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## INCREASE OF THE EFFICIENCY OF HEAT SUPPLY UNITS OF THE CHP PLANTS DUE TO THE CHOICE OF RATIONAL HEAT RELEASE MODES

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*The possibilities of increasing the efficiency of heat supply turbines of CHPPs due to the choice of rational modes of operation of network water heaters are analyzed. With the help of a software and computing complex developed at the Institute of Mechanical Engineering Problems of the National Academy of Sciences of Ukraine and adapted by the authors to the operating conditions of CHPP generating equipment with one or two network heaters, a set of calculation studies of various ways of connecting them depending on the outdoor air temperature is conducted in the paper. Areas of positive effect associated with increase in the electric power of the power plant have been established. The calculation study was carried out at typical power plant water consumption from 1000 t/h to 4500 t/h, in the range of changes in the outdoor air temperature from -11 °C to 10 °C (heating season) and more than 10 °C (hot water supply). The load change of the power unit was carried out due to the consumption of fresh steam at a constant pressure and temperature at the turbine inlet. As shown by the results of the operation of the T-100/120-130 heat supply turbine in operating conditions with one or two heat supply steam selections, in the area of positive outdoor air temperatures above 2 °C, it is advisable to use one lower selection (with the upper one turned off) for all network water consumption. At the same time, additional electric power in the area of outdoor air temperatures above 6 °C can be from 0.25 MW to 2.15 MW. However, when the outdoor air temperature is less than 2 °C, work with one lower heating selection becomes irrational. From the point of view of choosing rational modes of operation of turbine plants, the most important are the results of determining the optimal distribution of heat load between network heaters. The gain in electrical power of the turbine can be up to 2.46 MW in the nominal mode of operation with two heaters, and up to 7.84 MW in comparison with the use of single-stage heating. The nature of the influence of the distribution of the heat load indicates that during deprivation from the instructional uniform distribution of the heat load between network heaters, it is possible to obtain additional electricity.*

**Keywords:** *mathematical modeling, network water heater, modes of operation, combined heat and power plant, heat supply power plant.*

### Introduction

One of the ways to increase the efficiency of the combined heat and power plants (CHPPs) main generating equipment is the choice of rational modes of operation of turbine plants taking into account the consumption of thermal energy by the city's thermal networks, which is determined by the temperature schedule, the state of the thermal reserves, industrial consumers and directly by the thermal networks, which have significantly changed their thermal characteristics after putting them into operation [1–5]. This makes it necessary to adjust the modes of heat energy release in comparison with the given temperature schedule.

### Problem statement

The heat supply plant (HSP) of the CHPP works as part of a turbine plant, using a pair of reduced parameters from the upper and lower heat supply selections.

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The two-stage heat supply plant (Fig. 1) is connected to selection 6 of the medium-pressure cylinder (MPC), from which the steam of high pressure  $p_u$  (from 0.06 MPa to 0.25 MPa) is received by the network water heater of the upper stage NH-2, and to the adjustable selection 7 of the low-pressure cylinders (LPC), located between the MPC exhaust and the LPC inlet, from which steam with a pressure  $p_u$  (from 0.05 MPa to 0.20 MPa) is received by the network water heater of the lower stage NH-1 [6–10].

Heaters NH-1 and NH-2 are connected to the network water in sequence, which ensures the temperature of the network water at the outlet of the heat supply plant, which is determined by the consumer's temperature schedule. To regulate this temperature, latches 1–3 and jumpers with latches 4, 5 are used.

Regulation of the amount of released heat is carried out by a rotary diaphragm located at the entrance to the flow part of the LPC, driven by a servo motor. By turning the ring of the regulating diaphragm, the distribution of steam flows in NH-1 and the flow part of the LPC is carried out, on which the amount of heat given to the network water in the NH-1, and the power generation in the LPC, which is transmitted to the electric generator, depend.

At the same time, the increase in steam consumption in the NH is associated with a decrease in the steam consumption in the LPC, which leads to the operation of the LPC stages in the power consumption mode, in which electricity generation decreases. This makes it necessary to choose rational modes of NH heat release and coordinate their operation with the operation mode of the LPC.

**The purpose of the paper**

The purpose of the study is to analyze the operation of the heat supply plant using the example of the T-100/120-130 turbine and to determine the rational operating modes of the turbine when only the network heater of the lower stage is using and with the optimal distribution of the heat load between the heaters of the lower and upper stage.

**Research methods**

To increase the efficiency of the use of steam turbine plants of CHPPs, a mathematical model and a basic software complex for the study of axial turbines SCAT, developed at the IPMach of the National Academy of Sciences of Ukraine and adapted by the authors to the conditions of layout and operation of power equipment of powerful heat supply power plants, are used [2, 11–14].

When creating a flexible mathematical model of a power plant, focused on determining the optimal operation of a heat supply turbine plant, taking into account the maximum number of factors that affect economic indicators, it is necessary to choose a criterion for the efficiency of its operation. At the same time, the following values can be used as a criterion for operational efficiency in general case: the maximum amount of electricity produced –  $N_{sum}$ , minimum specific fuel consumption –  $q_{\Sigma}$ , minimum total fuel consumption –  $B_{\Sigma}$ , minimum payback period of the CHPP expansion project –  $\tau_{payback}$ , etc.

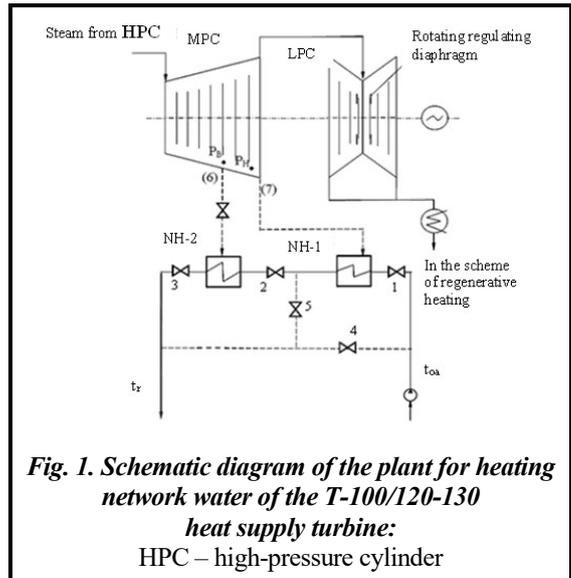
The analysis of the methods of finding a rational mode of operation showed that when solving the problems of distribution of thermal and electrical loads of CHPP turbines, the method of the experiment planning is considered to be the most effective [13].

To apply this method, a macro model of the thermal circuit was created, which is briefly described below and is a "black box" type dependency [11].

Depending on the stated problem, it is necessary to minimize or maximize the nonlinear objective function. The mathematical statement of the problem of finding the best parameters and profile of a thermal power plant can be written as follows

$$N = N(X, Y, Z); X \in XD, \tag{1}$$

where  $N$  is the objective function of the problem to be solved;  $X$  is a multitude of structural parameters of the power plant;  $Y$  is a multitude of defined parameters of the state (thermodynamic parameters and efficiency coeffi-



**Fig. 1. Schematic diagram of the plant for heating network water of the T-100/120-130 heat supply turbine:**  
HPC – high-pressure cylinder

cient);  $Z$  is a multitude of external factors (temperature of outdoor air, parameters and flow of steam in selections, total flow of steam);  $XD$  is an admissible area of  $N$  change, determined by the system of nonlinear constraints:

– in the form of equalities

$$\Phi_j(X_j, Y_j, Z_j) = 0, j = 1, 2, \dots, n_{TC}; \quad (2)$$

– in the form of inequalities

$$[F_{\min}] \leq F(X, Y, Z) \leq [F_{\max}]; [U_{\min}] < U(X, Y, Z) < [U_{\max}], \quad (3)$$

as well as restrictions on independent coupling parameters

$$Y_{\min i} < Y_i < Y_{\max i} \quad i = 1, 2, \dots, k_x; \quad (4)$$

and design parameters

$$X_{\min i} < X_i < X_{\max i} \quad i = 1, 2, \dots, k_x. \quad (5)$$

Equalities (2) include balancing conservation equations, flows, pressures, enthalpy changes for all  $n_{TC}$  elements of the power plant, on the basis of which the mathematical model of the power plant is built. In these equations, the index  $j$  defines subsets  $X$ ,  $Y$  and  $Z$ , belonging to  $j$ -th element of the thermal circuit. Design parameters of the power plant  $X$  include both discretely variable (number of included group pumps, characteristics determining the configuration of the thermal circuit, etc.) and continuously variable parameters. Solvability of system (1) (satisfaction of the relevant constraints of problem (2)) is achieved during modeling due to the features of the mathematical model of the power plant, associated with the possibility of constructing such a sequence of these equations, from which the sought  $Y$  can be calculated directly.  $\Phi_j$  – a set of balancing equations for all elements of the plant equipment.

Inequalities (3) determine the permissible range of change of  $F(X, Y)$ , ...,  $U(X, Y)$  of technological characteristics and parameters of the power plant.  $[F_{\min}]$ ,  $[F_{\max}]$ ,  $[U_{\min}]$ ,  $[U_{\max}]$  – minimum and maximum permissible values of these characteristics and parameters.

The composition of ratios forming the admissible area of problem (1) also includes restrictions (4) on the independent connection parameters of the power plant elements (according to the terminology of optimal design – state parameters and flow characteristics) and restrictions (5) on the design parameters that determine the area of the heat exchange surfaces, underheating, the number of pumps, the configuration of the vehicle, etc. This approach to solving the search problem (1), when the thermal circuit is calculated at each step (conditions (2) are satisfied), is commonly called modeling in the state space.

The purpose of conducting the research is to determine the production of electricity by the power unit at the given modes of operation of the heat supply turbine plant based on the consumption of fresh steam and the amount of heat released. In view of this, the criterion for evaluating the efficiency of the heat supply turbine operation is the additional electric power that is produced at the given mode of operation of the turbine.

Work efficiency assessment criterion

$$\Delta N_t = N_t - N_t^{\text{in}} = f(G_0, Q_{t1}, Q_{t2}, t_{\text{oa}}, G_{\text{nw}}),$$

where  $N_t$  is the turbine power under specified operating conditions, MW;  $N_t^{\text{in}}$  is the power of the turbine at the instructional characteristics of the operating mode, MW;  $G_0$  is the consumption of fresh steam, t/h;  $Q_{t1}$ ,  $Q_{t2}$  is the thermal load of network heaters no. 1 and no. 2, MW;  $t_{\text{oa}}$  is the outdoors air temperature, °C;  $G_{\text{nw}}$  is the consumption of network water, t/h.

## Results and discussion

The calculation study was carried out at the typical network water flow rate for a turbine plant  $G_{\text{nw}}$  from 1000 t/h to 4500 t/h, in the range of outdoor air temperature changes  $t_{\text{oa}}$  from -11 °C to 10 °C (heating season) and  $t_{\text{oa}} > 10$  °C (hot water supply) [6].

The load change of the power unit was carried out due to the consumption of fresh steam at a constant pressure and temperature at the turbine inlet. Fresh steam consumption for the considered loads in the adjustable range is equal to [10]:

$G_0=295$  t/h – the lower value of the turbine adjustment range;

$G_0=440$  t/h – nominal operating mode of the turbine;

$G_0=485$  t/h – operation of the turbine in the mode of maximum power and in the pure condensation mode (during the calculation study, the operation of the turbine without heat release was not taken into account).

The following modes of operation of the T-100/120-130 heat supply turbine were considered:

– when releasing heat from the lower selection using the NH-1

$$\bar{Q}_{t1} = Q_{t1}/Q_t = 1.0;$$

– when releasing heat from two selections, when network heaters NH-1 and NH-2 are turned on in sequence with equal heat load (standard mode of operation)

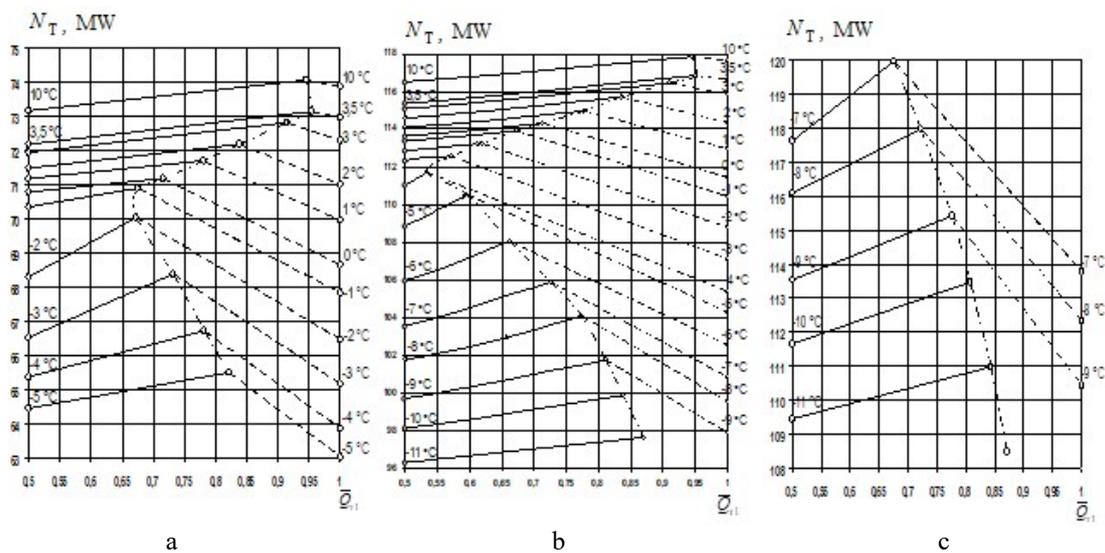
$$\bar{Q}_{t1} = Q_{t1}/Q_t = 0.5 \quad \text{or} \quad Q_{t1} = Q_{t2} = 0.5Q_t;$$

– with optimal distribution of heat load between them

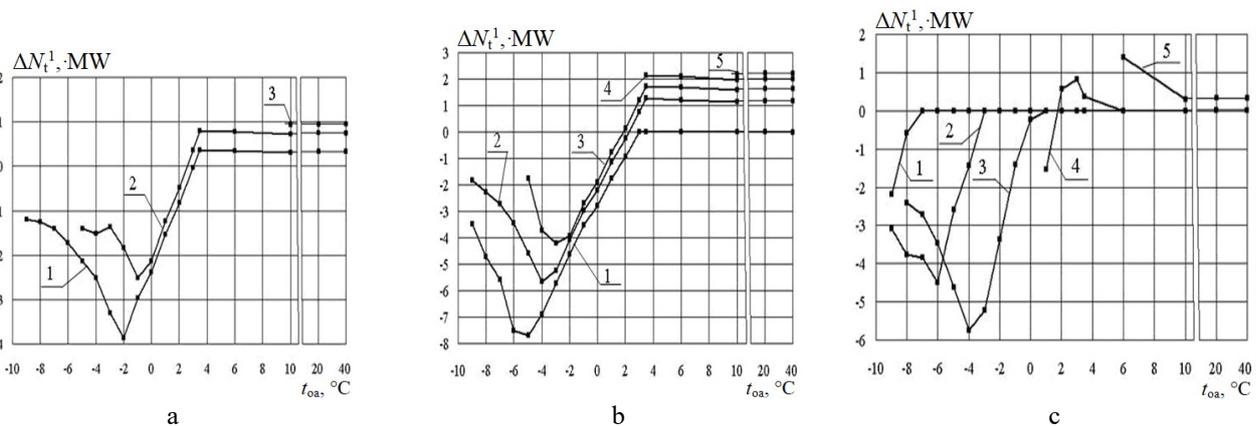
$$\bar{Q}_{t1} = \bar{Q}_{t1}^{\text{opt}},$$

where  $Q_{t1}$ ,  $Q_{t2}$  is the amount of heat supplied in the lower NH-1 and the upper NH-2 network water heaters;  $Q_t$  is the amount of heat released by the heat supply turbine.

Fig. 2 shows the result of a calculation study of the operation of network heaters of heat supply plant depending on the temperature of the outdoor air and the load of the T-100/120-130 turbine at the consumption of network water  $G_{nw}=2000$  t/h.



**Fig. 2. The influence of heat load distribution between network heaters of heat supply plant on the change in T-100/120-130 turbine power at network water consumption  $G_{nw}=2000$  t/h:**  
 a –  $G_0=295$  t/h; b –  $G_0^{\text{nom}}=440$  t/h; c –  $G_0^{\text{max}}=485$  t/h



**Fig. 3. The difference in turbine capacities when working with single-stage and two-stage heating of network water with equal distribution of the heat load:**  
 a –  $G_0=295$  t/h; b –  $G_0^{\text{nom}}=440$  t/h; c –  $G_0^{\text{max}}=485$  t/h;  
 1 –  $G_{nw}=1000$  t/h; 2 –  $G_{nw}=2000$  t/h; 3 –  $G_{nw}=3000$  t/h; 4 –  $G_{nw}=4000$  t/h; 5 –  $G_{nw}=4500$  t/h

To evaluate the effect when working with one or two network water heaters, the difference in turbine power  $\Delta N_t$  was considered. It was obtained when using one heater, when all the heat released to the consumer is formed in it ( $\bar{Q}_{t1} = Q_{t1}/Q_t = 1.0$ ), or when using two network heaters with an equal distribution of the heat released by the heaters ( $\bar{Q}_{t1} = 0.5$ ) at different temperatures of the outdoor air  $t_{oa}$  and other conditions being equal

$$\Delta N_t^1 = N_t^1 - N_t^2,$$

where  $\Delta N_t^s$  – power increase during the operation of a heat supply turbine plant with one lower network heater NH-1, MW;  $N_t^1$  is the power at single-stage heating of network water, MW;  $N_t^2$  – power with two-stage heating of network water and equal distribution of heat load between NH-1 and NH-2 heaters, MW.

Fig. 3 shows the variation of this difference for operating modes of the T-100/120-130 turbine at different flow rates of network water and fresh steam.

It can be seen that for  $G_0=295$  t/h (Fig. 3, a) at the outdoor air temperature  $t_{oa} \geq 6$  °C and  $G_{nw}=1000$  t/h turbine capacity with single-stage heating of network water exceeds capacity of one with two-stage heating by 0.25 MW, at  $3 \leq t_{oa} < 6$  °C the value of  $\Delta N_t^1$  reaches its maximum value and equals 0.4 MW. When the consumption of network water increases to  $G_{nw}=2000$  t/h (line 2) and  $t_{oa} \geq 6$  °C the power of the turbine plant with single-stage network water heating exceeded the power of one with two-stage by 0.73 MW, the maximum excess is 0.8 MW with  $t_{oa}=3.5$  °C. Zero value of  $\Delta N_t^1$  corresponds to  $t_{oa}=2.5$  °C (Fig. 3, a).

Further increase in network water consumption to  $G_{nw}=3000$  t/h (line 3) at  $t_{oa} \geq 10$  °C leads to an excess of turbine power in single-stage heating by about 1 MW.

At the values of the outdoor air temperature  $t_{oa} < 2.5$  °C, operation of the turbine with single-stage heating of network water is irrational and leads to a loss of power. So, for  $G_{nw}=1000$  t/h power loss with single-stage heating compared to one of two-stage heating can amount to 3.87 MW. This power loss occurs at  $t_{oa} = -2$  °C. For  $G_{nw}=2000$  t/h, similar power loss is 2.65 MW at  $t_{oa} = -0.7$  °C.

For the nominal mode of operation of the turbine (Fig. 3, b), the excess power during single-stage water heating in the range of changes in the temperature of the outdoor air  $t_{oa} > 2$  °C and network water consumption  $G_{nw} > 1000$  t/h is more significant.

For the loss of  $G_{nw}=1000$  t/h at  $t_{oa} \geq 3$  °C, zero power gain is observed.

The maximum gain in power is observed at  $t_{oa}=3.5$  °C (breaking point of the temperature graph) and corresponds to  $\Delta N_t^1=1.25$  MW for  $G_{nw}=2000$  t/h,  $\Delta N_t^1=1.7$  MW for  $G_{nw}=3000$  t/h,  $\Delta N_t^1=2.15$  MW for  $G_{nw}=4000$  t/h. At  $t_{oa} \geq 6$  °C a possible increase in power will be  $\Delta N_t^1=1.05$  MW for  $G_{nw}=2000$  t/h,  $\Delta N_t^1=1.55$  MW for  $G_{nw}=3000$  t/h,  $\Delta N_t^1=2$  MW for  $G_{nw}=4000$  t/h,  $\Delta N_t^1=2.15$  MW for  $G_{nw}=4500$  t/h.

At  $t_{oa}=2.3$  °C and  $G_{nw}=2000$  t/h capacity increase becomes equal to  $\Delta N_t^1=0$ , at  $t_{oa}=1.6$  °C and  $G_{nw}=3000$  t/h,  $\Delta N_t^1=0$ , at  $t_{oa}=3.5$  °C and  $G_{nw}=4000$  t/h  $\Delta N_t^1=2.15$  MW. Power gain range for  $G_{nw}=4500$  t/h is limited by temperature  $t_{oa} \geq 10$  °C.

For maximum load mode  $G_0^{\max}=485$  t/h (Fig. 3, c) with network water consumption  $G_{nw} \leq 4000$  t/h power increase is not observed in the entire change range of  $t_{oa} \geq 6$  °C. Only for  $G_{nw}=4500$  t/h,  $\Delta N_t^1=1.4$  MW. For the range of  $t_{oa} < 6$  °C, power increase to  $\Delta N_t^1=0.8$  MW can be obtained at  $t_{oa}=3$  °C, and at  $t_{oa}=1.8$  °C we have  $\Delta N_t^1=0$ .

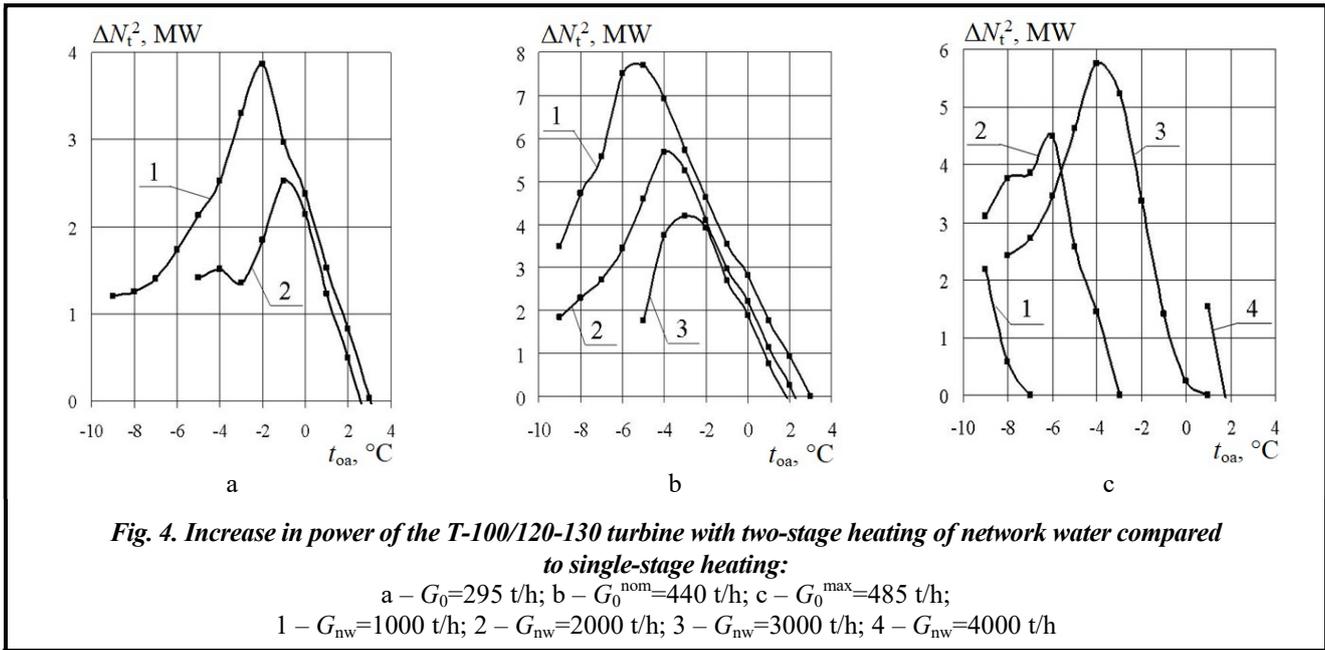
Given the narrow temperature range of the power increase, the error in determining the temperature  $t_{oa}$ , adjustment accuracy and other operational factors, it should be assumed that for mode of operation of the turbine at  $G_0^{\max}=485$  t/h and  $G_{nw} \leq 4000$  t/h capacity increase is insignificant ( $N_t=120$  MW for both options for using heating selections).

The use of single-stage network water heating is possible in the range of changes in the outdoor air temperature  $t_{oa} > -7$  °C for  $G_{nw}=1000$  t/h,  $t_{oa} > -3$  °C for  $G_{nw}=2000$  t/h,  $t_{oa} > 1$  °C for  $G_{nw}=3000$  t/h and  $t_{oa} > 1.8$  °C for  $G_{nw}=4000$  t/h.

In order to clearly assess the effect of the T-100/120-130 turbine plant with two-stage heating of network water, it is advisable to consider the difference in power

$$\Delta N_t^2 = N_t^2 - N_t^1$$

at different flow rates of network water (Fig. 4).



In the mode of operation of the turbine  $G_0=295$  t/h (Fig. 4, a), the maximum increase in capacity occurs at the network water flow rate of  $G_{nw}=1000$  t/h and outdoor air temperature of  $t_{0a} = -2$  °C. Turbine power when working with two-stage heating of network water with an equal heat load on both heaters ( $\bar{Q}_{t1} = 0.5$ ) is  $N_t^2=75.49$  MW, with single-stage heating ( $\bar{Q}_{t1} = 1.0$ ) the power of the turbine is equal to  $N_t^1=71.63$  MW. The maximum increase is  $\Delta N_t^2=3.86$  MW.

For nominal mode  $G_0^{nom}=440$  t/h (Fig. 4, b), the largest increase in the power of the turbine plant when working with two-stage heating of network water with an equal distribution of the heat load between heaters NH-1 and NH-2 reaches  $\Delta N_t^2=7.7$  MW at  $G_{nw}=1000$  t/h,  $t_{0a} = -5$  °C,  $Q_t=56.99$  MW, when  $N_t^2=118.38$  MW.

For maximum power mode  $G_0^{max}=485$  t/h (Fig. 4, c),  $\Delta N_t^2=5.76$  MW at  $G_{nw}=3000$  t/h,  $t_{0a} = -4$  °C,  $Q_t=167.47$  MW when  $N_t^2=118.45$  MW.

The comparison of the modes of operation of T-100/120-130 turbine with two-stage heating of network water at optimal distribution of heat load between network heaters and with their equal load (Fig. 5) is of greatest interest

$$\Delta N_t^{opt1} = N_t^{opt} - N_t^2,$$

where  $\Delta N_t^{opt1}$  – power increase during the operation of the heat supply turbine plant with the optimal relatively equal distribution of the heat load between network heaters, MW;  $N_t^{opt}$  – power produced by two-stage heating of network water with optimal heat load distribution between NH-1 and NH-2, MW, as well as with the option of single-stage heating of network water (Fig. 6)

$$\Delta N_t^{opt2} = N_t^{opt} - N_t^1,$$

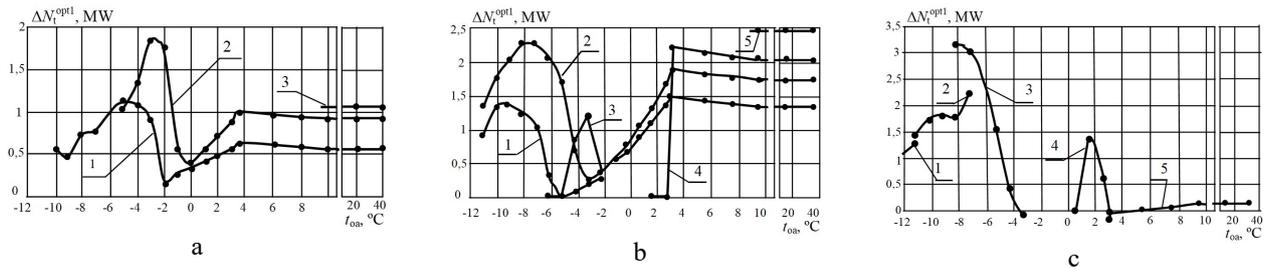
where  $\Delta N_t^{opt2}$  – power increase during the operation of a heat supply turbine plant with optimal distribution of the heat load between network heaters relative to single-stage network water heating, MW.

As can be seen from Fig. 5, the greatest gains are achieved in the nominal mode of operation of the turbine plant  $G_0^{nom}=440$  t/h.

So, at  $t_{0a}$  from  $-7$  °C to  $-8$  °C and  $G_{nw}=2000$  t/h  $\Delta N_t^{opt1}=2.3$  MW (line 2, Fig. 5, b), at  $t_{0a}=3,5$  °C and  $G_{nw}=4000$  t/h  $\Delta N_t^{opt1}=2.2$  MW (line 4, Fig. 5, b), at  $t_{0a} \geq 10$  °C and  $G_{nw}=4500$  t/h  $\Delta N_t^{opt1}=2.46$  MW (line 5, Fig. 5, b).

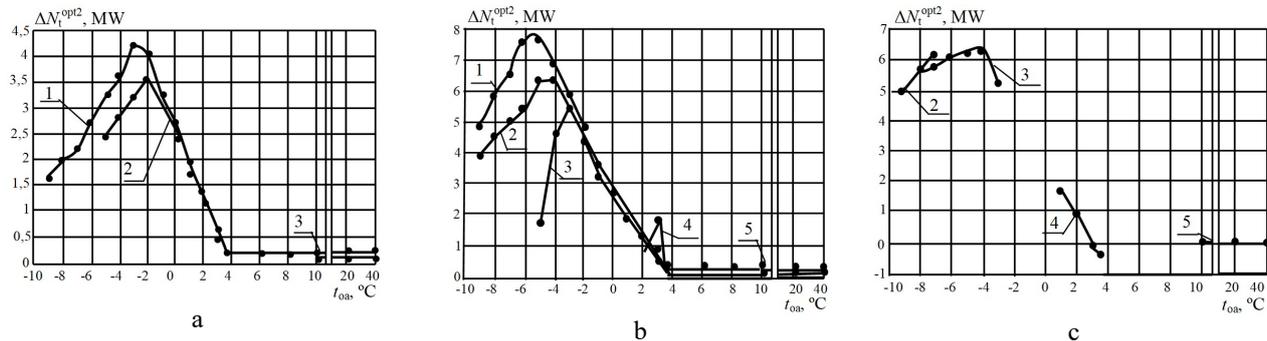
For the maximum mode of operation of the T-100/120-130 turbine plant  $G_0^{max}=485$  t/h these gains can be even greater (Fig. 5, c) and reach from 2.6 MW to 3.2 MW, but in a small range of  $t_{0a}$  from  $-6$  °C to  $-8$  °C.

It is obvious that the values of power gains will be even greater when compared with the option of single-stage heating of network water (Fig. 6).



**Fig. 5. The difference in turbine capacities when working with two-stage heating of network water with equal and optimal distribution of the heat load:**

a –  $G_0=295$  t/h; b –  $G_0^{nom}=440$  t/h; c –  $G_0^{max}=485$  t/h;  
 1 –  $G_{nw}=1000$  t/h; 2 –  $G_{nw}=2000$  t/h; 3 –  $G_{nw}=3000$  t/h; 4 –  $G_{nw}=4000$  t/h; 5 –  $G_{nw}=4500$  t/h



**Fig. 6. The difference in turbine capacities when working with two-stage network water heating with optimal heat load distribution between heaters and single-stage network water heating:**

a –  $G_0=295$  t/h; b –  $G_0^{nom}=440$  t/h; c –  $G_0^{max}=485$  t/h;  
 1 –  $G_{nw}=1000$  t/h; 2 –  $G_{nw}=2000$  t/h; 3 –  $G_{nw}=3000$  t/h; 4 –  $G_{nw}=4000$  t/h; 5 –  $G_{nw}=4500$  t/h

In the mode of operation of the turbine  $G_0=295$  t/h (Fig. 6, a), the maximum increase in capacity occurs with the consumption of network water of  $G_{nw}=1000$  t/h and outdoor air temperature  $t_{0a} = -3$  °C and is equal to  $\Delta N_t^{opt2}=4.2$  MW.

For nominal mode  $G_0^{nom}=440$  t/h (Fig. 6, b), the largest increase in the power of the turbine plant when working with two-stage heating of network water with optimal distribution of the heat load between heaters NH-1 and NH-2 reaches  $\Delta N_T^{opt2}=7.84$  MW at  $G_{nw}=1000$  t/h and  $t_{0a} = -6$  °C.

For maximum load mode  $G_0^{max}=485$  t/h (Fig. 6, c) with network water consumption  $G_{nw}=3000$  t/h and outdoor air temperature  $t_{0a} = -4$  °C the increase in power is  $\Delta N_T^{opt2}=6.33$  MW.

**Conclusions**

1. With the help of the software complex developed at the Institute of Mechanical Engineering Problems of the National Academy of Sciences of Ukraine, calculation studies were carried out to determine the possibilities of increasing the efficiency of heat supply turbines of the CHPPs due to the selection of rational modes of operation of network water heaters.

2. Calculation study of the operation of the T-100/120-130 heat supply turbine in operating conditions with one or two heating steam selections showed that in the region of positive outdoor air temperatures  $t_{0a} > 2$  °C at all network water flow rates  $G_{nw}$  it is expedient to work with one lower selection (when the upper one is disabled). At the same time, additional electrical power in the area of outdoor air temperatures  $t_{0a} \geq 6$  °C depending on  $G_{nw}$ , that is, on the amount of heat released into the heat network can be compared with the load regulated by the turbine operating instructions ( $Q_{t1}=Q_{t2}=0.5 \cdot Q_t$ ), from 0.25 MW to 2.15 MW. At  $t_{0a} < 2$  °C, work with one lower heating selection becomes irrational.

3. The most important, from the point of view of the organization of rational modes of operation of turbine plants, are the results of determining the optimal distribution of heat load between network heaters. It is shown that in this case, the gain in the electric power of the turbine can be up to 2.46 MW in the nominal mode of operation with two heaters, and, in comparison with the use of one-stage heating, up to 7.84 MW.

Thus, in the studied range of changes in the outdoor air temperature from  $-11\text{ }^{\circ}\text{C}$  to  $30\text{ }^{\circ}\text{C}$ , there is an optimum distribution of the load between the network heaters of the lower and upper stages, and the greater heat load should fall on the lower heat supply selection ( $\bar{Q}_{t1} > 0.5$ ).

4. The nature of the influence of distribution  $\bar{Q}_{t1}$  indicates that during deprivation from the instructional uniform distribution of the heat load between network heaters, additional electricity can be obtained, and therefore the optimal operation of the heat supply plant should be considered at different thermal and electrical loads of the turbine. This indicates the importance of creating software for the convenience of presenting the results of numerical studies for their use at the CHPPs during the operation of the power unit in the future, which will allow the operational staff to choose a rational distribution of loads between network heaters when changing modes of operation.

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## Підвищення ефективності теплофікаційних блоків ТЕЦ за рахунок вибору раціональних режимів відпуску теплоти

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Проаналізовано можливості підвищення економічності теплофікаційних турбін теплоелектроцентра-лей за рахунок вибору раціональних режимів експлуатації підігрівачів сітьової води. За допомогою програмно-обчислювального комплексу, розробленого в Інституті проблем машинобудування НАН України та адаптованого авторами до умов експлуатації генеруючого обладнання ТЕЦ з одним або двома мережевими підігрівачами, у роботі проведено комплекс розрахункових досліджень різних способів їх підключення залежно від температури зовнішнього повітря. Встановлено області позитивного ефекту, пов'язаного з підвищенням електричної потужності турбоустановки. Розрахункове дослідження виконано при типових для турбоустановки витратах сітьової води від 1000 т/год до 4500 т/год, в діапазоні зміни температури зовнішнього повітря від -11 °С до 10 °С (опалювальний сезон) і більше 10 °С (гаряче водопостачання). Зміна навантаження енергоблоку здійснювалася за рахунок витрати свіжої пари при постійному тиску й температурі на вході в турбіну. Як показали результати роботи теплофікаційної турбіни Т-100/120-130 в умовах експлуатації з одним або двома теплофікаційними відборами пари, в області позитивних температур зовнішнього повітря вище 2 °С при всіх витратах сітьової води доцільно використовувати один нижній відбір (при відключеному верхньому). При цьому додаткова електрична потужність в області температур зовнішнього повітря більше 6 °С може скласти від 0,25 МВт до 2,15 МВт. Проте при температурі зовнішнього повітря менше 2 °С робота з одним нижнім теплофікаційним відбором стає нераціональною. Із точки зору обрання раціональних режимів експлуатації турбоустановок найбільш важливими є результати з визначення оптимального розподілу теплового навантаження між сітьовими підігрівачами. Виграш у електричній потужності турбіни може становити на номінальному режимі експлуатації з двома підігрівачами до 2,46 МВт, а в порівнянні з використанням одноступінчатого підігріву – до 7,84 МВт. Характер впливу розподілу теплового навантаження вказує на те, що при відході від інструкційного рівномірного розподілу теплового навантаження між сітьовими підігрівачами можна отримувати додаткову електроенергію.

**Ключові слова:** математичне моделювання, підігрівач сітьової води, режими роботи, теплоелектроцентраль, теплофікаційна турбоустановка.

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