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THERMAL AND STRESS STATE OF THE STEAM TURBINE CONTROL VALVE CASING, WITH THE TURBINE OPERATION IN THE STATIONARY MODES

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The purpose of this paper is to determine the most stressful zones and assess the possibility of plastic deformations of the control valve casing in its crack formation zones, with the K-325 steam turbine operation in the stationary modes. The problem is solved in two stages. First, the steam flow characteristics in the steam distribution system and the casing temperature are determined. Then, the elastic stress-strain state of the casing of one of the two valve units (through which the steam consumption is always greater than through the other) is estimated using the values of the casing temperature field. The characteristics of steam flow in the steam distribution system and the thermal state of the control valve casing are determined numerically by the finite element method. The steam flow rates, temperature and pressure on the casing wall are determined based on the solution to the Navier-Stokes equation in a three-dimensional formulation. It is established that the steam temperature before the turbine control valves is practically the same as the one before the stop valve. In the casing itself, after the control valves, with the valves partially open, a significant drop in steam temperature may occur due to throttling. A significant decrease in the steam temperature in the control valve (by 100 °C) is observed at low power with a nominal vapor pressure after the boiler. The calculation of the elastic stress-strain state of the control unit casing was carried out using the finite element method based on the three-dimensional mathematical model for casing deformation. As a result, the stress state of the valve casing was obtained for the different operating modes of the turbine. It is shown that differences in stresses for different modes are associated with changes in the thermal state of the valve casing and the distribution of pressure on its walls. Zones of possible plastic deformations of the valve casing are established. In those zones, the elastic stresses exceed the yield strength of the material. The obtained results clearly show that the most dangerous mode in terms of the control valve casing static strength is not the turbine nominal mode of operation at a power of 320 MW, but a part-load operation mode, at 180 MW.

Keywords: *steam flow, steam distribution system, thermal stress state, valve casing.*

Introduction

Steam turbine control valves are the executive mechanisms of the steam distribution system. It is on their reliable operation that the stability of the turbine operation and its reliability, in the case of possible emergency situations, depends. Therefore, studies of the strength of control valve casings are important to ensure the reliable operation of the turbine and equipment in general.

In the modern K-325 steam turbine, there are two identical combined units [1], each consisting of a stop valve and two control valves located in a common casing. The latter is a complex shape casting (Fig. 1) with overall dimensions of 1.5×1.6×1.5 m. Under a pressure of 24 MPa, superheated steam with a temperature of 54 °C is fed through a branch into a steam inlet chamber (under the stop valve) and, after passing through the valve, further into the chamber above it, common to the two control valves located in the same casing. In the central chamber, the steam passes through a steam strainer, preventing weld flash and foreign inclusions from entering the valve unit and the turbine flow path. In the same casing, there are channels after the control valves, separated from the central chamber by thin bridges.

In this paper, one of the blocks with control valves CV1 and CV3 is considered, since steam consumption through this block is always higher than that through the other (control valves CV2 and CV4 are located in the second block operating parallel to the one under consideration).

Normal operation of a turbine is impossible without a reliable steam flow control system, which is why strict requirements are established for the operational reliability of control valves. As the operating experience shows, cracks appear in the valve steam receiving chamber and branches before the control valves in each of the casings, which requires significant repair and restoration work. This explains the actuality of assessing the strength of the K-325 steam turbine control valve casing.

The purpose of this paper is to determine the most stressful zones and assess the possibility of plastic deformations of the control valve casing in its crack formation zones, with the steam turbine operation in the stationary modes.

To determine the boundary conditions of the thermal strength problem (casing temperature and vapor pressure on its walls), the steam flow and thermal conductivity processes in the valve casing were considered together. Then the problem of estimating the elastic-stress state of the valve casing was solved.

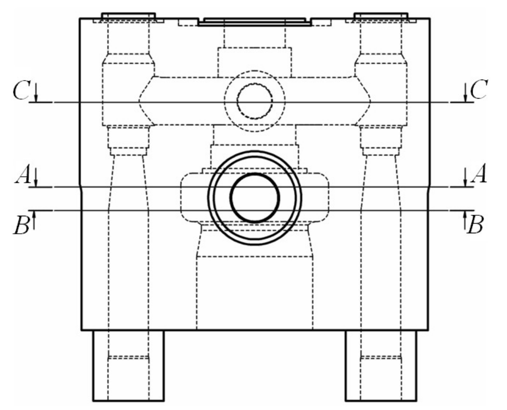

Fig. 1. General view of the control valve block

The valve operates in transitional and stationary operating modes [2]. The transitional modes are caused by the turbine start-up from a cold, hot, or warm state. In this work, the strength of the valve casing is investigated in seven stationary operating modes (see the table below). The design modes were the ones with a maximum and minimum turbine power of 320 and 30 MW; the ones for the case when the opening of the control valve CV3 takes place at 176 and 180 MW; the one with the maximum rate of steam flow through the valve at 240 MW; the part-load operation ones at 100 and 220 MW. The maximum steam flow at 240 MW (and not 320 MW) is due to the fact that the control system uses two blocks of stop and control valves. The steam flow through the second valve block (CV2 and CV4) increases significantly after 240 MW, which leads to a decrease in flow through the first valve block, despite the increase in the opening of control valves CV1 and CV3. At the same time, the opening rate of control valves CV2 and CV4 at 240 MW is higher (depending on turbine power) than that of control valves CV1 and CV3.

Design Modes

Power, MW	Steam flow though the valve block, kg/s	CV1 opening, mm	CV3 opening, mm	Pressure after CV1, MPa	Pressure after CV3, MPa
30	9.23	1.2	0	2.163	12.478
100	48.26	9.4	0	11.223	12.478
176	94.62	18.6	0	21.979	12.478
180	99.43	19.1	0.6	22.031	12.478
220	147.7	26.4	7.6	22.525	19.116
240	162.8	30.1	13.2	22.646	22.115
320	140.3	40.2	34.6	22.798	22.799

Assessment of the Valve Casing Thermal State

The flow of steam in the steam distribution unit of the K-325 steam turbine was modeled in the three-dimensional setting, taking into account the heat losses through the casing walls and thermal insulation. The thermal insulation is the valve outer casing made of 300 mm thick mineral wool (type M2B). In the calculations, it was taken into account as an additional solid with the properties of a heat insulating material. In the mathematical steam flow model, Reynolds-averaged Navier-Stokes equations for a viscous compressible heat-conducting gas were used. They are represented by the equations of steam continuity, change of momentum, and energy conservation [3, 4, 5]

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}\left(\rho U_j\right) = 0;$$

$$\frac{\partial \rho U_i}{\partial t} + \frac{\partial}{\partial x_j}\left(\rho U_i U_j\right) = -\frac{\partial p'}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu_{eff}\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\right];$$

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$$\frac{\partial}{\partial t}(\rho h_{tot})-\frac{\partial p}{\partial t}+\frac{\partial}{\partial x_j}(\rho U_j h_{tot})=\frac{\partial}{\partial x_j}\left(\lambda\frac{\partial T}{\partial x_j}\right)+\frac{\partial}{\partial x_j}(U_i\tau_{ij}).$$

where ρ is the gas (steam) density; t is the time; x_i, x_j are the Cartesian coordinates; U_i, U_j are the flow rate components; p is the pressure; μ_{eff} is the effective viscosity, $\mu_{eff}=\mu+\mu_t$; μ, μ_t are the coefficients of laminar and turbulent viscosity; T is the temperature; p' is the modified pressure

$$p'=p+\frac{2}{3}\rho k+\frac{2}{3}\mu_{eff}\frac{\partial U_m}{\partial x_m};$$

where k is the kinetic energy of turbulence; h_{tot} is the total enthalpy; τ is the effective stress; λ is the thermal conductivity.

Heat transfer both in the valve casing and thermal insulation is modeled by the heat equation

$$\frac{\partial}{\partial t}(\rho_m c_p T)=\frac{\partial}{\partial x_j}\left(\lambda_m\frac{\partial T}{\partial x_j}\right),$$

where ρ_m is the material density; c_p is the material heat capacity; λ_m is the material thermal conductivity.

In this case, the steam flow rate, pressure and temperature were given at the control valve block inlet. The boundary conditions at the valve block inlet were the flow rate values determined through the steam flow, its density and cross-sectional area. The flow rates on the valve casing walls were zero. The flow rates

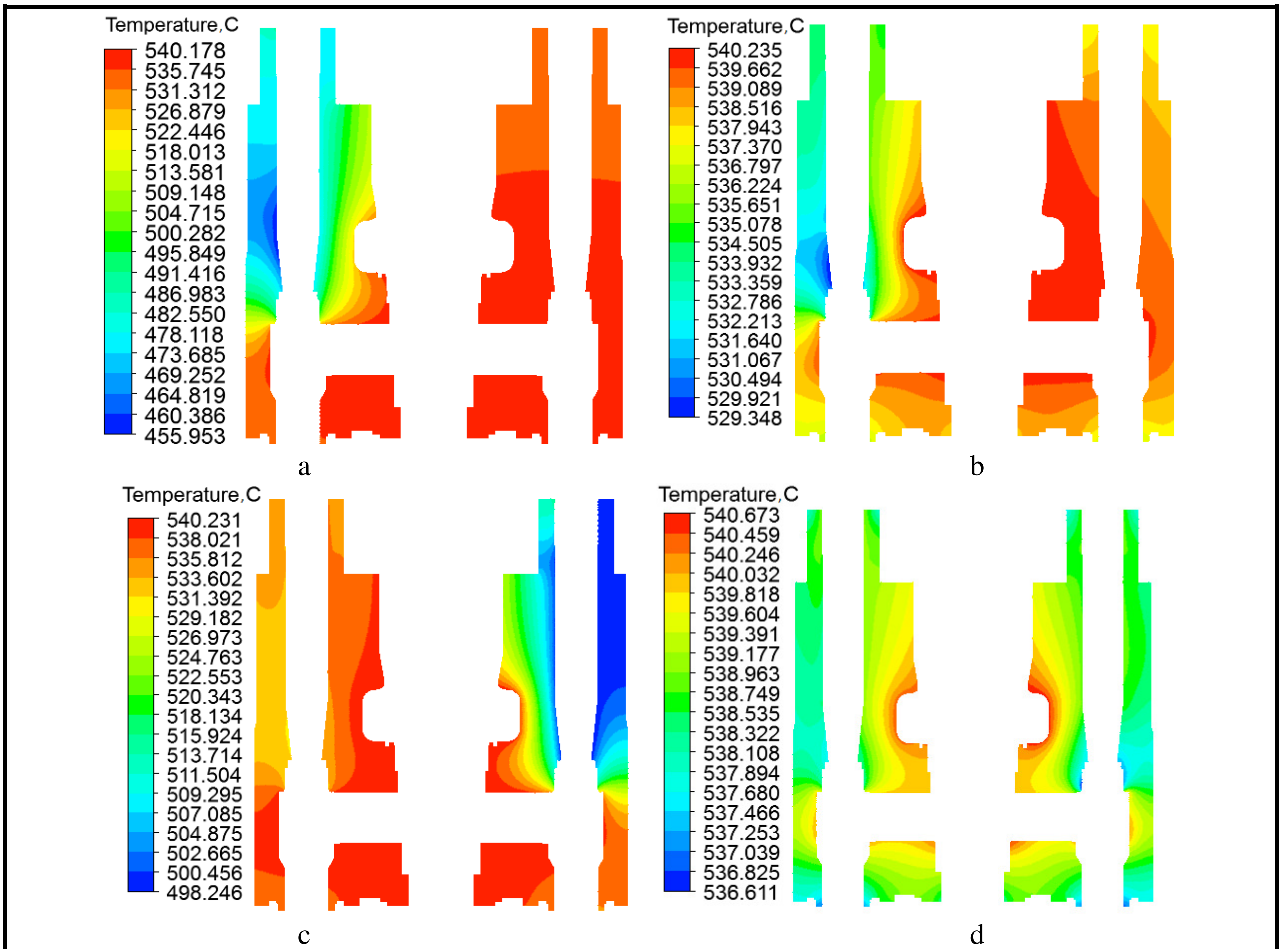

Fig. 2. Casing temperature at power:
a – 30 MW; b – 176 MW; c – 180 MW; d – 320 MW

on the valve casing walls were zero. At the block outlet, the conditions were represented by static steam pressure values. The casing inner surface temperature corresponded to the temperature of steam in the near-wall nodes. On the outer surface of the thermal insulation, the boundary conditions of free convection were given.

As a result of the numerical simulation of steam flow and thermal conductivity, the casing temperature values and steam characteristics (speed, pressure and temperature) in the valve block were obtained. The finite element model in the computational models under consideration consisted of 33 million elements and 9 million nodes with five cells (elements) in the boundary layer.

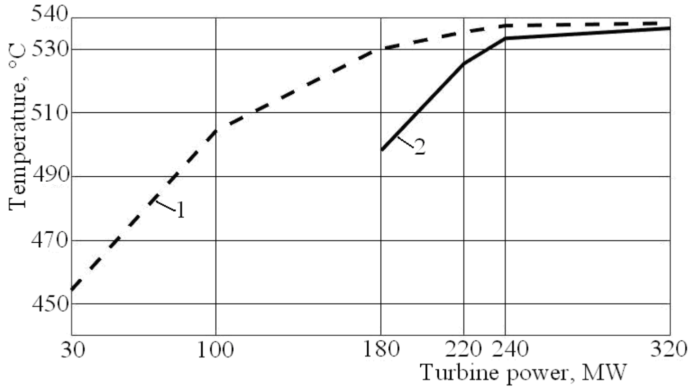

Fig. 3. Seat wall temperature after the control valves:
1 – seat temperature of control valve CV3;
2 – seat temperature of control valve CV1

From the calculation results shown in Fig. 2, it follows that the temperature field of the control valve casing is different for different stationary operating modes. The steam temperature before the turbine control valves is practically the same as that before the stop valve. In the casing itself, after the control valves, with the valves partially open, a significant drop in the steam temperature may occur due to throttling. Fig. 3 shows the temperature of outlet branch walls after the control valves (CV1, CV3) at different turbine power. A significant decrease in steam temperature in the control valves (by 100 °C) is observed at low power with a nominal vapor pressure after the boiler. According to the results of the assessment of the thermal state of the valve block casing under consideration, it can be said that the difference between the temperatures of the casing and steam is largely independent of the turbine power and the opening value of control valves CV1 and CV3, except for the outlet branches and control valve seats, where there is a significant decrease in temperature compared to that of steam. The influence of the temperature gradient, which reaches a value of 100 °C, is registered further when evaluating the elastic-stress state of the valve casing.

Assessment of the Valve Casing Elastic Stress State

The calculation of the elastic stress-strain state of the K-325 steam turbine control unit casing is performed using the three-dimensional mathematical model for casing deformation. The problem was solved by the finite element method. Full deformations at a point are described by the following equation [6]:

$$\{\varepsilon_n\} = \{\varepsilon^{th}\} + [D]^{-1}\{\sigma_e\},$$

where $\{\sigma_e\}$ is the vector of elastic stresses; $[D]$ is the stiffness matrix; $\{\varepsilon_n\}$ is the vector of elastic deformations; $\{\varepsilon^{th}\} = \Delta T\left[\alpha_x^{se}\ \alpha_y^{se}\ \alpha_z^{se}\ 0\ 0\ 0\right]^T$ is the vector of temperature deformation; $\alpha_x^{se}, \alpha_y^{se}, \alpha_z^{se}$ are the coefficients of thermal expansion.

The finite element model in the calculations under consideration consisted of 197 thousand elements and 289 thousand nodes. The boundary conditions in the form of the casing temperature and pressure on the internal surfaces of the casing were obtained in the calculations of the thermal state in the corresponding modes. Since the

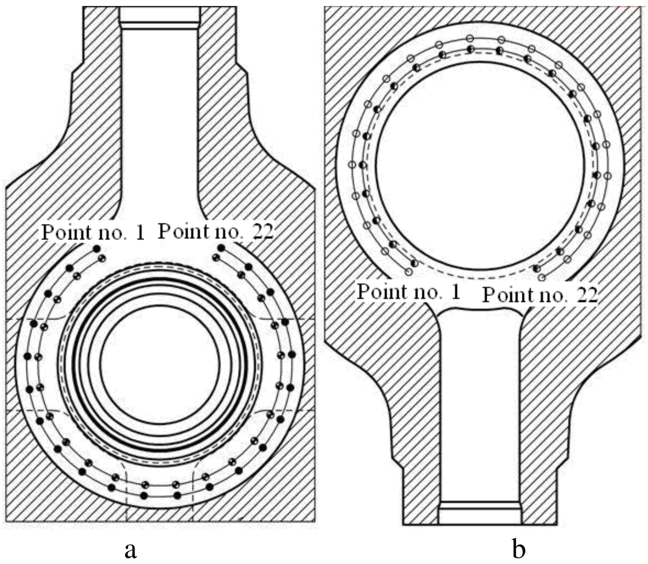

Fig. 4. Steam distribution chamber:
a – upper part, cross-section B-B; b – lower part, cross-section;
– fist circuit; – second circuit;
– third circuit; – fourth circuit

finite-element models for estimating the temperature state and the elastic stress state of the casing were not the same, the temperature values for the casing and pressure on its surface were interpolated to a new grid to determine the stress state.

As a result of the numerical solution to the problem, the stress state of the valve casing was obtained in different operating modes of the turbine. The assessment of elastic equivalent stresses was carried out on the surface of the steam inlet chamber. The results are given for the four circuits in Fig. 4, where the casing cross-sections A-A and B-B are shown (Fig. 1). These areas of the valve casing were chosen based on the fact that cracks appear in them during operation.

The calculations revealed that among the modes considered, the highest stresses are observed at a power of 180 MW. It was found that for this mode, in the first circuit, the stresses reach 100 MPa, whereas from the side of the control valve CV3 they are 75 MPa, and in the vicinity of the inlet branch, 90 MPa (Fig. 5, a). In the second circuit, the stresses increase up to 160 MPa in the central part of the chamber, from the side of the control valve CV1, up to 110 MPa, from the side of the control valve CV3, up to 90 MPa, and at the inlet branch, up to 135 MPa (Fig. 5, b). In the third and fourth circuits, a significant stress drop is observed: from the side of the control valve CV1, the stresses reach 100 MPa; from the side of the control valve CV3, 120 MPa; and in the rear part of the chamber, 30 MPa (Figs. 5, c–d).

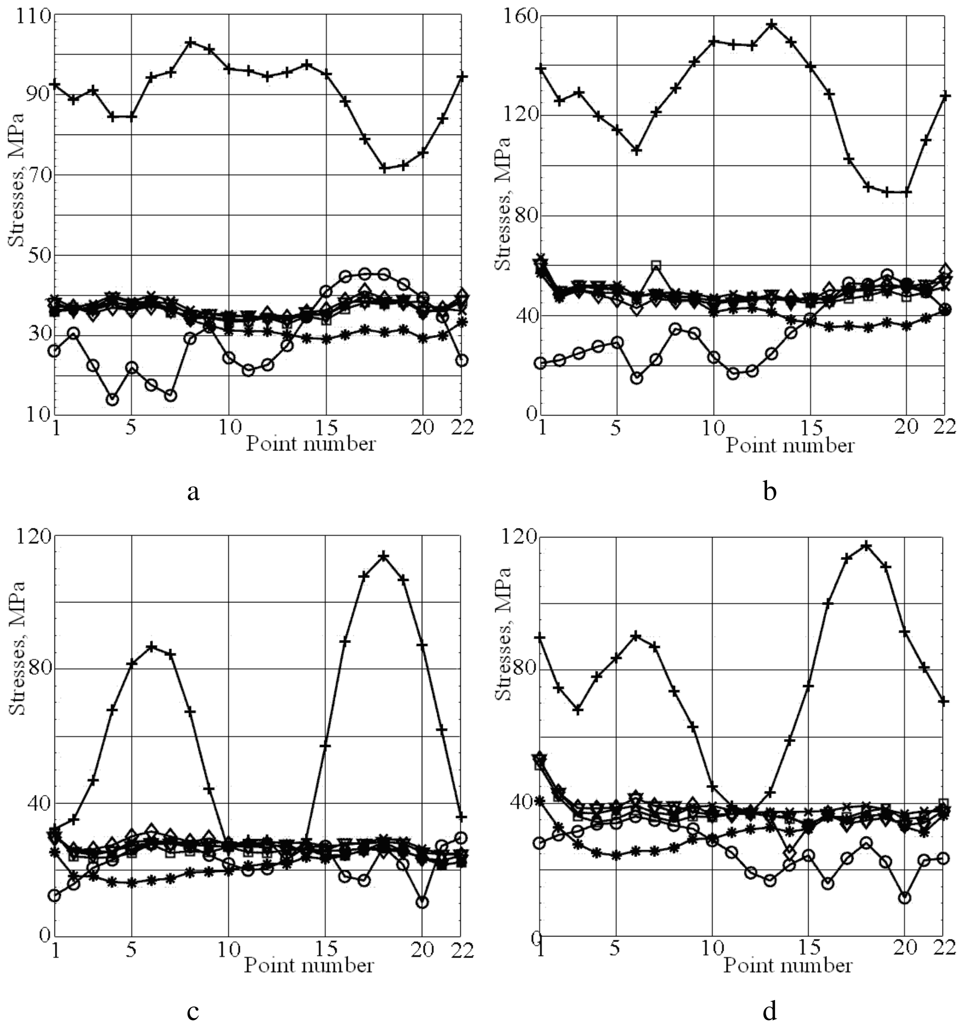

Fig. 5. Stress in the circuits:

а – first; б – second; в – third; г – fourth;

Turbine power: – 30 MW; – 100 MW; – 176 MW; – 180 MW; – 220 MW; – 240 MW; – 320 MW

With a power of 30 MW in the first and second circuits, the following stresses were obtained: from the side of the control valve CV3, 45 MPa in the first circuit and 58 MPa in the second one. At the same time, from the side of the control valve CV1, the stresses reach 20 MPa. For other operating modes, the stress state is almost the same. In the first circuit, the stresses reach 40 MPa with a slight decrease in the rear part of the chamber up to 35 MPa. In the second circuit, the stresses reach values of 45 MPa with an increase in the vicinity of the inlet branch up to 60 MPa. In the third circuit, the stresses reach 30 MPa, and in the fourth one, 40 MPa.

Maximum stresses in the valve casing are observed in the branches before the control valves CV1 and CV3. It should be noted that it is in these areas that the formation of cracks is observed during operation. The stresses at the four points, shown in Fig. 6, were determined depending on turbine power (Fig. 7). In the turbine operating mode corresponding to a power of 180 MW, the stresses at points 1 and 2 reach 190 MPa, and at points 3 and 4 they reach 160 MPa and 130 MPa, respectively. Accordingly, in the mode corresponding to a power of 30 MW, the stresses before the inlet to the control valve CV1 reach 100 MPa, and before inlet to the control valve CV3, 30 MPa. In the other modes, the stresses at the points under consideration are close to 80 MPa.

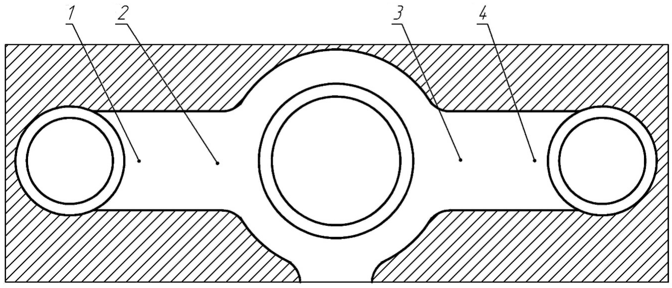

Fig. 6. Scheme of branches before the control valves
(cross-section C-C, Fig.),
points 1–4 refer to stress control places

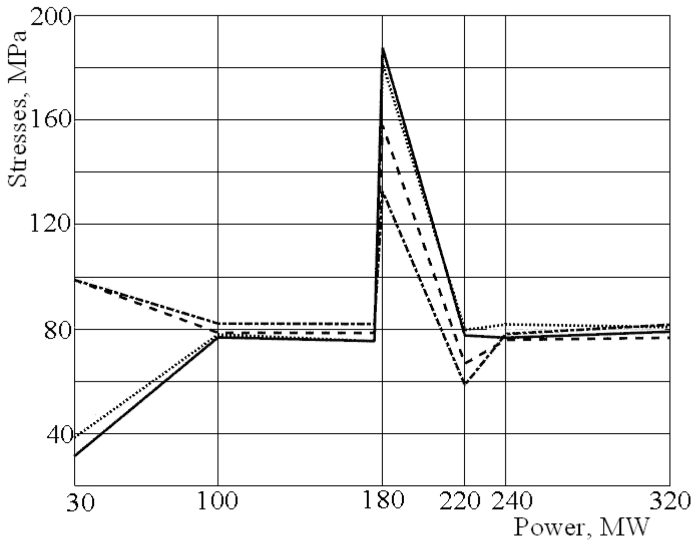

Fig. 7. Stresses at points:
– 1; – 2; – 3; – 4

The valve casing is made of 15H1M1FL steel with the yield strength of 168.2 MPa. Mechanical properties were taken for a metal temperature of 540 °C [7]. As follows from the results obtained, the equivalent stresses exceed the yield strength, which indicates the possible appearance of plastic deformations. This indicates the need to study the valve casing strength in the turbine operation mode corresponding to a turbine power of 180 MW in the elastic-plastic formulation. Differences in stresses in different operating modes are associated both with changes in the valve casing thermal state and pressure distribution on its walls. The results obtained can be used in solving the problem of cyclic strength.

Conclusions

A numerical analysis of the steam flow and heat conduction in the steam distribution system was carried out in the three-dimensional formulation for the stationary operating modes of the K-325 turbine with different power. The boundary conditions for solving the problem of the thermally stressed state are determined. It is revealed that with decreasing turbine power, the temperature of the walls of outlet branches decreases. The maximum temperature decrease to 455 °C is observed at a power of 30 MW. The largest stresses exceeding the material yield strength and causing the appearance of plastic deformations were detected at a power of 180 MW. For the first time, it is shown that a decrease in the casing temperature caused by steam throttling after the two control valves (by 50 °C for CV3 and 15 °C for CV1) increases the stresses in the casing more than the decrease in temperature only after one valve (PK1) by 100 °C. Based on the results obtained, it can be said that the most dangerous mode according to the criteria of static strength of the control valve casing is not the nominal operating mode of the turbine at a power of 320 MW, but a part-load operation mode, at 180 MW.

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Тепловий й напружений стан корпуса регулюючого клапана парової турбіни на стаціонарних режимах роботи

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Метою даної роботи є визначення найбільш напружених зон і оцінка можливості появи пластичних деформацій корпусу регулюючого клапана в місцях появи тріщин під час експлуатації на стаціонарних режимах роботи парової турбіни К-325. Задача розв'язується в два етапи. Спочатку визначаються характеристики течії пари в системі паророзподілу та температура корпусу. Потім оцінюється пружний напружено-деформований стан корпусу одного з двох блоків клапанів (через який витрата пари завжди більша, ніж через інший) з використанням значень поля температур корпусу. Характеристики течії пари в системі паророзподілу та тепловий стан корпусу регулюючого клапана визначаються чисельно методом скінченних елементів. Швидкості, температура та тиск пари на стінці корпусу знаходяться на основі розв'язання рівняння Нав'є-Стокса в тривимірній постановці. Встановлено, що температура пари перед регулюючими клапанами турбіни практично не відрізняється від температури перед стопорним клапаном. У самому ж корпусі за регулюючими клапанами за їх часткового відкриття може відбуватися суттєве зниження температури пара через дроселювання. Значне зниження температури пари в регулюючому клапані (на 100 °С) спостерігається на малих потужностях з номінальним тиском пари за котлом. Розрахунок пружного напружено-деформованого стану корпусу блоку регулювання здійснено з використанням методу скінченних елементів на основі тривимірної математичної моделі деформування корпусу. В результаті отримано напружений стан корпусу клапана за різних режимів роботи турбіни. Показано, що відмінності в напруженнях для різних режимів пов'язані зі змінами теплового стану корпусу клапана та розподілом тиску на його стінки. Встановлено зони можливих пластичних деформацій корпусу клапана, де пружні напруження перевищують межу плинності матеріалу. З отриманих результатів випливає, що найбільш небезпечним режимом за статичної міцності корпусу регулюючого клапана є неномінальний режим роботи турбіни за потужності 320 МВт, а частковий – за 180 МВт.

Ключові слова: *течія пари, система паророзподілу, термонапружений стан, корпус клапана.*

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INVESTIGATION OF THE STRESS STRAIN STATE OF THE LAYER WITH A LONGITUDINAL CYLINDRICAL THICK-WALLED TUBE AND THE DISPLACEMENTS GIVEN AT THE BOUNDARIES OF THE LAYER

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This paper proposes an analytical-numerical approach to solving the spatial problem of the theory of elasticity for the layer with a circular cylindrical tube. A cylindrical empty thick-walled tube is located inside the layer parallel to its surfaces and is rigidly fixed to it. It is necessary to investigate the stress-strain state of the elastic bodies of both the layer and tube. Stresses are given on the inner surface of the tube, and displacements, on the boundaries of the layer. The solution to the spatial problem of the theory of elasticity is obtained by the generalized Fourier method with respect to the system of Lamé's equations in the cylindrical coordinates associated with the tube and the Cartesian coordinates associated with the boundaries of the layer. Infinite systems of linear algebraic equations obtained as a result of satisfying the boundary and conjugation conditions are solved by the truncation method. As a result, displacements and stresses are obtained at various points of the elastic layer and elastic tube. Due to the selected truncation parameter for the given geometrical characteristics, the satisfaction of boundary conditions has been brought to 10^{-3}. *An analysis of the stress-strain state for the elastic body at different thicknesses of the tube, as well as at different distances from the tube to the boundaries of the layer is conducted. Graphs of normal and tangential stresses at the boundary of the tube and layer, as well as normal stresses on the inner surface of the tube are presented. These stress graphs indicate that as the tube approaches the upper boundary of the layer, the stresses in the elastic bodies of both the layer and tube increase, and with decreasing tube thickness, the stresses in the elastic body of the layer decrease, growing in the elastic body of the tube. The proposed method can be used to calculate structures and parts, whose design schemes coincide with the formulation of the problem of this paper. The analysis of the stress state can be used to select the geometrical parameters of the designed structure, and the stress graph at the boundary of the tube and layer can be used to analyze the strength of the joint.*

Keywords*: thick-walled tube in a layer, Lamé's equations, generalized Fourier method.*

Introduction

When designing composite structures and components whose calculation scheme is the layer with a built-in longitudinal circular tube, it is necessary to have an idea of the stress-strain state of the layer and tubes, as well as the stress in their joint. To achieve this, it is required that there be a method of calculation that would give an opportunity to obtain the result with the necessary accuracy.