of the root section of the blade (shaded in Fig. 1) $q = 132,8\,\text{MPa}$

and mass forces distributed over the volume of the shank, the equilibrium of which C for a certain volume υ can be determined by the formula:

$$C = \rho \upsilon \omega^2 (R_0 + Z^{1'}), \quad (1)$$

where: density of the material $\rho = 7{,}85\ \mathrm{kN \cdot s^2 / m^4}$, distance from the plane $Z^{1'} = 0$ to the axis of rotation of the rotor $R_0 = 812\,\mathrm{mm}$.

Table 1

ε_i^p, %	0,00	0,68	1,70	2,67	3,66
T, MPa	265,6	326,2	360,8	386,8	401,2

When calculating structures using the finite element method, the correct choice of an approximation grid is of great importance, which allows you to obtain stable results with a minimum number of unknowns. This is achieved mainly by the use of non-uniform grids, the thickening of which is carried out in the zones of probable stress concentrators. Fig. 2,a shows the breakdown of the shank into finite elements, used in the study of the influence of the nature of the load distribution on the surface of the industrial Part. Particular attention is paid to rounding zones, since it is in these places that maximum stresses should be expected. Two calculation options have been performed. In the first case, the surface load is applied strictly along the area of the root section of the scapula (Fig. 1), in the second, it is evenly distributed along the axis $Z^{3'}$ (the area of application is shown by hatching in Fig. 2,a). The shank is fixed against displacements $u_{1'}$ on the surface of the contact pad *ABCD*.

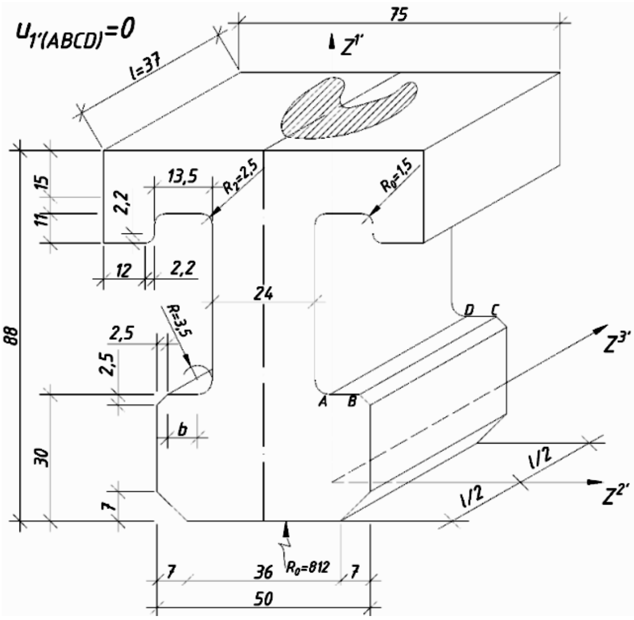
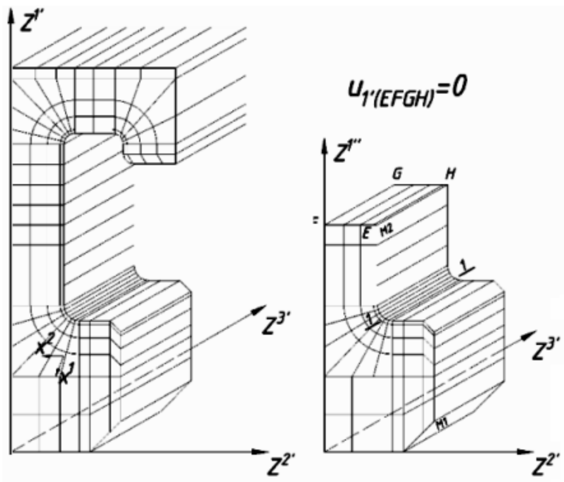


Fig. 1. General view of the shoulder blade

Table 2 shows the values of maximum tensile stresses $\tilde{\sigma}^{22}$ and the intensity of tangent stresses T recorded in the area of the galvanized transition between the shelf and the neck. Their comparison shows that taking into account the unevenness of the application of surface load does not have a noticeable effect on the level of maximum stresses. Moreover, in both variants of the calculation in the central part of the neck, the picture of the stress-strain state turned out to be close to homogeneous. The feature allows you to significantly simplify the design scheme of the shank and



(a) (b)

Fig. 2. Scheme of applying the load to the surface of the scapula

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include only the lower part of the neck and the shelf in it. Fixing against displacements $u_{1'}$ can be carried out along the cross-sectional area of the neck *EFGH*, and the load can be applied on the surface of the contact pad, indicated by hatching in Fig. 2, b. According to Table 2, the difference in the results calculated in accordance with this scheme is less than 1% in relation to those calculated earlier. The proposed calculation scheme, in addition to a significant reduction in the number of unknowns, provides an opportunity to study the effect on the value of maximum stress of nature Distribution of forces over the surface *ABCD*.

Table 2

Calculation scheme	$\tilde{\sigma}^{22}$, MPa	%	*T*, MPa	%
Fig. 1	683,5	-	333,6	-
Fig. 2 a	689,1	0,82	337,1	1,05
Fig. 2, b	679,5	0,58	331,4	0,66

According to the manufacturers of steam turbines, in accordance with the technological features of the manufacture and assembly of rotors, the interaction of the rim and disc can be carried out not along the entire length of the shelf *l*, but along some of its central part in size $a \le l$. This leads to an uneven distribution of contact forces along the axis $Z^{3'}$, the diagram of which for the maximum permissible case ($a = 0{,}84l$) is shown in Fig. 3.

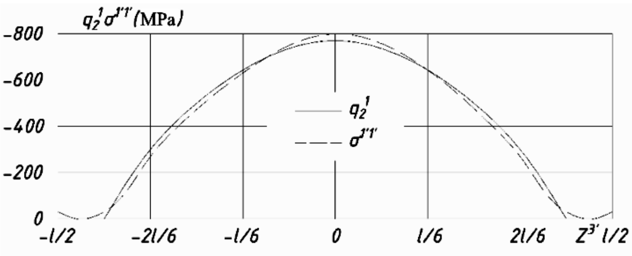

Fig. 3. Diagram of normal stresses $\sigma^{1'1'}$ built along the length of the shelf

Table 3 presents the maximum values $\tilde{\sigma}^{22}$ and T, calculated with different accuracy of the solution of the systems of equations determined ε by, the number of retained terms of the series M and the number of nodes of the grid region M1 and M2 along the axes x^1 and x^2 respectively. Based on these data, it can be concluded that stable results are achieved at $\varepsilon = 10^{-3}$, M=5, M1=9, M2=18.

Table 3

Calculation scheme	ε	M	M1	M2	$\tilde{\sigma}$, MPa	%	*T*, MPa	%
Fig. 2, b	10^{-3}	5	7	15	840,1	0,43	418,5	0,19
	10^{-4}				842,0	0,21	418,9	0,09
	10^{-5}				843,8	-	419,3	-
	10^{-3}	3	7	15	843,0	1,22	413,1	2,27
		5			840,1	0,87	418,5	0,99
		8			832,8	-	422,7	-
	10^{-3}	5	7	15	840,1	2,81	418,5	2,70
			9	18	820,7	0,44	409,5	0,49
			12	24	817,1	-	407,5	-

Thus, a significant increase in the number of unknowns (more than 1.5 times) due to an increase in the number of retained members of the series or the number of nodes of the grid region leads to a slight (less than 1%) change in the results. As shown by the analysis of the values of normal and tangent stresses on the stress-free surface of the shank, their value does not exceed 2% compared to the maximum values $\tilde{\sigma}^{22}$. The plot of normal stresses $\sigma^{1'1'}$, built along the length of the shelf and shown in Fig. 3 by a dotted line, is satisfactorily consistent with the plot of loads. The equilibrium of normal stresses, calculated in the plane *EFGH*, differs by less than 1% from the equilibrium of surface and volumetric forces.

The results of the calculation of the blade shank in two-dimensional and spatial formulation are shown in Fig. 4 in the form of diagrams $\tilde{\sigma}^{22}$ and T, constructed in the section I-I. A dotted line indicates the results of solving a flat problem, a solid line - a three-dimensional load under the action of a load unevenly distributed along the length of the shelf. Their comparison allows us to conclude that in the case of a uniform load, the determination of stresses can be carried out within the framework of a flat problem, since this leads to a relatively small (5-8%) error compared to the spatial setting. Taking into account the uneven nature of the distribution of forces along the length of the shelf allows you to significantly clarify the level of maximum stresses in comparison with a flat problem. Thus, the value of the intensity of tangential stresses T increased by more than 30% and crossed the yield strength of the material. The limit value of the angular velocity, corresponding to the elastic behavior of the material, is 2414 rpm, which is significantly lower than the nominal value.

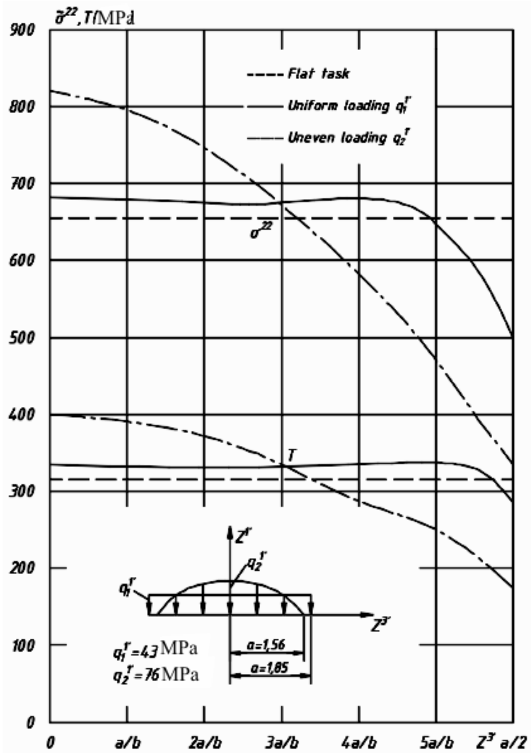

Fig. 4. Results of calculation of the shank of the blade in two-dimensional and spatial formulation

Modeling of the change in the pattern of the stress-strain state of the shank, due to the elastic-plastic behavior of the material in the process of loading, was carried out by solving a sequence of nonlinear problems with a gradual increase in the rotor rotation speed. Fig. 5 and 6 show plots $\tilde{\sigma}^{22}$ and ε_i^p, respectively, plotted in the section I-I for different values of angular velocity. $Z^{3'}$ As the plastic deformations increase, which ω= 3000 rpm are distributed over the entire length of the shelf. Fig. 7 shows a picture of the change in the area of plastic deformations in the plane $Z^{3'}=0$. Its characteristic feature is the tendency to develop plastic deformations in the depth of the shelf along the line, which forms an angle of the order of 45◦ with the surface of the shelf, which is fully consistent with the ideas about the operation of the T-shaped shank, set out in the work [6].

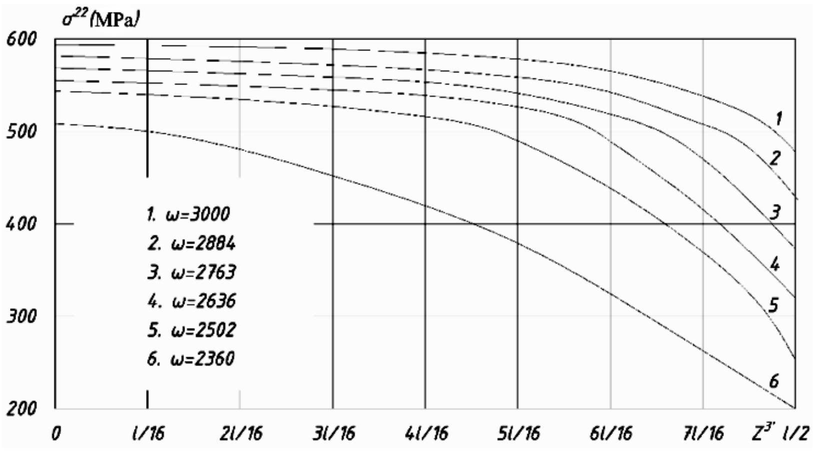

Fig. 5. Plots $\tilde{\sigma}^{22}$, respectively, are plotted in the section I-I for different values of angular velocity

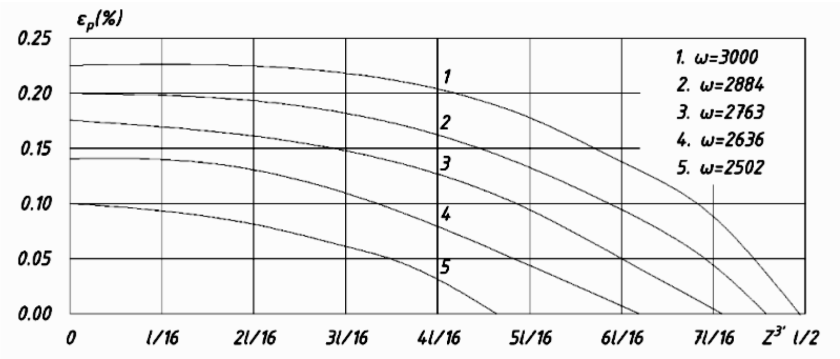

Fig. 6. Plots ε_i^p , respectively, are plotted in the section I-I for different values of angular velocity

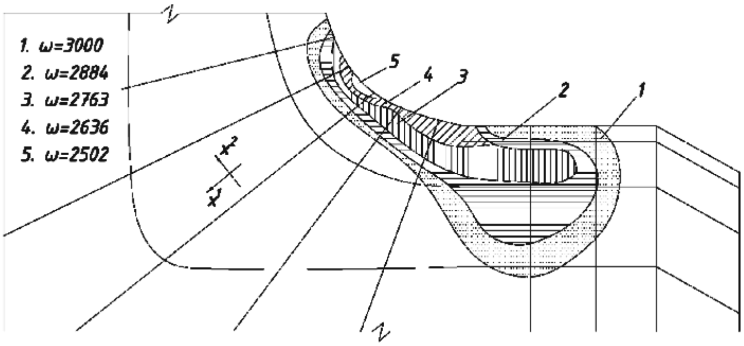

Fig. 7. Dependence of the width of the shelf to the level of plastic deformations.

Despite the relatively small amount of plastic deformations, their presence in such a crucial element as the shank of the scapula is quite undesirable. Reducing stresses and improving the operating conditions of the concentrator is achieved, as a rule, by increasing the radius of curvature *R*. However, it's increase leads to a decrease in the width of the shelf and a corresponding *b* increase in the intensity of contact forces. At the same time, the resources for increasing the overall dimensions of the lower part of the shank along the axis $Z^{2'}$ are quite limited by the dimensions of the disc rim. Therefore, for the new value of the rounding radius, *R*=4,5 mm two cross-section, the results of the calculations for both options are quite close and an increase *R* here leads to a decrease in stresses and deformations. However, in the central part of the area, which is located in the depths of the shelf and is shown in Fig. 7 with horizontal shading, a decrease in the width of the shelf causes a 2-fold increase in the level of plastic deformations.

In the first case, the total width of the lower part of the shank remained unchanged and the width of the shelf *b* decreased to 6 mm, in the second case, the width of the shelf remained unchanged (*b*=7 mm). $\tilde{\sigma}^{22}$ ε_i^p ω= 3000 rpm.

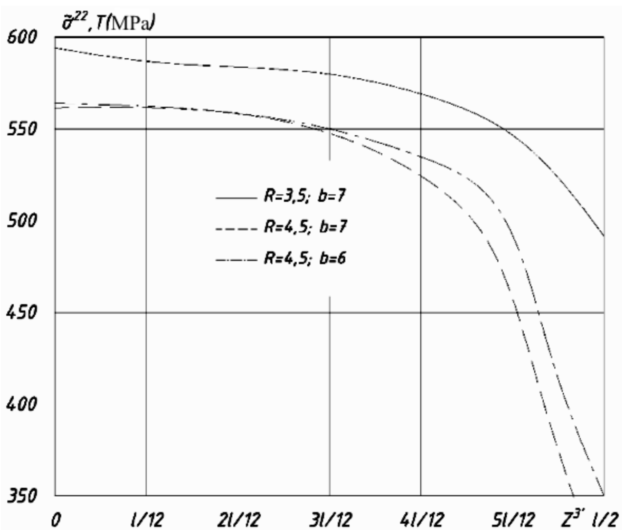

Fig. 8. Diagrams $\tilde{\sigma}^{22}$ in the section of I-I diagrams constructed at angular velocity

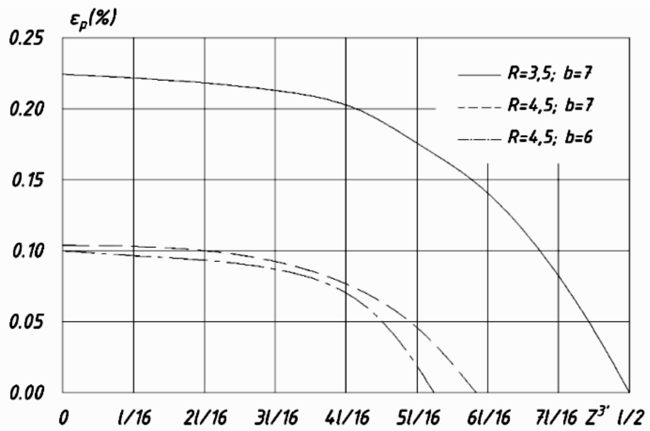

Fig. 9. Diagrams ε_i^p in the section of I-I diagrams constructed at angular velocity

Conclusion. Based on the above studies, it can be concluded that taking into account the conditions of interaction of the shank with the rim of the disc leads to a significant increase in the maximum values of stresses and straines, the reduction of which can be achieved by simultaneously increasing the radius of the galley junction and the width of the shelf.

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РОЗРАХУНОК Т-ПОДІБНОГО ХВОСТОВИКА ЛОПАТКИ РОТОРА ПАРОВОЇ ТУРБІНИ

Широкі можливості розробленого підходу [8, 9, 11, 12] ілюструються розв'язанням нової практично важливої задачі, пов'язаної із конкретними проектно-конструкторськими розробками.

До найбільш відповідальних несучих елементів роторів парових турбін відносяться хвостовики лопаток.В даній роботі представлений розрахунок Т-подібного хвостовика, на який діють сили, зумовлені обертанням ротора. Вони складаються з розподіленої за площею кореневого перерізу лопатки поверхневого навантаження та розподілених по об'єму хвостовика масових сил. Відповідно до технологічних особливостей виготовлення і збір ротора взаємодія обода і диска може здійснюватися не по всій довжині полки, а по деякій його центральній частині. Це призводить до нерівномірного розподілу контактних зусиль по осі вздовж хвостовика. Отримані результати розрахунку дозволяють зробити наступні висновки, що у випадку рівномірного навантаження визначення напружень можна проводити в рамках плоскої постановки, так як це призводить до порівняно невеликої (п'яти, восьми відсотків) похибки порівняно з просторовою постановкою. Врахування нерівномірного характеру розподілення навантаження по довжині полички дозволяє суттєво уточнити рівень максимальних напружень в порівнянні з плоскою задачею. Так, величина інтенсивності дотичних напружень збільшилась більш ніж на тридцять відсотків і переважила межу текучості матеріалу. Результати розрахунку хвостовика за межами пружніх властивостей матеріалу представлений в даній роботі і відображає розвиток зони пластичних деформацій в площині $Z^{3'}$=0. Незважаючи на порівняно невелику величину, їх наявність у такому відповідальному елементі, як хвостовик лопатки вельми небажана. Шляхом дослідження впливу геометричних параметрів хвостовика встановлено, що зниження рівня пластичних деформацій може бути досягнуто за рахунок одночасного збільшення радіусу галтельного переходу і ширини полиці.

Ключові слова: метод скінченних елементів, напіваналітичний метод скінченних елементів, напружено-деформований стан, пружне деформування, деталь кріплення поворотного пристрою, циліндричний корпус, дискретизація, пружно-пластична постановка.

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Широкі можливості розробленого підходу [8, 9, 11, 12] ілюструються розв'язанням нової практично важливої задачі, пов'язаної із конкретними проектно-конструкторськими розробками. До найбільш відповідальних несучих елементів роторів парових турбін відносяться хвостовики лопаток. В даній роботі представлений розрахунок Т-подібного хвостовика, на який діють сили, зумовлені обертанням ротора. Шляхом дослідження впливу геометричних параметрів хвостовика встановлено, що зниження рівня пластичних деформацій може бути досягнуто за рахунок одночасного збільшення радіусу галтельного переходу і ширини полиці.
Табл. 2. Іл. 6. Бібліогр. 16 назв.

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The broad possibilities of the developed approach [8, 9, 11, 12] are illustrated by the solution of a new practically important task related to specific design developments. The most important load-bearing elements of steam turbine rotors include blade shanks. This paper presents the calculation of a T-shaped shank, which is affected by the forces caused by the rotation of the rotor. By studying the influence of geometric parameters of the shank, it was found that a decrease in the level of plastic deformations can be achieved by simultaneously increasing the radius of the galley junction and the width of the shelf.
Table. 2. Fig. 6. Refs. 16.

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