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Heating modes and design optimization of cogeneration steam turbines of powerful units of combined heat and power plant

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An important scientific and technical problem of increasing the efficiency of CHP steam turbine units through the optimization of their operation modes and the creation of new highly efficient flow parts of cogenerating turbines is solved. Solutions to the problem of rational distribution of heat loads between the network heaters of cogeneration turbines during the heating period are presented. The calculations were performed using the software package *SCAT* which was developed in IPMach NAS of Ukraine. The carrying out of calculations of three-dimensional turbulent flows in flow parts of turbines using modern software systems is an effective direction of increased efficiency of power equipment. For the numerical research of three-dimensional currents, steam in the flow part of the steam turbine software package *IPMFlow* which is developed in IPMach NAS of Ukraine is used. With the use of software package *IPMFlow*, the researches of three-dimensional currents steam in the flow part of the medium pressure cylinder of the steam turbine of series T-100-130 are carried, which showed the feasibility of optimizing the geometry of the flow part in order to improve gas-dynamic characteristics of blades apparatuses.

Keywords: network heater, combined heat and power, temperature graph, turbine flow part, efficiency

INTRODUCTION

One of the most important components of the well-being of citizens is the complete, reliable and environmentally safe provision of society's needs with heat, electricity and other energy products. A significant part of the consumption of fuel and energy resources (FER) in the World is accounted for by thermal power stations (TPS) and combined heat and power (CHP) plants, which produce electrical and thermal energy [1, 2].

Due to the fact that there is a shortage in the World and the cost of natural gas, petroleum products and coal is increasing; therefore, more and more attention is paid to improving the efficiency of FER for modernizing and optimizing the operation of existing power equipment [3].

The fuel and energy complex (FEC) of Ukraine has a highly developed infrastructure of the electricity and oil-gas industries and a sufficient mine fund of the coal industry [4]. Ukraine's entry into the global energy market requires an increase in the efficiency of energy facilities in order to increase competitiveness in the European market [5]. Therefore, it's relevant to search and select rational operating modes of cogeneration plants, which allow, without increasing the fuel consumption, maintaining the existing heat load to receive additional electricity [6].

Also, one of the ways to increase the efficiency of energy equipment is to carry out calculations of three-dimensional turbulent flows in turbine flow parts using software systems that are based on numerical integration of differential equations of gas dynamics (*IPMFlow*, *ANSYS*, etc.) [7, 8]. Their main advantage is the high information content of the results, the analysis of which allows determining the directions and methods of gas-dynamic improvement of flow-through parts of energy machines.

Six power units with a capacity of 100 MW are operated at the Ukrainian CHP plants (turbines of the T-100-130 series of modification of CJSC "Ural Turbine Plant"). Most of the electricity generating units were released and put into operation in the 60–70s of the last century. More than 10 turbine units of this type operates on the territory of the European Union countries (Poland, Romania and others), which makes the task even more relevant.

In recent years, the conditions and modes of operation of turbo-installations have changed significantly. In this regard, it's necessary to improve the power characteristics of power units in a wide range of operating parameters.

It is necessary to consider the operation of the cogeneration turbine depending on the environment temperature. Since heating steam extraction for heat supply is carried out from the medium pressure cylinder (MPC), then it is necessary to explore and improve its flow part separately. This approach will allow to obtain a high gas-dynamic efficiency of the flow part at the design stage and to increase the efficiency of using fuel resources during exploitation.

In the paper computational research results using software complexes carried out (*SCAT* and *IPMFlow*), which were developed at the A. Podgorny Institute of Mechanical Engineering Problems of the National Academy of Sciences of Ukraine, is presented. The results of the investigation for rational operation modes of cogeneration turbines of CHP using the example of the turbine series T-100-130 by the redistribution of heat loads between the network heaters are given. Also, the results of calculations of three-dimensional flows in two modifications of the MPC of the cogeneration turbine of the T-100-130 series are given: T-120/130-12.8 – the existing turbine and T-125/150-12.8 – marking T-122/146-12.8 at the stage of draft design.

OPTIMIZATION OF COGENERATION TURBINE UNITS OPERATION MODES

Mathematical model

The basic mathematical model and the software package *SCAT* for calculating power installations are adapted to solve the set task [9] and verified based on the results of researches of thermal schemes of a large-capacity CHP plant. The object of verification is the turbine T-100/120-130 of power units No. 1 and No. 2 of PJSC "Kharkiv CHP-5" [10].

The results of the verification confirmed the feasibility of using information models of turbine installations for performing computational researches [11].

The mathematical formulation of the solution of the problem of a comprehensive search for the best parameters of a heat and power installation can be represented in this way.

Depending on the task, it's necessary to minimize or maximize the non-linear function of the goal

$$N = N(X, Y, Z),$$

$$X \in XD, \tag{1}$$

where XD – the permissible range of the function change N, by a system of nonlinear constraints is determined:

– in the form of equality

$$\Phi_j(X_j, Y_j, Z_j) = 0, \qquad j = \overline{1, n_{\mathrm{TS}}}; \tag{2}$$

– in the form of inequalities

$$[F_{\min}] \le F(X, Y, Z) \le [F_{\max}], \qquad [U_{\min}] \le U(X, Y, Z) \le [U_{\max}]; \tag{3}$$

– on the independent communication parameters

$$Y_{\min i} < Y_i < Y_{\max i}, \qquad i = \overline{1, k}; \tag{4}$$

– on constructive parameters

$$X_{\min i} < X_i < X_{\max i}, \qquad i = \overline{1, k}, \tag{5}$$

where X is a set of structural parameters of the energy unit, Y is a set of defined state parameters (thermodynamic parameters and efficiency), Z is a set of given external factors (external air temperature, etc.) Φ_j is a set of balance equations for all elements of the equipment installation.

The task of finding rational modes of operation of a heat and power installation is solved as Eq. (1) and is solved using the experiment planning method: the values of independent coupling parameters, design parameters and a set of values of discrete parameters (features) of a structural-layout installation are found for which the function of these parameters N reaches a minimum or maximum compliance (Eqs. (2)–(5)).

The use of the specified software complex allowed conducting a research the purpose of which is to determine the possibility of increasing the production of electricity by the power unit under given operating conditions of the cogeneration turbo generator on the consumption of fresh steam and the amount of heat supplied. Therefore, the criterion for evaluating the efficiency of a cogeneration turbine is the additional electric power generated under a given turbine operating mode.

The performance evaluation criteria of work are as follows:

$$\Delta N_t = N_t - N_t^i = \mathcal{F}(G_0, Q_{h1}, Q_{h2}, t_{ot}, G_{nw}), \tag{6}$$

where N_t is turbine power at specified operating conditions, MW, N_t^i is turbine power at instructive characteristics of the operating mode, MWG_0 is fresh steam consumption, t/h (1 kg/s = 3.6 t/h), Q_{h1}, Q_{h2} is the amount of heat that is transmitted to the water in the network heaters No. 1 and No. 2, MW t_{ot} is outdoor temperature, °C G_{nw} is network water consumption, t/h.

Optimization of operating modes on the example of the turbine T-100-130

As an example, the question about the generation of additional electricity from power units with turbines T-100/120-130 of PJSC "Kharkiv CHP-5" is considered. The principal thermal scheme is shown in Fig. 1. The goal is achieved through the rational distribution of heat loads between the network heaters of cogeneration turbines.

At the first stage, the task of determining the conditions of the additional electricity production is considered by using only one heater of a bottom rung $\bar{Q}_{hl}$ = 1.0 in comparison with the operation of two heaters at equal thermal load $\bar{Q}_{hl}$ = 0.5, which is regulated by the operating instructions.

The ranges of thermal loads at which there is a possibility of increasing the thermal efficiency of the turbine under consideration are determined according to the temperature schedule of the central regulation of heat supply for the Kharkiv region.

According to the results of computational research, Fig. 2 shows a graph of the change in the power of the turbine T-100/120-130, depending on the outdoor temperature t_{ot} and expenses of network water G_{nw} at values $\bar{Q}_{hl}$ = 1.0 and $\bar{Q}_{hl}$ = 0.5. Line A-A corresponds to the break on the temperature chart at t_{ot} = 3.5°C.

It is shown that in the range of variation of the outdoor temperature $-11 \le t_{ot} \le -2.0$°C, when the flow rate of network water is G_{nw} = 1000 t/h, the mode of operation of the turbine with two network heaters is rational with equal distribution of heat load between them ($\bar{Q}_{hl}$ = 0.5). When $t_{ot} > -2$°C and G_{nw} = 1000 t/h, the required temperature in the direct line is not maintained. For outdoor air temperature $t_{ot} \ge 2$°C with network water consumptions 2000, 3000, 4000 t/h the turbine power when operating with only one lower stage heater ($\bar{Q}_{hl}$ = 1.0) exceeds turbine power in modes with two series-connected heaters with $\bar{Q}_{hl}$ = 0.5.

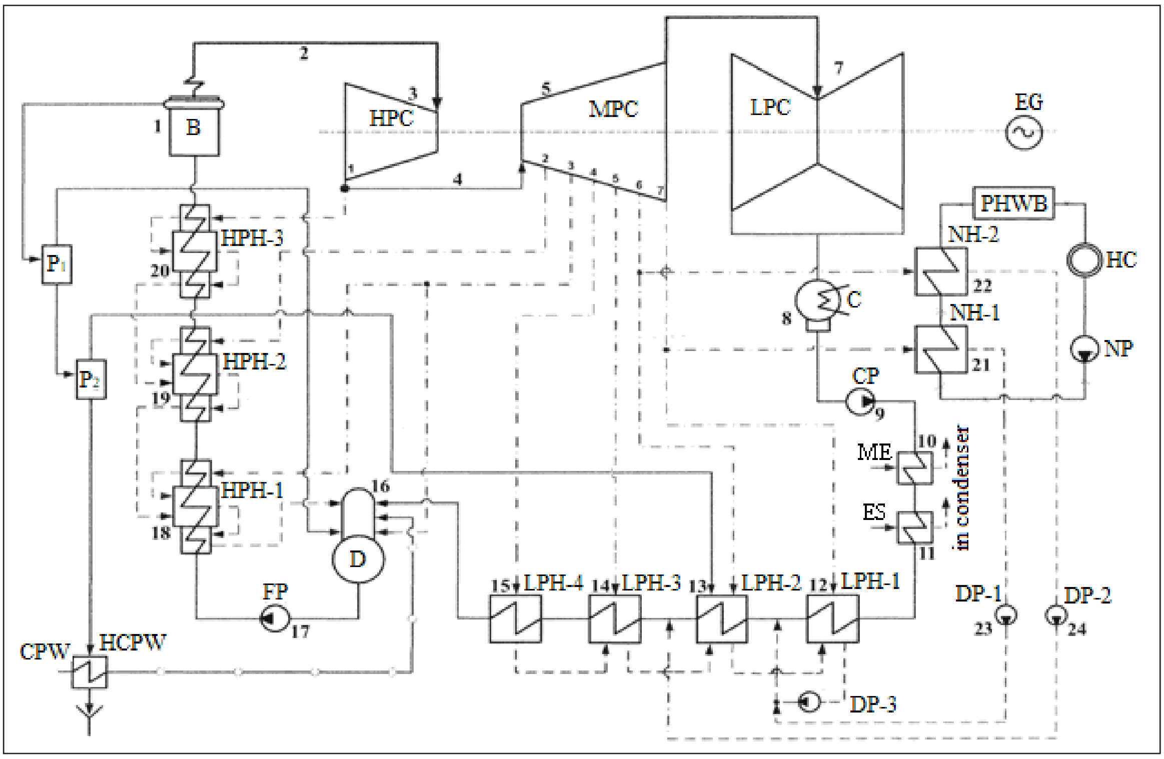

Fig. 1. The principal thermal scheme of the steam turbine T-100/120-130

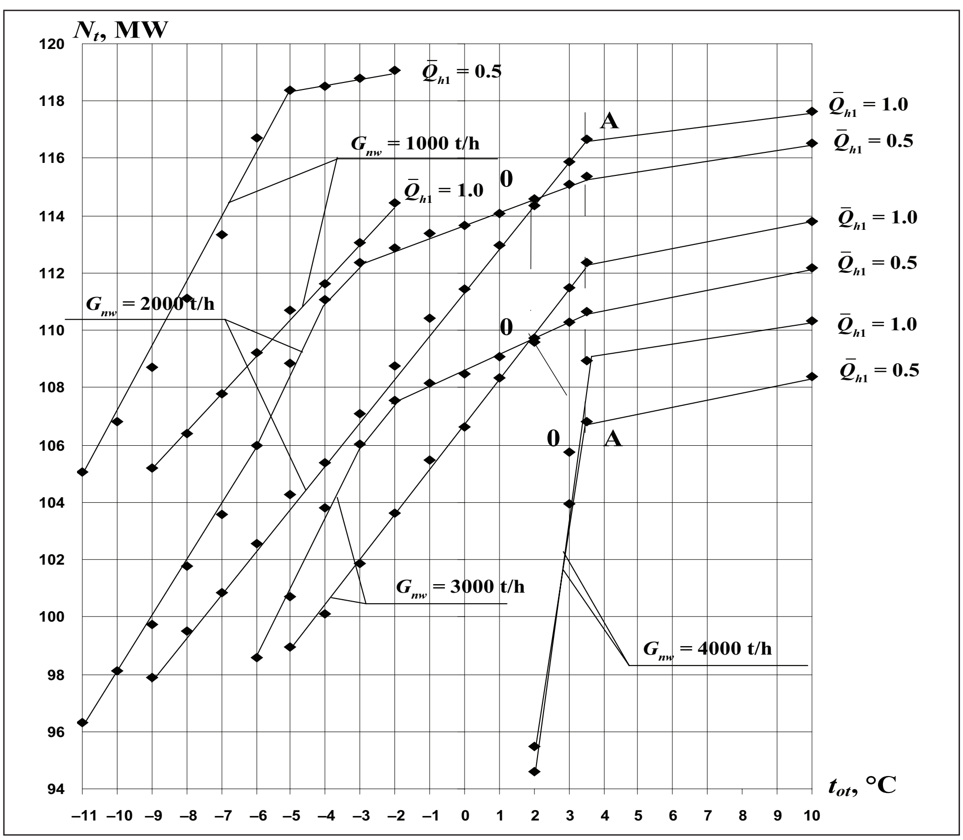

Fig. 2. Comparative characteristics of change in power of turbo-installation with one $\bar{Q}_{hl} = 1.0$ and two network heaters an equal distribution of heat load $\bar{Q}_{hl} = 0.5$

Thus, the condition $\Delta N_t > 0$ is fulfilled for the region of ambient temperature change of $2.0 \le t_{ot} \le 10$°C and higher, and this considered method ($\bar{Q}_{hl} = 1.0$) provides for obtaining additional electric power. In the region $t_{ot} \le 2$°C, the power growth does not occur ($\Delta N_t < 0$); therefore, it's advisable to work with two of series-connected network heaters of the lower NH-1 and the upper NH-2 with equal distribution of the heat load between them ($\bar{Q}_{hl} = 0.5$). No power increment $\Delta N_t = 0$ corresponds to the line with points of zero values (Fig. 2).

The analysis performed on the second stage was used to search for the optimal distribution of the heat load $\bar{Q}_{hl}^{opt}$ when the outside air temperature varies in the range $-11 \le t_{ot} \le 10$°C, which ensures an increase in the electrical power of the turbine unit, in contrast to the operating mode with two included heaters and an equal distribution heat load between them ($\bar{Q}_{hl} = 0.5$). The results are shown in Fig. 3.

The area of positive values of the gain in electric power zN_m, depending on the technological characteristics of the turbine, is in the following ranges:

I – $3.5 \le t_{ot} \le 10$°C; $1.45 \le G_{nw}/1000 \le 4.0$;
II – $0 \le t_{ot} < 3.5$°C; $1.2 \le G_{nw}/1000 \le 3.5$;
III – $-5 \le t_{ot} < 0$°C; $1.0 \le G_{nw}/1000 \le 3.25$;
IV – $-8.5 \le t_{ot} < -3$°C; $1.0 \le G_{nw}/1000 \le 3.05$;
V – $-11 \le t_{ot} < -7.8$°C; $1.0 \le G_{nw}/1000 \le 2.45$.

For each of the selected areas, depending on the consumption of network water, the gains of the electric power of the turbine are given in Table 1.

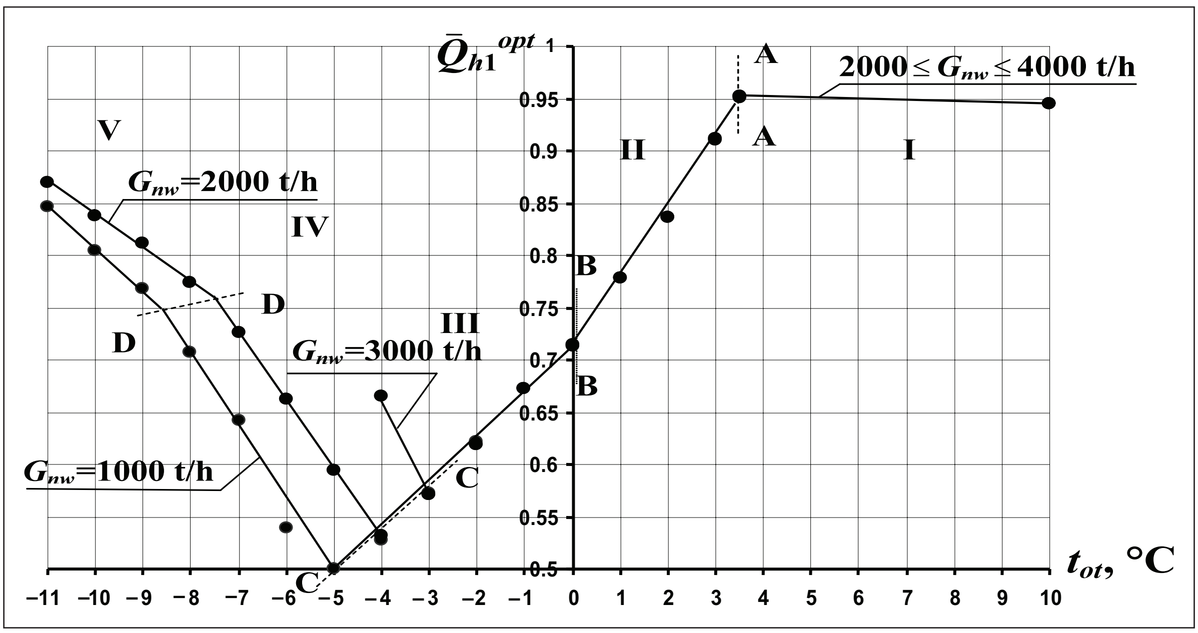

Fig. 3. The change in the optimal distribution of heat load between the network heaters NH-1 and NH-2, depending on the outdoor temperature: A-A, B-B, C-C, D-D – lines corresponding to the fracture on the temperature graph

Table 1. **The value of the increase in the electric power of the turbine T-100/120-130**

$\frac{G_{nw}}{1000}$	Area				
	I	II	III	IV	V
	Energy gain ΔN_t, MW				
1.0	–	–	0.26–0.07	0.33–1.37	1.33–0.90
2.0	1.33–1.48	1.35–0.67	0.54–0.69	1.69–2.30	2.04–1.33
3.0	1.73–1.87	1.68–0.75	0.57–1.21	1.21–0.85	–
4.0	2.03–2.5	–	–	–	–

According to Table 1, the maximum values of the gain in electrical power of the cogeneration turbine T-100/120-130 when operating with optimal distribution of heat load between the network heaters of the lower and upper stages are in the area I at the flow rate of network water G_{nw} = 4000 t/h, in areas IV and V at G_{nw} = 2000 t/h.

For simplicity of use at power generating units of the CHP, the results are processed by mathematical statistics methods and the corresponding regression equations are constructed.

GASDYNAMIC CALCULATIONS OF TURBINE FLOW PARTS

Technique of construction of the geometry of turbine blades

For the numerical research of three-dimensional currents, steam in the flow part of the steam turbine software package *IPMFlow*, developed in IPMach NAS of Ukraine is used [12]. For design of the geometry of the crown shoulder blade in the flow part of the axial turbine, the method of parameterization and analytical profiling of the shoulder blade is used. The blade is given as an arbitrary set of flat profiles, each of which is considered in a Cartesian coordinate system with the abscissa axis which is parallel to the turbine axis and the ordinate axis, which coincides with the front lattice (Fig. 4).

The input and output edges, as well as curves backrest and troughs are describing profiles. Input and output edges are circles, curves backrest – polynomials of 5th order, trough – polynomials of 4th order. They have the following view:

$$y(x) = \sum_{i=0}^{5} a_i x^i \quad a_i = const, \tag{7}$$

$$y(x) = \sum_{i=0}^{4} a_i x^i \quad a_i = const. \tag{8}$$

In forming the lattice, profiles are baseline data: b_x – profile width, α_1 – skeletal angle lattice on inlet, r_1 – radius of the inlet edge, α_{2ef} – effective angle of flow exit, r_2 – radius of the outlet edge, t – the lattice spacing, $\Delta\alpha_1$, $\Delta\alpha_2$ – angles “sharpening” of input and output edges, α_{2bwv} – angle “level” of backrest, $\alpha_\infty + \alpha_{2s} + \alpha_{2bev}$; 1_{bac}, 2_{bac}, 1_{tr}, 2_{tr} – point coupling the input and output edges with curves backrest and trough (Fig. 4). The coefficients of the curve (7), which describes backrest, are calculated by iteration from the relations:

$$\begin{cases} y'_{bac}(x_{1bac}) = tg(\alpha_1 + \Delta\alpha_1) \\ y''_{bac}(x_{1bac}) = \{y''_{bac,0}\} \\ y_{bac}(x_O) = y_O \\ y'_{bac}(x_O) = tg(\alpha_{co}) \\ y_{bac}(x_{2bac}) = y_{2bac} \\ y'_{bac}(x_{2bac}) = tg\{\alpha_{2s}\} \end{cases}, \tag{9}$$

where α_{2s} and $y''_{bac,0}$ are variable parameters for the relations (9). Their selection should provide the predetermined amount lattice of throat O and the minimum value of maximum curvature on the set of curves (7). Throat size is determined by the given values of the lattice spacing and effective angle

$O = t \cos \alpha_{2ef}$.

After determining the curve of the back and inscribing input and output edges, the coefficients

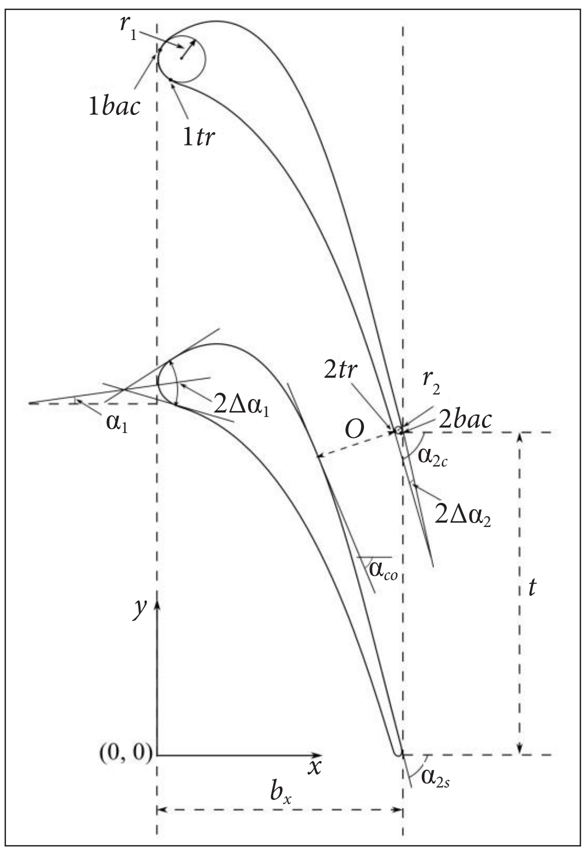

Fig. 4. Lattice of profiles

curve (8) for the trough is calculated by an iterative way by using the following relations:

$$\begin{cases} y_{tr}(x_{1tr}) = y_{1tr} \\ y'_{tr}(x_{1tr}) = tg(\alpha_1 - \Delta\alpha_1) \\ y''_{tr}(x_{1tr}) = \{y''_{tr,0}\} \\ y_{tr}(x_{2tr}) = y_{2tr} \\ y'_{tr}(x_{2tr}) = tg\alpha_{2c} \end{cases}, \qquad (10)$$

where x_{1c}, y_{1c}, x_{2c}, y_{2c}, are the coordinates of the touch curves trough with circles of input and output edges, which are determined by a given angle $\alpha_1 - \Delta\alpha_1$ on the input edge and varying angle α_{2c} on the output edge. The angle α_{2c} is selected in the range of α_{co} and α_{2s} thereby to provide a minimum value of maximum curvature curve trough.

Simulations of the MPC flow part of turbine of series T-100

Simulations of the medium pressure cylinder (MPC) flow part of turbine T-120/130-12.8-8MO were produced by using the software package *IP-MFlow*. The view of the turbine stages of the MPC flow part is shown in Fig. 5.

The results of thermal calculations of UTZ for calculation researches of obtaining the basic geometrical characteristics of the stages and the gas dynamic characteristics of the MPC flow part of turbine T-120/130-12.8 are taken.

Three-dimensional gas dynamic calculations were performed at the difference grid, consisting of about 500 thousand cells in each crown.

Painting flow around the blades apparatus in all stages is very favourably showed by an analysis of the calculation results. However, the graphs of distribution of static pressures on the blade surfaces (flow around the stator blades (SB) and rotor blades (RB) in the middle sections of the 8th degree is shown as an example) did not show any noticeable monotonic changes of pressure on the side rarefaction of profiles (Fig. 6), which is caused primarily by imperfection of the geometrical shape of the blades apparatus (discontinuities of the second derivative values on the backrest and trough). In the meridional plane in the peripheral sections of the flow, separation occurs, which in stages of the third compartment are the most observed (Fig. 7).

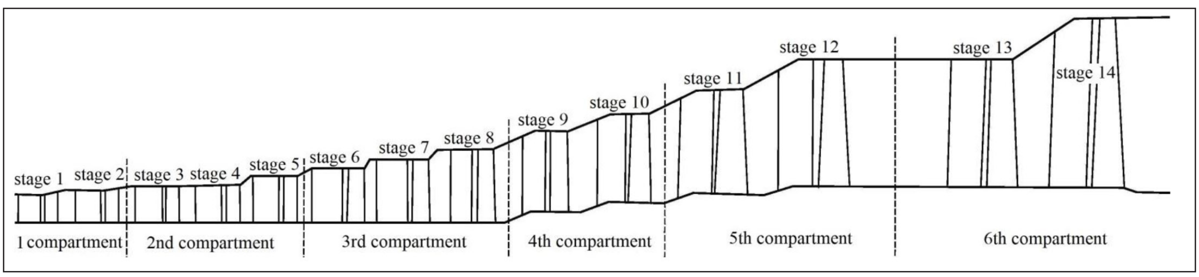

Fig. 5. Meridional of the projection initial of the MPC flow part of turbine T-120/130-12.8

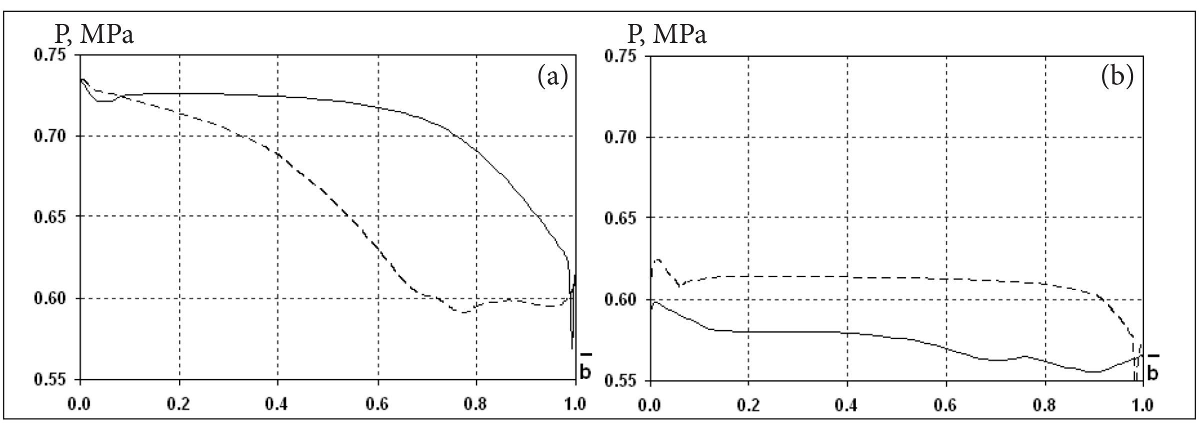

Fig. 6. Pressure distribution along the surface profile of the stators and rotor blades stages MPC of the turbine T-120/130-12.8: (a) – SB 8th stage, (b) – RB 8th stage

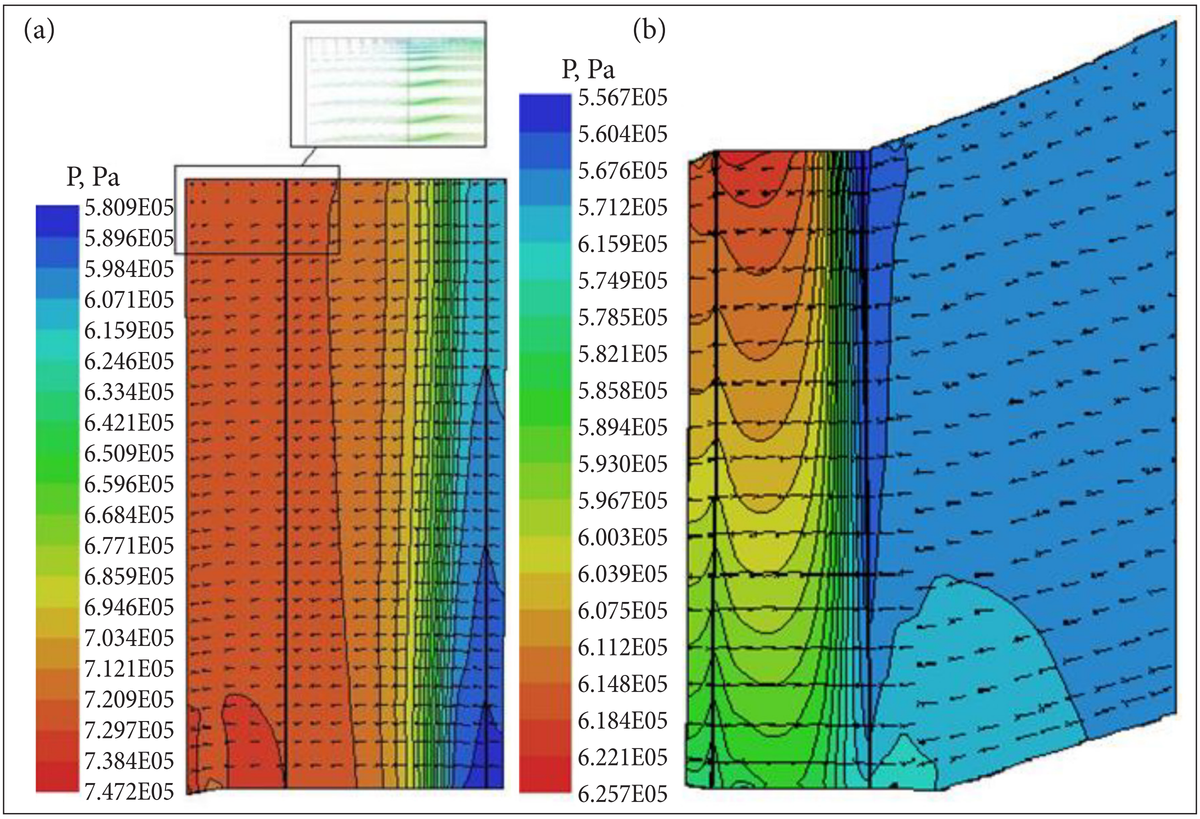

Fig. 7. Painting flow around in the meridional plane for SB and RB in the middle sections of 8th stage: (a) – SB, (b) – RB

The three-dimensional model of the flow part of the MPC of the T-125/150-12.8 new turbine in view of the shortcomings of the existing flow part was developed (Fig. 8).

Through the use of the smooth profiles of the graphics static pressure distribution on the blade surfaces have become more monotonous (Fig. 9). Flow separation in the meridional plane at the periphery is almost completely vanished (Fig. 10). The meridional section of advanced design was built similarly to the existing structure. The pattern of flow around is observed favourably by the results of visualizing (Fig. 10).

Computational researches of basic and improved designs of flow of the MPC showed the stock of increased efficiency by improving the geometry of the blades (Fig. 11).

All stages of the new construction of the MPC have a higher efficiency as it is presented in Fig. 11. The total efficiency of the proposed design is 92.3% without taking into account moisture and 90.1% with moisture taken into account, that is by 2.9% and 2.2%,

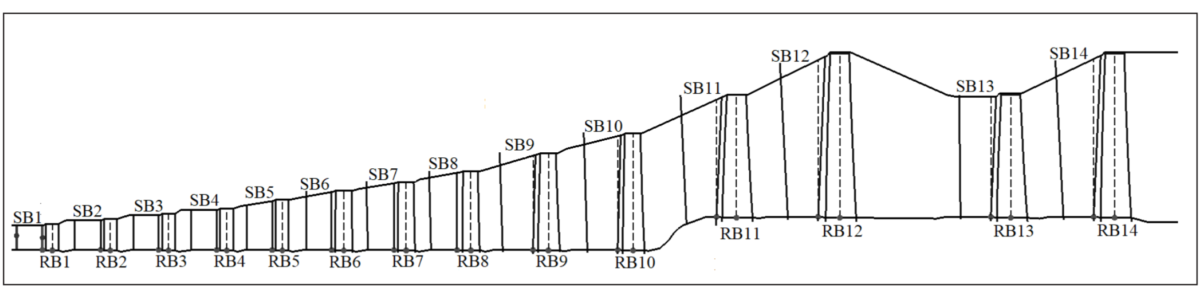

Fig. 8. Meridional projection of the improved flow part of the MPC of the turbine T-125/150-12.8

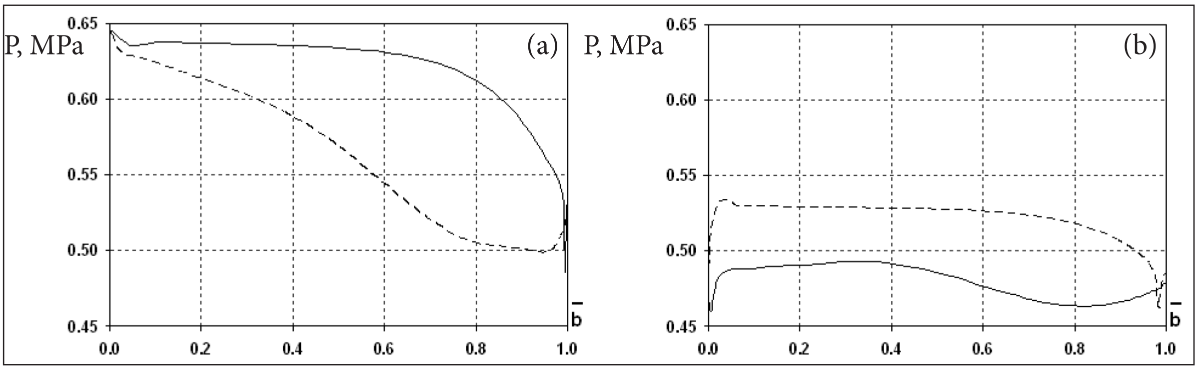

Fig. 9. Pressure distribution along the surface profile of the upgraded stator and rotor blade stages of MPC of the turbine T-125/150-12.8: (a) – SB 8th stage, (b) – RB 8th stage

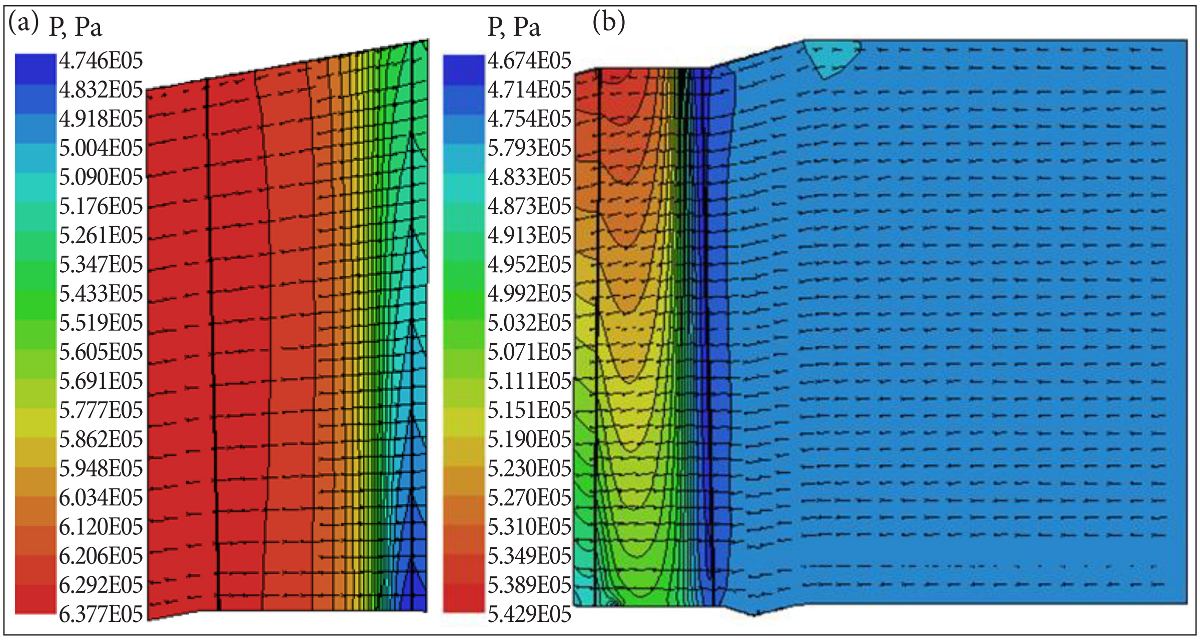

Fig. 10. Painting flow around in the meridional plane of upgraded SB and RB in the middle sections of 8th stage: (a) – SB; (b) – RB

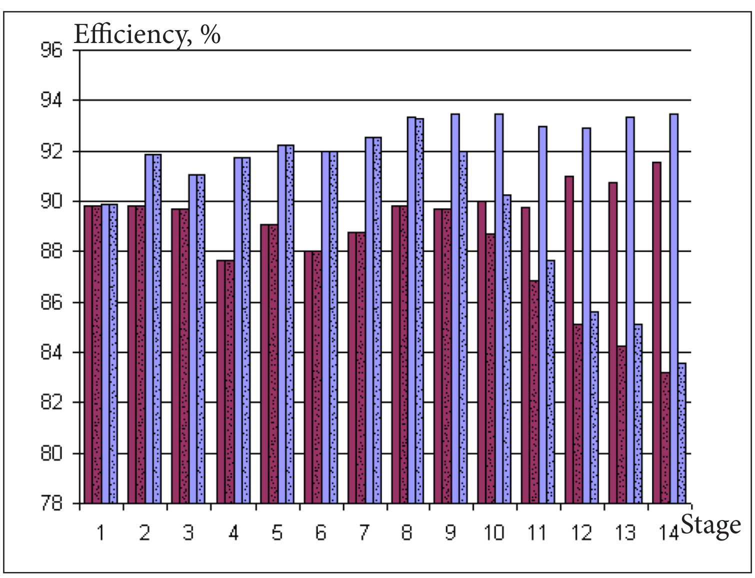

Fig. 11. Efficiency of stages of the MPC flow part of turbines T-120/130-12.8 and T-125/150-12.8:
– T-120/130-12.8,
– T-125/150-12.8,
– efficiency without moisture loss,
– efficiency including moisture losses [13]

respectively, more than in case of the original design.

DISCUSSION

A scientific and practical problem has been tasked and solved in the work, which is aimed at increasing the efficiency of high-power steam turbines by improving their operating modes and the gas-dynamic characteristics of the flow-through parts. The solution of this problem allowed to obtain the following scientific and practical results:

1. The method of operation of the cogeneration turbine T-120/130-130 with one network lower stage heater allows, in some cases, to provide a more economical mode of operation of the turbine compared to the instructive mode of heat load distribution between two heaters ($\bar{Q}_{hl}$ = 0.5) due to the rational selection of the flow of network water at specified temperatures of outdoor air according to temperature graph of the heating system. This is especially effective during scheduled maintenance work and in emergency situations associated with disconnecting the network upper stage heater. Thus, in the range of variation of the outdoor air temperature $2 \leq t_{ot} \leq 10$°C, the value of the gain in the electric power of the turbine is considered to be able to reach 2.13 MW.

Testing of this method at the cogeneration turbine T-100/120-130 of the power unit No. 1 of PJSC "Kharkiv CHP-5" confirmed the presence of a positive effect on the turbine operation modes with one-stage heating of the network water in the obtained spectra of changes in the normative parameters.

2. On the basis of the conducted technical and economic assessment, it has been established that using the method of optimal distribution of heat load between network heaters during the heating period (180 days), taking into account changes in the outdoor air temperature, $-11 \leq t_{ot} \leq 10$°C allows, in comparison with the adopted ($-Q_{hl}$ = 0.5), to significantly save, due to the additional production of electricity, fuel in power units with T-100/120-130 turbines in the amount of 3,500 tons of standard fuel, which is equivalent to saving 3 million m^3 of natural gas.

3. The methods that are proposed for producing additional electric power can be used for cogeneration turbines with the types of turbines T-50/60-130, T-100/120-130, T-180/210-130, T-250/300-240 and their modifications.

4. As a result of computational researches of the flow part the next problems are implemented:

– analysis and verification of the results;

– the 3*D* model was constructed and the flow of steam in the MPC of the turbine T-120/146-12.8 was performed and calculated;

– a series of verification calculations (over 300) was made, based on the analysis of the results the optimization of blading was held, and the geometry of the flow part was obtained, which is sufficient for correct playback characteristic profiles of the blade apparatus, etc.;

– the geometries of all sections of profiles designed of stator and rotor blades of MPS stages was constructed;

– pressure in front of and behind the compartments was maintained (with change no more than ± 5%);

– the profiles of rotor and stator blades in 12–15 stages were constant, in 17–25 stages – with a twist, and other constructive improvements (for maximum efficiency) were built.

5. In the calculations the following conditions are ensured:

– the diameters in root and the number of stages stored, also the length of the rotor according to the preliminary geometry of the MPC flow part of the turbine are unchanged; the degrees of reactivity in root of about 5% is accepted;

– width profiles and geometrical characteristics of the existing turbine stages T-120/130-12.8-8MO according to the preliminary flow part geometry of the new turbine T-125/146-12.8, including the radii of the input and output edges of the profiles, the number of blades, height of RB blades and output cross=sections of the blades SB, and others are preserved to fulfil strength conditions.

6. The peripheral lines of diaphragm channels of conical shape perform so that a straight line of peripheral contour in the meridian projection is located on the outlet edge of the blade SB about to the outlet edge of the previous blade RB. This shape of lines allows to increase efficiency

compared with "stepped" lines by 0.5–1.0% (depending on the degree of expansion). New profiles that are built using the monotonic curves of the 4th and 5th orders, as well as blades with a variable form of profiles, the use of which will increase the efficiency by 1.0–2.0%, are applied.

CONCLUSIONS

1. The results of numerical simulation showed that in accordance with the temperature schedules of the central regulation of heat supply, provided that the thermal loads are rationally distributed between the network heaters of the cogeneration turbine, an increase in electrical power without additional fuel combustion can be obtained. The exploitation of the cogeneration turbine T-120/130-130 with a change in the outdoor air temperature $2 \leq t_{ot} \leq 10$°C can give an increase in electrical power of 2.13 MW without additional fuel combustion.

2. The software package for carrying out gas-dynamic calculations of turbine flow parts made it possible to determine the degree of perfection of stator and rotor blades and propose technical solutions for improving the efficiency of the turbine flow part. Numerous studies of the two options of MPC steam turbines T-120/130-12.8 (existing turbine) and T-125/150-12.8 (new turbine) were performed. The new turbine through the use of modern profiles of the blades and seals, changing the shape of meridional contours efficiency of the MPC significantly are shown. Efficiency without losses from moisture increased by 2.9%, and efficiency with moisture taken into account increased by 2.2%, and the value was 92.3% and 90.1%, respectively.

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DIDELIŲ KOMBINUOTO CIKLO JĖGAINIŲ KOGENERACINIŲ GARO TURBINŲ ŠILDYMO REŽIMAI IR DIZAINO OPTIMIZAVIMAS

Santrauka

Išspręsta svarbi mokslui ir praktikai kombinuoto ciklo jėgainių garo turbinų efektyvumo didinimo problema optimizuojant jų veikimo režimus ir įdiegiant efektyvias kogeneracinių turbinų fluido srauto dalis. Pateikti šiluminių apkrovų racionalaus pasiskirstymo tarp termofikacinių turbinų tinklo šildymo būdai. Skaitiniai tyrimai vykdyti programinės įrangos paketu SCAT, kuris yra sukurtas Ukrainos nacionalinės mokslų akademijos Mechanikos inžinerijos problemų institute. Pagrįsta, kad trimačių turbulentinių srautų modeliavimas turbinų fluido srauto dalyse, naudojant šiuolaikines programinės įrangos priemones, yra veiksminga įrangos efektyvumo didinimo kryptis. Modeliuojant trimačių srovių garo srautinę dalį garo turbinoje panaudotas programinės įrangos paketas IPMFlow, kuris taip pat sukurtas Ukrainos nacionalinės mokslų akademijos Mechanikos inžinerijos problemų institute. Programinės įrangos paketas IPMFlow užtikrina patikimus tyrimus vidutinio slėgio cilindro trimačių srovių garo srautinės dalies turbinose T-100-130. Rezultatai parodė, kad galima optimizuoti srauto dalies geometriją, siekiant pagerinti turbinos disko dinamines charakteristikas.

Raktažodžiai: tinklo šildytuvas, kombinuotas ciklas, temperatūros grafikas, turbinos srauto dalis, efektyvumas