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STEAM TURBINE, GAS TURBINE, STEAM-GAS PLANTS AND ACCESSORY EQUIPMENT

Development of High-Powered Steam Turbines by OAO NPO Central Research and Design Institute for Boilers and Turbines

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Abstract—The article provides an overview of the developments by OAO NPO TsKTI aimed at improvement of components and assemblies of new-generation turbine plants for ultra-supercritical steam parameters to be installed at the power-generating facilities in service. The list of the assemblies under development includes cylinder shells, the cylinder's flow paths and rotors, seals, bearings, and rotor cooling systems. The authors consider variants of the shafting–cylinder configurations for which advanced high-pressure and intermediate-pressure cylinders with reactive blading and low-pressure cylinders of conventional design and with counter-current steam flows are proposed and high-pressure rotors, which can increase the economic efficiency and reduce the overall turbine plant dimensions. Materials intended for the equipment components that operate at high temperatures and a steam cooling technique that allows the use of cheaper steel grades owing to the reduction in the metal's working temperature are proposed. A new promising material for the bearing surfaces is described that enables the operation at higher unit pressures. The material was tested on a full-scale test bench at OAO NPO TsKTI and a turbine in operation. Ways of controlling the erosion of the blades in the moisture–steam turbine compartments by the steam heating of the hollow guide blades are considered. To ensure the dynamic stability of the shafting, shroud and diaphragm seals that prevent the development of the destabilizing circulatory forces of the steam flow were devised and trialed. Advanced instrumentation and software are proposed to monitor the condition of the blading and thermal stresses under transient conditions, to diagnose the vibration processes, and to archive the obtained data. Attention is paid to the normalization of the electromagnetic state of the plant in order to prevent the electrolytic erosion of the plant components. The instrumentation intended for monitoring the relevant electric parameters is described.

Keywords: turbine, parameters, cylinder, seals, blades, shafting, vibration stability, diagnostics, electrolytic erosion

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Starting from 2003, the concept of high-powered turbine plants for combined heat and power stations and nuclear power plants have been developed at OAO NPO Central Research and Design Institute for Boilers and Turbines (TsKTI). The research is being conducted under commercial agreements with the investment foundations of RAO ES (Energy without Borders Foundation), the Ministry of Energy of the Russian Federation, the Ministry of Industry and Trade of Russia, and AO Kontsern Rosenergoatom.

Designs of turbine assemblies and components have been devised that are of independent value and have prospects for further application in the future projects and in the plants in operation [1]. It is extremely important to point out that, when designing the above assemblies, the tasks were formulated in accordance with the technical specifications of the customers aimed at enhancing the economic efficiency, reliability, reparability, flexibility, and lifetime of the turbine plants at optimal cost prices, since the capital investments in construction of serial new-generation power-generating plants—designed for ultra-supercritical steam parameters as a rule—should not exceed the cost prices of the existing power-generating plants by 10–15%.

Having analyzed the demand of the domestic and foreign energy markets and the capabilities of the turbine-construction works of Russia, the designers selected the parameters of the prospective turbine plants presented in Table 1 from which follows that the capacities of the proposed turbines do not exceed the capacities of the already existing turbines; the pressure and temperature of the live steam, however, surpass the achieved values by 10–20 and 20–30%, respectively, which enables a 15% reduction in the cost of the generated energy. The increased steam parameters are not the only factor that ensures high economic and engineering characteristics of the turbine equipment.

Table 1. Technical data of turbines for high-powered power-generating plants

Parameter	K-660-28 turbine, HPC + IPC + LPC or HPC + IPC + 2LPCs	K-800-31 turbine, HPC + IPC + 2LPCs
Rated capacity, MW	660	800
Maximum electric demand, MW	700	860
Live steam:		
flow rate, t/h	1750 ± 50	1846 ± 50
pressure, MPa	28.4	31.0
temperature, °C	595	650
Reheat steam:		
initial pressure, MPa	7.5 ± 0.5 or 5.0 ± 0.5	7.5 ± 0.5
temperature, °C	610	650
pressure loss, % of the live-steam intermediate pressure	10	
Design cooling water temperature, °C	12	
Makeup water flow rate (into the condensate tank), % of the live steam flow rate	3	
Design pressure loss in the heat-exchange steam extraction pipes, % of the pressure in the extraction tube	5	
Steam distribution	Throttling	
HPR and IPR blading	Reactive, three-dimensional blades	
Efficiency, %:		
HPC	91.5	93.0
IPC	92.5	95.4
LPC	83.2	89.4
Last-stage blade length, mm	1200	1200/1320

With two LPCs in use, the efficiency of the turbine is 0.5% higher than that of the turbine with one LPC.

Advancements to the design of the assemblies responsible for the enhancement of the above characteristics also play an important role.

When developing the configuration of the plant and designing the turbine rotors and cylinders, the designers were guided by the following principles [1, 2]:

(1) The flows in the high-pressure and immediate-pressure cylinders (HPCs and IPCs, respectively) should be mutually compensated by arranging the cylinders opposite to each other to ensure a minimum axial force.

(2) The HPC and IPC shells have narrow flanges, which facilitates better heating and, consequently, ensures greater flexibility under low thermal stresses.

(3) The thermal expansion of the cylinders occurs along the horizontal parting line, which reduces the torques applied to the bearing bodies and transverse beams.

(4) The control may be effected by the jet and throttle (preferable) techniques.

(5) High separation pressure of 6.5–7.5 MPa allows a more compact configuration of the HPC, which makes possible a rigid rotor and minimum radial clearances.

(6) The rigid HP rotor ensures the first critical speed that exceeds the rated speed.

(7) The three-dimensional reactive blading of the high-pressure rotor (HPR) and the first stages of the immediate-pressure rotor (IPR) form a vertically variable cross-section considering the three-dimensional current of the flow.

(8) As the steam is not extracted from the HPC shell, the efficiency and reliability of the latter is enhanced.

(9) A system of forced steam cooling (FSCS) of the rotors and the HPC and IPC is provided.

A distinguishing feature of the low-pressure cylinder (LPC) design is the horizontal connection of the inner cylinder to the IPC shell and a special longitudinal guide connected to the lower part of the inner LPC shell to ensure better alignment of the rotor in the flow path under thermal expansion. In this way, optimal inlet clearances are ensured at all stages under all operating conditions.

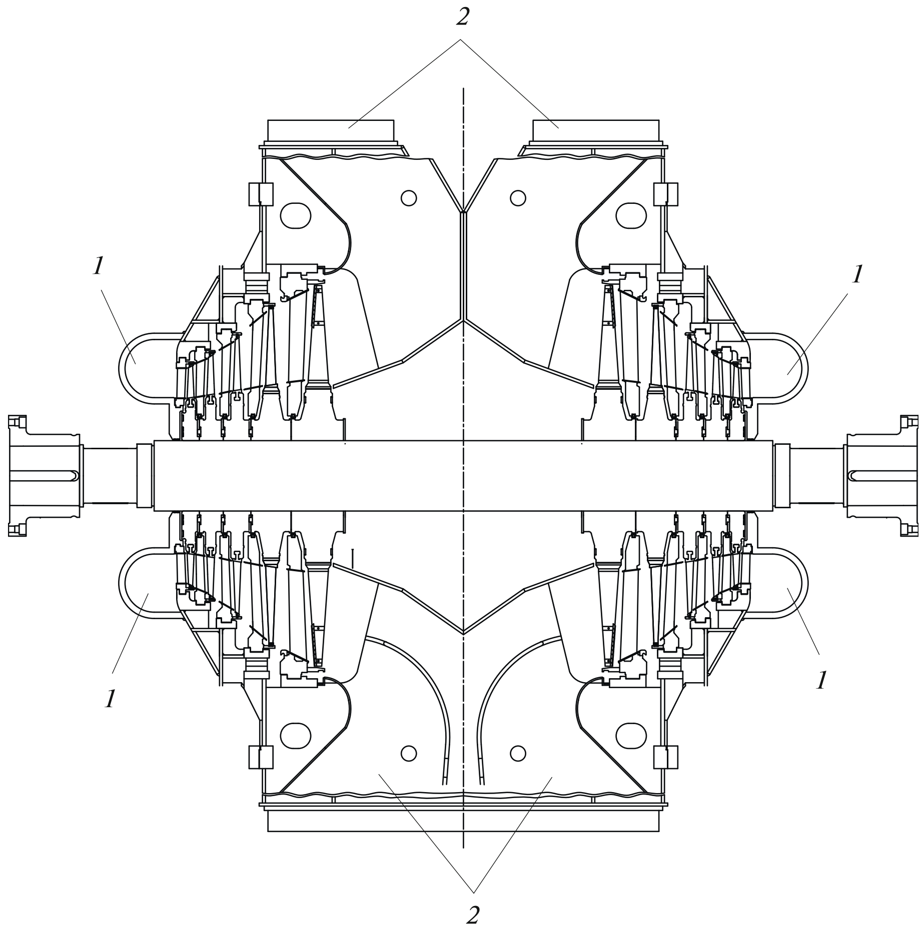

Fig. 1. Design of a counter-current flow LPC: *1, 2*—steam inlet and outlet.

In addition, an LPC with a reduced axial exhaust size was designed, which allows a reduction in the axial external dimensions of the LPC from 8010 to 7470 mm without impairing its efficiency [3]. Alongside the classical design of the LPC with divergent steam flows, a design with counter-current flows is proposed (Fig. 1). The main advantage of such a design is the absence of the end vacuum—atmosphere seals due to which the economic efficiency of the LPC is enhanced, its external axial dimensions are reduced, and the seal design is simplified [4].

Moreover, under low-consumption conditions, the counter-current flows formed in the exhaust tube do not get into the flow path. With the LPC of such a design, there is a probability of increasing the overall hydraulic resistance even under the exhausts with reduced overall dimensions, which necessitates studies on the impact of this resistance on the pressure losses.

However, with the LPC of such a design, the problem of ensuring the tightness of the horizontal parting remains unsolved as well. To solve this, renewable seals for flange joints were devised and tested on test benches at TsKTI. The basic component of the seals is a monocrystalline sealing material based on copper—nickel—aluminum (Cu—Ni—Al) alloy or nitinol (Ti + Ni) that exhibits the shape memory effect (SME). The shape-memory material is inserted into a groove in the flange in the form of a strand (Fig. 2). Upon heating, the material (the strand diameter) increases in volume and the strand is tightly pressed against the groove walls, thus reliably tightening the joint. During the subsequent cooling, the material does not change its shape any more, thus maintaining the tightness of the connection. The stresses that occur in the component from the shape-memory material under heating depend on the difference between the initial and final temperatures. Thus, the seals of this type will be effective in both the LPCs and IPCs [5].

Further, various shafting configuration variants were studied and comparative calculations of the dynamic behavior of the rotors and bearings were performed for each rotor with the conventional double support and a so-called scanty support when every two rotors rest on three supports. This enables a considerable reduction in the axial dimension of the turbine, which may prove to be essential in case if the need to

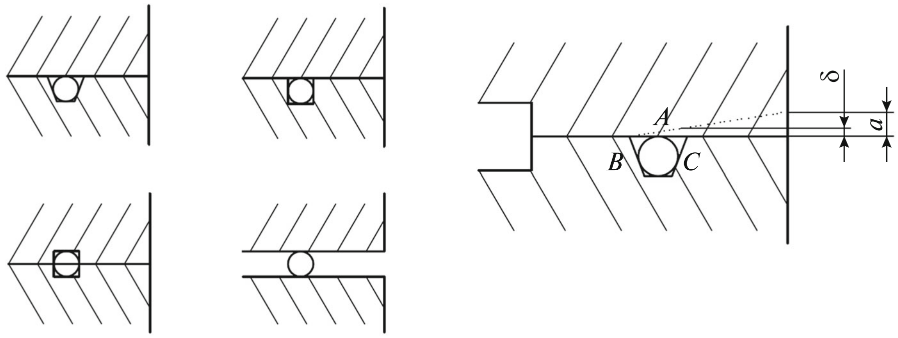

Fig. 2. Examples of designs of seals for steam-turbine flange joints: *A*, *B*, *C*—edges of the groove profile on the flange; *a*—possible opening of the flanges; δ—overlapping of the opening by the sealing material.

install the turbine on old foundations should arise when replacing the capacities or when the design requires arranging the assemblies crosswise, which considerably decreases the costs of the building and the crane facilities. For example, the K-1000-60/3000 plant installed at the Bushehr NPP had to be placed on the old foundation with smaller axial dimensions, which necessitated the use of the scanty-support structure.

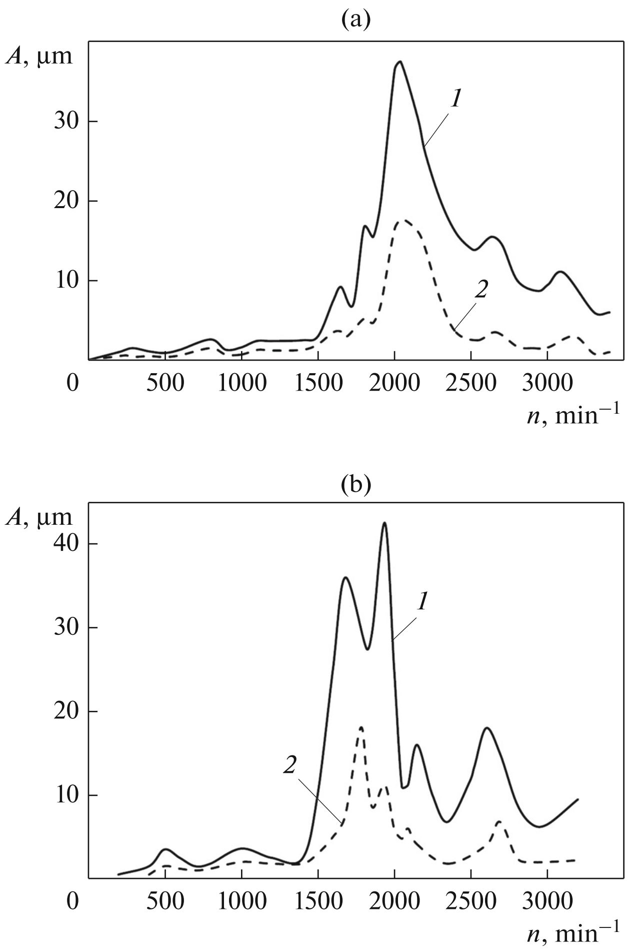

Fig. 3. Calculated dependence of the oscillation amplitude of shafting *A* on speed *n*: (a) front IPR support, (b) rear LPR1 support; *1, 2*—scanty- and multisupport shafting designs.

In Fig. 3, comparative amplitude-frequency characteristics are presented for the leading support of the intermediate-pressure rotor (IPR) and the tail support of the low-pressure rotor (LPR1) in the classical multi-support and potential scanty-support arrangement. From the figure, one can see that the scanty-support structure requires more thorough balancing of the shafting, which, however, is not a serious obstacle to the use of this structure [6], including the new-generation turbines for ultra-supercritical steam parameters.

The increase in the live steam parameters, particularly, the temperature, necessitates new high-chromium steel grades. In collaboration with the All-Russia Thermal Engineering Research Institute (VTI), the following materials were proposed:

Steam pipes and manifolds	DI-756, DI-82, DI-184
Turbine rotors	EP-291, EI-756, EI-802
Steam inlet and fittings	15Cr11MnPBL
Fixings	EI-993

Therefore, it seems to be reasonable to work out the measurements to reduce the temperature of the rotor metal that will allow the use of the known and already tested steel grades, e.g., P-2M (25Cr1Mn1P).

At TsKTI, systems for forced steam cooling of the rotors and diaphragms [7, 8] were developed and successfully used on approximately 70 energy-generating plants. Further, modernized FSCSs were developed for the ultra-supercritical steam parameter turbines; in these systems, the area of the cooling surface is increased and the cooling intensity is enhanced. The scheme to feed the cooling steam into the HPC and IPC is described in [8]. The distribution of the cooling steam flows in the IPC flow path is arranged in such a way that the cooling affects the largest possible area of the surface from the seal chamber and the dummy cyl-

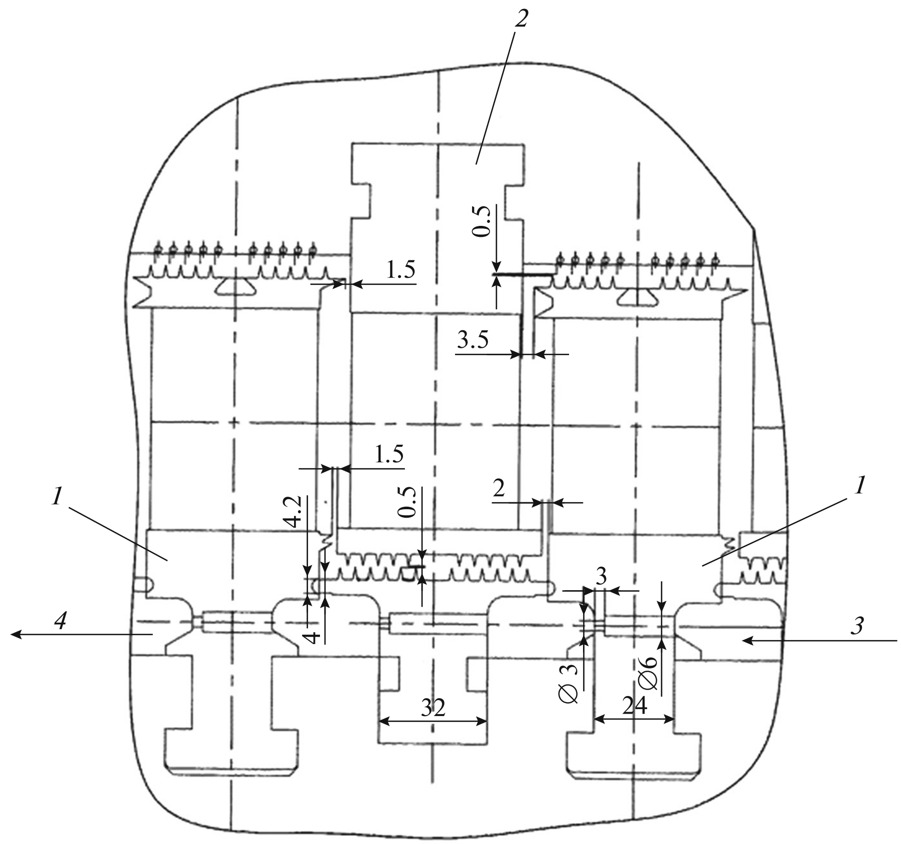

Fig. 4. Cooling circuit for the HPC stages: *1*—working blades; *2*—diaphragm; *3, 4*—cooling steam inlet and outlet.

inder through the entire rotor length. This design uses the properties of the reactive blading, namely, the pressure difference on the working blades under the action of which the cooling steam flows along the rotor, thus decreasing the temperature of the rotor surface and the blade roots by 50–70°C, which decreases the creep rate by 3–5 times.

The design of the cooling steam feed system is depicted in more detail in Fig. 4, from which one can see that the diaphragm seal chambers are also cooled sufficiently intensively, which prevents their warping. As a result, the leakage in the seals and partings are reduced and the losses of the steam consumed for cooling are compensated for. As numerous studies have shown, the efficiency of the turbine does not decrease, which allows neglecting the influence of the cooling system on the economic characteristics.

In Fig. 5, the calculated values of the temperature fields of the K-660-28 turbine rotor manufactured by LMZ are shown under the rated load with an integrated FSCS and without it.

One of the extremely serious problems that arises in the course of developing high-potential turbine is ensuring the vibration stability of the rotors that rotate under nonconservative circulatory forces of the lubricant layer and aerodynamic forces that develop in the flow path. The action of the above forces may, with a high probability, cause the development of the asynchronous precession of the rotor at a frequency close to 50% of the operating speed or a frequency that coincides with the first critical rpm of the rotor, the so-called low-frequency vibration (LFV).

The basic measures to suppress the LFV are as follows [9, 10]:

(1) the use of tilted pad bearings in which the circulatory forces that develop in the lubricant layer are minimal;

(2) a "rigid" rotor, i.e., a rotor whose first critical rpm is significantly higher than 50% of the rated speed or exceeds the latter; and

(3) creation of shroud and diaphragm seals that do not facilitate the development of considerable circulatory forces in the flow path, predominantly in the chambers of the above seals.

Numerous studies conducted by various authors, including specialists of TsKTI, showed that the circulatory excitation forces that develop in the seals are proportional to the dynamic change in the steam flow rate through the seals, which depends on the dynamic change in the gaps between the stationary and rotating surfaces. This is the so-called "criticality" of the gaps'

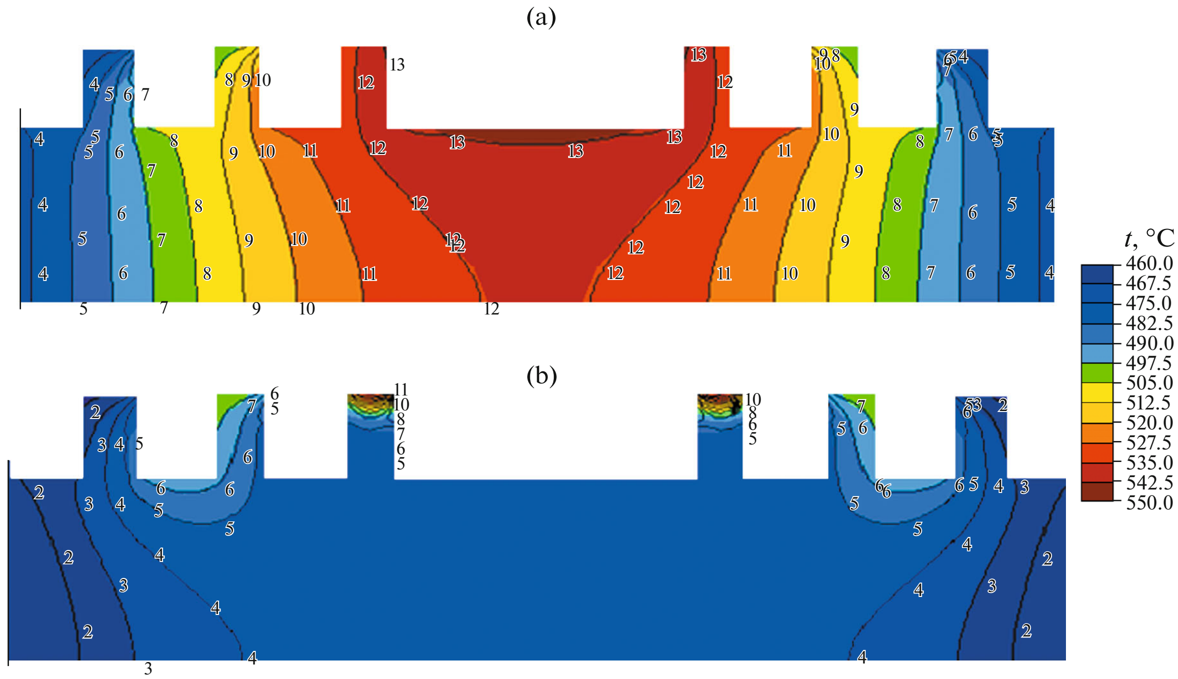

Fig. 5. Temperature fields of the intermediate-pressure rotor of the K-660-26.5 turbine by LMZ at the rated load (a) without the cooling system and (b) with the integrated FSCS.

influence on the steam flow rate. For example, in classical shroud seals, the 10% increase in the clearance results in an increase in the steam flow rate through the seals—depending on the design of the seal—by 5–20%.

The aim of the works conducted at TsKTI was to decrease the influence of the change in the clearances in the seals on the steam flow rate through the latter, i.e., on the flow coefficient [11].

In Fig. 6, the dependence of flow coefficient α_f on the reduced radial clearance δ/h—here, δ is the radial clearance and h is the ridge height—is shown. It can be seen from the figure that, at δ/h = 0.75–1.1, the increase in the clearance leads even to a reduction in the steam flow rather than to an increase, which does not contribute to the development of the circulatory excitation forces, i.e., either the sealing is neutral or it facilitates the aerodynamic stability of the rotating rotor. This somewhat paradoxical result was obtained in the course of multiple experimental studies with different ridge height to chamber width ratios c and can be explained by formation of parasitic eddies in the space between the ridges that block the flow through the seal. Such a seal design can be recommended for use in the assemblies where the LFV may occur.

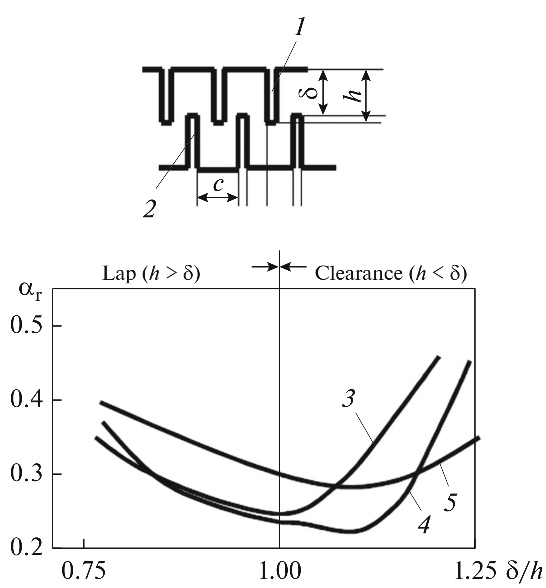

Fig. 6. Dependence of the steam flow coefficient through the seals with axial sealing clearances on the δ/h ratio: *1*, *2*—ridges on the stator and rotor seal parts; h/c: *3*—1.25; *4*—1.0; *5*—0.75.

There is also a problem of longevity of the seals related to the abrasion of the seal ridges under rubbing during the start-ups and stops of the plant when it runs through the critical speeds of the rotors, which results in increasing leakages. To resolve the problem, seals with variable radial clearances, the so-called two-position seals, were tested at TsKTI; the design of these seals is shown in Fig. 7. Such seals are used in the world practice [12]; researchers of TsKTI, however, tested the existing domestic seals adapted to the operation in the two-position mode.

Under the action of coil or plate spring *1*, seal segment *2* is forced back from rotor *3*; the plant is started

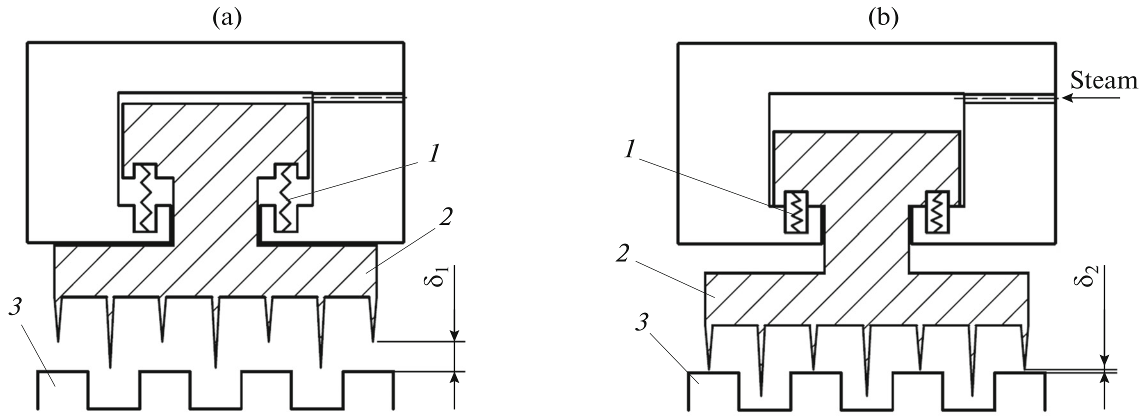

Fig. 7. Two-position seals with the (a) maximum and (b) minimum clearances.

and loaded at large radial clearances δ_1. Upon reaching the steady-state conditions, the steam is fed beneath the segment shoulders, the segment is pushed towards the rotor overcoming the spring pressure, and the seal operates at minimum clearances δ_2 under the design conditions [11]. The steam is fed into the segment chamber owing to the pressure difference on the diaphragm seal whose difference hardly exists under no-load conditions and increases as the load is taken. The chamber in front of the seal is always connected to the seal segment chamber. The efficiency is affected in this case to a negligible degree.

One of the problems of creating new-generation high-powered turbine plants is the reliability of pedestal and thrust bearings. The babbit bearing lining presently used in turbine construction is on the verge of exhausting its capabilities, because the mechanical properties of a babbit decrease at a temperature of 100°C and above and the material does not ensure the reliable and safe operation of the shafting to the right degree. Moreover, a babbit is a good electric current conductor and is frequently damaged by electrolytic erosion.

It should be taken into consideration that, as a result of thermal strains of the support system, the load on the pedestal bearings of high-potential plants may reach ±100% of the rated load per support. The load from the axial force on the thrust bearing is also variable, since this force has different values under different conditions, changes greatly upon load droppings, and cannot be calculated exactly even under rated conditions. These circumstances necessitate the bearings with the bearing surface of a highly reliable material, i.e., capable of functioning within a wider range of unit pressures.

Therefore, bearings were designed at TsKTI that use the antifrictional heat-resistant carbon-filled plastic grade UPFS devised by specialists of Prometei Company [13]. A pedestal bearing with this material was tested on the TsKTI test bench and the turbine in operation at the TsKTI thermal power station. The results of the tests proved to be most positive. The new material reliably withstands a unit pressure of 3.0 MPa and over with a reduced temperature of the bearing surface and lower power losses.

In Fig. 8, the distribution profile of temperature t over the bearing surfaces from babbit and carbon-fiber-reinforced-composite is shown. Comparison of the curves shows that the temperature of the carbon-fiber composite surface is considerably lower than that of the babbit surface at a higher unit pressure p without any distinct peak temperature point in the maximum pressure area. We should also note that the carbon composite is a dielectric that surpasses a babbit in its strength characteristics and can be recommended for use in the turbine construction at present and in the future.

The works conducted at TsKTI to reduce the erosion of the final LPC stages caused by impingements of the moisture droplets on the working blade surface are also of a certain practical interest. The droplets formed upon separation of the water film from the trailing edges of the guide blades impose the greatest danger. To fight this phenomenon, it is suggested to heat the guide blades by the steam fed into the inside of the blades from one of the previous stages or from the chamber upstream from the stage in question [14]. The influence of the steam extraction to heat the blades on the efficiency is negligible; it is compensated for by the increased economic effect. The works were conducted on the K-300-23.5 turbine at Stavropol State District Power Station.

In Fig. 9, the results of measuring the wear of the leading edges of the working blades are presented at the stages without heating and the stages with heated blades. The heating of the guide blades resulted in a considerable reduction of the erosion—by approximately 60% on the periphery and 25% in the dampening wire area. Due to the preserved blade cross-section and a less rough surface, the reliability and efficiency of the plant is ensured.

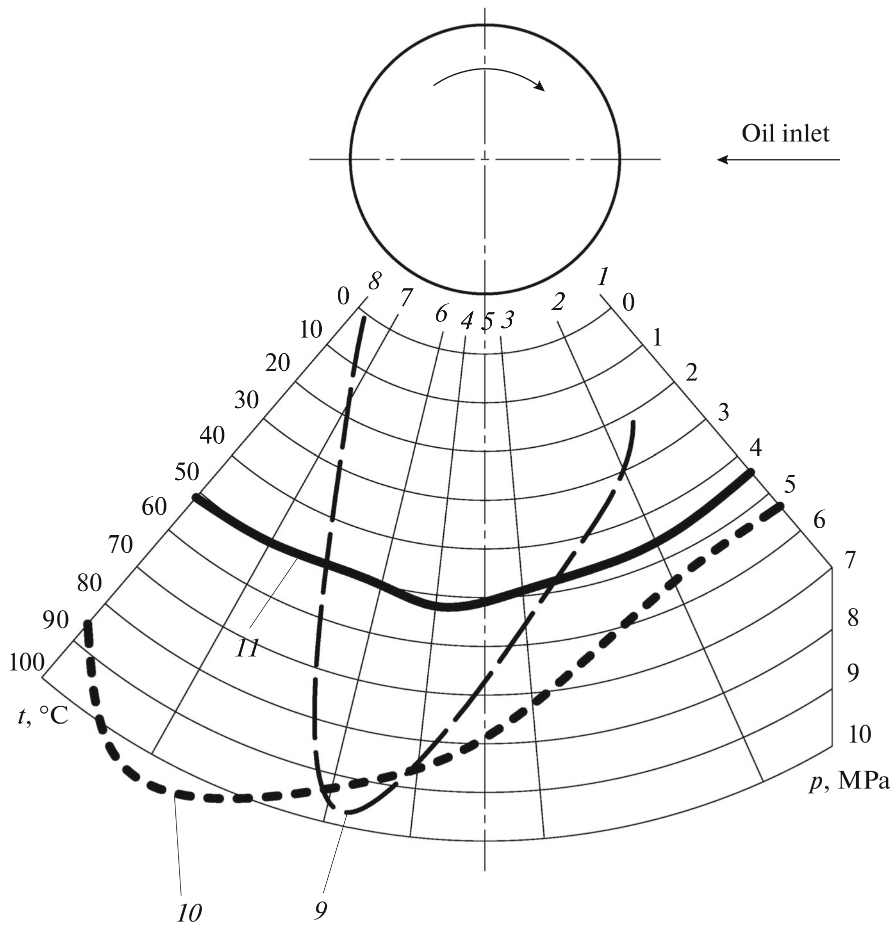

Fig. 8. Temperature distribution over the bearing surface of a carbon-fiber reinforced composite with the babbit bearing lining under a unit load on the bearing of 3.0 MPa: *1–8*—measurement point numbers; *9*—pressure in the lubricant layer; *10*—babbit (B-83) lining; and *11*—carbon-fiber composite.

It is obvious that the new-generation turbines must be equipped with up-to-date measuring and analyzing instrumentation based on the latest advancements in electronics, electrical engineering, and information technologies. At TsKTI, the main attention was paid to the development of the instrumentation for measuring the mechanical characteristics. The work resulted in the development of contactless eddy-current vibration transducers for the rotor journals, a phase transducer, a transducer to control the expansion of the rotor with respect to the stator, the VIDAS software package that uses the instruments of other manufacturers, viz., YaTsRF-VNIIEF, OOO NPO Vibrobit, Bentley Nevada, and Vibro-Meter [15], installed on numerous plants. It is being planned to develop in the near future all engineering means for measuring and analyzing the mechanical characteristics.

Software is used to monitor and automatically diagnose the condition of the equipment, to archive the data, and to perform the balancing calculations [16]. The latest developments are the assessment of the mutual support displacements caused by seasonal and operation factors calculating the reactions, stresses in the rotor journals, and compensating half-coupling alignment as well as the algorithm for diagnosing cracks in the rotor. The algorithm is based on the use of contactless eddy-current transducers with signal isolation by the reverse component during the rotation of the rotor on the shaft-turning device [17]. With a crack present, the readings of the vertically installed transducers begin to significantly exceed the readings of the horizontally placed transducers—the impact of the proper rotor weight on the behavior of the crack under rotation—which is one of the diagnostic signs of initiation and development of a transverse crack.

At TsKTI, the SKALA system was developed and repeatedly used; the system is intended for measuring static and dynamic displacements of the final LPC stage blades in order to diagnose potential disturbances in the operation of the blading and prevent emergencies caused by failure in blade ring components, such as blades, damping wire, and shrouding.

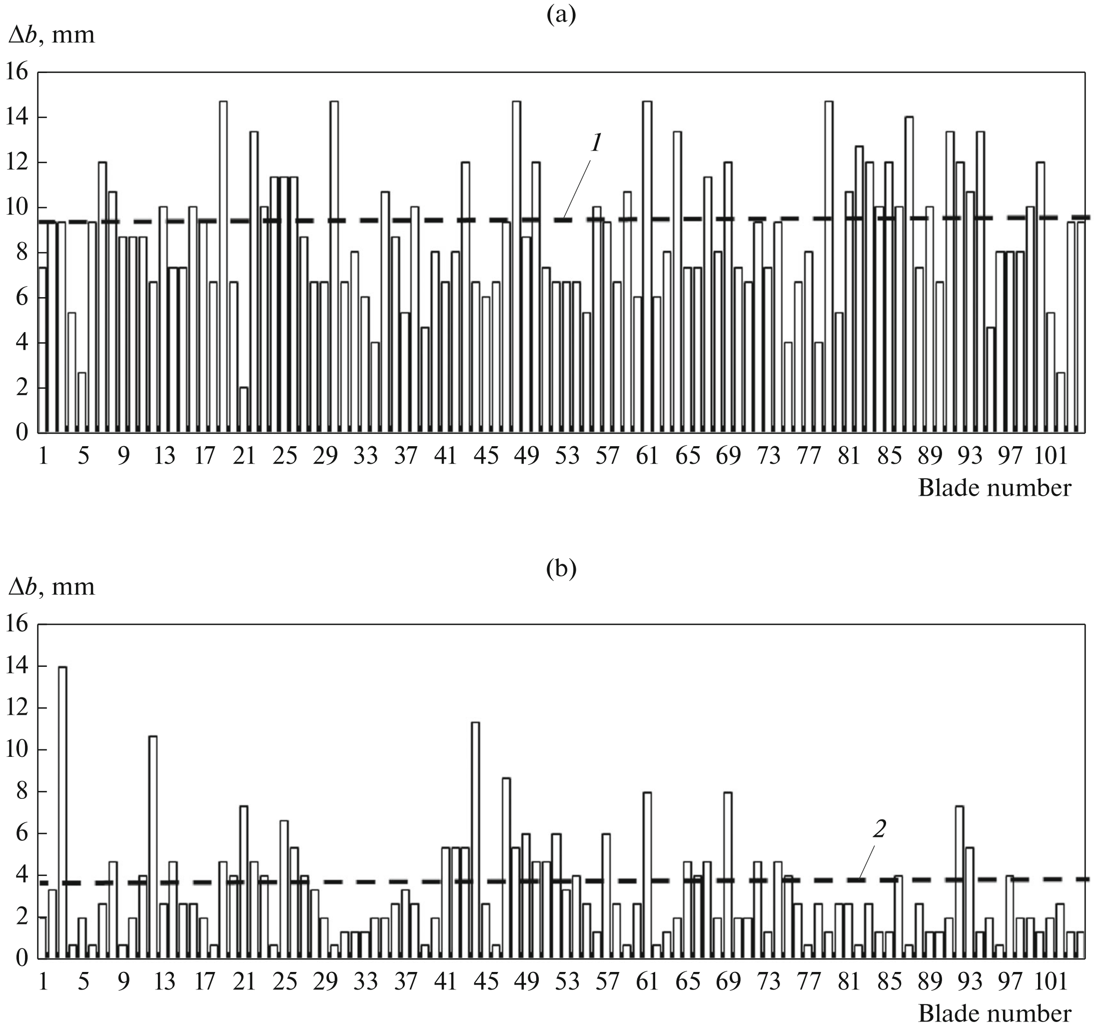

Fig. 9. Reduction in chord Δb of the working blades as a result of erosion on the periphery (a) without heating the guide blades and (b) with heated guide blades: Δb_{av} (mm): *1*—9.5; *2*—3.5.

The entire hardware of the SKALA system based on the discrete phase method, as well as the software, was developed by specialists of TsKTI [17].

In addition to diagnosis systems based on measuring and analyzing the mechanical characteristics, the MENTOR system was developed and applied at TsKTI to analyze thermal stresses in the rotors under transient conditions [16]. On the basis of thermal-engineering parameters, the system computes the current stresses, estimates the lifetime exhaustion, and makes recommendations on the parameter change rates in order to achieve thermal stresses of an optimal level.

For many years, specialists of TsKTI have been working on the solution of the problem of electroerosive damage to the turbine equipment, which becomes especially topical for the ultra-supercritical steam parameter turbines. For this purpose, a system for operational control of the electromagnetic condition of the turbine plant has been developed and produced [18]; it is a hardware–software complex intended for continuous control of the condition of the current-collecting device and resistors of the bearing journal insulation of the insulated bearings and the insulation of the oil films. The use of this system facilitates the normalization of the electromagnetic condition of the turbine plant and considerably reduces the risk of electroerosive damage, such as failure of a babbit, overheating and scoring of the journals, thrust collars and bearing pads, the wear of the seals and the rotor journals under the seals, and the control unit components and allows eliminating the influence of the electroerosive processes on the operation of other diagnosis and control systems.

Naturally, the range of the developments made by TsKTI to create new-generation turbines is not restricted to the assemblies and devices presented in the article; it obviously demonstrates, however, that the work on the concept of future turbines is already bringing tangible benefit, since the already existing achievements can be implemented not only in future projects but also on the plants in operation.

REFERENCES

1. V. E. Mikhailov, L. A. Khomenok, and I. A. Kovalev, "Steam turbines for high-capacity power units," in *Proc. Conf. Prospects of the Development of New Technologies in the Russian Power Industry, Moscow, Oct. 27–28, 2016* (Vseross. Teplotekh. Inst., Moscow, 2016).
2. I. I. Pichugin, L. A. Khomenok, I. A. Kovalev, O. A. Vladimirskii, Yu. Ya. Kachuriner, and I. V. Zaitsev, "The concept of the constructive profile of a new generation turbounit for ultra supercritical steam parameters," Nadezhnost Bezop. Energ., No. 3, 14–17 (2012).
3. V. K. Migai and E. I. Gudkov, *Design and Calculation of Output Diffusors of Turbomachines* (Mashinostroenie, Leningrad, 1981) [in Russian].
4. V. G. Orlik, I. A. Nosovitskii, and L. A. Khomenok, Patent RF Application of I. I. Polzunov Scientific and Development Association on Research and Design of Power Equipment (NPO TsKTI) No. 2016147235/06 (075849) (2016).
5. L. A. Khomenok, A. N. Remezov, I. A. Kovalev, V. S. Shargorodskii, S. Sh. Rozenberg, and V. I. Olimpiev, *Improving the Operation Efficiency of Steam Turbine Units of Thermal and Nuclear Power Plants* (PIPK, St. Petersburg, 2001) [in Russian].
6. V. I. Olimpiev, I. A. Kovalev, D. V. El'kin, S. A. Tikhomirov, and V. D. Gaev, "The issues of ensuring vibroreliability of shaftings of high-power steam turbines with a low-bearing support system," Tr. TsKTI, No. 283, 32–44 (2002).
7. L. P. Safonov, V. S. Shargorodskii, A. N. Kovalenko, L. A. Khomenok, S. Sh. Rozenberg, V. A. Shilin, A. P. Ogurtsov, N. N. Gudkov, and V. N. Mitin, "Introduction of forced cooling systems for elements of 200–800 MW turbines," Tyazh. Mashinostr., No. 1, 27–34 (1996).
8. L. A. Khomenok, V. V. Bozhko, I. S. Leonova, I. V. Zaitsev, and D. V. Prokhorov, "Extending the service life of steam turbines using the systems for cooling the thermally stressed parts of high- and intermediate-pressure cylinders," Therm. Eng. **59**, 199–203 (2012).
9. I. A. Kovalev, Candidate's Dissertation in Engineering (I. I. Polzunov Central Boiler and Turbine Inst., Leningrad, 1978).
10. A. G. Kostyuk, *Dynamics and Strength of Turbomachines: Textbook for Higher Education Institutions* (Mosk. Energ. Inst., Moscow, 2007) [in Russian].
11. V. G. Orlik, "Improvement of labyrinth seals in the process of the development of turbine construction," Elektr. Stn., No. 10, 15–25 (2007).
12. *Fundamentals of Modern Power Engineering*, Ed. by A. D. Trukhnii (Mosk. Energ. Inst., Moscow, 2008), Vol. 1 [in Russian].
13. V. E. Bakhareva, I. A. Kovalev, I. V. Lishevich, Yu. I. Monogarov, A. S. Sargsyan, D. G. Esperov, and S. S. Enrol'd, "Application of heat-resistant antifriction carbon plastics for bearings of steam turbines," Vopr. Mater., No. 4, 171–180 (2012).
14. N. V. Averkina, Yu. Ya. Kachuriner, V. G. Orlik, F. M. Sukharev, and M. A. Filaretov, "Experience of industrial application of guiding blade heating for erosion reduction of wet-steam turbine stages," Elektr. Stn., No. 2, 24–27 (2004).
15. I. A. Kovalev, "Development of the algorythm of functioning and defect detection for an automatic diagnostics system," Tr. TsKTI, No. 273, 27–33 (1992).
16. I. A. Kovalev, N. Yu. Isakov, and V. V. Bozhko, "Comprehensive diagnostics as a necessary tool for ensuring operational reliability of turbine units," Therm. Eng. **59**, 189–194 (2012).
17. I. A. Kovalev, V. G. Rakovskii, N. Yu. Isakov, and A. V. Sandovskii, "Development and improvement of the operating diagnostics systems of NPO CKTI works for turbine of thermal and nuclear power plants," Therm. Eng. **63**, 168–173 (2016). doi 10.1134/S0040601516030046
18. V. D. Kots and L. Z. Khain, "Electroerosive damage of steam turbine elements and methods of its prevention," Tr. TsKTI, No. 283, 198–202 (2001).

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