Proposals for the Modernization of TPP Turbogenerators

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Abstract. *The purpose of this work is to develop proposals for modernizing the design of turbogenerators with a capacity of 200–300 MW in order to increase their power. To increase the power, it is proposed to increase the generator electromagnetic parameters (induction in the magnetic circuit and linear load of the generator). At the same time, the possibility of replacing hydrogen, which cools the generator internal volume, on the air was studied, which is a global trend for turbogenerators of thermal power plants with a capacity of 200–300 MW. A prerequisite was the preservation of the turbogenerator dimensions. The set goals were achieved by changing the geometry of the stator core tooth zone and intensifying cooling. To determine the necessary changes in the tooth zone with a change in the electromagnetic and thermal characteristics of the turbogenerator, a mathematical model was developed that makes it possible to determine by calculation the required dimensions and configuration of the stator slots and gas cooler elements at condition that the permissible thermal characteristics are maintained. The modelling of the stator core was carried out in order to select the dimensions of the slots and channels of axial ventilation for the required mode of the coolers operation, subject to an increase in power.*

Keywords

Power increase, thermal power plant, tooth zone of the stator core, turbogenerator cooling, turbogenerator modernization.

1. Introduction

The global economic crisis has made it impossible to timely replace the power plants electrical equipment, which has worked out the period set by the manufacturer. First of all, this applies to turbogenerators - a complex and very expensive element of the electricity generation cycle. Therefore, in world practice, it is not the replacement, but the modernization of turbogenerators that is carried out. During modernization, their capacity is usually increased by 10–15 %. [1], [2], [3], [4], [5], [6], [7] and [8]. The issues of increasing power were almost always included in the programs for overhaul of turbogenerators. During the global economic crisis, it is necessary to introduce all possible methods to ensure energy saving and increase the volume of electricity production. Therefore, increasing the power of turbogenerators during their repairs and modernization has become mandatory for all world electrical companies, [2], [3], [4], [5], [9], [10], [11], [12], [13], [14] and [15]. As confirmation, we can indicate the results of the turbogenerator TVV-1000-2 modernization installed in the second unit of the Khmelnitsky Nuclear Power Plant (NPP) (Ukraine) carried out in 2021. During the overhaul of the electrical equipment of the unit, which was carried out by Westinghouse, the power of the turbogenerator was increased by 10 %, up to 1100 MW.

Increasing the power of turbogenerators is usually carried out by increasing the electromagnetic parameters (linear load, induction in the air gap and in all elements of the magnetic circuit), which requires additional control of the mutual influence of the proposed changes on the generator characteristics. It is necessary to carry out additional calculations and, first of all, to determine new thermal loads, [14] and [15].

The paper analyses the possibility of simultaneously increasing the power of a turbogenerator and replacing the cooling medium in its internal volume (replacing hydrogen with air), provided that such modernization is carried out at a block of Thermal Power Plants (TPP) without the involvement of specialists from the manufacturer. Such modernization is carried out with the preservation of the overall, installation and connecting dimensions of the generator; and allows you to keep the foundation and auxiliary systems unchanged: a system for supplying oil to bearings and water to coolers, gas pipelines. It should be noted that the replacement of the cooling medium (hydrogen with air) is a global trend for turbogenerators with a capacity of 200–300 MW, [1] and [2]. However, in world practice there are different opinions on the advisability of further operation of thermal power plants, on the advisability of extending their service life and on the construction of new units. There is an opinion about the need to close them due to the negative impact on the environment [4], [5] and [6]. Therefore, in order to establish the relevance of the topic of work, the relevance of developing proposals for the modernization of turbogenerators of the specified power range, we first of all conducted a study of the state and prospects for the development of thermal power industry.

The need to continue the operation of TPPs, that is, the need to modernize the turbogenerators of TPPs, can be argued:

- The construction of new Nuclear Power Plant (NPP) units, as the main sources of electricity in the near future, is a long process (7–8 years for the construction of one unit). During this period, it is necessary to maintain in working order turbogenerators that are in operation at nuclear power plants and thermal power plants.
- Modern energy networks need constant regulation of the active and reactive power balance. And only turbogenerators of TPPs can adequately maintain the parameters of the power system during peaks and dips in energy consumption. Regulation and maintenance of power system parameters by changing the operating modes of nuclear power plant generators is unacceptable for safety reasons, and using hydroelectric power plant generators is not enough due to their low power and therefore a small control range, [2], [4] and [8].
- Currently, the generators of thermal power plants operate in electrical networks, which include generators that operate from renewable energy sources. For example, wind turbine generators. These energy sources have unstable parameters, which requires constant regulation of the overall balance in the energy system. Such regulation is possible by changing the operating modes of TPP generators, [6], [9], [12] and [13].

The issues of modernization of TPP turbogenerators in the power range of 200–500 MW are dealt with by scientists from different countries. The main directions of these studies:

- Increasing the power of turbogenerators during their manufacture and modernization, [2], [4], [6] and [11].
- Improvement of systems and methods for cooling turbogenerators, [12], [15], [21], [22], [23] and [32].
- Condition monitoring and timely diagnostics of defects in turbogenerators, [14], [24], [30] and [31].
- Analysis of the destruction causes of the cores and insulation of the turbogenerators stator windings, [4], [11], [33], [34], [35], [36], [37], [38], [39] and [40].
- Condition monitoring and modernization of turbogenerators excitation systems, [7], [10], [29] and [30].
- Vibration control of the turbogenerators state [4] and [22].
- Improvement of the mathematical apparatus for the design of electrical machines, [41] and [42].

After the research, the goal of the work was formulated: the development of proposals for increasing the power of turbogenerators while replacing hydrogen, which cools the internal volume of the generator, with air while maintaining the overall, installation and connection dimensions.

Such a combination of issues that are simultaneously proposed to be resolved in the process of modernizing the turbogenerator has not been considered before.

2. Main Stages and Questions of the Conducted Research

2.1. The Following Was Done in the Work:

- The need for further operation of the TPP was confirmed; their role in the energy system is determined. This determines the relevance of carrying out work on the modernization of TPP turbogenerators.

- The range of possible increase in the turbogenerator power, which operate at TPP units, is determined, provided that the overall and connecting dimensions are maintained.
- Based on the results of assessing the thermal state of the turbogenerator modernized design, the power of which was increased by 20 %, the possibility of replacing the cooling medium of the generator internal volume (hydrogen with air) was shown.
- To determine the necessary changes in the geometric dimensions of the stator core toothed zone of an air-cooled turbogenerator and with the increased power, a model for calculating the core has been developed.

2.2. The Role and Prospects for the Use of TPP Turbogenerators in the General Scheme for Generating Electricity

The assessment of prospects and the choice of directions for the electric power industry development should be carried out in the context of the actual socio-economic and technical level of each specific country development. But it is possible to identify areas common to all countries. This is a search for new, environmentally friendly sources and technologies for generating electricity; a continuous increase in the volume of electricity generation, an increase in the efficiency of its transmission, distribution and consumption, a reduction in losses at all these stages, [3] and [13]. For the sustainable development of the electric power industry and ensuring the energy security of each country, it is necessary to build new units and improve the electrical equipment of nuclear power plants that are in operation, [4]. But in order to ensure a sustainable energy supply to consumers until the launch of new NPP units, until a sufficient number of high-capacity plants are created that will operate from renewable energy sources, or until new types of energy sources (for example, thermonuclear reactors) are discovered, it is necessary keep in working order TPP turbogenerators, [2], [3], [12] and [17]. The increase in the number and capacity of power plants that operate from renewable energy sources and are included in the overall energy system has raised new questions for ensuring its stability. It also requires constant regulation of the active and reactive energy balance in the network due to the uneven consumption of electricity. It has been proven that the best solution for this is the transfer TPP turbogenerators to non-nominal operating modes, [20].

The presence of coal deposits and a developed coal industry is also the reason for the continued use of coal TPPs. And, despite the understanding that coal TPPs cause the greatest harm to the environment, their operation continues all over the world and new power units are being built. Data on the new blocks of TPPs construction and on the electricity generation at TPPs in some countries are presented in Tab. 1.

Tab. 1: Examples of construction and use of TPPs in countries with their own coal resources.

Part of electricity received at TPPs in total electricity generation, in % (2020 data)	
Germany	39
Poland	92
China	67
Construction / Preparation for construction of new TPP units, MW	
Germany	5372 / 1580
Poland	3785 / 8785
China	115 500 / 45 200

For example, in the fall of 2020, in the energy balance of Germany, solar and wind stations accounted for almost 33 % of the total electricity generation, but in winter it was reduced to 20 % and the electricity deficit was provided by TPPs. A similar situation was noted in other European countries. In Bulgaria, Romania, Hungary, Czech Republic, Greece, Poland, Sweden in 2019–2022 the main part of the electricity was generated by gas and coal thermal power plants.

Therefore, in the last 2–3 years, a new understanding of the prospects for the operation of TPPs has appeared, and, accordingly, the need to improve TPP turbogenerators has been confirmed.

2.3. Identification of Components and Parts of Turbogenerators that will be Affected by an Increase in Power and Replacement of the Cooling Gas

When upgrading, it is necessary to evaluate the impact of the changes made on all components and elements of the turbogenerator, on maintaining their sufficient mechanical reliability and temperature indicators within the permitted limits. Let us consider the components and parts of turbogenerators, which will primarily be affected by an increase in electromagnetic loads and a decrease in the thermal performance of the cooling medium (heat transfer coefficient, specific density of the coolant, overall heat transfer coefficient), and formulate proposals to reduce the impact of changes:

- When increasing the power of a turbogenerator due to an increase in electromagnetic loads, new insulating materials should be used for the turbogenerators stators windings. Stator windings operate under severe conditions: high temperatures, mechanical stress, constant vibration activity, variable loads. Therefore, the issues of reliability, durability, reduction of insulation thickness are constantly being investigated. For example, it can be proposed to replace continuous compounded insulation with thermoset insulation made of glass-mica or glass-mica materials with epoxy binders. type "Monolith-1" or "Monolith-2M" (with additional crimping of the stator winding groove part). These types of thermoset insulation compared to mica thermoplastic compounded insulation have 2 times higher dielectric strength than thermoplastic. It reaches 30–34 kV$\cdot$ mm^{-1}. Also, they have higher moisture and oil resistance; they have smaller dielectric loss angle ($\mathrm{tg}\,\delta$) and better thermal conductivity, [30] and [36].

- Assessment of possible accidents of turbogenerators in world practice is carried out on the basis of statistical data on the operation of electrical equipment of power systems. In order to be ready to prevent failures and reduce the risks associated with defects in turbogenerators, it is necessary to know which of them are most common; a numerical indicator is needed, for example, the value of the specific downtime of the Turbogenerator (TG) (hours/(generator per year)). All researchers note that the most important (in terms of the risk of consequences) should include the destruction of the end parts of the stator core laminated packages, the destruction of bandages and the insulation of the stator windings frontal parts, [22], [25], [27] and [28].

  The destruction of lamellar stator cores usually occurs when the pressure inside the core decreases, when the its compression level decreases. In this case, the vibration increases, the wedges in the grooves are displaced, the shroud rings and the insulation of the windings (especially at the exit points from the grooves) are destroyed. Such problems are most often noted in two-pole generators, which include the turbogenerators studied in this work. To reduce the probability of destruction of both extreme and middle packages of core, one should calculate the changes interaction forces between steel sheets in the laminated core. Studies have shown that an increase in electromagnetic loads reduces the pressing force in the cores, [27]. Properly selected pressing force reduces the relative sliding of the sheets in the core. It has been established that it is necessary to choose the pressure in the laminated packs $p_0 \geq 1.5$ MPa for modernized turbogenerators and use gluing of the extreme core packages.

- Special attention was paid to the value of the induction in the tooth of the stator core. Basically, damage to the teeth of the outer packages was noted. But sometimes the damaged teeth were located not in the extreme stator packages, but in the middle ones. It has been established that due to a decrease in pressing pressure, the teeth of the laminated core are separated, and mutual displacements of individual steel sheets appear. These displacements depend on the magnitude of the magnetic induction in the teeth, which we propose to increase. Displacements increase fretting corrosion and internal friction, which damages the insulation of the sheets and can cause a "fire" in the steel. This is a serious accident; and it will be possible to restore the stator core only at the manufacturer's factory, [27].

- It must be taken into account that during operation the thermophysical properties of the cooling medium (air, hydrogen), the electrical properties of insulating materials, the properties of copper windings and steel of the magnetic core change due to "aging", [22], [23], [24] and [27]. Increasing electromagnetic loads can accelerate all these processes. Therefore, when determining the permissible limit for increasing the turbogenerator power, the main attention was paid to the stator core for which the largest number of failures during operation was noted. In the work, modelling of the stator core tooth zone was carried out, [12], [26] and [27].

2.4. Modelling and Selection of the Geometry of the Tooth Zone of the Stator Core, Taking Into Account the Features of the Cooling System

The use of modern models makes it possible to consider changes in parameters, connections and functional interaction of different elements and nodes; allows you to choose the best design solutions. At the same time, due to repeated comparison of possible options, it is possible to minimize the number of errors in the choice of solutions. This approach eliminates the need to create expensive physical models.

On Fig. 1 shows a sketch of the toothed zone of the basic turbogenerator laminated stator core.

When performing calculations, we used the data of turbogenerators TGV-200-2U3 with a capacity of 200 MW manufactured by the Electrotyazhmash plant (Kharkov, Ukraine), with hydrogen cooling of the internal volume of the machine, and with stator winding water cooling. Core segment material - steel M250-50A (according to EN-10106), stator winding - wire PSD-1 (TU 302.08.003), insulation of heat resistance class.

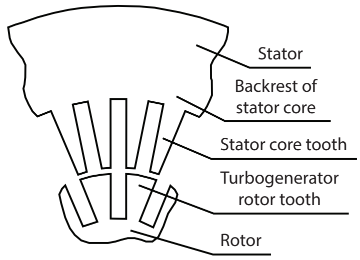


Fig. 1: Sketch of the toothed part of the three-phase synchronous turbogenerator magnetic circuit.

The process of modelling the stator core of a turbogenerator included three stages:

- Calculation of the geometry of the gas cooler tubes.
- Calculation of electromagnetic parameters with the introduction of parameters of the cooling medium (hydrogen and air).
- Determination of the thermal state of the TG with an increase in power and replacement of the cooling medium.

Cooling in the turbogenerator rotor - multi-jet, self-ventilation, direct, with the intake and emission of hydrogen into compartments alternating along the length. The rotor ventilation is not coordinated with the stator ventilation scheme. In the latest years for turbogenerators of the entire power range, a two-jet ventilation system is used, in which hydrogen enters the corners of the turns of the rotor winding and branches into two streams: one goes in the axial direction and is ejected into the gap through the holes at the ends of the slot wedges; the other enters the space under the bandage and is ejected through the holes into the gap. Hydrogen in gas coolers is cooled in a two-way scheme, which complicates the sealing system. The sealant is used to prevent hot hydrogen from flowing into the cold hydrogen collecting area. To increase the reliability of water supply to the stator groove, the ends of the winding must be hermetically connected to the bushings and to the tips of the fluoroplastic hoses. To increase the reliability of this connection, it is possible to propose the use of a "cone-sphere" design, which does not require rubber hoses for sealing, Fig. 2.

On Fig. 3 is shown a multi-jet radial cooling system for the rotor winding: a scheme for supplying hydrogen to the rotor slots in the axial (Fig. 3(a)) direction and a scheme of the gas movement (hydrogen or air) in the rotor array, Fig. 3(b).

To build a mathematical model for choosing a sufficient heat removal area, we use the fluid flow continuity equation (Bernoulli equation) and the equation of steady water movement.

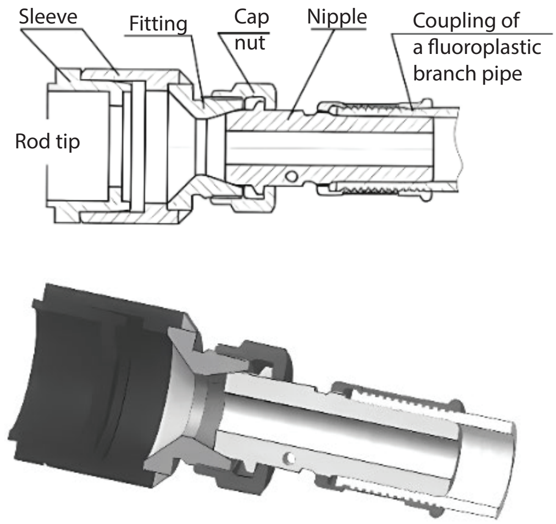
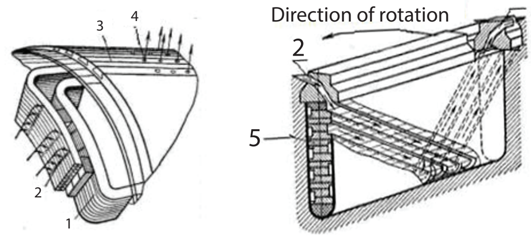


Fig. 2: Connection of the "cone-sphere" type for the heads of the rods of the stator winding and the tips of the fluoroplastic hoses.



(a) (b)

Fig. 3: Schemes of hydrogen supply to the conductors of the rotor winding.

To take into account the contamination of the surface (internal and external), the heat transfer coefficient was calculated using the equation of thermal resistance along the path of the heat flow, provided that $R_1/R_2 \geq 2$, where R_1 and R_2 - is outer and inner radii of the cooler tube, Fig. 4.

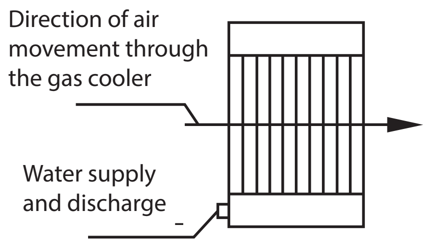


Fig. 4: Gas cooler with one-way movement of gas.

For heat exchanger tubes:

$$\frac{1}{k} = \frac{1}{\alpha_1} + \frac{\delta_{wal}}{\lambda_{wal}} + \gamma_1 + \gamma_2 + \frac{1}{\alpha_2}, \quad (1)$$

where α_1 and α_2 - heat transfer coefficients from the side of heat carriers (water and gas, respectively), W·m^{-2}; δ_{wal} - tube wall thickness, m; λ_{wal} - thermal conductivity of the tube wall material (aluminium or cupronickel), W·(m·°C)$^{-1}$; γ_1 and γ_2 - thermal resistance of the inner and outer layers of the heat exchanger tube (taking into account pollution), W·(m·°C)$^{-1}$.

All the coefficients that are listed here and below were selected in reference books depending on the operating temperature range, on the flow regime, the type of liquid in the cooler and its temperature, the hydraulic resistance of pipelines, and the roughness of the inner surface of the tubes. For our calculations, this range was determined +(60–200) °C and considered the water flow in the cooling system to be laminar.

We write down the volumetric flow rates of the coolant using the flux continuity equation, m^3 · s^{-1}:

$$V = \frac{G}{\rho} \cdot f \cdot V, \quad (2)$$

where V - coolant velocity in the tubes at the cooler inlet, m·s^{-1}; G - mass flow rates of the coolant, kg·s^{-1}; ρ - coolant density, kg·m^{-3}; f - the tube sectional area, m^2.

The tube sectional area $f = \pi \cdot d^2 \cdot 4^{-1}$, where d - hydraulic diameter of the cooling tube with fins, m:

$$d = 1,13 \cdot \sqrt{\frac{G}{\rho \cdot V}}. \quad (3)$$

Heat transfer coefficient for tube with fins:

$$k_T = \left(\frac{1}{\alpha_{int}} \cdot \frac{f_{air}}{f_{water}} + \frac{1}{\alpha_{tak}} \right)^{-1}, \quad (4)$$

where α_{int} - heat transfer coefficient from tube to water, kW·(m ·°C)$^{-1}$. The ratio $1/\alpha_{int}$ is determined by the characteristics of the cooling water and the water flux hydraulic diameter passing through the tube. The execution of tubes and their layout are determined by the technological capabilities of the manufacturer, as well as economic requirements. Therefore, we can assume that for the structures under consideration, the coefficient $1/\alpha_{int}$ = const. α_{tak} - heat transfer coefficient from tube to gas (air), kW·(m ·°C)$^{-1}$. The ratio $1/\alpha_{tak}$ characterizes the geometry and thermal conductivity of the fin's material of cooling tubes; f_{air} - surface area that gives off heat and is washed by air, m^2; f_{water} - surface area that is washed by water, m^2.

The ratio f_{air}/f_{water} determines the ratio of areas washed by gas and water, r.u. In the calculations, we assume that the air parameters in the coolers correspond to normal operating conditions. To improve mechanical reliability with an increase in air speed, we use a variant of a monolithic finning of the tube, instead of a spiral one, which was used in "hydrogen" machines, Fig. 5.

Tube material - cupronickel alloy, profile - tube DKRNM ($d \times 1$ MD 10-70-1), where d is the outer diameter of the cooling tube, mm.

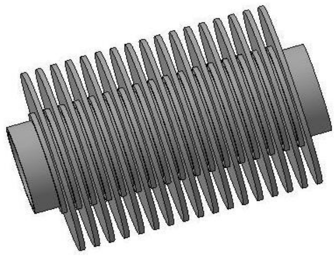


Fig. 5: Cooler tube finning option, which can be recommended when switching to air cooling.

The heat transfer coefficient of such a tube can be calculated, kW·(m ·°C)$^{-1}$:

$$\alpha_{int} = 0.023 \cdot Re_{water}^{0.8} \cdot Pr_{water}^{0.4} \cdot \frac{\lambda_{water}}{d_{int}}, \quad (5)$$

where Re_{water} and Pr_{water} - Reynolds and Prandtl numbers for the water flux, r.u.; λ_{water} - water thermal conductivity coefficient, kW·(m ·°C)$^{-1}$; d_{int} - cooler tube inner diameter, m.

The outer area of the pipe, which is washed by air, and the inner area of the pipes, which are washed by water, can be calculated, m^2:

$$f_{air} = \pi \cdot d_{out} \cdot L_\Sigma, \quad (6)$$

$$f_{water} = \pi \cdot d_{int} L_\Sigma, \quad (7)$$

where d_{out} and d_{int} - outer and inner diameters of the cooler tube, m^2; L_Σ - total length of cooler tubes, m.

The water pressure in the cooler tube can be calculated, Pa:

$$\begin{aligned} P_{water} &\approx \frac{Q_{water} \cdot \rho_{water} \cdot v_{water} \cdot L_\Sigma}{14,7 \cdot 10^2 \cdot d_{int}^4} \cdot 9.8 = \\ &= \frac{0.67 \cdot Q_{water} \cdot \rho_{water} \cdot v_{water} \cdot L_\Sigma}{10^2 \cdot d_{int}^4}, \end{aligned} \quad (8)$$

where Q_{water} - flux rate of water passing through the cooler tube, m^3·s^{-1}; ρ_{water} - water density at the cooler inlet at t_{1water}, kg·m^{-3}; v_{water} - coefficient of kinematic viscosity of water at the inlet to the cooler at a known temperature t_{1water}, m^2·°C^{-1}.

To calculate the area of heat removal depending on the type of refrigerant and load, provided that the dimensions of the cooler are maintained, the fluid flux continuity equation (Bernoulli equation) and the energy balance equation characterizing the steady-state movement of water under the condition of an internal steady-state balance were used:

$$\begin{cases} D = \dfrac{\left[0,41 \cdot \left(\frac{V_B \cdot l_T}{\nu_B}\right)^{0,6} \cdot \left(\dfrac{a_B}{a_T}\right)^{0,33} \cdot \left(\frac{V_B}{a_B}\right)^{0,33}\right]}{\alpha} \cdot C, \\ D = 2 \cdot \sqrt{\dfrac{(a \cdot b) - (1-k) \cdot (a \cdot b)}{n \cdot \pi}}, \\ \dfrac{0,5 \cdot k^{0,75} + k^2}{(k-1)} = \dfrac{\left(Z_{ABC} - 0,0314 \cdot \delta \cdot \dfrac{\rho_B}{2 \cdot (a \cdot b)^2}\right) \cdot (a \cdot b)^2}{0,5 \cdot \rho_B}, \end{cases} \tag{9}$$

where V_B - gas velocity at the cooler inlet, m·s^{-1}; l_T - is the effective length of one cooling tube, m; v_B - is the kinematic viscosity of the gas, m^2·s^{-1}; α_B and α_T are the coefficients of thermal diffusivity of the gas and the material of the cooling tube wall, respectively, which characterize the rate of change (equalization) of the temperature of the cooling medium in thermal processes, m^2·s^{-1}; ρ_B - gas density, kg·m^{-3}; a and b - width and depth of the cooler, m; n is the number of tubes in the cooler; δ is the angle of gas channel rotation in the middle of the cooler, rad. In the calculations, a linear cooler was considered, therefore $\delta = 1$; k is a correction factor that takes into account the number of transverse rows in the cooler. In calculations, the values of the coefficient k were taken equal to $k = 2.0$; ZABC is the aerodynamic resistance of the cooler section with a variable cross section, kg·m^{-2}.

The total area of the cooler tubes heat exchange surface, m_2:

$$S_{rel} = \pi \cdot D \cdot l_T. \tag{10}$$

This made it possible to simplify the final system of equations for calculating the heat removal area:

$$\begin{cases} D = \dfrac{\left[0,41 \cdot \left(\dfrac{V_B \cdot l_T}{\nu_B}\right)^{0.6} \cdot \left(\dfrac{a_B}{a_T}\right)^{0.33} \cdot \left(\dfrac{V_B}{a_B}\right)^{0,33}\right]}{\alpha}, \\ D = 1.6 \cdot \sqrt{\frac{a \cdot b}{n}}, \\ \dfrac{\left(Z_{ABC} - 0.0314 \cdot \dfrac{\rho_B}{2 \cdot (a \cdot b)^2}\right) \cdot (a \cdot b)^2}{0.5 \cdot \rho_B} = 4.84. \end{cases} \tag{11}$$

To assess the thermal state of the core, we calculated the specific heat transfer coefficient in the elements of the core, which was divided into two parts: the tooth part (α_k) and the back of the core (α_s), W·(mm^2·°C)$^{-1}$, (Fig. 1):

$$\alpha_k = \frac{1 + 0.25 \cdot V_k}{450} \cdot 1.3 \cdot \left(p_N^{0.8}\right), \tag{12}$$

where p_N - is the accepted overpressure of the gas in the turbogenerator (for hydrogen, the overpressure was assumed to be 0.05 MPa); V_k - is the gas velocity in the ventilation canals of the stator core tooth zone, m·s^{-1}.

The required gas velocity in the ventilation ducts of the stator core tooth zone V_k was determined using the geometric data of the generator and the thermophysical characteristics of hydrogen and air, m·s^{-1}:

$$V_k = \frac{L \cdot 10^6}{\left(\dfrac{k_s}{2k_s - 1}\right) \cdot n_{rs} \cdot b_{rs} \left[\pi \left(D_s + h_{ns}\right) - z_s \cdot b_{ns}\right]}, \tag{13}$$

where k_s - is the number of parallel fluxes in the core cooling system; n_{rs} - is the number of the core tooth ventilation channels; b_{r1} - is the diameter of the ventilation channel in the core tooth, mm; D_s - is the inner diameter of the stator core, mm; Z_s - is the number of stator slots; h_{ns} and b_{ns} - are the depth and width of the stator slot, mm; L - is the volume gas flux rate in the stator core ventilation duct, m^3·s^{-1}, which was determined using the flux continuity equation:

$$L = \frac{G}{\rho} = f \cdot V, \tag{14}$$

where V - is the speed of the coolant in the channels at the entrance to the tooth zone, m·s^{-1}; G - is the mass flow rate of the coolant, kg·s^{-1}; ρ - is the specific density of the coolant (hydrogen, air), kg·m^{-3}; f - section of the ventilation channel in the core tooth, m^2:

$$f = \pi \cdot \frac{d^2}{4}, \tag{15}$$

d - hydraulic diameter of the ventilation duct, m:

$$d = 1.13 \cdot \sqrt{\frac{G}{\rho \cdot V}}. \tag{16}$$

Using Eq. (12) and Eq. (13), we obtained the heat transfer coefficient α_k in the ventilation ducts of the stator core teeth for a turbogenerator with certain geometric and electromagnetic parameters:

$$\alpha_k = 1 + 0.25 \cdot \left(\frac{L \cdot 10^6}{\left(\dfrac{k_s}{2k_s - 1}\right) \cdot n_{rs} \cdot b_{rs} \left[\pi \left(D_s + h_{ns}\right) - z_s \cdot b_{ns}\right]}\right) \cdot \frac{1.3 \cdot \left(p_N^{0.8}\right)}{450}. \tag{17}$$

Similarly, we determined the heat transfer coefficient α_s in the ventilation ducts of stator core backrest, W·(°C mm^2)$^{-1}$:

$$\alpha_s = \frac{1 + 0.25 \cdot V_s}{450} \cdot 1.3 \cdot \left(p_N^{0.8}\right), \tag{18}$$

where V_s - is the speed of hydrogen (air) in the ventilation ducts of the stator core backrest, m·s^{-1}:

$$V_s = \frac{L \cdot 10^6}{\left(\frac{k_s}{2k_s - 1}\right) \cdot n_{rs} \cdot b_{rs} \cdot \pi \cdot (D_a + h_{ns})}, \qquad (19)$$

where D_a - is the stator core outer diameter, mm:

$$\alpha_s = \left[1 + 0.25 \cdot \frac{L \cdot 10^6}{\left(\frac{k_s}{2k_s - 1}\right) \cdot n_{rs} \cdot b_{rs} \cdot \pi \cdot (D_a + h_{ns})}\right] \cdot \frac{1.3 \cdot \left(p_N^{0/8}\right)}{450}. \qquad (20)$$

After determining the main coefficients that are necessary to calculate the parameters of the water coolers of a particular generator, the number and geometry of cooler channels were calculated to ensure sufficient heat removal (boundary conditions). The calculations were carried out taking into account the increased power of the turbogenerator, the accepted class of insulation heat resistance, with the established values of the external dimensions, stator core external and internal diameters. In calculations, it was assumed that the movement of the cooling medium (gas) is laminar ($Re < 2300$).

When modelling the generator core, the boundary conditions were the above dimensions and allowable mechanical stresses in the materials of the structural elements. The main limiting criterion was the allowable thermal load of the generator. When modelling, the finite element method (with the construction of a grid of parameters in the solid body of the model) and the SolidWorks software (SolidWorks Simulation application) were used. The features of the geometry of individual elements were taken into account (for example, the rounding of the corners of the ventilation ducts, which affects the nature of the movement of the coolant flow in the cooling system and ensures its laminar flux).

A model of the stator core segment of the TGV-200-2 turbogenerator with direct hydrogen (or air) cooling of the rotor and the internal volume of the machine has been created. In Fig. 6 the basic model of the core (without ventilation ducts) is given. Simulation of cores with channels and slots was performed similarly. Data TGV-200-2 with increasing power up to 250 MW, used for calculations, are given in Tab. 2.

Using the proposed model of the stator core segment $P_s = f(\alpha_k; \alpha_s)$ (Eq. (18) and Eq. (20)) the required diameters of axial ventilation channels were determined, taking into account the thermal load and electromagnetic indicators for turbogenerators with a capacity of 120–500 MW when cooled with hydrogen and air.

On Fig. 7 shows the distribution of thermal fields (change in temperature along the height of the teeth and backrest of the core) in the segment of the stator core. The increase in heat transfer from the core steel to the cooling gas is carried out by increasing the diameters of the ventilation canals. As a result, it was found that for TG with a capacity of 300 and 500 MW, even with hydrogen cooling, it is necessary to change the shape of the ventilation canals. It is necessary to use slits, which increases the heat release surface. Figure 4 shows the proposed changes in the shape of the channels for the passage of cooling gas.

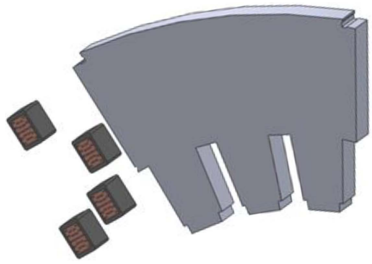


Fig. 6: 3D-model of the connecting elements (segment of the stator core of the turbogenerator TGV-200-2 and the winding), which was used in the simulation.

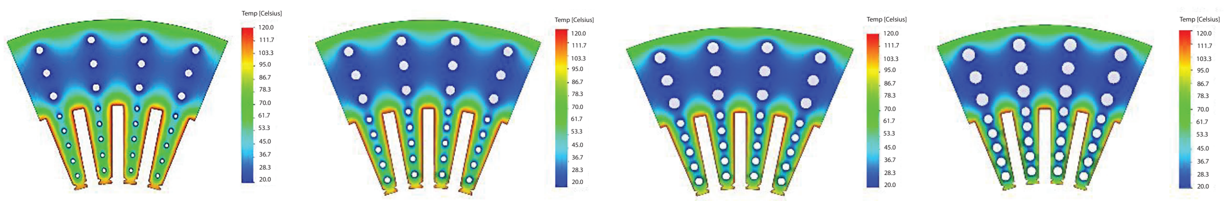
Tab. 2: Data of turbogenerator TGV-200-2 with increasing power up to 250 MW.

Name of parameters	Value
Power, MB·A/MW	294.12/250.0
Power factor ($\cos\varphi$), r.u.	0.85
Speed of rotation, rpm	3000
Voltage frequency, Hz	50
Number of phases	3
Stator current, kA	10.29
Efficiency, %	98.6
Stator voltage, kV	16.5
Stator winding scheme	≪star≫
Rotor current (calculated), A	3458
Excitation voltage (calculated), V	4373
Steel M270-50A, 0.5 mm thick of firm ≪Cogent≫ (Sweden)	

When constructing the model, the choice of the ventilation ducts diameter in the back and teeth of the core was carried out taking into account the provision of sufficient mechanical reliability of the teeth of the core according to the rules of the technology for stamping products from the thin sheet electrical steel (according to according to technical conditions of the plant ≪Electrotyazhmash≫, Kharkov, Ukraine).

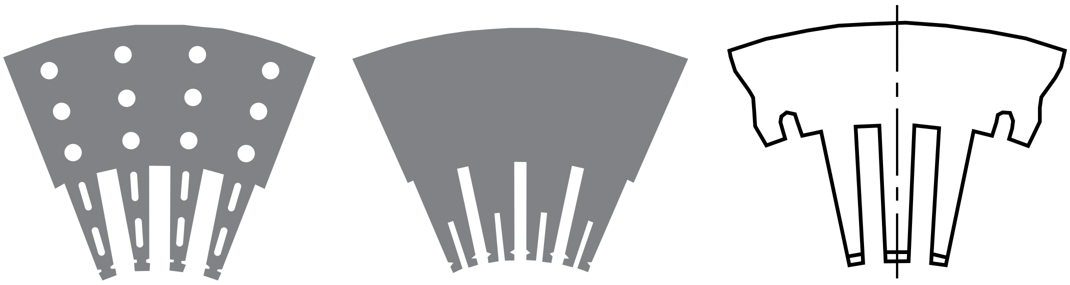
Additional thermal calculations were performed for the air-cooled generator, since, while maintaining the dimensions, the losses in steel and copper increase faster than the areas of the ventilation ducts.

It has been established that with an increase in power and with a simultaneous transition to air cooling, it is necessary to replace the round ventilation channels in the teeth of the stator core with channels of the “notch” type for the increases the heat removal area.



(a) 120 MW. (b) 200 MW. (c) 300 MW. (d) 500 MW.

Fig. 7: Models of temperature distribution in the stator core segment at different TG power.



(a) Non-through slits. (b) One slit. (c) Double through slits.

Fig. 8: Segments of the stator core with slits in the teeth.

The required flow rate of the cooling medium (air, hydrogen) can be determined, $\mathrm{m^3 \cdot s^{-1}}$:

$$V = \frac{Q}{c \cdot \theta_{\mathrm{B}}}, \tag{21}$$

where c - is the specific volumetric heat capacity of the cooling medium, $\mathrm{J \cdot (deg\ m^3)^{-1}}$: for air $c = 1005$, for hydrogen $c = 14270$, so its consumption will be less than the air consumption more than 14 times; $\theta_B = (t_r - t_c)$ - is the change in the cooling medium temperature; t_r - is the temperature at the inlet to the core ventilation canal, t_c - is the temperature at the turbogenerator core outlet, °C.

In the created model from the program (SolidWorks library) characteristics of turbogenerator elements materials and their properties were entered: coefficients of heat transfer of these materials and cooling environments at convective heat exchange; air cooling the surface of the charged package, as well as the value of losses:

$Q_{Fes} = 100$ W - losses in the steel of the stator core per one segment of the package;

$Q_{Cu} = 2650$ W - losses in the stator winding for one package at a capacity of 250 MW;

α_{air} - heat transfer coefficient from steel to air. In the calculations, it was taken $\alpha_{air} = 25\ \mathrm{W \cdot (m^2 \cdot K)^{-1}}$.

Middle air consumption per 1 kW of losses in the TG is equal to:

$$V = \frac{1000}{1100 \cdot \left(\frac{12}{30}\right)} = \left(\frac{0.03}{0.075}\,\mathrm{m^3}\right) \cdot \mathrm{s^{-1}}. \tag{22}$$

As an example, we present the calculation results for the turbogenerator TGV-200-2 with an increase in its power from 200 MW to 250 MW, the replacement of hydrogen with air in the turbogenerator internal volume, with a width of a single stator core package of 31.5 mm and a groove depth of 55 mm. The distance between individual core packages, which are made of 0.5 mm thick electrical steel sheets with varnish insulation, is 10 mm (data from the plant ≪Elektrotyazhmash≫, that manufactures such turbogenerators).

The results of comparing different options for the stator tooth zone (see Fig. 8) showed that when choosing a design with through slots in the teeth and reducing the thickness of the packages to 23.5 mm, the maximum temperature increase will be 10–15 °C less than in turbogenerator, in which the thickness of the outer package was 31.5 mm, and the slots were non-through (Fig. 9, graph 4 and graph 1, respectively). Temperature level 9 corresponds to the stator core outer diameter; level 4 corresponds to the beginning of the core stator tooth; level 1 - stator teeth crowns. For the considered TG the required speed of the cooling medium in the slotted ventilation channels was determined, Tab. 3. Permissible temperature of the stator core

Tab. 3: Calculation data of the stator core temperature with an increase in the turbogenerator power from 200 MW to 250 MW and a change in the cooling medium (hydrogen to air).

TG type	Cooling medium	Gas speed $m \cdot s^{-1}$	Heat transfer coefficient α (at 20 °C) $W \cdot (m^2 \cdot °C)^{-1}$	Maximum stator core temperature °C
TGV-200-2	hydrogen	3.75	55	63
TGV-250-2	hydrogen	3.75	55	77
TGV-200-2	air	3.75	21	84
TGV-250-2	air	3.75	21	101
TGV-250-2	**air**	**5.00**	**53**	**89**

was adopted according to the recommendations of the manufacturer +90 °C, (plant ≪ Electrotyazhmash≫, Ukraine)

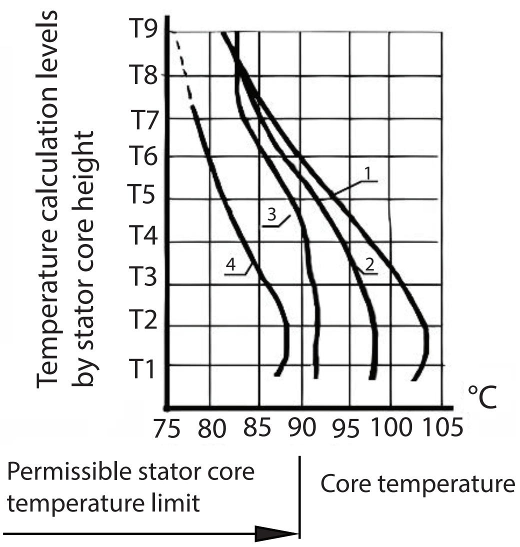


Fig. 9: Altitude temperature distribution turbogenerator tooth TGV-200-2 ($PN = 200$ MW, $\cos\varphi N = 0.85$).
1 - non-through cut in the tooth, the thickness of the outer package is 31.5 mm,
2 - a cut in the tooth through, the thickness of the outer package is 31.5 mm,
3 - non-through slot in the tooth, the thickness of the outer package is 23.5 mm,
4 - through the slot in the tooth, the thickness of the outer package is 23.5 mm.

It has been established that in order to fulfil this requirement (the operating temperature of the stator core is less than +90 °C), it is necessary to increase the speed of air passage through the ventilation ducts in the teeth and in the core back to 5 $m \cdot s^{-1}$. This speed is possible for the movement of the cooling gas without changing the nature of the flow (laminar).

3. Conclusion

- The prospects of maintaining the TPP turbogenerators in working order and, accordingly, the need to carry out work to modernize the turbogenerators were confirmed.
- The possibility of replacing hydrogen, which cools the internal volume of the generator and the rotor, with air in turbogenerators with a capacity of 200–300 MW has been proven, while increasing the power by 20–25 % and maintaining the dimensions of the generators through the use of new insulating materials and cooling intensification. This solution makes it possible to use the foundation existing at the station block, the turbine connection system and auxiliary systems.
- It has been established that with an increase in the turbogenerators power and with a simultaneous transition to air cooling, in order to increase their operational reliability, it is necessary to reduce the thickness of the extreme core packages of 25 %, make through cuts in the teeth of the stator core, and increase the speed of air passage through the axial ventilation channels by 33–35 % (from the standard value of 3.75 $m \cdot s^{-1}$ to 5 $m \cdot s^{-1}$) in order to reduce the tooth zone operating temperature to the limit temperature that to set by the manufacturer (+90 °C).
- To increase the mechanical reliability of the system for supplying water to the stator slots, it is proposed in modernized machines to use a "cone-sphere" connection for the heads of the stator winding rods and the tips of the fluoroplastic hoses. To intensify cooling, as well as to increase reliability, it was proposed to use four-section single-pass coolers with sections connected in series and use a cooler with monolithic fin instead of the cooler with spiral fin which has always been used in the generators with hydrogen-cooled.
- To select new design solutions, to change the geometry of the tooth zone of the stator core in the process of upgrading the turbogenerator, a model for calculating the active zone of the stator core was developed, which allows you to choose the best solution and minimize the number of calculations.

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Author Contributions

V.S. supervised the execution of work, summed up the results; A.M. collected materials at the turbogenerator factory; L.S contributed to the collection of materials for the article and to the documentation.

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