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STRENGTH AND SERVICE LIFE OF A STEAM TURBINE STOP AND CONTROL VALVE BODY

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The stability of operation of steam turbines depends (along with other factors) on the reliable operation of their steam distribution systems, which are based on stop and control valves. This paper considers the strength of the elements of the K-325-23.5 steam turbine valves, in whose bodies, after 30 thousand hours of operation, cracks came to be observed. Previously determined were the nature of gas-dynamic processes in the flow paths of the valves and the temperature state of the valve body in the main stationary modes of operation. To do this, a combined problem of steam flow and thermal conductivity in stop and control valves was solved in a three-dimensional formulation by the finite element method. Different positions of the valve elements were considered taking into account the filter sieve. The assessment of the thermal stress state of the valve body showed that the maximum stresses in different operating modes do not exceed the yield strength. Therefore, the assessment of the creep of the valve body material is important to determine the valve body damage and service life. Modeling the creep of the stop and control valves of the turbine was performed on the basis of three-dimensional models, using the theory of hardening, with the components of unstable and steady creep strains taken into account. The creep was determined at the maximum power of the turbine for all the stationary operating modes. The maximum calculated values of creep strains are concentrated in the valve body branch pipes before the control valves and in the steam inlet chamber, where in practice fatigue defects are observed. However, even for 300 thousand hours of operation of the turbine (with a conditional maximum power) in stationary modes, creep strains do not exceed admissible values. The damage and service life of the valve bodies were assessed by two methods developed at A. Pidhornyi Institute of Mechanical Engineering Problems of the NAS of Ukraine (2011), and I. Polzunov Scientific and Design Association on Research and Design of Power Equipment. (NPO CKTI) – 1986. The results of assessing the damage and the turbine valve body wear from the effects of cyclic loading and creep of the turbine in stationary modes for 40, 200 and 300 thousand hours show that the thermal conditions of the body in the steam inlet chamber are not violated (without taking into account possible body defects after manufacture). The damage in valve body branch pipes after 300 thousand hours of operation exceeds the admissible value, with account taken of the safety margin. At the same time, the damage from creep in stationary operating modes is about 70% of the total damage. The maximum values of damage are observed in the areas of the body where there are defects during the operation of the turbine steam distribution system. The difference between the results of both methods in relation to their average value is ~20%.

Keywords: *stop and control valve, steam distribution system, finite element method, thermal stress, creep, cyclic fatigue, service life.*

Introduction

Reliable operation of power turbines is associated with ensuring the strength of their elements and components. The stability of operation of a turbine significantly depends on the reliable operation of its steam distribution system, the main components of which are stop and control valves. Experience in the operation of steam turbine valves shows that there are places in their body where there is an accumulation of noticeable fatigue damage. Thus, [1] gives data on the operating time of some steam turbine stop and control valves. For example, on one of the K-200-130 turbines, the left and right stop valves of the high-pressure cylinder were dismantled after 210 and 230 thousand operating hours due to the intensive cracking of the body. In 2003, on the K-200-130 turbine that had operated for 110 thousand hours and started 275 times, a through crack was detected in the body of the high-pressure cylinder stop valve, which developed from the inner surface, having a length of 75 mm with a valve body wall thickness of 45 mm. The crack was processed and welded. During the inspection of a PT-50-130-4 TMZ turbine after 378 thousand hours of operation and 474 starts, a crack was detected on the inner surface of the stop valve.

Therefore, the study of the strength of steam distribution system elements is of great practical importance to ensure the reliable operation of the turbine and power equipment [2]. The formation of defects in the bodies of stop and control valves can lead to significant repair and restoration work.

The analysis of modern approaches to the estimated assessment of the reliability of control valves [3, 4] showed that one of the main problems in solving strength problems is the correct definition of boundary conditions (pressure on the walls of the body and its temperature). Most of the studies on control valves pay considerable attention to the efficiency of the steam distribution system to further optimize the flow paths of the valves and increase the efficiency of the power plant as a whole. Determining the boundary conditions to address strength issues requires a simultaneous solution of the problems of steam flow and thermal conductivity of the valve body elements, as these processes are interconnected. In [5], the authors determine the pressure on the walls of the body of the K-325-23.5 steam turbine stop and control valves and the valve body temperature. A general view of the system of stop and control valves of the K-325-23.5 steam turbine is shown in figure 1.

From the above it can be concluded that the study of the stress, creep, cyclic fatigue and assessment of the service life of the steam distribution system, using the results of determining the boundary conditions based on the study of steam flow and thermal conductivity, are relevant.

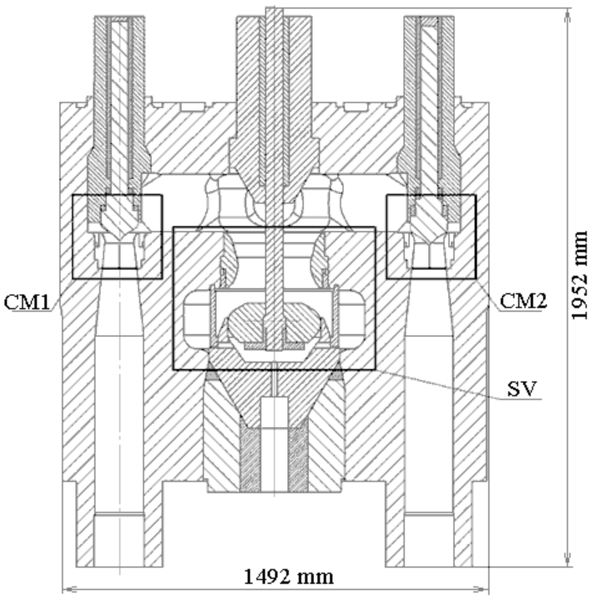

Fig. 1. Stop and control valves of the K-325-23.5 steam turbine

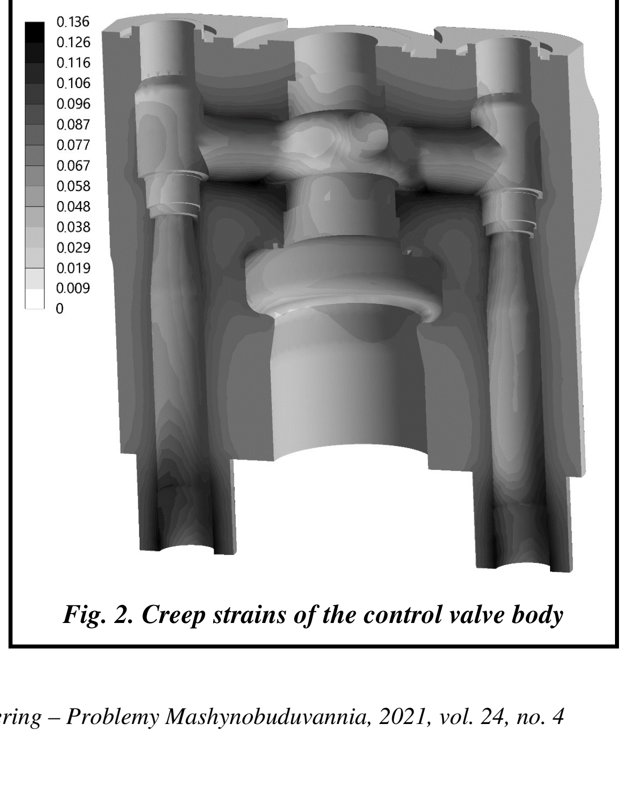

Fig. 2. Creep strains of the control valve body

Creep of a Stop and Control Valve Body

The results of [6] show that the thermal stresses of a stop and control valve body do not exceed their permissible values during the operation of the steam distribution system in stationary modes. To determine the causes of damage during the operation of the K-325-23.5 steam turbine valves, it is advisable to determine the creep of the body because creep deformations of the material can cause damage during prolonged valve operation. In this paper, the creep of the valve body in stationary operating modes is analyzed using three-dimensional finite element models with account taken of the results of the analysis of the thermal conductivity and thermal stressed state of the control valve body according to the method [7] based on the theory of hardening.

To substantiate the quality of the calculation model of the steam turbine control valve body, the analysis of the stress-strain state was performed with different mesh discretization. The analysis showed that the difference between the results when using the proposed mesh discretization (number of nodes – 286440; elements – 195280), in relation to the twice smaller mesh, is ~1.4%. This indicates that the use of a finite element model of the control valve body is acceptable in terms of the accuracy of the results obtained and efficiency of the use of computing resources.

The total strain is defined as follows:

$$\{\varepsilon_{tot}\} = \{\varepsilon^{el}\} + \{\varepsilon^{pl}\} + \{\varepsilon^{t}\} + \{\varepsilon^{cr}\},$$

where $\{\varepsilon^{el}\}$ is the elastic strain vector; $\{\varepsilon^{pl}\}$ is the plastic strain vector; $\{\varepsilon^{t}\}$ is the thermal strain vector; $\{\varepsilon^{cr}\}$ is the creep strain vector.

In this case, the creep strain is found as a scalar quantity from the equation

$$\varepsilon^{\rm cr} = \frac{C_1 \sigma_e^{C_2} t^{C_3+1} e^{-C_4/T}}{(C_3+1)} + C_5 \sigma_e^{C_6} t e^{-C_7/T},$$

where $C_1 \ldots C_7$ are constants; t is time; T is temperature.

Taking into account the elastic stresses and thermal state of the valve body [8, 9], the constants of the creep model were determined by the 15X1M1F steel diagrams, which corresponded to temperatures of 550 °C, 525 °C and stresses of 80 MPa, 60 MPa. The constants are given below.

C_1	C_2	C_3	C_4	C_5	C_6	C_7
1.4627×10^{-19}	3.2952	-0.4408	27260	3.0817×10^{-22}	1.1303	3196

The creep strains of the valve body were determined by the maximum power of the turbine for all stationary modes in an inhomogeneous thermal field. It can be noted that higher creep strains are observed on the inner walls of the valve body, while on the outer ones there is almost no creep (Fig. 2). The maximum strain values, which are approximately equal to the value of 0.136% at time t=200 thousand h (Fig. 3), are concentrated in the branch pipes before the control valves. In the central chamber of the valve body the intensity of strain is lower. Note that the accumulation of fatigue defects in practice is observed in the branch pipes before the control valves and in the steam inlet chamber.

Figure 4 shows the change in strain as a function of time at points A, B, C, D, E, F in the process of creep. It should be noted that the creep is more severe in the lower part of the steam inlet chamber. At points D and E, the creep strain does not exceed 0.03%. From the analysis of the results obtained, it can be noted that the steady-state creep occurs approximately after 30 thousand hours of operation of the steam distribution system.

Given that in practice significant fatigue defects in the valve body are formed after four years of operation (about 30 thousand hours) and based on the results obtained, we can conclude that the creep of the material is not the only factor that leads to the formation of data damage during this period.

The results obtained, allow us to note that significant creep strains are observed in the zones corresponding, in practice, to the places where damages of the valve body material are formed during operation. The creep strains, which are 0.136% for 200 thousand hours of operation, do not exceed permissible values (less than 1%).

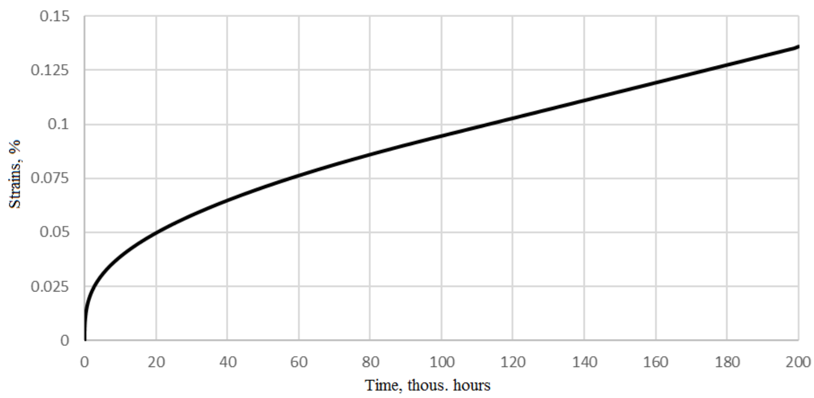

Fig. 3. Creep strains in the branch pipes before the control valves

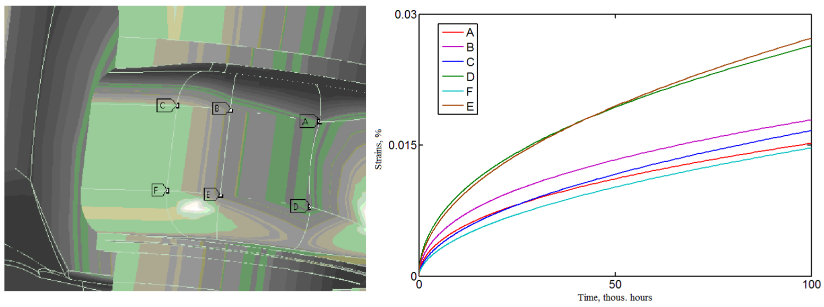

Fig. 4. Creep strain in the central chamber of the stop valve

Cyclic Damage and Service Life of the Valve Body

The calculation of the cyclic fatigue and service life of the valve body of the steam distribution system of the K-325-23.5 steam turbine was performed using the SOU-N MEB 40.1-21677681-52 technique [10] developed in IPMash NAS of Ukraine (2011), and the low-cycle fatigue with account taken of creep, using the RTM 108.021.103 technique [11] developed by NPO CKTI (1986). These techniques differ in their approaches to summarizing damage from cyclic loading and creep. In [10], the linear sum of static and cyclic damages is used, and in [11], the effect of creep is taken into account when determining the permissible number of load cycles.

During service life the high-temperature equipment of operating steam turbines undergoes various cyclic loads, and the stress amplitude may be lower or higher than the fatigue limit. The most common method of calculating damage under cyclic loads is based on the linear law of their accumulation. If N_i is the number of cycles before destruction at a stress of σ_1, then the fraction of the total durability by the number of cycles n_1 will be a function of n_i/N_i. According to the linear law, destruction occurs if $n_1/N_1+n_2/N_2+n_3/N_3+\ldots=1$.

The linear summation of damages has disadvantages due to the fact that the sequence of formation of damages from cyclic loads is not taken into account. Experiments show [12] that this law is applicable when cyclic loads with different stress levels throughout the service life are evenly distributed in small blocks (according to the number of cycles). At the same time, none of the other methods has a comprehensive and complete justification, given the variety of factors that affect the process and the final result of determining the damage (e.g., strain hardening (slander) or weakening, stress concentration).

According to SOU-N MEB 40.1-21677681-52 [10], the total damage П accumulated in the study area of the body, during operation under conditions of the combined action of creep and cyclic load is determined by the formula

$$\Pi = \Pi_{\rm st} + \Pi_{\rm cy} = \sum_{j=1}^{q} \frac{t_j}{t_{pj}} + \sum_{i=1}^{k} \frac{n_i}{N_{pi}},$$

where $\Pi_{\rm st}$, $\Pi_{\rm cy}$ are the static and cyclic damages.

Permissible total damage [П], taking into account safety factors will be written in the form

$$[\Pi] = \max\left\{\left([\Pi_{\rm st}]+[\Pi_{\rm cy}]\right), n_d\Pi\right\} = \max\left\{\left(\sum_{j=1}^{q} \frac{t_j}{[t_p]_j} + \sum_{i=1}^{k} \frac{n_i}{[N]_i}\right), n_d\Pi\right\}, \quad (1)$$

where $[\Pi_{\rm st}]$, $[\Pi_{\rm cy}]$ are the static and cyclic damages with account taken of safety factors; t_j is the operating time in the j-th stationary operating mode with the creep stresses σ_j^c; t_{pj} is the time before the onset of the limit state of a crack under the action of the creep stresses σ_j^c, which are determined by the endurance diagram of the material (Fig. 5); $[t_p]_j$ is the permissible operating time in the j-th mode under long-term strength conditions, determined by the endurance diagram of the material (Fig. 5), but with the creep stresses $n_{LS}\sigma_j^c$. For valve bodies, according to [13], it is accepted that n_{LS}=1.5; n_i is the number of i-th operating cycles; N_{pj} is the number of cycles before the occurrence of a crack in the i-th operating cycle, which corresponds to the strain amplitude ε_{ai}, and is determined by experimental fatigue diagrams for the isothermal symmetric stress cycle (Fig. 6); $[N]_i$ is the permissible number of cycles of the i-th type; n_d=5 is the safety factor for total damages; q is the number of stationary cycles characterized by the temperature T_j and steady-state creep stresses σ_j^c; k is the number of different operating cycles, which are characterized by different reduced strain amplitude ε_{ai}, which was determined in reducing asymmetric cycles to symmetrical isothermal cycles according to the method [11].

Figure 6 shows the used experimental diagrams of the valve body steel (15X1M1FL) for the isothermal symmetrical cycle in the form of the dependence of the strain amplitude ε_{ai} on the number of cycles N until the occurrence of cracks at temperatures of 400–565 °C [14].

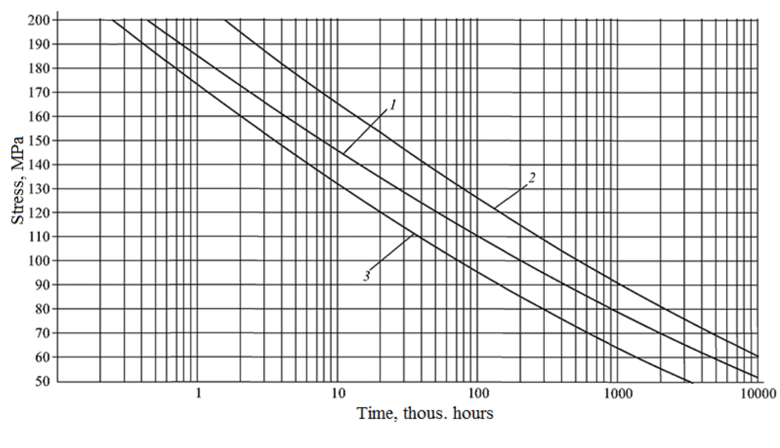

Fig. 5. Diagrams of average quality values of the 15X1M1F steel endurance at T=545 °C:
1 – centrifugally cast pipes; 2 – deformed pipes; 3 – cast structures

Figure 5 shows long-term strength diagrams for the 15H1M1F steel. These dependencies are obtained on the basis of the statistical processing of tests of a large number of samples of a particular material. They are actually medium-quality long-term strength diagrams for the 15H1M1F steel for different technological states [15]. The comparative analysis of the obtained data shows that the long-term strength for the same steel grade depends on the methods of its manufacture.

The value of the permissible number of cycles is obtained by the relation

$$[N]_i = \min(N_{1i}, N_{2i}),$$

where $N_{1i}=N_{pi}/n_N$ is the number of cycles before the crack initiates, with account taken of the safety factor in cycles n_N (according to [13] we accept $n_N=5$); N_{2i} is the number of cycles before the crack initiates, with account taken of the safety factor in the strain amplitude $n_\varepsilon \varepsilon_{ai}$, determined by the experimental fatigue diagrams for the isothermal symmetric stress cycle (Fig. 6). According to [11], it is accepted that n_ε=1.5.

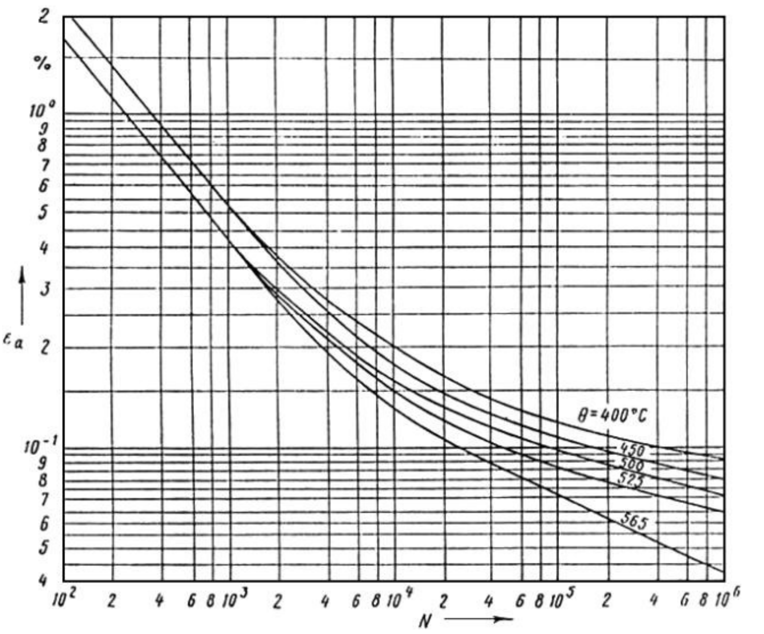

Fig. 6. Experimental fatigue diagrams for the 15X1M1F and 15X1M1FL steel grades for the isothermal symmetric cycle of stresses

According to the RTM 108.021.103 technique [11], the maximum number of cycles in the valve body area under study during operation under conditions of the combined action of creep and cyclic load is determined by the formula

$$N_A = \left[1 - \left|\frac{1.25\sigma^c}{\sigma_{LS}}\right|^q\right] \min\left\{\frac{N_1}{n_N}; N_2\right\}, \tag{2}$$

where q is the exponent in the long-term strength equation ($t = B\sigma^{-q}$) in the time interval (1–2)×10^5 of operation; N_1, N_2 is the number of cycles corresponding, in the fatigue diagram, to the amplitudes ε_a^r and $n_\varepsilon \varepsilon_a^r$; n_N=5, n_ε=1.5 are the safety factors in the number of cycles and strain.

The total damage Π in this case is determined by the equation

$$\Pi = \sum_{i=1}^{k} \frac{n_i}{N_{Ai}}. \tag{3}$$

At elevated temperatures, when the creep of the material occurs in the nominal operating mode, the strain amplitude, which is reduced to a symmetrical isothermal cycle, is determined by the equation

$$\varepsilon_a^r = C\varepsilon_a + \frac{1+\nu}{1{,}5E}\left(\min\{\sigma_{-1}; \sigma_{LS}\} - \min\{\sigma_N; \sigma_N^c\}\right),$$

where C is the coefficient of strain intensity concentration; ε_a is the strain amplitude; σ_{-1} is the fatigue limit at the design temperature; σ_{LS} is the long-term strength of the material over its service life at the design temperature; σ_N is the fatigue limit during the asymmetric stress cycle; σ_N^c is the fatigue limit with the asymmetric stress cycle during creep.

The fatigue limit with the asymmetric stress cycle during creep is determined by the formula

$$\sigma_N^c = \begin{cases} \min\left\{ \dfrac{\sigma_a \sigma_{LS}(\theta_1)}{|\sigma_{\max}|}; \dfrac{\sigma_a \sigma_{LS}(\theta_2)}{|\sigma_{\max} - 2\sigma_a|} \right\} & \text{at} \quad \sigma_{\text{a}} < \overline{\sigma}_{0,2}^c \\ \min\{\sigma_{LS}(\theta_1); \sigma_{LS}(\theta_2)\} & \text{at} \quad \sigma_{\text{a}} > \overline{\sigma}_{0,2}^c \end{cases},$$

where $\sigma_{LS}(\theta_1)$, $\sigma_{LS}(\theta_2)$ are the long-term strength limits over the service life at the appropriate temperature; $\sigma_{\max}$ is the maximum stress in the cycle; σ_a is the calculated stress amplitude.

The fatigue limit with the asymmetric stress cycle is determined by the equation

$$\sigma_N = \frac{\sigma_{-1}}{1 + \dfrac{\sigma_{-1}}{\sigma_{UTS}} \dfrac{1+r}{1-r}},$$

where r is the asymmetry coefficient of the stress cycle; σ_{UTS} is the tensile strength of the material

The stress-cycle asymmetry factor is calculated by the formula

$$r = \begin{cases} \max\left\{ \dfrac{\sigma_{\max} - 2\sigma_a}{\sigma_{\max}}; -1 \right\} & \text{at} \quad \sigma_{\max} > 0 \\ -1 \quad \text{at} \quad \sigma_{\max} < 0 \end{cases},$$

where the maximum stresses in the cycle are determined by the formula

$$\sigma_{\max} = \begin{cases} \overline{\sigma}_{i\max} \quad \text{at} \quad \overline{\sigma}_{i\max} < \sigma_{T1}^c \quad \text{and} \quad 2\sigma_a - \overline{\sigma}_{i\max} < \sigma_{T2}^c \\ \min\{2\sigma_a - \sigma_{T2}^c; \overline{\sigma}_{0,2}^c\} \quad \text{at} \quad \overline{\sigma}_{i\max} < \sigma_{T1}^c \quad \text{and} \quad 2\sigma_a - \overline{\sigma}_{i\max} > \sigma_{T2}^c \\ \sigma_{T1}^c \quad \text{at} \quad \overline{\sigma}_{i\max} > \sigma_{T1}^c \end{cases}.$$

The considered mathematical models [10, 11] are used in this paper to estimate the wear of the body of steam turbine stop and control valves from the action of cyclic loading and creep in stationary modes (excluding possible initial defects of the valves after manufacture).

Cyclic Strength Calculation Results

According to the methods described above, the cyclic damage of the body of the valves of the steam distribution system of the K-325-23.5 steam turbine due to cyclic load and creep in stationary operating modes and their service life have been determined.

Note that damage assessment was carried out without taking into account the possible initial defects of the valve body after its manufacture. As practice shows, after the manufacture of complex structural elements by casting methods, the formation of initial defects is possible. According to the normative documentation [16], it is allowed to use castings of steam turbine parts with defects up to 5 mm. Residual stresses appear during the casting process, which can also affect the stress state of a valve body.

Below is the number of admissible [17] cycles for the design service life of the turbine.

Cold starts	150
Hot starts	2000
Warm starts	1000
Load shedding to idle run	150
Unloading to the lower control-range limit	10,000

The number of unloadings to the lower control-range limit is regulated as no more than 10,000 per 200,000 hours of operation. The lower control-range limit for the K-325-23.5 steam turbine is 100 MW, the turbine being capable of operating for a long time in the load range from 100 to 325 MW. Having the results of thermal stress calculations for the 100, 176, 180, 220, 240 and 325 MW operating modes and the results of creep calculations, it is possible to estimate the cyclic damage and service life of the valve body in stationary modes, using the methods described above. The cyclical damage and service life assessments were performed for three design periods of operation – 40, 200 and 300 thousand hours. The period of 40 thousand hours was chosen as the probable period of formation of noticeable damage in the valve body; 200 thousand hours, as the design service life of the body; 300 thousand hours, as the beyond-design service life of the valve body.

The values of elastic stresses in different stationary operating modes, which were used to assess the damage, were obtained in the calculations of the thermal stress state [6], and are given in table 1. Note that the stress state of the valve body in different operating modes changes even under the same parameters of steam at the steam distribution system inlet. This is due to the fact that when the turbine power changes, the position of the control valves changes, and, as a consequence, throttling effects – a change in the valve-body wall temperature and stress state.

Table 1. Stresses (in MPa) in the body of the stop and control valves in turbine stationary operating modes

Capacity, MW	Chamber		Branch pipe	
	von Mises stress	Intensity	von Mises stress	Intensity
325	56.131	62.974	81.023	90.627
240	62.516	69.789	78.849	88.170
220	60,894	67.982	79.244	88.621
180	150.290	165.150	205.280	220.810
176	63.686	70.672	78.272	87.307
100	81.409	90.921	64.676	72.056
30	27.553	28.232	41.361	47.524

The values of stresses (in MPa) in the steady-state creep state of the material in the nominal operating mode, which were used to assess the damage, are given below.

Chamber		Branch pipe	
von Mises stress	Intensity	von Mises stress	Intensity
30.453	35.064	35.064	49.236

The results of the calculation of the total damage to the body of the control valves for the three periods of operation, using the two methods, are given in table 2. One of the methods (SOU-N MEB 40.1-21677681-52 [10]) uses a linear summation of damage due to cyclic loading and creep and in the second one (RTM 108.021.103 [11]), the effect of creep is taken into account when determining the admissible number of load cycles. The total damage according to the first method was determined by formula (1), and the second one, using relation (3), where the admissible number of cycles was determined by formula (2).

Table 2. Results of the calculation of the total damage to the control valve body

Valve body element	Design time								
	40 thou a year			200 thou a year			300 thou a year		
	$[П_{st}]$	$[П_{cyc}]$	$[П]$	$[П_{st}]$	$[П_{cyc}]$	$[П]$	$[П_{st}]$	$[П_{cyc}]$	$[П]$
SOU-N MEB 40.1-21677681-52									
Chamber	0.020	0.028	**0.048**	0.102	0.139	**0.241**	0.154	0.208	**0.362**
Branch pipe	0.108	0.036	**0.143**	0.538	0.178	**0.716**	0.807	0.267	**1.075**
RTM 108.021.103									
Chamber	–	–	**0.035**	–	–	**0.177**	–	–	**0.266**
Branch pipe	–	–	**0.176**	–	–	**0.879**	–	–	**1.319**

Assessing the valve body wear due to cyclic fatigue and creep in stationary modes of operation shows that after 40,000 and 200,000 hours of operation, the thermal conditions are not violated. The difference between the results of damage in the valve branch pipes before the control valves by different methods is ~20%. The results of research have shown that the main contribution to the accumulation of damage is made by the valve body creep. The results obtained show that the creep of the stop and control valve body contributes to the total damage in stationary modes of operation up to 70% of the total damage.

Taking into account the parameters of the complex distribution of temperature fields in the valve body and the pressure on its walls allowed us to obtain the value of damage in places that coincide with the zones of defects during the operation of the steam distribution system of the steam turbine. Taking into account these factors allowed us to give an effective assessment of the service life of the valve bodies when operating in stationary modes. The obtained values of the total damage to the valve body can be used in the modernization of the operating power units of thermal power plants or in the design of new steam distribution systems for steam turbines.

Conclusions

The creep of the valve body of the steam distribution system of the K-325-23.5 steam turbine at maximum power is considered using three-dimensional finite-element models. Creep strains were determined using the creep model according to the theory of hardening. The temperature and pressure distribution in the valve body was previously determined by the finite element method. To do this, the problem of steam flow in a system of stop and control valves was solved numerically on the basis of the Navier-Stokes equations, taking into account the position of the valves and the influence of the filter sieve. The obtained maximum creep strains after 40, 200 and 300 thousand hours of operation are observed in the branch pipes before the control valves and in the steam inlet chamber of the valve body, the strains being smaller than permissible even after 300 thousand hours of operation of the steam distribution system.

The service life of the valve body was determined taking into account the effect of stress changes due to temperature and pressure differences on the body walls with changes in operating modes for 40, 200 and 300 thousand hours of operation. The service life of the body was assessed by two different methods, which differ in the approaches to summing up the damage from the cyclic load and creep.

The results of research show that the conditions of thermal strength of the valve body in the area of the steam inlet chamber, with account taken of safety factors, are not violated. Creep damage in stationary operating modes is about 70% of the total damage. The maximum values of damage are observed in the areas of the body where there were defects (cracks) during the operation of the turbine steam distribution system. The maximum value of damage in the valve body branch pipes after 300 thousand hours of operation exceeds the limit value, taking into account the safety factor. The discrepancy of the results of the two methods in relation to their average value is about 20%.

The assessment of damage to the valve body is performed without taking into account possible initial defects and residual stresses of the valve, which may occur after manufacturing the body by casting methods. Taking these factors into account can significantly affect the results of the valve body strength assessment.

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Міцність та ресурс корпусу стопорно-регулювальних клапанів парової турбіни

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Стабільність експлуатації парових турбін залежить (поряд з іншими чинниками) від надійної роботи системи паророзподілу, основу якої складають стопорно-регулювальні клапани. В роботі розглядаються питання міцності елементів клапанів парової турбіни К-325-23,5, в корпусі яких після 30 тисяч годин експлуатації почали спостерігатись тріщини. Попередньо визначався характер газодинамічних процесів в проточній частині клапанів та температурний стан корпусу клапанів на основних стаціонарних режимах роботи. Для цього розв'язувалась сумісна задача течії пари та теплопровідності в стопорно-регулювальних клапанах у тривимірній постановці методом скінченних елементів. Розглядалось різне положення елементів клапанів з урахуванням фільтруючого сита. Оцінка термонапруженого стану корпусу клапанів засвідчила, що максимальні напруження на різних режимах роботи не перевищують межі плинності. Тому оцінка повзучості матеріалу корпусу клапанів є важливою для визначення його пошкодження та ресурсу. Моделювання повзучості корпусу стопорно-регулювальних клапанів парової турбіни виконувалось на основі тривимірних моделей з використанням теорії зміцнення. При цьому було враховано складові несталої та усталеної деформації повзучості. Повзучість корпусу визначена за максимальної потужності турбіни для всіх стаціонарних режимів роботи. Максимальні розрахункові значення деформацій повзучості зосереджені в патрубках корпусу перед регулювальними клапанами та в пароприймальній камері, де на практиці спостерігаються втомні дефекти. Проте навіть за 300 тисяч годин експлуатації турбіни (з умовною максимальною потужністю) на стаціонарних режимах деформації повзучості не перевищують допустимих значень. Пошкодження і ресурс корпусу клапанів оці-

нювалися за двома методиками, що створені в Інституті проблем машинобудування ім. А. М. Підгорного НАН України (2011 р.), та в Науково-виробничому об'єднанні з дослідження в проектуванні енергетичного обладнання ім. І. І. Ползунова (НВО ЦКТІ) – 1986 р. Результати оцінки пошкоджуваності і спрацювання ресурсу корпусу клапанів парової турбіни від впливу циклічного навантаження та повзучості турбіни на стаціонарних режимах роботи за 40, 200 та 300 тисяч годин свідчать, що умови термоміцності корпусу в області паропрриймальної камери не порушуються (без урахування можливих недосконалостей корпусу після виготовлення). Пошкодження в патрубках корпусу клапанів після 300 тисяч годин експлуатації перевищують граничне значення з урахуванням коефіцієнтів запасу. При цьому пошкодження від повзучості на стаціонарних режимах роботи складає біля 70% від сумарного. Максимальні значення пошкодження спостерігаються в зонах корпусу, де мають місце дефекти при експлуатації системи паророзподілу турбіни. Розбіжність результатів за обома методиками по відношенню до їх середнього значення становить ~20%.

***Ключові слова**: стопорно-регулювальні клапани, система паророзподілу, метод скінченних елементів, термонапруження, повзучість, циклічна втома, ресурс.*

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