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BLADE PROFILING FOR THE FINAL STAGES OF POWERFUL SHIP'S STEAM TURBINES

Problem statement. The search for technical solutions aimed at improving the reliability and efficiency of ship steam turbines during their design is an actual problem. One of the objectives of this problem is the improvement of the flow parts of ship turbines, which should be based on the optimization of all parameters in terms of aerodynamic and strength characteristics.

Analysis of research and publications. Research and development work of leading specialists in the field of aerodynamics of flow parts (and, in particular, the last stages of turbines), aimed at improving the reliability and efficiency of operation of steam turbines, are carried out using both calculated and experimental research methods.

The results of computational studies of two-dimensional flow in the channels of the nozzle and working gratings of steam turbines showed a significant effect of the shape of the blade profiles on the energy loss during flow past them [2, 5]. This is confirmed by the results of experimental studies [3], conducted in a wind tunnel and a steam-dynamic pipe.

Analysis of experience in designing and operating ship steam turbines [1,2,5,6] shows shortcomings in solving individual design problems, in particular, the lack of introducing special structural forms of the blades, which allow to reduce energy losses during flow.

Currently, a significant part of large-capacity gas carriers are equipped with steam-turbine units with a capacity of more than 30,000 kW, which include a low-pressure turbine (LPT), which largely determines the efficiency of the entire turbine [7]. A special place in the design of turbines occupy the last steps, which are the most loaded element of the turbine.

The purpose of this study is to develop recommendations for the design of the profiles of the working blades of the last steps of ship low-pressure steam turbines, taking into account the most optimal aerodynamic and strength characteristics.

Presentation of the main research material.

The absolute length of the working blades (from the root to the peripheral sections) of the last steps in modern steam turbine plant (30-50 MW) reaches 0.50-0.65 m. Such steps are called steps with relatively long blades or steps with high fanning $\theta = D_{cp}./\ell < 10$, where

D_{cp} - the average diameter of the blade, in [m];

ℓ - is the length of the blade in [m].

When $\theta<10$, it is already necessary to know and take into account the variability of the flow parameters along the radius and then the centers of gravity (c_g) of all cross sections along the blade height for the purpose of vibration reliability, it is necessary to locate on one straight axis of the blade coinciding with the radius of the turbine shaft.

The specific profile for each section of the scapula is designed separately: first, the root and peripheral sections, and then the middle section and intermediate sections. The required values of the areas of all cross-sections of the height of the blade are pre-determined on the basis of the allowable tensile stresses from centrifugal forces and bending stresses from vapor forces. Thus, for an already specified area (for example, the middle section of the blade), it is necessary to design a profile that corresponds to the lowest level of energy losses during flow around it with the input angles β_1 determined from the thermal calculation and the output β_2 of the flow when the Mach numbers are calculated for this section $M_{2т} = \frac{W_{2т}}{a}$ (where $W_{2т}$ - relative flow rate at the exit of the turbine stage, and a - the speed of sound). These conditions impose strict requirements on the magnitude of the moment of resistance of the section W and the section area F of the blade profile. To meet the aerodynamic and at the same time strength requirements can be quite difficult. Thus, Fig.1 shows an example of designing a relatively long working blade of a steam turbine.

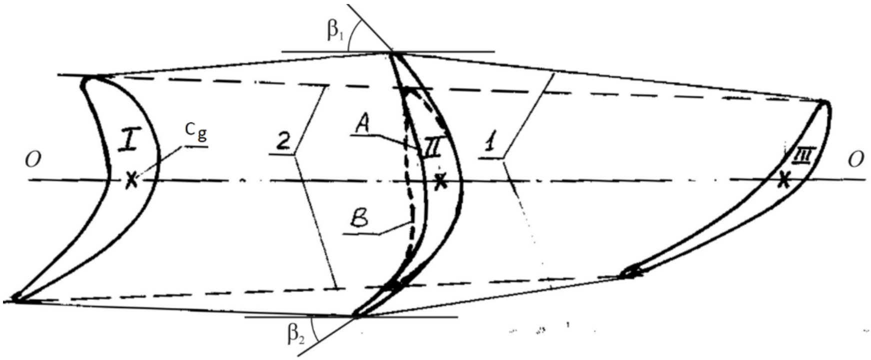

Fig. 1. An example of a preliminary design of a turbine blade

Profiles of the root (I), medium (II, A) and peripheral (III) sections are designed in accordance with the aerodynamic and strength (W and F) characteristics [3,4]. Having located the centers of gravity of sections on the axis of the blades O-O, we see that the lines passing along the inlet and outlet edges of the blades have a fracture in the middle part (line 1). Such a spasmodically changing shape of the blade negatively affects its vibration reliability [4], and also does not fit into the size of the radial gap between the rotating disk with the blades and the guide vane. Therefore, it is proposed to apply a biconvex profile (Fig. 1, B), developed in the Moscow Power Engineering Institute by one of the authors (Fig. 2, profiles 1,2) [3], whose width is within the limits of lines 2-2 (Fig. 1).), corresponding to a smooth change in the width of all cross sections of the blade along its height.

This form of the profile allows, with the necessary W and F, to reduce the width of the profile and to ensure compliance with the conditions of vibration reliability: line 2, var. In are straight and the width of the scapula gradually decreases from the root to the periphery.

An important advantage of the biconvex profile is the movement of the maximum bending stress from the thin input edge (zone C, Fig. 1), where one has to expect the stress concentration, to the top of the protrusion on the pressure side (vol. A, fig. 2) [3,4].

Dependencies of relative calculated coefficients of core losses $\overline{\zeta}_{пр}$.· (for the initial profile - curve 3 and for the proposed one - curve 1), depending on the number $M_{2т}$, are shown in Fig.3. .The results of an experimental study of the lattice of conventional profiles (3) and biconvex (profiles 1, 2, Fig. 2) are presented in Fig. 4, where the dependences of the coefficient of profile losses $\overline{\zeta}_{пр}$ on the number $M_{2т}$ at the exit of the grids are shown [3]. A comparison of these data shows that lattice 3 (initial profile) is characterized by increased energy losses (6–8%). For the remaining grids in the $M_{2т}$ = 0.5-1.35 range, the ratio of core losses is $\overline{\zeta}_{пр}$ = 3-4%.

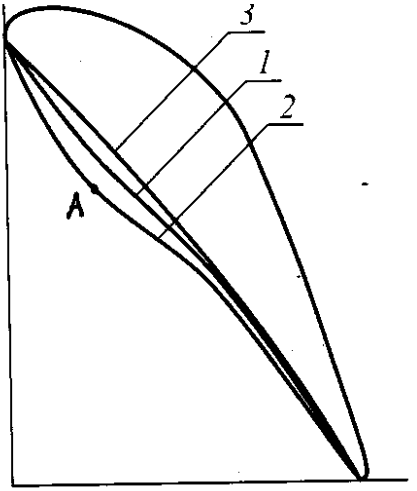

Fig. 2. Variants of the shapes of the profiles of the middle section of the blade

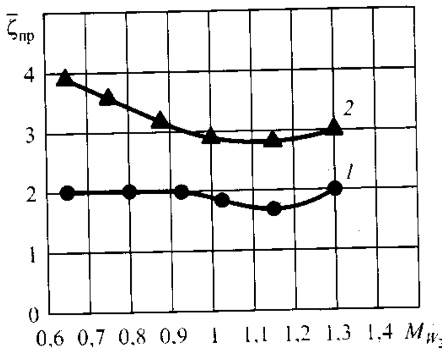

Fig. 3. The relative calculated coefficient of core energy losses for different variants of profiles

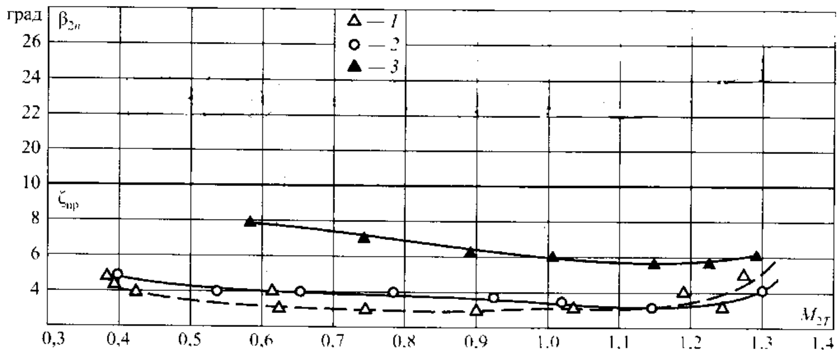

Fig.4. The results of experimental studies of turbine profiles

Findings

When developing the design of the working blades of the last stages of powerful ship turbines, the use of biconvex profiles allows you to:

- perform the optimal design of the blades of the last stages in terms of vibration reliability;
- to ensure the reduction of bending stresses from steam forces;
- To achieve a reduction in the level of profile energy losses in comparison with the standard profile by 2–3 times.

Conclusion

The introduction into the production of lenticular profiles for the working blades of the last stages of the TND of stationary steam turbines was partially carried out at the Kharkov Turbo Generator Plant.

The increase in the power of steam turbine installations of modern large-tonnage vessels has led to the need to increase the length of the working blades of the last stages of the low-pressure cylinder having a variable cross-sectional profile in height. In such turbines, it is advisable to use blades with a biconvex profile, which allows to optimize its geometric, aerodynamic and strength characteristics and to increase the efficiency.

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